DEVELOPMENT OF A TEST METHOD FOR CHARACTERIZING AN INDIRECT EVAPORATIVE COOLING MODULE FOR DATA CENTER COOLING APPLICATION

By

ABHIJIT R. BHOSALE

Presented to the Faculty of the Graduate School of

The University of Texas at Arlington in Partial Fulfillment

Of the Requirements

For the Degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

THE UNIVERSITY OF TEXAS AT ARLINGTON

MAY 2021

UNIVERSITY OF
TEXAS
ARLINGTON

ACKNOWLEDGEMENTS

I want to express my sincere gratitude to Dr. Dereje Agonafer for his continuous support, motivation, and encouragement for over one year with my research. I am thankful that he involved me in an important project, equipped me with resources, and helped me shape my thesis with his review and guidance.

I want to thank Dr. Amaya Miguel and Dr. Haji Sheikh for taking the time to be on my thesis committee.

I am forever thankful to Mr. Ashwin Siddarth, Mr. James Hoverson, Mr. Tyler Freeman, and Mr. Vibin Simon for being my mentors and guiding me throughout my work with great ideas and their knowledge of fundamentals.

My special thanks to my dear wife, Mrs. Apurva Bhosale, my mother, Mrs. Sanjivanee Bhosale, my father, Mr. Rajendra Bhosale, and the rest of my precious family members for the unconditional love and encouragement that helped me achieve my goals. I wish to continue making them proud.

May 07, 2021

Abstract

DEVELOPMENT OF A TEST METHOD FOR CHARACTERIZING AN INDIRECT EVAPORATIVE COOLING MODULE FOR DATA CENTER COOLING APPLICATION

ABHIJIT R. BHOSALE, MS (Mechanical Engineering)

The University of Texas at Arlington, 2021

Supervising Professor: Dr. Dereje Agonafer

Evaporative cooling is a smarter way of maintaining optimum operating conditions in data centers. Indirect Evaporative Cooling (IEC) provides the advantage of cooling without changing the humidity of the Data Center (DC) air. We analyze and develop a method of testing to help build IEC modules at larger scales with improved characterization that is intended to help DC engineers achieve stringent Power Utilization Efficiency (PUE) targets by expending minimum power.

We study the design parameters pertinent to sizing an IEC module used in DC cooling and develop a test method while technically selecting the components involved. The module is planned with centrifugal plug fans on both primary and secondary sides selected to deliver 8000cfm of air at 2.5 in-wg of static pressure, an air-to-air crossflow

heat exchanger, a water distribution system to deliver and distribute water to the wetted channels of the secondary side, a sump for collecting water, and a recirculation piping. The primary side air is set to simulate a typical DC return air. We use 6Sigma software as a tool to size the cabinet, visualize air velocities across different parts such as inlets and outlets, and observe effects for further controlling the outlet air on the secondary side, that is connected to an auxiliary unit designed to supply air at desired temperature and humidity to simulate ambient climatic conditions. The heat exchanger considered in this study is a commercially available plate heat exchanger made of Al 8009 alloy and epoxy coated of size of 48" x 48" x 48". Face velocities up to 600 FPM are considered. Based on the literature surveyed, we recommend two types of spray headers, namely full cone spray nozzle and 360 deg rotating type water sprinkler in arrangements that effectively wet the heat exchanger cross section.

Cooling capacity and cooling effectiveness can be documented for several air speeds across the heat exchanger channels. Additionally, evaporation rate, water consumption, and pressure drop for the secondary side can be estimated. Thus, we associate the cooling effectiveness with parameters related to the fan-motor power and water utilization.

We build correlations between several distinct parameters for rating the performance of an IEC module. The results from this design and test method will be invaluable in developing phenomenological models of IEC modules which can subsequently be used for developing IEC units at scale to help size and optimize equipment for Data Centers.

LEFT INTENTIONALLY BLANK

Table of Contents

LIST OF TABLES

LIST OF ILLUSTRATIONS

Chapter 1

INTRODUCTION

Servers in a Data Center (DC) are arranged in racks in a way to allow flow of cold air and hot air in separate channels. Thus, heat is drawn from the hot air side and replaced with cold air on the other side. Additionally, individual servers house fans that enable flow of cold air over itself to effectively dissipate the heat generated in its operation. This hot air is typically cooled using methods such as Direct Expansion (DX) type units (e.g., CRAC units) or Chiller based air conditioning systems. I.e., compressor-based equipment involving refrigerants. In this paper, these are further referred as Traditional Air-conditioning systems, "TAS", and Data Center as "DC".

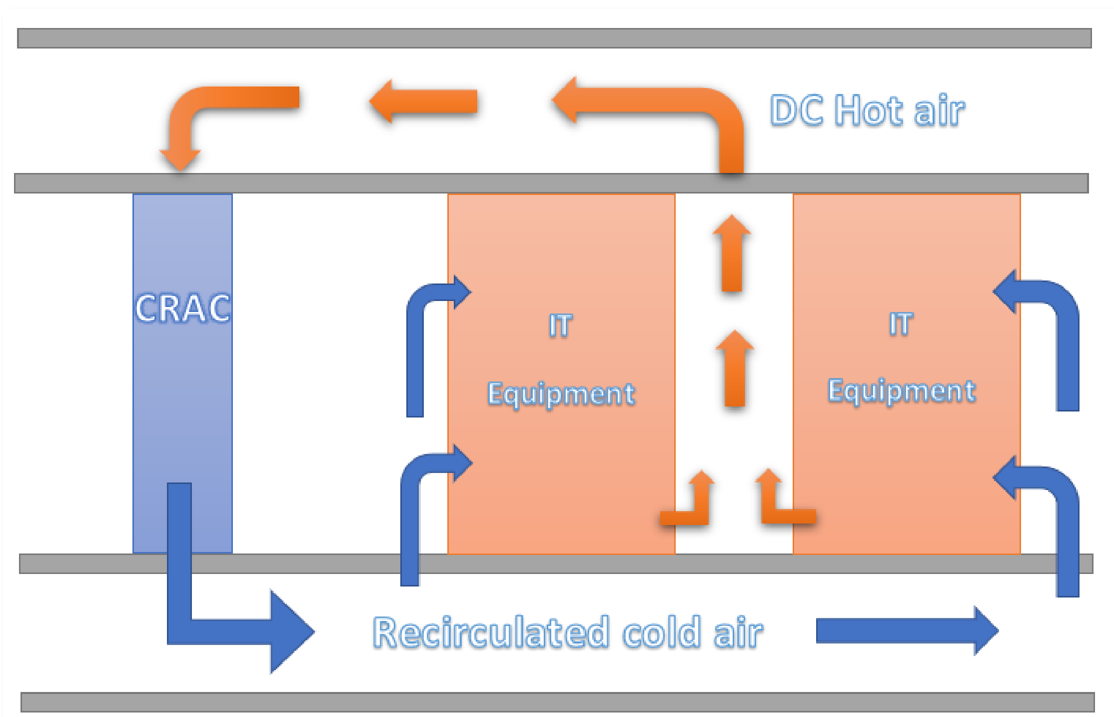


Figure 1: Air Cooled Data Center

1.1 **Evaporative Cooling in Data Centers:** DC engineers need to achieve strict Power Utilization Efficiency (PUE) targets. Apart from IT equipment, a major portion of the DC power is utilized for heat removal/ cooling. Traditional compressor-based air-conditioning systems (TAS) consume very high energy as compared to evaporative cooling. Therefore, in recent times, many companies are opting towards using evaporative cooling as the primary mode of operation. DC operating zone recommendation by the American Society of Heating Refrigerating and Air conditioning Engineers (ASHRAE) is considered as acceptable by the industry. Below is an illustration of DC operating zones, as recommended by ASHRAE in its TC9.9 "2011 Thermal Guidelines for Data Processing Environments" and a comparison with allowable operating conditions described by modern IT equipment suppliers, which clearly suggests a wider envelope for Evaporative Cooling [1, 25].

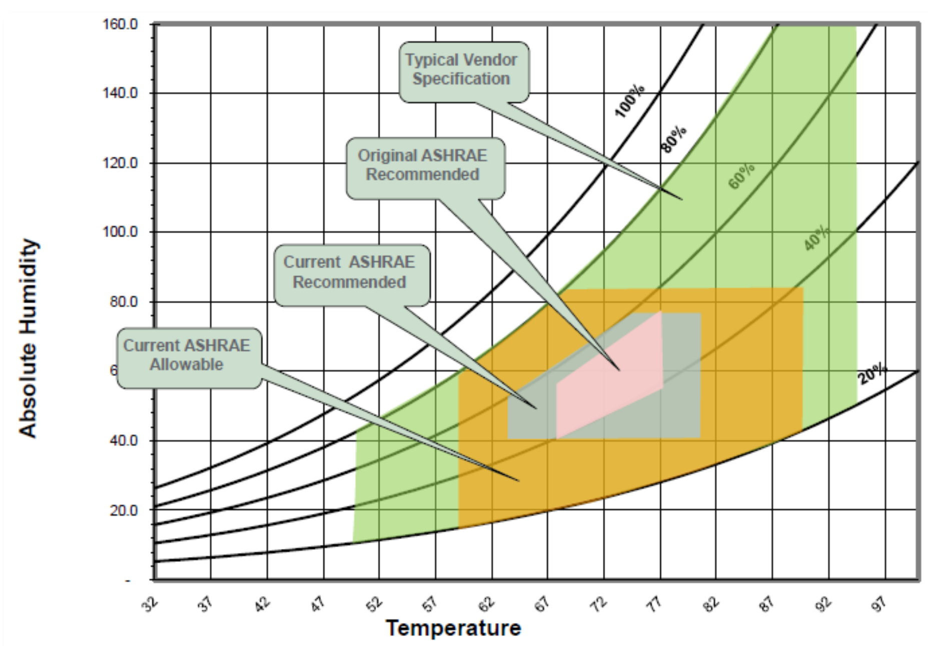


Figure 1-1: Recommended Data Center Operating Zone

As eminent from the above, evaporative cooling can be employed for larger number of hours resulting in greater power savings.

1.2 **Introduction to Indirect Evaporative Cooling (IEC):** Indirect evaporative cooling (IEC) is a system where a fluid is cooled by air on another side, without physically coming in contact, by the means of a heat exchanger. The fluid that brings in heat is considered primary, and the other side, secondary. Air on secondary side is obtained from the environment. It is further cooled by evaporation of water achieved by wetting the secondary side. This is achieved by spraying water in the secondary channel and forcing the movement of air in either con-current or countercurrent direction. Thus, the primary air is 100% sensibly cooled. Below is a typical crossflow plate heat exchanger that may be used in regular Air Conditioning applications.

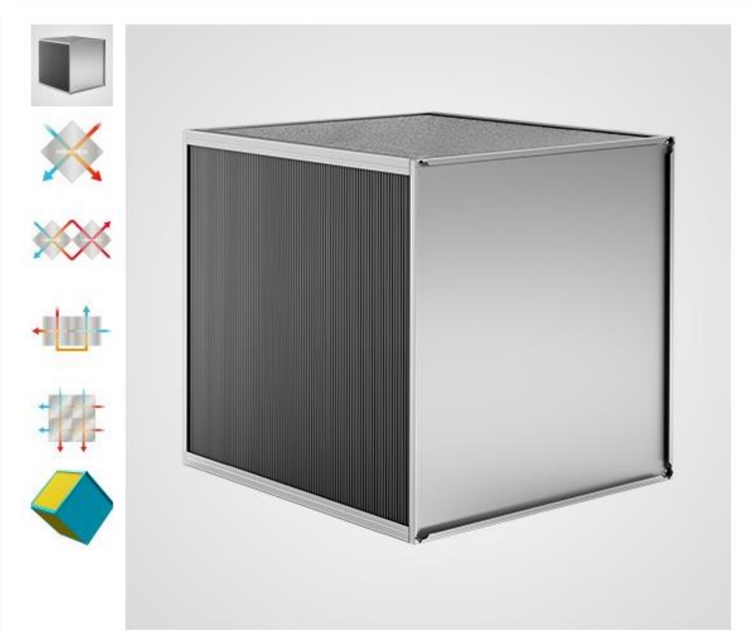


Figure 1-2-1: Crossflow Plate Heat Exchanger [2]

As explained above, on the primary side (can be also named as dry channel), air exchanges heat with the secondary side without mixing with each other. In the wet channel, secondary air is cooled by evaporation of the water film, which is formed by spraying water on the heat exchanger plates. In the wet channel, water film absorbs sensible heat of the air and converts it into the latent heat in the course of evaporation. As air carries more water vapor, it comes closer to its saturation point. In the process, the temperature of the water film and secondary air are decreased. Primary air flows in the alternative dry channels which is cooled by conduction of the separating plate between dry and wet channels [3].

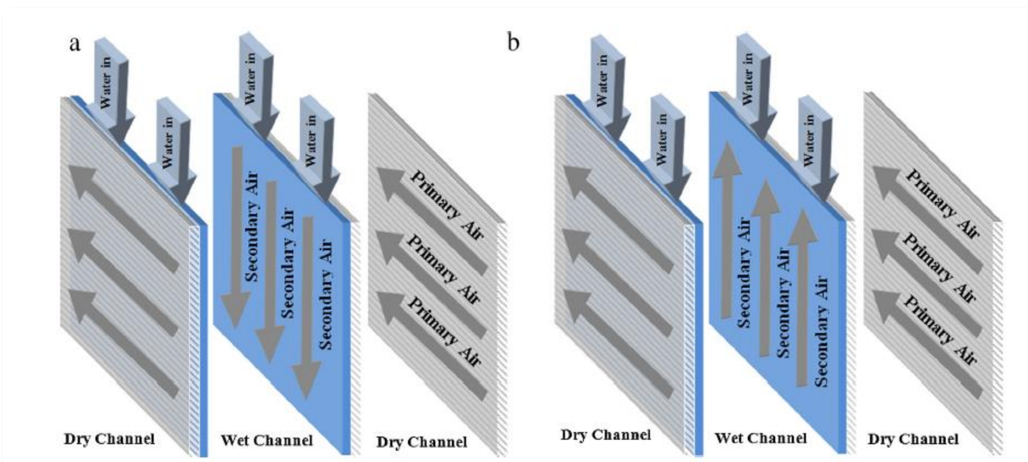


Figure 1-2-2: Dry and Wet Channels

1.3 **Indirect Evaporative Cooling (IEC) for Data Centers:** A smart DC cooling system is designed with both TAS and IEC. While IEC acts as a basic mode of operation, TAS can be triggered when temperature or humidity approaches the outer bounds of set operating limits. Thus, based on the IT equipment heat load and ambient conditions, DC cooling can be operated in the below 3 modes: Air-to-Air IEC in dry mode, Air-to-Air IEC in wet mode, and Direct Expansion (DX) or Chilled water cooling (TAS).

The ASHRAE journal March 2011 describes DC operation with IEC. In this design, IEC module is designed to recirculate the DC air. Hot air from DC is forced through the primary side of the DC. Once cooled, it is pushed back to the DC servers through air channels, usually installed under raised floors. In this way, there is no contamination from outside environment or humidity fluctuation. The journal also describes a makeup air humidification/ dehumidification unit, designed at minimum 0.25 air changes per hour (ACH). Additionally, a filtration unit, is described with 6-10 ACH to filter the mixture of fresh air and hot DC air as it is supplied to the IEC unit [4].

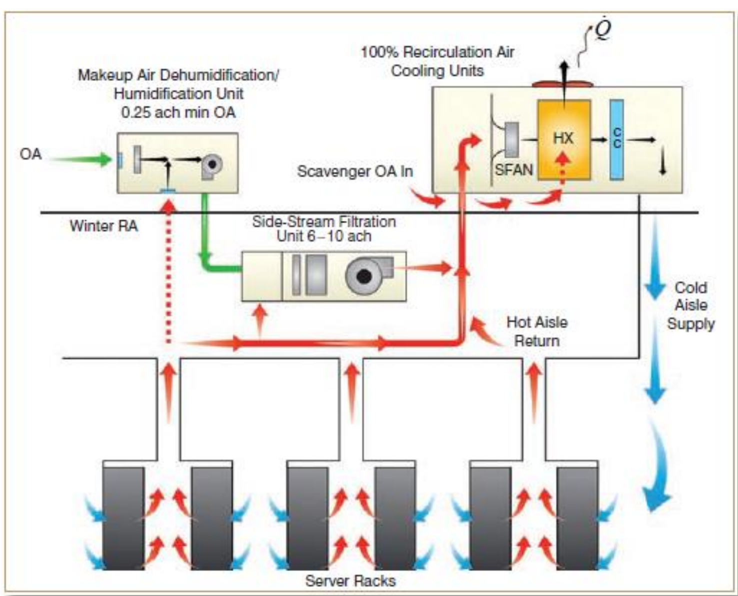


Figure 1-3: IEC System Design

Chapter 2

MOTIVATION AND STUDY OBJECTIVE

This study originated from project 21 of ES2 (Center for Energy-Smart Electronic Systems) and UTA's Electronic, MEMS and Nanoelectronics Systems Packaging Center (EMNSPC), which is aimed towards "Characterization and Compact Modeling of Wet Air-to-Air heat exchangers". The goal of this project is to provide best practices for using Indirect evaporative cooling technologies, develop deeper insight, and provide guidance in implementing, operating, controlling and maintaining cooling systems integrated with air-side economization for data center cooling. This study lays the foundation for this project by defining a smart and extensive test method which will be invaluable in developing phenomenological models of IEC modules which can subsequently be used for developing IEC units at scale to help size and optimize equipment for Data Centers.

