DEVELOPING AND VALIDATING THERMODYNAMIC MODELS FOR EVAPORATIVE COOLING EMPLOYED IN DATACENTERS

By

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Abstract

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Cooling of Data centers is of paramount importance for continuous and reliable operation of Servers housed by them. Considerable amount of total power is used to drive the cooling activity. Various different types of cooling methods have been implemented and are currently operating in various data centers. Evaporative (Indirect/Direct) Cooling is trending these days because of its ability to provide cooling at minimal cost. Previous studies have presented various cooling plants in data centers and studied them.

A dedicated model for Evaporative cooling is developed and validated. The data is computed assuming the developed cooling model operating at a particular location and subsequently computing the cooling capacity and performance parameters of a Datacenter such as PUE and Water consumption. Also the environmental data for temperatures at a particular location is gathered. Based on the data and results obtained from the models and by analyzing an actual Evaporative Cooling Unit we can validate our models. After validation of the models we can study the performance of the data center along the Thermal Guidelines created by ASHRAE.

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CHAPTER 1

INTRODUCTION

1.1 Data center cooling

Data centers are purpose-built facilities which house IT equipment which need to be running at most/all times. For instance, any activity that we perform on the internet, be it searching on Google, viewing videos on YouTube, or using Social Networking sites such as Facebook, LinkedIn, Instagram or literally anything that we view, post or download is obtained from data centers. All the information is stored in computers (aptly called servers) stacked in rows of several racks in a datacenter. Any command executed by us on our browser sends a request to these servers to access the information and thereafter the required information is relayed back to us. Now, if this information has to be accessed at any time of the day throughout the year, then the server has to be up and running at all times.

Now, when our personal computer is operating continuously for few hours it starts getting heated up and we can feel that by touch. Imagine the amount of heat liberated from servers which are operating continuously at all times. This heat generated is enormous and needs to be removed quickly in order to keep the servers operating. Therefore, Cooling becomes of paramount importance and needs to be supplied continuously.

There are different methods that are currently employed in data centers to cool their servers. We have Liquid Cooling, where water is circulated through tubes in server to collect and reject heat from the cold plate to the surrounding. Immersion Cooling practices immersion of the servers in a di-electric medium which is circulated through a heat exchanger to reject heat to the surrounding. But, Air Cooling is used widely because of less complexity and greater flexibility. Air Cooling has been in practice ever since the introduction of data centers.

1.2 Air Cooling Employed in Data centers

Traditional Data centers employ Computer Room Air Conditioning (CRAC) Units. These work on refrigeration cycle where compressor consumes most of the energy. The hot exhaust air is returned to the CRAC where air-to-refrigerant heat exchanger cools the air and supply the air to cold aisle. The evaporated refrigerant is the compressed by the compressor to increase its pressure and temperature. This compressed refrigerant loses its heat to the Cooling tower water there by condensing to liquid phase again where it is expanded to absorb heat from the hot exhaust air. The water absorbs the heat and rejects it to the atmosphere with the help of cooling tower.

Figure 1-1: CRAC Unit

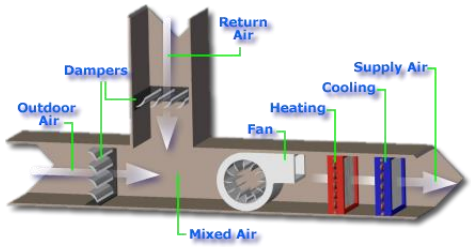


Figure 1-1: Airside Economizer

Air Side Economization is also being employed these days because of its ability to give better energy efficiency. Whenever ambient conditions are favorable to the data centers, the outside air is directly fed to the data center thereby consuming less energy for cooling resulting in better energy efficiency. But this practice invites the risk of contamination on the hardware and measures to prevent it have to be adopted.

Evaporative Cooling is another means of cooling the air by evaporating water into its stream to lower the dry bulb temperature of the air. However, in doing so the moisture content of the air is increased. There is a much higher risk of contamination due to high moisture content in the air and accordingly measures need to be adopted to prevent corrosion and other forms of contamination. However, the energy efficiency obtained is much better than the conventional CRAC employed data centers.

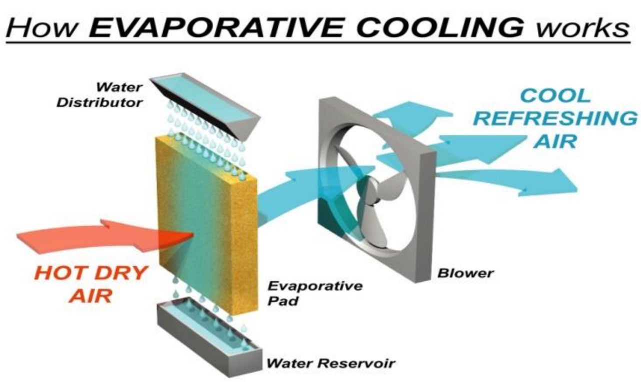


Figure 1-2: Evaporative Cooling

CHAPTER 2

WHY EVAPORATIVE COOLING?

2.1 Quality Metric

Power Usage Effectiveness: This is a metric defined by the Green Grid to measure the effectiveness of a data center. It is a ratio of Total power required by the facility to the IT compute power.

$$PUE = \frac{Total\ Facility\ Power}{IT\ Compute\ Power} = \frac{IT\ Compute\ Power + Cooling\ Power + Misc\ Power}{IT\ Compute\ Power}$$

This gives us a sense of understanding as to how much of power is spent on other activities which is not for the prime use. Therefore, we aim to reduce the Cooling Power to achieve better values of PUE without affecting the working of IT equipment.

2.2 Cooling Energy Breakdown for Conventional CRAC

For a conventional Computer Room Air Conditioning (CRAC) Unit schematic is shown in the figure below. Hot exhaust air is cooled by refrigerant. The refrigerant vaporizes and is compressed by the chiller compressor to increase its temperature and pressure. This heat is then released into the cooling tower loop and condensed refrigerant is passed through expansion valve to once again absorb the heat from datacenter hall. When all this energy is broken down, we see that Chiller compressor accounts for the majority of the energy.

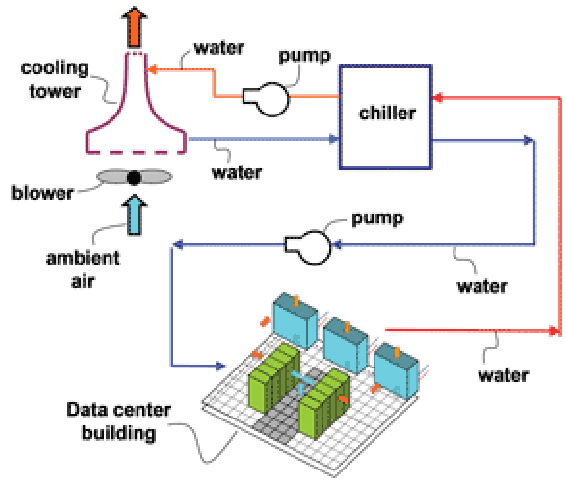


Figure 2-1: Schematic of data center employing CRAC

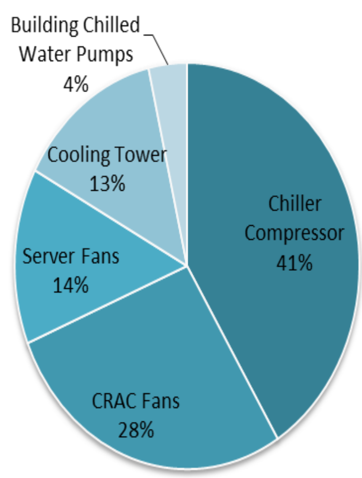


Figure 2-2: Cooling Energy Breakdown

We therefore can realize that chiller compressor accounts for the 41% of the cooling energy it consumes. We therefore need to look for ways that can help us reduce Cooling energy. We can do that by reducing Chiller compressor Energy or by eliminating it completely. If we increase the rack inlet temperature the chiller work will reduce but then again it poses questions on server reliability.

