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Evaporative cooling is used extensively for cooling in climates with medium to low humidity. In residential buildings the conventional mechanical systems tend to be both noisy and unsightly protuberances on buildings, whilst passive cooling systems incorporated into the building structure which employ chimneys and the like tend to be designed for the specific building and so may not have wide application. This study presents a proposal for a passive evaporative cooling system which makes use of natural ventilation at the building facade. The system makes use of the evaporative effect from water falling vertically along guides to produce a reduction in the temperature of the air entering the building. It can also be used as a design element in the building facade. Such a system provides an inexpensive, energy efficient, environmentally benign and potentially attractive cooling system.

A numerical study is presented to demonstrate the system efficiency and air flow rate through a building, making use of measured outside wind speed and direction, building geometry and surroundings. A design parameter of 0.03 is defined as an optimum volumetric airflow rate per square meter of the surface area of the water in the evaporative cooling system. The likely effect of the system on the indoor air temperature is discussed.

Thermal comfort level which is created by the passive evaporative cooling system as proposed is estimated and analysed. Then, the minimum climatic conditions required to achieve an appropriate comfort zone is presented. The present investigation has revealed that, theoretically this system is capable of providing thermal comfort in summer for occupants of the residential buildings in arid and semi-arid regions. The results of calculations indicate that, indoor environmental condition achieved by this system is only 5.4% over the satisfaction level.

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A PASSIVE EVAPORATIVE COOLING SYSTEM FOR RESIDENTIAL BUILDINGS

A thesis submitted as a part of the requirements for the degree of Doctor of Philosophy (PhD)

by

Zahra Ghiabaklou (M.Sc)

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The University of New South Wales

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1996

IN THE NAME OF GOD,

MOST GRACIOUS, MOST MERCIFUL

CANDIDATE'S DECLARATION

I hereby declare that this submission is my own work and that, to the best of my knowledge and belief, it contains no material previously published or written by another person nor material which to a substantial extent has been accepted for the award of any other degree or diploma of a university or other institute of higher learning, except where due acknowledgment is made in the text.

I also declare that the intellectual content of this thesis is the product of my own work, even though I may have received assistance from others on style, presentation and language expression.

Zahra Ghiabaklou

PUBLICATIONS ARISING FROM THIS STUDY

1. Ghiabaklou, Z., Ballinger, J. A. and Prasad, D. K. 1995. The Simulation of Naturally Ventilated Residential Buildings in Semi-Arid Regions. *Proceeding of the Annual Conference of the Australian and Newzeland Solar Energy Society*, 29 November- 1 December. Hobart, Tasmania.

2. Ghiabaklou, Z. and Ballinger, J. A. 1996. The Simulation of Naturally Ventilated Residential Buildings in Warm-Humid Climate. *Proceeding of the International Conference of the Modelling, Simulation and Optimisation*, May 6-9, Gold Coast, Australia.

3. Ghiabaklou, Z. and Ballinger, J. A. 1996. A Passive Evaporative Cooling System by Natural Ventilation. *J. Building and Environment.* in press.

4. Ghiabaklou, Z. 1996. Passive Evaporative Cooling Techniques. *J. Abady.* in press (in Persian).

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ABSTRACT

Evaporative cooling is used extensively for cooling in climates with medium to low humidity. In residential buildings the conventional mechanical systems tend to be both noisy and unsightly protuberances on buildings, whilst passive cooling systems incorporated into the building structure which employ chimneys and the like tend to be designed for the specific building and so may not have wide application. This study presents a proposal for a passive evaporative cooling system which makes use of natural ventilation at the building facade. The system makes use of the evaporative effect from water falling vertically along guides to produce a reduction in the temperature of the air entering the building. It can also be used as a design element in the building facade. Such a system provides an inexpensive, energy efficient, environmentally benign (not requiring ozone damaging gas as in active systems) and potentially attractive cooling system.

A numerical study is presented to demonstrate the system efficiency and air flow rate through a building, making use of measured outside wind speed and direction, building geometry and surroundings. A design parameter of 0.03 is defined as an optimum volumetric airflow rate per square meter of the surface area of the water in the evaporative cooling system. The likely effect of the system on the indoor air temperature is discussed.

Thermal comfort level which is created by the passive evaporative cooling system as proposed is estimated and analysed. Then, the minimum climatic conditions required to achieve an appropriate comfort zone is presented. The present investigation has revealed that, theoretically this system is capable of providing thermal comfort in summer for occupants of the residential buildings in arid and semi-arid regions. The results of calculations indicate that, indoor environmental condition achieved by this system is only 5.4% over the satisfaction level.

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LIST OF SYMBOLS

a	constant
α & β	the wind angle in degrees
ach	air changes per hour
a_r & b_r	terrain constants of the reference terrain
A	free area of inlets or outlets assumed equal (m^2)
A_e	the effective window area (m^2)
A_i	open inlet area (m^2)
A_o	open outlet area (m^2)
A_w	surface area of water (m^2)
C, m	constants
Ca	actual flow coefficient for large porosities
Cd	the discharge coefficient
C_{de}	effective discharge coefficient of inlet and outlet
C_{di}	discharge coefficient 0.60
Cp	pressure coefficients at inlet and outlet referenced to the site wind speed
ΔCp	the pressure coefficient difference across the inlet and outlet
C_{pm}	humid specific heat (kJ/kg (dry air). K)
CQ	the flow coefficient
c_p	specific heat at constant pressure (kJ/kg (dry air). K)
cf	conversion factor
D	diameter
E_s	cooling or saturation efficiency
f_{cl}	the ratio of the surface area of the clothed body to the surface area of the nude body
G	the natural log of the side ratio
h	height in the reference terrain (m)

H	height in building terrain where V_{ref} is required (m)
$\overline{h}$	average convection heat transfer coefficient (W/m^2. K)
h_c	convection coefficient
H_S	system height per story
I_{cl}	insulation of clothing in clo unit
k	thermal conductivity (W/m. K)
K_m	mass transfer coefficient (g/s. m^2)
L	length of the system (balcony)
ln	the natural logarithm
μ	viscosity (kg/s . m)
$\dot{m}_A$	mass flow rate of air (g/s)
$\dot{m}_B$	rate of mass transfer (g/s)
M/A_{DU}	metabolic rate (W/m^2)
n	the number of openings through which the air flows
η	mechanical efficiency
NCp	the normalised Cp
N_L	number of water lines per rows
N_R	number of rows
Nu	Nusselt number
$\overline{Nu}_D$	Nusselt number related to diameter
p	atmospheric pressure (kPa)
p_a	pressure of water vapour in ambient air (mmHg)
Pr	Prandtl number
Pr_s	Prandtl number at water temperature
p_w	partial pressure of water vapour in moist air (kPa)
Q	air flow rate (m^3/s)
ρ	mass density (kg/m^3)
Re	Reynolds number
Re_{Dmax}	Reynolds number based on the maximum fluid velocity occuring within the tube bank
RH	relative humidity

S	the side ratio
SCF	shielding correction factor
S_D	diagonal pitch (m)
S_L	longitudinal pitch (m)
S_T	transverse pitch (m)
T	absolute temperature K (K $^{o}C + 273.15$)
t_a	air temperature (^{o}C)
t_c	comfort temperature (^{o}C)
t_{cl}	temperature of clothing surface (^{o}C)
T_i	average temperature of indoor air in height h (K)
t_m	mean monthly outdoor temperature (^{o}C)
t_{mrt}	mean radiant temperature (^{o}C)
t_o	operative temperature (^{o}C)
T_o	dry-bulb temperature of outdoor air (K)
T_s	thermodynamic wet-bulb temperature of the entering air (K)
t_{db}	dry-bulb temperature (^{o}C)
t_{wb}	wet-bulb temperature (^{o}C)
ν	Kinematic viscosity (m^2/s)
V	wind speed (m/s)
V_h	the free stream wind velocity at roof height (m/s)
V_{max}	maximum velocity (m/s)
V_{ref}	site external wind speed at a reference height (m/s)
W	humidity ratio of moist air (mass of water per mass of dry air kg/kg)
W_1	the width of the wall for which Cp is sought
W_2	the width of the adjacent wall
W_i	inlet moisture (kg water/kg air)
W_o	outlet moisture (kg water/kg air)
W_s	moisture at saturation (kg water/kg air)
W_s*	humidity ratio of moist air at saturation point

x/h a wind shading factor with x the fetch length to the closest building of comparable height on the windward side and h the roof height of the building

Zv zone volume (m^3)

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CHAPTER 1

INTRODUCTION

In this study an attempt has been made to develop an energy efficient, environmentally safe and potentially attractive cooling system for residential buildings in arid and semi-arid regions. This system in which water passes over vertical guides in the form of a screen will provide passive evaporative cooling combined with natural ventilation for summer comfort in such climates. On the basis of this idea, the natural ventilation through the building in contact with the exposed water, will reduce indoor air temperature.

One of the main purposes of a building is to provide a comfortable environment for its occupants. Thus, there is a growing demand for space cooling in hot climates as people seek to raise their standard of living and improve work performance. This results in an increasing demand for energy during the day, competing with other more pressing needs such as refrigeration and the demands of the manufacturing sector.

The internal environment of a modern building can be made more comfortable no matter how uncomfortable the external environment is. Housing is often not oriented optimally for energy conservation, since excessive emphasis is placed upon aesthetics and repetitive use of fixed designs; again this results in energy inefficient design. Thus, current design of residential buildings leads to an energy expensive lifestyle.

Carroll (1982) showed that buildings in developed countries use up to 40% of primary energy, much of which is in the form of air-conditioning requiring electricity. The conversion of fossil fuels to electricity involves considerable expense and it is about 40% less efficient. Total energy consumption is expected to rise even higher as existing refrigerants are replaced with the thermally less efficient but environmentally friendly substitutes. This increasing consumption has created important problems for utilities which, in some countries, have had to build additional power plants, thus increasing the average cost of electricity. Developing countries in hot climates are facing serious peak demand problems as a result of such trends. In Iran, for example, the residential sector alone accounts for 61.6% of the electricity used in 1989 (Salehi, 1993).

This increase in energy consumption, has environmental side-effects associated with the increased CO_2 emissions, the ozone-depleting chloroflurocarbons used in air conditioners and depletion of the earth's limited resources. According to Kelly (1990), the air-conditioning industry consumes approximately 25% of the world's production of fully halogenated chloroflurocarbons (CFCs), which are responsible for the depletion of the earth's ozone layer. In addition, the burning of fossil fuels has a serious effect in that it releases sulfur dioxide and trioxides into the atmosphere, leading to increased acidity of rainfall which has a damaging effect upon vegetation and water supply. Pollution in the atmosphere has also resulted in air being difficult to breathe, especially for asthma sufferers.

One of the most widely used cooling devices in Tehran, is the single-effect direct mechanical evaporative cooler as shown in Figure 1.1.

A fan is used to draw outdoor air inwards through pads which are saturated with water dripping downwards from a water tough. The water is supplied by a small pump which raises the water from a sump at the bottom of the unit. Make-up water is supplied from the city supply through a float valve. A two speed fan motor is available in these units.

In addition to the electricity consumption associated with this cooling system, it has other disadvantages. Many types of evaporative coolers are located outside and so require ducting, which increases the cost of the system and lowers its efficiency. In the cases where these coolers are installed in facades of the building or attached to the windows, they are also unattractive in appearance.

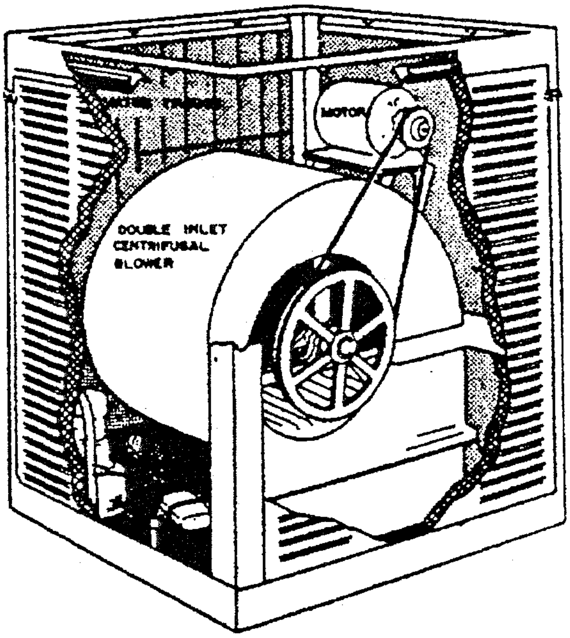


Figure 1.1: Conventional single-effect direct evaporative cooler (Yellot, 1981).

The challenge is to remedy these problems with particular reference to housing development. How can building development encompass energy efficiency, environmental stability and provide a comfortable living environment.

An obvious starting point may be to simply reduce consumption, that is, to use less electricity and move to more energy efficient building techniques.

The use of passive or renewable energies instead of fossil fuels is one effective alternative which can reduce the impact of these services on the environment. Passive cooling, for one, reduces electrical energy consumption compared to a mechanically operated system. Although passive cooling cannot maintain the temperature, relative humidity and air movement at a set level as can an air-conditioning system, it can provide some degree of comfort and requires very little or no electrical energy for operation.

In principle there are several possible ways of lowering the indoor temperatures by means of the natural conditions that exist both in arid and semi-arid environments. Passive evaporative cooling is one such technique. Evaporation occurs when the vapour pressure of water is higher than the partial vapour pressure in the adjacent air. During this phase change of water into vapour, sensible heat from the air is absorbed and the air temperature decreases while its moisture content increases. This is known as the direct evaporative cooling process and is a form of cooling is well suited to arid and semi-arid regions where the relative humidity is relatively low.

This study aims to assess the technical feasibility of a climate control strategy which could provide an energy efficient and environmentally benign system that satisfies human's physiological needs of thermal comfort through a simple natural process.

It is anticipated that such a cooling system would be inexpensive and potentially attractive. In addition a case can be made for the psychological benefit to people in a hot climate of the sound of falling water. The testing of performance, cost estimation and psychological aspects of the system on the occupants needs further research and will not be considered in this study.

1.1 Research Methodology

To design the most efficient combination of a building and its evaporative cooling system, a base case building model has been chosen for which parametric studies are undertaken. A numerical study is used to demonstrate the proposed evaporative cooling system efficiency and air flow rate through a building, making use of measured outside climatic data, building geometry and surroundings. A modified version of the computer simulation program CHEETAH (Delsante, 1995) will be used to predict the thermal behaviour of the building with a passive evaporative cooling system to explore the theoretical integration of such a system into the building fabric. A number of design alternatives and a parametric analysis is performed as a series of modifications to a base case which defines a particular set of construction, design and operating characteristics for the building air changes and efficiency of the system.

The level of thermal comfort created by the proposed system is defined using comfort temperature equations and Fanger's (1970) predicted mean vote (PMV) and finally, the climatic range and conditions required to achieve an appropriate comfort evaporative cooling is studied.

The question posed for this study is "the extent to which the passive evaporative cooling system as presented, is capable of providing thermal comfort in summer for occupants of the residential buildings in arid and semi-arid regions".

1.2 Research Outlines

The research outlines are as follow:

- Chapter 2 presents a literature review of current knowledge of various passive cooling systems. This background knowledge is used to present a proposal for a new passive evaporative cooling system which makes use of natural ventilation at the building facade.

- As this cooling system works only with natural ventilation, Chapter 3 investigates the theoretical aspects of calculating natural ventilation rate and particular methods of doing this. The results of this are then used to predict the airflow rate through a given building in relation to various parameters.

- Chapter 4 focuses on the method of calculating heat and mass transfer of the system. Since, there is no specific information or formulae relevant to calculating efficiency of this particular system, an extensive investigation and personal interviews were conducted which form the basis of this chapter. Based on the equations presented in this chapter, a parametric study was undertaken to investigate the effects of various parameters on the efficiency of the system.

- Chapter 5 begins by presenting the base case building and computer simulation program used in this research. Then, a parametric study is used to investigate the various parameters affecting on the ventilation strategy, system design and thermal behaviour of the building.

- Chapter 6 has two parts. The first part deals with the thermal comfort level achieved by the system and various affecting parameters. The second part defines suitable climatic range for the proposed cooling system.

- Chapter 7 presents the conclusion of the research and related recommendations for further work.

CHAPTER 2

REVIEW OF PASSIVE COOLING SYSTEMS

2.1 INTRODUCTION

Passive cooling systems are systems that require negligible electric or fossil fuel energy to remove heat from a person or a building to an environmental sink (Clark, 1989). The term "passive cooling" emphasises one end of that spectrum of conditions that bases human comfort on the natural climate (Cook, 1984). Passive space heating is driven only by the sun and was named first. "Passive cooling" is its counterpart. Unlike the single solar heat source of passive heating, passive cooling embraces several heat sinks and a wide variety of bioclimatic practices in building design.

In earlier times, when the supply of energy was greatly limited and the technology for modifying the internal environment mechanically was unavailable, the designers of buildings had to rely on given stratagems to maximise the comfort of the internal environment. Ancient builders had to contend not only with the same environmental factors with which we are faced, but also had greater limitation imposed through a restricted range of building materials. Even in the face of such restrictive technology and materials, the performance of their buildings proved incredibly good (Lesiuk, 1983).

The chapter presents a literature review of current knowledge of various traditional and modern passive cooling systems with special emphasis on passive evaporative cooling systems. The background knowledge of passive cooling types and resources is used to present a proposal for a new passive evaporative cooling system.

The term "passive" in the context of this study does not exclude the use of a fan or a pump when their application might enhance the performance. This term as explained by Givoni (1991), emphasises the utilisation of natural cooling sources, or heat sinks, for the rejection of heat from the building. If some power is needed to operate the system, that the heat transfer system is low cost and simple and that the ratio of power consumption to the resulting cooling energy is rather low.

2.2 PASSIVE COOLING RESOURCES

Passive cooling resources are the natural heat sinks of the planet, those thermal transfer opportunities that generally balance the continuous energy inputs from our sun (Cook, 1989).

Passive or hybrid cooling systems may be classified according to the type of cooling sources they utilise, by the modes of heat transfer and fluid flow, and by the type of energy storage material they employ.

The "sources of coolness" which may be utilised as described by Bahadori (1981) are:

1. Ambient air, at temperatures below or around the comfort range.

2. The sky, to which thermal radiation may be emitted for cooling.

3. The ground, where the daily and annual temperature fluctuations are damped out with depth.

4. Water, which can provide cooling either by a temperature change or by a solid to liquid or liquid to vapour phase change.

The heat transfer modes are: Convection, thermal radiation, evaporation or condensation, solidification or liquefaction, and conduction. The fluid flow may be forced, using fans or pumps, or generated by the wind and buoyancy forces.

Another classification of passive systems is by the type of materials employed to store the coolness. Just like solar energy, natural sources of coolness are of an intermittent nature. It is therefore essential to store the coolness from the period of its availability to the time of its demands. The storage of coolness may be done for a short period of a few hours or for several months. Materials suitable for the storage are:

1. Building mass, in which the coolness may be stored for a few hours.

2. Rock beds, in which coolness may be stored for a few hours or a few months.

3. The ground, in which coolness is stored generally from winter to summer.

4. Water, in which coolness may be stored for a few hours or a few months; in the form of ice, it is generally stored for several months.

5. Phase changing materials, where the solid-liquid phase change is of particular interest, both for daily and seasonal storage of coolness.

Table 2.1 summarises the classification of passive and hybrid cooling systems. Also included in this table, are the factors considered important for cooling load reduction, the first step to be taken for any successful passive cooling system design.

It is clear from this table that there are many options for natural cooling of buildings. The objective is, of course, to provide thermal comfort for the building resident in summer. As a rule, we try to employ as many sources of coolness that are available, and any modes of heat transfer or fluid flow which can most effectively transport the coolness to the building or transport the heat from the building to the heat sinks (Bahadori, 1986).

Table 2.1: Summary of the classification of passive and low energy cooling systems (Bahadori, 1981).

1. Cooling load reduction
   A. Geometry
   B. Orientation
   C. Structural shading
   D. Mass, Insulation, and Infiltration effects
   E. Colour and texture
   F. Landscaping

2. Types of cooling sources
   A. Air
   B. Sky
   C. Ground
   D. Water
      a. Sensible
      b. Liquid-vapour phase hange
      c. Solid-liquid phase change

3. Modes of heat transfer
   A. Convection
   B. Thermal radiation
   C. Boiling or Condensation
   D. Solidification or Liquefaction
   E. Conduction

4. Modes of fluid flow
   A. Buoyancy effects
      a. In water
      b. In air
         1. Regular chimney effects
         2. Solar chimney effects
   B. Wind effects
   C. Fans
   D. Pumps

5. Types of energy storage material
   A. Building mass - Several hours
   B. Rock beds - Several hours or months
   C. Ground - Several months
   D. Water
      a. Sensible - Several hours or onths
      b. Ice - Several months
   E. Phase changing materials (PCM)

2.3 PASSIVE COOLING SYSTEMS

In many hot countries, and also in countries with a temperate climate having hot summers, there is growing interest in utilising passive and low energy systems for cooling buildings, both residential and commercial. In developed countries this interest is motivated by the desire to conserve energy and to reduce the summer peak demand for electricity caused by air conditioning. In developing countries, where air conditioning is not applied at present on a large scale, the interest in passive cooling is motivated by the desire to minimise the heat stress experienced in buildings and its deleterious effect on health and productivity, as well as the desire to minimise the pressure for large scale installation and use of air conditioning (Givoni, 1991).

The alternative cooling methods are:

1. Evaporative cooling.
2. Convective cooling.
3. Radiant cooling.
4. Earth cooling.

2.3.1 Evaporative Cooling

The most widely used process for reducing the dry bulb temperature of air is evaporative cooling. When hot and dry air are brought into contact with water either by drawing the air over a wetted surface or spraying droplets into the air stream, evaporation will take place (Yellot, 1981).

Water evaporation is used extensively in arid regions to cool the indoor air of buildings. Such a system is usually operated during the day when the cooling is most needed for large quantities of hot air. However, evaporative cooling can also be operated in night hours by storing the night coolness in thermal storage for day time use (Givoni, 1981a).

Direct evaporative cooling is nearly a constant energy process in which sensible heat is converted to latent heat. On the psychrometric chart, evaporative cooling follows the constant wet-bulb lines (ie., it is essentially an adiabatic process in which no heat is added or removed from the air). Instead, sensible heat in the ambient air is converted to latent heat as water is evaporated into the air; this process occurs as air is forced through the system. The result is a reduction in room dry-bulb temperature, while room relative and absolute humidity are increased.

2.3.1.1 Traditional Passive Evaporative Cooling Systems

Evaporative cooling was known to the ancient Egyptians. Frescoes excavated a century ago and dating from about 2500 BC show slaves fanning jars of water to keep them cool.

The vessels were presumably porous enough to maintain wet surfaces to facilitate the evaporation process. A wall painting in Herculaneum of about 70 AD shows a leather water bottle used for cooling drinking water by similar means. The ladies of the court of Frances the first sent to Portugal for "earthen vessels which would render the water cooler and more healthful" (Watt, 1963).

Traditional building techniques are normally well adapted to the climate and to the way of life. Imported western technology has often replaced the established tradition in the design of the buildings.

Many cultures have used passive cooling systems to make their living spaces comfortable. For centuries man's experience with the natural environmental factors has led him to diverse settlement forms in different settings. The emergence of various types of habitation occurs largely within the parameters of interacting sociocultural, economic, technological and physical environmental factors. This is a human ecological phenomenon. That is, while man is adapting to the limitations of natural factors i.e., climate, topography, available resources to generate his living environment, 'meso-environment' (Fitch, 1972), or architecture; he acts actively upon nature through his cultural forces his technology, tools, beliefs, ideologies, economic system and imprints his norms upon the resulting forms. There is a whole range of examples which reflect how and why people organise their environment in various forms (Rapaport, 1969).

Warm air passing over water evaporates the water and, as a significant amount of heat is absorbed in the process, the air is cooled. The evaporated water is retained in the air thus increasing its humidity; for this reason evaporative coolers can only be used in relatively dry climates and are found in desert, composite and Mediterranean zones. These coolers are based on the evaporation of a thin film of water on a carrier over or through which air is passed. The simplest system is a wooden frame across which open weave matting vegetable fibre is stretched. When hung in front of windows in the path of the natural air flow, and kept damp, the matting humidifies and cools the air as well as filtering out the dust. Another simple system entails the use of large, porous earthenware pots filled with water which seeps through the walls of the pot moistening the outside and cooling the passing air as it evaporates. In wind catchers, beds of wet charcoal over which the air passes before entering the room, are sometimes used (Konya, 1980).

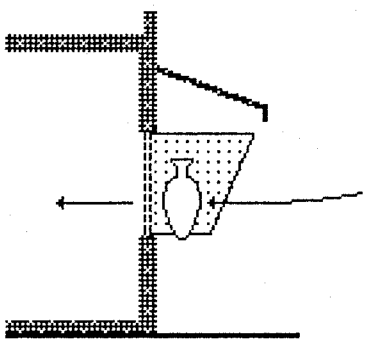


Figure 2.1: Traditional window evaporative cooler (based on Konya, 1980).

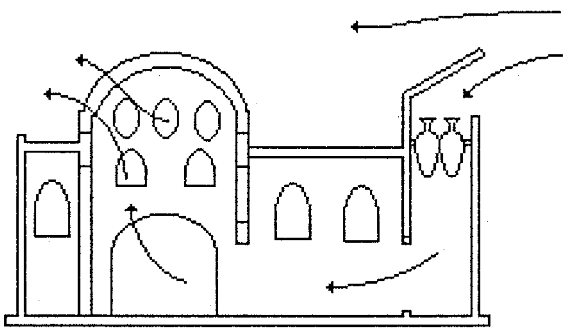


Figure 2.2: Traditional evaporative cooling by wind catcher (based on Konya, 1980).

The same principle can be used by channelling breezes over pools or water sprays before they enter buildings. To ensure that the cooled and humidified air enters the buildings, the pool should be contained between walls on two or, preferably, three sides.

A spray pond is more effective than a still pool of the same size and has the additional advantage that it not only cools the air but can also wash it; the water droplets stick to dust particles in the air which can then no longer remain in suspension (Konya, 1980).

The people of India still use the ancient "cocos tatti" of greater effectiveness. Doors are replaced in summer by tatties, frameworks similar to screen doors, covered with dried "khuss-khuss" grass. These are kept wet traditionally by coolies, but more recently by recirculating pumps and catch basing. Many are periodically soaked by balanced troughs at the top. Water trickling into these from supply pipes fills them until they over balance and dump their loads on the pads. Gravity then rights them for next cycle. The traditional Indian ventilating device, the broad ceiling hung cotton "punkah", or coolie-operated swinging fan, is occasionally operated in a moistened condition. Another Indian evaporative air cooler is the "therm antidote" known for over 50 years. Here a door is replaced by a wheel-like framework covered with khuss-khuss grass which is revolved and kept moist by coolies. In this, as in tatties, the grass lends a pleasant odour to the air (Watt, 1963).

Similar primitive uses of evaporative cooling are the canvas-covered canteens for soldiers' drinking water, the canvas "desert" water bags used widely in the United States and Australia, and the "ollas", or porous water jars, of the American-Indians and Mexicans. Western people preserved food in cloth-covered boxes or food coolers. A pan on top or a trough at the bottom kept the cloth damp by capillary action. Many such boxes were mounted outside kitchen windows so the wind encouraged evaporation. Simple comfort cooling followed. Occasionally, a window facing the wind was covered with burlap and moistened, or drapes in doorways dampened, or bed-sheets sprinkled with water for cooler sleeping.

The other case is a Texas porch semi-surrounded by falling water. On two sides perforated pipes attached to the cornice allowed a curtain of droplets to fall past the areas open to the air. The water fell into shrubbery which it irrigated. This arrangement's effectiveness depended, of course, upon the breeze. A similar device was repeatedly made by Peter the Great for one of his gardens in what is now St.Petersburg. A tree was piped so that a curtain of water fell from its outermost branches and surrounded, but did not wet, benches near the trunk (Watt, 1963).

Traditional buildings in Turkey sometimes feature a Selsebil, a kind of ornate fountain with water running down a sloping ceramic surface terminating in a small pool underneath. While water is sliding down it evaporates and helps the room to cool down (Erginbas, 1953). A measurement taken on a July afternoon in the cool room of sardab of the house of Diyarbakir indicated a $5^{o}C$ lower dry bulb temperature and 10% increase in relative humidity compared to that of outside (Imamoglu, 1980).

Thermal and airflow behaviour are conceptualised in traditional internal courtyard design. Their use is maximised year round in overheated arid conditions, while in overheated humid conditions optimisation occurs only during the periods of no rain. In these regions however they remain a constant source of cooling in well designed buildings. In double courtyards, building morphology and orientation further effect the heating and cooling of surfaces with a consequential effect on airflow patterns. External courtyards, aimed mainly at privacy, do not effect thermal regulation in the core of a building, but merely at the building's edge. Consequently orientation of site and building are crucial. Materials used in all circumstances (mass, live planting etc.) effect the radiation performance and hence the quality and volume of air in motion.

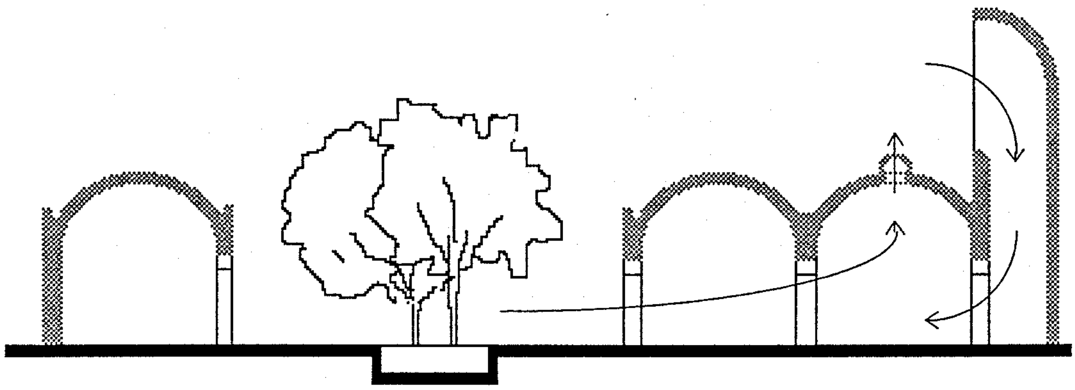


Figure 2.3: Evaporative cooling by typical courtyard airflow.

As more surface area is exposed to ambient climate in underheated conditions, thermal losses will be increased and generally courtyards in these locations are designed for natural day lighting, ventilation of internal spaces of buildings with a large floor plan, and for outdoor summer use (temperate) of the inhabitants to enjoy both seasonal weather and privacy. Courtyards height to plan ratio plays a large role in determining the quantity of cold air entrapment in the courtyard shaft. In underheated conditions this column of cold air is omnipresent, draining off the building's heat from its core in a 'lake-like' action, both day and night (Bowen, 1981).

Cooling of buildings by utilising the energy consumed in the process of water evaporation can be accomplished by two basically different approaches. The first is to cool directly by evaporation outdoor air which is introduced into the building. The air is thus humidified while its temperature is lowered and the indoor water content level is elevated above the outdoor level. This is direct evaporative cooling. The other approach is to cool by evaporation a given element of the building, which then serves as a heat sink and absorbs heat penerating into the building through its envelope or generated indoors.

This is indirect evaporative cooling. With such systems the indoor temperature is lowered without elevation of the indoor vapour content of the air (Givoni, 1984).

2.3.1.2 Direct Evaporative Cooling

In the day when cooling is most needed, this method can be used to cool large quantities of hot air (range 35-45 degree C). This approach cools directly by evaporating outdoor air introduced into the building. The air is humidified while its temperature is lowered and the indoor water content is elevated above the outdoor level (Givoni, 1986). If we take a tank of water equipped with a recirculating pump and spray water into the air, some of the water will evaporate. When it evaporates, heat is taken from the air. Therefore the air cools, and at the same time the moisture content increases.

Direct evaporative cooling is nearly a constant energy process. On the psychrometric chart, evaporative cooling follows the constant wet-bulb lines (ie., it is essentially an adiabatic process in which no heat is added or removed from the air). Instead, sensible heat in the ambient air is converted to latent heat as water is evaporated into the air; this process occurs as air is forced through the system. The result is a reduction in room dry-bulb temperature, while room relative and absolute humidity are increased.

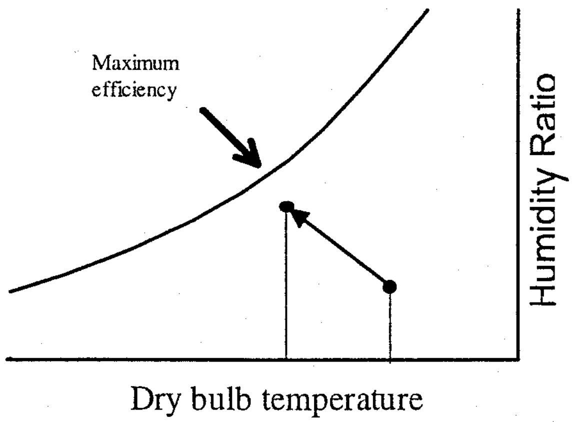


Figure 2.4: Evaporative cooling process on the psychrometric chart.

The skeleton psychometric chart presented in Figure 2.4, relates dry bulb temperature and humidity ratio. If we move up toward the saturation curve to the point of maximum efficiency, we will obtain a lower dry bulb temperature with more humid air. The effect of the increased relative humidity here on comfort is small compared to the reduction in dry bulb temperature.

2.3.1.2.1 The Use of Window Evaporative Pads

In many arid and desert regions, winds of rather high speed blow during the day time (mainly the afternoon hours) from a constant general direction, eg., from the sector between the northwest and the southwest in northern hemisphere. In such cases it is possible to cool small buildings by simple passive evaporative cooling devices. For instance, when wetted pads made of fibres are placed in windows facing strong winds they can provide natural direct evaporative cooling in the same way as the pad in the wall of a mechanical evaporative cooler. Their advantages are obvious in desert regions, where the afternoon winds are laden with dust. A secondary benefit of this device is that the wetted fibre is very effective in filtering dust out of the air penetrating through the window (Givoni, 1986).

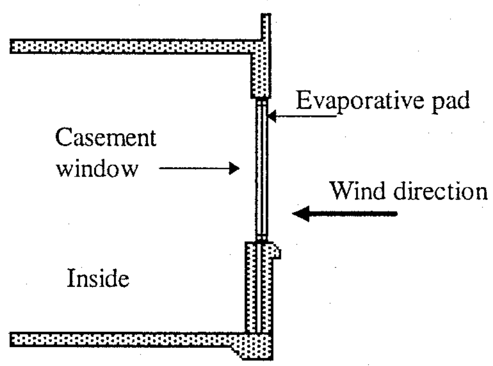


Figure 2.5: Direct evaporative cooling by window evaporative pad (based on Givoni, 1986).

To be effective, from the cooling viewpoint, the pads should provide a large surface area for evaporation, without creating too much resistance for the wind to penetrate through them. To provide the necessary surface area for the evaporation process, the pads have to be of a relatively large thickness, about 5-10 cm. The degree of saturation of the air passing through these home-made passive pads is usually lower than is the case with the pads used in mechanical devices. Temperature reductions of 40-50% of the DBT-WBT (dry bulb temperature-wet bulb temperature) differences can be expected (Givoni, 1991).

Disadvantages of window evaporative pad are visual blockage and reduced air flow speed in the building.

2.3.1.2.2 Performance of a Porch

When a porch is available on the windward side of the building, it is possible to put the evaporative pads in front of the porch. In order to provide daylight and not to block the view completely, windows and doors can be placed alongside the pads. If the doors are openable they provide access to the yard around the porch. In this way, the porch is turned into a front patio, where the air temperature is the coolest. From this patio the air enters the house proper through a connecting door (Givoni, 1991).

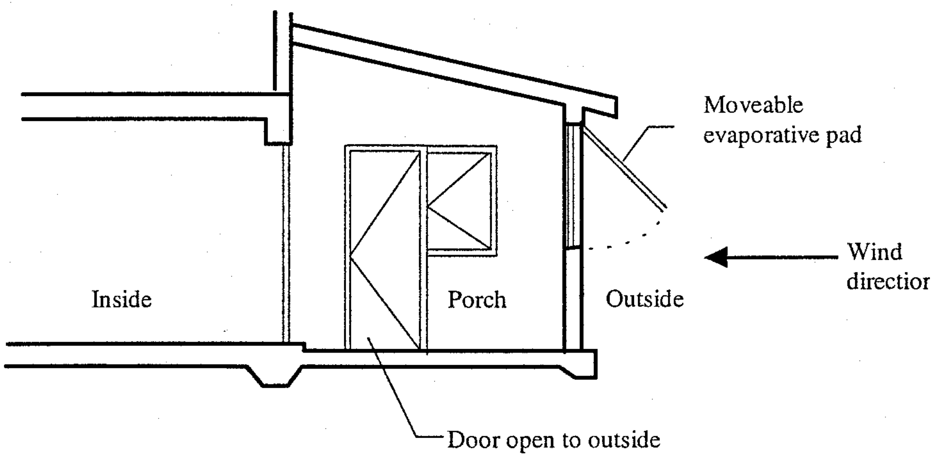


Figure 2.6: Evaporative pads installed in a porch on the windward side (based on Givoni, 1986).

2.3.1.2.3 Passive Evaporative Cooling Tower

Bahadori in 1985 proposed a new type of an evaporative cooling tower. A cross section of this tower which is attached to a residential building is shown in Figure 2.7. Compared with a conventional baud-geer (wind catcher), a more effective evaporative cooling can be accomplished, with the air entering the living space cooled appreciably.

Results of theoretical study conducted to show the effectiveness of this design. For a tower 5m high located in a region with a wind velocity of 5 m/s, the temperature of 40^{o}C and relative humidity of 15%, the air entering the tower can be cooled down to a temperature of 25^{o}C and a relative humidity of 65%. This conditioned air is admitted into a living space of 300 m^3 volume at a velocity of 0.1 m/s.

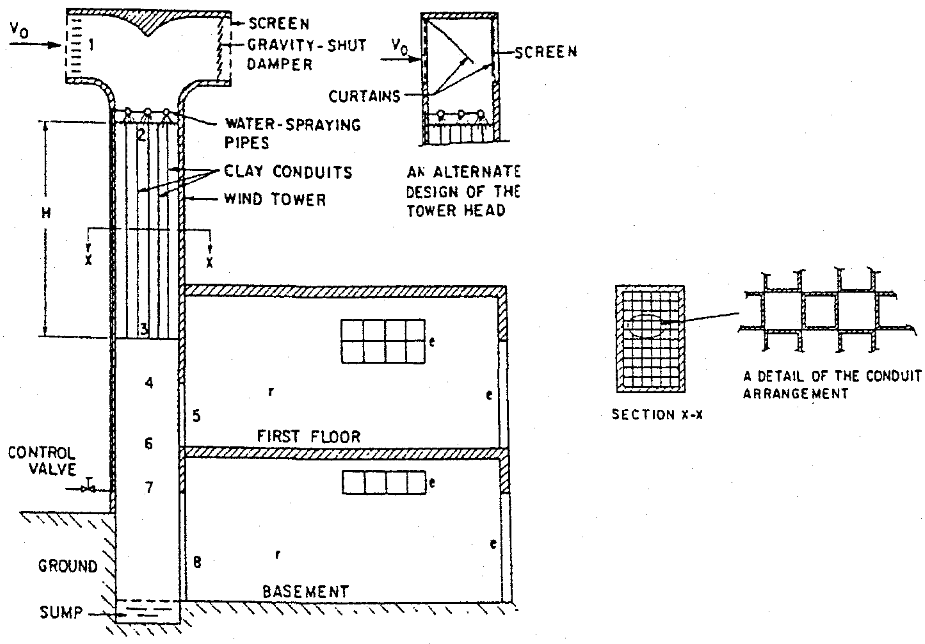


Figure 2.7: A cross section of the evaporative wind tower (Bahadori, 1985).

Passive cooling with natural draft cooling towers in combination with solar chimneys, has been proposed by William Cunningham and Lewis Thompson (1986). The test structure chosen for this work is a well-insulated frame building with 93 square metres of floor space and is part of ERL's (Environmental Research Laboratory) passive solar village in Tucson, Arizona, USA. The structure was retrofitted with an evaporative cooling tower. At the top, a 10 cm thick, vertical, wetted cellulose pad was used to cool the air. A plywood "X" baffle in the top acts to catch wind and direct it downward. A 90-watt pump recirculates the water over the pads from a sump at the base of the pads. Make-up water is added and regulated with a conventional float valve. Vertical pads were chosen over horizontal pads primarily because of the reduction in pumping requirements. A solar chimney is located at the opposite end of the house. The entire roof and attic space is used as a solar collector to assist the solar chimney. All of the return air is exhausted through the ceiling into the attic. The exhaust air makes two passes through the attic due to a baffle down the center of the ridge.

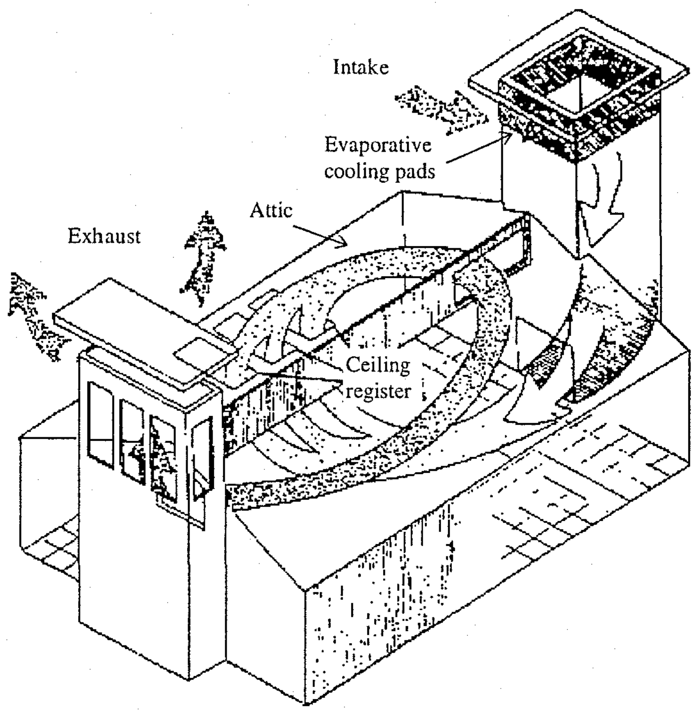


Figure 2.8: Air flow path through house (Cunningham & Thompson, 1986).

The solar chimney is intended to provide ventilation air movement within a structure, in combination with cooled downdraft chimney. In the downdraft chimney, air is cooled by an evaporative cooler, consisting of wetted packing material, which is an integral part of the chimney. The cooled air, heavier than ambient air, falls by gravity through the chimney into the structure to be cooled.

2.3.1.2.4 Construction of a Rock Wall Evaporative Pad

A rock wall can be constructed between two wire mesh panels held together by steel bars. The space between them filled by small sized rocks and wetted from the top. This replacement for the window's evaporative pad is more stable and more efficient with higher air flow rate (Givoni, 1986).

2.3.1.2.5 Rock Bed Evaporative Cooler

Robeson (1970) and John Yellot (1981) mentioned that many different combinations of direct evaporative coolers have been proposed to endeavour to make use of dry-bulb temperature reduction without paying a penalty in the form of a greatly increased Humidity Ratio. One of the first to archive success in this field was the Australian Commonwealth Scientific and Industrial Research Organisation (C.S.I.R.O), headquartered in Melbourne, Australia. Under the direction Roger Morse, past president of International Solar Energy Society (I.S.E.S.), Donald Pescod and Michel Woodridge, C.S.I.R.O. developed the concept of the rock bed regenerative (R.B.R.) cooler. This used two relatively shallow rock beds (A and B) and an evaporative cooler, with two blowers (1 and 2) and a damper to direct the air flow. Air from the cooled space is delivered by blower No. 1 through the evaporative cooler to rock bed A, which is cooled vigorously without adding moisture to the rocks. At the same time outdoor air is being drawn in through bed B for which the rocks have just been cooled. After a few minutes of operation in this mode, when the rocks in bed A have been cooled and those in bed B have been heated, the position of the damper is reversed and the hot outdoor air is drawn in through bed A and the evaporatively cooled discharge air is discharged through bed B. This relatively simple device proved to be effective in providing comfort in Australia's relatively dry regions and several hundred of them are installed in schoolhouses where the need for cooled make-up air is large.

2.3.1.2.6 Applicability of Direct Evaporative Cooling

Direct evaporative cooling is applicable only in arid regions, and where water is available. The maximum WBT is the main climatic criterion for the applicability of this system. The indoor maximum air temperature in reasonably insulated residential buildings, cooled by a direct evaporative cooling system, would be about 3-4 K above the maximum ambient WBT. Taking into account the effects on comfort of the higher indoor air speed and the higher humidity associated with direct evaporative cooling, it can be suggested that direct evaporative cooling is advisable only where and when the

WBT maximum in summer is not higher than about 22^{o}C or 24^{o}C and the DBT is not higher than 42^{o}C, in developed and in developing countries, respectively.

A drawback of direct evaporative cooling is the usage of a large quantity of high quality water, in addition to the real cost of the water. This factor may limit the application of direct evaporative cooling in arid regions with restricted water supply (Givoni, 1991).

Also some interesting studies in applicability of direct evaporative cooling has been carried out in Australia by Pescod (1968, 1971 and 1976).

2.3.1.3 Indirect Evaporative Cooling

Instead of cooling by evaporation the moist air which is introduced into the building, with high vapour content, it is possible to cool the roof of the building by evaporation (either by wetting the roof or by having a shaded pond over the roof). The building then is cooled by conduction across the roof. This is an indirect evaporative cooling by which indoor air and radiant temperatures are lowered without elevation of the indoor vapour content (3).

2.3.1.3.1 Roof Ponds

When a water pond, shaded and naturally ventilated, exists over an uninsulated roof, it provides passive and indirect evaporative cooling for the space below the roof. An insulated roof shades the pond from the sun. Openings in the roof allow air currents to pass over the pond during the summer. As water evaporates the pond will become cooler, and together with the ceiling structure will act as a heat sink for the interior of the building. During the winter the pond is drained and the roof openings are closed. The main disadvantage of this system is the cost of the double roof structure and waterproofing. A clever alternative to the above roof ponds is the roof pond with floating insulation. At night a pump sprays the water over the top of the insulation, and it cools by both evaporation and radiation. When the sun rises, the pumps and the water remains under the insulation, where it is protected from the heat of the day (Givoni, 1986; Lechner, 1991).

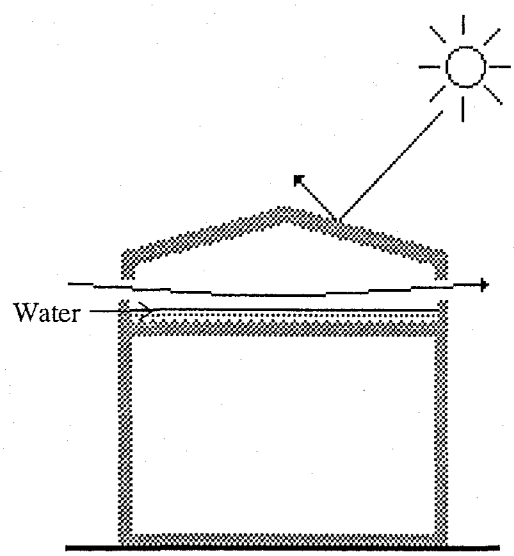


Figure 2.9: Indirect evaporative cooling by shaded roof pond (based on Lechner, 1991).

To bring the ceiling temperature as close as possible to that of the pond, the water must be in direct contact with the roof structure below, ie. no insulation may be installed between the water and the ceiling. This last factor has obvious repercussions for application of water ponds in regions with cool winters, where the roof should be insulated in winter. It also has implications for the options of ceiling treatment in spaces cooled by roof ponds. Any applied ceiling finish eg. acoustic treatments, should be detailed to allow maximum exposure of the structural slab to radiation, and should impair convective air flow as little as possible. Several techniques are possible for shading a roof pond. They differ in cost as well as in the efficiency of the evaporative cooling in the summer and in the thermal performance of the roof in winter.

There are three different types of shading and insulation for roof ponds:

1. Permanent insulated shading structure over the pond, continuously opened and ventilated in summer but closed in winter.
2. Shading by pebbles, with insulation embedded within the water pond under a layer of pebbles.
3. Insulation floating over the water, with water circulated above the insulation by pumping during the night.

In all of these systems the roof should be able to structurally support the water pond (about 200-400 kg/m^2, depending on the system) and be provided with perfect waterproofing (Givoni, 1984).

Davis and Schubert (1981) in their book, *Alternative Natural Energy Sources in Building Design*, described early evaporative roof pond systems. They pointed out that, water-cooled roofs were first experimented with in the late 1930s. They were almost immediately successful. At that time the roofs were simply flooded with 4 inches of water. The following figure shows the relative cooling effect of a roof pool or spray based on scientific tests.

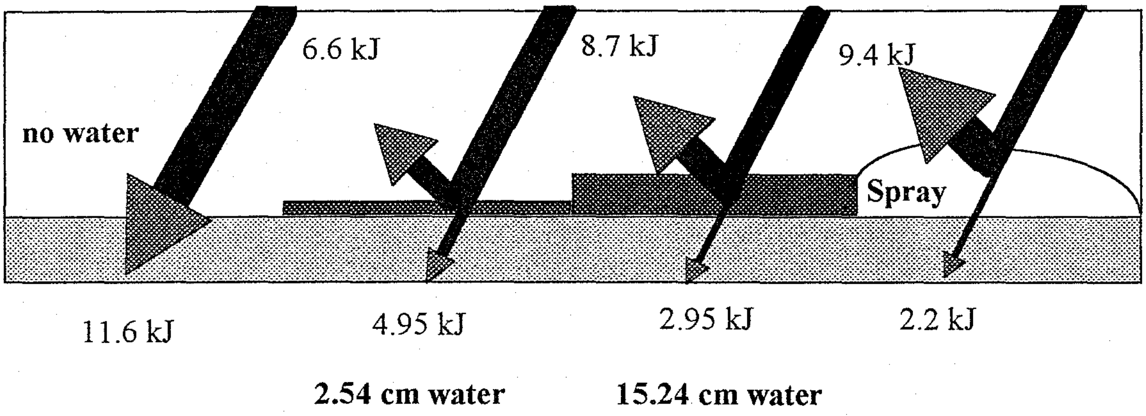


Figure 2.10: Relative effect of a roof pond and spray (based on Architectural Forum, June 1946).

The "Sky-therm" system (Hay & Yellot, 1969; Niles, Haggard & Hay, 1976), employs a roof pond and movable insulation. Their testing offered successful results for very hot, dry and humid areas. The "Sky-therm" process utilises six means of modulating the ambient temperature conditions: Solar heating; heat capacity; nocturnal radiation; water evaporation; radiation , evaporation and fan-coil operation; and fan-coil and pond-blower use. Hay (1973) described the structure of his first prototype in Phoenix, Arizona as follows: Corrugated metal sheets form the ceiling between upward-extending roof beam supports. A black plastic film above the metal sheets was crossed over the beams to waterproof the structure. 18 to 21 cm of water in transparent plastic bags was laid on lining between the beams. This amount of water provided the heat capacity effect of a foot of concrete with a weight equivalent of a 3-inch slab. Extruded aluminium trackings mounted a top the beams permitted insulation panels to be alternately positioned above the ponds or stacked two or three deep over a carport.

Givoni (1981b) presented an experimental study on the performance of different design options for utilising flat roofs as passive cooling elements in buildings. Radiant cooling, evaporative cooling and combination of both were the physical processes utilised. The experimental facilities in this series consisted of small boxes, 55 x 55 x 50 cm, made of panels of expanded polystyrene (5 cm), internal plywood (2.5 cm) and external plywood (0.5 cm). The roof consisted of a concrete plate, 4.5 cm thick. Different treatments were applied to the roofs in order to utilise them as cooling elements for the spaces below (the rooms), as follows:

1. Heavy insulation, to minimise the roof effect; a standard for comparison.
2. Just white painting, to reflect solar radiation.
3. Shading, while providing permanent ventilation.
4. An exposed water-pond, to provide continuous evaporative cooling without eliminating solar and longwave radiation effects.
5. A shaded water-pond.
6. A water-pond with operable insulation, exposed at night and insulated during the day.
7. A thicker roof with operable insulation, exposed at night and insulated during the day.
8. Insulation over the roof, covered by pebbles and a water-pond.

The following table shows average minimum and maximum temperature of the roof surface and indoor air, together with outdoor conditions, for the period 4-12 October 1980. It should be noted that, ΔMax. and ΔAve are equal to Max. and Ave. temperatures of the standard case minus, Max. and Ave. temperatures of the various cases.

Table 2.2: Maximum and minimum temperatures with roof cooling (Givoni, 1981b).

Measurements	Roof Surface					Indoor Air				
Cooling system	Min.	Max.	Ave.	ΔMax.	ΔAve.	Min.	Max.	Ave.	ΔMax.	ΔAve.
Outdoor air (DBT)	15.3	26.9	21.1	-		15.3	26.9	21.1	-	
Wet bulb (WBT)	14.0	19.0	16.5	-		14.0	19.0	16.5	-	
Heavy insulation (standard)	19.5	24.6	22.0	-		19.4	25.6	22.5	-	
White painting	12.3	30.8	21.5	+6.2	-0.5	14.2	30.4	22.3	+4.8	-0.2
Shading the roof	15.9	28.8	22.3	+4.2	+0.3	16.6	29.1	22.8	+3.5	+0.3
Exposed water pond	14.0	27.1	20.5	+2.5	-1.5	14.6	27.8	21.2	+2.2	-1.3
Shaded water pond	15.2	19.9	17.5	-4.7	-4.5	15.6	21.9	18.7	-3.7	-3.8
Water pond with operable insulation	13.7	19.7	16.7	-4.9	-5.3	14.7	21.4	18.0	-4.2	-4.3
Concrete with operable insulation	14.0	21.9	17.6	-3.4	-4.4	15.2	22.4	18.8	-3.2	-3.7
Insulation, pebbles and a water pond	17.4	21.4	19.4	-3.2	-2.6	17.8	22.5	20.1	-3.1	-2.4

In 1982 Givoni presented a mathematical model of the expected indoor temperatures in buildings cooled by different types of shaded roof ponds. From experimental data, he found that the overall cooling effect of an exposed water roof pond is comparable to that of white painting the roof, and the best cooling effect was achieved with water pond with operable insulation. By shading a roof water pond the heating effect of solar radiation is eliminated as well as the cooling effect of the nocturnal radiation. The only cooling process is evaporation, which brings the temperature of the water close to the ambient wet bulb temperature. The water consumption of a shaded pond is much lower than that of an exposed one because the effect of the absorbed solar radiation on the evaporation rate is eliminated. Furthermore, the water temperature of a shaded pond is much lower as compared with an exposed one, so that the additional cooling potential is achieved together with conservation of water. In this study three shaded water ponds were tested:

a. A roof pond with fixed shading above it (uninsulated).

b. A water pond shaded by pebbles, with insulation laid directly over the roof.

c. A water pond like (b) but with the insulation raised above the roof surface by a shallow pebble layer (about 1 cm).

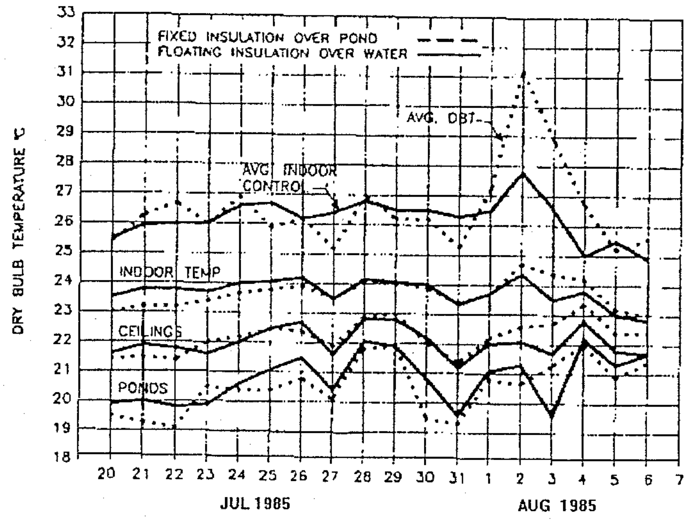
The results show the relationship between the computed indoor temperatures and the measured ones for the three types of shaded roof ponds mentioned above.

Performance of two test rooms, cooled by two different types of roof ponds, was investigated by Givoni in 1987. The ponds had two different means for preventing solar heating and for providing winter insulation:

i. An insulated fixed shade over the pond, with wind flow between the insulation and the water;
ii. A pond with floating insulation and with water circulation at night over the insulation.

The measured water, ceiling and indoor air temperatures, and the indoor air of a control test room, are shown in Figure 2.11.

The thermal performance of the two types of ponds was very similar, in spite of the basic differences in their design details and cooling processes. In the pond with fixed insulation and continuous evaporation, the cooling is produced only by evaporation which takes place day and night and consumes a large quantity of water. In the pond with floating insulation, the cooling effect is produced only at night, mainly by outgoing longwave radiation, and water consumption by evaporation is very small.



Fixed insulation over pond - - - - - - -

Floating insulation over water ————

Figure 2.11: Measured temperatures of the indoor air, ceiling and water, in two cooled by roof ponds (Givoni, 1991).

Furthermore, the pond with fixed insulation requires the construction of two roofs; the structural roof supporting the pond and a second, insulated roof, to shade the pond and to provide insulation during the winter. The space between the two roofs should be high enough to enable maintenance and repairs. On the other hand, the pond with floating insulation requires only the structural roof to support the pond. When repair is needed the insulating panels can be removed. It seems that this type is the preferable one, considering the simplicity of construction, the lower cost and the very significant conservation of water.