The principal objective of this study is to develop a method of testing, to help build IEC modules at larger scales and Improve characterization to help DC operators achieve stringent PUE targets. This study covers design parameters pertinent to sizing an IEC unit with technical selection of the components involved. We use information from existing research to develop optimum spray arrangement with spray header selection. The components selected in this study can create +40Deg F temperature on both Primary and secondary sides. Thus, with this test method, we can create a wide range of hot dry and hot humid conditions from ambient climatic conditions of about 70 deg F, which is a standard condition in the enclosure of a factory or warehouse building.

The proposed method makes efficient use of saturated exhaust air on the secondary side for humidification of ambient air to adjust humidity and simulate a set climatic condition. A unique water collection grid is designed to accommodate at-least 10 minutes of experimental runtime, to record results in stable conditions. Finally, the proposed method allows estimation and characterization of cooling effectiveness, cooling capacity, water evaporation, water distribution, power usage, and total pressure drop.

Chapter 3

LITERATURE REVIEW

3.1 **Ashrae-143 2015 standard:** This standard suggests a method of testing and rating the performance of packaged and semi packaged indirect evaporative cooling units. We use this standard as a starting point and look to develop a test method that can be applied at a bigger scale [5].

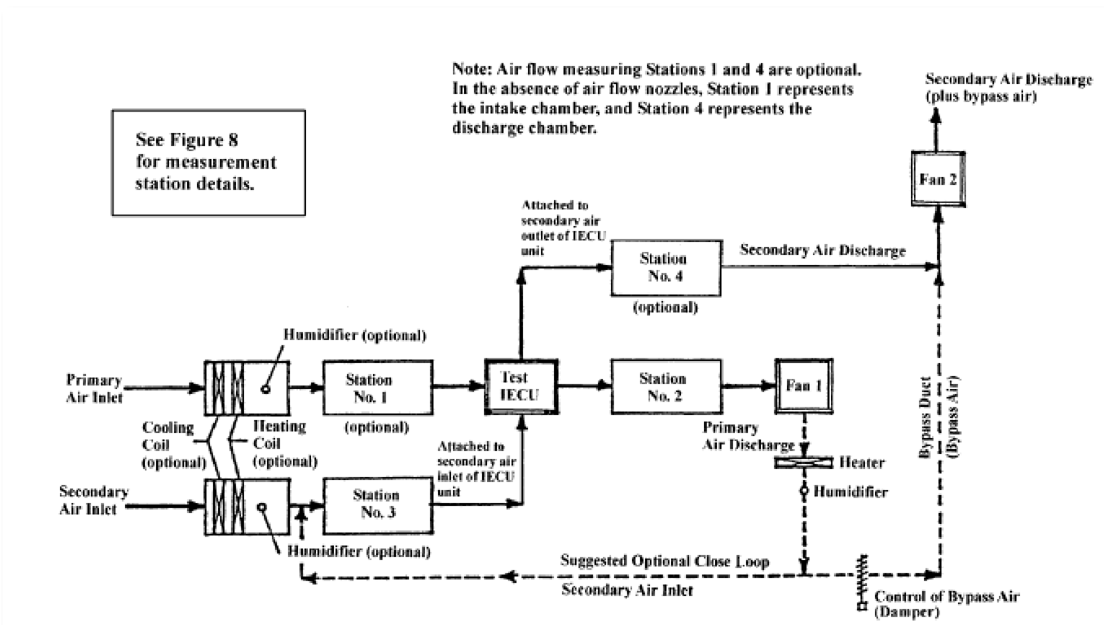


Figure 3-1: ASHRAE-143-2015 Test Method Schematic

3.2 **J.D. Palmer's Evaporative Cooling Design Manual:** This manual was funded by the United Stated Department of Energy, New Mexico Energy Minerals and Natural Resources Department, and Energy conservation and management department for

New Mexico schools and commercial buildings. It highlights a very crucial characterization of cooling effectiveness against air speeds for various Primary to Secondary air ratios and pressure drops. This forms a crucial base for our study of efficient, low-cost fan selection on the secondary side [6].

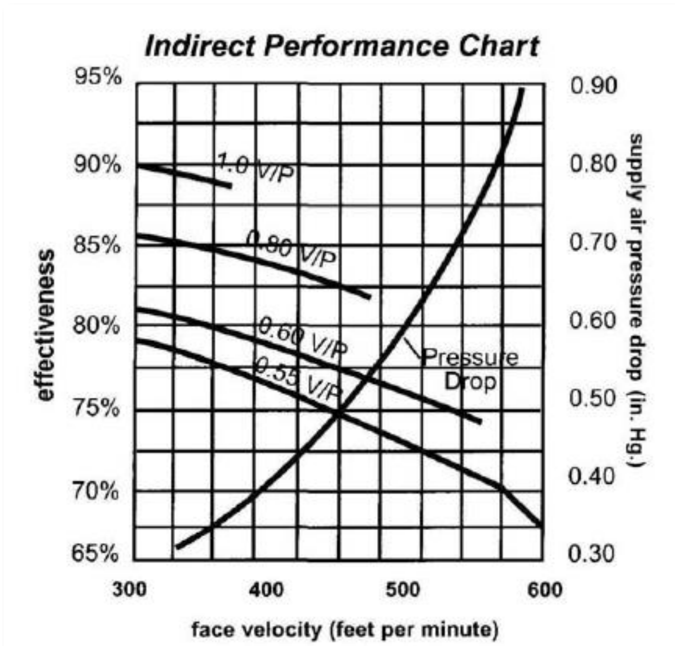


Figure 3-2: Cooling Effectiveness Vs. Heat Exchanger Face Velocities

3.3 **Spray Setup**: Reputed American nozzle manufacturer Bete's engineering information provides fundamental understanding on spray types, coverage, spray angle, height of installation, flow rate, and pressure requirements of various nozzle types, and help identify the most suitable options for an IEC module [7].

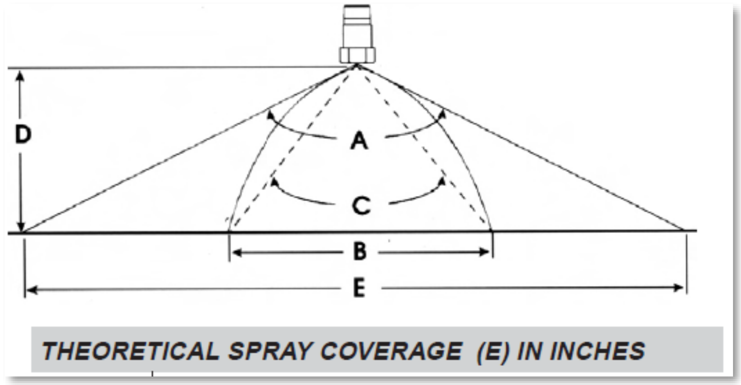


Figure 3-3: Theoretical and Actual Spray Coverage

3.4 **Experimental Study:** Tiezhu Sun's experiments on various nozzle types to identify water distribution provide further clarity into picking an ideal spray header and its optimum arrangement to maximize wetting of the Heat Exchanger plates. Yubio Sun's numeric and experimental work on Natural Draft cooling towers provides an analysis of different spray arrangements [8][9].

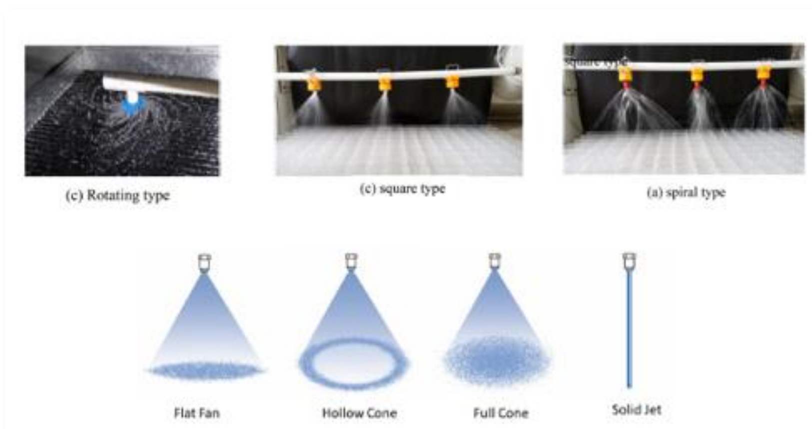


Figure 3-4: Various Spray Patterns

Chapter 4

IEC MODULE SIZING USING 6SIGMA SOFTWARE

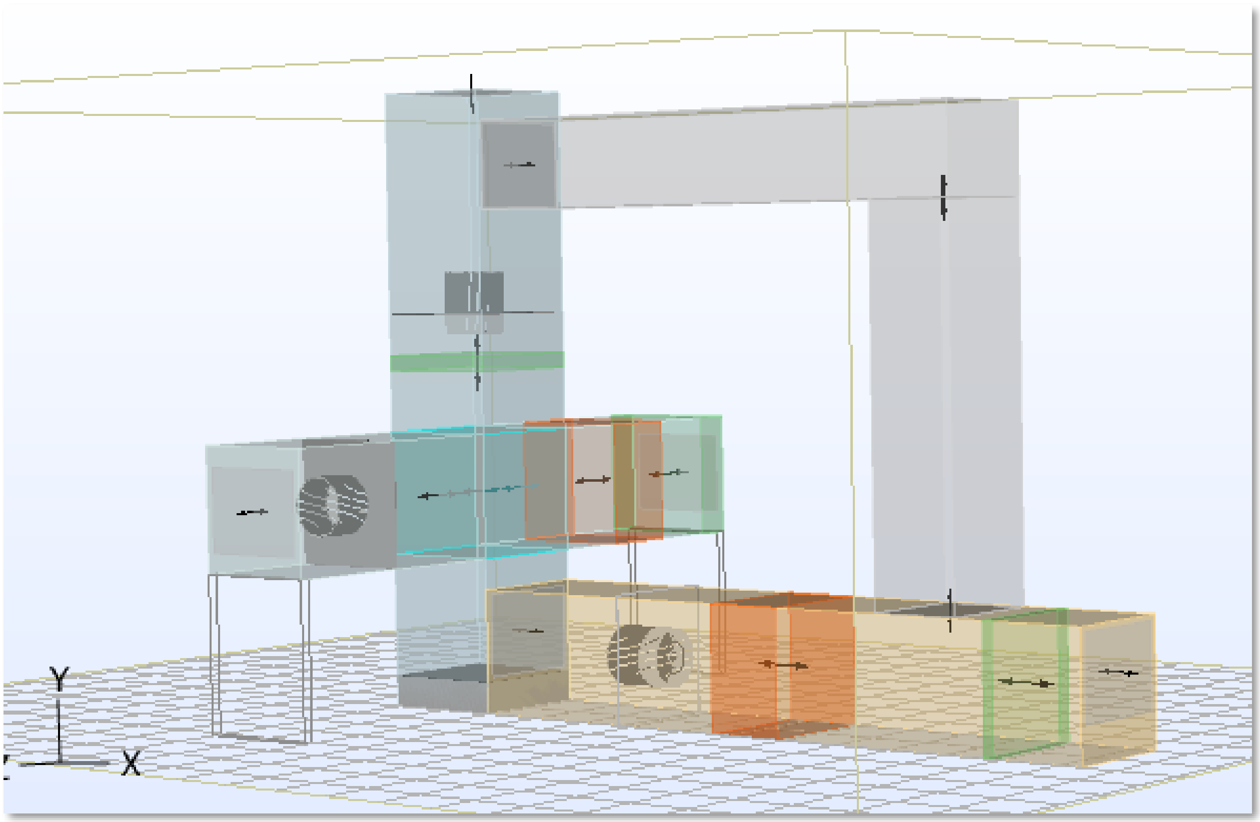


Figure 4-A: IEC and Test Model on 6Sigma software [10]

6Sigma is software developed by Future Facilities to size Data Center cooling equipment. The software lets us conduct CFD analysis of the designed equipment. We use this software to size the cabinets of our air handling units and visualize flow through their inlets, outlets, and recirculating ducts. In this way, we can estimate the resistances, face, and channel velocities for the discussed test method. As 6Sigma does not have Plate Heat Exchanger in its model tree, we do not conduct thermal analysis of the setup.

4.1 Primary Side:

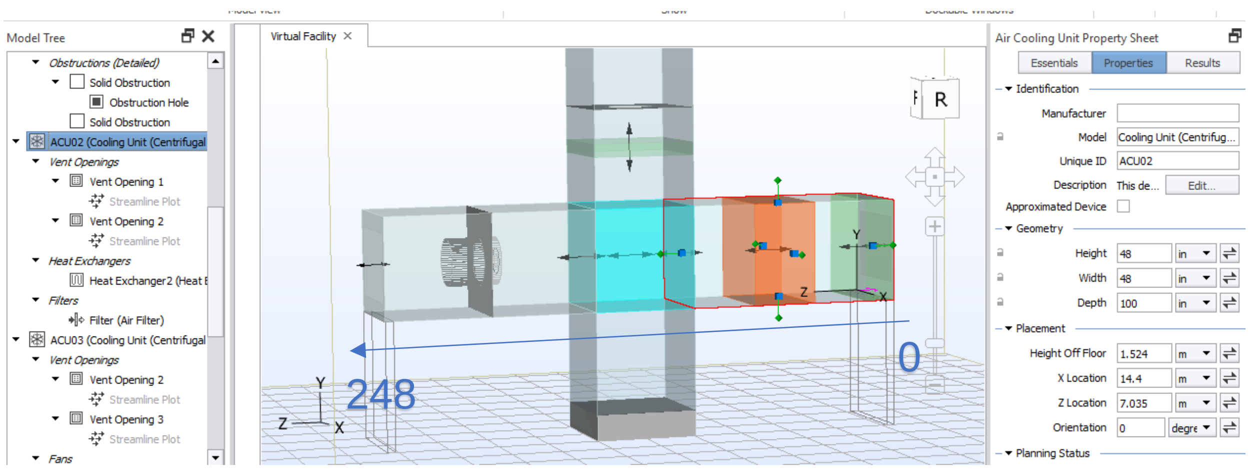


Figure 4-1: Primary Model on 6Sigma software [10]

Position	Component	Dimension (in)	Max. operating Pd (in-wg)	Max. operating Pd (Pa)
0	Vent	42 x 32	0.05	12.5
0-8	Airflow & Temperature measuring station	42 x 32 x 8	0.1	25
12-14	Prefilter	(12 x 12 x 2) x 4	0.4	100
40-70	Coil Heater	48 x 48 x 30	0.5	125
90-92	Position for Temperature sensors		0.05	12.5
100-148	Air-to-Air Plate Heat Exchanger	48 x 48 x 48	1.5	375
175-185	Reserved for VCD to control Airflow (optional)	48 x 48 x 10		0
190-192	Position for Temperature sensors		0.05	12.5
197-219	Fan + Fan wall	48 x 48 x 22		0
248	Vent	42 x 32	0.05	12.5
	Total Pressure Drop		2.7	675

