University of Southern Queensland
Faculty of Engineering and Surveying

COMPUTATIONAL FLUID DYNAMICS (CFD) INVESTIGATION OF THE THERMAL EFFICIENCY OF MICRO CLIMATE CONDITIONING OF A TYPICAL WORKSTATION

A dissertation submitted by

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towards the degree of

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ABSTRACT

Energy efficiency of buildings is becoming more critical, evidenced by the fact that building rating schemes and minimum efficiencies are now referenced in section J of the Building Code of Australia (BCA), and it would be reasonable to assume these requirements will be expanded in due course.

This project aims to investigate micro climate air conditioning as an alternative to conditioning the entire space in order to provide a more efficient system to be implemented in typical office workspaces. Conventional air conditioning systems typically are based on supplying conditioned air at low velocity from ceiling mounted registers. The basis for a micro climate system is to provide low air quantities at a higher velocity and slightly higher temperature locally at the workstation level. Computational Fluid Dynamics (CFD) computer programs provide a predictive tool to simulate the resultant thermal output from an air conditioning system. The effectiveness of a micro climate system was tested in a typical open plan office workstation. Phoenics CFD software was used to provide results on the expected thermal gradients in the space which was then compared against the data compiled from two data loggers located both at the workstation and the ceiling level.

Reasonable correlations between the CFD analysis and the data loggers output were achieved. A number of assumptions were made in order to develop the CFD model, which appear to have been substantiated particularly for the summer conditions. A number of variables were changed to improve the performance of the system; some of these were immediately implemented with favourable results.

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Date

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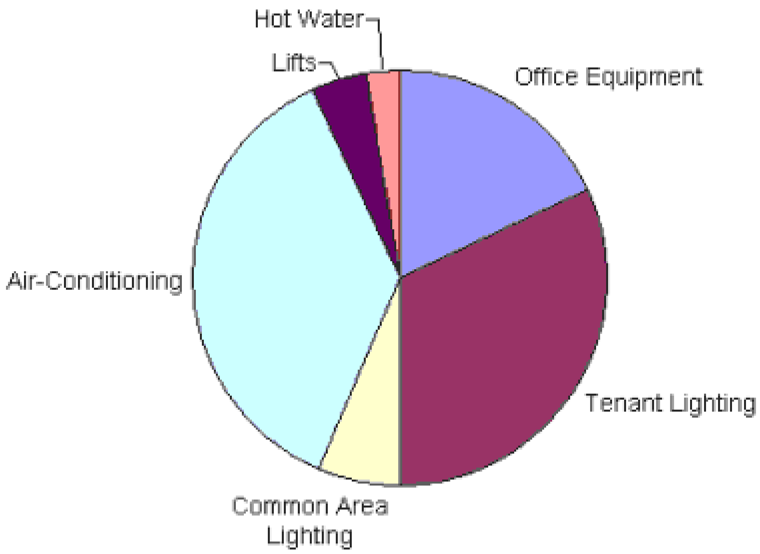
GLOSSARY OF TERMS

m = Meters
Q = heat gain by conduction (W)
U = overall heat transfer coefficient ($W/m^2.°K$)
A = area of the surface
ΔT = dry bulb temperature difference across the surface
Pa = Pascals
°C = degrees Celsius
mm = millimetres
L/s = Litres per second
DB = Dry Bulb temperature in °C
WB = Wet Bulb temperature in °C
min = minute
BCA = Building Code of Australia

CHAPTER 1

1.1 Introduction

Mechanical systems that make up the air conditioning plant consume as much as 40% of the total energy used in a building. When combined with the lighting systems the effective total energy used can be as high as 85% of entire building. Below is a pie chart representing the proportioning of energy costs in a typical commercial building.



An opportunity therefore exists to reduce a large proportion of energy usage in buildings simply by improving the efficiency of the air conditioning systems.
Meeting minimum efficiency targets for buildings are now mandatory as referenced in section J of the Building Code of Australia (BCA). The BCA nominates specific deemed to satisfy requirements both for the building fabric as well as the building services. In order to comply the proposed development must be a deemed to satisfy solution or achieve a lower value than the prescribed energy target when the building is modelled using approved energy modelling software.
Designers and developers must embrace energy efficient systems and technology in order to meet the targets set in the Australian Standards and the BCA. The following system analysis is designed to test the application of a system that represents a dramatic reduction in energy usage for buildings. If the same methodology can be utilised in larger developments a dramatic reduction in the energy would be required for air conditioning.

CHAPTER 2

2.1 Background and Literature Review

2.1.1 Introduction

Many authors have described various methods employed to air condition buildings. This chapter will provide a brief insight into some of the traditional types of air conditioning systems typically employed within office spaces. The factors that influence the heat load within the space will also be identified, as well as summarising a case study of thermal satisfaction of occupants within office spaces in geographically different locations.

2.1.2 Types of Air Conditioning Systems

Air Conditioning essentially provides a mechanical means of reducing the temperature of an internal space by rejecting the corresponding internal heat to the ambient air. Refrigerants are typically delivered via an expansion device to an evaporator where it boils until it is in a vapour state where it is compressed and the heat is expended using a condenser. This process is commonly referred to as the vapour compression cycle. There are a variety of ways whereby the heat is expended to the atmosphere such as water cooled towers or air cooled condensers. The refrigeration process also can be used to provide a cooling medium to water which can easily be reticulated throughout buildings to provide a temperature difference in the evaporator heat exchanger, these are commonly referred to as chilled water systems. A number of methods are utilised to distribute the conditioned air to the space. Constant air volume systems provide a fixed airflow regardless of changes in the conditions within the space. Variable air volume however modulates the airflow to respective areas based on sensing conditions within the space, typically these systems are coupled with variable speed fans on the air handling units.
Chilled Beams have been implemented in a number of buildings achieving classification of a 5 star rating. The systems typically supply chilled water through an exposed beam that conducts and radiates into the space. Care needs to be taken to ensure the temperature of the chilled beam does not drop below the dew point thereby eliminating any potential condensation within the space.

2.1.3 Internal Heat Load Sources

In order to develop an efficient micro climate air conditioning system it is essential to identify the factors that influence the heat load within a typical office space. The heat load can further be defined as either sensible or latent heat. Sensible heat refers to the heat energy that when added or removed from a surface will result in a measurable change in the dry bulb temperature. Latent heat however refers to the addition or removal of moisture from a substance. This may take the form of a change of phase of a substance eg. the boiling of water resulting in evaporation creating steam. The cooling load for a space is defined as the rate of heat that must be removed to maintain the required conditions, which are usually nominated as the dry bulb temperature and relative humidity.

The cooling load demand on an air conditioning system may consist of some or all of the following:

- Heat gain due to convection, conduction and solar radiation via external surfaces such as the roof, external walls, skylights and windows.
- Depending on the temperatures of adjacent spaces conduction may also transfer heat through partition walls, ceilings and floors.
- Internal sources that influence the heat load such as lights, people, equipment and appliances.
- Infiltration of outdoor air via windows, doors and small cracks in the building envelope.
- Outdoor air deliberately introduced into the space / system to achieve the required fresh air ventilation requirements.
- Heat gains associated with the fans and other mechanical equipment required for the air conditioning system.

The conduction identified in the above as influencing the heat load in the space can be approximated by the following equation.

$Q = U \times A \times \Delta T$

Where
Q = heat gain by conduction (W)
U = overall heat transfer coefficient ($W/m^2.°K$)
A = area of the surface
ΔT = dry bulb temperature difference across the surface

For composite surfaces the U factor equates to the inverse of the summation of the thermal resistances (R values) of each of the materials. The change in the azimuth angle of the sun throughout the day for different periods of the year will dramatically impact on the level of solar radiation on the external surfaces of the building. When the suns rays strike a surface at 90° the maximum heat energy is transferred to the surface whereas angles less than 90° refract a percentage of the heat energy away from the surface. The colour of the wall affects the emissivity and therefore also the heat energy exchange between the surface and the sun.
The operational hours of the office is also important in determining the internal heat load as the time lag for conduction of these external surfaces varies according to the ability of the structure to absorb heat. If for example the peak load on a space occurs in December at 3pm and the office only operates until 5pm, then if the thermal lag due to the structure exceeds 2 hours then the proposed system will not have to accommodate the peak load and hence the air conditioning plant may be reduced in size. Therefore as highlighted above the form of construction of the building inclusive of the materials used as well as the orientation of the external walls and use of shading devices on windows to reduce the solar heat load is very important in reducing the energy expended to air condition the space.

2.1.4 Thermal Comfort

W.P Jones (2001) defined comfort as ‘a subjective sensation that is expressed by an individual, when questioned, as neither slightly warm nor slightly cool.’ As with comfort there is no strict definition of indoor air quality (IAQ) however

Jones (2001) quoted ASHRAE Standard 62-1989 'for comfort, indoor air quality may be said to be acceptable if not more than 50% of the occupants can detect any odour and not more than 20% experience discomfort, and not more than 10% suffer from mucosal irritation, and not more than 5% experience annoyance, for less than 2% of the time.'

Comfortable indoor conditions depends on six variables namely air temperature, air velocity, relative humidity, radiant temperature, occupants clothing insulation and occupants activity level. High humidity promotes an environment whereby mould and bacterial growth can exist. The American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) recommend that relative humidity be kept under 60%. Low humidity levels can also cause irritations to the eyes and skin, and therefore lower levels should be maintained above 30%. Fisk (1981) defined a comfort equation for neutrality. Neutrality indicated that a person within the space is neither too hot nor too cold. The neutral equation is as follows:

$$\Theta rs = 33.5 - 3\ Icl - (0.08 + 0.05\ Icl)\ Ms$$

Icl = insulation value of clothing measured in 'clo' units
Ms = metabolic rate of the body

CFD analysis is capable of providing a predicative tool to enable cost effective analysis of alternative methods of providing comfort conditioning within buildings. The analysis often incorporates modifications to the building envelope to reduce the external heat load afforded due to solar and climatic conditions. This can be achieved by changing shading schemes or the internal fitout. Other analysis focus on the factors that affect the internal comfort which can be affected by the air distribution methods employed as well as the extent of internal heat generating equipment.

2.1.5 Existing Technology and Tests Conducted

Whilst research has been documented within the United States and Saudi Arabia and within selected locations throughout the Asian continents, detail specific to Australia and the south eastern areas of Queensland was limited. Therefore the information and data gained by referencing papers on foreign countries must be interpreted based on their relevancy to the local climatic conditions.

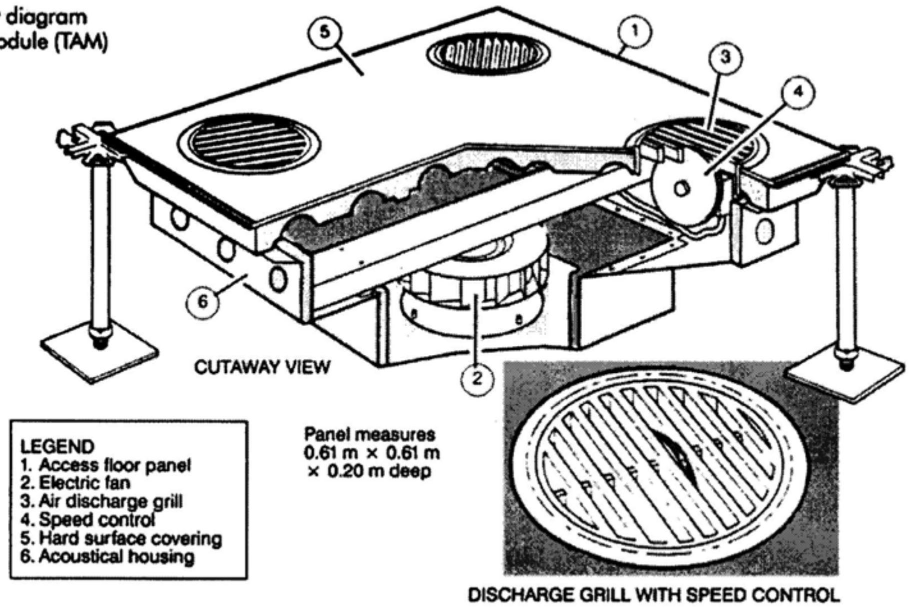
Tamara Erlandson et al (2003) conducted studies by using questionnaires on two cities which display distinctly different climatic conditions namely Kalgoorlie – Boulder (inland western Queensland) against Townsville which has a higher average relative humidity. The survey aimed to identify differences in responses to thermal comfort questions based on the climatic conditions of the city they live in. The results of the survey for Townsville showed a correlation between the level of work satisfaction and thermal sensation via the effectiveness of the air conditioning system. The Kalgoorlie – Boulder results indicated a correlation between thermal sensation due to a perceived lack of effective air conditioning and frequent illness. Townsville occupants appeared more adapted to their outdoor climatic conditions than Kalgoorlie – Boulder respondents, which may be due to limited residential air conditioning. The questionnaires of both locations indicated that the air movement in their respective workplaces was generally less than they would desire.

The findings of the study confirmed the theory that the use of appropriate clothing in the workplace in accordance with the outdoor conditions allows for a more efficient air conditioning system. This was evidenced by the Townsville respondents being more accommodating of fluctuations in temperatures.

Fisk et al., (1991) analysed task ventilation provided by way of underfloor air distribution (Tate System). The following describes the experimental setup and resultant findings from the tests.

A system referred to as the Tate System utilises adjustable supply air grilles mounted in the floor shown below in Figure 1.

Fig. 1 Cutaway diagram of a task air module (TAM)



As depicted in Figure 1 an Air Handling Unit (AHU) typically supplies 100% outside air to each of the sub floor supply air plenums. Fan Coil Units can be installed in areas with expected high internal heat loads to further cool the plenum air. A supply air plenum is formed via a metal framework suspending removable floor panels approximately 0.3 to 0.6m above the permanent floor. The task modules replace any given floor panel. The supply plenum can also house a fan which is controlled by a Variable Speed Drive (VSD) and adjusts its speed in response to the movement of recessed thumb wheels allowing the occupant to modulate the velocity of the supply air usually between 2 to 6 m/s. The grilles also can be rotated to deliver air in accordance with the occupants' requirements. The supply air temperature is a fixed temperature of 18°C which is approximately 5°C higher than the temperature in most U.S. HVAC systems. Air is typically exhausted from the space via ceiling mounted return air grilles connected to return air ducts or return air plenums. Tests were conducted on a standard 5.5m by 5.5m controlled environment chamber with three workstations and task air modules (TAM) installed within the partitioned confines of each of the workstations. The space had a ceiling height of 2.5m with an additional 0.6m for the underfloor plenum / ductwork and 0.5m for the ceiling void space. A number of tests were conducted on both furnished and unfurnished areas. The furnished test also included overhead lights with a total power of 500 W, 75 W of task lights at each workstation and also a personal computer emitting approximately 90 W each. A manikin was located in the seat of one of the workstations, using electric heater elements to simulate the 75 W typically released by an office worker.

The supply air was labelled with a tracer gas to measure the level of gas concentrations within the space and provide data by which the indoor airflow patterns could be analysed.

A summary of their findings are as follows.

1. A significant displacement flow pattern was evident when the task air module was supplied at the minimum air rate. As only one task air module was operated at one time during the tests, further tests would need to be conducted utilising all three TAM's analysing the displacement flow pattern.
2. The results of the age of air samples indicated that this system did not dramatically deviate from traditional ventilation methods. Based on the readings it was surmised that reasonable mixing of the room air resulted from the entrainment of room air with the air discharged from the supply air jets. It was noted that movement of occupants was not simulated in these tests and could vary the results expected in practice.
3. In almost all of the tests the air directed towards the occupant resulted in significantly lower age of the air as compared to other locations within the space. Therefore the supply direction is one of the most important operating parameters. A percentage of supply air should be directed toward the occupant to enhance the ventilation air supplied within their breathing zone. This is only relevant while the occupant remains seated at their workstation.
4. Overall the results indicate that with a supply air temperature of between 17°C to 19°C maintained a local temperature at the seated position of each workstation of 24°C.

Indoor Air Quality (IAQ) and thermal comfort in an office environment are important factors in achieving employee satisfaction and minimising absenteeism. Poor indoor conditions can affect the occupants' health, creating symptoms such as headaches, nose, throat, eye and skin irritation, nausea and drowsiness. Open plan offices typically are more densely populated than individual offices thereby creating higher internal heat loads and potentially more indoor contaminants. Open plan offices are usually controlled via a single thermostat negating the ability of occupants to individually control their local environment. Air distribution should be maintained with air velocities generally being below 0.2 m/s. Some research indicates that higher air velocities can provide more comfortable conditions especially in warm, humid conditions. With this it is also recommended that the occupants have control over their local air speed. ASHRAE standards nominate that the minimum outside air supply rate be 10 L/s.p (litres / second . person), which was reduced to 8.5 L/s.p.

NOTE: AS1668.2 2002 nominates 5 L/s.p however the current BCA calls AS1668.2 1991 which stipulates 7.5 L/s.p minimum. Therefore the current mandatory requirement is that nominated by the BCA (7.5 L/s.p). Personal systems can improve IAQ more when directed towards the occupants face however this solution can also provide discomfort from drafts.‘ Because of the opportunities for individual control, personal systems have been associated with improved satisfaction and thermal comfort, and a reduction in physical symptoms, as compared to traditional systems.’(9)

High efficiency filtration systems should also be used to eliminate outdoor contaminants from entering the conditioned space.

CHAPTER 3

3.1 Test Setup Description

3.1.1 Introduction

A micro climate system was installed in an Architectural office in Brisbane, providing an opportunity to test the effectiveness of the system by analysing actual temperatures in the space logged over a period of approximately four months. The following chapter describes the test setup.

3.1.2 Test Setup

The air conditioning system comprised of a direct expansion split system with the fan coil unit and air distribution ductwork mounted underside of the building. Appendix B shows a drawing of the open plan office space. The conditioned air is supplied via insulated plenums and delivered to the space by Ø50mm jets. The system has a capacity of 14kW and utilises inverter technology which varies the speed of the compressor and hence modulates the capacity of the system as required. The ducted system that was installed was a Daikin FDYP145DLV1 / RZP145DV1. A thermistor is located within the supply air ductwork to provide feedback to the system to control based on the setpoint selected on the wall mounted controller. The total air quantity according to the manufacturers data is 1070 L/s. Figure 2 below shows the installed ducted system.

Figure 2.

Two temperature and humidity data loggers were placed in the space to trend log the conditions both at the roof level and at a typical workstation. The data loggers are set to take readings at 5 minute intervals and are downloaded approximately every two weeks. One of the sensors is located on the workstation desktop approximately 500mm in front of the jet and 110mm below. The second data logger was placed on a beam at the roof level. The data loggers are a LASCAR EL USB which measures temperature ±0.1ºC and humidity to ±0.5%. The air conditioning system was set at 18ºC and the airflow at the jet was measured using an anemometer as 5m/s which equates to approximately 12.5 L/s. The system was set at a high fan speed. Each of the jets has the ability to adjust both the air quantity and the direction of airflow to meet the individuals' needs, see figure 3 below.

Figure 3.

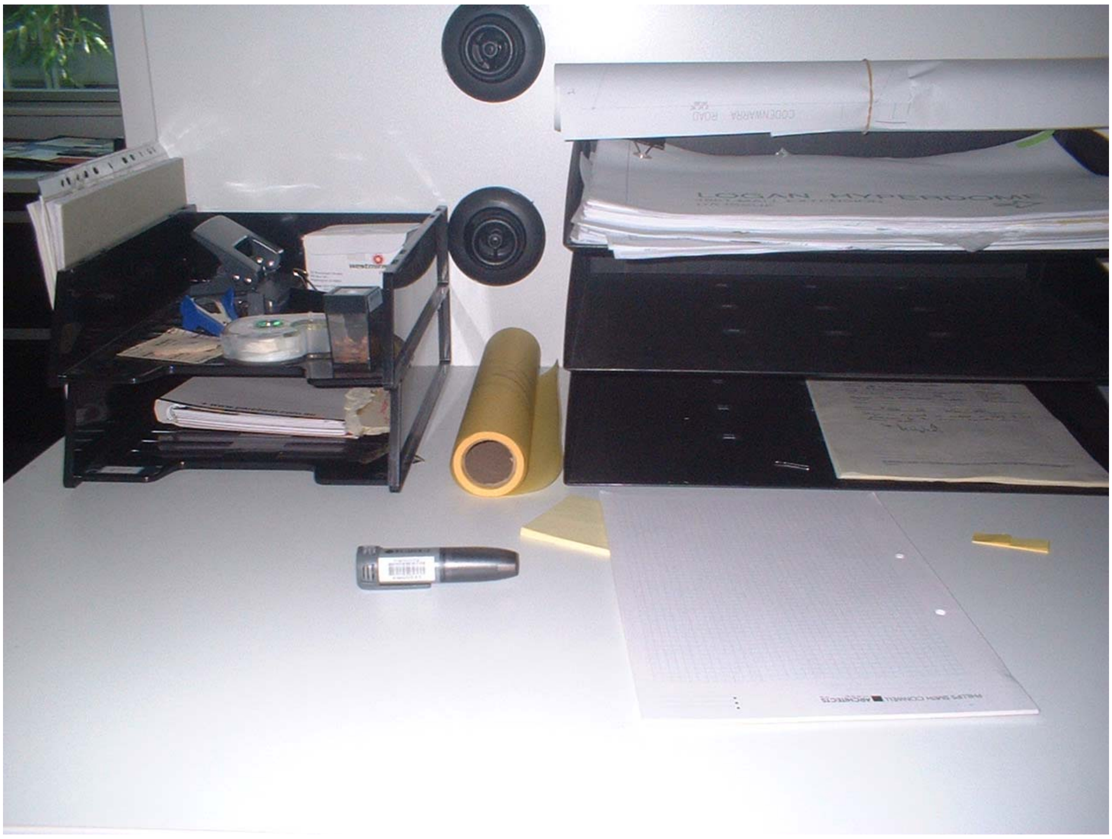


Figure 4 below shows the layout of the typical workstation.

Figure 4.

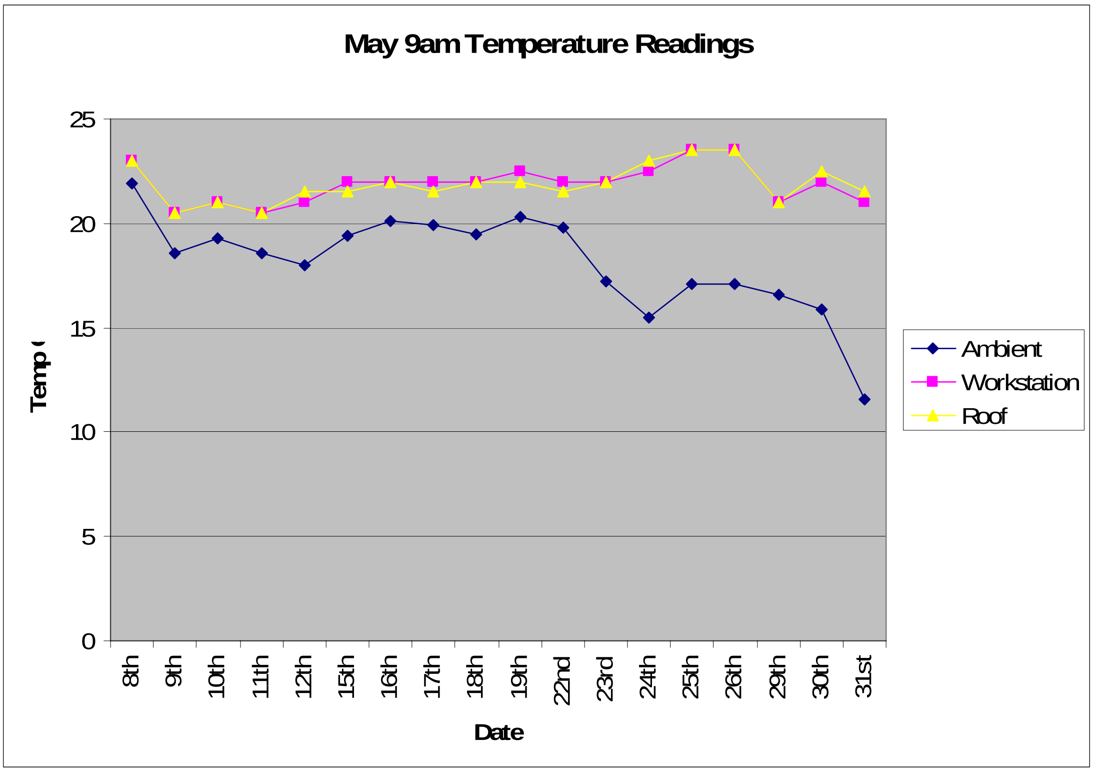
There are 30 workstations total in an open plan layout. Therefore based on two jets per workstation at 12.5 L/s at each jet the achieved supply air quantity was 750 L/s. Referencing the fan curve in the Daikin engineering data this equates to an external static pressure of approximately 150 Pa. Two relief air vents were installed in the roof, the damper was open at the point in time of testing. The intent of the air vents are to relieve the internal heat during cooling mode and are to be closed in heating mode to help contain the internal heat load. This was achieved via a manually operated damper which relies upon the occupants closing it in winter and reopening it in summer. This process could be automated if a supplementary control system was installed; however for an operation required only twice a year it was not warranted. Six 3.5kW wall mounted air conditioning units were installed on the southern wall to provide additional capacity for any peak loads that the micro climate system cannot handle.
A number of modifications have been made to the set up and control of the system to provide a better response in the space, which is further, described in the evaluation of the trend logs.

