DEVELOPING AND VALIDATING ENERGY MODELS FOR INDIRECT AND DIRECT EVAPORATIVE COOLING USED IN DATACENTERS

by

RAJESH KASUKURTHY

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Abstract

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Rajesh Kasukurthy, MS

The University of Texas at Arlington, 2016

Supervising Professor: Dereje Agonafer

In the present day to day life, Internet of things is everything, i.e., we need internet and telecommunication for almost every task of our life and these require data from data centers. These data centers provide various facilities like data processing, storage, and transmission, and maintenance, operations etc., for performing these tasks, a huge amount of power is consumed by the data center, which in turn generates large amount of heat. As these data centers are to be made operational throughout the year, cooling of data centers is of utmost importance. These cooling systems like air conditioners, economizers and evaporative cooling systems also require a considerable amount of power. Evaporative Cooling is one of the effective methods available for cooling data centers.

Developing and validating a thermodynamical model for evaporative cooling used in data centers. Environmental data such as temperature, humidity etc., for a particular location is used for computing the cooling capacity and performance (PUE and water consumption) of the data centers. For the purpose of validation, the actual data from the existing evaporative cooling unit is compared to the computed data from our model. The type of evaporative cooling system suitable for particular environmental conditions can also be known from this model.

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Chapter 1

Introduction

A data center is a facility that house the IT equipment used for data processing, storage, and transmission. Data that is used in day to day activities for example Internet of Thongs, a thing, in the Internet of Things, can be a person with a heart monitor implant, a farm animal with a biochip transponder, an automobile that has built-in sensors to alert the driver when tire pressure is low or any other natural or man-made object that can be assigned an IP address and provided with the ability to transfer data over a network [1]. So these data centers are kept secured and tightly controlled environment to make it running throughout the year. To maintain the controlled environment a lot of energy is used by the air conditioning systems.

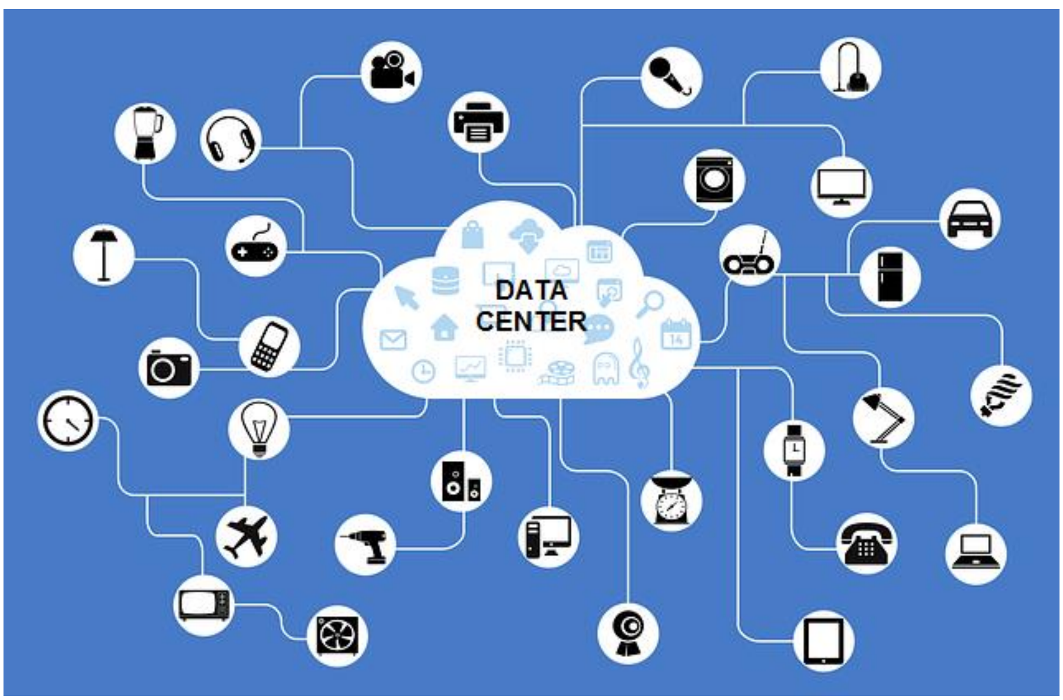


Figure 1-1 Representation of Internet of Things linked to data center [2].

1.1 Data Center Cooling

Data center is organized such that a numerus amount of electronic hardware can be cooled in an efficient manner. Several devises called servers are stacked in rows of several racks in a datacenter. The amount of heat liberated from servers which are operating continuously at all times 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. Figure 1-2 [3] shows the cold air flow in data center.

Figure 1-2 Schematic diagram of flow of air in cooling data center.

There are different types of mechanisms employed in cooling data center.

- Oil immersion cooling
    - Chilled oil supply cooling
    - Two stage liquid phase change cooling
- air cooling
    - free air cooling
    - Refrigerated air cooling

- Economizer cooling
- Evaporative cooling

1.1.1 Oil immersion cooling

In oil immersion cooling servers are immersed in the oil and the oil is being circulated to cool the servers. The heat exchange takes place through convection between the oil the servers. Figure 1-3 [4] shows process of oil immersion cooling.

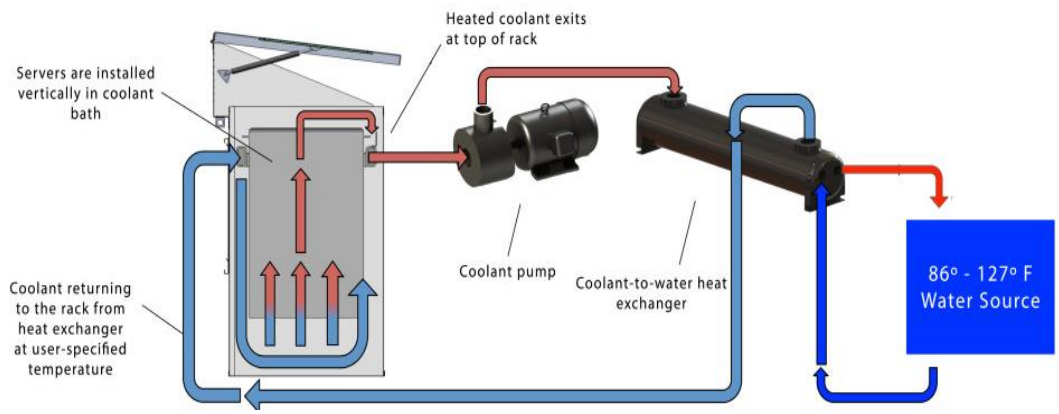


Figure 1-3 Diagram showing the processes of oil immersion cooling.

