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PASSIVE COOLING TECHNIQUE FOR TROPICAL SUMMER CONDITIONS

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1. INTRODUCTION

Evaporative cooling of direct and indirect type is one of the most effective and energy effecient passive method of keeping the room air temperature within comfortable range in hot and dry climatic regions. The most popular and in-expensive evaporative cooler used in India during hot summer months are drip type coolers. The basic principle of direct evaporative cooling is that when unsaturated air is exposed to a wet surface, some of the sensible heat of the air is utilized as latent heat for the evaporation of water. In this process, the temperature of the air decreases while the humidity of the air increases; the lowest temperature to which the air can be cooled by this process is obviously the wet bulb temperature.

The feeling of comfort is a resultant effect of dry bulb temperature, air movement and humidity level. The effect of all these factors have been combined into one factor to define a comfort index namely Tropical Summer Index (TSI). The significance of TSI for tropical summer condition in Central and Northeren India has been established by systematic studies [1].

The dry and wet bulb temperature of the air in a room fitted with an evaporative cooler depends on the interaction of the cool air from the cooler with the heat flux coming into the room from various components i.e. walls, roof, windows etc. In the present paper an analysis has been developed to study the overall effect of a room cooler on the TSI in a normal sized room (3m x 6m x 3m) which is exposed to the ambient conditions from all sides. Numerical calculations have been made for a typical summer day in New Delhi (Lat. 28° 55) to study the effect of the various parameters of the cooler (such as the packing factor, pad area and air flow rate) on the hourly variation of TSI. Optimised values of these parameters, which will maintain the TSI in thermal comfort range (25°C - 30°C), have also been determined.

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2. ANALYSIS

2.1. Rate of Thermal Flux Removal By An Evaporative Cooler

When the air at dry bulb temperature T_a and wet bulb temperature T_w passes through the pads of a drip type evaporative cooler, the temperature T_o of the air coming out of the cooler is given by [2,3].

$$(T_o-T_w)/(T_a-T_w) = \exp[(-h_D A_p F_p)/\dot{m}] \qquad (1)$$

where h_D is the mass transfer coefficient, A_p is the pad area, F_p is the packing factor (F_pA_p gives the net wet surface area in the pads) and $\dot{m}$ is the mass flow rate of the air. The rate at which heat is removed by the cooler from a room at temperature T_R is given by

$$\dot{Q}_o = \dot{m}\ C_R\ (T_R - T_o) \qquad (2)$$

C_R being the specific heat capacity of the moist air; assumed to be constant. With the help of Eq.(1), $\dot{Q}_o$ can be written as

$$\dot{Q}_o = \dot{m}\ C_R[(T_R-T_w)-(T_a-T_w).\exp[(-h_D A_p F_p)/\dot{m}]] \qquad (3)$$

Writing $\dot{m} = \rho A_p V_p = \rho A_F V_F$ and

$$h_D = (5.50+1.65\ V_p)\times 10^{-3}\ \mathrm{kg.m^{-2}\,sec^{-1}}* \qquad (4)$$

where ρ is density of the air, V_p is the velocity with which air moves though the pads and A_F, V_F are the fan area and fan velocity respectively, we get:

$$\dot{Q}_o = \rho A_F V_F C_R[(T_R-T_w)-(T_a-T_w).\exp[-(1.65+5.50A_p/A_F V_F)(F_p/1000\rho)]] \qquad (5)$$

2.2. Periodic Heat Flux Through A Building Element

Following Sodha et.al. [5], the periodic heat flux into a room as a result of conductron, direct radiation and internal heat load can be written as

$$\dot{Q}_R = \sum_{m=1}^{6} A_m h_m^i (\theta_m|_{x=x_m} - T_R) + A_w S(t) + Q_{int} \qquad (6)$$

Where the first term on the RHS corresponds to the heat flux conducted into the room from six sides of the room envelope, the second term corresponds to the direct radiation entering through the window area and the third term corresponds to the internal heat generation.

*[4] L.D.Berman (1961).

For the environmental conditions which vary periodically with periodicity of 24 hrs., the solar radiation, ambient temperature, room air temperature can be assumed to be given by a Fourier series expansion of the type.

$$f(t) = \sum_{n=0}^{\infty} f^n \exp(in\omega t) \qquad (7)$$

For the observed daily variation on a normal day first six hamonics are found to be adequate for the convergence of the series.