Natural cooling by thatched roofs in regions with summer rains, is an other subject which has been discussed by Givoni (1991). He explained that, Kimura and Yamazaki (1982) reported on the performance of thatched traditional roofs in Japan, where summer rains are frequent. When rain fell during the night, the thatched roof absorbed water and on sunny days evaporated it during the daytime at a rate which was enough to keep the indoor temperature at about 25^{o}C when the outdoor maximum was about 33^{o}C.

Other roof systems have been described by Crowther and Melzer (1979), Givoni (1977), and Hand (1980). One important advantage of most roof dissipater systems is that they may easily be adapted to provide solar heating. Heat loss rates for roof dissipaters have been discussed by Clark and others (1983) and Haines, Haves and Vollonk (1981). A validated sizing method for " Skytherm" roof pond cooling systems has been presented by Fleischhacker, Bentley and Clark (1982).

2.3.1.3.2 Water Pond on the Ground

Instead of having a pond over the roof it is possible to have the pond on the ground, next to the building, with a much larger depth of water. It is possible to maintain the water temperature near, or even below, the average diurnal WBT (wet bulb temperature), by fine spraying of the water over the insulation during the nights. Air tubes can be installed in the pond. When warmer air from the building's interior is circulated during the daytime hours through the tubes it is cooled by heat transfer to the water. The pond thus acts as a heat sink for the building with heat transfer by forced convection, while the cold is produced by natural energy sources (Givoni, 1991).

2.3.1.3.3 Applicability of Indirect Evaporative Cooling by Roof Ponds

The water temperature of a shaded roof pond, and that of the ceiling, follow approximately the average ambient WBT. The lowered ceiling temperature lowers the average indoor radiant temperature. For these reasons it is possible to apply roof pond cooling in places where the maximum WBT and DBT are higher than the applicability limits of direct evaporative cooling. The suggested maximum WBT for application of indirect evaporative cooling is 25^{o}C and the maximum DBT is 46^{o}C. In buildings cooled by a roof pond the indoor air speed is very low. If the indoor temperature is too high for comfort with still air, a higher indoor air speed can be provided by interior fans, thus improving the thermal comfort of the occupants.

Roof pond cooling can be applied only to a single-story building or to the top floor of a multi-story building (Givoni, 1991).

2.3.2 Convective Cooling

Convective cooling is applicable mainly in arid and desert regions, which have a large diurnal temperature range (above about 15 K) and where the night minimum temperature in summer is below about $20^{o}C$. In such regions it is possible to store the coolness of the night air in the structural mass of the building. The flow of outdoor air at night through the building can be induced naturally by the wind (where wind speed at night at the building site is sufficient e. g., above 2-3 m/s) or mechanically by an exhaust fan. During the following day the cooled mass serves as a heat sink, maintaining the indoor temperatures well below the outdoor level, hopefully within the comfort range. This temperature reduction can be achieved only when the building is well insulated, with the insulation external to the structural mass, and is not ventilated by the hot outdoor air during the daytime hours (Givoni, 1991).

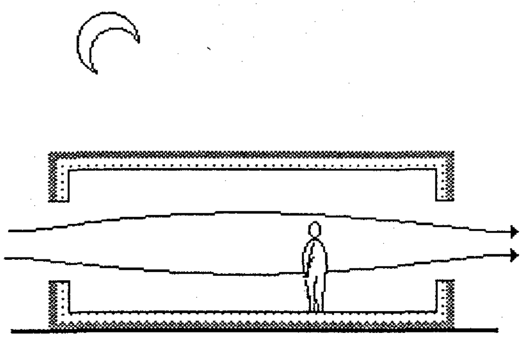


Figure 2.13: Cooling the mass of the building by night ventilation (based on Lechner, 1991).

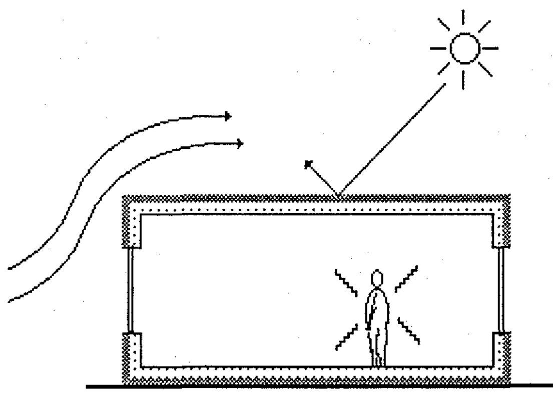


Figure 2.12: The convectively cooled mass acts as a heat sink during the day.

Alternatively it is possible to store night coolness by blowing outdoor air through a rock bed. The surface for the heat transfer between the air and the storage mass provided by the rock is much larger than that provided by the structural mass. As a result, the temperature of the rock in the morning can be lower than that of the building structure (Givoni, 1981a).

2.3.2.1 Traditional Convective Cooling Systems

Wind catchers or wind towers can be found in hot climate areas ranging from Pakistan through the Gulf States to Egypt and North Africa, and although the form and details may vary from region to region, the basic principle of catching unobstructed higher level breezes, remains the same. In some place the catchers are unidirectional and oriented to catch the favourable breezes, while in other place pivoted scoops and multi directional towers utilise winds from any direction (Konya, 1980). Wind towers or 'Baud-Geers', have been used for centuries in the hot arid regions of Iran and neighbouring countries to provide ventilation and natural cooling (Bahadori, 1978). Design of this system has been based primarily on the architect's personal experience; no published design criteria are available (Bahadori, 1981). The wind catcher is visually similar to a chimney with one end in the basement, running through the other floors with window openings at each floor. The upper part of the tower is divided into several air passages. The wind catcher functions by changing the temperature and thus the density of the air around it. This difference creates a draft, pulling air up or down the tower. Doors in the lower part of the tower open into the basement or into the main floor. The flow of air can be controlled by the opening or closing the doors from inside the rooms (Bahadori, 1978).

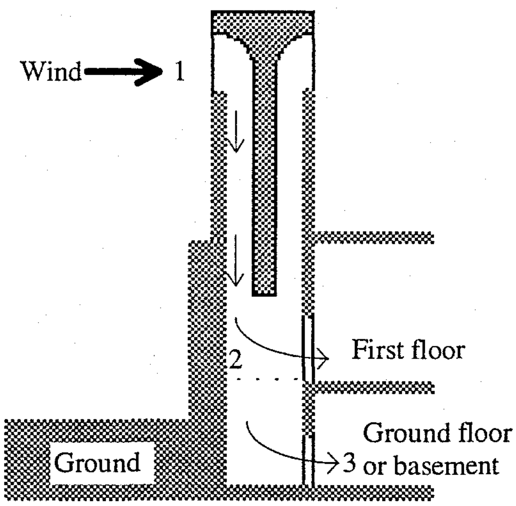


Figure 2.14: Creation of draft by a wind tower (based on Bahadori, 1981).

Wind towers can be employed in conjunction with curved roofs, which are another type of traditional convective cooling systems. The hot air that gathers under a curved roof is well above the living area of the room. In this way the room is kept more comfortable, and heat transfer from the roof to the room is limited because a high temperature is maintained next to the roof.

A curved roof is most effective when it incorporates an air vent. The operation of an air vent depends on the fact that when air flows over a cylindrical or spherical object, the velocity at the apex of the object increases; consequently the pressure at the apex decreases. If there is a hole at the apex of a domed or cylindrical roof, the difference in pressure induces the hot air under the roof to flow out through the vent.

An air vent is usually protected by a small cap in which there are openings that direct the wind across the vent. Since the functioning of the vent depends on air flowing over a curved surface, roofs with vents are oriented to present the maximum curve to the wind. In areas where the wind is a prevailing one cylindrical roofs are built with the axis of the cylinder perpendicular to the wind direction; in areas where the winds blow in all directions domed roofs are employed. Air vents are usually placed over the living rooms of buildings (Bahadori, 1978).

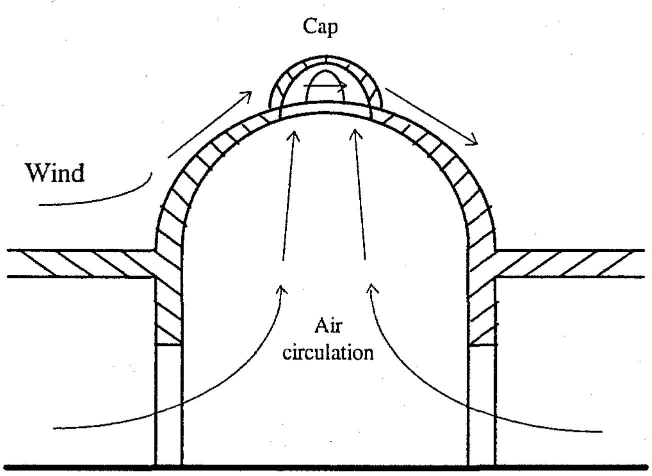


Figure 2.15: A dome roof with air circulation patterns over and under it (based on Bahadori, 1978).

2.3.2.2 System Designed for Short Term Storage

These systems may make use of the building mass, rocks and/or water as their storage material, and they make use of the night air as the main source of cooling.

2.3.2.2.1 Building Mass as the Storage Medium

When the building is insulated on the outside, nearly all of its mass may be used for the storage of coolness during the night. Night air has to be circulated over the walls, floor, ceiling, furniture, etc. to remove the heat transferred to them during the day, and cool them. This may be accomplished by providing a natural circulation of the ambient air through the building, for example, by opening the doors and windows and making use of the stack or wind effects.

Because there is an upper limit to the air circulation velocities in the building, the rate of heat transfer between the air and the building mass is limited. To overcome this obstacle, and hence to increase the amount of coolness which may be stored in the building mass, one may blow the ambient air through the passages provided in the floors, ceilings, walls, etc. during the night. The room air may be circulated through the same passages during the day. The following figure shows a design where concrete cored slabs are used for the storage of coolness.

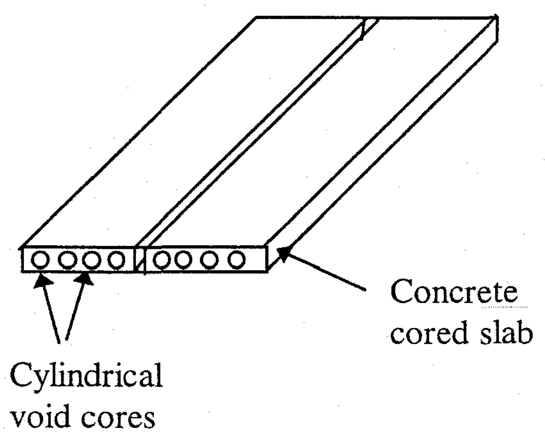


Figure 2.16: Concrete cored slab (based on Bahadori, 1981).

This design may be employed in two story as well as multi story buildings, in which case the slabs separate the two floors from each other. The heat transfer from each of the lower and upper rooms to the slab is by radiation, natural convection, if the room air is circulated through the slabs.

To have a more effective cooling of the slabs the night air may be cooled evaporatively or by thermal radiation losses to the sky, before passing through the passages.

Several design methods can be applied for improving the effectiveness of the structural mass for storage of night coolness in the summer, Givoni (1981a) described some of them:

- Increasing the surface area for that mass at the interface with the indoor space.
- Increasing the heat transfer rate by a higher air speed next to the materials of the structural mass.
- Increasing the thermal conductivity and heat capacity of the materials of the structural mass.

In considering the structural mass available for thermal storage it should be noted that building elements which are insulated from the indoor air, for example, floor covered by carpet should not be taken into account as storage of night coolness.

2.3.2.2.2 Rock Beds as the Storage Media

Two major drawbacks to the storage of night air coolness in the building mass are relatively small heat transfer area of the mass to be cooled and the low heat transfer coefficient between the air and this area. Provision of hollow cored slabs and circulation of high speed air through them somewhat overcomes these disadvantages. Rocks, with their large surface to volume ratio, provide a very effective storage, even though the heat transfer coefficient between them and the circulating air may still be low.

The following figure schematically shows a rock bed which may be employed for the storage of the night air coolness. During the night dampers A and B are at position 1. Night ambient air enters through the duct C, passes through the bed, and is exhausted back into the atmosphere. During the day dampers A and B are at position 2, and the room air enters the rock bed from duct F and the cooler air is supplied to the building through the duct E.

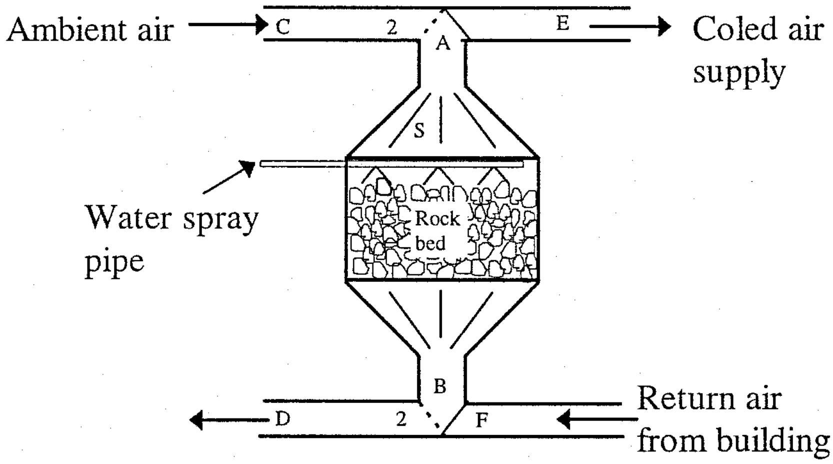


Figure 2.17: A rock bed storing the night air coolness (based on Bahadori, 1981).

2.3.2.2.3 Water as a Storage Media

When water is used to store the night air coolness it is best to utilise the evaporative cooling effects as well as the thermal radiation losses to the sky.

Figure 2.18 shows a completely passive design where the water stored in a water wall is cooled by the ambient air through several heat pipes. The pipes contain a fluid, for example, a freon, which can be evaporated and condensed at a constant temperature. The pipes are in good thermal contact with the water and are equipped with fins for a better heat transfer with the ambient air. When the water temperature is above the ambient air temperature, the liquid freon, cumulated at the base attached to the water wall, evaporates, travels the length of the pipe, and finally condenses at the end exposed to the ambient air. The condensate returns by gravity to the lowest point in the heat pipe. When the ambient air is at a higher temperature than the water in the water wall no evaporation

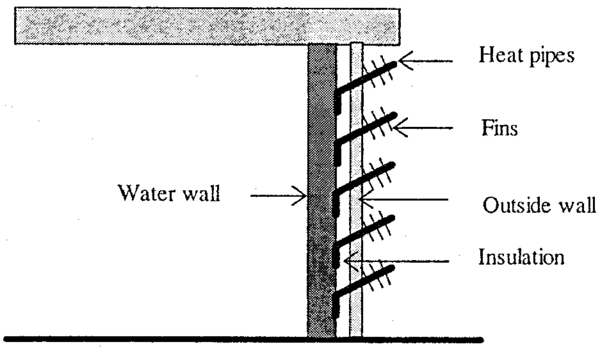


Figure 2.18: Water storage of night air coolness through heat pipe action (based on Bahadori, 1981).

and condensation take place. In this case, the amount of heat transferred through the heat pipes is negligible. To prevent an excessive heat loss from the water through the heat pipes, the water wall has to be drained in winter.

A design in which the heat losses from the water wall to the ambient air may be greatly reduced in winter is shown in Figure 2.19.

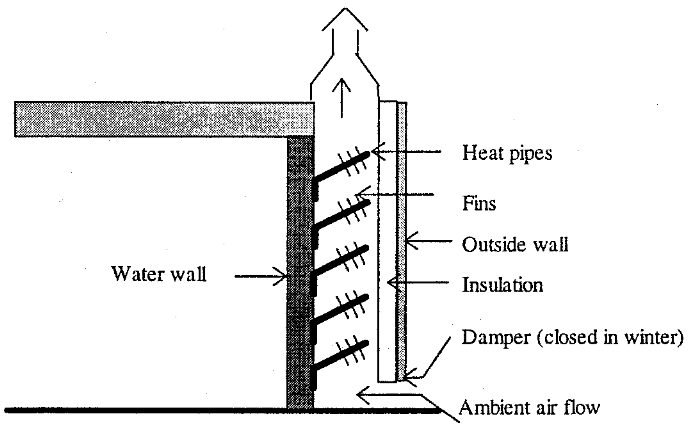


Figure 2.19: Storage of night air coolness in water through heat pipes (based on Bahadori, 1981).

The air flow over the heat pipes may be by a fan or by the chimney or wind effects. The air circulation by the wind may be accomplished by the proper choice of a ventilator cap. The damper at the bottom of the stack should be closed in winter (Bahadori, 1981).

2.3.2.2.4 Phase Changing Materials as Storage Media

A eutectic of many salt hydrates may be found to melt isothermally at a desired temperature. Such salt hydrates may be used to fill shallow trays or tubes stacked in an air duct. By blowing the cool night air through these containers and solidifying the phase changing material one can store the night air coolness in them. During the day the room may be cooled by circulating its air through the duct containing the phase changing material. The ambient air entering the duct at night may be further cooled by thermal radiation losses to the sky or by water evaporation, before reaching the containers.

2.3.2.2.5 Combination of Several Storage Media

It is possible to employ more than one material to store the night air coolness. The same method may be employed in active cooling systems. To reduce the peak cooling load of a building one may store the night air coolness in one or more of the materials mentioned before. The idea of coolness storage may be extended to completely active cooling systems in order to reduce the peak cooling load or reduce the energy consumption during the peak hours. In such cases the air conditioning system may be operated to cool the building structure or other masses.

2.3.2.2.6 Combination of Several Cooling Sources

The best passive or low energy cooling systems are those designs which use more than just one of the cooling sources described earlier. The water in the pool is cooled primarily by the cool ambient air and by evaporation. The cooled water flows down through a downcomer to the storage columns located at suitable places in the room. When the water in these columns is heated it rises and flows back to the pool. The flow of water is maintained by the thermosiphoning action.

2.3.2.3 Long Term Storage of Coolness

Many places requiring summer cooling have rather cold winters. In these areas the storage of the winter coolness for summer use seems to be an important and a natural proposition. The main source of coolness is, therefore, the winter ambient air, and the materials in which the coolness may be stored are water, in both liquid and solid phases, rocks and earth.

2.3.2.3.1 Storage of Coolness in Water

Because of the large amount of energy which has to be stored, and in order to reduce the heat gains during the storage period, and to utilise the thermal storage capacity of the ground, the winter cold water is stored in underground tanks, cisterns, reservoirs or aquifers. The following figure shows a two-well aquifer system. During winter, cold water is injected into the well A and at the same time an equal amount of warm water is removed from the well B, and is brought in contact with the ambient air (for example in a lake or a pond). This process can be continued until the aquifer is charged with cold water. In summer, the cold water may be removed from well A and the used water injected into the well B.

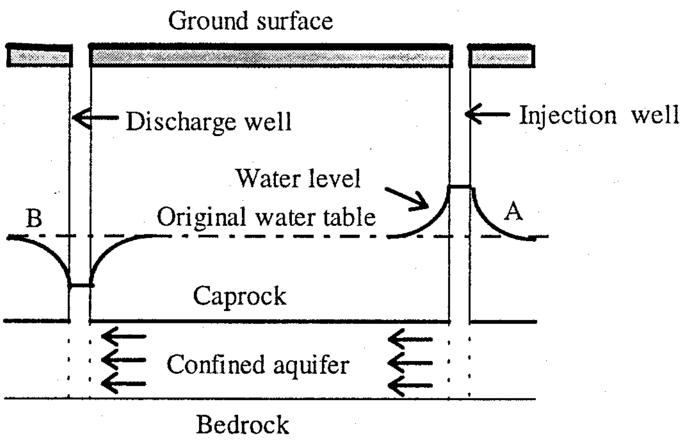


Figure 2.20: A two-well aquifer system (based on Bahadori, 1981).

2.3.2.3.2 Storage of Coolness in the Form of Ice

The storage of ice from winter for summer use is an old technology, and was utilised before the advent of mechanical refrigeration. To utilise the coolness stored in the ice one may submerge pipes in the tank at appropriate intervals. Antifreeze-added water may be circulated through these pipes. One problem to the passive ice making technique just described is the formation of ice around the heat pipes and reduction of rate of heat transfer between the water in the tank and the liquid in the heat pipe. This problem may be reduced by using flexing heat pipes, where the ice formed around the heat pipes is broken away, or by increasing the number of heat pipes to be employed.

2.3.2.3.3 Storage of Coolness in Rocks

The hollow spaces formed or found in the ground due to trenching, strip mining, rock quarrying, underground blasting and caves may be filled with crushed rocks and used for long term energy storage. The winter cold air is blown through the rock bed through air manifolds, and the warm air, leaving the bed, which may be at a temperature of about 25^{o}C, may be exhausted into the atmosphere, or be used for winter heating of buildings (with some reheating of the air), in greenhouses, or in any other application. During the summer, the ambient or the building air may be circulated through the rock bed, where it is both cooled and dehumidified.

2.3.2.3.4 Storage of Coolness in the Ground

The seasonal storage of winter coolness in the ground is similar to the seasonal or long term storage of solar energy in the ground. Basically, pipes are laid, horizontally or vertically, in the ground. During winter, cold water is withdrawn from a lake, pond, etc., is circulated through these pipes, and finally discharged to the same reservoir. In summer, the water to be cooled is circulated through the same pipes.

Another method to store the winter coolness in the ground is to use heat pipes. The ground space considered for the storage may be in the form of a cylinder, hemisphere, a cube, or any other shape. The heat pipes as well as the supply and return pipes, can be driven into the ground, thus eliminating the need for excavating the ground for piping. The supply and return pipes may be connected to their headers at a level about a meter below the ground level, and carry freeze protected water, or another suitable liquid. With the heat pipe action the rate of heat transfer from the ground to the ambient air can be very high, whereas the amount of heat transferred from the air to the ground is very low. In this way, the ground can be cooled very effectively and the coolness stored in it for a long time. To reduce the ground's heat gain by conduction, a layer of insulation may be placed between the ground level and the supply and return pipe headers. During the summer, a pump circulates the cooling fluid through the supply and return pipes, thus transferring heat to the cooled ground.

Another method is to provide air passages or ducts in the ground. the ambient air is circulated through these ducts in winter and cools the ground. the heated air may be exhausted back into the atmosphere, or it may be used in a building, a greenhouse, or in any other applications. During the summer, ambient air constituting the fresh air requirements of the building, or the building's air may be circulated through the ground, where it is both cooled and dehumidified.

2.3.2.4 Applicability of Convective Cooling

Nocturnal convective cooling would be the preferable strategy in regions where the diurnal temperature range in summer is large enough to enable significant reduction of the indoor air temperature below the outdoor maximum. From the climate aspect, nocturnal convective cooling as a building design strategy is preferable to comfort ventilation in regions where the day time temperatures in summer are above the upper limit of the comfort zone (with air speed of about 1.5 m/s), and it is applicable mainly in arid regions where the daytime temperature is between 30 and 36^{o}C and the night temperatures are below about 20^{o}C. In this situation, daytime ventilation is not desirable as it would raise the indoor temperature. The large diurnal temperature range, typical of arid regions, makes possible the reduction in the indoor daytime temperature to significantly below the outdoor level by night ventilation. However, in arid regions with daytime temperature above 36^{o}C, night ventilation would not maintain the indoor temperature at an acceptable level and other passive cooling systems should be considered during the too hot hours as supplements to convective cooling, such as evaporative cooling, earth cooling, or compression air conditioning. But even when an additional cooling system is provided, the design for and use of convective nocturnal cooling can significantly reduce the length of the periods and duration of the time when the operation of the additional cooling systems will be needed.

Some practical problems may limit the applicability of nocturnal convective cooling e.g., the need to open and close windows at prescribed times and issues of security, especially in public buildings (Givoni, 1991).

2.3.3 Radiant Cooling

Radiative cooling is a common phenomenon at the earth's surface. In fact, it is the only mechanism by which the earth can lose heat. Considering that the sun pours its energy on the earth at a rate of about 1.5 x 10^{19} kJ. (1.42 x 10^{19} Btu) per day, and that the average surface temperature is approximately constant over a number of years, it is obvious that a similar amount of energy per day must escape. Some of this abundant energy is of course, reflected back into space as visible light, and a small fraction is converted to chemical energy by photo-synthesis. But the largest part heats up the earth's surface, atmosphere, and oceans, and is ultimately emitted into space in the form of thermal infrared radiation.

The entire process is very complex and many paths can be followed by the energy flow originating from the sun and eventually escaping from the earth into space. Unequal warming of the land masses and oceans between the equatorial and polar regions drives the global weather system, but eventually all this agitation is degraded into thermal energy that radiates away from the earth's surface in the form of infrared radiation.

Fundamental physical principles ensure that each object emitting radiant energy also absorbs it. People radiate and absorb heat at the same time, usually at different rates. Buildings emit infrared energy as heat to the sky and pick it up, at different rates, from surrounding buildings and trees, clouds, and even the clear sky itself. Although humans are equipped with acute optical sensors in the visible region of the spectrum, we have no equivalent way of directly detecting currents in the sea of infrared radiation in which we are immersed. At most we feel these effects as a warm or cool sensation on the skin, but this non specific thermal feeling could equally arise from the convection of warm or cool air.

In effect, we have a sensory "blind spot" in the infra-red part of the electromagnetic spectrum that causes us to be unfamiliar with what is occurring in the world about us. The events taking place in this spectral region, of which we are generally unaware, are remarkably intense and of great importance in determining our thermal environment. On a hot summer day the noontime intensity of direct beam radiation from the sun is only about twice this value. On the same day the roof of a house could easily become heated by the sun to 65°C. Obviously infra-red heat transfer is not a negligible mechanism at the earth's surface. Indeed, it and the convective coupling between objects and the air are usually the dominant heat flow paths.

Increasing the air humidity slows the rate of radiative cooling through the atmosphere. As an example of the effects of humidity, it is well known that the day-to-night air temperature swing is much larger under arid conditions than it is when the humidity is high. The greater reduction of the arid night time air temperature is largely caused by the increased rate of net radiation away from objects under clear, dry atmospheric conditions.

Infrared radiation emitted from an object at the earth's surface can be absorbed and remitted many times by water droplets and atmospheric gases like carbon dioxide, water vapour, and ozone, before escaping into space (Martin, 1989).

All objects emit and absorb radiant energy, and an object will cool by radiation if the net flow is outward. At night the long wave infrared radiation from a clear sky is much less than from a building, and thus there is a net flow to the sky. Since the roof has the greatest exposure to the sky, it is the best location for a long wave radiator. Since only shiny metal surfaces are poor emitters, any non-metalic surface will be a good choice fir a long wave radiator. Painted metal (any colour) is especially good because the metal conducts heat quickly to the painted surface, which then readily emits the energy. Such a radiator on a clear night will cool as much as 8^{o}C below the cool night air. On humid nights the radiant cooling is less efficient but a temperature depression of about $3.9^{o}C$ is still possible. Clouds, on the other hand, almost completely block the radiant cooling effect (Lechner, 1991).

Direct radiant cooling, and indirect radiant cooling are two methods for this system.

2.3.3.1 Direct Radiant Cooling

Potentially the most efficient approach to radiant cooling is to make the roof itself the radiator. For example, an exposed concrete roof will rapidly lose heat by radiating to the night sky. The next day the cool mass of concrete can effectively cool a building by acting as a heat sink. The roof, however, must then be protected from the heat of the sun (Lechner, 1991).

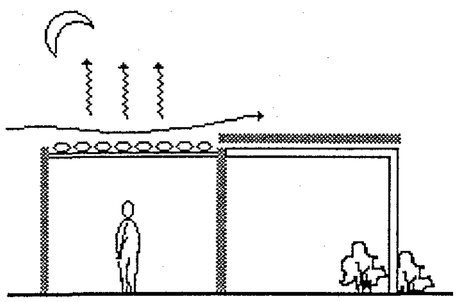


Figure 2.22: removing the insulation during the summer nights (based on Lechner, 1991).

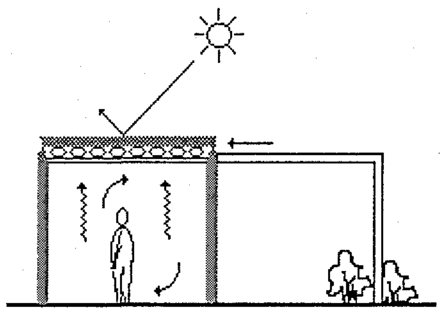


Figure 2.21: Summer day cooling by water bags (based on Lechner, 1991).

Hay (1978) has designed and built several buildings using this concept, except that he used plastic bags filled with water rather than concrete for the heat sink material. Another direct cooling strategy uses a lightweight radiator with movable insulation on the inside. With this system a painted sheet metal radiator, which is also the roof, covers movable insulation. At night this insulation is in the open position, and during the day the insulation is moved into the closed position to block the heat gain from the roof (Lechner, 1991).

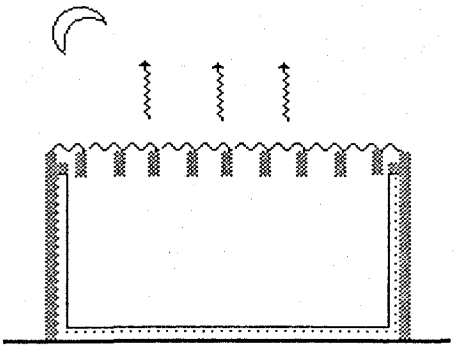


Figure 2.24: Radiative cooling the building by opening the movable insulation at night (based on Lechner, 1991).

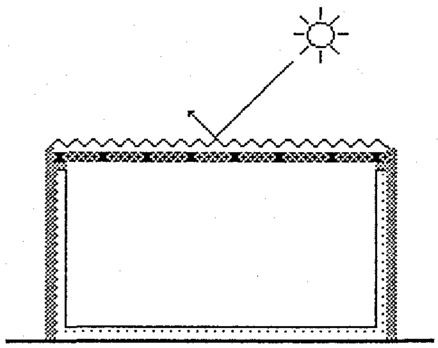


Figure 2.23: Closing the insulation during day time to keep the heat out (based on Lechner, 1991).

From the viewpoint of performance, this type of radiant cooling system (a high mass roof, either made of concrete or of a metallic roof with water bags, with movable insulation panels) can be the most efficient radiant cooling system because the mass of the roof is radiating at relatively high temperature, usually above the outdoor night air. Thus the cooling of the roof takes place also by convection, in addition to the radiant heat loss, and the total cooling rate of the mass is maximised. As the ceiling serves as a cooling panel the convective heat transfer from the interior space to the ceiling and then by conduction to the cooled mass is also maximised, in addition to the lowering of the indoor radiant temperature (Givoni, 1991).

Another direct cooling strategy uses a lightweight radiator with movable insulation on the inside. This avoids two of the problems associated with the above concept: heavy roof structure and a movable insulation system exposed to the weather. With this system a painted sheet metal radiator, which is also the roof, covers movable insulation. At night this insulation is in the open position so that heat from the building can migrate up and be emitted from the radiator. For the cooling effect to be useful during the day, sufficient mass must be present in the building to act as a heat sink. Also during the day the insulation is moved into the closed position to block the heat gain from the roof (Lechner 1991).

2.3.3.2 Indirect Radiant Cooling

The difficulty with movable insulation suggests the use of specialised radiators that use a heat transfer fluid. The painted metal radiator cools air at night, which is then blown into the building to cool the indoor mass.

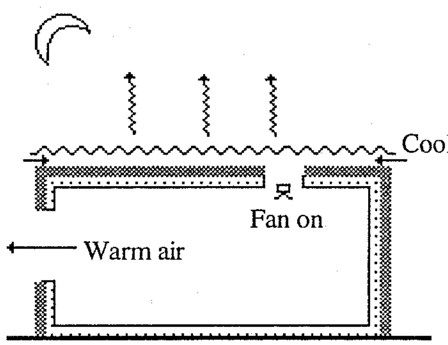


Figure 2.26: An example of indirect radiant cooling by blowing the cool air into the building at night (based on Lechner, 1991).

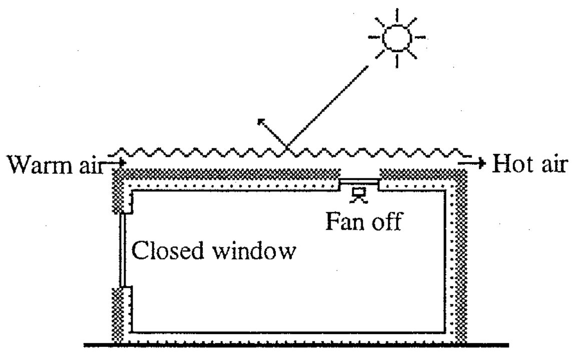


Figure 2.25: The cooled mass acts as a heat sink during the day (based on Lechner, 1991).

The next morning the fan is turned off, and the building is sealed. The cooled indoor mass now acts as a heat sink. The radiator is vented during the day to reduce the heat load to the building (Lechner, 1991).

2.3.3.3 Applicability of Radiant Cooling

The roof is the only element of a building with extensive exposure to the sky, unless the sky is blocked by surrounding higher buildings, and therefore it is the natural location for a nocturnal radiator, regardless of its type. When a high mass roof with operable insulation serves as the radiator, and the heat transfer to the space below is by conduction, radiant cooling is applicable only to single story buildings or to the upper floors of a multi story building. But even with a metallic radiator, and fan driven airflow which can be directed to any space, the radiator's size needed to cool a given space might be close to the floor area of that space, so the limit of application to a single floor is still a real one.

High mass roofs with operable insulation, either of concrete or with roof ponds, provide the functions of cold collection and of storage in one element and, therefore, these types of radiant cooling are effective in providing daytime cooling, practically at any region with low cloudiness at night, regardless of the air humidity.

A radiant cooling system which utilises a metallic radiator, with fan driven airflow underneath, can cool the night air below the ambient level. The main climatic requirement is again low cloudiness during the nights. Humidity is less important as long as the nights are clear. In arid regions a temperature drop of about 3-5 K, depending on the flow rate of the air exiting from the radiator, can be expected. In humid regions with a clear sky the expected temperature drop would be about 2-3 K.

In humid regions, moisture would be condensed out of the air while flowing under the radiator, thus also lowering the indoor humidity in addition to lowering the temperature. This cooled and partly dehumidified air can be used directly to cool a building during the evening and night hours in regions where the outdoor air is too warm for providing comfort by direct ventilation. If the building contains enough thermal storage also the daytime temperatures will be lowered (Givoni, 1991).

2.3.4 Earth Cooling

The temperature of the soil near the surface follows the air temperature. Specifically, the average monthly air temperature and the soil temperature near the surface are about the same (Lechner, 1991). However, the natural underground earth temperature at a depth of several metres is almost constant and close to the average annual air temperature. Therefore in summer it is always below the average ambient temperature and especially below the day time air temperature.

2.3.4.1 Methods of Soil Cooling

It is possible by special means to lower the earth temperature well below the natural temperature characterising a given location. At present two methods have been tried successfully by Givoni (1987) to lower the earth temperature by shading it while enabling water evaporation from the surface:

1. covering the soil with a layer of mulch, for example gravel or wood chips, at least 10 cm thick and, in regions with dry summers, irrigating it;

2. raising the building off the ground and facilitating evaporation from the shaded soil surface of water provided either by irrigation or by summer rains.

Once the soil's surface temperature in summer is lowered, its annual average temperature, as well as the temperatures of the layers below the surface, are also lowered. Experiments have demonstrated that it is possible to lower the earth surface temperature by about 8-10 K below the summer temperature of exposed soil. The following figures show seasonal and diurnal temperature patterns of cooled soil in Tallahassee, Florida. These figures demonstrate that the difference between the outdoor maximum air temperature and the cooled earth temperature in mid summer can be up to about 14-16 K in arid regions and up to 10-12 K in some hot humid regions. With such a temperature difference the soil can provide a heat sink for a building, especially in cooling the ventilation air.

Figure 2.27 demonstrates that even in a hot humid region such as Florida it is possible to cool the soil in summer to a level below the outdoor minimum temperature. The difference between the outdoor maximum and the cooled soil is largest during periods of ambient heat waves (eg., June 24-29 and July 17-18).

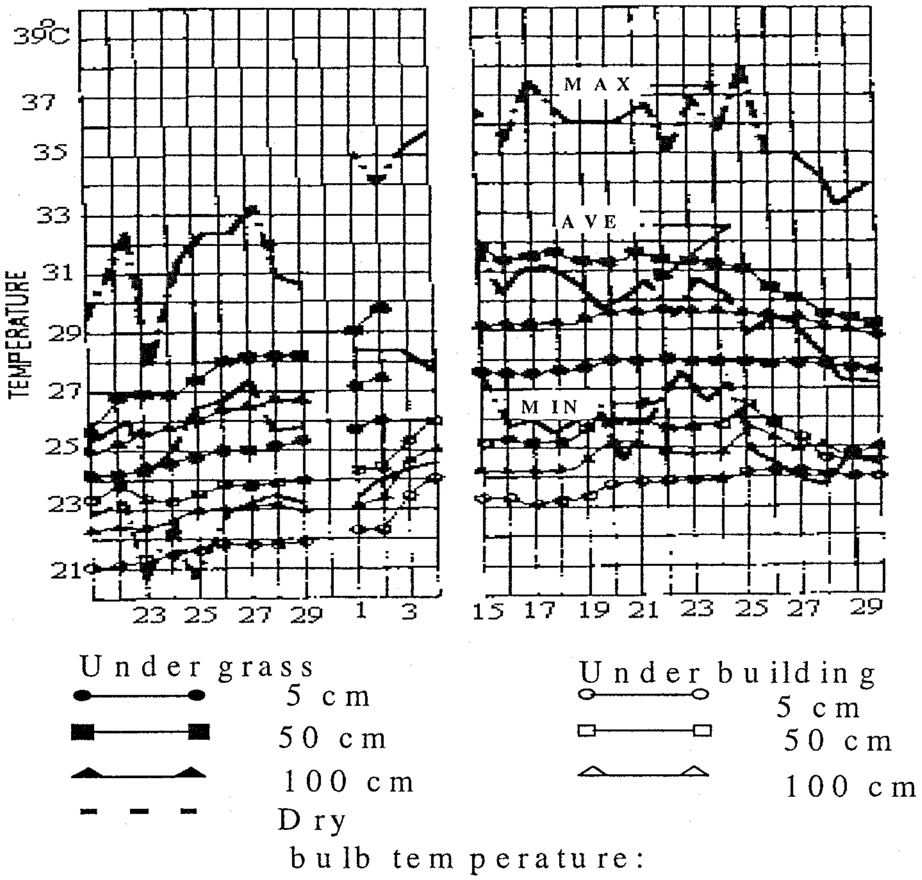


Figure 2.27: Seasonal (summer) temperature patterns of two soil areas in Tallahassee, Florida, at different depths (Givoni, 1991).

2.3.4.2 Direct Earth Cooling

When the underground soil temperature is cool enough it is possible to utilise it for cooling buildings by several methods. In the case of earth integrated buildings, in which the walls are bermed and the roof covered by earth, cooling of the earth mass adjacent to the building provides a direct passive cooling effect to the building. This approach would be most suitable in hot dry regions with mild winters. In such places this direct conductive cooling of the building will be very effective. Concerning the water availability issue, it should be noted that by effectively shading the soil surface with a layer that is thermally insulative but permeable to vapour (eg., wood bark mulching), effective soil cooling can be achieved with much lower water consumption than that needed to water a lawn, for instance.

In hot regions with cold winters, the direct conductive coupling of the indoor space with the surrounding soil through highly conductive walls and roof, although effective in summer, may be undesirable since it will cause a high rate of heat loss in winter (Givoni, 1991).

2.3.4.3 Indirect Earth Cooling

Another approach, an active one, is to insulate the building and to install air pipes in the soil, circulating through them the air from the building or the ventilation air. The cooler earth mass serves then as a heat sink to cool the air which is introduced into the building. Heat transfer to the cool soil can be by means of an array of pipes, usually of plastics, eg., of PVC. The air circulation may be through a closed circuit, or it may be outdoor air used for ventilation. Circulating the indoor air through air tubes embedded in the cool soil can keep the indoor temperature about 10 K below the outdoor average maximum air temperature. During a heat wave with abnormal high outdoor temperatures, the cooling effect of the tubes, embedded in the more stable earth, can be even greater (Givoni, 1991).

Sodha *et al.* (1985) have measured the performance of a similar system near Dehli, in India. In their study, air was blown through a section of an existing tunnel several meters below the ground level. The length of the test section was 80 m and its cross sectional area 0.528 m^2. Two fans, each of 500 watts, drew air through an inlet duct and out through an outlet duct, at a rate of 3.1 kg/s mass flow and speed of 4.89 m/s. The summer tests lasted 10 consecutive days in June 1983. The outdoor maximum temperatures during this period ranged from 36.5 to 42.5 ^{o}C and the minimum from 23.4 to 30.3 ^{o}C. The range of the outlet maximum temperatures during the same period was 25.7-28.2 ^{o}C and the outlet minimum 23.1-25.2 ^{o}C. The average daily cooling obtained during the test period was 512 kWh/day.

2.3.4.4 Applicability of Earth Cooling

In regions with a temperate climate, the natural temperature of the soil in summer at a depth of 2-3 m may be low enough to serve as a cooling source. In hot regions, on the other hand, the natural temperature of the soil in summer is usually too high to serve as a cooling source. However, it is possible by very simple means to lower the earth temperature well below the natural temperature characteristics of a given location.

In regions with hot summers but cold winters, the direct conductive coupling of the indoor space with the surrounding soil may cause a high rate of heat loss in winter. In these regions, indirect active coupling of the building to the cool soil, by circulating air through pipes embedded in the soil, can provide the required cooling (Givoni, 1991).

2.4 PROPOSED PASSIVE COOLING SYSTEM

Based on the knowledge arising from above study, for a semi-arid climate like Tehran the summer night temperature is not cool enough (below $20^{\circ}C$) to store the coolness of the night air in the structural mass of the building. Thus, convective and radiant cooling alone are not the effective methods of lowering indoor air temperature in such climate. In the big cities, multi storey buildings can not be covered by soil to take the benefit from the earth cooling.

Hence, evaporative cooling is the most appropriate way of cooling space for cities in such climates. This method of cooling can lower the indoor air temperature while increases the humidity level which is desirable because of uncomfortably low outdoor relative humidity.

This comprehensive literature survey has showed that roof ponds are the most frequently proposed indirect evaporative cooling systems. Although there are some benefits with these systems, there are also disadvantages, which reduce their attraction:

- The high cost of the double roof structure and water proofing.
- Functional limitation to single-story buildings, or the top floor of multi story buildings.
- Labor and difficulties to move insulators in some cases.

Evaporative cooling pads in windows and evaporative cooling towers are the other passive evaporative cooling methods. Using evaporative pad in the window, blocks the view and light through the window, and reduce the air flow inside the building. The cost of building a cooling tower and in some cases solar chimney which may encourage unwanted bacteria and insects particularly in wet cases, are major disadvantages of these systems.

Each passive evaporative cooling system has disadvantages which reduce its functions. There have been many different passive cooling systems proposed but never built. For instance, water applied as a cooling source, can also be a design element inside of a building. Therefore, it is proposed that "the use of water passing over vertical guides in the form of a screen will provide passive evaporative cooling combined with natural ventilation for summer comfort in dry climates. On the basis of this idea, the natural ventilation through the building, in contact with exposed water, will reduce indoor air temperature.

This study presents a possible method of cooling apartments in low rise multi-storey buildings by a simple water cascade associated with openings and balconies of the individual units. Figure 2.28 shows a system which includes evaporation of an exposed water film with the help of natural ventilation. Water is allowed to fall vertically over the elements such as nylon lines or other filaments thus exposing the maximum surface area of the water to the passing air flow. The arrangement of the lines is such that the water forms a curtain through which the air flows horizontally.

The water is raised to an upper level by a small pump from the reservoir in the bottom of the system and then allowed to flow down the lines and back to the reservoir. The air passing through the system is cooled and humidified. If contact between water and air is sufficient to bring the water and the exit air into equilibrium, the air leaving the system will become saturated at a temperature close to the wet bulb temperature of the exiting air. Since the water evaporated into the air is lost from the system, make-up water is supplied to the reservoir.

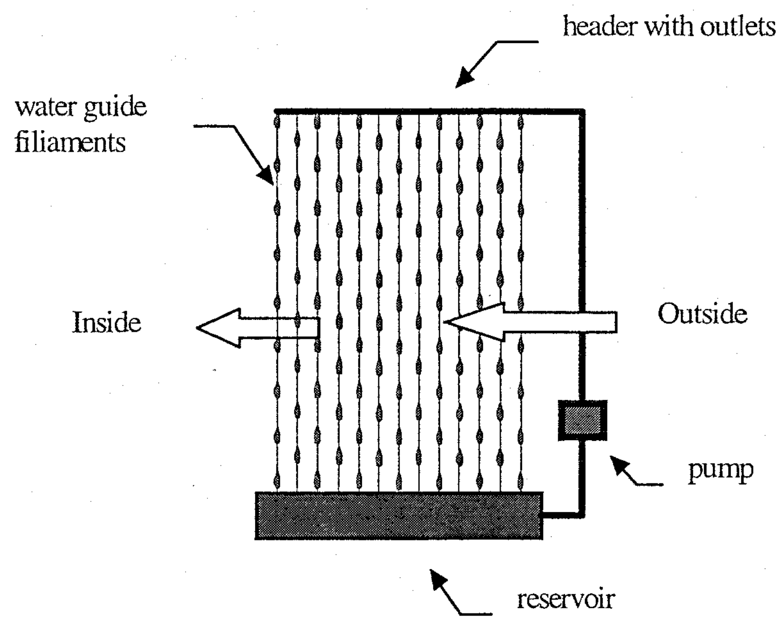


Figure 2.28: Schematic diagram showing evaporative cooling system.

Figures 2.29 to 2.32 give an impression of the appearance of the proposed cooling system.

Figures 2.33, 2.34 and 2.35 show examples of applications as works of art (all utilise water running down the outside of a fine diameter guide).

As this cooling system works only with natural ventilation, the following chapter will investigate the theoretical aspects of calculating natural ventilation rate and strategies.

Figure 2.29: Impression of the appearance of the proposed cooling system on the building facade.

Figure 2.30: Impression of the appearance of the proposed cooling system on the building facade.

Figure 2.31: Impression of the appearance of the proposed cooling system from inside.

Figure 2.32: Impression of the appearance of the proposed cooling system from inside.

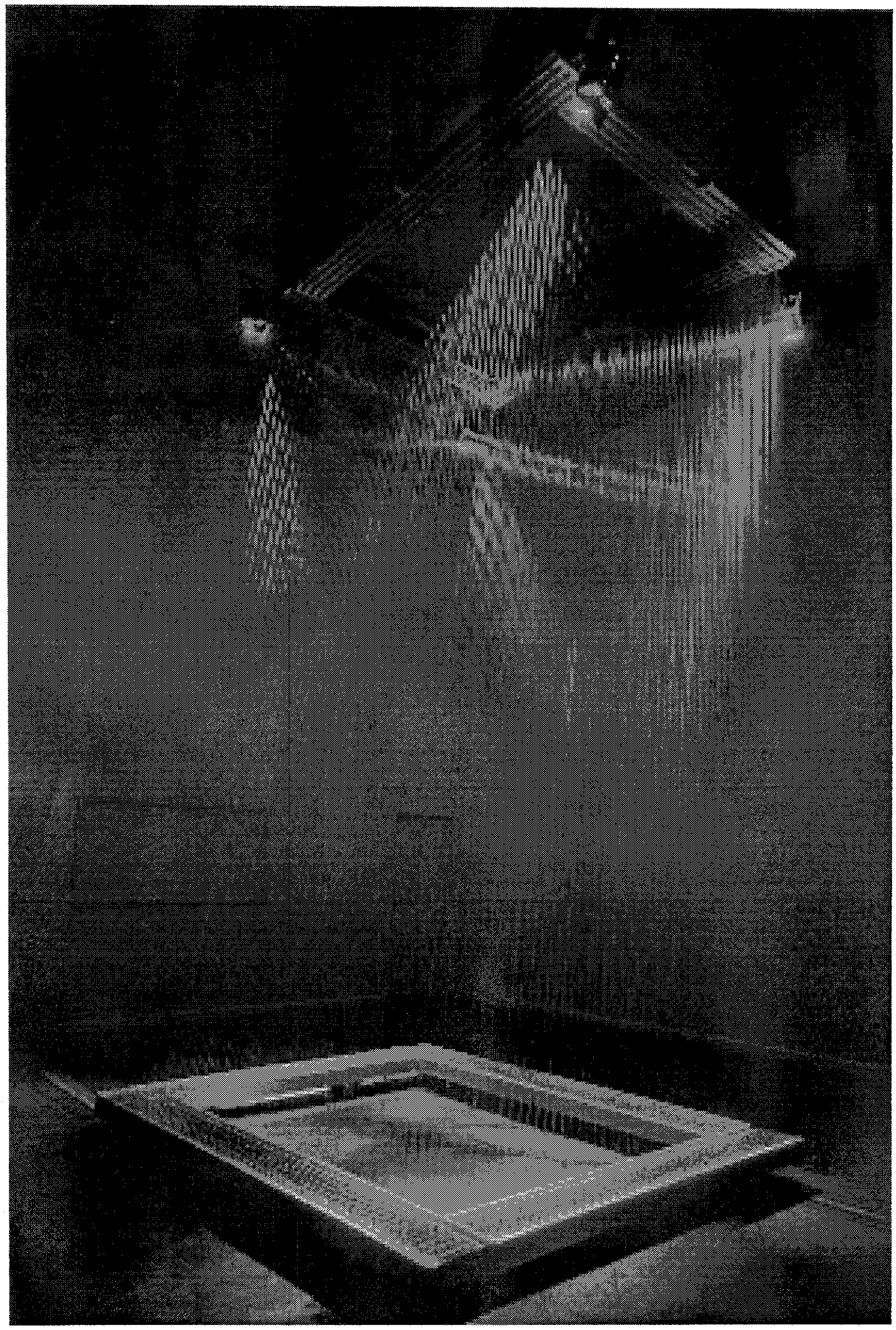


Figure 2.33: Water work by Jennifer Turpin, Sydney College of the Arts, Australia, 1991.

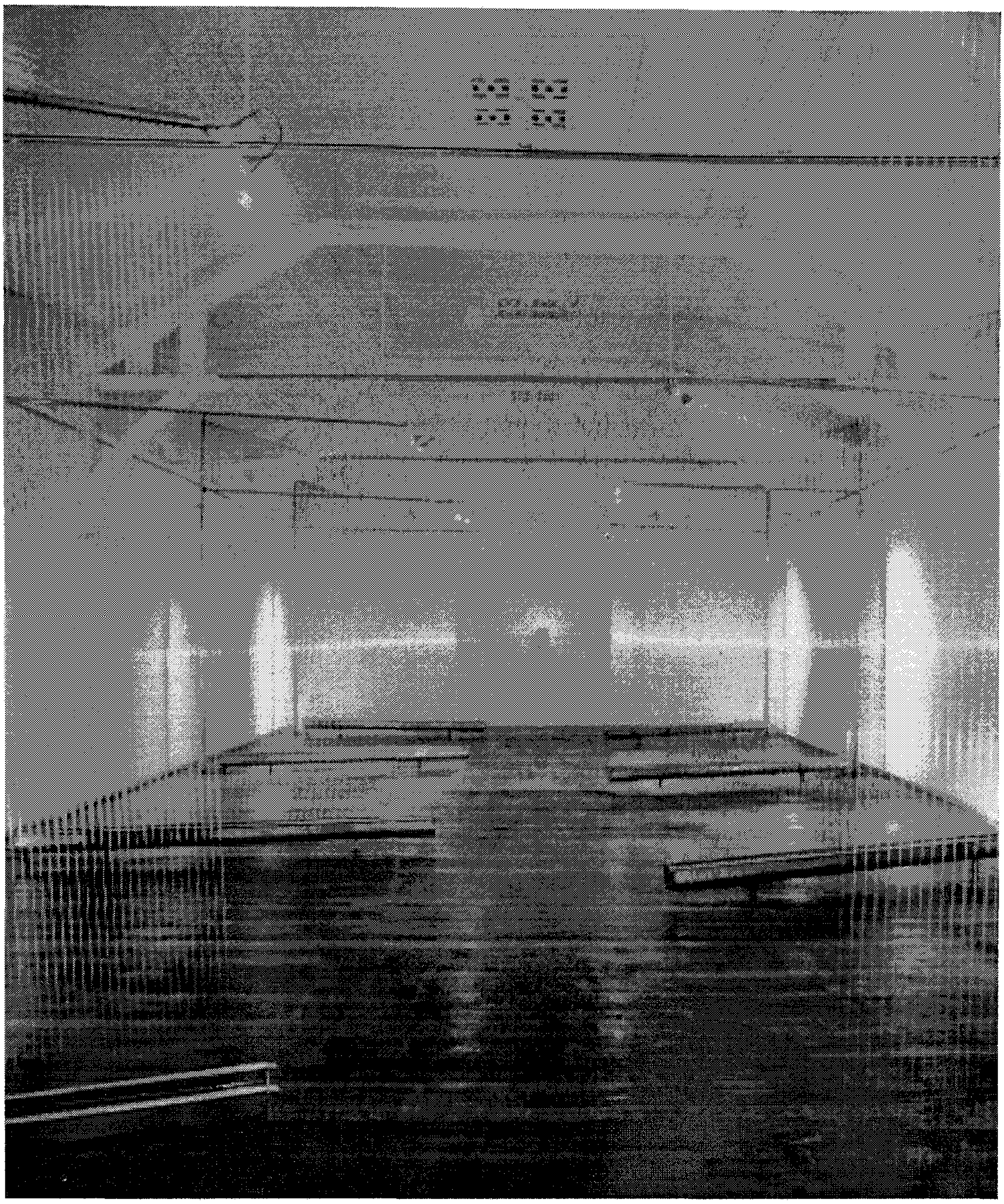


Figure 2.34: Water work by Jennifer Turpin, Annandale Galleries, Australia, 1992.

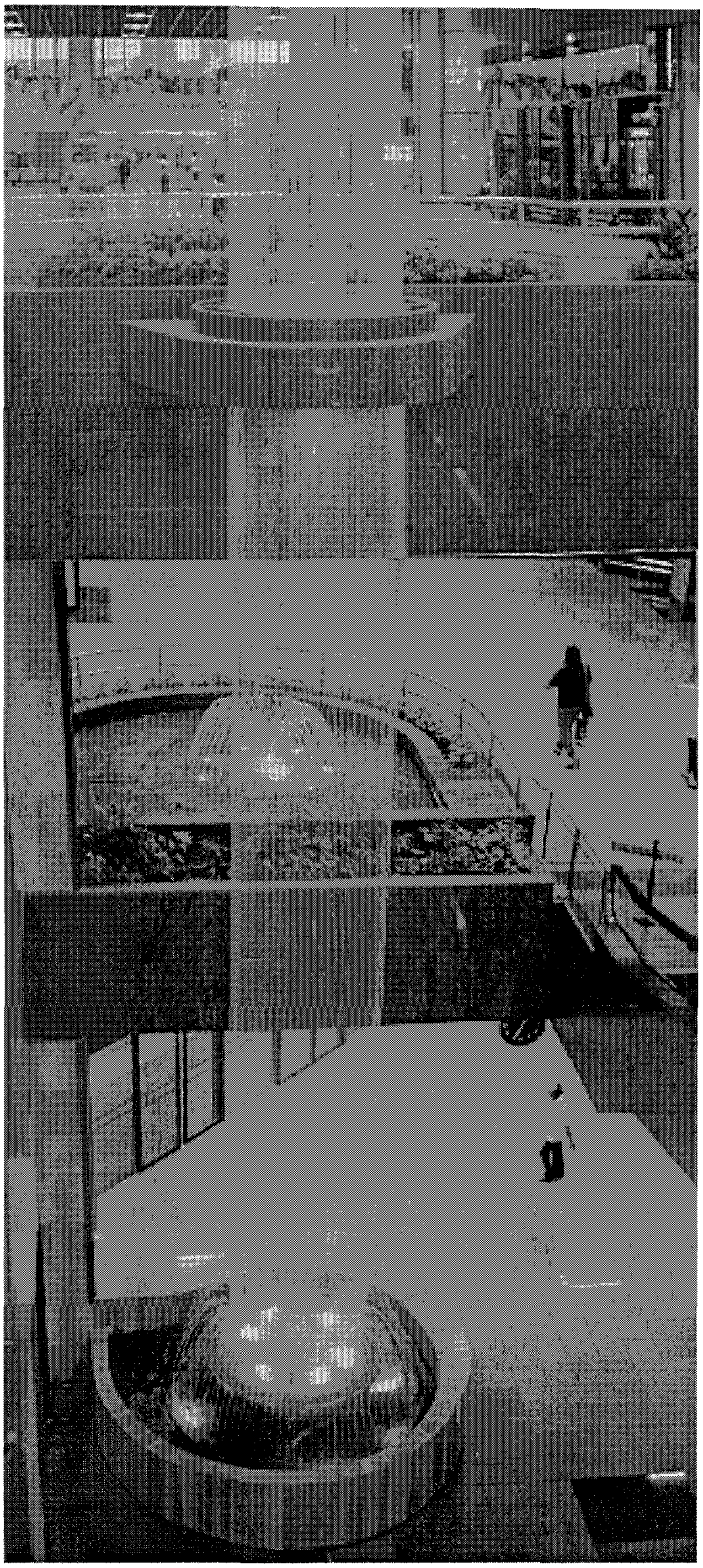


Figure 2.35: Water work in Singapore airport.

CHAPTER 3

NATURAL VENTILATION PRINCIPLES AND DESIGN IMPLICATIONS

3.1 INTRODUCTION

Natural ventilation occurs because of pressure differences acting on inlets and outlets of a space. This pressure difference can be created by wind or by a thermal chimney (stack ventilation). The pressure difference caused by winds may be steady (as in cross ventilation) or unsteady (as in turbulent ventilation). Steady wind-driven ventilation, i.e., cross ventilation, is usually the strongest mechanism and is produced when a prevailing wind direction creates distinct positive and negative (suction) pressures at the inlets and outlets of a volume.

