*Energy Convers. Mgmt* Vol. 24, No. 4 pp. 305 - 311, 1984
Printed in Great Britain. 

0196-8904/84 \$3.00+0.00


# EXPERIMENTAL VALIDATION OF A THERMAL MODEL OF AN EVAPORATIVE COOLING SYSTEM*

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(*Received* 29 *June* 1983)

**Abstract**—A periodic thermal model for an evaporative cooling system over the roof has been presented. Open roof pond, water film and flowing water layer are the the special cases of the analysis. The time dependency of solar radiation, ambient air, sol-air and room air temperatures has explicitly been taken into account by expressing as a Fourier series of time for a 24 h cycle. Experimentally observed air temperature of rooms, treated with and without evaporative cooling over the roof, has been found in good agreement with theoretical results.

Solar energy    Passive system    Evaporative cooling

## NOMENCLATURE

$b$ = breadth, m
$c$ = specific heat, J/kg°C
$c_w$ = specific heat of water, J/kg°C
$c_{air}$ = specific heat of air, J/kg°C
$d$ = thickness of water column, m
$h$ = heat transfer coefficient, W/m$^2$ °C
$k$ = thermal conductivity, W/m°C
$L$ = length of water path, m
$M_W$ = heat capacity of water, J/m$^2$ °C
$M_R$ = heat capacity of room air, J/°C
$\dot{m}$ = mass flowing per unit time, kg/s.
$n$ = number of harmonic
$p$ = partial pressure of water vapour at temperature $T$, N/m$^2$
$\dot{Q}(t)$ = amount of heat flux, W/m$^2$
$S(t)$ = intensity of solar radiation, W/m$^2$
$T_i$ = inlet water temperature assumed as $T_A(t)$
$T_A(t)$ = ambient air temperature, °C
$T_W$ = water temperature, °C
$u_0$ = water flow rate, m/s
$v_0$ = wind velocity, m/s
$x$ = position coordinate, m
$X$ = area, m$^2$
$\alpha$ = absorptivity of surface
$r$ = relative humidity
$\theta(x,t)$ = temperature distribution °C
$\rho$ = density, kg/m$^3$
$\rho_w$ = density of water, kg/m$^3$
$\omega$ = $2\pi$/period, s$^{-1}$
$\epsilon$ = emissivity of surface
$\Delta R$ = difference between long wave radiation incident on the surface from sky and surroundings and the radiation emitted by a black body at ambient air temperature
$\tau_1$ = fraction of solar radiation absorbed by water
$\sigma$ = Stefan–Boltzmann constant, $5.6697 \times 10^{-8}$ W/mK$^4$
$\epsilon_w$ = emissivity of water surface
$\tau_2$ = fraction of solar radiation absorbed by roof surface
$\phi_{1n}, \phi_{2n}$ = constants, equation (14)

*Subscripts*

$R$ = roof
$S$ = South wall
$W$ = West wall
$N$ = North wall
$E$ = East wall
$F$ = floor
$o$ = outside
$i$ = inside
$f$ = to floor
$f_i$ = outside of floor to ambient
$r$ = radiative
$e$ = evaporative
$c$ = convective
$D$ = door

*Superscripts*

$A$ = ambient
$R$ = roof
$S$ = South wall
$W$ = West wall
$N$ = North wall
$E$ = East wall
$F$ = floor

## 1. INTRODUCTION

The authors are engaged in the design of passive buildings appropriate for the hot and dry climate of North India. As a part of this effort, it was considered desirable to develop thermal models of various passive heating/cooling approaches in the building. The conventional approaches for the reduction of heat flux into the building through the roof (as about 60% of the heat flux comes through the roof in common designs at locations between 0 to 35° latitude) are: provision of insulation and/or false ceiling, shading of roof due to vegetable pergola, use of removeable canvas and the use of reflective paints over the roof. An unconventional and highly effective approach to the problem consists of utilization of evaporative cooling over the roof; this can be achieved by an open pond or thin film or flow of water over the surface of the roof.

Mainly steady state thermal models are available for the prediction of heat flux across the roof with an evaporative cooling system. These analyses do not consider the time dependence of meteorological

* The work is partially supported by the Department of Science and Technology, Government of India.

conditions and enable one to calculate only the average heat transfer through the fabric without considering the storage effect of building masonry (U-value method). In the present analysis we have considered the time dependency of solar intensity, ambient air, sol-air and room air temperatures by expressing their parameters as Fourier series of time over a 24 h cycle.