If we incorporate Evaporative Cooling, we can achieve required cooling with less energy requirements and therefore with much better PUE. With ASHRAE expanding its allowable envelope, the chances of using Evaporative Cooling as the primary form of cooling for most times during a year greatly increases. This allows us to be energy efficient for most duration of the year. In conditions where high values of Relative humidity pose a threat of contamination, we can use indirect evaporative cooling, where we can eliminate addition of moisture to keep the moisture content in acceptable limits.

2.3 Motivation

Kofi Annan once said “All of us have to share the earth’s fragile ecosystems and resources; each one of us has a role to play in preserving them. If we are to go on living together on this earth, we are all to be responsible for it”.

To be technologically advanced with least impact on nature should be every engineer’s goal. Water is an important resource for sustaining life on this planet. Accountability for water has to be taken into consideration for determining sustainability. There has to be a robust model for depending upon the sustainability of this method in terms of water consumption.

2.4 Objective

The Objective of this research is to develop thermodynamic models that can be relied upon to calculate the sustainability of the Evaporative Cooling at a particular location by determining Water consumption and pPUE (partial PUE) of a data center. Furthermore, depending on its geographic location the amount of water and cost of water varies and to come up with an analysis as to whether this method provides adequate cooling and at the same time is it sustainable enough to go ahead with this method

CHAPTER 3

PSYCHROMETRICS

3.1 Psychrometric Terminology

Dry-bulb Temperature: It is the temperature measured on thermometer exposed to air in a place sheltered from direct solar radiation. It is an indicator of heat content and is shown along the bottom axis of psychrometric chart.

Wet Bulb Temperature: It is a thermodynamic property of a mixture of air and vapor. It is measured by wrapping moist muslin around the bulb and exposing it to airflow. It is shown as slanted lines on psychrometric chart

Relative Humidity: It is the ratio of vapor pressure of moisture in the sample to the saturation pressure at the dry bulb temperature of the sample. It is shown as curved lines on psychrometric chart

Dew Point Temperature: It is defined as the saturation temperature of the moisture present in the sample of air. It is also that temperature at which vapor changes to liquid. It is shown on the vertical axis and increases as we go from bottom to top.

Specific Humidity: It is defined as the portion of mass of water vapor per unit mass of moist air.

Humidity Ratio: It is defined as the portion of mass of vapor per unit mass of dry air. It is represented on the vertical axis.

Specific Enthalpy: It is the sum of the internal energy of the moist air, including the heat of the air and vapor within. It is represented by slanted lines parallel to that of wet bulb temperature.

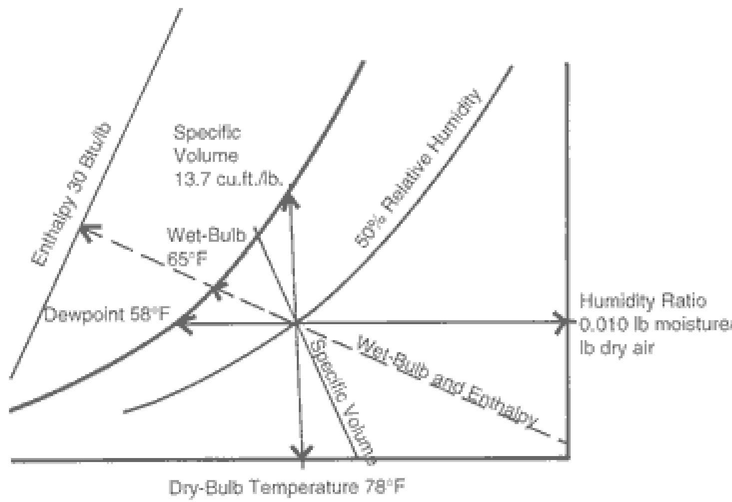


Figure 3-1: Psychrometric Chart

3.2 Processes on a Psychrometric Chart

All the processes that moist air can go through can easily be represented on a psychrometric chart. A psychrometric Chart is plotted for any particular value of pressure and other properties are calculated by co-relations and plotted on one chart. If we know any two properties of moist air we can figure out all other psychrometrics by just plotting the point on the chart. Depending on the end conditions of the moist air we can determine whether we require to heat, cool, humidify, dehumidify or any two process together need to be applied.

If we move to the right on the chart, we have undergone heating. Similarly, if we move to left on the chart we need to undergo cooling. Likewise, if we need to go to Right upper corner we need heating and humidifying simultaneously and similarly all other processes.

We need to look out for our air and determine whether it is in the permissible limits of the allowable envelope. If not we need to work on the air accordingly to bring the product air within the allowable limits specified by the Technical committee 9.9 of ASHRAE.

