OPTIMIZATION OF WATER AND ENERGY USE IN INDIRECT EVAPORATIVE COOLING SYSTEM BY CFD

By

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Abstract

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Data center cooling is becoming important to maintain environmental condition for ITE-Information technology equipment. Conventional evaporative wet cooling media lowers the temperature of the dry bulb temperature of incoming air in which heat transfer to water film and sensibly cool the product air by increasing moisture into air. Wet cooling media technique requires large quantity of clean water, which is reducing area where water is very less. Therefore, indirect evaporative cooling is good alternative. Chiller plants/Indirect Evaporative cooling(IEC) are used as a cooling source for larger data centers as they are more efficient in heat transfer without adding moisture into product air. In this study, Chilled water is supplied to the heat exchanger unit ASC- 8 row copper coils indirect heat exchanger. Hot air stream across the coils transfer heat energy to the chilled water inside the coil and excessive moisture in secondary air will be condensed before the outlet as it will reach to the dew point. Focus of this study is to identify the various configurations for cooling coils. Parametric study has been carried out to show the impact of both water and air velocities on coil performance. Variable primary flow of water in chilled water cooling unit has been studied to improve efficiency with reducing pumping power. Weather bin analysis of DFW area for several months has been done. Optimization of coil design has been studied by maintaining balance with coil face area, air pressure drop, and water pressure drop parameters. Numerical simulation data has been validated with Aztek indirect evaporative cooling unit.

Table of Contents

List of Illustrations

List of Tables

NOMENCLATURE

CFD	Computational Fluid Dynamics
DEC	Direct Evaporative Cooling
IEC	Indirect Evaporative cooling

CHAPTER 1 INTRODUCTION TO INDIRECT EVAPORATION COOLING

1.1 Why Evaporative cooling system?

The amount of energy needed for cooling the data center is huge and increasing day by day in cost and environmental impact. Where we heard about electronic components and processors, it obvious comes with heat when it works and to remove that heat is necessary to maintain efficiency of processor and. Therefore, cooling of the data center is essential and so does to maintain the cost. The most efficient and worldwide method for data center cooling is air-cooling. In air cooling, servers are installed in racks. Cool air passing through rack level cooling and dissipates the heat from the servers. Air cooling is not more effective technique for cooling as it creates contamination and need more power to run fan. Developers have moved to green cooling technique to save energy in unique way. Besides air cooling, evaporative water cooling, oil immersion cooling has gained rapid acceptance in data center cooling area. [1].

Evaporative cooling has been taking place over air-cooling system because of the cooling process depends on evaporation of water which is not harmful to environment as well as require less energy. Mechanical cooling system requires higher cost for installation and higher energy where HVAC convert 100% outside air into cool air. Evaporative cooling unit can be used in residential area, business building, and data center cooling. Evaporative cooling considers as green cooling system as it does not require any chemical reaction and does not depend on hazardous material. It depends on chilled water evaporative cooling unit. This cooling system can work efficiently in any weather. As shown in figure [1] dry bulb temperature from outside air enters through the system through fan. This air is passing through evaporative cooling pad and reduces the dry bulb temperature and this chilled air is passing to the datacenter cooling. Evaporative cooling unit has two distinguished system; Direct Evaporative cooling unit (DEC) and Indirect Evaporative cooling system(IEC).

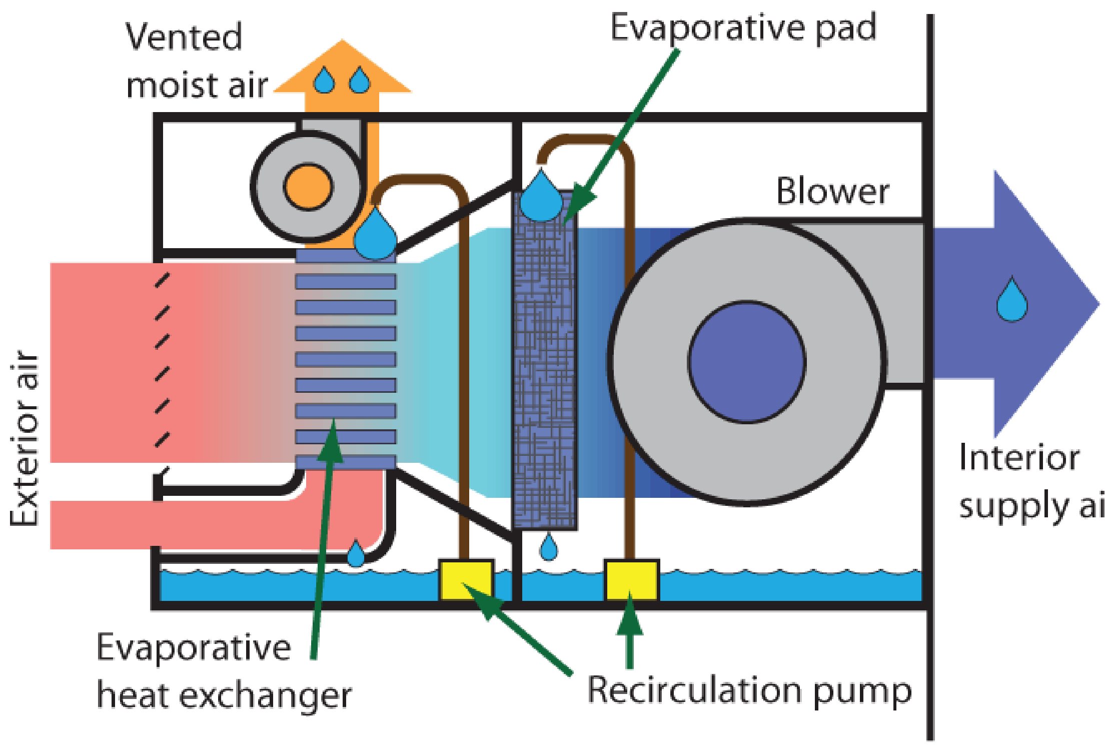

Figure 1 Evaporative cooling system[2]

1.2 Purpose of Direct Evaporative cooling system

Direct evaporative cooling is not new technique has been found in recent years. It has been in usage for primary cooling in house in urban dry places. Researchers studied to improve efficiency of DEC unit so that it can be used in commercial level. Direct evaporative cooling works on ambient dry bulb temperature passing through fan through wet cooling media. This wet cooling media cools the hot air till 80% efficiency. Heat absorbed by the wetted porous media which evaporates through water. This procedure is adiabatic process as shown in figure [3]. Cool air coming out is moistened because of direct contact through water. Direct evaporative cooling unit works on the principal that sensible heat is converting into latent heat. Sensible cooling capacity defines as to reduce the temperature where latent heat capacity helps to remove moisture content from the air. The latent heat follows the water vapor and diffuses into the air. There is various type of wetted media which are available in market like cellulose, fibers, or a spray of water. Efficiency of Evaporative cooling best describes by saturation effectiveness and system resistance curve. Saturation effectiveness is the difference between dry bulb temperature difference of incoming and outgoing air. An efficient wetted pad can reduce the air temperature by as much as 95% of the wet-bulb depression.

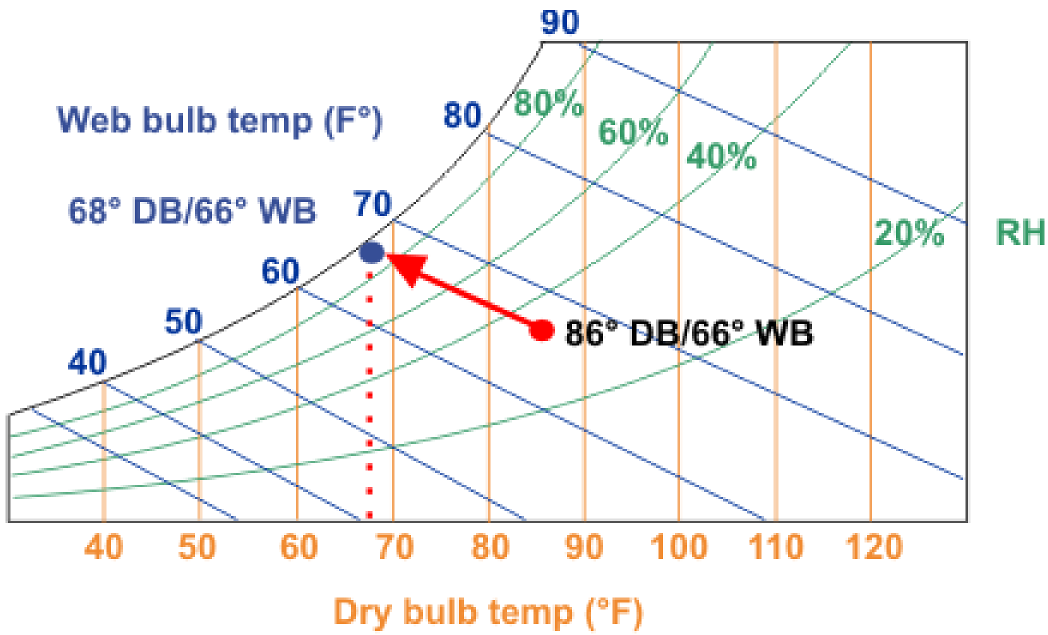

Figure 2 Adiabatic process of evaporative cooling system[4]

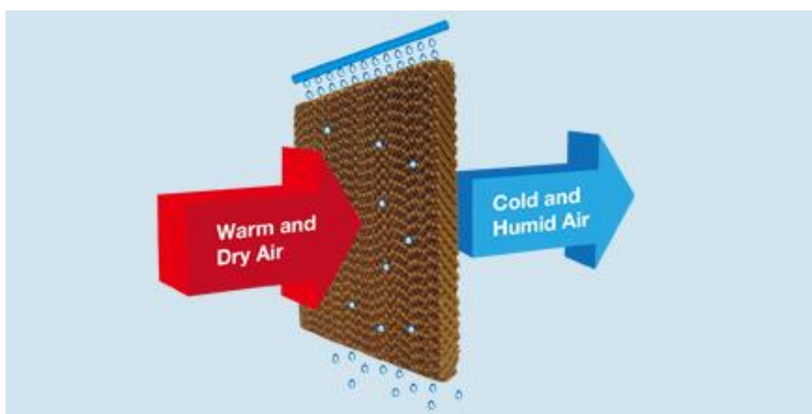

Figure 3 Evaporative cooling system [3]

1.3 Indirect Evaporative cooling system

Indirect evaporative cooling itself defining the effect of indirect evaporation of water. It cools down the outside dry bulb temperature without encountering water. As shown in figure [4], there are two wet and dry passages for cooling. Primary outside air passing through the dry passage and secondary cool air meeting water and reduce the primary air temperature. The surface of wet passages is wetted by spray water, so that water film evaporates into the secondary air and decreases the temperature of the wall. Therefore, heat is transferred from primary to secondary air without the introduction of moisture into the primary air stream. The air leaving the dry side of the cooler has a lower wet-bulb temperature than the ambient. Indirect evaporative cooling unit is not efficient as direct evaporative cooling so, in summer time evaporative cooler has been placed in series for higher efficiency. Indirect evaporative cooling overcome the disadvantages of Direct evaporative cooling unit of adding humidity in air. Although indirect evaporative cooling unit is not efficient as direct evaporative cooling.