An experimental study of the indoor air temperature of a room with evaporative roof cooling treatment has also been performed. Besides the roof which is studied, the heat gain/loss through different walls or floor has also been considered and the aggregate of their performances, which enables one to find the explicit expression for room air temperature. It is found that the observed room air temperature is in good agreement with the numerical results theoretically obtained (Fig. 4.). The open roof pond, water film and flowing water over the roof surface are the special cases of the model developed.

## 2. ANALYSIS

The schematic sketch of the system is shown in Fig. 1. The temperature distribution in the various walls, roof and floor is governed by the one dimensional equation of heat conductance viz.

$$k \frac{\partial^2 \theta_j}{\partial x^2} = \rho c \frac{\partial \theta_j}{\partial t} \tag{1}$$

For parameters relevant to common building materials, the characteristic length of the system is less than 15 cm; hence, if one considers the transverse dimensions to be larger than 50 cm, the heat conduction equation can be assumed one dimensional.

The temperature distribution in the various regions may be expressed as [1]

$$\theta_j(x,t) = A_{1j} + A_{2j}x + \mathrm{Re} \sum_{n=1}^{\infty} \{C_{jn} \exp(\alpha_{jn} x) + D_{jn} \exp(-\alpha_{jn} x)\} \exp(in\omega t) \tag{2}$$

where

$$\alpha_{jn} = -\alpha_j (1 + i) \sqrt{n}$$

and

$$\alpha_j = \left(\frac{\omega \rho_j \bar{c}_j}{2k_j}\right)$$

$A_{1j}$, $A_{2j}$, $C_{jn}$ and $D_{jn}$ are constants to be determined by appropriate energy balance conditions. Regions I, II, III and IV are between $(0 \le x \le x_1)$, $(x_1 \le x \le x_2)$, $(x_2 \le x \le x_3)$ and $(x_3 \le x \le x_4)$ of the masonry, respectively.

The energy balance for surfaces whose one side is exposed to outside environment and the other being in contact with room air, may be expressed as

$$-k \frac{\partial \theta}{\partial x}\bigg|_{x=0} = h_0 \left[\theta_{SA}(t) - \theta\bigg|_{x=0}\right] \tag{3}$$

and

$$-k \frac{\partial \theta}{\partial x}\bigg|_{x=X} = h_i \left[\theta(x=X,t) - T_R\right] \tag{4}$$

where

$$\theta_{SA} = \frac{\alpha_s S(t)}{h_0} + T_A - \frac{\epsilon \Delta R}{h_0}$$

commonly known as sol-air temperature.

In the case of multilayered wall/roof (Fig. 1), the energy balance conditions can be expressed by assuming continuity of heat flux and temperature across the interfaces, viz.

$$\theta_j (x = X_1,t) = \theta_{j+1} (x = X_1,t) \tag{5}$$

and

$$-k_j \frac{\partial \theta_j}{\partial x}\bigg|_{x=X_1} = -k_{j+1} \frac{\partial \theta_{j+1}}{\partial x}\bigg|_{x=X_1} \tag{6}$$

Based on these boundary conditions, one can write the expressions for heat flux coming into the room through the exposed walls/roof in terms of solar intensity, ambient and room air temperatures. However, if the floor is in contact with the ground, the additional boundary condition can be written in the form