For periodically varying environmental conditions the temperature distribution of the m^{th} building component is given by the periodic solution of the one dimensional heat conduction equation i.e.

$$\theta_m = (C_m^o\ x + D_m^o) + \sum_{n=1}^{\infty} [C_m^n \exp(\beta_m^n x) + D_m^n \exp(-\beta_m^n x)]\exp(in\omega t) \qquad (8)$$

Where $\beta_m^n = (in\omega/\alpha_m)^{1/2}$, α_m being the diffusivity of the m^{th} component and the constants C_m^n and D_m^n are determined from the boundary conditions.

$$-k_m(\partial\theta_m/\partial x)|_{x=0} = h_m^o(T_m^{SA} - \theta_m|_{x=0}) \qquad (9)$$

$$\text{and} \quad -k_m\ (\partial\theta_m/\partial x)|_{x=x_m} = h_m^i(\theta_m|_{x=x_m} - T_R), \qquad (10)$$

T_m^{SA} being the Solair temperature corresponding to the m^{th} building component, X_m being its thickness and h_m^o and h_m^i are the heat transfer coefficients on its outside and inside surfaces respectively. Solar radiation falling on each wall of the room has been calculated from the available radiation data on horizontal surface [6] by using Klucher formula [7]. The gain of heat through ground from the semi-infinit ground can be calculated by simply replacing h_m^o by $k_g\beta_g^n$ for the corresponding harmonic frequency [8]; k_g being thermal conductivity of the ground.

2.3. Energy Balance For A Room Fitted With An Evaporative Cooler

The energy balance equation for the air of the room fitted with an evaporative cooler will essentially consists of terms contained on the right hand side of Eq. (6). In addition, however one has to include the terms corresponding to the coolness created by the evaporative cooler. The complete equation can be written as

$$M_R C_R(\partial T_R/\partial t) = \sum_{m=1}^{6} h_m^i A_m(\theta_m|_{x=x_m} - T_R) - \rho A_F V_F C_R[(T_R-T_w)-(T_a-T_w).$$
$$\exp[-(1.65+5.5A_p/V_F A_F)F_p/(\rho\ 1000)]]$$
$$+\dot{Q}_{int} + A_w S\ (t) \qquad (11)$$

The term on left hand side corresponds to the net rate of energy change in room air, while on the right hand side the first term corresponds to the convective heat transfer from the inside surfaces, the second terms corresponds to the ventilation term maintained by the cool air provided by the cooler and the third and fourth terms correspond to the internal heat generation and the direct radiation respectively. In the present calculations however the last two terms have not been considered.

With their periodic representations of T_m^{SA}, T_R and θ_m given by Eq.(7) and Eq.(8) substituted in Eqs.(9) to (11), the unknown parameters including the room temperature T_R can be calculated.

2.4. Humidity Ratio And The Air Movement In The Room

Apart from the temperature of the room air, the other parameters of importance in determining the comfort conditions are the humidity of room air and its movement in the room. The humidity ratio W_o of the air coming out from the cooler is given by the relation [3].

$$(W_s-W_o)/(W_s-W_a) = \exp(-h_D A_p F_p/\dot{m}) \qquad (12)$$

Where W_a and W_s are the humidity ratios of the ambient air and the corresponding saturated air respectively.

Substituting for h_D from Eq.(4) in Eq. (13) one gets

$$W_o=W_s+(W_a-W_s)\exp[-(1.65+5.50\ A_p/(A_F V_F)\ F_p/(\rho 1000)] \qquad (13)$$

The rate at which the water vapour is added to the room air from the humidified air coming from the cooler is given by the relation.

$$A_F\ V_F\ (W_o - W_R) = V_R\ dW_R/dt \qquad (14)$$

Where W_R is the humidity ratio of the room at any given time t and V_R is the volume of the room.

The solution of eq.(14), subject to the intial condition that $W_R=W_{Ri}$ at t=0, is obtained as

$$W_R = W_o + (W_{Ri}-W_o)\exp[-t\ A_F V_F/V_R] \qquad (15)$$

The parameter $V_R/A_F V_F (=\tau)$ is the relevant time constant equivalent to the time of one air change. For a 3m x 6m x 3m sized room and fan air flow rate of 0.125m s, τ =7.2 min. While considering the hourly variations therefore, W_o itself can be taken as the humidity ratio W_R of the room air.

The air movement velocity V_C can be estimated from the relation $A_F\ V_F = A_C\ V_C$ where A_C is the cross sectional area of the room across the direction of air from the fan.

2.5. Thermal Comfort Conditions

For tropical summer conditions, Sharma [2] evolved the concept of Tropical Summer Index (TSI) as the thermal comfort index; this index is given by:

$$TSI = 0.308\ T_{Rw} + 0.745\ T_R - 2.06\ V_C^{\frac{1}{2}} + 0.841 \qquad (14)$$

Where T_{Rw} is the wet bulb temperature of the room air. It was found that for Indian subjects, a value of TSI ranging between 25°C to 30°C create comfortable condition. The values of T_R and W_R obtained from sections 2.3 and 2.4, can be used to obtain T_{Rw} with the help of ASHRAE Psychrometric charts. The values of T_{Rw}, T_R and V_C thus obtained can then be used to calculate TSI from the above relation.

3. RESULTS AND DISCUSSION

Dimensions of the various elements of the room and values of different building parameters used for the clculations are given in Table 1.

TABLE 1 Values of Different Parameters Used For Calculating Room Temperature T_R.