CHAPTER 4

4.1 Evaluation of Trend logging

The ambient temperatures were approximated using Brisbane meteorological data from the Australian Bureau of Meteorology website which provides daily weather temperatures at 9am and 3pm. The data was compiled from the tables which are included in appendix D. Appendix C includes all of the data logger output graphs. Figures 5, 6, 7 and 8 below provides graphs compiling the temperature taken from the data loggers at the workstation and roof levels compared against the ambient temperature at 9am and 3pm respectively for the duration of May 2006 to August 2006 inclusive.

Figure 5 May Temperature Readings



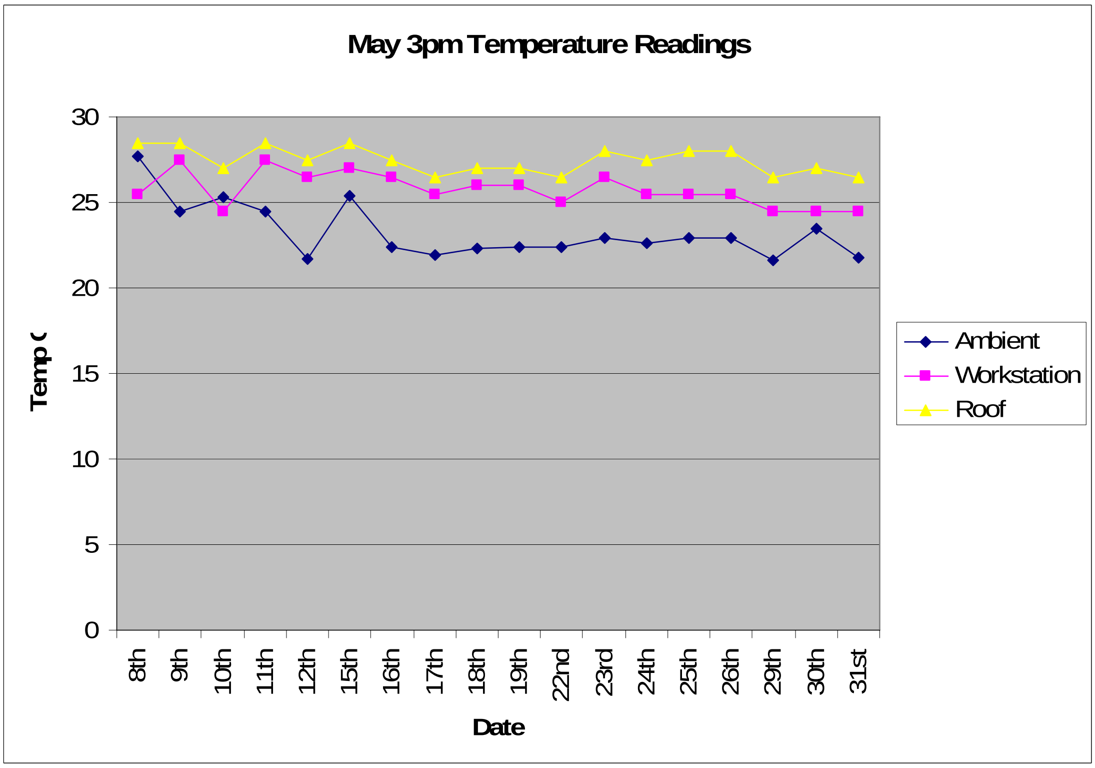
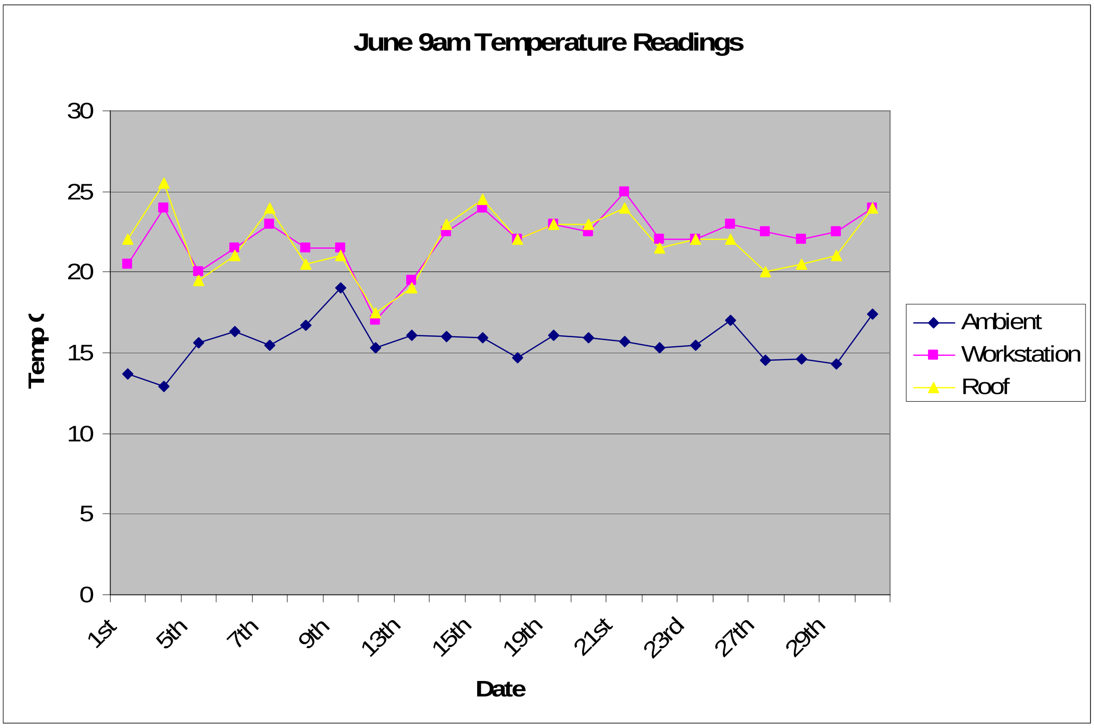


Figure 6 June Temperature Readings



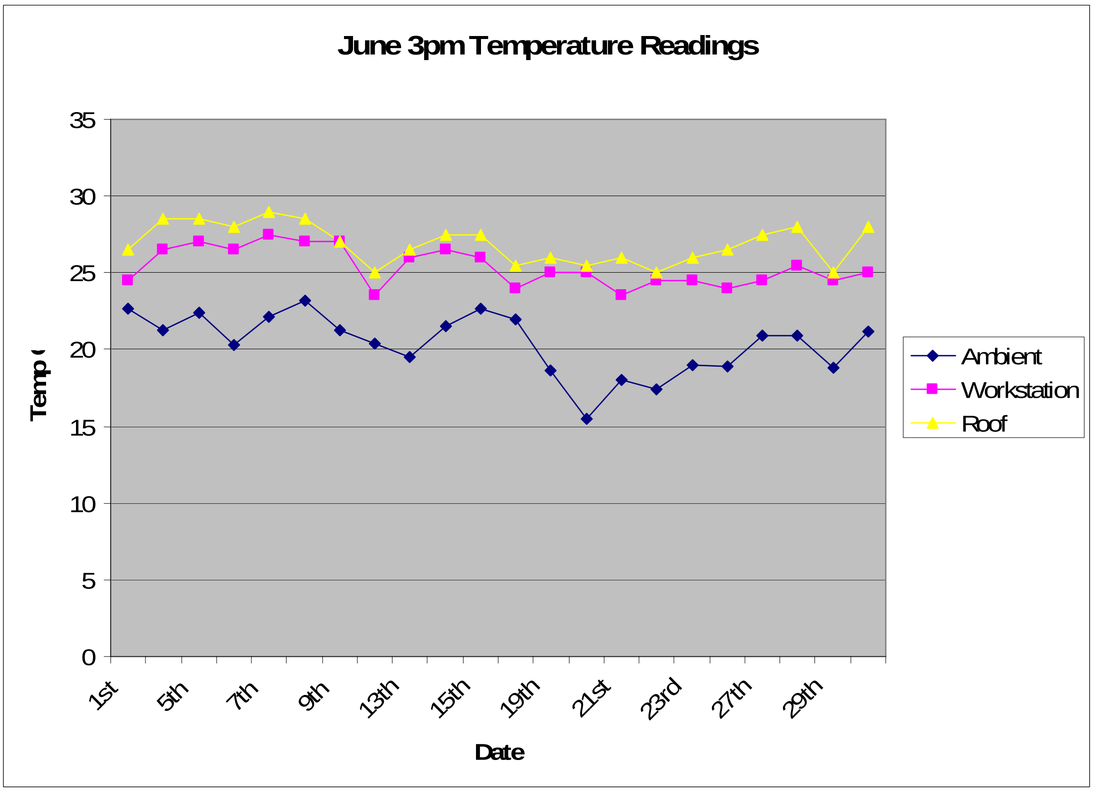
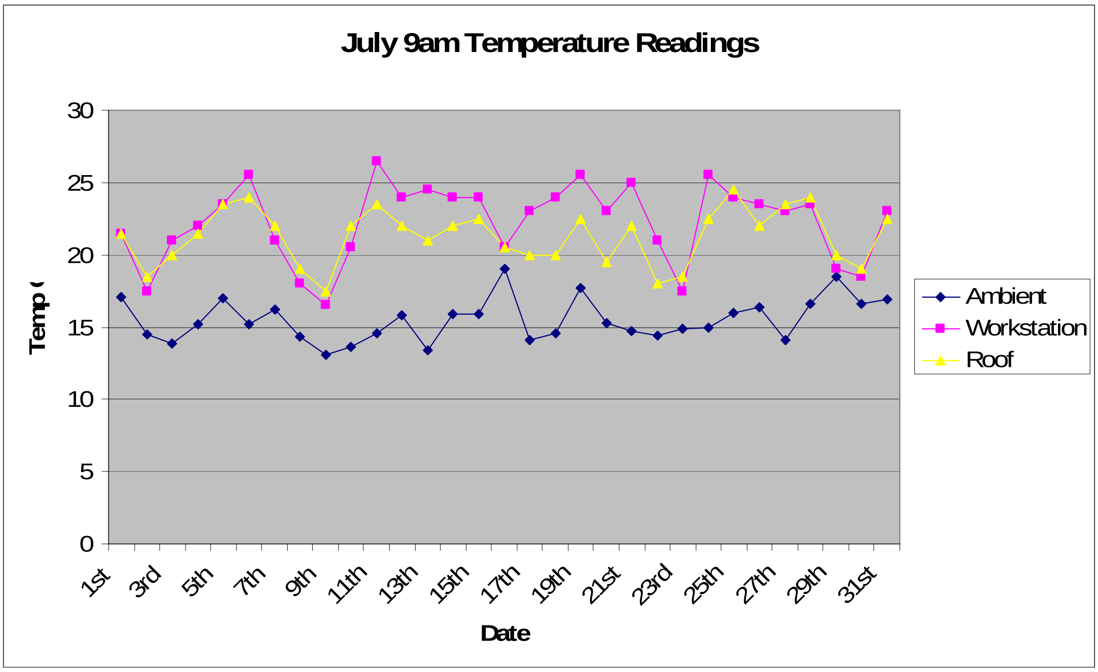


Figure 7 July Temperature Readings



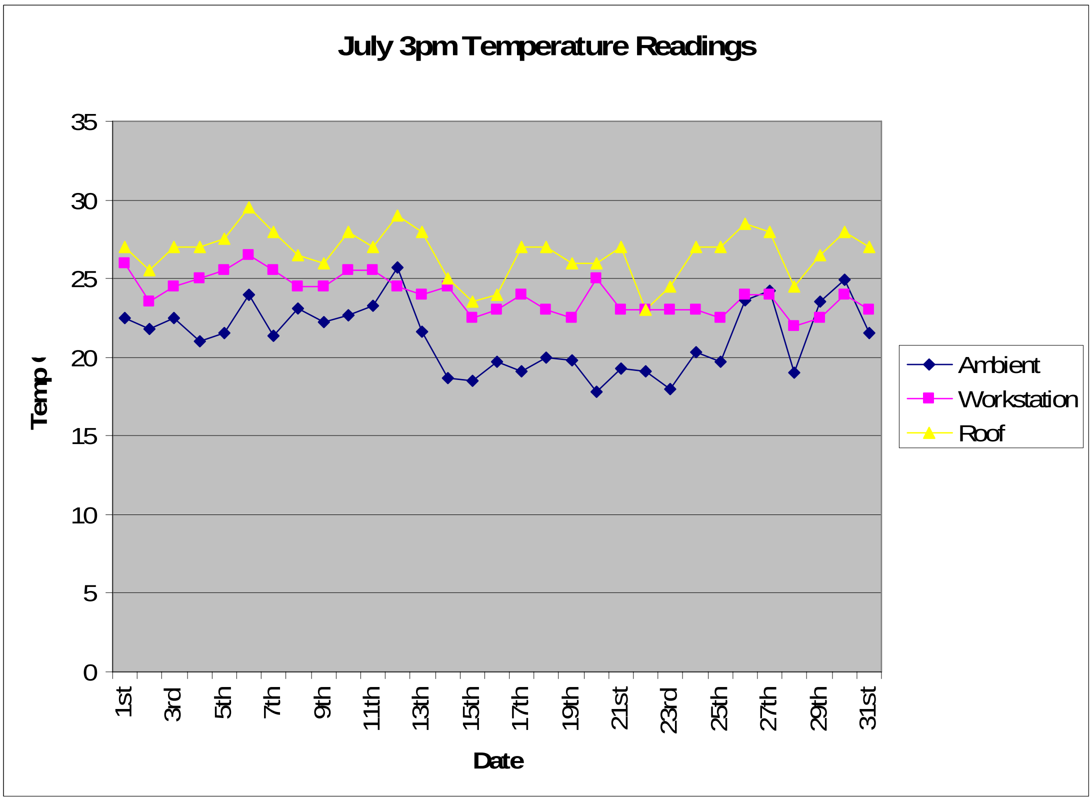
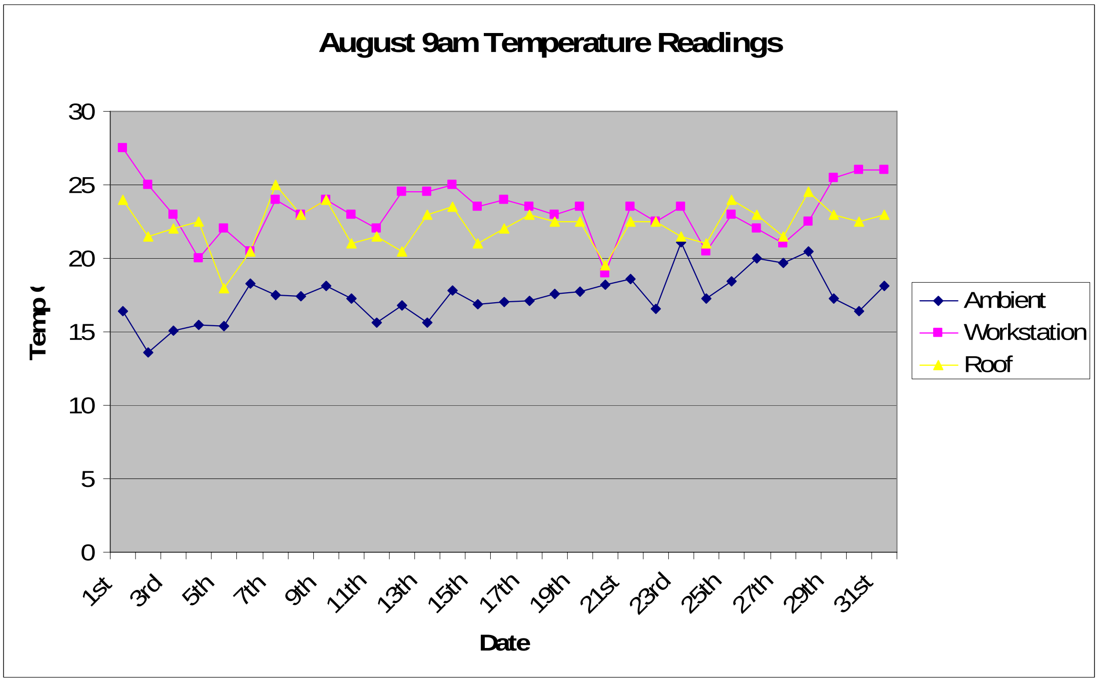
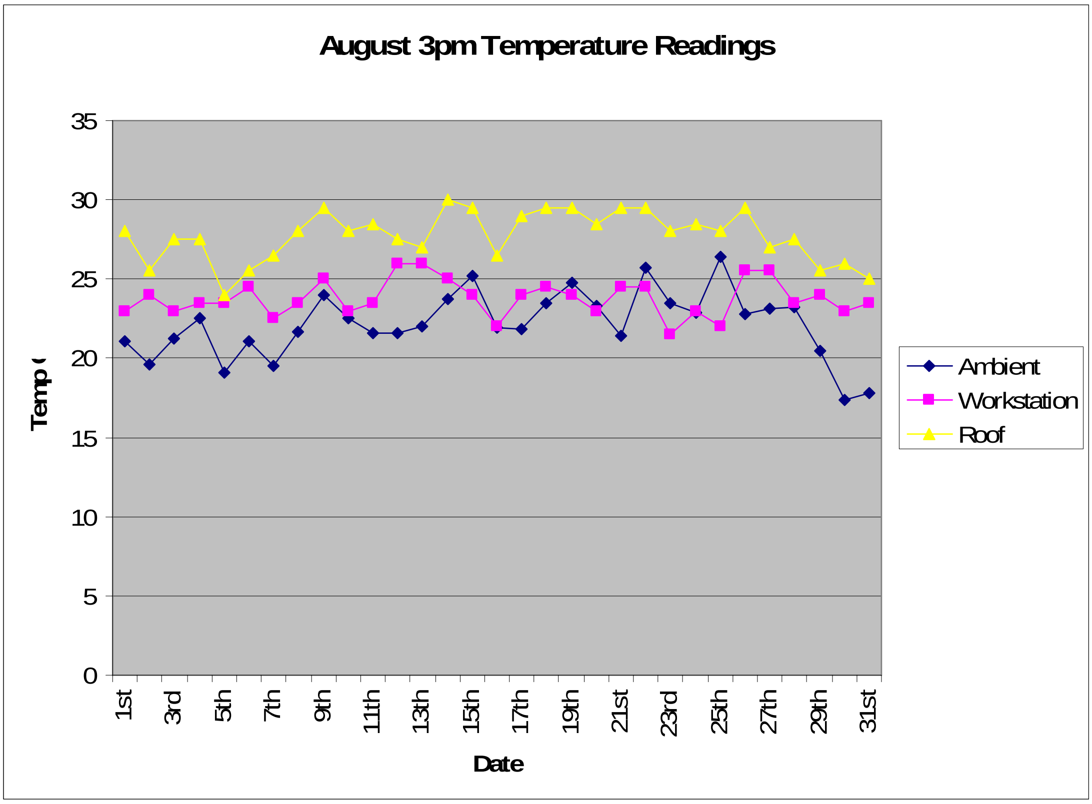


Figure 8 August Temperature Readings





From the graphs shown in Figures 5, 6, 7 and 8 it is apparent that for all the 9am readings the roof and workstation temperatures are reasonably close. Most of the ambient temperatures for 9am over the documented four months were in the range of 15ºC to 20ºC with the exception of July which had a number of readings under 15ºC. Typically at 3pm the roof temperature exceeded the workstation temperatures by approximately 2-3ºC. The most significant difference in temperatures was evident in the 3pm temperatures of August. The ambient temperatures at 3pm are predominantly between 20ºC to 25ºC. It was noted in the May and June readings (up until 20th June) that the relief vents were still in an open position thereby not retaining the internal heat load whilst in heating mode. The 3pm graphs after this date showed a slight average increase in the difference between the workstation and roof temperatures. Further scrutiny of the control system revealed an issue with the response of supply air temperature sensor. The Daikin optional sensor controls the air conditioning system until it is within 1ºC of the setpoint and then it averages the temperature of the supply air and the sensor housed in the wall mounted controller. A reading of the supply air temperature taken on the 6th June varied between 24°C to 33°C over a period of a few seconds. This variation was not evident from the trend logging because the readings were taken at 5min intervals. This control methodology caused the system to rapidly react to the supply air temperature causing the system to consistently overshoot the setpoint in heating and then cycle into cooling mode creating a every inefficient condition. An instantaneous temperature reading was taken at 6.15 pm on the 10th July showing the supply air temperature was at 30ºC with a general space temperature of 24ºC. The temperature logs for the roof in May and June show generally a trend of temperatures starting at approximately 18ºC at 8.00am and developing to a maximum of 26ºC to 27ºC at approximately 2.00pm. This then drops to approximately 16ºC around 5.00am to 6.00am. The same condition is seen for the workstation however the afternoon temperatures are approximately 2-3°C lower. These temperature changes appear to cycle in line with ambient conditions which can be evidenced by the fact that the building structure comprises light weight insulating media that provides little thermal mass. The lack of thermal mass does not provide any noticeable thermal lag.

Appendix C provides the data logger output graphs in their entirety. It was noted that the workstation readings on the data loggers had a noticeable change in the response after the supply air temperature sensor was disabled on the 10th July, in order to alleviate the problem with overshooting. From this day the workstation response appeared to average around 23ºC to 24ºC during the normal office hours of 8.00am to 6.00pm. The roof during the same period of time responded typically over a range of 17ºC to 27ºC.

A temperature difference of approximately 4ºC was measured on the 10/07/06 between the centre jets and the workstation located at the end of the duct run, i.e the supply air temperature was 4°C higher. This would indicate a substantial heat loss from within the ductwork. Perhaps a smaller run of duct distributed at a higher velocity would have minimised the losses in the duct. The workstations within the office space are constructed of MDF particle board, the void space between the workstations are the plenum for the supply air and were not insulated for two reasons, firstly the plenum is within the conditioned space and secondly it was assumed that the MDF would provide a reasonable insulator in itself and hence there was no need to further insulate the plenum. In feeling the lower levels of the workstations at various locations it was noted that they were quite cold perhaps indicating that some of the 4ºC loss may be also attributed to losses via conduction through or radiation from the particle board. The MDF had a higher density than expected hence any further systems of this kind should incorporate fully insulated supply air plenums.

Changes were made to the system to test the response in the colder months by adding some return air ductwork with a low level return air grille under a bench top located almost centrally in the space. The return air grille was 650mm x 650mm and in the installed condition had an average face velocity of 1.1m/s and therefore should deliver approximately 465 L/s of the air in the space back to the ducted system.

Static temperature readings were taken on the 15/08/06 showing that while the unit had a set point on the controller of 21°C the supply air at the jet was 16°C and the return air temperature was 26°C. The controller provides a readout of the temperatures that the system reads at either the controller or at the return air sensor (thermistor). On the above date the return air sensor was reading 24°C and the sensor in the controller was 29°C. The temperature measured in the space in general was between 27°C to 28°C. This indicates the variation in temperatures between the different sensors and the importance of selecting an appropriate location to sense from so that the system responds appropriately. Ideally for this application the system would provide a better response if the sensors did not average between the controller and return air (only using the return air for the winter conditions).