$$\theta (x \to \infty, t) \text{ is finite} \tag{7}$$

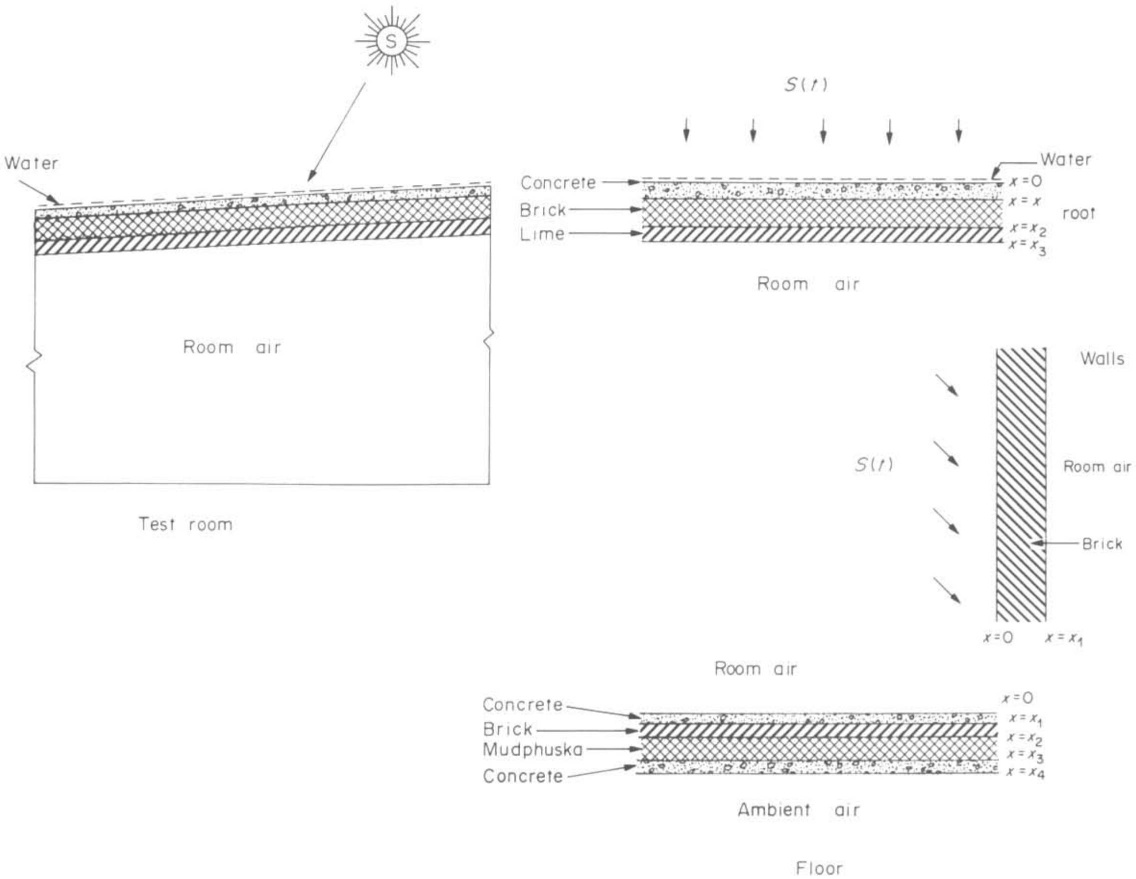
If the floor is not in contact with the ground, or in contact with the ambient air (Fig. 1), the energy balance conditions may be expressed as

$$-k \frac{\partial \theta}{\partial x}\bigg|_{x=X_1} = h_f \left[T_R(t) - \theta(x = X_1, t)\right] \tag{8}$$

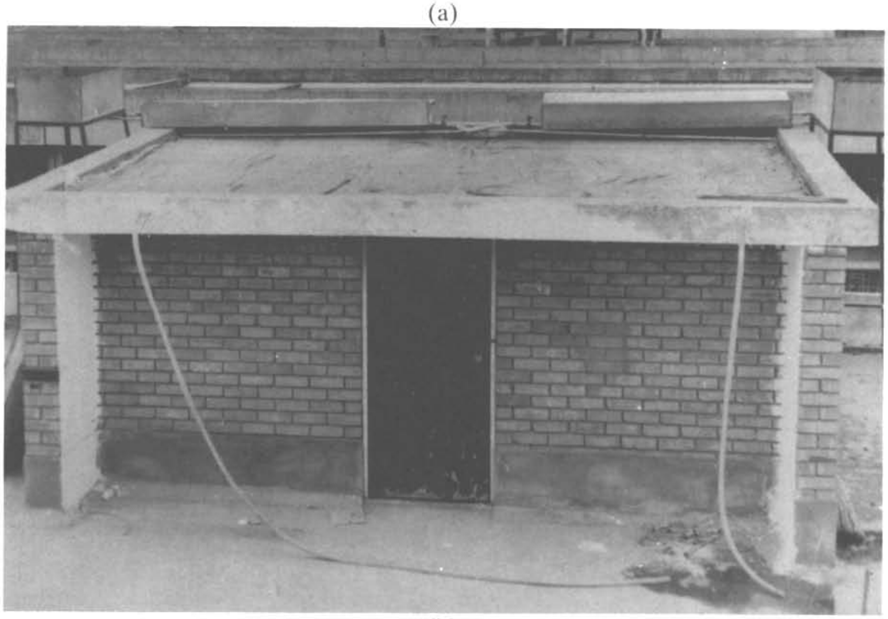
and

$$-k \frac{\partial \theta}{\partial x}\bigg|_{x=X_4} = h_{fi} \left[\theta(x = X_4, t) - T_A(t)\right] \tag{9}$$

