NET ZERO DESICCANT ASSISTED EVAPORATIVE COOLING FOR DATA CENTERS

by

David Okposio

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THE PURDUE UNIVERSITY GRADUATE SCHOOL
STATEMENT OF COMMITTEE APPROVAL

Dr. A.G. Agwu Nnanna, Chair
Department of Mechanical and Civil Engineering

Dr. Harvey Abramowitz, Co-Chair
Department of Mechanical and Civil Engineering

Dr. Chandramouli Viswanatha Chandramouli
Department of Mechanical and Civil Engineering

Dr. Xiuling Wang
Department of Mechanical and Civil Engineering

Approved by:
Dr. Chenn Zhou

Dedicated to the memory of my Father, Mathew Okposio

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LIST OF ABBREVIATIONS

ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
Alumina	Aluminum Oxide
Bi_2Te_3	Bismuth Telluride
$CaCl_2$	Calcium Chloride
C_p	Specific Heat Capacity
C_{PE}	Specific Heat of the Cooling Media
CRAC	Computer Room Air Conditioning System
DEC	Direct Evaporative Cooling
ε	Saturation Effectiveness of the Evaporative Cooling Unit
ε_{ff}	Dehumidification Effectiveness
H_2O	Water
h_a	Enthalpy of dry air
h_w	Enthalpy of Moisture in Air
h_{fg}	Latent Heat of Vaporization
HVAC	Heating Ventilation and Air-Conditioning
IDEC	Indirect Evaporative Cooling
LiBr	Lithium Bromide
$\dot{m}_a$	Mass Flow rate of dry air
$\dot{m}_w$	Mass flow rate of moisture content of air
$\dot{m}_{evap}$	Evaporation rate of Water on the Pad
$\dot{m}_{ad}$	Rate of Moisture Adsorption by desiccant material
$\dot{m}_{des}$	Rate of moisture removal from saturated desiccant material
$\dot{m}_{cond}$	Rate of Condensate Formation
MRC	Moisture Recovery Capacity
MRU	Moisture Recovery Unit
PA	Process Air
PUE	Power Utilization Efficiency
ρ	Density

Q_s	Sensible Heat loss/Sensible Cooling
Q_l	Latent Heat
Q_T	Total Heat
RH	Relative Humidity
SiO2	Silica Gel
SHF	Sensible Heat Factor
t_R	Residence Time
TEC	Thermoelectric Coolers
TEG	Tri-ethylene glycol
TDS	Total Dissolved Solids
T_{in}	Inlet Air Temperature
T_{pa}	Temperature of Process Air
V	Volume
V_{in}	Inlet Air Velocity
V_{RA}	Return Air Velocity
W	Humidity Ratio

ABSTRACT

Evaporative cooling is a highly energy efficient alternative to conventional vapor compression cooling system. The sensible cooling effect of evaporative cooling systems is well documented in the literature. Direct evaporative cooling however increases the relative humidity of the air as it cools it. This has made it unsuitable for data centers and other applications where humidity control is important. Desiccant-based dehumidifiers (liquid, solid or composites) absorb moisture from the cooled air to control humidity and is regenerated using waste heat from the data center. This work is an experimental and theoretical investigation of the use of desiccant assisted evaporative cooling for data center cooling according to ASHRAE thermal guidelines, TC 9.9. The thickness (depth) of the cooling pad was varied to study its effect on sensible heat loss and latent heat gain. The velocity of air through the pad measured, and to determine its effect on sensible cooling. The flow rate of water over the pad was also varied to find the optimal flow for rate for dry bulb depression. The configuration was such that the rotary desiccant wheel (impregnated with silica gel) comes after the direct evaporative cooler. The rotary desiccant wheel was split in a 1:1 ratio for cooling and reactivation at lower temperatures. The dehumidification effectiveness of a fixed bed desiccant dehumidifier was compared with that of a rotary desiccant wheel and a thermoelectric dehumidifier. A novel condensate recovery system using the Peltier effect was proposed to recover moisture from the return air stream, (by cooling the return air stream below its dew point temperature) thereby optimizing the water consumption of evaporative cooling technology and providing suitable air quality for data center cooling. The moisture recovery unit was found to reduce the mass of water lost through evaporation by an average of fifty percent irrespective of the pad depth.

1. INTRODUCTION

1.1 Overview

It is estimated that about five percent of the total global energy usage is by electronics. The information technology sector is expected to consume about 20% of all the world's electricity by 2025, whilst being responsible for over 5 percent of carbon emission. [1] The ASHRAE Technical committee 9.9 white paper of 2016, estimates that about 25% of electricity consumed by data centers is used to service cooling. [2] With the expansion of cloud computing and a massive increase in demand for online services, it is only natural to expect that the number of data centers continues to grow on a global scale. In the same vein as the digitalization of the world continues ,and an increasing number of connected devices, video streaming services, face recognition technology, surveillance camera footages, smart homes become more prominent on a global scale and as over a billion more people come online from third world countries, the computing and storage power required by existing data centers will increase accordingly and new data centers may need to be built in places where hitherto data centers have not been situated. As the need for energy to support these data centers rises, the need to find and use efficient cooling. sources will also increase direct consequence of this advancement is the cooling of the data centers [3]The first thermal guidelines for the cooling of data centers was published by ASHRAE in 2004 with prescribed temperatures ranging from 20^{o}C to 25^{o}C. These recommendations took into account maximal uptime and hardware lifespan. This temperature range allows for optimal usage and gives enough buffer room in the event of air conditioning failures. However, as the focus has shifted from just optimal uptime and hardware lifespan to power utilization efficiency, these guidelines have been revised accordingly. Nowadays it is commonplace to find recommended temperature sweet spot for most servers provided by manufacturers. The recommended temperature range today is between 18^{o}C to 27^{o}C. [4] These higher allowable temperatures mean that data centers and businesses with dedicated server rooms need not cool them as they once used to. It is therefore pertinent given the growth of data centers and companies with dedicated server rooms, that an alternative to conventional vapor compression cooling should be thoroughly investigated. Such an alternative should offer significant energy savings and provide suitable air quality for data centers.

The conventional cooling technology for data centers is the computer room air conditioning system. However, the cost associated with using CRAC units has led to many researchers investigating alternatives to conventional vapor compression systems. Some of the alternative approaches investigated include water cooling, in rack solutions such as affixing a heat exchanger to the rack doors for heat rejection and evaporative cooling in conjunction with containment strategy. It is becoming increasingly common to find different combinations of these methods being employed in the cooling of data centers. [5]The overall aim is always to improve power utilization efficiency (PUE).

The United States environmental protection agency in a report to congress in 2007 presented an alarming trend in energy consumption by data centers; with 61 billion kilowatt-hours consumed in 2006**.** Total electricity consumption by data centers continues to rise as highlighted by the United States Data Center Energy Usage Report. [5] It has been reported that about 38% of energy consumption by data centers results from HVAC systems, known as Computer Room Air conditioning system, CRACS. [6]

With current total energy consumption of data centers standing at around 90 billion kilowatt hours, 38 percent of which goes to HVAC systems, the need to improve data center air conditioning has never been more urgent. So, the energy consumption of data centers is a major operational problem for data center operators. This need has been recognized by researchers and business leaders across the world. Evaporative cooling has been proposed as a viable alternative cooling technology to CRACS. [4]

In terms of energy consumption on a global scale, data centers are surpassed only by the United states and China. Whilst this highlights the critical role of data centers in the global economy, the challenge for the Mechanical Engineer is to explore viable cooling solutions for data centers that are both efficient in terms of satisfying the cooling load requirement, environmentally friendly and economically viable in terms of cost savings. [3]

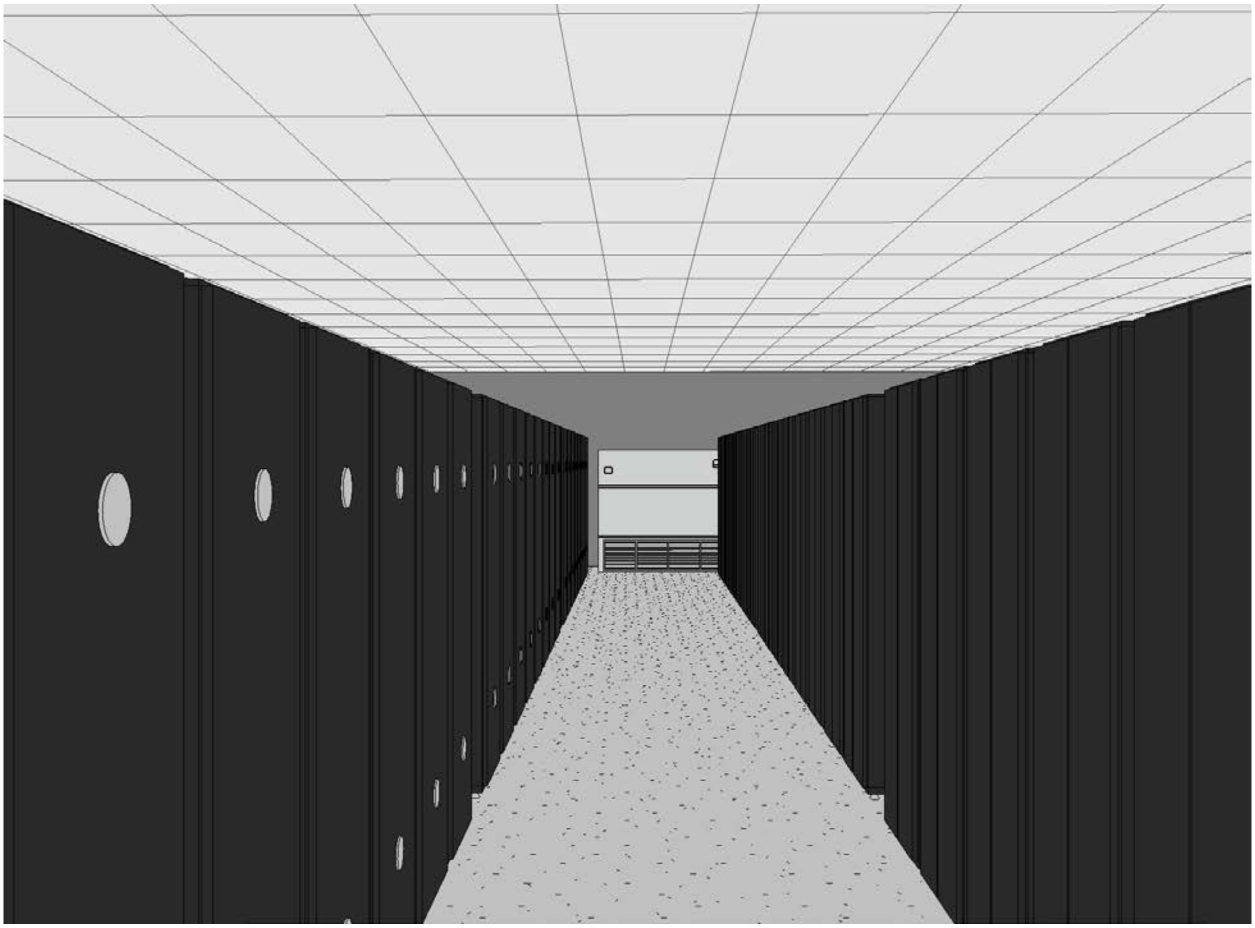

Figure 1. 1 Cold isle of a hard floor data center vapor compression cooled data center

1.2 Objective and Scope

The objective of this work is to investigate the suitability of desiccant based evaporative cooling technology for the cooling data centers, while minimizing the water consumption of the system by a condensate recovery process.

The use of desiccant assisted evaporative cooling technology is proposed as a way to significantly improve power utilization efficiency, PUE for data centers.

2. LITERATURE REVIEW

2.1 Evaporative Cooling

Evaporative cooling is a highly energy-efficient alternative cooling system free of the high energy cost associated with conventional vapor compression cooling systems. There are basically two types of evaporative cooling: direct or indirect evaporative cooling. It can be used as a stand-alone cooling system or as a hybrid system combined with conventional vapor compression cooling technology, with applications in various spheres such as homes, poultry farms and even data centers it offers distinct advantages as it is both clean, energy efficient, low-cost and devoid of the challenges that some refrigerants pose to the environment in conventional cooling systems. [7]

Typically, direct evaporative cooling as opposed to indirect evaporative cooling increases the humidity ratio of the process air (cooled air). This challenge is hardly encountered with indirect evaporative cooling, however the sensible cooling observed in direct evaporative cooling is better than what is obtained in indirect evaporative cooling. To make direct evaporative cooling suitable for certain applications, it is used in conjunction with a dehumidifier to keep the humidity ratio of the cooled space within the desired range. This challenge is solved by the aid of rotary desiccant wheels, which helps dehumidify the process air. [8]

Although direct evaporative cooling produces process air temperature suitable for human comfort and other applications it is disadvantaged by the introduction of moisture to the cooled space as mentioned above. Another critical challenge faced by evaporative cooling systems is the volume of water consumed. [9] Water consumption has been highlighted as a major impediment to the use of direct evaporative cooling, especially in cases where high volumetric air flow is required. Typically water consumption in direct evaporative cooling can arise from; leakages in the system, periodic bleeding of the system due to mineral build up and water losses due to evaporation, the latter is integral to the cooling process. [10, 11] Water losses due to evaporation on the pad is a major focus of this work. [12]

2.1.1 Psychrometrics of Evaporative Cooling

Evaporative cooling is an adiabatic process in which air is cooled while it absorbs moisture. Psychometrically speaking, during evaporative cooling, temperature and humidity changes along the lines of constant wet bulb temperature. Other notable changes include dry bulb temperature, specific volume, dew point temperature, relative humidity, and specific humidity. But as indicated above, no changes in wet bulb temperature and enthalpy are recorded. The dry bulb temperature refers to the temperature of the air as measured by a thermometer or a thermocouple, it is an indicator of the heat content of the air, whilst the wet bulb temperature indicates the moisture content of the air. [13] Taken together the difference between the dry bulb temperature and the wet bulb temperature is very important in evaporative cooling as it indicates the cooling efficiency of an evaporative cooling unit. Just as important is for this project is the dew point temperature which highlights the temperature at which the vapor content of the air begins to condense. In fact, at high relative humidity when the dew point is close to the dry bulb temperature, the cooling effect of evaporative cooling is less effective. In other words, the capacity of the air to hold moisture is significantly diminished.

