International Journal of
Thermal Energy and Applications

Vol. 1: Issue 2
www.journalspub.com

Theoretical Investigation on Transient Heat Transfer through Building for PV-powered Solar Summer Air Conditioning

Suhas B.G.*[1], Kavadiki Veerabhadrappa[2], Kiran Kumar U.[3]
[1,2]Assistant Professor, Department of Mechanical Engineering, B.M.S. College of Engineering, Bull Temple Road, Bengaluru, Karnataka, India
[3]Assistant Professor, Department of Mechanical Engineering, B.M.S. College of Engineering, Bull Temple Road, Bengaluru, Karnataka, India

ABSTRACT
The present work determines the transient heat transfer through wall for solar air conditioning study in summer. Suitable recirculation, indoor dry-bulb temperature, wet-bulb temperature, outdoor dry-bulb temperature and relative humidity are chosen. Considering the plaster on inside-wall and outside-wall construction, roof construction, floor construction, door size and thermal conductivities of glass, concrete, bricks, plaster, and the latent heat and sensible heat from the room are determined. Direct and diffuse radiation is obtained for the month of March, April and May. The sol-air temperature is estimated by considering these radiations, outdoor temperature and outside heat transfer coefficient. From the sol-air temperature, time lag of temperature difference between indoor and outdoor is calculated. Equivalent temperature difference method is also used to determine the temperature difference between these indoor and outdoor. The transient heat transfer coefficient is calculated by the values of temperature differences. Power required to run the solar panel is calculated from the determined values of heat transfer coefficient.

Keywords: *Diffuse radiation, direct radiation, inside heat transfer coefficient, sol-air temperature, solar panel*

Corresponding Author
E-mail: suhas_bg@yahoo.co.in

INTRODUCTION

In great-grandfather's day, air conditioning was confined to winter comfort; that is, to heating the home to a comfortable temperature in the winter by means of open fireplaces or stoves. Later, decided improvements were made in the heating equipment by the introduction of steam and hot water heating, but air conditioning was still confined to winter comfort. Less than 50 years ago, human beings were content to be comfortable in winter and uncomfortable during the humid heat of summer. Air conditioning has been defined as the process for treating air, so as to control simultaneously its temperature, humidity, cleanliness and distribution to meet the requirements of the conditioned space. Air conditioning systems for these hot and dry climates also require cooling of air below the ambient temperatures. In order to reduce cost and save electricity, solar air conditioning can be used. Solar energy provides to be given energy due to zero emission [1, 2]. From the summer air conditioning review, it found that the energy consumption of the systems with an energy recovery ventilation (ERV) increases as the exhaust airflow coefficient of performance (COP) has direct effect on running cost of refrigeration cycle; therefore, higher the COP lower will be the running cost [3]. The renewable and sustainable energy reviews found that possibility of using non-polluting materials save more than half of the primary energy. In recent years, becomes a power-hungry nation, it is observed that demand for power will grow assuming 9% annual increase in the usage of power for the next 10 years [4]. The increase in power demand is now at a faster rate as more people have started using the available electrical and electronic gadgets for comfort due to less power production, power failures, power cuts, become a common phenomenon in many places. In this scenario, saving the energy is at most important; therefore, time has come to develop energy-efficient air conditioning system. Several researchers developed solar cooling technology using absorption chiller [5, 6] or ejector cooling [7, 8] which were driven by the solar energy. In these types of cooling, it was required to drive adsorption or ejector chiller with high energy conversion efficiency at high temperature. As the cost of collector and heat-driven chiller is high, this kind of solar cooling is not economical. Therefore, it found that solar cooling by using solar panels is an alternative for the absorption cooling. For small residential and small commercial cooling, photovoltaic (PV)-powered cooling has been the most economical means of solar cooling technology. Recently, few researchers studied the solar cooling using vapor compression cooler driven by solar PV system, since solar PV system cost falls down very fast. Hartmann et al. [9] compared theoretically solar thermal and photovoltaic cooling for a small office building and found that the grid-tied PV system has lower cost of primary energy savings. In these solar cooling systems, the power grid will supply electricity for cooling when solar energy is not available. Huang et al. [10] developed a stand-alone solar air conditioner driven directly by solar PV. An air conditioner with 200 W AC power was driven directly by 430 W solar PV module.

Today, the most popular technologies are thermal driven and adsorption chillers in combination with solar thermal collectors. However, because of high costs of the sorption machines and collectors, the market for these technologies is growing very slowly. This economical reason increases the attractiveness of solar–electrical air conditioning systems. Therefore, the coupling of PV modules with an electrical-driven system of the type presents the concept of PV-based air conditioning. Therefore, the objective of the present work is to design an air cooling system based on PV panels.

