ALTERNATIVE COOLING TECHNOLOGIES FOR TELE-
COMMUNICATION DATA CENTERS: AIR COOLING
AND REAR DOOR HEAT EXCHANGERS

by

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ABSTRACT

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Telecommunication industry is currently emphasizing to a miniaturization and convergence. The need for smaller, smarter and faster devices is leading drastic increment in power density and accordingly, thermal management of such devices has become a challenge. The heat generated by the devices has to be dissipated in order to ensure proper and efficient functioning of the electronics. Electronics cooling ranges from device level to the room level cooling. Data Center is a facility (room) consisting of servers used for managing and transferring data or information. The thermal management of this facility is challenging due to the huge decrease in floor space for equipment and a tremendous increase in the power density of the servers without a compromise in the energy efficiency. The increase in server heat density waves a scope for developing improved cooling technologies without increasing the power consumption. It is commonly observed that 40% of the total data center energy is consumed by its cooling equipments. In our application, thermal management through air cooling is possible to a maximum of 16kW. Beyond 16kW, typical air cooling techniques leads to a substantial increase in the power consumption hence, a concept of hybrid cooling using

Rear Door Heat Exchanger (RDHx) is incorporated. RDHx is used for higher server density cabinets up to 25kW which helps in reducing the power consumption and efficient cooling. This study is therefore divided into two parts: air cooling and hybrid cooling.

In air cooling of Data center, suitable cooling accessories for various heat loads of the cabinets in a data center are determined. The accessories include the blanking panels and chimneys. Furthermore, impact of hot/cold air isolation is analyzed for high heat loads of the cabinets. It is observed that the accessories are effective for heat loads up to 12kW and the hot/cold air containment is effective for 16kW.

Furthermore, in case of RDHx, emphasis is placed on the analytical determination of the optimum heat load after determining the effectiveness of heat exchanger at the given operating conditions. Then, thermal analysis is performed to compare various heat loads of the cabinets with RDHx. Based on the results; a ‘best known method’ or ‘rule of thumb’ is verified that rear door heat exchanger could be 100% efficient in cooling the cabinets of heat loads up to 27kW. The impact of RDHx in different configurations is studied and compared. A computational study of the effectiveness of RDHx in side breathing network switches is analyzed. It is seen that there is a decrease in the global maximum temperature with the inclusion of RDHx and they are very effective in cooling the side breathing switches.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

NOMENCLATURE

ρ	- Density (kg/m^3)
u	- Velocity in x- direction (m/s)
h	- Specific enthalpy (J/Kg)
k	- Thermal conductivity (W/m-K)
T	- Temperature (K)
S_h	- Volumetric rate of heat generation (W/ m^3)
B_x	- Body force per unit volume in x- direction (N/m^3)
V_x	- Viscous force per unit volume in x-direction (N/m^3)
τ	- Shear force
ε	- Kinematic rate of dissipation (m^2/s^3)
P	- Pressure (Pa)
Q	- Heat load in kW.
$\dot{m}$	- Mass flow rate in kg/sec
C_p	- Specific heat at constant pressure in J/kg-K
ΔT	- Difference in temperature in K
ε	- Effectiveness of heat exchanger

CHAPTER 1

INTRODUCTION

The current telecommunications technology emphasizes on miniaturization and convergence. This industry can be classified into four main sectors: wired, wireless, satellite and other establishments. The wired systems requires networks of cables and wires for accessing devices like television, telephones(landline) and on the other hand, wireless systems transmit signals over networks of radio towers. The signals reach the customers directly through the antennas. For example, devices like cell phones, mobile computers, can now receive, store, interpret and send information or data at faster rate. There is a tremendous demand for faster, efficient and compact devices; new technology is being deployed for achieving the customer needs of faster data transfer. Due to this miniaturization of devices, the power density is increasing at an alarming rate. Therefore, thermal management of these devices is becoming more challenging. Data Center is a facility used for housing the equipments for storing and processing data. It typically consists of servers used for managing and transferring data or information. The heat produced by the servers is cooled by passing cold air through the cabinets but due to its layout, there is a mixing of cold and hot air and also recirculation of hot air within the room. This increases the inlet temperature of air entering the cabinets. The servers are vertically stacked in the cabinet racks for saving floor space of the room which leads to a mal distribution of air due to its height. Therefore, it can be inferred that the air flow has a major impact on the thermal management of Data Center.

1.1 Thermal management of Data Center and its importance

Heat is generated within the components of the device because of the flow resistance offered by the building blocks; transistors, resistors and capacitors. This heat has to be dissipated to ensure a proper and efficient working of the devices otherwise heat keeps increasing at a rate till the device fails to work or being operated.

Cooling allows lowering the high temperature due to the heat by transferring heat from hot fluid to the cold fluid. This is based on the first law of thermodynamics which is the law of conservation of energy. The temperature is moderated by reaching a steady state value which is acceptable for the device's operation.

The electronics cooling can be divided into various levels as shown in figure 1.1. Package at each level releases heat which has to be cooled. The power dissipation is increasing tremendously and thus thermal management is more challenging. Cooling of telecommunication systems ranges from cooling of chips, devices, cards, drawers to cabinets/racks and room.

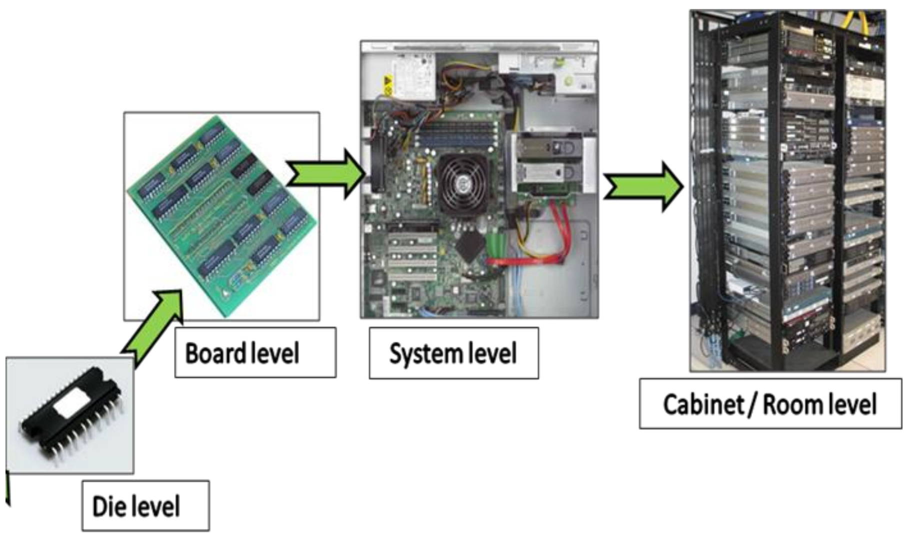


Figure 1.1 Cooling range from wafer level to room level

Thermal management of Data Center is becoming more challenging on an account of the huge decrease in floor space for equipment and a tremendous increase in the power density of the servers without a drop in the energy efficiency. American Society for Heating, Refrigerating and Air conditioning Engineers (ASHRAE) sets the standards for constructing the

Data Center TC9.9 [1] and also provides guidelines for the operating conditions required. Figure 1.2 shows the current trends in heat produced per foot print area.

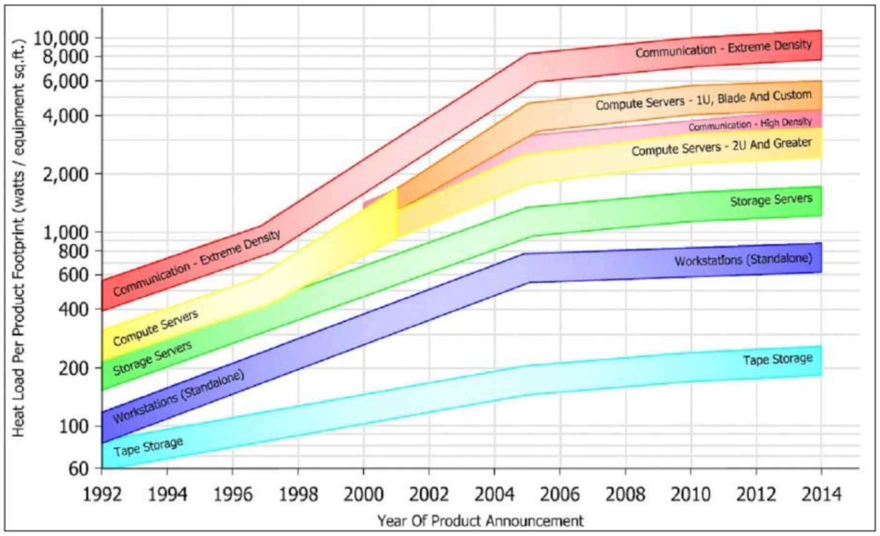


Figure 1.2 Current trends in heat loads of the Data Center

The current trend of increasing power in Data Centers at a warning rate leads to higher heat fluxes due to the increase in power dissipation and higher heat densities. From ASHRAE guidelines for datacom equipment [1], the predicted foot print is doubled and the heat loads for the storage servers are tripled for the same period (2000-2004). It is observed that these heat fluxes were 4000 W/ft^2 in 2006 which corresponds to a heat load of 27kW in a 19 inch racks, available commercially. However, there is further increase in the coming years and it is observed that the heat loads will increase to 5000W/ft^2 in 2010 and around 6000W/ft^2 in 2014 for 1U blade computer servers. This corresponds to a heat load of 35 kW and 42 kW respectively. Higher heat loads in Data Centers lead to hot spots, the regions with higher heat fluxes, which increase the temperatures of Data Center causing reliability issues. Therefore, cooling of Data Centers is of high priority.

1.2 Scope of the study

The study deals with the thermal management of the telecommunication Data Centers. It is divided into two parts, where the first part discusses the impact of air cooling and its advantages and disadvantages and in the second part, the importance of hybrid cooling using Rear Door Heat Exchanger is studied. This is used when the air cooling is consolidated for higher heat loads of the cabinets in Data Centers.

Chapter 2: Discusses the various cooling technologies used for cooling Data centers

Chapter 3: Numerical modeling techniques used for modeling Data Centers.

Chapter 4: Discusses the thermal impact of air cooled Data Centers and different accessories necessary for various heat load.

Chapter 5: Impact of Rear Door Heat Exchangers (RDHx) in cooling of data center.

Chapter 6: Conclusion and future work

CHAPTER 2

LITERATURE REVIEW

2.1 Air cooling techniques

There are different cooling techniques used in Data centers: Air and liquid cooling. Following is a collection of data on the different cooling techniques used in Data Centers.

The commonly used cooling techniques used are CRAC units. Roger Schmidt [2] in one of his papers collected different practices for Data Center thermal management in which he covered topics like new building design, perforated tile layouts in raised floor facilities, rack-related effects in raised-floor and non-raised-floor Data Centers, energy management and efficiency, and miscellaneous items such as experimental case studies, analytical and numerical modeling, and Data Center metrics.

Schmidt et.al [3] describes the factors that affect the cooling design of the computer and telecommunication equipment room. This review paper deals with the thermal management of computer Data Center and telecommunication rooms. New techniques for cooling which improve the reliability of the equipment have been mentioned.

Boucher et.al [4] has provided strategies for thermal management and energy performance of Data Centers with automatic control. The strategy used currently consists of a single sensory which controls the temperature of CRAC and also shows its heat dissipation. The drawback of this strategy is its inflexibility in showing how the cooling is provided to the computers and so it is not efficient in optimizing the Data Center. The three requirements that are used for actuating dynamically are 1) A distributed sensor network which indicates the local conditions of the Data Center. 2) Ability to vary cooling resources locally. 3) Having knowledge of how each variable affects the conditions of Data Center.

Sharma et.al [5] in his study described experimental results from heat transfer and fluid flow experiments in production level Data Center. Temperature distribution across the Data Center is measured by large distribution with sensors at CRAC unit air flow and supply temperature configurations.

Schmidt et.al [6] overviewed the necessary thermal environment of Data Center by a numerical model used for governing mechanisms. He discussed about the air recirculation, one of the major cooling problem.

Kakri et.al [7] focused on the techniques used for controlling the air flow distributions by using mathematical model and their effectiveness. These techniques are related to the plenum height, area of tiles, position of CRAC units, tiles and presence of the under floor blockages.

Patel et.al [8] dealt with the design of cooling Data Center and studied the use of Air conditioners in the numerical modeling. It involves the extraction of energy in Air conditioners used in Data Centers.

Patel et.al [9] had provided a vision for the use of such a sensor network coupled with standard CRAC components to enable more energy-efficient control architecture and noted that a 50% of energy was reduced by using cooling techniques added to the space utilization increase of Data Center.

Bash et.al [10] described about the dynamic thermal management of air cooled Data Centers in which the conventional method of CRAC units was used and its effect on cost was studied. It was observed that a total of 25% of savings was made in cost for space and power

Bedekar et.al [11] studied the effect of location of CRAC units in Data Center and concluded that performance of Data Center majorly depends on the location of CRAC units and also the flow rate of the air entering the cabinets.

Beaty et.al [12] and Schmidt et.al [13] predicts the low inlet temperatures for those racks which have clear path for hot air from racks to the CRACs. They also recommend placing the CRAC facing hot aisles rather than facing the cold aisles.

2.2 Cooling of Data Centers with high heat loads

Different techniques are incorporated for cooling Data Centers for avoiding higher inlet temperatures. These include hot aisle containment and cold aisle containment. This isolation has the following advantages as shown in the figure 2.1. The figure summarizes the pros of isolation and need for having it.

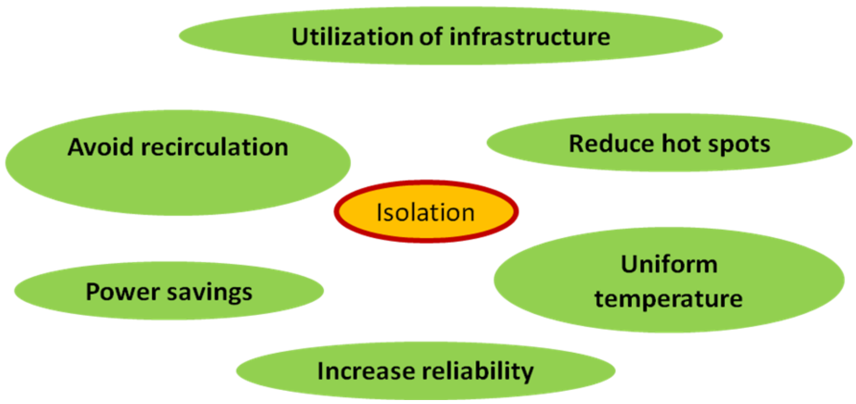


Figure 2.1 Advantages of isolation

2.2.1 Hot/cold aisle containment

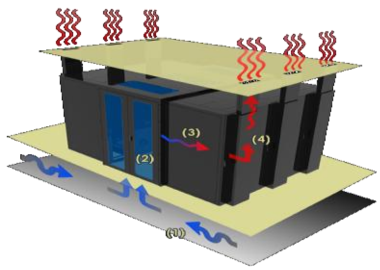
This technique is most widely used by companies in order to reduce the maximum temperature of the cabinet and also the inlet temperature of air entering the cabinets. The hot/cold aisle offers the benefits [14]:

1. Higher temperature can be set to cooling systems which saves energy and supplying the load with safe operating temperatures.
2. Reduction in humidification/ dehumidification costs.
3. Physical infrastructure utilization which enables right sizing resulting higher efficiencies.

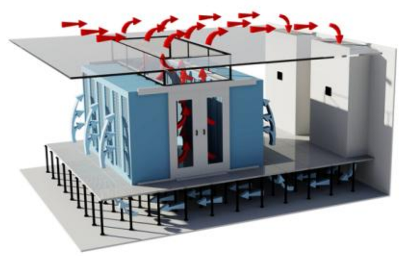
2.2.2 Hot aisle containment

Hot aisle containment is a technique followed for reducing the inlet temperature of the cabinets and also the recirculation of the air [14]. Various methods are incorporated for adopting this technique. Use of chimneys, ducts and doors are some of the methods described here.

Using of chimneys is the method used in which the hot air is exhausted directly in to the plenum located at the top of the chimney. Ducts are the still yet to be used in companies but its functioning is similar to that of the above just that the path of hot air is restricted for exiting. Other technique which is used is the restricting the hot aisle by placing doors or curtains for collecting the hot air. One of the above or the combination of two of them can be used for improving the cooling of Data Center. Each of the technique is shown in the figures [15].



(a)



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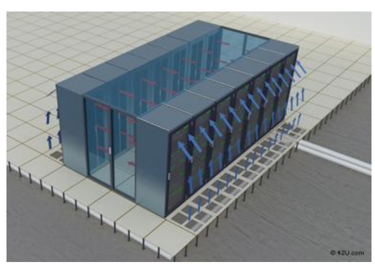


(c)

Figure 2.2 (a) Cabinet with chimney and a top plenum, (b) Data Center with hot and cold air containment, (c) Data Center with hot air containment

2.2.2 Cold aisle containment

In this technique, cold aisle is contained for reducing the inlet temperature of the air entering cabinets. Eliminating mixing cold and hot air saves energy consumption and results in higher exhaust temperatures which increases the cooling capacity of the Data Center and reduces the over cooling. The advantages of using cold air containment are:

1. More cold air is available for cooling and also power loss.
2. For maximizing cooling system efficiency, cold aisles are contained.

