Kara Mackay

**Department of Mechanical
Engineering**

MSc Energy Systems and the Environment

Energy Efficient Cooling Solutions for Data Centres

Declaration of Author's Rights

Acknowledgements:

Abstract

The Data Centre industry has become essential in modern day society and is currently experiencing a major growth period. Found in nearly every sector of the economy, data centres house electronic equipment that provide communications networking, processing and storage of data. Working twenty four hours a day, seven days a week they consume up to 25 to 50 times more electricity than a standard office space. It is desirable to reduce this energy consumption so as to save on energy costs and CO_2 emissions.

Removing the heat produced by the IT equipment inside a data centre is a very high priority for a data centre, and it was found that cooling the data centre uses nearly as much power as the IT equipment itself. It is the intention of this project to investigate different methods of reducing the power consumption of data centres by reducing the power consumption of the cooling equipment supporting them.

To begin with an assessment of the cooling system currently used in data centres was carried out and an investigation into the effects of increasing chilled water setpoints from 7°C to 11°C in 2°C increments showed that small amounts of energy savings were made. It was also found that when the chilled water setpoint was 11°C the heat transfer wasn't sufficient enough to maintain the inside of the data centre at 24°C so this temperature had to be raised to 26°C.

A further study introducing free cooling into the conventional cooling system of a data centre, using the same chilled water setpoints, showed that further energy savings could be made, but only if free cooling is enabled at an optimum temperature, which was found to be 10°C for all chilled water setpoints.

An initial investigation of using combined heating, cooling and power with and without free cooling was performed showing that there is potential for energy savings and cost savings to be made by using CCHP with no free cooling.

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Nomenclature

AC	Alternating Current
AHU	Air handling Unit
ATS	Advance 3/single transformer
CCHP	Combined Cooling, Heating and Power
CO_2	Carbon Dioxide
COP	Coefficient of Performance
CPU	Central Processing Unit
DC	Direct Current
EPBD	European Directive of Energy Performance in Buildings
GHG	Green House Gas
HVAC	Heating Ventilation and Air Conditioning
kW	Kilowatts
kWh	Kilowatt hours
LTHW	Low temperature hot water
MTHW	Medium temperature hot water
PDU	Power Distribution Unit
PLR	Part Load Ratio
PSU	Power Supply Unit
Tec	Temperature entering compressor
Tee	Temperature entering evaporator
Tlc	Temperature of water leaving compressor
Tlc	Temperature of water leaving evaporator
UPS	Uninterruptible Power Supply
US	United States

Nomenclature for Equations

C_p [kJ/kgK] Constant pressure specific heat

COP_{nom} [-] Chiller nominal Coefficient of Performance at current conditions.

COP_{rated} [-] Chiller rated Coefficient of Performance at current conditions.

COP_{ratio} [-] Chiller ratio at current conditions divided by the rated COP.

$Capacity$ [kJ/hr] Chiller capacity at current conditions.

$Capacity_{rated}$ [kJ/hr] Chiller rated capacity.

$Capacity_{ratio}$ [kJ/hr] Chiller capacity at current conditions divided by the rated capacity.

h [kJ/kg] Specific enthalpy

m [kg/s] Mass flow rate

η [-] Efficiency

Q [kJ] Heat transfer

$\dot{Q}_{load}$ [kJ/hr] Current load on the chiller.

$\dot{Q}_{met}$ [kJ/hr] Load met by the chiller.

$\dot{Q}_{rejected}$ [kJ/hr] Energy rejected by the chiller to the ambient.

$\dot{m}_{chw}$ [kg/hr] Flow rate of fluid entering the chilled fluid stream.

Cp_{chw} [kJ/kg.K] Specific heat of fluid entering the chilled fluid stream.

$\dot{m}_{cw}$ [kg/hr] Flow rate of fluid entering the cooling fluid stream.

Cp_{cw} [kJ/kg.K] Specific heat of fluid entering the cooling fluid stream.

$T_{chw,set}$ [℃] Desired outlet temperature of fluid in the chilled fluid stream.

$T_{chw,in}$ [℃] Temperature of fluid entering the chilled fluid stream.

$T_{chw,out}$ [℃] Temperature of fluid exiting the chilled fluid stream.

PLR [0..1] Chiller Part Load Ratio (the ratio of the current load to the rated load.

P [kJ/hr] Power drawn by the chiller at current conditions.

$FFLP$ [0..1] Fraction of full load power.

1 <u>Introduction</u>

The Data Centre industry is currently experiencing a major growth period and has been made essential in modern day society because of the shift in information management from being paper based to being electronic. Found in nearly every sector of the economy, from businesses to government and non government organisations, data centres house electronic equipment that provide communications networking, processing and storage of data. The requirement for data centres to work twenty four hours a day, seven days a week means that they are massive consumers of power and studies have shown that over the past five years there has been a significant growth in the number of data servers, along with an estimated doubling in the energy used by these servers and the power and cooling infrastructure that supports them.

Due to climate change Britain, and the World, are making great efforts to reduce Carbon dioxide (CO_2) and Green House Gas (GHG) emissions and much legislation has been introduced setting targets to be met by governments in the UK and abroad. The UK alone is committed to reducing carbon emissions by 20%, from the 1997 levels, by the year 2010 and on March 13 2007, a draft Climate Change Bill was published aiming to put in place a framework to achieve a mandatory 60% cut, from the 1990 levels, by 2050. Buildings are considered large consumers of energy and the building industry is seeing the impacts of such bills and legislations, as this is the area where it is seen that a significant amount of CO_2 and GHG emissions savings can be made. Because of this, new and refurbished buildings now have to comply with Part L (in England and Wales) or Section 6 (in Scotland) of the Building Regulations, which stipulates efficient energy use in buildings. As such, new and refurbished data centres need to take action now to reduce their energy consumption so as to be in line with current legislation, and to reap the benefits it incurs such as reduced energy costs and carbon emissions.

This report can be considered as four main sections. The first section constitutes the literature review and gives a general background of data centres and their systems. Firstly Part L of the Building Regulations for England and Wales is described, and the changes that have been introduced to help tackle climate change are detailed along with how data centres will be affected by these changes. Secondly a background of data centres, explaining the reasons why they have become critical to modern day society is detailed as well as the energy consumed by them in Britain and the US. A description of the inside of a data centre is given, which is critical to understanding

how and where the energy is required and distributed amongst the data centre systems, which not only includes the IT equipment but also the infrastructure systems that support the IT equipment. This section uses a diagram, specific to the case study used in the project, to show that a majority of the power consumed by the IT equipment is matched by the power consumed by the cooling equipment. Therefore, if the power consumed by the IT equipment increases, so too does the power consumed by the infrastructure supporting it. Also acknowledged in this section is the fact that electrical power consumption comes secondary to ensuring reliability, and satisfying performance requirements is the primary requirement when designing a data centre. The large power consumption by data centres makes them ideal for investigating energy efficient design measures that can save on electricity use and money, and opportunities for energy efficiency within data centres is also covered in this section. A detailed diagram of a "typical" data centre HVAC system is given in order to understand the systems analysed in this project. The basic principles of chillers are explained using the Carnot and Reverse Carnot Cycles and the Coefficient of Performance is derived. This first section aims to explain the current status of data centres and the technologies used in "typical" data centres.

The second section outlines the case study and introduces the program TRNSYS 16, which was used to investigate how to make a data centre more energy efficient using current technology practices within data centres.

A description of the case study model is given and the systems used within the TRNSYS 16 model are looked at in detail. The methodology used for investigating the impacts on energy efficiency is implemented in this section. The methodology looked at different scenarios of increasing the data centre chilled water temperature leaving the chillers, and of incorporating free cooling. The results for the system with and without free cooling are analysed, showing the COP of the chiller and the COP of the full system, and how it varies throughout the year. A comparison of the energy consumption of the different scenarios is given along with a CO_2 analysis. Lastly a comparison of the best case scenario and base case scenario performance of the system during a winter and summer day is given, demonstrating why increasing the chilled water setpoint leaving the chillers from 7°C to 9°C saves on power consumption and hence CO_2 emissions.

The third section looks at improvements that could be made to the data centre that can help further reduce energy consumption and CO_2 emissions. Suggestions made in this project are using phase change materials for thermal storage, utilising

combined cooling heat and power (CCHP) and fuelling this with bio diesel. Firstly an overview of energy storage is provided and the use of phase change material in thermal storage is discussed. A diagram helping to describe how phase change materials work and a brief explanation of how they could be of benefit in a data centre is given. CCHP is then discussed briefly as it is covered in more detail in the following section. Rough calculations were performed to demonstrate the savings in CO_2 emissions and cost benefits of using CCHP instead of electric chillers. The last part of this section investigates the savings in CO_2 that can be made if bio diesel were to be used to fuel CCHP. A brief study of the supply of bio diesel found that only 3% of the world's production is in the UK, and a list of potential suppliers in the UK is provided in the appendices. The methodology of how the CO_2 is calculated for bio diesel is given and the results are presented, showing that using high blends of bio diesel for the case study will give CO_2 savings.

The fourth section is concerned with combined cooling heat and power (CCHP) and gives a general background of CCHP followed by a description of the components that make up a combined heat and power unit. A brief history of absorption chillers followed by a diagram demonstrating the differences in the amount of heat rejection between electric chillers and absorption chillers, is provided. This diagram is important in understanding that the heat rejection by absorption chillers is larger than that of electric chillers and this has an effect on the performance of the dry coolers in the system. A quick explanation of the vapour absorption cycle is also provided followed by a description of how the CHP unit and the absorption chillers are combined in terms of the Carnot and Reverse Carnot Cycle.

The CCHP case study is presented in this section, which builds on the previous models built in TRNSYS 16. The new TRNSYS 16 model replaces the previous electric chiller, in the original TRNSYS 16 model, with a high temperature and a low temperature absorption chiller connected to an internal combustion generator. It was at this stage the decision was made to control the heat output of the generator to meet the cooling load of the absorption chillers. This was because it was undesirable to have excess heat produced by the generator, which would require a means of cooling before being rejected to the atmosphere. Two simulations were then run in order to get results for the CCHP system with and without free cooling and the term Part Load Ratio (PLR) is introduced and used to compare the two systems. The last part of this section provides graphs showing the power produced by the CCHP unit, power consumption of the whole system and CO_2 analysis of the two systems.

The final section is an overview of results for non free cooling, free cooling and CCHP without free cooling, for a chilled water setpoint of 7°C and indoor data centre temperature of 24°C. These scenarios were chosen because current practices in data centres use these temperatures.

2 Background

This project is concerned with increasing the energy efficiency of Data Centres, whilst decreasing their carbon emissions. Presently Britain, and the World, are making great efforts to reduce Carbon dioxide (CO_2) and Green House Gas (GHG) emissions, and much legislation, such as the Kyoto Protocol, has been introduced, which sets targets to be met by governments in the UK and abroad. The UK alone is committed to reducing carbon emissions by 20%, from the 1997 levels, by the year 2010 and on March 13, 2007, a draft Climate Change Bill was published aiming to put in place a framework to achieve a mandatory 60% cut, from the 1990 levels, by 2050.

The building industry is seeing the impacts of such bills and legislations, as this is the area where it is seen that a significant amount of CO_2 and GHG emissions savings can be made. The Building Regulations for England and Wales, in particular Part L, have been modified and it is important to outline these modifications for this project so as to see the standards expected of the Data Centre Building. There are different Regulations for Scotland, which will not be covered in this project, as the case study for this project is based in Wales.

2.1 The Building Regulations

The Building Regulations apply to most new buildings, and many alterations of existing buildings, in England and Wales. They also apply to domestic, commercial and industrial buildings and are made under powers provided in the Building Act 1984. Their primary purpose is to ensure the health and safety of people in and around buildings and they promote: (Communities and Local Government, Building Regulations Overview, 2007)

- Standards for most aspects of a building's construction, including its structure, fire safety, sound insulation, drainage, ventilation and electrical safety. Electrical safety was added in January 2005 to reduce the number of deaths, injuries and fires caused by faulty electrical installations.

- Energy efficiency in buildings. The changes to the regulations on energy conservation proposed on 13 September 2005 will save a million tones of carbon per year by 2010 and help to combat climate change.

- The needs of all people including those with disabilities. They set standards for buildings to be accessible and hazard-free wherever possible.

The current edition of the regulations is 'The Building Regulations 2000' (as amended) and the majority of building projects are required to comply with them (Office of the Deputy Prime Minister, creating sustainable communities, 2003).

2.2   Changes to Part L of the Building Regulations

Part L of the Building regulations for England and Wales is concerned with conservation of heat and power within domestic and commercial buildings and applies to both existing and new builds. Changes to Part L have arisen because of the UK Governments legal commitment to implement the European Directive of energy performance in buildings (EPBD).   The EPBD requires that all new and existing buildings are given an energy rating, and that all new buildings meet a minimum standard of energy efficiency.   These new measures are intended to reduce carbon emissions by one million tonnes per year by 2010, equivalent to the emissions from more than one million semi-detached homes (Barbour Index, 2006).

The final versions of the new Part L Approved Documents of the Building Regulations were published in March 2006 and took effect in April 2006. The four new approved documents are focused on the key goal of reducing CO_2 emissions from buildings. The new approved documents subdivide dwellings and non dwelling buildings into new and existing buildings categories as follows: L1A, New dwellings; L1B, Existing dwellings; L2A, New Buildings other than dwellings; L2B, Existing buildings other than dwellings.

The previous Requirements L1 and L2 have been consolidated in Part L, Schedule 1 to cover 3 distinct requirements: (Barbour Index, information services, 2006).

1. Limiting heat gains and losses: through thermal elements and other parts of the building fabric; and from pipes, ducts and vessels used for space heating.
2. Providing and commissioning energy efficient fixed building services with effective controls.
3. Providing sufficient information to the owner about the building, the fixed building services and their maintenance requirements so that the building can be operated in a way that will use no more fuel and power than is reasonable in the circumstances.

Also contained in the regulations are two new requirements, regarding thermal elements and a change to energy status, as laid out in Regulation 4. Regulation 4A covers the replacement and renovation of thermal elements in existing buildings and

Regulation 4B covers changes in a buildings energy status, and requires buildings to comply with the applicable clauses of Part L of Schedule 1. (Barbour Index, information services, 2006)

Included in the Part L Approved Documents are five new Regulations, which implement the Directive covering Articles 3-6 of the EPBD, grouped under Part VA of the regulations listed below: (Barbour Index, information services, 2006)

- Regulation 17A defines the methodology for calculating the energy performance of buildings (EPBD, Article 3).
- Regulation 17B sets a minimum energy performance requirement for buildings based on a target CO_2 emission rate (EPBD, Article 4).
- Regulation 17C is about new buildings meeting the target CO_2 emission rate (EPBD, Article 5).
- Regulation 17D covers consequential improvements to buildings which are being extended, or which are having fixed building services installed for the first time or extended (EPBD, Article 6); this means that the energy performance of the whole building is upgraded. The Regulation applies to existing buildings with a total useful floor area over 1000m^2.
- Regulation 17E gives the interpretation of the term 'building' as either a whole building or parts of it that have been designed or altered for separate use.

The means of checking that a building meets the required CO_2 emission rate is covered by three new Regulations grouped in Part VI: (Barbour Index, information services, 2006)

- Regulation 20B covers air pressure testing of buildings.
- Regulation 20C describes ways of confirming that the fixed building services have been commissioned correctly.
- Regulation 20D requires written evidence that the actual building's CO_2 emission rate has been calculated correctly for the building, as constructed, and thus that it complies with Regulation 17C.

The Approved Documents for new dwellings and buildings, L1A and L2A are similar to each other, containing sections on design standards, quality of construction and commissioning, providing information, model designs and definitions.

Buildings that are exempt from the building regulations are: "Buildings not frequented by people. These are detached buildings, into which persons do not

normally go or only go intermittently to inspect plant and machinery. The exemption applies only if the building is sited at least one and a half times its heights either from the boundary or from any point of a building into which people normally go". (London Borough of Richmond Upon Thames).

From this it is presumed that if a data centre were to be built with office spaces attached, the building regulations would have to be adhered to. However, if the data centre is a stand alone data centre then the building regulations may not necessarily apply to it. Another factor to remember with a data centre is that the computers have very high energy requirements, when compared to a standard building, therefore, the building may be required to meet building regulation standards, but the computers (i.e., the activity) inside the room may be exempt from Part L of the building regulations. For best practices it would be recommended that energy efficient measures be taken during the design phase of any data centre in order to make the data centre more economical and to reduce green house gas emissions.

2.3 Background Data Centres

Data centres are facilities that primarily house electronic equipment that provide communications networking, processing and storage of data for companies. Data centres are found in nearly every sector of the economy, from businesses to government and non governmental organisations, and are essential to modern day society especially since the World's economy has seen a shift in its information management from being paper based to being electronic. Presently the data centre industry is experiencing a major growth period. This has been stimulated by an increase in demand for data processing and storage of which some reasons for this are listed below;

- The financial services are seeing an increase in use of electronic transactions such as online banking and electronic trading on the stock market.
- There is a growth of internet services such as music downloads and online communications.
- There has been a growth of global commerce and services.
- The adoption of satellite navigation.
- Electronic tracking in the transportation industry.
- The transition of records keeping from paper based to electronic, for example medical records are all saved in a database, businesses now save employee

records electronically, and often companies are required by law to hold records electronically for a set amount of years.

Data centres have become common and critical to the functioning of businesses as they are relied on for information management, aiding business processes and communication. It is therefore a requirement that data centres work 24 hours a day, 7 days a week. Because of this they are massive consumers of power, up to 25 to 50 times more electricity is required than a standard office space (Pacific Gas & Electric Company, 2006). A study carried out in the United States by the Environmental Protection Agency (U.S EPA, 2007) has shown that over the past five years, the demand for computer resources has led to significant growth in the number of data servers, along with an estimated doubling in the energy used by these servers and the power and cooling infrastructure that supports them. This statement can be supported by the findings of Dr Koomey (2007), who found that in 2005, total data centre power consumption, including servers, cooling and auxiliary equipment, amounted to 1.2% of total US electricity consumption, US $2.7 billion and in 2006, U.S EPA (2007) found that the power consumed by data centres was 1.5% of total U.S electricity consumption costing $4.5 billion. In his studies Dr Koomey (2007) also noted that World wide data centre electricity consumption amounted to US $7.2 billion in 2005. One of the reasons for the increase in energy consumption by data centres, between 2000 and 2006, can be attributed to one type of server, the volume server, whose energy use more than doubled in this period, and is now responsible for 68% of the electricity consumed by IT equipment (U.S EPA, 2007). In the United Kingdom it was found that the average energy costs of running a corporate data centre is currently around £5.3 million a year and this figure is set to double over the next five years to £11 million (Barker, 2006). Not only has the energy consumed by the IT equipment risen over the past few years, but so has the power and cooling infrastructure that supports the data centre, which in itself consumes a significant amount of energy.

2.4 Inside a Data Centre

In order to understand the energy requirements of a data centre a description of the standard layout of a data centre will first be given. Firstly, data centres are not designed for humans but for computers, and as a result typically have minimal circulation of fresh air and no windows. They can be housed in either an existing building that has been retrofitted or a new building specifically constructed for the purpose and their size can range from a small room, or server closet, within a

conventional building to a large building purpose built to house IT equipment and supporting infrastructure.

Data centre rooms contain rows of racks containing IT equipment. The IT equipment consists of servers, storage devices and network equipment. A data centre also houses the relevant infrastructure equipment that supports the IT equipment such as the power delivery systems, which provide backup power, regulate voltage and convert alternating and direct current (AC/DC), and the cooling equipment such as Room Air Conditioning Units (RACUs), also known as Computer Room Air Conditioning (CRAC) units.

The electricity supplied to the IT equipment follows the path described;
Electricity is supplied to an uninterruptible power supply unit (UPS).
The UPS acts as a battery backup. This is to ensure that the IT equipment does not experience any power disruptions, which would cause disruption to businesses or loss of important data, both of which are very costly to businesses.
Electricity is converted from AC to DC in the UPS to charge the batteries.
The power from the batteries is then converted from DC to AC before leaving the UPS.
Power enters the power distribution unit (PDU) and is sent to the IT equipment in the racks.
In the servers the server power supply unit (PSU) converts electricity from AC to low voltage DC power.
Low voltage DC power is used by the servers' internal components (central processing unit (CPU), disk drives, memory, fans).
Storage devices and network equipment, facilitating the storage and transmission of data, also receive electricity.

Data centres are operated continuously and modern IT equipment generates high, concentrated heating loads that must be removed so the equipment can operate properly. It is therefore important, when designing a data centre that careful consideration is given to the control of the air flow. In most data centres, RACUs meet the cooling requirements and the entire air handling unit (AHU) is situated on the data centre floor. The AHU is responsible for conditioning and distributing the air throughout the data centre and consists of fans, filters, and cooling coils. The aim of an air management system is to reduce or eliminate mixing between the cold air supplied to the IT equipment and the hot air rejected from the IT equipment. There

are many designs and methods available to achieve a good air management system, such as hot aisle, cold aisle rack layouts, ventilated racks, flexible barriers, optimised supply and return grills/ floor tiles. However, the hot aisle, cold aisle configuration is becoming more widely used due to its ability to serve high density racks better than previous, more mixed flow configurations (Pacific Gas & Electric Company, 2006). Figure 1 shows a basic hot aisle cold aisle rack layout and also shows where the cold air enters and the hot air exits.
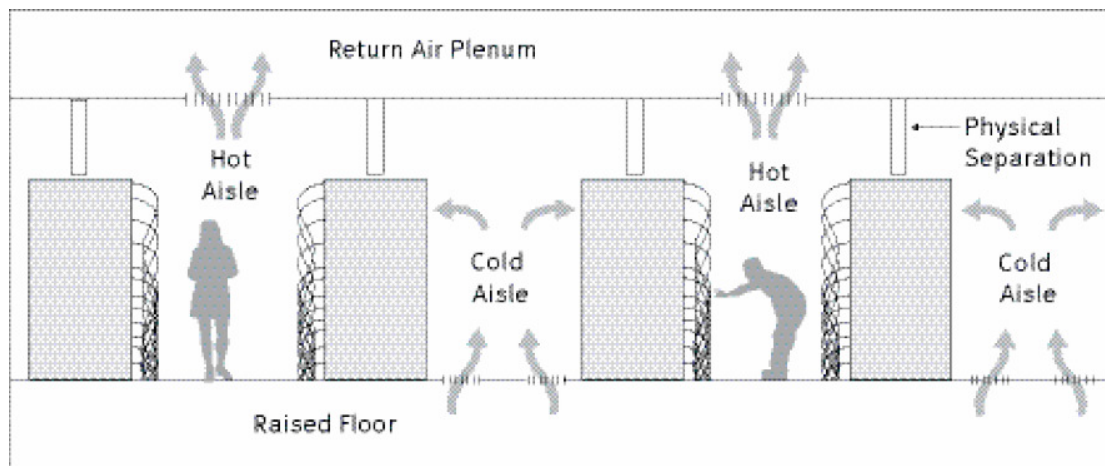
Figure 1: Hot aisle/ cold aisle rack configuration showing air flow. Source: Pacific Gas & Electric Company, 2006, High Performance Data Centres.

It can be seen from the diagrams that the data centre IT equipment is laid out in rows of racks separated by cold and hot aisles.   Isolation barriers (the physical separation), which prevent the hot air and cold air mixing and reduce air recirculation are also shown in Figure 1. The cold aisle is placed on the rack air intake side and the hot aisle is placed at the rack exhaust side. The cold air is drawn into the racks by the equipment from the front and the hot air produced by the equipment is rejected into the hot aisle where it is removed, thus achieving a front to back air flow pattern. When properly installed, with isolation barriers, the temperature from the hot aisle does not impact upon the temperature of the racks and hence the operation of the data centre. The system is designed to only supply cold air to the cold aisles and remove hot air from the hot aisles. Cold air is blown up through floor grills/ tiles/ vents to the cold aisle. The floor is raised and serves as both a duct and a wiring chase. Therefore, consideration must be taken to ensure that the air flow is not blocked by electrical equipment. The placing of floor grills or tiles supplying the cold air to the aisles is also of importance so that 'hot' spots do not occur and grills are not blocked. The air from the hot aisle rises and is removed, then taken to the RACUs where it is conditioned, by passing over coils containing chilled water that has been

pumped from chillers located outside the data centre room. The air is re-circulated internally in this manner and only a small percentage of outside air is admitted to the room to keep the zone positively pressurized. When designed correctly the air management system can reduce initial start up costs, operating costs and increase the data centres density capacity as well as reducing failure of equipment due to heat related processes.

2.5   Energy Use inside a Data Centre

Data centres are designed for a specific power density, watts per square foot (or meter). In the U.S a typical cabinet of equipment occupies 14 square feet (1.3 m²) of space and over the past several years power consumption in a rack has steadily increased. This is due to the replacement of pedestal servers, drawing 1 kW/rack, with 1U/2U pizza box servers at 3.5 kW/cabinet and blade implementations at 10 kW/cabinet (Data Centre Energy Dynamics, 2007). In addition to this, faster CPU's and larger memory chips contribute towards increasing power demands. The number of racks a data centre can support is limited by the power required per rack. A typical data centre measuring 10,000-20,000 sq.ft (square feet) (1860 m²) is designed for 50-100 watts/sq.ft (540-1080 W/m²), thus requiring 0.5 MW to 2 MW of power. As all the energy entering the data centre is converted to heat, cooling is required for the IT equipment, and as IT loads and hence heat loads increase, so does the floor space required for cooling equipment and the cost of cooling. This large amount of power is costly and difficult to obtain, and supporting the infrastructure needed, such as generators, ATS, UPS, and distribution equipment adds to this expense.

Another expense is also accrued through the backup equipment that is in place for power delivery and cooling. Figure 2 below shows this, using Chillers as an example. A typical data centre may have a load of 900 kW and use three chillers of 300 kW to meet this load, often this system would be called "N" and in this example N=3, the number of chillers needed for the data centre to work correctly. If one of these chillers was to fail, there is a spare chiller available for use, this is termed a plus one, and the system is now an N+1 system.

Chillers

N   +1
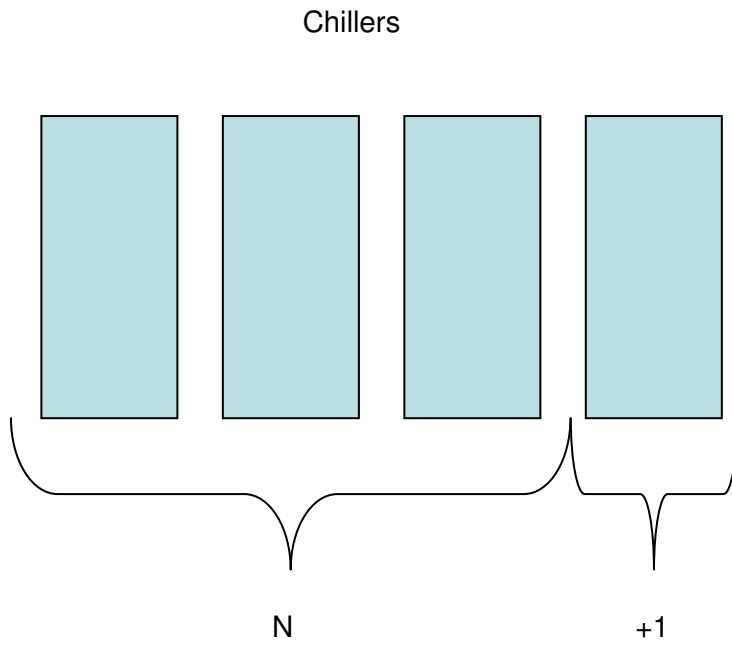
Figure 2: N+1 Cooling Plant

Figure 3 shows two cooling plants, both of which are an N+1 system. The second cooling system is in place in case something should happen to the first system, making it unusable. These two systems are termed 2N, i.e., a duplicate has been made.

N   N

Chillers   Chillers

N   +1   N   +1
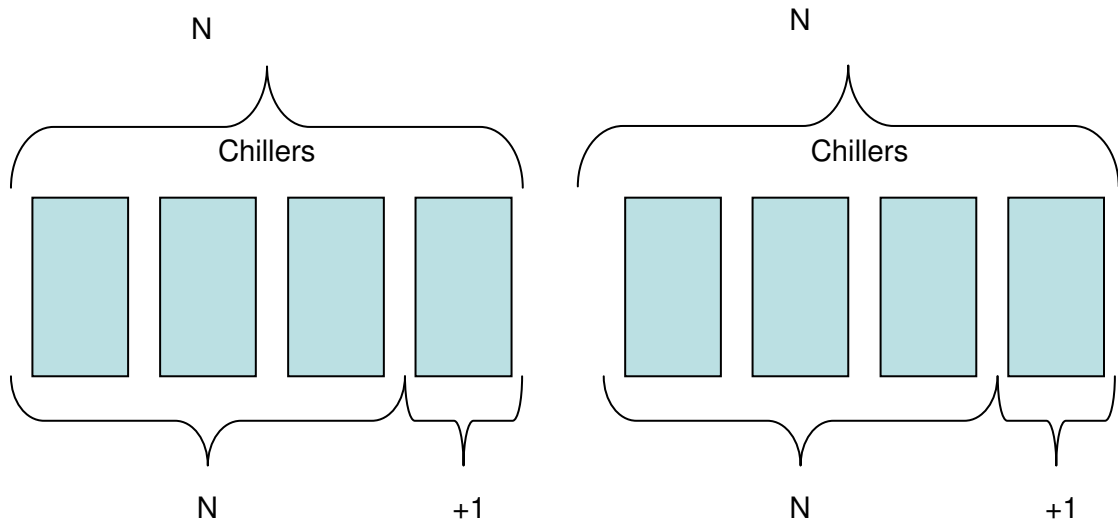
Figure 3: 2N cooling plant

Three key components are supplied with energy in a data centre; IT equipment, power delivery and cooling. If data centres were 100% efficient then all the power supplied would reach the IT loads. However, the two other components are also consumers of energy and the energy breakdown is shown in Figure 4. The diagram is based on a typical data centre with 2N power and N+1 cooling equipment, operating at about 30% of rated capacity, which makes this a conservative estimate.
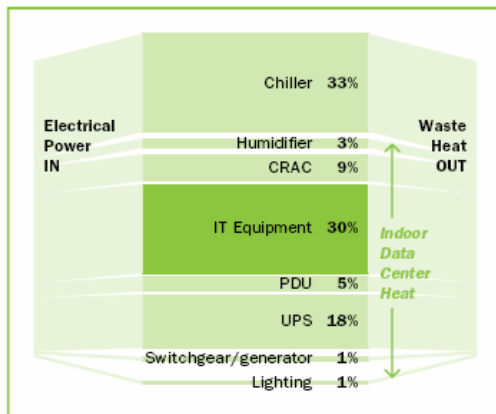
Figure 4: Data Centre Energy Consumption. Source: The Green Grid, 2007. Guidelines for Energy-Efficient Data Centres.