LOAD CALCULATION

The heat transfer takes place through the building structure (walls, roof, floor, etc.), due to the temperature difference between the conditioned indoor space of a building and outdoor ambient. This is known as fabric heat transfer and this includes sensible heat transfer through all the structural elements of a building. But it does not include radiation heat transfer through fenestration. Exact analysis of heat transfer through building structures is very complex, as it has to consider:

a. Geometrically complex structure of the walls, roofs, etc. consisting of a wide variety of materials with different thermophysical properties.
b. Continuously varying outdoor conditions due to variation in solar radiation, outdoor temperature, wind velocity and direction, etc.

Table 1 gives the room parameters that are considered for design.

Table 1. *Room parameters.*

Location	Bengaluru, Karnataka, India
Height of the room	31 m
Area of the room	120 m^2
Plaster thickness of inside wall	0.0125 m
Outside wall construction	0.2-m block concrete and 0.1-m brick
Roof construction	0.2-m RCC slab with 0.04-m asbestos cement board
Floor construction	0.2-m concrete
Occupancy	70 students
Door size	1.5 m × 2 m$m \times 2m$
Daily Temperature range	22°–37°C

Outside Air Sensible Heat (OASH)
Outside air sensible heat is given as

$$\text{OASH} = 0.02 \times V_{ven}(T_{d1}\text{-}T_{d2}) \times V_{ven}(T_{d1}\text{-}T_{d2}) \quad (1)$$

Where, V_{ven} is the volume of ventilation, V_{ven} = 0.3 × volume of room × correction factor, 0.3 (30%) is the percentage of recirculation chosen.

Outside Air Total Heat (OATH)
It is the sum of outside air sensible heat and outside air latent heat as given by the following equation:

$$\text{OATH} = 0.02\ V_{ven}(h_1 - h_2) \quad (2)$$

h_1, h_2 are taken from psychometric chart as shown in Figure 1.

Outside Air Latent Heat (OALH)
Outside air latent heat is given by

$$\text{OALH} = \text{OATH} - \text{OASH} \quad (3)$$

Room Sensible Heat (RSH)
It is the sum of lighting load and the sensible heat from person due to physical phenomena.

Room Latent Heat (RLH)
It is the latent heat from each person. Table 1 shows the room sensible heat, latent heat and total heat.

Total Sensible Heat (TSH)
It is the sum of outside air sensible heat and room sensible heat as given by

$$\text{TSH} = \text{OAHSH} + \text{RSH} \quad (4)$$

Total Latent Heat (TLH)
It is the sum of outside air latent heat and room latent heat as given by

$$\text{TLH} = \text{OALH} + \text{RLH} \quad (5)$$

Room Total Heat (RTH)
It is the sum of total sensible heat and total latent heat as given

$$\text{RTH} = \text{TSH} + \text{TLH} \quad (6)$$

Total Volume
It is defined as the sum of the supply, stack, crack and door volume as follows:
V_{supply} is the total volume of air supplied into the room.
V_{stack} is the volume of movement of air into and out of building.
V_{crack} is the volume of air entering from cracks of building.
V_{doors} is the volume of air entering and exiting through doors.

$$V_{total}=V_{supply}+V_{stack}+V_{crack}+V_{doors} \quad (7)$$

Where

$$V_{supply}=\frac{volume\ of\ room \times correction\ factor}{60}$$

$$V_{stack}=0.172 \times A_{one\ wall} \times \sqrt{(to-ti)h}$$

h is the height of room from the ground level.

