SOLAR DESICCANT EVAPORATIVE COOLING

A Final Year Project Report

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ABSTRACT

Vapor Compression Cooling systems are creating a lot of problems for mankind including destruction of ozone layer and global warming due to emission of greenhouse gases. Secondly they have substantially high energy requirements and are also rendered inefficient in humid conditions. Need of the hour is for some cheap environment friendly alternative that can run on recyclable energy. Therefore a new design of desiccant based evaporative cooling is proposed which runs primarily on solar energy. The use of desiccant system would dehumidify the air and solar air heater would be used for regeneration of desiccant system. This dehumidified air can be effectively cooled using a setup of evaporative coolers which only use water and air as their working fluids. The overall objective is to develop and test the proposed design to find out the potential of solar desiccant based cooling. An important task of this project is to make the whole apparatus economically viable and to have all the components in simple arrangements for easier use by the general public. The widespread adoption of such a system will lead to a green environment friendly cooling system that has very low electrical input or can be run entirely on solar energy.

ACKNOWLEDGMENTS

We would first of all thank 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. Abdur Rehman throughout the project.

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.

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. Abdur 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.

ORIGINALITY REPORT

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PRIMARY SOURCES

1	Abhishek Saxena, Varun, A.A. El-Sebaii. "A thermodynamic review of solar air heaters", Renewable and Sustainable Energy Reviews, 2015 Publication	2%
2	B. K. Maheshwari, Rajendra Karwa, S. K. Gharai. "Performance Study of Solar Air Heater Having Absorber Plate with Half-Perforated Baffles", ISRN Renewable Energy, 2011 Publication	1%
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TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

ABBREVIATIONS

SAH	Solar air heater
PCM	Phase change material
DEC	Direct evaporative cooler
DW	Desiccant Wheel
AC	Air Conditioner
FDM	Finite Difference Method
RPH	Rotation per hour
CMH	Cubic meter hour
IDEC	Indirect evaporative cooler
EER	Energy efficiency ratio
DW	Desiccant Wheel
NTU	Number of Transfer Units
CAD	Computer aided design
COP	Co-efficient of performance

NOMENCLATURE

T	Temperature	K
A	Area	m^2
Q	Heat transfer	W
h	Heat transfer co-efficient	W/m^2-K
k	Thermal conductivity	W/m.K
V	Speed	m/s
Nu	Nusselt Number	
Ra	Rayleigh Number	
Pr	Prandtl Number	
d	Thickness	m
C_p	Specific heat	J/kg-K
I	Solar irradiance	W/m^2
$\dot{m}$	Mass flow rate	kg/s
U	Overall loss co-efficient	W/m^2-K
g	Gravitational acceleration	m/s^2
Y_a	Absolute humidity ratio of air stream	
Y_d	Humidity ratio in equilibrium with desiccant	
D_e	Desiccant effective diffusivity	m^2/s
K_y	Gas-side mass transfer coefficient	Kg/m^2s
W	Desiccant adsorption mass	kg
X	Wheel thickness	m
z	Axial displacement through matrix	m

Greek letters

σ	Stephen Boltzmann constant	W/m^2-K^4
ε	Emissivity	
υ	Kinematic viscosity	m^2/s
τ	Transmissivity	
α	Absorptivity	
μ	Dynamic viscosity	kg/m.s
ρ	Density	Kg/m^3
η	Efficiency	

Subscripts

p	Absorber plate
g	Glass cover
i	Insulation
w	Wind
a	Ambient
s	Sky
m	Mean

CHAPTER 1: INTRODUCTION

Conventional vapor compression air-conditioning systems can provide comfortable environment only when sensible heat ratio is greater than 75%. When sensible heat ration drops below 75% (which means relative humidity increases), conventional vapor compression air-conditioning systems require more electrical energy, emit more CO2 and their efficiency reduces.

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

The building cooling load can be divided into sensible and latent load; conventional AC systems cannot effectively control the latent load. To remove the moisture, it cools the air below its dew point and then reheats it to room level resulting in a lot of wastage of energy. The conditions where latent load is overwhelming these two procedures for example overcooling and afterward reheating will enhance the utilization of electrical energy and outflow of CO2 remarkably.

