THERMAL DESTRATIFICATION OF AIR STREAMS TO IMPROVE THE COOLING PROVISIONING OF AIR-COOLED DATA CENTERS

by

ANTO JOSEPH BARIGALA CHARLES PAULRAJ

THESIS

Submitted in partial fulfillment of the requirements
for the degree of Master of Science in Mechanical
Engineering at
The University of Texas at Arlington
May 2019

Arlington, Texas

Supervising Committee:

Dr. Dereje Agonafer, Supervising Professor
Dr. Abdolhossein Haji-sheikh
Dr. Andrey Beyle

ACKNOWLEDGEMENTS

I would like to take this opportunity to express my sincerest appreciation to my supervising professor Dr. Dereje Agonafer for his exceptional support, guidance and inspiration throughout the course of my research work. The invaluable advice and timely support whenever required was the major driving force, which enabled me to complete my research work.

I would like to thank Dr. Abdolhossein Haji Sheikh and Dr. Andrey Beyle for taking their time for serving as my committee members. I would also like to take this opportunity to thank Mr. Mark Seymour of Future Facilities for all his expertise and continuous support and feedback during the projects. His industrial expertise has been important for my research. Also, I would like to thank Ashwin Siddarth, Rajesh Kasukurthy and other lab mates for their invaluable support while working at the EMNSPC labs.

I would like to thank all my friends in the EMNSPC team and my roommates who constantly supported me throughout my time here. Lastly, I would like to thank my parents Mr. Charles Paulraj Barigala Francis and Mrs. Jessia Margaret Selvaraj for their support both emotionally and financially, without which I would not have dreamt this far. I owe my love and gratitude to God almighty for giving me strength and keeping me motivated.

May 2019

ABSTRACT

THERMAL DESTRATIFICATION OF AIR STREAMS TO IMPROVE THE

COOLING PROVISIONING OF AIR-COOLED DATA CENTERS

Anto Joseph Barigala Charles Paulraj, M.S

The University of Texas at Arlington, 2019

Supervising Professor(s): Dr. Dereje Agonafer

Air side economization is an arrangement of duct, damper and automatic control system which together allow introducing outside air to reduce the mechanical cooling during mild or cold weather thereby decrease the energy consumption. The outside ambient air and heated return air from the information technology (IT) pod is mixed inside a dedicated space to achieve a target cold aisle operating temperature and thereby increase economization. Major constrain faced by the design engineers while designing the Mixing Chamber/ Plenum is the stratification of air stream due to the Temperature gradient in the mixed air stream and this stratification can be attributed to the short span of time and space that is available for the air streams to interact with each other. Thermal stratification can lead to coil freeze-ups, nuisance freeze-stat trips, energy wastage due to sensing error and poor indoor air quality and increases the cooling power. So, to achieve our objective we want to understand the fundamental physical phenomenon which causes mixing of any two fluids and thereby apply the knowledge to our test scenario. Literature review gave us the information about Richardson number a dimensionless number which is a ratio between buoyancy and flow shear stress and for improving the process of mixing the Richardson number should be minimized. The variables involved in minimizing the Richardson number are the velocity and temperature difference between the two fluid and the total vertical column height of the two fluid that needs to be mixed. This fundamental understanding will help us in determining the structural features needs to be achieve the desired mixing. The scope of the project is also extended to address the effect of thermal stratification after evaporative media cooling pad, in this scenario the process of segmentation of the pad for reducing the relative humidity also creates thermal stratification. Computational Fluid Dynamic analysis is carried out to report the proof of concept and thereby report the changes in the effectiveness of the mixing process at upstream of heating coil and downstream of cooling pad.

LIST OF ABBREVATIONS

P	Density (kg/m3)
k	Thermal Conductivity (W/m-K)
ε	Kinematic Rate of Dissipation (m2/s3)
V	Velocity (m/s)
µ	Viscosity (N/m2 s)
m	Mass flow rate (kg/s)
q	Heat load (W)
P	Power (W)
v	Volumetric Flow Rate (cfm)
IT	Information Technology
ASHRAE	American Society of Heating, Refrigeration and Air Conditioning Engineers
Re	Reynolds number
Cp	Specific Heat capacity (J/kg K)
P	Pressure (in of H2O)
DEC	Direct Evaporative Cooling Unit
DEC	Indirect Evaporative Cooling Unit
OA	Outside air damper
RA	Return air damper

LIST OF FIGURES

FIGURE PAGE

1. *Average Data center power allocation*……………………………………………………………………………….1

2. *Table of ASHRAE recommendations for air cooled data centers*……………………………………………….2

3. *Different psychrometric process*……………………………………………………………………………………..4

4. *Typical schematic of air-side economization*…………………………………………………………………….5

5. *Schematic of Direct Evaporative cooling*……………………………………………………………………………6

6. *Mixing of stratified layers with velocity shear*……………………………………………………………………7

7. *Data center at Mestex Dallas facility*…………………………………………………………………………………10

8. *Top view of Aztec 30 air handling unit*…………………………………………………………………………… .11

9. *Damper Angle vs percentage of maximum flow based on damper authority*……………………… 12

10. *Adiabatic mixing schematic with process line on the psychrometric chart*……………………… 13

11. *Adiabatic mixing of OA and RA on psychrometric chart*…………………………………………………14

12. *CFD model of the ASC 30 unit*…………………………………………………………………………………………16

13. *Result plane location for baseline model without static structure*……………………………………18

14. *Temperature plot of the baseline model without static structure*……………………………………19

15. *CFD model with static structure*………………………………………………………………………………… 20

16. *Different views of the static structure*…………………………………………………………………………21

17. *Temperature distribution for the model with static structure*……………………………………22

18. *CFD model of ASC 30 unit and the IT pod*……………………………………………………………………23

19. *Temperature plot when middle two pads were wetted*…………………………………………………24

20. *Server inlet air temperature distribution inside cold aisle*………………………………………………24

21. *Temperature plot when alternating pads were wetted*…………………………………………………25

22. *Server inlet air temperature distribution inside cold aisle*………………………………………………25

TABLE OF CONTENT

ACKNOWLEDGEMENT

ABSTRACT

LIST OF FIGURES

LIST OF ABBREVATIONS

CHAPTER – 1 – INTRODUCTION……………………………………………………………………………………01

CHAPTER – 2 – LITERATURE AND MOTIVATION……………………………………………………………04

CHAPTER – 3 – AIR HANDLER AND CFD MODEL……………………………………………………………10

CHAPTER – 4 – RESULTS AND DISSCUSSIONS…………………………………………………………………18

CHAPTER – 5 – CONCLUSION AND FUTURE WORK…………………………………………………………26

REFERENCE………27

APPENDIX……28

CHAPTER 1 – INTRODUCTION

Data centers are large group of networked computer servers typically used by organizations for the remote storage, processing, or distribution of large amounts of data. Data storage and data sharing has become vital in the growth of all industries. Storage and sharing of data are no longer happening locally, now it is doing globally. This exponential growth for the storage has increased the demand for the data centers. It became very crucial to run the Data center very efficiently in optimal space and resources and in the same time cost effectively. Cost reduction can be attained by using various techniques such as air side economization (ASE), Indirect evaporative cooling (IEC) and Direct evaporative cooling (DEC) and Direct Expansion Cooling.

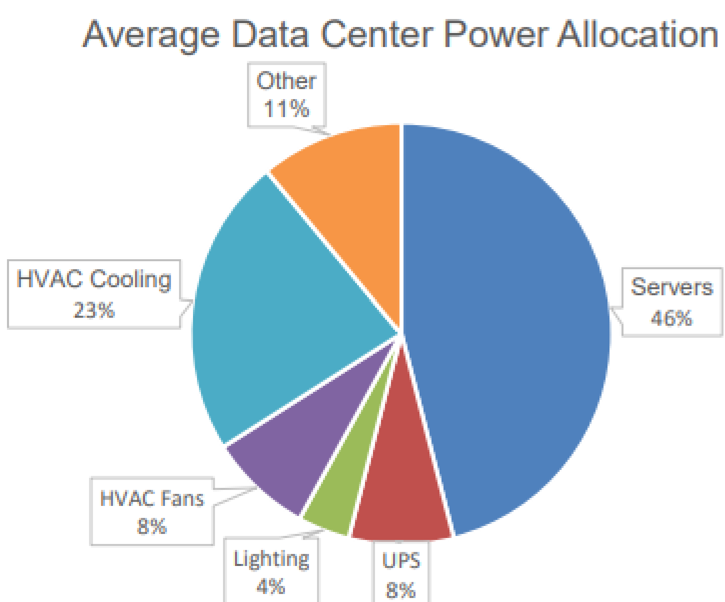

Figure – 1: Average Data center power allocation [1]

According to a report by US data centers consumed about 70 billion kilowatt-hours of electricity in 2014, the most recent year examined, representing 2 percent of the country’s total energy consumption [2]. Power Usage Effectiveness (PUE) is the most common method for measuring and quantifying the energy usage in a data center.

