studied and evaluated to determine the best combinations and thicknesses.

- The temperature in and around buildings can be tempered by the nature of the surrounding surfaces. Land forms and vegetation should be used to modify the microclimate. Studies have shown that pavements absorb 50% of the heat, bare ground absorbs 30%, while grass absorbs only 5% [6]. The use of plants provides shading for the building and reduces the effects of sandstorms.
- The use of shading devices prevents the direct sunshine on the building and results in the circulation of air around these devices which releases the absorbed heat and prevents its radiation inside the building. Special attention should be considered for shading all windows and entrances.
- Windows should be small and positioned carefully. Special glasses that filter out most of the sun's heat should be used. Solar radiation through glazed areas enters with little loss in heat energy. The heat is trapped and can increase the indoor temperature even above that outdoors. Reflected glare from the ground and other surfaces should be avoided.
- Orientation of the building is important in the effects of the climatic conditions. As the sun is either due east or west for most of the day, walls facing these directions should be smallest and windows should be limited on these walls.
- Earth-sheltered houses can reduce the energy cost. Underground structures in hot, arid regions have lower temperatures than conventional buildings, since the soil is a good insulating material: the temperature changes with depth very slowly.
- Evaporative air coolers evaporate water at ambient temperature into an airstream, so that the dry-bulb temperature is reduced along a line of constant wet-bulb temperature. This inexpensive means of cooling without mechanical refrigeration should find applications in the hot, arid environment of Kuwait.
- Use of efficient cooling systems. Systems should be designed to cool only used living spaces.
- Ventilation and air circulation can lower the effects of the high temperatures.
- The use of light colours on exterior walls reduces the absorption of heat as most of the sunlight is reflected.
- Air infiltration must be controlled, since the amount of heat entering a building as a result of infiltration can often be greater than the heat which comes through the materials; therefore, all openings should be airtight.
- Thermal competency of alternative designs of projects should be evaluated. The life-cycle costs of these alternatives should be used in the comparisons instead of the construction costs.
- Pricing of electricity should be revised to discourage waste of energy.
- Importing of materials should be limited to tested ones.

CONCLUSION

The true cost of the building is its cost over the period of its use and not its initial construction cost. It was shown that the operational cost of housing in the harsh hot, dry environment of Kuwait is very high. Therefore, it is important in the planning, design, and construction of housing to reduce the cost of air-conditioning operation, maintenance and installation, and the cost of maintaining the building due to the deterioration of its materials. Designs should be made with regard to the environment. The environmental aspects stated above should be considered to minimize the effects of a hot, arid environment on the building and its internal environment.

REFERENCES

1 *Achievement Data for 1984*, Ministry of Electricity and Water, Kuwait, 1985.
2 *Energy Conservation Programme*, Ministry of Electricity and Water, Kuwait, 1983.
3 Y. Al-Hajeri, The development of energy conservation in Kuwait, *Proc. First Conference on Energy Conservation, April 6 - 8, 1985, Kuwait*, Kuwait Engineering Society.
4 S. Fereig, Analysis of building costs in Kuwait, *Proc. International Congress on Housing: The Impact of Economy and Technology, November, 1981, Vienna, Austria*, Pergamon Press, Oxford, pp. 833 - 845.
5 O. Kayyali, Application of non-destructive methods in assessing concrete durability, *Proc. Third International Conference on the Durability of Building Materials and Components, August 12 - 15, 1984, Espoo, Finland*, Technical Research Centre of Finland (VTT), pp. 486 - 499.
6 A. Konya, *Design Primer for Hot Climates*, The Architectural Press Ltd., London, 1980.

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The Optimum Azimuth for a Solar Chimney in Hot Climates

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ABSTRACT

In a solar-heated chimney, used to promote air movement within a building, stored heat is the main source for warming air within the chimney. This creates a buoyancy pressure which draws air through the building. It is therefore important to study the effect of the orientation upon the amount of heat that can be collected by the chimney for better performance. A theoretical study was conducted using a finite difference technique which showed that the amount of heat that can be collected by a solar chimney is strongly dependent upon its azimuth.

