Understanding Cooling Delay in High Density Data Center

by

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ABSTRACT

With the ever-increasing demand for high-end services, technological companies have been forced to operate on high performance servers. In addition to the customer services, the company's internal need to store and manage huge amounts of data has also increased their need to invest in High Density Data Centers. As a result, the performance to size of the data center has increased tremendously. Most of the consumed power by the servers is emitted as heat. In a High Density Data Center, the power per floor space area is higher compared to the regular data center. Hence the thermal management of this type of data center is relatively complicated.

Because of the very high power emission in a smaller containment, improper maintenance can result in failure of the data center operation in a shorter period. Hence the response time of the cooler to the temperature rise of the servers is very critical. Any delay in response will constantly lead to increased temperature and hence the server's failure.

In this paper, the significance of this delay time is understood by performing CFD simulation on different variants of High Density Modules using ANSYS Fluent. It was found out that the delay was becoming longer as the size of the data center increases. But the overload temperature, *ie*. the temperature rise beyond the set-point became lower with the increase in data center size. The results were common for both the single-row and the double-row model. The causes of the increased delay are accounted and explained in detail manner in this paper.

ACKNOWLEDGEMENTS

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CHAPTER 1

INTRODUCTION

The data management is an important part of any organization. The ever increasing demand for on-line services, the digital content of media products and the company's necessity to meet its own demands of proper database management for future requirements have imposed the need for highly powerful and continuously operating data centers. The need to survive in this competitive market have forced even smaller organizations to have a sustained and highly reliable data center that can meet their demands. With the data that needed to be managed is increasing exponentially big corporations are investing huge amounts on expanding their data center capabilities. Microsoft has recently announced plans to invest for their data center expansion with an estimated amount of $350 million [1]. Facebook has announced to expand its data center capacity at Oregon for a total land area of around 307,000 sq. ft [2]. The largest member of the Internet world, Google, Inc. has invested in its data center facility a huge sum of $1.9 billion in 2006 and $2.4 billion in 2007 [3]. The data center infrastructure is a huge business such that with only meager rise in 2.4% investment has resulted in $143 billion spending. The trend is expected to rise and according to Gartner, forecasts have projected the investment to increase to 7% [4]. Although it is the norm for big players to expand their data center facilities with increasing requirements, the developing and smaller organizations can't afford the price to meet the requirements. They needed an alternative to sustain and provide proper service. Some recent innovations have been made in improved data center design facilities and alternative server technology.

The basic notion is to increase the power utilization per floor space area. This means that the processing capabilities and the number of servers needed to be increased in the limited containment. These high power usage data centers are commonly referred to as high-density data centers.

According to Gartner, the high-density zones/data centers are defined as volume containing servers that consume more than 10 kW per rack. It can also best described in terms of single rack as those racks that are minimum 50 % filled [5]. The idea is to opt for highly capable blade servers that can be packed in the racks to perform heavy workloads. With recent advancement in the infrastructure and cooling technologies, smaller companies can easily migrate towards deploying high density pods in their low-density infrastructure facility as stated in [6]. Traditionally the data center cooling is uniform and the controller circuit monitoring the overall temperature of the data center maintains it. Hence in that type of facility the mix of high-density racks and low density racks or remodeling the low density to upgrade it to high-density modules might not be viable option. But with advancements in server sensing and workload placements [7], the companies can shift towards the high-density modules. This results in high initial investment. Even what is more challenging is the need to maintain the facility to in order to sustain the operations devoid of interruptions, breakages, outages and permanent failures. Improper maintenance has resulted in serious damages in terms of money and reputations. A survey conducted by IDC, has found out that nearly 85% of the companies couldn't provide proper customer service, canceled application roll-outs and reallocation of resources to goals away from the planned strategy due to server failures, cooling failures and the poor performance of the data centers [8]. This resulted in huge losses in terms of financial benefits, customer retaining and market share loss. The main reason being, poor data center facility design and improper cooling management. All these have summed up to the frustrations of the organization and the customers as well.

The data center design facility should be sophisticated enough to sustain the operations. This includes proper air handling, server deployment, workload management and the cooler ability to respond to the sudden temperature rise of the servers. Intel in its paper [9], has laid five important facilities design aspects for a reliable and sustained data center infrastructure. This includes air management, thermal management, architectural considerations, electrical considerations and stranded capacities. The data center design is critical. It generally includes wire distribution, server rack placement, workload distribution, air management, cooler temperature and humidity control, cooler response, power utilization capacity and the outside air interaction. But the design can be broadly classified into two main categories, workload management and thermal management. The former is controlled by optimized algorithm to process the data and prevent overload of the servers or racks. It is dealt extensively by the software management personnel and reduces the workload stress on the servers. The thermal management is equally important to maintain the normal operations of the data center. The proper cooling systems, air flow management and the response time to the unexpected rise in server temperature and the room temperature are critical factors in determining the sustained thermal operations of a regular data center. In case of high-density data centers, the above-mentioned factors are delicate and a simple error can affect the life of the servers and hence the maintenance cost of the data center to a larger level.

It is clear by now that the energy costs to operate the data center is almost as high as cost to operate the servers. The high-energy consumption of the servers for continuous service providing and the continuous cooling energy consumed by the chillers to maintain the operations have increased rapidly over the period. This trend is foreseen to increase not even stabilize at the least. The cooling power required by the chillers is almost equal to the power consumed for server operation. The paper [10] on data center energy consumption during 2013 gives an alarming numbers. It is estimated that the power consumed by the data center is around 91 billion kilo watt-hours annually. This figure will spike to around 140 billion kilowatt-hours annually. This amount of power is equivalent to power generated from 50 power plants. But at the current period, technologies are incompetent to mitigate neither massive power consumption nor does the cooling technologies help to reduce the cooler power consumption. Hence considering the current technological trends and the continuous user demands for digital content and on-line services, the power consumption is not going to reduce. Hence the thermal management must ensure the sustained operations of the servers without any thermal failures and interrupted services, which in turn costs huge financial and capital losses.

The server failure is a major crisis with regards to the infrastructure management of the data centers. The server failure rate is quite unpredictable and can depend on lots of factors. But one important factor that can severely hamper their operations and permanently damage the servers is the surface temperature or the exit temperature of the cooling air from the server surface. If the temperature of the cooling air is quite not within the set bandwidth for the normal server operations, then it can critically damage the server operation. According to ASHRAE in [11] gives the maximum and minimum operating temperature ranges for the servers. Traditionally the operating the temperature range for the data servers existed between 18 ^{0}C to 27 ^{0}C. This situation has changed dramatically since then. With the advancements in server technology and proper cooling infrastructure inside the data center the temperature ranges have increased. Now the temperature ranges mostly depend on the server type and type of operations the servers performs inside the room. The data center can consist of different types of processing equipment for varied operations. The major classification as mentioned in the paper consists of 1U/2U networking, large networking, server equipment of high processing capabilities and the servers exclusively used for large data storage and management. The maximum temperature rating for all the equipment varies, and all lie closer to 40 ^{0}C to 60 ^{0}C. In our project the modeled server is 1U/2U networking servers. Hence as mentioned in the paper, the maximum exit temperature of the server can reach to 50 ^{0}C and 55 ^{0}C. The humidity level is also an important factor. But majority of the chillers are intelligently designed in such a way that the humidity level are brought down to acceptable range by cooling the incoming air to dew point temperature and hence the humidity level are brought down to acceptable range by default. Despite proper temperature settings the airflow management plays a highly important factor in ensuring the proper cooling of the servers by the coolant air. Proper air management must ensure the prevention of localized heating caused by both recirculation and bypassing the other servers. In small containment this phenomenon occurs even more predominantly. Hence careful assessment of airflow through the servers must be done to prevent such events. Improper airflow management can also cause high risk cooling decision issues. For example if the monitoring system detects the localized heating at some server, then the chiller will bring down the temperature of the recirculating air, because that will be the programmed decision of the chiller control system. While this may result in slight temperature decrement at the heated server, the other servers will be at high risk, because of the high cooling and hence the operational failure of other servers. Hence the air distribution plays a major role in determining the server failure rate because of the localized heating caused by bypass and recirculation of hot air within same zone of the servers.

The server failure rate heavily influences the liabilities in the data center infrastructure management. The servers are vulnerable to the temperature of the cooling air. Improper cooling caused by either the high temperature of the air or overcooled air are dangerous to the normal server operations. Even with the advent of intelligent cooling systems arriving to automatically detect the temperature rise beyond the threshold, the chiller response has to be immediate to bring down the temperature of the cooling air and hence ensure the longevity of the server operations. As stated above, the airflow management is a critical factor that can effectively carry out the chiller's response to the detected high temperatures. While there might be several reasons for server failures, the majority of the failure is caused by improper workload distribution [12, 13]. The impact of the failures is also huge and the numbers are so high that the cost incurred in repairing or replacing the working machines directly impacts the organization's income and the performance as well. In a study published by Google's I/O conference [14] on its 1800 servers, the failure rates of the servers are very high. The estimated failure rates are discussed in terms of the temporary machine failure to complete breakdown of the servers forcing it to be replaced. During the first performance year of the clusters, it is estimated that about 1000 machines will fail completely. In fact because of the overheating and thermal breakdown problems, most of the servers will fail within 5 minutes and will take about 1-2 days to recover.

In fact this is the major problem facing all the companies moving from low density to the high performance high-density data centers. The thermal management of the high heat emitting servers, deployed in the small containment volume, is very complicated. It is one of the critical and complicated tasks while designing the infrastructure of the data centers. The server placement might be one important factor. Proper methods of placing the server not only serves the purpose of effective performance of the data center as a whole [15, 16], but also efficient thermal management of the data center greatly reduces the power consumption [17] of the servers and the cooling systems and thus ensures the reliability of the operations. The airflow management is also one of the important techniques of stabilizing the temperature inside the data center.

This method of reducing the temperature of the servers after detected temperature rise/ decline is termed as reactive techniques [18]. This method of thermal management although works simple, it is various disadvantages while maintaining the high-density data centers. By the time, the thermostat detects the overall rise/drop in temperature of the data center or the temperature anomalies at the server exit; it would be late for the cooler system to bring the temperature back to normal operations. Hence the pro-active approaches [19, 20] are undertaken for earlier detection of the server temperature or the room temperature overall by speculating the possibility of the temperature variation before even the temperature reaches above or below the set point temperature. This type of approach ensures not only the smooth operation of the data servers, but the reduction of overall power consumption of the data center also ensures lower cooling cost and the longevity of the servers.

1.1 Objective

The predictive mechanisms or the pro-active approaches are very complicated to be performed in a high-density data center. Because of the smaller floor-space area, the temperature variations are very robust and chaotic, which makes the prediction complicated. The temperature variations can be caused by any of the factors such as improper temperature setting of the cooler, insufficient airflow or excessive airflow, by-pass flow and recirculation. The conjugate heat transfer mechanism from the server to the cooling air only complicates the flow mechanism that should be solved by Navier-Stokes equation. The high flow field in a smaller volume causes local turbulences. Combined with all this, the solution has to be derived for transient flow. Hence deriving an algorithm to predict the temperature variation makes it more complicated and can't be assured of its accuracy. Hence the pro-active approaches can't be efficiently employed to determine the temperature rise and effectively manage the thermal stability of the data center.

For the high-density data centers, the reactive approaches can be suitable option because of the insufficient development in the pro-active mechanisms. But as stated earlier, the reactive mechanisms have severe disadvantages in predicting the temperature variations earlier and countering them. Hence there is a need to determine the value of increased temperature above the set-point temperature and the reactive time of the cooler to counter the temperature rise inside the data center. The cooler has to detect the increased temperature above the set-point temperature and should counter the increased heat rise as early as possible. But usually there will always be lag in detection and time taken for the mechanical systems to reach the desired power setting to cool the incoming air. Hence the lag in time of the cooler to react to the increased temperature of the servers, reach the cooled system settings and stabilize the increasing temperature of the cooling air is termed as *cooling delay*.

Understanding this cooling delay is very critical in designing the infrastructure of the data center and fixes the power settings of the rack and the chiller systems. The delay time to react is directly proportional to the value of increased temperature above the set-point temperature. Longer the cooling delay time, higher the temperature is increased above the set-point temperature. This temperature value is directly proportional to the failure rate of the server.

The server failure rate is directly proportional to the temperature and the reaction time of the chiller or per say the time taken to bring down the temperature. The failure rate (FR) or the acceleration at which the failure will occur is defined using the Arrhenius equation [21] which can be calculated using the temperature above the set-point temperature and the cooling time delay of the chillers which is expressed as

$$FR = A * e^{\frac{-Ea}{KT}}$$

Where, A is the reaction rate constant

Ea is the activation energy

K_B is the Boltzmann constant which is equal to 1.38×10^{-23} J/K

T is the increased temperature above set-point temperature, which is expressed in Kelvin

Hence it is vital to detect the temperature rise of the servers and determine the cooling delay of the cooler systems. Else this will directly affect the server life and hence add upon the maintenance cost of the data center. With lots of companies migrating towards the high-density data center, it is critical to develop a means to effectively determine the cooling delay and offer some cooling techniques to immediately stabilize the rapid temperature variations of the high-density data center.

The CFD (Computational Fluid Dynamics) can be the best tool, which can be effectively implemented to predict the temperature variations inside the high-density data center [22, 23, 24, 25, 26]. The CFD method is one of the best options to evaluate the system performance under constant changes. A single professional can be employed to effectively model and analyze the system that makes it all the more inexpensive. While the real-time monitoring of the data center involves expensive installation of sensors and the data that is collected from them is localized, the CFD data can effectively give a comprehensive visualization of the entire activity and the data can be extracted at any desired point of the room without the need for installation charges. The improved design and the infrastructure create a what-if situation, which can't be monitored and analyzed with the expensive real-time monitoring systems. Hence this analysis tool offers potential efficiency gains for the organizations to analyze the system effects before the investing in bigger infrastructure.

The work done in this project considers three variants of data center with linear increase in the floor-space area. There is a need to analyze the cooling delay in each variants of data center. It is very logical to conclude that the cooling delay increases with increase in floor-space area because of the larger air volume and therefore the longer time for the cooling air to react to the temperature change. But there is a critical need to quantitatively determine the amount of cooling delay for each variation of the floor-space area. This can effectively help the companies to design the infrastructure of the data center and save lots of investment on maintenance.

Hence we use three variations of data center on each of a kind, ie. three variations on single-row model and three variations of double-row model and determine their relation with respect to the cooling delay time and the increased temperature above the set-point temperature before it is being stabilized. The size variation of each of the data center requires different airflow speed in order to prevent thermal nemesis such as recirculation and bypass flow. Therefore some suggestions on the airflow management for each of the models are also discussed in order to obtain stable thermal conditions during the normal operations.

CHAPTER 2

RELATED WORK

Numerous works have been previously done to describe about the power failures inside the data center, cooling outages and the temperature bandwidth for server operations and the proper thermal management for sustained operations. These works form the basis of the current research work as it comprehensively deals with the cooling, thermal management and the delay in cooling.

The standard conditions of the air temperature inside the IT room are computed at 24^0C, which is chosen as normal operating temperature for the current simulation [27, 28]. In the latter research paper, they developed a transient model for chiller operation failure and the thermal inertia of the pipe components. The developed models are processed through some computational software and are validated through experiments for that particular model. A proposal has been made for validation of the model for other data centers through CFD simulations. Like the previous work, a transient model for heat recirculation in order to predict the temperature rise of the servers and determine the hot spots inside the data center was developed and discussed in [29]. The faster and lightweight approach model predicts the temperature rise in few minutes compared to the high time consuming CFD simulations to determine the temperature rise. The paper also discusses the about predictions of hot air distribution inside the data center and its effect on incoming air for new servers. The comparative study done in [30] discusses about the different types of cooling methods currently employed in high density data centers and concludes on best methods of cooling in order to obtain maximum cooling efficiency. While operating a data center, the performance on workloads completed in unit time is as critical as the cost of operation. This is clearly discussed in [31] and [32]. In the former, they discuss about a developed robust scheme where the cost is greatly reduced by allowing a delayed workload performance from the data centers at different geographical locations. Conversely analyzing the paper, the performance of the data center can be increased by increasing the power usage of the servers. The power usage of the servers affects the heat emission and in turn results in increased data room temperature. In the latter, the deployment of the type of servers and the workload distribution between highly cooled servers placed near the CRAC and the low power-mode servers, which are harder for, cool air to reach and the effective cost benefits are discussed. This generally gives an idea about the airflow management for cost savings and high functionality of the data centers. Hence depending on the requirement of the data center, a trade-off has to be established between the cost and the heat emissions for the data center. The airflow management is critical factor in data center management. The server fans play a major role in immediately removing the high heat from the server surfaces. But the operation should be optimal such that the power consumption and hence the cost of cooling should be optimal, the rate of airflow expressed in CFM are all-important features for enhanced air circulation in data centers. All the above-mentioned factors are explained in detail in [33]. The ramifications of bypass air caused by improper airflow management in a hot-aisle, cold-aisle data center are discussed in detail in [34]. They lay claim that effective air flow management can be attained by proper pressure sensing system and the aisle containment can an additional practice that can be employed to achieve maximum cooling and reduce the chiller operating cost in the data center.

The simulation performed by CAE organization on a twenty-rack model using CFX model. The work gives general idea about the normal operating temperature and the set point temperature of the servers [35]. Another work done in [9] shows how the CFD simulation can be an effective method to analyze the physiology of data centers and predicts some hazards beforehand. Some of the factors that can efficiently managed by the CFD modeling are controlling air distribution, predicting bypass air flow, system failure prediction due to over-heating, some of the redundancies like same server cooling and failures caused by the obstructions in lower floor. From the analysis the best cooling practices and optimal configurations are put forth for proper thermal management of data centers and meet the future requirements of the operations.

The cooling outage and the server temperature rise are two important factors for a sustained thermal management in a data center. The former is discussed in detail in [36] and [37]. In the former, a zero-dimensional heat transfer model was developed to determine the temperature rise in the data center during the power outage as a function of time constant and evaluate the thermal performance of the data center during this time. The time constant factor was developed as a function of the data center size and the factors affecting these parameters were evaluated and analyzed. In the latter, the CFD model determines the temperature rise in different sections of the data center due to the chiller failure. Based on the performed simulations and other strategic techniques they give a general comparison of which method would be suitable to counter the temperature rise of the servers during the chiller failure. The proactive approach as mentioned in [38] describes the cooler response delay and the resulting overheating of the servers. Although their norm of cool air temperature and server set point temperature is not followed, the significance of time delay and its consequences gave the comprehensive knowledge about the two considered parameters. It is this work that is taken as an important contribution to the above discussed parameters and the research described here in this document, takes this work as a foundation for the rest of the research work.