These issues of regular Air Conditioners can be solved by utilizing an innovation called desiccant based evaporative cooling. This innovation is a mix of desiccant dehumidifier and evaporative coolers. The only power utilized in this framework is to drive the fans, water pumps and to recover the desiccant dehumidifier amid the recovery procedure.

Desiccant based cooling systems handle both the loads separately thus effectively controlling the whole process. Desiccant cooling systems are energy efficient and cost effective. Water and air are used as cooling medium which are totally environment friendly.

The problem with desiccant based cooling systems is their increased pressure drops and requirement of high regeneration temperatures for their continuous functioning. This consequently increases energy requirement.

Need of the hour is to bring design improvements to reduce the regeneration temperature requirements so that any low grade energy can be used to accomplish the deed. Using solar energy to regenerate the desiccant seems the most practical and cheap method.

CHAPTER 2: LITERATURE REVIEW

On the basis of working cycles Desiccant based air conditioning systems can be divided into three basic types [1]. These cycles vary from each other in the aspect that whether fresh air is supplied to the conditioned space or not.

Ventilation Cycle

In ventilation cooling cycle return air from the room is not circulated in the system, instead the all the supply air consists of fresh ambient air. The process starts from desiccant wheel; ambient air enters the desiccant wheel where moisture is removed so its temperature raises. Next this hot and dry air passes through a finned tube heat exchanger where its temperature is reduced without affecting the humidity level. Finally this supply air is passed through direct evaporative cooler where its temperature is lowered and water content is increased so as to bring it to the human comfort level. For the regeneration of desiccant wheel exhaust air from the room is heated and then passed through the desiccant wheel, instead of exhaust air from the room ambient air can also be used for this purpose. COP and specific cooling capacity of such system is low due to the high temperature and humidity value of ambient air as compared to return air from the room.

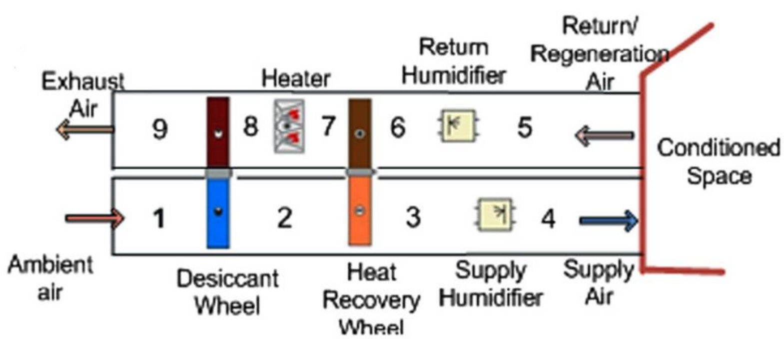

Figure 1 Schematic of Ventilation Cycle [1]

Recirculation Cycle

This is a modified form of ventilation cycle. In recirculation cycle supply air to the room consists solely of return air from the room. Return air is passed through desiccant wheel to decrease its humidity. In the process of dehumidification temperature of air rises, this hot air is passed through finned tube heat exchanger to lower its temperature. This reduction of temperature is done without increasing the specific humidity of air. To bring the air temperature and humidity level to human comfort level, air is passed through direct evaporative cooler. Regeneration of desiccant wheel is done using ambient air. COP and specific cooling capacity of this system is higher than the ventilation cycle due to lesser requirement of temperature and humidity reduction. The use of this cycle is limited due to absence of fresh air, however fresh air can be mixed with the return air to cater for the human needs.

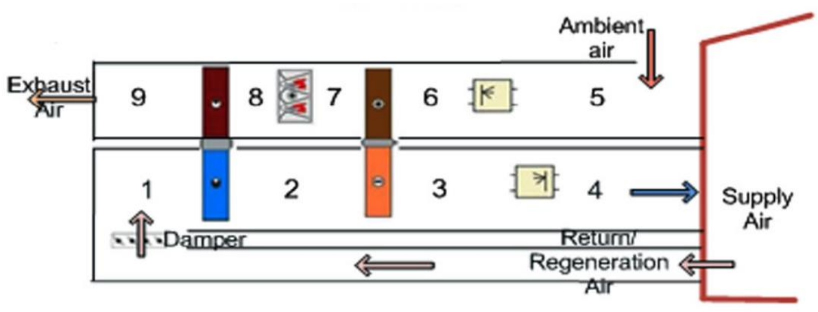

Figure 2 Schematic of Recirculation Cycle [1]

Dunkle Cycle

This cycle employs the best features of both ventilation cycle and recirculation cycle. An additional heat exchanger is employed to take advantage of both low temperature and higher cooling capacity of recirculation mode and ventilation mode. Dunkle cycle is also limited in use due to absence of fresh air. Ventilated Dunkle cycle is used to keep the system efficient as well as useable.