Table 4-1: Components of IEC on Primary Side

4.2 Secondary Side:

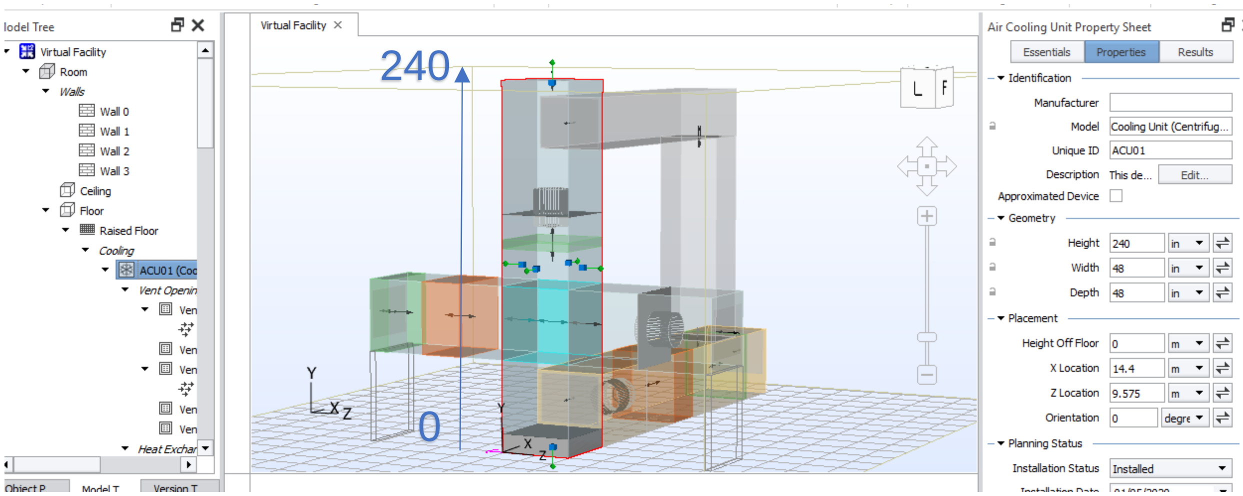


Figure 4-2: Secondary Model on 6Sigma software [10]

Position	Component	Dimension (in)	Max. operating Pd (in-wg)	Max. operating Pd (Pa)
0-12	Water Collection Grid	48 x 48 x 12		
14-46	Vent (Right Side @ suction)	42 x 32	0.05	12.5
60-108	Air-to-Air Plate Heat Exchanger (wet mode)	48 x 48 x 48	2	500
116-124	Water Spray Setup	48 x 48 x 3	0.25	62.5
132-138	Mist/ Drift Eliminator	48 x 48 x 4	0.8	200
138-140	Position for Temperature sensors		0.05	12.5
154-176	Fan + Fan wall	48 x 48 x 22		0
195-227	Motorized Volume Control Damper (VCD) - Right side	42 x 32	0.05	12.5
240	Motorized Volume Control Damper (VCD) - Top discharge	42 x 32	0.05	12.5
	Total Pressure Drop		3.25	812.5

Table 4-2: Components of IEC on Secondary Side

4.3 Auxiliary AHU:

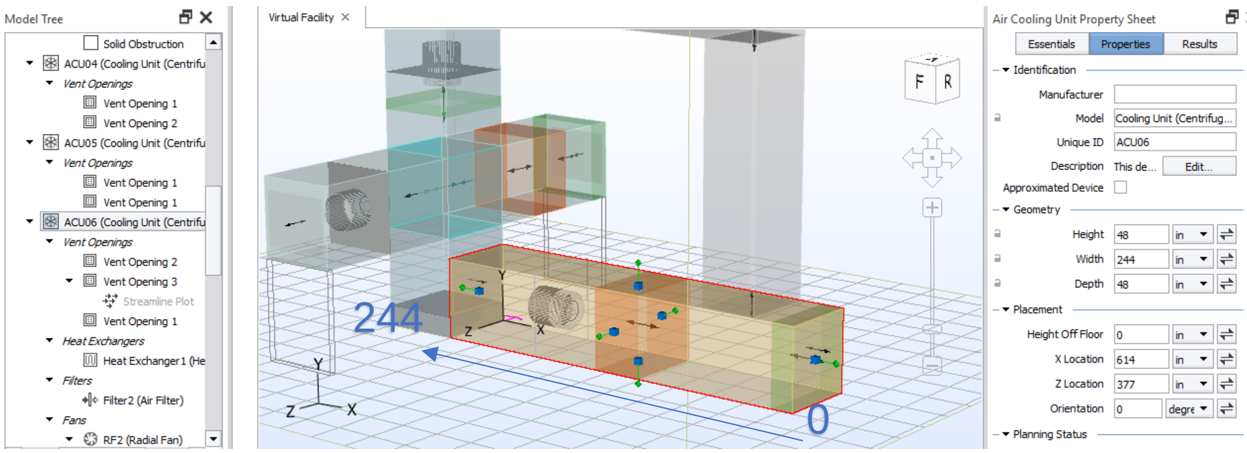


Figure 4-3: Auxiliary AHU Model on 6Sigma software [10]

Position	Component	Dimension (in)	Max. operating Pd (in-wg)	Max. operating Pd (Pa)
0	Motorized Volume Control Damper (VCD)	42 x 32	0.05	12.5
0-8	Airflow & Temperature measuring station	42 x 32 x 8	0.1	25
12-14	Prefilter	(12 x 12 x 2) x 4	0.4	100
28-60	Return air - Motorized Volume Control Damper (VCD) - Top	42 x 32	0.05	12.5
14-104	Air Mixing Chamber	48 x 48 x 90		
104-134	Coil Heater	48 x 48 x 30	0.5	125
178-200	Fan + Fan wall	48 x 48 x 22		0
220-230	Reserved for VCD to control Airflow (optional)	48 x 48 x 10		0
235-237	Position for Temperature sensors		0.05	12.5
244	Vent	42 x 32	0.05	12.5
	Total Pressure Drop		1.2	300

Table 4-3: Components of IEC in Auxiliary AHU

Chapter 5

TEST SETUP DESIGN

The Test Setup is designed by carefully assessing and technically selecting components suitable for airflow of 8000 – 9000 cfm with a total static pressure not exceeding 3 in-wg on each side and the auxiliary unit. Special attention has been given to the face velocities to be maintained below 600fpm as per ASHRAE recommendations and pressure loss of equipment at 1000 fpm for an extreme case.

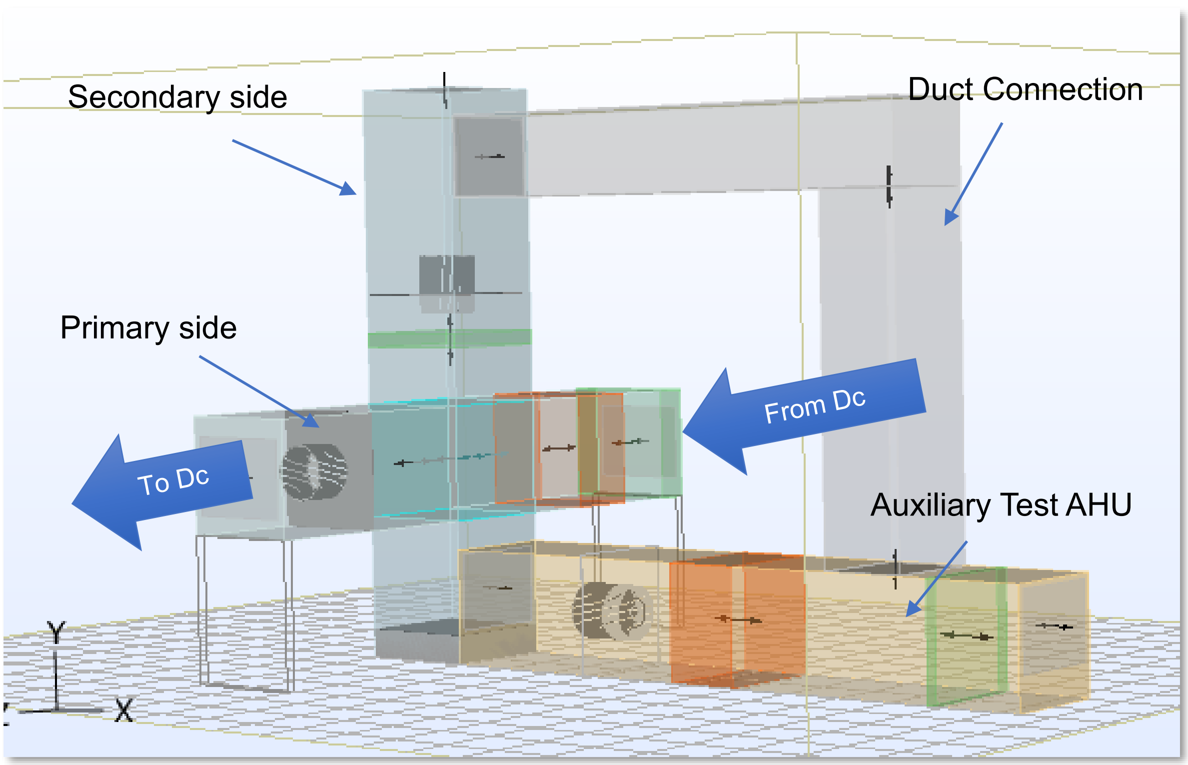


Figure 5-A: IEC and Test Model on 6Sigma software [10]

5.1 **Air-to-Air Plate Heat Exchanger:**

Our main interest is to build a module than can be used in multiple scales. Since DC Air Handling Unit (AHU) capacity can range from 20,000 to 100,000 cfm, we design a module for capacity 8,000 – 9,000 cfm. We begin by selecting a commercially available Plate Heat Exchanger (PHX) of an optimum size, capable of handling 8,000 - 9,000cfm of airflow. We select a reputed American manufacturer Heatex Inc.'s cross flow plate heat exchangers 48" x 48" x 48" with 5 and 10mm gaps between plates. Attached is a sample product selection report from manufacturer's selection software. This report provides an initial basis for more advanced calculations. It is to be noted that manufacturer's data cannot be entirely used as it does not truly consider the effect of different spray header arrangements. However, for developing a test method on a scalable IEC module, this selection software offers essential information of pressure drop in standard conditions. Pressure drop is likely to increase by the influence of water spray.

Air flow	CFM	8000	8000
Pressure drop (*)	inH_2O	0.809 (0.765)	0.911 (0.878)
Efficiency	- Wet	67	N/A
	- Dry	67	N/A
Effectiveness	- Wet	67	-
	- Dry	67	67
State before °F	Temperature	110°F	80°F
	Temp.wet bulb	86.6°F	68.1°F
	Relative humidity	40%	55%
	Absolute humidity	156.5 gr/lb*	84.16 gr/lb*
State after °F	Temperature	90°F	98.9°F
	Temp.wet bulb	82.4°F	73.4°F
	Relative humidity	73%	30.3%
	Absolute humidity	156.5 gr/lb*	84.15 gr/lb*
Air velocity (face/channel)	ft/min	514.27 / 1260.79	514.27 / 1260.79
Transferred power	BTU/h		161144.6

Figure 5-1-A: Heatex Software Selection at 8,000cfm

Air flow	CFM	10000	10000
Pressure drop (*)	inH_2O	1.343 (1.278)	1.536 (1.501)
Efficiency	- Wet	67	N/A
	- Dry	67	N/A
Effectiveness	- Wet	67	-
	- Dry	67	67
State before °F	Temperature	110°F	70°F
	Temp.wet bulb	86.6°F	55.7°F
	Relative humidity	40%	40%
	Absolute humidity	156.5 gr/lb*	43.42 gr/lb*
State after °F	Temperature	83.3°F	94.8°F
	Temp.wet bulb	80.9°F	64.5°F
	Relative humidity	90.4%	17.9%
	Absolute humidity	156.5 gr/lb*	43.42 gr/lb*
Air velocity (face/channel)	ft/min	642.84 / 1575.99	642.84 / 1575.99
Transferred power	BTU/h		266720.9

Figure 5-1-B: Heatex Software Selection at 10,000cfm

It is to be noted that as the airflow increases from 8,000 cfm to 10,000 cfm, the pressure drop on both primary and secondary sides increases significantly, also raising the face velocities beyond ASHRAE's recommendation of 500 fpm. Hence, we design a test setup at 8000cfm with the described plate heat exchanger [11].

5.2 **Primary, Secondary, and Auxiliary Fan Selection:**

Fans are primarily classified based on their impeller type as Axial or Centrifugal. Axial fans are high-volume low-pressure fans. For a given amount of airflow and static pressure, centrifugal fans perform at higher efficiencies. However, Axial fans comparatively occupy lesser space and are a good choice in low static pressure requirements. A commercially available Centrifugal Plug fan with backward curved centrifugal impeller is selected. Reputed manufacturer Ziehl-Abegg's fan is considered. This fan is selected to deliver 8000 cfm of airflow @ 2.6 in-wg at 1670 rpm. It comprises of a 6Kw, 3phase, IP 55 multi-speed electronically commutated (EC) motor. At maximum speed of 1860 rpm, the fan can deliver 8000cfm @ 4 in-wg or 9000cfm @ 3 in-wg static pressure [12]. Fan/ cabinet spacing guidelines are considered from a reputed manufacturer's (KRUGER Ventilation [23]) data, which follows Air movement and control association's (AMCA) standard 211. Manufacturers can choose to design with axial fans to reduce the overall cabinet size and cost. This however depends on IT equipment heat load and extreme ambient conditions of the location.

Figure 5-2-1: Centrifugal Plug Fan

fan data

SFP-class | SFP-value (P_{SFP})	- | Ws/m³	**3** | 1149
airflow volume (q_V)	ft³/min	9534.7
air velocity	ft/s	49.20
pressure, **stat. (p_{sF})** | tot. (p_F)	in.wg.	**2.500** | 3.037
electrical power input (P_{sys})	W	5169
system eff., **stat. ($\eta_{sF,sys}$)** | tot. ($\eta_{F,sys}$)	%	**54.2** | 65.9
fan speed **(n)** | max. (n_{max})	rpm	**1864** | 1860
fan speed, set value ($\%n_{max}$)	%	100
frequency **(f_{BP})** | (f_{max})	Hz	**60** | 60
voltage (U_{DP})	V	460
current (I_{DP})	A	6.79
acoustics, suction side **($L_{w(A),5}$)** | ($L_{w,5}$)	dB	**87** | 93
acoustics, pressure side **($L_{w(A),6}$)** | ($L_{w,6}$)	dB	**94** | 99
dimensions (w x h x d)	in	26.38 x 26.38 x 21.14
product weight (m_{pr})	lb	154.3
k-factor nozzle pres. (k)	-	308
differential pres. nozzle ($p_{sF\ nozzle}$)	Pa	2766

PF:PF_50; Ano:116180; STol:+-10 %

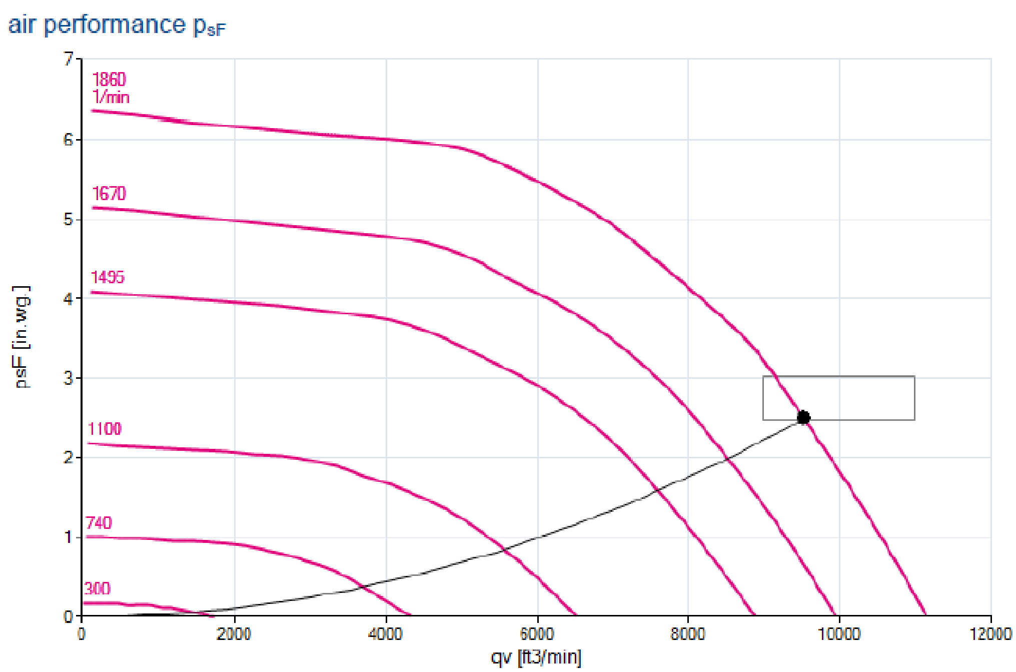


Figure 5-2-2: Ziehl Abegg fan selection [12]