PUE needs to be minimized, according to a survey a PUE of 1.65 is considered as an ideal average for the modern data centers.

The American society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE) recommends thermal guidelines for the safe operations of IT equipment in data centers. In 2011 ASHRAE released an updated thermal guideline for the data centers by publishing different classes.

Class[a]	Equipment Environmental Specifications for Air Cooling						
	Product Operations[b,c]					Product Power Off[c,d]	
	Dry-Bulb Temperature[e,g] ˚C	Humidity Range, Non-Condensing[h,i,k,l]	Maximum Dew Point[k] ˚C	Maximum Elevation[e,j,m] m	Maximum Temperature Change[f] in an Hour (˚C)	Dry-Bulb Temperature ˚C	Relative Humidity[k] %
Recommended (Suitable for all 4 classes)							
A1 to A4	18 to 27	-9˚C DP to 15˚C DP and 60% RH					
Allowable							
A1	15 to 32	-12˚C DP & 8% RH to 17˚C DP and 80% RH[k]	17	3050	5/20	5 to 45	8 to 80
A2	10 to 35	-12˚C DP & 8% RH to 21˚C DP and 80% RH[k]	21	3050	5/20	5 to 45	8 to 80
A3	5 to 40	-12˚C DP & 8% RH to 24˚C DP and 85% RH[k]	24	3050	5/20	5 to 45	8 to 80
A4	5 to 45	-12˚C DP & 8% RH to 24˚C DP and 90% RH[k]	24	3050	5/20	5 to 45	8 to 80
B	5 to 35	8% to 28˚C DP and 80% RH[k]	28	3050	NA	5 to 45	8 to 80
C	5 to 40	8% to 28˚C DP and 80% RH[k]	28	3050	NA	5 to 45	8 to 80

Figure – 2: Table of ASHRAE recommendations for air cooled data centers [3]

When external conditions are favorable for the cooling application of data centers, Air side economization (ASE) can be used for reducing the cost for the mechanical cooling in the data centers. In simple, Air side economization is a mechanism which is used to regulate the use of outside air for cooling. It is the most commonly used cooling systems. Air side economizers are used to supply the cold air from environment to the data center through ducts. The outside and inside temperature conditions are measured with the help of sensors. Another important concern for the ASE, the accuracy of the sensors which are used to measure the temperature and enthalpy or humidity of the outside air. Based on the dryness of outside air we might have to humidify or dehumidify the air.

CHAPTER 2 – LITERATURE AND MOTIVATION

2.1 PYSCHROMETRIC PARMETERS AND PROCESS

Dry Bulb Temperature (T_{db}): It is the temperature that is measured by a thermometer that is simply kept in the ambient. There are no external factors affecting its readings. On a psychrometric Chart, Dry Bulb Temperature is always found on the horizontal axis.

Wet Bulb Temperature (T_{wb}): The Wet Bulb Temperature unlike the Dry Bulb Temperature also accounts for the moisture content of the air. It represents the lowest temperature that can be achieved in the ambient. It is represented as parallel 45° lines on the psychrometric chart.

Dew Point Temperature (T_{dp}): The Dew Point Temperature is theoretically the lowest possible that can be achieved before condensation begins. It is represented as horizontal lines originating from the vertical axis on the right.

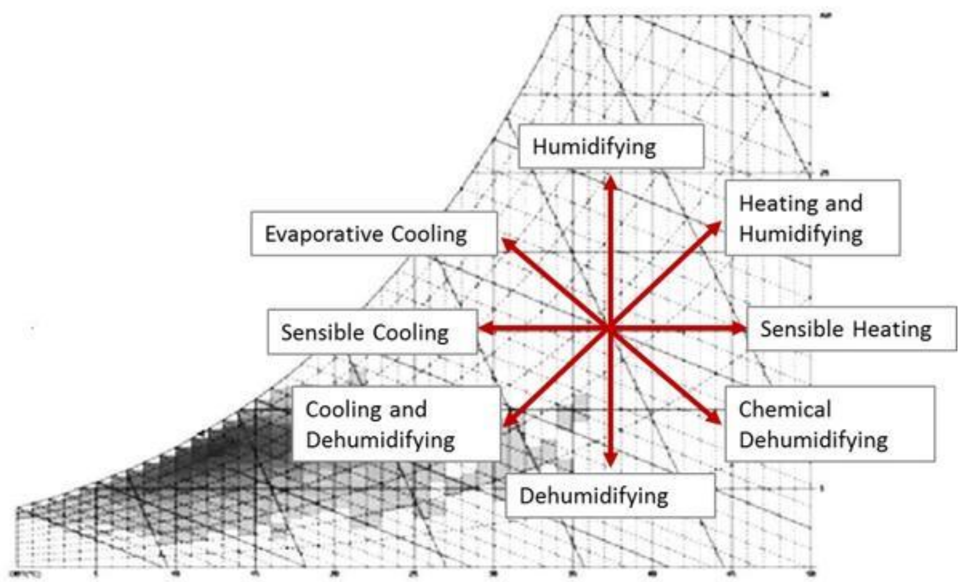

Figure – 3: Different psychrometric process [4]

Relative Humidity (R.H.): It represents the ratio of the amount of moisture present in the air to the maximum amount of vapor that can be present in air at that given temperature. It is represented in terms of percentage (%) and can be read on the psychrometric chart as the curved lines from 10% to 100% (saturation).

Specific Humidity: It is defined as the ratio of the mass of water vapor in air to the total mass of air. It is represented as percentage (%) and represented by horizontal lines just like Dew Point Temperature.

In ASE, when the outside air temperature and humidity is below the recommended region the return air is mixed with cold outside air.

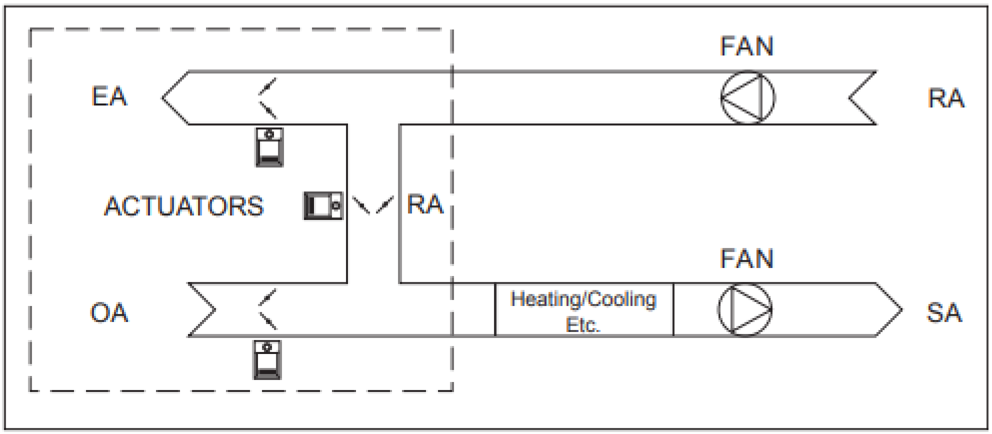

Figure – 4: Typical schematic of air-side economization [5]

2.2 TURBULENCE IN MIXING

Turbulence plays an important role in the mixing of two streams. Flow can be classified based on the Reynolds number as laminar when the Reynolds number is low or turbulent when the Reynolds number is high. Some characteristics of turbulence are irregularity, diffusivity, 3D vorticity fluctuations, dissipations and continuum.

There are various models to predict the nature of turbulent flow and for this study purpose one such model called K-ε turbulence model is used. Even though RANS K-ε turbulence model is better in accuracy than the K-ε turbulence model, the computational power requirements doesn't justify the need of higher magnitude of accuracy for our study.

2.3 DIRECT EVAPORATIVE COOLING

It is widely used in arid climatic regions where the temperature is high, and humidity is low. The air is passed through a porous media pad where water is passed through and when the air comes in contact with the pad the latent heat of air is used to evaporate the water and thereby the temperature of air is reduced at the same instance the humidity of the air is also increased.

