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MOISTURE EVAPORATION AND MIGRATION IN THIN POROUS PACKED BED INFLUENCED BY AMBIENT AND OPERATING CONDITIONS

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SUMMARY

The present paper investigates moisture migration in a thin porous bed filled with unconsolidated sand, unsaturated with water, and examines its cooling effect by water evaporation when used as a cooling device for room air-conditioning. An analytical model has been developed to simulate heat and moisture transport phenomena numerically and calculate the evaporation rate of water on which its cooling performance is dependent. For the case of a horizontal thin bed problem, with very small height and a relatively larger surface that is exposed to atmosphere air, one-dimensional, steady-state computation results have been obtained by focusing on the influence of ambient and operating conditions on the physical quantity fields in the porous packed bed.

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KEY WORDS: unsaturated porous media; thin packed bed; free evaporative cooling; moisture migration; heat transfer

1. INTRODUCTION

Porous materials are widely used in many aspects of engineering, agriculture and environmental protection because the mechanisms of mass and energy transport in those media are closely associated with various physical phenomena and practical problems that are eagerly expected to be explained or solved. Compared with a porous bed saturated with a single fluid, an unsaturated bed is much more complicated in its transportation processes as two-phase flow, heat and mass transfer with phase changes and other complex mechanical characteristics are involved in the pore spaces. Though some typical theories have been well established by Philip and DeVries (1957), Luikov (1966, 1975) and Whitaker (1977, 1980) and many important applications worked out in recent years by authors such as Vafai and Whitaker (1986), Vafai and Tien (1989), Cheng and Pei (1989), Udell (1983) and so on, the mathematical models need to be improved and many application areas remain to be explored.

Theoretically and experimentally, free evaporative cooling in or on an unsaturated porous packed bed that works as a part of roofs or walls in buildings is a highly promising project due to its engineering background and theoretical interests, but appear to have been little considered by researchers. This method of cooling may be used in designing a useful air-conditioning device that could be installed in private houses and public buildings to transfer heat flux from rooms to ambient air, thereby reducing the temperature of the room by the free evaporation effect in pore spaces and on the surface exposed directly to atmosphere. A remarkable advantage in acceptance of this simple device, instead of a conventional air-conditioner with thermodynamic cycle, is that first, no electricity is consumed to drive flowing media, secondly, only water energy at low grade

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is needed as its energy source and thirdly, moving parts are not necessarily involved when operating it. The study of cooling performance for horizontal beds when working as a roof (or part of it) and vertical beds when working as a wall (or part of it) of the building in the hot and dry summer, therefore, are of great importance to energy conservation.

It is worth noting that Sodha *et al.* (1989) studied evaporative cooling of a liquid water layer on a cinema roof with an air-conditioning system. By the cooling effect of direct evaporation on the free surface of the water layer, the indoor temperature was decreased on the basis of operating an electrical air-conditioning system with a thermodynamic cycle. As a result of adopting this supplementing means of evaporation cooling, the capacity of the air-conditioner could be reduced by 19%, and an energy saving of 16% achieved.

But Sodha's idea needs to be developed further, as some problems remain unsolved: first, the water has to be maintained on the roof of the building in the form of a thin layer, because a thick water layer is not advantageous to thermal conduction, which dominates the transport process, to transfer heat from the bottom side of the layer to the upper evaporating surface; secondly, the amount of evaporating water and moisture migration are limited, for water evaporation only occurs on the free surface of the water layer; thirdly, the manner of evaporation cooling of a single-phase liquid possibly may not be applied to the situation of the walls of a building, since it may be difficult to maintain a thin water layer on a vertical surface.

Thus, the application of evaporation cooling by a thin porous packed bed unsaturated with water is introduced to be able to overcome the shortcomings of Sodha's cooling method. A comparison between the two methods will be given later.

2. MATHEMATICAL MODEL

In the present study, a porous bed packed with unconsolidated sand is horizontally designed as a thin square with a metal plate as its bottom side which contacts the room air (see Figure 1) and an uncovered surface as its upper open side. Water enters the bed from a water tank by adjusting the height potential. The open surface is exposed to ambient air, in which ambient relative humidity, temperature, wind velocity and other climate conditions may change, and the bottom metal surface acts as the ceiling of the room, to which heat transfer enhancements, for instance a surface with fins or a fan to yield convective air flow, cold be made. When driven by meteorological data, all field gradients of physical quantities in the packed bed, such as temperature, water content, velocities of liquid and gaseous mixture, pressure, etc., begin to change, and moisture migration and heat transportation take place.

