COMPUTATIONAL STUDY AND OPTIMIZATION OF WET COOLING MEDIA IN DIRECT EVAPORATIVE HEAT EXCHANGERS

by

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Abstract

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Evaporative cooling media pads are the primary component of importance in direct evaporative cooling units. To increase the overall efficiency of direct evaporative cooling systems, a wet cooling media pad is considered for further optimization in design and operation. An installation technique called ‘staging’ of cooling media pads is examined and evaluated for its incremental cooling, water usage, and operational cooling efficiency. For Staging, a cooling media is divided into multiple vertical sections, and separate water distribution headers control each part. Staging of cooling media can reduce water consumption in data centers and potentially lessen the amount of cooling media required. Increasing the number of stages allows more precise control of humidity and temperature at the discharge.

The objective of the present study is to validate a computational model in Computational Fluid Dynamics (CFD) software in predicting the cooling efficiency of wet cooling pads. At the initial level of this project, a CFD model of cooling media will be developed and analyzed. Then, this CFD model will be validated with existing experimental data for GLASdek or CELdek cooling pads. The basic parameters of influence such as pad thickness, flute angle, water flow rate are considered for parametric studies and their impact on cooling media

saturation effectiveness, the pressure drop across the media and water consumption will be reported. Also, the degree of increase in cooling capacities as the cooling media wall is scaled/stacked up to accommodate higher cooling unit flow discharge will be investigated. Furthermore, this model will be further modified for ‘staged’ installations and analyzed through the simulations. This information will be helpful in understanding and improving wet cooling media performance.

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Chapter 1

INTRODUCTION

Today, Cooling Systems are a major component in almost every industry. Especially in Data Centers, IT and Electronic components must be maintained at certain temperature and humidity. With the advances in computing technologies need for larger data centers are increasing. More and more people are connecting to internet cloud, and that is resulting in growing number of data centers over the world. Every year with more innovations, the computing power of servers is also increasing. To enhance the life of these data centers and stop from complete failure, appropriate cooling techniques must be developed considering their high efficiency and less maintenance cost.

With the advent of innovative Data center cooling techniques, there has always been an additional cost associated with the operations. Cooling data centers efficiently with the usage of fewer resources and less power consumption is important. Evaporative cooling systems provides optimum cooling with minimum energy usage but is significantly dependent on resources like water and type of cooling media.

Figure 1-1 Facebook Data Center

1.1 Motivation

According to data published by datacenterknowledge.com [1], US data centers consumed about 70 billion kWh of electricity in 2014, which is representing 2 percent of the country’s total energy consumption. That’s equivalent to the amount consumed by about 6.4 million homes. Data centers consume a lot of energy produced, so efforts must be taken for energy efficiency improvement. Out of total energy consumed, 1/3rd of energy is spent on cooling systems. Improving cooling effectiveness in the data center can save a lot of energy. Hence understanding and study cooling systems in the data center is a must. According to data published, Energy efficiency improvements have saved over 40 billion kWh energy between 2010 and 2014.

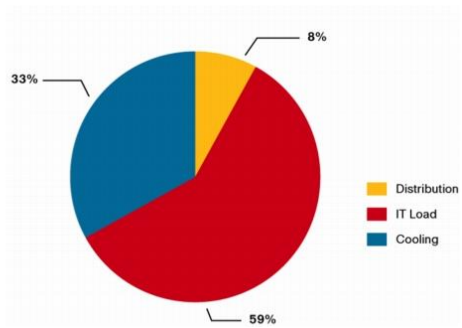


Figure 1-2 Energy Consumption in Data Centers

There are different types of cooling systems used in data centers like (Direct Evaporative cooling, Indirect Evaporative cooling, Free cooling, Economizer and Direct Expansion Cooling, etc.) In this project, we will discuss one of the cooling systems which are critical and widely used in data center cooling systems. Evaporative cooling systems are a proven solution within the industries such as food &beverage, Aerospace, Chemical & Pharmaceutical, logistics and warehouses, general manufacturing, education &modern offices.

1.2 Types of Cooling Systems used in Data Centers

The American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE) [2] publishes guidelines for operating range of the temperature and humidity for IT equipment. To reduce energy consumption and maintain high efficiency, so IT equipment must be kept at these allowable limits. Various types of cooling systems are used in combination to achieve required cooling. ASHRAE TC 9.9 standard is shown below, which describes allowable envelope of operating parameters like Temperature and Humidity.

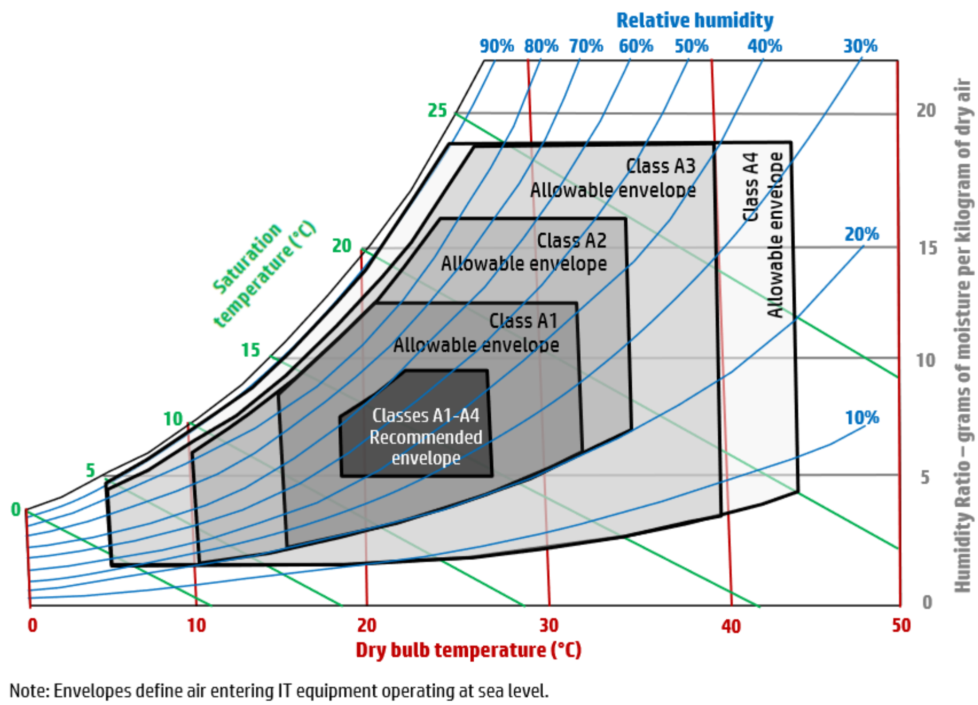


Figure 1-3 ASHRAE specifications per class [2]

As per the requirement and the outside air conditions, a particular cooling system or a combination of systems are used to maintain that air within the ASHRAE envelope.

Cooling Systems used in Data Centers:

- Free Cooling
- Air side Economizers
- Direct Evaporative Cooling
- Indirect Evaporative Cooling

a) **Free Cooling:** There are mainly three types of Free cooling,

i) *Air-side free cooling:*

In this kind of cooling system, The outside air is directly pumped into data center through filters. Filters can be replaced periodically to avoid air resistance due to containment build up. Sometimes a significant amount of air requires a cooling server, which can increase pumping power at the server level. Some particular condition these methods can save so much of energy that is why it's called as free cooling.

ii) *Adiabatic Free Cooling:*

It is a slightly different than airside free cooling. Air is brought into the chamber and passed through the water cooling tower and due to the effect of evaporation air is cooled. Then, this cooled air is supplied to the data center.

iii) *Water-Side Free Cooling:*

The cooling medium such as water or glycol is used passed through cooling tower instead of chillers or compressor to achieve free cooling. Water-side free cooling is most logical. Anyway, most data centers use chilled water to cool their systems. So it's easy to use water-side free cooling because the piping and the cooling towers are already in the plan. That makes the cost huge cost saving.

b) **Air-side Economizers:** Economizers, directly brings outside air into the building and provides to servers. Instead of, cooling it and circulating, exhaust air from servers is pulled outside. Sometimes outside, cold air is brought into mixing chamber and mixed with exhaust air so that its Temperature and humidity falls into desired range and then recirculated back to servers. Typical working of Economizers is shown below fig.

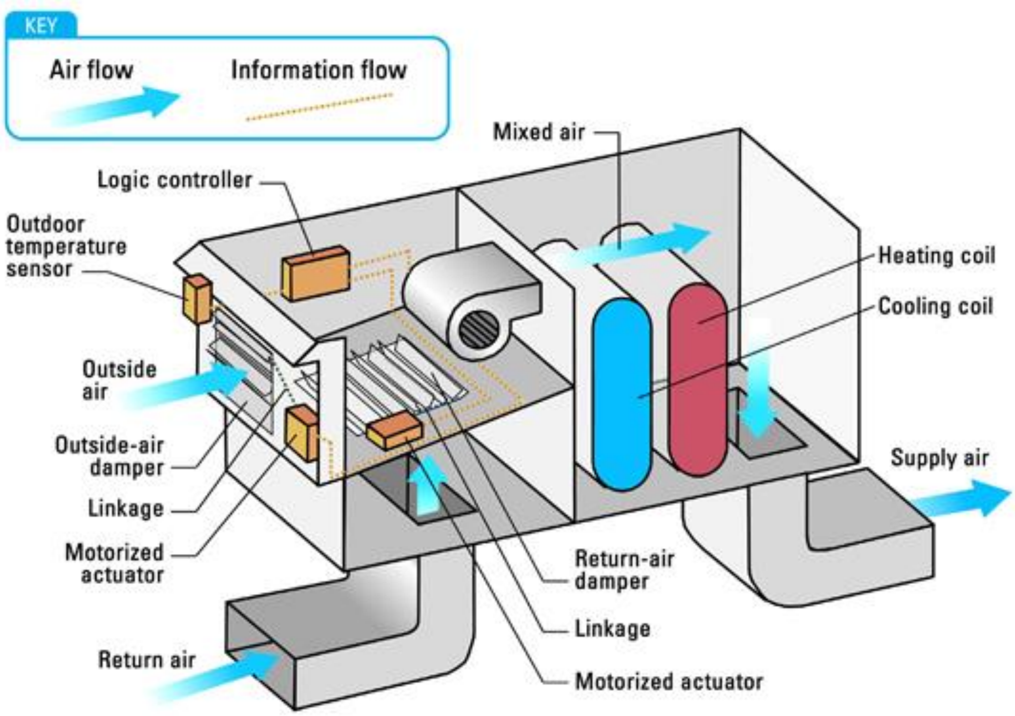


Figure 1-4 Air-Side Economizer [3]

c) **Direct Evaporative Cooling (DEC):** Evaporative cooling is a century old technique, used to achieve a very economical way of cooling. Water evaporation phenomenon is used to cool down the air. Previously, water is sprayed into the moving air, and due to water evaporation, the heat of air is rejected into the water, and as a result, water evaporates. The disadvantage was that the contact time between water and the air decreases, so the efficiency of the whole system is very less. To increase efficiency and improve contact between water and air, evaporative cooling pads were developed. The basic working of the typical evaporative cooler is very simple and as shown in fig below. Hot air came from one side and passed through the cooling pads, water flowed over the cooling pads through water distribution header and flows downwards due to gravity and stored down in the water reservoir. Fans pull the air from other side and due to pressure difference hot air flows over the wet surface of cooling pads through multiple channels. Air temperature decreases because of the water evaporation and cooled air come out from another side of the cooling pads.

