DESICCANT BASED EVAPORATIVE AIR CONDITIONING SYSTEM

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by

SHAH RUKH KHAN

M. HANZALA MUNAWAR

MAWAHID ALI KHOKHAR

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EXAMINATION COMMITTEE

We hereby recommend that the final year project report prepared under our supervision by:

SHAH RUKH KHAN	NUST201432730BSMME11114F
M. HANZALA MUNAWAR	NUST201432755BSMME11114F
MAWAHID ALI KHOKHAR	NUST201433164BSMME11114F

Titled: "**DESICCANT BASED EVAPORATIVE AIR CONDITIONING SYSTEM**" be accepted in partial fulfillment of the requirements for the award of BE Mechanical Eng. Degree with grade ____.

Supervisor: Mr. Hafiz M. Abdul Rehman, Lecturer SMME, NUST	______________________ Dated:
Committee Member: Dr. Zaib Ali, Assistant Professor, SMME, NUST	______________________ Dated:
Committee Member: Dr. Emad Uddin, HOD, SMME, NUST	______________________ Dated:

______________________ ____________

(Head of Department) (Date)

COUNTERSIGNED

Dated: ________________ __________________________

(Dean / Principal)

ABSTRACT

Conventional Vapor Compression system fails to provide comfort conditions in humid areas where it is important to control the latent load of the environment as it deals with the sensible load only. Moreover, the refrigerants used in the conventional system not only contribute in depleting the ozone layer in stratosphere but also directly contributing in the global warming when released in the atmosphere. Desiccant Based Evaporative Cooling system has the potential to be a very reasonable alternative to conventional system. It has the potential to deal with the sensible and latent load both but is also environmental friendly as it involved the use of only desiccant material and not any sort of refrigerant. Air can be conditioned in a cleaner and affordable method using a Desiccant Based Evaporative Cooling system. The objective of this work is to design and develop a solid desiccant based evaporative cooler to control both sensible and latent load in hot and humid environment. The system was developed and evaluated for the climate of Karachi, Pakistan and solved on TRNSYS® software. The results evaluated on the TRNSYS software have been compared with the literature review and with the results obtained from the simulator designed by NOVEL Aire. It is found that lower temperature of the conditioned space is achieved on recirculation mode of the cycle as compared to the ventilation mode. The system consists of solid Desiccant Based dehumidifier for the extraction of the moisture from the air, a heat recovery wheel to lower the temperature after desiccant wheel, a heater to heat the regeneration air and evaporative coolers.

PREFACE

We are extremely delighted to present you our report on successful experimental testing of desiccant based evaporative cooling in Pakistan Climatic conditions.

Desiccant based cooling has proved to be cost effective and environment friendly alternative to the conventional Vapor Compression air conditioner systems for space cooling applications. Though many research papers contain sufficient material and results of their desiccant systems, but no attempt has been made to obtain experimental results for Pakistan with desiccant based evaporative cooling system. This report demonstrates the output results of using desiccant cooling to obtain human comfort conditions. The material presented in this report before you is with the help of many research papers.

We would first of all thanks Almighty Allah, Who blessed us with the energy and strength to complete all the tasks successfully.

This tiresome work surely was not possible without the continuous guidance, encouragement and technical insight of Sir Hafiz M. Abd-ur-Rehman throughout the project. We are also extremely grateful to Dr. Safdar and Dr. Emad Uddin for their help in mathematical modelling and ANSYS. Dr. Adeel Waqas provided us with the tool to solve transient simulation (TRNSYS) and the desiccant wheel to perform experiment and we are extremely grateful to him.

Constant prayers, support, both emotional and practical, of our parents can't be ignored here.

We have tried our level best not to omit any topic so that you may find it easy in comprehending our system.

In case of any discrepancy or suggestion, do give us your feedback.

Mawahid Ali Khokhar	**Muhammad Hanzala Munawar**	**Shah Rukh Khan**
14bememkhokhar@smme.edu.pk	14bememmunawar@smme.edu.pk	14bemesrkhan@smme.edu.pk

ORIGINALITY REPORT

We hereby declare that no portion of the work of this project or report is a work of plagiarism and the workings and findings have been originally produced. The project has been done under the supervision and guidance of Sir Hafiz M. Abdul Rehman and has not been a support project of any similar work serving towards a similar degree's requirement from any institute. Any reference used in the project has been clearly cited and we take sheer responsibility if found otherwise.

Final Report

ORIGINALITY REPORT

15%	5%	12%	3%
SIMILARITY INDEX	INTERNET SOURCES	PUBLICATIONS	STUDENT PAPERS

PRIMARY SOURCES

1	M. Mujahid Rafique, P. Gandhidasan, Shafiqur Rehman, Luai M. Al-Hadhrami. "A review on desiccant based evaporative cooling systems", Renewable and Sustainable Energy Reviews, 2015 Publication	3%
2	D.B. Jani, Manish Mishra, P.K. Sahoo. "Solid desiccant air conditioning – A state of the art review", Renewable and Sustainable Energy Reviews, 2016 Publication	2%
3	Submitted to Heriot-Watt University Student Paper	1%
4	"Desiccant Heating, Ventilating, and Air-Conditioning Systems", Springer Nature, 2017 Publication	1%
5	Ge, T.S.. "A review of the mathematical models for predicting rotary desiccant wheel", Renewable and Sustainable Energy Reviews, 200808 Publication	1%

(Supervisor Signature)

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ABBREVIATIONS

CFC	Chlorofluorocarbon
VAC	Conventional Vapor Compression Air Conditioners
R/P	Split ratio
COP	Coefficient of Performance
RO	Reverse Osmosis
VCS	Vapor Compression System
RPH	Revolution per hour
VCR	Vapor Compression Refrigeration
FDM	Finite Difference Method
HAP	Hourly analysis Program
CMH	Cubic meter hour
NTU	Number of transfer unit
TRNSYS	Transient System Simulation Tool
FBD	Free Body Diagram
FS	Factor of Safety
FEM	Finite Element Method
DW	Desiccant Wheel
HRW	Heat Recovery Wheel
EC	Evaporative Cooler
CAD	Computer Aided Design

NOMENCLATURE

A	Cross sectional Area of the Airflow channel, desiccant or substrate layer	m^2
A_s	Surface area	m^2
BHP	Brake horse power	hp
C_p	Specific Heat	J/kg-K
C_h	Hot fluid heat capacity	J/kg
C_c	Cold fluid heat capacity	J/kg
C	Heat capacity ratio	
H	Convective heat transfer Coefficient	W/m^2-K
ΔH	Heat of adsorption or vaporization	J/kg
P	Perimeter length of the airflow channel	m
D	Diameter	m
d_e	Hydraulic diameter	m
D_e	Desiccant effective diffusivity	m^2/s
K	Thermal Conductivity	W/m.K
K_y	Gas-side mass transfer coefficient	$Kg/m^2 \cdot s$
M	Mass	kg
P	Power	W
Q	Heat transfer	J/s
q_{st}	Heat of sorption	J/kg
T	Temperature	K
T	Time	s
U	Overall Heat transfer coefficient	W/m^2-K
u	Air velocity	m/s
W	Desiccant adsorption mass	kg/kg
x	Wheel Thickness	m

Y	Humidity ratio	kg/kg
Y_a	Absolute humidity ratio of the air stream	kg/kg
Y_d	Humidity ratio in equilibrium with the desiccant	kg/kg
Y_g	Absolute humidity ratio of the air stream	kg/kg
Y_m	Humidity ratio in equilibrium with the matrix	kg/kg
Z	axial displace through matrix measured from period entrance	m
Greek Letters	**Full form**	
Ε	Effectiveness	
Ρ	Density	kg/m^3
Δ	the thickness of desiccant felt	M
Φ	Relative Humidity	kg/kg
ω	Wheel rotate speed	rad/s
Subscript	**Full form**	
1,2 …	State	
A	Dry air	
DW	Desiccant wheel	
D	Desiccant	
Db	Dry bulb	
ECpro	Evaporative Cooler process air	
ECreg	Evaporative Cooler regeneration air	
HRW	Heat recovery wheel	
M	Desiccant matrix or mass transfer	
V	Water Vapor	
w	water	
Wb	Wet bulb	

CHAPTER 1: INTRODUCTION

The amount of energy required to cool and bring a space at comfort level has been increasing every day. In building, most of the primary energy are utilized for the purpose of cooling and heating. The widely used method to condition air is the vapor compression refrigeration-based method. Providing the comfort conditions to the people not only means to control the sensible load capacity (Temperature Control) but also the latent load capacity (Humidity Control).

Sensible heat ratio is a factor that can be determined by taking the ratio of sensible load with the sum of sensible and latent load.

$$Sensible\ Heat\ Ratio = \frac{\text{Sensible Load}}{\text{Sensible + Latent Load}} \tag{1}$$

In order to control latent load, the conventional vapor compression system uses the process of condensation to condense the water vapors on the coils when the air is cooled below its dew point temperature and then reheated again up to the required supply conditions. Normally the vapor compression cycle used are working on 0.75 sensible heat ratio means 75% of the capacity is used in controlling the sensible loading while remaining 25% of the capacity is used in controlling the latent loading. So, it will provide the comfort conditions when the sensible heat ratio is greater than 0.75 [1].

Now it has certain drawbacks:

- The sensible heat ratio in hot and humid areas is found to be less than 0.75. In order to conditioned air in such areas the conventional vapor compression system can be used but it will require a large amount of electrical energy
- The excess use of Vapor Compression system has affected the environment in a harmful way. The ozone layer has depleted because of the CFCs used in a VAC system.
- Increasing electricity production in power stations due to high demand has resulted in many environmental issues, including global warming.

So, an alternative to the VAC systems is required which could make use of the renewable energies in order to minimize the emission of CO_2 and CFCs in to the environment. Evaporative cooling, one of the oldest method, is one technique to meet cooling demand of the building by utilizing evaporative cooing effect, with less power requirements, about one fourth to that of the VAC systems. Evaporative cooling is a simple, cost effective, environment friendly and energy saving technique for space cooling.

The evaporative coolers are best fit for temperature control when air humidity is low, like in hot and dry climate. While for both temperature and humidity control like in hot and humid climate, the effectiveness of the cooler drops remarkably. Therefore, it is used along with some other dehumidification system. One way to achieve comfort conditions for hot and humid climate is through Indirect evaporative coolers. However, the efficiency of indirect evaporative cooler is only around 60-70%.

A desiccant dehumidifier, whose purpose is to removes the moisture from the process air, can be used in conjunction with a VAC system, to allow the cooling system to function effectively. Such a system is called, **Hybrid Desiccant Cooling System**. The use of hybrid desiccant cooling system can not only control the latent load but also reduce running costs and power consumption.

But in order to meet the cooling requirements and be completely environment friendly, evaporative cooler used along with the desiccant dehumidifier is best alternative. Combined system is known as **Desiccant based Evaporative Cooling System**.

Table 1-A comparison among different Cooling Technique

Parameter	Mechanical Vapor Compression	Evaporative Cooling	Desiccant based evaporative cooling
Cost of operation	High	Low	Low
Input Energy Resource	Electricity, Natural Gas, Vapor	Low Grade Energy	Low grade energy e.g. solar energy, waste heat etc.
Latent load control	Average	Low	Accurate
Sensible load control	Accurate	Accurate	Accurate
Quality of indoor air	Average	Good	Very Good
System Installment	Average	Average	Slightly Complicated
Emission of greenhouse gases	High	Low	Low
Market potential	Dominate Air-Conditioning Market	Limited Application	Immature Technology with limited application
Cooling Medium	Refrigerants	Water	Water

CHAPTER 2: LITERATURE REVIEW

2.1. DESICCANT BASED EVAPORATIVE COOLING SYSTEM

The new technology of desiccant based evaporative cooling consists of two components i.e. A desiccant for the dehumidification and an evaporative cooler (Direct or Indirect). It significantly reduces the consumption of the electrical energy compared to the conventional vapor compression cycle [3]. It provides more economical, cleaner and accessible air conditioning.

The working mechanism of the system is such that when the hot and humid air enters in the system (ventilated or recirculated), its moisture is extracted by the desiccant. When the moisture is absorbed from this process air, its temperature further rises. The temperature is then lowered by using heat exchangers. For a continuous system, the moisture absorbed by the desiccant should be extracted (regenerated) out from it so that it is able to absorb more moisture from the process air. There is certain temperature needed to regenerate this desiccant that can be done by electric heater or waste heat (e.g. solar). Many researchers used solar energy to regenerate the desiccant [4-6]. Regeneration can be done effectively using solar energy.

Different desiccant material has different tendency to absorb the moisture [7]. There are two different types of desiccant i.e. solid and liquid. Solid desiccant that are commonly used are Silica gel and natural zeolite. While liquid desiccants are Lithium Bromide, Lithium Chloride and Activated Ammonia. There are certain advantages and disadvantages of using solid or liquid desiccant.

Table 2 - A comparison between solid and liquid desiccant system

Properties	Solid Desiccant	Liquid Desiccant
Regeneration Temperature	High	Low
Carry Over	Low	High
Compactness	High	Low
Capacity to hold moisture	Low	High
Cost	Low	High

2.2. CYCLE

The desiccant based evaporative cooling system consists of three main cycles as defined by Muzaffar Ali [9]:

2.2.1. Ventilation Mode

The process starts with dehumidification of the air (1-2), the moisture is extracted from the air, so the temperature of the system rises. Next the warm air passes through heat recovery wheel, where the temperature of the air drops (2-3). Then this air passes through the supply humidifier

(3-4) which can be direct or indirect evaporative cooler and supplied to the conditioned space. In the regeneration phase, the air from the conditioned space passes through humidifier (5-6) and then heated by the Heat Recovery wheel (6-7), then air passes through electric or solar assisted heater (7-8) to elevate its temperature to the point where it is able to regenerate (8-9) the desiccant material as shown in Fig. (1).

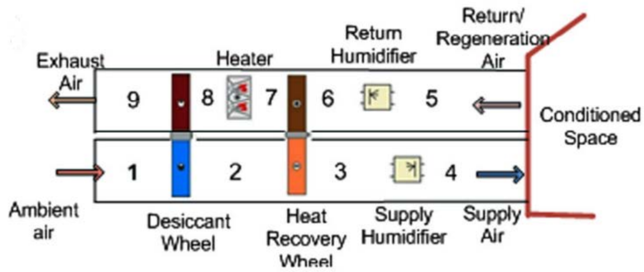

Figure 1 - Desiccant Based Evaporating Cooling System running on Ventilation Cycle

2.2.2. Recirculation Mode

The recirculation mode is also the variant of the standard ventilation configuration. In this system, return air is reused as the process air while the regeneration of the desiccant material is done by the ambient air in the outdoor conditions as shown in Fig. (2).

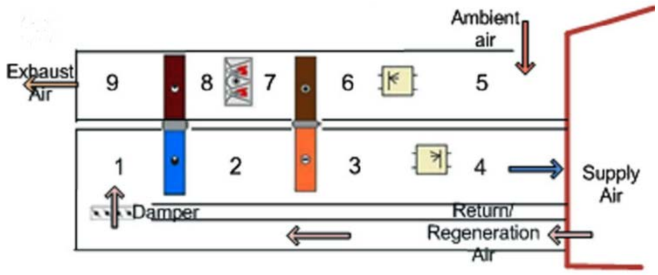

Figure 2 - Desiccant Based Evaporative Cooling System running on Recirculation Cycle

2.2.3. Dunkle Mode

This system takes the advantages of both the ventilation mode as well as recirculation mode. An additional heat recovery wheel is used to take advantage the lower temperature of the return air and provided to the supply air at the end as shown in Fig. (3). One of the major drawback of the system is that, it cut-off the supply of the fresh air in the conditioned space.

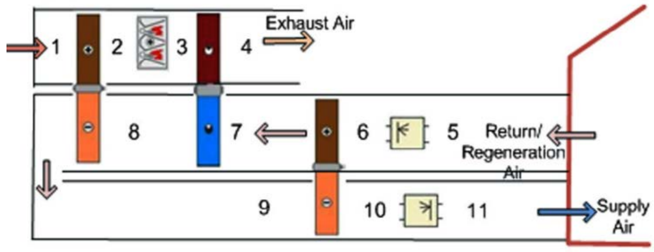

Figure 3- Desiccant Based Evaporative Cooling System running on DUNKLE Cycle

2.3.SOLID DESICCANT

Solid desiccant system mainly comprises of four different components which are desiccant wheel, rotary heat exchanger, cooling unit and regeneration source. Some of the studies are as follows:

Dhanes and Williams [10] modeled numerically the open cycle ventilation mode solid desiccant system and determine the effect of different parameters on the temperature and humidity of the conditioned space e.g. desiccant channel length, dehumidifier mass fraction etc. on the constant temperature of the desiccant.

Low grade heat energy for the regeneration purpose of the desiccant has been studied by Pesaran et al [11]. It reduces the cost for the working of the system and justify the economic feasibility of the system.

The drawback of large system is countered by Shelpuk [12] demonstrated hybrid dehumidifier systems having advanced solid desiccant and compact components to replace large and expensive solid desiccant cooling equipment available. The cost of desiccant system can be further reduced if it is widely accepted in the market.

Different cycles of desiccant based evaporative cooling is also studied by Jain et al. [13] made comparison of different cycles of the desiccant based evaporative cooling in hot and humid climate like in middle east and Psychometric evaluations are carried out for the different cycles in exact room conditions. It is evaluated that the Dunkle Cycle gives the best results in various outdoor conditions.

Parametric study on influence of the temperature and humidity of the different tropical climates on the desiccant based evaporative cooling is carried out by Camargo et al. [14]. The result showed that for evaporative cooling, lesser reactivation temperature and lesser R/P (reactivation air flow/ process air flow) leads to the best operation point.

Experimental investigation carried out on open cycle ventilation mode by Carpinlioglu et al. [15].

Operational parameters of the system e.g. regeneration temperature, rotational speeds of desiccant and rotary heat exchanger and mass flow rate through the system etc. has been determined.

