GENERAL METHODS FOR THE THERMAL DESIGN OF DATA CENTERS

BASED ON EVAPORATIVE COOLING

by

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THESIS

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ABSTRACT

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BASED ON EVAPORATIVE COOLING

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With the increasing load on the servers, the cost and energy required to cool a data center has been on the rise. This has kept the researchers to explore more efficient and economical cooling technologies for data centers. One such technology is evaporative cooling. The evaporative cooling systems are one of the most effective methods available for cooling the data centers. So, developing the generic methods for the thermal design of data centers is of utmost importance. This research aims to identify, develop and validate the thermal model for Direct Evaporative Cooling and Indirect Evaporative Cooling. By taking the environmental data of each location and server rack specifications as inputs, the performance and cost analysis are done. Strategies to cope with system component failures have also been considered by the model. A tool, that works on the identified methods has been developed. The tool gives the performance results, the cost analysis and a comparison between direct and indirect evaporative cooling, allowing the user to make decisions on the type of cooling that can be used for specific environmental conditions.
TABLE OF CONTENTS

ACKNOWLEDGEMENTS ........................................................................................................... i

ABSTRACT ................................................................................................................................. ii

LIST OF FIGURES ...................................................................................................................... vi

LIST OF TABLES ....................................................................................................................... vii

Chapter 1 INTRODUCTION ....................................................................................................... 1

1.1 Data Centers – An Overview .............................................................................................. 1

1.2 Data Center Cooling ............................................................................................................. 2

   Computer Room Air Conditioning (CRAC): ........................................................................ 3

   Air Side Economization (ASE): ......................................................................................... 4

   Evaporative Cooling: ........................................................................................................... 4

1.3 Why Evaporative Cooling .................................................................................................. 5

1.4 Scope of the work ............................................................................................................... 8

   Literature Review & Motivation ......................................................................................... 8

   Objective of the research .................................................................................................... 9

Chapter 2 PSYCHROMETRY .................................................................................................. 10

2.1 Psychrometric Terminologies ............................................................................................. 10

2.2 Psychrometric Chart ........................................................................................................ 11

Chapter 3 METHODOLOGY FOR THERMODYNAMIC MODEL ........................................... 13

3.1 Direct Evaporative Cooling ................................................................................................ 13
Chapter 9 References ................................................................. 39
APPENDIX A .................................................................................. 42
BIOGRAPHICAL INFORMATION ....................................................... 73
LIST OF FIGURES

Figure 1-1: Modular Data Center ................................................................. 1
Figure 1-2: Oil immersion cooling ............................................................... 2
Figure 1-3: CRAC system ........................................................................... 3
Figure 1-4: Air Side Economizer ................................................................. 4
Figure 1-5: Evaporative Cooling ................................................................. 5
Figure 1-6: Energy Consumption Breakdown ............................................ 6
Figure 1-7: Thermal Guidelines as per ASHRAE TC 9.9 .............................. 7
Figure 2-1: Psychrometric Processes .......................................................... 11
Figure 2-2: Cooling Processes ................................................................... 12
Figure 10: Direct Evaporative Cooling ....................................................... 14
Figure 11: Thermodynamic Process ............................................................ 14
Figure 12: Indirect Evaporative Cooling ..................................................... 18
Figure 13: Thermodynamic Process ............................................................ 19
Figure 14: Tool layout- initial .................................................................... 23
Figure 15: TMY3 Data for DFW ................................................................. 26
Figure 16: Results in tool........................................................................... 33
Figure 17: PUE comparison between DEC & IEC .................................... 34
Figure 18: Water Consumption for DEC .................................................... 34
Figure 19: Water Consumption for IEC ...................................................... 35
Figure 20: Temp. vs Water consumption - DEC ...................................... 36
LIST OF TABLES

Table 5-1: Inlet temperatures and ΔT ................................................................. 24
Table 6-1: Results for DEC - DFW .................................................................... 27
Table 6-2: Results for IEC-DFW ....................................................................... 28
Table 6-3: DEC- Portland .................................................................................. 29
Table 6-4: IEC-Portland .................................................................................. 30
Table 6-5: DEC-Minneapolis .......................................................................... 31
Table 6-6: IEC-Minneapolis .......................................................................... 32
Chapter 1

INTRODUCTION

1.1 Data Centers – An Overview

Data Centers are facilities that house IT equipment used to perform functions like data processing, storage, transmission and enabling swift access to the data. The IT equipment responsible for performing these operations are known as servers, which are the core for the Internet of Things. These servers are stacked on top of each other in the form of server racks. A data center can occupy anywhere from one room to a whole level of a building. The servers are stacked in several racks arranged in single rows forming aisles. Figure 1 shows a modular data center. Modular Data Centers are the modern systems of data center deployment that can be placed anywhere. Modular data center systems consist of purpose-engineered modules and components to offer scalable data center capacity with multiple power and cooling options [1].

Figure 1-1: Modular Data Center [2]
1.2 Data Center Cooling

Now, as the data centers are run continuously all the year, a large amount of heat is generated. To keep the servers operating and maintain their life, this heat needs to be removed quickly. Therefore, the cooling of data centers is of utmost importance.

There are various techniques that can be used to cool down the data centers. Firstly, there is liquid cooling in which water or other liquids are sent to the critical components through tubes. This is a highly effective method due to water’s high heat transfer capacity. But this method is very complex and dangerous.

Then we have immersion cooling, where the servers are submerged in a thermally conductive dielectric medium. The dielectric medium is usually oil. Although this method is a potential solution for green data centers [3], there are a lot of changes that should be made to the existing server designs and components. The following figure shows how the oil immersion cooling works.

![Oil immersion cooling diagram](image)

Figure 1-2: Oil immersion cooling [4]
Air Cooling is the most popular and widely used technique, due to its simplicity and flexibility. Air cooling is further divided into different methods.

- **Computer Room Air Conditioning (CRAC):**

  This is a traditional method to cool down the data center. CRAC unit is a device that monitors and maintains the temperature, humidity and air flow distribution inside the data center. The CRAC units work on the refrigeration cycle, and most of the power is consumed by the compressor. The hot exhaust air is sent to the CRAC units where the liquid refrigerant takes up the heat from the hot exhaust air. It is then compressed inside a compressor. The compressed evaporated refrigerant is condensed in the cooling tower and is sent back to the CRAC unit. The water inside the cooling tower takes up the heat from the refrigerant and ejects it into the atmosphere. These units are expensive and are not highly efficient. The following figure shows the working of a CRAC system.

![Traditional Cooling Diagram](image)

Figure 1-3: CRAC system [5]
• **Air Side Economization (ASE):**

Air side economization is being currently used in most of the data centers for being economical and highly efficient. In this method, whenever the ambient air conditions like dry bulb temperature and relative humidity are favorable, the outside air is directly used to cool down the IT room. Mechanical systems like fans could be used when some heating of the outside air is required. One of the drawbacks of this method is the introduction of contaminants present in air to the IT equipment. This will cause the equipment to fail and could lead to shut down. The working of an ASE is explained in the following figure.

![Figure 1-4: Air Side Economizer](image)

• **Evaporative Cooling:**

In this technique, cooling is achieved by evaporation of water. The hot outside air used as inlet to the data center, is introduced to a wet media pad or pipe where the water gets evaporated by taking the heat from the incoming air. This air is then sent to the IT room to cool down the servers. The air at the inlet of the IT room although being low in temperature has high moisture content. This moisture content could lead to the contamination of the
equipment, so counter measures must be taken to reduce the risk. There are two types of evaporative cooling: direct and indirect. A comprehensive study about these is shown in the upcoming chapters. The following figure shows the basic principle of evaporative cooling.

![Figure 1-5: Evaporative Cooling](image)

1.3 Why Evaporative Cooling

The traditional cooling facilities rely on a centralized chiller plant, computer room air handlers and cooling towers for facilitating ample cooling to the IT equipment. All these components involved in the cooling, consume power and it can be seen from the following figure the cooling energy breakdown of all the components involved. It is observed that the compressor work accounts for most of the power consumption.
The effectiveness of a cooling system is measured using a metric defined by Green Grid “Power Usage Effectiveness (PUE)”. It is the ratio of the Total power required by the facility to the IT compute power.

\[
PUE = \frac{\text{Total Facility Power}}{\text{IT Compute Power}} = \frac{\text{IT Compute Power} + \text{Cooling Power} + \text{Misc. Power}}{\text{IT Compute Power}} \tag{1-1}
\]

This tells us how much power is being consumed by other non-cooling activities. Ideally PUE should be 1, but in the real world for a system to be termed as effective the PUE should be close to 1.

As seen from figure 6, the PUE of a chiller plant-based model will be far greater than 1. So, to reduce the cooling energy we need to find new ways that can minimize the chiller work or eliminate it. This is where evaporative cooling can be used. With evaporative cooling, the same amount of cooling will be achieved with lesser cooling energy, thereby giving much better PUE.
Also, as ASHRAE has expanded the allowable envelope, the scope of using evaporative cooling as the primary cooling strategy throughout the year has increased. This will allow us to be more energy efficient for most part of the year.

The ASHRAE’s (American Society of Heating, Refrigeration and Air Conditioning Engineers) TC 9.9 published their first thermal guidelines for data processing environment in which the envelope of operational environmental conditions for IT equipment was specified in 2004 [7]. As per their latest update, the allowable envelopes have been expanded in response to the system designers and operators pushing the thermal limits for increased efficiency. The following figure shows the thermal guidelines published by ASHRAE TC9.9.