CHAPTER 3

MODELING AND MESHING

3.1 Model Description

The models were designed based on the realistic data centers recently built in Tempe, Arizona. In order to develop a realistic version of the model, two variety of the already functioning data center version were chosen. It consists of a single-row rack model with one hot aisle and one cold aisle. The other data center model is a two-rack model with two cold-aisles and one hot-aisle at the center. The chiller was placed below the raised floor plenum and the cold air was pushed through the cold air tile and it was recirculated back to the chiller through the hot air tile. The cold air was pushed from bottom section and the hot air was sucked through the hot air tile placed on the same floor. This reduces the unnecessary height of the data center room at the top.

The model was described in terms of floor space area, width of a single rack and the number of racks placed in each row. Some of the other important dimensions were chosen from the existing data center model at the IMPACT Lab, Arizona State University. This includes the vent size, the tile size, rack height, rack depth and the cooler dimensions. The distance between the tile and the rack; tile and the data center walls; bottom of the floor and the raised floor plenum are assumed of the reasonable values.

Based on the given floor space area and the room dimensions the width and the length of the room were calculated. This is shown in the appendix of this document, The calculated values of the length and the width of the data center room are used to render a two dimensional version of the data center with the top view showing the top part of the data center is shown in the figures down below

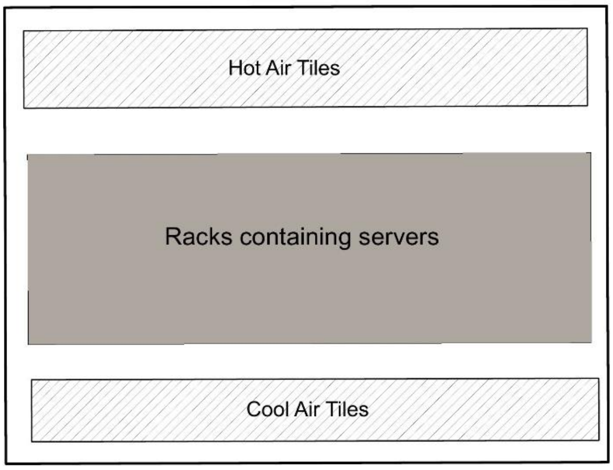

Figure 3.1: Top View Of The Single-Rack Data Center Model. The Cooler Floor Is Below The Operating Data Center

The Power Distribution Unit (PDU) is placed along with the Racks in the same row. As shown in the figure, the cool air from the chiller is forced through the cool air tiles by the cooler fans and distributed to the servers. After removing the heat form the server surfaces, the hot air is now pushed through the hot air tiles by the incoming cool air. The cycle repeats continuously in order to maintain the stable temperature inside the data center.

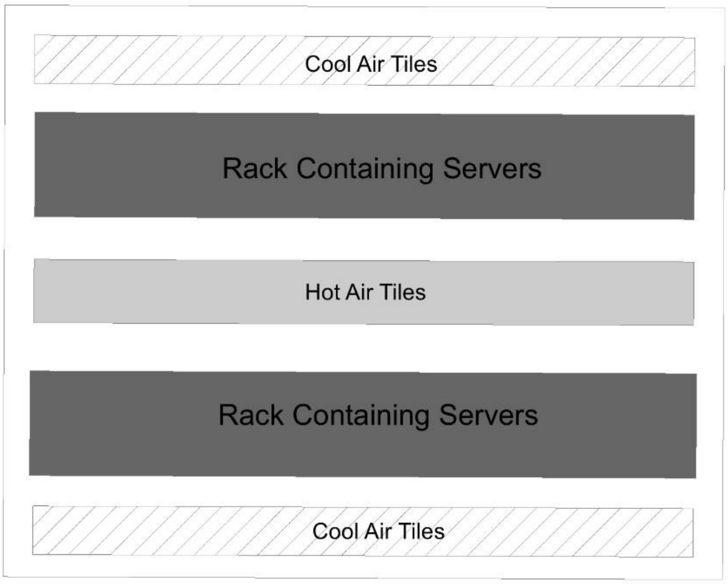

Figure 3.2: Top view of the double-row model

In this model, just like the IODMOD model, the each of the coolers placed underneath the Racks pushes the cool air on either side of the racks. But the hot air is pushed through the common hot air tile placed in the middle. Thus the hot air, which usually expands and rises because of the density change of the flowing air, has to be forced down through the hot air tile. The PDU is placed in each of the racks at the corner of the Racks.

Using the calculated and the assumed values,, the design was developed as three-dimensional model using Salome-Meca, which is open-source design and analysis software.

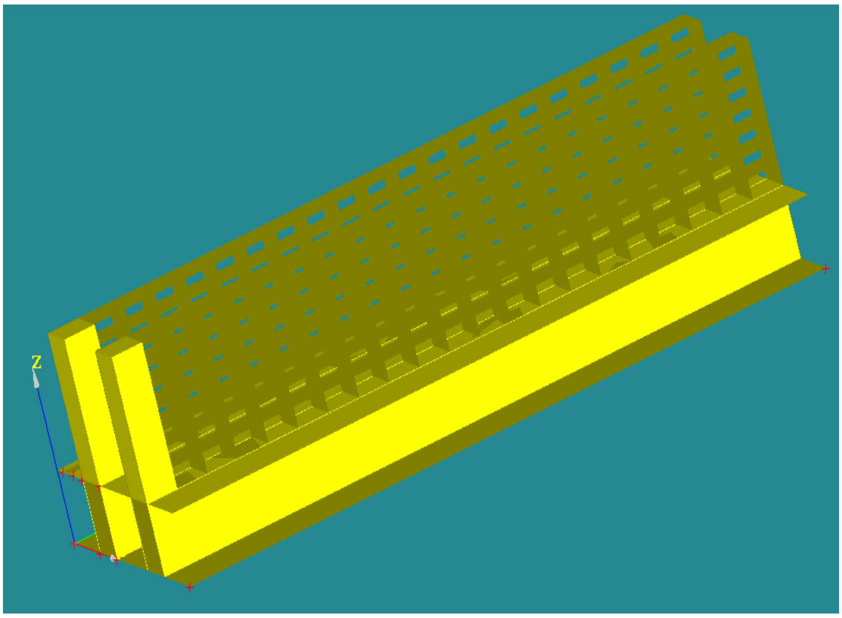

Figure 3.3: Three-Dimensional Model Of The Double-Row High Density Data Center

3.2 ANSYS Tool

ANSYS workbench is used to perform simulation in our case. ANSYS is a multipurpose engineering simulation software used to design, mesh and perform simulation for variety of applications including static and dynamic analysis, structural analysis, Heat Transfer, Fluid Dynamics and magnetohydrodynamics [39]. It has numerous interactive keys and push buttons, which will provide easier navigation for the user to design, mesh and carry simulation in its environment. Because of these features, the ANSYS is the most widely preferred simulation tool among the industrial personnel. The ANSYS module, which is used to perform the fluid dynamics simulation, is the Fluent. It is a very powerful general-purpose fluid simulation software which is used for variety of fluid dynamics problems including turbulent flows, heat transfer and compressible flows. Because of the advanced solver techniques and the interactive set up options, it is widely recognized and employed by industries to perform the required simulation. It has numerous solving techniques to suit variety of problems and resolves the iterations to converge faster. Multiple options enable the user to modify the mesh and improve the quality of solution. It also gives the user to develop their own codes using the TUI command window in order to modify the module to suit the current problem and define the nature of the parameter acting on the system. The options required for the set up are arranged in serial order, which helps the user to proceed sequentially and also navigate quickly between the features. The interactive tools make it easier to post-process the results and analyze them in detail with wide variety of option to inspect the different parameters affecting the simulation [40]. The results can also be exported in formats, which can be imported in other tools and can be used for data processing and presentation. Because of these feature and advantages, the model is developed, meshed and simulated in ANSYS workbench.

3.3 Geometric Modeling

The geometry was developed based on the given specifications, standard dimensional values, reasonable assumed values and the calculated values dependent on the previously mentioned values. Keeping the dimensions of the single-row and double-row models, the dependent lower capacity modules and high capacity modules, *ie*. smaller data center and larger data center were developed. The data center were called as High Density Modules and represented as HDM. The nomenclature consists of the general description of the type of the data center represented by HDM, followed by number of rows represented by 1 or 2 and the increased dimensional order represented by A, B and C. Hence the six data center developed were represented as HDM-1A, HDM-1B, HDM-1C, HDM-2A, HDM-2B and HDM-2C.

ANSYS Design Modeler was used to develop the three-dimensional models. It is a relatively simple tool with lots of user-interactive features to design all the models. The entire data centers were developed as single-rack model. From the room dimension calculation, it is found out that for each rack the dimension of the cooler is nearly 1.1 times. Hence a single-rack and single-cooler model was developed and the cooler power was inputted as such to have 1.1 times the power of a single-cooler. This type of developing a model greatly reduces the simulation time and the resources needed to perform the simulation.

Initially the dimensions of the servers, cooler, server fans, cooler fan and number of servers per rack are first calculated. The dimensions of these components are constant and can imported to develop other data center models. The dimensional calculation these components are shown in the appendix. Using these values, the server and the cooler frame are developed as solid models. The fan for both the server and the cooler are developed as thin surfaces. This thin surface can be designated as fan component in electronic materials in the design modeler. The height is kept as 84 inches and it house 8 servers inside it. Height of each server is calculated to be 10 inches with the server frame taking the rest of the length. The server fan was placed at the tail of the server frame and placed at 2 inches from the end. The cooler is 40 inches high and has an inch thick frame. The cooler fan is placed exactly placed at the center of the cooler frame. The distance between the cooler top and the server bottom is about 5 inches. Since the dimensions of the rack and the cooler is same for all the data center models, the developed and imported to the rest of the models

Having developed the working components of the data center model, the cool air tile-rack distance, tile-wall distance is assumed to have reasonable values from a set of standard operating high-density data centers. In each of the cases, the floor length dimension is increased by almost double and the tiles are placed further apart from the racks. But the dimension of the cool-air and hot air tiles, height of the raised floor plenum and the height of the data center is also the same in all the models. The width of the model is the same as the width of the full-size data center.

3.3.1 HDM-1A

The HDM-1A model has smallest length of the all models developed. It is compact and has just enough floor-space to accommodate the servers and the cooler. The distance between rack and the tiles is kept to 5 inches and the distance between the tiles and the wall is kept to be 3 inches. This dimension is sufficient to hold the servers with minimum air-space volume.

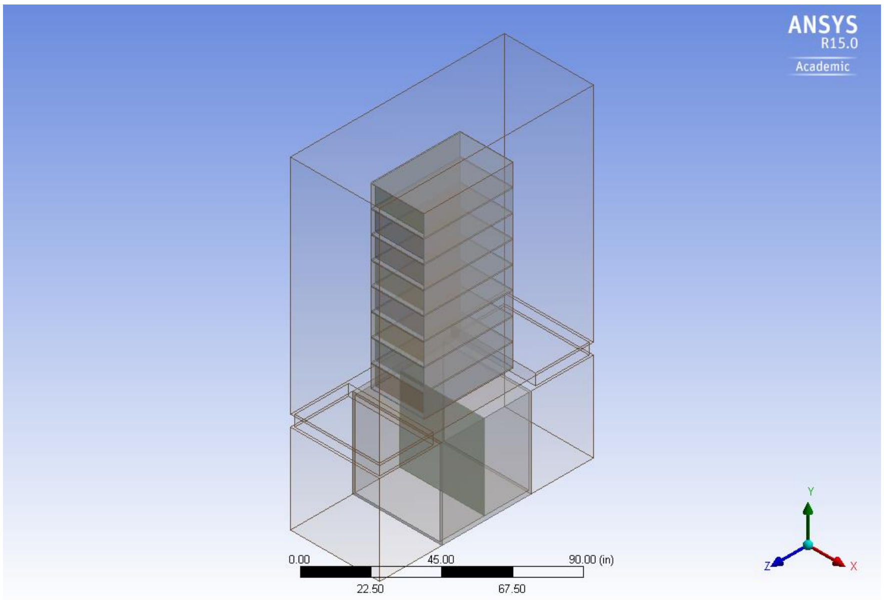

Figure 3.4: Three-Dimensional Rendering Of The HDM-1A Model Using ANSYS

3.3.2 HDM-1B

The HDM-1B is an existing model. It has the standard dimensions for the high-density data center. The dimensions are sliced to single-rack model and developed in the design modeler. The distance between the rack and the tile is maintained the length of the single tile. This value is taken from the existing IMPACT lab at Arizona State University. The tile and the walls are kept at 7 inches.

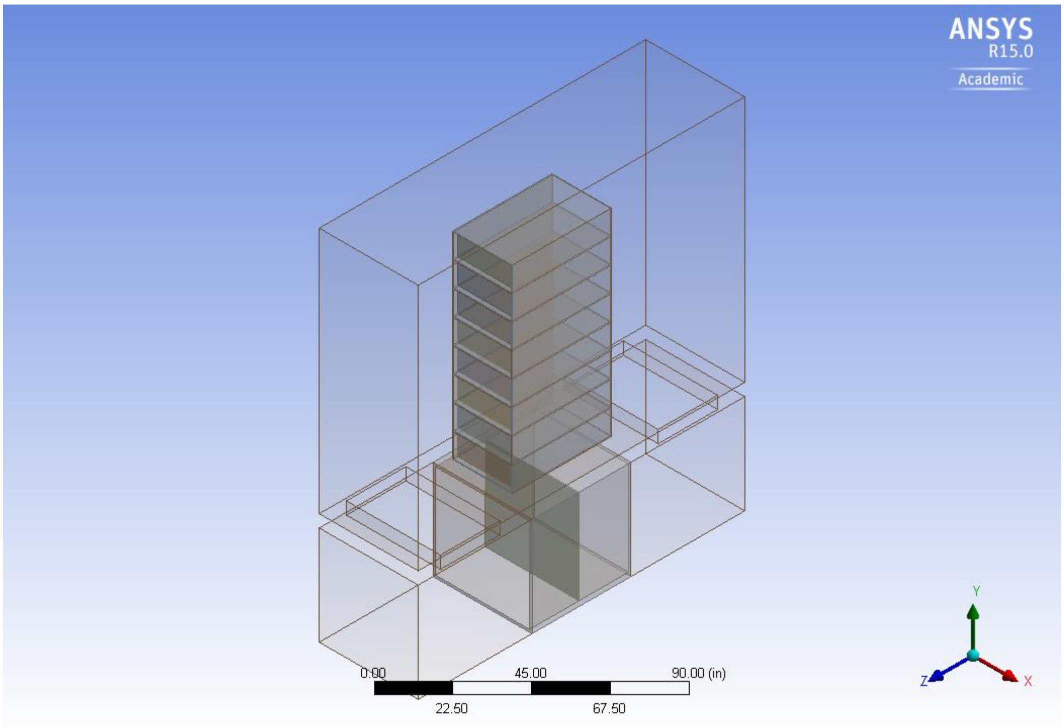

Figure 3.5: Three-Dimensional Rendering Of The HDM-1B Model Using ANSYS

During the development of the single-row models, the dimensions of the HDM-1B are kept as base reference and the other models are developed as double the size and half the size of the base model. This particular assumption will give a correlation between the size of the data center and the cooling delay parameters.

3.3.3 HDM-1C

The HDM-1C has the largest length of the data center models. It has almost double the floor space of the HDM -1B model. The dimensions of the rack and the tile are unchanged, rather the distance between the rack and the tile and the distance between the tile and the walls are increased to double the length of the floor. The distance between the rack and the tiles is kept as 36 inches and the distance between the tile and the end walls is kept as 10 inches.

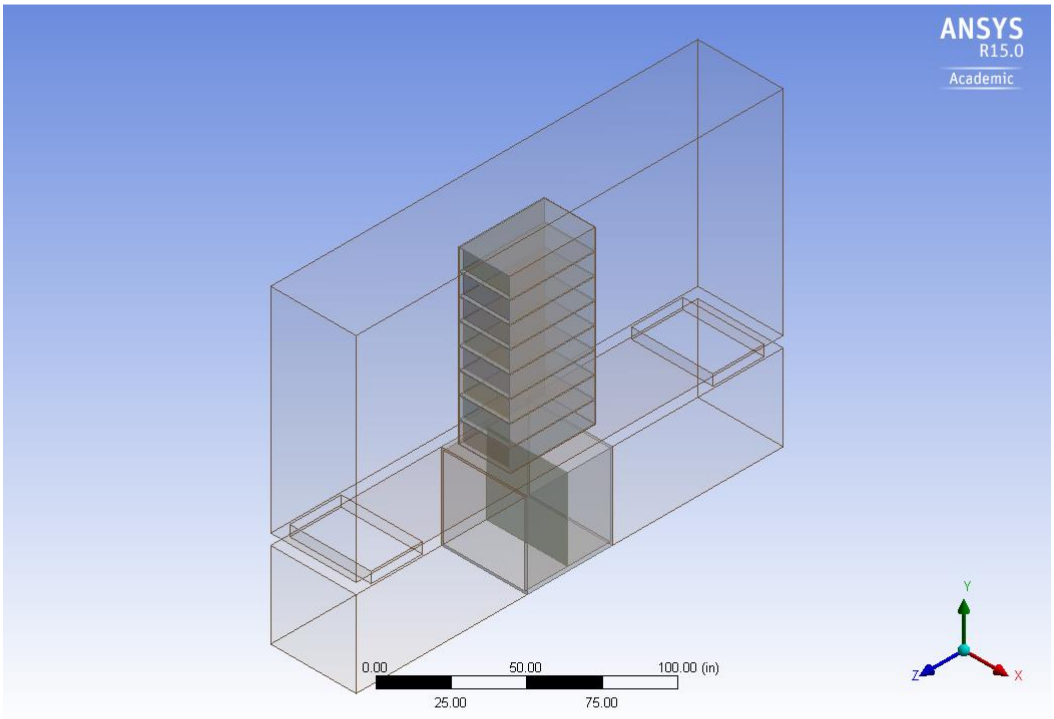

Figure 3.6: Three-Dimensional Rendering Of The HDM-1C Model Using ANSYS

3.3.4 HDM-2A

Just like the HDM-1A model, this has the least floor-space length of the double-row models, albeit the floor-space length is longer than the former. The distance between the hot-air tile and the racks is given as 10 inches and the distance between the cool-air tiles and the respective racks are given as same 10 inches. Similarly the distance between the cool-air tiles and the walls are kept as 5 inches,

Because of the small size of the data center, the air-volume is smaller and the transport of air is quick from the cooler to the racks. Between the racks, since the space is very small, the hot air gets concentrated at the middle. Hence the air handling in this model is very difficult.