Evaporative cooling effectiveness is not very significant in high humidity conditions (Daou et al. [16]). Desiccant cooling when combined with the evaporative cooling system can have the operating conditions in the high diversity of the climatic conditions. It has feasibility in different climatic conditions proven in terms of energy and cost savings.

Ge et al. [17] studied the two-stage rotary dehumidifier desiccant wheel which helps to lower the temperature of the regeneration of the desiccant material which enables to use low grade energy e.g. solar energy to regenerate the material.

Desiccant air conditioning system is theoretically modeled by Panaras et al. [18]. Different operating parameters e.g. weather conditions, air flow rate, regeneration temperature and cooling load effect on performance of overall system is determined.

For hot and humid climate, potential of Desiccant based evaporative cooling system was studied by Parmar and Hindoliya [19]. The COP of the system is calculated at different environmental conditions and concluded that the performance of the system is highly dependent on the humidity of the area. Higher the humidity, lower the COP of the system.

Heidarinejad and Pasdarshahri [20] developed the transient mathematical model of the system with heat and mass transfer system under various configuration of the system i.e. ventilation, recirculation, make up and mixed mode.

Mittal and Khan [21] evaluated the energy consumption of the desiccant system and compare it with the conventional system. They used the desiccant bed of silica gel. Compared to the conventional system, the electric savings is more than 19% in desiccant system.

Panaras et al. [22] experimentally observed the operational parameters and its effects on the performance of the system. For the same regeneration temperature, increase in the airflow enhance the thermal requirement of the system and thus decrease its COP. So, it is suggested to keep the flow rate minimum. Thus, minimum air flow rate should be selected. In case of increase in regeneration temperature, for ventilation cycle above than 80℃ regeneration temperature affects COP and for recirculation cycle optimum regeneration temperature is 60℃ while above it affects its performance.

The effect of ambient conditions on the Coefficient of Performance of the system in different ventilation and makeup mix configuration of the system by Heidarinejad and Pasdarshahri [23]. It is observed that the energy required in regeneration in recirculation is lower than the ventilation mode thus increasing the Coefficient of Performance (COP) of the recirculation configuration.

Optimization of the desiccant cooling system design with the conjunction of indirect evaporative cooler has been done by Goldsworthy and White [24]. The primary stream face area fraction S in commercial wheels for the desiccant is usually between 0.5 to 0.75.

Temperature of supply air and regeneration temperature increases when S is increased more than 0.75. A large increase in S occurs between T_r= 60℃ and T_r =70℃ over a range of T_{cool}.

Panaras et al. [25] use the control strategy and introduced thermostat and humidistat to ‘on/off’ the indirect evaporative cooler and humidifier. This system helps in conserving the electrical energy through these on/off conditions and studies the effect of each component with and without them.

Nobrega and Brum [26] develops the graphical methodology for designing different cooling cycles.

Chung and Lee [27] investigated the effect of 11 operational parameters on two different configurations of the system and it is concluded regeneration temperature is found to be the most dominant parameter in cost saving of the system while performance of system, maximum humidity ratio of dry air water, density of desiccant, mass fraction of desiccant in the wheel affect is negligible in comparison.

Effect of components effectiveness on the system is investigated numerically by Sphaier and Nobrega [28]. It is concluded that in ventilation mode, overall system performance of the system decreases by 40-50% with the decrease of 20-30% decrease in dehumidifier performance.

To predict the capacity of the desiccant wheel made of silica gel and the condition of the process air after the dehumidification, a neural network has been developed by Koronaki et al [29] for different climatic conditions.

Overall performance of the system and its components is investigated experimentally by Uckan et al. [30] in hot and humid climate. It is determined experimentally that ambient air at 31ºC can be cooled down to 19ºC and continuous supply of 25ºC can be maintained to the conditioned space.

Riffat and Zhu [31] concluded that the better performance is achieved with reasonable indoor air velocity which is 0.6 m. s^{-1}, and also by increasing thermal conductivity of heat pipe condenser and the container surface. The dimensions of the passage of the airflow and process to intake ratio as well as the velocity of air are main parameters.

Davis RA [32] concluded that for the same cooling load, evaporative cooler requires greater mass flow rate as compared to the conventional air conditioning system because of small enthalpy difference. Also, the humidity of the supply air in evaporative cooler is also much higher than conventional Air conditioners.

Gandhidasan P [33] concluded that evaporative coolers are generally used when Wet Bulb Temperature doesn’t exceed 25ºC frequently

Archibald J [34] concluded that evaporative cooling has high coefficient of performance when it works in dry climatic conditions.

Dezfouli et al. [35] compared solar assisted desiccant based evaporative cooling under two different configurations i.e. Ventilation and Recirculation Mode and concluded COP of 0.8 for ventilation and 1.6 under ventilation mode in Malaysia.

Katejanekarn and Kumar [36] used heat exchanger instead of direct evaporative cooler to lower the temperature of the warm air coming out of the desiccant dehumidifier without adding moisture to it.

Nelson et al. [37] studied two different models working on ventilation and recirculation configuration. They concluded that 95% of the energy use in regeneration can be obtained from solar energy using the collectors of area 45 m^2.

Pescod [38] used the cross flow indirect evaporative cooler in the desiccant system with water flowing in wet channel and air flowing in the secondary channel. All the key parameters i.e. spacing and dimensions of channels, velocity and flowrates of air and water as well protrusion details are used.

Suryawanshi et al. [39] concluded that in hot and dry climate, two stage evaporative cooler is 4.5 times more efficient than conventional air conditioning systems.

Kim and Jeong [40] utilize both direct evaporative cooler and indirect evaporative cooler in the system and concluded that this system saves up to 74-77% of the total energy used in operating the conventional air conditioning system.

Worek et al [41] concluded that by using the desiccant of 1M type, it can be regenerated at 165^{o}C and with staged regeneration fraction of 16%, system working under ventilation configuration operates at higher COP.

Table 3 - Summary of studies related to Desiccant Evaporative Cooling System

Author	Summary
Dhanes and Williams [10]	Effect of different parameters on temperature and humidity of conditioned space.
Pesaran et al. [11]	Low grade heat application for regeneration
Shelpuk [12]	Reduced cost of desiccant system
Jain et al [13]	Dunkle cycle is best in various climates
Camargo et al [14]	In evaporative cooling, lesser regeneration temperature and lesser R/P leads to best operation point
Carpinlioglu et al. [15]	Rotational Speed of wheel, mass fraction of desiccant and regeneration temperature determined
Daou et al. [16]	Feasibility of desiccant system in different climatic conditions.

Ge et al. [17]	Two stage rotary dehumidifier lowers the regeneration temperature
Panaras et al. [18]	Operation parameters effect on performance of the system (Theoretical)
Parmar and Hindoliya [19]	Higher the humid climate, lower the COP of system
Heidarinejad and Pasdarshahri [20]	Different cycles transient mathematical model with heat and mass transfer
Mittal and Khan [21]	Electrical savings is more than 19% in desiccant system compared to Conventional System.
Panaras et al. [22]	Increase of airflow at constant regeneration temperature, increase thermal consumption thus decrease COP (Experimental).
Heidarinejad and Pasdarshahri [23]	Recirculation mode has higher COP than Ventilation Mode.
Goldsworthy and White [24]	Optimize desiccant wheel with Indirect Evaporative Cooler and manipulating the area of the wheel.
Panaras et al. [25]	Control strategy to save electrical energy
Nobrega and Brum [26]	Graphical methodology for designing cycles
Chung and Lee [27]	Regeneration temperature is most dominant parameter regarding cost saving
Sphaier and Nobrega [28]	20–30% decrease in dehumidifier performance can lead to 40-50% reduction in the overall cycle performance
Koronaki et al. [29]	Analyze desiccant wheel capacity and outlet conditions after dehumidification
Uckan et al. [30]	Experimentally obtained 25℃ air in room from ambient air using desiccant system.
Riffat and Zhu [31]	Optimize indoor air velocity to be 0.6 m. s^{-1} and increased thermal conductivity of condenser and container to achieve better performance
Davis RA [32]	Evaporative cooler requires greater mass flow rate than conventional system under same load.

Gandhidasan P [33]	Evaporative cooler used under 25°C wet bulb temperature.
Archibald J [34]	Evaporative cooler has high COP in dryer climate
Dezfouli et al. [35]	Ventilation and Recirculation have COP of 0.8 and 1.6 respectively.
Katejanekarn and Kumar [36]	Heat exchanger to cool the air without adding humidity (moisture) to it.
Nelson et al. [37]	95% energy used in regeneration can be obtained through sun by collector area 45 m^2
Pescod [38]	Design the system in combination of Indirect Evaporative cooler (IEC) and determine the key components
Suryawanshi et al. [39]	Two stage evaporative cooler is 4.5 times more efficient than conventional system.
Kim and Jeong [40]	74-77 % energy saved with the combination of both direct and indirect evaporative cooler.
Worek et al [41]	By using the desiccant of 1M type, the regeneration temperature achieved to 165°C and with staged regeneration fraction of 16%, with high COP ventilation configuration.

2.4. LIQUID DESICCANT

There are six major components that constitute the system. These six components are dehumidifier, a regeneration heat source, a solution-to-solution heat exchanger, air-to-air heat exchanger. Dehumidifier in the system consists of an adiabatically working packed tower. From the absorber pool, this solution is pumped to the bottom of tower and into the plate type heat exchanger where cooling process takes place by cooling tower [42]

Some of the studies done by researchers on liquid desiccant cooling are mentioned below:

Oliveira et al. [43] proposed the cooling system working on the liquid desiccant named Lithium Bromide with conjunction of direct evaporative cooler and without recirculation of air. They concluded that this system has lower initial cost than conventional air conditioning system.

Kessling et al. [44] experimentally evaluated that liquid desiccant Lithium Chloride can be regenerated at less than 80°C working with indirect evaporative cooler.

Al-Sulaiman et al. [45] used the multistage evaporative cooling with liquid desiccant working in between the stages proposed two different procedures for the regeneration which is thermal (line heater) and Mechanical (RO process) and concluded that thermal saves more than 25% more energy than mechanical process.

Radhwan et al. [46] mathematically modeled solar assisted liquid desiccant system and observed the working of the system in the climate of Jeddah, Saudi Arabia.

2.5. HYBRID DESICCANT COOLING SYSTEM

Some of the studies done on hybrid desiccant system are:

Dhar and Singh [47] studied the different cycles that could be used for different type of weather conditions (hot humid and hot dry). It was concluded that more energy is actually saved as compare to conventional systems. Different factors that could affect the overall performance like mixing ratio, regeneration temperature or heat factor has also been studied.

Different air conditioning systems have been studied by Henning et al. [48] for different climatic zones. A 50% reduction in energy was observed by using a hybrid system. Operational costs also got reduced.

A comparative study was carried out by Dai et al. [49] of different VCS. The VCS under study were standalone, one associated with desiccant and evaporative cooling, and one associated with desiccant only. The COP was improved by 20-30% while the cold production improved by 38.8 to 76%.

Subramanyam et al. [50] studied the vapor compression system coupled with a desiccant wheel experimentally. The system performance was evaluated on the basis of different parameters. It was concluded that the best performance for high humid region was at 17.5 RPH.

Jia et al. [51] tried to minimize the use of electrical power by a solid desiccant cooling hybrid system by studying different operating parameter. It was observed that the system efficiency is increased by reducing the high-grade regeneration energy.

Liu et al. [52] combined a rotary desiccant wheel with a conventional refrigeration system to design a Dedicated Outdoor Air System. Different factors were studied to understand their significance on the COP of the system.

In order to increase the overall performance of dehumidification in an air conditioning system, Sayegh et al. [53] studied and compared two different methods. One is the heat exchanger cycle while the other one is a hybrid system that contains a desiccant wheel coupled with a VCR cooling coil. It was observed that hybrid system performs better than the heat exchanger cycle in areas where the relative humidity is low.

Hybrid systems of two different types were studies by Hong et al. [54]. Both of the system under study were desiccant based cooling system, one was using an evaporative cooler while the other a conventional vapor compression system as humidifier. Both systems physical, mathematical and numerical models have also been studied. The hybrid desiccant system was consuming less energy that the desiccant based vapor compression system when both of the systems were operated under the same conditions.

CHAPTER 3: METHODOLOGY

3.1. PROPOSED PROJECT

After the detailed literature review and study of different mode of desiccant systems, we came to the decision of prototyping a **solid** desiccant evaporative cooling system based on **recirculation** cycle for our final year project. The decision was based on solid desiccant advantages of easy handling and cost effectiveness; recirculation cycle performance in comparison to that of ventilation cycle and complexity of Dunkle cycle. We will use a Desiccant Wheel, followed by a heat exchanger and direct Evaporative Cooler.

3.1.1. Schematic

The complete schematic of our system is shown below:

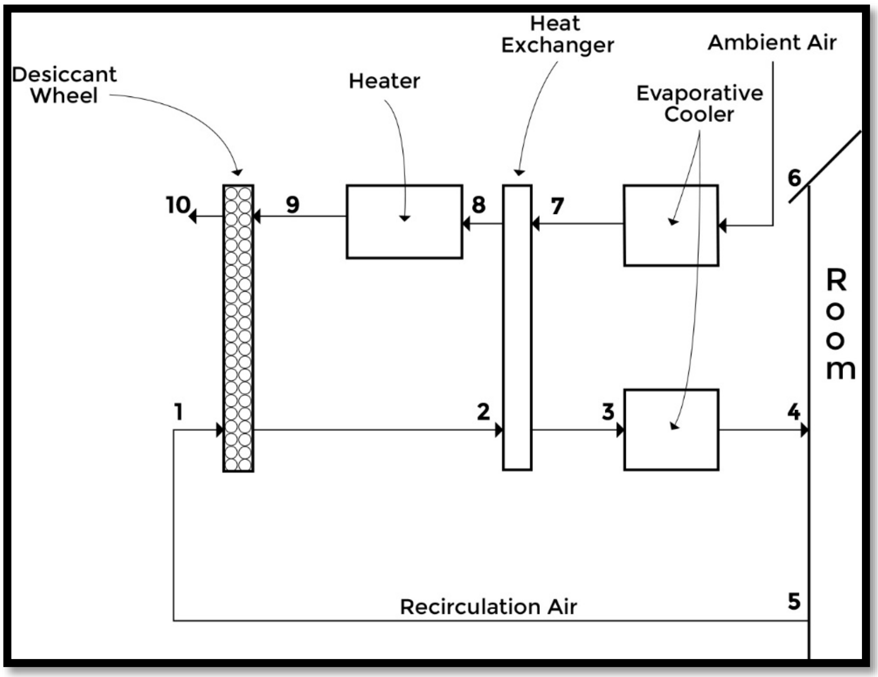

Figure 4 - Schematic of Desiccant Based Evaporative Cooling System

3.1.2. Performance Parameters

The performance of this system can be obtained using the experimental results.

$$\varepsilon_{DW} = \frac{\omega_1 - \omega_2}{\omega_1 - \omega_{2,ideal}} \tag{2}$$

$$\varepsilon_{HRW} = \frac{T_2 - T_3}{T_2 - T_6} \tag{3}$$

$$\varepsilon_{ECpro} = \frac{T_3 - T_4}{T_3 - T_{3w}} \tag{4}$$

$$\varepsilon_{ECreg} = \frac{T_5 - T_6}{T_5 - T_{5w}} \tag{5}$$

3.1.3. CAD Model

A CAD model of the Desiccant Wheel has been made. Desiccant Wheel manufactured by NOVEL Aire have been taken as references. The wheel is mounted on a shaft that will rotate with the help of a motor through belt drive. The shaft is supported by journal bearing from both sides. The frame shows the division of the process and regeneration air. An animation has also been made in order to better understand the working of a desiccant wheel.