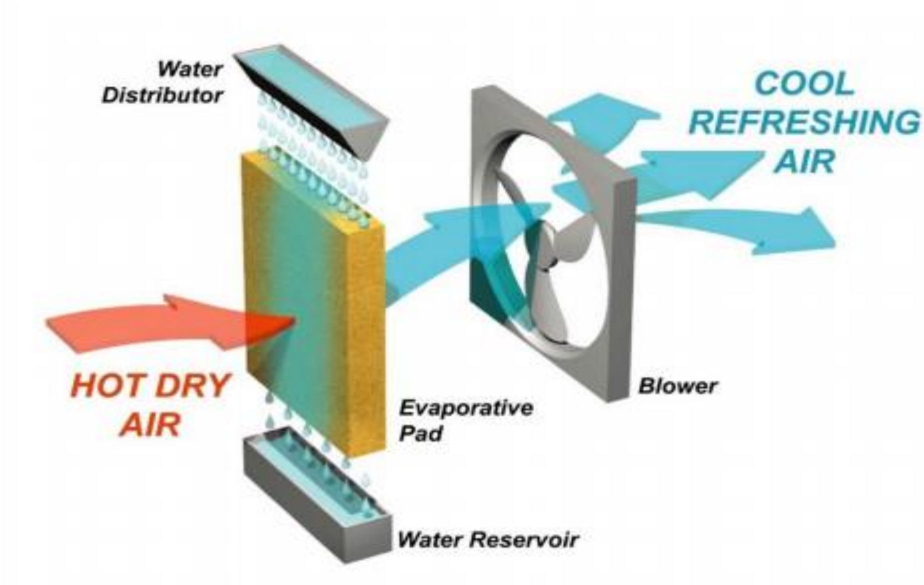

Figure – 5: Schematic of Direct Evaporative cooling [7]

2.4 RICHARDSON NUMBER

The first approach in studying mixing is by considering its geometrical aspects, which is the topology of the streamlines.

Naturally for mixing to happen the kinetic energy release of the fluids needs to exceed the potential energy gain, but a sizeable fraction of the kinetic energy release by the mixing process creates turbulence and thereby dissipated as heat energy [8].

Richardson Number is a dimensionless number which gives us the ratio of buoyancy to the flow shear stress. Mixing in a stratified fluid will occur when Richardson Number is minimized.

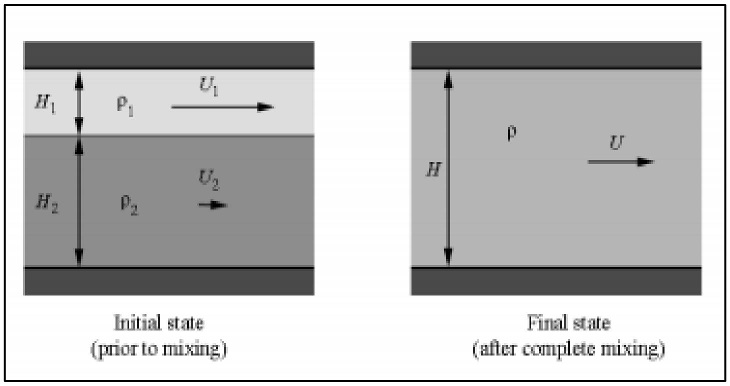

Figure – 6: Mixing of stratified layers with velocity shear [8]

$$Ri=\frac{\alpha * g * H * \Delta T}{|\Delta U|^2}$$

Where,

- Ri – Richardson number
- α – Thermal Expansion Coefficient
- g – Acceleration due to gravity
- H – Total characteristic length along the direction of stratification
- ΔT – Difference in temperature between two fluids
- $|\Delta U|$ – Difference in velocity between two fluids

2.5 MIXING EFFECTIVENESS

Mixing of two streams can be quantified using mixing effectiveness such as range effectiveness [9]

$$(E_{RT}) = \left(1 - \frac{T_{Max} - T_{Min}}{|T_{RA} - T_{OA}|}\right) * 100\%$$

where

- T_{max}-Maximum Temperature in the plane of measurement
- T_{min}-Minimum Temperature in the plane of measurement
- T_{RA}- Return Air Temperature

- T_{OA}- Outdoor Air Temperature

SUPPLY HEAT INDEX

SHI is a dimensionless parameter used to evaluate the thermal performance of a large-scale data center [10].

$$SHI = \frac{\sum(Ti - Tsup)}{\sum(Tr - Tsup)}$$

Where,

- T_i – Server Inlet Temperature
- T_r – Server Outlet Temperature
- T_{sup} – CRAC supply Temperature

This parameter is specifically used to evaluate the extent to which cold air and hot air mix inside the IT pods. They are scalable metric, applicable for potential application related to racks, rows of wide data center levels. Supply Heat Index (SHI) is used to quantify the stratification that occurs vertically across the cabinet.

2.6 EVAPORATIVE COOLING PAD

Evaporative cooling pads are porous pads which uses the evaporation to cool the warm outside air by passing it over the wetted pad. The efficiency of the cooling pad is

$$\text{Efficiency (\%)} = (T_{in} - T_{out})/(T_{in}-T_{wb}) * 100$$

Where, T_{in} and T_{out} are the temperature of the air entering and leaving the cooling pad and T_{wb} is the wet bulb temperature [11].

2.7 MOTIVATION FOR STUDY

Poor mixing of two air streams inside the air handling unit leads to air stratification results in freezing of coils, Poor indoor air quality and Freeze stat tripping nuisance. Coil freeze up problem arises when the water inside the coil goes below 32-degree Fahrenheit which leads to freezing of water. This happens due to localized spots of cold regions due to improper mixing of return and outside air.

Freeze stat tripping nuisance is caused due to the repeated tripping of the long element that is calibrated to trip if the temperature falls below certain set point temperature range. Coil inefficiencies could also occur because of the improper mixing because coils are tested with uniform velocity and in the actual scenario, the velocities are varying, and this can lead to degradation of coils. Improper mixing can also lead to uneven loading on the filters. Poor mixing also leads to failure for attaining uniform desired temperature inside the cold aisle in the data center. Poor mixing will also impact the cooling power consumption by over provisioning the cooling that is required. Improper mixing will also cause sensor errors which in turn will provide wrong feedback for the control system and thereby it will also lead all other problems that was discussed earlier.

CHAPTER 3 – AIR HANDLERS AND CFD MODELS

3.1 AIR HANDLING UNIT

Air handling Units (AHU) is an essential component of heating ventilation and Air conditioning (HVAC) systems. AHU are large metallic structures with ducts and vents to direct the conditioned air into the information Technology (IT) pods.

A mixing chamber or a plenum is a dedicated space upstream of heat and cooling coil where two streams of air are mixed. Generally, the mixing chambers are of short length and space which minimal space and time to achieve complete mixing of two air streams. The airflow rate inside the mixing chamber are controlled using the dampers based on the control strategy that is implemented.

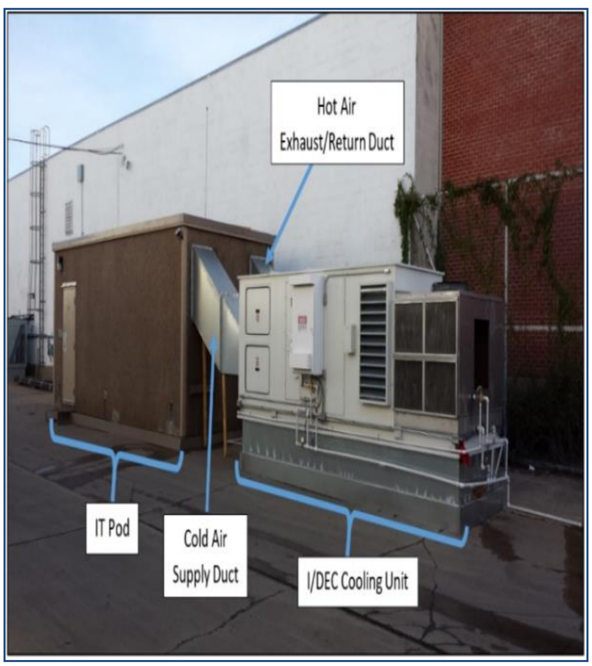

Figure – 7: Data center at Mestex Dallas facility [12]

In general AHU are used to control the temperature, humidity, air movement and the air cleanliness inside the IT pod.

Components inside an AHU are,

- Dampers
- Filters
- Mixing Chamber
- Heating and Cooling elements
- Humidifiers and Dehumidifiers
- Blowers and Fans

