Moisture Storage in Buildings

by

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Abstract

Commonly used building materials and furnishings have the ability to store or release moisture depending on the surrounding conditions. In hot and humid climates, the latent load represents a significant part of the total cooling load for a building, and moisture storage and release become important contributors to energy gains.

Moisture storage depends on the sorption properties of the materials, and moisture transport in the materials is a function of the diffusivities. Moisture transport is accompanied by heat gain or loss from the zone due to the heat of sorption of water vapor in the materials. The equations describing moisture transport are a net of coupled ordinary differential equations for heat and mass flows. Solving for moisture content can be accomplished using finite element or finite difference techniques. These methods are accurate, but computationally involved.

In this paper, a simplified model for moisture storage known as the " Effective Penetration Depth Model " is modified. Equilibrium isotherms and an effective material thickness are used to solve for the change in the moisture content of the building materials and room temperature and humidity ratio at each time step. Simulation results show that the modified method, compared with the finite difference solutions, is efficient to model moisture storage and transport in buildings. The newly modified model has the advantages of simplicity, ease of application and less computation time.

Acknowledgements

It is 2:43 a.m.; a time that is not unusual for me to be in this lab working on my research. I never thought that one day I will be here typing the last words of my thesis (the fruit of my hair loss). Another era of my life has passed to become a part of my past leaving behind 18 months loaded with precious moments and treasurable memories to be fond of.

During the past one and a half years, I became more familiar with the term "*Research*". It is staying up late nights staring at the computer screen for hours without accomplishing anything. It is staying up late nights fighting with the computer for not doing what you want it to do and expecting it to respond. It is then when you know that it is the time to go home. It is going through endless loops in " *Research Hell* " where you see your self working at a car wash or asking people " plastic or paper ? " in a super market. It is going through a maze of equations and calculations expecting to accomplish something until you find your self where you started. It is spending an extensive time working on a small graph and getting a computer bomb before you get the chance to save your work. A good researcher is that person who can go through all this without losing a piece from his mind.

This thesis comes as the fruit of days and nights of hard work and frustrations. Many people contributed to the completion of this research. Sandy, thank you for being my advisor, my teacher and my friend; the elevator discussions were really helpful. You motivated and encouraged me and built up my self confidence which helped me to produce this work. John, thank you for your support and suggestions which gave a flavor to this work. Jack, thank you for founding this lab. I do not remember that I saw you once without your " *Don't Worry, Be Happy* " smile. It made me realize that things can always get worse no matter how bad they are. Bill, thank you for the lab financial support and for making the lab feels like home. My special thanks go to the Maxwells for their care, concern and support.

I would like to thank the Solar Lab family which made my stay in Madison unforgettable. The " *Thermo Man* ", Osama, for helping me on my numerical problems and for being a good friend. Mark, Florian and Juergen for the wonderful memories. Ruth for saving me when I was drowning in the " *TRNSYS Dilemma* ".

Thank you Zaidoon, my best friend, for being there when I needed someone who cares to talk to. Bassem for being supportive, caring, loving and wonderful brother.

Finally, my love and gratitude goes to the most important people in my life. Mom and Dad, you weren't able to support me financially, however, you gave me your love which is more precious. It kept me going on when everything was turning against me. you taught me how to face my problems and never run away from them. I think that I'm the luckiest person in the world because you are my parents. Mom, Dad, your prayers have been answered and this work is dedicated to you.

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List of Tables

Nomenclature

Roman Letters

Symbol	Definition	SI-Units
A	surface area	[m^2]
a	regression constant	
Bi_m	mass transfer Biot number	
c	regression constant	
Cap	heat capacitance of the room air and furniture	[kJ/°C]
C_p	specific heat	[kJ/kg°C]
D_m	moisture Diffusivity	[m^2/hr]
e	regression constant	
$\dot{E}$	energy rate	[kJ/hr]
F	air mass magnification factor	
Fo	Fourier number	
g	regression constant	
h	enthalpy	[kJ/kg]
h_m	convective mass transfer coefficient	[kg/m^2hr]
h_t	convective heat transfer coefficient	[kJ/hr.m^2°C]

Continued

Symbol	Definition	SI-Units
k	thermal conductivity	[kJ/hr.m°C]
$\dot{m}$	mass flow rate	[kg/hr]
$\dot{m}''$	mass flux	[kg/m^{2}hr]
M	moisture mass	[kg]
P	atmospheric pressure	[N/m^2,Pa]
P_v	vapor pressure	[N/m^2,Pa]
q_T''	imposed heat flux	[W/m^2]
$\dot{Q}$	rate of energy flow	[kJ/hr]
R	heat transfer resistance	[°C.hr/kJ]
R_a	ideal gas constant for air (461.52)	[J/kg.K]
R_m	mass diffusion resistance	[hr.Pa/kg]
S_R	sum of the squares of the residuals	
t	time	[hr]
T	temperature	[°C]
U	internal energy	[kJ]
U	moisture content of the solid material	[kg/kg]
V	volume	[m^3]
Y_T	isothermal moisture capacity	
Z_ρ	thermo-gradient coefficient	[kg/kg.K]

Greek Letters

Symbol	Definition	SI-Units
α	solar absorptance	
χ	thickness participating in moisture exchange	[m]
δ	effective penetration depth	[m]
Δ	increment	
ϕ	relative humidity	[0 to 1.0]
λ_{vap}	heat of vaporization of water	[J/kg]
ρ	density	[kg/m^3]
ω	humidity ratio	[kg/kg]
$\dot{\omega}_I$	rate of internal moisture generation	[kg/hr]
ξ	moisture Permeability	[kg/m.hr.Pa]

Subscripts

Symbol	Definition
a	air
amb	ambient
c	ceiling

continued

Symbol	Definition
con	condensed
cond	conductive
conv	convective
des	design
diff	diffusion
dr	doors
equ	equipment
f	end of time step
fur	furniture
g	saturated vapor sate
i	node number
in	inlet
inf	infiltration
int	internal
j	solid material number
l	latent
lig	lights
m	mass transfer
mix	internal flows

Continued

Symbol	Definition
n	last node number
o	beginning of time step
p	pores
peo	people
pr	in terms of pressure
r	room
rd	desired room
rev	re-evaporation
s	sensible
sa	sol-air
sat	saturated condition
sd	solid
sin	sink
sor	source
sorp	sorption
sto	stored
sup	supply air
sur	surface
tot	total

Continued

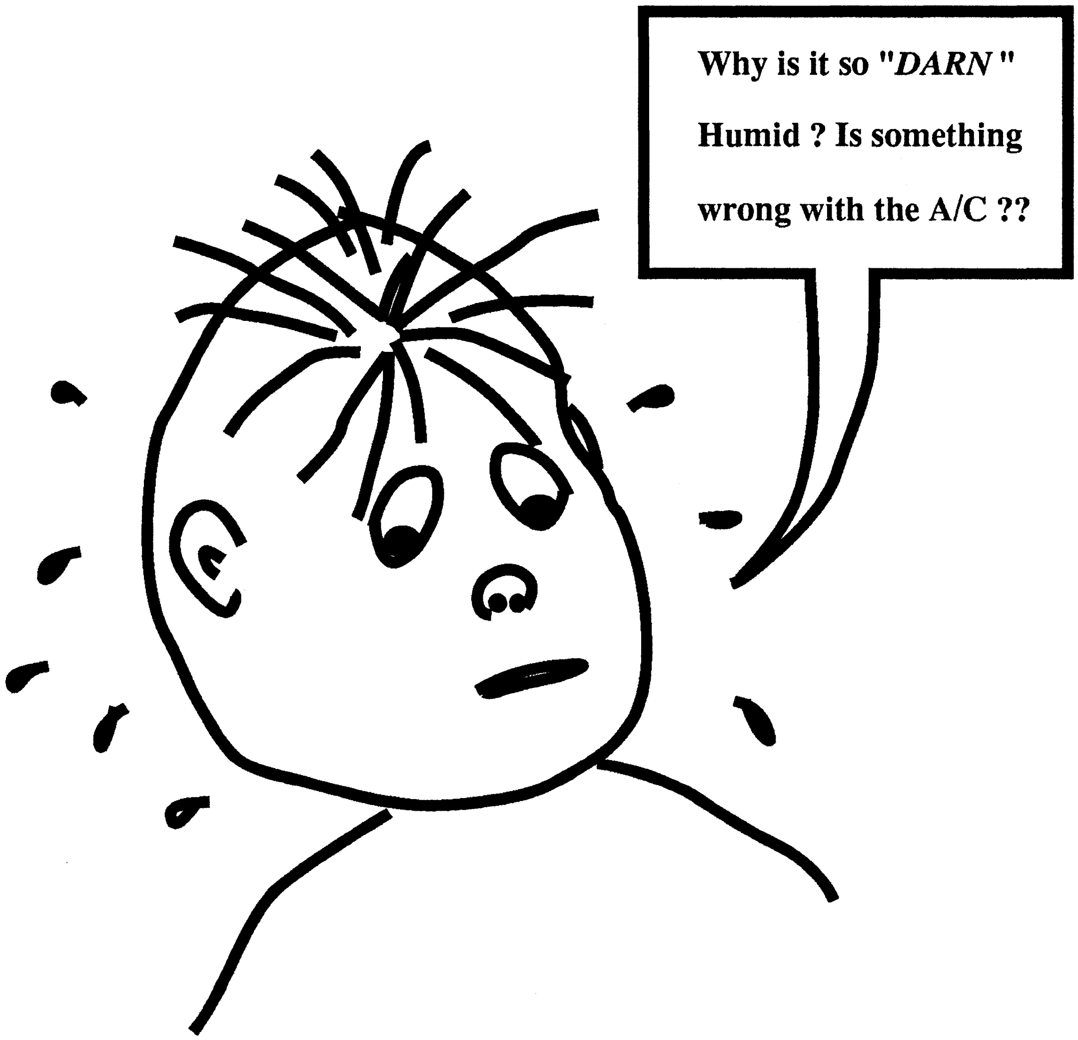
Symbol	Definition
T	net
v	ventilation
vap	vapor
wd	window
wl	wall
z	neighboring zone
zi	zone number

Superscripts

Symbol	Definition
b	regression constant
d	regression constant
f	regression constant
h	regression constant
*	surface condition

Miscellaneous

Symbol	Definition	
—	average value	
∇	vector operator	[1/m]
.	time derivative	
MEPD	Modified effective penetration depth	


Why is it so "*DARN*" Humid ? Is something wrong with the A/C ??

Chapter I

Introduction

Predicting indoor space conditions via computer simulation is a major concern in building science. Considerable effort has been paid to develop computer codes that are user friendly, simple, and accurate. These codes (TRNSYS [14], FSEC 1.1 [12,13], MOBAL3 [16] and HVACSIM [15], for example) are based on the heat and mass balance concepts. They can predict indoor conditions by solving the heat and mass transport equations.

Combined heat and mass transport is mathematically as well as physically difficult to describe. Moisture moves into porous materials whose moisture content is non-uniform and below saturation. Phase change is also associated with moisture transfer. Moisture is adsorbed as vapor, but it may condense as it propagates inside the material. When moisture is desorbed, it evaporates into the surrounding air by convective diffusion as it reaches the surface of the material.

The state of moisture inside the material may vary depending on the internal conditions of the material. Moisture may exist in the solid material in a vapor state, a liquid state, or a mixture of vapor and liquid. Vapor-liquid or liquid-vapor phase change is usually accompanied by heat generation or absorption which affects the temperature gradient inside the material.

Heat transfer in or out of the solid material occurs due to the temperature difference between the material and its surroundings. Similarly, mass transfer occurs due to a vapor pressure difference between the air in the material pores and the surrounding air.

Moisture is generated inside the building due to breathing, cooking, showering, etc, and therefore, it varies with the number of people in the building and their type of activity. Moisture is also brought into the building by infiltration and ventilation. This causes the indoor conditions to be a function of the outdoor environment.

Commonly used building materials are hygroscopic to some extent. The moisture stored in the furniture, walls, and structures can significantly affect the indoor conditions. When the cooling system is not in operation, part of the moisture brought into the room is adsorbed by the walls and furnishings affecting the room air relative humidity. Moisture adsorption is usually accompanied by heat generation inside the material causing its temperature to rise. On the contrary, moisture desorption is usually accompanied by heat absorption inside the material causing its temperature to drop. Moisture adsorption or desorption is accompanied by sensible gains or losses due to the vapor temperature difference between the surface air of the material and the room air. These gains or losses might affect the room temperature if they are significant. The temperature change due to moisture adsorption or desorption can affect the times when the system is in operation if the system is controlled by a thermostat (temperature level control); consequently, the time distribution of the load will change. When the system is in operation, some of the moisture stored in the walls and furnishings is released into the room air raising its relative humidity. As a result, the energy consumed by the cooling system to dehumidify the air increases.

This thesis studies the effect of moisture storage on indoor conditions, and presents a modified model that predicts indoor conditions taking moisture storage into account.

I.1 THE IMPORTANCE OF UNDERSTANDING AND MODELING MOISTURE STORAGE IN BUILDINGS

Space heating and cooling represent 21% of the overall energy consumed in the United States [17]. Commercial buildings are major consumers of cooling energy because they require cooling in the winter as well as in the summer. Therefore, significant effort has been put to develop more efficient methods for cooling.

The cooling load is defined as the energy required to maintain a space at a specific set temperature. The cooling load is divided into two parts:

a) Sensible load: The energy required to decrease the air temperature to a desired value.

b) Latent load: The energy required to dehumidify the air to a desired value.

When moisture is stored in the walls, furnishings and structures, mold is more likely to develop inside. This will cause structural damage, and bad indoor air quality that can lead to health problems [4,5]. However, designing a cooling system that is able to release some of the stored moisture will overcome this problem.

A cooling system can operate in different ways. By varying the system parameters, for example air and water flow rates, temperature set points, fan speed, etc, a combination of system parameters that allow the system to operate at the highest efficiency possible and lowest cost can be reached. This process is defined as optimization or optimal control. Therefore, in the process of optimization, estimation of the cooling load is a necessity.

Sizing a cooling system for a particular building is necessary before installation to insure that the system is large enough to maintain comfortable indoor conditions. Usually, a system is sized based on design conditions. The design conditions are the ambient and indoor temperatures and humidity ratios. Ambient design conditions can be described as the worst possible conditions and sizing a system according to these conditions will insure system normal operation under any condition. The indoor design conditions are the desired comfort conditions. Using the design parameters the sensible load due to infiltration can defined as

$$\dot{Q}_{inf,s} = \dot{m}_{inf}\, C_{p,a} \left(T_{des} - T_{rd}\right) \tag{1.1}$$

where

$\dot{m}_{inf}$ = infiltration air mass flow rate.

$C_{p,a}$ = constant specific heat of air.

T_{des} = the design ambient temperature.

T_{rd} = the desired room temperature.

The total design sensible load is the energy required to remove the net heat gain which is the gains from infiltration, walls, windows, doors, ceiling, people, and appliances plus heat stored in furnishings and structures

$$\dot{Q}_{des,s} = \dot{Q}_{inf,s} + \dot{Q}_{wl} + \dot{Q}_{wd} + \dot{Q}_{dr} + \dot{Q}_{c} + \dot{Q}_{peo,s} + \dot{Q}_{app,s} + \dot{Q}_{sto,s} \quad (1.5)$$

On the other hand, the latent load due to infiltration is defined as

$$\dot{Q}_{inf,l} = \dot{m}_{inf}\, \lambda_{vap}\, (\omega_{des} - \omega_{rd}) \quad (1.6)$$

where

λ_{vap} = latent heat of vaporization.

ω_{des} = the design ambient humidity ratio.

ω_{rd} = the desired room humidity ratio.

The total design latent load is defined as the energy required to remove moisture gains due to infiltration plus people and appliances plus moisture stored in furnishings and structures

$$\dot{Q}_{des,l} = \dot{Q}_{inf,l} + \dot{Q}_{peo,l} + \dot{Q}_{app,l} + \dot{Q}_{sto,l} \quad (1.7)$$

The total design cooling load is calculated as [17]

$$\dot{Q}_{des,tot} = \dot{Q}_{des,s} + \dot{Q}_{des,l} \quad (1.8)$$

The system size will be equal to the total design load in tons. As noticed from the above equations, heat and moisture storage are not considered, the design total load will be smaller and hence, the cooling system will be undersized.

In doing an economic analysis for a system, or predicting the energy consumption for future energy needs, estimating the cooling load accurately is necessary because it reflects the energy consumption of the system.

As noticed in some of the cases mentioned above, evaluation of the cooling load is important. The sensible load is defined as

$$\dot{Q}_s = \dot{m}_a C_{p,a} (T_{a,in} - T_{a,out}) \tag{1.9}$$

where

$\dot{m}_a$ = the cooling coil air mass flow rate.

$T_{a,in}$ = the inlet air temperature.

$T_{a,out}$ = the outlet air temperature.

and the latent load is defined as

$$\dot{Q}_l = \dot{m}_a \lambda_{vap} (\omega_{a,in} - \omega_{a,out}) \tag{1.10}$$

where

$\omega_{a,in}$ = the inlet air humidity ratio.

$\omega_{a,out}$ = the outlet air humidity ratio.

The cooling coil inlet air is usually a mixture of indoor and outdoor air, therefore the sensible and latent loads are functions of the indoor and outdoor conditions (temperatures and humidity ratios).

In commercial buildings, one of the cooling strategies is to cool the building during the day and ventilate during the night when there are no occupants. This will cause the room conditions to vary between day and night; therefore, to estimate the cooling load for a commercial building, knowledge of indoor conditions is important especially in the early cooling hours when the amount of moisture stored in the walls and furniture during the night is released into the room air. This will make the prediction of the indoor conditions critical.

Moisture removal creates the latent load, but has a little effect on the sensible load due to the relatively small amount of heat absorption or release. In hot and humid climates, the amount of moisture stored in furnishings and walls may be significant. Therefore, predicting the indoor conditions based on the assumption that the zone air is the only moisture storage media, can lead to errors in evaluating the instantaneous cooling loads. By neglecting the moisture stored in the furnishings and walls, the energy consumed in releasing this moisture is also neglected.

Moisture adsorption and desorption can also affect the indoor conditions, and neglecting moisture storage will lead to misprediction of these conditions. This will cause bad design, sizing, and control of a system since they are functions of the load, indoor and outdoor conditions. That is why moisture transport and storage in buildings is an issue.

I.2 OBJECTIVE

The objective of this research is to make an exclusive evaluation of a combined heat and moisture transfer model developed by Kerestecioglu *et. al.* 1988 [12] at the Florida Solar Energy Center. This model, known as the " Effective Moisture Penetration Depth ", uses the lumped capacitance method [8] to model moisture storage in buildings. In this model, and for short periods of time when the moisture condition of the inner layers of the material are not affected by surrounding conditions, moisture exchange is assumed to take place in the thin layer close to the surface of the material. Since this layer is very thin, the resistance to moisture diffusion is neglected. Further discussion on this model is in chapter II.

The goal of this evaluation is to modify the Effective Penetration Depth model to include a moisture diffusion resistance. The modification must not complicate the model that must be computationally simple, accurate, and easy to implement. The Modified Effective Penetration Depth model should use material sorption properties to calculate the moisture distribution inside the solid material and the amount of moisture transferred in or out of the material. The modified model will be used in predicting indoor conditions and cooling loads, and will be implemented in the building simulation subroutine TRNSYS-TYPE 19.

In Chapter II, different models for moisture transport and storage in buildings are introduced along with their governing equations. The modified model with its physical as well as mathematical description is also discussed. Chapter III consists of simulation results and comparisons of the solutions to validate the newly modified model. Finally, conclusions and some recommendations are provided in Chapter IV.

I.3 INTRODUCTION TO TRNSYS

Computers are the most powerful engineering tool of the generation. There is no question about the fact that there is a significant economic and possibly time expense in conducting an experiment, especially if it involves buildings and cooling systems. The costs of installing a cooling system in a building to test its performance and determine a control strategy is high. An inexpensive way to test a system and its performance is by computer simulation. Simulations have the advantage of giving the user the ability to change the system parameters if more than one system is being studied. Computer simulations are not only cost effective, but time saving too; for example, one year simulation can be done in few hours via computer.

TRNSYS 13.0 (Klein *et. al.*,1988) [14] is a transient simulation program developed at the solar energy laboratory at the University of Wisconsin-Madison. TRNSYS was originally developed to simulate solar heating systems, but later its range of application was expanded to include modeling HVAC, power cycles, and photovoltaic systems. This software is composed of different subroutines called " TYPEs ". Each TYPE represents a piece of equipment like a pump, cooling coil, or solar collector, and consists of equations that determine the performance of an equipment when solved. For every TYPE, there is a set of parameters, inputs, and outputs.

In order to simulate a system, different TYPEs are connected together in the configuration desired so that the output of one subroutine is an input to the other forming a cycle. Unlike the inputs, the TYPE parameters are constants and do not change throughout the simulation; the parameters plus the input will determine the performance of a system.

The system is described in a file called " deck ". In the deck, the equipment needed to form the system and their parameters, plus the simulation time step that ranges from seconds to hours are specified. The output of each piece of equipment are directed as input to another equipment in a way that satisfies the system configuration. In the deck, all the information needed to solve the equations that determine the performance of a system is provided. The system cycle is driven by a weather data file known as typical mean year (TMY) data (Hall *et. al.*, 1978) [7] that consists of hourly solar radiation, ambient temperature and humidity ratio, wind speed and others. At every time step, the equations that describe the performance of the system are solved so that the results at the first time step are the input for the next one.

As an illustration, consider the system described in Figure (1.1). The system consists

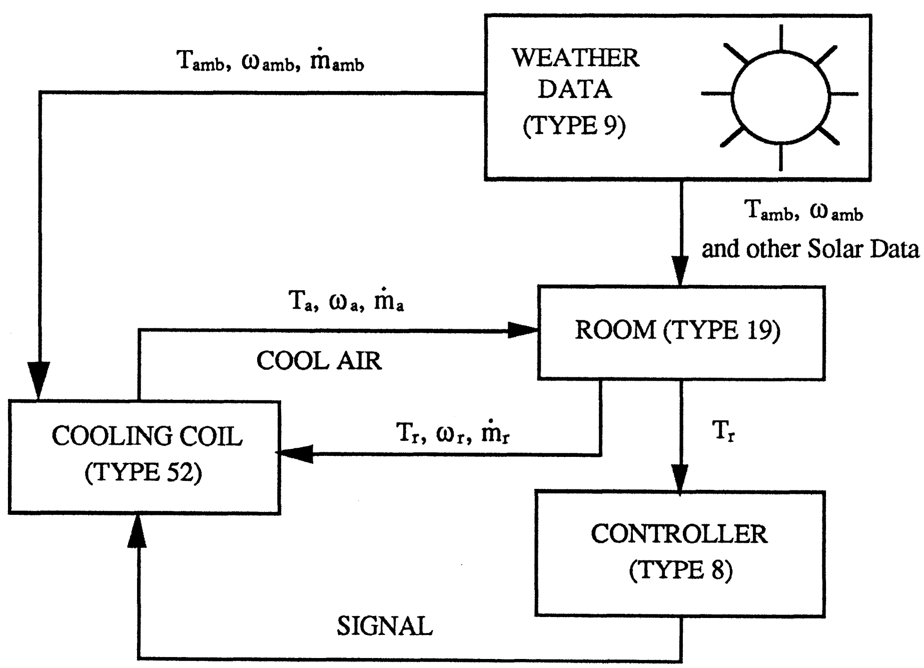


Figure 1.1 Simple TRNSYS model for space cooling.

of a cooling coil, room, and a controller. At the initial time step, The TMY data (TYPE 9) is read and analyzed by TRNSYS to determine the solar radiation, temperature, and humidity ratio and other data as needed. The weather data is passed as input to the subroutine (TYPE 19) that describes the room. TYPE 19 solves the equations required to determine solar gains and indoor conditions. The room temperature, an output of TYPE 19, is directed as input to the controller (TYPE 8) that determines whether the cooling system should be operated or not. The signal, output from TYPE 8, is passed to the cooling coil (TYPE 52). If cooling is needed, a mixture of ambient and room air are passed through the coil and cooled by rejecting heat to the cold water in the coil pipes. Then the air flow and temperature, output of TYPE 52, is passed as input to TYPE 19, and the cycle continues in the same fashion for the same time step until the input passed to every TYPE converges within a tolerance specified by the user at the beginning of the simulation. After the convergence, the simulation for the next time step starts and the simulation is completed after reaching a certain time specified by the user. The simulation period can range from few seconds to years.

I.4 INTRODUCTION TO TRNSYS-TYPE 19

TYPE 19 [18] is a one-zone model used to determine cooling and heating loads as well as indoor temperature and humidity. TYPE 19 uses ASHRAE heat transfer functions to model the room walls, ceiling, and floor.

Two control strategies are associated with this subroutine; the energy rate control or the temperature level control. In the later, the user specifies maximum and minimum temperatures and humidity ratios as set points for cooling and heating. TYPE 19 calculates the load (cooling or heating) required to keep the room condition between the set temperature limits.

In the temperature level control, a cooling or heating equipment is needed to condition the room. A controller is usually associated with the equipment to command heating or cooling according to the controller temperature set points.

Chapter II

Models for Moisture Storage

This chapter introduces some of the existing models for moisture transport and storage. The governing equations for each model are presented. The advantages and disadvantages for each model are discussed.

II.1 TRNSYS MODEL

TYPE 19 is a one zone model subroutine in TRNSYS that is designed to predict indoor conditions and the associated heating and air conditioning loads [18]. TYPE 19 utilizes an energy balance to determine the room temperature and the associated sensible load. Taking the room as a system, a sensible energy balance equation can be written as

$$\mathrm{Cap}\left(\frac{T_{rf} - T_{ro}}{\Delta t}\right) = \sum_{k=1}^{n} h_{t,k} A_k (T_{sur} - T_r) + \dot{Q}_{v,s} + \dot{Q}_{inf,s} + 0.3\,\dot{Q}_{peo,s} + \dot{Q}_{int,s} + \dot{Q}_{z,s} \quad (2.1)$$

where

Cap = a number representing the heat capacitance of the room air plus furniture.

T_{rf} = room temperature at the end of the time step.

T_{ro} = room temperature at the beginning of the time step.

Δt = time step.

$h_{t,k}$ = convective heat transfer coefficient of surface " k ".

A_k = area of surface k, where " k " is the surface number.

T_{sur} = surface temperature.

T_r = room temperature.

$\dot{Q}_{v,s}$ = ventilation sensible gains.

$\dot{Q}_{int,s}$ = internal sensible gains other than people or lights.

$\dot{Q}_{z,s}$ = sensible gains due to convection from neighboring zones.

In equation (2.1), a term that accounts for heat due to evaporation or condensation of moisture from or into the solid material is not included. Equation (2.1) is a first order difference equation and TRNSYS solves numerically for the room temperature at each time step.

With respect to moisture storage, TYPE 19 uses lumped moisture transport equations and assumes that moisture is only contained by room air. A mass balance on the room air can be written as

$$\rho_a V_a \frac{d\omega_r}{dt} = \dot{m}_{inf} (\omega_a - \omega_r) + \dot{m}_v (\omega_v - \omega_r) + \dot{\omega}_I + \frac{\dot{Q}_{peo,l}}{\lambda_{vap}} \qquad (2.2)$$

where

ρ_a = density of air.

V_a = volume of room air.

ω_r = room air humidity ratio.

ω_a = ambient air humidity ratio.

ω_v = ventilation air humidity ratio.

$\dot{m}_v$ = ventilation air flow rate.

$\dot{\omega}_I$ = rate of internal moisture gains (other than people).

Equation (2.2) is a first order differential equation that does not include a term that accounts for moisture storage in furniture, walls, and structures. TRNSYS solves for room humidity ratio numerically using modified Euler method.

The above model is computationally as well as mathematically simple since the equations are lumped and only one domain is involved. The disadvantage of this model is that it does not predict the indoor condition accurately since moisture storage in the building materials is neglected.

II.2 THE EFFECTIVE AIR MASS MULTIPLIER MODEL

The Effective air Mass Multiplier model [11] is similar to the TYPE 19 TRNSYS model except that it accounts for moisture storage in furniture, walls, and structures. In this model, the room air mass is increased by a factor, typically between 1 and 10, to account for the effects of moisture storage on the room humidity condition. By magnifying the air mass by this factor, the mass of moisture in the air is also magnified by the same factor. Therefore, the moisture that is stored in the solid material is modeled as being stored in the room air. Using this model, equation (2.2) becomes

$$F \rho_a V_a \frac{d\omega_r}{dt} = \dot{m}_{inf} (\omega_a - \omega_r) + \dot{m}_v (\omega_v - \omega_r) + \dot{\omega}_I + \frac{\dot{Q}_{peo,l}}{\lambda_{vap}} \qquad (2.3)$$

where "F" is the factor by which the room air mass is increased to account for moisture storage.

This model avoids using a domain other than the air, therefore, it is characterized by simplicity, quickness of computation, and ease of implementation because it can be easily introduced in most of the current existing codes.

A disadvantage of this model is that it does not consider the effect of moisture diffusion in the solid material. Moisture storage in the walls and furnishings is treated the same as moisture storage in air although diffusion in a solid is not the same as diffusion in air. This model does not accurately model the storage media. Storage in solids is totally different than storage in air and because of properties differences, the mass transport resistances in solids are different than the ones in air . Another disadvantage is that the amount of moisture stored is unknown and it differs from case to case depending on the room size, ventilation and infiltration flowrates, and ambient conditions. Consequently, the value of the factor "F" is arbitrary and must be determined empirically.

II.3 THE FINITE DIFFERENCE MODEL

In the Finite Difference model, detailed heat and moisture transfer equations are used for the solid domain, whereas lumped equations are used for the room air domain assuming it is well mixed. The solid domain is divided into a specified number of segments. A node is assigned to each segment so that the conditions of the segment (temperature, vapor pressure, moisture content and relative humidity) are that of the node as shown in

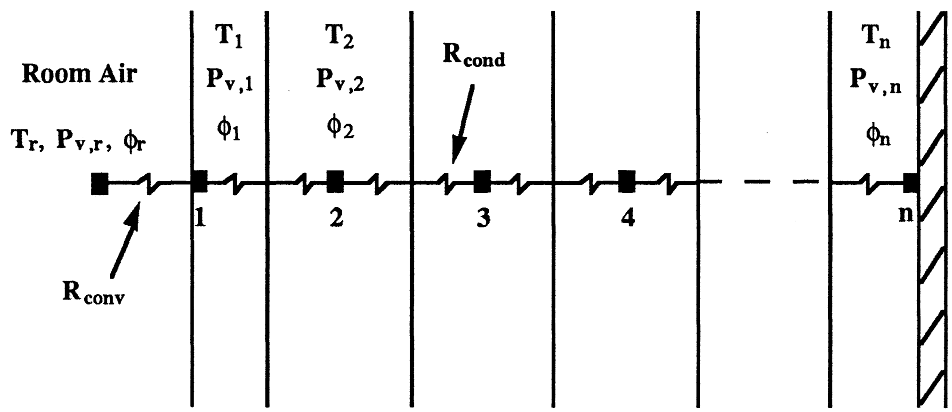


Figure 2.1 Finite difference layout.