There is other type of oil cooling where oil changes its phase to cool the servers. In this type of cooling volatile liquids are used. Natural or forced convection is used to circulate the oil in phase change cooling. Figure 1-4 [5] shows the two phase liquid cooling

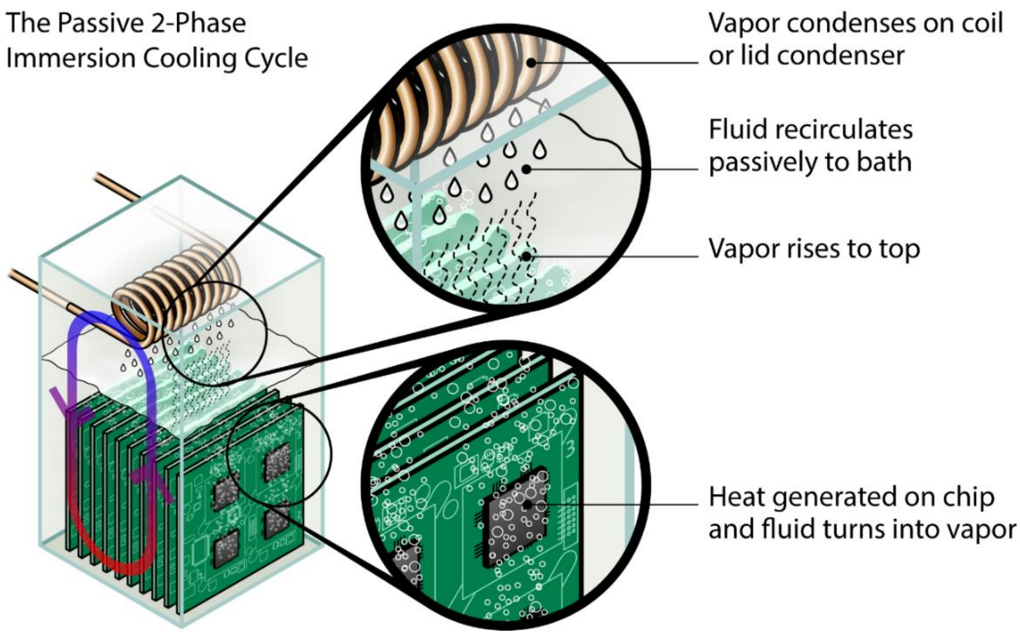


Figure 1-4 Diagram showing the processes of 2 stage liquid phase change cooling.

1.1.2 Air cooling

Traditional Data centers employ Computer Room Air Conditioning (CRAC) Units. The hot exhaust air is returned to the CRAC where heat exchanger occurs between air-to-refrigerant which cools the air and supply the air to cold aisle of the servers. The evaporated refrigerant goes to the compressor to increase its pressure and temperature. This compressed refrigerant transfers heat to the Cooling tower water there by condensed to liquid then it is send to the evaporator where it is expanded to absorb heat from the hot exhaust air. The absorbed heat by the water is rejected to the atmosphere with the help of cooling tower. In the cycle most of the energy provided for the cooling system is used by the compressor. So to replace the compressor several new cycles and cooling systems are designed like economizers.

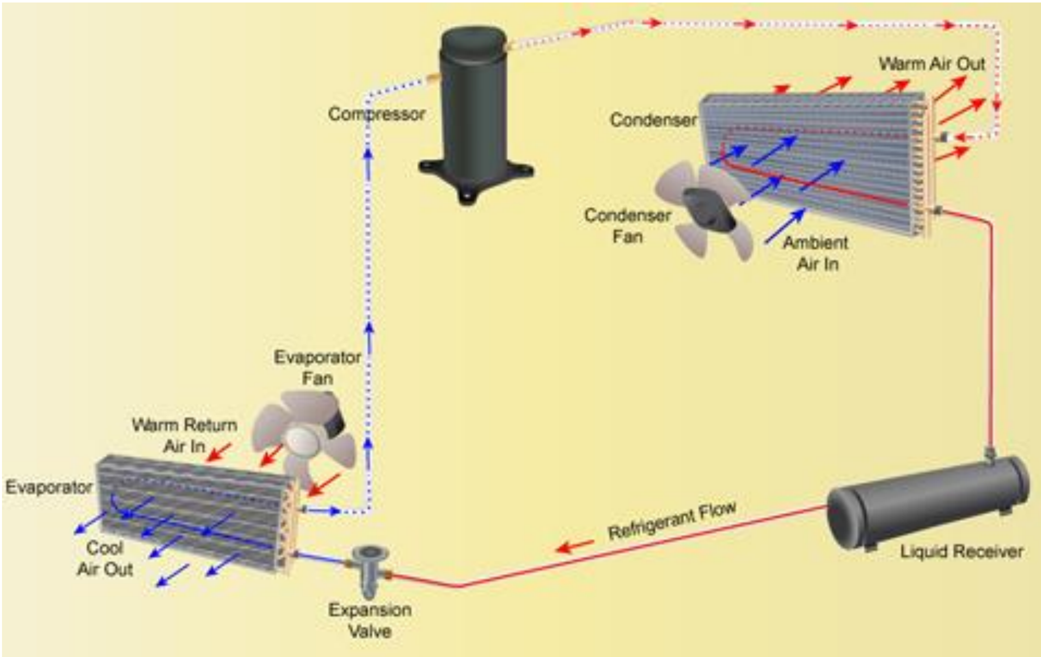


Figure 1-5 Layout diagram of stranded CRAC

Figure 1-5 [6] shows the compressor refrigeration air cooling. Air Side Economization is also being employed in data centers because it is efficient. Whenever ambient air conditions are favorable to the data centers such as dry and wet bulb temperature, and humidity, the outside air is directly fed to the data center thereby it uses no energy to cool the air again. This gives better energy efficiency. But this involves the risk of contamination of the hardware so prevention measures have to be adopted by the data center. The figure below shows how the economizer works.

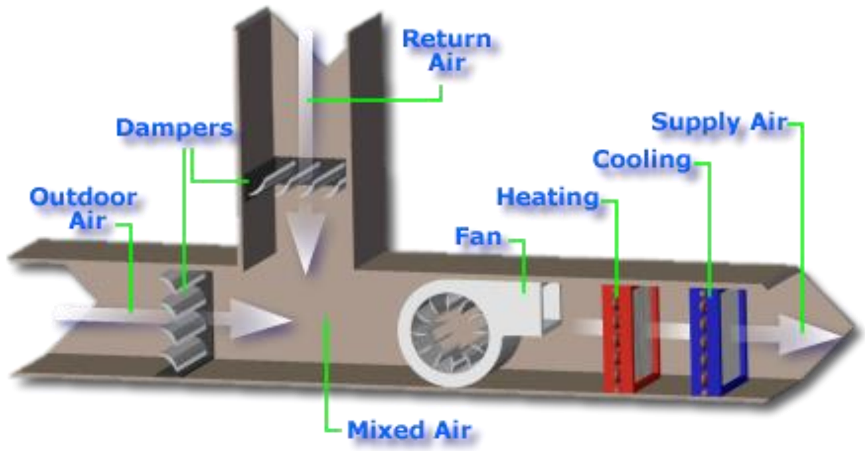


Figure 1-6 Air flow in economizer

Evaporative cooling is means of cooling the air by evaporating water into the air stream to lower the dry and wet bulb temperature of the air. However, the moisture content of the air is increased. Due to high moisture content in the air there is a much higher risk of contamination. So measures are need to be adopted to prevent corrosion and other forms of contamination. The energy efficiency is higher than compressor air cooling.

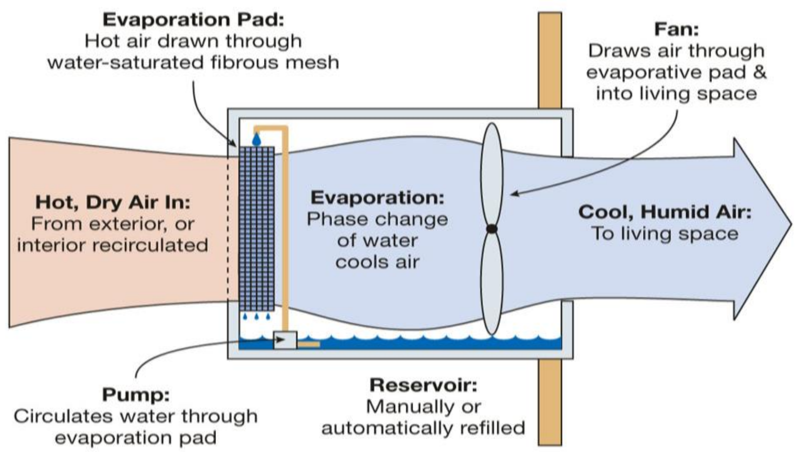


Figure 1-7 Evaporative cooling

Chapter 2

Psychrometry

2.1 Psychrometric Terminology

These definitions are preexisting and are been restated or stated from different sources [7], [8] and [9].