However, there are little inefficiencies when the containment is deployed in a room based perimeter cooling environment.

1. For adequate air distribution, required pressures and distances results in inefficiencies.
2. Limitations in density when cold air distribution through raised floor.
3. Predictability of the raised floor.

The table below summarizes the differences between hot and cold aisle containment [14].

Table 2.1 Cold aisle containment vs. hot aisle containment

Characteristic	Cold aisle containment	Hot aisle containment
Efficiency improvements	Yes	Yes
Increase in cold air supply	No	Yes
Room neutral solution	No	Yes
Ease of deployment with room cooling	Yes	No

2.3 Impact of Liquid cooling

Dr. Shlomo Novotny [16] in one of his white papers, explained in detail about the need of liquid cooling and its techniques. He mentioned about the advantages of rear door heat exchangers (RDHx) and its performance in data center. RDHx is a device placed for cooling the hot air released from the cabinets. Several types of liquid cooling solutions that are incorporated in Data Centers [17] are Modular cooling unit that contains cooling modules providing a cooling capacity of 30 kW. These utilize variable speed fans for controlling the heat load within the rack. Door units which contain sealed tubes of chilled water. Integrated rack based liquid cooling is a rack based cooling architecture integrating UPS power, CDU etc. for pumping water through the aluminum/plastic to cool servers, Liquid cooling heat exchangers for reducing the hotspots.

The limitations of under populated racks and spreading of heat loads can be overcome using liquid cooling. However, cost of using liquid cooling is justified for high heat loads (70 kW) as proposed by [17]. Thermal management of dense data center clusters using liquid cooling is described in [18]. The other technique which is feasible is the hybrid cooling which uses both air and water for cooling [19].

Schmidt et.al [20] describes a method for reducing the impact of the hot air recirculation with a water cooled heat exchanger attached to the rear door of the rack. Tests have been performed to analyze the impact of this heat exchanger in cooling the racks. It is noticed that heat exchangers can cool up to 50-60% of heat and also significantly reducing the exhaust temperature. Heat exchangers showed a significant advantage in the first time and annual cost within the Data Center.

Chio et.al [21] studied the operating characteristics of telecommunication equipments relative to the heat density and operating conditions and also the performance of the hybrid refrigeration system. A test was performed for a unit rack and observed that the heat transfer coefficient increases with the rise in fan speed. The cooling capacity decreases and power consumption increases with increase in the outdoor temperature. The maximum temperature should be maintained for the thermal reliability of telecommunication equipments.

Baer [22] made an effort to indentify the factors effecting the power and thermal densities and analyze the cooling system on the telecommunication spaces. The current cooling practice is to use air as a medium which uses ducts. Fluids and refrigerants are good alternatives for absorbing heat and transport of heat. Several techniques for cooling are mentioned.1) Fans, helpful for equipment that uses high discharge temperature.2) Smaller footprint air conditioners for reducing the installation space and improving reliability and saving the valuable floor space. 3) Enclosure mounted heat exchanger that uses water or dielectric fluid as a medium for absorbing the heat expelled by the equipment. 4) Ceiling mounted heat exchanger with movable fan trays, tube coils and fluid or refrigerant as the medium. 5) Fluid assisted heat sinks used for high density packaging. 6) Spray cooling method is used for cray computers. The future cooling system will be a combination of above mentioned techniques. It is inferred from the paper that the thermal and energy saving methods can reduce the operating costs.

Ellsworth et.al [23] dealt with the water cooling technology. He discussed about the air-water cooling and indirect water cooling systems. Water cooling techniques meet thermal challenges and provide knowledge which will be a base for the new water cooling designs.

Herb villa [24] expresses liquid cooling cabinet heat exchangers (LCP) as an ideal solution for removing excessive heat loads from the data center. This solution has to provide a sufficient floor space for installing these systems and also for additional power.

Kang et.al [25] describes liquid cooling system which can cool single or multiple heat sources. The system uses cold plates for taking in the heat from the heat sources and then transfer heat to air by forced convection. It is observed that the performance of liquid cooling is in accordance with the test data.

Sharma et.al [26] experimentally determined the performance of Data Center. Temperature distribution across the Data Center is measured by the sensors placed at the CRAC units for the air flow and supply temperature. The CRAC units are responsible for the cooling the exhaust hot air from the computer racks. The work done to distribute the cool air and extract the heat from the exhaust air is the energy consumed in cooling the Data Center. The chilled water cooling coil is used for the extracting heat from air and cools it within the range of

10-18^{0}C.An experimental setup of the data center is established with seven rows of servers and six CRAC units. The results obtained by the changing the power from 0-8 kW in steps of 2kW. The supply temperature is 20.8^{0}C. A comparison of conventional cooling to liquid cooling for a 1000 kW data center load is tabulated below [27].

Table 2.2 Comparison of convectional cooling and liquid cooling

	Cooling towers and pumps	Chiller	Chilled water pumps	Fan	Other	Total power (kW)	% savings
Traditional system (45^{0}F chilled water)	70	500	50	150	N/A	770	N/A
Liquid cooling with fans (55^{0}F chilled water)	70	425	50	100	N/A	645	16
Liquid cooling without fans (55^{0}F chilled water)	70	425	50	0	N/A	545	29
Liquid cooling directly couple (70-80^{0}F chilled water)	70	0	50	0	Room A/C - 245	365	

CHAPTER 3

CFD MODELING

3.1 Introduction

Computational fluid dynamics, a branch of fluid dynamics uses numerical methods and algorithms for solving fluid flow problems. The domain of interest in numerical model is represented as a mathematical model. The study includes modeling of system level electronics like telecommunication shelters and data centers using the commercially available software. The objective is to analyze the air-flow processes occurring within and around electronics equipment, by improving and optimizing the design of the required equipment. CFD software is governed by the differential equations to find the numerical solutions.

3.2 Governing equations

For solving heat transfer and fluid flow problems, three differential (governing) equations are solved for the conservation of mass, energy and momentum and solving these governing equations results in the numerical solution [28].

Following are the generalized cases for the conservation of mass, energy and momentum respectively.

$$\frac{\partial \rho}{\partial x} + \operatorname{div}(\rho u) = 0 \tag{1}$$

$$\operatorname{div}(\rho u h) = \operatorname{div}(k \operatorname{grad} T) + S_h \tag{2}$$

$$\frac{\partial}{\partial t}(\rho u) + \operatorname{div}(\rho u u) = \operatorname{div}(\mu \operatorname{grad} u) - \frac{\partial p}{\partial x} + B_x + V_x \tag{3}$$

3.2.1 Boundary conditions

The space where governing differential equations are solved is called the domain of integration. The solution is obtained by providing the boundary conditions for the solution domain. The properties of fluids like conductivity, density, viscosity, specific heat, expansivity and diffusivity have to be specified for solution. The conservation equations and the associated boundary conditions are solved by means of numerical integration for majority of cases. Flotherm, CFD code used for this study, uses finite volume method for solving these equations. The first step in solving these equations is to discretize the domain of integration into contiguous finite volumes. These finite volumes are also called as grid cells. The variables to be calculated are located at the centroid of these cells. The layout of velocities for grid in all the three dimensions is shown below.

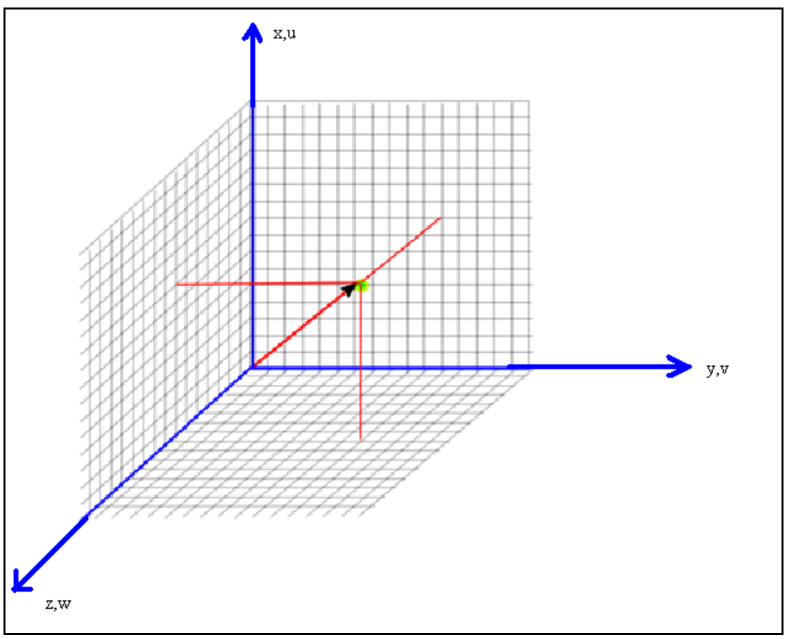


Figure 3.1 The grid showing the velocities

The finite volume method is advantageous in satisfying the equations for conservation of mass, energy and momentum over the entire domain of computation. The second step is to obtain discretized results in a set of algebraic equations that relates to the value of the variable in a cell to its value in the neighboring cell. As an example, if T is the temperature variable, it is obtained by the following equation [29]:

$$T = \frac{C_0 T_0 + C_1 T_1 + C_2 T_2 + C_3 T_3 + C_4 T_4 + C_5 T_5 + S}{C_0 + C_1 + C_2 + C_3 + C_4 + C_5} \tag{4}$$

Where, T_1, T_2, T_3, T_4, T_5, are the temperature values in the neighboring cells and the T_0 is the temperature value in the old time step, C_s is the coefficient that connect the cell values and S is the source term. The algebraic equations are solved for the field variables T, u, v, w, and ρ which implies that if the number of cells are n in the domain then a total of 5n equations are solved.

The equations are non-linear in reality as the coefficients are also functions of the above field variables but they appear to be linear. However, this is exploited when the equations are solved iteratively. For outer iteration, the values for the coefficients are solved for once and then considered constant while the equations are solved by inner iteration.

Generally, the solution is more accurate as the number of grid cells increase i.e., finer the grid, the better is the approximation of the governing equations. Therefore, more grid cells are used at the regions in the domain where the gradient of the variables is high. It is observed that the grid independent solution alone does not guarantee a solution that simulates close to the real world problem, as it is also affected by the factors like the accuracy of boundary conditions, turbulence model that affect the solution outcome and its accuracy.

The basic thermodynamic processes include constant volume process, constant pressure process, isothermal process, adiabatic or isentropic process and polytrophic process and each process is discussed briefly below.

- Constant Volume (Isochoric) process:
    - Temperature and pressure increases under a constant volume
    - It is an irreversible process
    - Energy considered is the internal energy.

- The relation is given as $\frac{T_2}{T_1} = \frac{P_2}{P_1}$ and the difference in entropy is calculated as

$$\Delta S = mC_v \log\left(\frac{T_2}{T_1}\right)$$

- Constant Pressure (Isobaric) process
  - Temperature and volume increases under a constant pressure
  - It is an irreversible process
  - There is an external work due to an increase in the volume and therefore internal energy is supplied by the heat
  - The relation is given as $\frac{T_2}{T_1} = \frac{V_2}{V_1}$
- Constant temperature (Isothermal) process
  - This process occurs when the substance is in thermal contact with the surroundings and so there is change in temperature.
  - There is no change in internal energy and enthalpy
  - It is a reversible process
  - The relation is given as $P_1V_1 = P_2V_2$ or $\frac{P_1}{P_2} = \frac{V_2}{V_1}$
- Adiabatic (Isoentrophic) process
  - In this process there is no transfer of heat between the system and surroundings
  - There is a change in temperature and internal energy
  - It is a reversible process
  - The relation is given as $\frac{P_1}{P_2} = \frac{T_1}{T_2}$ and the difference in volume is calculated as

$$\Delta V = mC_v(\Delta T)$$

- Polytrophic process
  - It is an irreversible process

In the present study, the solution domain is treated as open boundaries which are of constant pressure or symmetry boundary conditions where no air or heat flow between system and surroundings.

3.3 Turbulence modeling

A turbulent flow field is compounded with velocity fluctuations in all directions and infinite number of degrees of freedom. The flow is described as three dimensional, diffusive, dissipative and intermittent.

Solving Navier stokes (NS) equations for a turbulent flow is unachievable as these equations are elliptical, non-linear and also coupled with pressure or temperature with velocity. One of the solutions for this is to develop new partial differential equations (PDEs) for every term in the NS equations but this involves new correlations for unknowns.

Following are the equations solved for variables.

Conservation of mass

$$\frac{\partial \rho}{\partial \tau} + \frac{\partial \rho U_i}{\partial x_i} = 0 \tag{5}$$

Conservation of momentum

$$\frac{\partial \rho U_i}{\partial \tau} + \frac{\partial \rho U_i}{\partial x_i} = \frac{\partial}{\partial x_j}\left[\mu \frac{\partial U_j}{\partial x_j} - \rho \overline{u_i u_j}\right] - \frac{\partial P}{\partial x_j} + S_{ui} \tag{6}$$

Conservation of passive scalar

$$\frac{\partial \rho c_p T}{\partial \tau} + \frac{\partial \rho c_p U_j T}{\partial x_j} = \frac{\partial}{\partial x_j}\left[k \frac{\partial T}{\partial x_j} - \rho c_p \overline{u_j t}\right] + S_t \tag{7}$$

Where T is the scalar

Therefore, alternative to this is to develop PDEs for turbulent stresses and fluxes aided to modeling. Following are the different models used as per the complexity [30]:

- Algebraic or zero equation models

- One equation models: k-model or μ_ε first order model
- Two equation models: k-ε, k-kl, k-ω^2, low Re k- ε first order model
- Algebraic stress models or ASM second order models.
- Reynolds stress models or RSM second order models.

The two equation model is used for predicting the eddy viscosity on a cell by solving two differential transport equations. The common ways by which Flotherm models this Low Reynolds number turbulence flow regimes is by LVEL turbulence and the K-Epsilon model.

3.3.1 Automatic algebraic turbulence model

It does not consider any user defined velocity or length scale. The velocity scale considered to be the cell velocity. [31]

The length scale is calculated as

$$D = \sqrt{(|\nabla\phi|^2 + 2\phi)} \tag{8}$$

$\nabla\phi$=-1 with ϕ=0 at a wall

Then the length and velocity scales are also considered in addition to the boundary layer wall function to determine cell turbulence viscosities which vary from cell to cell in bulk flow.

3.3.2 LVEL K-Epsilon turbulence model

The model solves using two variables; the kinetic energy of turbulence (k) and the dissipation rate of k (denoted ε). The model requires the nearest wall distance (L), the local velocity (VEL) and the laminar viscosity. It is inexpensive and can be applied to three dimensional problems. From the eq. (8), we can calculate the maximum local length scale and the distance to the nearest wall as

$$L = D - |\nabla\phi| \tag{9}$$

This method is applicable for the electronic equipments placed close to each other and also to the walls of the enclosure as it predicts the turbulence viscosities for cells near the walls and provides high turbulent viscosities in the free stream.

3.3.3 K-Epsilon turbulence model

It is also called as two-equation model, used for fluid dynamics. It computes viscosity on a grid cell and not on viscosity due to the walls. It involves solving of transport equations for kinetic energy and rate of turbulent dissipation. This model is applicable for problems with shear layers and recirculating flows [31]. This is most preferable for free air streams like data centers and telecommunication shelters.

The following are transport equations [32]

$$\frac{\partial(\rho\varepsilon)}{\partial t} + \frac{\partial(\rho\varepsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_i}\right] + G_k + G_b - \rho\varepsilon \tag{10}$$

$$\frac{\partial(\rho\varepsilon)}{\partial t} + \frac{\partial(\rho\varepsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial \varepsilon}{\partial x_i}\right] + C_{1z}\frac{\varepsilon}{k}(G_k + C_{3z}G_b) - C_{2z}\rho\frac{\varepsilon^2}{k} \tag{11}$$

3.4 Flotherm smart parts for modeling

The following are the components used for modeling the telecommunication shelter and data centers.

- Enclosures
- Cuboids
- Fans
- Fixed flows
- Filters
- Recirculation devices

3.4.1 Enclosures

These are used for modeling many container shaped objects in electronic equipments. Generally, the size of the enclosure matches the solution domain. It is useful for modeling internal objects where the internal flow is considered.

3.4.2 Cuboids

These can be also created and any other palette like volumetric regions, monitor points, prisms etc., can be added to it as siblings. These cuboids can be collapsed for forming plates.

3.4.3 Fans

Fans are devices used for cooling the equipment by creating air to flow through the device. They are low pressure air pumps that use the power from motor for finding the flow rate for a given pressure. The torque supplied to the propeller is converted to increase the static pressure across the fan rotor and also the kinetic energy of the air particles.