Figure (2.1). The material conditions may vary along the depth of the material but are assumed constant within each segment.

II.3.1 Mathematical Description

The Finite Difference model is formed by dividing the solid domain into segments (nodes) as shown in Figure (2.1), and applying heat and moisture balances at the center of each segment or node. Starting with the energy balance equation [9]

$$\dot{E}_{in} - \dot{E}_{out} = \dot{E}_{sto} \qquad (2.4)$$

where

$\dot{E}_{in}$ = the rate of heat absorbed by the solid material.

$\dot{E}_{out}$ = the rate of heat rejected by the solid material.

$\dot{E}_{sto}$ = the rate of heat stored in the solid material.

a set of " n " equations of " n " unknowns can be obtained. For the first node, equation (2.4) can be written as [3]

$$\frac{T_r - T_1}{R_{conv}} + \frac{\lambda_{vap}}{A}\frac{dM_1}{dt} - \frac{T_1 - T_2}{R_1 + R_2} = \rho_{sd}\, C_{p,sd}\, \Delta x \frac{dT_1}{dt} \qquad (2.5)$$

where

T_1 = material temperature at the first node.

T_2 = material temperature at the second node.

ρ_{sd} = density of material " j ".

$C_{p,sd}$= constant specific heat of the solid material.

Δx = increment of material thickness.

$\frac{dM_1}{dt}$ = the change of solid moisture mass per unit time at the first node.

R_{conv} = convective heat transfer resistance.

R_1 = heat conduction resistance such that,

$$R_{conv} = \frac{1}{h_t} \qquad (2.5.1)$$

and

$$R_i = \frac{\Delta x}{2\,k} \qquad (2.5.2)$$

where

k = thermal conductivity of the solid material.

i = node number.

A general equation can be applied to the nodes that fall between the first and the last node [3]

$$\frac{T_{i-1} - T_i}{R_{i-1} + R_i} - \frac{T_i - T_{i+1}}{R_i + R_{i+1}} = \rho_{sd}\, C_{p,sd}\, \Delta x\, \frac{dT_i}{dt} \qquad (2.6)$$

Equation (2.5), includes a term that accounts for heat of sorption

$$\dot{Q}_{sorp} = \frac{\lambda_{vap}}{A} \frac{dM_1}{dt} \qquad (2.7)$$

because moisture is assumed to condense within the first node. However, the state of moisture after the first node is assumed to be sorbed, hence, equation (2.6) does not include the heat of sorption term.

The energy balance equation for the last node is written as [3]

$$\frac{T_{n-1} - T_n}{R_{n-1} + R_n} = \rho_{sd}\, C_{p,sd}\, \Delta x\, \frac{dT_n}{dt} \qquad (2.8)$$

where

n = the last node.

With respect to moisture storage, a general moisture balance equation can be written as

$$\dot{M}_{in} - \dot{M}_{out} = \dot{M}_{sto} \qquad (2.9)$$

where

$\dot{M}_{in}$ = the rate of moisture adsorbed by the solid material.

$\dot{M}_{out}$ = the rate of moisture desorbed from the solid material.

$\dot{M}_{sto}$ = the rate of moisture stored in the solid material.

In this model, the vapor pressure of water in air at equilibrium with the solid material was used as the moisture driving force. This vapor pressure can be expressed in terms of relative humidity and temperature [11]

$$P_v = \phi \exp\left(23.7093 - \frac{4111}{T_{a,p} - 35.5} \right) \qquad (2.10)$$

where

ϕ = relative humidity of air in the pores in equilibrium with the solid material.

$T_{a,p}$ = the temperature of air in equilibrium with the solid material in degrees Kelvin [K].

This temperature is assumed to be equal to the material nodal temperature.

The equilibrium air relative humidity can be obtained from the equilibrium moisture content of the material that can be presented in an exponential form [11] such that

$$U = a\,\phi^{b} + c\,\phi^{d} \qquad (2.11)$$

where

a, b, c, d= regression constants.

Equation (2.11) is usually referred to as " equilibrium moisture isotherm ". Regression constants of different building materials at T= 25 °C taken from Kerestecioglu *et. al.* (1988) [11] are provided in Table (2.1).

Table 2.1 Equilibrium moisture isotherms regression constants for different building materials.

Material Name	**ρ (kg/m³)**	**a**	**b**	**c**	**d**
Balsa Wood	125	0.014022	0.503004	0.027242	5.852653
Brick	1720	0.003744	22.184770	0.002230	0.255390
Cellular Concrete	510	0.015204	0.330670	0.040595	8.402223
Cellulose Building Paper	600	0.115488	5.794530	0.074729	0.655132
Cement Abestos Board	775	0.068560	10.145370	0.006334	0.203290
Cork	430	0.044540	9.070470	0.006964	0.585883
Doussie Wood	660	0.067990	3.441290	0.072336	0.370192
Leather	940	0.203852	0.408090	0.197957	6.898870
Linoleum	2.1	0.052246	0.546057	0.074477	7.927515
Mineral Wool Board	400	0.013022	14.113446	0.005810	0.490509
Pine Wood	530	0.075279	0.473989	0.088280	5.094006
Plaster Board	730	0.009620	0.202660	0.196697	11.476600
Polyester Foam	55.2	0.000400	11.598490	0.002020	1.054990
Rag Building Felt	500	0.155030	8.918695	0.051232	0.671865
Wood Fiber Board	610	0.076060	9.480125	0.072070	0.577840
"	870	0.109178	7.646888	0.102728	0.464374
"	960	0.105460	0.456258	0.120703	7.037468
Wood Particle Board	560	0.064765	0.516527	0.118435	8.472525
Wool Felt	200	0.021749	0.751252	0.038275	5.774993

The moisture content is defined as the ratio of moisture mass to the bulk mass

$$U \equiv \frac{M}{M_{sd}} \qquad (2.11.1)$$

where

M = the mass of moisture stored in the solid material.

M_{sd} = the mass of the solid material.

With a known " U ", equation (2.11) can be solved iteratively to obtain the relative humidity of the air in equilibrium with the solid material.

Starting at the first node equation (2.9) can be written as

$$\frac{dM_1}{dt} = \frac{P_{v,r} - P_{v,1}}{R_{m,conv}} - \frac{P_{v,1} - P_{v,2}}{R_{m,1} + R_{m,2}} \qquad (2.12)$$

where

$P_{v,r}$ = room air vapor pressure.

$P_{v,1}$ = equilibrium vapor pressure of air at the first node.

$P_{v,2}$ = equilibrium vapor pressure of air at the second node.

$R_{m,conv}$ = convective moisture transfer resistance.

$R_{m,1}$ = moisture diffusion resistance at the first node.

The moisture diffusion and convection resistances can be expressed as [1, 2]

$$R_{m,i} = \frac{\Delta x}{2\,\xi} \qquad (2.12.1)$$

and

$$R_{m,conv} = \frac{1}{h_m} \qquad (2.12.2)$$

where

ξ = solid material moisture permeability in terms of pressure.

Moisture permeability can be expressed in terms of the relative humidity of air in equilibrium with the solid in an exponential form [11] such that

$$\xi = e\,\phi^{f} + g\,\phi^{h} \qquad (2.12.1.1)$$

where

e, f, g, h= regression constants.

Regression constants for some building materials at T= 5 °C to 45 °C as taken from Keresteciogłu *et.al.* (1988) [11] are provided in Table (2.2).

The convective mass transfer coefficient " h_m " can be obtained from " Lewis relation " [10]

$$h_m = \frac{h_t}{C_{p,a}} \qquad (2.12.2.1)$$

In equation (2.12), the moisture diffusion resistance is used in terms of pressure,

Table 2.2 Moisture permeability regression constants for different building materials.

Material Name	ρ (kg/m³)	e	f	g	h
Balsa Wood	125	0.001223	3.325168	0.000104	0.326892
Brick	1720	0.000530	0.094560	0.000532	0.115540
Cellular Concrete	510	0.000334	2.599087	0.000690	0.093965
Cellulose Building Paper	600	0.000663	0.148310	0.001085	7.036223
Cement Abestos Board	775	0.000497	0.307816	0.000645	2.748160
Cork	430	0.000149	0.006497	0.000068	6.854995
Doussie Wood	660	0.000310	6.511740	0.000132	1.163469
Leather	940	0.000374	0.101853	0.000299	6.330303
Linoleum	2.1	0.000003	0.182210	0.000005	4.320802
Mineral Wool Board	400	0.001788	0.116999	0.001788	0.117080
Pine Wood	530	0.001174	13.406200	0.000122	2.336779
Plaster Board	730	0.003133	10.562230	0.000308	0.198397
Polyester Foam	55.2	0.000590	0.232459	0.000589	0.130896
Rag Building Felt	500	0.000403	0.102459	0.000990	6.084116
Wood Fiber Board	610	0.003980	3.317516	0.004303	0.019256
"	870	0.000117	0.076865	0.000438	8.931798
"	960	0.000080	0.093082	0.000127	3.384946
Wood Particle Board	560	0.000245	3.632870	0.000158	0.268519
Wool Felt	200	0.002290	0.271862	0.002290	0.271835

therefore, the convective mass transfer resistance has to be of the same unit since both resistances are used in the same equation. For the mass transfer coefficient to be in terms of pressure,

$$\dot{m}'' = h_m \left(\Delta\omega \right) \qquad (2.12.3)$$

but

$$\omega = 0.622 \frac{P_v}{P - P_v} \qquad (2.12.4)$$

By neglecting the vapor pressure in the denominator and substituting equation (2.12.4) in equation (2.11.4) the following expression is obtained

$$\dot{m}'' = h_{m,pr} \frac{0.622}{P} \left(\Delta P_v \right) \qquad (2.12.5)$$

where

$\dot{m}''$ = mass flux.

$h_{m,pr}$ = convective mass transfer coefficient in terms of pressure.

P = total atmospheric pressure.

ω = humidity ratio.

By comparing equations (2.12.5) and (2.12.3),

$$h_{m,pr} = h_m \frac{0.622}{P} \qquad (2.12.6)$$

A general equation for moisture mass change for the nodes falling between the first and the last node is written as

$$\frac{dM_i}{dt} = \frac{P_{v,i-1} - P_{v,i}}{R_{m,i-1} + R_{m,i}} - \frac{P_{v,i} - P_{v,i+1}}{R_{m,i} + R_{m,i+1}} \qquad (2.13)$$

and for the last node

$$\frac{dM_n}{dt} = \frac{P_{v,n-1} - P_{v,n}}{R_{n-1} + R_n} \qquad (2.14)$$

The total change in the moisture mass of one material will be the sum of the moisture mass changes at every node

$$\frac{dM_{tot,j}}{dt} = \frac{dM_{1,j}}{dt} + \frac{dM_{2,j}}{dt} + \ldots\ldots + \frac{dM_{n,j}}{dt} \qquad (2.15)$$

where

j = material number.

Substituting equations (2.12), (2.13), and (2.14) into equation (2.15), an expression relating the total moisture mass change of one material to the pressure difference is obtained

$$\frac{dM_{tot,j}}{dt} = \frac{P_{v,r} - P_{v,1,j}}{R_{m,conv,j}} \qquad (2.16)$$

If more than one material is involved, the total moisture mass change of all the materials combined will be

$$\frac{dM_T}{dt} = \sum_{1}^{j} \left(\frac{P_{v,r} - P_{v,1,j}}{R_{m,conv,j}} \right) \qquad (2.17)$$

The total moisture mass change can be either positive or negative. If it were positive, moisture is adsorbed, otherwise, moisture is desorbed.

In order to solve for the room conditions, equations that relates the total mass change to the room conditions are needed. These can be determined by writing lumped energy and moisture balance equations on the room air as shown in Figure (2.2).

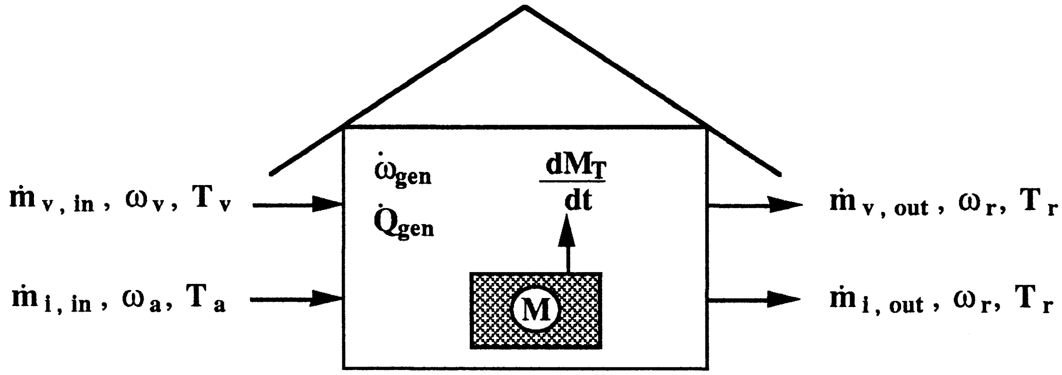


Figure 2.2 Room Energy and mass flows.

Starting from the general energy balance equation, the total load can be expressed as

$$\dot{Q}_{tot} = \sum_{in} \dot{m}h - \sum_{out} \dot{m}h + \dot{Q}_{cv} - \frac{dU}{dt} \qquad (2.18)$$

where

$\dot{Q}_{cv}$ = energy gains from several sources.

$\dot{Q}_{tot}$ = sensible plus latent loads.

Assuming a well mixed room air, the total load can be written as

$$\begin{aligned}\dot{Q}_{tot} = &\sum_{k=1}^{n} h_{t,k} A_k (T_{sur} - T_r) + \dot{Q}_{peo} + \dot{Q}_{int} + \dot{Q}_{z,s} \\ &+ \dot{m}_{inf}(\mathbf{h}_a - \mathbf{h}_r) + \dot{m}_v(\mathbf{h}_v - \mathbf{h}_r) - Cap\left(\frac{T_{rf} - T_{ro}}{\Delta t}\right) \\ &- \rho_a V_a \lambda_{vap} \frac{d\omega_r}{dt} - \frac{dM}{dt}\left\{C_{p,vap}(T_{sur} - T_r) + \lambda_{vap}\right\}\end{aligned} \qquad (2.18a)$$

where

$$\mathbf{h} = C_{p,a}(T_a - T_{ref}) + \omega(\mathbf{h}_g) \qquad (2.18a.1)$$

$$\mathbf{h}_g = \mathbf{h}_{g,ref} + C_{p,vap}(T_{in} - T_{ref}) \qquad (2.18a.2)$$

The latent load equation can be written as

$$\dot{Q}_l = \lambda_{vap}\Big\{ \dot{m}_{inf}(\omega_a - \omega_r) + \dot{m}_v(\omega_v - \omega_r) + \dot{\omega}_I - \frac{dM_T}{dt} - \rho_a V_a \frac{d\omega_r}{dt} \Big\} + \dot{Q}_{peo,l} \qquad (2.19)$$

Subtracting equation (2.19) from (2.18a) and substituting equation (2.18a.1) into (2.18a) the ventilation sensible load equation can be written as

$$\dot{Q}_{v,s} = \dot{m}_v (A_1 + B_1) \qquad (2.19.1)$$

where

$$A_1 = C_{p,a}(T_v - T_r) \qquad (2.19.1.1)$$

$$B_1 = \omega_r[\mathbf{h}_{g,ref} + C_{p,vap}(T_r - T_{ref})] - \omega_r[\mathbf{h}_{g,ref} + C_{p,vap}(T_r - T_{ref})] \qquad (2.19.1.2)$$

Similarly the infiltration sensible load can be written as

$$\dot{Q}_{inf,s} = \dot{m}_{inf}\left(A_1^{'} + B_1^{'}\right) \qquad (2.19.2)$$

where

$$A_1^{'} = C_{p,a}(T_a - T_r) \qquad (2.19.2.1)$$

$$B_1' = \omega_a\left[\mathbf{h}_{g,ref} + C_{p,vap}\left(T_a - T_{ref}\right)\right] - \omega_r\left[\mathbf{h}_{g,ref} + C_{p,vap}\left(T_r - T_{ref}\right)\right] \qquad (2.19.2.2)$$

The terms " B_1 " and " B_1' " are very small compared to the terms " A_1 " and " A_1' ", therefore, the moisture effect on the sensible load can be neglected. The sensible energy transferred with vapor to the room air from the walls and furnishings or vice versa is defined as

$$\dot{Q}_{vap,s} = \frac{dM}{dt} C_{p,vap}\left(T_{sur} - T_r \right) \qquad (2.19.3)$$

The above sensible energy term can be neglected due to the small temperature difference between the room air and the bounding surfaces. Therefore, the total sensible energy equation can be reduced to

$$\dot{Q}_s = \sum_{k=1}^{n} h_{t,k} A_k \left(T_{sur} - T_r\right) - Cap\left(\frac{T_{rf} - T_{ro}}{\Delta t}\right) + 0.3\,\dot{Q}_{peo,s} + C_{p,a}\left\{\dot{m}_{inf}\left(T_a - T_r\right) + \dot{m}_v\left(T_v - T_r\right)\right\} + \dot{Q}_{z,s} + \dot{Q}_{int,s} \qquad (2.20)$$

with the assumption that 70 % of the sensible energy gains from people is absorbed radiation. Consequently, the sensible energy balance equation can be written as

$$Cap\left(\frac{T_{rf} - T_{ro}}{\Delta t}\right) = \sum_{k=1}^{n} h_{t,k} A_k \left(T_{sur} - T_r\right) + 0.3\,\dot{Q}_{peo,s} + \dot{Q}_{int,s} + C_{p,a}\left\{\dot{m}_{inf}\left(T_a - T_r\right) + \dot{m}_v\left(T_v - T_r\right)\right\} + \dot{Q}_{z,s} \qquad (2.20.1)$$

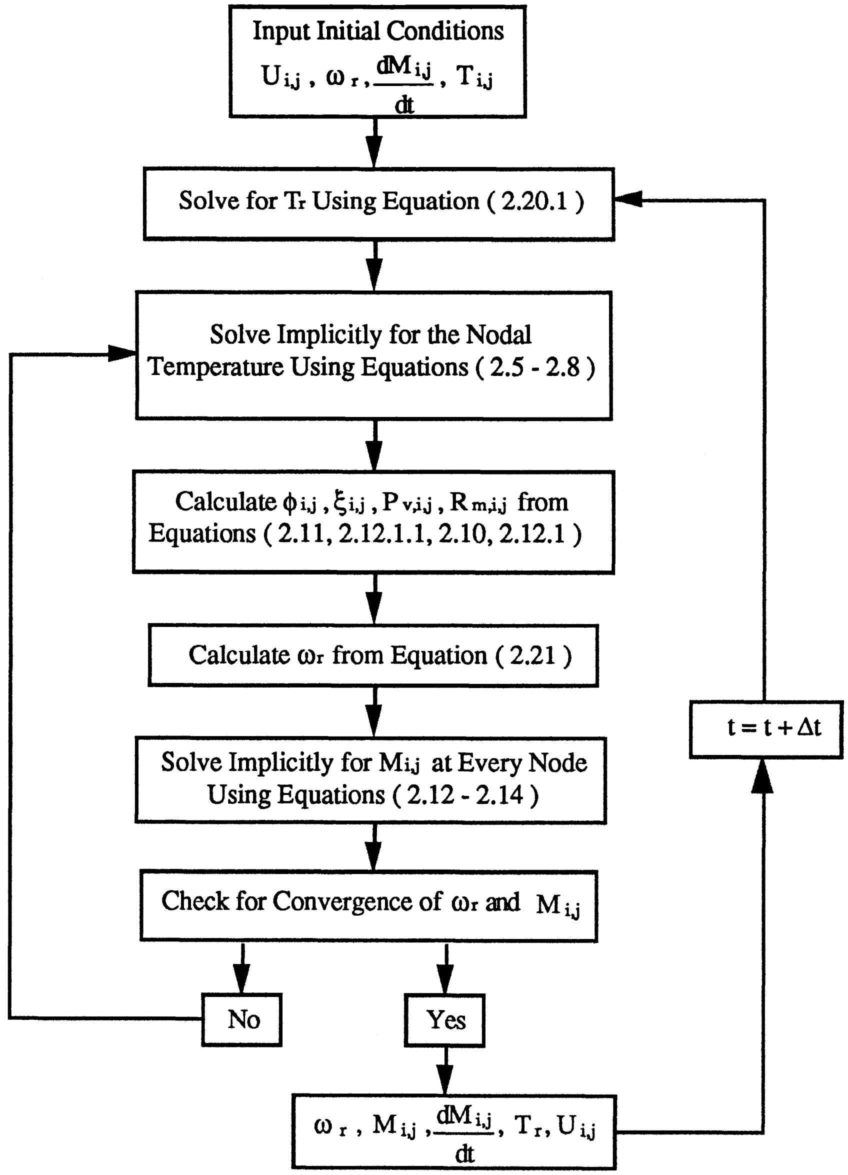


Figure 2.3 The Finite Difference model solution procedure.

and the mass balance equation on the room air can be written as

$$\rho_a V_a \frac{d\omega_r}{dt} = \dot{m}_{inf}(\omega_a - \omega_r) + \dot{m}_v(\omega_v - \omega_r) + \dot{\omega}_I + \frac{\dot{Q}_{peo,s}}{\lambda_{vap}} - \frac{dM_T}{dt} \qquad (2.20.2)$$

The following equation is obtained by substituting equation (2.17) into (2.20.2)

$$\rho_a V_a \frac{d\omega_r}{dt} = \dot{m}_{inf}(\omega_a - \omega_r) + \dot{m}_v(\omega_v - \omega_r) + \dot{\omega}_I + \frac{\dot{Q}_{peo,s}}{\lambda_{vap}} - \sum_{1}^{j} \frac{P_{v,1,j} - P_{v,r}}{R_{m,conv,j}} \qquad (2.21)$$

where $P_{v,r}$ is a related to the room humidity ratio through equation (2.12.4); therefore, in equations (2.20.1) and (2.21) are two independent equations with two unknowns, room temperature and humidity ratio (T_r, ω_r).

Equations (2.5) through (2.21) are solved simultaneously to determine the room humidity ratio; however, equation (2.20.1) is solved independently to determine the room temperature. The solution procedure is explained in the flowchart in Figure (2.3).

II.4 THE EFFECTIVE PENETRATION DEPTH MODEL

The Effective Penetration Depth model was developed by Kerestecioglu *et. al* [12] at the Florida Solar Energy Center. This model presents a simplified method for solving simultaneous heat and moisture transport equations in buildings. The building is composed of three domains, the air, the furniture, and the envelope (walls, floor, and ceiling). Heat and mass transport equations are solved for each domain. Lumped equations are used for the air domain, however, distributed energy equations and lumped or distributed moisture transport equations are used for the other domains.

For short periods of time when the inner regions of the solid material are not significantly affected by the bounding environment, a thin layer of the material that is close to the surface is assumed to participate in the moisture exchange process with the surroundings. If this layer is very thin, lumped equations can be used for the furniture and the envelope; however, if this layer is thick or full thickness of the material is considered, distributed equations must be used. The choice of using lumped or distributed equations depends on mass transfer Biot number [8] (the ratio of the diffusion resistance to the convection resistance) which is defined as

$$Bi_m \equiv \frac{h_m \delta_m}{\xi} \qquad (2.22)$$

where

δ_m = the effective material thickness (penetration depth).

If $Bi_m \leq 0.1$, then lumped equations may be used; otherwise, distributed equations

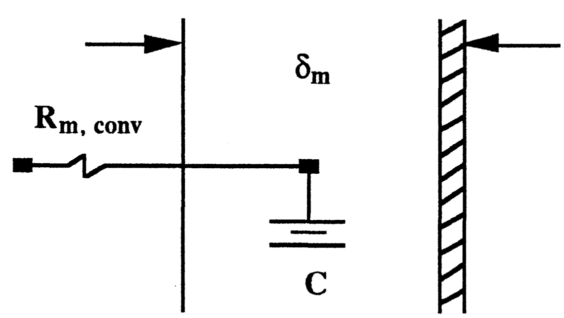


Figure. 2.4 The Effective Penetration Depth model

must be used. If lumped equations were used, then the moisture diffusion resistance inside the solid material is neglected and only the convective resistance is considered. Therefore, the Effective Penetration Depth model will be composed of a convective mass transfer resistance and a capacitor " C " that represents storage as shown in Figure (2.4).

In this model, distributed equations are used for heat transfer, while lumped equations are used for mass transfer.

For this model to be valid, the cyclic amount of the total moisture adsorbed and desorbed must be equal to zero

$$\int_{t_1}^{t_2} \frac{dU}{dt}\, dt \cong 0 \qquad (2.23)$$

where

U = the moisture content of the solid material.

t_1 = initial time.

t_2 = time at which equation (2.23) holds.

For equation (2.23) to hold, a steady state of moisture adsorption and desorption has to be reached; otherwise, the inner layers of the solid materials are affected by the moisture exchange process and therefore, the model fails.

The forcing functions (indoor and outdoor temperature and relative humidity) are usually of an oscillatory periodic nature and for this model to hold, the steady state has to be reached before the amplitude of the periodic function changes.

To understand the effective penetration depth concept further, consider Figures (2.5 (a) and (b)). These figures represent the moisture distribution in the solid material at a certain instant of time using the finite difference solution and lumped solution respectively. In the finite difference solution the moisture content " U " is a function of material thickness, however, in the lumped model " U " is a constant throughout the penetration depth. For

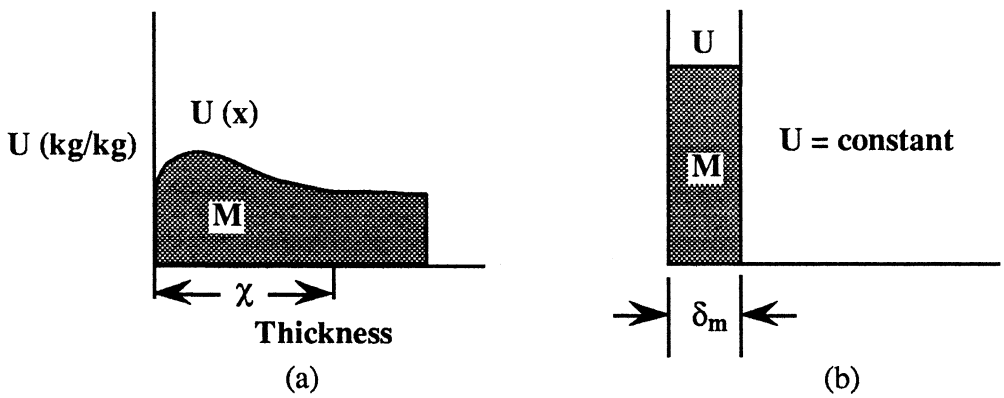


Figure 2.5 The effective penetration depth concept.

the two solutions to match, the change in the amount of moisture " M " stored in the solid material with time should be the same in both cases, therefore the following condition must be satisfied

$$\frac{dM}{dt} = \frac{d}{dt}\int_0^{\chi} U(x)\,dx = \frac{d}{dt} U\,\delta_m \qquad (2.24)$$

In this case, M is represented by the area under each curve. The value of " δ_m " is unique such that there is one and only one value of " δ_m " that satisfies equation (2.24) at all times.

II.4.1 Mathematical Description

The Effective Penetration Depth model is easy to describe mathematically since the moisture transport equations are lumped. The mass transport equation for the solid domain with indoor boundary can be described as

$$\left(A\,\rho_{sd}\,\delta\right)_j \frac{dU_j}{dt} = h_{m,j}\,A_j\left(\omega_r - \omega_j^*\right) \qquad (2.25a)$$

Moisture is also transported through the zone walls from or to the outside by diffusion. Therefore, the mass transport equation for the solid domain with outdoor boundary can be described as

$$\left(A\,\rho_{sd}\,\delta\right)_j \frac{dU_j}{dt} = h_{m,j}\,A_j\left(\omega_a - \omega_j^*\right) \qquad (2.25b)$$

where

ω_j^* = surface air humidity ratio in equilibrium with solid material " j "

The moisture content of the solid material can be represented as a function of the humidity ratio and temperature of the surface air in equilibrium with the solid material, $U = U\left(\omega^*, T^*\right)$, such that

$$\phi^* \cong \frac{\omega^*}{\omega_{sat}^*} \qquad (2.26)$$

where

ω_{sat}^* = the saturation humidity ratio of the surface air in equilibrium with the solid material and it can be defined as

$$\omega^*_{sat} = \frac{1}{R_a \rho_a T^*} \exp\left(23.7093 - \frac{4111}{T^* - 35.45} \right) \qquad (2.26.1)$$

where

R_a = ideal gas constant for air.

By substituting equation (2.26.1) into (2.26), a relation between ω^*, T^* and ϕ is obtained

$$\omega^* = \frac{\phi^*}{R_a \rho_a T^*} \exp\left(23.7093 - \frac{4111}{T^* - 35.45} \right) \qquad (2.27)$$

Substituting equation (2.27) into equation (2.11) and using the chain rule to differentiate the moisture content with respect to time

$$\frac{dU}{dt} = \frac{\partial U}{\partial \omega^*} \frac{\partial \omega^*}{\partial t} + \frac{\partial U}{\partial T^*} \frac{\partial T^*}{\partial t} \qquad (2.28)$$

Equation (2.28) can be expressed as

$$\frac{dU}{dt} = Y_T \frac{d\omega^*}{dt} - Z_\rho \frac{dT^*}{dt} \qquad (2.29)$$

where Y_T is defined as the isothermal moisture capacity

$$Y_T = \frac{\left(a\, b\, \phi^{*b} + c\, d\, \phi^{*d} \right)}{\omega^*} \qquad (2.29.1)$$

and Z_ρ is defined as the thermo-gradient coefficient

$$Z_\rho = -\left[\frac{1}{T^*} - \frac{4111}{\left(T^* - 35.45\right)^2}\right]\left(a\, b\, \phi^{*b} + c\, d\, \phi^{*d}\right) \qquad (2.29.2)$$

By substituting equation (2.29) into equations (2.25a) and (2.25b) the following relations are obtained

$$\left(A\, \rho_{sd}\, Y_T\right)\frac{d\omega^*}{dt} = h_{m,j}\, A_j\left(\omega_r - \omega_j^*\right) + \left(A\, \rho_{sd}\, \delta\, Z_\rho\right)_j \frac{dT_j^*}{dt} \qquad (2.30a)$$

$$\left(A\, \rho_{sd}\, Y_T\right)\frac{d\omega^*}{dt} = h_{m,j}\, A_j\left(\omega_a - \omega_j^*\right) + \left(A\, \rho_{sd}\, \delta\, Z_\rho\right)_j \frac{dT_j^*}{dt} \qquad (2.30b)$$

where equations (2.30a) and (2.30b) are for the solid domain of inside boundary and outside boundary, respectively.