Figure 4 illustrates that the power consumed by the IT equipment is matched by the power consumed by the cooling equipment and all the energy consumed by the data centre is rejected as waste heat into the atmosphere. Therefore, if the power consumed by the IT equipment increases, so too does the power consumed by the infrastructure supporting it.

In order to relate this diagram to the project, a diagram specific to this project was made, using preliminary power results from TRNSYS 16, and used as a comparison. Figure 5 shows the power breakdown of the systems used, and the systems themselves, in the TRNSYS 16 model that gave rise to Figure 6. HVAC (Heating Ventilation and Air Conditioning) is the term used in this diagram for the Chillers and all the pumps required for circulating the chilled water.
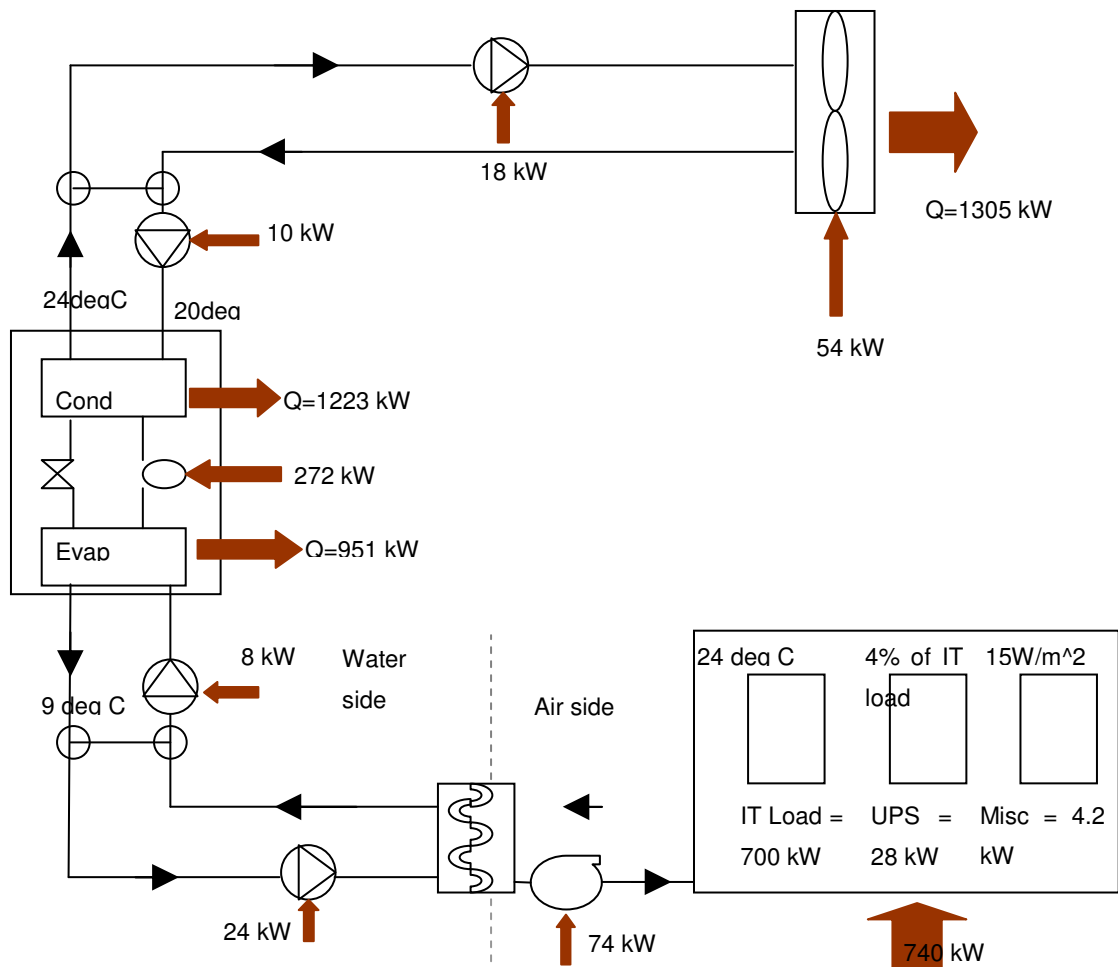
Figure 5: Model entered into TRNSYS for producing Energy Breakdown
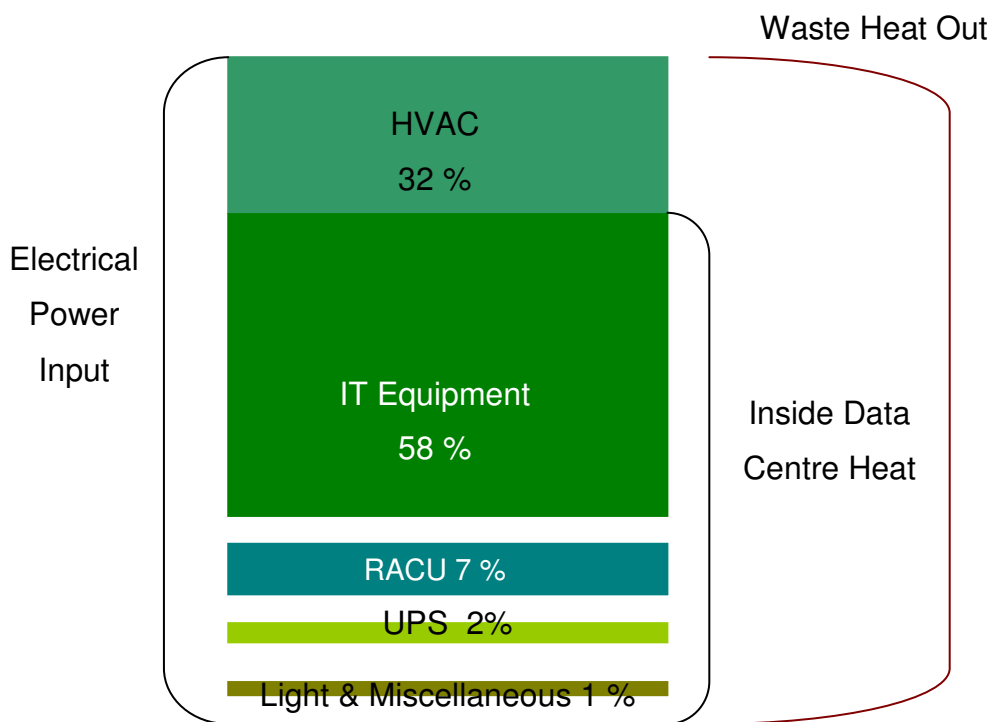
Figure 6: Data Centre Energy Breakdown based on Case Study

The power breakdown of Figure 5 and the assumptions used for lighting, miscellaneous and UPS input, is from initial facts and figures provided by WSP, for the case study used in this project. Although the electricity consumption between the case study data centre and the Green Grid data centre electricity consumption differ, it can still be seen that the Chillers and their accessories (water pumps, dry coolers, fans) and the IT equipment are the largest consumers of power entering a data centre.

There are many issues that can reduce the efficiency of data centres, these include: power distribution units and/or transformers operating below their full load capacities, air conditioners consuming extra power to drive air over long distances at high pressure or compensate for blockages under the floor, over sizing a UPS to avoid operating near its capacity limit, running a UPS system at low loads, reducing the efficiency of the equipment.

The primary requirement, when designing a data centre, has always been to ensure reliability and satisfy performance requirements. Electrical power consumption has been a secondary attribute, and equipment that could compromise the performance of the data centre will generally not be implemented. This is despite the fact that electrical energy costs over the life of a data centre may exceed the costs of the electrical power system, including the UPS, and even the cost of the actual IT equipment. The low priority given to the consumption of electrical power during the design phase, means that its management as an expense has also been of low priority, and as a consequence, few data centres are able to supply in depth information on the monitoring of their energy usage and costs. Some reasons for this are explained below (The Green Grid, 2007):

- Electrical bills are sent out long after charges are incurred and are seen as a necessary event that most people don't consider trying to influence.
- Electrical bills for the data centre may be incorporated within a larger electrical bill and are often not within the responsibility or budget of the data centre operating group.
- There are no clear links existing between particular decisions such as installing a new zone of equipment in the data centre or operational practices, and the increased cost of electricity.
- Tools for modelling the electrical costs in data centres are not widely available or commonly used during the design phase.

- Often sufficient information regarding energy cost consequences is not given to decision makers during the planning and purchasing phase of the project.

In order to avoid the high consumption of electricity by data centres in the future, the use of best practices, such as correctly sizing the physical infrastructure system to the load and designing an energy efficient system, should be employed. For this to be successful a multi pronged approach, enforced throughout the lifecycle of the data centre is necessary, incorporating energy efficient strategy, engineering, deployment, operations and organisation.

2.6 Energy Problems Faced by Data Centres

The amount of electricity used by data centre servers and infrastructure is increasingly becoming an important issue as reliance upon data centre services have become more widespread. Data has to be available constantly in order to enable businesses to operate and data loss can cost a company thousands. Therefore, one of the major criteria when designing a data centre is its ability to be resilient. If there is a loss of power to the data centre, servers will stop working, and the rise in temperature, resulting from the RACUs not working, will cause a significant rise in temperature of the data centre, damaging servers and important data can be lost. It is imperative that energy systems for a data centre are designed with suitable back up. Another problem facing data centres are the cooling requirements. Data centre equipment racks can produce very concentrated heat loads and it is important that this waste heat is removed so servers do not overheat.   Therefore, air flow management and cooling are very important factors in the design stage. It is also imperative that cooling is still available when changes to the power availability occur, or if there is a total loss of power. Because of this, cooling requirements for dense servers are driving up the power demand and nowadays increasing numbers of data centres are finding that they run out of power availability before they run out of floor space.

2.7 Opportunities for Energy Efficiency

Until recently there has been limited research carried out to ascertain energy reduction options available to data centres. Due to the growing concern for the environment and a desire to reduce CO_2 emissions, as well as new legislation being put into practice, such as changes to the building regulations, research into energy efficient data centres is becoming an increasingly important topic. The large power consumption by data centres makes them ideal for investigating energy efficient design measures that can save on electricity use and money. However, these measures are strongly dictated by the fact that the requirement for reliability and high power density outweighs that of energy efficiency.

Careful air management practices within a data centre can have a significant effect on energy efficiency. Some issues that are critical to the performance of a data centre include (Pacific Gas & Electric Company, 2006):

- Using best-practices air management, such as strict hot aisle/cold aisle configuration, can double the computer server cooling capacity of a data centre.

- An airside economiser is part of the HVAC equipment, and uses controls and supply and return air dampers to control outside air quantities. They are used in some applications to cool the indoor air temperature when the outdoor air temperature and humidity are sufficiently low enough to economically provide cooling without using the mechanical refrigeration cycle. Air management can reduce data centre cooling costs by over 60% when combined with an airside economiser.

- Removing hot air immediately as it exits the equipment allows for higher capacity and much higher efficiency than mixing the hot exhaust air with the cooling air being drawn into the equipment. Equipment environmental temperature specifications refer primarily to the air being drawn in to cool the system.

- A higher difference between the return air and supply air temperatures increases the maximum load density possible in the space and can help reduce the size of the cooling equipment required, particularly when lower-cost mass produced package air handling units are used.

- Poor airflow management will reduce both the efficiency and capacity of computer room cooling equipment. Examples of common problems that can decrease the usable capacity of a RACU by 50% or more are: leaking floor tiles/cable openings, poorly placed overhead supplies, under floor plenum obstructions, and inappropriately oriented rack exhausts.

Other considerations for increasing energy efficiency and 'greening' a data centre are: raising the temperature within the data centre room, in order to reduce the work having to be done by the chillers; using free cooling when available; energy storage facilities such as phase change materials for storage of coolth, or hydrogen fuel cells for storing electricity; procuring energy from a 'green' supplier; producing power and heat onsite (combined cooling heat and power) so as to save on electricity usage and costs.

This project intends to look at the options available for making data centres more energy efficient, whilst still maintaining high reliability within the data centre. Referring back to Figure 6 shows that the only variable capable of having a significant impact on energy use is the energy used for cooling the data centre, i.e., the performance of the chillers, as the IT loads cannot be changed without significant changes being made to servers, which are out with the scope of this project. Pumps, air conditioning units and UPS units are difficult to change without manufacturing redesign.

3 Description of some of the systems used in a "Typical" Data Centre

Buildings that use full air conditioning require some means of cooling and dehumidification. This is usually provided by an HVAC system that consists of chillers, which provide chilled water to RACUs that condition the air, and all the corresponding pumps and pipes to enable air conditioning to occur. To gain an understanding of the systems analysed in this project, a diagram showing the components that make up the HVAC system of a "typical" data centre is provided. A more detailed description of the systems is given in the subsequent chapters of the project.
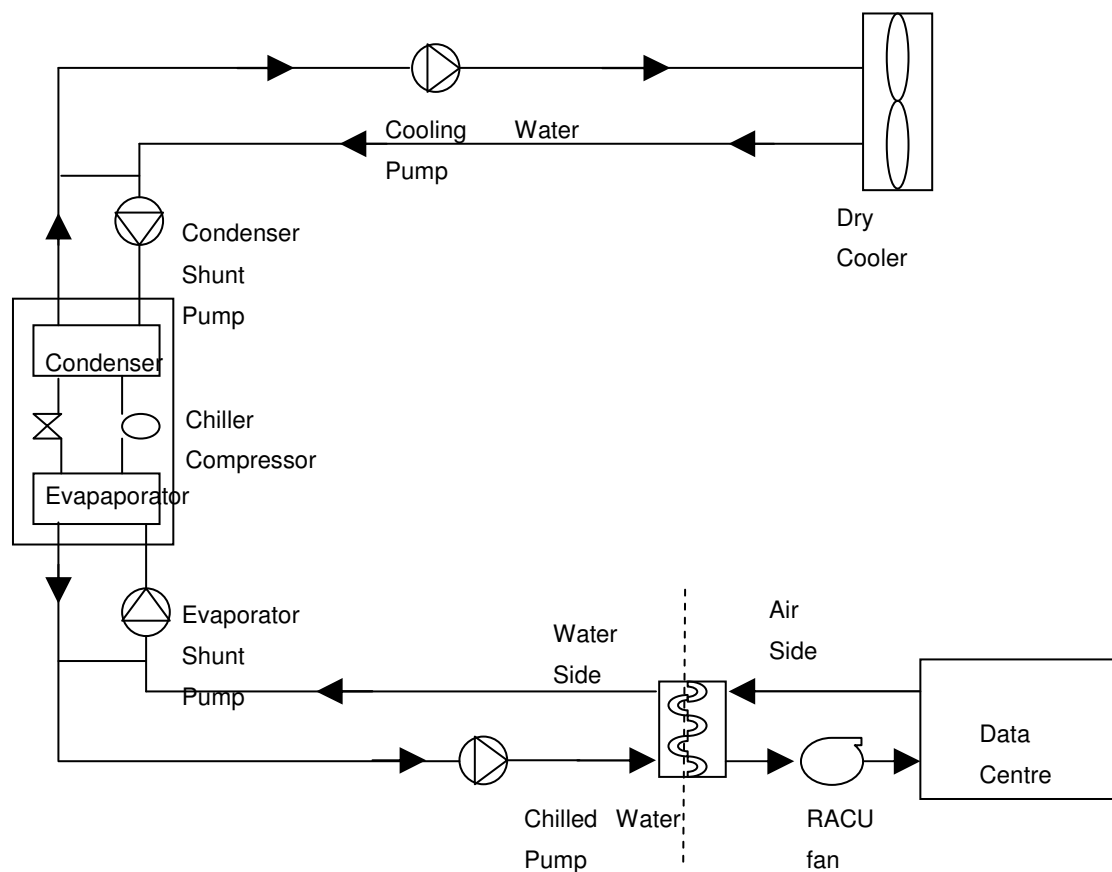
Figure 7: "Typical" HVAC system for a Data Centre

It can be seen in the diagram that the chillers are only one component of the overall HVAC system that is required for cooling a building. The dry coolers required for cooling the water exiting and entering the condenser of the chiller, are an important component, as are the components that make up the RACUs; RACU fans that

circulate cool air around the room, and the heat exchanger that removes the heat from the air in the room. The performance of the HVAC system is largely reliant on the pumps in the system, as the flow rate of the water has an effect on the rate of heat transfer in the system. The shunt pumps of the system are necessary to facilitate the manufacturer of the chillers requirement for certain temperatures and flow rates to be passing through the evaporator and condenser of the chiller at all times.

3.1   Chillers

Chillers use a vapour compression or absorption refrigeration cycle to remove heat from a liquid. This liquid is usually water which may also contain a small amount (approximately 20%) glycol and corrosion inhibitors. In mid to large size businesses chilled water is used to cool and dehumidify air. Sometimes other fluids such as oil can be chilled, but for this project only chilled water is considered. This project will be analysing two types of chillers, vapour compression chillers and absorption chillers. In order to gain an understanding of how these work the basic principles of chillers will be outlined.

3.1.1 Basic Principles of Chillers

The Clausius statement of the Second Law of Thermodynamics indicates that "it is impossible to operate a cyclic device whose sole effect is the energy transfer of heat from a cooler body to a hotter body" (Herold, K.E, 1996). However, a net heat transfer from a cooler body to a hotter body is possible if a cyclic device receives an energy input, either in the form of work in a vapour compression cycle, or heat in an absorption cycle. Refrigeration is the process of moving heat from one location to another using this method. Applications of refrigeration include commercial and industrial refrigeration and air conditioning systems.

Because of this a common design of vapour compression chillers uses a modified reverse Rankine cycle, which is itself a modification of the theoretically ideal Carnot cycle for fluids and processes.

3.1.2 Carnot Cycle

The Carnot Cycle can be used as an idealized energy conversion cycle for a simple steam plant. The Carnot Cycle for power generation is shown in Figure 8 on a temperature- entropy diagram.
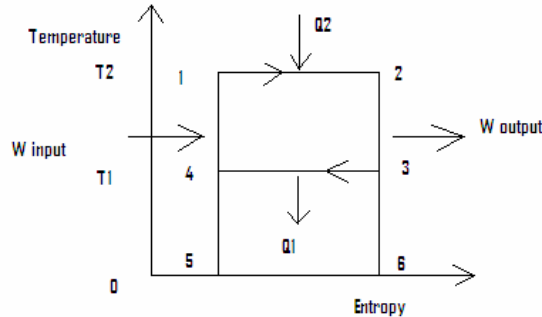
Figure 8: The Carnot Cycle for power generation. Source: Herold et al, 1996. Absorption Chillers and Heat Pumps.

All the energies are positive in the direction of the arrows, which is clockwise. Process 1-2 represents the isothermal addition of heat, which occurs in the boiler of the plant. Process 2-3 represents the isentropic production of work, which occurs in the turbine of the plant. Process 3-4 represents the isothermal rejection of heat Q1 at a temperature T1. Process 4-1 represents the isentropic, adiabatic input of work that occurs in the compressor of the plant. If all processes are assumed to be reversible then the area enclosed by 1234 represents the net amount of work produced (W) and the area enclosed by 3456 represents the amount of thermal energy Q1 rejected by the cycle, assuming that 5 and 6 are at temperature T= 0K. The sum of the two areas (1256) represents the amount of heat Q_2 supplied to the cycle as required by the first law.

$$Q_2 = Q_1 + W \qquad\qquad \text{Equation 1}$$

The efficiency is defined as the amount of work produced from a cycle divided by the heat supplied to the cycle.

$$\eta = \frac{W}{Q_2} \qquad\qquad \text{Equation 2}$$

For the Carnot Cycle shown in Figure 8, the Second Law of Thermodynamics states that for reversible operation the net entropy production is zero, so that;

$$\frac{Q_2}{T_2} - \frac{Q_1}{T_1} = 0 \qquad\qquad \text{Equation 3}$$

The maximum efficiency, available in terms of temperature only, can be found by modifying Equation 2, and is given by;

$$\eta = 1 - \frac{T_1}{T_2} \hspace{4cm} \text{Equation 4}$$

Where $T_1 = SinkTemperature$ and $T_2 = SourceTemperature$.

Although the practical power cycle will always be less than this ideal, for any particular combination of source and sink, an indication of the importance of temperature at which heat is transferred to the working fluid is given. It can be seen from Equation 4 that the lower the ratio of sink temperature to source temperature the higher the maximum attainable thermal energy will be. The sink temperature is usually outside of the engineers control but is unlikely to be far outside the range of 280K-310K (7°C-37°C). The source temperature is dependent on the design of the heating plant so the engineer can exercise some control over it. In principle the Carnot Cycle could be used for a steam power plant but in reality this is not the case as it has serious disadvantages such as having a low work ratio due to the large work input to the compressor, having serious practical difficulties associated with the compression process, and being limited to having a moderate thermal efficiency value as the critical point for steam occurs at a temperature of only 374°C, which is well below the metallurgical limit of around 600ºC -650ºC. Therefore, the Carnot Cycle is not used in practise. However, if the Carnot Cycle is reversed the system will be working as a heat pump cycle.

3.1.3 Reverse Carnot Cycle

Figure 9 shows a Reverse Carnot Cycle.
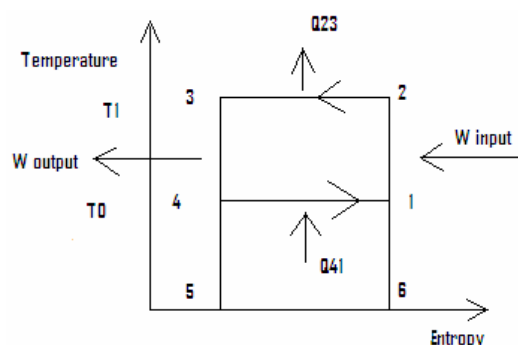
Figure 9: Refrigeration Cycle diagram & Reverse Carnot Cycle Diagram. Source: Herold et al, 1996. Absorption Chillers and Heat Pumps.

It can be seen that the direction of all the processes is reversed when compared to the Carnot Cycle.

The Coefficient of Performance (COP) describes the performance of a heat pump (or refrigerator) and is the ratio of the benefit obtained (amount of heat (Q) available at the high temperature), divided by the expenditure (net work, W).

$$COP_{HP} = \frac{|Q_{23}|}{|W_{12}| - |W_{34}|} = \frac{T_1 \Delta s}{(T_1 - T_0)\Delta s} = \frac{T_1}{T_1 - T_0}$$

Equation 5: (Heat Pump)

$$COP_R = \frac{|Q_{41}|}{|W_{12}| - |W_{34}|} = \frac{T_0 \Delta s}{(T_1 - T_0)\Delta s} = \frac{T_0}{T_1 - T_0}$$

Equation 6: (Refrigerator)

In both the heat pump and the refrigeration cycles it is desirable to have a high COP and the smaller the difference in temperatures between the reservoirs, the greater the COP will be.

3.1.4 Vapour Compression Chillers

Vapour Compression Chillers basically consist of a condenser, evaporator, compressor and expansion valve and a basic diagram of a refrigeration cycle used by a water cooled chiller is illustrated in Figure 10 along with a corresponding reverse Carnot Cycle.
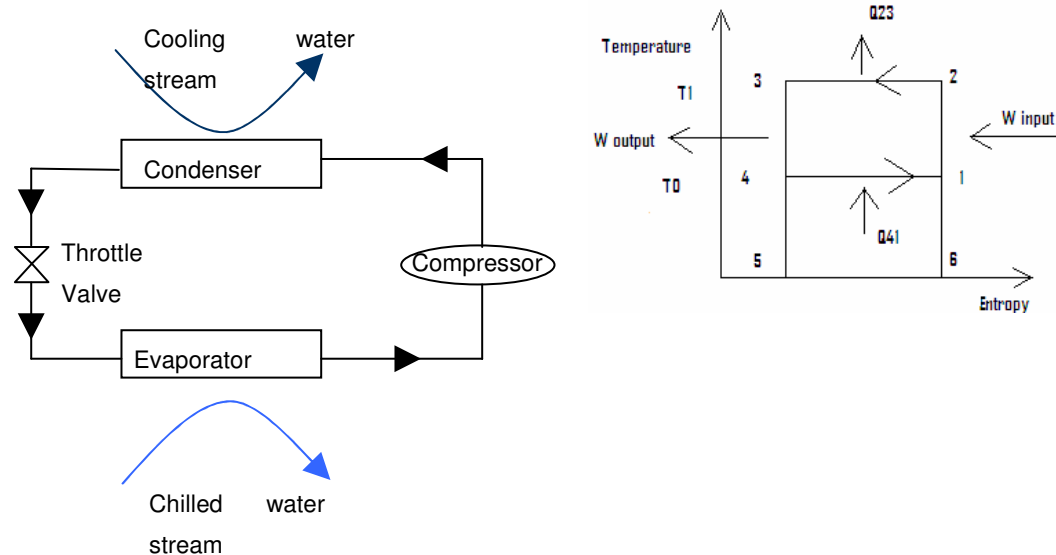
Figure 10: Refrigeration Cycle Diagram. Source: McLean R.C., 2006

Processes 1-2 and 3-4 are reversible, adiabatic and isentropic.
Processes 2-3 and 4-1 are reversible, isothermal and isobaric.

The refrigerant at point 1 has been evaporated to saturation point. Then, because of the Clausius statement, a compressor has to apply work to the fluid to raise the temperature so that net heat transfer can occur. The refrigerant, now at point 2, then passes through the condenser, where it is condensed back to a saturated liquid, point 3, and heat is rejected. From points 3-4 the refrigerant is expanded in the throttling valve before passing through the evaporator where heat from the surroundings is exchanged enabling the refrigerant to evaporate to the saturated vapour line. The cycle then repeats itself.

The compressor work is calculated by:

$$|W_{12}| = h_2 - h_1$$
<div align="right">Equation 7</div>

The condenser heating effect is:

$$|Q_{23}| = h_2 - h_3$$
<div align="right">Equation 8</div>

And the evaporator refrigeration effect is given by:

$$|Q_{41}| = h_1 - h_4$$
<div align="right">Equation 9</div>

Therefore the COP for a vapour compression refrigeration cycle is:

$$COP_R = \frac{Q_{41}}{W_{12}}$$
<div align="right">Equation 10</div>

It is desirable to have the chillers be able to extract as much heat as possible from its low temperature heat source, in this project the data centre, for a certain expenditure of work.

There are many variations of each of the components within a vapour compression chiller, and also variations on how they are combined, how they are controlled and in some cases enhancements to the components, such as multistage compressors to increase efficiency. The types of compressors chosen vary depending on the range of capacity. Small capacity sized chillers tend to use reciprocating piston and scroll compressors, intermediate capacity chillers use screw compressors and large size chillers tend to use turbo compressors. Turbo compressors are generally radial designs, typically referred to as centrifugal compressors, although axial designs are in limited use and are used for refrigerants that need extreme mass flow rates. Centrifugal compressors, referred to often as centrifugal chillers, are prevalent in large systems such as central and district systems, hospitals, hotels.

3.1.5 Cooling Towers and Dry Coolers

Vapour compression chillers require some form of cooling; this can be either air cooled or water cooled. Water cooled chillers incorporate the use of cooling towers and have a better thermodynamic effectiveness than air cooled chillers. This is because the heat rejection is at, or near, the air wet bulb temperature instead of the dry bulb temperature. Although water cooled chillers have a better thermodynamic effectiveness, there are health and safety issues regarding air borne diseases, such as legionnaires disease, therefore, for smaller applications, such as in this project, air cooled chillers are used, which use dry coolers for cooling the water on the condenser side. Dry coolers consist of fans that are able to increase or decrease their speed to cool the ambient air which passes over the condenser water, thus cooling it.

4   Case Study – Non Free Cooling and Free Cooling

4.1   Brief Outline of Case Study Data Centre

The case study chosen for this project was based on information kindly provided by WSP. The Data Centre is to be situated in Wales and is currently undergoing substantial restructuring and refurbishment, thus making it ideal for investigating how to make data centres more environmentally friendly. The main aim of the Data Centre is to collect and maintain important data about its customers. It relies heavily on computer systems to input and store all of its data and also relies on the internet for their customer services. It is therefore important that the infrastructure supporting the Data Centre runs effectively.

The dimensions and power consumption of the data centre will be used as inputs for the simulations in TRNSYS 16.

The assumptions made about the Data Centre are:

25 m length x 11 m width x 6 m height

Volume = 1650 m^3

Internal heat gains, excluding RACU fan power = All IT + lighting + miscellaneous = 740 kW.

4.2   Description of the Case Study model

Inside the Data Centre the electrical loads taken into account are made up of: 700 kW of IT, UPS losses of around 4% of the IT load (28 kW), and lighting and miscellaneous accounts for around 2% of the IT load (12kW).

