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REPORT NO. 90

THE EFFECT OF THERMAL MASS OF A STRUCTURE IN ENERGY EFFICIENT COMMERCIAL (AIRCONDITIONED) BUILDINGS

Results of research carried out as MERIWA Project No. E134 at the School of Architecture and Planning, Curtin University of Technology

by

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ABSTRACT

Solar designed buildings in temperate climates are said to save between 60% to 90% of the total energy demand. An integral part of building energy demand reduction to these levels is through the use of thermal mass. This can account for up to 30 to 50% of energy savings. Although many papers have been written as case studies, these percentages have never really been substantiated to the level required for confident use by building developers.

Data on the flow of energy from mass to air and the reverse is essential information that designers require to take advantage of the benefits of thermal mass and design appropriate mechanical systems.

A literature search established that most work relating to thermal mass effects had been carried out for conventionally designed buildings. This was not the case for solar designed buildings, and systems had not comprehensively been researched. Therefore the data, although interesting, has little relevance to solar designed buildings.

The rate of temperature flow , or energy flow, at different times of day, season, and at special air flow speeds, is the type of information needed after a concept of design is accepted.

A test rig was designed and built, which enabled a practical test procedure to be established to measure thermal inertia effects and generate usable information for design. Limited testing was undertaken on one sample of concrete during different seasons of the year.

Laboratory results were collected and analysed. They were compared with a built example, the Solar Energy Information Centre in South Perth, Western Australia.

The outcome of the study was that thermal mass to air heat transfer rates determined from test results compared favourably with the results of the case study. This proved that the test rig and procedure was both accurate and reliable for predicting the heat transfer rates for moving air across a mass surface.

In validating the test procedure and establishing its usefulness, it was apparent that there is a gap in theoretical knowledge, especially air film resistances at slow speeds. Also figures on surface roughness and other effects such as humidity have been largely undocumented and in most cases ignored in previous predictive studies of air to mass heat transfer in buildings. In future work, the test rig and procedure could form the basis of providing comprehensive information on materials to architects and engineers. Also, a new predictive formula may be developed which could incorporate these additional factors which to date have been treated as insignificant and ignored.

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1.0 INTRODUCTION

The objective of this study was to investigate the thermal mass effects on air temperature in commercial buildings that are designed to be energy efficient. Thermal mass is a term which applies to the thermal energy storage contained inside a building, which can be used to reduce the temperature swings, especially between day and night (Birrer,1983). Specifically, the knowledge sought was the ability to correctly predict the volume and area of thermal mass required to take advantage of energy efficient features in the design and thus to reduce the size of the mechanical air - conditioning plant.

With passive solar house design, a large amount of information is available in a "user friendly" form for architects and builders to use. However similar information for passive solar commercial buildings is not available and what is, may not in a form for easy understanding or application by the architectural practitioners.

The largest factor unidentified and unquantified is the thermal inertia effects of a structure and its thermal behaviour (Catani and Goodwin, 1977). Therefore, with this awareness of the inadequacy of knowledge, it was logical that an interest in further information in this area be developed.

1.1 Energy Consumption in Buildings

Over the past 10 years a small number of low energy, passive solar designed commercial buildings have been constructed in Australia. Although calculation methods and computer simulation models are available to predict their performance, engineers are often unable to effectively and efficiently design mechanical services for these new generation low energy designed buildings (Baverstock and McGeorge, 1987).

In conventional air-conditioned buildings, because of the small areas of the mass of the structure in contact with the air, engineers generally negate thermal mass effects in determining heating and cooling loads. With commercial buildings designed to be energy efficient, apart from solar exposure, air infiltration and the thermal efficiency of the building envelope, the largest factor determining the air temperature inside a building is the thermal inertia of the structure, particularly when there is sufficient mass in the building to cope with the loads (Birrer, 1983).

Designers, from practical experience, have commonly relied on the notion that night ventilation can lower the temperature of a structure to within 1°C - 2°C of the minimum outside air temperature in temperate climates, thereby enabling the thermal mass to lower the temperatures during the next day (Kusuda and Bean, 1981). Conversely in winter, heating of the mass during the day enables warmer temperatures at night and sufficient carry over for subsequent cold days (Heap, 1975).

Harnessing thermal inertia in commercial buildings in Australia has many implications. Solar buildings in Australia save between 60% and 90% of energy costs (Ballinger, 1988), and at least 30% can be saved by utilizing thermal mass in buildings (Birrer, 1983). Given that 19% of total energy use relates to supplying electrical power to buildings, it is possible that the use of thermal mass could save about 6% of total energy demand in Australia (Ballinger, 1988). Overall, based on a 60% saving, a conservative total of 12% of the total energy demand could be saved if all buildings were energy efficient buildings. Furthermore, since the majority of electricity power generation in Australia is used in buildings, and as this power is generated through burning coal, it can be seen that low energy building design utilizing thermal mass has enormous potential in reducing greenhouse gas emissions; that is apart from the reduction in total energy consumption.

1.2 Thermal Inertia in Buildings

As previously mentioned, there is a lack of user friendly information about thermal storage in low energy commercial buildings. However a number of papers were written and books published throughout the 1960s, 1970s and 1980s which provided valuable groundwork for future implementation of design strategies for passive solar buildings.

In the early 1960s in England, Pratt and Ball (1963), looked at the theory of heat transfer and thermal storage in detail. In the late 1970s many scientists in the USA were investigating buildings and providing strategies for reducing temperature swings (Balcomb, 1978a). A number of papers written by Balcomb covered all types of passive solar buildings and formed an important part in the US Department of Energy assessments of the utilisation of solar energy in the late 1970s (Balcomb, 1978b, 1978c, Balcomb et al, 1978, Balcomb & Wray, 1978).

In the 1980s there was a further world - wide scientific quest for more information and theory about solar design of buildings, with the work of Dexter (1980), Kusuda and Bean (1981), Lokmanhekim (1975), among others. Arumi (1981) documented evidence showing that theory was actually proven and supported the hypothesis that thermal mass can have a significant effect in reducing energy demands in commercial buildings, particularly in temperate climates. In Australia, Ballinger, (1983,1988,1989), Szokolay, (1975,1982), and Williamson and Coldicutt (1980) were producing similar material, which expanded knowledge about passive solar design strategies for buildings.

The use of thermal mass in conjunction with an optimized thermally efficient envelope was promoted as a viable approach, leading to at least 30% energy savings in office buildings constructed in South Africa (Birrer, 1983).

The importance of appropriate building design as the key factor to take advantage of thermal mass effects has been identified (Peavy et al, 1973). If inappropriately designed or if the building is a non-passive solar design, then thermal mass effects can be insignificant or even have adverse effects on comfort and energy demand.

In the context of thermal inertia effects, "environmental temperature" was an important theoretical concept. Many publications brought together the radiant heat effects on human comfort (Markus and Morris, 1980). It was stated that the combination of the effects of air speed, mean radiant temperature, as well as supplying air at certain temperatures, could provide alternatives for providing human comfort (Burberry, 1970). Using mass to advantage, by using radiant heat effects, was seen as a desirable strategy to saving energy in buildings (Oughton, 1980).

Because of the substantial effect of radiant temperatures on the air temperature and comfort of the space, and the greater exposure of mass surface in a space, the greater the effect of thermal inertia in helping to maintain temperature (Lee and Oberdick, 1981).

Energy efficiency was the underlying objective of reviewing thermal storage systems in buildings by Hariri and Ward (1988). They found that storage systems were wide and varied. Also their formulae are useful in providing a basis for calculating storage capacity and the diffusion of energy within the mass.

The relationship between thermal mass and its transfer characteristics within a thermal design was also seen as crucial to the energy efficiency of a building and the comfort of the occupants (Birrer, 1983, Balcomb, 1978a, Bahadori and Haghighat, 1985).

Although these papers described the characteristics and benefits of thermal mass, the useful and detailed design information and some basic fundamental facts about low - energy commercial buildings was needed and justified further research.

1.3 The Effect of New Building Technologies

Because of high interest rates and the high cost of on- site labour in most developed countries, speed of erection is a key factor in the architect's choice of materials in a building design. This must always be taken into account in the design of contemporary buildings, whether or not they are solar designed.

The tendency is therefore for buildings to become less massive, to be prefabricated using steel frame in preference to concrete and for the use of precast lightweight units instead of masonry walls. This is contrary to what most casual observers would expect as a feature of low energy buildings.

Since energy efficiency is only one ingredient of the design of a building, some compromises must be considered in the choice of materials and labour. It is mandatory for the solar conscious architect to take these new materials and techniques into account and maximise their thermal features.

Therefore, with the realities of architectural design practice in mind, the choice of materials must not only be appropriate structurally, but also maximise its thermal storage characteristics - when important - and incorporate resistance to heat transfer, particularly for external walls and the roof.

A different design approach is needed in reducing perimeter and skin loads of a building to an optimized minimum over the full year. This was evident in the design of the Solar Energy Information Centre (refer to Appendix 5) wherein the following new materials and passive design features were incorporated:

1. Exposed steel permanent formwork/ceiling - 'Bondek' - used as a thermal bank in contact with the air volume, instead of suspended acoustic ceilings.
2. Lightweight concrete wall systems - 'Siporex' - for insulated external walls such as fire rated party walls - in contact with the air in the building, and also exposed to external ambient air conditions in some cases.
3. Solid demountable partitions such as gypsum/polystyrene bead core, gyprock faced - to add thermal storage value.
4. Polystyrene - steel faced - spandrel and integrated roof panels - to reduce heat transfer on the exterior face of a building - both with insulating and mass characteristics.

The unresolved question is - how much benefit is achieved from using specific types of material in certain ways and positions.

1.4 Research Hypothesis

To validate the idea of using the thermal mass of a structure to affect air temperatures and thus reduce heating and cooling requirements, a specific design solution was considered a prerequisite to the proper testing of the hypothesis.

Once a certain design solution was assumed, testing was required to establish the extent that the ideas were successful. Answers were required to some basic questions:

1. How long does it take for the thermal mass of a structure to effect air temperature in a space under various air supply conditions of temperature and air speed? This was

considered to be a fundamental question.

2. What is the contribution of thermal mass in saving energy in an energy efficient and solar designed commercial building?

Only a methodical testing and monitoring program would provide the answers to these questions. To limit the field of study and to make the results manageable, a simple design concept for a test procedure had to be assumed prior to any predictive work being done.

1.5 Report format

Prior to embarking on developing methods to test thermal mass effects in commercial buildings a great deal of background was required to establish the current state of knowledge. A wide variety of subjects needed coverage, ranging from climatic information, human comfort, air flow rates for health, thermal mass effects, evaporative and ventilation systems and other associated technologies and energy efficiency figures for commercial buildings. Relevant information covered in section 2.0 helped establish parameters for a test procedure and the design brief for a testing apparatus.

Current theory had to be assessed. In section 3.0 evaluation of formulae for predictive work is described. This enabled informed decisions to be made about the detail design of the test rig.

A test rig was designed and constructed. Section 4.0 describes the rationale and the procedure involved in bringing the rig into existence.

Section 5.0 reports on the monitoring program and the analysis of the results. Since claims of 60% to 90% energy savings have been made, this section looks at the monitored results and extrapolates the probable energy conservation rate from the thermal mass effects, so comparisons could be made. Data from previous case studies also helped reach conclusions about this broad claim (Section 6.0).

Finally, conclusions are made in relation to the basic questions posed at the beginning of the study. The definitive result of the study was the establishment of a new test procedure to validate the idea of using thermal mass in reducing heating and cooling requirements in buildings. This is explained in Section 7.0.

This report finally makes a case for continuance of research in this area. Prospects for future work are identified in Section 8.0 .

2.0 BACKGROUND

A number of basic issues such as climate, energy efficiency in commercial buildings and factors affecting thermal mass were investigated at the start of this project. This chapter discusses the following:

1. Climatic Information
2. Energy Consumption of Commercial Buildings
3. Case Studies of Energy Efficient Buildings
4. Human Comfort
5. Air Flow Rates
6. Evaporative Cooling Systems
7. Heating and Cooling Loads
8. Relevant Data

These topics were included because their investigation could influence the direction of the research and the test program adopted. Their study helped formulate the hypotheses and provide comparisons with research results in order to validate conclusions made.

2.1 Climatic Information

The Australian Bureau of Meteorology (1975) considered the following proportion of the land area of Australia as having a temperate climate:

%	State or Territory
63	Western Australia
19	Northern Territory
46	Queensland
100	Victoria, Tasmania, South Australia, A.C.T. and New South Wales

Table 1 Areas of Australia in Temperate Climates

It is significant that low energy buildings operate most effectively in temperate climates and that the majority of the Australian population live in this defined area.

2.1.1 Temperate Climate

A temperate climate is said to exist between 27.5°S to 45°S latitude in the southern hemisphere (Australian Bureau of Meteorology, 1975). A temperate climate is moderate both in summer and winter with temperatures rarely becoming life threatening from extreme conditions. It is defined as in Table 2 (Australian Bureau of Meteorology, 1980) :

Summer		Winter	
day temp.	night temp.	day temp.	night temp.
30 - 35°	13 - 18°	18 - 25°	5 - 13°
		(A wide diurnal swing is evident)	

Table 2 Temperate Zone Average Temperature Range

Humidity is described as moderate (30% - 40%) and is higher in coastal regions.

Perth, Western Australia (latitude 32.5°S), has a recognised temperate climate; night ventilation cooling is successfully used in summer. According to the Bureau of Meteorology the mean January day - time temperature is 30°C and the night - time mean is 18°C.

From experience, night - time minimums must be below 22°C in order to achieve any night cooling benefit in summer. Alice Springs has a mean night time summer minimum of 21°C. Although this does not strictly qualify as a temperate climate, the same cooling techniques could be applied. Thus Alice Springs would be considered the northern-most city to almost qualify as temperate.

2.1.2 Definition of a Low Energy Building

A low energy office building designed to suit a temperate climate can be defined as follows :

A building that is located in a climate which requires summer cooling and where night time temperatures will provide human comfort when ambient air is introduced to cool the thermal mass over night. In addition to this, the climate exhibits opposite effects in winter, with outside temperatures and sunlight providing a substantial amount of warmth if it is transferred into the building during daylight hours.

2.2 Energy Consumption of Commercial Buildings

In Australia the energy consumption for a typical office building, varies between 335 - 710 MJ/annum/m^2 (BOMA, 1986). Converting these energy units to kWh/m^2 and translating them into energy costs, the typical office building energy consumption would be \$20 to \$43/annum/m^2.

In many countries targets for energy conservation have been set and energy use figures have been analysed. In Australia for instance, it has been demonstrated that buildings use 26% of the total energy demand and about 17% of the final energy demand (Brown et al, 1985).

Guide-lines, for energy consumption in offices, have been suggested as follows (BO MA ,1986);

-cooling 120 MJ/ annum/m^2
-heating 200 MJ/ annum/m^2
-hot water 5MJ/ annum/m^2

An index (MJ/annum/m^2) can be applied for various locations. For example, for Perth:

cooling	* 1.8 x 120	= 216	
heating	* 0.54 x 200	= 108	* indicates the
hot water	*1.0 x 5	= 5	index for Perth
Total		= 329 MJ/annum/m^2 or, 90 kWh/annum/m2 ($16.2 /annum/m^2) at 18¢ / kWh	

Plus lighting, lifts, ventilation, etc.
Allowance by B O MA = 240 MJ/annum/m^2 or 70 kWh/annum/m^2 (\$12.6/annum/m^2)

Total = <u>\$28.8/annum/m^2</u>.

In order to provide heating and cooling in a building, 60 kWh/m^2 per annum has been considered usual for efficiently designed office buildings in temperate climates (Swan et al 1981). This implies lower figures than the B O M A guideline-line of 90 kWh/m^2. Perth has a warm temperate climate and there is a higher cooling load which explains the extra energy demand recommended by B O M A.

2.3 Case Studies of Energy Efficient Buildings.

To the layman observer there may appear to be conflicting opinions about the extent of the effect of thermal mass on the energy efficiency of a building. The main cause for this confusion is the type of building that is being examined.

The key to obtaining a significant effect of thermal mass in an energy efficient building, is the design measures taken (Birrer, 1983). In Birrer's example cooling loads were reduced firstly by using a single duct variable air volume system (VAV), and further reduced by using the structure (exposed mass) and again by using cooled ventilation with the structural storage.

The data presented a significant drop in energy demands by good thermally efficient design of the building envelope, with a substantial energy reduction from incorporating and exposing the thermal mass of the structure to the inside of the building. Further reductions were obtained by ensuring the air supplied to the space passed through the mass of the structure.

Birrer presented a comparison of a conventional building with heat loads of 125 W/m^2 for cooling and 100 W/m^2 for heating and a building with thermal mass structure, wherein the loads were 50 W/m^2 and 40 W/m^2 respectively. The key to this concept was the exposure of thermal mass to the air supply system. The reduced peak load demand on energy resulted in 35% running costs savings in two buildings, the Standard Bank, Johannesburg, and the Cullian Electrical Division, near Johannesburg (Birrer, 1983).

Other techniques can be used to save energy in buildings. For example, solar heating in winter reduces the reliance on fossil fuel sources. The use of solar air collector systems has been extensively explored in the United States of America and well documented in the late 1970s (Balcomb, 1978b). In temperate climates some supplementary heating may be required at certain times. In colder climates it has been shown that the combination of solar air collection systems with structural thermal mass storage can save between 50% and 80% of the total energy bill (Balcomb, 1978c). Also the use of high efficiency storage systems such as phase change chemical tubes can ensure that discomfort on cold mornings caused by start up thermal lag can be minimised.

Since 1978, experience in Perth has been that thermal mass will help cool conventional buildings using night ventilation techniques, without undue cost penalties (Baverstock, 1983). The "Rokeby Road Offices" were designed in 1980 to have a lower capital cost and 50% energy savings (Baverstock, 1983). The passive solar effects were recognized by the tenants and the effects of the thermal inertia combined with rock stores, were useful but not as effective as expected as measured by the energy savings cost. Savings between \$2/m^2 and \$3/m^2 (1980 dollars) per annum were recorded for this 530 m^2 building while other case studies have since saved \$7/m^2-\$21/m^2 (refer to following examples).

A refurbishment and retrofit of a three storey office complex in Perth, the Russell Centre, between 1985 and 1987, used thermal mass cooling combined with a conventional air conditioning system (Baverstock and McGeorge, 1987). This project demonstrated similar results: capital costs were comparable with a conventional approach but substantial energy savings were made as is shown in Table 3:

Heat Load	External	Internal	Total
Original Old building	\$9#	\$2	\$11
Near Empty Old building (Mostly occupied)	\$9	\$13	\$22
Refurbished building (Mostly occupied)	\$2	\$13	\$15
	# energy bills in \$/annum/m^2		

Table 3 Energy Costs for the Russell Centre, Adelaide Terrace Perth 1985 -1987 (Baverstock and McGeorge 1987)

The total energy bills, as shown above, of the Russell Centre energy retrofit was \$15/ annum/m^2 (Baverstock and McGeorge, 1987). The savings were as high as 30% compared with a conventional office building. If further innovation occurred with this project, such as the incorporation of evaporative cooling systems to augment the ventilation/mass system, energy costs would have been further lowered. This was the case with the Solar Energy Information Centre (SEIC) in South Perth, built in 1989 (Baverstock and D'Cruz, 1992).

The total energy demand of the Solar Energy Information Centre is \$9/annum/m^2 (Baverstock , D'Cruz and Healy 1992). It was a saving of 69% that could be achieved compared with average non-low energy buildings, and 79% compared with the recommended upper range figure for Perth (BOMA ,1986).

The contribution of the thermal inertia towards this effect is generally acknowledged. After an analysis of designs, other researchers using simulation techniques have been able to evaluate the effects of thermal mass, and it has been stated that in an analysis of a building that used thermal mass : *"Existing configurations work well, with a very large reduction in energy use compared to typical buildings and generally good comfort conditions"* (Burt et al 1987, pp 227).

Case studies of conventional buildings and standards set by BOMA provided validated results for comparisons with solar buildings. A baseline for making comparisons was established by an engineer who provided figures for a typical conventional energy efficient building in Perth (Healey, 1991). A summary of annual energy figures for a building considered to be energy efficient is shown in Table 4 :

	Tenant Power	Mechanical House	Lighting & Power House	Total
5/8 to 2/9	7.45#	10.42	5.52	23.39
11/7 to 5/8	6.41	10.17	5.11	21.69
10/6 to 11/7	7.33	10.12	5.69	23.14
12/5 to 10/6	7.72	9.20	4.66	22.54
13/4 to 12/5	7.32	7.18	3.49	17.99
10/3 to 10/4	6.85	7.95	6.30	21.10
10/2 to 10/3	7.13	8.24	6.71	22.08
Jan to Feb	5.81	8.10	6.00	19.91

Markalinga House Perth, 1988 (Healey, 1991)
All figures in dollars per metre square per annum.

Table 4 Energy costs for Markalinga House, Perth WA

This study shows a total energy demand on average of $22/m^2/annum. Less efficient buildings as described by BOMA can create energy demands of up to $43/annum/m^2, as previously shown . These inefficient buildings are clearly and visibly identifiable in most cities, particularly those with the unshaded glass wall constructions popular in the 1980s.

From these case studies it was possible to obtain a picture of the type of energy savings possible from new solar designed buildings. The savings appeared to be between 30% and 90%.

Thus theory is borne out in practice and there is agreement in principle that massive buildings can produce large energy savings, provided the heat loads are controlled by efficient design of the thermal envelope. However, certain problems need to be overcome; that is start up time, discomfort during winter (after a cold night), and temperature build up after a hot night, when the mass has not been able to be cooled adequately by a night ventilation system (Kaushik et al., 1981).

Although overall energy figures from case studies indicated the value of the solar design strategy, the contribution of the thermal mass remained largely unquantified as a separate component. Therefore the part played by thermal inertia in these figures for solar buildings was open to further investigation and awaits more accurate predictive methods for incorporation into building designs.

Decision making on the design of buildings is usually acknowledged as pre-emptive. For instance, once decisions on mass are made by the designer, then glazing areas, shading, etc. can be designed (Burt et al 1987). However, basic constraints are needed for the designer to make these pre-emptive decisions. The quantity of mass, the area of mass in relation to the floor area, the approximate energy savings attributable to the thermal mass directly and indirectly, are all elements of knowledge which could assist the architect, as described in the introduction.

2.4 Human Comfort

The provision of energy to a building is largely used to enhance the comfort of the occupants. Energy conservation strategies must rely on minimizing the amount of energy supplied to allow these comfort levels to be attained. Before quantifying energy savings, it was necessary to understand the science of human comfort.

The human body is a naturally balancing system. Surface skin temperature can vary from 17°C to 40°C with a pain limit of 45°C. Given tolerable external air temperatures that will not threaten life, the human body will normally adjust to the environment and maintain an internal temperature of 36°C to 39°C (Olesen, 1982). However to investigate energy savings, the comfort of humans was found to be far more intricate and subtle.

Human beings achieve comfort in many ways. Low energy air-conditioning systems rely on this fact to take advantage of the thermal mass energy storage of a building (Birrer,1983). Therefore to maximise the advantages of passive and active cooling and heating of thermal mass within a building it is important to understand how human beings perceive comfort and how it is calculated.

Comfort parameters have been defined for humans while carrying on various activities and occupations (ASHRAE, 1985; AIRAH, 1980). For the purposes of this study it was assumed that most activity is likely to be sedentary. Key determinants are shown in Table 5:

Dry-bulb temperature	(°C)
Globe temperature to indicate radiation effects entering the room	(°C)
Mean radiant temperature of the spatial enclosure element	(°C)
Air movement	(m/s or ACH)
Relative humidity	(%)

Table 5 Key Parameters for Thermal Comfort

Many researchers have taken these parameters and upon assuming clothed humans have devised a composite guide known as effective temperature (ET) and environmental temperature (T_e) (Markus and Morris, 1980).

ET is the human comfort perceived temperature which takes into account the dry bulb temperature, the air movement effects on evaporation of the skin combined with the humidity effects. With passive solar designed buildings receiving direct solar radiation (particularly in winter), radiant effects must also be taken into account.

It is this corrected figure which is most important for passive solar buildings utilising solar efficient building envelopes, exposed thermal mass and high levels of air movement.

Graphs are available (Markus & Morris, 1980) which have been devised to balance the effects of the parameters defined above. For fully clothed humans with ventilation air speeds of up to 1.5 m/s, it is possible to ensure comfort by extrapolating from mean radiant temperatures to assess the acceptable ambient temperature internally (Burberry, 1970). For example, a mean radiant temperature of 28°C and air speed 1 to 1.5 m/sec is allowable when the ambient air is 25°C. Also, a mass temperature of 20°C and air movement of 1 to 1.5 m/sec will make a 28°C air temperature acceptable, provided the relative humidity is about 50%. Higher speeds can be useful with higher humidity.

It is therefore possible to postulate, that by changing the nature of cooling and heating systems, energy savings in the operation of buildings could be instituted. Specifically, this could be done by the following measures which take advantage of the way humans experience and perceive comfort :

1. Acceptance of higher humidities than that produced by refrigerative air conditioning systems in summer.
2. Higher air speeds moving over the human occupants.
3. Lower thermal mass temperatures in summer and conversely higher in winter using passive and active solar techniques.

This means that it is possible to utilise low energy systems as follows :

1. Indirect evaporative cooling for reduction of dry bulb temperatures and manageable increases in humidity in summer.
2. Solar air collection for supply of solar heated air in winter.
3. Higher speed recycled and economy cycle systems at higher temperatures to take advantage of natural evaporative effects on contact with the human occupants and therefore reduce supplementary cooling loads in summer.
4. Use of night ventilation in summer and solar air collection in winter to control thermal mass temperatures to stabilise the environmental temperature.

2.4.1 Comfort Air Temperature

The acceptable sensible comfort temperature for clothed adults was a question which needed to be addressed, prior to setting temperature parameters for further research or testing.

Thermal comfort conditions vary with the climate. According to Szokolay (1987), acceptable thermal comfort is given by the following formulae -

$$T_n = 17.6 + 0.31\, T_o \qquad \text{(EQ 2 - 1)}$$

$T_n = \pm 2$ degC (for the year)
$T_n = \pm 1.75$ degC (for the month) for $18.5 < T_n < 28.5$

Where T_o = the mean day and night temperature for the period of consideration (°C)
T_n = acceptable comfort temperature (°C)

According to this formula the following comfort levels are calculated;

Climate	Summer	Winter
Tropical	28.5	22.15
Subtropical	27.5	20.1
Temperate	27.1	18.5

Table 6 Summer and Winter Comfort temperatures (°C) in Australia

These temperatures are for non-moving air. Moving air will effect these comfort air temperatures with comfort being attained with higher temperature combined with higher air speed. Adjustment of humidity can also alter this acceptable air temperature. For instance, it is possible for a comfort level to be attained with lower temperature and higher humidity or higher temperature and lower humidity. Using air speed and controlled humidity in temperate climates, the set points of an air conditioning system could be 1.5°C to 2.0°C higher than generally accepted by air-conditioning engineers (Browne and Moller, 1987) .

2.4.2 The Effects of Clothing

Before accepting Szokolay's suggested comfort temperatures for non- moving air it was necessary to check to see if the guideline temperatures could be adjusted if the extent of clothing was varied to suit the season.

In an air-conditioned space it has been established that the amount of clothing will effect the comfort temperature for a person in that space. For instance, from charts produced by Humphreys (1976) the consideration of metabolic rate was introduced. Higher metabolisms require less temperature for comfort. In temperate climates, for an energy efficient strategy, it was considered that light clothing (0.5 clo) be worn and in winter warm clothes be worn (2 clo) and mid season ordinary dress be appropriate (1 clo).

A "clo" is a term which defines the level of clothing worn by an adult human. For instance light clothing (0.5 clo), relating to summer dress, would be a shirt or blouse. The level of clothing must be appropriate to the season and can be used to save on energy in buildings. For instance, assuming occupants of a building are wearing 2 clo, an engineer can design the comfort level to a minimum in winter (e.g. 18°C). Conversely, assuming 0.5 clo in summer a maximum of 29°C can be permissible. Thus energy savings are obviously possible by the promotion of appropriate dress for the seasons.

Summarizing :

Summer	(0.5 clo)	21°C - 29°C
Winter	(2 clo)	18°C - 22°C
Spring Autumn	(1 clo)	19°C - 27°C
(For a range of metabolisms & a range of climates)		

Table 7 Clothing and Temperature Levels for Comfort.

2.4.3 Mass Temperature

Thermal mass surrounding a space can effect the perceived temperature by humans. Mass emits or absorbs from the space, creating what is known as the environmental temperature (T_e) :

$$T_e = 1/3\ t_{ai} + 2/3\ tri\ (°C) \qquad \text{(EQ 2 - 2)}$$

where:

t_{ai} = Internal temperature (dry bulb) (°C)

t_{ri} = Mean radiant temperature of the surrounding surfaces (°C)

Surface temperatures of the mass are important to comfort. As a guide in conventional buildings it has been stated that provided the surface temperature of the radiant mass does not vary by 4°C more than the air temperature then comfort is maintained (DHC, 1980). Mass temperatures in conventional buildings are also recommended at 22 °C - 28 °C (DHC, 1980).

The review of the subject confirmed the strategy of having warmer mass than air temperatures in winter and conversely cooler mass than air temperature in summer.

2.5 Air Flow Rates

As already seen in the concept of effective temperature, a key factor in the perception of comfort is air flow over the human body. Therefore, a review of what is needed to achieve and enhance comfort was necessary. The air flow rates are important from three perspectives :

1. Human comfort (section 2.4)
2. Health - free air requirements.
3. Heat exchange with the structure - to maximise thermal mass effects.