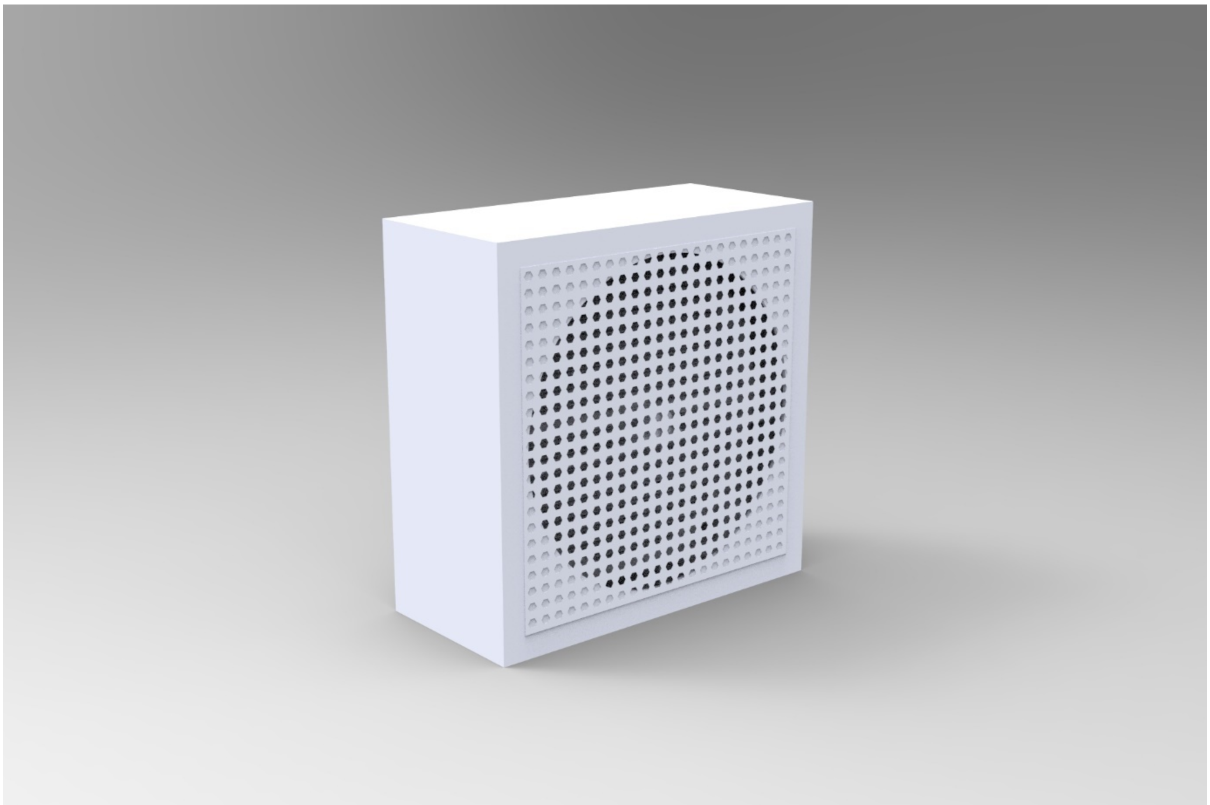

Figure 5 - Desiccant Wheel

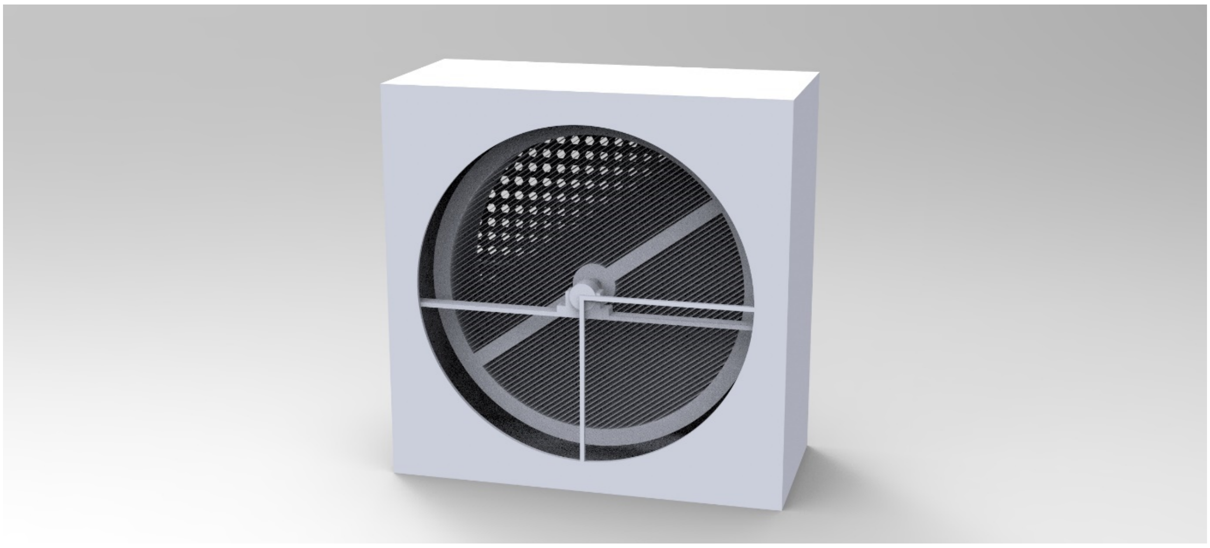

Figure 6 - Desiccant Wheel (Excluding Grill)

3.2. Mathematical Model

In order to establish a proper mathematical model to solve and understand the behavior of this project at different conditions. Following steps are first taken:

- First of all, we will divide the project into three Components:
  - Desiccant Wheel
  - Heat Exchanger
  - Direct Evaporative Cooler
- Make assumption and define the control volume and the factors on which the governing equations must depend
- Then we need to derive the governing equations.
- Boundary conditions must be defined in order to solve the equations
- An approach should be adopted to solve the mathematical model

3.2.1. Desiccant Wheel

Assumptions

- Heat conduction in the desiccant wheel is considered to be negligible, the plane between two channels is taken as adiabatic and impermeable
- The flow is considered to be laminar
- The air condition at the inlet are constant for the whole wheel surface but they are transient, they can vary with time
- The thermodynamic properties like density, specific heat of the dry air and desiccant material are taken as constant.
- In one revolution, it's assumed that the desiccant gets fully saturated in the process region and is completely recovered in the regeneration region.

Control Volume

We will be doing the analysis of the Desiccant wheel

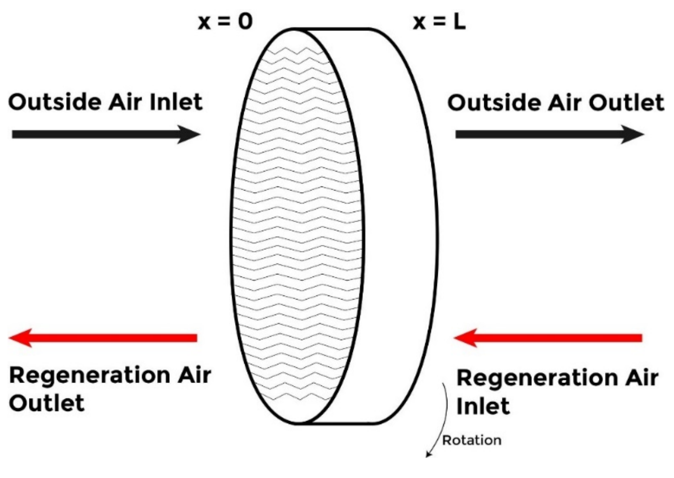

Figure 7 - Desiccant Wheel Schematic

Governing Equation

There are four main governing equations [55], the mass and energy balance equation for both the desiccant material and air. Although in cylindrical coordinates system, there are 3 coordinates but, in this case, we will consider the system to be one dimensional and study the heat and mass transfer along the depth only.

Moisture Conservation in Air

$$d_e \rho_a \left(\frac{\partial Y_a}{\partial t} + u \frac{\partial Y_a}{\partial z} \right) = K_y (Y_d - Y_a) \tag{6}$$

Energy Conservation in Air

$$d_e c_{pa} \rho_a \left(\frac{\partial T_a}{\partial t} + u \frac{\partial T_a}{\partial z} - \frac{k_a}{c_{pa} \rho_a} \frac{\partial^2 T_a}{\partial z^2} \right) = h(T_d - T_a) + c_{pv} K_y (Y_d - Y_a)(T_d - T_a) \tag{7}$$

Moisture Conservation in Desiccant

$$\delta \rho_d \left(\frac{\partial W}{\partial t} + D_e \frac{\partial^2 W}{\partial z^2} \right) = K_y (Y_d - Y_a) \tag{8}$$

Energy Conservation in Desiccant

$$\begin{aligned} c_{pd} \rho_d \delta \left(\frac{\partial T_d}{\partial t} - \frac{k_d}{c_{pd} \rho_d} \frac{\partial^2 T_d}{\partial z^2} \right) \\ = h(T_a - T_d) + K_y (Y_a - Y_d) q_{st} + c_{pv} K_y (Y_a - Y_d)(T_a - T_d) \end{aligned} \tag{9}$$

First Approach (Finite Difference Method)

Finite difference method (FDM) was the approached we use to solve the Partial Different equations.

Table 4 - Interval Ranges for FDM

RANGES	
Time (t) (Represented by "i")	$0 \leq t \leq 2.82$
Axial Distance (z) (Represented by "j")	$0 \leq z \leq 0.1$
i (for time, t)	0 to 2
j (for axial position, z)	0 to 2

After Applying FDM, the PDEs were reduced to the following

Moisture Conservation in Air

$$1.77 \times 10^{-3} Y_{a_{i+1,j}} + 0.838 Y_{a_{i,j+1}} - 0.5633 Y_{a_{i,j}} - 0.2757 Y_{d_{i,j}} = 0 \tag{10}$$

Energy Conservation in Air

$$\begin{aligned} 1.79 T_{a_{i+1,j}} + 841.504 T_{a_{i,j+1}} - 8.492 \times 10^{-3} T_{a_{i,j+1}} - 8.492 \times 10^{-3} T_{a_{i,j-1}} \\ - 799.97 T_{a_{i,j}} = 43.3 T_{d_{i,j}} + 1.1524 (Y_{d_{i,j}} - Y_{a_{i,j}})(T_{d_{i,j}} - T_{a_{i,j}}) \end{aligned} \tag{11}$$

Moisture Conservation in Desiccant

$$0.032W_{i+1,j} - 0.032W_{i,j} + 6.4W_{i,j+1} + 6.4W_{i,j-1} - 12.8W_{i,j} = 0.2757(Y_{d_{i,j}} - Y_{a_{i,j}}) \tag{12}$$

Energy Conservation in Desiccant

$$\begin{aligned} 32.27T_{d_{i+1,j}} - 32.27T_{d_{i,j}} &= 43.3\left(T_{a_{i,j}} - T_{d_{i,j}}\right) + 769478.7\left(Y_{a_{i,j}} - Y_{d_{i,j}}\right) \\ &\quad + 1152.43\left(Y_{a_{i,j}} - Y_{d_{i,j}}\right)\left(T_{a_{i,j}} - T_{d_{i,j}}\right) \end{aligned} \tag{13}$$

Relation Between $\mathbf{Y_d}$ and W

$$\frac{0.75}{W_{i,j}}\left(0.4 - 0.25W_{i,j}\right) = 4.09 \times 10^{-9} \times T_{d_{i,j}} \times Y_{d_{i,j}} \times e^{\frac{5196}{T_{d_{i,j}}}} \tag{14}$$

Because of high degree of non-linearity and the limitations of some software (MAPLE), we couldn't solve these equations and extract results. Therefore, in order to get results we had to go for another approach.

Second Approach (Experimental Interpolation)

We used a simulator created by NOVEL Aire Technology to solve the Desiccant Wheel. The simulator interpolates the given conditions and display the outlet conditions. The wheel manufactured by NOVEL Aire Technology were tested under different conditions and the results of those test and experiment were used to develop this software.

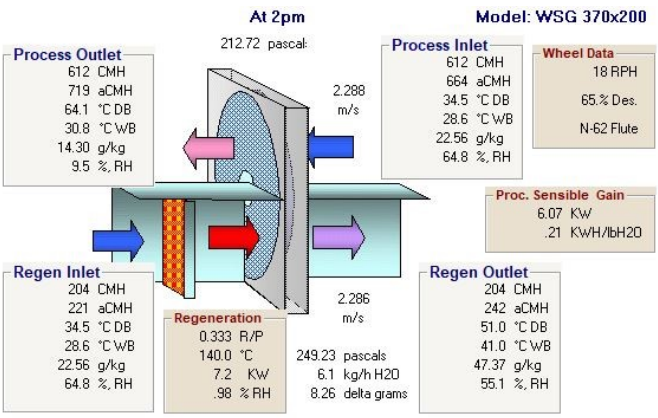

Figure 8 - NOVEL Aire Simulator

One Room Analysis (Karachi)

The Volume flow rate of air was calculated using **HAP software** for Karachi by considering a room of the following dimensions.

$$10\ ft \times 10\ ft \times 10\ ft$$

The needed volume flow rate came out to be **612 Cubic Meter per hour**.

- We decided to solve the simulation for one of Pakistan most humid region i.e. Karachi [9].
- The wheel **RPH** is taken to be **17.5** [2].
- After few hit and trials, it was found that the needed heater temperature for Karachi Climate is **140°C.** in order to reduce humidity up to certain extent.
- For regeneration a split ratio of 1/3 is considered
- Outdoor air is used for regeneration.

Home Analysis

Cooling load and required mass flow rate was then estimated for a complete home this time using HAP. It was observed that by only changing the diameter of the desiccant wheel, the same outlet conditions can be obtained without having to change the depth of the wheel. The diameter increased by two times in order to design the whole system for the complete house. Using the same flow rate, analysis was performed in three different cities of Pakistan and the behavior was observed on a monthly basis. The outlet temperature, humidity and required regeneration heater temperature were estimated

Following were the parameters set before performing the simulation

Table 5 - Simulation Parameters

Atmospheric Pressure	101325
Assumed Air Flow Rate (CMH)	4500
Humidity Criteria	11 to 14
Temperature Max (Leaving Desiccant Wheel)	60
Mass flow rate of air (kg/s)	1.53125
C_p of air kJ/(kg.K)	1.005
Required Final Temperature (Entering Room)	25

Third Approach (TRNSYS Analysis)

TRNSYS (Transient System Simulation Tool) is a simulation software basically used to simulate transient system. The Desiccant Based Evaporative Cooling system was designed on this software and was simulated by varying conditions. It was then compared with the results of our experimental interpolation approach.

Manish Mishra [56] studied ventilation and recirculation mode of the Desiccant Based Evaporative Cooling system. A lecture hall was considered with a Cooling capacity of 30kW. It was observed that the COP for recirculation mode is higher than the ventilation mode. The simulation was solved for Karachi, the flow rate for both the streams (process air and return air) was taken to be 2.5 kg/s. In the other case, the return air stream was taken to be 1.25 kg/s.

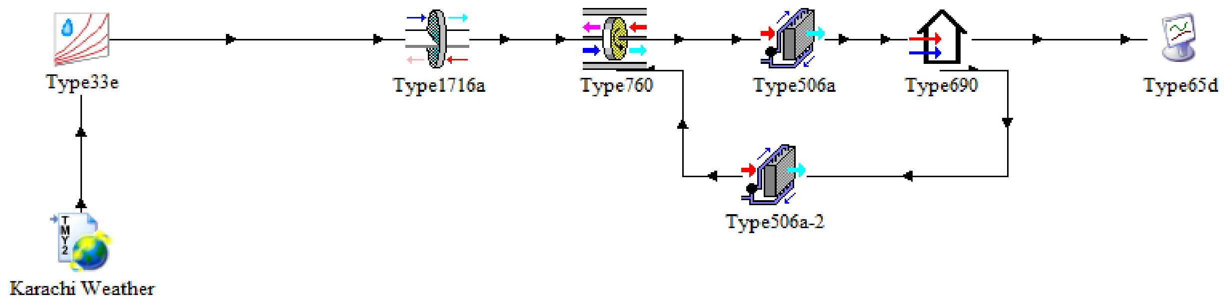

Figure 9 - TRNSYS Studio (Ventilation)

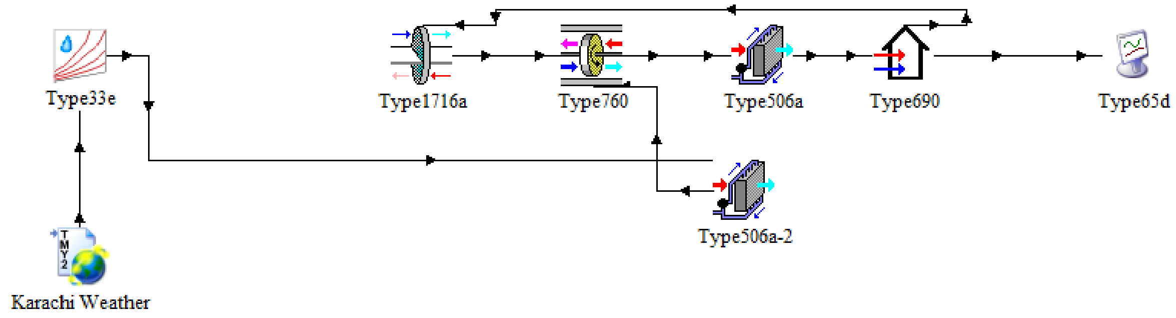

Figure 10 - TRNSYS (Recirculation)

List of components used is shown on the next page.

Table 6 - TRNSYS Components

No.	Component	TRNSYS Name
1	Weather Data Reader	Type109-TMY2
2	Psychrometric	Type33e
3	Desiccant Wheel	Type1716a
4	Heat Exchanger	Type760
5	Evaporative Cooler	Type506a
6	Conditioned Space	Type690

3.2.2. Heat Exchanger

In order to reduce the temperature of air at the outlet of desiccant wheel, a heat exchanger is needed. To accomplish the task of heat exchanger in a more cost effective way was to use an **indirect evaporative cooler**. Its calculations were calculated on the basis of "Karachi one room" results for the worst-case scenario (2pm).

Indirect Evaporative Cooler

There are two streams in the indirect evaporative cooler: -

i. The Hot air stream
ii. Water stream

The outlet air from the desiccant wheel is at a temperature of 63.4ºC and we have to reduce it to 40ºC. The water stream is taken at 25ºC, while the flow rate of water is assumed to be 0.045kg/s (2.6 l/min). We will solve it considering it act as a shell and tube heat exchanger using the following equations.

$$Nu = 0.023 \times Re^{0.8} \times Pr^{0.4} \tag{15}$$

$$h_a = \frac{Nu \times k}{D} \tag{16}$$

$$Nu = 0.332 \times Re^{0.5} \times Pr^{1/3} \tag{17}$$

$$R = \frac{1}{h_i A_i} + \frac{\ln(\frac{D_o}{D_i})}{2\pi k l} + \frac{1}{h_o A_o} \tag{18}$$

$$U = \frac{1}{R \times A_s} \tag{19}$$

$$Re_a = \frac{V_m\ D_h}{v} \tag{20}$$

$$NTU\ (Cross\ Flow\ Heat\ Exchanger) = \frac{1}{c-1} \ln\left(\frac{\varepsilon - 1}{\varepsilon c - 1}\right) \tag{21}$$

3.2.3. Direct Evaporative Cooler

After leaving the heat exchanger, the air is at around 40ºC with the same humidity as after leaving the desiccant i.e. 2.458 g/kg. The value of Wet-Bulb Temperature obtained from Psychometric chart is 22.59 ^{0}C.

3.3. STRUCTURAL DESIGN

Our aim is to design a shaft that will be able to rotate a desiccant wheel. According to the manual provided by NOVEL Aire, the mass of the wheel that we are using is around 50kg.

Following are the specifications of the wheel.

Table 7 - Wheel Specifications

Mass	110 kg
Weight	1079.1 N
Diameter	770 mm
Width	200 mm
Material	Galvanized Steel
Power Required	9.20 W

The angular velocity with which the wheel needs to rotate is 18 RPH (Revolution per Hour).