2.1.2 Types of Evaporative Cooling

There are two major types of evaporative cooling systems, namely: direct evaporative cooling systems and the indirect evaporative cooling system. Some configurations exist that are a combination of both types. But broadly speaking direct and indirect evaporative cooling is a suitable classification. [12]

2.1.3 Direct Evaporative Cooling

Direct evaporative cooling usually involves warm air blowing across a wetted pad as water runs over the pad from the reservoir. The warm air exchanges sensible heat for latent heat as it is cooled in the process. The goal of direct evaporative cooling is to reduce the dry bulb temperature of the process air to the wet bulb temperature of the air from the outside. [12]

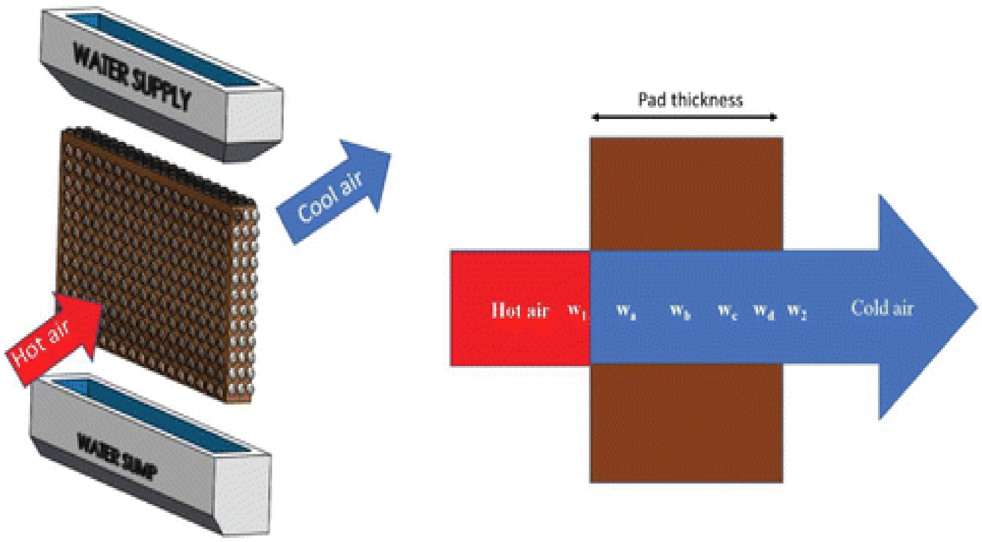

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Figure 2. 1 Mechanism of direct evaporative cooling

When water evaporates off the surface of the pad it draws energy from its surrounding which produces a considerable cooling effect. Evaporative cooling occurs when moisture is added to air that has a relative humidity of less than a hundred percent. The lower the relative humidity, of the air to be cooled the greater the temperature drop observed. This temperature drop is called the dry bulb depression. On a psychometric chart the increase in humidity ratio is seen as the warm air exchanges sensible heat for latent heat. This usually represents a movement along the enthalpy line on a psychometric chart. [13] Figure 2.1above shows the crossflow configuration of a direct evaporative cooling unit. The cooling occurs via a process of heat and mass transfer which takes place on the cellulose pad. [14, 15]

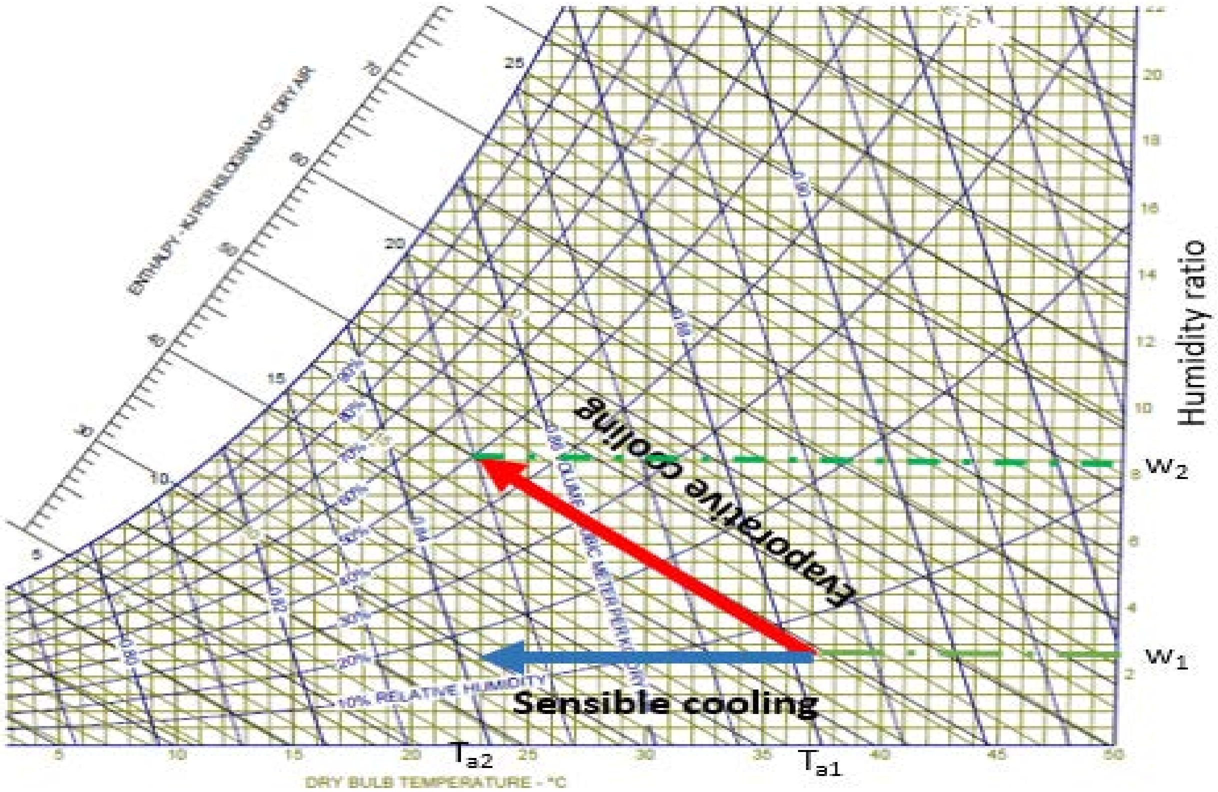

Figure 2. 2 Representation of direct evaporative cooling on a psychrometric chart

Direct evaporative cooling as an alternative or hybrid to mechanical vapor compression computer room air conditioning systems is a promising technology and is an area calling for more investigations.

2.1.4 Indirect Evaporative Cooling

The major difference between direct evaporative cooling and indirect evaporative cooling is the fact that the former increases the relative humidity of the air as it is cooled whilst the latter has little or no effect on the relative humidity of the air it cools. So, the air is sensibly cooled, and the humidity ratio is unchanged in the process. The primary idea of the indirect evaporative coolers is based on cooling by decreasing the inlet air sensible heat without changing its humidity ratio. It is represented by a horizontal line on the psychrometrics chart. [14, 16]. The usual practice is to use two airstreams separated by dry and wet channels. One of the air streams called the primary air stream is sensibly cooled with the help of a heat exchanger, while the secondary air carries away the heat energy from the primary air. With indirect evaporative cooling, a secondary air stream is cooled by direct evaporative cooling. The cooled secondary airstream goes through a heat exchanger where it cools the primary airstream. [17]

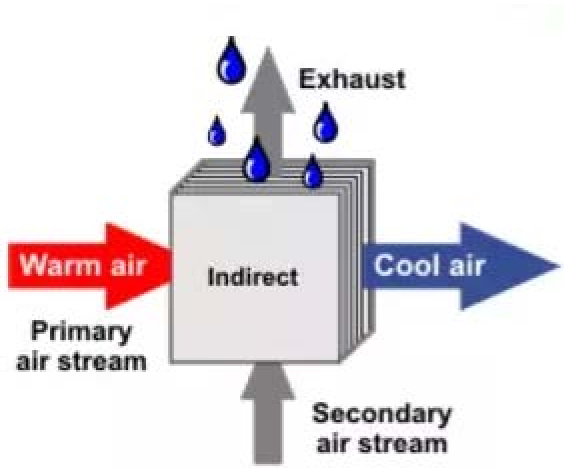

Figure 2. 3 Configuration of indirect evaporative cooling

Indirect evaporative cooling does not add moisture to the primary air stream. In indirect evaporative cooling systems, both the dry bulb and wet bulb temperatures of the air are reduced.

2.2 Desiccant Based Dehumidification

In section 2.1 above We mentioned that direct evaporative cooling increases the relative humidity of the cooled air. Depending on the desired application this trend can be a problem especially for servers and other electronic devices whose tolerance of moisture may not be very high, to avoid the bridging of circuitry. The use of a dehumidifier can help solve this problem. [18]

Moisture content or latent heat of air can be managed (controlled) either by condensing the water vapor or by the use of a suitable desiccant material for dehumidification. Now dehumidification is the removal of moisture from the air. It is necessary in spaces requiring low humidity such as pharmaceutical industries, hospitals, electronic assembly units, clean rooms and of course data centers. [19, 20]

Desiccants are substances that attract water-vapor molecules from the air through an adsorptive or absorptive process. Adsorption describes a desiccant that attracts airborne moisture without experiencing a phase change. These are usually solids. Examples include activated alumina, silica gel and zeolites (molecular sieve). It is however worth mentioning that composite desiccant materials exist and depending on the desired application can be very effective.

In collecting airborne moisture an absorptive desiccant material experiences physical or chemical changes. Some common examples include Solutions of Lithium Chloride, Lithium Bromide (LiBr), Triethylene glycol (TEG) in water, Calcium chloride ($CaCl_2$). It is not always feasible to reactivate absorptive desiccant materials. [21]

2.2.1 The Rotary Desiccant Wheel

The process air passes through the desiccant wheel which has been coated with desiccant material (Calcium Chloride, Silica gel, Lithium Bromide, Triethylene glycol (TEG) etc.). So that the air leaving the desiccant wheel becomes dehumidified. Dehumidification is the stripping of the water molecules from the process air stream due to vapor pressure difference between the humid air stream and the silica gel material. Certain factors are critical to the performance of the rotary desiccant wheel. There is a consensus among researchers that the performance of desiccant based cooling systems is hinged on the performance of the rotary desiccant wheel. [22, 23] Experimental analysis have been conducted on solid desiccant dehumidification wheels (the so called rotary desiccant wheel or enthalpy wheel) with emphasis on the effect of inlet humidity ratio on humidity ratio of the air stream exiting the desiccant wheel, it was found that the rotation speed is inversely proportional to sorption time. The effect of geometry of channels of a silica gel desiccant wheel, was also thoroughly investigated and the conclusion was that the geometry of the passage is critical to sorption rate. [24] Also, the triangular, sinusoidal and rectangular cross sections gave better dehumidification performance in that order. [24, 25, 26]

As mentioned above the process air is dehumidified on one side of the wheel and the desiccant material is reactivated or regenerated on the remaining portion of the wheel. This is the return or regeneration (reactivation) air stream. The ratio of the wheel dedicated to dehumidification and regeneration is critical to the performance of the wheel. Zendeboudi and Esmaeili [27] conducted an experimental investigation to determine the effect of the dehumidification and regeneration area on the performance of the wheel. Two desiccant wheels of diameter 55cm and thickness 18cm were used for this study. They found that, regardless of the desiccant material increase in area ratio resulted in better dehumidification at inlet humidity ratio of 10.63 g/kg for a 1:3 split the supply air was 8.92 g/kg whilst for a 1:1 configuration a superior humidity ratio of supply was obtained at 7.67g/kg. Jae *et al* [28] conducted a numerical simulation on a desiccant wheel with a close look at wheel rotational speed and the area of regeneration and dehumidification for a range from 60ºC to 150ºC. They found that as regeneration temperature decreases the regeneration section becomes a larger portion of the wheel, they also noted that an optimal rotational speed is obtained when moisture recovery capacity, MRC is at a maximum. They proposed that as is found in supplier's manual a split 1:3 is suitable for high regional temperature and split 1:1 more suitable for low regeneration temperature, and as inlet humidity increases the ratio of regeneration area to dehumidification area also increases. O'Connor *et al.* [29] presented a novel design of a rotary desiccant wheel for reduced air temperature. This work showed little pressure drop as opposed to the conventional rotary desiccant wheel. They [29] also reported an increase in the temperature of the air after the desiccant wheel. An idea corroborated by many other researchers. [29, 30, 31]. The dehumidification is an isenthalpic process.

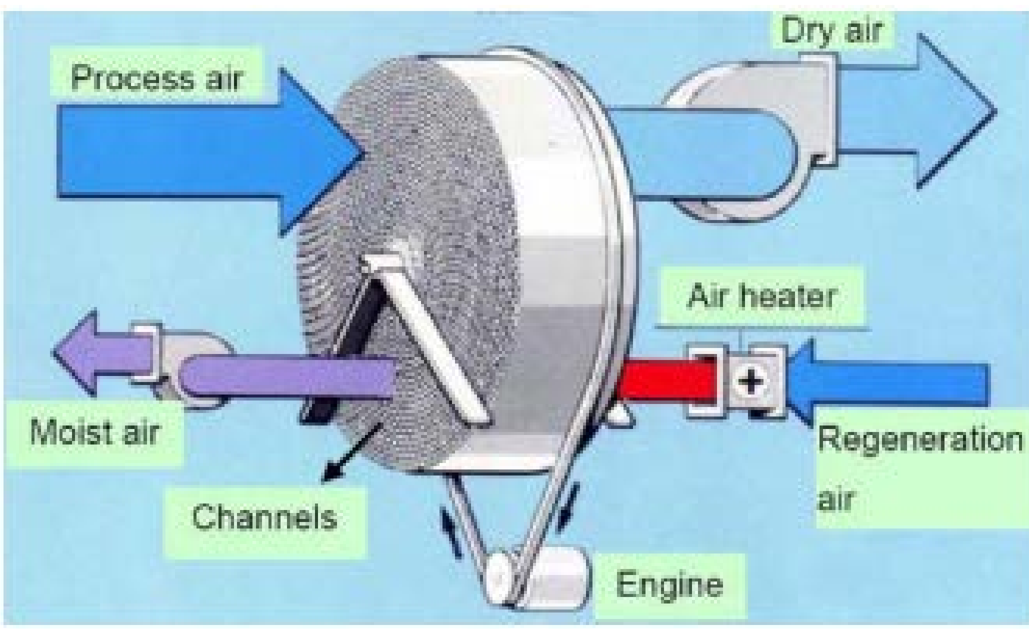

Figure 2. 4 Schematic of a rotary desiccant wheel

2.2.2 Condensate Recovery

Water is a most treasured natural resource; its responsible use is our collective responsibility. This is even more so for dry and arid regions, where evaporative cooling technology is most applicable. The water consumption of evaporative cooling technology is therefore one area where we can contribute to the responsible use of one of nature's most treasured gift to Mankind. Condensate recovery systems for HVAC systems is clearly a way to go in this regard. Salem A. *etal* [32] conducted a detailed review on HVAC condensate recovery systems, quality and concerns. They found that the quality of condensate recovery is relatively high, the water being almost mineral free and with very low content of total dissolved solids (TDS). They [32] highlighted certain meteorological factors that affect condensate recovery. These include ambient temperature, dew point temperature, ambient temperature, annual operating hours [33].

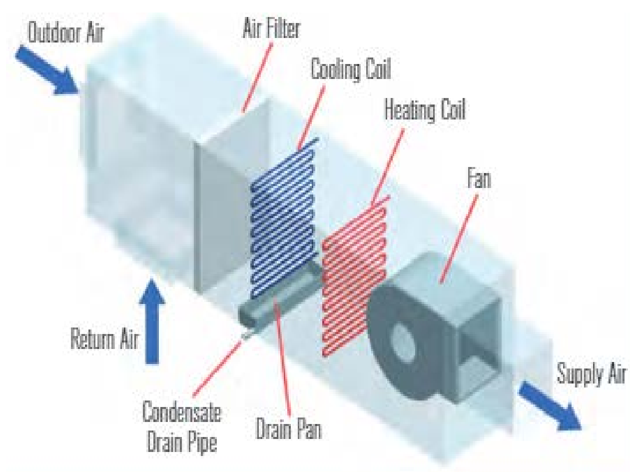

Figure 2. 5 A simple condensate recovery system [34]

The recovery of condensate has been shown to significantly improve the energy efficiency of buildings. Gawe *et al.* [34] worked on the quality of condensate from air handling units. They identified some of the on-site uses of condensate to include make up water for evaporative cooling processes among other uses. The idea behind atmospheric moisture harvesting by direct cooling is quite simple; ambient air meets a cold surface and the temperature difference facilitates heat transfer from the air to the surface resulting in temperature decrease of the air that is in close proximity to the surface, and condensation of the vapor that exceeds the moisture saturation capacity of the chilled air takes place. The condition of the process air after the cooling pad is usually cool and humid. Effective dehumidification implies that the exhaust air stream should be almost as humid as the process air stream. [35, 36, 37]

This work proposes a novel condensate recovery and re-use method. The mass of water available in the exhaust stream is a function of the mass of water evaporated, mass of water adsorbed, and the mass of water desorbed from the desiccant material in the return air stream. So, the following parameters shall be critically investigated.