Unsteady pressure differences also may be created by wind, such as changing pressure patterns over a windward wall with two widely spaced windows on the same wall. The fluctuating wind directions, typical in suburban or other rough terrain, create unsteady pressure fluctuations that can generate significant ventilation.

Another type of natural ventilation arises in rooms with only one window. Here minimal ventilation is created as some air enters the room at one time and a few seconds later some air exits because of the fluctuating static pressure of the wind. This pattern creates very minimal ventilation and will not be discussed further. The theoretical analysis of this type of "turbulent diffusion" ventilation is explained by Warren and Parkins (1984).

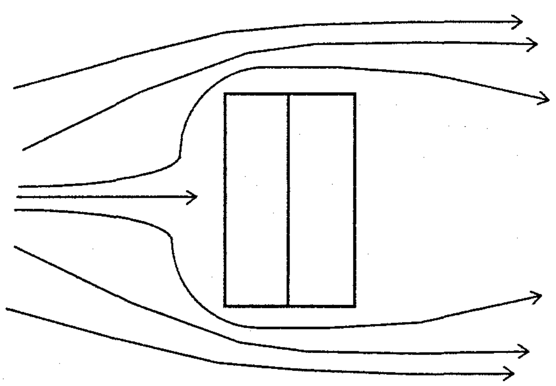
3.1.1 Air Flow Around Buildings

Extensive knowledge exists concerning mean wind loads on buildings. Newberry (1974), MacDonald (1975) and Cermak (1976) have presented comprehensive reviews of the subject. The dynamic loading of buildings under high gusts or tornados are poorly understood and not relevant to this study.

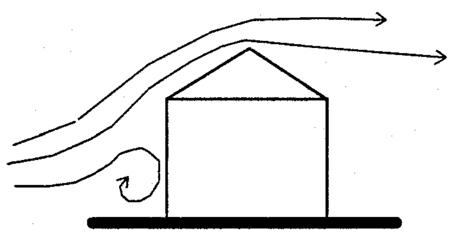
Prismatic buildings in wind are aerodynamically equivalent to three dimensional bluff bodies with sharply defined separation edges placed in a turbulent shear flow. The flow structure is complex and highly dependent on wind direction and building geometry. Architectural features such as eaves, canopies, parapets, wing walls and neighbouring buildings change the flow pattern around a simple building significantly. Also significant is the effect of landscaping (Landscape Planing, 1977 and White, 1954).

The flow past a simple building shape is shown in Figure 3.1 and the pressure distribution in Figure 3.2. The wind is slowed against the windward face and generates a positive pressure (i.e., greater than the free stream pressure) which diminishes towards the edges.

This cushion of pressure on the windward side diverts the air around the sides and over the roof of the building. Note the eddy near the lower part of the windward wall which creates an air flow opposite to the direction of the wind. The flow in this eddy spirals

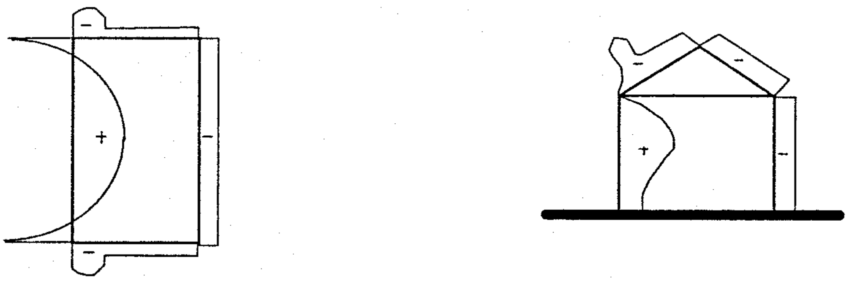


(a) Plan



(b) Elevation

Figure 3.1: Airflow around a building (based on Chandra, 1980).



(a) Plan (b) Elevation

Figure 3.2: Pressure distributions (based on Chandra, 1980).

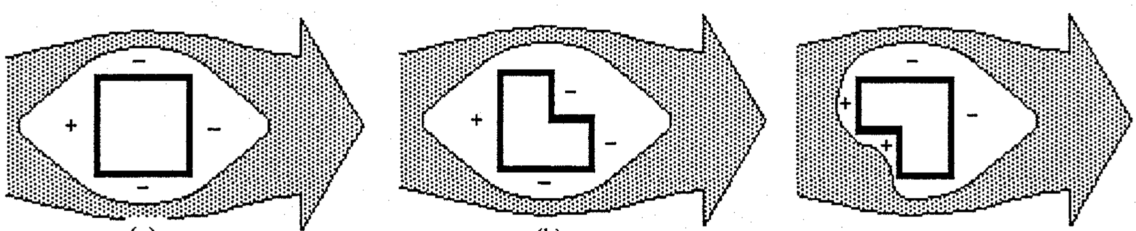
+ higher than free stream pressure

- lower than free stream pressure

along the sides creating the high suction shown on the side of the building (Figure 3.2a).

The separated flow at the back and the sides create a uniform suction as shown. Flow separates on the leeward roof surface. Flow may also separate at the leading edge of the windward surface, the probability increasing with decreasing roof pitch (flat roofs usually create vortices when exposed to oblique winds) and increasing building height (Chandra, 1980).

The following figures show the wind effect on the rectangular and the L shaped buildings. Plan (a) shows a low pressure zone along the sides parallel to the wind and on the leeward side of the building (Reed, 1953).



(a) (b) (c)

Figure 3.3: The effect of building shape on the air movement (based on Bowen, 1981a).

An L shaped structure (b) with a leeward projection, causes a large eddy in its recessed corner, with reversed airflow in its sheltered side (Evans, 1957). When the L shaped building (plan c) projects into the wind, the pattern alters with noticeable eddies on the leeward side (Newberry, 1974).

Wind direction and intensity can be modified by features around the building itself. Neighbouring buildings are a major perturbation obstacle. Their effects can be significant to channel the wind or to create a calm zone.

For an isolated building, the leeward wake extends roughly horizontally four times the ground to eaves height. Therefore, if the gap between buildings is smaller, those in the wake are poorly ventilated. This situation is often encountered when the buildings are in a row. But, if they are positioned in a staggered pattern, the flow of air is deflected on each building and induces contrasting high and low pressure zones. This configuration is also favourable for pedestrian comfort because in case (a) venturi effects increase air velocities.

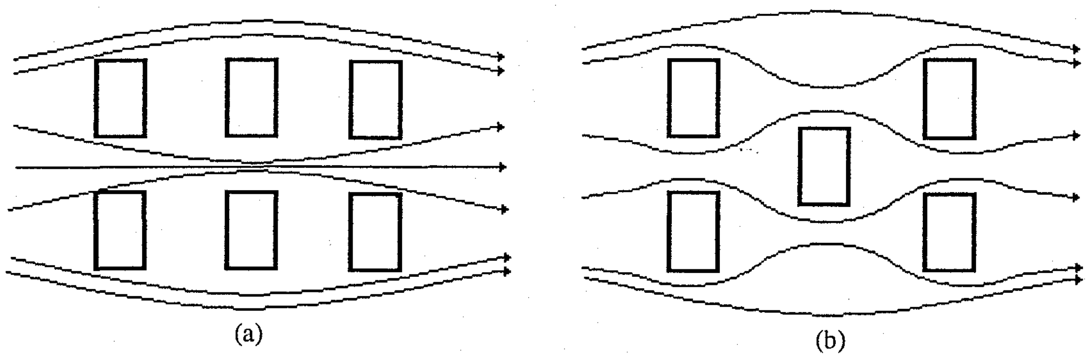


Figure 3.4: Building layout; (a) in line, (b) staggered (based on Fleury, 1990).

The building shape and height modify the air movement around the building. For a long building, most of the air is deviated over the top. For a tall building, air passes over the sidewalls. Roof forms also affect the size of the projected area. The depth of the wake will be increased with a pitched roof. However, predicting the effect of various heights and sizes of buildings on air movement, in a complex urban area, requires extensive skill and expertise (Fleury, 1990).

With increasing obliqueness in the wind direction, positive pressures can develop on both or one windward side depending on wind direction and barracking length-to-width ratio. Flow reattachment is possible if the length of the side along the wind direction is significantly greater than the length of the sides normal to the wind. Projecting canopies and overhangs modify the air flow, particularly if they are on the windward side. Usually, a pressure is created on the underside of the overhang and suction above it. Flow details are presented by previously cited references, and also Akins et al. (1979) who have collected extensive wind tunnel data on rectangular buildings.

3.1.2 Air Flow Through Buildings

Historically, the available airspeeds in ventilated rooms with different room geometry and window configuration have interested natural ventilation researchers. Studies were usually done by testing scale model buildings in wind tunnels. In the early 1950s, a comprehensive series of wind tunnel tests were conducted at the Texas A&M University using a uniform speed wind tunnel. A summary of those investigations is provided by Evans (1979). A detailed summary of the research results useful to the building designers is given by Reed (1953). Many of the airspeed patterns in rooms observed by the Texas A&M group have been summarised in pattern diagrams by Bowen (1981a).

3.2 VENTILATION STRATEGIES

3.2.1 Wind Towers and Solar Chimneys

Diverse strategies can be adopted to take advantage of the driving forces of natural ventilation. An example being, wind towers that draw upon the driving forces of the wind to generate air movement within the building (Karakatsanis et. All, 1986 & Bahadori, 1988). There are various systems based on this principle.

The wind-scoop inlet of the tower, oriented toward the windward side, captures the wind and drives the air down the tower. In this case, Figure 3.5 (a), wind driven air flow is created through a building's interior space when a high level wind-scoop inlet is located on the windward side and an outlet on the leeward side. A similar pattern, Figure 3.5 (b), may be expected with a low level inlet and a wind-scoop oriented leeward as an outlet, acting under low pressure suction (Bowen, 1981).

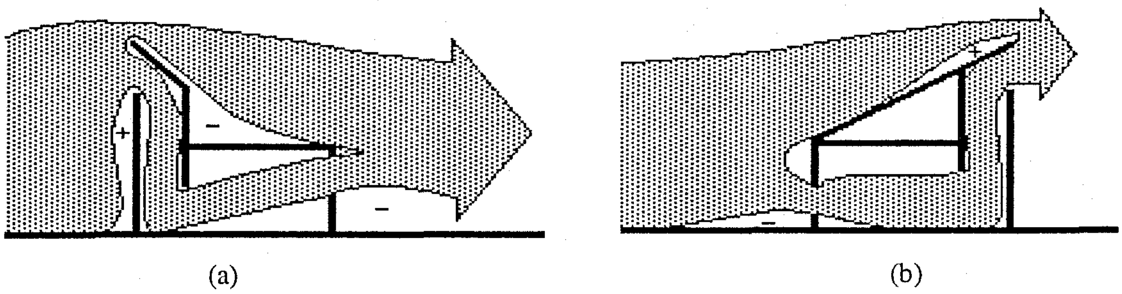


Figure 3.5: Natural ventilation by wind tower (based on Bowen, 1973).

Alternatively, the chimney cap can be designed to create a low pressure region at the top of the tower, and the suction initiates air flow up the chimney. A windward opening should be associated with the system for air inlet. The anabatic process benefits in this case from buoyancy of the warm inside air (Fleury, 1990).

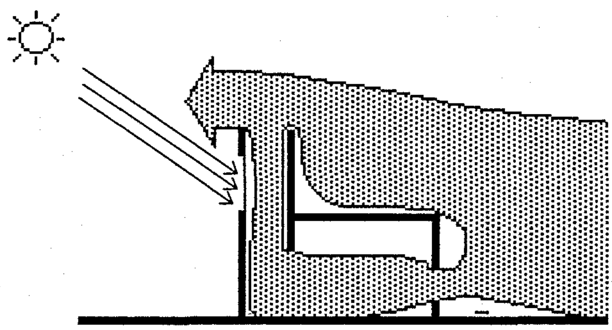


Figure 3.6: Natural ventilation by solar chimney (based on Bowen, 1973).

Solar chimneys use the sun to warm up an internal surface of the chimney. Buoyancy forces due to temperature difference help induce an upward flow along the plate. The chimney width should be closed to the boundary layer width in order to avoid potential backward flow (Bowen, 1973). The stairwell may serve as a chimney and so be completely integrated in the building architecture.

3.2.2 Windows Ventilation

The changes in airflow patterns caused by different types of windows were investigated by Holleman at Texas A&M (1951) as shown in Figure 3.7. Holleman found that fully open projection windows were capable of directing air at occupant level because of the slots. Such slots were also responsible for the good performance of casement windows for oblique wind incidences.

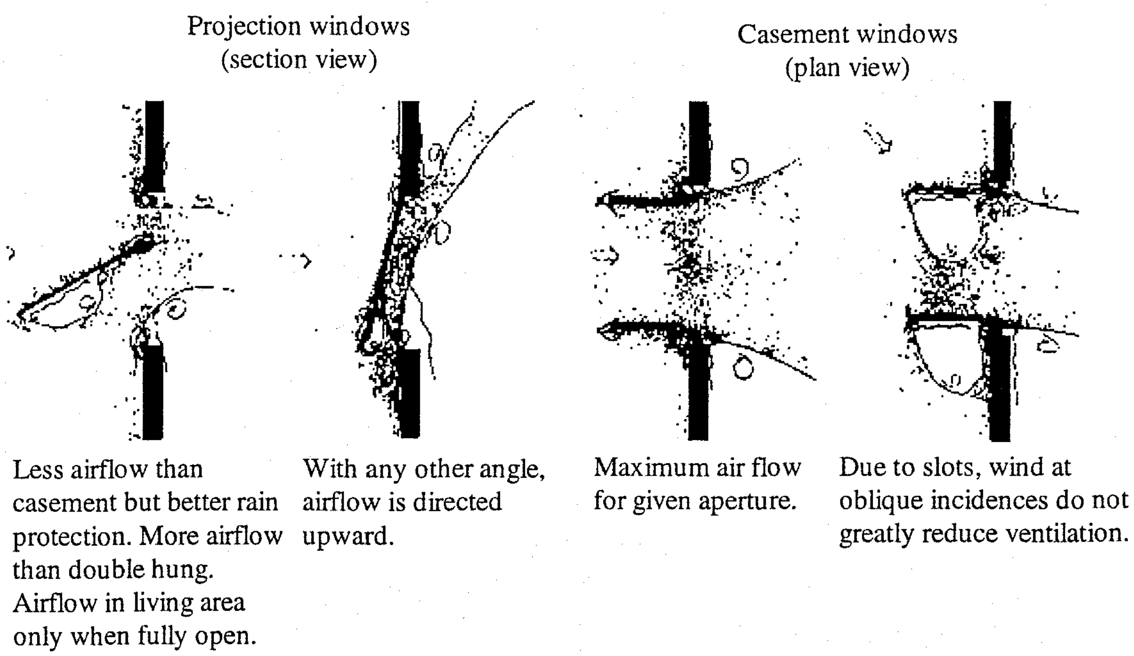


Figure 3.7: Airflow patterns through windows (Holleman, 1951).

In the 1960s Givoni conducted another thorough set of wind tunnel studies using a uniform wind tunnel. Many of the findings can be found in Givoni (1976). But many more interesting findings regarding airspeeds in building groups and buildings with courtyards, methods to cross ventilate double-loaded corridors, and building layout for apartment buildings to enhance ventilation are only cited in the original research report by Givoni (1968), *Ventilation Problems in Hot Countries.* Givoni demonstrated the usefulness of adjacent windows.

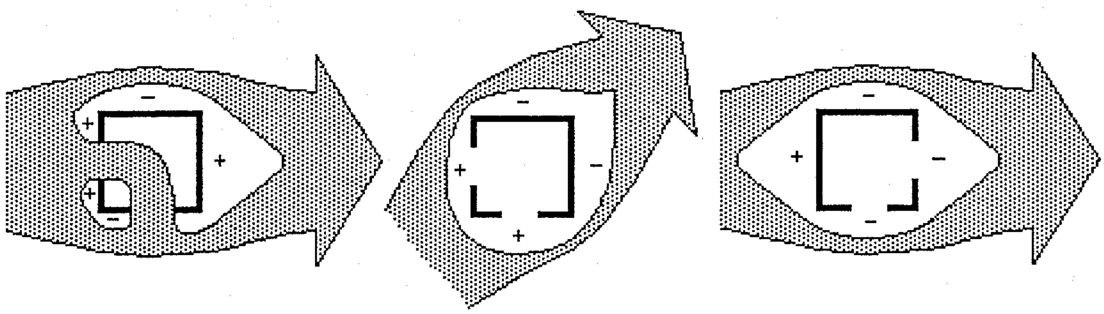


Figure 3.8: Natural ventilation by adjacent windows (based on Bowen, 1981a).

He found that rooms with windows on adjacent walls ventilated better than traditional cross-ventilated rooms with windows on opposite walls when the incident wind angle was perpendicular to the inlet. At oblique wind incidences (45° incidence angle to inlet) traditional cross-ventilated rooms performed better than rooms with adjacent windows.

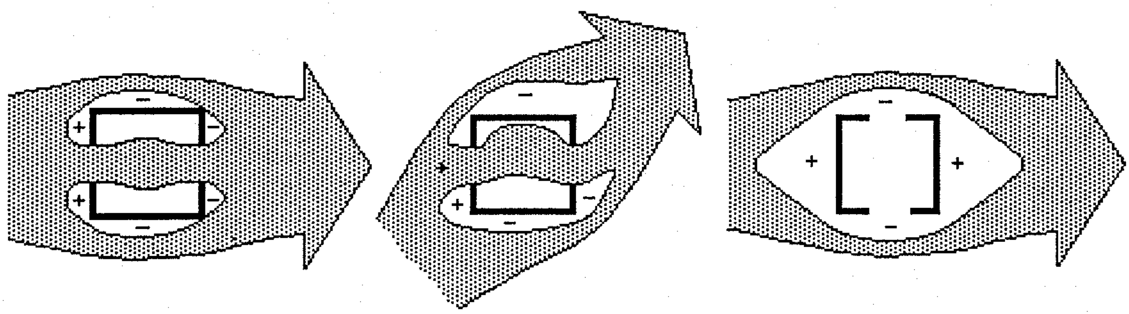


Figure 3.9: Natural cross ventilation (based on Bowen, 1981a).

The estimation of room air change rates, average room surface temperatures and room air temperatures enables one to predict the cooling or heat removal rate for natural ventilation. However, it is very important to note that the room air change rate may not be related to air flow rates through the openings. Consider normal wind incidence and windward and leeward openings directly in line with one another. Depending on the ratio of the areas of opening, the air can rush through without significantly mixing and entraining room air. As a result, little heat will be removed and circulation in many parts of the room will be poor. Staggered windward and leeward openings that force the air to turn are better for ventilation. For similar reasons, winds at an oblique rather than normal incidence provide better cooling if the apertures are not staggered (Chandra, 1980).

Givoni also provided an elegant solution to the problem of ventilating rooms with only one outside wall by proposing wing walls. These wing walls create distinct positive and negative pressures on the windows inducing good airflow in the room when the outside wall is a windward wall. Givoni found that for oblique winds, the wing walls created an average room airspeed of about 40% of outside wind compared with about 15% for the room without the wing wall. However, for perpendicular winds, Givoni did not record an appreciable increase the average room airspeed increased from 12% without wing walls to 18% with wing walls.

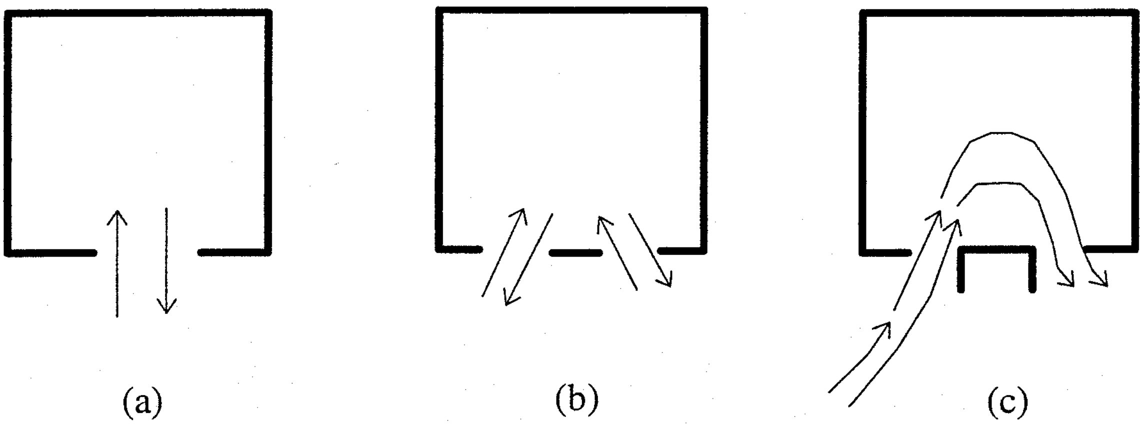


Figure 3.10: Natural ventilation by one side wall window (based on Chandra, 1980).

In the 1960s Sobin conducted another comprehensive wind tunnel study at the Architectural Association (London). Sobin was the first to use a boundary layer wind tunnel for natural ventilation studies. A boundary layer wind tunnel differs from the uniform speed wind tunnel used in aeronautical studies in that the former simulates both the variation of wind speed with height and the natural turbulence of the wind. Sobin published some of his findings in 1981, and published all the data of his several hundred tests in 1983. Sobin investigated many interesting window types and measured room airspeeds both in section and plan. One of his most interesting findings relates to window shape. He found horizontal windows (windows that are wider than their height) created greater wind speeds than vertical windows (windows that are higher than their width). This effect was more pronounced for oblique wind incidences. It is interesting to note that Givoni's apertures were also horizontal and he also found good performance at oblique wind incidences. Aynsley et al. (1977) continued airspeed measurements in a building with wind scoops.

Insect screening is a necessary consideration in ventilation in many parts of the world. Givoni (1976) found that screening entire balconies produced greater airspeeds in rooms than did screening the windows. Van Straaten (1967) measured the decrease in airflow caused by screens and found that it was dependent on the incident wind speed. For a 1.5 mph (0.7 m/s) wind, the airflow was reduced by 60%, whereas in a 6 mph (2.7 m/s) wind the reduction was only 28%. This difference is possibly due to the reduction of the wake region behind a cylinder as the Reynolds number increases.

Internal airspeeds can only be predicted by solving the three-dimensional turbulent flow equations, a difficult task that has been attempted by only a few (Nielsen, 1974; Ishizu and Kaneki, 1984). White of Texas A&M (1954) investigated airflow reductions caused by landscaping elements such as trees and hedges. He also discovered using solid paper and landscape moss models of trees and hedges, that certain landscaping schemes are advantageous for increasing ventilation in building (Evans, 1979).

3.2.2.1 Effect of Window Design on Natural Ventilation

The effect of various window configurations and architectural factors on indoor air movement, including wind orientation, cross-ventilation, inlet/outlet area ratio, inlet shape, window location and window accessories, was reported in an experimental wind tunnel study by Sobin (1980).

He conducted a test program to study the last four of these factors. The study was undertaken at the Department of Tropical Studies of the Architectural Association (London) during the period 1963-1966. The primary objective of the program was to systematically investigate the degree and manner in which various window configurations convert to kinetic energy of the natural wind into useful indoor air movement.

Particular emphasis was placed on examining the airflow effect of:

1. orientation to wind
2. cross-ventilation
3. the relative size of inlet and outlet openings
4. the shape of inlets
5. window location; and
6. the type of "window accessories" utilised.

The test schedule included some window configurations previously tested either qualitatively or quantitatively, together with a substantial number of window shapes and accessories not previously studied.

3.2.2.2 Orientation

The effect of orientation to wind or wind angle on ventilative cooling was found to vary with the physical characteristics of the window configuration used, and in particular the characteristics of window location, shape, size and accessories. Generally speaking, for the majority of window configurations tested, orientation of inlets at 90° to the wind provided the highest average indoor speed ratios, with airflow velocities dropping off rapidly with external wind shifts to either side of 90°. Significantly enough, however, it was found that certain combinations of inlet characteristics (especially shape), while providing substantially similar results with 90° wind are also capable of providing equal or better ventilative cooling in oblique (up to 45°) winds than they do in normal (90°) winds. This is a finding of particular importance since wind direction is of course rarely if ever constant. If window systems are to take maximum advantage of wind-powered ventilation, they should be selected where possible to provide a reasonably "broad band", not a strongly "peaked" directional response providing greater effectiveness under customary conditions in which the wind changes direction over a certain range of directions on an hourly, daily, or seasonal basis. This is the case even in areas of great directional constancy, such as trade wind locations, where, as on the coastal regions of Caribbean islands, directional shifts of up to 90° take place during each 24-hour period. These directional effects are described with respect to each of the window design characteristics discussed below.

3.2.2.3 Cross-Ventilation

Test results confirm that for optimum ventilative cooling, sufficient effective area of inlet and outlet openings is required, with the inlet/s located in a zone of positive pressure and the outlet/s in a zone of negative pressure. Rooms equipped with inlets only tend to provide very much reduced indoor speed ratios (though demonstrating somewhat improved performance in oblique winds), especially in the case of horizontally-shaped openings. The configuration with inlets only corresponds to the frequently encountered arrangement in which rooms are provided with windows on one side of a building only. The relative improvement produced by oblique wind can amount to as much as 250%, where the single opening is located on a windward facade, but the overall result even under these conditions at best amounts to only one third of the average speed ratios provided by a cross-ventilating configuration. Smoke-tracing investigations of "one-sided" configurations show that in oblique and normal winds, a single opening functions as both inlet and outlet. Motive power for indoor airflow thus originates in pressure differences across the opening (almost always small in 90° wind, but somewhat more substantial in oblique wind).

3.2.2.4 Inlet Outlet Area Ratio

Test results confirmed that where inlet and outlet opening are equal, as their areas increase, increases occur in the amount of indoor ventilative cooling they produce. Since, however, window sizes are not determined by ventilation alone but must also take into account other architectural factors such as daylighting, privacy, security, and solar control, a significant question for ventilation purposes is how best to distribute a given and usually limited amount of opening area. An important parameter here is the relative distribution of area as between the inlet/s and outlet/s. Sobin's preliminary test results suggest that for a given total opening area, the highest indoor air speed ratios throughout rooms are achieved when the ratio A_o/A_i is approximately 1.25, that is, when the inlet is slightly smaller than the outlet. Inlets substantially smaller than outlets produce high local velocities in the vicinity of the inlet itself, but lower speed ratios when results are averaged across the entire room. It thus appears advisable to provide approximately equal inlets and outlets, or a very slightly smaller inlet, where maximum ventilative cooling is required.

3.2.2.5 Inlet Shape

A review of test results suggests that inlet shape is the single most important window design parameter in determining the efficacy of wind driven ventilative cooling. Square and vertical inlet openings produce a sharply peaked or "narrow-range" response under conditions of changing wind direction, with both types attaining maximum performance in a perpendicular (90°) wind, but falling off rapidly in efficiency with even small departures of wind direction from the perpendicular. At 45°, for example, vertical inlet performance has decreased by more than 17%, that of square inlets by more than 26%.

On the other hand, horizontal inlets not only have a substantially higher average performance for all wind angles, but in contrast to square and vertical inlets, horizontal inlets actually improve their effectiveness in angled winds, producing two maxima at wind angles in the vicinity of 45° to either side of the perpendicular, while showing a relatively flat, or "wide-range" response throughout this 90° quadrant of wind angles (or orientations). The improvement of horizontal inlets in oblique compared to perpendicular wind angles can amount to 30% or better and depends on the relative opening sizes used. For example, given equal areas of inlet and outlet, where each opening is equal to 22% of the inlet and outlet wall areas respectively, the increase in average indoor speed ratio for horizontal inlets in a 45° wind compared to a 90° wind is typically on the order of 16%. Horizontal inlets were found to increase their performance in oblique winds in fully cross-ventilated rooms (openings in opposite walls), in diagonally-ventilated rooms (openings in adjacent walls), and in rooms with inlet openings only.

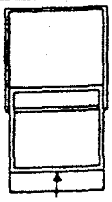
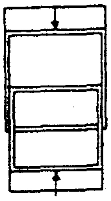
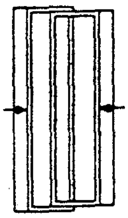
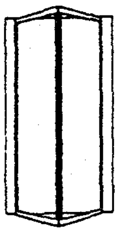
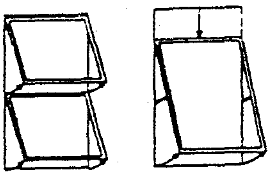
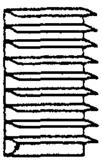
From the results of Givoni's ventilation study (1962), he reached the general conclusion that better ventilation is often achieved when the wind is oblique to the inlet. With respect to this conclusion, however, it should be noted that all inlet and outlet openings tested by Givoni were horizontally shaped; no "square" or "vertical" opening shapes were included in his tests. Another of Givoni's conclusions not supported by the results of the present study concerns his explanation for the superiority of oblique wind angles. Givoni suggests that when airflow has to change direction inside a room, as it must with wind oblique to the ventilation-axis (defined as a line drawn between the centre points of the inlet and outlet openings), a larger proportion of room volume becomes involved in the flow resulting in higher average velocities.

Test results from the Sobin's study (1980) indicate, however, that a change in direction of airflow inside a room does not necessarily lead to increased air movement, on the contrary it is often substantially reduced. Smoke-tracing shows, for example, that in oblique winds, strong directional changes take place inside rooms equipped with square or vertical inlets, yet these two inlet types consistently provide substantially lower average indoor speed ratios in oblique than in perpendicular wind, typically showing relative losses of 25% or more in a 45° wind.

A more comprehensive hypothesis, capable of explaining the full range of observed changes in performance with different opening shapes appears to require inclusion of at least two factors in addition to flow patterns: (1) the influence of wind angle on the effective inlet area, and (2) external wind-pressure distributions. It should also be observed that horizontally-shaped inlets tend to produce a broader, flatter, more "room-wide" jet or sheet indoor airflow than do vertical or square ("hole-in-a-wall") shaped inlets. This fact may help to explain, at least in part, their clearly superior ability to provide higher average amounts of ventilative cooling throughout the interior of rooms.

The following table illustrates the main characteristics of the various window styles as far as ventilation is concerned.

Table 3.1: Advantage and drawback of various window types (Fleury, 1990).

Window type	Advantage	Drawback
 Single Hung	Adjustment of the opening area. Air enters the openings and continues inside in the same direction as the outside wind.	Opening limited to 50 % of the window size. Winter leakage.
 Double Hung	Adjustment of the opening area. Adjustment of the sashes for directing air streams to a specific area.	Opening limited to 50 % of the window size. Winter leakage.
 Horizontal sliding	Adjustment of the opening area to direct air streams to a specific area.	Opening limited to 50 % of window size. Width/height small ratio does not favor hight efficiency for every wind direction.
 Side hunged	100 % openable. Sash can act as a wingwall and redirect the flow. Good sealing.	Difficult window treatment. Low market penetration
 Projection and awning	Excellent rain protection.	At low opening angles, the air flow is deflected upward, outside the occupied zone. Reduced opening area
 Jalousie	Take benefit of a wind direction. 100 % openable. Can direct the flow.	Excessive infiltration
 Basement and Hopper	Excellent for night ventilation and for air intake and exhaust. Can stay opened.	Reduced opening.

3.2.2.6 Window Location

Preliminary results suggest that in general, ventilative cooling performance is improved when the inlet and outlet are arranged so that the ventilation axis is parallel to the wind. This condition occurs either (a) when the inlet and outlet are located directly in line with one another on opposite walls of a room, with the wind perpendicular to the inlet; or (b) when the inlet and outlet are located in adjacent walls of a room, and an oblique wind passes successively through the inlet, through at least one corner of the room, then passes through the outlet. It should be noted that in both of these cases, the main tube or jet of airflow passes directly from inlet to outlet without changing direction inside the room. The only previous study to have examined the diagonally-ventilated room configuration (Givoni, 1962) and which reported a test result contrary to that attained in the present study, proceeded to use this result as a basis for concluding that ventilation is improved wherever airflow changes direction inside a room.

The Texas (White, 1954) studies found that while airflow was maximised by equal inlet and outlet areas, airspeeds in rooms were locally maximised (particularly near the inlet) if the outlet was slightly larger than the inlet. They found that while the outlet location did not affect the airflow pattern significantly, the inlet location controlled the airflow pattern. A high inlet directed airflow near the ceiling, whereas a low-to-medium height inlet directed airflow to the occupant levels. However, even a mid ceiling height level inlet at the second floor directed air to the ceiling. They also observed the "wall jet" effect. If the inlet is near a corner, the air tends to lair along the nearest wall.

3.2.2.7 Window Accessories

Window accessories have been traditionally designed to work as sun shading, privacy or security devices, not as airflow controls. However, window "equipment" designed to produce solar or rain protection, visual privacy, shielding and other non-aerodynamically related purposes can frequently have unintentioned, yet at times seriously deleterious effects on wind-powered ventilative cooling. Earlier studies have recognised this problem; the present results confirm its importance.

One instance of the unfavourable effect of window equipment revealed by present test results is the aerodynamic effect of fixed or movable horizontal and vertical louvres. The effect of primarily horizontal louvres or canopies on indoor airflow speeds and patterns is chiefly manifested in section. Horizontal louvres have the tendency, when adjusted to typical angles, to direct airflow toward the ceiling, thus greatly reducing ventilative cooling effectiveness within the room's occupied zone. However, the effect of primarily vertical louvres chiefly shows up on plan. For example, horizontal louvres used on vertical inlets do nothing to alter the basically peaked, "narrow-band" and symmetrical plan-response of this type of opening to changing wind angles, regardless of the blade setting angle. The same phenomenon is created by inlet accessories which incorporate vertical elements, which also tend to produce a strongly peaked, "narrow-band" directional response, providing maximum average indoor airflow when the vertical elements present the minimum degree of airflow resistance. i.e., when they are parallel to the wind. For example, when vertical louvres are set perpendicular to the plane of an inlet opening, they tend to cut off diagonal wind; yet when oriented at an oblique angle they sharply favour diagonal winds arriving at angles close to that same direction. By the addition to vertical control elements, it is also possible to "convert" the typical "broad-band" directional response of horizontal inlets, into the "narrow-band" response of square or vertical inlets.

In general, as the wind angle shifts, vertical "window furniture" produces an effective change in inlet area. The degree of change depends on the angular relationship between the accessory elements and wind direction. Increasing the angle increases "narrowing", and decreasing the angle increases effective inlet area where it is not possible or desirable to orient opening accessories to face the wind, flow-directing accessories such as louvres can be placed within or across the inlet opening to "turn" wind to enter a belladonna. But results show that considerable resistance losses of up to 50% or more are incurred by the use of such techniques, suggesting that wherever possible flow-directing accessories should be located adjacent to inlets. not within them.

3.2.2.8 Architectural Implications

The architectural implications of natural ventilation are discussed in detail in previous references. Some of the findings are summarised below by Chandra (1980):

a) Airflow is governed by three guiding principles
   - Air has inertia, i.e., air does not necessarily travel in the shortest path between an inlet and an outlet in an adjacent wall.
   - Moving air produces friction in contact with bodies and as a result slows down or forms into eddies.
   - Air moves due to pressure differences.

b) Local topographical conditions, landscaping and adjacent buildings can dramatically influence winds at the site. Thus, recorded wind speeds and direction data at a nearby meteorological station may be of little value.

c) A study of the external airflow patterns is necessary to determine the best locations for windows or other apertures. Inlets should be placed in the high pressure regions and outlets in the low pressure regions.

d) Window inlets must be so designed and placed as to provide maximum airspeeds at the desired locations (sitting level in living rooms, just above the bed level in bedrooms).

e) Window types greatly influence the airflow direction in a room. One of the best type of window for natural ventilation purposes seem to be awning or louvred windows which can be manually rotated to direct the wind to desired locations inside the room. Such windows will also generally provide the maximum aperture area, afford protection from the rain and allow installation of bug screens. Although window types affect the airflow direction, they do not significantly affect the overall ventilation rate.

f) The best location for an inlet is near the vertical and longitudinal centre of a wall which is perpendicular to the wind. This is where the pressure is the highest. If the inlets are too high or are near to the side edge of a wall, overhangs or wingwalls should be provided to induce a high pressure zone.

g) The outlet should be exposed to the eddy or wind shadow (eddies in Figure 3.1 a) which is the low pressure zone. The outlet may be placed high, near the ceiling, to take advantage of the stack effect, if any. However, it may be ill advised in locations without a prevailing wind direction, or in some coastal areas where the prevailing wind fluctuates between two opposite directions.

h) In many cases, oblique winds at 45° provide better or equally well ventilation as normal winds. Consider a cross-ventilated room with inlet and outlet windows directly in line with each other. Winds normal to the windows will have a tendency to move through the building without mixing well with the room air. Therefore, regions near the wall will not be well ventilated. Whereas, oblique winds will create a circulating airflow pattern in the entire room providing greater ventilation rates and lower local airspeeds. This has profound design implications in those humid areas where the prevailing summer breezes are easterly or westerly, the direction most difficult to shade and ventilate at the same time. If the prevailing breeze is easterly, one can orient the inlet wall to face south east or north west, directions which are easier to shade with long overhangs. Thus simultaneous shading and ventilation are possible .

i) Although windows on only one side of a room do not induce appreciable ventilation, it is possible to design vertical projections (wing walls) to induce appreciable ventilation under oblique winds.

j) For cross-ventilated rooms, <u>both</u> inlet and exit window sizes need to be increased to increase ventilation rates. However, making the inlet smaller than the outlet creates higher wind speeds near the inlet wall, which may be desirable.

k) The effect of indoor partitions is greatest when they are close to the inlet window. But on the average, the ventilation rate is not greatly reduced.

l) Fly screens a necessity in bug-infested hot and humid regions, result in a percentage decrease in total air flow that is greater (50-60 percent) at low wind speeds (1.5-2 mph) than at higher wind speeds (25 percent at 10 mph), for normal incidences. Wind direction effects are significant. Screening a whole balcony produces more ventilation than screening the window.

m) Extended eaves and end walls are very effective for ventilation with oblique winds for buildings on the ground. Their effectiveness is further increased when the building is elevated above the ground.

n) Hedges and trees can significantly aid or deter ventilation, depending on plant height and spacing from building.

3.3 ROOM AIRFLOW CALCULATION METHODS

An important contributor to errors in the thermal analysis of naturally ventilated buildings is inaccurate airflow predictions. These predictions are important for designers in countries where most buildings are naturally ventilated. Prediction models for natural ventilation range from simplified to comprehensive ones. Comprehensive procedures are usually of limited use to designers as they are laborious to implement and the required input data are difficult to obtain.

Estimation of the air speed in the room is important in the determination of comfort. So room air speeds and directions, in addition to air change rates, must be measured to calculate efficiency of proposed passive evaporative cooling system. At present, empirical correlations for average room air speeds as a function of external wind speed and direction exist only for a few isolated cases. See Givoni (1976) for square rooms at normal incidence with inlet/ outlet windows at opposite walls and Aynsley (1979) for rectangular buildings on ground or above-ground at normal incidence with multiple inlet and outlet apertures, all in the direction of the normal incidence wind. Correlations for oblique winds and the effect of partitions, fly screens and such architectural features as wing walls or canopies are scant. Caudill et al. (1951) and Evans (1959) have presented some data.

The following sections will present three methods of calculating wind-driven ventilation.

3.3.1 Wind-Driven Ventilation, Method I

Aynsley et al. (1977) have derived the airflow equations for simple cross-ventilated rooms and for rooms in series where all air flows through a series of openings. For a general formulation suitable for multiple windows in a building, a set of simultaneous nonlinear algebraic equations have to be solved; see Vickery (1981) or Walton (1983). In the following, the Aynsley formulation is discussed. The airflow in cubic meter per second in a room with a single inlet and outlet summarised by Chandra (1989) and is given by:

$$Q = C_{de} A_e V_{ref} \sqrt{C_{pi} - C_{po}} \qquad \textbf{(3.1)}$$

where:

- Q = air flow rate (m^3/s)
- V_{ref} = site external wind speed at a reference height (m/s)
- Cp = pressure coefficients at inlet and outlet referenced to the site wind speed V_{ref} at the reference height the subscripts i and o refer to inlet and outlet, respectively
- C_{de} = effective discharge coefficient of inlet and outlet
- A_e = effective area of inlet and outlet.

The $C_{de}A_e$ product is defined by:

$$C_{de}A_e = 1/\sqrt{1/C_{di}^2A_i^2 + 1/C_{do}^2A_o^2} \quad \textbf{(3.2)}$$

where C_d and A refer to the discharge coefficient and areas for the inlet and the outlet. The discharge coefficients vary depending on aperture size and location. A range of values are suggested by Aynsley et al. (1977). For small openings in the center of a wall a typical value of C_d would be 0.65. For equal inlet and outlet areas (A) having identical C_d, Equation 3.2 reduces to:

$$C_{de}A_e = C_dA/\sqrt{2} \quad \textbf{(3.3)}$$

In using Equation 3.1 one needs to estimate V_{ref} and Cp. The V_{ref} value is usually only available at airports and needs to be corrected for the terrain effect. The major problem is in estimating the pressure coefficients. Surface pressure coefficients have been measured by civil engineers for wind loading calculations, using solid models in boundary layer wind tunnels. For ventilation calculations, one needs Cp data for porous buildings. An investigations by Vickery et al. (1983) have shown that wall porosities less than 20% do not affect the solid body pressure distributions significantly for apertures in the wall. However, small roof level apertures can have a significant effect, and airflows computed from solid body Cp data can substantially overestimate the airflow in such situations.

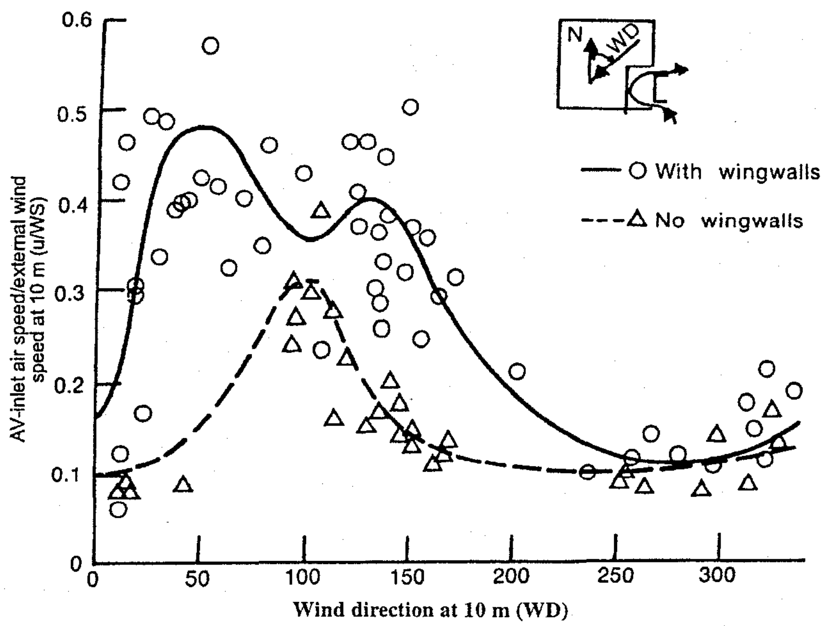
Cp data for isolated buildings are available from Davenport (1978), MacDonald (1975), Newberry and Eaton (1976), and Vickery et al. (1983). Allen (1983) has a comprehensive summary of different Cp data bases. The problem in using much of these data is that the Cp is usually referenced to an eaves height external wind speed, and wind speed data are available generally at the 10-m height. These Cp data were referenced to the 10-m height and have been summarised by Chandra (1983) for design calculations. It was found that the Colorado State University (CSU) Cp data predicted measured airflows in a typical one-story residence at Florida Solar Energy Centre (FSEC) to 18%. This design procedure also adapts the data of Lee, Hussain, and Soliman (1980) for reducing Cp in the presence of neighbouring buildings. Additional data on the shielding effect of neighbouring buildings may be found in Evans (1957), Vickery (1981), Aynsley et al. (1977), and others. In general, Evans (1957) found that the eddy zone of a ventilated building extended three to four eaves heights. Lee's data show that the reduction in Cp is negligible if the upstream building is farther than six eaves heights away for single-story buildings.

The Cp data from wind tunnels are obtained from strong wind conditions. The validity of such Cp data under low wind conditions is being examined by Ashley and Sherman (1984). They measured Cp in the field for some U.S. Navy housing in Hawaii and are now comparing the results of wind tunnel tests conducted in the tunnel at the Naval Civil Engineering Research Laboratories in Port Hueneme, California.

Airflow can be measured in the field or in the wind tunnel by the tracer gas dilution technique. This technique was first done by Van Straaten in 1955 and is reported in Van Straaten (1967). FSEC has performed tracer gas studies using SF6 and has monitored two residences. Field measurements (Chandra, 1983) indicate that in a conventional cross-ventilated residence with no significant window shading and with only 3.7% total open wind area covered by a 60% porosity screen, airflows on the order of 20-30 air change per hour (ach) are obtainable in site 10-m wind speeds of 7-10 mph (3-4 m/s).

The efficiency of wing walls in improving room airflow was demonstrated by full-scale experiments at FSEC. Room airflows were measured in a room with and without wing walls. The results are reported in Chandra et al. (1983) and are summarised here. The experiments were conducted in a 3.7 x 5.5 x 2.4m room with two widely spaced windows (actually holes in the walls). The apertures were each equal to 2.8% of the floor area. The room air changes per hour were measured with and without the wing walls for a variety of outside wind speeds (WS) and wind directions (WD). The air changes per hour figure was multiplied by the room volume to obtain airflow through the room. This number was then divided by one of the aperture areas. The result is an effective average inlet velocity (u), equal to that at the outlet, since the inlet and outlet areas are equal. This u can be divided by WS to obtain a dimensionless parameter indicative of the total airflow and the average airspeed at the apertures. A further benefit of the airflow data represented as u/WS is that this quantity is likely to be independent of reasonable variations in aperture size as long as the inlet to outlet area ratio is not changed.

Figure 3.11 plots u/WS against wind direction (WD) for all the experiments. The WD is clarified by the small drawing of the room shown in Figure 3.11. WD at 0, 90, and 135 indicate northerly, easterly, and southeasterly winds, respectively. A value of 90 for WD is thus perpendicular to the wall containing the window. As can be seen, there is significant scatter in the data. The trend is indicated by the solid and dashed lines through the data points.



WS = Wind speed
WD = Wind direction
u = Inlet air speed (airflow divided by inlet area)

Figure 3.11: Airflow data sets with and without wing walls obtained in full-scale experiments (Chandra, 1989).

Considering the data for the room with the wing walls, we can see that the airflow has a broad double peak. It peaks first when WD is 55 and then decreases to a low when WD is about 100; it increases to another peak when WD is 145 and then decreases again. The valley in the curve when WD is 100 is expected since there are no large pressure differences across the apertures for normal (90° to the wall) winds. However, the reduction in airflow when WD is around 100 is not severe. The reason for the substantial airflows, and the high, narrow peaks without wing walls, is the fluctuation in the natural wind direction.

WD fluctuates quite extensively. Measured standard deviations of WD over the 2-min. experiments based over the 10-s scan of the WD have ranged from 10° to 35°. This means the actual WD may fluctuate +40° or more about the mean value. For a mean easterly WD, the WD really alternates between east-northeast and east-southeast. This results in the two windows alternating as inlet and outlet. This occurrence has been visually confirmed by smoke tests and was also seen for northerly winds. We believe that this WD ulceration is the prime reason the windows with wing walls behave like those without wing walls for normal winds. This fluctuation also is the reason the curve tends to rise when WD is greater than 330; for those WDs there are times when the WD becomes greater than 360 (i.e., 0) and the wing walls become ineffective.

For the room without wing walls, the airflow rate peaks around WD equal to 100. This high, narrow peak in airflow is believed to be caused by WD fluctuations. The peak creates pulsating pressure fluctuation across the apertures, causing them to alternate as inlets and outlets. This high airflow rate is to be contrasted with low room airspeeds for similar geometry rooms as observed by Givoni (1976) in wind tunnel tests.

These field results show that ventilation in real buildings can be higher than that indicated in wind tunnel tests. This is because wind direction fluctuations, present in the real wind, are almost impossible to model in the wind tunnel.

3.3.2 Wind-Driven Ventilation, Method II

A structured procedure for calculating wind-driven natural ventilation rate proposed by Vickery (1983) and summarised by Swami and Chandra (1988) for wider dissemination of his results. The assumptions of his method are:

- No stack effect.
- No pressure drop inside building, negligible effects due to partitions.
- Perfect mixing.
- Wind profile can be described by power law.
- Use of pressure coefficient (Cp) data on an average wall basis for low-rise buildings.
- Valid for window or other wall apertures only, not for roof level apertures.

The procedure for calculating the flow through a cross ventilated building with one effective inlet and one effective outlet can be expressed as:

$$\mathbf{CQ = Q / (A_e\, V_{ref}) = Cd\, (\Delta\, Cp)^{1/2}} \qquad \mathbf{(3.4)}$$

where:

CQ	= the flow coefficient
Q	= the airflow rate (m^3/s)
A_e	= the effective window area (m^2)
V_{ref}	= the reference velocity (m/s) at the building height
Cd	= the discharge coefficient = 0.62 [recommended by Swami and Chandra (1987)]
ΔCp	= the pressure coefficient difference across the inlet and outlet.

and

$$\mathbf{A_e = A_o A_i / (A_o^2 + A_i^2)^{1/2}} \qquad \mathbf{(3.5)}$$

Where, A_o and A_i are the open outlet and inlet areas respectively (m^2) and the reference velocity is determinable from equation:

$$\mathbf{V_{ref} = V_{bH} = (10/h)^{b_r}\, (H/10)^{b_b}\, (a_b/a_r)\, V_{rh}} \qquad \mathbf{(3.6)}$$

where:

h = mast height in the reference terrain (m)

V_{rh} = wind speed in the reference terrain at height h (m/s)

a_r & b_r = terrain constants of the reference terrain (Table 3.3)

H = height in building terrain where V_{ref} is required (m)

a_b & b_b = terrain constants of the building terrain (Table 3.3)

$V_{ref} = V_{bH}$ = the reference velocity (m/s) at the height (H) in the building terrain.

Vickery (1983) constructed two identical models, one solid and another with varying wall porosities for the long walls. The airflow through the ventilated model was directly measured in the wind tunnel. The flow was then compared to that calculated using Cp data as measured on the identical solid-body model. For porosities larger than 21% wall porosity, the solid body Cp data overestimates the actual flow. The "through" flow through the porous building decreases the pressure difference between the windward and leeward sides and thus the actual flow is reduced. Vickery suggests a simple correction factor to account for this:

$$\mathbf{Ca = CQ/(1+CQ)} \qquad \mathbf{(3.7)}$$

Where, Ca is the actual flow coefficient for large porosities.

Therefore the airflow equation takes the form:

$$\mathbf{Q = Ca \,.\, V_{ref} \,.\, A_e} \qquad \mathbf{(3.8)}$$

The final equation for air changes per hour, ach, is:

$$\mathbf{ach = (Q \times 3600) / Zv} \qquad \mathbf{(3.9)}$$

Where, Zv is the zone volume (m^3).

For ach less than 3, ach = 3 should be used. Note that this value is based on experimental measurements of ach with windows fully open on windless nights (Swami and Chandra, 1988).

3.3.2.1 Pressure Coefficient Calculation

Air flows through a building due to wind-induced pressure differences acting across the inlets and outlets of a building. The stack effect can also cause natural ventilation. However, the stack effect is weak, and it works only in the daytime when ambient temperatures are generally higher than the desired indoor temperature and ventilation is not desired. Thus, stack ventilation is not discussed further in this method. The wind engineering and infiltration research communities have generated a large data base of building pressure coefficient (Cp) distributions by testing solid building scale models in boundary-layer wind tunnels. Natural ventilation researchers have proposed that the airflow through naturally ventilated buildings be also calculated from building Cp data .

3.3.2.2 Effect of Wind Angle and Building Geometry

The coefficient of' pressure varies considerably with the approach wind angle and, to a lesser extent, with the geometry of the building (i.e., side ratio and roof slopes).

Cp data, either mean or local, are usually given in terms of the wind angle for each of the four surfaces constituting the house. Most researchers have defined the wind angle with respect to the windward wall of the building, and Cp data for all four walls are tabulated with respect to the wind angle. The disadvantage of this approach is having to carry the wall number as an additional parameter for curve fitting. It was felt that defining the wind angle with respect to the actual surface for which Cp is sought, rather than any one surface, would be more appropriate and would be less cumbersome for curve fitting. Since all data are available for rectangular buildings, the data could easily be converted in terms of our wind angle definition. This would eliminate wall number as a dependent parameter.

The wind angle is defined to be the angle between the outward normal of a surface and the wind direction and is always a positive value between 0 and 180 degrees. Due to the symmetry of the data, the actual sign of the angle is unimportant.

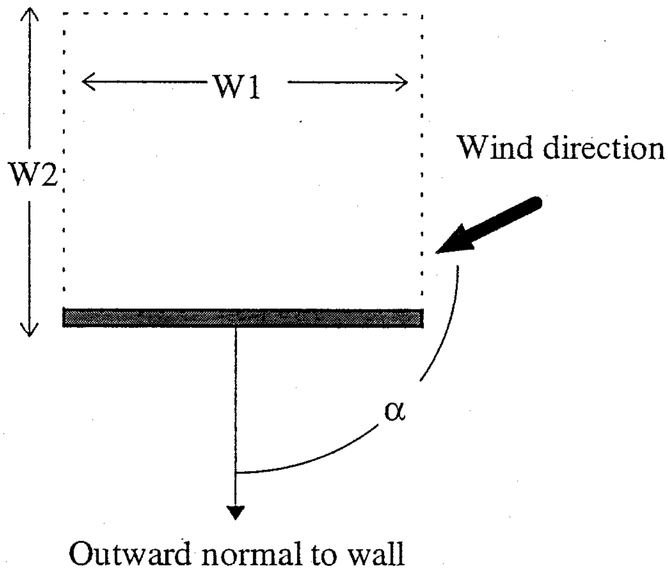


Figure 3.12: Conventions for wind angle α, and side ratio S (based on Swami & Chandra, 1988).

α = the angle between the wind direction and the outward normal to the wall

S = the side ratio (W_1/W_2)

W_1 = the width of the wall for which Cp is sought

W_2 = the width of the adjacent wall.

The solid line in the figure is the wall surface under consideration, and the dotted line indicates the rest of the building. To account for the effect of the adjacent wall, the parameter side ratio is defined and is another parameter influencing the Cp value. Data for all the surfaces were converted into this form. Samples of such a conversion are provided in Swami and Chandra (1987).

3.3.2.3 Normalised Pressure Coefficient (NCp)

Different researchers have referenced Cp based on velocities at different heights. Since it is proposed to use Cp referenced to the velocity at the building height, all Cp data in the literature must be re-referenced. To do this, the velocity profile of the study will have to be known a priori. This effort can be considerably simplified if Cp at different wind angles are normalised with respect to CP at a fixed wind angle. Since Cp at a wind angle of zero degrees is usually most reliable and this value is provided by most studies, all Cp are normalised with respect to the Cp at the wind angle of zero degrees. The normalised values thus become independent of the reference height, and it is only needed to reference the Cp at zero degrees to the building height. This will result in the value of normalised Cp at zero degrees to be 1.0 regardless of all other parameters, which facilitates curve fitting.

Swami and Chandra (1987) found that, wind angle (α) and building side ratio (s) are to significantly influence Cp, so the recommended correlation does not have roof angles as a variable. This could be due to some conflicting data as well as to the fact that only wall Cp distributions are being correlated.

With the significant parameters obtained, the actual form is chosen. The nature of the data imposed several constraints.

a) Regardless of all other parameters, the normalised Cp must always be equal to 1.0 for zero degrees wind angle.

b) The terms containing the roof angles in the equation must disappear from the equation when they are zero, leaving the rest of the equation intact.

c) Since the natural logarithm of the side ratio is the significant parameter this term will become zero for S=1. These terms must be chosen so that they do not affect the other terms of the equation. To abide by these constraints, terms containing side ratio as well as roof angles were combined with sine functions of wind angle so that these terms would vanish for wind angle of zero degrees. The final recommended equation is:

$$\mathbf{NCp = ln\ (C0 + C1.Sin(\alpha/2) + C2.Sin^2(\alpha) + C3.\ Sin^3(2.\alpha.G) + C4.Cos(\alpha/2) + C5.G^2.Sin^2(\alpha/2) + C6.Cos^2(\alpha/2))} \quad \textbf{(3.10)}$$

where:

NCp = the normalised C_p

ln = the natural logarithm

α = the wind angle in degrees

G = the natural log of the side ratio.

The coefficients of the equation are:

C0 = 1.248 C1 = -0.703 C2 = -1.175 C3 = 0.131 C4 = 0.769

C5 = 0.07 C6 = 0.717

The actual C_p is calculated by multiplying the normalised value by the C_p at zero incidence which is 0.6.

3.3.2.4 Effect of Surrounding Buildings

Surrounding buildings can have significant effects on the airflow through buildings. Correlations for change in Cp due to the presence of three specific surrounding patterns rectangular, hexagonal, and a single neighbouring building were carried out by Swami and Chandra (1987) from the data available in Wiren (1985). Since these are only specific effects, they are not presented here. However, correction factors were developed based on the generalised shielding coefficients of Sherman and Grimsrud (1982).

3.3.2.5 Correction for Shielding Effects

The factors for reduction in airflow due to shielding were calculated based on the generalised shielding coefficients of Sherman and Grimsrud (1982). Taking their Shielding class I to represent a totally unobstructed house, the correction factor to be applied for the other classes was calculated by taking the ratio of the Sherman and Grimsrud coefficients with respect to the unshielded class. The calculated correction factors are given in Table 3.2. Note that the correction factors given in the table should be applied to the ventilation flow rate and not Cp.

$$\textbf{Correct ach} = \textbf{ach} \cdot \textbf{SCF} \qquad \textbf{(3.11)}$$

where:

ach = air changes per hour

SCF = shielding correction factor.