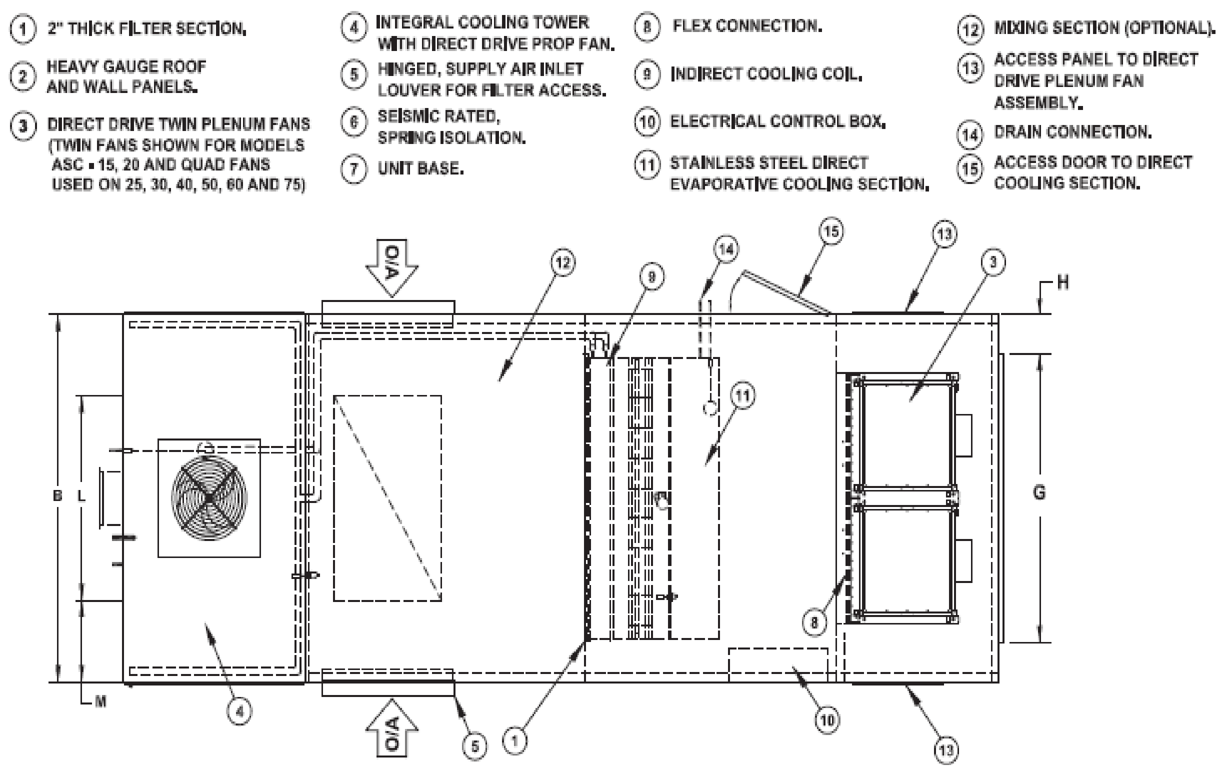

Figure – 8: Top view of Aztec 30 air handling unit [13]

3.2 DAMPERS

Dampers are the primary air deflectors inside the air handling unit. Simply, dampers are an array of flat plate structures arranged in a specific fashion with certain flexibility of movement to redirect the air and to control the volumetric flow rate of the air inside the air handler. Dampers are of two types based on their arrangement, parallel blade dampers and opposed blade dampers. Parallel blade dampers provide minimal pressure drop across the damper than compared to the opposed blade dampers.

For this study purpose, parallel blade dampers are chosen because they provide higher flowrate for lower open percentage at minimal pressure drop.

3.3 DAMPER AUTHORITY

Damper authority is crucial in correlating damper blade angles, velocity ratios and mixing volume ratios of OA and RA when utilizing motorized damper control and feedback configuration [14].

$$\text{Damper Authority (\%)} = \frac{\text{Open Damper Resistance}}{\text{Total System Resistance}} \times 100\%$$

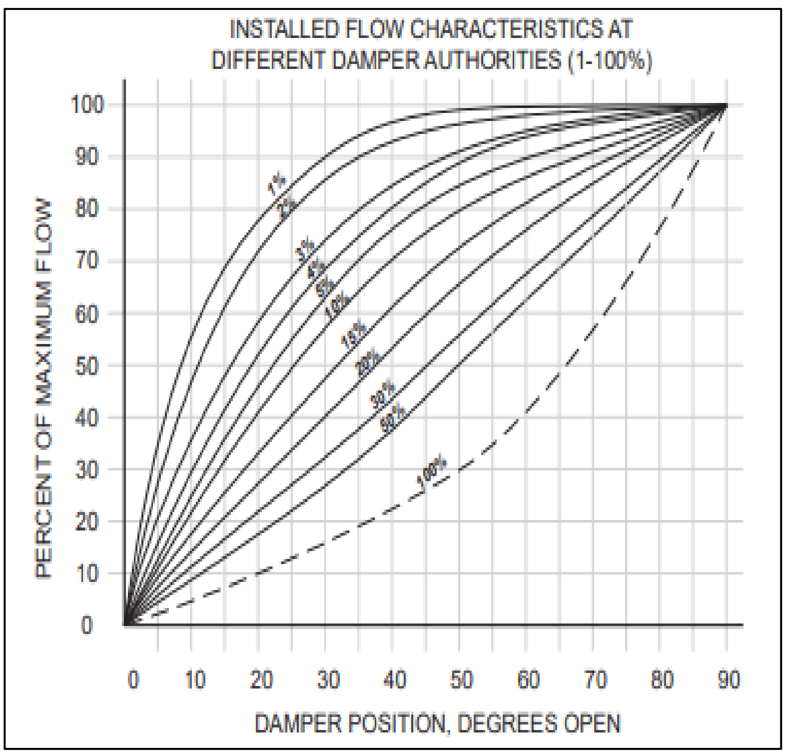

Figure – 9: Damper Angle vs percentage of maximum flow based on damper authority [14]

At lower damper angle the flow increases at higher rate and Damper authority is minimum on the total pressure drop in the system. Based on the Damper Authority percentage, we can find the combination of the damper angle for the OA & RA.

In order to find the damper angle combination, we must calculate the percentage of the two-stream mixing. Once we know their respective percentages, we can calculate the damper angle from the flow characteristic curve using the damper authority.

3.4 ADIABATIC MIXING

Adiabatic mixing is a psychrometric process where we mix two different air streams to obtain a third mixed air stream. The conditions of the resulting mixture can be obtained from the mass and energy balance.

Energy Balance: $m_1h_1+m_2h_2=m_3h_3$

Mass Balance: $m_1+m_2=m_3$.

In this study, we have chosen two streams at temperature of 37-degree Fahrenheit and 113-degree Fahrenheit at a relative humidity of 40% for the both streams. Outdoor air temperature was set at 37F because the lowest average temperature in a year in Dallas-Fort Worth area was 37F and the return air temperature was chosen as 113F because a typical class A4 has return air temperature up to 113F. So, these extreme cases where considered to study the mixing phenomenon.

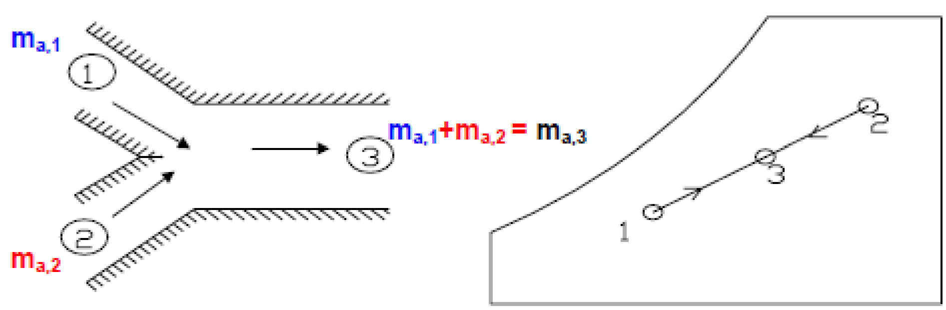

Figure – 10: Adiabatic mixing schematic with process line on the psychrometric chart [15]

The target temperature of the mixed air was to set 68F and all other conditions were obtained from the psychrometric chart by plotting and connecting the first and second air stream point through a straight line. By this plotting, we can find the percentage of each air stream i.e. OA/RA ratio can be found. The percentage of RA was found out to be 40% of the total volume.