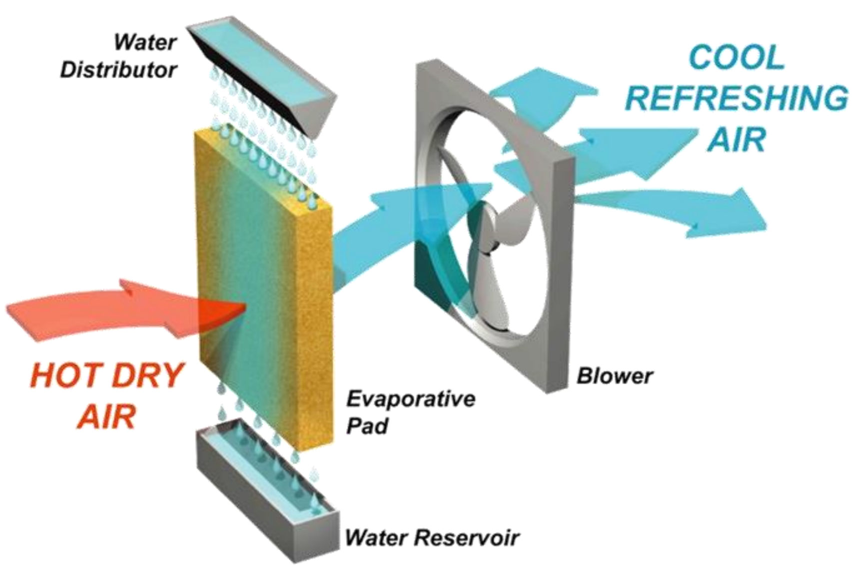


Figure 1-5 Direct Evaporative Cooling [4]

d) **In-Direct Evaporative Cooling (IEC):** In this type of cooling system, Outside air is cooled in the cooling tower, then this cooled secondary air is used to cool primary air using a heat exchanger, primary air is taken from inside of the facility. Then this cooled primary air is recirculated back to servers with the help of blowers. In IEC the primary air never mixed with outside air, so moister content in the air never increases. Both dry and wet bulb temperatures are reduced using this method. If outside air is somewhat cold, then heat exchanger can preheat the secondary air using exhaust air from servers.

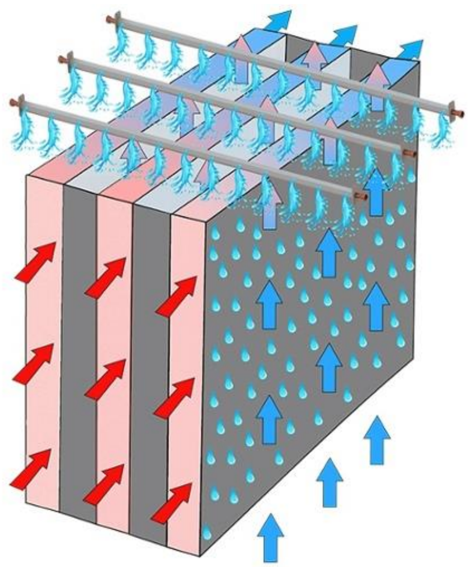


Figure 1-6 Indirect Evaporative Cooling [5]

Sometimes Indirect evaporative cooling is used along with DX cooling, the primary air is cooled using IEC to reach the desired level of temperature, if further cooling is required, then DX cooling is used to cool down the air further to achieve required cooling. IEC system often provides greater improved air quality (IAQ) than other systems.

1.3 Direct Evaporative Cooling (DEC)

Direct evaporative cooling is the easiest and most economical way to achieve air cooling. Passing outside air through thoroughly wet and saturated cooling pads (made up of cellulose) and air gets cooled due to the effect of evaporation. Then this cooled air recirculated server rooms. The performance of these systems largely depends on cooling pads. The evaporative cooling system provides a constant stream of air without adding any moisture content, unlike Mechanical cooling, they recirculate inside air to achieve cooling. Unlike the refrigeration system, evaporative cooling uses the addition of water into the air to lower the Temperature of air.

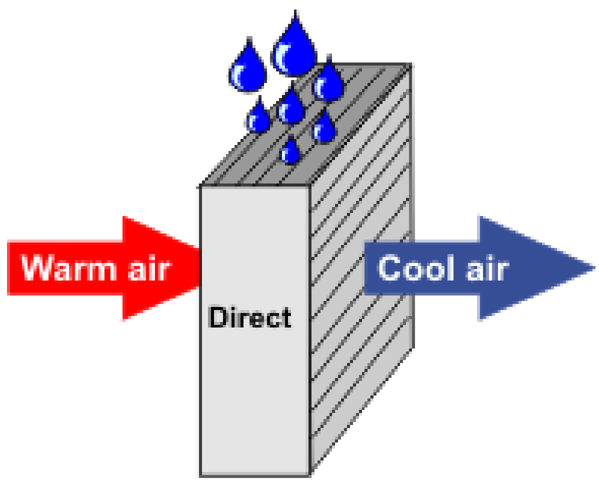


Figure 1-7 Basic Principle DEC [5]

Fundamental Principle is very simple to understand, the sensible heat from the air is used to provide energy to evaporate water. This energy is converted into sensible heat, which affects the temperature of the air. This process takes at constant enthalpy; hence it is called as an adiabatic process. The temperature drop and humidity increase are proportional to sensible heat reduction and latent heat gain respectively.

1.4 Types of Evaporative Cooling Pads

There are major two types of evaporative cooling pads. First made up of fibers (like Aspen, Cellulose, wood or synthetic). These are used in small applications. Another type is Rigid Media pads, usually employed in large applications. The primary difference between these type of pads is explained in the table below.

Type	Material used	Application & Advantages
Fiber Media	Aspen Cellulose Synthetic	• Used in small application with more than on Inlets • Usually very thin, up to 1 to 2 inches • High Maintenance & Replacement • Very Short Life • Aspen wood is durable for these type of application. • Cellulose and synthetic can be used in any sizes.
Rigid Media	Cellulose, Glass fiber or synthetic stacked corrugated sheet material	• Employed in large applications like Warehouses or data centers.
		• High initial investment cost.
		• Usually thick, available in various sizes, from 6 to 12 inches.
		• High efficiency, Air velocity is low, resulting in High humidity and lower temperature compared to Fiber Media
		• Longer work life compared to fiber media
		• Low maintenance.

Table 1-1 Different Types of Cooling Pads [6]

In this study, we are going to investigate only Rigid media, which are widely used in Data center cooling. Due to their high efficiency and low maintenance cost, rigid media are favored by industry experts in data center industry. Depending upon the application and use, we can choose the appropriate media. Different types of cooling media are shown below.

(a) Fiber Media – Aspen wood

(b) Rigid Media - Cellulose

Figure 1-8 Types of Evaporative Cooling Pads

Evaporative Cooling also depends on Environment factors, The Dry Bulb and Wet Bulb temperature of water, Temperature of air, Air Velocity but the most important factors are discussed below,

Three key things that drive the Evaporation:

a) Energy Input from moving air molecules to Water molecules at the surface

   (i.e. Increasing the kinetic energy so molecules can escape).

b) Diffusion of Water molecules

   (i.e. Spreading of water molecules into atmosphere).

c) Transport of Water vapor away from the water surface.

Chapter 2 – TESTING OF EVAPORATIVE COOLING PADS

For the validation of CFD model, Testing the Analytical model and Experimentally testing the Evaporative Cooling pads is very necessary. The experimental studies were carried out at The University of Texas at Arlington Facility. The former Students at EMNSPC carried out the experimental testing for various types of Cooling pads. The experimental studies will be validated with CFD data using System Resistance Curve. The scientist at the University of Shanghai for Science and Technology developed the Analytical Model for Rigid evaporative cooling media. We will use this analytical model to validate Experimental and CFD data.

2.1 Analytical Modeling

In the Study done by J.M. Wu, X. Huang, and H. Zhang, in 2005 [7]. The report shows the theoretical Analysis of Heat and Mass Transfer in Direct Evaporative Cooler. The numerical relation between Cooling efficiency and the geometric parameters of the cooling pad indicates that the dimensions of the cooling pads can affect the cooling performance. The paper shows, the simplified relation between cooling efficiency and cooling pad thickness.

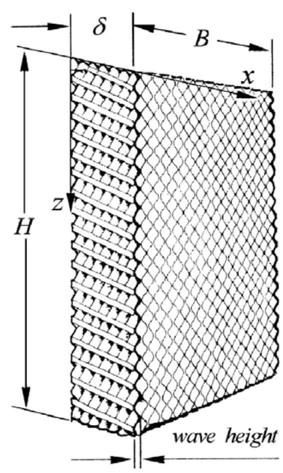


Figure 2-1 Rigid Media Cooling Pad

First, we will make the assumptions to develop The Analytical Model, to simplify the equation, we will neglect the least affecting factors.

The assumptions are as follows:

- Cooling pad material is wetted uniformly and fully.
- Convective heat transfer coefficient and mass transfer coefficient of moist air on the surface of water film are constant.
- Heat flux transferred from surrounding is neglected.
- Water-air interface temperature is assumed to be uniform and constant.
- Thermal properties of water and air are constant.
- Air temperature changes only in the flow direction.