Table 3.2: Correction factors for generalised shielding (Swami & Chandra, 1988).

Shielding class	Correction factor (SCF)	Description
I	1.0	No obstruction or local shielding
II	0.88	Light local shielding with few obstructions (e.g., a few trees or a shed in the vicinity)
III	0.74	Moderate local shielding; some obstructions within two house heights (e.g., thick hedge or fence and nearby buildings)
IV	0.57	Heavy shielding; obstruction around most of perimeter building or trees within five building heights in most directions (e.g., well developed tract houses)
V	0.31	Very heavy shielding; large obstruction surrounding perimeter within two house heights (e.g., typical downtown area)

3.3.2.6 Presence of Garage or Wing Walls

The presence of a garage wall or wing wall protruding from a wall will drastically affect the value of Cp depending on the approach wind angle. Studies by Chandra et al. (1983) show that for an angle of up to 90 degrees between the wing wall and the approach wind (as shown in the Figure 3.13), the value of Cp on the wall may be assumed to be the value at zero incidence. For angles in the positive direction beyond 90 degrees, the effect of the garage or wing wall is minimal and therefore no modification is suggested. For angles in the negative direction, as shown in Figure 3.13, the presence of the wing wall produces negative pressures as if the wind is approaching from the leeward side. In this case, it is suggested that the window areas of the wall may be added to the window areas of the leeward wall of the building.

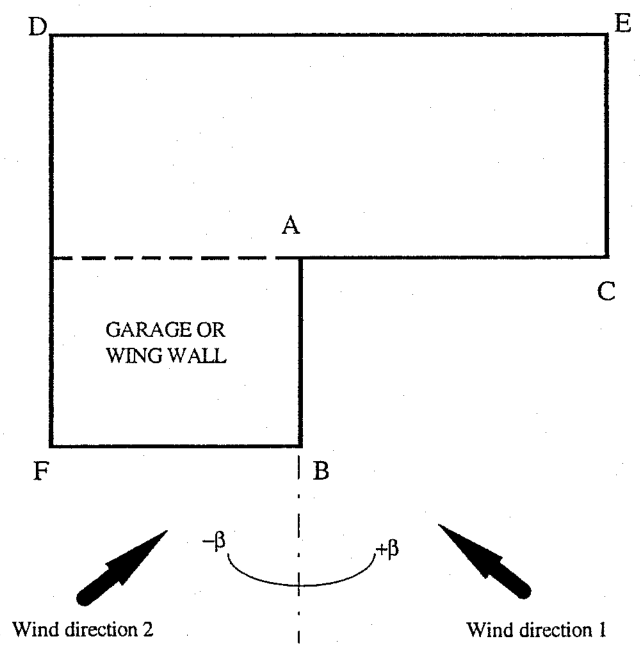


Figure 3.13: Correction/modification to Cp for the presence of garage or wingwalls (based on Swami & Chandra, 1988).

Correction/modification for wall AC should be as follows:

a) For β in the positive direction up to 90°, Cp may be taken as the value at zero incidence (i.e., Cp = 0.6).

b) For β in the positive direction greater than 90°, no correction is suggested.

c) For β in the negative direction up to -90°, include the apertures in wall AC as if they are in wall EC and use normal equations.

3.3.2.7 U-Shaped Buildings

Figure 3.14 shows a typical U-shaped building. Since measured data are unavailable for this common building shape also, common sense guidelines are recommended. The Cp of the wall forming the inner surfaces of the U should be modified as follows:

- For approach wind up to 45 degrees on both sides of line o-o, the Cp values of all the U-walls may be taken as the value at zero incidence since positive pressures will be experienced by those walls.

- For angles beyond 45 degrees and up to 60 degrees on both sides of line o-o, the wall facing away from the wind approach is likely to be experiencing suction conditions, while the other two walls are likely to be experiencing positive pressures.

- The wall facing away from the wind direction should be treated as if it were a leeward wall, and its aperture area should be added to the aperture area of the leeward wall of the building.

- The Cp for the other two walls of the U may be taken as Cp at zero incidence. For angles beyond 60 degrees, the flow is likely to bypass the U region, and all walls of the U will experience suction. Therefore, the areas of windows on these walls should be added to the window areas of the appropriate leeward wall. Figure 3.14 illustrates the different cases.

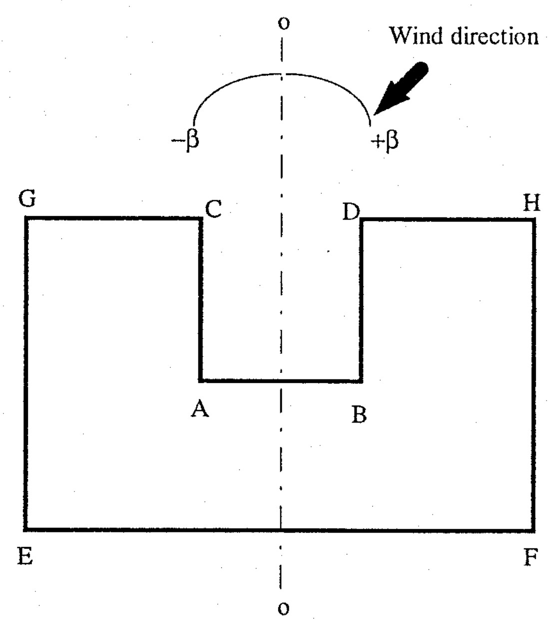


Figure 3.14: Modification to Cp for U-shaped buildings (based on Swami & Chandra, 1988).

The following modification to Cp for walls AB, AC, and BD is suggested:

a) For angles β up to ±45°, Cp for walls AB, AC, and BD may be assumed to be the value at zero incidence (i.e., Cp = 0.6).

b) For positive β up to 60°, walls AB and AC may be taken to be at zero incidence (i.e., Cp = 0.6). Window(s) on wall BD may be added to those in wall EF.

c) For negative β up to 60°, walls DB and AB may be taken to be at zero incidence (i e. Cp = 0.6). Window(s) in AC may be added to those in wall EF.

d) For angle β beyond ±60°, the apertures in all three walls should be treated as if they are in leeward region. Thus, add all the aperture areas in wall AC, AB, and BD and include them as areas in wall GE for $\beta > +60°$, and in wall HF for $\beta < -60°$.

3.3.2.8 Correction for Window Type

Area of the window is defined as the open window area. For sliding or hung windows, open window area is typically 40% of the rough opening in the wall. For fully operable windows (e.g., awing or casement windows) assume A_i to be the entire glazed area. The window may or may not have insect screening. Correction for window type and insect screening is presented below by multiplying the flow by the following factors:

1. Fully open awing window, no screen: 0.75
2. Awing window and 60% porosity insect screen: 0.65
3. 60% porosity insect screening: 0.85
4. No data available for blockage in casement windows when the winds are at an oblique angle.

3.3.2.9 Terrain Effects

Wind engineers have developed five standard terrain classifications, ranging from open ocean fronts to the center of large cities. The terrain enters into the calculation of the reference wind speed, since the terrain affects the shape of the approach wind velocity profile.

Another issue with terrain effects is whether the shape of the velocity profile affects the Cp directly. Akins (1976) conducted a systematic investigation of five velocity profiles of high-rise buildings and found that Cp dependence on terrain virtually vanishes if the Cp is defined with wind velocities at local height rather than at some fixed height . No one has yet conducted a systematic study for low-rise buildings, encompassing all five terrain classes.

Table 3.3: Terrain parameters for standard terrain classes (Swami & Chandra, 1988).

Class	b	a	Description
I	0.10	1.30	Ocean or other body of water with at least 5 km of unrestricted expanse
II	0.15	1.0	Flat terrain with some isolated obstacles
III	0.20	0.85	Rural areas with low buildings
IV	0.25	0.67	Urban, industrial or forest areas
V	0.35	0.47	Centre of large city

Most available data are for terrain classes II or III (see Table 3.3 for terrain classifications) and the data are conflicting. Thus, it has chosen to ignore the effect of velocity profile shape on Cp.

3.3.3 Wind-Driven Ventilation, Method III

A simplified analysis to calculate the air change rates through the openings in naturally ventilated desert buildings developed by Mathews et al. (1992).

The most commonly used simplified procedure is based on the following correlation equation, derived from measurements in buildings with diverse geometrise and window placement:

$$\mathbf{ach = a + b.V_{ref}} \tag{3.12}$$

where:

ach = the air change rate per hour

V_{ref} = the wind speed at a reference height

a & b = empirically determined coefficients.

Measurements however showed that an unsatisfactory correlation exists between air change rates and wind speeds when using the abovementioned equation. Errors resulting from this approach were discussed by Mathews and Richards (1990) and it was shown that the following equation, as a first approximation, should provide a better solution for inclusion in a thermal model:

$$ach.\frac{Vol}{A} = a_{new} + b_{new}.V \qquad \textbf{(3.13)}$$

This equation retains some of the more important characteristics of a building, such as the internal volume (Vol) as well a the total open areas (A). Measurements of natural ventilation flow rates were done for full scale and model buildings.

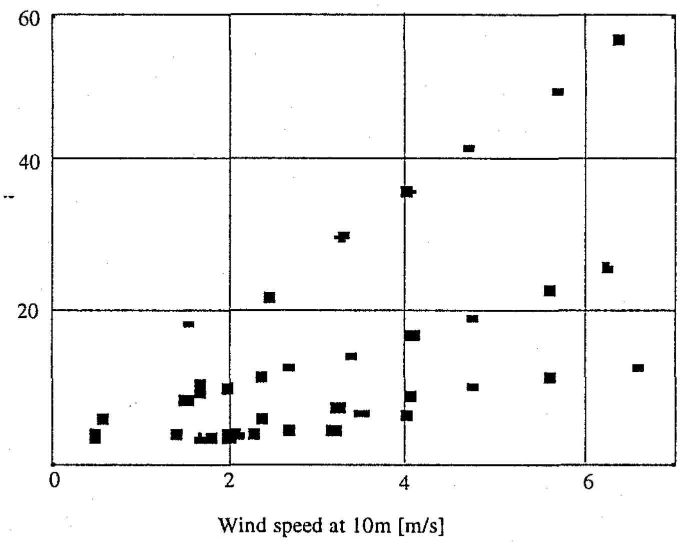


Figure 3.15: Measured natural ventilation flow rates (ach) versus the wind speed (V) measured at a reference height of 10 m (Mathews et. al., 1992).

These are given in Figures 3.15 and 3.16 respectively using the approaches suggested by Equations 3.12 and 3.13. A curve (Equation 3.13) was fitted to the data in Figure 3.16 which represents a good fit with a regression coefficient of 0.95. On the other hand, it would be senseless to try and fit a single curve (Equation 3.12) to the data in Figure 3.15. It is clear that the proposed Equation 3.14 is an improvement on the popular

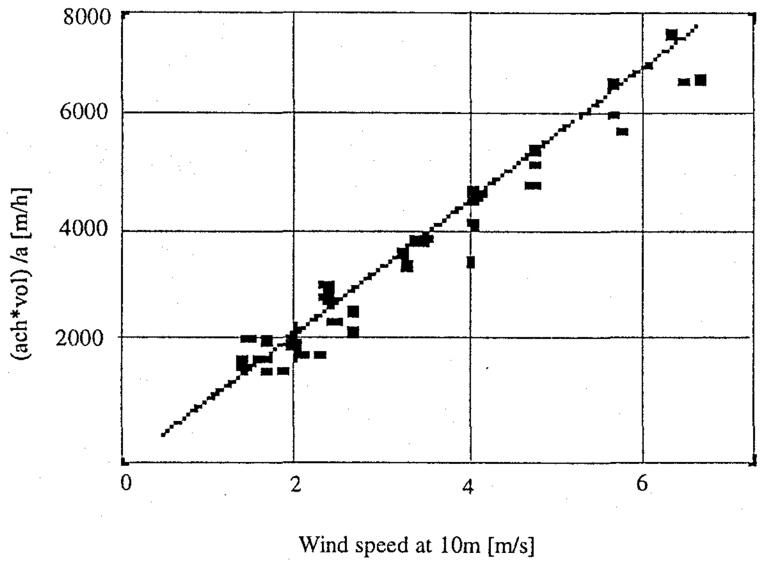


Figure 3.16: Measured values of (ach .Vol)/A versus the wind speed (V) measured at a reference height of 10 m (Mathews et. al., 1992).

Equation 3.13. The proposed Equation 3.14 was thus further developed by the Centre for Experimental and Numerical Thermoflow, Dept. of Mechanical Engineering, University of Pretoria, in South Africa.

The new equation attempts to account for building geometry, wind shading and direction, as well as internal volume. This equation applicable to a building with two windows open to the outside and situated on directly opposite walls of the building. The derivation of the equation is based on measurements of natural ventilation flow rates for full scale and model buildings. The equation for the non-dimensional ventilation parameter is:

$$\frac{ach.Vol}{C_d A.V_h} = 3500\tanh(2.8(abs(\sin\beta))^{0.85}).\tanh\left(0.1\left(\frac{x}{h}\right)^{0.8}\right) \tag{3.14}$$

where:

β = the wind direction as defined in Figure 3.17

x/h = a wind shading factor with *x* the fetch length to the closest building of comparable height on the windward side and *h* the roof height of the building

$V_h = V_{ref}$ = the free stream wind velocity at roof height (m/s).

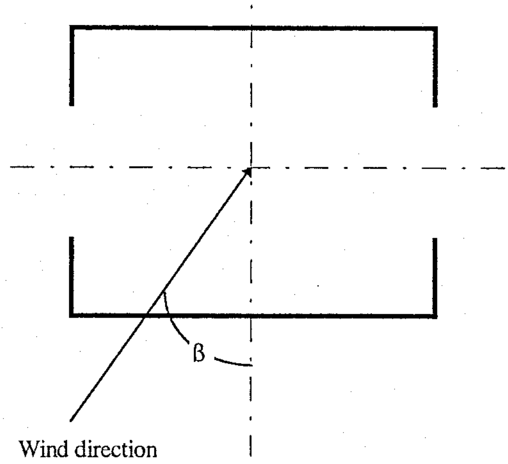


Figure 3.17: Definition of wind direction β.

The permeability coefficient C_dA for the building is calculated as:

$$C_d A = \frac{\prod_{i=1}^{n} (C_{d_i} A_i)}{\sqrt{\sum_{i=1}^{n} \left[\prod_{\substack{i=1 \\ i \neq j}}^{n} (C_{d_i} A_i)^2 \right]}} \tag{3.15}$$

where:

n = the number of openings through which the air flows

C_{d_i} = discharge coefficient, 0.60

A_i = the aria respectively of opening i.

Any number of internal walls with openings is allowed.

3.4 VENTILATION FROM THERMAL BUOYANCY (STACK EFFECT)

Temperature differences between the inside and outside cause density differences, causing pressure differences that drive infiltration. During the heating season, the warmer inside air rises and flows out of the building near its top. It is replaced by colder outside air which enters the building near its base. During the cooling season, the flow directions are reversed and generally are less significant because inside-outside temperature differences are smaller.

Stack effect ventilation is usually practised by designers in hot and arid climates e.g., Baer (1983) and Crowther (1980). In such climates daytime ventilation is impossible and the building gradually heats up during the day. At night the cool air flushes the building. The night time wind speeds are usually low; thus the wind effect is augmented by the stack effect by placing high outlets e.g., operable sky-lights and low inlets.

The stack must terminate above the roof peak so that the stack top is always under suction compared to the lower inlet level. Otherwise, a wind coming from the wrong way can introduce the hot stack air into the room (see Schubert and Hahn (1983) for ventilator designs that might be especially well suited for this purpose). One must also be careful in designing Trombe walls for stack ventilators in the summer. Unless the Trombe wall is insulated at the house side in the summer and is vented at a high point at the top, unwanted heating may result. The physics of stack ventilation are well explained in ASHRAE 1985, Chapter 22. In general, the stack effect is weak. As discussed by Chandra et al. (1983), the cross-ventilation airflow from a 2.7m/s wind can overcome that from a 2.4m stack at 43°C.

Qualitatively, the pressure distribution over the building in the heating season takes the form shown in Figure 3.18.

The point on the vertical surface where the interior and exterior pressures are equal is called the Neutral Pressure Level (NPL). Above this point for the normal case during the heating season the interior pressure is greater than the exterior; below, the greater exterior pressure causes air flow into the building.

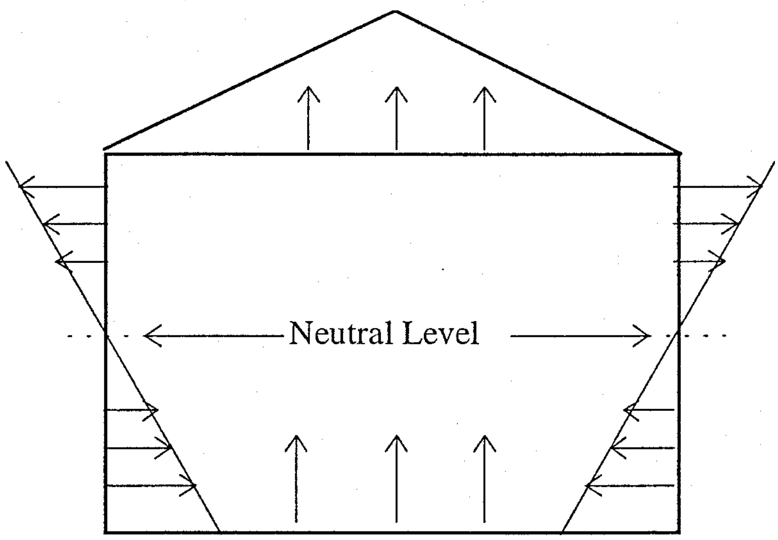


Figure 3.18: Pressure differences caused by stack effect for a typical structure. Arrows point from higher to lower pressure and indicate direction of air flow (ASHRAE, 1985).

The location of the NPL at zero wind speed depends on the distribution of openings in the shell. If this distribution is uniform over the shell area, the NPL will be at the mid height of the building, as shown in Figure 3.18. However, if the leakage is dominated by one large opening, the NPL will be found close to the height of that opening.

The NPL depends on the vertical distribution and resistance of openings to air flow. For only one opening or for an extremely large opening relative to others, the NPL is at or near the centre of the opening. For openings uniformly distributed vertically, the NPL is at mid height of the enclosure. Locating the NPL for simple enclosures with openings of known air flow characteristics is straight forward. For example, with two openings having areas A_1 lower and A_2 upper separated by vertical height H, the NPL *h*, measured from the lower opening is given by:

$$h = \frac{H}{1+[(A_1 / A_2)^2 (T_i / T_o)]} \tag{3.16}$$

In Equation 3.16, $T_i > T_O$ If $T_i < T_O$, the ratio T_i / T_O is inverted. If there is not significant building internal resistance, the flow caused by stack effect is:

$$\mathbf{Q = (\mathit{cf})\ A[h(T_i - T_O)/T_i\ ]^{1/2}} \tag{3.17}$$

where:

Q	= air flow (m^3/h)
A	= free area of inlets or outlets, assumed equal (m^2)
h	= height from lower opening to NPL (m)
T_i	= average temperature of indoor air in height h (K)
T_O	= temperature of outdoor air (K)
cf	= conversion factor, including a value of 65% for effectiveness of openings; this should be reduced to 50% if conditions are not favourable (*cf* = 10360).

Equation 3.17 applies when $T_i > T_o$. If $T_i < T_o$, replace T_i in the denominator with T_o and interchange the order of the terms in the numerator. The height h is calculated using Equation 3.16.

Greatest flow per unit area of openings is obtained when inlets and outlets are equal; Equation 3.16 is based on this equality. Increasing the outlet area over inlet area, or vice versa, increases air flow but not in proportion to the added area. When openings are unequal, use the smaller area in the equations, and add the increase, as determined from Figure 3.19.

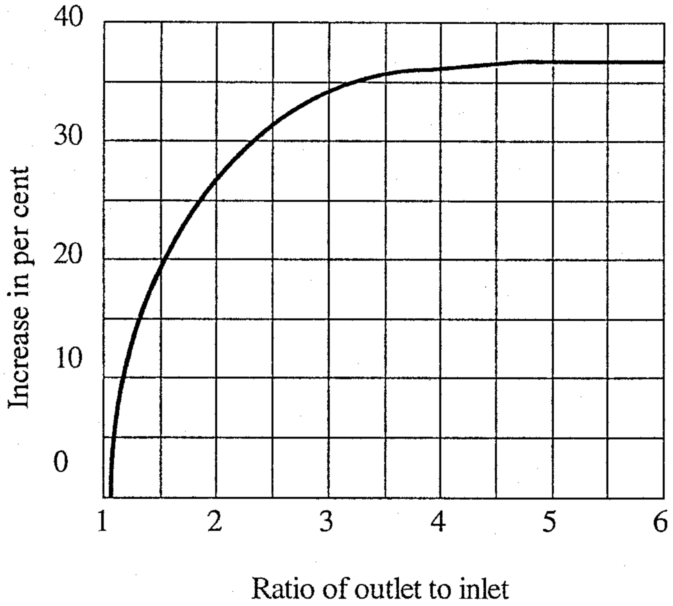


Figure 3.19: Increase in flow caused by excess of one opening over another (ASHRAE, 1985).

3.5 NATURAL VENTILATION GUIDE-LINES

A set of general rules which should be observed in designing for ventilation are summarised in ASHRAE (1985):

1. Systems using natural ventilation should be designed for effective ventilation regardless of wind direction. Ventilation must be adequate when the wind does not come from the prevailing direction.

2. Inlet openings should not be obstructed by buildings, sign-boards or indoor partitions.

3. Greatest flow per unit area of total opening is obtained by inlet and outlet openings of nearly equal areas.

4. The neutral pressure level tends to move to the level of any single openings, resulting in pressure reduction across the opening. Two openings on opposite sides of a space increase the ventilation flow. If the openings are at the same level and near the ceiling,

much of the flow may by-pass the occupied level and be ineffective in diluting contaminants at the occupied level.

5. There must be vertical distance between openings for temperature difference to produce natural ventilation; the greater the vertical distance, the greater the ventilation.

6. Openings in the vicinity of the NPL are least effective for thermally induced ventilation.

7. Openings with areas much larger than calculated are sometimes desirable when anticipating increased occupancy or very hot weather. The openings should be accessible to and operable by occupants.

8. When both wind and stack pressures act together, even without interference, estimated resulting air flow is not equal to the two flows separately. Flow through any opening is proportional to the square root of the sum of the squares of the two flows calculated separately.

CHAPTER 4

HEAT AND MASS TRANSFER OF THE SYSTEM

4.1 INTRODUCTION

This chapter focuses on the method of calculating the efficiency of the system. Since, there is no specific information or formulae relevant to calculating efficiency of this particular system, an extensive investigation and personal interviews were conducted which form the basis of this chapter. Based on the equations presented in this chapter, a parametric study was undertaken to investigate the effects of various parameters on the efficiency of the system.

The efficiency of the system is controlled by a number of factors such as the average heat and mass transfer coefficients. Assuming that the filaments carrying the water in the system are tube bundles, the convection heat transfer associated with cross flow over a bank of tubes is used to calculate average convection heat transfer coefficient. Then a mass transfer coefficient is conveniently defined using the log-mean humidity ratio difference to determine system efficiency.

4.2 FLOW ACROSS BANKS OF TUBES

Heat transfer to or from a bank (or bundle) of tubes in cross flow is relevant to numerous industrial applications, such as steam generation in a boiler or air cooling in the coil of an air conditioner. The geometric arrangement is shown schematically in Figure 4.1. Typically, one fluid moves over the tubes, while a second fluid at a different temperature passes through the tubes. In this section we are specifically interested in the convection heat transfer associated with cross flow over the tubes.

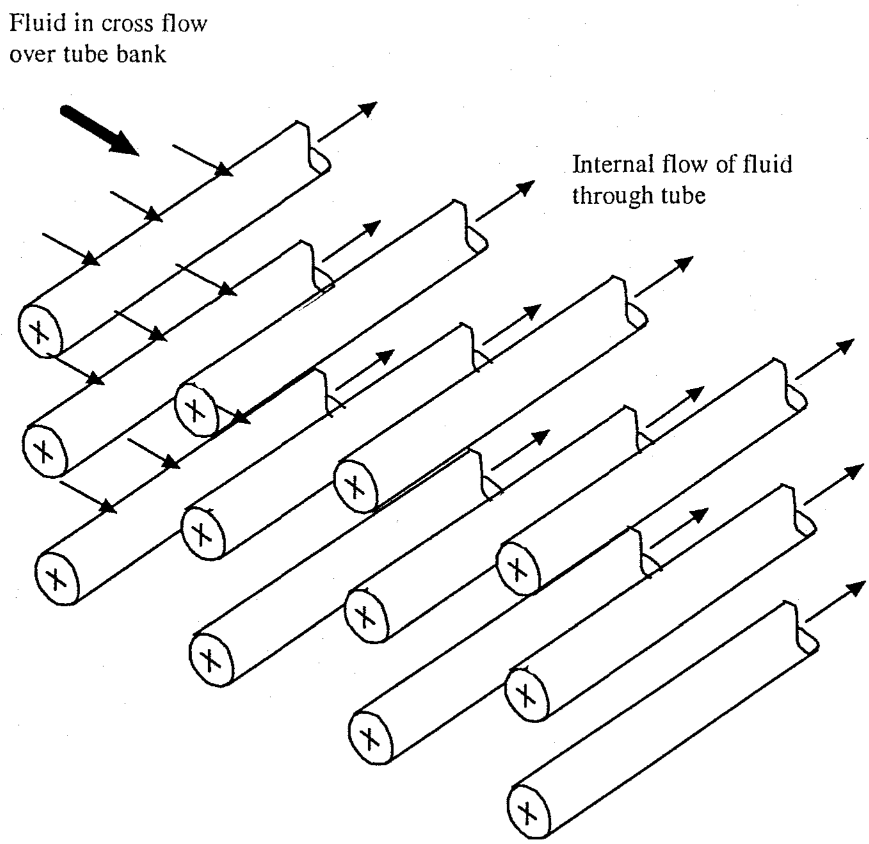


Figure 4.1: Schematic of a tube bank in cross flow (Incropera, 1990).

The tube rows of a bank are either staggered or aligned in the direction of the fluid velocity V (Figure 4.2). The configuration is characterised by the tube diameter D and by the transverse pitch S_T and longitudinal pitch S_L measured between tube centers. Flow conditions within the bank are dominated by boundary layer separation effects and by wake interactions, which in turn influence convection heat transfer.

Heat transfer from banks of tubes depends on flow conditions. A tube in the first row of the bank differs from a single tube as regards flow conditions. This is because of the presence of other tubes in the same row, and also the influence of the following (second) row. Therefore, heat transfer for the first row differs somewhat from heat transfer for a single tube. In the inner rows of the bank, the tubes are in contact with a flow of high degree of turbulence which leads to an increase in heat transfer (Zukauskas & Ulinskas, 1988).

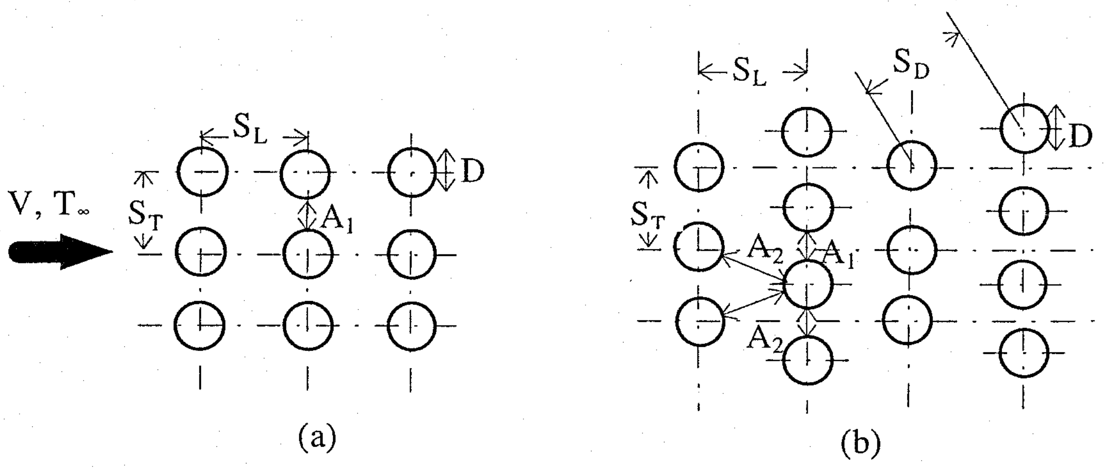


Figure 4.2: Tube arrangements in a bank; (a) aligned; (b) staggered (Incropera, 1990).

Even earlier investigations indicated that, from the heat transfer standpoint, the staggered arrangement is more effective. The in-line tubes tend to give a somewhat lower pressure drop and poorer heat transfer because the flow tends to be channeled into high-velocity regions in the center of the lanes between the tube rows (Fraas, 1989).

4.2.1 Local Heat Transfer Coefficients in Tube Banks

Both the heat transfer coefficient and the friction factor depend on the tube spacing. So many combinations of the tube spacing, both transverse and parallel to the direction of flow, are possible that a large number of curves is required. Two such sets of curves are given in Figure 4.4. In these curves the friction factor, or pressure-loss coefficient, is the fraction of a dynamic head lost per bank of tubes, with the dynamic head spaced on the nominal fluid velocity through the minimum flow-passage area between the tubes. Note that where the transverse spacing is large and the axial spacing is small the heat transfer coefficient and friction factor approach the values for a smooth passage.

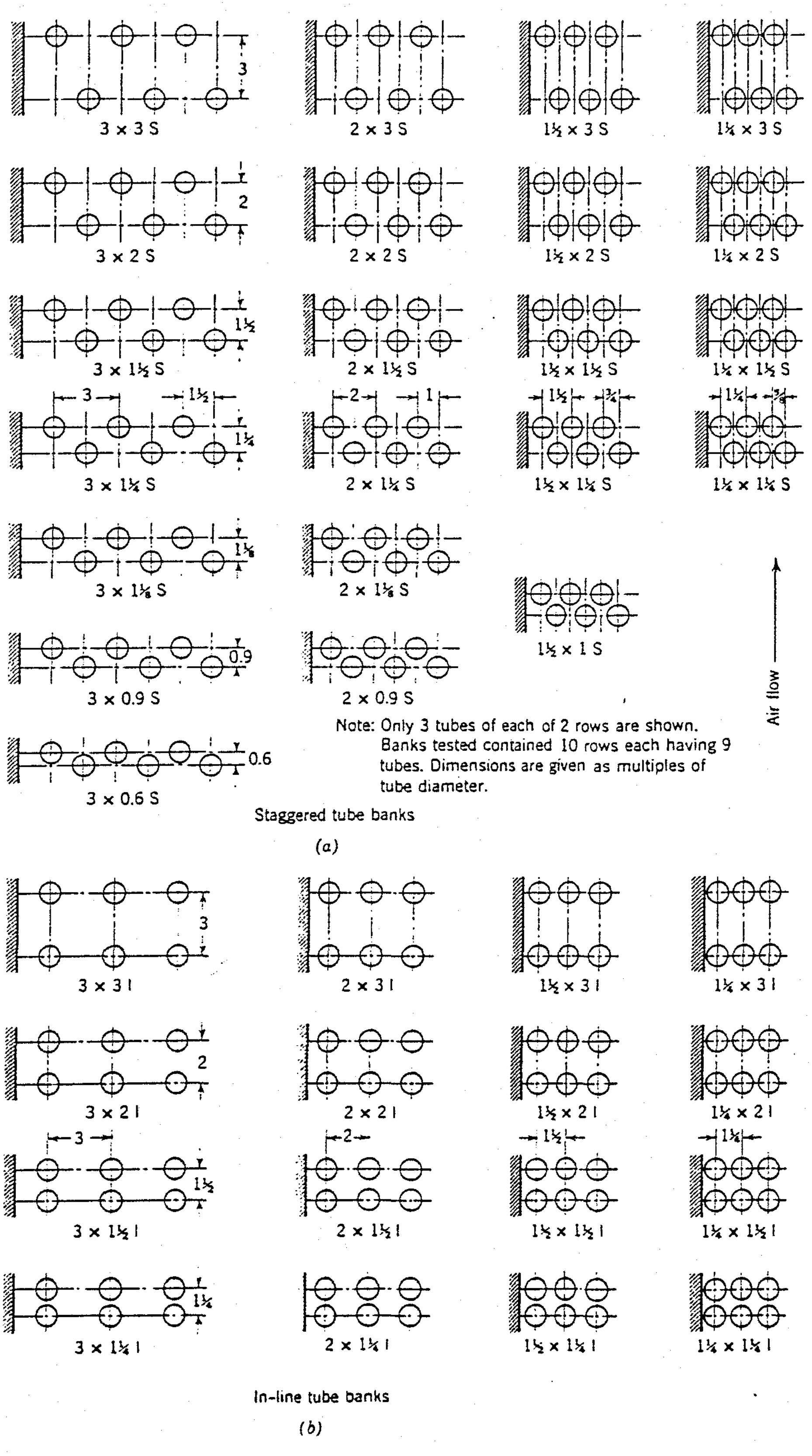


Figure 4.3: Tube bank arrangements used to obtain the data for Figure 4.4 (Fraas, 1989).

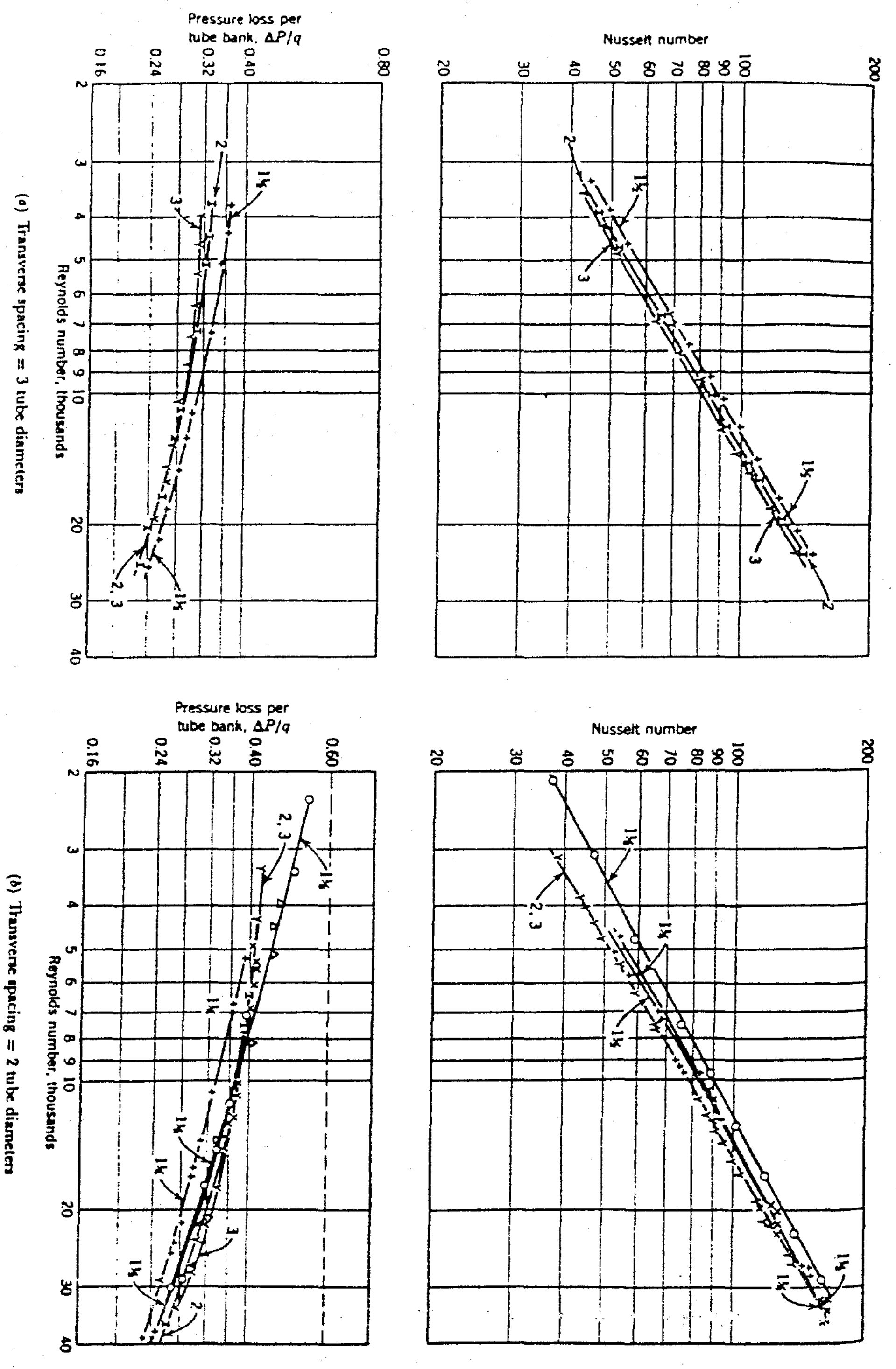


Figure 4.4: Heat transfer and pressure loss coefficients for cross-flow over staggered banks of bare tubes.

R_e and $\Delta p/q$ are based on the tube diameter and the mass flow rate through the minimum gap between the tubes; $\Delta p/q$ is the ratio of the pressure drop per bank to the dynamic head. The figures on the curves indicate the tube pitch parallel to the flow in the tube diameters. (A) Transverse spacing = 3 tube diameters; (b) transverse spacing = 2 tube diameters; (c) transverse spacing = 1 1/2 tube diameters; (d) transverse spacing = 1 1/4 tube diameters (Fraas, 1989).

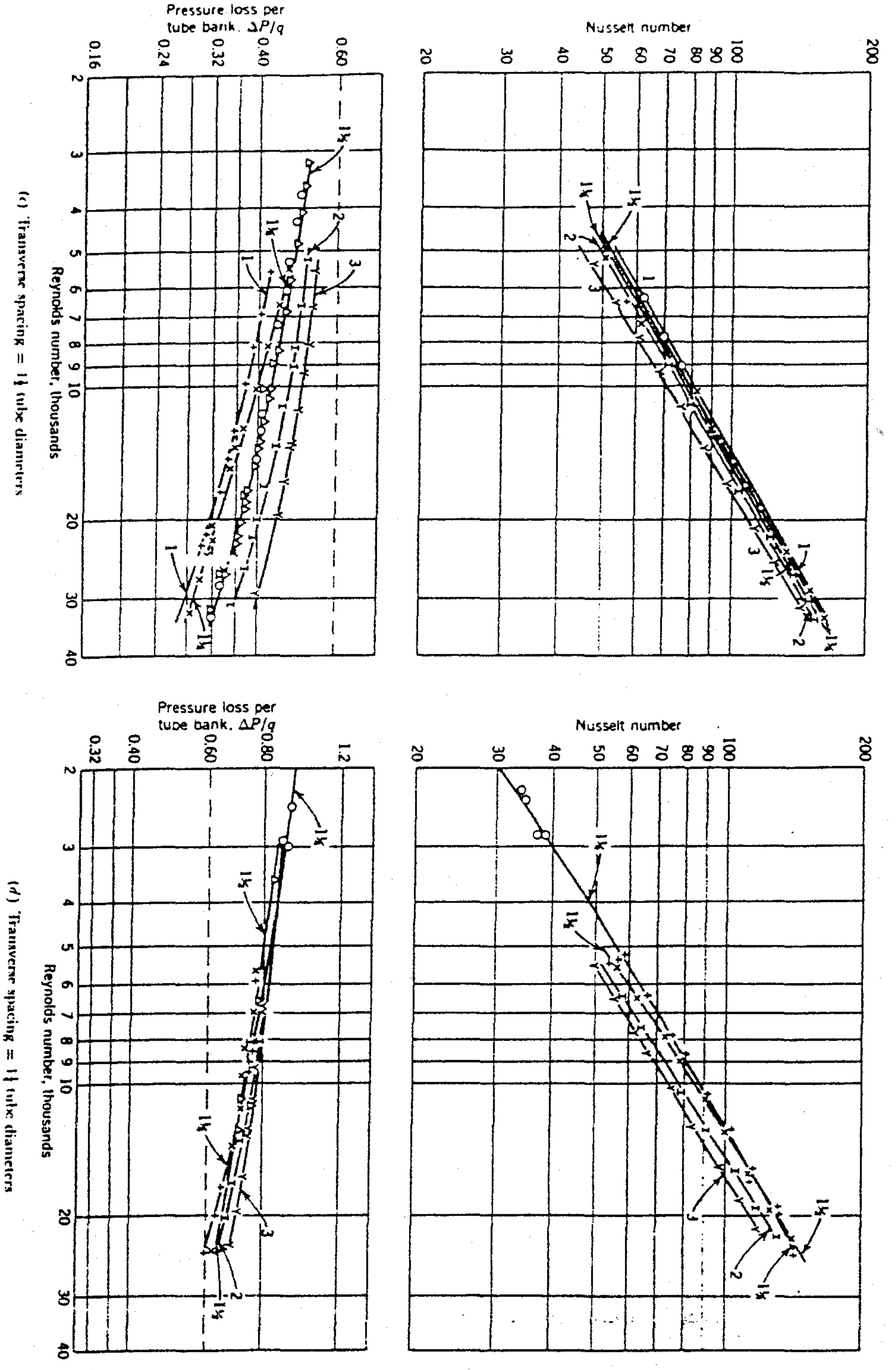


Figure 4.4: (Continued).

Where the transverse spacing is small (i.e., where the gap between the tubes is 1/4 or 1/2 of a tube diameter) the friction factor approaches unity (i.e., the pressure loss per bank approaches that for flow through a flow nozzle with no diffuser to recover the dynamic pressure). The resulting turbulence, of course, greatly increases the heat transfer coefficient and the pumping power required (Fraas, 1989).

More advanced investigations followed later. To start with, the influence of bank type and the relative transverse and longitudinal pitches on the intensity of heat transfer were investigated (Pierson, 1937, Kuznetsov, 1937 and Antuf'iev & Kozachenko, 1938) . The majority of experimental investigations showed that the heat transfer in a tube in a bank is greater than for a single tube and depends on the longitudinal and transverse pitches. With the development of the theory of similarity, the experimental results were generalised according to this theory and were expressed by the equation of similarity:

$$\mathbf{Nu = c\, Re^{m}} \qquad \mathbf{(4.1)}$$

where:

Nu = Nusselt number

Re = Reynolds number

Various values for "c" and "m" were proposed, depending on the pitch. This allowed the transition to a wider generalisation of data obtained by different authors.

Antufiev and Beletsky (1948) proposed single formulas for the calculation of heat transfer and developed nomograms based on these formulas. In Grimison's work (1937), based on Pierson's (1937) experimental data and investigations by other authors, tables of c and m were derived. When the generalisations of different authors were compared, discrepancies of several tens of percent were noticed. This was due in part to inadequacies of experiments and to the different methods of generalisation. Some investigations were conducted on real specimens, and often on fully heated models. Later, in the majority of experiments, the method of local modelling was employed, i.e., one tube in a row was either heated or cooled. The method of local modelling is based on the theory of similarity and is subject to special investigations (Kirpichev & Mikheiev, 1936).

When generalising the results in all the mentioned works, the diameter of the tube in a bank is taken as the characteristic dimension, and the velocity in the minimum flow area of the bank or of the transverse row of tubes as the design velocity.

Analysis of the functions proposed by different authors showed that, in the majority of cases, the choice of the determining temperature was the reason for the discrepancy. This is the temperature at which the physical parameters included in the similarity variables are evaluated.

Mikheiev (1968), after studying heat transfer from a single tube in cross flow and analysing the results of works by different authors regarding banks of tubes, came to the conclusion that the influence of temperature head and direction of heat flow on heat transfer is best accommodated when the physical quantities are evaluated at the free-stream temperature. Kuznetsov (1952) established that if the physical gas constants are evaluated at the free stream temperature, then the results can be generalised with sufficient accuracy without the need to introduce an additional parameter to account for the temperature factor. As was mentioned above, the choice of the determining temperature was often the reason for the different ways the influence of tube location in a bank on heat transfer was accounted for, and which affected the formulas.

The influence of the pitches of tube banks on heat transfer are described by Kirpichev & Eigenson (1936) and Bressler (1958). In the former, heat transfer for a tube was determined as a function of the distance to the turbulence generating grid. It was shown that heat transfer increases rapidly when this distance decreases. In the second work, it was experimentally established that heat transfer for a tube in a bank was not a function of the initial turbulence of the flow. Apparently, the bank created a corresponding turbulence mechanism.

Numerous investigations showed that heat transfer from the first row of tubes in a bank amounts to 60% of heat transfer from the third row, after which it begins to stabilise.

In the aforementioned works, heat transfer was studied at moderate (1 x 10^5) Reynolds numbers. Later, heat transfer of condensed banks was studied in more detail by Kays & London (1964) together with heat transfer at high Re. In investigations with air, it was found that at Re higher than 1.2 x 10^5 a sharp intensification of heat transfer takes place, which is indicated by the break in the heat transfer curve (Lyapin, 1956).

Works dedicated to the investigation of local heat transfer along the circumference of a tube have great significance for the understanding of the physical mechanism of heat transfer in tube bank systems in cross flow.

Kurzhilin and Schwab (1935) were the authors of the first work on a technique for the direct determination of local heat transfer for a single cylinder. Mikhailov (1939) studied in detail the variation of heat transfer across the circumference of a tube in a bank in a cross flow of air. Experiments with banks of tubes in a staggered arrangement showed that the distribution of heat transfer along the perimeter of a tube in an inner row was quite similar to the distribution of heat transfer in a single tube with maximum heat transfer at the frontal critical point. In in-line banks, a zone of weak circulation of the liquid was observed between the inner tubes; consequently, heat transfer from the front and rear parts of the tubes was insignificant. Therefore, unlike staggered banks, maximum heat transfer for in-line banks occurred at an angle 50 ° from the frontal point.

For airflow across tube bundles composed of 20 or more rows ($N_L \geq 20$), Zhukauskas (1972) has proposed a correlation for tube bundles which takes into account wide ranges of Reynolds numbers and property variations. The correlating equation takes the form:

$$\overline{Nu}_D = C\,\mathrm{Re}_{D\max}^{m}\,\mathrm{Pr}^{0.36}\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_s}\right)^{1/4} \qquad \textbf{(4.2)}$$

where:

$\overline{Nu}_D$ = Nusselt number

$Re_{D\,max}$ = Reynolds number based on the maximum fluid velocity occuring within the tube bank

Pr = Prandtl number

Pr_s = Prandtl number at water temperature

C, m = constants

and all properties except Pr_s are evaluated at the arithmetic mean of the fluid inlet and outlet temperatures; the constants "C" and "m" are listed in Table 4.1. For single cylinder the constants C and m are listed in Table 4.2.

Table 4.1: Constants of Equation 4.2 for the tube bank in cross flow(Zhukauskas, 1972)

CONFIGURATION	$Re_{D\,max}$	C	m
Aligned	$10\text{-}10^2$	0.80	0.40
Staggered	$10\text{-}10^2$	0.90	0.40
Aligned	$10^2\text{-}10^3$	Approximate as a single	
Staggered	$10^2\text{-}10^3$	(isolated)cylinder.	
Aligned	$10^3\text{-}2\times10^5$	0.27	0.63
Staggered	$10^3\text{-}2\times10^5$	$0.35(S_T/S_L)^{1/5}$	0.60
Aligned	$2\times10^5\text{-}2\times10^6$	0.021	0.84
Staggered	$2\times10^5\text{-}2\times10^6$	0.022	0.84

The Reynolds number $Re_{D,max}$ for the foregoing correlations is based on the maximum fluid velocity occurring within the tube bank.

$$Re_{D,\max} \equiv \frac{\rho V_{\max} D}{\mu} = \frac{V_{\max} D}{\nu} \qquad \textbf{(4.3)}$$

where:

ρ = mass density (kg/m^3)

V_{max} = maximum velocity (m/s)

D = diameter (m)

μ = viscosity (kg/s . m)

ν = kinematic viscosity (m^2/s)

Table 4.2: Constants of Equation 4.2 for the circular cylinder in cross flow (Zhukauskas, 1972).

Re_D	C	m
1-40	0.75	0.4
40-1000	0.51	0.5
10^3-2×10^5	0.26	0.6
2×10^5-10^6	0.076	0.7

For the aligned arrangement, V_{max} occurs at the transverse plane A_1 of Figure 4.2 (a), and from the mass conservation requirement for an incompressible fluid:

$$V_{max} = \frac{S_T}{S_T - D} V \tag{4.4}$$

where:

S_T = transverse pitch (m)

V = wind speed (m/s)

For the staggered configuration, the maximum velocity may occur at either the transverse plan A_1 or the diagonal plane A_2 of Figure 4.2 (b). It will occur at A_2 if the rows are spaced such that:

$$2(S_D - D) < (S_T - D) \tag{4.5}$$

Where, S_D is diagonal pitch (m).

The factor of "2" results from the bifurcation experienced by the fluid moving from the A_1 to the A_2 planes. Hence V_{max} occurs at A_2 if:

$$S_D = \left[S_L^2 + \left(\frac{S_T}{2}\right)^2\right]^{1/2} < \frac{S_T + D}{2} \tag{4.6}$$

Where, S_L is longitudinal pitch (m).

In which case it is given by:

$$V_{max} = \frac{S_T}{2(S_D - D)} V \tag{4.7}$$

If V_{max} occurs at A_1 for the staggered configuration, it may again be computed from Equation 4.4.

The need to evaluate fluid properties at the arithmetic mean of the inlet ($T_i = T_\infty$) and outlet (T_o) temperatures is dictated by the fact that the fluid temperature will decrease or increase, respectively, due to heat transfer to or from the tubes. If the fluid temperature change, $|T_i - T_o|$, is large, significant error could result from evaluation of the properties at the inlet temperature. If $N_L < 20$, a correction factor may be applied such that:

$$\overline{Nu}_D|_{(N_L<20)} = C_2 \overline{Nu}|_{(N_L \geq 20)} \tag{4.8}$$

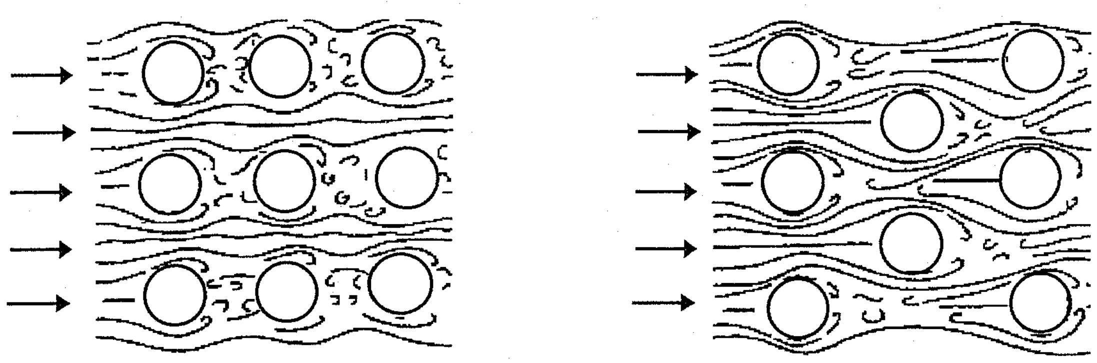
Where C_2 is given in Table 4.3.

Table 4.3: Correction factor C2 of Equation 4.8 for N_L<20 and Re_D>103 (Zhukauskas, 1972)

N_L	1	2	3	4	5	7	10	13	16
Aligned	0.70	0.80	0.86	0.90	0.92	0.95	0.97	0.98	0.99
Staggered	0.64	0.76	0.84	0.89	0.92	0.95	0.97	0.98	0.99

Aligned tubes beyond the first row are in the turbulent wakes of upstream tubes, and for moderate values of S_L convection coefficients associated with downstream rows are enhanced by turbulation of the flow.

Typically, the convection coefficient of a row increases with increasing row number until approximately the fifth row, after which there is little change in the turbulence and hence in the convection coefficient. However, for small values of S_L, upstream rows, in effect, shield downstream rows from much of the flow, and heat transfer is adversely affected. That is, the preferred flow path is in lanes between the tubes and much of the tube surface is not exposed to the main flow. For this reason, operation of aligned tube banks with $S_T/S_L > 0.7$ (Table 4.1) is undesirable. For the staggered array, however, the path of the main flow is more tortuous and a greater portion of the surface area of downstream tubes remains in this path. In general, heat transfer enhancement is favored by the more tortuous flow of a staggered arrangement, particularly for small Reynolds number (Re_D < 100).



(a) (b)

Figure 4.5: Flow conditions for (a) alined and (b) staggered (Incropera, 1990).

The average heat transfer coefficient in cross flow from the empirical correlation due to Hilpert (1933) is the form:

$$\bar{h} = \overline{Nu}\frac{k}{D} \quad \textbf{(4.9)}$$

where:

$\bar{h}$ = average convection heat transfer coefficient (W/m^2. K)

k = thermal conductivity (W/m. K)

4.3 SIMULTANEOUS HEAT AND MASS TRANSFER BETWEEN WATER-WETTED SURFACE AND AIR

A simplified commonly used equation for solving problems involving simultaneous heat and mass transfer was developed with the use of the Lewis relation and gives satisfactory results for most air-conditioning processes (ASHRAE Fundamentals, 1985).

A mass transfer coefficient is conveniently defined using the log-mean humidity ratio difference, as the driving potential:

$$\dot{m}_B = K_m A_W \left(\frac{W_i - W_o}{\ln\left(\dfrac{W_s - W_o}{W_s - W_i} \right)} \right) \qquad \textbf{(4.10)}$$

where:

$\dot{m}_B$ = rate of mass transfer (g/s)

K_m = mass transfer coefficient (g/s. m^2)

A_w = area of water (m^2)

W_i = inlet moisture (kg water/kg air)

W_o = outlet moisture (kg water/kg air)

W_s = moisture at saturation (kg water/kg air)

The coefficient, K_m, is defined as:

$$K_m = \frac{\bar{h}}{C_{pm}} \qquad \textbf{(4.11)}$$

where the humid specific heat, C_{pm} (kJ/kg (dry air). K), of the airstream is, by definition:

$$\mathbf{C_{pm} = (1 + W_i)\, c_p} \qquad \textbf{(4.12)}$$

Where, c_p is specific heat at constant pressure (kJ/kg (dry air). K).

The mass flow rate of air can be calculated by the following equation:

$$\dot{m}_A = \rho . Q \qquad \textbf{(4.13)}$$

Where ρ is mass density (kg/m^3) and Q is airflow rate (m^3/s).

Equation 4.14 can be written by substituting $\dot{m}_B$ in Equation 4.10 by $\dot{m}_A(W_o - W_i)$:

$$\dot{m}_A(W_o - W_i) = K_m A_W \left(\frac{W_i - W_o}{\ln\left(\frac{W_s - W_o}{W_s - W_i}\right)} \right) \qquad \textbf{(4.14)}$$

By calculating W_o from Equation 4.14, the dry-bulb temperature of the outlet air (T_o) can be computed and hence, the efficiency of the system will be known by Equation 4.15.

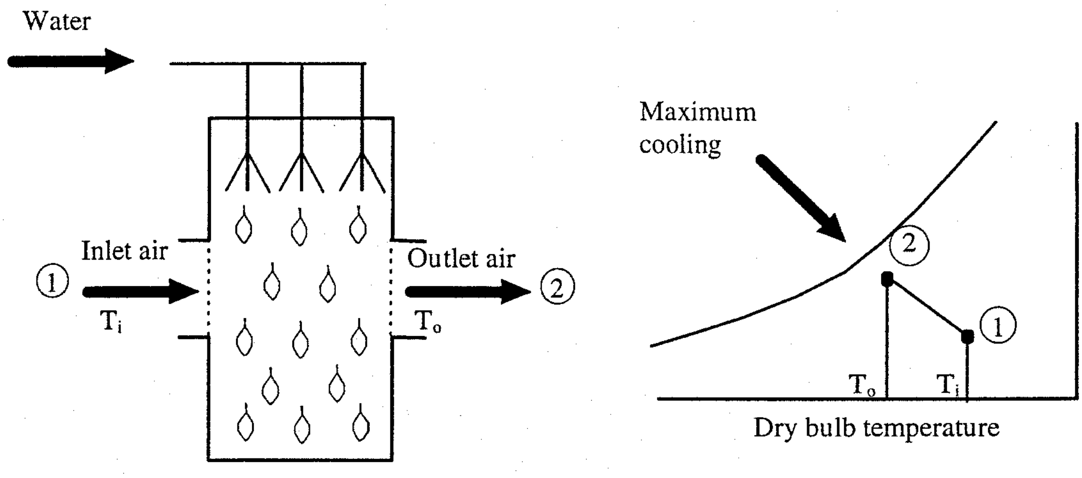
4.4 ADIABATIC SATURATION

The direct evaporative cooling utilises the easy conversion of sensible to latent heat. Non-saturated air is exposed to free and cold water, with both thermally isolated from other influences. Some of the air's sensible heat transfers to the water and converts to latent heat from the evaporation of some of the water. The latent heat follows the water vapour and diffuses into the air.

No heat is gained or lost, but the air temperature falls as its sensible heat evaporates water and converts into latent heat. The added vapour and latent heat raise the air's humidity and heat content, but since the vapour is at the air's saturation temperature, it does not impair the cooling of the air. This exchange of sensible for latent heat tends to progress until the air is saturated and air and water temperatures and vapour pressures equalise. This is called adiabatic saturation because no external heat is involved and saturation is approached purely by conversion of the air's existing sensible heat. In theory, the water temperature remains constant, neither raised nor lowered by contacting the air. This is the optimum arrangement; all evaporation serves to cool the air, none to cool the water (Watt, 1963).