1. INTRODUCTION

The cooling of dwellings is one of the major aims in hot climates. Several methods are encountered in traditional buildings, one of which is the use of air movement; wind towers and courtyard spaces are particularly common. Unfortunately, the air outside is sometimes so hot that ventilation is of no benefit until the evening. An arrangement as illustrated in Fig. 1 has been investigated [1, 2]. This could be used to promote evening ventilation, at which time the outside air is cool. This relies upon the collection within the fabric of a solar chimney of sufficient heat to induce cooling draughts through the building. Walls on the sunny side warm the air within the chimney which, when opened at the top and bottom, induces ventilation.

This work was aimed at determining the orientation at which the solar-heated chimney would give the best performance. Heat storage within a solar chimney at different azimuths for latitude 33°N was calculated for summer conditions, using a one-dimensional finite difference approximation.

2. ANALYSIS

2.1. Determination of solar radiation

Solar radiation reaches the outside wall of the chimney in three ways: radiation direct from the sun, diffused from the sky and reflected from the ground and surroundings, such as building surfaces. The intensity of solar radiation depends mainly on the latitude, the altitude above sea-level and sky clarity. The total solar radiation intensities on the outer wall of the solar chimney were determined, based on the radiation data published by CIBSE [3], as follows:

$$I_{tvd} = I_{Dvd} + 0.5(I_{dhd} + k_r I_{thd}) \qquad (1)$$

where the ground reflection factor k_r has a value of 0.5 for hot arid climates and 0.2 for other climates.

$$I_{Dvd} = k_{af} k_D I_{Dvb} \qquad (2)$$

The altitude correction factor, k_{af}, may be assumed as unity for altitudes between 0 and 300 m above sea-level. It can be calculated for other values from

$$k_{af} = 1.02 + 10^{-5}[2 + 5\,\mathrm{cosec}(h)]H \qquad (3)$$

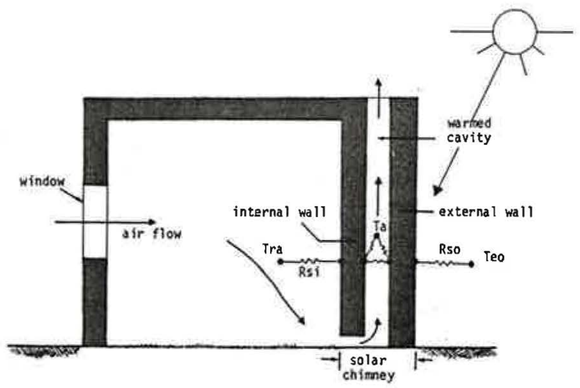

Fig. 1. Section through the room and 'solar chimney'.

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and the direct radiation factor value k_D is 1.1 for hot arid climates.

$$I_{dhd} = k_{af} k_d I_{dhb} \quad (4)$$

The diffuse radiation factor k_d has a value of 0.9 for arid climates.

$$I_{thd} = k_{af}(k_D I_{Dhb} + k_d I_{dhb}) \quad (5)$$

Equation (1) may be simplified, by substituting eqns. (2), (3), (4) and (5) into it and inserting the correction factors with their values, to the following form:

$$I_{tvd} = k_{af}(1.1 I_{Dvb} + 0.675 I_{dhb} + 0.275 I_{Dhb}) \quad (6)$$

It is apparent from Fig. 2, for latitude 33°N in summer that the peak total intensity received by a southern wall is 550 W/m^2 at noon, whereas a western wall at 16:00 receives 885 W/m^2.