The commonly used fan types in cooling the electronics are axial flow fans and centrifugal blowers. Fans and blowers are differentiated by the method involved for the air flow and also the system pressure at which they have to be operated. [34]

- Axial fans: Fan delivers air in the direction parallel to blade axis and is designed for delivery high flow rates

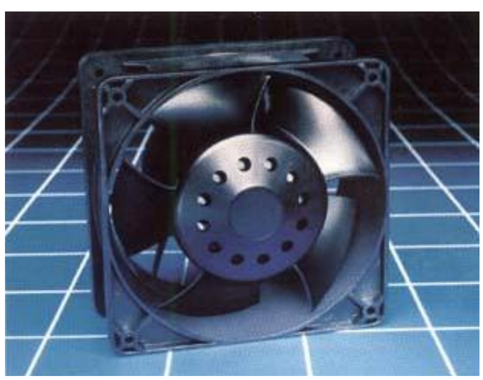


Figure 3.2 A typical fan

There are four types of axial flow fans and are based on increasing static pressure:

1. Propeller fans: It is also called as panel fan and most commonly used fans in industrial, commercial, institutional and residential applications.
2. Tube axial fans: These fans are applied for exhausting air from an inlet duct. It has fewer parts and so low in cost.
3. Vane axial fans: These fans are used for blowing air as well as exhausting air.
4. Two-stage axial flow fans: These fans have a configuration of two fans in series so that the pressures add up. These fans are very useful when higher static pressures are required but leads to noise and high tip speeds.

- Centrifugal blower: They deliver air in the direction perpendicular to the blower axis and are designed for working against high pressures but deliver low flow rates. These may have forward curved wheel, a backward curved wheel or of squirrel cage variety.

3.4.3.1 Basics of fans

Static pressure: When the piston moves upwards or down wards in a U-tube manometer, it produces a positive or negative static pressure relative to atmospheric pressure. A fan blowing air into the system produces a static pressure for overcoming the resistances. A fan exhausting from duct system produces negative pressure for overcoming the resistances.

The governing principle in selecting a fan is that the given fan can deliver one flow at one pressure in a particular system. Fan curve provides the user with information on the performance of the fan. It is plotted between the differential pressure (pressure drop which is $P_{atm} - P_s$) and the flow rate of fan. Fans generally operate at the operating point of the fan which is determined by the intersection of system curve and fan curve [35]. Figure [36] 2.3 represents the plot with operating points and different curves.

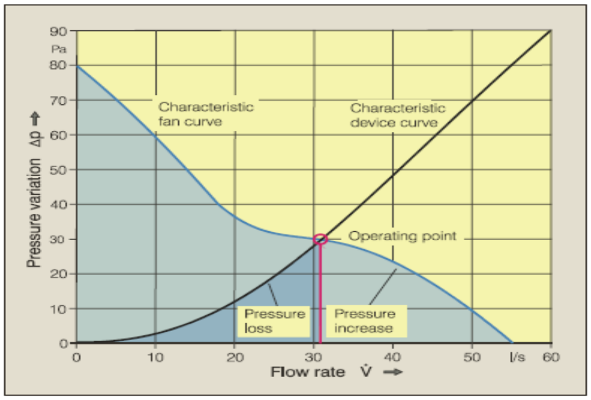


Figure 3.3 A plot showing the operating point and different curves

System curve or system impedance curve is a plot between the pressure drop and the flow rate and is generally proportional to each other i.e. the flow rate increases with the increase in the differential pressure. System curves vary with the geometry of the fan.

Fan curve is a plot between the pressure drop and the flow rate. The plot is obtained when the fan is run from a shut off position (where a maximum pressure drop and a zero flow rate is noted) to a free delivery position (where a zero pressure drop and maximum flow rate is noted). The fan operates at stable conditions because the difference in pressure drops as flow rate increases. When fans have to operate at restricted air flow path, the fan compensates the loss of pressure by increasing the static pressure. Generally operating point is obtained at 60-80% of the maximum flow rate of the fan, where the fan operates with its maximum efficiency. For selecting a fan, the flow rate is required and it is calculated using the basic heat transfer equation:

$$Q = \dot{m} \times c_p \times \Delta T \tag{12}$$

Where, $\dot{m} = \rho G$

A better approach for determining the required air flow is to find the optimum operating point using the air flow network analysis or using computational fluid dynamics software.

3.4.3.2 Fan Laws

The fan laws are used for determining the output of the given fan under other conditions of speed, density or changing the fan performance from one system to the other [37]. Listed below are the laws for fans that are geometrically similar. These laws are based on the following parameters:

- Speed of the fan(RPM)
- Volumetric flow rate(cfm)
- Static pressure(SP)
- Diameter of the wheel(D)

Law 1: Effect of RPM change

- cfm α RPM
- SP α RPM^2
- BHP α RPM^3

Law 2: Effect of fan size change (Constant volume)

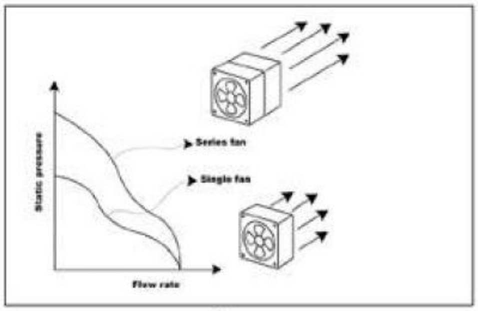
- cfm α D^2
- BHP α D^2
- RPM α $\frac{1}{D}$

Law 3: Effect of fan size change (Constant RPM)

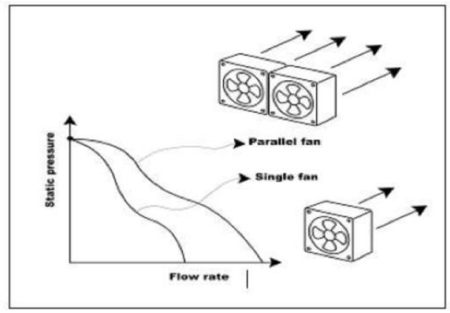
- cfm α D^3
- SP α D^2
- BHP α D^5

Fans in the CFD tool can be modeled as a 2D or 3D fan. Fans can be arranged in series or parallel configurations.

Fans when arranged in parallel the flow rate is added at the same pressure drop. Therefore, this configuration is used when higher flow rate is required by the system. On the other hand, when fans are arranged in series the pressure drop is added for the same flow rate and so this configuration is used when a higher pressure drop is required by the system. The figures show the plots for both parallel and series configurations of the fans [38].



(a)



(b)

Figure 3.4 Plots showing (a) series and (b) parallel configurations

3.4.4 Fixed flow

When the flow rate and temperature entering the device are known then a fixed flow is modeled through which the fluid is introduced to or extracted from the system. It should always be within the boundary of solution domain, surface of enclosure or on a non-collapsed cuboid. In chapter 4, a fancurve is used for selecting a fan and the below is the

3.4.5 Recirculation device

It is a combination of two parts: Extract and the supply. The working of recirculation device is to extract the flow from the solution domain at the given flow rate , cooling or heating the fluid and then re- supplying the treated fluid or flow at a different location. Both the parts are represented as a planar surface. They are modeled at the opposite sides of the cuboids or the boundary. The various methods for specifying air flow rate are:

1. Volume flow rate
2. Linear fan curve
3. Non-linear fan curve

Thermal properties: Any of the four can specified for air that passes through recirculation device. Following are the equations which calculate the temperature at the supply:

1. The change in temperature (ΔT) : $T_{Supply}=T_{Extract}+\Delta T$

2. Heat extraction rate: $T_{Supply} = T_{extract} + \frac{\text{Rate of heat extraction}}{(\text{Mass flow})\times(\text{Specific heat})}$

3. Heat exchanger

4. CRAC

Heat exchanger: It is also modeled as a recirculation device in the option of heat exchanger can be activated for entering the secondary inlet conditions. A heat exchanger can be modeled in two ways: simple heat exchanger and LMTD method with parallel/counter flow options.

a. Simple exchanger model

The air flows through the recirculation device can be regarded in contact with a secondary having a specified temperature. The heat flux extracted from the secondary added is determined$H(T_{Secondary} - T_{extract})$, the supply temperature is calculated by 2. The heat extraction rate is calculated as $\varepsilon C_{min} (T_{Liquid,inlet} - T_{air,inlet})$

Where ε is the heat exchanger effectiveness

C_{min} is the smaller heat capacity of the either fluids

$T_{coolant}$=$T_{liquid, inlet}$

$T_{extract}$= $T_{air, inlet}$

Thus, by the ε-NTU performance curve, the values for H and $T_{coolant}$ are computed iteratively.

b. LMTD model

This model is similar to the simple exchanger model but the fluid on the secondary side is not at a fixed temperature. Therefore, user mentions the flow rate and type of the fluid in parallel/counter flow type model.

3.4.6 Filters

These are devices that consists materials used for restricting the flow of air and capture the air borne materials and supply clean air to the system. They are generally located at places where the air is pushed in or pulled out of the system. It majorly depends on two factors: surface area of the filter and the speed of the fluid (air). Filters are widely used in electronics, computers, telecom, military area, medical field, power generation, industrial and HVAC. Selection of filter is an important step in designing as wrong selection can affect the electrical and thermal performance of the electronics housed by it. The main operating characteristics of a filter are its efficiency and pressure drop where efficiency is the percentage of air borne particulates that filter is capable of removing from the flow at a given velocity and pressure drop is a measure of force required to move air through the filter at the specified velocity. The system resistance is the sum of all the pressure drops in the systems including the pressure drop across the filter. This drop in pressure is a function of velocity of air and filter medium.

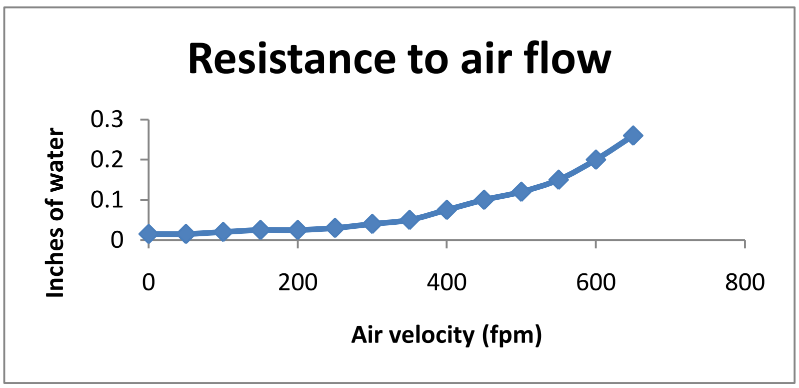


Figure 3.5 Plot showing the air flow resistance

CFD tool used for this study uses resistance for modeling the filter. These resistances can be collapsed/planar or non- collapsed/volumetric. The effect of pressure drop and velocity gradients are similar to that of the filter is established in the model which is possible by entering the loss coefficient and free area ratio for the filter. They can be specified in two ways: standard and advanced. The pressure drop in standard form is calculated as

$$\Delta p = \left(\frac{f}{2}\right) \times \rho \times \upsilon^2$$

In advanced option the loss coefficient is calculated as

$$k = \frac{A}{R_e} + \frac{B}{{R_e}^{index}}$$

And $R_e = \frac{[\rho \times \text{length scale} \times \text{velocity}]}{\text{viscosity}}$

Resistances are directly proportional to pressure drop and inversely proportional to the surface area and flow rate. These resistances can be placed in series or parallel configurations. When placed in series, the resistance increases and the flow rate is reduced. When placed in parallel, the resistance decrease and the flow rate is increase as the area is increased.

3.4.6.1 MERV ratings filters

MERV stands for Minimum efficiency reporting values, standardized by ASHRAE 52.2 for the selection of the filters. It is a number which is relative to the air filter's efficiency. Typically, the air filter is more efficient at removing particles when the MERV of the filter is higher. A maximum of 95% efficient filters are present [40]. The different MERV rated filters are used as per the requirements. The table below describes the different application and limitations of the each rated filter.

Table 3.1 MERV Rating Chart

MERV Rating	Dust Spot Efficiency	Typical controlled containment (pm)= 1 x 10^{-12} m	Typical applications and limitations
1-4	<20%	Carpet Fibers / Pollen >10 pm particle size	Minimal Filtration
5-8	<20- 35%	Cement dust 3-10 pm Particle size	Standard Industrial/ Commercial buildings
9-12	40-75%	Welding fumes	Superior Residential hospital laboratories
13-16	89->95%	Droplet Nuclei (Sneeze) 0.3 - 0.1 pm Particle Size	Surgery Superior Commercial Buildings
17 to 20	N/A	All Combustion Smoke	< 0.30 pm Particle Size Clean rooms

The MERV rated filters used in this study are MERV 8. These filters have an efficiency of 20-35% of collecting dust particles and they can collect particles sized as small as 3 microns.

CHAPTER 4

AIR COOLING OF DATA CENTER

4.1 Introduction

Data Centers are typical divided into four classes based on the environments required and Class 1 Data Center that presumes air conditioners for cooling is of interest in this study. ASHRAE [1] recommends the allowable conditions for air entering the cabinets. Figure 4.1 below shows the preferred temperature and humidity for a Data Center. ASHRAE guidelines recommend the server inlet temperatures (dry bulb) to be between 20- 25°C and the upper limit not to exceed 27°C with a relative humidity of 40-55%. Anyhow, these parameters do not ensure an optimum energy efficiency at which the Data Center is operated.

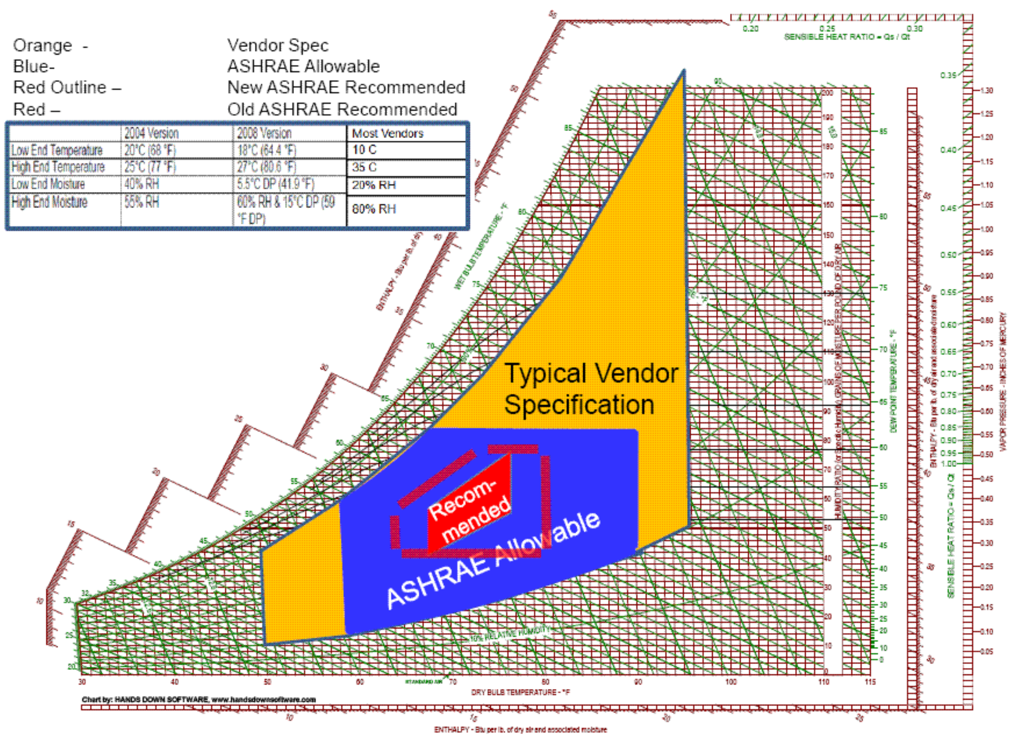


Figure 4.1: Recommended inlet conditions of air

4.2 Energy consumed in a Data Center

The energy consumed in Data Center is due to, UPS (uninterruptible power supply), power distribution units, IT equipments, cooling equipment, lighting and power backup [41]**.** The power (electric power) required varies from one Data Center to the other. The figure below exhibits the electrical requirement's breakdown in a Data Center due to the various loads in the Data Center. These loads are due to the direct expansion (DX) cooling system, the critical loads which include the all the loads due to the IT equipment i.e., the servers, computers, routers, storage devices, telecommunication equipment. Therefore, cooling of Data Center due to the heat released by the loads has become very important in Data Center.