Expressed conveniently in air changes per hour (ACH) the following rates were found to be generally accepted in temperate climates (DHC, 1980):

Note: Air speeds of 0.2 m/s to 0.5 m/s are generally acceptable for comfort and up to 1 m/s is considered an extreme maximum for office work (from DHC, 1980)

Function	Ventilation Rate
Human Health & Comfort	2-5 ACH
Night Ventilation of a Mass Structure	15-25 ACH

Table. 8 Desirable Air Flow Rates and Speeds.

The recommended air change rate for offices is defined as 0.5 to 1.0 ACH (Clarke, 1985). These figures are perhaps set as minimums to suit refrigeration cycle systems, given the knowledge that it is very inefficient and expensive to provide any better levels of ventilation for conventional air conditioning systems.

In Australia, an allowance of 0.5 ACH for fresh air intake is an accepted rate when calculating cooling loads (AIRAH, 1980).

It must be pointed out that ACH rates of 0 to 0.5 are common with conventional refrigeration cycle air conditioned office buildings in temperate climates. They may achieve comfort conditions, but they certainly do not satisfy proper health requirements and obviously do not provide enough air exchange to enable the system to take full advantage of the climate and thermal storage of the building structure.

The relationship between temperature and air speed is crucial for human comfort in solar designed buildings (ASHRAE, 1985). However, the crucial ingredient for low energy buildings is the use of ambient air, solar heated or evaporatively cooled, to not only keep air temperatures within a broad comfort range, but also to affect the thermal mass temperature of the building in order to level the loads between day and night.

2.5.1 Health

The usage of a space in a building determines the recommended fresh air exchange, as ventilation is required to reduce the build-up of water vapour and heat from people. It also restores the imbalance produced by oxygen deficiency and carbon dioxide and reduces the effects of air pollution such as cigarette smoke, and other air-borne pollutants.

To establish the air flow rates for health, reference was made to guidelines for fresh air values (DHC 1980), which covered residential type spaces, and 'typical air change rates' (Clarke, 1977). It was seen from these two sets of figures that kitchens require 20 to 30 ACH, bathrooms and WCs require 4 to 8 ACH and offices required a range of 0.5 to 1.0 ACH. However, practice and discussions with mechanical engineers indicate that these figures are rarely attained (Healey, 1991). Many small commercial buildings, for instance, do not have a fresh air intake capacity designed into their system and rely purely on fresh air coming through doors and infiltration through windows.

In commercial buildings minimum rates for health reasons are usually the determinant of fresh air rates. The number of people in a space determines the recommended forced ventilation, e.g. 12 m^2/person requires about 4 L/s (British Gas, 1981). The Chartered Institute of Building Services (CIBS,1976) indicates that a figure of 0.5 to 1.0 ACH was desirable. However it further required that this air flow rate be overridden by stipulating a maximum energy rate of heating per unit volume of air being introduced (CIBS, 1976). Naturally, in England if external temperatures are too low it would be imprudent to draw in the recommended rates of fresh air all the time. The energy restrictive figures imply shut-down times since they were much lower than the health requirement for fresh air.

The figures supported the basic hypothesis that low energy solar buildings require a different set of parameters to conventional buildings and therefore supported the need to allow for shut-down times in saving energy through the use of thermal mass.

2.5.2 Mass/Thermal Effects

From the investigation of case studies, the fundamental reason why thermal mass can be significant in lowering energy demand in buildings was heat flow between the air and mass (Birrer,1983). This interchange occurs while the air is moving, most of the time. In cooling a building using thermal mass, night time exhaust of the air in the building takes advantage of the cool outside air.

The Australian Department of Housing and Construction guidelines (DHC, 1980) give recommended ambient air change rates to night cool a light or heavy mass structure. Relating the mass of a building to the total volume of the building is an appropriate way of assessing the extent of thermal inertia in a structure. The classification of the weight of a structure is shown in Table 9:

Heavy	>180Kg/m^3
Medium	between 120 and 180Kg/m^3
Light	<120 Kg/m^3

Table 9 Building Structure Weights

A light - weight structure will require 10 -15 ACH at night for cooling, while a heavy - weight structure will require up to 25 ACH for effective night cooling (DHC,1980). The heavy weight structure will obviously have more thermal inertia, thus justifying the supply of more air and use of more energy to provide night cooling.

As discussed in the introduction, with emerging new building technologies and the tendency for lighter weight industrialised component buildings both now and predicted for the future, the aim of the designer would be to target for a medium mass building. However, the exposure of the air to a larger surface area of mass is more important than simply just designing for as much mass as possible (Clarke, 1977).

One method of increasing the area of thermal mass in contact with the air is to introduce a rock store (Baverstock, 1983). However mass cooling requires the air to pass relatively slowly over and through a rock bed. Recommended air speeds are 0.06 - 0.12 m/s to restrict the size of rock bed storage (Baker et al, 1984). An analysis of the volumes of mass required to cool an office building show that these speeds are unsatisfactory since they would produce unacceptable ventilation rates as outlined previously. A low energy office building cannot therefore rely on a rock storage system alone if the project is going to be cost effective. The building structure itself must provide the major source for the mass/air interchange.

To increase the thermal mass of a building, the use of phase change chemical tubes may make more sense because air flow rates can be increased. Tubes can be installed in the return air or supply duct to have a quicker reaction time. This means that only one air supply/return air system needs to be used, reducing costs in mechanical plant. The quicker response and higher charge per mass capacity, the smaller volumes of material are needed to be used (ICI Calor Alternative Energy, 1976).

Because of the good heat transfer between mass and air at slower speeds, the strategy of maximising the thermal inertia effects of a structure may have to incorporate regular shut-down times when outside conditions are unsatisfactory for economy cycle and when the mass is still at the comfort temperatures.

Rates of 15 ACH to 25 ACH are thus seen as the desirable rates of forced ventilation for night cooling. Tests done in the USA have shown that at least 15 ACH was necessary and effective for using ambient temperatures for passive cooling of buildings (Kusuda and Bean 1981).

During daylight hours 2 ACH to 5 ACH are seen as a desirable rate which both satisfy health requirements and introduce sufficient energy in the air to effect the thermal mass areas within the building. Addition of heat to the mass in winter and transferring coolness from evaporatively cooled air in summer was necessary in extreme conditions to justify this rate of air exchange.

A key to extracting more benefit from energy stored in thermal mass walls (enveloping a space) is to allow shut-down periods with 0.0 ACH to 0.06 ACH. With more than adequate fresh air intake for health during ventilation times, it was considered acceptable to assume stagnant periods, given that most conventional buildings are usually circulating stale air. In England heating periods have been cycled and savings instituted in housing energy bills (Desson, 1976). Many buildings in the USA have been analysed and energy management programs instituted to save energy (Rohles, et al 1984).

It is acknowledged by engineers that shut down operating modes do in fact work and it is the thermal mass of the building which establishes and maintains temperatures during shut-down times (Rudoy and Robbins, 1977).

Summarising the investigation of forced ventilation, it can be said that there are modes of operation required by a forced ventilation system in a low energy building (refer to Table 10) . These rates require confirmation by further research:

1	Night ventilation cooling	15 - 25 ACH
2	Supply of air from solar air collectors	2 - 5 ACH
3	Economy cycle fresh air ventilation during daylight hours (if outside temperatures are correct)	2 - 5 ACH
4	Internal circulation and shut down times.	0.06 - 1 ACH

Table 10 Forced Ventilation rates

2.6 Evaporative Cooling Systems

In order to improve the energy demand of low energy buildings, low energy cooling systems should be incorporated. Evaporative systems are less energy dependent than refrigerative systems and so the validity of using evaporative systems in low energy buildings was worthy of more detailed investigation. Temperate climates usually place a much larger demand on cooling in a commercial building. Since passive design strategies cannot provide 100% of cooling requirements, mechanical systems are needed. Indirect evaporative cooling can be a low energy way of providing this cooling. The heat exchange system used with an indirect system can reduce the humidity build-up problem with the supply air. This occurs because the water is evaporated by the exhaust air in the building. Therefore moisture build up problems associated with conventional evaporative systems are avoided.

An indirect evaporative cooling system was used in the Solar Energy Information Centre (Baverstock et al, 1992). The system used was the "Dricon" system. New methods combining desiccant materials are now being developed to make the system more effective in humid weather (McNab, 1990). Dehumidification methods are also currently being researched and developed extensively in the U S A (Meckler, 1989).

The Solar Energy Information Centre, utilises the simple indirect system without the desiccant dehumidification function. However, relatively high humidities are experienced at certain times of the year. The system operates reasonably successfully in relatively high humidity of up to 70%, which occurs occasionally at certain times of the year in Perth, Western Australia.

Elsewhere, direct and indirect evaporative systems have demonstrated a saving of 40% to 50% as compared with a conventional refrigeration cycled air-conditioned building in Colorado, USA (Schofield and Deschamps, 1980). This scale of saving is significant and should be included in an overall strategy, if the objective is to maximize energy savings in commercial buildings.

As with the American experience it has been recognized in Australia that significant energy savings are possible for conventional evaporative cooling in a wide range of temperate climates. Savings are documented as 40% to 80% in Australian temperate climates (Williamson and Coldicut, 1980).

The best use of these systems is in hot dry climates where direct as well as indirect evaporative coolers are used in providing comfort in the hot and dry climate of Phoenix, Arizona. (Gamero-Abarca and Yellot, 1983). The highest indoor temperatures achieved were 28.9°C in July and 29.4°C in August indicating that there is a great potential for achieving comfort and energy savings.

Throughout the world there are large populations living in temperate climates which could benefit from cheaper, low - energy evaporative cooling systems (Schofield and Deschamps, 1980). Therefore the potential for using this system and saving energy on a large scale is immense.

In Australia most of the temperate zone has a reasonably hot and dry climate and the system has equivalent application in many regions, however little practical information regarding the extent of its effectiveness is currently available. This makes it difficult for developers to use the system in their commercial developments.

The effectiveness of such systems relies on the success in meeting human comfort parameters. Air speed is crucial with evaporatively cooled air. As described in the Human Comfort section of this report (section 2.4) an air velocity of over 0.25 m/s is advisable. At this speed the upper limit of the comfort zone can be increased from 26 °C to 29° C. Ceiling fans for mixing may be used or otherwise the air supply system requires proper design to deliver air evenly through the offices at the desired speed. This adjustment in set points for the supply air temperature has an effect on the final energy efficiency of the system.

With a growing consciousness among professionals, the public and the Governments of the world about the "greenhouse effect" and the part to be played by energy efficiency in buildings, low energy systems such as evaporative coolers, whether direct or indirect or combined with desiccant dehumidification systems, are worthy of more research. The extent that thermal mass can contribute to the effectiveness of such systems and consequently the energy savings has not been quantified to date. The difficulty in segregating the components which make up the total energy savings is the main obstacle to obtaining this information.

2.7 Heating and Cooling Loads

The basic assumption with a low energy building is that the building envelope has been optimized to minimise heat loads in summer and to optimize heat loads in winter to take advantage of passive solar heat. Without a well optimized envelope the thermal inertia effect can be insignificant. Low heating and cooling loads are prerequisites for a thermal mass moderated office building as has been described in the introduction and the investigation of case studies.

Since the largest exposed element to the weather is the external wall area it is critical that the heat transfer through walls be given priority in any design process. It was shown that heat loads in buildings could be 21% to 43% less than calculated when simply using 'R' values for materials in determining heat transfer (Van Geem, 1987). The reason for this identified reduction in laboratory experiments was the thermal lag caused by thermal mass incorporated in the external wall. The experiments showed that insulation with mass added can be more effective in reducing heat flows than by insulation alone.

The following characteristics are suggested for wall construction when reducing heat transfer using thermal mass (Dexter, 1980):

1. Thermal mass can be used to delay 60% of the total annual wall heat load until later in the day.
2. Wall insulation should be outside the mass in buildings conditioned 24 hours a day.

Since engineers use semi dynamic methods to determine cooling loads, the heat loss/ gain of the internal wall surfaces are not usually calculated. However for a building utilising the effects of structural elements on the effective temperature, it is important to know the rate of heat flow at the surface of an exposed structural element. (Clarke, 1977).

There are other considerations that effect the heat transfer of the inside surface of an external wall. Thermal bridging was seen as a particular feature to be avoided in low energy designs. Heat losses can be higher by 10% to 21% due to this effect (Fang et al, 1984).

Optimization of the envelope of a building by varying the thickness, conductivity and density of a material is clearly understood. Numerous computer simulation programs can accurately predict the heat flows e.g.: ESP, (Clarke, 1977), DOE-2, (Curtis, 1981), CHEETAH, (Delsante,1985). Also, ASHRAE standards have enabled engineers to generate design charts recommending areas of glass, types of overhangs, and recommended U-values. Heat load analysis systems are well developed and reliable guidelines have been produced by engineers.

What is not readily predictable is the rate of energy flow from the thermal mass that can be used to reduce this heat/cooling load for passive/active solar buildings. Information of this type could change the direction and the type of mechanical plant and controls installed. Also it could reduce the peak load demand criteria for sizing the plant.

2.7.1 Guide-lines for Optimization of Building Envelopes

Engineers in temperate climates have produced collectively and individually for their own in-house use, guideline data in optimization of building envelopes. In housing in Australia large volumes of work have been sponsored by G.M.I. Council (GMI, 1984). Commercial buildings, apart from the creation of theoretical models, have not been an area of intense investigation, to create user friendly data to date. Such investigations are complex. However, in Perth, a consulting firm, Lincolne Scott prepared information on the Western Australian Water Authority building (completed in 1979) and published data and guidelines at an international Solar Energy Society meeting held in Perth (Lincolne Scott, 1979). Typical results are summarised in Table 11 below.

/

OTTV * U-value walls U value roofs ceiling	Lincolne Scott recommended : 99 W/m^2K *actual : 46.6 W/m^2 K* recommended : 2.1 W/m^2 K (for 3 storeys & under) 2.6 W/m^2 K (for 3 to 5 storeys) *actual :1.83 W/m^2 K* recommended: 0.57W/m^2 K *actual : 0.4 W/m^2 K* *Overall Thermal Transmission Value

Table 11 Thermal Characteristics of the W A Water Authority Building (John Tonkin Water Centre)

The authors also analysed the climate to see when plain fresh air ventilation known as economy cycle, could be used and produced guidelines for both heating and cooling. However it was noted that only office hours were analyzed (8 am to 7 pm). Use of passive solar systems and use of thermal inertia/night ventilation were not analysed specifically because it was not incorporated in the design.

The CSIRO has also produced information for Australia on heat loads through computer modelling (Spencer and Anson, 1973). This paper pointed out that 20 % of the initial cost of a building is attributable to an air-conditioning system installation. In addition to this, it showed that optimization of the building envelope is important to reduce capital costs in conventional buildings. With low energy solar designs it is critical.

New generation materials developed recently have given more emphasis to thermal considerations (Fang et al, 1984, Jurovics et al, 1985). The information provided though does not quantify the part that thermal mass can play in optimization of building envelopes, in a form which is easily usable by designers.

2.8 Relevant Data

Despite the wealth of information available there is little published material that actually defines completely the terms of reference and analysis in the form that is required to be useful to designers. The most simplistic and, apparently useful, is the table following (Burberry, 1970) :

Material	Thickness(mm)required for a U-Value of 1W/m^2°C	Temperature rise resulting from application of 1 kW/mm/m^2 for 1 minute
Concrete	830	0.04
Brickwork	700	0.06
Timber	120	0.68
Lightweight Concrete	250	0.20
Wood Work	83	1.40
Fibreboard	42	4.80
Expanded Polystyrene	25	96.0

Table 12 Typical Thermal Response of Some Common Materials

This data can be used to compare with theoretical models and the monitored information from a test rig. Some insight into response times such as time of inside temperature peak is also available (Balcomb, 1978b). The effect of thickness of a material on the thermal capacity has been explored also. Most mass materials such as concrete and brick for thickness of 100mm to 200mm reach their peak capacity of storage exposed. As thickness increases it has been shown that the diurnal heat capacity reduces (Balcomb, 1983). Although this information is not directly useful, it points to the fact that depth of mass, exposure to radiation and position, affect thermal storage capacity and are very important in the design of mass walls and ceilings. Many sources (DHC, 1980) give heat storage capacity of materials as follows:

Material	Volumetric Heat Capacity kJ/m^3
Concrete	2040
Brick	1340
Plasterboard	1040
Timber	900

Table 13 Thermal Capacity of some Common Materials

2.9 Summary

This chapter has explored the issues relevant to this research. Thermal mass was seen by other researchers to be a significant factor in saving energy in buildings, contributing as much as 30% in temperate climates. New buildings, which tend to be light weight, must employ thermal mass in an efficient and methodical way in order to take advantage of these savings. A review of comfort, climate, air flow rates and energy efficiency of case studies, helped to formulate the direction and parameters for the experimental work carried out in this study.

3.0 METHODOLOGY AND THEORY

The primary objective of the present research was to establish the rate of heat transfer from air to mass and vice versa with flowing air. A case for the research program was developed from the lack of information revealed by the literature study.

A secondary the objective was to develop a predictive method to calculate the energy savings due to maximization of this thermal mass/air transfer in commercial buildings.

A reliable experimental system had to be developed for laboratory testing. A test rig concept was formulated and current theory used to predict thermal transfers. Once data was collected, appropriate theory had to be applied to analyze the thermal mass contribution to the energy flows.

3.1 Development of a Brief for a Test Rig

In order to calculate the heat transfer from mass to air and vice versa, isolation of the material being studied was necessary. External influences had to be identified and quantified. The aim was to focus the heat transfer analysis at the air/mass interface with minimal disturbance from external influences to ensure that actual results could easily be compared with theoretical values . Even though a test apparatus had not been designed at this stage the environment assumed for the predictive calculations was as follows:

1. A thermally stable, possibly earth - sheltered, free from solar exposure, space in which to conduct the experiments.
2. A highly insulated chamber isolating the test panel face being measured, and the adjacent air, from heat transfer from the surrounding environment, however stable.

The first step was to identify the key requirements of the apparatus and the procedure advisable in its operation.

3.1.1 Factors Affecting the Heat Transfer

In order to quantify the contribution of energy to and from mass and air a convective component and radiative component needed to be calculated. The rate at which energy is supplied from the mass ultimately is governed by the diffusivity of energy within the material. However characteristics of the transfer are complex, being dependant on what the temperature differences are at the time of the monitoring, as well as the absorptivity and emissivity of the surrounding surfaces, and the air velocity of the air in contact with the mass.

3.1.2 Design of the Monitoring Program

Prior to embarking on the brief for the monitoring device the parameters needed to be established before the physical characteristics of the testing device could be finalized. A step by step process of setting parameters was as follows:

1. Determination of air speed settings.
2. Location of temperature recording positions.
3. Comfort targets for supply air.
4. Final determination of the scope of monitoring .

3.1.3 Air Speed Settings Determination

From the literature search and discussions with practising mechanical engineers a series of air speed settings were decided that were established for the test rig. These are shown in Table 15 :

DAY Air speed m/sec		NIGHT Air speed m/sec	
	Summer		
	Comment		Comment
0.0	Stagnant after n.v#	0.27	Slow moving nv
0.02	Slow after nv	0.36	Medium speed nv
0.04	Economy cycle	0.45	Fast moving nv
0.06			
0.09			
0.15			
0.21	Indirect evap cool		
0.28			
	Winter		
0.0	Stagnant after heat	0.0	Shutdown
0.02	Solar air collector		
0.04			
0.06			
0.09			
	Autumn /Spring		
0.0	Stagnant after nv	0.0	Stagnant after day
0.02	Slow moving	0.28	Slow moving nv
0.04		0.36	Medium speed nv
0.06	Econ. cycle or solar		
0.09	Heating		

Table 14 Air Speed Settings for monitoring
Night Ventilation (nv)

Note: Day-time temperatures will be achieved using fresh solar air collector heated air. The assumption being that comfortable air temperatures will be supplied only when a thermostat activates at the set temperatures.

3.1.4 Temperature Recording positions

The positioning of data collection points had to be carefully considered to ensure that measurements accurately depicted the status of the air to mass heat transfer at any instant.
The following data collection points were selected:

1. Laboratory air temperature adjacent to the rig (L)
2. Inlet air temperature to the rig (I) - fresh ambient or after conditioning of the ambient air.

3. Outlet air temperature from the rig (O).
4. Air temperature adjacent to the test panel (3 positions):
   (a) 50 mm from face of mass (t_{al})
   (b) mid point of chamber (t_{a2})
   (c) 50 mm from far side of chamber (ta3)
5. Test panel temperatures (2 positions). Inside the mass (t_{ml}) 10 mm from inside surface and inside test panel (t_{m2}), 20 mm from the outside surface.

3.1.5 Comfort Targets for Supply Air

The literature search indicated that indoor air temperatures of up to 29°C are acceptable provided air speeds are above 0.25 m/s (Burberry, 1970). The lower limit for sedentary activities with stagnant air would be about 18°C but more like 22°C in practice. Therefore the acceptable supply air temperature range - for testing purposes only - was chosen as 18°C to 30°C. for the whole year. This range was selected to be analysed in conjunction with the mass /air temperature relationships at various speeds. Because of the metabolic influence mentioned by Carruthers (1980) in a review of human comfort parameters, it was decided to use the range of 18°C to 26°C in winter and spring/autumn and a range of 20°C to 30°C in summer months.

3.1.6 Settings to Suit the Comfort Range

Selected temperature regimes, air flow rates as well as monitoring times selected before commencement are shown in Table 15 :

Season	Day/Night	Temperature	Air speed m/sec	Test times
Winter	day	18° -26°	0.04	08.00 - 12.00
			0.09	12.00 - 17.00
	night		shutdown	17.00 - 08.00
Spring /Autumn				
	day	21°	0.04	08.00 - 16.00
			0.09	08.00 - 16.00
			shutdown	16.00 - 24.00
Summer				
	day	26°	0.09	08.00 -20.00
			shutdown	20.00 - 24.00
	night	16° -22°	0.09	24.00 - 06.00
			0.18	24.00 - 06.00

Table 15 Comfort Conditions Settings

3.1.7 Humidity Influence on Comfort

Because of the lack of control over humidity for ambient air supplied buildings, there was little purpose in dictating humidity as a comfort condition with air temperature in the comfort range , apart from noting the humidity at the time of the experiments and noting any variance in mass/air temperature relationships in the experiment as a result. However, in the background study some attention was given to a new de-humidifying indirect evaporative cooling system. Such systems will improve low energy buildings by controlling humidity. This factor however was omitted from the testing due to a lack of suitable available technology. This factor should be considered in any future work (refer to section 8.0)

3.1.8 Integration of Preferred settings

Following the establishment of preferred settings for integration into a monitoring program, it was important to achieve the following objectives :

1. Finalize a design brief for a test rig and set parameters for experiments.

2. Test a sample panel to simulate air conditions for the various seasons that would be experienced in a solar designed building.

3. Produce data that would allow average rates of heat transfer from moving air to and from mass, to be predicted accurately for various conditions.

3.2 Elements for Testing and Parameters

To restrict the number of permutations and combinations, it was decided to selectively monitor the effects of air temperature against one concrete wall panel, with the final monitoring program as a guide.

Concrete was chosen because it is the predominant mass material used in commercial buildings in temperate climates (Birrer,1983).

3.2.1 Test Panel

A one - square - metre panel of 150 mm thick concrete vertically placed was chosen for the series of tests. The area of wall is the major exposed surface in a building and therefore the most important influence in the thermal inertia behaviour.
The concrete selected for the tests was a 2300kg/m^3 panel , produced from a 20MPa, 80mm slump, 20 mm diameter diorite aggregate mix.

3.2.2 Other Materials

The rig was to be designed to take floor and ceiling elements in addition to wall units for testing in isolation. It was also to be designed to take other common building materials. With future new - generation buildings a variety of materials may be used (Refer to Appendix 7)

3.2.3 Seasonal Conditions for Analysis

To predict the thermal behaviour of the building elements, it was necessary to categorize the climatic conditions within which low energy buildings in temperate climates have to operate. The main objective in setting the parameters was the use of ambient air conditions to the best advantage.
An analysis of Australian temperate climates in section 2.0 helped establish a set of temperature ranges and humidities. These parameters therefore were investigated and set as shown in Table 16 (Australian Bureau of Meteorology 1975, 1980) :

Season	Ambient mean air conditions temperature	humidity
Summer		
Day	21°- 36.6°	20% - 70%
	e.g. Flinders Is Alice Springs	
	Note: 20% relates to higher temperatures only	
Night	9°- 22.2 °	30% - 80%
Winter		
Day	6°- 21°	30% - 80%
	Note: 80% relates to higher temperatures only	
Spring/ Autumn		
Day	14.9°- 24°	44 - 80%
	e.g. Hobart to Brisbane	
Night	6.2°-12.7°	44- 80%

Table 16 Ambient Air Conditions for Testing

3.2.4 Target Temperatures for Testing

To reduce the amount of monitoring and calculation down to a manageable and yet still useful level and give a wide sample of results it was decided to monitor on days which came close to typical days as tabulated.

The rationale employed was that a range of days were to be selected which were considered cold, cool, warm and hot during the relevant seasonal period.

Table 17 (overleaf) shows the planned ambient air test conditions with targetted typical days for testing.

	Closest condition expected or forecast day max.	RH % as forecast	Preferred time for monitoring
Summer DAY	21°	20 - 70%	2 -4 pm
	25°	"	"
	30°	"	"
	37°	"	"
Summer NIGHT	(Min.)	30 - 80%	10pm - 4am
	9°	"	"
	13°	"	"
	18°	"	"
	22°	"	"
Winter DAY (For solar collection)	6°	30-80%	8am - 12 am
	10°	"	"
	17°	"	1pm - 3pm #
	21°	"	" #
Spring Autumn DAY			
	15°	44% - 80%	1pm - 4pm
	18°	"	"
	21°	"	"
	24°	"	"
NIGHT			
	6° ##	44% - 80%	8pm - 9pm
	8° ##	"	"
	10° ##	"	"
	13° ##	"	8pm - 12pm

(# or other times: e.g. night, to simulate actual temperatures encountered)
(##only if the day time was 24° or higher)

Table 17 Planned ambient air test conditions (targeted typical days for testing)

3.3 Chosen Monitoring Periods

Monitoring periods chosen encompassed the three types of seasons, for the following dates in 1990 - 1991 :

Winter	14 July to 31 August
Spring/Autumn	1 September to 30 November
Summer	1 December to 28 February

3.4 Prediction of Results from an Experimental Model

Constants and variables which were likely to effect the experiments and the way the heat transfer would occur in buildings had to be identified as the first step in the research, and analysed from first principles.

3.4.1 Variables

Definition of the variables and their ranges and whether they could be controlled practically and economically in laboratory conditions had to be scrutinized. The variables that could be controlled in the experiments were :

1. Air supply temperature
2. Air supply speed
3. Seasonally expected relative humidity
4. Seasonally expected outside air temperature

The variables that could not be controlled in the experiments were :

1. Air temperature inside the laboratory (L). However temperature variations could be minimized by location in a stable environment.
2. Actual relative humidity of the supply air at any specific time of the day or night.
3. The outside ambient environmental or air temperature and humidity at any specific time day or night on a structure.
4. Inside laboratory environmental temperature (T_{eo1})

The test rig was established in the earth sheltered basement of the School of Architecture, Building 201 at Curtin University, W.A. There was only southern glazing which was recessed and completely shaded late afternoon in summer. It is a heavy mass, concrete and solid brick building (Appendix 1).

Since no direct solar radiation penetrated the laboratory, T_{eo1} and L were taken to be equal. The rationale was that the mean radiant temperature of the laboratory would be very close to the air temperature inside the laboratory and the effect of the heat loss or gain from the rig was minimal because of the use of insulation. Therefore the subtle difference between air and mass temperature in the laboratory would be insignificant to the results of the experiments.

3.4.2 The Constants

Certain characteristics would remain constant throughout each experiment :
1. Heat storage capacity for each element (E).
2. The thermal conductivity (k) of each element in the experiments.

3.4.3 Experimental Objectives

Prior to conducting the experiments it was useful to frame the basic objectives. These were;

1. For a standard building element, to compare the thermal transfer of energy of the mass to the air with the laboratory readings at various air speeds (including stagnant air), temperatures and humidities.
2. To calculate the temperature gradient of the thermal mass across the cross sectional area of the chosen building element and compare with monitored results under various energy flow conditions in the test tunnel. This was to ascertain and quantify the external environmental effects on the experiments.
3. To establish the recovery time of air temperature close to the mass with regard to temperature at low air speeds or stagnant air conditions immediately after the following events :
   a. Night ventilation cycle periods.
   b. Solar air collection cycle periods.
   c. "Dricon" air-conditioned air cooling cycle periods.
   d. Economy cycle periods using ambient air for either heating or cooling.

3.4.4 Initial Theory and Calculations

To predict the behaviour of the material to bc tested it was decided to take the key temperatures and airflow rates to establish the following :

1. Rate of transfer of energy to the air (all seasons). Initial calculations assumed static air movement to simplify decisions about the final design of the test apparatus.
2. Anticipated rate of heat gain or loss due to laboratory temperature conditions (to the back of the test panel).
3. Temperature of the mass at the designated probe positions for the test panels t_{m1} (10mm in front inside surface) t_{m2} (20mm in from outside surface).
4. Air temperatures in the test rig adjacent to the test panels at stagnant periods - after night ventilation periods, (air speed, 0.0 m/s), after assumed ventilation periods and after solar air collection periods.

The theory used to predict behaviour of the rig was simple. The rate of transfer ('d') of energy from the mass to the air is given by Hairi and Ward,(1988) as:

$$d = [UA\ (t_{m1} - t_{a1}) - s - g\ ]\ \ (W/m^2) \qquad (EQ\ 3 - 1)$$

Where:
d = transfer of energy from mass to air
U = calculated 'U' value for each test panel ($W/m^2.K$)
A = area (m^2)
t_{m1} = temperature near the surface of the mass (° C)
t_{a1} = air temperature close to the mass area (° C)
s = instantaneous solar gains (W)
g= instantaneous internal gains (which were not used during stagnant periods)(W)

The menu of building elements chosen for initial calculations was as follows;

1. Concrete 150 mm thick panel (2040 kJ/m^3K thermal capacity)
2. Brick Panel 110mm thick (1600 kJ/m^3K thermal capacity).
3. "Styroplast" partition 40mm thick.
4. Light weight concrete 150 mm thick panel (281 kJ/m^3K thermal capacity).