The objective is to provide suitable air quality for data centers. And in addition to that to be able to recover water from the exhaust air stream which may be re-used for the evaporative cooling. In chapter three we will show a mathematical a thermodynamic analysis of this process.

2.3 Data Center Cooling

The aim of this work is to investigate the suitability of evaporative cooling coupled with desiccant based dehumidification for data centers. Presently many data center operators employ a variety of technologies to effectively cool data centers. Among many of the alternatives are water cooling, free cooling, evaporative cooling and containment strategy to enhance power utilization efficiency. An industry-based parameter for measuring the efficiency of data centers is the power utilization efficiency (PUE). [38] Table 2.1 below shows data center cooling technologies, classified as conventional and alternative cooling technologies.

Table 2. 1 Data center cooling technology

Conventional Cooling	**Alternative Cooling**
Computer room air conditioning	Water cooling [lake sourced]
	Free cooling
	Direct & Indirect Evaporative Cooling

The use of alternative cooling technologies such as water cooling and evaporative cooling has been proposed as alternatives to conventional vapor compression cooling systems. However, both alternatives are beset with setbacks such as the availability of water bodies for water cooling, and there are equally environmental concerns pertaining to the use of water cooling for data centers; some concerns have been raised about tampering with natural bodies and perhaps the possibility of a steady rise in temperature of the water body. [39]

Evaporative cooling on the other hand is most effective in arid climates, where water might be a scarce resource. Now, irrespective of climate, evaporative cooling is currently used as a means to reject heat from condensers in data centers. These uses have come under attack by environmentalists who express serious concern due to the chemical and water use of this method. [12, 3] The introduction of condensate recovery to desiccant assisted evaporative cooling is expected to minimize the water consumption of evaporative cooling and hence make this technology more usable.

2.4 ASHRAE guidelines for data centers

The ASHRAE technical committee is the body responsible for thermal guidelines for data centers. “ASHRAE TC 9.9 was formed in response to the lack of effective information transfer between the building, HVAC, and IT industries. Its mission is to be recognized by all areas of the Datacom industries as the unbiased engineering leader in HVAC and effective provider of technical Datacom information Prior to publication in 2004 of the first edition, there was no single source in the data center industry for ITE temperature and humidity requirements. In the second edition of the *Thermal Guidelines for Data Processing Environments* the recommended envelope was expanded to provide data center operators guidance on maintaining high reliability and also operating their data centers in the most energy efficient manner". [40]

This envelope was created for general use across all types of businesses and conditions. The second edition also introduced new allowable envelopes (A1 and A2), that expanded the maximum allowable dry-bulb temperature to 32°C and 35°C, respectively, with a maximum relative humidity limit of 80% and a minimum relative humidity limit of 20%. These envelopes also differed in the maximum allowable dew point temperature. These allowable envelopes offered data center operators the flexibility in using an operating envelope that matched their business need and to weigh the balance between the additional energy savings of the cooling system versus the deleterious effects that may be created on total cost of ownership by operating outside the recommended range". [41] These equipment classes are explained in ASHRAE thermal guidelines for data centers [41]

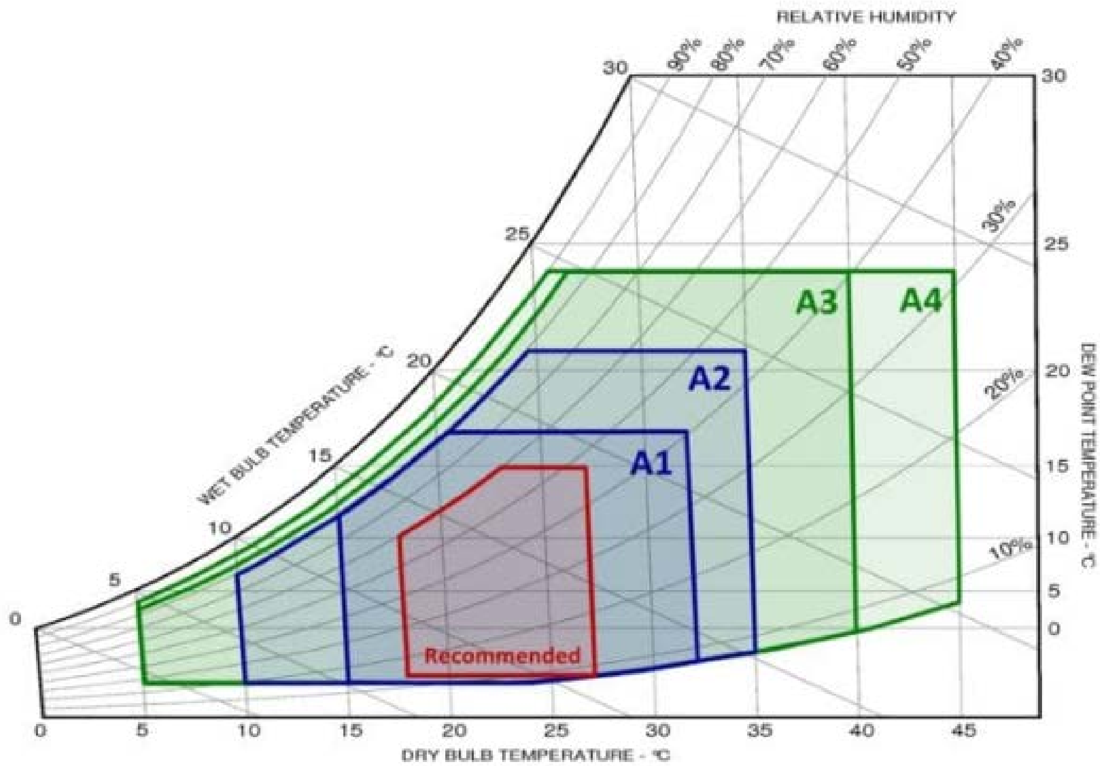

Figure 2. 6 ASHRAE thermal guidelines for different classes of data center equipment [41]

A lot of research dollars continue to be invested in enhancing the thermal management of data centers. A direct consequence of this is the increase in allowable maximum temperatures and a wider range of relative humidity, namely from about 20% to 80% as highlighted above. We now turn our attention to the materials and methods portion of this work.

3. MATERIALS AND METHODS

3.1.1 Materials

A six-inch diameter air duct attached to a Fantech FR 110 series was used to supply air to a direct evaporative cooling unit. The Direct evaporative cooling unit is composed of a cellulose cooling pad manufactured by Dial Inc. and water is run over the pad enclosed in an induct air filtration unit. The sensibly cooled air with increased relative humidity is now passed through a rotary desiccant wheel, which design is inspired by Dominc O'Connor *et al.* [29] Air of suitable quality is now supplied to a modular data center. Heat generation in the modular data center is stimulated by way of a space heater of controllable temperature. As cool air is supplied to the data center to cool same, warm air which is waste heat from the data center is now returned via another Fantech impeller fan series 100. The return air stream which carries waste heat from the data center is now used to reactivate the desiccant material impregnated in the rotary desiccant wheel. Warm and moisture rich air leaving the desiccant wheel is now passed through a moisture recovery unit (MRU), where the moisture content of the warm return air stream is stripped as it runs unto a cold plate cooled by the Peltier effect. Condensate is recovered and measured.

Humidity and temperature readings was monitored throughout the experiments. K-type thermocouples and Omega manufactured RH-USB probes was attached to the set up at various points of interest. Readings were documented via a bench-link data logger. The accuracy of the thermocouples is within ±1% accuracy while the humidity probes are accurate to within ±2.5%. To cool the cold plate where condensate was recovered, four thermoelectric modules were attached to a heat sink, where the hot side was convectively cooled to increase the heat transfer efficiency and to cool the aluminum metal sheet so attached. This was placed in the moisture recovery unit where; a box shaped MRU that allows for the interaction of the moist air with the cold surface. The quantity of moisture recovered was measured and recorded accordingly

The cellulose pads used were manufactured by Dial Inc. with reported porosity of 92%. The Pads were cut to 14 x 12 inches to fit into the induct air filtration unit adapted for this experiment. Pad thickness was varied between 2, 3 and 4 inches respectively.

A water distribution system was fabricated in house to distribute water over the cellulose pad via a ½ inch internal diameter pipe attached to a Watson Marlow Peristaltic pump. The water distribution pipe was placed at the middle of the water supply system to enable even distribution of water across the flow channels. This created a cross flow heat exchange system between the water molecules and the air being forced through the evaporative cooling pad. In the next section We show the optimization of water flow for enhanced sensible cooling. Next, a lab scale rotary desiccant wheel was fabricated using acrylic glass sheets and parts printed on a 3-D printer.

Figure 3. 1 Cellulose pads of thickness 2, 3 and 4 inches

Table 3. 1 Evaporative cooling pad dimensions

S/N	**Dimension (inches)**	**Pad Depth (inches)**
1	14 x 12	2
2	14 x 12	3
3	14 x 12	4

3.1.2 Materials for Dehumidification-Silica-Gel (SiO_2)

An adsorptive desiccant material, silica-gel is a hard, crystalline substance. It is reportedly [20] able to hold about 40% of its own mass in moisture, with porosity of about 50-70%. It has a bulk density range of 480-720kg/m3 and a specific heat capacity, C_p of 1.13kJ/kg-K. As highlighted above, the adsorptive nature of this desiccant material and its wide reactivation temperature range makes it suitable for our purpose. Shown in figure 3.2 below, the silica-gel used for this work has diameter of 3-5mm. It also has a wider temperature range for reactivation.

Figure 3. 2 Silica gel beads and silica gel beads impregnated into the cassettes

Each cassette was impregnated with about 111 grams of Silica gel beads of diameter 3-5mm.The rotary desiccant wheel was powered by a small electric motor of 50/60Hz with an average Rotary Speed of 10/12 revolutions per minute (rpm).

3.1.3 Thermoelectric Modules for Cooling of Cold plate

For thermoelectric modules generally, the thermodynamic conversion of heat to work involves four unique processes. Of these, two are irreversible, while the other two are reversible. The reversible processes are Seebeck/Peltier and Thomson effects [42]. The irreversible processes are the Fourier effect, which accounts for heat conduction, and the Joule effect, which accounts for electrical résistance. The Seebeck and Peltier effects are actually two physically relative effects in that where a temperature difference across a module leads to electric power generation in the Seebeck effect, the input of electric power - hence an imposition of potential difference across the module -- leads to the generation of a temperature difference on both sides of the module, known as the Peltier effect [43]. Thus, thermoelectric modules or devices can convert electrical energy into thermal energy and vice versa. This ability is based on the fact that temperature differences can set charge carriers in motion.

3.2 Methods

The approach taken is one of a simple thermodynamic modelling of the cooling process and the experimental verification of the same. “Two powerful analytical tools of the HVAC engineer are the conservation of energy or energy balance and the conservation of mass or mass balance”. [44] Following the above injunction we present a mathematical model of the process below

The mathematical model described below is designed to track the moisture consumption of the desiccant assisted evaporative cooling system so described. The process is clearly a three-stage process, namely direct evaporative cooling, rotary desiccant wheel dehumidification and reactivation using waste heat and the recovery of moisture from the exhaust air stream.

3.2.1 Analytical Model of the Desiccant Assisted Evaporative Cooling System

We first define the variables m_a as the mass flow rate of dry air. W as the humidity ratio of the air. So that W $= \frac{m_w}{m_a}$, where m_w is the mass flow rate of water.

We also define $m_{w,\ in}$ and $m_{w,\ out}$ as the flow rate of water entering and exiting the pad. Now the composition of air is given by $m_a(1+W)$

So that by conservation of mass we have

$$\dot{m}_a(1+W_1)+\dot{m}_{w,in}=\dot{m}_{w,out}+\dot{m}_a(1+W_2) \qquad (1)$$

The subscripts 1 and 2 indicate before and after the evaporative cooling pad respectively.
The enthalpy of air is given as $h_a + Wh_w$

Where $h_a = enthalpy\ of\ dry\ air$ and $h_w = enthalpy\ of\ moisture\ in\ air.$

We may now write an energy balance across the evaporative cooling pad as follows

$$\dot{m}_a(h_a+Wh_w)-\dot{m}_a(h_a+W_2h_w)=m_E C_{pE}\frac{dT}{dt}+m_w C_{pw}\frac{dT}{dt} \qquad (2)$$

Since mass is conserved $m_a\ is\ constant\ and\ at\ steady\ state\ \frac{dT}{dt}=0$

A critical information to derive from this is the mass of water evaporated by the process which we denote by $\dot{m}_{evap} = \dot{m}_a(W_2 - W_1)$ It is losses due to this particular variable that we are interested in for this work.

Next, we shift our attention to the dehumidification section of the process. We first write a mass balance on the desiccant wheel. We assume a 1:1 split between the dehumidification side and the regeneration side of the wheel.

$$\dot{m}_a(1 + W_2) - \dot{m}_a(1 + W_3) = \dot{m}_r \tag{3}$$

Where $\dot{m}_r$=mass of water adsorbed.
And the mass of water adsorbed by the desiccant is given by

$$\dot{m}_r = \dot{m}_a(W_2 - W_3) \tag{4}$$

The Supply air humidity ratio is W_2.

On the regeneration/return air stream side we have the following mass balance

$$\dot{m}_a(1 + W_3) - \dot{m}_a(1 + W_4) = \dot{m}_{rd} \tag{5}$$

Mass of water desorbed from the desiccant is

$$\dot{m}_{des} = \dot{m}_a(W_4 - W_3) \tag{6}$$

It is also important to note that flow of water in and out of the pad is conserved hence

$$\dot{m}_{w,in} = \dot{m}_{w,out} \tag{7}$$

Mass balance on the pad is given by

$$\dot{m}_a(1 + W_1) + \dot{m}_{w,in} = \dot{m}_{w,out} + \dot{m}_a(1 + W_2). \tag{8}$$

Finally, we write a mass balance for the moisture recovery unit

$$\dot{m}_a(1 + W_4) - \dot{m}_a(1 + W_5) = \dot{m}_{condensate} \tag{9}$$

We are particularly interested in $\dot{m}_a(W_2 - W_1)$ which is the mass of water evaporated.

$W_1 < W_{2,} W_3 < W_2, W_3 < W_4, and\ W_5 < W_4$

$\dot{m}_a(W_2 - W_3)$= mass of water adsorbed

$\dot{m}_a(W_3 - W_4) = mass\ of\ desorbed$

$\dot{m}_a(W_4 - W_5) = mass\ of\ water\ condensed$

We shall be investigating the relationship between the mass of water evaporated $\dot{m}_a(W_2 - W_1)$ and the mass of water desorbed $\dot{m}_a(W_3 - W_4) = mass\ of\ desorbed$.

3.2.2 Experimental Procedure

Air is heated up to simulate outdoor conditions and drawn in by the impeller Fantech FR 110 series Fan, as the water distribution system described above runs water evenly over the cellulose pad thereby creating a cross flow heat exchange configuration between the warm air and the water interacting on the evaporative cooling pad. It has been shown [45] that the residence time and pad depth are essential to the cooling efficiency of the evaporative cooling unit.

