Simulation of IST´s data center cooling system using Simulink

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Abstract

The objective of this thesis is to model the cooling system of IST´s data center using Simulink, 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 is considered to represent real life satisfactorily.

Keywords: Data Center, Simulink, Energy Efficiency

1. Introduction

A Data Center is a building (or part of a building) hosting information technology (IT) systems. They aree ssential in today's society, enabling companies and organizations to provide the most diverse type of internet-based services. 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 web-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 [8].

Due to the considerable amount of power required by these installations (from tenths of kilowatts to hundreds of megawatts for hyper scale data centers), 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, that shows a typical energy consumption breakdown in a data center.

Data centers worldwide represented 1.3% of the total yearly consumption (2% in the United States)in 2012 [7]. As information technologies become more and more essential for all kinds of economic activities, it would be expected that this consumption would increase correspondingly and therefore the need to improve the cooling efficiency of these installations becomes even more relevant.

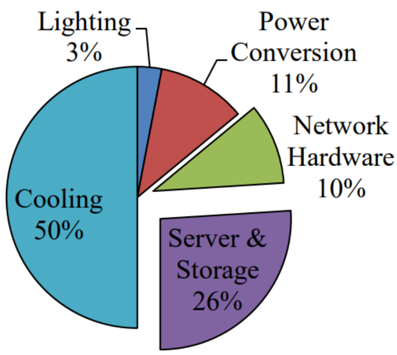

Figure 1: Typical Data Center energy consumption breakdown [11]. The IT power is represented in the "Servers and Storage" section.

2. Background

In order to perform their function correctly, data centers need large ammounts of electrical power 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.

Another important feature of a data center is the necessity to operate without interruption. To do this, the air inside the data center should remain within recommended limits of temperature and humidity in order to maximise reliability. ASHRAE is the leading authority in this matter it´s reccomendations are represented in figure 7.

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. 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. in addition to this, more and more plant operators are starting to use air and water economisers in order to harness the power of natural cooling - using ambient temperature to cool down the data center when possible.

2.1. IST Data Center

The IST data center is located in the Central pavillion of the Alameda campus, in Lisbon. This installation is responsible for keeping essential web based services for the IST's community, such as Fenix and the Técnico Webmail running.

Inside the data center, the servers are interspersed by the cooling units (as seen highlighted in blue in figure 2). 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 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.

3. Modelling and Software

Simulink was the main software used in this thesis to develop a physical model of the IST data center´s cooling system . Simulink is a graphical programming language,which allows the user to simulate the behaviour of complex physical systems.

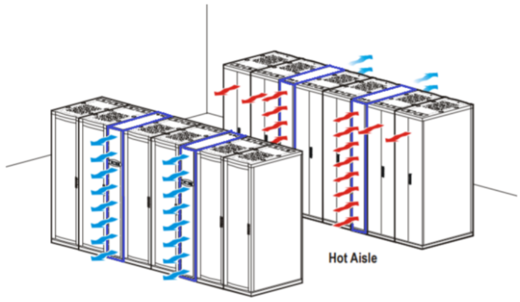

Figure 2: Illustration of a data center utilizing the INRow®RC cooling units. Adapted from [1].

3.1. 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,where most of the heat is generated by the IT equipment.

3.2. Cooling Unit/CRAH

The cooling 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 air of $1380l/s$.

All the cooling units were simplified as being a single air-water heat exchanger, uning the ε-NTU heat transfer method. This comes as a consequence of simplifying the room as a single point. The values of heat transfer coefficients and areas were chosen based on trial and error based on the available data.

3.3. 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.

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

3.3.1 Evaporator

The evaporator is a plate heat exchanger with exterior dimensions of approximately 0,62x0,262x0,3m.(see figure 3 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.

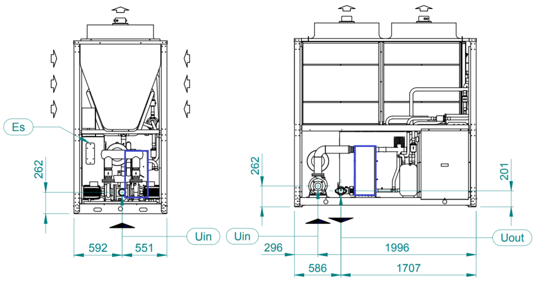

Figure 3: Chiller's dimensions in millimetres (evaporator highlighted in blue), taken from [4]

	Refrigerant	Water
Heat transfer area (m^2)	14	14
Fouling ($m^2 * K/W$)	0.0002	0.0002
Heat transfer coefficient ($W/(m^2K)$)	6000	4873

Table 1: 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[10] for the refrigerant and water sides of heat exchangers.