To predict a range of results it was decided to calculate the heat flow (W/m^2) expected under the basic conditions outlined as possible in temperate conditions for all seasons, and for stagnant air periods.

For the purpose of gauging the possible heat flows in the rig, it was decided to look at a number of temperature settings. From the investigation of temperate climate reported in section 2.0 and the subsequent investigation in this section, the following target settings were identified as conditions worthy of trying to simulate in the experiments:

Summer	t_m	*11°, 15°, 20°, 24°
	t_a	20°, 23° 26°, 30°
Winter	t_m	6°, 10°, 17°, 21°
	t_a	18°, 21°, 24°, 26°
Spring/ Autumn	t_m	17°#, 20°, 23°, 26°#
	t_a	18°#,21°,24°,26°#

#Negate; because the air and mass temperatures are the same or similar

Note: In temperate climates in summer, spring and autumn it is known among solar design specialists that night ventilation is required to keep mass temperatures down to manageable levels.

*t_m = temperature of the mass
t_a = temperature of the air in the chamber

Table 18 Target Temperature Settings for The Experiments

Once probable temperature differences were established, the energy flows of mass to air were established as shown overleaf in Table 19.

Summer	$t_{m1} - t_{a1}$	U-value	'd' W/m2
1. Concrete (150mm panel)	-9°, -8°, -6°	3.79	34.1,303, 22.7
2. Brick (110mmpanel)	“	3.86	34.7,309, 23.1
3. Styroplast (40mm)	“	2.14	193, 17.1, 12.8
4. Light weight concrete (panel)	“	0.87	7.8, 6.9, 5.2
Winter			
1. Concrete(150mm panel)	-12°, -11°, -7°, -5°	3.79	45.5, 46.7, 26.5 18.9
2. Brick (110mmpanel)	“	3.86	46.3, 42.5, 27.0, 19.3
3. Styroplast (40mm)	“	2.14	25.7, 23.5, 14.9, 10.7
4. Light weight concrete panel	“	0.87	10.4, 9.6, 6.1, 4.4.
Spring / Autumn			
1. Concrete (150mm panel)	-1°	3.79	3.79
2. Brick(110mmpanel)	“	3.86	3.86
3. Styroplast (40mm)	“	2.14	2.14
4. Light weight concrete (panel)	“	0.87	0.87

Table 19 Mass to Air Energy Flows

3.4.5 Anticipated Rate of Heat Gain or Loss Due to the Laboratory Temperature Conditions

The initial concept for the test rig was not to insulate the test panel (exposing it to laboratory air conditions). The temperatures in the basement for estimating the predicted heat flows were as follows (Evans, 1991):

Summer: 24°C
Winter: 15°C
Autumn/Spring: 18°C
(note: some earth shelter existing)

Predicted heat flows for different seasons were calculated as follows for concrete panel, brick panel, styroplast panel and light weight concrete panel. These are shown in Table 20:

Summer					
t_{m1}	11°	15°	20°	24°	
L					24° #
L - t_{m1}	+11.3°	+9°	+4°	NIL	
HEAT GAINS					
u.(L-t_{m1})	+40.3°	+34.1°	+15.2°	-	concrete
	+50.2°	+34.7°	+15.4°	-	brick
	+27.8°	+19.3°	+8.5°	-	styroplast
	+11.3°	+7.8°	+3.5°	-	l.w. concrete
Winter					
t_{m1}	6°	10°	17°	21°	
L					15° #
L - t_{m1}	+ 9°	+5°	-2°	-6°	
HEAT FLOW					
u.(L-t_{m1})	+34.1°	+18.9°	-7.6°	-22.7°	concrete
	+34.7°	+19.3°	-7.7°	-23.2°	brick
	+19.3°	+10.7°	-4.3°	-12.9°	styroplast
	+7.8°	+4.4°	-1.7°	-5.2°	l.w. concrete
Autumn/Spring					
t_{m1}	17°	20°	23°	26°	
L					18° #
L - t_{m1}	1°	-2°	- 5°	-8°	
HEAT FLOW					
u.(L-t_{m1})	+3.79°	-7.6°	-18.9°	-30.3°	concrete
	+3.86°	-7.7°	-19.3°	-30.9°	brick
	+2.14°	-4.3°	-10.7°	-17.1°	styroplast
	+0.87°	-1.5°	-4.4°	-7.0°	l.w. concrete

Laboratory temperature

Table 20 Predicted Heat Flows (W/m^2) for Different Seasons and Materials

3.4.6 Predicted Temperature of Mass at Probe Positions at the Beginning of Monitoring Periods

Working from the temperature gradients, the temperatures were estimated, assuming the heat flows from predicted laboratory room temperatures. Resistance figures used:

	1/R	R
Concrete 150 mm	14.40	0.06
Brick 110mm	5.88	0.17
Styroplast 40mm	2.00	0.50
L.w. Concrete 150 mm	1.14	0.88

Temperatures:

Summer Table 21 (a)				
t_{m1},	11.°	15°	20°	24°
t_{m2}	22.1°	22.7°	23.4°	24°
Winter Table 21 (b)				
t_{m1}	6°	10°	17°,	21°
t_{m2}	13.7°	14.3°	15.3°	15.8°
Spring/Autumn Table 21 (c)				
t_{m1}	17°	20°	23°	26°
t_{m2}	17.9°	18.6°	18.7°	19.1°

Note: Since the t_{m1} mass temperatures are assumed to have been reached and the laboratory temperature assumed as stable, the time factor has been omitted from the predictions.

Table 21 (a, b, c) Predicted Mass Temperatures at Start of Monitoring

3.4.7 Air Temperatures in the Test Rig Chamber.

Looking at various mass temperatures at t_{m1} and assuming a period of supply air temperature t_a, and then a period of stagnation, the energy transfer figures were converted into temperature gains and losses. With the rate of heat gain or loss calculated and the assumed mass temperature at t_{m1}, it was then possible to convert the temperature build - up or loss as an instantaneous rate (°C/sec).

Assuming the volume of air being heated or cooled as $1m^3$, then the mass of this air will hold 1.012g kJ/kgK (The density of air is $1.2Kg/m^3$ at 20°C)

Therefore, $1m^3$ of air will absorb the following amount of energy;

1.012 x 1.2 KJ/°K = 1.214 KJ/°K

now: d = W/m2 = J/s/m^2 where: d = Rate of energy flow

For an area of the panel of 1 m^2,

the rate of temperature rise (°C/Sec) is given by:

$$°C/sec = \frac{d}{1000 \times 1.214} \qquad (EQ\ 3-2)$$

a) Summer	Δt = -9°	Δt = -8°	Δt = -6°	
Concrete	0.028	0.024	0.018	
Brick	0.028	0.025	0.019	
Styroplast	0.015	0.014	0.0011	
L/w concrete	0.0064	0.0056	0.0043	
b) Winter	Δt = -12°	Δt = -11°	Δt = -7°	Δt = -5°
Concrete	0.037	0.038	0.022	0.016
Brick	0.038	0.035	0.022	0.016
Styroplast	0.02	0.019	0.012	0.0088
L/w concrete	0.0086	0.0079	0.0050	0.0036
c) Autumn Spring			Δt = -1°	
Concrete			0.031	
Brick			0.032	
Styroplast			0.0018	
L/w concrete			0.00072	

Table 22 (a, b & c) Calculated Rate of Temperature Rises or Falls °C/sec

3.4.8 Conclusions Prior to Commencing a Monitoring Programme

The time lag of the external influences obviously has an effect on the results after a period.

Just as the flow of energy through to the air was predicted then the energy flow relating to flow (°C/sec) at the outside surface and the flow through the thickness of the test panel could be calculated .

The heat flows due to laboratory conditions were combined with the heat flows to and from the test panel surface (t_{m1}) due to the test chamber conditions. It was obviously the nett effect of the heat flows which would interfere with the results of the experiments. Thus tallying the heat flow on the outside of the panel due to the laboratory temperature with the heat flow to the inside of the test rig :

Heat loss due to lab. temperature - $[U (L - t_{m1})]$

+ Heat flow due to air temperature experiment $[+ (U (t_{m1} - t_{a1})]$

$= [-U (L - t_{m1})] + [U (t_{m1} - t_{a1})]$

$= - U [(L - t_{m1}) - (t_{m1} - t_{a1})]$ *for each element tested in* W/m^2. *(EQ 3 - 3)*

The convention for heat flows associated with the test panels is shown in Figure 1, Table 23 is derived from this convention.

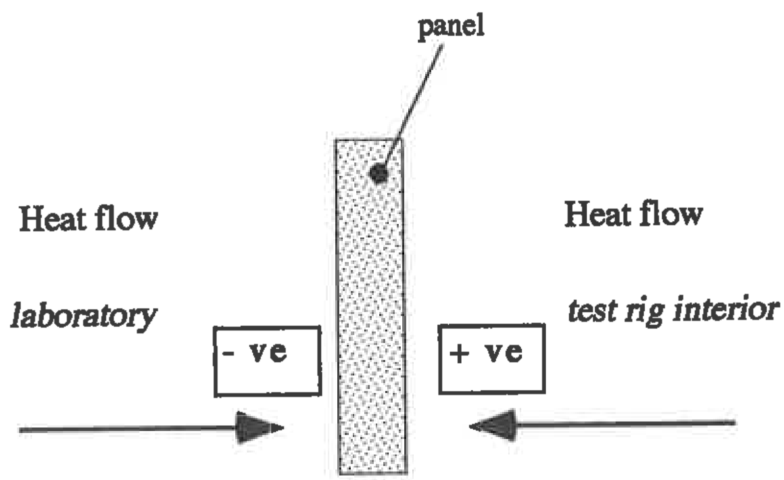


Figure 1 Convention for heat flows associated with the test panels.

	t_{m1}:t_{a1}/ L:t_{m1} Condition 1	t_{m1}:t_{a1}/ L:t_{m1} Condition 2	t_{m1}:t_{a1}/ L:t_{m1} Condition 3	t_{m1}:t_{a1}/ L:t_{m1} Condition 4
Summer	-9° / +13°	-8° / +9°	-6° / +4°	n/a*
Concrete	34.1 - 49.3 = -15.2	30.3 - 34 = -3.8	22.7 - 15.2 = 7.5	n/a
Brick	34.7 - 50.2 = - 15.5	30.9 - 34.7 = -3.8	23 - 15.4 = 7.7	n/a
Styrene	19.3 - 27.82 = -8.52	17.1 - 19.3 = -2.2	12.8 - 8.5 = 4.3	n/a
l/w Conc	7.8 - 11.3 = -3.5	6.9 - 7.8 = -0.9	5.2 - 3.5 = 1.7	n/a
Winter	-12° / +9°	-11° / +5°	-7° / -2°	-5° / -6°
Concrete	45.5 - 34.1 = 11.4	46.7 - 18.95 = 27.75	26.5 +7.6 = 41.6	18.9 + 22.7 = 41.6
Brick	46.3 - 34.3 = 11.4	42.5 - 19.3 = 23.2	27.0 + 7.7 = 34.7	19.3 + 23.2 = 42.5
Styrene	25.7 - 19.3 = 6.4	23.5 - 10.7 = 12.8	14.9 + 4.3 = 19.2	10.7 + 12.9 = 23.6
l/w Conc	10.4 - 7.8 = 2.6	9.6 - 4.4 = 5.2	6.1 + 1.7 = 7.8	4.4 + 5.2 = 9.6
Spring/ Autumn	-1° / +1°	-1° / -2°	-1° / -5°	-1° / -8°
Concrete	3.79 - 3.79 = 0	3.79 + 7.6 = 11.39	3.79 + 18.9 = 22.7	3.79 + 30.3 + 34.09
Brick	3.86 - 3.86 = 0	3.86 + 7.7 = 11.56	3.86 + 19.3 = 23.2	3.86 + 30.9 = 34.74
Styrene	2.14 - 2.14 = 0	2.14 + 4.7 = 6.84	2.14 + 10.7 = 12.84	2.14 + 17.1 = 19.24
l/w Conc	0.87 -0.87 = 0	0.87 + 1.5 = 3.64	0.87 + 4.4 = 5.27	0.87 + 7.0 = 7.87

* n/a since t_{a1} = L

Table 23 Predicted heat flows for each side of Test Panel in (W/m^2)

It was observed that the rate of heat flow from the rear of the test panel should be eliminated as far as possible. The loss or gain of heat from the experiment of up to 42.5 W/m^2 was considered too excessive. For instance, for brick, this is the equivalent of a temperature transfer of 0.035°C/second or 2.1°C/minute. This was unacceptable.

The main consequence of these figures was the need to add insulation to the rear of the panel. 150mm of polystyrene was therefore to be added to the rear of the test rig panels prior to experimentation proceedings to minimise heat loss gain from this source.

3.5 Relevant Formulae for Use in the Analysis of Data

In investigating the rates of heat transfer sampled from the data a number of formulae were needed:

1. Forced convective transfer from air to mass:

$h' = 0.99 + 0.2v$ for $v = < 5m/s$ (EQ 3 - 4)

where, h'= forced convection heat transfer coefficient

note : if v = 0.2 h' = 0.99 + 0.042 = 1.032 W/m^2.K

if v = 0.1 h' = 0.99 + 0.021 = 1.011 W/m^2.K

(ASHRAE,1980)

2. Radiative heat transfer air to mass:

$Q_R = \partial A_W E_g (T_g^4 - T_w^4)$ (EQ 3 - 5)

where ∂ = Boltzmann Constant = 5.67 x 10-8 W/m^2.K^4

E_g = Emissivity of the air

Aw = Area of the mass exposed to the air (m^2)
T_g = Temperature of the air (°C)
T_w = Temperature of the mass (°C) and:
E_g = 0.19 for 50% R.H.
= 0.10 for 10% R.H.
= 0.22 for 75 % R.H.
(ASHRAE, 1980)

3. Rate of heat loss/gain:

$$Q_{FS} = \frac{As(\,hc + Ehr\,)\,(\,Tai - Tsi\,)}{Rsi} \quad (W) \qquad \text{(EQ 3 - 6)}$$

Q_{FS} = total rate of heat gain/loss
As = Area of mass (m^2)
hc = convective coefficient = 5.8 + 4.1v ($W/m^2.K$)
E = Emittance of the mass surface
hr = radiative coefficient = 5.7 $W/m^2.K$ for 20°C
= 4.6 $W/m^2.K$ for 0 °C (ASHRAE,1980)
Tai = Temperature of inside air film (°C)
Tsi = Temperature of mass adjacent the air film (°C)
Rsi = surface / air resistance for moving air (m^2K/W)

4. Thermal bridge loss/gain (Fang, et al , 1984):

$$Q_{TB} = As.\,1/k\,.\,\Delta t \quad (W) \qquad \text{(EQ 3 - 7)}$$

where
As = area in contact with test panel (m^2)
k = conductivity of steel (W/m.K)
Δt = temperature difference between rig and outside (°C)

5. Heat storage :

$$Q_{TS} = C\,V\,.\,\Delta t \qquad \text{(EQ 3 - 8)}$$

where
C = specific heat of the material ($kJ/m^3.K$)
V = volume (m^3)
Δt = temperature difference between rig and outside (°C)

6 Heat Transfer due to Ventilation

$$Q_{AC} = V.\,C_V.\Delta t.ACH \qquad \text{(EQ 3 - 9)}$$

where,
V = volume of the chamber (m^3)
C_V = specific heat of air (kJ/m^3K)
Δt = temperature difference between inlet and outlet air (°C)
ACH = number of air changes per hour

7 Heat balance formula:

$$Q_{TS} = Q_{FS} + Q_{TB} + Q_{AC} + Q_{R} \qquad \textit{(EQ 3 - 10)}$$

Where:

Q_{TS} = thermal storage
Q_{FS} = heat losses/gains to and from the rig
Q_{TB} = thermal bridge losses/gains
Q_{AC} = ventilation losses/gains due moving air
Q_{R} = radiation loss/gain to surrounding surfaces

From the heat balance formula it was possible to quantify the amount of energy provided by the thermal mass of the test panel.

4.0 DESIGN OF THE TEST RIG

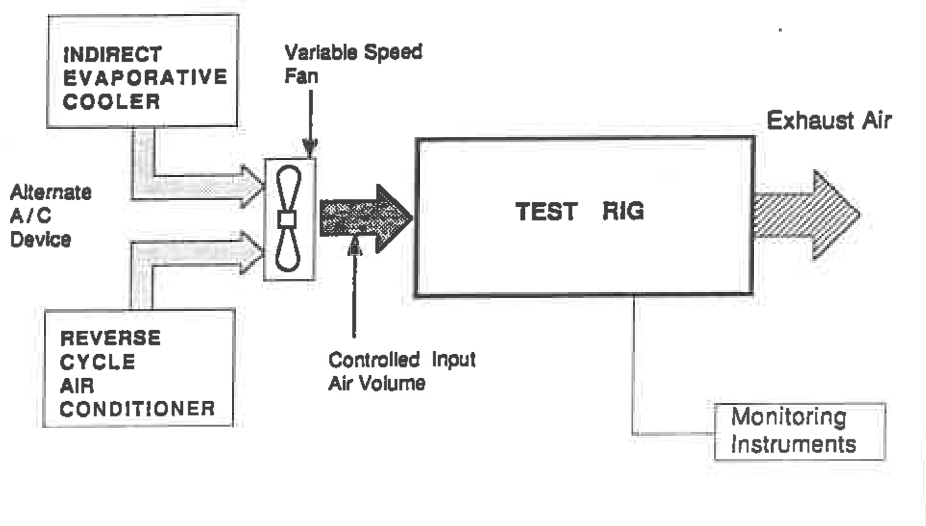
With the experimental parameters defined it was possible to design an apparatus which could generate data on heat flow for different wall floor and ceiling materials at various air flow rates and temperatures.

A design solution was needed. In the conceptual analysis it was considered, for ease of translation of results, that $1m^3$ of air against $1m^2$ of mass in an air isolated chamber would be a convenient and useful configuration. The problems of scaling, end, edge and corner effects were noted as potential problems with an apparatus of this type.

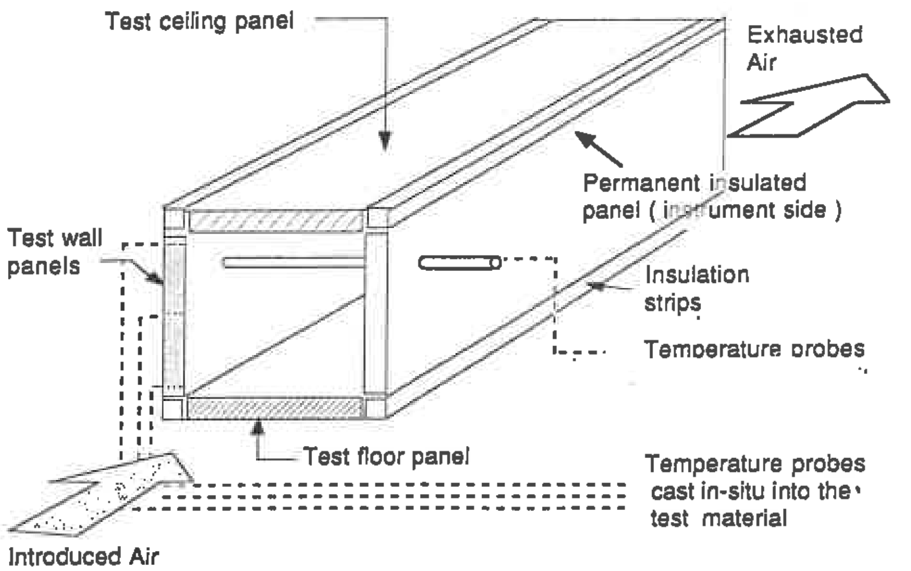
The design and construction of the test rig is shown in Figs 2 - 5 on the following pages.

The Test Rig was made up of the following components:

1. The heating/cooling equipment.
2. The air supply/control and distribution system.
3. The air chamber and support structure.
4. Air exhaust system.
5. The measuring system.
6. Ancillary information devices.



Test Rig Schematic



Test Tunnel

Figure 2 Test Rig: Schematic and Test Tunnel Diagrams

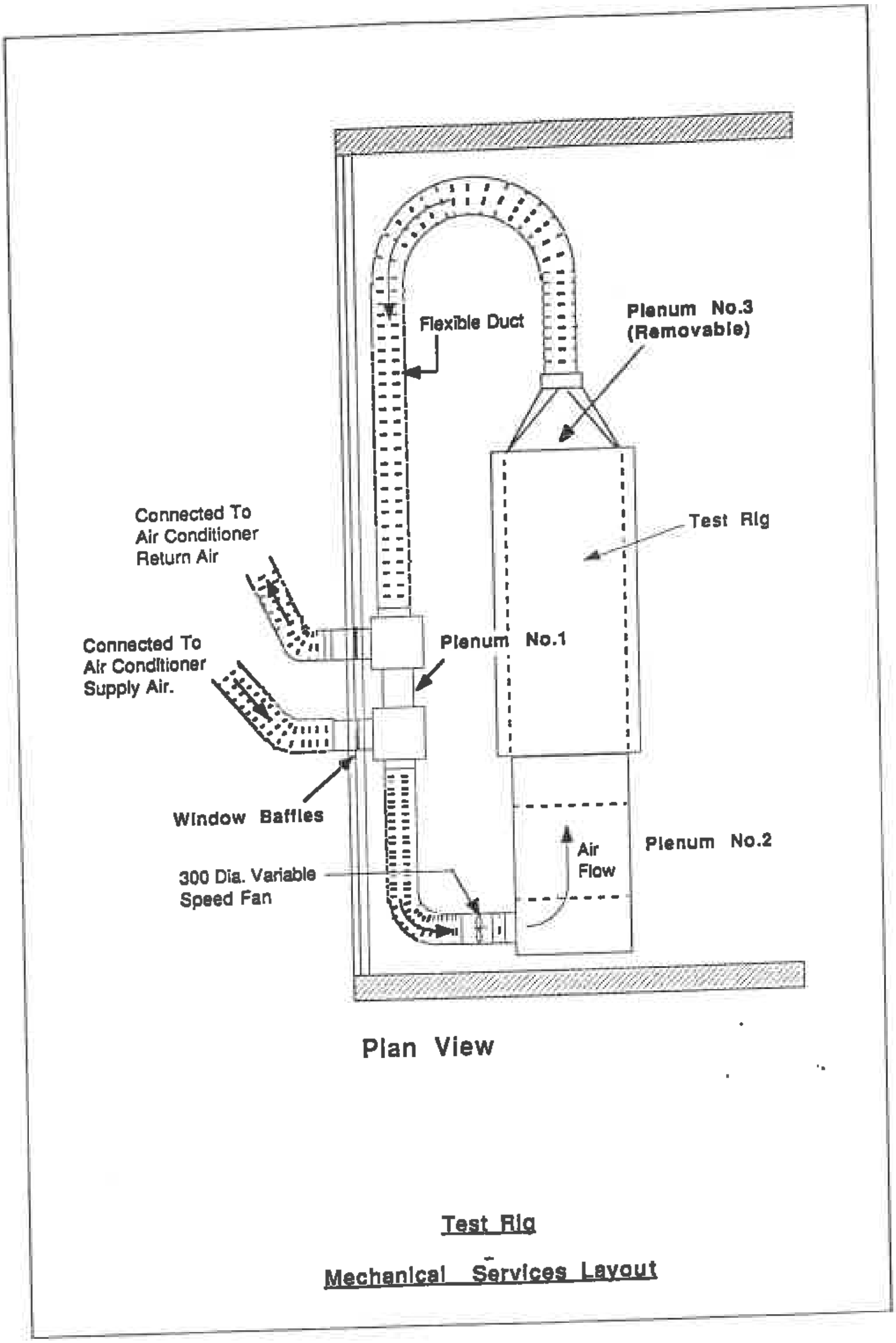


Figure 3 Test Rig : Mechanical Services Layout

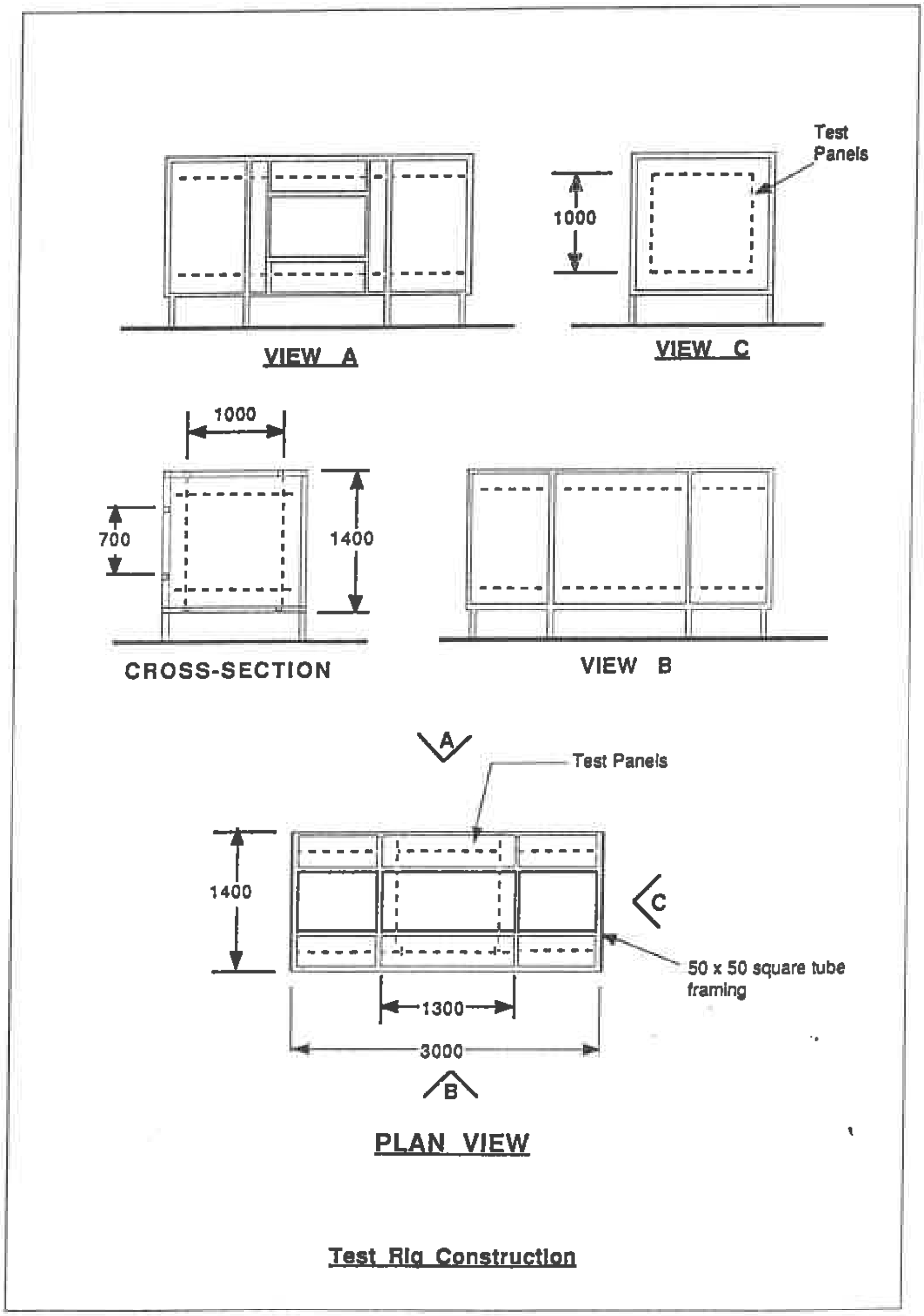


Figure 4 Test Rig: Frame Construction Details

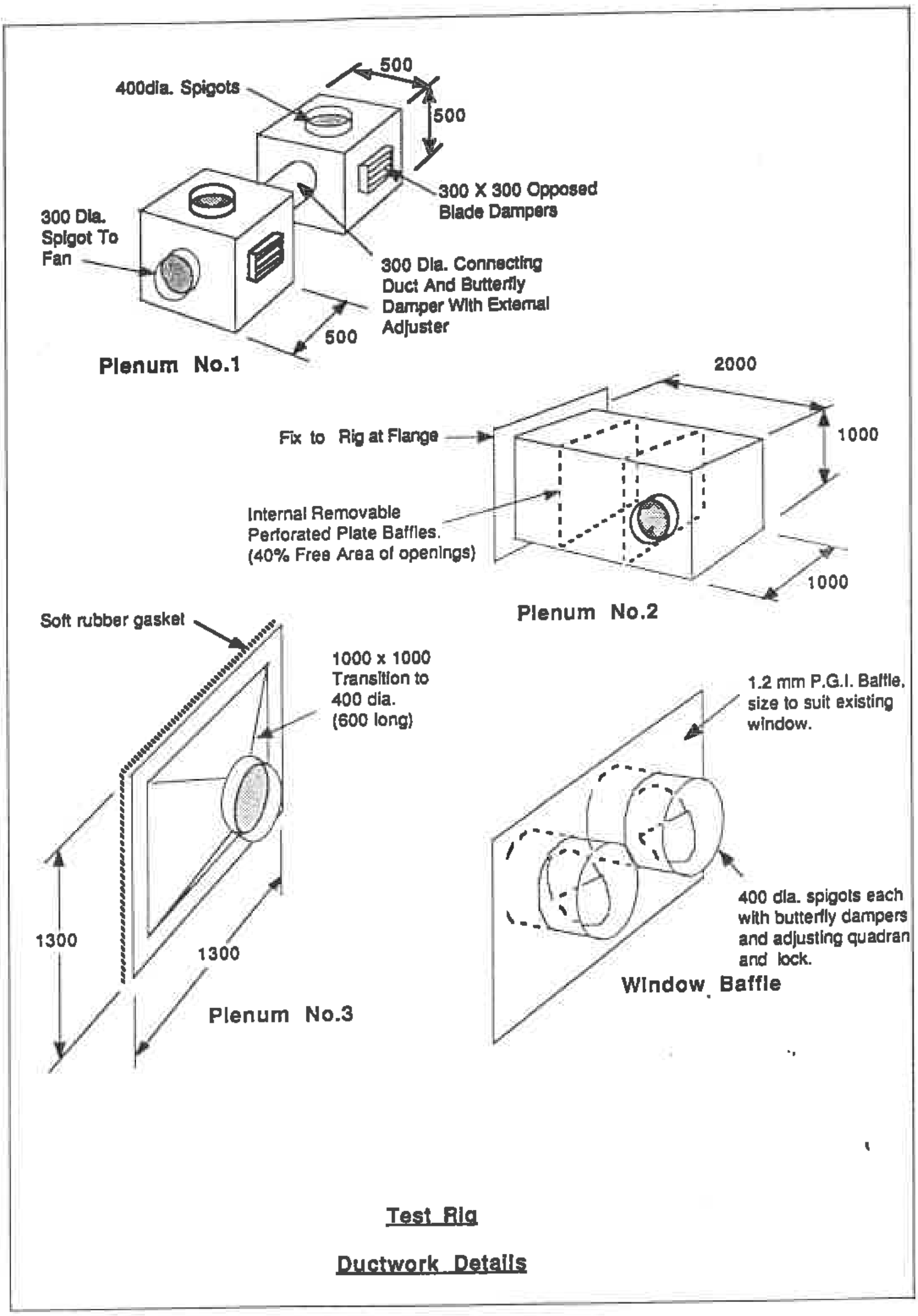


Figure 5 Test Rig: Ductwork Details

4.1 The Heating/Cooling Equipment

To add extra cooling of ambient air being supplied to the test chamber an evaporative cooler was incorporated into the apparatus. An indirect evaporative cooler known as "Dricon" (the "CHR" model) was chosen to supply straight ambient air ventilation or with indirect evaporative cooled air. In addition, a reverse cycle refrigerative air-conditioning unit was added to the system. This was for adjusting ambient air temperatures for winter and to help to handle the extreme temperatures in summer, when the "Dricon" was unable to handle all the cooling.