According to published paper, the correlation between cooling efficiency can be expressed as follows:

- Correlation of cooling efficiency (η):

$$\eta = 1 - \exp\left(-\alpha \frac{\delta}{V^{0.35}}\right) \qquad \left(\text{Where, } \alpha = \frac{A.\xi}{\rho_a . C_p}\right)$$

<u>Nomenclature</u>:

δ	Width of pad material
α	Material constant
V	Inlet air velocity
ξ	Porosity coefficient per unit pad volume
ρ_a	Density of air
C_p	Specific heat of air

As per the equation, the efficiency can be directly expressed in term of Geometrical properties of cooling media. To find the outlet properties of cooling media we can compare this correlation with classical saturation efficiency equation.

The classical Saturation efficiency equation is as follows:

- Saturation Efficiency relation (η):

$$\eta = \frac{t_1 - t_2}{t_1 - t_s} \times 100\%$$

Where,

t_1	Inlet air dry-bulb temperature
t_2	Outlet air dry-bulb temperature
t_s	Inlet air wet-bulb temperature

If we know the geometric properties of the rigid cooling media, the using first equation we can find the cooling efficiency of the evaporative cooler. Once we have a cooling efficiency of the cooling system and know Inlet properties of air (t1 & ts), then by comparing efficiency in the second equation, we can easily find outlet air dry bulb temperature.

Using above method, we can easily find the predicted values for outlet dry bulb temperature for the Rigid media evaporative cooling system. In this study, the efficiency correlation equations are developed for CELdek 7090 & GLASdek7079 rigid media. The material constants and porosity values are taken from manufacturers data sheet.

2.2 Analytical Results

The Analytical model can predict Cooling efficiency, Downstream RH & Downstream temperature with varying Pad thickness and Upstream velocity. First, keeping pad thickness constant at 100mm, Inlet Temperature at 32°C (305 K) and changing upstream velocity from 0.5 - 5 m/s, It can be observed from the above graphs that Cooling efficiency and Relative Humidity decreases and Downstream temperature increases as the upstream velocity increases.

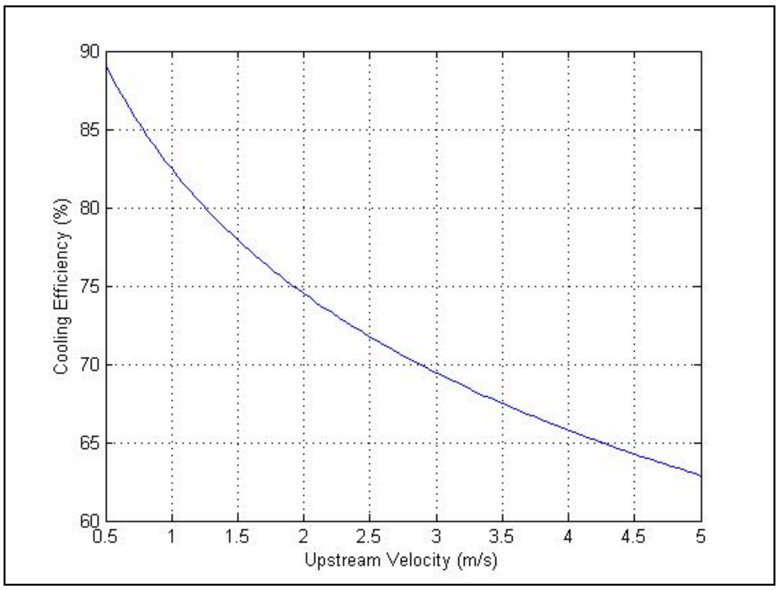


Figure 2-2 Cooling efficiency vs. Upstream velocity

(At constant cooling pad thickness 100mm)

We can see that, from above graph that, as inlet, velocity increases the contact time between water and air decrease and evaporation reduces. Hence the cooling efficiency decreases as the upstream velocity increases. So the results show us that if we control the inlet velocity of the air, we can control cooling efficiency. The second graph illustrates the change in Relative Humidity along with the change in upstream velocity. As the upstream velocity increases the Relative Humidity decreases.

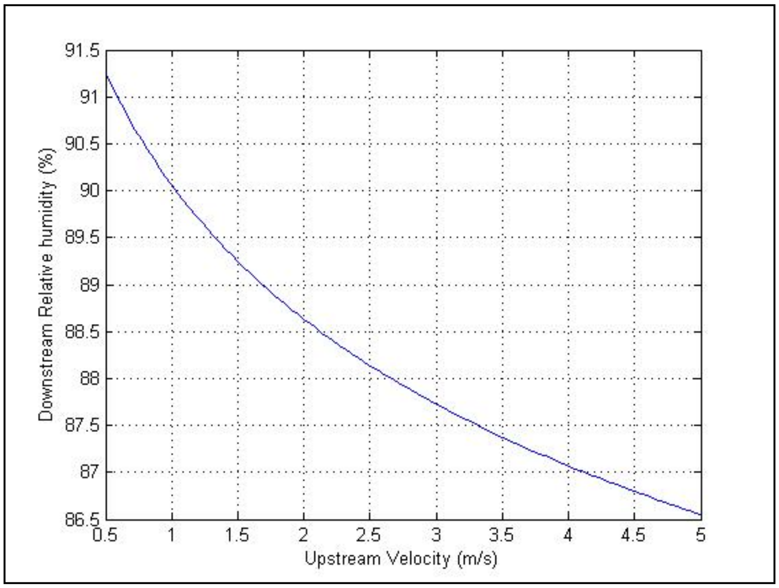


Figure 2-3 Downstream Relative Humidity Vs Upstream Velocity

(At constant cooling pad thickness 100mm)

The third graph shows that the Downstream dry bulb temperature increases as the upstream velocity increases.

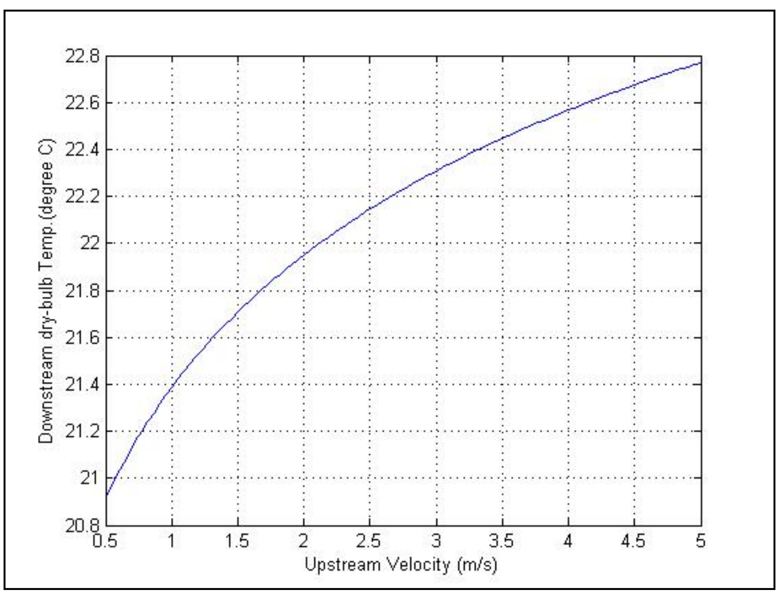


Figure 2-4 Downstream Temperature Vs Upstream velocity

(At constant cooling pad thickness 100mm)

Second, Keeping upstream velocity constant at 2 m/s, Inlet Temperature at 32°C (305 K) and varying pad thickness from 50 – 300 mm, It can be observed that the Cooling efficiency and Relative Humidity increases and Downstream temperature decreases as the pad thickness increases.

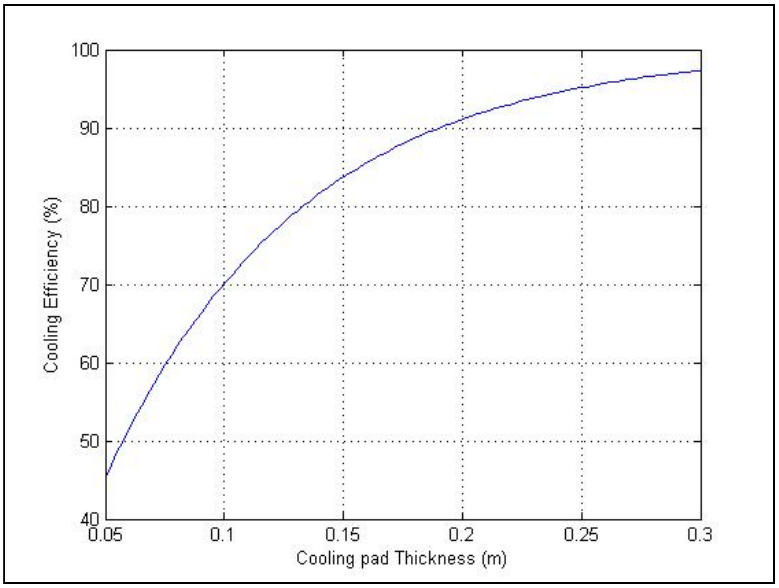


Figure 2-5 Cooling efficiency Vs Cooling Pad Thickness

(At constant upstream velocity 2 m/s)

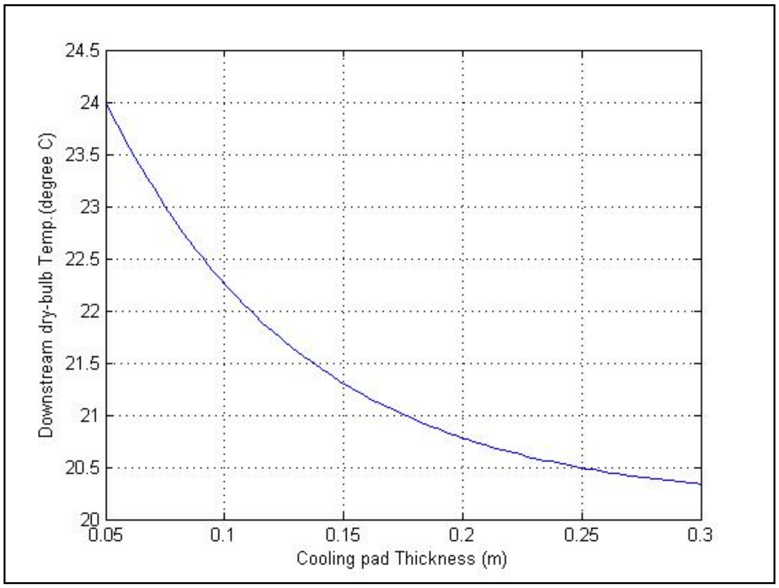


Figure 2-6 Downstream Temperature Vs Cooling Pad Thickness

(At constant upstream velocity 2 m/s)