All of these electrical loads are converted to heat within the data centre therefore the whole data centre produces 740 kW of heat, which is cooled by RACUs.   Figure 11 shows the typical power consumption of all the systems used in the initial TRNSYS 16 model, with no free cooling.
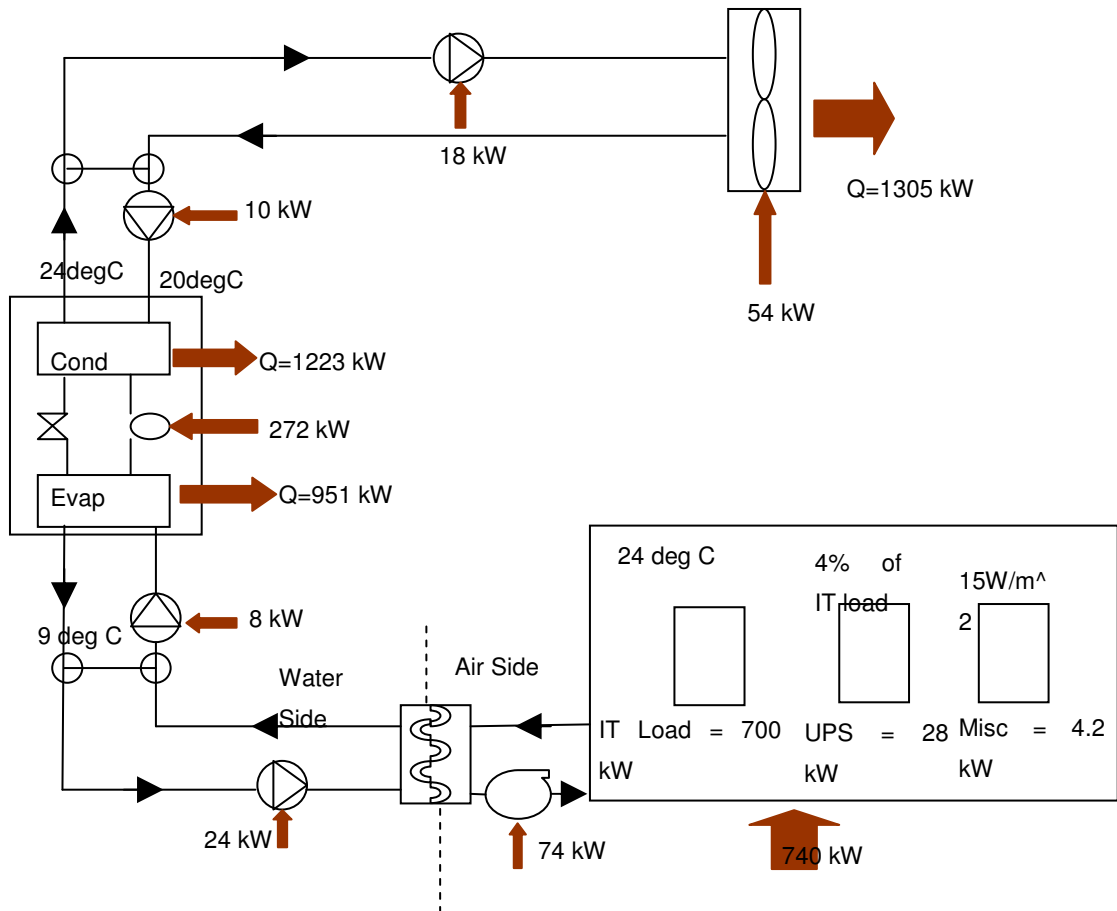
Figure 11: Initial system entered into TRNSYS showing power consumption

4.2.1 The Chillers used in the Case Study

The chillers used for the case study are vapour compression chillers. The choice of chillers to cool 740 kW is to have three chillers, two of which are capable of meeting the load, so each chiller is sized at 370 kW, giving a total of 1110 kW of capacity. This configuration is termed N+1. It is presumed that each chiller is served by a dry cooler, and they are laid out in parallel (Figure 12).
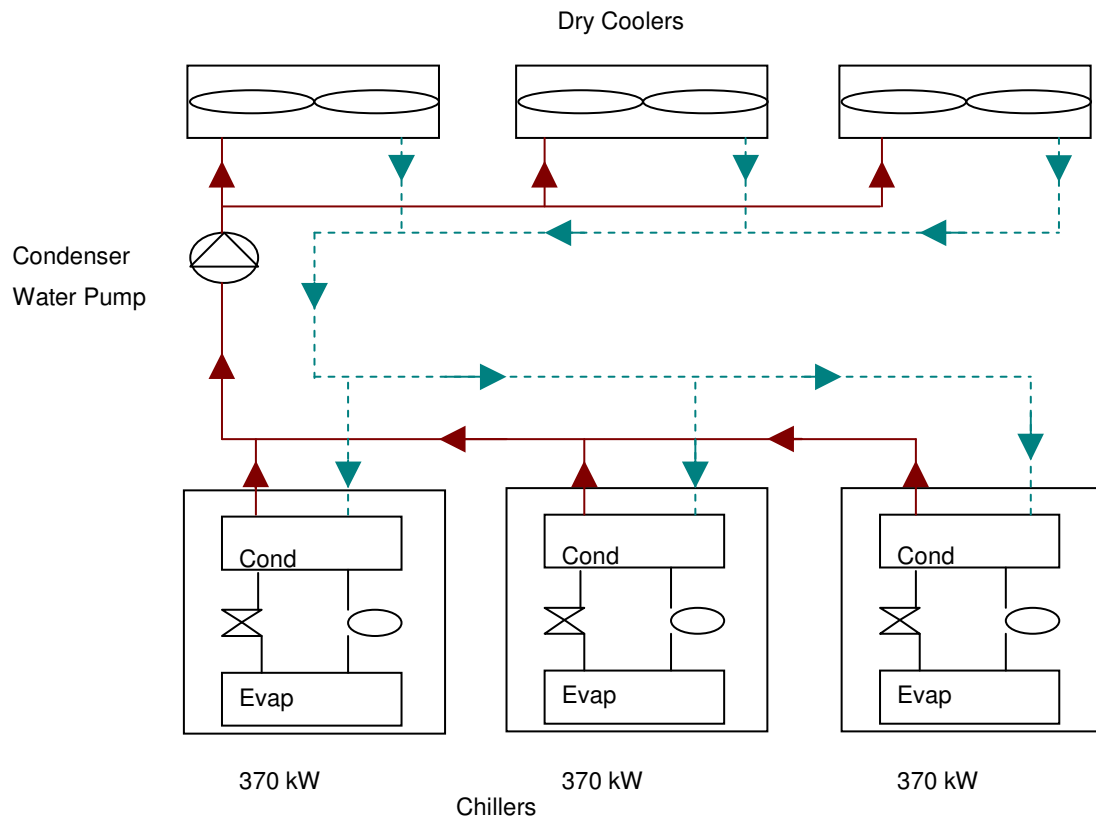
Figure 12: Diagram showing how chillers are connected to dry coolers

The condenser water pump is a constant volume pump that serves multiple dry coolers. The condenser in each of the chillers draws its own cooled water from the cooled water loop. The fans in the dry coolers are variable speed and each dry cooler has its own temperature control. Because the temperature controls are dependent on the ambient temperature, it is expected that all the dry coolers will be running at the same speed at any given time. The amount of dry coolers necessary is N+1.

Before continuing further with an in depth description of the case study the program TRNSYS 16 will be explained.

4.3   TRNSYS 16

TRNSYS 16 is a TRanNsient System Simulation program that is able to extensively simulate systems, including multi-zone buildings, and is used by engineers and researchers around the world to validate new energy concepts from simple domestic hot water systems to the design and simulation of buildings and their equipment and occupant behaviour as well as alternative energy systems such as wind, solar, photovoltaic, hydrogen.

TRNSYS 16 is made up of a suite of programs, the TRNSYS simulation studio the simulation engine (TRNDll.dll) and its executable (TRNExe.exe) and the building input data visual interface (TRNBuild.exe). The Editor is used to create stand alone redistributable programs known as TRNSED applications (TRNEdit.exe) (TRNSYS manual).

During this project only the TRNSYS 16 simulation Studio and the simulation engine were required for building the project and analysing the results.

The TRNSYS 16 simulation Studio is the main visual interface in which projects can be created by dragging and dropping components, from a menu, into the workspace. These components have global simulation parameters and inputs and outputs to which they can be connected to each other in order to build the model. Included in the simulation Studio is an output manager where the user is given control over which variables are integrated, printed and/ or plotted.

The simulation engine reads all the information on the simulation, which is known as an input file, and creates output files. It is called by an executable program, TRNExe.exe, which also implements an online plotter. This online plotter enables the user to view a number of output variables, which are user defined, during a simulation.

In this project TRNSYS 16 will be used to simulate three scenarios for cooling a data centre; cooling using vapour compression chillers (also referred to as electric chillers), cooling using vapour compression chillers with free cooling, and cooling using combined cooling heat and power (CCHP).

4.4 The TRNSYS 16 Model

4.4.1 The Building in TRNSYS 16

The building was modelled in TRNSYS 16 as a lumped capacitance building (Type 88) with internal gains. This means that the structure is subject to internal gains but the building loss coefficient is negligible, i.e., loads such, as solar gains, and losses, through fabric etc are taken as negligible. It was decided to model the building in this way because of the speed in which heating and/or cooling loads could be added to the simulation.

4.4.2 The Chillers in TRNSYS 16

For ease the chillers were modelled as one large water cooled chiller (Type 666), capable of meeting the load. The chiller uses performance data for the Carrier 19XR 1407kW (400 ton) Centrifugal VSD R-134a and has a rated power of 1.4 MW, although this is a small overestimate of the actual size required, TRNSYS 16 scales this performance data down, so the simulations actually have the correct installed capacity for the chillers. The rated COP is 4.59 and it is preferable to have the cooling water temperature returning to the chillers at a constant 20°C.   More information about the chiller used in the TRNSYS 16 model can be found in Appendix 1.

4.4.3 The Chiller and Dry Cooler in TRNSYS 16

Heat is rejected to the atmosphere, with help from the dry coolers, from the water of condenser of the chiller.   This water is termed cooling water, and the condenser needs a constant cooling water temperature of 20ºC. The diagram below illustrates how this is modelled in reality. This model was then translated into TRNSYS 16.
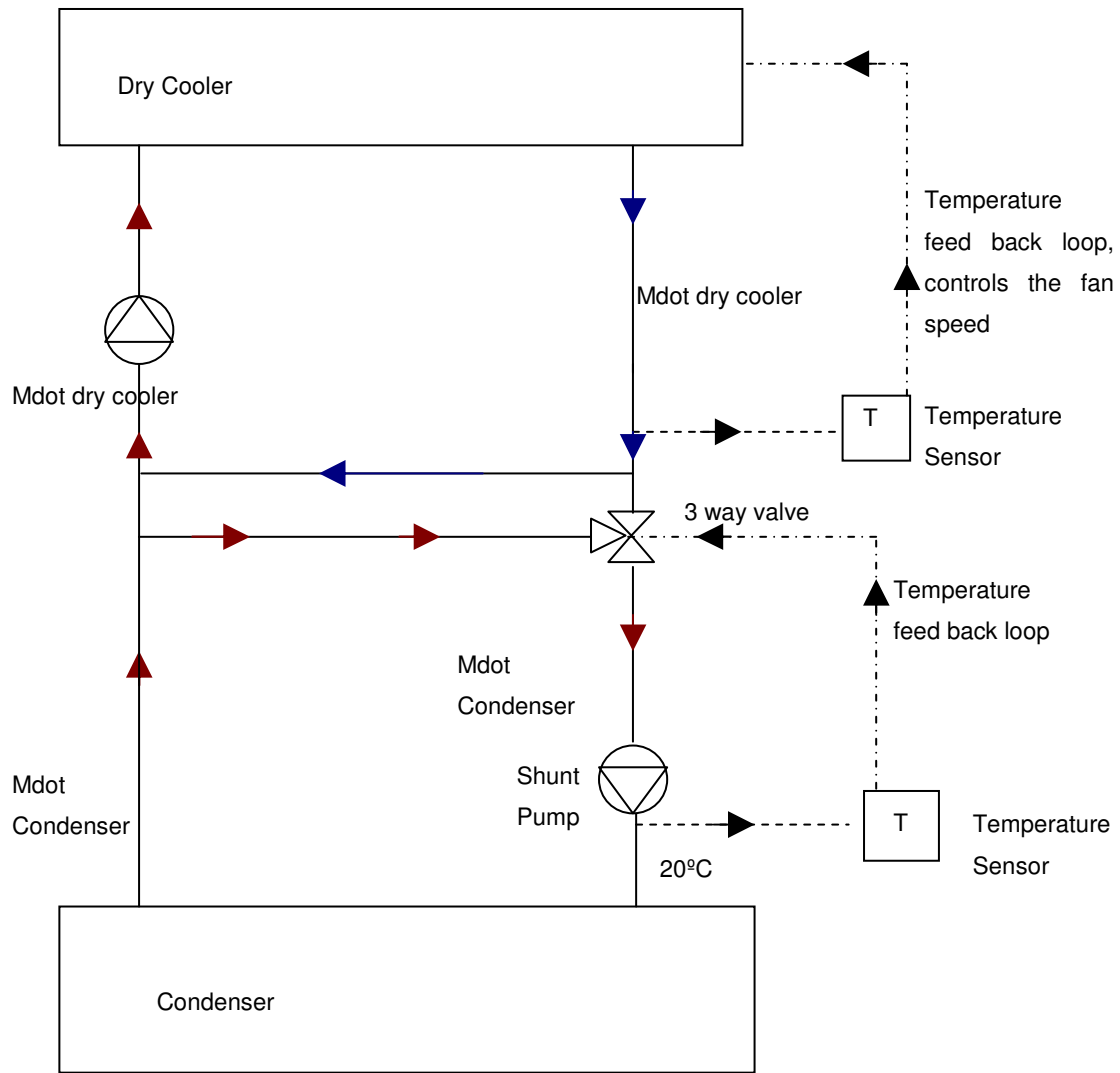
Figure 13: Basic diagram of Dry Cooler and Condenser

It can be seen from the diagram that there are essentially two loops. The first loop is the condenser loop, ensuring that the temperature entering the condenser is maintained at 20ºC. The second loop is the dry cooler loop which circulates water through the dry cooler. The temperature in both loops is monitored, the first temperature sensor makes sure that if the temperature rises above 20ºC in the condenser loop, the three way valve allows cooled water from the dry cooler to mix with the condenser water and cool it. The second temperature sensor controls the speed of the dry cooler fans, and hence the power consumed. The mass flow rates in the dry cooler and the condenser will be similar to each other as both the shunt pump and the main condenser water pumps are constant speed pumps.

The dry cooler is modelled as a dry fluid cooler (Type 511) in TRNSYS 16 and is based on the Leibert range (10 Fan Dry Cooler/ Fluid Cooler). The design inlet and

outlet air temperatures are 48.9ºC and 42.6 ºC respectively, assuming a ΔT of 6.3 ºC. The design parameters for the dry cooler can be found in Appendix 2.

4.4.4 The Room Air Conditioning Unit in TRNSYS 16

The whole data centre is cooled by RACUs, which must provide 740kW of total useful cooling. The RACUs were modelled as a single equivalent unit consisting of: -

- The cooling coil, which is modelled using a counter flow heat exchanger (Type 5). The UA value of the heat exchanger is dependent on the water flow rate and the equation for adapting the UA value was obtained using manufacturer data.

- A variable speed fan (Type 111).

The hot air from the data centre passes over the heat exchanger, containing the chilled water from the chillers, and is cooled down. This cool air is then blown, by the fans, up through air vents in the raised floor, to cool the data centre equipment.

Below is a simple diagram of the RACUs and air flow inside the data centre, and the water leaving and returning to the chillers.
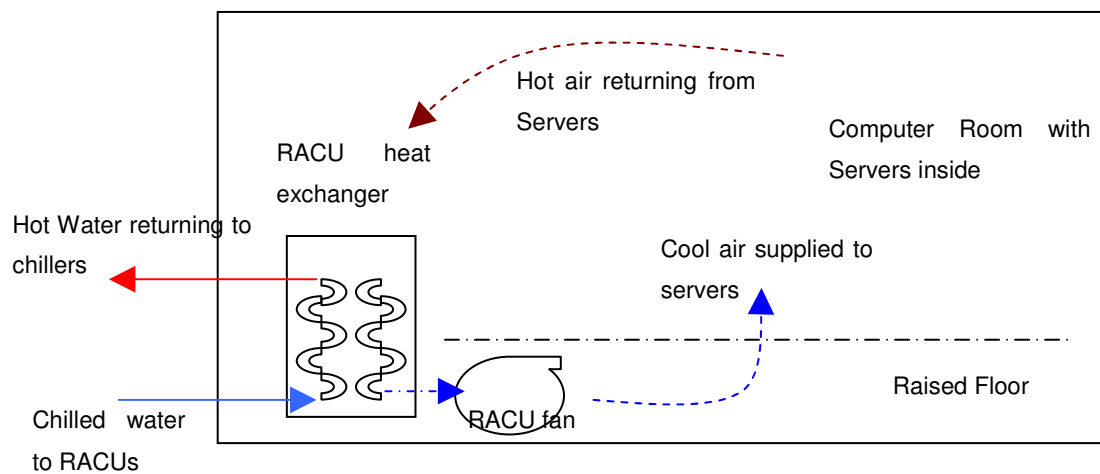
Figure 14: RACUs and RACU fans inside the data centre

In order to control the chilled water flowrate to the heat exchanger of the RACU a PID controller (Type 23) acts on a chilled water pump, modelled as a variable speed pump (Type 110). The pump has a rated flowrate of 135023.7 kg/hr = 37.5 kg/s, and a rated power (at full flowrate) of 31.5 kW, which adds 0.2°C to the chilled water temperature (in part-load conditions the power used by pumps is proportional to the cube of the flowrate so the power drops drastically and the ΔT added by the pump drops as well).

45

4.4.5 Other component in TRNSYS 16

Another important component that the simulation relies on is the weather data processor that reads a weather data file.

In order to link the main components of the data centre system together, mixers, controllers and equation blocks were required.

The equation blocks give the user the ability to define equations within the input file that are not in a component. The equations can then be used as inputs to other components, or as parameters, initial values of inputs, or derivatives if the equations are time independent (TRNSYS 16 manual). One main equation block was used in the simulation to make changing certain parameters easier and is especially useful for enabling free cooling in one easy step.

The output files from TRNSYS 16 were put into Excel so as to be analysed more in depth and to generate graphs.

5 Methodology for Investigating Improvements on Current Technology used in Data Centres

To investigate methods for improving energy efficiency in data centres using current technologies, i.e., electric chillers, a number of steps were taken: -

- A simulation of the full system without free cooling was undertaken.
- Simulations of the full system with differing chilled water inlet temperatures were run.
- Simulations of the full system with free cooling were run for different chilled water inlet temperatures.

5.1 Model of full system with and without free cooling

Current practice in a data centre is to have chilled water temperatures supplied at 7°C and a data centre room temperature set at 24°C. In order to quantify energy savings it was necessary to change both the chilled water set point temperature and the data centre room temperature. The chilled water set point temperatures were chosen as 7°C, to show the energy use of current practices, 9°C to give an idea of energy performance whilst still being able to maintain the data centre room temperature at 24°C. The chilled water set point was then increased to 11°C and it was necessary to increase the data centre room temperature by 2°C. This was done because it has been commented on, in the literature review, that energy savings could be made if the temperature inside the data centre is permitted to rise above 24°C.

When carrying out simulations of free cooling it was necessary to set an ambient temperature setpoint at which free cooling would be switched on , i.e., by letting the cooling water from the dry cooler pass through the heat exchanger before reaching the condenser of the chiller, essentially bypassing the chiller. In TRNSYS 16 this setpoint ambient temperature tells the program when to turn on free cooling, so if the ambient temperature is less than the setpoint ambient temperature, free cooling will be turned on. For each simulation the first ambient temperature at which the heat exchanger loop is opened is set at 1°C below the chilled water temperature, as full free cooling will occur at this temperature. The setpoint ambient temperature at which free cooling is switched on was raised by 4°C to allow full and partial free cooling to occur. Increments of 4°C were chosen as this seemed a reasonable amount at which to increase the temperature in order to obtain sensible results. The

temperature of the cooling water setpoint is the temperature of the water leaving the dry cooler when free cooling is enabled. It is set at 1°C less than the chilled water setpoint to ensure that the dry coolers will cool the water sufficiently to allow full free cooling.

Simulations of the full system (dry cooler, chillers, RACUs, data centre) with and without free cooling were carried out for different temperatures of chilled water set point and data centre room temperature. These temperatures are shown in Table 1:

Scenario	Chilled Water Setpoint (TChW) °C	Data Centre Room Temperature (TRoom) °C	Ambient Temperature Set point (used for Free Cooling) °C	Temperature of cooling water setpoint (leaving the dry cooler) °C
Scenario 0-1 (Standard Practice)	7	24	n/a	n/a
Scenario 0-2	9	24	n/a	n/a
Scenario 0-3	11	26	n/a	n/a
Scenario 1	7	24	6	6
Scenario 2	7	24	10	6
Scenario 3	7	24	14	6
Scenario 4	9	24	8	8
Scenario 5	9	24	12	8
Scenario 6	9	24	16	8
Scenario 7	11	26	10	10
Scenario 8	11	26	14	10
Scenario 9	11	26	18	10

Table 1: List of temperatures used in TRNSYS 16 simulations

Figure 15 shows a simplified version of how the dry cooler interacts with the chillers and the free cooling heat exchanger and in turn how these components interact with the RACUs inside the data centre. The model without free cooling simply omits the free cooling heat exchanger from it.
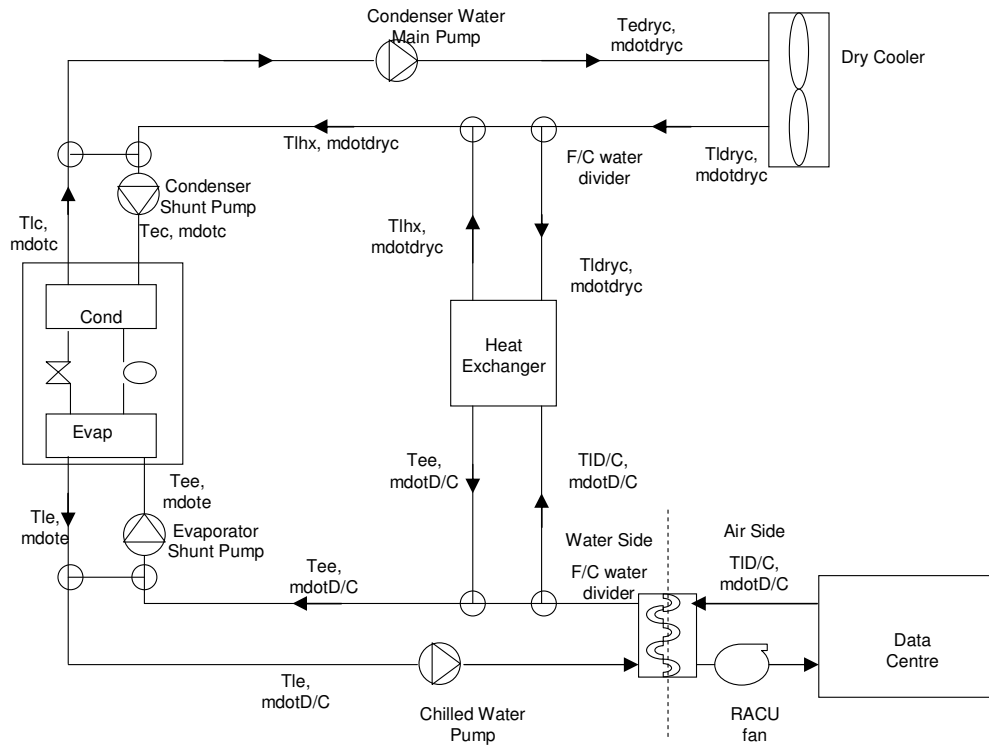
Figure 15: Simple model of data centre with chillers and free cooling

It can be seen that this model includes:

- Pumps and shunt pumps for increasing or decreasing the flow rate in the pipes and for circulating the water in the pipes.
- Pipe/ duct tee pieces, mixers and diverters for diverting or splitting the fluid flow, such as when the flow is to be diverted from the chiller and through the free cooling heat exchanger.
- Controllers for maintaining a control variable, in this case the data centre room temperature, the cooling water temperature and the speed of the dry cooler fan.

The model without free cooling simply omits the heat exchanger.

6 Results Analysis for Full System with no Free Cooling Allowance

Provided below is a recap of the Scenarios used in the TRNSYS 16 simulations.

Scenario	Chilled Water Setpoint (TChW) ºC	Data Centre Room Temperature (TRoom) ºC
Scenario 0-1 (Standard Practice)	7	24
Scenario 0-2	9	24
Scenario 0-3	11	26

6.1 COP analysis for system with no free cooling

Below are graphs showing the Chiller COP and Full System COP variation over the year for the three different scenarios.
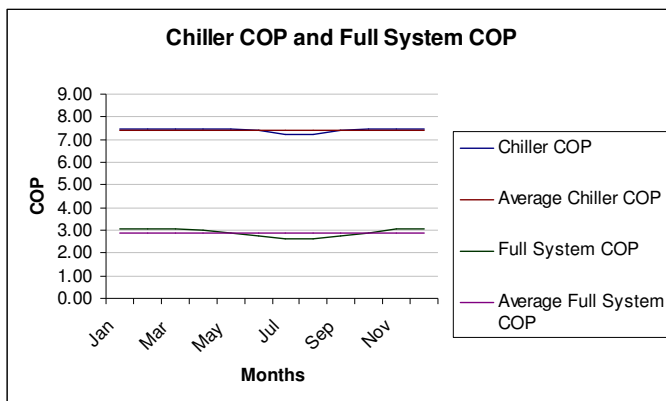
Scenario 0-1: Chilled water set point = 7 ºC
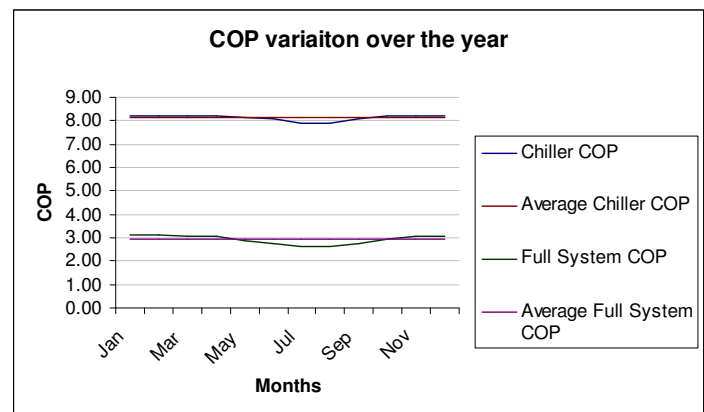
Scenario 0-2: Chilled water set point = 9 ºC
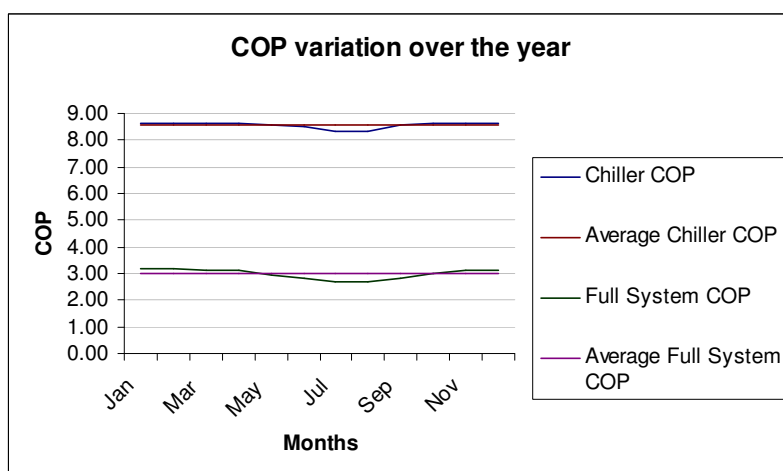
Scenario 0-3: Chilled water set point = 11 ºC, data centre room temperature = 26 ºC

Figure 16: COP variation of full system with no free cooling

There is a significant difference between the Chiller COP and Full System COP. This is because the chiller COP considers only the Chiller, whereas the Full System COP takes into consideration the Chiller and the HVAC system. As illustrated in Figure 17.
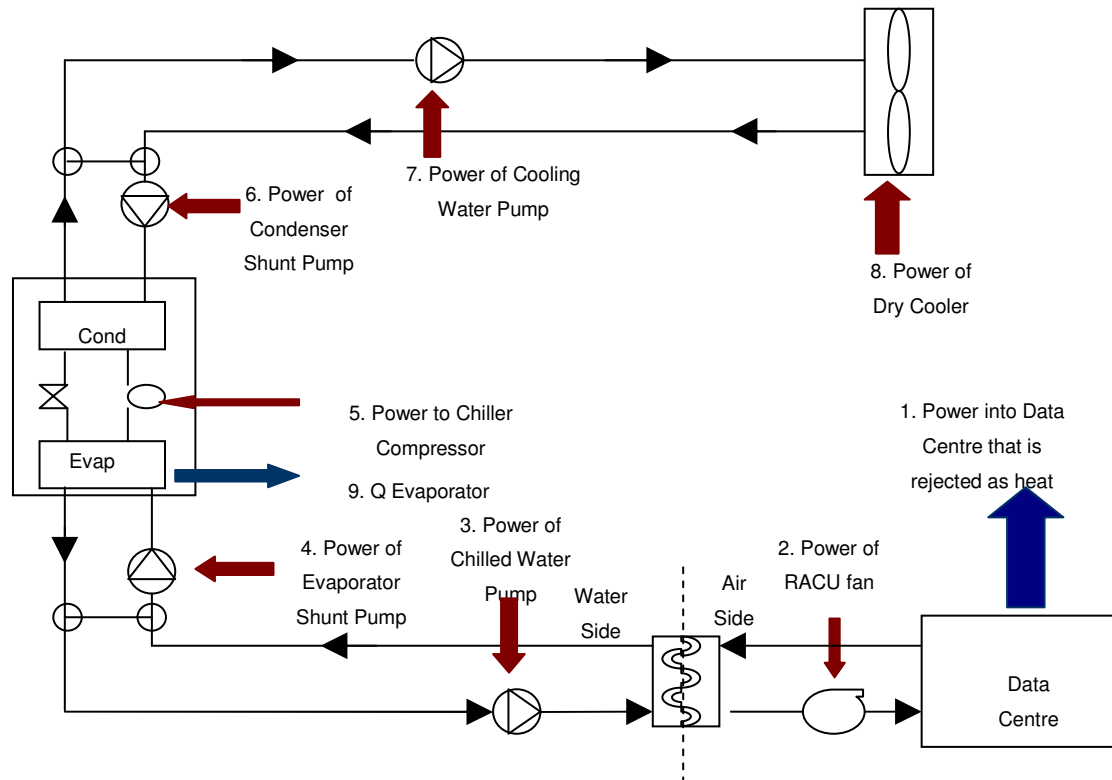
Figure 17: COP Diagram of System

Using Figure 17, the Chiller COP was worked out by:

$$COP = \frac{QEvaporator}{Powertocompressor} = \frac{No.9}{No.5}$$
Equation 11

Again using Figure 17 the Full System COP was calculated by:

$$COP = \frac{PowerinDataCentre}{PowerusedbyallHVAC} = \frac{No.1}{Nos.2+3+4+5+6+7+8}$$
Equation 12

It can be seen that the COP of the chiller increases as the temperature of the chilled water increases. It should also be noted that the COP decreases in the summer months, when the chillers have to work harder to provide cooling for the data centre.