A distributed heat transport equation is used for the solid domain using the heat conduction equation

$$\left(\rho_{sd}\, C_{p,sd}\right)\frac{\partial T_{sd}}{\partial t} = \nabla\left(k \,.\, \nabla T_{sd}\right) \qquad (2.31)$$

where

T_{sd} = temperature of the solid material.

Since equation (2.31) is a partial differential equation, the following boundary conditions are necessary to solve this equation

$$- k \, . \, \nabla T_{sd} = - q_T'' + h_t (T^* - T_r) + \lambda_{vap} \, h_m (\omega^* - \omega_r) \qquad (2.32a)$$

$$- k \, . \, \nabla T_{sd} = - q_T'' + h_t (T^* - T_a) + \lambda_{vap} \, h_m (\omega^* - \omega_a) \qquad (2.32b)$$

where

q_T'' = imposed heat flux.

and equations (2.32a) and (2.32b) are for the inside and outside boundary, respectively.

With respect to the indoor air domain, lumped energy balance and mass balance equations for a zone " zi " are presented as

$$\begin{aligned}(\rho_a \, V_a \, C_{p,a})_{zi} \frac{dT_r}{dt} &= \dot{Q}_{equ,s} + \dot{Q}_{fur,s} + \dot{Q}_{inf,s} + \dot{Q}_{lig,s} \\ &+ \dot{Q}_{mix,s} + \dot{Q}_{peo,s} + \dot{Q}_{sin,s} + \dot{Q}_{sor,s} \\ &+ \dot{Q}_{sup,s} + \dot{Q}_{v,s} + \dot{Q}_{wl,s} \qquad (2.33)\end{aligned}$$

where

$\dot{Q}_{equ,s}$ = energy gains due to equipment.

$\dot{Q}_{fur,s}$ = energy gained or lost by furniture.

$\dot{Q}_{lig,s}$ = energy gains due to lights.

$\dot{Q}_{mix,s}$ = energy gained or lost by internal flows.

$\dot{Q}_{sin,s}$ = energy removed by sinks.

$\dot{Q}_{sor,s}$ = energy gains from sources.

$\dot{Q}_{sup,s}$ = energy gained or lost by supply air.

$\dot{Q}_{wl,s}$ = energy flow between room air and walls.

and

$$(\rho_a V_a)_{zi} \frac{d\omega_r}{dt} = \dot{Q}_{equ,l} + \dot{Q}_{fur,l} + \dot{Q}_{inf,l} + \dot{Q}_{rev,l} + \dot{Q}_{mix,l} + \dot{Q}_{peo,l} + \dot{Q}_{sin,l} + \dot{Q}_{sor,l} + \dot{Q}_{sup,l} + \dot{Q}_{v,l} + \dot{Q}_{wl,l} + \dot{Q}_{con,l} \quad (2.34)$$

where

$\dot{Q}_{equ,l}$ = moisture gained from equipment.

$\dot{Q}_{fur,l}$ = moisture gained or lost by furniture.

$\dot{Q}_{rev,l}$ = moisture gained due re-evaporation.

$\dot{Q}_{mix,l}$ = moisture gained or lost by internal flows.

$\dot{Q}_{sin,l}$ = moisture removed by sinks.

$\dot{Q}_{sor,l}$ = moisture gained from sources.

$\dot{Q}_{sup,l}$ = moisture gained or lost by the supply air.

$\dot{Q}_{wl,l}$ = moisture added or removed by walls, floor and ceiling.

$\dot{Q}_{con,l}$ = moisture due to condensation.

The boundary condition equations (2.32a) and (2.32b) are solved by initializing the unknowns: T_r, T_i^*, ω_r, U and ω_i^*. Using these boundary conditions, equation (2.31) is solved to obtain T_i^* which is substituted in equation (2.33) to obtain T_r. With a known T_r, equations (2.30a), (2.30b,), and (2.34) are solved to obtain ω_i^* and ω_r, respectively. The solution procedure is repeated until ω_i^* and ω_r converge.

The Effective Penetration Depth model is computationally simple and time saving, however, there is no procedure to calculate the penetration depth. It must be determined experimentally, empirically or from detailed simulations.

II.5 THE MODIFIED EFFECTIVE PENETRATION DEPTH MODEL

Based on the Effective Penetration Depth model in Section 2.4, this model takes into consideration the moisture diffusion resistance. In this modified model, lumped transport equations are used for both the solid and the air domains. The Modified Effective Penetration Depth model is a combination of the Finite Difference and the Effective Penetration Depth models. It uses the effective penetration depth principle with one node placed in the middle of the penetration depth. The node conditions (temperature, vapor pressure, moisture content) are assumed to be the penetration depth internal conditions. Consequently, the Modified Effective Penetration Depth model is composed of a convective resistance, a diffusion resistance, and a storage capacitance as shown in Figure (2.6).

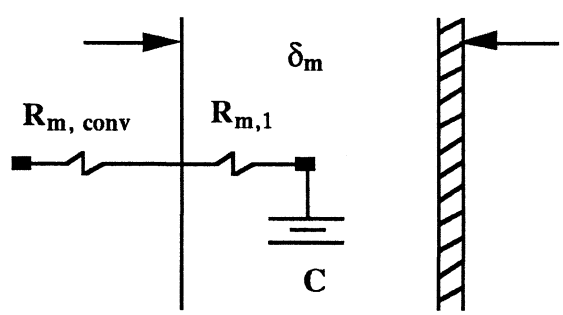


Figure 2.6 The Modified Effective Penetration Depth model.

II.5.1 Mathematical Description

The same restrictions in equations (2.23) and (2.24) still apply after modifying the Effective Penetration Depth model. The equations are very similar to the equations in the

Finite Difference model. However, in this model implicit solutions for node temperatures and moisture masses are eliminated since only one node is involved in the solution.

Unlike the Effective Penetration Depth model, the modified model does not take heat and moisture conduction through the room walls from or to the outside and hence, external walls are modeled as internal partitions.

Starting with the energy balance on the node, equation (2.5) can be written as

$$\frac{T_r - T_1}{R_{conv} + R_1} + \frac{\lambda_{vap}}{A}\frac{dM_1}{dt} = \rho_{sd}\, C_{p,sd}\, \Delta x \frac{dT_1}{dt} \tag{2.35}$$

Equation (2.24), the moisture balance on the first node [6], can be written as

$$\frac{dM_1}{dt} = \frac{P_{v,r} - P_{v,1}}{R_{m,conv} + R_{m,1}} \tag{2.36}$$

where

$$R_{m,1} = \frac{\delta_m}{2\,\xi} \tag{2.36.1}$$

If more than one material is involved in the simulation, the total moisture mass change will be the sum of the moisture mass changes for all the materials, hence, equation (2.17) becomes

$$\frac{dM_T}{dt} = \sum_{1}^{j}\left(\frac{P_{v,r} - P_{v,1,j}}{R_{m,conv,j} + R_{m,1,j}}\right) \tag{2.37}$$

To solve for the room conditions (temperature and humidity ratio), lumped energy and mass balance equations on the room air are used. Equation (2.20.1), the energy balance equation on the room air, stays the same. By substituting equation (2.37) into (2.20.2), the mass balance equation on the room air becomes

$$\rho_a V_a \frac{d\omega_r}{dt} = \dot{m}_{inf}(\omega_a - \omega_r) + \dot{m}_v(\omega_v - \omega_r) + \dot{\omega}_I + \frac{\dot{Q}_{peo,l}}{\lambda_{vap}} - \sum_1^j \frac{P_{v,r} - P_{v,1,j}}{R_{m,conv,j} + R_{m,1,j}} \qquad (2.38)$$

In the penetration depth model, the value of the effective material thickness is unknown, and is determined experimentally or by detailed simulations. Without the effective thickness, the penetration depth model is not very useful. In modifying this model, an effort was made to develop a method to calculate the penetration depth.

In order to maintain similarity between two unsteady-state transient heat transfer processes, equality of the ratio of material physical property to material geometric property is necessary [8]. This ratio is called Fourier number.

Similarly, in unsteady-state moisture transfer processes, equality of mass transfer Fourier number is a necessary condition for similarity if the boundary conditions are the same. Therefore, in estimating the penetration depth, the ratio of the rate of diffusion to the rate of storage of the solid material (mass transfer Fourier number) was assumed constant. Fourier number [9] is defined as

$$Fo \equiv \frac{D_m t}{\delta_m^2} \qquad (2.39)$$

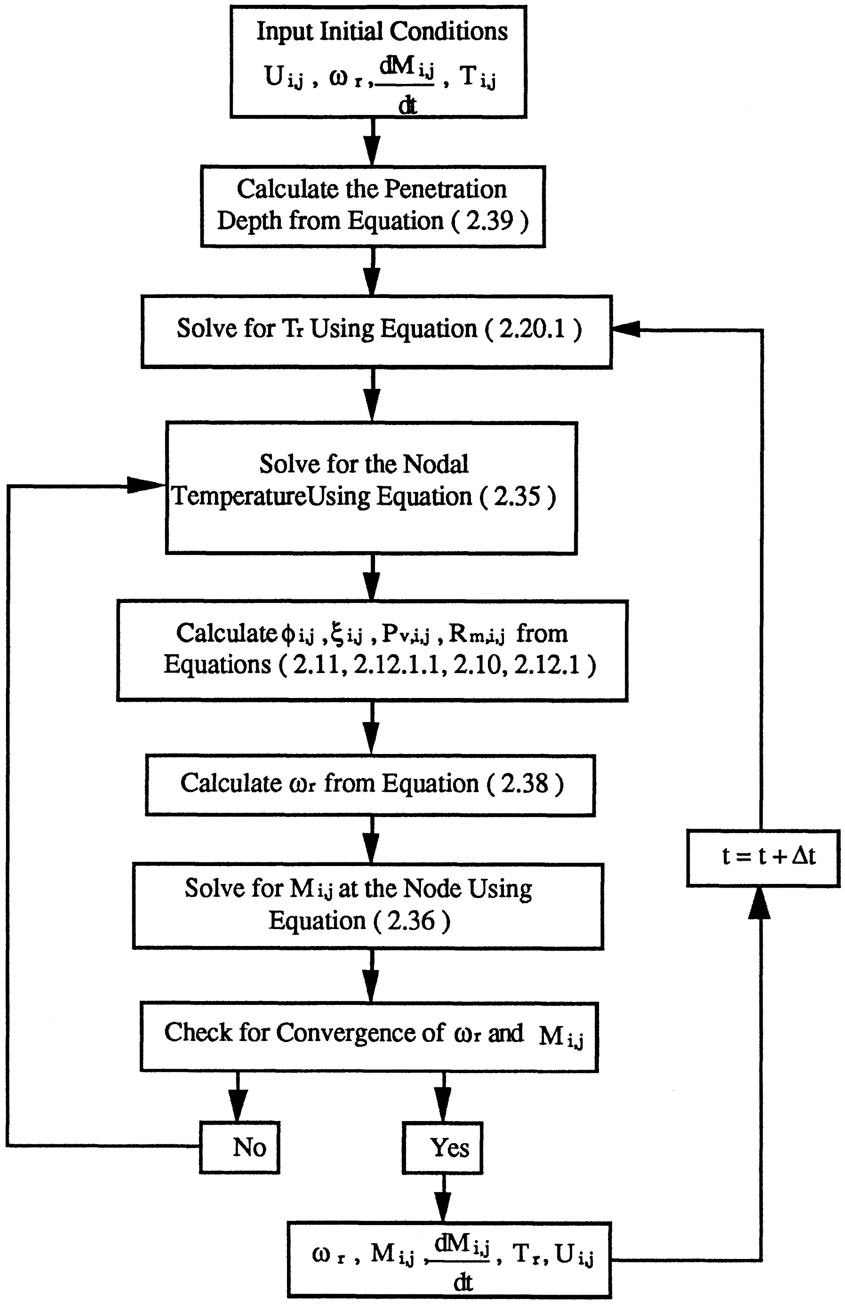


Figure 2.7 The Modified Effective Penetration Depth model solution procedure.

where

D_m = the moisture diffusivity of the solid material.

t = 24 hours, assuming a constant amplitude twenty four-hour periodic change of room and ambient conditions.

To obtain the value of the effective material thickness, knowing the value of " Fo ", the user will have to provide the value of the diffusivity " D_m " which is commonly available.

By assuming a constant Fourier number and moisture diffusivity, the penetration depth becomes automatically constant. Consequently, a change in the room conditions does not significantly affect the thickness of the penetration depth. The constant Fourier number assumption will be discussed in detail and validated in Chapter III.

The solution procedure for the Modified Effective Penetration Depth model is illustrated in the flowchart in Figure (2.7).

Chapter III

Model Testing and Comparisons

In the previous chapters, the Modified Effective Penetration Depth (MEPD) model was presented and mathematically described. In this chapter, TRNSYS simulations are used to compare the MEPD model with other models. Finite Difference TRNSYS simulations were avoided in the comparisons because they are computationally time consuming due to the iterative solutions. Finally, an evaluation of the model from an energy point of view is discussed.

III.1 MODEL TESTING

A FORTRAN subroutine was developed for the purpose of testing the MEPD model. Depending on the number of nodes specified, the subroutine will solve for the nodal moisture content and temperature and the room humidity condition at a specific room temperature at each time step . If the number of nodes specified in the parameters is greater than " 1 ", the subroutine will solve implicitly for the room humidity ratio using the Finite

Difference model, whereas if the number of nodes specified is equal to " 1 ", the subroutine uses the Modified Effective Penetration Depth model to solve implicitly for the room humidity ratio. Flow charts for both models were provided in Chapter II Figures (2.3) and (2.7).

The subroutine uses the ventilation and infiltration humidity ratios and room temperature as driving forces. These sinusoidal 24-hour periodic nature forcing functions were assumed to have a constant amplitude. Reaching a steady state of moisture adsorption and desorption, requires repeated constant amplitude 24-hour periodic driving forces curves; consequently, the model can be used for long term simulations because after reaching a steady state, the penetration depth will no longer be a function of time.

In testing the Modified Effective Penetration Depth model, two sets of forcing functions were assumed to simulate two different conditions as shown in Table (3.1). In the first condition, infiltration was neglected, while in the second condition, infiltration was included and the amplitudes of the temperature and ventilation humidity ratio curves were changed as shown in Figures (3.1) and (3.2).

Table 3.1 Parameters for the first and second testing conditions.

Parameter	**Condition 1**	**Condition 2**
Room volume (m^3)	180	105
Surface area (m^2)	216	142
Ventilation flow rate (kg/hr)	555	1135
Infiltration (kg/hr)	0	30
Heat transfer coefficient ($W/m^2\,^\circ C$)	9	3

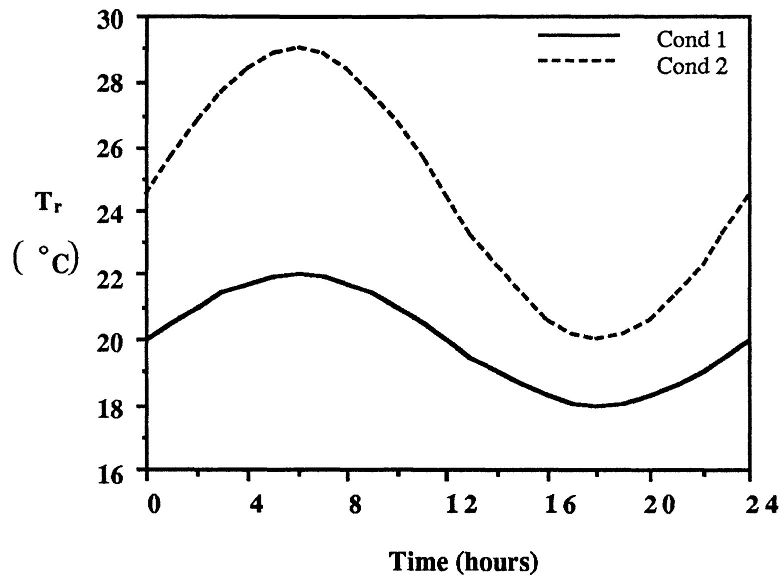


Figure 3.1 Room temperature forcing functions.

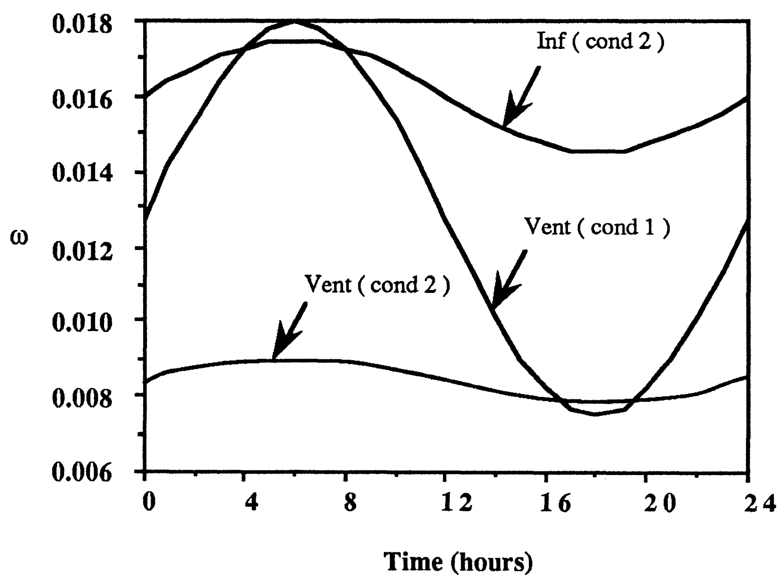


Figure 3.2 Ventilation and infiltration humidity ratio forcing functions.

The main purpose for changing the driving forces curves is to determine the effect of these forces on the value of the penetration depth.

Fiber wood, concrete and plaster board were the three different building materials used to evaluate the MEPD model. These materials are very common and they exist in every building structure. The equilibrium moisture isotherms for these materials are shown in Figure (3.3).

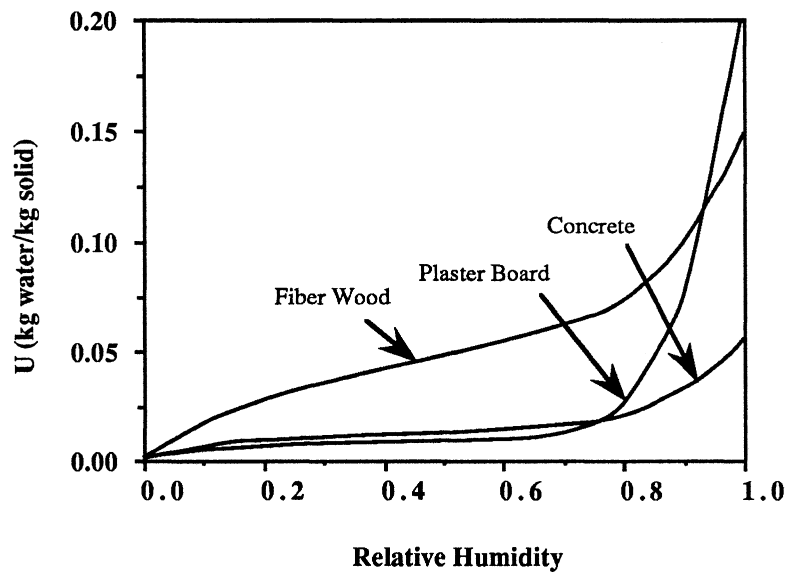


Figure 3.3 Equilibrium moisture isotherms for fiber wood, plaster board and concrete.

In the simulations for the first condition, a 180 m^3 room bounded by 25 mm thick fiber wood was modeled and subjected to the forcing functions shown in Figures (3.1) and (3.2). In the first simulation, the finite difference method with 15 nodes was used to solve for the moisture content and temperature of the wood at every node and the room humidity ratio. Figure (3.4) represents the moisture content curve at every hour for a period of 24 hours after reaching a periodic steady state.

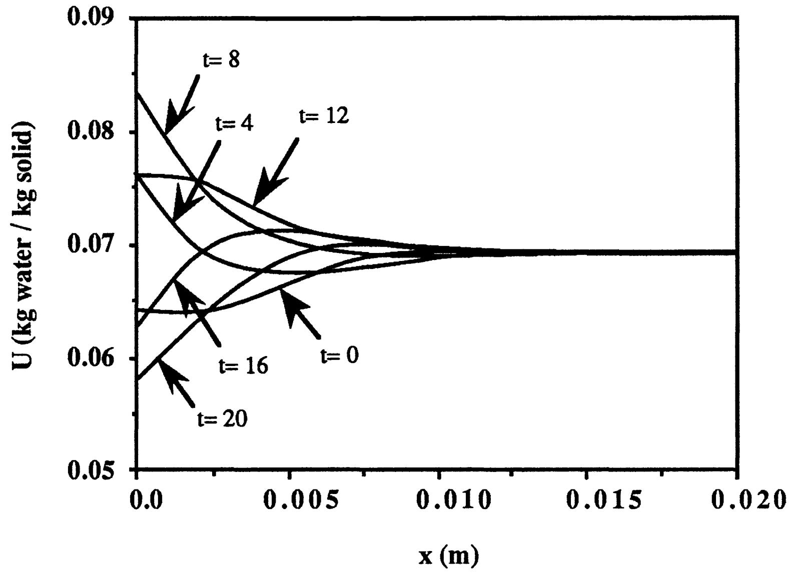


Figure 3.4 Fiber wood moisture content time curves.

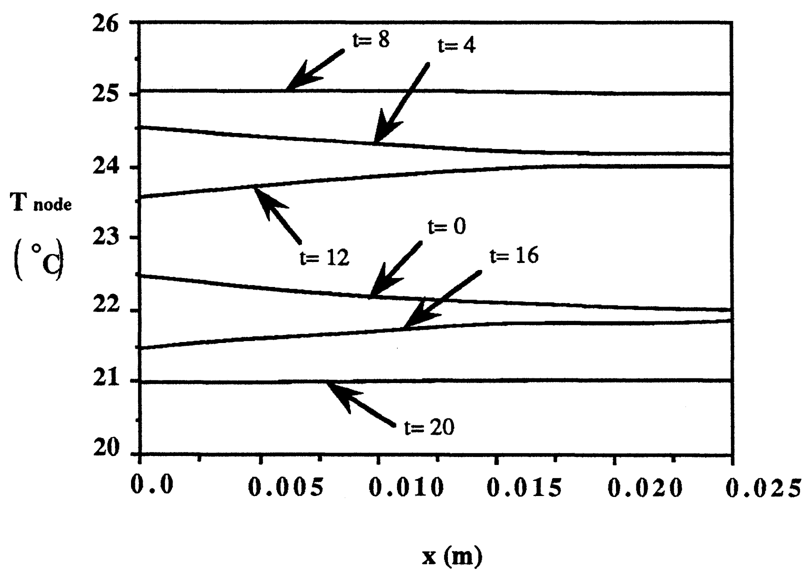


Figure 3.5 Fiber wood node temperature time curves.

As shown in Figure (3.4), approximately 15 mm of wood thickness were affected by moisture adsorption and desorption. Beyond 15 mm the moisture content of the wood is constant at all times and does not contribute in the moisture exchange process; therefore, at 15 mm wood can be assumed impermeable with regard to moisture transfer from the room air.

With respect to the heat conduction, the whole material thickness was affected by the room temperature fluctuation. Figure (3.5) shows the temperature time curves as a function of material thickness for a period of 24 hours after reaching a periodic steady state.

In the second simulation, the MEPD model was used. The full material thickness (25 mm) was used to solve for the node temperature since the whole thickness was affected by the boundary temperature change, whereas the moisture effective penetration depth was determined by trial and error. After several runs with different penetration depths, the MEPD model humidity ratio curve matched the Finite Difference model humidity ratio curve at an effective material thickness of " 3.3 mm ". Equation (3.1) which represents the sum of the squares of the residuals [3] was used to match the results from several runs.

$$S_R = \sum_{0}^{24} \left(\omega_{r,F.D} - \omega_{r,P.D}\right)^2 \qquad (3.1)$$

The best match was the one with the smallest " S_R ". The matching results are provided in Table (3.2).

Table 3.2 Matching results for different values of the penetration depth.

Penetration Depth (mm)	$S_R \times 10^5$	$S_R/S_{R,min}$
1	1.77	21.32
3.3	0.083	1
7	0.23	2.77
10	0.53	6.38
25	1.96	23.61

In the third simulation, the MEPD model was used with full material thickness (2.5 cm). The results of the three simulations are shown in Figure (3.6).

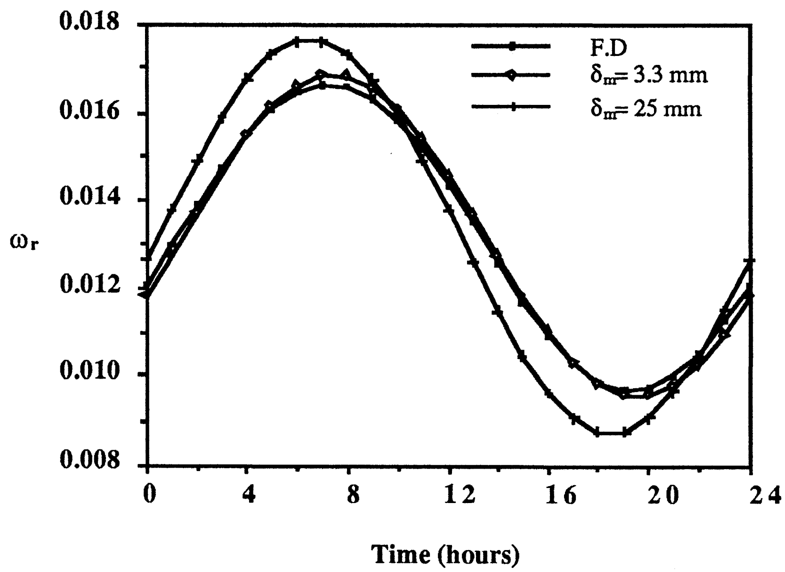


Figure 3.6 Room humidity ratio for fiber wood boundary.

In the above Figure, nearly a perfect agreement exists between the results of the MEPD model with " 3.3 mm " effective moisture penetration depth and the Finite Difference model, whereas discrepancy exists between the results of the MEPD model with full material thickness and the Finite Difference model.

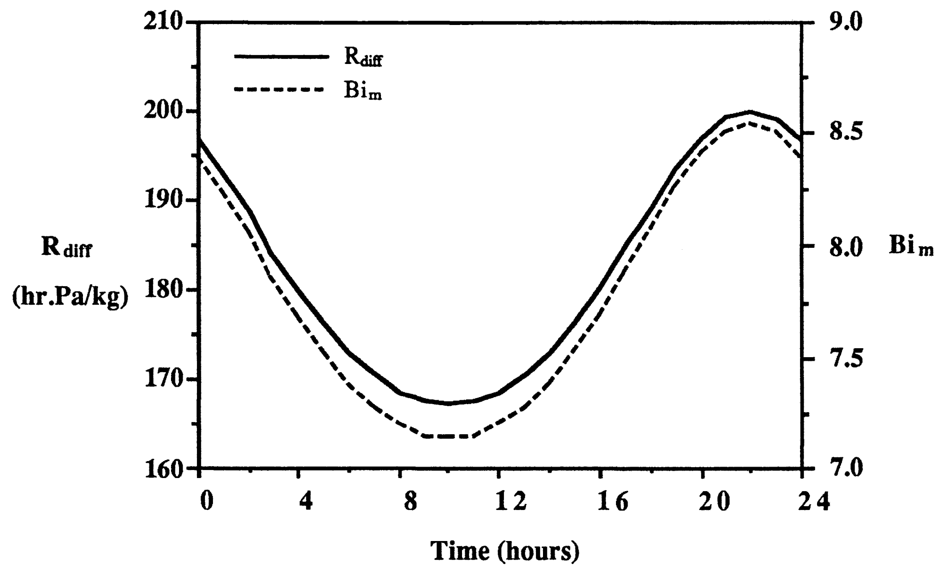


Figure 3.7 Moisture diffusion resistance and mass transfer Biot number for fiber wood boundary.

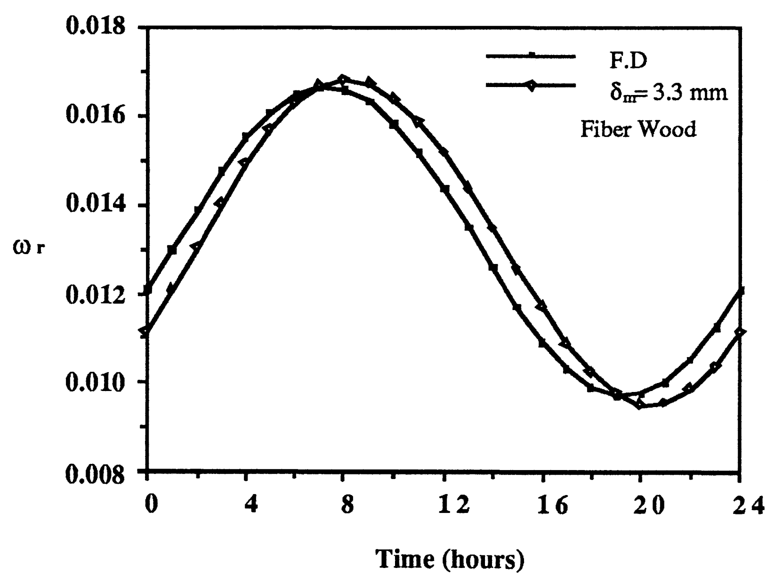


Figure 3.8 Room humidity ratio for fiber wood boundary without moisture diffusion resistance.