Dry-Bulb Temperature (DBT): It is the temperature of the air as measured by a standard thermometer or other temperature measuring instruments. It is an indicator of heat content and is shown along the bottom axis of psychrometric chart.

Wet-Bulb Temperature (WBT): It is temperature measured by wrapping moist muslin around the thermometer bulb and exposing it to airflow. It is shown as slanted lines on psychrometric chart.

Specific or Absolute humidity (ω): It is defined as the ratio of the mass of water vapor to the mass of dry air in a given volume of the mixture. It is represented on the vertical axis.

Relative humidity (Φ): It is defined as the ratio of the mole fraction of water vapor in moist air to mole fraction of water vapor in saturated air at the same temperature and pressure. It is shown as curved lines on psychrometric chart

Dew-point temperature (DPT): If unsaturated moist air is cooled at constant pressure, then the temperature at which the moisture in the air begins to condense is known as dew-point temperature of air.

Degree of saturation (μ): The degree of saturation is the ratio of the humidity ratio to the humidity ratio of a saturated mixture at the same temperature and pressure.

Enthalpy (s): The enthalpy of moist air is the sum of the enthalpy of the dry air and the enthalpy of the water vapor. Enthalpy values are always based on some reference value. For moist air, the enthalpy of dry air is given a zero value at 0ºC, and for water vapor the enthalpy of saturated water is taken as zero at 0ºC. It is represented by slanted lines parallel to that of wet bulb temperature.

Specific volume (υ): The specific volume is defined as the number of cubic meters of moist air per kilogram of dry air. From perfect gas equation since the volumes occupied by the individual substances are the same, the specific volume is also equal to the number of cubic meters of dry air per kilogram of dry air.

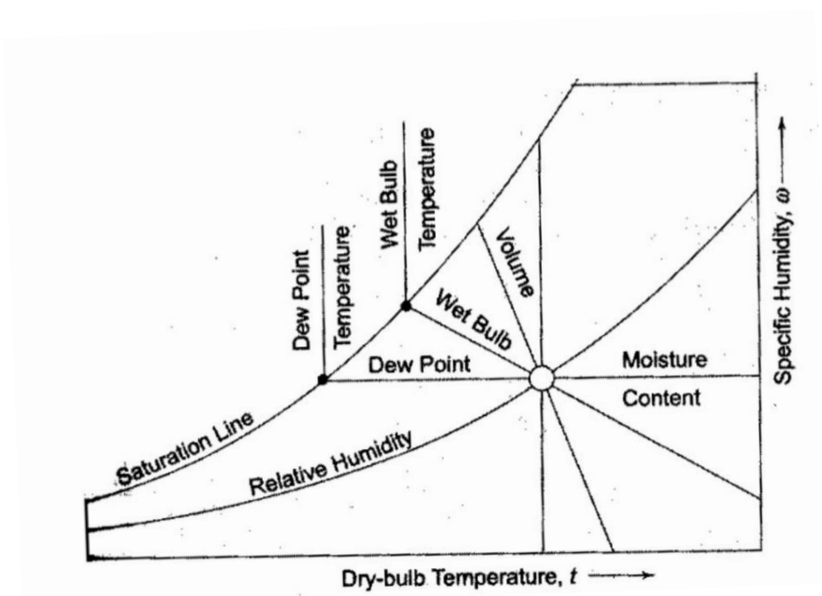


Figure 2-1 Psychrometric chart out line

2.2 Psychrometric Chart

A Psychrometric chart graphically represents the thermodynamic properties of moist air. Standard psychrometric charts are bounded by the dry-bulb temperature line and the vapor pressure or humidity ratio. The Left Hand Side of the psychrometric chart is bounded by the saturation line. Figure shows the schematic of a psychrometric chart. Psychrometric charts are readily available for standard barometric pressure of 101.325 kPa at sea level and for normal temperatures (0-50^{o}C). ASHRAE has also developed psychrometric charts for other temperatures and barometric pressures (for low temperatures: -40 to 10^{o}C, high temperatures 10 to 120^{o}C and very high temperatures 100 to 120^{o}C).

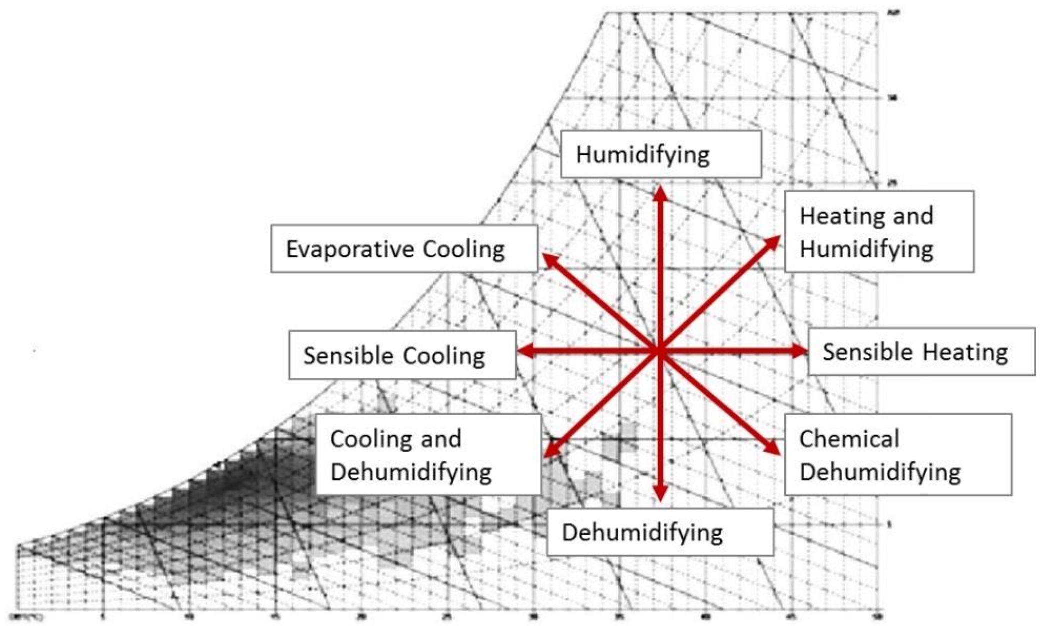


Figure 2-2 Processes in Psychrometric chart

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

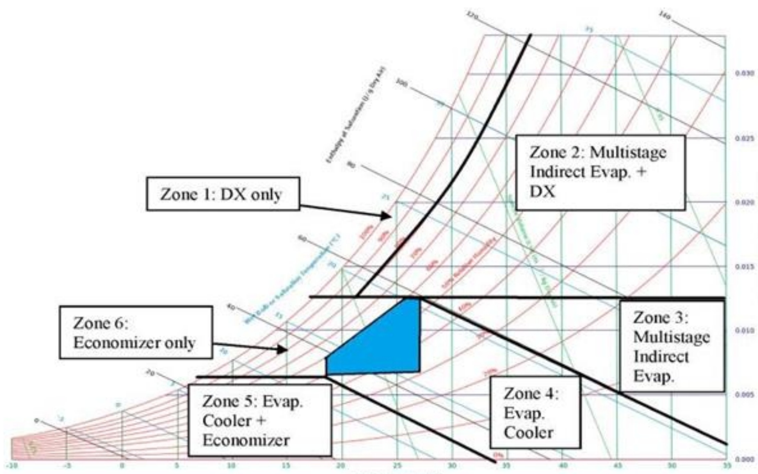


Figure 2-3 Psychrometric chart representing type of cooling system used for cooling

2.3 PUE

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. Figure 3-6 [5] shows the power distribution among the data center.