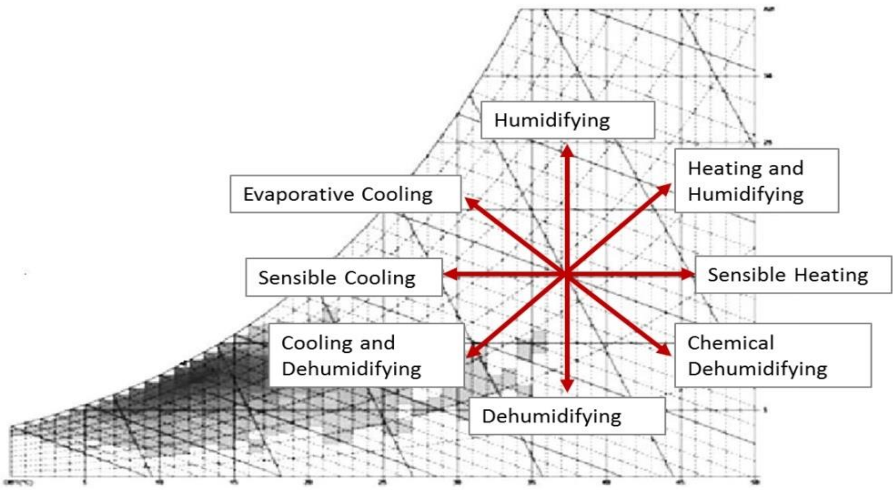


Figure 3-2: Various Processes on Psycrometric Chart

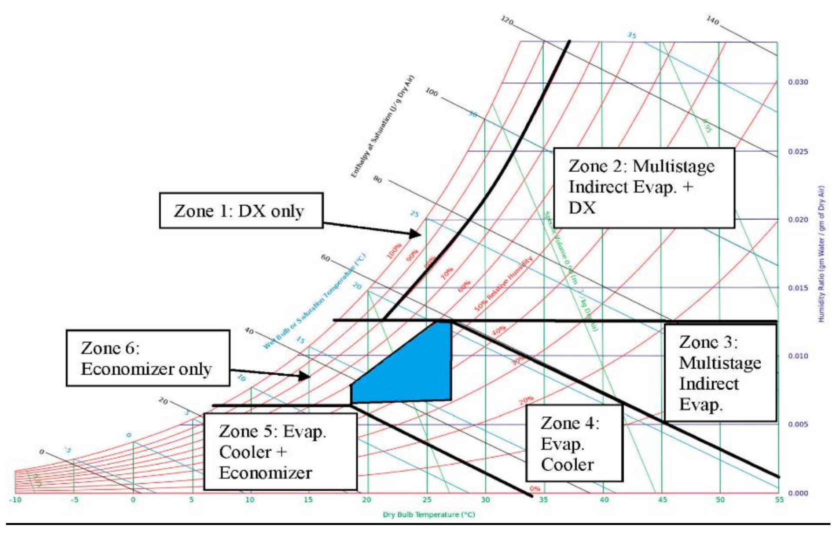


Figure 3-3: Cooling Processes to reach target zone

CHAPTER 4

DIRECT EVAPORATIVE COOLING

4.1 Introduction

Direct Evaporative Cooling is a way to cool the air by simply evaporating water into the air stream. It is also called adiabatic cooling because there is no heat addition or loss in the process. Simply energy from the hot outside air is used to evaporate the water which absorbs its latent heat of vaporization to change its phase. In doing so, the hot air reduces in its dry bulb temperature because of the energy lost by it to the water. The air also picks up the water evaporated in form of vapor and carries with it forward.

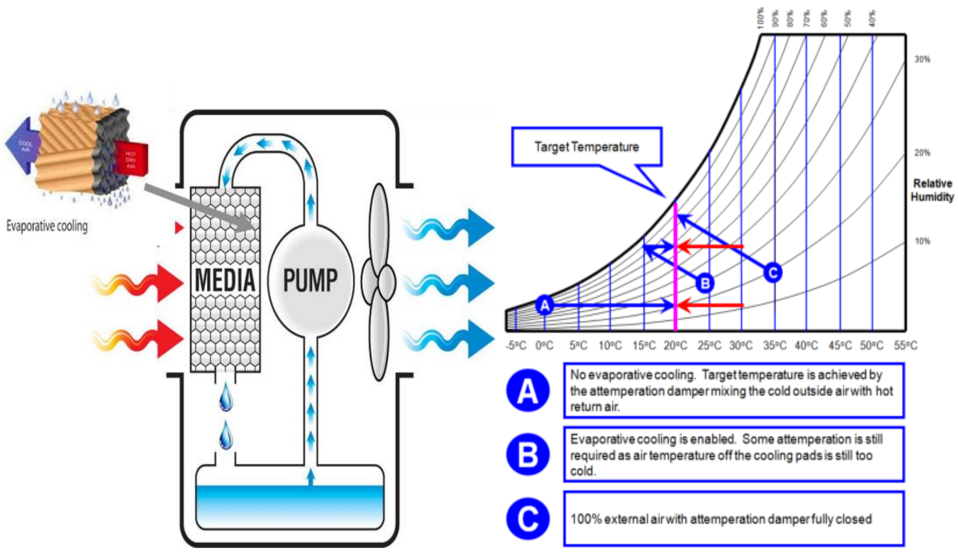
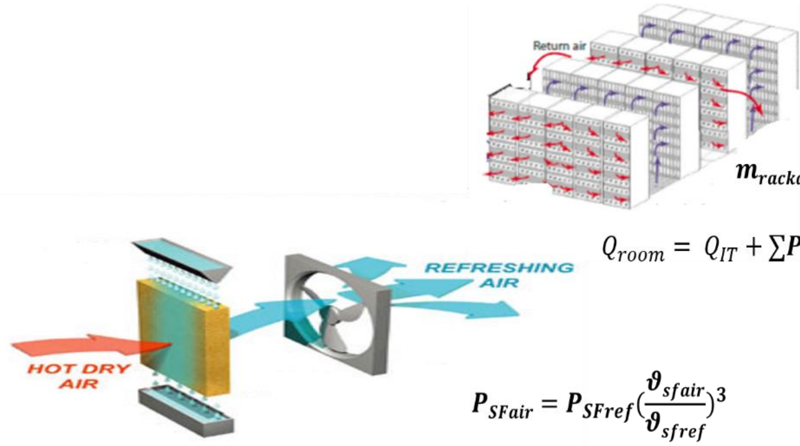


Figure 4-1: Direct Evaporative Cooling

Since, this is cooling with added humidity, the process moves on the upper left of the starting point as shown in figure. To achieve the target temperature it is mixed with return exhaust.

4.2 Energy Model



$$\boldsymbol{m}_{rackair} = \frac{\boldsymbol{Q}_{rack}}{\boldsymbol{C}_p(\Delta \boldsymbol{T}_{rackair})}$$

$$Q_{room} = Q_{IT} + \sum \boldsymbol{P}_{SYSFAN}(\boldsymbol{1} - \eta_{sysfan})$$

$$\boldsymbol{P}_{SFair} = \boldsymbol{P}_{SFref}\left(\frac{\boldsymbol{\vartheta}_{sfair}}{\boldsymbol{\vartheta}_{sfref}}\right)^3$$

$$\boldsymbol{P}_{CWpump} = \frac{\Delta \boldsymbol{p}_{CW}\boldsymbol{\vartheta}_{CW}}{\eta_{CWpump}}$$

Figure 4-2: Energy Model for Direct Evaporative Cooling

The development of this model has taken the approach of separating each level of a data center cooling system. The model is evaluated from the rack to the evaporative media as shown above.

4.2.1 Fan Model

The flow rates of the fans are calculated such that appropriate air is supplied to and extracted from the servers in a rack. Reference operating conditions for these fans have been obtained from fan and system curves from manufacturer data and literature. The fan curve can be transposed from the reference point to the required operating point using the fan law scaling for speed, pressure drop, and power of each component in the system.

4.2.2 Rack Model

The rack is composed of the IT hardware. The amount of heat dissipation from the racks is fixed, and is independent of fan power. There is additional heat dissipation due to the inefficiency of the system fan's operation. An inlet air temperature and air temperature rise across the rack is taken from published values for the model. The air inlet temperatures are varied from 15^{o}C to 45^{o}C in the intervals of 5^{o}C across the ASHRAE A4 envelope. The temperature rise across the rack is taken from published values which were experimentally found out. The mass flow rate of air required through the rack can hence be determined using the heat balance of

$$\boldsymbol{m_{rackair}} = \frac{\boldsymbol{Q_{rack}}}{\boldsymbol{C_p(\Delta T_{rackair})}}$$

The power demand for these server fans in operation can be determined through scaling of the fans' reference operating condition using the fan laws, The total power demand for the rack is the summation of server fans' power demand for all racks in the data center. An additional amount of heat dissipation from the inefficiency of this server fan's operation is taken into account with the IT heat dissipation for the total heat dissipation into the room determined by

$$Q_{room} = Q_{IT} + \sum \boldsymbol{P_{SYSFAN}(1 - \eta_{sysfan})}$$

4.2.3 Supply Fan

The duty of the Supply Fan is to supply the computer room and racks with a chilled air stream of a specified temperature, and at the flow rate, which will ensure sufficient provisioning for the racks. Using the mass flow rate of air required through the Supply Fan and the corresponding air temperatures, the volume flow rate of air can be calculated. Fan and system curves extracted from literature provided by the manufacturer for the particular supply fan.

$$\boldsymbol{P_{SFair} = P_{SFref}\left(\frac{\vartheta_{sfair}}{\vartheta_{sfref}}\right)^{3}}$$