2.2. Determination of the solair temperature

The determination of hourly solair temperature is important, because the heat transferred into the solar chimney depends upon the difference between the solair temperature and the outside surface temperature of the chimney. This may be obtained from the following equation:

$$T_{eo} = T_{ao} + R_{so}(aI_{tvd} - EI_e) \quad (7)$$

where the outside air temperature, T_{ao}, is obtained from the climatic data of El-Oued [4]. The outside total thermal resistance R_{so} is a combination of convective and radiative thermal resistances. From Fig. 3, the highest solair temperature for latitude 33°N on a west-facing wall is 77 °C and for a south-facing one is 55 °C.

2.3. Heat transfer

The walls of the solar chimney are designed to collect as much solar energy as possible. They can be built from any high thermal capacity building material. In this investigation, we have assumed a material with density 2000 kg/m^3, thermal conductivity 1.1 W/mK and specific heat capacity 900 J/kgK. The outer wall of the solar chimney is 0.1 m thick and the inner wall is 0.2 m thick.

The outside wall of the solar chimney receives solar radiation from the sky and the surroundings and has convective heat exchange with the ambient air. The outside total thermal surface resistance is taken as 0.05 m^{2}K/W. On the inner side of the room, the walls are assumed to be exposed to air at constant temperature and to be surrounded by surfaces at a mean radiant temperature equal to the air temperature. The convective heat loss up the chimney is considered small compared with radiative heat exchange between the two walls. These assumptions are made in order to simplify the analysis and are acceptable since the purpose of this study is to compare the magnitudes of heat storage within the walls of the chimney at different azimuths. The inside thermal surface resistance of the room is taken as 0.123 m^{2}K/W. The transient temperature is obtained by dividing the walls into slices.

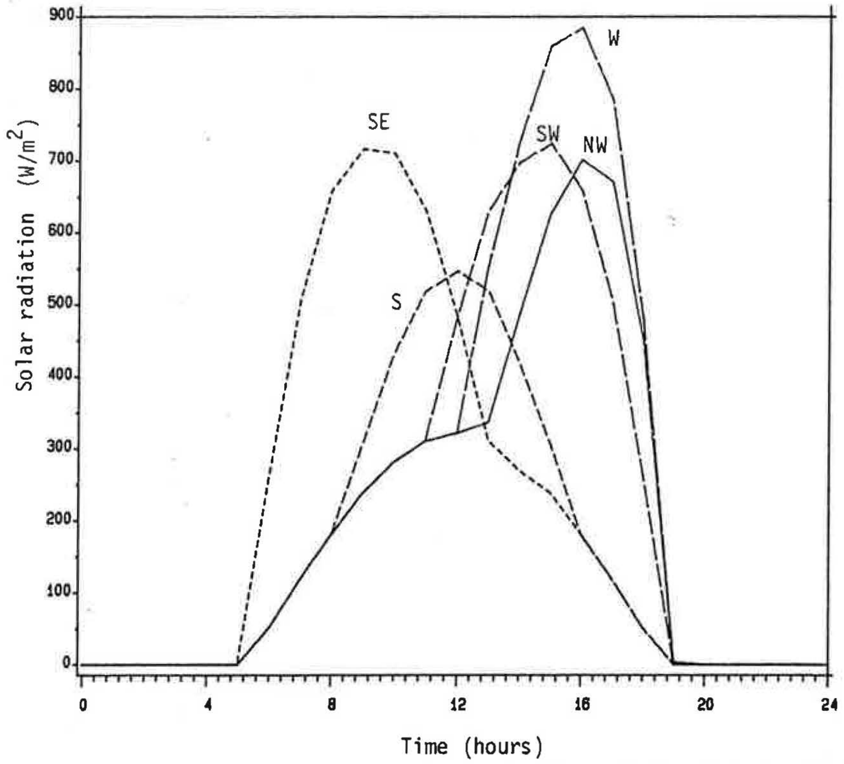

Fig. 2. Daily intensities of solar radiation received by vertical walls facing SE, S, SW, W, and NW at latitude 33°N.