Figure (3.7) shows Biot number and the moisture diffusion resistance for the MEPD model as a function of time. According to this figure, the moisture diffusion resistance cannot be neglected since mass transfer Biot number is always greater than " 0.1 ".

Simulations were made to determine the effect of the moisture diffusion resistance. Figure (3.8) presents a comparison between simulation results using the Finite Difference model, the MEPD model with 3.3 mm effective thickness. As shown in this figure, the two curves do not match due to the time shift, therefore, the moisture diffusion resistance cannot be neglected.

In the Effective Penetration Depth model, it was left to the user to provide the value of the penetration depth; however, in modifying this model, similarity between mass transfer processes was used by equating Fourier number for the purpose of finding the penetration depths of other materials.

After obtaining the moisture effective penetration depth for fiber wood, Fourier number " Fo " was calculated at every time step using equation (2.39) and then was averaged over 24 hours to obtain $\overline{Fo} = 11860$. The use of the average Fourier number to determine the effective penetration depths of other materials under the same boundary conditions is discussed later in this chapter.

Simulations were also made with a room bounded by concrete and a room bounded by plaster board using the Finite Difference model. Figures (3.9) and (3.10) show that only 9 mm of concrete and 3.5 mm of plaster board participate in moisture adsorption and desorption.

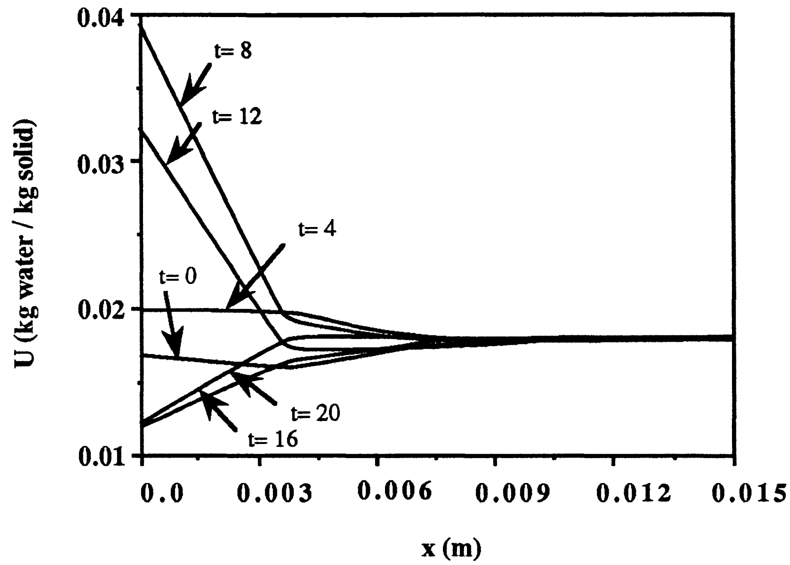


Figure 3.9 Concrete moisture content time curves.

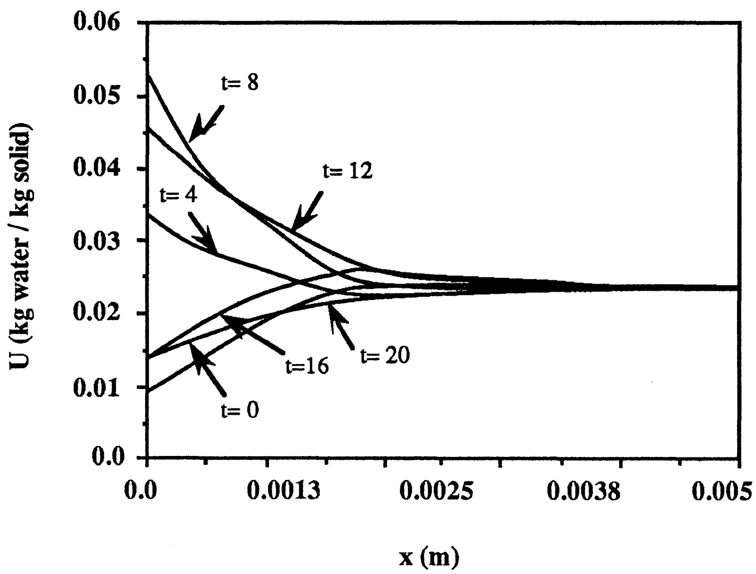


Figure 3.10 Plaster board moisture content time curves.

Similar to the fiber wood, temperature variations were observed within the full thickness of the concrete and plaster board walls as shown in Figures (3.11) and (3.12).

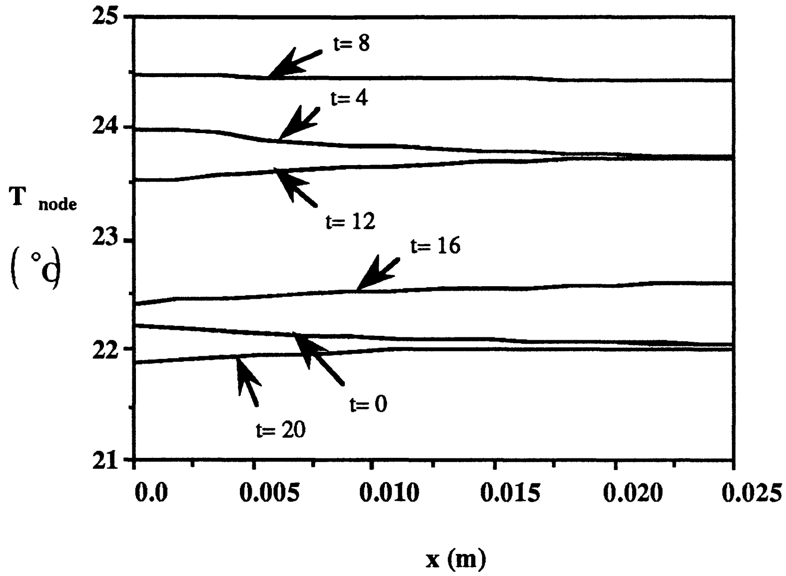


Figure 3.11 Concrete node temperature time curves.

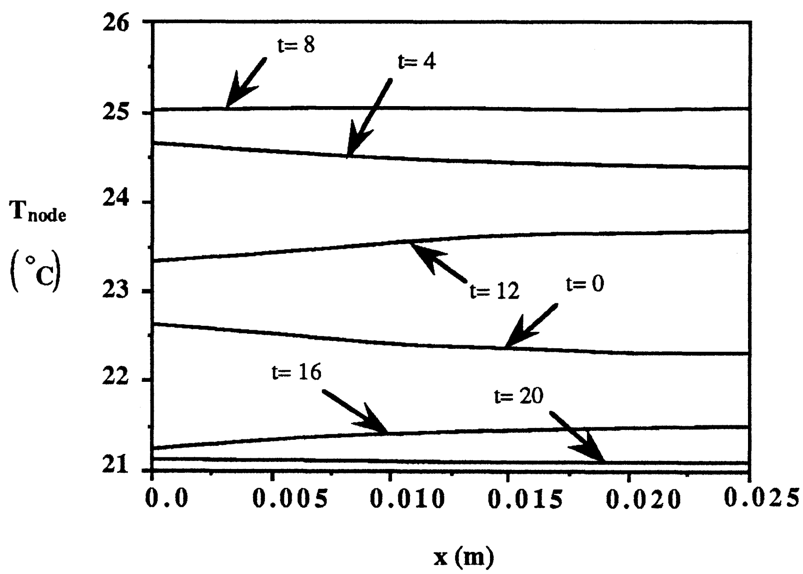


Figure 3.12 Plaster board temperature time curves.

Using the previously obtained average Fourier number ($\overline{Fo} = 11860$) and a constant moisture diffusivity, the effective penetration depths for concrete and plaster board were obtained using equation (2.37). With these moisture effective penetration depths, simulations were made using the MEPD model to examine the validity of equating Fourier number. Material moisture diffusivity for some of the commonly used building materials are provided in Table (3.3.).

Table 3.3 Moisture Diffusivity for different building materials.

Material Name	**Moisture Diffusivity** x 10^4 (m^2/hr)
Carpet	5.56
Concrete	8.47
Fiber Wood	51.04
Paper	4.37
Pine Wood	0.82
Plaster Board	4.70

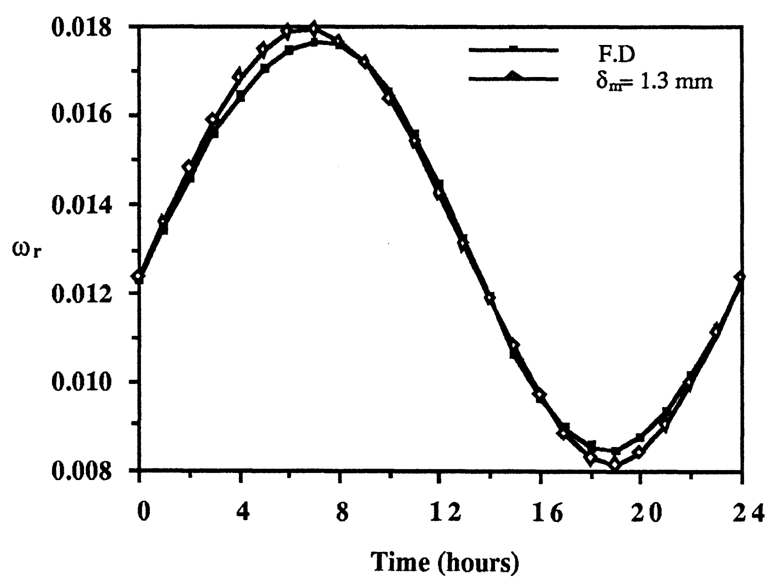


Figure 3.13 Room humidity ratio for concrete boundary.

The simulation results (room humidity ratio curves) obtained using the MEPD model for both, concrete and plaster board were compared with the Finite Difference model curves as shown in Figures (3.13) and (3.14). Figure (3.13) shows a very good agreement between the curves obtained from both models, whereas little discrepancy exists between the curves in Figure (3.14). A trial and error test with different penetration depths was made to determine whether a better fit exists; however, the penetration depth obtained using Fourier number resulted in nearly the best fit after matching the results using equation (3.1). Therefore, the constant Fourier number ($\overline{Fo}$ = 11860) is valid to determine the penetration depth of different materials under the same specific boundary conditions.

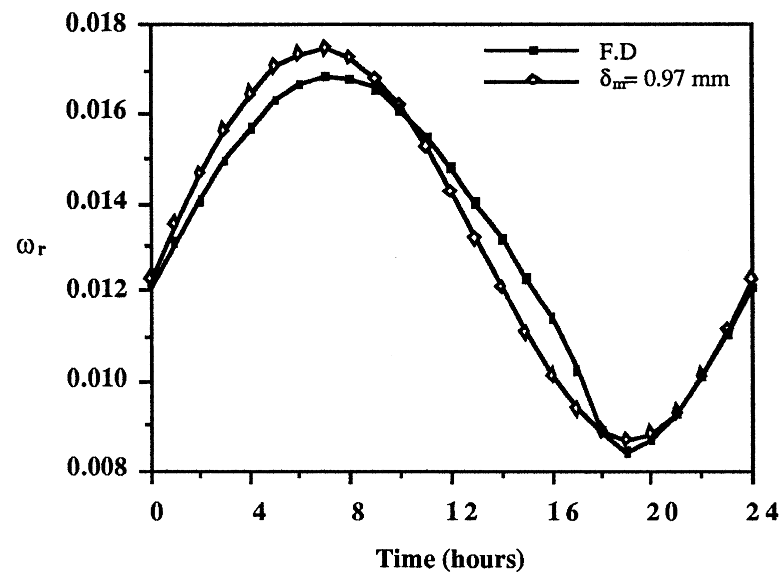


Figure 3.14 Room humidity ratio for plaster board boundary.

A test was made to determine the cause of the discrepancy between the curves in Figure (3.6). Figure (3.15) shows the amount of moisture stored in a room bounded by fiber wood as a function of time. In this figure, the finite difference and the " 3.3 mm "

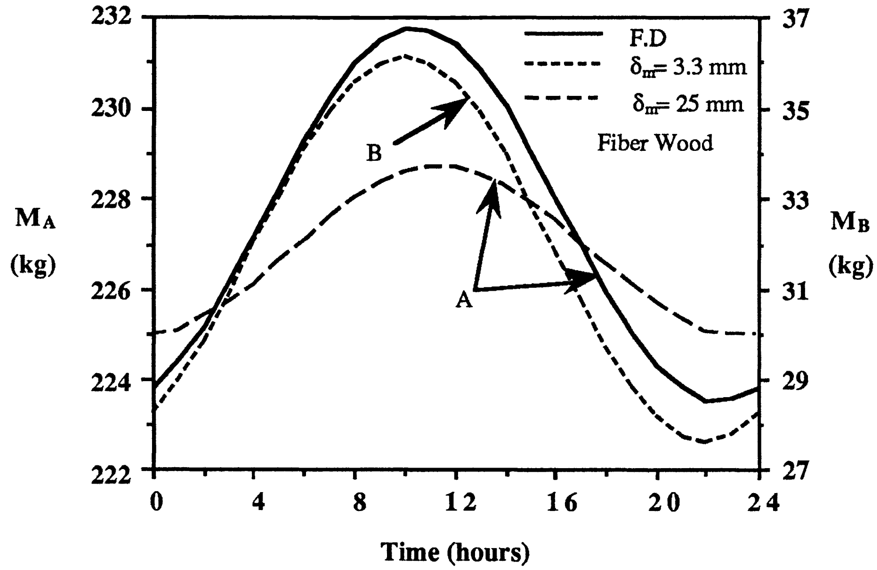


Figure 3.15 Moisture stored in room with fiber wood boundary.

penetration depth curves have approximately equal amplitudes (8.5 kg). These amplitudes are equal to the amount of moisture adsorbed and desorbed during the 24-hour period. However, the amplitude for the penetration depth of full material thickness curve is about 3.5 kg. The difference between the amplitudes (5 kg) caused the discrepancy in the humidity ratio curves. Therefore, the amplitude of the moisture storage curve decreases with increasing the material thickness due to an increase in the moisture diffusion resistance and consequently, lead to more discrepancies between the humidity ratio curves..

The effect of using the material full thickness on the room humidity conditions and cooling loads is presented later in this chapter.

The next step was to determine the effect of boundary conditions (forcing functions) change on Fourier number which causes the MEPD and the Finite Difference models to have the same solution. In this step, the same subroutine was used but with different boundary conditions as shown in Figures (3.1) and (3.2). The amplitudes of the ventilation humidity ratio curve and the room temperature curves were decreased and increased, respectively; moreover, infiltration humidity ratio curve was included.

Using the finite difference method, a simulation was made for a room bounded by fiber wood to obtain a room humidity ratio curve as a function of time. The effective penetration depth was then determined by trial and error using the MEPD model. After several simulations using different penetration depth values, results (humidity ratio curves) were matched using equation (3.1). The best match was obtained at a penetration depth of " 3.7 mm " and the results are shown in Figure (3.16).

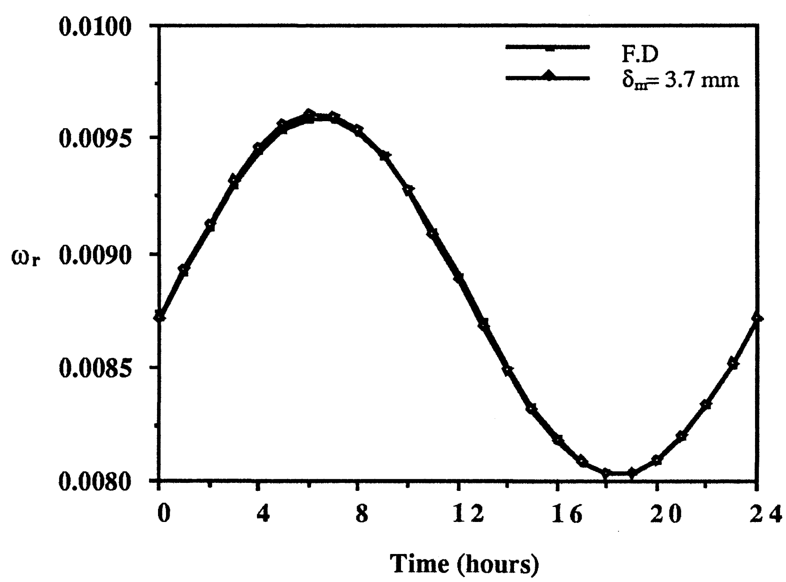


Figure 3.16 Room humidity ratio for fiber wood boundary - second condition.

The Fourier number was then calculated at every time step using equation (2.37) and averaged over 24 hours to obtain $\overline{Fo} = 9133$.

Figure (3.17) shows Fourier number curves obtained from both boundary conditions

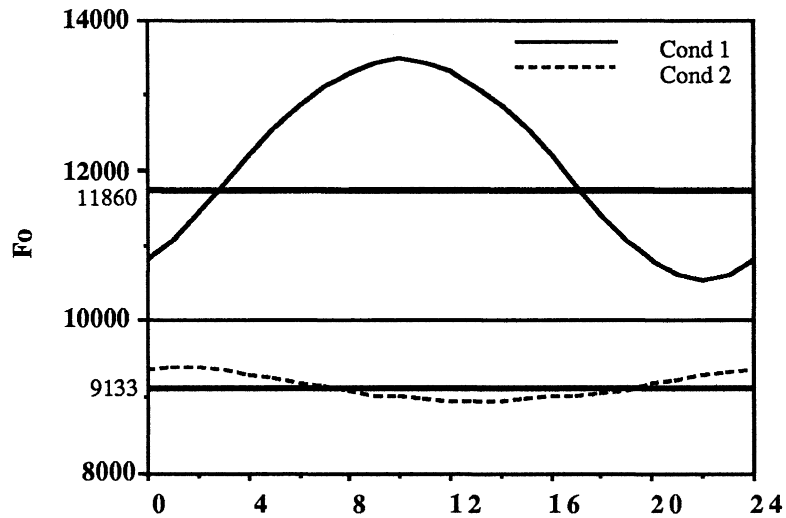


Figure 3.17 Fourier numbers for first and second conditions and their averages.

sets along with their averages. To determine the effect of Fourier number change on the material penetration depth, simulations were made for both, concrete and plaster board using the average Fourier numbers " $\overline{Fo}$ " of 11860, 9133 and 10000. Assuming constant diffusivity, each one of these Fourier numbers should result in a different penetration depth. To determine the significance of the changes in these depths, the room humidity ratio curves resulting from these simulations for both materials were compared.

Figures (3.18), (3.19) and (3.20) prove that although the penetration depth changes with the change in the boundary forcing functions and Fourier number, the effect of these changes on the room humidity ratio is not significant. Therefore, the constant Fourier number ($\overline{Fo}$ = 10,000) assumption is valid.

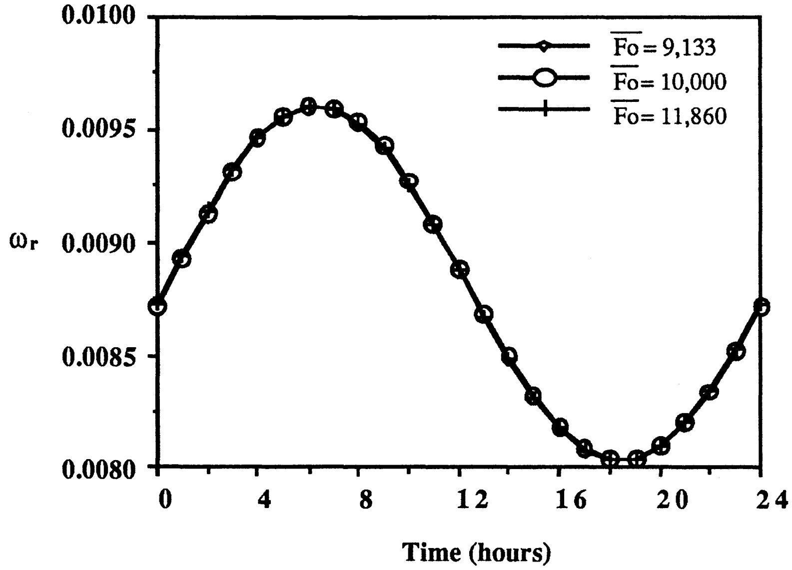


Figure 3.18 Room humidity ratio for fiber wood boundary using three different Fourier numbers.

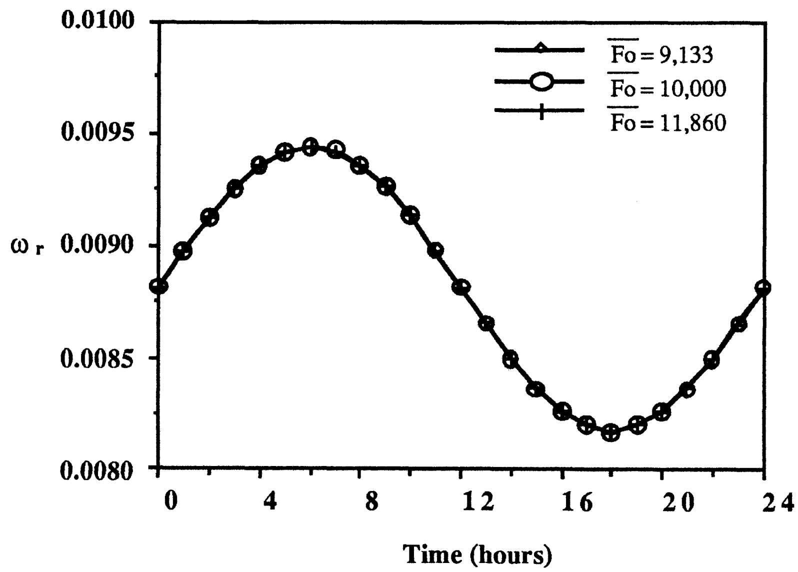


Figure 3.19 Room humidity ratio for concrete boundary using three different Fourier numbers.

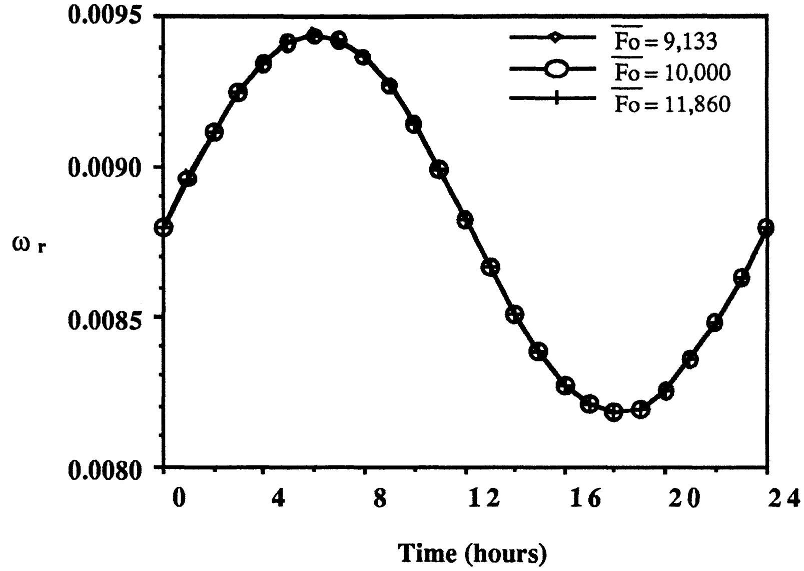


Figure 3.20 Room humidity ratio for plaster board boundary using three different Fourier numbers.

The room humidity ratio curves obtained from using $\overline{Fo}$ = 10000 for both, concrete and

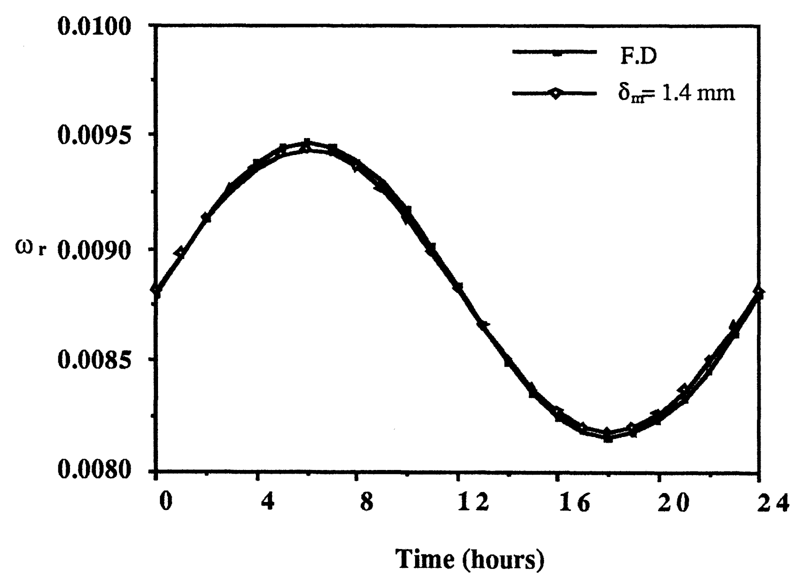


Figure 3.21 Room humidity ratio for concrete boundary - second condition.

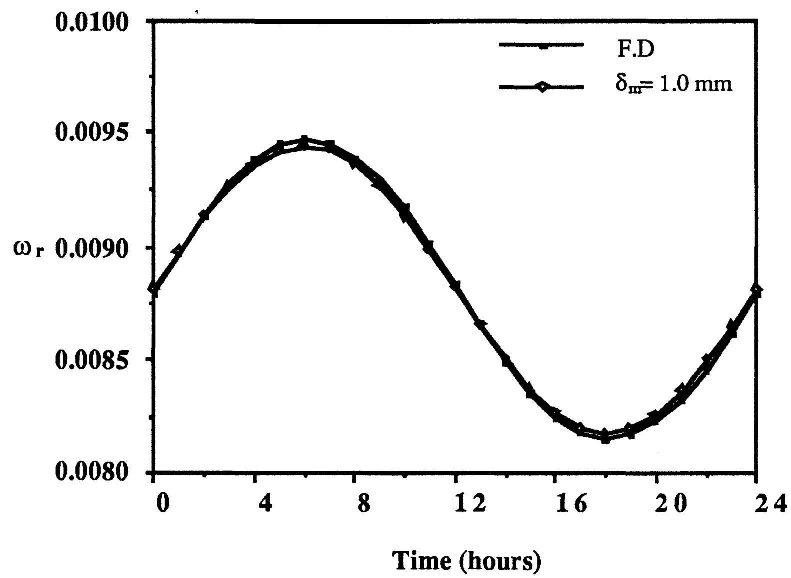


Figure 3.22 Room humidity ratio with plaster board boundary - second condition.

plaster board were then compared with the curves obtained from using the finite difference method. As shown in Figures (3.21) and (3.22), there is a very good agreement between curves obtained from both models for both materials. The effective moisture penetration depths for several building materials obtained using $\overline{Fo}$ = 10000 are provided in Table (3.4).

Table 3.4 Effective moisture penetration depths for different building materials.

Material Name	δ_m (mm)
Carpet	1.1
Concrete	1.4
Fiber Wood	3.5
Paper	1.0
Pine Wood	0.4
Plaster Board	1.0

The last step in testing the MEPD model was to determine whether equation (2.23) is satisfied. Hourly moisture mass changes $\left(\frac{dM}{dt}\right)$ for the three materials (fiber wood,

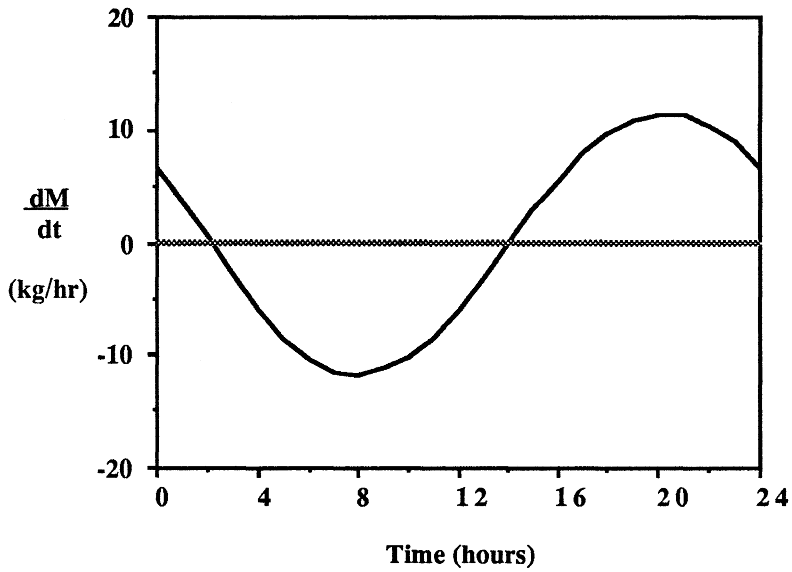


Figure 3.23 Hourly moisture mass change for fiber wood boundary.

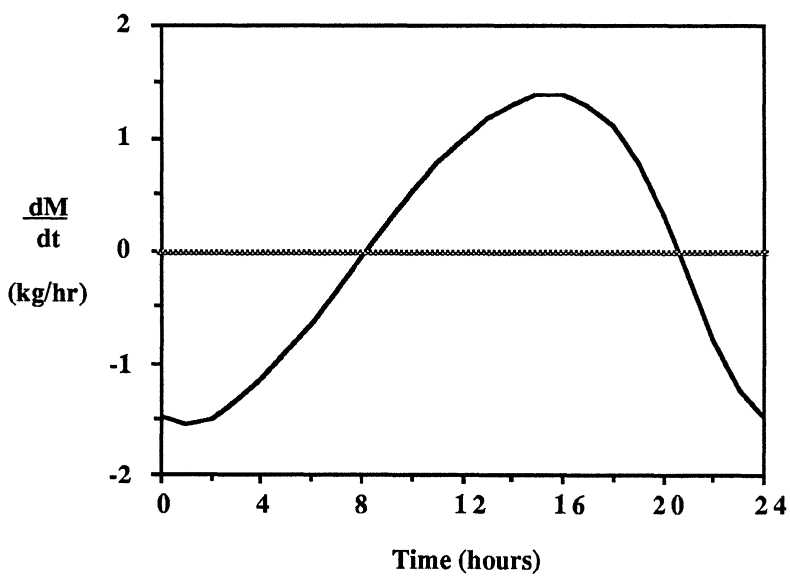


Figure 3.24 Hourly moisture mass change for concrete boundary.