So,

$$\omega = 18\,RPH = 18 \times \frac{2\pi}{3600} = 0.0314\,rad/s \tag{22}$$

As,

$$P = Torque \times Angular\,Velocity \tag{23}$$

$$Torque = T = \frac{9.20}{0.0314} = 292.84\,Nm \tag{24}$$

3.3.1. Free Body Diagram

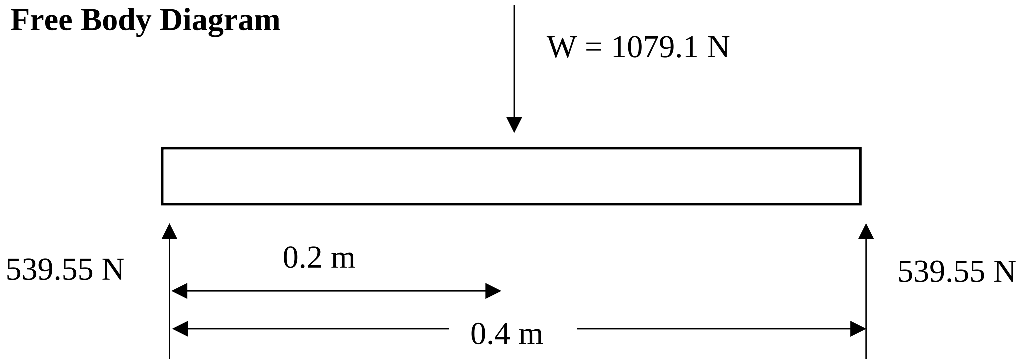

Figure 11 - Free Body Diagram

3.3.2. Calculations

$$\sum F_y = 0 \tag{25}$$

$$\sum M = 0 \tag{26}$$

$$1079.1 = A_1 + A_2 \tag{27}$$

Where

$$A_1 = Reaction\ at\ one\ point$$

$$A_2 = Reaction\ at\ other\ point$$

Applying Moment Equation at one point.

$$1079.1 \times 0.2 = A_2 \times 0.4 \tag{28}$$

$$A_2 = 539.55\ N \tag{29}$$

$$A_1 = 539.55\ N \tag{30}$$

On the basis of the obtained results, shear force and bending moment diagram were developed, the maximum shear and bending stress theory were analyzed under the given circumstances and the diameter of the shaft was determined.

3.4. FINITE ELEMENT METHOD

The fatigue analysis of the designed shaft was performed in order to verify its sustainability. ANSYS software was used.

3.4.1. Geometry

The shaft has been designed of 35 mm with stainless steel as material being used. In order to simplify the problem, instead of separately adding the weight of the Desiccant Wheel, a circular material has been added to the shaft with an approximately similar mass of the desiccant wheels. The added material is placed at the same point as the wheel position.

The material, added intentionally, dimension have been calculated using basic mass, volume and density equations.

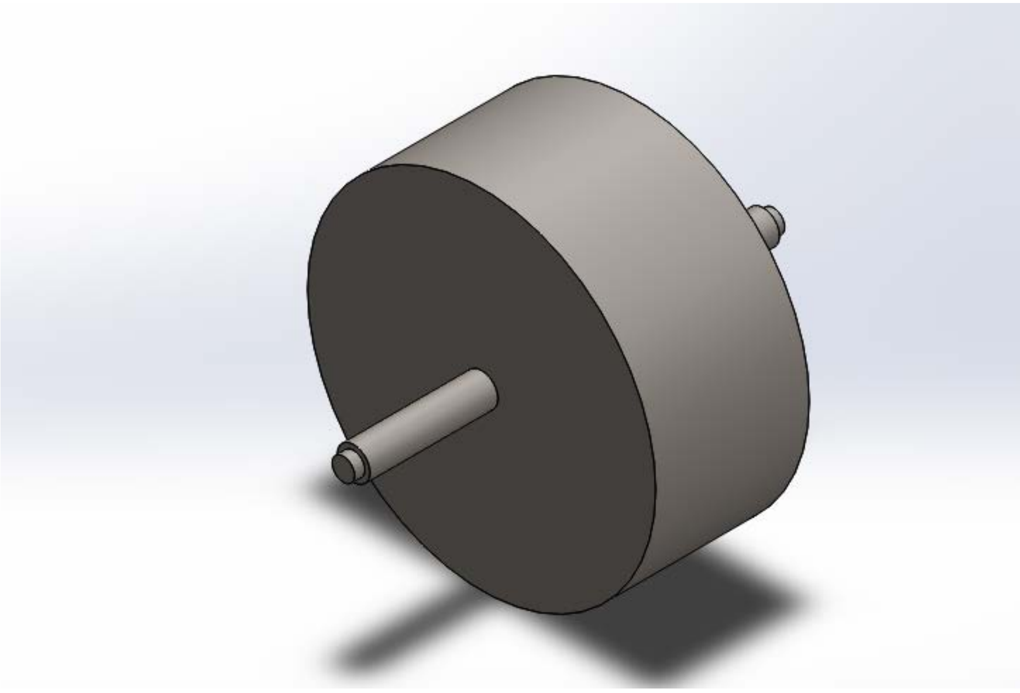

Figure 12 - Simplified Model of Shaft

$$m = \rho V \tag{31}$$

$$m = 110\ kg \tag{32}$$

$$\rho = density\ of\ galvanized\ steel \tag{33}$$

$$V = Volume\ of\ hollow\ cylinder = \frac{\pi}{4} \times ({d_2}^2 - {d_1}^2) \tag{34}$$

$$d_1 = Internal\ Diameter$$
$$d_2 = External\ Diameter$$

3.4.2. Mesh

Tetrahedron, fine meshes have been used.

$$No. of\ nodes\ =\ 67401$$
$$No. of\ Elements\ =\ 43490$$

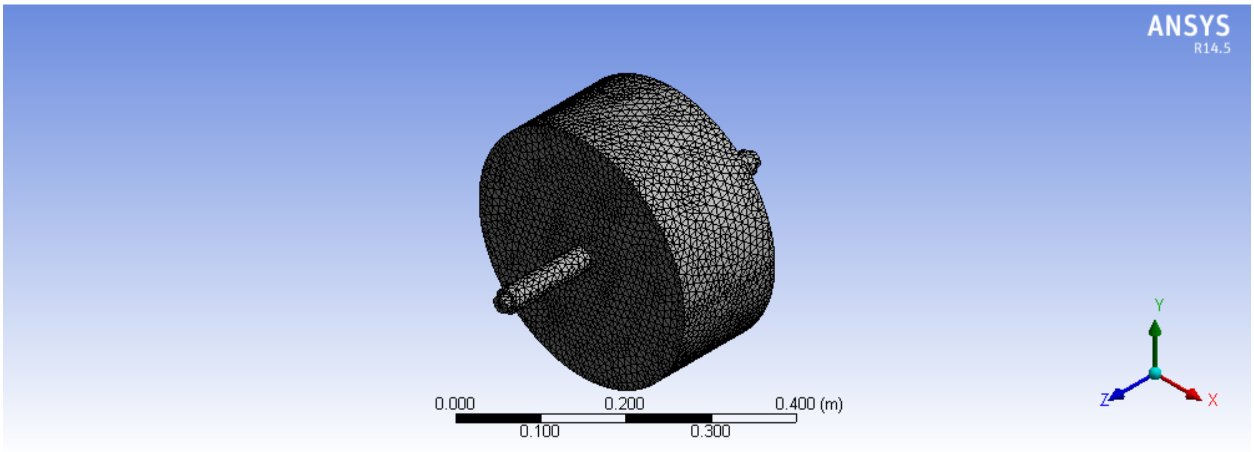

Figure 13 - Mesh of Geometry

3.4.3. Loading Conditions

A Moment and Angular Velocity has been given to the shaft.

$$M\ =\ 292.84\ Nm$$
$$Angular\ Velocity = \omega = 18RPH = 0.0314\ rad/s$$

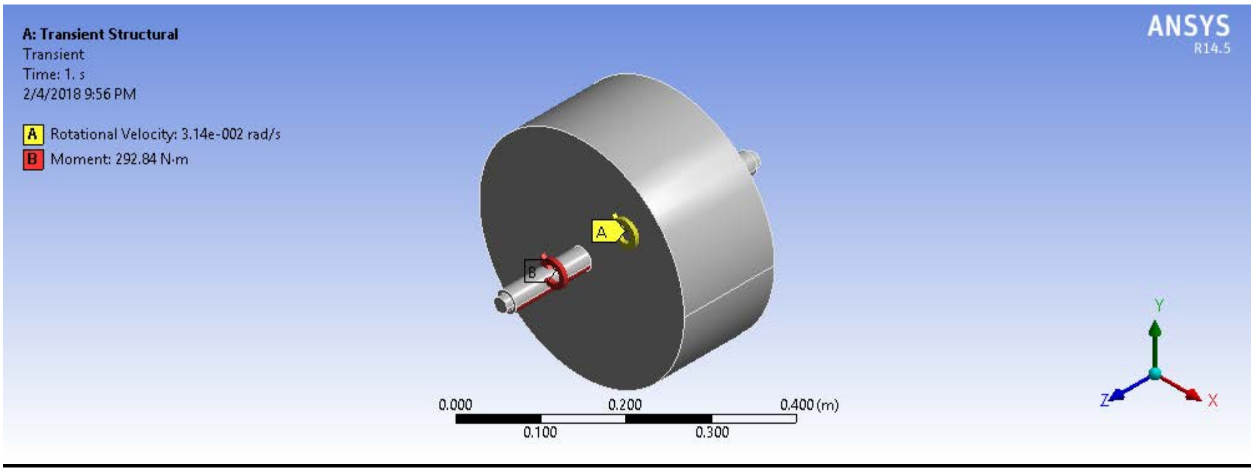

Figure 14 - Loading on Geometry

3.4.4. Stress Theory

Goodman Stress Theory has been utilized in the Fatigue Tool [57].

CHAPTER 4: RESULTS

4.1. Theoretical Results

4.1.1. Desiccant Wheel

Second Approach (Experimental Interpolation)

One Room (Karachi)

Table 8 - Experimental Results for Karachi

Time	Before Entering Desiccant Wheel				After Leaving Desiccant Wheel	
	Temperature (°C)	Relative Humidity (%)	Saturated Pressure (Pa)	Absolute Humidity (kg/kg)	Outlet Dry Temperature (°C)	Outlet Humidity (kg/kg)
12:00	35	55	5699.031372	0.019855668	63.7	0.01218
13:00	35	55	5699.031372	0.019855668	63.7	0.01218
14:00	34.5	64	5539.896546	0.02255404	**64.1**	**0.0143**
15:00	35	57	5699.031372	0.020601607	63.9	0.01278
16:00	34	64	5384.708507	0.021899993	63.6	0.01368

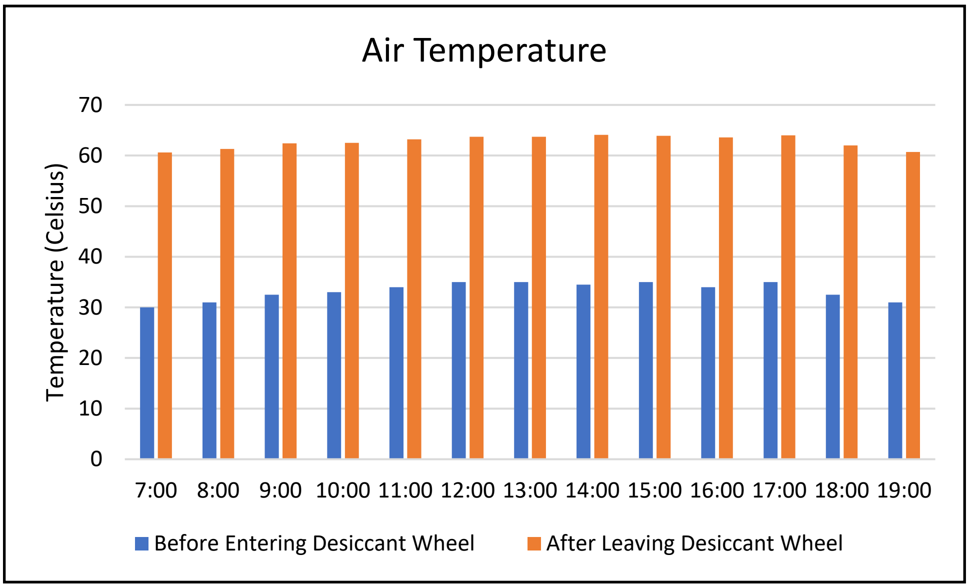

Figure 15 - Temperature of Air Before and After Desiccant Wheel

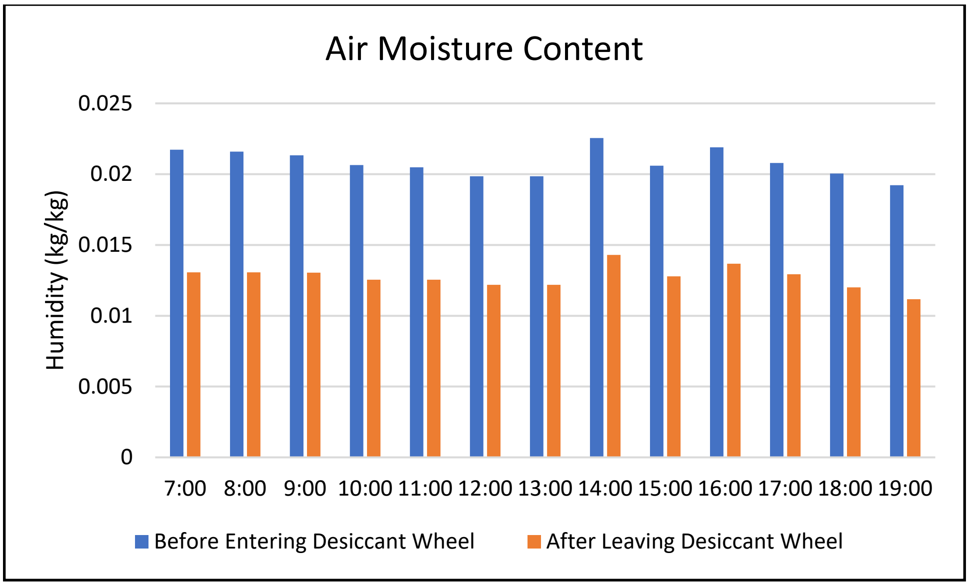

Figure 16 - Moisture of Air Before and After the Desiccant Wheel

HAP Results

The results used were of a home that "BGreen", a team we are part of, are designing and looking to build in Dubai. The required Air flow rate was taken from the analysis. Following are the analysis results.

Air System Information

Air System Name	Living+Kitchen	
Equipment Class	PKG ROOF	
Air System Type	VAV	
Number of zones	1	
Floor Area	55.4	m²
Location	Dubai, United Arab Emirates	

Sizing Calculation Information

Zone and Space Sizing Method:

Zone L/s	Peak zone sensible load
Space L/s	Individual peak space loads
Calculation Months	Nov to Dec
Sizing Data	Calculated

Central Cooling Coil Sizing Data

Total coil load	16.5	kW
Sensible coil load	15.6	kW
Coil L/s at Nov 1600	920	L/s
Max block L/s at Nov 1600	997	L/s
Sum of peak zone L/s	997	L/s
Sensible heat ratio	0.942	
m²/kW	3.4	
W/m²	298.3	
Water flow @ 5.6 °K rise	N/A	
Load occurs at	Nov 1600	
OA DB / WB	37.2 / 21.8	°C
Entering DB / WB	26.8 / 17.5	°C
Leaving DB / WB	12.8 / 11.9	°C
Coil ADP	11.2	°C
Bypass Factor	0.100	
Resulting RH	45	%
Design supply temp.	12.8	°C
Zone T-stat Check	1 of 1	OK
Max zone temperature deviation	0.0	°K

Preheat Coil Sizing Data

No heating coil loads occurred during this calculation.