Plenum No 1 is shown in Figure 5 consisting of the two boxes with 400 mm diameter, supply and exhaust opening on the top, 300mm dia. supply and return air openings to the side, blade dampers in each box for "bleed' control and a connecting butterfly "mixing" damper between. Plenum No 2 is a series of perforated baffles to evenly distribute the air supply across the $1m^2$ area of the test tunnel. Plenum 3 is used to channel exhaust air out of the test tunnel via the mixing boxes and has a window baffle for connections to the supply air equipment located (in ambient air conditions) outside the laboratory.

The key to the adjustment and control of supply of the air to the chamber was the introduction of a variable speed fan prior to the distribution box (Plenum No 3). The type of fan used was a "NETA ILE" 400/4 in line centrifugal fan with a VA 2.8 speed control with single phase motor.

A 2kW air conditioning unit was incorporated as an alternative, to pump warmer or cooler air into the system. It was important to have a method of adjusting ambient temperatures to simulate a range of ambient and comfort air temperatures. The reverse cycle was chosen to simulate solar air collection temperatures for winter, given that low energy buildings in temperate climates should use this method of heating to provide economic use of energy.

4.2 The Air Supply/Control and Distribution System

Once the air was supplied at the correct volume, speed and temperature it was necessary to ensure that the correct speeds were attained.

The speed variations were obtained by a variable speed fan. The experimental system was working within the space of $1m^2$ of air moving through a chamber and ducting required was established as (400mm diam.) to flow and return.

4.3 The Air Chamber - and Support Structure

In order to avoid the effects of conduction at the edges, the test panel, was centrally placed in the 3 metre tunnel. By having the edges of the panel abutting insulation and by measuring one flat surface in the tunnel at a time, the pitfalls of measurement discrepancies associated with scaling end, edge and corner effects caused by radiant temperatures associated with scale model techniques was avoided.

To study the effects of the thermal inertia and air temperature effects during shutdown times, a system of sliding baffles which could isolate the $1m^3$ of air against the $1m^2$ of test panel was incorporated. Panels of 150mm thick polystyrene were used for the walls of the tunnel. Polystyrene was chosen because of its high emittance, excellent R-rating and to minimise the temperature effects of the laboratory conditions.

A square hollow section was chosen to form a framework to house (externally) the styrene tunnel and the baffles to create an airtight configuration. The design had to be strong enough to take the weight of the test panel which was 350 kg.

4.4 Air Exhaust System

The choice of indirect evaporative cooling made it important to ensure that the exhaust air was channelled back through the cooler to ensure the indirect cooling cycle (see Figure 3). Balanced and measurable exhaust air flow was also necessary to be established and thus be able to monitor the air changes in the test tunnel.

Air changes were set between 0.06 to 25 ACH. Because of the isolation and relatively small areas of the test panels it was very important to accurately measure these air exchange rates, to ensure precise results.

4.5 The Measuring System

Nine sensors were necessary to measure temperature. (Figure 2 and Table 24 below). Data was logged using a Unidata Starlog portable data logger. This was used in conjunction with a microcomputer.

In order to calculate energy flows from the mass to the air inlet and outlet temperatures were measured, with probes placed at both ends of the test tunnel. To a lesser extent, because of the strategy of isolating the test panel from the laboratory air temperatures, a probe was included to measure the laboratory temperature close to the test.

The key temperature probes for the air close to the mass were labelled as t_{a1}, t_{a2} and t_{a3} at the mid point of the chamber, and at various distances (50mm, 500mm and 950mm) from the test panel. The other key temperatures were the mass temperatures, one 10mm from the inside surface of the test panel (tm1 installed from the laboratory side) and also 20mm from the outside surface (t_{m2}, on the laboratory side).

Semi conductor discrete measuring devices (AD590 analog device) were used for temperature recordings. Thermocouples were considered but rejected because of cold junction problems and accuracy in relation to the data logger selected. Table 24 summarizes the position and code references for the probes:

Probe No.	Position /Description	Position Code
0	Cold Point	CP
1	Inlet	I
2	Outlet	O
3	Air temp. close to mass(50mm)	t_{a1}
4	Air temp. mid point (500mm)	t_{a2}
5	Air temp away from mass (950 mm)	t_{a3}
6	Thermal mass inside (10mm)	t_{m1}
7	Thermal Mass outside(20mm)	t_{m2}
8	Ambient air temp.	AA
9	Laboratory temp.	L

Table 24 Temperature Probe Positions

4.6 Ancillary Information Devices

A probe thermometer was used to measure inlet temperatures in order to check conditions prior to logging. In addition to this it was important to measure the air flow from the distribution plenum into the tunnel using an electronic air flow manometer.

4.7 Construction of the Test Facility

The work was divided into sections :

1. Manufacture of the steel frame and erection.
2. Manufacture of the plenums and baffles.
3. Purchase and installation of air handling equipment.
4. Installation of the external enclosure.
5. Installation of the flexible ducting.

Appendix 1, shows the progress of the construction of the test rig, from the frame to the completed rig.

4.8 Commissioning of the Rig

The Dricon CH3 unit (water pump) reverse cycle air-conditioning unit, and the variable speed supply air and controller were connected to the electricity supply. The water pump required connection to a removable water supply for use during indirect evaporative cooling cycles.

Once this was in place the eight discrete measuring devices were calibrated. The devices were placed in a refrigerator and the temperature reduced as close as possible to 0° C. The devices were then placed in an oven and raised to 100° C. The process was repeated until results were consistent.

4.9. Data Generated

Data was recorded then transferred to a personal computer and tabulated. (Appendix 3 shows recordings at five minute intervals).

The information was then analysed and graphs produced for the following parameters :

1. Ambient air and laboratory temperatures;
2. Inlet and outlet temperatures;
3. Chamber air and Mass temperatures

(Refer to Figures 6 to 8, and Appendix 2.

The graphs and description of the comparisons of these temperatures for each of the experimental periods and discussion of results, are shown in Section 5.0.

5.0 MONITORING PROCEDURE AND RESULTS

This section describes the procedures, and the conditions during monitoring. Each experiment was analysed to establish the average rates of temperature rise or fall for the chosen periods. By tabulating the average results, interim findings were established. This allowed an impression of initial patterns to be formed and highlighted errors or discrepancies, or incorrect operating procedures during monitoring .

Once an initial pattern was established, a more detailed analysis of the data was done. The heat transfer was divided into the components that effect the flows between mass and air. Current theory was utilized to analyse the data (refer to Section 3.5). The radiative and convective component, thermal bridge transfer, and test rig transmission transfer, all had an influence and needed to be quantified, so the actual thermal mass contribution could be identified.

5.1 Monitoring Periods

The day and the time period for monitoring was chosen to closely coincide with the target conditions outlined in Table 16 (section 3.0). Monitoring proceeded in eight sessions and data was recorded. This enabled a number of conditions to be simulated (refer to Table 25 and Table 26).

Monitoring Period	Year	Dates & Time	Period	Season Simulation	Air Speed m/sec
1	1990	29/10 (15:00)	to 30/10 (08:15)	Winter	0.2
2	1990	30/10 (13:15)	to 31/10 (14:20)	Winter	0.2
3	1990	9/11 (10:45)	to 10/11 (08:00)	Autumn/ Spring	0.2
4	1990	12/11 (08:15)	start	Autumn/ Spring	0.1
		13/11 (11:50)	final		0.0
5	1990	28/11(15:15)	start	Summer	0.2
		30/11(10:10)	final		0.1
6	1990	30/11 (09:55)	to 1/12 (15:00)	Summer	0.1
7	1991	22/11 (09:00)	to 23/11 (09:30)	Summer	0.2
8	1991	23/11 (09:00)	to 24/11 (09:00)	Summer	shut dow

Table 25 Summary of Experimental Conditions (times and air speeds)

Period	Ambient air	Outlet air	Inlet air	Initial air	Initial mass	Lab
1	12.0-15.2	16.3-17.0	16.2-17.2	16.8	16.4	15.5 -17.0
2	12.7-20.0	17.2-18.7	16.5-18.8	18.1	16.6	16.6 -18.5
3	15.8-24.5	18.7-23.7	17.8-23.7	18.5	17.1	18.0 - 21.0
4	8.8-20.6	16.9-19.5	13.6-21.3	17.5	19.5	15.8 - 20.8
5	14.0-26.7	15.4-24.3	15.4-24.3	22.0	19.9	18.0 - 22.0
6	15.0-27.0	16.7-26.0	16.5-26.8	24.0	19.0	18.6 - 22.7
7	14.7-29.0	16.7-26.7	16.7-26.6	23.7	19.3	19.1- 23.8
8	13.0-21.3	19.0-21.2	17.6-20.8	19.0	20.6	18.0 - 21.0
	(AA)	(O)	(I)	(ta_1)	(tm_1)	(L)

Table 26 Summary of Temperature (°C) Ranges During the Experiments

5.2 Experimental Procedure

Air was passed through the tunnel and the temperatures were recorded and tabulated. The data were then computer analysed and graphed for the following conditions:

1. Ambient air temperature vs Laboratory temperature
(AA) (L)

This was done to compare the stability of the laboratory temperature as the outside temperature varied (refer to Figure 6).

2. Test tunnel air temperature vs mass temperatures
(ta_1 close to the mass) (tm_1 : near inside chamber surface and tm_2 : near outside surface)

This indicated how the mass effected air temperature and vice versa throughout the monitoring period (refer to Figure 7).

3. Inlet temperature vs Outlet temperature
(I) (O)

This was compared to indicate if heat energy was being transferred to and from the air by the mass. (refer to Figure 8)

The graphs were useful in determining temperature differences between the laboratory (L) and the air chamber (t_{a1}), for heat flow calculations which were necessary for detailed analysis of the logged data. Figures 6, 7, and 8 indicate the results from period 1. Appendix 2 contains similar graphs of the results for periods 2 - 6.

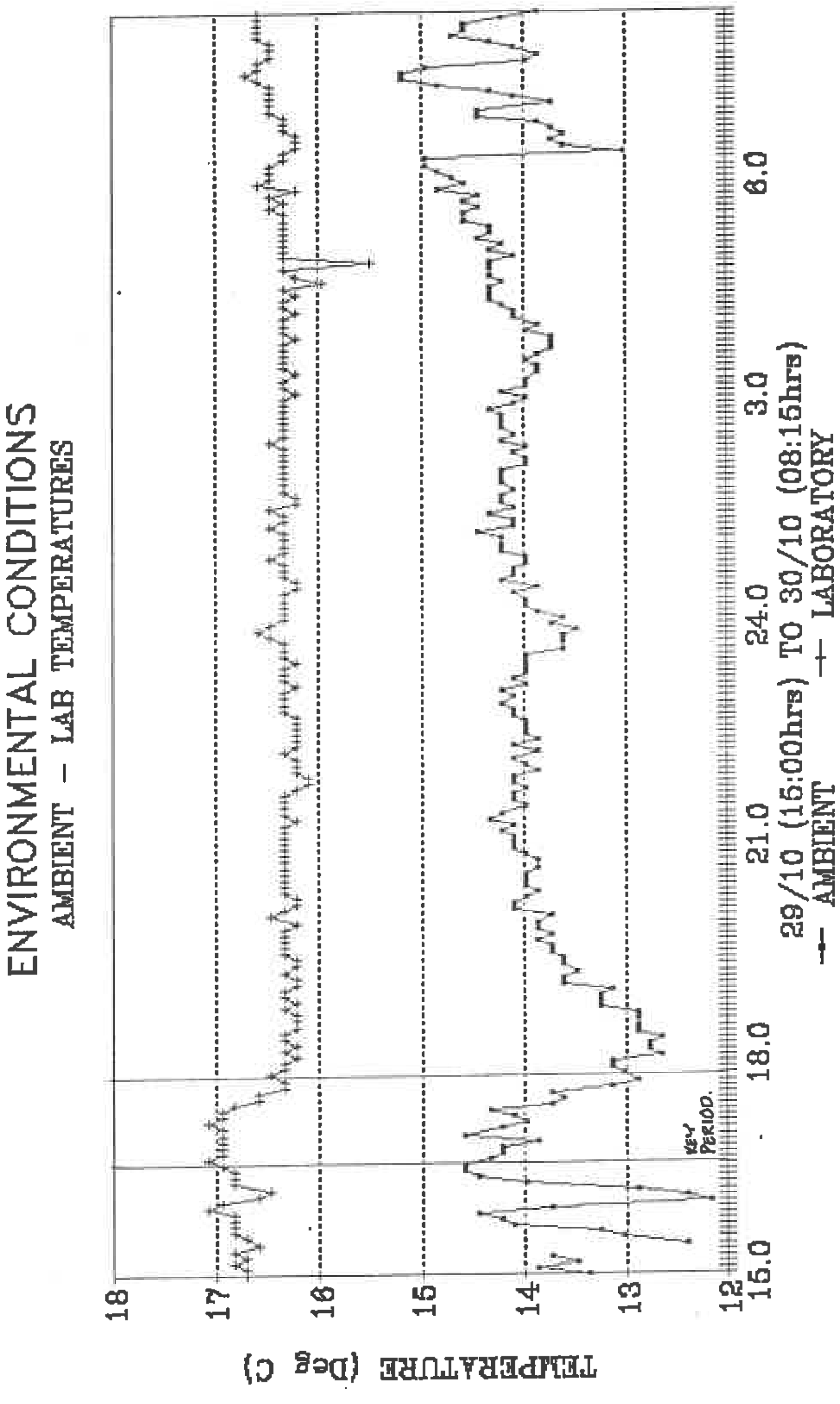


Figure 6 Ambient - Laboratory Temperatures 29/10 - 30/10

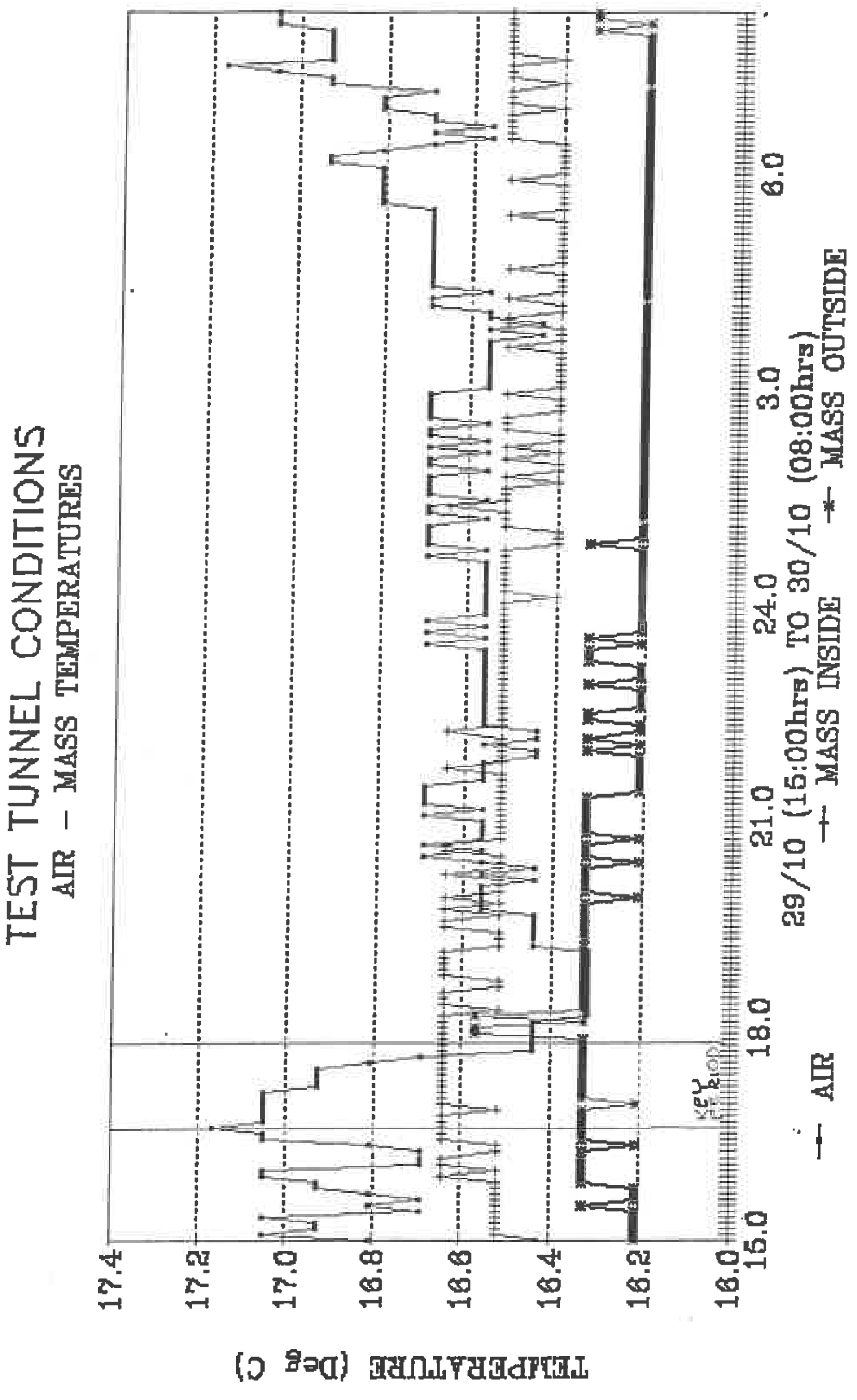


Figure 7 Air- Mass Temperatures 29/10 - 30/10

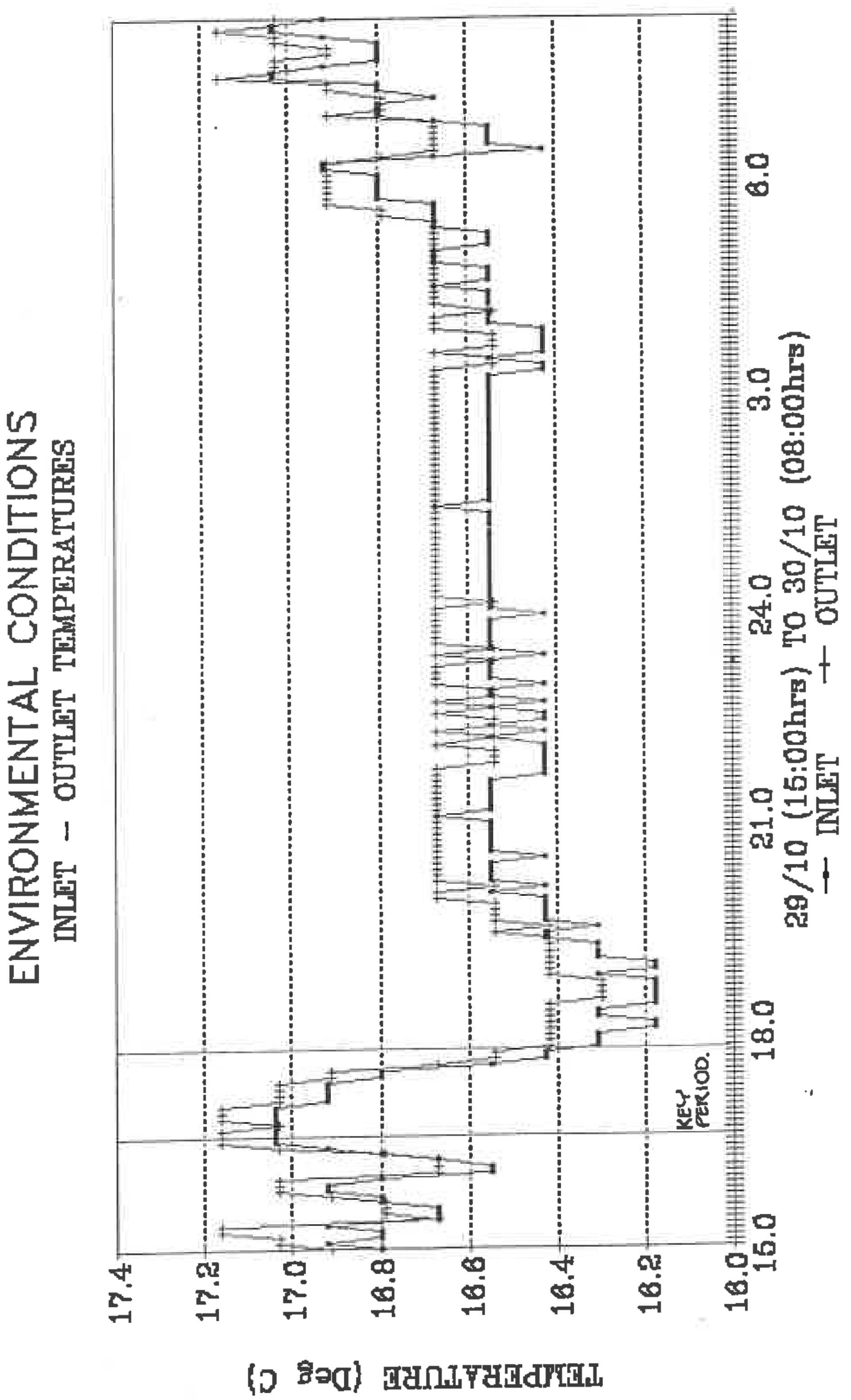


Figure 8 Inlet-Outlet Temperatures 29/10 - 30/10

5.3 Preliminary Analysis of Temperature Data

Initially, temperatures were analysed to estimate the rate of change of temperature of the mass and air in the test tunnel. Each period was investigated to ascertain a period where the rise or fall in the mass temperature was at a uniform rate. Using the graphs these key periods were readily selected. Relevant information was extracted from the data and tabulated (refer to table 27):

Period	Mass temp. range rise/fall °C	Key period of rise/fall Hours**	Av. supply (I)	Av.(ta_1-tm_1)°C temp °C
1	0.10 rise	1.25	16.90	0.25
2	0.75 rise	3.00	18.56	1.37
3	2.20 rise	5.00	23.00	4.50
4*	(a) 0.25 rise	2.00 rise	20.20	2.00 (0.1m/s)
	(b)1.50 fall	14.00 fall	-18.13	-0.04 (0.0m/s)
5	(a)1.30 rise	3.00	23.50	1.20
	(b)3.50 fall	4.00	-19.50	-1.50
6	(a) 3.00 rise	8.00	24.23	3.65
	(b) 3.00 fall	14.50	-21.00	-2.34

* omitted from further analysis due to unquantifiable ventilation load.
**(refer to graphs figure 6 to 8)

Table 27 Key Period Analysis

Periods 7 and 8 related to globe temperature measurements to evaluate the effect on the experiments. These were taken 12 months after the main monitoring was completed. Results showed that periods 1 - 6 were satisfactory for the purposes of the research and detailed analysis was not necessary for periods 7 & 8 (refer to section 5.5).

For each of the selected key periods the rate of temperature change (°C/sec) were calculated. For example for period 1 the rise in mass temperature = 0.1/1.25 °C/hour = 0.000022°C/sec. Once this rate of temperature change was established, the rate of change for the temperature difference between the mass and the air was extrapolated by dividing these figures by this average temperature difference, as follows:

$$0.000022/0.25 = 8.8 \times 10^{-5}\ °C/sec.$$

The other monitoring periods were similarly analysed for rates of change in mass temperature (tm_1). Table 28 shows rates for monitoring periods 1 - 6.

Monitoring Periods	1st	2nd	3rd	4th A	4th B	5th A	5th B	6th A	6th B
Air flow rate m/sec	0.2	0.2	0.2	0.1	0	0.2	0.2	0.1	0.1
Temp change (rise - fall)	0.10	0.75	2.20	0.25	1.5	1.3	3.50	3.00	3.00
Time period (hours)	1.25	3	5	2	14	4	8	2.5	12
Rise in mass temp (°C/sec)	2.2×10^{-5}	6.9×10^{-5}	1.2×10^{-4}	3.5×10^{-5}	3.0×10^{-5}	1.2×10^{-4}	2.43×10^{-4}	1.04×10^{-4}	5.7×10^{-5}
Mean Δt between mass/air (°C)	0.25	1.37	4.50	2.00	-0.04	1.20	-1.50	3.65	-2.34
Rate of temp. change of mass/ °C of Δt	8.8×10^{-5}	5.03×10^{-5}	2.66×10^{-5}	1.75×10^{-5}	7.50×10^{-4}	1.00×10^{-4}	1.62×10^{-4}	2.84×10^{-5}	2.45×10^{-5}

Table 28 Summary of the Rate of Mass Temperature Changes

1 Not steady flow period

2 Rate of change in-consistent

3 Mass started off 2° colder

4A Mass started over 2° colder

4B Mass started warmer initially but got colder after 30 mins., then stayed same temp

5A Steady indication of possible discrep-ancies

5B Not a period of constant air supply Reasonably steady fall of air & mass temps. Temps. started the same.

6A Mass started 2° colder

6B Mass started 0.3° warmer.

Notes on Monitoring Periods

5.4 Summary of Interim Findings

Data was reviewed progressively as each monitoring period was completed. This enabled certain procedural problems to be identified during monitoring periods 1 to 4, which are summarised as follows:

1. Mixing plenum dampers to be positioned to prevent air being introduced from the laboratory.
2. Supply air and return/exhaust air dampers to be closed or open to suit the experimental objective and ensure the correct air flow rate.

Interim recordings are shown in Table 29:

Air speed m/s	Rate of change of mass temperature °C/sec	Temperature rise/fall
0.2	3.70×10^{-5} to 1.00×10^{-4}	rise
0.2	1.62×10^{-4}	fall
0.1	1.75×10^{-5} to 2.84×10^{-5}	rise
0.0	7.50×10^{-5} *	fall

* stagnant period was for a cooling period
(note: these figures are summarised from Table 28)

Table 29 Interim Recordings

These interim findings indicated that the test rig was capable of producing data that could be used for further analysis and indicated its usefulness as a monitoring device. Further work was necessary to ensure that the data produced could lead to expanding knowledge of how air and thermal mass interact and how predictions about energy efficiency in buildings could be evaluated from the data produced.

5.5 Radiant Temperature Effects on the Experiment.

Once the initial set of recordings were examined and assessed to gauge the effectiveness of the rig, it was decided to do a test run with a globe temperature device added to the experiment, to compare with the discrete devices used for measuring air temperature. If there were discernible differences in the probe readings in the chamber, it would have meant that all experiments in the test rig should have measured globe temperature rather than air temperature and the experiments would need repeating.

Prior to the incorporation of the globe probe, it was noted that readings from the three probe positions in all experiments were extremely close and could be considered equal.

The central chamber probe was then modified to measure globe temperature. It was thus possible to ascertain the impact of globe readings adjacent to the mass material being tested both for moving air and stagnant air experiments.

The seventh and eight monitoring periods were run using a similar procedure to previous tests (1 - 6), with air moved through the rig at 0.2 m/sec from 09.00 (22.11) to 09.30 (23.11), and kept shutdown from 09.30 (23.11) to 09.00 (24.11). Refer to Appendix 2.

Investigation of the temperature readings of ta_1, ta_2, and ta_3 are shown in Appendix 3. There are no significant differences in these readings, even though ta_2 had been converted to measure globe temperature and ta_1 and ta_3 kept the same as the previous experiments. Once air movement started it was suspected that the greatest radiative effect was the transfer that occurred at the surface film interface. This needed confirmation.

5.6 Analysis of Data

Analysis of the data during selected periods of the monitoring was needed to interpret the energy flows involved in the test rig. The heat balance equation as defined by Equation 9 (EQ 3 - 9), section 3.0 was needed to quantify the unknown factor, Thermal Mass Contribution.

