SIMULATION OF A BUILDING HEATING, VENTILATING AND AIR-CONDITIONING SYSTEM

C.P. Botha

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ABSTRACT

Title: Simulation of a building heating, ventilating and air-conditioning system

Author: Cor Botha

Supervisor: Prof. E.H. Mathews

Department: Mechanical and Aeronautical Engineering

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Key terms: Integrated HVAC system simulation; Simulation model verification; HVAC system simulation applications; Optimising energy consumption; Control retrofits; Effect of equipment failure; Cooling coil fouling; Chiller failure.

Simulation is one of the oldest and also among the most important tools available to engineers. In the building Heating, Ventilating and Air-Conditioning (HVAC) community the availability and/or functionality of simulation tools is limited and it is difficult to determine whether the simulation models accurately represent reality. The purpose of this study was to accurately verify one such a simulation model and then to extend the study to two unique applications.

Comprehensive structural, comfort and energy audits were performed to construct a suitable simulation model with the aid of the control simulation package: QUICK Control. The model was then verified against measured building data to ensure an accurate representation of the actual dynamic building response. For the first application various control retrofits were evaluated and the highest potential for energy saving was found. Thereafter the model was implemented to investigate the change in indoor air conditions due to failure of HVAC equipment.

Heating, ventilating and air-conditioning in buildings consume a significant portion of the available electrical energy in South Africa. Of this energy up to 30% can be saved by improving the HVAC systems currently installed in the buildings. This could result in savings of up to R400 million. For the building used in this study it was found that up to 66% of the HVAC system's electrical energy consumption could be saved with a payback period of only 9 months. These savings could be achieved by implementing a setback control strategy with an improved time management procedure.

Predicting the impact of failing equipment is a difficult task because of the integrated dynamic effect every HVAC component has on the next. With the aid of a comprehensive integrated simulation model the implications of failing can be determined and necessary assessments and precautions can be taken. The results of this study showed that the air-conditioning system under investigation was approximately 100% over designed. Failure of up to 50% was allowable in the cooling equipment before any noticeable impact could be observed in the indoor climate. With further failure the required comfort conditions could not be sustained.

The substantial savings calculation and possibility of predicting climate deterioration would not have been possible without the aid of a comprehensive simulation package and model. This study clearly highlights the worth of integrated simulation.

SAMEVATTING

Titel:	Simulation of a building heating, ventilating and air-conditioning system
Outeur:	Cor Botha
Studieleier:	Prof. E.H. Mathews
Departement:	Meganiese en Lugvaartkundige Ingenieurswese
Graadkursus:	Meester van Ingenieurswese (Meganies)
Sleutel terme:	Integrated HVAC system simulation; Simulation model verification; HVAC system simulation applications; Optimising energy consumption; Control retrofits; Effect of equipment failure; Cooling coil fouling; Chiller failure.

Simulasie is een van die oudste en ook een van die mees belangrike hulpmiddels vir ingenieurs. In die lugversorgingsveld is simulasieprogramme nie geredelik beskikbaar nie en dit is moeilik om te bepaal of die simulasiemodelle die werklikheid akkuraat naboots. Die doel van hierdie studie was om een so 'n simulasiemodel akkuraat te verifiëer en dan aan te wend vir twee unieke toepassings.

'n Reeks breedvoerige ondersoeke was geloots op die struktuur, gemakstoestande en energieverbruik van 'n gebou om 'n simulasiemodel op te stel met die hulp van die simulasieprogram: QUICK Control. Die model was toe geverifiëer teen gemete waardes om 'n akkurate dinamiese respons te verseker. Vir die eerste toepassing was verskeie opgraderings beoordeel om die hoogste energiebesparingsmoontlikheid te vind. Daarna was die model aangewend om die verandering in interne toestande as gevolg van toerusting faling te ondersoek.

Lugversorgingsstelsels verbruik 'n aansienlike gedeelte van Suid-Afrika se beskikbare elektriese energie, waarvan tot soveel as 30% bespaar kan word deur opgraderings van die huidige beheerstelsels. Dit kan lei tot besparings van tot R400 miljoen. Hierdie studie het gewys dat, in hierdie spesifieke gebou, besparings van tot 66% van die elektriese energieverbruik moontlik is met 'n terugbetaaltyd van slegs 9 maande. Dit kan bewerkstellig word deur die gebruik van 'n "setback" beheerstrategie met verbeterde tydsbestuur.

Voorspelling van die impak van oneffektiewe toerusting is 'n moeisame proses as gevolg van die geïntegreerde dinamiese effek van al die komonente op mekaar. Met behulp van 'n breedvoerige geïntegreerde dinamiese model kan die implikasies van faling ondersoek word en nodige voorsorg kan getref word. Hierdie studie het daarop gedui dat die betrokke lugversorgingsstelsel ongeveer 100% oorontwerp is. Dus was faling van tot 50% toelaatbaar in die verkoelingstoerusting voordat enige merkbare verandering in die interne toestande waargeneem is. Eers met verdere faling kon die gemakstoestande nie gehandhaaf word nie.

Die berekening van besparings en voorspelling van toestandsagteruitgang sou nie moontlik gewees het sonder die hulp van 'n omvattende geïntegreerde simulasiepakket nie. Hierdie studie beklemtoon duidelik die waarde van geïntegreerde simulasie.

ACKNOWLEDGEMENTS

I would like to express my gratitude to Prof. E.H. Mathews for the opportunity to perform this study. His guidance, support and motivation throughout is greatly appreciated.

I would also like to thank Transfer of Energy Mass and Momentum International (Pty) Ltd for both their financial support and the use of their software package: QUICK Control.

Lastly, I would like to thank my parents, family and friends. Your loyalty, support and encouragement carried me all the way.

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NOMENCLATURE

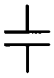
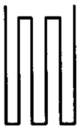
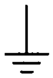
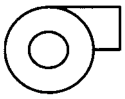
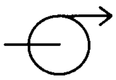
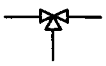
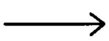
A	Area
a	Regression coefficient, Air
b	Regression coefficient
c	Regression coefficient
C	Capacitance
c	Regression coefficient, Specific heat capacity, Convective
D	Diameter
d	Regression coefficient, Hydraulic diameter
dp	Dew point
e	External, Out of
f	Flow, Fouling factor
h	Enthalpy, Pressure head, Convection coefficient
hm	High mass
i	Internal, Into
K	Non-dimensional coefficient
k	Conduction coefficient
l	Liquid
lm	Low mass
m	Mass, Mass flow
n	Rotational speed
P	Pressure
Pwr	Power consumption
p	Pressure
Q	Heat gain, Cooling capacity
R	Resistance
r	Radiative
S	Sensible heat ratio
s	Surface
ss	Steady state
T	Temperature

t	Temperature
U	Thermal conductance
V	Volume, Velocity
v	Ventilation
w	Humidity ratio
Δ	Change in
η	Efficiency
ρ	Density
μ	Viscosity

LIST OF ABBREVIATIONS

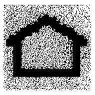
ACH	Air changes
AHU	Air handling unit
ASHRAE	American society of heating refrigeration and air-conditioning engineers
CO_2	Carbon dioxide
COP	Coefficient of performance
FCU	Fan coil unit
HSB	Human science building
HVAC	Heating ventilating and air-conditioning
LoanSTAR	Loan to Save Taxes And Resources
PA	Primary air unit
PID	Proportional integral differential
VAV	Variable air volume

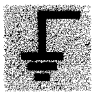
GRAPHICAL SYMBOLS

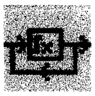
	Air flow
	Capacitor
	Cooling coil
	Control signal
	Dampers
	Earth
	Electrical heating element
	Fan
PID	Proportional integral differential controller
	Pump
	Resistor
STEP	Step controller
	Temperature input (current source)
	Temperature input (potential source)
T	Temperature sensor
	Three way valve
	Two way valve
	Water flow

SIMULATION MODEL SYMBOLS

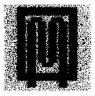
Air flow

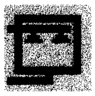

Building zone


Climate input

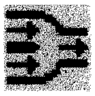

Controller

Control signal

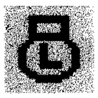

Cooling coil


Cooling tower


Electrical heater


Flow converge


Flow diverge


Liquid cooled chiller


Temperature sensor

Water flow

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CHAPTER 1 - Introduction

Commercial buildings consume a large percentage of a country's available electrical energy. By improving the energy management of the buildings up to R400 million per year may be saved in South Africa alone. The purpose of this study was to successfully simulate a HVAC system of one such a building. With the integrated simulation model the highest potential for energy saving could be found and accurate predictions could be made concerning the impact of failing equipment. In this chapter a brief background of the project is provided, the beneficiaries are identified and an outline of the study is stipulated.

1.1 Introduction

Simulation is one of the oldest and also among the most important tools available to engineers. It is supposed to be easier, quicker and less expensive to simulate a real life event than to actually observe it in practice. Through simulation the age-old question of: "What would occur if..." may be answered [1].

Usually the level of integration between the building, the heating ventilating and air-conditioning (HVAC) system and its control system is unreliable and modelling is somewhat suspect [2]. This implies that the validity and the integrity of the simulation model have to be verified.

The aim of this study was to accurately simulate a building currently in use with its HVAC system and control. The simulation model was verified to closely represent reality before being implemented for two unique applications: Finding the energy savings potential of the building and finding the impact of equipment failure on indoor conditions.

1.2 Motivation for this study

The building sector consumes roughly one third of the final energy used in most countries and it absorbs an even more significant share of the electricity [3]. In developed countries commercial buildings consume a large percentage of a country's electrical energy, approximately 57% of all electricity generated, whilst in less developed countries buildings account for 38% of the electricity consumed [4],[5]. The electricity use in developing countries has grown by more than 11% per year over the 1980's [6]. Electricity use in commercial buildings is reaching peak demand in the United States, Japan and some other wealthy, but less developed countries [7]. Meanwhile, South African studies show that 20% of the total municipal energy generated is utilised in commercial buildings [8].

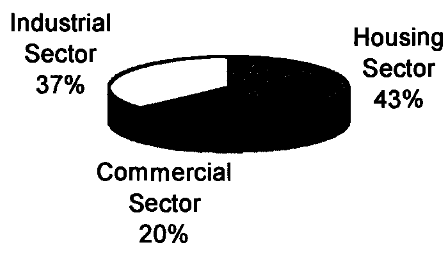

Figure 1-1 - Total electrical energy use in South African buildings

This results in energy expenditures of approximately R4000 million per year in inflation corrected terms [9]. Many countries without effective energy-efficient programmes for their buildings have gathered energy consumption data that can support such programme development. Audits of government office buildings in India suggest that 33% of the actual electricity use in a typical building could be saved through management and technological changes [10]. Similarly estimates from the former Soviet Union suggest that 25% of the energy used in existing buildings can be saved [11]. It is believed that 10% of all energy consumed in the world is expended by building air-conditioning systems and therefore a large potential for energy savings in heating ventilating and air-conditioning systems exists [12].

It is commonly agreed that energy savings of around 30% may be realised through the use of improved design and energy management practices, as well as through retrofit projects in existing commercial buildings [13]. If a 30% penetration in the industry with a 30% saving per building could be realised in South Africa it would result in energy savings of approximately R400 million per year [9],[13]. This money could be spent on more worthy causes.

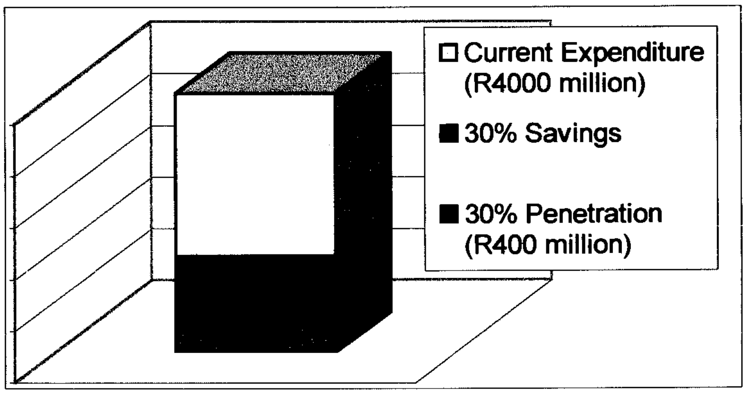

Figure 1-2 - Illustration of a possible R400 million saving out of a total of R4000 million expenditure

In order to calculate the potential for energy savings in South African commercial buildings, it was necessary to investigate similar procedures as those applied internationally. Numerous audit schemes have been performed abroad to determine the energy savings possible for their countries. These schemes consisted mainly of energy and comfort audits and provided the researchers with information on characteristics of energy use and the conditions of the indoor building environment. The outcome resulted in very useful information. It identified the areas in buildings with energy savings potential [14].

The successful American Texas, Loan to Save Taxes And Resources (LoanSTAR) programme, has the unique feature of measuring and reporting on energy savings from certain retrofits [15]. This feature was made possible by continuous yearly monitoring of buildings' energy channels. This approach has shown great benefits for large buildings, since it has proved to be a useful complement to the current energy-management control-systems. However, the savings achieved in smaller buildings of less developed countries do not justify the expenses of the costly continuous monitoring [16].

An important factor that has prevented accurate energy-savings potential prediction without continuous monitoring has been the lack of simple, low-cost and readily available analytical software tools [17]. Traditional design philosophy is based on load calculations only and is therefore not adequate to provide reliable, useful information on the energy performance of various designs. In order for an analytical tool to be reliable, a system approach that involves the building's lighting system, HVAC system and its controllers is necessary [12]. Until recently no such tools have been available.

Since the introduction of a new integrated HVAC simulation tool various simulation possibilities opened up [18]. One such an application is that of determining the effect fouling has on indoor air conditions and overall energy consumption. Many researchers have spent countless hours deriving the fundamentals surrounding fouling, but very little work has been done to determine the impact of the fouling on the entire HVAC system [19],[20]. One of the objectives of this study was to find the worth of the simulation package and model in diverse applications like this.

Another objective was to determine the energy savings potential of a building currently in use. A new holistic approach to retrofitting as well as a new analytical modelling technique which integrates the simulation of the building's heating ventilating and air-conditioning system with its control system were implemented in this study [21],[22].

1.3 Beneficiaries of this study

In order to determine the value of this study, the parties who will benefit the most from the work that was performed must be identified. For the beneficiaries listed below, the criterion for a successful study is discussed, along with the manner in which the results could be implemented, and the potential impact thereof.

The University of Pretoria

The building used for this project was the Human Sciences building on campus at the University of Pretoria. As building owners the University will be satisfied with the project if substantial low cost energy saving retrofits could be found. They will also be pleased if the impact of the building's equipment failure could be quantified and predictions of deteriorating indoor conditions could be assessed.

As primary beneficiary the university may implement the results of this study directly to capitalise on the potential monetary savings proposed in this text. Furthermore, this project may also spark similar studies to be implemented in other facilities of the university.

By not wasting money on unnecessary energy consumption, the monetary savings can be applied to more worthy causes.

Consulting engineers and building managers

For the engineer and building manager to predict the effect of fouling or the potential impact of energy retrofits a simulation method, like the one proposed in this study, is imperative.

The primary objective of this study is to accurately simulate the HVAC system of the building with all its integrated control parameters and dynamic responses. Similar studies will allow the engineer to not only apply retrofits to preinstalled systems but also design a building and its HVAC system components with detailed predictions of system responses.

With an integrated tool such as this the engineer and manager will be able to provide a better service. This in turn will result in more projects, which ultimately means more money.

ESKOM

The postponement of a new power station at a typical cost of R16 billion can be achieved by promoting energy efficient buildings nation wide [23]. HVAC simulations of this nature will not only be valuable but necessary to obtain these energy efficient buildings.

1.4 Outline of study

The scope of the study can be illustrated schematically as follows:

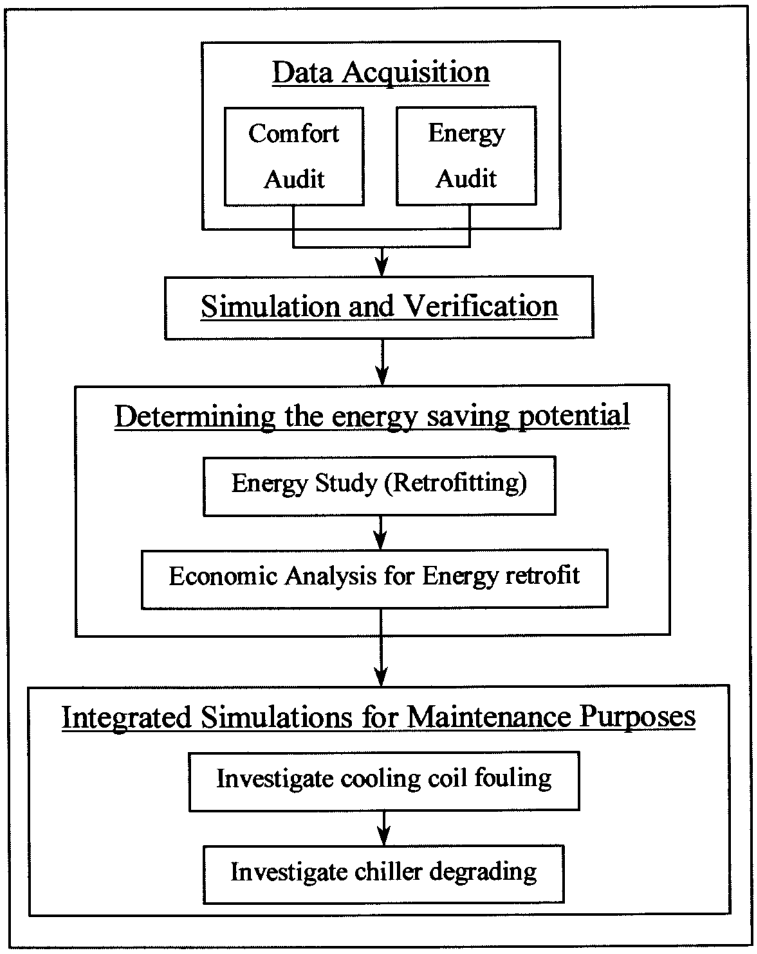

Figure 1-3 - Schematic illustration of the methodology followed during this study

1.4.1 Data acquisition

Acquiring data about the current building system involved the measurement of two distinct elements of the building: The comfort conditions and the energy consumption. Both these elements were measured over a period of time and a suitable test sample was selected for the verification of the simulation model.

Most audits fall into three categories, namely walk-through, mini- and maxi audits [14]. A walk-through audit is the least costly and therefore the most popular audit performed. It identifies preliminary savings by making a visual inspection of the building facility, maintenance and operation activities. The mini-audit requires tests and measurements to quantify energy use and losses and temperature ranges. Both the aforementioned audits allow collection of information to determine the need for a more detailed analysis. In order to accurately predict the energy savings potential of any building it is necessary to perform a maxi-audit [24]. The maxi-audit is the detailed step. It contains an evaluation of how much energy is used for each building function such as air-conditioning, lighting, etc. It also requires a model analysis like a computer simulation to determine energy usage patterns and predictions on a year-round basis, taking into account such variables as weather data.

Comfort Audit

The aim of the comfort audit was to evaluate the prevailing indoor air-conditions. These conditions were needed for the retrofit calculations to ensure that the indoor temperatures would not fall outside the allowable human comfort range [25]. The objective was to save on total energy consumption without deterioration of the indoor conditions when retrofit options were considered. Where necessary the conditions were even improved prior to retrofitting.

Energy Audit

The aim of the energy audit was to obtain an end-user breakdown of the energy consumption of the building and thereby identify areas with large energy savings potential. To determine the major end-users the electrical energy consumption of the HVAC system, the lighting system and other diverse equipment were measured. The focus was then shifted to only the HVAC system. The end-user breakdown was obtained for the various components such as the pumps, fans, chillers, cooling towers and heaters. Previous studies have shown these components to possess a large potential for energy savings [12]. Furthermore, knowing exactly how much energy is consumed by each of the HVAC components, the impact of failing equipment could also be assessed better.

1.4.2 Simulation and Verification

A suitable integrated simulation model of the current HVAC system was constructed with the use of the analytical modelling software package: QUICK Control* [26]. The aim was then to improve the simulation model until the simulated conditions were sufficiently close to resembling reality. Measured data was compared with simulated data. This verification study involved the correct modelling of the various system components, control parameters, load conditions, climatic data, building structural information, etc. Calculation of various multidimensional regressions produced the component modelling input parameters, while the structural data, control strategies and load conditions were evident from a series of walk-through and maxi audits. Only after the simulation model was sufficiently verified, were the potential energy savings retrofits and fouling effects evaluated.

1.4.3 Determining the potential for energy saving

The procedure for finding the energy savings potential of the building consisted of two steps [18]. Firstly, a series of applicable retrofit options were considered and implemented in the simulation model. Then the cost of each of the options was evaluated in conjunction with the potential savings and the best option was highlighted.

Energy study (Retrofitting)

With the acquired data as a basis for decision making a few possible retrofit options were considered. These options mainly involved altering the control parameters and

* Software developed by TEMM International (Pty) Ltd and used with special permission.

control strategies of the HVAC system to save as much electrical energy as possible whilst still maintaining acceptable indoor comfort levels. Major structural alteration options were ignored.

The options considered were:

- Current system with updated control variables
- Air bypass control
- Reset control
- Setback control
- Improved operating times
- Economiser cycle
- CO_2 control

These options will be discussed in greater detail in further sections of this text.

Economic Analysis

In the previous subsection the potential savings were established for each of the retrofit options. The economical analysis then involved the evaluation of these savings by means of a straight payback period. Since the current system is not performing as desired the first retrofit option included updating the system and restoring it to its rated functionality. The monetary expenses for the improvement retrofits were then weighed against the restoration cost to calculate the straight payback period [14].

1.4.4 Integrated simulations for maintenance purposes

HVAC system maintenance has always been a point of concern for building owners [19]. As an extension of the study into integrated simulation of building energy consumption the impact of failure of HVAC components was investigated. The simulation models from the previous sections were used to simulate the fouling of equipment, and then the results and impact of failure on both comfort conditions and energy consumption were evaluated.

For this study fouling of the coils along with failing of the chillers were investigated.

1.5 References

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CHAPTER 2 - Data acquisition

To construct a realistic simulation model, every aspect of the Human Science building was investigated. One of the objectives of this study was to find the potential for energy savings in the building. In order to find this potential the current performance of the building elements was measured and evaluated through a combination of a comfort audit and an energy audit. By knowing the exact conditions of the building the effect of equipment failure could also be weighed more accurately.

2.1 Introduction

In order to correctly construct and verify a simulation model for the building in question, a thorough auditing scheme was needed. In some studies it was proved that there is a direct relationship to the cost of the audits (amount of data collected and analysed) and the number of energy saving opportunities to be found [1].

The scheme used in this study contained three distinct elements:

- walk-through audits to collect structural information, control parameters, etc.
- a comfort audit to find indoor air conditions
- an energy audit for quantifying the consumption of the system components

Most audits fall into three type categories, namely, preliminary walk-through, mini-audit and maxi-audit [1].

Preliminary Walk-Through

This type of audit is the least costly and identifies preliminary energy savings. A visual inspection of the facility is made to determine maintenance and operation energy saving opportunities plus collection of information to determine the need for a more detailed analysis.

Mini-Audit

This type of audit requires tests and measurements to quantify energy users or temperature ranges and then determines the economic implications should changes be made.

Maxi-Audit

The maxi-audit goes one step further from the mini-audit. It contains an evaluation of how much energy is consumed by each of the system components or exactly what the supply and return air conditions to the building zones are. It then implements a numerical solution technique like a computer simulation to predict energy and air condition responses for given input variables like weather data. This is a comprehensive and integrated study of all the applicable variables.

2.2 Building description

The Human Sciences building is a 22 storey, north-south oriented building consisting of both offices and lecture halls. The top 16 floors contain the offices. These offices are arranged symmetrically about the centreline of the building with one half facing to the north and the other half facing to the south. Preconditioned air is supplied to the office block and then treated individually by each of the office occupants by means of Fan Coil Units (FCUs) installed in the offices. Chilled water is supplied to these units from the main water cooling plant situated on the ground floor. Electrical heating elements are also installed in the units should heating be necessary in the offices.

The lower five floors constitute the lecture halls. The 46 lecture halls of varying size and volume are mostly situated about the perimeter of the building with doors facing a central lobby area. Conditioned air is provided to the lecture halls by means of a chilled water constant volume system to handle the total of 4265m^2 of air-conditioned floor area. Escalators in the lobby area do not constitute internal load strain on the air handling system since the lobby area does not receive treated air. The building houses approximately 1600 people per weekday.

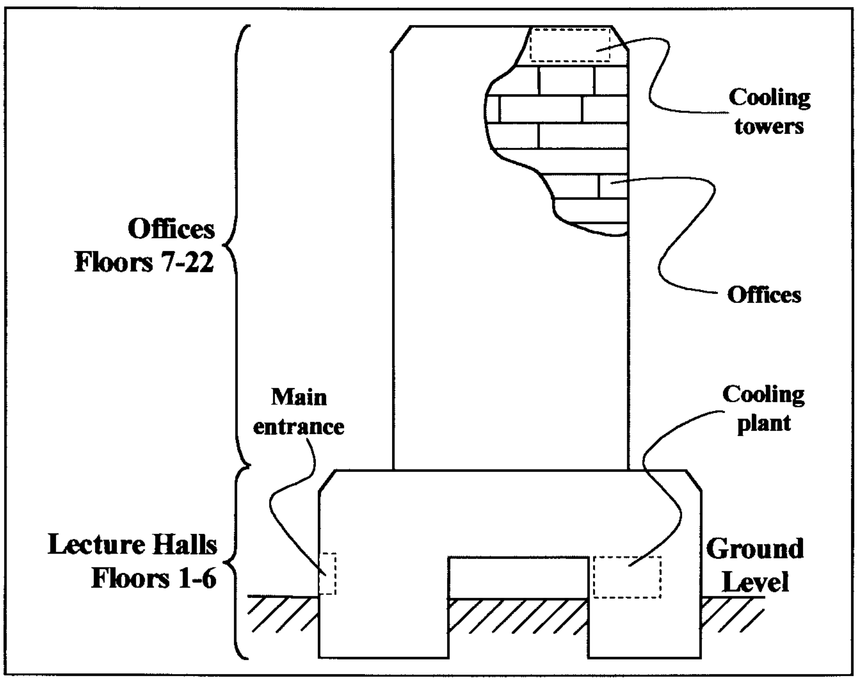

Figure 2-1 – Schematic layout of the building floor distribution

Lighting throughout the building is done with fluorescent tubes while most of the offices are equipped with personal computers and the lecture halls with overhead projectors. Other diverse load inducing equipment include photocopiers and coffee machines (about one of each per office floor). To correctly simulate the zone occupation it was assumed that the building houses one person per small office and two occupants per large office. Lecture schedule tables were used to construct the internal load distribution models for the lecture halls.

The Heating Ventilating and Air-Conditioning (HVAC) system has a rated cooling capacity of 1300kW and consists of 14 Air Handling Units (AHUs) spread out over four floors. Two of these are dedicated to the offices and the FCUs. This means that the cooling plant is able to provide 300W/m^2 of cooling, which is grossly oversized [2]. The system operates between 5am and 11pm throughout the entire week.

The remaining 12 AHUs are situated on floors 2 to 5 and supply a constant volume of air to the lecture halls. These units are numbered and their locations are as follows:

AHU	Location (Level)	Model/Make	Supplies to
1	1	MZ 80	Level 3 North
2	1	MZ 80	Level 3 South
3	1	HAH 108	Level 4 West
4	1	HAH 81	Level 1
5	2	MZ 100	Level 4 North
6	2	MZ 100	Level 4 South
7	2	HAH 108	Level 2
8	5	HAH 81	Level 6
9	5	HAH 108	Level 4 Centre
10	5	HAH 108	Level 4 Centre
11	5	MZ 80	Level 4 North East
12	5	MZ 80	Level 4 South East

Table 2-1 – Air handling unit models and locations

The AHUs all operate in a similar manner. Air is blown through a cooling coil in the unit before being split into predetermined volume fractions to each of the lecture halls. Prior to entering the lecture hall, the air is re-heated to the desired temperatures and supplied to the halls. Through return ducts running down long central shafts in the building the air is then returned to the AHUs where at this point fresh air is drawn in from the outside at a mixture ratio of 20% fresh air to 80% return air. Since no economiser system is present, this mixture ratio remains constant.

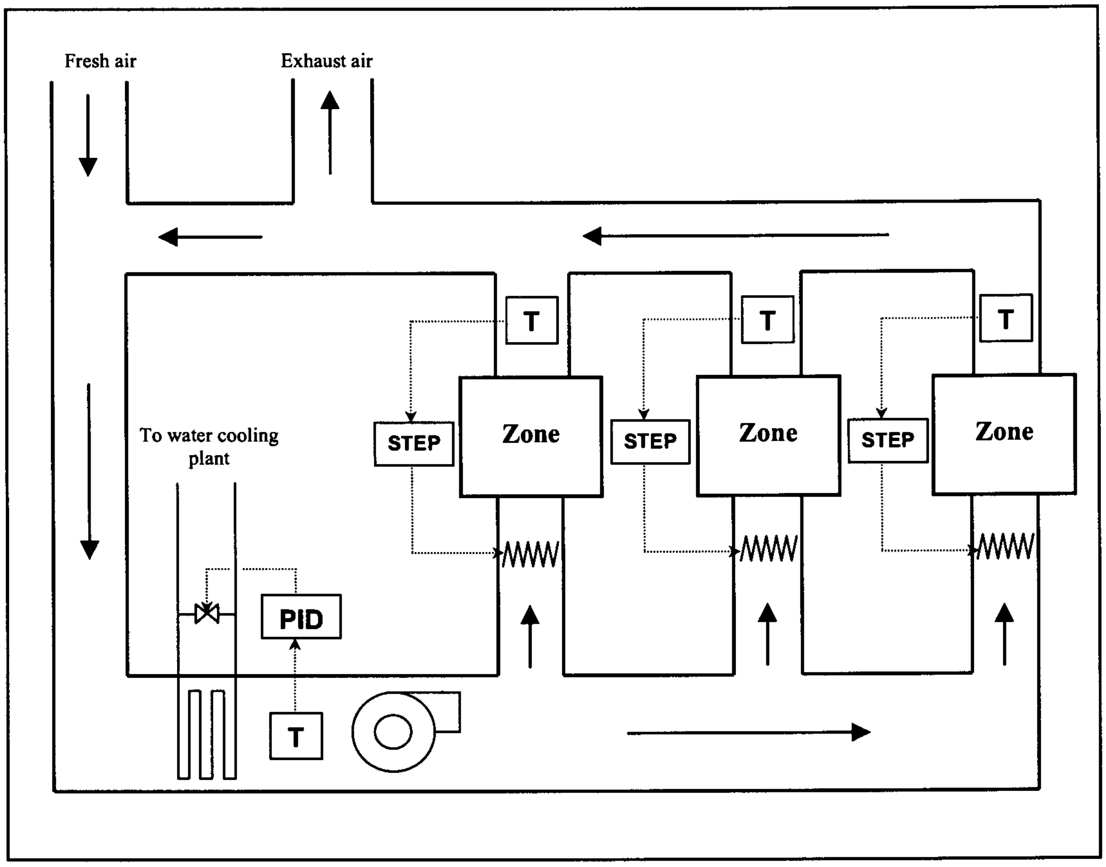

Figure 2-2 – Schematic layout of the standard air handling unit

The AHUs are set to supply a constant down-coil air temperature. This temperature setpoint is set sufficiently low to account for the maximum cooling load in any one of the supported lecture halls. Temperature control of the hall is then achieved on a per zone basis through heaters situated in the supply air ducts. The control variables for these heaters are obtained from temperature sensors placed in each of the return ducts of the control zones and then controlled with step controllers. Clearly this presents an obvious potential for saving energy since unnecessary re-heating might be avoided.

In some of the handling units (AHU 1,2,5,6,11 and 12) a coil bypass system is used. This implies that the flow of supply air through the coil is controlled with a bypass damper system that allows for better control of down coil air temperature. Should no cooling be required, no air is passed through the coil.

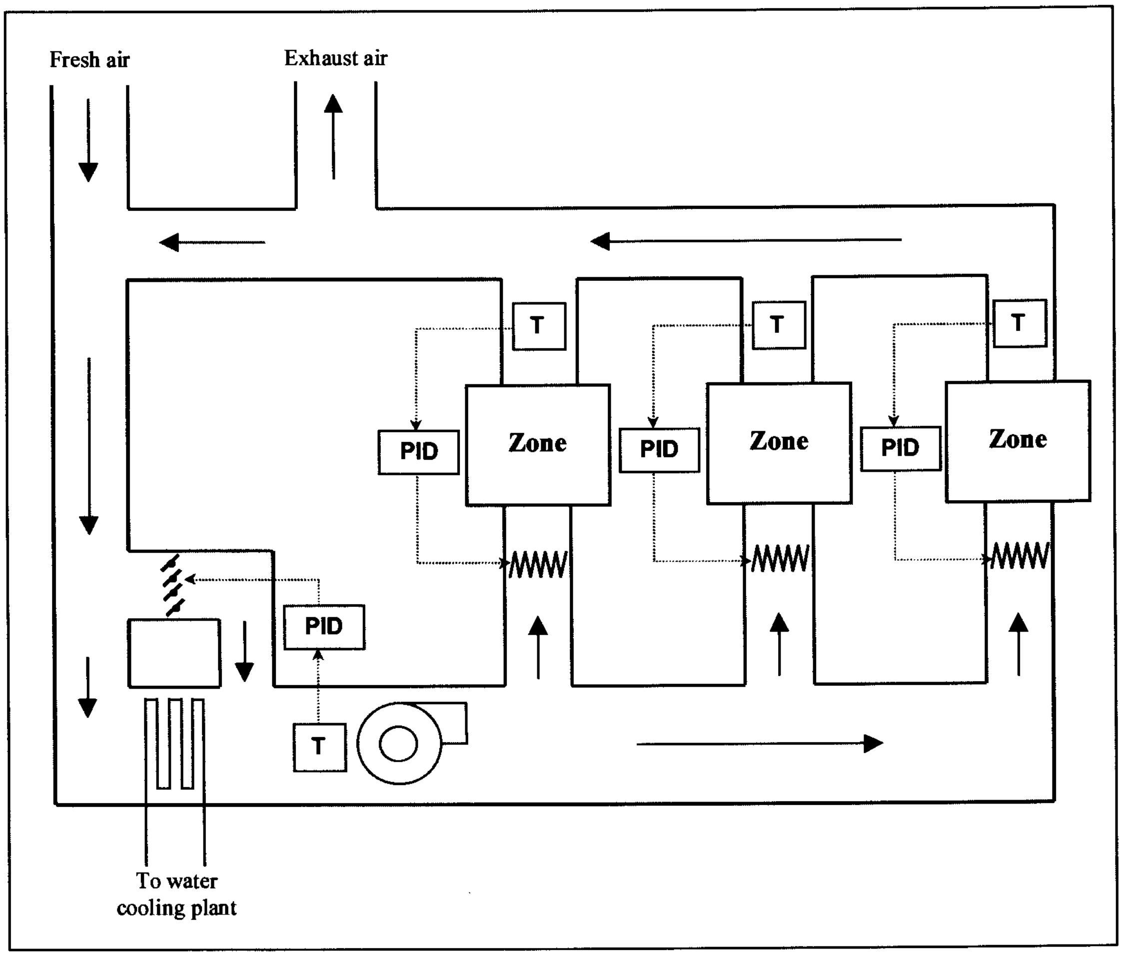

Figure 2-3 – Schematic layout of an air handling unit with coil bypass control

As already mentioned, each office is conditioned with its own FCU together with heated outside air from two AHUs located on the roof of the building. These units make use of 100% fresh air and supply air that is heated to the minimum temperature of 10°C to account for the assumed load conditions in the offices. Should the user then prefer a warmer air supply, the FCU electrical heating element can simply be switched on or inversely, should colder air be desired, the cooling coil can be implemented.