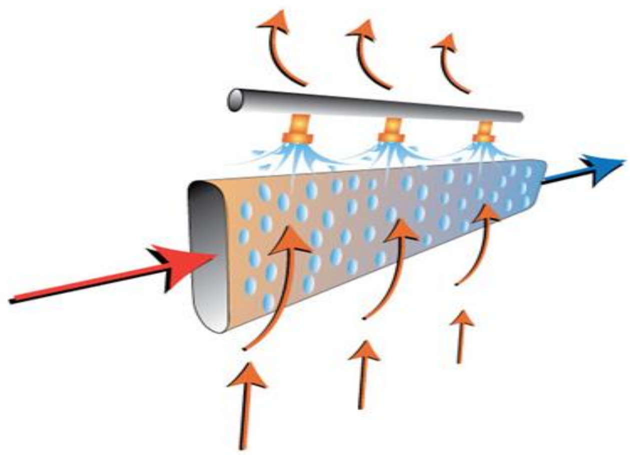

Figure 4 Indirect evaporative cooling system[5]

1.4 Combine Direct-Indirect Evaporative Cooling System

To overcome the disadvantages of direct and indirect evaporative cooling, there is another option for efficient cooling without adding humidity is Direct-Indirect evaporative system. In this system, dry bulb temperature from outside air passing through direct evaporative cooling unit and after this air passing through indirect evaporative cooling to remove moisture from air through latent cooling. A two-stage unit as shown in figure [5] idea helpful during the summer time for higher cooling in hot region. Most of the time, in two-stage system, indirect evaporative cooling unit has been placed before direct evaporative cooling system. First stage cools the outside hot air without adding moisture in it. Second stage cools this air by increasing humidity. Combine stage improve efficiency than individual unit. This unit reduces temperature till 25℃.

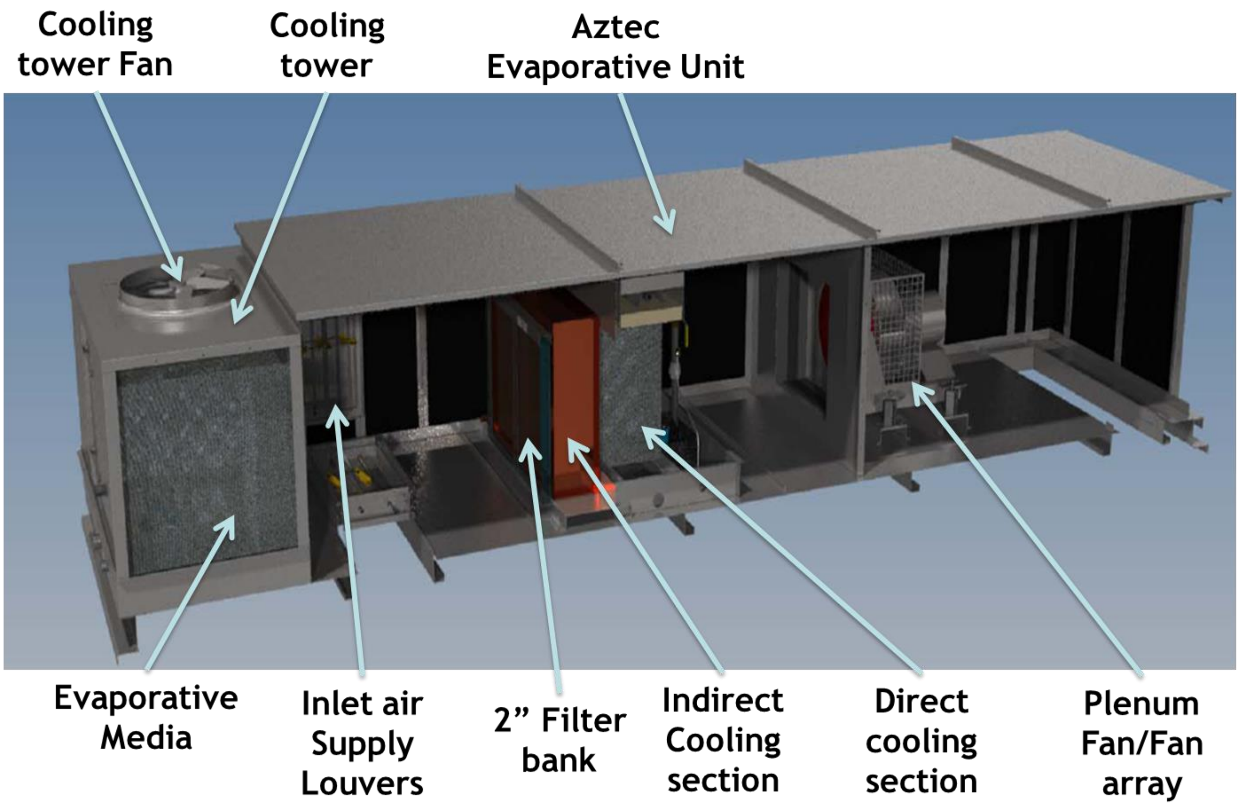

Figure 5 Combine direct-indirect evaporative cooling system [21]

1.5 ASHRAE Thermal Guidelines for Data Center Environment

ASHRAE thermal guidelines provides information about implications of efficiency for ITE cooling on data center operation. Guidelines provides for various temperature and humidity chart to implicate cooling desire. This update also defined additional two data center classes increasing the number of data center classes to four. Table 1 and Figure 6 show the 2011 Thermal Guidelines for Data Processing Environments – Expanded Data Center Classes and Usage. The Thermal Guidelines apply to the inlet air conditions to the IT equipment. Since 2008, the recommended range for temperature and humidity of inlet air conditions were expanded, enabling increased number of economizer hours and reduced mechanical cooling. The industry now recognizes that outside air can be used with economizers to vastly decrease mechanical cooling in data center implementations, that there is room to exploit alternate renewable and sustainable cooling technologies like airside and water-side economization[5].

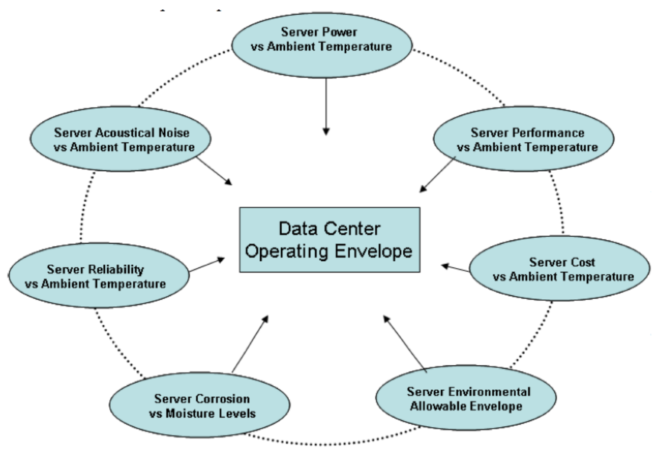

Figure 6 Data center operating envelope [5]

Table 1 2011 ASHRAE thermal guidelines [5]

Classes (a)	Equipment Environmental Specifications							
	Product Operations (b)(c)					Product Power Off (c) (d)		
	Dry-Bulb Temperature (˚C) (e) (g)	Humidity Range, non-Condensing (h) (i)	Maximum Dew Point (˚C)	Maximum Elevation (m)	Maximum Rate of Change(˚C/hr) (f)	Dry-Bulb Temperature (˚C)	Relative Humidity (%)	Maximum Dew Point (˚C)
Recommended (Applies to all A classes; individual data centers can choose to expand this range based upon the analysis described in this document)								
A1 to A4	18 to 27	5.5ºC DP to 60% RH and 15ºC DP						
Allowable								
A1	15 to 32	20% to 80% RH	17	3050	5/20	5 to 45	8 to 80	27
A2	10 to 35	20% to 80% RH	21	3050	5/20	5 to 45	8 to 80	27
A3	5 to 40	-12˚C DP & 8% RH to 85% RH	24	3050	5/20	5 to 45	8 to 85	27
A4	5 to 45	-12˚C DP & 8% RH to 90% RH	24	3050	5/20	5 to 45	8 to 90	27
B	5 to 35	8% RH to 80% RH	28	3050	NA	5 to 45	8 to 80	29
C	5 to 40	8% RH to 80% RH	28	3050	NA	5 to 45	8 to 80	29

1.6 Understanding of Psychrometric chart

The ASHRAE Psychometric chart is a graphical form of the thermodynamic data of air. It helps to understand thermodynamic properties of air and the air-conditioning system process better. The following data is shown on the psychometric chart. (i) Dry bulb temperature (ii) Wet bulb temperature (iii) Relative humidity (iv) Saturation temperature/dew point (v) Enthalpy or total heat (vi) Humidity ratio (moisture amount) (vii) Specific volume. As per the Psychrometric chart, the dew point of the incoming air is 57 °F. If the temperature reduces due to cooling unit, air reduces the capacity of holding moisture, so it increases relative humidity and moisture starts condensing at dew point on the cooling coil.