Figure 1-7: Thermal Guidelines as per ASHRAE TC 9.9
1.4 Scope of the work

- **Literature Review & Motivation**

With the increase in web-based services and cloud computing there has been a drastic increase in the demand for data centers. With increase in demand the cost of acquisition and leasing of data center facility and servers from vendors has increased. This made organizations and researchers to develop models and strategies that will assist in building their own data center facilities. So, in the recent years, a lot of research has been going on in the field of evaporative cooling. A number of thermal models have been presented by numerous authors and organizations. One of first organizations to design their own data center was Facebook [8] in 2011. For this thermal design Facebook followed the black-state approach and assumed everything will be working at maximum efficiency. They custom designed the model so that it could run on Air Side Economization throughout the year as the weather conditions were favorable at Prinville, Oregon. The thermal models developed were delivered as a part of the Open Compute Project [9].

Thermal models for Direct and Indirect Evaporative cooling were developed and presented as pre-design and design tools by [10]. The pre-design tools give the first estimation of cooling potential and hours of discomfort. More detailed thermal models that go from the chip inside the server to the cooling tower have been developed and presented in [11], [12], [13], [14]. In all these papers, highly detailed models from the chip level to server level to rack level till the cooling tower level have been discussed. Along with the thermal models, influence of server inlet temperatures has been discussed in [11]. Many other factors that have an impact on cooling have been discussed in the other three papers.

Holistic thermodynamic models were developed using system and component physics so that the energy consumption and heat transfer phenomenon inside the data center can be
predicted by [15]. These holistic models helped technologists to understand and identify the amount of energy used for cooling purposes among others. All these thermodynamic models till now discuss only about the heat transfers, energy consumption and effectiveness by the data center. One of the important resources that has not been taken into account is water consumption. Some light was thrown in to this direction by [16], where heat transfer models were prepared to validate the performance of the evaporative cooling pad. After a couple of years in 2015, [17] developed models that consider water consumption as an important factor while calculating the operational costs for the data center.

All the thermodynamic models discussed till now, account for almost all the major components of the data center. These models are specific, or custom made by the researchers and organizations to serve their purposes. This work seeks to identify, develop and establish the general methods to be followed for the thermal design of a data center based on evaporative cooling. All these models will be used to develop a tool that provides performance and cost analysis for each type of evaporative cooling for each geographic location, making it easier for the customer to make executive decisions.

- **Objective of the research**

  1. Identifying and establishing the general methods for the thermal design of a data center while accounting for water consumption.

  2. Adding redundancy to the existing models.

  3. Developing a tool (an application) that gives performance and cost analysis.
Chapter 2
PSYCHROMETRY

This chapter contains all the psychrometric terminologies that will be observed and used throughout this report. The psychrometric chart and its significance in supporting the claim of using evaporative cooling has also been presented in this chapter.

2.1 Psychrometric Terminologies

The definitions have been restated from [18] and [19].

- **Dry Bulb Temperature (DBT):** The temperature of air around us, measured with a normal dry thermometer. It is one of the most important parameters. It indicates the heat content and is depicted by the horizontal axis of the psychrometric chart.

- **Relative Humidity (φ):** It gives the amount of moisture content present in the air. It is the ratio of vapor pressure of moisture to the saturation pressure at the dry bulb temperature.

- **Wet Bulb Temperature (WBT):** It is the temperature measured by wrapping a wet cloth around the thermometer wick and introducing it to air flow. It is represented by the slanted lines on the psychrometric chart.

- **Dew Point Temperature (DPT):** It is the temperature at which the condensation of moisture begins, when the air is cooled at constant pressure.

- **Humidity Ratio (ω):** It is the ratio of mass of water vapor to the mass of air in any given volume of mixture. On the psychrometric chart, it is represented by the vertical axis.

- **Degree of Saturation (μ):** It is the ratio of air humidity ratio to humidity ratio of saturated moist air at same temperature and pressure.
2.2 Psychrometric Chart

The psychrometric chart represents the physical and thermodynamic properties of moist air, graphically. A psychrometric chart is plotted for a particular temperature. If the dry bulb temperature and relative humidity are known for a particular pressure, all the remaining properties can be found out from the chart.

The psychrometric chart with the allowable regions and recommended zones was shown in figure 1-7. So, depending upon the outside air temperature if it’s in the allowable regions, we can decide whether we need to cool down the air, or heat it, or humidify it or dehumidify it or sometimes a combination of two processes to bring the air into the recommended zone. The following figure shows the processes we can do on the outside air to bring it into the recommended zone.

![Psychrometric Processes](image.png)

Figure 2-1: Psychrometric Processes [20]
According to [21], TMY3 data combined with psychrometric bin analysis shows the type of cooling can be effectively used. If the air is outside the allowable regions, then appropriate methods must be used to bring it within the allowable regions as specified by ASHRAE TC 9.9. The following psychrometric chart shows the type of cooling system that can be used for cooling.

Figure 2-2: Cooling Processes

According to psychrometric bin analysis performed by [21], using evaporative cooling using evaporative cooling will give better results than CRAC and Direct Expansion Cooling.
Chapter 3

METHODOLOGY FOR THERMODYNAMIC MODEL

A thermodynamic model has been developed for both Direct and Indirect Evaporative Cooling. Each model is derived based on the first law of Thermodynamics.

\[ Q = mC_p \Delta T \]  \hspace{1cm} (3-1)

Each sub-model has been considered as a black-box model and only heat and mass transfer has been considered. The thermodynamic cost model that accounts for the energy and water consumption costs have been calculated on annual basis. The boundary conditions are assumed to be the same for both the cooling strategies. Quasi - steady state conditions are assumed to exist at the operating points. The weather data for each city has been taken from the TMY3 data [22]. The commercial utility rates for water and electricity have been taken from the available data for each city. It is also assumed that the all the servers are running at the same performance level and at maximum loading conditions. All the fans are equally efficient, and all the pumps are running at their maximum speed. Now that all the assumptions being made have been listed, thermodynamic model for each cooling strategy will be developed.

3.1 Direct Evaporative Cooling

- Principle & Working
In DEC, the water comes into direct contact with the supply air stream through a water spray or wetted cooling media. The fan shown in figure 10 draws the air from outside through the wet media pad and sends the cool air to IT room. A pump is used to keep the media pad wet at all times. The water absorbs the heat from the air and evaporates thereby cooling the air stream.
In DEC, the dry bulb temperature decreases, and humidity increases while the wet bulb temperature remains the same due to adiabatic cooling. The following figures show the working of a direct evaporative cooling system and the process diagram on the psychrometric chart.

Figure 3: Direct Evaporative Cooling [23]

Figure 4: Thermodynamic Process [24]
- **IT Room Model**

  The heat generated inside the IT room will be due to the servers and inefficiencies of the server fans.

  \[ Q_{serv} = P_{serv} - P_{sf} \]  
  \[ (3-2) \]

  As a rack is composed of a number of servers, assuming each rack has the same number of servers, the heat dissipated “\(Q_{rack}\)” from each rack will be same. If the temperature rise between the racks is given by “\(\Delta T_{rack}\)”, then the mass flow rate of air through the rack can be calculated by

  \[ m_a = \frac{Q_{rack}}{C_p \cdot \Delta T_{rack}} \]  
  \[ (3-3) \]

  The inefficiency of the server fans can be determined through fan laws. As this model concentrates to provide the general methods, the server fan inefficiencies are assumed to be minimal, enabling us to consider that the heat generated in the IT room to be the heat generated by the servers alone. This means the total heat generated can be equated to the IT compute power.

  \[ Q_{room} = P_{IT} \]  
  \[ (3-4) \]

  Equation (3-1) can also be used to calculate the rack and IT heat load but as this research aims to give cooling solutions to those who want to setup a data center based on evaporative cooling, exact values for the variables will not be available.

- **Pump model**

  The power requirement for the water pump can be determined by using the pressure drop in the circulating water loop, flow rate required and pump efficiency.

  \[ P_{wp} = (\Delta p_{wp} \cdot v_{wp})/\eta_{wp} \]  
  \[ (3-5) \]

  It can also be calculated using the following equation

  \[ P_{wp} = P_{ref} \times (V_w/V_{ref})^3 \]  
  \[ (3-6) \]
\( P_{\text{ref}} \) and \( V_{\text{ref}} \) can be obtained from the manufacturer’s data. \( V_w \) is the required flow.

- **Supply Fan model**

For the supply fan / blower, redundancy has been introduced. This is to make sure the system keeps working in case of any contingencies. The power required by the supply fan depends on the flow rate that needs to be delivered. According to the fan laws and data from manufacturer, the power for the supply fan is calculated using

\[
P_{Sfan} = n * P_{\text{ref}} * (V_{SfanAir} / V_{Sfref})^3
\]  

(3-7)

- **Water Consumption Model**

The water consumed to cool down the inlet air is calculated using the mass balance equation. The outside air temperature and relative humidity are taken from the TMY3 data. The minimum air exit temperature that can be obtained through the media pad is given by [24] as

\[
t_{\text{min}} = t_d - (SE_{\text{media}} * (t_d - t_w))
\]  

(3-8)

Here “\( SE_{\text{media}} \)” is the saturation effectiveness of the media pad being used. It depends on the material being used varies from 60%-93%. “\( t_w \)” is the wet bulb temperature of outside air. The exit air temperature from the cooling pad is taken as the supply air temperature for the IT room and hence it should be within the operating conditions.

The rate of evaporation is given by product of mass flow rate of air “\( m_a \)” and the difference between the humidity ratio of air before “\( \omega_{a,i} \)” and after “\( \omega_{a,e} \)” the wet media pad.

\[
m_{\text{evap}} = m_a(\omega_{a,e} - \omega_{a,i})
\]  

(3-9)

The set of formulas required to calculate the wet bulb temperature and humidity ratio have been shown in Appendix A.
• **PUE**

PUE is a measure of performance of the cooling system.