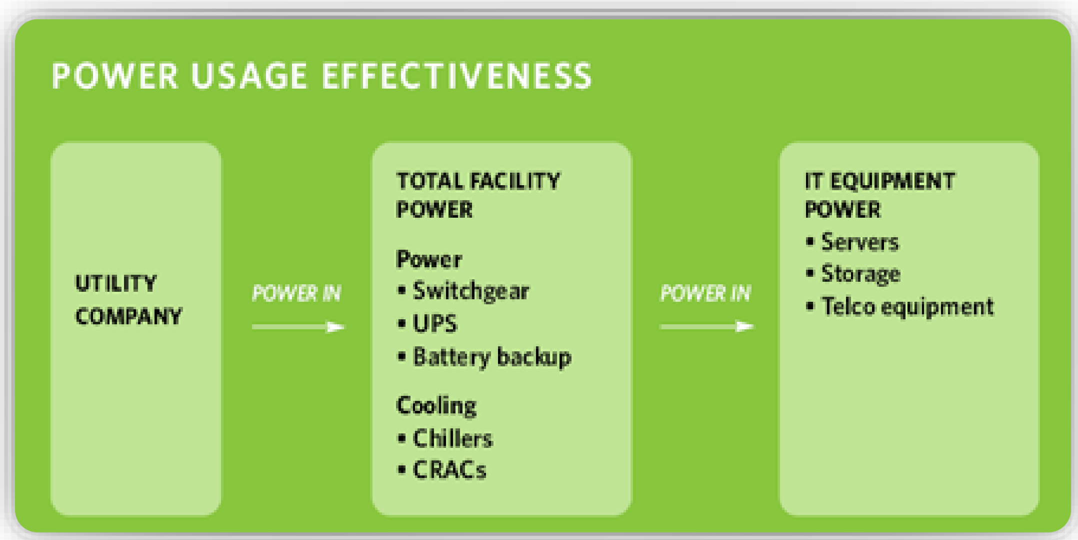


Figure 2-4 Power distribution chart

Power Usage Effectiveness (PUE):

$$\mathrm{PUE} = \frac{\text{Total Facility Power}}{\text{Compute Power}} = \frac{\text{Compute Power+Cooling Power+Misc. Power}}{\text{Compute Power}}$$

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.

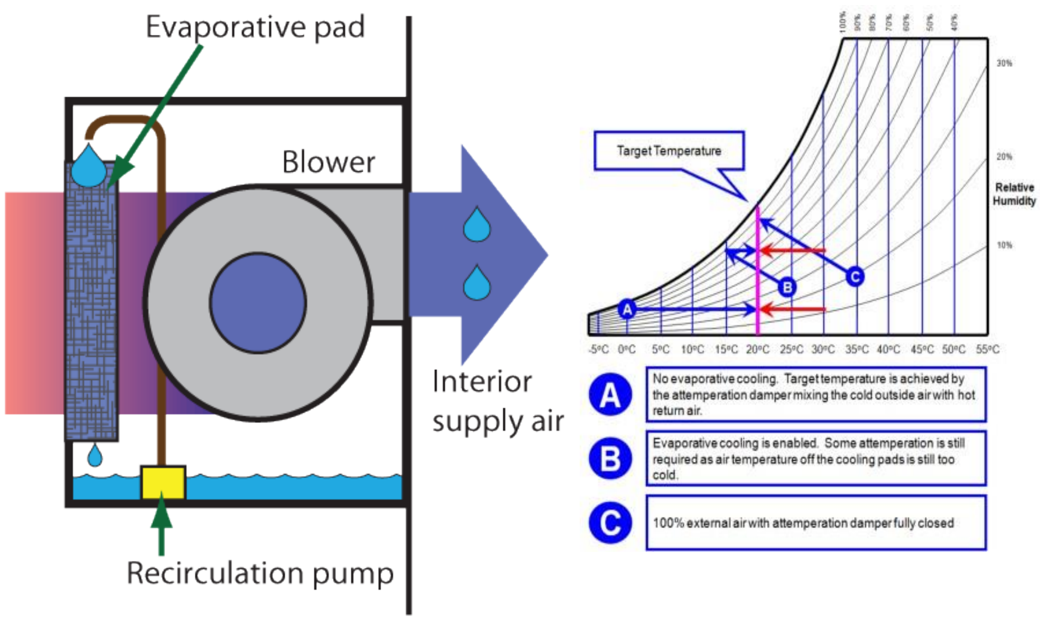
Ideal PUE =1.0

Chapter 3

Direct Evaporative Cooling

This model has been developed and is in practice in the industry. Dodia N has already designed the layout of formulation of the cycle and this is the improved version that concentrates on the fan efficiencies in the cycle[16]. This also adds more information on the efficiency and PUE of the evaporative cooling.

Direct evaporative cooling (DEC) water is sent directly into the supply airstream with a spray or wetted media. As the water absorbs heat from the air, it evaporates and cools the air. In direct evaporative cooling the dry bulb temperature is lowered but the wet bulb temperature remains unchanged. The latent heat of water is observed from the hot air. The water vapor mixes with the air and makes it more humid.



(a) (b)

Figure 3-1 (a) Direct Evaporative Cooling, (b) Psychrometric chart representation

In operation, the blower pulls air through a permeable, water-soaked pad. As the air passes through the pad, it is filtered, cooled, and humidified. A recirculation pump keeps the media wet, while air flows through the media. To make sure that the entire media is wet, more water is usually pumped than can be evaporated and excess water drains from the bottom into a sump. An automatic refill system replaces the evaporated water.

3.1 Model

This model is developed on the energy flow basis of data center cooling. This model considers the cycle from the entry of the ambient air to the exit of the exhaust air form the server room. For this the cycle has been divided into three sub levels where energy is calculated.

The cycle starts with the ambient air entering into the evaporating pad. The water is pumped from the top of the cooling media, which drifts down to the sink at the bottom of the media. The water evaporates to cool the air. Excess amount of water is sent in to the media so that the cooling pad is wet at all times. Water is cycled to the top from the sunk tank and makeup water is added every time. The cool air is then passed through filters to reduce the moister content in the air and also to purify the air from IT equipment pollutants. Large fans are used to force the air into the server room with the required pressure.

The air entering into the server room is fed into the cold isle. The server fans help to suck the cold air from the cold isle side to hot isle side and prevents the back pressure. This hot air from the hot isle is sucked by the exhaust fan.