To calculate the water-side heat transfer coefficient (assumed constant), a number of correlations existing in [3] were investigated. However, since many of the parameters are not known, a value calculated by Hsieh and Lin [9] for a plate heat exchanger with similar dimensions and the same refrigerant fluid of $h_{ref} = 6000 W/m^2K$ was used.

3.4. Condenser

The chiller's condenser is one of the fin and tube kind. Hot refrigerant flows through the inside of

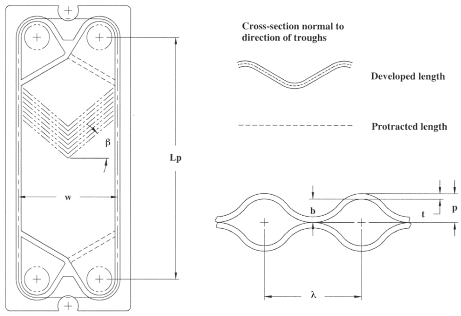

Figure 4: Plate Heat Exchanger dimensional parameters, from Ayub [3]

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.

	Refrigerant	Water
Heat transfer area (m^2)	18.85	220
Fouling ($m^2 * K/W$)	0.0002	0.0002
Heat transfer coefficient ($W/(m^2K)$)	173470	110

Table 2: Heat transfer areas, fouling factors and coefficients for the chiller's condenser

Kays and London [12] 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 $587 m^2/m^3$, a heat surface area of $220 m^2$ is obtained. This area concerns the airflow. The chosen fouling factor for both refrigerant and air flow was the same as in section 3.3.1.

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

The heat transfer coefficient for the refrigerant can be estimated using equation 1 [5], for fully developed refrigerant flow undergoing dropwise condensation in a horizontal tube.

$$h_{dc} = 51104 + 2044T_{sat} \quad if\ 22 < T_{sat} < 100\ °C$$
$$h_{dc} = 255510 \quad if\ T_{sat} > 100\ °C \tag{1}$$

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 $30 - 50°C$, 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 $80°C$), so an average condensation temperature of $60°C$ was assumed, resulting in a heat transfer coefficient of $h_{dc} = 17367\ W/(m^2K)$

3.5. Comparison with real life

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.

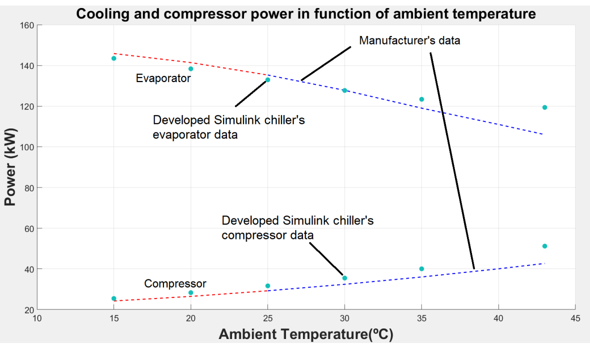

Figure 5: Compressor power consumption and evaporator cooling power in function of ambient temperature. Manufacturer's data from [4]

As can be seen in figure 5, the model closely follows the manufacturer's data for the range of 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.

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.

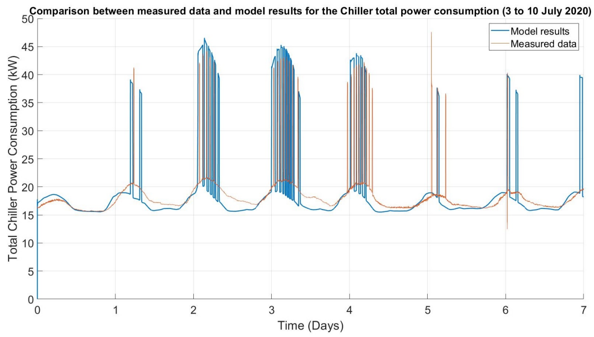

Figure 6: Comparison between the measured and simulated chiller power consumption values

One can see in figure 6 both the measured chiller power consumption, and the chiller power consumption calculated by the model (total chiller 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 value 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.