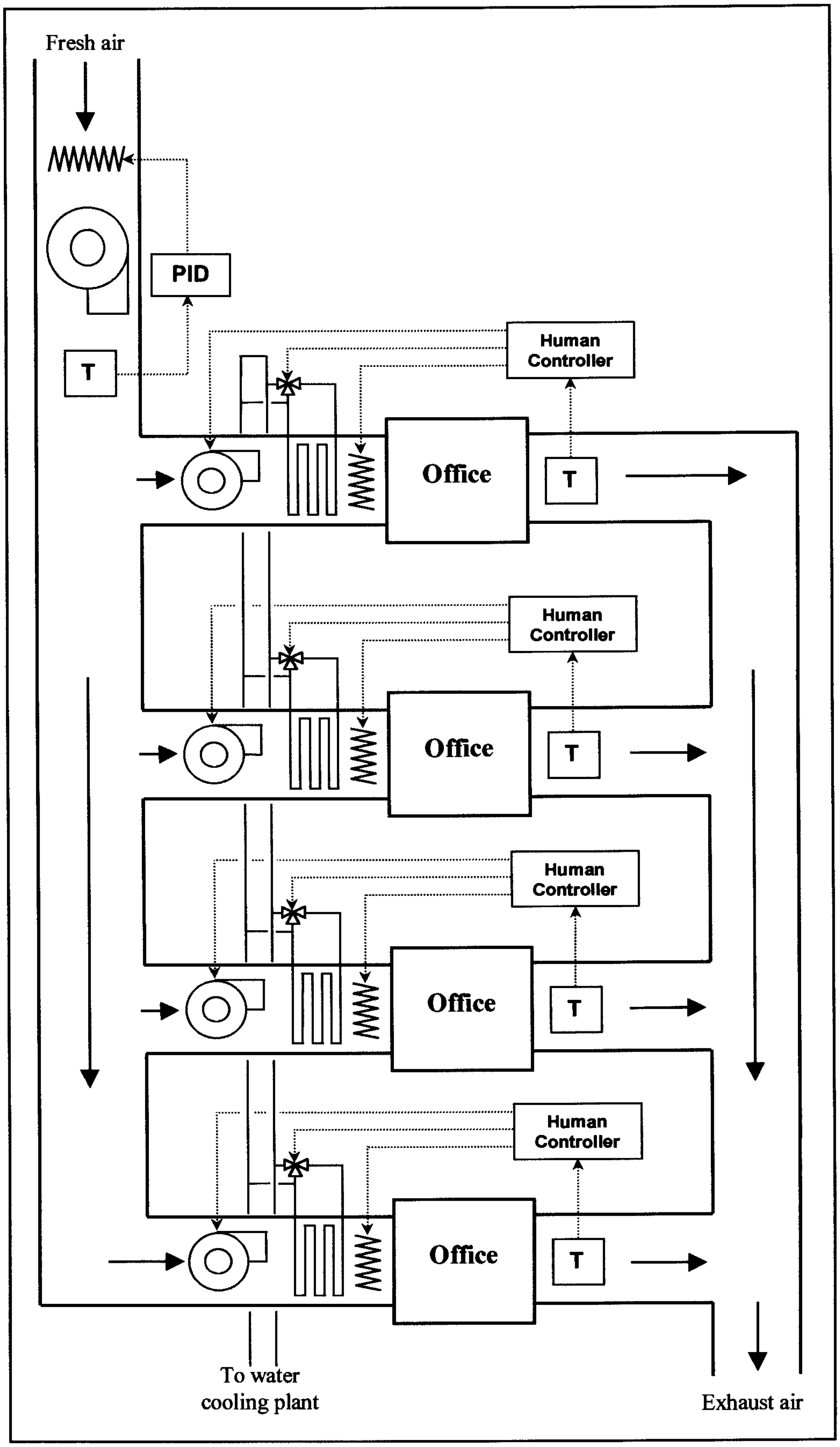

Figure 2-4 – Schematic layout of the fan coil unit air distribution and control

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Two Trane RTHA 215 water cooled chillers (rated cooling capacity of 650kW each), located on the ground level of the building, supply chilled water to the cooling coils of the AHUs and the FCUs. Two supply pumps run in series from 5am in the morning to 11pm at night for seven days a week to circulate evaporator water at a constant supply rate. The chillers are controlled with Proportional Integral Differential (PID) controllers. The input variables for these controllers are obtained from temperature sensors located directly down stream from the evaporators and the setpoint is currently set at 7°C.

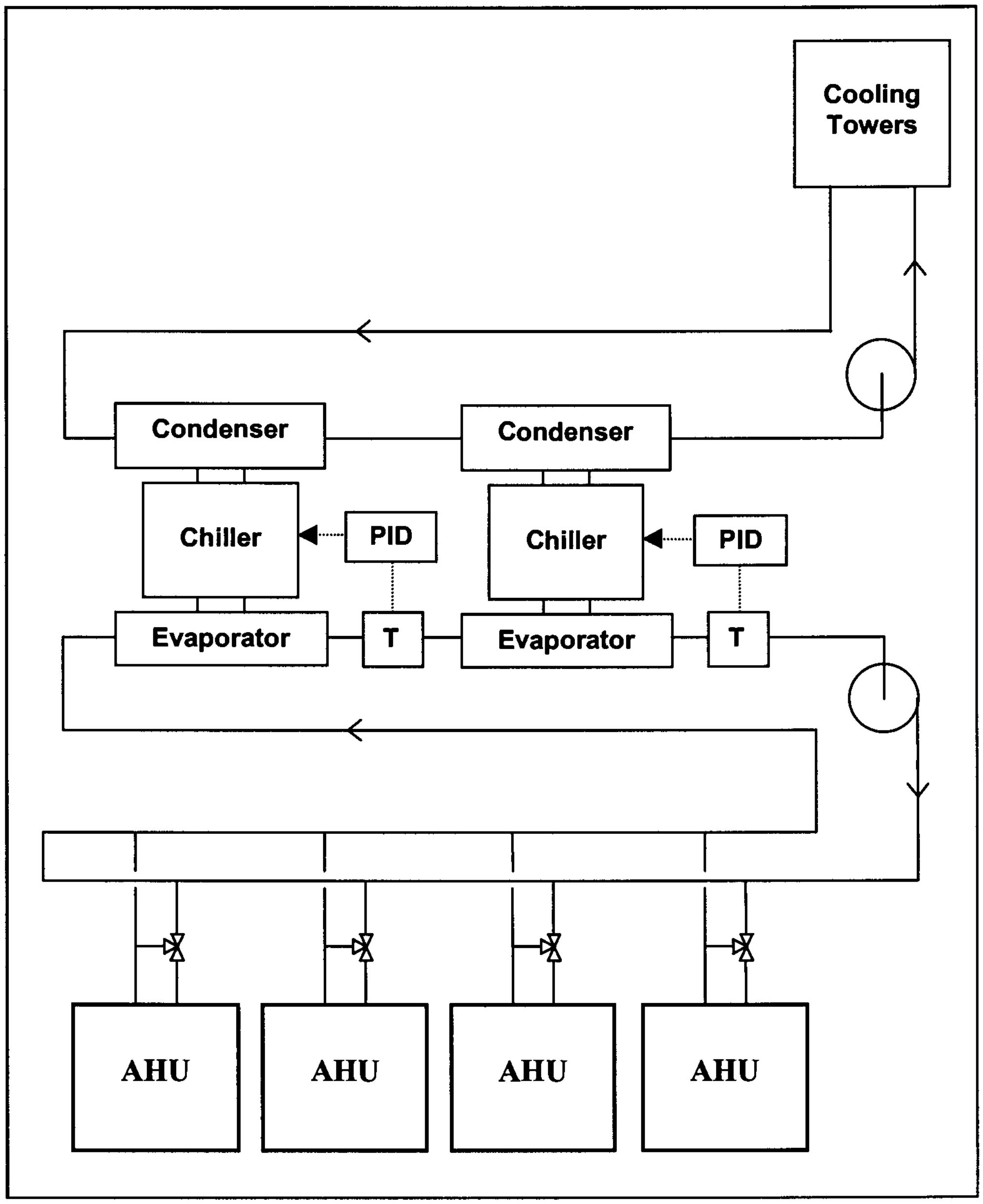

Figure 2-5 – Schematic layout of the water distribution system

The Human Science building has two Baltimore Aircoil VNT-150 cooling towers installed on the roof of the building to cool the chiller condenser water. They too are operational from 5am to 11pm. Since fans are the only electrical power consuming elements in the cooling towers, the assumption could be made that the towers have a constant electrical load throughout the day.

2.3 Comfort audit

Comfort is defined as the state in which building occupants express satisfaction with their working environment. The human body maintains its ideal operating temperature using the processes of convection, conduction, radiation and evaporation [3]. The aim of the comfort audit was to evaluate the current indoor air conditions in the Human Science building. Measurements were required to assess current conditions to the applicable standards and also for the verification of the simulation model. These indoor conditions, being satisfactory in accordance with the code, could then be used as standards for the evaluation of retrofit options [4]. The effect of fouling would also be highlighted with a better defined reference indoor climate.

2.3.1 Preliminary walkthrough audit

In the walkthrough audit problem areas regarding air conditions in the Human Science building were identified. These included areas like out of order HVAC components and neglected maintenance. Furthermore all the outstanding information about airflow, control parameters, setpoints, etc. could be collected.

One of the recognised problem areas was out of order damper control units for coil bypass control. The units were either broken or tampered with. Another problem was leakage in AHU 8. For verification of the model these inputs, along with opinions from the occupants with regards to the indoor comfort levels, were very useful.

2.3.2 Comfort measurements

Dry-bulb air temperature

Dry-bulb temperature is the single most important comfort index. According to the American Society of Heating Refrigeration and Air-Conditioning Engineers (ASHRAE), the best temperatures for humans are 24.4°C in summer and 21.7°C in winter [4]. Generally anything between 21°C and 24°C is considered to be comfortable [5]. Temperature measurements were taken in the supply and return ducts to and from each of the 14 AHUs to determine the actual indoor temperature conditions in each of the zones. In most of the zones the dry bulb temperatures were not satisfactory.

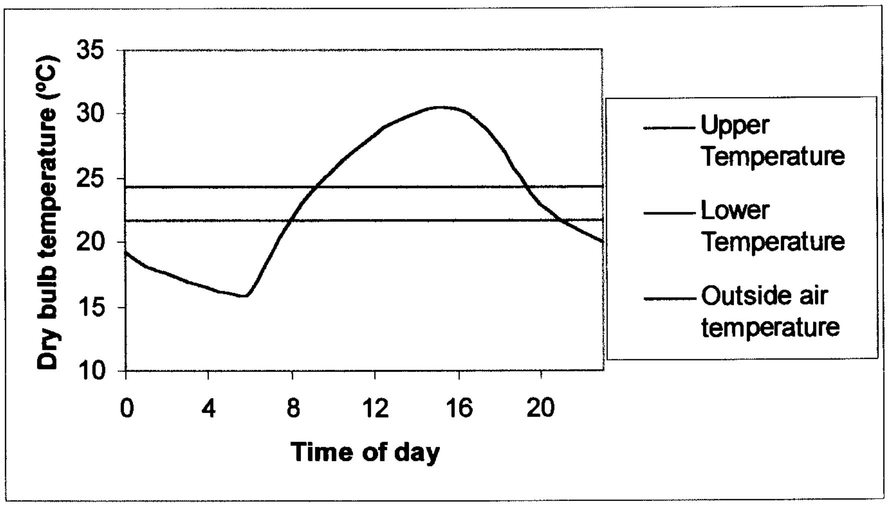

Figure 2-6 – Outside dry bulb air temperature vs. human comfort band

Relative air humidity

The best humidity conditions for humans are between 30% and 70% relative air humidity [5]. The walkthrough audit with the aid of a portable humidity probe revealed the average humidity levels in the building to be above 50% relative humidity throughout the testing period. Weather data for Pretoria and surrounding areas also showed that the expected humidity for the testing period would be 63.9% during summer and 57.6% during winter [6]. Since no humidity control is present in the building (only cooling coils that might lower the absolute humidity) the humidity measurements were neglected and assumed to be in the comfortable range.

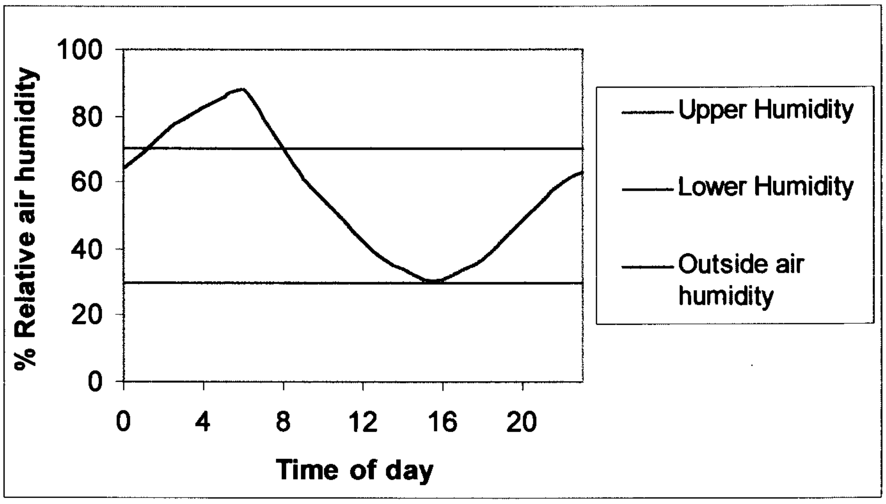

Figure 2-7 – Outside relative air humidity vs. human comfort band

Fresh air supply rate

Enough fresh air from the outside must be supplied to reduce the accumulation of body odours and other smells to a socially acceptable level. Thus more fresh air is required in areas of higher occupancy. The fresh air not only replaces old and contaminated air, but it also serves the necessary role to dilute the concentration of carbon dioxide to the maximum allowable value of 1300 parts per million [7].

To verify that the system adheres to these specifications the air mass flow rates to each of the zones were also measured. This in accordance with the air movement rate inside the zones and the mass fraction of fresh supply air to the zones was furthermore compared to standards and national building health regulations. Acceptable flow rates in buildings are between 5 $l/s/m^2$ and 8 $l/s/m^2$ [5]. The recommended fresh air supply is 10 l/s/person [7].

Air movement

Air velocity or movement plays an important role in maintaining thermal comfort by influencing the convective and evaporative heat transfer rate of a person's body [8].

Some scripts recommend that the air velocity in a room should not exceed 1.5 m/s in an air-conditioned room, unless the temperatures are above 26°C [9].

2.3.3 Interpretation of comfort audit results

AHU	Zone supply air		Zone return air		Water (l/s)
	Mass flow (kg/s)	Temperature (°C)	Mass flow (kg/s)	Temperature (°C)	
1	4.05	18.6	3.13	21.3	2.67
2	4.05	19.4	3.13	25.4	2.67
3	5.2	17.7	4.13	20.5	3.33
4	4.01	20.9	3.26	21.6	2.38
5	4.96	24.7	3.97	20.0	3.15
6	4.96	24.7	3.97	20.6	3.15
7	4.85	14.6	3.97	20.2	2.82
8	3.78	21.5	3.02	22.1	2.25
9	5.31	15.9	4.25	16.0	3.37
10	5.31	13	4.25	19.7	3.37
11	4	22.6	3.25	18.9	2.48
12	4	16.5	3.25	17.3	2.48
13	5.1	-	0	-	14.49
14	5.1	-	0	-	14.49

Table 2-2 – Average supply and return air temperatures, and air and water flow rates

From table 2-2 it is clear that many indoor temperatures lie outside the desired comfort range. The poor control may be attributed to load concentrations, poor control sensor placement and in some cases malfunctioning of equipment. This implies that before any retrofit options may be considered, the system model has to be upgraded to adhere to the required standards. Should any improvements to the system be evaluated afterwards, the basis for evaluation will then be the upgraded model.

The air supply of the building is set to supply a constant volume mixture of 20% fresh air and 80% return air to the zones. This means that during maximum cooling load conditions (full capacity of occupants) in the lecture halls the fresh air supply will be 10.2 l/s/person. This is acceptable [7].

In the Human Science building the air velocity in the lecture halls never exceeds 0.8 m/s which also falls within the required specification.

To determine the rate at which the air in a zone is replaced, which is an indication of the responsiveness of the HVAC system to changes in loads, the volume flow rate of air supplied is divided by the air-conditioned volume in the zone. ACH values of 10 to 15 are generally assumed to be acceptable [9].

$$ACH = 3600_{\frac{Seconds}{Hour}} \frac{Air \quad Supply}{Zone \quad Volume} = 3600 \frac{\dot{m}}{\rho V}$$

The following table shows the rate of air replacement.

AHU	Volume (m^3)	Air supply (m^3/s)	ACH (1/Hour)
1	976.8	4.05	14.9
2	976.8	4.05	14.9
3	1097.3	5.2	17.1
4	1268.9	4.01	11.4
5	1048	4.96	17.0
6	1048	4.96	17.0
7	1543.6	4.85	11.3
9	1048.45	5.31	18.2
10	1048.45	5.31	18.2
11	730.35	4	19.7
12	730.35	4	19.7

Table 2-3 – Zone air changes

The average rate at which the air in the zones are replaced is 16.3 changes per hour. This is somewhat above the recommended values, but satisfactory to ensure an adequate response should sudden cooling or heating load shifts occur [9].

2.3.4 Conclusion

The HVAC system is capable of supplying sufficient air at a low enough temperature to all the zones. The setpoints however are in many cases too low. The constant volume fresh air supply was satisfactory in all the zones according to standards, and the air replacement rates were high enough to ensure a good air quality in all the zones. Therefore, to conclude, after adjusting and upgrading the control equipment to full working order the comfort levels in die Human Science building should be acceptable.

2.4 Energy audit

The aim of the energy audit was to perform a total end-user breakdown of the energy consumption of the building and to identify certain areas where energy could be saved [1]. The data was also necessary to verify the simulation models in order to perform accurate retrofits and accurately predict the effects of fouling.

The following measurements were taken:

- Voltages (V)
- Currents (A)
- Energy Consumption (kV)
- Energy Demand (kVA)
- Power Factor

For the purpose of this study the electrical energy distribution of the building was subdivided into the following categories:

- Lighting system
- HVAC system
- Other diverse equipment

From there the energy consumption of the HVAC system was further subdivided to determine the energy consumption of the chillers, pumps, fans, cooling towers and heaters.

2.4.1 Walkthrough audit

Although the HVAC system was the primary concern, other diverse equipment such as lifts, computers, printers, photocopiers, lights, etc. were also accounted for to accurately determine the effect any retrofit options might have on the overall energy consumption [10].

During the walkthrough audit it was immediately noted that the lighting system in the building would play a major role in the breakdown energy consumption. The whole building was fitted with 40,70 and 100 watt fluorescent lights that were rarely switched off. Fluorescent bulbs dissipate heat energy with a 55% and 45% convective and radiant heat transfer rate respectively [11].

On average a single desktop computer induces a load of 250 watt in the building. Of this load 78% is convective heat and 22% radiant [12]. The Human Science building has approximately 310 desktop computers installed in the offices that are switched on for an assumed 90% of the time. This accounts for about 70 kW of continuous electrical power consumption and roughly the same amount of internal heat gain.

The elevators are operated with 6 x 10 kW motors. It can be assumed that elevators demand electrical power at 75% of its rated capacity for 85% of the time [13]. This constitutes a significant portion of the diverse equipment power consumption.

2.4.2 Energy measurements

Logging equipment was installed on the electrical supply lines to the air handling units and the other power consuming equipment. The figure below illustrates the

HVAC Component	Consumption per day (kWh)
Chillers	1162
Cooling Towers	509
Pumps	1090
Fans	1017
Heaters	3489
Total	7268

Table 2-4 – Average daily power consumption of the HVAC components

2.4.3 Interpretation of measurements

As can be seen from the figures, the heating ventilating and air-conditioning equipment play a major role in the overall power consumption of the building (54%). What is even more significant is the fact that the heating elements consume the largest amount of energy allocated to the HVAC system (almost 26% of the total amount of electrical power consumed by the entire building). These results confirm that significant savings are possible with more efficient control to prevent unnecessary re-heating. It also indicates that fouling of the coils and chillers might possibly prevent the re-heating altogether.

At this point it is evident that with an improved control strategy the heaters will be used less often and in turn less cooling will be needed from the chillers.

2.5 Conclusion

Due to the age of the building (built in 1972) and poor maintenance over a prolonged period of time the signs of extreme neglect are clear. Many of the HVAC components are barely working. This means that there is no sense in simply trying to save energy on the current system. The system has to be repaired to meet the necessary comfort requirements before the effect of retrofitting may be investigated. Only then can the potential for energy saving be evaluated.

The energy audit revealed that a large amount of electrical energy may be saved by simply preventing re-heating. This and the fact that the HVAC system is already fully installed in the building suggest that only control strategy improvement should be investigated. This then allows for retrofitting without excessive expenditure on structural modifications.

2.6 References

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CHAPTER 3 - Simulation and verification

Before the simulation model may be used for any application, the model had to be set up and verified to ensure an accurate representation of the actual building. The Human Science building was modelled with 13 building zones and each of the HVAC components were customised to be an exact mathematical model of the components used in the building. After a few modifications to the simulation model the simulation output was sufficiently close to the measured values to assume that future simulations will produce accurate results when new retrofits and fouling predictions are considered.

3.1 Introduction

The whole of the HVAC system with its control strategies were modelled and simulated on a computer with the software package: QUICK Control* [1]. With the package the best possible recommendations to save energy in the building and detailed predictions for maintenance purposes could be found with a series of comprehensive integrated simulations. The model was verified as being correct by comparing simulated outputs to the measured results as discussed in the previous chapter.

Zone number	Air handling unit represented
1	4
2	7
3	1
4	2
5	5
6	6
7	1
8	11 & 12
9	9 & 10
10	8
11	13
12	14

Table 3-1 – Zone numbering

Since air handling units 9 and 10 were so similar in both structural layout, control strategy and load conditions, it was decided to model these two units together as one building zone. By following this method of modelling, the computer simulations became much more stable and the simulation time decreased dramatically [1]. The same reasoning was followed with AHU 11 and 12. In the end the HVAC system was

* Software developed by TEMM International (Pty) Ltd and used with special permission.

modelled with 12 zones in stead of 14. The office floors were divided into a north and a south side, and represented by zone 11 and 12. The numbering is shown in table 3-1.

All the structural and climatic data for the zones were entered into the simulation database. This included: the building structure, wall composition, weather data, load conditions, etc. A passive load calculation simulation was then executed. The results (heating and cooling loads) of this preliminary evaluation were then used as input parameters and starting values for the solutions of an integrated dynamic simulation of the HVAC system and its control.

3.2 Outdoor conditions

The outdoor conditions were measured to ensure that the model would have the correct external input when running the simulations. Outside temperatures as well as the relative outside humidities were recorded. By supplying the simulation package with the correct humidity information, the correct moisture levels could be simulated with cooling of the air through the coils.

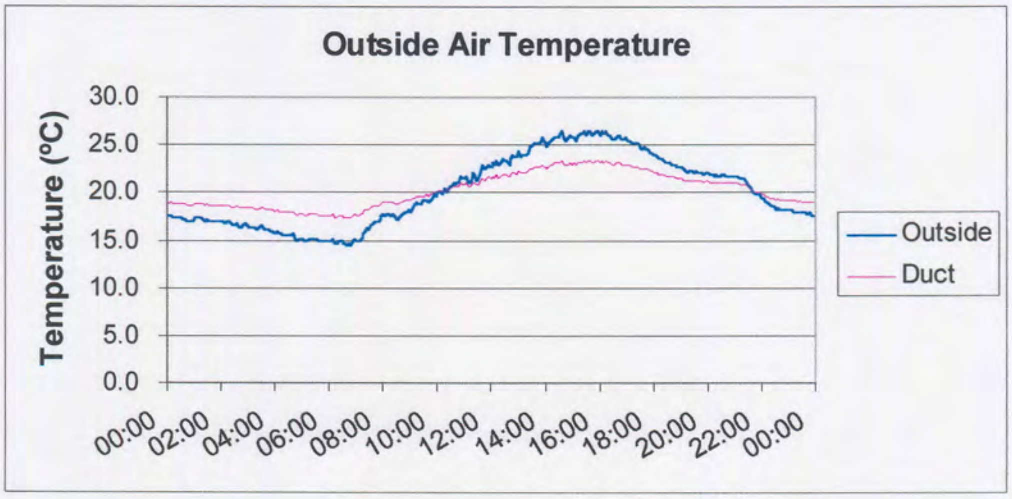

Figure 3-1 – Outside air temperature damping due to thermal mass of intake shafts

The outside air to the HVAC system is drawn into the AHUs through long shafts running down the centre of the building. These shafts and the walls that support them have a huge bulk mass that acts as a temperature variant damper. (It cools warm air and warms cool air like a regenerator.) For this reason the temperature and humidity in the intake directly prior to mixing with return air were also recorded.

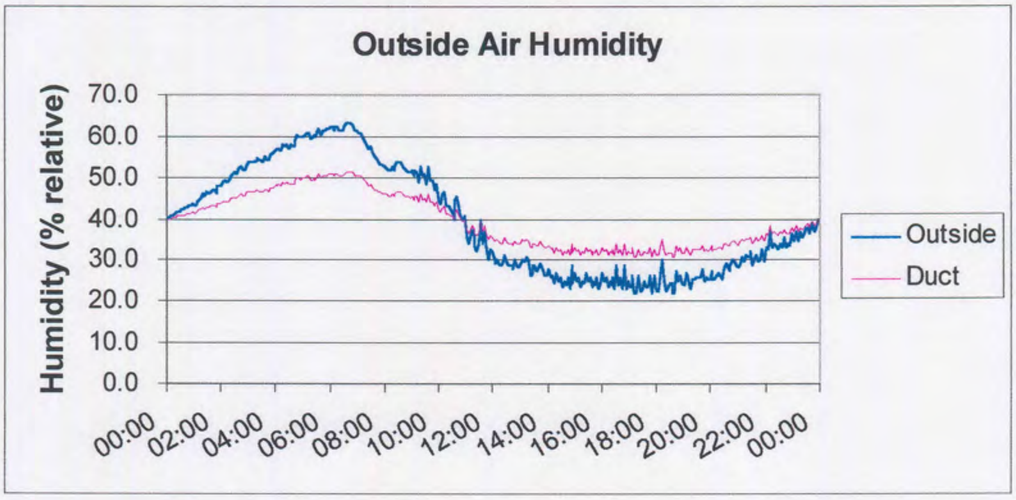

Figure 3-2 – Outside air humidity damping due to thermal mass of intake shafts

As can be seen from the humidity measurements (figure 3-2), the humidity level of the air entering the air handling units falls within the required comfort band of 30% to 70% relative humidity. This is common for Pretoria and thus the humidity considerations were omitted from the study [2].

3.3 Characterisation of simulation model

Characterising the various building and HVAC system components requires a large number of inputs. These consist of building structural data, internal load, outdoor climatic data, as well as all the HVAC component data and control parameters. The required data was obtained from drawings, technical data sheets and measurements.

Measurements required for characterising and verification purposes were taken over a period of two weeks. To correctly simulate the damping effect of the intake shafts (see previous paragraph), a thirteenth zone was added to the simulation model. Climatic (outside) air was then first drawn through this zone before being divided into the correct mass fractions for the rest of the system.

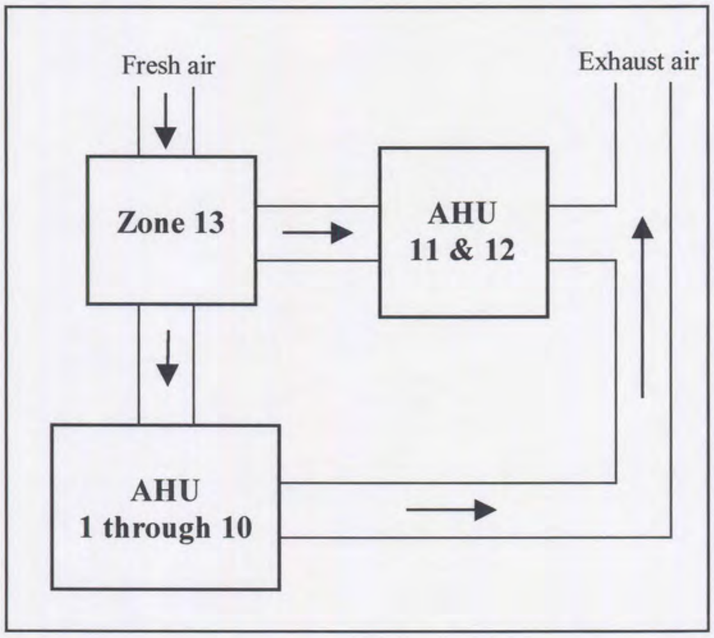

Figure 3-3 – Schematic layout of air-flow through the building zones

Zones 1 through 10 were allocated to the lecture halls while zones 11 and 12 represented the north and south offices respectively. The office walls and floors inside each zone were treated as partitions adding to the thermal mass of the zone and thus establishing the phase lag and temperature swing dampening effect on the envelope loads. Building structure data for each zone was obtained from building drawings and the walkthrough inspections. All of these were read into the simulation database.

The input data needed for the HVAC system chiller, pump and fan models was obtained from suppliers' performance data sheets. Due to the age of the system (installed in 1972) original data sheets for the coils, heaters and cooling towers were no longer available and were mainly substituted with data sheets from similar equipment, measurements and assumptions.

Control strategy inputs included operating times and control parameters for most system components and were obtained from the HVAC system operating manuals. Parameters not available from the manuals were again obtained from measurements.

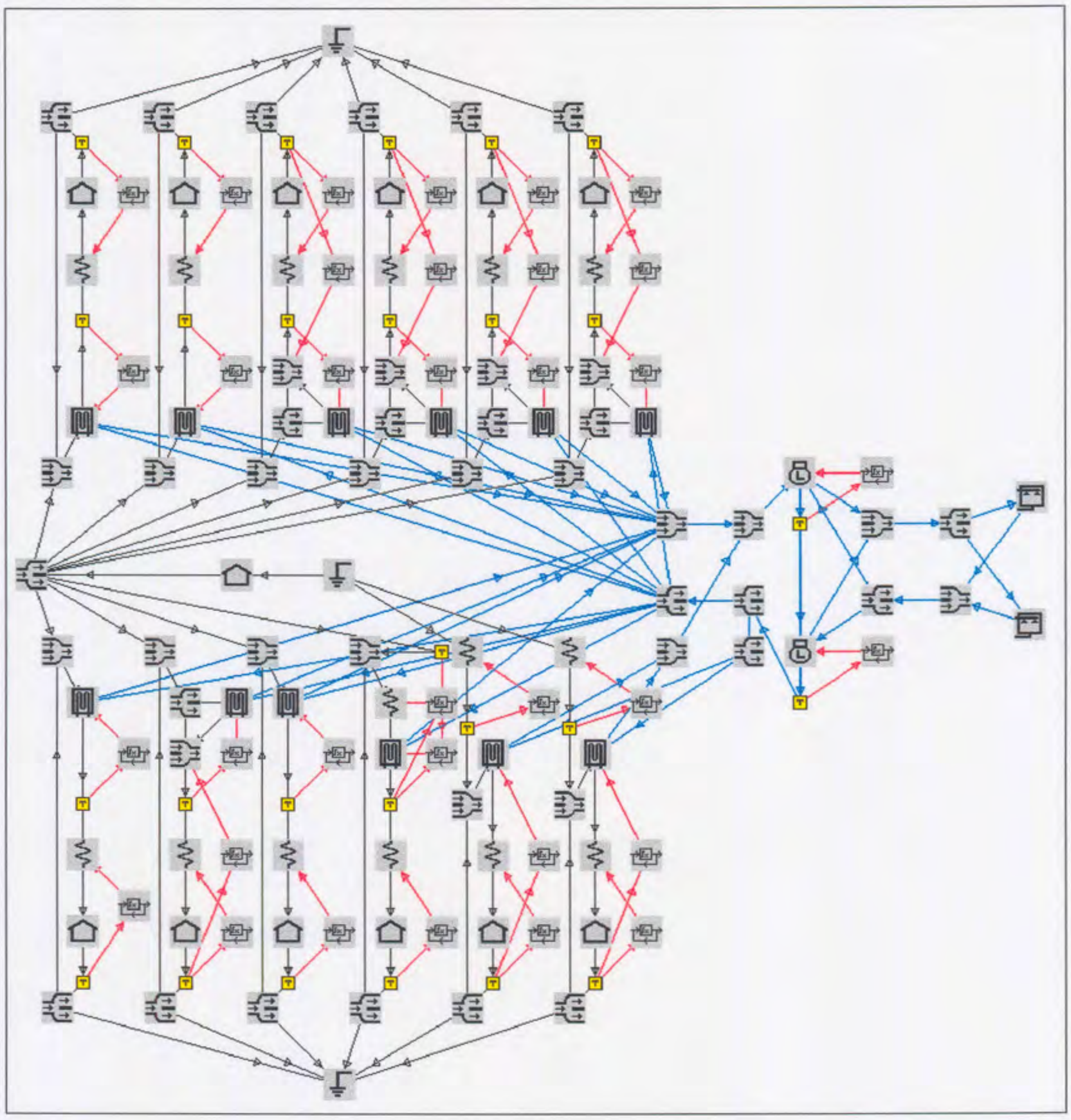

Figure 3-4 – Simulation model used for verification of the current building HVAC system

The final simulation model used for verification of the system is shown above. In this model the fans and pumps were omitted since no Variable Air Volume (VAV) system is present. After some extensive measurements it was found that the pressure loss over the flow systems stayed constant irrespective of changes due to controller outputs. To make the simulation model less bulky the assumption was made that both the fans and the pumps would consume a constant amount of electrical energy throughout the year [3],[4]. This assumption was verified with measurements.