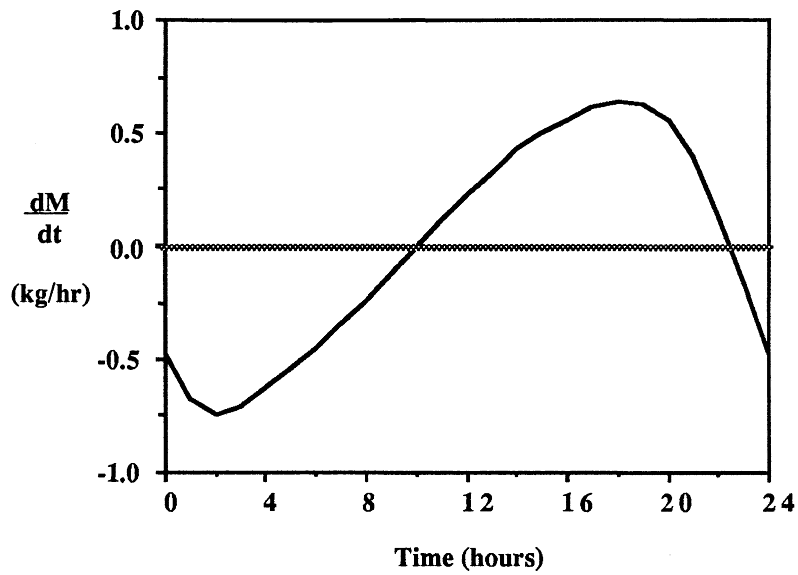


Figure 3.25 Hourly moisture mass change for plaster board boundary.

concrete and plaster board) were plotted versus time as shown in Figures (3.23), (3.24) and (3.25).

In each of these figures, the area under the curve is equal to " zero ", therefore, the cyclic integral of moisture adsorption and desorption is equal to " zero " and equation (2.23) is satisfied.

III.2 THE EFFECT OF MOISTURE STORAGE ON COOLING SYSTEM ENERGY CONSUMPTION.

In the previous section, it was shown that the MEPD model agrees with the Finite Difference model for predicting the indoor conditions. In this section, the MEPD model is implemented in TRNSYS-TYPE 19. Simulation results from using the existing TRNSYS model and modified TRNSYS model are compared to determine the significance of moisture storage in furnishings and structures and its effect on occupant comfort and the cooling system energy consumption.

III.2.1 The Effective Air Mass Multiplier Model Results

The TRNSYS deck described in section 1.4 was used to run simulations. In the first case, room air was assumed to be the only moisture storage media (current TRNSYS model). In the second case, the room air mass was magnified five times (F= 5) to account for moisture storage in furniture and structures. In this case, TRNSYS-TYPE 19 was modified for this purpose by replacing equation (2.2) by equation (2.3). Simulations were run using repeated Miami weather data for a typical day in July. In both cases, the room is cooled from 6 a.m. to 5 p.m. with a mixture of recycled 90 % indoor and 10 % outdoor air, respectively. The cooling system is controlled by a thermostat to keep the room at a temperature between 24 °C.and 26 °C. During the night, the room is vented with 100 % outdoor air. This is very similar to a cooling strategy used in some commercial buildings in order to conserve energy. The TRNSYS deck used in the following simulations is listed in Appendix " A ".

Figures (3.26) and (3.27) compare the indoor conditions for both cases after reaching a steady state. Magnifying the air mass had very little effect on the room temperature, however it significantly affected the room humidity ratio. By increasing the room air mass, the sharp drops and rises of the room humidity ratio were replaced by gradual rises and drops due to the increase in the moisture stored in the room air.

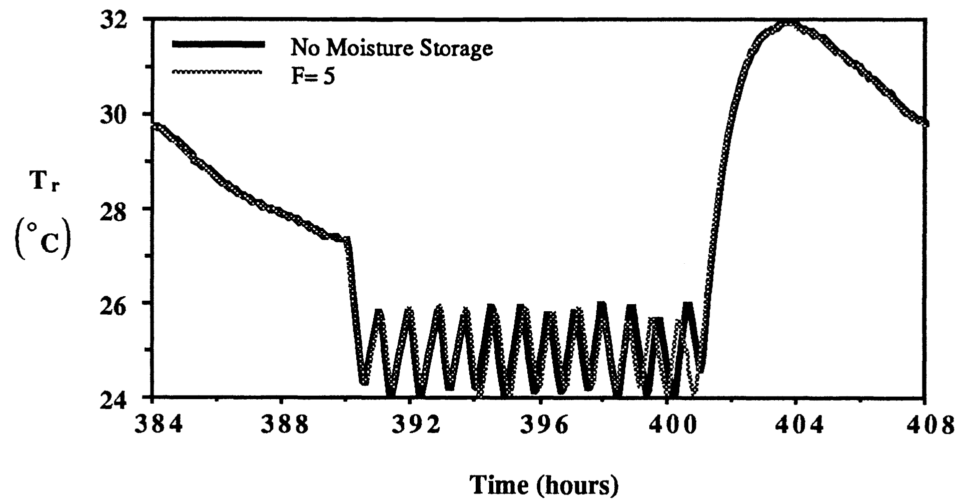


Figure 3.26 Room temperature before and after air mass magnification.

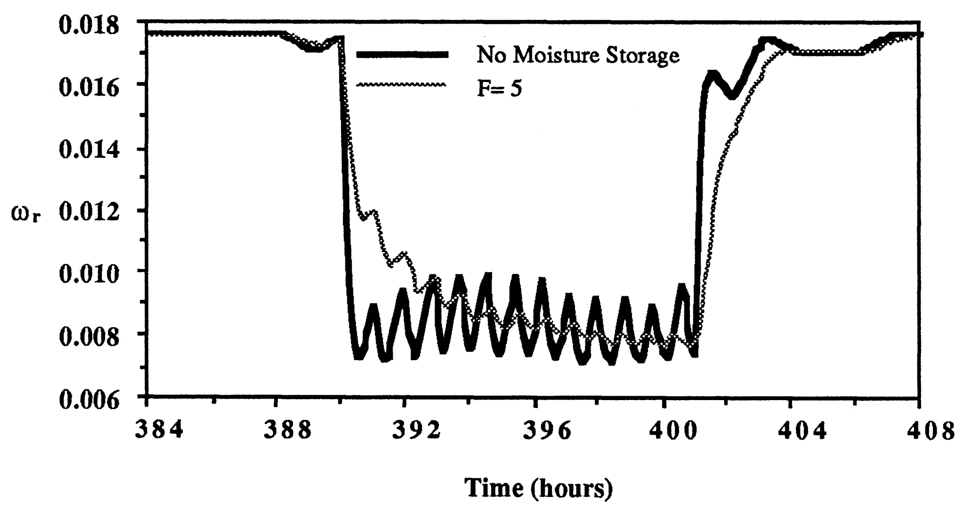


Figure 3.27 Room Humidity ratio before and after air mass magnification.

Figures (3.28), (3.29) and (3.30) show the sensible, latent and total load time distributions. As shown in these figures, the artificial magnification of the air mass had

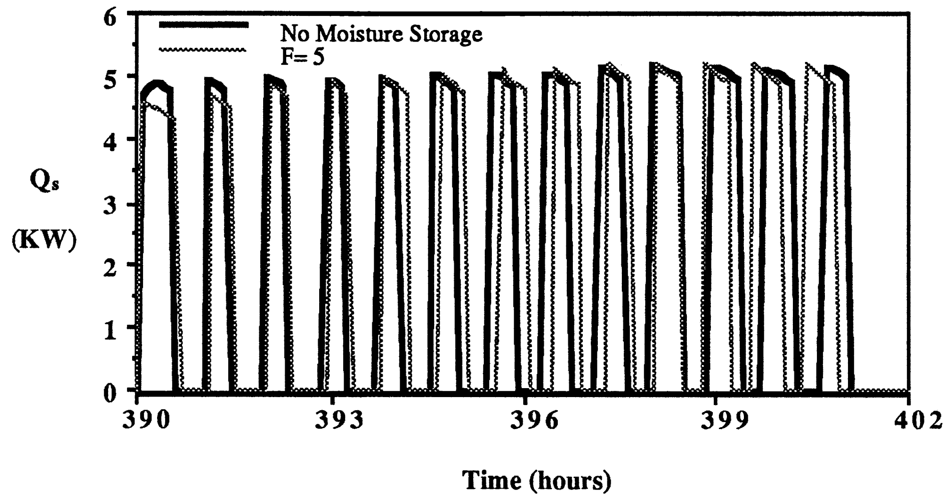


Figure 3.28 Sensible load distribution before and after air mass magnification.

little effect on the sensible load, but substantial effect on the latent load; especially during early cooling hours when the air is loaded with moisture stored during the night hours as a result of outdoor air ventilation.

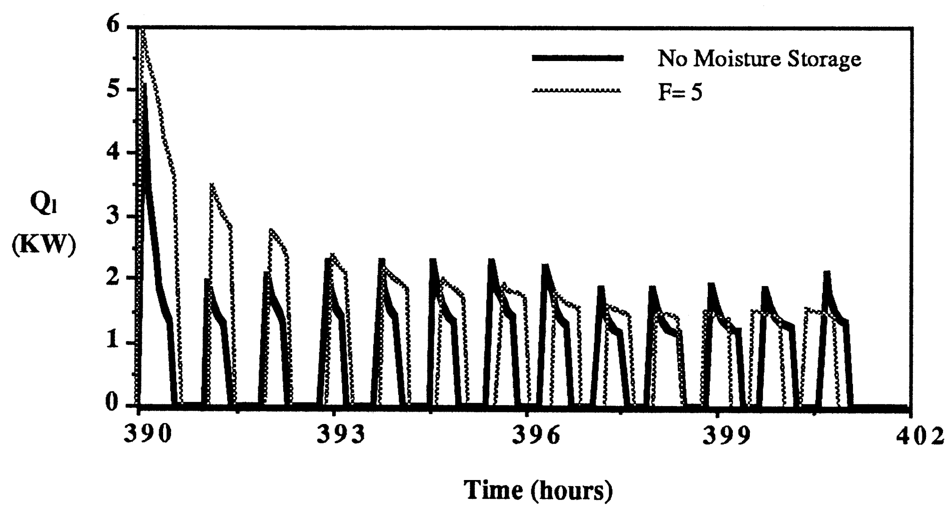


Figure 3.29 Latent Load distribution before and after air mass magnification.

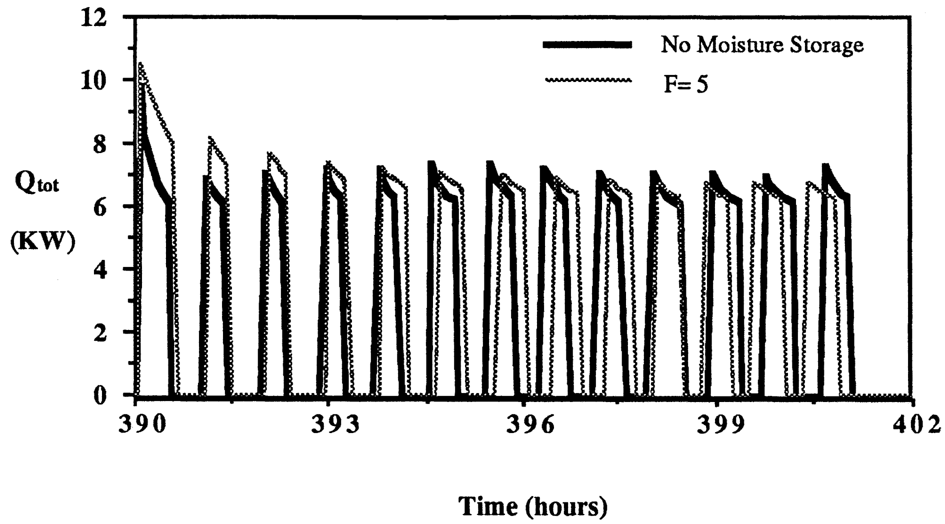


Figure 3.30 Total load distribution before and after air mass magnification.

The effect of moisture storage is seen in Tables (3.5) and (3.6). These tables present the cooling loads and the percent differences based on the cooling loads obtained by

Table 3.5 Cooling loads for magnifying the air mass by a factor of " 1" and " 5 ".

Air Mass Multiplier	Qs (kw)	Ql (kw)	Qtot (kw)
No Moisture Storage	324.25	109.63	433.88
F= 5	323.12	142.20	465.32

Table 3.6 Percent differences in the loads based on the loads due to air mass magnification.

Air Mass Multiplier	% Diff(sen)	% Diff(lat)	% Diff(tot)
No Moisture Storage	0.35	22.90	6.75
F= 5	-	-	-

magnifying the air mass by a factor of " 5 ". The percent difference in the total load was small since the latent load is represented only by approximately one third of the total load.

III.2.2 TRNSYS Modification

TRNSYS-TYPE 19 was modified to include the MEPD model. In this modification, the calculation procedure described in the flow charts in Figures (2.3) and (2.7) were used and equations (2.1) and (2.2) were replaced by the equations specified in these flow charts. This modification allows TRNSYS to model moisture storage in room air as well as its furnishings and structures. Moreover, the heat of sorption that may affect the room temperature was included.

The new TYPE 19 has the ability to model moisture storage in up to 25 different materials at the same time. The user has the choice of using either the Finite Difference or the MEPD model by specifying the number of nodes. If the finite difference solution is chosen, the solution procedure in Figure (2.3) is used, otherwise, the solution procedure in Figure (2.7) is used. Although the use of the MEPD model is recommended due to its computational simplicity, the user may also choose a combination of both models, i.e use one-node model for one material and Finite Difference model for another; however, this will increase the computation time because iterative solutions are used. The new TYPE 19 is listed in Appendix " B ".

To be compatible with the new TYPE 19, the TRNSYS deck used in the previous simulations was modified to include number of nodes, material properties (diffusivity, specific heat, thermal conductivity, material thickness, surface area and some necessary regression constants). The modified TRNSYS deck is listed in Appendix " A ".

The new deck was used to run simulations using the MEPD model for a room with concrete walls, fiber wood ceiling and furniture and carpeted floor. Measurements were made in 7 of the Solar Laboratory rooms in order to develop a relationship between the room volume and the walls and furnishings surface areas. The data obtained were linearly regressed and the following equation was obtained

$$A_{sur} = 2.13\ V + 43 \tag{3.2}$$

where $V \geq 35\ m^3$.

Using equation (3.2), the total surface area in a room of a given volume can be estimated.

III.2.3 Comparisons

The modified TRNSYS deck was used to run simulations for a partially operated cooling system (night ventilation strategy) in order to determine the effect of moisture stored in room furnishings and structures on indoor conditions cooling loads during the day.

Figures (3.31) and (3.32) compare the indoor conditions for the MEPD model and current TRNSYS model (no moisture storage in furnishings and structures). As shown in Figure (3.31), the effect of moisture storage on the indoor temperature during cooling hours was not significant. A slight time distribution difference was observed later during the day due to the different rates of temperature drop and rise.

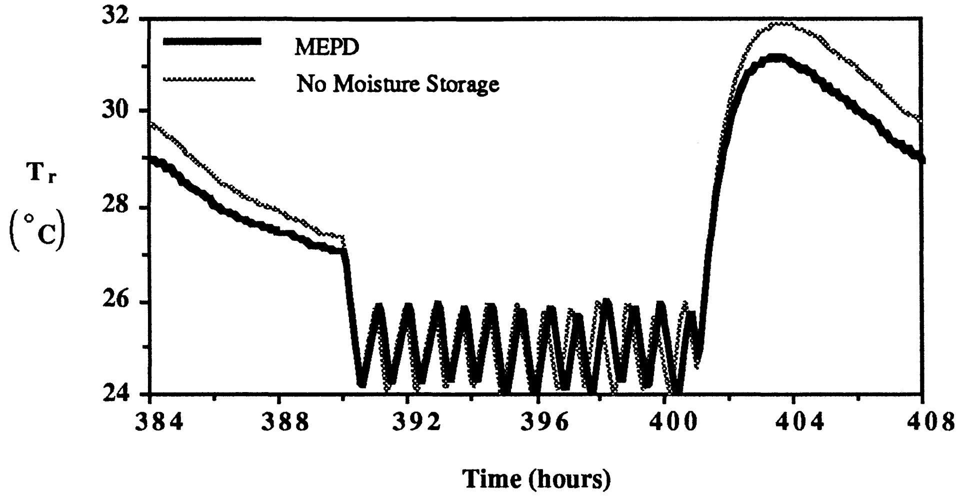


Figure 3.31 Room temperature for the MEPD vs. old TRNSYS models.

Figure (3.32), shows the effect of moisture storage on the room humidity ratio. In the

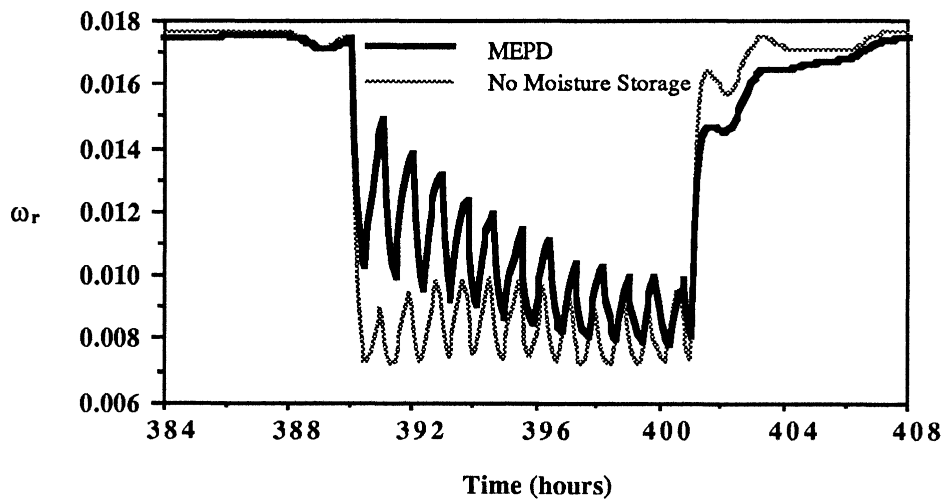


Figure 3.32 Room humidity ratio for the MEPD vs. old TRNSYS models.

case of no moisture storage (F= 1), the room humidity ratio drops quickly when cooling starts, whereas it drops gradually in the case of moisture storage (MEPD model). There is a great discrepancy in the room humidity ratio between both models; especially in the first seven hours of cooling. During the night, when the room is vented with 100 % outdoor air, the room furnishings and structures tend to adsorb the moisture in this highly humid air. In this case, the furnishings and structures behave like a capacitor or electric battery. The more the electricity stored in the battery, the longer the time it takes to be completely discharged. Similarly, the more the moisture stored in the room air, furnishings and structures during the night the longer the time it takes to dehumidify the room air. This explains the quick and gradual drop in the room humidity ratio.

Figure (3.33) compares the sensible load distribution for both models. In this figure, the integrated sensible load is not significantly affected by moisture storage, however, the slight time shift in the temperature curve produces a slight time shift in the sensible load distribution.

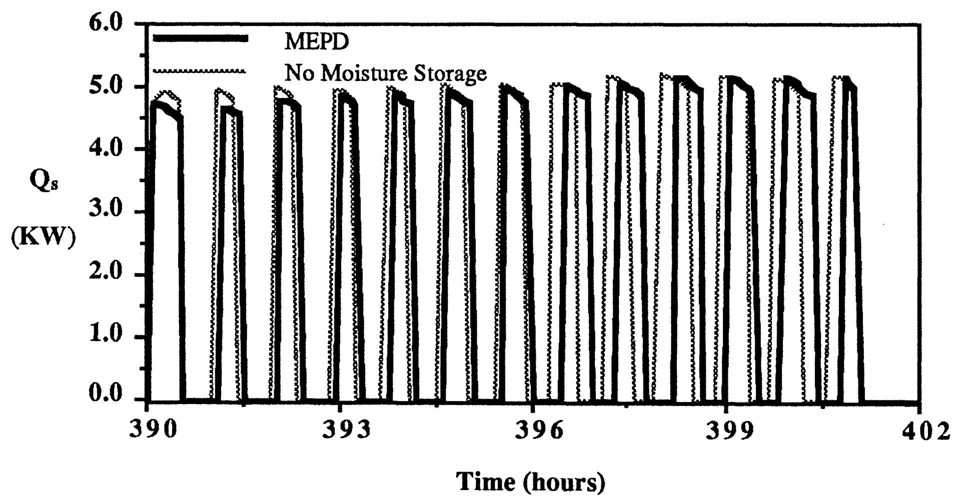


Figure 3.33 Sensible load distribution for the MEPD vs. old TRNSYS models.

Moisture storage during the night increases the energy consumed to dehumidify the room air in the morning. As shown in Figure (3.34), the latent load is greater in the

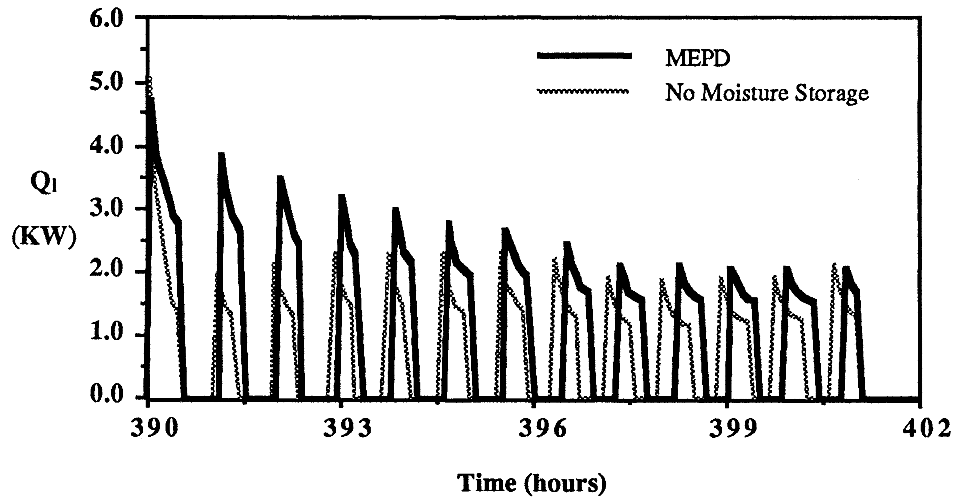


Figure 3.34 Latent load distribution for the MEPD vs. old TRNSYS models.

case of moisture storage especially in the early cooling hours. This produces peak difference accompanied by a time distribution difference.

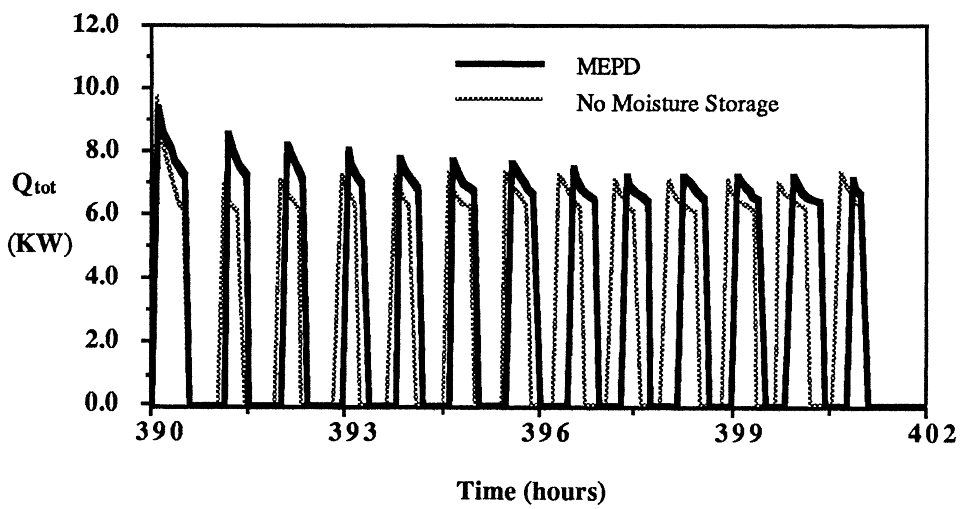


Figure 3.35 Total load distribution for the MEPD vs. the old TRNSYS models.

Figure (3.35) shows the total load distribution with and without moisture storage. The difference in the latent load does not significantly affect the value of the total load, however, it affects the time distribution of the curve.

The MEPD model results were compared to the Effective Air Mass Multiplier model (F= 5) results. Figure (3.36) compares the temperature curves of both models. In this figure, a time distribution difference in the late cooling hours still exits due to the difference in temperature drop and rise rates. This is due to the differences in the storage media properties and diffusion resistances.

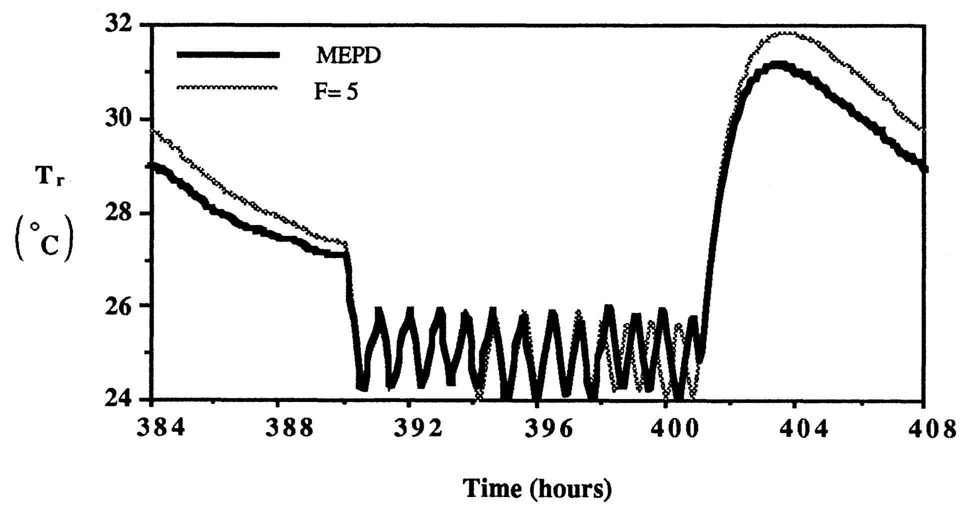


Figure 3.36 Room temperature for the MEPD vs. Air Mass Multiplier models.

Figure (3.37) shows the difference in the humidity ratio curves for the two models. Although there is a gradual drop in the humidity ratio in both models, there are peak and time distribution differences due to the difference in the way moisture storage is modeled.

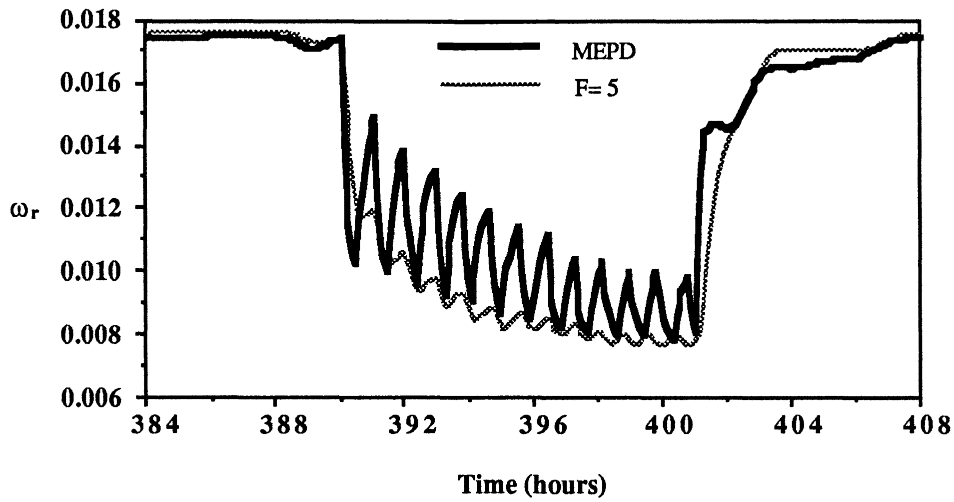


Figure 3.37 Room humidity ratio for the MEPD vs. Air Mass Multiplier models.

Figures (3.38), (3.39) and (3.40) show the sensible, latent and total load distribution curves, respectively. With respect to the sensible loads, there are load time distribution differences in the late cooling hours due to temperature drop and rise rates differences.

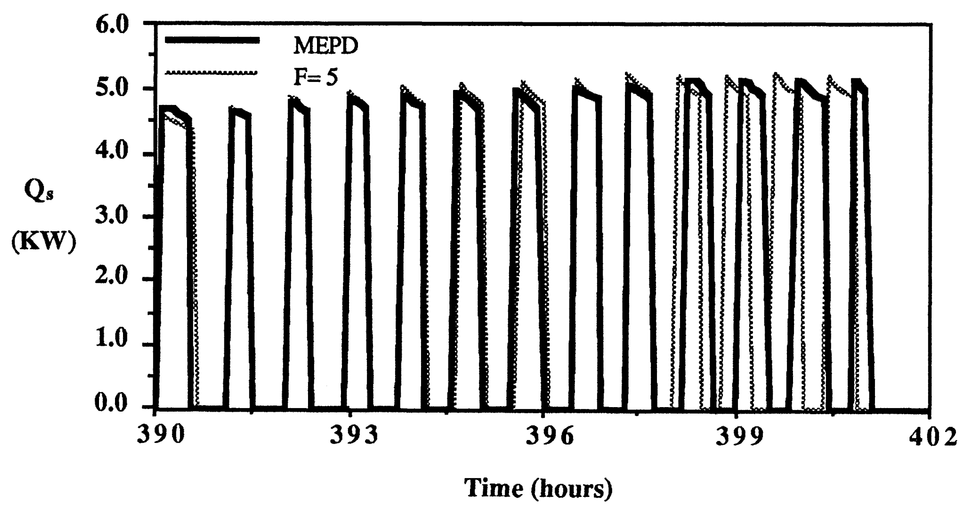


Figure 3.38 Sensible load distribution for the MEPD vs. Air Mass Multiplier models.

