Simulation of IST´s data center cooling system using Simulink

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Mechanical Engineering

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Procrastination makes easy things hard, hard things harder.

Mason Cooley

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Writing this thesis would not have been possible with the help of many people, to which I am most grateful.

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Resumo

Data Centers, ou Centros de Dados, são instalações que permitem o bom funcionamento de todo o tipo de atividades *online*. Este tipo de equipamento consome quantidades consideráveis de energia a nível mundial (1.8% do consumo total de eletricidade nos EUA em 2014). Dependendo do seu tamanho, podem ser uma instalação por si só, ocupando edifícios inteiros, ou ser parte integrante de outros edifícios, como é o caso do *Data Center* do Instituto Superior Técnico. Este representa uma grande parte do consumo total de energia do Pavilhão Central, que por si já é o segundo edifício com maior consumo elétrico do campus Alameda desde 2015.

Para entender melhor como aumentar a sua eficiência energética, a modelação física de sistemas térmicos revela-se uma poderosa ferramenta.

Pretende-se com esta Tese modelar o sistema de arrefecimento do Centro de Dados do IST, por forma a entender melhor quais as melhores estratégias de diminuição de consumo de energia.

Foram testadas 3 potenciais medidas de melhoria da eficiência energética do *Data Center*. Estas medidas serão a implementação de um sistema de aproveitamento do potencial de arrefecimento do ar atmosférico, quando as condições climatéricas assim o permitirem. Também será explorado o impacto de alterar diferentes pontos de controlo de temperatura, como a temperatura da saída de água do *Chiller* e a temperatura do ar da sala do *Data Center*. Além disso, será avaliado o potencial energético de mudar o presente *Chiller* por um do tipo água-água assim como um sistema de contenção de ar quente para os servidores. Finalmente, ter-se-á em consideração a tendência de crescimento desta instalação para perceber quais as potenciais necessidades de expansão a nível de equipamento de arrefecimento.

O modelo será desenvolvido através do uso da plataforma *Simulink* e das suas componentes de modelação de sistemas térmicos (*Simscape*). Numa primeira fase, os vários componentes individuais do sistema serão modelados individualmente. Quando a sua precisão for satisfatória, serão por fim agregados num sistema completo, que será também calibrado ao longo de sucessivas etapas até se revelar robusto e preciso.

Palavras-chave: Data Center, Simulink, Eficiência Energética

Abstract

Data centers are a type of installation which allows for the storage, and processing of data. This allows for organisations to offer all types of services online. This type of facility represents a considerable share of the total energy consumption around the globe (1.8 % of all energy consumption in the USA in 2014). Depending on their size, data centers can occupy their own building or be part of a bigger building, as is the case of Instituto Superior Técnico's data center. This installation represents a large share of the total power consumption of the central building, which itself is the building with the highest energy consumption of the Alameda Campus. To better understand how to increase the energetic efficiency, physical modelation of thermal systems reveals itself as a powerful tool.

The objective of this Thesis is to model the cooling system of the aforementioned data center, in order to better understand what are the most cost effective energy consumption reduction strategies. The considered measures will be the implementation of an air economiser within the overall data center HVAC system. In addition to this, the impact of changing the server room temperature and the chilled water temperature setpoints will also be explored. Moreover, the impact of changing the current chiller for a water-water one and implementing a hot air containment system within the server room will also be considered. Finally, an additional consideration will be done on the further scalability of this data center in terms of available cooling capacity.

The model will be developed using Simulink and it's thermal modelling resources (Simscape). In a first phase, the several multiple components will be modelled. When they are considered to represent accuratelly enough their real life counterparts, they will be put together on a complete system which will also be calibrated until it´s considered to represent real life satisfactorily.

Keywords: Data Center, Simulink, Energy Efficiency

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Nomenclature

Greek symbols

α Volume to area ratio. (m$^3/m^2$)

β Angle of throughs in a heat exchanger (°)

Δ Variation operator.

η Efficiency.

κ Thermal conductivity coefficient. (W/m.K)

Λ Gradient operator.

μ Molecular viscosity coefficient. (Pa.s)

Φ Absolute humidity. (g/m^3)

ϕ Ratio between the developed length and protracted length of the troughs in the heat exchanger.

ρ Density. (kg/m^3)

ε Heat Exchanger effectiveness.

Roman symbols

A Area. (m^2)

b Heat exchanger dimentional parameter.

C Heat capacity rate. (J/K)

c_p Specific heat. (kJ/kg.K)

d Diameter. (m)

f Frequency. (Hz)

h Heat transfer coefficient. (W/m$^2.K$)

L Length. (m)

Nu Nusselt number.

P Power. (W)

p Pressure. (Pa)

Pr Prandtl number.

q, Q Rate of heat transfer. (W)

Re Reynolds number.

T Temperature. (ºC)

V Voltage. (V)

Abbreviations

$ASHRAE$ American Society of Heating, Refrigerating and Air-Conditioning Engineers.

CFD Computational Fluid Dynamics.

$CRAC$ Computer Room Air Conditioner.

$CRAH$ Computer Room Air Handler.

DC Data Center.

EER Energy efficiency ratio.

$HVAC$ Heating Ventilation and Air Conditioning.

IST Instituto Superior Técnico.

IT Information Technology.

$MERV$ Minimum Efficiency Reporting Values.

NTU Number of transfer units.

PUE Power Usage Effectiveness.

Chapter 1

Introduction

1.1 Motivation

A Data Center is a building (or part of a building) hosting information technology (IT) systems. They are essential in today's society, enabling companies and organizations to provide the most diverse type of internet-based services, computational services, among others. Recent development of this kind of services led the existence of some type of data center in the present days at most medium to big sized companies and organisations (like IST). These can be of varying size depending on the expected load those services would have to sustain.

Recently, it has become more common for companies to subcontract other companies to handle of these kinds of IT needs. Companies such as Amazon Web Services (AWS), Google, IBM and Microsoft are the biggest players in this field, with each owning 45 or more hyperscale-size data centres, as of 2017[1]. Their growth relative to other types of IT infrastructure is illustrated in Figure 1.1.This growth is due partly to the superior energy efficiency of larger scale Data Centres and the harnessing of natural/free cooling instead of compressor based cooling system architectures.

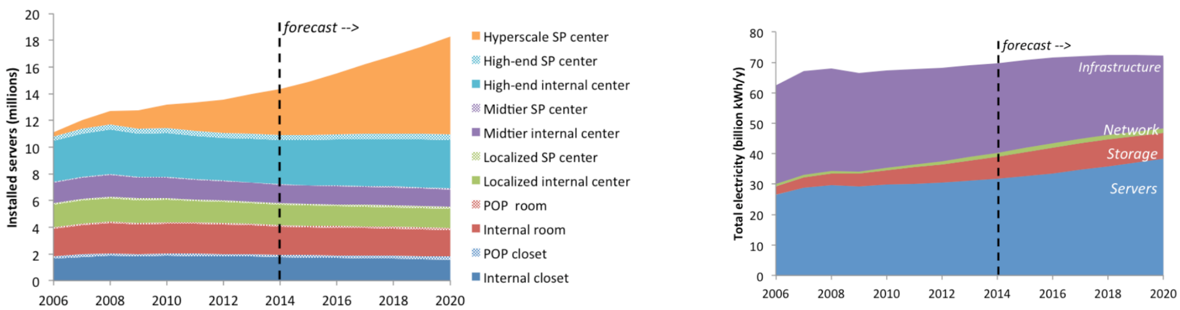

Figure 1.1: Data Center energy consumption by sectors [2]

Data Centers require different supporting systems. The electrical side of operations makes sure the energy to power the computers and other devices is supplied at varying voltages. It is also important to make sure that the data center can communicate with the exterior - as such, a communication network is essential between the computers themselves and to the outside world. Finally, due to the considerable amount of power required by these installations (from tenths of kilowatts to hundreds of megawatts), it becomes necessary to dissipate the heat associated with the energy consumption to the exterior in the most efficient way possible. Cooling a data center typically consumes around 40% to 50% of the total power consumption in this kind of facility, as depicted in figure 1.2, that shows a typical energy consumption breakdown in a data center.

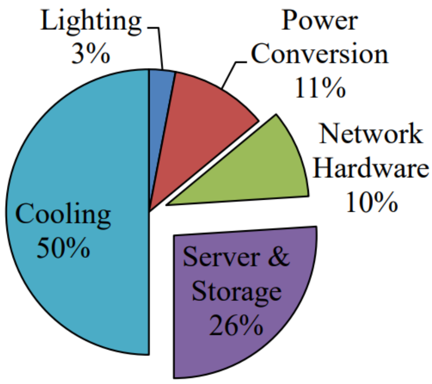

Figure 1.2: Typical Data Center energy consumption breakdown [3]. The IT power is represented in the ”Servers and Storage” section.

Data centers worldwide represented 1.3% of the total yearly consumption (2% in the United States) in 2012 [4]. As information technologies become more and more essential for all kinds of economic activities, it would be expected that this consumption would increase correspondingly. In fact the consumption has remained relatively stable for the last years, mainly because the energy efficiency in this field has improved faster than the energy demand. According to [5], although there is today around 12 times as much internet traffic as there was in 2010, the total data centre energy consumption has remained relatively constant, as represented in figure 1.3.

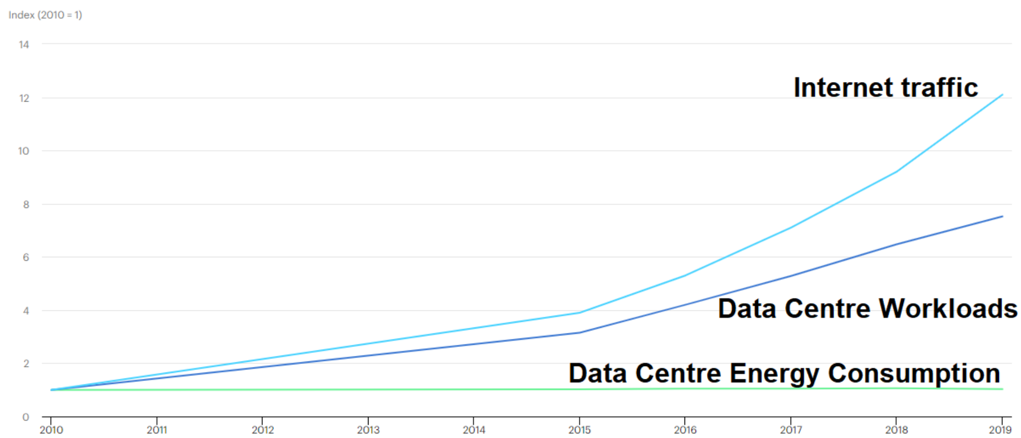

Figure 1.3: Global data center trends [5]

To measure how efficient an Data Center is, the most popular metric is known as Power Usage Effectiveness (PUE):

$$PUE = \frac{Total\ Facility\ Power\ Consumption\ (kW)}{IT\ Equipment\ Power\ Consumption\ (kW)} \tag{1.1}$$

,where $Total\,Facility\,Power\,Consumption$ includes the IT power, as well as that used in cooling and the rest of the power consumed in the facility. Ideally, one wants a PUE as low as possible, or an overhead as small as possible when compared with the power consumed by the servers. It can be seen in figure 1.4 that the average worldwide PUE has been decreasing, as more and more energy saving measures are implemented and the cooling efficiency is increased.

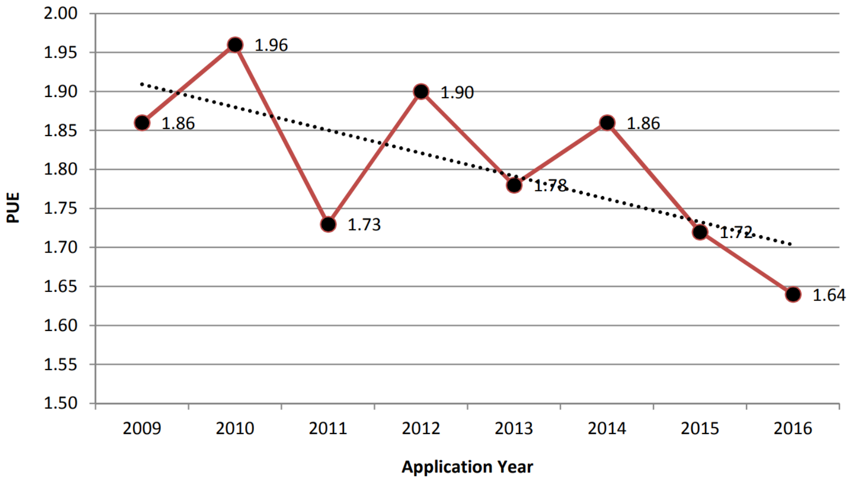

Figure 1.4: Average PUE for 289 Data Centers in Europe and the USA [6]

To keep up this increasing efficiency trend new energy saving methods need to be developed, in order to keep the carbon footprint of this essential type of infrastructure low. Heating, ventilation and air conditioning (HVAC) system modelation is a far more cost effective way to improve the operational efficiency of the system than replacing existing equipment [7], since one can find the optimal control for a given system and its loads. In addition to this, as the different available softwares become increasingly more complex and detailed, one can model the most diverse components of HVAC systems to a considerable level of detail.

Trčka and Hensen [8] categorize the different available types of HVAC modelling softwares into tools for pipe/duct sizing, tools for equipment sizing and selection, tools for energy performance analysis, tools for system optimisation, tools for control analysis and control optimisation and simulation tools for real-time performance optimization. All these different instruments allow, in their own way, for better and more efficient operation and design of HVAC equipment.

1.2 Topic Overview

Although data centers are a relatively recent type of installation, there is a considerable amount of accumulated knowledge from engineers working in this field, as well as a dynamic research activity on this subject and all it's branches.

The American Society of Heating Refrigeration and Air-Conditioning Engineers (ASHRAE) annually includes in their publications considerations about the best practices regarding Data Centre design and operations based on the experience of engineers in the field. This is an evidence of the importance of this thematic worldwide. ASHRAE[9] defines the basic concepts used throughout the industry. In addition, their recommendations for the most adequate operational parameters for data centers are regarded as the best reference, and are followed with great attention throughout the industry.

Khalaja and Halgamuge [10] defines five thermal management methodology levels in order to increase the efficiency of a Data Centre:

- **Chip level**- By optimising the frequency and voltage at which a processor chip runs at, the power it consumes (typically $P \propto Vf^2$) is minimised. Furhermore, by increasing the heat transfer coefficient between the processor and the cooling medium, it is possible to considerably reduce the power to cool down the chips. Typically servers have been air-cooled but innovation in this field has led to the research with single-phase cooling, as well as two-phase cooling. Here refrigerants can either be in direct or indirect (heat pipes) contact with the chip, taking advantage of the high heat transfer coefficient that phase change allows.

- **Server level**- By balancing the amount of workload assigned to each server in order to avoid the formation of hot spots, consequently increasing the cooling efficiency. Khalaja and Halgamuge [10] further subdivides this category into Workload Dispatching, Workload Migration and Dynamic Component Deactivation, based on how the assignment of computational tasks to each server is done.

- **Rack**- CFD investigation has been done in order to find more adequate designs for server racks, as well as optimisation of placement of servers within racks in order to improve the airflow and cooling performance. By implementing these kinds of strategies, heat dissipation could be increased by 50% [11].

- **Plenum**- Investigation has been done on ways to control the pressure distribution across the plenum, in order to achieve a more uniform airflow, by controlling the area and shape of the plenum

- **Room**- To take the maximum advantage out of a Data Centre cooling system it is imperative to have the servers receive an uniform airflow. One of the easiest ways to achieve this is through the implementation of Cold or Hot-air Containment, where the cold and hot air zones of a data center are separated physically to prevent undesired mixing. Niemann et al. [12] found that energy savings as high as 15% could be achieved when using this technique.

Since the objective of this Thesis is to develop a model of the IST data center, it becomes relevant to explore existing literature on the topic of modelation of this type of systems as well.

Several authors also worked on high level modelling, where the whole data center cooling system is modelled. Dayarathna et al. [13] summarises the main investigation done in this field up to 2015.

Wemhoff et al. [14] developed a modular language, where different blocks representing different components, such as Cooling Towers, Chillers, etc. can be joined to create more complex systems, such as a complete data center. The relevant variables are then calculated using mass and energy balances.

Fu et al. [15] developed a data center package for Modelica, an open source modular programming language and environment. The authors investigated the feasibility of the implementation of economisers in the context of data center and their performance for different levels of server load.

Lapussan et al. [16] and Kassas [17] explored the Simulink enviroment for designing HVAC systems for buildings. Lapussan et al. [16] used the Lumped Capacitance approach, designing each individual block, while Kassas [17] used the Simscape set of modelling tools, available by default in Simulink.

Despite all this investigation, none was found using Simulink for the specific case of a data center.

1.3 Objectives

The objectives set for this Thesis are primarily, to develop a SIMULINK/Simscape model that accurately represents the working principles of IST data center. Such model should be able to output different relevant variables, such as mass flows for the different fluids (air, water and refrigerant), temperatures, power consumption, among others. These shall be dependent on the input parameters, such as heating load and ambient temperature/humidity.

All these variables shall then be compared to measured data of the real system to determine whether the model´s accuracy.

After the aforementioned model is found to be sufficiently accurate, different energy saving strategies will be tested using the base model, in order to have an insight into which would be the most beneficial for the considered data center.

1.4 Thesis Outline

This Thesis comprises 5 chapters. Chapter 1, the present chapter, introduces the subject of the present study.

Chapter 2 explores the existing literature regarding data center cooling as well as industry best practices.

In Chapter 3, the different steps taken to achieve the model are laid out sequentially. The necessary calculations, modelling choices and necessary simplifications are explained in detail.

Chapter 4 addresses the different possible energy saving measures and their individual impact is measured using the developed model. The results are then presented graphically, and the necessary considerations are made.

Finally, in Chapter 5 the outcome of this Thesis is laid out and the most important findings are summed up. In addition to this, economic considerations are also taken into account here for the most promising energy saving measures.

Chapter 2

Data Center Operation and Theoretical Background

2.1 Data Center Overview

As mentioned earlier, Data Centres enable a considerable part of the human population to access the internet and the services it provides. To do this, the Data Centres host large quantities of servers, in order to process and store large amounts of data.

To perform their function correctly, power needs to be supplied to the servers and the associated heat generated consequently dissipated, in order to keep the components at an adequate temperature. Up until recently, servers were cooled exclusively using air as a cooling medium, through the use of small fans driving the air flow through the components.

With power densities rising as high as $200W/cm^2$ in modern servers [18], new cooling solutions have been proposed and have been increasingly adopted by some companies. As a consequence, there is now a distinction between the older ”legacy” Data Centres, where the cooling medium is air and newer methods, such as water cooling and phase change systems, where a water or a boiling fluid loop removes heat directly from the chip. These take advantage of the much higher heat transfer rates, resulting in a more efficient system. However, these systems still remain a minority, and will therefore not be the focus of this work. As a consequence, ”Legacy” Data Centers will simply be referred as ”Data Centers”.

<table>
<tr><th>Classes</th><th>Information Technology Equipment</th><th>Environmental Control</th></tr>
<tr><td>A1</td><td>Enterprise Servers, Storage Products</td><td>Tightly Controlled</td></tr>
<tr><td>A2</td><td rowspan="3">Volume Servers, Storage Products, Personal Computers,Worstations</td><td>Some Control</td></tr>
<tr><td>A3</td><td>Some Control, Use of Free Cooling Techniques When Allowable</td></tr>
<tr><td>A4</td><td>Minimal Control</td></tr>
</table>

Table 2.1: Information Technology equipment classes [19]

Another important feature of a data center is the necessity to operate without interruption. To do this, the operating parameters should remain within recommended limits in order to maximise reliability ans minimise downtime. Table 2.1 and Figure 2.1 illustrate ASHRAE recommended environmental conditions for Data Centres depending on their class, which is tied to their type of use.

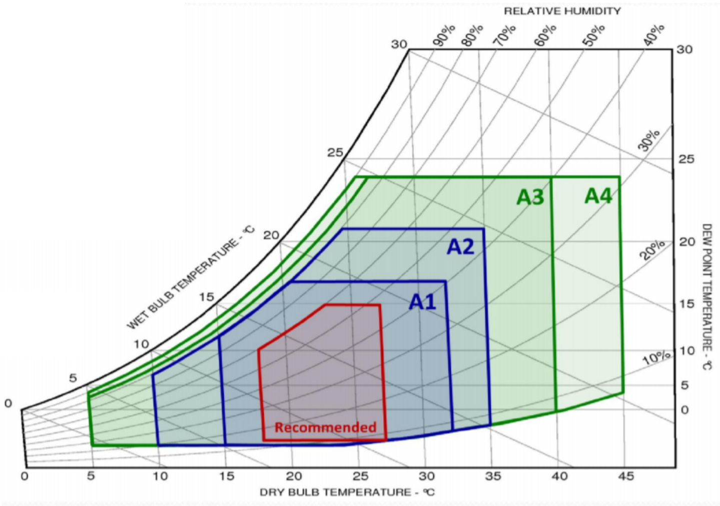

Figure 2.1: Recommended and allowable thermal envelopes for Data Centres in a psychrometric diagram [9].

Humidity should be kept above a minimum threshold, in order to avoid electrostatic discharge through the air, which tends to occur in very dry environments and can damage electronic equipment. On the other hand, it should also remain under a maximum of around 90% to steer clear from the possibility of condensation occurring, which could cause short circuits and damage electronic equipment as a consequence.

To achieve the best possible level of space utilisation, servers are stacked on top of each other, housed on what are normally called ”server racks”.

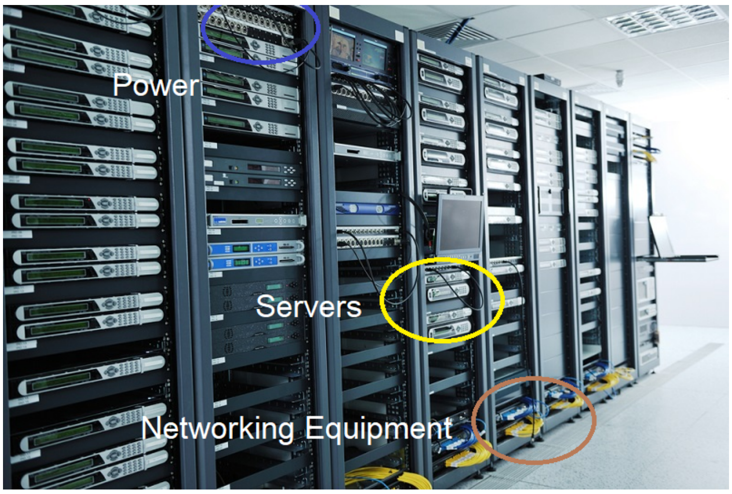

Figure 2.2: Typical server racks [20]

Figure 2.2 exemplifies a typical server rack. The servers themselves are housed in the middle section, while the networking equipment is typically at the bottom and power delivery apparatus in the top section.