The building zones were constructed and in every time step of the simulation the relevant heating and cooling loads were calculated. Furthermore each of the components were characterised by means of specific regression coefficients obtained from multidimensional curve fits.

3.4 The building zone model

Before the dynamic effect of the HVAC components may be calculated, the response of heating and cooling loads in each of the zones have to be found. The software tool uses an electrical analogy to model the heat transfer processes in the building for the accurate prediction of the thermal performance of the building zones [5].

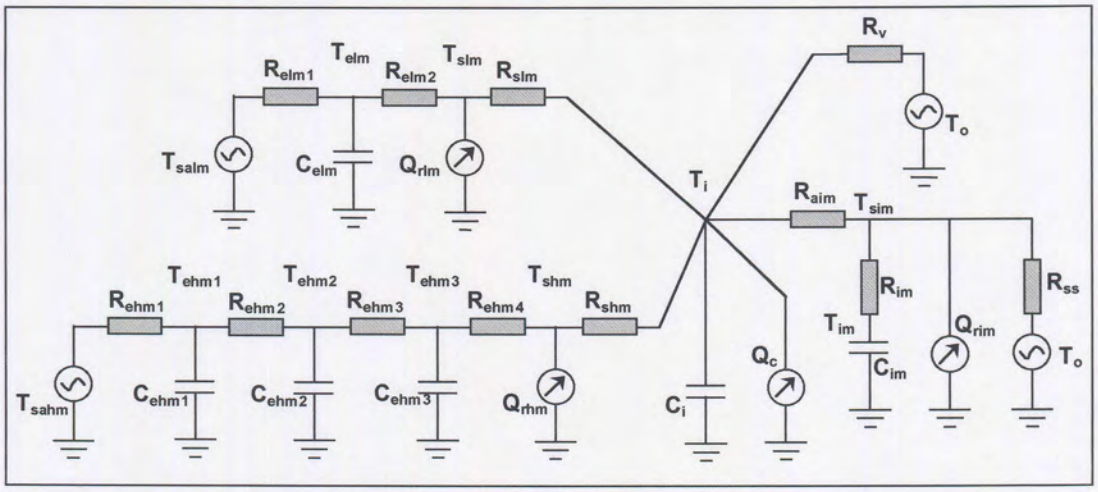

Figure 3-5 – Zone model electrical analogy

The model makes use of a six node electrical circuit to simulate the different heat flow paths. In figure 3-5 the air node is treated as a separate node for which the capacitance may be calculated by:

$$C_i = V\rho c_p$$

with:

V = volume of the building zone

ρ = the density of the air

c_p = the specific heat capacity of the air

Convective heat gain is simply added to the air node and is represented by Q_c. The resistance due to ventilation, infiltration and environmental control are all modelled together with:

$$R_v = \frac{1}{V\rho c_p (ACH)}$$

where ACH is the air changes per second. The internal masses are combined and represented by a single capacitor (C_{im}), while the heat gain through the floor is treated separately with the steady state resistance R_{ss}.

The remaining two circuits model two distinct structural heat flow paths, namely a low-mass path (a single node for structures like glass, wood, etc.) and a high-mass path (a triple node for heavy structures). Through the use of an analytical optimisation technique the thermal resistance and capacitance for each of the paths can be found [6]. Radiative heat gain (Q_r) is weighed according to surface area and act directly on the surface. The governing equations for each circuit can then be derived by adding the currents together at each node.

In total nine nodes remain to be solved at every time step through the simulation process and the solutions are determined by using the relevant psychometric relationships to deal with the moist air properties.

3.5 The HVAC component database

To perform the integrated simulation all the data of the HVAC components as well as their control parameters were needed. The software package simulates the dynamic response of a component by means of characteristic functions. By evaluating the component data sheets the necessary regression coefficients for the functions were found.

For this project it was necessary to develop models for the chillers, cooling towers, cooling coils, fans and pumps. Since heaters usually work at 100% effectiveness and almost no (negligible) airflow pressure drop, the heaters did not need a specific component model. Sensible heat gains were simply added to the air stream as were specified by the controllers.

The component characteristics that were read into the database were found according to the following mathematical regressions:

3.5.1 Chillers

The liquid cooled chillers are characterised by developing regression equations that describe both the cooling capacity and the power consumption. The coefficients in the equations were then read into the component database of QUICK Control*.

The explicit equations are as follows:

$$Q_e = f(a_0 + a_1 m_{l(condenser)} + a_2 m_{l(evaporator)} + a_3 t_{li(condenser)} + a_4 t_{li(evaporator)})$$

$$Pwr = f(b_0 + b_1 m_{l(condenser)} + b_2 m_{l(evaporator)} + b_3 t_{li(condenser)} + b_4 t_{li(evaporator)})$$

$$t_{le(condenser)} = t_{li(condenser)} + \frac{Q_e + Pwr}{m_{l(condenser)} c_{pl}}$$

$$t_{le(evaporator)} = t_{li(evaporator)} - \frac{Q_e}{m_{l(evaporator)} c_{pl}}$$

For determining the drop in pressure over the condenser and evaporator of the chiller, the following equations were solved:

$$\Delta P_{l(condenser)} = c_0 + c_1 m_{l(condenser)} + c_2 m^2_{l(condenser)}$$

$$\Delta P_{l(evaporator)} = d_0 + d_1 m_{l(evaporator)} + d_2 m^2_{l(evaporator)}$$

* Software developed by TEMM International (Pty) Ltd and used with special permission.

With:

Q_e = cooling capacity (kW)

Pwr = total power consumed by the compressor (kW)

$m_{l(condenser)}$ = mass flow rate of water through the condenser (kg/s)

$m_{l(evaporator)}$ = mass flow rate of water through the evaporator (kg/s)

$t_{li(condenser)}$ = temperature of water entering the condenser (kg/s)

$t_{li(evaporator)}$ = temperature of water entering the evaporator (kg/s)

c_{pl} = specific heat capacity of the liquid (kJ/kg.K)

a_n = regression coefficients for cooling capacity

b_n = regression coefficients for power consumption

c_n = regression coefficients for water pressure drop over the condenser

d_n = regression coefficients for water pressure drop over the evaporator

By performing a multidimensional curve fit with the aid of a spreadsheet program all the regression coefficients were found (see Appendix B-4).

3.5.2 Cooling towers

To characterise the cooling towers two sets of correlation coefficients were necessary. One set was for the relationship between thermal conductance, water mass flow and water temperature difference. The other was for the correlation between pressure rise and water mass flow. The regression coefficients were again obtained from a curve fit over data from the performance data sheets for the specific cooling towers.

The equations are as follows:

$$Q = UA\Delta h_{average} = m_a\left(h_{ae} - h_{ai}\right) = m_l c_{pl}\left(t_{li} - t_{le}\right)$$

with

$UA = a_0 + a_1 m_l + a_2 t_{li} + a_3 m_l^2 + a_4 t_{li}^2 + a_5 m_l t_{li}$

and $\Delta h_{average} = \left(\dfrac{(h_{si} - h_{ai}) - (h_{se} - h_{ae})}{\ln \dfrac{(h_{si} - h_{ai})}{(h_{se} - h_{ae})}} \right)$

And for preassure rise: $dP_l = -(b_0 + b_1 m_l + b_2 m_l^2 + ... + b_k m_l^k)$

The power consumption of the cooling tower is simply expressed by $Pwr_{total} = Pwr_{fans} + Pwr_{pumps}$.

For all the above equations the enthalpy values may be found by means of the following empirical equations and by solving all the necessary equations simultaneously.

$$h_{se} = 10^3 \begin{pmatrix} 16.66326 + 4.701617 t_{li} - 0.11237 t_{li}^2 + 0.004991 t_{li}^3 - 0.17197 p_{barometric} \\ - 0.01364 p_{barometric} t_{li} + 0.000493 p_{barometric} t_{li}^2 - 2.9 \times 10^{-5} p_{barometric} t_{li}^3 \end{pmatrix}$$

$$h_{si} = 10^3 \begin{pmatrix} 16.66326 + 4.701617 t_{le} - 0.11237 t_{le}^2 + 0.004991 t_{le}^3 - 0.17197 p_{barometric} \\ - 0.01364 p_{barometric} t_{le} + 0.000493 p_{barometric} t_{le}^2 - 2.9 \times 10^{-5} p_{barometric} t_{le}^3 \end{pmatrix}$$

With:

Q = cooling capacity (kW)

UA = thermal conductance (kW/°C)

h_{si} = specific enthalpy of saturated air entering at t_{le} (kJ/kg)

h_{se} = specific enthalpy of saturated air entering at t_{li} (kJ/kg)

h_{ai} = specific enthalpy of air entering at inlet (kJ/kg)

h_{ae} = specific enthalpy of air leaving at outlet (kJ/kg)

Pwr = electrical power consumed by the fans and pumps (kW)

dP_l = static pressure rise (Pa)

$p_{barometric}$ = barometric pressure of air (Pa)

m_a = mass flow rate of air (kg/s)

m_l = mass flow rate of water (kg/s)

c_{pl} = specific heat capacity of water (kJ/kg.K)

t_{li} = temperature of water into the cooling tower (°C)

t_{le} = temperature of water leaving the cooling tower (°C)

a_n = correlation coefficients for UA versus mass flow and temperature

b_n = correlation coefficients for pressure rise versus mass flow of water

3.5.3 Fans

Fans were characterised by developing regression equations that describe the non-dimensional pressure and efficiency coefficients against non-dimensional flow coefficients. By again performing a polynomial curve fit over the specific input variables for the selected fan the regression coefficients could be found.

The non-dimensional coefficients derived by employing the Buckingham Pi theorem are found as follows [7]:

$$K_f = \frac{m}{\rho n D^3}$$

$$K_h = \frac{P_{total}}{\rho n^2 D^2}$$

These two coefficients can be used to caracterise any specific fan by simply expressing the relationship between the coefficients as a polynomial function. The efficiency can be expressed in much the same way:

$$K_h = a_0 + a_1 K_f + a_2 K_f^{\,2} + ... + a_k K_f^{\,k}$$

$$\eta_{fan} = b_0 + b_1 K_f + b_2 K_f^{\,2} + ... + b_k K_f^{\,k}$$

In most HVAC applications axial fans are used to draw air from large volumes and then discharge the air back to a large volume again. A typical practical case is that of a condenser fan in an air-cooled condenser. For axial fans it was therefore assumed that the dynamic pressure difference across the fans will be negligible.

For centrifugal fans however the dynamic pressure difference is important. Air is ususally draw from large volumes (like the Humans Science building lecture halls) with negligible flow speed and discharged into ducts. Inside the air ducts the flow speed is much higher. To mathematically model a centrifugal fan the following equation may be used to express the total pressure:

$$P_{total} = dP_a + \frac{m_a^2}{2\rho_a A_e^2} = K_h \rho_a n^2 D^2$$

The total heat gain to the air passing through the fan will be equal to the total shaft power of the fan if the motor is mounted outside the air stream. If the motor is mounted in the air stream the total power supplied to the motor must be added to this. The motor efficiecy can also be written as a function of the flow coefficient K_f.

With:

K_h = dimensionless pressure head coefficient

K_f = dimensionless flow coefficient

η_{fan} = fan efficiency (%)

n = rotational speed of the fan (rpm)

P = pressure rise (Pa)

ρ = air density (kg/m³)

D = fan rotor diameter (m)

m = mass flow rate of the air (kg/s)

a_n = correlation coefficient of pressure versus flow

b_n = correlation coefficients for efficiency versus flow

The coefficients a and b are then calculated for the operating range of the fan, using data from the fan performance data sheets or the fan curves. The values are plotted against each other and the governing equations are determined using a trend line.

3.5.4 Pumps

The derivation for pumps is the same as for fans except for the following:

- No psychometric property relations are needed to calculate the fluid density. The liquid was assumed to be incompressible and the density was taken throughout as $\rho_l = 1000\,\text{kg/m}^3$. Similarly the specific heat coefficient at constant pressure was assumed to be $c_{pl} = 4187$ J/kgK [8],[9].
- The drive motor may not be situated within the liquid flow. Therefore no extra heat gain was to the water stream due to motor inefficiency [9].
- HVAC pumps usually have pipes connected to both the inlet and discharge ports. The dynamic pressure difference over the pump is therefore negligible [9],[10].

Thus solve the following explicit equations:

$$K_f = \frac{m_l}{\rho_l n D^3}$$

$$K_h = a_0 + a_1 K_f + a_2 K_f{}^2 + \ldots + a_k K_f{}^k$$

$$dP_l = K_h \rho_l n^2 D^2$$

$$\eta_{pump} = b_0 + b_1 K_f + b_2 K_f{}^2 + \ldots + b_k K_f{}^k$$

$$Q_l = \frac{(1 - \eta_{pump}) m_l dP_l}{\rho_l \eta_{pump}}$$

$$Pwr = \frac{k m_l dP_l}{\rho_l \eta_{pump} \eta_{motor}}$$

With:

K_h = dimensionless pressure head coefficient

K_f = dimensionless flow coefficient

η_{pump} = pump efficiency (%)

n = rotational speed of the pump (rpm)

P = static pressure rise (Pa)

ρ_l = water density (kg/m³)

D = pump rotor diameter (m)

k = number of pumps in cascade

m_l = mass flow rate of the water (kg/s)

Q_l = rate of heat gain to the water (kW)

Pwr = power required to drive the pumps (kW)

a_n = correlation coefficient of pressure versus flow

b_n = correlation coefficients for efficiency versus flow

3.5.5 Electrical heaters

Modelling of the heaters followed directly from basic principles. Since no pressure was lost due to flow over the heating elements the air pressure stayed constant. No moisture was added or subtracted from the air stream, so the humidity ratio also stayed constant. In general electrical heating elements are also 100% effective.

The only variant property was the increase in specific enthalpy:

$$h_{ae} = h_{ai} + \frac{Q_a}{m_a}$$

Where:

h_{ae} = specific enthalpy of air leaving the heater (kJ/kg)

h_{ai} = specific enthalpy of air entering the heater (kJ/kg)

Q_a = heater capacity (kW)

m_a = mass flow of air through the heater (kg/s)

3.5.6 Cooling coils

Solving the flow of air and liquid through a cooling coil is a much more comprehensive exercise than the other components. The psychometric properties of the air passing through the coil have to be taken into account. This means that two scenarios exist: a wet coil or a dry coil. If a coil is "dry" the air is simply cooled with the humidity ratio staying constant. However, if the coil is wet, moisture is extracted from the air stream and condensed on the coil fins.

This last scenario requires a number of simultaneous equations to be solved through an iterative process.

According to supplier data sheets the regression coefficients for the convection coefficient could be found by fitting the following polynomial equation over the design specifications:

$$h_0 = a_0 + a_1 V_a + a_2 V_a^2 + \ldots + a_k V_a^k$$

If the convection coefficient was not specified by the suppliers, it was determined through the so-called Dittus-Boetler equation [8]:

$$h_0 = 0.023 \frac{k_l}{d_0} \left[\frac{\rho_l V_l d_0}{\mu_l} \right]^{0.8} \left[\frac{\mu_l c_{pl}}{k_l} \right]^{0.4}$$

The air and water pressure drops were characterised by:

$$dP_a = -(b_0 + b_1 V_a + b_2 V_a^2 + \ldots + b_k V_a^k)$$

$$dP_l = -(c_0 + c_1 m_l + c_2 m_l^2 + \ldots + c_k m_l^k)$$

The next step was to determine whether the coil would be wet or dry. This was done by calculating the dew point temperature of the air at its specific entering state and then comparing it to the supply temperature of the chilled water. The dew point temperature is calculated through an empirical formula:

$$t_{dp} = 6.54 + 14.526\ln(P_w) + 0.7389(\ln(P_w))^2 + 0.09486(\ln(P_w))^3 + 0.4569P_w^{0.1984}$$

If this temperature is below the water entering temperature the coil will be dry (no change in humidity ratio of the air). Then:

$$t_{ae} = \frac{\left[e^{h_0A_0\left(\frac{1}{m_ac_{pa}}-\frac{1}{m_lc_{pl}}\right)}-1\right]t_{li} - \left[\frac{m_ac_{pa}}{m_lc_{pl}}-1\right]t_{ai}}{e^{h_0A_0\left(\frac{1}{m_ac_{pa}}-\frac{1}{m_lc_{pl}}\right)} - \frac{m_ac_{pa}}{m_lc_{pl}}}$$

$$t_{le} = t_{li} - \frac{m_ac_{pa}}{m_lc_{pl}}(t_{ae} - t_{ai})$$

If t_{dp} is higher than the water entering temperature the coil will be wet and the following equations will have to be solved simultaneously:

$$t_{ae} = t_{ai} - \frac{w_{ai} - w_{ae}}{w_{ai} - w_{adp}}(t_{ai} - t_{adp})$$

$$S = \frac{h_{ad} - h_{ae}}{h_{ai} - h_{ae}}$$

$$h_{ae} = h_{ai} + \frac{h_0A_0}{m_aS}\frac{(t_{le} - t_{ai}) - (t_{li} - t_{ae})}{\ln\left(\frac{(t_{le} - t_{ai})}{(t_{li} - t_{ae})}\right)}$$

$$t_{le} = t_{li} - \frac{m_a}{m_lc_{pl}}(h_{ae} - h_{ai})$$

$$w_{ae} = \frac{h_{ae} - t_{ae}}{2501 + 1.805t_{ae}}$$

$$h_{ad} = t_{ai} + w_{ae}(2501 + 1.805t_{ai})$$

Some of these equations are empirical and only applicable for air. The variables and constants are:

h_0 = convection coefficient (W/m^{2}K)

V_a = air stream velocity (m/s)

k_l = conduction coefficient (W/mK)

d_0 = diameter (m)

ρ_l = density (kg/m^3)

μ_l = viscosity (kgm/s^2)

c_p = specific heat capacity (J/kgK)

dP_a = air pressure drop through the coil (Pa)

dP_l = water pressure drop through the coil (Pa)

P_w = air vapour pressure

t_{dp} = dew point temperature of the air (°C)

t_{ae} = air temperature of air leaving the coil (°C)

t_{ai} = air temperature of air entering the coil (°C)

t_{le} = water temperature of water leaving the coil (°C)

t_{li} = water temperature of water entering the coil (°C)

h = enthalpy of the air with same the subscripts as the temperatures (kJ/kg)

w = humidity ratio of the air with same the subscripts as the temperatures (kg_{vapour}/kg_{air})

S = sensible heat ratio

A_0 = total heat transfer area (m^2)

m = mass flow rate (kg/s)

a_n = correlation coefficient for convection coefficient versus air stream velocity

b_n = correlation coefficients for air pressure drop versus air stream velocity

c_n = correlation coefficients for water pressure drop versus water mass flow rate

3.6 Verification study

The integrated simulation tool and the simulation model of the building were verified by comparing predicted indoor temperatures and HVAC component energy consumption with measured values for a specific day.

The results had to meet the following criteria:

- The average zone temperature error had to be within 1 °C
- The electrical power consumed by the AHUs had to be within 10%
- The electrical power consumed by the chillers had to be within 10%

A summary of the results is as follows:

Simulated temperatures vs. Measured temperatures				
Zone	**Average error (°C)**	**Max error (°C)**	**Time steps within 2°C (%)**	**Time steps within 1°C (%)**
1	0.56	2.88	83.3	81.4
2	0.43	1.44	99.4	84.2
3	0.12	2.64	98.1	83.3
4	0.59	1.95	100	84.7
5	0.21	1.48	100	100
6	0.27	1.84	100	85.6
7	0.57	4.61	98.9	81.7
8	0.06	2.17	98.6	95.8
9	0.82	2.95	88.9	83.6
10	0.52	1.02	87.3	82.2
11	N/A	N/A	N/A	N/A
12	N/A	N/A	N/A	N/A
13	0.02	0.25	100	100

Table 3-2 – Summary of comparison of measured and simulated temperatures

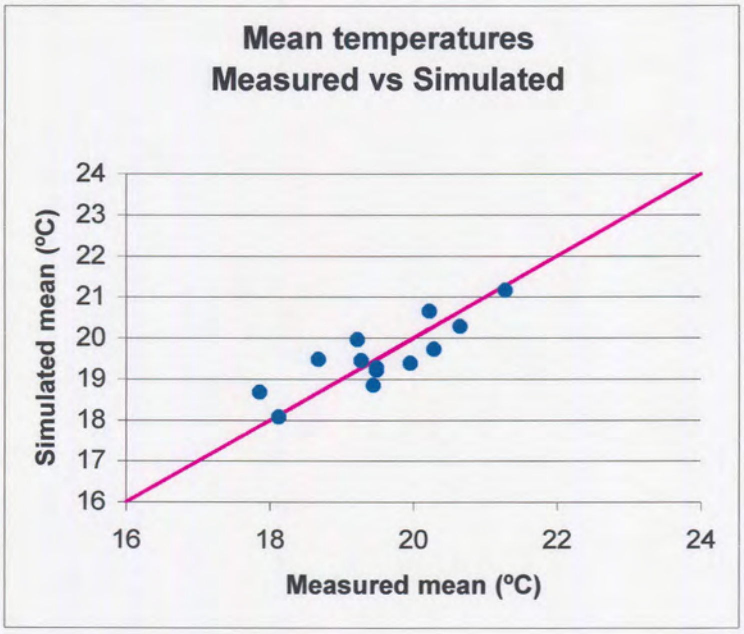

Figure 3-6 – Comparison of simulated and measured mean temperatures

The maximum errors do not influence the credibility of the computer model since it occurred due to uneven spikes (start up peaks) and large temperature differences in the early hours of the morning. During that time the HVAC system is not operational. The mean temperatures however were all within the requirements for the project.

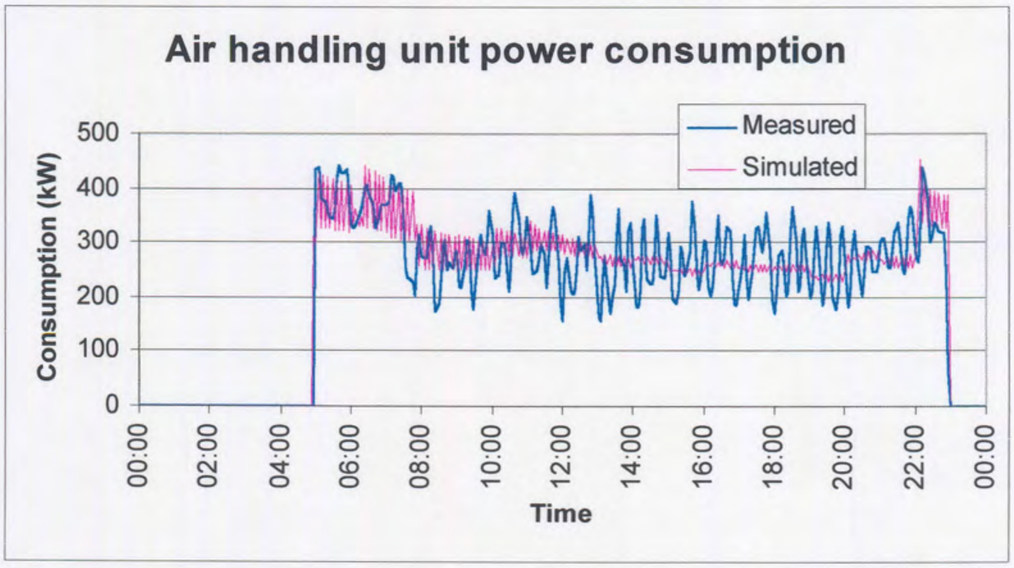

Figure 3-7 – Comparison of measured and simulated power consumption of the AHUs

Figure 3-7 shows the measured and predicted electrical power consumption of the AHUs. The results are sufficiently accurate with a maximum daily error of only 2.2% on total consumption (kWh). Figure 3-8 shows the simulated and measured chiller power consumption for the verification day. As can be seen step loading and unloading could be simulated successfully. There is only a small phase difference between the loading times of the results, which indicates that the simulation model represents the actual building closely. The measured and predicted energy consumption for the 24-hour day differed by only 0.61%. This was also sufficiently accurate.

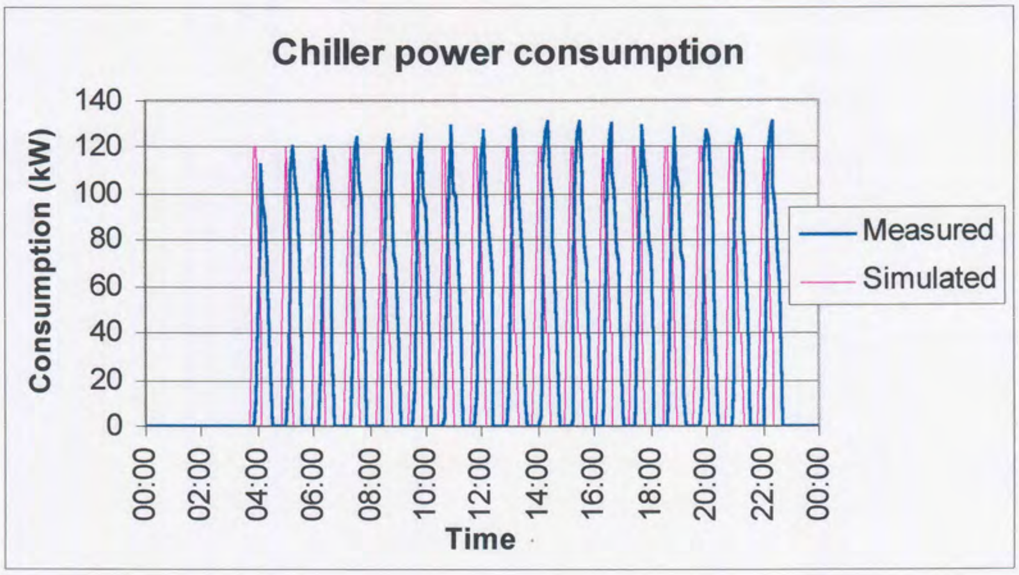

Figure 3-8 – Comparison of measured and simulated power consumption of the chillers

The verification study results show that the dynamics of the building, HVAC system and its controls were simulated with commendable accuracy. With the accuracy of the simulation tool successfully verified, it could then be applied to retrofit investigations with the assurance of achieving credible predictions.

3.7 References

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CHAPTER 4 - Application 1: Potential for energy saving

After verifying the simulation model the model was upgraded to support the required setpoints in each of the building zones. To find the potential for energy saving six retrofit options were implemented and evaluated according to two criteria: electrical energy saving and straight payback period. It was found that by improving the system operation with air-bypass, reset and setback control in conjunction with improved operating times the highest energy saving potential could be realised.

4.1 Introduction

With the simulation model verified, the first application that was considered was a retrofit investigation to obtain the energy saving potential of the building. The investigation was focussed specifically on the HVAC system because it is responsible for more than 54% of the building's total energy consumption. The savings potential as far as the lighting schedule is concerned is fairly straightforward. Simply switch off unnecessary lights by either employing a timer module or a motion detector system in the lecture halls.

Various HVAC retrofit options were evaluated. These included:

- adjusting indoor temperature set points to more acceptable levels
- air bypass control for the AHUs
- reset control on the AHUs
- setback control (motion detector driven) in the lecture halls
- improved HVAC system start-stop times
- economiser cycle combined with all of above
- CO_2 control

The retrofit energy savings potential was evaluated by comparing new energy consumption figures to that of the present system.

4.2 Retrofit simulations

4.2.1 Updated current system

To obtain a realistic comparison, the current energy consumption figures were calculated for a system with updated setpoint values and acceptable indoor air temperatures. All the applicable setpoints were adjusted to meet the requirements stipulated in the Chapter 2. Since the system had to do more heating and cooling in order to support the new load conditions, more electrical energy was consumed by the entire HVAC system:

HVAC Component	Consumption per day (kWh)
Chillers	1262
Cooling Towers	553
Pumps	1184
Fans	1104
Heaters	3789
Total	7890

Table 4-1 – Average daily power consumption of the HVAC components in the updated system

4.2.2 Air bypass

The first energy saving retrofit improvement investigated was air bypass of the AHU cooling coils. This system comprises damper control units fitted to the AHUs to regulate the portion of supply air to bypass the cooling coil. Currently these units are fitted to the AHUs supplying five lecture facility zones (zone 3, 4, 5, 6, and 8) and are either not operational or insufficient to efficiently regulate the mass flow rates required. The air bypass retrofit consists of the repair and upgrading of these units and installation of the control strategy in the other AHUs if applicable.

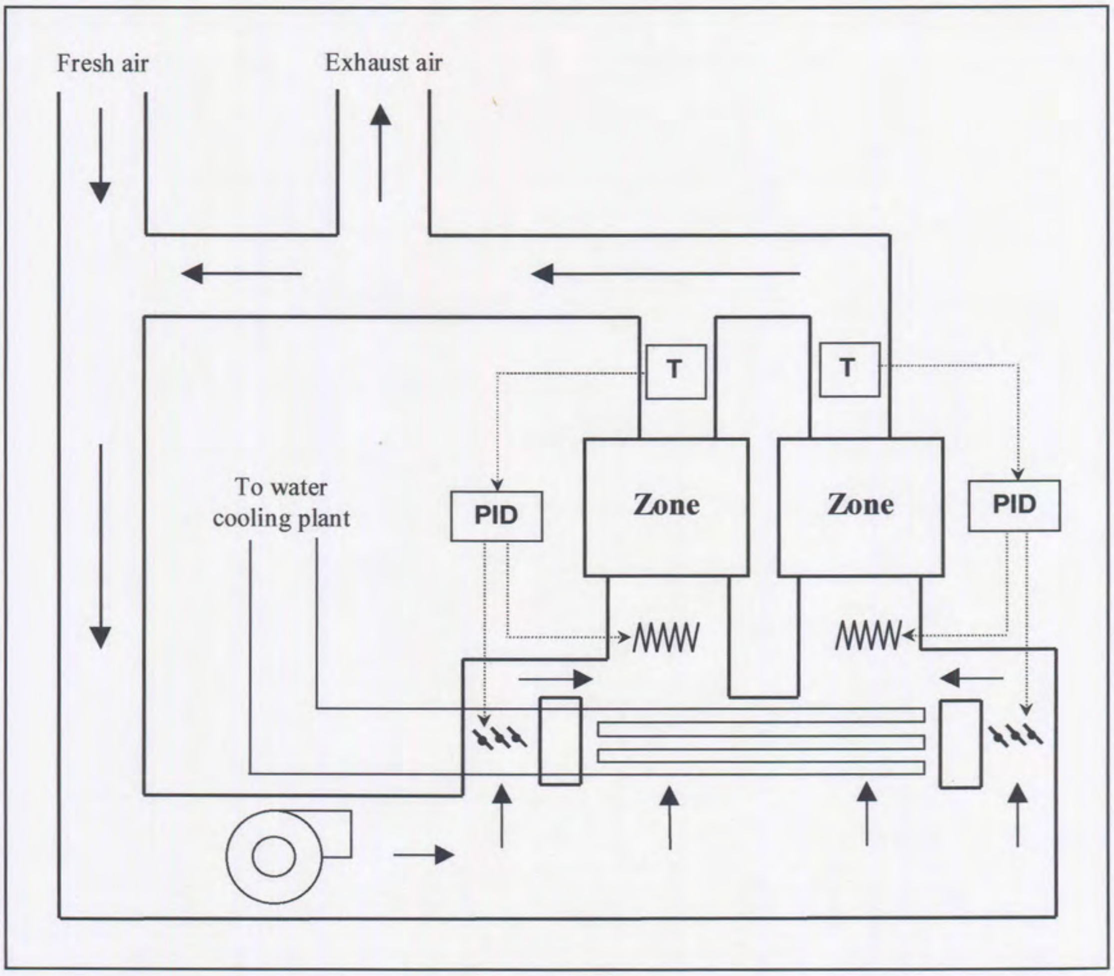

Figure 4-1 – Schematic layout of the standard Air bypass system

This option could account for much more savings should different load conditions be introduced to the system. The control strategy is especially appropriate for multi zoned air supply. If one of the lecture halls would have a large cooling load, that coil would have to supply air at a sufficiently low temperature to support the load. If the adjacent halls had a lower load, re-heating would have to be done to maintain that hall to its setpoint. With air-bypass implemented, this last hall's supply air would not be cooled, it would not require re-heating and electrical energy would be saved.

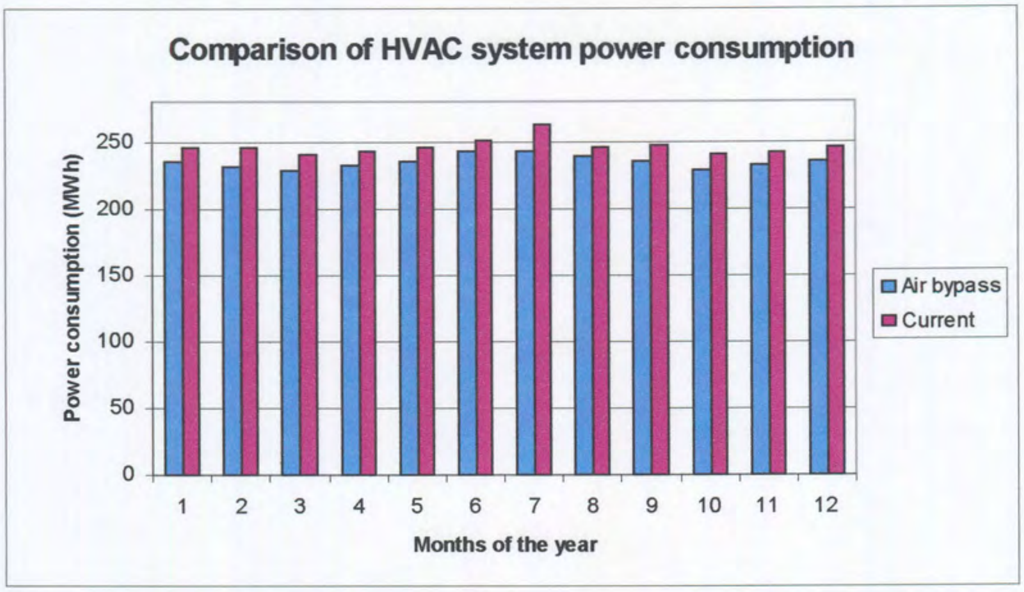

Figure 4-2 – Energy consumption of the Air bypass retrofit

The current and retrofitted system year simulation results are shown in figure 4-2. The total yearly energy consumption came to 2819 MWh, which constitutes a 4.6% saving. This is clearly not substantial, however this retrofit being relatively inexpensive may be an option and will be reconsidered in the economic analysis.