4.2.4 Evaporative Media

The cold airstream to the supply fan is supplied by evaporative media. The hot and dry air flows over the media which is constantly sprayed with water with the help of a circulating pump. This hot and dry air, then evaporates the water thereby lowering its dry-bulb temperature enough to provide cooling to the Computer Room. The air at exit of the media contains relatively much higher humidity in comparison to the air that enters the media. The lowest dry-bulb temperature possible for the hot air to achieve is its own wet-bulb temperature where the air is completely saturated. The amount of cooling possible also depends on the saturation efficiency of the evaporative media.

$$T_B = T_A - \eta(T_A - T_{wb1}) \qquad Q_C = m\ Cp_a\ (T_A - T_B)$$

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

$$\boldsymbol{P_{CWpump} = \frac{\Delta p_{CW}\vartheta_{CW}}{\eta_{CWpump}}}$$

The amount of water evaporated can be accounted for, by measuring the humidity ratio gain at the end of the evaporative media and multiplying it by the mass flow rate of the air

$$m_{evap} = m_{rack} \times (w_2 - w_1)$$

A number of assumptions are implied for this model. These include the following assumptions:

1. Sufficient resources for cooling exists

2. Each rack in the computer room is operating at the same performance level with the same loading conditions of worst case scenario at 98% CPU utilization

3. No losses exist in the system, requiring that the full heat load dissipated within the data center system must be released to the ambient environment.

4. The facility has a total IT capacity of 25 kW.

4.3 Model Validation

The development of this model has been based on similar previous work conducted by Breen et al. For the model outlined in this research to be considered appropriate, the operating conditions determined using the current model must correlate with the parameters specified in the Baseline DC.

The Baseline DC is an IT Pod of 25 kW at outdoor facility of MESTEX, Dallas. It employs a hybrid direct/indirect evaporative cooling unit with following specifications

- AZTEC DIRECT/INDIRECT EVAPORATIVE UNIT
    - Model: ASC-15-2A11-00-HLS
    - SUPPLY FAN: 10HP, 6250 CFM
    - COOLING TOWER FAN:2 HP, 5000 CFM
    - CIRCULATING PUMP: 3 HP, 67GPM

- DIRECT PUMP: 1 HP, 16 GPM

- IT POD
  - 3 FULLY POPULATED RACKS WITH HP SE1102 SERVERS
  - TOTAL LOAD: 25 KW

Table 4-1: Model Validation

	BASELINE	**MODEL**	**%ERROR**
TOTAL LOAD	25KW	25KW	0
SUPPLYFAN POWER	0.791 kW	0.69kW	-12.5%
DIRECT PUMP	0.83 kW	0.75	-7.5%
COOLING POWER	1.621	1.44	-11%
PUE	1.065	1.057	-0.75%
WATER CONSUMPTION	Not Metered	0.13	-

4.3.1 Validation Parameters

The Baseline DC can be summarized as a typical air cooled data center, with Direct Evaporative Cooling. The data center computer room contains rack units of 25 kW heat dissipation. The supply air temperature to the room has been assumed as 25°C.

The computer room is served by 1 AZTEC unit. Data for the AZTEC unit were taken from manufacturer data for a suitable unit.

4.3.2 Results and Model Comparison

The results of the validation model run using the parameters defined above are compiled in Table 1, with comparison to the Baseline DC. The validation model shows good agreement against the Base DC across the range of parameters that are considered in Table 1. Some variation in the Supply Fan power demands exists for the Supply fan. As the power demand of the Supply Fan is also dependent on the pressure in the cold aisle, with no specific pressure difference specified for the Base DC, the variation here can be considered acceptable. The total cooling system power demand, illustrates an acceptable variation between the Baseline DC and Model DC. The model development outlined above has been shown to describe acceptable performance parameters for this model to be valid.

4.4 Servers Considered for Study

The servers considered for study are the 1.5 RU Open Compute Servers. A study on effects of Rack Inlet Temperature on the performance of a chiller based Cooling System was done experimentally. Values for individual components and all temperature measurements were taken during this experiment. The values for density and temperature rise across the racks at different rack inlet temperatures were taken from this experiments. The study done to validate the model and the study done to calculate the performance of data center were done on different servers. Models were validated using HP SE 1102 servers. Once validated, models hold good for all servers and hence to reduce complexity and save time experimentally calculated data was used to calculate the performance across ASHRAE A4 envelope.

Table 4-2: RIT vs ΔT

Rack Inlet Temperature	Delta T
15 °C	17.5 °C
20 °C	17 °C
25 °C	17.5 °C
30 °C	10.75 °C
35 °C	7.62 °C
40 °C	4.85 °C
45 °C	4 °C

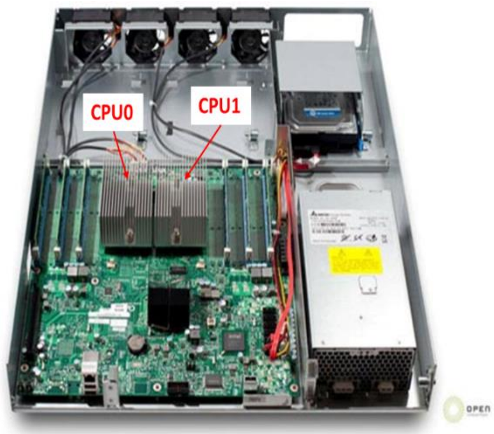


Figure 4-3: Open Compute Server

Using these values we can figure out the volumetric flow rate of air required to remove the heat dissipated from the server and accordingly find all the other parameters.

4.5 Performance across ASHRAE A4 Envelope

The server rack inlet temperature is varied from 15°C to 45°C in the interval of 5°C and performance of the data center is monitored for all parameters. 2 cities –San Jose and Dallas-Fort Worth are taken into consideration and calculated for performance in terms of pPUE and Water Consumption.

4.5.1 Economizer Mode

If the ambient temperature is below the set point, the air is mixed with the exhaust air in right proportions to reach the set point. During this mode water is not circulated over the media and the pump does not consume any power.

On occasion where ambient temperature is higher than the set point, the outside air is mixed with return exhaust air in such proportions that hot air flows over met media lowering its dry bulb temperature to the set point. The psychrometric values are calculated by in-built functions of MS-Excel.

4.5.1.1 Calculations

Calculations for pPUE and Water consumption has been made in following way,

Q= 25 kW; density = 1.185 kg/m^3 delta T = 17^{o}C

If ambient temperature is less than set point temperature, Direct Evaporative Cooling mode is not operated and Water consumption is Zero.