Supply Fan Sizing Data

Actual max L/s at Nov 1600	997	L/s
Standard L/s	997	L/s
Actual max L/(s-m²)	18.00	L/(s-m²)
Fan motor BHP	2.98	BHP
Fan motor kW	2.22	kW

Outdoor Ventilation Air Data

Design airflow L/s	42	L/s
L/(s-m²)	0.75	L/(s-m²)
L/s/person	4.16	L/s/person

Figure 17 - HAP Results for a Complete House

Karachi Results

Table 9 - Karachi Results (Full House)

Months	Desiccant Wheel Inlet		Desiccant Wheel Outlet			Cooling Load Required (kW)
	Temperature	Humidity%	Heater Temperature	Temperature	Absolute Humidity	
April	29	59	70	39.1	11.74	21.70
May	31	67	120	52.9	12.67	42.94
June	32	69	130	56.1	13.9	47.86
July	31	72	130	55.3	13.48	46.63
August	30	73	120	52.3	13	42.01
September	30	70	110	49.7	12.86	38.01
October	29	58	70	39.1	11.5	21.70

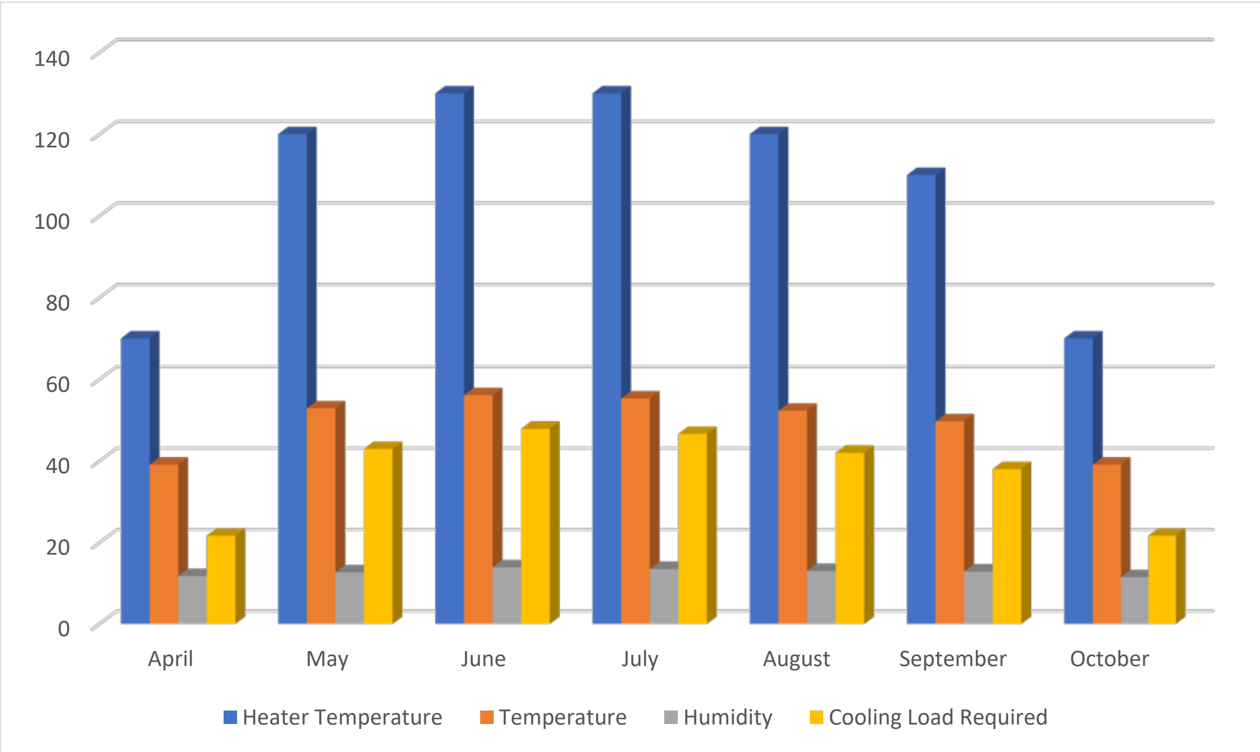

Figure 18 - Graphical Representation (Karachi Results)

Lahore Results

Table 10 - Lahore Results (Full House)

Months	Desiccant Wheel Inlet		Desiccant Wheel Outlet			Cooling Load Required (kW)
	Temperature	Humidity%	Heater Temperature	Temperature	Absolute Humidity	
April	27	44	66	35.8	7.18	16.62
May	32	37	66	39.9	8.75	22.93
June	33	48	100	49	10.63	36.93
July	31	70	130	55.2	13	46.47
August	31	74	130	55.5	13.98	46.94
September	29	69	110	48.7	11.59	36.47
October	26	61	70	36.7	9.57	18.01

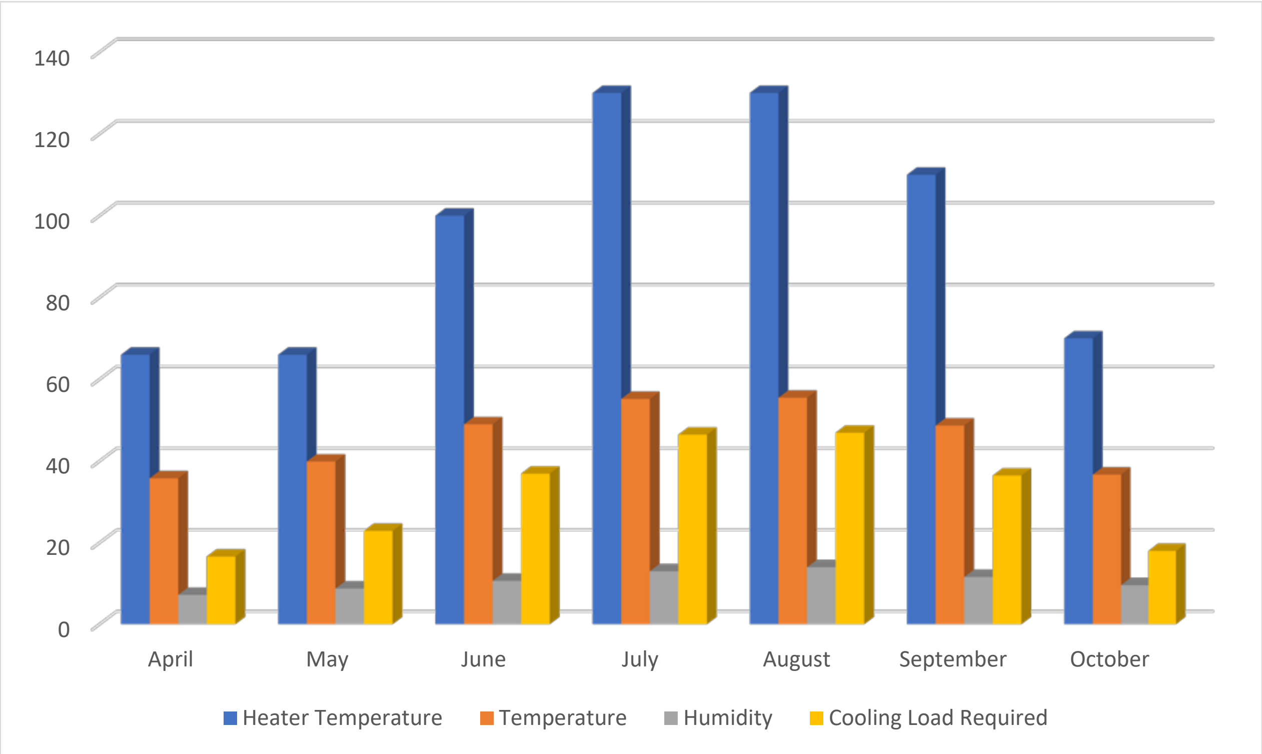

Figure 19 - Graphical Representation (Lahore Results)

Islamabad Results

Table 11 - Islamabad Results (Full House)

Months	Desiccant Wheel Inlet		Desiccant Wheel Outlet			Cooling Load Required (kW)
	Temperature	Humidity%	Heater Temperature	Temperature	Absolute Humidity	
May	29	41	66	37.5	7.78	19.24
June	31	45	66	39.3	12.23	22.01
July	30	67	110	49.5	12.15	37.70
August	29	74	110	49	12.64	36.93
September	28	68	80	41.1	12.13	24.78

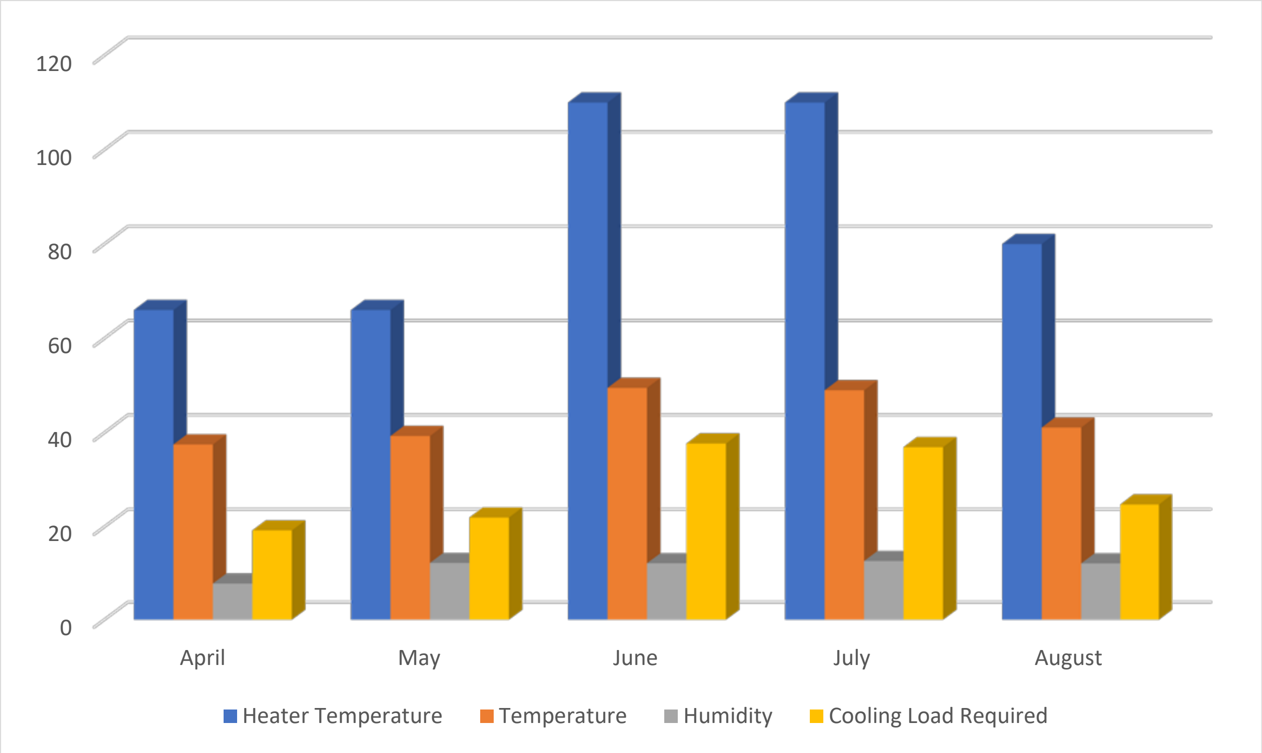

Figure 20 - Graphical Representation (Islamabad Results)

Third Approach (TRNSYS Analysis)

Ventilation Cycle

The simulation was time was set from March to September. The mass flow rate was taken to be 2.5 kg/s for the Process air and 1.25 kg/s for the Regeneration Air. Weather data of Karachi was imported. The maximum temperature observed was **26.08º C** with a relative humidity between 80% to 90% at the outlet of Evaporative Cooler.

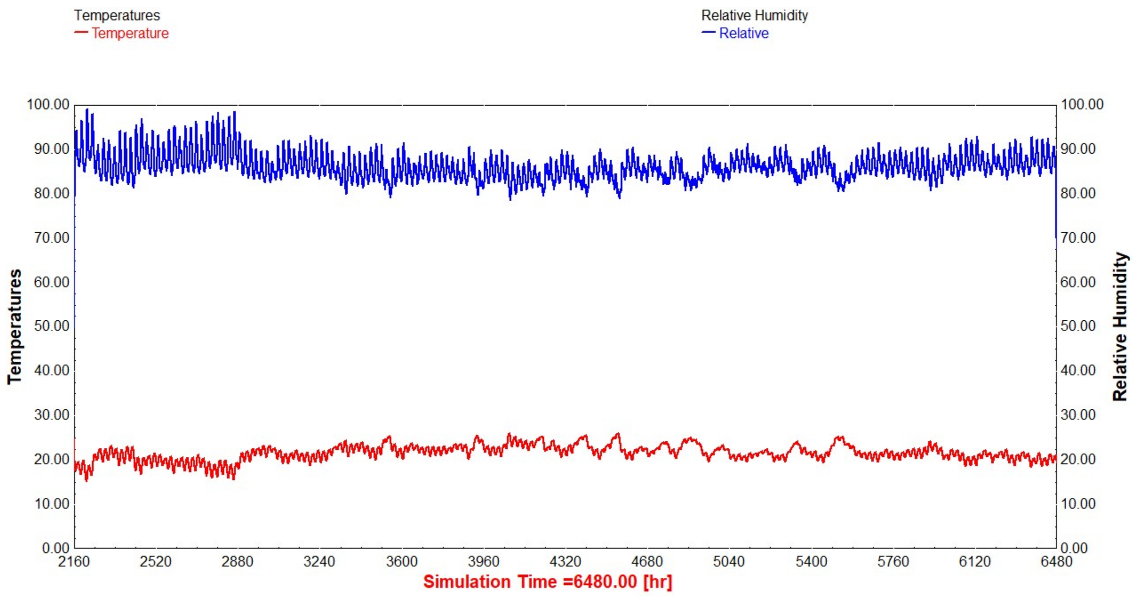

Figure 21 - Simulation Results Ventilation Cycle

Recirculation Cycle

Under same condition, simulation was performed for the Recirculation Cycle. The maximum temperature came out to be 19.2^{o} C but the relative humidity was found to be higher than Ventilation Cycle

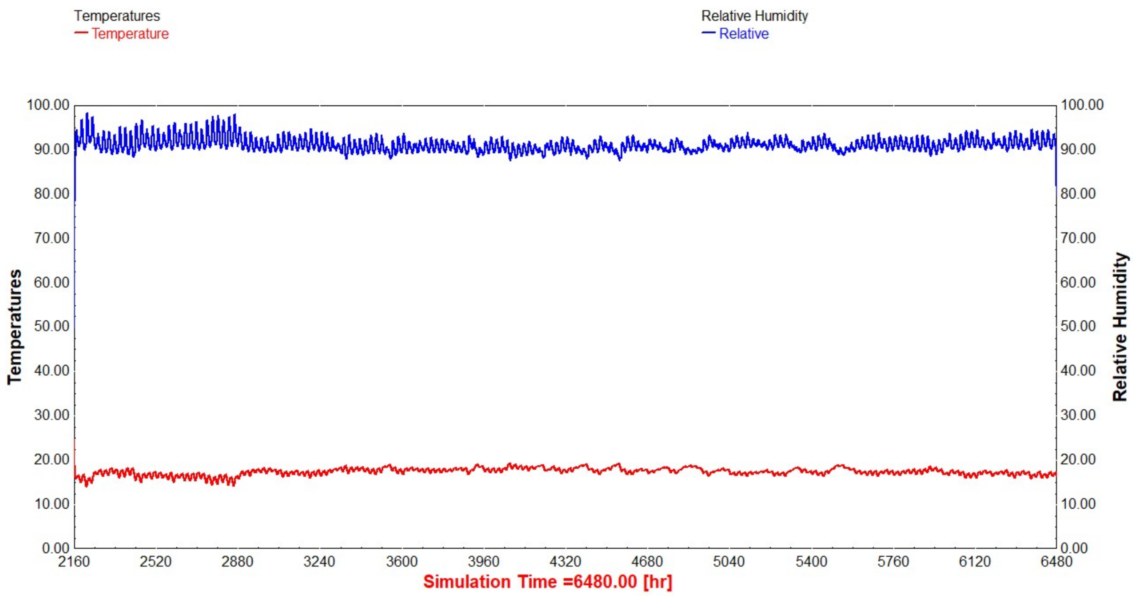

Figure 22 - Simulation Results (Recirculation Cycle)

The ventilation cycle was further studied under different condition.

Air Conditioning using Single Evaporative Cooler (No Desiccant Wheel)

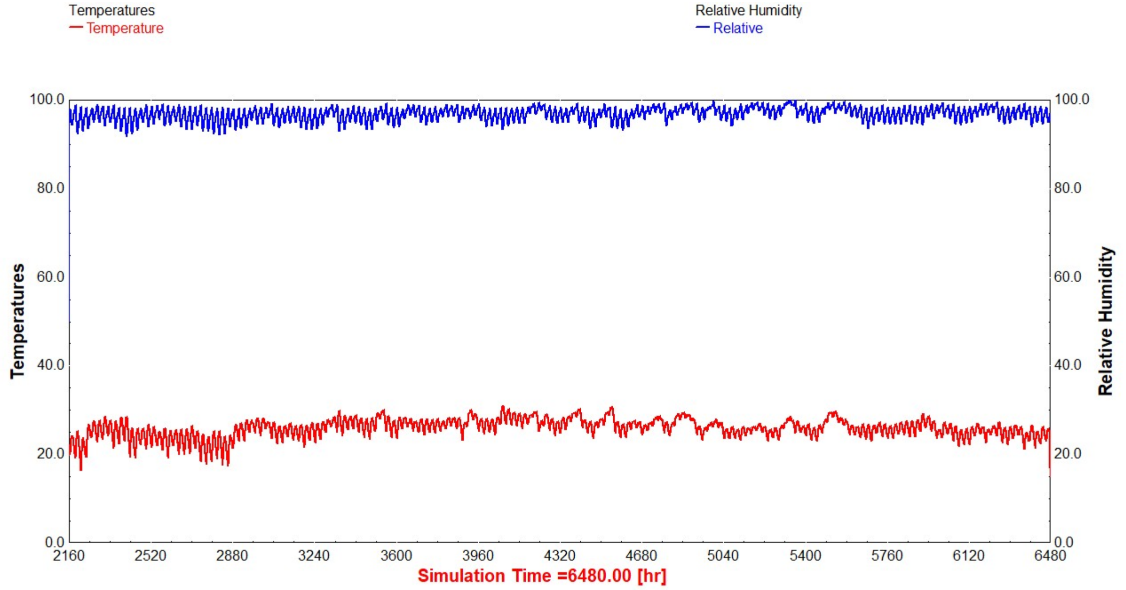

Figure 23 - Air Conditioning using a single Evaporative Cooler (No Desiccant Wheel)

Desiccant Based Air Conditioning without Heat Exchanger (Ventilation Mode)

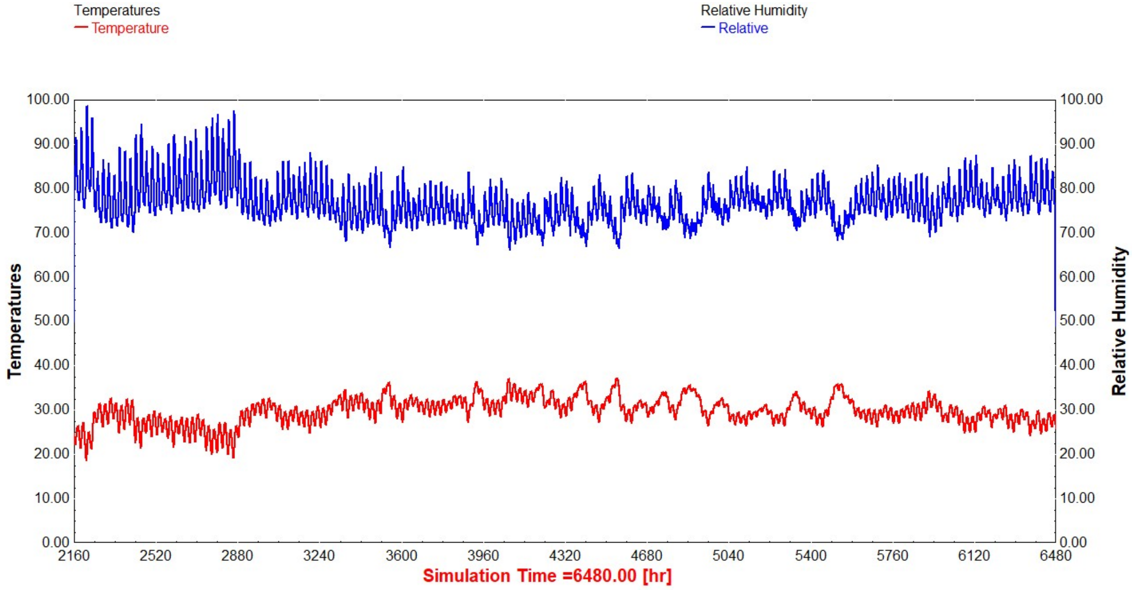

Figure 24 - Desiccant Based Air Conditioning without Heat Exchanger (Ventilation Mode)