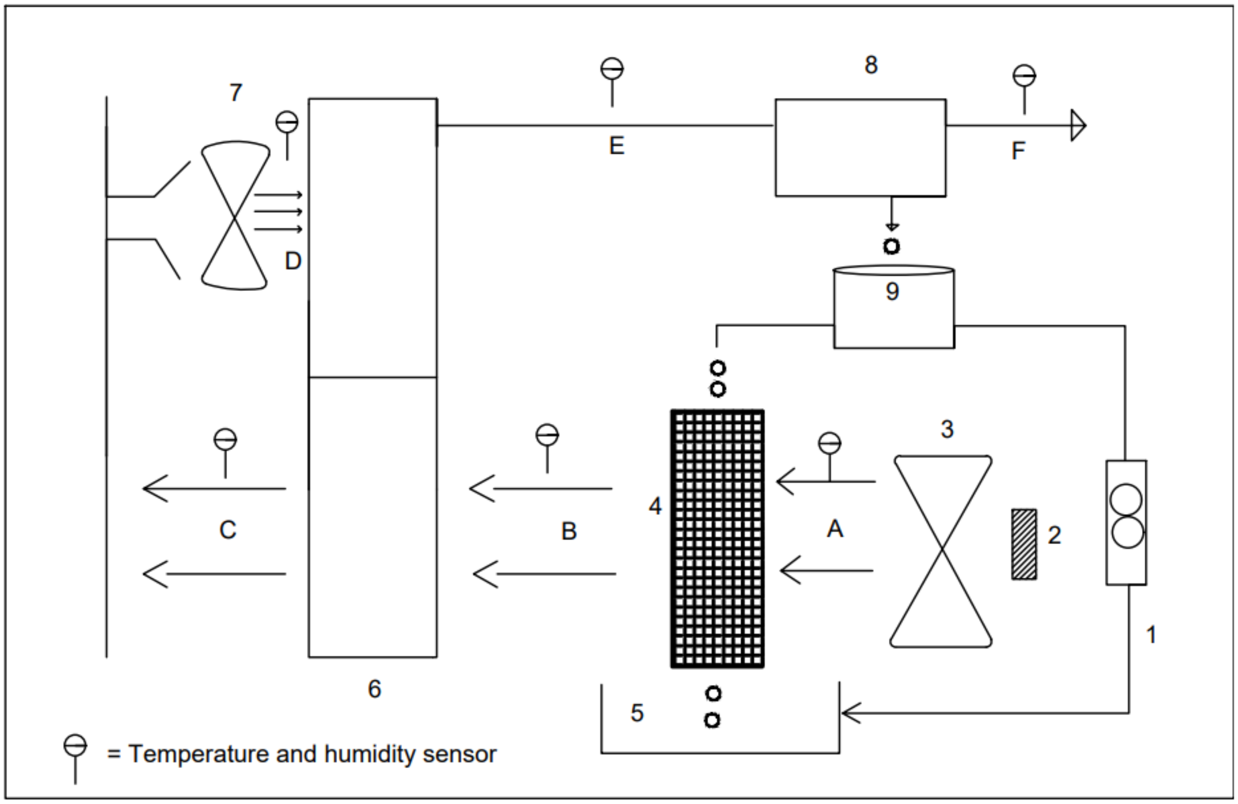

Figure 3. 3 Schematic of experimental set up

Notations: A-inlet air stream, B-evaporatively cooled air stream, C-dehumidified air stream to data Center-Warm air from data center-warm and humid airstream, F-Exhaust air stream.1-Flow meter,2-Electric heater,3-inlet fan,4- evaporative cooling pad,5-water sump,6-rotary desiccant wheel,7-return air fan,8-moisture recovery unit,9-water reservoir

3.3 Optimal Flow Rate of Water Over the Cellulose Pad

A simple experiment was conducted to determine the optimal flow rate that effectively wets the cellulose pad. Attached to a Watson Marlow Peristatic pump is a half inch internal diameter pipe running water over the pad. Varying the flow rate from 0.32 to 0.55kg/min, it was found that at a flow rate of 0.46kg/min the greatest cooling effect was observed. So, holding all other variables such as pad thickness, inlet air speed and inlet air conditions constant, a flow rate of 0.46kg/min was determined to be the most effective for this set-up. Figure 3.4 below shows the optimal flow rate for enhanced cooling effect for this experimental set up.

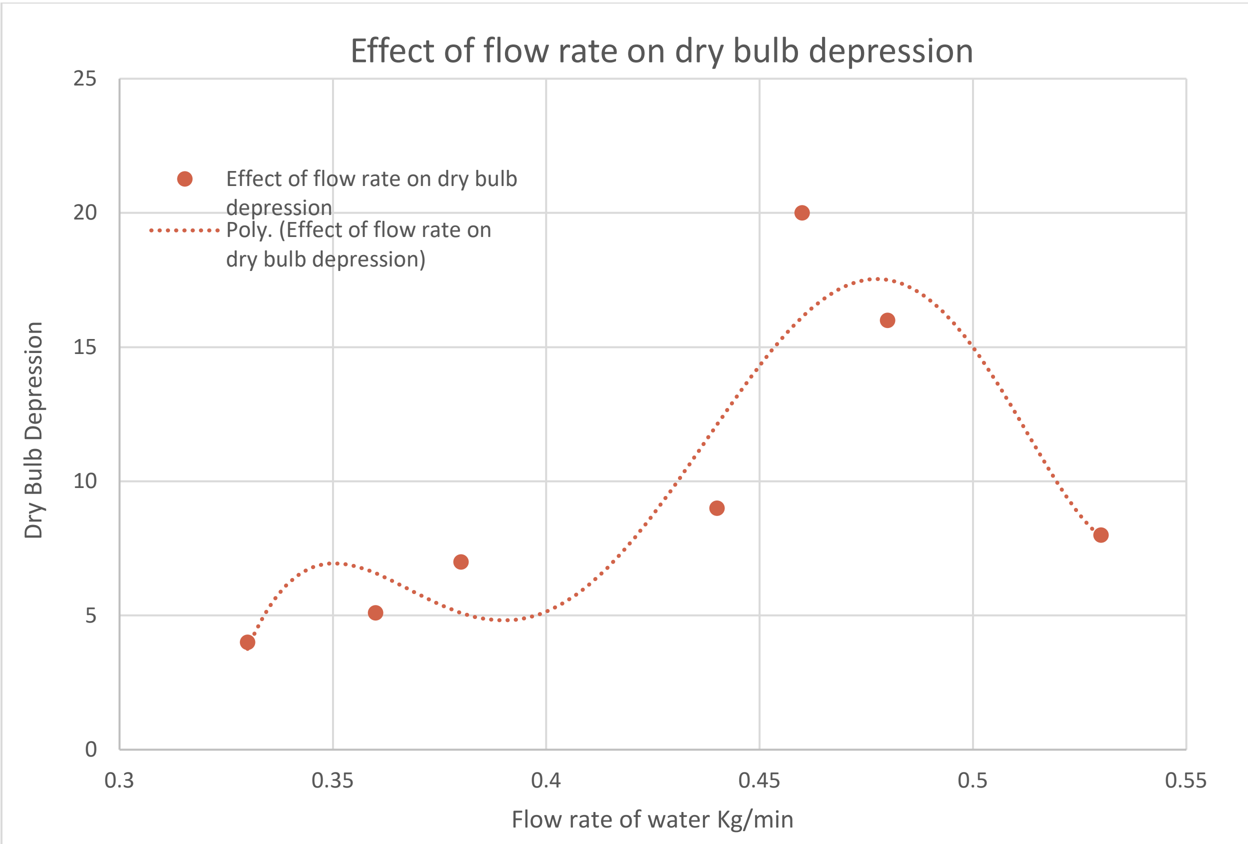

Figure 3. 4 Optimal flow rate of water over the pad

4. RESULTS AND DISCUSSION

In this chapter the results from experiments conducted using the set up schematically described in chapter three and shown in figure 3.3 are analyzed and discussed. The experiments validate the mathematical model and parametric studies conducted are here presented.

During the evaporative cooling process, heat and mass transfer occurs, the air supplies the energy needed to vaporize the water. Therefore, there’s transfer of mass in the form of increase in humidity or water vapor content of the air

We shall look at the cooling stage, the water consumption through evaporation, dehumidification and condensate recovery, in that sequence.

4.1 Dry bulb depression

In this work we have shown that evaporative cooling can be used to sensibly cool warm air. We look at some of the sensible cooling experiments conducted and the results obtained below. In exchanging sensible heat for latent heat, certain factors influence the quantity of heat loss by the warm air entering the evaporative cooling unit.

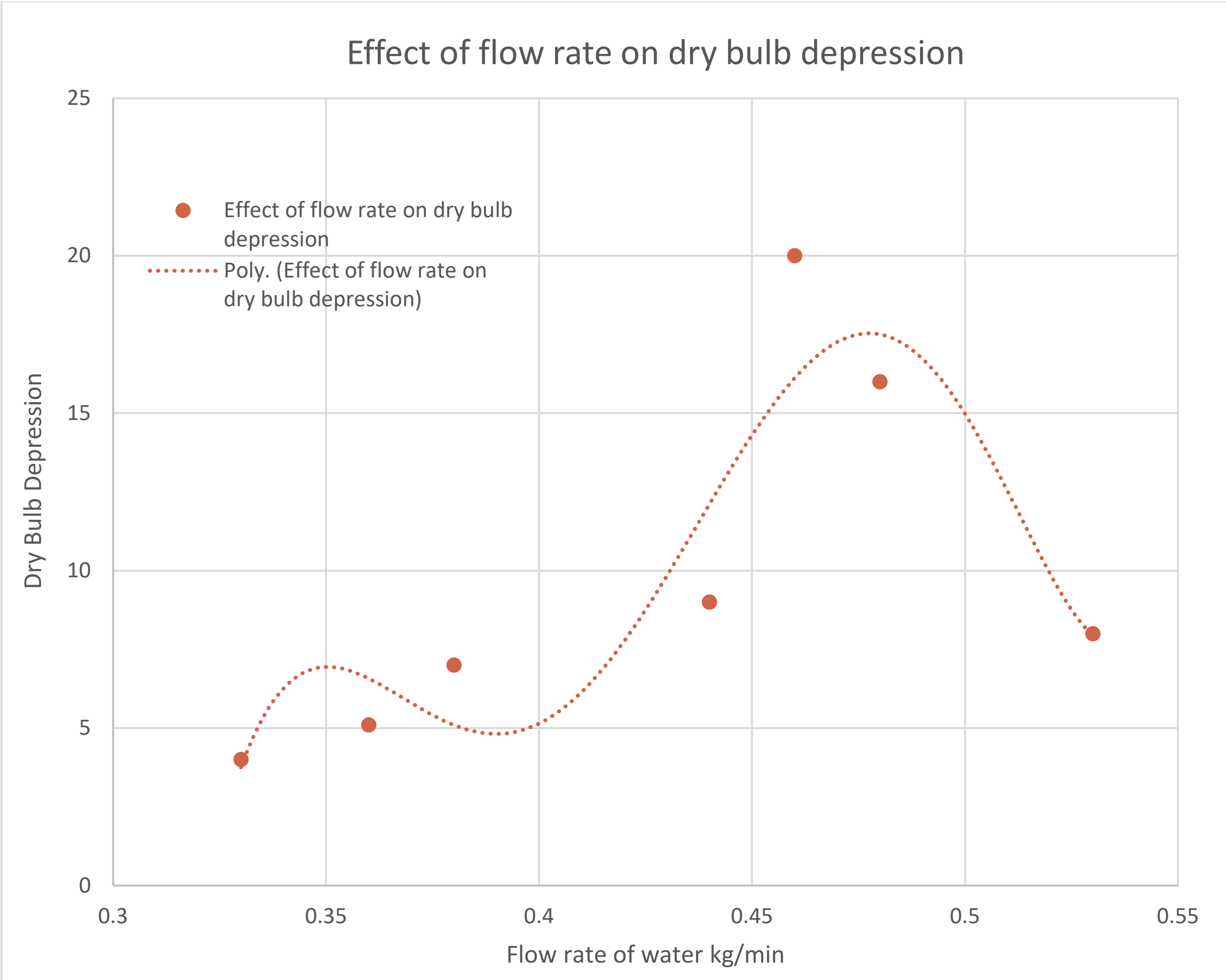

Figure 4. 1 Effect of flow rate of water on dry bulb depression

Using the set up depicted in figure 3.3 in chapter three above the mass flow rate of water running over the evaporative cooling pad also shown in figure 4.1 above was varied from 0.32kg/min to 0.52kg/min to determine a suitable flow rate for water over the pad. The sensible cooling observed increased as the mass flow rate of water over the pad increased. This is easily understood, when We consider the fact that evaporatively cooled air essentially exchanges sensible heat from the ambient with latent heat from the water flowing over the pad. As the flow rate of water over the pad increases, the highly wettable cellulose pad now holds more water molecules and hence the enhanced sensible cooling noted. For this particular set up, a peak was obtained at about 0.46kg/min. This approaches the theoretical temperature difference attainable given the cellulose pad dimensions. This theoretical maximum is the difference between the inlet air dry bulb temperature and the cooled air wet bulb temperature. The temperature readings were collected using k type thermocouples attached to a data logger.

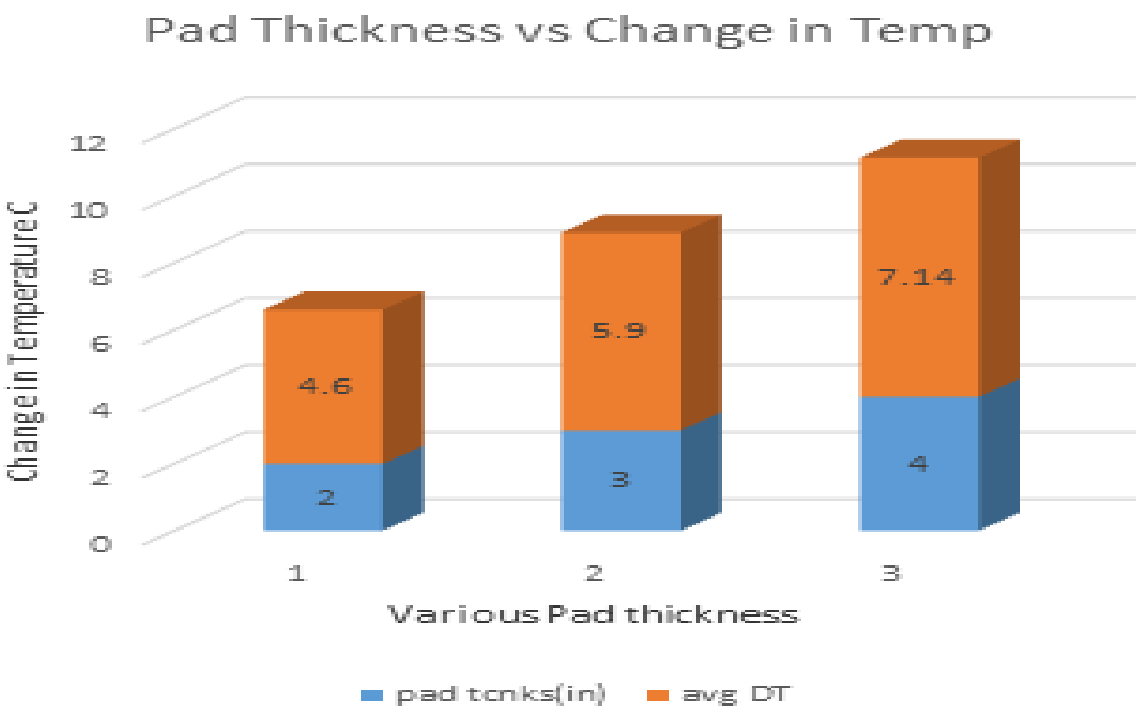

Figure 4. 2 Effect of pad thickness on dry bulb depression

4.1.1 Effect of Frontal air velocity on dry bulb depression

Using an anemometer, the speed of the inlet air was measured and varied from 0.86m/s to 2.56 m/s to determine the effect of inlet air speed on the cooling effect observed. At an inlet air speed of 0.86m/s the change in temperature (dry bulb depression) was recorded as 6.5ºC and at an inlet air speed of 1.1m/s a dry bulb depression of 7.6ºC was observed and finally the maximum dry bulb depression was observed at an inlet speed of 2.56m/s. The speed of the air being forced through the pad accelerates the rate of evaporation and hence delivers superior cooling at higher inlet air speeds.