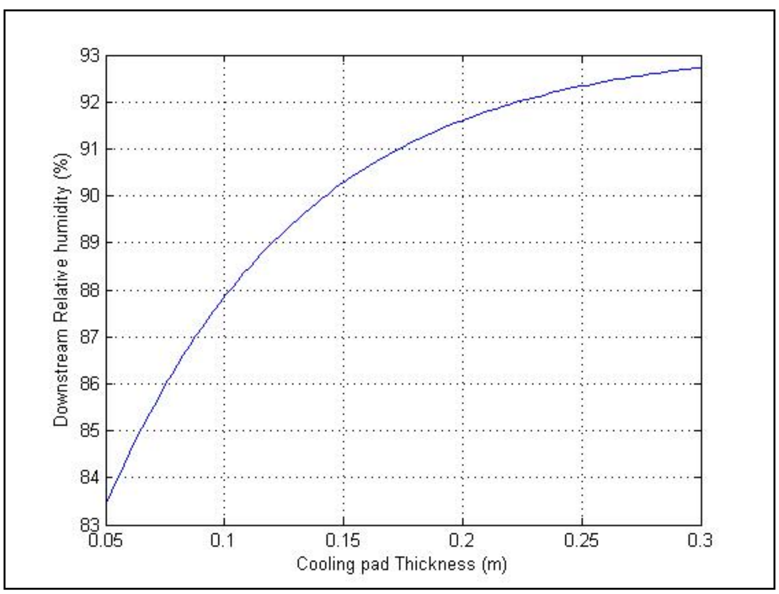


Figure 2-7 Downstream Relative Humidity Vs Cooling Pad Thickness

(At constant upstream velocity 2 m/s)

2.3 Experimental Testing

Fig 2.3-1 shows the experimental setup. In Figure 2.3-1 shows a three ducts of size (0.61 m x 0.61 m x 1.83 m) attached to an airflow bench. The inlet and outlet ducts have Dwyer A-302F-A pressure sensor mounted on them to measure static pressure. A plastic egg crate light diffuser is mounted on inlet and outlet duct. RF Code R155 tags Temperature Humidity sensors are mounted on each of these diffusers. (see Figure 2.3-3). The DEC media is placed in the center section. It also has water reservoir, water pump, water distribution header, and water level regulator. Figure 2.3-2 shows the center section of the duct, the DEC media, the reservoir (sump), the pad for water distribution and the water supply.

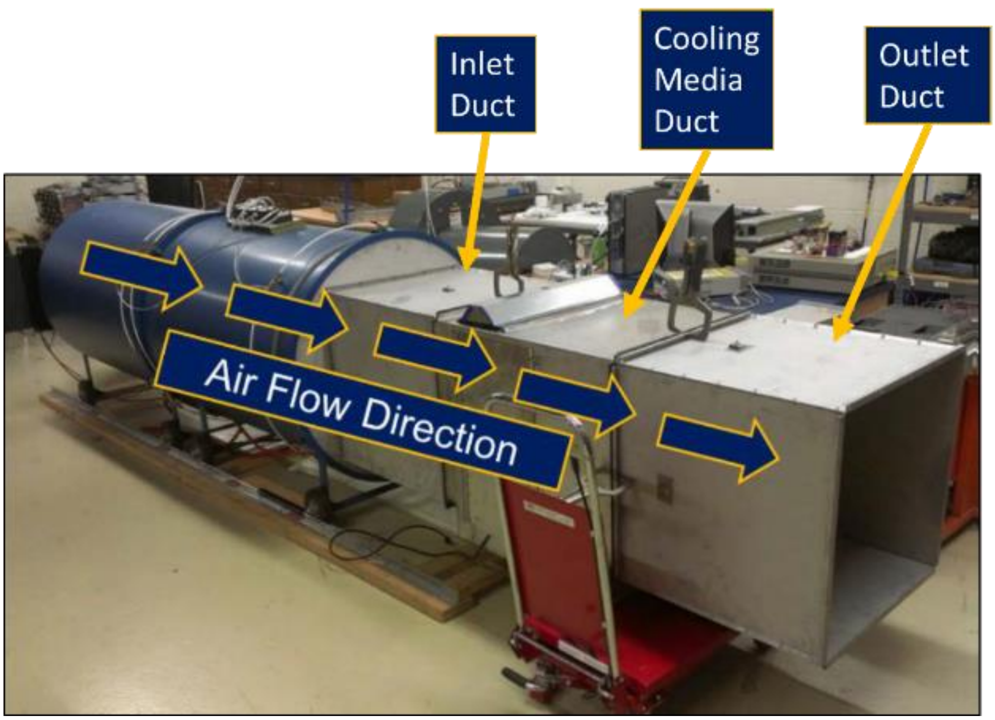


Figure 2.3-1 A three section rectangular duct attached to an airflow bench [10]

The test setup facility is available in UT Arlington, Wolf Hall, 109 facilities. The air flow direction is from left to right.

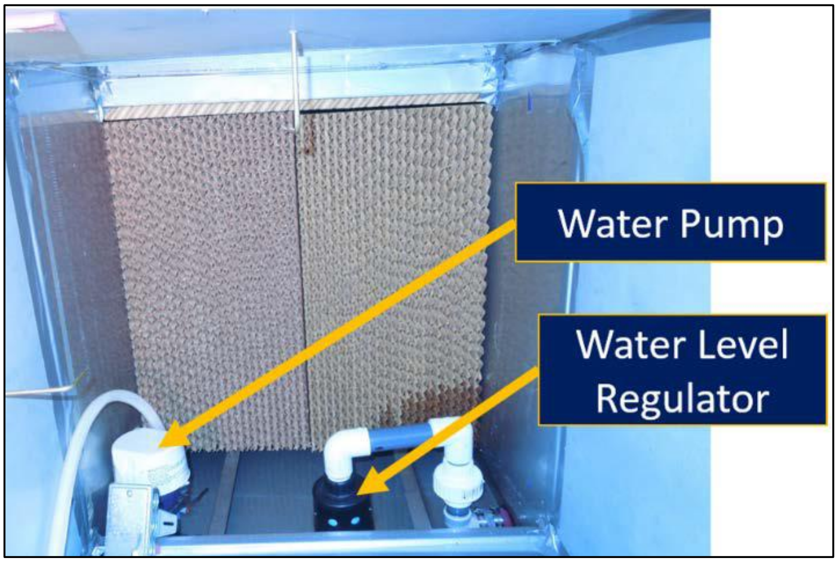


Figure 2.3-2 Water level regulator, water pump, and pressure taps

downstream of the duct [10]

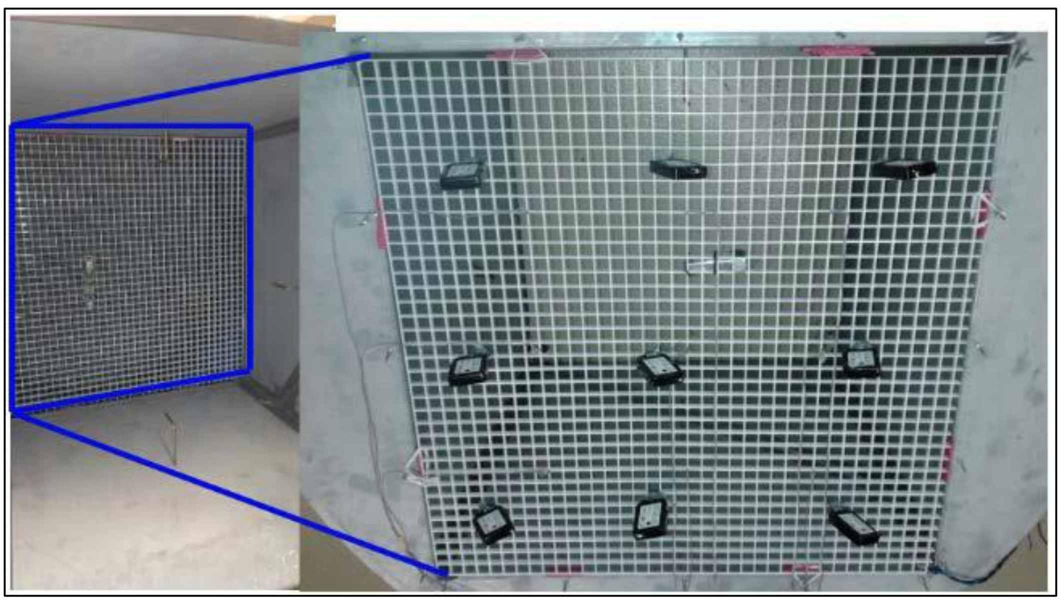


Figure 2.3-3 Humidity-temperature sensors

(Figure adapted, with permission, [8] © 2015 IEEE)

2.4 Experimental Results

The figure 2.4-1 shows, the Upstream and Down Stream Temperature reading over the time. The upper lines show the upstream temperature, which is at higher temperature. While the lower lines represent the downstream temperature which are low temperature compared to upstream air. The red circled area represents the time require to full wet media. It takes about 15 mins thoroughly wet media and then actual cooling starts.

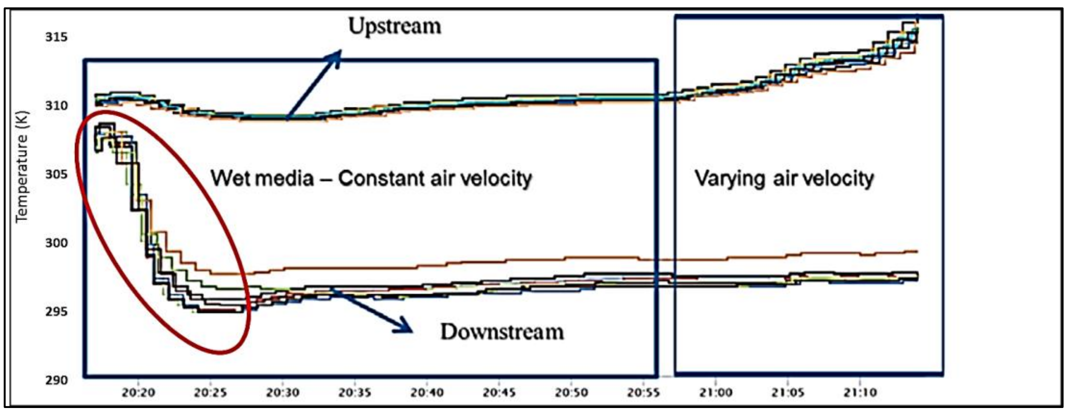


Figure 2.4-1 Temperature readings upstream and downstream of cooling media over time. (Figure adapted, with permission, [8] © 2015 IEEE)

The figure 2.4-2 show the saturation efficiency over the flow rate. The graph indicates that the cooling efficiency is independent of the water flow rate. The evaporation is the surface phenomenon. Even if the flow rate is increased the surface area of the water thin film remains constant. This implies that the evaporation is independent of the water flow rate.