5.3 **Airflow and Temperature Measurement:**

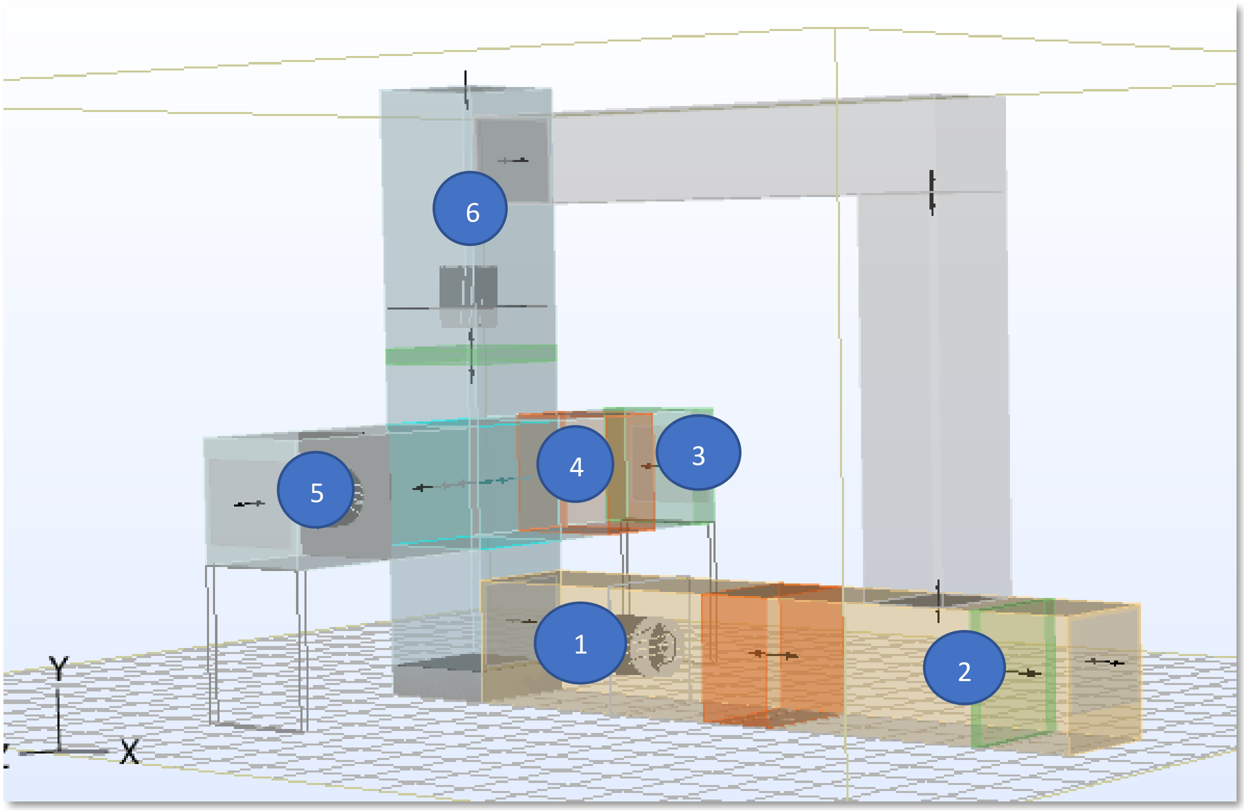


Figure 5-3-1: IEC and Test Model on 6Sigma software [10]

Airflow and temperature measurement stations are planned at various points for proper inspection of set conditions and accuracy of various calculations pertaining to performance rating. On the position 2 and 3, Airflow and Temperature measurement is through in-duct measuring stations. We select a commercially available product from a reputed manufacturer. On positions 1, 4, 5, and 6 Airflow measurement is through fan's inbuilt module (except 4) and Temperature measurement through thermocouple sensors.

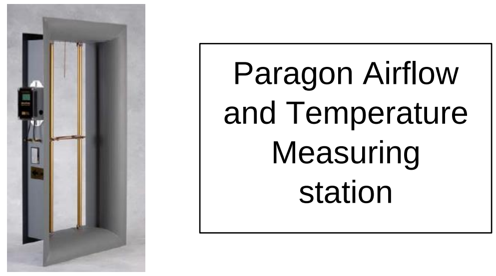
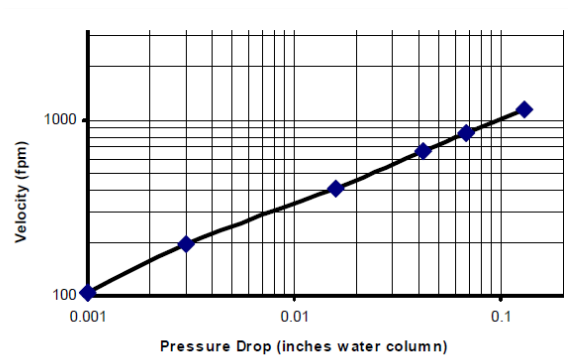


Figure 5-3-2: Paragon Measuring Station

A commercially available airflow station is considered from manufacturer Paragon controls. Accuracy of this station is reported within ±0.5% of actual flow through the velocity range of 200 to 1,200 fpm when installed in accordance with published recommendations and within ±5% at a velocity of 100 fpm. Our test method is designed for face velocities of 200 to 500 fpm. The measuring station selected is tested in accordance with ANSI/ AMCA 610-06/ 611-06 standards. Its operating temperature range is - 32 to 122°F (0 to 50ºC) [13]. Temperature sensor and transmitter's operating range - 30 to 130 F. Pressure drop reported by the manufacturer is 0.13 in-wg @ 1148fpm. Additionally, thermocouple sensors that are used to measure temperature and humidity on position 1, 4, 5, and 6 can be selected with accuracy of +-1%.



Pressure Drop (inches w.c.)	Velocity (fpm)
0.001	104.5
0.003	196.3
0.016	404
0.042	661.9
0.068	836.8
0.13	1148.6

Figure 5-3-3: Pressure Drop of Paragon Station [13]

5.4 **Water Spray Setup:**

Water spray arrangement can be done from top to bottom or bottom to top flow. In case of top to bottom, the secondary air acting is countercurrent. Therefore, it creates diffusion in the water spray and forces water droplets to join the water thin film on heat exchanger plates. Our main interest is to maximize the wetting of heat exchanger plates with minimum use of water by maximizing the residence time of water. In the top to bottom approach, based on spray header manufacturer's data, the spray manifolds shall be aligned vertically above the heat exchanger at a height of 8 to 16 inches.

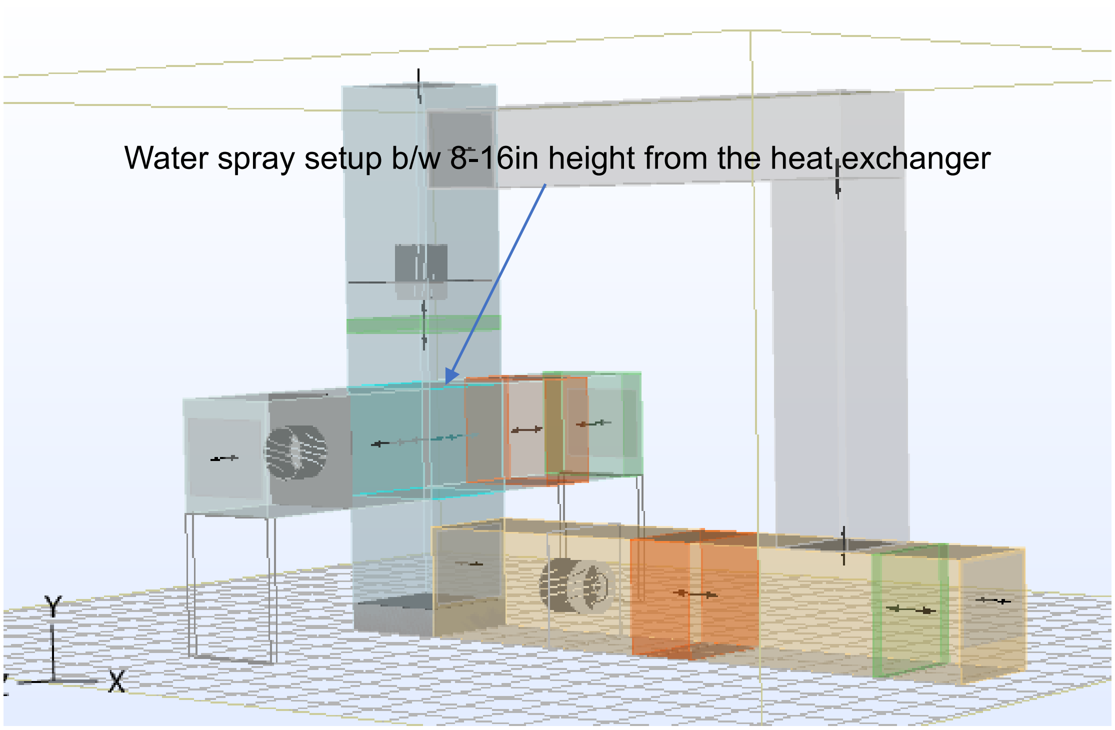


Figure 5-4-1: IEC and Test Model on 6Sigma software [10]

Based on previous literature review and manufacturer's data, we analyze several types of spray headers that are commercially available. For the purpose of evaporative cooling, usually an atomizing type nozzle is the most preferred option. However, this shall not be a good choice for Indirect evaporative cooling as it creates very small droplets that can be easily drawn through by the pressure of the secondary fan. Smaller/ non-uniform coverage will cause inconsistent plate wetting. Therefore, we need to select a spray header that can cover a large area uniformly with considerably larger droplets. Several other factors such as spray angle, height of installation, coverage, flowrate, and pressure affect the overall performance of the secondary side of IEC module [7][8].

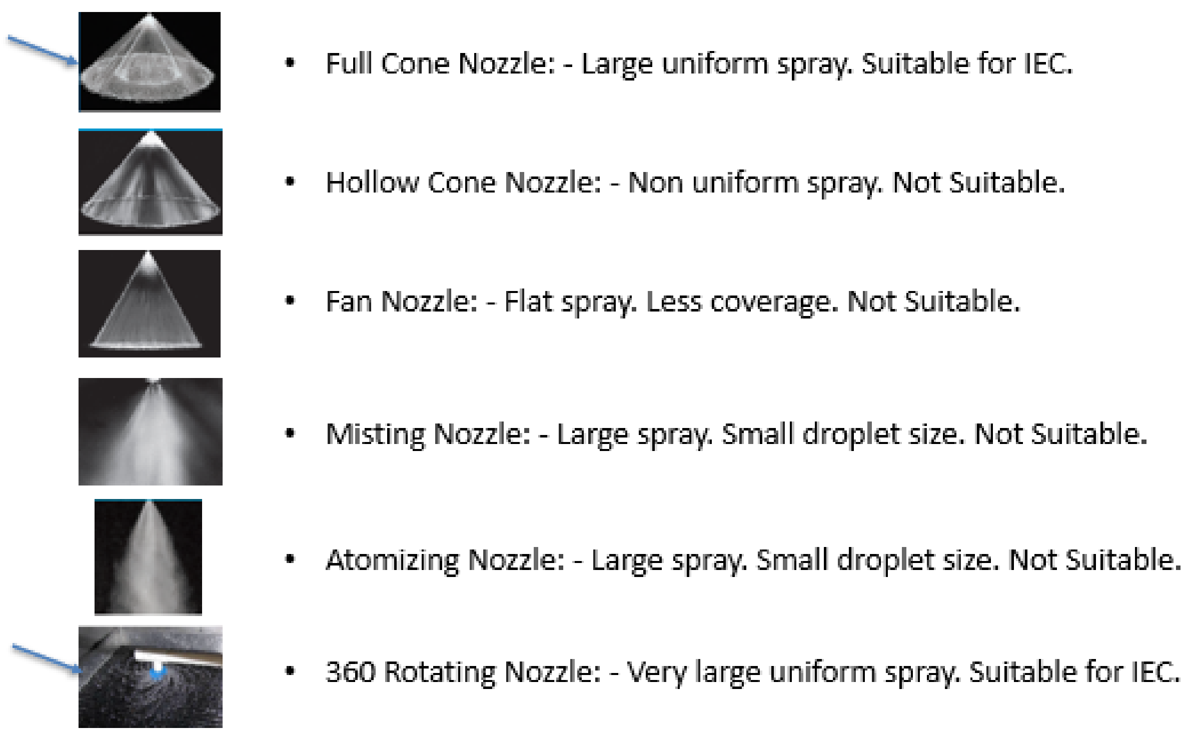


Figure 5-4-2: Spray Patterns

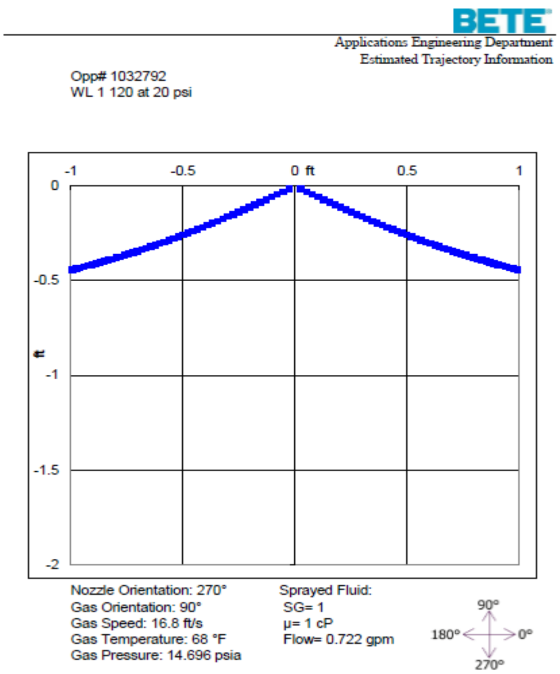
Therefore, for our 48 x 48 x 48 in Heat exchanger, we finalize 1) Full cone type nozzle, and 2) 360-degree rotating header.

Type	Model	Flow (GPM)	Press. (PSI)	Installation Ht. (in)	Coverage Dia. (Ft)	No. of headers
Full Cone	WL 120	1.08	20	8-10	2	6 (2 manifolds)
360 Rotating	3/32'' inverted	1.1	20	12	15	1 (1 manifold)

Table 5-4-1: Spray Header Selection

5.4.1 **Full Cone Nozzle**

- 2 Manifolds with 3 nozzles each.
- Flow and pressure measurement through gauges attached on top of each nozzle



Manufacturer's selection software suggests coverage diameter of 2+ Ft for the selected nozzle at 1000 fpm air speed across the manifold.

Figure 5-4-3: Bete Nozzle Software Selection [7]

We use an online tool to calculate the manifold length and diameter at the selected flow rate and pressure. Each manifold shall hold 3 nozzles, equally spaced from each other.