The total power consumed in Direct Evaporative Cooling is given by

\[ P_{\text{total}} = P_{\text{fan}} + P_{wp} + Q_{\text{room}} \]  
(3-10)

Now, PUE is calculated using equation (1-1),

\[ PUE = \frac{P_{\text{total}}}{Q_{\text{room}}} \]  
(3-11)

• **Cost model**

The cost of water consumption is determined by calculating the water consumed for the whole year and then multiplying it with the commercial rate of water and adding the base charge for the particular city.

\[ C_{\text{water}} = C_{\text{base}} + (U_{\text{water}} \times \text{Rate}) \]  
(3-12)

Here, “\( U_{\text{water}} \)” is the units of water consumed in gallons per year.

The cost of energy consumption is also calculated in a similar manner. The energy rates vary widely through the state. For the purpose of validation of the case study, commercial rates have been taken for each city.

Detailed formulas for calculation and conversions have been shown in Appendix A.
3.2 Indirect Evaporative Cooling

- **Principle & Working**

In IEC, the water doesn’t come in direct contact with the air stream. A heat exchanger is used to lower the temperature of the inlet air stream. Inside the heat exchanger, a secondary air stream cools down the primary inlet air stream. The secondary air stream is cooled down by the water stream. In this type of cooling, both the dry bulb and wet bulb temperatures of the inlet air get decreased, while the humidity of the air remains the same. This thermodynamic process is known as sensible cooling. In IEC, the return air is mixed with the inlet air to achieve the target inlet temperatures. Although this method does not increase the humidity of the primary air stream, which is beneficial. But when compared to DEC, it costs more and is less efficient. The main components of an indirect evaporative cooling are supply fan that brings the primary air into the system, a cooling tower that acts as the heat exchanger and the IT room. The following figure shows the working of IEC.

![Figure 5: Indirect Evaporative Cooling](image-url)
The following figure shows the thermodynamic process of sensible cooling. As there is no humidity being added in this process, the process line moves straight to the left as seen in the figure.

![Thermodynamic Process](image)

Figure 6: Thermodynamic Process [26]

- **IT Room Model**

The IT room model for indirect evaporative cooling is similar to that of direct evaporative cooling as there will be no change in the setup of the IT room.

Hence, the heat load of the IT room is equal to the power requirement of the IT room.

\[ Q_{room} = P_{IT} \]  \hspace{1cm} (3-13)
**Supply fan model**

The model for the supply fan for primary air will be similar to the model used in direct evaporative cooling. Redundancy is again introduced here and the equation for power consumed by the supply fan is

$$P_{Sfan} = n \times P_{ref} \times \left( \frac{N_{SfanAir}}{N_{Sref}} \right)^3$$  \hspace{1cm} (3-14)

**Cooling Tower Model**

The cooling tower is the main component of IEC that differentiates it from DEC. The cooling tower acts as a heat exchanger. It consists of a water pump that regulates the water flow, fan that regulates the air flow inside the cooling tower and a wet media that cools down the secondary air stream.

The heat load on the cooling tower is calculated as,

$$Q_{CT} = Q_{room} + \eta_{wp} \times P_{wp} + \eta_{sf2} \times P_{sf2}$$  \hspace{1cm} (3-15)

Power required for running the water pump and supply fan depend on the mass flow rate of air calculated from the heat load. These can be calculated in a similar as shown previously for DEC.

**Water Consumption Model**

The water consumed inside the cooling tower depends on the effectiveness of the wet media. Its effectiveness depends on the material being used and it varies from 60% to 90%. The amount of water being consumed is found out by multiplying the mass flow rate of air with the increase in the humidity ratio of secondary air.

$$m_{evap} = m_a (\omega_{a,e} - \omega_{a,i})$$  \hspace{1cm} (3-16)

The detailed calculations have been shown in Appendix A.
• **PUE**

The total power consumed by the Indirect Evaporative Cooling System is

\[ P_{\text{total}} = P_{\text{fan}} + Q_{\text{CT}} + Q_{\text{room}} \]  \hspace{1cm} (3-17)

Thus,

\[ PUE = \frac{P_{\text{total}}}{Q_{\text{room}}} \]  \hspace{1cm} (3-18)

• **Cost Model**

The cost model for IEC is exactly similar to that of DEC and the same set of equations will be used to do the cost analysis.
Chapter 4

THE TOOL – ECT

Now, after the general methods for thermal design of a data center that will be operating on evaporative cooling systems have been identified, developed & established and have been nicely documented in one place as a single body, it is time to put the face onto that body.

The tool “ECT” stands for “Evaporative Cooling Tool” and should not be confused with “Electroconvulsive Therapy” formerly known as electroshock therapy.

The tool was developed using MATLAB’s “App Designer”. The functionality of the tool is based on the models mentioned till last chapter. All the models were coded using MATLAB, so that the cumbersome analytical equations can be avoided, and the computation time could be reduced. The tool developed can be exported as a standalone application with a “.exe” extension and the user can install the application from the setup. All the code that provides functionality for the tool has been shown in Appendix A.

The figure 14 shows how the tool looks like when it is opened. The tool can take up to three TMY3 data files of in “.xlsx” format as inputs. It also gives the option for the user to input the power required for each rack and the number of racks. After hitting the calculate button, the tool does the performance and cost analysis for DEC & IEC for each city. The results are displayed in form of tables and some comparison analysis are shown in the graphs.

The text area on the bottom left gives the assumptions made in the models, so that the user is aware of the data that has been considered while doing the calculations. From the results obtained, a user can make effective decisions on whether they should implement evaporative cooling system or not based on the performance & cost analysis, water & energy consumptions as they are bound by economical and geographical conditions.
Figure 7: Tool layout- initial
Chapter 5
Validation

After the development of the tool, the models have to be validated. For this, a scenario has been considered where a customer plans to migrate to evaporative cooling. Three locations have been chosen for this study. First is Dallas Fort Worth, which is hot and moist. Second is Portland, which has a cold and marine climate and Minneapolis has a cold and humid climate as per [27]. It is being assumed that the user will have 5 racks each of 5kW power requirement. For this case study, the IT room inlet temperatures and the temperature rise across the racks have been taken from previous experimental data performed for CRAH units by [28]. The server used in the experimental setup was 1.5 RU Open Compute Servers. The fans are assumed to be running at 100% efficiency, and 100% outside air is being used in case of DEC.

Table 5-1: Inlet temperatures and $\Delta T$

<table>
<thead>
<tr>
<th>Rack inlet temperature</th>
<th>$\Delta T$</th>
</tr>
</thead>
<tbody>
<tr>
<td>°C</td>
<td>°C</td>
</tr>
<tr>
<td>15</td>
<td>17.5</td>
</tr>
<tr>
<td>20</td>
<td>17</td>
</tr>
<tr>
<td>25</td>
<td>17.5</td>
</tr>
<tr>
<td>30</td>
<td>10.75</td>
</tr>
<tr>
<td>35</td>
<td>7.62</td>
</tr>
<tr>
<td>40</td>
<td>4.85</td>
</tr>
<tr>
<td>45</td>
<td>4</td>
</tr>
</tbody>
</table>
Using the TMY3 data for the three cities and the rack inlet temperatures and the temperature rise across the racks, all the other required parameters were calculated by the tool. The calculations along with the formulas have been shown in Appendix A.

The performance and cost analysis for the three cities was done using the tool and comparisons on water consumption in each area for DEC and IEC were shown in the graphs, which have been shown in the next chapter.
Before the results are displayed, the TMY 3 data for DFW is shown as an example to let the reader know how the TMY3 data looks like. All the unnecessary columns have been hidden.

![Figure 8: TMY3 Data for DFW](image)

The dry bulb, relative humidity, dew point and pressure values are taken from the TMY3 data for the whole year.
### Table 6-1: Results for DEC - DFW

<table>
<thead>
<tr>
<th>Temp °C</th>
<th>PUE</th>
<th>Water ‘gal’</th>
<th>Power ‘kWH’</th>
<th>Water Cost ‘$’</th>
<th>Electricity Cost ‘$’</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>1.0323</td>
<td>48463</td>
<td>226060</td>
<td>229.14</td>
<td>10173</td>
</tr>
<tr>
<td>20</td>
<td>1.0322</td>
<td>47743</td>
<td>226060</td>
<td>225.91</td>
<td>10173</td>
</tr>
<tr>
<td>25</td>
<td>1.0323</td>
<td>49147</td>
<td>226070</td>
<td>232.21</td>
<td>10173</td>
</tr>
<tr>
<td>30</td>
<td>1.0333</td>
<td>83550</td>
<td>226290</td>
<td>386.33</td>
<td>10183</td>
</tr>
<tr>
<td>35</td>
<td>1.0358</td>
<td>119360</td>
<td>226840</td>
<td>546.74</td>
<td>10208</td>
</tr>
<tr>
<td>40</td>
<td>1.0424</td>
<td>167100</td>
<td>228290</td>
<td>760.63</td>
<td>10273</td>
</tr>
<tr>
<td>45</td>
<td>1.0523</td>
<td>208870</td>
<td>230460</td>
<td>947.78</td>
<td>10371</td>
</tr>
</tbody>
</table>

The above table shows the results as obtained in the tool for Dallas Fort Worth Area when Direct Evaporative Cooling system is used for the whole year. The table shows the PUE, amount of water consumed, and the energy consumed and their respective costs.

It can be seen that the PUE is close to 1, thereby stating that the models pretty accurate. Another major trend that must be taken into account is that, as the rack inlet temperature increases, the mass flow rate required to cool down the data center also increases. This leads to an increase in the energy and water consumption and their respective costs.
The table shown below is for DFW area when Indirect Evaporative Cooling is used throughout the year. This also follows a similar trend of increased power consumption with increase in the rack inlet temperatures. The water consumption remains almost the same for all rack inlet temperatures. It can be observed that the PUE in case of IEC is more than PUE of DEC, thus proving the rule of thumb that DEC is more efficient than IEC.