4.2.3 Reset control

To save more energy reset control was investigated as a retrofit option. This means that in the air-bypass system's AHU supply air temperature is controlled from return air temperature input and not from supply air temperature sensors. Cooling control is achieved by regulating the chilled water flow rate through the AHU coil. This will minimise the amount of cooling and re-heating required.

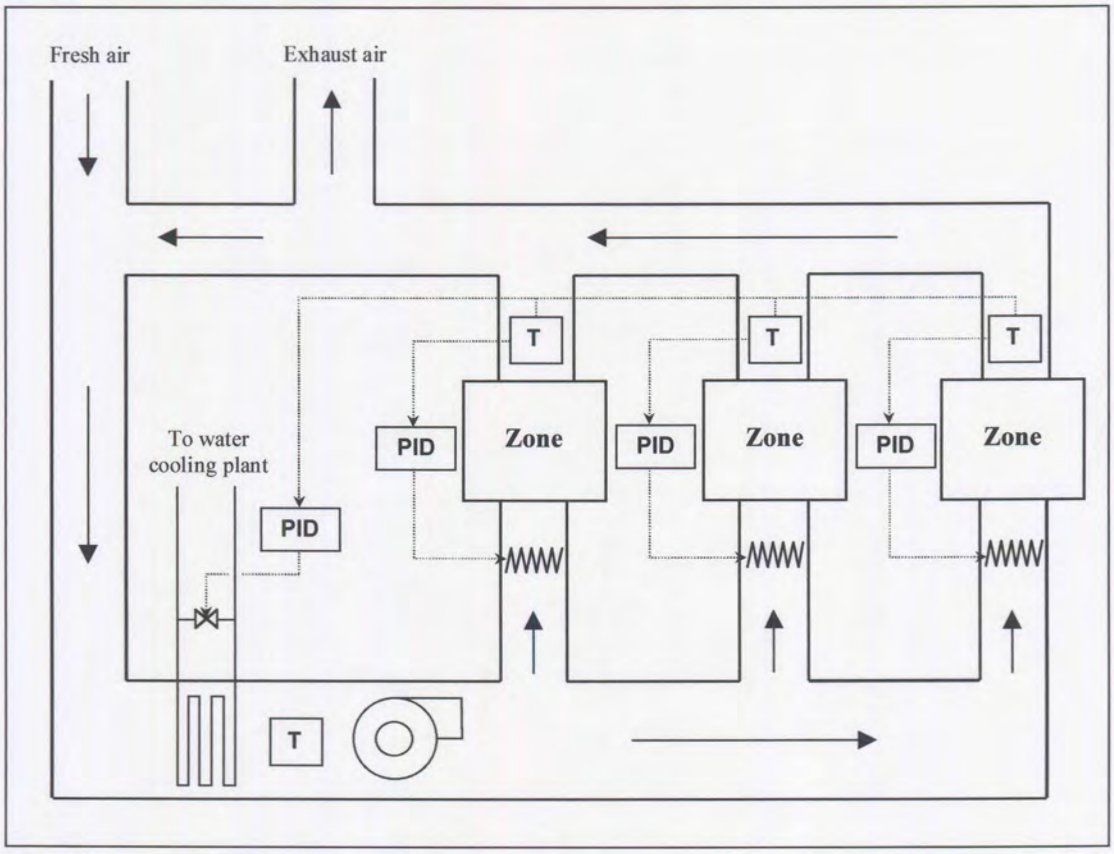

Figure 4-3 – Schematic layout of the standard reset control system

With the system as shown it is likely that either one of the controllers may saturate at high output and thereby maximise simultaneous heating and cooling in stead of minimising it. To overcome this problem the setpoints for the controllers were set at sufficiently different temperatures. (Example: When a zone is cooling down from a high temperature, as the return air dry bulb temperature decreases, the coil will have to stop cooling before the heater starts heating).

Furthermore, due to the multi-zoned nature of some of the zones, a digital system will be required to identify and control the chilled water supply rate to account for the zone with the largest cooling load (highest return air temperature) at any particular time. Control of the indoor temperature for the other zones was achieved through re-heating.

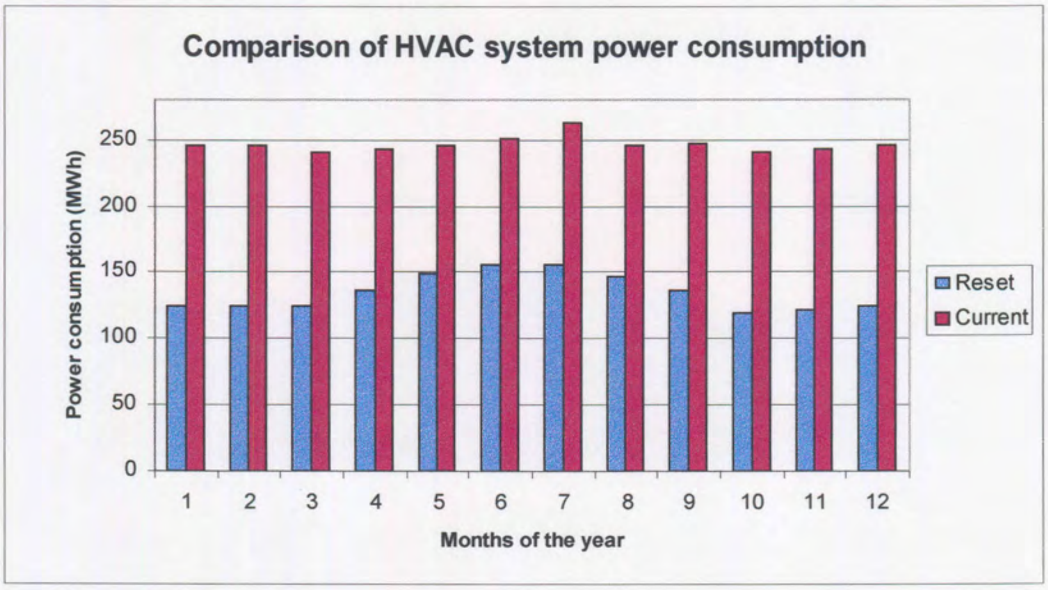

Figure 4-4 - Energy consumption of Reset control retrofit

The year simulation results are shown in figure 4-4. The results showed a substantial energy saving of 45% on the HVAC system's total energy consumption. This is a saving of 1 309 MWh per year.

4.2.4 Setback control

Setback control involves selectively relieving HVAC component setpoints. A typical example would be the relaxation of return air temperature constraints in unoccupied zones implemented through the use of motion detectors. It can be viewed as an extension of the air-bypass system together with reset control.

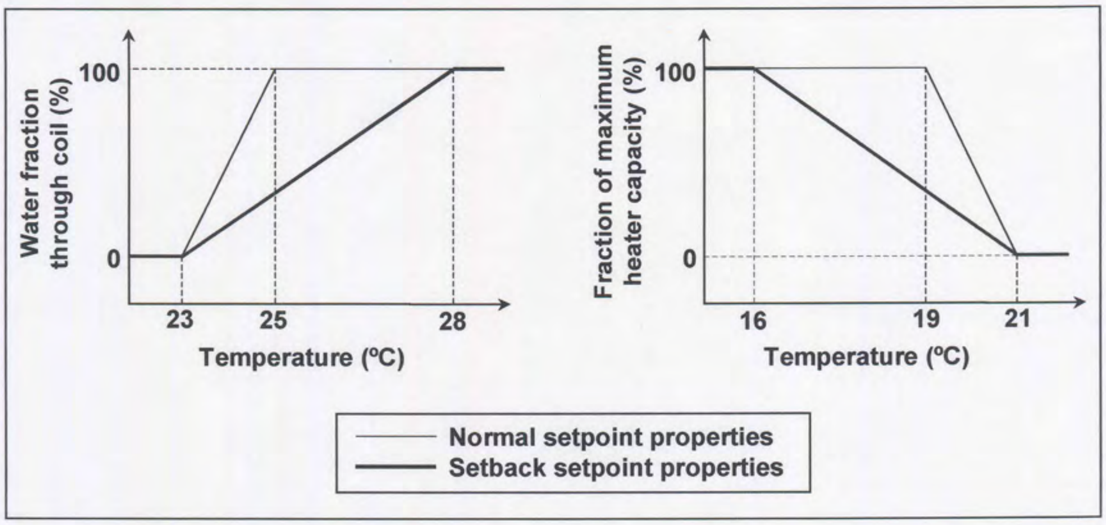

Figure 4-5 – Setpoint values for Setback control

Return air set points were allowed to drift between 16°C and 28°C when the zones were unoccupied. This range was chosen to assure that the lag period (time constant) of the system was insignificant enough not to influence comfort levels should sudden changes in internal thermal load occur.

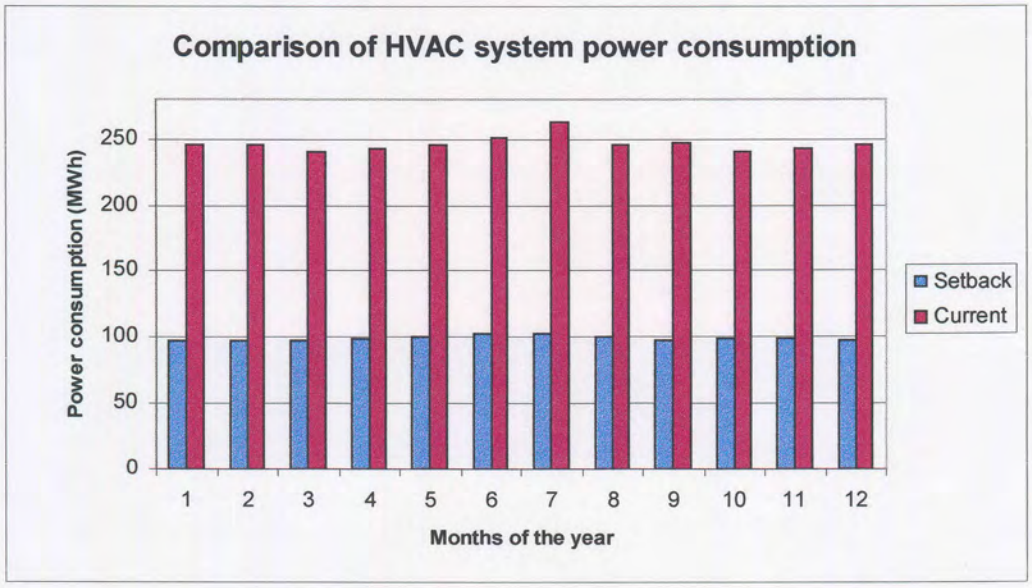

Figure 4-6 - Energy consumption of Setback control retrofit

An integrated retrofit simulation showed setback control to provide substantial annual energy savings of 60% or 1742 MWh (simulation results are shown in figure 4-6).

Bearing in mind that digital control is most likely to form part of the retrofit on account of its relative low cost, versatility and robust nature, this option poses to be an even further possibly lucrative choice worth evaluating in the economic analysis.

4.2.5 Improved HVAC system start-stop times

In all above simulations the HVAC system was in operation every day of the year as in the present system. For the intended use of this building this is clearly unnecessary. These retrofit simulations comprised of switching off of the system on obvious vacant days e.g. Sundays and public holidays.

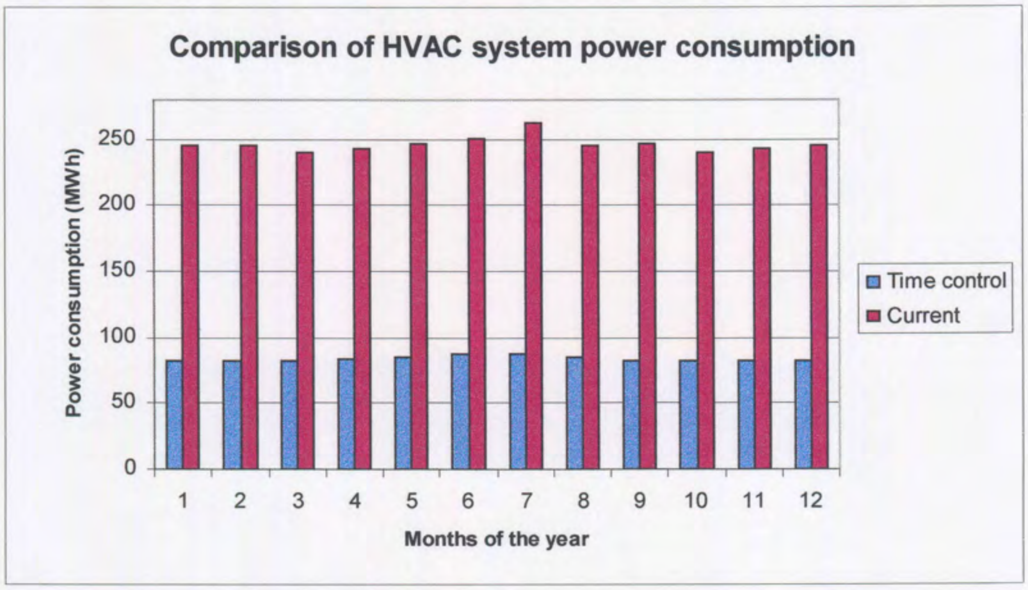

Figure 4-7 - Energy consumption of Time management retrofit

The improved start-stop time retrofit simulations predicted annual energy savings of 66% or 1 906 MWh. Bearing in mind the relatively inexpensive nature of this retrofit (involves the addition of timers or mere manipulation of operation set-points in the case of digital control) this retrofit option poses to be the most lucrative thus far.

4.2.6 Economiser control

To illustrate its cumulative effect, fresh air economising was investigated in simultaneous application with air-bypass, reset and setback control and improved HVAC system start-stop times. This was the sensible approach since economiser control without the other retrofits (like reset and setback control) would be costly and impractical.

With the economiser cycle the amount of fresh air into the system is controlled. This proposes to save on unnecessary cooling of supply air if cool outside air can be bled into the system in stead. Conversely available warm air can also be used. Economiser setpoints were chosen at 22°C and 23°C dry bulb for cool and warm intake air respectively.

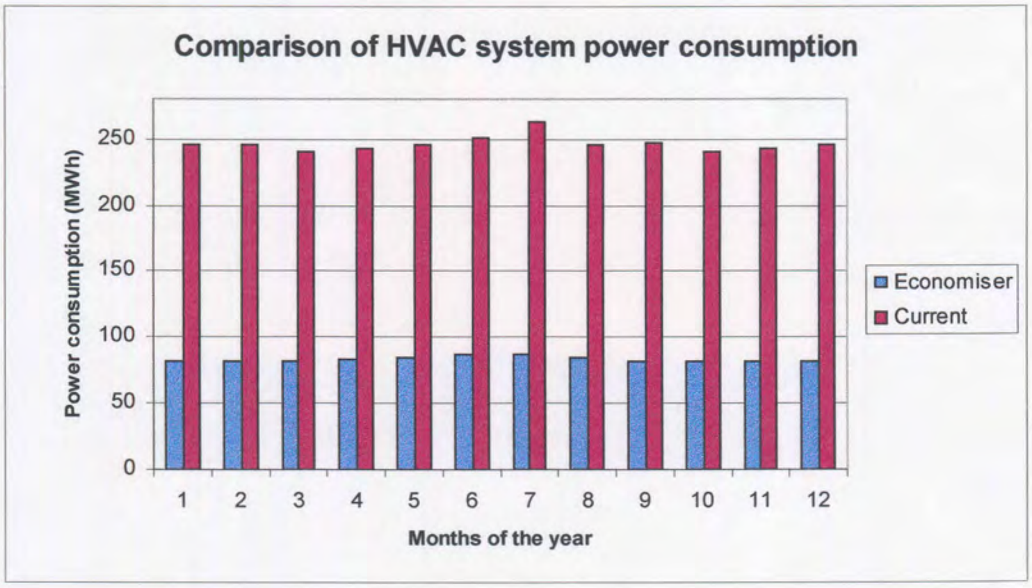

Figure 4-8 - Energy consumption of Economiser control retrofit

Resulting predicted savings came to 66% or 1 918 MWh annually. Compared to the other options economiser control adds but little value in terms of energy saving. This is mainly due to the specific climatic conditions not lending to cooler air during summer or warmer air during winter being drawn in from outside. Since expensive structural modifications will have to be made to the building the economiser cycle might prove impractical.

4.2.7 CO_2 control

An improvement on the above economiser system is through carbon-dioxide control. This system minimises fresh-air intake by controlling oxygen and odour levels through the measurement of CO_2 concentration. This control strategy was again simulated in conjunction with the first four to determine its cumulative effect.

In accordance with the world health organisation, CO_2 concentration levels are limited to 1300 ppm (parts per million) while 600 ppm is recommended [1]. As the outside air in this case has relatively high CO_2 concentration levels, between 700 ppm and 950 ppm , a level of 900 ppm was chosen for the indoor air.

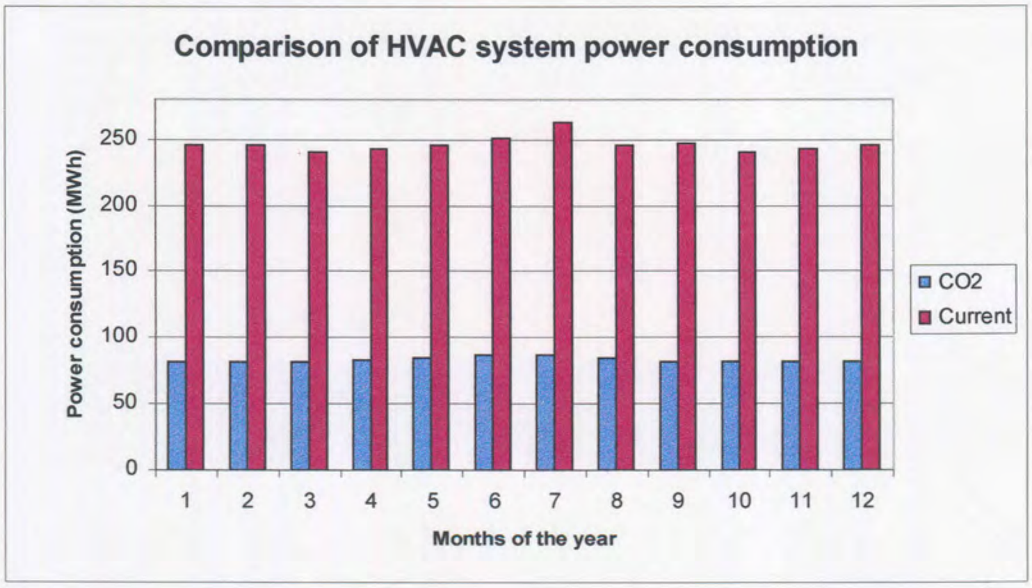

Figure 4-9 - Energy consumption of CO_2 control retrofit

The CO_2 retrofit option resulted in predicted savings of only 66%. Similar to the economiser option, CO_2 control is expected to have a marginal effect on the energy efficiency of the building.

4.2.8 Summary of retrofit options

The retrofit options investigated showed that the Human Science building has considerable energy savings potential. The results are summarised in figure 4-10. As shown, HVAC system energy savings of as much as 66% are attainable without compromising indoor air comfort levels. A 66% saving in HVAC energy constitutes a saving of more than 35% of the total building energy consumption.

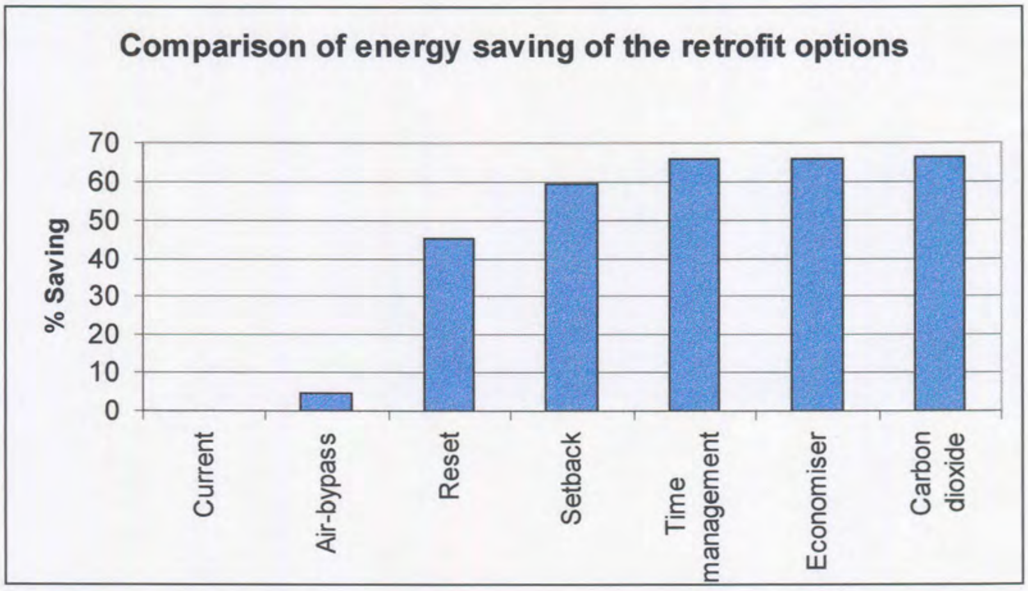

Figure 4-10 – Retrofit option saving summary

The following section focuses on determining the most economically viable retrofit option.

4.3 Economic analysis

An economic analysis was conducted to weigh up retrofit cost with expected monetary savings. This was done on the assumption that the same electricity tariff will be charged regardless the amount of electricity required. In other words, if half the normal amount of energy is consumed, the energy bill at the end of the year will be half the previous one.

The current energy consumption bill for 2653 MWh amounts to R270 035 per annum. Since the system is not in full working condition and the indoor air conditions are outside the desired human comfort range, the system has to be upgraded. This upgrade will introduce an added running cost of R25 511, which increases the current cost by 9.4% to R295 546. During the economic analysis the retrofit savings were weighed against this running cost.

The straight payback period was calculated as follows [2]:

$$Packback \quad Period_{months} = 12_{months}\left(\frac{Initial \quad Cost}{Savings}\right)$$

Retrofit option	Running cost (R/year)	Saving (R/year)	Implementation cost (R)	Payback period (months)
Current system	295 546	0	1 600	-
Air bypass control	281 998	13 548	135 900	120.4
Reset control	161 801	133 745	137 500	12.3
Setback control	119 218	176 328	145 000	9.9
Time management	100 269	195 277	146 600	9.0
Economiser control	100 355	195 191	204 500	12.6
CO_2 control	92 326	203 220	379 500	22.4

Table 4-2 – Economic comparison of retrofit options

The implementation costs were calculated and estimated according to manufacturer and supplier price lists and selection guides. The details are stipulated in Appendix C.

A summary of the results is shown in the figure below with the straight payback period compared in months. The air-bypass system alone had a relatively long payback period (120.4 months) due to the high implementation cost needed to upgrade the current system. It is therefore not shown in figure 4-11. However if air-bypass control is installed together with the other retrofit options the scenarios are as follows:

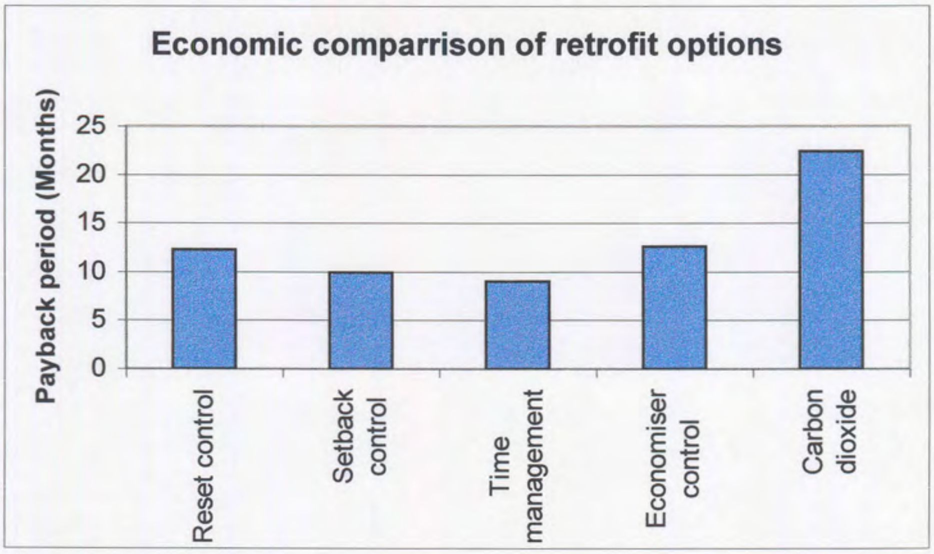

Figure 4-11 – Straight payback periods of the retrofit options

4.4 Conclusion

The Human Science building has a large potential for saving on electrical energy consumption. The heating ventilating and air conditioning system consumes 54% of the total electrical energy supplied to the building. Of this up to 66% can be saved by employing some of the retrofit options investigated.

From the results it is clear that the application of better time management will result in a straight payback period of only 9 months. This option, having the smallest payback period, is thus recommended as the retrofit option of choice. Installation of this option includes upgrading of the current system and implementation of air-bypass control, reset control and setback control. Up to R195 000 can be saved annually.

4.5 References

[1] ASHRAE, ANSI/ASHRAE Standard 55-1981, *Thermal environmental conditions for human occupancy*, Atlanta, Georgia, 1981.

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CHAPTER 5 - Application 2: Effect of equipment failure

Another application for the verified simulation model was to investigate the effect of fouling on indoor climate and energy consumption. In the Human Science building the major role playing components are the cooling coils and the chillers. By reducing the efficiencies of both these components it could be established that the HVAC system was capable of maintaining the indoor temperatures at acceptable levels up to 50% failed equipment. With this application the worth of the integrated simulations was highlighted since this prediction method could also be implemented for other scenarios in a similar way.

5.1 Introduction

Another application of the verified simulation model and simulation software package was to determine the effect of fouling of HVAC equipment on the entire HVAC system. In high risk applications correct prediction of fouling may save money or even lives [1].

The term 'fouling' may be explained by 'becoming less efficient' due to a time dependant phenomenon like contamination. Many textbooks are devoted to theoretical prediction of the fouling effect but in most cases the occurrence is examined in an isolated environment [2]. Often fouling does not play such an important role with the heat exchanger under investigation as it does with the system it is installed in.

Fouling is defined by [1]:

$$fouling \quad factor = \frac{normal \quad efficiency}{fouled \quad efficiency}$$

or $$f = \frac{\eta_{normal}}{\eta_{fouled}}$$

The purpose of this case study was to evaluate the influence of fouling in an integrated HVAC system. Since the simulation model and necessary software were already available, modifying the virtual system to investigate fouling was elementary.

The study consisted of two analyses: Firstly the efficiencies of the coils in the Humans Science building were reduced and the indoor air conditions were evaluated. Secondly the coils were restored to full working capacity and then the effect of chiller failing was evaluated.

5.2 Coil fouling

Fouling in coils may occur for a number of reasons, but usually age and/or neglect are to be blamed. The foulant may be crystalline, biological material, the products of chemical reactions including corrosion, or particulate matter [2]. Especially in cooling coils, as is the case with the Human Science building, moisture is condensed on the fins and hence an effective filter is formed that contracts dust and other contaminants from the working fluid (air passing through the coil). If the coils aren't cleaned regularly, the coil effectiveness is reduced by two mechanisms. Firstly a reduced heat transfer coefficient, and secondly a greater pressure drop over the coil area [3].

As is the case with the air flowing through the coil, the water flow heat transfer may also degenerate. Impurities in the chilled water from the cooling plant accumulate on the tube walls in the coil which decreases the heat transfer coefficient and heat transfer area and again increases the pressure drop in the water due to a reduced flow area [2],[3]. Maintenance for both these problems is often neglected.

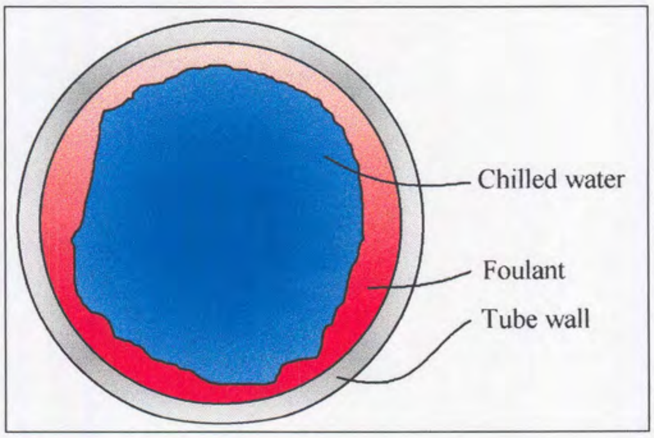

Figure 5-1 – Foulant deposit on the inside of a heat exchanger tube

Another form of reducing the effectiveness of the coils is by means of rust or corrosion. Because of the harsh environment in which these coils have to operate the coil materials are under tremendous mechanical and chemical strain. Temperatures may vary considerably over short periods of time and often the coil fins have to deal with high oxygen supply and wet conditions. This leads to oxidation of the fins [4],[5]. Again the heat transfer properties are influenced.

After a brief study it was found that the effect of fouling in only one cooling coil has the same effect on its specific zone indoor conditions as when all the coils are fouled together. The main difference being only that the energy consumption of the chillers is much higher. Thus it was decided to perform the simulation studies with all the coils failing by the same amount (equal fouling factor). This resembled reality much better since the Human Science building is not cleaned very often and subsequently the coils are fouled by roughly the same amount at any given time.

The following scenarios were investigated:

- Coils in 100% working condition
- Coils operating at 50% of full capacity (Fouling factor of 2)
- Coils operating at 25% of full capacity (Fouling factor of 4)

The Human Science building only incorporates cooling coils and therefore the effect of coil fouling was not investigated for heating coils. This allows for some further studies.

5.3 Results

The following figure shows the average temperatures in the 12 zones after the coil efficiency have been reduced. The detailed results for the individual zone temperature response are shown in the appendixes.

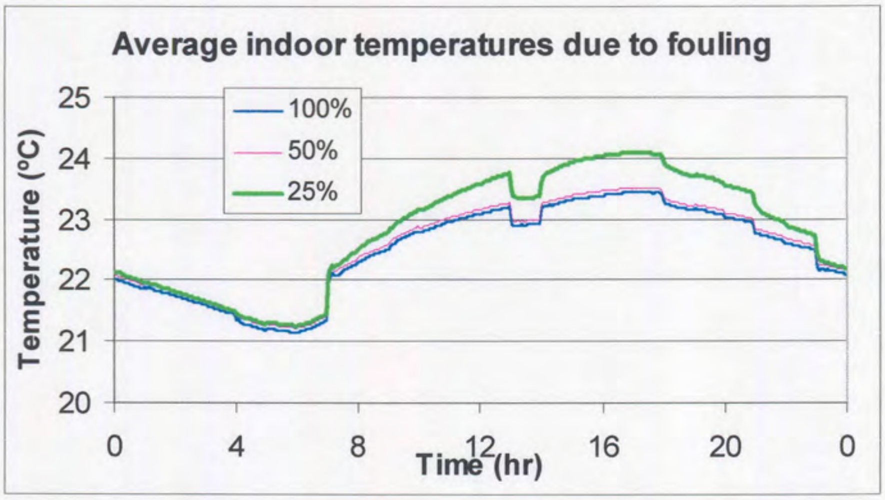

Figure 5-2 – Comparison of indoor temperatures at different coil efficiencies

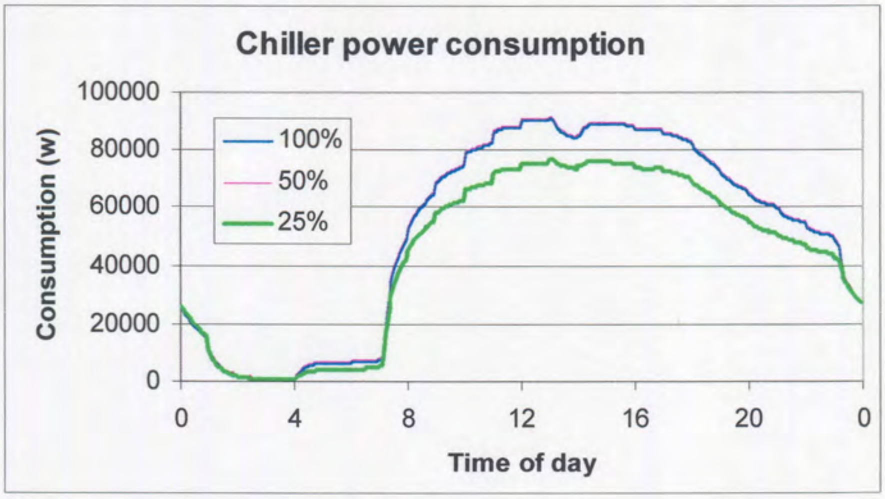

Figure 5-3 – Comparison of chiller power consumption at different coil efficiencies

From the graphs it is clear that the effect of the fouling is only realised when the coils have failed by 75% of rated capacity. This is a significant discovery since it proves that maintenance on the system would only be necessary after the coils have lost more than half their efficiency. If a suitable time dependant function could be found to predict the rate of fouling the necessity for coil maintenance could be estimated before unnecessary and costly cleaning operations are performed. This might save money in the long run [6].

In the case of the Humans Science building a possible reason for the coils not responding to fouling too soon is the fact that the whole system is over designed by quite a large margin [7]. The HVAC system is therefore more than capable of meeting the control specifications even though the coils are transferring heat at 50% of rated capacity. What was significant though was that fact that the heating elements did not do as much heating as when full working conditions were simulated. This can be attributed to unnecessary re-heating being avoided. At first glance the system actually seems less energy reliant.