A heat balance calculation was carried out for each of the key periods as follows:

Monitoring Period No 1 Relevant Data
Conditions:
02. m/sec Heating of Mass over 2 hours (16.00 - 18.00)
Average temperature difference (Δt) (°C) between air and mass = 0.25C°
Average Δt between chamber and laboratory (t_{al} - L) = 0.2°C
Average Δt between Outlet and Inlet (I -O) = 0.996°C

Note: the average temperature differences were calculated from the logged data (refer to typical read out as shown in Appendix 3, Figures 6,7, and 8, and Appendix 2 for the resultant graphs)

Figure 9 shows the parameters involved in Monitoring period 1

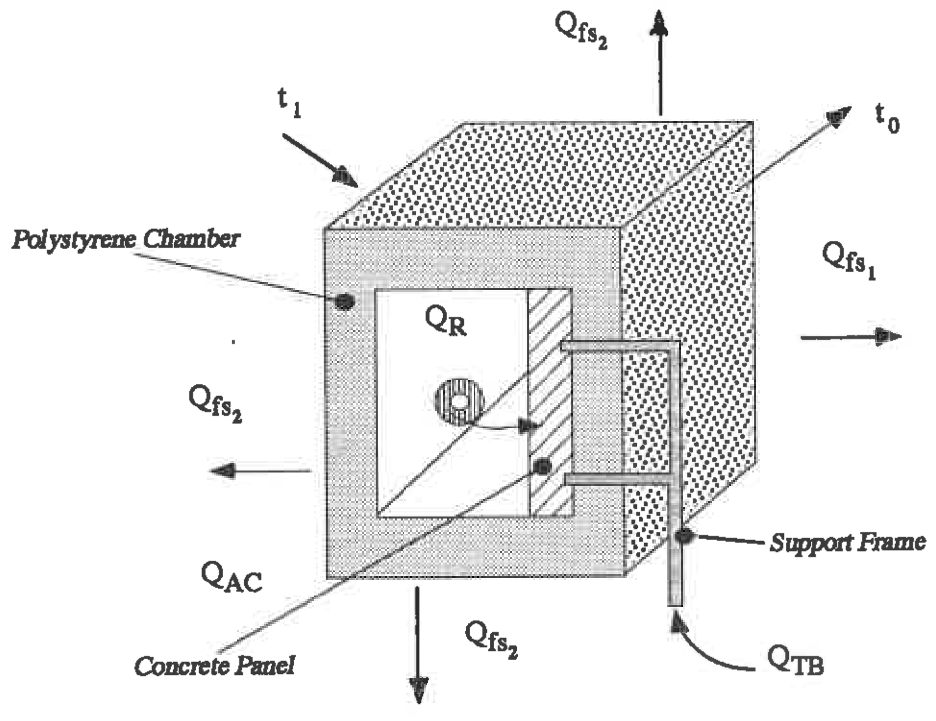


Figure 9 Monitoring Period 1 Heat Balance

1. Heat Loss/Gain from Test Rig (Q_{fs}):

Calculation of U values:

(a) CONCRETE PANEL

$$Rsi = \frac{1}{(hc + E\,hr)} = 0.085 \quad (m^2K/W)$$ (refer EQ 3 - 6)

where, Rsi = inside surface/air resistance for the concrete panel

Total Resistance (Rt):

For the concrete panel and the polystyrene insulated backing

inside air film	= 0.085	
concrete	= 0.107	
air cavity	= 0.350	(20 mm air gap test panel & insulation)
polystyrene	= 4.280	
outside air film	= 0.110	
Rt	= 4.930	

$U_{concrete}$ = 1/Rt = 0.203 W/m^2K

(b) polystyrene PANELS

Total Resistance (Rt)

inside air film	= 0.085
polystyrene	= 4.280
outside air film	= 0.110
Rt	= 4.475

$U_{polystyrene}$ = 1/Rt = 0.223 W/m^2K

$Q_{conduction} = A \cdot U \cdot (t_{a1} - L)$ (refer to EQ 3 - 6)

where,

A = Area of the test panel = 1 (m2)

U = U value of the test panel ($W/m^2.K$)

t_{a1} = internal temperature of the test rig (°C)

L = Laboratory temperature(°C)

Also the rig , $Q_{conduction} = Qfs = Q_{fs1} + 3Q_{fs2}$ *(EQ 5 - 1)*

where,

Qfs_1 = conduction from the concrete panel (W)

Qfs_2 = conduction from the polystyrene chamber (W)

Qfs = (1 x 0.203x 0.2) + 3 (1 x 0.223 x 0.2) = 0.04 + 0.13
= 0.17 W

2 Heat Loss due to Thermal Bridge (Q_{TB})

Q_{TB} = As . C . (t_{a1} - L) (refer to EQ 3 - 7) where,

C = l/k = 0.032/47 = 0.00068 ($W/m^2.K$)

l = thickness of the steel (m)

k = conductivity (m K/W)

Therefore,

Q_{TB} = 0.032 x 0.00068 x 0.2
= 0.0000002 W

which is insignificant

3. Heat Loss/Gain due to Chamber Moving Air (Q_{AC}):

$Q_{AC} = C_v \cdot v\,(I - O)$ (refer to EQ 3 - 9)

where,

C_v = Specific Heat of Air (kJ/m^{3}K)

v = velocity of the air = 200 (litres/sec)

O = air outlet temperature (°C)

I = inlet air temperature (°C)

Q_{AC} = 1.196 x 200 x 0.0996 = 23.82 W

4. Radiation Loss/Gain due to Air in the Chamber (Q_R):

$Q = \partial A_w \, Eg\,(Tg^4 - Tgw^4)$ (refer to EQ 3 - 5)

$= 5.67 \times 10^{-8} \times 0.19\,((273 + 16.93\ ^*)^4 - (273 + 16.65\ ^*)^4)$

= 0.29 W

* T_g and T_w were calculated from logged data (refer to Appendix 2 and Figures 6, 7, and 8)

The overall Heat Balance is determined by:

$Q_{TS} = Q_{FS} + Q_{TB} + Q_{AC} + Q_R$ (refer to EQ 3 - 10)

= 24.28 W

The average temperature difference between the air and the mass during the period of the experiment ($t_{a1} - t_{m1}$) = 0.25 °C, therefore; Heat Flow (H_F) over the 1 m^2 of test panel for 1° C temperature difference is 97.16 W/m^{2}K (determined by dividing by the temp. difference).

The calculations for period 1 serve as the sample calculation (Refer to Appendix 4 for the calculations for the remainder of the monitoring periods).

5.7 Summary of Results from Test Rig

A summary of the heat flow for the 6 monitoring period is shown in Table 30:

Period	Heat /Cool	Duration Hours	Air speed m/sec	Δt (t_{a1}-t_{m1}) °C	Av. rate of heat flow /1m^2 of mass (Q_{TS}) W/m^2	Heat Transfer Coefficient(H_F) W/m^{2}K
1	Heat	1.25	0.2	0.25	+24.28	+97.16
2	Heat	3.00	0.2	1.37	+27.07	+20.48
3	Heat	5.00	0.2	4.50	+16.67	+3.70
4	Heat	2.00	0.1	2.00	+8.82	+4.41
5A	Heat	4.00	0.2	1.20	+9.31	+5.82
5B	Cool	8.00	0.2	-1.50	-25.34	-16.89
6A	Heat	2.50	0.1	3.65	+14.15	+3.87
6B	Cool	12.00	0.1	-2.34	-18.84	- 8.05

Table 30 Summary Analysis of Monitoring Results

The inconsistency in the results especially for monitoring Periods 1, 2, 3 and 4 as discovered previously had procedural problems that were identified progressively as each experiment and preliminary analysis was completed. Air leakages in the rig were subsequently rectified as explained in section 5.4. Results of monitoring periods 1 to 4 were omitted from any further comparisons with case studies.

Results from periods 5A, 5B, 6A, and 6B formed the basis of comparing case study information with the test results.

From test rig data shown in Table 30 rates of heat transfer for period 5A, 5B, 6A and 6B were chosen for use in comparisons with the case study information as shown in Table 31 and Table 32:

Air speeds m/sec	Heat Flow/Unit Area (Q_{TS})W/m²*	Heat Transfer Coeff.(H_F) W/m²K *	Average Δt between air & mass °C
0.1	14.15	3.87	3.65
0.2	9.31	5.82	1.20

* the test panel was 1m² in area (these results were extracted from Table 30)

Table 31 Heating Periods Analysis

Air speeds m/sec	Heat Flow/Unit Area (Q_{TS})W/m²*	Heat Transfer Coeff.(H_F) W/m²K *	Average Δt between air & mass °C
0.1	-18.84	-8.05	-2.34
0.2	-25.34	-16.89	-1.50

* the test panel was 1m² in area (these results were extracted from Table 30)

Table 32 Cooling Periods Analysis

These results were reviewed by comparing the results with accepted surface resistances and heat transfer coefficients as follows:

Surface Air resistances m² K/W	Air speed	Heat Transfer Coefficient
0.12	for still air	8 W/m²K
0.08	for 0.5m/s	13 W/m²K
0.04	for 3 m/s	23 W/m²K
0.03	for 6 m/s	35 W/m²K

(AIRAH, 1980).
Table 33 Resistance and Transfer Coefficients

The experimental results ranged from 3.87 to 16.89 W/m^{2}K, while AIRAH figures indicated that results should have been approximately 9 to 10 W/m^{2}K for air speeds of 0.1 to 0.2 m/sec. The testing showed that there were noticeable variations to the expected rates of transfer. These variations could be due to mean temperatures being used in the calculation. Therefore, the differing temperature fluctuations and varying time periods, effected the calculated mean transfer coefficient.

In addition to this, there were other factors which were not taken into account because of the scope chosen for this study. For instance, the effect of humidity was ignored because the case study building do not control humidity as does many current state of the art low energy solar designed buildings. Also, the effect of the surface roughness was seen as a factor that could effect the rate of transfer at different air flow rates and temperature.

Further research and analysis would be needed to more accurately identify and quantify the factors determining the heat transfer rate (refer to section 8.0, Discussion).

6.0 CASE STUDY ANALYSIS

How much energy can be saved by using thermal mass to the maximum advantage?

To provide the answer, a comparison of data from the experiments with the information from a case study was necessary.

6.1 Energy Demand in a Solar Efficient Building

The Solar Energy Information Centre was chosen for comparison because of the availability of data and was a good example of a new generation solar designed building which maximised the use of thermal mass. It also had a proven energy performance.

General knowledge of energy demand of conventional buildings was useful in determining the comparative extent of savings achieved. It enabled the proportion of energy used for the various functions of the building, such as heating, cooling, lighting, and general power to be estimated.

Local practice (Healey, 1991) indicated that a typical conventional, non solar designed building in Perth, 1000 m^2 in area, would have an energy bill proportioned as shown in Table 34 :

Function	Energy Consumption kWh p.a.	% of Total Energy
Lifts	2 608	2%
House Lighting	5 218	4%
Toilet Exhaust	2 609	2%
Heating	26 091	20%
Cooling	54 791	42%
Tenants Power	39 137	30%

Table 34 Energy Demand in Office Buildings

These figures were comparable to other sources such as AIRAH (Brown and Moller, 1987).

The proportion of energy used for heating, cooling and house power in the SEIC building, which is 1000m^2, was calculated using actual energy bills.

Function	Load (kWh p.a.)
Cooling	13043
Heating	3720
Other House	4347
Tenants	1,388

Table 35 Energy Demand in the SEIC Office Building

Guidelines for use in buildings were reviewed to check how these figures compared.

As seen in the background study, the demand for cooling and space heating with an energy efficient conventional building, in Perth, should be as follows (Brown et al, 1985):

Energy demand for cooling - 216 MJ/m^2 p.a.
Energy demand for heating - 108 MJ/m^2 p.a.

This translates to the following for a 1000 m^2 building to:

Cooling 60,000 kWh p.a. Heating 30,000 kWh p.a.
Table 36 Cooling and Heating Requirements for Perth

These figures approximately compare to the break up provided by Healey. Also, according to Healey the inherent savings from using solar air collection for heating and indirect or dry cycle evaporative systems for cooling (as with the SEIC building) should amount to 70% for cooling and 80% for heating. For the 1000 m^2 SEIC building the energy bills should be (Table 37):

Cooling 18,000 kWh p.a. (70% saving) Heating 6,000 kWh p.a. (80% saving)
Table 37 Predicted Heating and Cooling for the SEIC building

It can be seen, with the SEIC, that an unidentified source was providing energy:

<u>Cooling (a)</u>:	
Expected energy load	18,000 kWh p.a.
Actual energy used	13,043 kWh p.a.
Nett energy supplied by unidentified source	4,957 kWh p.a.
<u>Heating(b)</u>:	
Expected energy load	6,000 kWh p.a.
Actual energy used	3720 kWh p.a.
Nett energy supplied by unidentified source	2280 kWh p.a.
Table 38 (a, b) Energy Analyses for the SEIC building	

This represented 27.5% for cooling and 38% for heating provided by this unidentified source.

The background study had indicated that the 30% savings were possible by using thermal mass in a temperate climate, as compared with conventional approaches (Birrer, 1983). The savings in the SEIC building
could be attributed to the thermal storage of the structure. Further investigation was needed to confirm this assumption. A heat load analysis of the SEIC building was done for comparison with the energy bill analysis.

6.2 Thermal Mass Area

The SEIC has an thermal mass effective area of 578 m^2. This area is effective because it is exposed to the internal office air volume. It is composed of the following:

Element (thickness)	* Adjusted Exposed Area (m^2)
1. 40 mm solid cement filled partitions :	40
2. 110 mm service core walls concrete bricks :	88
3. 150 mm suspended floor slab and soffit:	450
Total Effective Area :	578

* the adjusted area is the equivalent volume of 150 mm thick concrete

6.3 Comparative Air Speeds for the SEIC

Prior to calculating the heat loads and the heat balances for winter and summer it was necessary to know what the air speeds across the mass were in the SEIC building.

Determination of duct air speeds and distribution rates of the air over the thermal mass was carried out with assistance from a mechanical engineer (Healey, 1991). By relating duct air speed to the air change rate (ACH) and estimating the resultant air speed across the thermal mass, it was possible to relate air speed to ACH for the SEIC building as follows :

Supply Air Rates from fan system:

Summer :	night flush	12000 L/sec =	12 m^3/sec
	high speed	6000 L/sec =	6 m^3/sec
	low speed	4000 L/sec =	4 m^3/sec
Winter	high speed	6000 L/sec =	6 m^3/sec
	low speed	3000 L/sec =	3 m^3/sec

The ventilation rates for the building were calculated as follows:

$$ACH = \frac{\text{supply air rate x 3600 sec}}{\text{volume of the building}}$$

eg. Summer (high speed)

$$= \frac{6 \times 3600}{2700} = 8 \text{ ACH}$$

Then, duct air speeds were determined by:

$$\text{Air Speed} = \frac{\text{rate of supply from the fan}}{\text{area of the supply air duct}}$$

(eg. high speed)

$$= \frac{6}{2.25}$$

$$= 2.6 \text{ m/sec}$$

For the rate of distribution of air across the mass it was then possible to calculate:

$$\text{Air speed across mass} = \frac{\text{air speed in the duct x area of the supply register}}{\text{area of mass in the building}}$$

eg. Summer (high speed) :

$$= \frac{2.6 \times 15}{578}$$

Also, night flush :

$$= 0.067 \quad m/sec$$

$$= \frac{5.2 \times 15}{578}$$

$$= 0.134 \quad m/sec$$

Table 39 shows the air flow rates and speeds for the SEIC building;

Seasons	Winter		Summer		
Air Speed	high	low	night ventilation	high	low
Supply Air Rate m^3/sec	6	3	12	6	4
Infiltration Rate ACH	8(4)*	4(2.5)*	16	8	53
Duct Air Speed m/sec	2.6	1.3	5.2	2.6	1.7
Air Speeds over Mass m/sec	0.067** (0.1) ***	0.034** (0.025)***	0.134** (0.2) ***	0.067** (0.1) ***	0.044** -

* Average air infiltration rate due to shutdown periods.
** Actual air speeds in the SEIC
*** Preferred air speed for comparison with test results(SEIC heat transfer rates need adjustment for direct comparisons to test rig results, eg:
0.1m/sec adjustment factor = 1.033,
0.2m/sec adjustment factor = 1.066. Refer to the heat balance calculations in Section 6.0. These factors were calculated from heat transfer coefficient figures in Table 33; where the heat transfer coefficient increases $1W/m^2K$ for every 0.1 m/sec increase in airspeed for air speeds between 0 to 0.5 m/sec)
The factors were calculated as follows:
0.067 to 0.1: factor = 1 + (0.1 - 0.067) = 1.033
0.134 to 0.2: factor = 1 + (0.2 - 0.134) = 1.066

Table 39 The Heat Transfer due to Air Movement in the SEIC Building

6.4 Heating /Cooling Load Analysis of the SEIC

Typical July and January days were analysed to establish the rate of transfer of heat energy from mass to air and vice versa in the SEIC.

6.5 Heating Analysis for a Typical July Day

According to Szokolay (1982) , in Perth, the winter average temperature increase needed is:

$$\frac{3426}{31} = 110.5 \text{ Degree hours / Day}$$

Given that heating is only possible using the sun between 8am and 6pm (approximating to daylight as well as working hours) it meant that it was possible to calculate the amount of heat supplied to the building to heat the air. The following shows a heat load analysis for an average winter day :

For moving air: (refer to EQ 3 - 9)

$Q_{AH} = V.\,C_{V}.\,\Delta t.\,ACH$ (W)

where, Q_{AH}=Total rate of heating for air
V = Air volume of the buildin (m³)
Cv = Specific heat (kJ/m³K)
Δt = temperature rise needed
= degree hours/day / number of hours of heating (°C)
ACH = Air changes per hour

Now,

V = 2700 (m³)

$$\Delta t = \frac{\text{deg hours /day}}{\text{hours of heating}} = \frac{110.5}{10.9^{*}}$$

Cv = 1.196 (kJ/m³K)

* hours of heating, i.e. the number of sunshine hours

Assuming that there is 1 ACH, it can be seen that:

$$Q_{AH} = \frac{2700 \times 1.196 \times 1000 \times 110.5 \times 1}{3600 \times 10.9}$$

$$= 9093 \text{ W}$$

In winter the average number of air changes per hour (ACH) in daylight hours is between 2.5 to 4 ACH. When the maximum solar collection occurs in the building, the infiltration rate would be about 4 ACH on average during daylight hours. Abnormally cold overcast days, when daytime mean temperature is less than 17°C, results in less than an average of 2.5 ACH (Healey, 1991). Therefore, heat needed to handle ventilation loads and maintain this balance is an average between 22.7 kW and 36.3 kW for a fresh air intake of 2.5 to 4 ACH on the basis of degree-hours of heating required.

However, the average maximum heat gain, provided by the solar air heating systems in the building is limited. Experience at the SEIC has shown that an increase in air flow, above the 3 ACH on a continuous basis when combined with 1 ACH of ventilation (i.e. total 4 ACH) occurring through the air heat exchanger, has resulted in heat losses rather than gains (Healey, 1991).

Also, Table 39 shows that the rates, when tallied, closely compare with the 0.1 m/sec air flow rate used in the experiments using the test rig.

Prior to calculating the total heat load on a typical winter day, the assumptions were made based on available data, as follows:

1. The sources of heat to be quantified in thermal calculations are:

   (a) Direct solar gain through windows and skylights
   (b) Solar air collector heat raising internal temperature.
   (c) Thermal mass storage from the previous day or period.

2. Minimum Total Air Heating requirement (Q_{AH}) for the SEIC (daylight hours) for

different ventilation rates:

= 4 546 W (0.5 ACH)
= 9 093 W (1 ACH)
= 22 732 W (2.5 ACH) *
= 27 279 W (3 ACH)
= 30 007 W (3.3 ACH) **
= 31 825 W (3.5 ACH)
= 36 372 W (4 ACH)

** This is the amount of heating supplied by passive solar energy, active solar air collection and thermal mass carry over for a typical July day, to maintain a temperature equilibrium The 4 ACH rate of fresh air was adjusted to account for the heat recovery contribution by the heat exchanger (0.7 x 1ACH), i.e. 4 - 0.7 = 3.3 x 9 093 = 30 007 W.

* This is the amount of heating supplied by the total system on a day then there is intermittent air flow and shut down periods, averaging 2.5 ACH.

3. Apart from heated fresh air being introduced to the building, additional heat was contributed by the air collector and other sources. This was identified in monitored data from the SEIC building (refer to Appendix 6). It was observed that the supply air from the air heater was an average of 3°C higher than the internal air temperature, during heating hours.

6.5.1 Calculation of Heat Gains

(1) Direct Gain

Solar gains were calculated by multiplying the area of glazed element such as windows by the average solar gain heat factor for the element at that orientation (Szokolay, 1982).

Skylight	=	+ 11 277 W
Windows	=	+ 1 673 W (north/east)
		+ 990 W (north/west)
Total	=	+13 940 W

(2) Indirect Gain : Supplementary Heat from the Air Collector

The quantity of heat gained from the solar air collector was maximised at 4 ACH average over the heating hours. Data showed that the air collector could counteract the ventilation heat load as well as adding some extra heat to the building, when the ventilation rate was 4 ACH. At higher speeds it was found that the heat contribution rapidly decreased and would in fact start to cool the building instead of heating. This was because the air collector operates on a 100% fresh air system.

Apart from heating fresh in-coming air to room temperature the air collector is able to add extra heat to raise the room temperature. This extra heat was quantified as follows :

Extra heat gain, $Q_{AE} = V.\ C_v.\ \Delta t.\ ACH$ (Refer to EQ 3- 9)

where,
ACH = 3 (optimum rate through the air collector)
V = 2700 (m3)
Cv = Specific Heat of air =1.196 (kJ/m^3K)

$\Delta t = 3$ °C average temperature difference between room and air collector (as recorded by the SEIC, refer to Appendix 6).

$$= \frac{2700 \times 1.196 \times 1000 \times 3 \times 3}{3600}$$

$$= +8073 \text{ W}$$

(3) Internal Gains

People	+1800W
Lights	+624 W
Appliances	+240 W
Total	+ 2664 W

Total Gains (Q_T) = 13940 + 8073 + 2664 = + 24677 W

6.5.2 Calculation of Heat Losses

(1) Building Fabric Losses

Daylight hours : Building losses (adjusted to include night time losses stored) from the formula:

$Q_C = U.A.(t_i - t_o)$ (refer to EQ 3 - 6)

Where,

Q_C= heat loss from the building envelope (W)

U = Transmission rate (U - value) (W/m^2K)

t_i = average inside air temperature (°C)

t_o = outside air temperature (°C)

Results from the calculations are as follows:

Roof	- 3086 W
Rear Wall	- 5749 W
Spandrels	- 767 W
Total	- 9602 W

2) Air infiltration

There are two sources of air infiltration in the SEIC:

1. Forced ventilation (Q_{V1}) due to heat recovery losses through the non-solar air collector fresh air source:
2. Air Leakage through doors and openings (Q_{V2}).

(1) Forced Ventilation (Q_{V1}) :

30% x 1ACH was the chosen rate due to the maximum efficiency of the exhaust air heat recovery system:

$Q_V = V. C_v. \Delta t. ACH$ (refer to EQ 3 - 9)

Δt = temperature difference between average internal SEIC air temperature and the outside ambient temperature during heating hours.

= 3°C approximately

ACH = 0.3

Therefore,

$$Q_{V1} = \frac{2700 \times 1.196 \times 1000 \times 3 \times 0.3}{3600} = -807 \text{ W}$$

(2) Air Leakage (Q_{V2}):

0.5 ACH was assumed since the building was well sealed. This is a common assumption used in determining heat loads in buildings (AIRAH, 1980). Therefore, air infiltration through doors and openings was estimated at 50% of the 1ACH of ventilation load calculated for forced ventilation (Q_{V1}):

$$Q_{V2} = \frac{0.5 \times 2700 \times 1.196 \times 1000 \times 3}{3600} = -1345 \text{ W}$$

$$Q_V = -1345 - 807 = -2152 \text{ W}$$

$$\underline{\text{Total losses } (Q_{TL}) = -9602 - 2152 = -11754 \text{ W}}$$

6.6 Total Load

Total nett gain = 24677 - 11754 = + 12923 W.

This nett heat gain only occurs when radiation gains are sufficient and the air intake is averaged at the 2.5 ACH. If the fresh air intake increases then this heating rate would diminish. These heat loads formed the basis of a heat balance calculation (refer to section 6.8).

6.7 Cooling Load Analysis for a typical January day

The average summer temperature reduction required for Perth (Szokolay 1982) is : 19.19 degree hours /day.

The cooling for a building in Perth due to ventilation during working hours is as follows:

$$Q_{AC} = V.\, C_V.\, \Delta t.\, ACH \qquad \text{(refer to EQ 3 - 9)}$$

where,

Q_{AC} = Total cooling load during daylight working hours due to fresh air intake.

ACH = 8

$$Q_{AC} = \frac{2700 \times 1.196 \times 1000 \times -19.19 \times 8}{3600 \times 10}$$

$$Q_{AC} = -13770 \text{ W}$$

6.7.1 Heat Gains

The major source of heat gain in summer is due to air infiltration and solar gains and conductivity through the roof walls and windows. Internal gains due to lights, office appliances and equipment is a minor influence on the total heat load, but it should be accounted for in the total.

(1) Air Infiltration

$$Q_V = V.\, C_V.\, \Delta t.\, ACH \qquad \text{(refer to EQ 3- 9)}$$

where,

Q_V = air infiltration load
ACH = 0.5
Δt = average temperature difference between inside and outside air
= 3°C
Q_V = + 1345 W

(2) Conductivity and Solar Gains

Q_C = passive solar gains:
skylights = 7003 W
windows = 5986 W
building skin : roof = 874 W
spandrels = 217 W
glass = 2433 W

Total Q_C = + 16513 W

(3) Internal Loads

Q_I = + 2664 W (lights, people and computers, as previously)

6.7.2 Heat Losses

The cooling in the SEIC building is provided from 3 sources:
1. Night cooling of the mass
2. Thermal storage carry over from the previous 24 hours
3. Indirect evaporative cooling

<u>(1) Night Cooling of the Mass</u>
Cooling supplied directly to the building by the night ventilation can be calculated by :

$Q_{NV} = V.\ C_V.\ \Delta t.\ ACH$ (refer to EQ 3 - 9)
= - 14352 W
where,
ACH = 16 (at 0.134 m/sec: refer to Table 39)
Δt = 3 averaged over 8 hours (°C) (as seen from data obtained from the SEIC; refer to Appendix 6)
Also,
Q_{NV} = - 7176 W
when,
ACH = 8 (at 0.067 m/sec: refer to table 39)

<u>(2) Thermal storage carry over from the previous 24 hours</u>
This energy contribution is what is needed to be determined and is the unknown that had to be estimated from a heat balance calculation.

<u>(3) Indirect evaporative cooling</u>
Cooling by the evaporative system was designed to handle the ventilation loads, given that the thermal mass and night ventilation system was designed to offset the daily heat gains and solar load. To calculate the extra cooling provided by the evaporative cooler was difficult to determine:
Q_{EC} = Extra cooling provided by the evaporative cooler above the ventilation load = unknown (W)
(refer to cooling balance calculations in Section 6.8.2)

6.8 Heat Balance Calculations for the SEIC

A heat balance calculation was required for both summer and winter. This was needed because the thermal mass carry over, had to be calculated by solving two equations; the overall heat balance, and the proportion attributed as the unidentified source from the analysis of the energy bills for the SEIC as follows :

6.8.1 Heat Period Balance

The winter Heat Balance is given by:

$$Q_{AH} = Q_{AI} + Q_V + Q_C + Q_I + Q_{TS} + Q_{AE} + Q_{SR} \qquad \text{(EQ 6 - 2)}$$

where,

Q_{AH} = Total heating requirement of the building during daylight hours (W)

Q_{AI} = Heating by the air collector to maintain temperatures at a constant condition (W)

Q_V = Air infiltration load through doors, windows and dampers (W)

Q_C = Conduction loads (W)

Q_I = Internal loads (W)

Q_{TS} = Heating provided from thermal mass carry over (W)

Q_{AE} = Extra heat provided by the air collector over and above the ventilation load (W)

Q_{SR} = Solar radiation (W)

Now, it has already been established that the unidentified source was providing 38% of the heating over the full year (refer to section 6.1). This contribution has to be attributable to the extra heating, provided by the air collector and the thermal mass carry over.