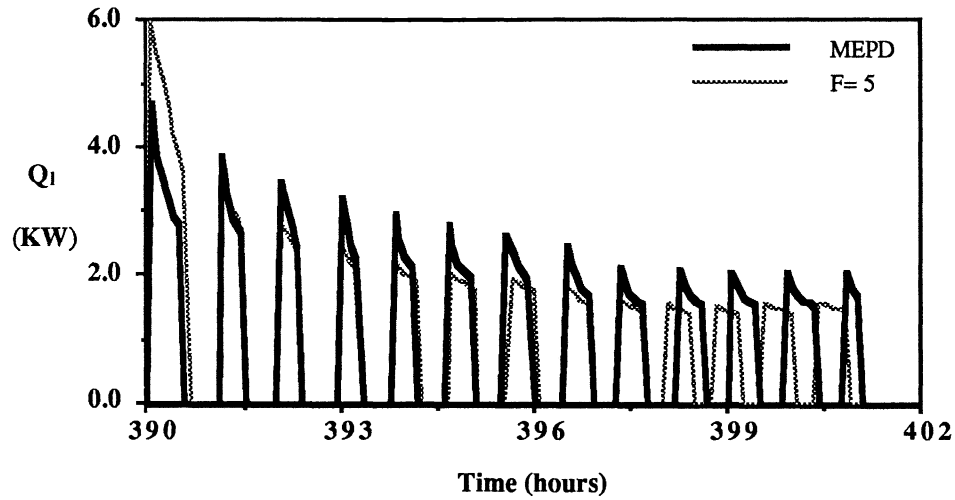


Figure 3.39 Latent load distribution for the MEPD vs. Air Mass Multiplier models.

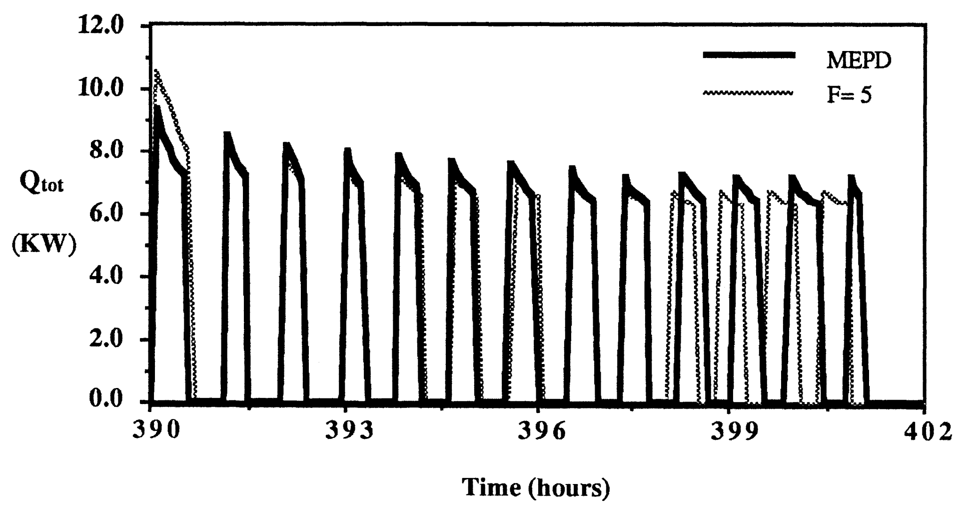


Figure 3.40 Total load distribution for the MEPD vs. Air Mass Multiplier models.

In Figure (3.39), there is a difference in the peak load in the early cooling hours between both models, however, differences in the loads tend to get smaller by time. Moreover, there is a difference in the load time distribution in the late cooling hours.

With respect to the total load, the difference in the time distribution of the sensible and the differences in the peak loads and time distribution of the latent loads caused a total load peak and slight time distribution differences. This leads more cooling load for the Effective Air Mass Multiplier model and vice versa for the MEPD model.

Simulations were made with the MEPD model using full material thickness. Using the full thickness of the material increases the moisture diffusion resistance which makes the amount of moisture adsorbed and desorbed very small. This is analogous to neglecting moisture storage in the building furnishings and structures. Figure (3.41) shows the humidity ratio curves for the calculated penetration depth and the full material thickness. As expected, using full materials thicknesses caused sharp drops and rises in room humidity ratio similar to the case when moisture storage was neglected. This is due to the small amount of moisture exchanged with the room air.

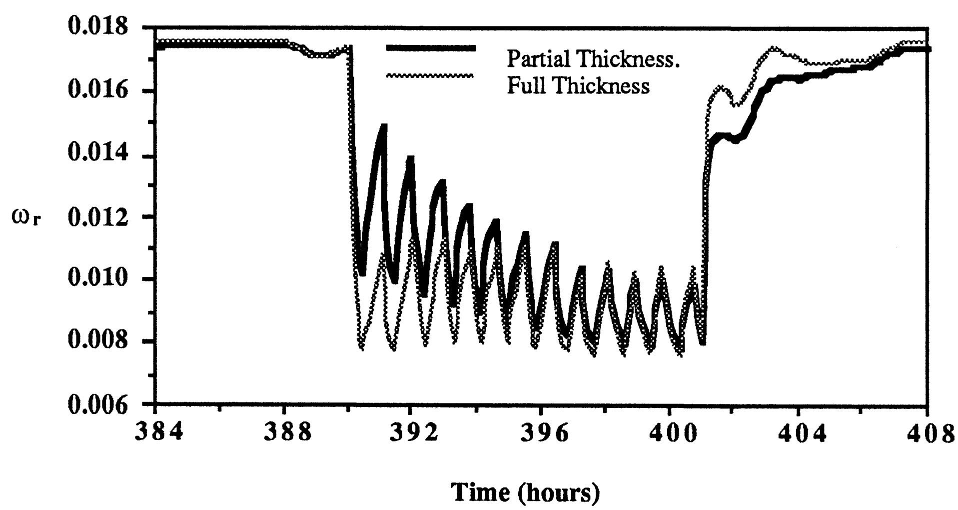


Figure 3.41 Humidity ratio for the MEPD model with partial and full material thickness.

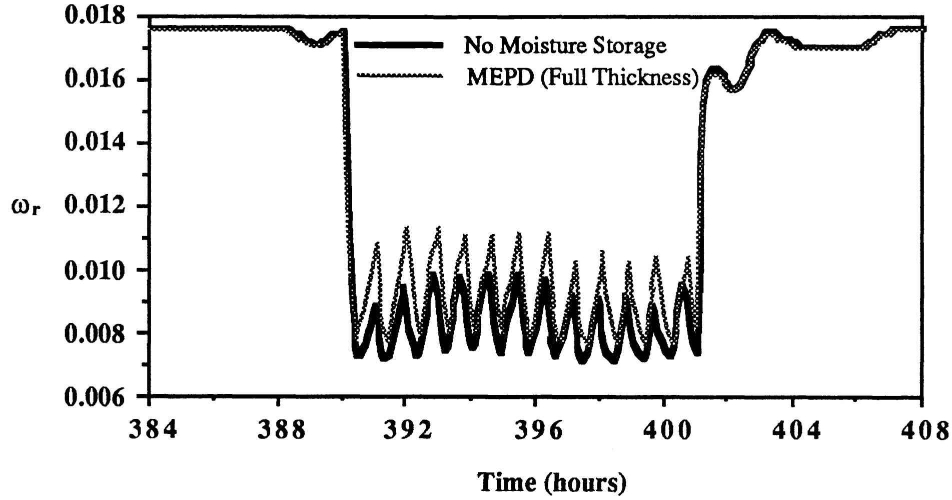


Figure 3.42 Humidity ratio for the MEPD with full material thickness vs. old TRNSYS models.

Figure (3.42) compares the humidity ratio of the MEPD model with full material thickness to the old TRNSYS model. As shown in this figure, there is a good

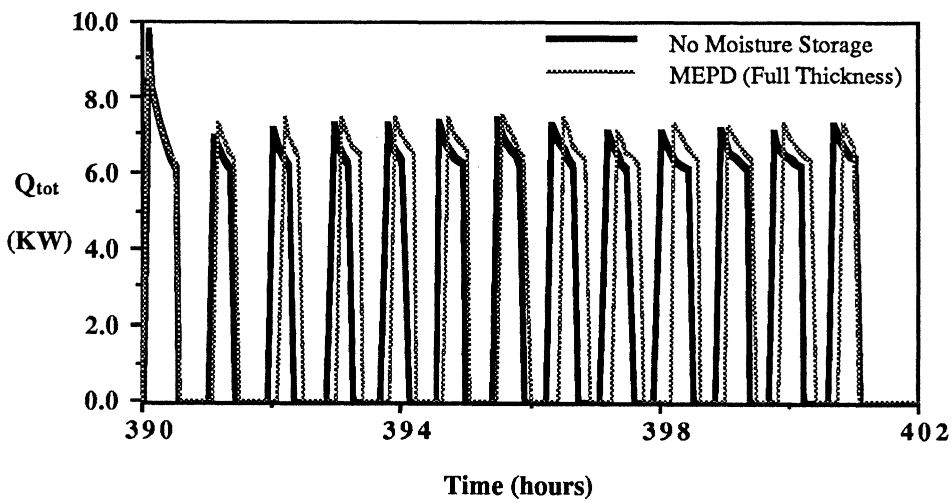


Figure 3.43 Total load distribution for the MEPD with full material thickness vs. old TRNSYS models.

expected agreement between results from both modes.

With respect to the total integrated load, there is a very good agreement in the value of the integrated load and the load time distribution between the MEPD model with full material thickness as the moisture penetration depth and the old TRNSYS model as shown in Figure (3.43). The cooling loads results and the percent differences based on the MEPD model loads are presented in Tables (3.7) and (3.8), respectively.

As shown in Table (3.7), moisture storage had some effect on the integrated value of the sensible load; however it had a significant effect on the integrated value of the latent

Table 3.7 Cooling loads for the MEPD and Effective Air Mass Multiplier models.

Model	Qs (kw)	Ql (kw)	Qtot (kw)
MEPD	296.78	139.52	436.30
No Moisture Storage	324.25	109.63	433.88
F= 5	323.12	142.2	465.32
MEPD (Full Thickness)	296.31	116.01	412.32

Table 3.8 Percent differences in the loads based on the MEPD model loads.

Model	% Diff(sen)	% Diff(lat)	% Diff(tot)
MEPD	-	-	-
No Moisture Storage	9.25	21.42	0.55
F= 5	8.87	1.92	6.65
MEPD (Ful l Thickness)	0.15	16.85	5.50

load. The effect of moisture storage on the total load was very small since the percent difference in the sensible load was significant. This is mainly due to a numerical error resulting from the different number of iterations the system has to go through before convergence is reached. The difference in the value of the sensible load was made up by a difference in the value of the latent load. Increasing the mass of moisture in the air, narrowed down the margin of difference for the latent load between the Effective Air Mass Multiplier and the MEPD models by an approximate factor of " 10 "; however, it did not significantly affect the sensible nor the total loads.

Increasing or decreasing the air mass might lead to a match in the results of both models, however, there still be differences in the load time distribution and peak loads that could affect the optimal control of the cooling system which affects the cooling energy costs. On the other hand, the percent differences in the latent and total loads increases when using the MEPD model with full material thickness.

In the last simulation, the MEPD model was used, but the cooling strategy was such that the system is in continuous operation rather than partial operation (night ventilation). This will keep the moisture stored in the room furnishings and structure approximately constant.

Figure (3.44) shows the temperature curves for both strategies. In this figure, the room temperature is oscillating between 24 $^{\circ}$C and 26 $^{\circ}$C which are the maximum and minimum controller temperature settings. There are no peak differences, however, there are time distribution differences, therefore, people will be at the same level of temperature comfort in both strategies.

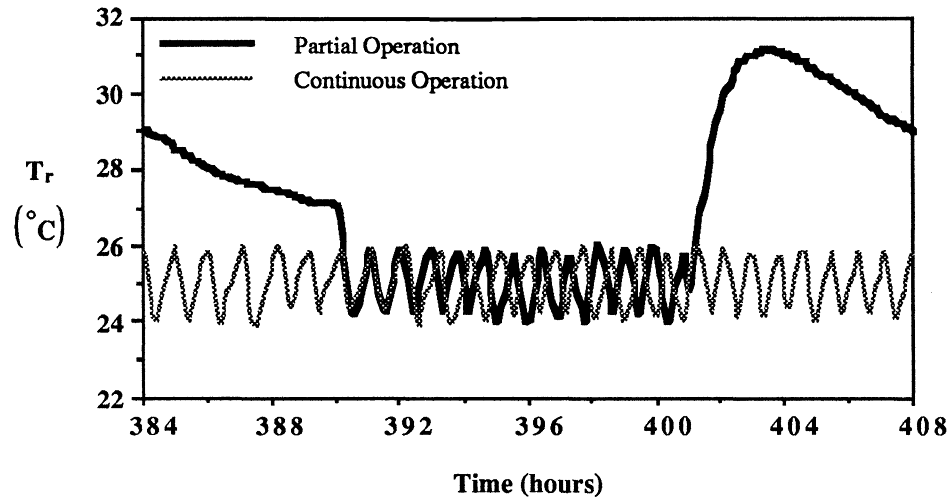


Figure 3.44 Room temperature for partial vs. continuous operations.

The humidity ratio curves for both strategies are shown in Figure (3.45). In the partial cooling strategy, although the temperature is in the comfortable region, the humidity ratio level that is high enough to cause occupant discomfort during the first 3 hours of cooling. This can be seen in Figure (3.46) that shows the summer time comfort region on

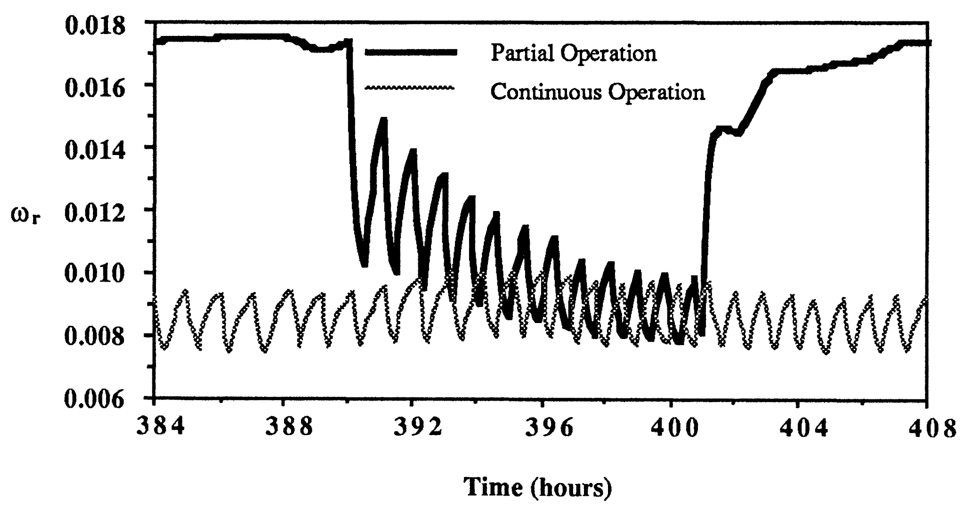


Figure 3.45 Room humidity ratio for partial vs. continuous operations

the psychometric chart as published by ASHRAE [1]. The peak temperatures and humidity ratios of first four cooling hours do not fall within the specified region of comfort, whereas

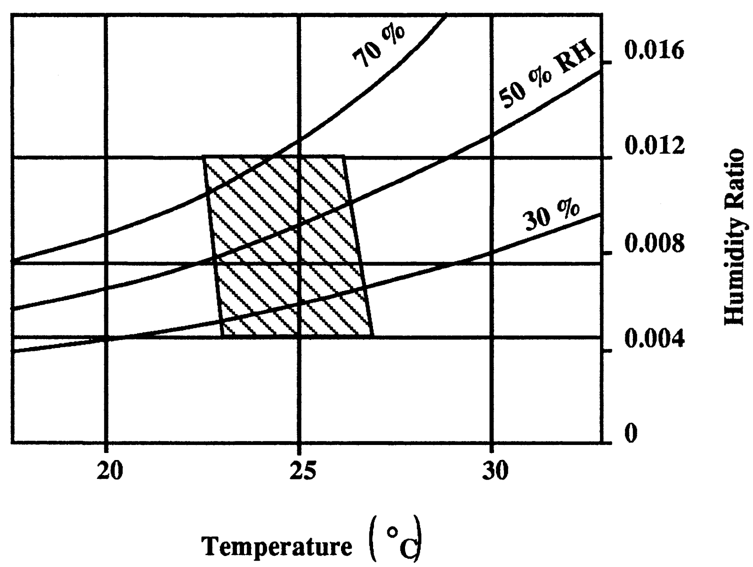


Figure 3.46 Summer time comfort region for people dressed in typical summer clothing.

they fall at the boundary of that region in the fifth hour. Therefore, energy and dollar savings are coupled with discomfort. To avoid these discomfort periods, it is necessary to start cooling the space at least four hours before it is occupied. This strategy will increase the energy and dollar consumption, however, the cost in this case is still lower compared to the continuous operation strategy cost. Moreover, the early cooling peak loads will occur during the time when the electricity rates are low.

Figures (3.47), (3.48) and (3.49) show the sensible, latent and total loads distribution for both strategies, respectively. For the partial operation strategy and in the early cooling hours, the total load peaks due to moisture storage in the room during the night, whereas the load for the continuous operation strategy is almost uniform at all times.

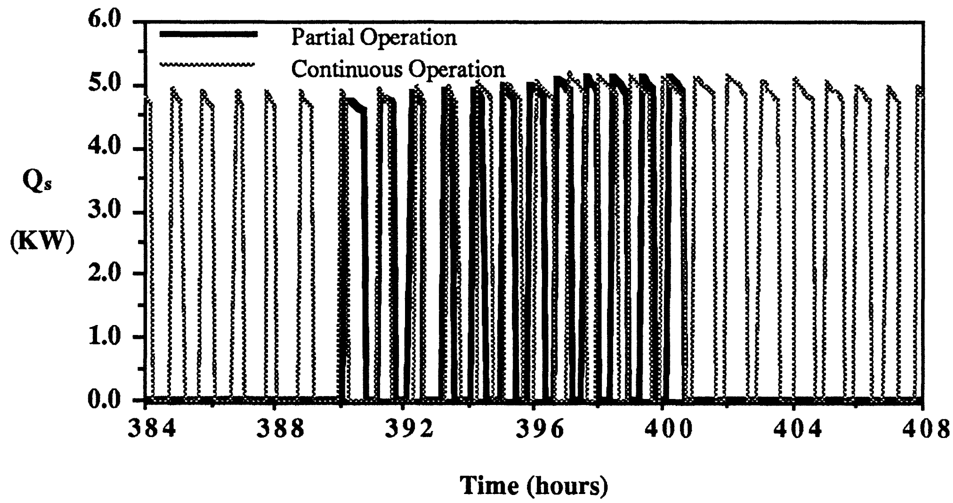


Figure 3.47 Sensible load distribution for partial vs. continuous operations.

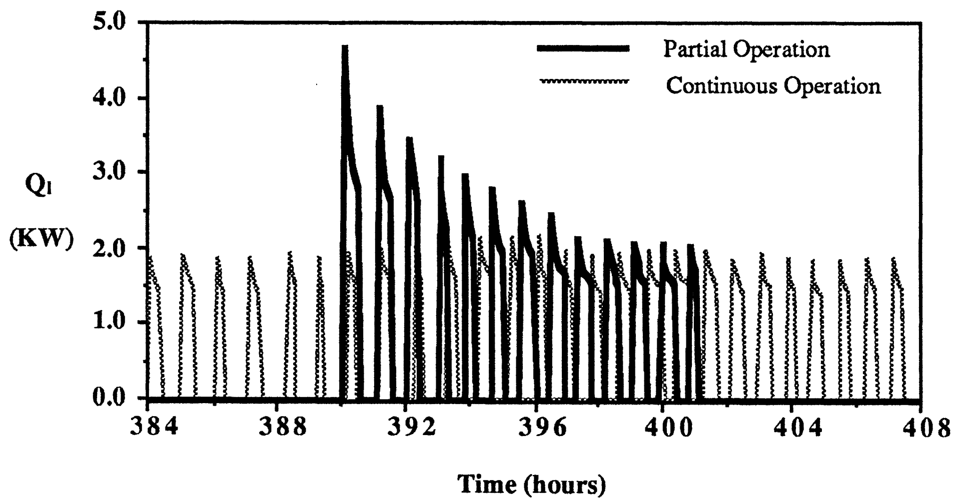


Figure 3.48 Latent load distribution for partial vs. continuous operations.

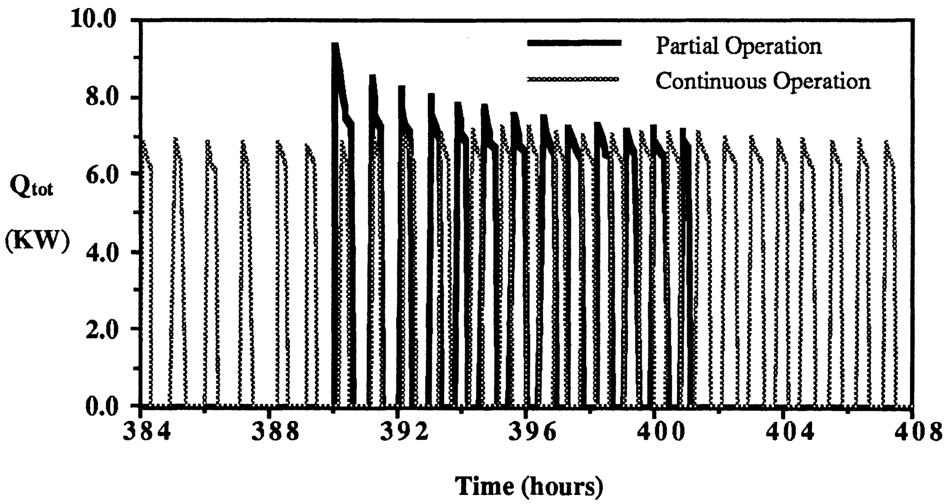


Figure 3.49 Total load distribution for partial vs. continuous operations.

Tables (3.9) and (3.10) present the cooling loads for both strategies and the percent differences based on the partial operation strategy. Keeping the cooling system in

Table 3.9 Cooling loads for partial and continuous operation strategies.

Strategy	Qs (kw)	Ql (kw)	Qtot (kw)
Partial operation	296.78	139.52	436.30
Continuous Operation	532.52	180.80	713.32

Table 3.10 Percent differences in the loads based on the partial operation strategy loads.

Strategy	% Diff(sen)	% Diff(lat)	% Diff(tot)
Partial operation	-	-	-
Continuous Operation	79.43	29.58	63.50

continuous operation will result in more than a 100 % increase in the system energy consumption in this case.

Figure (3.50) shows the moisture stored in the room furnishings and structures as a function of time for both strategies. In the partial operation, the moisture is adsorbed

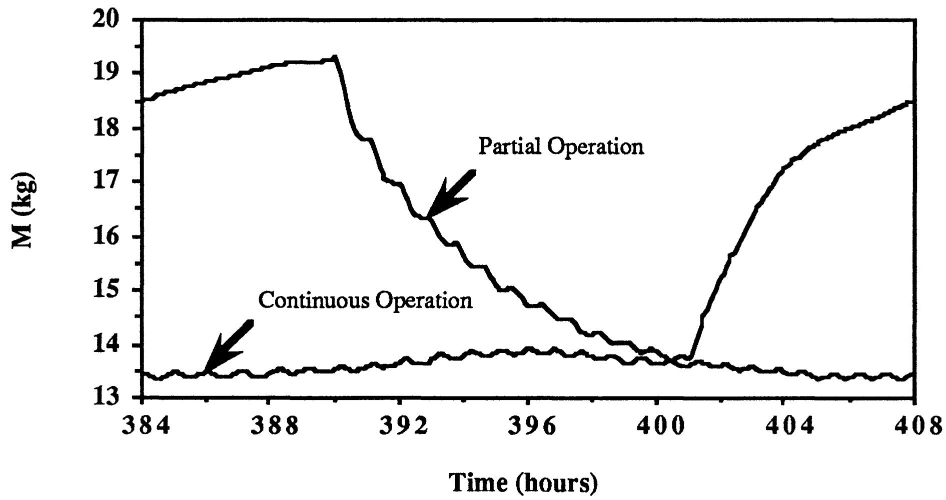


Figure 3.50 The moisture stored for partial vs. continuous operations.

during the evening and night and released during the day. The total amount of moisture released to the room air during the day is between 4 and 5 kg. This is approximately 5 times the amount of moisture that exists in the room air at any time.

In the continuous operation strategy, the amount of moisture stored is almost constant; hence, the amount of moisture exchanged with the room air is very small and does not significantly affect the indoor conditions nor the associated cooling loads.

Chapter IV

Conclusions and Recommendations

In the previous chapter, the Modified Effective Penetration Depth (MEPD) model was tested and its efficiency in modeling moisture storage in buildings was established. After the task of developing and testing this model was fulfilled, a set of conclusions and recommendations can be drawn to set up a track for further future model developments.

IV.1 CONCLUSIONS

Taking moisture storage into account can reduce the errors in the calculated indoor conditions and the associated latent load; however, more simulations are needed to determine whether the effect of moisture storage on the total load changes with changing the room and system parameters.

The effective air mass multiplier model might be efficient in some cases if the factor

" F " by which the air mass is increased were chosen properly; however, this could be a tough task since there is no basis for calculating this factor. Eventhough results from this model might match results from the MEPD model, differences in the load time distribution and peak loads are unavoidable.

Predicting the load time distribution is important for designing an optimal control strategy for a cooling system. For example, one of the objectives of the system optimization is to avoid the peak loads or force them to occur when the electricity rates are cheaper. Here moisture storage plays an important role since it affects the time distribution of the loads and their peak values

Energy and dollar savings are often associated with discomfort. Due to moisture storage, and in early cooling hours, periods of occupant discomfort are more likely to occur. These periods can be eliminated by an early start of cooling which will increase the system energy cost and consumption, and therefore, shift the early hours peak loads to the time when electricity rates are low.

A disadvantage of the partial cooling strategy is that the moisture adsorption and desorption can cause swelling and shrinkage of the building materials, respectively. Therefore, cyclic moisture adsorption and desorption could cause early structural damage, which reduces the lifetime of the material. Therefore, new optimal control strategies must take into account earlier start of cooling and minimize moisture adsorption and desorption to prevent material structural damage.

IV.2 RECOMMENDATIONS

The Modified Effective Penetration Depth model was theoretically proven to calculate the indoor conditions more accurately; however, this was done via computer simulations. In order to validate this model experimentally, TRNSYS results must be compared to real experimental data (room conditions and cooling loads); furthermore, case studies with bigger systems and different climates are recommended to determine if the total load percent difference is a function of system and space size and outdoor conditions.

This model was developed under the assumption that the forcing functions are of sinusoidal nature with constant amplitudes. This assumption can be avoided if a variable penetration depth were used. This can be done by using a moisture diffusivity that is a function of temperature to calculate the penetration depth at each time step.

Appendix A

TRNSYS Decks

This appendix contains the decks used in TRNSYS simulations before and after TYPE 19 modification.

The following deck represents a simple model for space cooling.

```
Nolist
*****************************************************************************************
* This deck simulates an air conditioned room taking moisture storage into account.  The cooling system is
represented by a ccoling coil operating from 6 a.m. to 5 p.m. after which ambient air is admitted into the room for
energy saving purposes. This deck was developed before TYPE19 was modified.
*****************************************************************************************
Sim 0. 408. .08333
Width 132
Tolerances -.01 -.01
Limits 25 4 25
*
UNIT 9 TYPE 9 Card Reader
*****************************************************************************************
* Parameters:
* 1- number of parameters to be read.
* 2- the time interval at which data is provided.
```

```
* 3- first parameter
* 4- factor by which first parameter is multiplied.
* 5- amount added to first parameter.
* 6- second parameter.
* 7- factor by which second parameter is multiplied.
* 8- amount added to second parameter.
* 9- third parameter.
* 10- factor by which third parameter is mutiplied.
* 11- amount added to third parameter.
* 12- fourth parameter.
* 13- factor by which fourth parameter is multiplied.
* 14- amount added to fourth parameter.
* 15- logical unit number of input data.
* 16- formatted reading (>0), free foramtted reading (<=0).
*************************************************************************************
Par 16
5 1 1 1 0 2 1 0 3 .1 0 4 .0001 0  20 1
(9X,F4.0,1X,F4.0,1X,F4.0,1X,F6.0,1X,F2.0)
*
UNIT 16 TYPE 16 Radiation Processor
*************************************************************************************
* parameters:
* 1- radiation mode.
* 2- tracking mode (fixed surface).
* 3- day of year of the start of the simulation.
* 4- latitude (degrees).
* 6- solar constant.
* 7- treat simulation time as solar time (if <0).
*************************************************************************************
*Par 7
2 1 181 25 78  4871 0  -1
*************************************************************************************
* inputs:*
* 1- radiation on horizontal surface.
* 2- time of last radiation reading.
* 3- time of next radiation data reading.
* 4- ground reflectance.
```

* 5- slope of surface or tracking axes (degrees)

* 6- azimuth angle (gamma) of surface or tracking axes.

* 7-12 azimuth angle of therest of the surfaces.*

****

Input 12

9,2 9,19 9,20 0,0 0,0 0,0 0,0 0,0 0,0 0,0 0,0 0,0

0. 0. 0. 0.3 90. 0. 90. -90. 90. 180. 90. 90.

*

*UNIT 14 TYPE 14 Forcing Function

** Outside air flow into the coil time function

****

* parameters:

* 1- time.

* 2- flow

* 3-12 time/flow.

****

Par 12

0. 1000. 6. 1000. 6. 100. 17. 100. 17. 1000. 24. 1000.

*

UNIT 15 TYPE 14 FORCING FUNCTION

* Room air flowing into the coil time function

****

* parameters: same sequence as unit above

Par 12

0. 0. 6. 0. 6. 900. 17. 900. 17. 0. 24. 0.

*

UNIT 17 TYPE 14 FORCING FUNCTION

* Water flowing into the coil time function

****

* parameters: same sequence as above unit

Par 12

0. 0. 6. 0. 6. 1900. 17. 1900. 17 0. 24 0.

*

UNIT 8 TYPE 8 ROOM THERMOSTAT

****

* parameters:

* 1- number of oscillations in a time step before outputs stick.