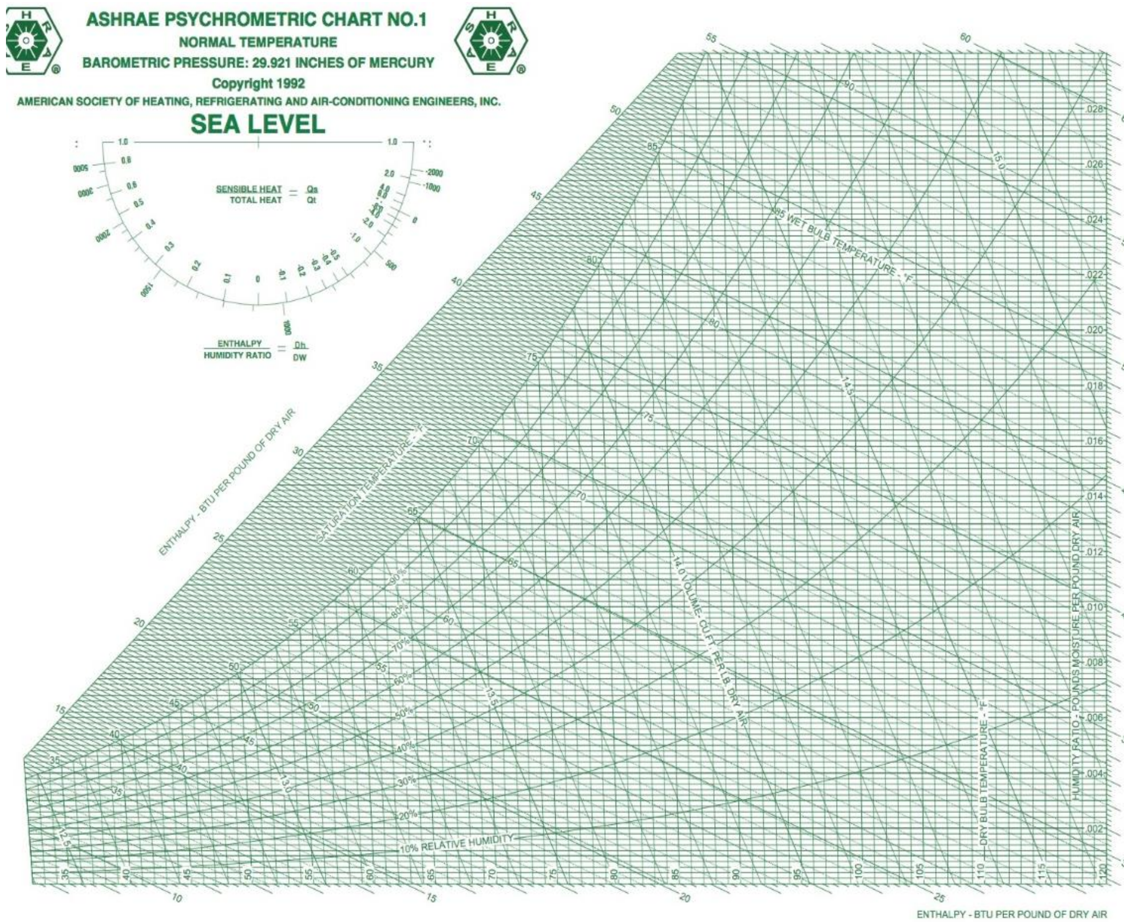

Figure 7 Psychrometric chart [12]

In the cooling unit, moisture content of incoming air remains constant after entering the unit. Moisture content at different temperature is shown in figure 8 in psychrometric chart on vertical axes. Even when the air gets to the room temperature of 75°F, its moisture amount is still constant and its relative humidity is 71%. The same amount of moisture results in a lower percentage relative humidity at 70°F than at 62°F.

1.7 Chilled Water cooling system

As shown in figure 9, chilled water cooling coil system is more efficient and cheaper option for green cooling system with less water consumption and less mechanical equipment. In this system, chilled water from the cooling tower approximately at 45 °F passin through copper coil. Dry bulb temperature from outside passing through this unit. Where chilled water cools outer surface of the copper coil sensible so when hot air passing through this chilled pipe surface it transfers heat to pipe and cools down. The cooling coil is made of copper tubing bent into a serpentine shape with aluminum fins bonded to the copper tubing to increase the heat transfer area. The air handler also contains air filters that remove impurities from the air that is being drawn over the coil by the fan. The fan is also called a blower. A motor drives the blower via a drive belt that has a V section. The air handler may also be furnished with a heating coil that adds heat to the air when heat is required. Most chilled water air handlers contain a section called a mixing box. The mixing box is a sheet metal section with two openings in it as shown in figure 9. There is a duct connected to each opening and a damper located within each opening. One duct is used to bring return air from the conditioned space back to the air handler. The second duct is connected to the outdoors and is used to introduce outdoor air for ventilation purposes. This is an energy saving device in four pipe systems and a necessity in two pipe systems. In buildings with two-pipe systems, the building may be circulating hot water to provide heat, while some spaces with high internal loads require cooling. Under these circumstances, outdoor air is the only medium available to provide cooling.

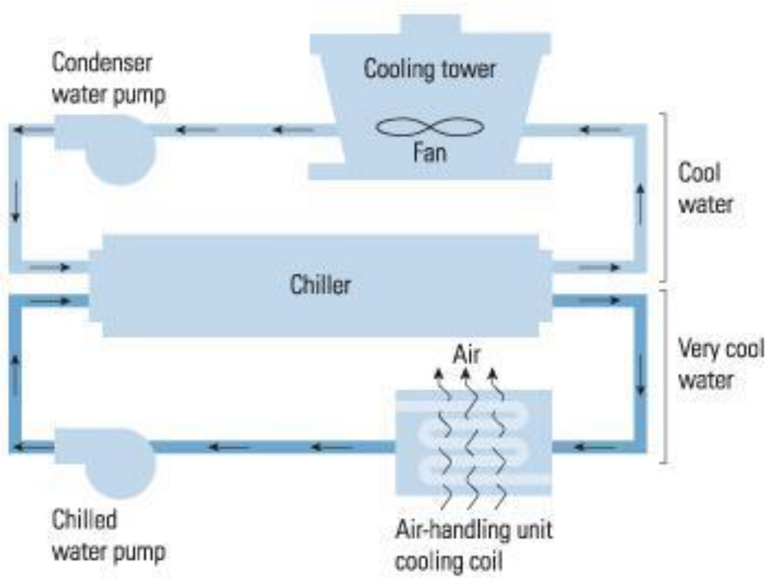

Figure 8 Chilled water cooling coil system [13]

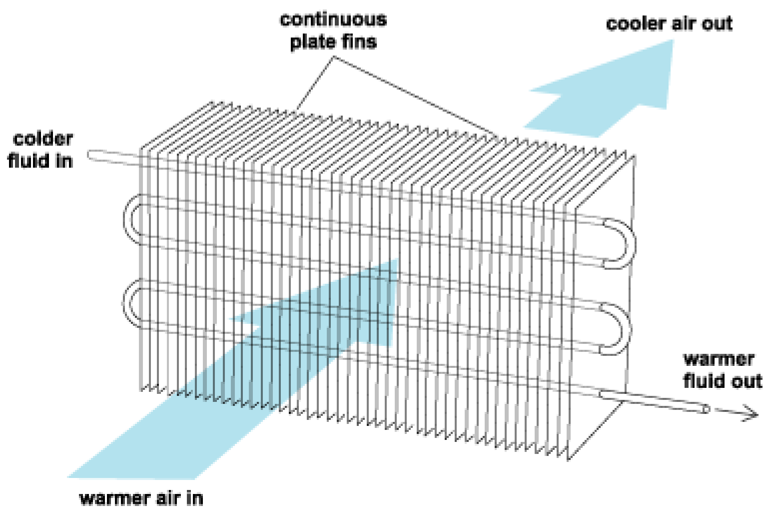

Figure 9 How water-air heat exchanger is working [14]

Some region in USA, as show in figure 10, USA has been divided as per the weather region. Central part of USA like Texas, Utah, Arizona are dry state which require direct and indirect both system to maintain moisture during cooling where eastern part of USA is moist cool region which can be work on indirect evaporative cooling system. Though, south eastern part like Florida, Tennessee is higher moist hot area where direct evaporative cooling system may increase moisture in data center. So, it requires IEC unit as well as chilled water cooling unit to maintain cooling in data center.

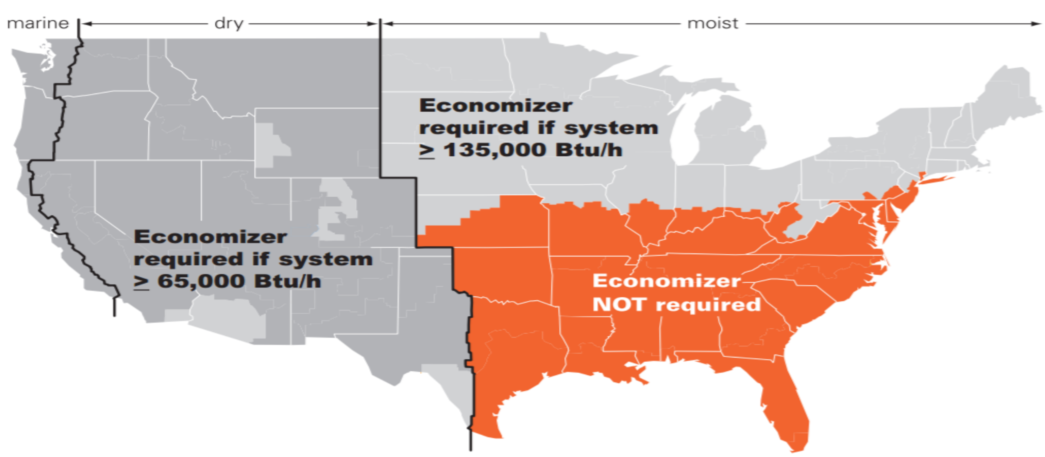

Figure 10 USA map categorized by weather [23]

1.8 Problem Facing with chilled water cooling coil design

We facing problem with chilled water system is to provide sufficient water to each air handler unit. In data center cooling, water supply to each handler is in parallel path. Water will choose the least resistance path to flow which will end up supplying higher pumping power to resistive path. Total resistance of the coil depends on length, diameter, and pumping power. Balancing valves are installed on each branch to add resistance to flow to guarantee that each branch receives the volume of water it was designed to handle as shown in figure 11.