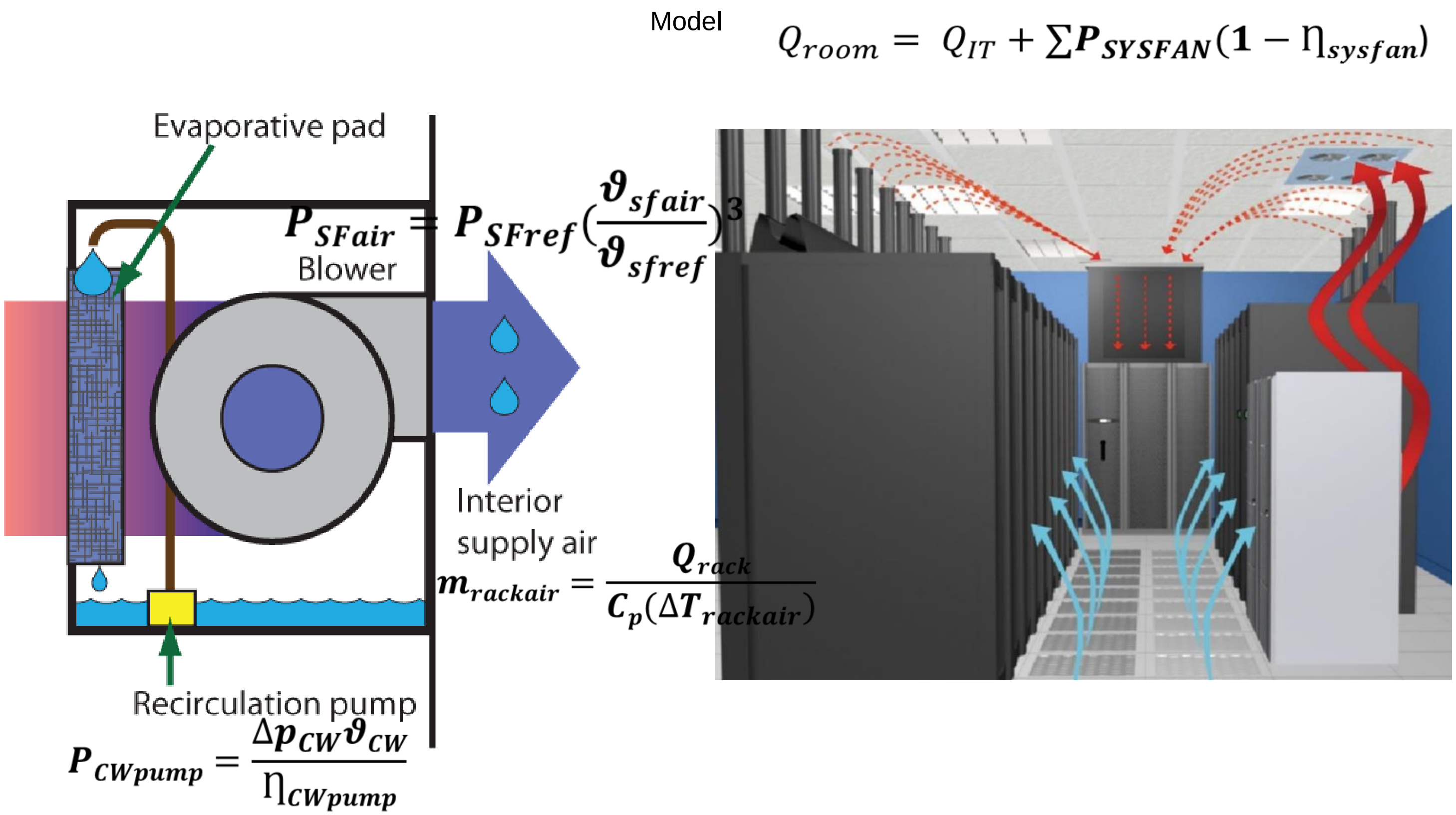


Figure 3-2 Energy flow model diagram

3.1.1 Assumptions and considerations

Assumptions for this model are

1. Each rack in the computer room is operating at the same performance level with the same loading conditions that is at 98% CPU utilization.
2. 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.
3. The facility has a IT capacity of 25kW.
4. Adequate cooling resources.
5. All server fans are equally efficient.

3.1.2 Server heat

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

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

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

3.1.3 Fan energy

We used fans at several steps of the cooling cycle. The electrical energy supplied to the fan is used to produce a pressure head but it also liberates heat due to the friction and current. The efficiency of the fan can be understood by the fan curves.

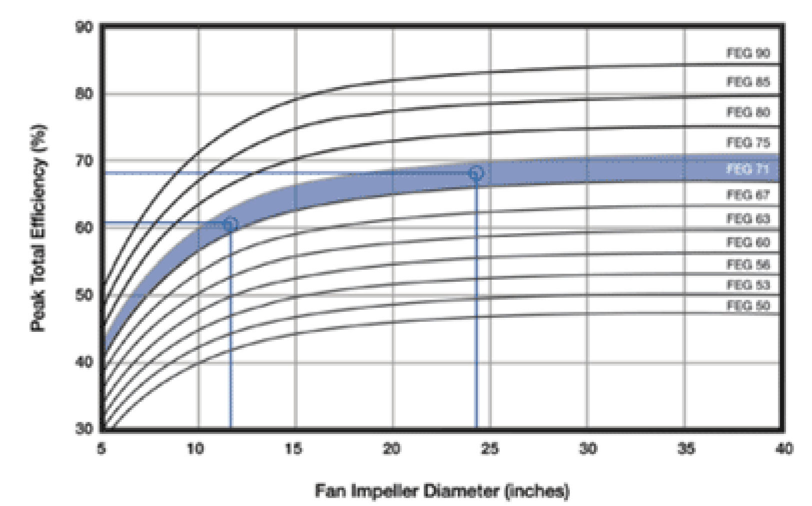


Figure 3-3 Fan efficiency curve

From figure 3-3 [11] we can see that the fan efficiency increases as the diameter of the fan impeller increases. It also depends on the fan efficiency grade (FEG) of the selected fan. So the server fan efficiency can be taken as 45% approximately. Figure 3-4 [11] shows the relation between pressure air flow and efficiency.

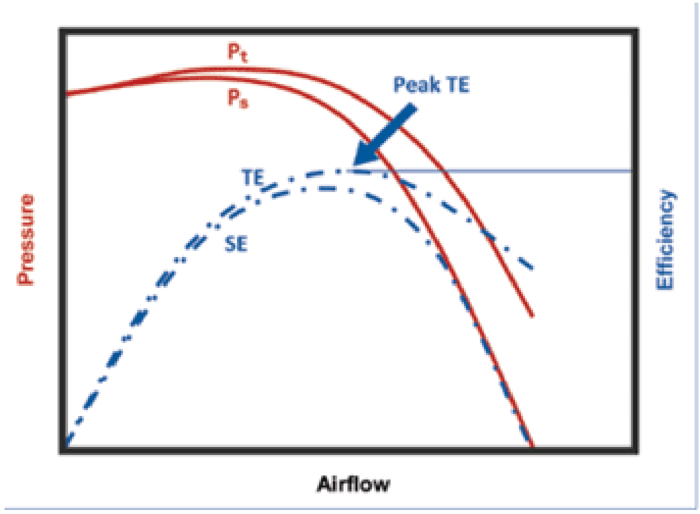


Figure 3-4 Relation among pressure, airflow and efficiency

The specifications of the server fan I have considered is

Voltage: 12V

Current: 1.88A

Speed: 13900 rpm

Power of fan = 22.56W

Figure 3-5 Server fan

The efficiency of the fan can be calculated by the formula

$$\eta_f = \frac{d_p\, q}{P}$$

Where

η_f = fan efficiency (values between 0 - 1)

d_p = total pressure (Pa)

q = air volume delivered by the fan (m^3/s)

P = power used by the fan (W, Nm/s)

Supply fan power is determined based on the amount of air required. 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.