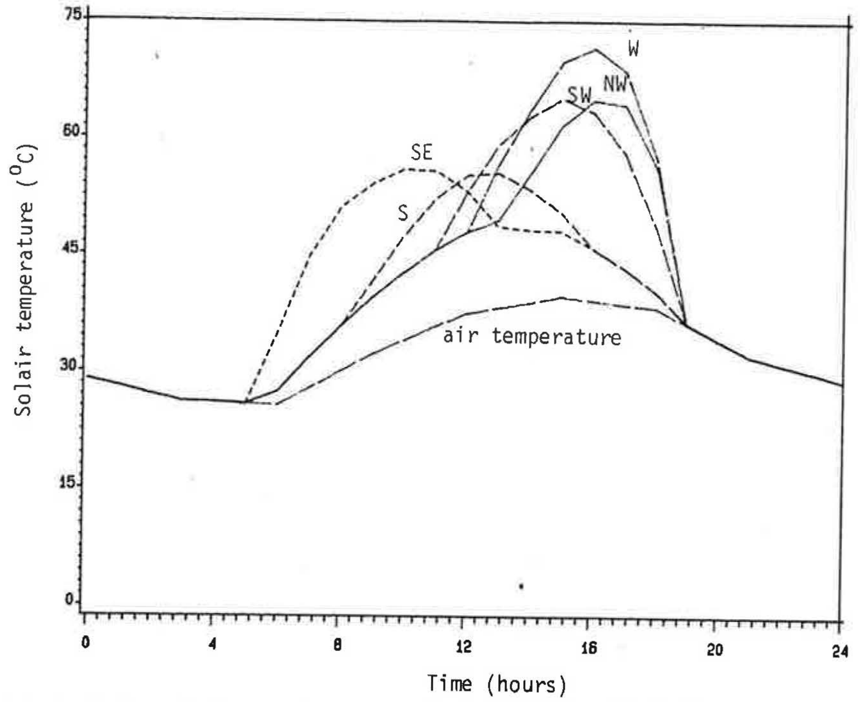

Fig. 3. Daily solair temperatures on vertical walls facing SE, S, SW, W and NW at latitude 33°N.

Assuming a one-dimensional heat flow through the walls, a heat balance equation may be established as follows:

Heat entering a node (n)

— Heat leaving that node

= Heat stored in it in a given time

which leads to:

$$T_{(n,\,t+1)} = \text{fact}_1 T_{(n-1,\,t)} + \text{fact}_2 T_{(n,\,t)} + \text{fact}_3 T_{(n+1,\,t)} \quad (8)$$

where fact$_1$, fact$_2$ and fact$_3$ are coefficients whose values are given as follows:

$$\text{fact}_1 = \frac{C_{(n-1,\,n)}\,\mathrm{d}t}{\rho\,\delta x\,c_p} \quad (9)$$

$$\text{fact}_2 = 1 - \frac{[C_{(n-1,\,n)} + C_{(n,\,n+1)}]\,\mathrm{d}t}{\rho\,\delta x\,c_p} \quad (10)$$

$$\text{fact}_3 = \frac{C_{(n,\,n+1)}\,\mathrm{d}t}{\rho\,\delta x\,c_p} \quad (11)$$

The stability and accuracy of the numerical solution depends upon the time and the distance increment, which should be chosen so that the time step

$$\mathrm{d}t \leqslant \frac{\rho\,\delta x\,C_p}{C_{(n-1,\,n)} + C_{(n,\,n+1)}} \quad (12)$$

For surface nodes the thickness of the slice is taken as half the actual slice.

The heat transfer within the chimney occurs by convection and radiation. The convective heat transfer coefficient may be obtained from the equation for a vertical flat plate given by Arpaci [5] as follows:

$$h_t = 0.1(GrPr)^{1/3} K_a/Z \quad (13)$$

where Gr and Pr are the Grashof and Prandtl numbers.