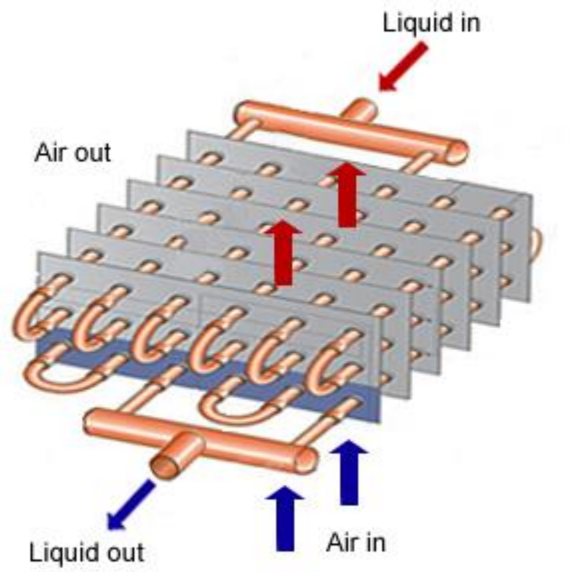

Figure 11 Chilled water colling coils system [11]

This study is to develop a CFD model for an Indirect Evaporative Heat Exchanger and validate with existing experimental data. Parametric studies and design optimization of compact heat exchanger by considering coil water flow rate, geometry and inlet air flow rates. Variable flow rate study has been considered for detailed analysis and reduce pumping power. Using weather data analysis of Dallas/Fort worth area, study has been carried out for chiller cooling unit.

Lewis factor is important relation for heat and mass transfer in cooling tower unit. J. Klopper & D. Kroger[10] had investigated effect of Lewis factor in heat and mass transfer analysis of evaporative cooling and noticed that Lewis factor must be specified explicitly. They had done experimental study with different Lewis number at different temperature and humidity. They have concluded that evaporation of water effect the Lewis number and it changes the outlet temperature. Their research concluded that one should select Lewis number wisely for higher efficiency. Lewis factor is ratio of thermal and mass diffusivity as shown in eq (1).

$$Lewis\ Factor = \frac{\alpha}{D} \tag{1}$$

Indirect evaporative cooling is one type of heat exchanger. Effectiveness of heat exchanger depend on number of transfer unit(NTU). Hsu et. Al. during their experiment they found that cooling effectiveness of each configuration increases with increasingly dry channel NTU and reaches maximum values at large NTU.[15] Results showed that it has almost no effect on the co-current and counter-current configurations as shown in figure 12 and its degrading effect on efficiency of ross flow is accelerated when the ratio of dry-passage length to that of the wet passage is large.

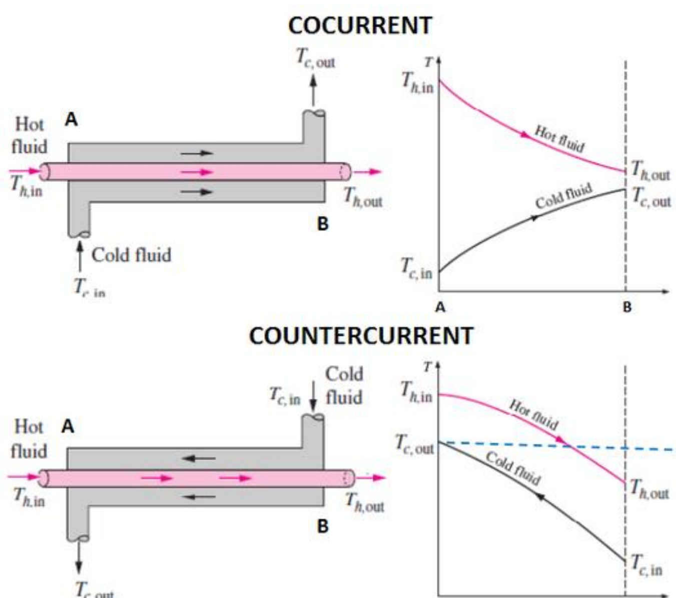

Figure 12 Co-current and Counter-current heat exchanger flow [16]

Pescod carried out one study using plastic pipes in indirect evaporative cooling unit with small prostution [17]. Pescod expected less heat transfer will be placed between plate and air because of low thermal conductivity of plastic. Though, it found that efficiencies of wet surface had given higher efficiencies of heat exchanger; which proved that wet surface gives higher efficiencies than dry surface.

Dreyer experimented three different setup for indirect evaporative cooling and its lewis factor: In first model, he took variable Lewis factor for indirect evaporative cooling unit and studied for saturation effect in secondary air; second model ran taking Lewis number unity and found negligible effect on spray water evaporation and secondary air never reaches to or above saturation, so he simplified model takin water temperature constant in crossflow heat exchanger with initial design purpose [18].

Chilled water cooling unit has been taking place in all over the world from data center cooling to residential building as cooling unit. Hydeman found out optimized design for operating condition to reduce pumping power by increasing efficiency of cooling unit [19]. His optimized model based on overall power consumption of plant, a plurality of chiller plant and pumps. Moreover, he had considered arrangements of water pumps for condenser. His experiment showed that when the projected energy savings exceeds the energy saving threshold value, sending the optimized chiller plant subsystems output to a building control system.

During the experiment one another aspect of study for indirect evaporative cooling unit had been focused. Tulsidasani had performed experiment to optimized coefficient of performance for IEC unit by considering air and water velocity [20]. Analytical and experimental study of optimum value of COP had been compares and validated summer months of May and June; the agreement is satisfactory. Hence, this simple analysis can be used to develop the optimum IEC size for maximum cooling performance. It is found that there exists an optimum value of process air velocity for which COP is maximum. The maximum value of COP increases with decreasing wet air velocity without significantly accepting process air cooling.

Moisture content of the air poses serious risk for damaging ITE of data center. Thus, Morentsen studied about moisture modelling of incoming air in room and heat and moisture transfer in walls by applying them as fluid walls [8]. In a 3D configuration, the impact of different boundary conditions are investigated and the results are discussed. The changes of boundary conditions that are studied are velocity, moisture and temperature conditions for room air. They had concluded that at higher temperature, when outlet temperature reaches to dew point it condense excessive water droplets.

Moving forward to the main research of this paper is based on chilled water cooling coil. X. Tang has given important results from his experimental setup of 8-row chilled water cooling coil[9]. He had found out the physical model involved with solution of transient energy solved with partial differential equation. Temperature distributions in the direction of water flow are handled by using a finite-volume aapproximation to the partial differential equations. A fin efficiency method is utilized to characterize the temperature distribution in fins in the air flow direction.

2.1 Project Methodology

Considering literature study and ASHRAE thermal guidelines along with real life experimental setup from MESTEX unit, this research showing CFD modelling of chilled water cooling (water-air heat exchanger) and validation with MESTEX unit. This paper also considers water and energy saving methods for implementing further. This study has been divided into two parts: (i) moisture interaction model and validation (ii) 8-row copper coil chilled water cooling unit and validation. Chapter 4 shows CFD validation through moisture interaction and Chapter 5 describes second part of this study.

Motivation of this research are listed below:

- Identifying the cooling range under various water mass flow rates to guide you to optimize the pumping power
- Study the effectiveness for different weather condition/ places demand for different cooling requirement to make compact model
- To find out the optimum range of air mass flow rate and water mass flow rate in order to reduce energy

CHAPTER 3 CFD MODELLIG FOR INDIRECT COOLING UNIT

CFD modelling of indirect cooling unit is based on species transport equation. In this research study commercial CFD software ANSYS Fluent is used for modelling approach. Indirect cooling unknit depends on relative humidity and cooling so it requires study of moisture interaction in system. This chapter is based on species of H_2O in air and how it affects when temperature reduces. This work gives validation of moisture interaction model with the experimental model.

3.1 Moisture interaction model and CFD validation

Indirect cooling working on sensible and latent cooling system. Chilled water cools the surface sensibly where air transfer temperature to the chilled water and cools. When air temperature reduces, relative humidity increases and it reaches to the dew point where species of H_2O starts condensation which calls latent cooling.

This study based on a room which has fluid wall at the inlet and outlet side at 0℃ and outside dry bulb temperature enters through hole as shown in fig 13 at 20℃ with 27% relative humidity. Study shows the species transport and heat transfer through walls. Mortensen's model can be used implementing following assumptions [8].

- Air and water flows are steady
- Negligible conduction in flow direction for water
- Water and air velocities are uniform
- Negligible energy storage within the air

Accounting these assumptions for indirect cooling unit, we can use Mortensen's model for validation purpose in this research. This model is for quite a flow over water surfaces. So turbulence model can be used for CFD modelling.

3.2 Moisture Modelling test case and Flow condition

To validate the species transport model, a test case with simple geometry is considered here. Two fluid walls placed at the inlet and outlet side where air is passing through the hole.