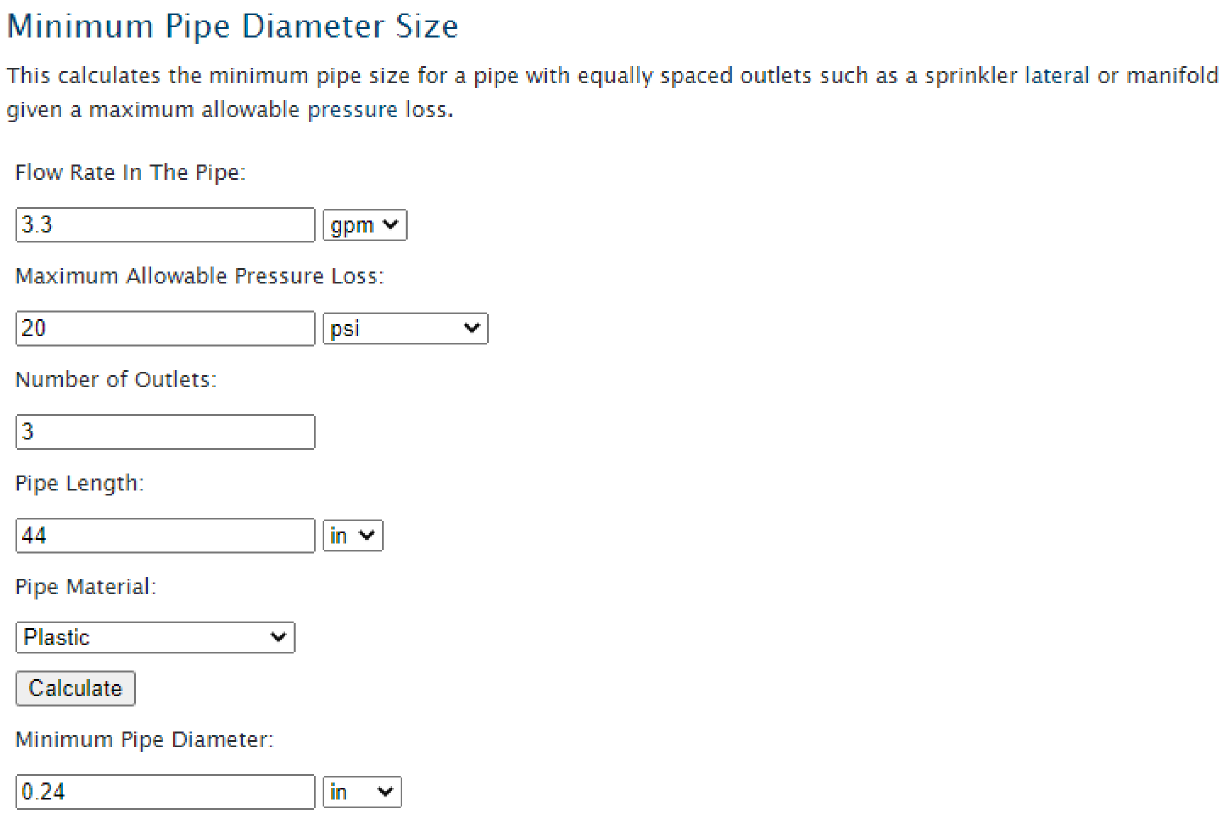


Figure 5-4-4: Manifold Sizing [14, 24]

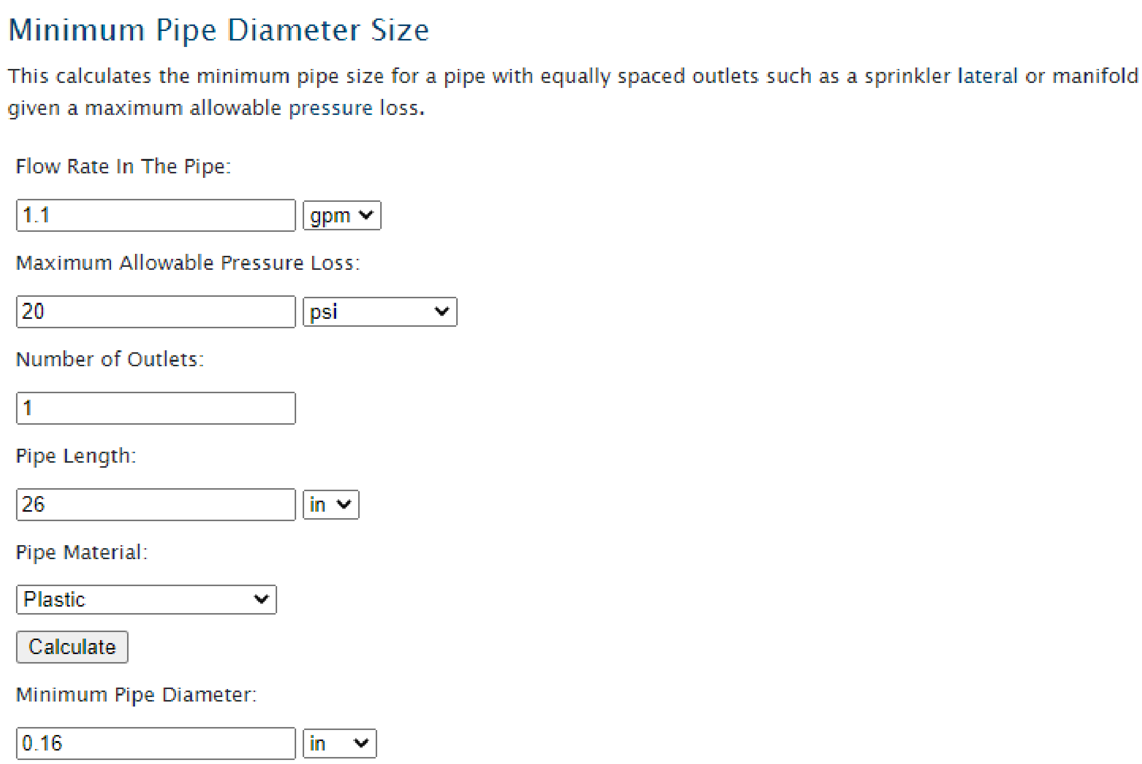
Source: http://irrigation.wsu.edu/Content/Calculators/General/Pressure-Loss-With-Outlets.php

5.4.2 **360 Degree Rotating Spray Header**

	PSI	
	20	25
#5 Nozzle - Beige (5/64")		
Flow (GPM)	0.75	0.84
Diameter at 3.0 ft. ht.	30.0	31.0
Diameter at 6.0 ft. ht.	32.0	32.5
#6 Nozzle - Gold (3/32")		
Flow (GPM)	1.10	1.25
Diameter at 3.0 ft. ht.	31.0	31.4
Diameter at 6.0 ft. ht.	34.0	34.5

Figure 5-4-5: 360 Degree Rotating Spray Header Selection [14]

We use the same online tool to size the manifold. In this case, we have only one spray header installed.



Source: http://irrigation.wsu.edu/Content/Calculators/General/Pressure-Loss-With-Outlets.php

Figure 5-4-6: Manifold Sizing [14, 24]

5.5 Electric Coil Heaters:

Electric Heaters are selected on 1) Primary side of the IEC module to simulate hot DC return air; 2) Auxiliary AHU to simulate hot (and humid) ambient condition for the secondary side. Both heaters are designed to sensibly heat air from 70F to 110F in 3 coil-steps with total installed power of 102 Kw [15]. This power is controlled to achieve the desired heating.

Duct Heaters			Dimensions (in)						CFM	FPM	Electrical Data					Stages			Drawings	Thermostat
			Heater			Control Box														
Item	Qty	Model (See Legend)	W	H	D	A	B	C			P (kW)	VAC-Ph	I (A)	Ctrl V	Signal	Qty	kW	A		
1	Tag(s): EDH-1, EDH-2																			
	2	DF CI00HB	50.00	48.00	6.00	9.00	49.50	10.00	8000	500	102.0	460-3	128.0	24	Mod 0-10V	3X	34.00	42.67	M203420009-A E203420009-A	None
	Elec. Opts.: FC-CA-SF-AC-MC-TR-TF-CF-PDN-HECB-STC8-TXM-SCR-CGC Mech. Opts.: N1-BCC-CC																			

Figure 5-5-1: Neptronic Heater Selection [15]

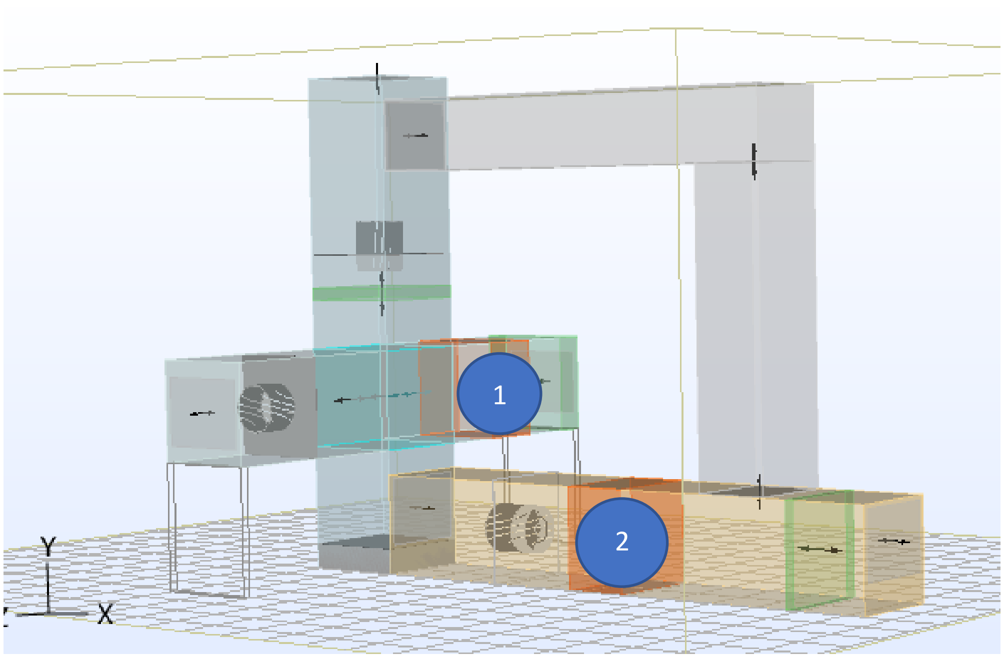


Figure 5-5-2: IEC and Test Model on 6Sigma software [10]

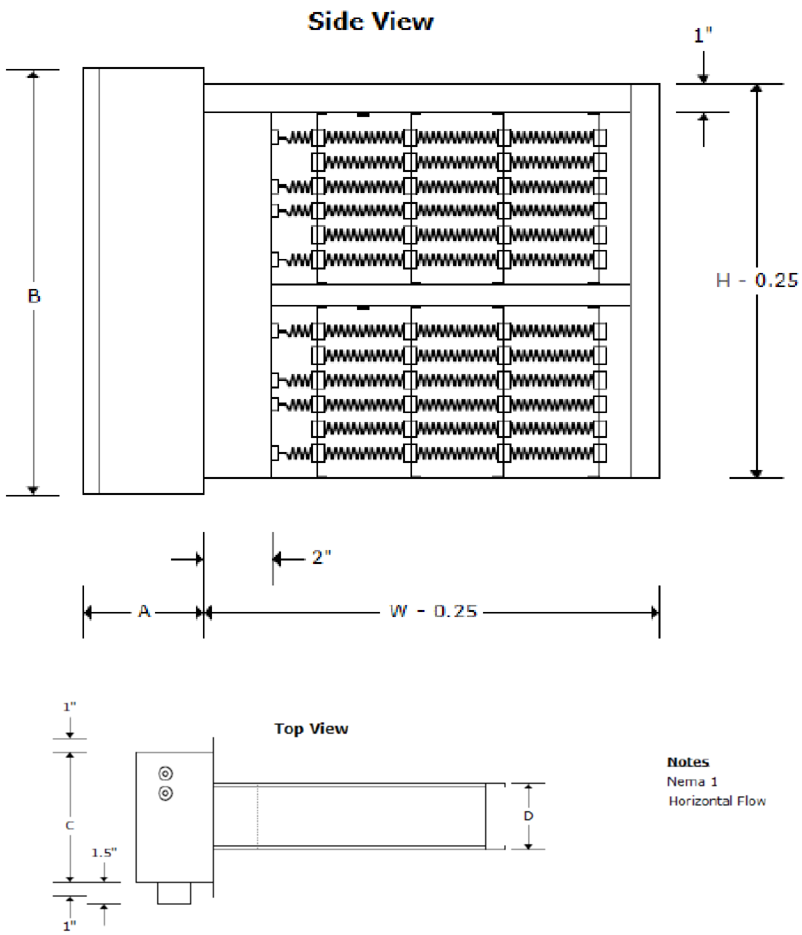


Figure 5-5-3: In duct Heater General Drawing [15]

Heat required (KW) = (Airflow (cfm) x dT (F))/3160 [Source: Marley Engineered products]

5.6 Prefilters:

Standard MERV (Minimum Efficiency Reporting Value) 8 air filter selected, assembled from 12 x 12 x 2 in panels [16]. Filters are installed at the suction of primary side and auxiliary AHU to protect against dust particles entering with the airstream.

SB, ZL, HV and MR-11 STANDARD SIZE FILTERS

SIZE W x H x D NOMINAL	SIZE W x H x D EXACT	RATED VELOC-ITY FPM	INITIAL RESIST. IN. W.G.				MEDIA SQUARE FEET				SIZE W x H x D NOMINAL MM	RATED VELOC-ITY M/H	INITIAL RESIST. PASCALS			
			SB	ZL	HV	MR11	SB	ZL	HV	MR11			SB	ZL	HV	MR11
12 x 12 x 2	11-1/2 x 11-1/2 x 1-3/4	500	.34	.30	.27	.31	2.31	2.89	4.33	4.33	305 x 305 x 51	9150	84.5	74.6	67.1	77.1

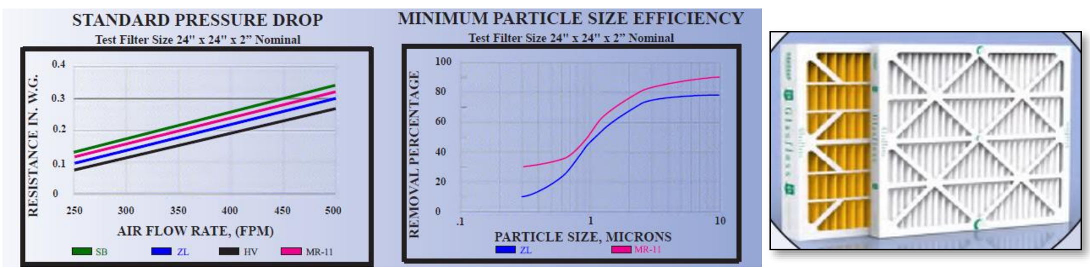


Figure 5-6-1: Air Prefilter [16]

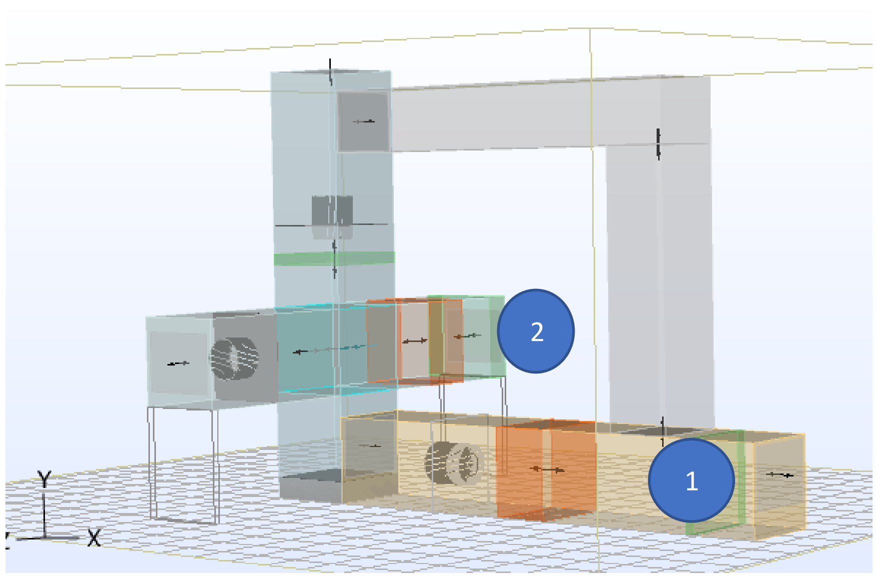


Figure 5-6-2: IEC and Test Model on 6Sigma software [10]

5.7 Mist/ Drift Eliminator:

Drift eliminator protects the water droplets from carryover due to countercurrent airflow. This not only saves precious water, but also allows us to estimate the water resident in the system. We select a commercially available Mist eliminator of size 48 x 48 x 2 (or 4) inches.

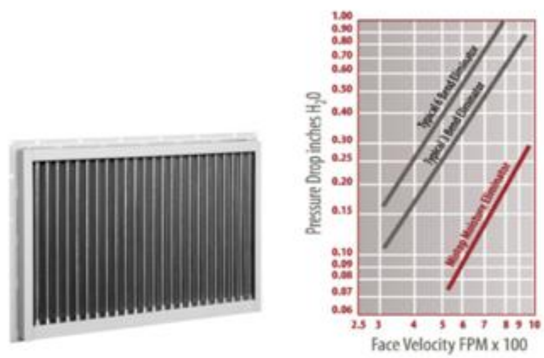


Figure 5-7-1: Mist/ Drift Eliminator [17]

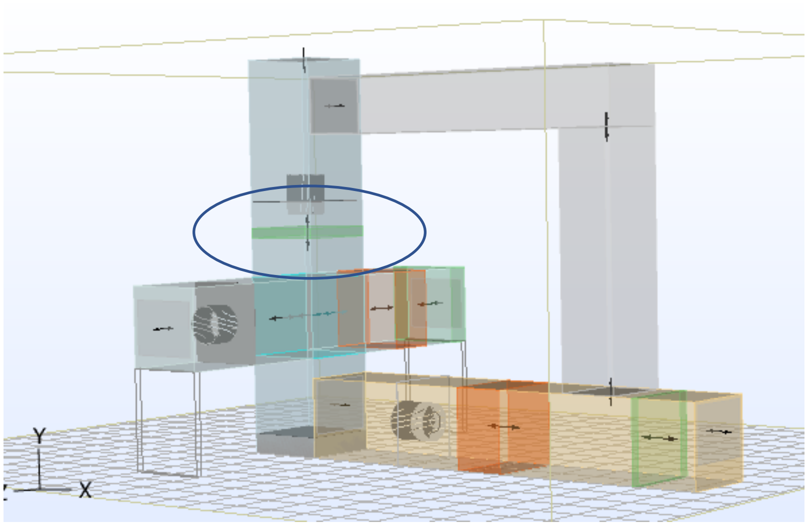


Figure 5-7-2: Model on 6Sigma software [10]

5.8 Motorized Volume Control Damper (VCD):

We select commercially available VCDs. These dampers should have a modulating operating mode that enables them to constantly change the opening based on control signals. 4 VCDs are selected to vary the mixture of return and fresh air as per set condition.