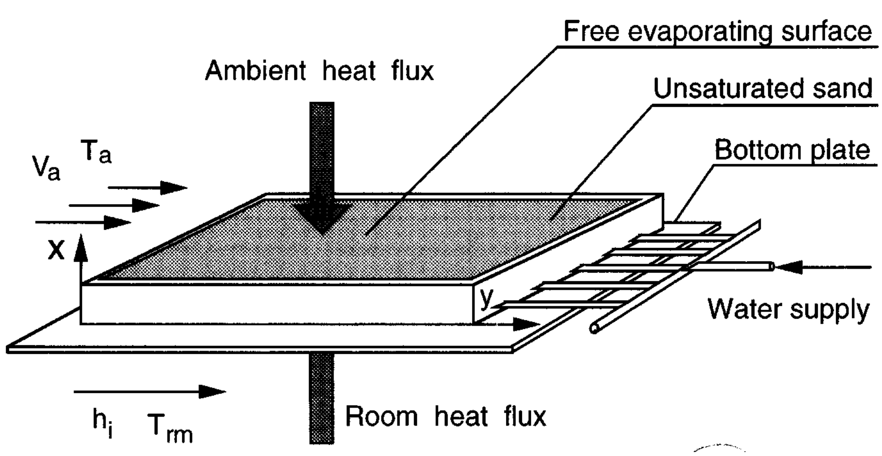

Figure 1. Construction of the porous packed bed

As the results of evaporation affect the free surface, the temperature of the upper boundary decreases to the lowest level in the porous layer and thermal gradients influence moisture transfer, which also affects heat flux. Inside the packed bed, however, the main driving forces for moisture motion are from capillary and molecular diffusion mechanisms, which help to bring the evaporation effect on the free boundary surface to the interior of the sand bed, so that ambient relative humidity plays an important role in determining water infiltration flow and water vapour diffusion. The temperature difference between the bottom plate and the room air could reach a significant level that generates a cooling effect for the room, depending on the ambient conditions. In order to evaluate the cooling performance of the evaporating bed, other parameters, for example the ambient and room temperatures, the wind velocity and the heat transfer coefficient of the metal surface, should also be focused on.

The steady-state mathematical model presented here in the coordinate system shown in Figure 1 is a general expression for the porous packed bed unsaturated with water to describe the simultaneous migration of heat, moisture and a gas mixture of air and vapor, excluding the effect of water vapour diffusion within the pores (Zhang *et al.*, 1993a, b). These are as follows:

$$\frac{\partial}{\partial x}(\rho_1 \varepsilon_1 u_1) + \frac{\partial}{\partial y}(\rho_1 \varepsilon_1 v_1) = -\dot{m} \tag{1}$$

$$\frac{\partial}{\partial x}(\rho_g \varepsilon_g u_g) + \frac{\partial}{\partial y}(\rho_g \varepsilon_g v_g) = \dot{m} \tag{2}$$

$$u_1 \frac{\partial u_1}{\partial x} + v_1 \frac{\partial u_1}{\partial y} - \frac{\dot{m}}{\rho_1 \varepsilon_1} u_1 = -\frac{g D_1}{K_1}\frac{\partial \varepsilon_1}{\partial x} - \frac{g}{K_1}(\varepsilon_1 u_1 - \varepsilon_g u_g) + \nu_1 \left(\frac{\partial^2 u_1}{\partial x^2} + \frac{\partial^2 u_1}{\partial y^2}\right) - g \tag{3}$$

$$u_1 \frac{\partial v_1}{\partial x} + v_1 \frac{\partial v_1}{\partial y} - \frac{\dot{m}}{\rho_1 \varepsilon_1} v_1 = -\frac{g D_1}{K_1}\frac{\partial \varepsilon_1}{\partial y} - \frac{g}{K_1}(\varepsilon_1 v_1 - \varepsilon_g v_g) + \nu_1 \left(\frac{\partial^2 v_1}{\partial x^2} + \frac{\partial^2 v_1}{\partial y^2}\right) \tag{4}$$

$$u_g \frac{\partial u_g}{\partial x} + v_g \frac{\partial u_g}{\partial y} + \frac{\dot{m}}{\rho_g \varepsilon_g} u_g = -\frac{1}{\rho_g}\frac{\partial P}{\partial x} - \frac{g}{K_g}(\varepsilon_g u_g - \varepsilon_1 u_1) + \nu_g \left(\frac{\partial^2 u_g}{\partial x^2} + \frac{\partial^2 u_g}{\partial y^2}\right) - g\beta(T - T_{cw}) \tag{5}$$

$$u_g \frac{\partial v_g}{\partial x} + v_g \frac{\partial v_g}{\partial y} + \frac{\dot{m}}{\rho_g \varepsilon_g} v_g = -\frac{1}{\rho_g}\frac{\partial P}{\partial y} - \frac{g}{K_g}(\varepsilon_g v_g - \varepsilon_1 v_1) + \nu_g \left(\frac{\partial^2 v_g}{\partial x^2} + \frac{\partial^2 v_g}{\partial y^2}\right) \tag{6}$$

$$\rho_1 \varepsilon_1 c_1 \left(u_1 \frac{\partial T}{\partial x} + v_1 \frac{\partial T}{\partial y}\right) + \varepsilon_g c_a \left(u_g \frac{\partial}{\partial x}(\rho_a T) + v_g \frac{\partial}{\partial y}(\rho_a T)\right) = \frac{\partial}{\partial x}\left(k_m \frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k_m \frac{\partial T}{\partial y}\right) - \dot{m} \times \gamma \tag{7}$$

In the partial differential equations (1)–(7), the relative movement between liquid phase and gaseous phase is included in the Darcy resistance term, which is in balance with other forces such as the capillary force, the viscosity resistance, gravity, buoyancy, the pressure gradient, etc., in the process of unsaturated infiltrating flow of the fluid under a temperature gradient.