Table 6-2: Results for IEC-DFW

<table>
<thead>
<tr>
<th>Temp °C</th>
<th>PUE</th>
<th>Water ‘gal’</th>
<th>Power ‘kWH’</th>
<th>Water Cost ‘$’</th>
<th>Electricity Cost ‘$’</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>1.0329</td>
<td>282510</td>
<td>226210</td>
<td>127.7</td>
<td>10180</td>
</tr>
<tr>
<td>20</td>
<td>1.0329</td>
<td>282510</td>
<td>226210</td>
<td>127.7</td>
<td>10179</td>
</tr>
<tr>
<td>25</td>
<td>1.0329</td>
<td>282510</td>
<td>226210</td>
<td>127.7</td>
<td>10180</td>
</tr>
<tr>
<td>30</td>
<td>1.0340</td>
<td>282510</td>
<td>226440</td>
<td>127.7</td>
<td>10190</td>
</tr>
<tr>
<td>35</td>
<td>1.0365</td>
<td>282510</td>
<td>226990</td>
<td>127.7</td>
<td>10214</td>
</tr>
<tr>
<td>40</td>
<td>1.0431</td>
<td>282510</td>
<td>228440</td>
<td>127.7</td>
<td>10280</td>
</tr>
<tr>
<td>45</td>
<td>1.0530</td>
<td>282510</td>
<td>230610</td>
<td>127.7</td>
<td>10378</td>
</tr>
</tbody>
</table>
In the following table, a similar trend as that of Table 6-1 can be seen. As the rack inlet temperature increases, the water consumption and power required to cool the data center increase.

Table 6-3: DEC- Portland

<table>
<thead>
<tr>
<th>Temp °C</th>
<th>PUE</th>
<th>Water ‘gal’</th>
<th>Power ‘kWH’</th>
<th>Water Cost ‘$’</th>
<th>Electricity Cost ‘$’</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>1.0554</td>
<td>20740</td>
<td>231130</td>
<td>104.94</td>
<td>10401</td>
</tr>
<tr>
<td>20</td>
<td>1.0552</td>
<td>20432</td>
<td>231130</td>
<td>103.56</td>
<td>10399</td>
</tr>
<tr>
<td>25</td>
<td>1.0555</td>
<td>21033</td>
<td>231160</td>
<td>106.25</td>
<td>10402</td>
</tr>
<tr>
<td>30</td>
<td>1.0693</td>
<td>35756</td>
<td>234180</td>
<td>172021</td>
<td>10538</td>
</tr>
<tr>
<td>35</td>
<td>1.1024</td>
<td>51081</td>
<td>241440</td>
<td>240.87</td>
<td>10865</td>
</tr>
<tr>
<td>40</td>
<td>1.1904</td>
<td>71513</td>
<td>260700</td>
<td>332.40</td>
<td>11732</td>
</tr>
<tr>
<td>45</td>
<td>1.3224</td>
<td>89391</td>
<td>289600</td>
<td>412.50</td>
<td>13032</td>
</tr>
</tbody>
</table>
The following table shows the results for Portland city when IEC is run throughout the year. The trend observed is similar to table 6-2.

<table>
<thead>
<tr>
<th>Temp °C</th>
<th>PUE</th>
<th>Water ‘gal’</th>
<th>Power ‘kWH’</th>
<th>Water Cost ‘$’</th>
<th>Electricity Cost ‘$’</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>1.0639</td>
<td>57636</td>
<td>233000</td>
<td>270.24</td>
<td>10485</td>
</tr>
<tr>
<td>20</td>
<td>1.0639</td>
<td>57636</td>
<td>232960</td>
<td>270.24</td>
<td>10483</td>
</tr>
<tr>
<td>25</td>
<td>1.0641</td>
<td>57636</td>
<td>233030</td>
<td>270.24</td>
<td>10486</td>
</tr>
<tr>
<td>30</td>
<td>1.0796</td>
<td>57636</td>
<td>236430</td>
<td>270.24</td>
<td>10939</td>
</tr>
<tr>
<td>35</td>
<td>1.1169</td>
<td>57636</td>
<td>244590</td>
<td>270.24</td>
<td>11007</td>
</tr>
<tr>
<td>40</td>
<td>1.2158</td>
<td>57636</td>
<td>266270</td>
<td>270.24</td>
<td>11982</td>
</tr>
<tr>
<td>45</td>
<td>1.3643</td>
<td>57636</td>
<td>298770</td>
<td>270.24</td>
<td>13445</td>
</tr>
</tbody>
</table>
Table 6-5 shows the results for Minneapolis when Direct Evaporative Cooling is run throughout the year. The trend is again same as in case of other DEC systems.

<table>
<thead>
<tr>
<th>Temp °C</th>
<th>PUE</th>
<th>Water ‘gal’</th>
<th>Power ‘kWH’</th>
<th>Water Cost ‘$’</th>
<th>Electricity Cost ‘$’</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>1.0554</td>
<td>22641</td>
<td>231130</td>
<td>113.45</td>
<td>10401</td>
</tr>
<tr>
<td>20</td>
<td>1.0552</td>
<td>22304</td>
<td>231100</td>
<td>111.95</td>
<td>10399</td>
</tr>
<tr>
<td>25</td>
<td>1.0555</td>
<td>22960</td>
<td>231160</td>
<td>114.89</td>
<td>10402</td>
</tr>
<tr>
<td>30</td>
<td>1.0693</td>
<td>39032</td>
<td>234180</td>
<td>186.89</td>
<td>10538</td>
</tr>
<tr>
<td>35</td>
<td>1.1024</td>
<td>55760</td>
<td>241440</td>
<td>261.83</td>
<td>10865</td>
</tr>
<tr>
<td>40</td>
<td>1.1904</td>
<td>78064</td>
<td>260700</td>
<td>361.75</td>
<td>11732</td>
</tr>
<tr>
<td>45</td>
<td>1.3224</td>
<td>97581</td>
<td>289600</td>
<td>449.19</td>
<td>13032</td>
</tr>
</tbody>
</table>
The same trend as observed in other IEC systems can be observed in the following table.

Table 6-6: IEC-Minneapolis

<table>
<thead>
<tr>
<th>Temp °C</th>
<th>PUE</th>
<th>Water ‘gal’</th>
<th>Power ‘kWh’</th>
<th>Water Cost ‘$’</th>
<th>Electricity Cost ‘$’</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>1.0639</td>
<td>62192</td>
<td>233000</td>
<td>290.65</td>
<td>10485</td>
</tr>
<tr>
<td>20</td>
<td>1.0637</td>
<td>62192</td>
<td>232960</td>
<td>290.65</td>
<td>10483</td>
</tr>
<tr>
<td>25</td>
<td>1.0641</td>
<td>62192</td>
<td>233030</td>
<td>290.65</td>
<td>10486</td>
</tr>
<tr>
<td>30</td>
<td>1.0796</td>
<td>62192</td>
<td>236430</td>
<td>290.65</td>
<td>10639</td>
</tr>
<tr>
<td>35</td>
<td>1.1169</td>
<td>62192</td>
<td>244590</td>
<td>290.65</td>
<td>11007</td>
</tr>
<tr>
<td>40</td>
<td>1.2158</td>
<td>62192</td>
<td>266270</td>
<td>290.65</td>
<td>11982</td>
</tr>
<tr>
<td>45</td>
<td>1.3643</td>
<td>62192</td>
<td>298770</td>
<td>290.65</td>
<td>13445</td>
</tr>
</tbody>
</table>
This is how the results look in the tool. The tabs named DEC1, IEC1 etc. stand for DFW, Portland and Minneapolis.

Figure 9: Results in tool
This is the temperature vs PUE graph that compares the DEC and IEC systems for DFW area.

It can be seen that the PUE of IEC is greater than PUE of DEC. This has been the case for each city.

![Figure 10: PUE comparison between DEC & IEC](image)

The following graph shows the varying water consumption rate in the three cities for DEC.

![Figure 11: Water Consumption for DEC](image)
The following graph shows the varying water consumption rate in the three cities for IEC.

![Temp vs Water Consumption - IEC](image)

**Figure 12: Water Consumption for IEC**

It can be seen from the above two figures that the water consumption rate is very much greater in DFW compared to other two cities as DFW is a hot region and the other two are cold.

A general trend that can be observed in all of the tables is that the water and energy consumption are almost linear from 15°C to 25°C and increase drastically after that.
The graph shown in figure 20 is a screenshot of the results obtained from the tool.

Figure 13: Temp. vs Water consumption - DEC
Chapter 7

CONCLUSION

The general methods for the thermal design that have been identified developed and established are reliable and have been validated as shown in the results. It was observed that with increase in the inlet temperatures, the water and power consumption increased. The water consumption model developed here is complete and considers gives the exact results.

The tool that has been developed here is ready to be shipped as a standalone application. As observed from results, the PUE is best at 20°C for both the cooling systems for each city. This temperature is well within the ASHRAE recommended region.

Finally, it can be concluded that the tool which is based on the established methods gives reliable results and it was observed that DEC is more efficient than IEC, which is exactly how it should be.
Chapter 8

FUTURE WORK

In future, these methods can be improved by researching in the following areas:

- Models for air flow systems can be introduced in the IT room.
- Analytical models at server levels can also be introduced.
- Models for thermal mixing of entering and return air can also be a good addition to the existing model.
- Adding new features to the UI and increasing its functionality and making it more user friendly.
Chapter 9 References


http://www.airhandlers.net/.


J. E. Fernandes, "Minimizing power consumption at module, server, and rack levels within a data center through design and energy efficient operation of dynamic cooling solutions," ProQuest Dissertations and Theses, 2015.

APPENDIX A

The following code was used in MATLAB “APP DESIGNER” to develop the tool’s GUI and Functionality.