4.4.1 Saturating Efficiency

In the direct evaporative process an air stream is cooled by the evaporation of water into the air, as shown in Figure 4.6 (a). The addition of water vapour increases the latent heat of the air-vapour mixture. As the overall process is adiabatic this increase is offset by a sensible heat reduction and a consequent lowering of the dry-bulb temperature of the air. During the cooling process the wet-bulb temperature of the air remains constant as shown on the psychrometric chart in Figure 4.6 (b). This chart also shows that maximum cooling is achieved when the outlet air is saturated implying equal dry-bulb and wet-bulb temperatures. The lowest possible dry-bulb temperature of air leaving a direct cooler is therefore equal to the wet-bulb temperature of the inlet air.



Direct cooler (a)

Psychorometric chart (b)

Figure 4.6: The direct evaporative cooling process (Mathews et al., 1994).

The inlet and outlet temperatures of the direct cooler are used to define its performance factor. This factor is also called the saturation efficiency because it indicates how close the outlet conditions on the psychrometric chart approach the saturation curve. Equation 4.15 defines the performance factor as the reduction in the dry-bulb temperature of air flowing through the cooler divided by the difference between the dry-bulb and wet-bulb temperatures of the incoming air (Watt, 1963)

$$E_s = \frac{T_i - T_o}{T_i - T_s} = \frac{W_i - W_o}{W_i - W_s} \tag{4.15}$$

where:

E_s = cooling or saturation efficiency

T_i = dry-bulb temperature of the inlet air (K)

T_o = dry-bulb temperature of the outlet air (K)

T_s = thermodynamic wet-bulb temperature of the entering air (K).

From this equation it can be seen that a performance factor of 1 indicates maximum possible cooling of the air flow. The dry-bulb temperature of the saturated outlet air is then equal to the wet-bulb temperature of the inlet air.

Figure 4.7 presents a summary of the steps involved in calculating the cooling efficiency of the system.

Figure 4.7: Calculation procedure for the cooling efficiency.

$$V_{\max} = \frac{S_T}{S_T - D} V$$

$$\mathrm{Re}_{D,\max} \equiv \frac{\rho V_{\max} D}{\mu} = \frac{V_{\max} D}{\nu}$$

$$\overline{Nu}_D = C \,\mathrm{Re}_{D\max}^{m} \,\mathrm{Pr}^{0.36} \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_s} \right)^{1/4}$$

$$\bar{h} = \overline{Nu} \frac{\kappa}{D}$$

$$C_{pm} = (1 + W_i)\, cp$$

$$K_m = \frac{\bar{h}}{C_{pm}}$$

$$\dot{m}_B = K_m A_W \left(\frac{W_i - W_o}{\ln\left(\frac{W_s - W_o}{W_s - W_i} \right)} \right)$$

$$\dot{m}_A (W_o - W_i) = K_m A_W \left(\frac{W_i - W_o}{\ln\left(\frac{W_s - W_o}{W_s - W_i} \right)} \right)$$

$$E_s = \frac{T_i - T_o}{T_i - T_s} = \frac{W_i - W_o}{W_i - W_s}$$

CHAPTER 5

SYSTEM EFFICIENCY AND INDOOR ENVIRONMENTAL CONDITIONS

5.1 INTRODUCTION

This chapter starts with a presentation of the base case building and computer simulation program used in this research. Then, a parametric study is undertaken to investigate sensitivities of the indoor environmental conditions due to the various design parameters. In order to calculate indoor environmental conditions, an intensive study was focused on the following three issues:

- First, ventilation strategies and the various parameters affecting the airflow rate through the building,
- second, system design and the various parameters affecting the evaporative cooling efficiency,
- third, building design and the various parameters affecting the thermal behaviour of the building.

5.2 DESCRIPTION OF THE BASE CASE

A three storey rectangular shaped building structure is assumed for simplicity with dimensions 10 x 10 x 2.8 m for each floor. The building is of cavity wall construction with a tiled concrete floor and conventional plaster ceiling. Balconies are located on the south facade with a 1.5m depth and a width which depends on the system design. The orientation, size and construction of the windows and the balconies in the main facade can be altered.

A parametric analysis is performed as a series of modifications to a base case which defines a particular set of construction, design and operating characteristics for the building zones. Table 5.1 summarises the construction of the zones for the base case.

Table 5.1: Construction material for the base case.

Name	Construction	Thickness (mm)	U-Value (W/m^2-K)
Exterior wall	Brickwork	110	0.7
	Air space	-	
	Polystyrene	35	
	Brickwork	110	
	plastering	10	
Floor	Ground	-	0.4
	Concrete, standard	100	
	Carpet	20	
Floor between zone	Concrete, standard	100	1.5
	Carpet	20	
Roof	Ceramic tiles	13	0.9
	Concrete, Scoria	60	
	Glass fibre	40	
	Concrete, standard	150	
	plastering	13	
Window	Glass	3.0	7.0

A number of design alternatives for various building structure and different evaporative cooling efficiencies, ventilation strategies, and different climatic types have been analysed which will be presented in the following sections.

5.3 COMPUTER SIMULATION PROGRAM

Simulations are numerical calculations that can give an indication of the thermal performance of a building and are more flexible than physical experiments. They are, however, relatively quick and inexpensive and can produce information on the effect of design variables on the system performance with a series of calculations all using exactly the same loads and weather data. Simulations are uniquely suited to parametric studies and thus offer to the process designer the capability to explore the effects of design variables on long-term system performance. They offer the opportunity to evaluate effects of system configuration and alternative system concepts. They have the advantage that the weather used to drive them is reproducible, allowing parametric and configuration studies to be made without uncertainties of variable weather. By the same token, a system can be operated by simulation in a wide range of climates to determine the effects of weather on design.

The aim in modelling the thermal performance of the building is to construct a satisfactory model, with which to predict performance, with the least complexity possible. The methods considered range from simple empirical methods through idealised analytic tools to detailed computer simulations. All the models try to reach a compromise between realism of the model and efficiency of calculation.

A large number of programs have been developed, many of them in North America. Some well-known ones that are readily available include BLAST, SERI-RES, NBSLD, DOE-2, DEROB and TRNSYS and QUICK from South Africa. In Australia there has been similar activity on a smaller scale, which has produced programs such as ZSTEP, TEMPER, BUNYIP and CHEETAH from CSIRO, and TEMPAL from the University of Melbourne. The simulation program CHEETAH (Delsante, 1987), which carries out the hour-by-hour calculations, using a building data file and a climatic data file, has been developed from program ZSTEP (Delsante and Spencer, 1983).

Selection criteria to chose a suitable program for this study were:

- easy availability in Australia
- availability of natural ventilation and evaporative cooling simulation
- documentation sufficient for thorough understanding of the model
- availability of feedback and technical advice from the author which was necessary for specialised development of the program.

Based on the above mentioned parameters, a modified version of the computer program CHEETAH which was specially developed by Dr. Delsante in 1995 for this project was used to predict the thermal behaviour of the building with the passive evaporative cooling system in this study. The program is fully described in Appendix A.1.

5.4 CLIMATIC DATA

The simulation for Tehran is carried out for the hottest week of the year, July 1977. Climatic data DATSAV2 was obtained from the National Climatic Data Centre (NCDC) in the U.S.A (see Appendix A.2).

The DATSAV2 climatic data was processed for use in CHEETAH by the author. Some data such as solar azimuth, solar altitude, diffuse solar irradiance on a horizontal plane and direct solar irradiance on a plane normal to the beam which were not included in the original data have been generated using the computer program TRNSYS. Also in the original data some hours were missing or not recorded and in these cases, the missing data were completed by averaged values.

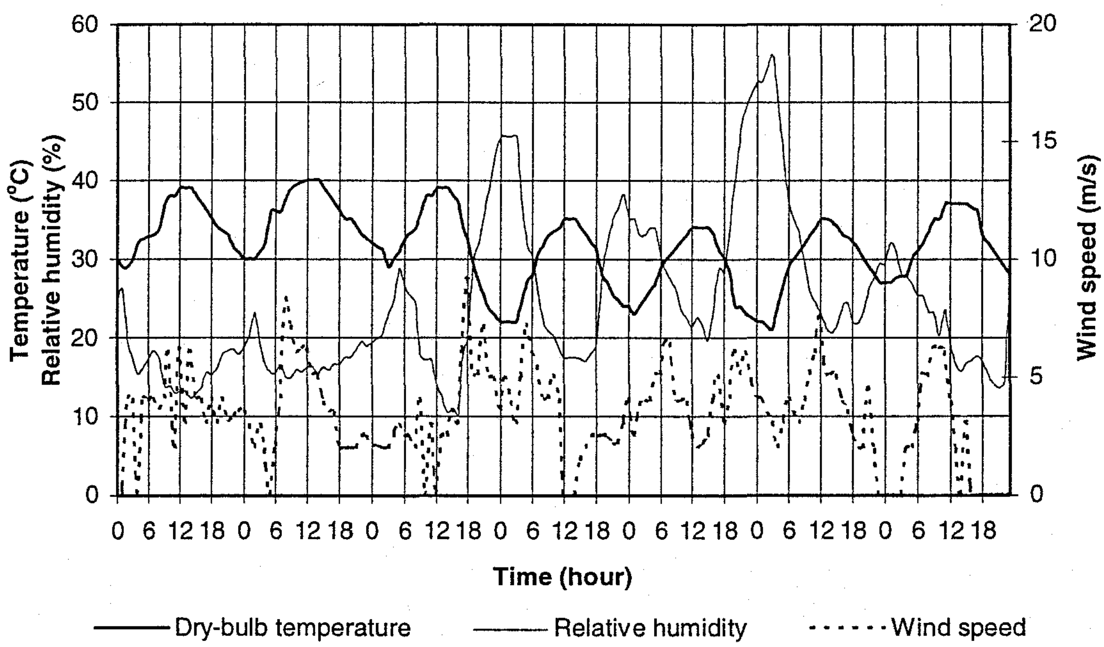


Figure 5.1: Outdoor environmental condition of Tehran for simulation period.

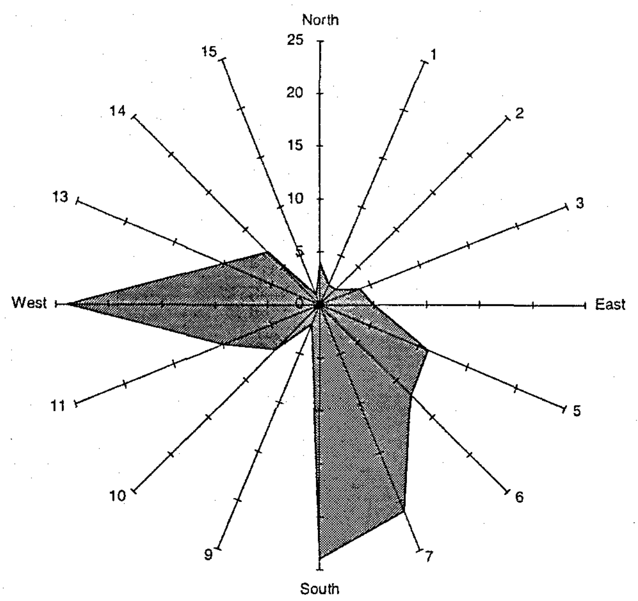


Figure 5.2: Prevailing wind direction of Tehran for simulation period.

Figures 5.1 and 5.2 show the outdoor environmental conditions of simulation period for Tehran from 13 to 19 July 1977. A general explanation of climatic conditions of Tehran and monthly average environmental parameters are presented in Appendix A.3.

The following sections will analyse the various strategies and parameters which influence airflow through building and evaporative cooling efficiency.

5.5 VENTILATION STRATEGY AND THE EFFECT OF VARIOUS PARAMETERS ON THE AIRFLOW RATE THROUGH THE BUILDING

Air movement in buildings plays an important role in energy saving, thermal comfort and indoor air quality because the distributions of temperature and contaminant concentrations are directly dependent on the airflow pattern. Air movement in buildings is affected by many parameters, such as the locations and size of openings, building shape and orientation and also local shielding and terrain parameters. It is difficult to deal with the variations of each parameter in a single experimental study. With the development of mathematical models, the numerical simulations have been widely used in indoor airflow study because of its flexibility in dealing with different design conditions and its ability to estimate the indoor ventilation rates. A number of mathematical models have been considered to investigate the airflow affected by above mentioned parameters. A few examples are presented in Chapter 3.

A mathematical model of natural ventilation which considered various design parameters and has been well verified by Swami and Chandra (1988) is employed in the computation to predict the air flow rate through the building and also the number of air changes per hour (ach). All calculations including the ventilation rate and the efficiency of the system based on hourly recorded climatic data are computed by the program EXCEL. A series of analyses of the effect of various parameters on the natural ventilation rate is presented in the following sections.

5.5.1 The Effects of Shielding and Terrain Factors

Airflow through buildings can be affected significantly by shielding and terrain factors. The terrain factor has a direct effect on reference velocity (V_{ref}) and the shielding factor (SCF) on the numbers of air changes through building.

Case 1 is one where the building is located in a light local shielding with few obstructions (SCF = 0.88) and a flat terrain with some isolated obstacles (from Table 3.3, a = 1 and b = 0.15). Case 2 assumes the same building in a moderate local shielding with some obstructions within two house heights (SCF = 0.74) and a rural area with low buildings (a = 0.85 and b = 0.20). In case 3, there is a building in a heavy shielding with obstruction around most of perimeter building or trees within five times building height in most directions (SCF = 0.57) and an urban, industrial terrain or forest areas (a = 0.67 and b = 0.25). The final case looks at a building in a very heavy shielding with a large obstruction surrounding perimeter within two house heights (SCF = 0.31) and centre of large city (a = 0.47 and b = 0.35). The selected case for simulation has SCF = 0.74 , a = 0.67 and b = 0.25.

In all cases, it is assumed window openings of 11.2 m^2 located on the south and 2.24 m^2 (20% of southerly opening) on the northerly facades.

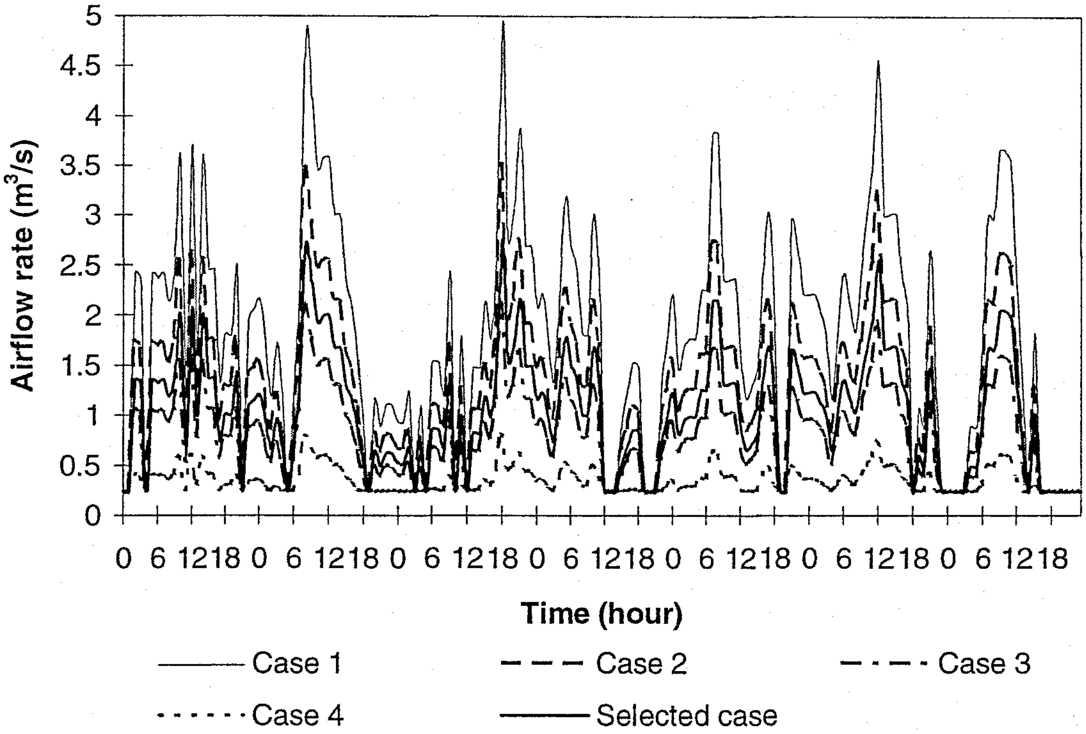


Figure 5.3: Influences of various shielding and terrain coefficients on the airflow rate through building.

Figure 5.3 compares the influences of various shielding and terrain coefficients on the airflow rate through a building. A plot of the results over a 168-hours period, in this figure, shows that although the minimum of airflow rate 0.233 m^3/s is the same for the all cases where the wind speed is zero, the maximum air flow rate decreases from the case 1 to 4. As the shielding and terrain classifications, obtain a lighter class, the ventilation rate proportionally increases.

5.5.2 The Effect of Building Geometry

Most data in literature as described in Chapter 3, are for rectangular buildings only, however, the selected model (Swami and Chandra, 1988), is capable of calculating natural ventilation through other house plans forms.

The effect of various building geometry on the natural ventilation rate is considered in this section. Four different building shapes as shown in Figure 5.4 are chosen for analysis and in all cases, windows are assumed to be located in the north and south facades.

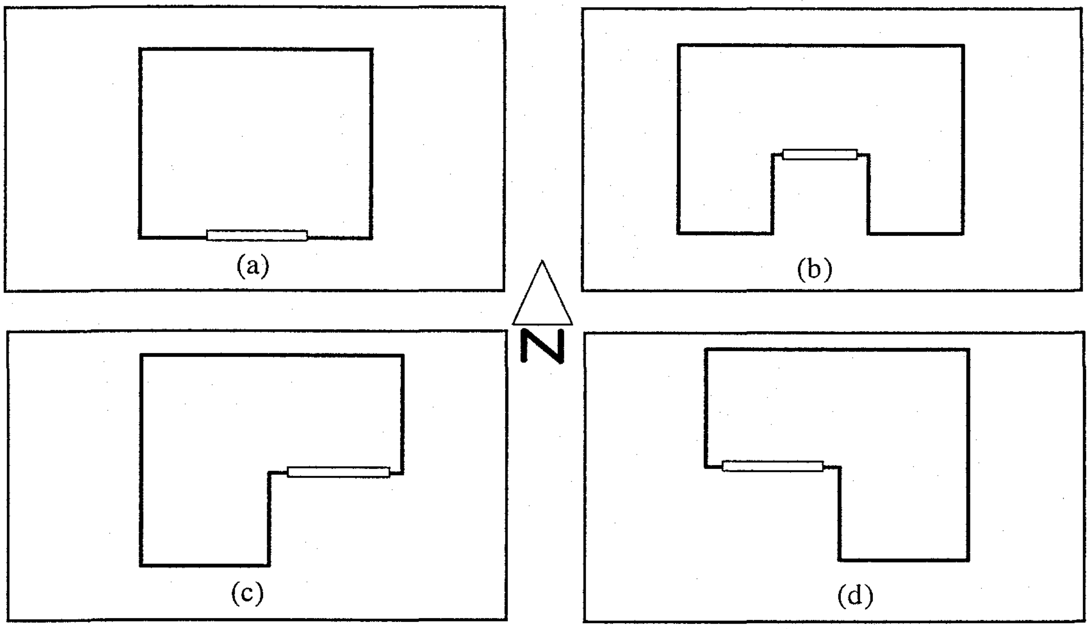


Figure 5.4: Various building shape for airflow rate calculation.

The first one is a building of rectangular shape and the second is a building with the same area and volume but a "U" shaped floor plan. The third and fourth building are both "L" shaped, but in a different configuration.

The simulation results show that, the computed airflow rates for the given building shapes do not vary by very much. However, the average airflow rate for a rectangular building is 1.01 m^3/s which is slightly lower than an average airflow rate of 1.02 m^3/s for a "U" shaped building. The reason is that, in "U" shaped buildings, pressure coefficient values (Cp) for wind angles up to $\pm 60^{\circ}$ from normal to the surface, may be assumed to be the value at zero incidence. The prevailing wind directions in summer time in Tehran are through south and south east, therefore, most of the Cp values will be taken to be at maximum value of 0.6.

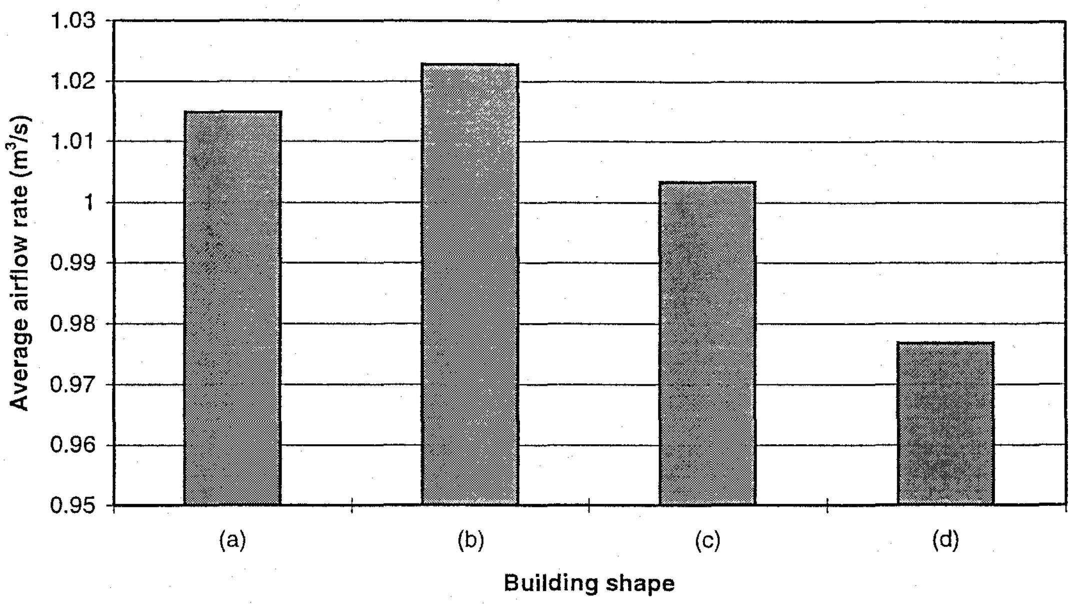


Figure 5.5: The effect on the airflow rate by differences in building shapes.

As can be seen from Figure 5.5, the airflow rate achieved from the two different configurations of the "L" shaped buildings are not the same. The third case (c) has an average airflow rate of 1.00 m^3/s while, the fourth one (d), has a lower average airflow rate of 0.98 m^3/s. This is also because of the south and south east prevailing wind directions in Tehran. Thus, Cp values for wind angles up to $+90^{\circ}$ in this building shape is taken as the maximum value at zero incidence.

A comparison between previously discussed cases shows that, in the given climatic condition, the "U" shaped and rectangular shaped floor plan obtain more airflow rate than the "L" shaped floor plans.

5.5.3 The Effect of Window Design

It is well known that westerly and easterly windows are difficult to manage in summer especially in hot regions. Therefore, the window locations in this study are assumed to be fixed on the north and south facades. According to Equation 3.5 , the effective area of the windows (A_e) depends on the opening area of the inlet and outlet. The evaporative cooling system is assumed to be installed on the south facade of the building because of the south-eastern prevailing wind direction in summer time in Tehran. Hence, to obtain the most evaporative cooling effect, it is assumed to have maximum opening area (fully open window in balcony) in south facade. As Tehran has very cold winters, maximising the size of southerly window has also a beneficial effect on the indoor conditions in winter due to the low sun altitude from the south.

It should be noted that the total area of the glazing is not necessarily open but, the opening can be operated to control the amount of airflow through the building.

Influence of the various sizes of north facing opening area on the interior airflow rate is shown in Figure 5.6. In all cases, the south facing window is assumed to have a 11.2 m^2 opening area. For the northerly opening outlet area of window (A_o), six alternatives have been studied:

- 10% of southerly opening area ($A_o = 1.12\ m^2$ and $A_e = 1.11\ m^2$)
- 20% of southerly opening area ($A_o = 2.24\ m^2$ and $A_e = 2.2\ m^2$)
- 30% of southerly opening area ($A_o = 3.36\ m^2$ and $A_e = 3.22\ m^2$)
- 40% of southerly opening area ($A_o = 4.48\ m^2$ and $A_e = 4.16\ m^2$)
- 50% of southerly opening area ($A_o = 5.6\ m^2$ and $A_e = 5\ m^2$)
- 60% of southerly opening area ($A_o = 6.72\ m^2$ and $A_e = 5.76\ m^2$).

It can be seen from Figure 5.6 that, the value of airflow rate increases with the size of the opening area of the window. In regions such as Tehran; increasing the day time natural ventilation rate is not always recommended because it not only introduces the hot outdoor air inside the building but also has an opposite effect on the cooling efficiency which will be discussed on Section 5.6.3.

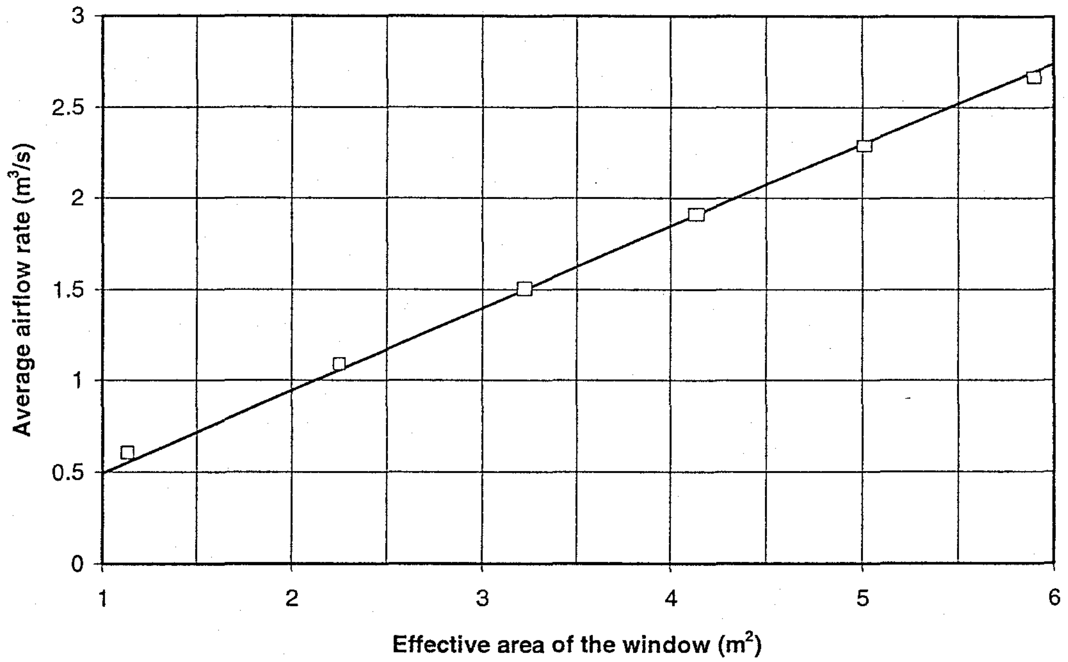


Figure 5.6: The effect of opening area of the window on the airflow rate through the building.

5.6 EVAPORATIVE COOLING EFFICIENCY AND THE EFFECTS OF VARIOUS DESIGN FACTORS

The method of computing heat and mass transfer rates from the surface of the evaporative cooling system and also the efficiency of the system are presented in Chapter 4. The efficiency of the system as explained before, depends on two main factors: climatic condition and the system design. Some of the required climatic data namely relative humidity (RH), wet-bulb temperature (t_{wb}) and the humidity ratio of moist air at saturation point (W_s*) which were not included in the existing climatic data, were determined by the equations of the psychrometric chapter of ASHRAE (1985). The relative humidity RH is given by:

$$\mathbf{RH = (p_w/p_{ws}) \times 100} \qquad (5.1)$$

Where p_w is the partial pressure of water vapour in moist air (kPa) and defined by:

$$\mathbf{p_w = (W \times p)/(0.622+W)} \qquad (5.2)$$

where:

W = humidity ratio of moist air (mass of water per mass of dry air, kg/kg)

p = atmospheric pressure (kPa).

The term p_{ws} represents the saturation pressure of water vapour at the given temperature in Pa. This pressure is a function only of temperature and differs slightly from the vapour pressure of water in saturated moist air and is defined by:

$$\mathbf{p_{ws} = EXP(C_1/T + C_2 + C_3.T + C_4.T^2 + C_5.T^3 + C_6.\ln(T))} \qquad (5.3)$$

where

$C_1 = -5800.2206$

$C_2 = 1.3914993$

$C_3 = -0.04860239$

$C_4 = 0.41764768.10^{-4}$

$C_5 = -0.14452093.10^{-7}$

$C_6 = 6.5459673$

where, T is absolute temperature, K (K = ^{o}C + 273.15).

Thermodynamic wet-bulb temperature, t_{wb}, as defined is a unique property of a given moist air sample independent of measurement techniques utilised and is given by:

$$\mathbf{t_{wb} = \left[((t_{db}\ 1.8)+32)\ \ln(4 + 0.058\ RH))/4.1414653\right]-17.7776} \qquad (5.4)$$

where in this equation, t_{wb} and dry-bulb temperature, t_{db}, both are in °C.

The humidity ratio of saturated moist air at thermodynamic wet-bulb temperature, W_s^* is given by:

$$W_s^* = \left(W\,(2501+1.805 \times t_{db} - 4.186 \times t_{wb}) + t_{db} - t_{wb}\right) / (2501 - 2.381 \times t_{wb}) \qquad (5.5)$$

The other factors such as thermal conductivity, k (W/m.K), kinematic viscosity, ν (m^2/s), mass density, ρ (kg/m^3), Prandtl number, Pr and Prandtl number at water temperature, Pr_s, are generated for various hourly temperature ranges based on thermophysical properties of air and water from Table A.1 and Table A.2 in Appendix A.4.

5.6.1 The Effect of Wind Direction

As the wind direction changes over time, this section examines the effect of various wind directions on the system efficiency. Natural ventilation occurs through windows by the pressure differences acting on inlets and outlets. Once a window subjected as an inlet area, by changing the wind direction, becomes as an outlet area. Therefore, ventilation will take place and does not matter through which direction.

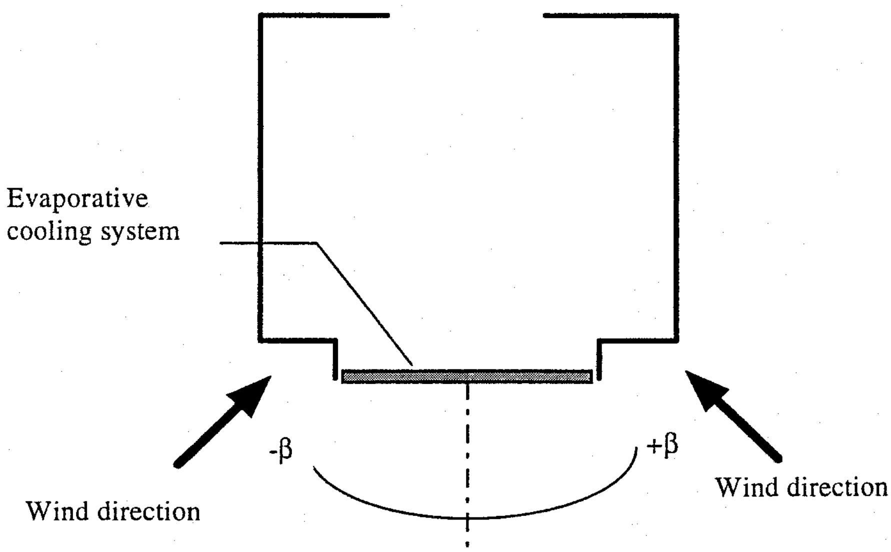


Figure 5.7: Schematic of acceptable wind directions through evaporative cooling system.

However, for calculating the efficiency of the evaporative cooling system, only the wind directions of $-90° < \beta < 90°$ as shown in Figure 5.7, are considered because, the wind directions beyond these, will not have cooling effect inside the building. In fact, in case of $\beta \leq -90°$ and $\beta \geq 90°$, natural ventilation occurs inside the building and there will be a value for air changes rate, but the cooling efficiency will be set at zero because air flows from the other side of the building where there is no evaporative cooling system. Moreover, where the numbers of air changes ≤ 3, in low wind periods (based on experimental measurements of Swami and Chandra, 1988), the system's efficiency is assumed to be zero at these times.

If it is assumed that the evaporative cooling system is installed in southern balcony and it is arranged in three rows with water lines of 3mm in diameter and transverse pitches of 9mm, as shown in Figure 5.8(a), the effect of four different wind directions on the average efficiency of the system is examined in this section. It is essential to use the reference velocity at the height of the building terrain because the wind speeds in climatic data are in the reference terrain at height 10m.

Figure 5.10 indicates that, the average cooling efficiency of the system rises as the wind direction increases up to $\pm$ 90°. In fact when the wind direction is perpendicular to the system, it passes only through 3 rows. While the wind direction changes and tends to an oblique direction, it should pass through more rows. The maximum efficiency occurs when the wind direction is $\pm 90°$, and it passes through maximum numbers of rows. In spite of maximum efficiency for the system itself which occurs at this angle, there will be no pressure differences across the openings of the building (south and north facing windows) for wind directions of $\pm 90°$. Therefore efficiency of the system at $\pm 90°$ will be set at zero and as mentioned before only the wind directions of $-90° < \beta < 90°$ will be considered to calculating the evaporative cooling efficiency of the system.

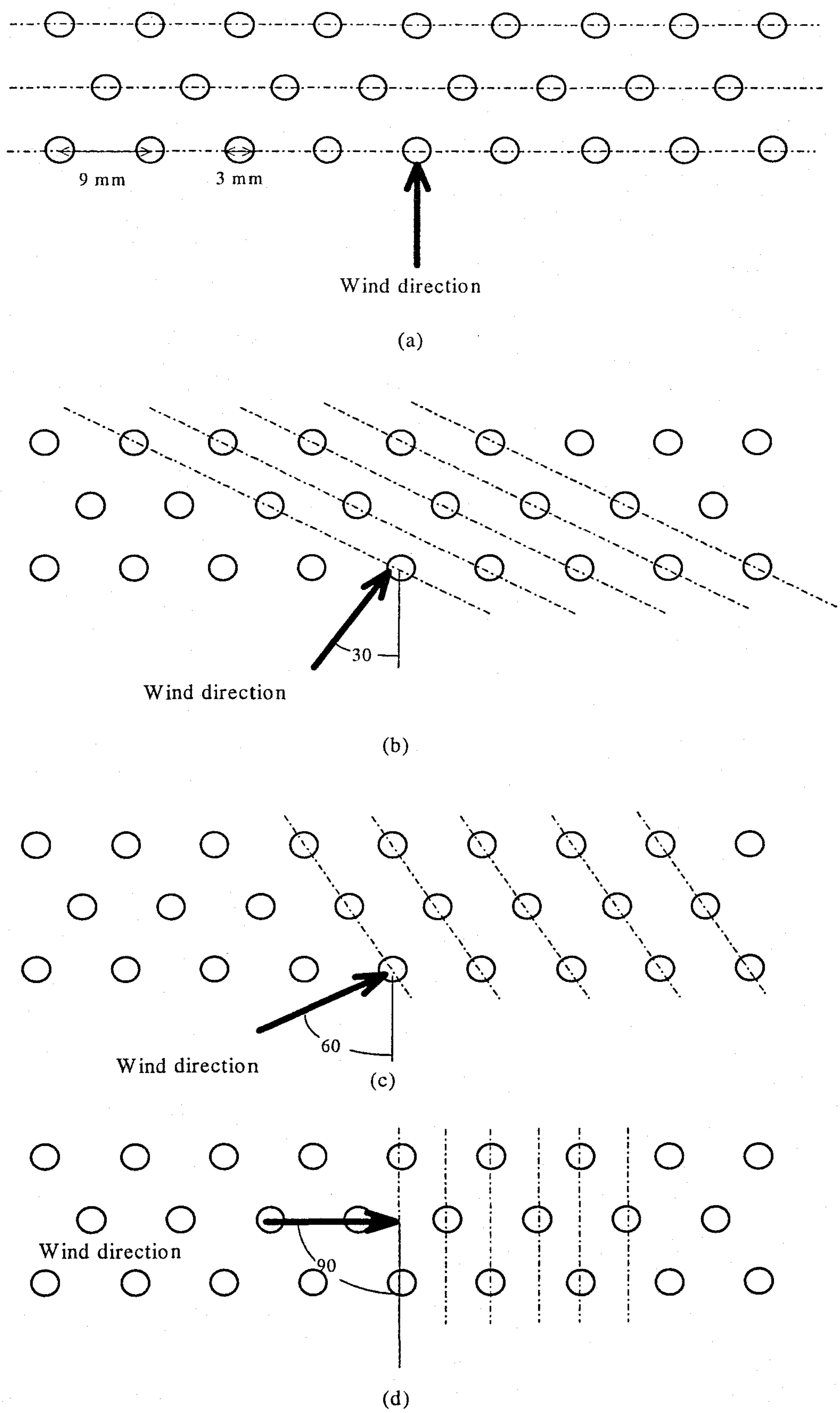


Figure 5.8: Number of rows with various wind direction.

Although the maximum efficiency happens in oblique directions, a conservative minimum efficiency is chosen to continue the rest of the study.

5.6.2 The Effect of Filament Arrangement

An essential part of system analysis is the determination of the heat transfer coefficient which has a direct effect on the system efficiency, and this depends on the geometric arrangement and size of the filaments as well as the climatic condition. The effects of various system designs on the evaporative efficiency of the system are represented in this section.

The water surface area of the system is one of the most important factors which has a direct effect on the cooling efficiency and is associated with diameter, height and the number of water lines in the system.

$$\mathbf{A_w = N_R.N_L.D.\pi.H_S} \qquad \mathbf{(5.6)}$$

where:

A_w = surface area of water

N_R = number of rows

N_L = number of water lines per row

D = water line diameter

H_S = system height per story

and

$$\mathbf{N_L = L / S_T} \qquad \mathbf{(5.7)}$$

where:

L = length of the system (balcony)

S_T = transverse pitch

It is obvious that with the same number of water lines, the surface area of water varies in proportion to the diameter. If the transverse pitches (S_T) are defined as multiples of the water line diameter, this would mean that as the diameter increases, the numbers of water lines decrease, but the surface area of water remains constant. Thus from the Equations 5.6 and 5.7 , the water surface area becomes independent of the diameter.

$$\mathbf{S_T = \mathit{a} \,.\, D} \tag{5.8}$$

Thus, A_w can be written as:

$$\mathbf{A_w = N_R \,.\, (L \,/\, \mathit{a}) \,.\, \pi \,.\, H_S} \tag{5.9}$$

Where in both equations, a is a constant.

The water volume of the system can be calculate by the following equations:

$$\mathbf{V_W = N_R.N_L.(D/2)^2.\pi.H_S} \tag{5.10}$$

From Equation (5.6), Equation (5.10) becomes dependent of the diameter:

$$\mathbf{V_W = D/4 \,.\, A_w} \tag{5.11}$$

Therefore, if the transverse pitch is assumed as multiple of the diameter, for any diameter, the water surface area is the same while the water volume will increase by increasing the diameter.

The nature of the situation can significantly influence the saturation efficiency. Although it is obvious that the larger surface area of water has a greater cooling efficiency, from an examination of the effects of various system designs and a careful analysis of the results, it is assumed to have a constant water surface area of $35.2m^2$. Therefore the following assumptions can be fixed for the all the parametric analyses performed:

- The system is assumed to have a staggered arrangement.
- The lines of water are arranged as an equilateral triangle ($S_T = S_D$).
- The height of the system is equal to the height of the balcony (2.8m).
- The diameters of the filaments are 1.5mm.

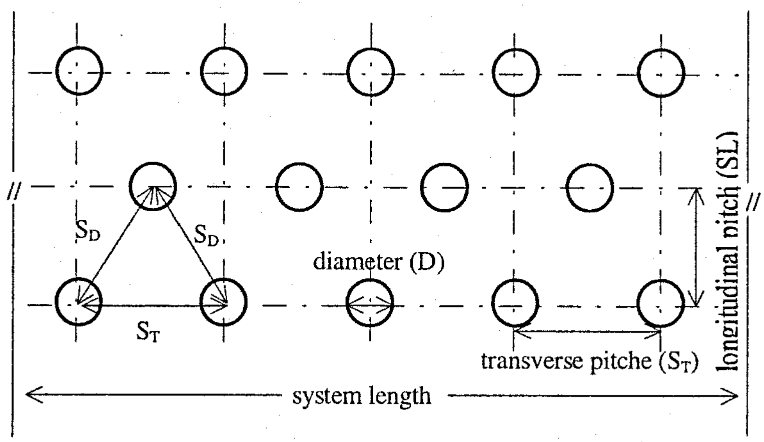


Figure 5.9: Horizontal section of the system.

The first analysis examines the effect of various diameters of water lines on the efficiency of the system. In this case, the system being modelled is assumed to have diameters of D1 = 3mm, D2 = 5mm and D3 = 7mm. The transverse pitches of the system are assumed to be 3D in three rows with the system length of 4m as shown in Figure 5.9.

Table 5.2 shows the transverse pitches (S_T), numbers of water lines (N_L), water surface area and water volume for the three alternatives.

Table 5.2: System characterisation.

Water line diameter (D) m	Transverse pitch (S_T) m	Number of water line per row (N_L)	Number of rows (N_R)	Water surface area (A_w) m^2	Volume of water (V_w) m^3
0.003	0.009	445	3	35.2	0.026
0.005	0.015	267	3	35.2	0.044
0.007	0.021	191	3	35.2	0.061

Figure 5.10 shows a comparison of the effect of three different diameters of the water lines on the efficiency of the system. As can be seen, the reduction in diameter causes better results and a higher efficiency because, the air which crosses the system should pass through tighter pitches and so has a better opportunity to be cooled. For the case of 3mm diameter the average efficiency is 55.32% while for 5mm and 7mm diameters it decreases to 54.4% and 53.4%. From the point of view of water flow rate , although the water surface area in the three cases is the same, the volume of water increases when the diameters increase. Therefore considering the energy consumption for pumping a large volume of water and because of less cooling efficiency, a diameter of 3mm is chosen to continue the analysis of the system.

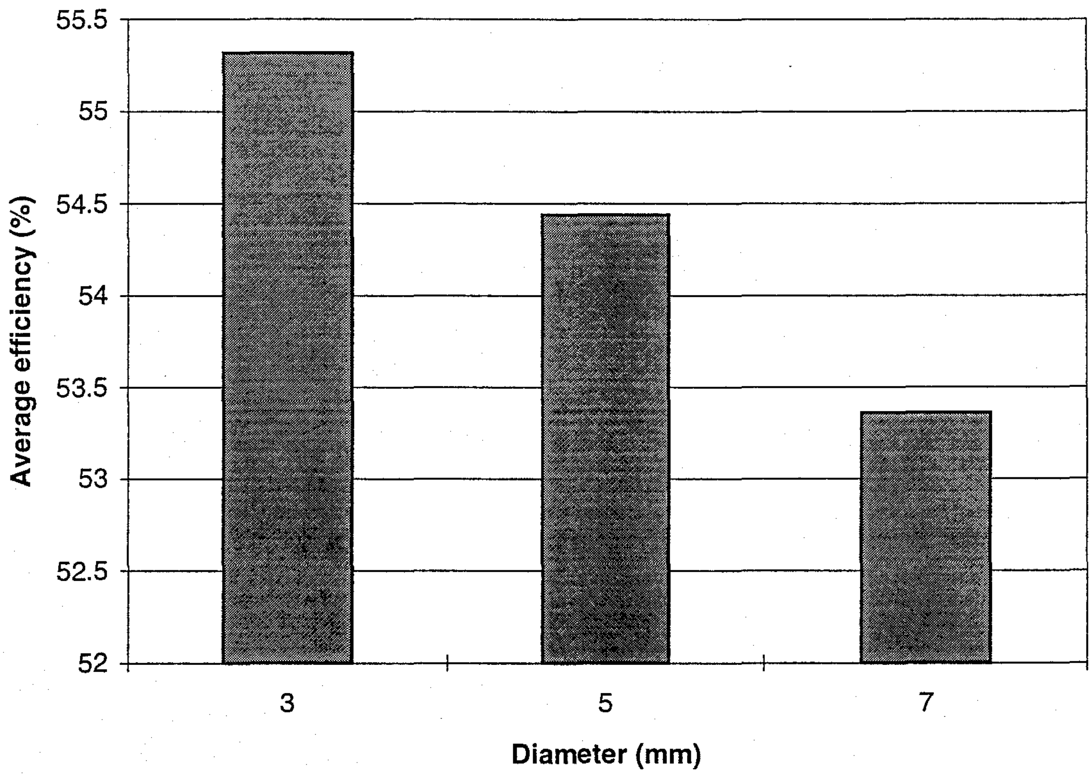


Figure 5.10: The effect on the system efficiency of different diameters of water line.

The influence of various pitches of the water lines on the cooling efficiency, is the subject of the second analysis. This case involves comparing three different transverse pitches of

$S_T1 = 3D$, $S_T2 = 4D$ and $S_T3 = 5D$. To provide the same surface area of water, the width of the system varies as the distances between the transverse pitches change. The widths of the system are L1 = 4m, L2 = 5.3m and L3 = 6.7m respectively. This case is also modelled for the three rows.

Figure 5.11 indicates that the cooling efficiency of the system, as explained in Section 4.4.1, decreases when the distance of the water lines increases. Therefore the best performance can be achieved where the distances between the centres are small (in this case $S_T = 3D$). The results represent the average efficiencies of 54.0% for pitches of 4D and 53.3% for the pitches of 5D while that which remains for 3D is the same as in the previous section.

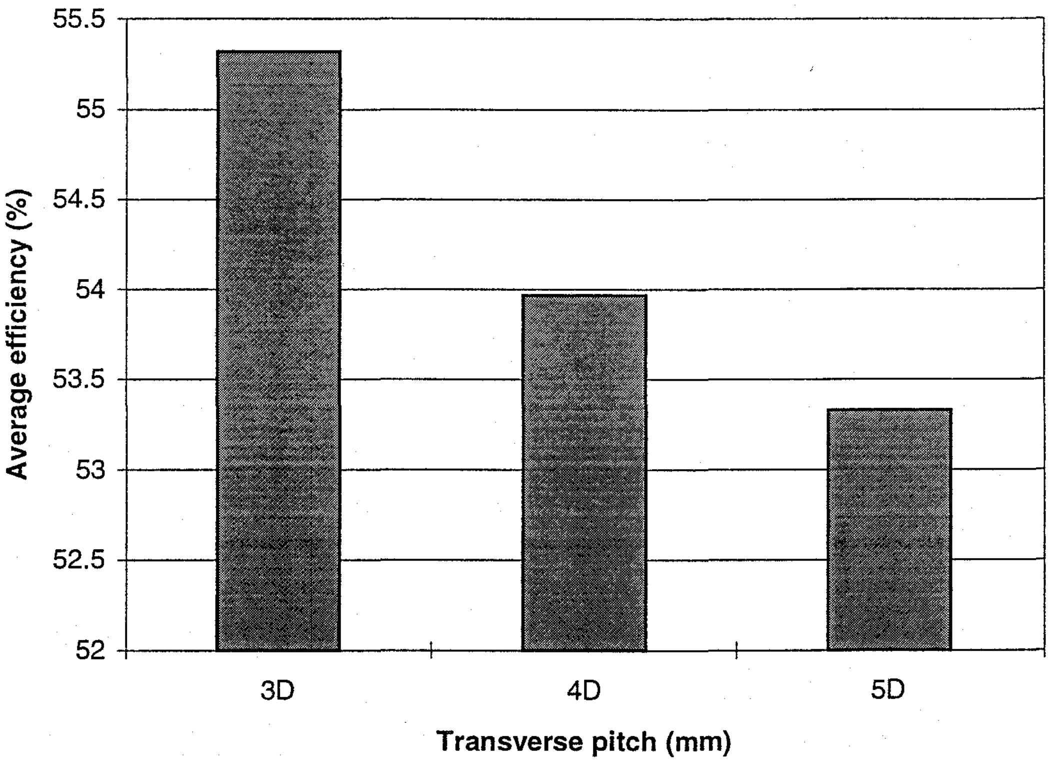


Figure 5.11: The effect on the efficiency of various transverse pitch.

The third analysis examines the system efficiency by varying the transverse pitch of the water lines and the number of rows in the system. This case involves three different pitches and rows but the same width of 4m. S_T1 = 3D with 3 rows, S_T2 = 4D with 4 rows and S_T3 = 5D with 5 rows.

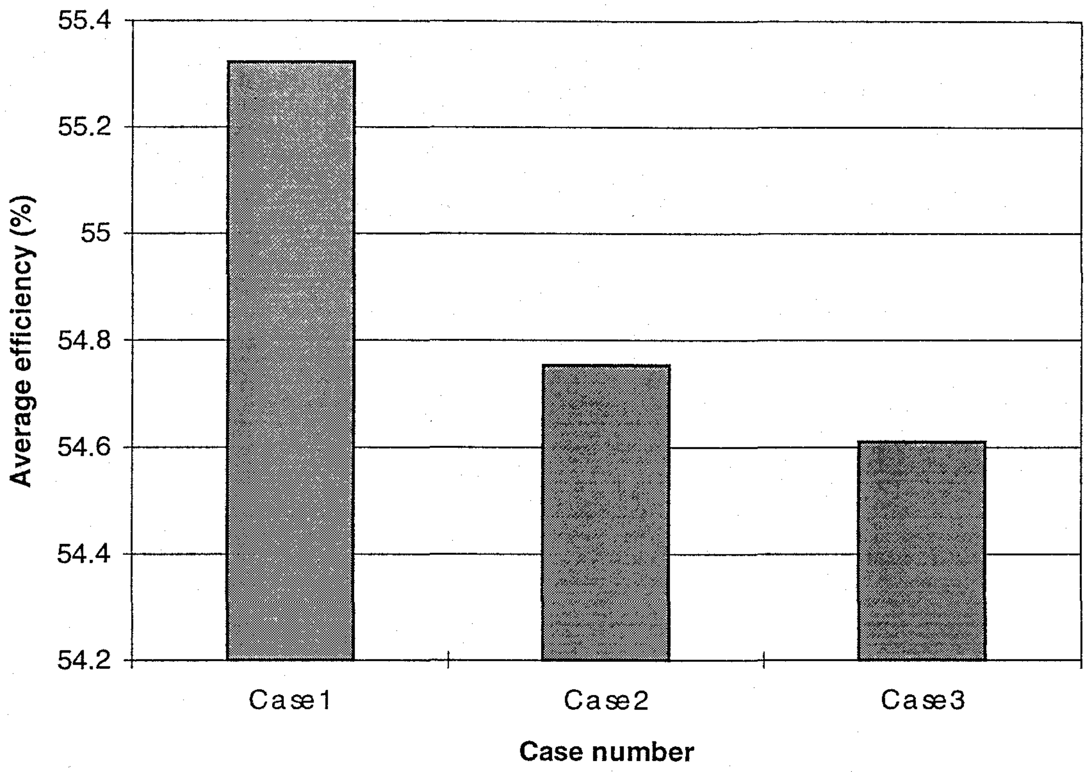


Figure 5.12: The effect on the efficiency of various pitches and rows with the same width.

Figure 5.12 shows that the smaller transverse pitches with the lesser number of rows have greater efficiency than the larger transverse pitches with more rows. The average efficiencies for cases 1 to 3 are 55.3%, 54.8% and 54.6% which the differences are not noticeable particularly for cases 2 and 3. However, in comparison with the fourth analysis, which is of a system with various numbers of rows and a different width of N_R1 = 3 with L1 = 4m, N_R2 = 4 with L2 = 3m and N_R3 = 5 with L3 = 2.4m, but the same pitches of S_T = 3D. as Figure 5.13 shows, a system with more numbers of rows and smaller width, is more efficient than a system with wider length and fewer rows (56.0% for 4 rows and 56.4% for 5 rows).

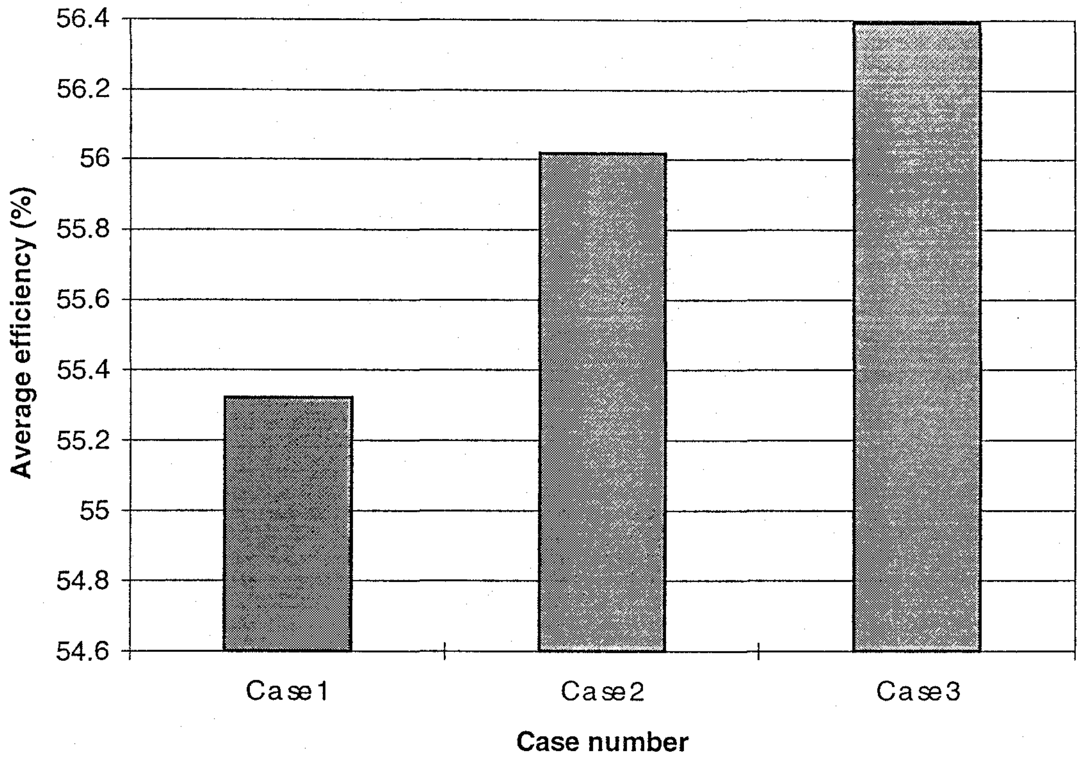


Figure 5.13: Effect of various numbers of rows with variable width on the efficiency.

Table 5.3 summarises the results of the simulations. In all previously discussed cases, the average cooling efficiencies vary from 53.3 % to 56.4 %. The lowest case belongs to a system with large transverse pitches (S_T=5D) while a strategy with a small diameter and pitches with more rows is the best one. In addition, an increase in the number of rows results in a reduction of the system's width which gives flexibility in buildings designs for situations with narrow facades. The case with 3mm diameters of water lines and pitches of 3D is preferred because of their better efficiency. However, it depends on the design situation to select the number of rows of the system. The case number 1 is chosen to continue the analysis of the influences of the other factors on the system's efficiency.

Table 5.3: Simulation results of effect of various parameters on the cooling efficiency of the system.

Case number	Diameter of water lines, D (mm)	Transverse pitches, S_T	Numbers of rows, N_R	Width of the system, L (m)	Average cooling efficiency
1	3	3D	3	4	55.3%
2	5	3D	3	4	54.4%
3	7	3D	3	4	53.4%
4	3	4D	3	5.3	54.0%
5	3	5D	3	6.7	53.3%
6	3	4D	4	4	54.8%
7	3	5D	5	4	54.6%
8	3	3D	4	3	56.0%
9	3	3D	5	2.4	56.4%

5.6.3 The Effect of the Air Flow Rate

As it is mentioned before, airflow rate through a building can be controlled by changing the size of window opening. Thus, to evaluate the various ventilation strategies and their influence on the system efficiency, the same airflow rates as that described in Section 5.5.3 have been chosen for the parametric study relating to system's efficiency. As can be seen from the Figure 5.14, the efficiency of the system drops as the airflow rate increases. The reason is that, the air passing through the evaporative cooling system, has a better opportunity to be cooled and saturated during times of low flow rate. Thus, there will be an improvement in the efficiency of the system in lower flow rates.

However, this does not indicate a better result in room air temperature, because, with a lower flow rate, the evaporatively cooled air will not have a sufficient circulation inside the building. To determine the optimum room air temperature related to both airflow rate and efficiency of the system, a study has been made which is described in following section.

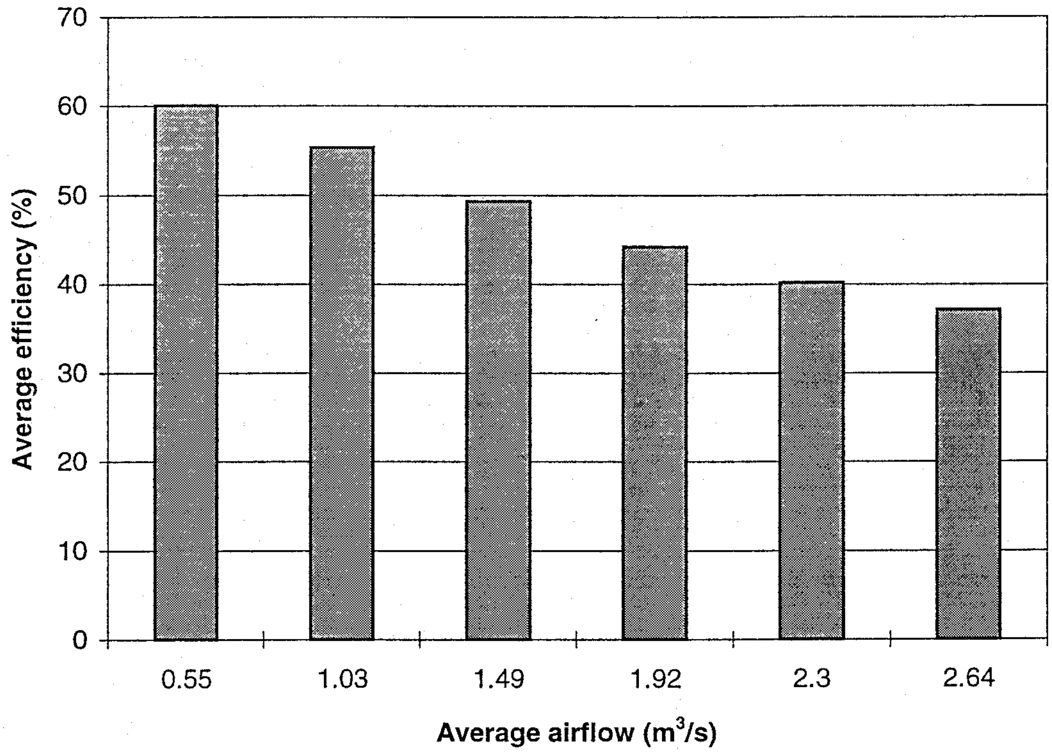


Figure 5.14: Influence of various airflow rates on the efficiency of the system.