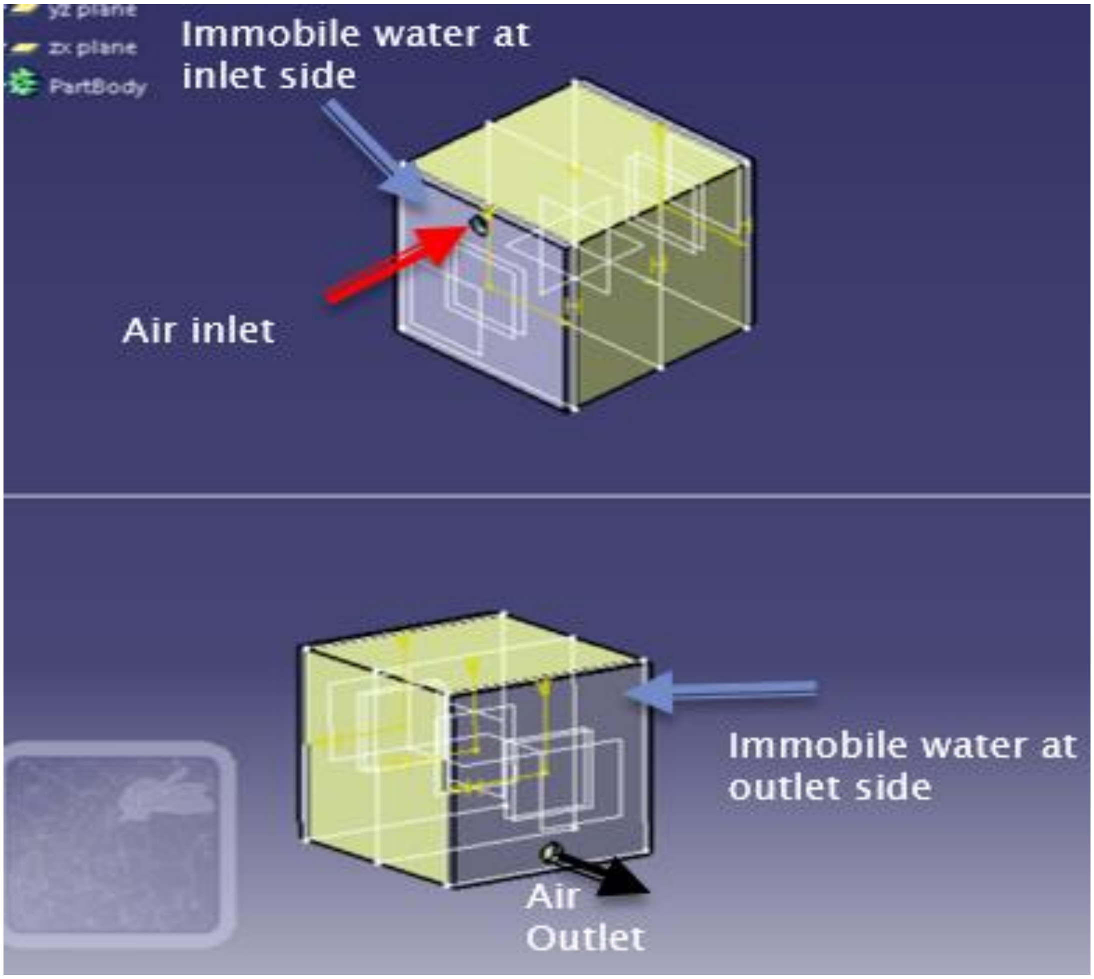

Figure 13 CAD model of fluid walls and room [8]

Geometry with all dimensions of the test case is shown in figure 13. All dimensions are given in m. Flow conditions are given in table 2.

Table 2 Flow conditions [8]

AIR INLET	
Temperature	20° C
Relative Humidity	27%

Mass fraction of water	0.00979 kg
Velocity	0.167 m/s
Fluid	
Water	Immobile
Temperature	0° C

Species transport equation has been used for moisture content in air.

3.3 Governing Equations

ANSYS Fluent solves continuity, momentum and energy equation along with species transport. Equation 2 represents continuity equation used for this case [7].

$$\frac{\partial \rho}{\partial t} + \nabla.\left(\rho \vec{V}\right) = S_{DPM} + S_{Other} \tag{2}$$

Where,

ρ = density of fluid phase

$\vec{V}$=velocity in vector form

S_{DPM}=Discrete phase model source

S_{other}=additional mass source

Momentum equation is shown in equation 3 for this case. Here acceleration due to droplet particle and viscous forces are considered [7].

$$\frac{\partial \rho \vec{V}}{\partial t} + \nabla.\left(\rho \vec{V}\vec{V}\right) = -\nabla p + \nabla.\tau + \rho \vec{g} + \vec{F}_{DPM} \tag{3}$$

Where,

p=Static pressure

τ=stress tensor

$\vec{g}$=gravitational acceleration

$\vec{F}_{DPM}$= DPM force acceleration

Water vapor in the air is modelled using species transport in ANSYS Fluent. Species transport equation uses convection-diffusion phenomena to transport water vapor in the air. The general equation is shown in equation 4 [7].

$$\frac{\partial \rho Y_i}{\partial t} + \nabla.(\rho \vec{v} Y_i) = -\nabla.\vec{J}_i + S_i \tag{4}$$

Where,

Y_i=local mass fraction of each species

$\vec{J}_i$=diffusive flux

S_i=creation of species by DPM

Heat transfer is governed by energy equation which is shown in equation 5 [7].

$$\frac{\partial \rho E}{\partial t} + \nabla.\left(\vec{v}(\rho E + p)\right) = \nabla.(k_{eff}\nabla T - \sum_j h_j \vec{J}_j + \bar{\bar{\tau}}_{eff}.\vec{v}) \tag{5}$$

Where,

E=enthalpy

k_{eff}=effective thermal conductivity

$\vec{J}_j$=diffusive flux due to species

3.4 CFD model validation and results

Using above governing equations for the flow simulation with enabling species transport equation with steady state analysis of k-ε turbulence model has been carried out. Hydraulic diameter is 0.0127 and turbulence intensity is 6% where Reynolds number is higher than 10^5 which lid to consider k-ε turbulence model[7] with enhanced wall treatment to derive condensation near wall. Thermal and mass diffusivity has been considered constant at 1.9*10^$^{-5}$ cm^2/s. Viscocity and density are mass weighted for calculation.

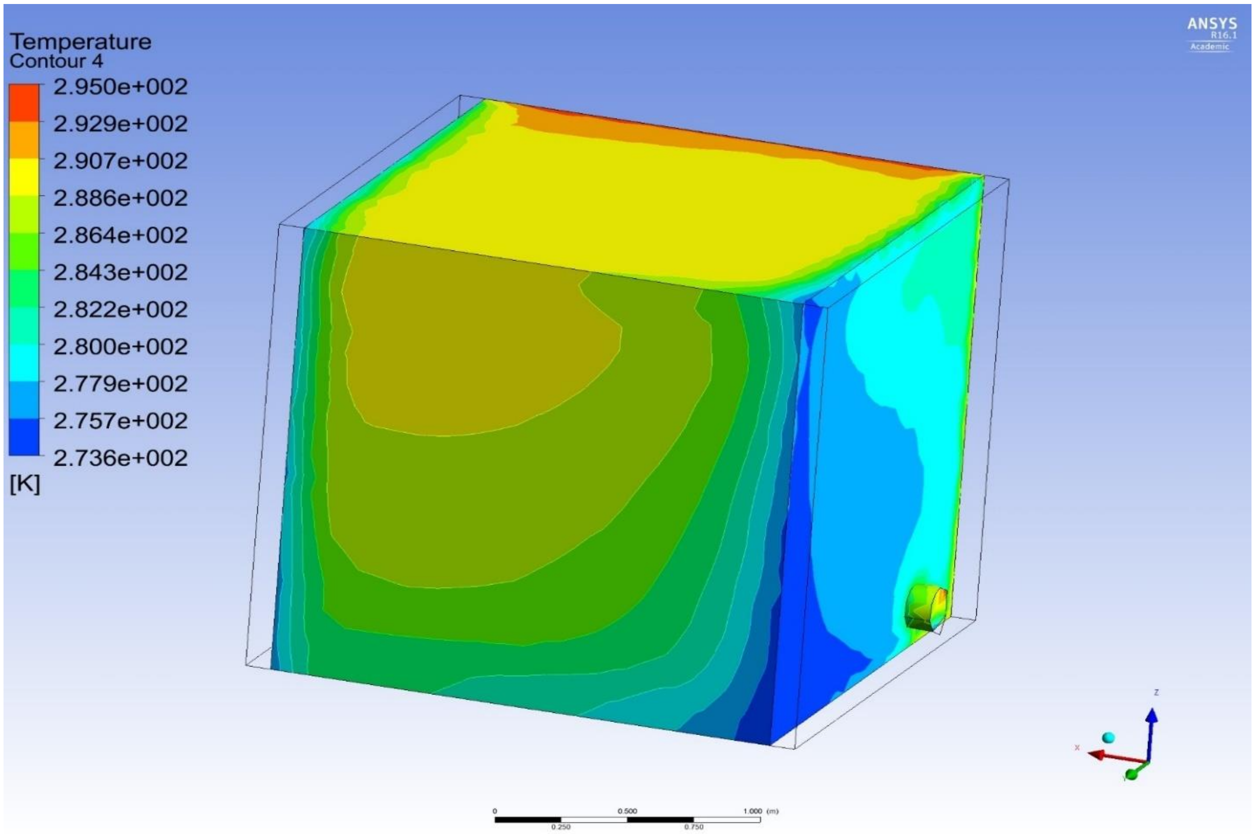

Figure 14 Heat transfer between air-water

As we can see in figure 14 that applying 293K temperature at inlet air reduced temperature at 284.3K. which shows almost 10K reduction. At the outlet side water started condensing which helps to reduce moisture content from air till 0.0076. Heat transfer rate of this system is 28.93K. This results show that CFD model for indirect cooling unit is validated with existing data. Likewise, relative humidity increases up-to 45% [8]. <u>It proves that Species transport model can be used further for this research work.</u>

4.1 CAD Model and dimension

ASC 8-row chilled water cooling coils has been designed in CATIA V5 CAD software using solid surfaces which are shown in figure 15. Dimension has been takes as shown in Table 3. Coils are made up of copper because of higher thermal conductive material.

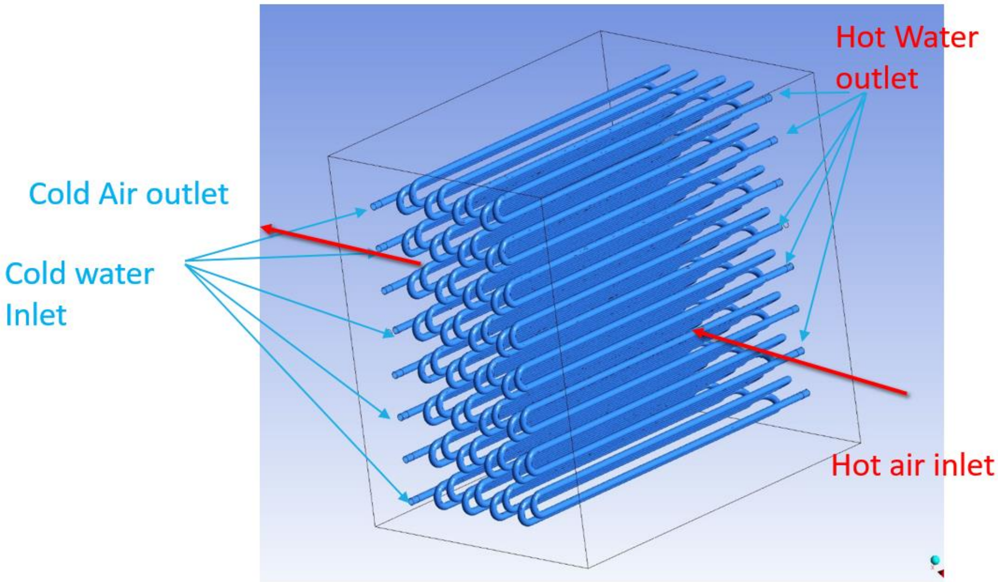

Figure 15 Baseline 8-row coil model [9]

Table 3 Dimension of baseline 8-row coil model [9]

Coil face width	0.6096m
Coil face height	0.6096
Coil face depth	0.264
Tube diameter	0.0127m

4.2 Flow condition

Flow conditions are shown in table 4. Baseline case of air velocity is taken as 0.6m/s. case is baseline case 8 row coils in parallel.