6.2 Power breakdown and CO_2 analysis of the systems with no free cooling

Figure 18 shows the power breakdown of the system in the three different scenarios and Table 2 shows the annual energy consumed and the corresponding tonnes of CO_2 released by the chillers over the period of a year. Table 3 shows the annual energy consumed and the corresponding tonnes of CO_2 released by the system over the period of a year.
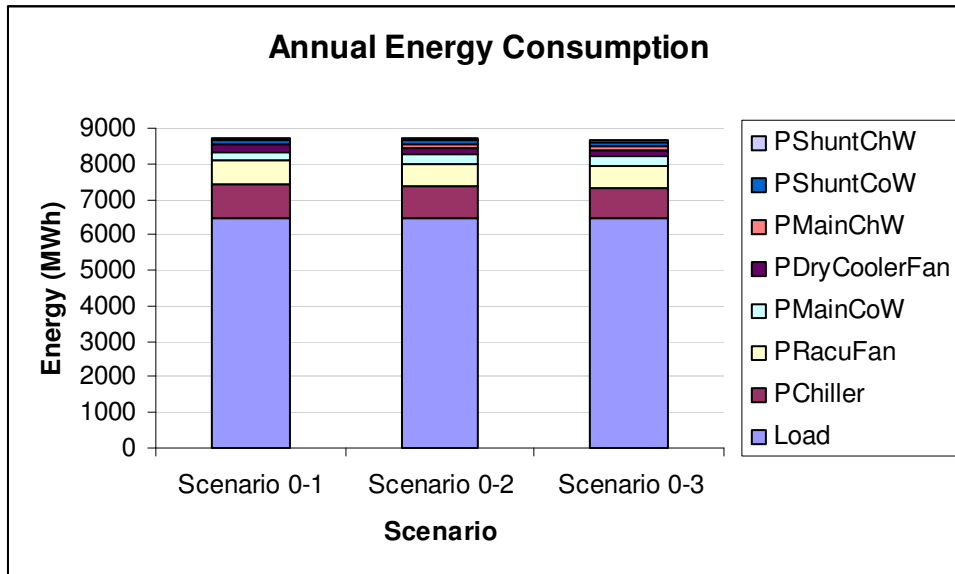
Figure 18: Annual Energy Consumption of system with no free cooling

Where:-

Load = Power consumed by the IT equipment, UPS and Miscellaneous

PChiller = Power consumed by the chiller

PDryCoolerFan = Power consumed by the dry cooler fan

PMainChW = Power consumed by the main chilled water pump

PMainCoW = Power consumed by the main cooling water pump

PShuntCoW = Power consumed by the chilled water shunt pump

PRacuFan = Power consumed by the RACU fan

PShuntChW = Power consumed by the chilled water shunt pump

Annual Carbon Emissions of the Chiller			
	Energy Consumed by Chiller (MWh)	Tonnes CO_2/year	CO_2 Saving
Scenario 0-1	2241	965	
Scenario 0-2	2228	960	0.5%
Scenario 0-3	2183	940	2.6%

Table 2: Chiller CO_2 savings for system with no free cooling

Annual Carbon Emissions for the full system			
	Total Energy Consumed (MWh)	Tonnes CO_2/year	CO_2 Saving
Scenario 0-1	8720	3750	
Scenario 0-2	8710	3745	0.14%
Scenario 0-3	8670	3725	0.66%

Table 3: CO_2 savings for system with no free cooling

CO_2 analysis was performed using: -

CO_2 emitted from Gas = 0.19 kg CO_2/ kWhr (Carbon Trust)

CO_2 emitted from Electricity = 0.43 kg CO_2/ kWhr (Carbon Trust). Note that the carbon emissions for electricity and gas, used throughout this thesis, are taken from the Carbon Trust. The Carbon Trust, 2007, states;

"In the case of electricity consumption it is in most cases impossible to identify with any certainty the source of generation that has been reduced or displaced. For this reason the Climate Change Levy Negotiated Agreements and the ETS are using an average carbon intensity factor for the estimation of carbon dioxide emissions from the consumption of electricity. This has been fixed at 0.43 kgCO_2/kWh."

The CO_2 savings made by the chiller are larger than the CO_2 savings made by the full system. This is because the full system includes the IT load, which is very large and not influenced by the HVAC system.

In both tables it can be seen that Scenario 0-3 consumes the least energy of the three scenarios. The second best scenario for energy consumption is Scenario 0-2 using less energy than Scenario 0-1. The power of the chiller decreases as the chilled water set point increases, as does the power of the dry cooler. The power of the chiller decreases because the temperature difference between the chilled water temperature returning to the evaporator and leaving the evaporator has decreased (i.e., ΔT). It can also be seen that when the chilled water temperature is increased by 2°C, CO_2 savings of less than 0.14% are given. However, if the chilled water temperature and room temperature of the data centre are increased to 11°C and 26°C respectively, then the increase in CO_2 savings is four times as much, at 0.66%. Table 4, below, quantifies the energy consumed in monetary terms, using the cost of electricity as £0.07 pence per kWh (Supplied by WSP).

	Total Energy Consumed (MWh)	Cost per year of electrical power	Saving
Scenario 0-1	8720	£611,000	Between Scenario 0-1 and Scenario 0-2: £1,000
Scenario 0-2	8710	£610,000	Between Scenario 0-2 and Scenario 0-3: £3,000
Scenario 0-3	8670	£607,000	Between Scenario 0-1 and Scenario 0-3: £4,000

Table 4: Cost Analysis of System with no free cooling Energy Use

The saving between Scenario 0-2 and Scenario 0-1 is minimal at £1,000. The biggest savings are made between the Scenario 0-3 and Scenario 0-1. There is also a significant saving made between Scenario 0-3 and Scenario 0-1.

Thus, it can be concluded that energy, carbon emissions and cost savings are made when the temperature of the chilled water setpoint is increased from 7°C. However, if the setpoint is increased to 11°C then the temperature of the data centre has to be increased, as the chilled water is unable to maintain the room temperature at 24°C. In practice raising the data centre temperature above 24°C may cause problems with the servers and thus the energy saving potential of increasing the water setpoint to 11°C may not be taken up.

6.3 Comparison of winter and summer days for systems with no free cooling

Having established that the best case scenario for energy saving was by increasing the chilled water temperature, the performance of the system during a winter and summer day is looked at using chilled water temperatures of 7°C and 9°C.
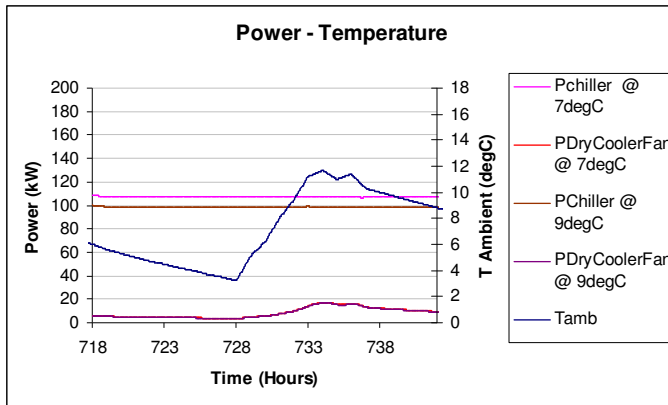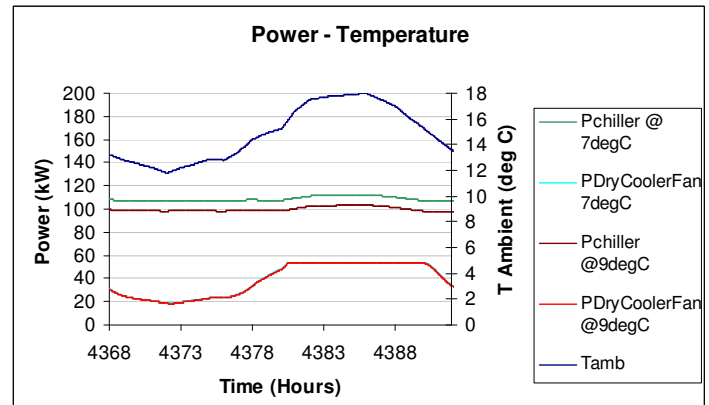
Winter   Summer

Table 5: Winter and Summer Comparison of no free cooling

From these graphs it can be seen that the power of the dry cooler doubles its power, until it reaches maximum power, in the summer day. This is because the dry cooler has to work harder to maintain the temperature entering the condenser at 20°C, as explained earlier using Figure 13: Basic diagram of Dry Cooler and Condenser. As the ambient temperature increases in the summer so too does the power of the chiller. During the winter and summer months the power of the chiller in both cases remains fairly constant. However, it can be seen that the power of the chiller at 7°C is higher than the power of the chiller at 9°C. The reason for this is; the temperature lift that the chillers see between the hot source (condenser side of the chiller, assumed to be 20°C) and the cold source (evaporator side of the chiller), increases as the chilled water setpoint (temperature entering the evaporator) decreases. The chiller, like all refrigeration machines, uses more energy as the temperature lift increases between the hot temperature source and the cold temperature source.

7   Results for Full System with free cooling

Below is a recap of the Scenarios used during the TRNSYS 16 simulations.

Scenario	Chilled Water Setpoint (TChW) ºC	Data Centre Room Temperature (TRoom) ºC	Ambient Temperature Setpoint (used for Free Cooling) ºC	Temperature of cooling water (leaving the dry cooler) ºC
Scenario 1	7	24	6	6
Scenario 2	7	24	10	6
Scenario 3	7	24	14	6
Scenario 4	9	24	8	8
Scenario 5	9	24	12	8
Scenario 6	9	24	16	8
Scenario 7	11	26	10	10
Scenario 8	11	26	14	10
Scenario 9	11	26	18	10

7.1   Energy Consumption and CO_2 Emissions Comparison

The results in this section show the energy consumption and the CO_2 emissions for chilled water setpoints of 7°C and 9°C and the corresponding setpoint ambient temperature at which to switch on free cooling. The results for chilled water setpoint of 11°C can be found in Appendix 4, it has been decided not to show these results in the main body of the report because the temperature of the data centre would have to be raised to 26°C which at this present time causes controversy within the industry due to the sensitivity of computer servers to changes in temperature. It should be

noted that the load of the Data Centre remains constant throughout the scenarios at 6482.4 MWh/year.

Chilled Water Setpoint = 7°C
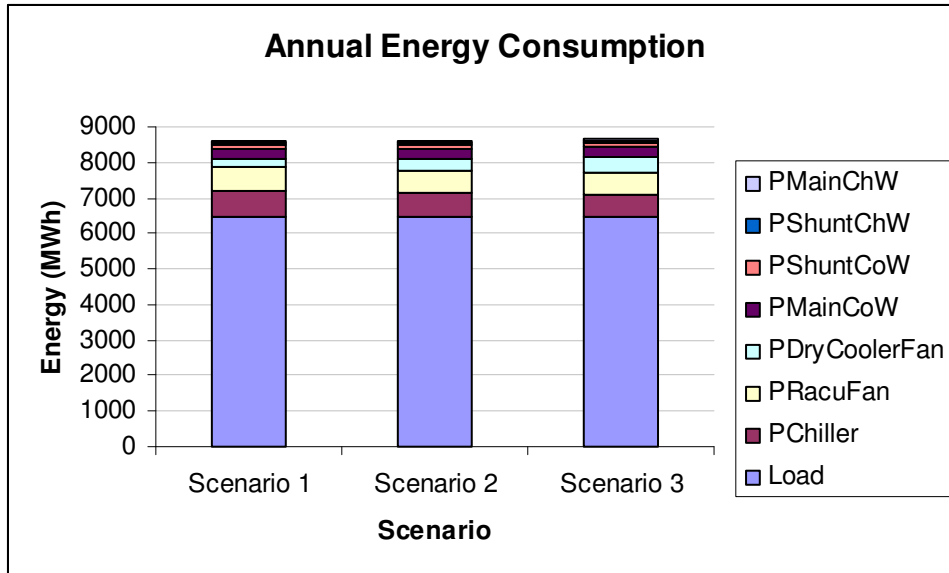
Figure 19: Annual Energy Consumption with free cooling, Chilled Water Setpoint = 7°C

Where:-

Load = Power consumed by the IT equipment, UPS and Miscellaneous

PChiller = Power consumed by the chiller

PDryCoolerFan = Power consumed by the dry cooler fan

PMainChW = Power consumed by the main chilled water pump

PMainCoW = Power consumed by the main cooling water pump

PShuntCoW = Power consumed by the chilled water shunt pump

PRacuFan = Power consumed by the RACU fan

PShuntChW = Power consumed by the chilled water shunt pump
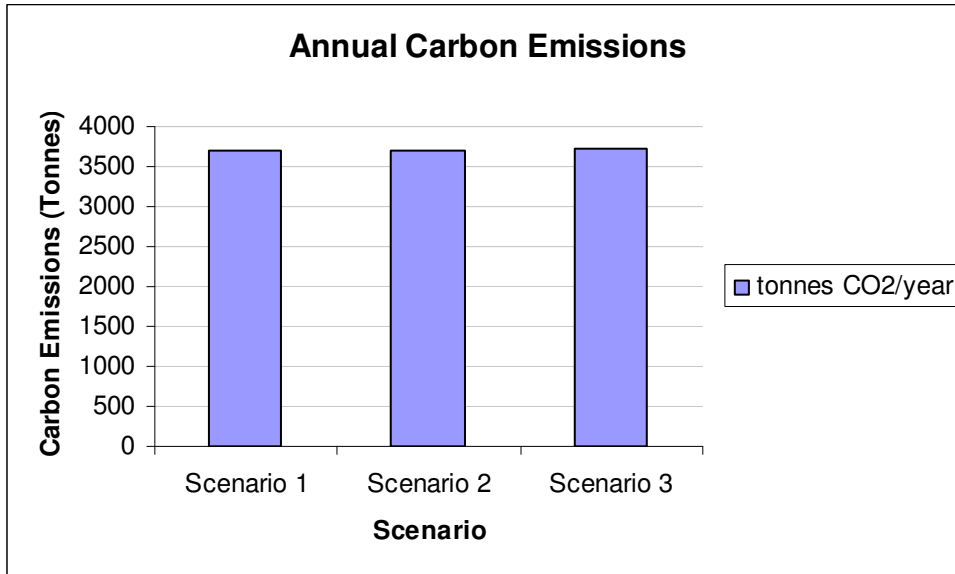
Figure 20: Annual Carbon Emissions with free cooling, Chilled Water Setpoint = 7°C

Annual Carbon Emissions			
	MWh(elect)/year	Tonnes CO_2/year	CO_2 Saving
Scenario 1	8605	3700	
Scenario 2	8600	3698	0.05%
Scenario 3	8640	3715	-0.4 %

Table 6: Annual Energy Consumption and Carbon Emissions for Chilled Water Setpoint = 7°C

CO_2 analysis was performed using: -

CO_2 emitted from Gas = 0.19 kg CO_2/ kWhr (Carbon Trust)

CO_2 emitted from Electricity = 0.43 kg CO_2/ kWhr (Carbon Trust). Note that the carbon emissions for electricity and gas, used throughout this thesis, are taken from the Carbon Trust. The Carbon Trust, 2007, states;

"In the case of electricity consumption it is in most cases impossible to identify with any certainty the source of generation that has been reduced or displaced. For this reason the Climate Change Levy Negotiated Agreements and the ETS are using an average carbon intensity factor for the estimation of carbon dioxide emissions from the consumption of electricity. This has been fixed at 0.43 kgCO_2/kWh."

It can be seen that the best scenario for free cooling with a chilled water setpoint of 7°C is Scenario 2, where free cooling is turned on at an ambient temperature of 10°C. This is because for total free cooling the ambient temperature needs to be 6°C. However, there are fewer occasions when the ambient temperature is 6°C than when it is 10°C. Therefore, when the ambient temperature is set to 10°C, partial free cooling is available, thus the chillers don't need to work as much. When the ambient temperature is set to 14°C the dry cooler fan has to use more power to cool the water

to 6°C, thereby negating any use from partial free cooling. This is illustrated in the table below.

	Energy Consumption of Chiller (MW/h)	Energy Consumption of Dry Cooler Fan (MW/h)
Scenario 1	743	261
Scenario 2	636	364
Scenario 3	604	436

Table 7: Energy consumption of chiller and dry cooler at 7°C

From the table it can be seen that as the chilled water setpoint increases, the dry cooler fan energy consumption increases. At 14°C (Scenario 3), the energy consumption of the dry cooler fan outweighs the energy savings made by the chiller.

Chilled Water Setpoint = 9°C
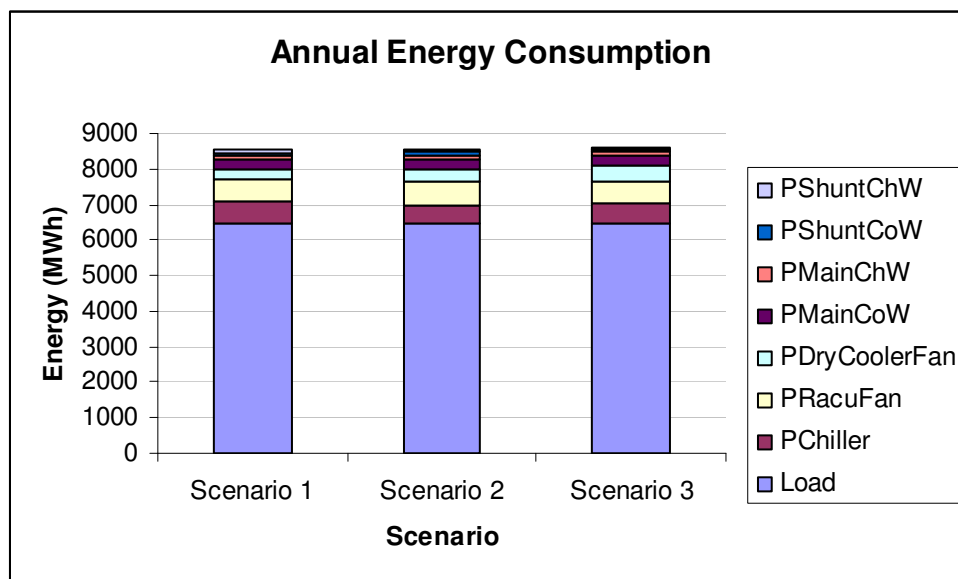
Figure 21: Annual Energy Consumption with free cooling, Chilled Water Setpoint = 9°C

Where:-

Load = Power consumed by the IT equipment, UPS and Miscellaneous

PChiller = Power consumed by the chiller

PDryCoolerFan = Power consumed by the dry cooler fan

PMainChW = Power consumed by the main chilled water pump

PMainCoW = Power consumed by the main cooling water pump

PShuntCoW = Power consumed by the chilled water shunt pump

PRacuFan = Power consumed by the RACU fan

PShuntChW = Power consumed by the chilled water shunt pump
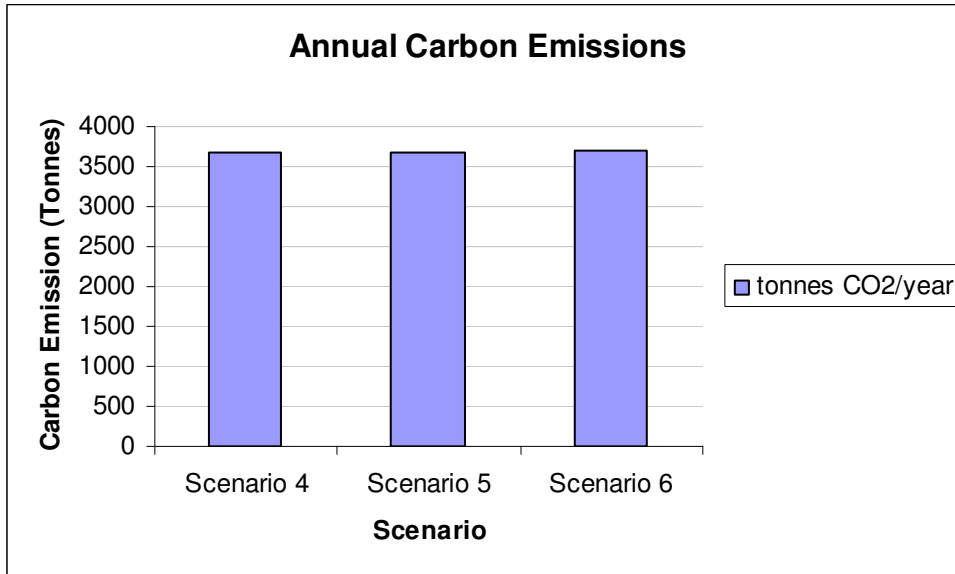
Figure 22: Annual Carbon Emissions with free cooling, Chilled Water Setpoint = 9°C

Annual Carbon Emissions			
	MWh(elect)/year	Tonnes CO_2/year	CO_2 Saving
Scenario 4	8530	3670	
Scenario 5	8560	3680	-0.3%
Scenario 6	8630	3710	-1%

Table 8: Annual Energy Consumption and Carbon Emissions for Chilled Water Setpoint = 9°C

CO_2 analysis was performed using: -

CO_2 emitted from Gas = 0.19 kg CO_2/ kWhr (Carbon Trust)

CO_2 emitted from Electricity = 0.43 kg CO_2/ kWhr (Carbon Trust). Note that the carbon emissions for electricity and gas, used throughout this thesis, are taken from the Carbon Trust. The Carbon Trust, 2007, states;

"In the case of electricity consumption it is in most cases impossible to identify with any certainty the source of generation that has been reduced or displaced. For this reason the Climate Change Levy Negotiated Agreements and the ETS are using an average carbon intensity factor for the estimation of carbon dioxide emissions from the consumption of electricity. This has been fixed at 0.43 kgCO_2/kWh."

From these results it can be seen that the best temperatures to have chilled water setpoint and ambient temperature setpoint, yielding the least energy consumed a year and therefore producing the least CO_2 emissions, are; Chilled Water Setpoint = 9℃ and ambient temperature setpoint at which free cooling is turned on is 8℃. During the simulations it was found that the optimum ambient temperature set point was 10℃ for chilled water setpoints of 7℃ and 11℃. It can therefore be suggested that increasing the ambient temperature setpoint at which free cooling is switched on from 8℃ to 10℃, for the chilled water setpoint of 9℃, would be beneficial, to allow for some free cooling within the system.

7.2 COP variation over the year for the system with free cooling

Having run a number of simulations for different chilled water set points with corresponding ambient temperatures at which to turn on free cooling, it was found that the best results were achieved when the ambient temperature, at which free cooling was turned on was 10°C. The results of COP variation over the year for the three scenarios of differing chilled water setpoint and ambient temperature of 10°C are shown in this section.

Chilled Water Setpoint = 7°C, Data Centre Room Temperature = 24°C.
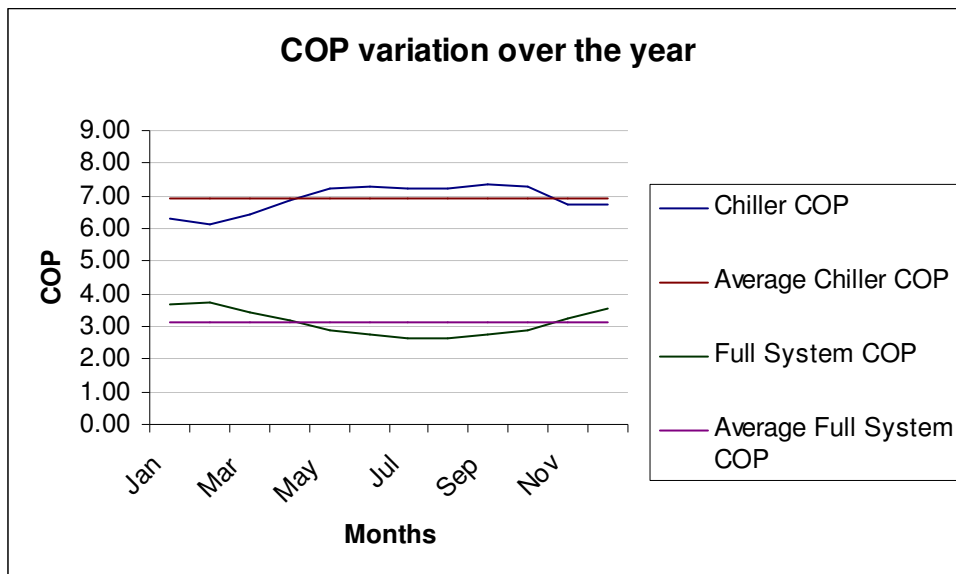
Figure 23: COP variation over the year for system with free cooling, TChW = 7°C

Chilled Water Setpoint = 9°C, Data Centre Room Temperature = 24°C.
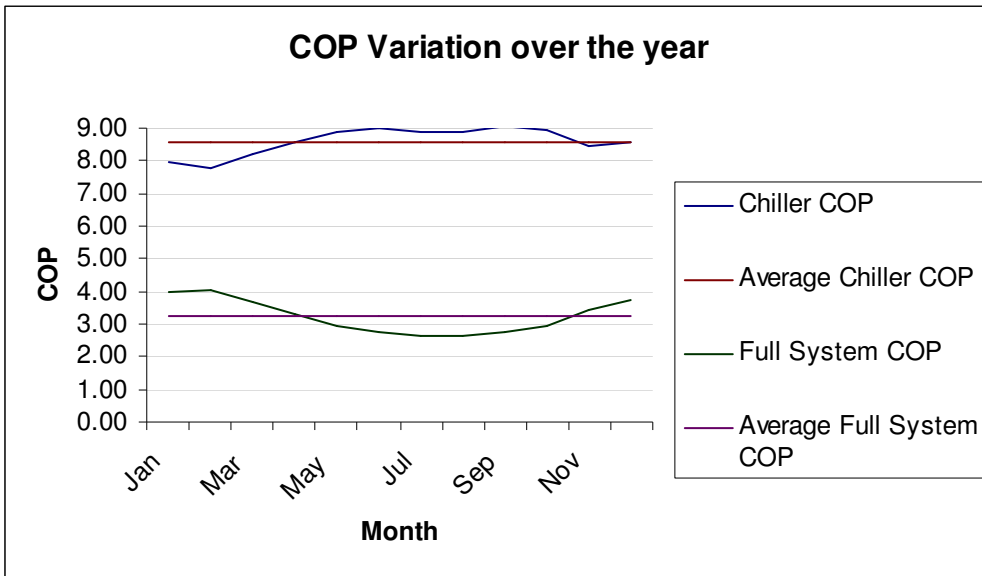
Figure 24: COP variation over the year for system with free cooling, TChW = 9°C

Chilled Water Setpoint = 11°C, Data Centre Room Temperature = 26°C,
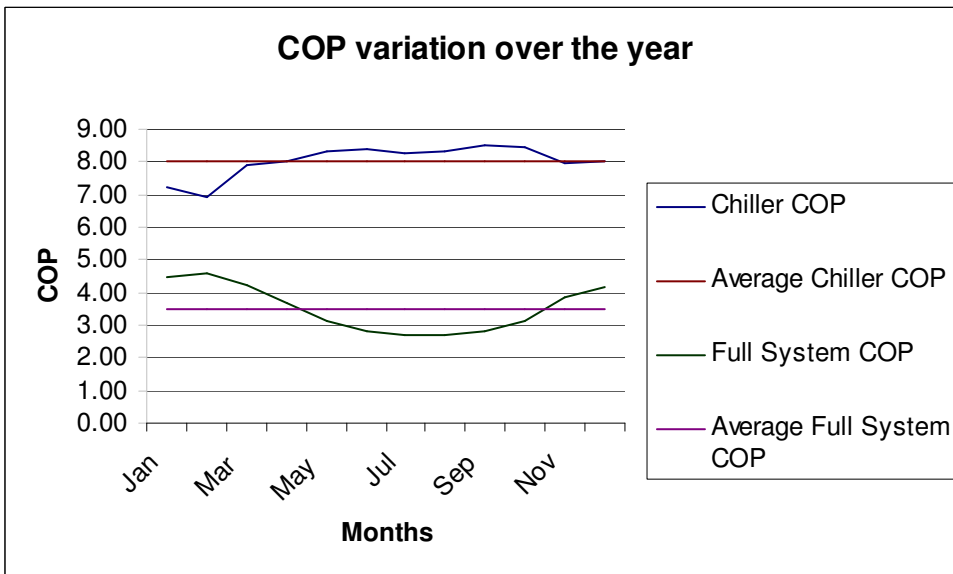
Figure 25: COP variation over the year for system with free cooling, TChW = 11°C

The chillers are "ramped down" and "ramped up" in the winter time depending on the cooling demand. As the chillers are ramped down, the part load ratio decreases, affecting the COP of the chillers. This makes the COP of the chillers low during the winter months. Therefore the part of the COP curve that is of most interest to this project is between the summer months of May to October. The system COP is high during the colder months and decreases in the warmer months (summer time).

Remembering that the

$$COP = \frac{PowerinDataCentre}{PowerusedbyallHVAC} = \frac{No.1}{Nos.2+3+4+5+6+7+8}$$

Equation 12.

In this equation the numbers represent the power consumption of parts of the HVAC system, and equation explains why the system COP is decided by the chiller power and the dry cooler fan power. During the winter months the power of the chiller is reduced due to free cooling and the power of the dry cooler is also lower, during the periods when free cooling is not turned on. Therefore, the system COP is higher in the winter. During the summer months, the dry cooler fan power and the chiller power at high, making the system COP lower during this period. It can be seen that the best chiller performance, based on the COP variation over the year, is achieved when the chilled water setpoint is 9°C and the ambient temperature at which free cooling is switched on is 10°C.

The full system COP variation over the year had the highest values for the chilled water temperature setpoint of 11°C. The second best COP performance was when the chilled water setpoint was set to 9°C.

It can therefore be concluded that the best temperature for energy savings and system performance is to have the chilled water setpoint at 9°C and the ambient temperature at which free cooling is turned on 10°C. It should also be noted that the most energy savings for all scenarios occurred when the ambient temperature at which free cooling was turned on was 10°C.

7.3 Comparison of winter and summer days for systems with free cooling

Having established that the most energy savings can be made for all scenarios when the ambient temperature at which free cooling is turned at is 10°C, a look at the performance of the system during a winter and summer day for a chilled water setpoint of 7°C is looked at. This chilled water temperature was chosen as it is the temperature currently used in data centres.
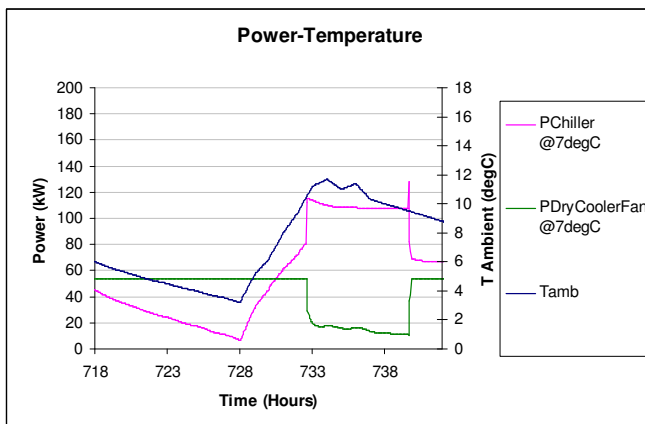
Winter
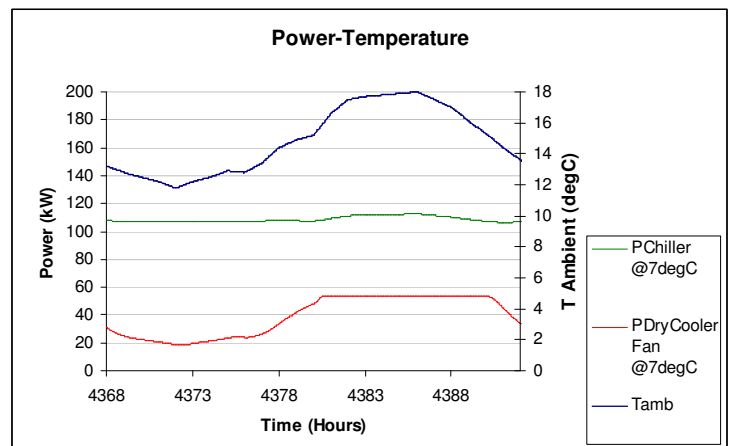
Summer

Table 9: Winter and Summer Comparison of free cooling

With the chilled water setpoint at 7°C it can be seen, from the graphs, that during the winter time the power of the dry cooler is at a maximum until the temperature rises. This can be explained by the fact that during the winter, when the temperature is below 10°C, free cooling is switched on. Therefore the dry coolers have to run at full power in order to cool the chilled water to as close to 7°C as possible. Once the temperature starts to increases the chillers come back into use, hence the rise in chiller power, and reduces the temperature mechanically back to 7°C. At this stage the dry coolers now only have to cool the water to 20°C, which is the input cooling water required for the chillers. Thus the dry coolers have to do less work and the power required is reduced. As the ambient temperature increases in the summer so too does the power of the chiller and the dry cooler. No free cooling is present in the summer as the temperatures are not low enough to enable free cooling. As with the no-free cooling scenarios, the power of the dry cooler rises as the ambient temperature rises until it reaches maximum power, in the summer day. This is because the dry cooler has to work harder to maintain the temperature entering the condenser at 20°C.