5.6.4 Optimum Indoor Airflow

A series of simulations is performed in order to assess the impact of the ventilation rate and evaporative cooling efficiency on the building thermal behaviour. The optimum ventilation rate to reach to the lowest indoor air temperature has been evaluated by simulating the thermal behaviour of a three storeys building (100 m^2 surface area x 2.8 m height in each storey), assumed to be constructed as the reference case according to the Table 5.1. The building has a 11.2 m^2 fully opened window and a balcony with 4m width and 1.5m depth facing south and a variable window opening size on the northern facade which controls the airflow rate inside the building.

The building is assumed to be ventilated with six different airflow rates related to the northerly opening size of the window referred to the previously mentioned section. To simplify the comparative analysis, the simulations are performed only for the middle storey of the building.

The simulation results for 168 hours of a hottest week of the year as illustrated in Figure 5.15 show that, the lowest indoor air temperature achieved when the average airflow rate is at 1.03 m^3/s where the northerly window has an opening area of 2.24 m^2 (see Section 5.5.3).

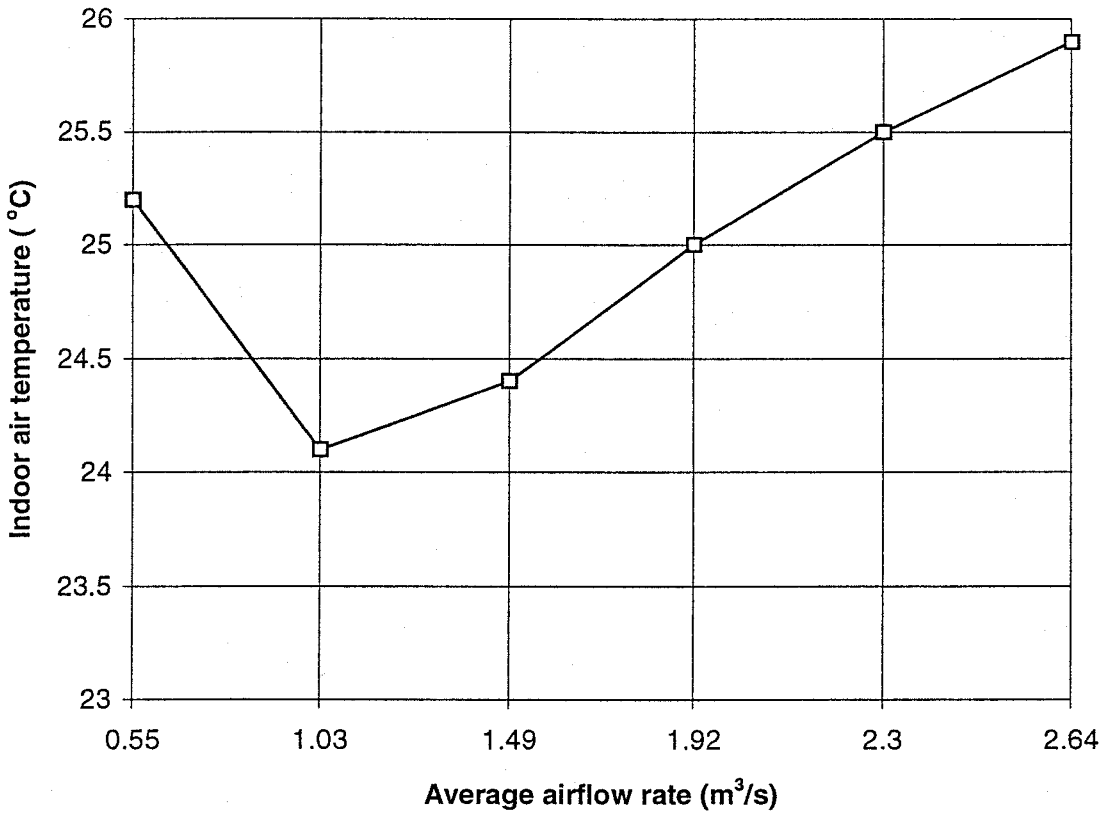


Figure 5.15: Indoor air temperature with various airflow rates for the base case building.

Although, at this flow rate the evaporative cooling efficiency of the system itself, is not at a maximum, however, the maximum cooling effect on the building is achieved due to this flow rate.

A study has also been made of whether the volume of building will have an effect on the indoor temperature conditions with exactly the same system. For this purpose, two buildings with a volume of 548.8 m^3 (14x14x2.8 m) and 137.2 m^3 (7x7x2.8 m) are subjected to the same process of hourly simulation as the base case building.

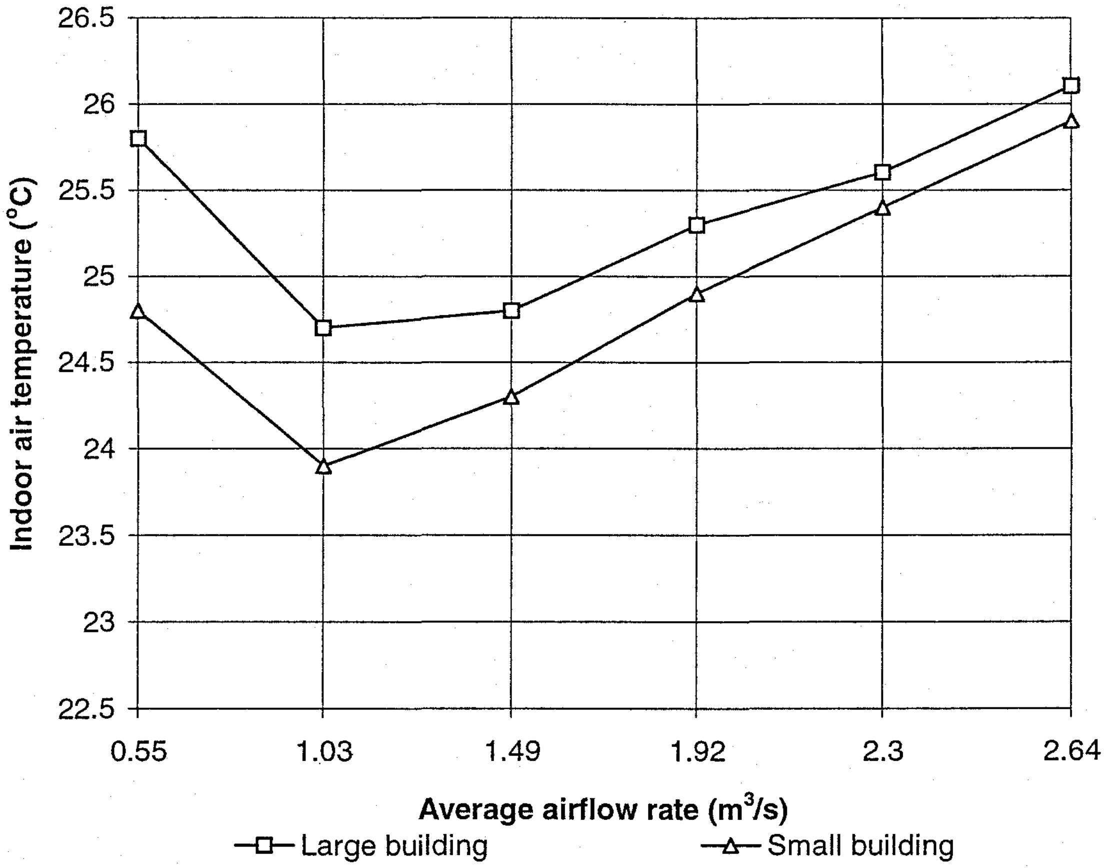


Figure 5.16: Indoor air temperature with various airflow rates for the small and large buildings.

As can be seen from the Figure 5.16 result shows that both large and small buildings have a minimum temperature in the flow rate of 1.03 m^3, the same as the reference building. It means that the cooling effect obtained from the evaporative cooling system, is independent of the building volume as it depends on the airflow rate and the system performance.

An additional series of hour-by-hour simulations was performed for each of the above mentioned buildings (base case, large and small buildings) with double the evaporative cooling size (double water surface area), to find out whether a similar behaviour occurred as in the previously discussed cases.

The results of the simulations are given in Figure 5.17. In the base case building it is observed that the lowest indoor air temperature is related to the airflow rate of 1.49 m^3/s. A similar behaviour can be seen for the two other buildings although the temperature differences are not very sensitive.

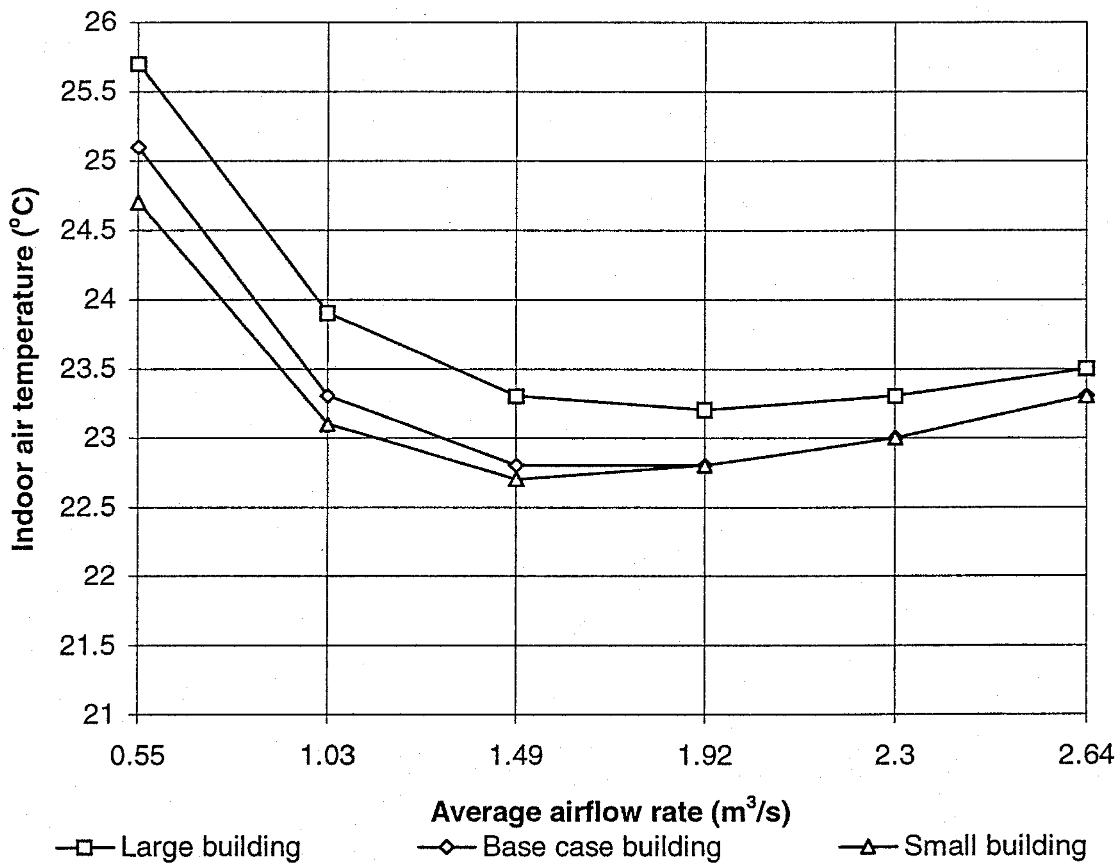


Figure 5.17: Indoor air temperature with various airflow rates and double water surface area for the base case, large and small buildings.

From these results it is concluded that a design parameter of 0.03 can be defined as an optimum volumetric airflow rate per square metre of surface area of the water in the system which takes the form:

Design parameter 0.03 = Optimum Q / A_w **(5.12)**

With this factor as a constant, it will be possible to calculate the optimum airflow rate due to the given water surface area of the system and also the suitable opening size of the windows.

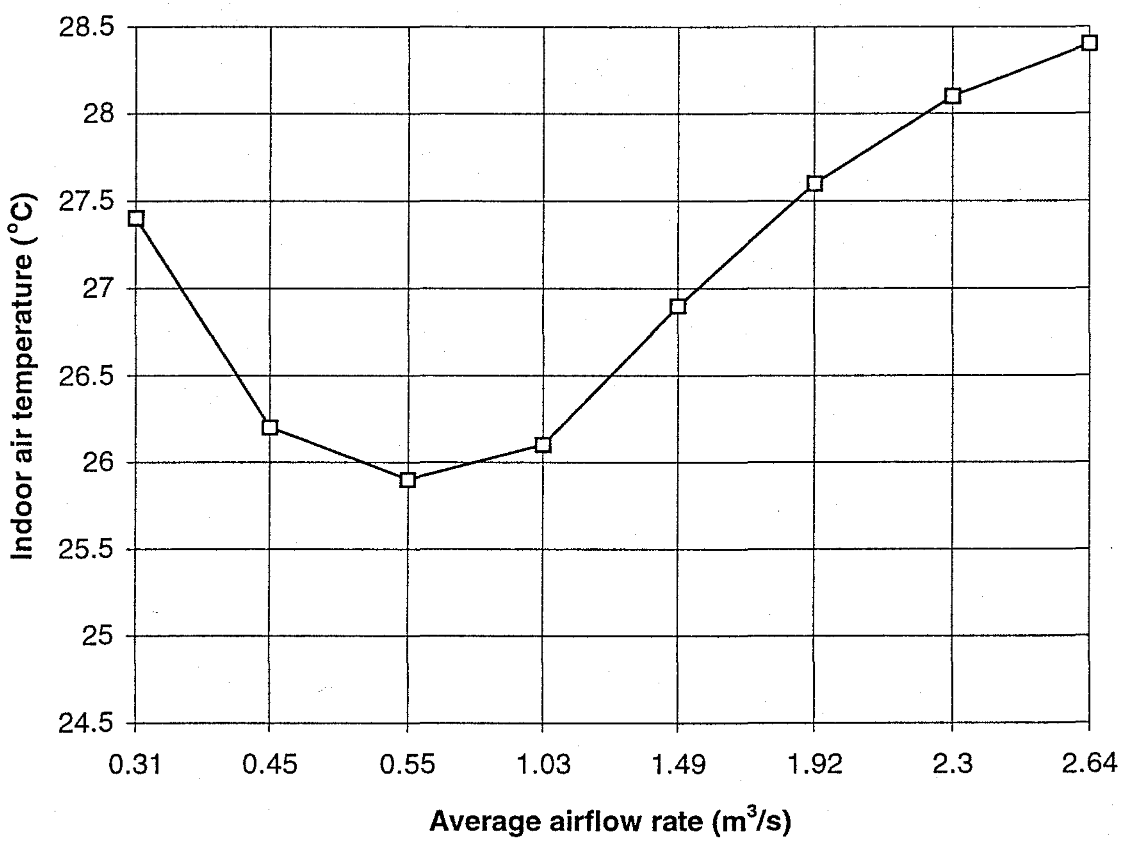


Figure 5.18: Indoor air temperature with various airflow rates and half water surface area for the base case building.

For example, for a half surface area of the water (35.2÷2 = 17.6 m^2), an optimum airflow rate of 0.55 m^3/s is required to obtain the lowest indoor air temperature which is independent to the volume of the building. The results of simulation for the base case building which support the above mentioned statement, are shown in Figure 5.18.

A summary of various buildings size and opening areas with average indoor airflow rates and also inside air temperature is illustrated in Table 5.4. When a comparison is made with this table for the same size of evaporative cooling system, it can be seen that the minimum indoor air temperature occurs in the small building. Even so the performance of the system itself is not related to the building size, and so it will have more cooling effect on smaller buildings. Furthermore, it is also evident that by increasing the surface area of water by 100%, the indoor air temperature in the base case decreases only about 1.3°C.

Table 5.4: Results of optimum airflow rate simulation and its impact on the indoor air temperature.

Building size	Effective area of openings (m^2)	Average optimum airflow rate (m^3/s)	Surface area of water (m^2)	Average indoor air temperature (°C)
Base case	2.2	1.03	35.2	24.1
Small	2.2	1.03	35.2	23.9
Large	2.2	1.03	35.2	24.7
Base case	4.16	1.92	70.4	22.8
Small	4.16	1.49	70.4	22.7
Large	4.16	1.92	70.4	23.2
Base case	1.11	0.55	17.6	25.9

5.7 INFLUENCE OF VARIOUS PARAMETERS ON THE INDOOR AIR TEMPERATURE

The thermal response of the building for the chosen airflow and evaporative cooling system is shown in Figure 5.19. This figure compares indoor air temperature of the three zones of the unoccupied base case building and outdoor air temperature.

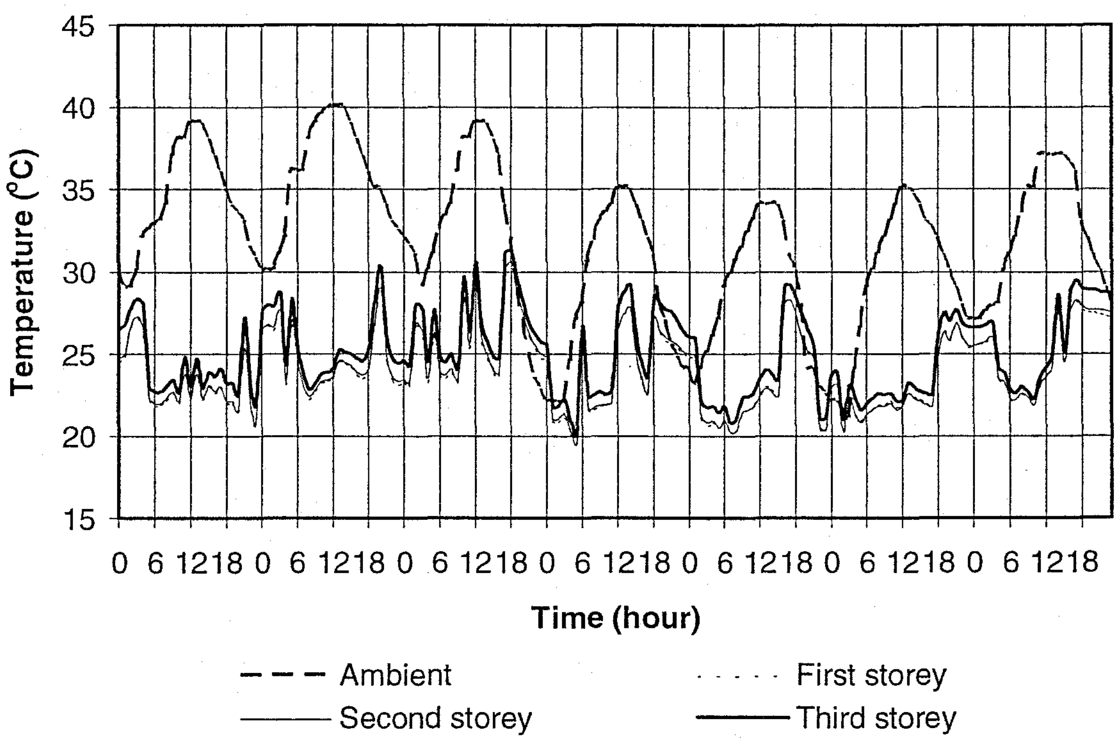


Figure 5.19: Comparison between internal and ambient temperature.

As can be seen from Figure 5.19, the temperature difference between the first and second storey is quite small. The average temperature for the first storey is 24.0°C and for the second storey is 24.1°C while for the third or top storey it is 25.0°C. The most likely explanation for this is that there is solar radiation on the roof in this zone. The air temperature in the third storey can be altered by varying the roof design which will be discussed later. When a comparison is made between the average outdoor and indoor air temperatures, the results show a considerable difference of about 7.8°C for the first and second storey and 6.8°C for the third storey, which is a significant improvement.

As can be seen from the figure above, evaporative cooling is more efficient in the day time, than the night time. This is because of the lower humidity level during the day in comparison to the night. As a general statement it can be said that, the evaporative cooling systems can provide a lower indoor air temperature in warm climates where the air humidity level is low. The effect of no evaporative cooling system at night will be discussed later in Section 5.7.4.

Figure 5.20 illustrates the effect of continuous ventilation on the indoor air temperature with no evaporative cooling system and the hourly variation of the indoor air temperature of the second storey of the base case when the evaporative system is installed.

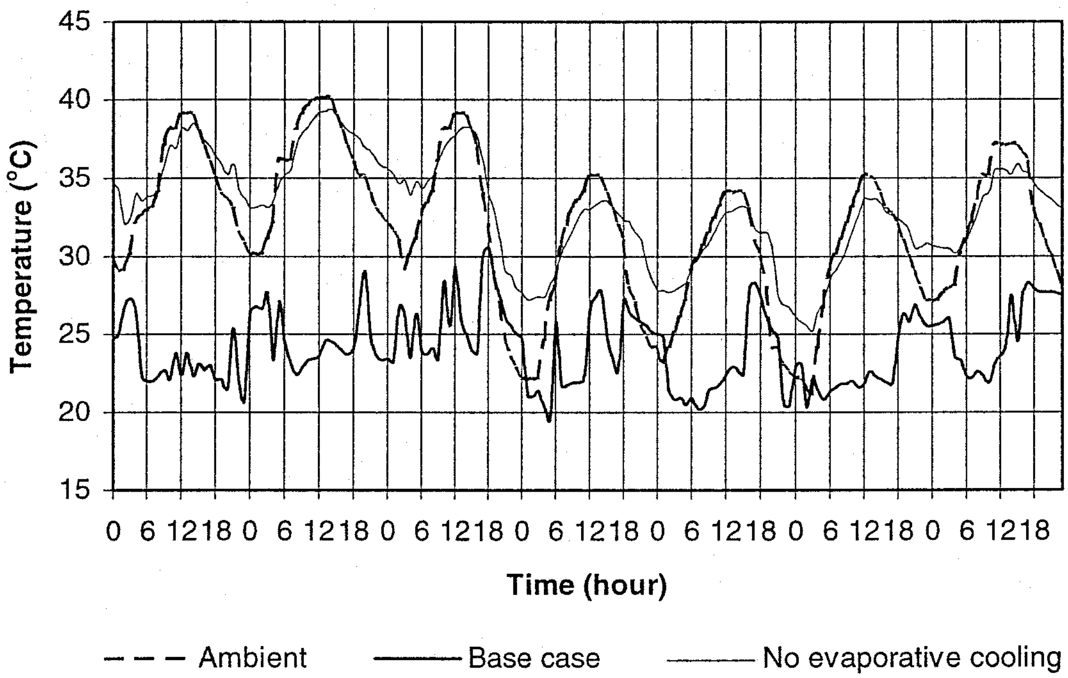


Figure 5.20: Comparison between ambient and internal temperature with and without evaporative cooling system.

It shows that the average of daily maximum inside air temperature with no evaporative cooling system is only 1.3°C lower than the average maximum ambient temperature. During the hottest hours of the week when the evaporative cooling system is installed however, the inside average maximum air temperature is reduced to 25.4°C which is 11.7°C less than the incoming ambient air temperature.

5.7.1 Effect of Walls and Roof Type

In the base case and in all previously discussed cases, the walls are assumed to be insulated with an average thermal transmittance of U = 0.7 W/m^2.K. Figure 5.21 shows the effect of wall insulation on the indoor air temperature of the second storey of the building. This figure compares an uninsulated wall with an average thermal transmittance of U = 2.4 W/m^2.K, a base case of current design and a highly insulated wall design which have an average thermal transmittance of U = 0.4 W/m^2.K.

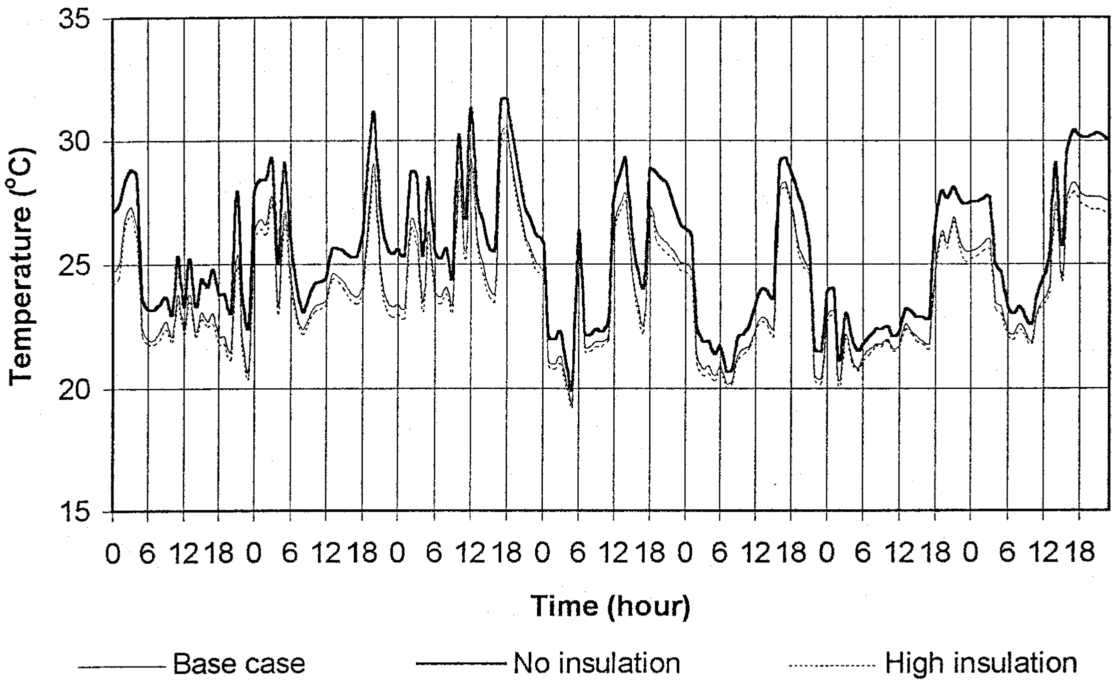


Figure 5.21: Indoor air temperature with various wall insulation.

The results show that the average temperature for the highly insulated case is about 0.4°C lower than the base case while this difference for an uninsulated case is about 1.4°C higher.

The effect of roof type and insulation on the indoor air temperature of the top storey is shown in Figure 5.22. The average level of air temperature for the highly insulated case with a thermal transmittance of U = 0.4 W/m^2.K and 100mm insulation (glass fibre) is 24.5°C which is 0.5°C lower than that at base case with 40mm insulation and thermal transmittance of U = 0.9 W/m^2.K. An examination of the results shows that for an uninsulated case with a thermal transmittance of U = 2.7 W/m^2.K, there will be a considerable temperature rise of about 2°C in compared to the base case.

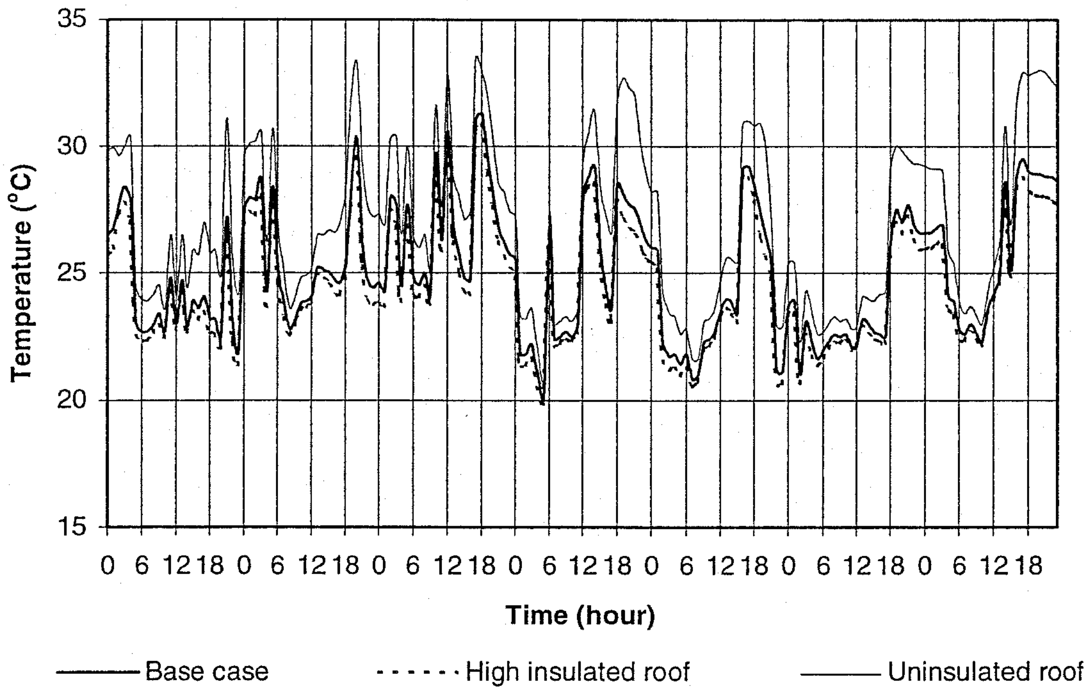


Figure 5.22: Effect of various roof insulation on the indoor air temperature.

5.7.2 The Effect of Interior Thermal Capacity

In all previously discussed cases it was assumed that there were no internal walls or furniture. It is well known that an interior thermal capacity is necessary in hot and arid or semi-arid climates. The thermal mass is simulated by different areas of brick wall. It is assumed that the wall consists of a 110mm brickwork with 10mm plastering on both sides of the wall with a total U-value of 3.2 W/m^2.K.

Figure 5.23 indicates that a 50 m^2 internal wall will reduce the average maximum room air temperature on the second storey by 1°C, and a 100 m^2 wall, by a further 0.7 $^\circ$C.

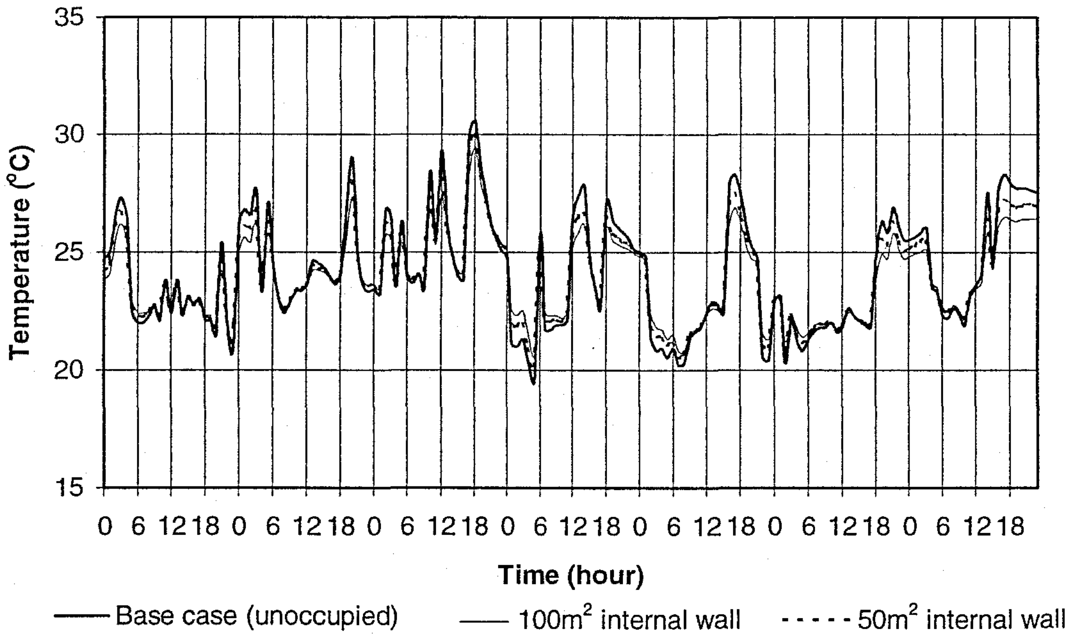


Figure 5.23: Influence of interior thermal capacity on the indoor air temperature.

5.7.3 Effect of Occupation

It has been assumed there are four occupants who provide continuous heating of 4px75 watt per storey, a continuous domestic heating of 250 watt for refrigerator and water heating, domestic heating of 500 watt from cooking (17.00-19.00 PM) and the other domestic heating such as light and television (17.00-23.00 PM) provide 200 watt free heat distributed over the day as follows:

- 00.00 - 17.00: 550 W per storey
- 17.00 - 19.00: 1050 W per storey
- 19.00 - 24.00: 800 W per storey

As can be seen from Figure 5.24 the influence of occupants and domestic energy is quite small.

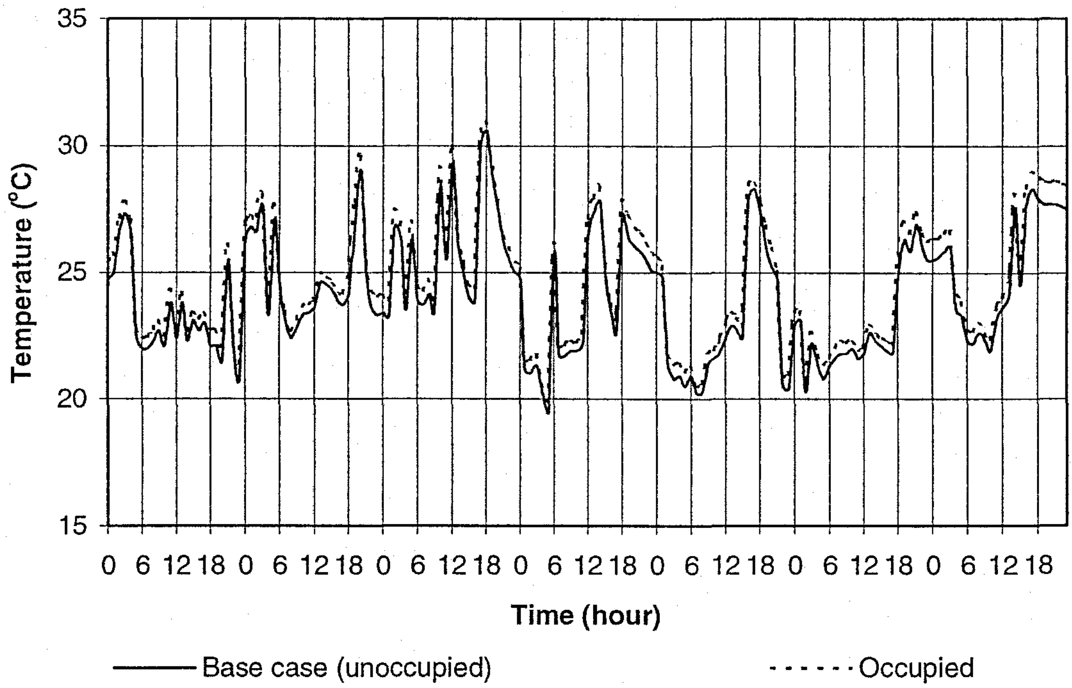


Figure 5.24: Influence of occupancy on the indoor air temperature.

5.7.4 Effect of No Evaporative Cooling at Night Time

During the night, the cool outdoor air can be used for passive cooling of the building in regions where the temperature difference between day and night time is about 20°K (Givoni, 1991). On the other hand, in regions such as Tehran the night time temperatures in summer are not cool enough to store in thermal mass and reduce the interior air temperature of the building during the following day. However, this section examines the effect of natural ventilation with no evaporative cooling system at night time (10 PM to 6 AM) on the indoor air temperature in a building with 100 m^2 interior wall.

Figure 5.25 shows the indoor air temperature on the second storey on the hottest week of the year. As can be seen, there is a temperature rise during the night when the evaporative cooling system does not operate. Looking at average level of indoor air temperature shows that there is a temperature rise of 1.2°C compared to the base case.

However, this temperature rise does not influence the day time temperature by very much.

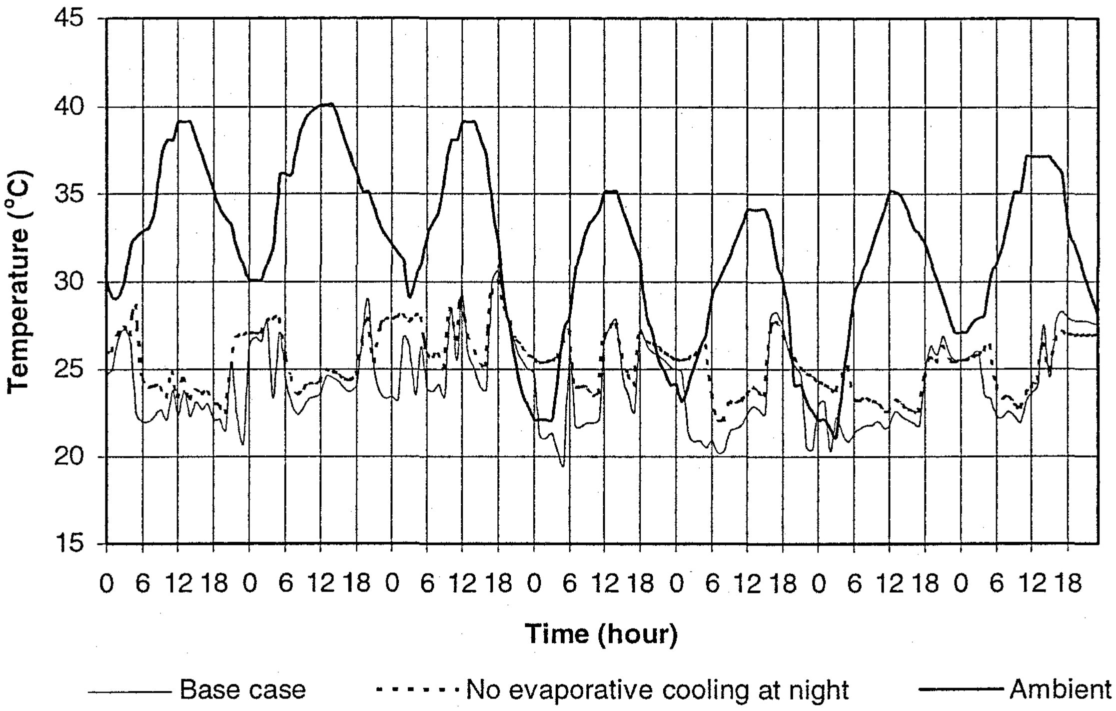


Figure 5.25: Effect of no evaporative cooling at night time on the indoor air temperature.

The following cases are designed to examine the effects of various night time ventilation rate with no evaporative cooling on the indoor air temperature. In the first case, the ventilation rate is reduced from the optimum flow rate to 3 ach.

It can be seen in Figure 5.26 that the indoor air condition pattern is similar in the day time to the base case. But, for the first three nights where the ambient temperature is high, a decrease in airflow rate causes a reduction in room air temperature during night. Obviously the improvement of the indoor condition is due to control of the introduction of outdoor warm air into the building by reducing the airflow rate. On the other hand, in periods where the night outdoor temperatures are cooler, this case gives rise to a worse condition. The average indoor air temperature in this case is 25.2°C which is 1.1°C higher than the base case but, the average maximum daytime temperature remains almost same as the base case.

The second case examines the effect of no ventilation at night time. As can be seen from Figure 5.26, the case with 0 ach at night, provides a lower temperature than the previous one for the first three nights and a slightly higher temperature for the other nights.

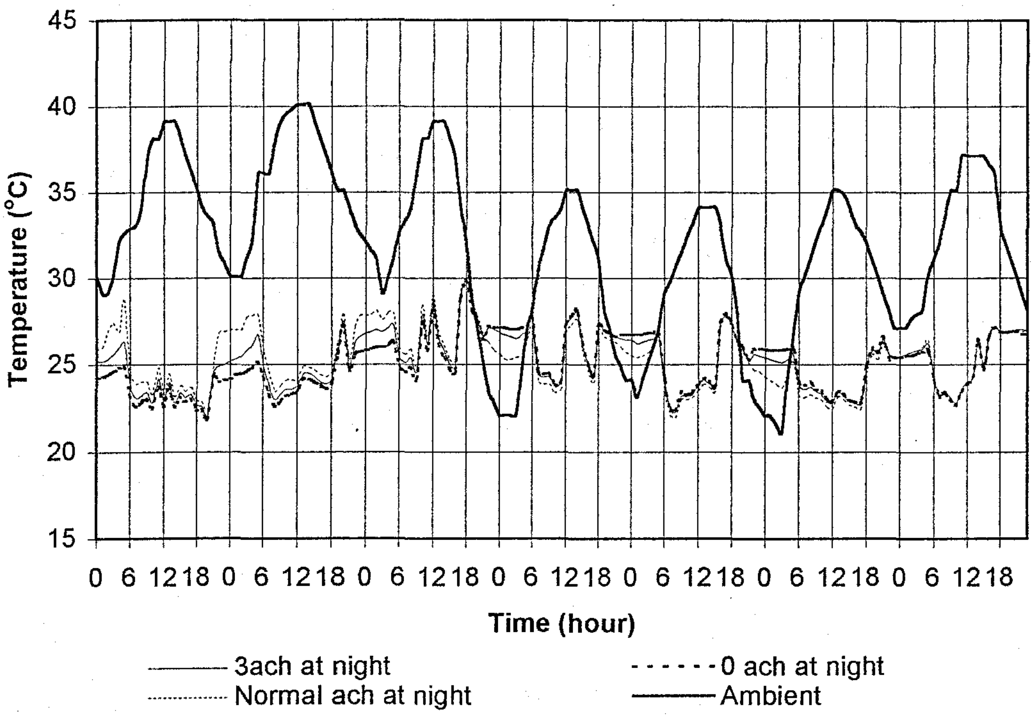


Figure 5.26: The effect on the indoor air temperature of various night time air change rate.

Therefore it can be concluded that, increasing the numbers of air changes, causes a rise in indoor air temperature for the first three nights where, the outdoor air temperature is grater than about 28°C. On the other hand, in the temperature ranges below 28°C, increasing in the number of air changes, provides a lower indoor air temperature.

5.7.5 Final Case Presentation

The final case sets out the hourly temperatures of the hottest week of the year for a three storey building with outer walls of U = 0.7 W/m^2.K and roof of U = 0.4 W/m^2.K. The building is assumed to be occupied as explained in Section 5.7.3 and have 100 m^2 internal wall as a thermal capacity. It is also assumed that the evaporative cooling system does not operate from 10 PM to 6 AM and the numbers of air changes per hour for this period are same as the base case with optimum airflow rate values. Details of building sections and materials are presented in Appendix A.5.

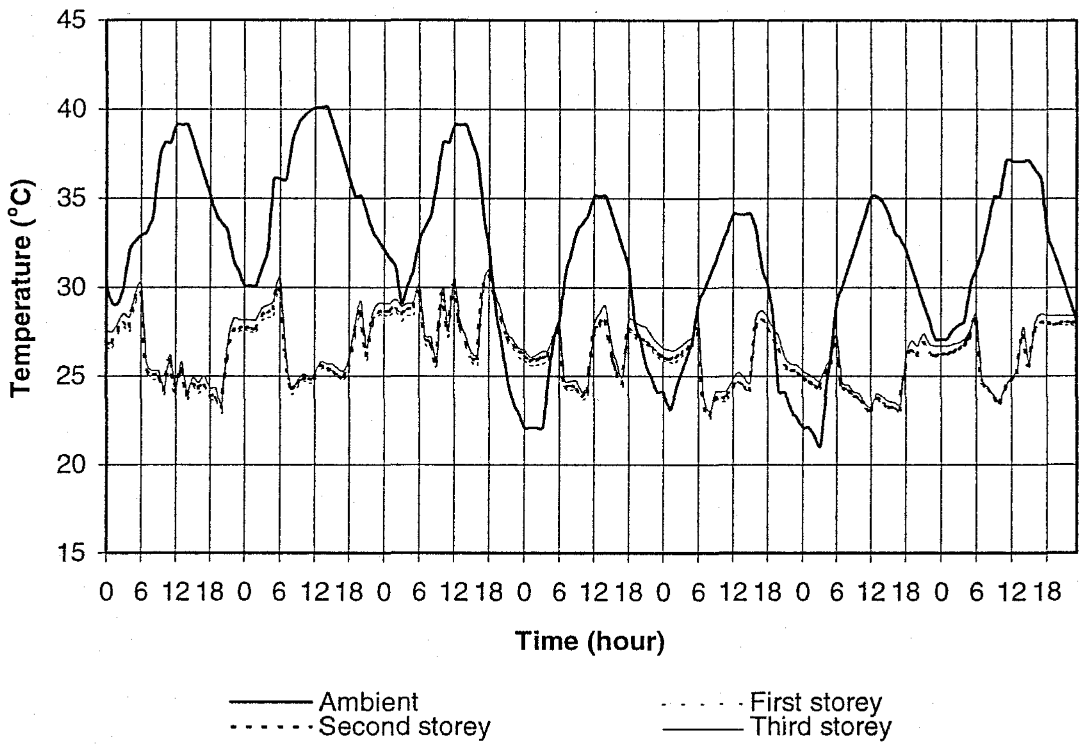


Figure 5.27: Comparison between internal and ambient temperature for the final case.

Figure 5.27 demonstrates indoor air temperature of the three zones of the building based on the above mentioned parameters compared to the outdoor air temperature. Table 5.5 gives the weekly average outdoor and indoor (middle storey) air temperatures for the day and night time. A comparison between the average daily indoor and outdoor air temperature shows a 8.6°C lower indoor air temperature. As a final case, results of simulation indicate that by the proposed passive evaporative cooling system, average of daily maximum ambient temperature which is 37.1°C, will be reduced at the same time to 25.5°C which is 11.6°C lower than the outdoor air temperature.

Table 5.5: Average air temperatures of the ambient and the final case.

	Average temperature (^{o}C)	Daily average temperature (^{o}C) (6 AM to 10 PM)	Nightly average temperature (^{o}C) (10 PM to 6 AM)	Average of maximum daily temperature (^{o}C)
External	31.8	34.2	27.8	37.1
Final case	26.2	25.6	27.1	25.5

Results also demonstrate a $1.5^{o}C$ lower temperature for the average indoor daily temperature compared to the night time. The thermal comfort level provided by this cooling system and the various factors influencing the thermal comfort will be discussed in the next chapter.

5.8 CONCLUSION AND RECOMMENDATIONS

The effects of various parameters on the airflow rate through the building, efficiency of the system and the indoor air temperature were considered in this chapter. It was found that, the airflow rate through the building can be significantly affected by shielding and terrain factors rather than the building shape. The value of the airflow rate through the building can be increased by increasing the size of opening area of the windows. On the other hand, increasing the airflow rate has a negative effect on the evaporative cooling efficiency. The results indicated that, the maximum cooling effect on the building was achieved at the optimum airflow rate. In this study, a design parameter of 0.03 was defined as an optimum volumetric airflow rate per square meter of the surface area of the water in the evaporative cooling system which is independent to the volume of the building.

In the study of the effect of wind direction on efficiency, it was found that, an oblique wind direction was more efficient than the perpendicular one. However, a conservative minimum efficiency due to perpendicular direction was chosen for calculation. Also, only wind directions of $-90^{o}<\beta<90^{o}$ were considered to calculate the evaporative cooling efficiency of the system.

Simulation results of the effect of filament arrangement on the efficiency of the evaporative cooling system showed that, a system with smaller water line diameter (3mm) and also smaller transverse pitches (3D) had a higher efficiency than the others. Even when the number of rows were taken into consideration, a system with smaller pitches and the less number of rows was more efficient than the larger pitches and more rows. However, in comparison between the systems with various numbers of rows and different width, a system with more rows but smaller width was more efficient than a system with wider length and fewer rows.

The results of the study of the effect of various parameters on the indoor air temperature showed that design changes can significantly improve the thermal performance of a building. By controlling the zone parameters that influence the interaction between the air conditioned by an evaporative cooling system and the building's thermal mass, one can actively reduce indoor air temperature. The reduction in average zone air temperature is significant; the analysis indicated as much as 2.5^{o}C difference from one case to another.

The conclusion of the simulations confirmed that a good building design in the context of this study must have thermal insulation in walls and roof corresponding to at least 35 mm mineral wool or cellular plastics, and also a high internal capacity.

Evaporative cooling system can be turned off at night time and the building and the ventilation rate can be altered to obtain the desirable condition. The effect of various rates of night time ventilation on the thermal comfort of occupants will be discussed later in Chapter 6.

CHAPTER 6

THERMAL COMFORT

6.1 INTRODUCTION

The present chapter has two parts. The first discusses the conditions necessary for optimal thermal environments for human beings and the methods of evaluating a given thermal environment. The aim of this part is the estimation of the thermal comfort in environment which is created by the proposed passive evaporative cooling system. The second part presents the climatic range for comfort evaporative cooling. In this part the minimum climatic conditions required to achieve an appropriate comfort zone will be studied.

6.2 PART I: INDOOR THERMAL COMFORT ACHIEVED BY THE SYSTEM

Thermal comfort is defined as "that condition of mind which expresses satisfaction with the thermal environment" (ASHRAE, 1992). If a group of people is subjected to the same room climate, it will not be possible, due to biological variance, to satisfy everyone at the same time. One must then aim at creating optimal thermal comfort for the group, i.e., a condition in which the highest possible percentage of the group is in thermal comfort (Fanger, 1970).

Since people have variable preferences, a simple definition of comfort is almost impossible. However, ignoring all factors but temperature sensations, it seems certain that most people are comfortable when maintaining normal body temperature without expending effort to counteract either the gain or loss of body heat (Watt, 1963).

6.2.1 Physiological Comfort Conditions

Physiologically uncomfortable conditions in arid climates are mainly caused by the extreme heat and dryness. In hot dry conditions a person is comfortable when his/her body is able to dissipate to the surroundings all the heat it produces, including heat lost by evaporation from the skin and from the respiratory system. Under normal sedentary conditions in a comfortable environment an average person gains or loses heat in three ways in approximately the following proportions (Saini, 1980):

- Convection 30 per cent,
- Radiation 45 per cent,
- Evaporation 25 per cent.

Only a minimal amount of heat is dissipated by conduction. Loss of heat by convection and radiation can take place only when the air and surroundings are at less than skin temperature. Loss by evaporation depends upon the relative humidity of the surrounding air and the rate of air movement. Air movement will assist in cooling the body only when the air temperature is less than that of the skin and the relative humidity of the atmosphere is not very high. Consequently, up to a point beyond which heat alone becomes a factor of discomfort, hot dry air tends to be more comfortable than hot humid air (Saini, 1980).

The ideal occurs therefore when the body's heat losses exactly balance its metabolic heat generation so that bodily temperature neither tends to increase nor decrease. Comfort thus results when the environment accepts bodily heat exactly at its rate of production. This point of balance necessarily varies with individual activity, metabolism, and amount of clothing (Watt, 1963).

The human thermoregulatory system depends on the maintenance of a body temperature of about 37°C without excessive demands on the body to increase heating by shivering or by exercise, or to increase cooling by evaporation (Linacre and Hobbs, 1977).

The sensible heat loss can be altered by a variation of the sub-cutaneous blood flow and thus of the skin temperature; latent heat loss can be increased by sweat secretion, and internal heat production can be increased by shivering or muscle tension. These mechanisms are extremely effective and ensure that the heat balance can be maintained within wide limits of the environmental variables. Maintenance of the heat balance is, however, far from being a sufficient condition for thermal comfort. Within the wide limits of the environmental variables for which heat balance can be maintained, there is only a narrow interval which will create thermal comfort (Fanger, 1977).

Human thermal sensation is related to the state of the body's thermoregulatory system, the degree of discomfort being greater the heavier the load on the effector mechanism. Experiments by Yaglou (1927) indicated a correlation between the skin temperature and the sensation of thermal comfort; later and more complete studies by Gagge et al. (1937) and Winslow et al. (1937) showed a correlation between thermal sensation and skin temperature, independent of whether the subjects were nude or clothed. Therefore, it was generally accepted for a long time that the physiological conditions for comfort were that a person had a mean skin temperature of 33-34°C and that sweating (or shivering) did not occur. This was later confirmed in experiments by Fanger for sedentary subjects. At activities with a higher metabolic rate than sedentary ones, man prefers a lower mean skin temperature and prefers to sweat (Fanger, 1967 and McNal et al. 1967). By setting up a heat balance equation for the human body, Fanger (1970) then used comfort values of skin temperature and sweat secretion (both as a function of the metabolic rate) to derive his comfort equation.

Gagge et al. (1967, 1969 and 1973) found that cold discomfort is related to skin temperature while warm discomfort is more closely related to the wettedness of the skin, defined as the relation between the actual evaporation from the skin and the maximum possible evaporation from a completely wet skin. Gagge later used these observations in his derivation of the New Effective Temperature scale (ET*).

6.2.2 Effective Temperature (ET*)

Results of tests carried out by Gagge and his co-workers (1971) in environmental chambers have shown that skin wettedness is an excellent predictor of discomfort. In the ET* scale both skin temperature, t_S, and skin wettedness, w, are used to define the thermal state of a person. The skin wettedness, w, is the ratio of the actual evaporative loss at the skin surface to the maximum loss that could occur in the same environment, i.e. when the skin is completely wet This scale has been adopted by ASHRAE (1985) to replace the old ET scale.

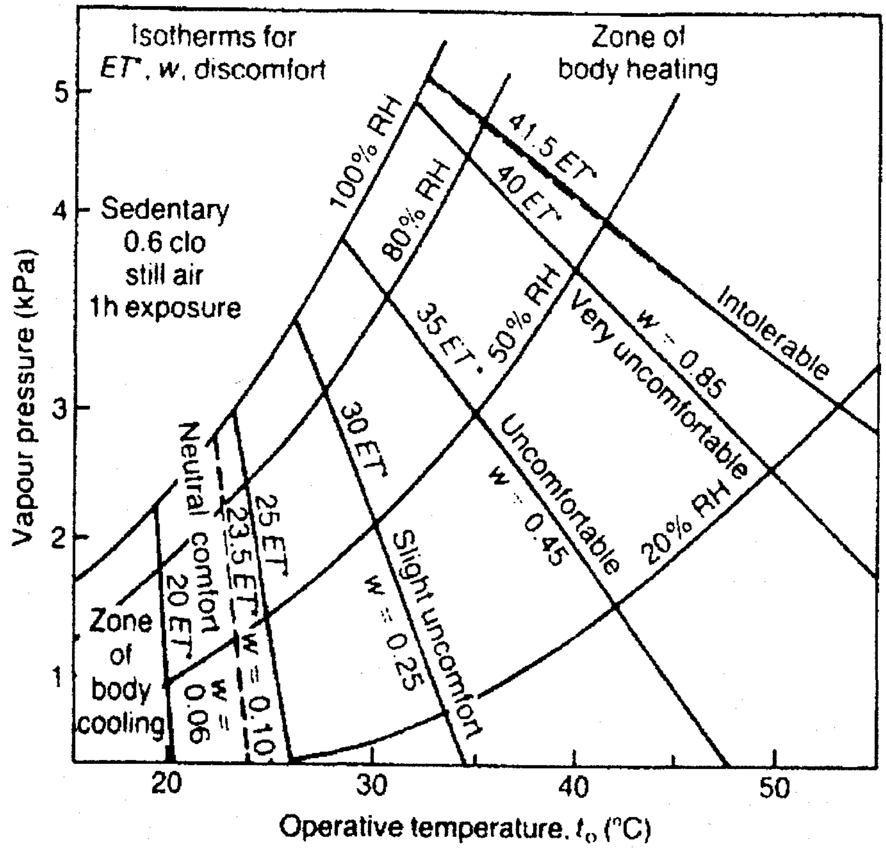


Figure 6.1: The new effective temperature scale-lines of constant ET* (ASHRAE, 1985).

The ASHRAE effective temperature (ET*) is the dry-bulb temperature of uniform enclosure at 50% relative humidity (rh) in which people have the same net heat exchange by radiation, convection and evaporation as they do in varying humidities of the test environment. The ET* scale is based on equal air and operative temperatures, a clothing resistance of 0.6 clo, an air speed of 0.2 m/s, a sedentary activity (≈1 met) and an exposure time of 1 h. As shown in Figure 6.1 the new effective temperature for thermal comfort at these conditions and an rh of 50% is 23.5°C for w = 0.06, i.e. no sweating. The figure also gives the values of ET* for different skin wettedness ratios. The principle used to develop ET* has been extended to higher levels of activitiy, representing higher levels of sweating. These are discussed in the ASHRAE handbook (1985).

6.2.3 Operative Temperature (t_o)

The ASHRAE (Standard 55-1981) uses the operative temperature as the environmental variable for evaluating thermal comfort at different activity and clothing insulation levels. It is defined as the uniform temperature of a radiantly black enclosure in which an occupant would exchange the same amount of heat by radiation plus convection as in the actual non-uniform environment. The ASHRAE standard specifies environmental conditions that are acceptable to 80% or more of the occupants. It is mainly applicable to sedentary activity (≈1.2 met) with normal winter or summer clothing ensembles, i.e. 0.8-1.2 clo winter clothing or 0.6-0.8 clo summer clothing. The acceptable range of operative temperature and humidity for winter and summer by the shaded areas in the psychrometric chart of Figure 6.2. The figure shows an overlap of the winter and summer zone in the range of t_o = 23-24°C because in this region people in summer clothing would tend to feel slightly cool while those in winter garments would be near the slightly warm sensation.