CHAPTER 5

5.1 CFD Model Description

5.1.1 CFD Software

CFD or Computational Fluid Dynamics has dated back as early as in the 1960's. It has only been in the 1980's through to the 1990's that the industry started to expand and with advances in computer technology it has provided a powerful computational tool. CFD software currently is used for predicting the outcome of changes in plant allowing greater efficiencies in designs of power plant, chemical reactors, processing equipment, vehicle and aeronautical design, just to name a few. It is expected in the future that medical research may utilise CFD analysis for predicting fluid or blood flow throughout the body or the brain.

CFD mathematically predicts fluid or heat flow through the use of mathematical models solved for each individual node or point on a mesh created by the software. The finer the mesh grid, the greater the computation time required to solve it.

Flair is a special purpose version of Phoenics which is designed specifically for heating / cooling and ventilating buildings. The original version of Phoenics was first available on the market in 1981.

5.1.2 Flair Model Description

Flair CFD software was used to model the open plan space, displaying the pressure, velocities and temperature distribution within the space based on the input data. Flair uses an editor package that allows data to be input in order to assemble a three dimensional model. It is within the editor that all of the properties are set for each individual component. Boundaries are set to constrain the domain also having its properties input via the editor. Once the model is developed complete with all of the components within the space the solver is run based on the mesh and number of iterations defined in the main menu. Whilst the solver is running a graphical display shows the variation of spot values as well as the change in error. The optimum outcome is to achieve a convergence of the spot values and a zero error change, hence a straight line on the graphical display. The estimated time required to either achieve the specified convergence or complete the number of iterations is also displayed. A number of assumptions were made in order to develop a workable model. The most significant assumption was that the shading on the building was such that there was no appreciable solar radiation through the thin line of windows. Infiltration was assumed for the purposes of the CFD model as being negligible, in reality a small percentage of outside air would leak through the building structure. The extent of infiltration would be dependant on how well sealed the building structure is. Obviously the model provides static representations of the space and therefore does not account for any transient movements of people throughout the open plan office. The aim of this project however is to test the response of a microclimate system on a typical workstation, the model developed also shows the effect of each individual workstation on the overall office space. Therefore the response can be estimated on both a micro and macro level.

The ambient conditions dictate the data that needs to be input to confirm the operation of such a system under various climatic conditions. For this reason the input data had to be modified for either summer or winter conditions.

5.2 Summer CFD Input Data

The summer input data was as described below:
Domain

- X dimension for the length of the building = 32m
- Y dimension for the width of the building = 9.8m
- Z dimension for the height of the building = 4.5m
- Buoyancy was turned on using the density difference method whereby a reference density of 1.29 kg/m^3 is used based on the density not being constant.
- The number of iterations was 100.
- Global convergence was set to 0.1%, which means that the solution will stop when the errors in all of the solved equations have fallen below the set value.
- Properties were selected as air using the ideal gas law.

Objects

- Walls are set as an adiabatic energy source with an initial temperature of 35°C.
- Floor was set as an adiabatic energy source with an initial temperature of 30°C.
- Ceiling was set as an adiabatic energy source with an initial temperature of 40°C.
- Jets are defined as an inlet with a velocity of 5 m/s and a supply temperature of 15°C. At a velocity of 5 m/s the jets deliver 12.5 L/s.
- Computer monitors were defined as being a heat source of 100 W with an initial temperature of 30°C.
- The people were denoted as in a sitting position and as a heat source of 160 W.
- The task lights were input as 100 W with an initial temperature of 50°C.

The data referenced above was taken from drawings of the office as well as estimates of heat loads associated with people and lights in the space. The following calculation was performed in order to estimate the expected supply air temperatures based on varying ambient conditions.

For the purposes of the analysis of a system that replicates the installed system the inlet supply temperature is dependant on the capacity of the air conditioning system. From the Daikin engineering data for the 14kW inverter ducted system (FDYP145DLV1 / RZP145DV1) for indoor conditions of 25°C DB and 18°C WB and an outdoor temperature of 30°C DB the total capacity is 14.5kW and a sensible capacity of 11.4kW.

Using the formulae ΔT = sensible capacity (W) / (Air Quantity (L/s) x density of air (1.213kg/m^3))

$\therefore \Delta T = 11400\ W / (750\ L/s \times 1.213 kg/m^3)$
$= 12.5°C$

Therefore for an ambient temperature of 27.5°C the supply air temperature at full capacity would be 15°C. When the ambient temperature is 32.5°C the supply air temperature is 20°C.

Note: A number of changes were made to the variables to see what effect it had on the space, some of which are listed below. The results are further described in Chapter 6 CFD evaluation.

1. The jets were modified to test the resultant model based on delivering 10°C air to the space instead of 15°C. This exceeds the capability of the installed system however it was tested to confirm how much of an effect it would have on the space.
2. The supply air velocity and hence the overall supply air quantity was increased. The temperature was maintained at 15°C and the velocity was changed from 5m/s to 7m/s. At a velocity of 7 m/s the jets delivered 17.5L/s. With 30 workstations and 2 jets per workstation the total air quantity increased from 750 L/s to 1050 L/s.
3. The outlets velocity was modified from 1.5m/s to 3.5m/s increasing the exhaust rate out of the 450 x 450 grilles from approximately 305 L/s to 705 L/s for each of the two grilles. This was to simulate the effect of installing exhaust fans on the roof instead of just relieving the pressurised air out of static vents.
4. A return air grille shown as an outlet at low level in the space was installed. The grille size was 650 x 650 and the velocity was set at 1.1m/s.
5. The supply air temperature was changed to 20°C to simulate an ambient temperature of 32.5°C.

5.3 Winter CFD Input Data

The winter input data was as described below:

Domain

- X dimension for the length of the building = 32m
- Y dimension for the width of the building = 9.8m
- Z dimension for the height of the building = 4.5m
- Buoyancy was turned on using the density difference method whereby a reference density of 1.29 kg/m^3 is used based on the density not being constant.
- The number of iterations was 100.
- Global convergence was set to 0.1% which means that the solution will stop when the errors in all of the solved equations have fallen below the set value.
- Properties were selected as air using the ideal gas law.

Objects

- Walls are set as an adiabatic energy source with an initial temperature of 10°C.
- Floor was set as an adiabatic energy source with an initial temperature of 10°C.
- Ceiling was set as an adiabatic energy source with an initial temperature of 25°C.
- Jets are defined as an inlet with a velocity of 5 m/s and a supply temperature of 30°C. At a velocity of 5 m/s the jets deliver 12.5 L/s.
- Computer monitors were defined as being a heat source of 100 W with an initial temperature of 30°C.
- The people were denoted as in a sitting position and as a heat source of 160 W.
- The task lights were input as 100 W with an initial temperature of 50°C.

The inlet supply temperature for heating is dependant on the heating capacity of the air conditioning system i.e in reverse cycle. From the Daikin engineering data for the 14kW inverter ducted system (FDYP145DLV1 / RZP145DV1) for indoor conditions of 20°C DB and an outdoor temperature of 10°C DB the total capacity is 16.6kW and a sensible capacity of 13.1kW.

Using the formulae ΔT = sensible capacity (W) / (Air Quantity (L/s) x density of air (1.213kg/m^3))

$\therefore \Delta T$ = 13100 W / (750 L/s x 1.213kg/m^3)
= 14.4°C

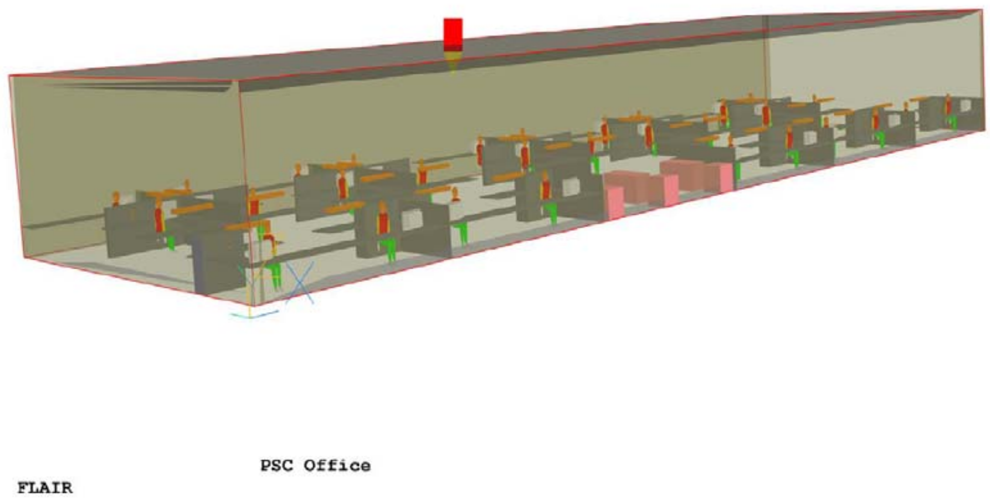
Therefore for an ambient temperature of 10°C the supply air temperature at full capacity would be 24.4°C.

CHAPTER 6

6.1 CFD Model Evaluation

The full model as depicted in the editor can be seen in figure 9 below.

Figure 9 Model Displayed in the Editor

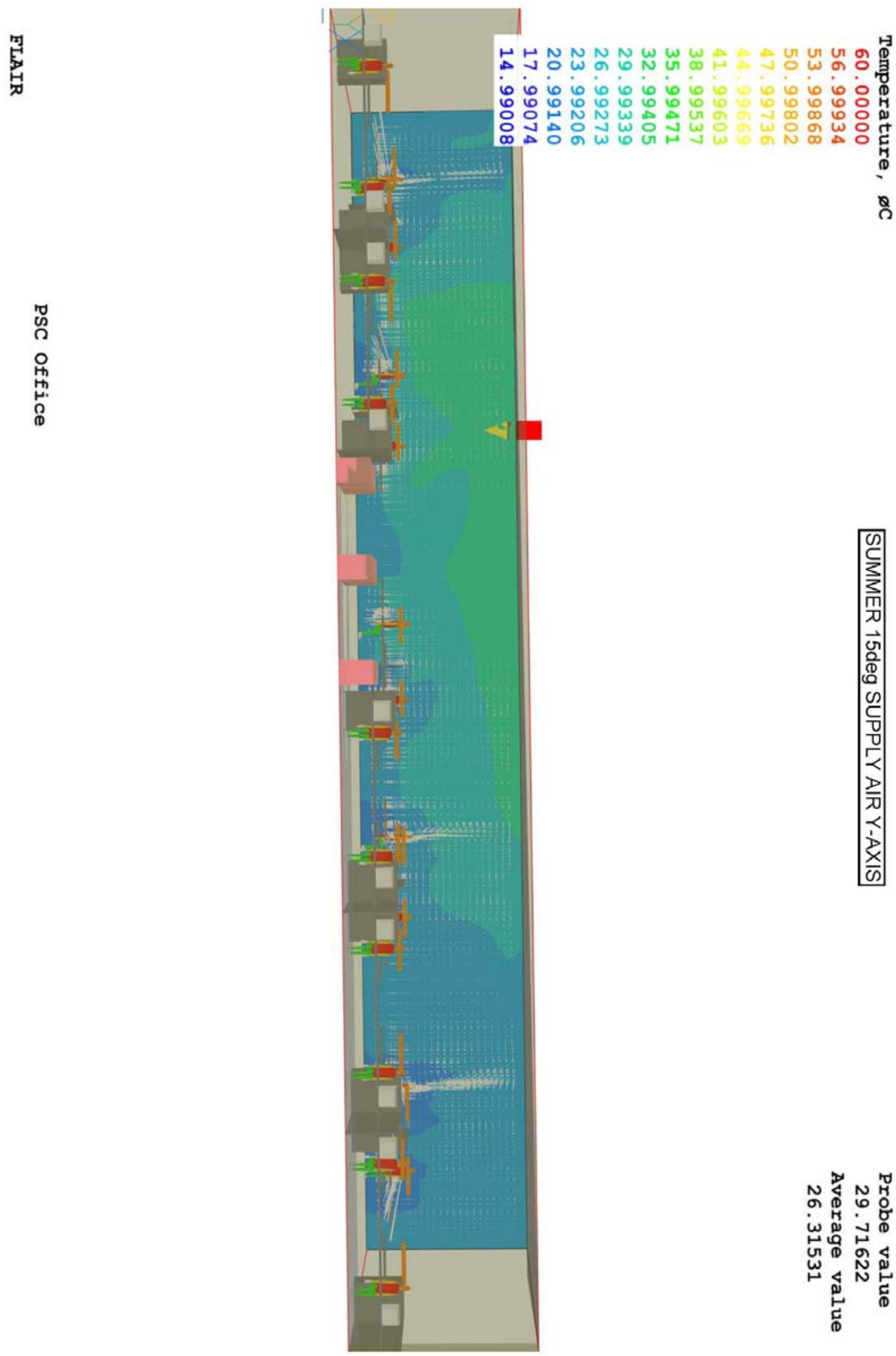


Temperature based on Summer Conditions

At the conditions measured in the space i.e. 15°C supply air at 5 m/s velocity at each jet, the summer output generally shows a temperature distribution varying between 15°C to 35°C. The results indicated that the temperatures were at 17°C up to approximately 0.5m, increasing from 23°C to 26°C at 0.5m up to 1.0m. From 1.0m to 1.5m up to the ceiling the temperatures increase from 26°C up to 35°C. The temperatures at the upper head height of the workstations are higher than what would typically be expected for comfort conditions. The corridor area towards either end of the building was between 27°C and 28°C. Temperatures of up to 38°C were shown above the area with the equipment.
The displayed flair output for these conditions are shown in Figures 10 and 11 below.

Figure 10 Summer Output 15ºC Supply Air X Axis

Figure 11 Summer Output 15ºC Supply Air Y Axis



Decreasing the supply air temperature to 10°C had the expected result of lowering the temperature at low level however the temperature at around 1.2m is still elevated at approximately 29°C. The lower level temperatures of up to approximately 1.2m local to the workstations ranged between 16ºC to 19ºC. The CFD model was based on the assumption that the airflow from the respective supply air jets was directed in either the positive or negative Y direction depending on the location on the workstation. The jets however are moveable and hence the actual 29ºC line could be higher than 1.2m in practice. Also the capacity required to deliver 10°C supply air would increase the size of the system to 18.2 kW sensible based on an ambient temperature of 30°C DB. That is provided the system can deliver a low enough air off the coil temperature to achieve a 10°C supply. The displayed flair outputs for these conditions are shown in Figure 12 and 13 below.

Figure 12 Summer Output 10°C Supply Air X-Axis

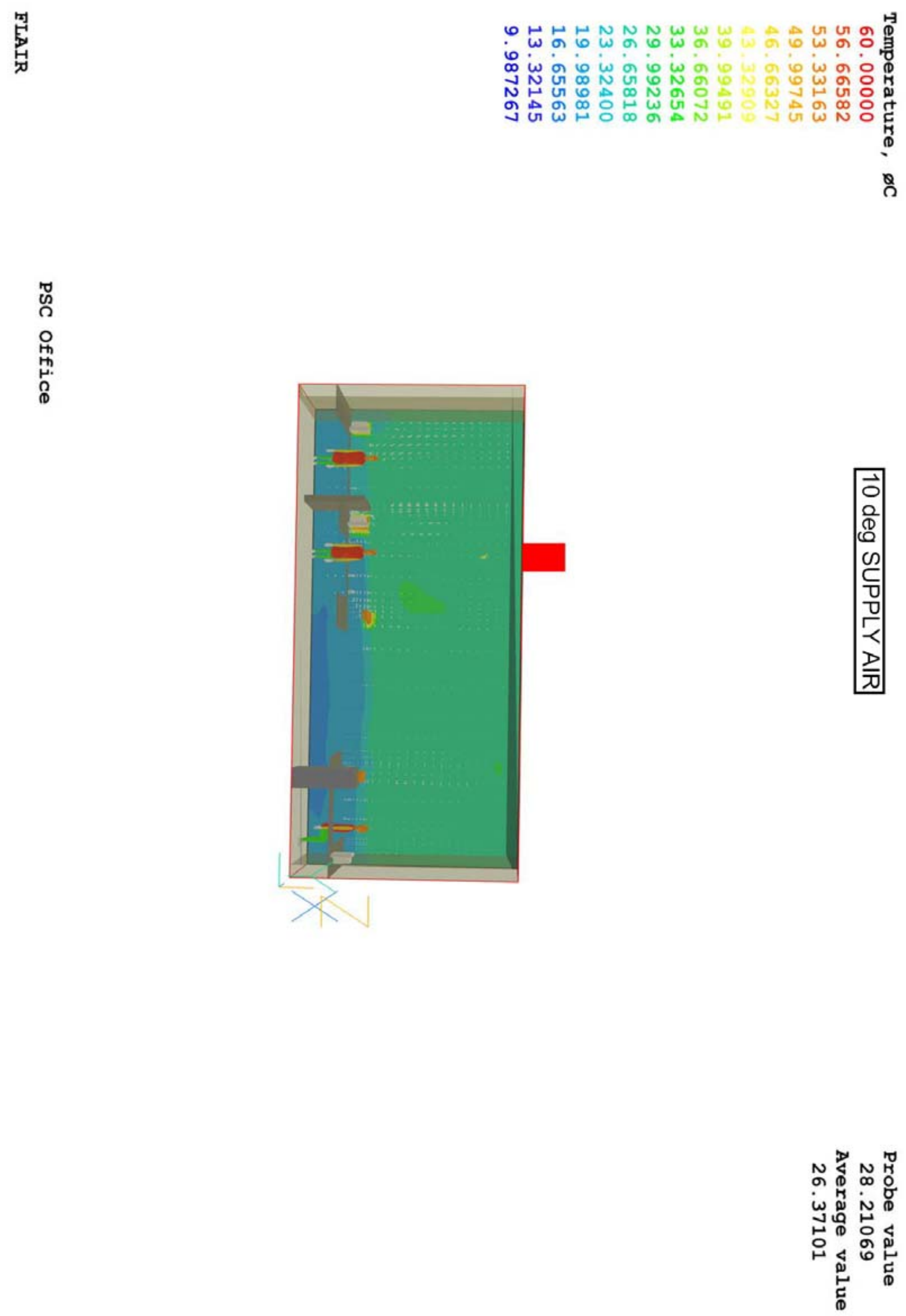
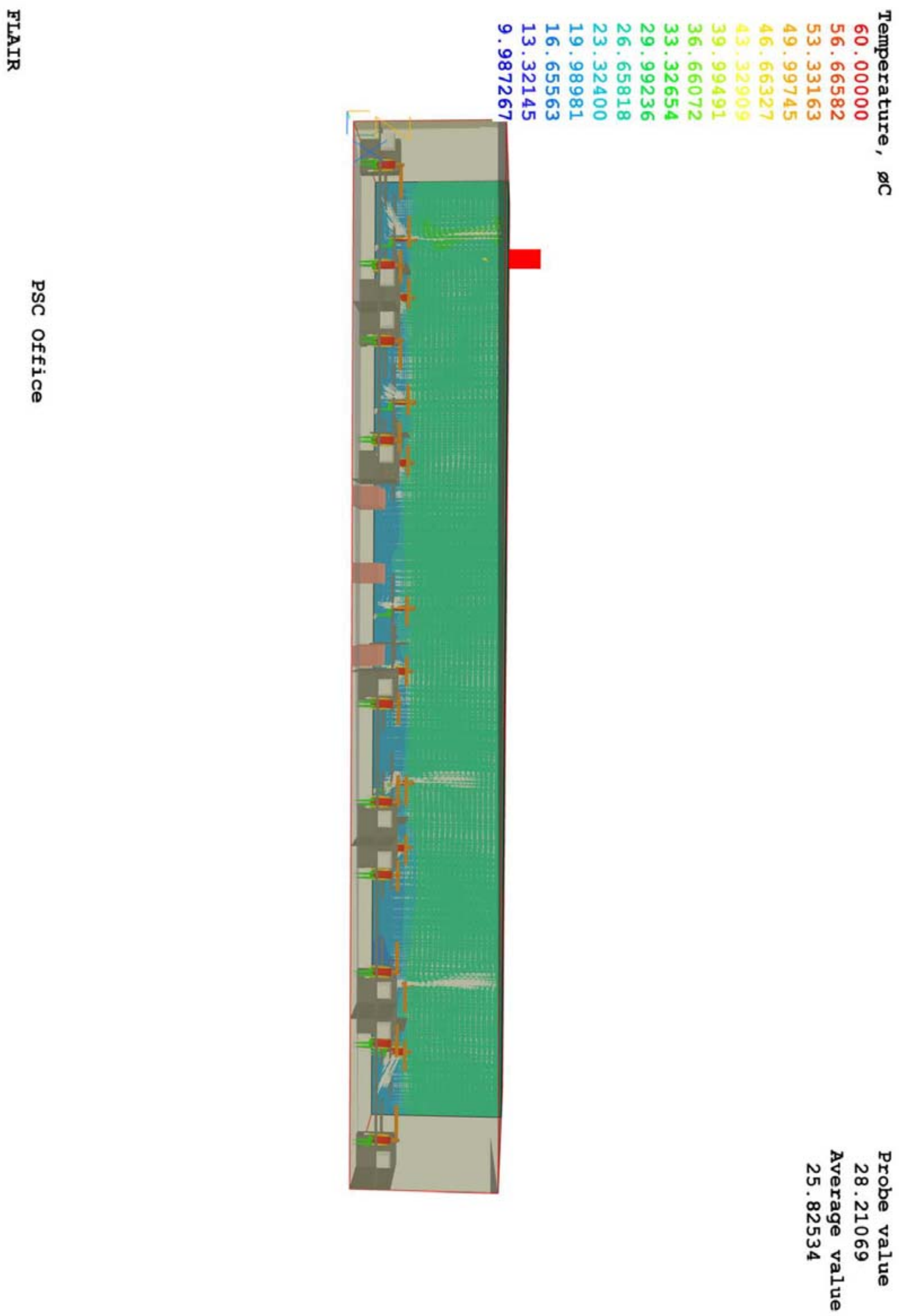


Figure 13 Summer Output 10ºC Supply Air Y-Axis



Increasing the supply air velocity from 5 m/s to 7 m/s maintaining a supply temperature of 15ºC dramatically modified the temperature distribution in the space. At 7m/s the supply air quantity at each jet is approximately 14 L/s requiring a total air quantity of 840 L/s for the office. The temperature variation was between 15°C to approximately 30°C. The temperature range was 23°C to 26°C at approximately 1.5m showing a dramatically improved thermal range for the workstation spaces. The follow on effect of increasing the supply air quantity would be the need for a greater capacity system.

The displayed flair outputs for these conditions are shown in Figure 14 and 15 below.