One of the simplest measures taken to minimise cooling costs is to implement hot or cold air containment of server racks, as illustrated in figure 2.3. These types of layout avoid the creation of hot spots - areas where the temperature is much higher than the rest of the room, as well as avoiding unnecessary mixing of cold and hot air. This increases the cooling efficiency, because the temperature difference between the hot air and the cooling fluid increases, increasing the heat transfer rate accordingly.

In the hot air containment case, cold air is supplied to the room (the methods to cool the air will be discussed further ahead), which then enters the servers and exits as hot air into a containment aisle that is, in turn, connected back to the cooling device. On the other hand, a data center with a cold air containment type layout usually has raised floors through which the cold air flows from the cooling device to the contained cold aisle. The hot air then exits from the back of the server and is fed to the room, and then back to the cooling unit.

Niemann and Avelar [12] compared the hot and cold air containment techniques using a theoretical model in an effort to conclude on which has more energy saving potential. It was determined that for an adequate working environment for both workers and machines, the hot air containment strategy proved superior, achieving energy savings as high as 15%

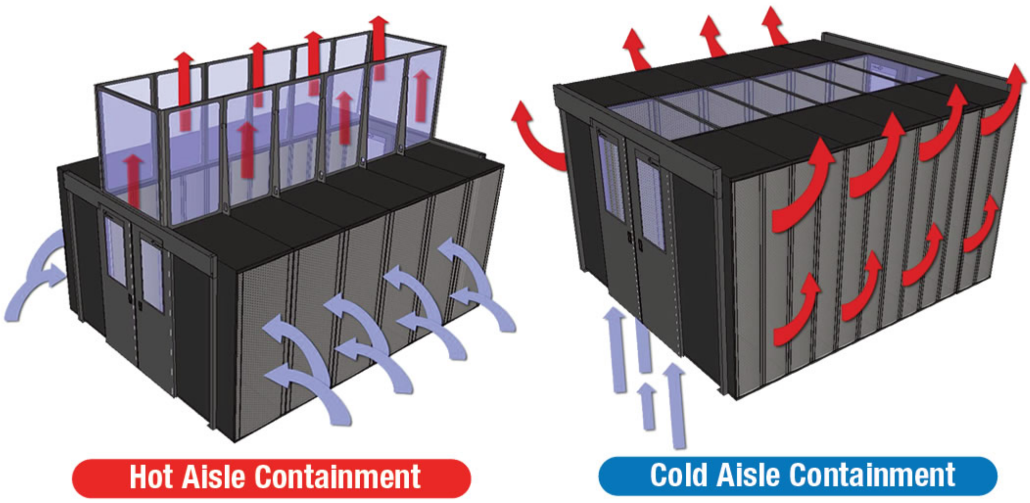

Figure 2.3: Cold and Hot air containment representation, from [21]

There are many different methods to cool the air inside a data center. Inside the data center Room, one can have the so-called as computer room air handler (CRAH) or a computer room air conditioner (CRAC). The first one is usually consisted of a cooling coil, where cold water flows and a fan pushing air through the coil. The cold water usually comes from either a cooling tower or an air or water-cooled chiller. The usual arrangements are represented in figure 2.4

A CRAH, on the other hand, also sits inside the server room, but takes advantage of a refrigeration cycle to cool the air. The evaporator in this cycle removes the heat from the air which is then transferred to the water loop as the refrigerant cools down in the condenser. In this case, the water can be cooled in a cooling tower, for example.

The decision of implementing a system using a CRAH or CRAC, or deciding between using chillers

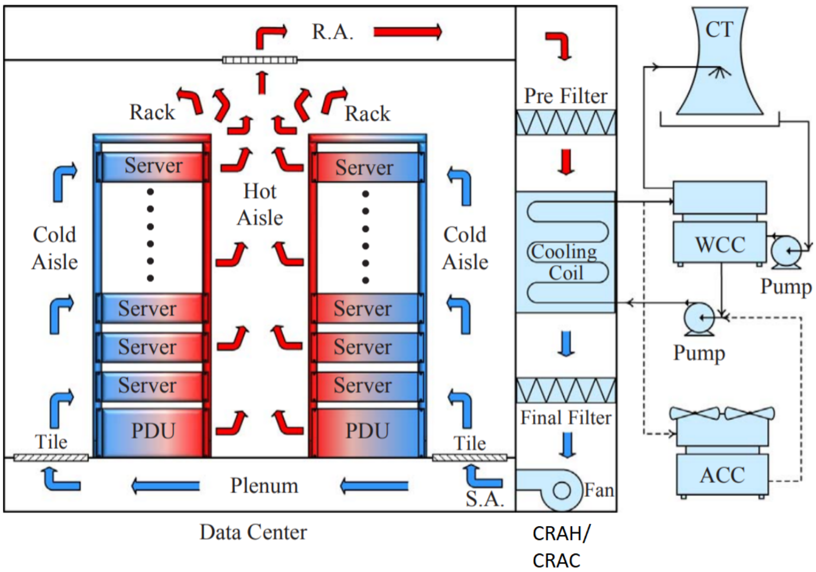

Figure 2.4: Typical Data Centre cooling system layout [10]

or cooling towers is dependent on a number of factors, like the climate and economic considerations. In cooler climates, one can have a chiller-less installation, relying only on cooling towers with no necessity for a refrigeration cycle for the majority of the year for example. This would be harder in a more temperate climate such as Portugal's. In addition to this, the size of the data center is also important - for bigger data centers, the cooling efficiency is more relevant, since it will consume much more energy to be kept at an acceptable temperature. Moreover, since keeping the installation without interruptions is essential, redundancy is also a factor to consider when deciding between the number of different cooling systems or how many units of the same system to implement for a single Data Centre.

When designing a data center, the calculation for total cooling requirements, one has to take into consideration many different heat sources, the most relevant one being the IT power. The different factors for a typical data center are summarized in table 2.2.

	Heat output calculation (W)
IT Equipment	Same as IT Load
Power Supply (UPS)	0.04*(Power system rating) + 0.06*(IT Load)
Power Distribution	0.02*(Power system rating) + 0.02*(IT Load)
Lighting	21.53 * (floor area, in m^2)
People	100*(Max number of personnel in DC)

Table 2.2: Typical Cooling loads in a Data Centre, from [22]

2.2 Fundamentals of Heat Transfer

The adequate way to divide Heat Transfer mechanisms into different types is Convection, Conduction and Radiation:

Conduction is defined as transfer of thermal energy through matter by transfer of molecular kinetic energy with no net (macroscopic) displacement of the molecules and is more relevant for solids.

Convection, on the other hand is defined as the macroscopic transfer due to the movement caused within a fluid either under the driving force of a turbomachine or by the tendency of hotter and therefore less dense material to rise, and colder, denser material to sink under the influence of gravity. It is therefore more relevant in gases an liquids.

Since the temperatures are not very high, radiation heat transfer is negligible when studying a Data Centre.

One of the foundational equations that describe conduction heat transfer processes is the Fourier Law (Equation 2.1). It states that the rate of conduction heat transfer through a medium is equal to the negative of the thermal conductivity constant (k) multiplied by the temperature gradient (∇T).

$$\vec{\mathrm{q}} = -k\nabla T \tag{2.1}$$

Combining Equation 2.1 with the energy balance to an infinitesimal control volume $dx.dy.dz$, where the only mechanism of heat transfer present is conduction with the definition energy accumulation in a control volume $dx.dy.dz$, ,$\rho dV C_p \frac{dT}{dt}$, and the volumetric heat rate generation $q.dV$ where ρ and c_p are the material's density and specific heat respectively, one can obtain the general unsteady heat conduction equation for 3 dimensional Cartesian coordinates:

$$\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) + \frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right) + q_V = \rho c_p \frac{\partial T}{\partial t} \tag{2.2}$$

Adapting this equation for the case under study, and provided the necessary boundary conditions, one can calculate a temperature distribution in a unsteady system.

The convection heat transfer rate (Q) is given by equation 2.3, where h is the convection heat transfer coefficient, A is the surface area and ΔT is a temperature difference between the solid and the fluid. Coefficient h is dependant of the specific case, and has to be calculated accordingly using the correct empirical correlation.

$$Q = hA\Delta T \tag{2.3}$$

To model the heat transfer in complex systems, one must divide such system into several simpler subsystems between which the heat transfer takes place, and only then can the equations mentioned previously be applied. This process of discretization will be explained further along this thesis.

2.3 Fundamentals of Thermodynamics - Refrigeration Cycle

One of the fundamental components of the IST Data Center installation is the Air-Cooled Chiller. This device removes heat from the chilled water loop by using a refrigeration cycle, schematized in figure 2.5. A refrigeration cycle takes advantage of the fact that latent heat transfer (due to phase change) is much more efficient than sensible heat transfer. By regulating the pressure of a fluid finds itself one can control it's boiling and condensation temperatures. This becomes very useful in many fields, specially in HVAC systems.

A typical refrigeration cycle uses the following main components:

- **Evaporator (evolution 4-1 in Figure 2.5)** - The refrigerant enters here as a mix between a liquid and a vapour and boils until it becomes a saturated vapour, increasing the fluid's enthalpy. The process of phase change allows very high heat transfer coefficients, which in turn, allows a large amount of heat to be removed from the chilled water loop using a relatively small mass flow rate. These are usually components with a very high surface area to volume ratio, in order to maximise the heat transferred.

- **Compressor (1-2)** - After the leaving thee evaporator, the refrigerant flows into the compressor where the pressure increases, and, consequently, so does the temperature. Since this is not an isentropic process, the entropy increases due to irreversibilities. The compressors used in a refrigerant cycle can be of the scroll, reciprocating, rotary, among others. The chiller used in IST's data center is one of the scroll type (Figure 2.6). It works using two spiral parts - one fixed and one static. As one of them rotates, it traps some refrigerant between it and the static part, compressing the refrigerant.

- **Condenser (2-3)** - The heat admitted into the cycle in the evaporator needs to be rejected to the outside environment. This is done in the condenser (seen in the top part of Figure 2.7 and Figure 2.8). Here, the refrigerant enters as a high pressure, high temperature superheated vapour and is cooled down in an isobaric process into a mix of vapour and liquid, or saturated liquid. As a consequence of this process, the temperature decreases. In the case of IST's Data Center chiller, the condenser is cooled by air. It is composed of several coils running in parallel with fins in between them, in order to maximise the surface area and, therefore, the amount of transferred heat. The air is forced through the condenser by 2 axial fans, flowing perpendicularly to the refrigerant.

- **Expansion Valve (3-4)** - This component drops the pressure of the circulating refrigerant through the use of a sudden change in area. By reducing the opening area, one can decrease the pressure of the circulating fluid, and as a consequence decreasing the temperature as well. This process is, for engineering processes, considered isenthalpic.

2.4 Heat Exchanger modelling - $\varepsilon - NTU$ model

In an air cooled data center, heat is transferred sequentially between the air in the Data Centre and a cold water flowing in the cooling unit, water and refrigerant in the evaporator and, finally, between the refrigerant and the outside air in the condenser. The broad designation for each of these components, the cooling unit, the evaporator and the condenser is heat exchanger. This type of device enables heat transfer between two fluids that can either be liquids, gases or both. To model this process many

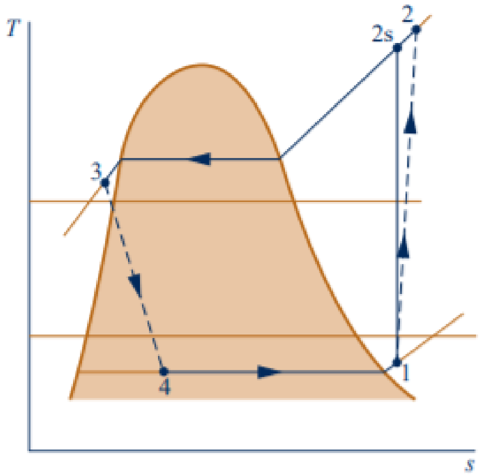

Figure 2.5: Temperature - Entropy plot of a typical refrigeration cycle [23]

Figure 2.6: Typical scroll compressor and illustration of mechanism [24]

Figure 2.7: Typical air cooled Chiller [25]

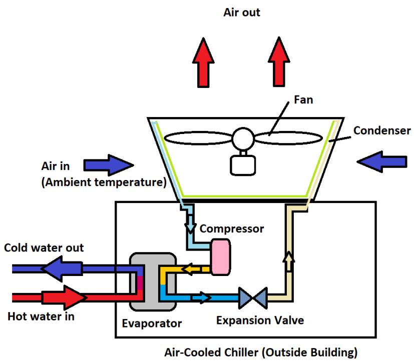

Figure 2.8: Representation of the Chiller's refrigerant cycle and components

methods exist but the $\epsilon - NTU$ method is by far the most common and as such will be briefly explained here.

The simplest category into which one can divide Heat Exchangers is connected to the relative flow directions of the circulating fluids. The three most common are, according to [26],

- **Parallel-Flow** - In this type of heat exchanger, the two fluids flow parallel to each other and in the same direction, as depicted in figure 2.9a).
- **Counter-Flow** - Contrary to a paralell flow heat exchanger, in a counter flow heat exchanger, the two fluids circulate in opposite directions, as seen in figure 2.9b).
- **Cross-Flow** - Finally, in a cross-flow heat exchanger, the two fluids flow perpendicularly to each other (figure 2.9 c) and d)).

To find out the amount of heat transferred in a heat exchanger (q), one must first find out its effectiveness ε.

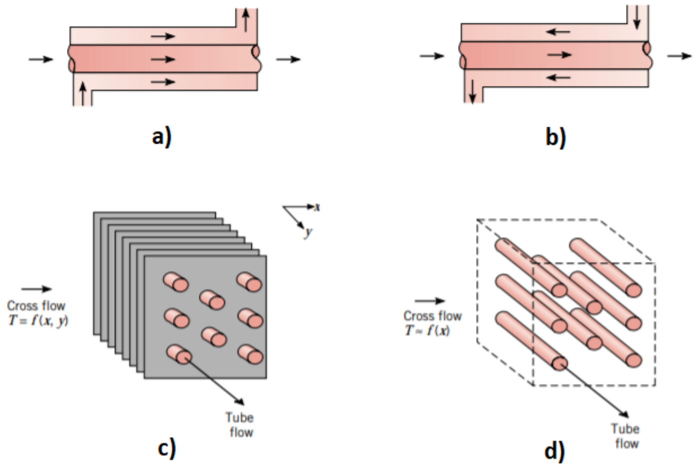

Figure 2.9: Types of Heat Exchangers, according to the relative flow direction [26]

$$\varepsilon = \frac{q}{q_{max}} \Rightarrow q = \varepsilon q_{max} \tag{2.4}$$

q_{max}, on the other hand, is defined as the greatest amount of heat transfer rate, possible in an infinitely long counter-flow heat exchanger.

$$q_{max} = C_{min}(T_{h,i} - T_{c,o}) \tag{2.5}$$

Where subscripts h and c are the hot and cold fluids, respectively, and o and i stand for the outgoing and entering conditions.

One can show that $\varepsilon = f(NTU, \frac{C_{min}}{C_{max}})$, where C_{min} and C_{max} are the smallest and largest heat capacity rates of the flows involved in the process. NTU on the other hand, is the *number of transfer units*, and is defined as $NTU = \frac{UA}{C_{min}}$, where U is the overall heat transfer coefficient and A is the fluid's contact area. Consequently, if these are known, as well as the type of flow, one can use one of the equations derived by [26] to calculate ε and then q.

2.5 Fundamentals of Thermodynamics - Cooling tower

A cooling tower can be thought of as a special type of heat exchanger where both fluids are in contact, that allows water to be cooled down by the atmospheric air mostly through evaporative but also latent cooling (these concepts will be further explained later in this section). It is used to remove heat from the most varied industrial devices in a economic way, since it harnesses the natural cooling power of the atmospheric air. In the HVAC context they can be used together with water cooled chillers or individually, depending on the design and if air temperature and humidity allow it.

As a result of using the atmospheric air to cool down the water flow, a cooling tower's performance is heavily dependent on the air's humidity and temperature - the more humid the air is, the harder it is to cool down the water, since the evaporative part of the heat transfer decreases.

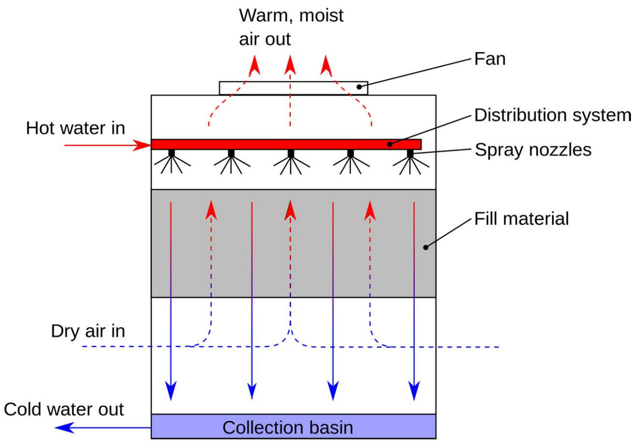

Figure 2.10: Illustration of a cooling tower, [27]

Figure 2.10 schematizes a cooling tower's operation. Warm waters is introduced from the top in the spray nozzles, which turn the continuous liquid into tiny droplets in order to increase the contact surface area between the air and water flows. As the hot water droplets fall by gravity through the sinuous path of the fill material, they become into intimate contact with the ascending cold air, which absorbs a percentage of the water entering the cooling tower that evaporates partially by cooling down. At the same time, sensible heat may also be transferred from the hot water to the cold air by convection, which reinforces the water temperature reduction. As this happens, the warm water releases heat that is absorbed by the ascending air, leaving the cooling tower at a warm and moist state. The remaining water flow is collected in the collection basin, and returns to the part of the facility that needs cooling.

The total cooling power provided by a cooling tower can be subdivided into its sensible and latent heat transfer components. The sensible heat transfer translates into an increase of the outlet air temperature ($\dot{m}_4$ in Figure 2.11), while the latent component is responsible for the increase in humidity of that same airflow (as well as for the water temperature decrease), due to the moisture evaporated from the water. The advantage of using a cooling tower is that the latent heat transfer is much more efficient than the sensible component.

The mass balances for the dry component of the air and water flows are:

$$Air: \ \dot{m}_{a3} = \dot{m}_{a4} \tag{2.6}$$

$$Water: \ \dot{m}_1 + \dot{m}_5 + \dot{m}_{v3} = \dot{m}_2 + \dot{m}_{v4} \tag{2.7}$$

In equation 2.7, $\dot{m}_{v3}$ and $\dot{m}_{v4}$ are the water vapour flow rates within the air respectively at the inlet and outlet.

The energy conservation equation applied to a cooling tower in steady state, in the absence of mechanical work and neglecting variations of kinetic and potential energies, on the other hand, can be expressed by equation 2.8.

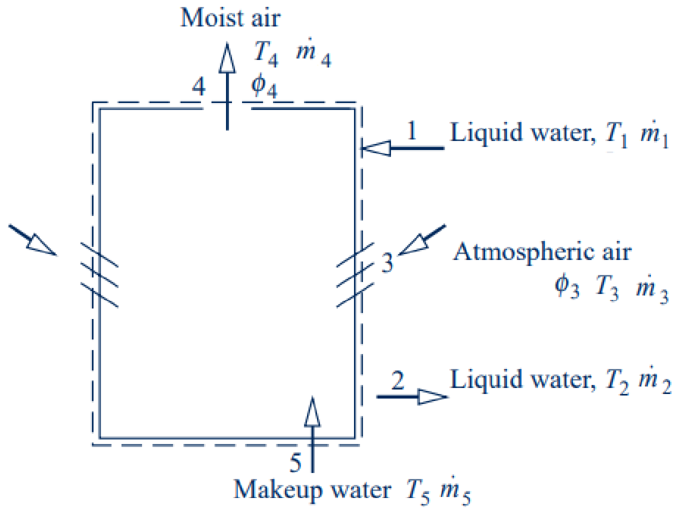

Figure 2.11: Illustration of a cooling tower´s mass and energy balance variables, adapted from [23]

$$0 = \dot{m}_1 h_{w1} + (\dot{m}_a h_{a3} + \dot{m}_{v3} h_{v3}) + \dot{m}_5 h_{w5} - \dot{m}_2 h_{w2} - (\dot{m}_a h_{a4} + \dot{m}_{v4} + h_{v4}) \tag{2.8}$$

Most authors divide cooling towers according to the driving force of the air flow and the relative flow directions of water and air. Regarding the first category, one can make the distinction between:

- **Natural Draft** - Here, the air flows up purely by it's buoyancy. As it warms up, it rises and exits the top of the cooling tower. These are very common when the needed capacity is very high, and have a characteristic parabolic shape.

- **Mechanical Draft** - in these, the air is forced to ascend through the cooling tower using axial fans typically positioned at the top of this kind of device.

Finally, cooling towers can also be characterised as cross-flow or counter-flow depending on the relative flow direction of the air and water flows (figure 2.9).

To characterise a cooling tower's performance, the following variables are used: approach and range. The first is the difference between the cooled water and the wet bulb temperature. In other words, the smaller the approach, the more efficient the cooling tower is. Conversely, wet bulb temperature is defined as the temperature measured by a thermometer whose bulb is wetted. Since the water around it tends to evaporate, it introduces a cooling effect and as such, the wet bulb temperature is always smaller than the ambient temperature.

Range, on the other hand, is defined as the difference between inlet and outlet water temperatures and is mostly dependent of the entering water temperature, since the air and water flows should remain relatively close to design values in order to maintain the highest level of efficiency.

2.6 Economisers

Free cooling in the HVAC context consists in harnessing the natural cooling potential of the outside air (or even water) in order to cool down a space or equipment, therefore avoiding the necessity of power-intensive refrigeration cycles.

Khajala and Halgamude [10] divides economisers in the context of data centers into the following categories:

- **Air side economizers**
  - Direct air side economizers
  - Indirect air side economizers
  - Hybrid air-side economizers
- **Water side economizers**
  - Direct water-cooled system
  - Air-cooled system
  - Cooling tower system

Of these, the most common (judging by the amount of the works published in the literature on the subject) are the direct air economiser and water side with cooling tower economisers (figure 2.12), and as such, the main focus will fall on them.