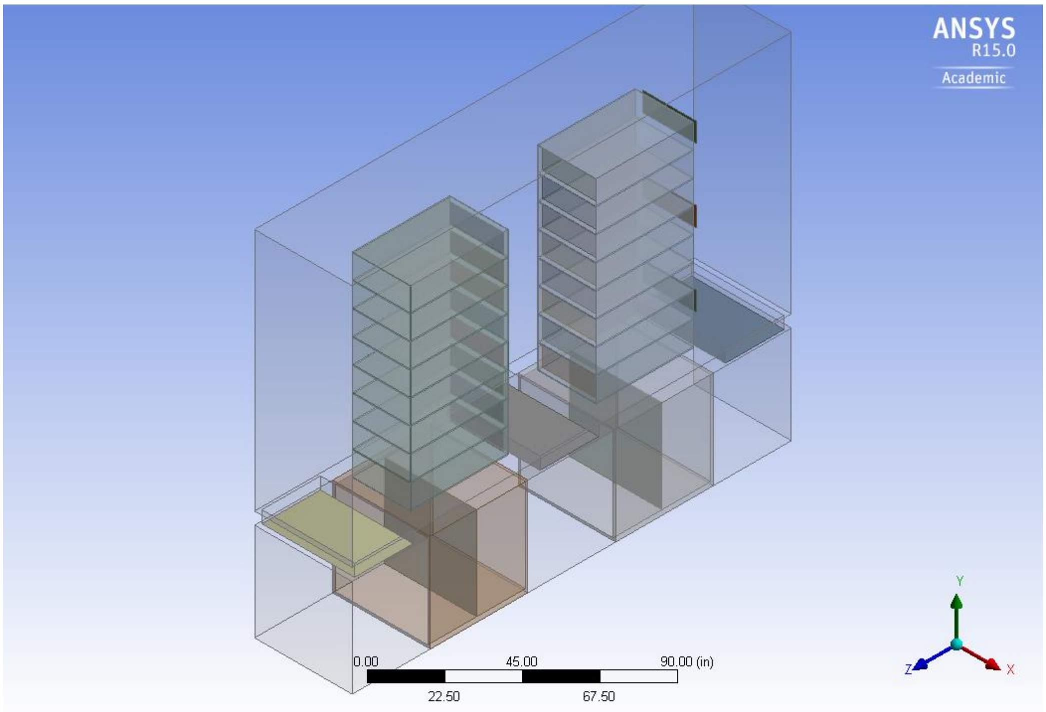

Figure 3.7: Three-Dimensional Rendering Of The HDM-2A Model Using ANSYS

3.3.5 HDM-2B

The HDM-2B model is also an existing two-row model. The dimensions were taken as it is and sliced to have the single-rack version of the original data center model. The distance between the racks and the hot-air tiles is kept as 24 inches, which is a standard two feet tile size. The distance between the racks and the respective cool-air tiles is also kept the same dimension as the two feet tile dimension. The distance between the cool-air tiles and the end of the walls is maintained as 10 inches.

Similar to the HDM-1B model, the HDM-2B model is the base model over which the other models were developed. The HDM -2A model has about half the floor-space length and the HDM -2C is about twice the floor-space length. This gives a base reference to understand the cooling delay over the room dimensions.

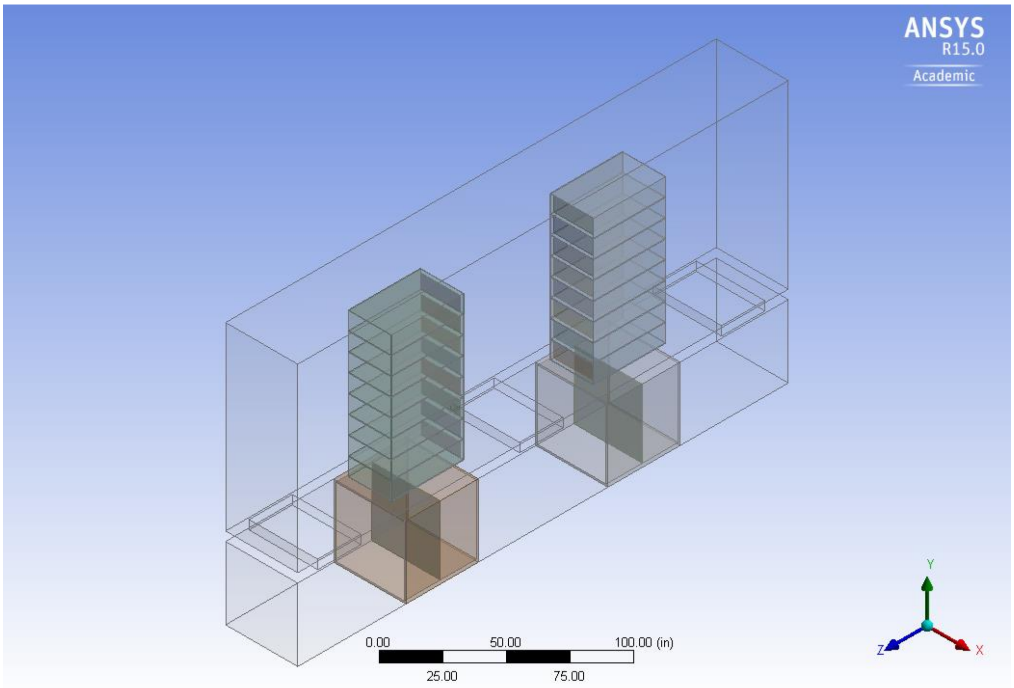

Figure 3.8: Three-Dimensional Rendering Of The HDM-2B Model Using ANSYS

3.3.6 HDM-2C

It has about twice the floor-space length of the HDM -2B model. The larger air-volume helps the cooling air and the hot air to move freely and are partially driven by the density changes of the air, which is caused by the temperature rise of the servers inside the data center.

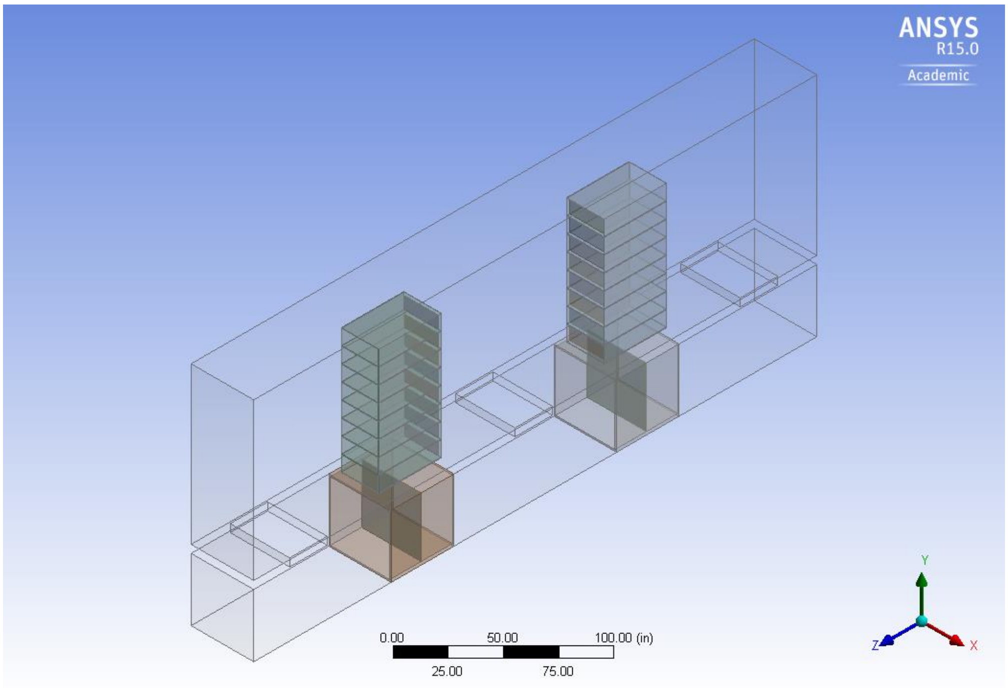

Figure 3.9: Three-Dimensional Rendering Of The HDM-2C Model Using ANSYS

3.3.7 After Treatment

After the models were developed, the solid portions of the data center, *ie* the servers and the cooler were removed from the model. It was removed to simplify the simulations and get the required results can be easily obtained. The simulation was performed to understand the cooling delay and the heat transfer of the air, the solid part are removed. The servers also produce constant heat flux for the surface and hence their boundary will remain constant. Hence removing the solid part will not affect the simulation results. Thus the model becomes simplified and the calculations become faster.

3.4 Meshing

The mesh was performed using ANSYS mesh module. It is a relatively simple mesh module with lots of easy-to-access options, which can be easier to mesh the model.

The model is meshed to have reasonable number of elements such that the mesh is not too high to complicate the calculations and not too low for solution inaccuracy. Hence all the models were meshed to have close to three hundred thousand elements. This is an optimum value and the mesh quality is verified in Fluent, which has an orthogonality feature to report poor quality meshing. Hence the mesh model was verified for good quality meshes and the mesh is saved.

The boundary conditions are an important aspect for any simulation. It is the part where the user-input parameters are defined and the simulation can be ensured that it has necessary input parameter to carry out the simulation. The rack servers, cooler walls, the server fans and the cooler fans are the boundary parameters. The boundary conditions are named so that the Fluent can identify the regions when the models are imported. Hence the boundary conditions defined are shown in the table below

Table 3.1: Boundary Conditions Of The Components Of The Data Center

Component	**Type of surface**	**Boundary name**
Server	Solid walls	Heat input
Cooler	Solid walls	Heat removal
Server fans	Fluid surface	Fan server
Cooler fans	Fluid Surface	Fan cooler

The keyword Heat and Fan are key for the Fluent what type of boundary conditions needs to be imposed on the specified region. This simplifies the user task to specify what type of boundary conditions what is the parameter that would be given as an input to the specified regions. A sample picture of the mesh model is shown in the figure below to get a picture of the nature of the mesh that has been rendered to the model. The shape of the mesh is tetrahedral and the same type of mesh is used for all the models. The advantage of this type of mesh is that it is robust in nature and the calculations are performed faster than the other type of meshes.

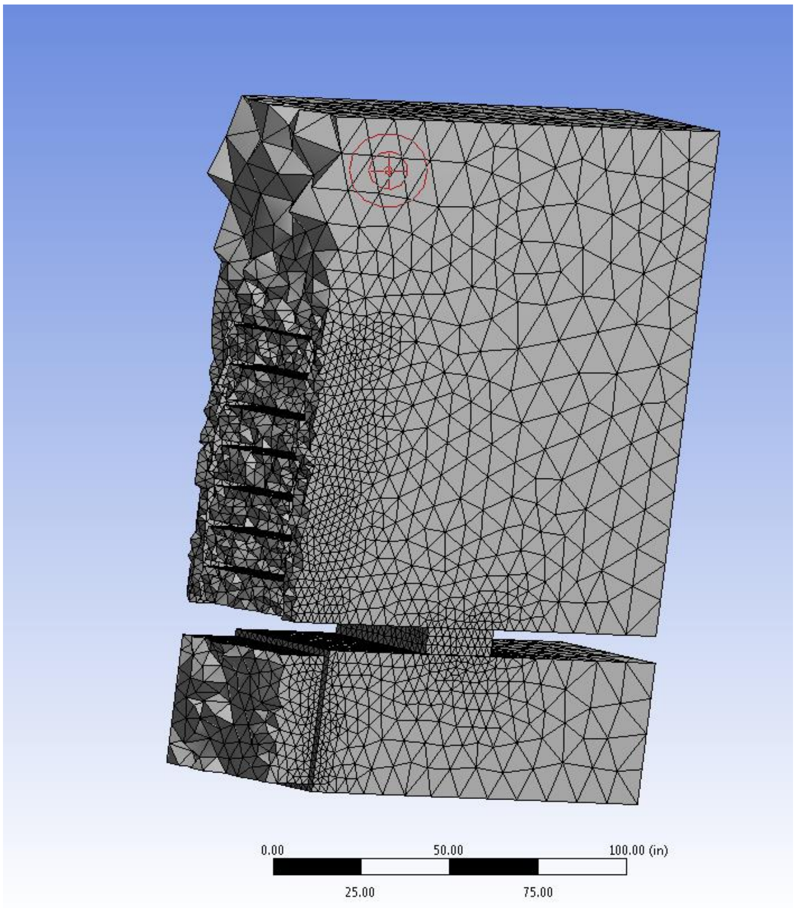

Figure 3.10: Mesh Generated For Single-Row Model

In the above figure, the model is sliced to half and one half is shown. It can be noted that the mesh density is maximum near the server walls and the cooler surface. The tetrahedral elements are programmed to be crowded near the smaller dimensional regions. In our case, the servers are packed close together and hence they have smaller dimensional values. It is also in our best interest to study the parameter variations near the server walls. Hence the mesh type is chosen and imported into the FLUENT module.

CHAPTER 4

NUMERICAL SIMULATION

CFD simulations offer some advantages of employing them over the method of physically testing the model for their validity. The physical models if they are large enough, then the cost of construction and maintenance are huge. Since there are six physical models with varying dimensions, it costs immense amount of resources and time to verify the conditions of the model physically. Hence the CFD simulation can be effectively employed to assess and derive a rough estimate of the solution. The solution of one CFD model can be verified with the existing physical model and the same conditions can be extrapolated to determine the cooling delay in other data center models. The momentum and the energy equations of the Navier-Stokes method which is the commonly used partial differential equation to solve flow and temperature problems, helps to arrive at an approximate solution easily and thus gives us an insight into the physics of the flow and the thermal variations in the data center.

Although commercial packages are available specifically to solve the data center conundrum, a fundamental fluid analysis is necessary to determine and understand the micro causes of the temperature rise and flow pattern resulting from the micro-turbulences. This will offer greater control for the user, over the simulation parameters and a deeper understanding of the interdependency of the physical parameters. Hence ANSYS Fluent is chosen because of its simplicity and user-friendly interface.

The simulation was set up and carried out using ANSYS workbench. A single rack model width wise configuration was chosen to minimize the computational resources and the simulation time. Single fluid and single-phase model was chosen to perform the simulation.

4.1 Turbulence Modeling

The flow is modeled using standard k-ε turbulence model. The Reynolds number is calculated to be more than 10^6, hence the flow is assumed to be fully turbulent. The model that is going to be simulated consists of forced convection by the cold air combined with the partial natural convection where the airflow is slightly low and the temperature gradient is sufficient enough to raise the warm air above the incoming cold air. Also the constant heat output at the server surface causes a temperature shear layer, which acts a natural temperature insulation layer against the incoming the cold air. These two properties of natural convection of the hot air and the insulation layer formed by the shear of the temperature at the server surface forms a crucial part in choosing the necessary flow model for the simulation. There are numerous turbulence flow models that are available for implementation. Out of all the turbulence models, the widely used models and those models that are available in Fluent are

- Spalart-Allmaras
- k-ε model
- k-ω model

Each model offers certain advantages and is subjected to their own adversities. Hence proper criteria are considered while considering the turbulence models, that which suits the problem statement and the offers certain advantage while carrying out the simulation.

The standard k-ε model is a system of two equation model which solves for the kinetic energy, k term and the rate of dissipation of the kinetic energy, ε term after using the additional Boussinesq term added so that the fluctuation terms can be linked to the mean flows [41]. The Boussinesq term is used to define the changes in air density to the temperature changes and hence the model is suitable for implementation as it correctly defines the natural convection term along with the flow equations. The model can account for the shear layers caused by the flow although it can fully account for near wall treatments [42]. This turbulence model is well suitable for fully turbulent flows and is widely implemented turbulence model in the industry to solve the myriad of turbulent problems because of its relative simplicity to implement and can converge at relatively lesser iterations. For these reasons it is even considered by many professionals and employed in the industries [41, 42, 43].

In our model, it is fully turbulent as it is stated earlier and there is partial natural convection to the temperature gradient change and change in air density for the same reason. There is no separated flow in the model as in case of the airfoil simulations nor there is an adverse pressure gradient in the problem statement. As a result it is wise to consider the standard k-ε model to be implemented to solve for the turbulent flow. Also since six models are to be simulated it is economical to arrive at the convergence of the steady state solution earlier so that the iterations can be completed quickly. Additionally it is computationally economical to implement so that at the minimum usage of the computational resources, the solution can be arrived. Hence the standard k-ε model is the preferred model to solve the turbulent equation over the Spalart-Allmaras and the k- ω model. To augment the accuracy of the simulation and to properly address the effect of turbulence and natural convection, the wall normal boundary conditions and the buoyancy effect options were selected.

The heat flux is given as a constant boundary condition at the server surfaces The Fans were modeled to have pressure jump and the pressure jump was calculated based on the exit velocity of air. The temperature monitoring points were placed at the inlet and exit of both the servers and the cooler. A SIMPLE model is effective in solving. While performing the transient simulation, a fixed time step of 1second was chosen.

In Fluent, the server power value is given as heat flux boundary condition at the surface of the servers. The heat flux value was calculated from the rack power value, which was described in the model specification and is shown in the Appendix. Some assumptions were made before the boundary conditions were given as inputs in Fluent. The power of the rack is the combined heat emission of the individual servers. The server walls are all heat emitting surfaces. Hence from the outer surface area of the servers, the heat flux of each server is calculated. Similar procedure was followed to calculate the heat removal flux of the coolers. During heating condition, the server heat flux value was increased to 2.5 times the initial value and operated. The boundary conditions for the server and the cooler at the steady conditions, heating conditions cooling conditions are shown below

Table 4.1: Boundary Conditions Of The Server And The Cooler For Different Operating Conditions

Operating conditions	**Server (BC: Heat Flux in W/m2)**	**Cooler (BC: Heat Removal Flux in W/m2)**
Steady	360	1240
Heating	900	1240
Responsive cooling	900	3100
Cooling	360	3100

4.2 Transient Simulation

The transient simulation was carried out using ANSYS FLUENT. The temperature was monitored at the inlet and the exit of the server and the cooler. The simulation was performed for five different conditions of server and the cooler. The conditions are steady state operating condition, continuous heating condition, responsive cooling condition, continuous cooling condition and backs to steady state operation.