Figure 14 Summer Output 15ºC Supply Air at 7m/s

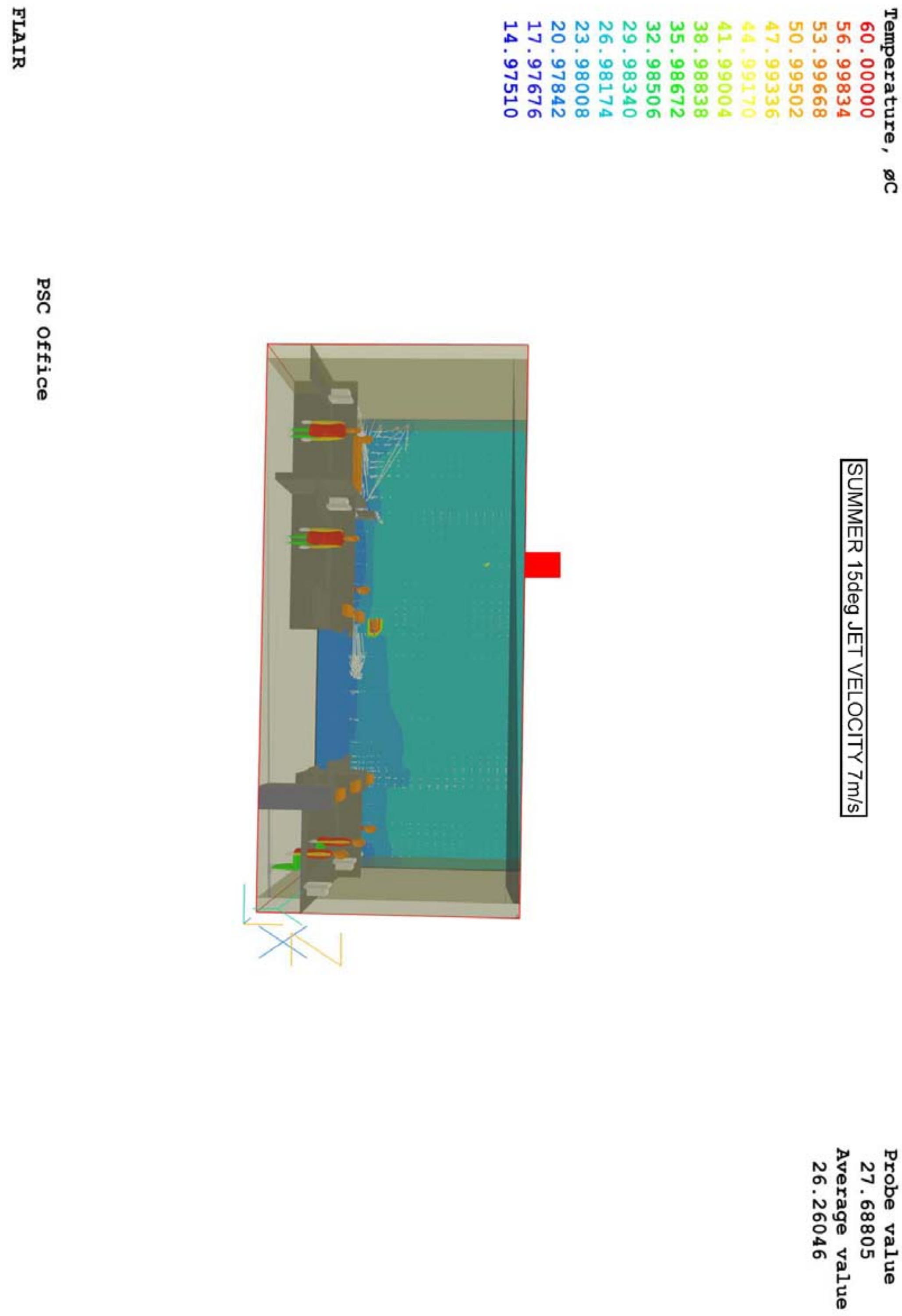
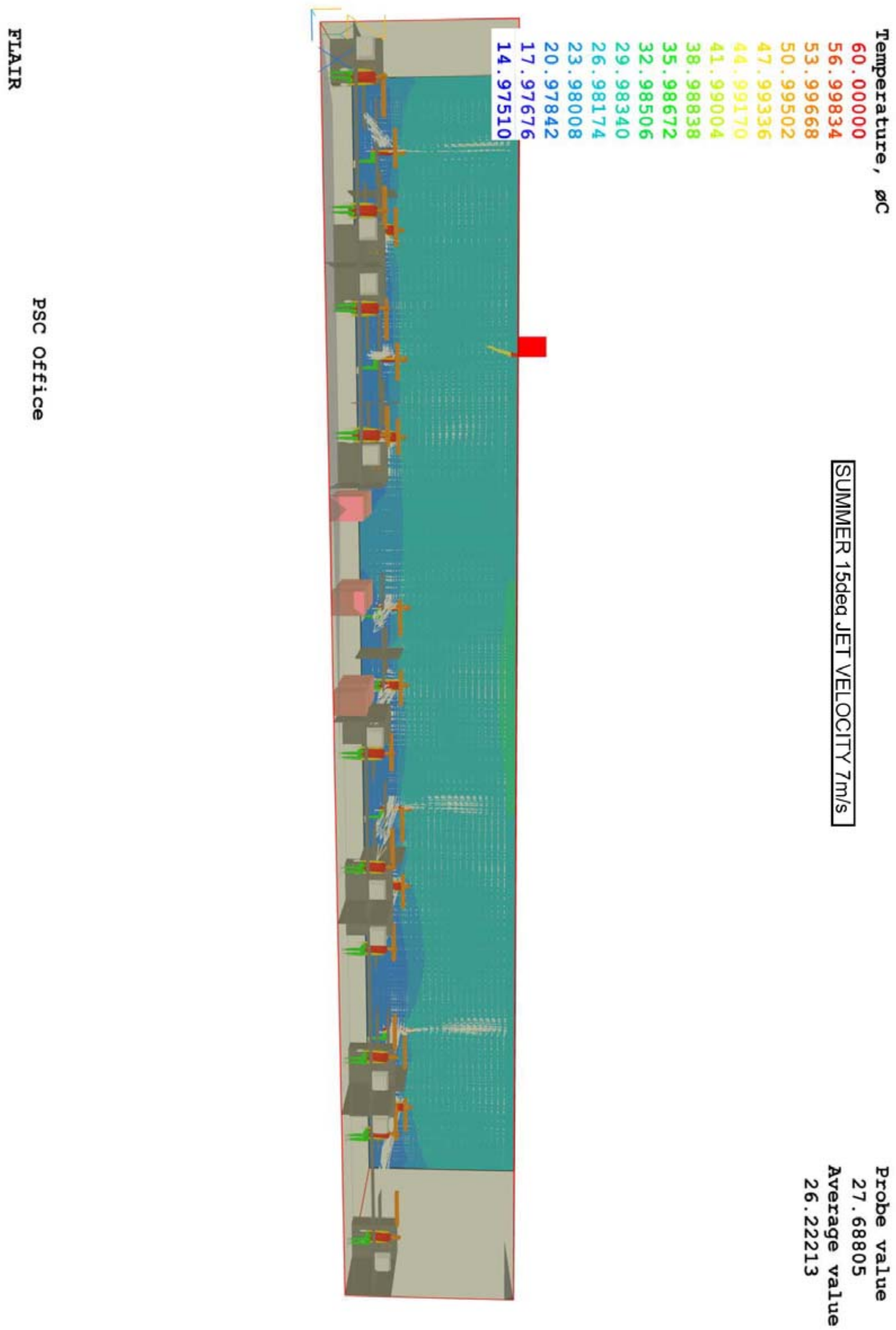


Figure 15 Summer Output 15ºC Supply Air at 7m/s Y-Axis



Increasing the outlet velocity to 3.5m/s had the overall effect of increasing the comfort zone i.e. temperatures at head height was typically between 23°C to 26°C. The response was not as effective as lowering the supply air temperature to 10ºC. The modification to the CFD model however did not account for the infiltration into the space as make up air which would be at ambient conditions. Therefore the actual space would exhibit slightly higher temperatures in reality. This model was performed to test the effect of installing an exhaust fan instead of only having static vents. At 3.5m/s for the 450mm x 450mm exhaust grilles equates to exhausting 710 L/s for each grille. This in effect would induce approximately 670 L/s of outside air into the space and has not been accounted for in this model, being the difference between the total exhaust and supply air quantity.
The displayed flair outputs for these conditions are shown in Figure 16 and 17 below.

Figure 16 Summer Output 15°C Supply Air at 5m/s Outlet 3.5m/s

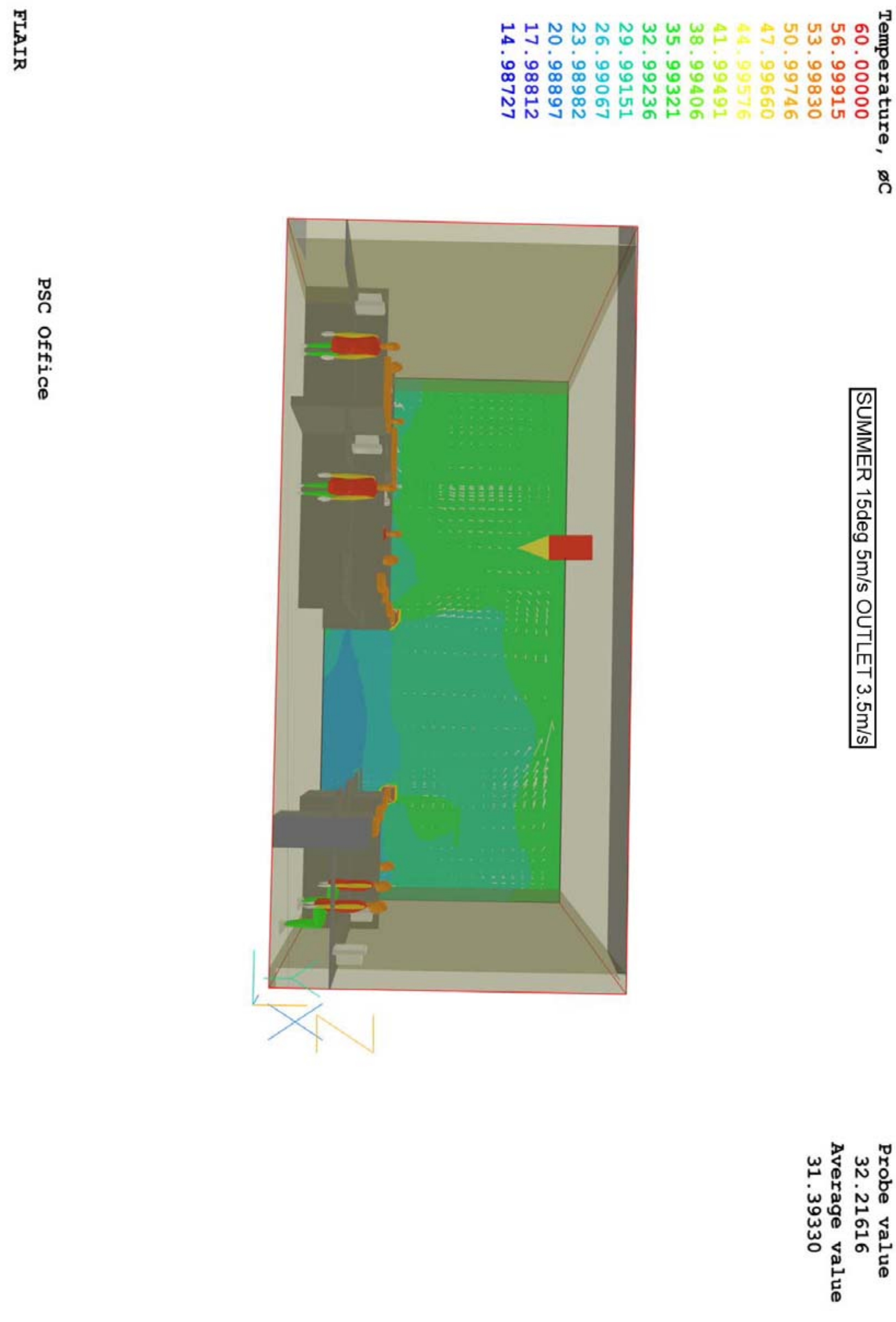
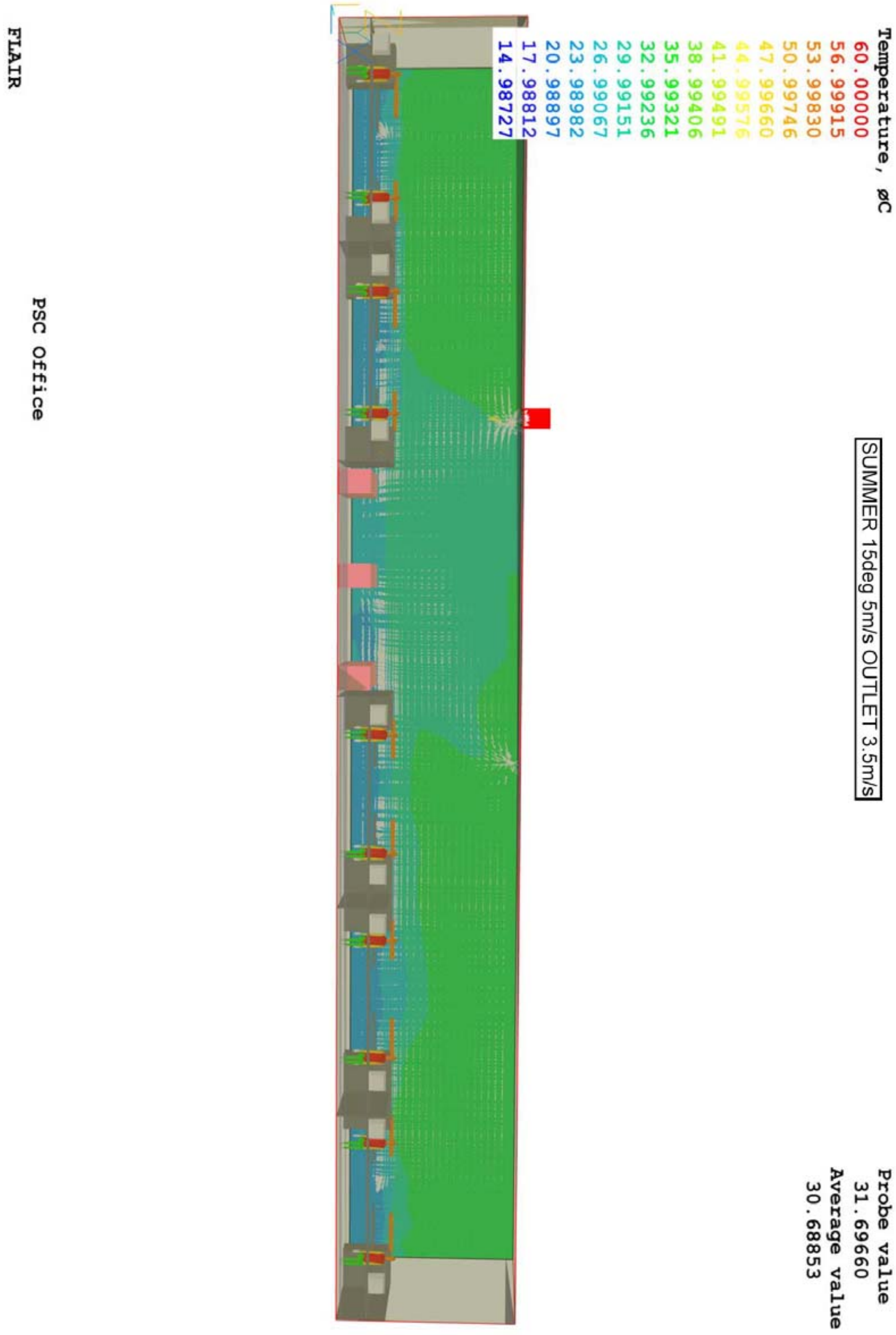


Figure 17 Summer Output 15ºC Supply Air at 5m/s Outlet 3.5m/s Y-Axis



The supply air inlets set to 20°C had the predictable result of increasing the temperature in the space in general. In particular the region from 0.8m to 1.2m was typically around 27°C to 33°C. The average temperature over the entire volume was 47°C. A slice taken through the Y axis depicts the higher temperatures being present showing the warmer air movement towards the exhaust grille outlets. The 20ºC supply air would depict the expected response to a 32.5ºC ambient temperature.
The displayed flair outputs for these conditions are shown in Figure 18 and 19 below.

Figure 18 Summer Output 20ºC Supply Air at 5m/s X-Axis

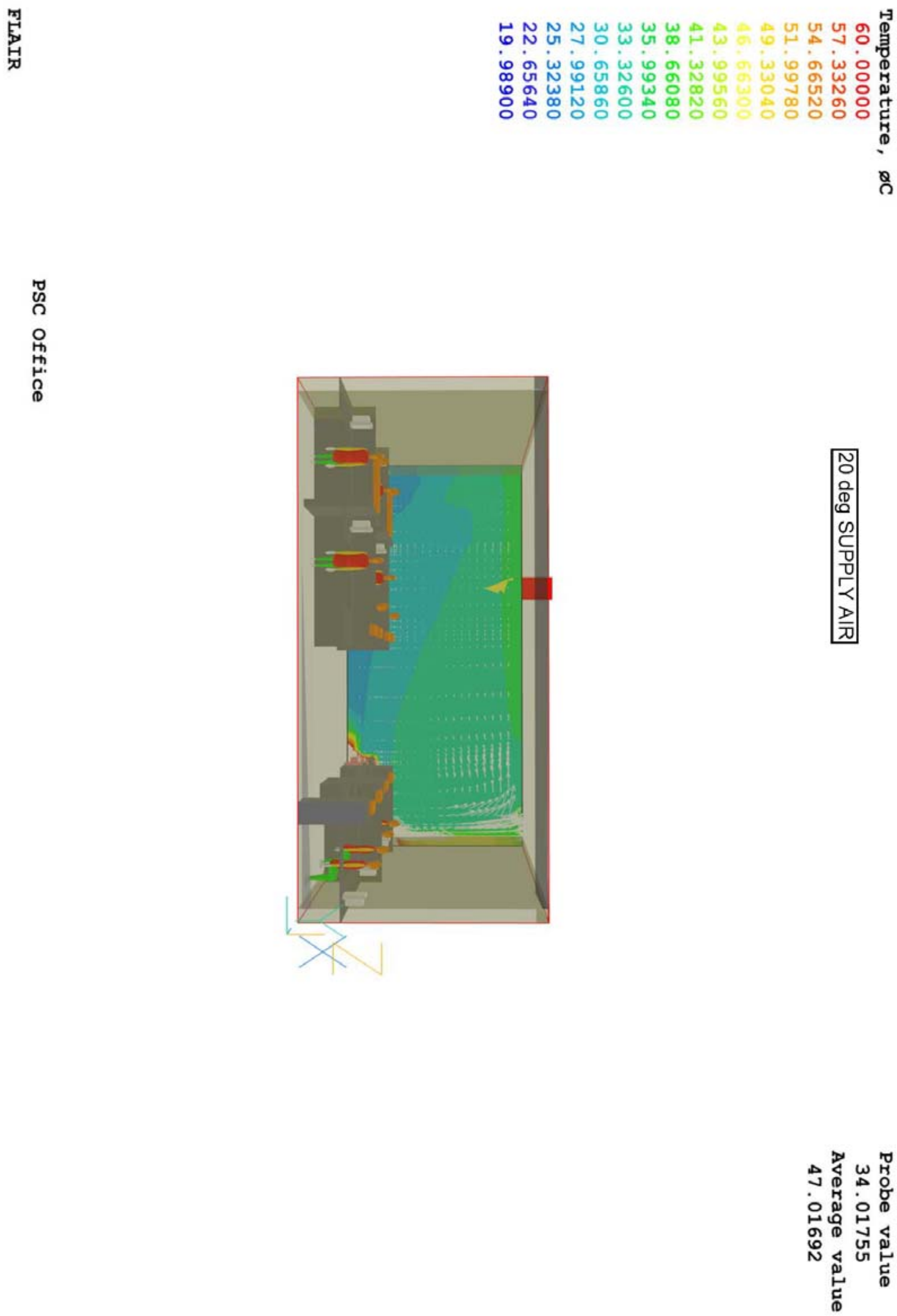
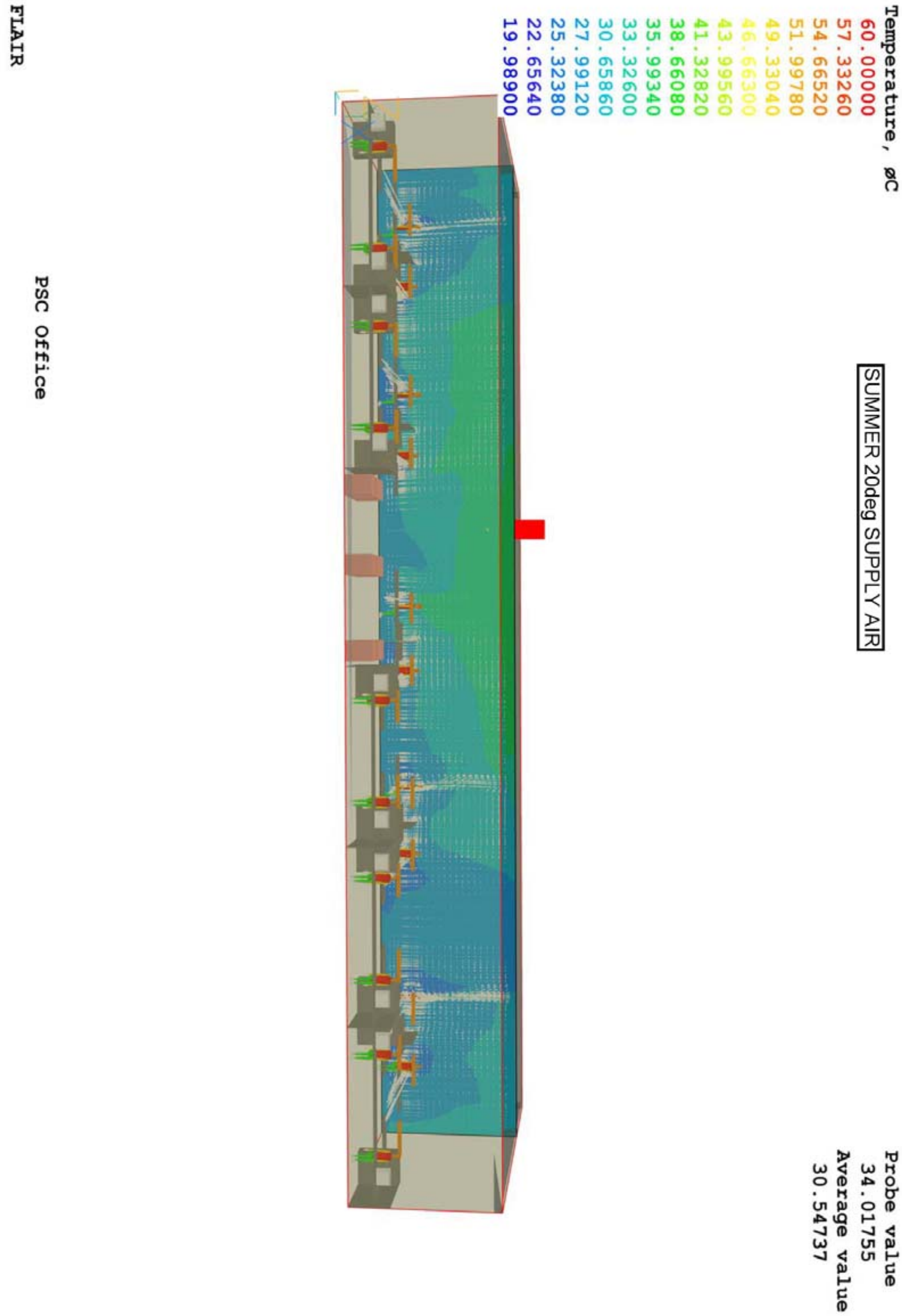


Figure 19 Summer Output 20ºC Supply Air at 5m/s Y-Axis



Temperature based on Winter Conditions

The winter CFD models depicted in Figures 20 to 23 generally provided results indicating the temperature in the space would become quite warm. In particular the results shown in figures 20 and 21 for the condition of 25ºC supply air ranged between 31ºC to 45ºC. The areas typically around the workstations were between 31ºC and 38ºC. There clearly was a discrepancy in this model and has therefore been disregarded. This was further evidenced by the fact that the output shown in figures 22 and 23 displayed lower temperatures generally in the space using a supply air temperature of 30ºC. The CFD model of the 25ºC supply air at winter conditions did not have any allowance for relief out of the space. In reality the building structures are not completely sealed allowing some leakage of air. Without allowing for this condition the results are incorrect hence the uncharacteristic models shown in Figure 20 and 21. For the 30°C supply air the temperatures around the workstations was typically between 27°C to 33°C. The higher level temperatures increased accordingly with temperatures up to 41°C at the roof. The 30°C supply air temperature theoretically would be achievable with ambient temperatures as low as 15°C.