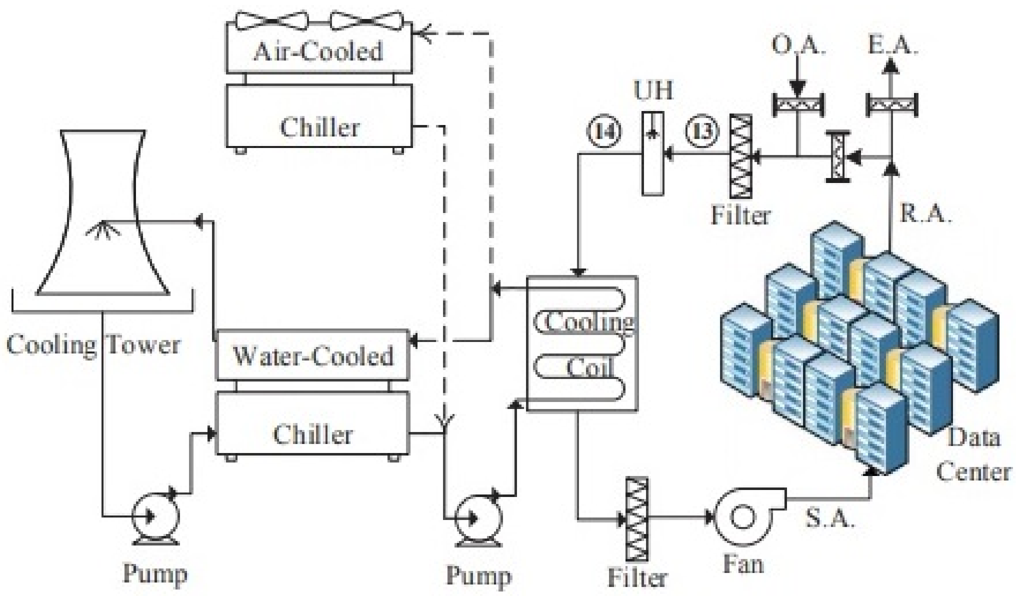

Figure 2.12: Cooling tower system and direct air-side economizer [10]

2.6.1 Direct air-side economisers

A direct air-side economizer attempts to reduce energy consumption in data centers by inserting outside ambient temperature air in the data center, when the outside air enthalpy is lower than the data center room enthalpy. This can lead to the necessity to humidify or purify the entering air in dry or polluted areas, respectively. Notwithstanding, the energy savings should make up for the increase in filtering and humidification costs.

When the ambient temperature is deemed to be low enough, outside air is circulated into the data center, circumventing the need to use the Chiller. Figure 2.13 shows how avoiding the use of compressor-based cooling, one can significantly decrease a data center's PUE.

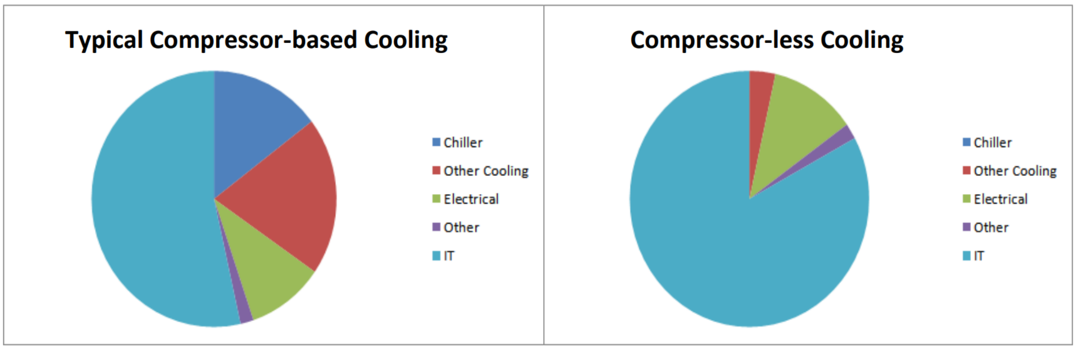

Figure 2.13: Comparison between typical compressor based and free cooling-based data centres' energy consumption breakdown, taken from [28]

Deymi-Dashtebayaz and Namanlo [29] explored the feasability of the implementation of an air-side economizer for a data center in several locations in Iran. The authors concluded that a reduction in PUE as high as 12% could be achieved, and payback times in 7 out of the 10 studied locations were under 1.5 years, suggesting a high potential in energy savings.

Lee and Chen [30] conducted a study on the viability of this type of free cooling in 17 locations worldwide, where climates of nearly all types were considered. The two closest locations to Lisbon were Brest, France and Albacete, Spain. These two locations observed potential yearly energy savings of respectively 32 and 9%. Albacete, due to its drier climate, presented higher humidification costs, hence the lower energy savings.

Figure 2.14 illustrates how a direct air-side could be implemented in the context of a data center. Outside air is circulated into the data center room using a fan equipped with a speed controller, in order to adjust the flow entering the building based on a pre-defined setpoint. According to Lee and Chen [30], the most common control methods for this type of systems are fixed temperatures, differing indoor and outdoor temperatures, fixed enthalpy, and indoor and outdoor enthalpy differential controls. In this thesis, the temperature inside the data center was chosen as the variable to mantain, and the outside air temperature as the deciding factor on which to use or not the air economizer.

One potential problem that comes with this type of economizer is the introduction of dust and pollutants inside a room full of electronic equipment running on a permanent basis. To avoid this, filters can be installed in order to achieve a sufficient level of cleanliness inside the server room. Shebabi et al.

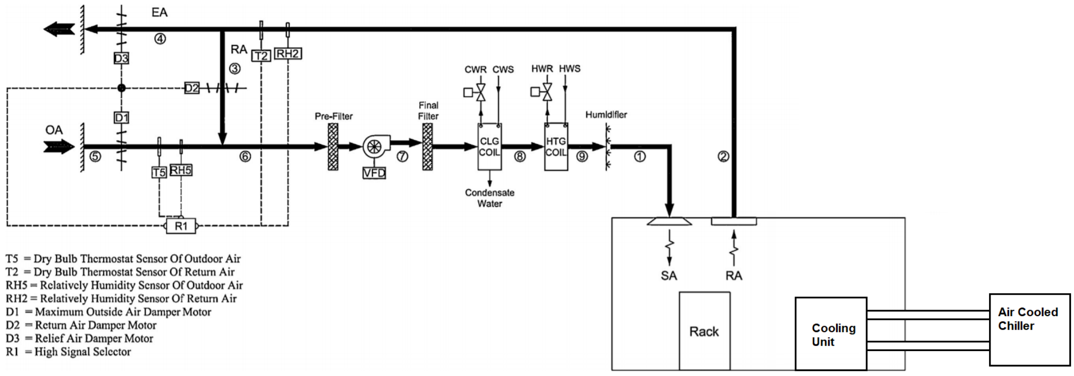

Figure 2.14: Data Center with air-side Economizer layout. Adapted from [30]

[31] found that a by implementing an adequate filtering system like the one in figure 2.14, the adequate operating conditions can be achieved. According to the authors, contamination and humidity levels were within those recommended by ASHRAE.

ASHRAE studied the impact of the implementation of an air-side economiser on the gaseous and particulate contamination levels in [32]. According to such study, MERV 11 or preferably, MERV 13 filters (figure 2.15) are the most adequate for filtering air entering a data center, and should suffice in keeping away most of the dust and particulates to a point where reliability is not affected.

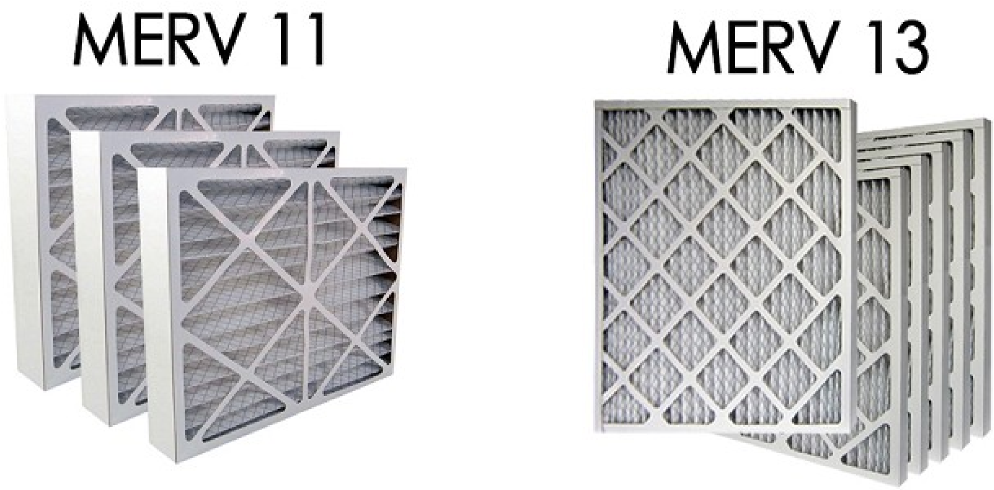

Figure 2.15: Example of MERV 11 and MERV 13 - type filters. From [33]

2.6.2 Cooling tower system economizer

As illustrated in figure 2.12, in a cooling tower system economiser, the water is cooled in the chiller, which is in turn cooled in the cooling tower. In the case that ambient conditions allow it, the data center water loop can be cooled exclusively or partially at the cooling tower, using a heat exchanger instead of the water cooled chiller to interface between the two water loops. This type of cooling layout is very common in larger data centres [10].

Thanks to the flexibility that this setup allows for, it has become the second most common type of economiser in the United States of America, after direct air economisers [34]. By designing a system that uses pure free cooling, pure compressor-based cooling and a mix of the two, it is possible to harness the natural cooling power for a large part of the year.

Garday [35] found that after implementing this type of economiser, the power consumed on cooling was reduced by 85% by avoiding the chiller use from late fall to early summer, when temperatures are lower and the free cooling is enough to cool the considered data center.

2.7 ISTs data center cooling system overview

The IST data center is located in the Central pavillion of the Alameda campus, in Lisbon. This installation is responsible for keeping essential services for the IST's community running.

The cooling system layout is illustrated in figure 2.16. The server floor, located in Floor -1 has a floor area of $119.6m^2$ and is populated with the server racks and eight cooling units, as seen in Figure 2.17 . It is possible to see the server's air outlets and inlets are all pointing in the same direction, in order to maximize the temperature differential across the cooling unit's coils.

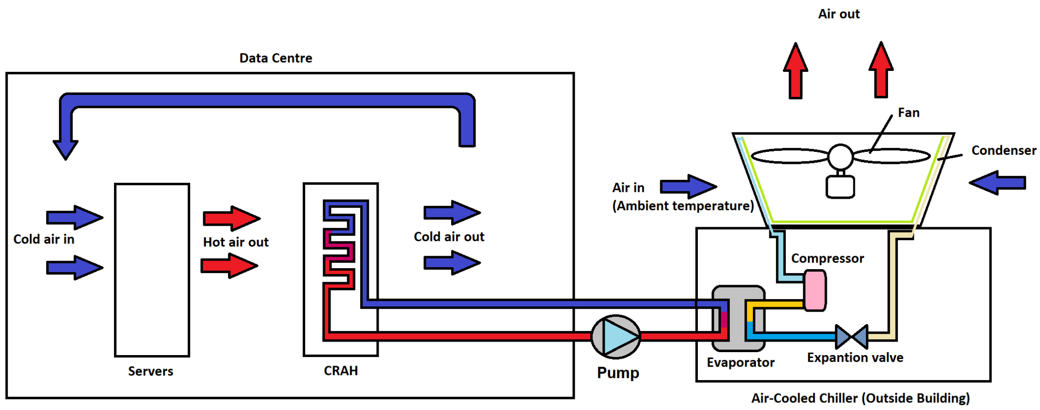

Figure 2.16: Diagram of IST's data Center´s cooling system

The servers are interspersed by the cooling units (as seen highlighted in blue in figure 2.18 and 2.17). There are eight of these INROW ®RC cooling units for the whole data center. These take the cold air from the hot aisle and cool it down and drive it into the zones requiring cooling, as illustrated in figure 2.18. Each of these units has a design cooling capacity of 17kW for a air ΔT of 11ºC across the cooling unit. As the water absorbs the heat from the hot air, its' temperature rises. The water then heads to the cooling distribution unit (CDU) and from there to one of the two chillers (one for redundancy), located in the roof of the Central pavilion. Each chiller has 2 centrifugal pumps assembled in parallel, which circulate the water through the water loop. The water is then cooled in the chiller's evaporator from a temperature of approximately 15 to 10 ºC, and redirected back to the server room downstairs.

The chiller cools the water down using a refrigeration cycle, explained in detail in section 2.3. This is done in the evaporator, where the water transfers heat to the low pressure/temperature side of the refrigerant cycle.

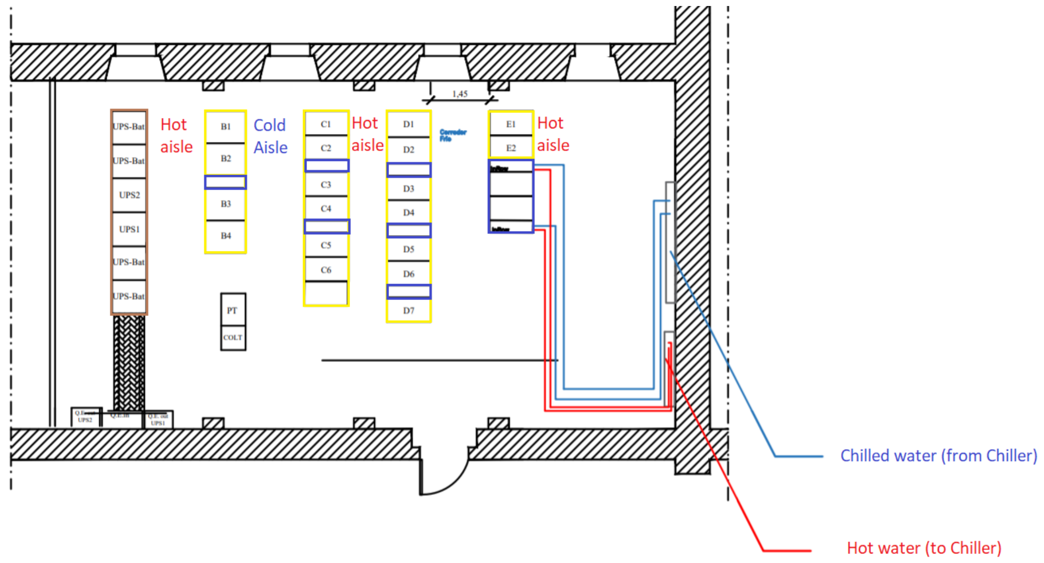

Figure 2.17: Data floor architectural plan

Finally, in the condenser, the refrigerant releases the heat to the outside environment (ambient air).

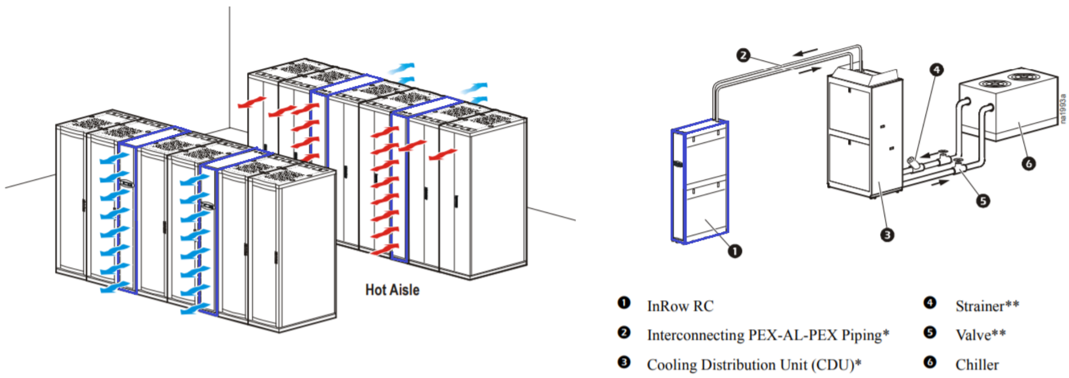

Figure 2.18: Illustration of a data center utilizing the INRow®RC cooling units. Adapted from [36].

Chapter 3

Modelling and Software

3.1 Software

In this section, the software used in the context of this thesis will be presented and the steps taken to develop the model will be explained.

3.1.1 MATLAB

MATLAB is a programming language and computing environment developed by *Mathworks*. Due to the widespread use in academic and even practical engineering situations, MATLAB became the most obvious choice for developing this model.

3.1.2 SIMULINK

Simulink is a visual programming environment inside MATLAB. It is a graphical programming language, which means that rather than the user need to develop by writing lines of a code, this is done by grouping together different blocks, each performing a different function. These, when associated in a certain way can describe the behaviour of complex dynamical systems.

Using the appropriate toolboxes made available by *Mathworks*, the user can develop models in the most diverse engineering fields. Using the different resources available to the user, one can simulate anything from an electrical circuit to a rocket or self-driving car.

3.1.3 Simscape

Simscape is a Toolbox within Simulink that focuses mainly on components to model physical systems. This means that the user can use components defining several physical phenomena without having to define them using the corresponding equations (Figure 3.2). By passing on the correct constant values for a component (Figure 3.2), one can represent it's real behaviour satisfactorily from an engineering point of view.

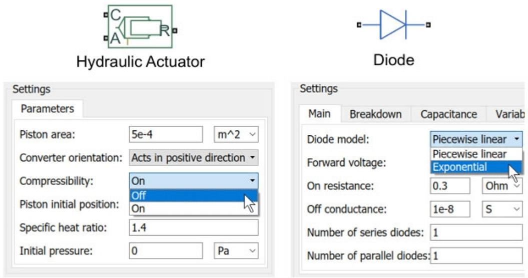

Figure 3.1: Example of Simscape blocks and their tunable parameters

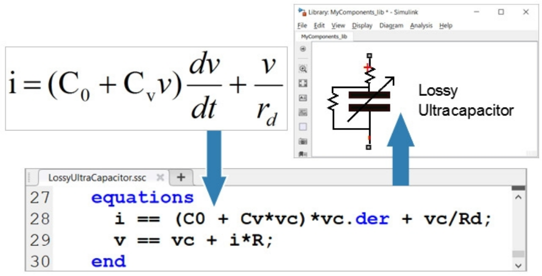

Figure 3.2: Example of how the modeling works ”behind the scenes”

By grouping different components together, it’s possible to represent a complex system, as is the case of the model displayed in figure 3.3, where a gas turbine cycle is modelled. The blocks and connections between them are color-coded, representing different physical domains such as, mechanical, thermal, electrical and much more.

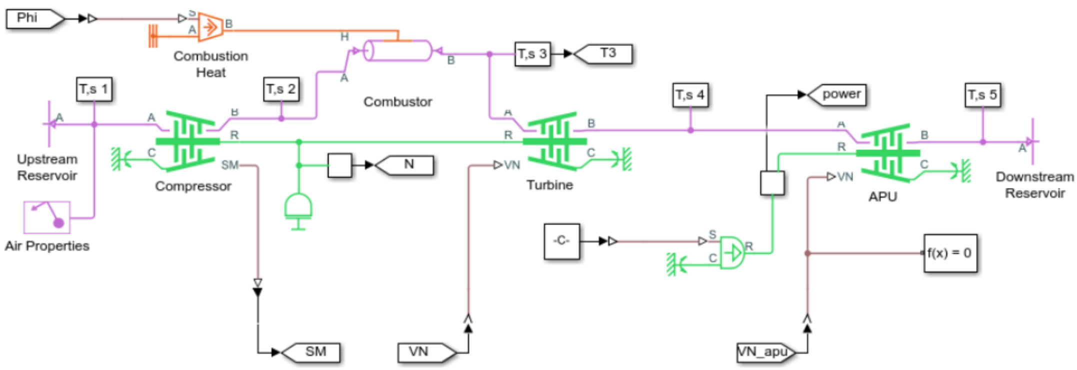

Figure 3.3: Gas Turbine model using Simscape [37]

3.2 Numerical model

3.2.1 Server Floor

From a thermal network standpoint, the server floor is comprised of the room itself (the physical place), as well as the IT and power distribution equipment.

Heat is generated by the operation of IT equipment, as well as power delivery systems, lighting and human presence. To estimate the total heating requirements, the data provided in table 2.2 was used. There, one can see that the majority of the cooling requirements are due to the IT equipment, while power distribution systems, lighting and human presence accounts for a non-negligible increase of cooling requirements.

This system interacts with the exterior, through heat losses by wall conduction and convection, as well as air leakage. Table 3.1 illustrates the characteristics of both wall, roof and floor materials and what type of room is on the other side.

	Material	Thickness (m)	Thermal Conductivity $(W/m.K)$	Convection Coefficient (W/m^2K)	Adjacent Room
North Wall	Brick	0.3	1	10	Warehouse
South Wall	Stone	-	-	-	Ground
East Wall	Stone	1	3	10	Street (semi buried)
West Wall	Stone	1	3	10	Corridor
Floor	-	-	-	-	Ground
Ceiling	Slab and Mortar Layer	0.3	1.7	1	Offices

Table 3.1: Thermal properties of the server room

The data for the thermal conductivity for the different materials were taken from [38]. For the convection coefficient, for the vertical walls (North, East and West), a typical value of $10W/m^2K$ was used.

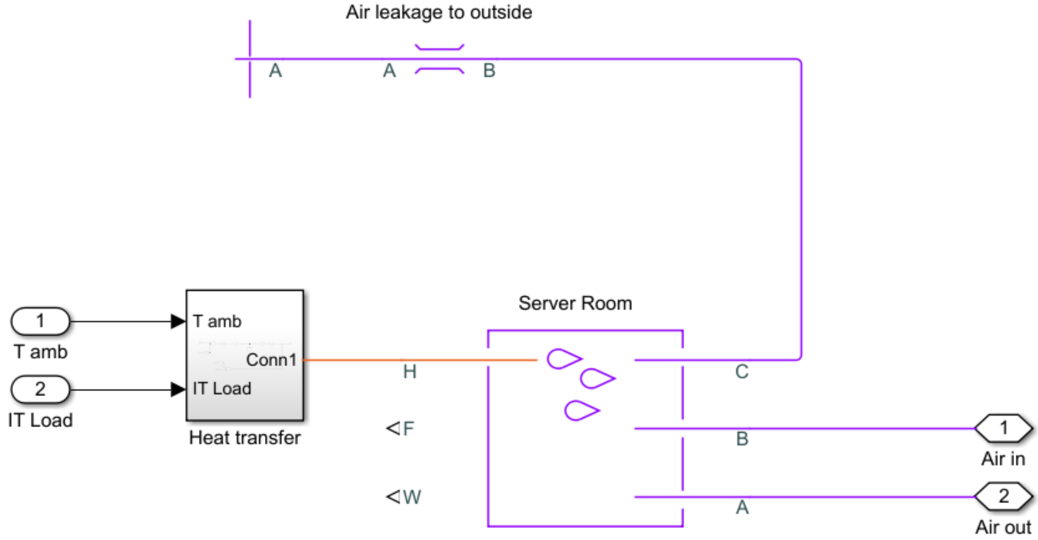

Figure 3.4: Room subsystem in Simulink

Figure 3.4 represents how the server room is modelled in Simulink. Starting from the left, the heat transfer is calculated in the "Heat transfer block", where the properties expressed in table 3.1 are used in conjunction with the heat transfer subsection of Simscape. The server room itself is represented by the "Server Room" block, with an air inlet and outlet, which, in turn, interface with the cooling unit block. There is also a small amount of air leakage to the outside, modelled by the "Air leakage to outside" block.