Heater power consumption			
Capacity	100%	50%	25%
Fouling factor	0	2	4
Consumed (kWh per day)	4282	4135	3245
Saving	0%	3.4%	24.2%

Table 5-1 – Heater consumption at different coil efficiencies

In the case of the 25% efficient coils the results are misleading. The air handling units are unable to meet the required comfort conditions as was specified on the controllers. Since the purpose of employing the HVAC system in the first place was to maintain the indoor air conditions at acceptable comfort levels, the results found with the inoperative coils are unacceptable. Thus, even though it might seem to be a more energy effective system, the occupants will not be comfortable.

5.4 Chiller fouling

As is the case with the coils, the chillers also become less efficient over time. This can mainly be ascribed to either the compressor failing or the evaporator and condenser fouling [8]. In the case of the Human Science building a screw compressor is used to compress the working fluid (in this case R-22) and transfer heat from the evaporator to the condenser by means of a thermodynamic cooling cycle. Wear and tear on the compressor blades cause leakage and back flow which in turn causes the coefficient of performance (COP) of the chillers to decrease [7]. At some point the COP of the chillers becomes so low that they are unable to produce chilled water at the rate required for the HVAC system. Then the failing can be experienced with changes in indoor climate.

$$COP = \frac{Q}{Pwr}$$

where:

Q = The amount of cooling performed on the chilled water

Pwr = The amount of electrical energy consumed by the chiller compressor [7]

Fouling of the evaporators and condensers is very similar to fouling of the cooling coils. Again the units become less efficient over time as regular maintenance is neglected. Contaminants and impurities cause the heat transfer coefficients to decrease and the pressure drop to increase [2],[3].

The simulations for determining the effect of failing chillers were performed in the same way as with the coils. Both the chillers were degraded by the same fouling factors. This implies that the overall COP of the chillers is also degraded by that same factor.

The following scenarios were again investigated:

- Chillers in 100% working condition
- Chillers operating at 50% of full capacity (Fouling factor of 2)
- Chillers operating at 25% of full capacity (Fouling factor of 4)

5.5 Results

Plots of the average temperatures in the zones and the chiller power consumption is shown in figure 5-4 and 5-5.

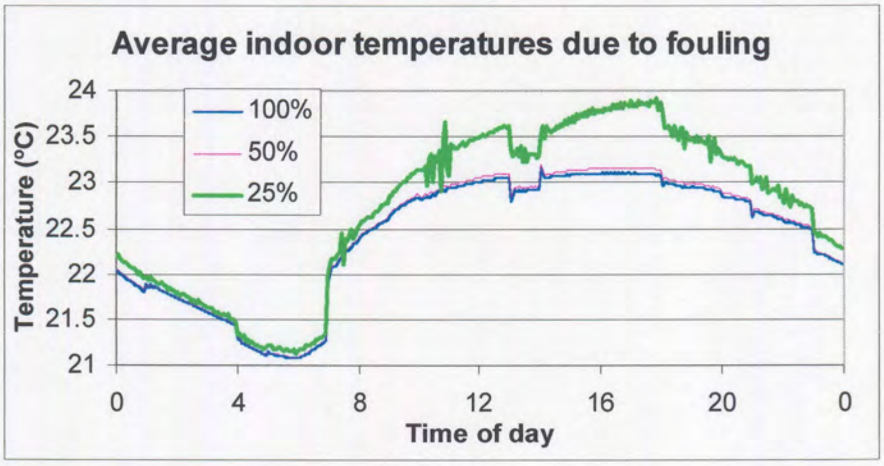

Figure 5-4 – Comparison of indoor temperatures at different chiller efficiencies

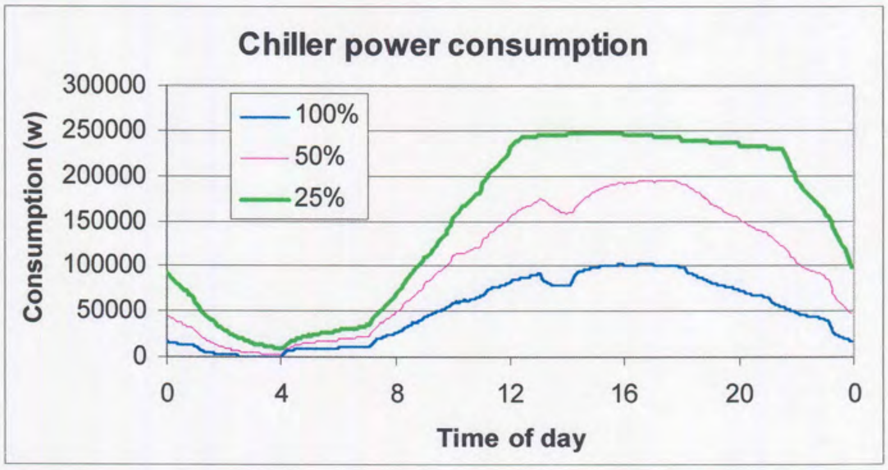

Figure 5-5 – Comparison of chiller power consumption at different chiller efficiencies

From the figures it is evident that the chiller failing also only starts making an impression on the indoor temperatures after the COP of the chillers was reduced by more than half. This means that the chillers are indeed over designed since they are able to maintain the chilled water temperature at the required level (7°C) up to $f = 2$. The impact of the 50% degenerated COP is only visible on the electricity bill.

When the COP was reduced further, to 25% of rated capacity, the chillers were unable to produce chilled water at the specified rate. Only then the coils received sub standard water supply, the air could not be cooled sufficiently and the indoor temperatures rose to over the required comfort levels. As soon as the HVAC system is unable to maintain the specified comfort requirements, the system becomes obsolete.

As was the case with the coils, the heaters did not operate as much with the failed chillers as they did with the 100% working condition chillers. The same reasoning could be employed. Since the chilled water could extract less heat from the supply air (reduced temperature difference) less re-heating was required.

Heater power consumption			
Capacity	100%	50%	25%
Fouling factor	0	2	4
Consumed (kWh per day)	4282	4111	3874
Saving	0%	4.0%	9.5%

Table 5-2 – Heater consumption at different chiller efficiencies

The same notion is proposed as with the fouling prediction of the coils: If a mathematical function (empirical or otherwise) could be found to predict the deterioration rate of the chillers, a suitable maintenance scheme may be derived to streamline maintenance and improve the savings potential of the building [6].

5.6 Conclusion

From the simulation data collected it is clear that the system is grossly over designed. Both the coils and the chillers may be allowed to operate at half the design efficiency before any significant changes in indoor climate are realised. It shows that initially much smaller HVAC components could have been implemented. This would not only reduce the initial installation cost, but also the maintenance and, if necessary, replacement cost.

With the components failing, less heating is required. At first glance the system might actually seem more energy efficient. This is however not the case. Less energy is consumed, but the system is unable to meet the comfort requirements as were specified in the previous chapters.

From a safety point of view the fouling effect does not seem to have such a big impact on the building or its occupants. The maximum temperature found in any of the zones is 31.1°C which is uncomfortable but not unsafe for an office/lecture building. The lowest temperature was 9.2°C during one of the winter months. Again, since the building does not support temperature sensitive equipment this temperature is very uncomfortable but safe for human occupants [9]. The building has a relatively large thermal mass that acts as an effective regenerator. Even without heating and cooling the minimum temperatures are never allowed to dip too low and visa versa the maximum temperatures never rise too high.

This application of the model clearly points out the worth of the simulation package and the correctly verified simulation model. Without the aid of the simulation tool it would have been impossible to predict the effect of fouling on the indoor climate. This method of prediction might also be implemented in other scenarios, for example high-risk applications like laboratories or chemical plants. In these types of applications the benefits of accurate prediction is obvious.

5.7 References

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[8] T.R. Bott, R.A. Walker, *Fouling in heat transfer equipment*, Chemical Engineering Journal, No.251, pp. 391-395, 1971

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CHAPTER 6 - Closure

In this chapter a brief summary of the other chapters is given. The results of the two simulation applications are discussed and a few suitable recommendations are made for future work.

6.1 Summary

Simulation is one of the oldest and also among the most important tools available to engineers [1]. In the HVAC community the availability and/or functionality of simulation tools is limited though. Furthermore it is difficult to determine whether the simulation models accurately represent reality. The purpose of this study was to accurately verify one such a simulation model and then to extend the study to two distinct applications: Determining the energy savings potential of the Human Science building and investigating the effect of failing HVAC components on the air conditions and energy consumption of the building.

Heating, ventilating and air-conditioning in buildings consume a significant portion of the available electrical energy in South Africa [2]. Of this energy up to 30% can be saved by improving the HVAC systems currently installed in the buildings. This could result in savings of up to R400 million [3]. By accurately simulating control retrofits on the system the highest potential for energy saving could be found.

Predicting the impact of failing equipment is also a difficult task because of the integrated dynamic effect every component has on the next. With the aid of a comprehensive integrated simulation model the implications of failing can be determined and the necessary assessments and precautions can be taken. The verified simulation model was used to determine whether these effects could be predicted successfully.

This project followed the following procedure:

- Firstly all the applicable data about the building was collected and analysed.
- Then a suitable simulation model was constructed and extensively verified.
- This verified model was then applied to perform the above mentioned two studies.

6.2 Results

During a series of audits (comfort and energy) the building in question was extensively measured and analysed. It was found that the building was in poor condition and a few possibilities for energy savings were already identified. During the data acquisition stage it was decided that since the HVAC system is fully installed, only control retrofits would be considered for the energy saving study.

The HVAC model was then constructed with the aid of the simulation package: QUICK Control*. The model, consisting of 13 zones, contained all the components of the HVAC system which were individually constructed and verified to accurately represent reality. After some alterations to control parameters and component specifications the simulated outputs were sufficiently close to the measured outputs. This means the model was verified and ready to be used for the two case studies.

The first of the case studies was finding the potential for energy saving of the Human Science building. After the model's control parameters were updated to adhere to comfort regulations, six retrofit options were applied to the model. The results of the retrofits revealed that up to 66% of the building's HVAC energy can be saved by simply installing an improved control system.

After a detailed economic analysis the option with the lowest straight payback period was chosen as the one to implement in the building. The chosen option was improved time management with reset and setback control. It has a straight payback period of only 9 months with a predicted saving of R195 000 per year.

The second application of the verified simulation model was evaluating the effect of fouling of HVAC components. The coil and chiller efficiencies in the system were reduced and the output of comfort conditions and energy consumption were analysed. It was found that the system had the capacity to maintain the indoor air conditions at acceptable levels up to a fouling factor of 2 (50% less efficient).

* Software developed by TEMM International (Pty) Ltd and used with special permission.

The energy consumption effect of the failing equipment was also investigated. With the coils fouling the heaters consumed up to 30% less electrical energy. In the case of the chiller COP reduction up to 10% less heating was done. Since at these conditions the indoor climate falls outside the acceptable levels the reduction in heating load could not be attributed as savings though. Only with the aid of the integrated simulation was it possible to find how much fouling was acceptable.

These case studies clearly show the worth of the integrated simulation. The substantial savings and possibility of predicting climate deterioration would not have been possible without the aid of a comprehensive simulation package and model.

6.3 Recommendations for further work

During the project some problems were encountered. One major concern was finding component models for some of the components that were outdated and old. Many assumptions were made to incorporate some of the components but verifying the models was a difficult and tedious task. If the simulation process is to be streamlined an investigation should be launched into modelling such components in a more robust way.

The simulation package is very comprehensive. It could be recommended that a few obvious assumptions are made by the program to avoid tedious details that might have little effect on the simulation outputs. This presents the opportunity for many studies since extensive verification will be needed.

The study was only performed on one scenario, namely a commercial office/lecture facility. Even though both the simulation applications proved much worth it is recommended that more case studies are performed to further determine the prediction ability of the simulations. Other retrofit options and other equipment fouling may be investigated. Especially high-risk applications where the fouling might have a larger impact should be looked into.

Furthermore only cooling coils and chillers were investigated in this project. The effect of other equipment like pumps, fans, heating components, etc should also be studied.

6.4 Conclusion

All the objectives of the study were achieved. A simulation model was successfully constructed and verified to accurately simulate the heating ventilating and air-conditioning system of a commercial building. The model was also used to firstly find the energy saving potential of the building through control retrofits and secondly to find the dynamic impact of failing equipment on the entire integrated system. Judging by this project every building owner, contractor and engineer should employ this simulation method for accurate condition and energy prediction.

6.5 References

[1] J. Lebrun, *Simulation of HVAC systems,* Renewable Energy, Vol. 5, Part 2, pp.1151-1158, 1994.

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Appendix A - Building data

A.1 Air Handling Unit details

The following table contains some of the detailed information of the various AHUs in the Humans Science building.

AHU	Model	Serial No.	Volume	Supply (m^3/s)	Return (m^3/s)	Outside (m^3/s)	Water (l/min)	Water (l/s)	Mass fraction	Cooling capacity	Air changes
1	MZ 80	7216813	976.8	4.05	3.24	0.81	160	2.67	7.8	74.4	14.93
2	MZ 80	7216814	976.8	4.05	3.24	0.81	160	2.67	7.8	74.4	14.93
3	HAH 108	7216747	1097.3	5.2	4.16	1.04	200	3.33	9.8	92	17.06
4	HAH 81	7216818	1268.9	4.01	3.21	0.80	142.8	2.38	7.0	66.2	11.38
5	MZ 100	7216816	1048	4.96	3.97	0.99	189.2	3.15	9.2	87.9	17.04
6	MZ 100	7216815	1048	4.96	3.97	0.99	189.2	3.15	9.2	87.9	17.04
7	HAH 108	7216994	1543.6	4.85	3.88	0.97	169	2.82	8.3	78.5	11.31
8	HAH 81	73H2130	655	3.78	3.02	0.76	135.2	2.25	6.6	62.7	20.78
9	HAH 108	73H2131	1048.5	5.31	4.25	1.06	202	3.37	9.9	93.8	18.23
10	HAH 108	73H2132	1048.5	5.31	4.25	1.06	202	3.37	9.9	93.8	18.23
11	MZ 80	73H2133	730.4	4	3.20	0.80	148.5	2.48	7.3	68.9	19.72
12	MZ 80	73H2134	730.4	4	3.20	0.80	148.5	2.48	7.3	68.9	19.72
							Total	34.11			
PA 1				5.1	0	5.1	869.3	14.49	50		
PA 2				5.1	0	5.1	869.3	14.49	50		
							Total	28.98			

Table A-1 – AHU details

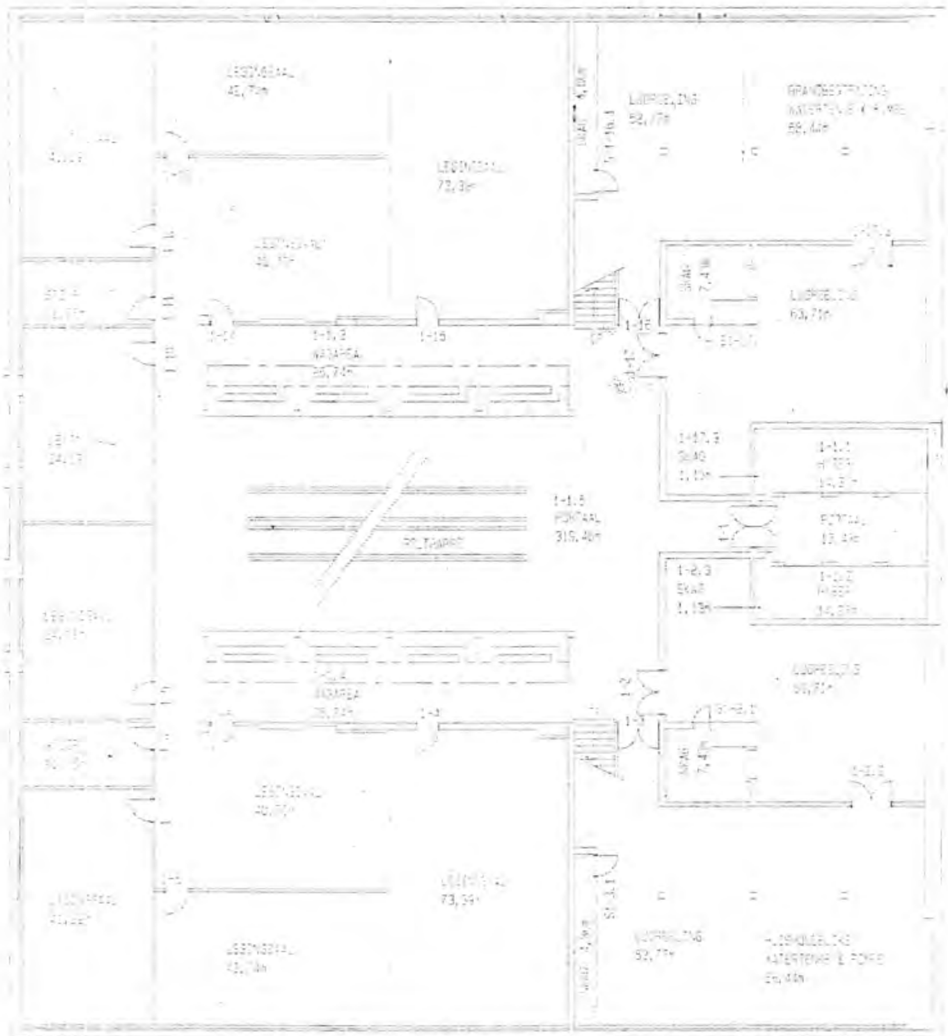
A.2 Zone details

Zone	AHU	Location	Supplies to floor	Room Number	Heating Capacity	Lighting System
1	4	1_2	1	15	6	1375
				14	3	660
				13	3	800
				12	3	800
				10	3	640
				4	6	1375
				5	3	660
				6	3	800
				7	3	800
				9	3	640
				Total	36	8550
2	7	2_2	2	14	1	330
				15	5	1935
				14.1	1	500
				12	5	1935
				13	1	500
				9.1	4	1760
				10	3	1000
				11	5	1760
				5	1.5	600
				4	6	1935
				5.1	1.5	600
				8	4	1760
				7	5	1760
				6	6	1935
				Total	49	18310
3	1	1_16	3 (N)	3_17	4	750
				3_18	4	750
				4_21	8	300
				3_15	10	1875
				3_14	8	1500
				Total	34	5175
4	2	1_3	3 (S)	3_21	4	750
				3_20	4	750
				4_40	8	300
				3_23	10	1875
				3_24	8	1500
				Total	34	5175
5	5	2_17	4 (N)	4_5	9	960
				4_3	36	4200
				Total	45	5160
6	6	2_2	4 (S)	4_16	9	960
				4_1	36	4200
				Total	45	5160
7	3	1_17	4 (W)	4_2	48	4900
8	11+12	5 (N), 5 (S)	4 (E)	4_9	30	2140
				4_10	12	1220
				4_12	30	2140
				4_11	12	1220
				Total	84	6720
9	9+10	5 (N), 5 (S)	4 (C)	4_6	18	2800
				4_7	18	2800
				4_8	18	2800
				4_15	18	2800
				4_14	18	2800
				4_13	18	2800
				Total	108	16800
10	8	5 (S)	6	-	18	1760
				-	9	960
				-	1	160
				-	4	320
				-	4	320
				Total	36	3520
11	PA 1	Offices (N)	7_22 (N)		40	56320
12	PA 2	Offices (S)	7_22 (S)		40	56320

Table A-2 – Zone location, heater capacity and internal loads

A.3 Building floor plan

The following pages show the floor plan for the first nine floors of the Human Science building. The remaining upper office floors are all similar to the last floor shown.



GEESTESWETENSKAPPE
VLAK 1
4029

SKAAL 1 - 100 (A1)
97-05-05

REV & DAT	WYSIGING AMENDMENT
A	

GEESTESWETENSKAPPEGEBOU
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DEPARTMENT OF FACILITIES AND SERVICES

ELANDSPOORT 356 JR
GEDEELTE 399

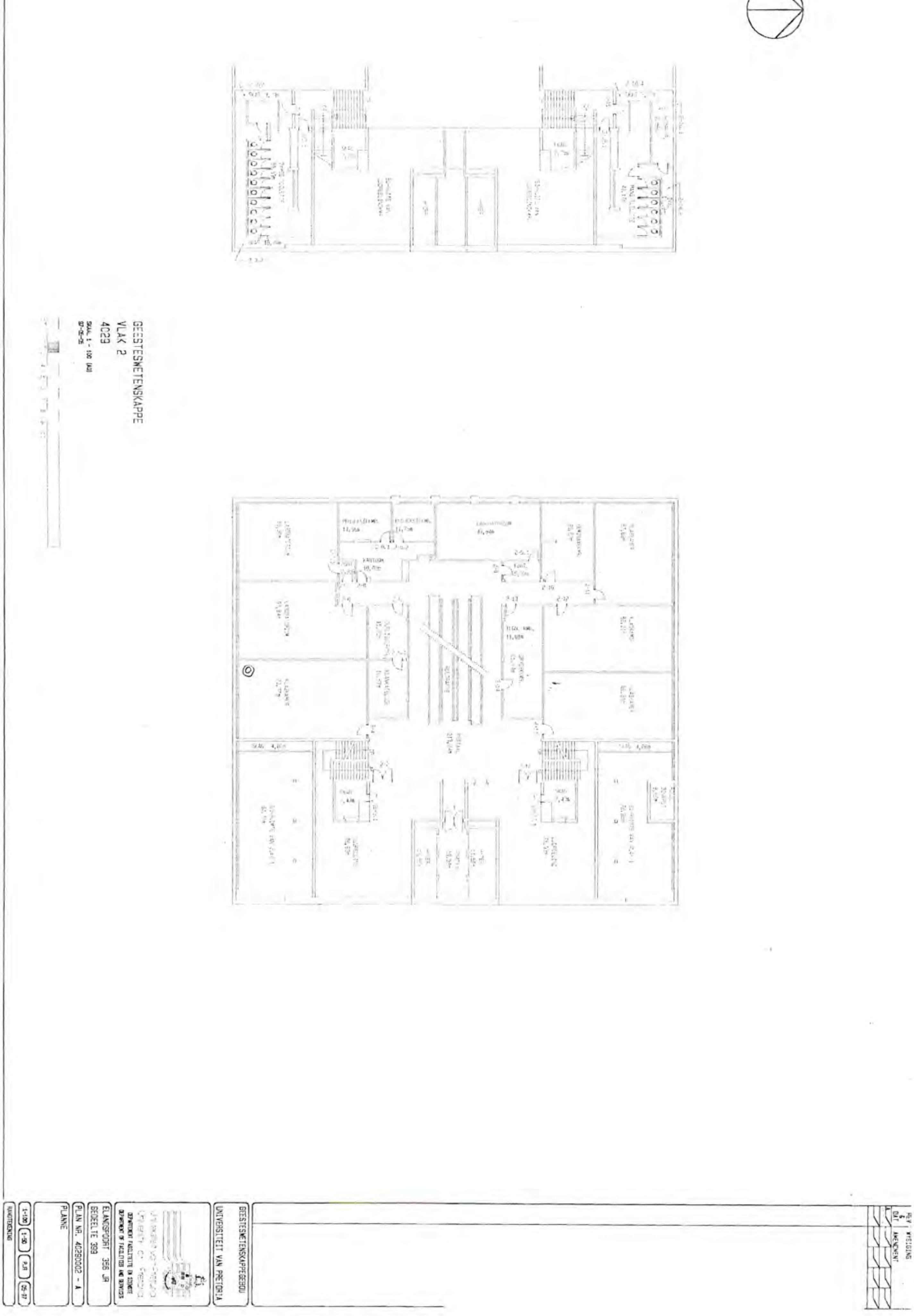
PLAN NR. 40290001 - A

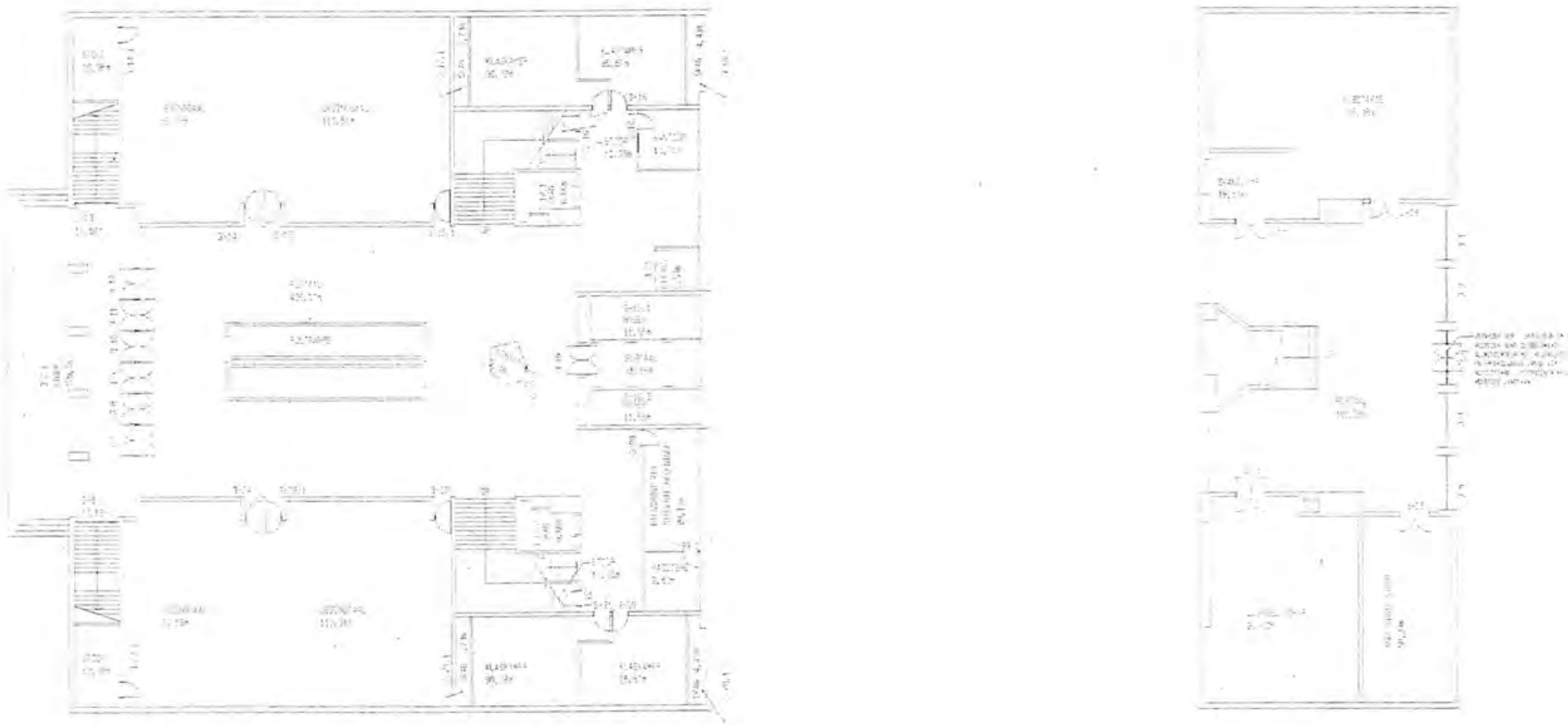
PLANNE

1-100 | 1-50 | PJR | 05-97

HANDTEKENING

UNIVERSITEIT VAN PRETORIA
UNIVERSITY OF PRETORIA
YUNIBESITHI YA PRETORIA





GEESTESWETENSKAPPE
VLAK 3
4029

SKAAL 1-100 (A0)
97-09-12

REV & DAT	WYSIGING AMENDMENT

GEESTESWETENSKAPPEGEBOU

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DEPARTMENT OF FACILITIES AND SERVICES

ELANDSPOORT 356 JR
GEDEELTE 399

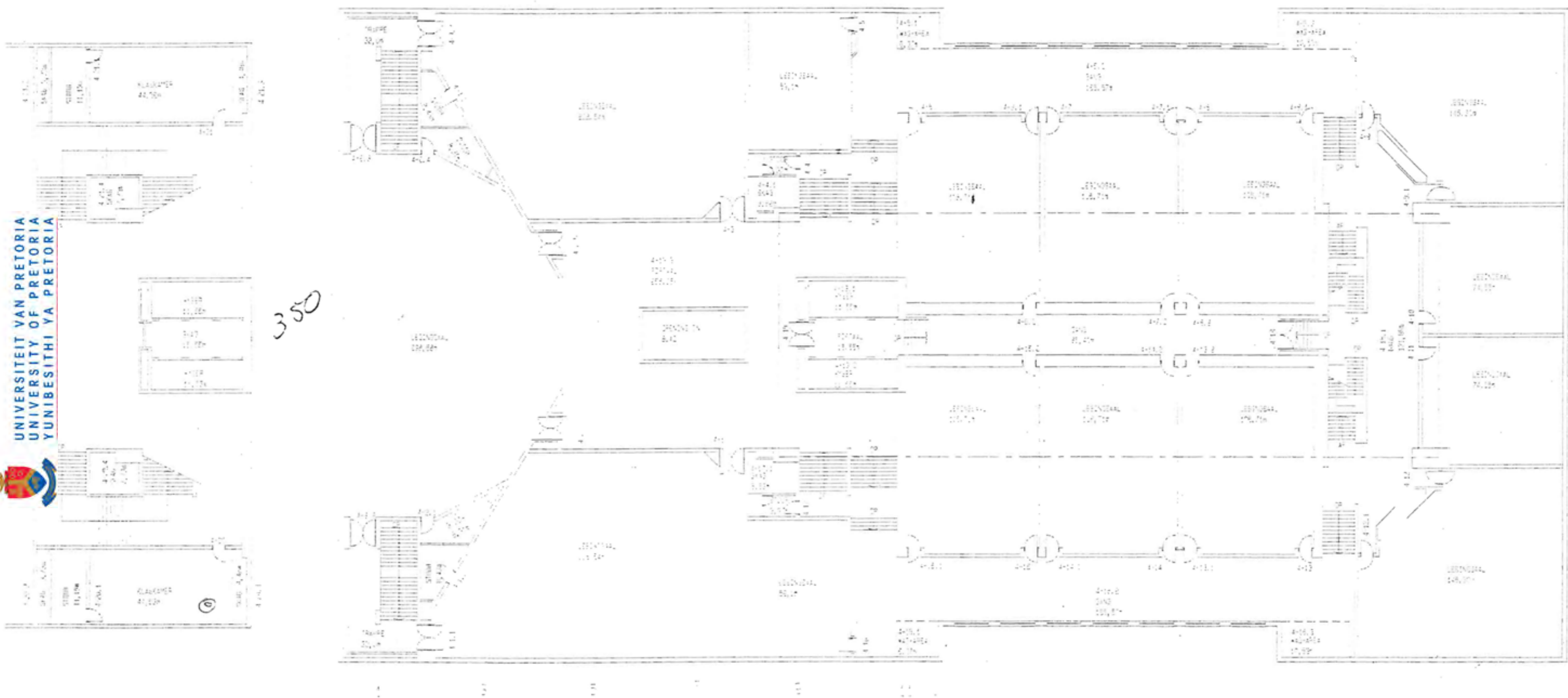
PLAN NR. 4029C003 - B

PLANNE

1-100 | 1-50 | PJP | 09-97

HANDTEKENING

350



GEESTESWETENSKAPPE
VLAK 4
4029

SKAAL 1 - 100
97-05-05

REV & DAT	WYSIGING AMENDMENT

GEESTESWETENSKAPPEGEBOU

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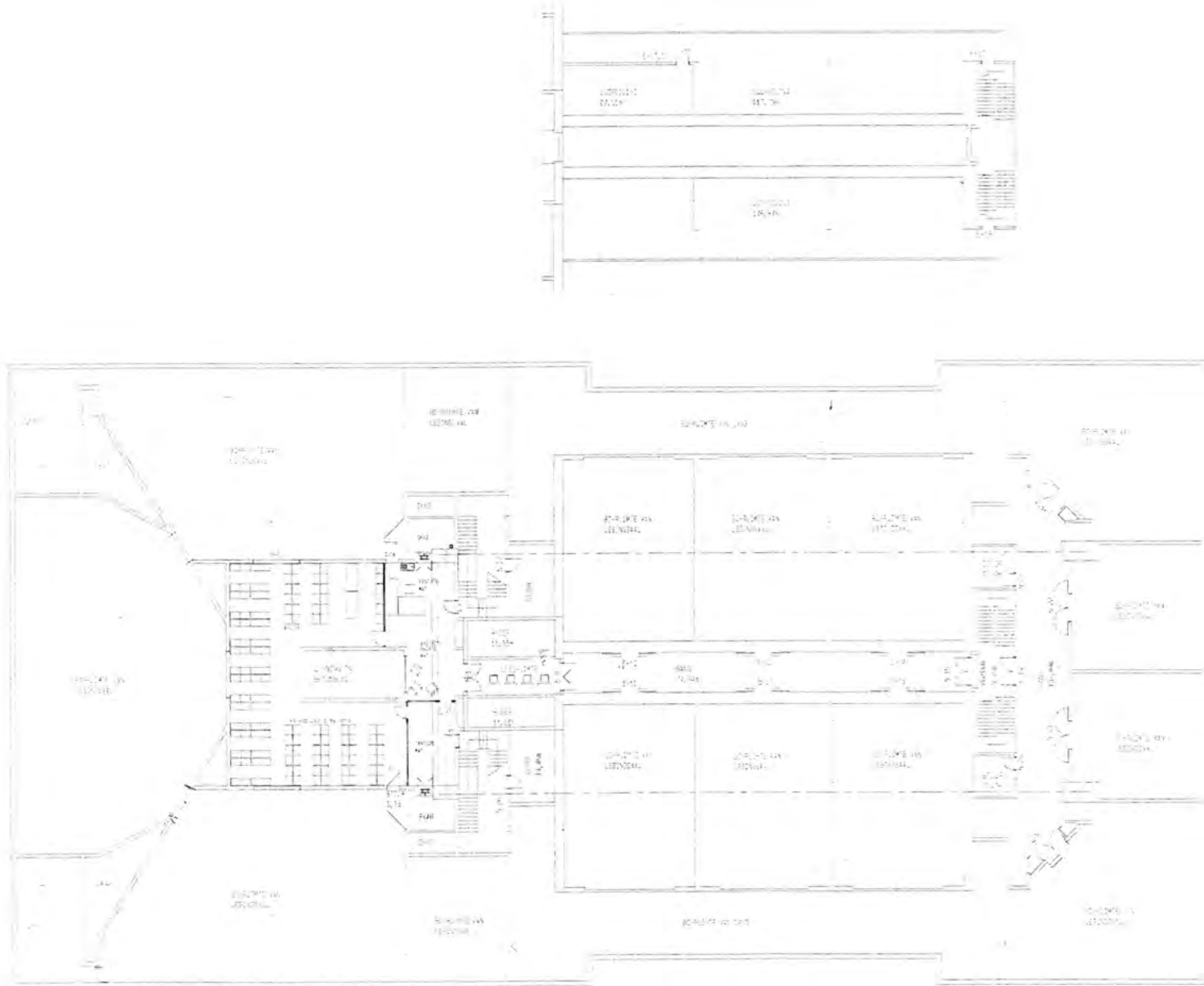
ELANDSPOORT 356 JR
GEDEELTE 399