The radiation heat transfer coefficient, h_r, between two parallel walls is

$$h_r = 5.8 C_{F_{12}} \quad (14)$$

where $C_{F_{12}}$ is a configuration factor which depends on the emissivity, the areas and the relative view (or 'shape') factor F_{12} of the two surfaces.

$$C_{F_{12}} = \left[\frac{1}{E_1} - 1 + \frac{A_1}{A_2}\left(\frac{1}{E_2} - 1\right) + \frac{1}{F_{12}}\right]^{-1} \tag{15}$$

where the subscripts 1 and 2 refer to the first and second walls. F_{12} is the relative view factor obtained from O'Callaghan [6] and Wong [7] for example.

2.4. Determination of heat storage

The total stored heat within the walls of the solar chimney can be obtained by calculating the heat stored in each slice over a given time, dt, and then taking the total of all slices as follows:

$$H_{s(n)} = \rho c_p\, \delta x\, A(\overline{T}_{(n)} - T_r) \tag{16}$$

where

$$\overline{T}_{(n)} = (T_{(n-1,\,n)} + T_{(n,\,n+1)})/2 \tag{17}$$

so that the whole stored heat is

$$H_T = \sum_{1}^{n} H_{s(n)} \tag{18}$$

Figure 4 shows the energy stored by the solar chimney as a function of time in July for latitude 33°, and Fig. 5 shows the maximum energy plotted against azimuths. From Figs. 4 and 5, the energy content of a west-facing chimney is greater than that obtained with one facing south. It is interesting to note that the peak energy content is reached at about 20:00 when the outside air is approaching a comfortable temperature.

3. CONCLUSION

The heat storage of a solar chimney was found to be strongly dependent upon the orientation. It was found that a west-facing solar-heated chimney receives more energy than one facing south at latitude 33°. This is because the more one goes south, the more solar radiation is received on east and west walls and less on the south. So if a solar-heated chimney is to be used to induce evening ventilation in a building in low latitudes, it should go on a west wall.

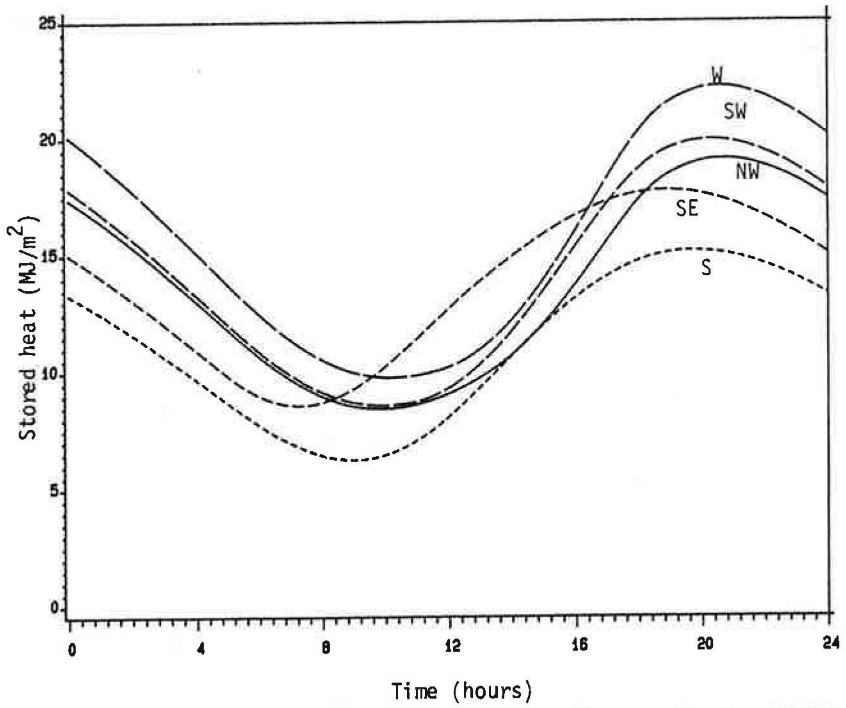

Fig. 4. Heat stored within a chimney against time with respect to T_r = 30 °C.