Therefore, m = 25/(1.05*17) = 1.46 kg/s = 1.46/0.0004719 = 2616 cfm

Psupplyfan = 7.5*(2616/6250)^3 = 0.54 kW Pdirectpump= 0.75 kW

Pcooling = 0.75 + 0.54 =1.29 kW

pPUE = (25 +1.29)/25 = 1.051

Humidity ratio at inlet = 0.01135 Humidity ratio at inlet = 0.015437

Evaporation loss = 1.46*(0.015437 – 0.01135) = 0.0005967kg/s = 0.021 m^3/hr

4.5.1.2 Results

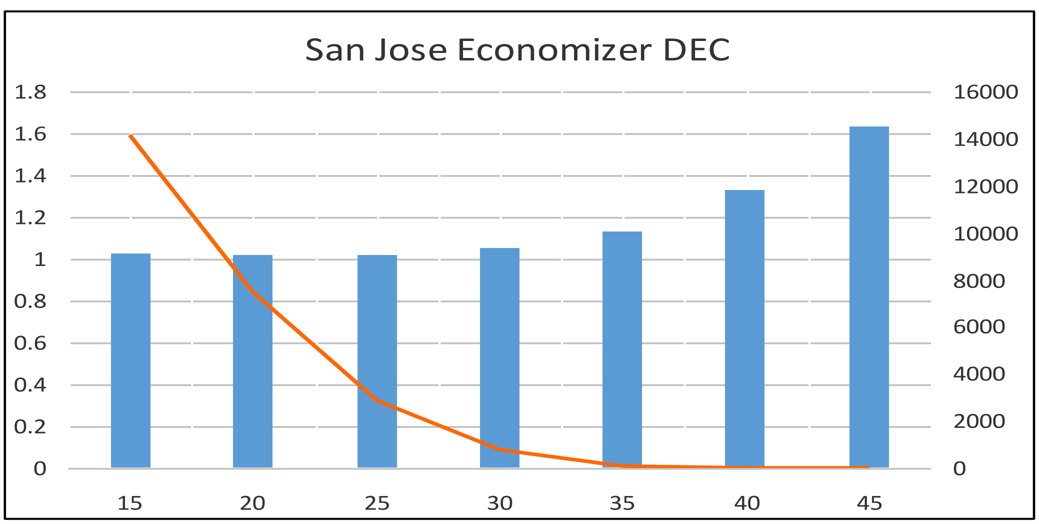
As rack inlet temperature increases from lower temperature to a higher temperature, the mass flow rate required to cool the server increases. Therefore, the power required by the supply fan increases which overall increases the Total cooling power. The economizer mode is operated for longer duration of the year as rack inlet temperature increases. For both the cities economizer mode varies due to the ambient conditions and therefore water consumption varies across both cities.

<u>San Jose</u>

Table 4-3: San Jose Economizer Mode

San Jose Economizer				
Temp	pPUE	Water(gal	Econ(hrs/\	Econ (%)
15	1.027	14125	4433	50.60502
20	1.017	7515	7219	82.40868
25	1.014	2888	8329	95.07991
30	1.048	776	8710	99.42922
35	1.127	55	8758	99.97717
40	1.329	0	8760	100
45	1.636	0	8760	100

Graph 4-1: (pPUE,WC) at different rack inlet temperatures for San Jose

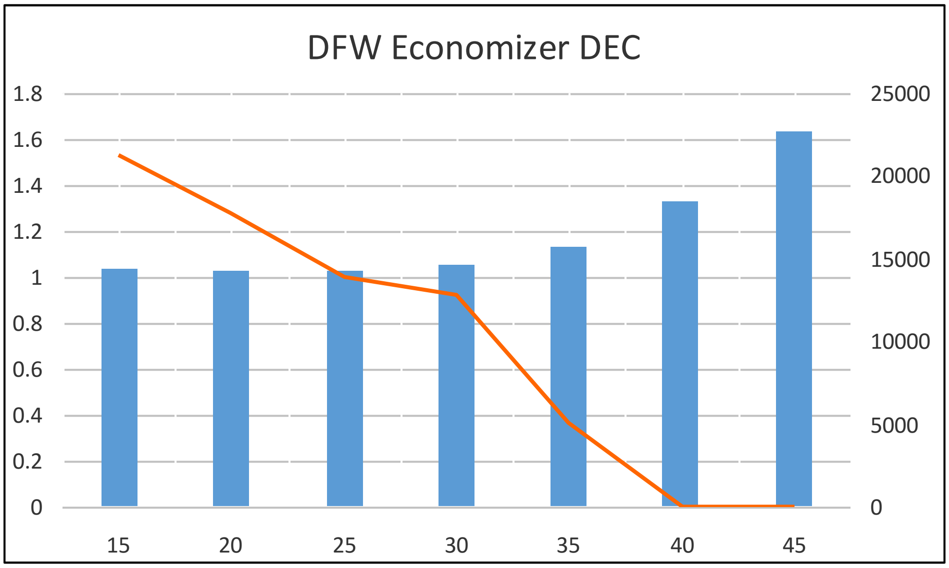


DFW

Table 4-4: DFW Economizer Mode

DFW Economizer				
Temp	pPUE	Water(gal)	Econ(hrs/yr)	Econ (%)
15	1.033	21243	4433	50.60502
20	1.028	17799	7219	82.40868
25	1.023	13850	8329	95.07991
30	1.052	12856	8710	99.42922
35	1.128	5107	8758	99.97717
40	1.329	38	8760	100
45	1.636	0	8760	100

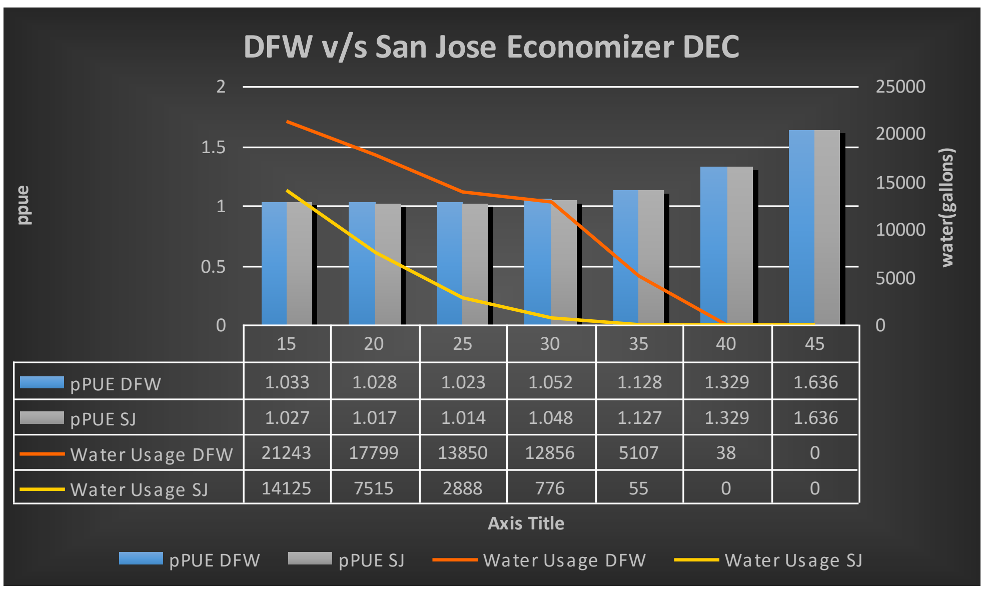
Graph 4-2: (pPUE,WC) at different rack inlet temperatures for DFW



San Jose V/S DFW Economizer

Comparing San Jose and DFW on terms of pPUE and Water consumption we can conclude that San Jose fairs better than its counterpart on all fronts. We also see an interesting observation that, both cities show their best pPUE at rack inlet temperature.

Graph 4-3: (pPUE,WC) at different rack inlet temperatures for both cities



CHAPTER 5

INDIRECT EVAPORATIVE COOLING

5.1 Introduction

Indirect Evaporative Cooling is a way to cool the air by sensibly cooling the air stream. It is the adiabatic evaporative cooling process that provides a stream of cool humid air which sensible cool the ambient hot air. In doing so, the hot air reduces in its dry bulb temperature because of the energy transfer to the secondary stream of evaporatively cooled air. The air also does not pick up any moisture.