PLAN NR. 40290005 - A

PLANNE

1-100 | 1-50 | PJR | 05-97

HANDTEKENING



GEESTESWETENSKAPPE
VLAK 5
4029
SKAAL 1 - 100 (A0)
97-09-12

REV & DAT	WYSIGING AMENDMENT

GEESTESWETENSKAPPEGEBOU

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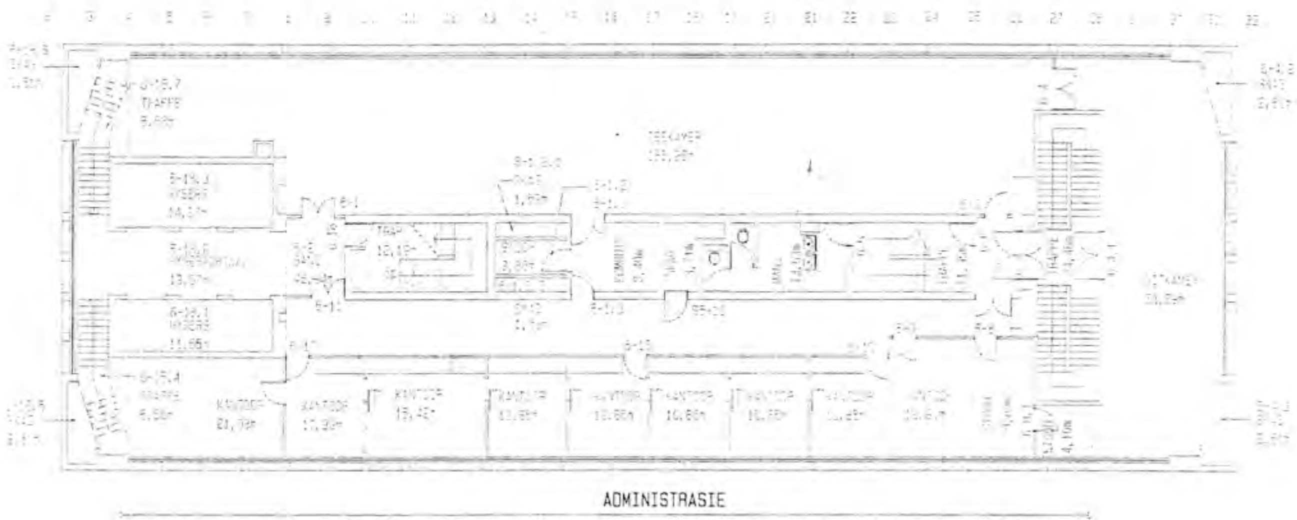
ELANDSPOORT 356 JR
GEDEELTE 399

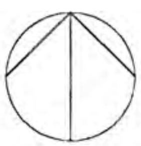
PLAN NR. 40290005 - B

PLANNE

1-100 1-50 PJR 09-97

HANDTEKENING





GEESTESWETENSKAPPE

VLAK 6

4029

SKAAL 1-100 (A1)
97-09-12

REV & DAT	WYSIGING AMENDMENT
A	

GEESTESWETENSKAPPEGEBOU

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GEDEELTE 399

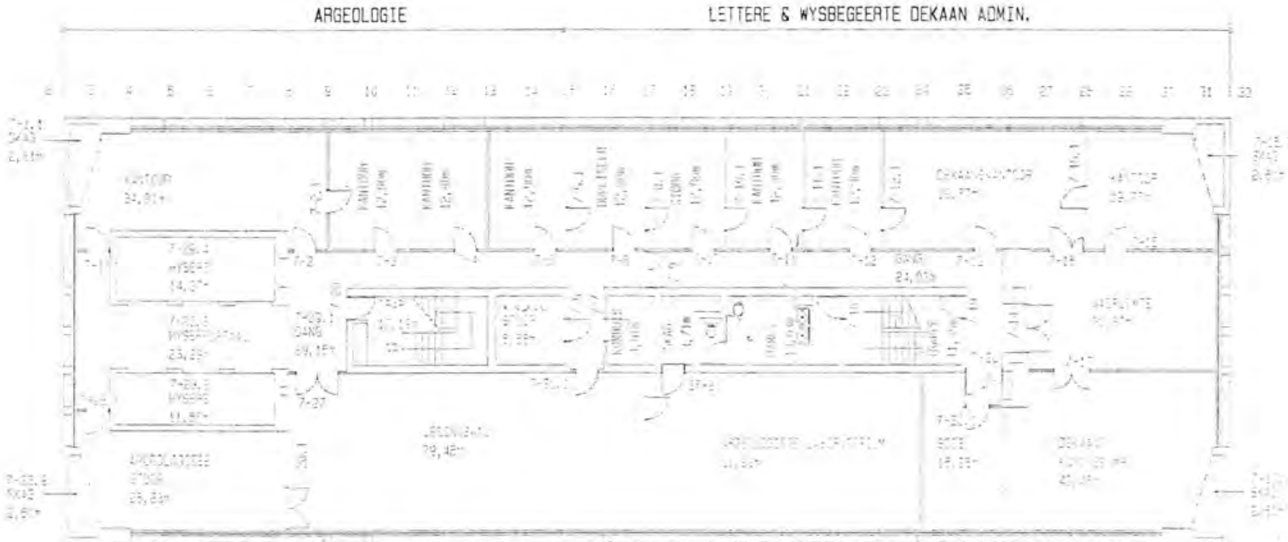
PLAN NR. 40290006 - A

PLANNE

1-100 | 1-50 | PJR | 05-97

HANDTEKENING

REV & DAT	WYSIGING AMENDMENT
A	



GEESTESWETENSKAPPE
VLAK 7
4029

SKAAL 1-100 [A1]
97-09-12

0 1 2 3 4 5 6 7 8 9 10

GEESTESWETENSKAPPEGEBOU

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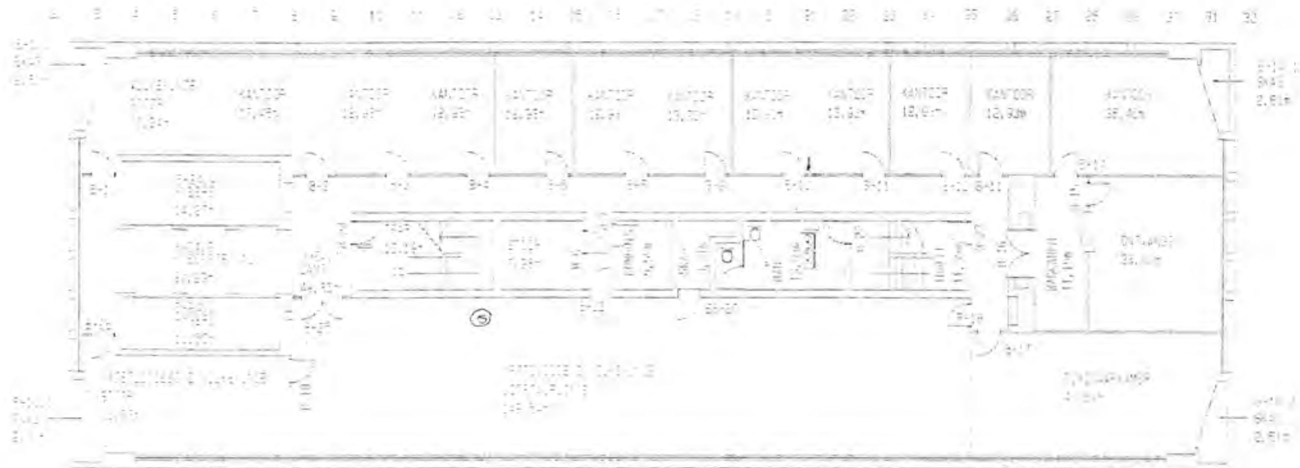
GEDEELTE 399

PLAN NR. 40290007 - B

PLANNE

1-100 | 1-50 | PJR | 06-97

HANDTEKENING



GEESTESWETENSKAPPE
VLAK 8
4029
SKAAL 1-100 (A2)
97-09-12

REV & DAT	WYSIGING AMENDMENT
A	

GEESTESWETENSKAPPEGEBOU
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ELANDSPOORT 356 JR
GEDEELTE 399

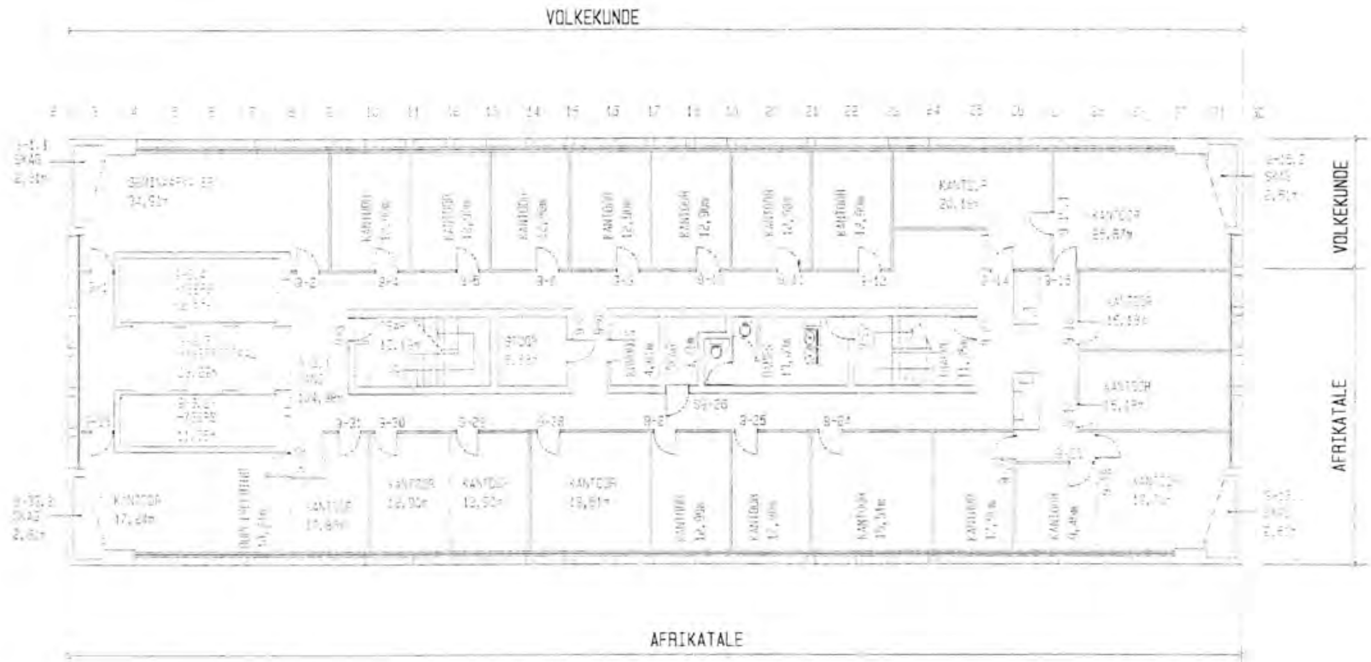
PLAN NR. 40290008 - 8

PLANNE

1-100 | 1-50 | PJR | 09-97

HANDTEKENING

REV & DAT	WYSIGING AMENDMENT
A	



GEESTESWETENSKAPPE
VLAK 9
4029

SKAAL 1-100 (A1)
97-09-12

GEESTESWETENSKAPPEGEBOU
UNIVERSITEIT VAN PRETORIA

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DEPARTMENT OF FACILITIES AND SERVICES