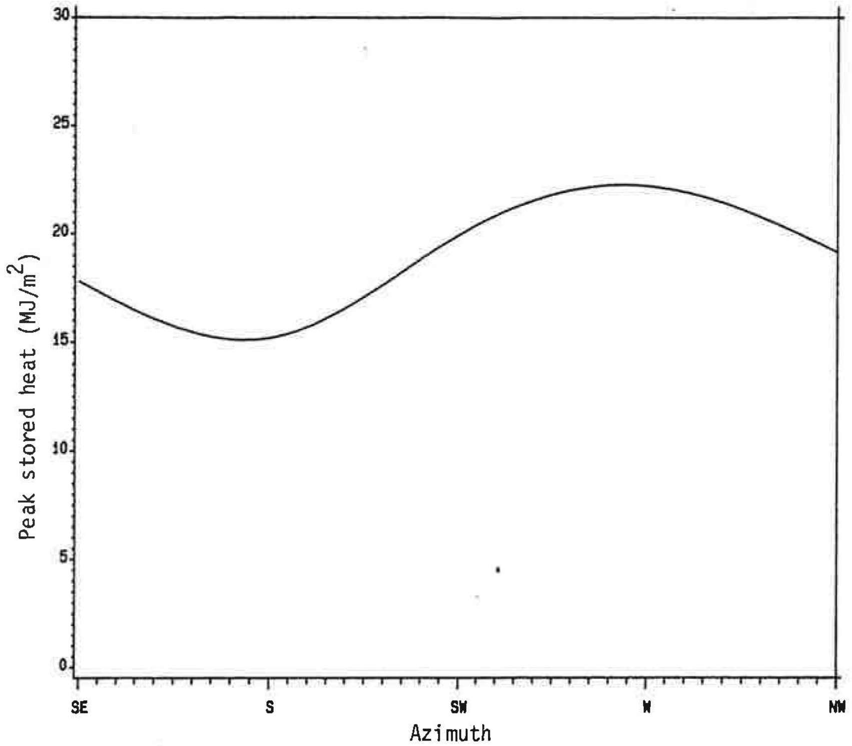

Fig. 5. Peak stored heat against azimuth for latitude 33°N.

LIST OF SYMBOLS

a	surface absorptivity
A	area (m^2)
$C_{(n-1,\,n)}$	thermal conductance between the nodes, $(n-1)$ and n (W/m^{2}K)
c_p	specific heat capacity of the node n (J/kg K)
dt	time increment (s)
E	surface emissivity (0.9 for ordinary building material)
h	solar altitude (degrees)
h_c	surface convection coefficient inside the cavity (W/m^{2}K)
H	altitude above sea-level (m)
$H_{s(n)}$	heat stored within a single slice of a wall (MJ/m^2)
H_T	total stored heat in the chimney (MJ/m^2)
I_{dhb}	basic diffuse irradiance on a horizontal surface (W/m^2)
I_{dhd}	design diffuse irradiance on a horizontal surface (W/m^2)
I_{Dhb}	basic direct irradiance on a horizontal surface (W/m^2)
I_{Dvb}	basic direct irradiance on a vertical surface (W/m^2)
I_{Dvd}	design direct irradiance on a vertical surface (W/m^2)
I_e	longwave radiation loss from black surfaces (W/m^2)
I_{thd}	design total irradiance on a horizontal surface (W/m^2)
I_{tvd}	design total irradiance on a vertical surface (W/m^2)
K_a	thermal conductivity of the air (W/m K)
k_D	direct radiation coefficient (1.1 for arid climates)
k_{af}	altitude correction factor
k_d	diffuse radiation coefficient (0.9 for arid climates)
k_r	ground reflection factor (0.5 for arid places)
R_{si}	total inside surface thermal resistance on the room side (m^{2}K/W)
R_{so}	total outside surface thermal resistance (m^{2}K/W)
T_{eo}	solair temperature (°C)
$T_{(n,\,t+1)}$	future temperature of the node n predicted from the known temperatures (°C)
T_a	air temperature within the chimney (°C)
T_{ra}	room air temperature (°C)
T_r	reference temperature (30 °C)
Z	height (m)
δx	distance increment (m)
ρ	density of the material of the node n (kg/m^3)