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

3.1.4 Water Pump

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 calculated by measuring the humidity ratio gain at the end of the evaporative media and multiplying it by the mass flow rate of the air

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

3.1.5 Evaporating 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. The efficiency of the media varies from 60% to 90% depending on the number of cycles and time period.

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

3.3 Model Validation

3.3.1 Server taken in to 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.

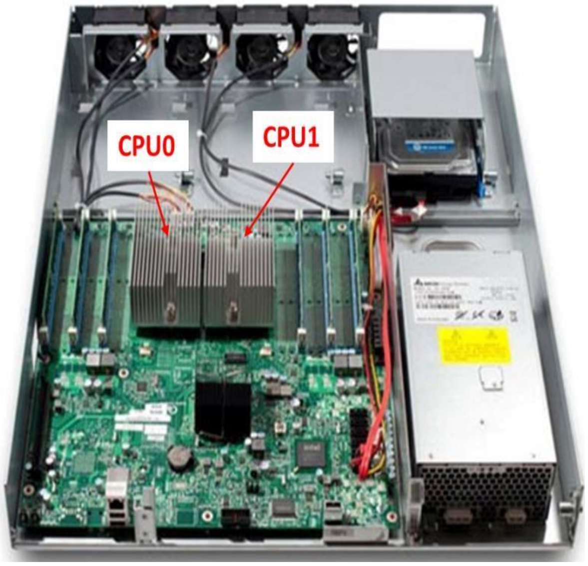


Table 3-1 Inlet Temparatures and Delta T

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

3.3.2 MESTEX Model comparison

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: ½ HP, 67GPM
- DIRECT PUMP: 1/3 HP, 16 GPM
- 3 FULLY POPULATED RACKS WITH HP SE1102 SERVERS
- TOTAL LOAD: 25 KW

Table 3-2 Model comparition

	BASELINE	**MODEL**	**%ERROR**
Total Load	25KW	25KW	0
Supply fan Power	0.791 kW	0.71kW	-11.5%
Direct Pump	0.83 kW	0.75	-7.5%
Cooling Power	1.621	1.45	-10%
PUE	1.065	1.0578	-0.745%
Water Consumption	Not Metered	0.135	-

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.

3.3.3 Special case

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.

3.4 Calculations

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

Q= 25 kW;

Density = 1.185 kg/m^3

ΔT = 17^{o}C

Q_{sf} = Q+∑12*(1-0.45) = 25k+1.58k = 26.58k

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

Therefore, m = 26.58/(1.05*17) = 1.489 kg/s = 1.489/0.0004719 = 3155 cfm

$P_{supplyfan}$ = 7.5*(3155/6250)3 = 0.94 kW

$P_{directpump}$= 0.75 kW

$P_{cooling}$ = 0.75 + 0.94 =1.69 kW

PUE = (25 +1.69)/25 = 1.067

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

3.5 Results

As rack inlet temperature of the air increases, the mass flow rate required to cool the server increases. Therefore, the power consumed by the supply fan increases which overall increases the Total cooling power consumption. For both the cities economizer mode varies due to the ambient conditions and therefore water consumption varies across both cities.

Table 3-3 San Jose Economizer for DEC

SAN JOSE

San Jose Economizer				
Temp	PUE	Water(gal)	Econ(hrs/year)	Econ (%)
15	1.029	14135	4433	50.60502
20	1.019	7555	7219	82.40868
25	1.016	2895	8329	95.07991
30	1.05	781	8710	99.42922
35	1.129	56	8758	99.97717
40	1.332	0	8760	100
45	1.64	0	8760	100

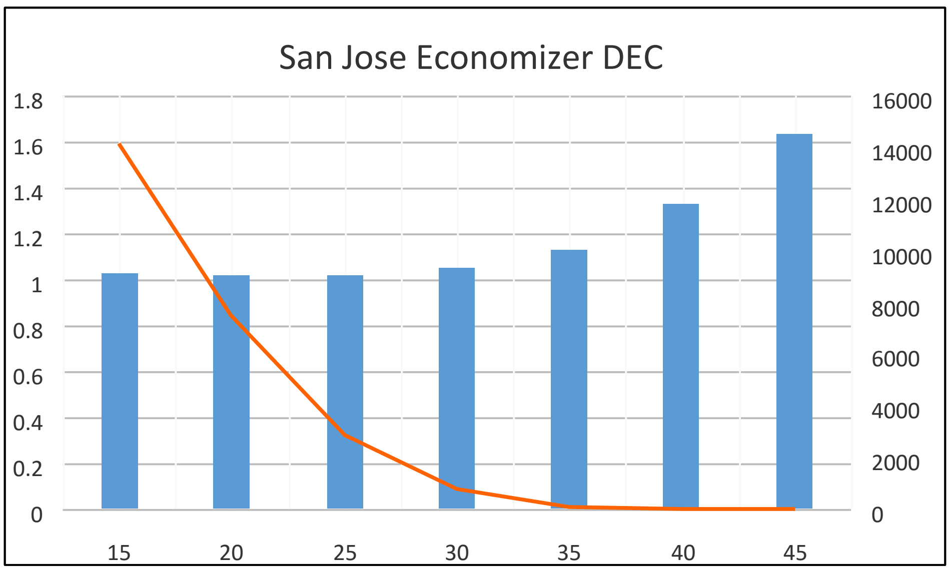


Figure 3-7 San Jose Direct Evaporative Cooling

DFW

Table 3-4 DFE Economizer for DEC

DFW Economizer				
Temp	PUE	Water(gal)	Econ(hrs/yr)	Econ (%)
15	1.035	21253	4433	50.60502
20	1.029	17819	7219	82.40868
25	1.025	13865	8329	95.07991
30	1.053	12864	8710	99.42922
35	1.129	5111	8758	99.97717
40	1.332	39	8760	100
45	1.638	0	8760	100

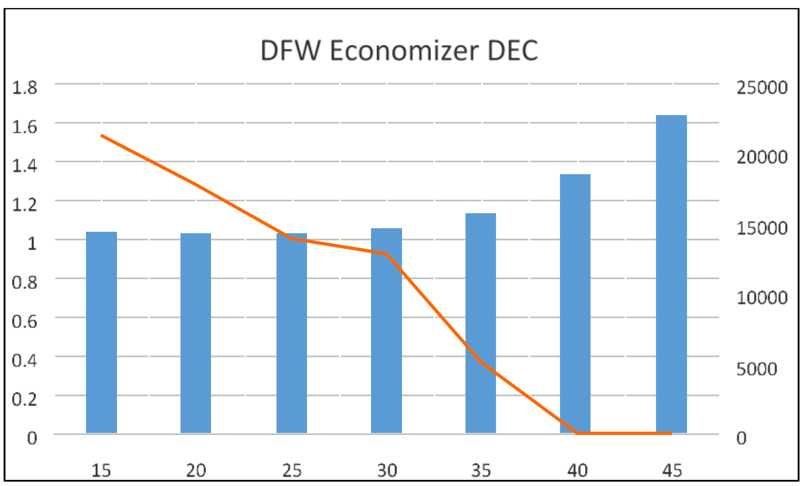


Figure 3-8 DFW Direct Evaporative Cooling

Chapter 4

Indirect Evaporative Cooling

Indirect evaporative cooling (IDEC) lowers the temperature of air through a heat exchanger, in which a secondary airstream is cooled by water stream like in the direct evaporative cooling and which in turn cools the primary air. The cooled air never comes in direct contact with water so, it is not polluted. In indirect evaporative cooling system both the dry bulb and wet bulb temperatures of the air are reduced.