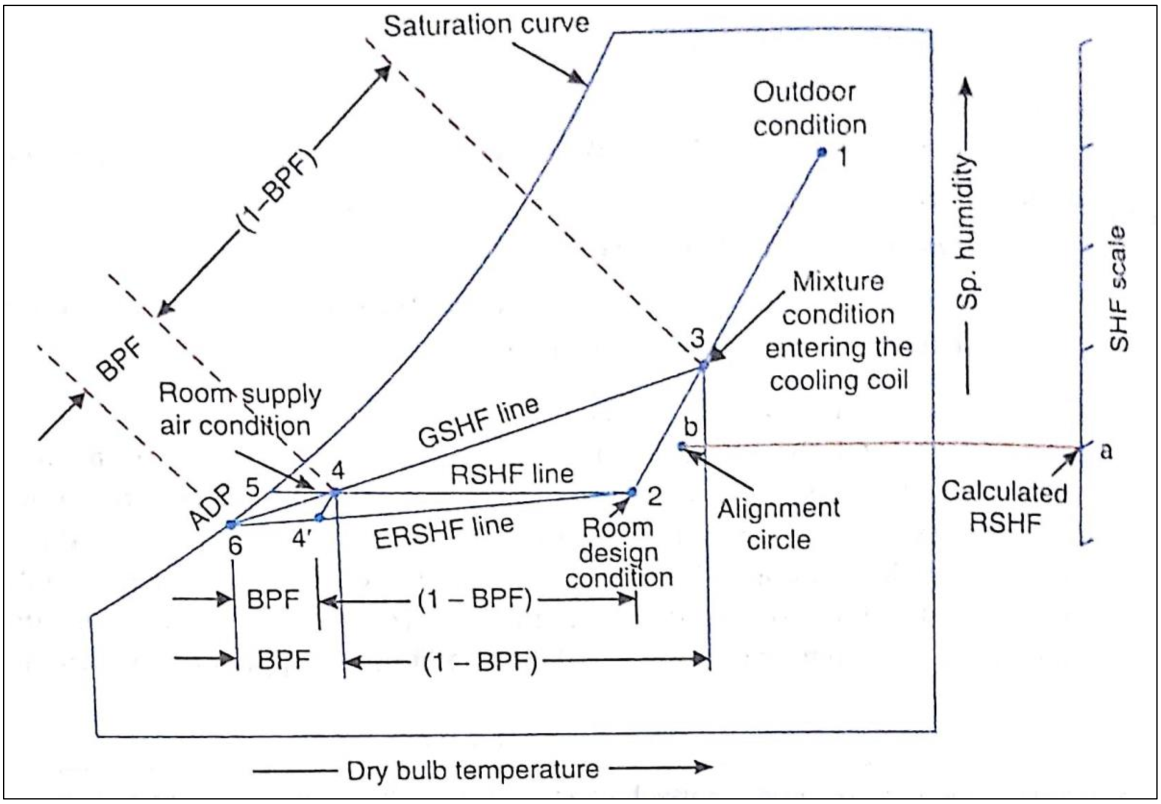

Fig. 1. *Psychometric chart [11].*

Bypass Factor (BPF)
Bypass factor (BPF) is the percentage of air that travels through a tube and fin coil without touching any coil surface as given in Equation (8). By considering the ratio of recirculated air and total air, the supply is located in the psychometry chart:

$$\text{BPF} = \frac{Td3-ADP}{Td4-ADP} \qquad \frac{Td3-ADP}{Td4-ADP} \tag{8}$$

Determine hs and ADP from the psychometry chart.

Effective Room Sensible Heat (ERSH)
It is the sum of all sensible heat gains that occur in the room including the gain due to the portion of the ventilation air which is bypassed as given in Equation (9)

$$\text{ERSH} = \text{RSH} + \text{BPF} \times \text{OASH} \tag{9}$$

Effective Room Latent Heat (ERLH)
It is the sum of all latent heat gains that occur in the room including the gain due to the portion of the ventilation air which is bypassed as given by the following Equation (10)

$$\text{ERLH} = \text{RLH} + \text{BPF} \times \text{OASH} \tag{10}$$

Effective Room Total Heat (ERTH)
It is the sum of effective room sensible heat, effective room latent heat and building orientation as given by Equation (11)

$$\text{ERTH} = \text{ERSH} + \text{ERLH} + \text{Building orientation} \tag{11}$$

Sol-Air Temperature
Sol-air temperature is a variable used to calculate the cooling load of the building and to determine the total heat gain through exterior surface, as given by Equation (12)

$$t_e t = t_o + \frac{\alpha_b I_T}{h_o} \tag{12}$$

Where $t_{o=}t_o$ is the outside temperature, I_T= is the total radiation.

Calculation for total radiation is given later.

$h_{o=}h_o$ is the outside heat transfer coefficient w/m^2 and $\alpha = \alpha$ is the absorbtivity.

Decremental Factor
(λ) and time lag (ϕ)

Decrement factor is the reduction in cyclical temperature on the inside surface compared to the outside surface. The time delay due to the thermal mass is known as time lag as given by Equation (13)

$$\Delta t_d = (t_{em}\text{-}t_e)+\lambda(t_{e\tau-\phi}\text{-}t_{em}) \quad \Delta t_d = (t_{em}\text{-}t_e)+\lambda(t_{e\tau-\phi}\text{-}t_{em}) \tag{13}$$

Where t_{em} is the mean sol-air temperature, λ is the decrement factor, and $t_{e\tau-\phi}$= is the mean sole air temperature (ϕ) hours ago

Total Radiation

Global radiation on daily basis monthly average (H_g) of a particular day is given as follows:

$$\frac{H_g}{H_o} = a + b\left(\frac{\omega_s}{\omega_{s,max}}\right) \tag{14}$$

Where $\frac{\omega_s}{\omega_{s,max}}$ is the ratio of sunset hour to maximum sunset hour in the particular month.