Table 4 Flow domain condition for baseline model [9]

Boundary Condition	
Air Inlet Temperature	27℃
Air velocity(m/s)	0.6
Relative Humidity (%)	34.9
Water Inlet Temperature	2℃
Water Inlet Velocity	0.5
Relative Humidity at Outlet	57.8%

After applying equation 2,3,4 and k-ε turbulence model to baseline case model it reduces temperature of air from 27℃ to 17.8℃ which is showing almost 10℃ reduction in outlet side as shown in figure 16. Results are validated with experimental study from ref as shown in table 5.

Reduction in Temperature	Temperature outlet Air℃	Water outlet Temperature℃
Numerical Study	10.10	12.20
Experimental study [9]	10.45	12.85
Error (%)	-3.49	-5.32

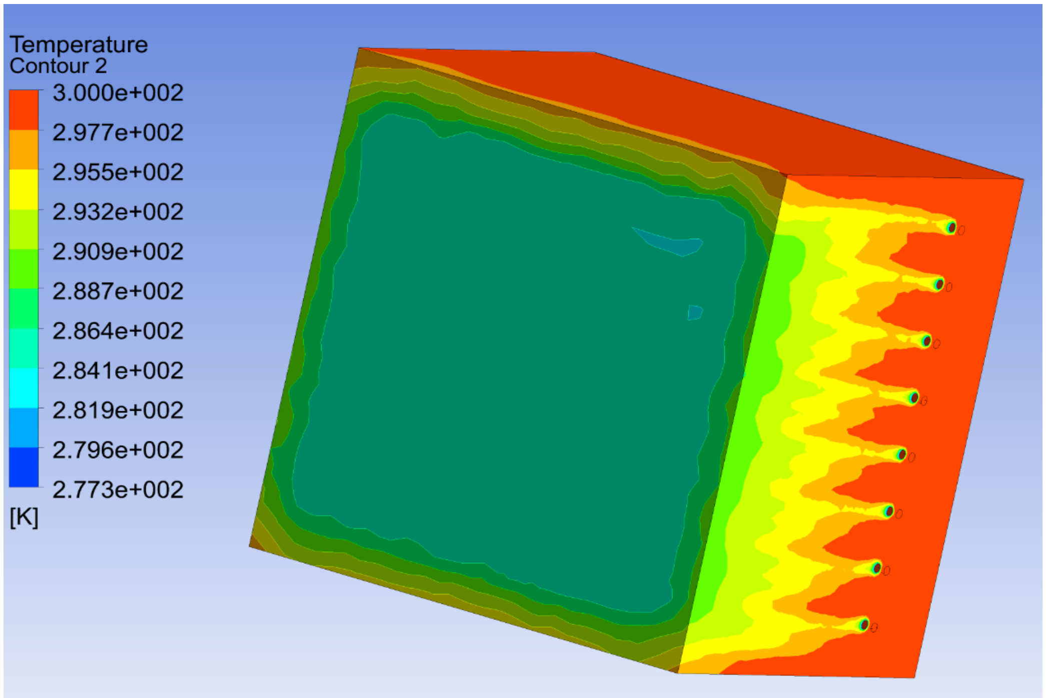

Figure 16 CFD Analysis of baseline case

Baseline case study proves that applied CFD modelling is appropriate and gives expected results. Moving further to make this study more realistic, research continues with validating CFD model with MESTEX unit with boundary condition. Flow condition has been takes as shown in table 5.

Table 5 Boundary condition for MESTEX unit [21]

Air Inlet Velocity	0.6m/s
Water inlet velocity	0.8m/s
Relative humidity	35.8%

Applying CFD model it reduces temperature in summer weather from 90°F to 75°F as experimental setup.

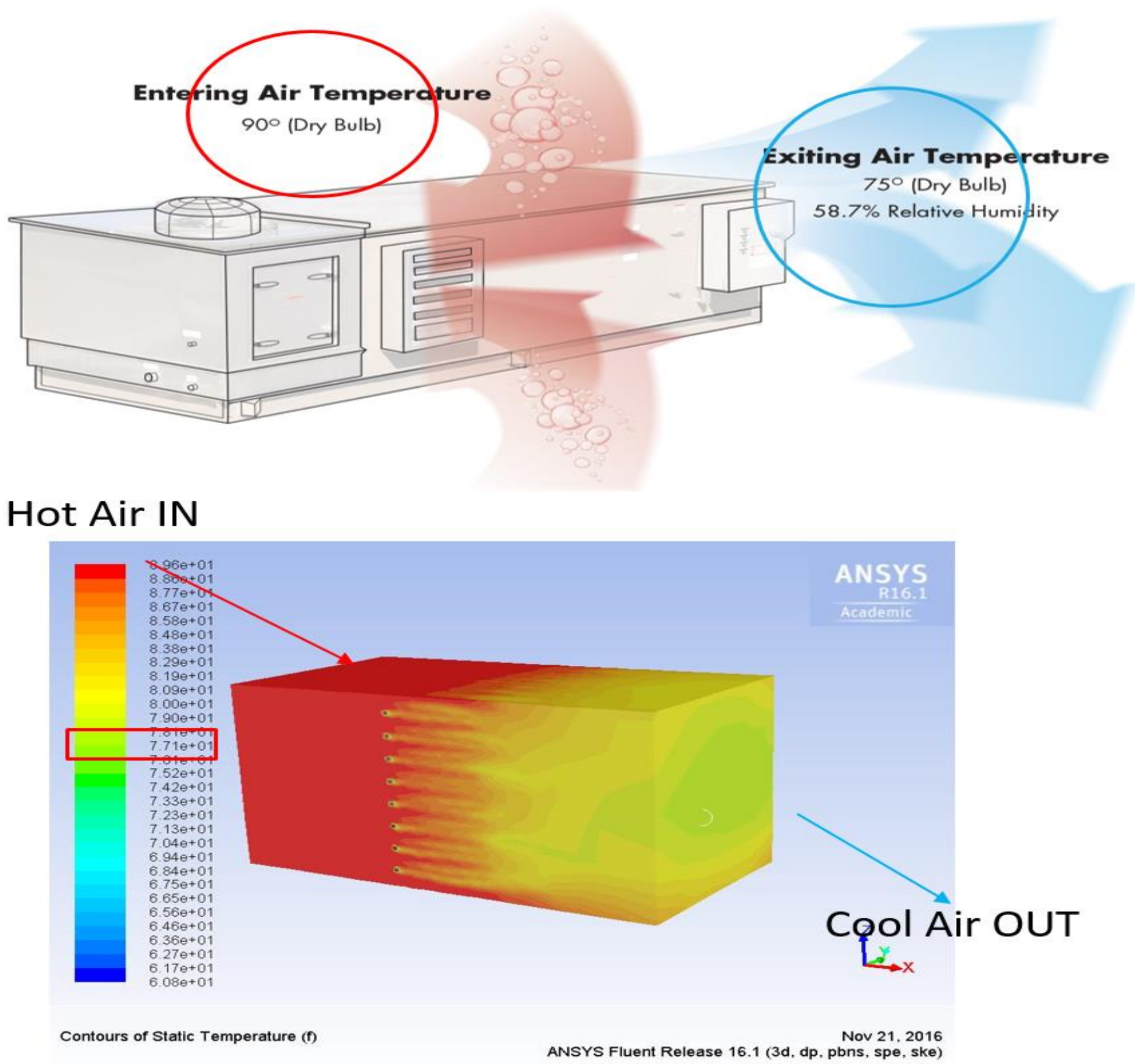

Figure 17 Validation with MESTEX modular unit [21]

As show in table 6, temperature difference between experimental and analysis is 2.66% which can be neglected as during process they use air filters which reduce the velocity. We can say that this research has been validated with experimental setup. We can use this flow condition for further study.

Table 6 Percent Error in validation

	Air Temperature outlet ℉	**Humidity (Rh%)**
CFD	77.10	63
Experimental	75	58.7

Error(%)	2.66	7.32

4.3 Grid Independence Study

Grid independence study has also been carried out for baseline case of Indirect cooling design and temperature of 310K and 0.6 m/s. 6 million cell is threshold value in this research work. Up to 6 million cells, junction temperature reduces significantly which is shown in figure 18. After this limit temperature change is within an agreement. After 6 million cells, it remains constant. For future study of this research work 6 million cells have been used for meshing.