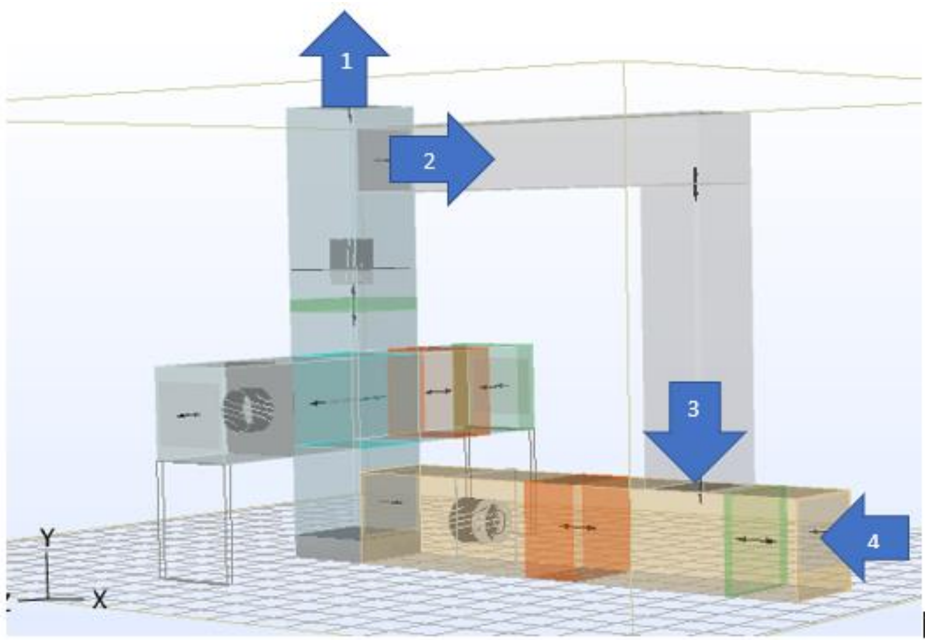


Figure 5-8-1: IEC and Test Model on 6Sigma software [10]

Figure 5-8-2: Motorized Volume Control Damper [18]

5.9 Water Collection Grid:

We design a plastic water collection sump – 48 x 48 x 12 inches, divided in 36 equal grids, aligned vertically below the Heat Exchanger. The 36 grids allow us to study water distribution across the heat exchanger [19].

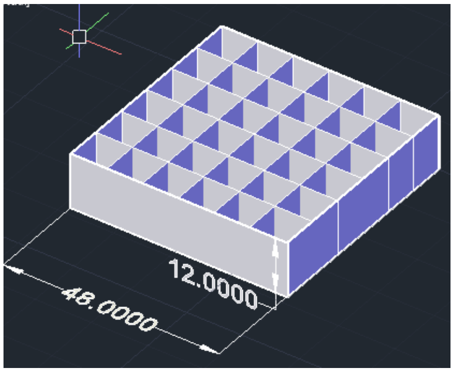


Figure 5-9-1: Water Collection Grid

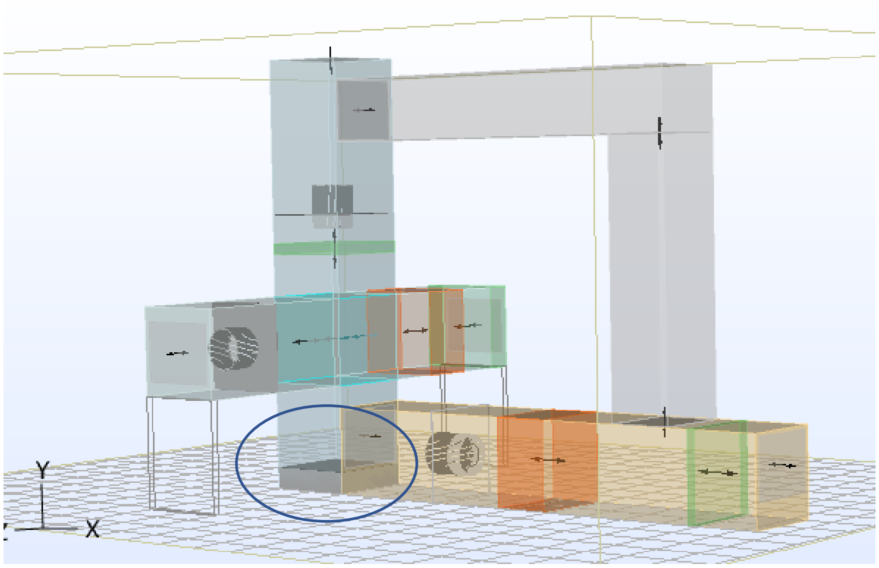


Figure 5-9-2: IEC and Test Model on 6Sigma software [10]

Chapter 6

CALCULATIONS AND REPORTS

6.1 Test Setting:

This setup should create temperature and humidity conditions, emulating a typical DC, to measure and report performance under several parameters.

Primary Side				
Ambient		Process	Test Condition	
Temp (F)	R.H (%)		Temp (F)	R.H (%)
70	30	Sensible heating	90	16
70	30	Sensible heating	100	12
70	30	Sensible heating	110	9.05

Table 6-1-1: Sample DC condition Setpoints on Primary Side

Secondary Side				
Ambient		Process	Test Condition	
Temp (F)	R.H (%)		Temp (F)	R.H (%)
70	30	Not Applicable	70	30
70	30	Return Saturated Air mixing + Sensible heating	90	50
70	30	Return Saturated Air mixing + Sensible heating	110	60

Table 6-1-2: Sample DC condition Setpoints on Secondary Side

6.2 Process Path of Secondary Air:

Secondary air process is a mixture of adiabatic and sensible cooling. As air enters the secondary side, before reaching the heat exchanger, it is cooled adiabatically by the water spray. In this process, air gains humidity and approaches its saturation point. As air is forced through the heat exchanger channels, it exchanges heat with the thin water film on the heat exchanger plate of secondary side, that gains heat through conduction from the primary airstream. As we know the primary air is hot DC return air, the secondary side airstream can be seen as a forced laminar flow over an isothermal plate. The convection heat transfer to secondary air stream due to primary side can be calculated using Newton's law of cooling [22].

$$\dot{Q} = h_{plate} A \left(T_w - T_\infty \right)$$

$$\dot{Q}_{plate} = 0.664 \cdot \left(Pr \right)^{1/3} \cdot \sqrt{Re_L} \cdot \frac{kA}{L} \Delta T$$

Figure 6-2-1: Heat Convected from Hot Isothermal Plate to Forced Laminar Flow [22]

While secondary air gains heat, its water carrying capacity increases. This air, being in continuous influence of the water spray, is constantly cooled adiabatically. Below is general schematic to understand the process path of secondary air in an IEC module.

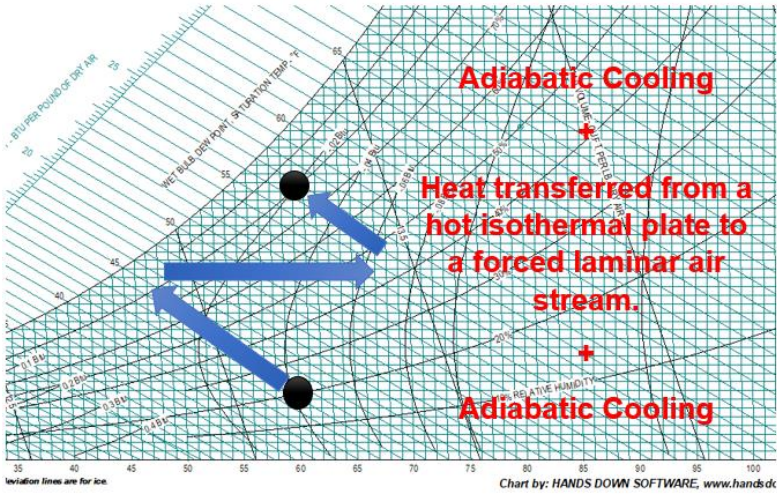


Figure 6-2-2: Process plot on Carrier Handsdown Software [20]

6.3 Control of condition on secondary side: Let us consider a case study where we want to create 110 F @ 35 % R.H with +-1% for 8000 cfm of air. Please note that these numbers and processes are samples to explain the process path of secondary air. In this designed test method, secondary side air shall be measured at the designated measuring stations. Further control settings shall be dependent on measurements from these stations.

	Temp (F)	R.H (%)
primary side air set temperature	110	9
ambient air	70	30

	1	
	Air at the inlet of Secondary side at the start of the cycle	
	Temp (F)	R.H (%)
cycle 1	70	30

Table 6-3-1: Secondary Side – Auxiliary AHU Return Air Mixing

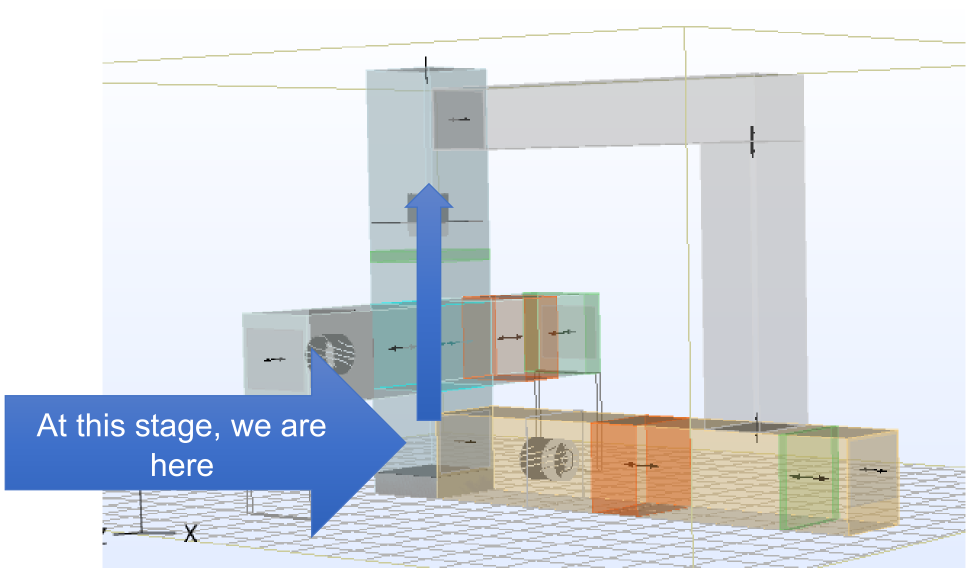


Figure 6-3-1: IEC and Test Model on 6Sigma software [10]

Air entering the secondary side is adiabatically cooled due to the influence of water spray.

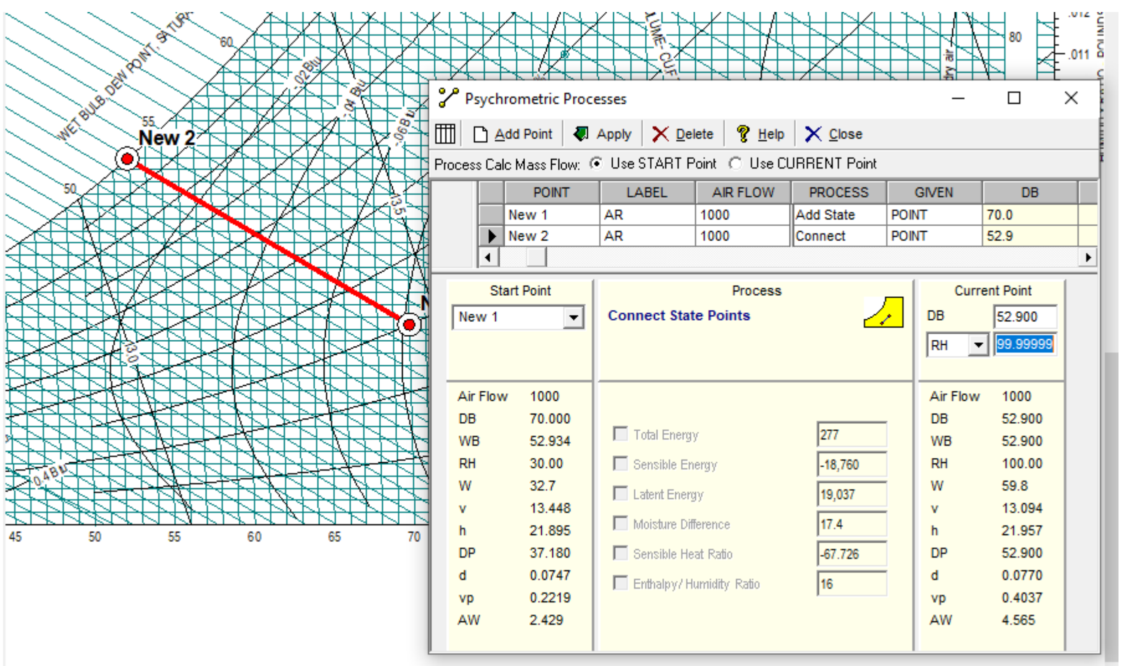


Figure 6-3-2: Process plot on Carrier Handsdown Software [20]

	1		2		3	
	Air at the inlet of Secondary side at the start of the cycle		Adiabatic cooling		heat transfer due to convection	
	Temp (F)	R.H (%)	Temp (F)	R.H (%)	Temp (F)	R.H (%)
cycle 1	70	30	52.9	100	81.5	37

Table 6-3-1: Secondary Side – Auxiliary AHU Return Air Mixing

As air reaches the heat exchanger plates, it gains heat convected by hot primary side plate. Therefore, in large number of miniscule steps, air gains heat sensibly and again cools adiabatically.