8 Case Study – Non Free Cooling and Free Cooling - Conclusion

8.1 Full System with Non Free Cooling

It was found that the COP of both the system and the chiller improves as the chilled water temperature is increased. The differences between system the terms Chiller COP and System COP are established in this part of the project. It was found that the Chiller COP is higher than the system COP because it considers only the chiller, whereas the system COP includes all the pumps and fans that make up the HVAC system.   The power breakdown and CO_2 analysis of the system showed that as the chilled water setpoint temperature increased the energy consumption of the system decreased. Very few CO_2 emissions savings were made when increasing the chilled water temperature from 7°C to 9°C (0.14%) but when the chilled water temperature was raised to 11°C, and the data centre room temperature increased to 26°C four times as much CO_2 savings were achieved when compared to having a chilled water temperature of 7°C.

A comparison of winter and summer days for chilled water setpoints of 7°C and 9°C showed that in summer the dry cooler doubled its power consumption, when compared with winter, until it reached its maximum. This is because the dry cooler has to reduce the temperature to 20°C, which uses more energy when the ambient temperature is higher in the summer than the winter.

The power of the chiller at 7°C is higher than that at 9°C because the chiller has to work more to maintain the lower chilled water temperature.

8.2 Full System with Free Cooling

The results presented in the main body of the report are for chilled water setpoints of 7°C and 9°C. The results for 11°C can be found in the appendix. A number of simulations were run to establish the optimum ambient temperature setpoint at which free cooling can be turned on at, and which chilled water setpoint temperature would give the most energy savings. Having run these simulations it can be concluded that the best ambient temperature at which to switch on free cooling, in all scenarios, is 10°C. This is because at 7°C and 9°C partial free cooling occurs which reduces the cooling effect required by the chillers, and hence energy consumption.

When implementing a chilled water setpoint of 7°C, more power was required by the system when compared to having a setpoint of 9°C. This is because in order to

achieve a chilled water setpoint of 7°C, the dry cooler fan has to use more power to reduce the temperature to 7°C than it does to reduce it to 9°C. Also, there are fewer days when the ambient temperature is low enough to enable full free cooling when the chilled water temperature is set at 7°C. If the ambient temperature setpoint at which to turn on free cooling is set too high above that of the chilled water setpoint, it was found that the dry coolers have to use more power to cool the water to the chilled water setpoint. This increased the full system energy consumption considerably, thereby negating any use of partial free cooling.

A comparison of the COP values for the full system and the chillers found that the best chiller performance, based on the COP variation over the year, is achieved when the chilled water setpoint is 9°C and the ambient temperature at which free cooling is switched on is 10°C. The full system COP variation over the year had the highest values for the chilled water temperature setpoint of 11°C. The second best COP performance was when the chilled water setpoint was set to 9°C.

In conclusion, the best ambient temperature at which free cooling was turned on was 10°C. It can be proposed that for a data centre not willing to increase the inside data centre temperatures above 24°C but still want to utilise free cooling, that a chilled water setpoint of 9°C and an ambient temperature at which free cooling is turned on at is 10°C, be implemented in order to achieve energy savings and improve system performance.

9 Improvements to the Data Centre

Having looked at different ways of increasing the energy efficiency of a data centre, such as increasing the chilled water temperature leaving the chillers to enhance performance, and demonstrating the benefits of free cooling during the winter months, other options were also found that could increase the data centre energy performance further. These were: -

- Using thermal storage to further enhance free cooling.
- Implementing Combined Cooling Heat and Power (CCHP) to meet the data centre energy needs.
- Considering bio-fuels for use with CCHP.

This chapter gives an overview of thermal energy storage using phase change materials and information about bio-diesel. A brief introduction to CCHP is given and a more in depth study of CCHP use in a data centre is carried out using TRNSYS 16 in the next chapter.

9.1 Thermal Energy Storage Using Phase Change Materials

Energy can be stored in numerous ways. In its primary form energy has been stored on Earth for millions of years. Fossil fuels such as coal, oil and gas are stored underground and solar energy is captured and stored by plants and potential energy of water through evaporation and precipitation. After extraction of this energy, such as fuel, there is even more storage in the form oil tanks, wood piles etc.   There is a growing requirement for energy storage as prices of fossil fuels rise, consumption of energy throughout the day varies and without storage waste from production of energy is produced.

There are many types of storage available for different uses. These include thermal energy storage, pumped water storage, compressed air storage and biogas storage. Although there are many types of storage available this project will concentrate on the storage of thermal energy and more specifically chilled water, also referred to as coolth. Thermal energy storage refers to a number of technologies capable of storing energy in thermal reservoirs, which may be maintained at a temperature above or below that of the ambient temperature for later reuse. Today the principle application of thermal energy storage is the production of ice, chilled water or eutectic solution at night, which is then used for cooling environments during the day. Other thermal energy storage technologies store heat, usually from solar collectors, for later use in

space heating or domestic hot water. They can also be used to balance the energy demand between day time and night time.

Phase change thermal control devices are used to either thermally control a medium or store energy by utilizing a material that undergoes a change of phase. There are a number of phase change transformation classes and they are listed below: (Humphries and Griggs 1977)

- Solid-liquid transformations (melting/ freezing). E.g. melting or freezing of ice or liquid. This is of great importance as most classes of materials undergo this type of transformation without large volume changes whilst releasing or absorbing relatively large quantities of energy.

- Liquid-gas transformations (vaporization). E.g. boiling of water. This transformation requires heavy and complicated pressure vessels or special design features such as internal bellows because of the large changes in volume that occur.

- Solid-gas transformations (sublimation). E.g. sublimation of solid carbon dioxide at atmospheric pressure, "dry-ice". Large volume changes also occur in this transformation.

- Solid-solid transformations. The energy change here can be significant and a number of materials display this phenomenon in a temperature range near their melting point.

- Liquid-liquid transformations. This process absorbs and releases very little energy and because of this it is questioned whether or not this is a true class of phase change.

Figure 26 shows a typical phase change device in its simplest form.
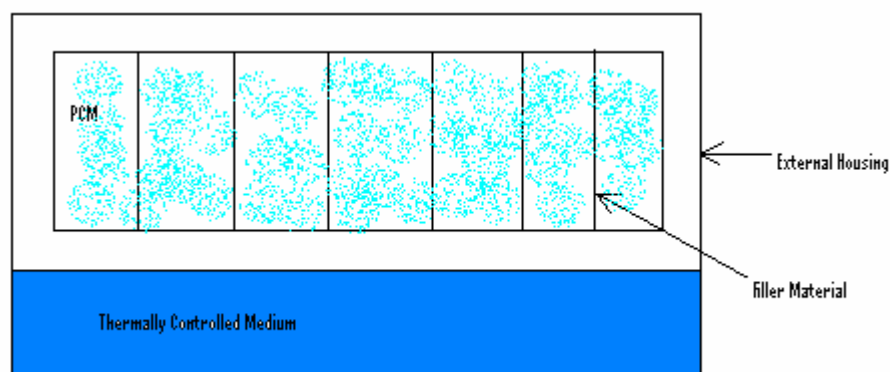
Figure 26: A Phase Change Device

The phase change device has no moving parts and consists of an external housing with the phase change material inside, which can be interspersed with or without filler material. It is a passive device that is usually in close thermal contact with the medium that it is thermally controlling. There are numerous geometrical configurations with a number of different filler configurations that this device can be applied in, such as; thermal energy storage, waste heat recovery, off peak power utilization, heat pump systems, space applications, laptop computer cooling, cooling suits, telecom shelters, cooling drinks.

Phase change materials are materials that have a high latent heat of fusion. They are capable of storing or releasing large amounts of energy by melting and solidifying at certain temperatures and have been used in thermal storage for many years because of their unique characteristics of being able to have large amounts of heat absorption/ release during the phase change process with only small changes in temperature variation.

There are two types of PCMs, liquid to gas and liquid to solid. However, only the liquid to solid PCMs are used for thermal storage. Although liquid to gas PCMs have a higher heat transformation than liquid to solid PCMs, they require large volumes or high pressures to store the materials when they are in their gas form, making them impractical for use as thermal storage. The liquid to solid PCM's initially perform like conventional storage materials with their temperature rising as they absorb heat. However, unlike conventional storage materials, when they reach the temperature at which the phase change occurs they absorb large amounts of heat without a significant rise in temperature. When the ambient temperature around the PCM falls, the PCM solidifies, releasing its stored latent heat. PCMs also fall into two categories; organic compounds such as polyethylene glycol, and salt based products such as Glauber's salt. The most commonly used compounds are salt hydrates, fatty acids, esters and paraffins such as octadecane and recently research into using ionic liquids as PCMs has been carried out. (Paré, D & S Bilodeau, 2007)

Although this technology has the potential to be used in data centres, time limits prevented it to be fully investigated. However, it can be suggested that at night, when the ambient temperature is lower than in the day, coolth can be produced and then stored for use during the day to provide free or partial free cooling to the data centre.

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9.2 CCHP

CHP has been reported over 100 years ago, although recently its use has been limited to large installations. The development of small scale, pre packaged CHP units, with high efficiencies, have seen their successful applications in smaller installations. CHP can now be seen as a major component in the strategy to reduce CO_2 emissions by the UK Government, and the Labour Party committed themselves to a target of 10 GW of installed CHP by 2010 (IMechE, 1998). Presently the majority of the UK capacity of CHP is located in medium to large scale systems serving industrial sites. Schemes of 10 MW and upwards account for nearly 80% (IMechE, 1998) of the total UK cogeneration capacity and currently generate a sixth of the total electricity used by industry (IMechE, 1998). The amount of CHP used to serve buildings and community heating is rising and in 1996, 100 MW of capacity was installed. CHP is now established as a mainstream energy technology for a range of commercial and public buildings, such as hospitals and leisure centres and hotels.

Conventional power stations generate electricity and reject heat as waste. This wastage plus transmission losses gives rise to a poor overall efficiency with high electricity costs. Therefore, an argument in favour of CHP is it is possible to produce heat and electricity more efficiently than using electricity delivered from the national grid. This CHP, produced locally minimises distribution losses and allows heat output from the plant to be used for space or process heating.

The diagram below shows the primary energy consumption of a conventional scheme compared with CHP, to satisfy the same heating and electrical demand.
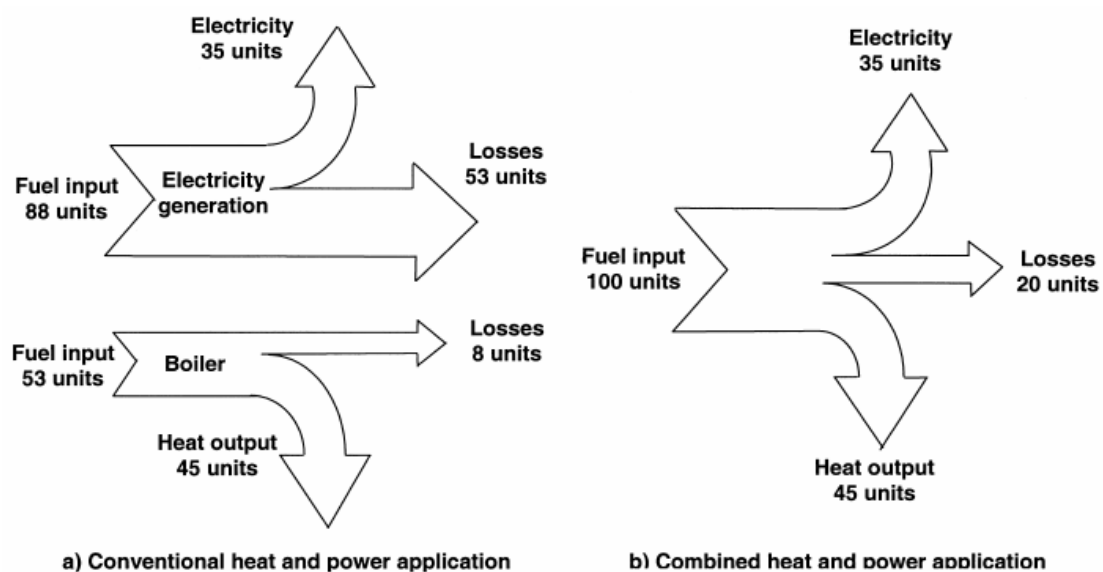
Figure 27: Sankey diagrams showing primary usage for conventional and combined heat and power applications. Source: Maidment, G.G., Prosser, B. 2000, The Use of CHP & Absorption Cooling in cold storage.

It can be seen that CHP uses less primary energy (up to 41%), and has less losses than the conventional scheme, because of this there will be cost advantages associated with CHP. There are also environmental advantages in the fact that there is less energy consumption, meaning that less non renewable energy reserves are used and pollution is reduced.

9.2.1 Basic CCHP Calculations

Preliminary calculations show that using combined cooling heat and power in a data centre has energy, cost and CO_2 saving potential when compared to conventional power and cooling methods. Below is a comparison between the two scenarios, one using solely electricity for its power and cooling needs and one using combined cooling heat and power. The calculations used to arrive at these results can be found in Appendix 5.

Annual Energy Cost	Cost per year of electrical power	Cost per year of gas	tonnes CO_2/year
Traditional	£ 710,000		4362
CCHP		£ 325,000	4026

Table 10: Preliminary Cost and CO_2 Calculations CCHP

CO_2 analysis was performed using: -

CO_2 emitted from Gas = 0.19 kg CO_2/ kWhr (Carbon Trust)

CO_2 emitted from Electricity = 0.43 kg CO_2/ kWhr (Carbon Trust). Note that the carbon emissions for electricity and gas, used throughout this thesis, are taken from the Carbon Trust. The Carbon Trust, 2007, states;

"In the case of electricity consumption it is in most cases impossible to identify with any certainty the source of generation that has been reduced or displaced. For this reason the Climate Change Levy Negotiated Agreements and the ETS are using an average carbon intensity factor for the estimation of carbon dioxide emissions from the consumption of electricity. This has been fixed at 0.43 kgCO_2/kWh."

The potential cost savings that can be made using CCHP, when compared to buying electricity from the National Grid are **£385,000** per year. According to the preliminary study there are CO_2 savings to be made of approximately **8%**. Refer to Appendix 5 for the information used for calculating these savings.

It can be seen that there are potential CO_2 savings and cost savings that can be made; therefore a study using TRNSYS 16 was undertaken, in the next chapter to further investigate the potential of combined cooling heat and power.

9.3 Biodiesel

Bio-diesel is a fuel oil derived from biological sources, mainly vegetable oils and waste cooking oils and greases. It is an alternative fuel source that can be used in place of diesel with little or no modifications necessary to engine equipment. Bio-diesel is considered to be carbon neutral because the quantity of CO_2 released during combustion is the same as that absorbed by the plant during photosynthesis. This differs from fossil fuels in that, although both originating from organic matter, the carbon in fossil fuels has been locked away for millions of years, and when released during combustion, disrupts carbon dioxide levels in the atmosphere. In reality, bio-diesel is not completely carbon neutral as there are carbon emissions associated with processing the fuel and other possible emissions of green house gases from, for example, fertiliser use. There may also be carbon costs with transportation of bio-diesel and the technologies used. Although bio-diesel also produces green house gases, such as carbon monoxide, SOx, and hydrocarbons, it has been found that these emissions are less than those produced by diesel.

Bio-diesel is made through a chemical process called transesterfication, where the vegetable and/ or animal oils, fats, and greases are combined with methanol, in the presence of sodium or potassium hydroxide, which acts as a catalyst. This process has two products, methyl esters, the chemical name for bio-diesel, and glycerine, an alcohol with many useful properties, which gives rise to many commercial applications, such as in the cosmetics industry. Bio-diesel must be produced to strict industry specifications to insure proper performance and although bio-diesel contains no petroleum it can be blended with petroleum diesel to make different levels of bio-diesel purity.

This part of the project will be investigating the amount of CO_2 produced when using different blends of bio-diesel to power a CCHP plant being used in a data centre. It will focus on the amount of bio-diesel needed to power the CCHP over the period of a year and look into potential suppliers in the United Kingdom and Europe. The calculations were carried out with the help of the Bio-diesel Simulator, created by Masters Students for their group project. (www.esru.strath.ac.uk/EandE/Web_sites/06-07/Biodiesel/simulator.htm)

9.3.1 Bio-Diesel Supply Options for the UK

The pie chart in Figure 28 shows the World production of bio-diesel.
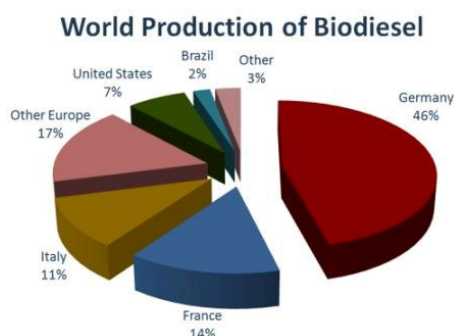
Figure 28: World Production of Biodiesel. Source: (www.esru.strath.ac.uk/EandE/Web_sites/06-07/Biodiesel/biodiesel2.htm)

It can be seen that the UK plays a minor role in the production of bio-diesel, less than 3% of the World production is produced in the UK. However, bio-diesel is beginning to be promoted within the UK. This can be seen in Scotland, where there is investment from INEOS Enterprises, almost £70 million, to build Europe's largest bio-diesel production plant at their Grangemouth site. The Scottish Executive has backed this investment with a Regional Selective Assistance of £9 million, which will help safeguard jobs. It is intended that this site will produce around a quarter of the UK's total bio-fuel needs by 2010 and provide opportunities for Scottish agriculture to provide the raw materials (Scottish Executive, News, 10/2006). Popular sources of bio-diesel include:

- Virgin oil feedstock, including rapeseed oil (Europe), soybean oil (USA), palm oil (S.E Asia), also, sunflower, canola, castor bean and many other crops.
- Waste vegetable oil from both domestic and food industrial use.
- Animal fats and fish oils.

A list of potential suppliers of bio-diesel can be found in Appendix 6.

9.3.2 Methodology to calculate the CO_2 savings possible with bio-diesel

A study looking at the amount of diesel, bio-diesel blend B2, B5, B20, B50 and B100 needed for fuelling a 1 MW CHP unit for one year, was undertaken.

With information, using a Bio-Diesel Simulator produced by students of the Energy Systems and The Environment MSc, as part of their group project (www.esru.strath.ac.uk/EandE/Web_sites/06-07/Biodiesel/biodiesel2.htm), it was possible to work out the amount of CO_2 produced and hence the savings possible when using bio-diesel.

The Gross Calorific Value of No.2 Diesel was taken as 46 MJ/Kg and the Gross Colorific Value of Bio-Diesel was taken as 35 MJ/Kg. (www.kayelaby.npl.co.uk/chemistry/3_11/3_11_4.html).

The density of No. 2 diesel used in the calculations was 850 kg/m^3 and the density of pure bio-diesel was taken as 880 kg/m^3 (www.esru.strath.ac.uk/EandE/Web_sites/06-07/Biodiesel/simulator.htm).

Presuming that bio-diesel is a blend of pure bio-diesel and diesel (in percentage of volume), it was possible to work out the calorific values of B2, B5, B20 and B50. It was then possible to work out the value of carbon emissions produced by the diesel used in the bio-diesel blends. Refer to Appendix 8 for the Excel spreadsheet for calculations.

<u>Calculation Method</u>

Firstly the Energy in Joules per year was calculated (E).

The mass of bio diesel required was calculated using:

$$Mass = \frac{E}{CalorificValueofFuel}$$

Equation 13

From this it was possible to calculate the CO_2 emissions using:

1kg of Diesel = 3.1kg of CO_2

1kg of pure bio-diesel = 2.8 kg of CO_2. This figure is the amount of CO_2 released by the bio-diesel during combustion, however, it is presumed that this is carbon neutral as the CO_2 produced is reabsorbed by the growing the crops for bio-diesel.

The above figures were taken from the Bio-diesel Simulator, created by Masters Students for their group project. (www.esru.strath.ac.uk/EandE/Web_sites/06-07/Biodiesel/simulator.htm).

9.3.3 Results and Analysis

Figure 29 shows the amount of CO_2 produced by diesel when using different blends of bio diesel, at different power outputs required of the CHP engine.
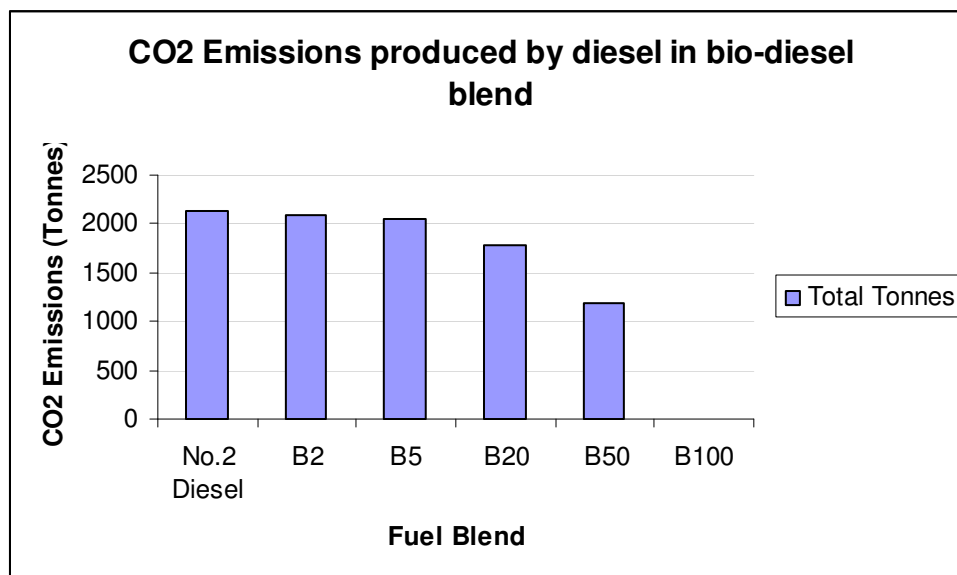
Figure 29: CO_2 production of diesel when using blends of biodiesel

It can be seen from Figure 29 that there are CO_2 emissions produced by the diesel when using blends of bio-diesel, and the reductions in the carbon emissions were small when using B2 and B5 blends of bio-diesel, when compared with No.2 Diesel. This is because bio-diesel B2 and B5 are a blend of 2% bio-diesel and 98% diesel, and 5% bio-diesel and 95% diesel respectively, which means that the diesel proportion is quite large.

There were only zero CO_2 emissions when pure bio-diesel was used, as the carbon produced by combusting pure bio-diesel can be considered as carbon neutral, and there is no diesel in a B100 (pure bio-diesel) mixture. However, to make a completely fair comparison between using diesel and bio diesel, a study comparing the CO_2 emissions during the production stage of the two fuels should be undertaken, which is out with the scope of this study.

Therefore, it would be advisable to either use blends of B20, B50 or B100 bio-diesel to see a significant amount of CO_2 emissions savings. However, it would not be advisable to commit to using bio diesel unless a secure supply of high blends of bio diesel or B100 are available.

10 Combined Cooling Heat and Power (CCHP)

10.1 Background CCHP

In recent years it has become standard practice to consider CHP in the early design stage of a commercial building and it is now becoming considered for wider applications, particularly within industrial building design. New initiatives from the UK Government, encouraging reduced energy use, have encouraged this further.

Trigeneration also referred to as combined cooling, heating and power (CCHP), can be considered as the generation of mechanical power (usually converted to electricity), heat and cooling from a single heat source. Trigeneration works using the same principles as cogeneration (combined heat and power, CHP), with added absorption chillers.   These absorption chillers use the waste heat from the cogeneration plant and convert it to useful energy. Not only can absorption chillers be integrated with cogeneration systems but they can also operate with industrial waste heat streams.

10.2 CHP Package Components

A CHP package generally consists of:
- Prime mover and its fuel supply system – converts fuel or thermal energy to shaft energy. These conversion devices are usually reciprocating internal combustion engines, combustion turbines, and steam boiler-turbine combinations (cogen systems and engine and turbine drives).
- Generator.
- Waste heat recovery system.
- Electrical and thermal transmission and distribution systems.
- Connections to the building mechanical and electrical services.

Generator

The most common prime movers in the smaller cogeneration plants (under 15 MW) (ASHRAE, 2000) are reciprocating engines, otherwise known as internal combustion engines. These are either Spark ignition Otto Cycle engines or compression ignition cycle engines.

The Spark ignition Otto Cycle engines use natural gas, liquefied petroleum gas (LPG) and other gaseous fuels and are usually produced in sizes up to 13 MW. These

engines usually operate in the range of 360 rpm to 1200 rpm, with some up to 1800 rpm, and the specific operating speed selected depends on the size, brand, generator, and the desired length of time between complete engine overhauls (ASHRAE, 2000). They are usually selected to operate for a minimum of 15,000 to 30,000 hours between minor overhauls and up to 50,000 hours between major overhauls. The larger, lower speed engines correspond with higher operating hours.

The generator is normally a synchronous unit in a small scale CHP unit, which allows the generating plant to operate in isolation from the grid and other generating plant. As it is synchronous it can also be operated in parallel with the grid so that the general load can be supplemented by the grid if demand is greater than the system output. Conversely, if the load is less than the output, power can be exported to the grid, or the output modulated. However, the sale price for exported electricity is low and normally just covers the basic production cost.

Waste Heat Recovery System
Heat produced by the generator can either be rejected directly to the atmosphere, if this occurs and no heat is used for other applications the unit cannot be considered combined heat and power, or can flow through a heat exchanger.
Heat exchangers are used to recover the heat rejected from the engine jacket and exhaust gas, during generation, and provide water temperatures of generally between 75ºC and 90ºC for heating. Some systems may be configured to produce medium temperature hot water (MTHW) at temperatures of up to 130ºC (Maidment, G.G. & Prosser, B., 2000).

The electrical and thermal transmission and distribution systems and the connections to the building mechanical and electrical services are important factors to consider when implementing a CCHP package, but are not detailed in this report.

10.3 Absorption Chillers

The basis of absorption design is the affinity of some vapours, such as ammonia (NH_3) for water, or water for Lithium Bromide (LiBr). After Faraday had demonstrated the absorption principle of refrigeration in 1824, using a bent glass tube charged with ammonia (NH_3) and silver chloride, the first absorption design was applied in industry by Nicolle, who patented his ice making machine in 1861 and subsequently went on to design refrigerating machinery for ships. Nicolle's refrigeration units operated

using NH_3 and were first used for freezing meat to be shipped from Australia to Britain. The COP of these units was very low (less than 0.2) and it wasn't until the 1960's that LiBr/water chillers driven by hot water or steam were marketed seriously. This technology was then ruled out for most of the world, Japan being the exception, in the 1970's because of energy efficiency issues. Japan at this time had invested in a gas expansion policy so bought absorption technology, and developed two stage direct fired units. They are now the world leaders in absorption technology, with 6000 plus absorption chillers (IMechE, 1998) being produced annually. After Japan, China, Korea and the USA follow the market. In Japan the chillers tend to be direct fired, whereas in the USA the chillers tend to be used in conjunction with CHP. The market is much lower in the UK, which is reported to have 200 LiBr absorption chillers (IMechE, 1998).

Absorption chillers offer advantages over vapour compression cycle chillers because the cooling effect is driven by heat energy instead of mechanical energy. Also, they do not use atmosphere harming halogenated refrigerants or require lubricants that are known to degrade heat and mass transfer. The various equipment can be either direct-fired, by combustion of fuel, or can be directly heated by various waste fluids, or can be heated by steam or hot water (from either direct combustion or from hot waste fluids) (ASHRAE Handbook, 2002, Refrigeration).

The disadvantages of absorption chillers are their low COP, usually less than 2 for High Temperature Hot Water (HTHW) chillers, and less than 1 for Medium or Low Temperature Hot Water (M/ LTHW) chillers, which results in much heat needed to drive the absorption process. This heat has to then be rejected into the atmosphere, which requires a large amount of cooling power, i.e., dry coolers. This is illustrated in simple terms in Figure 30 and Figure 31 comparing electric chiller heat rejection and absorption chiller heat rejection.

Total heat rejected = 1.25 MW

Electric Chiller
COP = 4

Electrical input = 0.25 MW

Heat rejected to chillers
from data centre = 1 MW
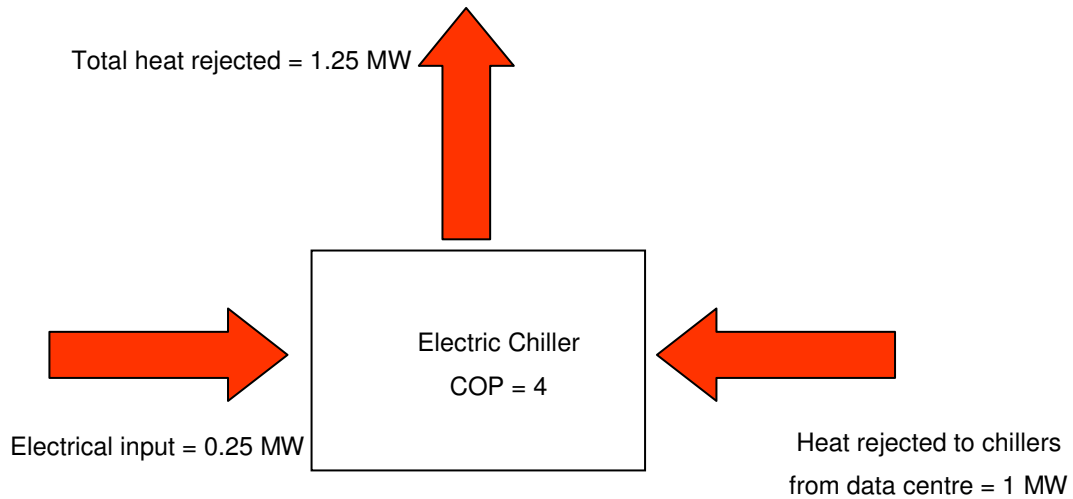
Figure 30: Electric Chiller Heat Rejection

It can be seen that the electricity used to drive the electric chiller is ultimately rejected into the atmosphere as heat along with the heat rejected from the data centre.