Finally, it is worth noting that by modelling the room as a "point", all effects resulting from temperature gradients are not simulated.

3.2.2 Cooling Unit/CRAH

The cooling units inside the Data Centre are ARC100, from APC (figure 2.18). There are eight of them distributed throughout the server floor. In each row, the cooling unit air inlets are facing the hot zone, and the outlet is facing the cold zone.

Each unit has cooling capacity of up to $34kW$ (for an entering water and air temperatures of $6°\,C$ and $37.8°\,C$ respectively). The flow of air is forced by 8 axial fans each consuming $115W$ at full load resulting in a total nominal volumetric flow rate of $1380l/s$. These are controlled by the difference in temperature between the inlet and outlet air flows. Water is circulated through the system using the chillers' built-in centrifugal pumps. Each cooling unit is equipped with a valve to regulate the water flow depending on the defined control strategy and temperature setpoints.

By observing the trendlines of the air flow and water flow through the cooling unit, one can see they are practically constant, even on a monthly basis (figure 3.5 and figure 3.6). The occasional spikes are due to data acquisition errors.

Due to the proprietary nature of the control algorithms for the valves and fans, the water flow was assumed to be constant, while the air flow controlled by the temperature in the room.

All the cooling units were also simplified as being a single air-water heat exchanger, using the ε-NTU heat transfer method. This comes as a consequence of simplifying the room as a single point. Since there are no temperature differentials across the room, all cooling units would be operating at the same fan speed and valve opening, so a distinction between individual cooling units would make no sense.

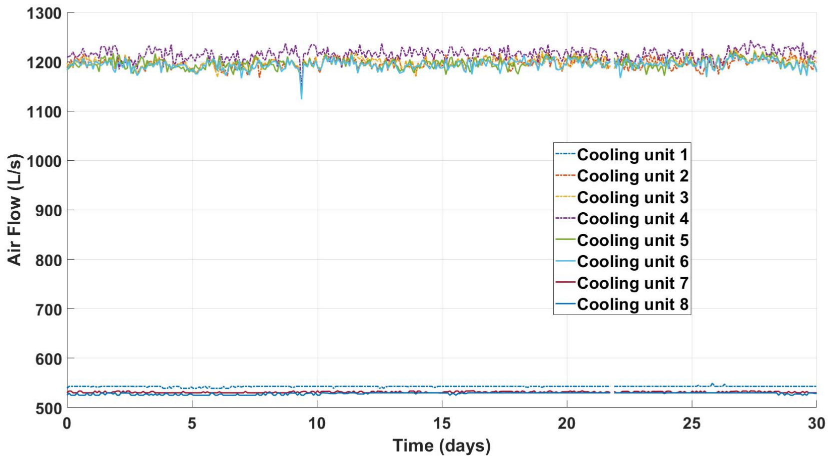

Figure 3.5: Airflow for all cooling units over a period of 30 days.

The fan power consumption is calculated by dividing the total airflow by the number of fans for all the cooling units (64) and then using the power law for fans ($P \propto Q^3$) and the nominal power consumption at full load provided by the manufacturer to calculate the total power consumption.

3.2.3 Air-coooled chiller

Warm water coming from the cooling units in the server floor is cooled down in the two chillers at the top of IST's Central pavillion. These two are used alternatively - one is kept for redundancy. The chiller's model is a Tetris 10.2 from Blue Box, which has a nominal cooling capacity of $110\ kW$, and energy efficiency ratio (EER) of 2.86 for an ambient air temperature of $35°\,C$ and inlet/outlet water temperatures of $12°\,C$ and $7°\,C$, respectively. The relevant components and it's characteristics are listed in table 3.2.

The working principles of a refrigeration were discussed previously, in section 2.3. The chiller sub-

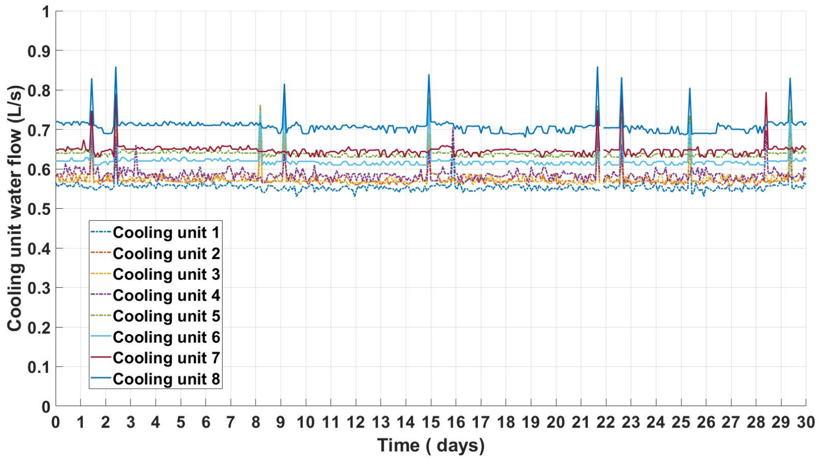

Figure 3.6: Water Flow for all cooling units over a period of 30 days.

model is therefore subdivided in a condenser, evaporator, compressor, expansion valve and liquid deposit.

	Quantity	Type	Nominal Air/ Water Flow (L/s)	Power Consumption /Release (kW)	Volume (L)
Evaporator	1	Stainless steel brazed-welded plate heat exchanger	5.26	110	-
Condenser	1	Finned-coil heat exchanger with copper tubes and aluminium fins.	11667	145	-
Fan	2	Axial Fans	11667	3.3	-
Expansion Valve	1	-	-	-	-
Compressor	2	Hermetic orbiting scroll compressors, connected in parallel	-	35	-
Expansion Tank	1	-	-	-	300

Table 3.2: Chiller components and data, from [25]

To accurately represent each sub-component of the chiller, some data from the technical datasheet were used. However, some important details were not available, such as the refrigerant mass flows, heat transfer coefficients and contact areas and dimensions in the condenser and evaporator, to name a few. As a consequence, additional calculations and some simplifying assumptions had to be made. These will be detailed below.

Evaporator

The evaporator is a plate heat exchanger with exterior dimensions of approximately 0,62x0,262x0,3m.(see figure 3.7 for its location in the chiller). It has 8 parallel plates in order to maximize the heat transfer area. Each plate is made of stainless steel, in order to reduce conduction resistance. Insulation is applied around this device in order to reduce the amount of heat exchanged to the exterior to a minimum. As a consequence, for the purpose of this model, the evaporator was considered to be adiabatic and the conduction resistance was ignored.

To correctly model the behaviour of the evaporator, the available ε-NTU Heat Transfer block made

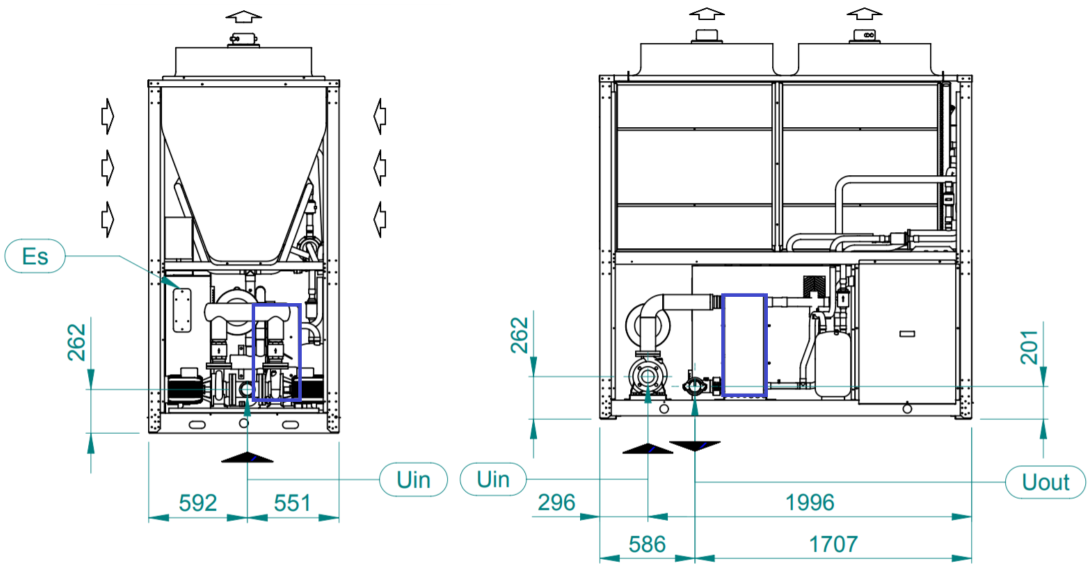

Figure 3.7: Chiller's dimensions in millimetres (evaporator highlighted in blue), taken from [25]

available by Simscape was used to calculate the amount of heat transfer. As the name suggests, the block uses the model laid out in section 2.4 and several variables are required to be determined, such as heat transfer coefficients for both sides, heat transfer contact area, fouling factor, among others.

	Heat Surface Area (m^2)	Fouling Factor ($m^2 * K/W$)	Heat Transfer Coefficient ($W/(m^2K)$)
Refrigerant Fluid	14	0.0002	6000
Water	14	0.0002	4873

Table 3.3: Heat transfer areas, fouling factors and coefficients for the chiller's evaporator

Since the manufacturer does not provide a value for the contact area, an estimation is used instead using the outer dimensions and number of plates. The fouling factors were taken from Incropera and Lavine [26] for the refrigerant and water sides of heat exchangers.

Due to the typical secrecy on the manufacturer's side regarding technical specifications, as well as a relative scarcity of investigation work on this type of devices for the purpose of evaporators, it becomes a hard task to determine the heat transfer coefficient on the refrigerant side. Bottgor [40] developed a correlation for this type of heat exchangers, as expressed by equation 3.1.

$$h_{ref} = 0.0675(k_l/d_e)[Re_l^2 h_{fg}/L_p]^{0.4124}(p/p_{cr})^{0.12}(65/\beta) \tag{3.1}$$

where h_{ref} is the convection coefficient for the refrigerant side, k_l is the thermal conductivity of the refrigerant fluid in it's liquid state, Re_l is the Reynolds number for the flow, h_{fg} is the specific heat of evaporation for the considered refrigerant. p is the pressure at which the refrigerant flows in the evaporator and p_{cr} is the critical pressure of that same refrigerant.

Hsieh and Lin [41] conducted an experiment to determine the heat transfer and pressure drop characteristics of a plate heat exchanger with similar dimensions and the same refrigerant fluid. For similar operating conditions of $Re = 6000$, the refrigerant-side heat transfer coefficient was determined to be

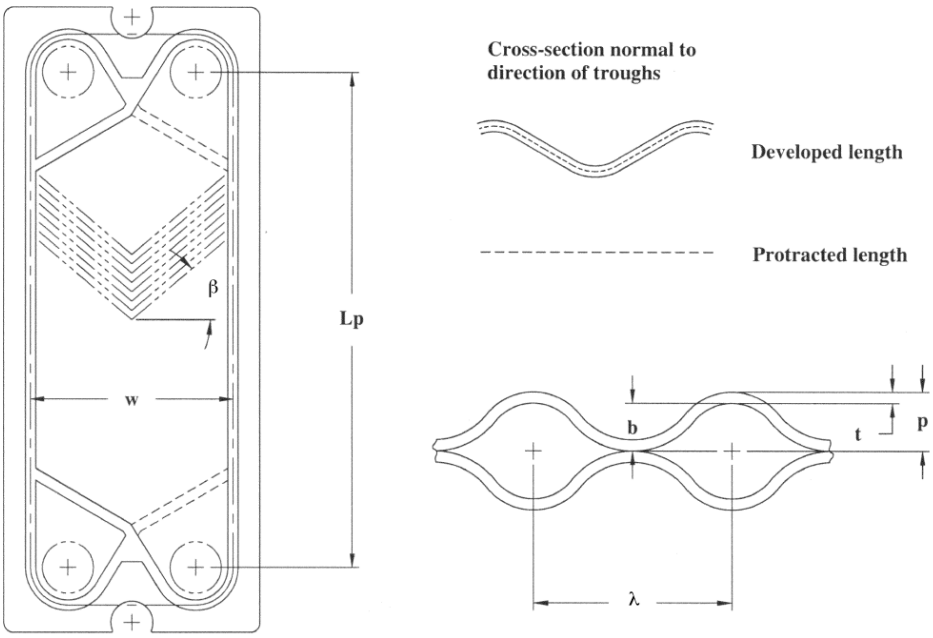

Figure 3.8: Plate Heat Exchanger dimensional parameters, from Ayub [39]

$h_{ref} = 6000 W/m^2K$.

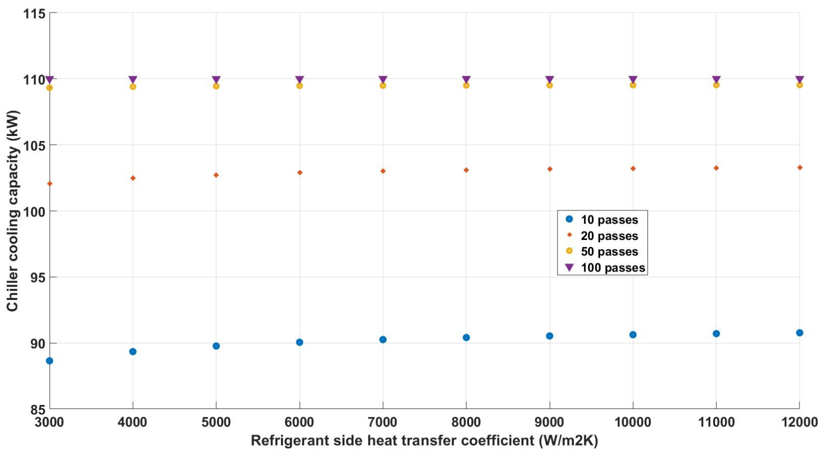

Figure 3.9: Impact of the refrigerant side heat transfer coefficient and number of passes on the chiller´s cooling capacity

Figure 3.9 illustrates how changing the refrigerant side heat transfer coefficient affects the whole chiller's cooling capacity. As one can see there doesn't seem to be a very big dependence on this parameter, so the value used was $h_{ref} = 6000 W/m^2K$.

Since Simscape does not have effectiveness data for plate heat exchangers, the default effectiveness data for shell and tube heat exchangers was used instead. Again looking at figure 3.9, for the nominal cooling capacity of 110 kW at an inlet water temperature of 12ºC and an ambient air temperature of 35ºC, the number of passes which seems to represent reality to a more accurate degree seems to be 100. As a consequence, this was the number of chosen passes.

Condenser

The chiller's condenser is one of the fin and tube kind. Hot refrigerant flows through the inside of tubes, which are cooled by air at atmospheric temperature forced by the two fans. Due to the lower heat transfer coefficients associated with air, this type of heat exchangers need larger heat transfer areas in order to operate satisfactorily.

	Heat Surface Area (m^2)	Fouling Factor (m^2K/W)	Heat Transfer Coefficient ($W/(m^2K)$)
Refrigerant Fluid	18.85	0.0002	173470
Air	220	0.0002	110

Table 3.4: Heat transfer areas, fouling factors and coefficients for the chiller's condenser.

Kays and London [42] developed a methodology to determine heat transfer coefficients for this kind of heat exchangers. Firstly, to obtain an estimation for the surface area, the volume can be calculated first and the area subsequently using the α factor, which is based on the dimensions of tube, tube spacing, and fins. The area for the refrigerant fluid was approximated using the diameter of each tube, their length and the number of tubes (Heat Transfer Area = $N\Pi DL$).

Using a α factor of $587m^2/m^3$,(figure 3.10), a heat surface area of $220m^2$ is obtained. This area concerns the airflow. The chosen fouling factor for both refrigerant and air flow was the same as in section 3.2.3 - see table 3.4.

Using the manufacturer nominal airflow of $13.813 \quad kg/s$, one obtains $Re = 2104$ and consulting figure 3.10, it is possible to get a coefficient $j_H = 0.008$ and an air-side heat transfer coefficient of $h_a = j_H \frac{\dot{m}}{\sigma A_{fr}} \frac{c_P}{Pr^{2/3}} = 110W/m^2K$

The heat transfer coefficient for the refrigerant can be estimated using equation 3.2 [43], for fully developed refrigerant flow undergoing dropwise condensation in a horizontal tube.

$$\begin{aligned} h_{dc} &= 51104 + 2044T_{sat} \quad if\ 22 < T_{sat} < 100\ ^{\circ}C \\ h_{dc} &= 255510 \quad if\ T_{sat} > 100\ ^{\circ}C \end{aligned} \tag{3.2}$$

Where h_{dc} is the dropwise heat transfer coefficient and T_{sat} is the saturation temperature for the pressure at which the refrigerant condenses.

Since the chiller has no measurements for the refrigerant cycle pressures and temperatures, and no information is given in the user's manual about this, an educated assumption must be made for this parameter. By measuring the coil's temperature, it can be seen it floated around $30 - 50^{\circ}C$, varying through the day depending, on the ambient temperature. The refrigerant temperature inside the tube must obviously be higher than this. On the outlet of the compressor, however, the refrigerant temperature was much higher (around $80^{\circ}C$), so an average condensation temperature of $60^{\circ}C$ was assumed, resulting in a heat transfer coefficient of $h_{dc} = 17367\ W/(m^2K)$

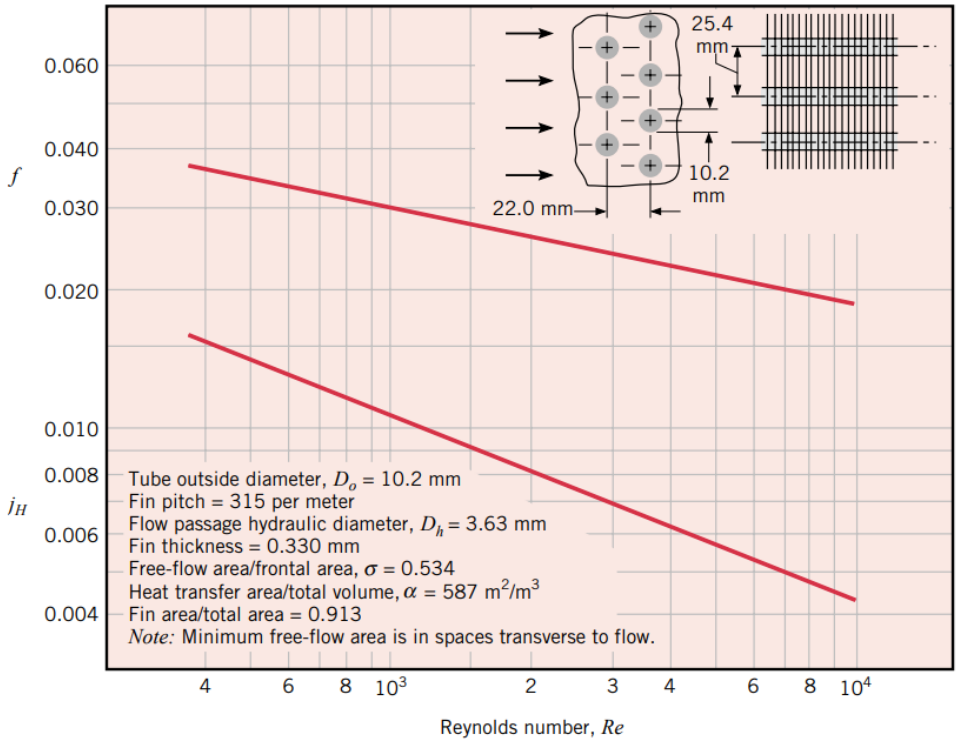

Figure 3.10: Heat transfer and pressure drop characteristics for a fin and tube heat exchanger,[26] [42]

Fans

The two fans are controlled by a simple relay. This means that they always work at full power. According to the manufacturers guide, the fans consume $3kW$ at full load, pushing $11000\ l/s$ through the condenser coils, at a temperature and humidity depending on ambient conditions.

Expansion Valve

The expansion valve is modelled as a variable area restriction, using the `Variable Local Restriction` block from Simscape. By varying the area, it controls the pressure drop, and consequently, the temperature downstream from the valve, in order to keep it at a set temperature (typically around $3C$ for R410a based systems).

Minimum Area (mm^2)	Maximum Area (mm^2)	Cross Sectional Area at Inlet/Outlet (mmˆ2)	Discharge Coefficient	Laminar Flow Pressure Ratio
60	150	1000	0.8	0.99

Table 3.5: Expansion valve characteristics

The discharge coefficient and laminar flow constants are respectively (according to *Mathworks*) the ”ratio of the actual to the theoretical mass flow rate through the restriction” and ”ratio of the outlet to the inlet port pressure at which the flow regime is assumed to switch from laminar to turbulent”. Both were left as the default values. The minimum and maximum and cross sectional areas, however, were chosen through an iterative process in order to represent the real life behaviour as best as possible.

Compressor

To increase the pressure of the refrigerant flow to the desired value at the inlet of the condenser, the Chiller has 2 screw compressors in parallel, on a on/off basis in order to keep the water flow at the outlet at (around) 10ºC. When possible, only one compressor is used, and the second one turns on automatically if only one compressor is insufficient to meet the cooling demand.

The compressors are of the hermetic orbiting scroll type, which work by "trapping" the refrigerant between a rotating and a stationary spiral, as represented in Figure 2.6.

To replicate the behaviour of this component, the `Mass Flow Rate Source` Simscape block is used, where a cross sectional area at inlet and outlet of $.001m^2$ was defined as the most adequate value, after trying different values. Power consumption is determined by product of the difference in specific enthalpy and mass flow of refrigerant (equation 3.3). The compression process, expressed by equation 3.3, is assumed isentropic, so the efficiency needs to be known to calculate the total power consumption.

$$P_{is} = \dot{m}(h_B - h_A) \tag{3.3}$$

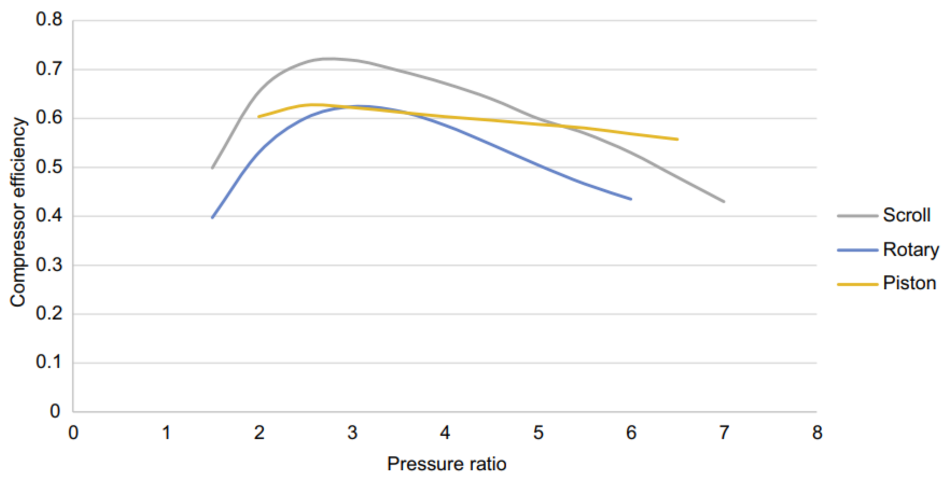

Figure 3.11: Typical efficiencies for different compressors under different pressure ratios, Fconver[44].