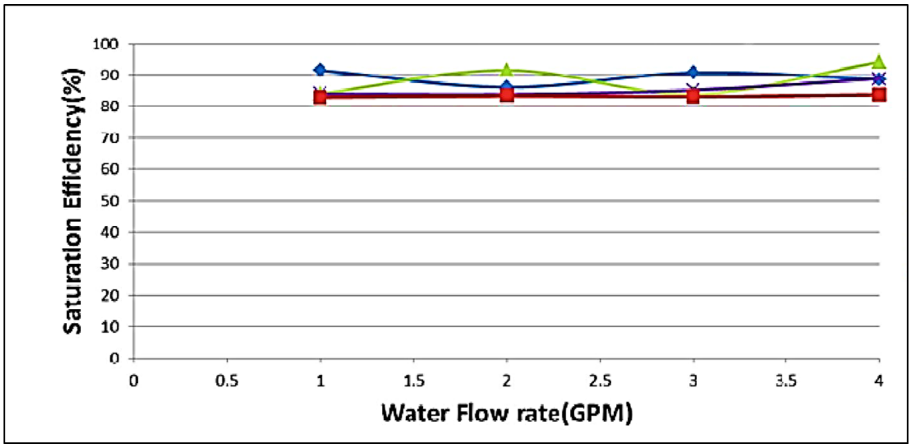


Figure 2.4-2 Saturation Efficiency over Flow rate

Chapter 3 – CFD MODELING OF EVAPORATIVE COOLING PADS

The main aim of this study is to develop the CFD model to predict the performance of the Cooling pads. Before this, to find the performance characteristics the cooling pads needs to be tested experimentally. The experimental tests consume a lot of time and money. The setup takes a couple of days, and the test run takes even more days. So the idea was to develop a computational CFD model so that the performance characteristics can be calculated as very quickly before using it in actual application. This model will also help while developing the new cooling pads. The cost of prototyping will be considerably lower if we use this model during product development.

3.1 Modeling & Geometry

The modeling is done using 3D CAD software in SOLIDWORKS. The dimension is same as the cooling media used in Experimental and Analytical testing. We used Munters CELdek media, some of the parameters are taken from product data sheet. The design parameters are shown as follows:

Modeling: SOLIDWORKS

Media: Munters CELdek Media

Height: 24 in

Width: 24 in

Thickness: 12 in

Air Flute Angle: 45°

Water Flute Angle: 15°

Base Sheet: Cellulose

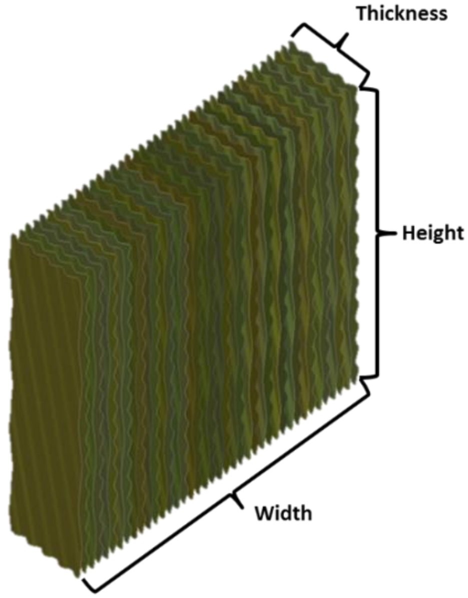


Figure 3.1-1 Cooling Media CAD Model

3.2 CFD Problem Setup & Boundary Conditions

The CFD model is set up in ANSYS Fluent package. The geometry and CAD model is imported using IGES files. The following conditions are used in setting up the CFD model.

a) **Multiphase Flow:** This model is based on Evaporation technique. The evaporation phenomenon involves mixing of two or more phases. The ANSYS Fluent has its own Evaporation model built in it, but we cannot use that model. Because the evaporation that we are trying to model is taking place at almost room temperature. The inbuilt evaporation model requires the saturation temperature values. In our case, the fluids are never reaching the saturation temperature (i.e. In case of water it is 100 degrees Celsius). So the multiphase model is used with Species transport model to define evaporation. To model evaporation parameters, we used the User Defined Function

(UDF) that was provided by the ANSYS. In this model, there are two uniform phases (Air & Water) & One discrete phase (air with water molecules or moist air).

b) **Species Transport:** Fluent can Model the mixing of two species using transport theorem. The Mixture of Water and Air is defined using species transport. (Mixture template - Air and Water).

c) **Eulerian Wall Film (EWF):** The flow of water over the cooling media surface is modeled using Eulerian Wall Film (EWF) model. It is assumed that, When the water flows over the cooling pads, it forms a thin film of water, which is constant over the whole media.

d) **Enhanced Wall Treatment:** The Enhanced Wall Treatment is near wall function. It is used as Boundary wall function with the two-layer approach so that the transition is smoother.

e) **Viscous Model:** The standard K-epsilon two equation turbulence model is used in this problem. It's a two equation model, which means two extra transport equation to represent turbulent properties. It is the most common turbulence model.

f) **General:** It is a transient study, pressure based and energy equation is on.

g) **User Defined Function (UDF):** The User Defined Function is developed with the help from ANSYS. In this model our fluid is never reaching the saturation temperature, so to calculate evaporation the saturation pressure is determined. Also to define inlet properties like Inlet Humidity, Defining Inlet Profile of Moist Air, Calculating Diffusion Coefficient we used UDF.

The Inlet Conditions are as follows:

Inlet Air Temperature - 310 K

Inlet RH of Air - 35%

Velocity – 4 m/s

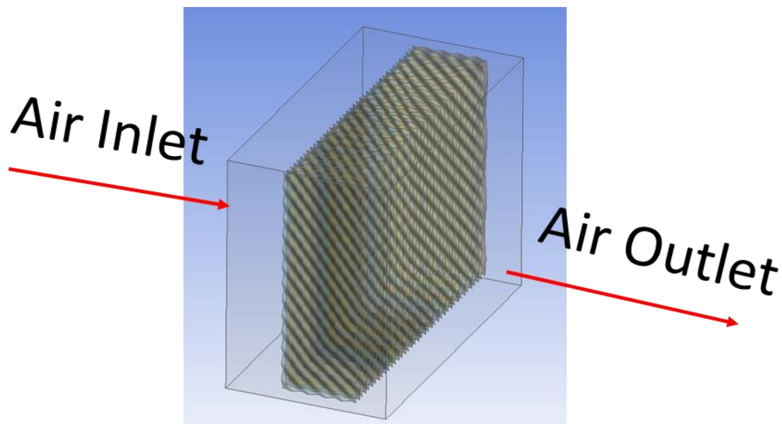


Figure 3.2- 1 Computational Domain

3.3 Results

The Mesh Sensitivity Analysis is done to calculate the accuracy of the CFD model. The pressure drop across media is noted down against the number of nodes. The rest of the boundary conditions are maintained same of the Analysis. The same simulation was run just by changing the number of nodes (i.e. Increasing Mesh count). The figure 3.3-1 shows the Pressure drop against Nodes graph. At about 2.1 million cells the graph starts to converge. The cell number is much higher as compared to the typical CFD simulations that we usually do. Since this CFD model is Multiphase flow, the boundary between two fluids needs to finally mesh. Also, the Geometry of the cooling pads has NURBS (Non-Uniform Rational B-spline Surfaces), and The cooling pad thickness is very thin, the number of cells increases.

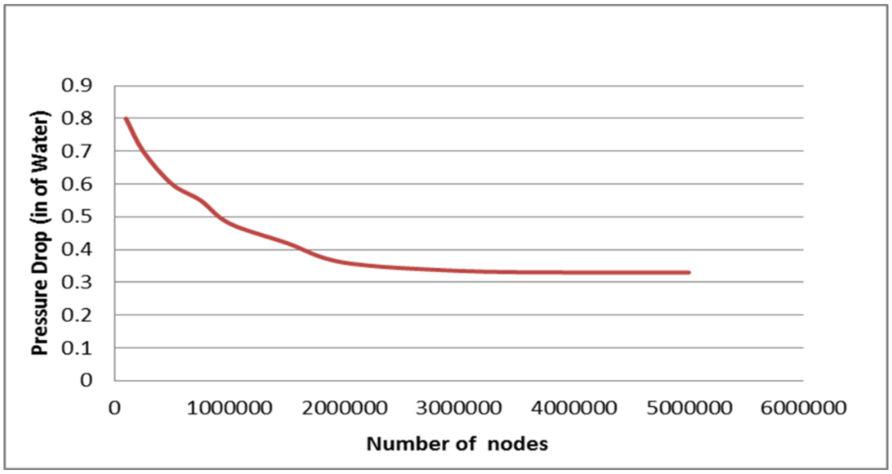


Figure 3.3-1 Mesh Sensitivity Analysis

Figure 3.3-2 shows the Temperature counter obtained from the results. The temperature drop from 310k to 301K is derived from given conditions. The temperature profile shows the decrease along the direction of flow which is as per the expected results.

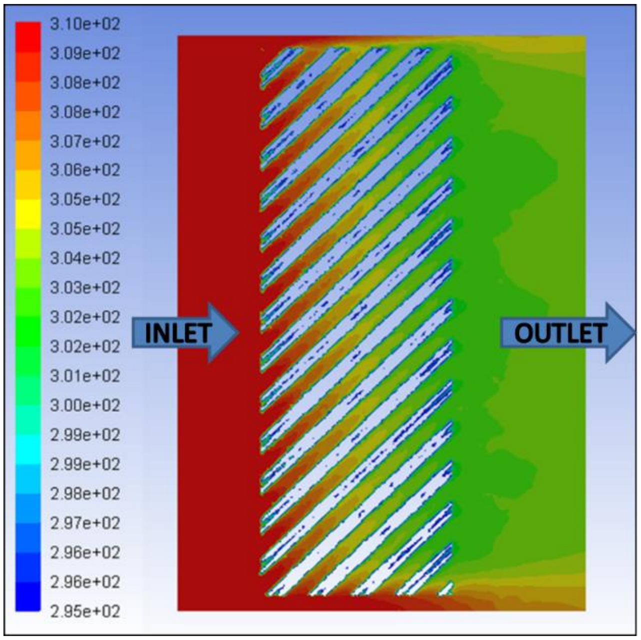


Figure 3.3-2 Temperature contour

The figure 3.3-3 shows the Relative Humidity change across the cooling media. The relative humidity increases along the flow direction. Relative Humidity increases from 35% to 80% as shown in the figure.