For the periodic nature of meteorological parameters, one can write Fourier series expansions for a 24 h cycle, for solar intensity, ambient air, room air and sol-air temperatures in the form




(a)



(b)

Fig. 1. (a) Schematic view of the room, roof, walls and floor. (b) Photographic view of the roofs with and without cooling system.

$$S(t) = S_O + \mathrm{Re} \sum_{n=1}^{\infty} S_n \exp(in\omega t) \tag{10}$$

$$T_A(t) = T_{AO} + \mathrm{Re} \sum_{n=1}^{\infty} T_{An} \exp(in\omega t) \tag{11}$$

$$T_R(t) = T_{RO} + \mathrm{Re} \sum_{n=1}^{\infty} T_{Rn} \exp(in\omega t) \tag{12}$$

$$\theta_s(t) = \theta_{SO} + \mathrm{Re} \sum_{n=1}^{\infty} \theta_{Sn} \exp(in\omega t) \tag{13}$$

where $\omega = 2\pi$/period. Therefore, the heat flux entering the room through different walls can be expressed as

$$\dot{Q}_w(t) = X\left\{u(\theta_{SO} - T_{RO}) + Re \sum_{n=1}^{\infty} h_i [\phi_{1n}\theta_{sn} - \phi_{2n} T_{Rn}] \exp(in\omega t)\right\} \quad (14)$$

where

$$\frac{1}{u} = \frac{1}{h_0} + \sum_j \frac{X_j}{K_j} + \frac{1}{h_i}$$

and $\phi_{1n}$, $\phi_{2n}$ are constants to be determined by energy balance conditions.

Following Tiwari *et al.* [2], the heat flow through the roof on which an evaporative cooling system is maintained, can be expressed as follows. If the flowing water layer is of thickness $d$, the energy balance condition for water moving over the roof along the $y$ direction (Fig. 2) can be expressed as

$$\left(bd\,\rho_w c_w \frac{\partial T_w}{\partial t} + \dot{m}_w c_w \frac{\partial T_w}{\partial y}\right) dy$$

$$= [\tau_1 S(t) - Q_r - Q_c - Q_e + h_1 (\theta_1 \Big|_{x=0} - T_w)]\, bdy \quad (15)$$

where

$$h_1 = \frac{K_w}{L_w}\left[0.14\,(\text{Gr.P})^{1/3} + 0.644\,(\text{Pr})^{1/3}\,(\text{Re})^{1/3}\right] [4]$$

$$Q_r = h_r (T_W - T_A)$$

$$Q_c = h_c (T_W - T_A)$$

$$Q_e = 0.013\, h_c\, [p(\bar{T}_w) - \gamma p(\bar{T}_A)]$$

$$h_r = \epsilon_w \sigma\, [(\bar{T}_W + 273.15)^4 - (\bar{T}_A + 261.15)^4] / (\bar{T}_W - \bar{T}_A)$$

$$h_c = 5.678\,(1.0 + 0.85\,\Delta v). \quad [5]$$

$$\Delta v = v_o - u_o \quad (16)$$

$\bar{T}_W$ and $\bar{T}_A$ are, respectively, the average values of $T_W$ and $T_A$; Gr, Pr and Re are Grashof, Prandtl and Reynolds numbers, respectively.