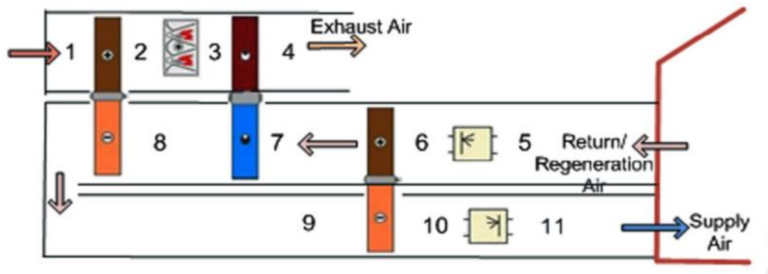

Figure 3 Schematic of Dunkle Cycle [1]

Desiccant Wheel

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. Regeneration can be done effectively using solar energy.

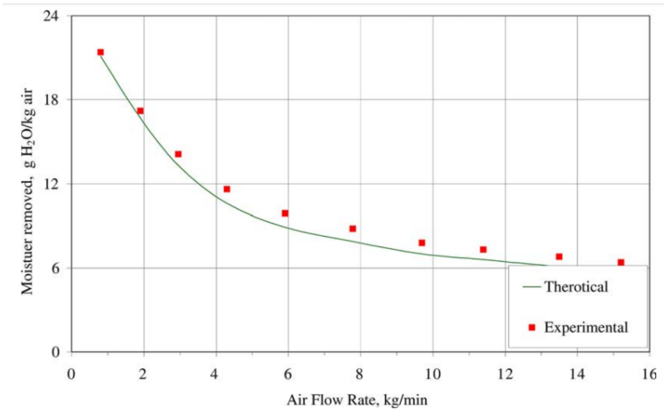

Figure 4 Graph relating mass flow rate of air to moisture removal capacity [2]

The figure 4 [2] relates air flow rate to moisture adsorption in process air side. Increasing the air flow rate reduces the moisture absorbed.

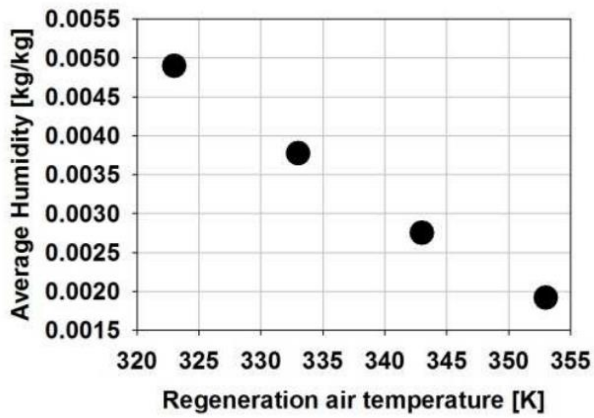

Figure 5 Graph showing effect of regeneration air temperature on outlet humidity [3]

The figure 5 [3] relates regeneration temperature with average humidity of process air after passing through desiccant wheel. For higher regeneration temperatures, the humidity is better controlled.

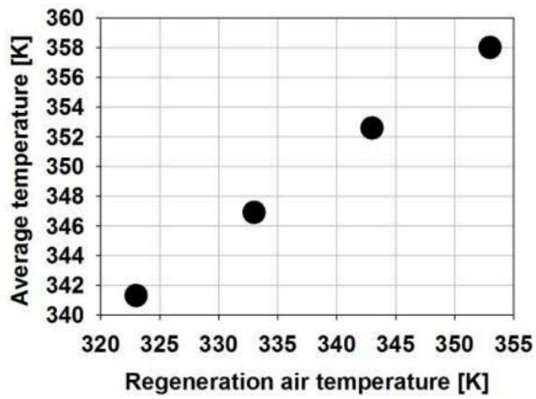

Figure 6 Graph showing effect of regeneration air temperature on outlet temperature [3]

The figure 6 [3] relates regeneration temperature with average exit temperature of process air. Higher regeneration temperature results in higher temperature of dry air at desiccant wheel exit.