Write this code inside a public function.

methods (Access = public)

function openfile1(app)
    global file
d= uigetfile('*.*');
global tD
tD= xlsread(file,'AF3:AF8762');
global tDP
tDP= xlsread(file,'AI3:AI8762');
global rH
rH= xlsread(file,'AL3:AL8762');
global pr
pr= xlsread(file,'AO3:AO8762');
end

function openfile2(app)
    global file2
d= uigetfile('*.*');
global tD2
tD2= xlsread(file2,'AF3:AF8762');
global tDP2
tDP2= xlsread(file2,'AI3:AI8762');
global rH2
rH2= xlsread(file2,'AL3:AL8762');
end
global pr2
pr2= xlsread(file2,'AO3:AO8762');
end

function openfile3(app)
    global file3
tfile3= uigetfile('*.txt');
global tD3
tD3= xlsread(file3,'AF3:AF8762');
global tDP3
tDP3= xlsread(file3,'AI3:AI8762');
global rH3
rH3= xlsread(file3,'AL3:AL8762');
global pr3
pr3= xlsread(file3,'AO3:AO8762');
end

function openfile4(app)
    global file4
tfile4= uigetfile('*.txt');
global tD4
tD4= xlsread(file4,'AF3:AF8762');
global tDP4
tDP4= xlsread(file4,'AI3:AI8762');
global rH4
rH4= xlsread(file4,'AL3:AL8762');
global pr4
pr4= xlsread(file4,'AO3:AO8762');
end
function DEC1(app)
    global pr
    global tD
    global rH
    global tDP
    a=app.RackPowerEditField.Value;
    b=app.NumberofRacksEditField.Value;
    qRoom=a*b %IT Load in kiloWatts

    nn=1;
    k=1;
    for j=15:5:45

        if(nn==1)
            delTrack= 17.24; % Temperature difference.
        end
        if(nn==2)
            delTrack= 17.5; % Temperature difference.
        end
        if(nn==3)
            delTrack= 17; % Temperature difference.
        end
        if(nn==4)
            delTrack= 10; % Temperature difference.
        end
        if(nn==5)
            delTrack= 7; % Temperature difference.
        end
        if(nn==6)
            delTrack= 5; % Temperature difference.
        end
if(nn==7)
    delTrack= 4; % Temperature difference.
end

% To Calculate Wetbulb, tW

\[
    tW = (-5.806 + 0.672 \times tD - 0.006 \times tD \times tD + (0.061 + 0.004 \times tD + 0.000099 \times tD \times tD) \times rH + (-0.000033 - 0.000005 \times tD - 0.0000001 \times tD \times tD) \times rH \times rH);
\]

% minimum outlet temperature of wet media pad is
% Rigid media with 93% efficiency is considered

\[
    tO = tD - (0.93 \times (tD - tW));
\]

% To calculate Humidity Ratio, we calculate vapor pressure

\[
    c1 = -1.04403 \times 10^{-4};
    c2 = -1.12946 \times 10^{-10};
    c3 = -2.70223 \times 10^{-2};
    c4 = 1.289036 \times 10^{-5};
    c5 = -2.478068 \times 10^{-9};
    c6 = 6.5459;
\]

\[
    tTD = (1.8 \times tD) + 32 + 459.667; \text{ Temp in Rankine}
\]

% Saturation Pr. in psi is

\[
    pWSbar = \exp((c1/tTD)+c2+(c3\times tTD)+(c4\times tTD\times tTD)+(c5\times tTD\times tTD\times tTD)+(c6\times \log(tTD)));
\]

% at dry bulb temp

\[
    pWS = pWSbar \times 68.9475729; \text{ in mbar}
\]

\[
    tTO = (1.8 \times tO) + 32 + 459.667; \text{ minimum temp in Rankine}
\]

% Vapor Pressure in psi is

\[
    pWbar = \exp((c1/tTO)+c2+(c3\times tTO)+(c4\times tTO\times tTO)+(c5\times tTO\times tTO\times tTO)+(c6\times \log(tTO)));
\]

\[
    pW = pWbar \times 68.9475729; \text{ in mbar}
\]

% Humidity Ratio

\[
    w = 0.62198 \times pW / (pr-pW);
\]

\[
    wS = 0.62198 \times pWS / (pr-pWS);
\]
%c=wS-w;
for i=1:8760
    if (wS(i)-w(i))>=0
        c(i)=wS(i)-w(i);
    else
        c(i)=0;
    end
end
c=transpose(c);
%display(c);

%mass of air is
m1=qRoom/(1.005*delTrack); % kg/s

m1CFM= m1*2118.88/1.2250; %ft^3/min
%disp "CFM"

%water consumed is
wCd=m1.*c*3600/1000; %m^3 per hour.
totalwCd= sum(wCd);
%disp("water consumed in litres for 1 month is")
totalwCd=totalwCd*1000;
%app.Results_DEC.value=totalwCd
%disp(totalwCd)

%pump in sump: Power is 1300 Watts
pPump= 1.300; %kW

pRef=0.7457; % kW or 1 HP just for calculations
\[v_{\text{Ref}} = 6500; \text{ CFM}\]
\[n = 2; \text{ number of blowers}\]
\[p_{\text{Blower}} = n \times p_{\text{Ref}} \times (m_{\text{1CFM}} / v_{\text{Ref}})^{3}; \text{ pRef and vRef could be obtained from the manufacturer.}\]

\[\% \text{disp}("Total power is")\]
\[p_{\text{Total}} = q_{\text{Room}} + p_{\text{Pump}} + p_{\text{Blower}}; \text{ Total Power Consumed}\]
\[p_{\text{TotalkWh}} = p_{\text{Total}} \times 8760;\]
\[\% \text{app.Result_Dec.table_as_cell} = p_{\text{TotalkWh}}\]
\[\% \text{disp}(p_{\text{Total}})\]

\[\% \text{disp}("PUE is")\]
\[\text{PUE} = p_{\text{Total}} / q_{\text{Room}}; \text{ PUE}\]
\[\% \text{app.Result_Dec.table_as_cell} = \text{PUE}\]
\[\% \text{disp}(\text{PUE})\]

\[b_{\text{Charge}} = 12.03; \text{ Base Charge, Depends on meter diameter, 3" dia. taken. USD.}\]
\[q_{\text{Water}} = \text{totalwCd} \times 0.264172; \text{ Water consumed in Gallons}\]

\[v_{\text{Charge}} = 4.48 \times (q_{\text{Water}} / 1000); \text{ in gallons, rate is 4.48 per 1000 gallons.}\]
\[m = 1;\]
\[c_{\text{Water}} = 0.01 \times b_{\text{Charge}} + v_{\text{Charge}}; \text{ m = no. of months}\]
\[\% \text{app.Result_Dec.table_as_cell} = c_{\text{Water}}\]
\[\% \text{disp}("water cost in USD is")\]
\[\% \text{disp}(c_{\text{Water}})\]

\[\% \text{Electricity cost is}\]
\[e_{\text{Rate}} = 4.5; \text{ in cents}\]
\[c_{\text{Elec}} = p_{\text{Total}} \times 8760 \times e_{\text{Rate}};\]
\[\% \text{disp}("elect cost in USD is")\]
\[c_{\text{Elec}} = c_{\text{Elec}} / 100;\]
\[\% \text{app.Result_Dec.table_as_cell} = c_{\text{Elec}}\]
\[\% \text{disp}(c_{\text{Elec}})\]
TTTable (k, :, :, :, :, :) = [j, PUE, qWater, pTotalkWh, cWater, cElec];
app.Results_DEC.Data = TTTable;
k = k + 1;

nn = nn + 1;
end
end

function IEC1(app)
global pr
global tD
global rH
global tDP
a = app.RackPowerEditField.Value;
b = app.NumberofRacksEditField.Value;
qRoom = a * b; % IT Load in kiloWatts

nn = 1;
k = 1;
for j = 15:5:45

if(nn == 1)
    delTrack = 17.24; % Temperature difference.
end
if(nn == 2)
    delTrack = 17.5; % Temperature difference.
end
if(nn == 3)
    delTrack = 17; % Temperature difference.
end
if(nn == 4)
    delTrack = 10; % Temperature difference.
end

end
if(nn==5)
    delTrack= 7; % Temperature difference.
end
if(nn==6)
    delTrack= 5; % Temperature difference.
end
if(nn==7)
    delTrack= 4; % Temperature difference.
end

%To Calculate Wetbulb, tW
\[ tW = (-5.806+0.672*tD-0.006.*tD.*tD+(0.061+0.004.*tD+0.000099.*tD.*tD).*rH+(-0.000033-0.000005.*tD-0.0000001.*tD.*tD).*rH.*rH); \]

%mass of air is
\[ m1=qRoom/(1.005*delTrack); \% \text{kg/s} \]

\[ m1\text{CFM}= m1*2118.88/1.2250; \% \text{ft}^3/\text{min} \]
%disp "CFM";

%Cooling tower Calculations
qCT=qRoom+0.30*0.8+0.3*0.37285;
mAir=qCT/(1.005*10); \% \text{kg/s} \%change in temp of water is assumed to be 10.
mAirCFM= mAir*2118.88/1.2250 ;%
% disp("in CFM");
%Cooling tower Fan specs
pRefCT=1.5; \%kW
vRefCT= 8500; \%CFM assumed
pCTFan = pRefCT*(mAirCFM/vRefCT)^3;
The inlet air temperature is taken as the exhaust air temperature from the data hall and the mass flow rate is assumed to be the same as flow rate through the data hall.