Global radiation on hourly basis monthly average (H_O) if there would be clear sky is given as follows

$$H_o = \frac{24}{\pi} I_{sc}\left[1 + 0.033\left(\frac{360n}{365}\right)\right](\omega_s Sin\phi\ Sin\delta + Cos\phi\ Cos\delta\ Sin\omega_s) \tag{15}$$

δ is declination angle and is given by

$$\delta = 23.45\ Sin\left[\frac{360}{365}(284+n)\right] \tag{16}$$

Diffused radiation on hourly basis monthly average (H_d) of a particular day is given by

$$\frac{H_d}{H_g} = 0.8677 - 0.7365\left(\frac{\omega_s}{\omega_{s,max}}\right) \tag{17}$$

Global radiation on hourly basis monthly average (I_g) of a particular day is given by

$$\frac{I_g}{I_o} = \frac{H_g}{H_o}(a' + b' cos\omega_s) \tag{18}$$

Where $a' = 0.409 + 0.506\ Sin(\omega_s - 60)$ and $b' = 0.6609 - 0.4767\ Sin(\omega_s - 60)$

Global radiation on hourly basis daily average (I_O) if there would be clear sky is given by

$$I_o = I_{sc}\left[1 + 0.033\left(\frac{360n}{365}\right)\right](\omega_s Sin\phi\ Sin\delta + Cos\phi\ Cos\delta\ Sin\omega_s) \tag{19}$$

Diffuse radiation on hourly basis monthly average (I_d) can be calculated by

$$\frac{I_d}{H_d} = \frac{I_o}{H_o}(a'' + b'' cos\omega_s) \tag{20}$$

Where $a" = 0.4922 + \{0.27/(H_d/H_g)\}$ and $b" = 2(1-a)(Sin\omega_s - \omega_s cos\omega_s)/(\omega_s - 0.5\ Sin2\omega_s)$

Total radiation (I_T) on the solar panel can be calculated by

$$I_T = I_g r_g(\tau\alpha_b)_g + \{\ I_d I_d + (I_g + I_d) r_r\}(\tau\alpha_b)_d \tag{21}$$

The total radiation obtained from Equation (22) is substituted in Equation (13) to obtain sol-air temperature.

Equivalent Temperature Difference Method

The equivalent temperature difference (Δt_e) is given by

Δt_e= [(outside max. temperature − Inside design temperature) − 8.3] − [(outside max.

temperature) − (min. outdoor temperature) 11.1] (22)

Equivalent temperature difference method corrections:

1. An outdoor daily range of a dry-bulb temperature of 11.1℃
2. Outside and inside design temperature difference of 8.3℃
3. Dark color roofs and walls with absorptivity is 0.3
4. Specific heat of the construction material is 1.005 kJ/Kg-K
5. The values of t_o and t_i are additive to Δt. Hence add or subtract the Difference of $t_o - t_i$ and 8.3℃.

Inside Heat Transfer Coefficient

The final equation for effective room total heat is given as

$$\text{ERTH} = [\frac{1}{h_0}+\frac{1}{h_i}+\frac{KA}{\Delta X}]\times[\frac{\Delta t_{d+\Delta t_e}}{2}] \quad \textbf{(}23)$$

By simplifying Equations (23), Equation (24) is obtained

$$\frac{1}{h_i}\frac{2ERTH}{[\Delta t_d+\Delta t_e]}-\frac{1}{h_o}-\frac{kA}{\Delta x} = \frac{2ERTH}{[\Delta t_d+\Delta t_e]}-\frac{1}{h_o}-\frac{kA}{\Delta x} \quad (24)$$

METHODOLOGY TO CALCULATE THEORETICAL HEAT TRANSFER COEFFCIENT AND NUMBER OF PANELS REQUIRED

Following is the procedure to calculate the theoretical heat transfer coefficient and the number of solar panels required:

- Suitable recirculation, indoor dry-bulb temperature, wet-bulb temperature, outdoor dry-bulb temperature and relative humidity are chosen for the study.
- Calculate the outside air sensible heat and outside air total heat by Equations (1) and (2). Outside heat latent heat can be calculated by Equation (3).
- Considering the plaster on inside–wall and outside-wall construction, roof construction, floor construction, door size and thermal conductivities of glass, concrete, bricks, plaster, and the latent heat and sensible heat from the class room are determined.
- Room total heat is calculated from the room sensible heat, latent heat, and the enthalpy potential between the conditioned space and recirculation by Equation (6).
- Effective room total heat is determined from the calculated values of room sensible heat, room latent heat, outside room sensible heat, outside room latent heat and total volume of air supplied.
- Direct and diffuse radiation is obtained for the month of March, April and May by Equations (15)–(22). The sol-air temperature is estimated by considering those radiations, outdoor temperature and outside heat transfer coefficient by Equation (14).
- From the sol-air temperature, time lag of temperature difference between indoor and outdoor is calculated by Equation (23).
- Equivalent temperature difference method is also used to determine the temperature difference between the indoor and outdoor by Equation (24).
- The two different temperature differences are added and the transient heat transfer is estimated from the effective room sensible heat and those temperature differences.
- From these calculated parameters, inside heat transfer coefficient is determined. Power required to run the solar panel is calculated from the heat transfer coefficient values.

RESULTS AND DISCUSSION

Validation

The total radiation calculated for the month of March, April and May is shown in Figure 2, and is compared with the National Renewable Energy Laboratory’s (NREL) data. From the figure, it is found that the total radiation in March is higher compared with that of April and May. This is due to the variation of direct sunlight on the earth surface and phenomenon due to the tilt in the earth axis relative to the plane of its orbit. The length of the day varies when the earth axis tilts and results in variation in the total radiation obtained for each month [12].

To validate the present theoretical work, the results obtained for inside heat transfer coefficients on March are compared with the experimental, and presented by Phiraphata et al. [13]. The correlation is given in Equation (26). Their experimental set-up was composed of a PV panel with aluminum plate on the upper layer and the lower layer. The inclination angle and air gap of channel was fixed at 30°. The channel width was 0.7 m, and the length was 1.2 m. The upper part of the top wall was exposed to the heat flux value of 220 W/m^2 up to the length of 1.6 m, while the other walls of the channel are assumed to be adiabatic. Figure 3 shows the comparison with the correlation developed by Phiraphata et al. [13], and the present results obtained on March 15. The characteristic length of the room for upper wall surface was taken for validation. It is observed that the present result agrees well with the correlation developed by Phiraphata et al. [13]. It is found that the average Nusselt number determined from the present work deviated by 17.5% from the correlation.

Total Radiation

Total radiation plotted against the number of days in the month of March, April and May month as shown in Figure 4.

From the graph, it is observed that the total radiation is high on the 13th day of April month and the lowest on the 14th day of March as shown in the graph.

Inside Heat Transfer Coefficient

The variation of heat transfer coefficient with time is shown in Figure 5. It is observed that the heat transfer coefficient is high between 8 and 9 PM, because as the considered room is of thick wall, the solar radiation incident on the wall is absorbed during the day time, and the absorbed heat cannot lose to outdoor even during the night. Hence the heat absorbed by the wall is transferred into the room. Thus, the inside heat transfer coefficient increases during the night time compared to day time.