Figure 20 Winter Output 25ºC Return Air at 1.1m/s X-Axis

Figure 21 Winter Output 25ºC Return Air at 1.1m/s Y-Axis

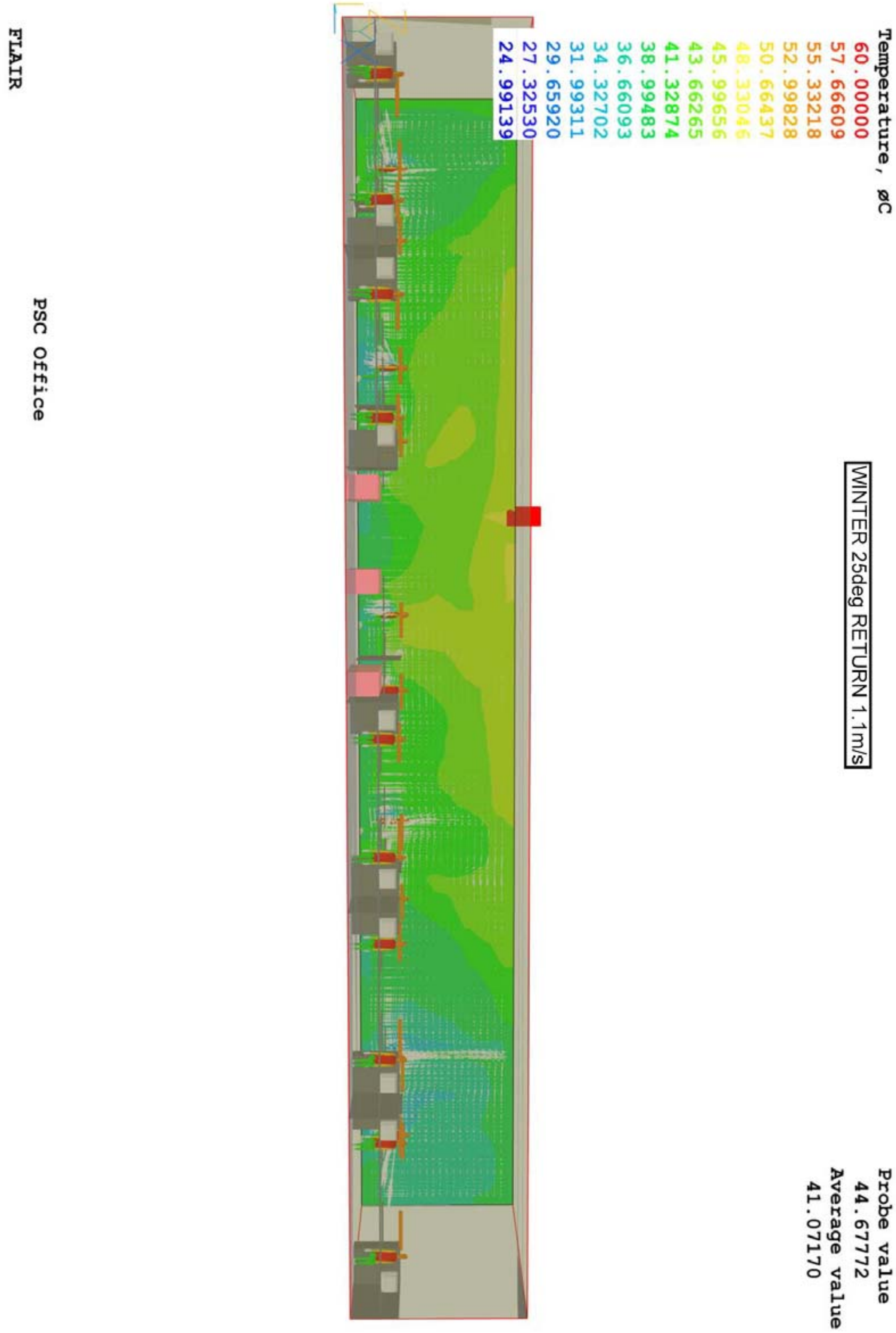


Figure 22 Winter Output 30ºC Return Air at 1.1m/s X-Axis

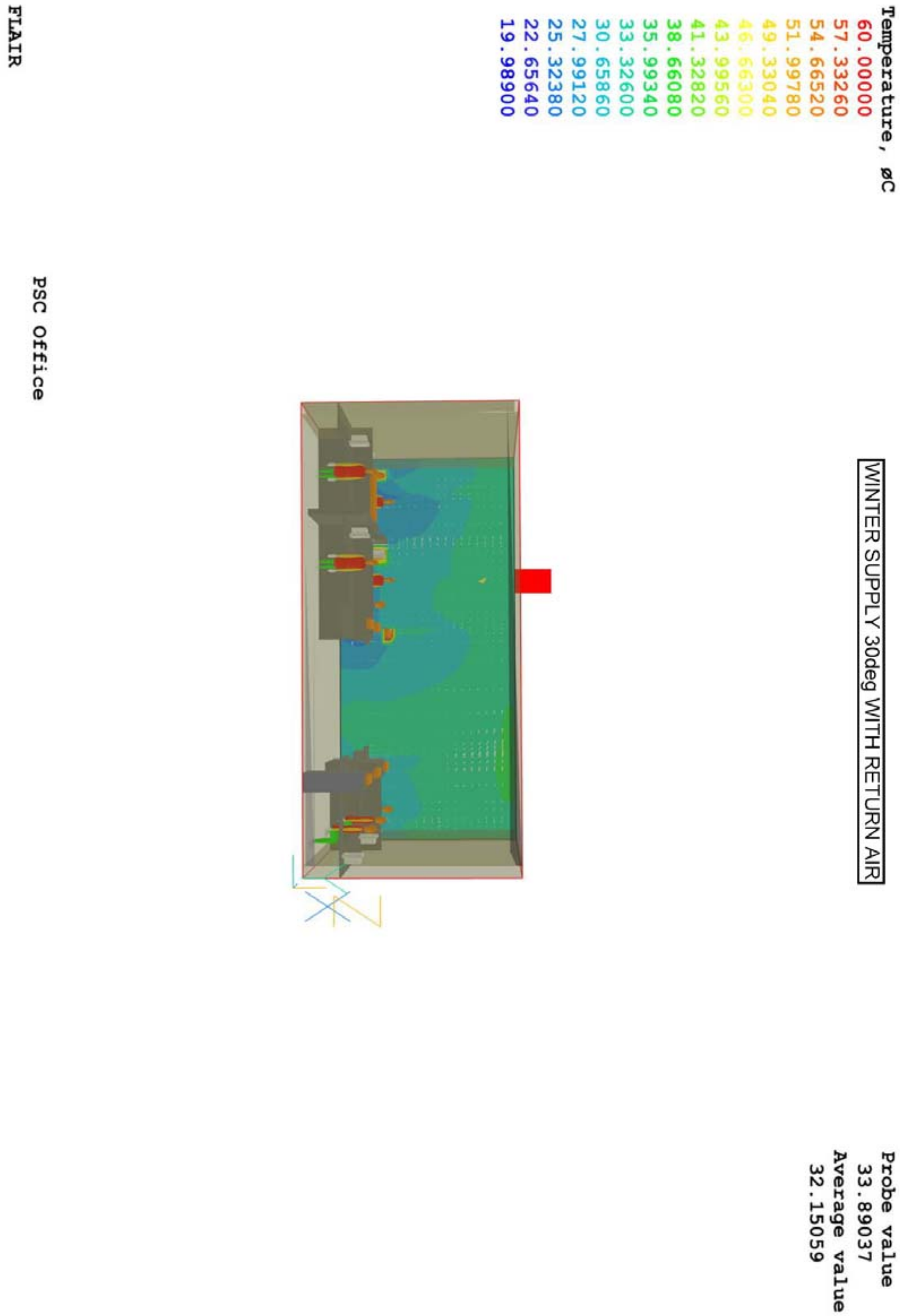
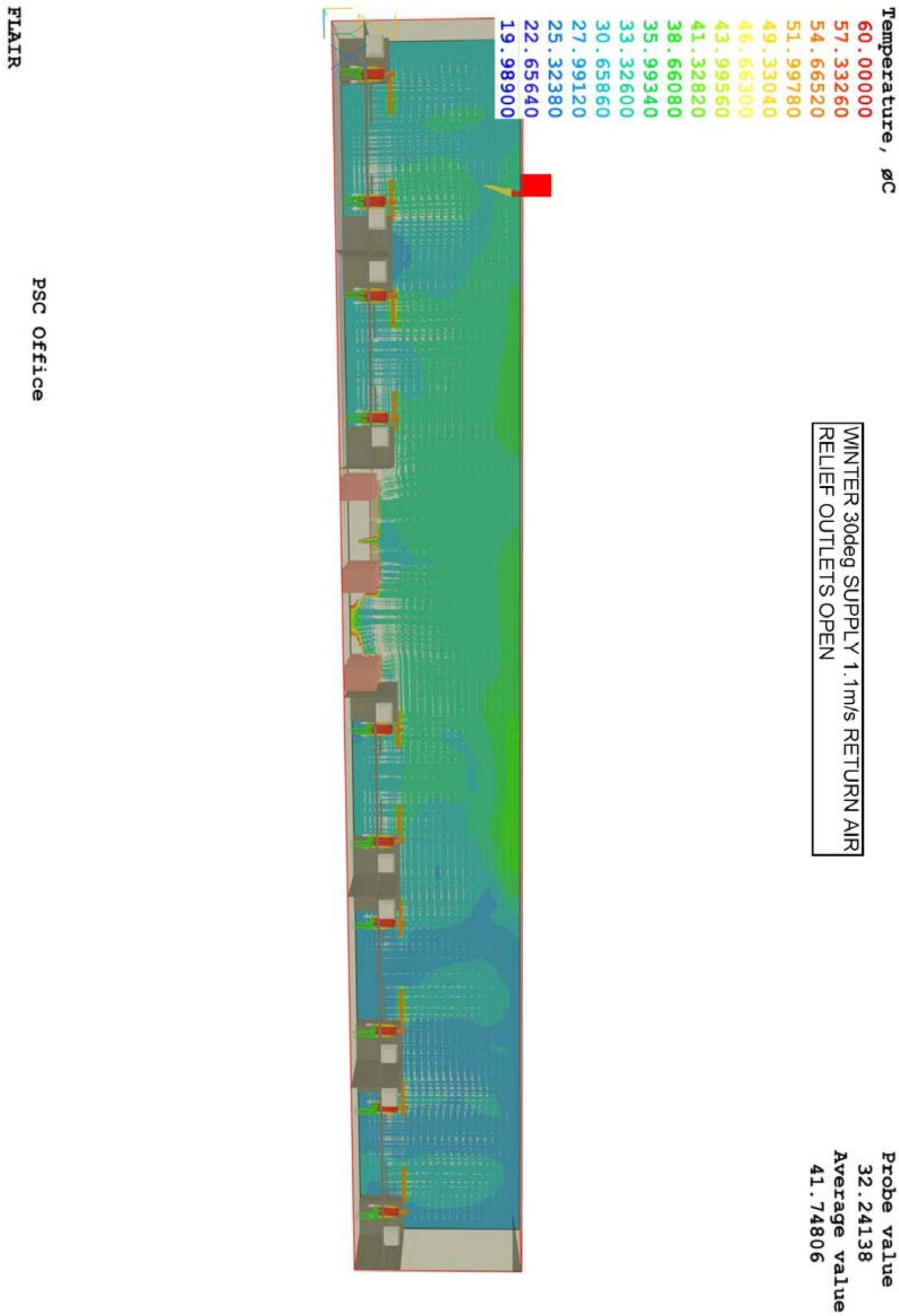


Figure 23 Winter Output 30ºC Return Air at 1.1m/s Y-Axis



Pressure

Analysis of the summer outputs of the CFD models with respect to pressure provided an insight into the effectiveness of pressurising the space. The pressure outputs shown in figures 24 to 27 typically indicate that the pressure stratifies in increasing values with increasing height (Z values) achieving a maximum pressure difference of 5 Pa. The objective of the system is to induce positive pressure in the space combined with the thermal gradients present to expel the warmer air out of the space via the open roof mounted vents. This could be improved by providing either more supply air or by using roof mounted exhaust fans, which would draw the excess air required by infiltration or leakage into the space in general. A higher supply air quantity would have the negative effect of increasing the total air quantity that needs to be conditioned and thereby would result in a larger capacity system being required. This approach would be contrary to the concept of providing an energy efficient system and therefore would not be recommended.

Putting a return air outlet at low level in the space resulted in a pressure differential in the space of approximately 3 Pa this was slightly less than the full displacement system (approximately 5 Pa) and therefore did not stratify and release the hot air as effectively as the other solutions.

Figure 24 Summer Output 25Pa Supply Air X-Axis

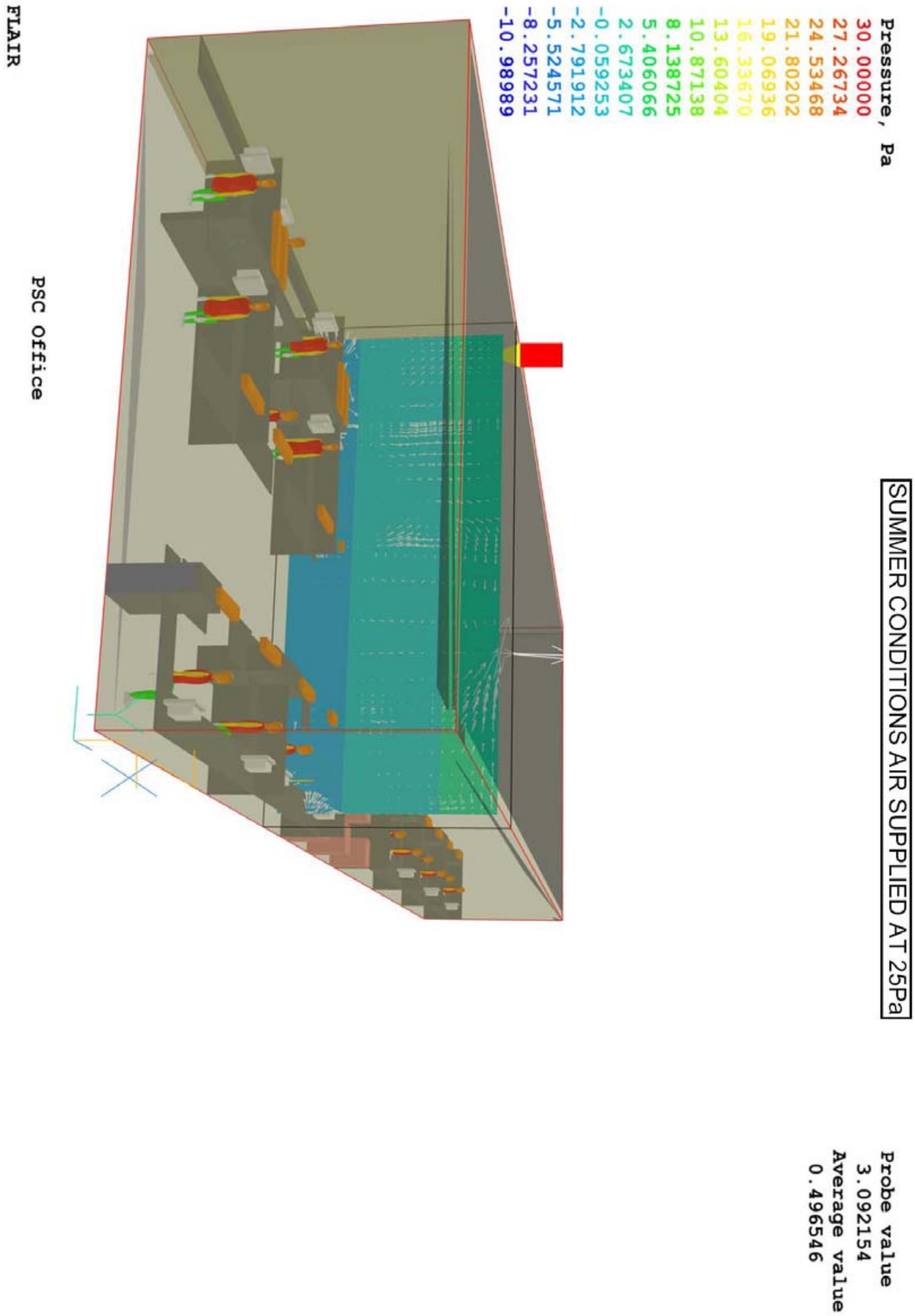


Figure 25 Summer Output 25Pa Supply Air Y-Axis

Figure 26 Summer Output 25Pa Supply Air at 10ºC

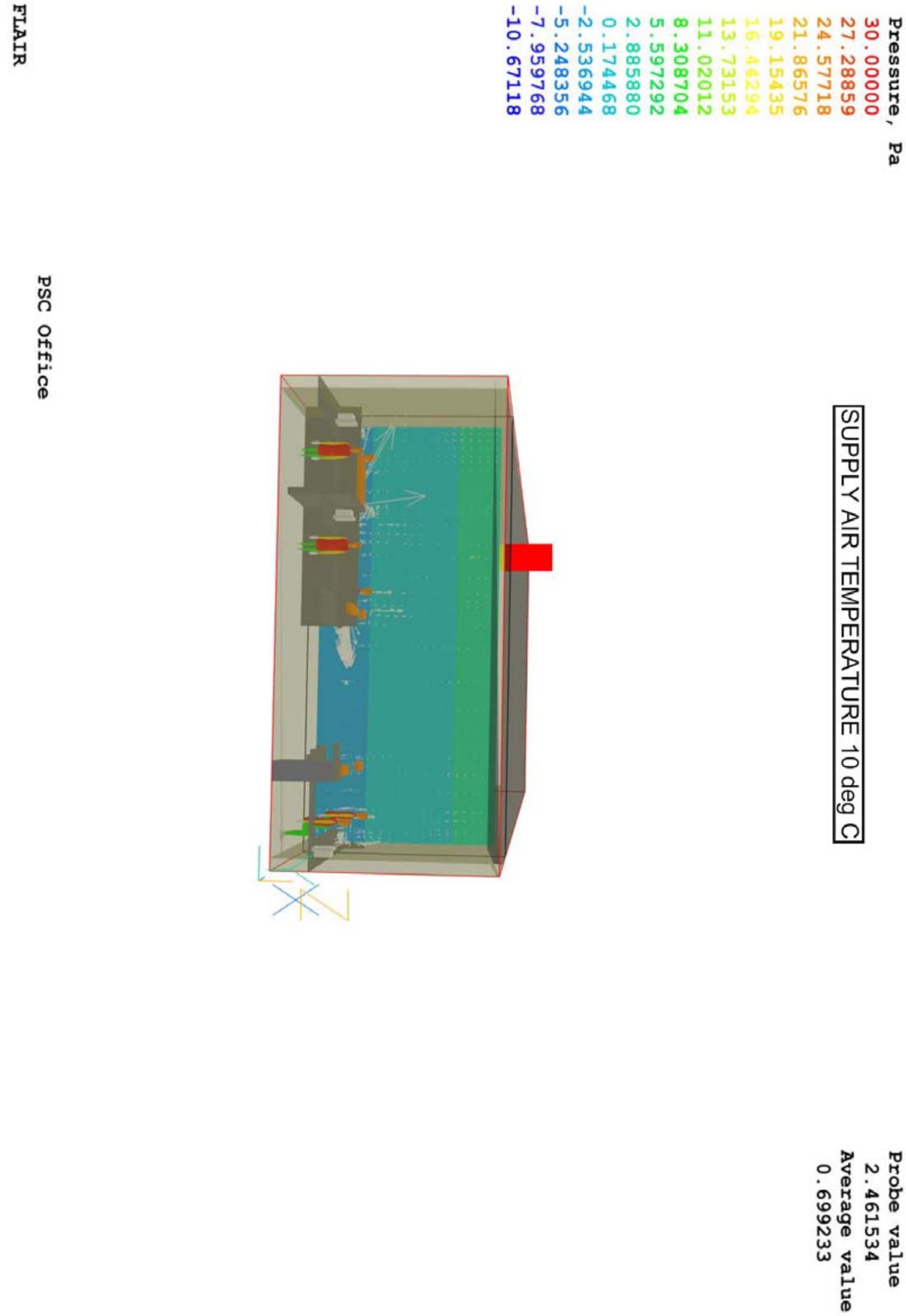


Figure 27 Winter Output Supply Air at 25ºC and Return Air 1.1m/s

CHAPTER 7

7.1 Comparative Analysis of CFD vs Data Loggers

The CFD outputs compared to the results gained from the data loggers provided some good correlations with the summer conditions however there were some notable discrepancies with the winter results.

As noted in Chapter 4 in the evaluation of trend logging there were some noticeable differences in the supply air temperature between the centre and perimeter workstations possibly correlated to thermal losses through the MDF particle board. The CFD analysis however was based on the assumption that all of the supply air quantities and temperatures were reasonably consistent. If for example a number of the supply air jets were closed it would increase the static pressure on the fan and therefore decrease the supply air quantity. Therefore the supply air will not necessarily be consistent as assumed. This assumption clearly leads to inaccuracies in the CFD analysis. The May ambient temperatures provided some summer data for comparison. The ambient temperatures ranged between 18°C to 28°C. The CFD analysis for 15°C supply air at 5m/s showed roof level temperatures between 32°C to 35°C. The results of the roof mounted data logger indicated the actual conditions where somewhat lower at approximately 28°C. The actual workstation temperature ranged between 24°C to 27°C whilst the CFD output showed values ranging between 23°C to 26°C. The difference in the results for the roof may be attributed to incorrect values being entered in the CFD analysis based on any of a number of assumptions made. Assumptions were made of the heat load associated with the computers, people, walls, roof and floor. Also the CFD model is based on 100% occupancy whereas in practice the space would not always be fully occupied. The practical application allows jets to be directed according to the user's preference or even closed off via way of a butterfly damper. The six wall mounted splits also were intermittently operated which would also influence the temperatures in the space.

The July ambient temperatures provides a good range for comparison of the winter values given they ranged between 9°C to 24°C. The data logger trends indicated a range between 26°C and 28°C at the roof level while the CFD analysis provided results between 38°C to 41°C. The data logger at the workstation indicated the actual temperatures between 23°C and 25°C and the CFD analysis had a range of 25°C to 27°C. Again the assumed variables with the space as well as the level of occupancy would be the most probable reason for the difference in the roof level values. Once again the workstation values showed a reasonable correlation between the actual and CFD values.

CHAPTER 8

8.1 Conclusions

The results of the CFD provided a reasonable approximation based on the assumptions made. Limitations were placed on the accuracy of the trend logging due to only two temperature loggers being available and also the space had supplementary wall mounted systems which would operate during either peak heating or cooling conditions. Further the comparative graphs were limited by the available data on ambient conditions in that historical data was only for 9am and 3pm each day.

The jet diffuser when pointed directly at the occupants can provide a feeling of cooler air based on the air quantities. For that reason a number of the occupants in the test case were directing the jets away from themselves in winter. The CFD results indicated that the optimum results are with a supply air of approximately 15ºC in summer providing as much pressurisation of the space as possible. This is evidenced by the improved conditions at 7m/s, whereas in winter the system has a more advantageous result when some return air is taken back to the system.

The models showing the pressure values indicated higher pressure differentials for the 100% outside air system than the modified system with return air. The subsequent temperature profiles indicated better overall thermal conditions in summer with the 100% outside air system and improved conditions for winter with the return air. The study provided results that support the theory that pressurisation of the space allows the internal heat to be relieved out of the roof mounted grilles and not returned into the conditioner as a heat load that would require additional capacity.

In winter however this internal heat load can be returned to the conditioner to reduce the heating capacity required. Therefore it is recommended that the outside air ductwork be increased in size to reduce the pressure drop and a damper be installed in the return ductwork to be closed off in summer and opened in winter. A controls system could be incorporated to sense ambient temperatures and automatically close the damper.

Further to this a sensor should be placed at low level in the space to control the system as opposed to the return air. This would provide more stable control even though the setpoint may not be entirely accurate. The setpoint may need to be reduced by a few degrees to what is being sensed at the remote sensor to provide the required supply air temperature.

8.2 Commercial Barriers

The implementation of micro climate systems is dependant on the availability of space for the installation of ductwork to distribute air to the respective workstations. In the case of the test office a naturally ventilated carpark was directly underside of the space allowing some of the ductwork to be run exposed underside of the building and because it is only a single storey building it was reasonably easy to install the relief grilles in the roof. Unique solutions would need to be incorporated into the preliminary building design to allow a method of distributing air to the jets in the workstation as well as relieving at high level. This could be achieved by having both a suspended floor and ceiling to provide a concealed space to run the ductwork. The commercial reality is that concealed spaces that add little to the occupied area need to be minimised in buildings. The loss of a few hundred millimetres on each floor of a multi storey building could result in one less floor for the same overall height. With a micro climate system however the request for both available floor and ceiling space may require less room than a conventional system ducted from within the ceiling space because the air quantities are much less.