The exit air temp is \( t_{AE} \)

\[
 t_{AE} = \frac{q_{CT}}{(m_{Air}\text{CFM} \times 1.005)} + t_D
\]

Temp after cooling tower

\[
 t_{At} = t_D + (0.75 \times (t_D - t_W))
\]

To calculate Humidity Ratio, we calculate vapor pressure

\[
 c_1 = 1.04403 \times 10^4; c_2 = -1.12946 \times 10^1; c_3 = -2.70223 \times 10^{-2}; c_4 = 1.289036 \times 10^{-5}; c_5 = -2.478068 \times 10^{-9}; c_6 = 6.5459;
\]

\[
 t_{TD} = (1.8 \times t_D) + 32 + 459.667; \text{%Temp in Rankine}
\]

Saturation Pr. in psi is

\[
 p_{WSbar} = \exp\left(\frac{c_1}{t_{TD}} + c_2 + c_3 \times t_{TD} + c_4 \times t_{TD} \times t_{TD} + c_5 \times t_{TD} \times t_{TD} \times t_{TD} + c_6 \times \log(t_{TD})\right);
\]

\% at dry bulb temp

\[
 p_{WS} = p_{WSbar} \times 68.9475729; \text{%in mbar}
\]

\[
 t_{AE} = (1.8 \times t_{AE}) + 32 + 459.667; \text{%minimum temp in Rankine}
\]

Vapor Pressure in psi is

\[
 p_{Wbar} = \exp\left(\frac{c_1}{t_{AE}} + c_2 + c_3 \times t_{AE} + c_4 \times t_{AE} \times t_{AE} + c_5 \times t_{AE} \times t_{AE} \times t_{AE} + c_6 \times \log(t_{AE})\right);
\]

\[
 p_{W} = p_{Wbar} \times 68.9475729; \text{%in mbar}
\]

Humidity Ratio

\[
 w = 0.62198 \times \frac{p_{W}}{(pr-p_{W})};
\]

\[
 w_S = 0.62198 \times \frac{p_{WS}}{(pr-p_{WS})};
\]
for i=1:8760
    if (w(i)-wS(i))>=0
        c(i)=w(i)-wS(i);
    else
        c(i)=0;
    end
end
c=transpose(c);

%Pump in sump for cooling tower: Power is 1300 Watts
pPump= 1.3; %kW

pRef=0.7457; % kW or 1 HP just for calculations
vRef=6250; % CFM
n=2; % number of blowers. 2 are being used to make sure even if 1 fails, the other keeps running.
pBlower= n*pRef*(m1CFM/vRef)^3; % pRef and vRef could be obtained from the manufacturer.

%Total power consumed is sum of power pump, power blower, power cooling tower and power IT room.

% disp("Total Power consumed is")
pTotal= pPump+pBlower+pCTFan+qRoom;
pTotalkWh=pTotal*8760;

% disp(pTotal)
% disp("PUE is")
pUE=pTotal/qRoom;

% disp(pUE)

% water consumed is
wCd = vRef * 1.2250 / 2118.88 * c * 3600 / 1000; % m^3 per hour.
totalwCd = sum(wCd);
% disp("water consumed in litres for 1 month is")
totalwCd = totalwCd * 1000;

% disp(totalwCd)

bCharge = 12.03; % Base Charge, Depends on meter diameter, 3" dia. taken. USD.
qWater = totalwCd * 0.264172; % Water consumed in Gallons
vCharge = 4.48 * (qWater / 1000); % in gallons, rate is 4.48 per 1000 gallons.
m = 1;
cWater = m * (bCharge + vCharge); % m = no. of months

% disp("water cost in USD is")
% disp(cWater)

% Electricity cost is
eRate = 4.5; % in cents
cElec = pTotal * 8760 * eRate;
% disp("elect cost in USD is")
cElec = cElec / 100;

% disp(cElec)

TTable(k, :, :, :, :, :) = [j, pUE, qWater, pTotalWh, cWater, cElec];
app.Results_IEC.Data = TTable;
k = k + 1;
nn = nn + 1;
function DEC2(app)
    global pr2
    global tD2
    global rH2
    global tDP2
    a=app.RackPowerEditField.Value;
    b=app.NumberofRacksEditField.Value;
    qRoom=a*b %IT Load in kiloWatts

    nn=1;
    k=1;
    for j=15:5:45
        if(nn==1)
            delTrack= 17.24; % Temperature difference.
        end
        if(nn==2)
            delTrack= 17.5; % Temperature difference.
        end
        if(nn==3)
            delTrack= 17; % Temperature difference.
        end
        if(nn==4)
            delTrack= 10; % Temperature difference.
        end
        if(nn==5)
            delTrack= 7; % Temperature difference.
        end
    end
end
end
if(nn==6)
    delTrack= 5; % Temperature difference.
end
if(nn==7)
    delTrack= 4; % Temperature difference.
end

%To Calculate Wetbulb, tW

```matlab
tW2=(-5.806+0.672*tD2-0.006.*tD2.*tD2+(0.061+0.004.*tD2+0.000099.*tD2.*tD2).*rH2+(-0.000033-0.000005.*tD2-0.0000001.*tD2.*tD2).*rH2.*rH2);
```

%minimum outlet temperature of wet media pad is
%Rigid media with 93% efficiency is considered

tO2= tD2-(0.93.*(tD2-tW2));

%To calculate Humidity Ratio, we calculate vapor pressure

c1=-1.04403*10^4; c2=-1.12946*10; c3=-2.70223*10^-2; c4=1.289036*10^-5; c5=-2.478068*10^-9; c6=6.5459;

tTD2=(1.8*tD2)+32+459.667; %Temp in Rankine

%Saturation Pr. in psi is

```matlab
pWSbar=exp((c1./tTD2)+c2+(c3.*tTD2)+(c4.*tTD2.*tTD2)+(c5.*tTD2.*tTD2.*tTD2)+(c6.*log(tTD2))); % at dry bulb temp
pWS=pWSbar*68.9475729; %in mbar
```

tTO2=(1.8*tO2)+32+459.667; %minimum temp in Rankine

%Vapor Pressure in psi is
\[ p_{\text{W bar}} = \exp\left(\frac{c_1}{t_{\text{TO2}}} + c_2 + (c_3 \cdot t_{\text{TO2}}) + (c_4 \cdot t_{\text{TO2}} \cdot t_{\text{TO2}}) + (c_5 \cdot t_{\text{TO2}} \cdot t_{\text{TO2}} \cdot t_{\text{TO2}}) + (c_6 \cdot \log(t_{\text{TO2}}))\right) ; \]

\[ p_{\text{W}} = p_{\text{W bar}} \cdot 68.9475729; \text{ in mbar} \]

% Humidity Ratio

\[ w = 0.62198 \cdot \frac{p_{\text{W}}}{(p_{\text{r2}} - p_{\text{W}})} ; \]
\[ w_{\text{S}} = 0.62198 \cdot \frac{p_{\text{WS}}}{(p_{\text{r2}} - p_{\text{WS}})} ; \]

% \( c = w_{\text{S}} - w ; \)

for \( i = 1:8760 \)

\[
\text{if } (w_{\text{S}}(i) - w(i)) >= 0
\]
\[ c(i) = w_{\text{S}}(i) - w(i) ; \]

\[
\text{else}
\]
\[ c(i) = 0 ; \]

\[
\text{end}
\]
end

c = transpose(c) ;
% display(c) ;

% mass of air is
\[ m_1 = \frac{q_{\text{Room}}}{1.005 \cdot \text{del Track}} ; \text{ kg/s} \]

\[ m_1 \text{CFM} = m_1 \cdot 2118.88 / 1.2250 ; \text{ ft}^3/\text{min} \]
% disp "CFM"

% water consumed is
\[ w_{\text{Cd}} = m_1 \cdot c \cdot 3600 / 1000 ; \text{ m}^3 \text{ per hour.} \]

\[ \text{total}w_{\text{Cd}} = \text{sum}(w_{\text{Cd}}) ; \]
% disp("water consumed in litres for 1 month is")
\[ \text{total}w_{\text{Cd}} = \text{total}w_{\text{Cd}} \cdot 1000 ; \]
%app.Results_DEC.value=totalwCd
%disp(totalwCd)

%Pump in sump: Power is 1300 Watts
pPump= 1.300; %kW

pRef=0.7457; % kW or 1 HP just for calculations
vRef=6500; %CFM

n=2; % number of blowers
pBlower= n*pRef*(m1CFM/vRef)^3; % pRef and vRef could be obtained from the manufacturer.

%disp("Total power is")
pTotal= qRoom+pPump+pBlower; %Total Power Consumed
pTotalkWh=pTotal*8760;
%app.Results_DEC.table_as_cell=pTotalkWh
%disp(pTotal)

%disp("PUE is")
PUE=pTotal/qRoom; %PUE
%app.Results_DEC.table_as_cell=PUE
%disp(PUE)

bCharge= 12.03; %Base Charge, Depends on meter diameter, 3” dia. taken. USD.
qWater=totalwCd*0.264172;% Water consumed in Gallons

vCharge= 4.48*(qWater/1000) ; %in gallons, rate is 4.48 per 1000 gallons.
m=1;
cWater=m*(bCharge+vCharge); %m = no. of months
%app.Results_DEC.table_as_cell=cWater
%disp("water cost in USD is")
%disp(cWater)
Electricity cost is

eRate=4.5; % in cents
cElec = pTotal * 8760 * eRate;
% disp("elect cost in USD is")
cElec = cElec / 100;
% app.Results_DEC.table_as_cell = cElec
% disp(cElec)
TTable (k, :, :, :, :, :) = [j, PUE, qWater, pTotalWh, cWater, cElec];
app.Results_DEC_2.Data = TTable;
k = k + 1;
nn = nn + 1;
end
end

function IEC2(app)
global pr2
global tD2
global rH2
global tDP2
a = app.RackPowerEditField.Value;
b = app.NumberofRacksEditField.Value;
qRoom = a * b; % IT Load in kiloWatts

nn = 1;
k = 1;
for j = 15:5:45
    if(nn == 1)
        delTrack = 17.24; % Temperature difference.
    end
    if(nn == 2)
delTrack= 17.5; % Temperature difference.
end
if(nn==3)
    delTrack= 17; % Temperature difference.
end
if(nn==4)
    delTrack= 10; % Temperature difference.
end
if(nn==5)
    delTrack= 7; % Temperature difference.
end
if(nn==6)
    delTrack= 5; % Temperature difference.
end
if(nn==7)
    delTrack= 4; % Temperature difference.
end