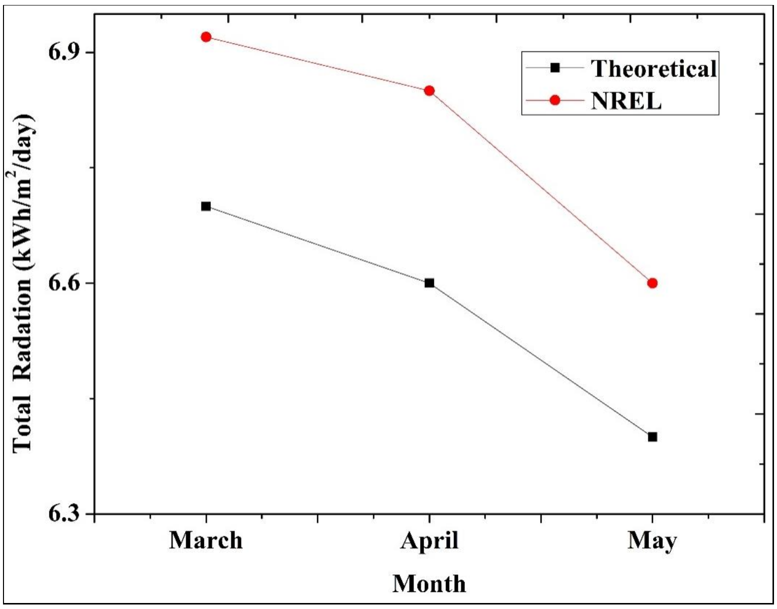

Fig. 2. *Comparison of standard and calculated total radiations.*

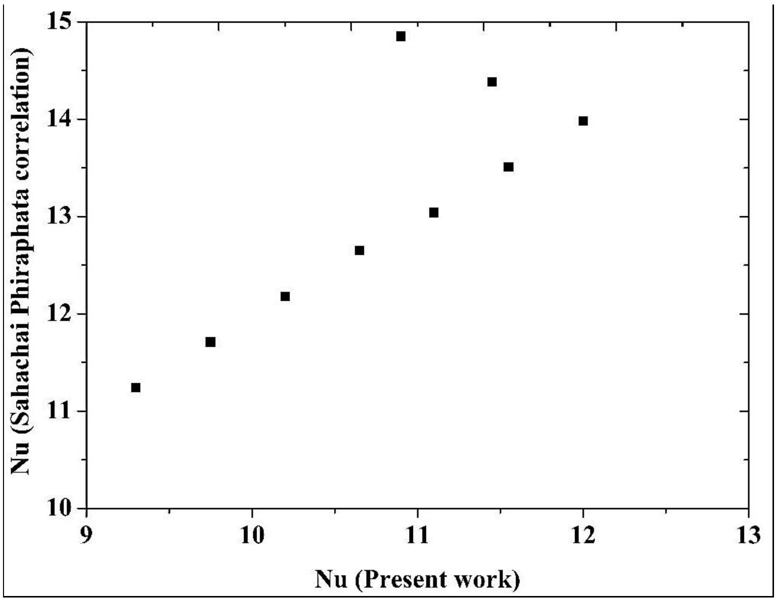

Fig. 3. *Validation of present work with literature correlation.*

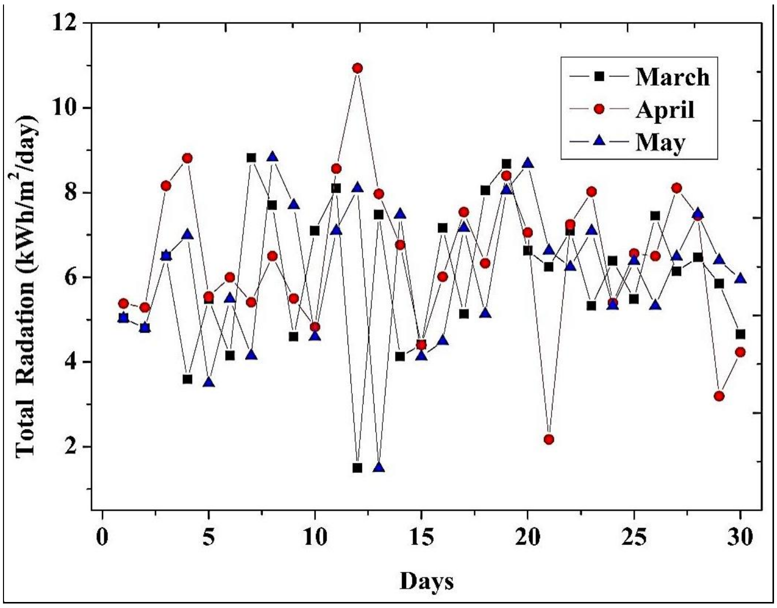

Fig. 4. *Total radiation vs. days.*

The average heat transfer coefficient with days is as shown in Figure 6. It is observed that the average heat transfer coefficient increases each day and reaches maximum at the end of the month compared to the first day of each month. This is because the inside heat is not completely transferred outside the room as some amount of heat will be retained due to thick wall. This results in increasing the average heat transfer coefficient day by day.

Solar Panels

The number of solar panels depends on the power required for an air conditioning for a particular duration. Inside heat transfer coefficient is obtained from Equation (24) for the month of March, April and May, and the area of the room is given in the room parameters, and the temperature difference is measured at that instant of time. The average heat transfer coefficient is considered to calculate the power and number of panels required as follows:

$$P = h_i \times \text{(Area of installation)} \times \Delta T \quad (25)$$

P = power required
h_i =Inside heat transfer coefficient
ΔT = Temperature difference

$$P = 66.82 \times 120 \times 20$$

$$P = 160 \text{ kW}$$

Considering 16 hours for a 250 W, gives 250 8 = 2 kW, that is, 2000 W in a day per 250 W rated solar panel. By multiplying 2 kW by 30 days in a month each, 250 W rated panel will produce about 60 kW in an average month. The number of solar panels is calculated as follows:

$$No. of\ Panels = \frac{Power\ obtained\ from\ Heat\ transfer\ coefficient}{Average\ power\ required\ per\ month} \quad (27)$$