The maximal average air speed in the occupied zone is specified by the standard as 0.15 m/s in winter and 0.25 m/s in summer environments. However, in summer, the comfort zone (see Figure 6.2) may be extended beyond 26°C operative temperature if the average air speed is increased by 0.275 m/s per 1 K of increased temperature to a maximum of 0.8 m/s. Air speeds of 0.8 m/s are unacceptably high in normal occupancy as loose paper, hair and other objects may be blown about at this speed.

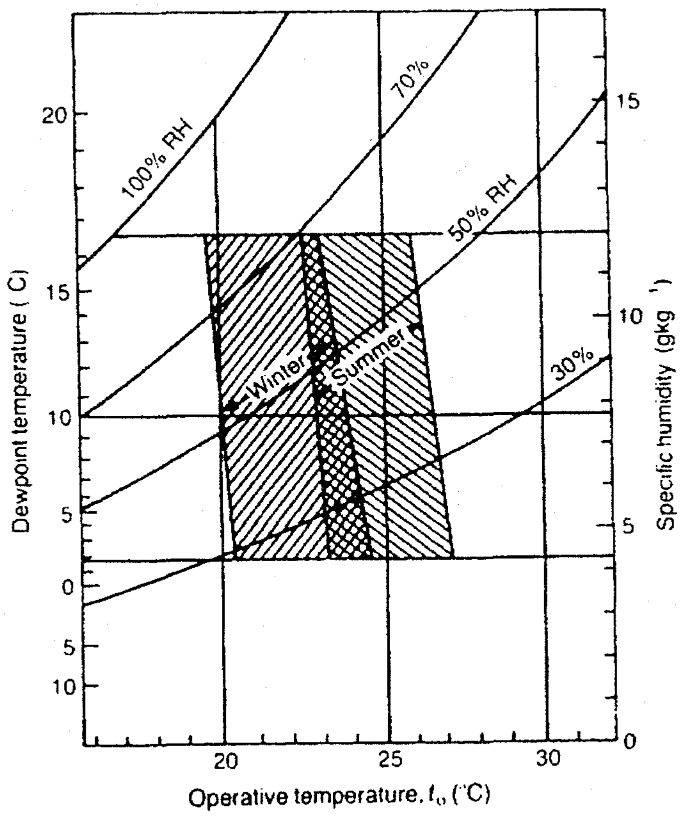


Figure 6.2: Acceptable ranges of operative temperature and humidity for winter and summer clothing and sedentary activity, ≈ 1.2 met (ASHRAE, 1985).

The comfort zone temperature in Figure 6.2 should be decreased when the activity level is higher than sedentary, i.e. M>1.2 met The operative temperature for activity range 1.2<M<3 is obtained using (ASHRAE, 1985):

$$t_0 \text{ active} = t_0 \text{ sedentary} - 3(1+clo)(met-1.2) \tag{6.1}$$

The minimum allowable temperature is 15°C.

6.2.4 Variables Influencing Comfort

The most important variables which influence the condition of thermal comfort are:

(a) Air temperature
(b) Mean radiant temperature
(c) Relative air velocity
(d) Humidity (water vapour pressure in the ambient air)
(e) Activity level
(f) Thermal resistance of clothing.

Thermal comfort can be achieved by many different combinations of the above variables and therefore also by the use of many fundamentally different technical systems.

Air temperature is the most commonly used measure of thermal comfort but it is not the complete definition. It is called shade temperature when describing the outside or ambient temperature and dry bulb temperature when used to refer to the indoor air temperature.

Mean radiant temperature is defined as the weighted average temperature of all the exposed surfaces in a given space (obtained by averaging the temperatures of the surfaces to which the body is exposed, weighted by the solid angle subtended by each). For example, the mean radiant temperature of a room in winter will be considerably influenced by a large uncurtained window which is very cold.

Relative air velocity or air movement is an important aspect of thermal comfort, especially in warm or hot climates. Air moving over the skin increases heat loss by convection and if the skin is wet further cooling results from evaporation.

Air movement accelerates convective heat dissipation from the skin, as long as the air temperature is less than the skin temperature. It also increases the evaporative heat loss. If the air temperature is slightly above skin temperature, there may be a convective heat input, but the evaporative cooling effect is grater, thus the net result is cooling. Beyond a certain point the convective heat input may be greater than the evaporative cooling effect. The utilisation of air movement for cooling purposes is restricted by its non-thermal effects. General subjective reactions are:

< 0.25	m/s	unnoticed
0.25 - 0.5	m/s	pleasant
0.5 - 1.0	m/s	awareness of air movement
1.0 - 1.5	m/s	draughty
> 1.5	m/s	annoyingly draughty

Under hot conditions 1.0 m/s is normally considered pleasant and 1.5 m/s may be acceptable. Under cold conditions 0.25 m/s should not be exceeded, but less than 0.1 m/s would create a feeling of stuffiness (Szokolay, 1987).

Humidity is usually an important consideration only when the air temperature is close to or above the upper limits of thermal comfort and the relative humidity is above 70 percent or below 30 percent. When the relative humidity levels are high then air movement becomes an important mechanism for removing the heat the body generates.

Activity (metabolic rate), the rate of energy production of the body is described by the 'met' unit. This factor is characteristic of different activities; Table 6.1 gives some typical values. They range from 41 W/m^2 for sleeping to 200 W/m^2 and over for prolonged heavy physical work or athletics. Walking slowly on the level has a value of 116 W/m^2, sitting, 58 W/m^2 and standing 70 W/m^2. The "sitting" rate is defined as the basic unit of activity, and is equivalent to 1 'met'. Any activity, of X W/m^2, is equivalent to X/58 in met units (Markus & Morris, 1980).

The value of η depends on the mechanical efficiency of the activity. Table 6.1 also gives a typical range. It will be seen that for many common activities it is 0 or low; it never exceeds 0.2.

Table 6.1: Metabolic rate of different activities with mechanical efficiency and relative still air velocity - i.e. velocity induce by the activity itself (based on Fanger and ASHRAE, 1985).

Activity	Metabolic rate, M/A_{DU} (W/m^2)	Mechanical efficiency, η	Relative velocity in still air (m/s)
Sleeping	41	0	0
Reclining	47	0	0
Sitting	58	0	0
Standing, relaxed	70	0	0
Walking, level, at 3.2 km/h	116	0	0.9
Walking, level, at 4.8 km/h	151	0	1.3
Walking, level, at 6.4 km/h	221	0	1.8
Walking, 15° upward slope, at 3.2 km/h	267	0.1	0.9
House cleaning	116-198	0-0.1	0.1-0.3
Typing	70-81	0	0.05
Gymnastics	175-233	0-0.1	0.5-2.0
Dancing	140-256	0	0.2-2.0
Sawing by hand	232-280	0.1-0.2	0.1-0.2
Heavy machine work (e.g. steel forming)	204-262	0-0.1	0.02

Clothing, transfer of dry heat between the skin and the outer surface of the clothed body is quite complicated, involving internal convection and radiation processes in intervening air spaces, and the conduction through the cloth itself. The clothing unit of 'clo' was introduced by Gagge et al. (1942) and it represents the amount of insulation required to maintain a sedentary person indefinitely comfortable in an environment of 21°C and 10 cm/s air movement. Typical values in clo units are given in Table 6.2; it will be seen that these range from 0, for nude conditions, to about 4.0 for heavy Arctic uniform. Normal outdoor winter clothes will range from 1.5 to 2.0; whilst the light clothes likely to be worn in the warm tropics will range from about 0.3 to 0.5.

Table 6.2: Values of I_{cl} and f_{cl} for various clothing ensembles (based on Fanger, 1970 and Gagge & Nevins, 1977).

Clothing ensemble	Insulation, I_{cl} (clo)	Ratio of surface area of the clothed body to surface area of nude body, f_{cl}
Nude	0	1.0
Shorts	0.1	1.0
Typical women's tropical ensemble	0.15-0.25	1.05
Typical men's tropical ensemble	0.3-0.4	1.05
Women's light summer clothing	0.4-0.5	1.05
Men's light summer clothing	0.4-0.6	1.1
Typical business suit	1	1.15
Typical business suit + cotton coat	1.5	1.15
Heavy wool pile ensemble	3-4	1.3-1.5

6.2.4.1 The Influence of Certain Special Factors on Thermal Comfort

Everyone is not alike. How then is it possible, from an equation, to specify one particular temperature which will provide comfort. Fanger's answer (1977) is that this temperature does not necessarily satisfy everyone. It gives, however, combinations of the variables which will provide comfort for the greatest number of people.

6.2.4.1.1 Age

It has often been claimed that due to the fact that metabolism decreases slightly with age, the comfort conditions based on experiments with young and healthy subjects cannot be used as a matter of course for other age groups. Studies by Fanger (1970), Rohles and Johnson (1972) and Griffiths and McIntyre (1973) with the young and the elderly showed, however, no difference in the comfort conditions for the different age groups.

The lower metabolism in elderly people seems to be compensated by a lower evaporative loss.

6.2.4.1.2 Men and Women

The climate chamber studies of Kansas State University (Nevins et al. 1966, Rohles, 1974 and Rohles et al., 1975) and at the Technical University of Denmark (Fanger, 1970 and, Fanger and Langkilde, 1975) showed no significant differences between the comfort conditions for men and women. Women's skin temperature and evaporative heat loss are slightly lower than those for men, and this balances the slightly lower metabolism of women (Olesen and Fanger, 1973 and, Fanger and Langkilde, 1975). But in practice women may tend to wear less clothing and be slightly more sensitive to cold (Gagge and Nevins, 1976).

6.2.4.1.3 Seasonal and Circadian Rhythm

As adaptation seems to have only minor effects on man's thermal preference, there is a reason to expect major differences between comfort conditions in winter and in summer. This was confirmed by an investigation where results of winter and summer experiments showed no difference (McNall et al., 1968).

It is reasonable to expect the comfort conditions to alter during the day as the internal body temperature has a daily rhythm, a maximum occurring late in the afternoon and a minimum early in the morning. Fanger investigated this experimentally by comparing the preferred temperature for subjects in the morning and in the evening (Fanger et al., 1974), during a normal 8-hour simulated working day and during the night shiftwork. No significant difference was found in the ambient temperature preferred during a 24-hour period, provided that the activity, clothing and the other environmental parameters were the same.

6.2.4.1.4 Colour

It seems to be a commonly accepted idea that colour in a room influences the feeling of warmth. The idea was put forward that by using warm colours (red and yellow) on walls or by the use of reddish lighting, a psychological feeling of heat could be conveyed to people, so that thermal comfort could possibly be maintained at lower ambient temperatures. Similarly in summer cold colours should be aimed at, or blue lighting used.

The results of experimental studies by Houghten et al. (1940) and Fanger et al. (1977), are in agreement with the fact that the colour of the surrounding surfaces has no influence on man's heat loss. As colour has no thermal influence on man, any influence on the thermal sensation must therefore be of a psychological nature.

6.2.4.1.5 Climate and Weather Effects

It is widely believed that, by exposure to hot or cold surroundings, people can acclimatise themselves so that they prefer other thermal environments, and that the comfort conditions vary in different parts of the world, depending on the outdoor climate. However, McNall et al. (1968) could observe little variability between summer and winter when the insulation values of clothing were held constant under laboratory conditions. From this Fanger (1970) concluded that the adjustments of acclimatisation could only affect degrees of strain experienced under thermal stress, but not the level of neutrality itself. Fishman and Pimbert (1979) on the other hand, have found that neutrality indeed may be seasonally constant even in real life situations, but only because clothing could be adjusted by individuals to give levels of warmth.

Field surveys in non-airconditioned places throughout the world including those in Australia show that people will adjust to given environmental conditions and temperature neutrality (i.e. the absence of thermal sensations at minimal physiological thermo-regulatory activity) depends upon the usual thermal levels encountered (Humphreys

1975, 1976, and Auliciems, 1977, 1983). The following sections will present various methods of thermal comfort prediction.

6.2.5 Thermal Comfort Prediction, Based on Mean Outdoor Temperature

By analysing a large number of field surveys, Humphreys (1976, 1978 and 1981) has shown that the neutral thermal comfort vote of occupants in both controlled and free running buildings is determined by the mean indoor and outdoor temperatures that they experience. The relationship between the mean indoor temperature and the comfort temperature in Figure 6.3 shows that a fixed operating temperature is more a control convenience than a comfort requirement.

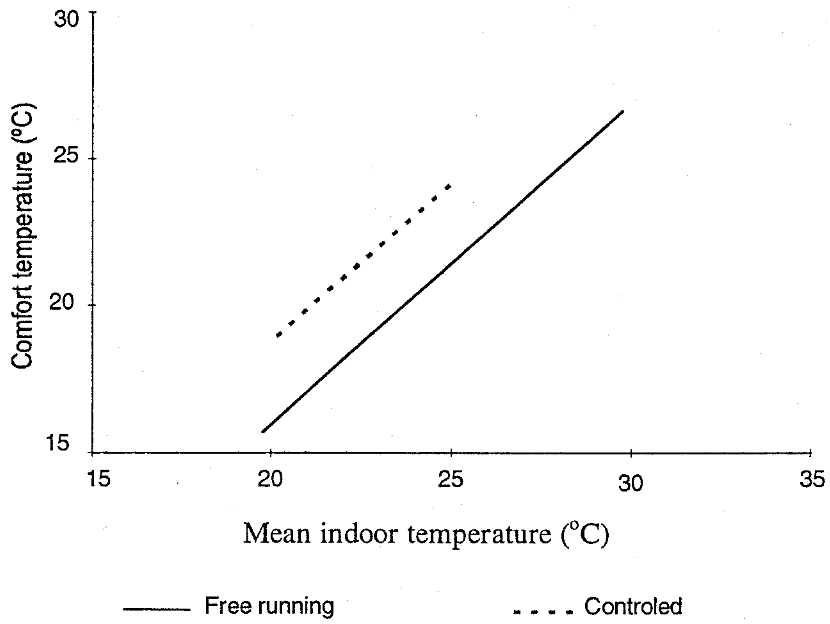


Figure 6.3: The relationship between the mean indoor temperature and comfort temperature (Humphreys, 1981).

The relationship between the mean outdoor temperature and the comfort temperature is strong for free running buildings, where the occupants are usually more able to adapt to the conditions, and weaker, but still significant, for controlled buildings. Humphreys equations for the relationship between the comfort temperature, t_c, and the mean monthly outdoor temperature, t_m, are given below for the two building types:

Free running: $$t_c = 0.53t_m + 11.9\ (^{\circ}C) \quad (6.2)$$

Controlled: $$t_c = 0.16t_m + 18.6\ (^{\circ}C) \quad (6.3)$$

Neither of Humphreys' equations on their own is a basis for a climatically derived comfort temperature for controlled buildings. Occupants of free running buildings are able to adapt to conditions in way which will probably not be available to occupants of controlled buildings, and the equation for controlled buildings is based on a situation where the indoor temperature is held relatively static, which will be a strong influence on the comfort vote. Although occupants of controlled buildings are not generally subjected to varying temperatures, Figure 6.3 shows that they will adapt to a range of different fixed temperatures.

Auliciems and deDear (1986) analysed a large number of field surveys, excluding some of Humphreys data, such as that for school children, and including some more recent studies, and drived the following comfort temperature equation based on mean outdoor temperature:

$$t_c = 0.31t_m + 17.6\ (^{\circ}C) \quad (6.4)$$

with the proviso that $18.5 < t_c < 28.5\ ^{\circ}C$

Nicol (1995) presented a similar equation using regression of t_c on t_m for thermal comfort in Pakistan.

$$t_c = 0.38\ t_m + 17.0\ (^{\circ}C) \quad (6.5)$$

Based on Equations 6.2, 6.4 and 6.5, the thermal comfort temperature for Tehran with annual outdoor mean temperature of 17.1°C will be 21.0°C, 22.9°C and 23.5°C respectively. By the same token, the summer comfort temperature with mean summer temperature of 29°C will be 27.3°C , 26.6°C and 28.0°C. Taking the average of the above temperatures, the annual comfort temperature for Tehran will be about 22.5°C and the summer comfort temperature will be around 27.3°C.

Szokolay (1987) has argued that the width of the comfort zone is taken as ± 2 K about the thermal neutrality, if t_m is an annual mean temperature and ± 1.75 K if t_m is a monthly mean. According to him, the comfort zone can be plotted on the psychrometric chart, by the following procedure:

1. find the annual mean temperature, t_m,
2. find the thermal neutrality (thermal comfort temperature) t_c from the above expression,
3. plot this t_c on the chart, on the 50% RH curve,
4. mark the lower: $L = t_c - 2$ and upper: $U = t_c + 2$ limits on the 50% RH curve,
5. draw the corresponding standard effective temperature lines, as the side boundaries,
6. mark the upper humidity ratio boundary at the 12 g/kg level and the lower boundary at the 4 g/kg level.

This comfort zone will be valid for lightly clothed people at sedentary work. For heavier physical activities the t_c should be adjusted:

for light work	(210 W): -2 K
for medium work	(300 W): -4.5 K
for heavy work	(400 W): -7 K

Figure 6.4 shows the summer comfort zone for Tehran on the psychrometric chart with an summer mean temperature of $t_m = 29\,^{\circ}\mathrm{C}$.

From Equation 6.4:

$$t_c = 0.31 \times 29 + 17.6 = 26.6\,^{\circ}\mathrm{C}$$

The limits will be:

$L = 26.6 - 2 = 24.6$

$U = 26.6 + 2 = 28.6$

The slope expression becomes:

0.025 x (24.6 - 14) = 0.26 K/(g/kg)

0.025 x (28.6-14) = 0.36 K/(g/kg)

From the psychrometric chart:

humidity ratio for L = 9.5 g/kg

humidity ratio for U = 12 g/kg

Thus the base-line intercepts of the two boundaries will be:

24.6 + (9.5 x 0.26) = 27.1

28.6 + (12 x 0.36) = 32.9

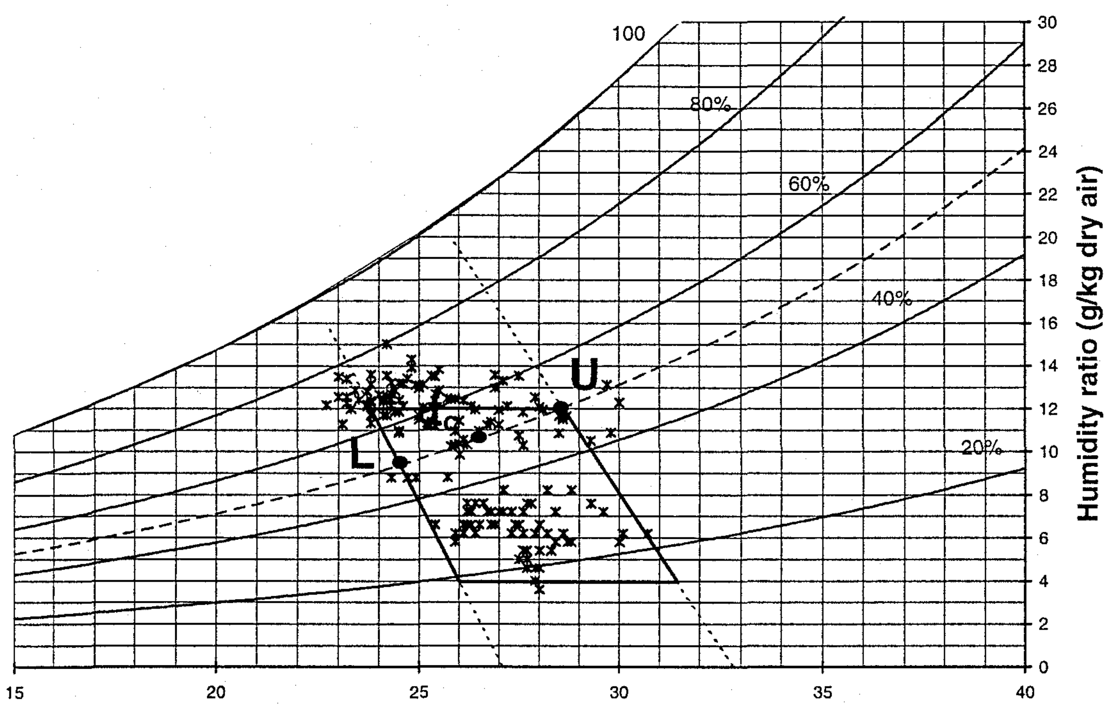


Figure 6.4: Tehran summer comfort zone on the psychrometric chart.

Indoor air temperature and humidity ratio were achieved in the final case, in Chapter 5, are also plotted in this figure. As can be seen, indoor air temperatures are almost in summer comfort range. Considering this chart is based on no significant air movement, in the cases where the humidity ratio are above comfort range, the actual air movement inside the building, will provide a comfortable condition.

6.2.6 Thermal Comfort Prediction, Based on Predicted Mean Vote (PMV)

The most widely accepted index of thermal sensation today is Fanger's "Predicted Mean Vote" (PMV) (1970). The comfort index used throughout this study is based on the analytical model developed by Fanger. This model accounts for the energy exchange between the human body and the environment. The comfort index, PMV, given by the model is a function of:

- Metabolic heat production,
- Clothing insulation,
- Dry-bulb temperature,
- Mean radiant temperature,
- Air velocity,
- Water vapour pressure.

As a measure of the thermal sensation, the commonly used seven point psycho-physical ASHRAE scale is used by Fanger:

+3	hot
+2	warm
+1	slightly warm
0	neutral
-1	slightly cool
-2	cool
-3	cold

The scale is symmetrical around the zero point, so that a positive value corresponds to the warm side and a negative value to the cold side of neutral. Fanger's method suggested that an individual whose thermal sensation was warmer than slightly warm (+1) or cooler than slightly cool (-1) would be dissatisfied with the environment and would be likely to complain.

In this study, the time-dependent PMV which is a complex mathematical expression involving activity, clothing and the four environmental parameters, was generated from Fanger's expression:

$$\textbf{PMV} = \left(0.352\, e^{-0.042(M/A_{DU})} + 0.032\right) \Big[M/A_{DU}\,(1-\eta) - 0.35\left(43 - 0.061\, M/A_{DU}\,(1-\eta) - p_a\right) - 0.42\left(M/A_{DU}\,(1-\eta) - 50\right) - 0.0023\, M/A_{DU}\,(1-p_a) - 0.0014\, M/A_{DU}\,(34 - t_a) - 3.410^{-8} f_{cl}\left((t_{cl} + 273)^4 - (t_{mrt}+273)^4\right) - f_{cl}\, h_c\,(t_{cl} - t_a)\Big] \qquad \textbf{(6.6)}$$

where:

M/A_{DU}= metabolic rate (W/m^2)

η = mechanical efficiency

p_a = pressure of water vapour in ambient air (mmHg)

t_a = air temperature (^oC)

f_{cl} = the ratio of the surface area of the clothed body to the surface area of the nude body

t_{cl} = temperature of clothing surface (^oC)

t_{mrt} = mean radiant temperature (^oC)

h_c = convection coefficient.

The t_{cl} is determined by the equation:

$$\mathbf{t_{cl}} = 35.7-0.032\, M/A_{DU}\,(1-\eta) - 0.18\, I_{cl}\Big[3.410^{-8} f_{cl}\left((t_{cl} + 273)^4 - (t_{mrt}+273)^4\right) + f_{cl}\, h_c\,(t_{cl} - t_a)\Big] \qquad \textbf{(6.7)}$$

and h_c by:

$$\mathbf{h_c} = \begin{cases} 2.05\,(t_{cl} - t_a)^{0.25} & \text{for } 2.05\,(t_{cl} - t_a)^{0.25} > 10.4\sqrt{v} \\ 10.4\sqrt{v} & \text{for } 2.05\,(t_{cl} - t_a)^{0.25} < 10.4\sqrt{v} \end{cases} \qquad \textbf{(6.8)}$$

Where I_{cl} is insulation of clothing in clo unit and v is relative air velocity (m/s).

Equation 6.7 is a transcendental equation which can only be solved by an iterative process.

6.2.6.1 Predicted Percentage of Dissatisfied (PPD)

From the experimental data available to him, Fanger correlated the percentage ratio of the people who were dissatisfied with the thermal environment with the predicted mean vote. This ratio was called the predicted percentage of dissatisfied (PPD) and is shown plotted against PMV in Figure 6.3. The figure shows a symmetrical curve with a minimum value of 5% corresponding to the lowest percentage of dissatisfied subjects, i.e. in an ideal environment. As the environment deviated from optimum thermal comfort condition, the dissatisfaction increased. At a PMV or group mean sensation of slightly warm (+1) or slightly cool (-1) the PPD was 27%, and when the PMV reached warm (+2) or cool (-2) the predicted percent of dissatisfied was approximetely 77%.

As Figure 6.5 shows it is impossible to satisfy all persons in a large group sharing a collective climate. Even with a perfect environmental system, which creates absolutely uniform conditions in the occupied zone, one cannot attain a PPD value lower than 5% for similarly clothed people in the same activity. Figure 6.5 may be represented by the expression (Awbi and Savin, 1984):

$$\mathbf{PPD = 5+20.97\,|PMV|^{1.79}} \qquad \textbf{(6.9)}$$

for $|\mathrm{PMV}| \leq 2$.

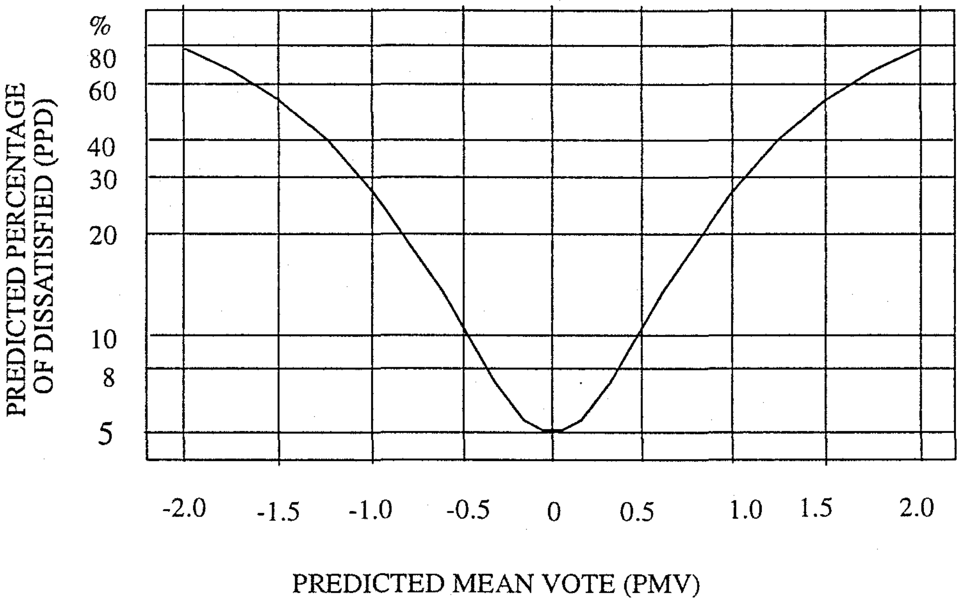


Figure 6.5: Predicted Percentage of Dissatisfied (PPD) as a function of Predicted Mean Vote (PMV) (based on Fanger, 1970).

6.2.7 Simulation Results of Thermal Comfort

In this section, the relative influence of some of the individual variables on the predicted mean vote will be considered. This influence is determined by differentiating Equation 6.6 with respect to the variables.

The following constants were assumed for the coefficients in the PMV equation:

- metabolic activity level M/ADU = 60 W/m^2,
- clothing value (I_{cl}) = 0.5,
- efficiency (η) = 0.0,
- surface ratio (f_{cl}) = 1.1.

Since calculating the mean radiant temperature (t_{mrt}) is very difficult and a complicated process, a conservative assumption of $t_{mrt} = t_a$ is made for simplicity. Since the largest surface area of the window of the building is shaded by the balcony and the evaporative cooling system is installed on the same side (south) of the building, t_{mrt} will be lower than t_a. Therefore, the actual indoor condition will be more comfortable than the calculated values. The model described above is used to simulate the performance of the proposed evaporative cooling system, and the results are discussed below.

At night time when the system does not operate, there is no evaporation through the cooling system. Therefore this is taken into consideration by assuming no additional moisture from 10 PM to 6 AM. Thus, the partial pressure of water vapour in room air (p_a), is calculated from ambient air in PMV equation.

Figure 6.6 shows the calculated PMV values for the second storey of the final case as described in Section 5.7.5. As can be seen from this figure, only 5.4% of the total hours in the hottest week of the year are over +1 PMV. The indoor air temperature, relative humidity and relative air velocity profiles are also shown in Figure 6.6. A comparison between the PMV curve and the other environmental parameters shows that the PMV curve is affected by the temperature variation more than any other factors. As can be seen from this figure, the PMV profile is similar to that of the temperature.

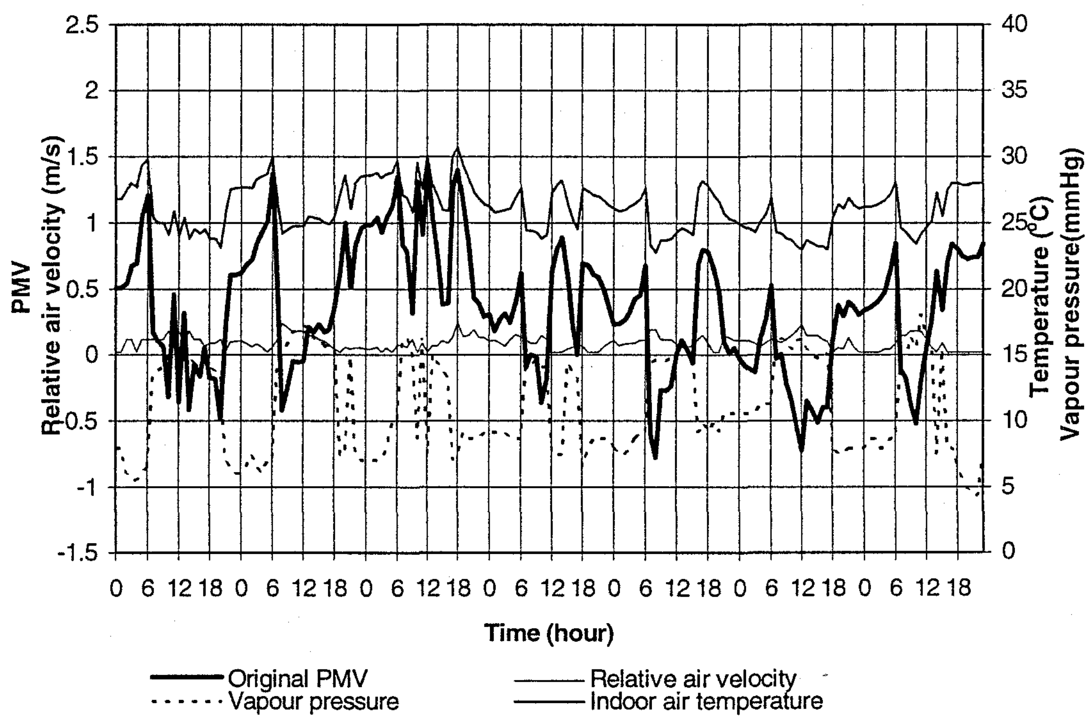


Figure 6.6: Indoor thermal comfort level of PMV.

If the three environmental parameters of air temperature, air velocity and humidity level are assumed to be constant, the PMV profile will be set at a steady line. The following figures demonstrate the behaviour of the PMV profile by assuming two constant environmental parameters.

Figure 6.7 shows the PMV profile by an average relative air velocity of v = 0.09 m/s and an average air temperature of t_a = 26.2^{o}C. As can be seen from the figure, the PMV profile follows exactly the same behaviour as the vapour pressure of p_a.

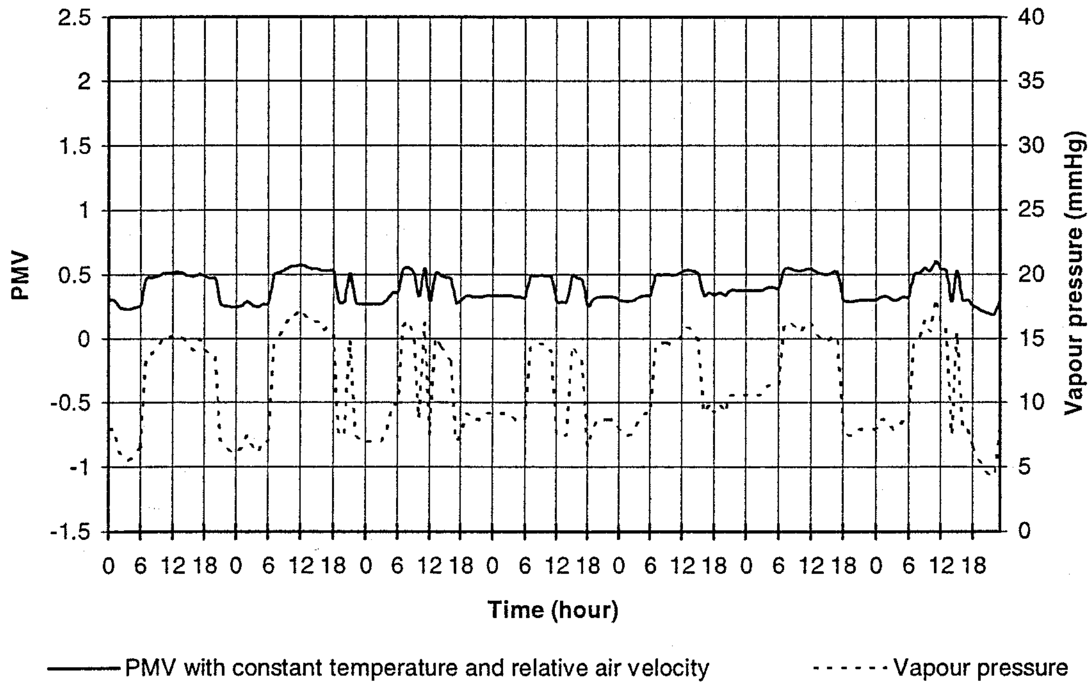


Figure 6.7: The effect of vapour pressure variation on the PMV.

Figure 6.8 demonstrates the same relative roles for the air temperature by fixing the relative air velocity and vapour pressure of p_a = 11 mmHg at the average level.

In order to analyse the effect of air velocity variation on the PMV level, Figure 6.9 demonstrates an environmental condition with a constant temperature of 26.2^{o}C and vapour pressure of p_a = 11 mmHg. As can be seen from the figure, air velocity variation has an opposite effect on the PMV level. At the same time, when the air velocity increases, the PMV value decreases.

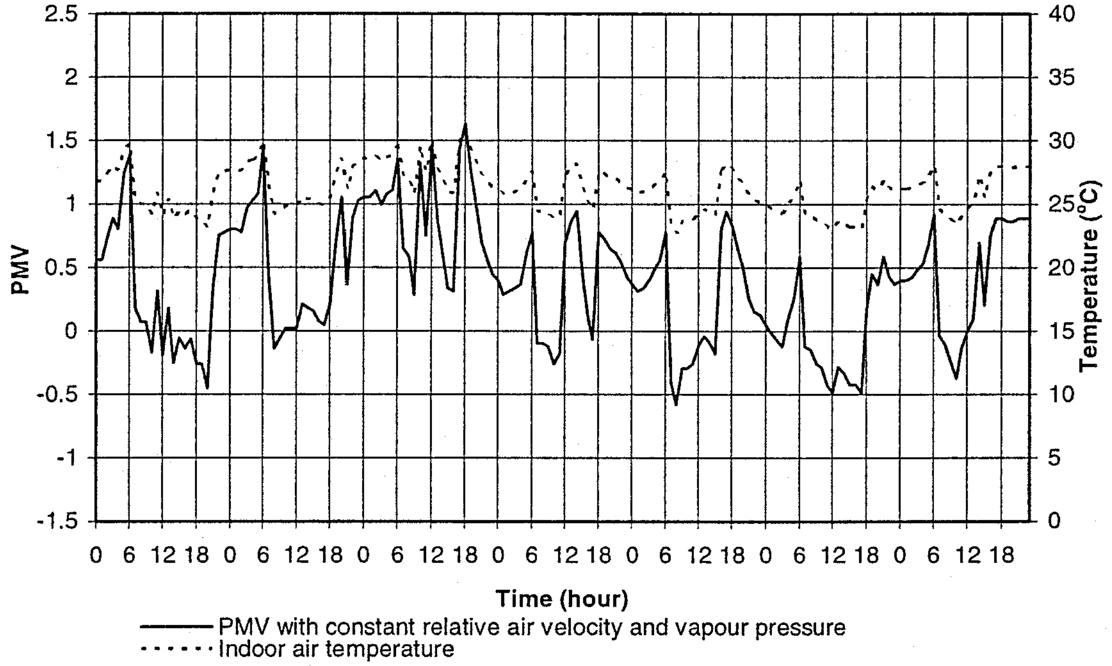


Figure 6.8: The effect of temperature variation on the PMV.

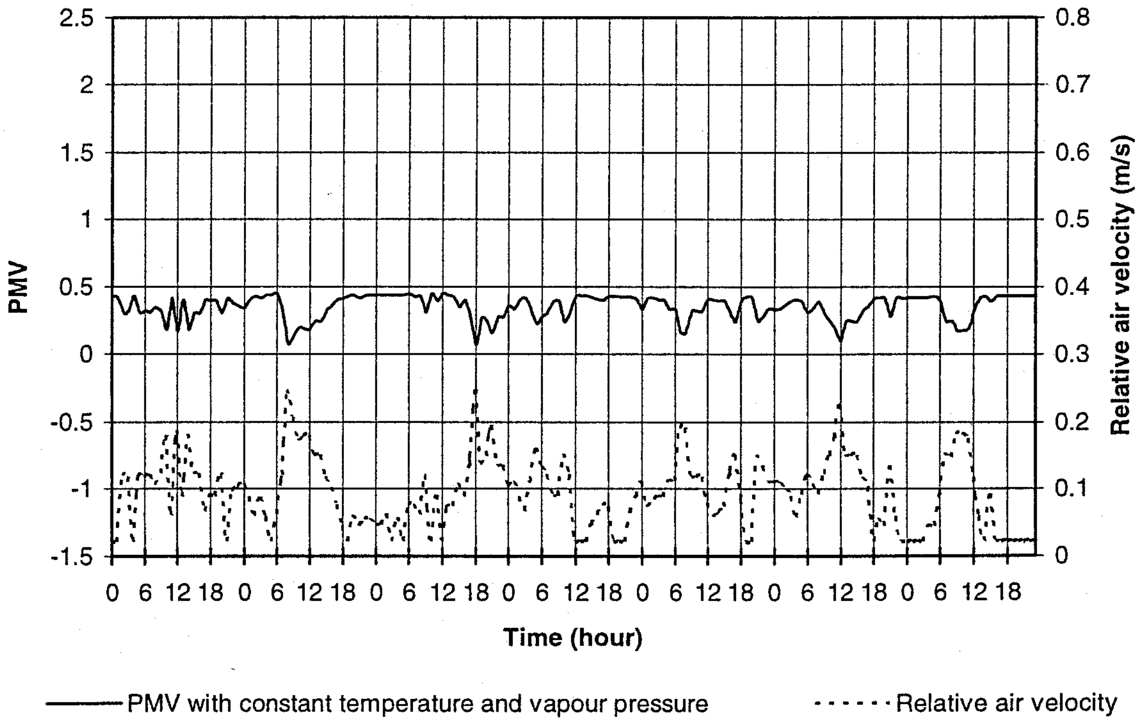


Figure 6.9: The effect of air velocity variation on the PMV.

Since in this study thermal sensation and comfort level are analysed from the passive cooling point of view, environmental parameters can not be set at a constant value. Common knowledge says that with increasing the humidity level in warm conditions discomfort level increases too. This is correct when the indoor air temperature is maintained at a constant level. However, in this study, indoor air humidity level increases where water surface area of the evaporative cooling system increases. By increasing the water surface area of the system, indoor air temperature will decrease as described in Chapter 5. Figure 6.10 illustrates the effect of double water surface area of the system on the PMV level.

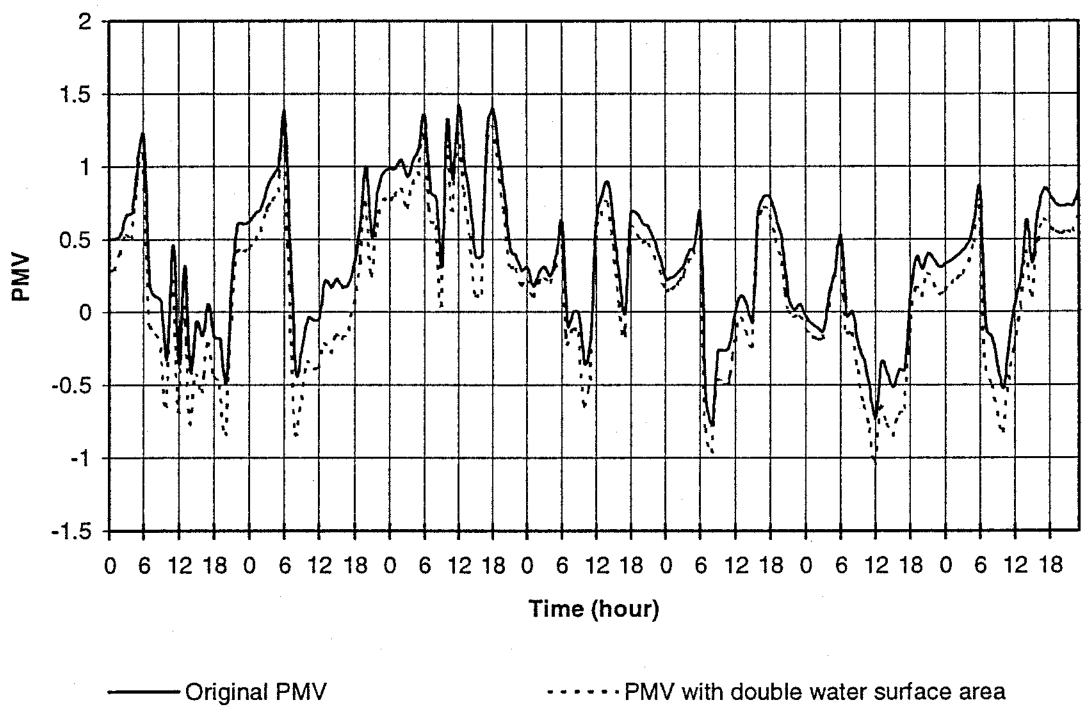


Figure 6.10: The effect of double water surface area of the system on the PMV level.

The average thermal comfort level of PMV slightly improved by double water surface area in the system. However, the improvement does not pay the cost of double water surface area of the system.

The following cases examine the effect of air changes per hour (ach) and air movement on the thermal comfort level. As explained in Chapter 5, increasing in the airflow rate through the building, which increases ach, will cause a rise on average indoor air temperature. Figure 6.11 shows how higher (double) airflow rate (by doubling effective area of the windows) can increase indoor air temperature in ambient temperature above 23°C but cause a reduction on the indoor air temperature in ambient temperature below 23°C. The same effect is true for the PMV profile as can be seen from Figure 6.12.

Evidently, air movement improves comfort at warm temperatures and increases discomfort in the cold. It should be noted that however, there is a difference between the airflow rate through the building or ach and the relative air velocity inside the building. It is possible to have a same value for the airflow rate or air changes per hour (ach) with the different relative air velocities. The airflow rate through the building depends on the effective area of the window (A_e) as well as the wind speed and direction. As it is shown in Equation 3.5, A_e can be maintained at a constant value but a variable opening area of the inlet and outlet.

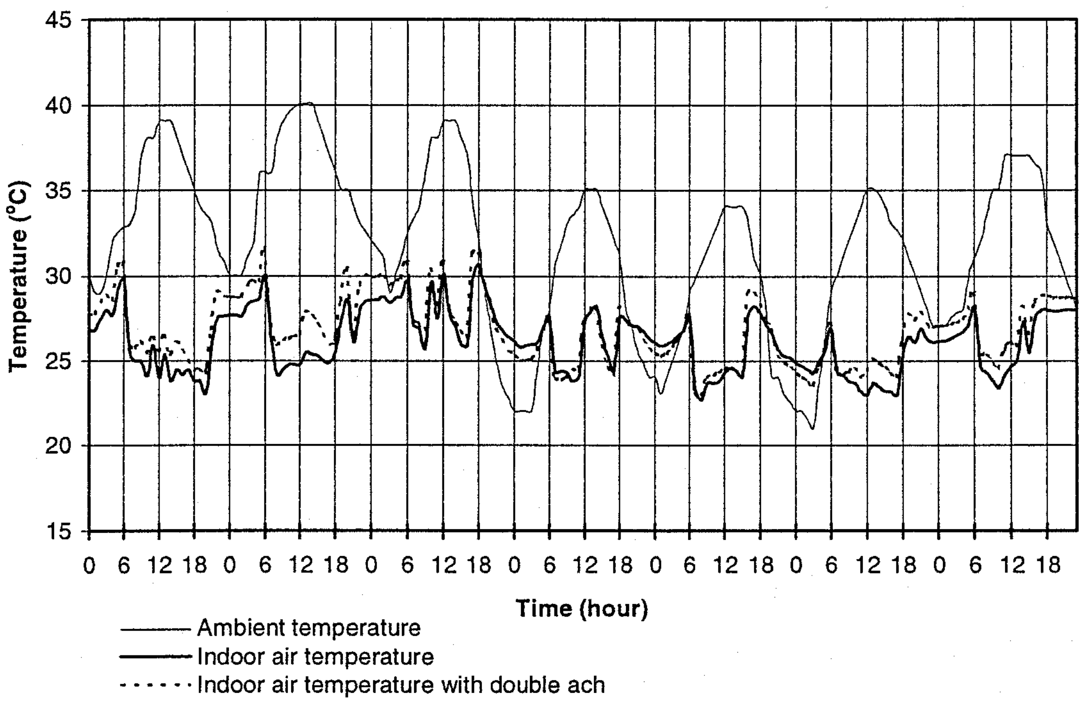


Figure 6.11: The effect of double airflow rate on the indoor air temperature.

Therefore, for the same amount of the airflow rate through the building, the opening area of the inlet and outlet can be changed to obtain the different relative air velocities inside the building. This is used to examine the effect of higher air velocity on the thermal sensation of comfort.

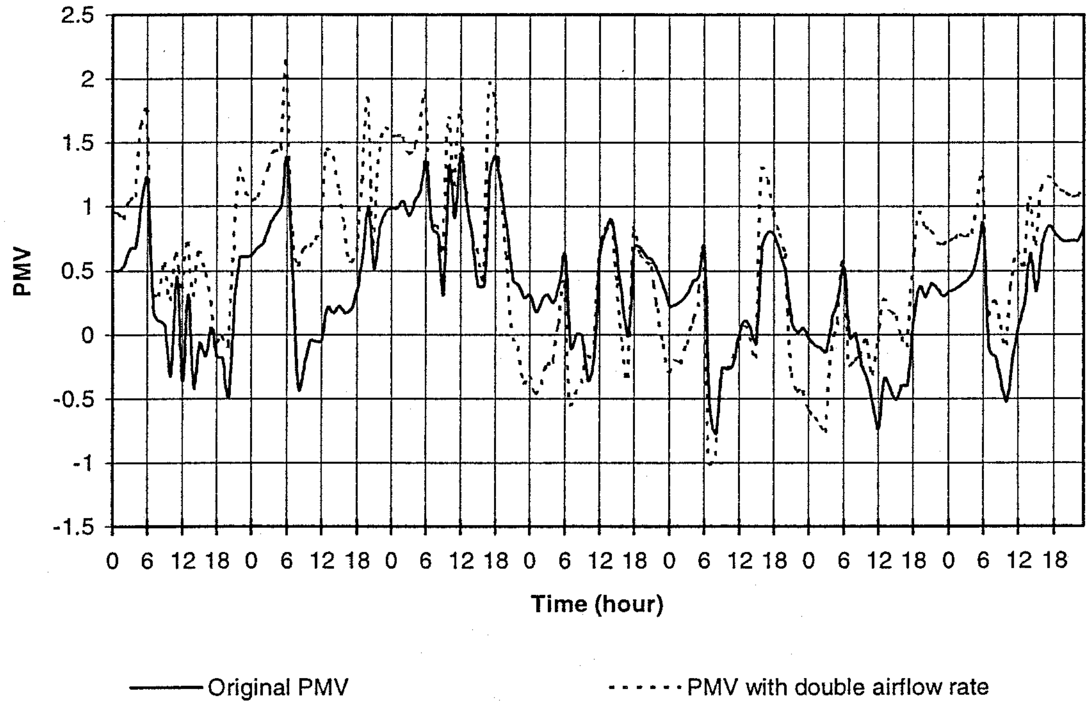


Figure 6.12: The effect of double airflow rate on the PMV level.

Figure 6.13 shows PMV affected by a double relative air velocity while the values of the ach are the same as the original case. The double relative air velocity profile is also shown in Figure 6.13. As can be seen from this figure, higher relative air velocity, can slightly improve comfort in warm temperature but cause a sense of cold in lower air temperatures.

In high velocities forced convection exists and the convective heat transfer coefficient of h_c in Equation 6.8, become a function of the velocity. On the other hand, for lower air velocities, the heat transfer from the outer surface of the clothed body takes place by free convection, so that h_c becomes a function of the temperature difference, $(t_{cl}-t_a)$. Fanger's model proposed to use free convection formula for $v < 0.1$ m/s, but, since temperature difference is also involved, the results of simulations in this study show that, free convection takes place in relative velocities less than about 0.075 m/s. Thus an increase in relative velocity up to this number will not have an influence on the convective heat transfer coefficient of h_c and as a result on PMV.

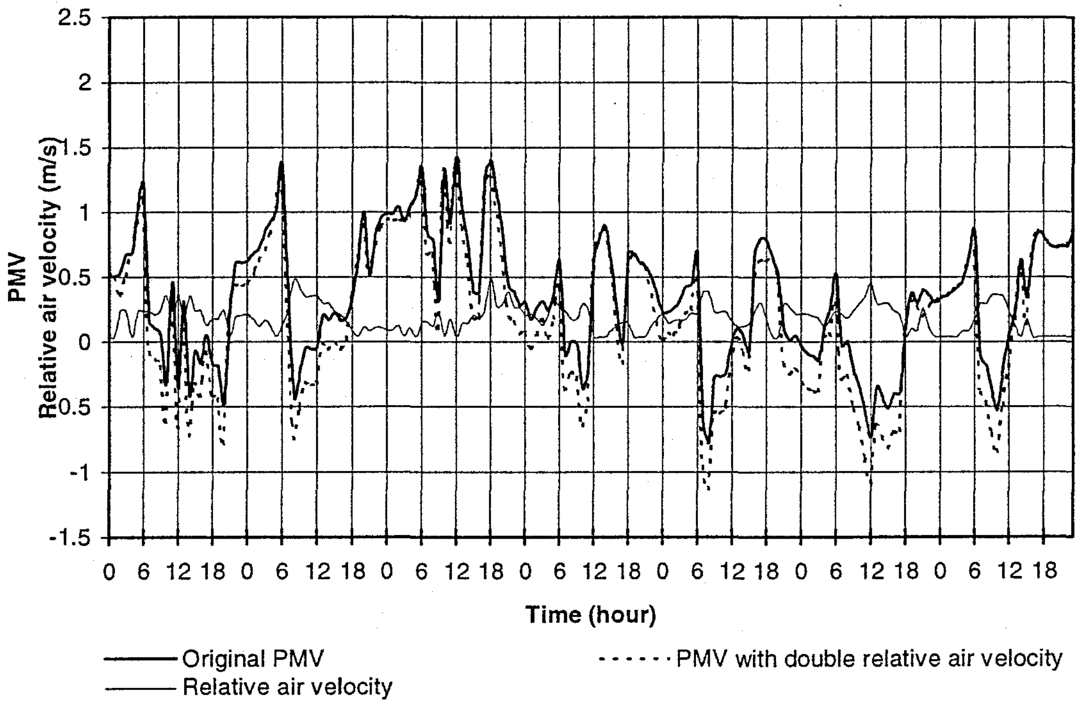


Figure 6.13: The effect of double relative air velocity on the PMV.

The following investigation demonstrates how the two key properties of clothing value and metabolic rate, affect thermal sensation and comfort as judged by the PMV index.

During sleeping hours, the metabolic rate (M/ADU) is assumed to decrease from 60 W/m^2 to 35 W/m^2. Moreover, the clo-value for a person who is lies is greater than the clo-value for a standing person in the same clothing ensemble, and also there might be a cover on the person at night. Therefore, it is assumed that the clo-value increases from 0.5 clo to 1 clo during sleeping hours (10 PM to 6 AM) and fcl from 1.1 to 1.15. Figure 6.14 shows the modified PMV based on the clo value and metabolic rate of sleeping hours. The average PMV value for the last case is +0.23.

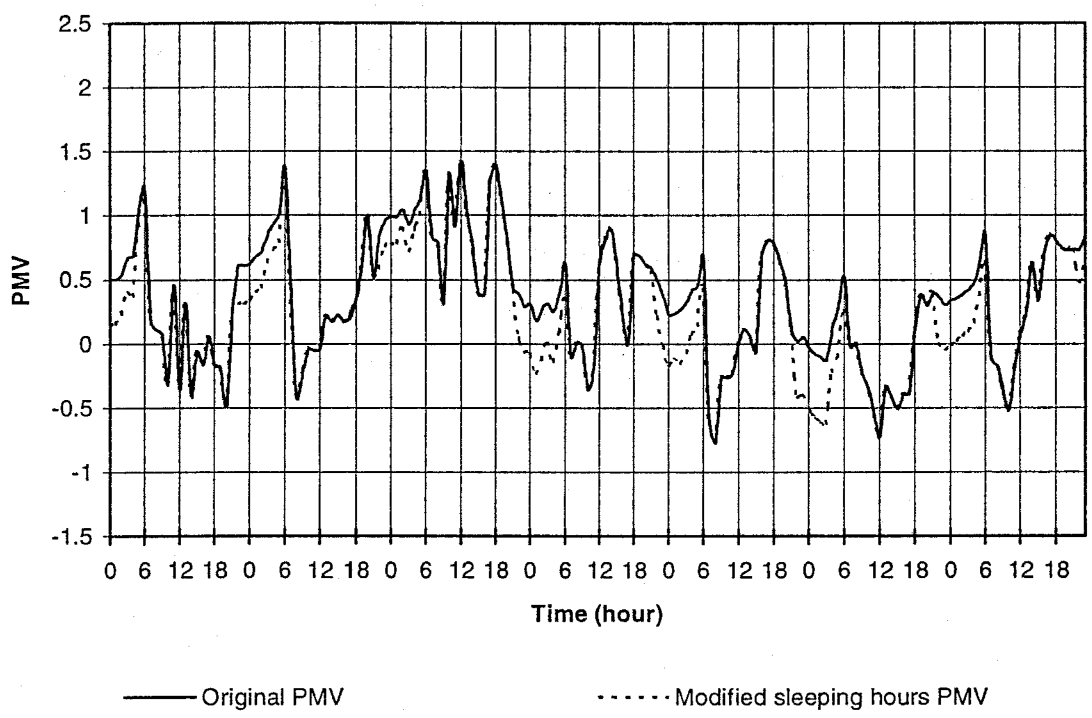


Figure 6.14: The effects of changing metabolic rate and clo value of the sleeping hours on the PMV.

In conclusion, the results of this part of the study indicate that, the proposed passive evaporative cooling system can provide a satisfactory thermal comfort condition for occupants of a residential building in the given climatic situation. This thermal comfort level is achieved with all the previously discussed conservative assumptions and may be further improved without those assumptions. Also it should be noted that the simulation is carried out for the hottest week of the year.

6.3 PART II: CLIMATIC RANGE FOR COMFORT EVAPORATIVE COOLING

The effectiveness of evaporative cooling depends on weather conditions. System design is affected by the prevailing outdoor dry bulb and wet bulb temperatures, as well as the application of the system. Evaporative cooling performs well obviously, only where summer provides adequately dry air. The minimum required weather differs with the type. The earliest authorities believed that direct evaporative coolers required a climate having average noon relative humilities in July (for the northern hemisphere) of 40 per cent or less. Those coolers operate best when wet-bulb levels are 23.9°C or less and local dry-bulb temperatures above 32.2°C warrant cooling equipment (Watt, 1963).

Giles (1955), has suggested a great range of comfort: three weather zones permitting respective "good", "fair" and "fringe" results. The respective number of hours that dry-bulb temperatures exceed 32.2°C and wet-bulb temperatures exceed 21.1, 22.8 and 24.4°C respectively are combined on the basis that hours of suitable operation in each category should exceed 2/3 of total hot hours. The suggested scale are:

a) Good Results:

$$\{(\text{hours over } 32.2^{\circ}\text{C db})-(\text{hours over } 21.1^{\circ}\text{C wb})\}/\,(\text{hours over } 32.2^{\circ}\text{C db})=2/3 \quad \textbf{(6.10)}$$

b) Fair Results:

$$\{(\text{hours over } 32.2^{\circ}\text{C db})-(\text{hours over } 22.8^{\circ}\text{C wb})\}/\,(\text{hours over } 32.2^{\circ}\text{C db})=2/3 \quad \textbf{(6.11)}$$

c) Fringe Results:

$$\{(\text{hours over } 32.2^{\circ}\text{C db})-(\text{hours over } 24.4^{\circ}\text{C wb})\}/\,(\text{hours over } 32.2^{\circ}\text{C db})=2/3 \quad \textbf{(6.12)}$$

Geographical application of this formula would be valuable, although it lacks clear relation to comfort zone analysis. One assumption is clear; fringe results here require minimum wet-bulb depressions of 7.8°C. This clearly includes some "relief cooling".

6.3.1 Comfort Cooling and Relief Cooling

The real need is for analysis based upon the comfort chart, indicating whether given climates allow evaporative cooling to achieve the comfort zone or not. Suitable climates could then be termed the "comfort range", permitting "comfort cooling"; less advantageous climates would constitute the "relief range", allowing "relief cooling" (Watt, 1963).