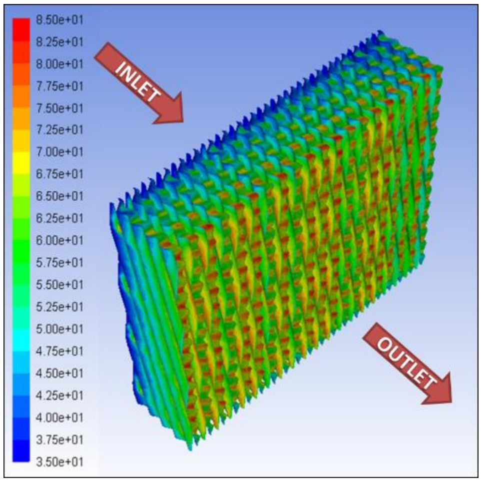


Figure 3.3-3 Relative Humidity Contour - (%RH)

The figure 3.3-4 shows the pressure drop across the cooling media along the direction of flow. As the flow goes on, we can see that the pressure is decreasing, which results in increasing the pressure drop.

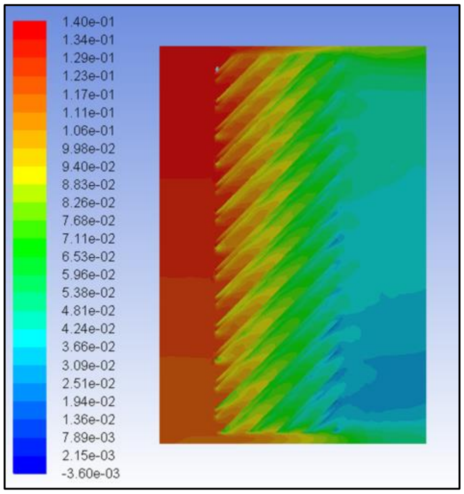


Figure 3.3-4 Pressure change across media

Figure 3.3-5 Shows the system resistance curve from Experimental and CFD results. For the accuracy of the model the validating system resistance is critical. The curve demonstrates that the excellent validation between CFD & Experimental data.

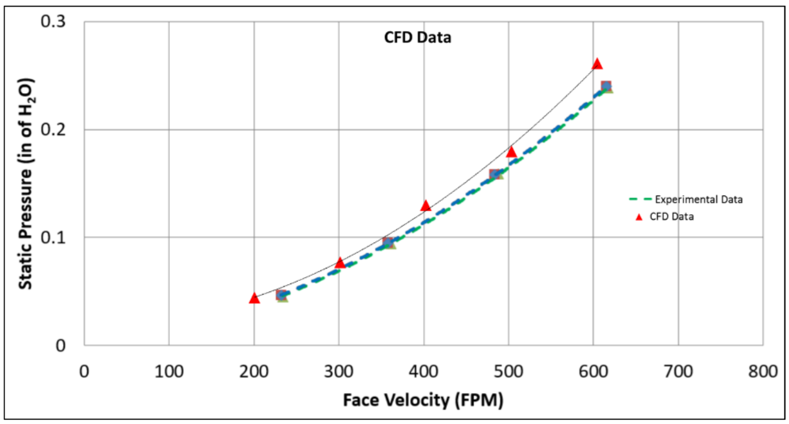


Figure 3.3-5 Pressure Drop (System resistance curve)

3.4 Validation

The table 3-1 shows the experimental results and predicted results from the CFD model and Analytical model. As we can see that the results are very close to all three models.

Temperature in °K						
Case No	Inlet Data	Experimental Result	Predicted Result			
			Analytical	*Error*	*CFD*	*Error*
1	288	281.68	282.14	0.46	280.3	-1.38
2	295	290.52	289.89	-0.63	291.25	0.73
3	308	298.44	298.09	-0.35	299.6	1.16

Table 3-1 Validation & Results

The figure 3.3-6 shows the pressure drop and efficiency curves for different thickness cooling pads. The graph shows that as the thickness increases the contact time between the water and air increases. As the contact, time increases the evaporation increases, hence as the thickness increases the cooling efficiency increases. Also, as the thickness increases the pressure drop also increases. So optimizing the pressure drop and efficiency is a must, otherwise, we will end up consuming lot of power unnecessarily. The next chapter discusses the different ways to optimize efficiency and pressure drop.

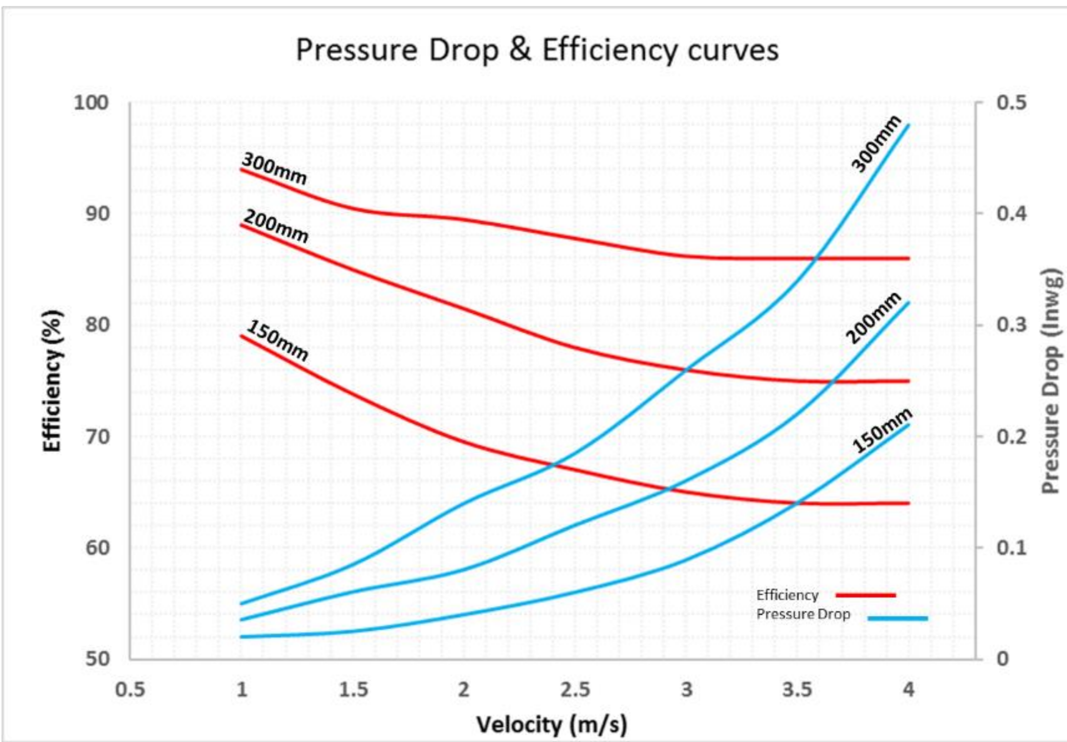


Figure 3.3-6 CFD Results - Different Thickness

Chapter 4 – COOLING MEDIA OPTIMIZATION

The evaporative cooling is used in various industries. The systems must operate in different conditions at different times. The cooling needs are never constant. The cooling requirements can change according to seasons, Day & Night and according to load. The cooling system must adapt to these changes. Also, In the data center industry, the IT equipment must be maintained at a certain level of humidity and temperature. These temperature and humidity needs vary very quickly. The cooling system needs to be optimized according to these requirements. That's why new technique called staging is developed.

4.1 Staging of Evaporative Cooling Pads

Staging of Cooling Media

Cooling media is divided into multiple vertical sections or Staking two or more section together, and separate water distribution headers control each section. Various configuration of vertically or Series staged media may be achieved by dividing the full width of the media into two, three, four or more number of equal sections and providing individually controlled water distribution headers.

Example: For three stages, if the media is divided into two equal sections, the media is either 0%, 50% or 100% wet at any given time.

In the data center, cooling requirement changes according to IT load and outside environment. (e.g. Day-night, winter-summer, sunny-cloudy) So the Precise control over humidity and temperature is needed. Staging can also reduce total water and electricity consumption of the system, which eventually reduces overall Power Usage Effectiveness (PUE). Life of cooling media can be increased using staging since Staging reduces scale build up on DEC media.

Advantages of using Staging:

a) It can provide precise control over Humidity & Temperature.
b) It can reduce Electricity & Water consumption.
c) It can increase the life of cooling media.
d) It can provide better control over Cooling System.

Different types of Staging:

1) **Split Distribution:** The figure 4.1-1 shows [9] the representation of the split distribution system. These systems allow water to drain out quickly. The service life of cooling media increases using this type of staging. It is advised to use these kind of system for the small application.

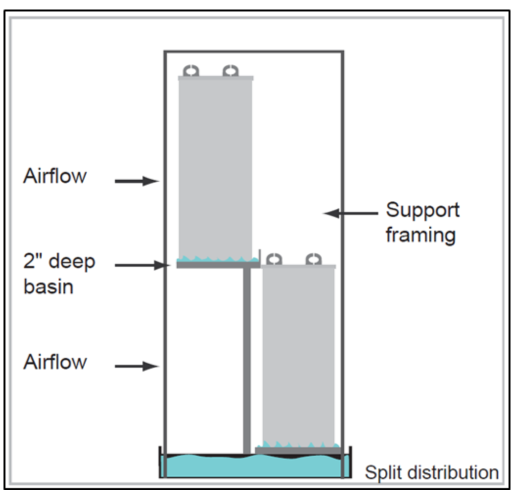


Figure 4.1-1 Split Distribution System [9]

2) **Vertical split:** Figure 4.1-2 [9] shows the vertical split system. The media is vertically divided into multiple sections, and separate water distribution header controls each section. This system allows better control over humidity and temperature. According to temperature and humidity need, each section is activated. It reduces water and electricity consumption when less cooling is required.