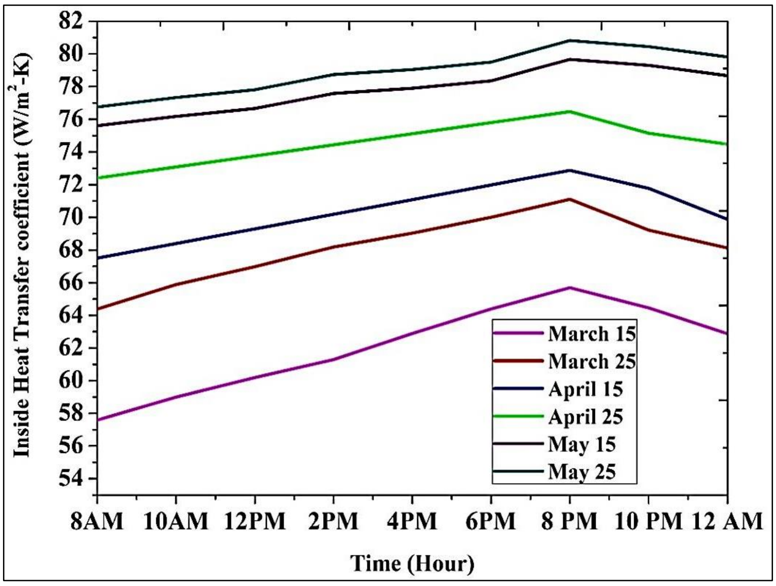

Fig. 5. *Inside heat transfer coefficient vs. time (hour).*

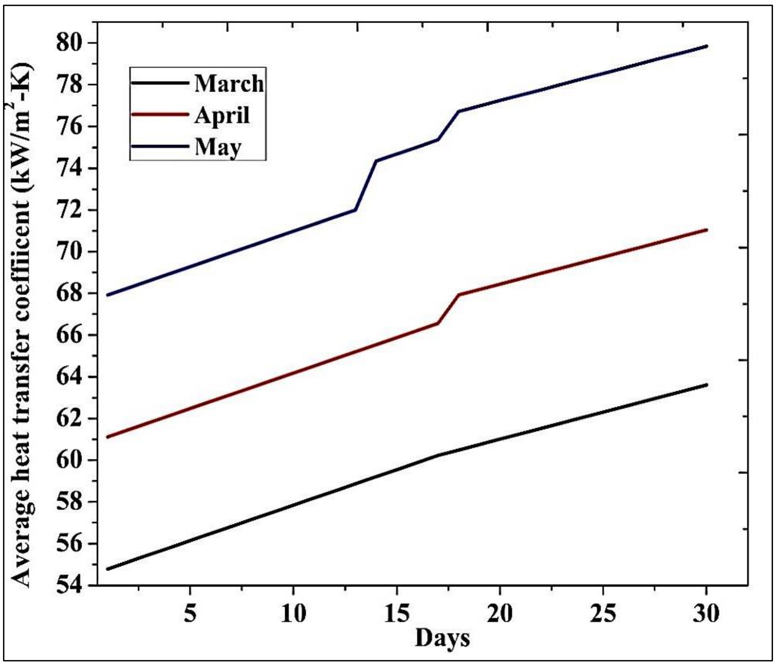

Fig. 6. *Average heat transfer coefficient vs. days.*

$$No.of\ Panels = \frac{160\ kW}{60\ kW} = 2.667$$

Therefore, three solar panels are required to generate 160 kW of power in an average month.

CONCLUSION

The present work determines the transient heat transfer through wall for solar air conditioning study in summer. Direct and diffuse radiation is obtained for the month of March, April and May. The sol-air temperature is estimated by considering those radiations, outdoor temperature and outside heat transfer coefficient. From the sol-air temperature, time lag of temperature difference between indoor and outdoor is calculated. Equivalent temperature difference method is also used to determine the temperature difference between the indoor and outdoor. The two different temperature differences are added and the transient heat transfer coefficient is estimated. Power required to run the solar panel is calculated from the determined values of heat transfer coefficient. The following are the conclusions that are arrived during the present study:

- It is observed that the total radiation is high on the 13th day of April month and the lowest on the 14th day of March.
- It is observed that the heat transfer coefficient is high between 8 and 9 PM because as the considered room is of thick wall, the solar radiation incident on the wall is absorbed during the day time, and the absorbed heat cannot lose to outdoor even during the night.
- It is also observed that the average heat transfer coefficient increases each day and reaches maximum at the end of the month compared to the first day of each month.
- Three solar panels are required to generate 160 kW of power in an average month.