4. Results

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

According to the existing literature, the potential energy savings when using air side economisers 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 7. 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.

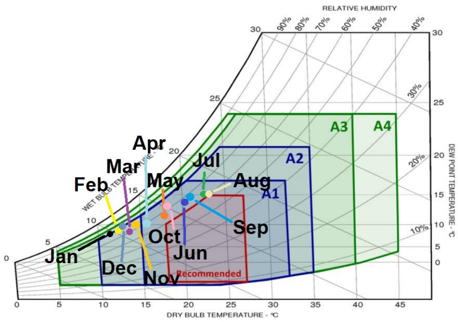

Figure 7: Comparison between Lisbon´s monthly average temperature and humidity with ASHRAE's recommended environment conditions inside data centers. Graph from [2] and climate data from [6]. The different areas' represent different reccomendations for different uses. The red area is the most inclusive.

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

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 $10m^3/s$.

For a case where the air side economiser is set to kick in when the outside air temperature is below 19.5ºC, 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 [2]. This is represented in figure 8. 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.

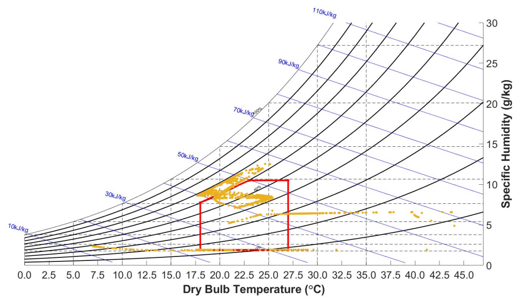

Figure 8: Simulated psychometric conditions inside the data center (orange dots), when using an air-side economiser. The area delimited by the red line is the recommended thermal envelope by ASHRAE for data centers [2].

Finally, to better understand how the temperature setpoint for the air economiser affects the energy consumption, several values were tried, as represented in figure 9. 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.

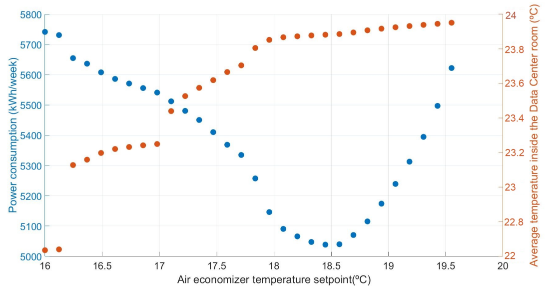

Figure 9: Impact of the economizer cutoff temperature on the average data center temperature and energy consumption in the considered July 2020 week

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.

By running the simulation for four different weeks with different weather conditions intended to 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).

4.2. 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. 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. 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.

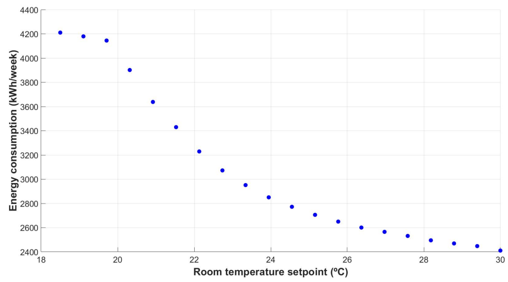

Figure 10: Impact of the air temperature setpoint inside the data center on the Chiller and cooling unit energy consumption.

Figure 10 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 11. 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.

After simulating for different seasons, it was 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

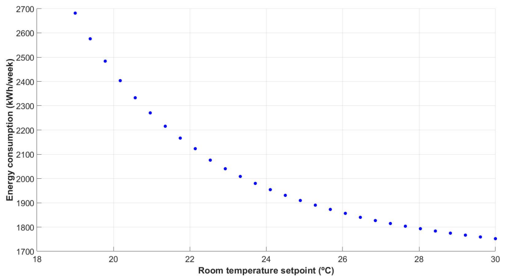

Figure 11: Impact of the Chiller's water temperature setpoint on the total energy consumption (during a winter week).

savings for a small investment.

4.3. Chilled water temperature setpoint

The chiller´s 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.