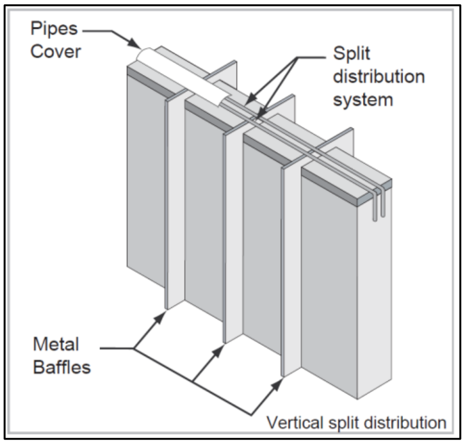


Figure 4.1-2 Vertical Split System [9]

3) **Series split:** Figure 4.1-3 [9] shows the Series arrangement. The cooling media is arranged in series. This system allows higher efficiency & Increases pressure drop across the whole system.

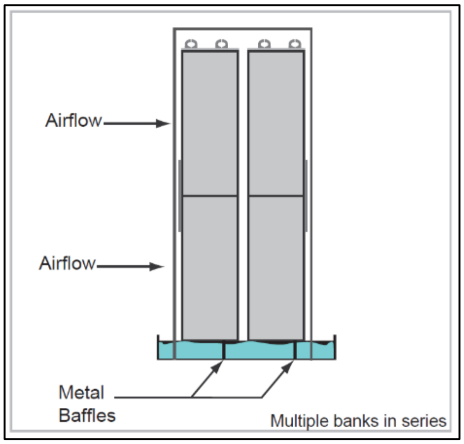
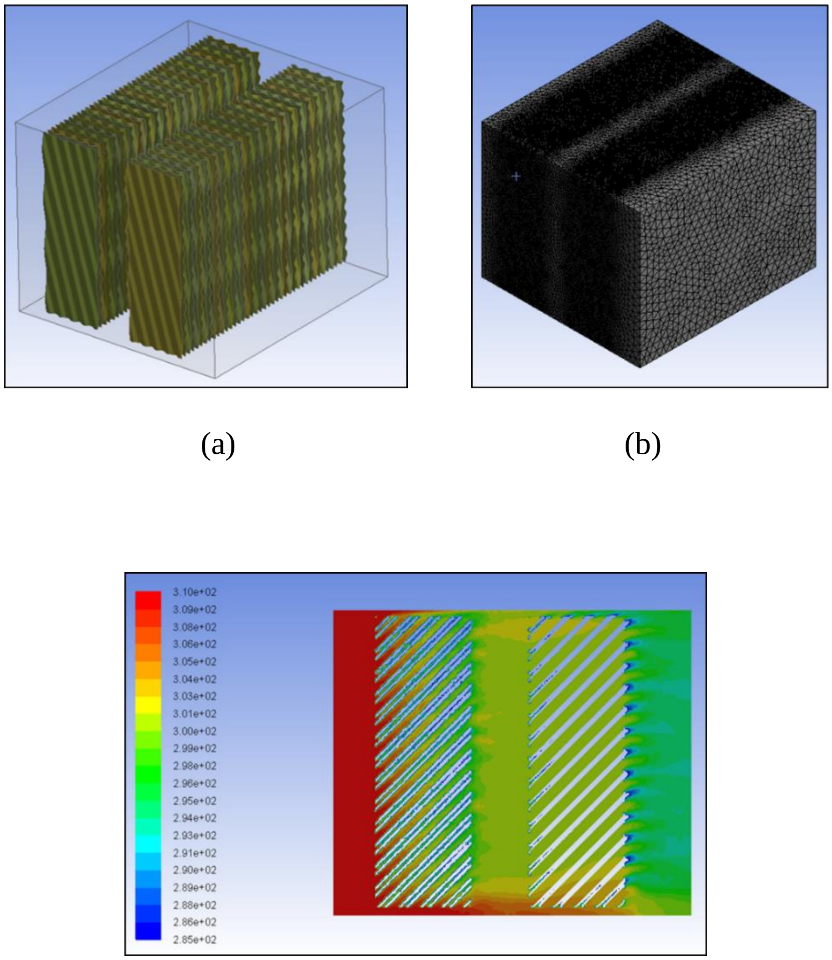


Figure 4.1-3 Series Split System [9]

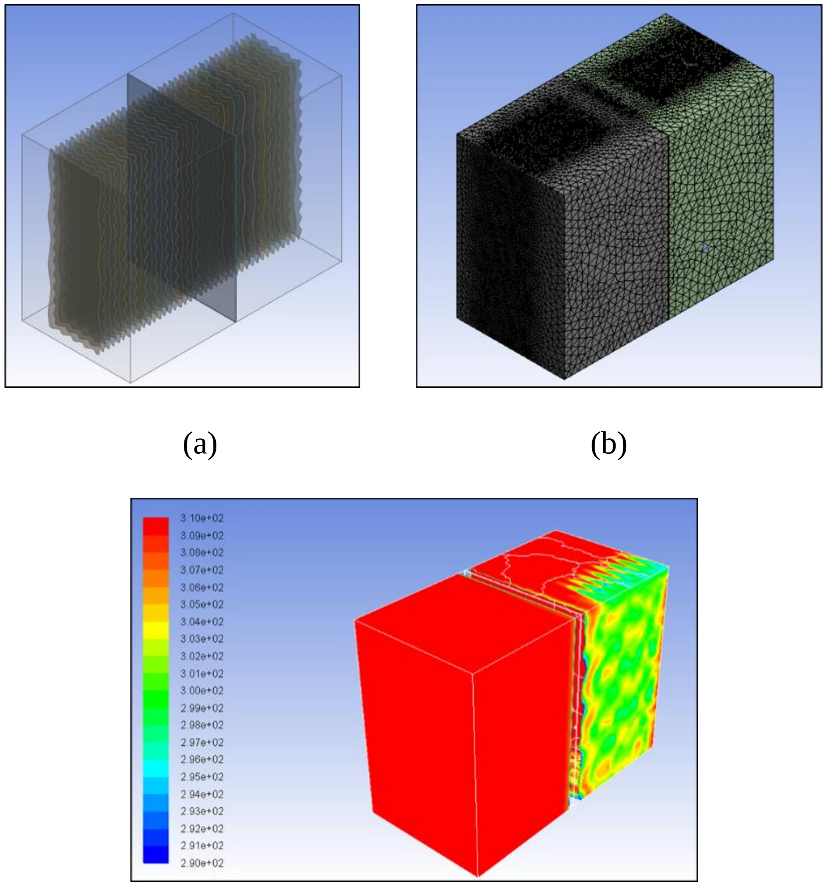
Some of the models and results are shown below using CFD model developed earlier. This indicates that the CFD model developed can be utilized for different models.



(a) (b)

(c)

Figure 4.1-4 Ansys Series Split system (a) 3D CAD Model (b) ANSYS Meshing (c) Temperature contour



(a) (b)

(c)

Figure 4.1-5 Ansys Vertical Split system (a) 3D CAD Model (b) ANSYS Meshing (c) Temperature contour

4.2 Flute Angle Optimization

Flute angles are the angles of the water and air channels with respect to air flow directions. As shown in the fig below, the flow direction is horizontal from left to right. The water channels are shown in blue color, and air channels are shown in brown color.

The flute angle optimization study is done using comparative study. The different models are built using SOLIDWORKS for models (30X30, 45X45, 30X60 & 45X30). The Stiff water angles mean water drains out quickly. The time requires to wet media is less. The fewer air channels angle means less pressure drop. But depending upon application the Air and Water angle is chosen. The CFD model is used to calculate the pressure drop and efficiency curves to compare different models with different flute angles. The figure 4.2-2 & 3 shows the results for the different models. Depending upon the application we can choose appropriate media with correct flute angle to optimize efficiency and pressure drop.

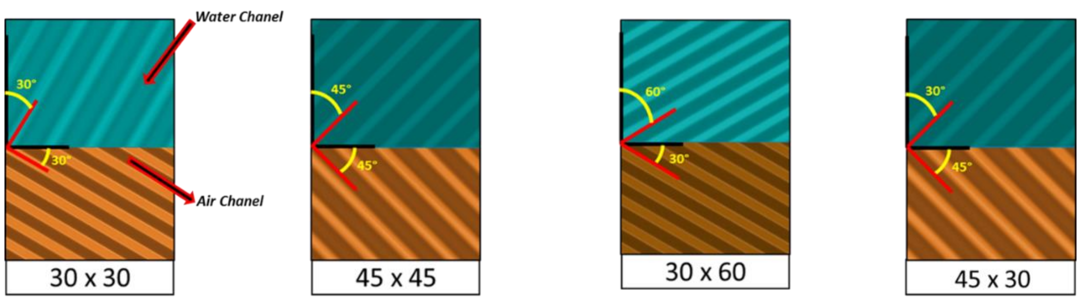


Figure 4.2-1 Flute angles in Rigid Media

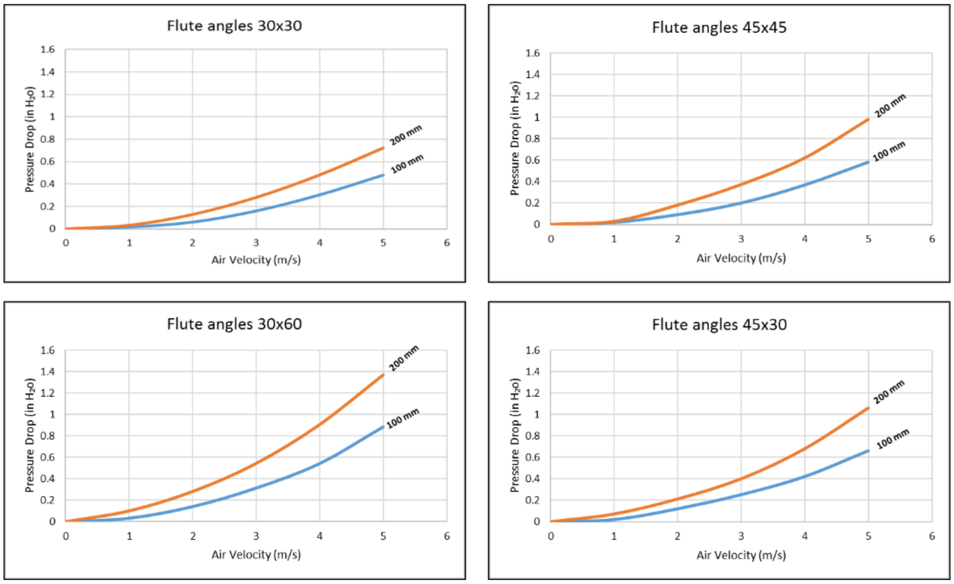


Figure 4.2-2 Pressure Drop Curves for Different flute angles

(Pad thickness 100 & 200 mm)