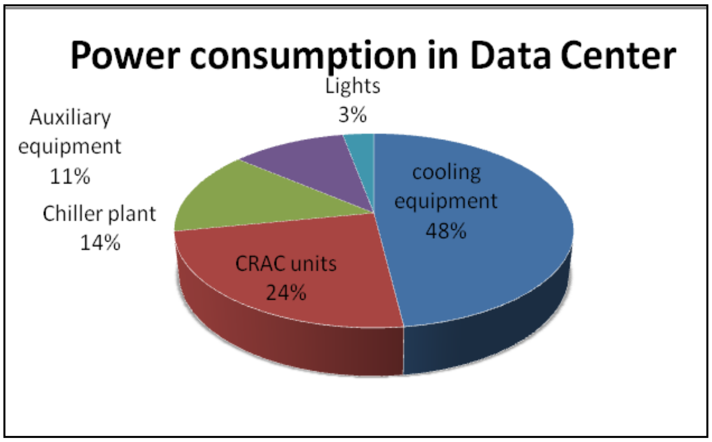


Figure 4.2 Power consumed by the equipments in Data Centers

For energy efficient Data Center, power effectiveness usage (PUE), cooling plant efficiency (CPE) and Data Center infrastructure efficiency (DCIE) are calculated by considering the power of the equipment, total power and power of cooling plant. Lower the value of PUE, higher is the efficiency. These ratios help in improving the Data Center's efficiency in operation and the design and processes. These are determined as

$$PUE = \frac{Total\ facility\ power}{IT\ equipment\ power} \tag{1}$$

$$DCIE = \frac{IT\ equipment\ load}{Total\ facility\ power} \times 100\%$$

(2)

Green grid [41] indicates that when PUE value is 1.2 and DCIE is 83%, Data Centers are most efficient however; these values are amended to 2.5 and 50% respectively for average working of datacenter. The IT equipment power corresponds to the loads of compute, storage and network equipment along with the switches, monitors and workstations. The total facility power is the power utilized solely by the Data Center room. It includes (a) power delivery components like UPS, switch gears, generators, PDUs, (b) cooling systems components like chiller plants, CRAC units, DX units, pumps and also cooling towers.

4.3 Construction of Data Center

Generally, Data Centers are cooled by the conventional technique of air cooling. In this technique air conditioners are used for cooling the air. According to Roger Schmidt et.al [43], Air flow management has a major impact on thermal environment. The most important requirement for Data Centers is the maintenance of inlet temperature and relative humidity to the specified values. In order to maintain such an environment commonly two ways are used for supplying air into the Data Centers: Over head supply and under floor supply. Both these techniques use hot/cold aisle layout. Hot aisle contains the hot air exhausted through the cabinets and cold aisle contains cold air that enters the cabinets.

a. Under floor configuration (Raised floor configuration):

In this configuration, cabinets are placed on a raised floor. Typical under floor Data Centers comprises of

1. Server racks/cabinets
2. Perforated tiles
3. CRAC (computer room air conditioning) units

The racks are arranged in rows and separated by perforated tiles. The inlets of the cabinets face the tiles through which air enters the cabinets supplied by CRAC units.

b. Over head supply configuration

In this configuration, air enters the inlet of the cabinets that are arranged in rows inlet facing the cold aisle, through the diffusers in the ceiling and exits through the walls of the Data Center. A comparison of both the configurations is made below.

The Data Center layout considered for this study is a raised floor configuration in which air is supplied through the CRAC units and it enters the cabinets through the perforated tiles and hot air released in the cabinets is sent back to the CRAC units and again cool air is supplied beneath the raised floor.

Table 4.1 Comparison of overhead and under floor supply configurations

Parameter	Over head supply	Under floor supply
Capacity	Limited by space, aisle velocity	Limited by free area of floor tiles
Temperature	Uniform	Bottom: cold and top: hot
Aisle capping	Hot or cold aisle	Hot or cold aisle
Energy cost	Best	Worst

The major challenge in using this configuration is the mixing of hot and cold air caused to the recirculation of air from the cold aisle to hot aisle thereby increasing the inlet temperatures of the cabinets.

4.4 Data Center numerical modeling

A half symmetric model of Data Center is shown in figure 4.3 which comprises of 20 cabinets [44] in a room of 44' in length, 20' in width and 12' in height. Cabinets are arranged in four rows A, B, C and D, having six, six, four and four cabinets/racks respectively. The inlets/front sides of cabinets of two adjacent rows face each other and are separated by perforated tiles of 2' in length and width. This aisle is called the cold aisle as the cold air is circulated through these tiles and the aisle facing the rear side of the cabinet is the hot aisle which contains the hot exhausted air. Each cabinet is 3.28’ in length, 1.96’in width and 6.56’in height. The cabinets are vented front and vented back and partially populated to 75%. Each server is considered to be 1U high. The Data Center used in this study has three CRAC units which act as a heat exchanger which receives hot air from the cabinets and supplies cold air back to the cabinets.

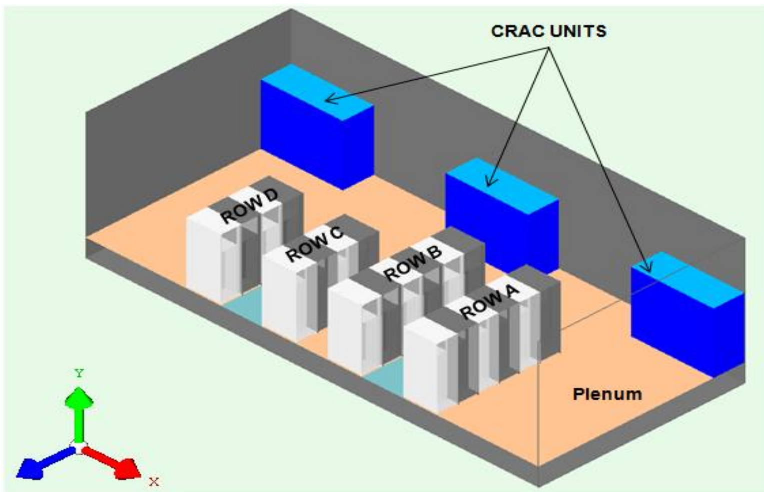


Figure 4.3 Isometric view of the Data Center

The objective is to determine the suitable cooling accessories that meet the design specification for the prescribed heat loads. The accessories considered for analysis are blanking panels and chimneys. Blanking panels prove to be advantageous when the racks are under populated. These restrict the path of air flow and avoid the recirculation of air within the cabinet which will in turn increase the cabinet temperature. Chimneys are convenient for letting the air take the prescribed way for maintaining the temperature of the cabinets. The study can be divided into sections: in the first section a comparative study is performed for finding the effect of blanking panel and chimney under different heat loads of the cabinets and later the effect of containment is observed. Several designs of experiments are complied for determining the operating power range for blanking panel and chimneys.

Following are assumptions considered for determining the maximum heat load of the cabinet

- The inlet temperature is assumed as 15°C
- The maximum difference in temperature in each cabinet is 45°C
- The maximum possible exhaust temperature of cabinet is 60°C
- Flow rate of air entering each cabinet is 965 cfm

Therefore the heat load is calculated using the formula $Q = \dot{m} \times C_p \times \Delta T$, which compiles to 24.4kW which is the maximum possible heat load for the following assumptions.

4.4.1 Determination of suitable cooling accessories for prescribed heat loads

Based on these assumptions, the heat load considered are 4kW, 8kW, 12kW and 16kW The analysis was performed four different configurations.

1. Cabinet with vented rear door and without chimney and without blanking panel
2. Cabinet with vented rear door and without chimney and with blanking panel
3. Cabinet with solid rear door and with production chimney and with blanking panel
4. Cabinet with vented rear door and with production chimney and with blanking panel

Each of the configurations is simulated for all the defined heat loads which totals to 16 experiments. The chimney inspected is specified by the company [45].Figure 4.4 represents the chimney modeled along with the blanking panel. However, placement of blanking panels depends on the cases run.

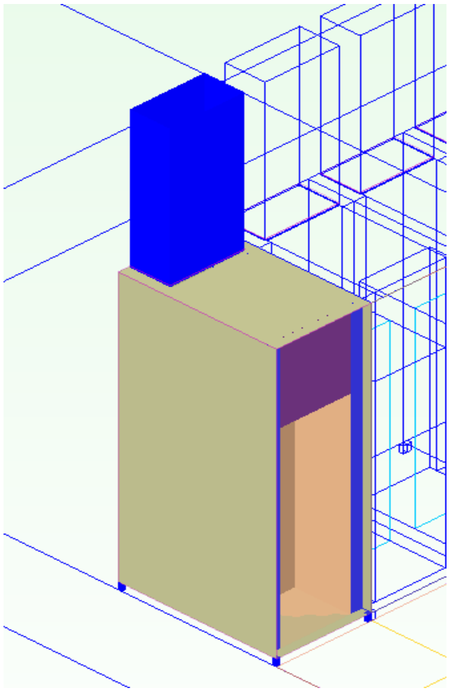


Figure 4.4 A model of chimney with blanking panel

4.4.1.1 Results

Following are the results obtained for the above configurations. For plotting the thermal plots the mid length of the C2-D2 shown below, is considered.

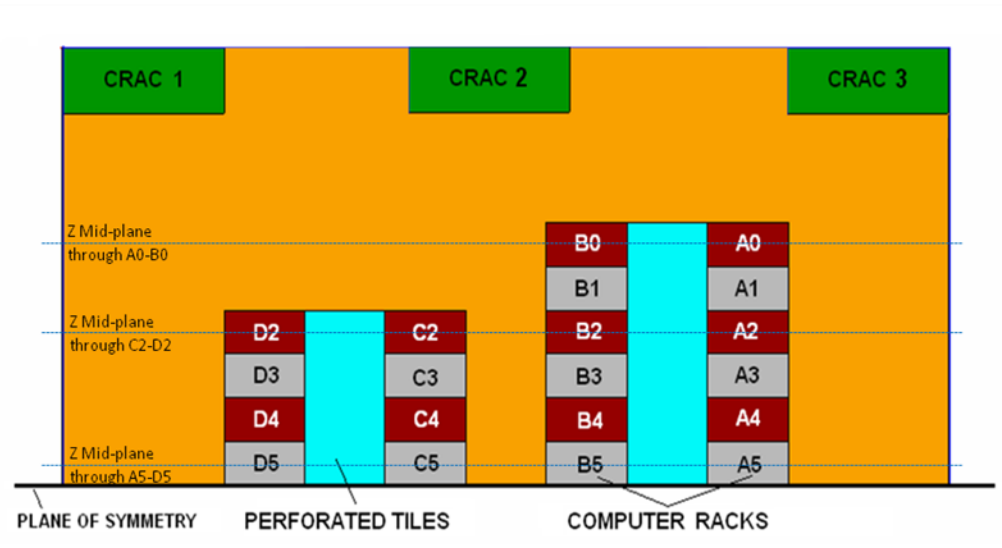


Figure 4.5 Top view of Data Center depicting the rows, and the plenums for reference

Case 1:

For configuration 1, following are the observations:

- For a power of 8 kW, all cabinets show a maximum temperature less than 60°C
- Inlet temperature is less than 30°C for all cabinets except the cabinets near the CRAC
- For power of 12 kW, all cabinets show maximum temperature less than 60°C, except end cabinets.
- For all power ratings inlet temperature remains close to each other for cabinets farthest away from the CRAC

Therefore, no blanking panel is required for 8 kW heat load cabinets. The following figures depict the thermal plots of this configuration which shows the necessity of blanking panels at higher heat loads. These plots are recognized at the mid length of the cabinet C2-D2.

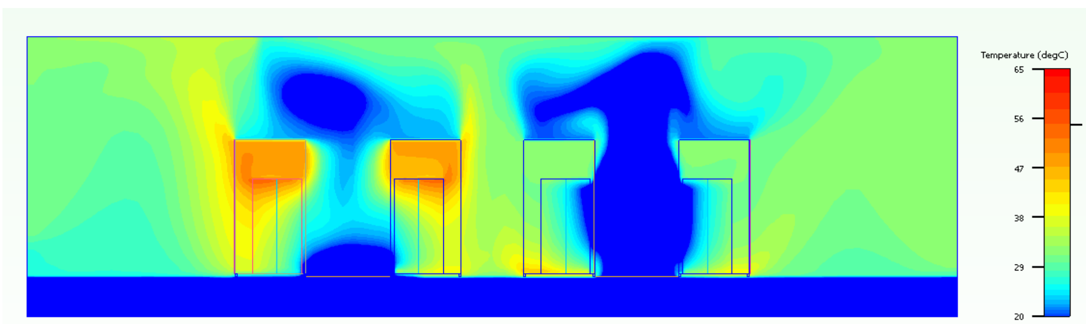


Figure 4.6 Temperature plot depicting the effect of cabinet with vented rear door and without chimney and without blanking panel for 8kW heat load

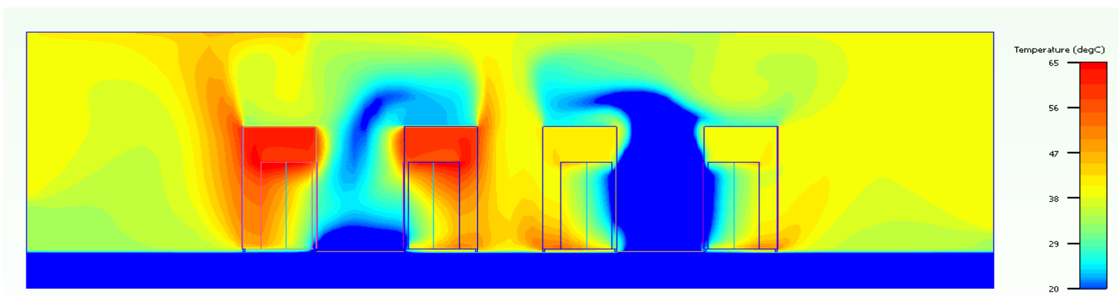


Figure 4.7 Temperature plot depicting the effect of cabinet with vented rear door and without chimney and without blanking panel for 12 kW heat load

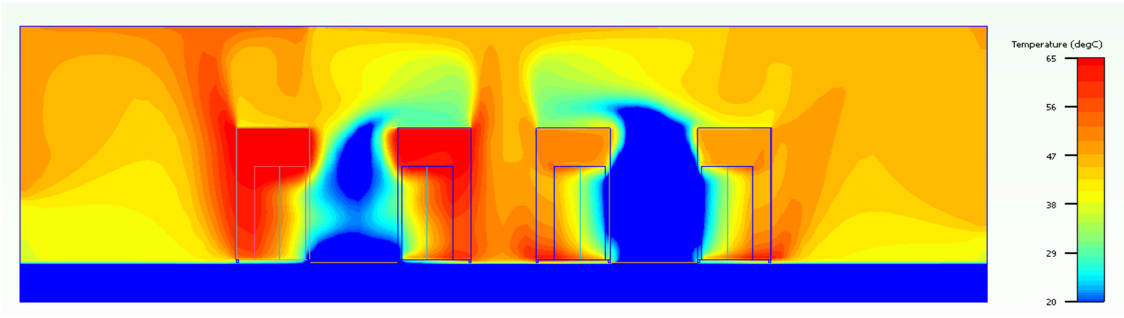


Figure 4.8 Temperature plot depicting cabinet with vented rear door and without chimney and without blanking panel for 16 kW heat load

Case 2: The effect of blanking of blanking panel is analyzed in configuration 2.

- The blanking panel cuts off the recirculation of hot air from the rear to the front side, inside the cabinet.
- By using a blanking panel the operating range for server heat load can be raised to 12 kW.
- For 12 kW the inlet temperature is less than 35°C

The following figures represent the thermal plots for this configuration for 12kW and 16kW to clearly show the importance of blanking panel. These plots are recognized at the mid length of the cabinet C2-D2

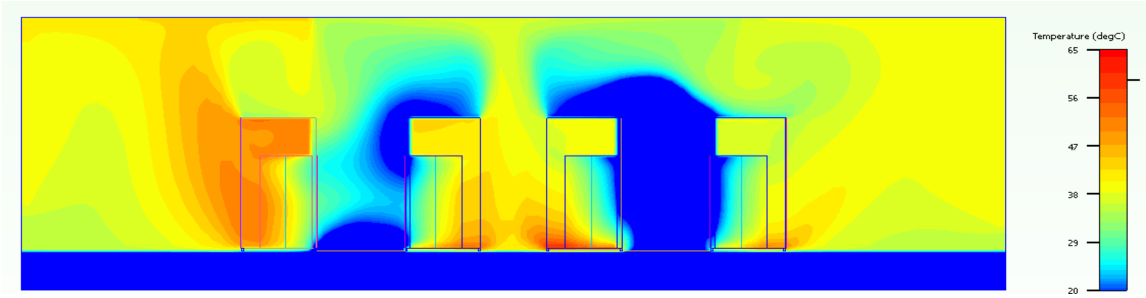


Figure 4.9 Temperature plot depicting cabinet with vented rear door and without chimney and with blanking panel for 12kW heat load

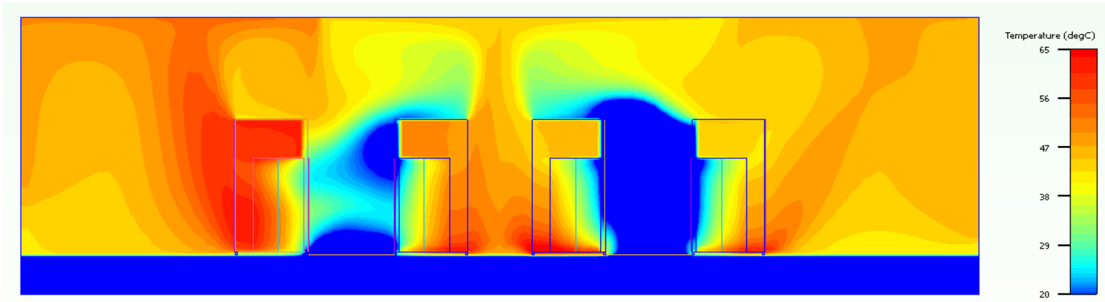


Figure 4.10 Temperature plot depicting cabinet with vented rear door and without chimney and with blanking panel for 16kW heat load

It is observed that though blanking panel is required but it is not effective for a high heat load of 16kW.