We have applied the present model to the following physical and geometrical conditions.

(1) The porous packed bed is installed horizontally with its height H much less than its length L and width W ($H < 0{\cdot}05$ m, $L > 2$ m, $W > 2$ m).
(2) The upper surface of the bed is exposed to a uniform atmosphere environment, which means the driving conditions for the evaporation cooling are the same over the whole of the free surface.
(3) The lower metal plate (made of steel or copper) contacting the room air is also regarded as an uniform boundary which is shared by the bottom sand layer totally saturated with liquid water of 3–5 mm in height.
(4) The water can be supplied continuously by maintaining a constant water potential with a water tank.

For the thin porous bed mentioned above, the regions of boundary layer flow and heat transfer affected by vertical walls are so small that only the migration of heat and moisture in the x-direction need be considered and a one-dimensional numerical simulation is accurate enough to express the changes of field quantities in it. Under this restriction, the mathematical model can be simplified by omitting the viscosity and inertia terms, but retaining the mass inertia term relating to the evaporation rate from phase change, which is important for unsaturated problems. Hence, the two-dimensional model reduces to

$$\frac{\mathrm{d}}{\mathrm{d}x}(\rho_l \varepsilon_l u_l) = -\dot{m} \tag{8}$$

$$\frac{\mathrm{d}}{\mathrm{d}x}(\rho_g \varepsilon_g u_g) = \dot{m} \tag{9}$$

$$\left(\varepsilon_l - \frac{\dot{m} K_l(\varepsilon_l)}{\rho_l \varepsilon_l g}\right) u_l = -D_l(\varepsilon_l)\frac{\mathrm{d}\varepsilon_l}{\mathrm{d}x} - K_l(\varepsilon_l) + \varepsilon_g u_g \tag{10}$$

$$\left(\varepsilon_g - \frac{\dot{m} K_g(\varepsilon_g)}{\rho_g \varepsilon_g g}\right) u_g = -\frac{K_g(\varepsilon_g)}{\rho_g g}\frac{\mathrm{d}P}{\mathrm{d}x} - K_g(\varepsilon_g)\beta(T - T_{cw}) + \varepsilon_l u_l \tag{11}$$

$$(\rho_l \varepsilon_l c_l u_l + \rho_g \varepsilon_g c_g u_g)\frac{\mathrm{d}T}{\mathrm{d}x} = \frac{\mathrm{d}}{\mathrm{d}x}\left(k_m \frac{\mathrm{d}T}{\mathrm{d}x}\right) - \dot{m}\times\gamma \tag{12}$$

Some nonlinear terms appear in the momentum equations, which can be handled by finite difference methods and all velocity variables involved are real velocities in each individual phase. The corresponding boundary conditions for the controlling equations can be written as:

$x = 0$:

$$P = P_a \tag{13}$$

$$\varepsilon_g = \varepsilon_{g,i} \tag{14}$$

$$-k_m \frac{\mathrm{d}T}{\mathrm{d}x} + \varepsilon_l \rho_l c_l u_l T = h_i(T_{rm} - T) \tag{15}$$

$x = H$:

$$P = P_a \tag{16}$$

$$\varepsilon_l \rho_l u_l + \varepsilon_g \rho_v u_g = h_m(\rho_{vs} - \rho_{v\infty}) \tag{17}$$

$$-k_m \frac{\mathrm{d}T}{\mathrm{d}x} + (\varepsilon_l \rho_l c_l u_l + \varepsilon_g \rho_v c_v u_g)T = h_0(T - T_a) + (\varepsilon_l \rho_l u_l + \varepsilon_g \rho_v u_g)\gamma - q_r \tag{18}$$

Concerning the physical properties, data for the unsaturated hydraulic conductivity K_l and water diffusivity in the porous medium D_l are used from experimental curves (Jury and Miller, 1974) and the relations $K_l - \varepsilon_l$ and $D_l - \varepsilon_l$ are fitted as the following formulae in the range of $\varepsilon_l = 0{\cdot}25$–$0{\cdot}38$:

$$K_l = 9{\cdot}48 \times 10^{12{\cdot}53\varepsilon_l - 10} \tag{19}$$

$$D_l = 5{\cdot}88 \times 10^{9{\cdot}18\varepsilon_l - 9} \tag{20}$$

Subject to continuous conditions for gas-phase flow, a Darcy resistance for the gaseous mixture is added, and a gaseous infiltrating conductivity K_g corresponding liquid hydraulic conductivity K_l, is defined to demonstrate the blocking mechanism of the porous matrix to the infiltrating flow of the mixture. To obtain the property data, we hoped to establish a relation between K_g and K_l, which has been formulated in equation (19). For this purpose, a relation for saturated properties has been derived according to Kozeny and Ergun's theory (Bear, 1972) assuming that solid particles in the porous matrix are spherical:

$$K_{s,g} = \left(\frac{\varepsilon_g}{1 - \varepsilon_s}\right)^3 \left(\frac{\varepsilon_s}{1 - \varepsilon_g}\right)^{4/3} K_{s,l} \tag{21}$$