Ash (1955) suggests that the true measure of evaporative cooling's "comfort range", not its total range, is its ability to maintain conditions below 25.6 ET, near the comfort zone's upper limit. This is the point where perspiration begins in most lightly clothed persons, indicating that body convective and radiant cooling are no longer adequate. He acknowledges that 25.6 ET is achieved by different means in different climates In dry climates it results mostly from adiabatic saturation based on large wet-bulb depressions.

Accordingly 25.6 ET can be achieved by either *cooling* effectiveness or *ventilating* effectiveness, depending upon the climate. The contrasting conditions below achieve almost identical comfort:

Table 6.3: The conditions which achieve almost identical comfort (Watt, 1963).

	Dry Climate	Average Climate
Outdoor dry-bulb	44.4°C	33.9 °C
Outdoor wet-bulb	22.2°C	25.6 °C
Wet-bulb depression	22.2°C	8.3 °C
Washed air enters at	26.7°C	27.2 °C
Room air temperature	29.4°C	28.3 °C
Washed air exhaust	32.2°C	29.4 °C
Room air velocity	0.2 m/s	1 m/s
Room ET:	25.6	25.6

These nearly opposite conditions create identical sensations of indoor comfort low air movement in the first case being compensated by 50 percent indoor rh, and high velocity in the latter being offset by 80 percent rh. This last humidity may damage some materials but is acceptable for human comfort.

This illustrates that evaporative coolers can improve indoor comfort in almost all climates, provided that high indoor air velocities and relative humidities are acceptable.

6.3.2 Evaporative Cooling Comfort Chart

Watt (1963) proposed to relate local climatic conditions positively to the comfort zone. Evaporative cooling will thus be recommended only where it usually achieves conditions with that area's comfort zone. In this recommended geographical range it will deliver "comfort cooling" and in a limited zone outside it, "relief cooling".

Since no natural definition exists, the following limits for the "comfort range" are proposed. Any locality whose climate allows average evaporative coolers to meet these requirements is thus considered the "comfort range" and its cooling is thus classified as "comfort cooling". The minimum requirements are:

1. that direct evaporative coolers have average saturating efficiencies of 70 percent or more, and the cooled air enters rooms without prior heat gain.

2. that cooled air induces a maximum average indoor air velocity of 1 m/s.

3. that cooled air gains at least 3.3°C indoors before its discharge.

4. that cooled spaces average 1.7°C above cooler discharge temperature and have average humidities of 70 percent rh or below.

5. that cooled spaces average at least $4.4^{o}C$ below outdoor dry-bulb temperature to counteract entering radiant heat and provide a differential below outside.

6. that cooled spaces average an effective temperature no higher than upper limit of the ASHRAE summer comfort zone adjusted for local latitude. (The final requirement varies with latitude)

For example, in Pittsburgh's 41°N latitude, the comfort chart's origin (Figure 6.15), the summer comfort zone ranges between 17.8 ET and 26.1 ET, with its centre at 21.7 ET. Thus, in this latitude average direct evaporative coolers must satisfy these requirements and achieve an indoor effective temperature of 26.1 ET or below, to provide "comfort cooling".

With the necessary final indoor conditions known for each region, Figure 6.15 helps determine the maximum permissible local outdoor wet-bulb temperature and the minimum average outdoor wet-bulb depression required for such performance. The former becomes the maximum permissible design wet-bulb temperature for the locality, and the latter is converted by a simple (and not entirely satisfactory) ratio into the required minimum design wet bulb depression.

6.3.2.1 Design Temperatures

Design temperatures, published to aid designers of air conditioning systems, consist of dry-bulb and wet-bulb temperatures which form a quick and easy index to local climatic conditions since they represent respectively the hottest and most humid weather commonly encountered. In general, they represent temperatures usually exceeded in only 1 to 5 percent of total hours in average summers, but as matters of local estimate and custom, not of calculation from weather records, they have uncertain accuracy. The ASHRAE Guide's compendium of design temperatures is perhaps the best. Design wet-bulb depression is simply design dry-bulb temperature less the design wet-bulb value.

6.3.2.2 Calculating Technique

Figure 6.15 is an adapted thermometric chart showing the interrelation of summer dry-bulb, wet-bulb and air velocity conditions in creating effective temperatures. The ASHRAE old comfort zone for 41°N latitude has been superimposed upon it, its upper limit on 26.1 ET. This zone moves bodily up the curve 0.56 effective temperature (one line) for each 5° reduction in latitude. The theoretical percentage of persons feeling comfortable at any temperature is indicated.

The chart is arranged as a nomograph in order to solve comfort problems; straight lines drawn between related dry-bulb and wet-bulb values indicate the resulting effective temperatures where they cross velocity lines.

To solve evaporative cooling's outdoor problems, Watt (1963) has added shorter vertical scales indicating direct evaporative cooler washed-air temperatures at saturating efficiencies of 70 to 90 percent. Thus, lines connecting appropriate outdoor values on the dry- and wet-bulb scales indicate on the new ones the temperature of cooled-air entering conditioned rammers.

To calculate the outdoor climatic conditions necessary to achieve comfort cooling as defined, the following method is proposed by Watt (1963):

1. Select outdoor dry-bulb temperature (ODB) and outdoor wet-bulb temperature (WB) from local air conditioning design data.

2. Draw outdoor condition line from ODB to WB.

3. Select cooler saturating efficiency column. Where it crosses outdoor condition line, read the washed air room-entering temperature (WAE).

4. Select washed air indoor temperature gain, usually 2.8-5.6°C (5-10 F), and locate washed air room-exhaust temperature (WAX) on left that far above WAE, and draw connecting line.

5. Locate washed air indoor mean temperature (WAIA) midway between WAE and WAX.

6. Draw horizontal line left from WAIA to find the average indoor dry bulb temperature (IDB).

7. Select estimated indoor wet-bulb temperature (IWB), usually 1.1°C (2 F) above WB.

8. Draw indoor condition line from IDB to IWB.

9. Select an estimated room average air velocity line.

10. Read indoor comfort level in Effective Temperature degrees, where indoor condition line crosses room air velocity line. Compare with comfort zone adjusted for local latitude.

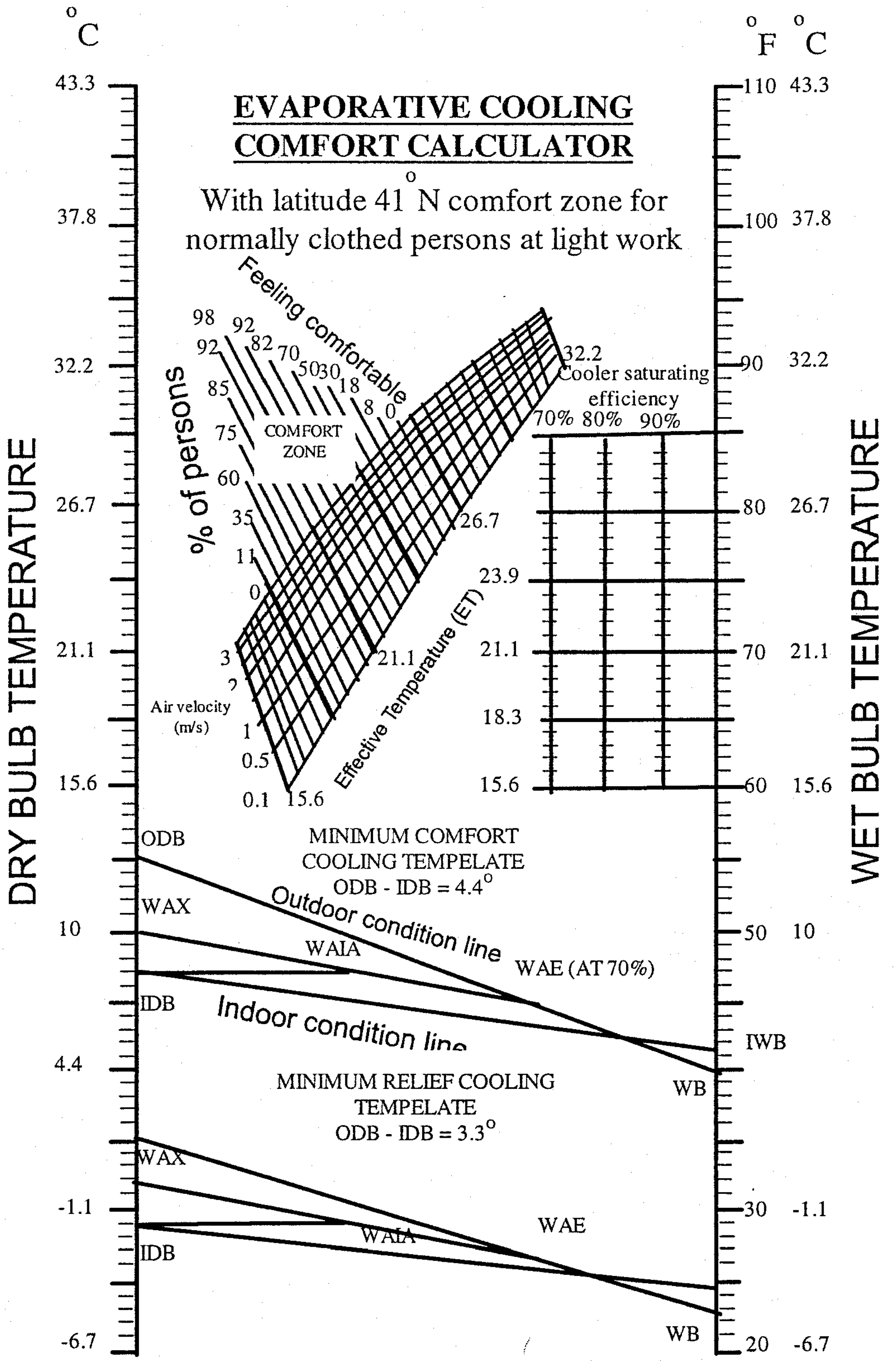


Figure 6.15: Comfort evaporative cooling chart (based on Watt, 1963).

6.3.3 Comfort Zone for Tehran and Various Locations

In order to calculate the summer comfort zone in Tehran with latitude 35.41^{o}N, the comfort zone in this latitude, has been bodily moved up the curve 0.56 effective temperature (one line). Thus, the summer comfort zone for Tehran, ranges between 18.3 ET and 26.7 ET, with its centre at 22.2 ET.

Design temperatures of dry-bulb and wet-bulb represent respectively the average hottest and most humid weather of the simulation period. Also, air velocity and efficiency of the system are assumed at a conservative averaged minimum level at the hottest time.

The assumptions of calculating comfort zone are:

- Outdoor design dry-bulb temperature = 37.1^{o}C
- Outdoor design wet-bulb temperature = 20.7 ^{o}C
- Room air velocity = 0.1 m/s
- Cooling saturating efficiency = 70%

Figure 6.16 shows the thermal comfort level achieved by the evaporative cooling system and above assumptions in the given climatic condition of Tehran. As can be seen from this figure, the Effective Temperature is 23.9 ET and 70% of people feel comfortable in this climatic region.

In order to determine the geographical performance of evaporative cooling system, Watt (1963) presented the minimum climatic conditions allowing average direct evaporative coolers in each region to achieve their appropriate comfort zone. By definition of design temperatures, the localities meeting these requirements probably will enjoy comfort zone cooling at least 80 to 90 percent of all hot hours.

In all cases, comfort cooling requires a minimum design depression of 12.2^{o}C and a maximum wet-bulb temperature based on latitude as presented in Table 6.4. Comfort cooling by direct evaporative cooling in the 35. 41^{o}N (Tehran) latitude requires design wet-bulb temperature not over 25.4°C. Evaporative cooling outside this limit cannot achieve the required 26.7 ET, and is recommended for relief cooling only.

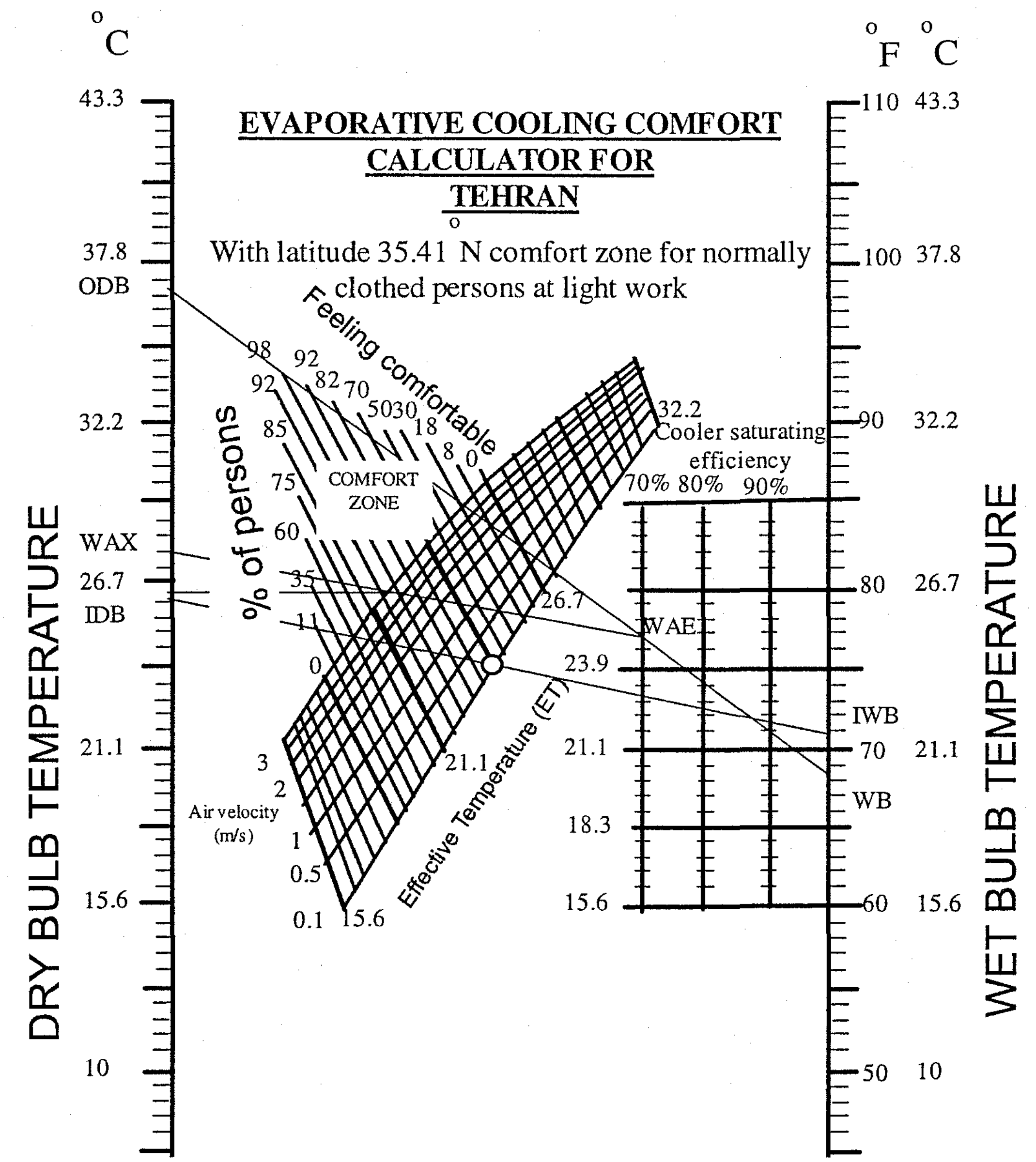


Figure 6.16: Comfort evaporative cooling chart for Tehran.

Similarly, since each change of 5° of North latitude shifts the comfort zone 0.56 ET, evaporative coolers in about 43-47°N, must achieve 25.6 ET, requiring design wet-bulb temperatures of 24.2°C or below. Correspondingly, comfort cooling in the latitudes about 38-42°N, must make 26.1 ET or below for which 24.8°C is the maximum permissible design wet-bulb temperature. In latitudes about 33-37°N, comfort cooling must achieve 26.7 ET or better, requiring an 25.4°C maximum design wet-bulb temperature. At approximate latitudes of 28-32 and 26-27°N, should respectively achieve 27.2 and 27.8 ET, and wet bulb temperatures not over than 25.9°C and 26.5°C.

Table 6.4: Recommended evaporative cooling climate (Watt, 1963).

	For comfort cooling			For relief cooling		
Approximate latitude	Maximum indoor ET (°C)	Minimum outdoor design WB depression	Maximum outdoor design WB (°C)	Maximum indoor ET (°C)	Minimum outdoor design WB depression	Maximum outdoor design WB (°C)
47-43°N	25.6	12.2	24.2	26.7	9.4	26.1
42-38	26.1	12.2	24.8	27.2	9.4	26.7
37-33	26.7	12.2	25.4	27.8	9.4	27.2
32-28	27.2	12.2	25.9	28.3	9.4	27.8
27-26	27.8	12.2	26.5	28.9	9.4	28.3

6.4 CONCLUSION

In this chapter thermal comfort issues were studied from two points of view. Firstly, estimating thermal comfort sensation levels achieved inside the building by the proposed system, and secondly, suitable climatic ranges for the system.

By ASRAE (1992) definition, thermal comfort was defined as "that condition of mind which expresses satisfaction with the thermal environment". Specific related terms of Effective Temperature (ET) and Operative Temperature (t_O) were explained as two scales of evaluating thermal comfort zone. Also, the environmental variables and certain special factors affecting thermal comfort were determined.

A summer thermal comfort temperature of 27.3°C was defined for Tehran based on the equations proposed by Humphreys (1981), Auliciems (1986) and Nicol (1995). Also, a summer comfort zone was defined on the psychrometric chart based on the Szokolay's (1987) procedure.

A predicted mean vote (PMV) equation proposed by Fanger (1970) was adopted in order to calculate indoor thermal comfort levels achieved by the proposed evaporative cooling system.

The results of the calculations indicate that, the PMV value for the final case in Chapter 5 was only 5.4% over the satisfaction level of +1. The results of examinations showed that, the PMV profile was affected by the temperature variation more than any other environmental factors. In the study of the effect of humidity ratio on the comfort level, it was found that, by doubling the water surface area of the system which increases indoor humidity level, PMV level slightly improved. However, this was not a considerable improvement in comfort level.

Investigation showed that airflow rate and relative air velocity have different effects on the PMV values. Increasing the airflow rate through the building, increases the number of air changes per hour (ach), and increases the sense of heat in the day time. On the other hand, increasing relative air velocity slightly improves the PMV value. Changing metabolic rate and clo value during sleeping hours showed a reduction in PMV level at night time.

In part two, climatic range of effective evaporative cooling was studied. It was stated that evaporative coolers operate best in climatic conditions with wet-bulb temperature of 23.9°C or less and dry-bulb temperature above 32.2°C. Suitable climates were termed as "comfort range", permitting "comfort cooling" and less advantageous climates would constitute the "relief range", allowing "relief cooling". An evaporative cooling comfort chart which proposed by Watt (1963) was adopted in order to determine the comfort cooling range in various climatic conditions. The evaporative cooling chart for Tehran showed that, by using this cooling system, 70% of people would feel comfortable in this climatic region. Also a table was presented for recommended evaporative cooling climates for various latitudes and design temperatures.

CHAPTER 7

CONCLUSION

The study has investigated a possible method of a passive evaporative cooling system for designing more energy efficient residential buildings. It was found at a theoretical level that, the use of water passing over vertical guides in the form of a screen which makes use of natural ventilation at the building facade will provide summer comfort in dry climates. This research has provided an insight into the potential application of a passive design concept which can be used to achieve a more satisfactory living environment. The advantages of such a design approach lies in the fact that comfort conditions can be achieved simply and economically by utilising of the natural processes and energies, through climate responsive building design.

The study has demonstrated that the proposed passive cooling system is capable of providing a satisfactory level of comfort for the occupants in hot summers of Tehran. This system can be used in a wide climatic range as is clarified in previous chapter, to create a thermally comfort cooling.

Wagga Wagga, a semi-arid location in New South Wales, an eastern state of Australia is an example within this climatic range. In another investigation (Ghiabaklou and Ballinger, 1996) it was demonstrated that the suggested cooling system significantly lowers indoor air temperature of a building as present study in this region.

As the cooling system works only with natural ventilation, the theoretical aspects of calculating natural ventilation rate, strategies and guide-lines where investigated in this study. It was found that, the airflow rate through the building can be significantly affected by shielding and terrain factors rather than the building shape. The results of investigation indicated that, the maximum cooling effect on the building was achieved at the optimum airflow rate. A design parameter of 0.03 was defined as an optimum volumetric airflow rate per square meter of the surface area of the water in the evaporative cooling system which is independent of the volume of the building.

Simulation results of the effect of different filaments arrangement on the efficiency of the evaporative cooling system showed that, a system with smaller water line diameter and also smaller transverse pitches had a higher efficiency than the others. Even when the number of rows were taken into consideration, a system with smaller pitches and the less number of rows was more efficient than the larger pitches and more rows. However, in comparison between the systems with various numbers of rows and different width, a system with rows of smaller width was more efficient than a system with wider length and fewer rows. The results of studying the effect of various parameters on the indoor air temperature showed that design changes can significantly improve the thermal performance of a building.

In order to calculate the indoor thermal comfort level achieved by the proposed evaporative cooling system, various thermal comfort equations were used. On the basis of the research question of "the extent to which the passive evaporative cooling system as proposed, is capable of providing thermal comfort in summer for occupants of the residential buildings in arid and semi-arid regions", the results of calculations indicated that, indoor air temperature of the final case was almost in the summer comfort zone and the PMV value was only 5.4% over the satisfaction level of +1.

The results of examinations showed that, the PMV profile was affected by the temperature variation more than any other environmental factors. In the study of the effect of humidity ratio on the comfort level, it was found that, by doubling the water surface area of the system which increases indoor humidity level, PMV level slightly improved. However, this was not a considerable improvement in comfort level.

Investigation showed that airflow rate and relative air velocity have different effects on the PMV values. Increasing the airflow rate through the building, which increases the number of air changes per hour, which increases the sense of overheating in the day time. On the other hand, increasing relative air velocity slightly improves the PMV value. Changing metabolic rate and clo value during sleeping hours showed a reduction in PMV level at night time.

It was found that evaporative coolers operate best in climatic conditions with wet-bulb temperature of 23.9°C or less and dry-bulb temperature above 32.2°C. The evaporative cooling chart for Tehran showed that, by using this cooling system, 70% of people feel comfortable in this climatic region. Also a table was presented for recommended evaporative cooling climates for various latitudes and design temperatures.

The system proposed and analysed in this study provides a potentially inexpensive, energy efficient, environmentally safe and attractive cooling method. Whilst it has not been possible to construct such a system due to financial constraints, it would seem that on a theoretical basis the system is feasible and could have wide application in medium density housing. It can be installed independently on the facade of the building, say across a balcony, to provide a passive cooling method for multi story residential buildings.

Equations and charts have been presented, which enable the calculation of the saturating efficiency of the system due to natural ventilation through openings. Further work is still required however to incorporate the calculation routines into suitable simulation models that would allow the building designer to make full use of this cooling device in building designs. The actual system performance, cost estimation and psychological aspects of the system on the occupants are other aspects of this proposal which need further research.

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APPENDIX

APPENDIX A.1 COMPUTER PROGRAM CHEETAH

The program calculates hourly temperatures, and heating and cooling energy requirements, using hourly recorded climatic data for periods ranging from one day to a full year. A response-factor method is the basis of the calculation. First, a calculation is made of the frequency response (i.e., the response to sinusoidal temperatures and heat flows of any given frequency) of each building section (walls, floors and so on). While still in the frequency domain, a heat balance is struck for the whole building, enabling the frequency response of each zone to the temperatures and heat flows acting on the building (called the drivers) to be calculated. Linear system theory is then used to convert the zone frequency response to a transient response-that is, the response to a unit inclined step, square, or triangular pulse-resulting in so-called total-zone response factors. These response factors are calculated only once, at the beginning of the calculation. Then, at each hour, the drivers are calculated from the climatic data and other information, and are approximated by an appropriate type of simple pulse. The total response of the each zone to all the drivers is then calculated by adding the response to each driver in turn, taking into account the accumulated responses from previous hours (Delsante, 1987).

The ventilation model used in CHEETAH has been replaced with calculated data and the evaporative cooler effectiveness or efficiency is also red in each hour from the calculated values in the weather data. In fact, two series of data which are numbers of air changes per hour (ach) and evaporative cooling efficiency of the system (E_s), should be provided to input CHEETAH to obtain indoor environmental temperature.

APPENDIX A.2 CLIMATIC DATA

Climatic data from DATSAV2 was obtained from the National Climatic Data Centre (NCDC) in the U. S.A. The DATSAV2 surface database is composed of worldwide surface weather observations collected and stored from sources such as the Automated Weather Network (AWN) and the Global Telecommunications System (GTS) collected since 1973. It includes such codes as synoptic, airways, METAR (Meteorological Aviation Routine Weather Report), AERO (Aviation Routine Weather Report), SMARS (Supplementary Marine Reporting Station), as well as observations from automatic weather stations. The DATSAV2 surface database consists of blocked variable length records in 8-bit ASCII character format. The maximum length of data recorded is 1,000 characters and the maximum block length is 8,192 characters (USAFETAC Handbook, 1986).

APPENDIX A.3 CLIMATIC CONDITIONS OF TEHRAN

The outdoor climate of Tehran, a semi-arid location in Iran with latitude 35.41°N has been chosen to show the influence of various design parameters. The climatic conditions of the four seasons can be summarised as follows:

- Spring (April - Jun): Temperate to warm days, cool nights with western wind.
- Summer (July - September): Hot and dry days, temperate nights with south-east wind.
- Autumn (October - December): Temperate to cool sunny days, cold nights with north and north-west winds.
- Winter (January - March): Cold snowy days with very cold nights and western winds.

Figure A.1 shows the monthly maximum, average and minimum outdoor temperatures. In the year 1977 the highest average temperature (July) was 36.8 °C and the lowest (January) -1.1°C.

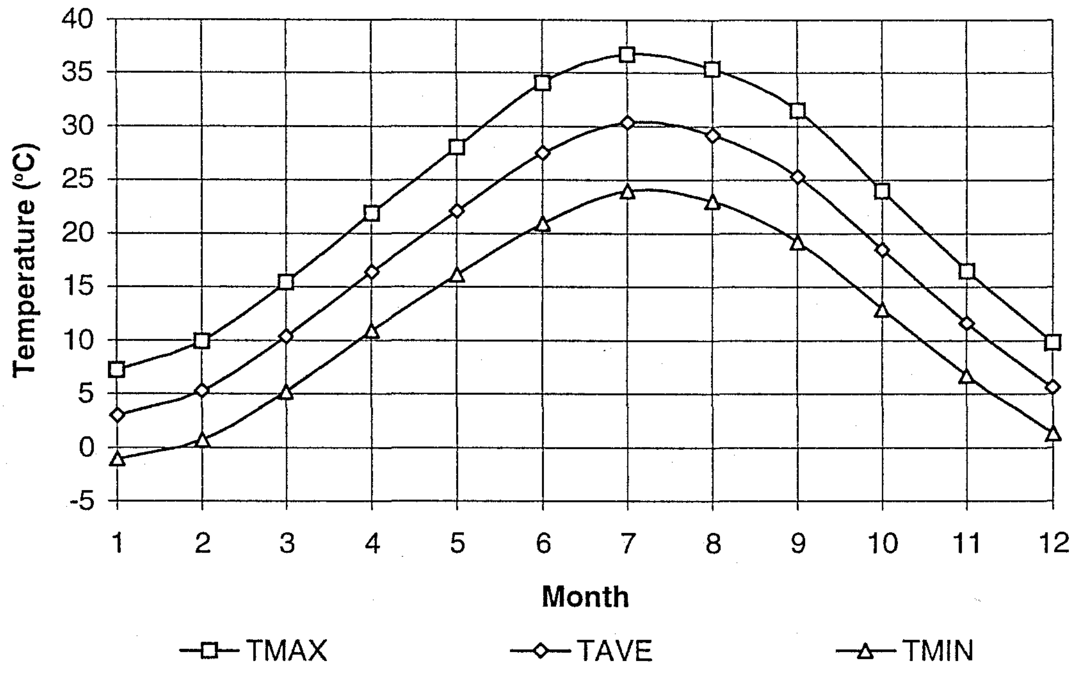

Figure A.1: Monthly outdoor air temperature.

The monthly maximum, average and minimum relative humidity are shown in Figure A.2.

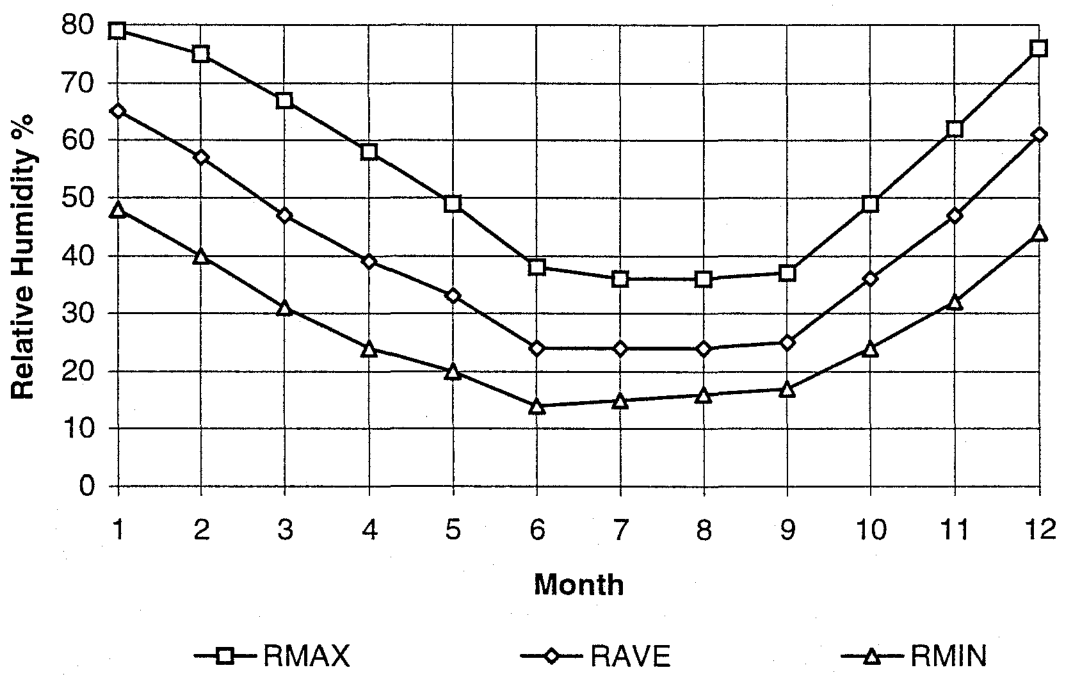

Figure A.2: Monthly outdoor relative humidity.

The monthly average of solar radiation on a horizontal surface (per day) can be seen in Figure A.3.

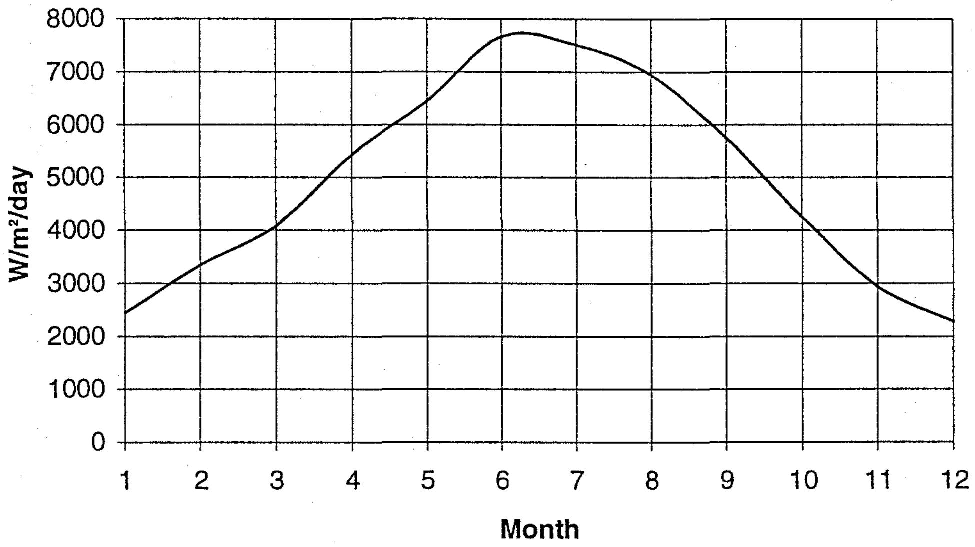

Figure A.3: Monthly average solar radiation on horizontal surface.

Figure A.4 shows the monthly average wind speed and prevailing wind direction.

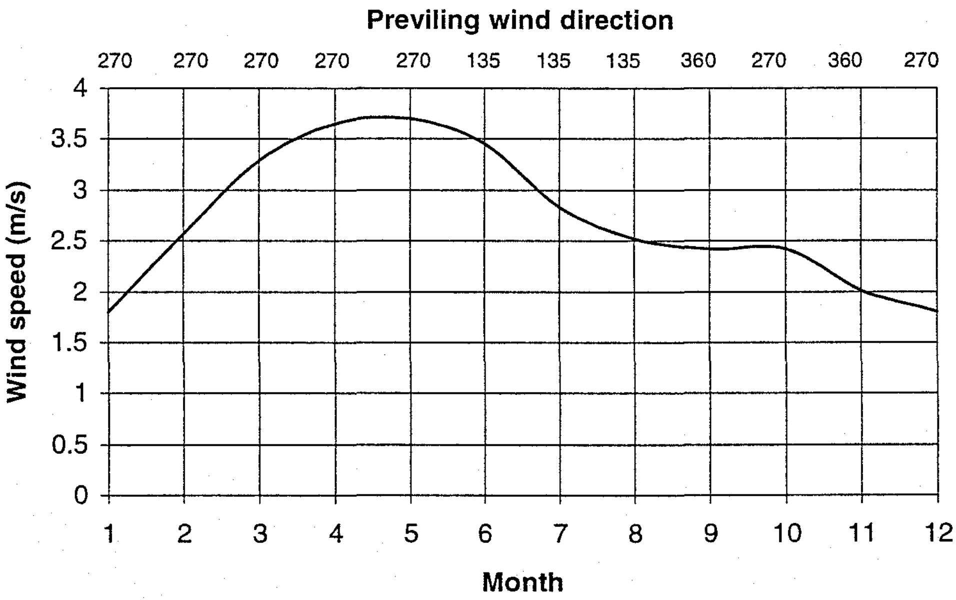

Figure A.4: Monthly average wind speed and prevailing wind direction.

The climate for this particular location is typical for most semi-arid regions. The results presented in this study should therefore be representative for these regions.

APPENDIX A.4 THERMOPHYSICAL PROPERTIES OF AIR AND SATURATED WATER

Table A.1: Thermophysical properties of air at atmospheric pressure (Incropera and Witt, 1990).

T (K)	ρ (kg/m^3)	c_p (kJ/kg · K)	$\mu \cdot 10^7$ (N · s/m^2)	$\nu \cdot 10^6$ (m^2/s)	$k \cdot 10^3$ (W/m · K)	$\alpha \cdot 10^6$ (m^2/s)	Pr
Air							
100	3.5562	1.032	71.1	2.00	9.34	2.54	0.786
150	2.3364	1.012	103.4	4.426	13.8	5.84	0.758
200	1.7458	1.007	132.5	7.590	18.1	10.3	0.737
250	1.3947	1.006	159.6	11.44	22.3	15.9	0.720
300	1.1614	1.007	184.6	15.89	26.3	22.5	0.707
350	0.9950	1.009	208.2	20.92	30.0	29.9	0.700
400	0.8711	1.014	230.1	26.41	33.8	38.3	0.690
450	0.7740	1.021	250.7	32.39	37.3	47.2	0.686
500	0.6964	1.030	270.1	38.79	40.7	56.7	0.684
550	0.6329	1.040	288.4	45.57	43.9	66.7	0.683
600	0.5804	1.051	305.8	52.69	46.9	76.9	0.685
650	0.5356	1.063	322.5	60.21	49.7	87.3	0.690
700	0.4975	1.075	338.8	68.10	52.4	98.0	0.695
750	0.4643	1.087	354.6	76.37	54.9	109	0.702
800	0.4354	1.099	369.8	84.93	57.3	120	0.709
850	0.4097	1.110	384.3	93.80	59.6	131	0.716
900	0.3868	1.121	398.1	102.9	62.0	143	0.720
950	0.3666	1.131	411.3	112.2	64.3	155	0.723

Table A.2: Termophisycal properties of saturated water (Incropera and Witt, 1990).

TEMPERATURE, T (K)	PRESSURE, P (bars)^b	SPECIFIC VOLUME (m^3/kg) $v_f \cdot 10^3$	v_g	HEAT OF VAPORIZATION, h_{fg} (kJ/kg)	SPECIFIC HEAT (kJ/kg · K) $c_{p,f}$	$c_{p,g}$	VISCOSITY (N · s/m^2) $\mu_f \cdot 10^6$	$\mu_g \cdot 10^6$	THERMAL CONDUCTIVITY (W/m · K) $k_f \cdot 10^3$	$k_g \cdot 10^3$	PRANDTL NUMBER Pr_f	Pr_g	SURFACE TENSION, $\sigma_f \cdot 10^3$ (N/m)	EXPANSION COEFFICIENT, $\beta_f \cdot 10^6$ (K^{-1})	TEMPERATURE, T (K)
273.15	0.00611	1.000	206.3	2502	4.217	1.854	1750	8.02	569	18.2	12.99	0.815	75.5	-68.05	273.15
275	0.00697	1.000	181.7	2497	4.211	1.855	1652	8.09	574	18.3	12.22	0.817	75.3	-32.74	275
280	0.00990	1.000	130.4	2485	4.198	1.858	1422	8.29	582	18.6	10.26	0.825	74.8	46.04	280
285	0.01387	1.000	99.4	2473	4.189	1.861	1225	8.49	590	18.9	8.81	0.833	74.3	114.1	285
290	0.01917	1.001	69.7	2461	4.184	1.864	1080	8.69	598	19.3	7.56	0.841	73.7	174.0	290
295	0.02617	1.002	51.94	2449	4.181	1.868	959	8.89	606	19.5	6.62	0.849	72.7	227.5	295
300	0.03531	1.003	39.13	2438	4.179	1.872	855	9.09	613	19.6	5.83	0.857	71.7	276.1	300
305	0.04712	1.005	29.74	2426	4.178	1.877	769	9.29	620	20.1	5.20	0.865	70.9	320.6	305
310	0.06221	1.007	22.93	2414	4.178	1.882	695	9.49	628	20.4	4.62	0.873	70.0	361.9	310
315	0.08132	1.009	17.82	2402	4.179	1.888	631	9.69	634	20.7	4.16	0.883	69.2	400.4	315
320	0.1053	1.011	13.98	2390	4.180	1.895	577	9.89	640	21.0	3.77	0.894	68.3	436.7	320
325	0.1351	1.013	11.06	2378	4.182	1.903	528	10.09	645	21.3	3.42	0.901	67.5	471.2	325
330	0.1719	1.016	8.82	2366	4.184	1.911	489	10.29	650	21.7	3.15	0.908	66.6	504.0	330
335	0.2167	1.018	7.09	2354	4.186	1.920	453	10.49	656	22.0	2.88	0.916	65.8	535.5	335
340	0.2713	1.021	5.74	2342	4.188	1.930	420	10.69	660	22.3	2.66	0.925	64.9	566.0	340
345	0.3372	1.024	4.683	2329	4.191	1.941	389	10.89	668	22.6	2.45	0.933	64.1	595.4	345
350	0.4163	1.027	3.846	2317	4.195	1.954	365	11.09	668	23.0	2.29	0.942	63.2	624.2	350
355	0.5100	1.030	3.180	2304	4.199	1.968	343	11.29	671	23.3	2.14	0.951	62.3	652.3	355
360	0.6209	1.034	2.645	2291	4.203	1.983	324	11.49	674	23.7	2.02	0.960	61.4	697.9	360
365	0.7514	1.038	2.212	2278	4.209	1.999	306	11.69	677	24.1	1.91	0.969	60.5	707.1	365
370	0.9040	1.041	1.861	2265	4.214	2.017	289	11.89	679	24.5	1.80	0.978	59.5	728.7	370
373.15	1.0133	1.044	1.679	2257	4.217	2.029	279	12.02	680	24.8	1.76	0.984	58.9	750.1	373.15
375	1.0815	1.045	1.574	2252	4.220	2.036	274	12.09	681	24.9	1.70	0.987	58.6	761	375
380	1.2869	1.049	1.337	2239	4.226	2.057	260	12.29	683	25.4	1.61	0.999	57.6	788	380
385	1.5233	1.053	1.142	2225	4.232	2.080	248	12.49	685	25.8	1.53	1.004	56.6	814	385

APPENDIX A.5 DETAIL AND CONSTRUCTION MATERIALS FOR THE FINAL CASE

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*************************************************************
                          CHEETAH

   CSIRO DIVISION OF BUILDING, CONSTRUCTION AND ENGINEERING

              Version 1.33EV February 1995

*************************************************************

                   JOB NAME: FINAL-AL

                   SITUATED: SPECIAL

               CLIMATIC DATA: A hot week

                  DATE RUN: 10 APR 1996

***********************************************************************
                     SUMMARY OF INPUT DATA
                     ---------------------
                       BUILDING SECTIONS
                       -----------------
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NAME	TYPE	CONSTRUCTION	THICKNESS (mm)	U-VALUE (W/sqm.K)
WINDOW 3MM	WINDOW	GLASS	3.	7.0
WINDOW DBL GLZ	WINDOW	GLASS	3.	3.2
		AIRSP UV NR	-	
		GLASS	3.	
BV WALL UNINS	EXTERNAL WALL	BRICKWORK	110.	2.7
		AIRSP V NR	-	
		PLASTERBOARD	10.	
CB WALL UNINS	EXTERNAL WALL	BRICKWORK	110.	2.4
		AIRSP V NR	-	
		BRICKWORK	110.	
		RENDERED PLASTER	10.	
BV WALL RFL	EXTERNAL WALL	BRICKWORK	110.	0.8
		AIRSP V R VERT	-	
		AIRSP UV R VERT	-	
		PLASTERBOARD	10.	
CB WALL R1 INS	EXTERNAL WALL	BRICKWORK	110.	0.7
		AIRSP V NR	-	
		POLYSTYRENE/UF	35.	
		BRICKWORK	110.	
		RENDERED PLASTER	10.	
DOOR EXTERNAL	EXTERNAL WALL	TIMBER(HARDWOOD)	3.	2.9
		AIRSP UV NR	-	
		TIMBER(HARDWOOD)	3.	
ROOF TILES	ROOF	ROOFING TILES	19.	6.2
FLAT ROOF INS	ROOF	CERAMIC TILES	13.	0.4
		CONCRETE, SCORIA	60.	
		GLASS FIBRE INS.	100.	
		CONCRETE, STNDRD	150.	

		RENDERED PLASTER	13.	
CSOG, CARPET	FLOOR AND/OR GROUND	GROUND	-	0.4
		CONCRETE, STNDRD	100.	
		CARPET/UNDERLAY	20.	
CSOG, TILES	FLOOR AND/OR GROUND	GROUND	-	0.5
		CONCRETE, STNDRD	100.	
		CERAMIC TILES	15.	
GROUND ALONE	FLOOR AND/OR GROUND	GROUND	-	0.5
CAR TIMB FLR,AB	FLOOR BETWEEN ZONES	TIMBER(HARDWOOD)	19.	1.5
		CARPET/UNDERLAY	20.	
TIL TIMB FLR,AB	FLOOR BETWEEN ZONES	TIMBER(HARDWOOD)	19.	3.5
		CERAMIC TILES	15.	
CAR CONC FLR,AB	FLOOR BETWEEN ZONES	CONCRETE, STNDRD	100.	1.5
		CARPET/UNDERLAY	20.	
TIL CONC FLR,AB	FLOOR BETWEEN ZONES	CONCRETE, STNDRD	100.	3.8
		CERAMIC TILES	15.	
CAR TIMB FLR,BL	CEILING BETWEEN ZONES	CARPET/UNDERLAY	20.	1.5
		TIMBER(HARDWOOD)	19.	
TIL TIMB FLR,BL	CEILING BETWEEN ZONES	CERAMIC TILES	15.	3.5
		TIMBER(HARDWOOD)	19.	
CAR CONC FLR,BL	CEILING BETWEEN ZONES	CARPET/UNDERLAY	20.	1.5
		CONCRETE, STNDRD	100.	
TIL CONC FLR,BL	CEILING BETWEEN ZONES	CERAMIC TILES	15.	3.8
		CONCRETE, STNDRD	100.	
CEILING UNINS	PARTITION BETWEEN ZONES	PLASTERBOARD	13.	3.9
CEILING INS	PARTITION BETWEEN ZONES	GLASS FIBRE INS.	75.	0.6
		PLASTERBOARD	13.	
PB PART BETW Z	PARTITION BETWEEN ZONES	PLASTERBOARD	10.	2.1
		AIRSP UV NR	-	
		PLASTERBOARD	10.	
BRK PART BETW Z	PARTITION BETWEEN ZONES	RENDERED PLASTER	10.	3.2
		BRICKWORK	110.	
		RENDERED PLASTER	10.	
DOOR BETWEEN Z	PARTITION BETWEEN ZONES	TIMBER(HARDWOOD)	3.	2.6
		AIRSP UV NR	-	
		TIMBER(HARDWOOD)	3.	
ZONE AIR EXCH	PARTITION BETWEEN ZONES	ZONE AIR EXCH.	-	0.3
PB PART IN Z	PARTS/FURN WITHIN ZONE	PLASTERBOARD	10.	2.1
		AIRSP UV NR	-	
		PLASTERBOARD	10.	
BRK PART IN Z	PARTS/FURN WITHIN ZONE	RENDERED PLASTER	10.	3.2
		BRICKWORK	110.	
		RENDERED PLASTER	10.	
FURNITURE	PARTS/FURN WITHIN ZONE	FURNITURE	-	5.0

****

SUBDIVISION OF BUILDING INTO ZONES   NUMBER OF ZONES: 3

ZONE 1 (ZONE 1)

BUILDING SECTION NAME	AREA (sq m)	RELORTN	ADJZN	ABSPT	SHDCF	HSR %	HSD %	VSR %	VSD %	SOWVSR
WINDOW 3MM	6.7	N			0.90	0	0	0	0	
WINDOW 3MM	11.2	S			0.90	54	0	0	0	
CB WALL R1 INS	21.3	N		0.60		0				
CB WALL R1 INS	16.8	S		0.60		0				
CB WALL R1 INS	28.0	E		0.60		0				
CB WALL R1 INS	28.0	W		0.60		0				
CSOG, CARPET	100.0									
CAR CONC FLR,BL	100.0		2							
BRK PART IN Z	100.0									

ZONE 2 (ZONE 2)

BUILDING SECTION NAME	AREA (sq m)	RELORTN	ADJZN	ABSPT	SHDCF	HSR %	HSD %	VSR %	VSD %	SOWVSR
WINDOW 3MM	6.7	N			0.90	0	0	0	0	
WINDOW 3MM	11.2	S			0.90	54	0	0	0	
CB WALL R1 INS	21.3	N		0.60		0				
CB WALL R1 INS	16.8	S		0.60		0				
CB WALL R1 INS	28.0	E		0.60		0				
CB WALL R1 INS	28.0	W		0.60		0				
CAR CONC FLR,AB	100.0		1							
CAR CONC FLR,BL	100.0		3							
BRK PART IN Z	100.0									

ZONE 3 (ZONE 3)

BUILDING SECTION NAME	AREA (sq m)	RELORTN	ADJZN	ABSPT	SHDCF	HSR %	HSD %	VSR %	VSD %	SOWVSR
WINDOW 3MM	6.7	N			0.90	0	0	0	0	
WINDOW 3MM	11.2	S			0.90	54	0	0	0	
CB WALL R1 INS	21.3	N		0.60		0				
CB WALL R1 INS	16.8	S		0.60		0				
CB WALL R1 INS	28.0	E		0.60		0				
CB WALL R1 INS	28.0	W		0.60		0				
FLAT ROOF INS	100.0			0.70						
CAR CONC FLR,AB	100.0		2							
BRK PART IN Z	100.0									

DIRECTION NORTHERNMOST WALL FACES: 0. DEGREES

****

BUILDING USAGE DATA

ZONES OF INTEREST: 1 2 3

HEAT FLOW TO ZONES (Watts)

	=====PERIOD 1======			=====PERIOD 2======			=====PERIOD 3======		
	TIME ON	TIME OFF	HEAT FLOW	TIME ON	TIME OFF	HEAT FLOW	TIME ON	TIME OFF	HEAT FLOW
ZONE 1	0000	1700	550.	1700	1900	1050.	1900	2400	800.
ZONE 2	0000	1700	550.	1700	1900	1050.	1900	2400	800.
ZONE 3	0000	1700	550.	1700	1900	1050.	1900	2400	800.

ZONE VOLUMES AND INFILTRATION RATES

ZONE No.	VOLUME (cub. m)	INFILTRATION RATE (air changes/hour)
1	280.0	0.5
2	280.0	0.5
3	280.0	0.5

****** Efficiency file name: e-night.txt
****** Ventilation file name: a-20.txt

HOURLY RESULTS

JUL 1900 DAY 13

TIME	EXTERNAL	TEMPERATURE ZONE 1	ZONE 2	ZONE 3
0	30.1	26.6	26.8	27.6
1	29.1	26.6	26.8	27.5
2	29.1	27.4	27.5	28.0
3	30.1	27.8	28.0	28.5
4	32.1	27.5	27.7	28.3
5	32.6	29.1	29.3	29.7
6	32.9	29.6	29.8	30.2
7	33.1	25.2	25.4	25.7
8	34.1	24.8	25.0	25.3
9	37.1	24.8	25.0	25.3
10	38.1	23.9	24.1	24.3
11	38.1	25.6	25.9	26.2
12	39.1	23.9	24.0	24.3
13	39.1	25.2	25.4	25.8
14	39.1	23.7	23.8	24.1
15	38.1	24.4	24.5	24.9
16	37.1	24.0	24.2	24.6
17	36.1	24.4	24.5	25.0
18	35.1	23.7	23.8	24.3
19	34.1	23.6	23.8	24.3
20	33.6	22.9	23.1	23.5
21	33.1	25.8	25.9	26.7
22	31.6	27.5	27.5	28.2
23	30.9	27.5	27.6	28.2

JUL 1900 DAY 14

TIME	EXTERNAL	TEMPERATURE ZONE 1	ZONE 2	ZONE 3
0	30.1	27.6	27.7	28.2
1	30.1	27.5	27.7	28.2
2	30.1	27.5	27.6	28.2
3	31.1	28.1	28.3	28.8
4	32.1	28.3	28.5	29.0
5	36.1	28.4	28.7	29.2
6	36.1	29.8	30.0	30.5
7	36.1	26.0	26.3	26.5
8	38.1	24.0	24.2	24.3
9	39.1	24.3	24.5	24.7
10	39.6	24.7	24.8	25.1
11	39.9	24.6	24.8	25.0
12	40.1	24.7	24.8	25.1
13	40.1	25.4	25.5	25.8
14	40.1	25.3	25.4	25.7
15	39.1	25.2	25.3	25.7
16	38.1	24.9	25.0	25.4
17	37.1	24.8	24.9	25.3
18	36.1	25.3	25.5	26.0
19	35.1	27.1	27.3	27.9
20	35.1	28.5	28.6	29.2
21	34.1	25.9	26.1	26.6
22	33.1	27.8	28.0	28.5
23	32.6	28.4	28.5	29.1

JUL 1900 DAY 15

TIME	EXTERNAL	TEMPERATURE ZONE 1	ZONE 2	ZONE 3
0	32.1	28.4	28.6	29.1
1	31.6	28.4	28.6	29.1
2	31.1	28.6	28.8	29.3
3	29.1	28.1	28.4	28.9
4	30.1	28.4	28.7	29.1
5	31.1	28.4	28.8	29.2
6	32.6	29.4	29.7	30.1
7	33.4	26.8	27.1	27.3
8	34.1	26.6	26.9	27.2
9	36.1	25.6	25.8	26.0
10	38.1	29.3	29.6	30.0
11	38.1	27.2	27.5	27.8
12	39.1	29.8	30.1	30.5
13	39.1	27.6	27.9	28.2
14	39.1	26.7	27.0	27.3
15	38.1	25.8	26.0	26.3
16	37.1	25.6	25.9	26.2
17	34.1	29.7	30.0	30.3
18	32.1	30.5	30.7	31.0
19	29.1	29.1	29.3	29.7
20	27.1	28.1	28.4	28.7
21	25.1	27.0	27.3	27.5
22	23.6	26.6	26.8	27.1
23	22.9	26.1	26.4	26.6

JUL 1900 DAY 16

TIME	EXTERNAL	TEMPERATURE ZONE 1	ZONE 2	ZONE 3
0	22.1	25.9	26.2	26.5
1	22.1	25.6	25.8	26.1
2	22.1	25.6	25.9	26.2
3	22.1	25.7	26.0	26.4
4	24.6	25.9	26.1	26.5
5	27.1	26.8	27.0	27.3
6	28.1	27.4	27.6	28.0
7	30.6	24.3	24.4	24.7
8	31.9	24.2	24.4	24.7
9	33.1	24.2	24.3	24.7
10	33.6	23.7	23.8	24.1
11	34.1	23.9	24.1	24.4
12	35.1	27.2	27.3	27.9
13	35.1	27.8	27.9	28.5
14	35.1	28.1	28.2	28.9
15	34.1	26.2	26.4	26.9
16	33.1	25.1	25.2	25.7
17	32.1	24.4	24.5	25.0
18	31.1	27.5	27.6	28.2
19	28.1	27.2	27.4	28.1
20	27.1	26.9	27.1	27.9
21	25.6	26.8	27.0	27.8
22	24.9	26.5	26.7	27.3
23	24.1	26.1	26.3	26.9

	JUL 1900	DAY 17		
		TEMPERATURE		
TIME	EXTERNAL	ZONE 1	ZONE 2	ZONE 3
0	24.1	25.9	26.1	26.6
1	23.1	25.8	25.9	26.5
2	24.1	25.8	26.0	26.5
3	25.1	26.1	26.2	26.7
4	26.1	26.3	26.5	27.0
5	27.1	26.7	26.8	27.3
6	29.1	27.5	27.6	28.1
7	29.9	23.2	23.3	23.6
8	30.7	22.6	22.7	23.0
9	31.5	23.7	23.7	24.1
10	32.4	23.6	23.7	24.1
11	33.2	23.7	23.8	24.2
12	34.1	24.3	24.3	24.8
13	34.1	24.6	24.6	25.2
14	34.1	24.4	24.4	24.9
15	34.1	24.1	24.1	24.6
16	33.1	27.8	27.7	28.3
17	31.1	28.2	28.2	28.7
18	30.1	27.7	27.8	28.3
19	28.1	27.0	27.1	27.7
20	24.1	26.5	26.6	27.2
21	24.1	25.6	25.7	26.1
22	23.1	25.3	25.3	25.7
23	22.6	25.1	25.2	25.6

	JUL 1900	DAY 18		
		TEMPERATURE		
TIME	EXTERNAL	ZONE 1	ZONE 2	ZONE 3
0	22.1	24.8	24.9	25.3
1	22.1	24.6	24.7	25.1
2	21.6	24.4	24.5	24.9
3	21.1	24.2	24.3	24.7
4	24.1	25.0	25.1	25.5
5	26.6	25.7	25.7	26.2
6	29.1	26.9	26.9	27.3
7	30.1	24.3	24.3	24.6
8	31.1	24.1	24.2	24.5
9	32.1	23.8	23.8	24.1
10	33.1	23.7	23.7	24.0
11	34.1	23.2	23.2	23.4
12	35.1	23.0	23.0	23.3
13	35.1	23.8	23.7	24.0
14	34.8	23.5	23.5	23.8
15	34.1	23.3	23.2	23.6
16	33.1	23.2	23.2	23.5
17	32.7	23.1	23.0	23.4
18	32.1	25.5	25.4	26.0
19	31.1	26.5	26.4	27.0
20	30.1	26.2	26.1	26.7
21	29.1	27.0	26.9	27.4
22	28.1	26.4	26.3	26.8
23	27.1	26.1	26.1	26.7

	JUL 1900	DAY 19 TEMPERATURE		
TIME	EXTERNAL	ZONE 1	ZONE 2	ZONE 3
0	27.1	26.2	26.2	26.7
1	27.1	26.2	26.2	26.7
2	27.6	26.3	26.3	26.8
3	27.9	26.5	26.5	26.9
4	28.1	26.7	26.7	27.1
5	30.1	27.1	27.2	27.6
6	31.1	28.1	28.1	28.4
7	32.1	24.6	24.6	24.8
8	33.6	24.2	24.3	24.4
9	35.1	23.8	23.8	23.9
10	35.1	23.4	23.4	23.6
11	37.1	24.2	24.2	24.4
12	37.1	24.7	24.7	24.9
13	37.1	25.1	25.1	25.4
14	37.1	27.4	27.3	27.8
15	37.1	25.5	25.5	25.8
16	36.6	27.6	27.5	28.0
17	36.1	28.0	28.0	28.5
18	33.1	27.9	28.0	28.4
19	32.1	27.9	27.9	28.4
20	31.1	27.9	27.9	28.4
21	30.1	27.9	28.0	28.4
22	29.1	27.9	28.0	28.4
23	28.1	27.8	28.0	28.4

MONTHLY SUMMARY

MEAN TEMPERATURE AND MEAN OF DAILY TEMPERATURE RANGES FOR JUL 1900

	EXTERNAL	ZONE 1	ZONE 2	ZONE 3
TEMP	31.8	26.1	26.2	26.6
RANGE	12.1	5.2	5.1	5.3

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