<u>Case 3:</u>

In configuration 3, the effect of blanking panel is investigated for the cabinet with solid rear and a production chimney and plots are shown for the heat loads of 12 and 16kW heat loads at the mid length of C2-D2 cabinets

- For 12 kW the maximum cabinet temperature is less than 60°C.
- Inlet temperatures are lesser than 35°C for the case of 12 kW.
- Cabinet with blanking panel, chimney and solid back gives temperature results which are similar to cabinets with blanking panel and vented back (No chimney).

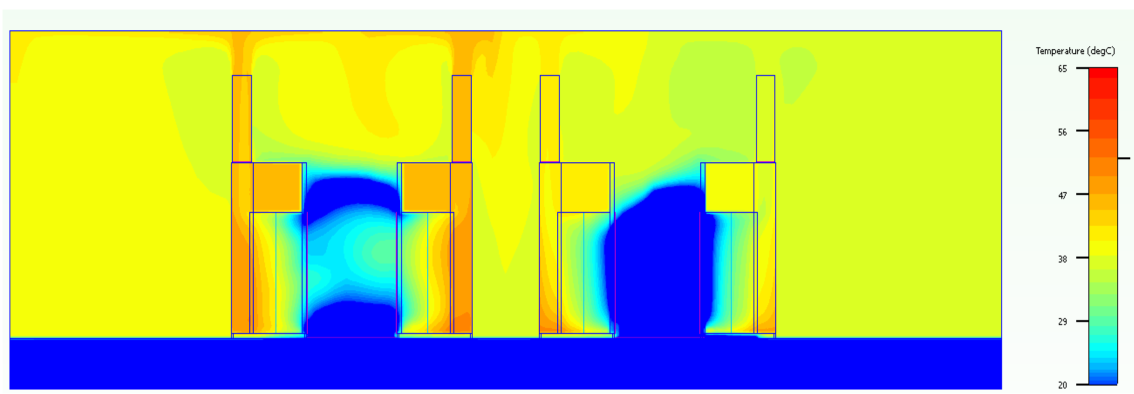


Figure 4.11 Temperature plot depicting cabinet with solid rear door and with production chimney and with blanking panel for 12 kW heat load

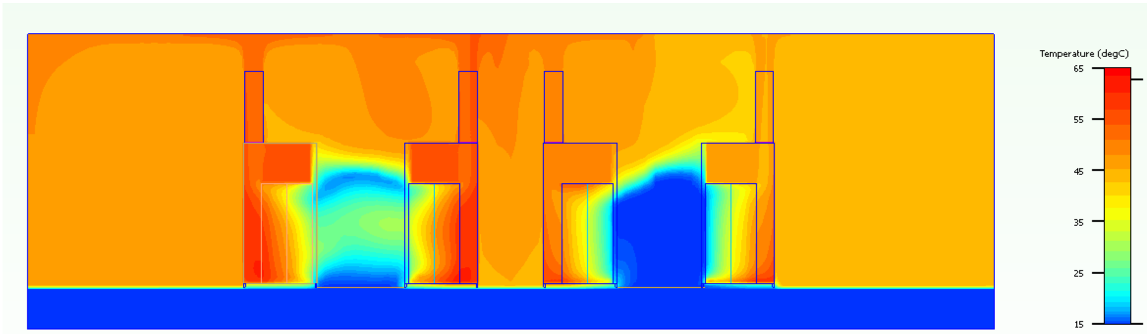


Figure 4.12 Temperature plot depicting cabinet with solid rear door and with production chimney and with blanking panel for 16 kW heat load

<u>Case 4:</u>

In configuration 4, Cabinet with vented rear door and with production chimney and with blanking panel is considered and it is noticed

- Decrease in maximum cabinet temperatures
- Cabinets close to the CRAC seem to be the limiting factor.
- Maximum cabinet temperatures less than 60°C is observed for most cabinets for 12 kW heat load.
- Inlet temperature is less than 35°C for most cabinets (for 12kW).

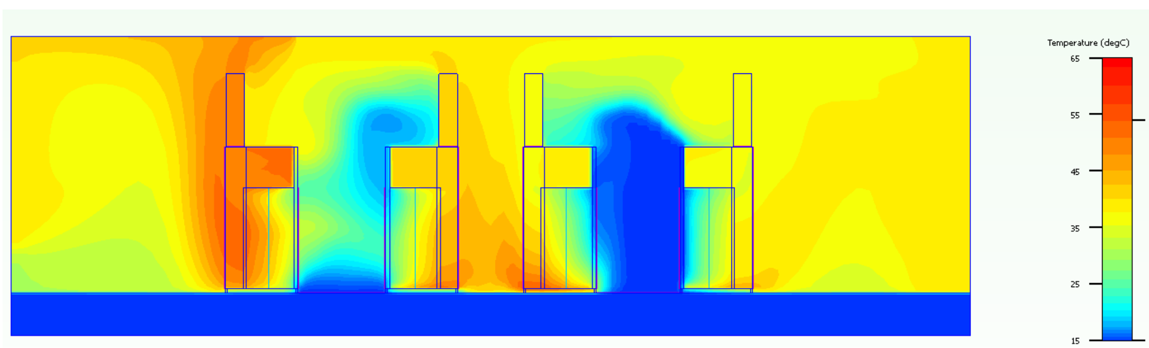


Figure 4.13 Temperature plot depicting cabinet with vented rear door and with production chimney and with blanking panel for 12kW heat load

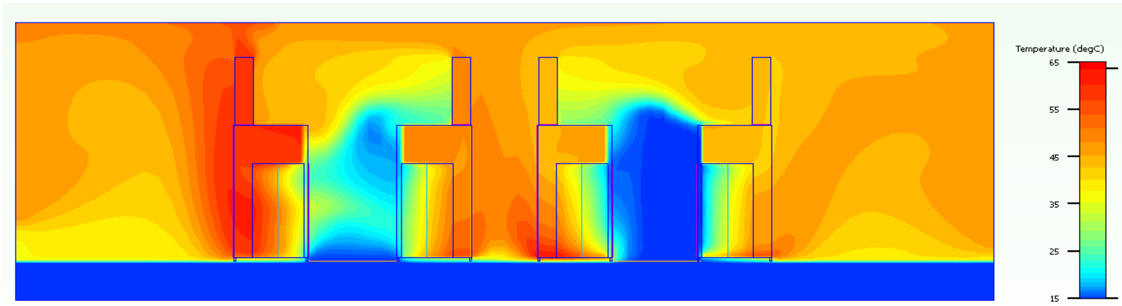


Figure 4.14 Temperature plot depicting cabinet with vented rear door and with production chimney and with blanking panel for 16kW heat load

From the above analysis it can be concluded blanking panels and chimneys are most effective for heat loads up to 12kW. Table below summarizes the effect of these accessories in different configurations considered. The table is read in the following way: for e.g. configuration1 (in parentheses) is suitable for heat loads only for 4 and 8kW heat loads.

Table 4.2 Effectiveness of blanking panel and chimney in different cases

Heat Loads (kW)	Vented back Without accessories Case (1)	Vented back with blanking panel Case (2)	Solid back with production chimney and blanking panel Case (3)	Vented back with production chimney and blanking panel Case (4)
4	Yes	Yes	Yes	Yes
8	Yes	Yes	Yes	Yes
12	No	Yes	Yes	Yes
16	No	No	No	No

4.4.2 Impact of kick plate

There are stands situated on which the cabinets rest. Due to the circulation of hot and cold air through the cabinets, some air is escaped through the gap between the plenum and the cabinet. Hence, it attributes to an increase in inlet temperatures. The figure below represents the vector plot depicting the mixing of hot and cold air beneath the cabinet. This can be avoided by placing a kick plate at the bottom of the cabinet facing the inlet. Hence, kick plate is used for avoiding the mixing of air.

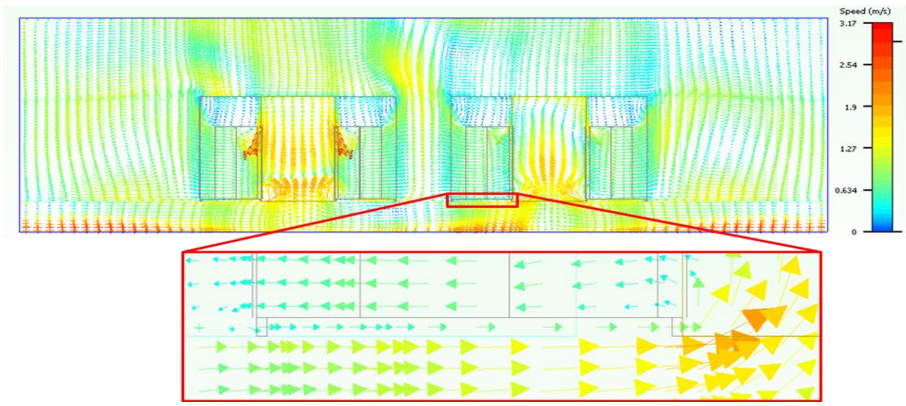


Figure 4.15 A Vector plot depicting the infiltration of air

4.4.3 Impact of cabinet fans in cooling Data Centers

Data center cabinets are thermally analyzed by placing a fan tray consisting of fans at the top of the cabinet. Flow rate for server fans is 965 cfm and no blanking panel is used. The figure below represents the cabinet with cabinet fans placed at the top end of the cabinet.

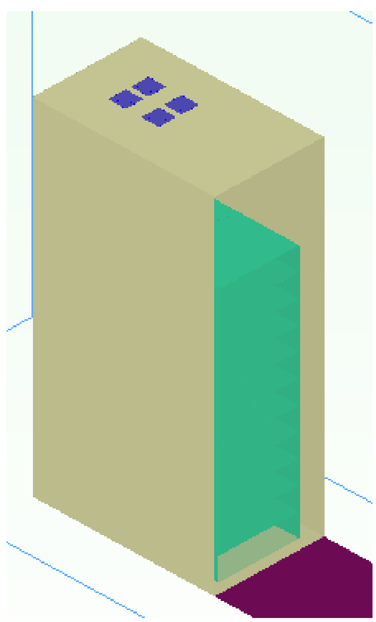


Figure 4.16 Cabinet with four fans placed in a fan tray

A sum of the flow rates of these four fans is considered for a single fan that is placed at the cabinets in the above mentioned Data Center. An isometric view of the Data Center is showed in figure 4.17.

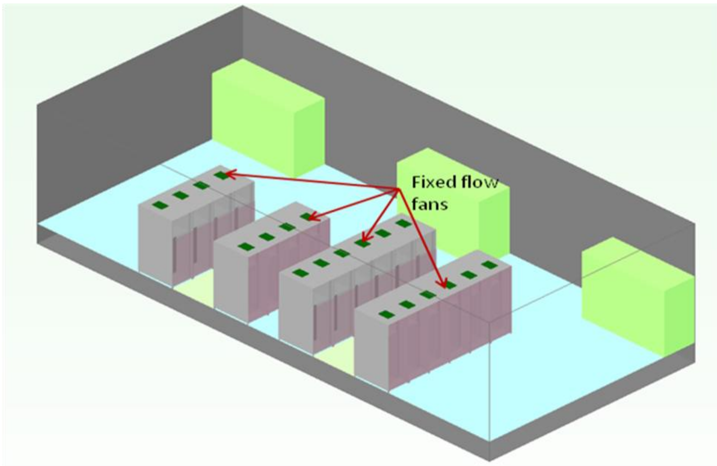


Figure 4.17 An isometric view of the Data Center

4.4.3.1 Results

Data center is thermally analyzed for heat loads of 12, 16, and 20 kW respectively. The maximum inlet temperatures for all the three cases are plotted below.

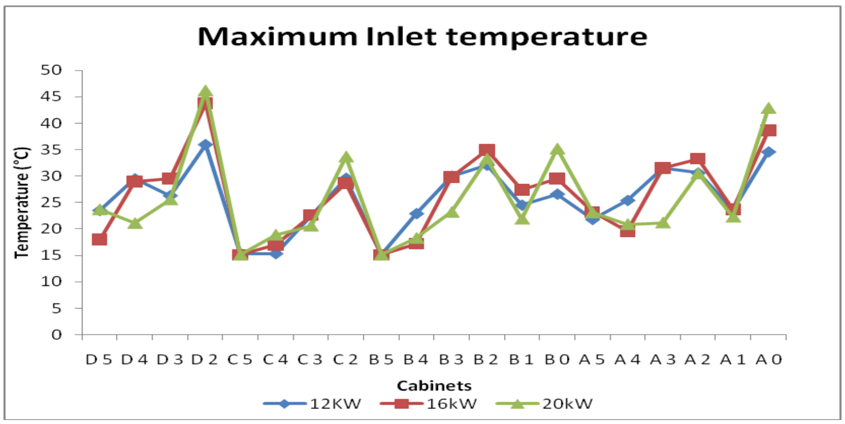


Figure 4.18 Plot depicting the maximum inlet temperatures for 12kW, 16kW and 20kW heat loads

From the above graph it is observed that the fans are effective for cooling cabinets of 12kW. A higher temperature is observed at cabinets located near the CRAC units and this is due to the inlet of high velocity air. The thermal plots below depict the maximum temperatures in the Data Center. It is clear from the thermal plots that the temperature in the Data Center increases with the heat load.

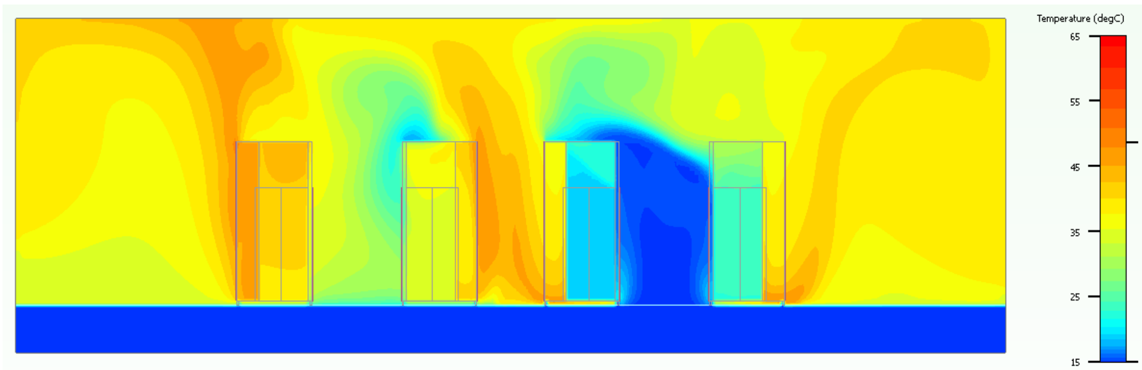


Figure 4.19 Thermal plot for cabinet with cabinet fans for 12kW

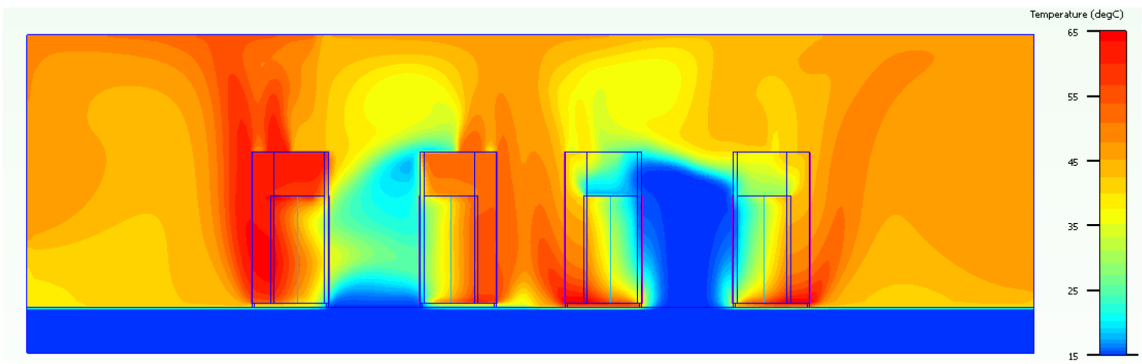


Figure 4.20 Thermal plot for cabinet with cabinet fans for 16kW

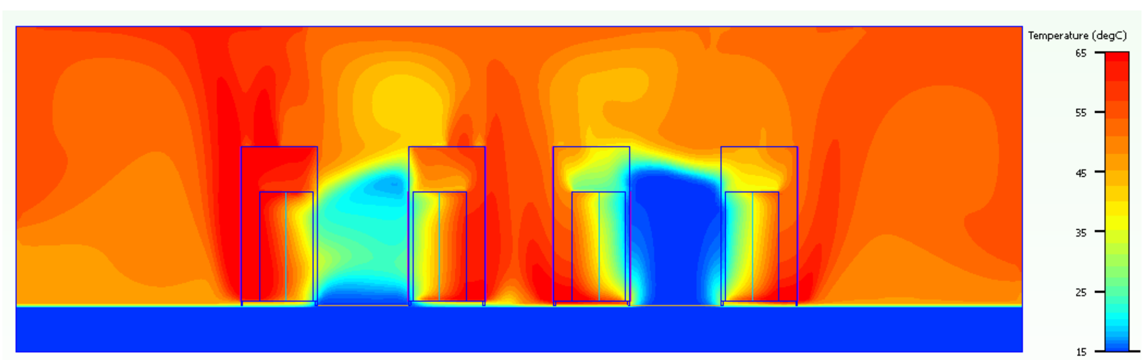


Figure 4.21 Thermal plot for cabinet with cabinet fans for 20kW

4.4.4 Investigation of effect of containment for high heat loads

Furthermore, analysis is performed for finding the suitable method for cooling these higher loads and containment technique is incorporated for these heat loads. The designs of experiments for this part of the study are:

Table 4.3 Design of Experiment (DOE)

Case	Rear door	Chimney	Blanking Panel	Power rating	Containment
1.1	Solid	Production	Yes	16 kW	Hot aisle
1.2	Solid	Production	Yes	20 kW	
2.1	Solid	Production	Yes	16 kW	Hot aisle with a duct
2.2	Solid	Production	Yes	20 kW	
3.1	Vented	No	Yes	16 kW	Hot and cold aisle
3.2	Vented	No	Yes	20 kW	
4.1	Vented	No	Yes	16 kW	Cold aisle
4.2	Vented	No	Yes	20 W	

4.4.4.1 Results

<u>Case 1:</u> In this configuration, a plenum is placed at the exit of the chimney which allows the hot air from the chimney to directly enter the CRAC units without mixing with cold air. Figure 4.18 shows the model of Data Center with hot aisle containment

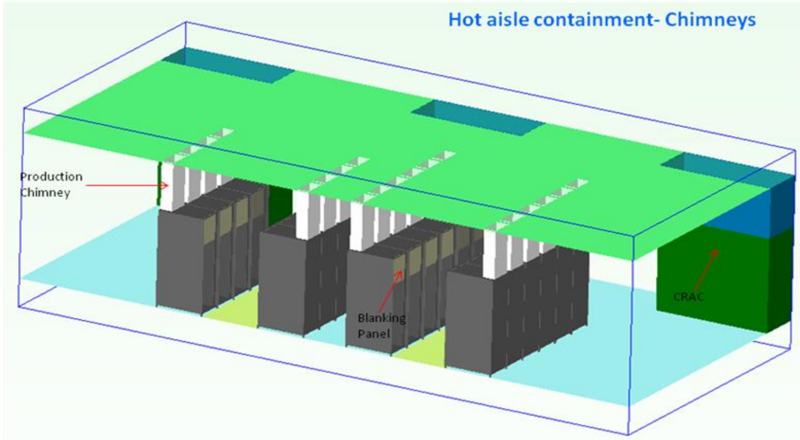


Figure 4.22 Data Center model with hot aisle containment using chimneys and plenum

A thermal plot is plotted for visualizing the effect of the hot air containment and the maximum global temperature in the Data Center.

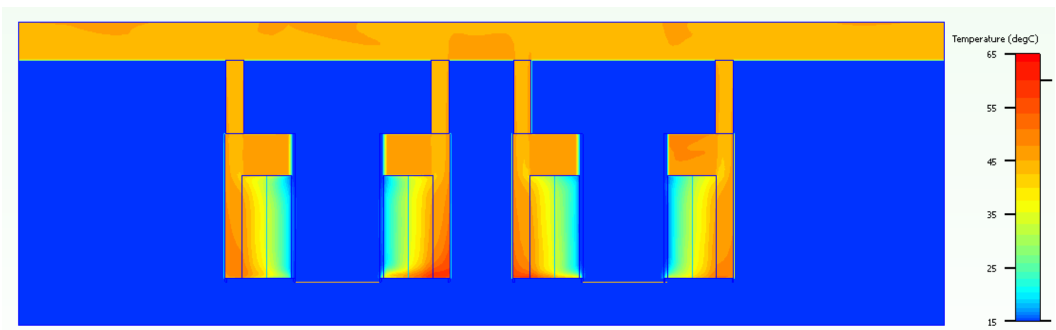


Figure 4.23 Thermal plot for cabinet with a chimney and a top plenum for hot aisle containment for 16 kW heat load

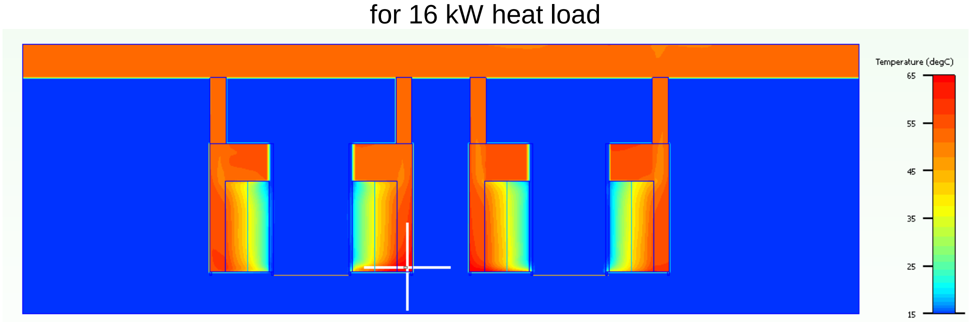


Figure 4.24 Thermal plot for cabinet with a chimney and a top plenum for hot aisle containment for 20 kW heat load

Case 2: Hot aisle containment with a duct for restricting the path for the air above the chimney is considered. Figure 4.20 represents the Data Center with ducting for flow of hot air

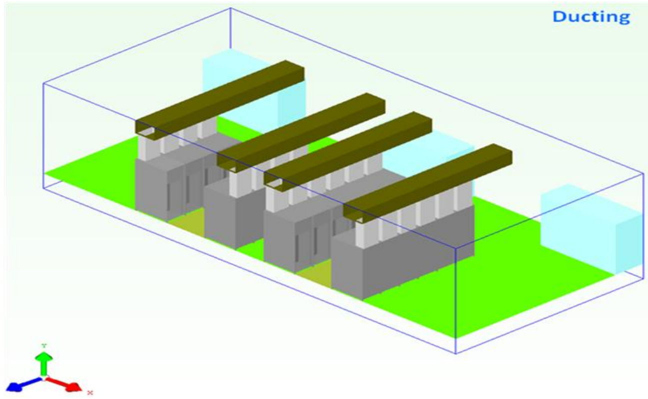


Figure 4.25 A Data Center model representing the chimney with ducting

The effect of ducting is shown in the figure 4.21 and 4.22 for 16 and 20 kW heat loads.

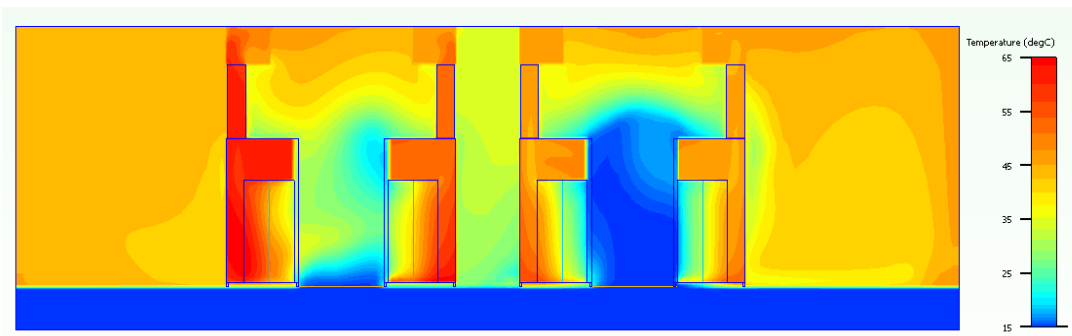


Figure 4.26: Thermal plot for cabinet with a chimney and a duct for hot aisle containment for 16kW heat load

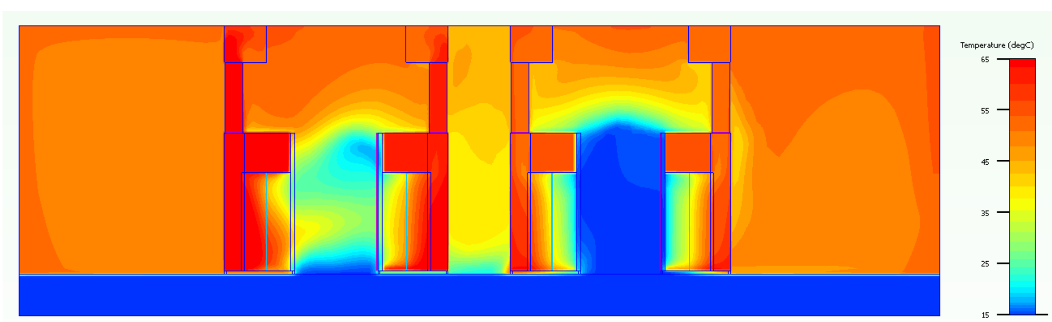


Figure 4.27 Thermal plot for cabinet with a chimney and a duct for hot aisle containment for 20 kW heat load

Case 3: Cabinet with both hot and cold aisle containment where the both the cold and hot aisle are restricted for reducing the global maximum temperature and maintain the inlet temperatures of the cabinets. Figure below depicts Data Center with such a configuration

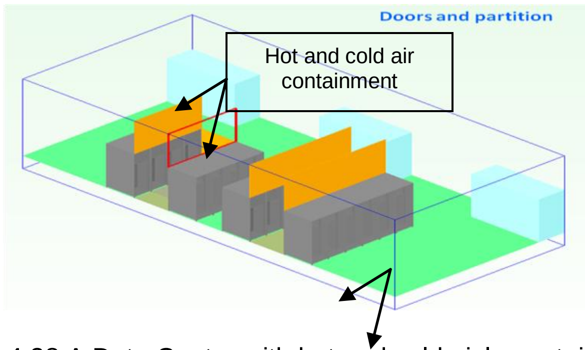


Figure 4.28 A Data Center with hot and cold aisle containments

These configurations are generally for maintaining the inlet temperatures of the cabinets and avoid the recirculation of the hot and cold air which increases the temperature. The following figures depict the effect of this configuration on Data Centers.

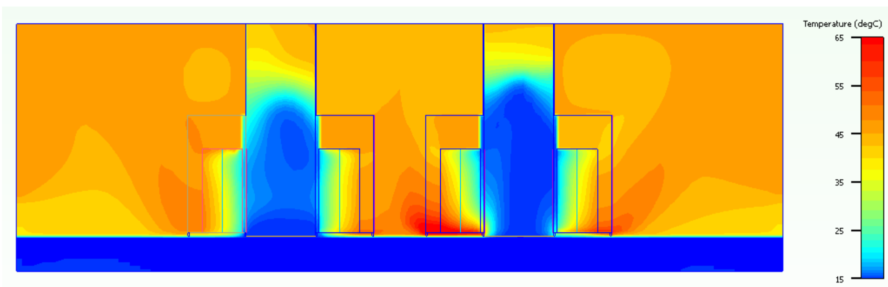


Figure 4.29 Thermal plot depicting the hot and cold aisle containment of the Data Center for 16kW heat load.

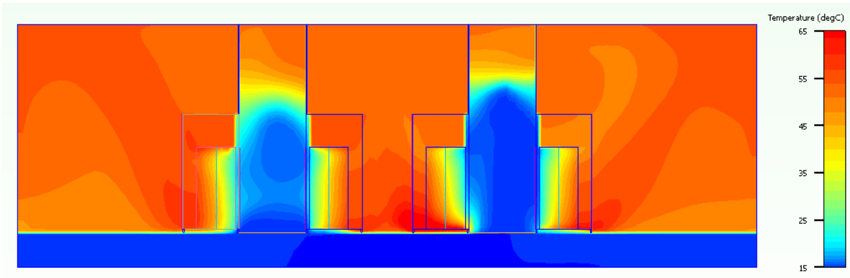


Figure 4.30 Thermal plot depicting the hot and cold aisle containment of the Data Center for 20kW heat load.

Case 4: The Data Center with cold aisle containment is considered for finding the thermal impact on the temperatures of the cabinets. The schematic of the configuration is shown below.

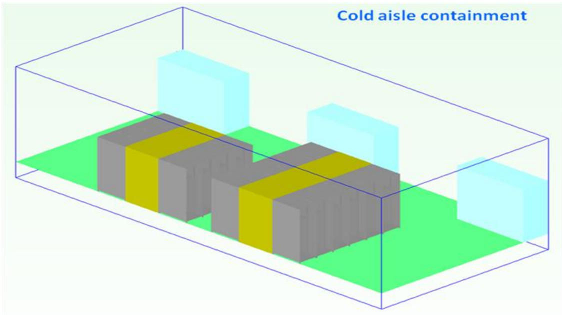


Figure 4.31 A Data Center model with the cold aisle containment

The next two figures represent the thermal plots of the Data Center with this configuration.

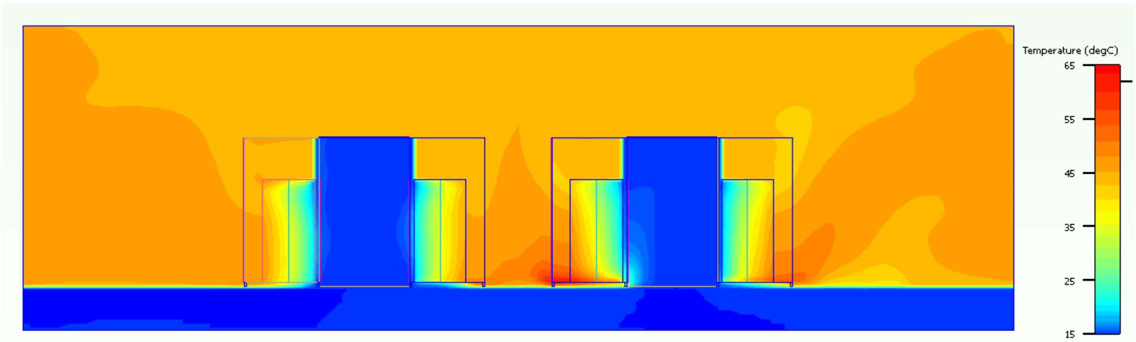


Figure 4.32 Thermal plot depicting the cold aisle containment for 16kW heat load Data Center.

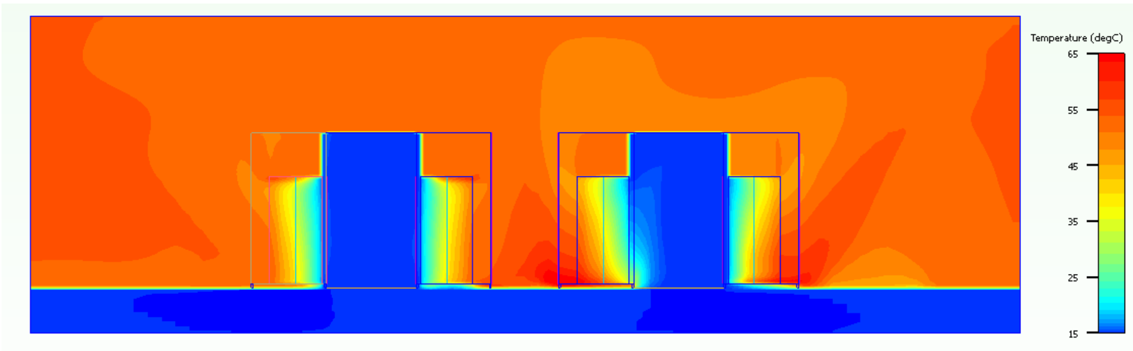


Figure 4.33 Thermal plot depicting the cold aisle containment for 20 kW heat load Data Center.

CHAPTER 5

REAR DOOR HEAT EXCHANGER: EFFECT, PERFORMANCE AND APPLICATION IN DATA CENTER

5.1 Introduction

Cooling of data center is more challenging as the heat loads increase. As the heat density of cabinets increase, efficiency of air cooling decreases as the high temperature gradients contribute to ineffectiveness of air cooling for high powered clusters. Cooling of racks is primarily managed by under populating the racks or spreading the heat loads. These units can cool up to 3KW/rack [16].

As heat dissipated by the racks increases, the hot spots in the cabinets also increase. Therefore, a hybrid cooling technique is incorporated for better cooling of the cabinets. Hybrid cooling solution is a combination of both air and liquid cooling. The air cooling is associated with a liquid to air heat exchanger.