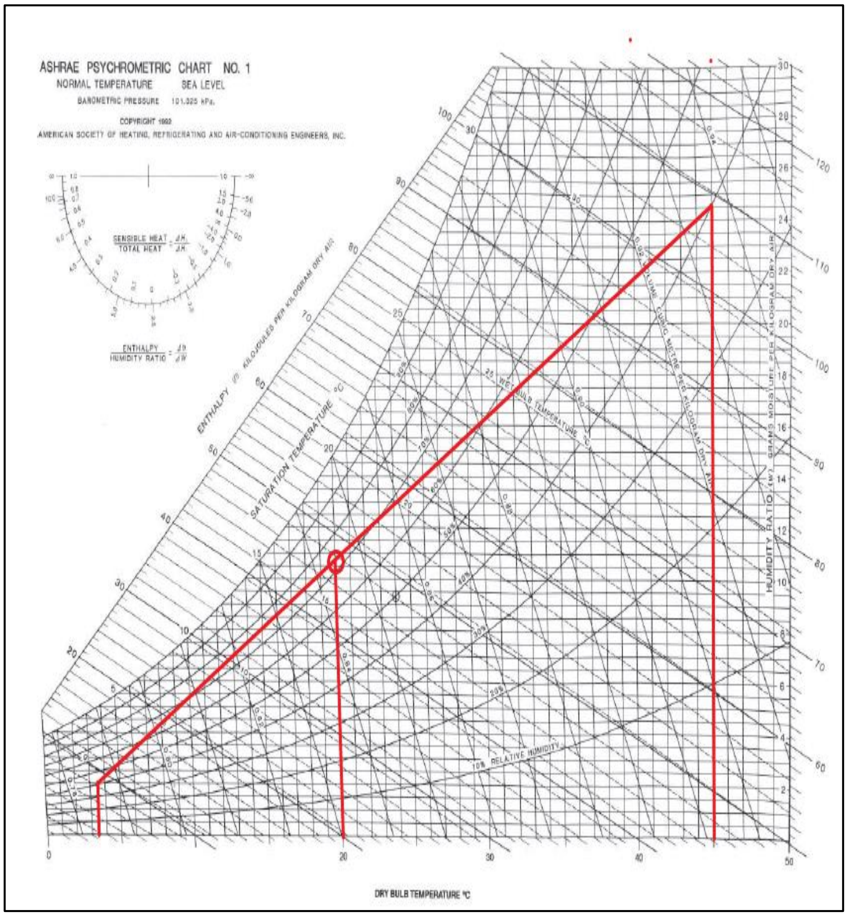

Figure – 11: Adiabatic mixing of OA and RA on psychrometric chart

3.5 GRID SENSITIVITY ANALYSIS

Grid sensitivity analysis is carried out to make the model accuracy independent of the mesh size. By carrying out grid sensitivity analysis we can find the minimum number of grids that is needed for the model, increasing the grid count after which is irrelevant to the accuracy of the model and it just increases the computation power.

Grid sensitivity for this study was carried out on dampers, since majority of the model is open and plain. The grid sensitivity analysis was also carried out for the static structure which was brought in the study.

It was found out that the grid independence was achieved at 60 cells in each direction for all three direction for the dampers and this led to overall grid count of half a million grids.

3.6 COMPUTATIONAL MODEL

The AHU was modeled based on MESTEX Aztec ASC 30 unit. This unit has a total evaporative face area of 4320 in^2. The volumetric flow rate for this unit is 15000 CFM. This unit was chosen because we needed to provide cooling for the standalone IT pod that was designed after the standard models from the open compute project. This CFD model was created in software called 6Sigmaroom from future facilities.

Geometrical Specification

- AHU Chamber size – 72.5 * 92 * 204 inches
- Supply air vent Size – 70 * 20 inches
- Return air vent size – 53.5 * 32.5 inches
- Outdoor air vent size – 36.5 * 60 inches
- Damper size – 0.5 * 10 * 36.5 inches

Fan Specification

- Fan Diameter – 18.2 inches
- Fan Flow rate – 6250 CFM
- Number of Fans – 4

Evaporative Pad Specification

- Number of cooling pads – 4
- Pad thickness – 12 inches
- Pad dimension – 15 * 72 inches

The flow rate through the cooling pad is 0.134 CFM of water at a temperature of 60F. The viscous resistance coefficient and inertial resistance coefficient was set to 3 and 30. The evaporative efficiency of the pad was set to 90%.

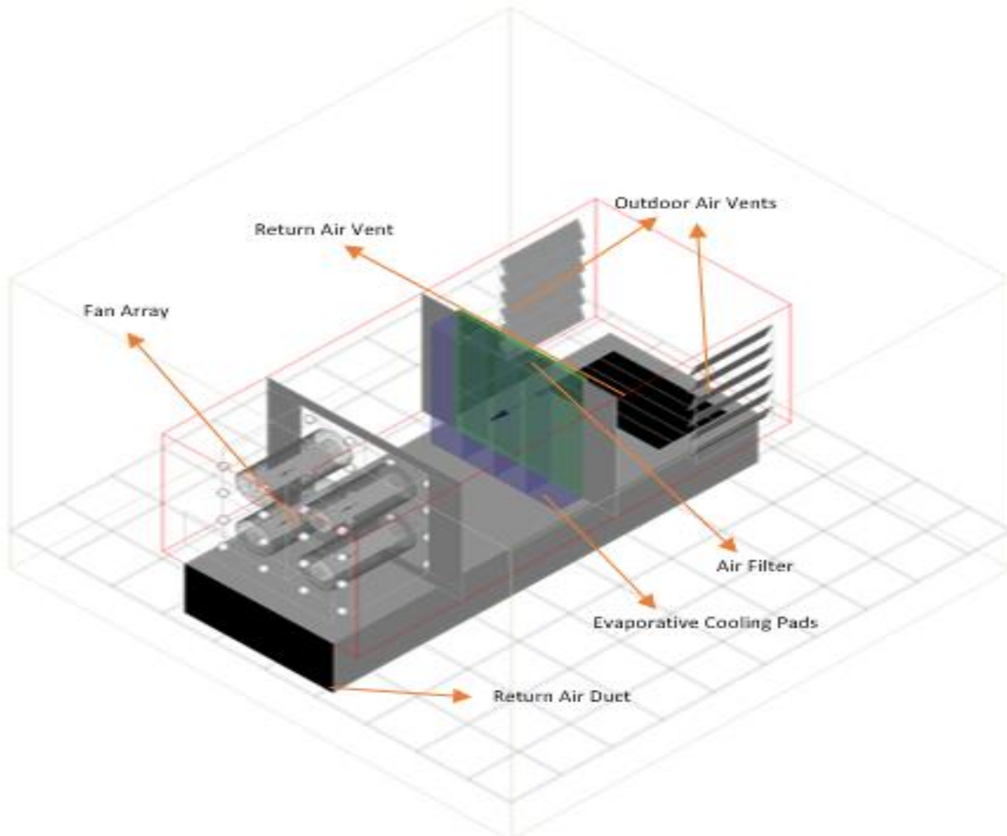

Figure – 12: CFD model of the ASC 30 unit

3.7 INFORMATION TECHNOLOGY POD

IT pod is a dedicated space to accommodate various computational equipment's for various purposes. These pods are generally cooled using AHU's and there are guidelines and recommendations stated for properly removing and maintaining the temperature and humidity within these IT pods.

This study uses an IT pod which is based on the Open Compute Project. In 2009, Facebook realized it needed an better energy efficient data centers to meet the demand for the storage, by dedicated work they created an energy efficient data center in 2011 and they decided to share their knowledge along with other industry leaders to the public, thereby making this project open for everyone [16]. IT pod in this model has 12 cabinets with each cabinet of 7.5KW.

3.8 BOUNDARY CONDITIONS

- Outside Air Temperature: - 37F
- Return Air Temperature: - 113F
- Target Mixed Air Temperature: - 68F
- k-ε Turbulence model
- Number of Fans – 4
- Total flowrate – 15000 CFM
- Unstructured Grid
- Total Cell Count – 0.5 million

CHAPTER 4 – RESULTS AND DISSCUSION

4.1 CASE STUDY - 1

The effect and influence are studied upstream of heating and cooling coil. Here the hot return air is mixed with cold outside air. This case compares the effects of mixing without any static structures and with placing a static structure.

4.2 MODEL WITHOUT STATIC STRUCTURE

Model without any static mixer is the baseline model which is exactly modeled after the Aztec 30 unit from the Mestex. The result plane depicts the temperature plot and the plane is plotted just before the air filter. This is where the mixing must be complete to avoid any localized spots which could lead to coil freeze-up.

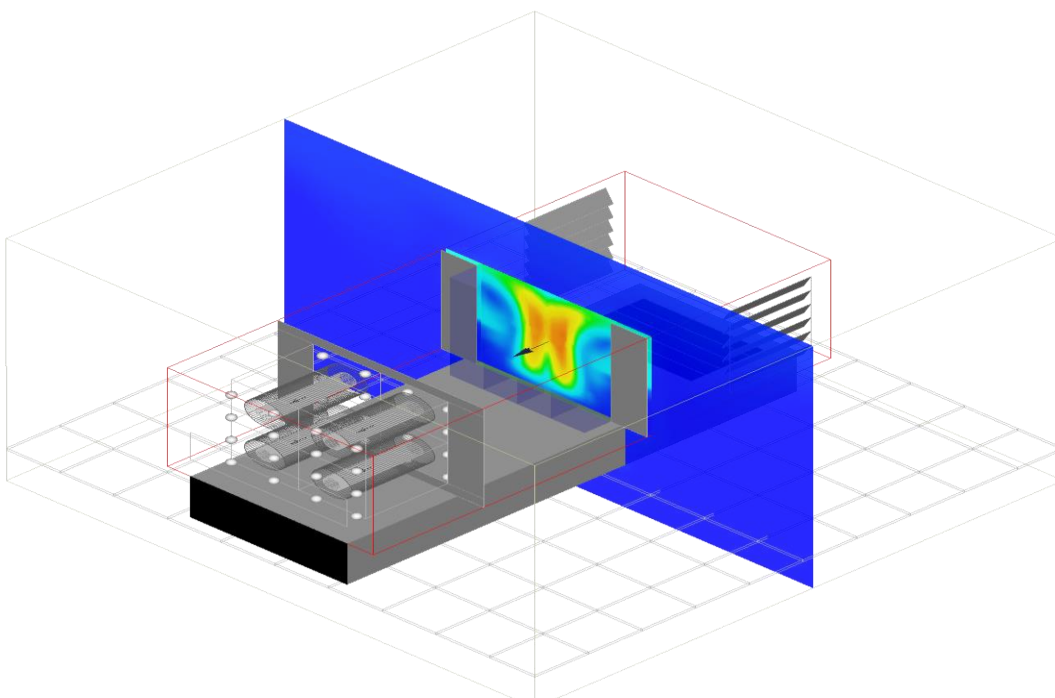

Figure – 13: Result plane location for baseline model without static structure

Temperatures are measured by placing an array of sensors at the inlet of the vents for the outside air and return air. Temperature sensor are also place on the result plane to capture the temperature variation across the plane.