I) Steady state conditions:

The server power is given in terms of heat flux which was calculated as 360 W/m^2, specified in the model description and the cooler is operated at 1240 W/m^2 which is the same as the rack power. The fan speed is adjusted for optimal airflow so as to provide maximum heat removal. Also it is continually monitored for by-pass and recirculation phenomenon, which can be dangerous for sustained data center operation. The input and output temperature of the server is monitored for about 100 seconds and observed for constant temperature plot. This ensures constant heating and cooling conditions.

II) Heating conditions:

The server heat flux was increased to 900 W/m^2, *ie.* 2.5 times its normal operating power. The temperature at the cooler inlet and the server output was monitored continuously and was allowed to rise steadily. It was carried until the set point temperature, which was kept at 313 K. Until this point, the cooler will be operated at its initial cooler power.

III) Responsive cooling:

The cooler heat removal flus were increased to 3100 W/m^2, *ie.* 2.5 times its initial operating value and the server power is kept the same. The operating conditions are maintained for approximately 100 seconds.

IV) Steady cooling:

The server heat flux was reduced to its initial operating value of 360 W/m^2 with the cooler heat removal flux maintained at 3100 W/m^2. The trend was continued until the temperature output of the servers was reduced to its normal operating conditions. After this the cooler heat removal flux was reduced and the simulation was carried out for 100 more time steps.

4.3 Fan Speed Variation

The fan speed of both the cooler and the servers are adjusted in each of the models. The size variations of each data center require the optimal airflow pattern for maximum cooling efficiency. For example the smaller data, if there is not enough cooler fan speed, the problem of hot air recirculation persists at the top deck of the rack. If another case, if the cooler fan speed is maintained high, then by-pass occurs which results in continuous heating of the lower part of the rack irrespective of increasing the cooler power.

Table 4.2: Boundary Conditions Of The Server And Cooler Fans

Models	**Server Fan Pressure Jump (in Pa)**	**Cooler Fan Pressure Jump (in Pa)**
HDM-1A	2	4.5
HDM-1B	2.5	8
HDM-1C	3.5	5.5
HDM-2A	1	6.5
HDM-2B	1	8
HDM-2C	1.5	10

Hence different fan speeds are tested for each model and the optimal values are shown below in the table. The fan speed is varied until at normal operations, a steady temperature is observed at the inlet and the outlet of both the server and the cooler is observed. This speed is maintained constant throughout the simulation.

The selection of proper monitor planes and monitor points are critical in order to assess the difference conditions inside the room like uniform air flow distribution, temperature distribution, local heat spots and the local heat recirculation zones. In our case, proper selection of monitoring plane is critical to assess the airflow distribution based on which the fan speed is determined. Hence, the three-dimensional flow is taken to consideration before selecting the monitor zones. But monitoring the 3D model is tedious, as visual inspection near the servers can be hindered by the flow parameters surrounding it. But the velocity-vector flow of the air can be visually assessed from the three-dimensional flow field. In assessing the 3D flow, it was seen that the flow didn't deviate much from ZYplane. So at any given planar flow, the air didn't deviate from this plane throughout the entire simulation. This might be because of the relatively small size of the room and high velocity of the air. This ensures that the flow direction is almost steady throughout the simulation. Hence the monitor plane was chosen normal to the flow direction. The X-plane was chosen to determine the fan speed of the air by monitoring the airflow distribution inside the room. To assess the exit temperature of the server, the monitor points are placed at 2 cm away from the server-end based on the experimental setup at the IMPACT lab.

4.4 Fan Speed Calculation For Each Model

4.4.1 HDM-1A

It is a very compact model with minimal air volume space available for heat removal from the servers. Hence maximum heat has to be removed from the server surface. Increasing the server fan pressure enables this. This pressure must also not be so higher so that a free recirculation operation is hindered. Hence an optimal value was set. Since the server fan is acting with higher pressure to remove heat, the cooler fan pressure has to be even more higher so that the cool air doesn’t gets recirculated within the lower deck. Also if the cooler pressure is low and the fan pressure is already acting at higher value, there is a high probability of hot air recirculation at the top deck. This is very dangerous as this will result in server operation failure. Hence the cooler pressure is increased until more air flows to the top of the servers. But this also causes some portion of the cool air to bypass the servers. Nevertheless, the server outlet temperature is maintained constant at steady operating conditions.

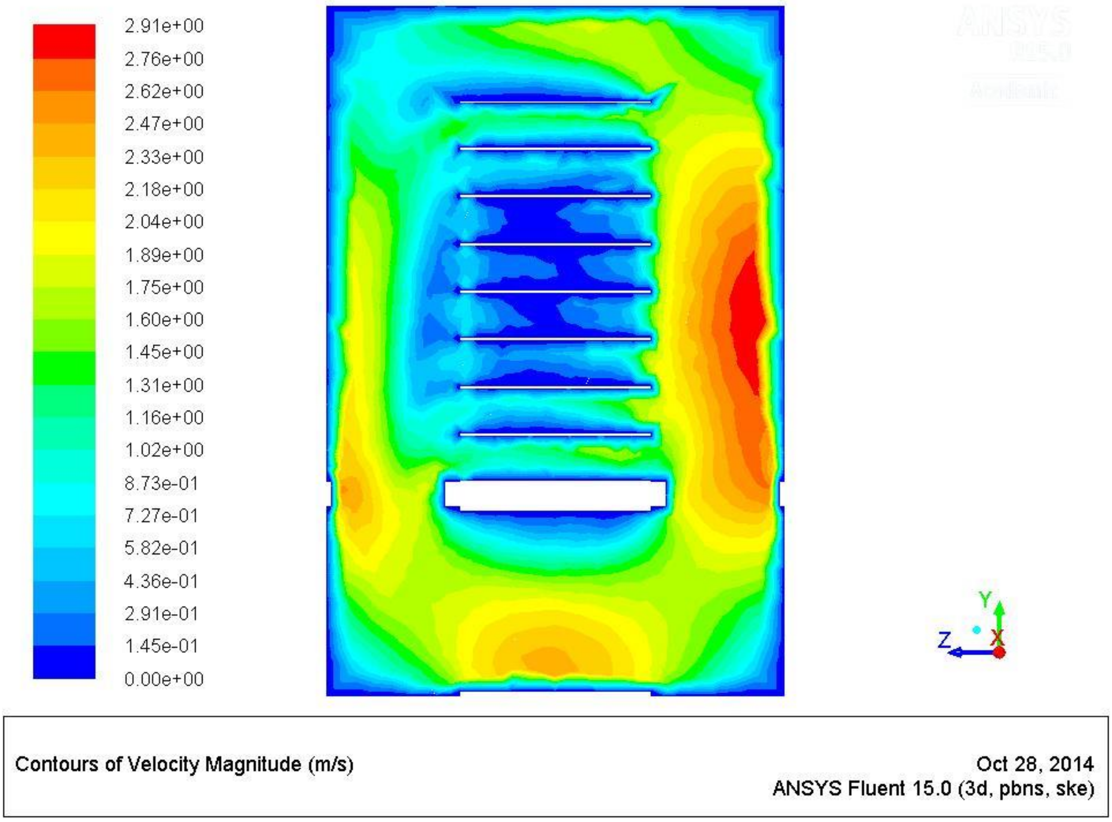

Figure 4.111: Velocity Profile Of The Recirculating Air Inside The HDM-1A Model During The Steady Operating Conditions

4.4.2 HDM-1B

It has approximately double the floor space area as that of HDM -1A. After certain trials, keeping the server fan speed constant and approximately doubling the cooler speed gives an optimal airflow circulation. In this circulation patter, the by-pass air is reduced and most of the air is concentrated towards the middle portion of the rack. This ensures a steady heating and cooling conditions at the server and the cooler level.

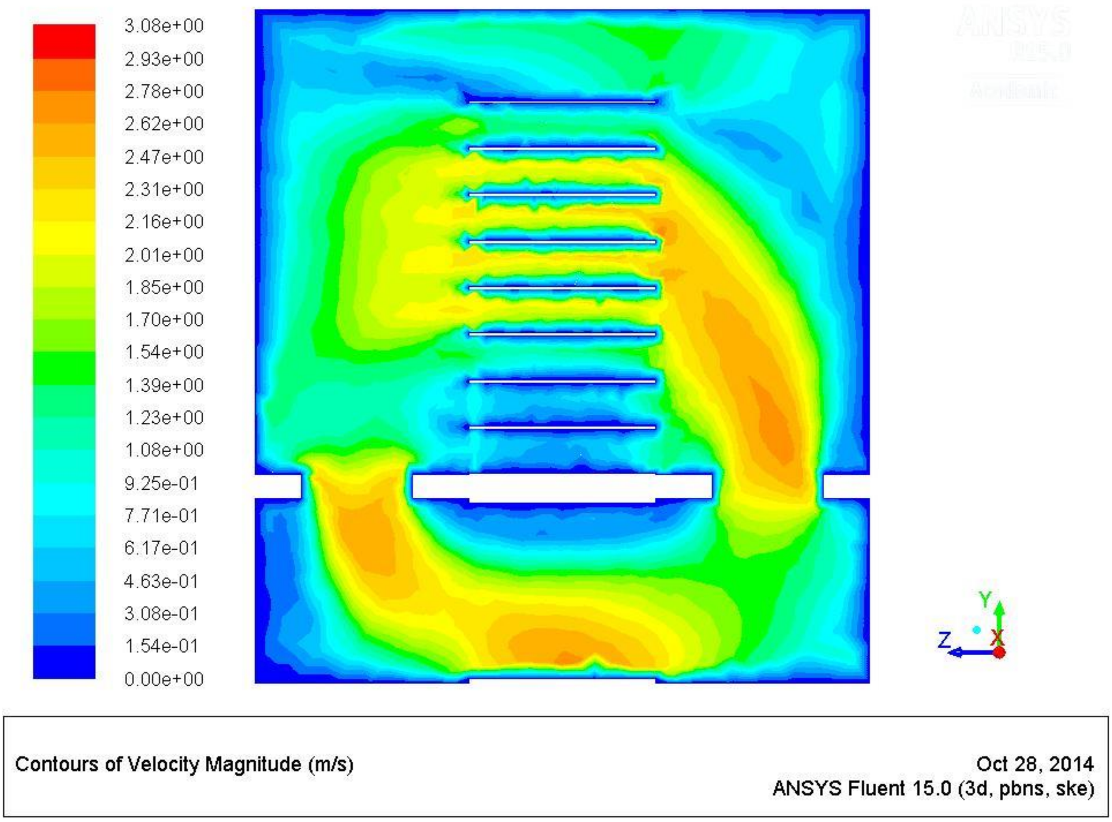

Figure 4.2: Velocity Profile Of The Recirculating Air Inside The HDM-1B Model During The Steady Operating Conditions

4.4.3 HDM-1C

Although this model is double the size of the HDM -2A, like in the previous case, doubling the cooler fan speed doesn't stabilize the server temperature. Since the data room is large, the server fans are operated at higher-pressure values to force away the hot air emitted from the server surface. The cooler air inlet is placed little farther from the rack. Hence, if the cooler fan is operated at higher pressure, most of the air is by-passed resulting insufficient cooling air to the lower servers of the rack. Hence the cooler pressure is reduced to force maximum air to the middle portion of rack.

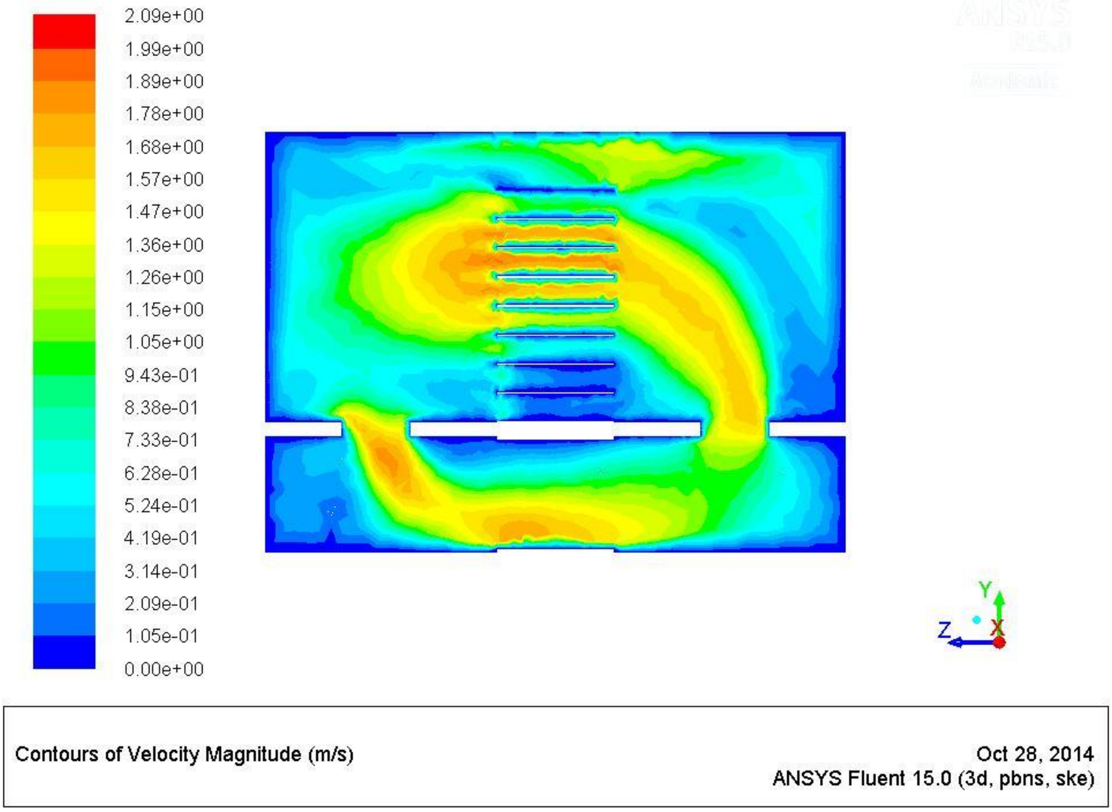

Figure 4.312: Velocity Profile Of The Recirculating Air Inside The HDM-1C Model During The Steady Operating Conditions

4.4.4 HDM-2A

Cooling of the double-aisle model is complicated, because the air at the hot aisle in the middle tends to rise up because of the buoyancy effects. Hence the server fans must operate at a pressure to force the hot air down. But it shouldn't be high enough to force the hot air from one rack to the server outlet of the second rack. Hence after long trials, the server fan speed was optimized to achieve a proper recirculation. The cooler fan pressure was also increased to ensure that there is no recirculation because of the hot air rising. Hence it is maintained at higher-pressure values than the single aisle model for the same version.

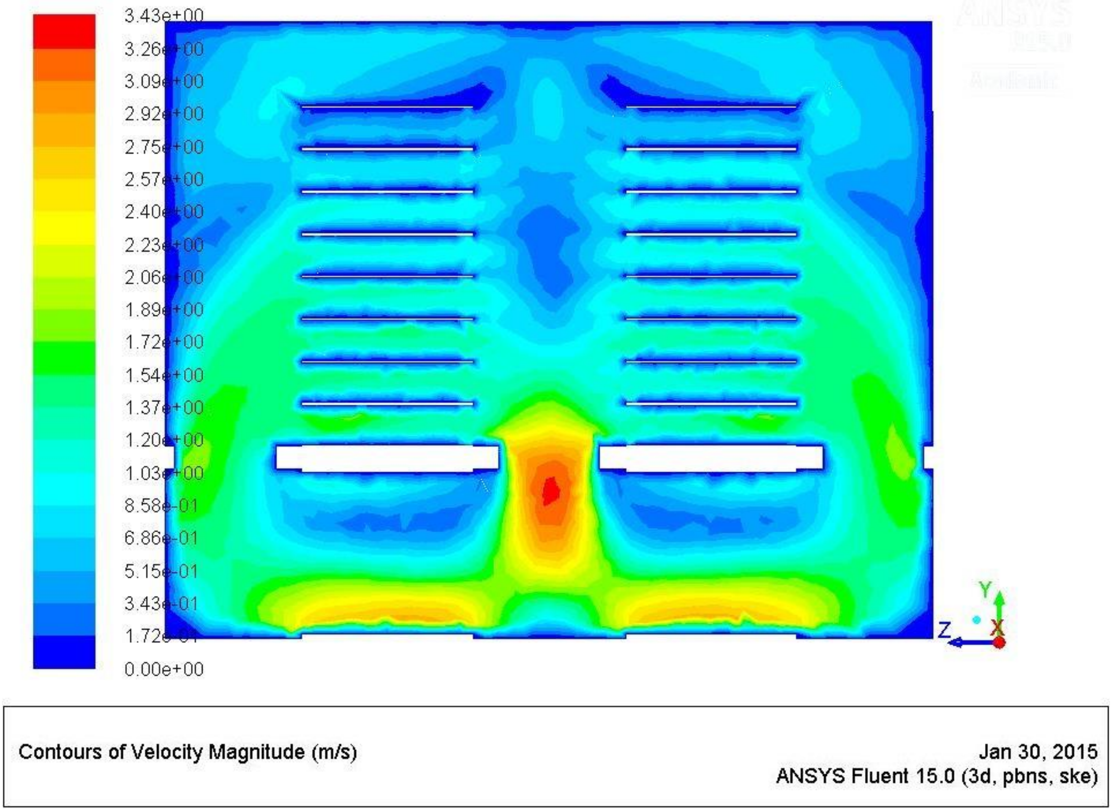

Figure 4.4: Velocity Profile Of The Recirculating Air Inside The HDM-2A Model During The Steady Operating Conditions

4.4.5 HDM-2B and HDM-2C

The server fan pressure value of the HDM-2B is maintained at same pressure value as the HDM-2A, but the cooler fan speed is increased to allow cooler air to reach for the top portion of the rack. In all the cases, the recirculation of hot air is major concern and hence the cooler is increased in linear steps as the floor space area is increased. The server fan pressure of the HDM-2C is increased little to force the hot air out because of larger space.