Liquid cooling is an efficient alternative for air cooling. Generally, water is used as a coolant in this technique. Owing to its high heat capacity in comparison to air, it is a very efficient method of transporting heat. On a volumetric basis, it removes 3500 times much heat than air. Heat is extracted from the racks by the liquid (water). By using this technique, the utility of CRAC units can be reduced. It is noted that 80% of IT area can be reduced by using liquid cooling. The study is performed in different steps:

1. Verify the 'Rule of thumb' for heat loads of 18kW and 20kW cabinets in data center
2. Analyze the effectiveness of heat exchangers for different configurations of data center for a heat load of 18kW.
3. To determine the impact of heat exchanger in side breathing switches.

5.2 Types of a heat exchanger

Heat exchanger is device used for efficient heat transfers from one medium to the other. Its applications include in space heating, refrigeration, air conditioning, power and chemical plants. It transfers heat without transferring the fluid carrying the heat. Heat exchangers are classified by their flow as follows [46]:

- Parallel flow: The fluid enters from the same end and travels parallel.
- Counter flow: The fluid enters from the opposite ends. This is considered to be the most efficient way of cooling as it can transfer most of the heat. The fluids are perpendicular to the one another through the exchanger. As the surface area of wall between the fluids increase, efficiency of the heat exchanger increases and minimizes the resistance.

Different types of heat exchanger include:

1. Shell and tube type heat exchanger

   It consists of series of tubes, of which one set has fluid to be cooled/heated and the other set has the fluid that runs above it to provide/absorb heat.

   These tubes are longitudinal finned, plain. Fins on tubes increase the surface area of heat exchanger to provide a higher efficiency. It is apt for higher pressure application.

   The design features of this heat exchanger considered are

   - Tube diameter: smaller the diameter, economical and compact is the heat exchanger. For determining the diameter, the space, cost and fouling nature of the fluids is considered.
   - Tube thickness: It is determined for providing
     - Space for corrosion
     - Resistance for flow induced vibration
     - Axial strength
     - Strength to withstand internal tube pressure

- Strength for over pressure in the shell (Buckling strength)

— Tube length: Ideal heat exchanger has a smaller shell diameter and a long tube.

2. Plate heat exchangers
3. Adiabatic wheel heat exchanger
4. Plate fin heat exchanger
5. Phase change heat exchanger
6. HVAC air coils
7. Direct contact heat exchangers

5.3 Thermodynamics of heat exchanger

Heat exchanger works on the law of thermodynamics. According to the first law of thermodynamics when the body is heated, it dissipates heat to the surrounding till the system and surrounding are in equilibrium. If the heat exchanger has pipes of length L and a heat capacity of C_i and mass flow rate j_i. Let T_1 and T_2 be the temperatures of the pipes.

Then the thermal energy for a small volume u along the distance x is given by

$$\frac{du}{dt} = \Upsilon(T_1 - T_2)$$, where Υ is the thermal connection

5.3.1 Determining the effectiveness of the heat exchanger

The effectiveness of the heat exchanger is calculated with the following assumptions.

1. Heat exchangers operates at steady state conditions
2. Heat losses to and fro in the exchangers are neglected (adiabatic)
3. Temperature of each fluid is uniform
4. Uniform thermal resistance.
5. Heat conduction in the fluids and walls are negligible

The heat loads for both fluids are calculated by considering below input variables.

- Air inlet temperature is32^0C
- Water inlet temperature is 18^0C

- By using the basic energy equation

$$Q = \dot{m} \times C_p \times \Delta T$$

[1]

The properties of air (hot fluid) and water (cold fluid) are as shown in table 1.

Table 5.1 Input values of Air and water for calculating the heat loads.

Parameters	Air(h)	Water(c)
Inlet temperature(Ti)(^{0}C)	32	18
Δ T(Outlet-Inlet)(^{0}C)	20	5.8
Outlet temperature(To)(^{0}C)	52	23.8
Flow in gpm	-	11.89
Flow in cfm	2000	1.5813
Converted in m^3/sec	0.9332	0.000741
Density in kg/m^3	1.028	998.77
Mass flow rate(kg/sec)	0.96	0.74
Specific heat J/kg-K	1005	4182
Heat capacity W/K	964	30937
Heat load(Q) in W	18000	18000

From the table above, Q_h>Q_c,

From the formula

$$Effectiveness(\varepsilon) = \frac{Q_h(T_{hi} - T_{ho})}{Q_{min}(T_{hi} - T_{ci})} = \frac{Q_c(T_{co} - T_{ci})}{Q_{min}(T_{hi} - T_{ci})}$$

[2]

$\Rightarrow Qc = Qmin\ (from\ the\ above\ table)$

$$\therefore (\varepsilon) = \frac{Q_c(T_{co} - T_{ci})}{Q_{min}(T_{hi} - T_{ci})}$$

[3]

$$\Rightarrow (\varepsilon) = \frac{(T_{co} - T_{ci})}{(T_{hi} - T_{ci})}$$

[4]

$$(\varepsilon) = 0.58$$

This effectiveness is considered as an average value for all the flow rates, which are varied for different heat loads. The various flow rates for different heat loads are tabulated in table 2. Also, the conductance is calculated by the formula

$$Conducatnce\ (h) = \varepsilon \times C_{min} \times C_p(\mathrm{min})$$

[5]

Where, C_{min} is the minimum heat capacity of air or water and the specific heat of the same.

5.4 Modeling of rear door heat exchanger in flotherm

Heat exchanger is modeled as a recirculation device. The computational model of Data center considered for modeling is a half symmetry model. The data center is of 44' in length, 20' in width and 12' in height. It consists of 20 cabinets arranged in four rows. Rows are named as A, B, C and D. Rows A and B consists of six cabinets each, C and D four each. Space is provided for accommodating cabinets in future. These cabinets are of length 3.28', width 1.96' and height 6.56'. The cabinets are vented front and vented back. These cabinets are placed on the plenum that consists of perforated tiles to allow air to enter the cabinets. These tiles are 2' in length and width. The cabinet in each row are placed such that the front doors of cabinet face the perforated tiles. Since cold air enters through these tiles it is called as cold aisle and the rear side from where the hot air is exhausted as hot aisle.

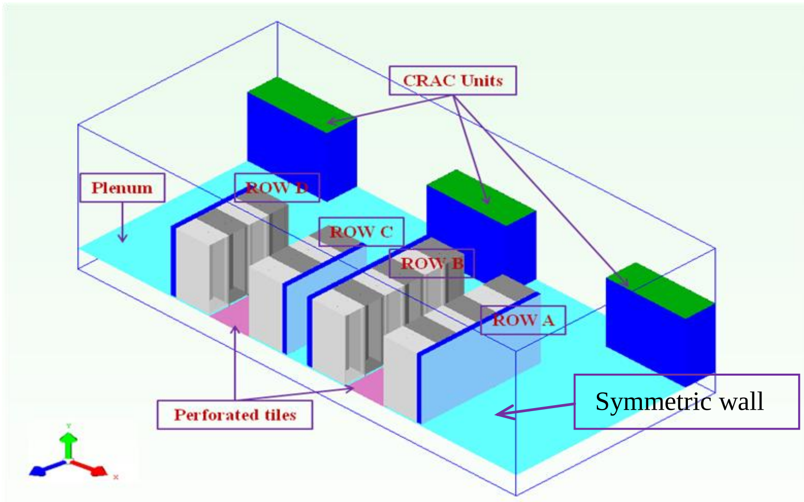


Figure 5.1 Isometric view of data center

The data center consists of CRAC units for supplying air to and extracting from the cabinets. The operation of CRAC units is varied as per the requirement. Rear door heat exchanger is modeled as a recirculation device with the water inlet temperature of 18^{o}C [47]. Following are the design of experiments for analyzing the working of rear door heat exchanger.

5.4.1 Rear Door Heat Exchanger

Rear door heat exchanger is one of the efficient techniques for cooling and in conservation of energy. It is a passive cooling technique which has less moving parts, no fans and no electricity needed (except for the pump to circulate water), making it more energy efficient and saving the total power of the data center. It is noted that the rear door heat exchangers can save more than 50% on the total cost of data center cooling compared to remaining techniques. The main advantage of using rear door heat exchangers is the reduction of the hotspots caused to the higher gradients in temperatures [47] and for increasing the heat density in the cabinets which helps in using high performance data center. RDHx are mounted at the rear side of the cabinet to cool the air exhausted from cabinets. The basic principle of operation of a heat exchanger is it transfers heat without transferring the fluid carrying the heat [48]. Water is supplied and returned through the Cooling distribution units (CDU). Water supplied must meet the requirements for optimal cooling performance and system reliability.

These specifications include quality, temperature, pressure and flow. The required flow rate and temperatures are specified in [48]. The water to air type heat exchanger is used in this study. The cooling performance of RDHx is given in figures 5.1 and 5.2, [49]. It is noted that the RDHx can cool 100% up to 27KW of heat load.

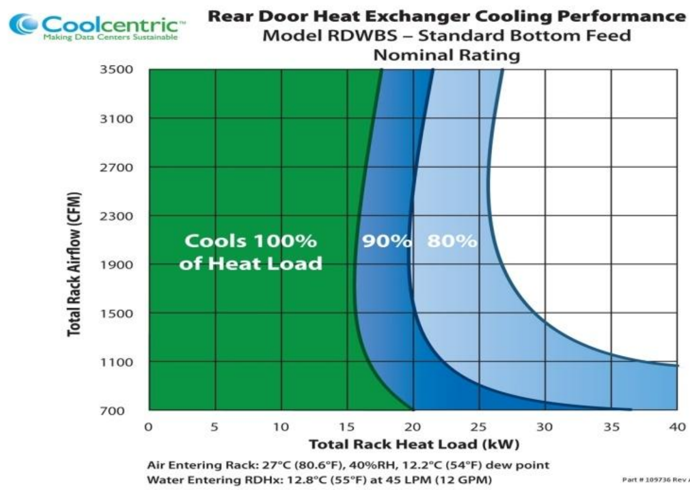


Figure 5.2 RDHx-Cooling performance for nominal cooling.

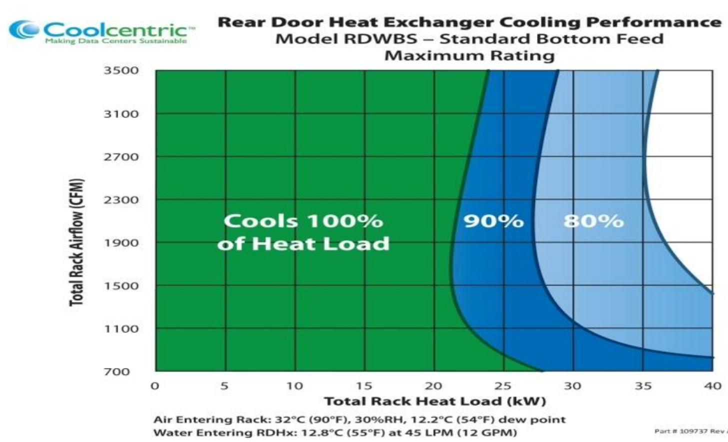


Figure 5.3 RDHx-Cooling performance for maximum cooling.

5.4.2 Rule of Thumb verification

The objective of this part of study is to verify the rule of thumb "rear door heat exchangers can be 100% efficient in cooling the data centers for heat loads of 27kW" [48]. The effectiveness of the heat exchanger is determined to be 0.58. This effectiveness is considered to be constant for all the experiments and only flow rates for the different heat loads are varied for finding the effectiveness of the heat exchanger. The flow rates of 18 and 20kW heat loads are calculated as tabulated below.

Table 5.2 Flow rates for the various heat loads

Case	Heat load (kW)	Mass flow rate (kg/sec)	Volumetric flow rate (cfm)	Conductance (W/K)
1	18	0.95	2000	559.19
2	20	0.95	2000	559.19

5.4.2.1 Results

The thermal plots in the following figures show the effect of RDHx in data centers for cabinets of different heat loads.

Case 1

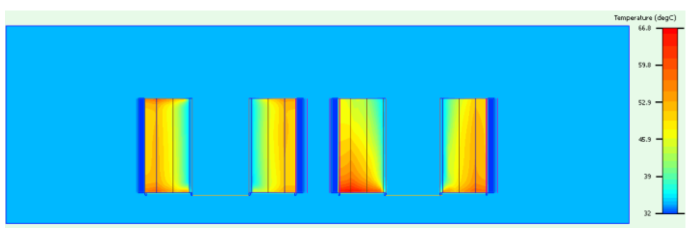


Figure 5.4 Temperature plot for case 1: 18 KW heat load

From the above thermal plot, it is observed that the heat exchangers can cool 100% efficiently and the inlet temperature at a steady state condition is 34^{o}C.

Case 2

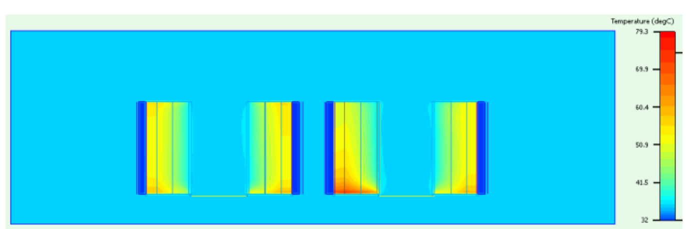


Figure 5.5 Temperature plot for case 2: 20 KW heat load

A similar trend is observed as in case 1. The inlet temperature a steady state condition is 34°C. The higher heat loads attribute to an increase in the inlet temperatures in the data center. Hence, the Rule of thumb is verified.

5.4.3 Impact of RDHx in different configurations

Additionally, different configurations are modeled for determining the impact of heat exchangers in data centers. Different configurations of data center are considered and studied for comparing the effectiveness of heat exchanger. Table below lists the design of experiments.

Table 5.3 Design of experiments

Case	Heat load (KW)	Flow rate (cfm)	Conductance (W/K)	CRAC unit	CRAC unit
1	18	2000	--	Yes	No RDHX
2	18	2000	559.19	Yes	With CRAC unit
3	18	2000	559.19	No	Fixed flow fans at the CRAC
4	18	2000	559.19	No	No Under floor Plenum

The different scenarios are analyzed for determining the effect of heat exchanger.

5.4.3.1 Results

Below are the graphs plotted for comparing the extract and supply temperatures for cases 2, 3 and 4 as case 1 is the baseline of typical cooling with CRAC units and no RDHx.

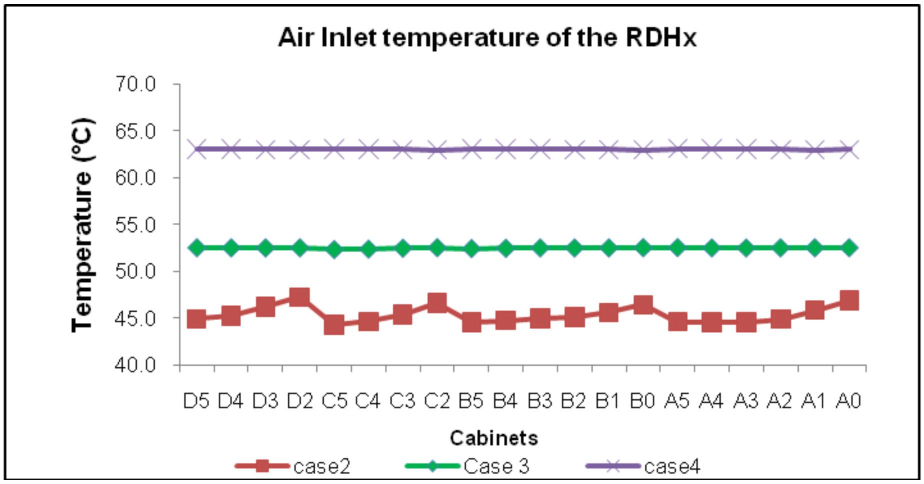


Figure 5.6 The extract temperatures in the data center

On comparing the extract temperatures for cases 2, 3 and 4, it is noticed that the temperature is least in case 2 which incorporates both heat exchanger and CRAC units for cooling and as expected the maximum extract temperature is recorded in case 4 where there is no mechanical supply of air for the cabinets. Similarly, the supply temperature to the data center from the RDHx is plotted as shown in figure 5.7.