Ventilation Mode with reduced effectiveness of Heat Exchanger

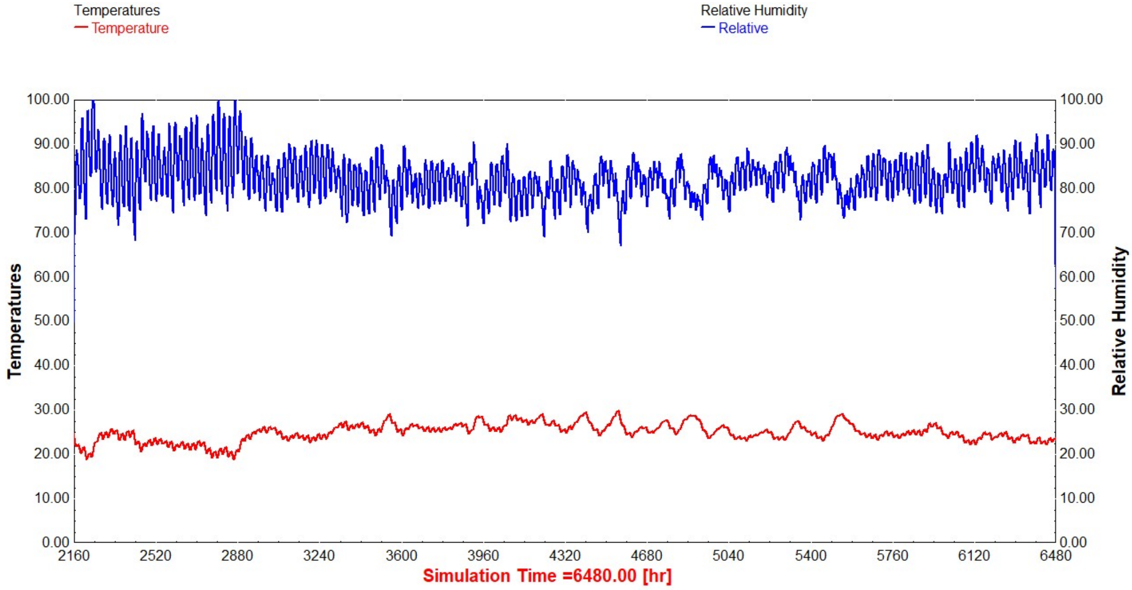

Figure 25 – Heat Exchanger Effectiveness Reduced (Ventilation Mode)

Process Air Flow Rate 0.6 m/s (Ventilation Mode)

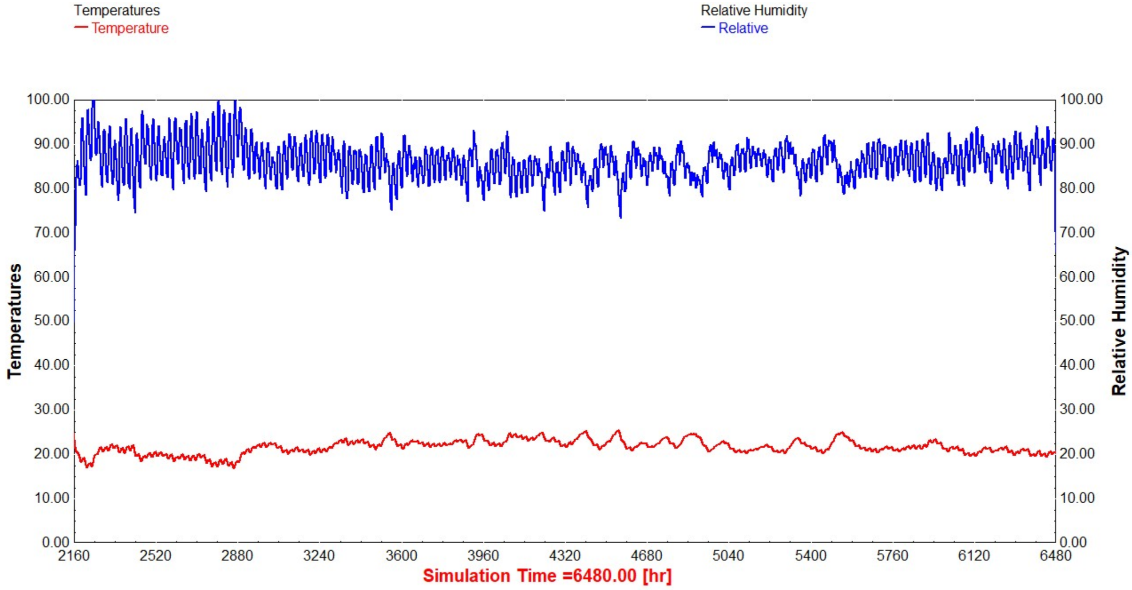

Figure 26 - Process Air 0.6 m/s (Ventilation Mode)

4.1.2. Heat Exchanger

Air Data

$$Mass\ flow\ rate\ of\ Air\ =\ \dot{m}_a = 0.1064\ kg/s \tag{35}$$

$$Outlet\ Temperature\ of\ Air\ =\ T_a = 63.4\ ℃ \tag{36}$$

$$Specific\ Heat\ of\ Air\ =\ c_{p,a} = 1.007\ kJ/(kg.K) \tag{37}$$

$$Density =\ \rho_a = 1.059\ kg/m^3 \tag{38}$$

$$Thermal\ Conductivity = k_a = 0.02808\ W/(m.K) \tag{39}$$

$$Kinematic\ Viscosity =\ \nu_a = 1.896\ \times 10^{-5}\ m^2/s \tag{40}$$

$$Pr_a = 0.7202 \tag{41}$$

Water Data

$$Mass\ flow\ rate\ of\ Water\ =\ \dot{m}_w = 0.3\ kg/s \tag{42}$$

$$Outlet\ Temperature\ of\ Water\ =\ T_w = 25\ ℃ \tag{43}$$

$$Specific\ Heat\ of\ Water\ =\ c_{p.w} = 4.184\ kJ/(kg.K) \tag{44}$$

$$Density =\ \rho_w = 997\ kg/m^3 \tag{45}$$

$$Thermal\ Conductivity = k_w = 0.607\ W/(m.K) \tag{46}$$

$$Kinematic\ Viscosity =\ \nu_w = 8.94\ \times 10^{-7}\ m^2/s \tag{47}$$

$$Distribution\ Pipe\ Diameter\ (Assumption) = 0.75\ in = 0.01905\ m \tag{48}$$

$$Area\ of\ Cross-Section = A_{c,w} = \frac{\pi}{4}(0.01905)^2 = 2.85 \times 10^{-4}\ m^2 \tag{49}$$

$$Pr_w = 6.14 \tag{50}$$

Pipe Data

$$Outer\ Diameter\ =\ D_o = 1\ inch\ =\ 2.54\ \times 10^{-2}\ m \tag{51}$$

$$Length\ =\ 12\ inch\ = 30.48 \times 10^{-2}\ m \tag{52}$$

$$Number\ of\ pipes = n = 8 \tag{53}$$

$$Gauge = 21 = 0.873\ mm \tag{54}$$

$$Inner\ Diameter\ =\ D_i =\ 0.024527\ m \tag{55}$$

$$Area\ of\ Cross-Section = A_{c,a} = \frac{\pi}{4}(0.0245247)^2 = 4.72 \times 10^{-4}\ m^2 \tag{56}$$

$$Inner\ Surface\ Area = A_i = \pi \times 0.024527 \times 30.48 \times 10^{-2} = 0.02348\ m^2 \quad (57)$$

$$Outer\ Surface\ Area = A_o = \pi \times 0.0254 \times 30.48 \times 10^{-2} = 0.0243\ m^2 \quad (58)$$

Solution

$$Velocity\ of\ Air = v_a = \frac{0.1064}{8 \times 1.059 \times 4.72 \times 10^{-4}} = 26.61\ m/s \quad (59)$$

$$Reynolds\ Number = Re_a = \frac{26.61 \times 0.024527}{1.896 \times 10^{-5}} = 34441.344\ (turbulent) \quad (60)$$

$$Nu = 0.023 \times Re^{0.8} \times Pr^{0.4} = 0.023 \times 34441.344^{0.8} \times 0.7202^{0.4} = 85.97 \quad (61)$$

$$h_a = \frac{Nu \times k}{D} = 85.97 \times \frac{0.02808}{0.024527} = 98.42\ W/(m^2.K) \quad (62)$$

$$Velocity\ of\ Water = v_w = \frac{0.045}{997 \times 2.85 \times 10^{-4}} = 0.1583\ m/s \quad (63)$$

$$Reynolds\ Number = Re_w = \frac{0.1583 \times 0.01905}{8.94 \times 10^{-5}} = 33.73\ (laminar) \quad (64)$$

$$Nu = 0.332 \times Re^{0.5} \times Pr^{1/3} = 0.332 \times 33.73^{0.5} \times 6.14^{1/3} = 3.53 \quad (65)$$

$$h_w = \frac{Nu \times k}{D} = 3.53 \times \frac{0.607}{0.01905} = 112.47\ W/(m^2.K) \quad (66)$$

$$R = \frac{1}{98.42 \times 0.0243} + \frac{\ln(\frac{0.0254}{0.0245})}{2\pi \times 54 \times 30.48 \times 10^{-2}} + \frac{1}{112.47 \times 0.02348} = 0.797 \quad (67)$$

$$U = \frac{1}{R \times A_s} = \frac{1}{0.797 \times 10 \times 0.0243} = 5.162\ W/(m^2.K) \quad (68)$$

$$C_h = m_a \times C_{p,a} = 0.1064 \times 1.005 = 0.1069 \quad (69)$$

$$C_c = m_w \times C_{p,w} = 0.3 \times 4.184 = 1.2552 \quad (70)$$

$$NTU = \frac{A_s \times U}{C_{min}} = \frac{0.0243 \times 5.162}{.1069} = 1.17 \quad (71)$$

$$\varepsilon = 1 - e^{\left\{\frac{NTU^{0.22}}{c}\left[e^{-cNTU^{0.78}} - 1\right)\right\}} = 0.67 \quad (72)$$

$$Q_{max} = C_{min} \times (T_h - T_c) = 0.1069 \times (63.4 - 25) = 4.104\ kW \quad (73)$$

$$Q_{actual} = Q_{max} \times \varepsilon = 4.104 \times 0.67 = 2.75\ kW \quad (74)$$

$$T_{a,2} = \left(T_{a,1} - \frac{Q_{actual}}{C_h}\right) = 63.4 - \frac{2.75}{0.1069} = 37.68\ ℃ \quad (75)$$

4.1.3. Direct Evaporative Cooler

DATA

$$Effectiveness = \varepsilon = 0.90$$
$$Dry\ Bulb\ before\ entering\ the\ cooler = T_{db,1} = 40℃$$
$$Wet\ Bulb\ before\ entering\ the\ cooler = T_{wb} = 22.59℃$$
$$Dry\ Bulb\ after\ leaving\ the\ cooler = T_{db,2} = ?\ ℃$$

$$T_{db,2} = T_{db,1} - 0.90(T_{db,1} - T_{wb}) = 40 - 0.90(40 - 22.59) = 24.331℃ \tag{76}$$

The process can be made more efficient if the return air from the room (at optimum room temperature) is used instead of water. The temperature of desiccant outlet air will decrease and the temperature of return air will increase. This return air is being used to regenerate the desiccant. An already high temperature return air will require less energy to get heated, ultimately reducing the heater power consumption. TRNSYS software was used to perform simulation under this situation.

4.1.4. Structural Design

Shear Force and Bending Moment Diagram

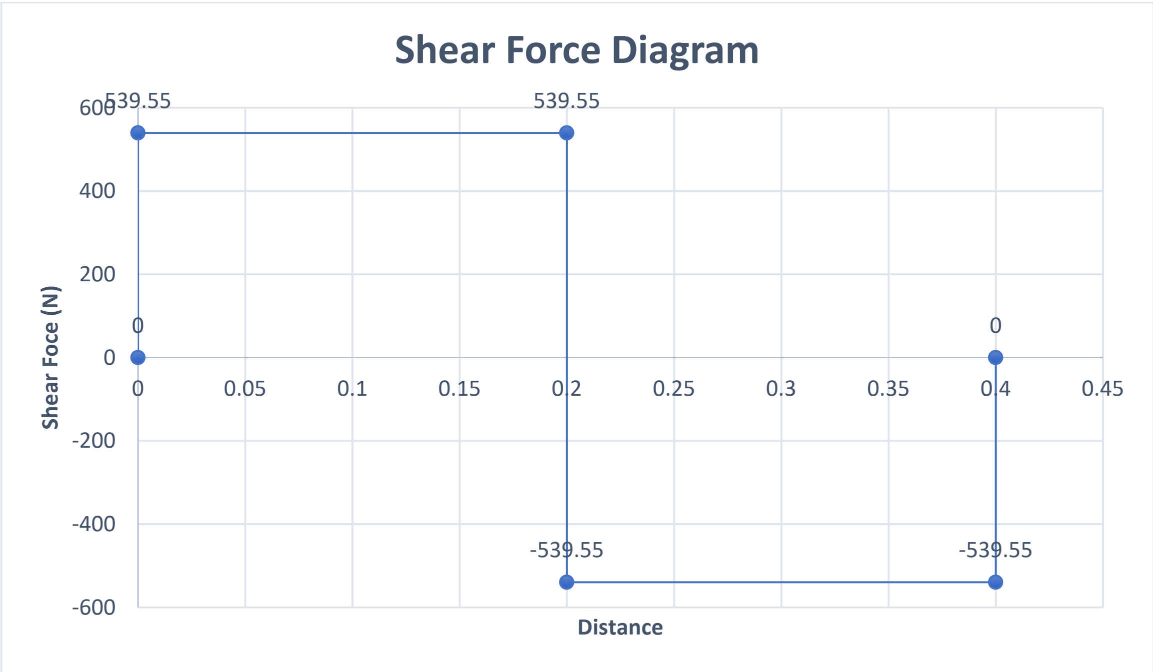

Figure 27 - Shear Force Diagram of Shaft

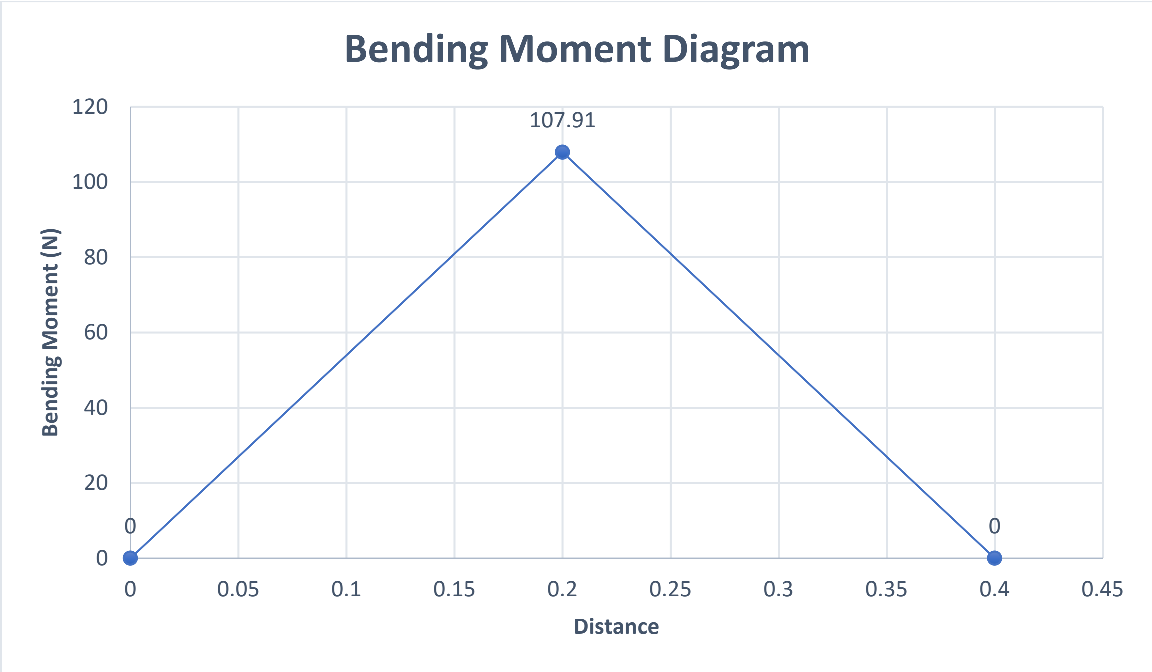

Figure 28 - Bending Moment Diagram of Shaft

Design Parameters

$$Yield\ Strength\ of\ Stainless\ Steel = \sigma_b = 200MPa \tag{77}$$

$$Shear\ Strength\ of\ Stainless\ Steel = \tau = 0.7 \times 500MPa = 140\ MPa \tag{78}$$

If we use a factor of safety

$$FS = 3 \tag{79}$$

$$Allowable\ Yield\ Strength\ of\ Stainless\ Steel = \sigma_b = \frac{200}{3} MPa = 66.67\ MPa \tag{80}$$

$$Allowable\ Shear\ Strength\ of\ Stainless\ Steel = \tau = 0.7 \times \frac{200}{3} MPa = 46.67\ MPa \tag{81}$$

Maximum Shear Stress Theory

$$T_e = \sqrt{M^2 + T^2} = \sqrt{107.91^2 + 298.84^2} = 317.72\ Nm \tag{82}$$

$$d^3 = 16 \times \frac{T_e}{\pi\tau} = \frac{16 \times 317.72}{\pi \times 46.67 \times 10^6} = 3.47 \times 10^{-5} \tag{83}$$

$$d = 32.6\ mm \tag{84}$$

Maximum Normal Stress Theory

$$M_e = \frac{M + \sqrt{M^2 + T^2}}{2} = \frac{107.91 + \sqrt{107.91^2 + 298.84^2}}{2} = 212.82\ Nm \quad (85)$$

$$d^3 = 32 \times \frac{M_e}{\pi \sigma_b} = \frac{32 \times 212.82}{\pi \times 66.67 \times 10^6} = 3.25 \times 10^{-5} \quad (86)$$

$$d = 32\ mm \quad (87)$$

So, (according to Eq. 27) we will go for a shaft with a diameter of more than 32 mm. Let's select one with a diameter of 35 mm.