By paying regard to the general relation between saturated and unsaturated status $K_s = (v/g)K$, we can obtain

$$K_g(\varepsilon_g) = \Gamma(S)K_l(\varepsilon_l) \tag{22}$$

in which a converting factor $\Gamma(S)$ has been derived as

$$\Gamma(S) = (1 - S)^3 \left(\frac{1 - \phi}{1 - \phi(1 - S)}\right)^{4/3} \frac{v_l}{v_g} \tag{23}$$

Where ϕ stands for porosity and S for saturation:

$$S = \frac{\varepsilon_l}{\varepsilon_l + \varepsilon_g} = \frac{\varepsilon_l}{\phi} \tag{24}$$

The apparent thermal conductivity for the unsaturated bed is to be satisfied with the pattern of parallel heat conductivity:

$$k_m = \varepsilon_s k_s + \varepsilon_l k_l + \varepsilon_g k_g \tag{25}$$

The molecular diffusion coefficient D_v is from Krischer–Rohnalter's expression in the work of Philip and De Vries (1957) and the density of saturated water vapour from the work of Mayhew and Rogers (1976):

$$D_v = 5{\cdot}893 \times 10^{-6} T^{2{\cdot}3}/P \tag{26}$$

$$1/\rho_{vs} = 194{\cdot}4 \exp[-0{\cdot}06374(T - 273{\cdot}15) + 0{\cdot}1634 \times 10^{-3}(T - 273{\cdot}15)^2] \tag{27}$$

Heat and mass transfer coefficients for the upper boundary are

$$h_m = h_o/(\rho_g c_g Le^{2/3}) \tag{28}$$

3. RESULTS AND DISCUSSION

Under the boundary conditions (13)–(18), one-dimensional, steady-state, numerical computations for the present mathematical equations have been carried out by finite difference methods adopting an upwind scheme for the convection term and a centred difference scheme for the diffusion term. The calculation is regarded to converge only if the temperature difference between the inner nodes in two iterative steps is less than 0·001°C. In order to ensure that every coupled quantity, such as temperature, velocity, phase content, evporation rate, pressure etc., matches well with the others in convergence rate, under-relaxation factors ranging from 0·2 to 0·8 are chosen for adjusting their values according to their changing rate in different iterative steps.

The standard values among the variational parameters in our calculation are on the basis of common climate conditions in the summer and include a constant coefficient of forced convective heat transfer in the lower boundary: RH = 60%, $T_a = 35$°C, $V_a = 3$ m s^{-1} and $h_i = 10$ W m^{-2} °C^{-1}. The reference height of the evaporating cooling bed under investigation is 25 mm and the calculated results for the influence of height on cooling effects are included in the paper.

The temperature profiles in the horizontal thin porous packed bed are shown in Figures 2–5. The migrations of heat and moisture in the unsaturated sand are dominated by atmospheric conditions that are mainly ambient relative humidity (measured by dry and wet bulb temperatures) RH, ambient temperature T_a and wind velocity V_a, and also by the extent of heat transfer enhancement on the metal bottom plate, which can be characterized by convective heat transfer coefficient h_i. When those parameters change with time, the temperature field in the bed changes also, which leads the cooling effect of the bed to vary under different situations.

Of all those conditions, RH is the most important influence on the movement of moisture in both the liquid and vapour states and the evaporation rate of water. During the dry summer season, the relatively lower temperature of the cooling bed is expected to produce a meaningful temperature difference between the