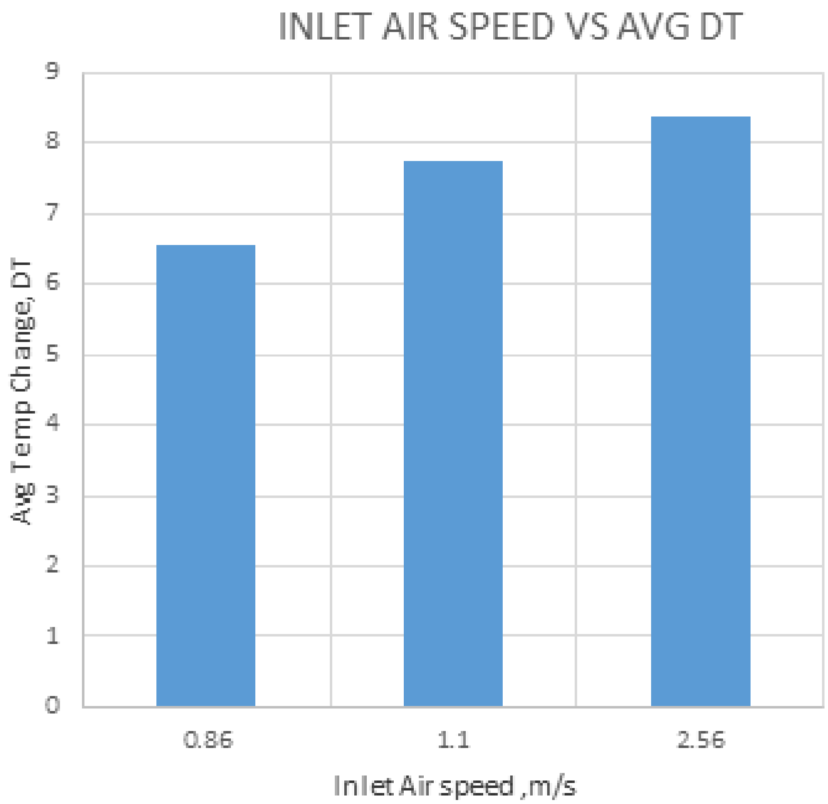

Figure 4. 3 Effect of frontal air velocity on dry bulb depression

4.2 Effectiveness of the direct evaporative cooling unit

The effectiveness of the direct evaporative cooling unit is defined as the ratio of the dry bulb depression to the wet bulb depression. Now the dry bulb depression refers to the temperature drop across the pad (dry bulb temperature) as the air goes through the pad. Essentially this describes the actual cooling produced in relation to the theoretical maximum attainable. Most commercially available evaporative coolers report cooling effectiveness values between 80% to 93 %.

The following equation describes the effectiveness, sometimes called the saturation efficiency of the evaporative cooling unit. $T_{in,db}$ is the dry bulb temperature of the inlet air, $T_{pa,db}$ is the dry bulb temperature of the process air (cooled air) while $T_{pa,wb}$ is the wet bulb temperature of the process air.

$$\varepsilon = \frac{T_{in,db} - T_{pa,db}}{T_{in,db} - T_{pa,wb}} \tag{10}$$

It was observed that the pad thickness and hence the residence time was positively correlated with the effectiveness of the direct evaporative cooling section of the experiment. For all three pads (2,3 and 4 inch) the effectiveness was found to range from 45% to over 80% for the 4-inch pad.

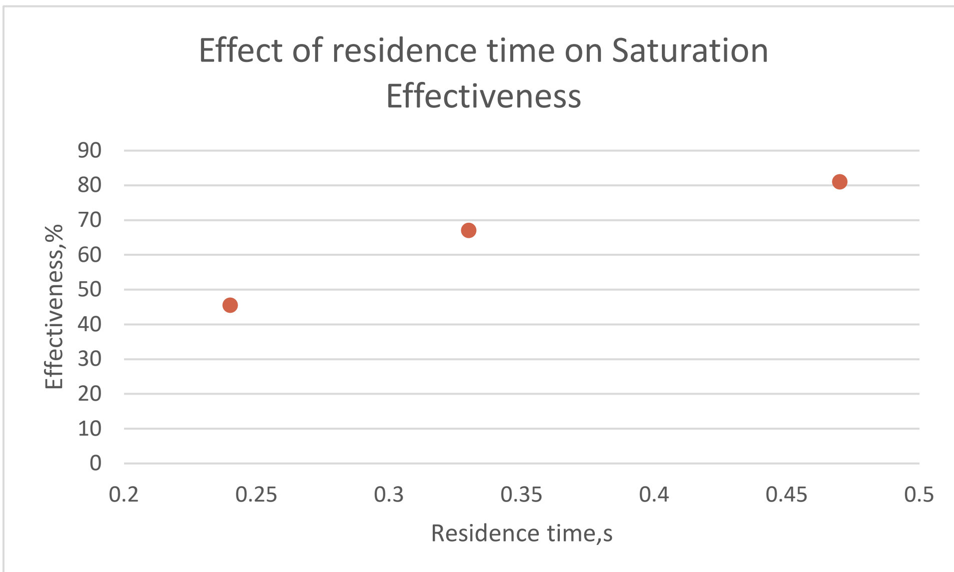

Figure 4. 4 Effect of residence time of inlet air on effectiveness of the evaporative cooling unit

By the same logic, since the pad depth affects the residence time as earlier shown, the effectiveness was also compared with the depth of each pad and figure 4.5 highlights the relationship so obtained.

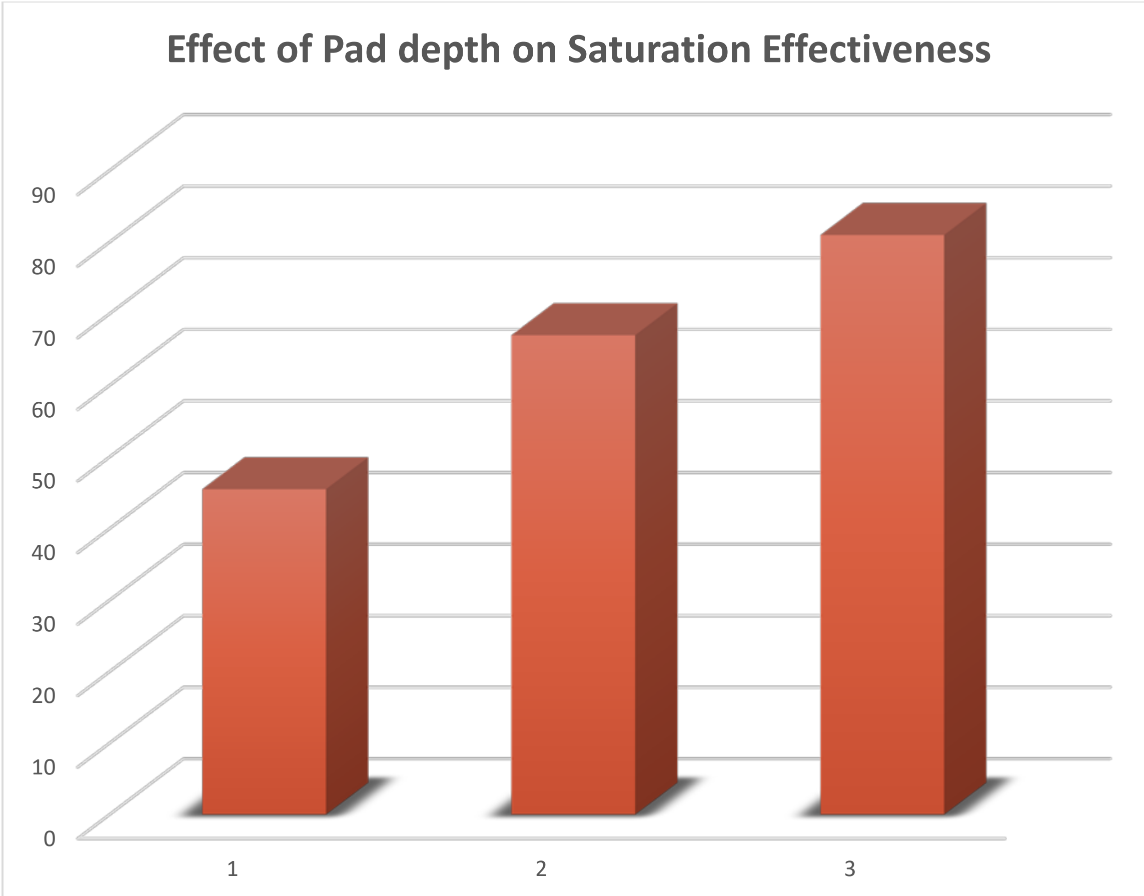

Figure 4. 5 Effect of pad thickness on the effectiveness of the evaporative cooling unit

4.3 Mean Residence Time

The residence time of the air as it goes through the pad was calculated using the mean residence time formula. $t_R = \frac{\rho V}{\dot{m}}$

Table 4. 1 Mean residence time based on pad thickness and optimal velocity

Pad Thickness (In.)	2 Inch Pad	3 Inch Pad	4 Inch Pad
Residence time(s)	0.24	0.33	0.47

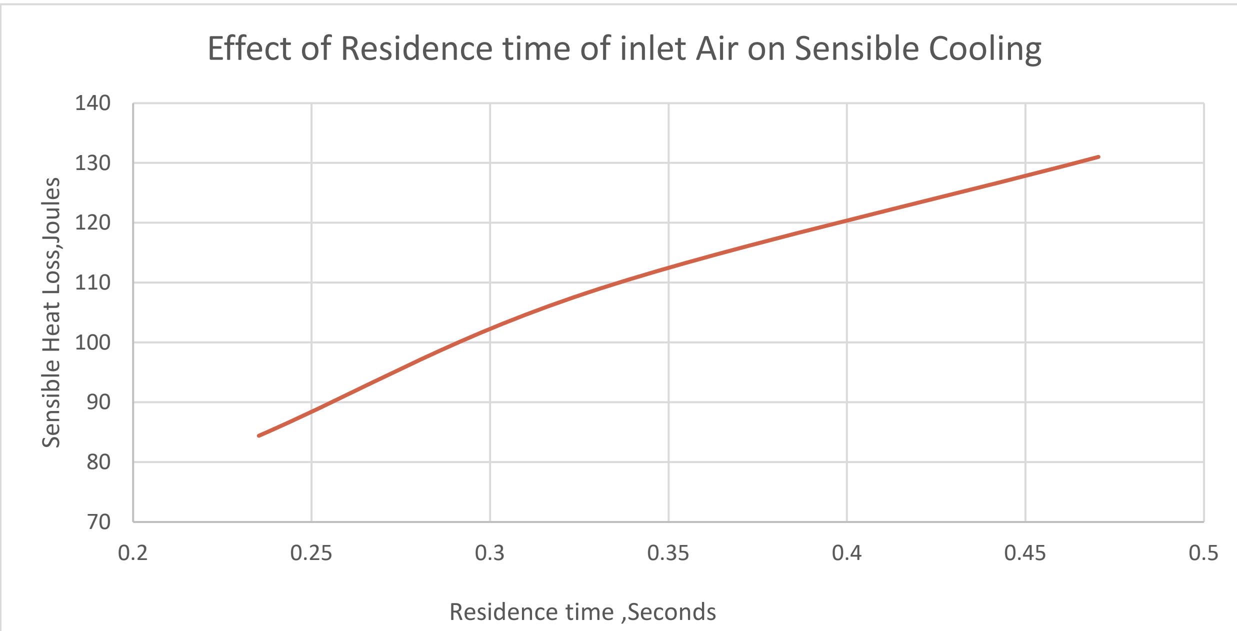

Figure 4. 6 The effect of residence time on sensible cooling

As expected, holding all other variables constant, the greater the residence time of air on the pad, the greater the sensible cooling realized. This is also impacted by the condition of the inlet air in terms of humidity and temperature. That is, the greater the humidity of the inlet air the less the sensible cooling realized as the air has less capacity to hold moisture.

4.4 Thermally Suitable Air Quality for Data Centers

We set out to show that evaporative cooling coupled with desiccant based dehumidification can produce thermally suitable air quality for data center cooling purposes. In terms of suitability our focus is on the temperature and relative humidity of the supply air. Whilst direct evaporative cooling increases the relative humidity as sensible heat is converted to latent heat, the attendant increase in relative humidity of the air is offset by the use of a rotary desiccant wheel. Using the same set up shown in figure 3.3, chapter three above but with a fixed bed desiccant based dehumidifier, and with inlet air condition varying from 31°C to 33°C and relative humidity varying from 33% to about 38%, the desiccant assisted evaporative cooling unit was allowed to run for about twelve hours to observe the suitability of the air quality produced in accordance with the thermal guidelines for data center cooling. Looking at fig 4.7, We observe that the first 4.5 hours produced air that is suitable in both temperature and relative humidity and as the desiccant's capacity to absorb moisture diminished, the relative humidity of the supply air began to climb, all the time rising to about 63%.

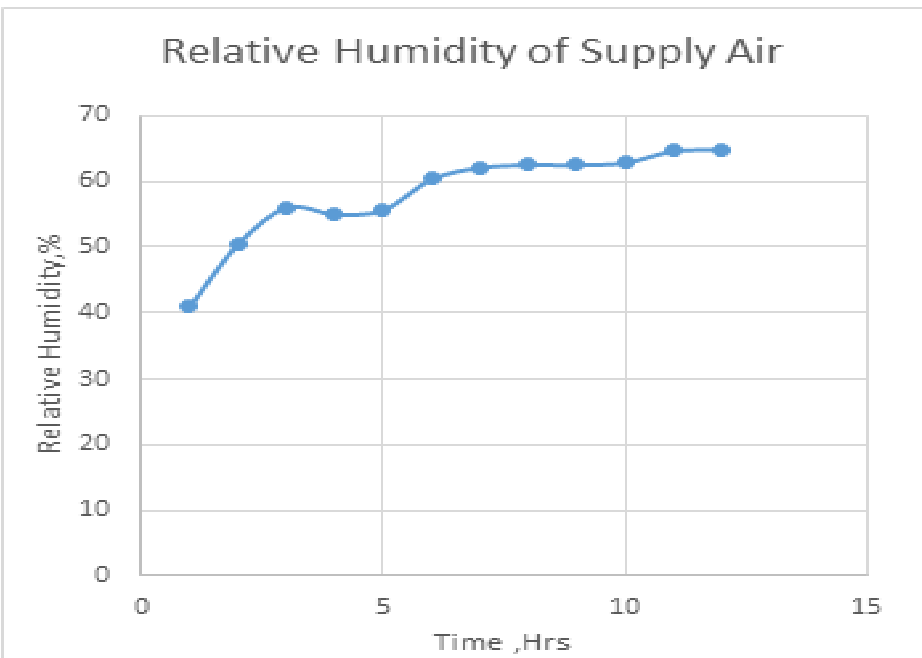

Figure 4. 7 Relative humidity of supply air documented for 12 hours

Even at a global maximum of 63% shown in figure 4.7 above, this is still very suitable for data centers, as modern-day servers and other data center equipment are now designed to withstand such relative humidity, RH values. Now turning our attention to figure 4.8 below and with inlet temperatures staying relatively the same, the desiccant assisted direct evaporative cooling system was allowed to run for the same period of time (about twelve hours) and the average inlet and supply air dry bulb temperatures measured and documented below. It was found that the dry bulb depression improved with time. This is due in part to the wettability of the pad and saturation efficiency of the direct evaporative cooling system. At an inlet dry bulb temperature of 31^{o}C, the dry bulb depression started out at about six degrees Celsius in the first hour and went on to about 9^{o}C in the sixth hour and by the 12th hour the dry bulb depression was at a peak of 12^{o}C.