Total heat rejected = 2 MW

Absorption
Chillers
COP = 1

Heat Input 1 MW

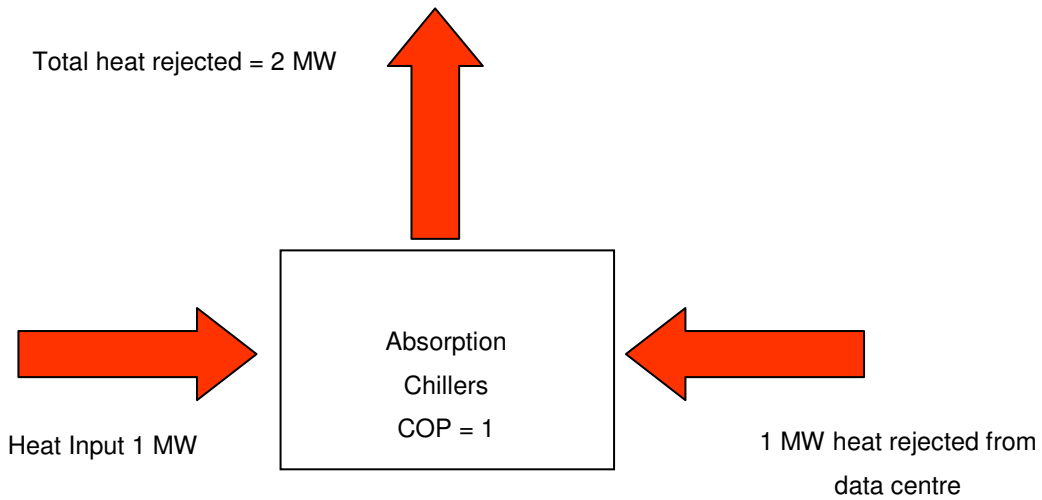1 MW heat rejected from
data centre

Figure 31: Absorption Chiller Heat Rejection

The diagram above shows that the heat input used to drive the absorption chillers is rejected along with the heat rejected from the data centre. It can be seen that when using electric chillers, with a COP of 4, the total heat rejected is less than the total heat rejected when using absorption chillers with a COP of 1.

10.3.1 Vapour Absorption Cycle

The vapour absorption system is similar to a vapour compression system, with a condenser, an evaporator and a throttle valve, the difference being the compressor is replaced by an absorber, a generator and a liquid pump as shown in Figure 32.
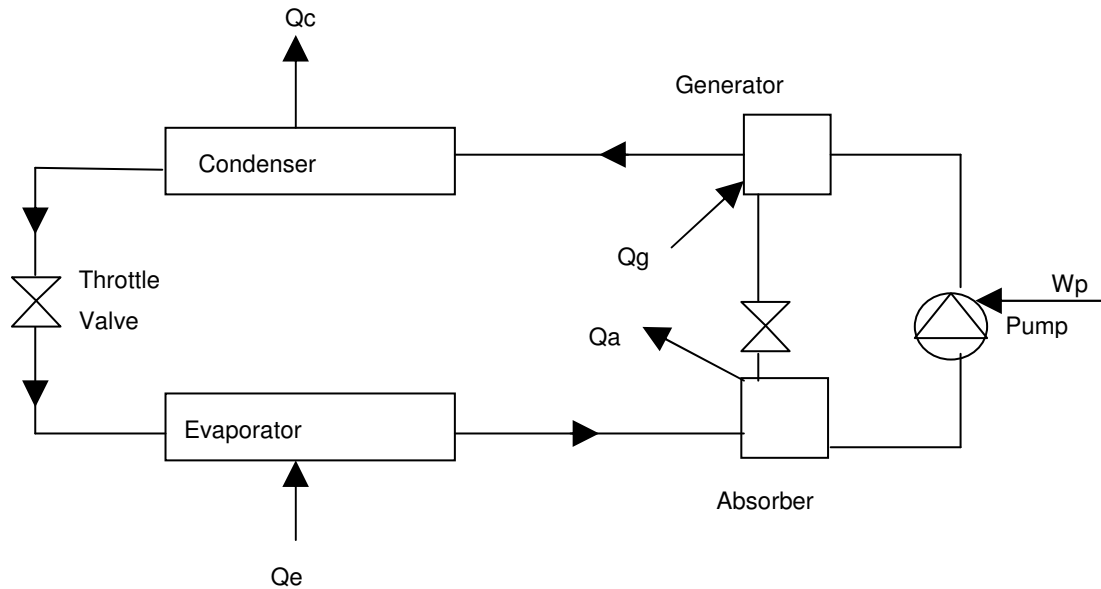
Figure 32: Vapour Absorption Cycle. Source: McLean R.C., 2006

The work done by pumping the liquid solution is small compared to the work used by the compressor in the vapour compression system. The absorption process works using both the Carnot and Reverse Carnot cycle and is exothermic, requiring heat to be continuously removed from the absorber.

Absorption cooling works because of the strong affinity of some chemicals to dissolve in one another. For example, a strong solution of lithium bromide in water will draw water vapour from its surroundings to dilute the solution. This affinity is used in absorption cooling to draw the refrigerant, in this case water, from a conventional evaporator into an absorber. The weakened solution is then pumped to a higher pressure, to the generator. Heat is applied and the water is driven off to a conventional condenser. The re-strengthened solution is now recycled to the absorber. Heat is rejected from both the absorber, as it gets hot as it absorbs the refrigerant, and from the normal condenser (ETSU, 2001).

10.3.2   Combining CHP with Absorption Chillers

In order to have combined cooling heat and power, a CHP engine must work in conjunction with an absorption chiller. The heat from the CHP engine drives the absorption chillers. The next section describes the combining of the Carnot Cycle and Rankine Cycle, explaining in more detail the how the two work together.

10.3.2.1   Combined Carnot Cycle and Reverse Carnot Cycle

The Carnot Cycle and Reverse Carnot Cycle can be combined into one device, Figure 33.
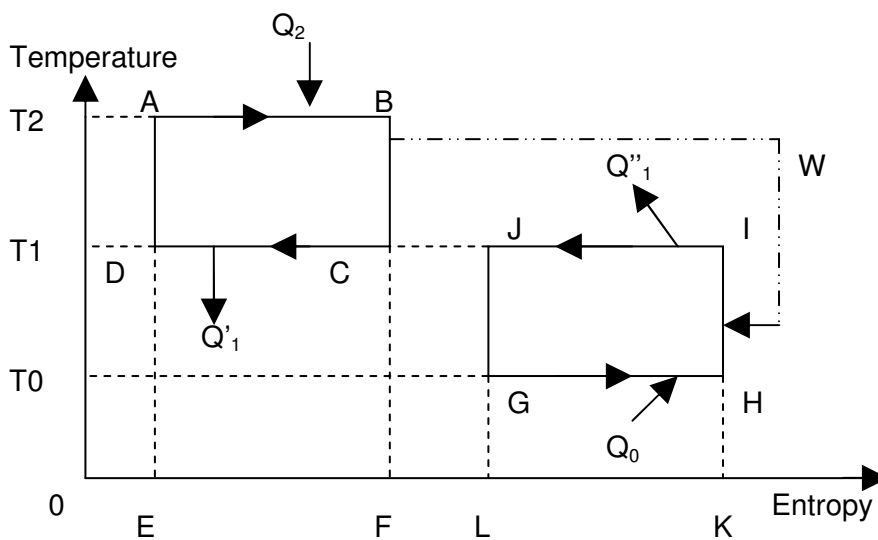
Figure 33: Carnot Cycles for a combined power- generation/heat-pumping facility such as an absorption heat pump. Source: Herold et al, 1996. Absorption Chillers and Heat Pumps.

This device represents a heat pump that is driven by the input of heat only and assumes that the work produced by the Carnot cycle for power generation (Figure 8, in Chapter 3) is equivalent to the amount of work required by the Carnot cycle for heat pumping (Figure 9, in Chapter 3). The temperature level of heat supplied at $T0$ is raised to $T1$ by using the thermodynamic availability of the high temperature energy supplied at $T2$. Waste heat produced by the power generation portion of the cycle is rejected at $T1$ along with the heat produced by the heat pump cycle.

11 Case Study - CCHP

The case study data centre used for CCHP is the same as that used previously. It is situated in Wales and the information was kindly provided by WSP.

As with the first simulations the dimensions and power consumption of the data centre will be used as inputs for the simulations in TRNSYS 16.

The assumptions made about the Data Centre are:

25 m length x 11 m width x 6 m height

Volume = 1650 m^3

Internal heat gains, excluding RACU fan power = All IT + lighting + miscellaneous = 740 kW.

11.1 Description of the Case Study - CCHP model

Inside the Data Centre the electrical loads taken into account are made up of: 700 kW of IT, UPS losses of around 4% of the IT load (28 kW), and lighting and miscellaneous accounts for around 2% of the IT load (12 kW).

All of these electrical loads are converted to heat within the data centre therefore the whole data centre produces 740 kW of heat, which is cooled by RACUs.

All parameters and systems used are the same as before, with the only difference being the replacement of the water cooled chillers with two absorption chillers connected to a CHP unit. Figure 34 illustrates the theory behind CCHP.
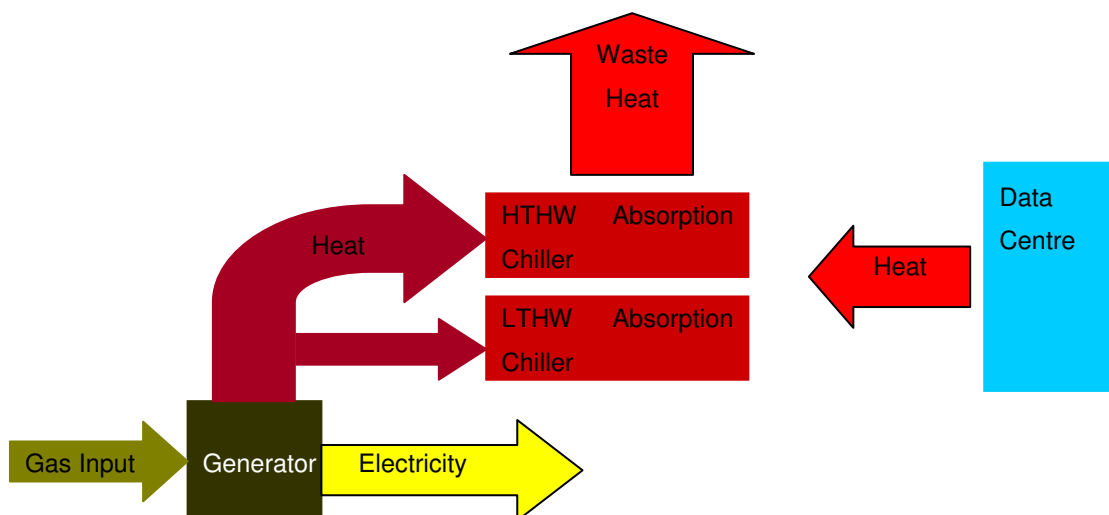
Figure 34: CCHP model

This diagram shows that a generator, fuelled by gas, produces heat and electricity. The heat is used to drive the absorption chillers and the electricity is used in the data centre. The electrical input to the data centre is converted into heat that must be removed. The absorption chillers produce chilled water for the RACUs, which remove the heat from the data centre, to the absorption chillers, and provide cooling for the data centre. As can be seen, heat is then rejected from the absorption chillers to the atmosphere.

11.2 The CHP unit and Absorption Chillers used in the Case Study - CCHP

Two absorption chillers are required in the case study, a High Temperature Hot Water (HTHW) Absorption Chiller and a Low Temperature Absorption (LTHW) Chiller. The HTHW absorption chiller uses the high temperature from the exhaust and the LTHW absorption chiller uses the low temperature from the jacket cooling.

11.3 The TRNSYS 16 Model

The parameters and inputs for the CCHP simulations are the same as previously, with the difference being that two absorption chillers replace the water cooled chiller and a CHP unit is connected to both the absorption chillers as illustrated in Figure 35 and Figure 36.
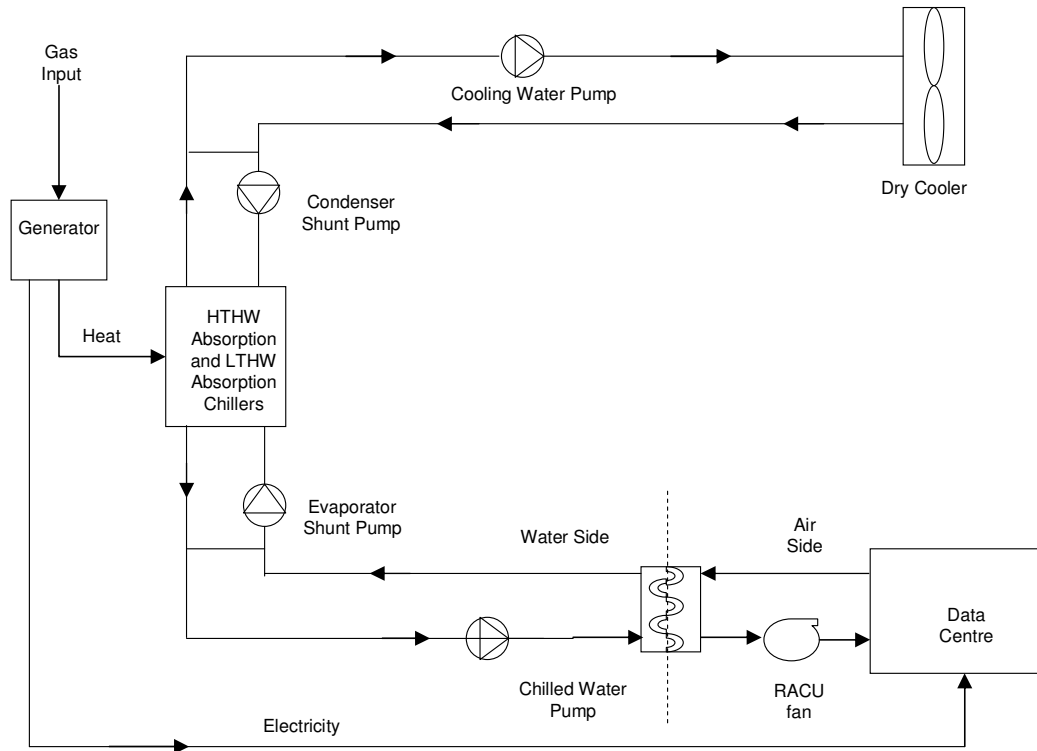
Figure 35: CCHP diagram of TRNSYS 16 Model with no free cooling
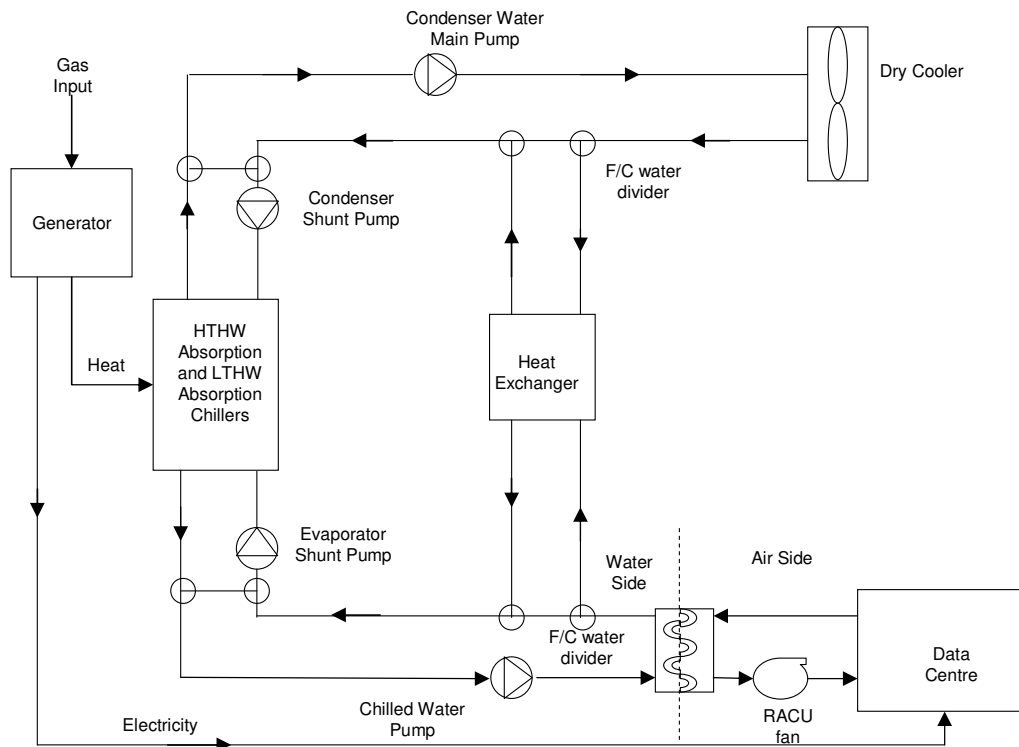
Figure 36: CCHP diagram of TRNSYS 16 Model with free cooling

11.3.1   The Absorption Chillers in TRNSYS 16

The absorption chillers (Type 107), in TRNSYS 16, use a normalized catalogue data lookup approach to model a single-effect hot-water fired absorption chiller. "Hot Water-Fired" indicates that the energy supplied to the machine's generator comes from a hot water stream. Because the data files are normalized, the user may model any size chiller using a given set of data files (TRNSYS 16 manual). The absorption chillers output is reliant on the amount of heat that is provided to them by the generator, therefore, if less heat is provided to drive the cooling effect, less cooling will be produced. Thus the generator is used in TRNSYS 16 to control the output of the absorption chillers, which is particularly important when utilising free cooling.

11.3.2   The Generator in TRNSYS 16

The generator used in this project was the TRNSYS 16 internal combustion engine (Type 907). The model relies on an external data file which contains efficiency, air flow rate (fraction of rated flow rate) and heat transfer data (fraction of total energy output) as a function of the intake temperature and the part load ratio (power over rated power). The information used to produce these data files was kindly provided

by WSP and can be found in Appendix 7. Figure 37 is a diagram showing how the generator interacts with the absorption chillers in TRNSYS 16.
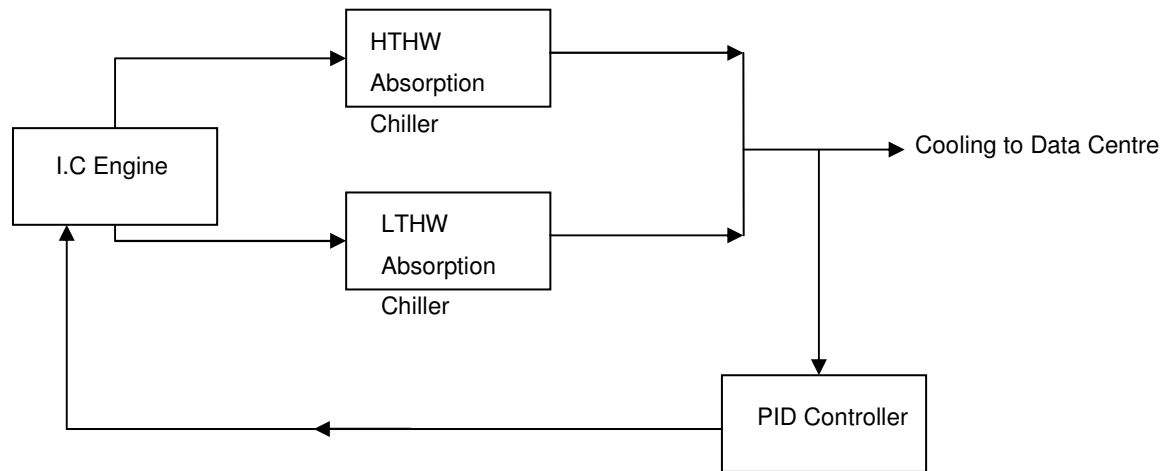
Figure 37: Generator & Absorption Chiller Interaction

It was decided at this stage to control the heat output of the generator to meet the cooling load of the absorption chillers. This was decided as it was undesirable, at this stage, to have excess heat produced by the generator, which would require a means of cooling before being rejected to the atmosphere. Figure 37 shows how a proportional, integral and derivative (PID) controller is employed, in the TRNSYS 16 program, to control the heat output of the generator. The PID controller (Type 23) calculates the control signal required to maintain the controlled variable at the setpoint (TRNSYS 16 manual). In this case, the PID controller is used to calculate the cooling produced by the absorption chillers in order to control the output of the generator.

12 Methodology for Investigating CCHP in Data Centres

To investigate the benefits of implementing CCHP in data centres two simulations were run. The first simulation run was of the data centre with free cooling and the second simulation was run without free cooling. It was decided that currently used chilled water temperatures of 7°C would be used in order to maintain the data centre room temperature at 24°C. The ambient temperature at which free cooling was turned on was set at 10°C, as this was found to be the optimum temperature, from the simulations run previously.

13 Results Analysis for CCHP

13.1 Results with no free cooling

The graph below shows the part load ratio (PLR) of the CHP unit. The PLR is the ratio of output from a piece of equipment, in this case the CHP unit, to the equipments rated output. Thus it is desirable to have a high PLR, as this means that the equipment is working as close to or at its rated output.

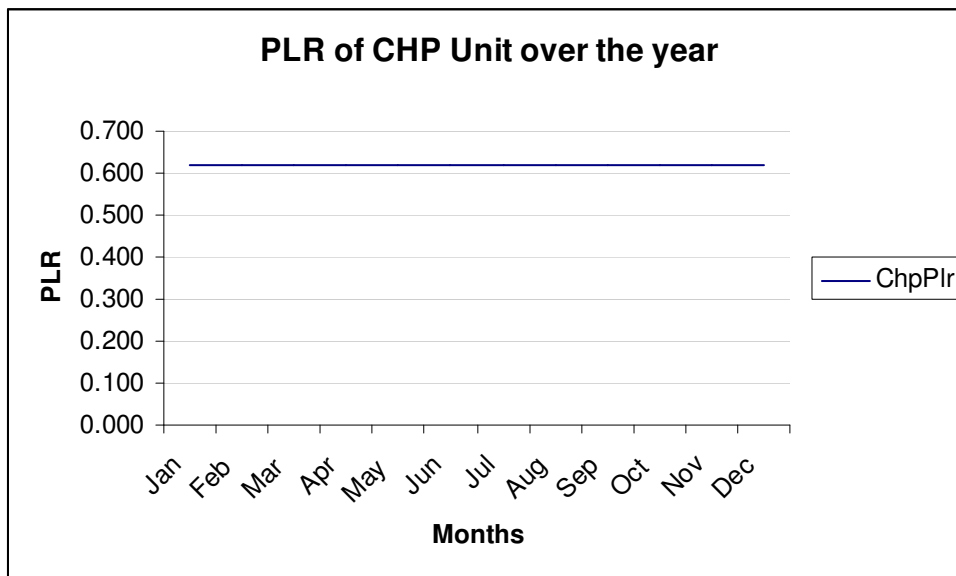Figure 38: PLR variation over the year of CHP unit - no free cooling

It can be seen that the PLR of the CHP unit is the same each month at a little over 0.6. This means that the CHP unit is working at around 60% of its capacity at all times.

Figure 39 shows the electrical energy produced by the generator and the electrical energy required by the data centre for each month over the period of a year.

Electrical Energy Produced- Electrical Energy Required

(bar chart showing Electrical Energy (MWh) on the y-axis from 0 to 800 and Month on the x-axis from Jan to Nov, comparing "Electricity required" and "Electricity produced")
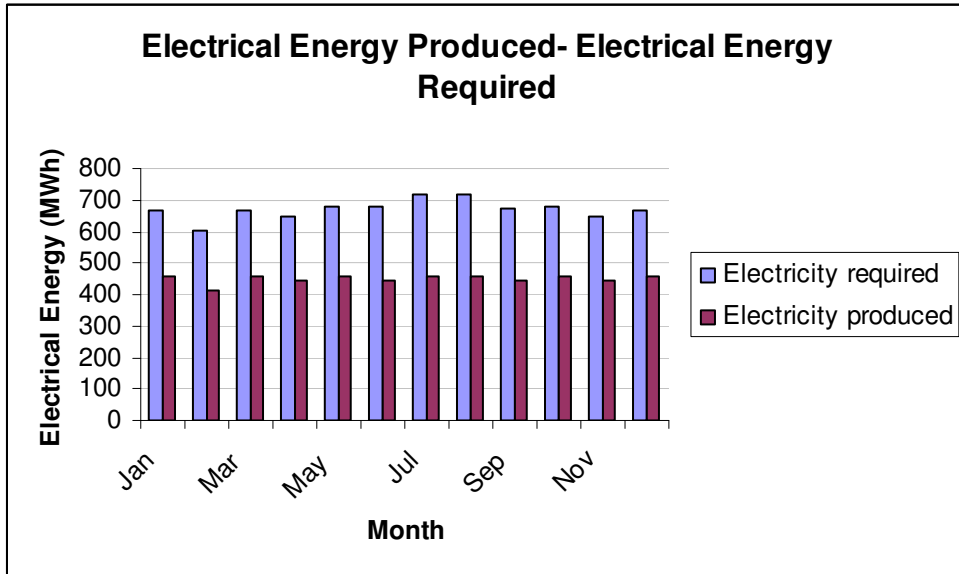
Figure 39: CCHP no free cooling - Energy Produced - Energy Required

It can be seen that the electrical energy produced by the generator does not meet the electrical energy requirements of the data centre, which means that power will have to be purchased from the National Grid. It can also be seen that the most electricity is generated in the summer months of July and August, when the ambient temperatures are high. This is because the cooling water returning to the condenser from the dry cooler is at higher temperatures than in the winter, so the heat transfer required between the condenser and the evaporator inside the absorption chillers is higher in the summer, thus the absorption chillers need a higher input heat, from the CHP unit, to drive the absorption process that can deal with this increase in heat transfer. This higher supply heat to the absorption chillers comes from making the generator work at a higher heat output and hence electrical output. There is quite a noticeable dip in the energy produced and the energy required during the month of February because there are only 28 days in this month.

13.2 Results with free cooling

Figure 40 shows the PLR of the CHP unit when the system is utilising free cooling. The chilled water temperature setpoint = 7°C and the ambient temperature at which free cooling is turned on is 10°C.
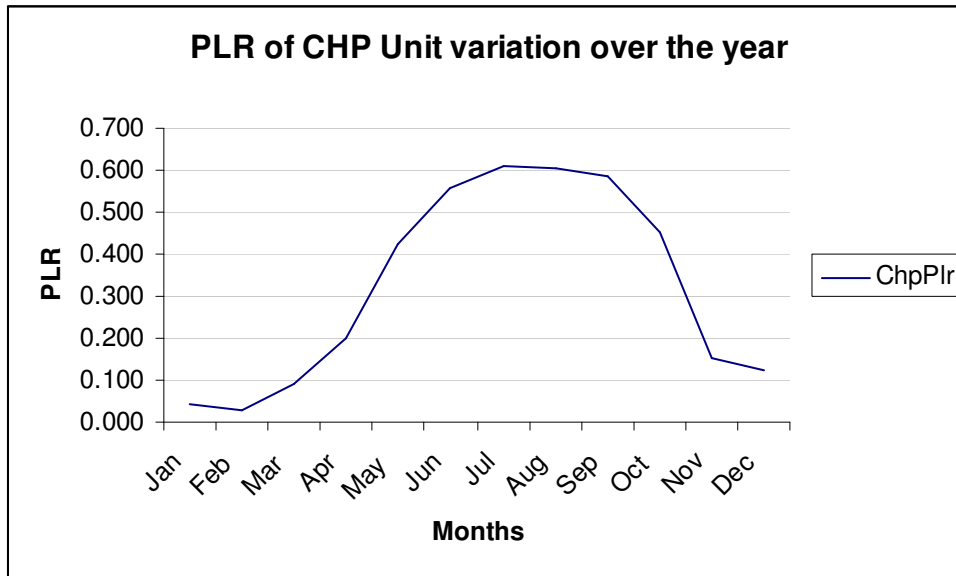
Figure 40: PLR variation over the year of CHP unit with free cooling

It can be seen that the PLR of the CHP unit is very low during the winter months, increasing as the ambient temperature increases into the summer months where it reaches a maximum of around 0.6. As explained previously, it is desirable to have a high PLR and during the months of January to March the CHP unit is working at less than 10% of its full capacity. The reason that this is occurring is because of the choice of control system used, which is to control the heat output of the generator to meet the cooling load of the absorption chillers. It is deduced, from analysing the PLR results, that free cooling is occurring during the winter months because the CHP unit is having to produce a very little heat to drive the absorption chillers, and as it is the heat requirement of the absorption chillers which control the CHP unit it can be said that the chillers have a low heat transfer rate, owing to the free cooling.

Electrical Energy Produced- Electrical Energy Required
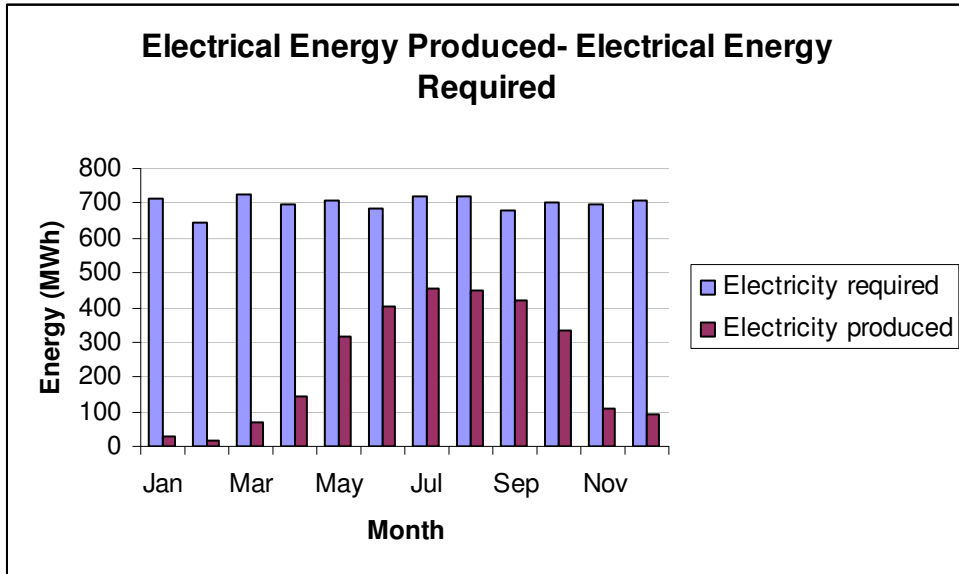
Figure 41: CCHP with free cooling – Energy Produced - Energy Required

The graph above shows the monthly electrical energy produced by the generator and the electrical energy required by the data centre. The energy produced by the generator does not meet the electricity energy requirements of the data centre, especially during the winter months which means that power will have to be purchased from the National Grid. This is because the free cooling is active during the winter months, as the ambient temperature is cold enough to enable free cooling. This means that the absorption chillers have less cooling to provide and hence the generator output is reduced because it is the heat output of the generator that controls the absorption chillers cooling capacity. It can also be seen that the highest electricity amount is generated in the summer when the generator is working at full output.