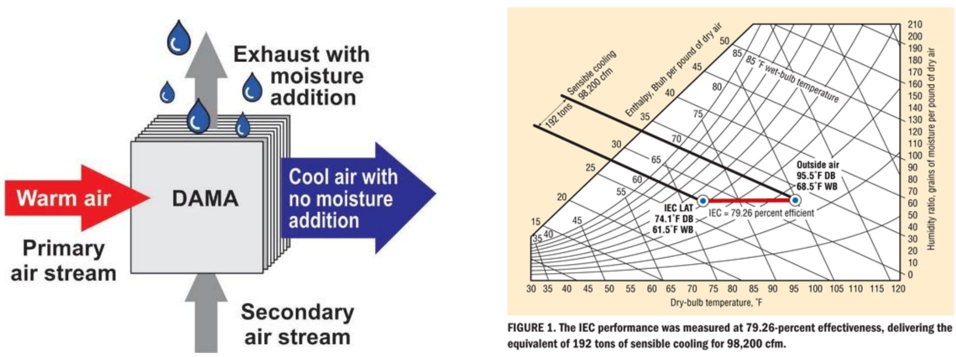


FIGURE 1. The IEC performance was measured at 79.26-percent effectiveness, delivering the equivalent of 192 tons of sensible cooling for 98,200 cfm.

Figure 5-1: Indirect Evaporative Cooling

Since, this is cooling with no added humidity, the process moves on the left of the starting point as shown in figure. To achieve the target temperature it is mixed with return exhaust if needed.

5.2 Energy Model

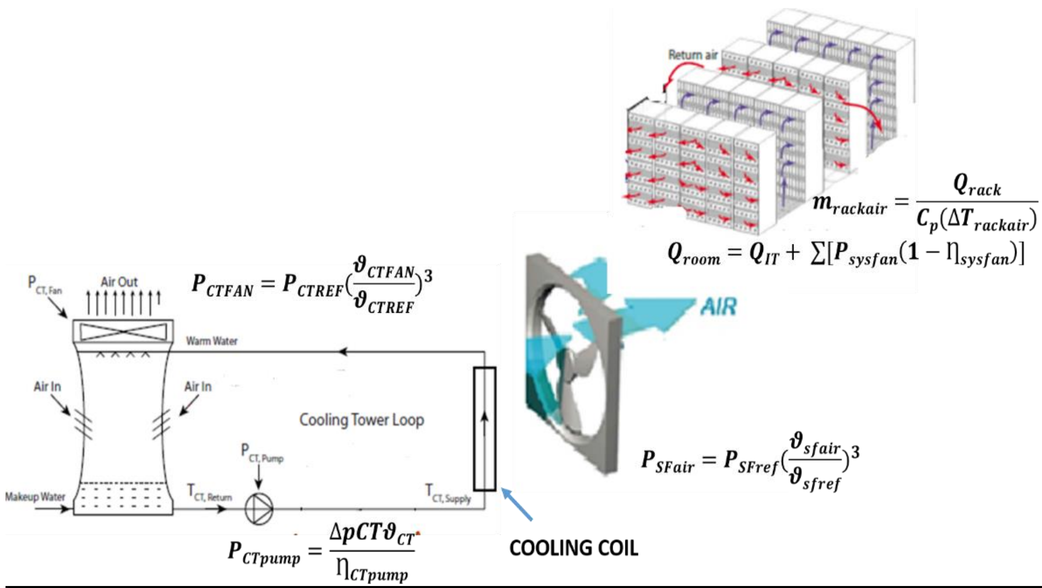


Figure 5-2: Energy Model for Indirect Evaporative Cooling

The development of this model has taken the approach of separating each level of a data center cooling system. The model is evaluated from the rack to the cooling tower as shown above.

5.2.1 Fan Model

The flow rates of the fans are calculated such that appropriate air is supplied to and extracted from the servers in a rack. Reference operating conditions for these fans have been obtained from fan and system curves from manufacturer data and literature. The fan curve can be transposed from the reference point to the required operating point using the fan law scaling for speed, pressure drop, and power of each component in the system.

5.2.2 Rack Model

The rack is composed of the IT hardware. The amount of heat dissipation from the racks is fixed, and is independent of fan power. There is additional heat dissipation due to the ineffi- ciency of the system fan's operation. An inlet air temperature and air temperature rise across the rack is taken from published values for the model. The air inlet temperatures are varied from 15^{o}C to 45^{o}C in the intervals of 5^{o}C across the ASHRAE A4 envelope. The temperature rise across the rack is taken from published values which were experimentally found out. The mass flow rate of air required through the rack can hence be determined using the heat balance of

$$\boldsymbol{m}_{rackair} = \frac{\boldsymbol{Q}_{rack}}{\boldsymbol{C}_{p}(\Delta \boldsymbol{T}_{rackair})}$$

The power demand for these server fans in operation can be determined through scaling of the fans' reference operating condition using the fan laws, The total power demand for the rack is the summation of server fans' power demand for all racks in the data center. An additional amount of heat dissipation from the inefficiency of this server fan's operation is taken into account with the IT heat dissipation for the total heat dissipation into the room determined by

$$Q_{room} = Q_{IT} + \sum \boldsymbol{P}_{SYSFAN}(\boldsymbol{1} - \eta_{sysfan})$$

5.2.3 Supply Fan

The duty of the Supply Fan is to supply the computer room and racks with a chilled air stream of a specified temperature, and at the flow rate, which will ensure sufficient provisioning for the racks. Using the mass flow rate of air required through the Supply Fan and the corresponding air temperatures, the volume flow rate of air can be calculated. Fan and system curves extracted from literature provided by the manufacturer for the particular supply fan.

$$\boldsymbol{P_{SFair} = P_{SFref}(\frac{\vartheta_{sfair}}{\vartheta_{sfref}})^3}$$

5.2.4 Cooling Tower

The cold air stream to the supply fan is supplied by passing the ambient air over the cooling coil which is supplied by chilled water by the cooling tower. The hot air flows over the media which is constantly sprayed with water with the help of a circulating pump. The water is cooled in the cooling tower by rejecting its heat to the atmosphere. The amount of cooling possible also depends on the saturation efficiency of the evaporative media.

$$Q_C = m\ Cp_a\ (T_A - T_B)$$

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

$$\boldsymbol{P_{CWpump}} = \frac{\Delta \boldsymbol{p_{CW}\vartheta_{CW}}}{\eta_{\boldsymbol{CWpump}}}$$

The amount of water evaporated can be accounted for, by measuring the humidity ratio gain at the end of the evaporative media and multiplying it by the mass flow rate of the air

$$\boldsymbol{m_{evap} = m_{CT} \times (w_2 - w_1)}$$

A number of assumptions are implied for this model. These include the following assumptions:

1. Sufficient resources for cooling exists

2. Each rack in the computer room is operating at the same performance level with the same loading conditions of worst case scenario at 98% CPU utilization

3. No losses exist in the system, requiring that the full heat load dissipated within the data center system must be released to the ambient environment.

4. The facility has a total IT capacity of 25 kW.

5.3 Model Validation

The development of this model has been based on similar previous work conducted by Breen et al. For the model outlined in this research to be considered appropriate, the operating conditions determined using the current model must correlate with the parameters specified in the Baseline DC.