In a narrow temperature range, the saturation vapour pressure of water ($p$) can be expressed by a linear relation of the form [6]

$$p(T) = R_1 T + R_2$$

where $R_1$ and $R_2$ are two constants to be determined from the saturation vapour pressure data [7] by a least square curve-fitting method. Hence, equation (15) can be expressed as

$$M_w \frac{\partial T_w}{\partial t} + \dot{m}_w c_w \frac{\partial T_w}{\partial t} = b\,H\,(T_s - T_w) + b\,h_1\,(\theta_1 \Big|_{x=0} - T_w) \quad (17)$$

where

$$T_s = \frac{1}{H}[\tau_1 S(t) + H_1 T_A(t) - R_0 R_2 (1-r)]$$

$$H = h_r + h_c + R_0 R_1$$

$$H_1 = h_r + h_c + \gamma R_0 R_1$$

$$R_0 = 0.013\, h_c$$

and $M_w = b.d.\, c_w.\rho_w$

Equation (17) is a general energy balance equation which can be simplified by putting

(i) $\dot{m}_w = 0$ for open roof pond and (ii) $M_w = 0$, $\dot{m}_w = 0$ for water film, spray, gunny bag system.

The water temperature ($T_w$) as a function of $y$ can be obtained from equations (17) with the initial boundary condition for water temperature viz.

$$T_w = T_i \text{ at } y = 0 \quad (18)$$

The temperature of water can also be assumed a periodic function of time as

$$T_w(y,t) = T_{wo}(y) + \text{Re} \sum_{n=1}^{\infty} T_{wn} \exp(in\omega t) \quad (19)$$

The amount of heat flux entering the room through the roof with an evaporative cooling system can be expressed as

$$\dot{Q}(y,t) = h_i [\theta \Big|_{x=X_4} - T_R] \quad (20)$$

Assuming a uniform surface temperature along the $y$-direction, the mean heat flux (averaged over $y$) into the room can be obtained as

$$\dot{Q}_{\text{roof}}(t) = X_R \frac{1}{L}\int_0^L \dot{Q}(y,t)\, dy$$

$$= X_R h_i \left[A_{14} X_4 + A_{24} - T_{RO}\right] + h_i \text{Re} \sum_{n=1}^{\infty} X_R \left\{ C_{4n} \exp(\alpha_{4n} X_4) + D_{4n} \exp(-\alpha_{4n} X_4) - T_{Rn}\right\} \exp(in\omega t) \quad (21)$$

where $A_{14}$, $A_{24}$, $C_{4n}$ *and* $D_{4n}$ are unknown constants and can be determined from the energy balance conditions. The interaction of room air to ambient air through infiltration and ventilation can be expressed as [8]

$$\dot{Q}(t)_{\text{inf/vent}} = V_0^A + V_1^A (T_R - T_A) \quad (22)$$

where the $V$'s are defined in Appendix I.

The heat transmission through a door can be expressed as

$$\dot{Q}_D(t) = h_D\,(T_R - T_A)\,X_D \tag{23}$$

Opening the door increases the number of air changes per second, and this effect is included in the $\dot{Q}(t)_{\text{inf/vent}}$ term.

## 3. ENERGY BALANCE FOR INSIDE ROOM AIR

Now, one can write the energy balance equation for room air temperature, which is mainly composed of components like heat flux entering the room through walls, roof and door and air changes due to infiltration/ventilation, viz.

$$M_R \frac{dT_R}{dt} = \dot{Q}_w(t) + \dot{Q}_R + \dot{Q}_D(t) - \dot{Q}_F(t) - \dot{Q}(t)_{\text{inf/vent}} \tag{24}$$

where the LHS represents the change in the heat content of room air. Using equations (10) to (13) and (15) to (21) in the energy balance condition from (3) to (9) one obtains

$$T_{Ro} = \Big[X_s U_s \theta^s_{so} + X_w U_w \theta^w_{so} + X_N U_N \theta^N_{so} + X_E U_E \theta^E_{so} + X^N_D h_D \theta^N_{so} + X_R U_R \theta^R_{so} + V^A_0 T_{Ao} + X_D h_D \theta^S_{so}\Big] \Big/ \Big[X_s U_s + X_w U_w + X_N U_N + X_F h_F + X_E U_E + X_R U_R + V^A_0 + X_D h_D\Big]$$

and $R_{Rn} = B_{2n}/B_{1n}$
where

$$B_{1n} = inwM_R + X_R \Phi^R_{1n} + X_s \Phi^s_{1n} + X_w \Phi^N_{1n} + X_N \Phi^N_{1n} + X_E \Phi^E_{1n} + V^A_1 + X_F \Phi^F_{1n} + X^N_D h_D + X^S_D h_D.$$

$$B_{2n} = X_R \Phi^R_{2n} \theta^R_{sn} + X_s \Phi^s_{2n} \theta^s_{sn} + X_w \Phi^w_{2n} \theta^w_{sn} + X_N \Phi^w_{2n} \theta^N_{sn} + X_E \Phi^E_{2n} \theta^E_{sn} + V^A_1 T_{An} + X^N_D h_D \theta^N_{sn} + X_F \Phi^F_{2n} T_{An} + X^s_D h_D \theta^s_{sn}$$

$T_{Ro}$ is the average and $T_{Rn}$ is the time dependent part of the room air temperature.