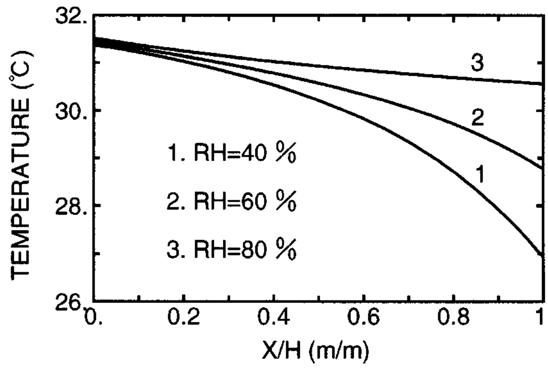

Figure 2. Temperature versus ambient relative humidity

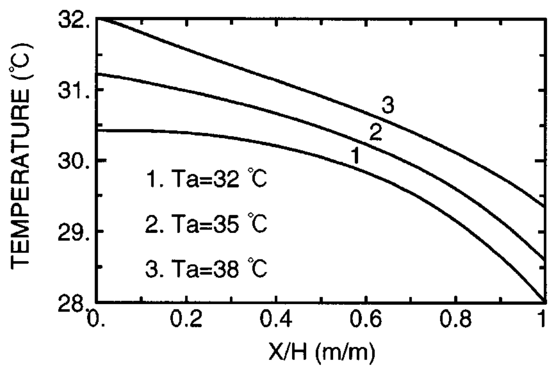

Figure 3. Temperature versus ambient temperature

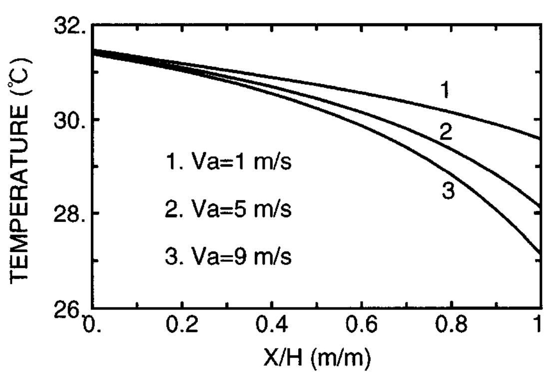

Figure 4. Temperature versus ambient wind velocity

bottom plate and the room air, so that it can work as effectively as possible as an alternative to air-conditioning or as a kind of supplement for the centralized air-conditioning system that may exist in a building. The lower the relative humidity, the better is the porous bed's effectiveness for cooling. T_a and V_a also have some influence on the transport of heat and mass in the bed, but the former's action in affecting the temperature field in the porous sand layer is more obvious than that of the latter. A relatively bigger change in temperature could be found at the point $(X/H) = 0$ in Figure 3, as we have set the ambient and

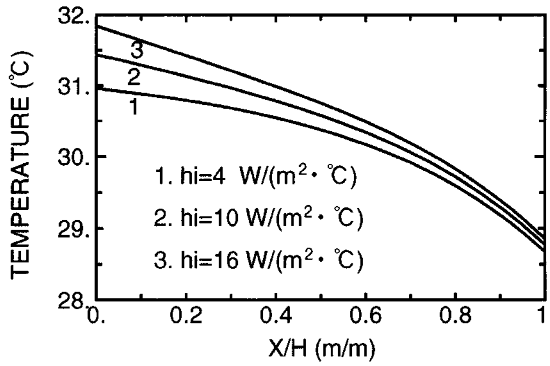

Figure 5. Temperature versus convective heat transfer coefficient

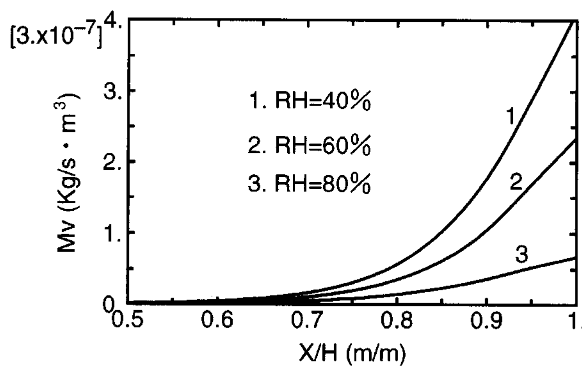

Figure 6. Evaporation rate of water versus ambient relative humidity

room temperatures to the same value in the calculation, which allows us to show the initial potential for effective cooling in a very hot summer. The higher the ambient temperature and the bigger the temperature difference between the cooling plate and the room air, the better is the cooling effect. As V_a increases, free evaporation is enhanced on the upper boundary surface, which is at a lower temperature than the bottom surface but is not regarded as affecting the temperature of the bottom plate directly, the water evaporation reduces to zero since water saturation $S = 1$ at this point. In the case of heat transfer enhancement on the metal plate contacting the room air, the plate temperature rises with increments in h_i, which is to say that more heat can be removed from the room. Thus the cooling bed is recommended for use at higher values of the convective heat transfer coefficient.

Figures 6–9 demonstrate the changes of evaporation rate of water with ambient and operating conditions. As displayed in Figure 6, when RH is somewhat larger, water evaporation becomes weak both in and on the porous packed bed, and M_v (i.e. $\dot{m}$) curves have a tendency to be flat when RH is over 80%. The same phenomenon can be detected in curve 3 of Figure 2, which illustrates a linear regularity. These data imply that the heat transfer process is dominated by thermal conduction, since moisture movement, both for convection and diffusion, is quite small at higher RH, which results in very little capillary and diffusive potential in the porous layer. From Figures 7 and 9, it can also be observed that the rate of water evaporation is not very sensitive to changes of T_a and h_i, but is sensitive to changes in V_a (see Figure 8). Though a large V_a can take a lot of evaporating mass away from the upper free surface on which the latent heat of phase change certainly increases for further lowering of the temperature, it makes much more ambient heat flux enter the bed too, from which the cooling effect does not benefit, so higher ambient wind velocity is not an optimum operating condition for the cooling bed.