* 2- diable first stage heating.

```
* 3- minimum heat source temperature for source utilization.
* 4- room temperature above which room to be cooled.
* 5- room temperature below which 1st stage heating is commanded.
* 6- room temperature below which second stage heating is commanded.
* 7- dead band temperature difference.
*****************************************************
Par 7
7.  0. 10. 26. 15. 10. 2.
*****************************************************
* inputs:
* 1- room temperature.
* 2- first stage heat source temperature.
*****************************************************
Input 2
19,1   9,3
18.   20.
*

* ambient air flow into coil
AAF=[8,3]*[14,1]
*
* room air flow into the coil
RAF=[8,3]*[15,1]
*
* water flow into the coil
WF=[8,3]*[17,1]
*
UNIT 11 TYPE 11 Mixing Valve
*
Par 1
* mode
  6
************************************************************************************
* inputs:
* 1- inlet '1' temperature.
* 2- inlet '1' humidity ratio.
* 3- inlet '1' mass flow rate.
```

```
* 4- inlet '2' temperature.
* 5- inlet '2' humidity ratio.
* 6- inlet '2' mass flow rate.
****************************************************************************************
Input 6
9,3 9,4   AAF  19,1 19,2  RAF
18. 0.0075 0.   10.  .008  0.
*
UNIT 50  TYPE 52 Cooling Coil
****************************************************************************************
* parameters:
* 1- mode (simple/detailed).
* 2- units (SI/English).
* 3- number of heat exchanger rows (passes >4).
* 4- number of parallel tubes in each row of tubes.
* 5- tube lenght (duct height).
* 6- duct width.
* 7- outside tube diameter.
* 8- inside tube diameter.
* 9- tube material thermal conductivity.
* 10- fin thickness.
* 11- fin spacing.
* 12- number of fins.
* 13- thermal conductivity of fin material.
* 14- fin mode (rectangular/annular).
* 15- distance between centers of tubes in a row.
* 16- distance between centerlines of tube rows.
****************************************************************************************
Par 16
2 1 6 15 1.2 .4 .02 .019 853.2
.000332 .00138  700  853.2  1 .025 .0508
****************************************************************************************
* inputs:
* 1- inlet air dry bulb temperature.
* 2- inlet air humidity ratio.
* 3- air mass flow rate.
* 4- water inlet temperature.
```

```
* 5- water mass flow rate.
*********************************************************************************************
Input 5
11,1  11,2 11,3  0,0  WF
0.  0.  0. 4. 0.
*
UNIT 19 TYPE 19 New Room
* Zone
*********************************************************************************************
* parameters:
*  1- mode (energy rate/temperature level control).
*  2- units (SI/English).
*  3- room air volume.
*  4- constant air change per hour.
*  5- proportionality constant for air change due to indoor-outdoor temperature difference.
*  6- proportionality constant for ai change due to wind effect.
*  7- capacitance of room air and furnishings.
*  8-  Factor by which the air mass is magnified.
*  9- number of surfaces.
* 10- initial room air temperature.
* 11- initial room air humidity ratio.
*********************************************************************************************
Par 11
2 1 105. .1 0.017 0.049 1800. 1. 7 25. .0075 5
*********************************************************************************************
Input 11
Ta Wa Tv  mv  Wv   WI  Npepl Iact QIR  Qint  W
********************************************************
9,3 9,4 50,1 50,3 50,2  0,0  0,0   0,0  0,0   0,0   9,5
0.  0.  5.   648. .0075 0.05 2.   4.   5000. 1000  0.
* Walls
Par 16
1 1 14. .7 .4 2 81 2 -1 15. 3 -1 21. 4 -1 21.
*
Input 4
16,6  16,11  16,14  16,17
0. 0. 0. 0.
*
```

```
* Floor and Ceiling
Par 14
5 2 35 .7 .4 3 37
6 2 35 .7 .4 3 37
*
* Window
PAR 9
7 5 1 1 .8 30. 2 3 6
*
Input 6
16,6 16,7 0,0 0,0 0,0 0,0
0. 0. .8 12.3 .7 .3
*
* View Factors
Par 17
1 3  5. 7. 1 4 3 2 6 5 1 7 1 1. .5 1. 1.
*
UNIT 28  TYPE 28 Simulation Summary
*
Par 15
dt    ts tf   output
*******************************************************************************************
.0833 0. 240. 0  2  0 -4 0 -4 0 -4 0 -4  0 -4
*
Input 5
19,1 19,2   50,7 50,6   19,4
*
End
```

*Modified TYPE 19 parameters:

```
UNIT 19 TYPE 19 New Room
* Zone
*******************************************************************************************
* parameters:
*  1- mode (energy rate/temperature level control).
```

* 2- units (SI/English).

* 3- room air volume.

* 4- constant air change per hour.

* 5- proportionality constant for air change due to indoor-outdoor temperature difference.

* 6- proportionality constant for ai change due to wind effect.

* 7- capacitance of room air and furnishings.

* 8- heat transfer coefficient.

* 9- number of surfaces.

* 10- initial room air temperature.

* 11- initial room air humidity ratio.

* 12- number of solid materials in the room.

* 13- number of nodes to use.

* 15- material thermal conductivity.

* 16- material specific heat.

* 17- material moisture diffusivity.

* 18- initial node moisture content.

* 19- material heat transfer area.

* 20- material thickness.

* 21- material density.

* 22-29 regression constants.

* 29-92 same as 13-29, but for different materials.

****

Par 60

2 1 105. .1 0.017 0.049 1800. 3. 7 25. .0075 3

*

* Fiber Wood (ceiling)

****

1 .17 2300. .005104 .073 126. .0508 610. .07606 9.480125 .07207 .55784 .003980 3.317516 .004303 .019256

*

* Carpeting (floor)

****

1 .05 800. .0005567 .04 35. .02 500. .15503 8.918695 .051232 .671865 .000403 .102459 .00099 6.084116

*

* Concrete (walls)

****

1 .6 835. .0007248 .0181 71. .0508 510. .015204 .33067 .040595 8.402223 .000334 2.599087 .00069 093965

Appendix B

Modified TYPE 19

This appendix contains *TRNSYS-TYPE 19* after it has been modified to account for moisture storage in the walls and furniture using the Modified Effective Penetration Depth or Finite Difference models.

```
---------------------------------------------------------------------------------------
      SUBROUTINE TYPE19 (TIME,XIN,OUT,DUM1,DUM2,PAR,INFO)
C--------------------------------------------------------------------------------------
C
C     This subroutine simulates a single zone building using heat transfer functions. It predicts the indoor
C     conditions and the associated heating and cooling loads. This subroutine has been recently modified
C     to account for moisture storage in the zone walls and furniture. It uses a one node with an effective
C     moisture penetration depth and material equilibrium moisture isotherms to calculate the amount of moisture
C     stored in a zone using the  vapor pressure as a driving force. This  subroutine can  model up to  25
C     different materials at the same time. The subroutine solves simultaneously for the room temperature
C     and humidity ratio. This subroutine will solve for moisture stored using the Finite Difference model if the
C     number of nodes specified is greater than " one ".
---------------------------------------------------------------------------------------
C     Variables definitions:
C--------------------------------------------------------------------------------------
C
C     A1= regression constant.
C     A2= regression constant.
C     ALPHA= absorptivity.
C     B1= regression constant.
C     B2= regression constant.
C     C1= regression constant.
C     C2= regression constant.
```