% To Calculate Wetbulb, tW

%mass of air is
m1=qRoom/(1.005*delTrack); % kg/s

m1CFM= m1*2118.88/1.2250; % ft^3/min
% disp "CFM";

% Cooling tower Calculations
qCT=qRoom+0.30*0.8+0.3*0.37285;
mAir=qCT/(1.005*10); % kg/s %change in temp of water is assumed to be 10.
mAirCFM= mAir*2118.88/1.2250 ;%

% disp("in CFM");
%Cooling tower Fan specs
pRefCT=1.5; %kW
vRefCT= 8500; %CFM assumed
pCTFan= pRefCT*(mAirCFM/vRefCT)^3;

%The inlet air temperature is taken as the exhaust air temperature from the
data hall and the mass flow rate is assumed to be the same as flow rate through the
data hall
%The exit air temp is tAE
%tAe= qCT/(mAirCFM*1.005)+tD

%Temp after cooling tower
tAe= tD2+(0.75.*(tD2-tW2));

%To calculate Humidity Ratio, we calculate vapor pressure
c1=-1.04403*10^4; c2=-1.12946*10; c3=-2.70223*10^-2; c4=1.289036*10^-5; c5=-2.478068*10^-9; c6=6.5459;
tTD=(1.8*tD2)+32+459.667; %Temp in Rankine

%Saturation Pr. in psi is

pWSbar=exp((c1./tTD)+c2+(c3.*tTD)+(c4.*tTD.*tTD)+(c5.*tTD.*tTD.*tTD)+(c6.*log(tTD))); % at dry bulb temp
pWS=pWSbar*68.9475729; %in mbar

 tAE=(1.8*tAe)+32+459.667; %minimum temp in Rankine

%Vapor Pressure in psi is
\[
pWbar = \exp\left(\frac{c1}{tAE} + c2 + c3 \cdot tAE + c4 \cdot tAE \cdot tAE + c5 \cdot tAE \cdot tAE \cdot tAE + c6 \cdot \log(tAE)\right);
\]
\[
pW = pWbar \cdot 68.9475729; \ % \text{in mbar}
\]

%Humidity Ratio
\[
w = 0.62198 \cdot \frac{pW}{(pr2 - pW)};
\]
\[
wS = 0.62198 \cdot \frac{pWS}{(pr2 - pWS)};
\]

for \( i = 1:8760 \)

\[
\text{if } (w(i) - wS(i)) \geq 0
\]
\[
c(i) = w(i) - wS(i);
\]
\[
\text{else}
\]
\[
c(i) = 0;
\]
\[
\text{end}
\]
\[
\text{end}
\]
\[
c = \text{transpose}(c);
\]

%Pump in sump for cooling tower: Power is 1300 Watts
\[
pPump = 1.3; \ % \text{kW}
\]

\[
pRef = 0.7457; \ % \text{kW or 1 HP just for calculations}
\]
\[
vRef = 6250; \ % \text{CFM}
\]
\[
n = 2; \ % \text{number of blowers. 2 are being used to make sure even if 1 fails, the other}
\]
\[
\text{keeps running.}
\]
\[
pBlower = n \cdot pRef \cdot (m1 \text{CFM/vRef})^3; \ % \text{pRef and vRef could be obtained from the}
\]
\[
\text{manufacturer.}
\]

%Total power consumed is sum of power pump, power blower, power cooling
%tower and power IT room.
\% disp("Total Power consumed is")
\text{pTotal} = \text{pPump}+\text{pBlower}+\text{pCTFan}+\text{qRoom};
\text{pTotalWh} = \text{pTotal} \times 8760;

\% disp(\text{pTotal})
\% disp("PUE is")
\text{pUE} = \text{pTotal}/\text{qRoom};

\% disp(\text{pUE})

\% water consumed is
\text{wCd} = \text{vRef} \times 1.2250/2118.88. \times \text{c} \times 3600/1000; \% \text{m}^3 \text{ per hour.}
\text{totalwCd} = \text{sum(wCd)};
\% disp("water consumed in litres for 1 month is")
\text{totalwCd} = \text{totalwCd} \times 1000;

\% disp(\text{totalwCd})

\text{bCharge} = 12.03; \% \text{Base Charge, Depends on meter diameter, 3" dia. taken. USD.}
\text{qWater} = \text{totalwCd} \times 0.264172; \% \text{Water consumed in Gallons}
\text{vCharge} = 4.48*(\text{qWater}/1000); \% \text{in gallons, rate is 4.48 per 1000 gallons.}
m = 1;
\text{cWater} = m*(\text{bCharge}+\text{vCharge}); \% m = \text{no. of months}

\% disp("water cost in USD is")
\% disp(\text{cWater})

\% Electricity cost is
\text{eRate} = 4.5; \% \text{in cents}
\text{cElec} = \text{pTotal} \times 8760 \times \text{eRate};
\% disp("elect cost in USD is")
cElec=cElec/100;

% disp(cElec)

TTable (k, :, :, :, :, :) = [j,pUE, qWater, pTotalWh,cWater, cElec];
app.Results_IEC_2.Data= TTable;
k=k+1;
nn=nn+1;

end
end

function DEC3(app)

global pr3
global tD3
global rH3
global tDP3
a=app.RackPowerEditField.Value;
b=app.NumberofRacksEditField.Value;
qRoom=a*b; %IT Load in kiloWatts

nn=1;
k=1;
for j=15:5:45

if(nn==1)
    delTrack= 17.24; % Temperature difference.
end
if(nn==2)
    delTrack= 17.5; % Temperature difference.
end

end
end  
if(nn==3)  
    delTrack= 17; % Temperature difference.  
end  
if(nn==4)  
    delTrack= 10; % Temperature difference.  
end  
if(nn==5)  
    delTrack= 7; % Temperature difference.  
end  
if(nn==6)  
    delTrack= 5; % Temperature difference.  
end  
if(nn==7)  
    delTrack= 4; % Temperature difference.  
end  

%To Calculate Wetbulb, tW  
tW3=(-5.806+0.672*tD3-
    0.006.*tD3.*tD3+(0.061+0.004.*tD3+0.000099.*tD3.*tD3).*rH3+
    (+0.000033-
    0.000005.*tD3-
    0.0000001.*tD3.*tD3).*rH3.*rH3) ;

%minimum outlet temperature of wet media pad is  
%Rigid media with 93% efficiency is considered  
tO3= tD3-(0.93.*(tD3-tW3));

%To calculate Humidity Ratio, we calculate vapor pressure  
c1=-1.04403*10^4; c2=-1.12946*10; c3=-2.70223*10^-2; c4=1.289036*10^-5; c5=-2.478068*10^-9; c6=6.5459;  
tTD3=(1.8*tD3)+32+459.667; %Temp in Rankine

%Saturation Pr. in psi is
\[
pWSbar = \exp\left(\frac{c_1}{tTD3} + c_2 + c_3 \cdot tTD3 + c_4 \cdot tTD3 \cdot tTD3 + c_5 \cdot tTD3 \cdot tTD3 \cdot tTD3 + c_6 \cdot \log(tTD3)\right); \quad \% \text{at dry bulb temp}
\]
\[
pWS = pWSbar \cdot 68.9475729; \quad \% \text{in mbar}
\]
\[
tTO3 = (1.8 \cdot tO3) + 32 + 459.667; \quad \% \text{minimum temp in Rankine}
\]

\[% \text{Vapor Pressure in psi is}
\]
\[
pWbar = \exp\left(\frac{c_1}{tTO3} + c_2 + c_3 \cdot tTO3 + c_4 \cdot tTO3 \cdot tTO3 + c_5 \cdot tTO3 \cdot tTO3 \cdot tTO3 + c_6 \cdot \log(tTO3)\right);
\]
\[
pW = pWbar \cdot 68.9475729; \quad \% \text{in mbar}
\]

\[% \text{Humidity Ratio}
\]
\[
w = 0.62198 \cdot \frac{pW}{pr3-pW};
\]
\[
wS = 0.62198 \cdot \frac{pWS}{pr3-pWS};
\]

\[% c = wS - w;
\]
\[
\text{for } i = 1:8760
\]
\[
\quad \text{if } (wS(i)-w(i)) \geq 0
\]
\[
\quad \quad c(i) = wS(i)-w(i);
\]
\[
\quad \text{else}
\]
\[
\quad \quad c(i) = 0;
\]
\[
\quad \text{end}
\]
\[
\text{end}
\]
\[
c = \text{transpose}(c);
\]
\[
\% \text{display}(c);
\]

\[% \text{mass of air is}
\]
\[
m1 = qRoom/(1.005 \cdot \text{delTrack}); \quad \% \text{kg/s}
\]
\[
m1CFM = m1 \times 2118.88/1.2250; \quad \% \text{ft}^3/\text{min}
\]
%disp "CFM"

%water consumed is
wCd=m1.*c*3600/1000; %m^3 per hour.
totalwCd= sum(wCd);
%disp("water consumed in litres for 1 month is")
totalwCd=totalwCd*1000;
%app.Results_DEC.value=totalwCd
%disp(totalwCd)

%Pump in sump: Power is 1300 Watts
pPump= 1.300; %kW

pRef=0.7457; % kW or 1 HP just for calculations
vRef=6500; %CFM
n=2; % number of blowers
pBlower= n*pRef*(m1CFM/vRef)^3; % pRef and vRef could be obtained from the manufacturer.
%disp("Total power is")
pTotal= qRoom+pPump+pBlower; %Total Power Consumed
pTotalkWh=pTotal*8760;
%app.Results_DEC.table_as_cell=pTotalkWh
%disp(pTotal)