To approximate the real power consumption, a typical efficiency for this type of compressor can be approximated $\eta_{compressor}$, as represented in figure 3.11

$$P_{total} = \frac{\dot{m}(h_B - h_A)}{\eta_{compressor}} \tag{3.4}$$

Expansion Tank

In typical chillers, this component (typically a flash chamber) is located before the compressor and separates by gravity the liquid and vapour parts of the refrigerant flow, in order to avoid liquid refrigerant entering the compressor and possibly damaging it. It is modelled using the `Receiver Accumulator` Simscape block, whose parameters were chosen iteratively in order to represent as well as possible the real life chiller.

3.2.4 Piping

To connect the water coming from the chillers in the top of the central building to the data center in the bottom floor there are pipes. The main type of piping is 3" ($76.2mm$ inside diameter and $1.14mm$ of thickness) galvanized steel pipe. The total length (one way) is approximately 50 meters, while the height difference is 15 meters. According to [38], the typical roughness for pipes of this type of material is $4mm$, while the thermal conductivity is $20W/mk$.

3.2.5 Water Pumps

Each chiller is equipped with two Lowara centrifugal pumps, with a minimum of 200 and maximum of 520 l/s of volumetric flow rate and a minimum of 13 and maximum of 22.9 m of head. It has a nominal power consumption of 1.85kW. These specifications were taken from the label present at each one of the pumps installed at the IST Data Center Chiller, figure 3.13.

Since the flow is kept approximately constant, it was decided to consider the power consumption as a constant through time as well, as can be seen in Figure 3.12.

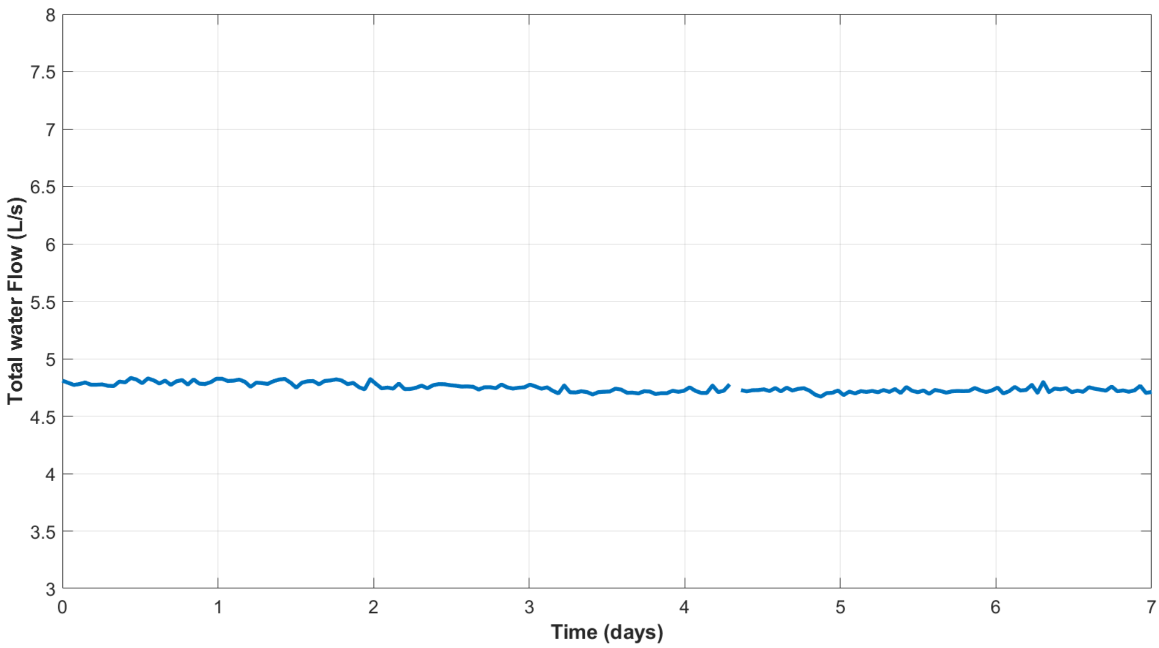

Figure 3.12: Water flow for the considered period (3 to 10 July 2020).

3.2.6 Comparison of model with real life

In order to proceed to the practical applications of this model, the most important variables were compared to actual experimentally measured values, in order to guarantee a high level of accuracy.

Figure 3.13: Pump technical specifications

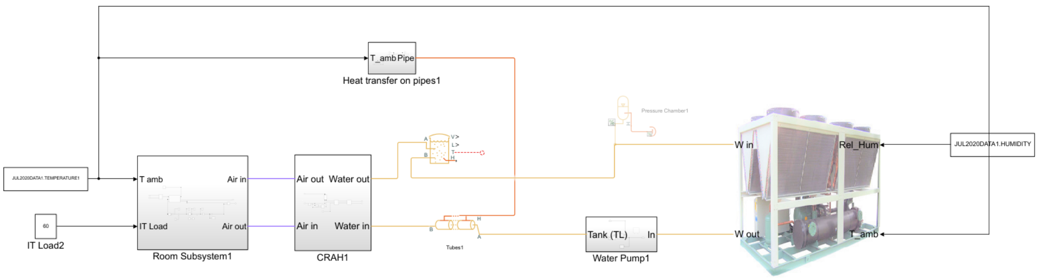

Figure 3.14: Overall view of the Simulink model

Chiller performance comparison

Firstly, the chiller cooling power and compressor consumption, as well as the COP (coefficient of performance) at full load (both compressors running) and an evaporator water outlet temperature of 10ºC was analysed. Data from the manufacturer's data sheet for nominal working conditions was compared to the model results for the same operational conditions *i.e.* same ambient temperature, full load on both cases and then comparing the cooling power and compressor power.

As can be seen in figure 3.15, the model closely follows the manufacturer's data for the range of

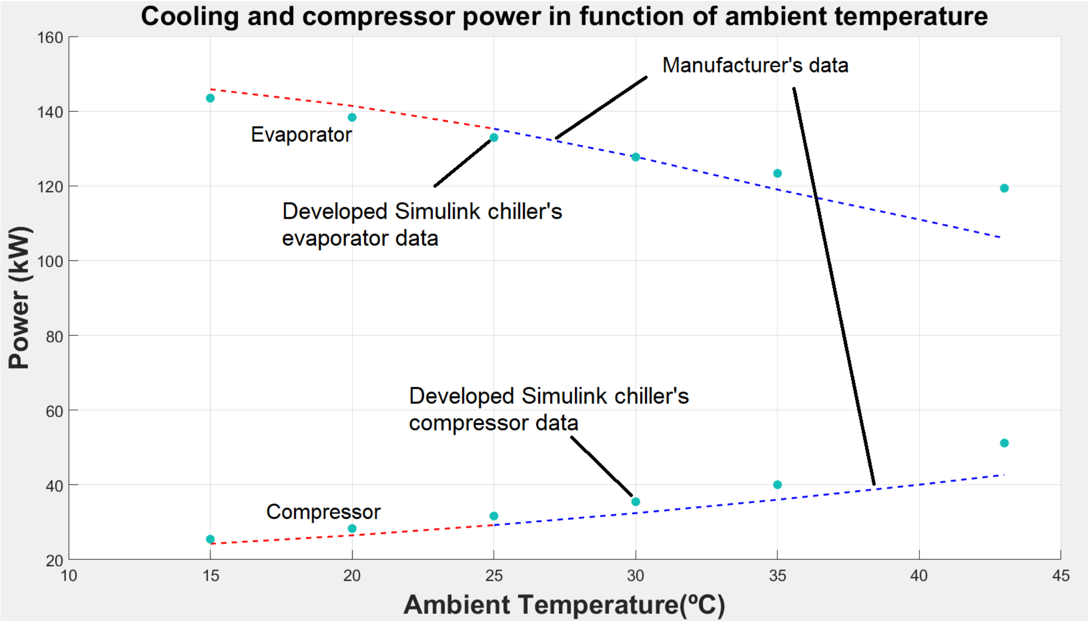

Figure 3.15: Compressor power consumption and evaporator cooling power in function of ambient temperature. Manufacturer's data from [25]

available temperatures. At higher temperatures (above $40° C$), the model results deviates slightly from the manufacturer's data, which can be due to a number of factors, such as the heat transfer coefficients and air flow being taken as constants. In any case, since these extreme conditions are unusual for long periods in Lisbon, this should not introduce a significant error to the final results.

July 2020 measurements

As a way to certify that the model represents the real behaviour of the system with enough accuracy, measurements of the refrigeration cycle temperatures, as well as total chiller power consumption were made. These measurements took place from the 3rd to the 10th of July 2020, using the equipment provided by the Campus Sustentável.

The electrical current was measured at the chiller electrical panel, using a triphasic power meter logger (Figure 3.18), which outputs a total power consumption. The measured data is represented in Figure 3.16. This measurement includes the chiller compressor, two axial fans and water pumps.

Looking at figures 3.16 and 3.20 it is possible to see that the power consumption increases with the outside air temperature and that during the hottest parts of the day, the second compressor has to operate momentarily in order to provide the desired cooling capacity, therefore resulting in a higher power consumption.

The refrigeration cycle temperatures were measured in four different thermodynamics states of the cycle, using four thermocouples (applied as shown in figure 3.19 connected to 2 dataloggers). The selected states (points) used for these measurements were at the evaporator inlet and outlet, as well as mid-way in the condenser coil and at the exit of the condenser. The point selected for the condenser was not at the inlet, because here the refrigerant is in a superheated state, and consequently it is not possible to correlate a temperature to a specific pressure. With this data, one can confirm that the

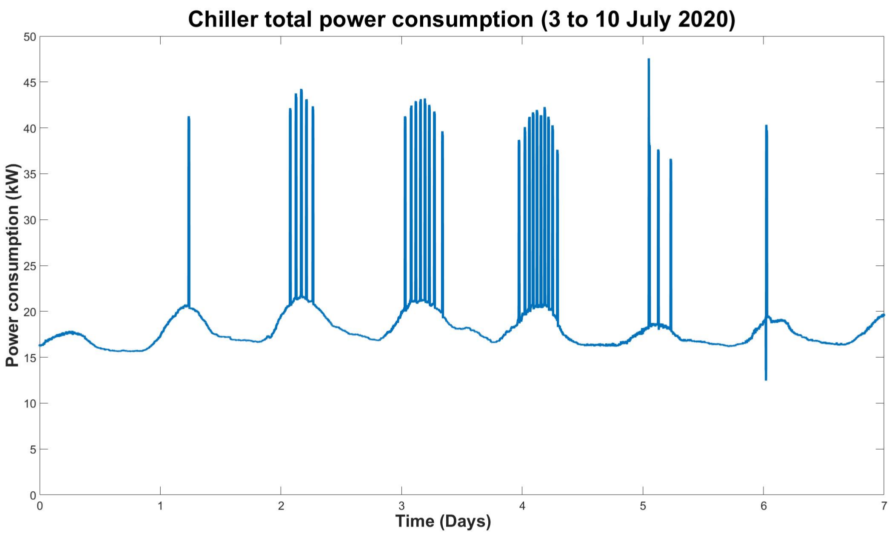

Figure 3.16: Chiller power consumption from the 3rd to the 10th of July 2020.

operating pressures observed in the actual system are similar to those observed in the model.

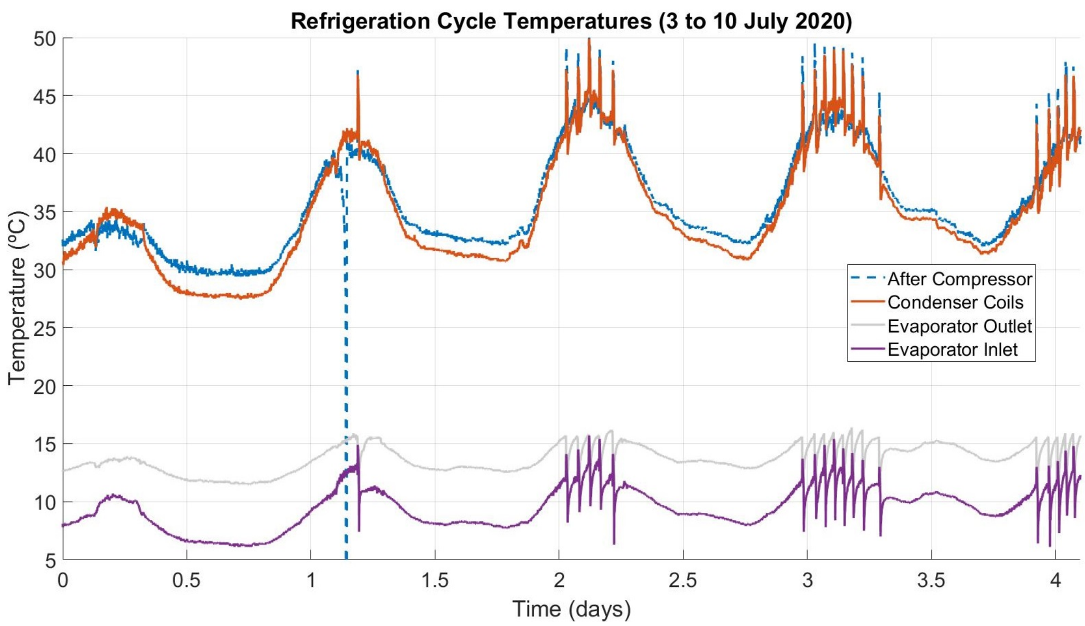

Figure 3.17: Refrigeration cycle temperatures from the 3rd to 10th of July 2020

The measured temperatures in the four thermodynamic states of the refrigerant cycle are represented in figure 3.17 Unfortunately, the data from the last days were corrupted and lost and, therefore, no data exists for that period.

The week's temperature and humidity data were taken from [45], an online database for weather records. The registered data for the period between the 3rd and the 10th of July 2020 is represented in figure 3.20.

One can see that this was a relatively warm week, where temperatures in the hottest time of the day often reached close to 37 to 38℃. It should also be said that, since the measurements were taken somewhere else in Lisbon, this could add to the global error, although probably not in a significant way.

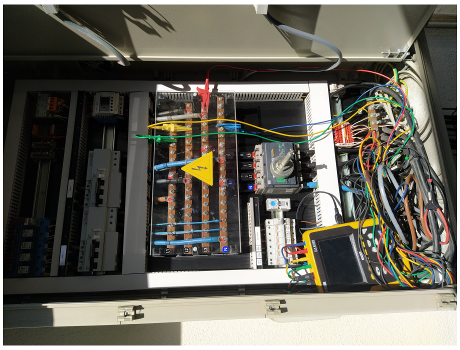

Figure 3.18: Electrical panel and amperemeter.

Figure 3.19: Thermocouple applied to condenser coil, and insulating tape on top of it.

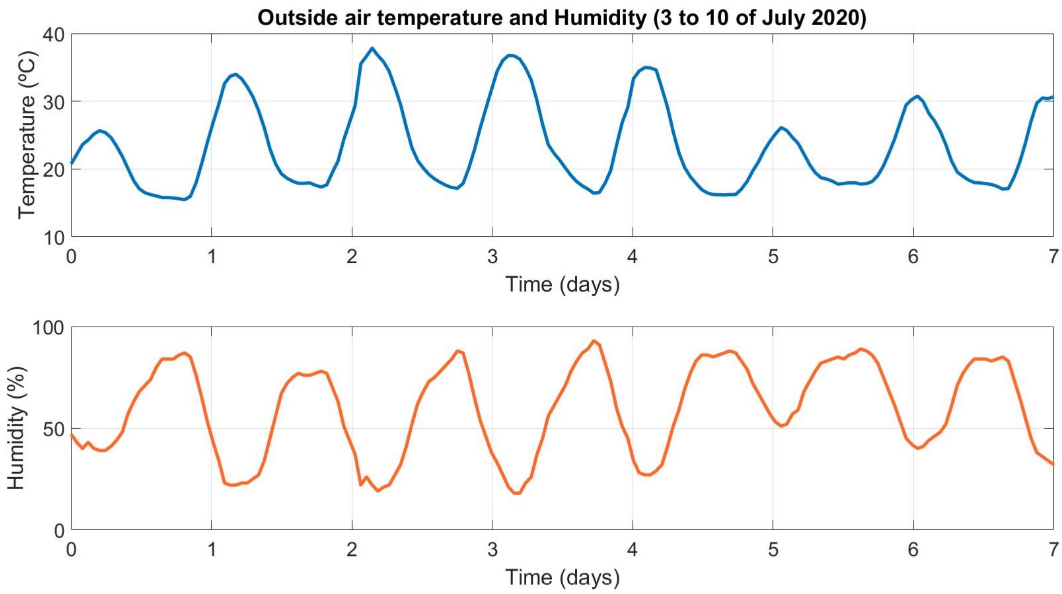

Figure 3.20: Temperature and humidity in Lisbon for the period of the 3rd to the 10th of July 2020

To complete the model simulation, the last needed input refers to that of the IT load on the servers. These data were taken from the DSI (Departamento de Serviços Informáticos of IST) data monitoring system and is represented in figure 3.21. One can see that the retrieved data shows three unusually low points. This is due to measurement error and, therefore, the value for the affected points was assumed to be the average between the two surrounding points. Figure 3.22 represents the resulting graph with the post-processed (corrected) data.

Comparison between model results and July 2020 measurements

After collecting all the necessary data, it becomes possible to run the model and compare its behaviour with real life measured data.

One can see in figure 3.23 both the measured chiller power consumption, and the chiller power consumption calculated by the model (total chiler power consumption is defined as the sum of the power consumed by the compressor, condenser fans and water pumps). The model results follow the real life

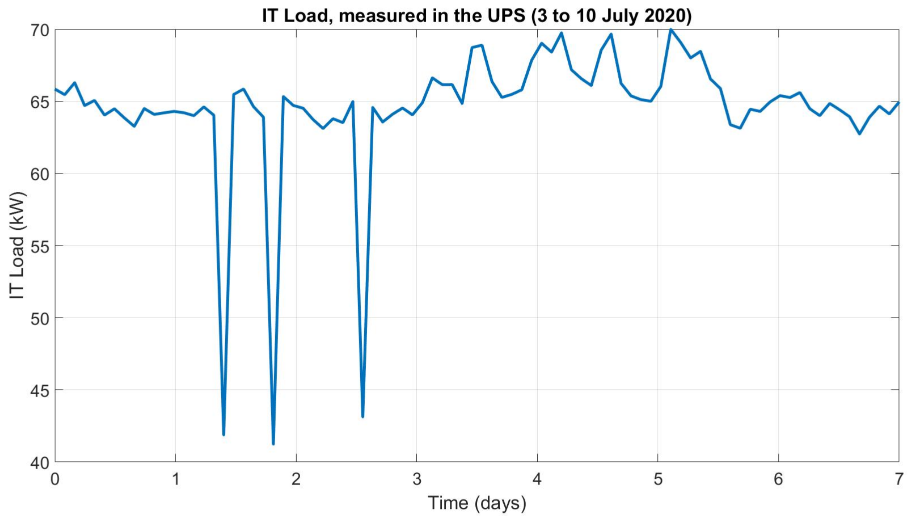

Figure 3.21: IT load, from the 3rd to the 10th of July 2020

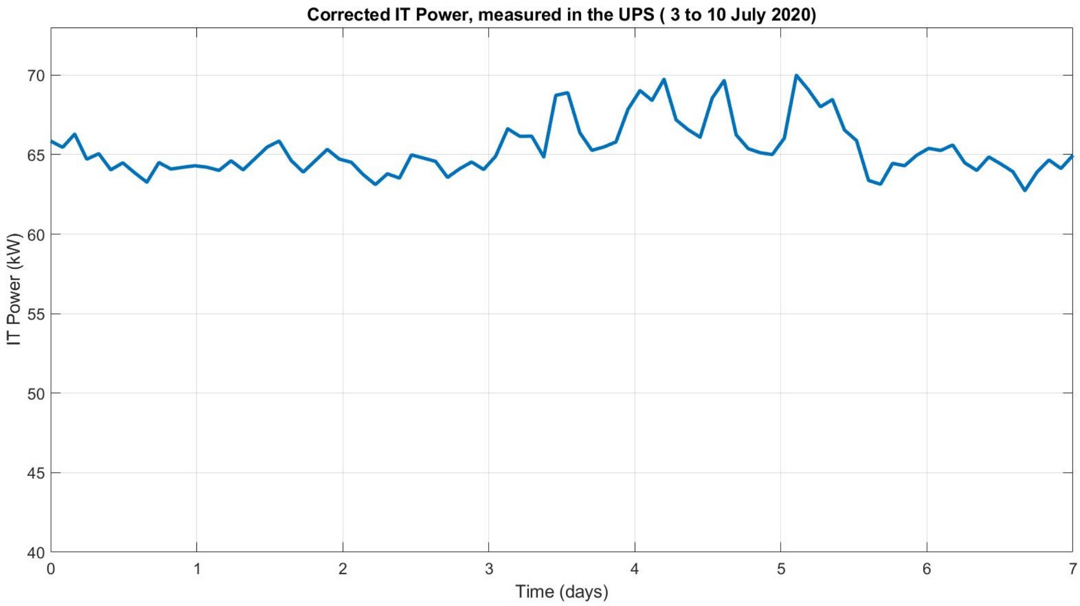

Figure 3.22: IT load, from the 3rd to the 10th of July 2020 (corrected)

value pretty closely. The observable small discrepancies are due to a number of factors including the measuring error in the input variables, simplifications in the model, as well as a inability of the model to be completely faithful to the control methods used for many of the components, since they tend to be proprietary and are not fully explained in detail by the equipment's manufacturers. Having said this, the week's measured chiller power consumption is $3085.64kWh$, while the one calculated using the model results is $3228.21kWh$, which results in a error of about 4.6% for the component that represents the majority of the power consumption in the whole data center system.