FEM

Fatigue Analysis was performed on the shaft by applying a weight equivalent to the of Desiccant Wheel at the center of shaft. Momentum and Angular Velocity were loaded on the shaft.

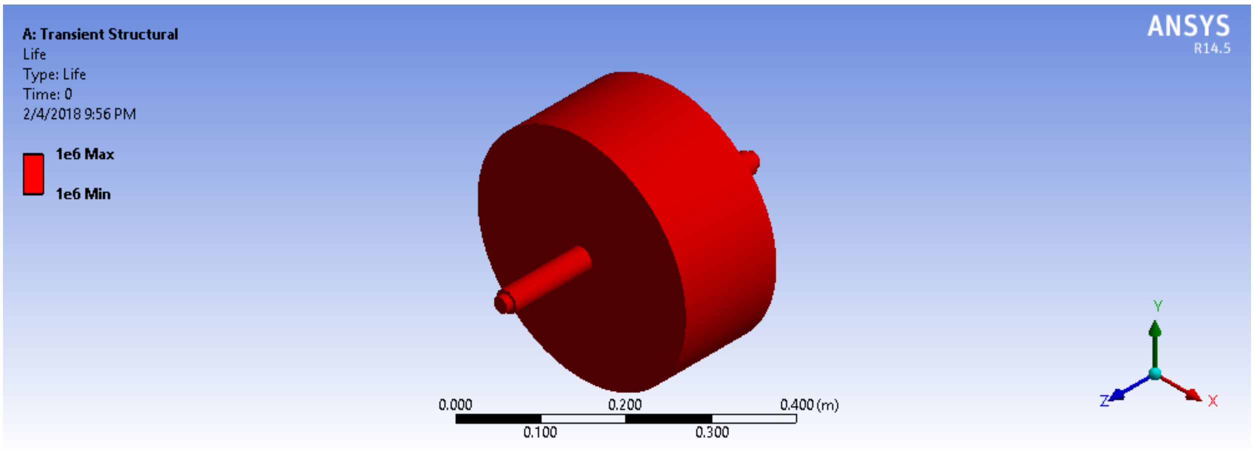

Figure 29 - Fatigue Life of Shaft

It can be observed from Fig. 20 that the life of shaft is infinite at the given conditions

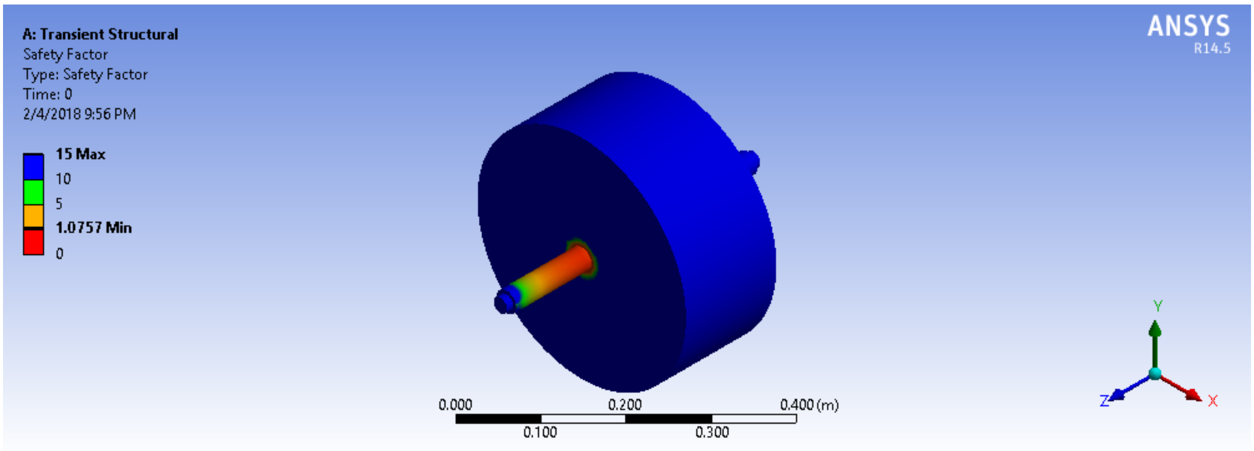

Figure 30 - Safety Factor

From Fig. 21 It can be seen that the point near the load, where the wheel is fitted show a lower factor of safety.

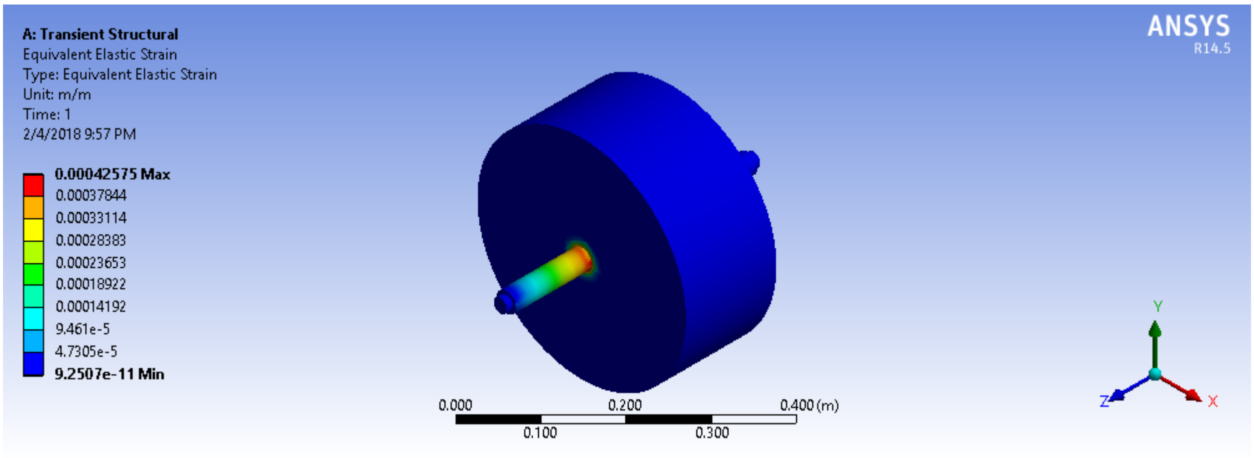

Figure 31 - Von Mises Strain

From Fig. 22, the maximum equivalent strain is seen near the point the wheel is attached with the shaft.

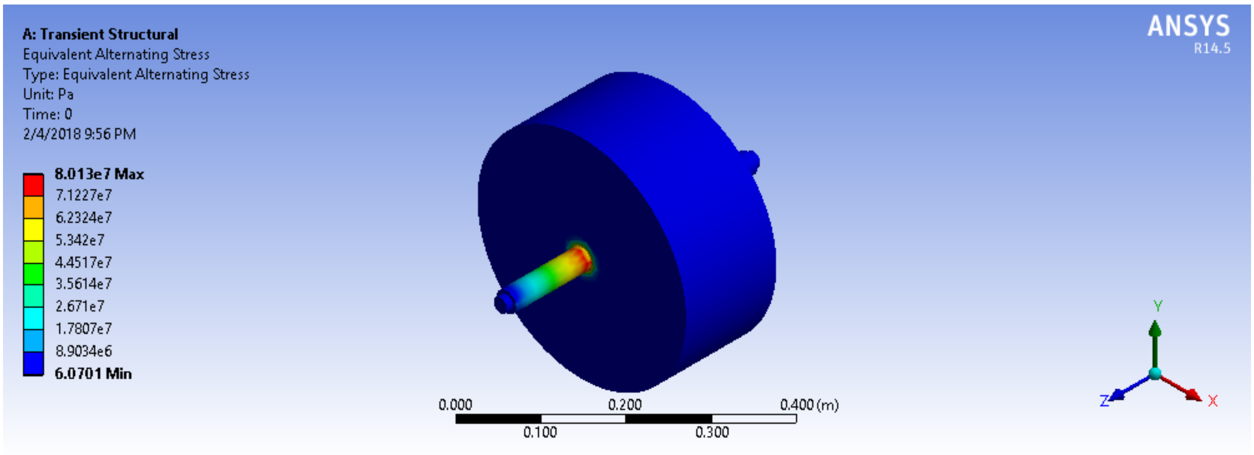

Figure 32 - Von Mises Stress

From Fig. 23, the maximum equivalent stress is seen near the point the wheel is attached with the shaft

4.2. EXPERIMENTAL RESULTS

The project was also tested in three parts:

1. Desiccant Wheel
2. Heat Exchanger
3. Direct Evaporative Cooler

4.2.1. Desiccant Wheel

Experiment was conducted on the desiccant wheel for three days to determine its performance on different conditions. The drop in the humidity of air is shown as below:

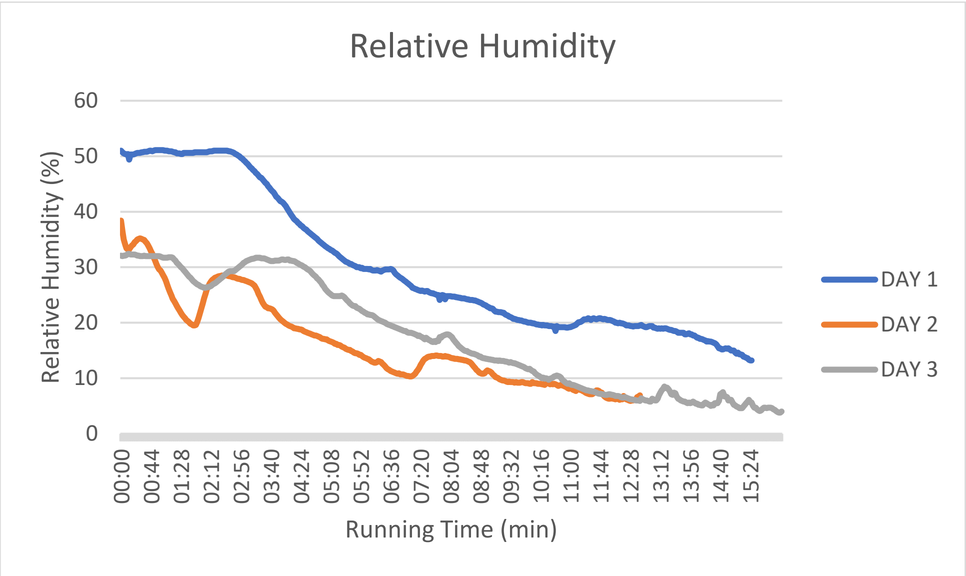

Figure 33 - Humidity drop

Absorbing the moisture from the air caused the temperature of the air to rise. The recorded rise in temperature for various condition is shown:

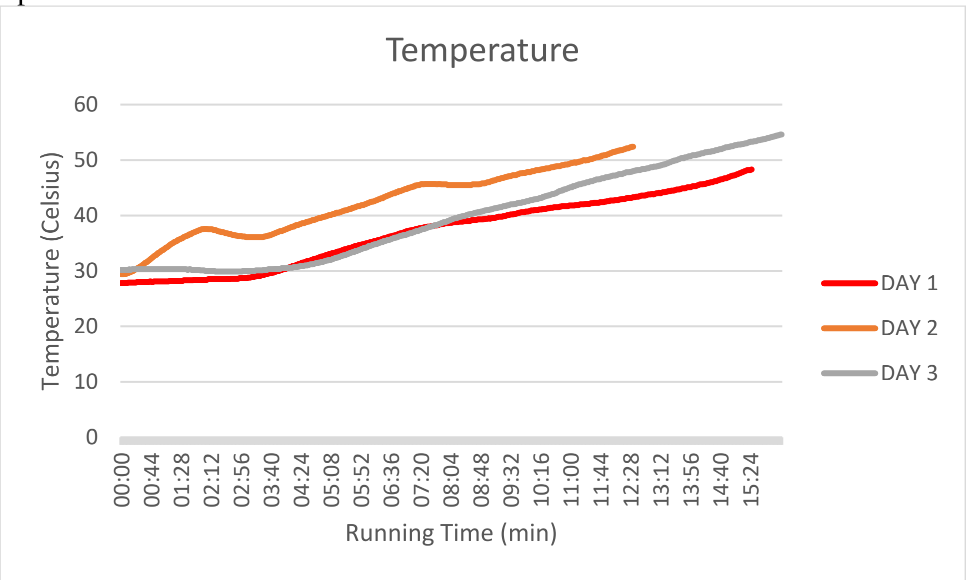

Figure 34 - Temperature Rise

4.2.2. Heat Exchanger

After passing through the desiccant wheel, the hot low humid air is passed through the indirect cooler to cool it down without adding humidity. The average temperature difference we achieved by our exchanger is shown:

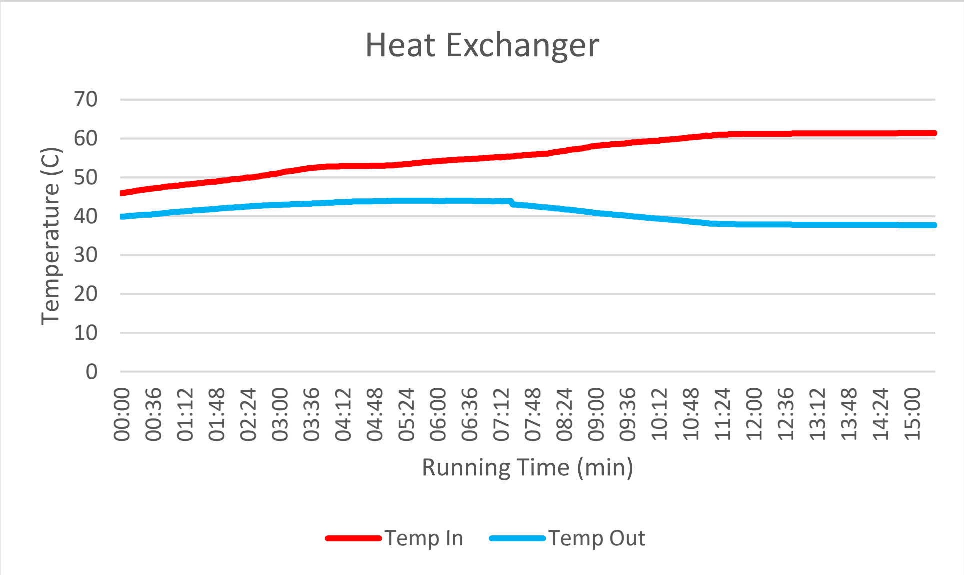

Figure 35 - Temperature Exchange

4.2.3. Direct Evaporative Cooler

In the final stage, the less hot low humid is passed through the humid evaporative pads to add moisture and cool down the air. The result is as shown:

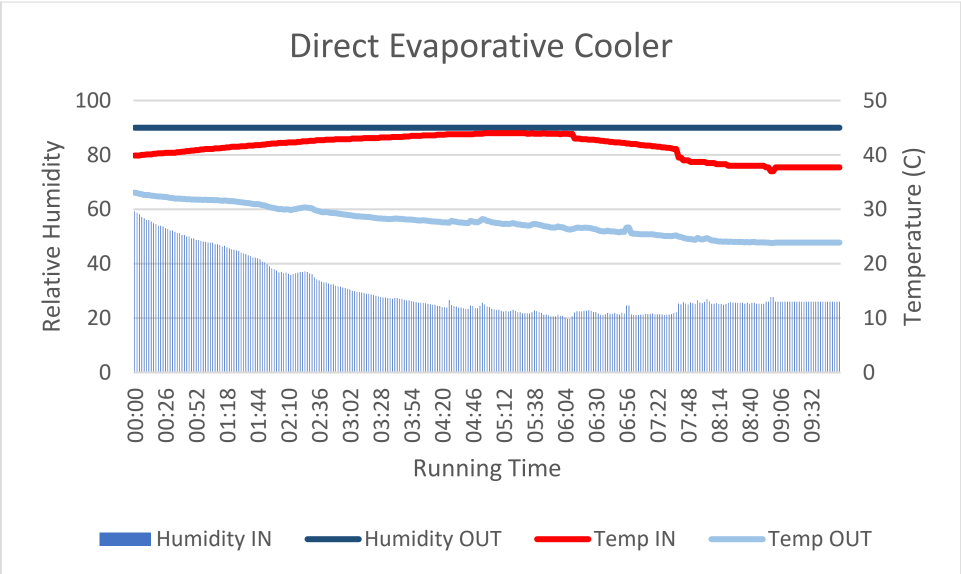

Figure 36 - Evaporative Cooler

CHAPTER 5: CONCLUSION AND RECOMMENDATIONS

5.1. THEORETICAL DISCUSSION

Using the TRNSYS and NOVEL Aire Simulator, simulations under different conditions and configurations were performed.

Daou et al. [16] said that the Evaporative Cooling alone is not that much significant in Area where the Humidity in air is already very high. Using the weather conditions of Karachi (a high humid region in Pakistan) a simulation was carried out using a single Evaporative Cooler. It was observed that the temperature didn't decrease much, it was seen to be higher than 30^{o} C in some times of the day (fig. 26). The reason is that the evaporative cooler cools air by adding moisture into it. The lower the moisture content in air, the higher tendency it has to cool using an evaporative cooler. This is the reason why Evaporative Cooler fails in high humid region.