Careful coordination of the location of jets is critical to ensure the higher velocity air distributed at the workstation level does not simply blow paperwork etc off the table. This should hopefully be eliminated by the fact that the user has control over the direction of the jet ensuring it is positioned to minimise any disruption to the workstation.

Another potential barrier is the resistance to embracing alternative means for air conditioning from the perspective clients. Often clients will opt not to take a risk from complaints due to the perceived lack of capacity and air movement in the space. People generally are far more comfortable with adopting the approach of ‘that’s how it has always been done’ because it is a tried a tested method.

CHAPTER 9

9.1 Recommended Further Work

Install a number of data loggers throughout the space at varying locations and heights to further either support or disprove the representation shown in the CFD outputs.

Testing of the system should be conducted over the peak periods of summer to establish the effectiveness at high ambient temperatures.

The temperature sensor for the system should be mounted within the space and a damper installed in the return air ductwork. This would allow the system to operate using return air in the colder months and then close the damper so that 100% outside air is used in the summer periods. This should be tested to confirm the warmer return air provides a more efficient operation in winter, and the use of 100% outside air provides more pressurisation in summer. The resultant pressurisation relieves the internal heat out of the high level exhaust grilles.

As noted in the conclusions it would be beneficial to install a controls system that has the ability to automatically close the roof mounted dampers in response to ambient temperatures. This was initially thought to have been not justified because it only required operation twice a year, however during the first change of season in the office in the test case the damper remained open for the start of winter indicating that unless a direct maintenance responsibility is assigned the system will not be touched. Various manufacturers provide such controls, one such company is Innotech Controls that supply a range of control systems that incorporate a number of digital and analogue input and outputs.

Research should be conducted on the implementation of like systems in multi storey buildings and the suitability of the same.

Upon implementation of any of the above modifications feedback would need to be sought from the occupants of the space to ensure the changes still meet their needs.

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Appendix A

University of Southern Queensland

FACULTY OF ENGINEERING AND SURVEYING

ENG 4111/4112 Research Project

PROJECT SPECIFICATION

For: **BRYCE NAPPER**

Topic: COMPUTATIONAL FLUID DYNAMICS (CFD) INVESTIGATION OF THE THERMAL EFFICIENCY OF MICRO CLIMATE CONDITIONING OF A TYPICAL WORKSTATION.

Supervisor: Ruth Mossad

Enrolment: ENG 4111 – S1 2006
ENG 4112 – S2 2006

Project Aim: The project aims to investigate the efficiency of micro climate conditioning compared with conventional air conditioning systems for a typical office workstation.

PROGRAMME: **ISSUE B, 21st MARCH 2006**

1. Research the factors that influence the heat load within typical office workstations.
2. Literature review of current methods employed to air condition office workstations.
3. Conduct CFD analysis of the temperature distribution in an office workspace, conditioning only the spaces that are occupied. The analysis shall provide optimum values for key variables such as air quantities and supply air temperature.
4. Evaluate the performance of a micro climate system as compared to air conditioning the entire volume of the space.
5. Analyse the micro climate system and identify potential commercial barriers that could prevent the implementation of such a system.
6. Based on the findings comment on the design that provides the optimum performance.

If time permits

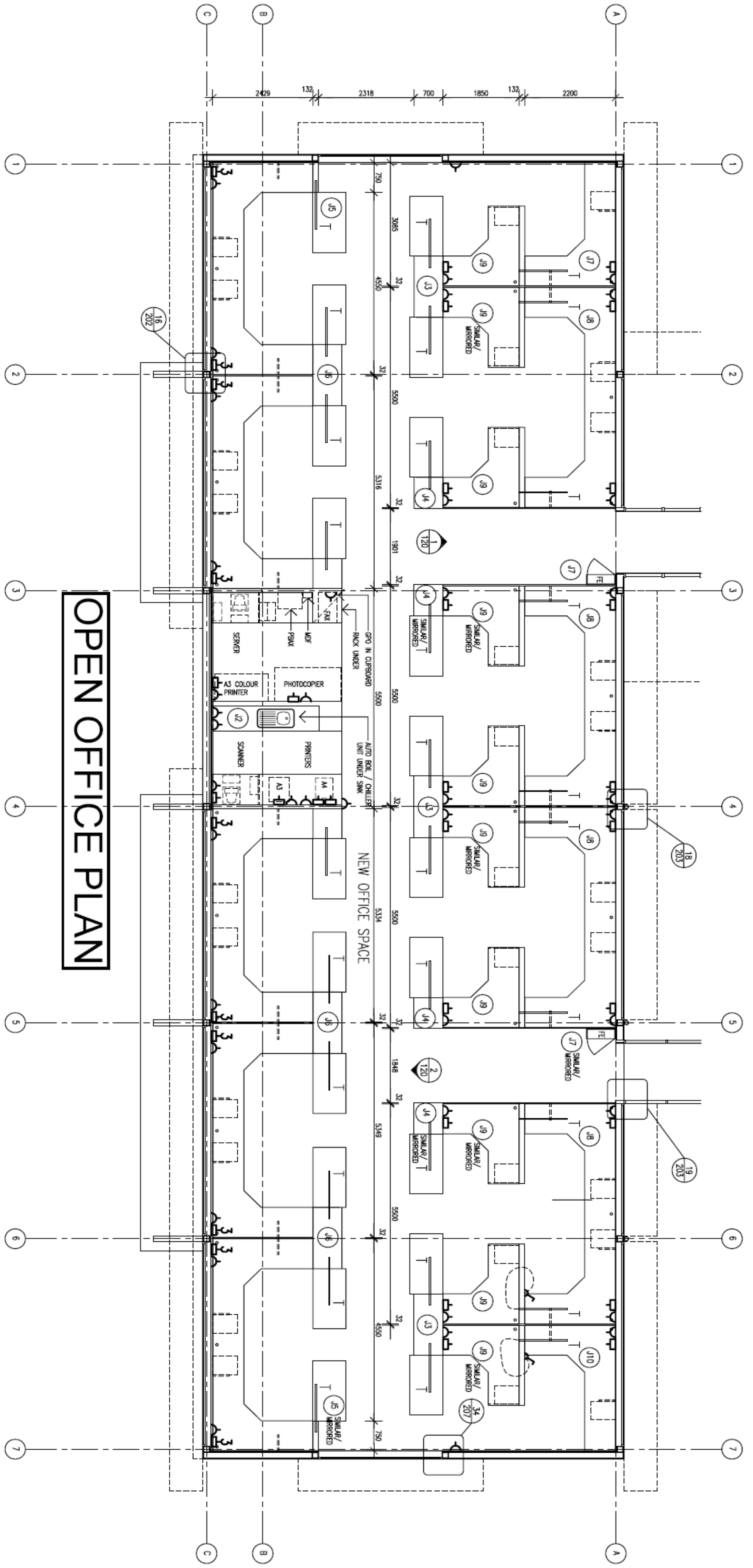
7. Design an improved micro climate system and conduct a CFD analysis of the modified system.
8. Log actual temperature data on an installed air conditioning system to further support or disprove the findings.

AGREED

_________________ (Student) __________________________ (Supervisor)

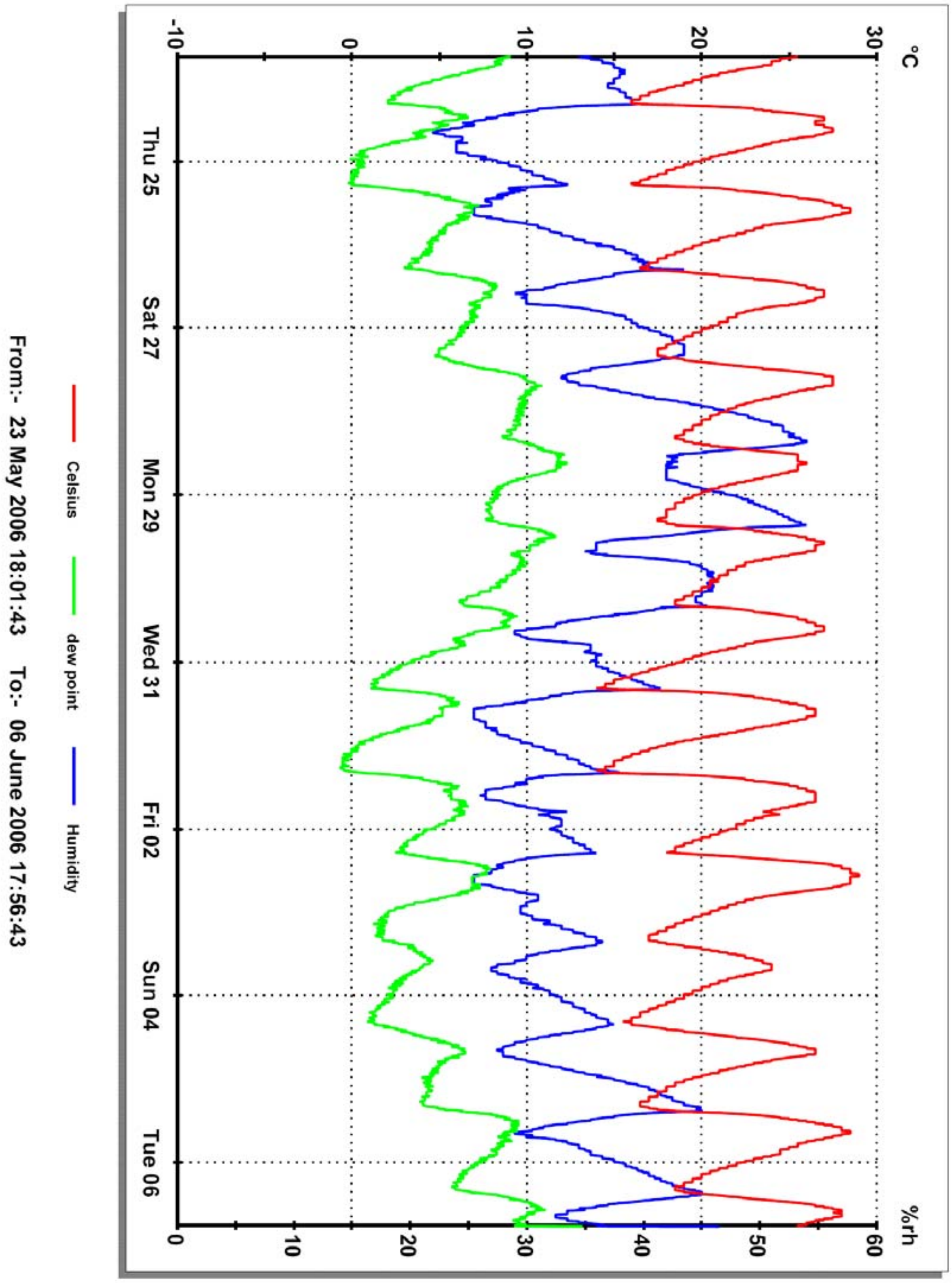
__/__/__ __/__/__

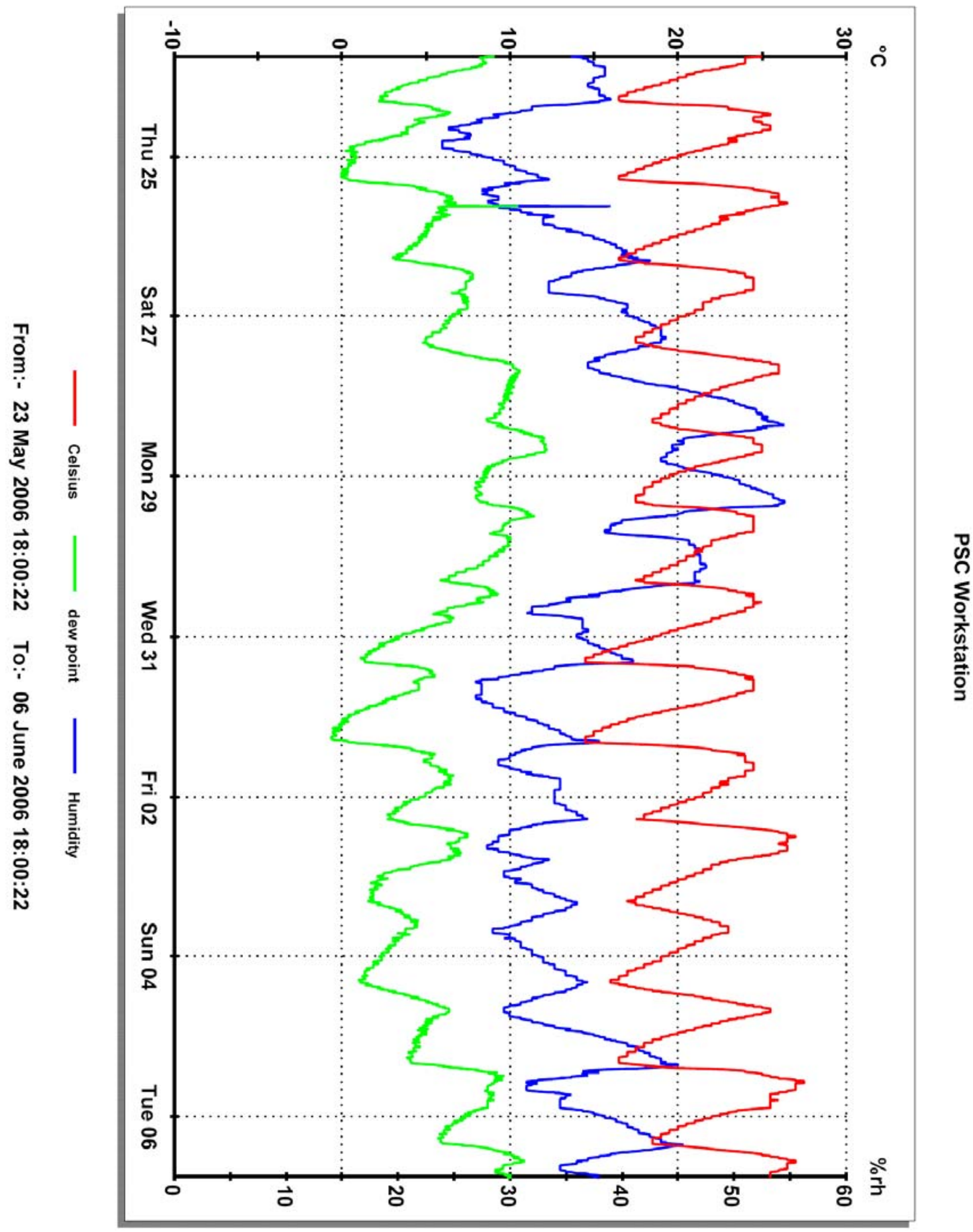
Appendix B



OPEN OFFICE PLAN
NEW OFFICE SPACE
SERVER
PBAX
MDF
FAX
RACK UNDER
GPO IN CUPBOARD
A3 COLOUR PRINTER
PHOTOCOPIER
SCANNER
PRINTERS
AUTO BOIL / CHILLER UNIT UNDER SINK
SIMILAR/ MIRRORED