To estimate the thermal mass contribution in isolation, the following derived formula was used:

$$Q_{TS} + Q_{AE} = 0.38\,(Q_{AI} + Q_V + Q_C + Q_I + Q_{SR}) \qquad \text{(EQ 6 - 3)}$$

From the heat load analysis for a total 4 ACH fresh air intake (3.3 ACH effective fresh air intake; refer to section 6.5) there was sufficient data to solve the unknowns and establish the contribution of the thermal mass to the daily heating requirements:

Q_{AH} = + 30007 W
Q_{AE} = +8073 W
Q_I = +2664 W
Q_V = - 2152 W
Q_C = - 9602 W
Q_{AI} = unknown variable
Q_{SR} = + 13940 W

$$Q_{TS} + 8073 = 0.38\,(Q_{AI} - 2152 - 9602 + 2664 + 13940) \qquad \text{(refer to EQ 6 - 3)}$$

$$Q_{TS} = 0.38\,(Q_{AI} - 2152 - 9602 + 2664 + 13940) - 2691$$

$$= 0.38\,(Q_{AI} + 5388) - 8073$$

$$= 0.38\,Q_{AI} - 6025$$

Therefore,

$$+30007 = Q_{AI} + (-2152) + (-9602) + 2664 + (0.38\,Q_{AI} - 6025) + 8073 + 13940$$

$$Q_{AI} = \frac{6898}{1.38} = 4998\ W$$

$$Q_{TS} = 0.38\,(+4998) - 6025 = -4125W \times 1.033^{*} = -4261\ W$$

(heat being absorbed by the cooler mass)
(*air speed adjustment for 0.1 m/sec ; refer to Table 39)

As already described, the average temperature difference between the mass and the air during heat periods was 1.5°C. Thus dividing the area of mass (578 m^2) into the heat storage energy calculated gave the rate of transfer in W/m^2 and adjusting for the temperature difference:

Winter Heating at 0.1 m/sec:

= 7.37 W/m^2 of mass area
= 4.91 W/m^2K (for Δt of 1.5 °C)

6.8.2 Cooling Period Balance

The following formula was used to establish the thermal balance:

$$Q_{AC} = Q_{EC} + Q_V + Q_C + Q_I + Q_{TS} + Q_{NV} \qquad \text{(EQ 6 - 4)}$$

From the unknown source, 27.5% of the cooling was provided. This had to be the combined effect of night ventilation cooling and thermal inertia carry over. Therefore, the unidentified source can be expressed as follows:

$$Q_{TS} + Q_{NV} = 0.275\,(Q_{EC} + Q_V + Q_C + Q_I) \qquad \text{(EQ 6 - 5)}$$

where,

Q_{AC} = Total cooling required by the total system = -13770 (W)
Q_{EC} = Extra cooling provided by the evaporative cooler above the ventilation load = unknown (W)
Q_V = Average air infiltration load through doors and windows (daylight hours) = +1345 (W)
Q_C = Heat loads due to solar load and conductivity = +16513 (W)
Q_I = Internal loads = +2664 (W)
Q_{NV} = Cooling provided by night ventilation = -14352 (W)
Q_{TS} = Thermal storage = unknown (W)

$Q_{TS} = 0.275\,(Q_{EC} + 1345 + 16513 + 2664) + 14352$ (refer to EQ 6 - 5)
$= 5644 + 14352 + 0.275\,Q_{EC}$

$= 0.275\,Q_{EC} + 19996$

Now,

$$-13770 = Q_{EC} + 1345 + 16513 + 2664 + (0.275\,Q_{EC} + 19996) - 14352$$

(refer to EQ 6 - 4)

$$QEC = \frac{-26166}{1.275} = 20522 \text{ W}$$

Q_{TS} = 0.275 QEC + 20522
= - 5644 + 20522
= 14878 W for 0. 134 m/sec
= 15859 W for 0.2 m/sec
(allowing for the adjustment factor; refer to Table 39)

= 10798 W for 0.067 m/sec
= 11154 W for 0.1 m/sec
(allowing for the adjustment factor; refer to Table 39)

Thermal Storage cooled at night = 15859 W for 578 m^2 of mass
= 27.44 W/m^2 for 3°C (Δt)
= 9.15 W/m^2K (0.2 m/sec)

Thermal Storage cooled at night = 11154 W for 578 m^2 of mass
= 19.29 W/m^2 for 3°C (Δt)
= 6.43 W/m^2K (0.1 m/sec)

7.0 CONCLUSIONS

As a result of the data collected from the test rig and the data collected and analysed from the case study information, conclusions could be made by direct comparisons.

7.1 Solar Energy Information Centre : Thermal Mass Performance.

The rate of heat transfer from thermal mass is summarized in Table 40 and Table 41. The overall performance of the SEIC building is shown in Table 42 :

Air speed m/sec	Heat Flow W/m^2	Heat Transfer Coefficient W/m^2K	Average Δt between mass & air °C
0.1	7.37	4.91	1.5

Table 40 Heating Periods Thermal Mass Heat Transfer Rate

Air speed m/sec	Heat Flow W/m^2	Heat Transfer Coefficient W/m^2K	Average Δt between mass & air °C
0.1	19.29	6.43	3
0.2	27.44	9.15	3

Table 41 Cooling Periods Thermal Mass Heat Transfer Rate

Building Area m^2	Mass to: Floor Area ratio	Contribution to Savings%	\$ m^{-2}/ p.a. saved*
1,000	0.578	heat 38%	0.52
1,000	0.578	cool 27.5%	1.14

* estimates of savings from total energy bills at the SEIC which are extremely low compared to conventional buildings (Baverstock et al,1992).

Table 42 Overall Thermal Mass Performance for SEIC Building

7.2 Test Rig versus a Built Structure

In order to make conclusions about thermal effects, results from the case study had to be compared with the test rig results (refer to the summary in section 5.0) This proved that the test rig can be reliably used in further research in modelling thermal mass characteristics of building materials, under simulated conditions. In addition, it was possible to estimate energy savings, enabling the financial advantage of using a mass material in a low energy building, to be quantified.

When the results were adjusted for temperature differences and air speeds, the SEIC estimates compared favourably with the test rig results, as can be seen in the following summary:

Information Source	Air speed m/sec	Heating Rate W/m^2 *(Δt)	Heat Transfer Coefficient W/m^{2}K *
Test Results	0.1	14.15(3.65°C)	3.87
SEIC Data	0.1	7.37 (1.5°C)	4.91

* for 1 m^2 of test panel

Table 43 Comparison of Test Results with Case Study: Heating Mass

Information Source	Air speed m/sec	Heating Rate W/m^2 *(Δt)	Heat Transfer Coefficient W/m^{2}K *
Test Results	0.1	18.84(2.34°C)	8.05
	0.2	25.34 (1.5°C)	16.89
SEIC data	0.1	19.29 (3°C)	6.43
	0.2	27.44 (3°C)	9.15

* for 1m^2 of test panel

Table 44 Comparison of Test Results with Case Study: Cooling Mass

Given the approximation of methods of calculating the heat flows, some discrepancies were expected. However other reasons could be given and these are discussed in Section 8.0.

In the process of producing the data for comparison work, a number of observations were made, which would be useful to building designers:

1. Thermal mass storage, when combined with night ventilation cooling can save 38% of air conditioning energy bills in a temperate climate such as in Perth, W.A. Also, thermal mass storage when combined with solar air collection can save 27.5% of space heating bills.
2. The average air temperature is 3°C lower than the average mass temperature during ventilation on a typical summer night. Also, during a typical winter day the solar collector can supply air at an average of 1.5°C above mass temperature.
3. The rate of heating from the surface of thermal mass is approximately 7.4 to 14.2 W/m^2 whilst the temperature difference between the air and the mass averaging 1.5 to 3.65°C. Test rig results as well as the case study indicated a heat transfer rate of 3.87 to 4.91 W/m^{2}K.
4. The rate of cooling of the surface of thermal mass is approximately 18.8 to 27.4 W/m^{2}K whilst an average temperature difference of 1.5 to 3°C is maintained during night ventilation. This translates into a heat transfer coefficient of 6.4 to 16.9 W/m^{2}K, which can be used to estimate the rate of heat extraction, and therefore help a designer establish the reduction of cooling needed the following day.
5. The energy savings can amount to \$0.84/m^{2}p.a. during cooling at 0.1m/sec and \$0.52/m^{2}p.a. for heating (based on 18c/kWh).
6. The recommended area of exposed thermal mass which is equivalent to 150mm thick structural concrete (2400 kg/m^3, 2040kJ/m^{3}K storage capacity), should be 578 m^2 for a 1000m^2 building. This should translate into a mass to floor area ratio of 0.6 or greater.

Presentation of this information could be useful to designers and is summarised and tabulated as shown in Table 45 and Table 46:

Ratio of exposed mass to building	Air speed m/sec	Average total heat transfer W/m^2	Average heat transfer coefficient W/m^2K	Average. Δt mass and air	Energy saving %	Dollar Savings $\$/m^2$.p.a area
0.6	0.1	11.41**	4.39**	2.6	27.5	0.52

Table 45 Thermal Mass Contribution to Heating

Ratio of exposed mass to building	Air speed m/sec	Average total heat transfer W/m^2	Average heat transfer coefficient W/m^2K	Average. Δt mass and air	Energy saving %	Dollar Savings $\$/m^2$.p.a area
0.6	0.1 - 0.2	18.84 to 27.44 *	6.43 to 16.89 *	3	38	0.84

Table 46 Thermal Mass Contribution to Cooling

* figures relate to the different air speeds
** figures determined by adjusting for the temperature differences and averaging the limits of the results of testing and case study.

The data generated from the research helped to quantify the air heating and cooling energy savings as a proportion of the total energy savings in a solar design approach.

The question that arises from this is; How much is saved by using thermal mass in a solar designed building compared with a conventional building? With the benefit of the case study information examined in section 2.0, an attempt at quantifying the energy savings was possible. This is discussed in the following section.

8.0 DISCUSSION

8.1 The Research Program

This study has shown that the laboratory test procedure, devised for this research, can be used with confidence to simulate the thermal mass effects in office buildings. Test rig data supports this conclusion by comparisons with a built example which utilizes thermal mass to reduce energy demand.

The process of comparing data from the rig with results from monitoring of the Solar Energy Information Centre (SEIC), by reducing heat flows to a heat transfer coefficient, proved to be a useful way of comparing results. The heat transfer coefficients are in a form which can be readily used by engineers to allow for the heating and cooling contribution during daily office hours from low energy systems in conjunction with thermal mass.

By analysing the mass to air transfer rates for air flow rates at typical speeds for heating and cooling, it was relatively simple to translate data into predicted energy savings for energy efficient buildings.

The research established a set of facts about the maximum direct benefit of using thermal mass in commercial buildings in a typical temperate climate like Perth, Western Australia. It was shown that a reduction of 27.5% of energy bills can be made in winter, as well as 38% of cooling bills in summer. This proved that the use of thermal mass in the overall design strategy is a significant factor. This conclusion was supported by previous indications as revealed in the literature search (Birrer, 1983).

Apart from the direct benefit of the energy savings due to the thermal storage factor, a central claim by others that energy efficient, solar designed buildings can save 60% to 90% of energy demand in comparison with conventional buildings, was validated.

With an energy demand of \$8 to \$9/m^2 p.a. in the SEIC building compared with accepted BOMA guidelines of \$28 to \$43/m^2 p.a. for the normal range of energy demand, along with a possibility of a \$80/m^2 p.a. for a highly inefficient building, this claim is justified.

In clarifying what energy savings can be expected from the solar design approach, the contributing factors to the possible total were segregated and quantified. This was produced from the data generated, previous case studies, and previous work in the field, a proportional analysis of these factors and the possible range of savings was prepared.

These estimates are shown in Table 47 overleaf :

Area of Savings	Estimated Energy $/m^2$ p.a.	Savings Percentage
1. Optimal Design of the Building Envelope (As determined by previous analysis of a case study, the Russell Centre Baverstock and McGeorge,1987).	4 - 16	15 - 20%
2. Direct Use of Thermal Mass (refer to section 7.1 and Birrer,1983)	4 - 24	30%
3. Use of Solar/Low Energy Plant in conjunction with the Thermal Mass (Estimated in the SEIC building).	2 - 8	6 - 10%
4. Energy Efficient Lighting and Use of Natural Light (estimated to be 30% to 50% in the SEIC building)	1 - 8	5 - 10%
5. Tenants Frugal Use of Appliances and Officc Equipment (The analysis of Markalinga House showed tenants bills of up to $7/m^2$ while the SEIC were as low as $2/m^2$).	5 - 13	6 - 17%
7. Energy Management of Common Area (conservative estimate)	1 - 2	3%
Overall Savings Possible	17 - 71	Range of 60 - 90%

Table 47 Estimated Energy Savings in Solar Designed Buildings

8.2 Suggestions for Future Work

Further work is needed in a number of areas and the test rig and the associated testing procedure could be used to generate data.

Many variables are involved in accurately predicting the rates of heat transfer that occur between thermal mass and the air in a low energy building. It was seen in section 3.0 that some factors are difficult to measure, and generally it is common practice among building scientists to not account for them in analysing heat transfer in buildings.

These other variables, apart from air speed and temperature differences were not included in this study because they were not considered a major influence on heat transfer calculations in determining heat loads on buildings. For example, the effect of surface roughness was neglected in the monitoring and analysis. Also, the effects of humidity and permeability of the test material was omitted from calculations of the heat transfers.

Owing to the discrepancies between the test results and the case study, a further development would be to measure these usually considered insignificant variables and current theory could be reviewed to include these factors in predictive calculations for thermal mass effects.

Future work should quantify these effects by the measurement of them and make comparisons with the major influences on the experiments. For example, the use of a Tally Surf, which is a device to minutely measure indentations of a surface or the use of Microscopy techniques could be used to determine the extent of surface roughness on a test material. Similarly, accurate measurement of air humidity as well as moisture content of the building material would assist in quantifying this influence on the overall heat transfer rate. Once all the factors could be integrated into a predictive model then it would be possible to produce design data, such as heat transfer coefficients for a number of materials (refer to Appendix 7 for a suggested menu).

Results of the testing demonstrated the need for further research into surface air resistances at slow air speeds. Test results indicated heat transfer coefficients which were different to maximum rates published by AIRAH (1980). Some of the rates from the research were lower than expected and this is attributed mainly to the averaging of fluctuating heat transfer rates over a period, as well as the combined effect of the usually considered insignificant effects, previously mentioned. However, the higher than expected rates indicated that surface to air heat transmission resistance and the factors which affect it requires much closer investigation in order to explain the phenomenon at air speeds of 0.1 to 0.2 m/sec.

Further work would support the development of a predictive method, by the comprehensive monitoring of a built example. Information available from case studies was useful but limited. More detail would help validate any continued testing program. For instance, it is known that it is virtually impossible to maintain an even distribution of air flow at a consistent speed over the exposed thermal mass areas in any building. More comprehensive monitoring of a built example would provide valuable data for different locations within a building, leading onto more specific information for designers to use thermal mass more effectively.

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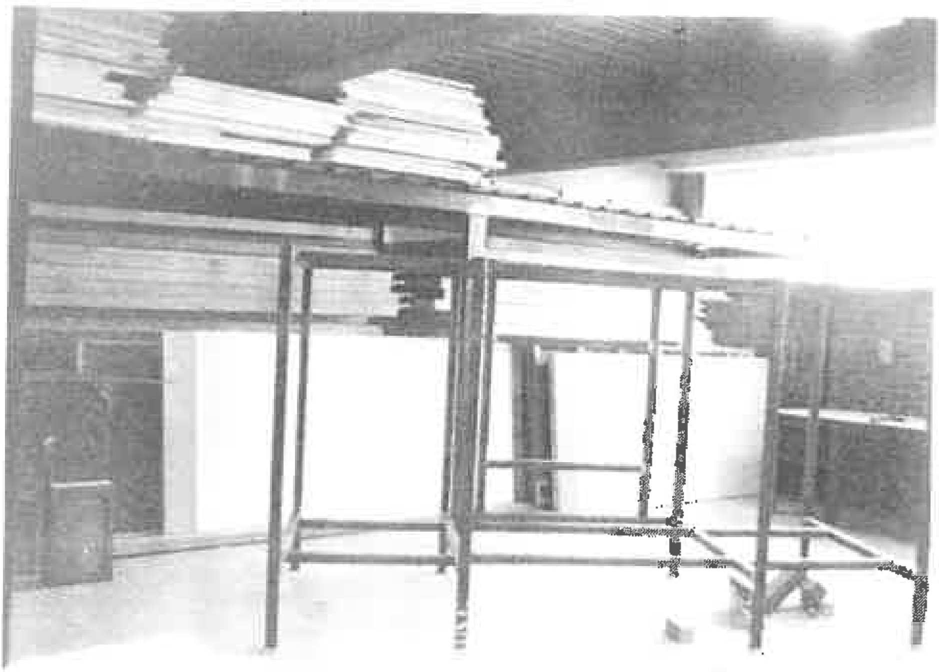
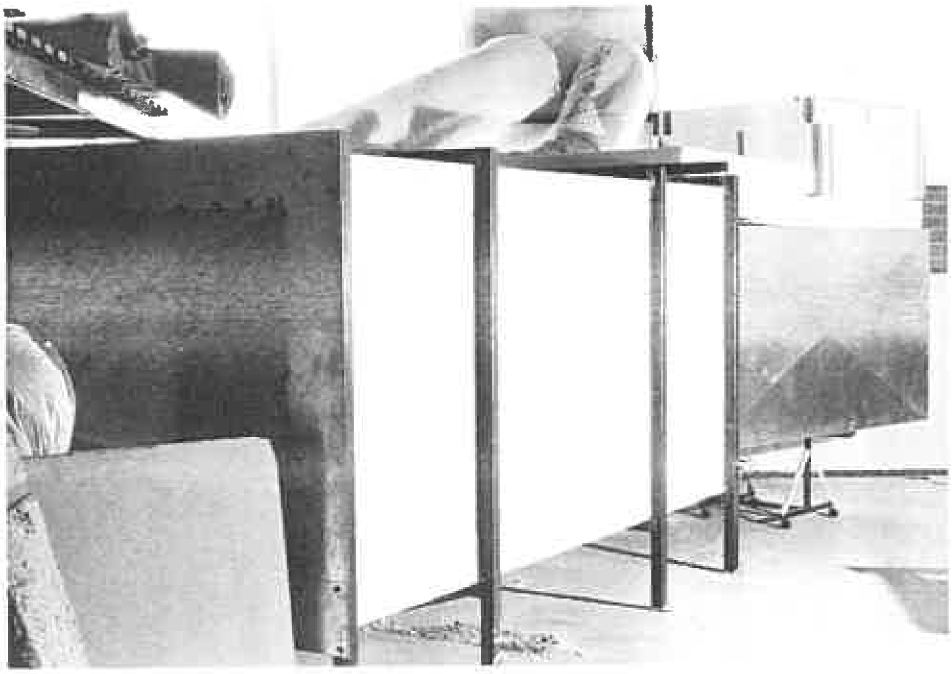
10.0 APPENDICES

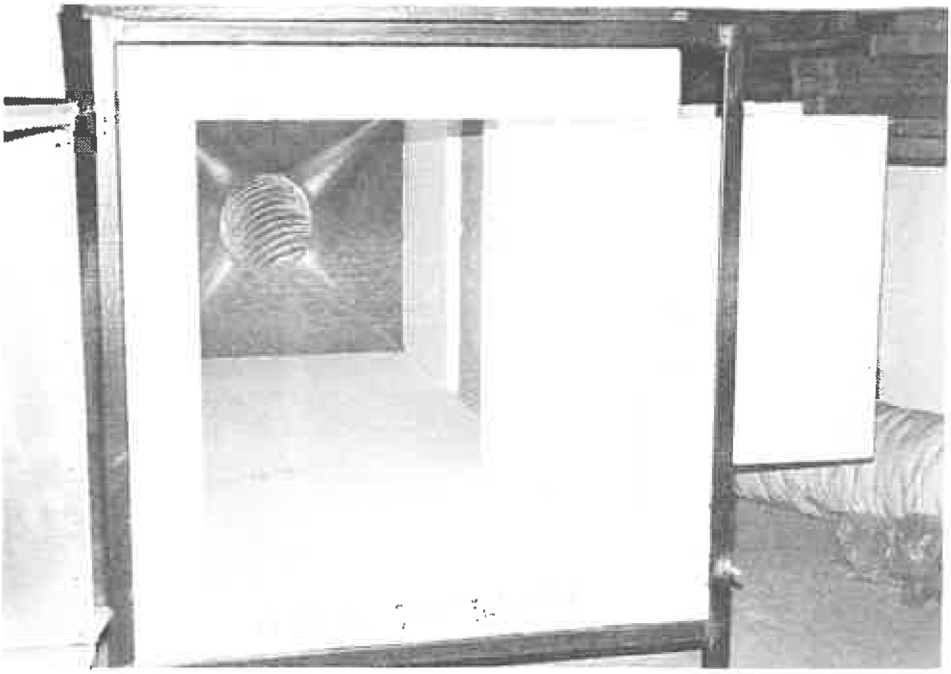
Appendix 1 Location of the Laboratory: The external view showing the south face of the building



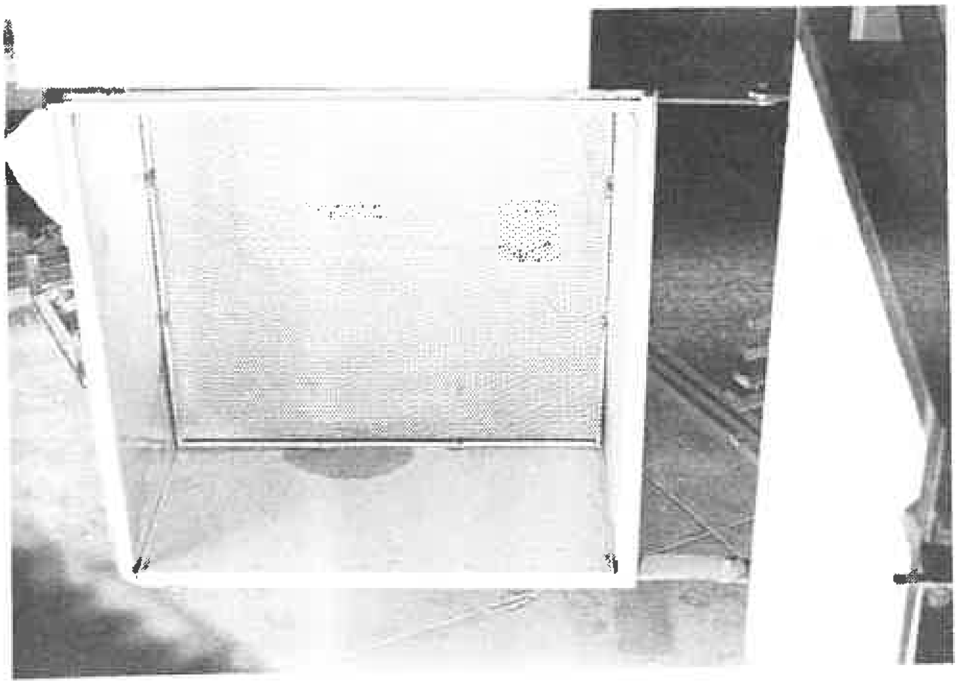
Appendix 1 cont. Views of the Support Frame: Inside the laboratory.



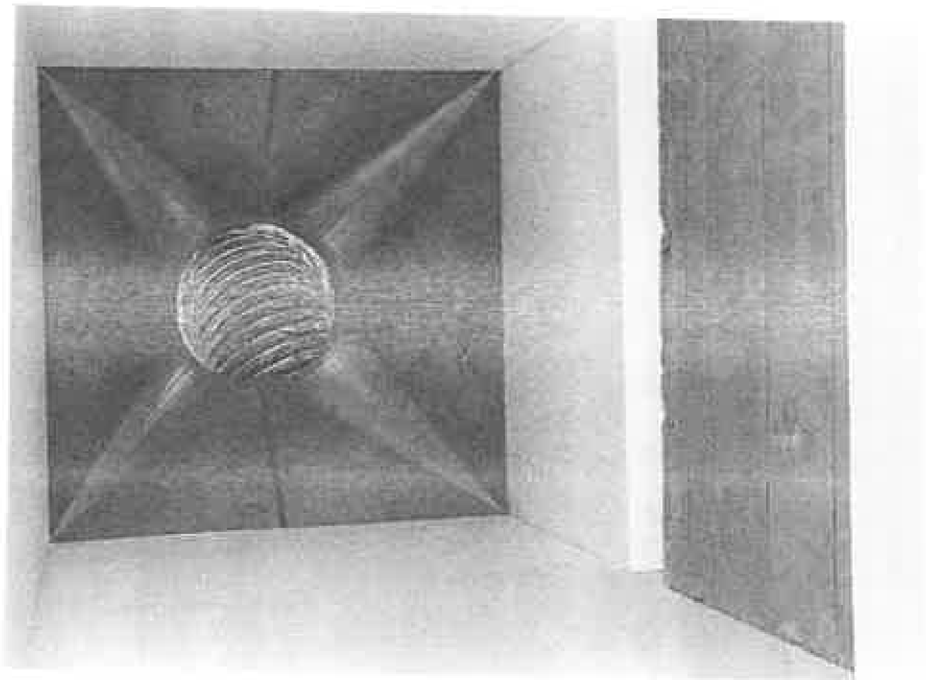
Appendix 1 cont. The Tunnel Complete: The polystrene fixed inside the frame.



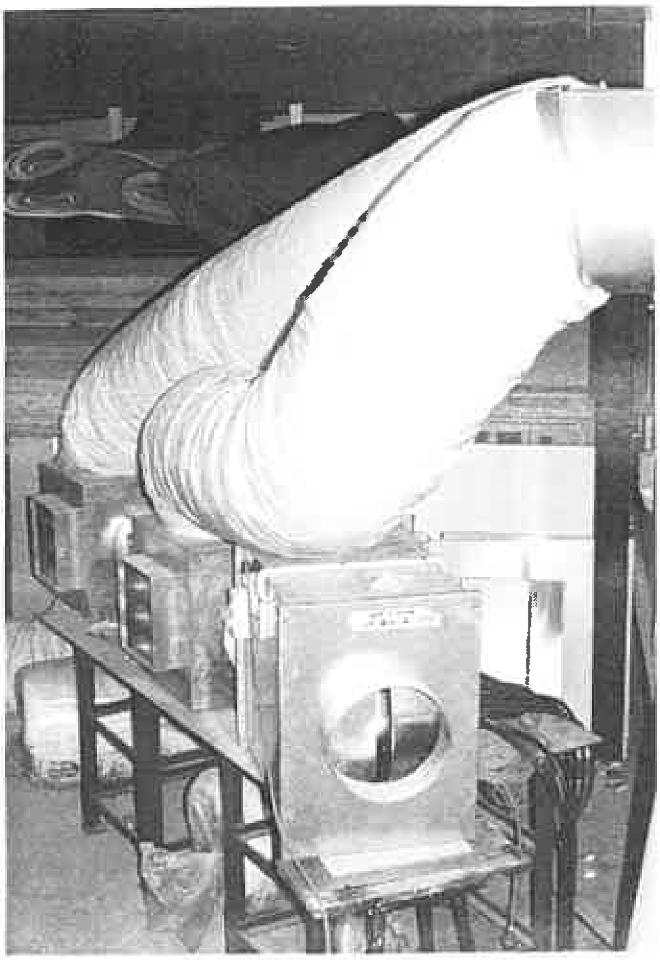
Appendix 1 cont. Inside the Tunnel: View of the shut - off dampers in an open position



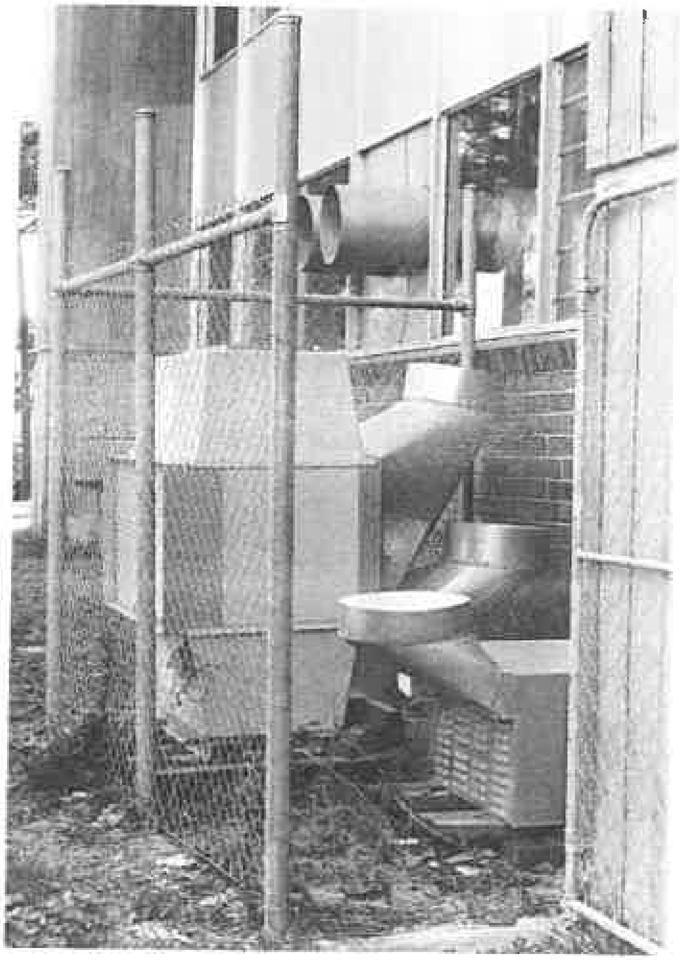
Appendix 1 cont. Air Distribution Plenum: Perforated baffle to push the air evenly across the cross section of the tunnel.