Temperature range is a useful parameter in design the control strategy for the air handling unit. Temperature range is defined as the difference between the maximum temperature and minimum temperature in the plane of measurement or interest.

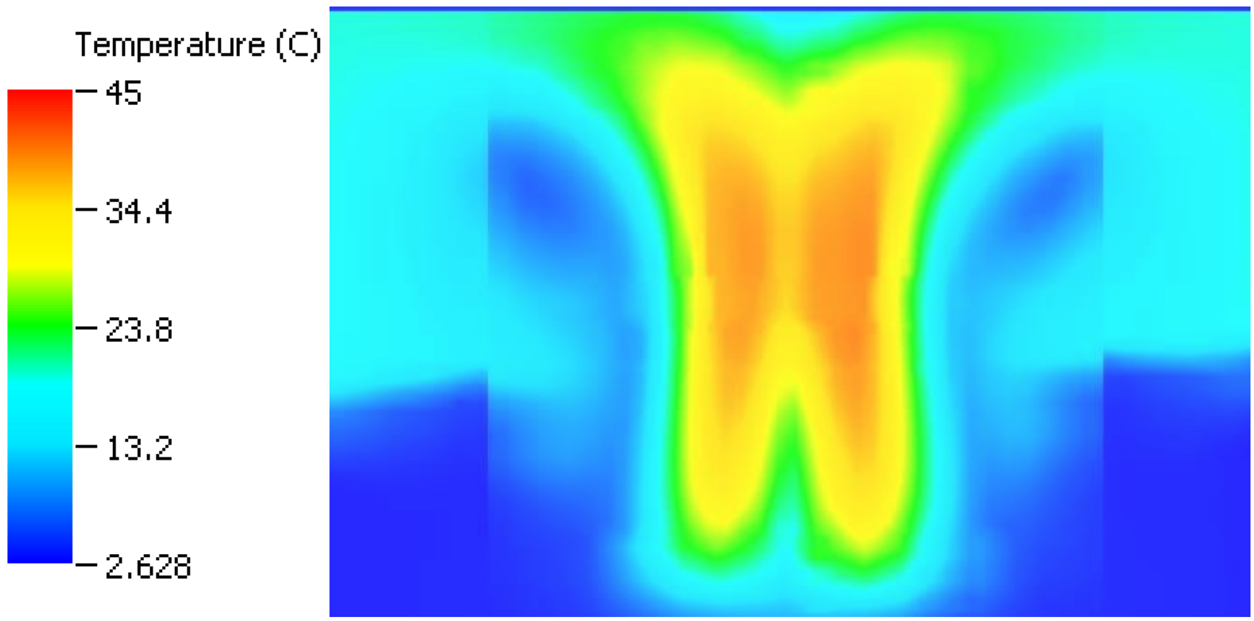

Figure – 14: Temperature plot of the baseline model without static structure

The temperature plot shows that hotspots are concentrated at the middle of the mixing chamber since the cold outside air gushes from the sides. It clearly shows that the mixing chamber has poorly mixed the air streams

The calculated range effectiveness for this model was 21% and the overall pressure drop that was measure across the fans were 3.34 inches of water. Thee pressure drop was calculated by placing a differential pressure sensor after the fan array with reference sensors before the fan array. The temperature range was 33°C.

4.3 STATIC MIXERS

Static mixers are fixed structures that are placed inside the mixing chamber to improve the mixing process by redirecting and recirculating the air streams within the mixing chamber thereby increasing the effective space and time for the mixing process. Addition of such static structure within the AHU's mixing chamber will come at a cost of pressure drop across the structure. Most cases the benefit of adding the structure overcomes the cost of pressure drop it creates.

4.4 MODEL WITH STATIC STRUCTURE

After the baseline study was done, another model with an indigenous static structure was modelled and analyzed using the computational fluid dynamics software. This static structure was modeled based on the insights gained from the Richardson number. Richardson number needs to be minimized to improve the mixing process and in order to do so the stream of air should be split into smaller jets and the jets needs to be in alternating fashion i.e. one hot jet and one cold. By doing this we are effectively reducing the characteristic length of the fluid and thereby we are also increasing the shear stress between the fluid layer, which creates a velocity difference between the fluids.

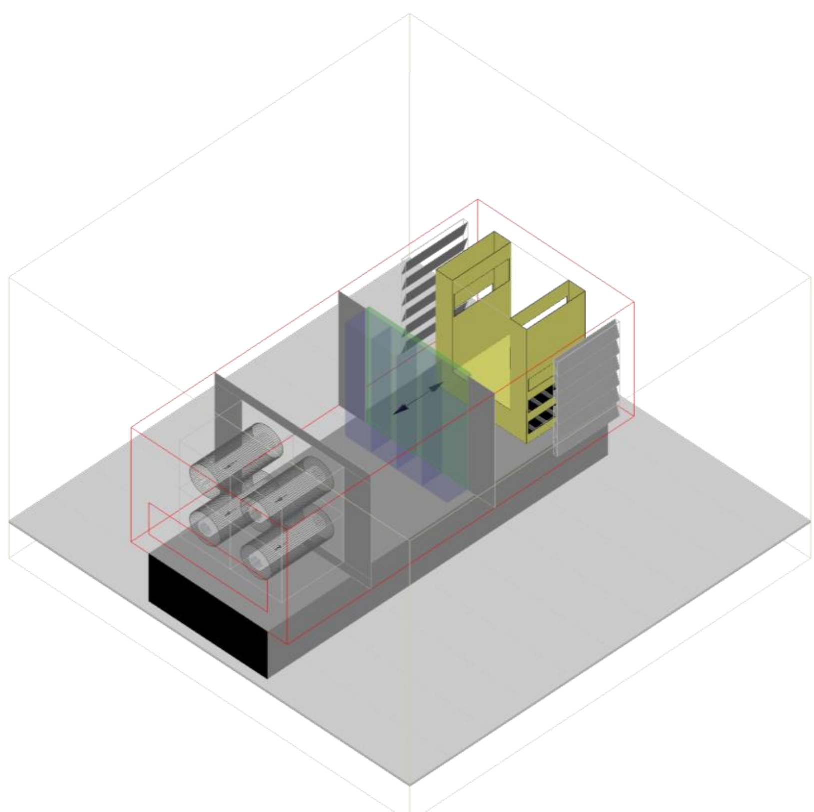

Figure – 15: CFD model with static structure

The Static structure is placed on top of the return air vent. The structure is a double inducing duct with three vents on either side facing the outdoor air vent and vent sizes are 200 * 825.5 mm and the structure has three such vents on either side.