## 4. EXPERIMENT

To verify the analytical model, a simple experiment was performed for the stationary water film case. Two identical rooms of dimensions 5.2 × 2.5 × 1.72 m were constructed. The roof of both rooms is tilted slightly at an angle of 10°. Figure 1a schematically represents the cross-sectional view of the rooms. The roof is made of traditional construction (Fig. 1) composed of reinforced cement concrete (5 cm), brick (10 cm) and lime plaster (5 cm). The walls are made of brick of thickness 20 cm. The rooms are located on the roof of the terrace of the main building, so the floors of the experimental test rooms are composed of concrete-brick, mudphuska* and concrete exposed to ambient air on the other side.

The roof of one room was treated with the evaporative cooling system, while the other was untreated for comparison. Two G.I. pipes were spread over the upper end of the roof with tiny holes in it along the length of the room. One end of each pipe was connected from a constant level water tank. Water through tiny holes in pipes spreads over the roof. As the roof is slightly tilted at an angle due to gravitational force, the water spreads over the roof. The flow rate from the pipes was so controlled that it keeps the roof surface just wet.

As a matter of fact, after a few hours it was observed that an extra smooth surface is needed just to wet the roof by this free movement arrangement of water. To overcome the problem of dry spots over the surface, thin jute cloth was spread on the roof. It was observed that, with the help of jute cloth and pipes of tiny holes, a uniform water layer over the roof can be maintained. The reason behind this is the capillary action and ability of jute cloth/gunny bag to retain the water. Both rooms were exposed to the outside environment for a few days to attain the steady state condition. The temperatures of both rooms were measured by thermocouples, keeping the other end of the junction in an ice box. The thermo e.m.f. of the thermocouples were measured by a potentiometer of least count 0.1°C. The hourly variation of solar insolation and wind speed were measured by Kipp and Zonon pyranometer and an aminometer, respectively. The room air temperature and ambient air temperature were noted for 1 h intervals. The relative humidity of the outside air was measured with the help of dry/wet bulb thermometer.

The heat and mass transfer from wetted jute cloth is identical to that from a free water surface at the same temperature. Hence, by neglecting the heat capacity of jute cloth, the analytical model for stationary water layer case ($M_w = \dot{m}_w = 0$) is applicable in this system.

The photograph of the treated room is shown in Fig. 1b.

## 5. NUMERICAL RESULTS AND DISCUSSION

Numerical calculations, to verify the experimentally observed results, are performed. The variation of solar radiation on different walls are calculated from the data observed for the horizontal surface using Liu

* Mudphuska is a mud mortar mixed with hay in 35 kg/m$^3$ proportion.

and Jordan relations [9]. These values, then Fourier analysed for a 24 h cycle, and the corresponding Fourier coefficients are shown in Table I.

The following sets of parameters have been used in the calculations.

1. Thermal conductivity

| | Concrete | Brick | Lime plaster | Mud-phuska |
|---|---|---|---|---|
| Thermal conductivity (W/m°C)→ | 0.72 | 0.71 | 0.731 | 0.52 |
| Density (kg/m$^3$) → | 1858.0 | 1922.4 | 1446.0 | 2050.6 |
| Specific heat→ (J/kg°C) | 655.2 | 837.4 | 836.8 | 1840.0 |

2. Heat capacity of room air = 37328.3 J/°C

$h_0$ = 22.78 W/m$^{2}$°C(corresponding to average wind velocity 10.2 km/h).

$h_i$= 8.4 W/m$^2$°C for bare roof surface with heat flow downwards

= 8.3 W/m$^2$°C for treated roof surface with heat flow upwards

= 6.3 W/m$^2$°C for vertical walls.

For floor $X_1$ = 5 cm; $(X_2 - X_1)$ = 10 cm; $(X_3 - X_2)$ = 10 cm and $(X_4 - X_3)$ = 5 cm.