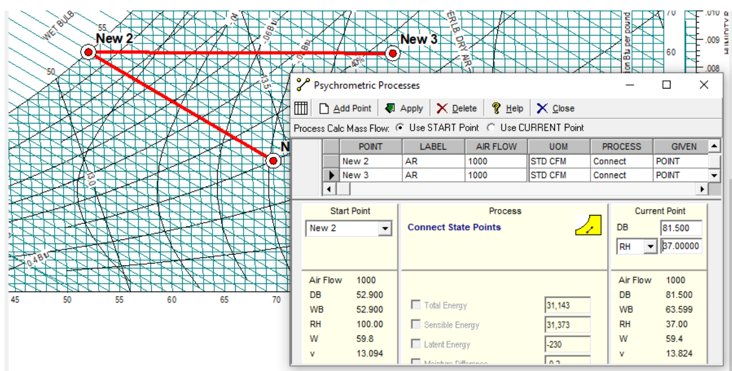


Figure 6-3-3: Process plot on Carrier Handsdown Software [20]

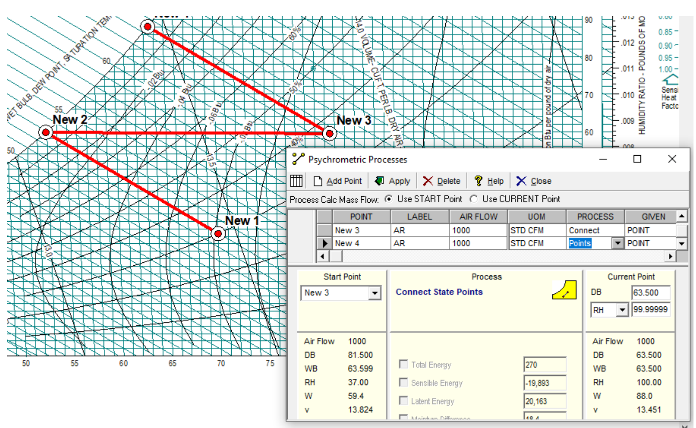


Figure 6-3-4: Process plot on Carrier Handsdown Software [20]

	1		2		3		4	
	Air at the inlet of Secondary side at the start of the cycle		Adiabatic cooling		heat transfer due to convection		Adiabatic cooling	
	Temp (F)	R.H (%)	Temp (F)	R.H (%)	Temp (F)	R.H (%)	Temp (F)	R.H (%)
cycle 1	70	30	52.9	100	81.5	37	63.6	100

Table 6-3-1: Secondary Side – Auxiliary AHU Return Air Mixing

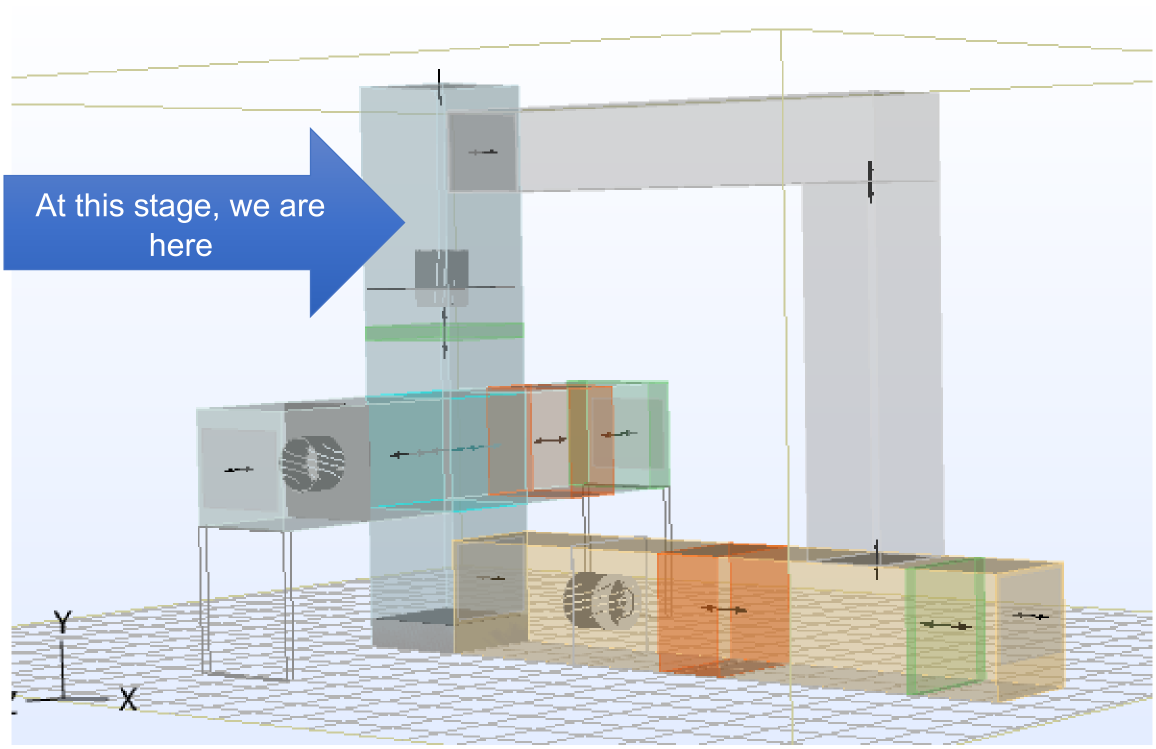


Figure 6-3-5: IEC and Test Model on 6Sigma software [10]

From this point, we exhaust certain amount of air and recirculate the rest back to the auxiliary unit to create a mixture with the ambient air. For this sample calculation, we assume 75% of return air and balance 25% of ambient fresh air. As discussed above, 4 motorized Volume Control Dampers (VCD) are selected to modulate the return air and fresh air percentage in the system. We use an online tool to calculate the result of this mixture [21].

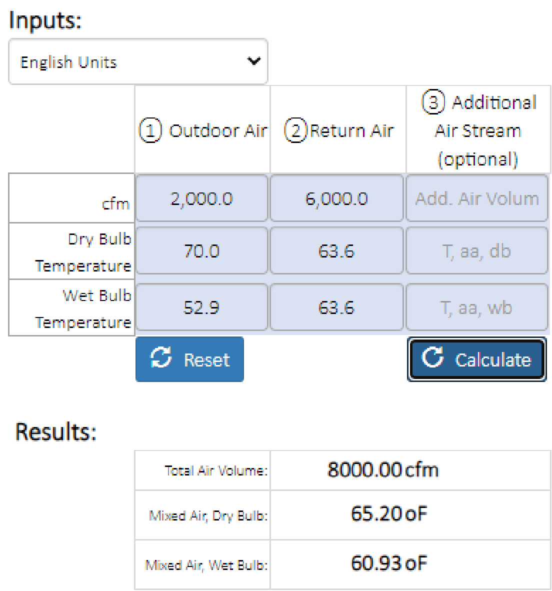


Figure 6-3-6: Return and Fresh Air Mixture [21]

	Air at the inlet of Secondary side at the start of the cycle		Adiabatic cooling		heat transfer due to convection		Adiabatic cooling		Return air VCD opening	Ambient air VCD opening	Air Mixture in Auxiliary AHU	
	Temp (F)	R.H (%)	Temp (F)	R.H (%)	Temp (F)	R.H (%)	Temp (F)	R.H (%)	75%	25%	Temp (F)	R.H (%)
cycle 1	70	30	52.9	100	81.5	37	63.6	100	6000 cfm	2000 cfm	65.2	78.6

Table 6-3-1: Secondary Side – Auxiliary AHU Return Air Mixing

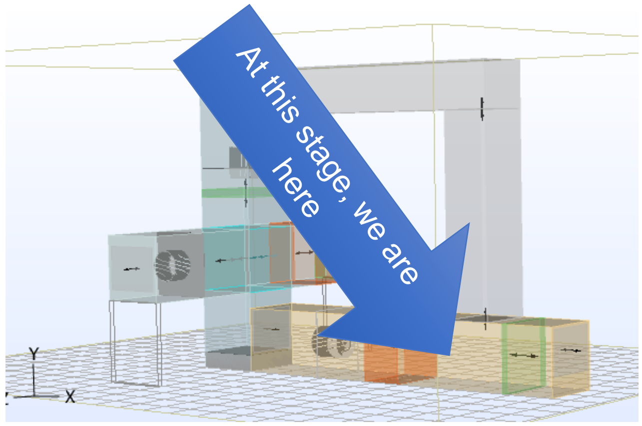


Figure 6-3-7: IEC and Test Model on 6Sigma software [10]

This air mixture is heated by the auxiliary electric heater that is set to sensibly heat the air to 110 Deg F as a starting assumption of this sample calculation. This heated air enters the secondary side suction as beginning of cycle 2.

	1		2		3		4		5		6		7	
	Air at the inlet of Secondary side at the start of the cycle		Adiabatic cooling		heat transfer due to convection		Adiabatic cooling		Return air VCD opening	Ambient air VCD opening	Air Mixture in Auxiliary AHU		Sensible heating set to 110F	
	Temp (F)	R.H (%)	Temp (F)	R.H (%)	Temp (F)	R.H (%)	Temp (F)	R.H (%)			Temp (F)	R.H (%)	Temp (F)	R.H (%)
cycle 1	70	30	52.9	100	81.5	37	63.6	100	6000 cfm	2000 cfm	65.2	78.6	110	19.2
cycle 2	110	19.2	74.9	100	92.5	57	79.48	100	6000 cfm	2000 cfm	77.13	81.8	110	29.5

Table 6-3-1: Secondary Side – Auxiliary AHU Return Air Mixing

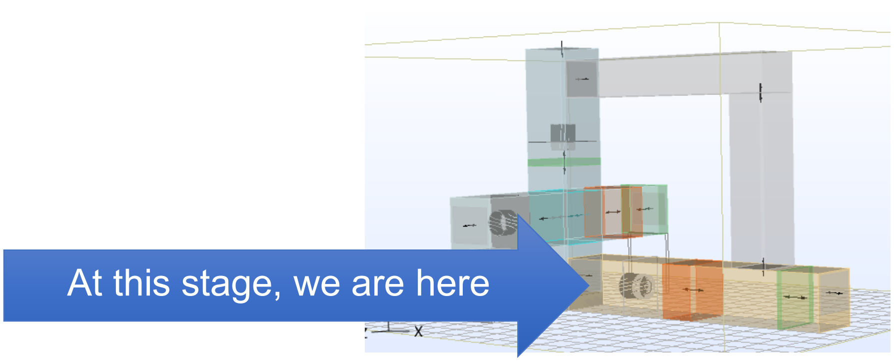


Figure 6-3-8: IEC and Test Model on 6Sigma software [10]

In this way a desired condition can be achieved by running various cycles. For a uniform and stable condition, it is advisable to achieve small increments in temperature and humidity by mixing the return and fresh air with close proportions. Due to response time for VCD opening adjustment, a stable condition can be achieved with slow increments over a period of 5-6 minutes.

	1		2		3		4		5		6		7	
	Air at the inlet of Secondary side at the start of the cycle		Adiabatic cooling		heat transfer due to convection		Adiabatic cooling		Return air VCD opening	Ambient air VCD opening	Air Mixture in Auxiliary AHU		Sensible heating set to 110F	
	Temp (F)	R.H (%)	Temp (F)	R.H (%)	Temp (F)	R.H (%)	Temp (F)	R.H (%)			Temp (F)	R.H (%)	Temp (F)	R.H (%)
cycle 1	70	30	52.9	100	81.5	37	63.6	100	6000 cfm	2000 cfm	65.2	78.6	110	19.2
cycle 2	110	19.2	74.9	100	92.5	57	79.48	100	6000 cfm	2000 cfm	77.13	81.8	110	29.5
cycle 3	110	29.5	81.183	100	95.6	63.5	84.45	100	6000 cfm	2000 cfm	80.88	82.58	110	34
cycle 4	110	34	83.67	100	96.8	66.5	86.56	100	6000 cfm	2000 cfm	82.45	82.87	110	35.5
cycle 5	110	35.5	84.58	100	97.3	67.5	87.35	100	6000 cfm	2000 cfm	83.05	82.98	110	36.5
cycle 6	110	36.5	85	100	97.5	68	87.71	100	5700 cfm	2300 cfm	82.61	80.48	110	35

Table 6-3-1: Secondary Side – Auxiliary AHU Return Air Mixing

As we can see above, the desired temperature and humidity was achieved in cycle 4. As the humidity starts to deviate, air mixture was adjusted in cycle 6 to maintain the humidity at the set condition of 35%. This adjustment is achieved by varying the VCD opening with the help of a programmable controller.

6.4 Measurements

Measurements of airflow, differential pressure, temperature, and humidity are taken across the 6 designated stations for further calculations. Parameters are numbered according to the station. E.g.: 2T_db = Dry bulb temperature recorded at station 2, which is at suction of auxiliary AHU.

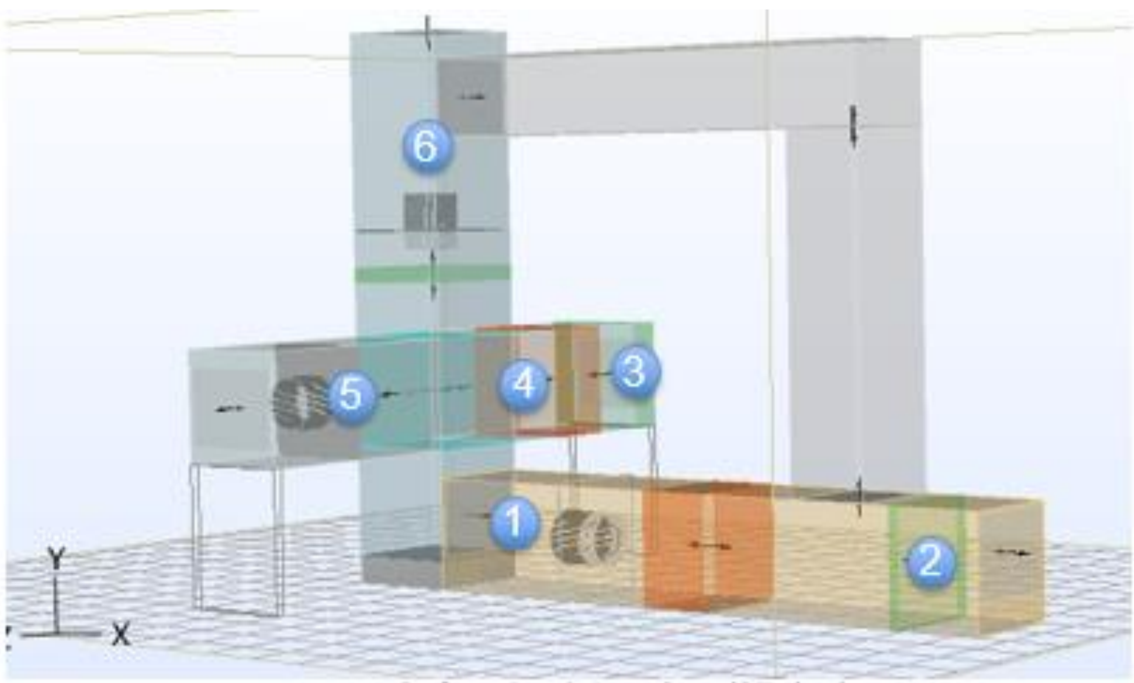


Image Source: Screenshot from 6Sigma design. Author – Abhijit Bhosale

Figure 6-4-1: IEC and Test Model on 6Sigma software [10]

Below table describes sample (assumed) readings to demonstrate implementation of the test method.

Primary Suction				Primary Heater					Primary Fan (near discharge)				Aux. Suction					Aux. discharge/ Secondary Suction				Secondary Fan (near discharge)				
3	3	3	3	4	4	4	5	5	5	5	5	2	2	2	2	1	1	1	1	Hx Face Velocity (fpm) = 1_cfm/ 16 sq.ft (Face area of Hx)	1	6	6	6	6	6
T_Db	T_Wb	R.H (%)	cfm	T_Db	T_Wb	R.H (%)	T Db	T Wb	R.H (%)	cfm	dP	T Db	T Wb	R.H (%)	cfm	T Db	T Wb	R.H (%)	1_cfm	1_Fv	DP	T Db	T Wb	R.H (%)	cfm	DP
70	52.9	30	8000	90	60.8	16	76	61.9	40	8000	2.8	70	52.9	30	2000	110	86.79	40	8000	500	1.8	87.7	87.7	100	8000	3.1
70	52.9	30	8000	90	60.8	16	79	63.4	40	8000	2.75	70	52.9	30	2000	110	86.79	40	7000	437.5	1.78	87.9	87.9	100	7000	3
70	52.9	30	8000	90	60.8	16	80	63.4	40	8000	2.7	70	52.9	30	2000	110	86.79	40	6000	375	1.7	89.14	89.14	100	6000	2.75

Table 6-4-1: Measurements Table

6.5 Calculations: Below are IEC module performance calculations on sample (assumed) measurement data to demonstrate implementation of the test method.