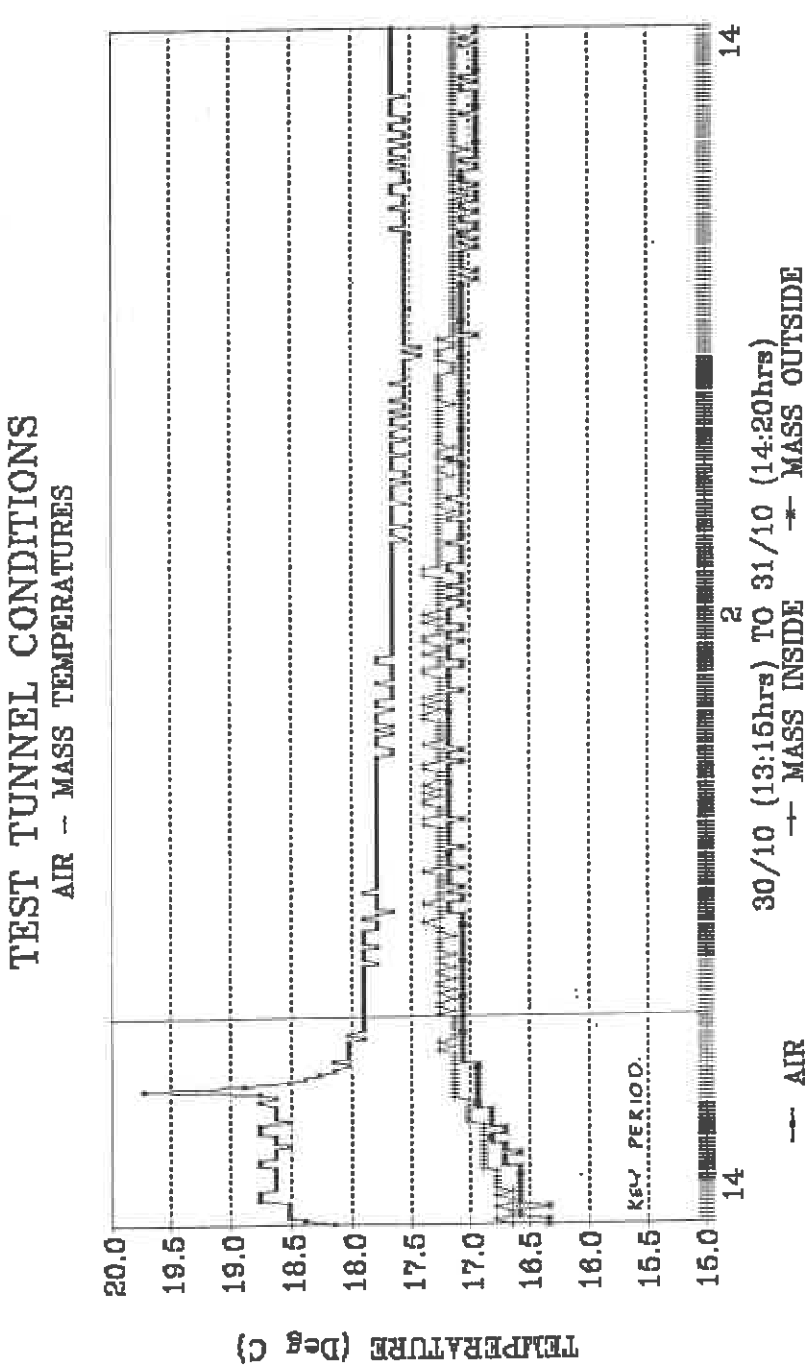
Appendix 1 cont. The Concrete Test Panel: In the middle of the length of the tunnel.



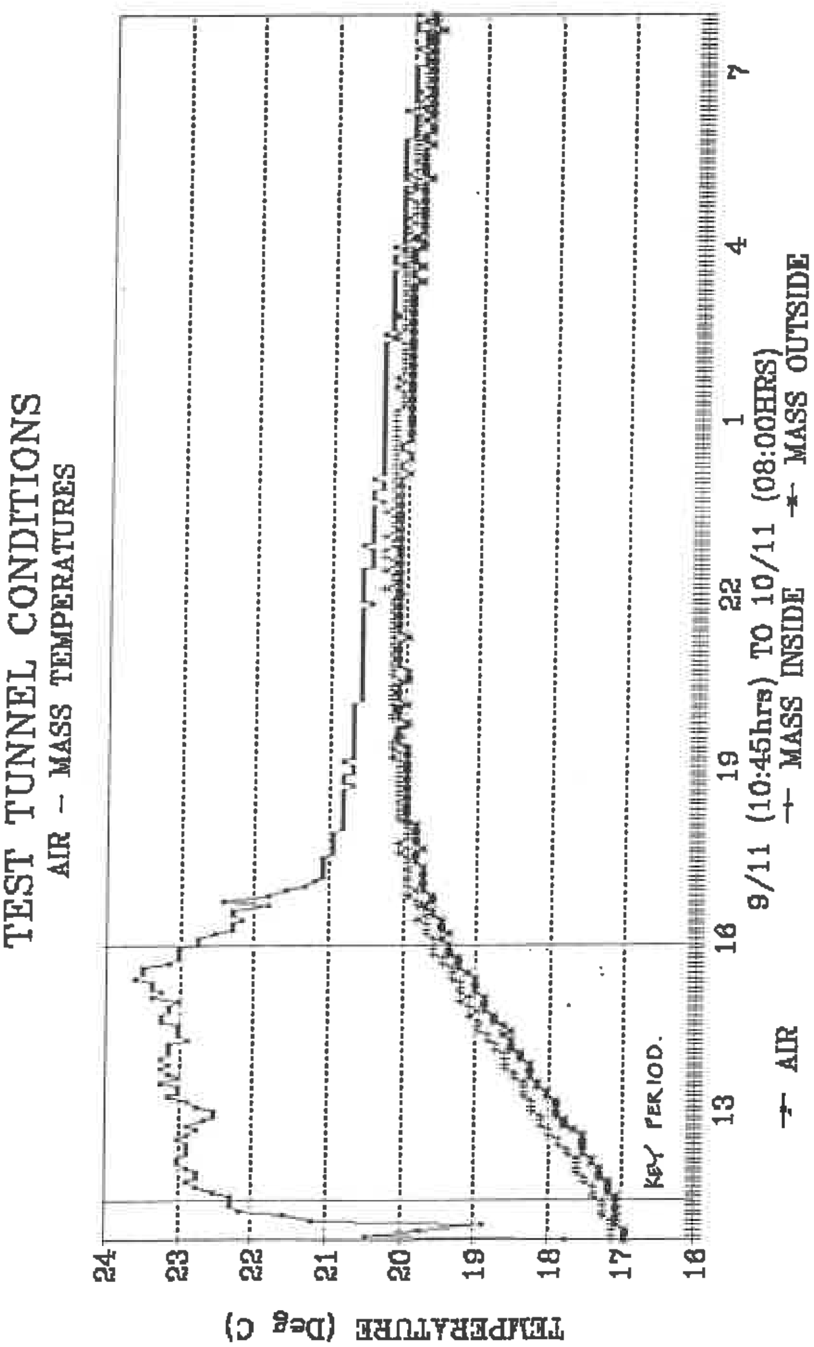
Appendix 1 cont. The Supply Air and Return Air Ducts: From the rig to widows of the laboratory.



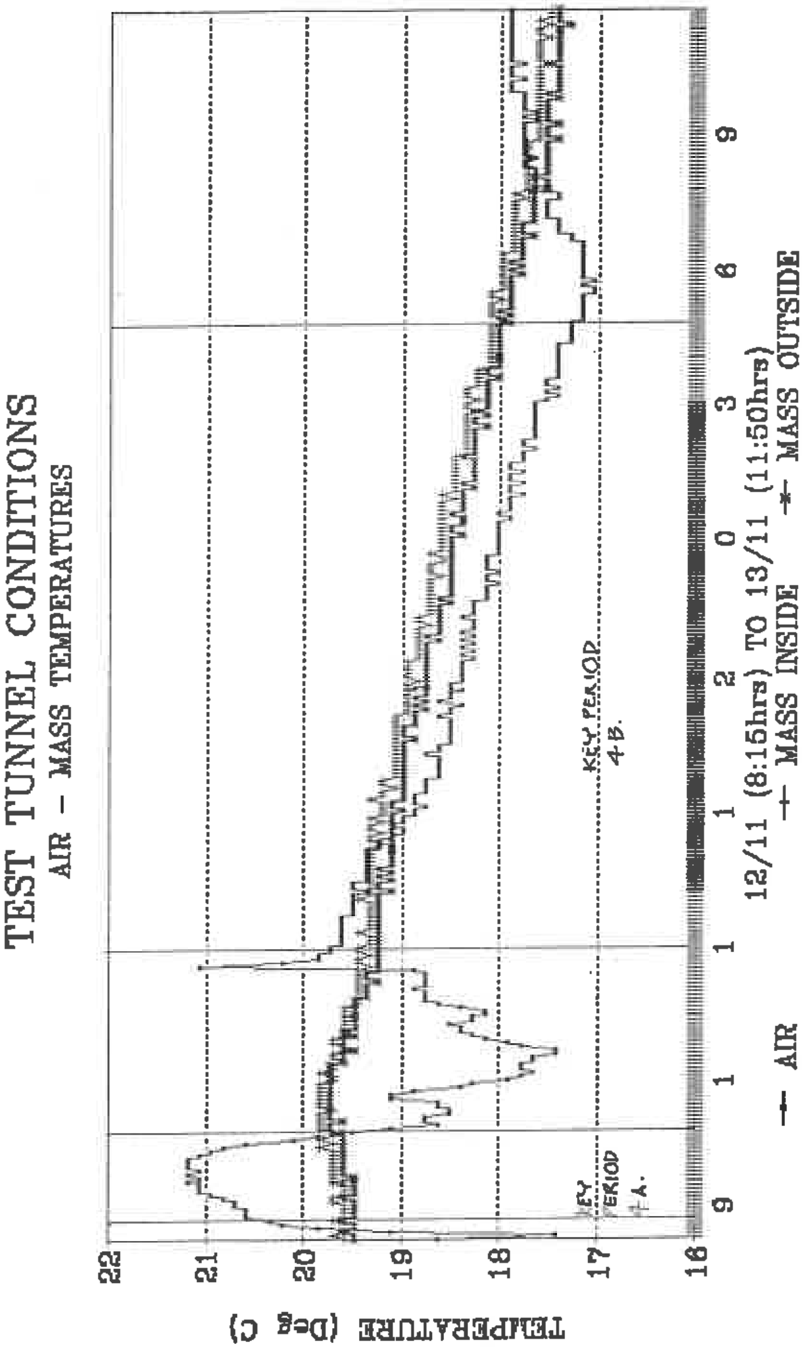
Appendix 1 cont. Mechanical Equipment: Installed immediately outside the laboratory.



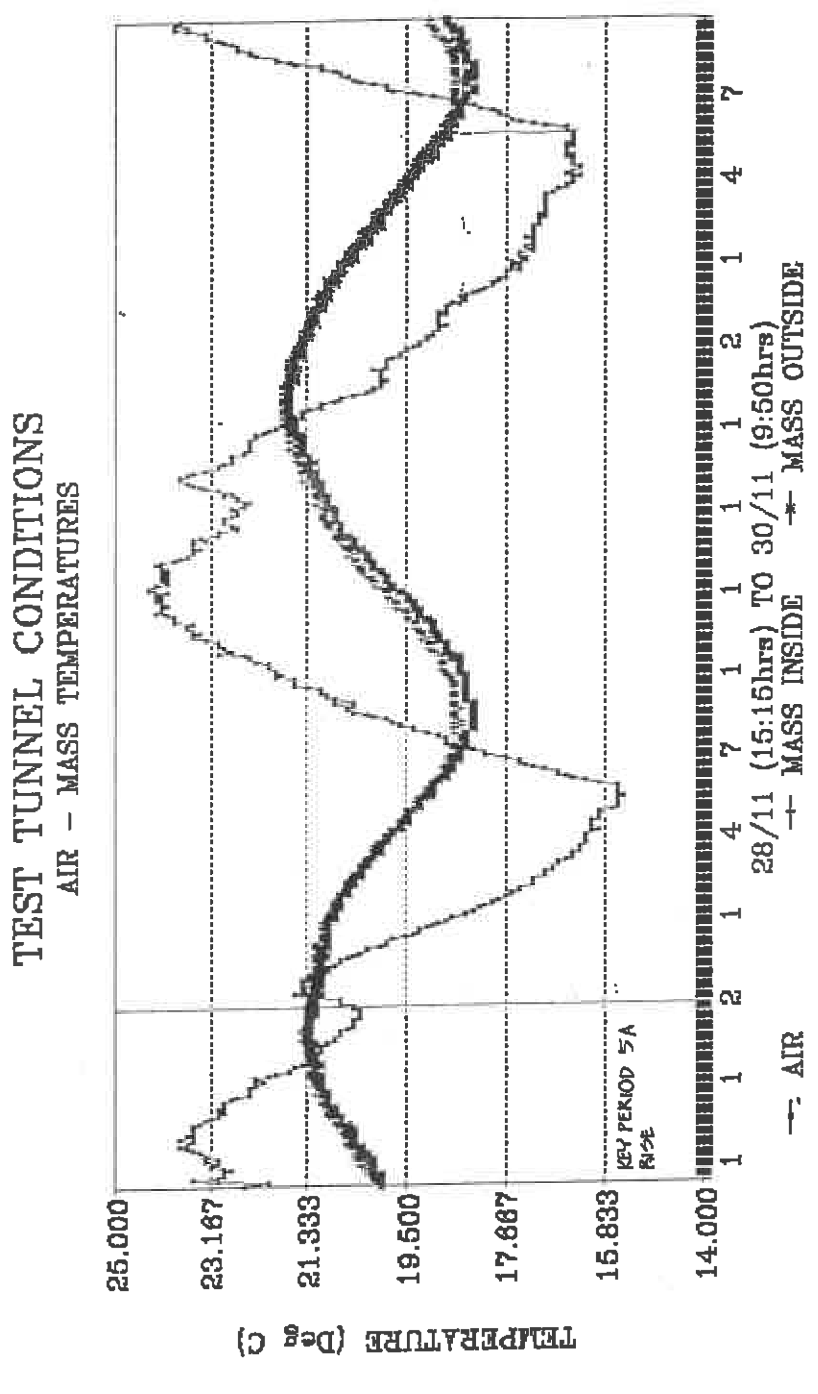
Appendix 2 Graphical data for Monitoring Period 2 - 6 (Period2)



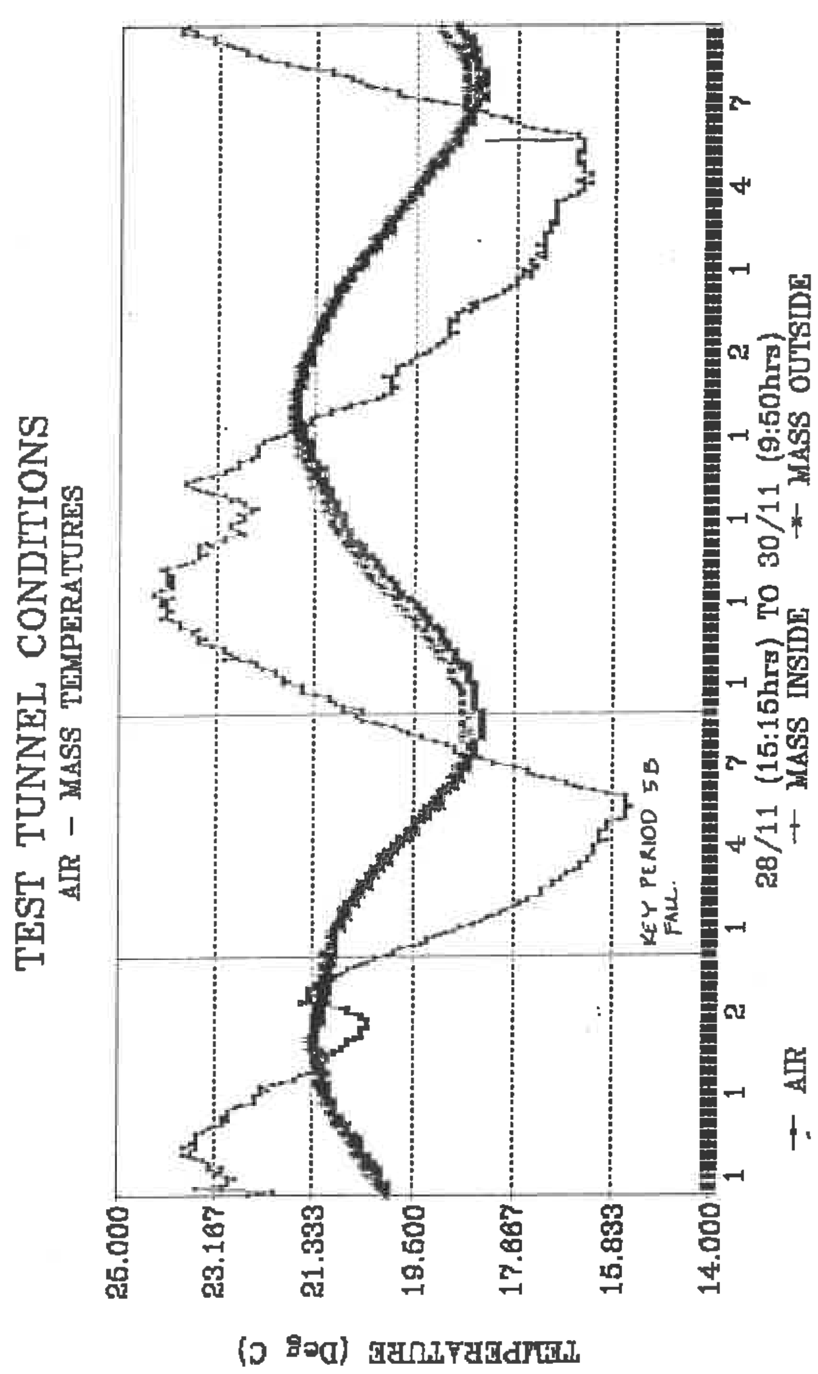
Appendix 2 cont: Graphical data for Monitoring Period 3



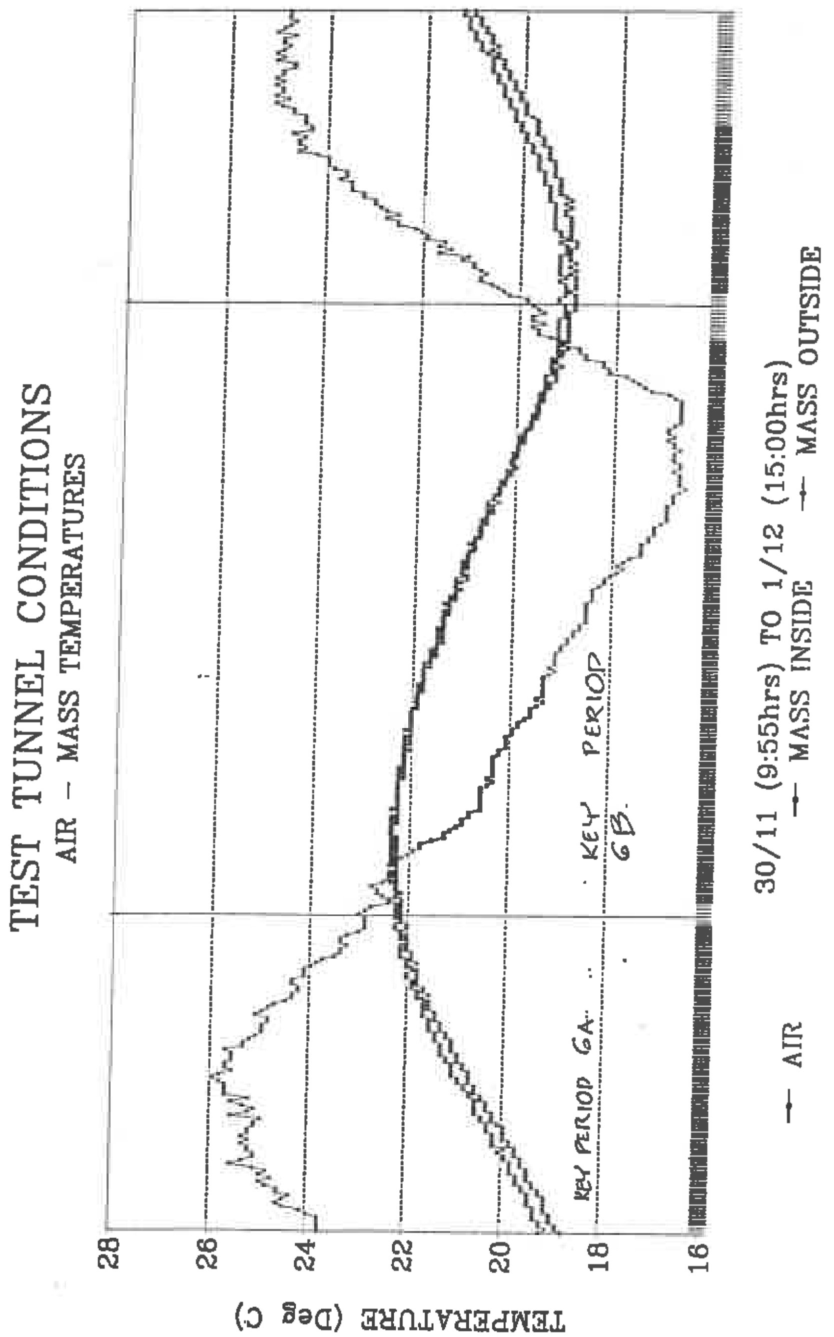
Appendix 2 cont: Graphical data for Monitoring Period 4



·Appendix 2 cont: Graphical data for Monitoring Period 5A



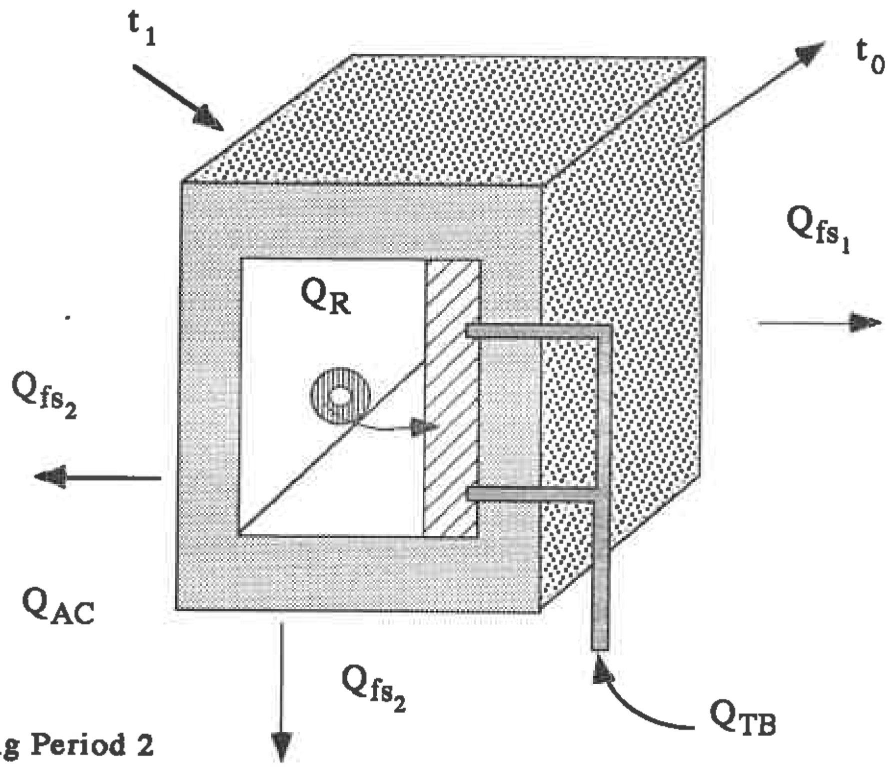
·Appendix 2 cont: Graphical data for Monitoring Period 5B



Appendix 2 cont: Graphical data for Monitoring Period 6A 6B

		Amb.	Inlet	Outlet	TA1	TA2	TA3	TM1	TM2	Lab
13:15	20/11	24.51	22.18	21.93	21.78	20.46	21.82	18.35	18.27	20.42
13:20	20/11	25.12	23.77	23.64	23.62	23.26	23.51	18.6	18.39	20.9
13:25	20/11	25	23.89	23.76	23.75	23.62	23.63	18.6	18.39	21.39
13:30	20/11	25.37	24.14	24.13	23.87	23.99	23.88	18.6	18.39	20.9
13:35	20/11	26.34	24.63	24.62	24.36	24.35	24.36	18.72	18.39	21.03
13:40	20/11	25.61	24.38	24.38	24.24	24.23	24.24	18.72	18.51	21.15
13:45	20/11	25.24	24.26	24.38	24.11	24.11	24.12	18.72	18.51	21.15
13:50	20/11	26.1	24.63	24.5	24.36	24.35	24.36	18.72	18.51	21.27
13:55	20/11	26.83	24.99	24.86	24.73	24.72	24.6	18.84	18.51	21.39
14:00	20/11	25.73	24.75	24.62	24.48	24.6	24.48	18.84	18.63	21.27
14:05	20/11	25.49	24.5	24.38	24.24	24.35	24.24	18.96	18.63	21.27
14:10	20/11	26.46	24.75	24.74	24.48	24.35	24.6	18.96	18.63	21.39
14:15	20/11	25.73	24.63	24.5	24.48	24.48	24.36	18.96	18.76	21.39
14:20	20/11	26.46	24.87	24.86	24.61	24.6	24.72	19.08	18.76	21.39
14:25	20/11	26.34	25.12	25.11	24.85	24.96	24.96	19.08	18.76	21.39
14:30	20/11	26.59	24.99	24.99	24.73	24.84	24.84	19.21	18.88	21.51
14:35	20/11	27.44	25.61	25.35	25.22	25.21	25.21	19.21	18.88	21.63
14:40	20/11	26.46	25.24	25.11	24.97	25.09	24.96	19.21	19	21.63
14:45	20/11	25.61	24.75	24.62	24.61	24.72	24.6	19.33	19	21.63
14:50	20/11	26.59	24.99	24.86	24.73	24.72	24.72	19.33	19.12	21.63
14:55	20/11	27.07	25.48	25.35	25.1	25.09	25.09	19.45	19.12	21.63
15:00	20/11	26.83	25.48	25.23	25.1	25.21	24.96	19.45	19.12	21.87
15:05	20/11	26.95	25.48	25.35	25.1	25.21	25.09	19.57	19.12	21.87
15:10	20/11	27.68	25.73	25.6	25.59	25.33	25.33	19.57	19.24	21.75
15:15	20/11	26.71	25.24	25.23	25.1	25.33	25.09	19.57	19.36	21.75
15:20	20/11	27.2	25.48	25.35	25.22	25.21	25.21	19.57	19.36	21.75
15:25	20/11	27.44	25.73	25.6	25.59	25.57	25.33	19.69	19.48	21.87
15:30	20/11	27.32	25.73	25.48	25.34	25.33	25.33	19.69	19.48	21.75
15:35	20/11	27.56	25.73	25.6	25.59	25.57	25.45	19.82	19.48	21.99
15:40	20/11	26.95	25.48	25.35	25.22	25.33	25.21	19.82	19.6	21.75
15:45	20/11	27.07	25.61	25.48	25.22	25.33	25.33	19.94	19.73	21.99
15:50	20/11	27.44	25.73	25.6	25.59	25.57	25.45	19.94	19.73	22.11
15:55	20/11	27.56	25.97	25.97	25.83	25.69	25.81	20.06	19.73	22.11
16:00	20/11	27.44	25.85	25.6	25.59	25.69	25.45	20.06	19.85	21.87
16:05	20/11	27.68	25.73	25.6	25.34	25.57	25.45	20.06	19.85	21.87
16:10	20/11	27.81	25.61	25.48	25.22	25.33	25.33	20.18	19.85	21.87
16:15	20/11	27.93	25.73	25.6	25.34	25.57	25.45	20.18	19.97	21.75
16:20	20/11	28.17	25.73	25.6	25.34	25.57	25.45	20.3	19.97	21.75
16:25	20/11	28.17	25.61	25.48	25.34	25.57	25.33	20.3	20.09	21.75
16:30	20/11	27.93	25.85	25.84	25.59	25.69	25.45	20.43	20.09	21.63
16:35	20/11	28.29	25.85	25.6	25.59	25.69	25.45	20.43	20.21	21.99
16:40	20/11	28.41	25.73	25.6	25.59	25.69	25.45	20.43	20.33	21.87
16:45	20/11	28.29	25.73	25.6	25.59	25.69	25.45	20.55	20.09	21.87
16:50	20/11	28.66	25.97	25.97	25.71	25.82	25.57	20.55	20.33	21.99
16:55	20/11	28.78	25.85	25.84	25.59	25.69	25.45	20.67	20.33	21.87
17:00	20/11	31.34	25.73	25.6	25.34	25.69	25.33	20.67	20.45	21.87
17:05	20/11	30.61	25.73	25.6	25.59	25.69	25.33	20.67	20.45	21.87
17:10	20/11	31.22	25.61	25.48	25.34	25.33	25.21	20.79	20.57	21.87
17:15	20/11	29.76	25.73	25.48	25.34	25.57	25.33	20.79	20.7	21.75
17:20	20/11	27.81	25.73	25.6	25.59	25.57	25.45	20.91	20.57	21.87
17:25	20/11	27.44	25.61	25.48	25.34	25.33	25.33	20.91	20.7	21.87
17:30	20/11	27.44	25.61	25.48	25.34	25.57	25.33	20.91	20.7	21.87
17:35	20/11	27.07	25.48	25.35	25.22	25.33	25.21	20.91	20.82	21.87
17:40	20/11	27.07	25.48	25.35	25.22	25.21	25.21	21.04	20.82	21.87
17:45	20/11	26.95	25.24	25.23	25.1	25.21	24.96	21.04	20.94	21.87
17:50	20/11	26.95	25.12	25.11	24.97	25.21	24.96	21.04	20.94	21.75

Appendix 3 Typical Temperature Recordings for Global Measurements



Monitoring Period 2
Conditions:
0.2 m/s Heating of Mass over 3 hours (14.00 to 17.00 - 30.10.90)

Average Δ t between Mass & Air = 1.37
Average Δ t between Chamber & Laboratory = 0.2
Average Δ t between Outlet & Inlet = 0.11

Period 2 Data Summary

Q_{fs} = + 0.17W

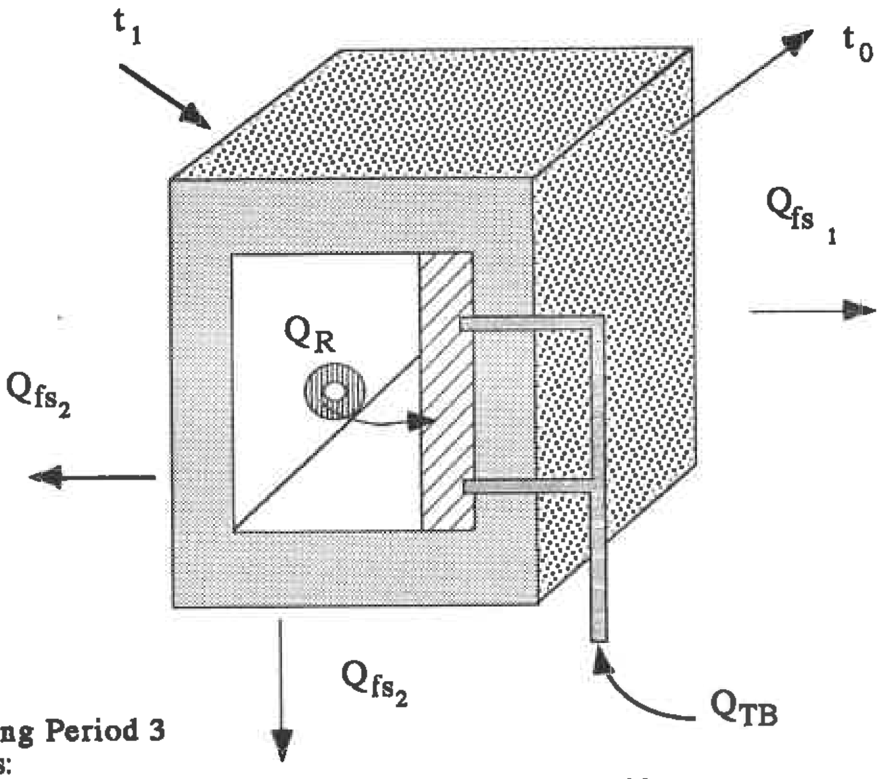
Q_{AC} = + 26.31W

Q_{R} = + 1.59W

Q_{TS} = 28.07 W/m^2

Q_{TB} was considered to be insignificant.