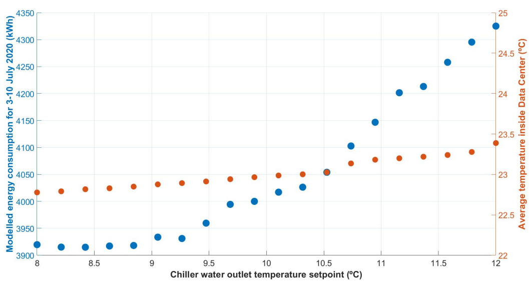

Figure 12: Impact of the Chiller's water temperature setpoint in a summer week.

Figure 12 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.

Similarly to the results presented in figure 12, 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.

4.4. 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.

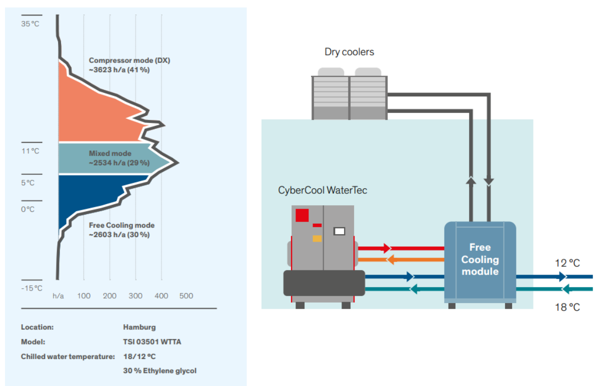

Figure 13: Example of a water-water chiller setup using Dry coolers [13]

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 13. 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 7, 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 13 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, from the manufacturer's (Marley SPX) engineering data [13]. 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 14 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.

As figure 14 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.

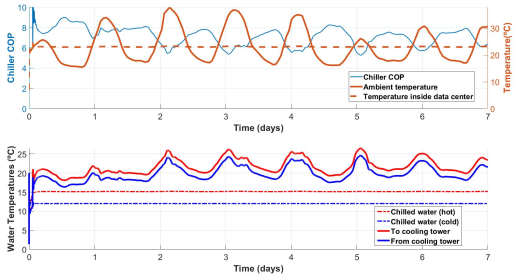

Figure 14: Results of the cooling system main variables when using a water cooled chiller

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. 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.

In IST's 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.. 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. 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. 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. The cold aisle temperature is kept nearly constant using this method.

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 hot air containment was also performed. Figure 15 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.

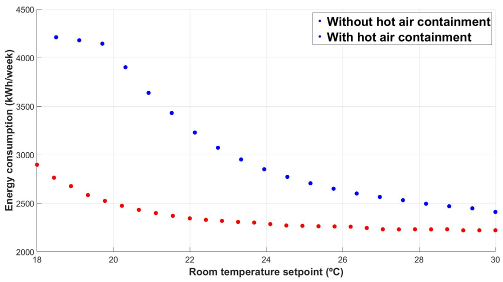

Figure 15: Comparison of the room temperature setpoint effect on weekly energy consumption when implementing a hot air containment system (3 to 10 July 2020)

By implementing the hot air containment, the amount of airflow needed to cool down the room for the same air temperature setpoint can be reduced. For the considered July 2020 week, the airflow reduction is of about 50%.

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.

5.1. Capacity analysis

Finally, it is worth investigating what is the capacity limit for the current cooling setup. The results are illustrated in figure 16.

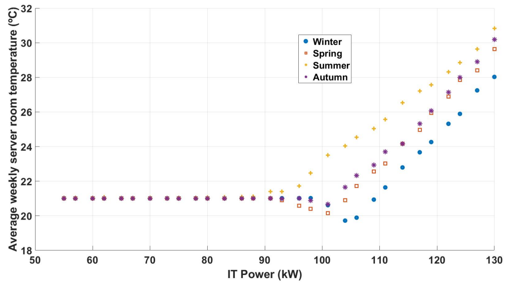

Figure 16: Comparison of room temperature setpoint effect on weekly energy consumption (3 to 10 July 2020)

The results, as laid out in figure 16, 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.

6. 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.

6.1. 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.

6.2. 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 done cheaply, since the materials used (steel profiles and sheet acrylic) and there only being a small number of 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.

6.3. 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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