ELANDSPOORT 356 JR
GEDEELTE 399

PLAN NR. 40290009 - B

PLANNE

1-100 | 1-50 | PJR | 09-97

HANDTEKENING

Appendix B - Verification study

B.1 Supply temperature verification

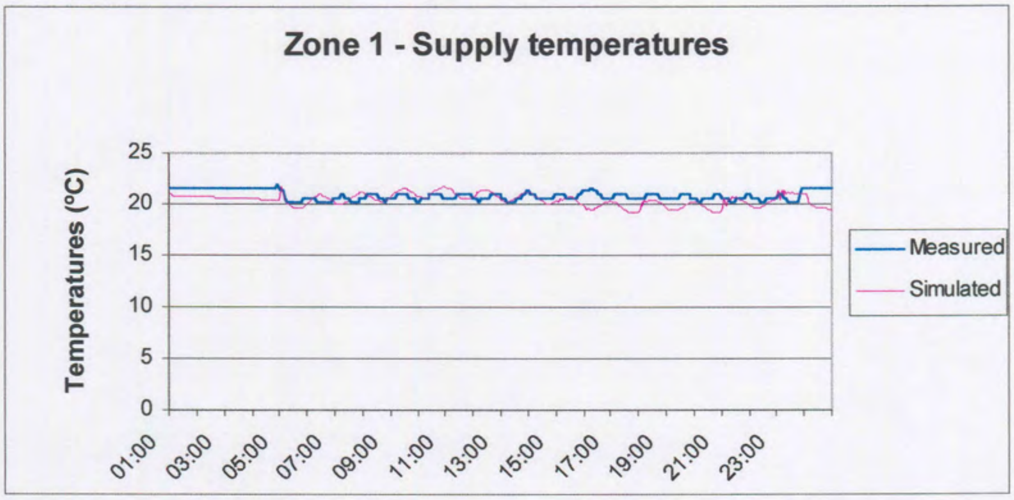

Figure B-1 – Zone 1 supply temperature verification

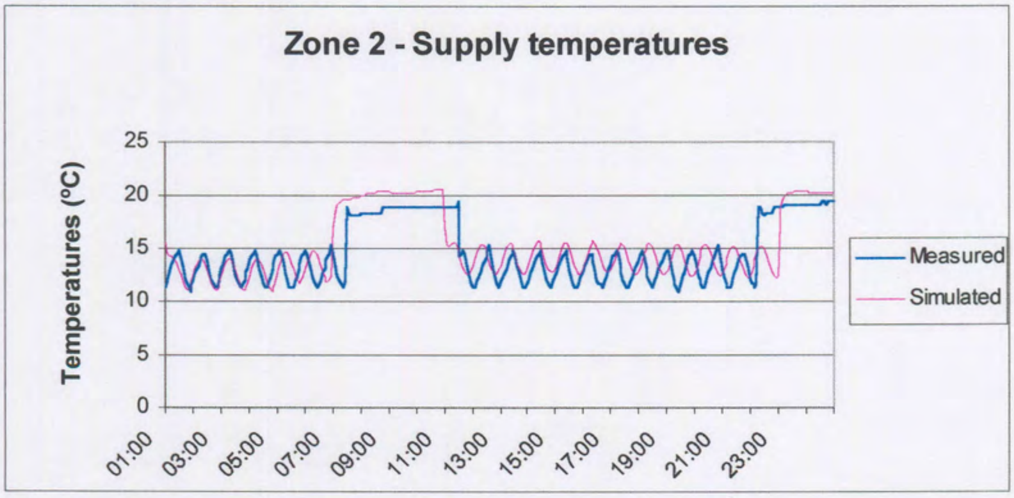

Figure B-2 – Zone 2 supply temperature verification

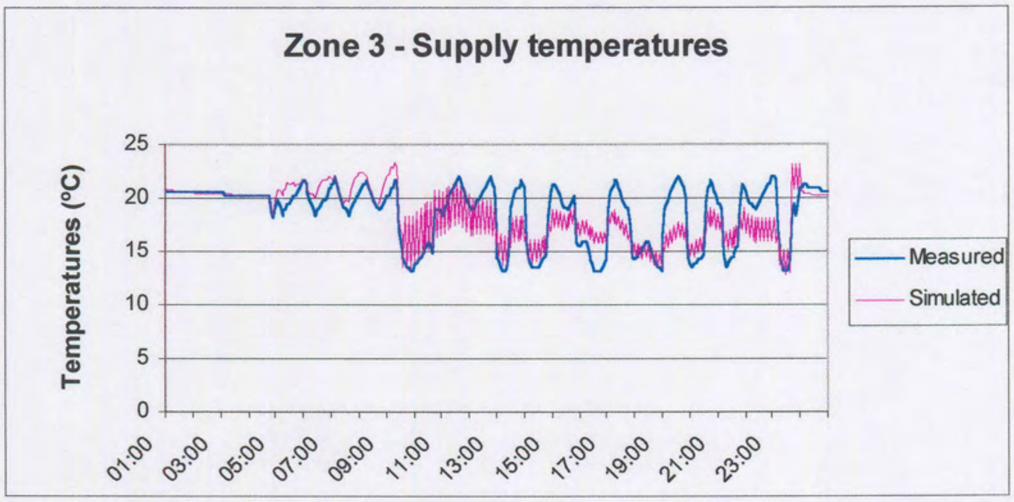

Figure B-3 – Zone 3 supply temperature verification

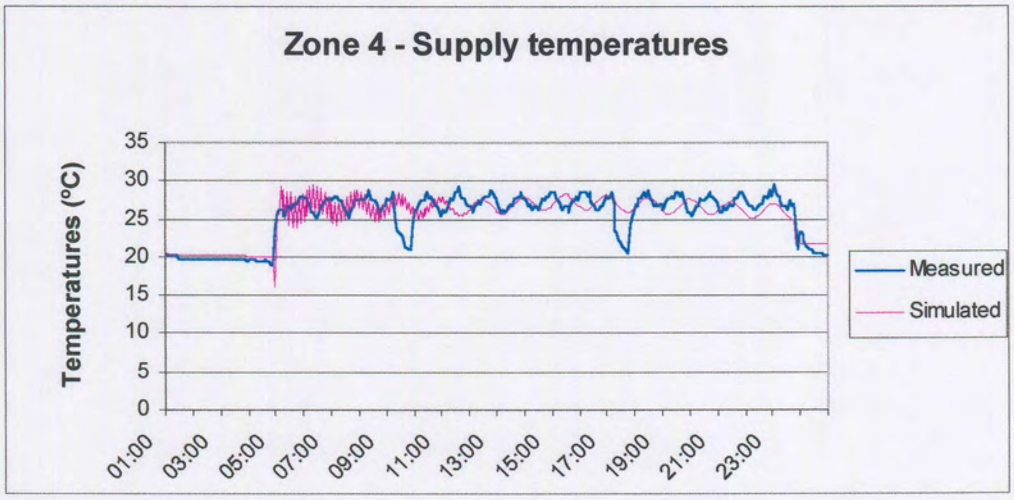

Figure B-4 – Zone 4 supply temperature verification

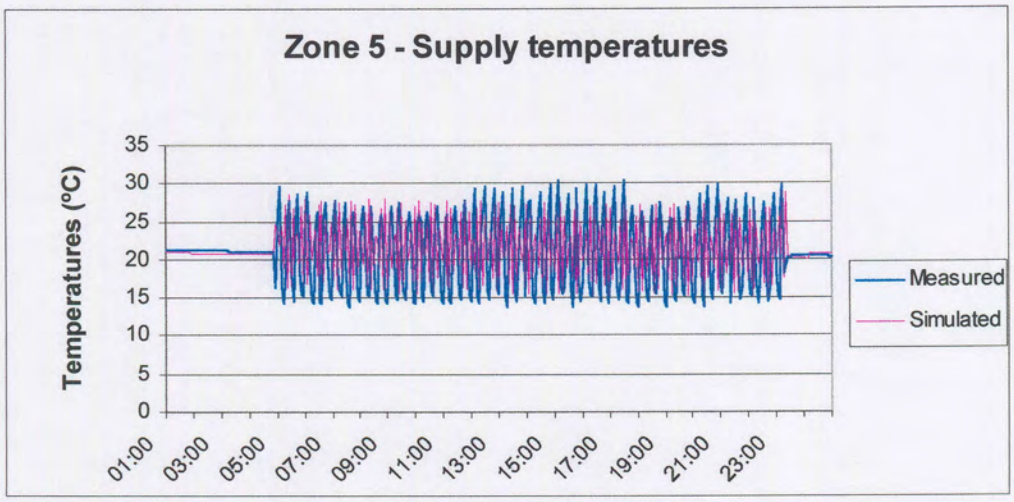

Figure B-5 – Zone 5 supply temperature verification

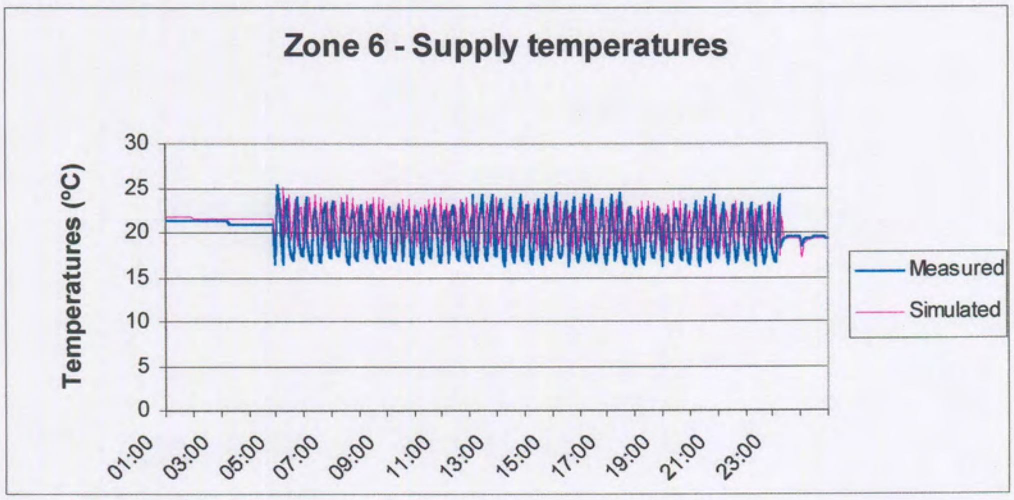

Figure B-6 – Zone 6 supply temperature verification

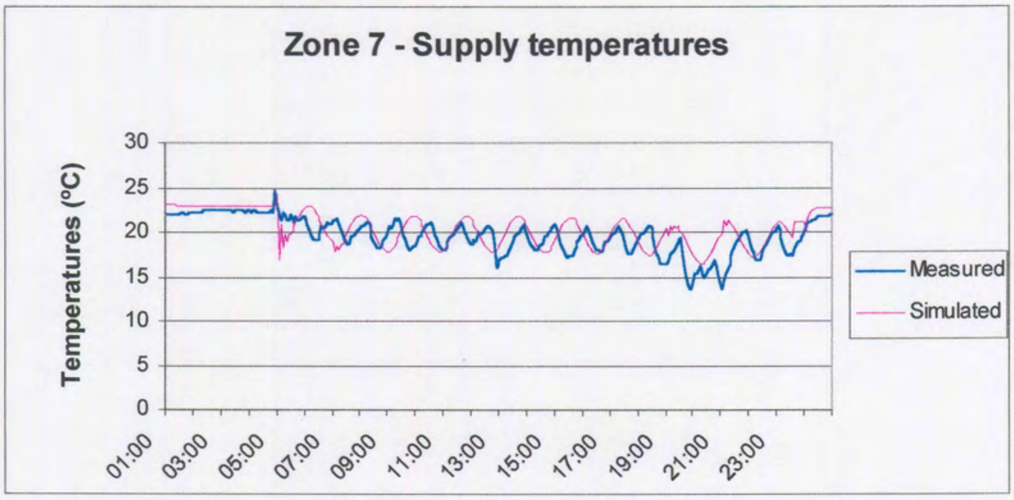

Figure B-7 – Zone 7 supply temperature verification

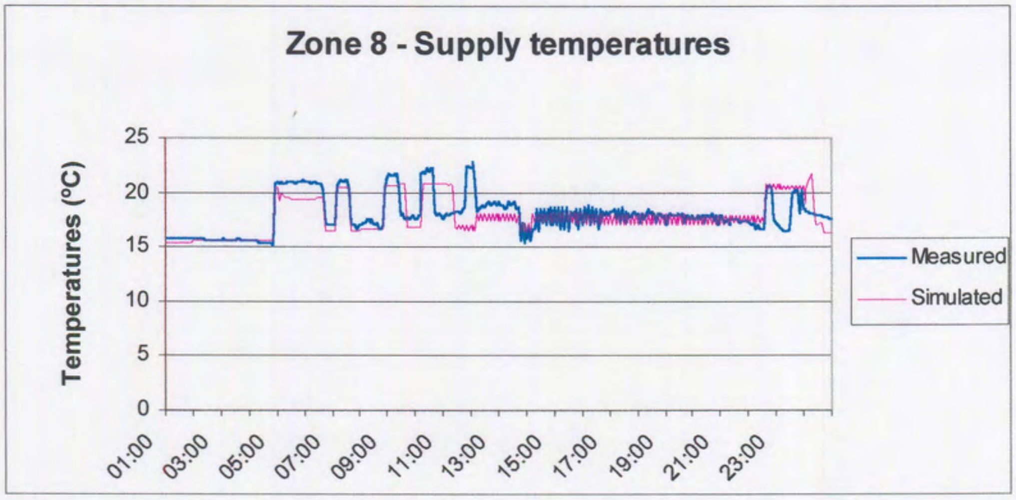

Figure B-8 – Zone 8 supply temperature verification

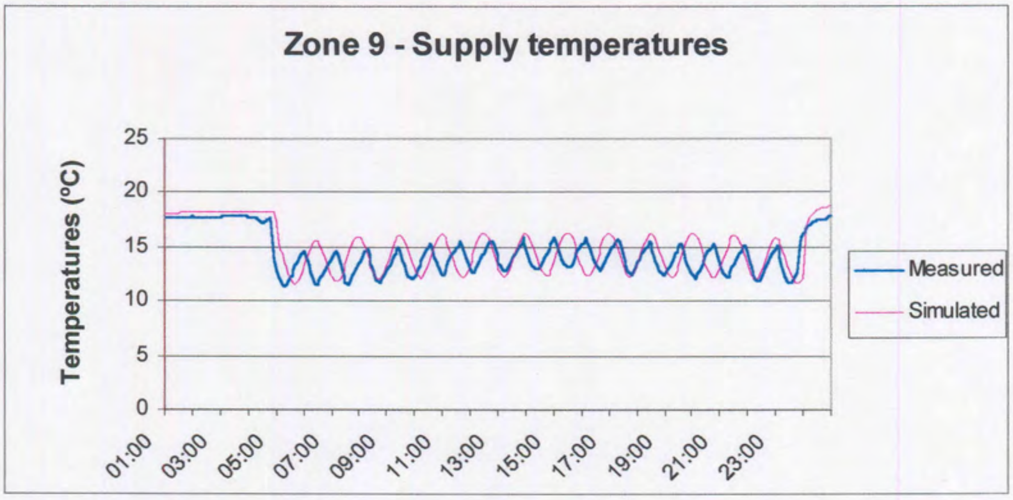

Figure B-9 – Zone 9 supply temperature verification

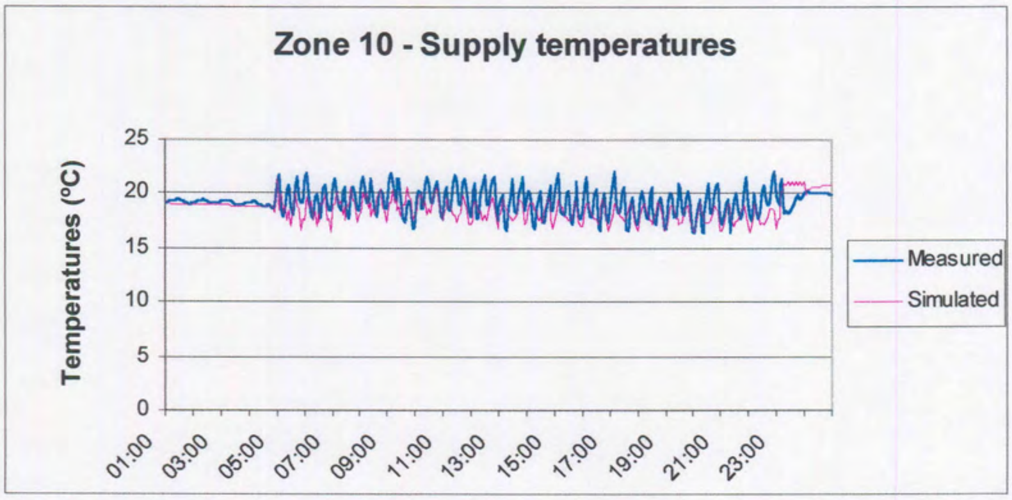

Figure B-10 – Zone 10 supply temperature verification

B.2 Return temperature verification

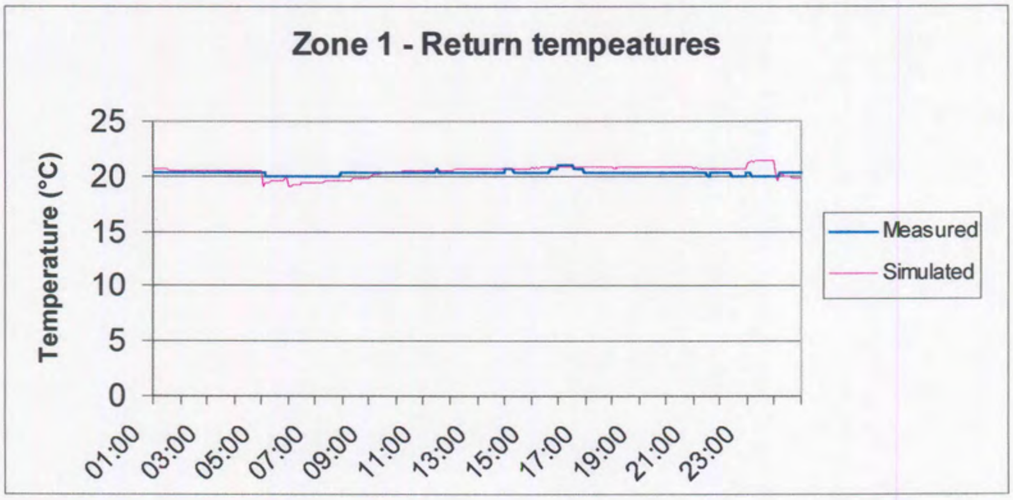

Figure B-11 – Zone 1 return temperature verification

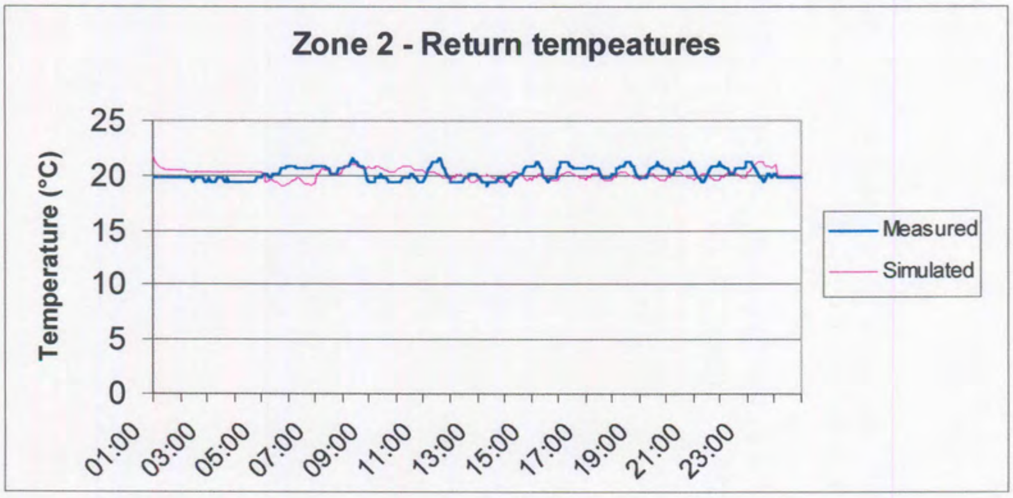

Figure B-12 – Zone 2 return temperature verification

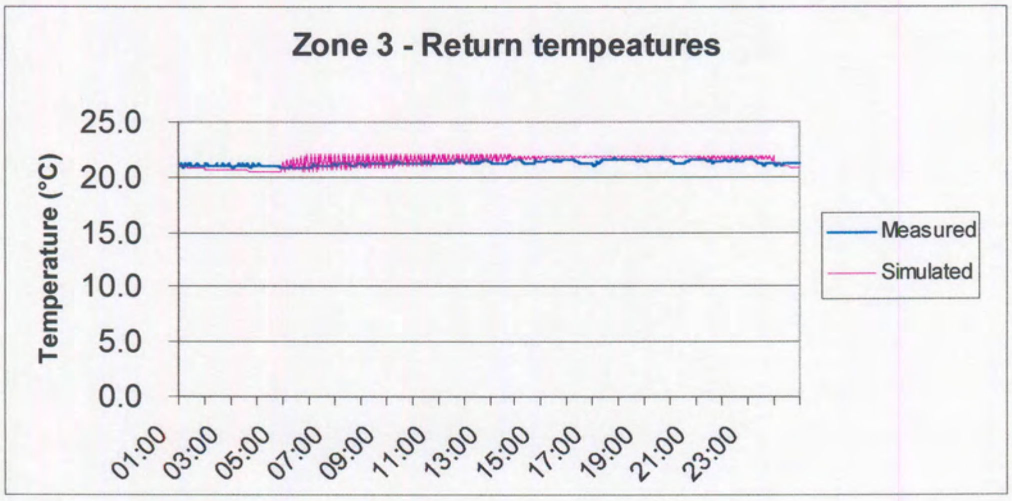

Figure B-13 – Zone 3 return temperature verification

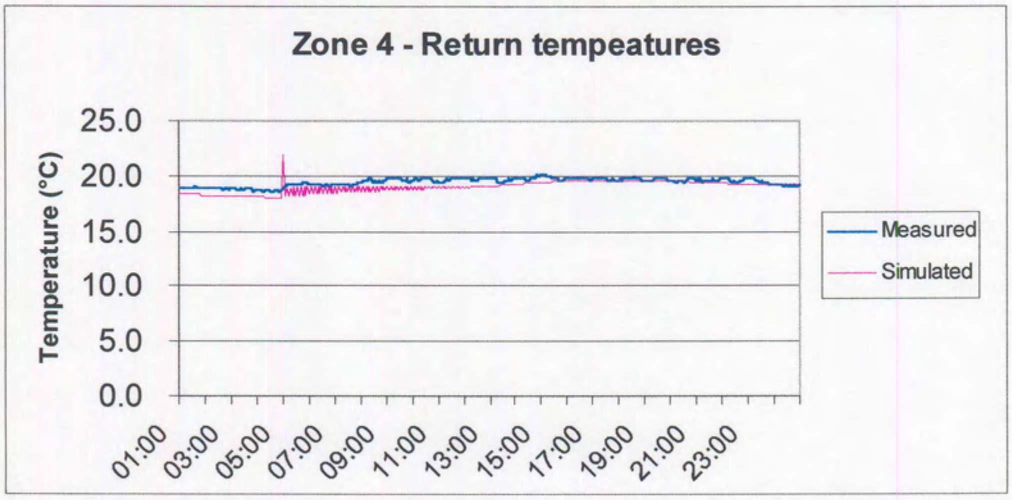

Figure B-14 – Zone 4 return temperature verification

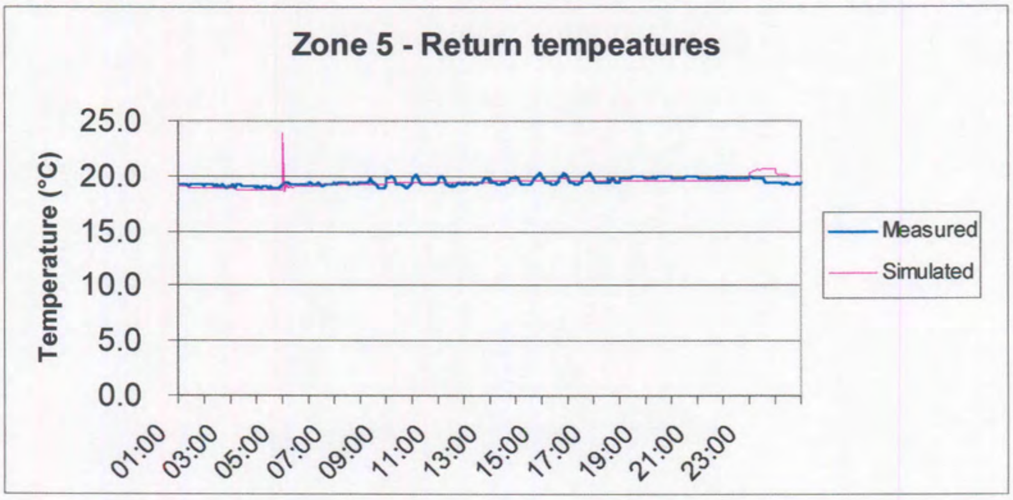

Figure B-15 – Zone 5 return temperature verification

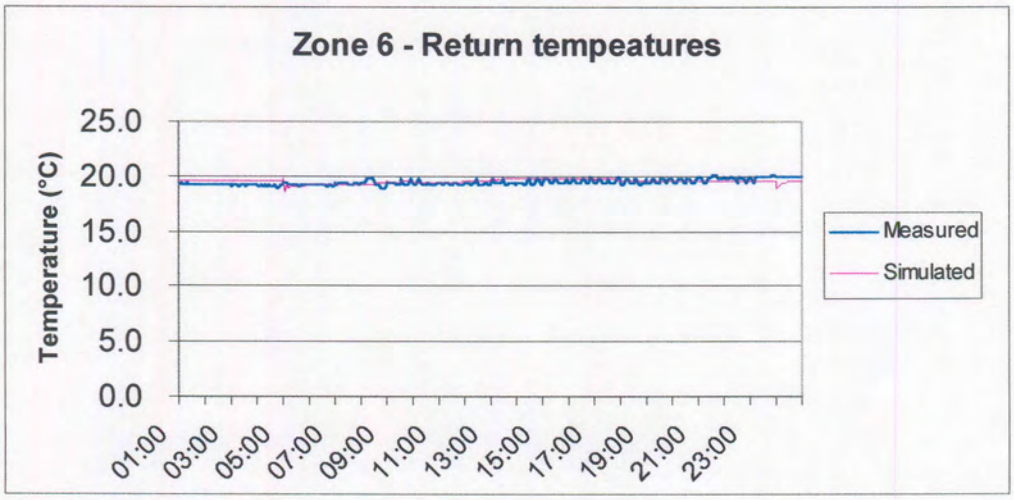

Figure B-16 – Zone 6 return temperature verification

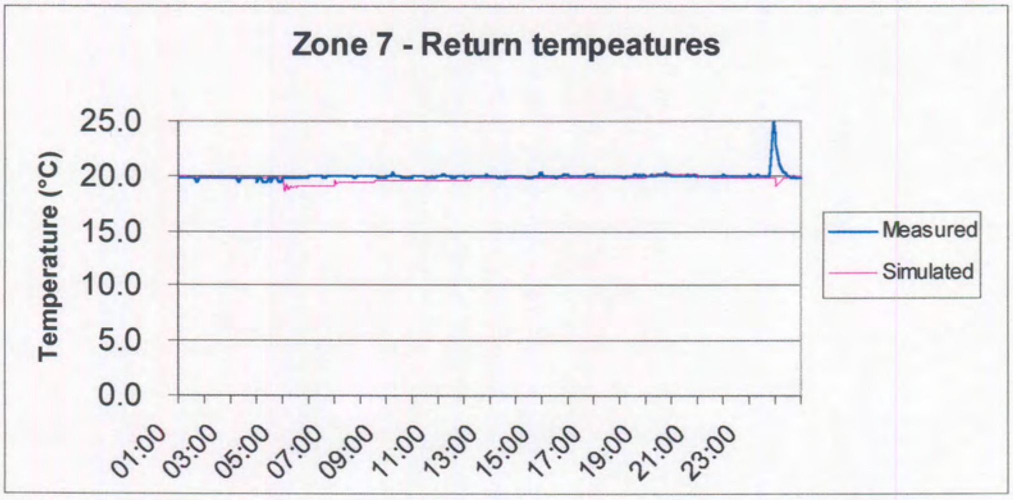

Figure B-17 – Zone 7 return temperature verification

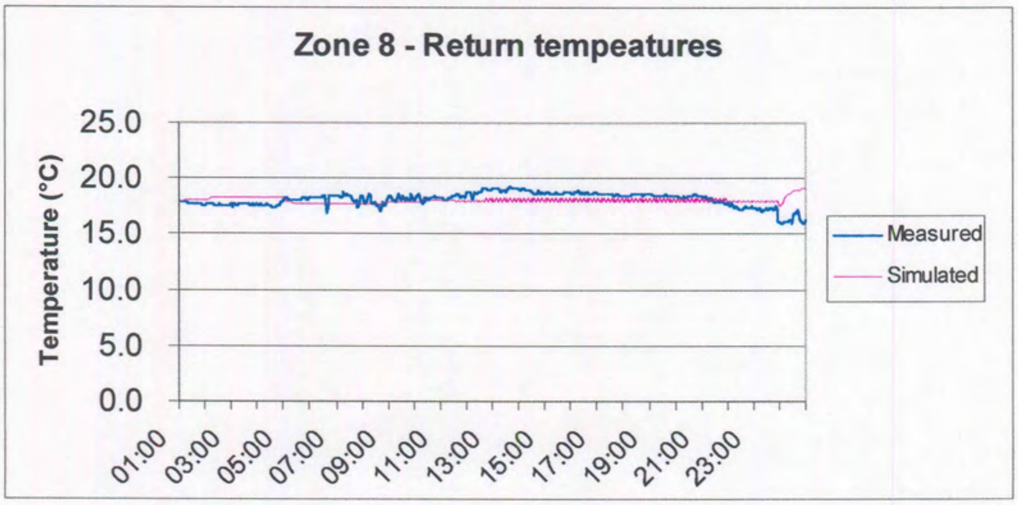

Figure B-18 – Zone 8 return temperature verification

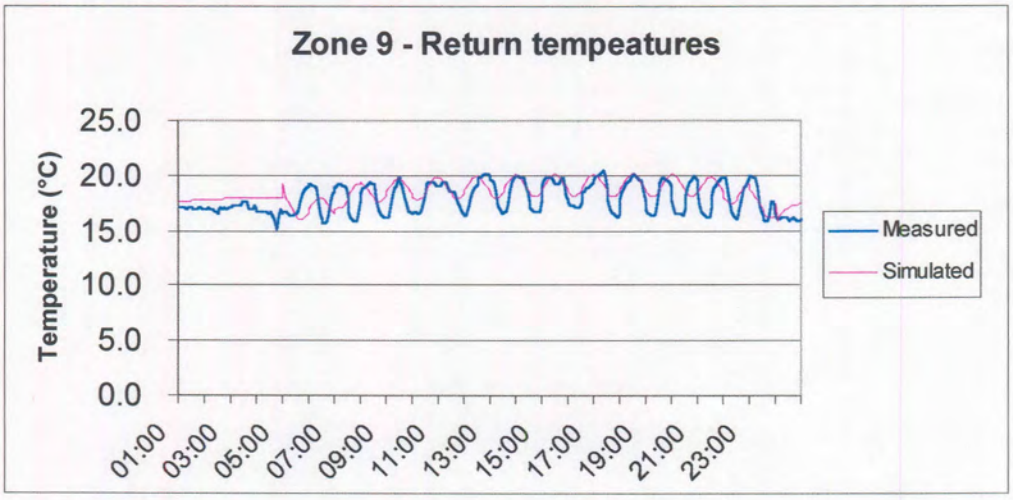

Figure B-19 – Zone 9 return temperature verification

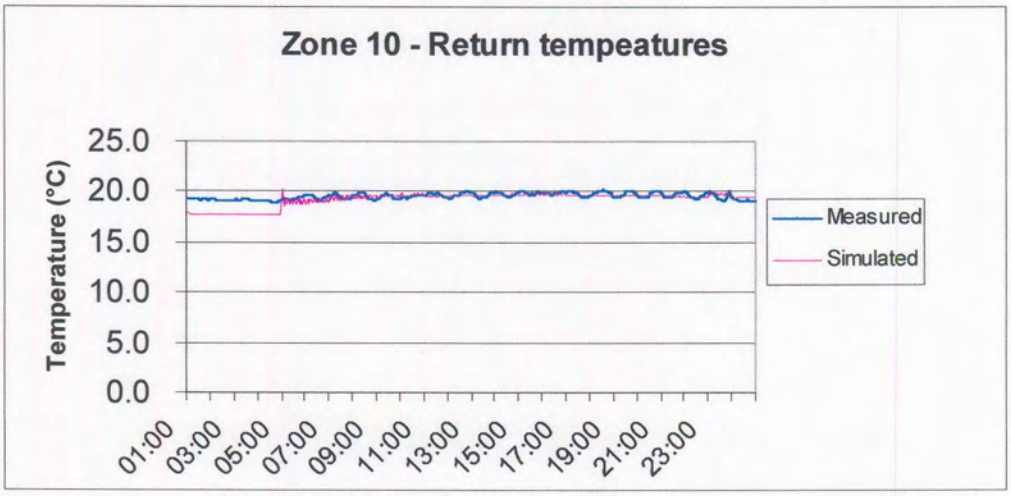

Figure B-20 – Zone 10 return temperature verification

B.3 Zone 13 verification

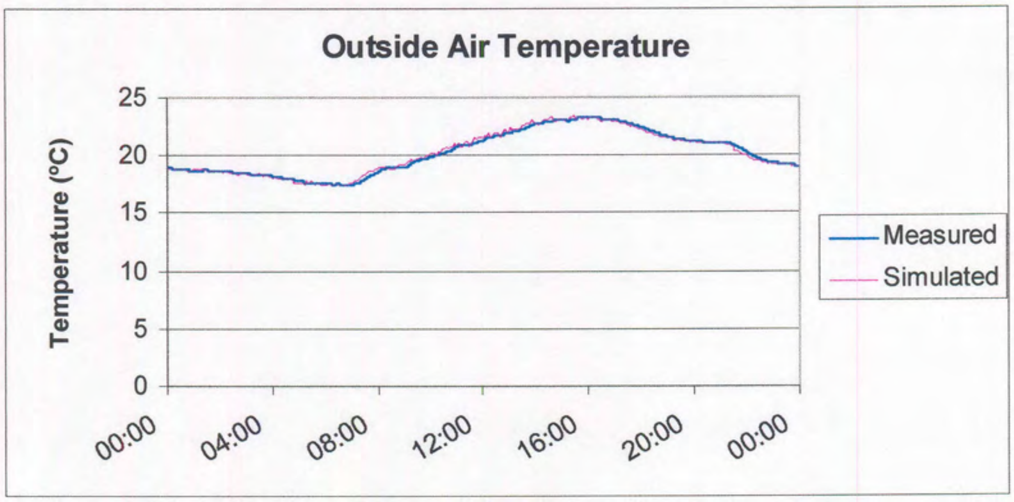

Figure B-21 – Zone 13 temperature verification

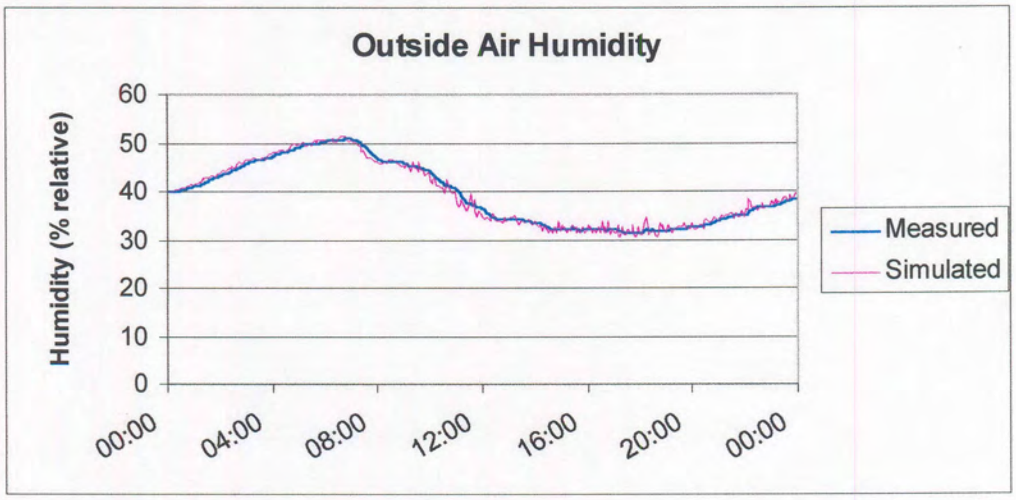

Figure B-22 – Zone 13 humidity verification

B.4 Multidimensional regressions

The regression coefficients were found with the aid of a spreadsheet program and a numerical solver utility.

Inputs				
N	800	rpm	rotational speed of the fan	
D	0.8	m	rotor diameter	
Q [m^3/s]	**P_{tot} [Pa]**	**$n_{eff.}$**	**K_h**	**K_f**
2	710	0.65	0.001548	0.004883
4	595	0.81	0.001297	0.009766
6	300	0.6	0.000654	0.014648

Output	
a0	0.001406
a1	0.069196
a2	-8.22857
b0	0.12
b1	146.432
b2	-7759.46

Table B-1 – Fan – Donkin BCC-2 800/1,0

Inputs						
N	1450	Rpm	rotational speed of the pump			
D	0.38	M	rotor diameter			
Q [m^3/h]	Q [l/s]	Head [m]	P_{tot} [Pa]	$n_{eff.}$	K_h	K_f
50	13.8889	47	461070	0.45	1.355956	0.628422
200	55.5556	42	412020	0.78	1.211705	2.513687
300	83.3333	28	274680	0.65	0.807804	3.770531

Output	
a0	1.280945
a1	0.168332
a2	-0.077924
b0	0.2
b1	0.4535108
b2	-0.088626

Table B-2 – Pump – KSB ETA 125-40

Mass flow through evaporator	Mass flow through condenser	Condenser leaving temperature	Evaporator leaving temperature	Condenser entering temperature	Evaporator entering temperature	Cooling capacity	Cooling capacity	Error	Compressor power consumption	Compressor power consumption	Error
58.3	37.5	25	5	20	10	597.5	602.473	24.7	96.9	94.98865951	3.65
		30	5	25	10	574	573.285	0.51	110.9	111.3390206	0.19
		32	5	27	10	563.4	561.61	3.21	117	117.879165	0.77
Regression Coefficients		35	5	30	10	546.7	544.097	6.78	126.8	127.6893816	0.79
		40	5	35	10	515.5	514.909	0.35	144.2	144.0397427	0.03
a0	0.148227	41	5	36	10	510.3	509.071	1.51	146.7	147.3098149	0.37
a1	4.7521949	42	5	37	10	504	503.233	0.59	150.2	150.5798871	0.14
a2	5.9080749	43	5	38	10	497.3	497.396	0.01	153.6	153.8499593	0.06
a3	-5.8376215	44	5	39	10	491.5	491.558	0	157.1	157.1200315	0
a4	19.6429	45	5	40	10	485.3	485.72	0.18	160.6	160.3901037	0.04
		46	5	41	10	479.1	479.883	0.61	164.1	163.6601759	0.19
Total error	267.02025	25	6	20	11	617.4	622.116	22.2	96.9	95.00556432	3.59
b0	0.0061264	32	6	27	11	583.3	581.252	4.19	117	117.8960698	0.8
b1	0.229639	35	6	30	11	566.4	563.74	7.08	126.7	127.7062864	1.01
b2	0.3567852	40	6	35	11	535.1	534.551	0.3	144.2	144.0566475	0.02
b3	3.2700722	41	6	36	11	529.8	528.714	1.18	146.7	147.3267197	0.39
b4	0.0169048	42	6	37	11	523.6	522.876	0.52	150.2	150.5967919	0.16
		43	6	38	11	517.3	517.039	0.07	153.7	153.8668641	0.03
Total error	46.774099	44	6	39	11	511.1	511.201	0.01	157.2	157.1369363	0
		45	6	40	11	504.6	505.363	0.32	160.7	160.4070085	0.09
		46	6	41	11	498.5	499.526	1.05	164.2	163.6770807	0.27
		25	7	20	12	637.2	641.759	20.8	96.9	95.02246913	3.53
		30	7	25	12	613.8	612.571	1.51	110.9	111.3728302	0.22
		32	7	27	12	603.1	600.895	4.86	117	117.9129746	0.83
		35	7	30	12	586.1	583.382	7.39	126.7	127.7231912	1.05
		40	7	35	12	554.7	554.194	0.26	144.2	144.0735523	0.02
		41	7	36	12	549.4	548.357	1.09	146.7	147.3436245	0.41
		42	7	37	12	543.1	542.519	0.34	150.2	150.6136967	0.17
		43	7	38	12	536.9	536.681	0.05	153.7	153.8837689	0.03
		44	7	39	12	530.6	530.844	0.06	157.2	157.1538411	0
		45	7	40	12	524.3	525.006	0.5	160.7	160.4239133	0.08
		46	7	41	12	518	519.169	1.37	164.2	163.6939855	0.26
		25	8	20	13	657.1	661.402	18.5	96.9	95.03937394	3.46
		30	8	25	13	633.6	632.213	1.92	110.9	111.389735	0.24
		32	8	27	13	623	620.538	6.06	116.9	117.9298794	1.06
		35	8	30	13	605.8	603.025	7.7	126.6	127.740096	1.3
		40	8	35	13	574.2	573.837	0.13	144.2	144.0904571	0.01
		41	8	36	13	568.9	568	0.81	146.7	147.3605293	0.44
		42	8	37	13	562.6	562.162	0.19	150.2	150.6306015	0.19
		43	8	38	13	556.2	556.324	0.02	153.8	153.9006737	0.01
		44	8	39	13	549.9	550.487	0.34	157.3	157.1707459	0.02
		45	8	40	13	543.6	544.649	1.1	160.8	160.4408181	0.13
		46	8	41	13	537.3	538.812	2.28	164.3	163.7108903	0.35
		25	9	20	14	676.3	681.044	22.5	96.9	95.05627875	3.4
		30	9	25	14	653.5	651.856	2.7	110.8	111.4066398	0.37
		32	9	27	14	642.9	640.181	7.39	116.8	117.9467842	1.32
		35	9	30	14	625.8	622.668	9.81	126.5	127.7570008	1.58
		40	9	35	14	594	593.48	0.27	144.2	144.1073619	0.01
		41	9	36	14	588.6	587.643	0.92	146.7	147.3774341	0.46
		42	9	37	14	582.3	581.805	0.25	150.3	150.6475063	0.12
		43	9	38	14	575.9	575.967	0	153.8	153.9175785	0.01
		44	9	39	14	569.6	570.13	0.28	157.4	157.1876507	0.05
		45	9	40	14	563.2	564.292	1.19	160.9	160.4577229	0.2
		46	9	41	14	556.8	558.454	2.74	164.4	163.7277951	0.45
		25	10	20	15	696.5	700.687	17.5	96.9	95.07318356	3.34
		30	10	25	15	673.3	671.499	3.24	110.8	111.4235446	0.39
		32	10	27	15	662.3	659.824	6.13	116.8	117.963689	1.35
		35	10	30	15	645.9	642.311	12.9	126.5	127.7739057	1.62
		40	10	35	15	613.9	613.123	0.6	144.2	144.1242667	0.01
		41	10	36	15	608.5	607.285	1.48	146.7	147.3943389	0.48
		42	10	37	15	602.1	601.448	0.43	150.3	150.6644111	0.13
		43	10	38	15	595.7	595.61	0.01	153.8	153.9344833	0.02
		44	10	39	15	589.3	589.773	0.22	157.4	157.2045555	0.04
		45	10	40	15	582.9	583.935	1.07	160.9	160.4746277	0.18
		46	10	42	15	570.1	572.26	4.66	168	167.0147722	0.97

Table B-3 – Chiller – Trane RTHA 215

Appendix C - Retrofit study

C.1 Retrofit cost estimation summary

The following table is identical to table 4-2. It shows the total cost of each of the retrofit options considered with the comparative payback periods. The savings are discussed in more detail in Chapter 4.

Retrofit option	Running cost (R/year)	Saving (R/year)	Implementation cost (R)	Payback period (months)
Current system	295 546	0	1 600	-
Air bypass control	281 998	13 548	135 900	120.4
Reset control	161 801	133 745	137 500	12.3
Setback control	119 218	176 328	145 000	9.9
Time management	100 269	195 277	146 600	9.0
Economiser control	100 355	195 191	204 500	12.6
CO_2 control	92 326	203 220	379 500	22.4

Table C-1 – Economic comparison of retrofit options

C.2 Current system update

To find a suitable basis for evaluation, the current HVAC system has to be upgraded to supply acceptable indoor air conditions in all the zones. No energy will be saved during this retrofit, only control setpoints will be adjusted at typical labour cost.

Component	Amount	Unit cost	Total cost
Labour hours	8	R 200	R 1 600

Table C-2 – Cost to update the current HVAC system

C.3 Air bypass

For the air bypass retrofit some structural modifications will have to be made to the building. The cost of the digital control system along with the installation will amount to almost R 90 000.

Component	Amount	Cost per unit	Price
Zone sensors	50	R 120	R 6 000
Supply duct sensors	12	R 104	R 1 248
Pneumatic converters	18	R 570	R 10 260
Relay controllers for the heaters	167	R 12	R 2 004
Display unit	2	R 4 875	R 9 750
Programming	1	R 9 000	R 9 000
Installation	1	R 60 000	R 60 000
Total			**R 135 900**

Table C-3 – Cost of the air bypass system

C.4 Reset control

To implement the reset control retrofit only eight hours of labour cost will be added to the installation. The consultant will have to install the control system to incorporate feedback from the zone temperature sensors and not the supply duct sensors.

Component	Amount	Cost per unit	Price
Zone sensors	50	R 120	R 6 000
Supply duct sensors	12	R 104	R 1 248
Pneumatic converters	18	R 570	R 10 260
Relay controllers for the heaters	167	R 12	R 2 004
Display unit	2	R 4 875	R 9 750
Programming	1	R 9 000	R 9 000
Installation	1	R 60 000	R 60 000
Labour hours	8	R 200	R 1 600
Total			**R 137 500**

Table C-4 – Cost of the reset control retrofit

C.5 Setback control

The cost of the setback control retrofit is very similar to that of the reset control, except for the motion detector input. At least one of each of these units will have to be installed in every lecture hall.

Component	Amount	Cost per unit	Price
Zone sensors	50	R 120	R 6 000
Supply duct sensors	12	R 104	R 1 248
Pneumatic converters	18	R 570	R 10 260
Relay controllers for the heaters	167	R 12	R 2 004
Display unit	2	R 4 875	R 9 750
Programming	1	R 9 000	R 9 000
Installation	1	R 60 000	R 60 000
Labour hours	8	R 200	R 1 600
Motion detectors	50	R 150	R 7 500
Total			**R 145 000**

Table C-5 – Cost of the setback control retrofit

C.6 Time management

Like with the "current system" retrofit only eight hours of labour cost is added to the retrofit cost to incorporate the improved time management retrofit.

Component	Amount	Cost per unit	Price
Zone sensors	50	R 120	R 6 000
Supply duct sensors	12	R 104	R 1 248
Pneumatic converters	18	R 570	R 10 260
Relay controllers for the heaters	167	R 12	R 2 004
Display unit	2	R 4 875	R 9 750
Programming	1	R 9 000	R 9 000
Installation	1	R 60 000	R 60 000
Labour hours	16	R 200	R 3 200
Motion detectors	50	R 150	R 7 500
Total			**R 146 600**

Table C-6 – Cost of the time management control retrofit

C.7 Economiser control

To install the economiser control system some structural modifications will have to be made to the building in order to regulate the volume of airflow through the intake ducts. New damper units with controlled actuators will have to be installed on each of the AHUs. The labour cost of the installation should also be taken into account.

Component	Amount	Cost per unit	Price
Zone sensors	50	R 120	R 6 000
Supply duct sensors	12	R 104	R 1 248
Pneumatic converters	18	R 570	R 10 260
Relay controllers for the heaters	167	R 12	R 2 004
Display unit	2	R 4 875	R 9 750
Programming	1	R 9 000	R 9 000
Installation	1	R 60 000	R 60 000
Adjustment Labour hours	16	R 200	R 3 200
Installation Labour hours	6	R 1 600	R 9 600
Motion detectors	50	R 150	R 7 500
Damper units and actuators	12	R 4 000	R 48 000
Economiser temperature sensors	3	R 100	R 300
Total			**204 500**

Table C-7 – Cost of the economiser control retrofit

C.8 CO_2 control

CO_2 control utilises the economiser control system to regulate the fresh air flow rate into the building. The only extra cost is that of the CO_2 sensors. These units are quite expensive, so the final installation cost of the CO_2 control system is very high.

Component	Amount	Cost per unit	Price
Zone sensors	50	R 120	R 6 000
Supply duct sensors	12	R 104	R 1 248
Pneumatic converters	18	R 570	R 10 260
Relay controllers for the heaters	167	R 12	R 2 004
Display unit	2	R 4 875	R 9 750
Programming	1	R 9 000	R 9 000
Installation	1	R 60 000	R 60 000
Adjustment Labour hours	16	R 200	R 3 200
Installation Labour hours	6	R 1 600	R 9 600
Motion detectors	50	R 150	R 7 500
Damper units and actuators	12	R 4 000	R 48 000
Economiser temperature sensors	3	R 100	R 300
CO_2 sensors	50	R 3 500	R 175 000
Total			**R 379 500**

Table C-8 – Cost of the CO_2 control retrofit

Appendix D - Effect of equipment failure

D.1 Zone temperatures after coil fouling

The temperatures show is the simulation results obtained from a typical week day simulation. The weather data used for the simulation was the average outdoor temperatures and humidities for Pretoria.