Different cycles have already been studied for Desiccant Based Air Conditioning [13] [23]. The Dunkle cycle was said to be the most efficient cycle. Because of Dunkle cycle high complexity, regeneration and ventilation cycles are preferred. The recirculation cycle showed better results than the ventilation cycle (fig. 24 and fig. 25). Dehumidifying and cooling the same air required less energy to achieve a lower temperature, in ventilation cycle the process air is the ambient air which is every time at the same conditions (outdoor conditions). While in recirculation the conditioned space air is reconditioned. It is usually better to incorporate some ventilation with recirculation.

Sphaier and Nobrega [28] concluded that the overall performance of the system widely depends on the effectiveness of Heat Exchanger or Energy Recovery Wheel. Simulations were solved in which the same ventilation configuration was studied, once without the energy recover wheel and the other time with an energy recovery wheel but with decreased effectiveness. It was observed that the temperature in the conditioned space had increased (fig 27 and fig. 28). This is because, we know that when air gets dehumidified the temperature increase. Before entering into the evaporative cooler, it is needed that the dew point of air is kept low so that air can be cooled to maximum extent. Thus, it is mandatory to reduce the temperature of air before allowing it to go through the evaporative cooler, either using a heat exchanger, energy recovery wheel or an indirect evaporative cooler. The humidity shouldn't vary much in this process otherwise the evaporative cooler won't be able to reduce temperature effectively. So, we can conclude that this stage is very critical and requires a device with high effectiveness for sensible heat transfer. Another simulation was performed by directly using the outdoor air in the energy recovery wheel, without passing it through the evaporative cooler to let it cool down first before exchanging heat with process air. The conditioned air temperature came out to be higher. As stated earlier, the greater the decrease in temperature during this process, the better will be the results. Using air directly from outdoor will decrease temperature less than the air that go through the evaporative cooler first.

Riffat and Zhu [31] concluded that the performance of the system is good at a process air velocity of 0.6 m/s. On using velocity 0.6 m/s, it was observed that the fluctuation in temperature and humidity value has decreased. Thus, giving an approximately constant conditioned air temperature and humidity in the conditioned space throughout the day (fig 29).

5.2. EXPERIMENTAL DISCUSSION

Comparing both the theoretical and experimental results for Islamabad, we can see that the experimental results are quite similar to that of theoretical. It can be seen from the theoretical results that the output humidity of the desiccant wheel depends on the regenerative air temperature. During the experimental testing the regenerative temperature was kept constant at 110^{o}C and because the testing was done in the month of May, the humidity got reduced down to 5-6% (fig 33) and the temperature of air rose to around 60^{o}C (fig 34) . For the NOVEL Aire Simulation, the heater temperature could be varied, and it was observed that the optimum regeneration air temperature is 70^{o}C for the month of May, the relative humidity gets reduced to 7-10% and the outlet air temperature rose to 38-40^{o}C. The prototype was tested in different days, and the results obtained were slightly different from each other. It can be concluded that the performance of a desiccant wheel depends on the outdoor conditions of air. In order to get a specific conditioned air, the parameters of the desiccant wheel like the regeneration air temperature, mass flow rate of air and the RPH of the desiccant wheel will need to be adjusted. The Indirect Evaporative Cooler plays a vital role in the whole cycle. It is responsible to reduce the temperature of the air output from the desiccant wheel without altering the moisture content. Reducing the dry bulb temperature of the air will also reduce the wet bulb temperature, so the more temperature the indirect evaporative cooler decrease (fig 35), lower temperature can be achieved at the outlet of the direct evaporative cooler (fig 36).

5.3. FUTURE RECOMMENDATIONS

Following are the future recommendations:

1. Regeneration of the desiccant needs outside power source. It can be improved by using solar driven regeneration by using Fresnel lens to regenerate solid desiccant.
2. Zero Carryover liquid are introduced which can be used for the desiccant based evaporative cooling of the conditioned space. As Liquid Desiccant will require low Regeneration Temperature compared to the solid desiccant, so it’ll be more efficient and require less energy for operation.
3. Efficient Liquid Desiccant e.g. Lithium Chloride (LiCl) can be used with indirect dehumidifier integrated in the configuration which eliminates the carryover effect in the liquid desiccant.
4. There are three different cycles for the system i.e. Ventilation, Recirculation and Dunkle Cycle. Dunkle and Recirculation Cycle can be used in order to increase COP of the system.
5. A Control System can be developed to operate the system in different configuration according to the conditions of ambient air. For example in the month of May or June when desiccant isn’t necessary would be turned off by the system.

REFERENCES

[1] S Davanagere, A Sherif, Y. Goswami A feasibility study of a solar desiccant air-conditioning system**—Part II: Transient simulation and economics Int J Energy Res, 23 (2) (1999), pp. 103-11**6

[2] M. Mujahid Rafique, P. Gandhidasan, Shafiqur Rehman, Luai M. Al-Hadhrami, **A review on desiccant based evaporative cooling systems, Renewable and Sustainable Energy Reviews**

[3] Haddad K, Ouazia B, Barhoun H. **Simulation of a desiccant-evaporative cooling system for residential buildings. In: Third Canadian solar buildings conference, Fredericton, N.B.; 2008; p. 1–**8.

[4] Halliday SP, Beggs CB, Sleigh PA. **The use of solar desiccant cooling in the UK**: a feasibility study. Appl Therm Eng 2002;22(12):1327–38.

[5] Enteria N, Yoshino H, Mochida A, Takaki R, Satake A, Yoshie R, Mitamura T, Baba S. **Construction and initial operation of the combined solar thermal and desiccant cooling system**. Sol Energy 2009;83(8):1300–11.

[6] Enteria N, Yoshino H, Satake A, Mochida A, Takaki R, Yoshie R, Baba S. **Development and construction of the novel solar thermal desiccant cooling System incorporating hot water production**. Appl Energy 2010;87(2):478–86.

[7] Enteria N, Mizutani K. **The role of the thermally activated desiccant cooling technologies in the issue of energy and environment**. Renewable Sustainable Energy Rev.2011;15(4):2095–122.

[8] Collier, R., F. Arnold, and R. Barlow. **Overview of Open-Cycle Desiccant Cooling Systems and Materials.** No. NREL/TP-631-1065. National Renewable Energy Lab.(NREL), Golden, CO (United States), 1981.

[9] Ali, Muzaffar, Vladimir Vukovic, Nadeem Ahmed Sheikh, and Hafiz M. Ali. **"Performance investigation of solid desiccant evaporative cooling system configurations in different climatic zones."** Energy Conversion and Management 97 (2015): 323-339.

[10] Charoensupaya, Dhanes, and William M. Worek. "**Parametric study of an open-cycle adiabatic, solid, desiccant cooling system.**" Energy 13.9 (1988): 739-747.

[11] Pesaran, Ahmad A., Terry R. Penney, and Alvin Warren Czanderna. **Desiccant cooling: state-of-the-art assessment**. No. NREL/TP-254-4147. National Renewable Energy Lab., Golden, CO (United States), 1992.

[12] Shelpuk, Benjamin. "**The technical challenges for solid desiccant cooling.**" Heat Recovery Systems and CHP 13.4 (1993): 321-328.

[13] Jain, Sanjeev, P. L. Dhar, and S. C. Kaushik. "**Evaluation of solid-desiccant-based evaporative cooling cycles for typical hot and humid climates**." International Journal of Refrigeration18, no. 5 (1995): 287-296.

[14] Camargo, J. R., E. Godoy Jr, and C. D. Ebinuma. "**An evaporative and desiccant cooling system for air conditioning in humid climates.**" Journal of the Brazilian society of mechanical sciences and engineering 27.3 (2005): 243-247.

[15] Çarpinlioglu, Melda Özdinç, and Murtaza Yildirim. "**A methodology for the performance evaluation of an experimental desiccant cooling system.**" International communications in heat and mass transfer 32.10 (2005): 1400-1410.

[16] Daou, Kadoma, R. Z. Wang, and Z. Z. Xia. "**Desiccant cooling air conditioning: a review.**" Renewable and Sustainable Energy Reviews 10.2 (2006): 55-77.

[17] Ge TS, Li Y, Wang RZ, Dai YJ. **Experimental study on a two-stage rotary desiccant cooling system.** Int J Refrig 2009;32 498-08.

[18] Panaras G, Mathioulakis E, Belessiotis V, Kyriakis N. **Theoretical experimental investigation of the performance of a desiccant air-conditioning system**. Renew Energy 2010; 35:1368–75.

[19] Parmar H, Hindoliya DA. **Performance of solid desiccant based evaporative cooling system in warm and humid climatic zone of India**. Int J Eng Sci Tech 2010; 2:5504–8

[20] Heidarinejad G, Pasdarshahri H. **The effects of operational conditions of the desiccant wheel on the performance of desiccant cooling cycles**. Energy Build 2010; 42:2416–23.

[21] Mittal V, Khan BK. **Experimental investigation on desiccant air-conditioning system in India**. Front Energy Power Eng 2010; 4:161–5.

[22] Panaras G, Mathioulakis E, Belessiotis **V. Solid desiccant air-conditioning Systems-Design parameters**. Energy 2011;36 2399-06.

[23] Heidarinejad G, Pasdarshahri H. **Potential of a desiccant-evaporative cooling system performance in a multi-climate country**. Int J Refrig 2011; 34:1251– 61.

[24] Goldsworthy M, White S. **Optimization of a desiccant cooling system design with indirect evaporative cooler**. Int J Refrig 2011; 34:148–58.

[25] Panaras G, Mathioulakis E, Belessiotis **V. Proposal of a control strategy for desiccant air-conditioning systems**. Energy 2011; 36:5666–76.

[26] Nobrega CEL, Brum NCL. **A graphical procedure for desiccant cooling cycle design**. Energy 2011; 36:1564–70.

[27] Chung JD, Lee DY. **Contributions of system components and operating conditions to the performance of desiccant cooling systems**. Int J Refrig 2011; 34:922–7.

[28] Sphaier LA, Nobrega CEL. **Parametric analysis of component effectiveness on desiccant cooling system performance.** Energy 2012; 38:157–66

[29] Koronaki IP, Rogdakis E, Kakatsiou T. **Thermodynamic analysis of an open cycle solid desiccant cooling system using artificial neural network**. Energy Convers Manag 2012; 60:152–60.

[30] Uckan I, Yilmaz T, Hurdogan E, Buyukalaca O. **Experimental investigation of a novel configuration of desiccant based evaporative air conditioning system**. Energy Convers Manag 2013; 65:606–15.

[31] Riffat SB, Zhu J. **Mathematical model of indirect evaporative cooler using porous ceramic and heat pipe.** Appl Therm Eng 2004;24(4):457–70.

[32] Davis RA. **Evaluation of advanced evaporative cooler technologies; Pacific Gas and Electric (PG&E) customer energy management emerging technologies program**. In: Report no.: 491-04.7; 2004.

[33] Gandhidasan P. **A simplified model for air dehumidification with liquid desiccant**. Sol Energy 2004;76(4):409–16.

[34] Archibald J.**A new desiccant evaporative cooling cycle for solar air-conditioning and hot water heating.** In: Forum proceeding. American Solar Energy Society and American Institute of Architects; 2001. p.237–242.

[35] Dezfouli MMS, Mat S, Sopian K. **Comparison simulation between ventilation and recirculation of solar desiccant cooling system by TRNSYS in hot and humid area**. In: Proceedings of the seventh international conference on renewable energy sources, Kuala Lumpur, Malaysia, ISBN:978-1-61804-175-3; 2013.

[36] Katejanekarn T, Kumar S. **Performance of a solar-regenerated liquid desiccant ventilation pre-conditioning system**. Energy Build 2008;40(7):1252–67.

[37] Nelson JS, Beckman WA, Mitchell JW, Close DJ. **Simulations of the performance of open cycle desiccant systems using solar energy**. Sol Energy 1978;21(4):273–8.

[38] Pescod D. **A heat exchanger for energy saving in an air-conditioning plant.** ASHRAE Trans 1979;85(2):238–51.

[39] Suryawanshi SD, Tarup M. Chordia, Nikita Nenwani, Harish Bawaskar, Suraj Yambal. **Efficient technique of air-conditioning**. In: Proceedings of the world congress on engineering, vol. III WCE 2011, London, U.K.; 2011.

[40] Kim MH, Jeong JW. **Development of desiccant and evaporative cooling based 100% outdoor system**. AEI, American society of civil engineers; 2013. p. 506–515.

[41] Worek WM, Zheng W, Belding WA, Novosel D. **Simulation of advanced gasfired desiccant systems**. ASHRAE Trans 1991;97(2):609–14.

[42] K. Gommed, G. Grossman, **Experimental investigation of a liquid desiccant system for solar cooling and dehumidification**, Solar Energy, Volume 81, Issue 1,2007, Pages 131-138, ISSN 0038-092X

[43] Oliveira AC, Afonso CF, Riffat SB, Doherty PS. **Thermal performance of a novel air conditioning system using a liquid desiccant**. Appl Therm Eng 2010;20 (13):1213–23.

[44] Kessling W, Laevemann E, Kapfhammer C. **Energy storage for desiccant cooling systems component development.** Sol Energy 1998;64(4-6):209–21.

[45] Al-Sulaiman FA, Gandhidasan P, Zubair SM. **Liquid desiccant based two-stage**

evaporative cooling system using reverse osmosis (RO) process for regeneration. Appl Therm Eng 2007;27(14):2449–54.

[46] Radhwan AM, Elsayed MM, Gari HN. **Mathematical modeling of solar operated liquid desiccant-evaporative air conditioning system**. Eng Sci 1999;11(1):119–41.

[47] Dhar PL, Singh SK. **Studies on solid desiccant based hybrid air-conditioning**. Appl Thermal Eng 2001; 21:119–34.

[48] Henning HM, Erpenbeck T, Hindenburg C, Santamaria IS. **Potential of solar energy use in desiccant cooling cycles.** Int J Refrig 2001; 24:220–9.

[49] Dai YJ, Wang RZ, Zhang HF, Yu JD. **Use of desiccant cooling to improve the performance of vapor compression air conditioning.** Appl Thermal Eng 2001;21 1185–05.

[50] Subramanyam N, Maiya MP, Murthy SS. **Application of desiccant wheel to control humidity in air-conditioning systems.** Appl Thermal Eng 2004; 24:2777–88.

[51] Jia CX, Dai YJ, Wu JY, Wang RZ. **Analysis on hybrid desiccant air-conditioning system.** Appl Thermal Eng 2006;26 2393-00.

[52] Liu W, Lian Z, Radermacher R, Yao A. **Energy consumption analysis on a dedicated outdoor air system with rotary desiccant wheel**. Energy 2007; 32:1749–60.

[53] Sayegh MA, Hammad M, Faraa Z. **Comparison of two methods of improving dehumidification in air conditioning systems**: hybrid system (refrigeration cycle – rotary desiccant) and heat exchanger cycle. Energy Procedia 2011; 6:759–68

[54] Hong H, Feng G, Wang H. **Performance research of solar hybrid desiccant cooling systems**. Procedia Environ Sci 2012; 12:57–64.

[55] Ge TS, Li Y, Wang RZ, Dai YJ. **A review of the mathematical models for predicting rotary desiccant wheel**. Renewable and Sustainable Energy Reviews. 2008 Aug 1;12(6):1485-528.

[56] Jani DB, Mishra M, Sahoo PK. **Simulation of solar assisted solid desiccant cooling systems using TRNSYS**. Proceedings of International ISHMT-ASME Heat and Mass Transfer Conference December. 2013 Dec:28-31.

[57] Gujar RA, Bhaskar SV. **Shaft design under fatigue loading by using modified Goodman method**. International Journal of Engineering Research and Applications. 2013 Jul;3(4):1061-6.

[58] Zhai, Chaoqin. **Performance modeling of desiccant wheel design and operation**. Diss. Carnegie Mellon University, 2008.

APPENDIX I: DATA USED IN FDM

RPH	Revolution per hour	17 [2]
$\boldsymbol{d_e}$	Hydraulic Diameter (m)	2.082e-3 [58]
$\boldsymbol{\rho_a}$	Air Density (kg/m^3)	1.2041
$\boldsymbol{c_{pa}}$	Specific Heat of Air (J/kgK)	1.005
$\boldsymbol{k_a}$	Thermal Conductivity of Air (W/m.K)	0.0257 [58]
$\boldsymbol{h}$	Air Side Convection Heat Transfer Coefficient (W/m^2.K)	43.3 [58]
$\boldsymbol{\delta}$	Thickness of Desiccant (m)	6.5e-5 [58]
$\boldsymbol{\rho_d}$	Desiccant Density (kg/m^3)	700 [58]
$\boldsymbol{D_e}$	Desiccant effective Diffusivity (m^2/s)	0.036 [58]
$\boldsymbol{q_{st}}$	Heat of Sorption (J/kg adsorbate)	2791000 [58]
$\boldsymbol{c_{pv}}$	Specific Heat of Desiccant (J/kgK)	1000 [58]
$\boldsymbol{u}$	Air Stream Velocity (m/s)	16.70
$\boldsymbol{k_d}$	Thermal Conductivity of Desiccant (W/m.K)	0
$\boldsymbol{K_y}$	Gas side mass transfer coefficient (kg/m^2.s)	0.2757 [58]

APPENDIX II: DATA TO SOLVE HEAT EXCHANGER

Overall Heat Transfer Coefficient	120	$W/m^2.℃$
Specific Heat of Air	1.005	$kJ/kg.℃$
Specific Heat of Water	4.184	$kJ/kg.℃$
Effectiveness of HE	0.70	
Effectiveness of EC	0.90	

APPENDIX III: TRANSYS COMPONENTS

No.	Component	TRNSYS Name
1	Weather Data Reader	Type109-TMY2
2	Psychrometric	Type33e
3	Desiccant Wheel	Type1716a
4	Heat Exchanger	Type760
5	Evaporative Cooler	Type506a
6	Conditioned Space	Type690

APPENDIX IV: WHEEL SPECIFICATIONS

Mass	110 kg
Weight	1079.1 N
Diameter	770 mm
Width	200 mm
Material	Galvanized Steel
Power Required	9.20 W