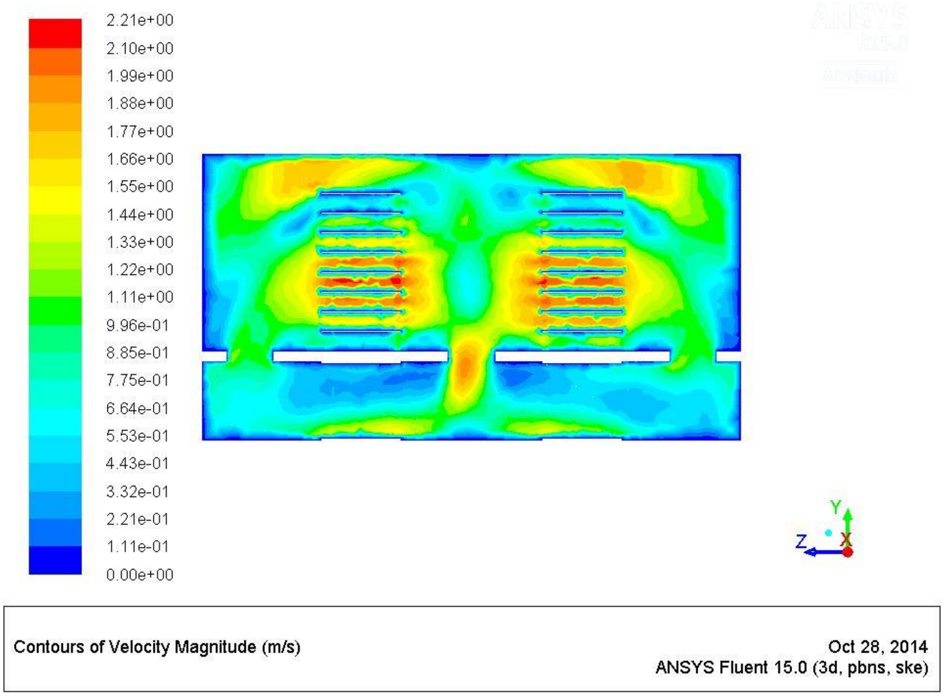

Figure 4.513: Velocity Profile Of The Recirculating Air Inside The HDM-2B Model During The Steady Operating Conditions

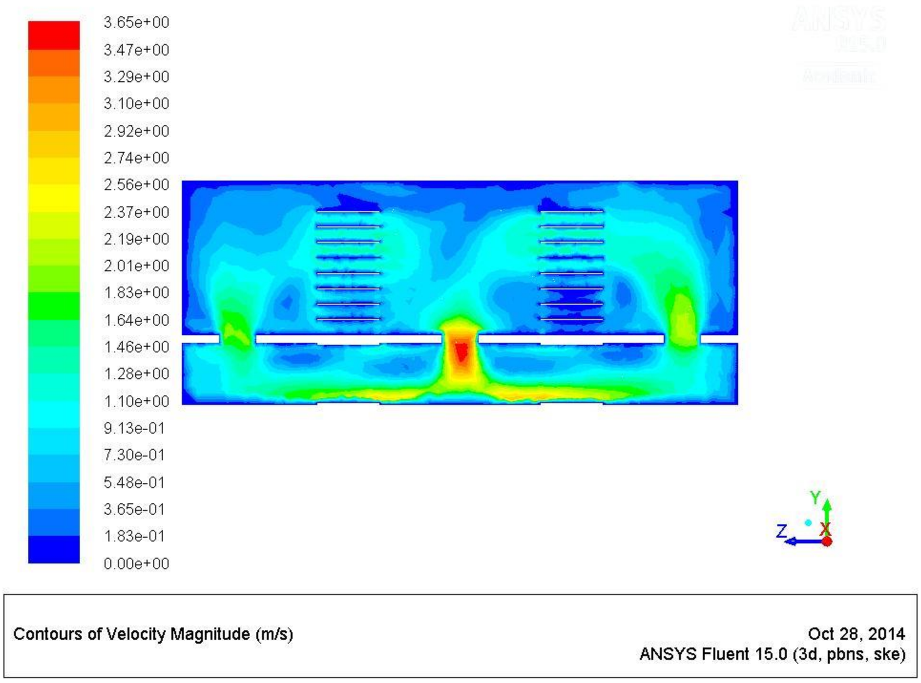

Figure 4.6: Velocity Profile Of The Recirculating Air Inside The HDM-2C Model During The Steady Operating Conditions

4.5 Temperature Monitor Points

The temperature monitor points are placed at the exit of all the servers. Technically the confined space of the server will generate more heat and it is natural to monitor the temperature at those regions. But the simulation is model is carried out to emulate the operating conditions of the IMPACT lab where the temperature sensors are placed at the server exit. So in Fluent software, the monitor points are placed at the exit of all the servers. The temperature data is collected during the steady and the transient conditions for all the servers. But while determining the cooling delay and the overload temperature values, the server temperature which had the maximum value during the constant cooling and heating conditions was chosen, considering the fact the server with maximum temperature values will require more attention while monitoring the servers. Hence depending on the airflow pattern, the temperature was monitored for different servers in each of the models.

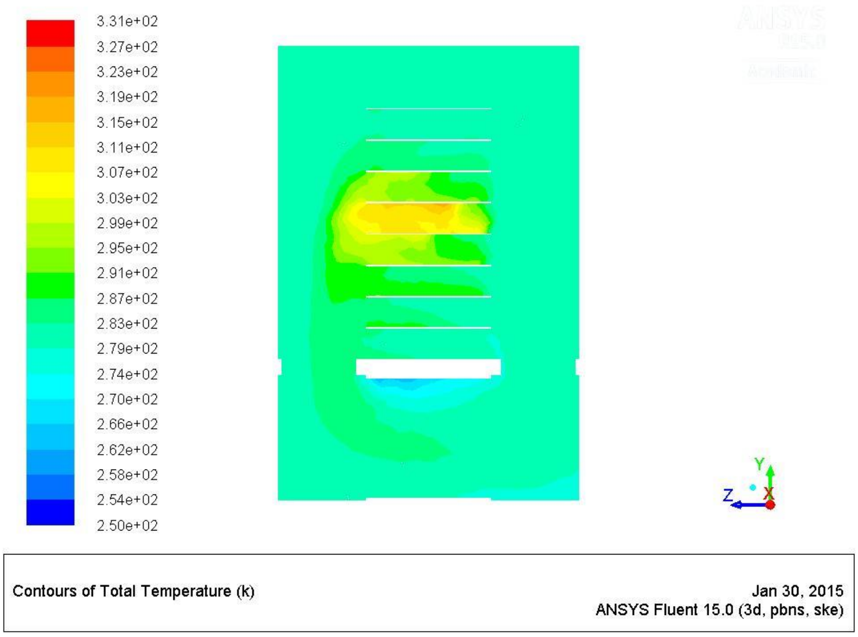

Figure 4.7: Temperature Contour Plots For HDM-1A Model

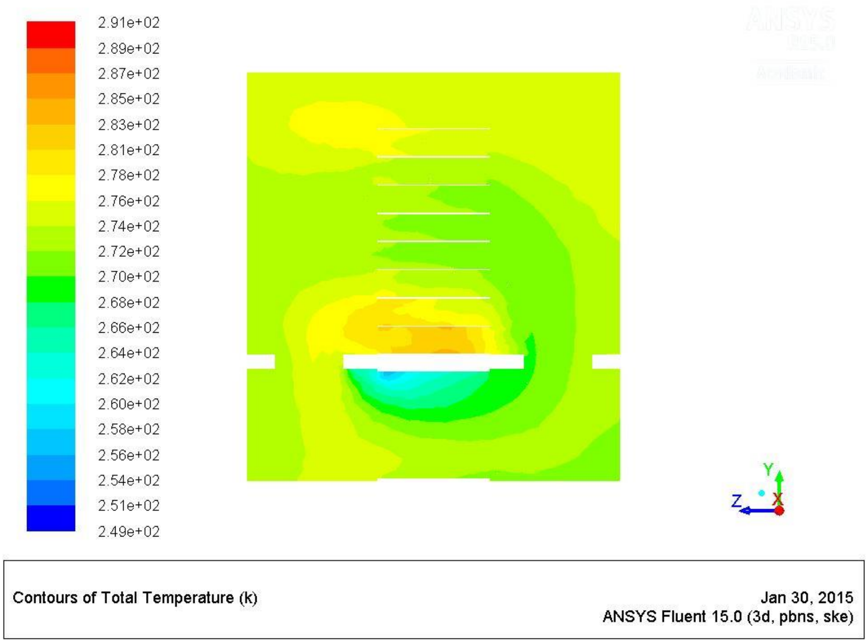

Figure 4.814: Temperature Contour Plots For HDM-1B Model

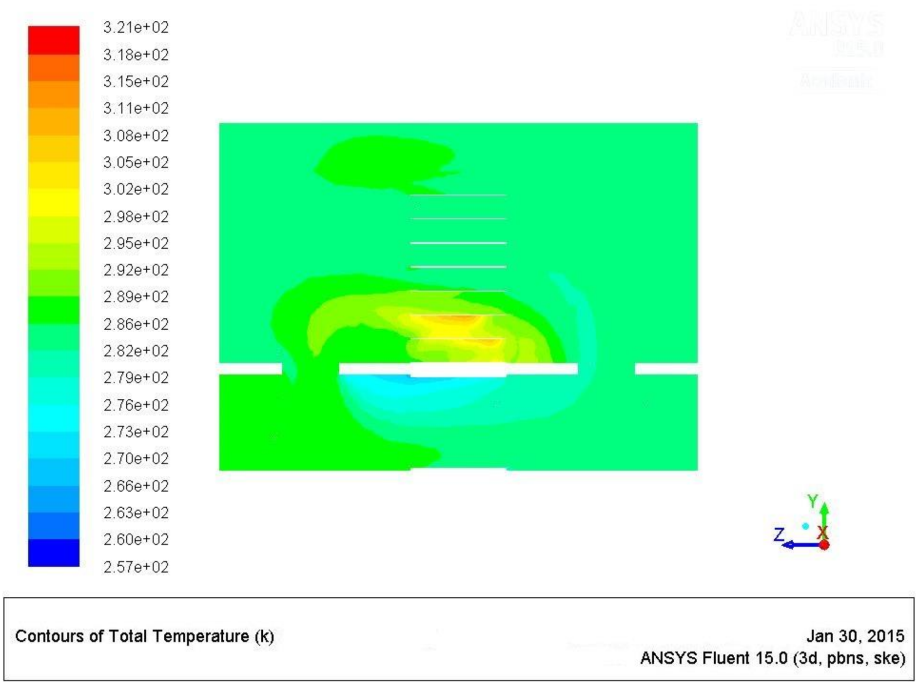

Figure 4.9: Temperature Contour Plots For HDM-1C Model

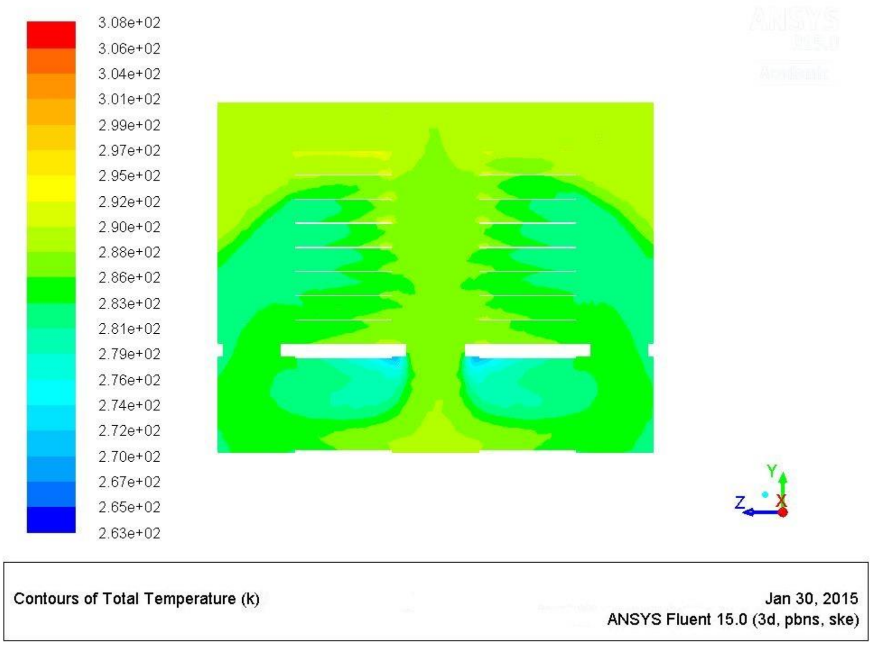

Figure 4.10: Temperature Contour Plots For HDM-2A Model

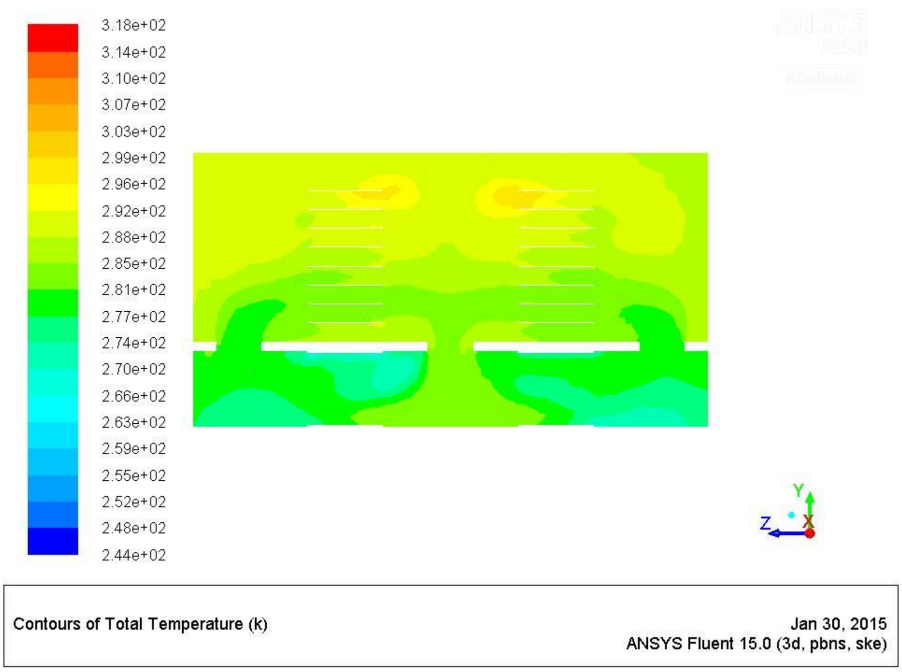

Figure 4.11: Temperature Contour Plots For HDM-2B Model

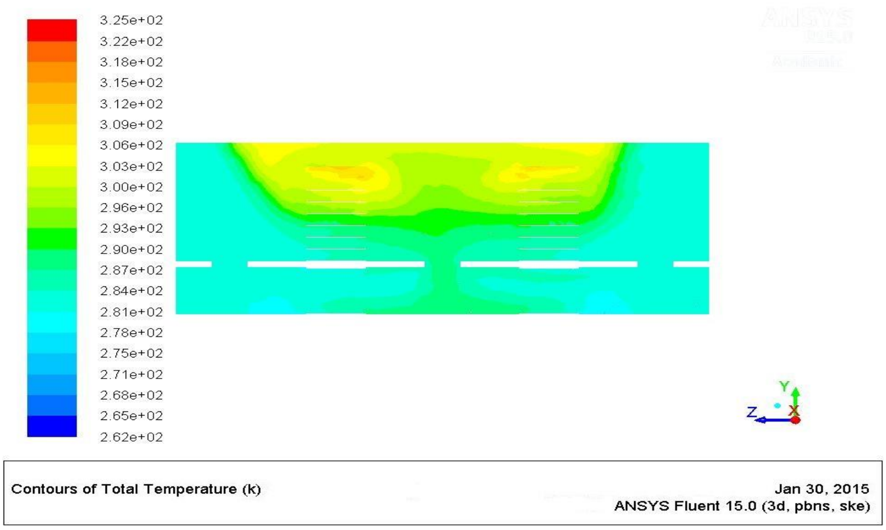

Figure 15: Temperature Contour Plots For HDM-2C Model

The temperature plots gives a general idea of the temperature distribution inside the data center. The monitor points can be set at the exit of all the servers and the one of the server exit temperature of most heated server can be represented to study the temperature changes with the changes in server power and cooler power. In HDM-1A, the most heated zone is around the middle of the rack near the server 4. While in case of HDM-1B and HDM-1C, the most heated zones are at the bottom of the racks. For all the double-row models, the top most servers of both the racks were the most heated. Therefore the monitor points are placed at the exit of the server 1 of both the racks. Thus the exit temperature data of the most heated servers are chosen and collected.

4.6 Mesh Refinement Study

In case of this CFD simulation, the meshes are created and the solution is obtained by solving the necessary Partial Differential Equation using the finite element method. There are some limitations using this method. The finer, the mesh gets, *ie*. by increasing the number of nodes in the mesh, it is hoped that the solution becomes more accurate, but the cost of obtaining such accurate results in terms of computational resources is larger.

Usually for smaller systems, the mesh is refined further and further until the results remain unchanged. At that instance, the earliest mesh with the same solution can be chosen and the simulation can be carried out. But in larger systems, the number of meshes will be much larger and the computational time will be longer. So performing mesh independent study is a tedious option. It is a decision for a computational engineer to make trade-off between the accuracy and the computational time.

In this thesis work, the mesh refinement is carried out for a single case study and it is represented in this section. The mesh elements were divided into half and the number of mesh elements in this model increased to 585,960. As expected, there were variations in the cooling delay values and the overload temperature values, but the simulation time increased by two and a half times the original simulation time. The values are compared and are represented in the table below

Table 4.3: Change In Results And Simulation Time Due To Mesh Refinement

Values	**Before Refinement**	**After Refinement**	**Change in %**
Mesh Elements	295,263	585,960	
Cooling Delay (s)	28	32	14.2
Overload Temperature (K)	1.6	1.8	15.9
Simulation Time (s)	81000	202500	149.99

From the values, it is clear that albeit there are variations in the values, the simulation time is very longer and the time will be larger with further refinement. For this refinement study, the initial model took almost 23 hours to for complete simulation and with doubling the number of mesh elements, the total run time of the simulation took around 57 hours. At this point, the trade-off needs to be done between the accuracy and the simulation time.

To determine the discretization error caused by finite time and space resolution, the Grid Convergence Index (GCI) is used to determine the error percentage. The GCI is performed in order to determine how much the numerical solution changes with the refinement and the deviation are represented in terms of error band [44]. One major advantage is that unlike other error analysis techniques, this method doesn't require actual solution to determine the discretization error. The solution obtained from the two or more refinements are computed using the approach given by Roache [45] and error band is used to determine how much the numerical solution is different from the asymptotic value. The GCI is computed using the following equation.

$$\mathrm{GCI}_{21} = \frac{F_s \left|\varepsilon\right| r^p}{\left(r^p - 1\right)}$$

Where F_s is the Factor of safety.