$h_F$ = 8.4 W/m$^2$°C for floor

and $h_D$ = 0.45 W/m$^2$°C for door.

The values of heat transfer coefficients have been calculated from relevant expressions given in Ref. [10].

3. Areas in (m$^2$)

$X_R = 15.43, X_S = 9.08,\ X_W = 4.3,\ X_N = 7.67$

$X_E = 4.3,\ X_F = 14.7,\ X_D^N = 1.85, X_D^S = 1.024.$

4. $L$ = 5.2 m

$b$ = 296 m

$d$ = 0.005 m (thickness of jute cloth)

$c_w$ = 4190.0 J/kg°C

$u_o$ = 0.0 m/s

$R_1$ = 325.17 N/m$^2$°C

$R_2$ = −5154.89 N/m$^2$

$\rho_w$ = 1000 kg/m$^3$

$\tau_2$ = 0.54

$\gamma$ = 0.27 (averaged over the day).

Figure 1 presents the schematic view of the system. There are two identical rooms, one treated with the evaporative cooling system while the other is untreated. Two doors of dimensions 1.85 and 1.04 m$^2$ are in the north and south walls, respectively. The roof, wall and floor is made of traditional construction except that the exposed roof surface is extra treated with reinforced cement. The rooms were fairly closed for a few days during the experiment and one air change per hour is assumed due to infiltration of air as suggested in Ref. [8].

Figure 2 presents the cross-sectional view of the moving water layer as depicted in Fig. 1.

Figure 3 presents the variation of partial vapour pressure with temperature; marked points are the experimental values obtained from steam tables [7]. The continuous line is plotted by a least square curve fitting method. It is reasonably good in the small temperature range of interest to express partial vapour pressure by a linear relation.

The variation of ambient and room air temperature with and without evaporative cooling system is shown in Fig. 4. It is seen that the variation of room air temperature without roof treatment is sufficiently high and even larger than the ambient air temperature during the day. It is due to the fact that there is more heat gain through the roof and walls and low heat losses to the environment during the sunshine period. Marked points (△) are the correspondingly observed values. Room air temperatures with the roof treated with the evaporative cooling system

Table 1. Fourier coefficients for daily variation of solar radiation on different surfaces and ambient air temperature on 4th June, 1981 at Delhi (Latitude 28°.5′ N). $S_n'$ is amplitude and $n$ is phase factor of solar radian

| Fabric | $n$ | 0 | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|---|---|
| Roof | $S_n'$ (W/m$^2$) | 307.16 | 470.6 | 191.37 | 10.172 | 21.121 | 4.62 | 12.84 |
| | $n$(radian) | | 3.17 | 0.1504 | 2.728 | 3.91 | 4.217 | 0.926 |
| South wall | $S_n'$ (W/m$^2$) | 78.76 | 117.33 | 41.98 | 5.65 | 6.64 | 11.48 | 4.26 |
| | $n$(radian) | | 3.113 | 0.006 | 4.252 | 0.5861 | 3.1 | 6.057 |
| West wall | $S_n'$ (W/m$^2$) | 146.18 | 224.19 | 128.98 | 106.31 | 62.22 | 24.13 | 13.958 |
| | $n$(radian) | | 3.654 | 1.453 | 5.562 | 2.994 | 0.591 | 4.180 |
| North wall | $S_n'$ (W/m$^2$) | 64.75 | 93.44 | 28.6 | 18.37 | 25.3 | 13.25 | 4.72 |
| | $n$(radian) | | 2.677 | 4.987 | 5.467 | 1.239 | 3.678 | 3.745 |
| East wall | $S_n'$ (W/m$^2$) | 162.7 | 249.17 | 141.08 | 105.65 | 50.86 | 9.601 | 1.701 |
| | $n$(radian) | | 2.499 | 4.574 | 0.315 | 2.683 | 4.586 | 4.039 |
| Ambient air temperature | $T'_{An}$ (°C) | 28.97 | 7.081 | 1.156 | 0.396 | 0.22 | 0.217 | 0.33 |
| | $An$(radian) | | 3.724 | 0.181 | 0.732 | 2.512 | 4.586 | 5.174 |