Indirect evaporative cooling dose not humidify the air, but costs more than direct evaporative cooling system and operate at a lower efficiency when compared to direct evaporative coolers. Figure 4-1 and 4-2 [10] indicates indirect evaporative cooling

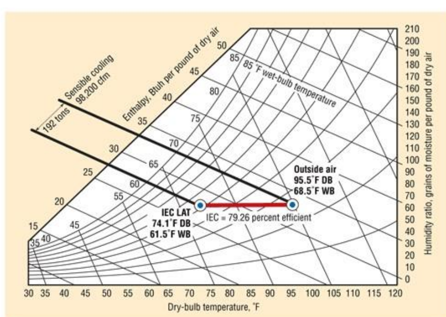


FIGURE 1. The IEC performance was measured at 79.26-percent effectiveness, delivering the equivalent of 192 tons of sensible cooling for 98,200 cfm.

Figure 4-1 Psychrometric chart for IDEC

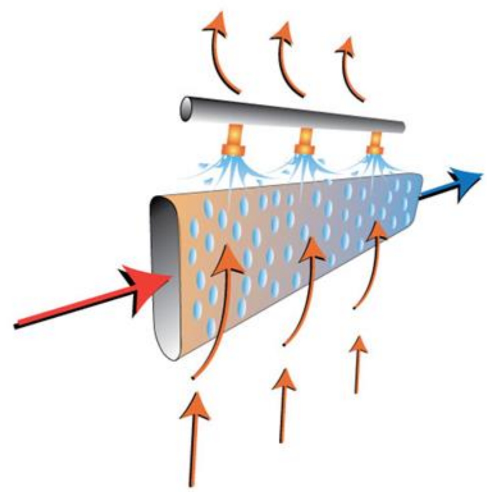
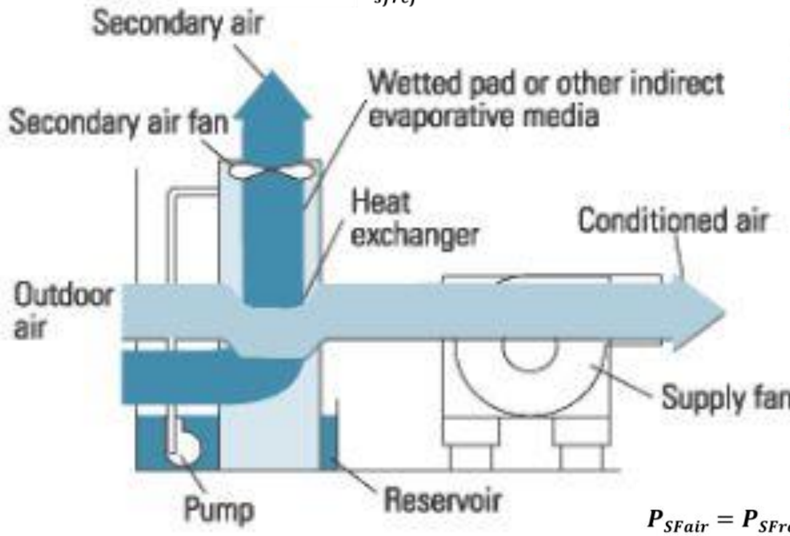
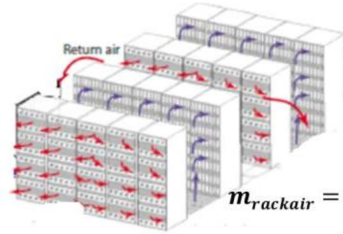


Figure 4-2 Indirect Evaporative cooling pad

4.1 Model

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





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

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

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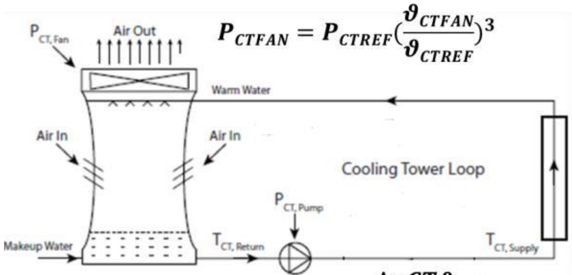
$$P_{SFair} = P_{SFref}(\frac{\vartheta_{sfair}}{\vartheta_{sfref}})^3$$



$$P_{CTFAN} = P_{CTREF}(\frac{\vartheta_{CTFAN}}{\vartheta_{CTREF}})^3$$

$$P_{CTpump} = \frac{\Delta pCT\vartheta_{CT}}{\eta_{CTpump}}$$

This model is developed by separating energy levels of the cooling system. The model starts at rack level heat released. The amount of heat released by the servers is calculated to find the amount of cool air needed to supply. Then power drawn by the fan is determined. Then comes the indirect evaporative cooling system.

In this indirect evaporative cooling supply air dose not directly contact the water or the environment. Secondary air is used for cooling the primary air. This secondary air undergo direct evaporative cooling in which latent heat is used to cool the secondary air and in turn primary air. So the efficiency of indirect evaporative media is less than the direct cooling.

4.1.1 Server heat

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

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

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

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

4.1.2 Fan Energy

The fan efficiency derived for direct evaporative cooling system is used in the indirect evaporative cooling model.

Primary fan power is determined based on the amount of air required. 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.

$$P_{PFair} = P_{PFref}\left(\frac{\vartheta_{Pfair}}{\vartheta_{Pfref}}\right)^3$$

Secondary fan power is also determined by the amount of cooling required. As the efficiency of the indirect evaporative media is less direct that is approximately 70% and it depend on the heat exchange material and the cooling pad efficiency. But the power consumed by the fan can be found by the same formula

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

4.1.3 Water pumping energy

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.

$$Q_{ct} = mC_p(T_i - T_o)$$

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

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

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

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

4.2 Model Validation

4.2.1 MESTEX model

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: ½ HP, 67GPM
- DIRECT PUMP: 1/3 HP, 16 GPM
- 3 FULLY POPULATED RACKS WITH HP SE1102 SERVERS
- TOTAL LOAD: 25 KW

Table 4-1 IDEC model

	BASELINE	MODEL	%ERROR
TOTAL LOAD	25KW	25KW	0
SUPPLY FAN POWER	0.791 kW	0.76 kW	-3.9%
COOLING TOWER FAN POWER	0.039 kW	1.5 kW*	*Modelled for fixed fan speed for Cooling Tower
CIRCULATING PUMP POWER	2.20	2.28kW	+3.6%
PUE(instantaneous)	1.12	1.17	+5%
WATER CONSUMPTION (instantaneous)	Not metered	0.3 GPM	-

4.2.2 Model comparison

The results of the validation model run using the parameters defined above are compiled in Table 5, 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 5. 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.

4.2.3 ASHRAE A4 Envelope

The server rack inlet temperature is varied from 15^{o}C to 45^{o}C in the interval of 5^{o}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. The ASHRAE envelopes are shown in the figure 4-3 [6] below.