The Baseline DC is an IT Pod of 25 kW at outdoor facility of MESTEX, Dallas. It employs a hybrid direct/indirect evaporative cooling unit with following specifications

- AZTEC DIRECT/INDIRECT EVAPORATIVE UNIT
    - Model: ASC-15-2A11-00-HLS
    - SUPPLY FAN: 10HP, 6250 CFM
    - COOLING TOWER FAN:2 HP, 5000 CFM
    - CIRCULATING PUMP: 3 HP, 67GPM
    - DIRECT PUMP: 1 HP, 16 GPM
- IT POD
    - 3 FULLY POPULATED RACKS WITH HP SE1102 SERVERS
    - TOTAL LOAD: 25 KW

Table 5-1: Model validation

	BASELINE	MODEL	%ERROR
TOTAL LOAD	25KW	25KW	0
SUPPLY FAN POWER	0.791 kW	0.71 kW	-10.25%
COOLING TOWER FAN POWER	0.039 kW	1.5 kW*	*Modelled for fixed fan speed for Cooling Tower
CIRCULATING PUMP POWER	2.20	2.25kW	+2%
PUE(instantaneous)	1.12	1.17	+5%
WATER CONSUMPTION (instantaneous)	Not metered	0.3 GPM	-

5.3.1 Validation Parameters.

The Baseline DC can be summarized as a typical air cooled data center, with Direct Evaporative Cooling. The data center computer room contains rack units of 25 kW heat dissipation. The supply air temperature to the room has been assumed as 25°C. The computer room is served by 1 AZTEC unit. Data for the AZTEC unit were taken from manufacturer data for a suitable unit.

5.3.2 Results and Model Comparison

The results of the validation model run using the parameters defined above are compiled in Table 1, with comparison to the Baseline DC. The validation model shows good agreement against the Base DC across the range of parameters that are considered in Table 1. Some variation in the Supply Fan power demands exists for the Supply fan. As the power demand of the Supply Fan is also dependent on the pressure in the cold aisle, with no specific pressure difference specified for the Base DC, the variation here can be considered acceptable. The total cooling system power demand, illustrates an acceptable variation between the Baseline DC and Model DC. The model development outlined above has been shown to describe acceptable performance parameters for this model to be valid.

5.4 Servers considered for study

The servers considered for study are the 1.5 RU Open Compute Servers. A study on effects of Rack Inlet Temperature on the performance of a chiller based Cooling System was done experimentally. Values for individual components and all temperature measurements were taken during this experiment. The values for density and temperature rise across the racks at different rack inlet temperatures were taken from this experiments. The study done to validate the model and the study done to calculate the performance of data center were done on different servers. Models were validated using HP SE 1102 servers. Once validated, models hold good for all servers and hence to reduce complexity and save time experimentally calculated data was used to calculate the performance across ASHRAE A4 envelope.

Table 5-2: RIT vs ΔT

Rack Inlet Temperature	Delta T
15 °C	17.5 °C
20 °C	17 °C
25 °C	17.5 °C
30 °C	10.75 °C
35 °C	7.62 °C
40 °C	4.85 °C
45 °C	4 °C

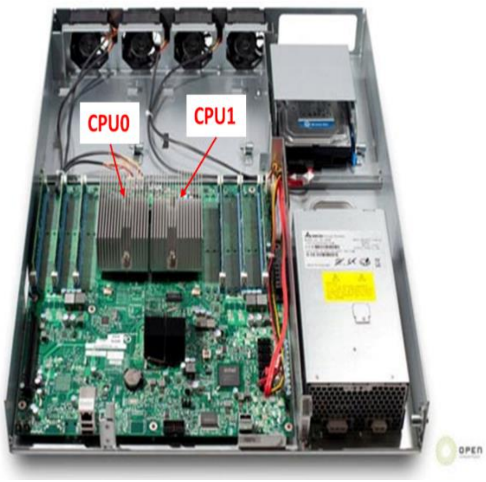


Figure 5-3: Open Compute Server

Using these values we can figure out the volumetric flow rate of air required to remove the heat dissipated from the server and accordingly find all the other parameters.

5.5 Performance across ASHRAE A4 Envelope

The server rack inlet temperature is varied from 15°C to 45°C in the interval of 5°C and performance of the data center is monitored for all parameters. 2 cities –San Jose and Dallas-Fort Worth are taken into consideration and calculated for performance in terms of pPUE and Water Consumption.

5.5.1 Economizer Mode

If the ambient temperature is below the set point, the air is mixed with the exhaust air in right proportions to reach the set point. During this mode water is not circulated over the media and the pump does not consume any power.

On occasion where ambient temperature is higher than the set point, the outside air is mixed with return exhaust air in such proportions that hot air flows over met media lowering its dry bulb temperature to the set point. The psychrometric values are calculated by in-built functions of MS-Excel.

5.5.1.1 Calculations

Calculations for pPUE and Water consumption has been made in following way,

Q= 25 kW; density = 1.185 kg/m^3 delta T = 17^{o}C

If ambient temperature is less than set point temperature, Indirect Evaporative Cooling mode is not operated and Water consumption is Zero.

Whenever Indirect mode is in operation, the cooling tower is put into operation and the fan speed of cooling tower is fixed in our calculations.

Therefore, m = 25/(1.05*17) = 1.46 kg/s = 1.46/0.0004719 = 2616 cfm

Psupplyfan = 7.5*$(2616/6250)^3$ = 0.54 kW

Pctfan = 1.5*$(5000/5000)^3$ = 1.5 kW Pcirculatingpump= 2.25 kW

Pcooling = 2.25+ 1.5 + 0.54 =4.24 kW

pPUE = (25 +4.24)/25 = 1.169

Humidity ratio at inlet = 0.01135 Humidity ratio at inlet = 0.015437

Evaporation loss = 5000*0.0004719*(0.015437 – 0.01135) = 0.00965kg/s = 0.035 m^3/hr

5.5.1.2 Results

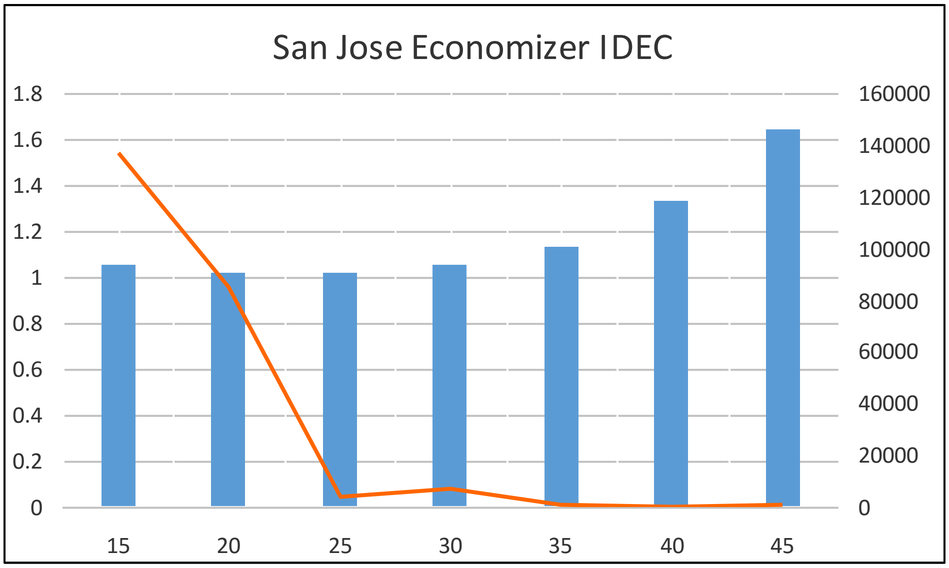
As rack inlet temperature increases from lower temperature to a higher temperature, the mass flow rate required to cool the server increases. Therefore, the power required by the supply fan increases which overall increases the Total cooling power. The economizer mode is operated for longer duration of the year as rack inlet temperature increases. For both the cities economizer mode varies due to the ambient conditions and therefore water consumption varies across both cities.