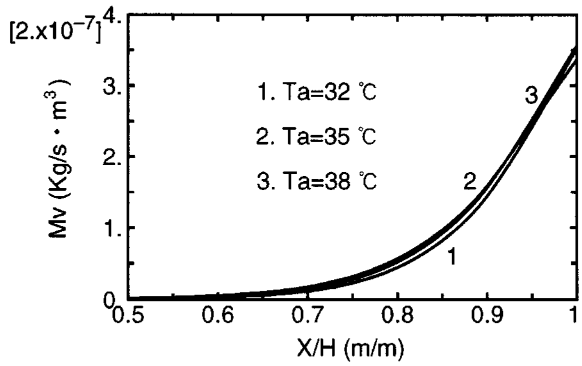

Figure 7. Evaporation rate of water versus ambient temperature

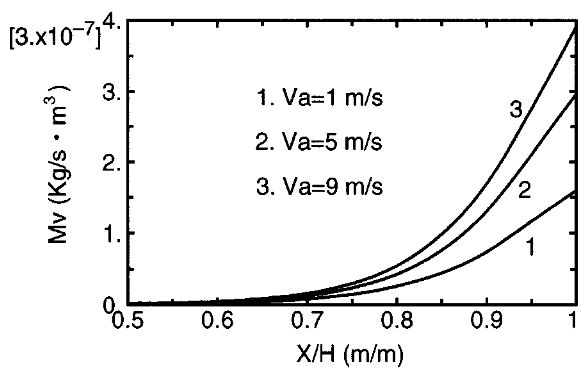

Figure 8. Evaporation rate of water versus ambient wind velocity

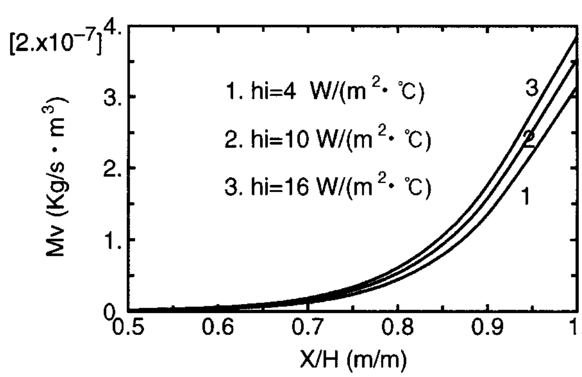

Figure 9. Evaporation rate of water versus convective heat transfer coefficient

The one-dimensional velocity distributions for the gas-phase are similar to the curves of evaporation rate, which are profiled in Figures 10–13. From these obvious convective movements can be observed. We have also studied how the height H, a construction parameter, influences the performance of the cooling bed. As shown in Figures 14–16, it seems that a relatively thinner bed is advantageous in improving cooling behavior, since the temperature of the metal bottom plate is lower when H declines. But any H less than 10 mm is not recommended because the sand bed may be saturated with water, which makes practical use difficult.