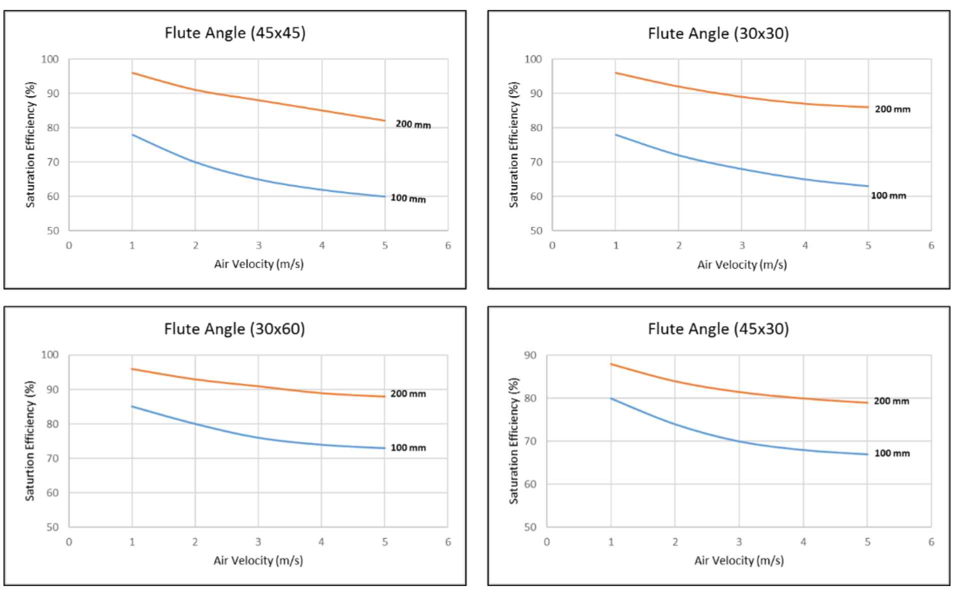


Figure 4.2-3 Efficiency Curves for Different flute angles

(Pad thickness 100 & 200 mm)

Chapter 5 – DISCUSSIONS, CONCLUSION & FUTURE WORK

In this study, the computational study and Optimization of Wet Cooling Media is conducted using simulation technique in ANSYS Fluent. The validation shows that the model can easily predict the performance of Evaporative Cooling System.

5.1 Conclusion

- This CFD modeling technique of Evaporative cooling can help predict the performance of evaporative pads.
- This Study enables us to understand the effect of Staging and Flute angles on the efficiency of cooling pads before experimenting them, which eventually save a lot of time and money.

- For Staging of Cooling Media:

  Vertically Split → Good control over Humidity and Temperature can be achieved using this method. This approach can also reduce water consumption.

  Series Split → This method is useful for low velocity and high-efficiency applications.

- For Cooling Media with different flute angles:

  45x45 → Suitable for use where a right balance between efficiency and pressure drop is required.

  30x30 → Ideal for applications where air crossing speed is high and stiff water angle avoid the release of water droplets.

5.2 Future Work

- Experimentally testing various types of staging of Cooling Media.
- Record and compare CFD results with experimental results.
- Experimentally testing Cooling media with different flute angles for further validation.
- To find out a numerical relation between flute angle and pressure drop for geometric optimization.
- Create a design using this CFD model for Less pressure drop and better efficiency.

Appendix A

ANSYS Code – User Define Function (UDF)

Calculating Diffusion Coefficient:

```
#include "udf.h"
static real Psat_of_temp(real t);
static real op_pressure=101325.0;
static real Mwater=18.0;
static real Mair=28.8;
/* relative humidity of the incoming air, 0.5 indicates 50
% */
real inlet_phi = 0.5;
DEFINE_DIFFUSIVITY(watervapor_diffusivity, c, t, i)
{
  real pressure, Tcell, diff_coeff;
  pressure = op_pressure + C_P(c,t);
  Tcell = C_T(c,t);
  diff_coeff = (0.926*pow(10,-
6.0)/(pressure*0.001))*pow(Tcell,2.5)/(Tcell+245.0);
  return diff_coeff;
}
```

Defining Inlet Profile of Moist Air:

```
/* The following profile sets the water vapor mass fraction
at the inlet based on T and p */
DEFINE_PROFILE(inlet_water_vapor,thread, i)
{
  real Psat, Pwater, Pair, rr, omega, inletrhum;
  face_t f;
  if(!Data_Valid_P()) return;
#if !RP_HOST
  begin_f_loop(f, thread)
    {
      Psat = Psat_of_temp(F_T(f,thread));
      inletrhum = inlet_phi;
      Pwater = Psat * inletrhum;
      /* Corrected Pair - missing F_P ! */
      Pair = F_P(f,thread)+ op_pressure - Pwater;
      rr = Mwater/Mair;
      omega = rr * Pwater/Pair;
      F_PROFILE(f, thread, i) = omega/(1.0+omega);}
  end_f_loop(f,thread);
#endif  }
```

Calculating Saturation Vapor Pressure:

```
/* Calculate saturation pressure at a given temperature */
real Psat_of_temp(real T_Kelvin)
{
  real n4, n3, n2, n1, c0, c1, c2, c3, c4, c5;
  real Tmin, Tmax, T, i, saturation_pressure;

  n4 = -0.0000154539;
  n3 = 0.0014359522;
  n2 = 0.0110544681;
  n1 = -0.0111461694;
  c0 = 0.6141199339;
  c1 = 0.0440621262;
  c2 = 0.0014401567;
  c3 = 0.0000268954;
  c4 = 0.0000002720;
  c5 = 0.0000000028;

  Tmin = 273.16;
  Tmax = 368.0;
if(T_Kelvin < Tmin)
    T_Kelvin = Tmin;
  if(T_Kelvin > Tmax)
    T_Kelvin = Tmax;

  /* Convert temperature from Kelvin to Celsius scale */
  T = T_Kelvin - 273.15;
  if(T < 1.0)
    {
      saturation_pressure = 611.65 + 49.31*T;
    }
  else
    {
      /* Calculation saturation pressure in Kpa */
      i=1.0/(T+0.001);
      saturation_pressure =
T*(T*(T*(T*(T*(c5)+c4)+c3)+c2)+c1) + c0
     + i*(i*(i*(i*(n4)+n3)+n2)+n1);
      /* Convert saturation pressure from Kpa to Pa */
      saturation_pressure = saturation_pressure*1000.0;
    }
 return saturation_pressure;}
```

Appendix B

Matlab Code – Analytical Calculation for Efficiency Correlation

```
%% STAGING OF COOLING MEDIA

%% Nomenclature:
% B - Width of pad module (m)
% Cp - Specific Heat of Air (J/kg.K)
% F - Surface Area (m^2)
% fa - Air humidity ratio (Kg water vapor/kg dry air)
% fs - Saturated air humidity ratio (Kg water vapor/kg dry
air)
% Ga - Air mass flow rate (kg/s)
% H - Hieght of pad module (m)
% hc - Convective heat tranfer coeff. (W/m^2.K)
% hm - Convective mass trasfer coeff. (Kg/m^2.s)
% L - Char. dimension of air channle in pad moldule (m)
% Le - Lewise Number
% Nu - Nusselt Number
% Pr - Prandlt Number
% Re - Reynolds Number
% r - Latent heat of water vapourization (J/kg)
% ta - Air dry bulb temp (C)
% ts - Air wet bulb temp (C)
% tw - Water film temp (C)
% V - Air frontal vel of pad module (m/s)
% alpha & Beta - Integrated factors
% aelta - Pad thickness (m)
% phiq - Latent heat transfer rate (W)
% phix -  Sensible heat transfer rate (W)
% eff - Cooling efficiency
% rhoa - Air density (Kg/m^3)
% nu - Kinematic viscosity (M^2/s)
% sigma - Pore surface coeeficient per unit pading volume.
(m^2/m^3)
% (** Pore surface coeff is taken from product
handbook_Munters corporation)


%% Efficiency Correlation

% Varriying Velocity
%V=linspace(1,4,100);
```

```
V=2;
A=42.9; % Material constant taken from munters corporation
handbook
rhoa=1.225;
Cp=1000;
sigma=440;
delta=0.138;
alpha=((A*sigma)/(rhoa*Cp));
eff=1-(exp(-((alpha*delta)./(V.^0.35))));
% figure
% plot(V,eff)



%% Calculation of Leaving air dry-bulb temperature
t1=27.21; %entering dry-bulb temp
ts=20.15; %entering wet-bulb temp
t2=t1-(eff*(t1-ts));
% figure
% plot(V,t2)



%% Calculation of Specific Humidity Ratio
Pt=1.0132*10^5; % Outside atm Pressure
Pvswb=0.0317*10^5;  %Partial pressure of water vapour at
wet-bulb temp (using steam table)
Pv=Pvswb-(((Pt-Pvswb)*(t1-ts))/(1527.4-1.3*ts)); % Partial
pressure of water vapour at inlet
W=(0.622*Pv)/(Pt-Pv); % Specific humidity of inlet air
h=(1.005*t1)+W*(2500+1.88*t1);  % Process enthalpy
W1=(h-(1.005*t2))./(2500+1.88*t2); % Specific Humidity of
outlet air
Pv1=((W1./0.622)*Pt)./(1+(W1./0.622));  % Partial pressure
of water vapour at outlet
Pvs2=0.0249*10^5;
Rh=Pv1/Pvs2; % Inlet Relative humidity
% figure
% plot(V,W)
% figure
% plot(V,W1)
```

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[10] Thesis Dissertation (PhD - Betsegaw Kebede Gebrehiwot, UTA)

Biographical Information

Tejas Vijay Bhongale (Feb 16th, 1990) is from Satara, India. He completed his Bachelors in Mechanical Engineering from Modern Education Society's College of Engineering, Pune in June 2012. Tejas worked as a Quality Assurance and Customer Support Engineer from February 2013 to November 2014 at TRW Sunsterring Wheels, Pune. He joined The University of Texas at Arlington in Spring 2015 for Master of Science program in Mechanical Engineering. Tejas worked as Graduate Research Assistant for Biomedical Technologies Division from September 2015 to December 2016 at The University of Texas at Arlington Research Institute (UTARI), Fort Worth, TX. He joined EMNSPC team in Fall 2015 and started working on various projects. Tejas graduated from The University of Texas at Arlington on November 28, 2016.