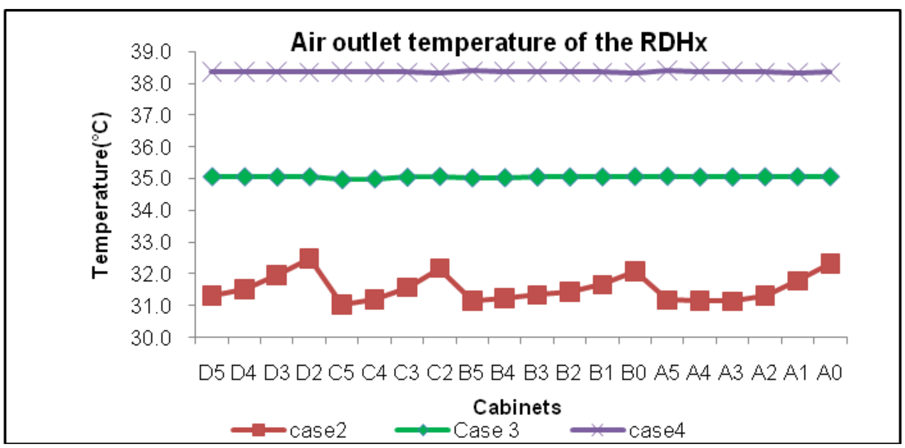


Figure 5.7 The supply temperatures in the data center

The results are very similar to that of the figure 5.6 and this is due to the no optimum utilization of heat exchanger due to the CRAC units. The thermal plots below depict the need of RDHx. Figure below is the thermal plot for case 1 where the cabinets are placed with CRAC units and without any RDHx.

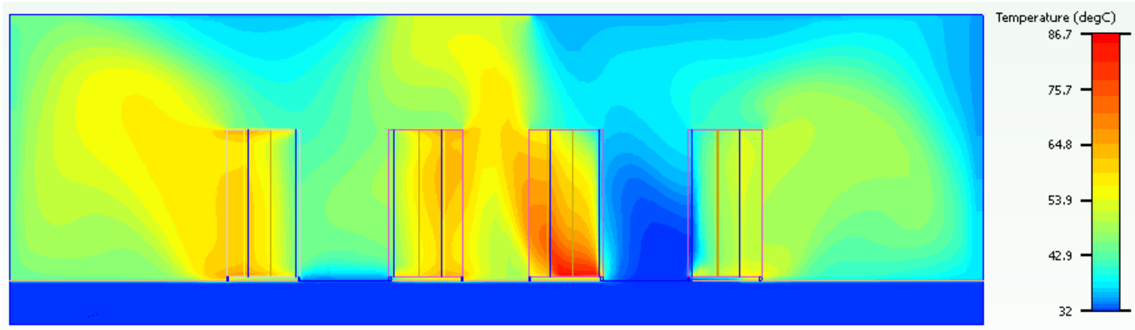


Figure 5.8 Temperature plot for case 1:18 KW heat load with CRAC unit and no RDHx

It is noticed that the temperatures are high and a high temperature in the hot aisle which mixes with the cold air and raises the temperature. This effect of recirculation is reduced by placing a RDHx. Figure below shows a reduction in the global maximum temperature due to RDHx.

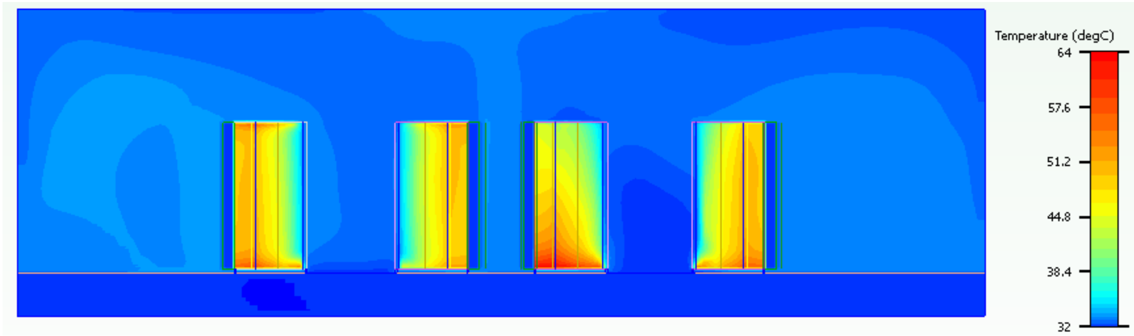


Figure 5.9 Temperature plot for case 2:18 KW heat load with CRAC unit and RDHx

Now, the CRAC units are replaced by axial fans which supply the required amount of air into the room. The figure below shows the effect of heat exchanger in such a configuration.

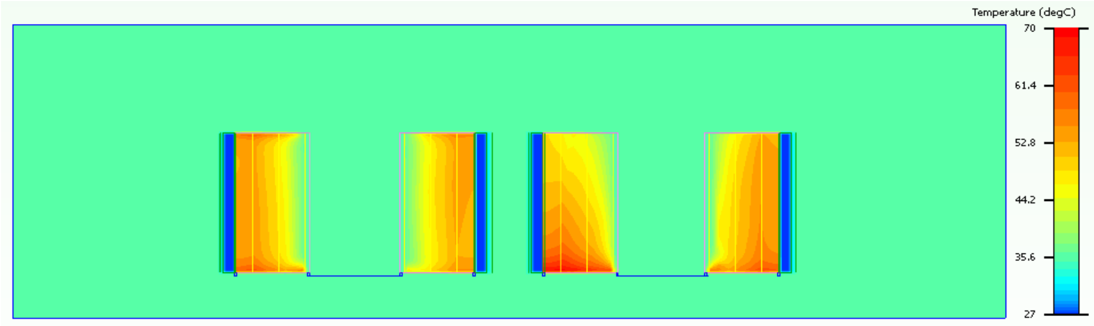


Figure 5.10 Temperature plot for case 3:18 KW heat load with fans and RDHx

The underfloor plenum is got rid off in the next case for allowing space for infrastructure. The figure shows the thermal plot of this configuration.

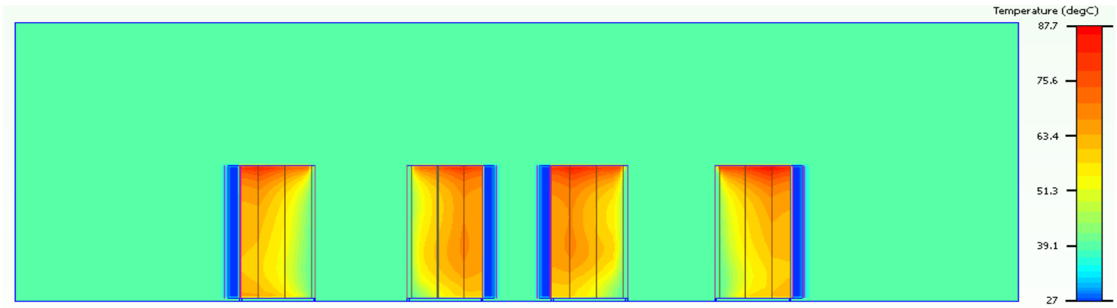


Figure 5.11 Temperature plot for case 4: No plenum and with RDHx

From the figures 5.8 to 5.11, it is observed, from case 2, the effect and need for having heat exchanger is understood. Also, it can be concluded the heat exchangers reduce the temperature of room.

5.4.4 Impact of RDHx in side breathing switches

In addition to previous scenarios, the effect of RDHx is also studied in side breathing switches. The current trend in data centers, the miniaturization of devices with a tremendous increase in the heat densities led to new approach for the server manufacturers. One of such approaches which creates problem in thermal management are the side breathing switches which allows air to flow from side to side. Prevention of hot air recirculation in a cabinet with side breathing switches is the current challenge. The cable management has to be performed for the proper air flow through the cabinets.

In the side breathing switches, air flows from the right side of the cabinet and exhausts through the left. There are different series of network switches that are developed like Cisco 6500 and 9500 series. The prerequisite for using these switches is to ensure sufficient volume of air for cooling the electronics. The model used for this study is Cisco 6503 E network switches. In this configuration, the air enters the server from right side of the server and leaves through left side. The aim of this part is to investigate the effect of heat exchangers in data center cabinets with side breathing switches [50].

Side breathing switches are the network switches in which air enters and leaves from the side of the servers. The figure below represents the network switch commercially available [51].

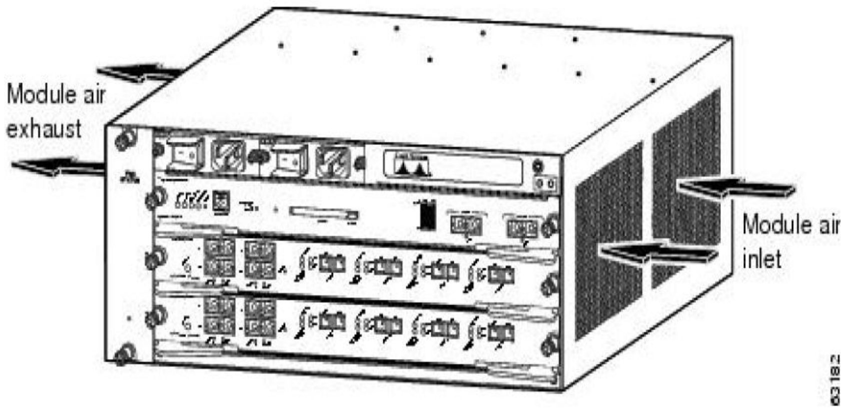


Figure 5.12 CISCO 6503E Side breathing switch

These switches have to be supplied with a sufficient volume of air to cool the supervisor engines, modules and power supplies. There should a minimum gap of 6” between the cabinet walls and inlet/exhaust of the equipment. The difference in temperature across the network switch is considered to be $10^{o}C$.

A commercially available network switch is utilized in this study in which air enters from the side of the cabinet and leaves from the other side. For supplying the required amount of air to flow into the network switch, cabinets of width 800 mm is recommended. The cabinets are 100% populated. The ambient is at $27^{0}C$. The typical data center model is considered in which 20 cabinets are placed. The cabinets consist of front-back configuration for air flow. The width of the cabinet is 800 mm. The number of servers per rack are 8(75% populated). The following are design variables considered for the calculating the flow rate required for the cabinets.

- Heat load per rack(Q)=950W x 8(servers)= 7.6 KW
- $\Delta T = 10^{0}C$*
- The flow rate($\dot{m}$)is then calculated by using the formula

$$Q = \dot{m} \times C_p \times \Delta T$$

= 1335.3 cfm

The heat load of the cabinet is considered as 7.6kW as per the standards of the manufacturers. The figure below represents the cabinets with heat exchangers.

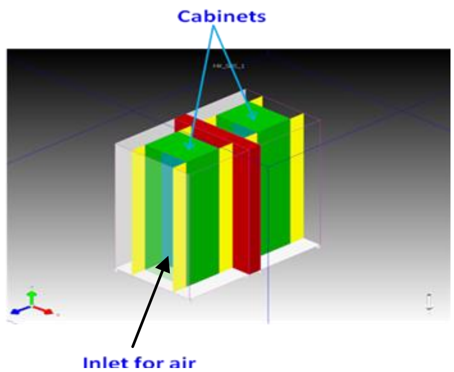


Figure 5.13 Cabinets with heat exchanger in the middle of the cabinets.

Blanking panels are provided for the restricting the wastage of air in the cabinets. The air enters from the side and leaves from the other. Heat exchanger helps in reducing the inlet temperature for the adjacent cabinet which in turn helps in a reduction in power consumption.

5.4.4.1 Results

The effect of heat exchanger is studied in two cabinets of the room. The figure below shows the effect of RDHx in side breathing switches.

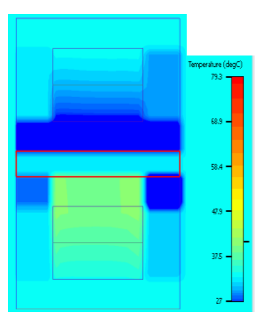


Figure 5.14 Temperature plot for side breathing switches

From the figure above, the importance of the heat exchanger is studied. It is noticed that employing heat exchanger provides the ambient temperature required for the adjacent cabinets.

CHAPTER 6

CONCLUSION AND FUTURE WORK

6.1 Conclusion

Summarizing based on the heat loads for the telecommunication Data Centers, guidelines are formulated for appropriate use of accessories based on heat loads.

From the thermal analysis performed in an air cooled Data Center considered, it is observed that the accessories: blanking panels, chimney and cabinet fans, are effective for heat loads up to 12 kW and containment proves to be effective for 16 kW heat loads and cold aisle containment shows the best results among all the cases as it has least global maximum temperature compared to all the cases. Also, kick plate aids to avoid mixing of hot and cold air beneath the cabinets. However, it is observed that the air cooling consolidates with the increase in heat loads of the cabinets. Therefore, air cooling is alternated with the hybrid cooling for cooling the Data Centers with high heat loads.

It is verified that for heat loads of 18 and 20 KW cabinets of data centers, the Rear Door Heat Exchangers can cool 100% efficiently. Additionally, it is noted that the effectiveness of heat exchangers help in reducing the maximum global temperature of the room but its effectiveness is reduced when CRAC units are placed along with it.

The effect of heat exchangers is analyzed in different configurations and it is recommended that RDHx can help in providing space for the infrastructure by avoiding under floor plenum for the supply of air. It is recommended that heat exchangers can be effective in cooling side breathing switches and as well help in significant power savings. It is also recommended that heat loads over 16kW can be accommodated with a RDHx. Obviously; the RDHx can be utilized for the entire heat load up to 27kW.

6.2 Future work

In the future, total cost ownership (TCO) needs to be studied for to give a more appropriate recommendations on RDHx. Also, the study can be extended to optimize the water inlet temperature for the RDHx. Furthermore, focus can be laid on the side breathing switches for higher heat loads.

APPENDIX A

DETERMINING LOSS COEFFICIENT AND FILTER CHARACTERIZATION IN FLOTHERM

A.1 Calculating The Loss Coefficient For The Resistance

The loss coefficient is determined by the below equation

$$k = \frac{1}{\beta^2}\{1 + [(0.5)(1-\beta)^{0.75}] + [(1.414)(1-\beta)^{0.375}]\}$$

Where k = loss coefficient

β= the opening

Table below shows the different loss coefficient for various free opening.

Table A.1: The loss coefficient values

open area in %	1- (Open area/100)	Loss coefficient
45	0.55	8.2
50	0.5	6.6
61	0.3875	4.3
73	0.27	3.0
80	0.2	2.4
90	0.1	1.8

A.2 Filter Characterization

Steps followed for characterization the filter are discussed considering a MERV 11 filter.

1. Determining the pressure and velocity values: These are determined experimentally

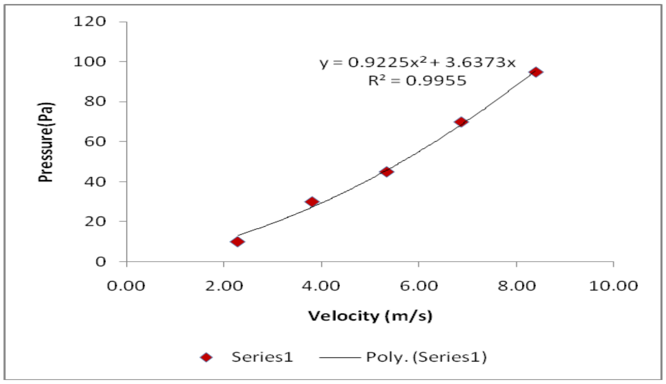
   The values of pressure and velocity of a MERV 11 filter are tabulated below.

Table A: Static pressure drop and volumetric flow rate for the sample filter

Pressure (in H_2O)	**Pressure (Pa)**	**Flow rate (cfm)**	**flow rate in (m3/s)**	**Velocity (m/s)**	**Area (in2)**	**Area (m2)**
0.04	9.9632	750	1.51	2.29	1024	0.660644
0.12	29.8896	1250	2.52	3.81	1024	0.660644
0.18	44.8344	1750	3.53	5.34	1024	0.660644
0.28	69.7424	2250	4.54	6.86	1024	0.660644
0.38	94.6504	2750	5.54	8.39	1024	0.660644

2. Plotting the pressure vs. velocity graph: The obtained values are plotted. The fitted curve should be a second order polynomial and the intercept should be fixed to 0. This enables the equation to be compared with pressure. The graph below represents the graph providing the trendline represented by a second order equation:

$$y= 0.9225\, x^2+3.6373\, x \qquad \text{eq.(i)}$$



3. The equation thus obtained is compared to the standard equation of the pressure drop

$$\Delta P = [A.\mu/2.l].v + [B.\rho/2.l].v^2 \qquad \text{eq.(ii)}$$

Where,

Variable	Symbol	Value
Dynamic viscosity (kg/m s)	μ	1.86×10^{-6}
Density (kg/m^3)	ρ	1.16
Characteristic length (m)	l	1
Pressure drop(kg /m s^2)	ΔP	
Velocity (m/s)	v	

4. Lastly, substituting the determined values in Flotherm

Values of A and B

A	7.06E+05
B	1.30E+00

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BIOGRAPHICAL INFORMATION

Poornima Mynampati, an aspiring student, received her Bachelor's degree in Mechanical engineering from the Jawaharlal Nehru Technological University, India in 2004-2008. Her interest in Thermal Engineering made her choose Mechanical engineering for her Master's in Fall- 2008 at the University of Texas, Arlington, Texas. The same passion resulted in perusing thesis in thermal management of Data Centers and also outdoor shelters leading to a Master's degree in Mechanical engineering. During the course of her study, she had an opportunity to collaborate with industry professionals which helped her to bridge the academic knowledge with the industry. She has been a strong participant in professional activities managing five publications.