13.3 Comparison of the CCHP without free cooling and with free cooling

13.3.1 Typical winter and summer day – No free cooling

The graphs in Table 11 show the power usage and free cooling controls over the period of a typical winter day and a typical summer day of CCHP without free cooling.
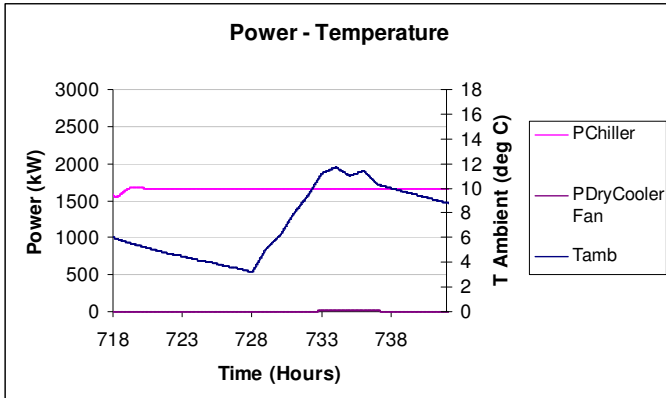
Typical Winter Day
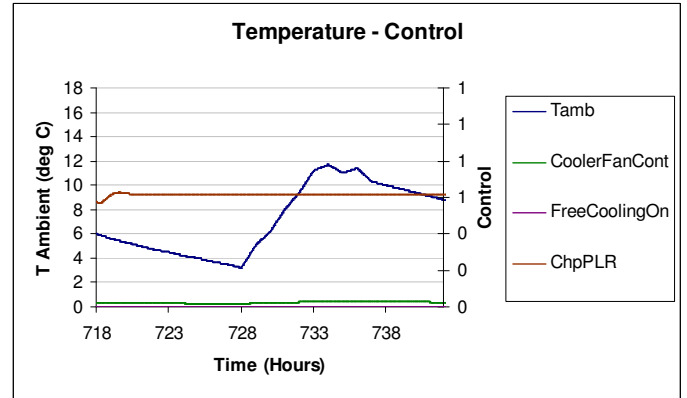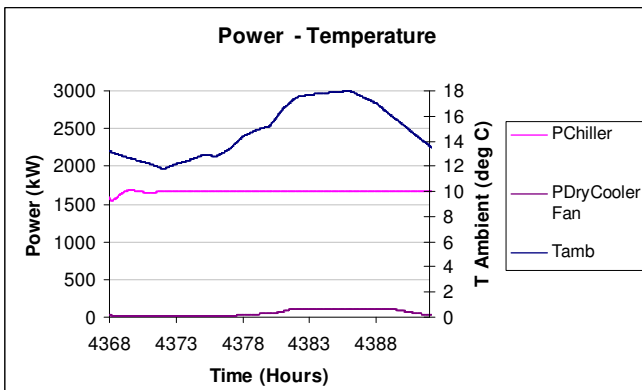
Typical Winter Day
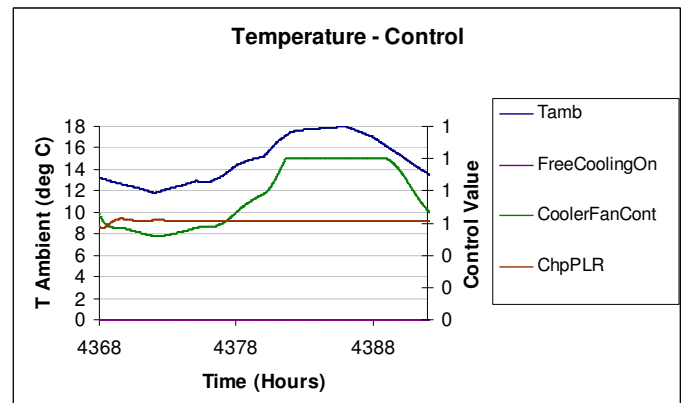
Typical Summer Day

Typical Summer Day

Table 11: Typical Summer & Winter Day - CCHP no free cooling

Where:-

PChiller = Heat input to the chiller

PDryCoolerFan = Power consumed by the dry cooler fan

Because there was no free cooling in this simulation it can be seen that the control system for the free cooling and the dry cooler fan remains at zero, i.e., turned off. The chiller PLR in both the summer and the winter day has a value of 1. The power of the dry coolers changes significantly between summer and winter and the power of the chillers remains constant at just above 1500 kW.

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13.3.2   Typical winter and summer day – Free cooling

The graphs in Table 12 show the power usage and free cooling controls over the period of a typical winter day and a typical summer day of CCHP with free cooling.
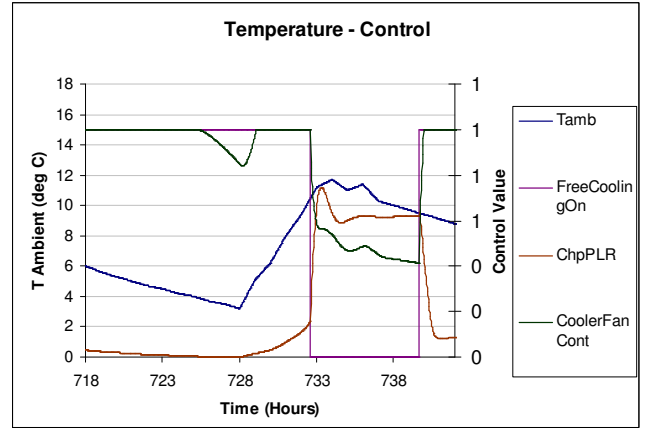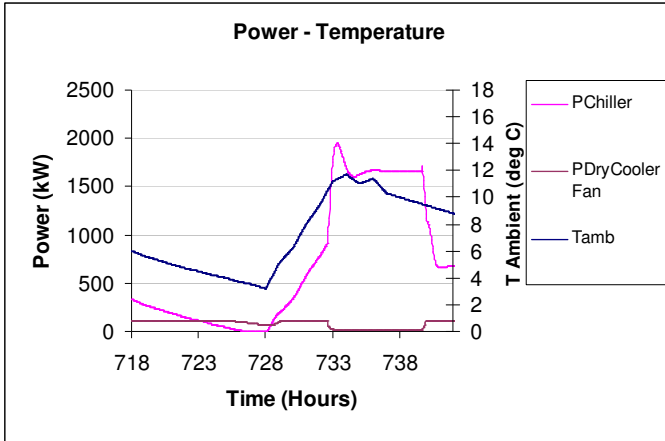
Typical Winter Day
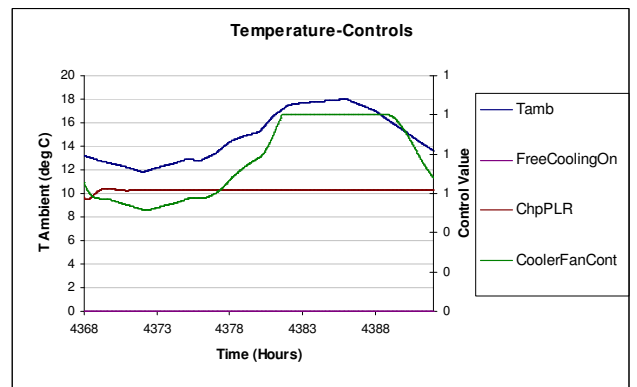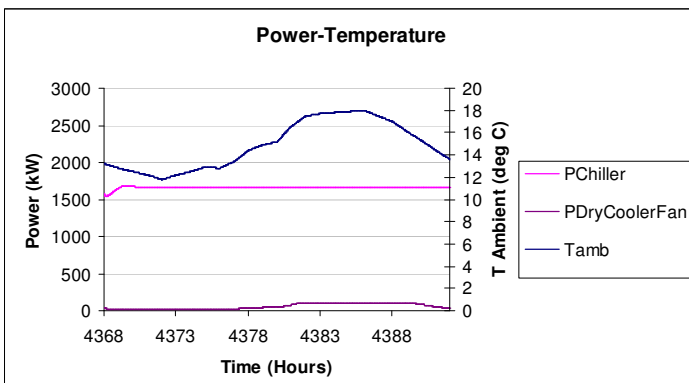
Typical Winter Day

Typical Summer Day

Typical Summer Day

Table 12: Typical Winter & Summer Day- CCHP with free cooling

Where:-

PChiller = Heat input to the chiller

PDryCoolerFan = Power consumed by the dry cooler fan

It can be seen from the graphs above that in the winter, there is full free cooling when the temperature is below 7°C. As the temperature rises above 7°C partial free cooling occurs and the chiller power starts to increase as it is required to do work to provide chilled water. It can also be seen that the dry cooler power is at maximum during the periods of free cooling and reduces in power when the chiller comes back on. The temperature-control graph shows that the control system chosen for turning on free cooling is working, as when the ambient temperature reaches 10°C the controls go to zero, turning off free cooling. The chiller PLR is zero during the

periods of free cooling, and increases to around 1 when free cooling is turned off. During the summer days there is no opportunity for free cooling and the chiller performs like it would if it had no free cooling capability.

13.4 Energy Consumption and CO_2 Emissions Comparison

The results in this section show the energy consumption and the CO_2 emissions for CCHP with and without free cooling.

Annual Gas Consumption
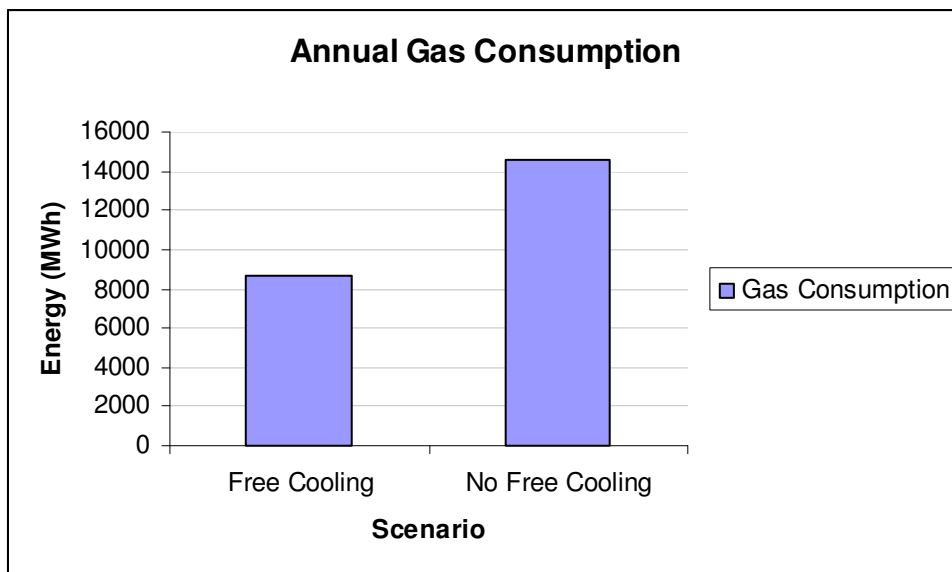
Figure 42: Annual Gas Consumption

Figure 42 shows the annual gas consumption of the system with and without free cooling. It can be seen that the system with no free cooling uses almost double the amount of gas as the system with free cooling.
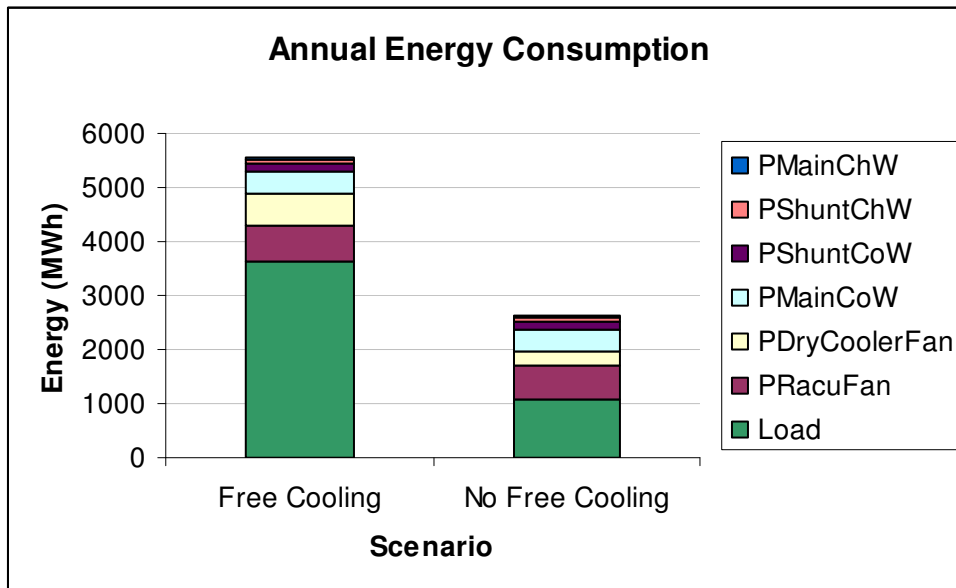
Figure 43: Annual Energy Consumption

Where:-

Load = Net Load, which is the electricity used by the data centre equipment (IT plus Miscellaneous) minus the electricity produced by the CHP unit.

PDryCoolerFan = Power consumed by the dry cooler fan

PMainChW = Power consumed by the main chilled water pump

PMainCoW = Power consumed by the main cooling water pump

PShuntCoW = Power consumed by the chilled water shunt pump

PRacuFan = Power consumed by the RACU fan

PShuntChW = Power consumed by the chilled water shunt pump

Figure 43 shows that when using free cooling the power consumed is almost double that than if free cooling were not utilised. There is a large difference in the power consumed by the load because it was decided to offset some of the load consumed with the electricity produced by the generator. It can be seen that the load in the system with no free cooling is a third of that of the load in the system with free cooling. This again owes to the choice of controlling the output of the generator by the cooling capacity of the absorption chillers. With free cooling the output of the generator is reduced in line with the cooling capacity of the absorption chillers, hence the electrical output of the chillers is also reduced. Hence more power input is required for the load in the system with free cooling. It can also be seen that the power consumed by the dry cooler is larger when free cooling is enabled and is another large contributing factor for the increase in power consumption of the whole system with free cooling.

Putting Figure 42 and Figure 43 together gives rise to Figure 44 which better illustrates the total energy consumption of both systems, where it can be seen that the system with free cooling uses less energy in total, than the system with no free cooling.
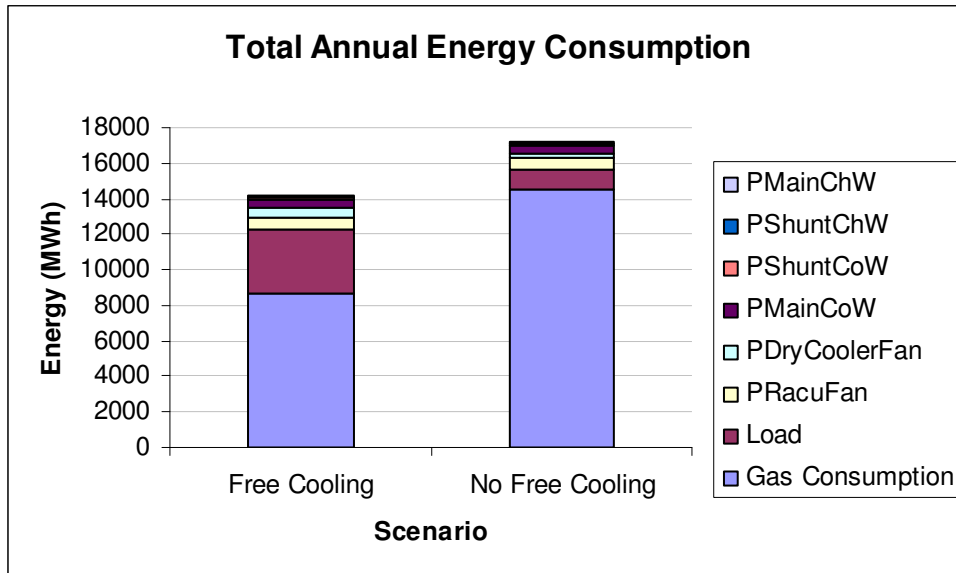
Figure 44: Total Annual Energy Consumption

Where:-

Load = Net Load, which is the electricity used by the data centre equipment (IT plus Miscellaneous) minus the electricity produced by the CHP unit.

PDryCoolerFan = Power consumed by the dry cooler fan

PMainChW = Power consumed by the main chilled water pump

PMainCoW = Power consumed by the main cooling water pump

PShuntCoW = Power consumed by the chilled water shunt pump

PRacuFan = Power consumed by the RACU fan

PShuntChW = Power consumed by the chilled water shunt pump

Figure 45 shows the annual carbon emissions produced by system with and without free cooling.

Annual Carbon Emissions
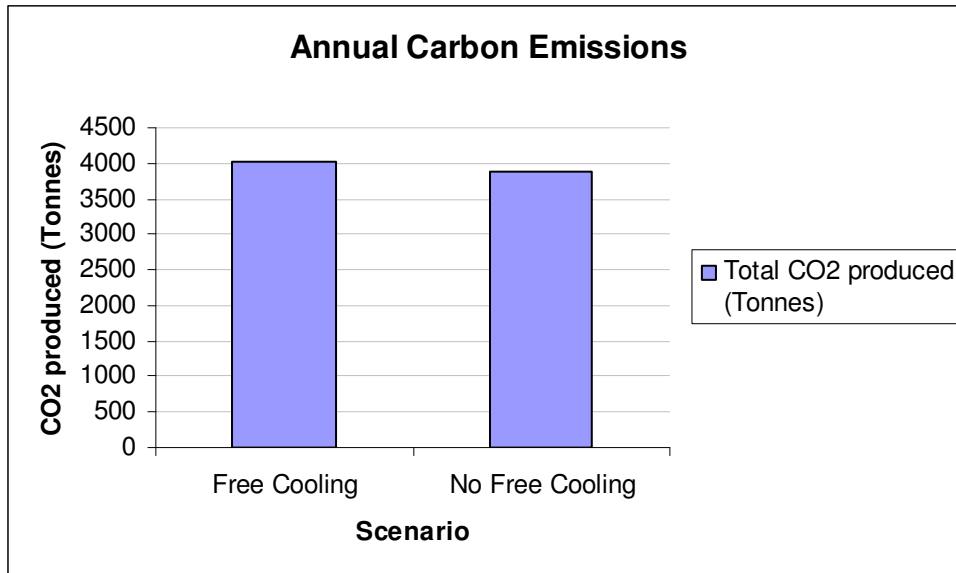
Figure 45: CCHP Annual Carbon Emissions

CO_2 analysis was performed using: -

CO_2 emitted from Gas = 0.19 kg CO_2/ kWhr (Carbon Trust)

CO_2 emitted from Electricity = 0.43 kg CO_2/ kWhr (Carbon Trust). Note that the carbon emissions for electricity and gas, used throughout this thesis, are taken from the Carbon Trust. The Carbon Trust, 2007, states;

"In the case of electricity consumption it is in most cases impossible to identify with any certainty the source of generation that has been reduced or displaced. For this reason the Climate Change Levy Negotiated Agreements and the ETS are using an average carbon intensity factor for the estimation of carbon dioxide emissions from the consumption of electricity. This has been fixed at 0.43 kgCO_2/kWh."

It can be seen that more CO_2 is produced by the system that makes use of free cooling.

The table below shows the figures that produced the graph above.

The gas input to the CHP unit is the total gas required for that year to power the generator. The electricity deficit to be met by the grid was calculated by taking away the total electricity produced by the generator in the year, from the total electricity requirement of the system for the year. Because gas and electricity have different CO_2 emissions when used the CO_2 emissions of each was calculated and then added together to give the total carbon emissions that would be produced by the system.

Annual Carbon Emissions						
	Gas Input to CHP Unit (MWh)	Electricity Deficit to be met by the national Grid (MWh)	CO_2 produced from the Gas (Tonnes)	CO_2 produced from Electricity (Tonnes)	Total CO_2 produced (Tonnes)	CO_2 Saving between the two systems (Tonnes)
CCHP without Free Cooling	14550	2620	2764	1130	3894	140
CCHP with Free Cooling	8650	5550	1644	2390	4034	

Table 13: Annual Carbon Emissions

CO_2 analysis was performed using: -

CO_2 emitted from Gas = 0.19 kg CO_2/ kWhr

CO_2 emitted from Electricity = 0.422 kg CO_2/ kWhr

The above analysis shows that more electricity is met by CCHP when free cooling is disabled. Although more gas is required to power the CCHP unit without free cooling, less CO_2, around 116 tonnes less, is produced overall as gas produces less CO_2 per kWhr than electricity. The CO_2 analysis is specific to the project, and therefore the fact that the generator is being controlled by the heat required to drive the absorption chillers has a large impact on the results. Other control systems chosen, such as running the CHP unit at full load and finding alternative ways to use the waste heat, whilst utilising the full electrical output, could result in very different CO_2 results.

13.5 Electrical Power Produced by CCHP

Table 14 shows the electrical power produced by CCHP with free cooling enabled and disabled.
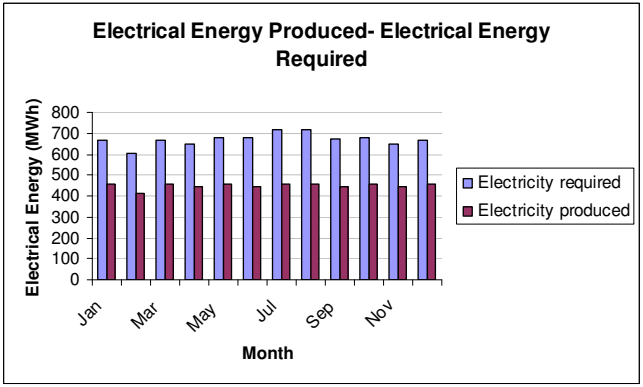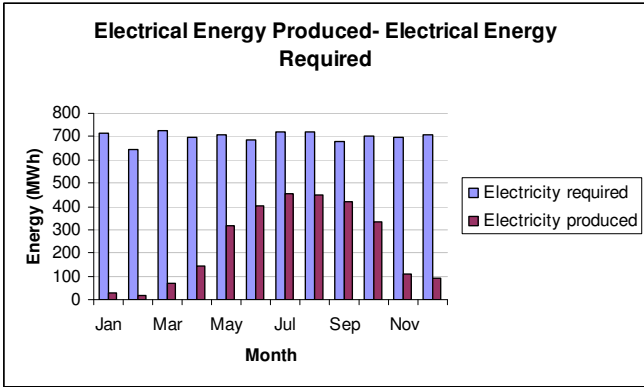
CCHP- Free Cooling Enabled

CCHP- No Free Cooling

Table 14: Power Produced-Power Required

It can be seen that the CCHP unit does not produce enough electricity to meet the data centre requirements. The CCHP unit without free cooling produces a fairly steady amount of electricity over the year. The CCHP unit with free cooling on the other hand produces more electricity over the summer months. This is because the generation unit is used to control the chillers, by increasing or decreasing the heat energy supplied to the absorption chillers. In the winter time, when free cooling is enabled, the absorption chillers are not required constantly. Therefore the heat energy supplied to them reduces, and when the generator produces less heat it also produces less electricity. In the summer, it is necessary for the absorption chillers to work at full output, so they have a high heat requirement to drive them and the generators produce more heat to meet this demand, and hence more electricity.

The analysis of CCHP with and without free cooling has shown that it would not be conducive for businesses, or environmentally, to use CCHP in conjunction with free cooling, with the control system chosen in this project. In these simulations the generator was controlled to meet the heat requirement of the absorption chillers. Therefore if the heat requirement reduced, so too did the heat and electrical output of the generator, which meant that very little of the electricity demand of the system was met by the generator. One alternative that could be looked at that may increase the electricity produced by the generator and further decrease CO_2 emissions, would be to run the generator at full capacity. However, one question that this poses is how should excess heat produced be used in an energy efficient way. One suggestion is to use it for space heating and cooling (using absorption chillers) if offices are in close vicinity to the data centre.

14 Case Study- CCHP- Conclusion

Having run two simulations of the data centre utilising CCHP with and without free cooling it can be concluded that free cooling is detrimental for energy savings when used with CCHP, with the heat output of the generator controlling the absorption chillers. This is due to the fact that during the winter months the ambient temperature is cold enough to enable free cooling, which means that the absorption chillers have less cooling to provide and the generator output is reduced to follow the heat requirements of the absorption chillers.   As the generator does less work it produces less heat and power, so it is necessary for the data centre to purchase electricity from the National Grid so as to meet the electrical demand. It can also be seen that the highest electricity amount is generated in the summer when the generator is working at full output.

When using the control strategy of running the generators to meet the absorption chillers heat demand the gas consumption, for the system with free cooling, is half of that without the free cooling and the electricity production is also half. There is a considerable deficit of energy produced by the generator compared to energy required by the data centre. When the amount of electricity that has to be acquired from the national grid is combined with the amount of gas used by the generator, the results show that more energy is consumed and more CO_2 is produced in comparison to the system using no free cooling.

It has been shown that by meeting the heating load for the Absorption chillers with free cooling does not provide any benefits to the system performance. Benefits to the overall system are only made when CCHP is utilised without free cooling. However, it could be suggested that the CHP unit is run to produce its full output, which will provide more electricity to meet the data centre load, and the excess heat can be used for heating other buildings, if the data centre is located in close proximity to these other buildings, or dumped into the atmosphere if necessary, although this is not ideal as extra cooling equipment would be required to accommodate this.

15 Results Comparison – An Overview

A comparison of three scenarios taken from each section is presented here. The three scenarios are: -

- Standard Practice: System where the chilled water setpoint temperature = 7°C and there is no free cooling.
- Free Cooling: System with the chilled water setpoint temperature = 7°C, incorporating free cooling which is programmed to turn on at an ambient temperature = 10°C.
- CCHP: System using CCHP, with no free cooling allowance and a chilled water setpoint temperature = 7°C.

The temperature inside of the data centre for the above three cases is maintained at 24°C. These scenarios were chosen because current practices in data centres use these temperatures.

The graph below shows the annual energy consumption for the three different scenarios outlined above, it should be noted that for CCHP system with no free cooling, the gas consumption has been included in the annual energy consumption.
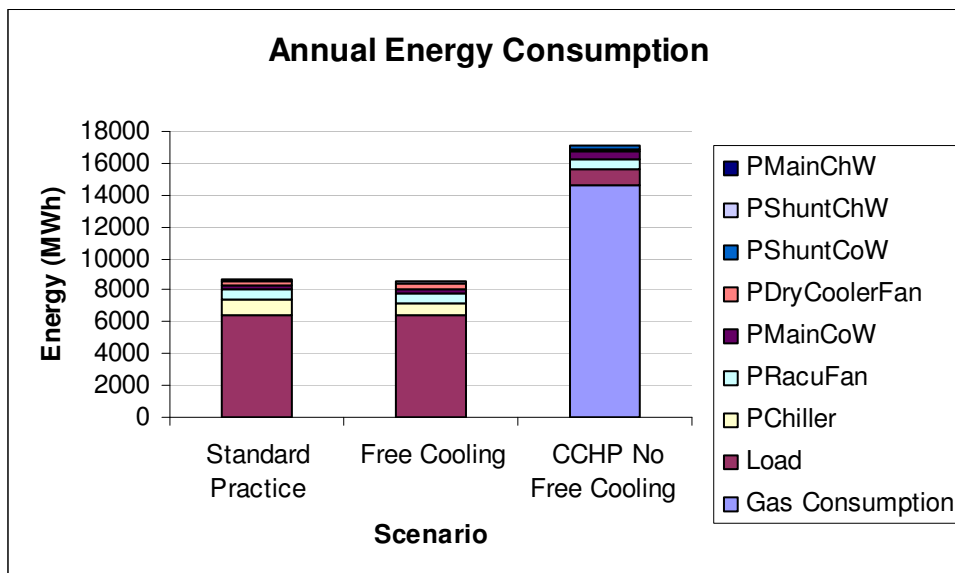
Figure 46: Annual Energy Consumption – Overview

Where:-

Load = Net Load, which is the electricity used by the data centre equipment (IT plus Miscellaneous) minus the electricity produced by the CHP unit.

Gas Consumption = Gas consumed by CCHP

PChiller = Power consumed by the chiller

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PDryCoolerFan = Power consumed by the dry cooler fan

PMainChW = Power consumed by the main chilled water pump

PMainCoW = Power consumed by the main cooling water pump

PShuntCoW = Power consumed by the chilled water shunt pump

PRacuFan = Power consumed by the RACU fan

PShuntChW = Power consumed by the chilled water shunt pump

The scenario with CCHP has a significantly lower load than the other two scenarios because it was decided to offset some of the load consumed, with the electricity produced by the generator. The largest differences between the standard scenario and the free cooling scenario are the chiller power and the dry cooler power. It was found that, with free cooling available the power consumption of the chiller is reduced but the power consumption of the dry cooler is increased, when compared with the scenario with no free cooling.

An overview of the annual carbon emissions is supplied below.
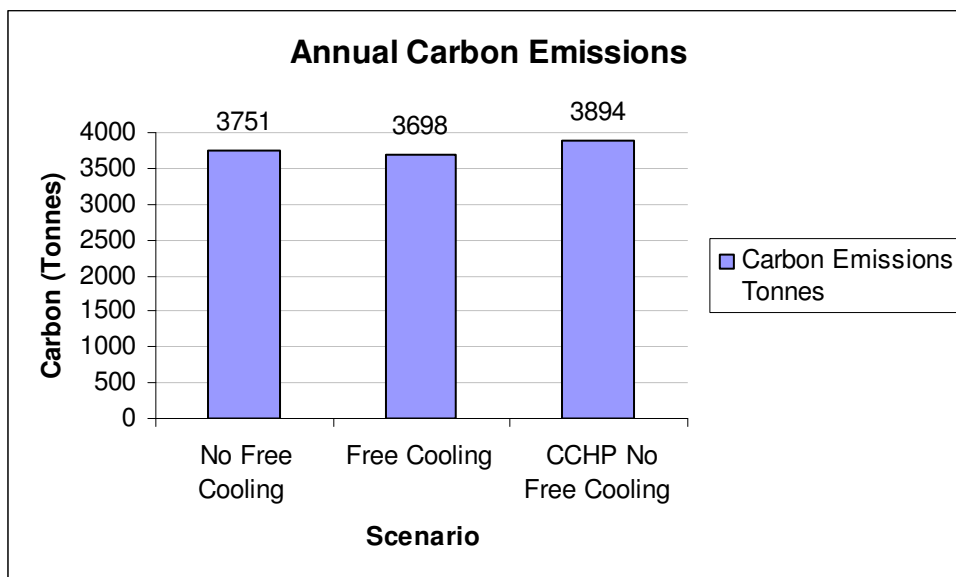
Figure 47: Annual Carbon Emissions – Overview

It can be seen that using electric chillers with free cooling produce the least carbon emissions. However, it should be noted that the control system used for the CCHP has an effect on the output of the system, and using a different control system could result in reduced CO_2 emissions.