Size of Room	- 3m x 6m x 3m with long axis in the E-W direction
Thickness of Walls/Roofs	= 20 cm
Thermal conductivity of the Walls Material	= 1.099 $wm^{-1}\,°K^{-1}$
Thermal conductivity of the Roof Material	= 2.216$wm^{-1}\,°K^{-1}$
Specific Heat of the Walls Materials	= 800 J/Kg °C
Specific Heat of the Roof Materials	= 800 J/Kg °C
Specific Heat of Water	= 4200 J $Kg^{-1}\,°C^{-1}$
Specific Heat of Air	= 1008 J $Kg^{-1}\,°C^{-1}$
Density of Walls Material	= 1920.0 Kg m^{-3}
Density of Roof Material	= 2288.0 Kg m^{-3}
Latent Heat of vaporation of water	= 2402 K.J.kg^{-1}
Absorptivity of the walls and Roofs Outer surface	= 0.6
Heat Transfer coefficient on Outer Surface of Walls/Roof	= 19.0 W $m^{-2}\,°C^{-1}$
Heat transfer coefficient on Inner Surfaces of Walls/Roof	= 6.27 W $m^{-2}\,°C^{-1}$

It is seen from Eq. (5), that the cooling performance of the cooler depends on the packing factor F_p, fan power $A_F V_F$ and pad area A_p. It is also evident from this equation that the cooling performance decreases for both small and large values of $A_F V_F$, suggesting that for a given value of A_p and F_p, there exists an optimum value of $A_F V_F$ for which the cooling performance of the cooler is maximum. To appreciate this effect, calculations were first made for only average environmental conditions of a representative day of May. (i.e. S = 303 W/m^2, T_a = 31.2°C and relative humidity=31.9%). Fig 1 shows the effect of $A_F V_F$ on TSI for different combinations of A_p and F_p. As expected, it is seen that TSI shows a minimum and the value of this minimum decreases with the increasing values of A_p and F_p;. and that for larger values of A_p and F_p, the minimum is obtained at higher values of the fan power.

The hourly variation of the various input parameters (S, T_a and T_w for a representative day of May) and different response parameters (T_o T_R and TSI) for the value of $A_F V_F$ for which the average performance of the cooler is optimum ($A_F V_F$ = 0.45 m^3/s for A_p = 3.0 m and F_p = 25.0), is shown in Fig. 2. The hourly variation of the room temperature in the absence of the evaporative cooler, has also been plotted.

Except for a small duration i.e. between 14.00h and 18.00h, the TSI inside the room is seen to be well with in the comfort range. Even during this period, the maximum deviation from the comfort range is only 0.5°C.

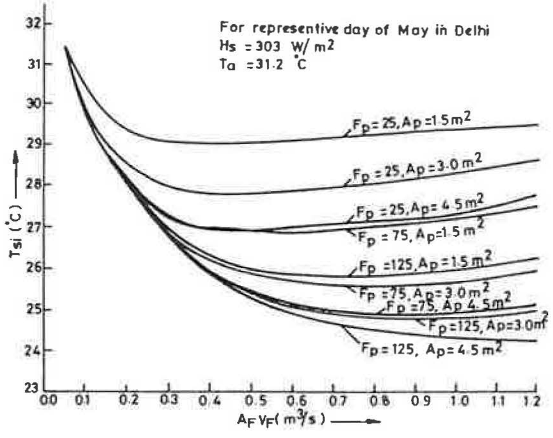

Fig.1. Variation of TSI with air flow rate of fan A_F V_F for different combination of F_p and A_p.

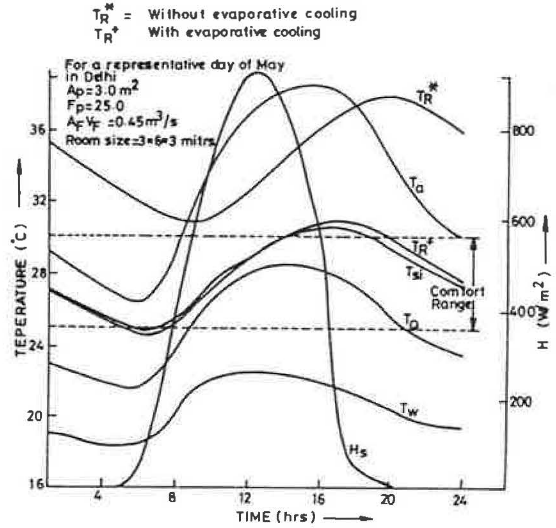

Fig. 2. Hourly variation of the metereological parameters (S, T_a and T_w) and the response parameters (T_R, T_o TSI).

4. CONCLUSIONS

Use of an evaporative cooler for maintaining thermal comfort inside buildings during hot dry summer months in tropical climates needs an optimisation of various parameters aporopriate for different type and sized buildings. The analysis developed in this paper allows the determination of the optimum values of these parameters alongwith the calculation of the Tropical Summer Index.

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