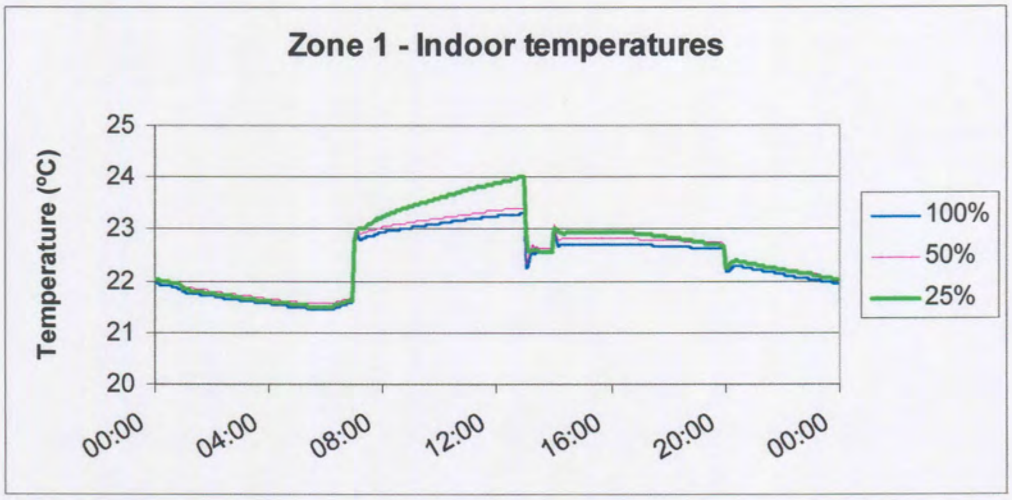

Figure D-1 – Simulation temperatures at different coil efficiencies for zone 1

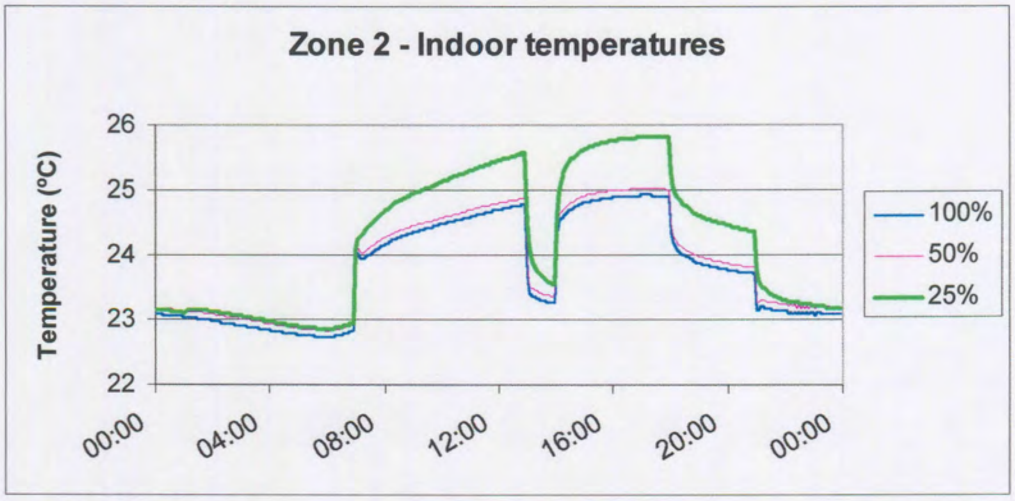

Figure D-2 – Simulation temperatures at different coil efficiencies for zone 2

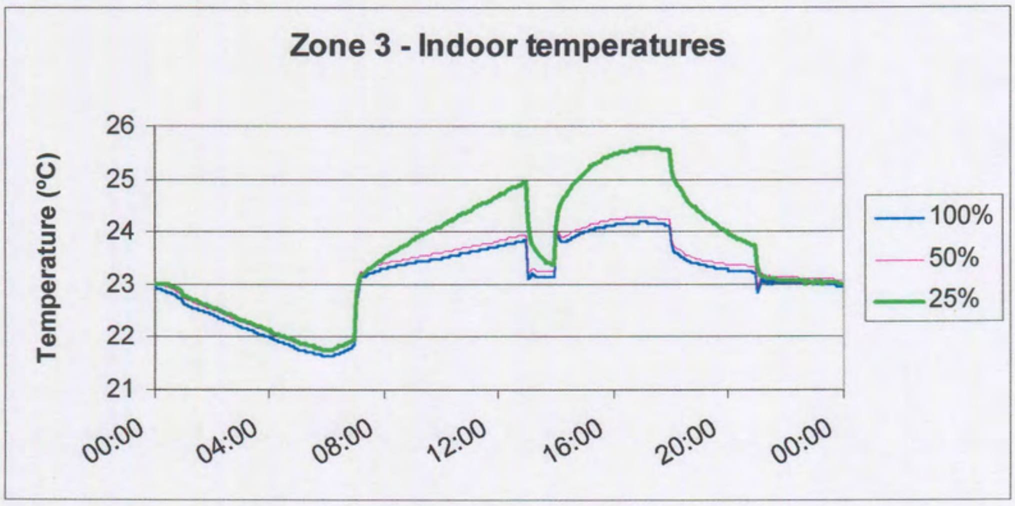

Figure D-3 – Simulation temperatures at different coil efficiencies for zone 3

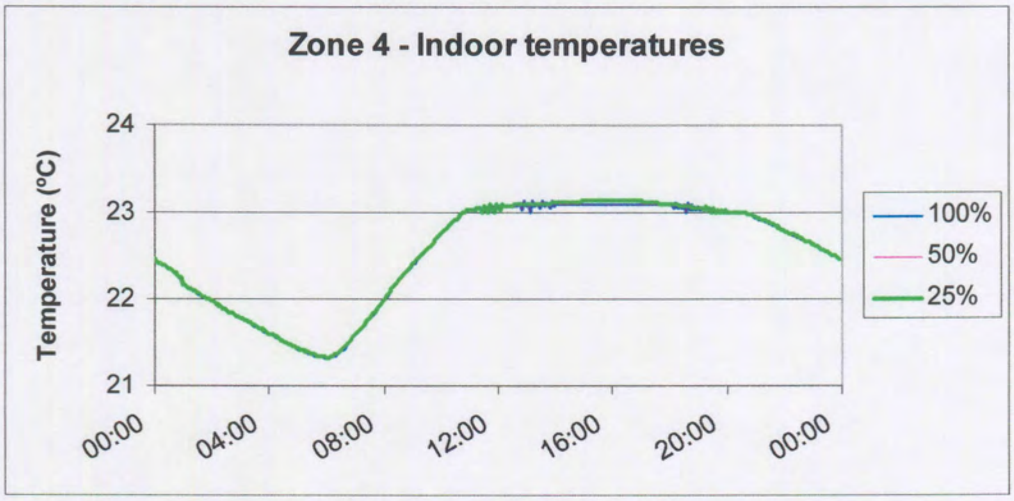

Figure D-4 – Simulation temperatures at different coil efficiencies for zone 4

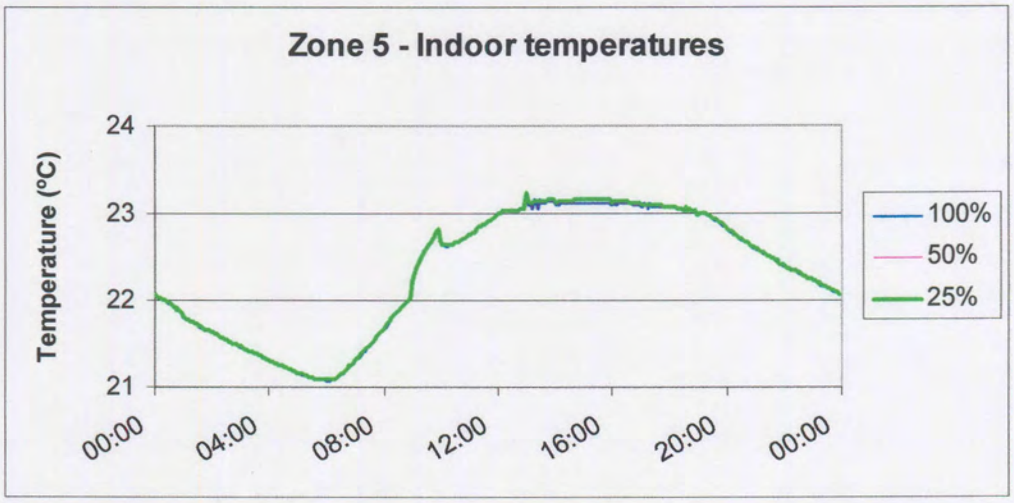

Figure D-5 – Simulation temperatures at different coil efficiencies for zone 5

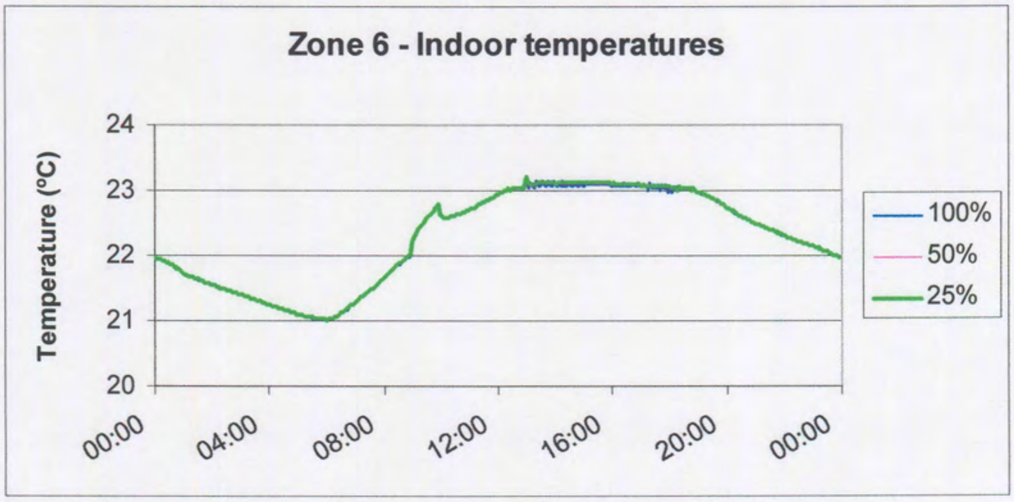

Figure D-6 – Simulation temperatures at different coil efficiencies for zone 6

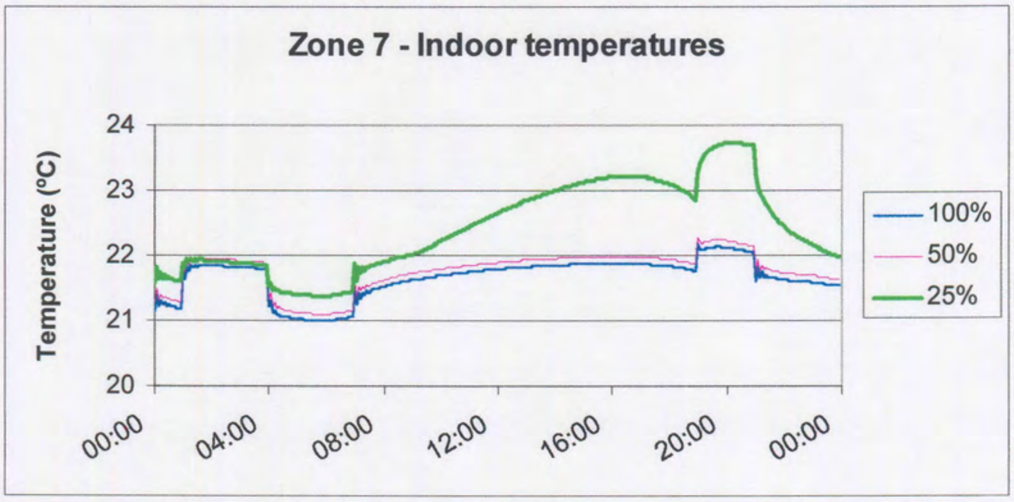

Figure D-7 – Simulation temperatures at different coil efficiencies for zone 7

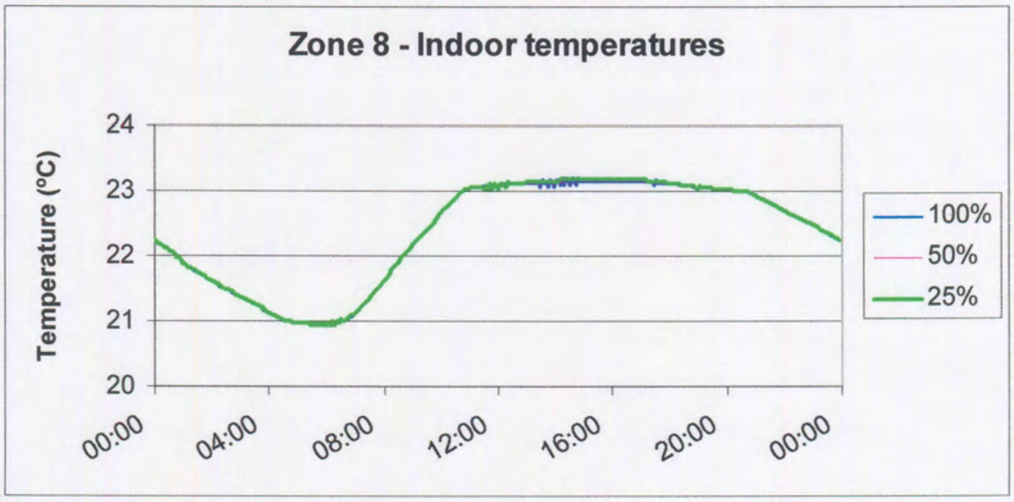

Figure D-8 – Simulation temperatures at different coil efficiencies for zone 8

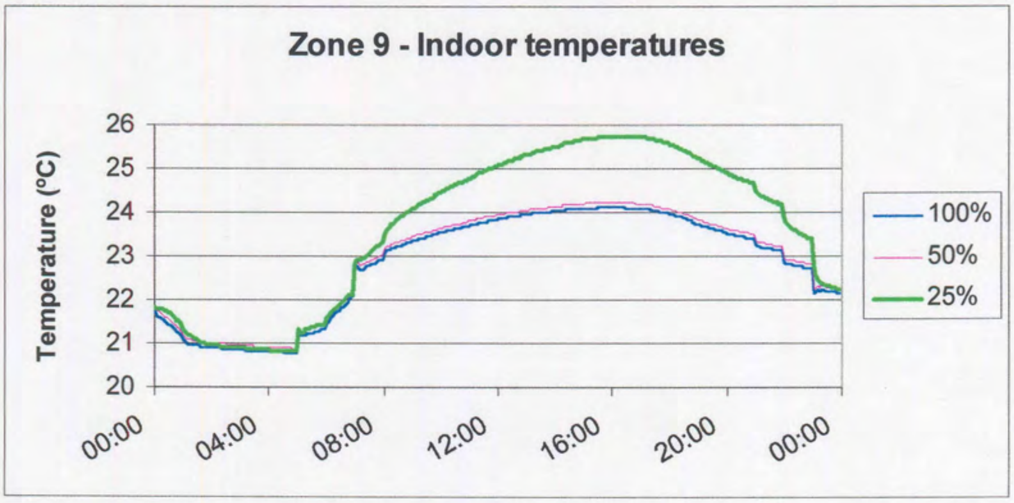

Figure D-9 – Simulation temperatures at different coil efficiencies for zone 9

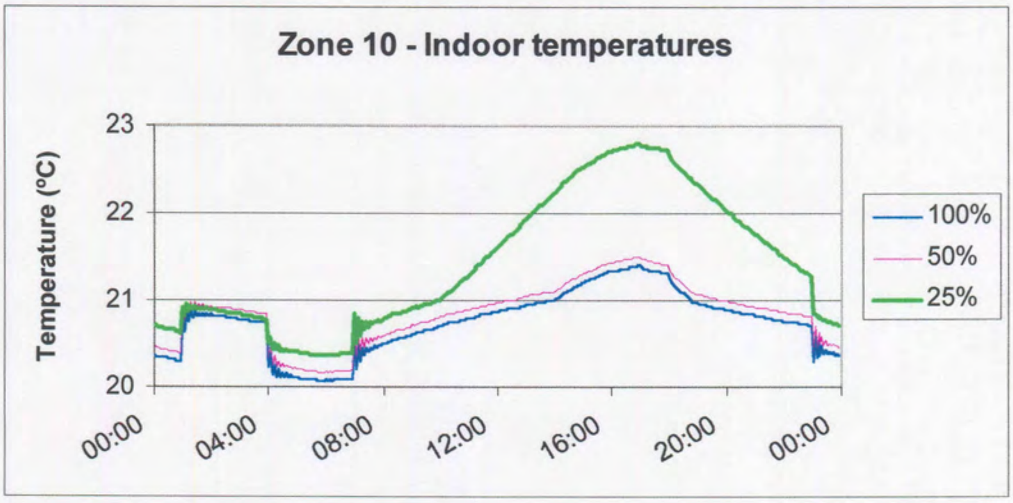

Figure D-10 – Simulation temperatures at different coil efficiencies for zone 10

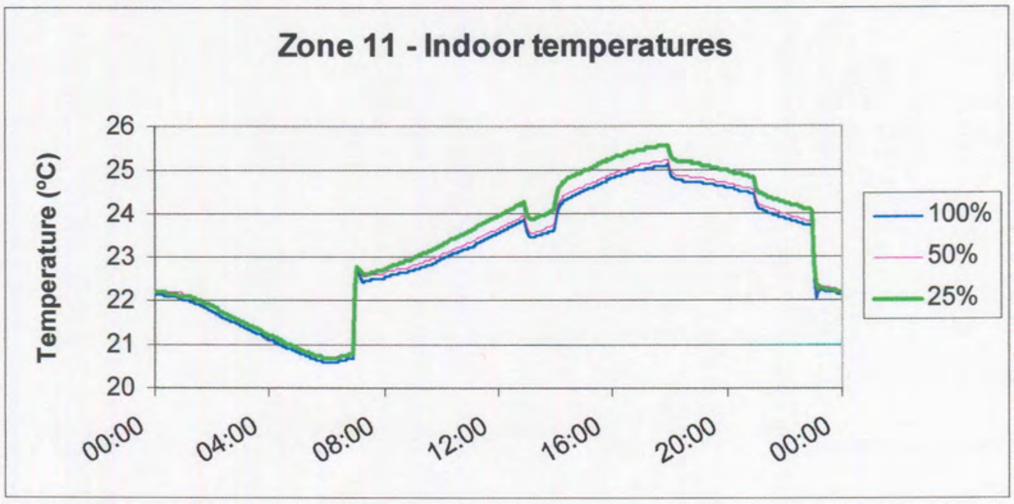

Figure D-11 – Simulation temperatures at different coil efficiencies for zone 11

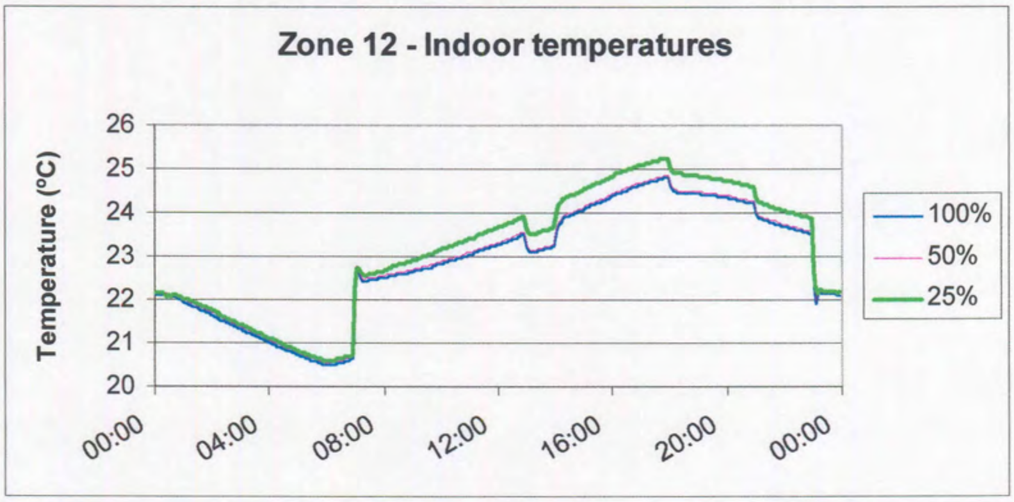

Figure D-12 – Simulation temperatures at different coil efficiencies for zone 12

D.2 Zone temperatures after chillers failed

The temperatures show is the simulation results obtained from a typical week day simulation. The weather data used for the simulation was the average outdoor temperatures and humidities for Pretoria.

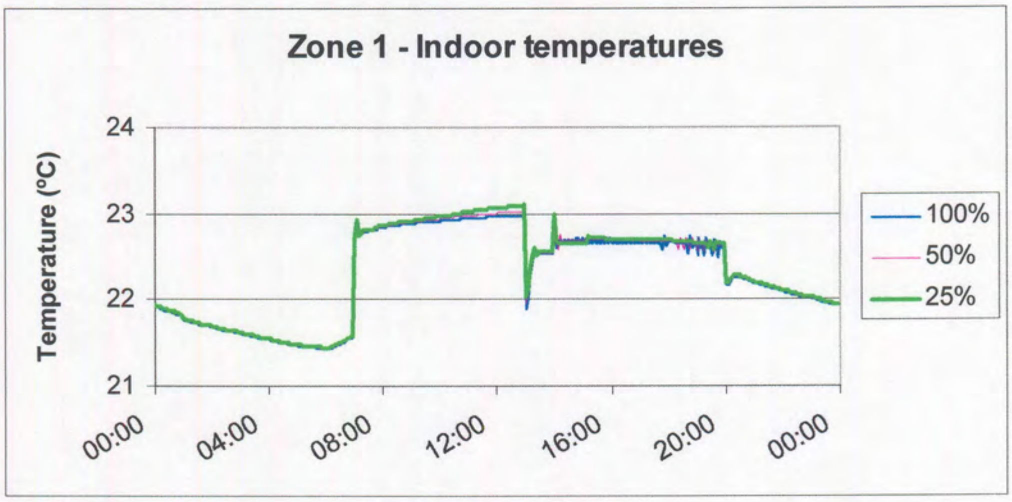

Figure D-13 – Simulation temperatures at different chiller efficiencies for zone 1

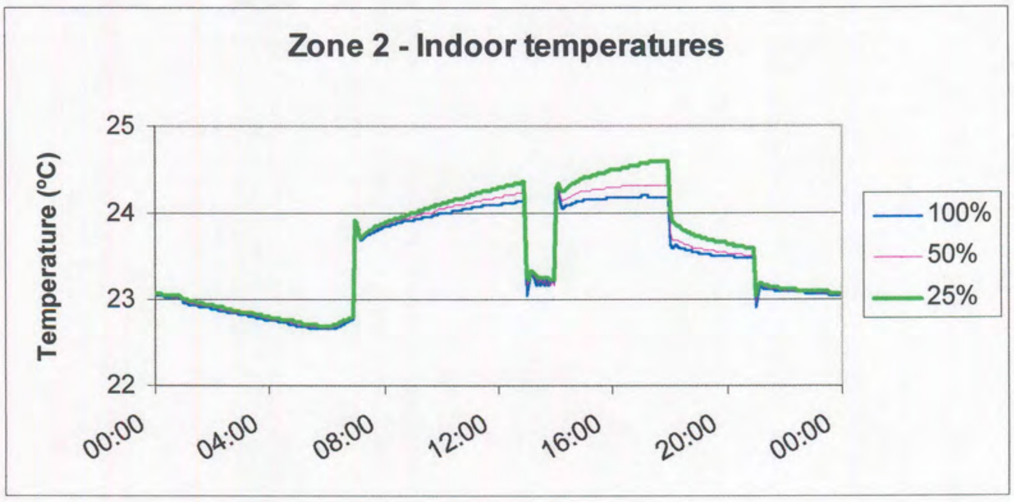

Figure D-14 – Simulation temperatures at different chiller efficiencies for zone 2

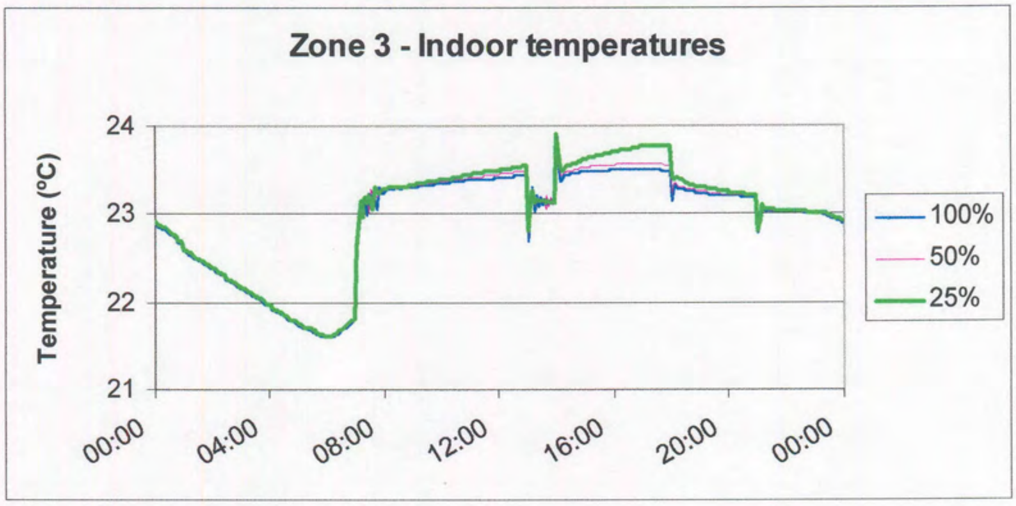

Figure D-15 – Simulation temperatures at different chiller efficiencies for zone 3

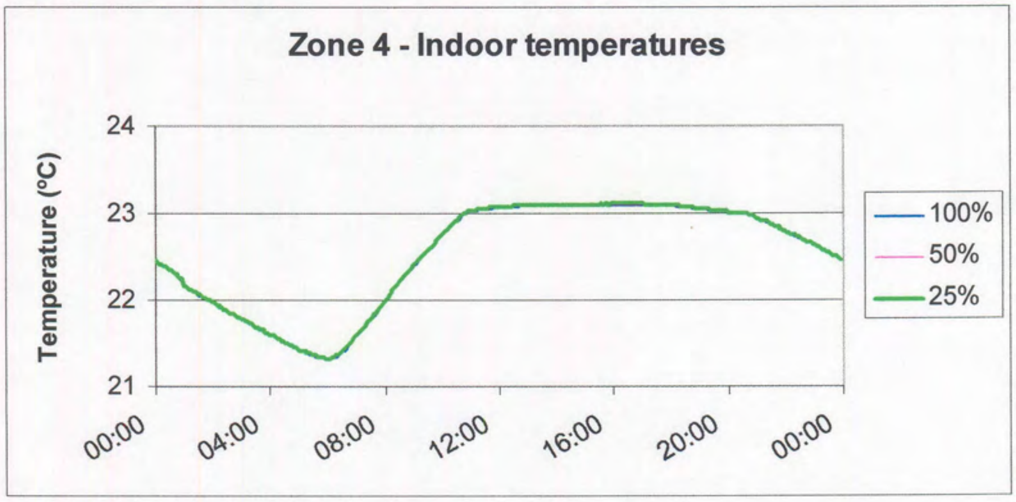

Figure D-16 – Simulation temperatures at different chiller efficiencies for zone 4

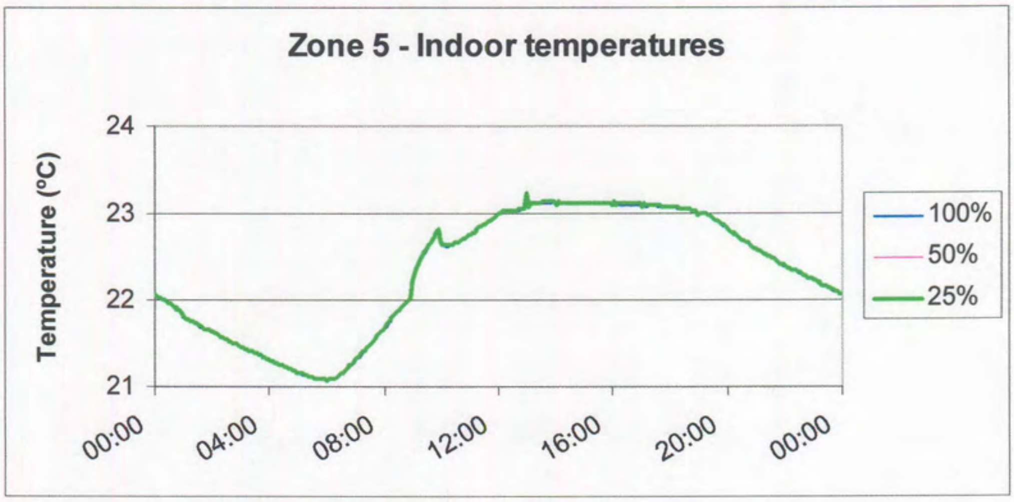

Figure D-17 – Simulation temperatures at different chiller efficiencies for zone 5

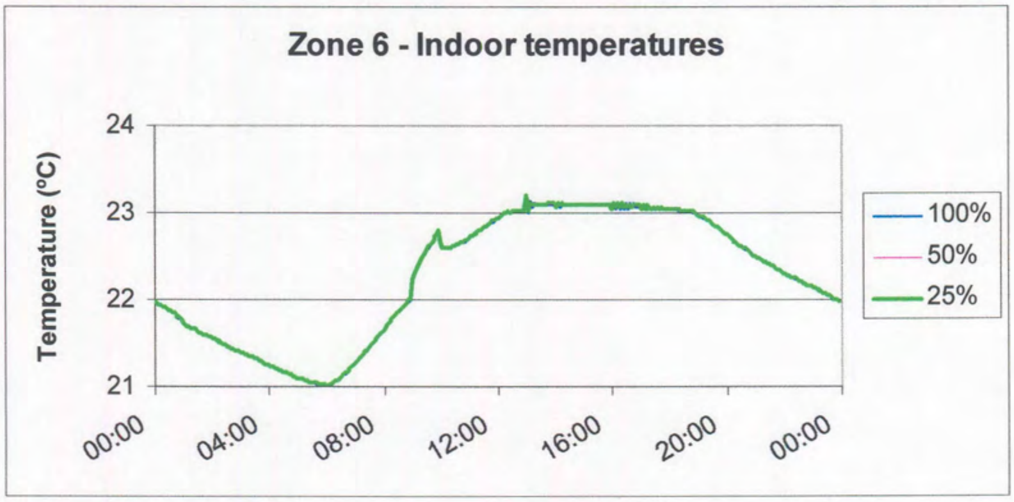

Figure D-18 – Simulation temperatures at different chiller efficiencies for zone 6

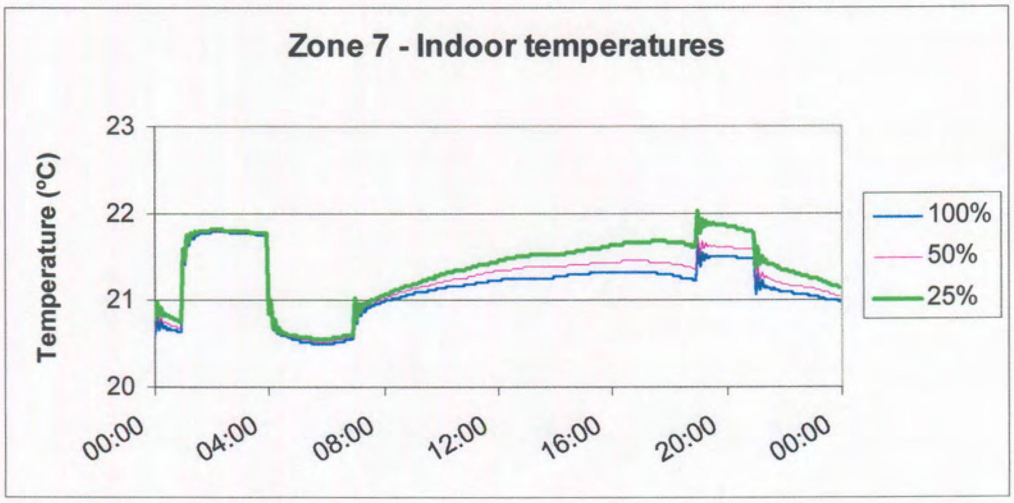

Figure D-19 – Simulation temperatures at different chiller efficiencies for zone 7

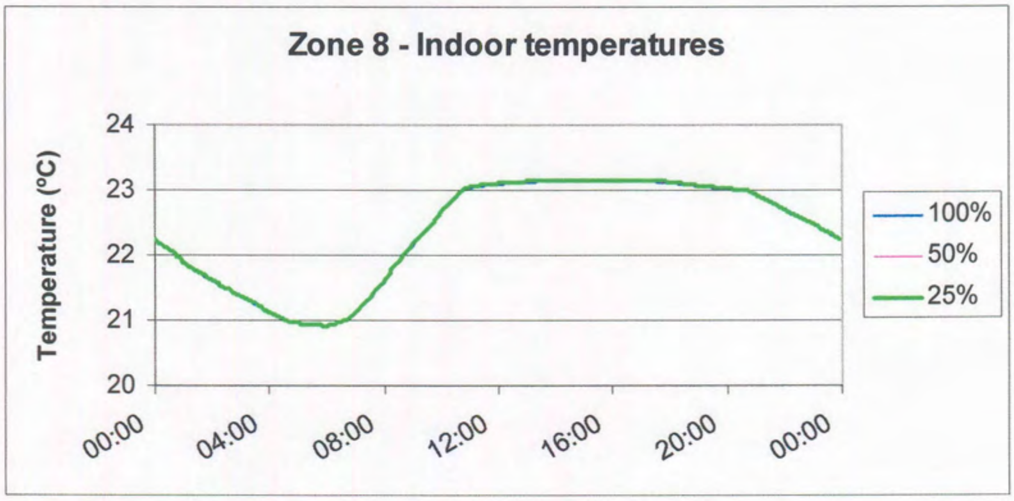

Figure D-20 – Simulation temperatures at different chiller efficiencies for zone 8

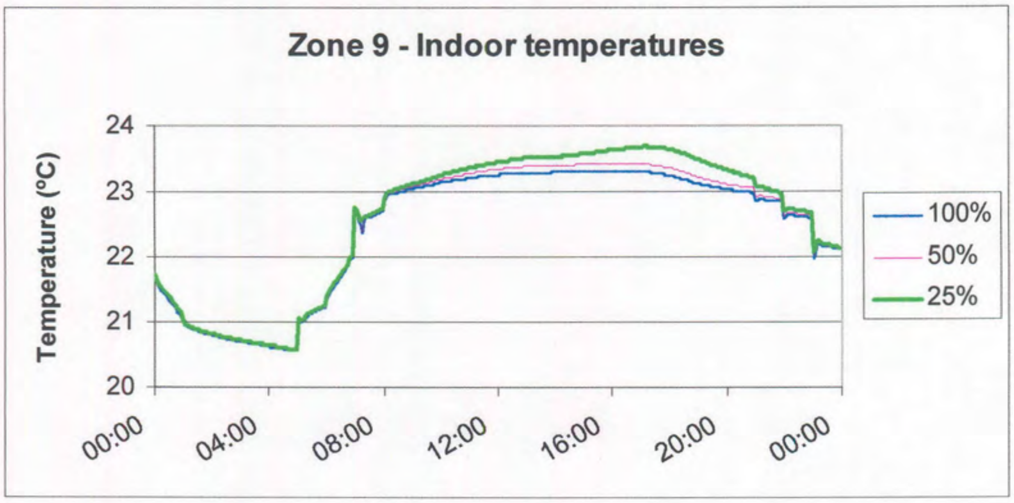

Figure D-21 – Simulation temperatures at different chiller efficiencies for zone 9

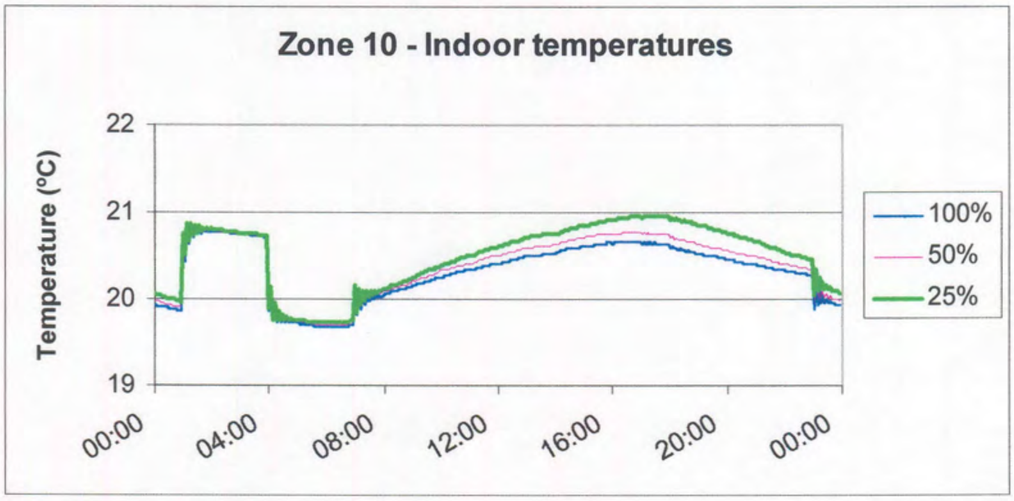

Figure D-22 – Simulation temperatures at different chiller efficiencies for zone 10

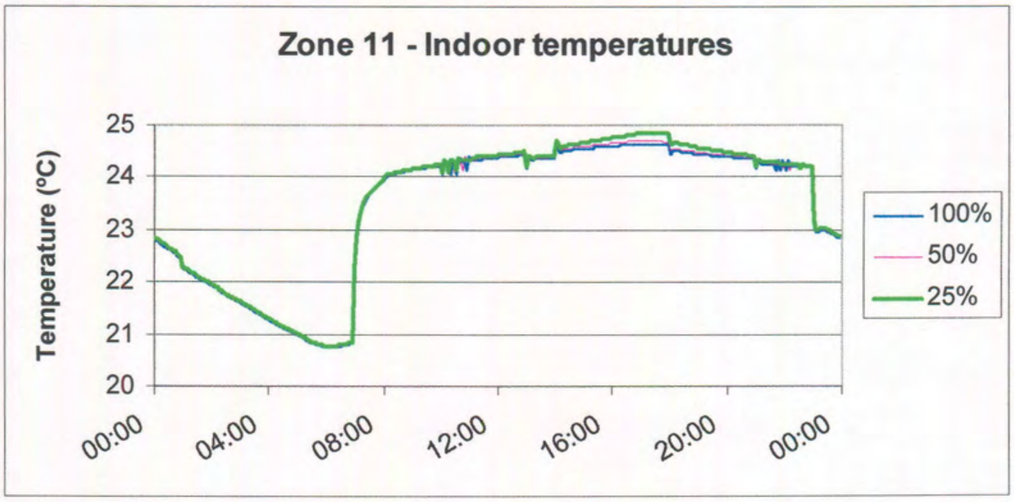

Figure D-23 – Simulation temperatures at different chiller efficiencies for zone 11

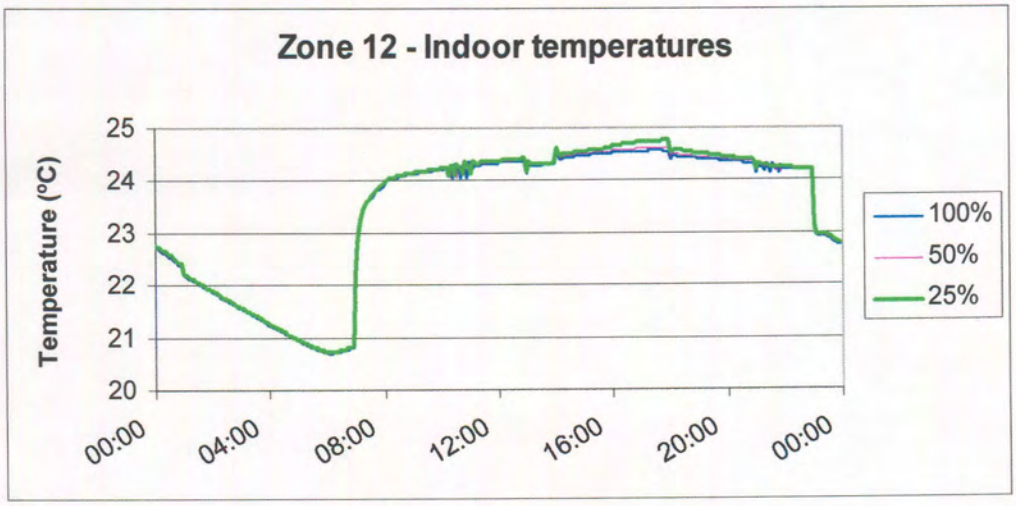

Figure D-24 – Simulation temperatures at different chiller efficiencies for zone 12