Appendix 4 Analytical calculations for Monitoring Period 2 - 6



Monitoring Period 3
Conditions:
0.2 m/s Heating of Mass over 5 hours (10.45 to 14.45 - 9.11.90)

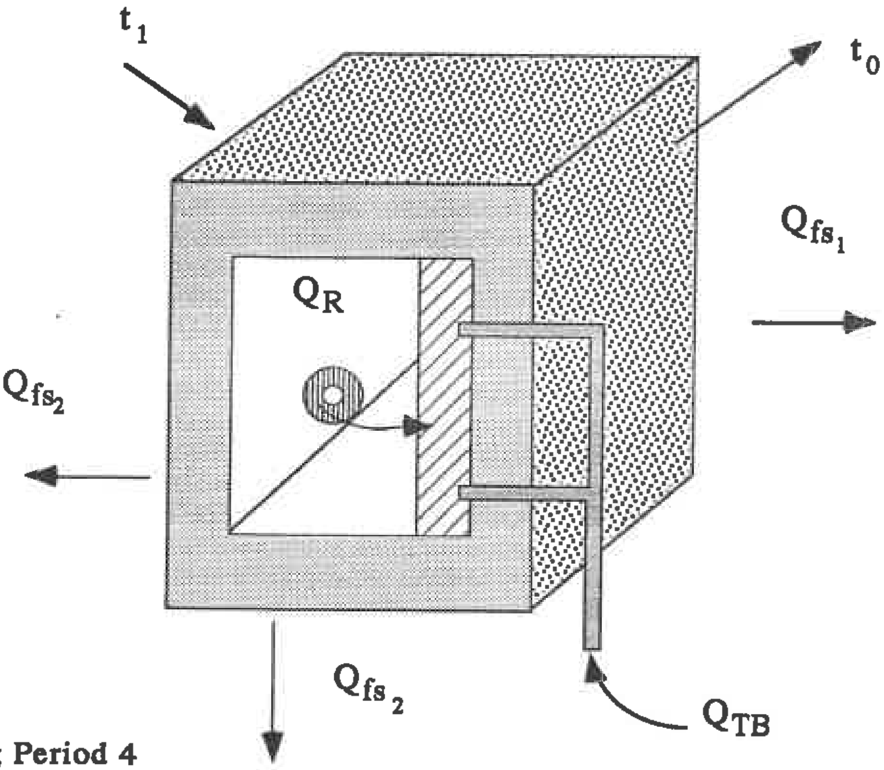
Average Δ t between Mass & Air = 4.5
Average Δ t between Chamber & Laboratory = 2.48
Average Δ t between Outlet & Inlet = 0.04

Period 3 Data Summary

Q_{fs} = + 2.16W
Q_{AC} = + 9.57W
Q_R = +4.94W
Q_{TS} = 16.67 W/m^2

Q_{TB} was considered to be insignificant.

Appendix 4 cont: Analytical calculations for Monitoring Period 2 - 6



Monitoring Period 4
Conditions:
0.1 m/s Heating of Mass over 2 hours (08.15 to 11.15 - 12.11.90)

Average Δ t between Mass & Air = 2.0
Average Δ t between Chamber & Laboratory = - 0.7
Average Δ t between Outlet & Inlet = 0.055°

Period 4 Data Summary

Q_{fs} = - 0.61W

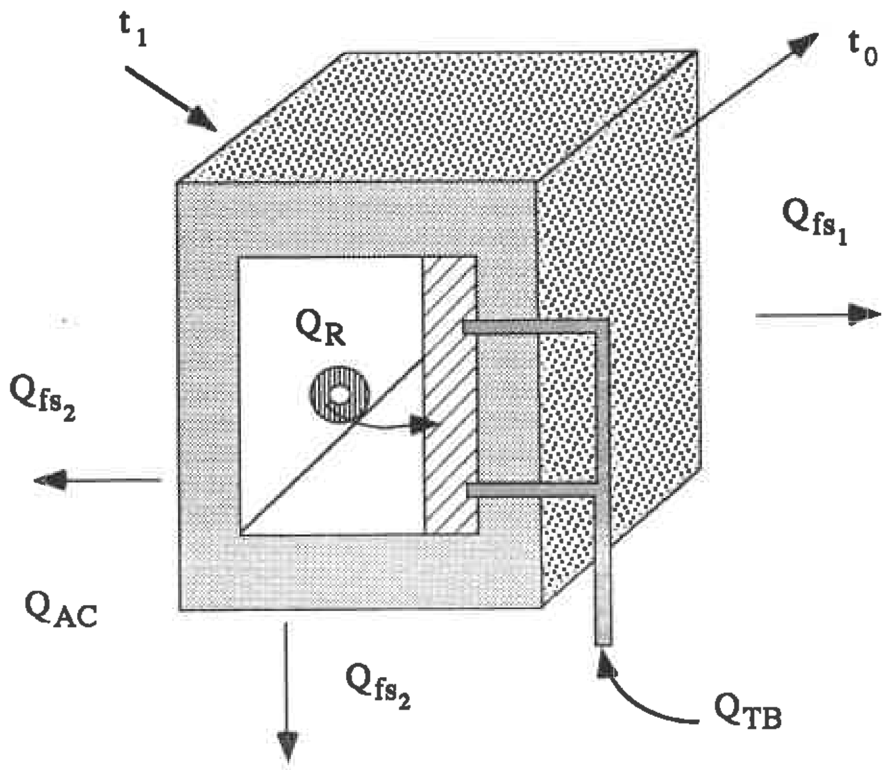
Q_{AC} = + 6.57W

Q_R = - 0.541W

Q_{TS} = 8.82 W/m^2

Q_{TB} was considered to be insignificant.

Appendix 4 cont: Analytical calculations for Monitoring Period 2 - 6



Monitoring Period 5A
Conditions:
0.2 m/s Heating of Mass over 3 hours (15.15 to 19.15 - 12.11.90)

Average Δ t between Mass & Air = 1.6
Average Δ t between Chamber & Laboratory = 1.2
Average Δ t between Outlet & Inlet = 0.03

Period 5A Data Summary

Q_{fs} = + 1.05W

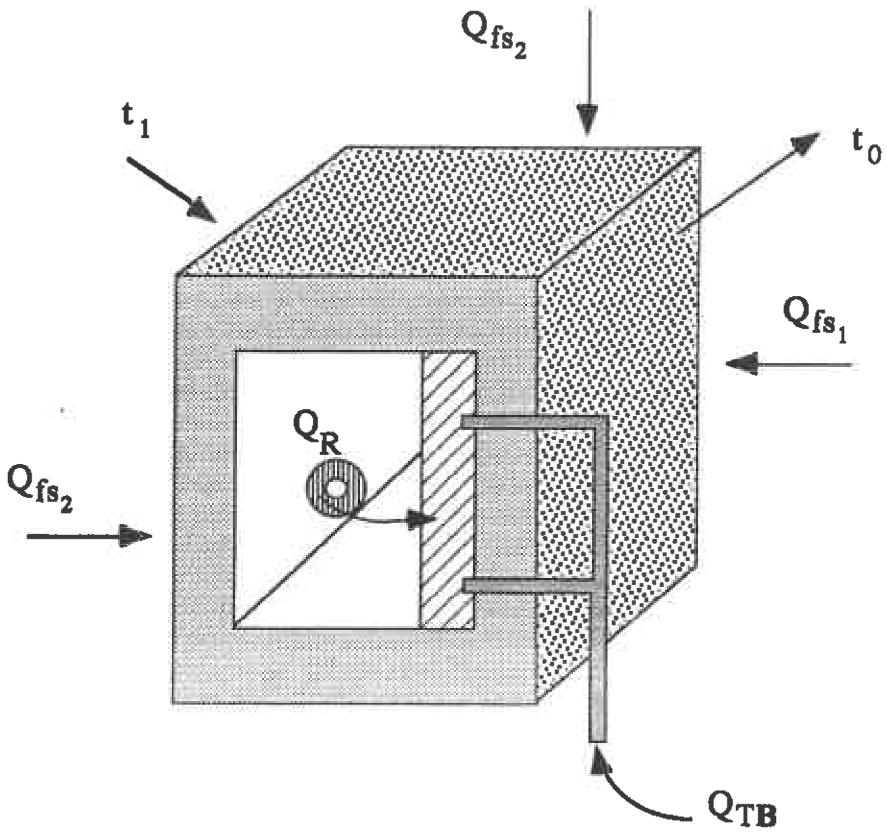
Q_{AC} = +7.18W

Q_R = +1.08W

Q_{TS} = 9.31 W/m^2

Q_{TB} was considered to be insignificant.

Appendix 4 cont: Analytical calculations for Monitoring Period 2 - 6



Monitoring Period 5B
Conditions:
0.2 m/s Cooling of Mass over 8 hours (22.05 to 06.05 - 29.11.90)

Average Δ t between Mass & Air = - 1.5
Average Δ t between Chamber & Laboratory = -1.3
Average Δ t between Outlet & Inlet = -0.095

Period 5B Data Summary

Q_{fs} = - 1.13W

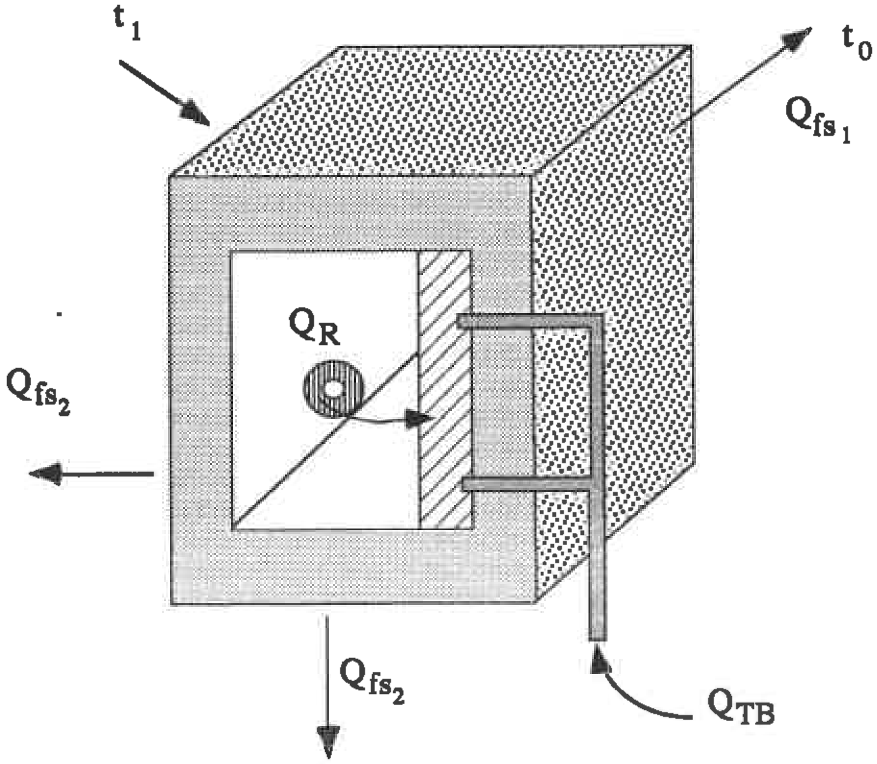
Q_{AC} = - 22.72W

Q_R = -1.49W

Q_{TS} = -25.34 W/m^2

Q_{TB} was considered to be insignificant.

Appendix 4 cont: Analytical calculations for Monitoring Period 2 - 6



Monitoring Period 6A
Conditions:
0.1 m/s Heating of Mass over 8 hours

Average Δ t between Mass & Air = 3.65
Average Δ t between Chamber & Laboratory = 2.3
Average Δ t between Outlet & Inlet = 0.028

Period 6A Data Summary

Q_{fs} = + 1.56W

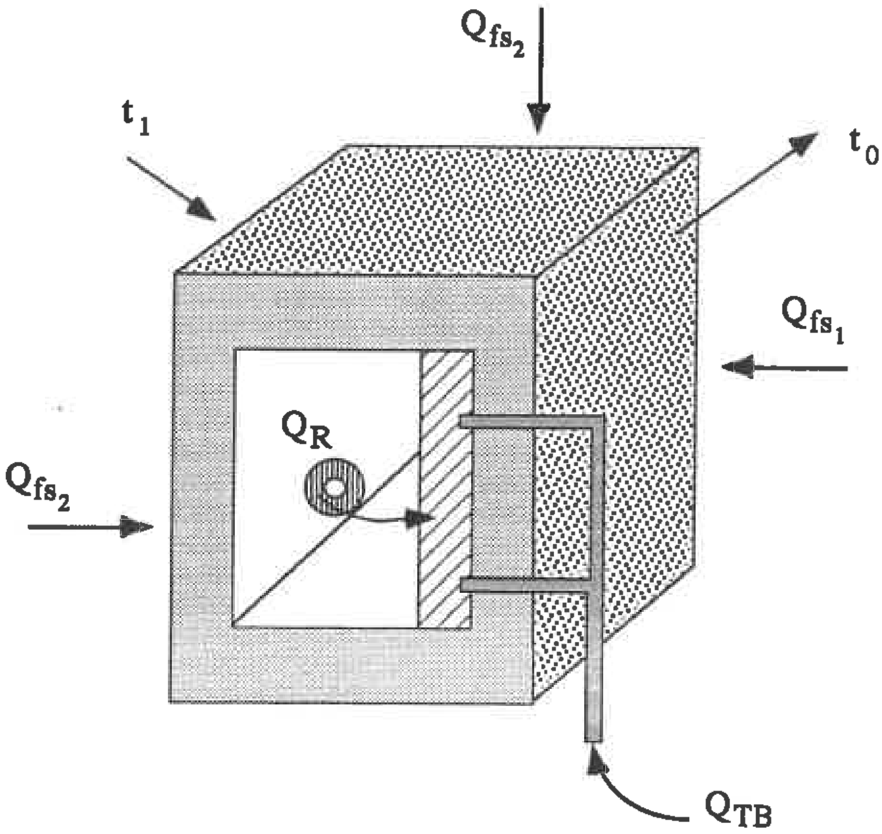
Q_{AC} = +8.61W

Q_R = + 3.98W

Q_{TS} = 14.15 W/m^2

Q_{TB} was considered to be insignificant.

Appendix 4 cont: Analytical calculations for Monitoring Period 2 - 6



Monitoring Period 6B
Conditions:
0.1 m/s Cooling of Mass over 14.5 hours

Average Δ t between Mass & Air = - 2.34
Average Δ t between Chamber & Laboratory = -2.25
Average Δ t between Outlet & Inlet = 0.12

Period 6B Data Summary

Q_{fs} = - 1.96W

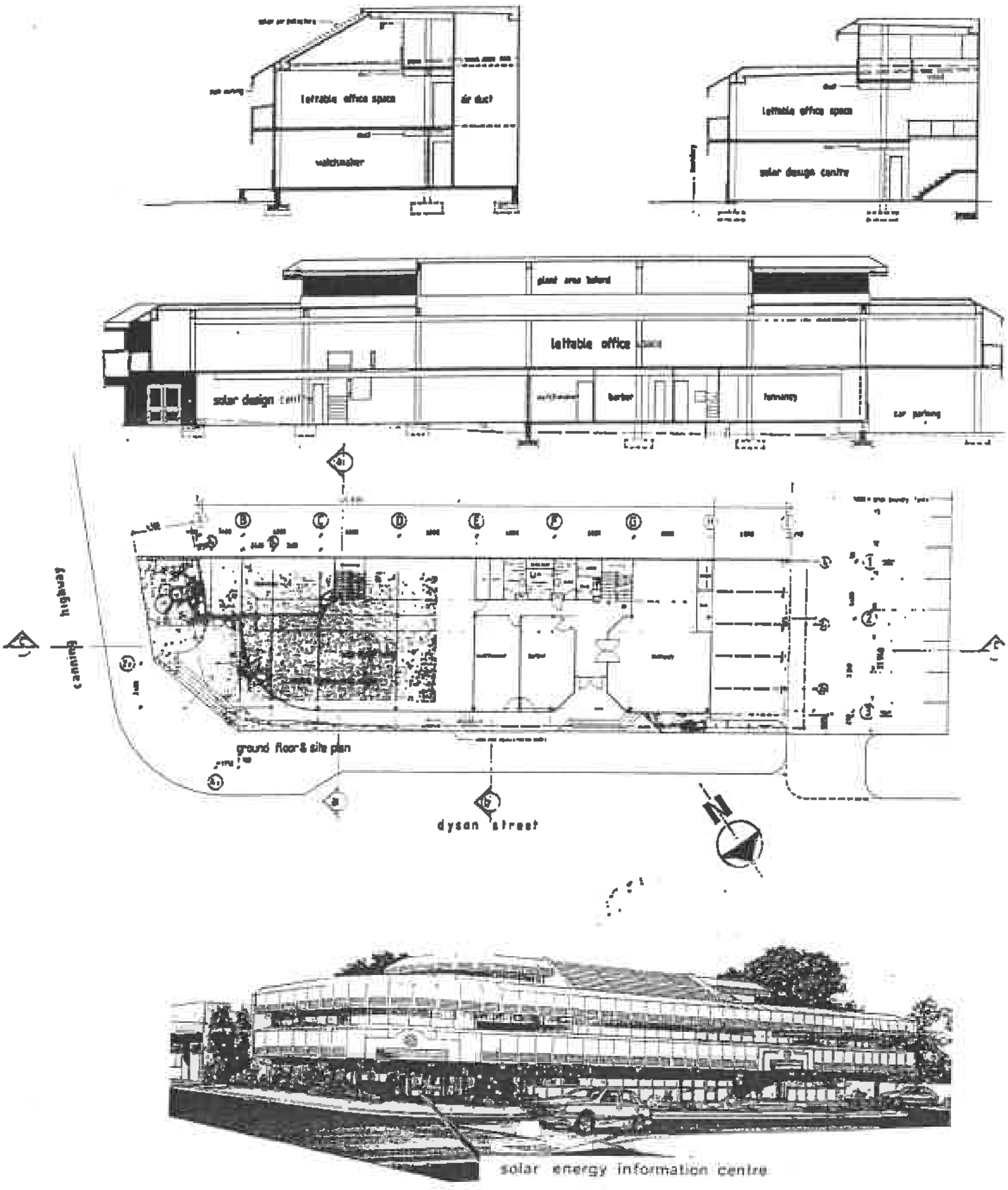
Q_{AC} = - 14.35W

Q_R = - 2.53W

Q_{TS} = 18.84 W/m^2

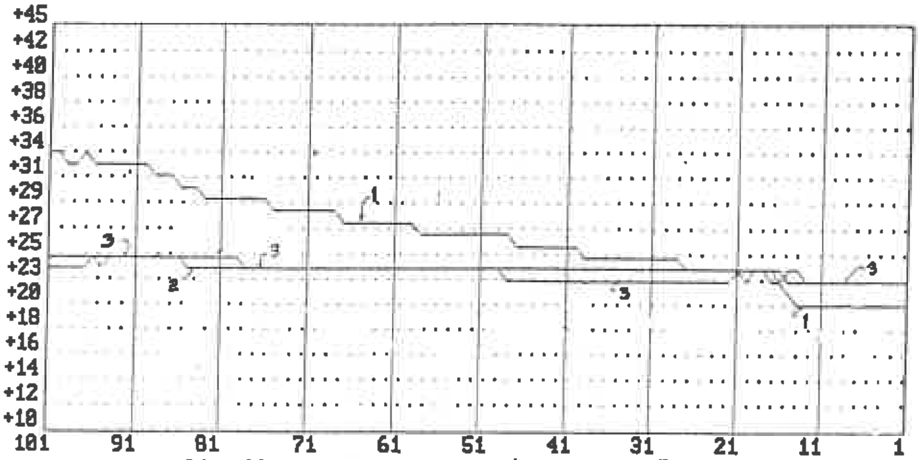
Q_{TB} was considered to be insignificant.

Appendix 4 cont: Analytical calculations for Monitoring Period 2 - 6



Appendix 5
Sections, Plan and Perspective Sketches of Solar Energy Information Centre

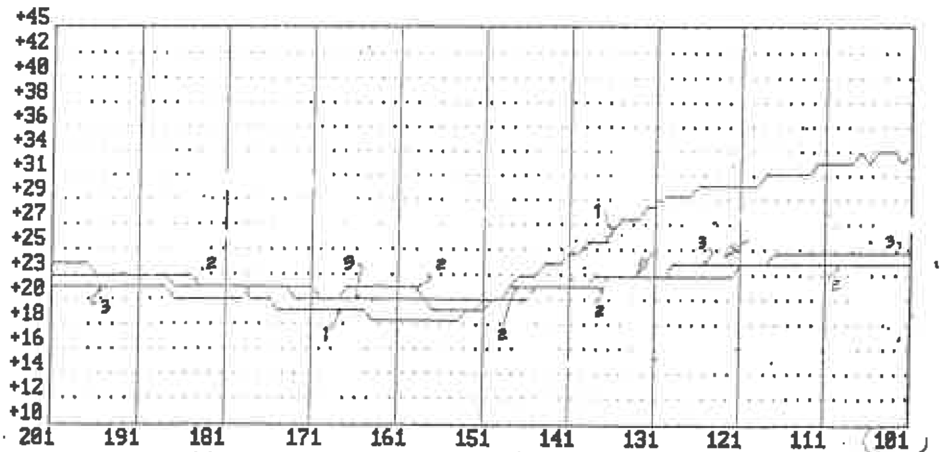
Plot for: SOLAR DESIGN Mon 12-Mar-1990 08:53:03
Trend #01: OUTSIDE AIR TEMP System 101 Start : 1
Trend #02: RETURN AIR TEMP System 102 End : 101
Trend #03: SUPPLY AIR TEMP System 104 Samples: 500



[<-- Oldest] Sample No.'s [Newest -->]

Interval: 00:10 Elapsed Time: 00/00:10 Last Sample: 12/08:24 Plots shown: 3

Plot for: SOLAR DESIGN Mon 12-Mar-1990 08:55:02
Trend #01: OUTSIDE AIR TEMP System 101 Start : 101
Trend #02: RETURN AIR TEMP System 102 End : 201
Trend #03: SUPPLY AIR TEMP System 104 Samples: 500



[<-- Oldest] Sample No.'s [Newest -->]

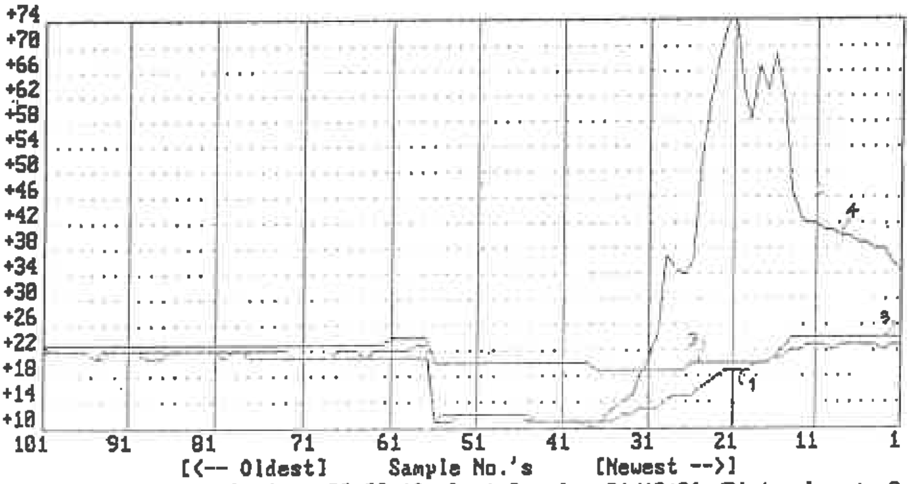
Interval: 00:10 Elapsed Time: 00/16:50 Last Sample: 12/08:24 Plots shown: 3

Typical Summer Records

Appendix 6 SEIC Summer temperature record

Plot for: SOLAR DESIGN
Trend #01: OUTSIDE AIR TEMP — System 101
Trend #03: SUPPLY AIR TEMP — System 104
Trend #04: SOLAR DUCT TEMP AVE — System 106

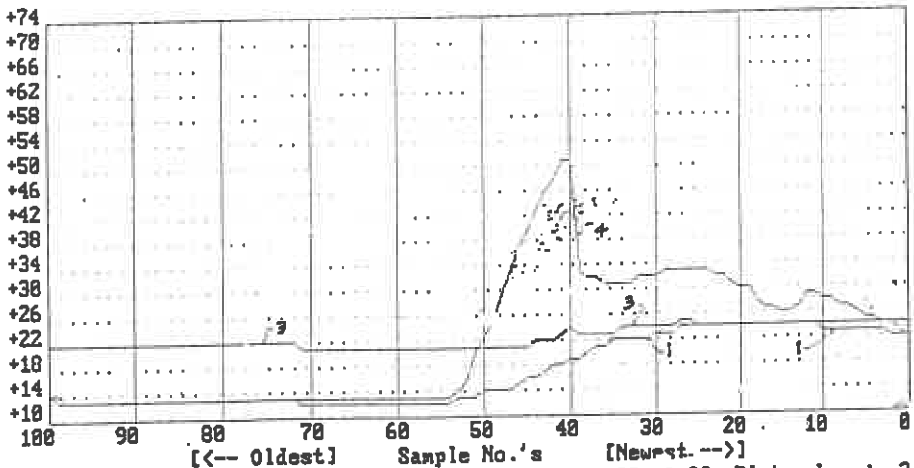
Fri 01-Jun-1990 15:42:03
Start : 1
End : 101
Samples: 500



Interval: 00:10 Elapsed Time: 00/00:10 Last Sample: 01/13:36 Plots shown: 3

Plot for: SOLAR DESIGN
Trend #01: OUTSIDE AIR TEMP — System 101
Trend #03: SUPPLY AIR TEMP — System 104
Trend #04: SOLAR DUCT TEMP AVE — System 106

Wed 23-May-1990 10:44:01
Start : 0
End : 100
Samples: 500



Interval: 00:10 Elapsed Time: 00/00:00 Last Sample: 23:16:20 Plots shown: 3

Typical Winter Records

Appendix 6 cont SEIC winter temperature records

C1 150mm straight concrete with skim coat plaster finish
C2 150mm straight concrete finish 25mm vermiculite spray
C3 150mm straight concrete with skim coat plaster finish with textured latex type paint
C4 150mm with Bondek form work as exposed ceiling
C5 150mm with Bondek and 25mm vermiculite spray
C6 150mm with Bondek form work as exposed ceiling painted with textured paint finish
C7 metal strip suspended ceiling insulated with R2 insulation
C8 suspended gyprock ceiling with R2 insulation.

CEILINGS

FLOORS

F1 150mm concrete with quarry tiles
F2 150mm concrete with direct stick carpet
F3 150mm concrete with commercial carpet and underlay
F4 150mm concrete with carpet tiles with rubber backing
F5 150mm concrete with vinyl sheeting
F6 150mm precast prestressed light weight concrete floor units

W1 150mm concrete spandrel (lightweight) - concrete exposed smooth finish
W2 150mm concrete spandrel (lightweight) - concrete exposed smooth finish with painted surface
W3 50mm concrete spandrel(lightweight) - concrete exposed smooth finish with plaster and painted
W4 50mm concrete spandrel (lightweight) concrete exposed smooth finish with wall paper finish.
W5 150mm concrete spandrel with exposed concrete rough texture and ribbed surface
W6 110mm brick spandrel face inside the building
W7 110mm brick spandrel face inside the building with painted finish
W8 110mm brick spandrel face inside the building with plaster and painted
W9 10mm brick spandrel face inside the building with rough bagged finish unpainted
W10 Precast prestressed lightweight wall units (150mm thick)
W11 40mm thick partition styrene bead/plaster infill core gyprock or plasterboard surfaced (painted)
W12 40mm thick partition styrene bead/plaster infill core gyprock or plasterboard surfaced (wall paper)
W13 40mm thick partition styrene bead/concrete infill core cell fibre cement board surfaced (painted)
W14 40mm thick partition styrene bead/concrete infill core cell fibre cement board surfaced (wallpaper)
W15 35mm light weight honeycomb paler core plywood faced (clear finish)

WALLS

Appendix 7 Building Materials for future study