<u>**Appendix C**</u>

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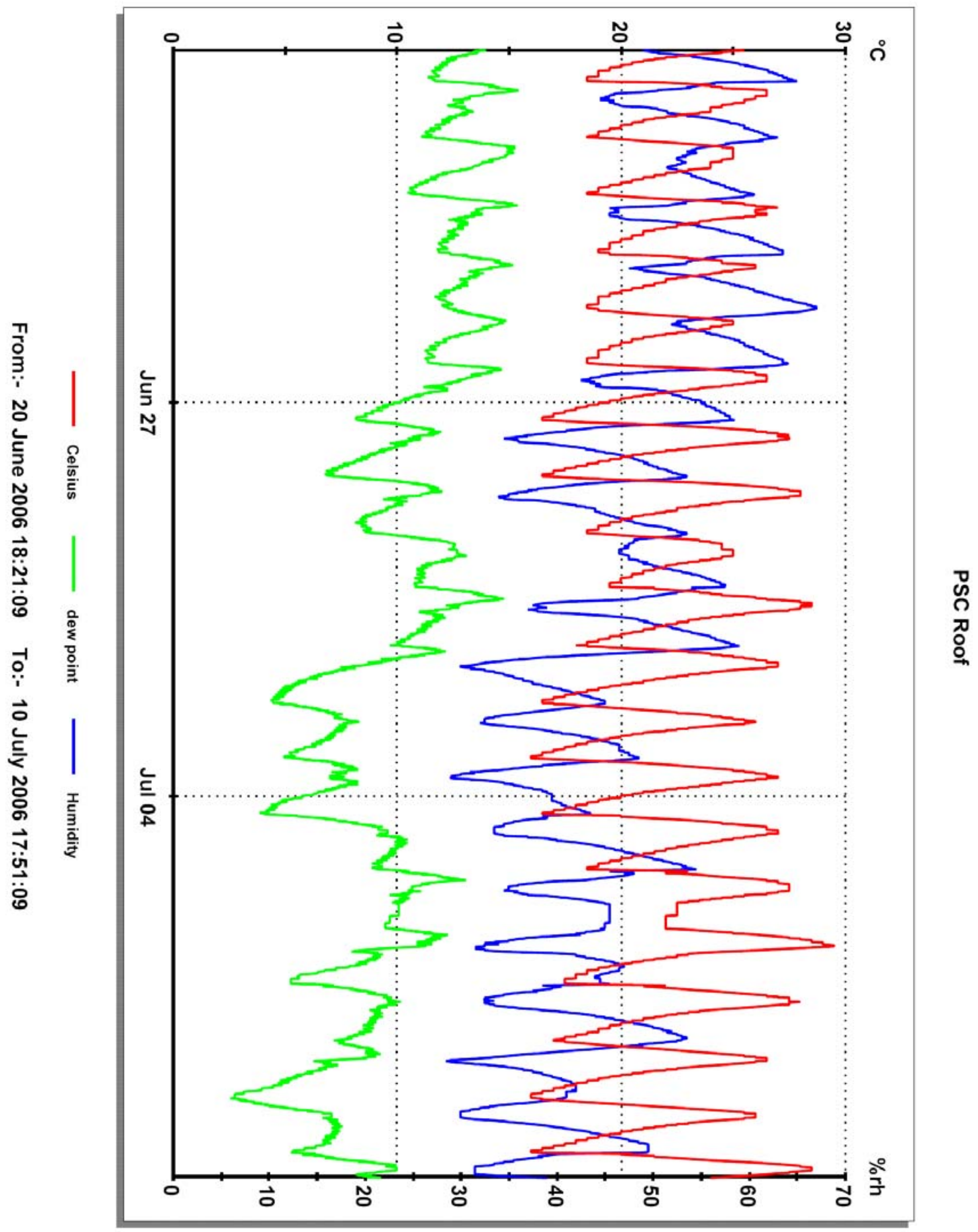
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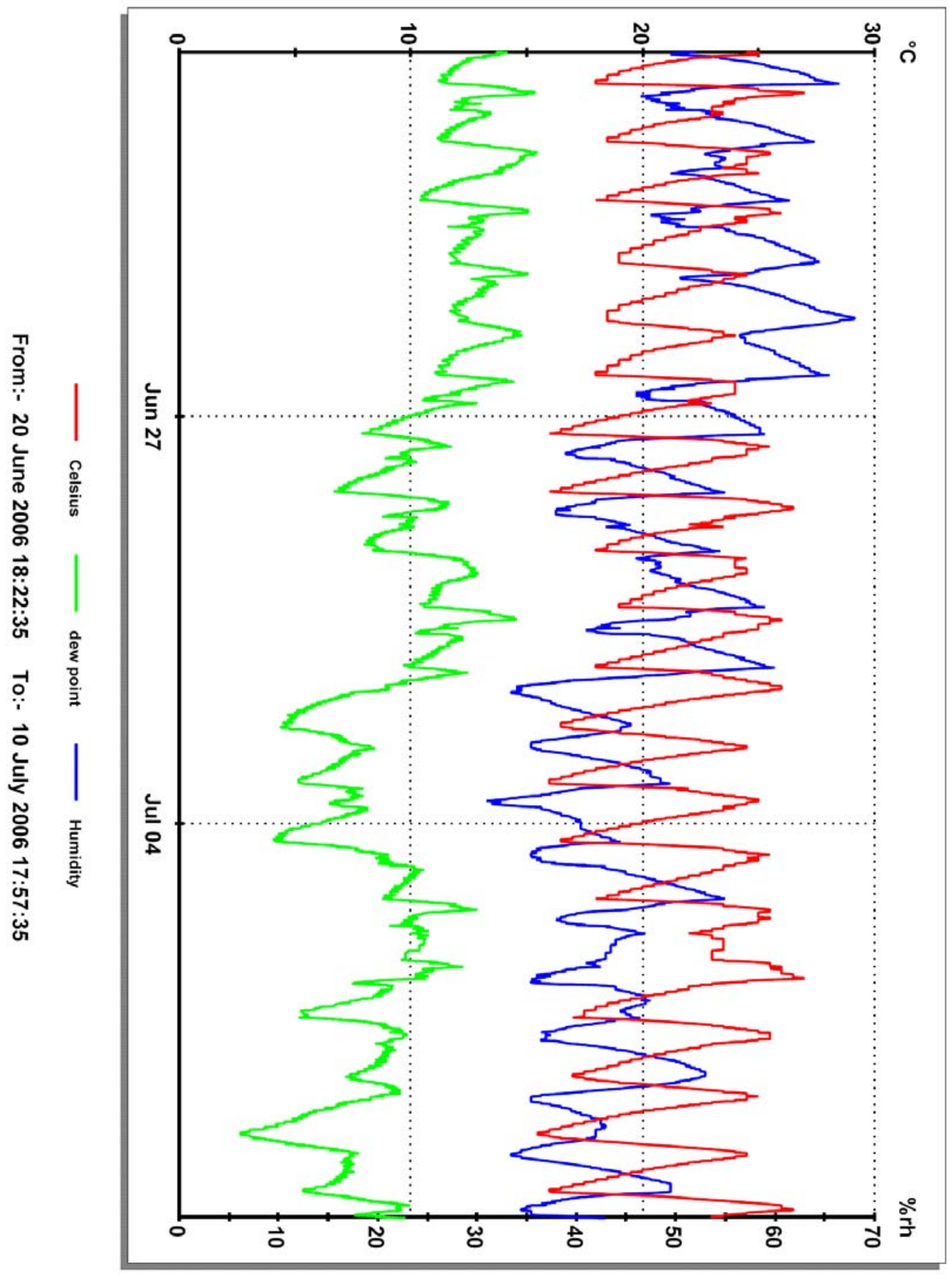
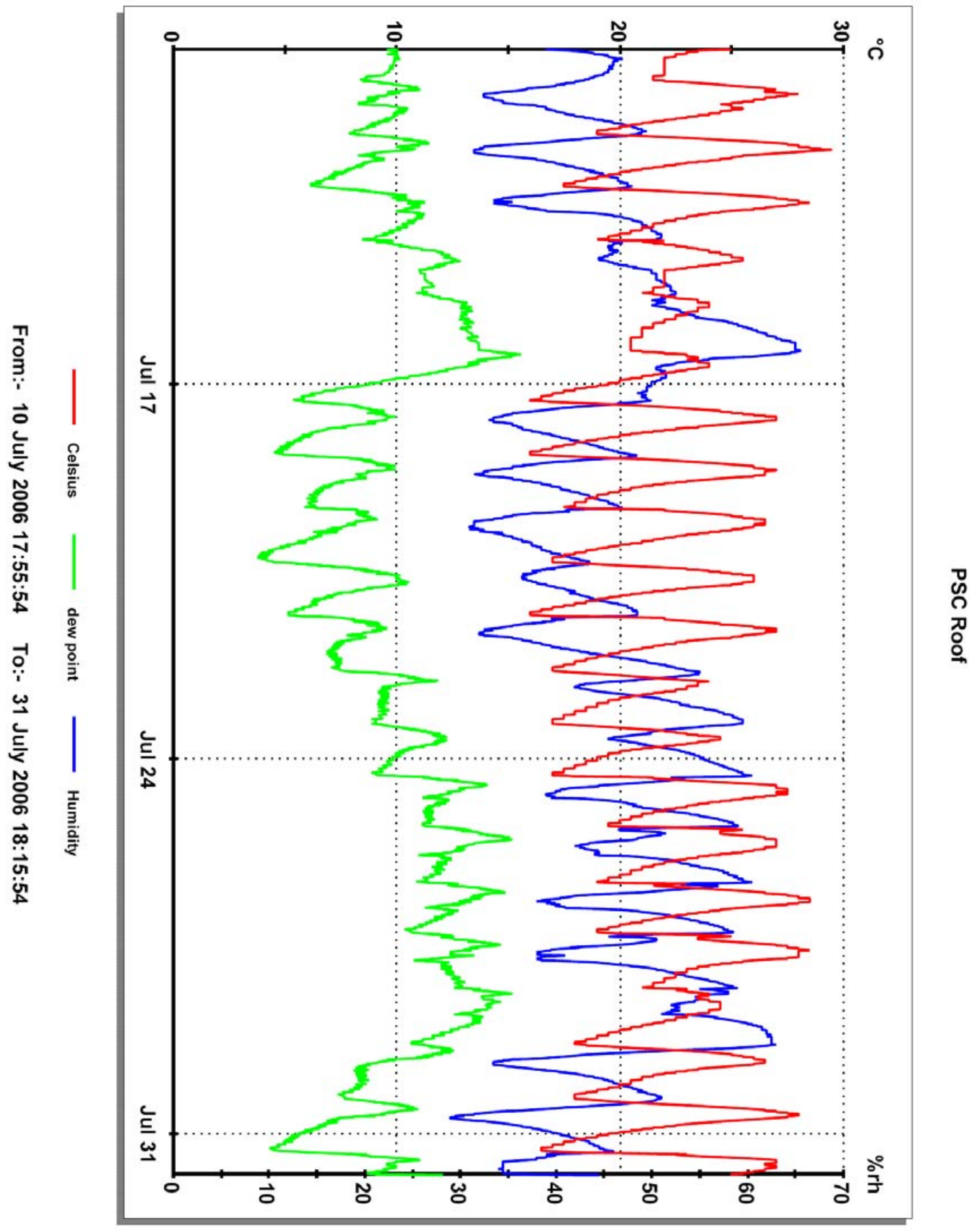
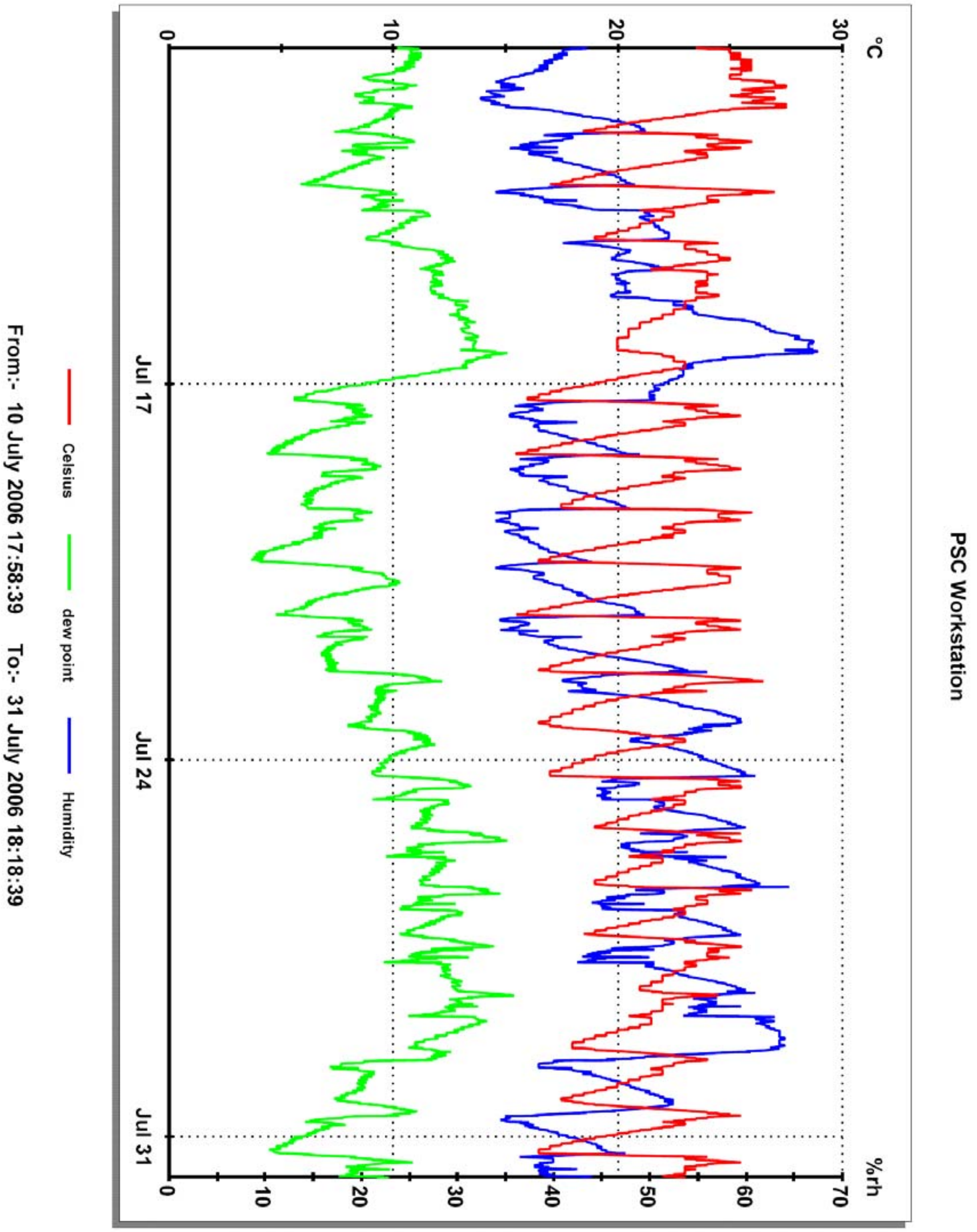
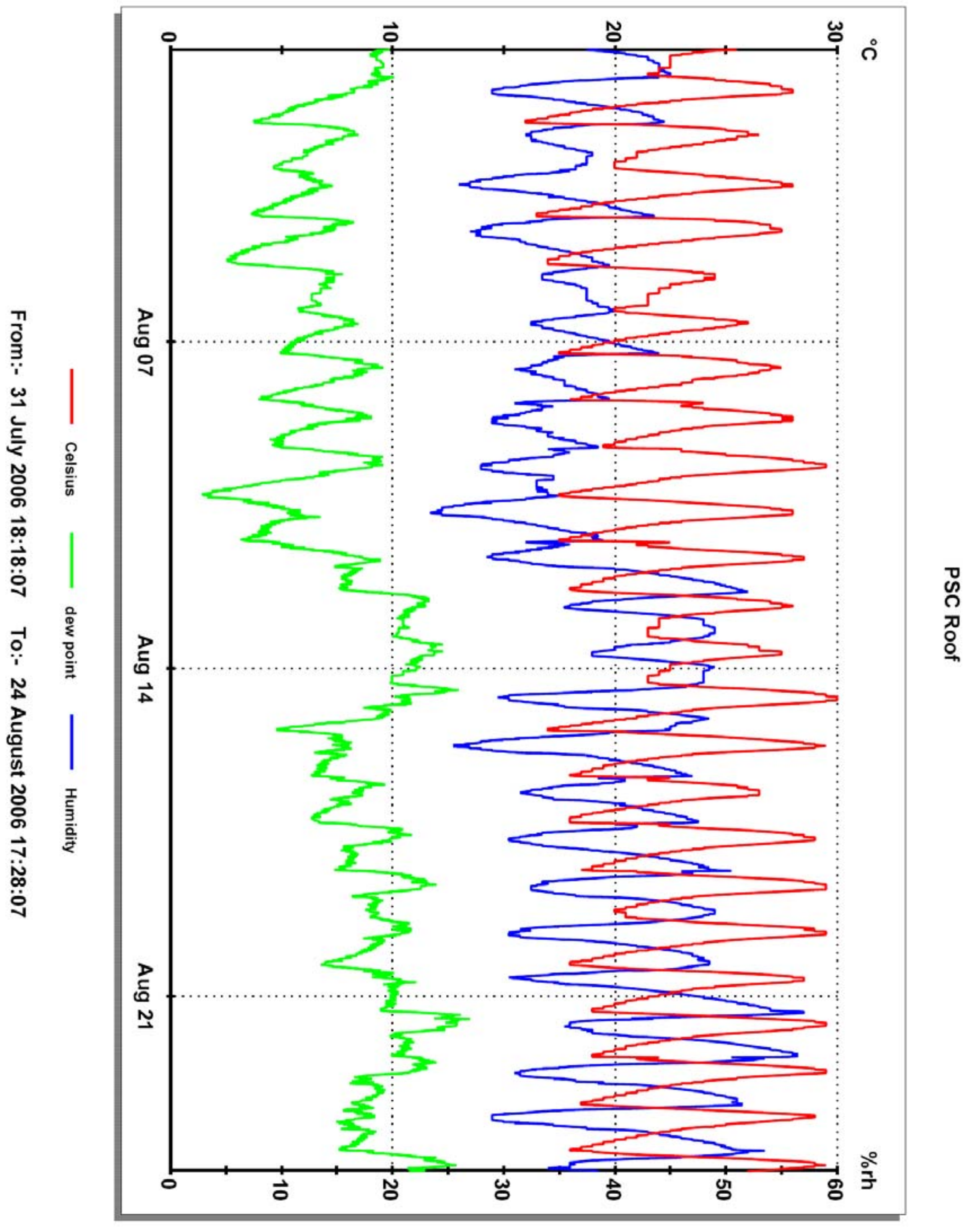
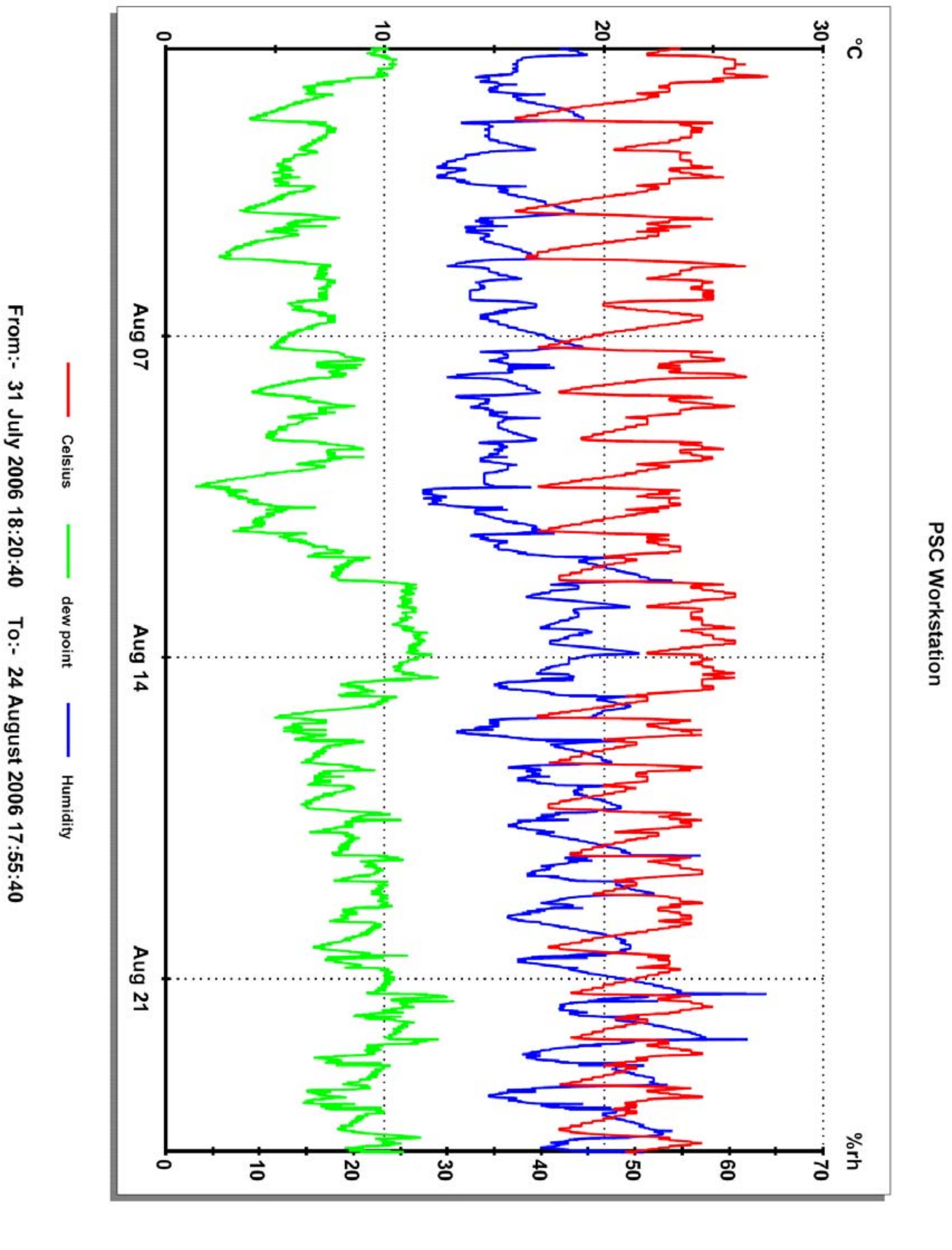
Humidity
From:- 20 June 2006 18:21:09 To:- 10 July 2006 17:51:09


PSC Workstation
°C
%rh
0
10
20
30
0
10
20
30
40
50
60
70
Jun 27
Jul 04
Celsius
dew point
Humidity
From:- 20 June 2006 18:22:35 To:- 10 July 2006 17:57:35


PSC Roof
°C
%rh
0
10
20
30
40
50
60
70
Jul 17
Jul 24
Jul 31
Celsius
dew point
Humidity
From:- 10 July 2006 17:55:54 To:- 31 July 2006 18:15:54


PSC Workstation
°C
%rh
0
10
20
30
40
50
60
70
Jul 17
Jul 24
Jul 31
Celsius
dew point
Humidity
From:- 10 July 2006 17:58:39 To:- 31 July 2006 18:18:39


PSC Roof
°C
%rh
30
20
10
0
60
50
40
30
20
10
0
Aug 07
Aug 14
Aug 21
Celsius
dew point
Humidity
From:- 31 July 2006 18:18:07 To:- 24 August 2006 17:28:07


PSC Workstation
°C
%rh
Aug 07
Aug 14
Aug 21
Celsius
dew point
Humidity
From:- 31 July 2006 18:20:40 To:- 24 August 2006 17:55:40

Appendix D

Brisbane, Queensland
May 2006 Daily Weather Observations

Most observations from Brisbane City, but some from Brisbane Airport.

Australian Government
Bureau of Meteorology

Date	Day	Temps Min	Temps Max	Rain	Evap	Sun	Max wind gust Dirn	Max wind gust Spd	Max wind gust Time	9am Temp	9am RH	9am Cld	9am Dirn	9am Spd	9am MSLP	3pm Temp	3pm RH	3pm Cld	3pm Dirn	3pm Spd	3pm MSLP
		°C	°C	mm	mm	hours		km/h	local	°C	%	eighths		km/h	hPa	°C	%	eighths		km/h	hPa
1	Mo	16.8	26.6	5.8	5.2	10.3	WNW	46	14:12	20.8	63	1	WSW	13	1012.1	26.2	34	3	WSW	15	1007.8
2	Tu	11.9	25.5	0	5.0	10.6	W	37	13:01	20.5	44	1	WSW	9	1011.3	25.2	27	0	W	17	1008.6
3	We	11.8	25.2	0	7.2	10.4	SW	20	09:02	19.3	45	0	SW	7	1016.9	24.3	26	0	NNW	6	1014.5
4	Th	10.9	26.2	0	6.4	10.1	SE	22	11:22	19.6	43	0	SSW	6	1019.9	25.0	42	1	ESE	11	1017.7
5	Fr	13.0	26.0	0	4.0	9.6	ENE	22	14:36	19.8	64	2	SW	9	1020.4	24.7	50	1	NE	11	1016.1
6	Sa	13.2	27.4	0	3.2	10.1	SSW	22	14:16	18.8	70	0	WSW	11	1017.9	27.2	27	1	WSW	7	1013.0
7	Su	13.3	26.8	0	4.0	10.1	NE	24	15:11	19.6	61	1	SSW	6	1017.4	25.3	53	1	NE	13	1012.7
8	Mo	13.8	28.5	0	4.2	9.3	WSW	26	15:24	21.9	64	2	SSW	4	1016.8	27.7	21	5	WNW	13	1013.7
9	Tu	13.3	23.4	0	5.4	10.0	NNE	22	13:39	18.0	48	1	SSW	9	1021.1	22.4	55	2	NE	11	1017.9
10	We	12.5	26.9	0	4.6	7.9	W	31	10:57	19.3	66	2	SW	6	1020.0	25.3	19	7	SW	9	1016.5
11	Th	10.9	25.3	0	4.8	9.4	E	28	14:17	18.6	46	1	SW	6	1023.8	24.5	47	1	E	15	1020.7
12	Fr	14.4	23.6	0	4.0	5.7	E	31	13:39	18.0	73	7	S	9	1024.3	21.7	62	3	E	15	1021.5
13	Sa	13.4	24.4	0	2.8	9.7	ESE	28	13:39	18.9	70	5	SSW	9	1024.5	24.0	48	3	NNE	9	1019.8
14	Su	12.6	26.4	0	4.0	10.0	ENE	26	15:35	18.3	74	1	SW	7	1021.7	26.0	32	1	NW	4	1017.9
15	Mo	12.9	26.0	0	3.8	9.3	SE	41	17:55	19.4	67	1	SSW	11	1022.6	25.4	39	1	SSW	9	1019.9
16	Tu	15.5	23.7	4.0	4.4	9.2	ESE	33	12:17	20.1	61	2	SSW	9	1026.5	22.4	51	6	ESE	11	1024.5
17	We	14.8	23.8	0.2	4.8	6.5	SE	31	10:49	19.9	63	2	SW	9	1027.0	21.9	54	7	SE	17	1023.9
18	Th	14.7	24.4	0	3.8	9.1	SE	35	16:09	19.5	67	1	SW	7	1025.7	22.3	57	6	ESE	15	1022.8
19	Fr	15.0	25.0	0	4.2	7.0	E	24	14:35	20.3	66	5	SSW	7	1024.5	22.4	63	5	E	11	1022.1
20	Sa	14.2	26.5	0	3.8	9.8	E	20	15:56	19.0	79	1	SW	6	1024.6	24.9	47	1	NE	13	1021.2
21	Su	14.0	25.0	0	3.6	9.8	E	28	15:57	19.1	69	1	SSW	9	1026.6	23.3	51	3	ESE	13	1023.0
22	Mo	13.2	24.7	0	4.8	7.5	W	20	21:09	19.8	65	6	SW	9	1023.6	22.4	64	7	ENE	11	1017.9
23	Tu	15.3	24.3	0	3.6	6.0	NW	24	12:13	17.2	60	7	SW	7	1016.9	22.9	19	3	W	13	1013.1
24	We	7.1	23.3	0	3.0	9.8	W	30	15:47	15.5	37	1	SW	9	1014.8	22.6	13	0	W	17	1010.8
25	Th	8.5	23.9	0	4.0	10.0	SW	19	11:25	17.1	39	1	SW	6	1016.2	22.9	26	6	SE	7	1013.3
26	Fr	9.7	24.8	0	3.4	8.2	NNE	17	15:07	15.5	54	3	SW	6	1019.1	22.6	32	6	NE	9	1015.9
27	Sa	10.6	24.3	0	3.6	8.7	NE	19	14:12	16.1	48	4	SSW	7	1019.3	21.4	51	4	NE	11	1016.0
28	Su	11.8	23.2	0	4.0	8.1	ESE	28	16:26	17.8	64	1	SSW	11	1019.7	20.4	56	6	NE	9	1015.7
29	Mo	11.4	23.5	0	2.0	9.2	NNE	20	14:24	16.6	65	5	WSW	9	1020.0	21.6	48	1	NE	9	1016.4
30	Tu	8.7	24.0	0	3.0	9.7	W	24	14:27	15.9	53	0	SW	7	1019.6	23.5	18	0	W	11	1015.4
31	We	5.0	22.4	0	4.0	9.7	WSW	24	13:28	11.6	55	6	SW	6	1018.4	21.8	19	1	W	9	1013.7
Statistics for May 2006																					
	Mean	12.4	25.0		4.1	9.1				18.4	59	2		7	1020.4	23.7	40	2		11	1016.9
	Lowest	5.0	22.4		2.0	5.7				11.6	37	0	SSW	4	1011.3	20.4	13	0	NW	4	1007.8
	Highest	16.8	28.5	5.8	7.2	10.6	WNW	46		21.9	79	7	WSW	13	1027.0	27.7	64	7	#	17	1024.5
	Total			10.0	128.6	280.8															

Temperature, humidity, wind, pressure and rainfall observations are from Brisbane {station 040913}. Cloud, evaporation and sunshine observations are from Brisbane Aero {station 040842}

Brisbane Airport is located about 12 kilometres north east of the Brisbane City site, and closer to the coast. The evaporation, sunshine and cloud values should be used as a guide only.

IDCJDW4019.200605 Prepared at 16:20 GMT on 2 Jul 2006

Users of this product are deemed to have read the information and accepted the conditions described in the notes at http://www.bom.gov.au/climate/dwo/IDCJDW0000.pdf

Brisbane, Queensland
June 2006 Daily Weather Observations

Most observations from Brisbane City, but some from Brisbane Airport.