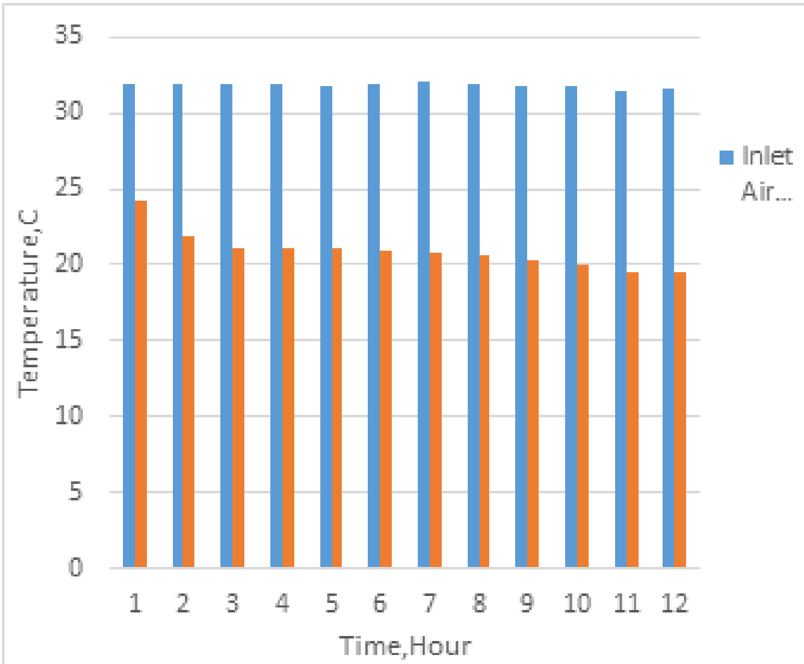

Figure 4. 8 Temperature evolution monitored for 12 hours

4.5 Heat and Mass Transfer

The overarching thought in this work has been the fact that the evaporative cooling process is essentially a conversion of sensible heat to latent heat. The sensible heat involves the change in temperature which takes place as the warm air goes through the pad and becomes cool, and, while the latent heat involves the change of state of water running over the cellulose pad to vapor. This phase change by the water running over the pad also represents mass transfer. We present an analysis below of the heat and mass transfer of the direct evaporative cooling section of this work.

4.5.1 Sensible Heat Loss [Q_s]

The sensible heat Q_s is the product of the heat capacity ($\dot{m}_a C_p$) and the temperature difference of the air across the pad. This shows the energy lost by the air to the water, for evaporation of the water from the surface of the pad.

$$Q_s = \dot{m}_a C_p (T_{in} - T_{pa}) \tag{11}$$

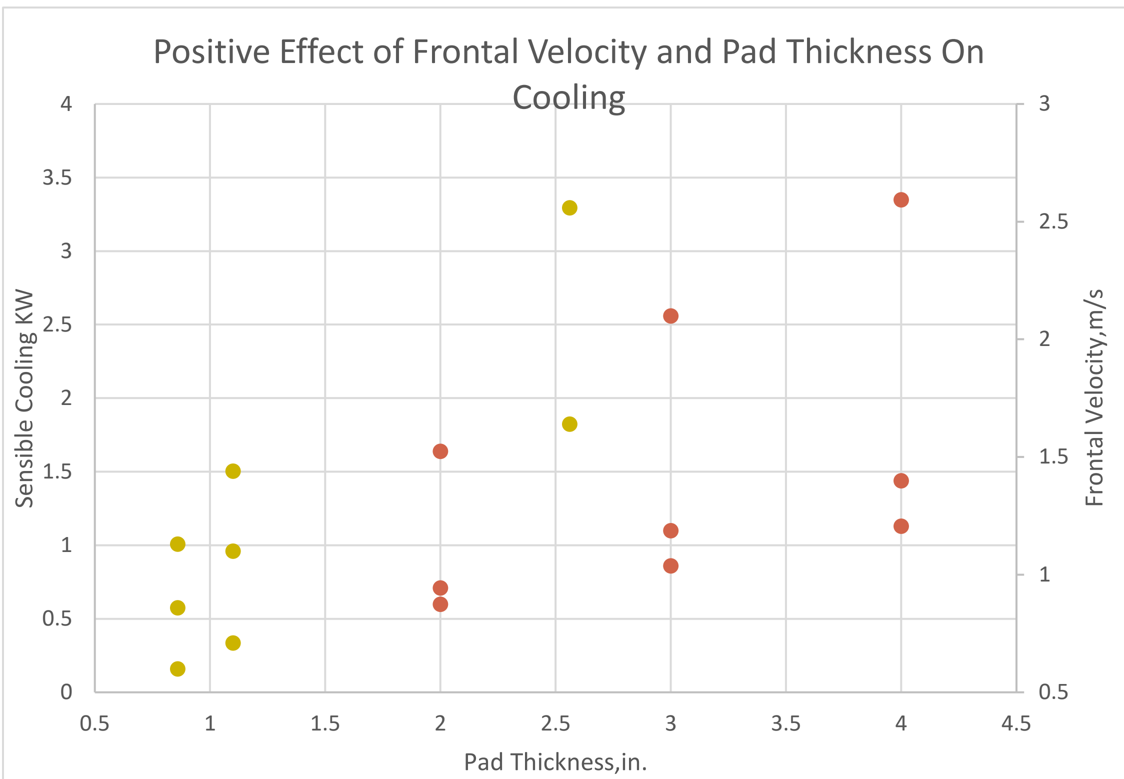

Figure 4. 9 Effect of pad thickness and frontal velocity on cooling

In Figure 4.9 above we see the relationship between Sensible cooling, pad depth and frontal air velocity (Velocity of the inlet air). Recall that three pads of thickness 2 ,3 and 4 inches were used for these experiments. Holding pad thickness as the independent variable, the sensible cooling improved the thicker the pad. Shown by the equation above the Sensible cooling is the product of the mass flow rate of the air entering the pad and the specific heat capacity of the air and the temperature gradient between the inlet air temperature ,T_{in} and the process air temperature(that is the cooled air) T_{pa}.The sensible cooling ranged from about 1kW on the 2 inch pad with a frontal inlet air velocity of 0.86m/s yielding a mass flow rate of 0.11kg/s to 3.4kW on the 4 inch pad with frontal inlet velocity of 2.56m/s and yielding a mass flow rate of 0.33kg/s. It is also noteworthy that when the pad depth is held constant the frontal inlet air velocity is varied the cooling sensible cooling is enhanced as well. This is shown for pad depth 4 in. below. The increased velocity induces better heat transfer as the air and water interaction enhances the process of conversion of sensible heat to latent heat, thereby enhancing the sensible cooling.

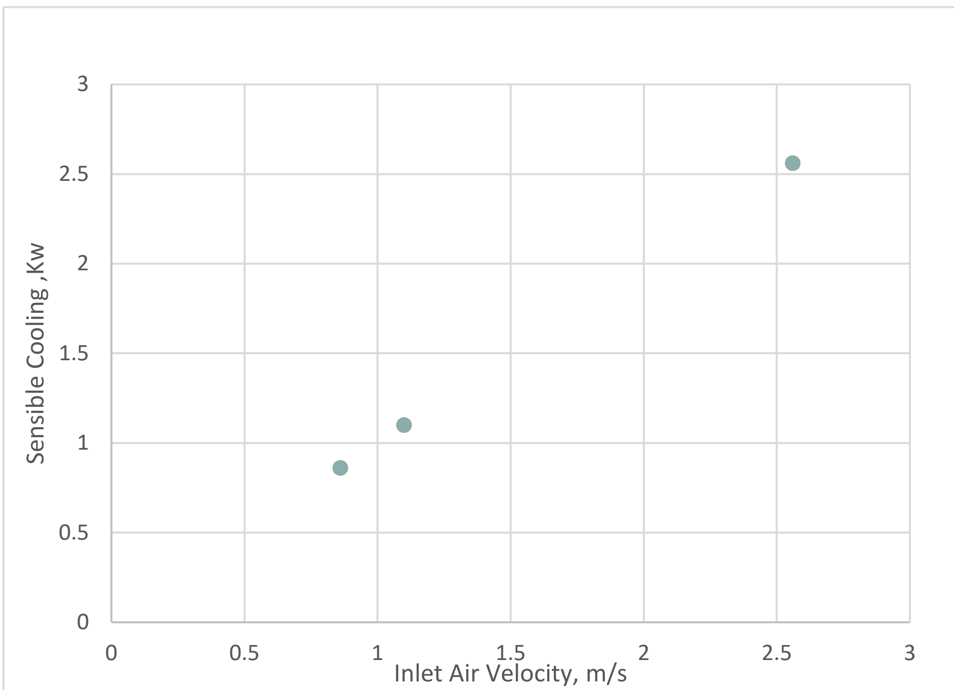

Figure 4. 10 Effect of inlet air velocity on sensible cooling

Here we see the direct relationship between the sensible heat loss by the warm air passing through the pad and the frontal air velocity. Recall that the sensible heat loss is given by equation 4.3.1.The right hand side of equation 4.3.1 is essentially the product of the mass flow rate of dry air entering the pad ,and specific heat capacity of dry air and the temperature gradient between inlet air and process air(cooled air).The mass flow rate of the air is dependent on the inlet air velocity and given by the equation $\dot{m}_a = \rho A V$ where the density and cross sectional area are held constant, an increase in frontal velocity therefore leads to an increase in sensible heat loss.

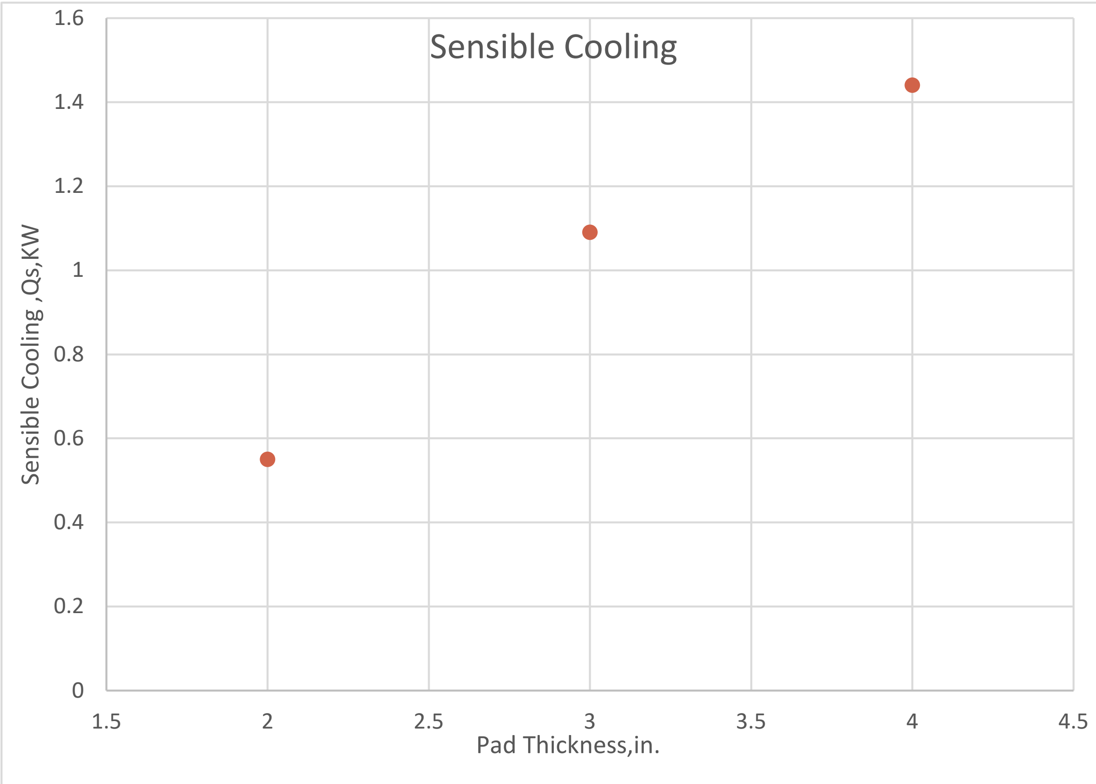

Figure 4. 11 Sensible heat cooling as a function of pad thickness

4.5.2 Latent heat gain

The Sensible Heat loss is directly related with the Latent heat gain. We mentioned this in Chapter 2 when We described the configuration of the evaporative cooling system as bearing close resemblance to a cross flow heat exchanger. The cellulose pad, being the heat exchange medium. Analytically, the sensible heat loss by the air entering the cellulose pad is converted to latent heat gained by the water as it undergoes a phase change.

We compute the latent heat gain using the equation below; the product of the latent heat of vaporization of water, the mass flow rate of dry air and the change in humidity ratio of the air as it passes through the pad. Figure 4.12 shows that as the Sensible heat loss (cooling) increases so does the latent heat gain. Ideally this increase should be in step, but due to experimental errors we see the overall pattern of the relationship with the expected deviation from the ideal.