%disp("PUE is")
PUE=pTotal/qRoom; %PUE
%app.Results_DEC.table_as_cell=PUE
%disp(PUE)

bCharge= 12.03; %Base Charge, Depends on meter diameter, 3” dia. taken. USD.
qWater=totalwCd*0.264172;% Water consumed in Gallons
vCharge = 4.48*(qWater/1000); %in gallons, rate is 4.48 per 1000 gallons.
m=1;
cWater=m*(bCharge+vCharge); %m = no. of months
%app.Results_DEC.table_as_cell=cWater
%disp("water cost in USD is")
%disp(cWater)

%Electricity cost is
eRate=4.5; % in cents
cElec= pTotal*8760*eRate;
%disp("elect cost in USD is")
cElec=cElec/100;
% app.Results_DEC.table_as_cell=cElec
%disp(cElec)
TTable (k, :, :, :, :, :)=[j,PUE, qWater, pTotalkWh,cWater, cElec];
app.Results_DEC_3.Data= TTable;
k=k+1;
nn=nn+1;

end
end

function IEC3(app)
global pr3
global tD3
global rH3
global tDP3
a=app.RackPowerEditField.Value;
b=app.NumberofRacksEditField.Value;
qRoom=a*b; %IT Load in kiloWatts
nn=1;
k=1;
for j=15:5:45

if(nn==1)
    delTrack= 17.24; % Temperature difference.
end
if(nn==2)
    delTrack= 17.5; % Temperature difference.
end
if(nn==3)
    delTrack= 17; % Temperature difference.
end
if(nn==4)
    delTrack= 10; % Temperature difference.
end
if(nn==5)
    delTrack= 7; % Temperature difference.
end
if(nn==6)
    delTrack= 5; % Temperature difference.
end
if(nn==7)
    delTrack= 4; % Temperature difference.
end

%To Calculate Wetbulb, tW

\[ tW3= -5.806 + 0.672 \cdot tD3 - 0.006 \cdot tD3 \cdot tD3 + (0.061 + 0.004 \cdot tD3 + 0.000099 \cdot tD3 \cdot tD3) \cdot rH3 + (-0.000033 - 0.000005 \cdot tD3 - 0.0000001 \cdot tD3 \cdot tD3) \cdot rH3 \cdot rH3; \]
%mass of air is
m1=qRoom/(1.005*delTrack); % kg/s

m1CFM= m1*2118.88/1.2250; %ft^3/min
%disp "CFM";

%Cooling tower Calculations
qCT=qRoom+0.30*0.8+0.3*0.37285;
mAir=qCT/(1.005*10); % kg/s %change in temp of water is assumed to be 10.
mAirCFM= mAir*2118.88/1.2250 ;%
% disp("in CFM");
%Cooling tower Fan specs
pRefCT=1.5; %kW
vRefCT= 8500; %CFM assumed
pCTFan= pRefCT*(mAirCFM/vRefCT)^3;

%The inlet air temperature is taken as the exhaust air temperature from the
%data hall and the mass flow rate is assumed to be the same as flow rate through the
data hall
%The exit air temp is tAE
%tAe= qCT/(mAirCFM*1.005)+tD

%Temp after cooling tower
tAe= tD3+(0.75.*(tD3-tW3));

%To calculate Humidity Ratio, we calculate vapor pressure
\[ c1=1.04403\times10^4; \quad c2=-1.12946\times10; \quad c3=-2.70223\times10^{-2}; \quad c4=1.289036\times10^{-5}; \quad c5=-2.478068\times10^{-9}; \quad c6=6.5459; \]
\[ tTD=(1.8\times tD3)+32+459.667; \quad \%Temp in Rankine \]

%Saturation Pr. in psi is
\[
pWSbar = \exp\left(\frac{c_1}{tTD}\right) + c_2 + (c_3 \cdot tTD) + (c_4 \cdot tTD \cdot tTD) + (c_5 \cdot tTD \cdot tTD \cdot tTD) + (c_6 \cdot \log(tTD))
\]

% at dry bulb temp

\[
pWS = pWSbar \cdot 68.9475729; \text{ in mbar}
\]

\[
tAE = (1.8 \cdot tAe) + 32 + 459.667; \text{ minimum temp in Rankine}
\]

% Vapor Pressure in psi is

\[
pWbar = \exp\left(\frac{c_1}{tAE}\right) + c_2 + (c_3 \cdot tAE) + (c_4 \cdot tAE \cdot tAE) + (c_5 \cdot tAE \cdot tAE \cdot tAE) + (c_6 \cdot \log(tAE))
\]

\[
pW = pWbar \cdot 68.9475729; \text{ in mbar}
\]

% Humidity Ratio

\[
w = 0.62198 \cdot pW / (pr3 - pW);
\]

\[
wS = 0.62198 \cdot pWS / (pr3 - pWS);
\]

\[
\text{for } i = 1:8760
\]

\[
\text{if } (w(i) - wS(i)) \geq 0\]

\[
c(i) = w(i) - wS(i);
\]

\[
\text{else}
\]

\[
c(i) = 0;
\]

\[
\text{end}
\]

\[
\text{end}
\]

\[
c = \text{transpose}(c);
\]

% Pump in sump for cooling tower: Power is 1300 Watts

pPump = 1.300; % kW

pRef = 0.7457; % kW or 1 HP just for calculations
vRef=6250; %CFM
n=2; % number of blowers. 2 are being used to make sure even if 1 fails, the other keeps running.
pBlower= n*pRef*(m1CFM/vRef)^3; % pRef and vRef could be obtained from the manufacturer.

%Total power consumed is sum of power pump, power blower, power cooling tower and power IT room.

% disp("Total Power consumed is")
pTotal= pPump+pBlower+pCTFan+qRoom;
pTotalkWh=pTotal*8760;

% disp(pTotal)
% disp("PUE is")
pUE=pTotal/qRoom;

% disp(pUE)

%water consumed is
wCd=vRef*1.2250/2118.88.*c.*3600/1000; %m^3 per hour.
totalwCd= sum(wCd);
% disp("water consumed in litres for 1 month is")
totalwCd=totalwCd*1000;

% disp(totalwCd)

bCharge= 12.03; %Base Charge, Depends on meter diameter, 3” dia. taken. USD.
qWater=totalwCd*0.264172; % Water consumed in Gallons
vCharge= 4.48*(qWater/1000); %in gallons, rate is 4.48 per 1000 gallons.
m=1;
cWater=m*(bCharge+vCharge); %m = no. of months
% disp("water cost in USD is")
% disp(cWater)

%Electricity cost is
eRate=4.5; % in cents
cElec= pTotal*8760*eRate;
% disp("elect cost in USD is")
cElec=cElec/100;

% disp(cElec)

TTable (k, :, :, :, :, :)=[j,pUE, qWater, pTotalWh,cWater, cElec];
app.Results_IEC_3.Data= TTable;
k=k+1;
nn=nn+1;

end
end

end

Write this code in the call back functionality of the call button.

DEC1(app);
IEC1(app);
DEC2(app);
IEC2(app);
DEC3(app);
IEC3(app);
%for graphs

tableData_DEC = get(app.Results_DEC,'Data');
tableData_DEC2 = get(app.Results_DEC_2,'Data');
tableData_DEC3 = get(app.Results_DEC_3,'Data');
etableData_DEC4 = get(app.Results_DEC_4,'Data');
plot(app.UIAxes, tableData_DEC(:,1), tableData_DEC(:,2));
plot(app.UIAxes_2, tableData_DEC(:,1), tableData_DEC(:,3));
plot(app.UIAxes_2, tableData_DEC2(:,1), tableData_DEC2(:,3));
plot(app.UIAxes_2, tableData_DEC3(:,1), tableData_DEC3(:,3));
plot(app.UIAxes_2, tableData_DEC4(:,1), tableData_DEC4(:,3));

etableData_IEC = get(app.Results_IEC, 'Data');
tableData_IEC2 = get(app.Results_IEC_2, 'Data');
tableData_IEC3 = get(app.Results_IEC_3, 'Data');
etableData_IEC4 = get(app.Results_IEC_4, 'Data');
plot(app.UIAxes, tableData_IEC(:,1), tableData_IEC(:,2));
plot(app.UIAxes_3, tableData_IEC(:,1), tableData_IEC(:,3));
plot(app.UIAxes_3, tableData_IEC2(:,1), tableData_IEC2(:,3));
plot(app.UIAxes_3, tableData_IEC3(:,1), tableData_IEC3(:,3));
plot(app.UIAxes_3, tableData_IEC4(:,1), tableData_IEC4(:,3));

legend(app.UIAxes,{'DEC', 'IEC'}, 'location', 'northwest');
legend(app.UIAxes_2,{'City1', 'City2', 'City3'}, 'location', 'northwest');
legend(app.UIAxes_3,{'City1', 'City2', 'City3'}, 'location', 'northwest');

app.NoteTextArea.Value='For DEC: Rigid media of 93% effectiveness. Pump is of 0.8kW. 2 supply fans of 1/2 HP are considered. For IEC: The cooling tower fan is of 1kW, pump is of 0.8kW, HX efficiency is 75%, 2 supply fans of 1/2 HP, *Selection of equipments is based on the required CFM'
BIOGRAPHICAL INFORMATION

Raghvendra Parinam received his Bachelor’s Degree in Mechanical Engineering from GITAM University, Hyderabad, India in 2015. In May 2018 he graduated from University of Texas at Arlington with a Master’s of Science degree in Mechanical Engineering. His research was focused on Thermal Modelling of Data Centers for Evaporative Cooling. He worked as a Research Assistant in Electronics, MEMS and Nanoelectronics Packaging Center while working on various industry-funded projects and in NSF funded Industry University Cooperative Research Center called Center of Energy-Smart Electronic Systems (ES2).