Ɛ is the relative error.

r is the grid refinement ratio

p is the order of convergence

The calculations are shown in the appendix. Using the above equation and the required parameters, the GCI for Overload temperature is 63.6% and the GCI for

Cooling Delay is 56.8%. Thus based on the study, it is concluded that the Overload temperature value is 1.6 K with error band of 63.6% and the cooling delay is 28 s with error band of 56.8%

CHAPTER 5

RESULTS AND DISCUSSION

After performing the simulations of heating and cooling conditions for all the models, the inlet and the outlet temperature of the most heated server is plotted against time. The cooler inlet and outlet temperature are also plotted. The temperature-time plots for HDM-1A, HDM-1B, HDM-1C, HDM-2A, HDM-2B and HDM-2C data center are shown in the figures below. The monitor points were placed at the entry point and the exit of the servers and the cooler as well.

The simulation is carried out for steady state condition, heated server condition, responsive cooling condition, steady cooling and finally normal operating conditions.

5.1 HDM-1A Model

The simulation was performed for the HDM-1A model using the given heat flux values and the determined fan speed values. The top most server had the most heat dissipation. Hence the inlet and the outlet temperature of this server was monitored. The temperature monitor points for the cooler are placed at the inlet and the outlet of the cooler openings.

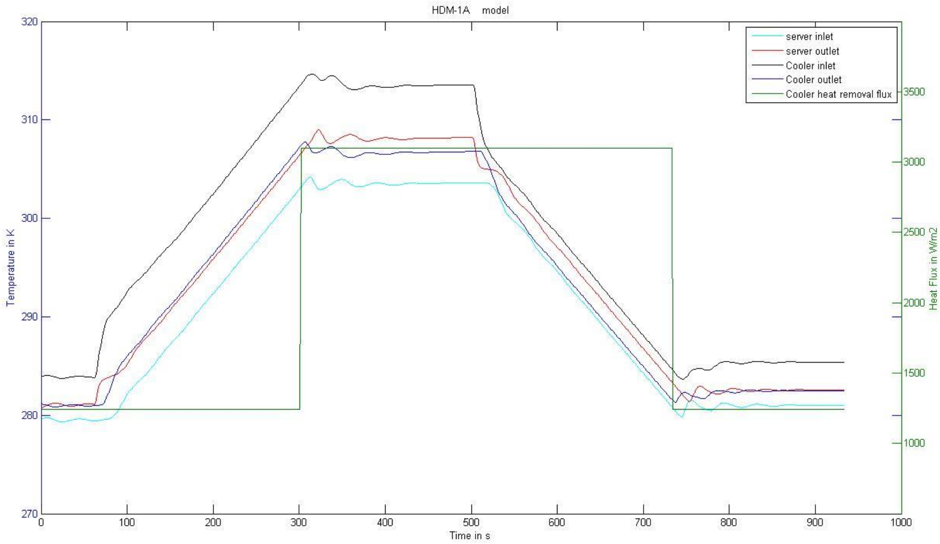

Figure 5.1: Temperature Plot Showing The Transient Temperature Cycle Of HDM-1A Model

From the plot, we can observe that the cooler inlet temperature is maximum than the top most servers. The data room is very small and hence the hot air was pushed out from the server surface by the server fans and it was removed by the combined effect of low pressure created at the cooler inlet and the high pressure from the flowing cool air. Initially, during the steady state, the temperature trend of all the components of data center remains the same. But as the heat of the server is increased, temperature at the server outlet responds immediately with a steep temperature rise. Inlet temperature of the cooler rises little later, but the temperature gradient rise is steeper than the server outlet. Again it validates our initial suggestion of combined heat at the cooler inlet. The temperature at the server inlet and the cooler outlet increases steadily. When the temperature at the cooler inlet reaches approximately 313 K, the cooler was operated with 2.5 times its initial value. The steady temperature rise observed at the cooler outlet changes and starts to cool down. The server inlet and the cooler inlet then start to cool down. The delay is observed at the server outlet. The temperature keeps on rising and the trend is observed despite the enhanced cooling of the cooler. It takes 20 seconds before the temperature rise is stopped and it stabilizes. Then the server power is brought down and the cooler operates to bring the temperature conditions back to its initial operating conditions.

5.2 HDM-1B Model

It has bigger floor-space area than the HDM-1A model. Hence the cool air is spread out over a large volume. This results in higher temperature at the top part of the rack. Hence when the server heat flux is increased, the outlet temperature of the server increases very steeply before the temperature rise gradient decreases. When the cooler power is increased to counter the server heat rise, the outlet temperature of the server doesn't increase much but the temperature rise remains for a longer period. Once the temperature stabilizes at this point, the server heat is decreased and the cooler cools down the room temperature to its normal operating range.

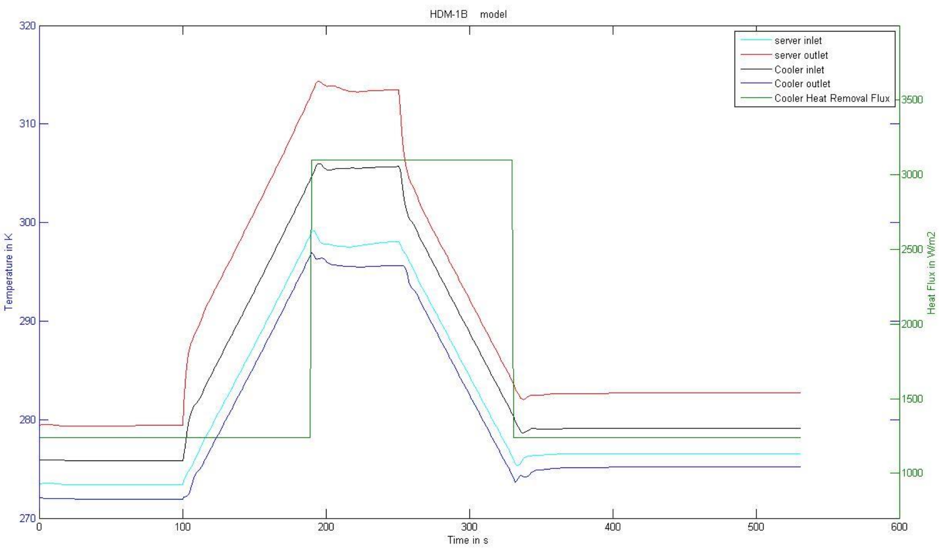

Figure 5.2: Temperature Plot Showing The Transient Temperature Cycle Of HDM-1B Model

5.3 HDM-1C Model

This version of the model is the larger among all taken case studies and has approximately double the floor space area of the HDM -1B model. The air volume inside this room is very high and the monitor points shows fluctuations in temperature reading because of the improper recirculation of the cooling air. The large volume of the room results in partial recirculation of hot air and also insufficient cooling air supply to the bottom part of the rack. Hence an optimal air circulation was difficult to achieve. The top part of the rack was monitored because recirculation is more dangerous phenomenon than the by-pass because the cooling is partially aided by the server fans.

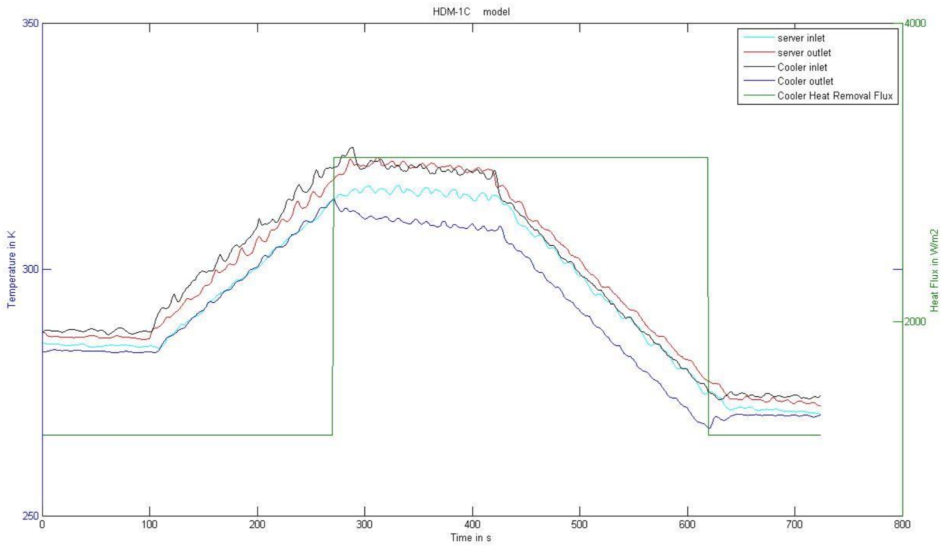

Figure 5.3: Temperature Plot Showing The Transient Temperature Cycle Of Hdm-1c Model

When the server heat is increased, there is an immediate temperature rise at the server outlet and the cooler inlet. But the temperature doesn't rise as steeply as in previous cases. This is because of the larger space. It takes longer to reach the set point temperature of 313 K. The temperature increase after the set point value is relatively low when compared against the smaller data room, but the delay to cool the servers and stabilize the temperature rise is longer.

5.4 Overload Temperature And Cooling Delay For Single-Row Models

The values of the cooling delay in terms of seconds and the overload temperature, *ie*. the increased temperature above the set-point limit is expressed in terms of Kelvin shown in the Table 5. The cooling delay values for each of the model is determined from the temperature plots of the models. The temperature keeps on rising after the cooler power is increased to counter the increased server power. The time delay taken for the responsive cooling to steady the increasing temperature is determined to be the cooling delay time. This doesn't include the downward trend of the temperature as this might be mistaken as the cool air temperature reading of the monitor point and this might lead to the error. This measurement of the time points from the start of the cooler operation to the time point where the temperature stabilizes and reads almost around the same value is considered as the cooling delay time. Then the overload temperature is also determined from the same zone of each of the plots. The overload temperature is determined as the difference between the temperature at the server exit and the highest temperature read at the server exit. This difference is taken as the overload temperature. Usually the responsive cooling is initiated after the temperature at the server exit reaches to 313 K. Technically, any temperature read greater than 313 K is considered as the overload temperature. But for our study just to determine the amount temperature overload on the servers, this particular parameter is taken to consideration so that this will be useful for future calculations on the server reliability. The determined values of the cooling delay and the overload temperature are shown in the table below.

Table 5.1: Cooling Delay And Overload Temperature Value Of The Single-Row Models

MODELS	Area (ft^2)	Power/Area (W/ft^2)	Delay time (s)	Overload Temperature (K)
HDM 1A	26.67	187.47	20	2.8
HDM 1B	36.67	187.47	28	1.6
HDM 1C	46.67	107.14	43	1.1

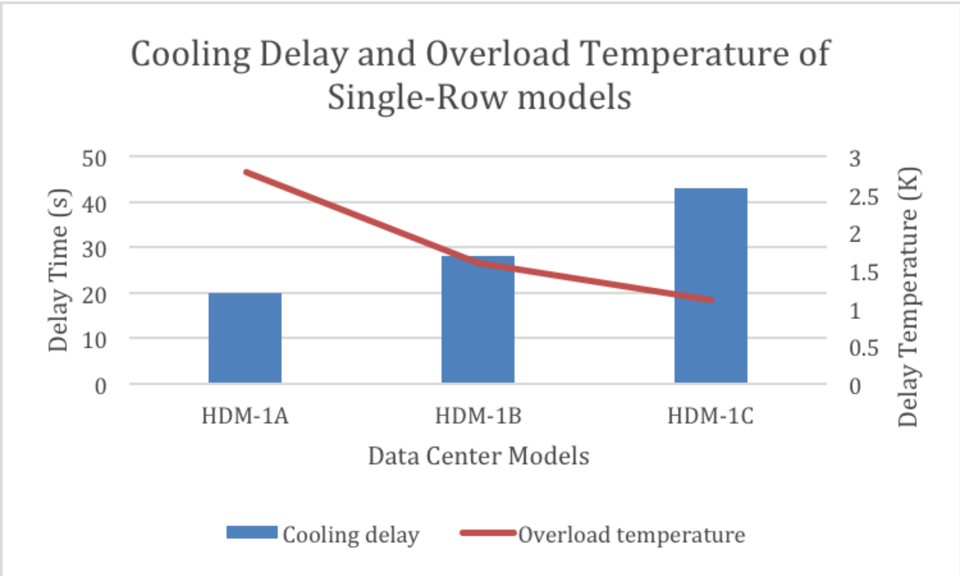

Figure 5.4: The Cooling Delay And The Overload Temperature For The Single-Row Models

The plot of the cooling delay and the overload temperature is shown in the figure. The figures are plotted from the date points using the Microsoft Excel 2013. Plotting the data gives a deeper meaning and helps us understand the trend in the time delay and temperature delay patterns with the change in the dimensions of the data center model. First we can observe that as the dimensions of the data center model is increased, there is an increase in the cooling delay time and then there is a decrease in the overload temperature values. The size of the data center model in particular the air-volume plays a major role in the determining both the values. Since the fan speed of both the server and the cooler fairly remains the same during the entire simulation, it is reasonable to assume that the air velocity plays a minimal role in the determining the values, but the recirculation pattern and amount of air per volume plays a significant and critical role in affecting these values.

For a smaller data center, the air-volume is equivalently smaller. The interacting air, *ie*. the volume of air which is always in contact with the server and the cooler surface and are subjected to temperature changes is large. Therefore the delay time will be shorter for the server temperature because of the rapid changes in the air temperature. Taking a look at the overload temperature, the temperature increase is relatively large enough. The temperature shear layer of the server air can explain this. As the incoming fresh cool air constantly removes the air, the server to the cool air can easily dissipate the heat. Hence the temperature gradient will be always large enough. Hence within the short time of the temperature delay, the server temperature increases relatively higher than the larger data center models.

The larger data center model provides contrasting results, *ie*. the cooling delay of the servers is longer and the overload temperature is smaller compared to the smaller data center models. This can also be explained in terms of air-volume. For a larger data center, the interacting air volume is comparatively lower. The larger air volume has a higher thermal capacitance and hence the thermal change of the air is low. This results in longer time for the data room air to react for the change in temperature of the servers. Hence the cooling delay is longer for the higher dimensional data center. On similar stances, since there is a larger air-volume, the non-interacting air, cools down the interacted air, which had a rise in temperature. Hence the temperature rise or the overload temperature of the server is lower than the smaller high-density data centers. Thus these inferences were made possible by observing the plot and recollecting from the physics and thermodynamics of the airflow. These inferences were verified by performing few more iterations with the different data center models and with different heating and cooling conditions.

The trend of the cooling delay and the overload temperature also follows a pattern. The pattern can be assumed to be any polynomial or exponential form. While the cooling delay shows an increasing or positive trend, the overload temperature shows a decreasing trend. It can be concluded that as the dimensions of the data center is doubled the rate of the increase of the cooling delay and the rate of decrease of the overload temperature varies exponentially or in any polynomial form, but not in a linear form in either of the cases.

5.5 HDM-2A Model

The air circulation pattern in a double row model is very complicated in general. Since the hot air forced through the bottom vent against its buoyancy effect, the fans must operate the air at higher pressure. Because of the single hot-aisle configuration, the hot air from the outlet of both the racks is mixed together creating a high pressure and high temperature zone. Hence the temperature fluctuations are high, making it very complicated to collect stable temperature readings.

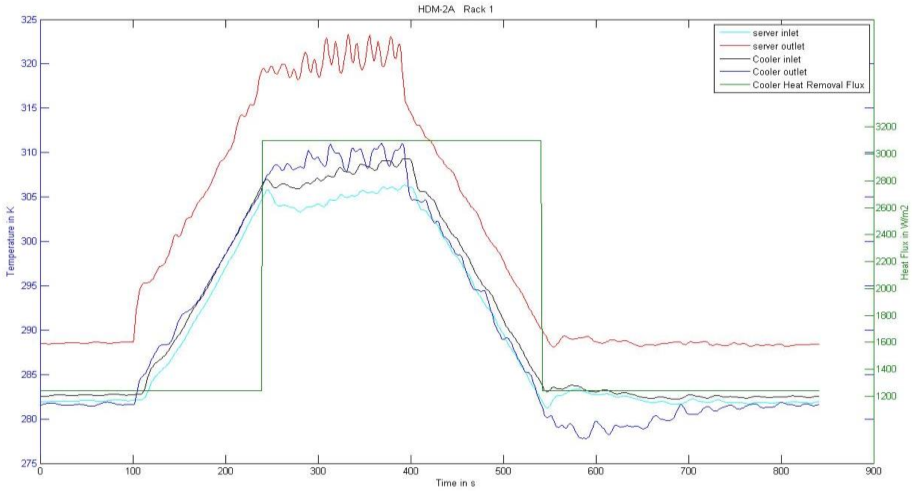

Figure 5.5: Temperature Plot Showing The Transient Temperature Cycle Of Row 1 Racks Of HDM-2A Model

The hot air, which is driven by the combined effects of fan blow pressure and server heat, is not equally distributed between the coolers. When the high pressure of the hot aisle air enters the low-pressure zone at the cooler inlet, the continuously circulating air is driven towards one of the coolers. As a result, one row of servers is cooled more than the other. Hence temperature pattern over the period varies for both the racks. . At normal operating conditions, the temperature of the cooler and the server is fairly constant, but the operating temperature greatly varies between the Rack 1 and Rack 2. In this model, we can clearly observe that rack 1 is insufficiently cooled compared to rack 2. In Rack 1, the maximum temperature is observed at the outlet of the topmost server. But in Rack 2, the maximum temperature is observed at the cooler inlet. Therefore, during the server-heating mode, outlet temperature of the server 1 is monitored for its set point temperature.