It is also possible to compare the measured water temperatures at the inlet and outlet of the data center with the model's results. The measured water temperature was not directly measured - instead each cooling unit´s inlet and outlet water temperature sensor measurements was averaged through in order to obtain unique values for water inlet and outlet temperatures. The comparison is done in figure

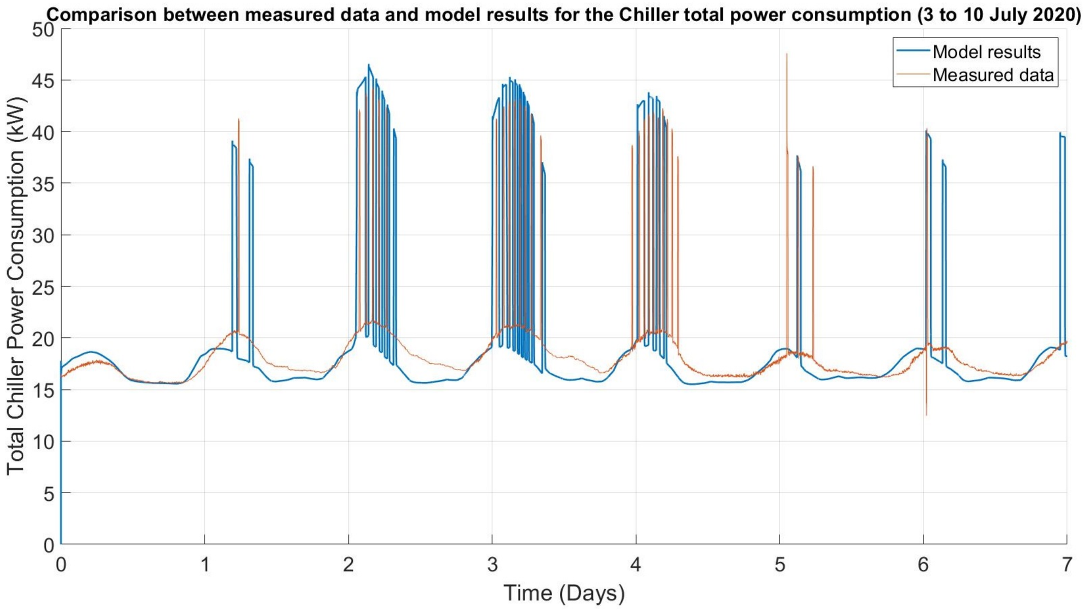

Figure 3.23: Comparison between the measured and simulated chiller power consumption values

3.24.

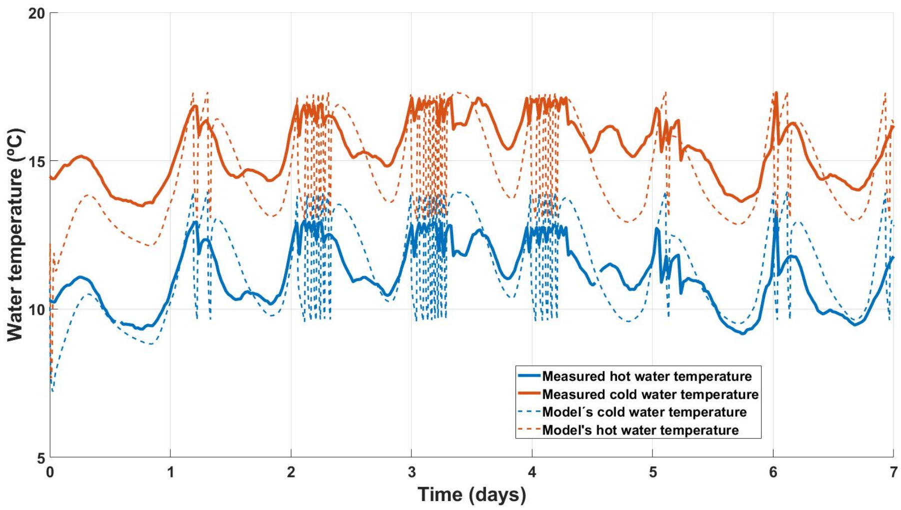

Figure 3.24: Comparison between the measured and simulated chiller inlet and outlet water temperatures

One can see that, while there are some noticeable differences in the peak temperatures, it still corresponds to the real life behaviour in average while the trend is captured. This can be due to the fact that the hot and cold measured water temperatures are not the result of the averaging of all the cooling unit´s readings.

Finally, the model refrigerant temperatures were compared to the measured data using the thermocouples. Figure 3.25 illustrates this comparison between the two data series. At a first glance one immediately notices that the model refrigerant temperature values are slightly lower than the ones obtained by observed in the measurements. This is because the thermocouples were placed on the outside of

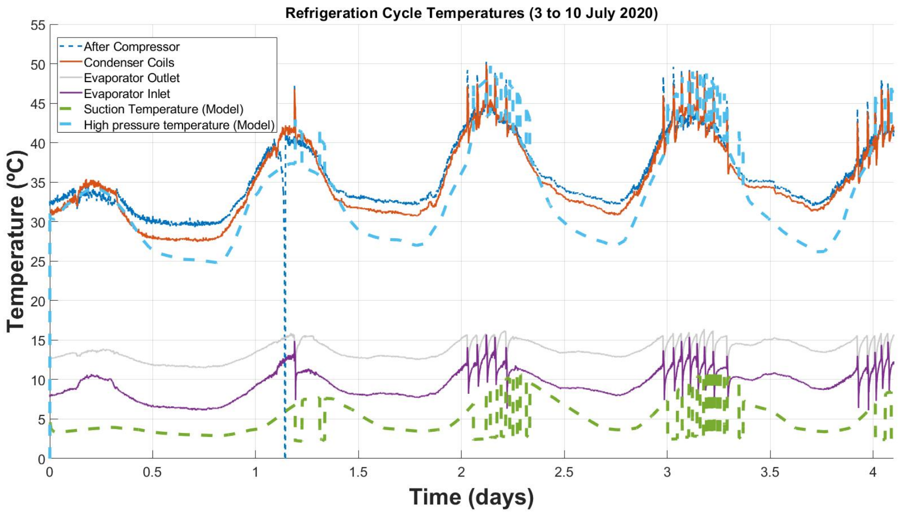

Figure 3.25: Comparison between the measured and simulated chiller refrigerant temperatures at different thermodynamic states

the pipes, whereas the model provides the refrigerant temperature itself. Since the pipe thickness is not known exactly, the comparison was left as is, without any correction for the thermal resistance introduced by the pipe heat conduction.

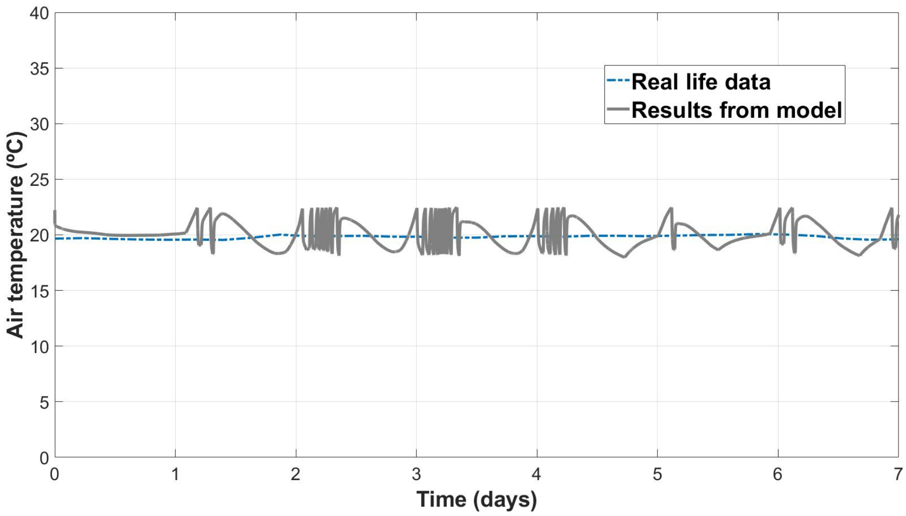

Figure 3.26: Comparison between the measured and the model results for the server room air temperature.

Figure 3.26 represents the comparison between the model air temperature and the measured one inside the data center. One can see that the model results seem to deviate slightly from the real life data because the cooling unit control method was not possible to be replicated due to its proprietary nature.

At this point, the developed model is thought to replicate the real life behaviour of the cooling system to a satisfactory degree taking into account the available data. Some of the variables´ behaviour, which are near constant, such as the water flow were easier to recreate. The fans power consumption was also relatively easy to recreate using the affinity laws.

On the other hand, The chiller's transient behaviour was more challenging to recreate since much of it's data was unavailable or hard to measure and educated guesses had to be taken regarding many of its components. Nonetheless, the chiller model (which represents the cooling systems most significant share of power consumption) can be said to follow real life power consumption accurately, as seen if figure 3.24.

Some of the simplifications, such as ignoring the temperature gradients across the server room can also induce a certain amount of error to the final results.

All in all the model can be said to be able to predict real life's behaviour with enough confidence for one to be able to use in in the next chapters as a tool to predict potential energy saving measures' impact.

Chapter 4

Results presentation and discussion

In this chapter, different potentially-saving energy measures are explored using the developed model. In each case, slight modifications to the system are carried out and impacts analysed.

According to Tschudi [28], in order to reduce existing data centers compressor based cooling, the most promising measures to reduce energy consumption for existent data centers were observed to be free air/water cooling systems. Using this type of technology, one can harness the cooling potential of lower ambient temperatures, avoiding the necessity for the use of a power-hungry compressor, which is an intensive energy consumer.

In the following sections, some popular energy saving measures regularly presented in the literature, as well as some of the Campus Sustentável suggestions and some thought of by the Author of this Thesis based on his experience in this field will be explored in order to figure out their energy saving potential.

4.1 Air side Economizer

The existing literature for the implementation and potential energy savings for air-side economisers is laid out in section 2.6.1. According to the existing literature, the potential energy savings is considerable for zones in similar climate characteristics to that of Lisbon. In addition to this, the average atmospheric air conditions in Lisbon throughout the year are very similar to the ones considered acceptable by ASHRAE for data centers, as schematized in figure 4.1. The average yearly weather conditions in Lisbon indicate that costs related with (de-)humidification would be minimal throughout the year and, therefore, this is a good indicator for the suitability of this potential solution.

As illustrated in figure 2.14, this solution would consist of installing an air inlet and outlet dedicated to the data center. This would introduce the so-called free cooling to the data center, which would turn it possible to avoid the necessity to use the chiller for long periods throughout the year, when the atmospheric conditions are favourable, potentially resulting in significant energy savings.

For the purposes of this thesis, the week of 3 to 10 of July 2020 was used as a reference, since recent data exists for that time period, making any comparison more trustworthy. The chosen control

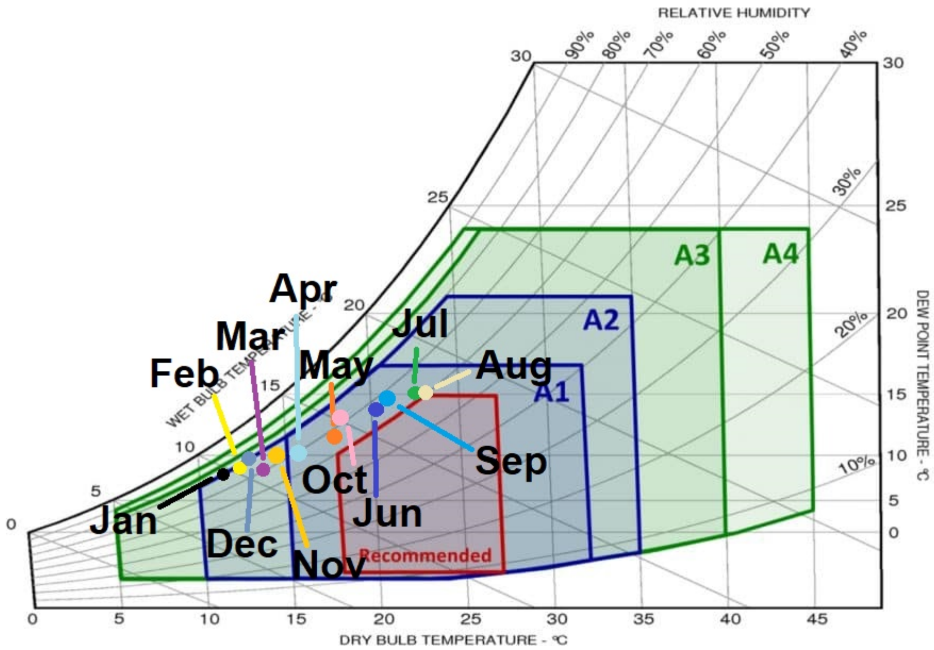

Figure 4.1: Comparison between Lisbon´s monthly average temperature and humidity with ASHRAE's recommended environment conditions inside data centers. Graph from [9] and climate data from [46]. The different areas' meaning is explained in table 2.1.

method for this simulation was temperature based, where the outside temperature is compared to the temperature setpoint inside the data center room.

Figure 4.2 shows the relevant system temperatures, as well as the volumetric mass flow introduced into the data center. The figure results evidence that, as soon as the outside temperature drops below 19.5ºC (aribitrarily chosen setpoint), the air economiser kicks in and the chiller is turned off, as the cooling power provided by the cold air circulating into the room is enough to keep it at a an adequate temperature. As a preliminary design choice, the maximum airflow was limited to $10\,m^3/s$.

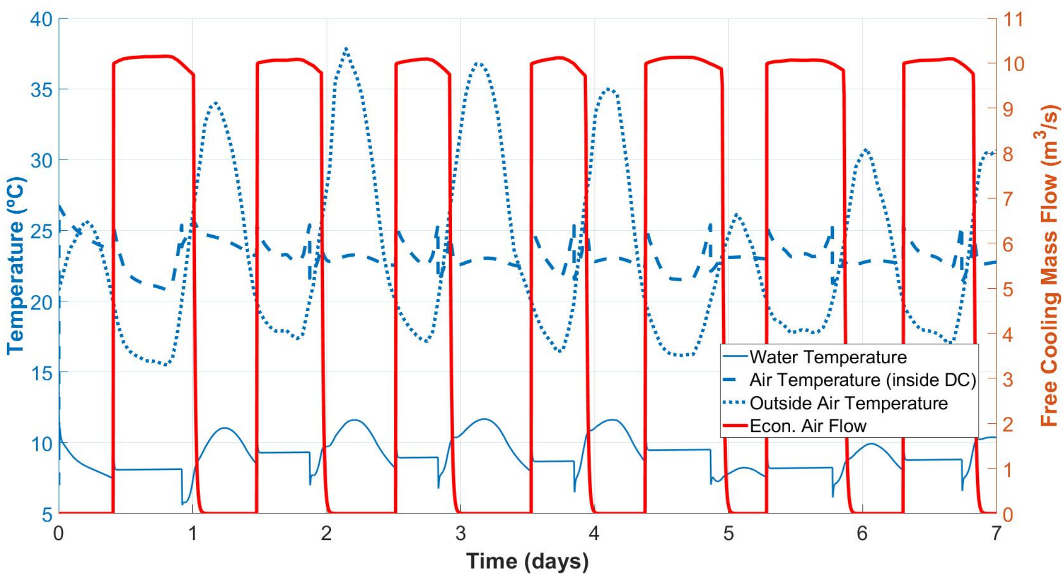

Figure 4.2: Model results for the implementation of an air economiser

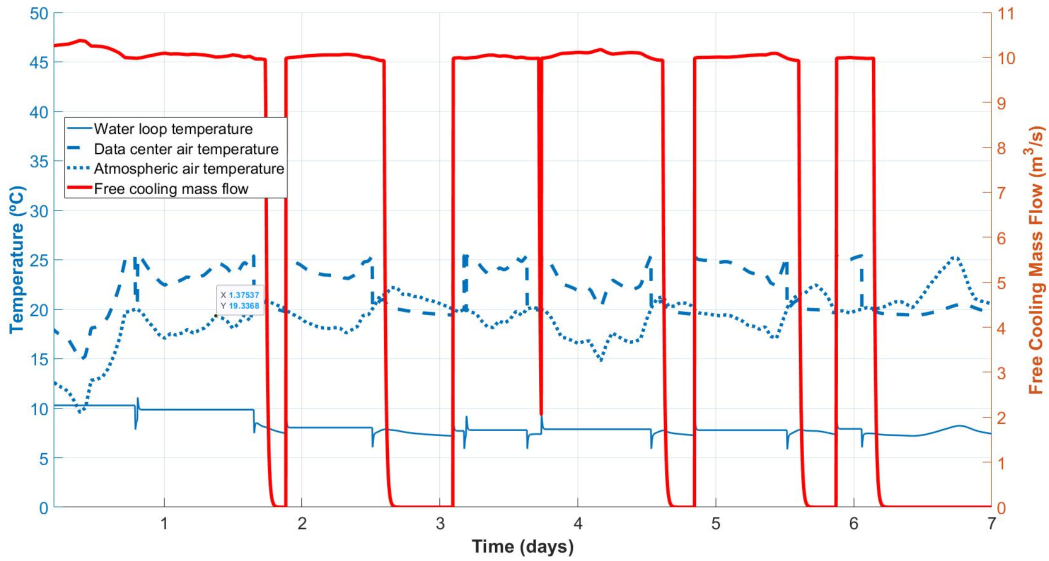

Figure 4.3: Model results for the implementation of an air economiser for a winter week

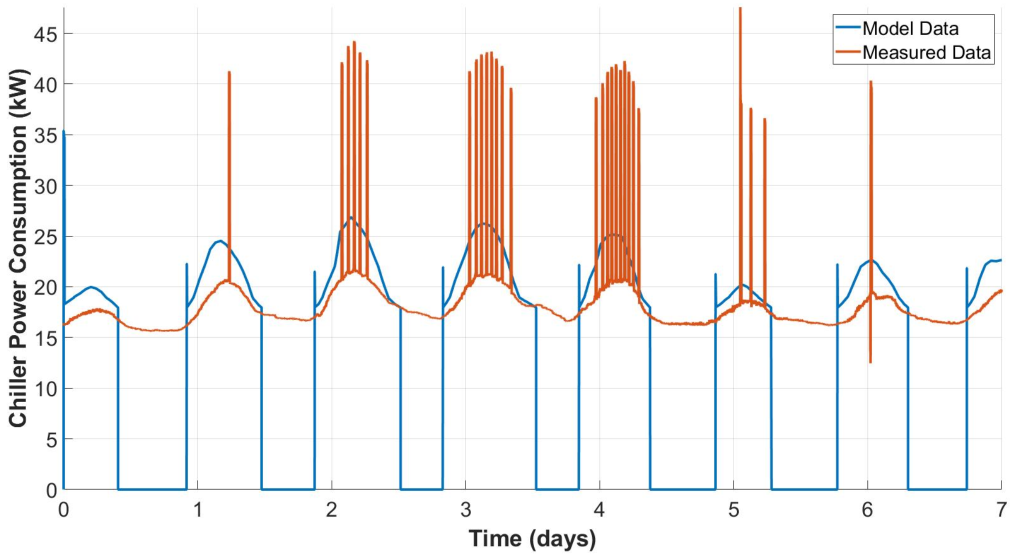

Figure 4.4: Comparison of the modelled chiller power consumption with an air side economiser and July 2020 measurements

Figures 4.2 and 4.4 both illustrate to a situation where the air side economiser is set to kick in when the outside air temperature is below 19.5ºC. For this case, the total energy consumption (chiller and air economiser fan) was $2126kWh$ for the whole week - this represents a 33% energy consumption decrease, when compared to the energy consumption yielded by the measurements done without the air side economizer in July 2020 (one of the warmest months of the year). The energy consumed by the fans inside the servers in the server floor was not considered because there was no direct way to measure it.

It is expected that in days where temperatures are lower *i.e.* most of the winter and spring/autumn, the energy savings would be much higher.

It is also important to add that, even taking into account the adverse conditions in the considered week when the measurements took place, by implementing an air economiser, the system would be able to keep the room near or at the recommended environmental conditions for data centers (according to the simulations), as specified by ASHRAE [9]. This is represented in figure 4.5. One can see that some of the point stray far away from the recommended area along the temperature axis, but these points occur during the first few steps of the simulation, where the model is slightly unstable. Generally, most of the points are inside or close to the recommended area, which is quite conservative itself.

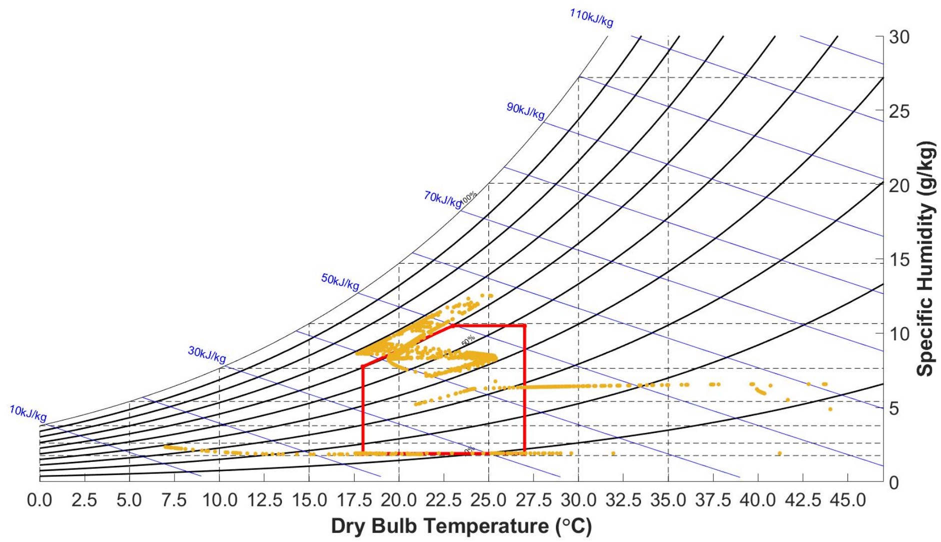

Figure 4.5: Simulated psychometric conditions inside the data center (orange dots), when using an air-side economiser. The area delimited by the red line is the recommended thermal envelope by ASHRAE for data centers [9].

Finally, to better understand how the temperature setpoint for the air economiser affects the energy consumption, several values were tried, as represented in figure 4.6. As the results seem to indicate, increasing the temperature cutoff setpoint for the economiser control seems not to reduce the overall energy consumption. However, the controller architecture seems to not be able to keep the room temperature at the desired setpoint once the temperature cutoff setpoint is higher than the temperature setpoint of the room. This makes sense, because if the system is trying to keep the room at the 20ºC temperature setpoint but is introducing air at 22ºC, then it will obviously not be able to keep the room at the desired setpoint no matter how much air it brings from the exterior.

This means that depending on the chosen room temperature setpoint, one should choose the cutoff temperature accordingly, in order not to lose control of the temperature inside the data center when temperatures are close to the cutoff temperature. Ideally, one would want the maximum temperature setpoint to be slightly lower than the temperature setpoint inside the room, so the economiser can regulate the temperature satisfactorily.