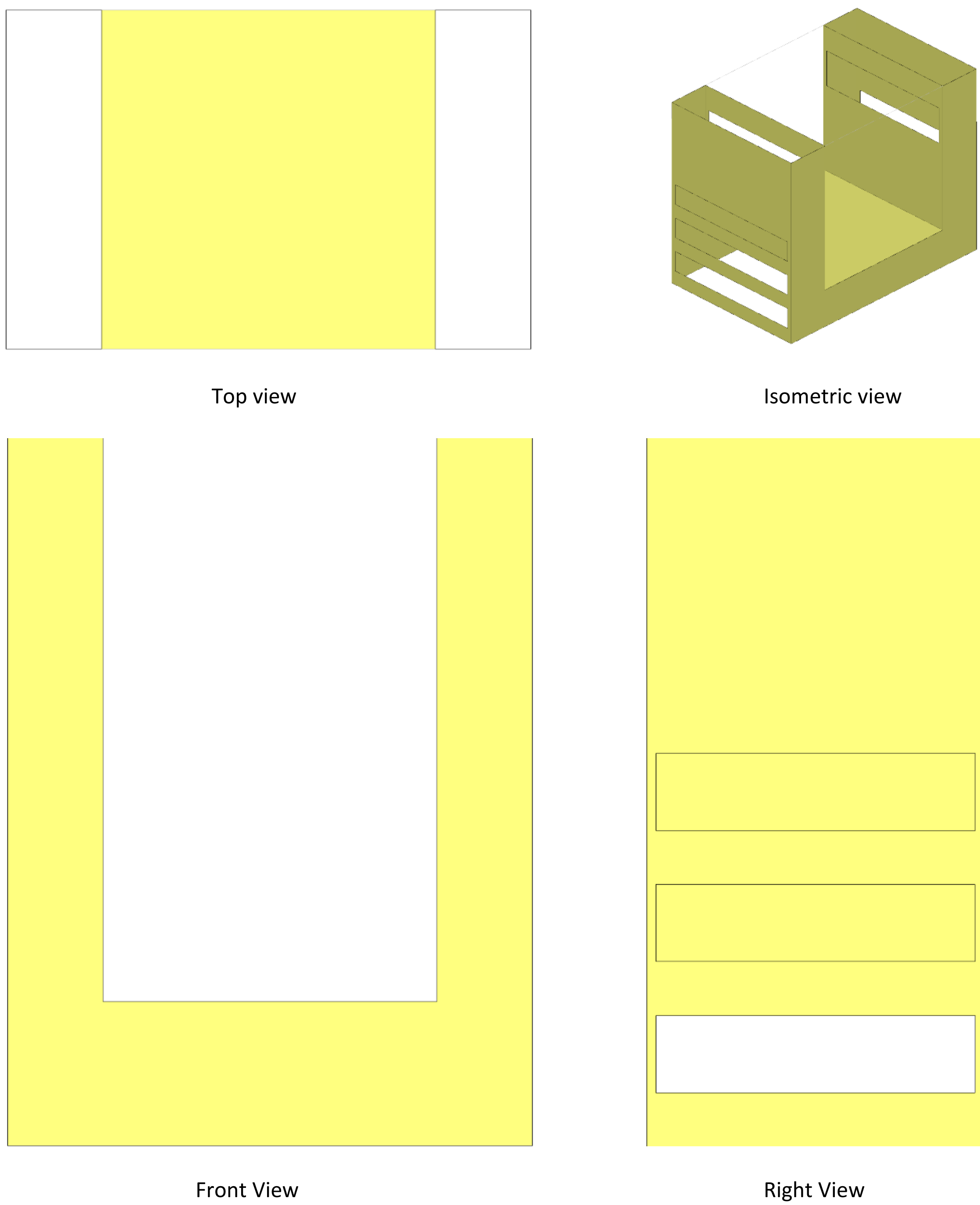

Figure – 16: Different views of the static structure

The result plane is plotted just before the cooling pad and the temperature plot shows signs of better mixing. This improvement could be attributed to velocity difference between the fluids.

The overall dimension of the static structure was 1842 * 876 * 3683 mm

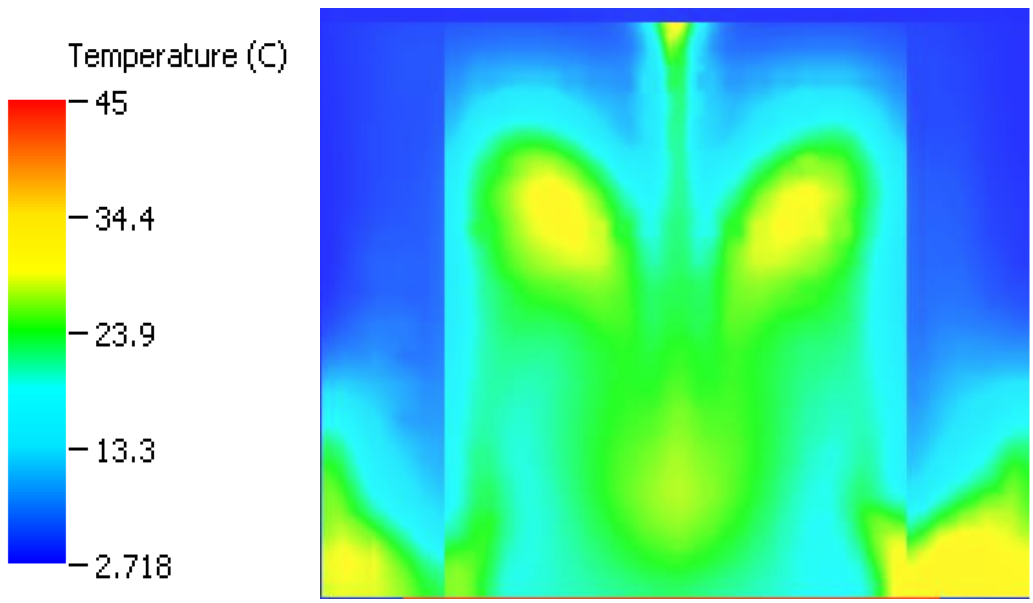

Figure – 17: Temperature distribution for the model with static structure

There is an increase of approximately 13% mixing effectiveness noted in the preliminary result. The temperature range of the stratified layer is also minimized from 33°C to 27°C.

The increase is achieved by placing double inducing ducts within the mixing chamber and thereby breaking up the return air into streams.

The overall pressure drop for the baseline case is 3.34 inches of water and the overall pressure drop for the modified case is 3.38 inches of water. So, there is an increase of 0.04 inches of Water pressure due to the static structure.

4.5 CASE STUDY – 2

This case study is carried out address the stratification that occur due to vertical sectioning of the evaporative cooling pads. Evaporative cooling pad sectioning will increase the stages of cooling. But this benefit comes at a cost of stratification problem due to partial portion of the pad provides cold air compared to the remaining portion.

Sectioning or Segmentation of the pad is carried out by running a MATLAB code to get maximized stages with given number of sections. For this case, the number of sections was considered as four and the code gave an output of 15 stages. The width each section was found out to be 6-inches, 12-inches, 18-inches and 24-inches. In this study, effect of stratification was also analyzed for pads sectioned equally.

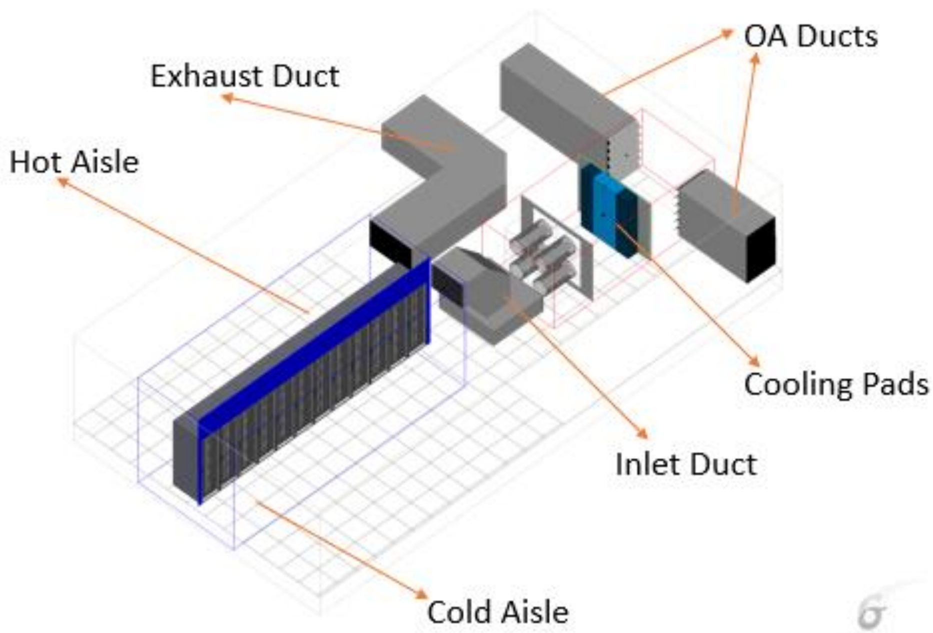

Figure – 18: CFD model of ASC 30 unit and the IT pod

To observe the effect of stratification, two sections of the pads was wetted and other two was not wetted. By doing so we can observe the stratification. Now we can improve the mixing by alternating the wetted pads.

The boundary condition for this study was set to 30 degree Celsius as the outside air temperature. The return air is not mixed with the outdoor as the outside air temperature is slightly above the recommended zone temperature.

The first model, the middle two pads were wetted and the and the other two was not wetted. The water flow rate provided for wetting the pads was 1 US gal/min and the temperature of the water was 15.6 degree Celsius.