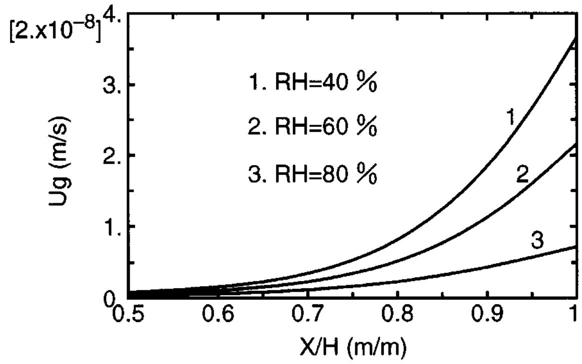

Figure 10. Gas-phase velocity versus ambient relative humidity

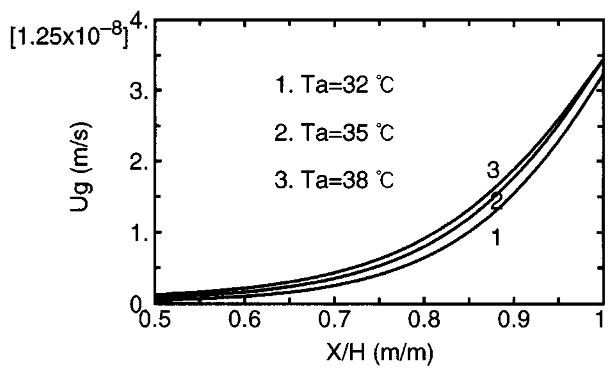

Figure 11. Gas-phase velocity versus ambient temperature

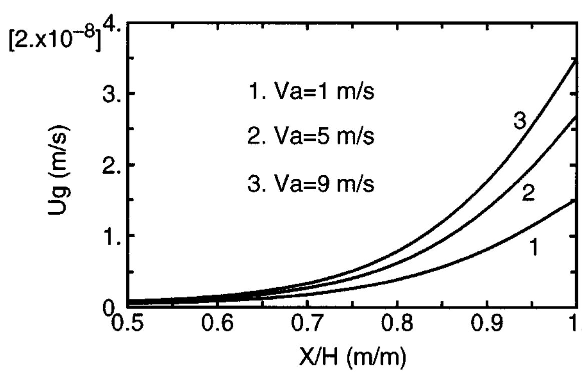

Figure 12. Gas-phase velocity versus ambient wind velocity

From the mathematical model calculations for the thin bed problem, two regions with an approximately central interface are numerically observable. The evaporating rate of water and the bulk velocity of the gas phase are almost constant with reduction of the relative humidity and remain approximately zero in the region $X/H < 0{\cdot}5$, since the porous packed layer in this part is nearly saturated with water, which is demonstrated by the curve of water content ε_l in Figure 17. Above this region, dominated by thermal

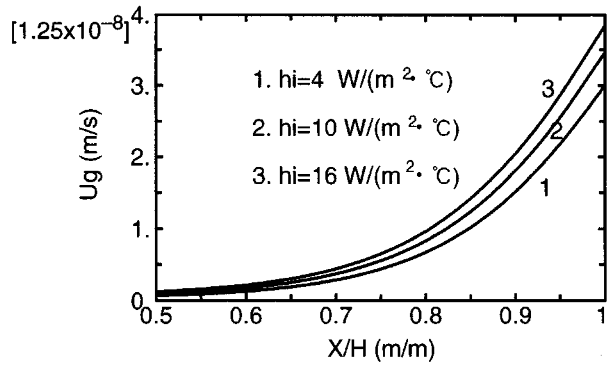

Figure 13. Gas-phase velocity versus convective heat transfer coefficient

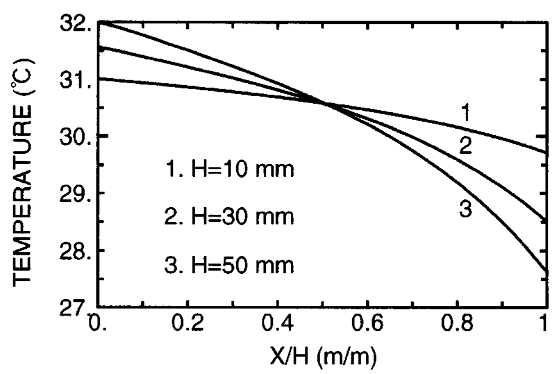

Figure 14. The influence of bed height on temperature field

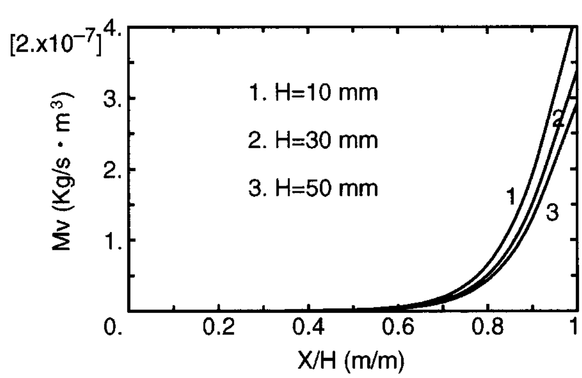

Figure 15. The influence of bed height on evaporation rate of water

conduction, evaporation and velocity curves show a nonlinear tendency, which explains that the transportation process of heat and moisture is dependent upon the phase change rate of water, gas and liquid phase convective flows and so on, as well as conduction. But it seems that ambient and operating conditions have very little influence on the velocity change of liquid water, so that all U_1 velocity curves almost coincide with each other.