6.5.1 Cooling Effectiveness and Capacity

cooling effectiveness	**Cooling capacity**
C_E = (4Tdb-5Tdb) / (5Tdb-4Twb)	C_q (BTU) = 1.08 x 5cfm x (4Tdb-5Tdb)
C_E	C_q
0.95	207360
0.60	95040
0.52	86400

Table 6-5-1-A: IEC Performance Calculation

4Tdb = Entering air Temp_dry bulb

5Tdb = Exiting air Temp_dry bulb

4Twb = Entering air Temp_wet bulb

5cfm = Airflow (cfm) recorded at exit

6.5.2 Water Evaporated

Water Evaporated				
humidity density of 1T_Db Lb/cub.ft	humidity density of 6T_Db Lb/cub.ft	water content (kg/min) = (hd x cfm)/2.2		Total Water Evaporated (kg/min) = W_hd6 - W_hd1
hd1	hd6	W_hd1	W_hd6	W_Ev
0.0687	0.0713	249.82	259.27	9.45
0.0687	0.0712	218.59	226.55	7.95
0.0687	0.071	187.36	193.64	6.27

Table 6-5-1-B: IEC Performance Calculation

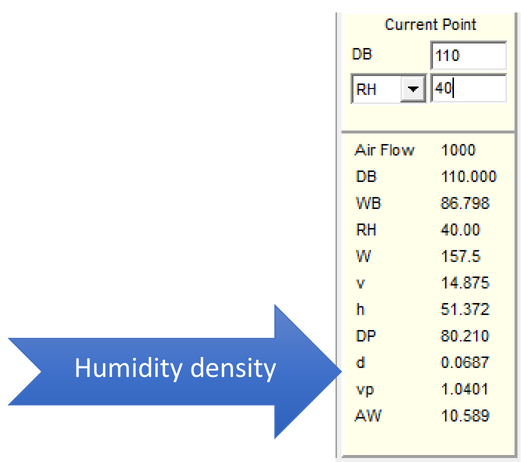


Figure 6-5-2: Carrier Handsdown Software [20]

1T_db = Secondary side entering air Temp_dry bulb

6T_db = Secondary side exiting air Temp_dry bulb

6.5.3 Water Consumption

Water Consumption									
Mass of Water spray: Nozzle 1 (kg/min)	Mass of Water spray: Nozzle 2 (kg/min)	Mass of Water spray: Nozzle 3 (kg/min)	Mass of Water spray: Nozzle 4 (kg/min)	Mass of Water spray: Nozzle 5 (kg/min)	Mass of Water spray: Nozzle 6 (kg/min)	Total Mass of Water Spray (kg/ min) W_s = sum (Nz1 to 6)	Mass of water Collected in grid(kg/min)	Total Water consumption (kg/min)= W_Ev + W_g	Mass of Water resident on Heat Exhanger plates (kg/min) W_r = W_s - (W_g+W_Ev)
W_Nz1	W_Nz2	W_Nz3	W_Nz4	W_Nz5	W_Nz6	W_s	W_g	W_c	W_r
4.158	4.158	4.158	4.158	4.158	4.158	24.948	14.6	24.05	0.893
4.158	4.158	4.158	4.158	4.158	4.158	24.948	12.6	20.55	4.393
4.158	4.158	4.158	4.158	4.158	4.158	24.948	12	18.27	6.675

Table 6-5-1-C: IEC Performance Calculation

6.5.4 Fan Characteristics

Fan Characteristics										
Primary Fan airflow (cfm)	Total Pressure drop (in-wg)	Velocity Pressure (in-wg)	Primary Fan static pressure (in-wg) = Total Pressure - Velocity pressure	Primary Fan absorbed Power (watts)	Secondary Fan airflow (cfm)	Total Pressure drop (in-wg)	Velocity Pressure (in-wg)	Secondary Fan static pressure (in-wg) = Total Pressure - Velocity pressure	Secondary Fan absorbed Power (watts)	Total Fan Power absorbed (watts)
P_cfm	P_Tp	P_vp	P_sp	P_pw	S_cfm	S_Tp	S_vp	S_sp	S_pw	F_pw
8000	2.8	0.3	2.5	3400	8000	3.1	0.35	2.75	3600	7000
8000	2.75	0.25	2.5	3350	7000	2.85	0.25	2.6	3500	6850
8000	2.7	0.2	2.5	3325	6000	2.75	0.35	2.4	3300	6625

Table 6-5-1-D: IEC Performance Calculation

6.6 Reports

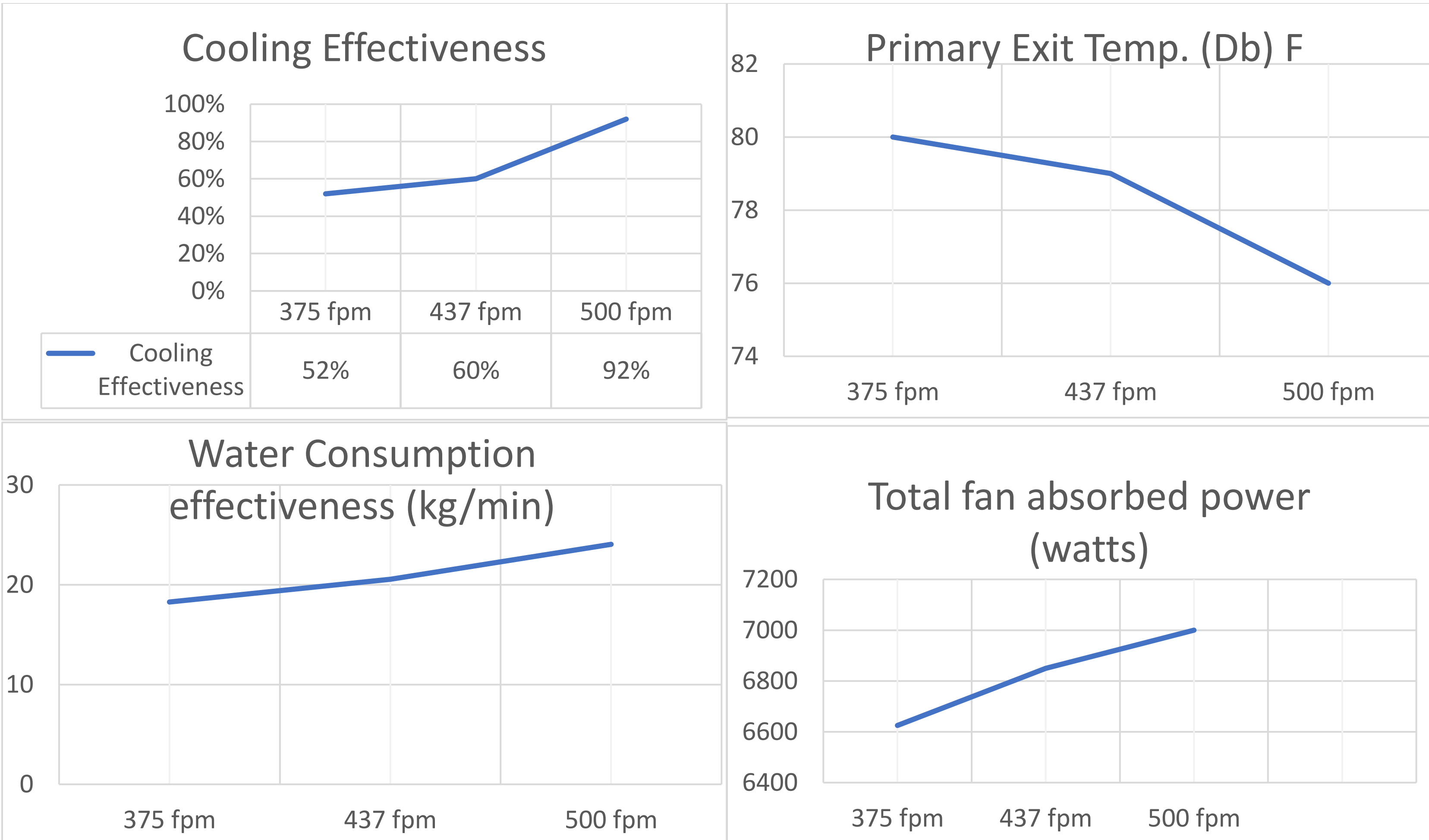
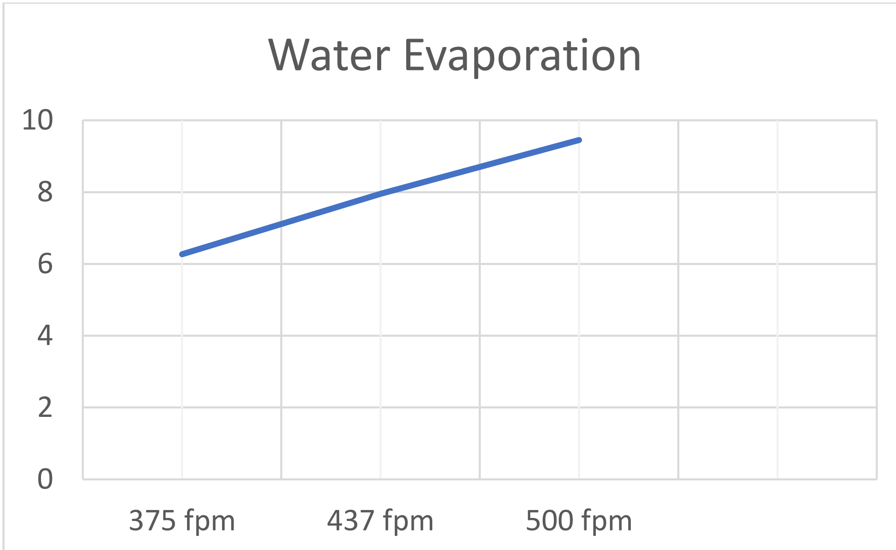


Figure 6-6-1: IEC Module Performance Plots on Sample (Assumed) Measurements and Calculations



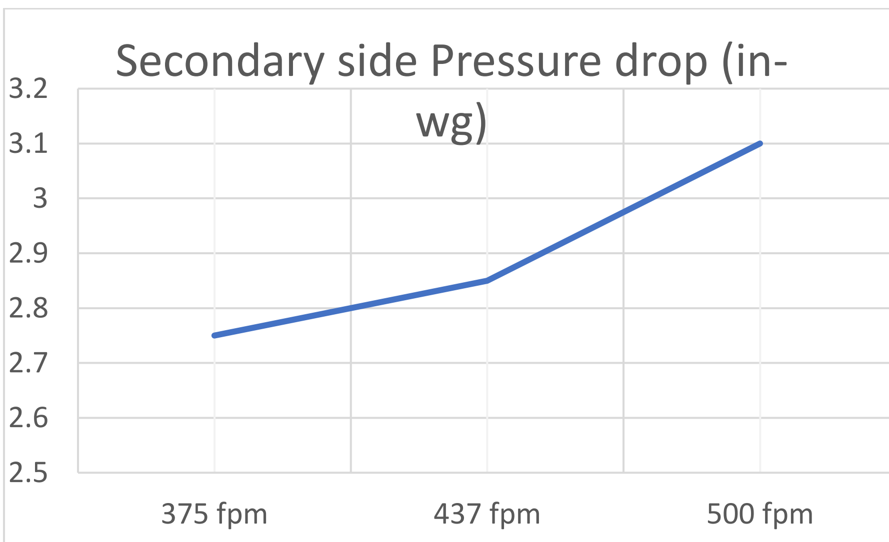


Figure 6-6-2: IEC Module Performance Plots on Sample (Assumed) Measurements and Calculations

Chapter 7

SUMMARY

Characterization of primary side exit temperature, cooling effectiveness, secondary side power consumption, and water usage allows DC operators to maximize PUE. For e.g., in the given sample calculations, while operating at 6000cfm of secondary air, we can achieve 80F on the primary side. This stays within the ASHRAE's recommended zone. In this choice, the operator saves 9% on fan power and 35% on water consumption. The Test method lets us create stable conditions of hot-humid environment. Evaporative cooling efficiency drops as the conditions become and hot and humid. Thus, this method allows testing the outer operating limits of IEC modules for DC. Various spray methods can be employed and tested to maximize cooling effectiveness. This Test method can be used to scale IEC modules in the multiples of 8000cfm in non-stacked arrangement of crossflow plate heat exchangers.

Chapter 8

UNCERTAINITIES AND FUTURE STUDY

8.1 Uncertainties

Controller and VCD response time gap may cause small but constant fluctuation in the conditions. This can be minimized by slowing the mixing process. Water resident on heat exchanger plates cannot be estimated accurately as a small portion may be resident on mist eliminator or vent walls.

8.2 Future Study

This method can be developed to study effects of 2 level stacked arrangement of Crossflow Plate Heat Exchangers. Various types of spray methods can be employed to investigate and find better cooling effectiveness.

REFERENCES

[1] HIGH EFFICIENCY INDIRECT AIR ECONOMIZER-BASED COOLING FOR DATA CENTERS, WHITE PAPER 136 (SCHNEIDER), BY WENDY TORELL. THE EXPANDING OPERATING ENVIRONMENT.

[2] HEATEX OFFICIAL WEBSITE, PRODUCT DATA, AND ENGINEERING INFORMATION AVAILABLE ON: WWW. HEATEX. COM/PRODUCTS/ CROSS-FLOW

[3] NOVEL MODELING OF AN INDIRECT EVAPORATIVE COOLING SYSTEM WITH CROSS-FLOW CONFIGURATION BY GHASSEM HEIDARINEJAD AND SHAHAB MOSHARI.

[4] 2011 DUNNAVANT ASHRAE JOURNAL DATA CENTER HEAT REJECTION INDIRECT AIR SIDE ECONOMIZER CYCLE.

[5] ANSI/ASHRAE STANDARD 143-2015, METHOD OF TEST FOR RATING INDIRECT EVAPORATIVE COOLERS.

[6] EVAPORATIVE COOLING DESIGN GUIDELINES MANUAL FOR NEW MEXICO SCHOOLS AND COMMERCIAL BUILDINGS. AUTHOR: J. D. PALMER, P.E., C.E.M. NRG ENGINEERING, FUNDED BY: UNITED STATES DEPARTMENT OF ENERGY AND NEW MEXICO ENERGY MINERALS AND NATURAL RESOURCES DEPARTMENT ENERGY CONSERVATION AND MANAGEMENT DIVISION

[7] BETE PRODUCT CATALOGUES AND ENGINEERING INFORMATION.

[8] EXPERIMENTAL INVESTIGATION OF WATER SPRAYING IN AN INDIRECT EVAPORATIVE COOLER FROM NOZZLE TYPE AND SPRAY STRATEGY PERSPECTIVES. TIEZHU SUN A, XIANG HUANG A, YI CHEN B, HONG ZHANG A.

[9] NUMERICAL AND EXPERIMENTAL STUDY ON SPRAY COOLING SYSTEM DESIGN FOR COOLING PERFORMANCE ENHANCEMENT OF NATURAL DRAFT DRY COOLING TOWERS, YUBIAO SUN.

[10] 6SIGMA DATA CENTER COOLING DESIGN AND SIMULATION SOFTWARE BY FUTURE FACILITIES, USA.

[11] PRODUCT SELECTION BY HEATEX SELECTION SOFTWARE HTTPS :// HEATEXSELECT. HEATEX. COM.

[12] ZIEHL ABEGG PRODUCT INFORMATION AND SOFTWARE SELECTION OUTPUT.

[13] PARAGON CONTROLS AIRFLOW AND TEMPERATURE MEASUREMENT PRODUCT INFORMATION.

[14] MANIFOLD SIZING CALCULATOR AVAILABLE ON WASHINGTON STATE UNIVERSITY WEBSITE SOURCE: HTTP://IRRIGATION.WSU.EDU/CONTENT/ CALCULATORS/ GENERAL/ PRESSURE-LOSS-WITH-OUTLETS.PHP

[15] HEATER PRODUCT SELECTION AND GENERAL ARRANGEMENT DRAWING BY NEPTRONIC, CANADA.

[16] MERV 8 PRE-FILTER PRODUCT SELECTION AND ENGINEERING INFORMATION FROM GLASFLOSS PRODUCT CATALOGUE.

[17] MIST ELIMINATOR PRODUCT SELECTION AND ENGINEERING INFORMATION FROM MANUFACTURER “MISTOP”.

[18] VOLUME CONTROL DAMPER (VCD) PRODUCT IMAGE FROM PRUDENTAIRE WEBSITE: HTTPS :// WWW. PRUDENTAIRE. COM/.

[19] WATER COLLECTION GRID DESIGNED USING AUTOCAD 2018 STUDENT VERSION.

[20] HEATING AND COOLING PROCESS PLOT USING CARRIER VERSION HANDSDOWN SOFTWARE FREE PACKAGE.

[21] AIR MIXTURE CALCULATION ON: HTTPS: //WWW. ADICOTENGINEERING. COM/ AIR-MIXING-CALCULATOR.

[22] HEAT CONVECTION FROM HOT ISOTHERMAL PLATE TO FORCED LAMINAR FLOW THEORY AND CALCULATION ON EFUNDA.COM, INCLUDING ONLINE CALCULATOR: HTTPS: //WWW. EFUNDA.COM/ FORMULAE/ HEAT_TRANSFER/ CONVECTION_FORCED/CALC_LAMFLOW_ISOTHERMALPLATE. CFM.

[23] CABINET SIZING OF CENTRIFUGAL PLUG FAN GUIDELINES FROM KRUGER VENTILATION BNB SERIES PRODUCT CATALOGUE. HTTPS ://WWW. KRUGERFAN. COM/.

[24] SYSTEMS WITH MANIFOLDS BY MARIO DONINELLI (CALEFFI).

[25] HVAC COOLING SYSTEMS FOR DATA CENTERS BY A. BHATIA, CED ENGINEERING.