Tables 4.1 and 4.2 illustrate the atmospheric conditions, as well as average air temperature and humidity inside the data center as well as energy consumption for four different weeks in four different seasons, intended to represent the behaviour of the cooling system over a typical year. One can see that for the winter, spring and autumn weeks, one can reduce the total energy consumption signifi-

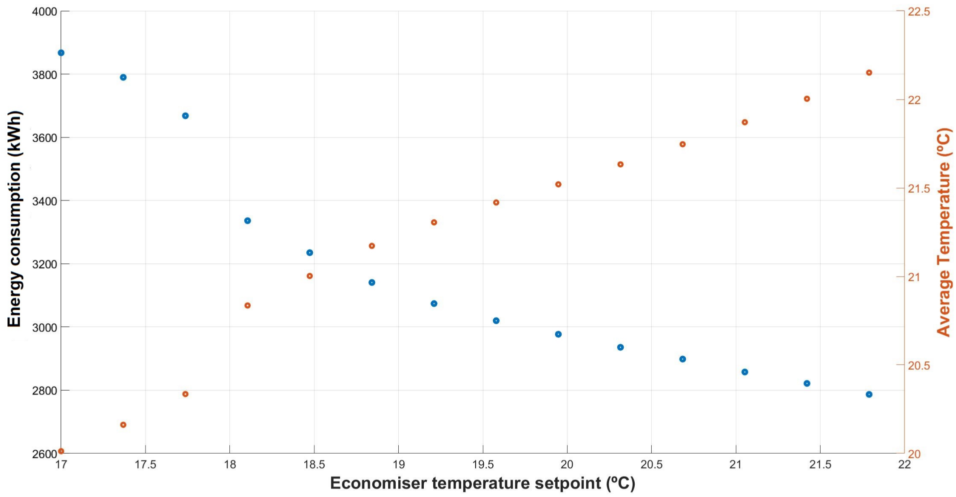

Figure 4.6: Impact of the economizer cutoff temperature on the average data center temperature and energy consumption in the considered July 2020 week

	Average atmospheric temperature (ºC)	Average atmospheric humidity (%)	Average server room temperature (ºC)	Average server room humidity (%)
Winter	8.45	70.18	20.06	68.00
Spring	12.37	77.59	19.6	50.11
Summer	23.37	59.37	21.33	57.43
Autumn	15.39	85.00	20.82	30.90

Table 4.1: Atmospheric air conditions and inside the data center for four different weeks, when using an air economiser

cantly. In the case of the winter week, one could even eliminate the chiller's share of the facility's energy consumption, while still keeping acceptable levels of temperature and humidity inside the server room.

It is worth noting that for a winter week, the use of a chiller can be completely reduced, since the temperature is much lower than the air temperature setpoint inside the data center. By averaging each week's energy consumption over an entire year, the predicted energy consumption over a typical year is 58059 kWh, which represents a 66.21% reduction in energy consumption for the same time period (171871kWh).

	Chiller energy consumption (kWh)	Economizer fan energy consumption (kWh)	Cooling unit energy consumption (kWh)	Total energy consumption (kWh)
Winter	0	218.12	0	218.12
Spring	139.82	587.69	50.94	778.46
Summer	2094.0	343.13	761.28	3198.4
Autumn	166.04	812.25	70.97	1049.3

Table 4.2: Energy consumption of the data center breakdown when using a air economiser.

4.2 Temperature setpoints adjustments

The cooling units inside the data center and the chiller are controlled respectively by the air temperature inside the Data Center and the outlet water temperature in the chiller's evaporator. By exploring different values for these setpoints, it may be possible to find more energy efficient operating conditions.

4.2.1 Room air temperature setpoint

The temperature inside IST server room is at this time kept at around 20ºC. This is in within ASHRAE recommended operational conditions for data centers (Figure 2.1). However, it is definitely on the conservative side and, therefore, potential energy savings could be achieved by allowing the room to be kept at a slightly higher temperature, without compromising the reliability of the servers. This would consequently result in a reduced electricity bill at the end of the year, since the chiller second compressor would have to kick in less times in order to keep the room at the desired temperature.

In this thesis, the intricacies of the cooling units control methods were not be explored in detail, since it's details are proprietary. As explained in section 3.2.2, the fan control method is modelled as a system for which the input is the room's temperature, which is kept at a constant value, by regulating the airflow passing the cooling coils. Therefore, to study the impact of regulating the temperature inside the data center, the control method was simplified, and several temperature setpoints were tried for the week of 3 to 10 of July.

While it is expected that the chiller will consume the same amount of energy, since the water temperature is unchanged, the cooling fans should consume less energy, since for the same flow, a greater heat transfer rate will be achieved because the temperature differential between the water and air increases.

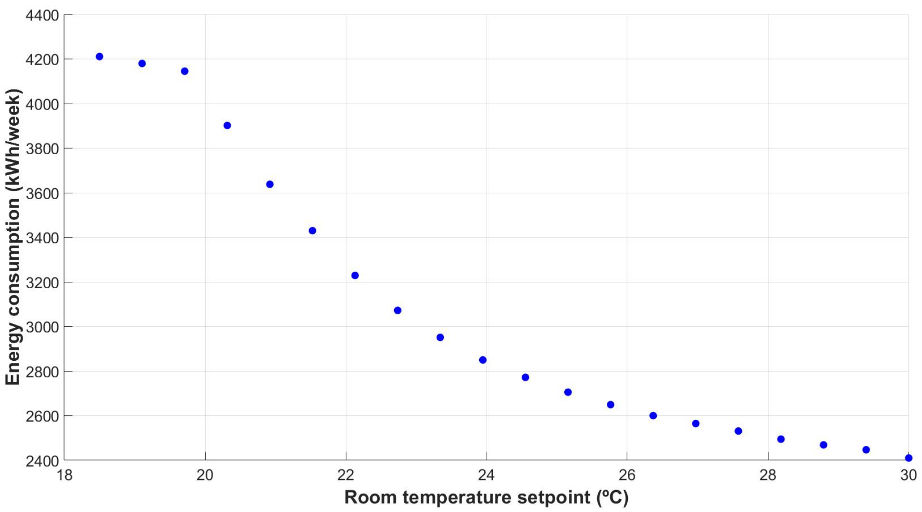

Figure 4.7: Impact of the air temperature setpoint inside the data center on the Chiller and cooling unit energy consumption.

Figure 4.7 shows the impact of changing the air temperature setpoint in the energy consumption for the week of the July 2020 measurements. It is possible to see how, for a higher temperature setpoint, the energy consumption reduces significantly in the range of 20 to 26 ºC (about 35%), and then the energy saving potential reduces as the temperature setpoint increases past 27ºC. The maximum recommended temperature by ASHRAE is 27ºc, which would make this an adequate value to take into consideration. These results suggest that a significant energy consumption reduction could be achieved for a relatively small investment.

In addition to exploring the impact that changing the air temperature setpoint had during the summer, the case of a winter week was also considered (from the 25th of January to the 1st of February 2021). The results are presented in figure 4.8. One can see, not surprisingly, that there is also a considerable reduction in the total energy consumption with the increase of the data center room air temperature setpoint. For the same interval as seen in the summer example (20 to 26ºC), the reduction in energy consumption may be 22% - a smaller amount but still considerable.

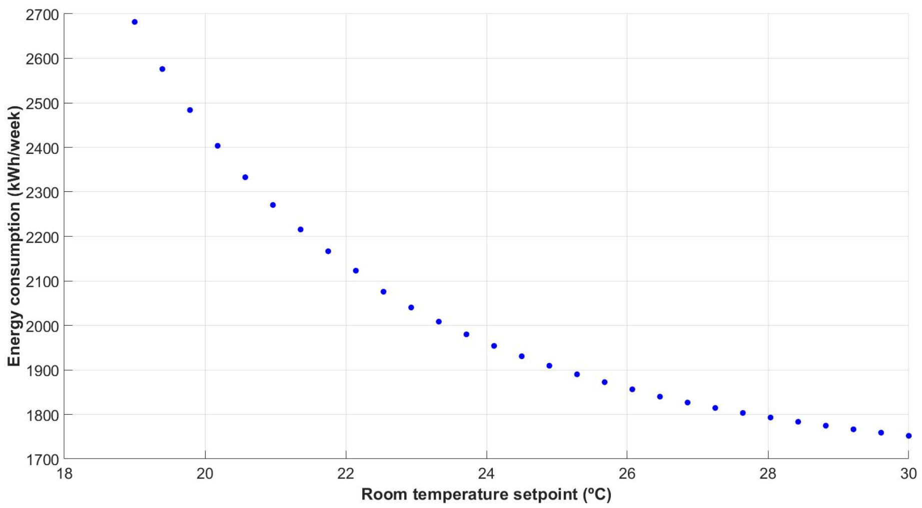

Figure 4.8: Impact of the Chiller's water temperature setpoint on the total energy consumption (during a winter week).

Tables 4.3 and 4.4 show the impact of changing the air temperature setpoint on the energy consumption of the system as well as the environmental conditions inside the server room. It is possible to see that for all the different seasons, the temperature and humidity remain relatively unchanged and within the ASHRAE recommended conditions (minimum of 10% humidity for a temperature of 25ºC). Regarding the energy consumption, this solution presents an average yearly energy consumption of 156091 kWh, which represents a 10% energy consumption reduction when compared to the present setup. This means that, depending on the complexity to change the algorithm in the cooling units, this could represent significant energy savings for a small investment.

It is therefore recommended the pursuit of this analysis in this area, in order to find out how to change the parameters in the cooling units controller in order to achieve a higher temperature inside the room and therefore the aforementioned energy savings.

	Average atmospheric temperature (ºC)	Average atmospheric humidity (%)	Average server room temperature (ºC)	Average server room humidity (%)
Winter	8.45	70.18	24.89	16.89
Spring	12.37	77.59	24.16	16.99
Summer	23.37	59.37	26.55	11.04
Autumn	15.39	85.00	25.00	16.48

Table 4.3: Atmospheric air conditions and inside the data center for four different weeks, when using a 25ºC air temperature setpoint

	Chiller energy consumption (kWh)	Cooling unit energy consumption (kWh)	Total energy consumption (kWh)
Winter	2489.0	267.2	2751.7
Spring	2542.7	301.2	2843.9
Summer	3037.2	482.6	3519.6
Autumn	2584.9	310.8	2894.8

Table 4.4: Energy consumption breakdown for the cooling system when using a 25ºC air temperature setpoint.

4.2.2 Chilled water temperature setpoint

As explained in section 3.2.3, the chiller compressor is controlled by the evaporator outlet water temperature - one of the compressors operates constantly as a base load, and the second starts when the water temperature raises above 13ºC in order to keep the temperature between 10 and 13ºC.

Since the chiller represents the highest energy consumption for the whole system, it is also relevant to study how adjusting the water temperature setpoint in the might influence the system. It is expected that by raising it, the chiller would show a lower power consumption. On the other hand, the cooling units fans would have to operate at higher rotating velocities (to drive more air) to keep the room at an acceptable temperature.

The different setpoint values were obtained by varying the lower bound of the control. In other words, when the temperature setpoint is set at 9 ºC, for example, the second compressor kicks in when the water temperature reaches 12.5ºC and turns off as soon as the water temperature reaches 9ºC again.

Figure 4.9 shows how adjusting the water temperature setpoint affects the weekly energy consumption in a typical summer week (3 to 10 July 2020). By increasing the water temperature setpoint, the tradeoff between the decrease in chiller energy consumption and the increase in the fan speed favours the latter, meaning that there does not seem exist an advantage to increase the water temperature setpoint.

To better understand if increasing this setpoint would result in energy savings when considering more seasons of the year, the results illustrated in tables 4.3 and 4.3 were obtained. Similarly to the results presented in 4.9, increasing the water temperature setpoint to 13ºC resulted in a yearly energy consumption of 187220.2 kWh, which represents an increase of 8.9% in the energy consumption when compared to the current setup. On the other hand, by decreasing the setpoint to 8ºC, the yearly consumption is also increased to 175609kWh, which represents a 2% increase in energy consumption as well. This seems to indicate that the current setpoint of 10ºC is probably the most adequate for the current setup.

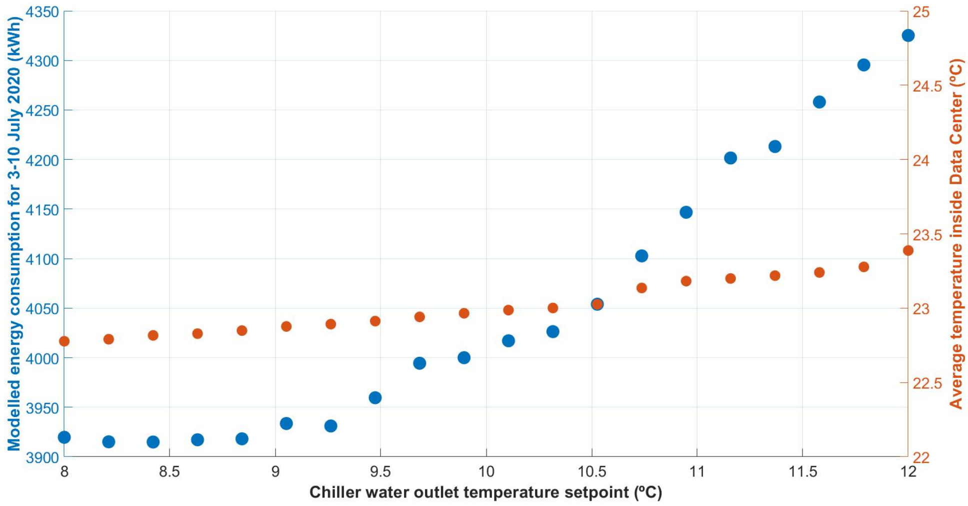

Figure 4.9: Impact of the Chiller's water temperature setpoint in a summer week.

	Average atmospheric temperature (ºC)	Average atmospheric humidity (%)	Average server room temperature (ºC)	Average server room humidity (%)
Winter	8.45	70.18	19.63	41.47
Spring	12.37	77.59	20.20	37.28
Summer	23.37	59.37	21.92	26.43
Autumn	15.39	85.00	21.64	26.43

Table 4.5: Atmospheric air conditions and inside the data center for four different weeks, when using a 12ºC water temperature setpoint

	Chiller energy consumption (kWh)	Cooling unit energy consumption (kWh)	Total energy consumption (kWh)
Winter	2486.4	522.3	3008.7
Spring	2536.3	1255.9	3792.1
Summer	2774.8	1553.3	4328.0
Autumn	2576.2	657.0	3233.3

Table 4.6: Energy consumption breakdown for four different weeks.

4.3 Water Cooled Chiller with Cooling Tower

In addition to the previously explained potential measures, replacing the existing air cooled chiller by a water cooled one might be a worthwhile investment from an energy saving standpoint. Firstly because Water cooled chillers present typically a much higher COP than air cooled ones due to the better heat transfer characteristics of water compared with air. This means that water cooled chillers are much more energy efficient. Secondly because the current chiller setup's cooling capacity might become insufficient in a medium to long time frame if the needs for IT services in the IST community increases, which is not a far fetched assumption.

In this type of setup, the chiller's evaporator would be connected to the water loop going to the data center, while the condenser would be connected to a cooling tower or dry cooler, which would form a second water loop.

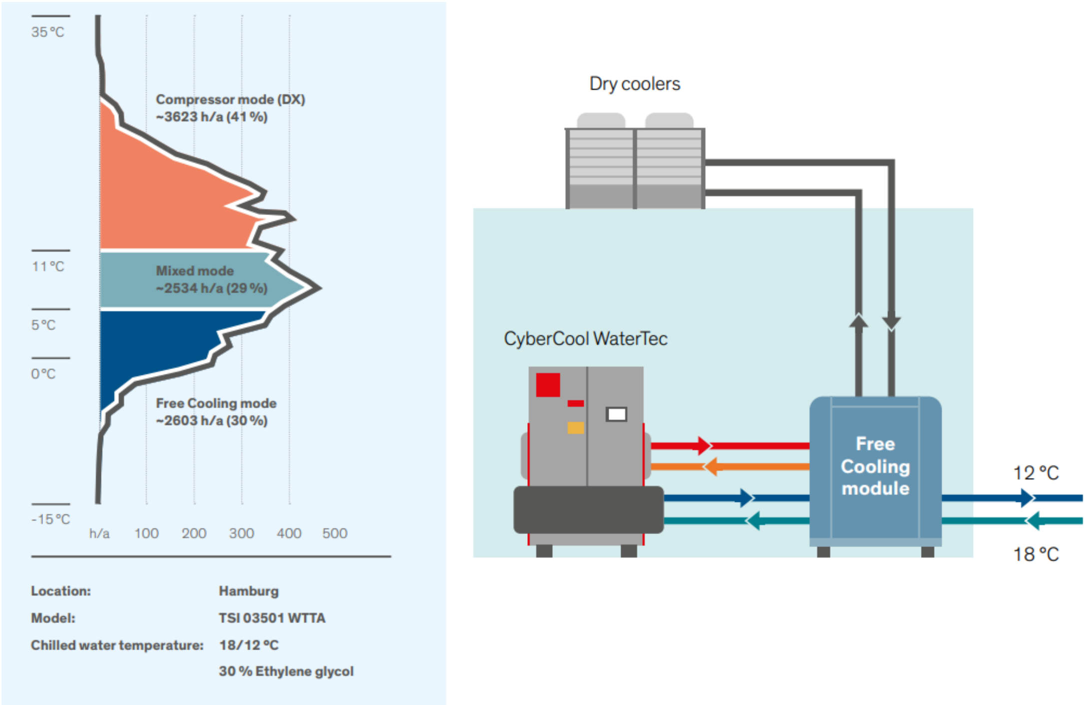

Figure 4.10: Example of a water-water chiller setup using Dry coolers [47]

In parts of the world where the the wet-bulb temperature is low enough for a long enough stretch of time during a typical year, the water could potentially bypass the chiller being directed straight to the cooling tower or dry cooler and enabling therefore the possibility to harness the natural cooling power of the ambient air, avoiding compressor-based cooling. This mode of operation is known as water side economiser and is illustrated in 2.14. This mode of operation would be controlled by the ambient wet-bulb temperature, since this is the minimum temperature to which a cooling tower can cool down an incoming water flow. In the case of the dry cooler, only the dry bulb temperature should be considered when choosing between the free cooling and compressor based cooling operation modes.

The design approach temperature then determines how close one can get the cooling tower's water outlet temperature to the wet bulb temperature under which it is operating. As represented in figure 4.1, the average monthly wet bulb temperature almost never goes below 10ºC in Lisbon, so this would translate into a very low number of economiser operation hours for the considered location.´

Figure 4.10 shows an example of a chiller manufacturer's data relative to the operating modes of a water cooled chiller in Hamburg, Germany. This specific manufacturer recommends a temperature setpoint of 5ºC for the system to operate entirely in the free cooling mode. If the atmospheric air temperature is between 5ºC and 11ºC the system operates in a mix of free and compressor based cooling and above 11ºc the cooling is entirely compressor based. It is possible to see that in Lisbon weather, the free cooling mode would be used in a very reduced amount of time using this setup since temperatures under 11ºC occur rarely throughout the year.

The cooling tower, was modelled to represent one of the model NX1010K-1, with a nominal cooling capacity of 425 kW, for a wet bulb temperature of 25ºC, as seen in figure 4.11, from the manufacturer's engineering data [47]. Although the fan speed in the cooling tower and the water pump speeds for both water loops could also be modulated, here they were taken as constants (the maximum 9 m^3/s of airflow in the cooling tower and 5 and 10 L/s for the server room and cooling tower water loops, respectively), because it introduced some numerical instability that proved difficult to solve when the control was introduced. This extra control could potentially further increase the efficiency of the system.

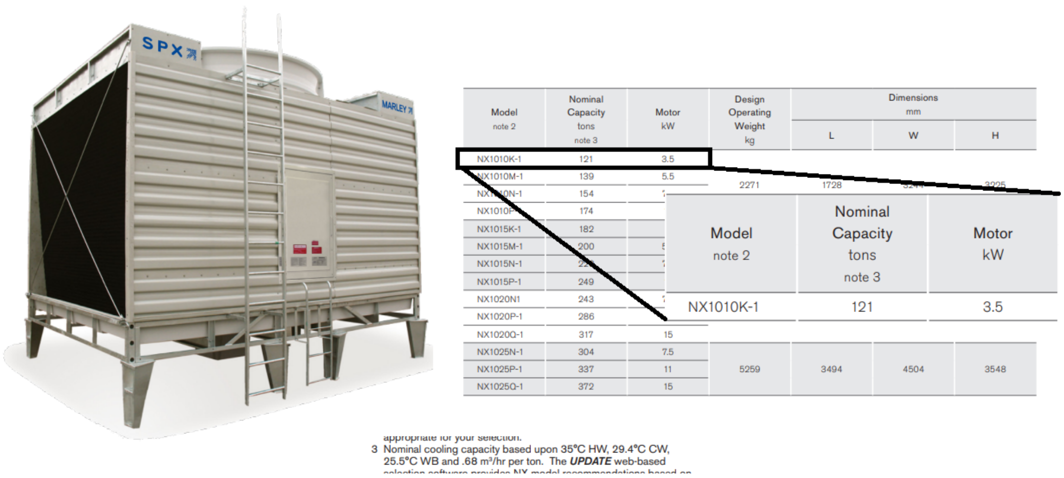

Figure 4.11: Marley NX1010K-1 cooling tower and technical data [48]

Figure 4.12 shows the results for a water cooled chiller connected to a cooling tower. The chiller's specifications were taken arbitrarily, except for its COP (7), which is around the common value for this type of equipment. The chiller compressor is controlled by the evaporator outlet temperature, using a Variable Frequency Drive (VFD). This is a common control strategy in modern chillers.

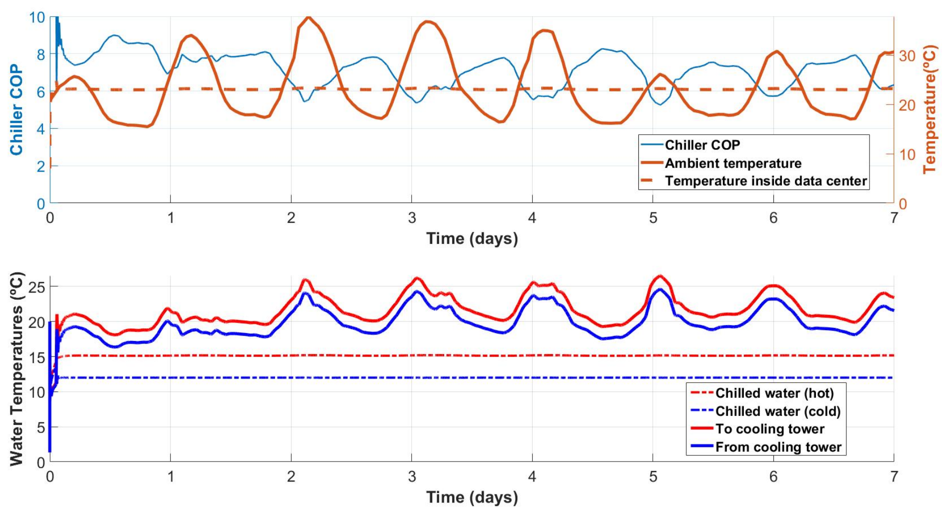

Figure 4.12: Results of the cooling system main variables when using a water cooled chiller

As figure 4.12 shows, the water temperature coming out from the cooling tower fluctuates during the day according to the wet bulb temperature, showing an offset of around 2 to 5 ºC, depending on the ambient conditions. This in in line with most new cooling towers. The chiller then modulates the refrigerant flow rate according to the water temperature to compensate it. For the considered week, the power consumption was of 3278.2 kWh, a reduction of about 20% when compared to the present air cooled chiller.