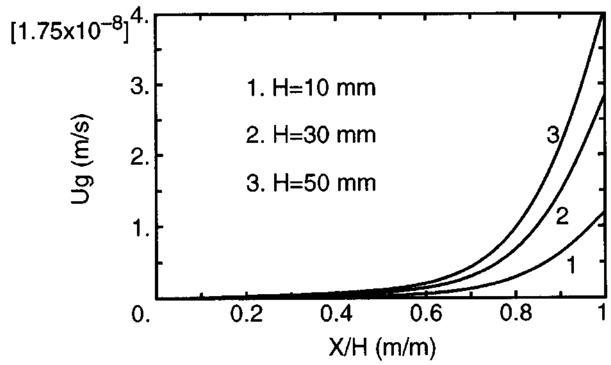

Figure 16. The influence of bed height on gas-phase velocity field

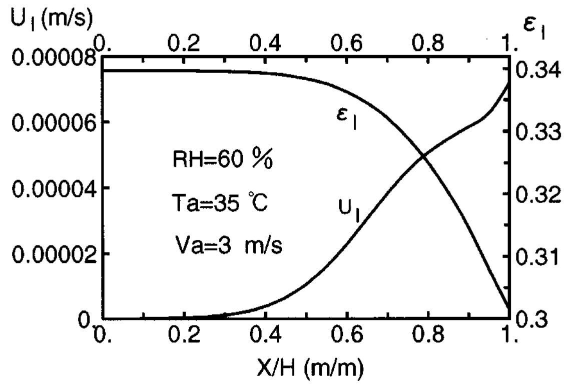

Figure 17. Liquid-phase velocity and water content in the porous bed

Table 1. Comparison of Q_i between Sodha's and present methods

Symbol (unit)	Ambient Value	Q_i (W m^{-2})	
		Sodha's	present
	40	41·83	66·10
RH (%)	60	26·87	52·20
	80	13·71	35·60
	30	25·32	50·96
T_a (°C)	35	26·87	52·20
	40	28·48	53·78
	1	25·71	51·22
V_a (m s^{-1})	3	26·87	52·20
	5	28·14	53·55
Standard values	RH 60	T_a 35	V_a 3

The cooling capacities illustrated in Table 1 indicate how effectively the cooling bed works, and some comparisons are made between the methods of Sodha's water layer cooling and present porous bed cooling. The heat load extracted from the room by effective evaporative cooling is largely dependent on ambient parameters, which are the initial driving conditions. If the evaporation is enhanced on the free surface, it will also be strengthened in the porous bed and the inner temperature gradient will change, so benefiting the capillary and diffusive action in the porous matrix; thus a greater heat flux can be absorbed from the room by the packed bed. This will result in an increased amount of evaporative cooling. The influence of ambient temperature and wind velocity on the cooling capacity are not as obvious as that of ambient relative humidity.

4. CONCLUSION

The concept of free evaporative cooling by a thin porous bed filled with unconsolidated sand unsaturated with water, which can be used as a cooling roof or wall for public buildings and private houses, has been developed. This simple device needs no electricity or moving parts when operated to generate significant air-conditioning effect for the room. It works by supplying liquid water as the energy source to maintain continuous water evaporation. Therefore, it is a very promising energy-saving device for room cooling in summer.

The mathematical model for this problem is formulated in terms of the physical properties of unsaturated porous sand. A Darcy resistance for gas-phase momentum equations is defined by introducing a new physical property called gaseous infiltrating conductivity, which is formulated in terms of liquid hydraulic conductivity and a converting factor depending on water saturation and kinematic viscosity of liquid and gas. This resistance mechanism is important for gas phase infiltration in the porous medium.

The boundary parameters, such as ambient relative humidity RH, ambient temperature T_a, wind velocity V_a and convective heat transfer coefficient h_i between the metal bottom plate and the room air, greatly influence changes in the inner physical quantities, especially temperature gradient and liquid water evaporation rate, but excluding liquid phase content and velocity.

When RH drops and T_a rises, evaporation both on the free surface and within the bed increases, which makes a big contribution to the change in temperature gradient that affects the evaporation. As a result, the heat load of the room is extracted by effective evaporative cooling of the porous packed bed. V_a seems to have little influence on the temperature of the metal bottom plate, but h_i does. A higher h_i value is recommended for operation of the cooling bed in a hot and dry climate.

NOMENCLATURE

c = specific heat (J kg^{-1} K^{-1})
D_l = diffusivity of water in porous materials (m^2 s^{-1})
D_v = molecular diffusivity of water vapour in air (m^2 s^{-1})
g = acceleration of gravity (m s^{-2})
h = convective heat transfer coefficient (W m^{-2} K^{-1})
h_m = convective mass transfer coefficient (m s^{-1})
H = vertical height of porous bed (m)
k = thermal conductivity (W m^{-1} K^{-1})
k_m = apparent thermal conductivity (W m^{-1} K^{-1})
K_g = infiltrating conductivity of gaseous mixture (m s^{-1})
K_l = hydraulic conductivity of water (m s^{-1})
L = horizontal length of porous bed (m)
L_e = α/D_v = Lewis number
$M_v, \dot{m}$ = mass rate of phase change per unit volume (kg m^{-3} s^{-1})
P = pressure (Pa)
q_r = solar radiation (W m^{-2})
Q = heat flux, heat load (W m^{-2})
RH = relative humidity (%)

S = saturation (%)
T = temperature (K, °C)
u = velocity component in the x-direction (m s^{-1})
v = velocity component in the y-direction (m s^{-1})
W = horizontal width of porous bed (m)

Greek letters
ε = phase content, emissivity (%)
ϕ = porosity (%)
ρ = density (kg m^{-3})
ν = kinematic viscosity (m^2 s^{-1})
γ = latent heat (J kg^{-1})
α = thermal diffusivity (m s^{-1})
β = thermal expansion coefficient (K^{-1})
Γ = converting factor defined in formula (11)

Subscripts
a = air, ambient
cw = cooling wall
g = gaseous mixture
i = inside surface
l = liquid water
o = outside surface
rm = room
s = solid, saturated
v = vapour of water
vs = vapour saturated
∞ = environment

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