$$Q_l = \dot{m}_a h_{fg}(W_2 - W_1) \tag{12}$$

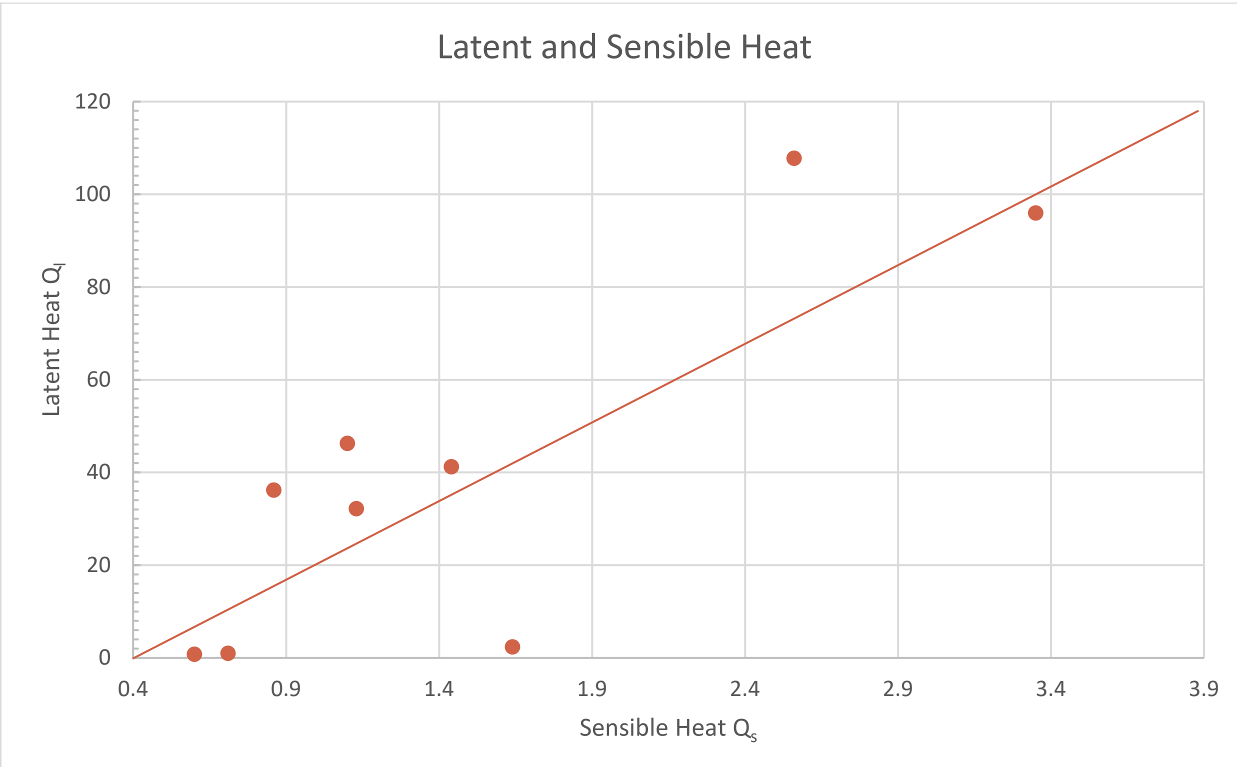

Figure 4. 12 Relationship between latent heat and sensible heat

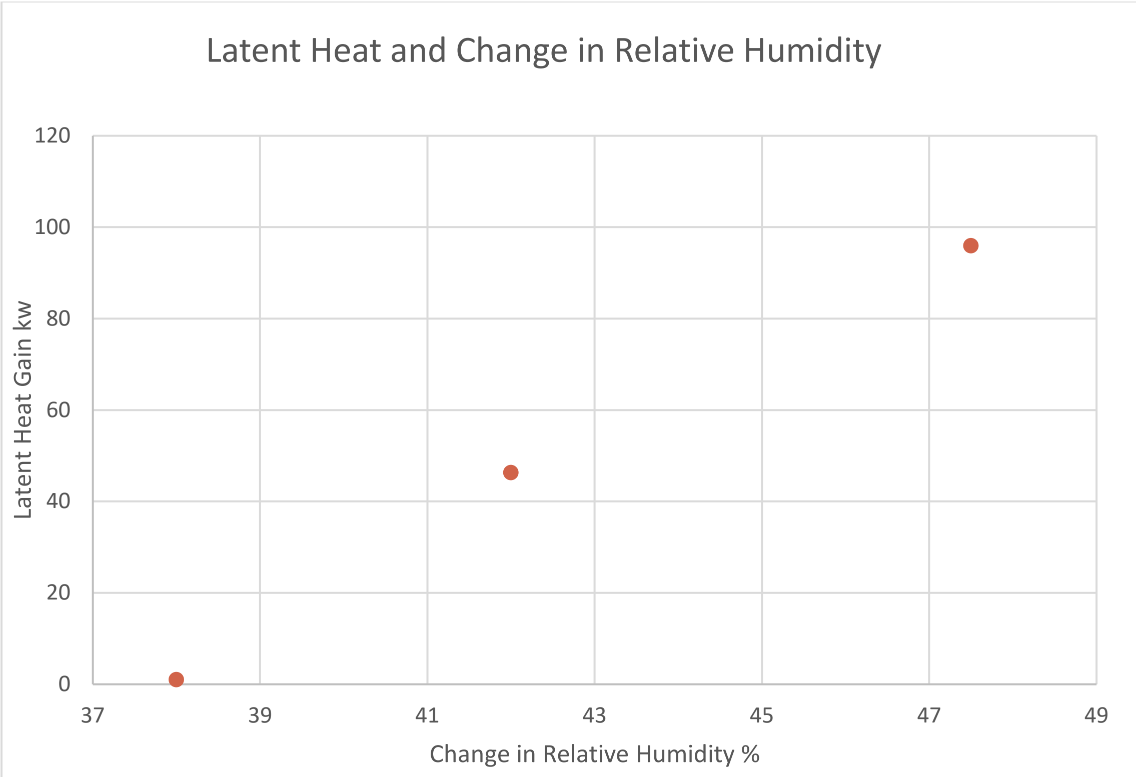

Figure 4. 13 The relationship between relative humidity change and latent heat gain

Now the latent heat gain is a function of the relative humidity of the process air. Figure 4.13 above shows the effect of change in relative humidity on the latent heat gain. Recall that the pad depth is positively correlated with sensible cooling which as in fig above is also positively impacted by latent cooling. The latent load for the thinnest pad, that is the 2-inch pad was found to be below 1kW, the latent load for the 2-inch pad stood at about 46 kW as relative humidity increased from 37% to 69%, an increase of about 42 points. This makes sense because the increase in relative humidity is a necessary consequence of the sensible cooling, which is also positively related to latent heat gain, an exchange in a manner of speaking. Hence the latent heat of 97kW observed at 4-inch pad is due to the significant rise in relative humidity of the air after the pad.

We know that this is an indication of the capacity of the air to hold moisture. And since the inlet condition of the air for all three pads are largely the same, and holding other variables constant, the 4-inch pad, with a greater surface area gave rise to a higher relative humidity for the process air and hence the latent heat value of about 98kW.

$$\dot{m}_{evap} = \dot{m}_a(W_2 - W_1) \tag{13}$$

4.6 Water losses through evaporative cooling

As mentioned earlier evaporative cooling is driven by the heat and mass transfer that takes place on the pad. The mass transfer is essentially the water molecules evaporated on the pad, which is responsible for the sensible cooling observed. However, as the sensible cooling is enhanced, there is a corresponding loss of water through evaporation and one of the goals of this work is to recover the greater portion of water lost through evaporation by a condensate recovery process using the Peltier effect

Typically, water consumption in evaporative cooling takes place via two means: the periodic bleeding of the system to prevent mineral build up thereby impacting on the air quality and the water losses through evaporation. The latter being an integral part of the evaporative cooling process. We highlighted the importance of pad depth in the cooling process in section 4.1 above. The residence time of the air on the pad and the surface area of the pad coupled with the impact of the frontal velocity on sensible cooling were investigated, now we make the vital connection between the sensible heat loss Q_s(Sensible Cooling)by the air and latent heat gain Q_l by the water running over the pad .

Figure 4.14 shows the relationship between the rate of evaporation on the pad and the thickness of the pad. We see that as the pad depth increases the rate of evaporation increases as well, holding the frontal velocity constant and the flow rate of water over the pad constant. The relationship between the water consumption and pad depth is a positive one. This is to be expected since the depth of the pad also positively impacts the Sensible cooling experienced. The mass rate of evaporation is given by equation below.

This highlights the change in humidity ratio of the inlet air as it goes through the pad. The mass rate of evaporation is the product of this difference and the mass flow rate of the air. Consequently, the mass rate is positively impacted by both pad depth and the inlet air velocity. As the inlet air velocity increases the mass rate of evaporation increases as well. Note in figure 4.14. below that at the highest inlet air speed of 2.56m/s and a pad depth of 4 inches the mass rate of evaporation on the pad obtained was about 39.6g/s. contrasted with the mass rate of evaporation of the 2-inch pad at an inlet velocity of 0.86m/s which yielded 0.33g/s. The effect of both frontal inlet velocity and pad depth is positive.

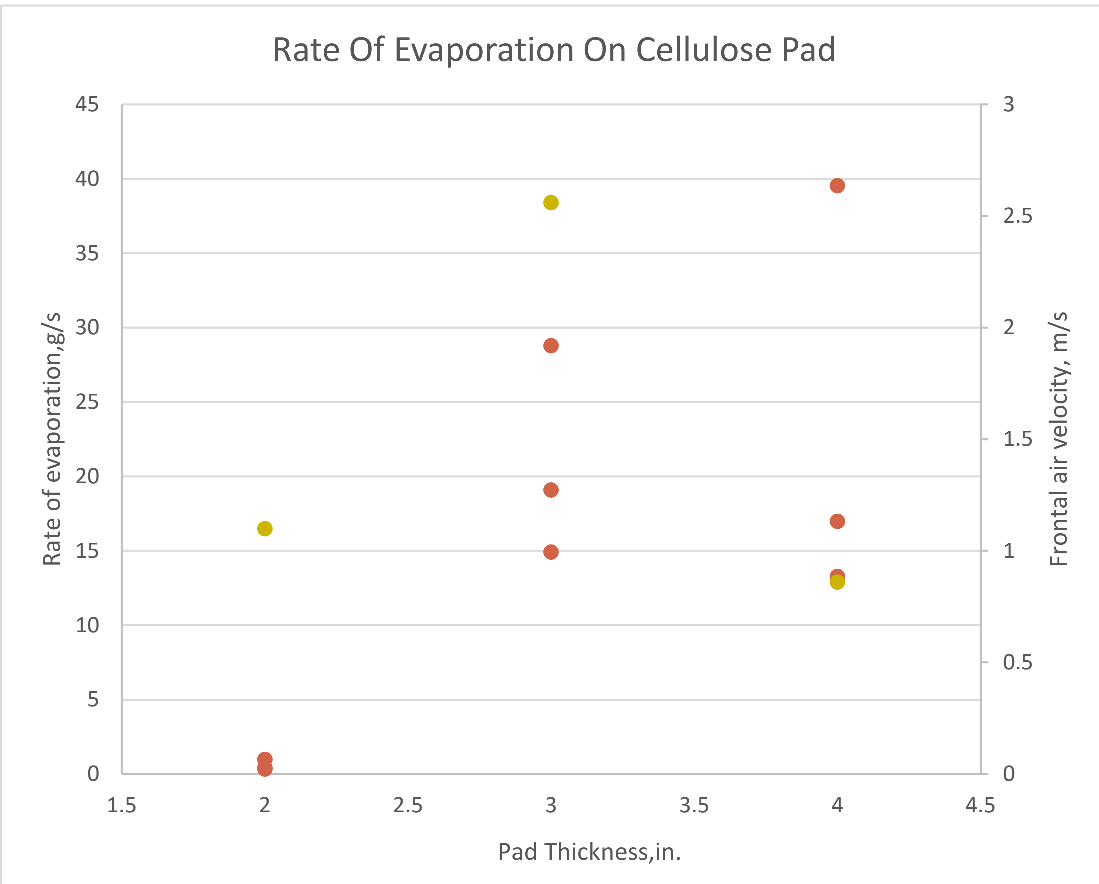

Figure 4. 14 Rate of evaporation due to pad depth(thickness) and frontal velocity

4.7 Rotary Desiccant wheel dehumidification

To manage the humidity of the cooled space, a rotary desiccant wheel was used to make the supply air suitable for our desired purpose. We are interested in the dehumidification capabilities of the silica gel impregnated wheel. Here We have investigated the dehumidification capabilities of silica gel using a static bed dehumidifier, an in house fabricated rotary desiccant wheel with a 1:1 dehumidification and reactivation ratio and both compared with a thermos electric cooler dehumidifying unit. We introduced the concept of dehumidification effectiveness and the adsorption of all three are compared using equation 4.3.0 and the result is shown in figure 4.15 below. This is a measure of the percentage change in relative humidity of the air as it goes through the dehumidifier. And it is the ratio of the change in humidity to the relative humidity of the process air multiplied by a hundred.

This metric is an estimate of how much of the moisture content of the air the dehumidification unit is able to remove. Here we have compared the thermo-electric cooler using the Peltier effect with a fixed bed desiccant dehumidifier and a rotary desiccant wheel dehumidifier. The use of percentages normalizes the comparison process so that each experiment is conducted with a set of comparable but not necessarily identical conditions. The relative humidity of the inlet air and outlet or supply air is measured with the aid of humidity probes accurate to within ±2.5% obtained from Omega Engineering. A set of k-type thermocouples were also attached to track the temperature of the room, the cold plate and the temperature of the inlet and outlet air streams.

$$\epsilon_{ff} = [W_2 - W_3]/W_2 * 100. \tag{14}$$

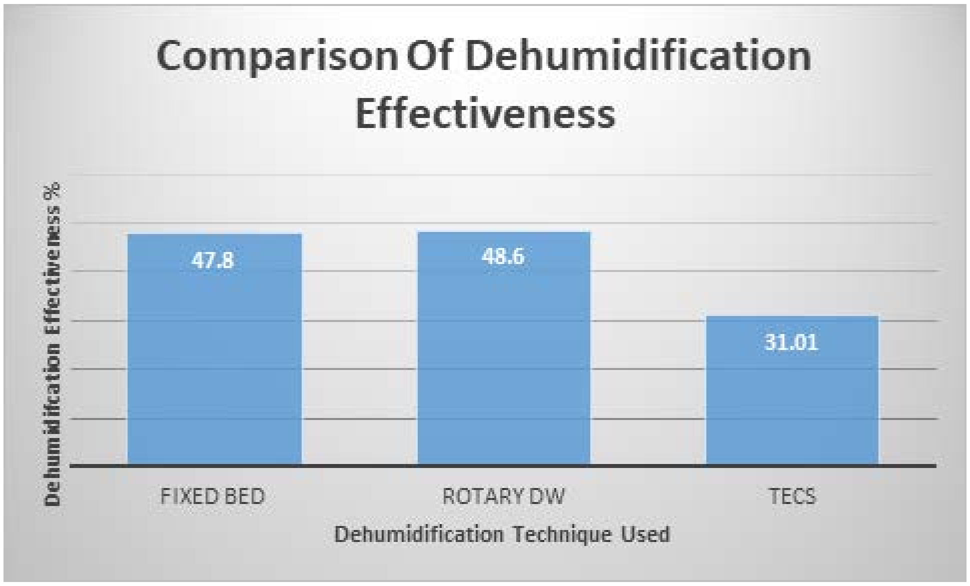

Figure 4. 15 Dehumidification effectiveness

4.8 Effect of process air relative humidity on adsorption

In section 4.5 above we explored the interrelationship between the pad depth and sensible cooling. We also made the connection between the latent heat and sensible heat gain by the water as it changes to vapor. Recall that the need for a dehumidification unit in direct evaporative cooling is consequent upon the relative humidity of the process air. Now the desiccant material employed for this work, Silica gel has been known to hold about forty percent of its weight in moisture. This means that as the relative humidity of the process air increases the amount of moisture to be adsorbed by the desiccant material increases. Holding all other variables constant, the pad depth increases the process air relative humidity and this in turn decides the moisture content of the process air. The higher the process air relative humidity the higher the adsorption rate of the desiccant material subject to reactivation and the adsorption capacity of the desiccant.