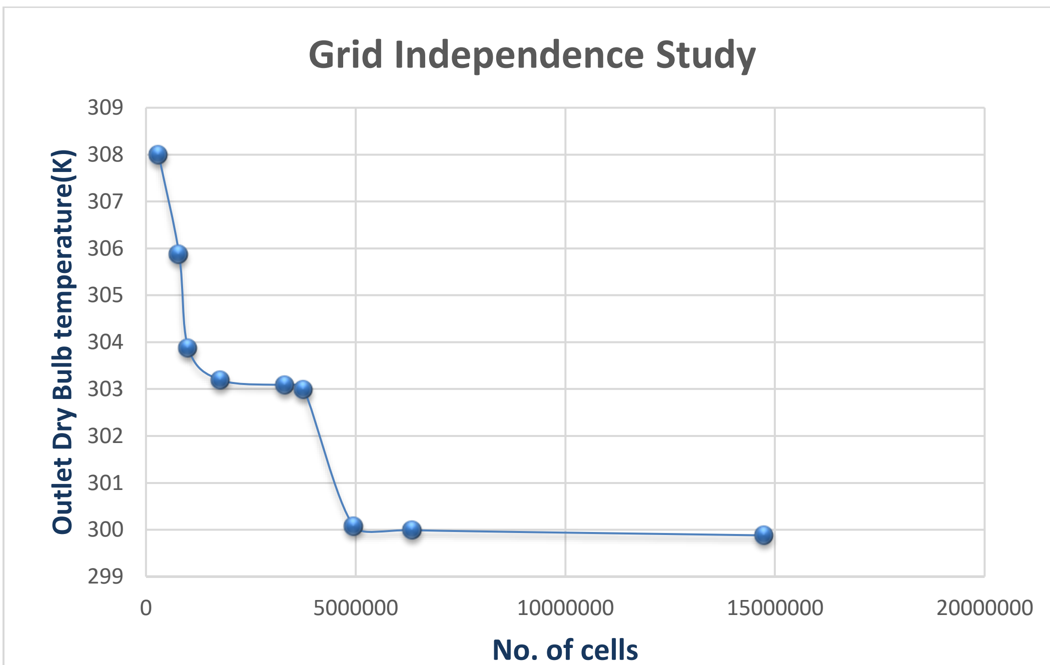

Figure 18 Grid independence study

CHAPTER 5 Parametric and optimization results

5.1 Parametric study

As shown in fig 19, air mass flow rate has been taken in range from 0.05 to 0.3m/s and water mass flow rate has been taken from 0.01 to 0.2 kg/s. It shows that as mass flow rate increasing efficiency decreases and so vice-a-versa. It helped to find out recommended zone by optimum range for air mass flow rate is 0.05-0.2 kg/s and for water mass flow rate is 0.04-0.17 kg/s. Parametric study by CFD reduced the cost of experiment and gives exact results in less timing.

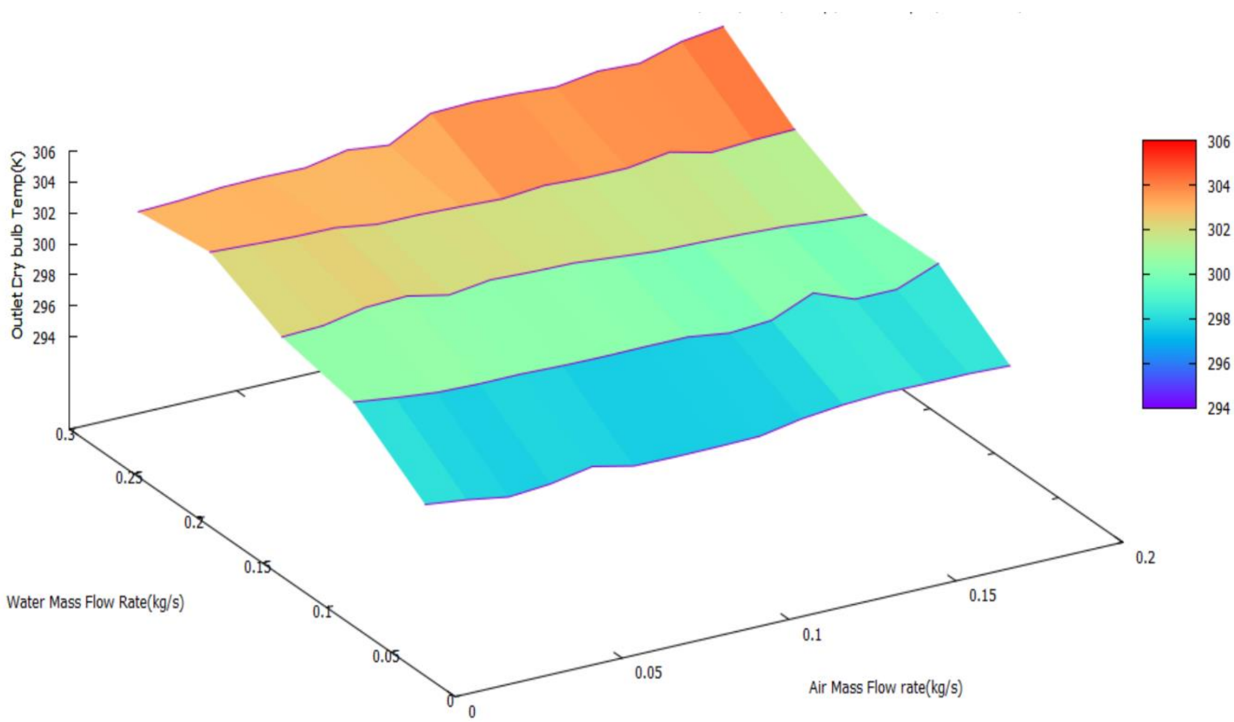

Figure 19 Air mass flow rate vs water mass flow rate

5.2 Saturations effectiveness curve

There are two ways to find out the efficiency of cooling system. 1. Saturation effectiveness 2. System resistance. Here, we have focused on saturation effectiveness. Saturation effectiveness is a mathematical representation of efficiency of the system which is shown in equation 6. It requires three quantities: inlet temperature and relative humidity and outlet temperature.

$$\varepsilon_e = \frac{t_1 - t_2}{t_1 - t'} 100\% \tag{6}$$

Where,

ε_e =saturation effectiveness,%

t_1= dry-bulb temperature of entering air, °F

t_2= dry-bulb temperature of leaving air, °F

t'= wet bulb temperature of entering air, °F

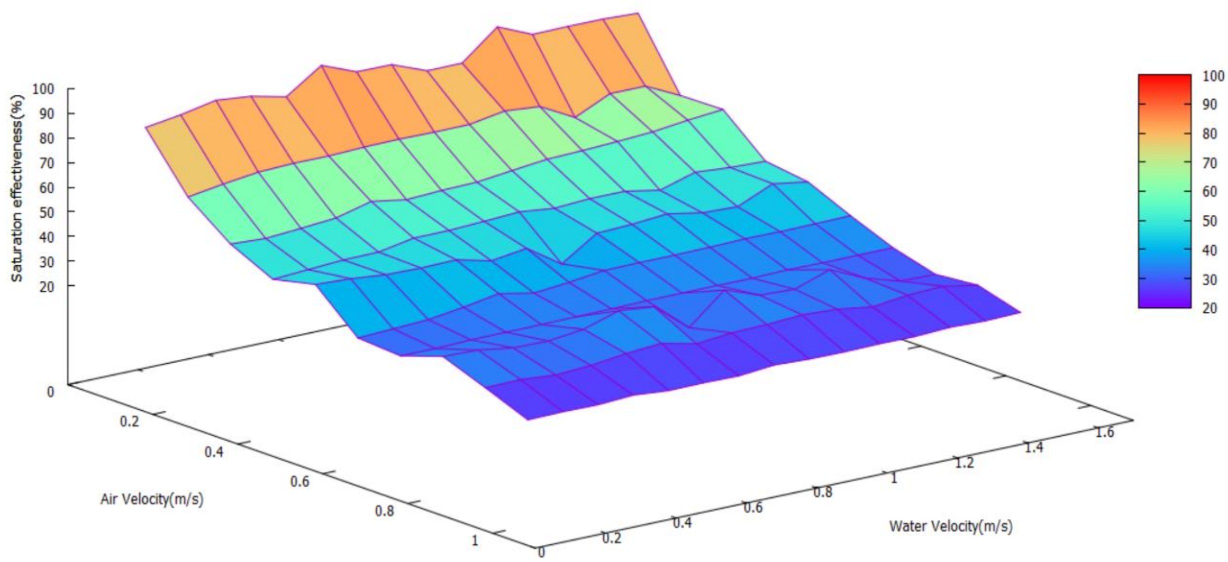

Figure 20 Saturation effectiveness curve for overall system

As shown in figure 20, at different air velocity and different water velocity outlet temperature changes. Saturation effectiveness curve helpful for different climate. It helps to find out how much data center requires cooling as per ASHRAE guidelines at different outlet temperature. It helps to save energy in winter timing when temperature is low outside, requirement of the air mass flow rate and water mass flow rate reduces as well.

Moreover, this curve can be helpful in compact modelling of DEC and IEC in Flowtherm, six-sigma room or Flovent software for easy setup as shown in figure [21] and figure [22] .

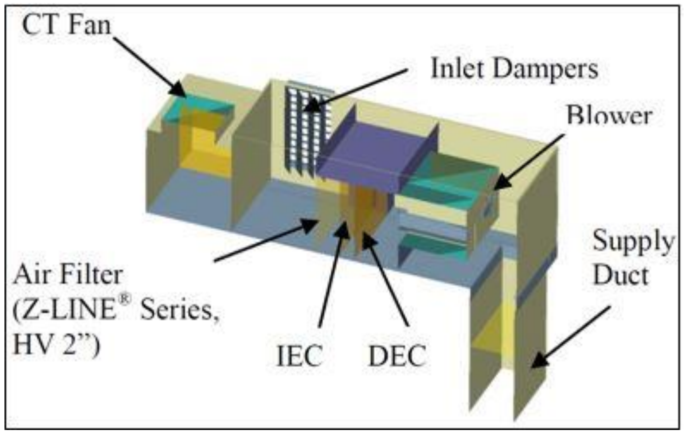

Figure 21 Compact modelling of IEC & DEC in Flow-therm [22]

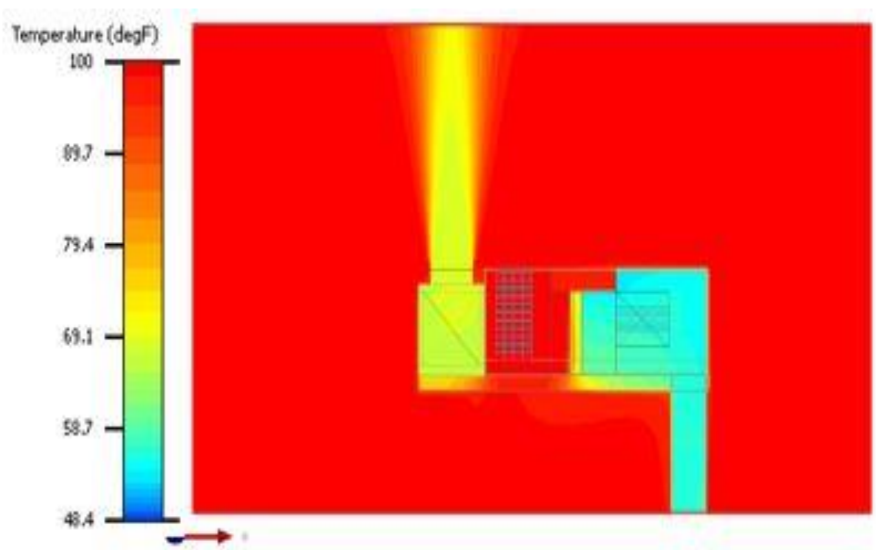

Figure 22 CFD analysis after applying 75% SE to unit [22]

5.3 Variable primary flow study

Variable primary flow is little be complicated for understanding and higher for efficiency. In chilled water cooling coils, there are just one inlet of water from cooling tower. It requires higher pumping power for water to reach at high water head and requires less power for bottom pipe which has lower head.