$S_n = S_n' \exp(-i_n)$ and $T_{An} = T'_{An} \exp(-i_{An})$

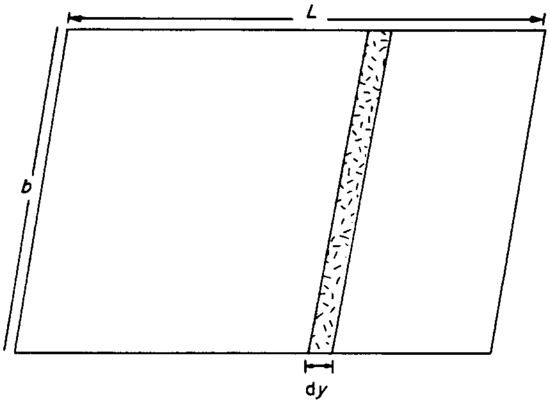



Fig. 2. Cross-sectional view of the moving water layer over the roof.

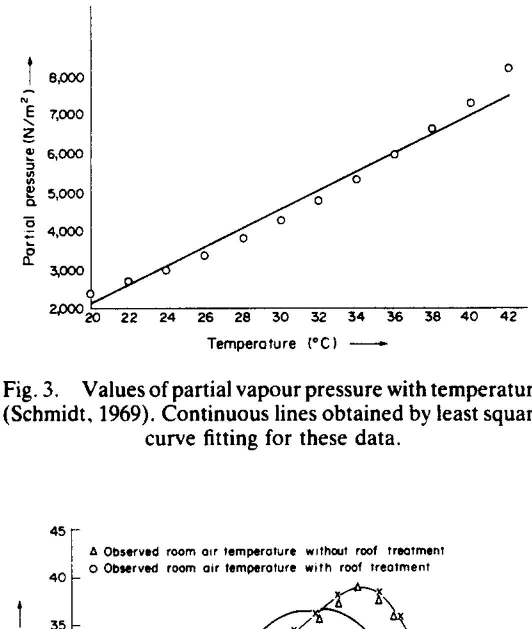



Fig. 3. Values of partial vapour pressure with temperature (Schmidt, 1969). Continuous lines obtained by least square curve fitting for these data.

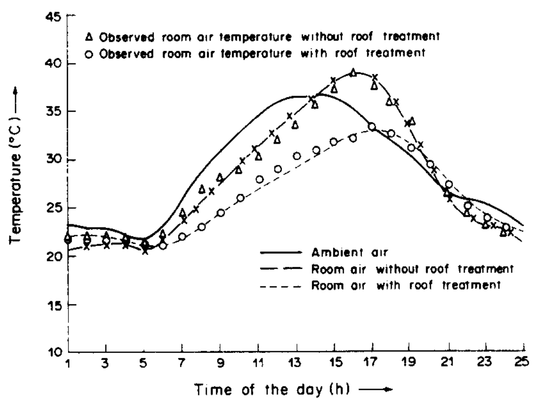



Fig. 4. Variation of ambient air (——) and theoretically calculated rooms air temperatures with (----) and without (– – –) evaporative cooling system over the roof. Marked points (⊙) and (∇) show the correspondingly observed room air temperatures.

obtained by present theory is shown by curve (----). It is seen that the room air temperature is substantially lower than that of the other room with the untreated roof. As a matter of fact, due to free evaporation of water over the roof, the roof ($x=0$) surface temperature significantly decreased even below the room air temperature. Hence, the roof loses heat to the environment rather than gaining. However, if it is decided to use the evaporative cooling system, it is good to construct the roof with some better conducting material to enhance the heat transfer to the environment from room air. Some attention should also be given to reduce heat transfer through the walls and due to air changes in the room.

Therefore, it may be concluded that the periodic theory successfully evaluates the heat transfer through (a) roof with evaporative cooling, and (b) different fabrics.

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## APPENDIX I

The values of $V$'s are

$$V_O^A = 2463\ M_R\ (N^A + N_O^A)\ .\ \Delta R_H/c_{\text{air}}$$

and

$$V_1^A = M_R\ (N^A + N_O{}^A)\ (c_{\text{air}} + 1.88\ \Delta R_H)/C_c$$

where $N^A$ is the number of air changes per hour due to the door ventilation and $N_o^A$ that due to the opening of window; $\Delta R_H$ is the difference in relative humidity between inside and outside. The superscript A refers to air changes from room to ambient air.