A cost analysis was undertaken to determine the cost benefits that could be provided by implementing energy efficient systems.

Cost Analysis					
	Gas Consumed (MWh)	Cost of gas (£)	Electricity Consumed (MWh)	Cost of Electricity (£)	Total Cost of Energy (£)
Standard Practice	0	0	8723	611,000	611,000
Free Cooling	0	0	8600	602,000	602,000
CCHP No Free Cooling	14550	220,000	2620	184,000	404,000

Table 15: Cost Analysis – Overview

This cost analysis was carried out using: -

Cost of Electricity = £0.07 per kWh

Cost of Gas = £0.01536 per kWh.

It can be seen that the cost of running the CCHP system is significantly smaller than running electric chillers. The CCHP system can save up to £200,000 a year. Although the energy consumption and CO_2 emissions are higher for the CCHP system, the cost of gas is seven times less than that of electricity, hence the cost savings shown. It can also be seen that using free cooling in conjunction with electric chillers also provides cost savings.

16 __Conclusion__

The aim of this study was to investigate the options available to data centres for making them more energy efficient. In their current period of growth, data centres are going to have to carefully consider their energy consumption so as to be in line with current legislation. Having carried out an extensive literature review it was found that one way of reducing energy consumption is to make changes to the cooling plant. This thesis concentrated on the chilled water loop used for cooling the data centre. Firstly vapour compression chillers were analysed with differing chilled water setpoints, then free cooling was incorporated, and a study of increasing the chilled water setpoints was undertaken. Secondly, a study of the system using absorption chillers and CHP was undertaken, with and without free cooling. Lastly a comparison of the energy performance for these differing parameters is given.

The literature review performed ascertains what a data centre is and the types of IT and operating systems that are used within them. It also provides an understanding of the current UK legislation brought into practise to tackle climate change. It was found that data centres should adhere to Part L of the Building Regulations for England Wales, especially if they are incorporated into office buildings. However, data centres are unique in that they are built for a singular specific purpose, so the building regulations need to be carefully consulted before embarking on building a data centre.

The literature review helped to focus the study towards chillers after having established that the power consumed by the cooling equipment makes up a sizeable chunk of energy consumption by a data centre. This section acknowledges the fact that if the power consumed by the IT equipment increases, so too does the power consumed by the infrastructure supporting it and when designing a data centre the electrical power consumption comes secondary to ensuring reliability and satisfying performance requirements.

An introduction to the HVAC systems was given, illustrating the technologies used in "typical" data centres. With the aid of the Carnot and Reverse Carnot Cycle, the basic principles of chillers are explained and the Coefficient of Performance was derived.

The second section detailed the case study and introduced the program TRNSYS 16, which was used to investigate how to make a data centre more energy efficient using current technology practices within data centres.

A description of the case study model was provided and the systems used within the TRNSYS 16 model were explained in detail. The methodology used for investigating the impacts on energy efficiency for differing scenarios of increasing the data centre chilled water temperature leaving the chillers and of incorporating free cooling was implemented in this section.

The results analysis for the system without free cooling showed that the COP of the chiller and the full system was highest when the chilled water setpoint was 11°C, inferring that this is when the system performed at its best.

A comparison of the energy consumption of the different scenarios for the system without free cooling was given along with a CO_2 analysis. These results concluded that as the chilled water setpoint increased so too did the energy savings and CO_2 reductions.

The results of the system with free cooling found that there was an optimum ambient temperature setpoint at which to switch on free cooling, which is 10°C. This was the case for all chilled water setpoints. This is because partial free cooling occurred when the temperature at which to switch on free cooling was 10°C and the chilled water setpoints were set at 7°C and 9°C. This partial free cooling reduced the cooling effect required from the chillers, and hence energy consumption of the chillers and consequently the whole system. When the chilled water temperature was set at 11°C, setting the ambient temperature setpoint to 10°C yielded the least energy consumption in the simulations.

A comparison of the best case scenario and base case scenario performance of the system during a winter and summer day was given, demonstrating why increasing the chilled water setpoint leaving the chillers from 7°C to 9°C gave savings on power consumption and hence CO_2 emissions.

The third section investigated improvements that could be made to the data centre to achiever further energy and CO_2 reductions. Suggestions were using phase change materials for thermal storage, utilising CCHP and then fuelling the CCHP with bio diesel. A description of phase change materials and coolth storage is provided and it was suggested that at night, when the ambient temperature is lower than in the day, coolth could be produced and then stored for use during the day to provide free or partial free cooling to the data centre. CCHP was then briefly described as it was covered in further detail later on in the project. Rough calculations performed demonstrated that savings in CO_2 emissions and cost benefits could be made if CCHP were to be utilised instead of electric chillers. The section regarding bio diesel investigated the possibility achieving CO_2 savings if bio diesel were to be used to fuel

CCHP. The methodology of how the CO_2 was calculated for bio diesel was given and the results presented showed that using B20, B50 or B100 bio diesel for the case study would achieve significant CO_2 savings. A brief look at the supply of bio diesel found that only 3% of the world's production is in the UK and it would be advisable to use bio diesel only if there is a secure supply available.

The last section of this thesis investigated the effects of CCHP on the systems of the data centre and gave a technical overview of combined heat and power and absorption chillers. TRNSYS 16 was utilised to simulate the CCHP. The model used a PID controller to control the heat output of the generator to meet the cooling load of the absorption chillers. This control system was chosen because it was considered undesirable to have excess heat produced by the generator. It must be noted that the choice of control system affects the performance of the CCHP and other control systems which could be used will give different results. The part load ratio was used to compare the CCHP with and without free cooling and it was found that the system with the highest PLR throughout the year was the system with no free cooling. A comparison of the power produced and the power required was given showing again that the system with no free cooling performs better than the system with free cooling. Also presented in the results are a CO_2 analysis and a comparison of energy consumption between the two systems.

The final section gave an overview of results for non free cooling, free cooling and CCHP without free cooling, for a chilled water setpoint of 7°C and indoor data centre temperature of 24°C. These scenarios were chosen because current practices in data centres use these temperatures.

It can therefore be concluded that there are a variety of options available to data centres for reducing energy consumption. The first half of this project showed that increasing the data centre chilled water setpoint from 7°C by 2°C increased the performance of the chillers, resulting in energy savings and carbon emission savings. Further increasing the chilled water setpoint by another 2°C resulted in further savings, but it was necessary to increase the temperature inside the data centre by 2°C. Incorporating free cooling further enhanced the performance of the chillers, at all chilled water setpoint temperatures and there was found to be an optimum temperature at which to turn on the free cooling, this was 10°C. This is because the power consumption of the dry coolers increases during free cooling, and above 10°C

the power consumption of the dry coolers exceeded that of what the chillers would have used without free cooling.

The second half of this project showed that there was no carbon emissions savings made when using CCHP, compared to using electric chillers. However, the annual running cost was potentially £200,000 less than that when using electric chillers. The results of the CCHP unit is subject to the control strategy chosen and it can be concluded that meeting the cooling load of the absorption chillers can be improved upon.

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www.kayelaby.npl.co.uk/chemistry/3_11/3_11_4.html
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Appendix 1- Chiller Performance

The chiller selected for use in TRNSYS 16 was a 'Water Cooled Chiller'. This cools a fluid stream on the evaporator side and rejects heat to a fluid on the condenser side as shown in Figure 48. In reality more than one chiller is used, to ensure cooling is still available should one break.
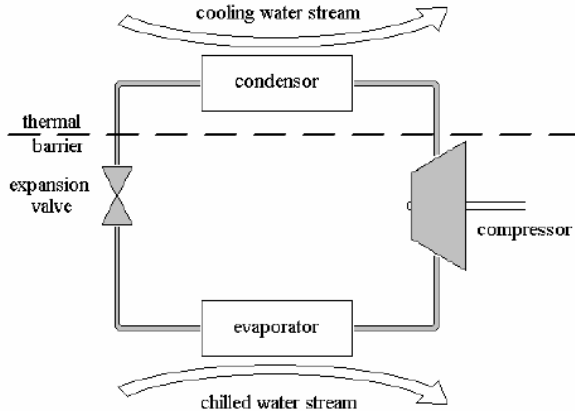

Figure 48: Schematic Diagram of a Single Stage Water Cooled Chiller. Source: TRNSYS 16 Manual

TRNSYS 16 performs its calculations by looking up data files, known as TRNSYS DynamicData. This data is contained in two files; one containing information on the chillers capacity ratio and COP ratio, for varying values of chilled water set point temperature (in °C), and the other containing information about the values of the chillers fraction of full load power for varying values of part load ratio. The data files were constructed from the performance data for the Carrier 19XR 1407kW (400 ton) Centrifugal VSD R-134a.

An energy balance on the system was performed in order to find out the mass flow rates of the working fluids, which was then entered into the TRNSYS 16 model.

The chilled water flow rate and cooling water flow rates for the chiller were calculated by performing an energy balance on the system:

$$Q = \dot{m} * Cp * \Delta T \hspace{4cm} \text{Equation 14}$$

Firstly, in TRNSYS 16, the chillers nominal COP and Capacity ratio is calculated by performing a call to the DynamicData (first file) with the current cooling water (sink) temperature and the chilled water setpoint temperature:

$$COP_{nom} = COP_{rated} * COP_{ratio} \hspace{3cm} \text{Equation 15}$$

$$Capacity = Capacity_{rated} * Capacity_{ratio} \hspace{2cm} \text{Equation 16}$$

TRYNSYS 16 then calculates the chillers load using Equation 17:

$$Q_{load} = \dot{m}_{chw} * Cp_{chw} * (T_{chw,in} - T_{chw,setpoint})$$

Equation 17

The Part Load Ratio is calculated by:

$$PLR = \frac{\dot{Q}_{load}}{Capacity}$$

Equation 18

The load met by the chiller is limited to the capacity of the machine if the calculated PLR is greater than unity. Once a valid PLR has been calculated (between 0-1), the DynamicData is called upon again, this time to specify the second data file, and the chillers power draw is given by:

$$P = \frac{Capacity}{COP_{nom}} FFLP$$

Equation 19

A corrected COP is then calculated as:

$$COP = \frac{\dot{Q}_{met}}{P}$$

Equation 20

The energy rejected to the cooling fluid stream, by the chillers, is:

$$\dot{Q}_{rejected} = \dot{Q}_{met} + P$$

Equation 21

The outlet temperature of the chilled fluid stream is:

$$T_{chw,out} = T_{chw,in} - \frac{\dot{Q}_{met}}{\dot{m}_{chw} Cp_{chw}}$$

Equation 22

The outlet temperature of the cooling fluid stream is:

$$T_{cw,out} = T_{cw,in} - \frac{\dot{Q}_{rejected}}{\dot{m}_{cw} Cp_{cw}}$$

Equation 23

The chillers capacity was originally set as 1500kW and the rated COP was set at 6.04. One simulation was run in order to get information about the mass flow rate, power and COP of the Chiller when it is only working with the RACUs.

Appendix 2- Dry Cooler

In TRNSYS 16 the fluid flow rate (water) was calculated using:

$$\dot{m}_{CoWmainpump} = \frac{\left(\dfrac{Load_{building}*1.2}{4.19}\right)}{6} = \frac{kg}{hour}$$

Equation 24

Where

$\dot{m}_{CoWmainpump}$ = mass flow rate of the cooling water through the cooling water main pump (kg/hour)

$Load_{building}$ = building load (kJ/hour)

This means that the dry cooler will be able to meet an extra 20% of the building load, included in this 20% is the power of the pumps.

The air flow rate through the dry cooler is calculated by:

$$\dot{m}_{fandrycooler} = (n*drycooler)*38.9*1.19*3600 = \frac{m^3}{s}$$

Equation 25

The power of the dry cooler is calculated by:

$$P_{drycooler} = (n*drycooler)*16.1*3600 = kW$$

Equation 26

Where

$\dot{m}_{fandrycooler}$ = mass flow rate of cooling water through the dry cooler (kg/hour)

$P_{drycooler}$ = power of the dry cooler (kW)

n = number of dry coolers

$drycooler$ = power of the dry cooler (kJ/hour which is converted into KW in the equations)

These equations were used for sizing the dry coolers and defining their rated characteristics.

Appendix 3- Design Parameters used in TRNSYS 16

The design parameters for the simulations are: -

Data Centre

25 m length x 11 m width x 6 m height

Volume = 1650 m^3

Building surface area = 982 m^2

Building capacitance = 50000 kJ/K

Internal heat gains, excluding RACU fan power = All IT lighting + lighting + miscellaneous = 740 kW (2520000 kJ/hr).

Building loss coefficient is negligible, i.e., loads such, as solar gains, and losses, through fabric etc are taken as negligible in TRNSYS program.

Room Air Conditioning Units (RACUs)

Rated fan power = 315000 kJ/hr = 87.5 kW

Rated flow rate = 307617 kg/hr = 85.5 kg/s

Design temperatures are: 24ºC returning from the data centre, 15 ºC after the cooling coil in the RACU, 16 ºC after the RACU fan.

Total air flow rate is 362424 kg/hr (= 90.7 kg/s = 76.2 m^3/s).

TRNSYS 16 models the RACU as one but in reality 6 to 10 actual RACUs would be used.

For the water side, the equivalent UA of the RACU is calculated using a regression on the water flowrate:

UA [kW/K] = -12.444438 + 34.3267977*x - 15.3908699*x**2 + 3.27705672*x**3 - 0.26623961*x**4

Where x is the chilled water flowrate in kg/s. The equation above is for one RACU with a design load of 76.5 kW so it is multiplied by (740/76.5) to model a series of parallel RACUs matching the total load of 740 kW.

The water flowrate is set by a PID controller that keeps the return air (or the room temperature in the model) constant at 24°C.

Appendix 4- Energy Consumption and CO$_2$ Emissions Comparison for Chilled Water Setpoint = 11°C

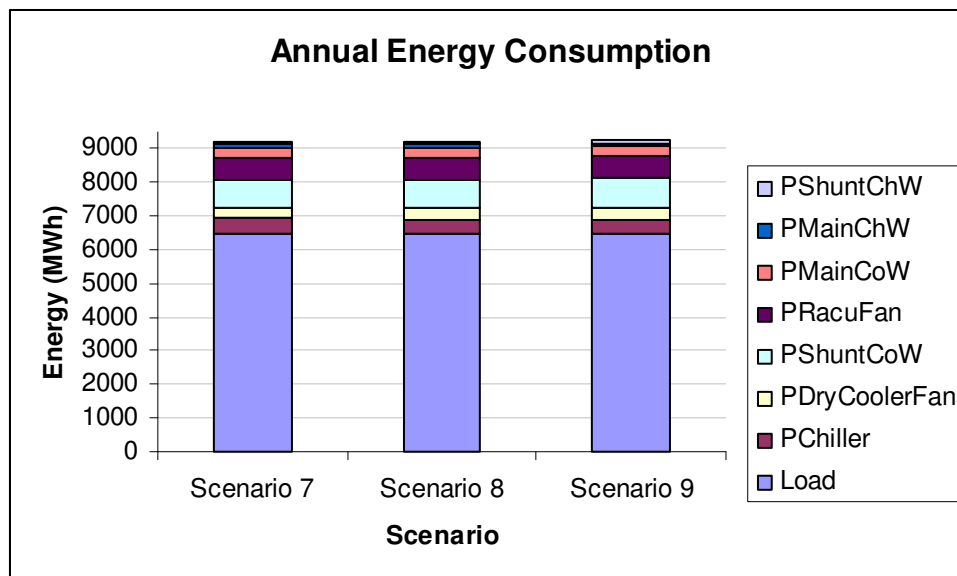| Scenario	Chilled Water Setpoint (TChW) ºC	Data Centre Room Temperature (TRoom) ºC	Ambient Temperature (used for Free Cooling) ºC	Temperature of cooling water (leaving the dry cooler) ºC
Scenario 7	11	26	10	10
Scenario 8	11	26	14	10
Scenario 9	11	26	18	10

Figure 49: Annual Energy Consumption with free cooling, Chilled Water Setpoint = 11°C

Where:-

PChiller = Power consumed by the chiller

PDryCoolerFan = Power consumed by the dry cooler fan

PMainChW = Power consumed by the main chilled water pump

PMainCoW = Power consumed by the main cooling water pump

PShuntCoW = Power consumed by the chilled water shunt pump

PRacuFan = Power consumed by the RACU fan

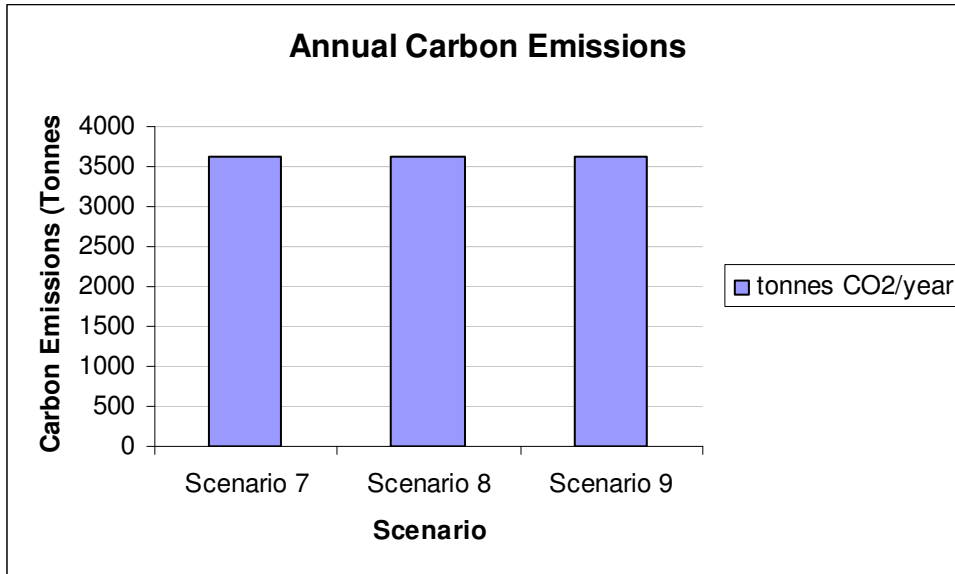PShuntChW = Power consumed by the chilled water shunt pump

Figure 50: Annual Carbon Emissions with free cooling, Chilled Water Setpoint = 11°C

Annual Carbon Emissions			
	MWh(elect)/year	Tonnes CO_2/year	CO_2 Saving
Scenario 7	1919	810	
Scenario 8	1932	820	-1%
Scenario 9	1967	830	-2%

Figure 51: Annual Energy Consumption and Carbon Emissions for Chilled Water Setpoint = 11°C

CO_2 analysis was performed using: -

CO_2 emitted from Gas = 0.19 kg CO_2/ kWhr (Carbon Trust)

CO_2 emitted from Electricity = 0.43 kg CO_2/ kWhr (Carbon Trust). Note that the carbon emissions for electricity and gas, used throughout this thesis, are taken from the Carbon Trust. The Carbon Trust, 2007, states;

"In the case of electricity consumption it is in most cases impossible to identify with any certainty the source of generation that has been reduced or displaced. For this reason the Climate Change Levy Negotiated Agreements and the ETS are using an average carbon intensity factor for the estimation of carbon dioxide emissions from the consumption of electricity. This has been fixed at 0.43 $kgCO_2$/kWh."

Appendix 5- Preliminary Comparison Calculations for CCHP

Annual Energy Cost - Traditional		Annual Energy Cost - CCHP	
1158	kWe total site power demand	886	kWe total site power demand
8760	h/year 24/7 operation	8760	h/year 24/7 operation
10144080	kWh(elect)/year	7761360	kWh(elect)/year
£0.07	per kWhr (elect)	2.73	Gas input/electrical power ratio
£710,085.60	per year electrical power	21188512.8	kWh(gas)/year
		£0.01536	per gross kW(gas)
		£325,455.56	per year gas
		£384,630.04	saving c/w Grid
		54%	energy cost saving

Annual Carbon Emissions (Elect)		Annual Carbon Emissions (Gas)	
10144080	kWe total site power demand	21188512.8	kWh(gas)/year
0.43	kg CO_2/kWhr	0.19	kg CO_2/kWhr
4361.9544	tonnes CO_2/year	4025.81743	tonnes CO_2/year
		336.136968	Saving CO_2/year
		7.71%	% Savings of CO_2/year

The calculation for the power consumption for the traditional data centre was based on requiring 272 kW electricity for powering the chiller, 18 kW electricity for the chilled water and cooling water shunt pumps, 54 kW for the dry cooler and 74 kW for the RACU fan.

The site power (kWe) is less for the CCHP because the power required for the electric chillers is not included as absorption chillers using heat from the CHP unit are used. It should also be noted that the heat output from the CHP unit is enough to provide cooling for the absorption chiller chosen which has a COP of 0.8.

The cost of electricity (£0.07) was kindly provided by WSP.

The cost of gas (£0.01536) was kindly provided by WSP as was the gas input/ electrical power ratio.

Appendix 6- Potential Suppliers of Biodiesel

Name	Bio-diesel Type	Quantity (that can be purchased in)	Contact Details
Global Commodities UK Ltd	B100	1000 Litres (in an IBC) 32,000 Litres (tanker)	4 The Old Foundry, Market Street Shipdham Thetford Norfolk IP25 7LZ Tel: 01362 821 582 E-mail: info@globeco.co.uk
Argent Energy (UK) Ltd			The Rural Centre, West Mains, Ingliston, Newbridge, Midlothian, EH28 8LT T: 0131 472 4131 F: 0131 472 4139 info@argentenergy.com
Biofuels Corporation Trading Limited			16 Earls Nook Belasis Hall Technology Park

			Billingham
		122	Teesside
			TS23 4EF
			Telephone: +44 (0) 1642 371 600
			FAX: +44 (0) 1642 565 265
Eco Bio-Diesel Ltd	B100		Unit 5 Forbes Court,
			Billington Road
			Burnley
			Lancashire
			BB11 5UB
			Telephone : 01282 831777
			Fax : 01282 831777
			Email : info@ecobiodiesel.co.uk
Greenergy International Limited			London office
			Greenergy International Limited
			198 High Holborn
			London
			WC1V 7BD
			Tel +44 (0) 20 7404 7700
			Fax +44 (0) 20 7400

			4799
		123	Email mail@greenergy.com
			Edinburgh office
			Greenergy Bioenergy Ltd
			18b Liberton Brae
			Edinburgh
			EH16 6AE
			Tel +44 (0) 131 672 9290
			Fax +44 (0) 131 672 9299

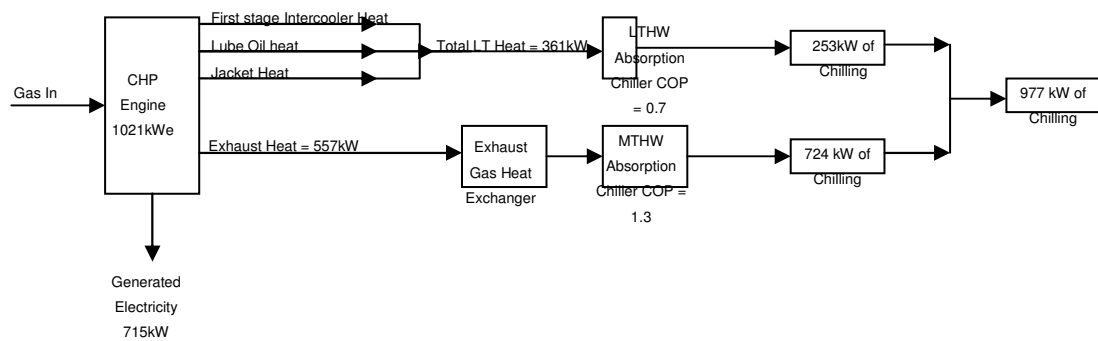# Appendix 7- Generator in put information for TRNSYS 16

Energy Balance of CHP Unit

This information regarding the CCHP unit has been provided in a quote to WSP.

Energy Balance at 100% Load
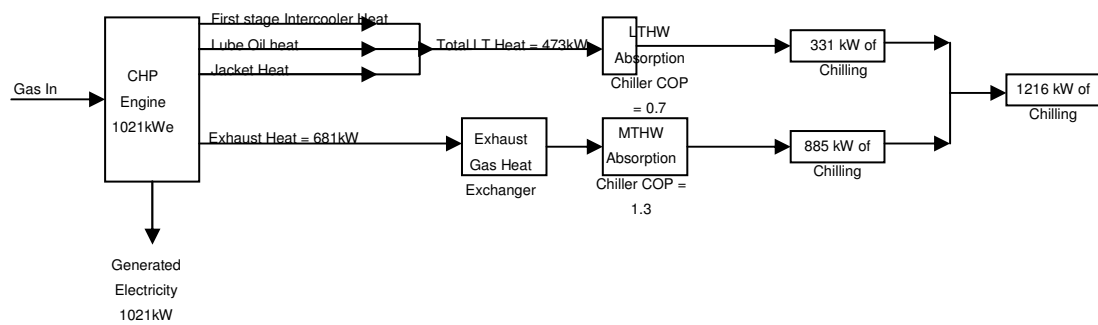
Energy Balance at 70% Load

Manufacturers Data

This information was supplied about the generator unit in a quote to WSP.

Performance and Efficiency		100%		80%		40%	
Fuel Input	kW	2161	100.0%	1770	100.0%	995	100.0%
Mechanical Shaft Power	kW	838	38.8%	670	37.9%	335	33.7%
Electrical Output	kW	798	36.9%	638	36.0%	319	32.1%
Heat Output from Jacket Water and Oil	kW	629	29.1%	510	28.8%	304	30.0%
Heat Output from Exhaust Gases (120°C)	kW	379	17.5%	319	18.0%	191	19.2%
Total Useable Heat Output	kW	1008	46.6%	829	46.8%	495	49.7%
Total Useable Energy	kW	1806	83.6%	1467	82.9%	814	81.8%
Intercooler Heat Output	kW	141	6.5%	127	7.2%	82	8.2%
Radiated and Unaccounted for Heat	kW	33	1.5%	31	1.8%	21	2.1%

Temperature and Flows		100%	80%	40%
Fuel mass flow	kg/hr	179.3	146.9	82.6
Fuel volume flow	nm^3/hr	224.1	183.6	103.2
Ventilation air volume flow (incl. comb .air)	nm^3/hr	16700	16700	16700
Combustion air mass flow	kg/hr	4801	3853	2123
Combustion air volume flow	nm^3/hr	3707	2975	1639
Exhaust gas mass flow (wet)	kg/hr	4980	4000	2206
Exhaust gas volume flow (wet)	nm^3/hr	3900	3133	1728
Exhaust gas volume flow (wet) @ 120°C	m^3/hr	5544	4453	2456
Jacket water flow	m^3/hr	80	80	80
Intercooler water flow – minimum	m^3/hr	50	50	50
Secondary water flow – minimum	m^3/hr	89	89	89
Maxium return water inlet temperature	°C	80	80	80
Secondary water outlet temperature	°C	90.0	88.2	84.9
Maximum intercooler water inlet temperature	°C	55	55	55
Intercooler water outlet temperature	°C	57.7	57.4	56.5
Exhaust gas temperature before cooler	°C	378	390	414

Appendix 8 – Bio-diesel Calculations Excel Spreadsheet

Bio Diesel Calculations

Seconds in a year	31536000	
Gross Calorific Value of Diesel J/Kg	46000000	Source: www.kayelaby.npl.co.uk/chemistry/3_11/3_11_4.html
Density of No. 2 Diesel fuel (kg/m^3)	850	Source: www.onlineconversion.com/density_common.htm

	Watts=joules/second	Energy (Joules) per year (E)	Mass (Kg)=E/calorific value
No.2 Diesel	1000000	3.1536E+13	685565.2174
	Production of CO2 (kg)	Tonnes of CO2	
No.2 Diesel	2125252.174	2125.252174	

	Energy (Joules) per year (E)	Mass (Kg)=E/calorific value	Volume (m^3)=mass/density
B2	3.1536E+13	688906.3756	809.9063903
B5	3.1536E+13	693970.6228	814.9977954
B20	3.1536E+13	720277.9308	841.4461808
B50	3.1536E+13	778370.4492	899.8502303
B100	3.1536E+13	895456.275	1017.563949
	Tonnes of CO2		
B2	2091.421272		
B5	2040.143231		
B20	1773.768549		
B50	1185.552678		
B100			

B2	Volume of Diesel Fuel	Mass of Diesel Fuel=density of diesel*volume	Production of CO2 (kg)
B5	793.7082625	674652.0231	2091421.272
B20	774.2479056	658110.7198	2040143.231
B50	673.1569446	572183.4029	1773768.549
B100	449.9251152	382436.3479	1185552.678

1kg No. 2 Diesel=		3.1	kg CO2
B2 Bio-diesel	Cp (J/kg)	45776902.52	
B5 Bio-diesel	Cp (J/kg)	45442845.8	
B20 Bio-diesel	Cp (J/kg)	43783099.07	
B50 Bio-diesel	Cp (J/kg)	40515412.72	
B100 Bio-diesel	Cp (J/kg)	35217800	

1kg of biodiesel=		2.8	kg of CO2
B2 Bio-diesel	Density (kg/m^3)	850.6	
B5 Bio-diesel	Density (kg/m^3)	851.5	
B20 Bio-diesel	Density (kg/m^3)	856	
B50 Bio-diesel	Density (kg/m^3)	865	

| B100 Bio-diesel | Density (kg/m^3) | | 880 |

B2	Volume = 1m^3	mass = volume*density	Joules=joules/kg * mass
Biodiesel	0.02	17.6	619833280
Diesel	0.98	833	38318000000
	Combined Joules	Calofiric Value (J/kg)	
Biodiesel	38937833280	45776902.52	

B5	Volume = 1m^3	mass = volume*density	Joules=joules/kg * mass
Biodiesel	0.05	44	1549583200
Diesel	0.95	807.5	37145000000
	Combined Joules	Calofiric Value (J/kg)	
Biodiesel	38694583200	45442845.8	

B20	Volume = 1m^3	mass = volume*density	Joules=joules/kg * mass
Biodiesel	0.2	176	6198332800
Diesel	0.8	680	31280000000
	Combined Joules	Calofiric Value (J/kg)	
Biodiesel	37478332800	43783099.07	

B50	Volume = 1m^3	mass = volume*density	Joules=joules/kg * mass
Biodiesel	0.5	440	15495832000
Diesel	0.5	425	19550000000
	Combined Joules	Calofiric Value (J/kg)	
Biodiesel	35045832000	40515412.72	