Australian Government
Bureau of Meteorology

Date	Day	Temps		Rain	Evap	Sun	Max wind gust			9am						3pm					
		Min	Max				Dirn	Spd	Time	Temp	RH	Cld	Dirn	Spd	MSLP	Temp	RH	Cld	Dirn	Spd	MSLP
		°C	°C	mm	mm	hours		km/h	local	°C	%	eighths		km/h	hPa	°C	%	eighths		km/h	hPa
1	Th	5.9	23.9	0	2.0	9.7	ENE	19	15:13	13.7	45	1	SSW	6	1014.8	22.5	27	2	ENE	11	1009.9
2	Fr	6.8	22.9	0	3.4	9.7	W	24	21:13	12.9	48	1	S	6	1013.2	21.3	31	2	ENE	11	1008.6
3	Sa	9.5	21.5	0	5.0	10.0	WSW	46	12:05	15.8	45	1	W	15	1012.5	21.0	28	1	W	19	1010.7
4	Su	7.9	23.8	0	6.6	9.9	WSW	20	10:03	17.1	41	1	SW	7	1020.0	23.1	29	1	W	4	1018.2
5	Mo	8.7	23.5	0	2.6	9.7	E	20	16:00	15.6	57	1	SW	7	1024.2	22.4	33	2	NE	7	1021.4
6	Tu	10.7	22.3	0	4.2	8.0	ENE	28	14:16	16.3	60	1	SSW	11	1026.2	20.3	45	1	ESE	13	1023.6
7	We	8.8	22.8	0	3.2	8.9	S	22	09:37	15.5	60	1	SW	9	1026.5	22.1	29	1	S	6	1022.6
8	Th	10.7	23.9	0	4.0	9.0	E	22	15:53	16.7	66	2	WSW	9	1026.8	23.2	41	2	ENE	7	1023.4
9	Fr	13.0	22.7	0	3.4	3.5	W	52	06:30	19.0	65	7	SSW	9	1027.0	21.3	62	7	ESE	15	1023.7
10	Sa	16.4	21.7	1.2	2.4	0.0	NW	50	21:15	18.1	93	7		Calm	1021.2	21.7	82	7	NE	7	1014.9
11	Su	17.5	20.3	12.0	1.2	5.0	W	46	14:23	17.7	62	6	WSW	13	1013.7	19.2	39	2	W	17	1012.9
12	Mo	10.4	21.3	0	4.0	9.2	SW	43	21:44	15.3	52	2	SW	9	1024.5	20.4	43	2	SE	9	1023.4
13	Tu	11.3	21.8	0	4.4	6.7	SSW	28	13:54	16.1	54	4	SSW	11	1029.7	19.5	61	5	SSW	11	1028.1
14	We	11.8	23.1	0	2.6	9.4	SSE	26	11:58	16.0	72	1	SW	7	1030.8	21.5	42	1	SE	13	1027.8
15	Th	12.7	23.7	0	3.4	8.3	SW	19	11:36	15.9	75	1	SW	9	1029.4	22.7	41	1	NNW	9	1024.0
16	Fr	9.1	22.7	0	2.8	9.9	W	24	10:29	14.7	65	1	SW	7	1024.1	22.0	26	0	WSW	7	1019.7
17	Sa	7.3	21.5	0	2.8	7.4	ESE	30	14:42	14.0	52	2	SW	11	1024.7	19.7	50	4	SSE	13	1021.8
18	Su	12.5	19.9	0	3.2	1.3	WSW	41	06:29	15.5	72	7	SSW	9	1025.3	16.6	78	7	SE	9	1023.6
19	Mo	13.4	20.5	1.0	1.6	5.3	SSE	35	11:47	16.1	80	5	SW	7	1026.5	18.6	69	7	E	9	1024.0
20	Tu	13.7	16.8	1.2	2.2	0.7	SE	37	16:41	15.9	80	7	SW	7	1025.9	15.5	93	8	S	7	1023.5
21	We	13.6	19.5	20.8	0.0	0.9	SSE	26	15:00	15.7	89	7	S	7	1023.9	18.0	73	6	SE	13	1022.2
22	Th	13.3	17.7	3.6	1.8	1.5	SE	26	13:11	15.3	86	8	SW	7	1024.4	17.4	76	7	SE	15	1022.1
23	Fr	13.2	21.0	1.4	0.4	3.5	E	28	12:12	15.5	82	6	WSW	6	1024.4	19.0	73	6	ENE	7	1021.7
24	Sa	13.6	22.9	0.4	1.8	3.6	E	19	14:25	15.9	90	7	SW	6	1024.0	21.2	69	1	NE	9	1020.8
25	Su	13.5	23.1	0.2	2.2	8.5	ENE	20	14:41	16.2	84	5	SW	7	1025.6	20.1	65	3	E	13	1022.4
26	Mo	13.1	22.6	0	2.6	5.5	SSE	28	19:09	17.0	79	5	SW	7	1024.0	18.9	64	5	E	13	1021.1
27	Tu	9.3	21.7	1.8	3.0	9.1	SSW	19	09:27	14.5	66	1	SW	11	1025.9	20.9	36	1	E	7	1022.7
28	We	8.7	22.4	0	2.8	9.3	SE	30	14:03	14.6	68	2	SW	7	1028.2	20.9	46	1	E	13	1026.3
29	Th	12.8	19.0	0	3.2	0.5	E	24	16:34	14.3	79	7	S	7	1030.0	18.8	62	7		Calm	1025.9
30	Fr	14.2	23.0	0.2	1.8	7.0	N	22	20:29	17.4	75	7	SW	9	1025.3	21.2	55	1	NE	9	1019.6
Statistics for June 2006																					
	Mean	11.4	21.8		2.8	6.4				15.8	68	3		8	1024.1	20.4	52	3		10	1021.0
	Lowest	5.9	16.8		0.0	0.0				12.9	41	1		Calm	1012.5	15.5	26	0		Calm	1008.6
	Highest	17.5	23.9	20.8	6.6	10.0	W	52		19.0	93	8	W	15	1030.8	23.2	93	8	W	19	1028.1
	Total			43.8	84.6	191.0															

Temperature, humidity, wind, pressure and rainfall observations are from Brisbane {station 040913}. Cloud, evaporation and sunshine observations are from Brisbane Aero {station 040842}

Brisbane Airport is located about 12 kilometres north east of the Brisbane City site, and closer to the coast. The evaporation, sunshine and cloud values should be used as a guide only.

IDCJDW4019.200606 Prepared at 13:40 GMT on 7 Jul 2006

Brisbane, Queensland
July 2006 Daily Weather Observations

Most observations from Brisbane City, but some from Brisbane Airport.

Australian Government
Bureau of Meteorology

Date	Day	Temps		Rain	Evap	Sun	Max wind gust			9am						3pm					
		Min	Max				Dirn	Spd	Time	Temp	RH	Cld	Dirn	Spd	MSLP	Temp	RH	Cld	Dirn	Spd	MSLP
		°C	°C	mm	mm	hours		km/h	local	°C	%	eighths		km/h	hPa	°C	%	eighths		km/h	hPa
1	Sa	10.6	22.8	0	2.2	9.9	WNW	37	13:17	17.1	70	1	SW	6	1018.9	22.5	22	1	W	17	1014.6
2	Su	8.0	22.4	0	5.2	9.6	W	20	13:23	14.5	58	1	SW	7	1018.2	21.8	31	1	WSW	7	1014.5
3	Mo	8.3	22.9	0	2.6	7.7	E	22	15:57	13.9	56	3	SW	7	1018.2	22.5	28	3	WSW	9	1015.2
4	Tu	8.7	22.7	0	3.2	8.5	SSE	31	13:49	15.2	54	1	SW	11	1021.6	21.0	48	6	SSW	11	1019.6
5	We	11.9	23.4	0	2.8	9.8	SSE	26	11:46	17.0	68	1	SW	11	1026.2	21.5	45	1	ESE	11	1023.1
6	Th	10.4	24.2	0	3.0	10.0	W	22	13:18	15.2	72	1	WSW	7	1025.7	24.0	33	1	WNW	9	1020.2
7	Fr	10.7	23.4	0	4.0	8.8	SE	22	12:01	16.2	56	3	SW	9	1025.4	21.4	46	2	SE	9	1022.5
8	Sa	10.0	23.5	0	3.4	10.0	SSW	22	15:28	14.3	63	1	SW	6	1025.7	23.1	19	1	SW	11	1022.0
9	Su	7.5	22.8	0	2.6	9.7	N	19	16:30	13.1	49	1	SW	7	1026.1	22.2	33	1	NNE	9	1022.6
10	Mo	7.6	23.3	0	3.2	9.8	NE	20	15:35	13.6	47	1	SW	9	1025.0	22.7	41	1	ENE	7	1020.9
11	Tu	9.5	24.2	0	2.4	9.2	NNE	22	16:29	14.6	64	1	SSW	6	1022.7	23.3	39	3	NE	9	1017.5
12	We	10.3	26.0	0	3.2	9.9	W	20	14:16	15.8	63	0	S	6	1019.2	25.7	27	1	SW	7	1015.4
13	Th	9.5	23.5	0	2.8	4.5	SE	28	18:47	13.4	66	7	SW	7	1021.8	21.6	51	7	E	13	1020.1
14	Fr	13.3	19.3	0	3.4	0.0	SSE	26	15:54	15.9	63	8	SSW	7	1026.3	18.7	63	8	ESE	15	1022.8
15	Sa	12.6	19.8	0	1.6	1.0	NNE	30	12:49	15.9	84	7	S	2	1021.2	18.5	80	7	NNE	9	1016.7
16	Su	15.8	21.2	1.8	1.0	2.8	W	46	22:37	19.0	89	6	N	7	1014.9	19.7	59	6	WSW	13	1012.9
17	Mo	10.7	20.4	0	4.4	8.9	W	37	03:00	14.1	56	4	SW	9	1017.5	19.1	40	2	WSW	11	1013.6
18	Tu	6.8	21.7	0.2	3.8	9.2	WSW	24	14:01	14.6	62	1	WSW	9	1017.6	20.0	35	3	W	9	1014.6
19	We	11.4	22.1	0	3.0	9.7	ESE	39	12:52	17.7	49	1	SSW	13	1022.1	19.8	39	1	SSE	17	1021.5
20	Th	10.4	19.6	0	4.8	6.2	SSE	33	10:37	15.3	55	2	SSW	13	1026.6	17.8	56	7	S	11	1024.6
21	Fr	9.3	21.6	0	3.2	9.7	ESE	35	16:38	14.7	56	1	SW	9	1029.5	19.3	40	2	SE	13	1026.4
22	Sa	11.3	21.2	0.4	3.0	5.5	ESE	33	13:23	14.4	82	5	SSW	7	1030.7	19.1	50	4	ESE	15	1027.1
23	Su	12.1	20.5	0.6	2.8	0.8	E	28	11:44	14.9	79	7	SW	6	1027.3	18.0	63	7	ESE	15	1022.8
24	Mo	11.6	21.7	0	2.0	7.0	SSW	50	03:37	15.0	79	7	SW	6	1023.3	20.3	55	4	ENE	13	1018.9
25	Tu	13.3	21.1	2.6	3.2	4.2	E	22	14:52	16.0	88	7	SW	7	1023.7	19.7	66	7	E	13	1021.4
26	We	14.0	24.3	0.2	0.6	8.5	SSW	20	11:15	16.4	81	6	SW	7	1024.2	23.6	48	4	SE	7	1020.1
27	Th	12.2	24.3	0	2.4	7.2	ENE	17	17:11	14.1	94	1	SW	7	1022.2	24.2	48	7	NE	9	1017.7
28	Fr	14.0	19.5	9.6	3.6	0.0	SSW	22	10:34	16.6	95	8	SW	9	1016.4	19.0	90	8	SSW	7	1013.3
29	Sa	10.2	24.3	16.8	2.0	10.2	SW	39	13:46	18.5	68	1	SSW	6	1019.5	23.5	30	1	W	24	1017.1
30	Su	10.9	26.3	0.2	3.6	10.1	NE	20	15:48	16.6	64	0	SSW	9	1022.4	24.9	22	1	NNE	9	1018.4
31	Mo	9.5	22.4	0	3.6	5.1	WNW	33	12:02	16.9	60	6	NE	6	1020.1	21.5	42	7	NW	7	1015.9
Statistics for July 2006																					
	Mean	10.7	22.5		3.0	7.2				15.5	67	3		7	1022.6	21.3	44	3		11	1019.2
	Lowest	6.8	19.3		0.6	0.0				13.1	47	0	S	2	1014.9	17.8	19	1	#	7	1012.9
	Highest	15.8	26.3	16.8	5.2	10.2	SSW	50		19.0	95	8	SSW	13	1030.7	25.7	90	8	W	24	1027.1
	Total			32.4	92.6	223.5															

Temperature, humidity, wind, pressure and rainfall observations are from Brisbane {station 040913}. Cloud, evaporation and sunshine observations are from Brisbane Aero {station 040842}

Brisbane Airport is located about 12 kilometres north east of the Brisbane City site, and closer to the coast. The evaporation, sunshine and cloud values should be used as a guide only.

IDCJDW4019.200607 Prepared at 16:24 GMT on 2 Sep 2006

Brisbane, Queensland
August 2006 Daily Weather Observations

Most observations from Brisbane City, but some from Brisbane Airport.

Australian Government
Bureau of Meteorology

Date	Day	Temps Min	Temps Max	Rain	Evap	Sun	Max wind gust Dirn	Max wind gust Spd	Max wind gust Time	9am Temp	9am RH	9am Cld	9am Dirn	9am Spd	9am MSLP	3pm Temp	3pm RH	3pm Cld	3pm Dirn	3pm Spd	3pm MSLP
		°C	°C	mm	mm	hours		km/h	local	°C	%	eighths		km/h	hPa	°C	%	eighths		km/h	hPa
1	Tu	9.6	21.8	0	4.0	10.1	WNW	24	13:13	16.4	53	1	SW	7	1019.1	21.1	29	1	WNW	11	1015.3
2	We	8.4	21.1	0	3.4	6.4	NE	20	13:59	13.6	54	7	SW	7	1018.9	19.6	34	7	NNE	7	1015.1
3	Th	6.1	21.9	0	2.8	9.2	SSE	43	05:59	15.1	48	2	SW	9	1017.5	21.2	28	1	SSE	11	1014.6
4	Fr	8.5	22.7	0	3.6	10.1	WSW	31	12:17	15.5	53	1	SW	11	1019.8	22.5	24	2	SW	9	1015.6
5	Sa	10.5	21.1	0	4.6	9.4	SSW	43	14:35	15.4	50	1	WSW	13	1018.7	19.1	45	7	SSW	20	1016.4
6	Su	13.2	22.5	0	4.4	7.4	SSW	37	13:24	18.3	47	1	SW	9	1021.3	21.1	39	7	S	11	1020.0
7	Mo	11.0	21.6	0	4.0	7.2	S	31	14:55	17.5	49	6	SW	7	1025.8	19.5	44	7	S	17	1024.5
8	Tu	11.4	23.0	0	3.6	8.5	NNW	65	11:13	17.4	44	1	SSW	13	1027.8	21.7	36	6	SSW	11	1025.1
9	We	11.7	25.2	0.2	3.6	10.2	W	28	14:30	18.1	46	0	SW	13	1026.1	24.0	29	1	WSW	15	1020.0
10	Th	6.4	23.4	0	4.0	10.2	SSE	28	14:51	17.3	31	0	WSW	13	1020.0	22.5	21	1	SSW	11	1019.4
11	Fr	7.9	23.4	0	4.8	10.2	NNE	22	15:00	15.6	48	1	SW	9	1026.0	21.6	47	1	NNE	11	1023.3
12	Sa	10.4	24.6	0	3.4	10.0	ENE	22	15:10	16.8	70	1	SW	6	1026.5	21.6	48	1	NE	13	1022.4
13	Su	11.0	23.6	0	3.8	10.1	NNE	24	16:03	15.6	75	5	SW	7	1025.2	22.0	52	1	NNE	11	1020.5
14	Mo	10.5	26.2	0	3.0	10.4	WNW	30	12:14	17.8	67	0	S	2	1021.6	23.7	43	1	NNE	11	1016.8
15	Tu	9.2	25.6	0	4.0	10.7	WSW	30	12:49	16.9	39	1	SSW	6	1020.5	25.2	16	1	SSW	7	1017.0
16	We	10.4	23.7	0	4.0	10.5	SE	31	13:53	17.0	56	1	SSW	9	1025.1	21.9	41	1	E	17	1023.1
17	Th	10.0	22.9	0	4.0	9.6	ENE	24	16:36	17.1	60	1	SSW	7	1027.4	21.8	37	2	ENE	15	1023.4
18	Fr	11.5	24.3	0	4.0	9.6	N	22	13:25	17.6	63	3	SW	6	1025.8	23.5	45	4	NE	9	1021.3
19	Sa	10.5	26.7	0	3.8	10.5	NE	20	14:50	17.7	63	1	SW	9	1023.8	24.8	35	1	NE	11	1019.4
20	Su	9.5	25.6	0	5.2	10.5	ENE	28	15:31	18.2	54	0	SW	9	1024.2	23.3	50	1	ENE	15	1020.0
21	Mo	11.9	24.6	0	4.0	9.3	NNE	28	16:00	18.6	72	1	SW	6	1022.6	21.4	59	5	NNE	11	1017.7
22	Tu	10.7	26.9	0	3.8	10.3	NNE	24	15:12	16.6	54	1	WSW	9	1019.9	25.7	31	2	NNE	11	1013.9
23	We	10.2	26.3	0	4.0	10.7	ENE	26	14:13	21.1	31	0	SW	9	1019.1	23.5	33	0	E	19	1016.5
24	Th	10.9	24.9	0	5.2	10.3	N	26	15:35	17.3	61	1	SW	6	1020.9	22.9	60	1	NE	15	1015.8
25	Fr	13.6	28.4	0	4.0	10.6	WSW	28	18:00	18.4	75	1	W	6	1014.8	26.4	49	2	NE	15	1009.0
26	Sa	13.9	25.6	0	3.6	8.5	ENE	31	16:12	20.0	60	7	WSW	7	1017.2	22.8	55	3	ENE	15	1014.8
27	Su	14.6	23.4	0	4.8	9.5	E	24	14:06	19.7	66	2	S	6	1021.8	23.1	55	1	ENE	15	1018.4
28	Mo	16.7	23.9	20.2	4.6	4.8	SW	33	03:37	20.5	80	6	S	7	1020.1	23.2	62	2	ENE	13	1017.0
29	Tu	14.9	22.5	24.2	4.0	2.8	SW	24	15:56	17.3	81	7	SSW	7	1022.8	20.5	58	7	SSW	11	1020.9
30	We	13.1	24.6	0	1.4	3.9	ESE	31	13:26	16.4	84	6	WSW	9	1026.0	17.4	90	8	E	11	1023.8
31	Th	14.5	21.1	10.6	3.6	1.8	ESE	41	11:24	18.1	90	7	SE	11	1026.2	17.8	91	7	E	19	1023.3
Statistics for August 2006																					
	Mean	11.1	24.0		3.9	8.8				17.4	58	2		8	1022.3	22.1	44	2		12	1018.8
	Lowest	6.1	21.1		1.4	1.8				13.6	31	0	S	2	1014.8	17.4	16	0	#	7	1009.0
	Highest	16.7	28.4	24.2	5.2	10.7	NNW	65		21.1	90	7	#	13	1027.8	26.4	91	8	SSW	20	1025.1
	Total			55.2	121.0	273.3															

Temperature, humidity, wind, pressure and rainfall observations are from Brisbane {station 040913}. Cloud, evaporation and sunshine observations are from Brisbane Aero {station 040842}

Brisbane Airport is located about 12 kilometres north east of the Brisbane City site, and closer to the coast. The evaporation, sunshine and cloud values should be used as a guide only.

IDCJDW4019.200608 Prepared at 13:19 GMT on 7 Sep 2006

Users of this product are deemed to have read the information and accepted the conditions described in the notes at http://www.bom.gov.au/climate/dwo/IDCJDW0000.pdf

Appendix E

Compiled Data Logger and Meterological Data **May**

9am Readings
May

	Ambient	Workstation	Roof
8th	21.9	23	23
9th	18.6	20.5	20.5
10th	19.3	21	21
11th	18.6	20.5	20.5
12th	18	21	21.5
15th	19.4	22	21.5
16th	20.1	22	22
17th	19.9	22	21.5
18th	19.5	22	22
19th	20.3	22.5	22
22nd	19.8	22	21.5
23rd	17.2	22	22
24th	15.5	22.5	23
25th	17.1	23.5	23.5
26th	17.1	23.5	23.5
29th	16.6	21	21
30th	15.9	22	22.5
31st	11.6	21	21.5

3pm Readings
May

	Ambient	Workstation	Roof
8th	27.7	25.5	28.5
9th	24.5	27.5	28.5
10th	25.3	24.5	27
11th	24.5	27.5	28.5
12th	21.7	26.5	27.5
15th	25.4	27	28.5
16th	22.4	26.5	27.5
17th	21.9	25.5	26.5
18th	22.3	26	27
19th	22.4	26	27
22nd	22.4	25	26.5
23rd	22.9	26.5	28
24th	22.6	25.5	27.5
25th	22.9	25.5	28
26th	22.9	25.5	28
29th	21.6	24.5	26.5
30th	23.5	24.5	27
31st	21.8	24.5	26.5

Compiled Data Logger and Meterological Data **June**

9am Readings
June

	Ambient	Workstation	Roof
1st	13.7	20.5	22
2nd	12.9	24	25.5
5th	15.6	20	19.5
6th	16.3	21.5	21
7th	15.5	23	24
8th	16.7	21.5	20.5
9th	19	21.5	21
12th	15.3	17	17.5
13th	16.1	19.5	19
14th	16	22.5	23
15th	15.9	24	24.5
16th	14.7	22	22
19th	16.1	23	23
20th	15.9	22.5	23
21st	15.7	25	24
22nd	15.3	22	21.5
23rd	15.5	22	22
26th	17	23	22
27th	14.5	22.5	20
28th	14.6	22	20.5

3pm Readings
June

	Ambient	Workstation	Roof
1st	22.7	24.5	26.5
2nd	21.3	26.5	28.5
5th	22.4	27	28.5
6th	20.3	26.5	28
7th	22.1	27.5	29
8th	23.2	27	28.5
9th	21.3	27	27
12th	20.4	23.5	25
13th	19.5	26	26.5
14th	21.5	26.5	27.5
15th	22.7	26	27.5
16th	22	24	25.5
19th	18.6	25	26
20th	15.5	25	25.5
21st	18	23.5	26
22nd	17.4	24.5	25
23rd	19	24.5	26
26th	18.9	24	26.5
27th	20.9	24.5	27.5
28th	20.9	25.5	28

29th	14.3	22.5	21
30th	17.4	24	24

29th	18.8	24.5	25
30th	21.2	25	28

Compiled Data Logger and Meterological Data **July**

9am Readings
July

	Ambient	Workstation	Roof
1st	17.1	21.5	21.5
2nd	14.5	17.5	18.5
3rd	13.9	21	20
4th	15.2	22	21.5
5th	17	23.5	23.5
6th	15.2	25.5	24
7th	16.2	21	22
8th	14.3	18	19
9th	13.1	16.5	17.5
10th	13.6	20.5	22
11th	14.6	26.5	23.5
12th	15.8	24	22
13th	13.4	24.5	21
14th	15.9	24	22
15th	15.9	24	22.5
16th	19	20.5	20.5
17th	14.1	23	20
18th	14.6	24	20
19th	17.7	25.5	22.5
20th	15.3	23	19.5
21st	14.7	25	22
22nd	14.4	21	18
23rd	14.9	17.5	18.5
24th	15	25.5	22.5
25th	16	24	24.5
26th	16.4	23.5	22
27th	14.1	23	23.5
28th	16.6	23.5	24
29th	18.5	19	20
30th	16.6	18.5	19
31st	16.9	23	22.5

3pm Readings
July

	Ambient	Workstation	Roof
1st	22.5	26	27
2nd	21.8	23.5	25.5
3rd	22.5	24.5	27
4th	21	25	27
5th	21.5	25.5	27.5
6th	24	26.5	29.5
7th	21.4	25.5	28
8th	23.1	24.5	26.5
9th	22.2	24.5	26
10th	22.7	25.5	28
11th	23.3	25.5	27
12th	25.7	24.5	29
13th	21.6	24	28
14th	18.7	24.5	25
15th	18.5	22.5	23.5
16th	19.7	23	24
17th	19.1	24	27
18th	20	23	27
19th	19.8	22.5	26
20th	17.8	25	26
21st	19.3	23	27
22nd	19.1	23	23
23rd	18	23	24.5
24th	20.3	23	27
25th	19.7	22.5	27
26th	23.6	24	28.5
27th	24.2	24	28
28th	19	22	24.5
29th	23.5	22.5	26.5
30th	24.9	24	28
31st	21.5	23	27

Compiled Data Logger and Meterological Data

August

9am Readings
August

	Ambient	Workstation	Roof
1st	16.4	27.5	24
2nd	13.6	25	21.5
3rd	15.1	23	22
4th	15.5	20	22.5
5th	15.4	22	18
6th	18.3	20.5	20.5
7th	17.5	24	25
8th	17.4	23	23
9th	18.1	24	24
10th	17.3	23	21
11th	15.6	22	21.5
12th	16.8	24.5	20.5
13th	15.6	24.5	23
14th	17.8	25	23.5
15th	16.9	23.5	21
16th	17	24	22
17th	17.1	23.5	23
18th	17.6	23	22.5
19th	17.7	23.5	22.5
20th	18.2	19	19.5
21st	18.6	23.5	22.5
22nd	16.6	22.5	22.5
23rd	21.1	23.5	21.5
24th	17.3	20.5	21
25th	18.4	23	24
26th	20	22	23
27th	19.7	21	21.5
28th	20.5	22.5	24.5
29th	17.3	25.5	23
30th	16.4	26	22.5
31st	18.1	26	23

3pm Readings
August

	Ambient	Workstation	Roof
1st	21.1	23	28
2nd	19.6	24	25.5
3rd	21.2	23	27.5
4th	22.5	23.5	27.5
5th	19.1	23.5	24
6th	21.1	24.5	25.5
7th	19.5	22.5	26.5
8th	21.7	23.5	28
9th	24	25	29.5
10th	22.5	23	28
11th	21.6	23.5	28.5
12th	21.6	26	27.5
13th	22	26	27
14th	23.7	25	30
15th	25.2	24	29.5
16th	21.9	22	26.5
17th	21.8	24	29
18th	23.5	24.5	29.5
19th	24.8	24	29.5
20th	23.3	23	28.5
21st	21.4	24.5	29.5
22nd	25.7	24.5	29.5
23rd	23.5	21.5	28
24th	22.9	23	28.5
25th	26.4	22	28
26th	22.8	25.5	29.5
27th	23.1	25.5	27
28th	23.2	23.5	27.5
29th	20.5	24	25.5
30th	17.4	23	26
31st	17.8	23.5	25