REFERENCES

1 A. Bouchair, D. Fitzgerald and J. A. Tinker, Passive solar induced ventilation, *Proc. 8th Int. Conference on Alternative Energy Sources, Miami Beach, Florida, U.S.A., 1987*, to be published by Hemisphere Publ. Corp., New York.
2 A. Bouchair, D. Fitzgerald and J. A. Tinker, Moving air using stored solar energy, *Proc. 13th National Passive Solar Conference, 1988, Cambridge, MA*, ASES, pp. 33 - 38.
3 *CIBSE Guide*, Chartered Institution of Building Service Engineers, London, 1986.
4 Climatic data from the Algerian Meteorological Office in Algiers.
5 C. Y. Warner and V. S. Arpaci, An experimental investigation of turbulent natural convection in air at low pressure along a vertical heated flat plate, *Int. J. Heat Mass Transfer, 11* (1968) 397 - 406.
6 P. W. O'Callaghan, *Building for Energy Conservation*, Pergamon Press, Oxford, 1978.
7 H. Y. Wong, *Handbook of Essential Formulae and Data on Heat Transfer for Engineers*, Longman, London, 1977.

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Predicting Vapour Content of the Indoor Air and Latent Loads for Air-conditioned Environments: Effect of Moisture Storage Capacity of the Walls

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ABSTRACT

A time-dependent model is described for predicting the indoor vapour content in a room and its corresponding latent loads. The model takes into account the moisture storage capacity of the walls.

In order to perform an exploratory investigation on the impact of this phenomenon, the model is applied to an air-conditioned space in which vapour gains from occupants are present during daily periods. Analytical results show that the moisture storage capacity of the walls significantly damps diurnal variations in relative humidity, especially when the moisture generation is markedly time-dependent. The effect of both moisture storage capacity and occupancy patterns is analyzed and briefly discussed. Results indicate that, for many air-conditioning applications in which relative humidity is not directly controlled but allowed to vary within the physiological limits for man's thermal comfort, improved benefits to both cooling equipment sizing and operating costs occur when latent loads computations take into account the moisture storage capacity of the walls.

INTRODUCTION

The control of the indoor vapour content of an air-conditioned space, and hence of the relative humidity, has a considerable importance in thermal building design.

Although humidity has a rather moderate effect on man's thermal comfort [1], it may often be responsible for the uncomfortable feeling of dryness that arises in the breathing system as well as for frequency of common colds and other respiratory diseases. Further effects on man's comfort and health are also well known: high relative humidity (≥70 - 75%) may favour the development of moulds or the growth of the dust mite [2, 3], while low relative humidity, by contrast, may cause uneasiness and discomfort due to electrostatic discharges [2].

Generally speaking, in air-conditioning practice it is possible to make a clear distinction between two extreme cases:

(1) when the indoor relative humidity is maintained at a steady value by means of air-conditioning equipment; and

(2) when relative humidity is not controlled but allowed to undergo wide variations due to several factors such as weather conditions, building characteristics, moisture generation and ventilation.

In many practical situations, however, moderate relative humidity variations are permissible inside buildings. This fact represents an intermediate case between the above two extreme situations; for instance, a typical design condition is to maintain the indoor relative humidity at values ranging from 40% to 60% by properly selecting the cooling capacity of the air-conditioned equipment able to meet this requirement without providing any direct control during normal operation.

In current air-conditioning design practice, the cooling load calculations are ordinarily carried out conservatively treating the latent cooling loads as instantaneous gains [4], while other calculation procedures use a dynamic mass balance equation between the vapour generation and dilution by means of ventilation, which disregards the moisture storage capacity of the walls [5, 6]. Thus, the

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