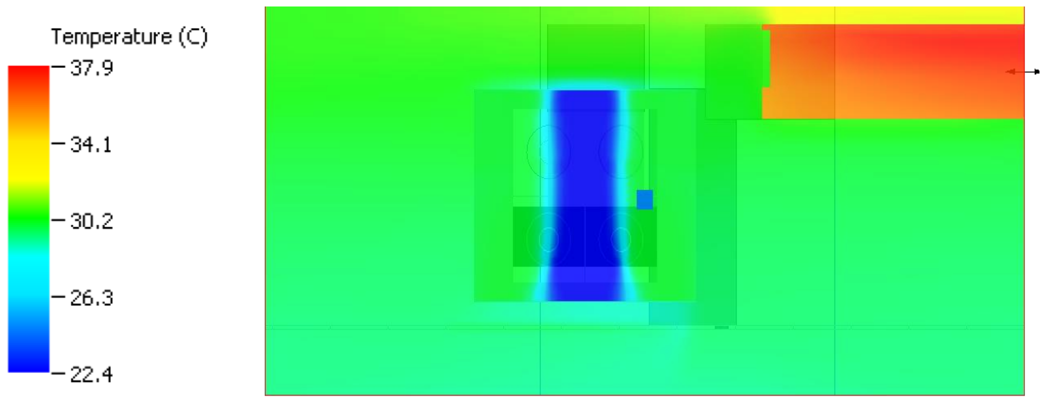

Figure – 19: Temperature plot when middle two pads were wetted

The result plane shows us that there is cold spot concentrated in the middle, the effect of stratification happening in the AHU is observed inside the cold aisle of the IT pod and the stratification is quantified by looking at the SHI value. For an ideal data center, the SHI value should be zero.

Figure – 20: Server inlet air temperature distribution inside cold aisle

The SHI for this case was found to be 0.115. Certain servers had higher inlet temperature because of the circulation within the cold aisle.

The other scenario that was modeled was by wetting the first and third pad. This was done in order to minimize the characteristic length of the fluids and thereby increase the mixing. Temperature profile was plotted just upstream of the fan array.

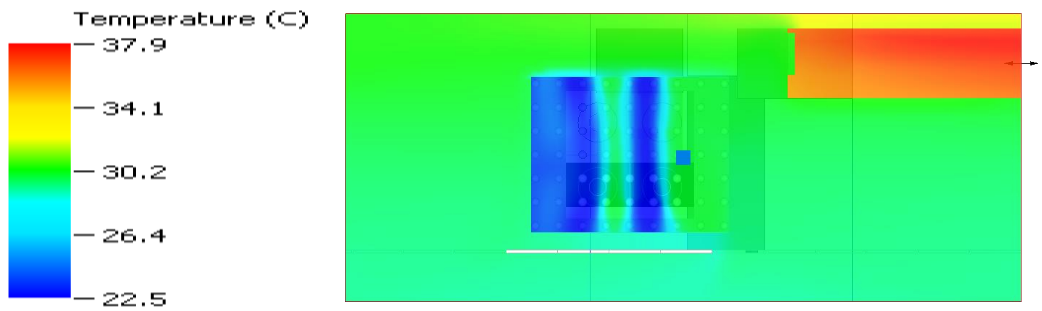

Figure – 21: *Temperature plot when alternating pads were wetted*

This case as better mixing compared to the previous model since the effective characteristic length of the fluid column was reduced and this improvement was transitioned to the cold aisle and the SHI value was reduced.

Figure – 22: Server inlet air temperature distribution inside cold aisle

The SHI value for this scenario was 0.112. The temperature at the inlet of the servers near the inlet is high because of low pressure region that is formed in the cold aisle due to recirculation of hot air into cold aisle and this recirculation is driven by negative pressure difference. The study shows us that the mixing process is better when we reduce the characteristic length of the fluid column.

CHAPTER 5 – CONCLUSION AND FUTURE WORK

The effect of thermal stratification was studied through the case studies that was simulated using the computational fluid dynamics software. Thermal stratification exists inside the air handlers at various parts under different operating conditions. When addressing the thermal stratification, static mixers were looked at in order to improve the mixing effectiveness, which was the parameter that was used to quantify the stratification. In the second case study we also looked at the progression of stratification inside the air handler to the cold aisle where it really matters as the server inlet air temperature will be higher than the ASHRAE recommended range. From the results it is evident that the we can improve the mixing in the stratified fluids by increasing the shear stress between fluid layers which will in turn increase the shear velocity. The mixing effectiveness increased in the first case study model with static structure which agreeing with the hypothesis that by increasing the shear stress between the fluid layer increases the mixing.

Second case study on the effect of stratification after the evaporative cooling pad also shows signs of improvement when alternating pads were wetted. This agrees with the inference from Richardson number that by reducing the effective characteristic length of the fluid column increase the mixing. Previous studies showed that influence of dampers on the process of mixing and this study showed the effect of static structures on the mixing process. The next phase of study would be to experimentally validate these hypothesis and designs. Other designs could also be stipulated and tested. The other study that could be carried out is by combining the previous study and this study.

REFERENCE

[1] Thesis Dissertation – M.S. Pavan Kaulgud, UTA

[2] Energy Consumption by Data Centers, http: ://www.datacenterknowledge.com/archives/2016/06/27/heres-how-much-energy-allus-data-centers-consume/.

[3] ASHRAE TC9.9 - Data Center Power Equipment Thermal Guidelines and Best Practices - https://tc0909.ashraetcs.org/documents/ASHRAE_TC0909_Power_White_Paper_22_June_2016_REVISED.pdf

[4] https://knowledge.autodesk.com/support/revit-products/getting started/caas/simplecontent/content/psychrometric-charts-part-1.html

[5] Damper Application guide – 1 - https://www.yorkland.net/downloads/Belimo_DamperApplicationsGuide.pdf

[6] Thesis Dissertation – PhD Betsegaw Kebede Gebrehiwot, UTA

[7] Richmueller, P. B. (2012). D.I.Y. Inspired Evaporative Cooler Design for Remote Military Applications. Retrieved December 12, 2016, from https://muellerdesignlab.wordpress.com/2012/04/20/diy-evaporative-cooler-design/

[8] Environmental fluid mechanics: Current issues and future outlook
By B. Cushman-Roisin, C. Gualtieri & D.T. Mihailovic´

[9] "Thermal Mixing of outdoor and return airflows in typical Air-Handling units" by Harry J. Sauer

[10] "Thermal metrics for data centers: a critical review" by Alfonso Capozzolia, Gianluca Seralea, Lucia Liuzzoa, Marta Chinnicib

[11] Future facilities 6Sigmaroom Help – Evaporative cooling pad

[12] Data center facility at Dallas by Mestex, Inc.,

[13] Technical Guide for ASC - 2 by Mestex, Inc.,

[14] The Different Technologies for Cooling Data Centers – White paper # 59 by Tony Evans

[15] Adiabatic mixing - https://nptel.ac.in/courses/112105129/pdf/R&AC%20Lecture%2028.pdf

[16] Open Compute Project - https://www.opencompute.org/

APPENDIX – A

```
% This code calculates width of direct evaporative cooling media sections
% that will maximize the number of stages for a given number of sections.
clear; clc; close all;
%% Inputs
Ns = 4; % number of sections
L = 60; % total width of the DEC media (length unit can be m, in, ...)
%%
L1 = L/sum(1:Ns); % smallest section width
Ln = L1*[1:Ns]; % all sections
%% Initialize
C = cell(1,Ns);
a=[];
q = 0;
Tb = zeros(100,Ns+1);
%% Loop
for i = 1:Ns
C{i} = nchoosek(Ln,i); %Find possible combinations
[row,col] = size(C{i});
if i == 1;
disp('Width of each DEC section (same unit as input total length) ')
Width = C{1}
else
end
for k = 1:row
m = sum(C{i}(k,:));
a = [a;m];
q = q+1;
%% 1 & 0 indicate which pump is on. (4 pumps considered)
for ij = 1:col
if C{i}(k,ij)==Width(1);
Tb(q,1) = 1;
elseif C{i}(k,ij)==Width(2);
Tb(q,2) = 1;
elseif C{i}(k,ij)==Width(3);
Tb(q,3) = 1;
elseif C{i}(k,ij)==Width(4);
Tb(q,4) = 1;
end
end
end
```

```
end
b = round(a./L*100*10)./10;
Lb = length(b);
%% Results
% column 1 is the stage number
% columns 2 through 5 indicate which pump section needs to be on
% column 6 is percentage of wetted media
Tb = [ [1:Lb]' Tb(1:Lb,1:Ns) b];
[Stages, ia, ic] = unique(Tb(:,Ns+2));
Number_of_Unique_Stages = length(Stages)
figure('Color',[1 1 1]);
set(gca,'Color',[1 1 1]);
plot(a,b,'.')
xlabel('Width of DEC Media')
ylabel('Wet DEC Width (%)')
axis([0 L 0 100])
%% Print
formatSpec = '%5.0f %5.0f %5.0f %5.0f %5.0f %5.1f \n'; %
fprintf(formatSpec,Tb')
SortedTb = sortrows(Tb,Ns+2)
```

Biographical Information

Anto Joseph Barigala Charles Paulraj received his bachelor’s degree in Mechanical Engineering from Anna University, India in June 2016. He received his master’s degree in Mechanical Engineering from University of Texas at Arlington in May 2019. His research focused on energy efficient operation of cooling systems in data centers. Anto served as a research assistant in Electronics, MEMS and Nanoelectronics Packaging Center and worked on various industry-funded projects as a graduate student researcher in NSF funded Industry University Cooperative Research Center called Center of Energy-Smart Electronic Systems (ES2).