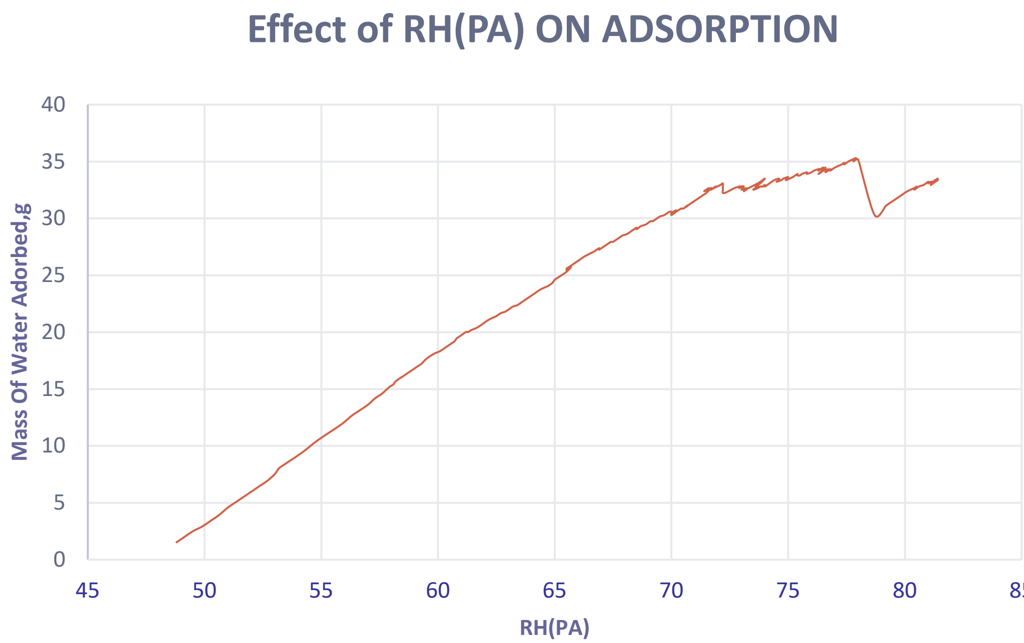

Figure 4. 16 Effect of process air relative humidity (RH) on adsorption

Holding inlet air velocity constant at 2.56m/s it was found that the rate of adsorption in grams per second was highest for the three-inch pad for our set up. One would have expected the adsorption rate to be either the same or highest for the 4-inch pad since the relative humidity of the process air (cooled air) using the 4-inch pad is the highest. Using the 4-inch pad some moisture content of the air does pass through the desiccant wheel but not significant enough to alter the air quality by a significant amount, and since reactivation is a continuous process using the rotary desiccant wheel this shouldn't pose a problem.

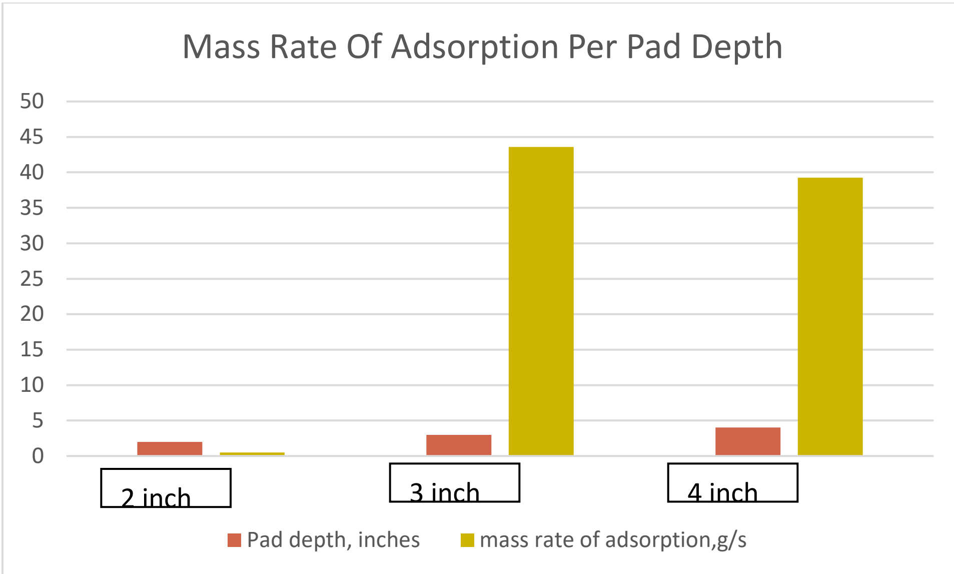

Figure 4. 17 Effect of pad thickness on adsorption rate

4.9 Reactivation of Desiccant Material

Having dehumidified the supply air to the desired relative humidity range between forty-five percent to nearly 80 percent dependent on equipment class, the rotary desiccant material needs to be reactivated. The reactivation temperature used here was 45^{o}C. Since the desiccant wheel dehumidification, reactivation ratio of 1:1 was used. Recall that splitting the wheel into a 1:1 ratio allows for reactivation at lower temperatures. This allowed for a steady reactivation of the silica gel material. The chart below shows the relationship between the adsorption capacity of the desiccant material and the desorption due to the reactivation process. The pattern of the chart shows that the amount of moisture recovered from the desiccant material is a function of the amount of moisture adsorbed by the desiccant in the process of dehumidification. In the same vein, the relationship between the relative humidity values of the process air, dehumidified air and reactivation air stream is show in figure 4.19. Figure 4.18-4.19 clearly shows that the reactivation air stream is impacted by the process airstream. The moisture content of the reactivation air stream is seventy-five percent that of the process air stream while the supply air stream relative humidity hovered around fifty percent that of the process air stream.

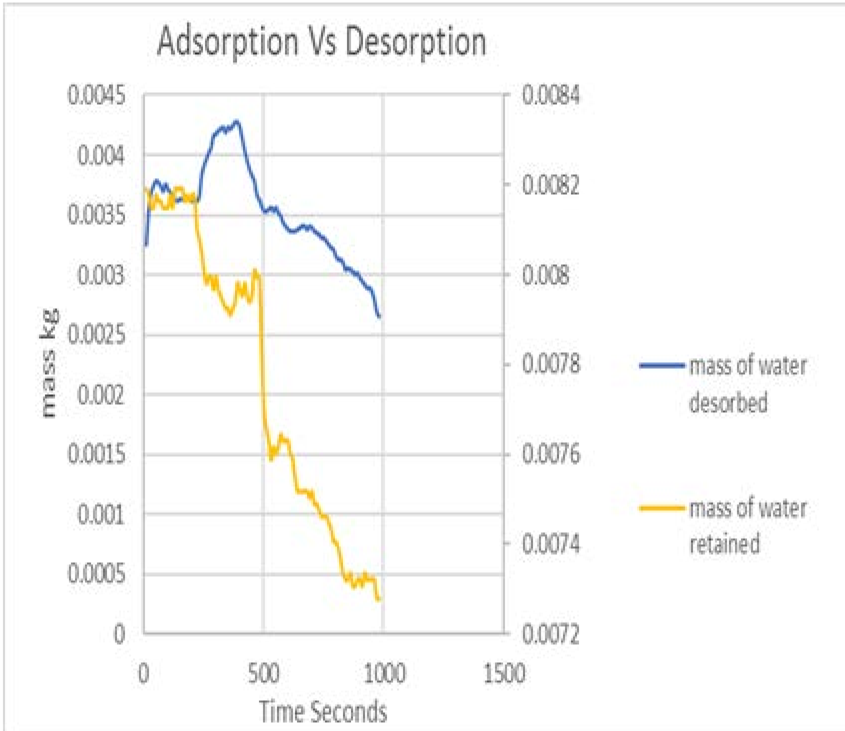

Figure 4. 18 Relationship between adsorption and desorption

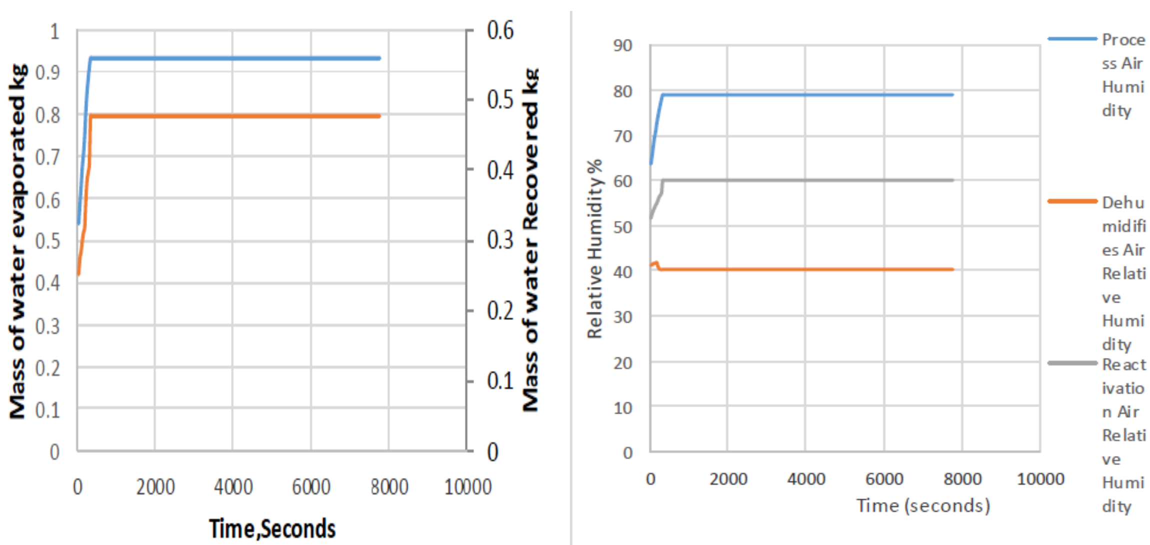

Figure 4. 19 Relative humidity of process air, supply air, and reactivation air

4.10 Condensate Recovery

Now to the pinnacle of this work which is condensate recovery from the exhaust air stream. Now as the desiccant is reactivated, the return air stream is enriched with moisture. Figure 4.19b above shows a relative humidity of over sixty percent. By running this air stream on to a cold plate (a heat sink attached to thermoelectric coolers) the moisture rich exhaust air is cooled below its dew point and condensates begin to form. By taking the mass of the moisture recovery unit at the start of the experiment and at the end of the experiment is taken, holding density constant, the volume of condensate is easily obtained This validates the mathematical model presented in chapter 3 above, and by recovering condensate, we minimize the water consumption of the desiccant assisted evaporative cooling unit.

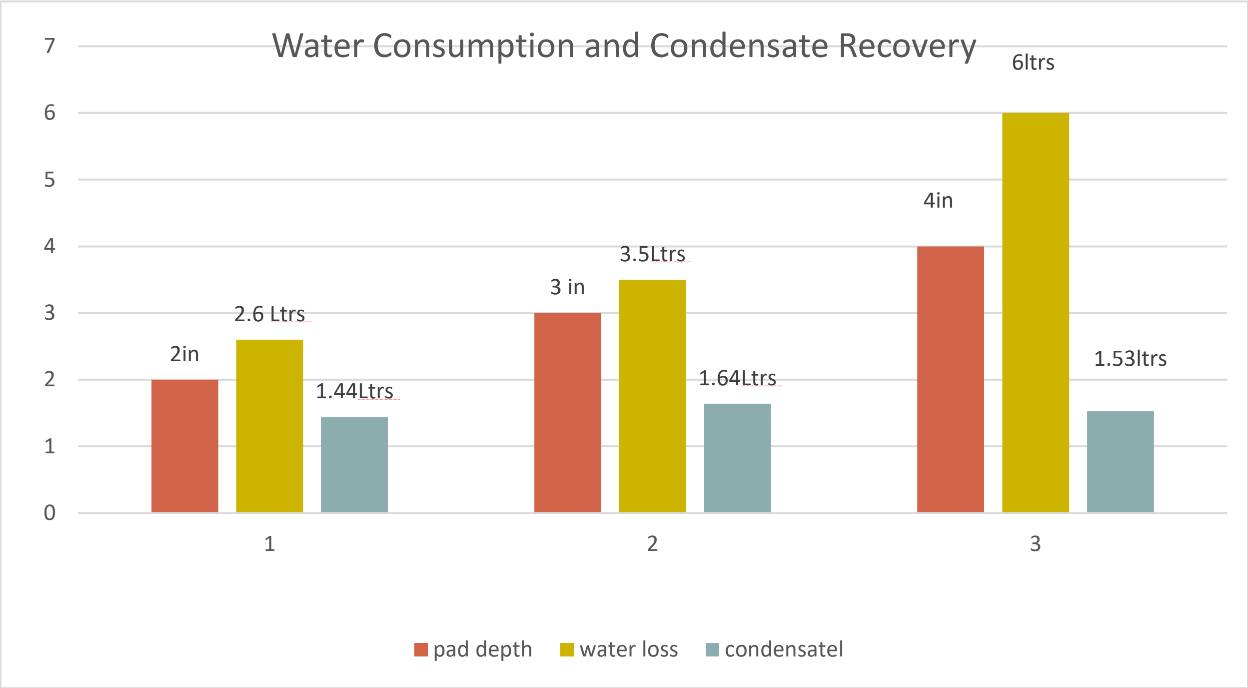

Figure 4. 20 Water consumption and condensate recovery

Now the Condensate recovered is a function of the relative humidity of the exhaust air stream, which in turn depends on the depth of the pad, that is the surface area for the effective heat and mass transfer which takes place on the cellulose pad. As shown in figure 4.20 above we see that on the condensate recovery system recovered over fifty percent for both the 2-inch pad and the 3-inch pads, the 4-inch pad resulted in a 26% moisture recovery. Hence for this lab scaled experimental investigation into condensate recovery from desiccant assisted evaporative cooling technology, the 3-inch pad provided optimal sensible cooling and condensate recovery. Further investigation should be carried out to test for scalability.

5. CONCLUSION

This work has shown that direct evaporative cooling system coupled with desiccant based dehumidification can produce air quality suitable for data centers.

In this work a desiccant assisted direct evaporative cooling system was studied for data centers using pad depths of 2, 3 and 4 inches. The effectiveness of the direct evaporative cooling unit was found to be 46%,67% and 81% accordingly for the aforementioned pad depths.

A rotary desiccant wheel fabricated in-house was used for dehumidification and was found to have dehumidification effectiveness of 49%, using a 1:1 cooling and reactivation ratio. When Compared with a fixed bed desiccant dehumidifier the dehumidification effectiveness was about the same until saturation of the fixed bed. Compared with a TEC, the desiccant based dehumidification system provided superior dehumidification

The desiccant assisted direct evaporative cooling system so contrived was found to produce thermally suitable air quality for data center cooling, according to the dictates of ASHRAE technical committee 9.9. [2, 41, 40] By providing suitable air quality for data centers, the desiccant assisted evaporative cooling system can significantly improve PUE, by reducing cooling cost. The factors affecting the sensible and latent heat transfer of the process were thoroughly investigated, and a thermodynamic analysis presented in chapter 3.

The sensible heat factor was found to be directly related to the pad depth and was constant irrespective of the inlet air velocity. By Cooling the warm and humid return/reactivation air stream below its dew point temperature, a novel condensate recovery method for evaporative cooling units was presented. The system so contrived recovered on average 50% of the moisture lost due to evaporation

It can therefore be used either as a stand alone or in conjunction with Conventional Vapor compression cooling systems. The water consumption of the system was also carefully studied, and it was found that the water consumption of direct evaporative cooling is an integral part of the process. A novel condensate recovery system was also presented, using simple thermoelectric devices to cool the moist exhaust airstream to below its dew point temperature.

The moisture so recovered from this process can be reused for the evaporative cooling process thereby minimizing the water consumption of evaporative cooling. This process should come in very handy in dry and arid regions where evaporative cooling is both highly effective based on the capacity of the air in such regions to hold a higher volume of water per kg of air and the attendant scarcity of water in such locations. The parameters driving moisture consumption were also studied and the power consumption of the process analyzed to show the advantage in terms of energy consumption over conventional CRACS.

Operational efficiency of the cooling systems is perhaps the most significant advantage that a direct evaporative cooling unit as detailed in this work presents for data centers. With fewer parts to maintain thereby needing less maintenance cost as compared to conventional vapor compression mechanical cooling systems which require a highly skilled workforce to maintain and more training cost for maintenance staff or lake or river sourced water cooling techniques which are prone to vandalism and controversial in terms of effect of the temperature of water being returned to the water bodies. Evaporative cooling coupled with desiccant based dehumidification and moisture recover, minimizes water consumption and can results in significant cost savings in terms of lower power cost and lower maintenance and personnel training costs. While this lab-scaled experimental work has shown superior results with the desiccant based direct evaporative cooling technology, the application of the condensate recovery proposed here must be investigated and appropriate sizing of cooling unit investigated before replicating these results.

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