1. San Jose

Table 5-3: San Jose in Economizer Mode

SAN JOSE ECONOMIZER				
Inlet Temp	pPUE	W.C in gallons(annual)	Economizer Mode(Hrs/year)	Economizer Mode (%)
15	1.05	137105	4433	50.60502283
20	1.02	84800	7219	82.4086758
25	1.015	3700	8329	95.07990868
30	1.05	7250	8712	99.45205479
35	1.13	430	8758	99.97716895
40	1.33	0	8760	100
45	1.64	0	8760	100

Graph 5-1: (pPUE,WC) at different rack inlet temperatures

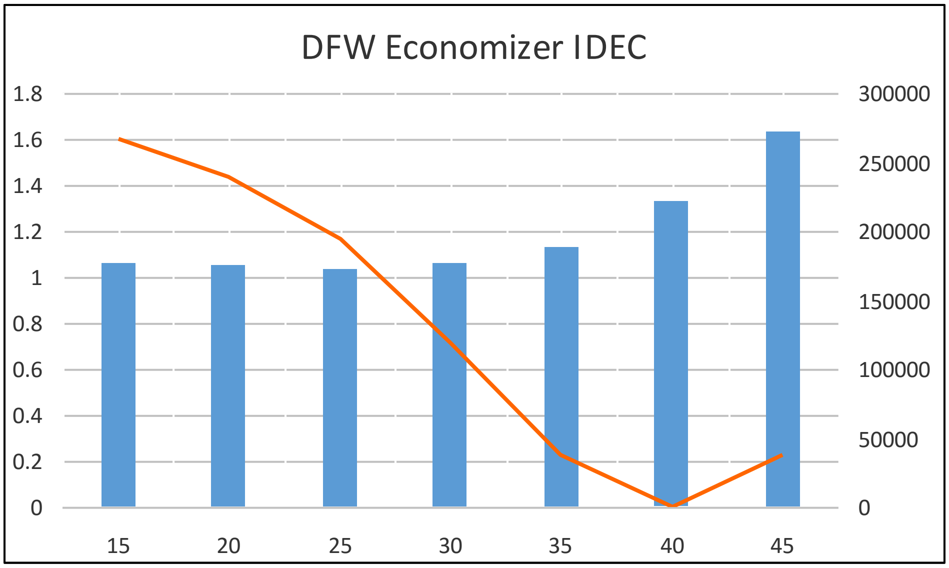


2. DFW

Table 5-4: DFW in Economizer Mode

DFW ECONOMIZER MODE				
Inlet Temp	pPUE	W.C in gallons(annual)	Economizer Mode(Hrs/year)	Economizer (%)
15	1.06	265954	2986	34.08675799
20	1.048	239288	4318	49.29223744
25	1.036	194449	5998	68.47031963
30	1.057	118895	7643	87.24885845
35	1.129	37708	8523	97.29452055
40	1.33	252	8760	100
45	1.6387	37708	8760	100

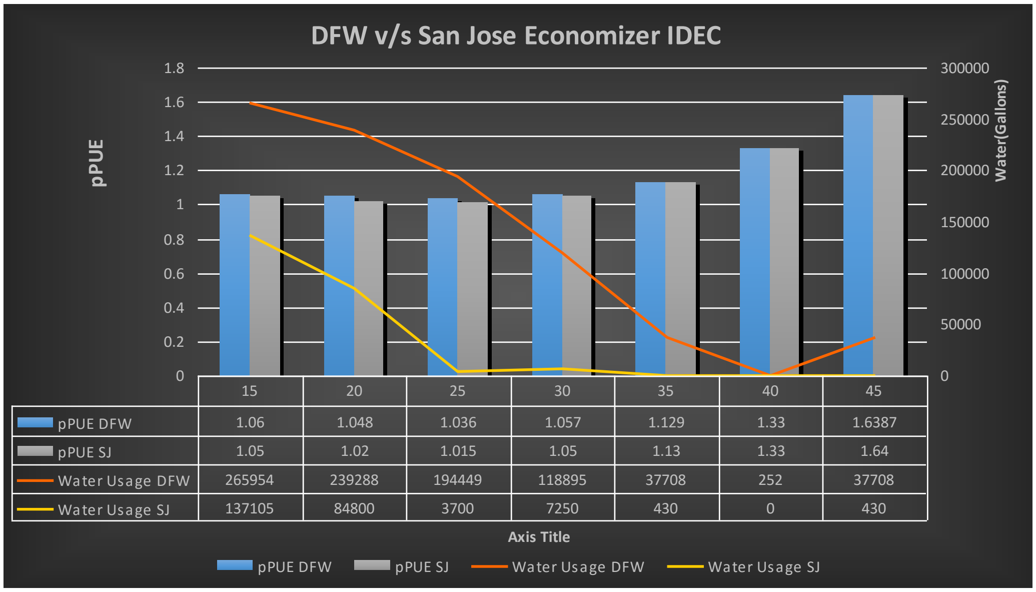
Graph 5-2: (pPUE,WC) at different rack inlet temperatures



San Jose V/S DFW Economizer

Comparing San Jose and DFW on terms of pPUE and Water consumption we can conclude that San Jose fairs better than its counterpart on all fronts. We also see an interesting observation that, both cities show their best pPUE at rack inlet temperature.

Graph 5-3: (pPUE,WC) at different rack inlet temperatures for both cities



CHAPTER 6

CONCLUSIONS AND FUTURE WORK

Using the developed models we can compute the pPUE of a data center and Water consumption at any location, provided we have the local weather data. Since the model is validated with an error in pPUE of less than 1% for Direct Evaporative Cooling, we can have a good agreement between the measured and calculated values. The Water consumption, however was not metered and need to be validated.

In future, this model can be improved for Indirect Evaporative Cooling, by calculating the variation of cooling tower fan needed in accordance with the weather conditions. In doing so, the evaporation of water will be less at lesser speeds and more accurate.

References

1.) 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 andTheses pp. 145.2015

2.) https://en.wikipedia.org/wiki/Psychrometrics

3.) From chip to cooling tower data center modeling: Part I Influence of server inlet temperature and temperature rise across cabinet
T.J. Breen, Ed.J. Walsh, J. Punch, A.J. Shah, Cullen Bash
DOI: 10.1109/ITHERM.2010.5501421 Conference: Thermal and Thermomechanical Phenomena in Electronic Systems (ITherm), 2010 12th IEEE Intersociety Conference on
Source: IEEE Xplore

4.) Images Courtesy : www.google.com

Biographical Information

Niravkumar Dodia received his bachelor's degree (BE) in Aeronautical Engineering from the Aeronautical Society Of India, New Delhi in 2013. He qualified his Master of Science Degree in Mechanical Engineering at the University of Texas At Arlington in January 2014.

During his master's program he was a part of the Sustainable cooling team and his work was focused on Developing and Validating Thermodynamic Models for Evaporative Cooling Employed in Data centers. He has also assisted his team mates on other projects such as Quantifying Airflow through a server experiment