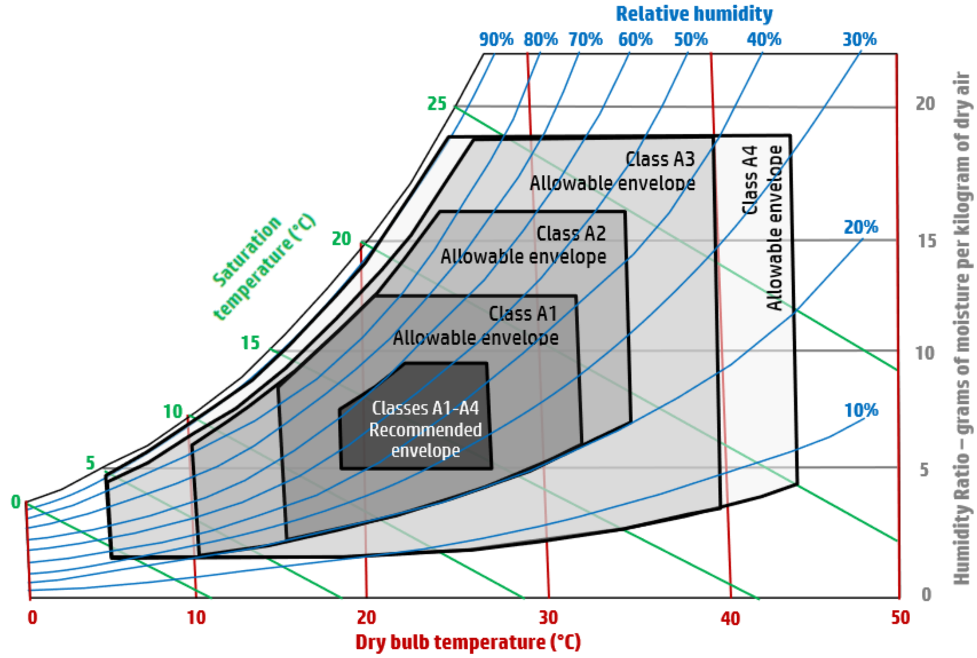


Figure 4-3 ASHRAE envelope [15]

4.4 Calculations

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

Q= 25 kW;

Density = 1.185 kg/m^3

ΔT = 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

$P_{supplyfan}$ = 7.5*(2616/6250)^3 = 0.54 kW

P_{ctfan} = 1.5*(5000/5000)^3 = 1.5 kW

$P_{circulatingpum}$p= 2.25 kW

$P_{cooling}$ = 2.25+ 1.5 + 0.54 =4.24 kW

PUE = (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

4.5 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.

San Jose

Table 4-2 San Jose Economizer for IDEC

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	430	8760	100

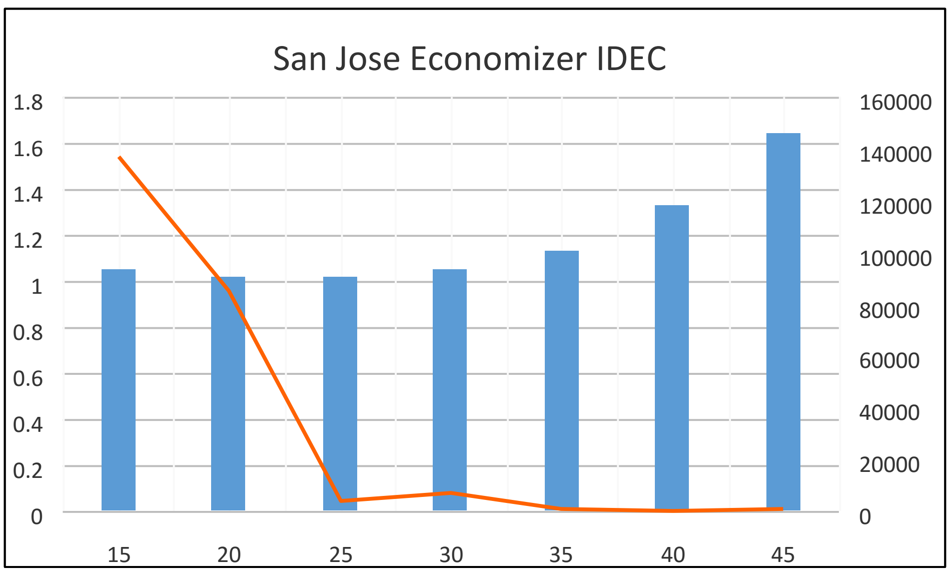


Figure 4-4 San Jose IDEC

DFW

Table 4-3 DFW economizer for IDEC

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

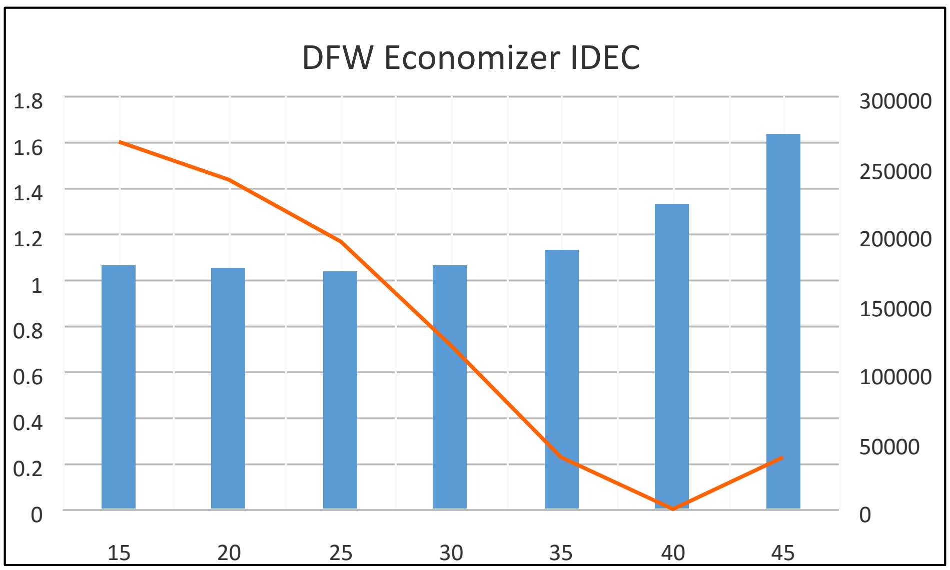


Figure 4-5 DFW IDEC

Chapter 5

Conclusion

This model is used to calculate PUE and water consumption for the given place and at given temperatures. As an example San Jose and DFW are done as the part of the study.

DFW (DIRECT MODE)

1. PUE is best for 15ºC Rack Inlet Temperature and increases with increase in Rack Inlet Temperature.
2. Water usage increases with increase in Rack Inlet Temperature.

SAN JOSE (DIRECT MODE)

1. PUE is best for 15ºC Rack Inlet Temperature and increases with increase in Rack Inlet Temperature
2. Water usage increases with increase in Rack Inlet Temperature

SAN JOSE (INDIRECT MODE)

1. PUE is best for 15ºC Rack Inlet Temperature and increases with increase in Rack Inlet Temperature.
2. Water usage is almost same for all Rack Inlet Temperature.

DFW (INDIRECT MODE)

1. PUE is best for 15ºC Rack Inlet Temperature and increases with increase in Rack Inlet Temperature
2. Water usage is almost same for all Rack Inlet Temperature.

Finally, we can conclude by saying that

(PUE)Direct economizer < (PUE) Indirect Economizer < (PUE) Direct < (PUE) Indirect

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Biographical Information

Rajesh Kasukurthy received his bachelor degree (B. Tech) in Mechanical Engineering from Jawaharlal Nehru Technological University Kakinada, University College of Engineering Vizianagaram, Andhra Pradesh, India. In the final year of bachelors Rajesh developed a Four Bar Spherical Mechanism for a defined path. For perusing his masters, Rajesh joined University of Texas at Arlington, Texas, US in August 2014. Evaporative cooling used in Data Center cooling is his main area of research. Rajesh received his Master’s degree(MS) in Mechanical Engineering from University of Texas at Arlington in August 2016. Rajesh is admitted in to Ph.D. program for fall 2016 in to The University of Texas at Arlington.