C D1= regression constant.
C D2= regression constant.
C DENS= air density.
C DENSMAT= density of the solid material.
C DIFUSE= solid material moisture diffusivity.
C DMT= nodal rate of moisture mass change.
C DMTOT= total rate of moisture mass change.
C DXM= grid size for material moisture content finite differencing.
C DXT= grid size for material temperature finite differencing.
C FE= relative humidity.
C FLWI= infiltration mass flow rate.
C FLWV= ventilation mass flow rate.
C HTA= heat transfer area of the solid material.
C HTC= convective heat transfer coefficient.
C HVAP= latent heat of vaporization.
C IPEPL= activity level of the people in the zone.
C K1= constant air change per hour.
C K2= proportionality constant for air change due indoor-outdoor temperature difference.
C K3= Proportionality constant for air change due to wind effects.
C KEY= parameter for testing room temperature and humidity ratio convergence.
C M1= variable used to test for loop convergence.
C MAIR= mass of zone air.
C MFUR= mass of furniture, walls, floor and ceiling.
C MW= nodal moisture mass at the current time step.
C MW0= nodal moiature mass at previous time step.
C MWTOTAL= total moisture mass stored in the zone.
C NMAT= number of solid materials in the simulation.
C NODE= number of nodes for finite differencing.
C NPEPL= number of people in the zone.
C NS= number of surfaces in the zone.
C PERM= material moisture permeability.
C PVS= summation of the ratio of the vapor pressure to the moisture convective+conductive
C resistance.
C PV= vapor pressure.
C PVR= room air vapor pressure.
C QABS= heat of vaporization or condensation.
C QINT= internal gains from other than people or lights.
C QLAT= latent load.
C QLIGHT= gains from lights.
C QLINF= latent gains due to infiltration.
C QLVENT= latent gains due to ventilation.
C QPLAT= latent gains due to convection from people.
C QPSEN= sensible gains due to convection from people.
C QSINFL= sensible gains due to infiltration.
C QSNET= net sensible load.
C QSVENT= sensible gains due to ventilation.
C QZONE= gains from neighbouring zones.
C RA= moisture conductive resistance.
C RC= moisture convective resistance.
C REST= summation of the moisture convective + diffusion resistances of all the solid materials

```
C       involved in the simulation.
C      RV= ideal gas constant for air.
C      T0= initial room air temperature.
C      T0MAT= material nodal temperature at the previous time step.
C      TAMB= anbient temperature.
C      TMAT= material nodal temperature at the current time step.
C      TMAX= maximum allowable zone temperature ( set point for cooling ).
C      TMIN= minimum allowable zone temperature ( set point for heating ).
C      TNODE= total number of nodes in all the materials combined.
C      TR= room temperature.
C      TVENT= ventilation air temperature.
C      U= nodal moisture content of the solid material.
C      U0= initial nodal moisture content of the solid material.
C      UC= maximum possible nodal moisture content of the solid material.
C      W= rate of internal moisture gains from other than people.
C      WAMB= ambient air humidity ratio.
C      WR0= initial room air humidity ratio.
C      WR= room air mumidity ratio at the end of the current time step.
C      WRO= room air humidity ratio of the previous time step.
C      WVENT= ventilation air humidity ratio.
C________________________________________________________________________________

      LOGICAL STIME,SHIFT
      REAL*8 G,GLAST
      REAL MAIR,K1,K2,K3,M1(100,25)
      REAL MW0(100,25),MW(100,25),DMT(25),MFUR(25)
      REAL MWTOTAL
      INTEGER WMODE

      DIMENSION XIN(50),OUT(20),PAR(201),INFO(10),T(15),HCI(15)
      DIMENSION RHO(15),AREA(15),X(15),NCF(15,3),LIST(15),TEQ(15)
      DIMENSION ALPHA(15),DENSMAT(25),A1(25),B1(25),C1(25),D1(25)
      DIMENSION A2(25),B2(25),C2(25),D2(25),U(100,25),PV(100,25)
      DIMENSION B(10,15),C(10,15),D(10,15),IS(15),IPAR(15),INP(15)
      DIMENSION FV(15,15),FH(15,15),HR(15,15),Z(16,16),SUMFV(15)
      DIMENSION SOL(15),TSOL(15),SIGMA(2),TCONV(2),CP(2),ECONV(2)
      DIMENSION WCONV(2),HCON(2),HVAP(2),DENS(2),RV(2),HTA(25)
      DIMENSION HCOEF(2),QPSEN(11),QPLAT(11),LOUT(10),IOUT(10)
      DIMENSION DXT(25),DXM(25),RES(25),RA(100,25),NODE(100),U0(25)
      DIMENSION PERM(100),TMAT(0:100,25),CPT(25),TC(25),ERX(100),MW2(100,25)
      DIMENSION T0MAT(100,25),UC(25),T1MAT(100,25),DIFUSE(25),ERX1(25)

      COMMON /SIM/ TIME0,TIMEF,DELT
      COMMON /STORE/ NSTORE,IAV,S(5000)
      COMMON /LUNITS/ LUR,LUW,IFORM

      DATA IUNIT/0/,SIGMA/2.0411E-07,1.7122E-09/,TCONV/273.15,459.67/
      DATA LIST/1,2,3,4,5,6,7,8,9,10,11,12,13,14,15/
      DATA ECONV/3.6,3.41/,CP/1.012,.24/,WCONV/1.,0.447/
      DATA HCOEF/9.58,0.47/,IPAR/15*0/
```

```
      DATA  HCON/3.6,0.1761/,HVAP/2468.,1061./,DENS/1.204,0.075/
      DATA  RV/461.52,85.76/
      DATA  QPSEN/60.,65.,75.,75.,90.,100.,100.,100.,120.,165.,185./
      DATA  QPLAT/40.,55.,95.,75.,95.,130.,205.,180.,255.,300.,340./
      DATA NSFTM/8/,DSMIN/.125/,NSMAX/15/
C
   ITER=-1

   IF (INFO(7) .EQ. 0) THEN

     OUT(10+NOUTS+1) = AMAX1(OUT(10+NOUTS+1),OUT(9))
     OUT(10+NOUTS+2) = AMIN1(OUT(10+NOUTS+2),OUT(10))

   END IF

   IF(IUNIT.EQ.INFO(1)) GO TO 120
   IUNIT=INFO(1)

C Input the main parameters.

10  MODE=INT(PAR(1)+0.1)
    IU=INT(PAR(2)+0.1)
    VOL= PAR(3)
    K1= PAR(4)
    K2= PAR(5)
    K3= PAR(6)
    CAP= PAR(7)
    HTC0= PAR(8)
    NS= INT(PAR(9)+0.1)
    T0= PAR(10)
    WR0= PAR(11)
    NMAT= PAR(12)
    LP= 1
    TNODE=0.

C Input the parameters for each solid material involved in the simulation.

    DO 5 ISO= 1,NMAT

     NODE(ISO)= PAR(12+LP)
     TC(ISO)= PAR(13+LP)
     CPT(ISO)=PAR(14+LP)
     DIFUSE(ISO)= PAR(15+LP)
     U0(ISO) = PAR(16+LP)
     HTA(ISO)= PAR(17+LP)
     DXT(ISO)= PAR(18+LP)
     DENSMAT(ISO)= PAR(19+LP)
     A1(ISO)= PAR(20+LP)
     B1(ISO)= PAR(21+LP)
     C1(ISO)= PAR(22+LP)
```

```
      D1(ISO)= PAR(23+LP)
      A2(ISO)= PAR(24+LP)
      B2(ISO)= PAR(25+LP)
      C2(ISO)= PAR(26+LP)
      D2(ISO)= PAR(27+LP)

      LP= LP+16

C Calculating the total number of nodes for all the solid materials combined.
C
      TNODE= TNODE + NODE(ISO)

5     CONTINUE

      NPAR= 16*NMAT + 12
      IN=11
      NTC=0
      IF(MODE.EQ.2) GO TO 12
      TMIN=PAR(NPAR+1)
      TMAX=PAR(NPAR+2)
      WMIN=PAR(NPAR+3)
      WMAX=PAR(NPAR+4)
      NPAR= NPAR+4
12    IP=NPAR
C
C  Wall and window parameters.
C
      ATOT=0.

      DO 50 K=1,NS

        I=INT(PAR(IP+1)+0.1)
        ITYPE=INT(PAR(IP+2)+0.1)
        AREA(I)=PAR(IP+3)
        ATOT=ATOT+AREA(I)

        IF(INFO(7).GT.-1) GO TO 13

C  Error checks.

        IF(I.LT.1 .OR. I.GT.NSMAX) GO TO 600
        IF(ITYPE.GT.5) GO TO 700
C
13      IPAR(I)=IP
        INP(I)=IN
        IEQ=0

        IF(ITYPE.GT.0) GO TO 17

C  Use parameter list from previously defined surface.
```

```
      IEQ=INT(ABS(PAR(IP+2))+0.1)

      IF(INFO(7).GT.-1) GO TO 16

C  Error checks.

      IF(IEQ.LT.1 .OR. IEQ.GT.NSMAX) GO TO 600
      IF(IPAR(IEQ).NE.0) GO TO 15

      I=IEQ
      GO TO 1200

15    IF(IS(IEQ).LT.1 .OR. IS(IEQ).GT.5) GO TO 700
C
16    IP=IPAR(IEQ)
      ITYPE=IS(IEQ)
      NPAR=NPAR+3
C
17    IS(I)=ITYPE
      GO TO (20,20,20,30,40) ,ITYPE
C
C  ASHRAE wall or roof.
C
20    RHO(I)=PAR(IP+4)
      ALPHA(I)=PAR(IP+5)
      ICOEF=INT(PAR(IP+6)+0.1)

      IF(ICOEF.EQ.4) GO TO 21

      NTABLE=INT(PAR(IP+7)+0.1)

C Calling table of ASHRAE wall, roof and floor coefficients.

      CALL TABLE(ICOEF,NTABLE,IU,NB,NC,ND,B(1,I),C(1,I),D(2,I))

      HCI(I)=HCOEF(IU)
      GO TO 22
21    HCI(I)=PAR(IP+7)
      NB=INT(PAR(IP+8)+0.1)
      NC=INT(PAR(IP+9)+0.1)
      ND=INT(PAR(IP+10)+0.1)
22    ND=ND+1
      NCOEF=NB+NC+ND
      NTC=NTC+NCOEF
      NMAX=MAX0(NB,NC,ND-1)
      D(1,I)=1.

      DO 25 J=1,NMAX
```

```
      IF(ICOEF.LT.4) GO TO 23
      IF(J.LE.NB) B(J,I)=PAR(IP+10+J)
      IF(J.LE.NC) C(J,I)=PAR(IP+10+NB+J)
      IF(J.LE.ND-1) D(J+1,I)=PAR(IP+10+NB+NC+J)
C
23      IF(J.GT.NB) B(J,I)=0.
      IF(J.GT.NC) C(J,I)=0.
      IF(J.GT.ND) D(J,I)=0.
C
       B(J,I)=B(J,I)/HCI(I)
       C(J,I)=C(J,I)/HCI(I)-D(J,I)

      IF(ITYPE.NE.2) GO TO 25

       C(J,I)=C(J,I)-B(J,I)
       B(J,I)=0.

25    CONTINUE

     NP=7

      IF(ICOEF.EQ.4) NP=9+NCOEF
      IF(ITYPE.EQ.1) IN=IN+1
      IF(ITYPE.EQ.3) IN=IN+3

     NCF(I,1)=NB
     NCF(I,2)=NC
     NCF(I,3)=ND

     GO TO 45
C
C  Exterior surface with conduction as input.

30    RHO(I)=PAR(IP+4)
     HCI(I)=PAR(IP+5)
      C(1,I)=-1.
     NP=5
     IN=IN+1

     GO TO 45
C
C WINDOW

40    WMODE=INT(PAR(IP+4)+0.1)
     RHO(I)=1.-PAR(IP+5)
     HCI(I)=PAR(IP+6)
      C(1,I)=-1.
     NI=INT(PAR(IP+7)+0.1)
     NP=7+NI
```

```
      IN=IN+4+NI

      IF(WMODE.EQ.2) IN=IN-1
C
45    IF(IEQ.GT.0) GO TO 50

      NPAR=NPAR+NP
50    IP=NPAR

      IGEOM=INT(PAR(IP+1)+0.1)
      IP=IP+1

      IF(INFO(7).EQ.-1) GO TO 55
C
C Look for optional outputs.

      NCK=IP

      IF(IGEOM.EQ.0) GO TO 53

      GO TO (51,52) ,IGEOM

51    NCK=NCK+10+INT(PAR(IP+10)+0.1)*6

      GO TO 53

52    NCK=NCK+NS*(NS-1)/2
C
53    NOUTS=0

      IF(INFO(4).EQ.NCK) GO TO 120
C
C Optional output parameters in sets of '3'.

      NOUTS=INT(PAR(NCK+1)+0.1)
      INFO(6) = 10 + NOUTS + 2

      DO 54 L=1,NOUTS

      IL=NCK+(L-1)*2+1
      LOUT(L)=INT(PAR(IL+1)+0.1)
      IOUT(L)=INT(PAR(IL+2)+0.1)

54    CONTINUE

      GO TO 120
C
55    CONTINUE

C Initial call of simulation.
```

```
C
C Check for complete set of surface numbers.

   DO 56 I=1,NS

    IF(IPAR(I).GT.0) GO TO 56

    GO TO 1200

56  CONTINUE

C Number of time steps per hour, limit temperature.
C Histories to NSFTM sets.

   NDELT=INT(1./DELT+0.1)
   NSHFT=MIN0(NDELT,NSFTM)
C
C Determining exchange factors and set up coefficient matrix at beginning of simulation.
C
   IF(IGEOM.NE.1) GO TO 57

C View factors for parallelpiped geometry.

   CALL ENCL(PAR(IP+1),PAR(IP+11),FV,INFO)
   IP=IP+10+INT(PAR(IP+10)+0.1)*6

   GO TO 60
C
C View factors entered by user if IGEOM = 1
C Or defined based upon area ratios if IGEOM = 0

57  J2=NS-1

   DO 59 I=1,J2

    J1=I+1
    FV(I,I)=0.

    IF(IGEOM.EQ.0) FV(I,I)=AREA(I)/ATOT

      DO 59 J=J1,NS

       IF(IGEOM.NE.0) GO TO 58

       FV(I,J)=AREA(J)/ATOT

      GO TO 59
58       IP=IP+1
       FV(I,J)=PAR(IP)
```

```
59   FV(J,I)=FV(I,J)*AREA(I)/AREA(J)

     FV(NS,NS)=0.

     IF(IGEOM.EQ.0) FV(NS,NS)=AREA(NS)/ATOT
C
60   DO 62 I=1,NS

      SUMFV(I)=0.

       DO 61 J=1,NS

        IF(I.EQ.J) GO TO 61

        SUMFV(I)=SUMFV(I)+FV(I,J)

61       Z(I,J)=-RHO(J)*FV(I,J)

62   Z(I,I)=1.+Z(I,I)
C
     NOUTS=0

     IF(INFO(4).EQ.IP) GO TO 65
C
C  Optional output parameters in sets of '3'.

     NOUTS=INT(PAR(IP+1)+0.1)

      IF(NOUTS.EQ.0) GO TO 65
      IF(NOUTS.GT.10) GO TO 800

     DO 64 L=1,NOUTS

      IL=IP+(L-1)*2+1
      LOUT(L)=INT(PAR(IL+1)+0.1)
      IOUT(L)=INT(PAR(IL+2)+0.1)

       IF(LOUT(L).LT.1 .OR. LOUT(L).GT.5) GO TO 1000
       IF(IOUT(L).LT.1 .OR. IOUT(L).GT.NS) GO TO 1100

64   CONTINUE
C
65   CALL INVERT(NSMAX+1,NS,Z,IFLAG)

      IF(IFLAG.EQ.1) GO TO 500
      IF(IGEOM.EQ.0) GO TO 68
C
C  Print table of view factors.

     WRITE(LUW,1300) INFO(1),INFO(2),(LIST(J),J=1,NS)
```

```
      DO 66 I=1,NS
        WRITE(LUW,1301) I,(FV(I,J),J=1,NS)
66    CONTINUE
C Check for incomplete enclosure.
      DO 67 I=1,NS
        IF(SUMFV(I).GT.0.98 .AND. SUMFV(I).LT.1.02) GO TO 67
        WRITE(LUW,1400) INFO(1),INFO(2),I
67    CONTINUE
C
C Net exchange factors, FHATS
68    DO 69 I=1,NS
        DO 69 J=1,NS
          FH(I,J)=0.
          DO 69 K=1,NS
69    FH(I,J)=FH(I,J)+Z(I,K)*FV(K,J)
C
C Walls, windows, etc.
      SUMHCI=0.
      HRMAX=4.*SIGMA(IU)*(T0+TCONV(IU))**3
      DO 75 I=1,NS
        DO 70 J=1,NS
70        HR(I,J)=FV(I,J)*HRMAX
        SUMHCI=SUMHCI+HCI(I)*AREA(I)
75    CONTINUE
      DO 80 I=1,NS
        DO 78 J=1,NS
78        Z(I,J)=C(1,I)*HR(I,J)/HCI(I)
```

```
      Z(I,I)=1.-C(1,I)*SUMFV(I)*HRMAX/HCI(I)

80    Z(I,NS+1)=C(1,I)
C  Room.

      DO 85 J=1,NS

85    Z(NS+1,J)=HCI(J)*AREA(J)

      Z(NS+1,NS+1)=-SUMHCI-2.*CAP/DELT
C
C Calling subroutine to invert matrix.

      CALL INVERT(NSMAX+1,NS+1,Z,IFLAG)

      IF(IFLAG.EQ.1) GO TO 500
C
C  TYPECK call to store information in S-array

      NSTOR=2*NS**2+(NS+1)**2+NTC*NSHFT+NS+4

      IF(MODE.EQ.1) NSTOR=NSTOR+(NS+1)**2
      IF(INFO(4).NE.IP) IP=IP+2*NOUTS+1
      INFO(6)=10+NOUTS

C Allocating spaces for variables to be stored in the S-array.

      INEED= TNODE*4 + 2*NMAT + 4
      INFO(10)= NSTOR + INEED

      CALL TYPECK(1,INFO,IN,IP,0.)

      ISTORE= INFO(10) - 1
      KSTORE= ISTORE + NSTOR

C Equating the room air temperature and humidity ratio to the initial values.

      WR=WR0
      TR=T0

C Initializing room humidity ratio, material nodal moisture content and temperature.
C Initializing the rate of total moisture mass change to "0".

      DMTOT=0.
      NN=1

C Do from material '1' to material 'N'.

      DO 86 LP= 1,NMAT
```

```
C If using one-node model, then use the Fourier number equation to determine the penetration depth.

    IF (NODE(LP).EQ.1) THEN
      DXM(LP)= SQRT (DIFUSE(LP)*24./10000.)

C  If the penetration depth is greater  than  the  material  full  thickness, then use the full thickness as the
C  penetration depth

      IF (DXM(LP).GT.DXT(LP)) DXM(LP)= DXT(LP)

    ELSE

      DXM(LP)= DXT(LP)/(NODE(LP)-1)

    END IF

C Initializing nodal temperatures, moisture contents and rate of moisture mass changein the solid material.

      DO 87 N=1,NODE(LP)

       T0MAT(N,LP)=TR

       DV=1.

C  If  using the finite difference model, then the size of the first and the last segments will be equal to half  the size
C  of any other segment.

         IF (N.EQ.1.OR.N.EQ.NODE(LP)) DV=2.

C If using the one-node model, then use the bulk mass of the whole lump.

         IF (NODE(LP).EQ.1) DV=1.

       DMT(LP)=0.
       MW0(N,LP)=U0(LP)*DXM(LP)
     .     *DENSMAT(LP)*HTA(LP)/DV

C Storing the initial values in the S-array.

       S(KSTORE+NN)= T0MAT(N,LP)
       S(KSTORE+NN+1)= MW0(N,LP)

       NN= NN+2

87     CONTINUE

C Storing the nodal rate of moisture mass change in teh S-array.

    S(KSTORE+NN)= DMT(LP)
```

```
      NN=NN+1

86    CONTINUE
C Storing the total rate of moisture mass change and the initial room humidity ratio in the S-array.

      S(KSTORE+NN)= DMTOT
      S(KSTORE+NN+1)= WR0

      WR=WR0
      S(ISTORE+3)=TR
      S(ISTORE+4)=WR

C  Initialize temperature history, separate histories for each time step if less than DSMIN HR

      IPTR=ISTORE+4
      TAMB=XIN(1)

      DO 90 I=1,NS

        IF(IS(I).GT.3) GO TO 90

       NB=NCF(I,1)*NSHFT
       NC=NCF(I,2)*NSHFT
       ND=NCF(I,3)*NSHFT
       NMAX=MAX0(NB,NC,ND)

         DO 88 J=1,NMAX

          IF(J.LE.NB) S(IPTR+J)=TAMB
          IF(J.LE.NC) S(IPTR+NB+J)=T0
          IF(J.LE.ND) S(IPTR+NB+NC+J)=T0

88       CONTINUE

       IPTR=IPTR+NB+NC+ND

90    CONTINUE

      IPTR1=ISTORE+NTC*NSHFT+4
      IPTR2=IPTR1+NS*NS
      IPTR3=IPTR2+NS*NS

C  Store exchange factors, radiative coefficients, and inverted Z-matrix.

      DO 95 I=1,NS

       DO 95 J=1,NS

        IPTR1=IPTR1+1
        IPTR2=IPTR2+1
```

```
      S(IPTR1)=FH(I,J)
95    S(IPTR2)=HR(I,J)
      NP1=NS+1
      DO 97 I=1,NP1
       DO 97 J=1,NP1
        IPTR3=IPTR3+1
97    S(IPTR3)=Z(I,J)
      IF(MODE.EQ.2) GO TO 355
C  Store inverted Z-materix for constant room temperature, mode '1'.
      DO 105 I=1,NS
        DO 100 J=1,NS
100       Z(I,J)=C(1,I)*HR(I,J)/HCI(I)
       Z(I,I)=1.-C(1,I)*SUMFV(I)*HRMAX/HCI(I)
105   Z(I,NS+1)=C(1,I)
      DO 110 J=1,NS
110   Z(NS+1,J)=0.
      Z(NS+1,NS+1)=1.
C  Calling subroutine to invert matrix..
      CALL INVERT(NSMAX+1,NS+1,Z,IFLAG)
      IF(IFLAG.EQ.1) GO TO 500
      IPTR4=IPTR3
      DO 115 I=1,NP1
       DO 115 J=1,NP1
        IPTR4=IPTR4+1
115   S(IPTR4)=Z(I,J)
      GO TO 355
C
C  Retrieve stored information after first internal iteration, only calculate quantities taht change.
```

```
C
120  IF(TIME.LT.TIME0+DELT/2.) GO TO 355

     ITER=ITER+1
     ISTORE=INFO(10)-1
     IPTR1=ISTORE+NTC*NSHFT+4
     IPTR2=IPTR1+NS*NS
     IPTR3=IPTR2+NS*NS
     IPTR4=IPTR3

      IF(MODE.EQ.1) IPTR4=IPTR3+(NS+1)**2

     IPTR5=IPTR4+(NS+1)**2
     KSTORE= ISTORE + NSTOR
     LSTORE= ISTORE + NSTOR + KSTORE

C  If the first call of the routine in the same time step, then read nodal temperatures, moisture contents, rate of
C  moisture mass changes and room humidity ratio from the S-array.

      IF (INFO(7).EQ.0) THEN

     NN=1

C  Do this procedure from material 'first' to material 'last'.

     DO 121 LP=1,NMAT

C  Read values from S-array for all nodes..


        DO 122 N=1,NODE(LP)

         S(LSTORE+NN) = S(KSTORE+NN)
         S(LSTORE+NN+1) = S(KSTORE+NN+1)
         NN=NN+2

122      CONTINUE

      S(LSTORE+NN) = S(KSTORE+NN)
      NN=NN+1

121  CONTINUE

     S(LSTORE+NN) = S(KSTORE+NN)
     S(LSTORE+NN+1) = S(KSTORE+NN+1)

     END IF

C  Retrieving values stored in the S-array.
```

```
      NN=1

      DO 123 LP=1,NMAT

       DO 124 N=1,NODE(LP)

        TMAT(N,LP) = S(LSTORE+NN)
        T0MAT(N,LP) = TMAT(N,LP)
        MW(N,LP) = S(LSTORE+NN+1)
        MW0(N,LP) = MW(N,LP)

        NN= NN+2

124   CONTINUE

      DMT(LP) = S(LSTORE+NN)
      NN=NN+1

123   CONTINUE

      DMTOT = S(LSTORE+NN)
      WRO = S(LSTORE+NN+1)

      IF(INFO(7).GT.0 .OR. ITER.GT.0) GO TO 125

      S(ISTORE+1)=S(ISTORE+3)
      S(ISTORE+2)=S(ISTORE+4)

125   TRI=S(ISTORE+1)
      WRLAST=S(ISTORE+2)
      ISTORE=ISTORE+4
      TR=TRI

      IF(INFO(7).GT.0 .OR. ITER.GT.0 .OR. CAP.LT.1.E-06) TR=OUT(1)

      NP1=NS+1

      DO 130 I=1,NP1

       DO 130 J=1,NP1

        IPTR3=IPTR3+1

130   Z(I,J)=S(IPTR3)

      IF(ITER.GT.0) GO TO 158

      DO 140 I=1,NS

       DO 140 J=1,NS
```

```
      IPTR1=IPTR1+1
      IPTR2=IPTR2+1
      FH(I,J)=S(IPTR1)

140   HR(I,J)=S(IPTR2)
C
C  Time varying inputs.

      DO 155 I=1,NS

       SOL(I)=0.

155   TSOL(I)=0.

C  Room.

      TAMB=XIN(1)
      WAMB=XIN(2)
      TVENT=XIN(3)
      FLWV=XIN(4)
      WVENT=XIN(5)
      WGEN=XIN(6)
      NPEPL=XIN(7)
      IPEPL=XIN(8)
      QLGHT=XIN(9)
      QINT=XIN(10)
      W=XIN(11)
C
      QSPEPL=0.
      QLPEPL=0.

      IF(IPEPL.LT.1 .OR. IPEPL.GT.11) GO TO 157

      QSPEPL=ECONV(IU)*NPEPL*QPSEN(IPEPL)
      QLPEPL=ECONV(IU)*NPEPL*QPLAT(IPEPL)
157   HCO=HCON(IU)*(5.7+3.8*W*WCONV(IU))

C  Calculating the room air mass.

      MAIR=VOL*DENS(IU)
      IDELT=INT(AMOD(TIME,1.)/DELT+0.1)
      IDELT=IDELT*NSHFT/NDELT+1
158   FLWI=MAIR*(K1+K2*ABS(TAMB-TR)+K3*W)
      QZONE=0.
C
      DO 200 I=1,NS

       ITYPE=IS(I)
       IP=IPAR(I)
```

```
      IN=INP(I)

      GO TO (160,161,162,170,180) ,ITYPE
C  ASHRAE wall.
C
C    Exterior.

C  If more than one iteration, then GO TO 200

160  IF(ITER.GT.0) GO TO 200

     TSA=TAMB+XIN(IN+1)*ALPHA(I)/HCO

     GO TO 164

C    Interior partition.

161  IF(ITER.GT.0) GO TO 200

     TSA=0.

     GO TO 164

C  Connected to another zone.

162  TSA=XIN(IN+1)
     TZ=XIN(IN+2)
     UAC=XIN(IN+3)
     QZONE=QZONE+UAC*(TZ-TR)

     IF(ITER.GT.0) GO TO 200

164  NB=NCF(I,1)
     NC=NCF(I,2)
     ND=NCF(I,3)
     NCOEF=NB+NC+ND
     NMAX=MAX0(NB,NC,ND)
     IPB=NB*(IDELT-1)
     IPC=NB*NSHFT+NC*(IDELT-1)
     IPD=(NB+NC)*NSHFT+ND*(IDELT-1)
     STIME = NDELT.LE.NSFTM .OR. AMOD(TIME,DSMIN).LT.DELT/2.
     SHIFT = INFO(7).EQ.0 .AND. ITER.EQ.0 .AND. STIME

     IF(.NOT.(SHIFT)) GO TO 167
C
C  Shift  previous  temperature history once each hour, separate histories for each time step if less than or equal to
C  DSMIN HR.

     DO 165 J=2,NMAX
```

```
      JB=IPB+NB-J+2
      JC=IPC+NC-J+2
      JD=IPD+ND-J+2

       IF(J.LE.NB) S(ISTORE+JB)=S(ISTORE+JB-1)
       IF(J.LE.NC) S(ISTORE+JC)=S(ISTORE+JC-1)
       IF(J.LE.ND) S(ISTORE+JD)=S(ISTORE+JD-1)

165  CONTINUE
C
167  SUMTB=TSA*B(1,I)
     SUMTC=0.
     SUMTD=0.

     DO 168 J=2,NMAX

      IF(J.LE.NB) SUMTB=SUMTB+B(J,I)*S(ISTORE+IPB+J)
      IF(J.LE.NC) SUMTC=SUMTC+C(J,I)*S(ISTORE+IPC+J)
      IF(J.LE.ND) SUMTD=SUMTD+D(J,I)*S(ISTORE+IPD+J)

168  CONTINUE

      IF(STIME) S(ISTORE+IPB+1)=TSA

     ISTORE=ISTORE+NCOEF*NSHFT
     X(I)=SUMTB-SUMTC-SUMTD
     GO TO 200
C
C  Conduction as input.

170  IF(ITER.GT.0) GO TO 200

     X(I)=XIN(IN+1)/AREA(I)/HCI(I)
     GO TO 200
C
C  Window.

180  WMODE=INT(PAR(IP+4)+0.1)

     GO TO (181,182) ,WMODE

181  GT=XIN(IN+1)
     GB=XIN(IN+2)
     GD=GT-GB
     TAU=XIN(IN+3)
     TAUD=PAR(IP+5)
     TAUB=0.

     IF(GB.GT.0.) TAUB=(GT*TAU-GD*TAUD)/GB
```

```
   QWSB=AREA(I)*GB*TAUB
   QWSD=AREA(I)*GD*TAUD
   UG=XIN(IN+4)

    IF(UG.GT.0.) UG=1./(1./HCI(I)+1./UG+1./HCO)

   TEQ(I)=S(IPTR5+I)
   QWT=UG*(TAMB-TEQ(I))

   GO TO 183

182  QWST=XIN(IN+1)
   QWSB=XIN(IN+2)
   QWSD=QWST-QWSB
   QWT=XIN(IN+3)/AREA(I)
183  X(I)=QWT/HCI(I)

    IF(ITER.EQ.0) GO TO 184

C  Modify for radiation input after the first internal iteration.

   X(I)=X(I)-C(1,I)*TSOL(I)/HCI(I)

   GO TO 200
C
184    IF(QWSB.LT.1.E-06) GO TO 190
C
C   Distribute beam radiation according to user input.

   NI=INT(PAR(IP+7)+0.1)
   IPR=IN+4

    IF(WMODE.EQ.2) IPR=IPR-1

   DO 185 J=1,NI

    LOC=INT(PAR(IP+7+J)+0.1)

185  SOL(LOC)=SOL(LOC)+XIN(IPR+J)*QWSB/AREA(LOC)

C   Distribute diffuse input according to net exchange factors
C   Windos are transparent.

190    IF(QWSD.LT.1.E-06) GO TO 200

   DO 195 J=1,NS

    IF(IS(J).EQ.5) GO TO 195
    TSOL(J)=TSOL(J)+FH(I,J)*(1.-RHO(J))*QWSD/AREA(J)
```

```
195  CONTINUE
C
200  CONTINUE

      IF(ITER.GT.0) GO TO 225
C
C  Total absorbed radiation.
C  Lights and people distributed with an even flux.
C  Windows transparent to solar and lights, opaque to Infrared.

     QEX1=0.7*QSPEPL/ATOT
     QEX2=QLGHT/ATOT

     DO 210 I=1,NS

      TSOL(I)=TSOL(I)+QEX1

       IF(IS(I).EQ.5) GO TO 210

      TSOL(I)=TSOL(I)+(1.-RHO(I))*SOL(I)+QEX2

       DO 205 J=1,NS

205       TSOL(I)=TSOL(I)+RHO(J)*SOL(J)*FH(J,I)
     .  *(1.-RHO(I))*AREA(J)/AREA(I)

210  CONTINUE
C
C  Modify X-vector for radiation input.

     DO 220 I=1,NS

      X(I)=X(I)-C(1,I)*TSOL(I)/HCI(I)

220  CONTINUE

C  Computing the room temperature.
C
C  Room.

225    X(NS+1)=-QZONE-2.*CAP/DELT*TRI-FLWV*CP(IU)*(TVENT-TR)
     .  - FLWI*CP(IU)*(TAMB-TR)-QINT-0.3*QSPEPL

C
C  Compute new room temperature.

     TR=0.

     DO 230 J=1,NP1
```

```
230  TR=TR+Z(NS+1,J)*X(J)

     T(NP1)=TR
     TRF=2.*TR-TRI
     QSENS=0.
     ISEN=0

     IF(MODE.EQ.1 .AND. (TR.LT.TMIN.OR.TR.GT.TMAX)) GO TO 250
C
C  Compute surface temperatures.

     DO 240 I=1,NS

      T(I)=0.

      DO 240 J=1,NP1

240  T(I)=T(I)+Z(I,J)*X(J)

     GO TO 270
C
C  Determine auxiliary in mode '1'.

C
C   Retrieve constant room temperature matrix.

250  DO 255 I=1,NP1

      DO 255 J=1,NP1

      IPTR4=IPTR4+1

255  Z(I,J)=S(IPTR4)
C
     X(NP1)=TMIN

     IF(TR.GT.TMAX) X(NP1)=TMAX

     DO 260 I=1,NP1

      T(I)=0.

      DO 260 J=1,NP1

260  T(I)=T(I)+Z(I,J)*X(J)

     TR=T(NP1)
     TRF=TR
     ISEN=1
C
```

```
270  QCONV=0.

     DO 275 I=1,NS

      SUM=TSOL(I)

        DO 273 J=1,NS
273       SUM=SUM+HR(I,J)*(T(J)-T(I))

      TEQ(I)=TR+SUM/HCI(I)
      S(IPTR5+I)=TEQ(I)

275  QCONV=QCONV+HCI(I)*AREA(I)*(T(I)-TR)

     QSVENT=FLWV*CP(IU)*(TVENT-TR)
     QSINFL=FLWI*CP(IU)*(TAMB-TR)
     DELU=CAP*(TRF-TRI)

     IF(ISEN.EQ.1) QSENS= QZONE+QCONV+QSVENT+QSINFL+0.3*QSPEPL
    ;              +QINT-DELU/DELT
C
C  Iterate internally if poor energy balance.

     QSNET=(QZONE+QCONV+QSINFL+QSVENT
    .     +0.3*QSPEPL+QINT)*DELT
     QBAL=QSNET-DELU-QSENS*DELT
     QCK=ABS(QSNET)+ABS(DELU)+ABS(QSENS)*DELT

     IF(((ABS(QBAL).LT.0.02*QCK).OR.(ABS(TR-OUT(1)).LT.1.0E-3))
    .   .AND.(ABS(TR- OUT(1)).LT.0.01).OR.(ITER.EQ.5)) GO TO 280
C
C  Use secant method.

     G=TR-OUT(1)

     IF(ITER.GT.0) GO TO 278

     TRLAST=OUT(1)
     GLAST=G
     OUT(1)=TR

     GO TO 120

278  DG=(G-GLAST)/(OUT(1)-TRLAST)

     TRLAST=OUT(1)
     GLAST=G
     OUT(1)=TRLAST-G/DG

     IF (ABS(OUT(1)-TRLAST) .LT. 1.0E-3) OUT(1) = TR
```

```
      GO TO 120
C
280   ISTORE=INFO(10)-1
      S(ISTORE+3)=TRF

      IF(.NOT.(STIME)) GO TO 295
C
C Compute  equivalent  inside  temperatures for ASHRAE walls and store along with Sol-Air and surface
C   temperatures, once each time step if less than 1/8 hr.

      ISTORE=ISTORE+4

      DO 290 I=1,NS

        IF(IS(I).GT.3) GO TO 290

       NB=NCF(I,1)
       NC=NCF(I,2)
       ND=NCF(I,3)
       IPC=NB*NSHFT+NC*(IDELT-1)
       IPD=(NB+NC)*NSHFT+ND*(IDELT-1)
       NCOEF=NB+NC+ND
       S(ISTORE+IPC+1)=TEQ(I)
       S(ISTORE+IPD+1)=T(I)
       ISTORE=ISTORE+NCOEF*NSHFT

290   CONTINUE

          DO 292 LP=1,NMAT

          DO 293 N=1,NODE(LP)
          MW2(N,LP) = MW0(N,LP)
293       CONTINUE

292       CONTINUE

C Initializing the room air humidity ratio to the humidity ratio of the previous time step.

      WR=WRO
      KEY= 0
      ITERATION=0

C  Solve for nodal temperature, nodal moisture content and nodal rate of moisture mass
C  change until the node moisture mass and room humidity ratio converge.

      DO WHILE (KEY.LT.NMAT+1)

      WR1= WR
```

```
C
C  Solving for the nodal material temperatures using Gauss-Seidle iteration technique.

295    HTC= HTC0

C  Sepecifying the number of loops according to the number of materials.

   DO 297 LP=1,NMAT

C  Calculating the heat transfer resistance, Volume, heat capacity and the resistance at the first node which
 C  includes a convective term.

C  If using  finite difference  model, then  divide the  material into segments as specified and use the same
C  segment thickness in solving for the noda moisture content.

     IF (NODE(LP).NE.1) THEN

       DXT(LP)= DXT(LP)/( NODE(LP)-1)
       DXM(LP)= DXT(LP)

     END IF

    OMEGA= DENSMAT(LP)*CPT(LP)*DXT(LP)*(1/(HTC*3600)
  ;        + DXT(LP)/(2*TC(LP)*3600))

    OMEGA= DELT/OMEGA
    ETA= HVAP(IU)*1000*DELT/(DENSMAT(LP)*CPT(LP)*DXT(LP)/2)

    GAMA= 2*HTC*3600*DELT/(DENSMAT(LP)*CPT(LP)*DXT(LP))
    BETA= TC(LP)*3600*DELT/(DENSMAT(LP)*CPT(LP)*DXT(LP)**2)


C  If using the one-node model, then solve uisng the specified equation.

     IF (NODE(LP).EQ.1) THEN

       TMAT(1,LP)= (T0MAT(1,LP)+ TR*OMEGA
  ;          + DMT(LP)*ETA/2)/(OMEGA + 1)

C  Else, solve implicitly for the nodal temperatures using the specified equations.

     ELSE

C  Iterate until error <=.1%.

          DIFF=5.

     DO WHILE (DIFF.GE..1E-4)
```

```
          DIFF=0.

C  Solving for the nodal temperatures.

          DO 300 N=1,NODE(LP)

          T1MAT(N,LP)=TMAT(N,LP)

C  If solving for the first node, then use the following equation.

          IF (N.EQ.1) THEN

           TMAT(N,LP)= (T0MAT(N,LP)+2*BETA*TMAT(N+1,LP)
     .      +DMT(LP)*ETA+ GAMA*TR)/ (1+GAMA+2*BETA)

C  If solving for the last node, then use the following equation.

          ELSE IF (N.EQ.NODE(LP)) THEN

           TMAT(N,LP)= (T0MAT(N,LP)+2*BETA*TMAT(N-1,LP))
     .            / (1+2*BETA)

C  Else, if solving for nodes between first and last, then use the following equation.

          ELSE

           TMAT(N,LP)= (T0MAT(N,LP)+ BETA*(TMAT(N-1,LP)
     .      + TMAT(N+1,LP)))/(1+2*BETA)

          END IF

C  Checking for node temperature convergence using the 'NORM' method.

         DFF= (TMAT(N,LP)-T1MAT(N,LP))**2
         DIFF= DIFF+ DFF
300        CONTINUE

           DIFF= SQRT(DXT(LP)*DIFF)

       END DO

       END IF

297  CONTINUE
C
C  Latent loads.
C
C  Initialize some variables.
C
```

```
      ILAT=0
      QLAT=0.
      PVS=0.
      REST=0.

C  Start with first material specificatoins.

      LP=1

C  Setting the loops according to the number of materials.
C  Do loop from material 'first' to material 'last'.
       DO 305 LP= 1,NMAT
C
C  Do loop from first node to last node.

       N=1

       DO 307 N=1,NODE(LP)

        DV=1.
C
C  If solving for the first node, then divide the bulk mass by '2'.

         IF (N.EQ.1.OR.N.EQ.NODE(LP)) DV=2.
C
C  If using one-node model, then calculate the penetration depth from Fourier number equation.
C
         IF (NODE(LP).EQ.1) THEN

          DXM(LP)= SQRT ( DIFUSE(LP)*24./10000.)
         DV=1

C  If the penetration depth is greater than the material ful thickness, then use the full thickness as the penetration
C  depth.

          IF (DXM(LP).GT.DXT(LP)) DXM(LP)= DXT(LP)

         END IF
C
C  Calculating the bulk mass or the mass of the material involved..

        MFUR(N)= DENSMAT(LP)*DXM(LP)*HTA(LP)/DV
C
C  Calculating the nodal moisture content.
C
        U(N,LP)= MW(N,LP)/MFUR(N)

C  Calculating the maximum possible moisture content.that that occurs at equilibrium relative humidity of C  '1'.

        UC(LP)= A1(LP) +C1(LP)
```

```
C  If moisture content is greater than maximum allowable, then 'STOP'.

      IF (U(N,LP).GT.UC(LP)) GO TO 1402

C  Calculating the equilibrium relative humidity at every node.using Newton's method.
C  Initialize the equilibrium relative humidity.

      FE= .001
      ERROR=5.

C  Do loop until error <= 1%.

      DO WHILE (ERROR.GT..001)

       FE1=FE
       DD1= A1(LP)*FE**B1(LP) + C1(LP)*FE**D1(LP)
       DD2= A1(LP)*B1(LP)*FE**(B1(LP)-1)
     ;     +C1(LP)*D1(LP)*FE**(D1(LP)-1)

       FE= FE - (DD1-U(N,LP))/(DD2)
       ERROR= ABS ((FE-FE1)/FE)

      END DO

C  if equilibrium relative humidity > 1, 'STOP'..

      IF (FE.GT.1.0) GO TO 1402

C  Calculating the equilibrium vapor pressure at every node..

      PV(N,LP)= FE*EXP(23.7093-4111/(TMAT(N,LP)+237.55))


C  Calculating the nodal moisture permeability.

      PERM(N)= A2(LP)*FE**B2(LP) + C2(LP)*FE**D2(LP)
       PERM(N)= PERM(N)*.0075/(1000.)

C  Calculating the miosture conductive resistance ( R= thickness/permeability ).

      RA(N,LP)= DXM(LP)/PERM(N)
      RA(N,LP)= RA(N,LP)/HTA(LP)

307   CONTINUE

C  Calculating the moisture convective resistance

     N=1
```

```
C  Convective heat transfer coefficient.

     HTC= HTC0

C  Calculating the convective mass transfer coefficient using 'Lewis Relation'.
C  HM= HTC/CP

       HM= HTC*3.6.*.622/(CP.*101325.)

C  Calculating the convective resistance.

     RC= 1/(HM*HTA(LP))

     RES(LP)= 1/RC

C  If using one node only, the total resistance is a combination of a convective and a conductive terms.

      IF (NODE(LP).EQ.1) RES(LP)= 1/(RC+RA(1,LP)/2)

C  Collecting terms and calculating the sum of the reciprocals of the reciprocal of the conductive resistance at
C  every node.

     PVS= PVS + PV(1,LP)*RES(LP)
     REST= REST + RES(LP)

305  CONTINUE

     AA= (FLWI+FLWV)
     BB=(FLWI*WAMB+FLWV*WVENT+WGEN+QLPEPL/HVAP(IU))

C  Calculating the room air humidity ratio.

      WR= ((BB+PVS)*DELT + WRO*MAIR)
    ;     /( (AA+101325*REST/.622)*DELT + MAIR )

       PVR= 101325.*WR/.622

C  Solving for the nodal moisture mass.

     LP=1

C  Solve from first to last material.

     DO 310 LP= 1,NMAT

C  If using the Modified Penetration Depth model, then use the following equation.

      IF (NODE(LP).EQ.1) THEN

       MW(1,LP)= MW0(1,LP) + DELT*(PVR-PV(1,LP))*RES(LP)
```

```
      ELSE

C  else, solve implicitly for nodal moisture mass.

      ER1=5.

      DO WHILE(ER1.GE.1E-4)

      ER1=0.
      N=1
        DO 312 N=1,NODE(LP)

         M1(N,LP)=MW(N,LP)

C  If solving for the first node, then use the following equation.

         IF (N.EQ.1) THEN

         MW(N,LP)= MW0(N,LP)+ (2*(PV(N+1,LP)-PV(N,LP))
     .          /(RA(N+1,LP)+RA(N,LP))
     .          - (PV(N,LP)-PVR)*RES(LP))*DELT

C  Else, if solving for th elast node, then use the following equation.

         ELSE IF (N.EQ.NODE(LP)) THEN

         MW(N,LP)= MW0(N,LP) -2*DELT*(PV(N,LP)-PV(N-1,LP))
     .              /(RA(N,LP)+RA(N-1,LP))

C  else, if solving for the nodes in between, then use the following equation.

         ELSE

         MW(N,LP)= MW0(N,LP)+2*DELT*( (PV(N+1,LP)
     .    -PV(N,LP))/(RA(N+1,LP)+RA(N,LP))
     .    -(PV(N,LP)-PV(N-1,LP))/(RA(N,LP)+RA(N-1,LP)) )

         END IF

C  Checking for the nodal moisture mas convergence using the 'NORM' method.

         ERS= (MW(N,LP)-M1(N,LP))**2
         ER1= ER1 + ERS

312      CONTINUE

      ER1= SQRT(DXM(LP)*ER1)
```

```
      END DO

      END IF

      ERX1(LP)= 0.

      Do 313  N= 1,NODE (LP)

        ERX(N)= ( MW (N,LP)-MW2 (N,LP))**2
        ERX1(LP)= ERX1(LP) + ERX(N)

313     CONTINUE

        ERX1(LP)= SQRT ( ERX1(LP)*DXM(LP))

310   CONTINUE

      DMTOT=0.
      MWTOTAL=0.

      DO 314 LP=1,NMAT

C  Calculating the nodal rate of moisture mass change for every material.

      DMT(LP)= (PVR-PV(1,LP))*RES(LP)

        DO 315 N=1,NODE(LP)

C  Calculating the new total moisture mass for all the materials combined.
        MWTOTAL= MWTOTAL + MW(N,LP)
        MW2(N,LP) = MW(N,LP)

315     CONTINUE

C  Calculating the rate of total moisture mass change.

      DMTOT= DMTOT + DMT(LP)
      DMT(LP)= DMT(LP)/HTA(LP)

314   CONTINUE

C  Checking for the room temperature convergence.

        KEY1=0

      DO 316 LP=1,NMAT

      IF ( ERX1(LP) L.E.1E-4 ) THEN
       KEY(LP)= 1
      ELSE
```

```
     KEY(LP)= 0
    END IF

       KEY1= KEY(LP) + KEY1

316 CONTINUE

C  Checking for the room humidity ratio convergence.

     KEY2= 0
     CHECK2= ABS ( (WR-WR1)/WR )

    IF ( CHECK2.LE.1E-4 ) KEY2= 1

     KEY= KEY1 + KEY2

     ITERATION= ITERATION +1

C  If more than 40 iterations were performed for this procedure, then 'STOP'.

     IF( ITERATION.GT.40 ) GO TO 1405

    END DO

    IF(MODE.EQ.2 .OR. (WR.LT.WMAX.AND.WR.GT.WMIN)) GO TO 320

    ILAT=1

    IF(WR.GT.WMIN) GO TO 318

    WR=WMIN
    WRF=WR

    GO TO 320

318  WR=WMAX

    WRF=WR

320  QLINF=FLWI*(WAMB-WR)*HVAP(IU)
    QLVENT=FLWV*(WVENT-WR)*HVAP(IU)
    QLGEN=WGEN*HVAP(IU)

    IF(ILAT.EQ.1) QLAT=QLINF+QLVENT+QLGEN+QLPEPL

    ISTORE=INFO(10)-1
    S(ISTORE+4)=WRF

C  Storing the new nodal temperatures, moisture mass, rate of moisture mass change, total rate of moisture mass
C  change and room humidity ratio in the S-array.
```

```
      KSTORE= ISTORE + NSTOR
      NN=1

      DO 325 LP=1,NMAT

       DO 330 N=1,NODE(LP)

        S(KSTORE+NN)= TMAT(N,LP)
        S(KSTORE+NN+1)= MW(N,LP)
        NN=NN+2

330    CONTINUE

       S(KSTORE+NN)= DMT(LP)
       NN=NN+1

325   CONTINUE

      S(KSTORE+NN)= DMTOT
      S(KSTORE+NN+1)= WR

C  Outputs.

355     OUT(1)= TR
        OUT(2)= WR
        OUT(3)= QCONV
        OUT(4)= 0.3*QSPEPL
        OUT(5)= QSINFL
        OUT(6)= QSVENT

        IF(MODE.EQ.2) GO TO 360

        OUT(7)= QSENS
        OUT(8)= QLAT

        IF(QSENS.GT.0.)
     .   OUT(9)=AMAX1(QSENS+AMAX1(QLAT,0.),OUT(10+NOUTS+1))

        IF(QSENS.LT.0.)
     .   OUT(10)=AMIN1(QSENS+AMIN1(QLAT,0.),OUT(10+NOUTS+2))

360     IF(NOUTS.EQ.0) RETURN

C
C  Optional outputs.
C
      DO 450 L=1,NOUTS

       LO=LOUT(L)
```

```
      I=IOUT(L)
        GO TO (370,380,390,400,410) ,LO
C
C  Inside surface temperature.
370  OUT(10+L)=T(I)
      GO TO 450
C
C  Equivalent inside room temperature.
380  OUT(10+L)=TEQ(I)
      GO TO 450
C
C  Absorbed radiation gains.
390  OUT(10+L)=AREA(I)*TSOL(I)
      GO TO 450
C
C  Energy convected to the room.
400  OUT(10+L)=HCI(I)*AREA(I)*(T(I)-TR)
      GO TO 450
C
C  Thermal radiation transfer.
410  OUT(10+L)=AREA(I)*HCI(I)*(TEQ(I)-TR)-TSOL(I)
C
450  CONTINUE
      RETURN
C
C  Errors.
500     WRITE(LUW,501) INFO(1),INFO(2)
501     FORMAT(//2X,22H***** ERROR ***** UNIT,I3,5H TYPE,I3/4X,
    .  38HMATRIX IS SINGULAR, SIMULATION STOPPED/)
        STOP
600     IERR=IP+1
        WRITE(LUW,601) INFO(1),INFO(2),IERR,NSMAX
601     FORMAT(//2X,22H***** ERROR ***** UNIT,I3,5H TYPE,I3,
    .  12H PARAMETER #,I4,/4X,22HSURFACE # MUST BETWEEN
    .  6H 1 AND,I3,21H - SIMULATION STOPPED/)
        STOP
700     IERR=IP+2
        WRITE(LUW,701) INFO(1),INFO(2),IERR
```

```
701   FORMAT(//2X,22H***** ERROR ***** UNIT,I3,5H TYPE,I3,
   . 12H PARAMETER #,I4,/4X,25HSURFACE TYPE MUST BETWEEN
   . 29H 1 AND 5 - SIMULATION STOPPED/)
      STOP
800   IERR=NCK+1
      WRITE(LUW,801) INFO(1),INFO(2),IERR
801   FORMAT(//2X,22H***** ERROR ***** UNIT,I3,5H TYPE,I3,
   . 12H PARAMETER #,I4,/4X,24HONLY 10 OPTIONAL OUTPUTS
   . 29H ALLOWED - SIMULATION STOPPED/)
      STOP
1000  IERR=IL+1
      WRITE(LUW,1001) INFO(1),INFO(2),IERR
1001  FORMAT(//2X,22H***** ERROR ***** UNIT,I3,5H TYPE,I3,
   . 12H PARAMETER #,I4/4X,28HONLY 5 OPTIONAL OUTPUT TYPES
   . 33H ARE ALLOWED - SIMULATION STOPPED/)
      STOP
1100  IERR=IL+2
      WRITE(LUW,1101) INFO(1),INFO(2),IERR
1101  FORMAT(//2X,22H***** ERROR ***** UNIT,I3,5H TYPE,I3,
   . 12H PARAMETER #,I4/4X,24H SURFACE # NOT DEFINED -
   . 19H SIMULATION STOPPED/)
      STOP
1200  WRITE(LUW,1201) INFO(1),INFO(2),I
1201  FORMAT(//2X,22H***** ERROR ***** UNIT,I3,5H TYPE,I3,/4X,
   . 7HSURFACE,I3,36H IS NOT DEFINED - SIMULATION STOPPED/)
      STOP
1300  FORMAT(///6X,'VIEW FACTOR SUMMARY FOR UNIT',1X,I2,
   .   ' TYPE',1X,I2//2X,'I',3X,'J=',15(4X,I2)/)
1301  FORMAT(1X,I2,5X,15(1X,F5.3))
1302  FORMAT(//)
1400  FORMAT(//2X,24H***** WARNING ***** UNIT,I3,5H TYPE,I3,/4X,
   . 47HINCOMPLETE ENCLOSURE - SUM OF VIEW FACTORS FROM,/4X,
   . 7HSURFACE,I3,35H TO OTHER SURFACES IS LESS THAN ONE)
1402  WRITE(LUW,*) ' "U" MORE THAN MAX ALLOWED FOR MATERIAL',LP
      STOP
1405  WRITE(LUW,*)'NO CONVERGENCE, SMALLER TIME STEP RECOMMENDED'
      STOP
      END
```

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