Variable primary flow is for to reduce pumping power and less water usage. In this research, there are two batches at lower velocity and optimum range velocity as shown in figure 22.

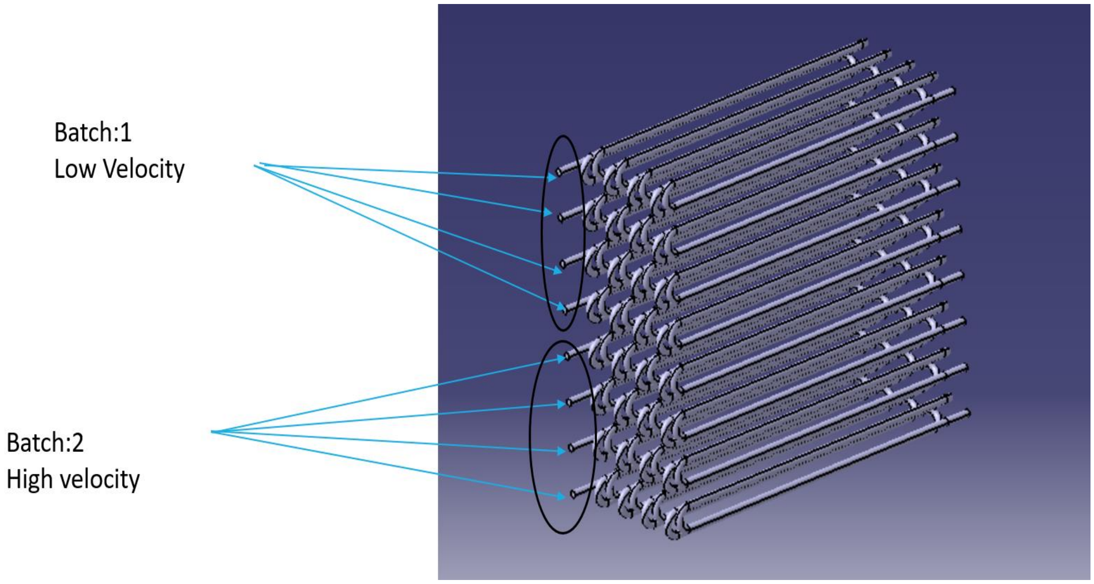

Figure 23 Variable primary flow setup in coils

As shown in Table 7, Batch 1 and Batch water velocity has been distributed. Applying this condition for analysis and optimization of water. Variable flow can be distributed by applying water pump to each coils to control.

Table 7 Velocity distribution

Batch 1 Water Velocity	**Batch 2 Water Velocity**
0.1	0.2-1.5
0.2	0.2-1.5
0.3	0.3-1.5
0.4	0.4-1.5
0.5	0.5-1.5

0.6	0.6-1.5
0.7	0.7-1.5
0.8	0.8-1.5
0.9	0.9-1.5
1	1-1.5

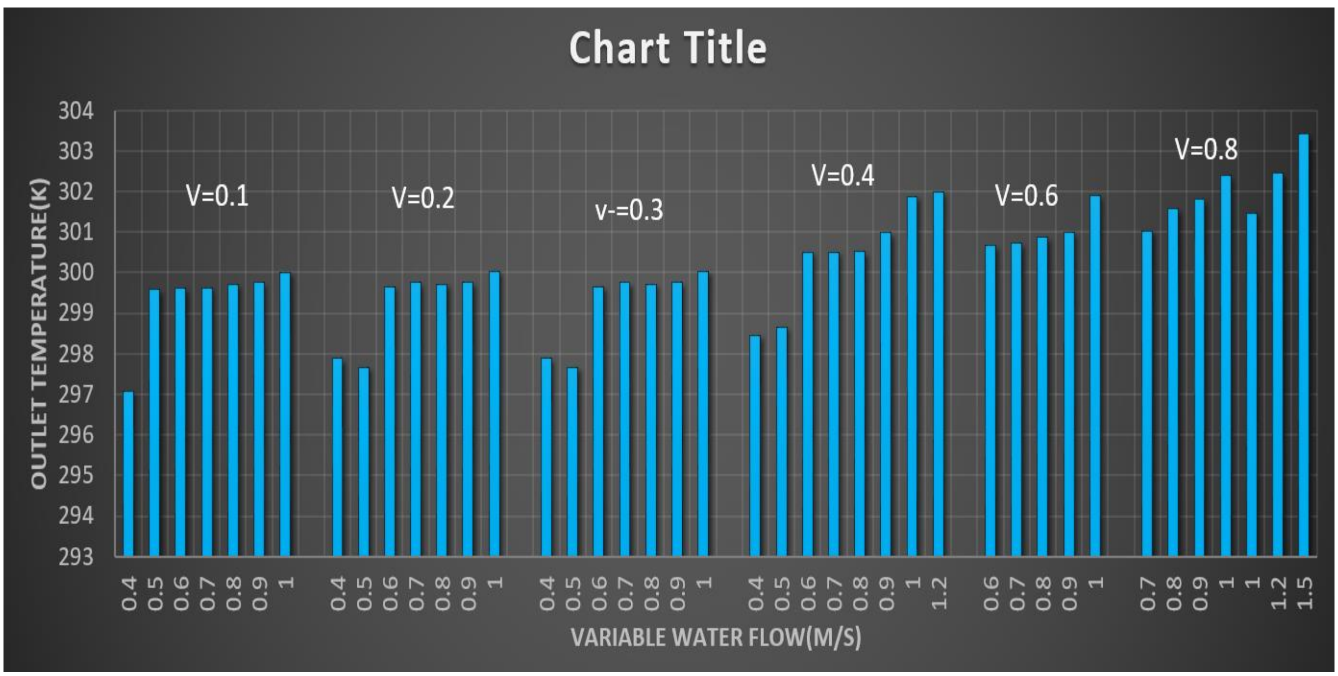

Figure 24 Variable water flow vs outlet temperature

As show in Figure 24, graph indicates outlet temperature by considering different temperature in coils. Indirect cooling coils give 75% maximum saturation effectiveness which means it can reduce temperature 10℃ at highest level. Inlet dry bulb temperature was taken in calculation was 310K. Graph shows that even after changing the flow rate in coils it does not change the efficiency of unit it maintains constant in optimum range of water velocity shown in Figure 20. At constant water velocity v=0.4m/s, outlet temperature of air was 298K where appling variable flow in coils at batch 1: v=0.1 and batch 2: v=0.4; temperature at outlet side is 297K. so we can say that reduction in pumping power will be 37.5%.

CHAPTER 6 CONCLUSION OF CHILLED WATER COOLING MEDIA AND FUTURE SCOPE

6.1 Conclusion

- ✓ Optimum thermal performance can be achieved at air mass flow rate from 0.05 to 0.2 kg/sec where current air mass flow rate is 0.5 kg/sec which saved energy almost 50%
- ✓ Optimum water mass flow rate obtained at 0.05 to 0.3 Kg/sec; it requires 1500 GPH pump instead of 2000 GPH
- ✓ IEC unit can be operated efficiently with variable primary flow with 37.5% low pumping power
- ✓ Shape optimization gives 13% of pumping power reduction which is operational efficiency and manufacturing cost is same because material volume is same

6.2 Future scope

- Indirect chilled water cooling is efficient system. During winter, dry air is passing through the system which can be harmful to the ITE; so, it is beneficial if we run system with DEC unit to maintain humidity.
- Cooling coil efficiency not only depends on water and air temperature, it also depends on number of coils, coil diameter, face area of coil, ratio to cooling length to diameter. It is important to find out most efficient design through parametric study in CFD for real-life implementation.
- Heat transfer performance varies with different cross section shape. Study of different cross section like oval, helical shape should be carried out through analytical and CFD
- Analytical study for chilled water cooling unit should be carried out and validated with experimental results from MESTEX unit
- Study should be carried out through steady state and transient level and compare for better results

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BIOGRAPHICAL INFORMATION

Mansi Vijay Prajapati was born in Ahmedabad, India in October 1992. She has pursued her Bachelor's degree (BE) in Aeronautical Engineering from Gujarat Technological University(GTU), India in 2010. She completed her Master of Science in Mechanical Engineering from the University of Texas at Arlington.

She is very enthusiastic and hard working towards her goals and dreams. She has this one goal to be an Astronaut, which she had decided when she was 10-year-old. Still she is dedicated to her dream. She has great interest in area of Thermal and Fluid science. She has conducted various project from undergraduate to till today to explore knowledge in different area. She had researched on "CFD study on Non-planar box wing- A conceptual design", which later on became journal paper in conference ITMAE-2014. She had studied on high lift configuration of aircraft through CFD and by doing these projects she made her knowledge stronger in CFD, fluid dynamics and structural analysis. Besides study, she was active member in management of events in national symposium held in collage.

Mansi joined EMNSPC team in Fall 2015. She has shown a great interest in Indirect evaporative cooling. She has been working on Indirect Evaporative cooling for data center application for Mestex Company. This work includes "Optimization of Water and Energy Use in Indirect Cooling Systems Using CFD" which is most important in today's world where we know we are facing water crisis. This work promises to open many optimization studies to help improve the unit and installation work. This research shows the 37.5% reduction in pumping power. She has presented this work in IMAPS-2016 Thermal management conference where she got great exposure. Her research interests includes thermal analysis, fluid design, thermal management and reliability.

She is active member of ASME and SWE. She is an extra ordinary person with kind heart and influential personality which motivates others as well for achieving their goals.