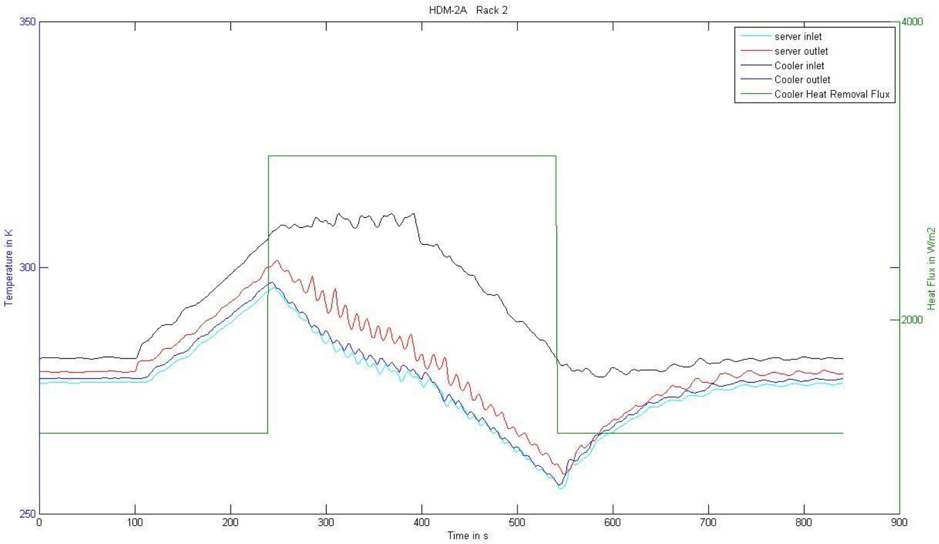

Figure 5.6: Temperature Plot Showing The Transient Temperature Cycle Of Row 2 Racks Of HDM-2A Model

When the cooler power is increased for the responsive cooling, the there is a little delay observed at the Rack 1, before its drops down a little bit and starts to stabilize. But there is no delay observed at any of the components of the Rack 2 side. When the server power is reduced, despite the sub-cooling of the Rack 2 components, the trend is maintained for the Rack 1 to reach its initial temperature conditions. This doesn't cause any operational damage to the servers because when the cooler power is reduced, the servers start to reach its initial temperature.

5.6 HDM-2B Model

The HDM-2B model has bigger floor space area than the HDM-1B model and hence the airflow is optimized so that majority of the cooling air reaches the mid-part of the racks. Unlike the HDM-2A model, the temperature variations in all the components of both the servers are almost similar. This means that the airflow is evenly distributed between the servers. A partial reason might be because of the larger room size, which allows air to flow freely subjected to less pressure variations. But because of high server heat emissions, the temperature readings are oscillatory.

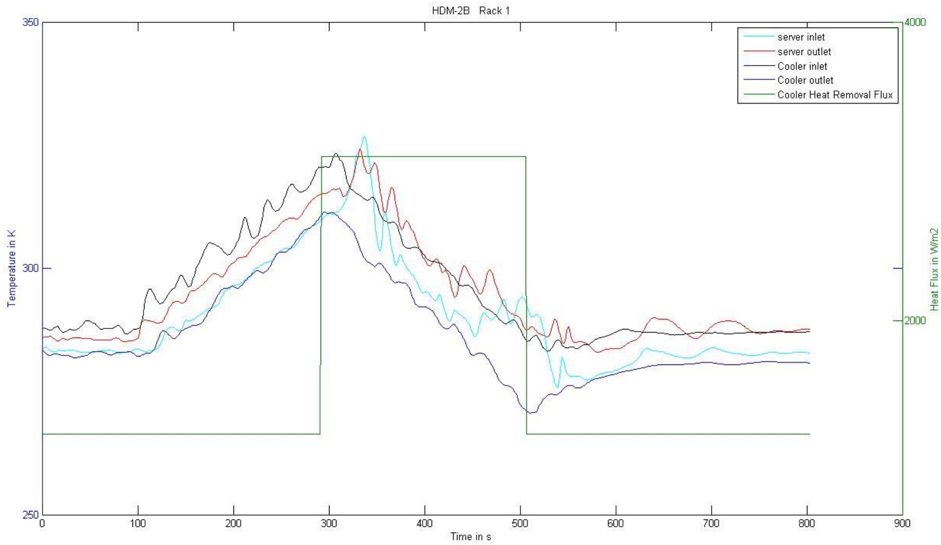

Figure 5.7: Temperature Plot Showing The Transient Temperature Cycle Of Row 1 Racks Of HDM-2B Model

During the initial steady conditions, the temperature readings are quite normal. But after the server power rise, the temperature readings are oscillatory and steadily increase with fluctuations. After the cooler power was increased to counter the server temperature rise, the server temperature increases before stabilizing. The stabilization stage was not carried out for a long time because the fluctuations are so high and they were allowed for steady cooling once the stabilization trend was observed. After the server heat flux was brought down, there is a sudden spike in temperature rise at the server inlet and outlet. The possible reason might be introduction of recirculating hot air into the top servers. Hence during the heating conditions, the hot air should have circulated within the top servers because of the buoyancy effect of the hot air. Then once the temperature reaches the initial operating conditions, the cooler heat removal flux is brought down to its initial operating conditions, during which the temperature reading stabilizes with minimal fluctuations.

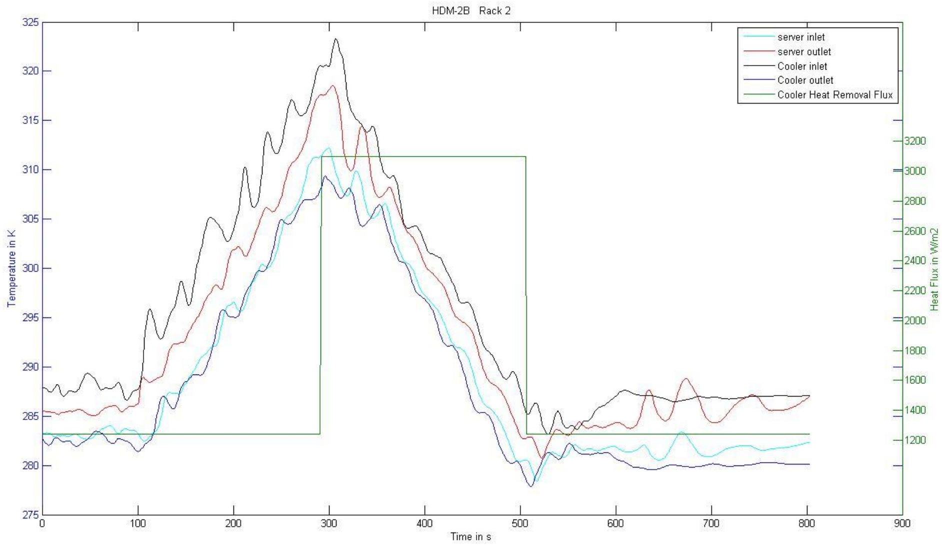

Figure 5.8: Temperature Plot Showing The Transient Temperature Cycle Of Row 2 Racks Of HDM-2B Model

5.7 HDM-2C Model

When looking at the temperature plots of the HDM-2C model, we observe that there minimum fluctuations. The operating temperature range of both the racks are of same range. Hence it can be inferred that the air flow distribution is equal between the racks. During the initial operating conditions, the temperature from the server and the cooler remains fairly the same with minimum fluctuations. But as the server power is increased, the temperature reading oscillates a lot. The time taken to reach the set point temperature is higher than the HDM-2A and HDM-2B models. Hence as the data center size increases, the response of the servers for the heating and cooling conditions is slower.

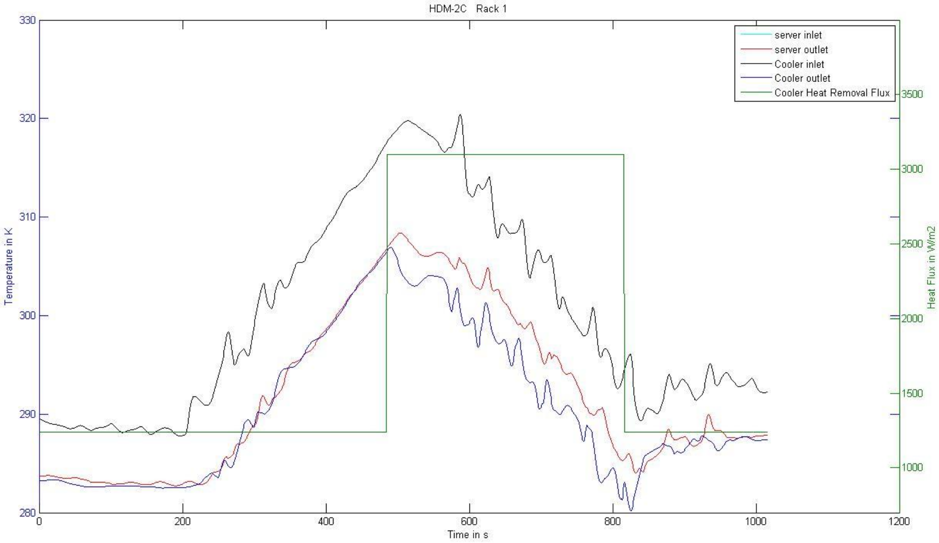

Figure 5.9: Temperature Plot Showing The Transient Temperature Cycle Of Row 1 Racks Of HDM-2C Model

The time taken to reach the set point temperature also increases linearly with the data center size increased. After the set point temperature has been reached the cooler power was increased to stabilize the temperature inside the data room. The temperature and time delay of the Rack 1 is longer and larger than Rack 2. The expelled hot air from the Rack 2 mixed with the hot air from the Rack 1, which might be read by the temperature monitor placed at the exit of server 1 of Rack 1. Hence it results in some delay between the Racks.

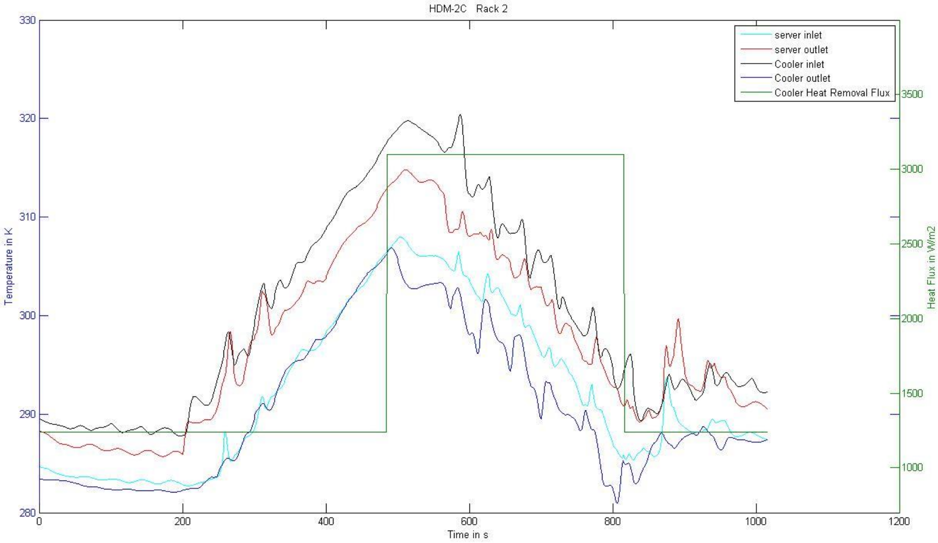

Figure5.10: Temperature Plot Showing The Transient Temperature Cycle Of Row 2 Racks Of HDM-2C Model

5.8 Overload Temperature And Cooling Delay For Double-Row Models

The cooling delay and the temperature data follows the same pattern as the single-row models. What is even more amazing is the pattern between the racks of the same data center model also follows the same pattern as the smaller to the larger data center models. The longer the cooling delay, the smaller is the overload temperature. The cooling delay between the racks of the same data center model is differs by smaller value and it becomes larger as the size of the data center increases. Similarly the contrasting pattern is seen in case of the overload temperature. The difference in overload temperature between the racks of the same data center model is large for smaller data center models and the difference becomes smaller as the data center size increases. The same explanation as given to the single-row models can be suitable to the double-row models too. As the size of the data center model increases, the cooling delay increases and the overload temperature decreases.

Table 5.2: Cooling Delay And Overload Temperature Value Of The Double-Row Models

Models	Area (ft^2)	Power/ Area (W/ft^2)	Racks	Time delay (s)	Temperature delay (K)
HDM-2A	47.78	104.63	Rack 1	25	3.82
			Rack 2	26	3.45
HDM-2B	63.89	78.26	Rack 1	38	3.09
			Rack 2	42	2.98
HDM-2C	80	62.5	Rack 1	59	2.76
			Rack 2	66	2.01

The inequality in the cooling delay and the overloaded temperature of the racks of the same data center model is due to the slight unbalanced distribution of the cool air to the servers from the cooler. This unbalance is extrapolated as the size of the data center increases. The initial unbalance is caused by the pressure difference of the recirculation air flowing from the server exit to the inlet of the cooler. Since the fans are operating at constant output speed and the cooler inlet vent is the same for both the coolers, which is relatively small, there is a small pressure difference created a few inches down the inlet of the cooler vent. This pressure difference is sufficient enough to create a difference in mass flow rate of the recirculation air between the coolers. As the fans of the cooler operate at the same speed throughout the simulation, the unbalance in flow is maintained throughout the entire process and this is the primary reason for the difference between the racks of the same data center model to have a difference in cooling delay and overload temperature values.

The cooling delay and the overload temperature for the Rack 1 of each of the models is shown in Figure 5.11: Cooling Delay And Overload Temperature Of Row 1 Of Double-Row Models below

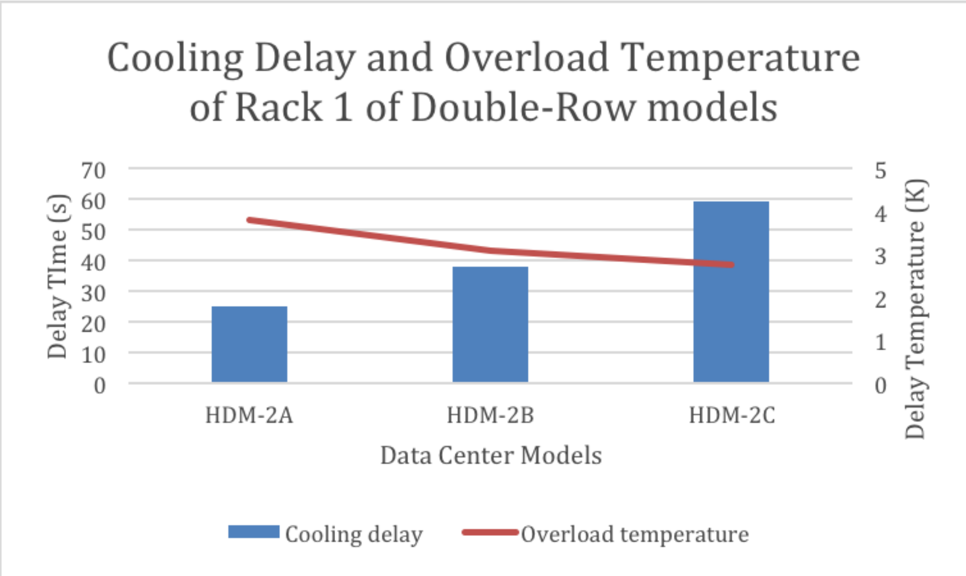

Figure 5.11: Cooling Delay And Overload Temperature Of Row 1 Of Double-Row Models

From the Figure 5.11: Cooling Delay And Overload Temperature Of Row 1 Of Double-Row Models we can observe that the cooling delay and the overload temperature of Rack 1 of the double-row models follows the same pattern as the single-row models. The cooling delay increases as the data center size doubles and the overload temperature decreases with the size. But when compared to the values of the single-row models, the values of the double-row models both in terms of the cooling delay and the overload temperature is higher. This can be attributed to the size of the data center and the heat density of the recirculation air respectively. The bigger size of the data center changes the air temperature slowly, but the server exit of both the rows face each other and hence the temperature values read by the monitor points are higher. The higher heat densities of the data center especially at the server exit makes the hot expelled air interact at the monitor points and hence reads the overload temperature value higher than the single-row models.

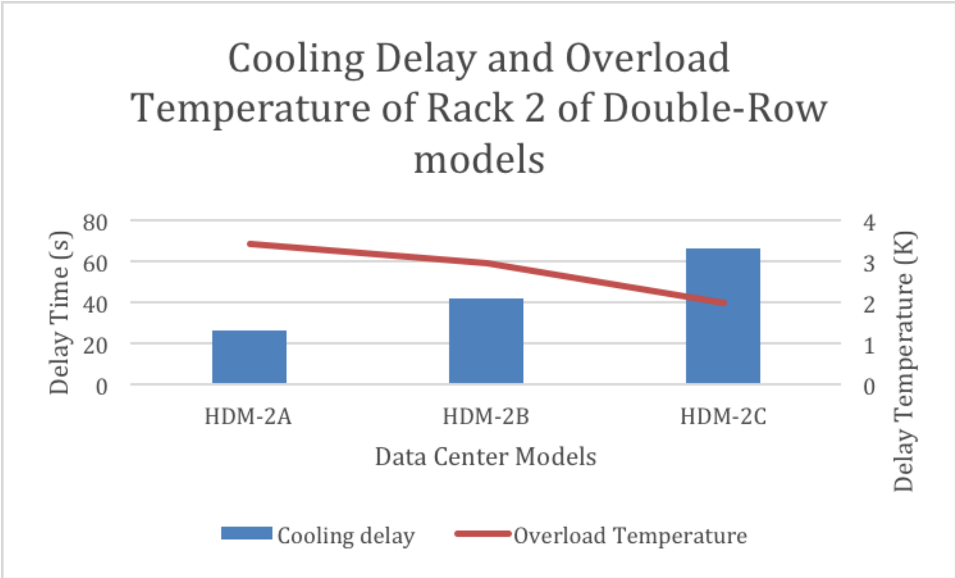

Figure 5.1216: Cooling Delay And Overload Temperature Of Rack 2 Of Double-Row Models

Comparing the figures we infer different observation. Albeit the overload temperature decreases for the both the Rack 1 and Rack 2, the pattern of downtrend differs in both the cases. The rate of decrease of overload temperature is high initially and then slows down in case of the former and the overload temperature rate is slower initially and then it decreases faster as the size of the data center increases. Any of the factors such as air recirculation pattern, pressure difference and the heat density of the data center can cause this change in decrease pattern.

CHAPTER 6

CONCLUSION

In this work, we established a relationship between the cooling delay, overload temperature and the dimensions of the data center model. Total of six high-density data center models were modeled using ANSYS Workbench with three single-row models and three double-row models. These models were designed based on increasing dimensions with each model having twice the dimension width wise, while the length and height of the data center remains same in all the models. The simulation was carried out using ANSYS Fluent in five stages, steady state condition, heating condition, responsive cooling, steady cooling and back again to steady state operation. The temperature was measured at the inlet and the outlet of all the servers and also similarly measured at the inlet and the outlet of the cooler vent. The cooling delay and the overload temperature for each of the models was determined from the transient temperature plots. These values are plotted against the dimensions of the models.

The plots give us some understanding of the cooling delay and the overload temperature and how these parameters trend with the increasing dimensions of the high-density data center. The cooling delay for each of the models increases as the dimensions of the data center increases but the overload temperature for each of the models decreases as the dimension of the data center increases. This is applicable to both the single-row and double-row models. Similarly matching the similar racks of all the models of the double-row model also shows a similar trend.