	Average atmospheric temperature (ºC)	**Average atmospheric humidity (%)**	**Average server room temperature (ºC)**	**Average server room humidity (%)**
Winter	8.45	70.18	21.42	19.33
Spring	12.37	77.59	21.5	19.25
Summer	23.37	59.37	20.89	21.2
Autumn	15.39	85.00	21.53	19.21

Table 4.7: Atmospheric air conditions and inside the data center for four different weeks, when using a water cooled chiller

	Chiller energy consumption (kWh)	**Cooling unit energy consumption (kWh)**	**Total energy consumption (kWh)**
Winter	1942.8	348.4	2653.9
Spring	2229.5	668.9	2898.4
Summer	2749.2	635.3	3384.6
Autumn	2546.0	672.0	3218.0

Table 4.8: Energy consumption breakdown for four different weeks, for a setup with a water cooled chiller

After doing the same analysis as in the other energy saving measures (represented in tables 4.7and 4.8) the yearly energy consumption, when using a water cooled chiller instead on an air cooled one was calculated to be of 157989 kWh, a reduction of 8% when compared with the air cooled chiller setup. This might seem like an underwhelming number at first, but considering that the current chillers will have to be replaced at some point, replacing them by this kind of chiller could make sense on the long term.

4.4 Compressor with Variable Frequency Drive

In this section an alternative control method for the chiller compressors is proposed and its potential effect explained.

The control strategy implemented presently is a simple one: one of the chiller's compressors is permanently turned on to cover the base cooling load, and the second one (in parallel with the first) is turned on when the temperature reaches a predetermined value of around 12ºC and turns off when the temperature reaches again 10ºC.

As stated in the report of Campus Sustentável [49], the current chiller's compressor control method is not appropriate when the cooling load increases, since the intermittent on/off behaviour leads to very dramatic current peaks, as seen in figure 3.16, taking a heavy toll on the compressor's lifetime.

As an alternative to the current control method, one could instead use a compressor controlled in a smoother way, using a VSD. The controller input could be the evaporator water exit temperature, which would be measured and the value compared to a setpoint. Depending on how far the measured temperature is to the setpoint, the controller commands the compressors to work at a set speed that drives a determined refrigerant mass flow rate. When the mass flow rate increases, so does the cooling capacity and the evaporator outlet temperature drops.

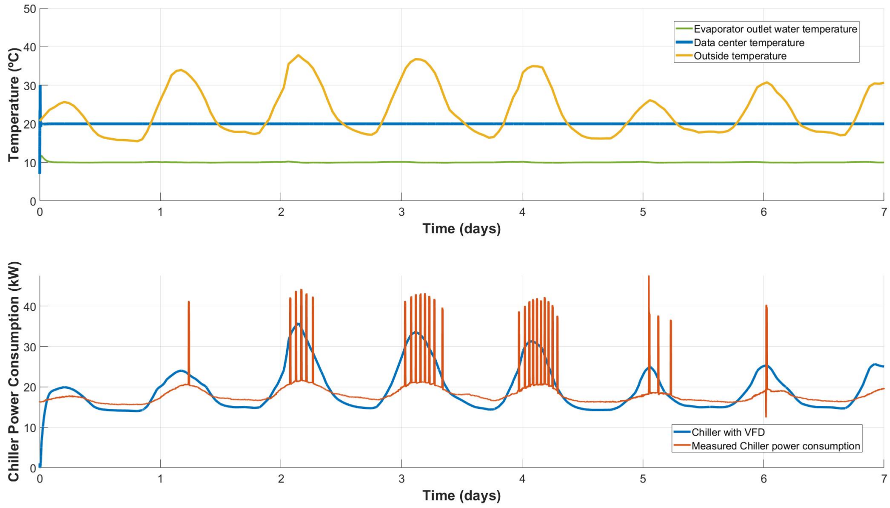

Figure 4.13: Comparison of chiller power consumption with a VFD and real life (for the week of 3 to 10 July 2020)

As seen in figure 4.13, the VFD controller allows for a smoother control of the water temperature. In addition to this, as a consequence of this control method, the compressors' power consumption is also much smoother.

	Average atmospheric temperature (ºC)	Average atmospheric humidity (%)	Average server room temperature (ºC)	Average server room humidity (%)
Winter	8.45	70.18	19.99	20.8
Spring	12.37	77.59	19.99	20.5
Summer	23.37	59.37	18.64	20.3
Autumn	15.39	85.00	20.00	19.8

Table 4.9: Atmospheric air conditions and inside the data center for four different weeks

In terms of power consumption, this control method showed an equivalent total energy consumption for the week, so the decision to have the compressor changed should be based on the potential life-time increase.

	Chiller energy consumption (kWh)	Cooling unit energy consumption (kWh)	Total energy consumption (kWh)
Winter	1958.7	749.6	2708.3
Spring	2178.1	700.2	2878.2
Summer	3017.4	650.2	3667.6
Autumn	2328.8	698.6	3027.4

Table 4.10: Energy consumption breakdown for four different weeks.

Tables 4.9 and 4.10 show how implementing this type of control affect the system.

4.5 Hot air containment

In most Data Centers, the racks are turned back-to-back with two parallel rows of servers outputting the hot air to the same row, as illustrated in figure 2.3 and 4.16. This allows for a higher temperature through the cooling coils, leading to a higher temperature difference between the coils and the air and, therefore, a cooling efficiency improvement, since less flow is needed for the same heat transfer rate. In the case of the IST data center, since the cooling units are interspersed in between the server racks, there is no need for the implementation of a plenum to direct the air into a centralised CRAC or CRAH units as is common practice in most legacy Data Centers (illustrated in figure 4.15)

In IST data centre case, in order to implement a hot air containment type solution, the top part of the server racks would have to be covered, to prevent the mixing of the hot air in the hot aisle with the cold air in the remaining parts of the room, as represented in figure 4.14. This type of installation is usually done using perspex panels or thin metal sheets shaped according to the servers layout in the room. The structure could be done using extruded metal profiles. In one of the ends of the hot aisle, there would be a door, to allow maintenance of the servers. Figure 4.15 illustrates what an installation like this would look like. The insulation does not have to be perfect, and some leakage is acceptable, but the most important is that the hot and cold air zones remain somewhat separated.

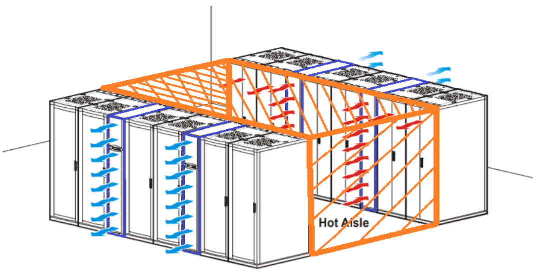

Figure 4.14: Illustration of a possible hot air containment method in the case of IST

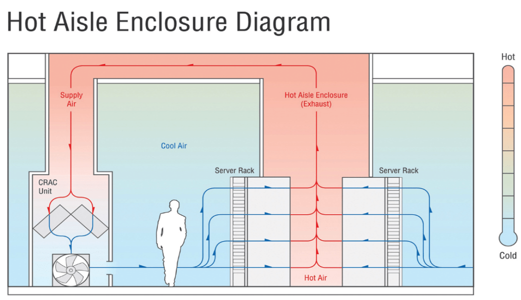

Figure 4.15: Diagram of a typical hot air containment system

From figures 4.14 and 4.16 one can see that cold air leaving the cold aisle enters the server racks, which supply the entering airflow with a constant heat flux equivalent to the IT Load. The air then flows into the hot aisle and from there goes to the cooling units (CRAH) and is cooled down back into the cold aisle. It is necessary to point out that the leakage between the cold and hot aisles was ignored, due to the difficulty in modelling this phenomenon. In any case, if the installation is built with sufficient care to avoid significant leakage, this should not impact the real life results considerably.

As a preliminary control approach, the airflow through the cooling units is controlled by the temperature in the cold aisle, in order to keep it at an acceptable temperature, since it is this temperature that affects the performance of the IT equipment. As one can see in figure 4.17, the cold aisle temperature is kept nearly constant, while the hot aisle temperature oscillates more.

In the case of Figure 4.17, the temperature in the hot aisle is kept at 25ºC- within the recommended values by ASHRAE. However, by varying the temperature setpoints, it is possible to get a broader view on how this setpoint influences the energy consumption. A comparison between the cases with and without

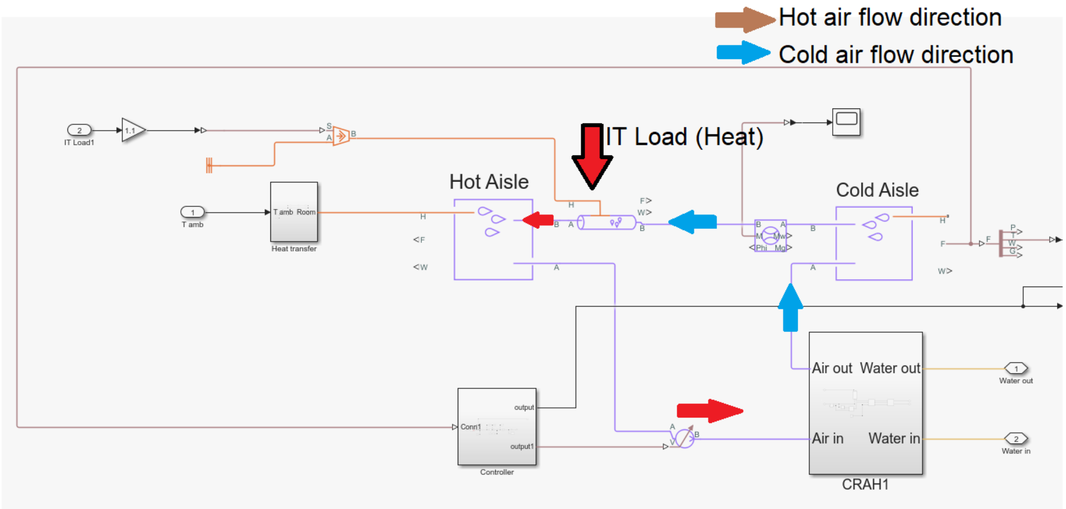

Figure 4.16: Illustration of the implementation a hot air containment system in Simulink.

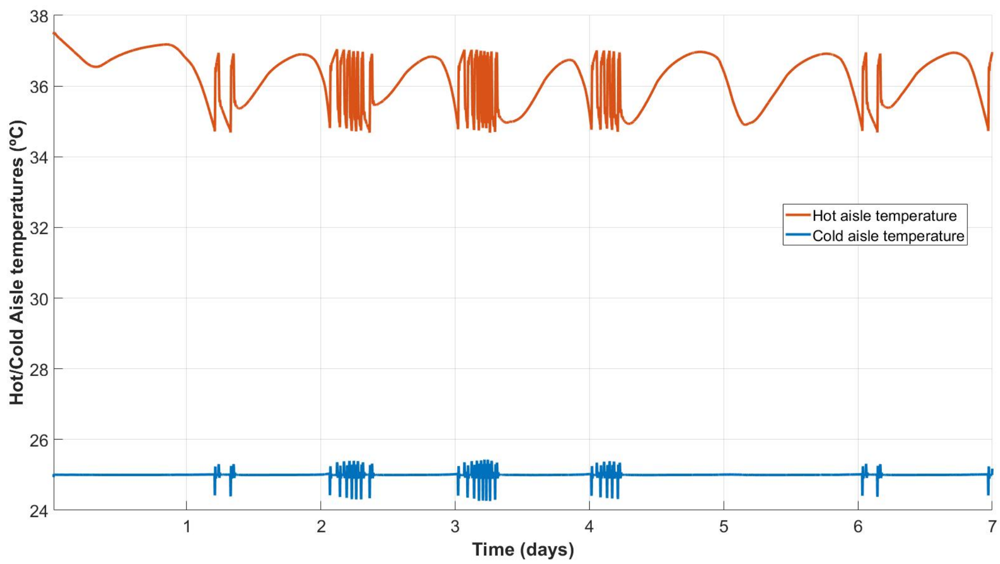

Figure 4.17: Hot and cold aisle temperatures

hot air containment was also performed. Figure 4.18 illustrates how varying the cold aisle temperature setpoint affects the overall energy consumption for the considered summer week (3 to 10 July 2020). As one can see, similarly to the results for the case without hot air containment, the higher the temperature setpoint is, the lower the energy consumption will be also in the case with air containment. In addition to this, it is possible to see that for lower temperatures values as setpoints, the energy consumption reduces more significantly in the case without hot air containment than in the case where this solution was implemented. In any case, if the temperature setpoint is chosen to remain the same, implementing this hot air containment measure might result in a shorter payback period than if the temperature is set higher than it currently is.

Finally, in figure 4.19, it is shown how implementing the hot air containment reduces the amount of airflow needed to cool down the room for the same air temperature setpoint. For the considered week,

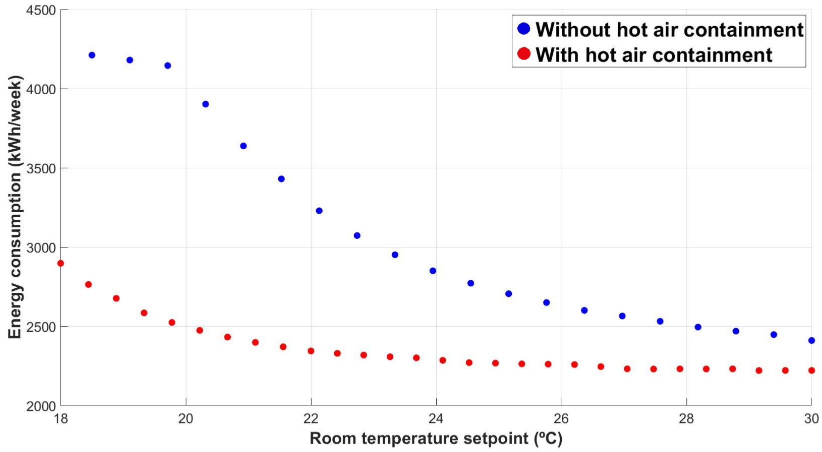

Figure 4.18: Comparison of the room temperature setpoint effect on weekly energy consumption when implementing a hot air containment system (3 to 10 July 2020)

the airflow reduction is of about 50%.

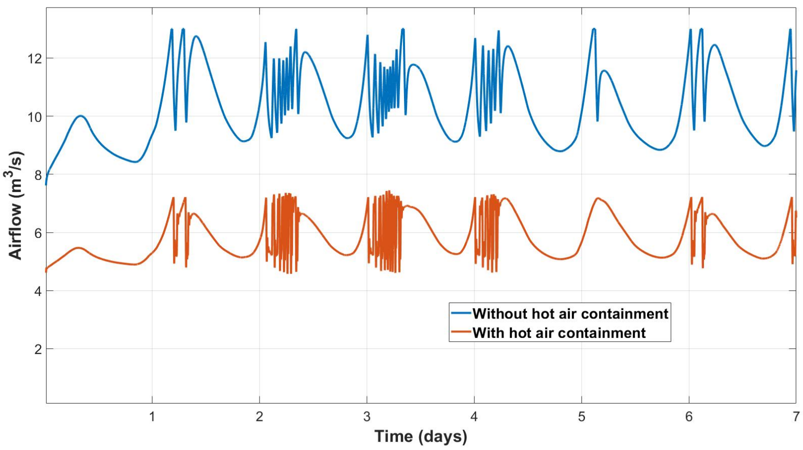

Figure 4.19: Comparison of cooling unit airflow with and without hot air containment.

Tables 4.11 and 4.12 show the impact of implementing this energy saving solution on the cooling system. One can see that the humidity and temperature remains within the limits for the cold aisle, which is the one that is of interest for this case, since it is from there that the air comes from to cool down the servers. It is noteworthy to point out that while the humidity is within the recommended limits, this variable is heavily dependent on the initial conditions since air exchange with the exterior is not considered. In any case, this type of separation between cold and hot air volumes translates into a yearly energy consumption of 145192.3 kWh, a 15% reduction in the yearly energy consumption.

4.6 Capacity analysis

Finally, it is worth investigating what is the capacity limit for the current cooling setup. The manufacturer's data suggests that the cooling capacity is 110kW for the design conditions, but the server room,

	Average atmospheric temperature (ºC)	Average atmospheric humidity (%)	Average server room temperature cold aisle(ºC)	Average server room temperature hot aisle(ºC)	Average server room humidity hot aisle(%)	Average server room humidity cold aisle(%)
Winter	8.45	70.18	22.18	31.49	18.53	28.6
Spring	12.37	77.59	22.21	31.53	19.31	29.6
Summer	23.37	59.37	22.11	31.48	18.52	29.4
Autumn	15.39	85.00	22.11	31.48	19.56	29.1

Table 4.11: Atmospheric air conditions and inside the data center for four different weeks when using a hot air containment system

	Chiller energy consumption (kWh)	Cooling unit energy consumption (kWh)	Total energy consumption (kWh)
Winter	2485.3	97.6	2582.9
Spring	2525.3	106.3	2631.6
Summer	3027.0	170.7	3197.6
Autumn	2616.3	109.6	2725.9

Table 4.12: Energy consumption breakdown for four different weeks when using a hot air containment system

pipework, etc, exchange heat with the outside air as well. In addition to this, the performance of the chiller varies depending on how far it is operating from the design conditions. Having said this, while the value of 110kW can be used as a rule of thumb for the cooling capacity limit for this system, a simple analysis was performed using the developed model for the data center. The results are illustrated in figure 4.20.

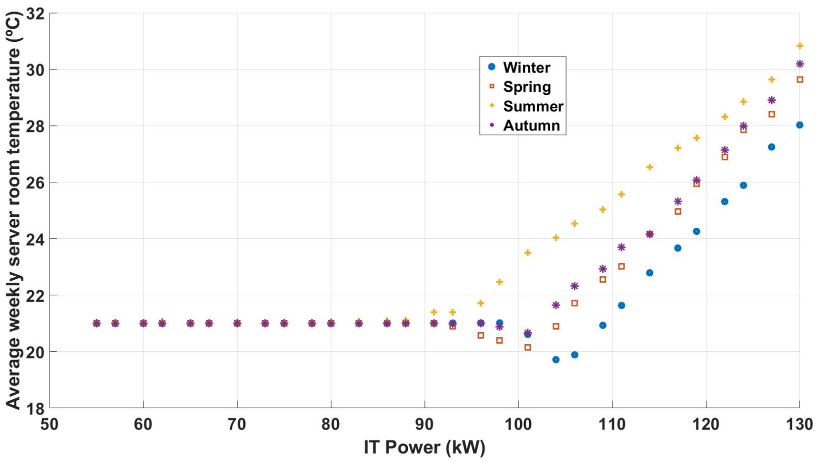

Figure 4.20: Comparison of room temperature setpoint effect on weekly energy consumption (3 to 10 July 2020)

For this analysis, only a summer week was considered, because this would be the hardest conditions for the chiller to operate in, as it's COP reduces with an increase in the outdoor temperature. The results, as laid out in figure 4.20, suggest that for the summer week, at around 95kW the average temperature throughout the week starts to increase linearly with the IT load, until it reaches 127kW, at which point the cooling system is unable to keep the server room within the recommended envelope suggested by ASHRAE. This would suggest that were IST to start to increase it´s IT power needs, it would be advisable to consider expanding the cooling capacity after the IT load reaches 110kW, which is the chiller's nominal cooling capacity. Alternatively, since there are 2 chillers in this installation (one for redundancy), it could be of interest to upgrade the piping to allow these two to work in parallel, since at the moment it is only designed for the operation of one chiller.

Chapter 5

Conclusions

The main objective for this thesis was to develop a reasonably accurate model representing IST's cooling system using a Simulink and Simscape - its physical modelling toolbox and to analize and propose energy efficiency measures in the Data Center, which may result in significand energy savings for IST.

5.1 Challenges

The biggest challenges when trying to achieve this objective were among others, to find the relevant characteristics of the cooling system equipment. In addition to this, the lack of precedent for the modelling and academic investigation on data centers or of most HVAC equipment using this software proved itself a challenge on multiple occasions.

Furthermore, the software proved to be rather cumbersome to work with, in multiple occasions, changing small parameters resulted in numerical errors. This meant that obtaining the results was often a frustrating and time consuming activity. This is not to say that the software is not adequate to simulate HVAC systems, only that the extensive detail the user can introduce into their models might also introduce a proportionate amount of troubleshooting.

5.2 Achievements

The main achievement for this thesis was to successfully gather the relevant technical data relative to the operation of IST Data Center cooling system and translating this knowledge into a working model in Simulink. The model accepts multiple inputs such as IT Load, outside temperature and humidity. In addition to this, it satisfactorily represents the behaviour of complex subsystems, for example the chiller.

Additionally, the model and it's sub-components are modular in nature - by aggregating the different sub-components (e.g. chiller, server room, etc), one could potentially represent a different data center without too much working effort (except the previously mentioned potential troubleshooting) . This could open the door to further work in modelling data centers and other systems relying in HVAC systems in Simulink.

Furthermore, the model allows tuning an immense variety of different parameters - from the refrigerant fluid's viscosity to the cooling unit's control methodology. This allows one to see how these parameters affect the system and as a consequence gain a better understanding of how the system behaves.

In the results presentation and discussion section, several potential energy saving measures were explored. The most promising ones were found to be the adjustment of the air temperature setpoint in the room, free cooling using an air economiser and hot air containment. This is because of their calculated energy savings as well as relative low cost to deploy. Changing the air setpoint should be the most straightforward one, depending on the level of effort make these changes on the cooling units' software. Implementing a hot air containment system could also be one cheaply, since the materials used (steel profiles and sheet acrylic) and there only being two server rows to isolate. On the other hand, installing an air economiser would be a more complex project, where an external contractor would have to be contracted, but the potential energy saving potential makes this measure worth further attention.

Finally, replacing the current air cooled chiller and changing the water setpoint, as well as replacing the compressor for a VFD control based one didn't show much potential from an energy saving prespective.

5.3 Future Work

As the need for web-based services grows, the cooling necessities should also increase accordingly. Therefore, there should be more research done on the cooling side of the IST Data Center to keep it up to date with standard industry practices. Other modelling softwares besides than Simulink could be used, such as Modelica, an open source modelling software for which there is a specific package directed for Data Centers. In addition to this, some of the energy saving measures recommended here are preliminary but have shown an interesting potential and deserve a more intensive study, as is the case of the water-cooled chiller cooling system approach.

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