NOMENCLATURE

A	Area of wall
k	Thermal conductivity of wall (kW/m-K)
h	Heat transfer coefficient (kW/m^2-K)
h_1	Indoor enthalpy (kJ/kg)
h_1	Outdoor enthalpy (kJ/kg)
H_g	Global radiation on daily basis monthly average on a particular day (kJ/m^2-day)
H_0	Global radiation on daily basis monthly average (kJ/m^2-day)
H_d	Diffused radiation on daily basis monthly average (kJ/m^2-day)
I	Radiation
I_{sc}	Solar constant (W/m^2)
I_g	Global radiation on hourly basis monthly average on a particular day (kJ/m^2-h)
I_0	Global radiation on hourly basis monthly average (kJ/m^2-h)
I_d	Diffused radiation on hourly basis monthly average (kJ/m^2-h)
Nu	Nusselt number
n	Number of days
r_g	Tilt factor for global radiation
r_d	Tilt factor for diffused radiation
r_r	Tilt factor for reflected radiation
t	Temperature (°C)
t_e	Sol-air temperature (°C)
Δt	Temperature difference (°C)
Δt	Wall thickness (m)
V	Volume flow rate of air (m^3/s)

Subscript

d	Diffused
d_1	Dry-bulb temperature at indoor (°C)
d_2	Dry-bulb temperature at outdoor (°C)
d_3	Dry-bulb temperature at mixing condition (°C)
d_4	Dry-bulb temperature at room supply condition (°C)
e	Sol-air
e_m	Mean sol-air
$e\tau - \phi$	time lag
g	Global
r	Reflected

λ	Decremental factor
Ven	Ventilated

Greek Symbol

α	Thermal diffusivity (m^2/s)
α_b	Absorptivity
δ	Declination angle
ω_s	Hour angle for sun set
ϕ	Latitude angle
$\tau\alpha_b$	Transmisivity–absorptivity product

REFERENCES

[1] Axaopoulos PJ, Theodoridis MP. Design and experimental performance of a PV ice-maker without battery. *Sol Energy*. 2009; 83: 1360–1369p.
[2] Cabrera FJ, Fernandez-García A, Silva RMP, Perez-García M. Use of parabolic trough solar collectors for solar refrigeration and air-conditioning applications. *Renew Sustain Energy Rev*. 2013; 20: 103–118p.
[3] El Tom OMM, Omer SA, Taha AZ, Sayigh AAM. Performance of a photovoltaic solar refrigerator in tropical climate conditions. *Renew Energy*. 1991; 1: 199–205p.
[4] Kattakayam TA, Srinivasan K. Thermal performance characterization of a photovoltaic driven domestic refrigerator. *Int J Refrig*. 2000; 23: 190–196p.
[5] Lazzarin RM. Solar cooling: PV or thermal? A thermodynamic and economical analysis. *Int J Refrig*. 2014; 39: 38–47p.
[6] Taha AZ. The oversizing method of estimation in PV systems. *Renew Energy*. 1995; 6: 487–490p.
[7] Toure S, Fassinou WF. Cold storage and autonomy in a three compartments photovoltaic solar refrigerator: experimental and thermodynamic study. *Renew Energy*. 1999; 17: 587–602p.
[8] Kaplanis S, Papanastasiou S. The study and performance of a modified conventional refrigerator to serve as a PV powered one. *Renew Energy*. 2006; 31: 771–780p.
[9] Hartmann N, Glueck C, Schmidt FP. Solar cooling for small office buildings: comparison of solar thermal and photovoltaic options for two different European climates. *Renew Energy*. 2011; 36(5): 1329–1338p.
[10] Huang BJ, Hsu PC, Wu MS, Ho PY. System dynamic model and charging control of lead-acid battery for stand-alone solar PV system. *Sol Energy*. 2010; 84: 822–830p.
[11] Khurmi RS, Gupta JK. *A Textbook on Refrigeration and Air Conditioning*. Eurasia Publishing House; 2006.
[12] Liu BYH, Jordan RC. The interrelationship and characteristic distribution of direct, diffuse and total solar radiation. *Sol Energy*. 1960; 4: 1–19p.
[13] Phiraphata S, Prommasa R, Puangsombut W. Experimental study of natural convection in PV roof solar collector. *Int Commun Heat Mass Transf*. 2017; 89: 31–38p.

Cite this Article: Suhas B. G., Kavadiki Veerabhadrappa, Kiran Kumar U. Theoretical Investigation on Transient Heat Transfer through Building for PV-powered Solar Summer Air Conditioning. *International Journal of Thermal Energy and Applications.* 2019; 1(2): 42–52p.