The cooling delay and the overload temperature of the models is greatly affected by the thermodynamic properties of the air. In single-row data center models, the increase in cooling delay of the models with the increase in dimensions the model is caused by the amount of air-volume inside the room. Air can assumed as slow responsive fluid and hence larger the volume of the air, the slower the response of the air to the overall change of air temperature within the data center. Hence larger volume of air causes slower response to the changes in server heat dissipation. Therefore larger data center have relatively higher cooling delay values compared to the smaller data center models.

In case of the double-row data center models, the cooling delay increases with the increases in dimension of the data center model. The values of the cooling delay are also slightly higher than the single-row counterparts. This is because of the higher volume of the air inside the data center room. Therefore the air responds even more slowly compared to the single-row models and hence higher cooling delay values. The difference in cooling delay values between the racks of the same model is caused by the uneven distribution of air. This is because of the pressure difference created at the cooler inlet vent created by the cooler fan operation.

The overload temperature, which is the difference in temperature of the air at the start of the responsive cooling and the highest temperature read by the monitor point during the responsive cooling, decreases as the size of the data center increases. This is also proportional to the air-volume inside the data center. Larger the air-volume, slower is the thermal response of the air inside the room and hence the smaller the temperature rise of the air. Hence smaller data center have higher overload temperature values than the larger data center.

In case of the double-row models, the overload temperature values are larger than the single-row models of the same dimensions. Even though the cooling delay is longer than their counterparts, the overload temperature is also higher because of the interacting hot air from the server exit of both racks. As a result, the overall temperature of the air in the hot aisle increases and the hence the overload temperature value increases.

But the cooling delay values and the overload temperature values follows the same pattern as the single-row models, which the cooling delay becomes longer as the dimension increases and the overload temperature value decreases with the increase in data center dimension.

In both the cases of cooling delay and the overload temperature, they follow a polynomial or exponential form. In case of cooling delay it is a positive form and in case of overload temperature, it takes a negative form.

CHAPTER 7

FUTURE WORK

The work has been performed three variations of the dimensional change. The simulation can be carried out for few more dimensional changes in order to accurately determine how the trend varies, as in whether the trend varies in parabolic or quadratic or exponential form. Thus the collection of few more data by performing the more simulation on few more dimensional variation of the data center will enable us to predict which form of the equation, the cooling delay and the overload temperature undertakes and thus a proper equation can be derived. Thus this equation will serve as a basic foundation in designing the future data center models.

The cooling delay and the overload temperature can be a useful parameter in determining the server failure rate. The server failure rate as stated earlier can be found using the Arrhenius equation. The time parameter and the temperature values can be used as the essential data to determine the server failure rate and thus the reliability of the servers and the data center maintenance cost can be estimated. This can be used as an essential element in server work allocation and hence proper thermal management of the data center.

In all our data center models, the servers are operated at same conditions, *ie*. they all operate at the same increased power consumption or at the normal operation state. Practically this will not be the case in an actual working data center. Some of the servers will be in high power consumption mode and some will be in lower power state and even few more will be operating at full workload conditions. Hence simulations can be performed emulating the real time conditions and determine the cooling delay and the overload temperature values and can compare against the determined values of this simulation. Albeit this is a large work, it can give us realistic values and can be useful for the data center designing.

REFERENCES

[1] "Microsoft Kicks Off $350M Data Center Expansion in Virginia | Data Center Knowledge." [Online]. Available: http://www.datacenterknowledge.com/archives/2014/06/13/microsoft-kicks-350m-data-center-expansion-virginia/. [Accessed: 30-Oct-2014].

[2] "Facebook DataCenter, Servers and Infrastructure FAQ." [Online]. Available: http://www.datacenterknowledge.com/the-facebook-data-center-faq/. [Accessed: 31-Oct-2014].

[3] "Google Data Center FAQ, Part 2 | Data Center Knowledge." [Online]. Available: http://www.datacenterknowledge.com/google-data-center-faq-part-2/. [Accessed: 31-Oct-2014].

[4] "6 Models Of The Modern Data Center - InformationWeek." [Online]. Available: http://www.informationweek.com/big-data/hardware-architectures/6-models-of-the-modern-data-center/d/d-id/1269326. [Accessed: 31-Oct-2014].

[5] "What Does High Density Mean Today?" 08-Nov-2011.

[6] N. Rasmussen and V. Avelar, "Deploying High-Density Pods in a Low-Density Data Center."

[7] L. Wang, S. U. Khan, and J. Dayal, "Thermal aware workload placement with task-temperature profiles in a data center," J. Supercomput., vol. 61, no. 3, pp. 780–803, Jun. 2011.

[8] "IDC: Data center power, space and cooling problems disrupt enterprises - FierceEnterpriseCommunications." [Online]. Available: http://www.fierceenterprisecommunications.com/story/idc-data-center-power-space-and-cooling-problems-disrupt-enterprises/2012-11-15. [Accessed: 31-Oct-2014].

[9] E. Overview, "Facilities Design for High density Data Centers," no. January, 2012.

[10] P. D. Nrdc and J. W. Anthesis, "Data Center Efficiency Assessment Scaling Up Energy Efficiency Across the Data Center Industry : Evaluating Key Drivers and Barriers," no. August, 2014.

[11] M. C. Facilities, D. Centers, T. Spaces, and E. Equipment, "Data Center Networking Equipment – Issues and Best Practices Table of Contents Introduction."

[12] R. Sahoo, "Failure data analysis of a large-scale heterogeneous server environment," *... Networks, 2004 ...*, pp. 1–10, 2004.

[13] P. Gill, N. Jain, and N. Nagappan, “Understanding network failures in data centers: measurement, analysis, and implications,” *ACM SIGCOMM Comput. Commun. …*, 2011.

[14] “Failure Rates in Google Data Centers | Data Center Knowledge.” [Online]. Available: http://www.datacenterknowledge.com/archives/2008/05/30/failure-rates-in-google-data-centers/. [Accessed: 30-Nov-2014].

[15] J. Choi and Y. Kim, “Modeling and managing thermal profiles of rack-mounted servers with thermostat,” *… , 2007. HPCA 2007. …*, pp. 205–215, 2007.

[16] R. Somegawa, “The Effects of Server Placement and Server Selection for Internet Services.”

[17] D. Yang, X. Fang, and G. Xue, “ESPN: Efficient server placement in probabilistic networks with budget constraint,” *2011 Proc. IEEE INFOCOM*, pp. 1269–1277, Apr. 2011.

[18] I. Rodero, E. K. Lee, D. Pompili, M. Parashar, M. Gamell, and R. J. Figueiredo, “Towards energy-efficient reactive thermal management in instrumented datacenters,” *2010 11th IEEE/ACM Int. Conf. Grid Comput.*, pp. 321–328, Oct. 2010.

[19] E. K. Lee, I. Kulkarni, D. Pompili, and M. Parashar, *Proactive thermal management in green datacenters*, vol. 60, no. 2. 2010, pp. 165–195.

[20] S. Microsystems and S. Diego, “Proactive Temperature Management in MPSoCs.”

[21] S. Sankar, M. Shaw, K. Vaid, and S. Gurumurthi, “Datacenter Scale Evaluation of the Impact of Temperature on Hard Disk Drive Failures ,” vol. 1, no. 212, 2010.

[22] N. M. S. Hassan, M. M. K. Khan, and M. G. Rasul, “Temperature Monitoring and CFD Analysis of Data Centre,” *Procedia Eng.*, vol. 56, pp. 551–559, 2013.

[23] J. Cho and B. Kim, “Evaluation of air management system’s thermal performance for superior cooling efficiency in high-density data centers,” *Energy Build.*, vol. 43, no. 9, pp. 2145–2155, Sep. 2011.

[24] “Computational Fluid Dynamics Modeling for Operational Data Centers.”

[25] “Data Center CFD Modeling vs. Real Time Monitoring.” [Online]. Available: http://www.42u.com/measurement/CFD-modeling.htm. [Accessed: 04-Dec-2014].

[26] P. Services, “Statement Of Work Professional Services Data Center Cooling Analysis Using Computational Fluid Dynamics 1. 0 Executive Summary Data Center Cooling Analysis using CFD.”

[27] D. Moss, “Data center operating temperature: the sweet spot,” *A Dell Tech. White Pap. Round Rock, Texas*, 2011.

[28] M. Kummert, “Transient thermal analysis of a data centre cooling system under fault conditions,” *… Exhib. Build. …*, pp. 1299–1305, 2009.

[29] G. Varsamopoulos, M. Jonas, J. Ferguson, J. Banerjee, and S. K. S. Gupta, “Using transient thermal models to predict cyberphysical phenomena in data centers,” *Sustain. Comput. Informatics Syst.*, vol. 3, no. 3, pp. 132–147, Sep. 2013.

[30] L. Landau, “No Title,” *Zhurnal Eksp. i Teor. Fiz.*, no. December, 1937.

[31] Y. Yao and L. Huang, “Data centers power reduction: A two time scale approach for delay tolerant workloads,” *INFOCOM, 2012 …*, no. Section IV, 2012.

[32] C. Bash and G. Forman, “Cool Job Allocation: Measuring the Power Savings of Placing Jobs at Cooling-Efficient Locations in the Data Center.,” *USENIX Annu. Tech. Conf.*, no. June, pp. 17–22, 2007.

[33] W. Angelis, “Cooling Solutions for Dense Packed,” 2005.

[34] D. Kennedy, “Ramification of Server Airflow Leakage in Data Centers with Aisle Containment,” 2012.

[35] “ansys cfd data center cooling.” [Online]. Available: https://caeai.com/resources/cfd-analysis-data-center-cooling. [Accessed: 31-Oct-2014].

[36] K. Khankari, “Rate of heating analysis of data centers during power shutdown,” *ASHRAE Trans.*, vol. 117, pp. 212–221, 2011.

[37] W. Paper, “Data Center Temperature Rise During a Cooling System Outage.”

[38] E. K. Lee, I. Kulkarni, D. Pompili, and M. Parashar, *Proactive thermal management in green datacenters*, vol. 60, no. 2. 2010, pp. 165–195.

[39] “ANSYS Fluent.” [Online]. Available: http://www.ansys.com/Products/Simulation+Technology/Fluid+Dynamics/Fluid+Dynamics+Products/ANSYS+Fluent. [Accessed: 30-Oct-2014].

[40] “ANSYS CFX and Fluent CFD Solutions.” [Online]. Available: http://www.mallett.com/ansys-cfd-solutions.php. [Accessed: 30-Oct-2014].

[41] M. Engineering, “TURBULENCE MODELS AND,” 2011.

[42] “Tips for Selecting Turbulence Models.” [Online]. Available: http://www.innovative-cfd.com/turbulence-models.html. [Accessed: 27-Dec-2014].

[43] "Use of k-epsilon and k-omega Models -- CFD Online Discussion Forums." [Online]. Available: http://www.cfd-online.com/Forums/main/75554-use-k-epsilon-k-omega-models.html. [Accessed: 27-Dec-2014].

[44] Slater, John W., "Examining Spatial (Grid) Convergence." [Online]. Available: http://www.grc.nasa.gov/WWW/wind/valid/tutorial/spatconv.html. [Accessed: 31-Mar-2015].

[45] P. J. Roache, "Quantification of Uncertainty in Computational Fluid Dynamics," *Annu. Rev. Fluid Mech.*, vol. 29, no. 1, pp. 123–160, 1997.

APPENDIX A

ANSYS CALCULATIONS

All the models were designed meshed in ANSYS academic version 14 and the fluid simulation was carried out using ANSYS Fluent. The models were developed based on the dimensional data provided by IO Data Center, Tempe, Arizona.

The provided dimensional data were provided in terms of inches and feet. So the initial calculations were performed using that standard system. While the thermal and flow conditions were later converted to SI units, the length is represented in terms of Inches and feet.

1. Geometric Calculations And Design Modeling:

The specifications of the floor area and the server power were provided in the blue print and the room dimensions were calculated from them. Some the dimensional data like the tile dimensions and the height of the room were all taken from the IMPACT lab at Arizona State University, Tempe. The dimension calculations are shown below.

Room Dimensions Calculation:

Given Details:

Floor area of the room = 432 ft^2

Length of the floor = 42 ft

Assumptions:

Height of the room = 100 in

Height of the raised plenum = 42 in

Cross-section of the floor tiles = 2 x 2 ft^2

Calculations:

Width of the floor = $\frac{432}{42}$ = 136 in or 11’ 4”

Rack Height Calculations:

Width of each Rack = 24 in (given)

Length of each Rack = 40 in (assumed)

Calculations:

Total height of all racks = 2400 RU (given)

1 Rack Unit (RU) = 1.75 in

2400 RU = 4200 in

Height of one Rack = $\frac{4200}{50}$ = 84 in

<u>No. of Racks:</u>

Single-Row models = 18

Double-Row models = 24

<u>Cooler Dimensions:</u>

Cooler cross-section = 40 x 40 in^2

Cooler length = 40 in

Bases on the calculated values, the geometric modeling was done using ANSYS Design Modeler and the 3D CAD model was develop. The servers and the coolers were developed as framed solids and the solid region was suppressed to provide the fluid boundary. Therefore the computational resources required to solve the solid region can be saved and the simulation time can be improved. The fans were developed as thin surfaces and were later defined as fans in the boundary conditions.

<u>2. Meshing:</u>

Meshing is an important factor in determining the accuracy of the results and also the simulation time. As the number of mesh-elements increases, the solution tends to be more accurate. But the time to solve the iteration also increases. Therefore always while developing and meshing a model, the designer has to always make a trade-off between the accuracy of the results and the simulation time. In our developed mesh model, the minimum number of meshes required to achieve reasonable results were chosen. This was verified by checking the mesh quality in ANSYS Fluent. The mesh quality is performed by checking the orthogonality values of the meshes. The orthogonality values range from 0 to 1. If the orthogonality is far too low, which is closer to zero, then the mesh quality if determined as poor and the solution will be inaccurate. The closer the orthogonality value to the value 1, then the quality of mesh is determined as better. In each of the models, the orthogonality was around 10^{-1}. Hence the developed mesh is determined as good and the mesh is imported to Fluent.

In the mesh model, the required boundary surfaces are chosen and named. This allows the user to define the necessary simulation parameters during the pre-processing of the model. The outer walls of both the servers and the cooler is chosen as heat input surfaces. The developed fan surfaces are named as fan surface. The vents are not given any boundary names, because they are included in the fluid region and hence no special treatment is required for them.

<u>4. Numerical simulation in fluent:</u>

The simulation was carried out using ANSYS Fluent. It is a simple, very powerful tool with myriad of options that can be switched ON depending on the problem statement and has an extremely user-friendly environment. Aster the mesh model is imported in the Fluent, the dimensions and the mesh quality are checked as stated in the previous section. Then the thermodynamic properties of air, which is a single-phase working fluid, are given and the buoyancy effects are included. Although the effects of buoyancy could be neglected in smaller data center because the effects of natural convection are negligible as a result of forced convection, its effects are significant in larger data centers. Therefore the option is switched on.

Boundary condition calculation:

The power of the racks is given in terms of Heat Flux in W/m2. It is defined in terms of flux because the heat source has to be defined in terms of heat per unit area in Fluent. There are few assumptions made before the boundary conditions are calculated. All the servers equally contribute the total power. Similarly the heat contributed by each server is contributed by all surfaces of the server. The total heat of the rack is countered by all the cooler power of the cooler. Hence the negative of total heat flux of the server gives the value of the cooler flux.

Heat Flux of Servers:

Surface Area of Servers:

Volume of the server = 24 x 10.8 x 40 in^3

Surface area of top face, A_1 = 40 x 24 = 960 in^2

Surface area of side face, A_2 = 40 x 10.8 in2 = 432 in^2

Total surface area of each server, $A_{ser} = 2 (A_1 + A_2) = 2760$ in^2

Total surface area in SI units = 1.74 m^2

Heat Flux of Servers:

Heat of each rack = 5000 W

Heat of each server = $\frac{5000}{8}$ = 625 W

Heat Flux of each server = $\frac{625}{1.74}$ = 359.2 W/m^2

Therefore the total heat flux of each server is approximated as 360 W/m^2

Surface area of cooler:

Volume of the cooler = 39 x 39 x 40 in^3

Surface area of each side , A_{ii} = 39 x 40 = 1560 in^2

Total surface area of cooler, A_{cool} = 6240 in^2

Total surface area in SI units = 4.03 m^2

Heat Flux of Cooler:

Heat flux of cooler = $\frac{5000}{4.03}$ = 1240.69 W/m^2

Required cooling load on the cooler = 1.3 x total power of rack

= 1612.9 W/m^2

Cooler per each rack = $\frac{14}{18}$ = 0.78

Actual heat flux of cooler = 1238.06 ≈ 1240 W/m^2

The calculated heat flux values of the server are given as the surface heat flux condition to all the servers. For the cooler, the calculated values are given as negative values in the input for the cooler surface.

The fan speed can be defined in terms of pressure jump. The pressure jump was determined only through number of simulation trials. After the iteration trials, the determined pressure jump value is given as input boundary condition for the fans. The fan direction is checked just to ensure the airflow direction is maintained in the required way. Since the fans are considered as interior fluid regions, Fluent adjusts the interface surface between the room and fan fluid regions.

After performing the simulation, the results were monitored by observing the temperature and velocity plots. The contour plots are also monitored just to observe the flow pattern of the air. The results were collected and plotted using Matlab.

Grid Convergence Index (GCI):

The GCI is calculated using the equation

$$\mathrm{GCI}_{21} = \frac{F_s |\varepsilon| r^p}{(r^p - 1)}$$

Where,

F_s = 3 for comparing two grids (used in this study)

= 1.25 for comparing three or more grids

$\varepsilon = \dfrac{f_1 - f_2}{f_1}$, f_1 & f_2 are the solutions from two grids

$r = \dfrac{h_2}{h_1}$, h is the grid spacing with h_1 being finer mesh, r=2

$p = \dfrac{\ln\left(\dfrac{f_3 - f_2}{f_2 - f_1}\right)}{\ln(r)}$. Since there is no third refinement, the theoretical order of convergence for p is chosen, which is p=2

GCI for Cooling Delay:

$$\varepsilon = \frac{28-32}{32} = -0.142$$

$$\text{GCI} = \frac{3\left|-.142\right|2^2}{2^2-1}\ 100\% = 56.8\%$$

GCI for Overload Temperature:

$$\varepsilon = \frac{1.6-1.8}{1.6} = -0.159$$

$$\text{GCI} = \frac{3\left|-.159\right|2^2}{2^2-1}\ 100\% = 63.6\%$$