Desiccant Material

Different desiccant material has different tendency to absorb the moisture [4]. 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 1 Comparison between Solid and Liquid Desiccant

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

The figure 7 [4] shows dehumidification performance of three different desiccants i.e. silica gel, polymer and zeolite with different inlet relative humidity at 50° C and 80° C regeneration temperature respectively.

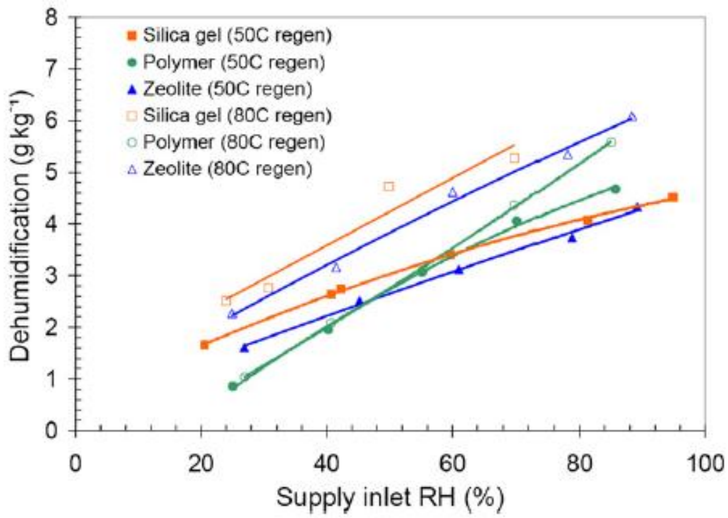

Figure 7 Graph showing effect of inlet RH on dehumidification capacity of various desiccants at different temperatures [4]

At lower regeneration temperature, the performance of silica and polymer are comparable whereas at higher regeneration temperatures, silica outperforms polymer and zeolite in dehumidification. Moreover, silica has consistent performance at various inlet velocities.

Hence silica is selected as desiccant material for following reasons:

- Best dehumidification at higher regeneration temperatures
- Cheap and readily available
- Consistent performance at different inlet velocities

For achieving the cooling of the dehumidified air after it exits the desiccant wheel. It is passed through a radiator and then a direct evaporative cooler.

Heat Exchanger

After passing through desiccant wheel temperature of air becomes quite high. As the cooling capacity of direct evaporative cooler is limited by the wet bulb temperature of supply air stream, this supply air needs to be cooled before entering DEC. this reduction in wet bulb temperature is done by using a finned tube heat exchanger. Water flows inside the tubes while air is passed over the finned tubes, thus transferring heat from hot air stream to the cold water flowing inside tubes.

Various researchers have undertaken the study of effects of different indirect evaporative coolers on the overall performance of desiccant based cooling system.

Pandelidis et al. [5] did the comparison of air conditioning systems with different evaporative cooler. They studied three desiccant air-conditioning systems with different evaporative coolers: (1) the cross flow Maisotsenko cycle heat and mass exchanger, (2) the regenerative counter flow Maisotsenko cycle heat and mass exchanger and (3) the standard cross cross flow evaporative air cooler. They concluded that system with cross flow M-Cycle is the most complicated solution but it allows obtaining the lowest supply air temperatures and largest cooling capacities, system with standard cross flow evaporative cooler is the cheapest and simplest but it can only be applied to the places with low latent load requirement.

Farmahini-Farahani and Heidarinejad [6] studied the improvement in the effectiveness of evaporative cooling by pre-cooling using nocturnally cooled water. They used a cooling coil unit to pre cool the air entering the Direct-Indirect evaporative cooler. By using the cooling coil unit they achieved a temperature reduction of more than 10^o C. They also found that effectiveness of the system increases by an average value of 9% as compared to a Direct-Indirect evaporative cooler working solely.

Al-Juwayhel et al. [7] studied the performance of one, two and three stage evaporative coolers. There one stage system consisted of an IDEC or a DEC, two stage system was a combination of IDEC and DEC and three stage system was combination of IDEC, DEC and mechanical vapor compression system. Same water was used for both IDEC and DEC. It was concluded that two stage system gives best energy efficiency ratio EER. Highest effectiveness was achieved by three stage system while single stage DEC was least effective.

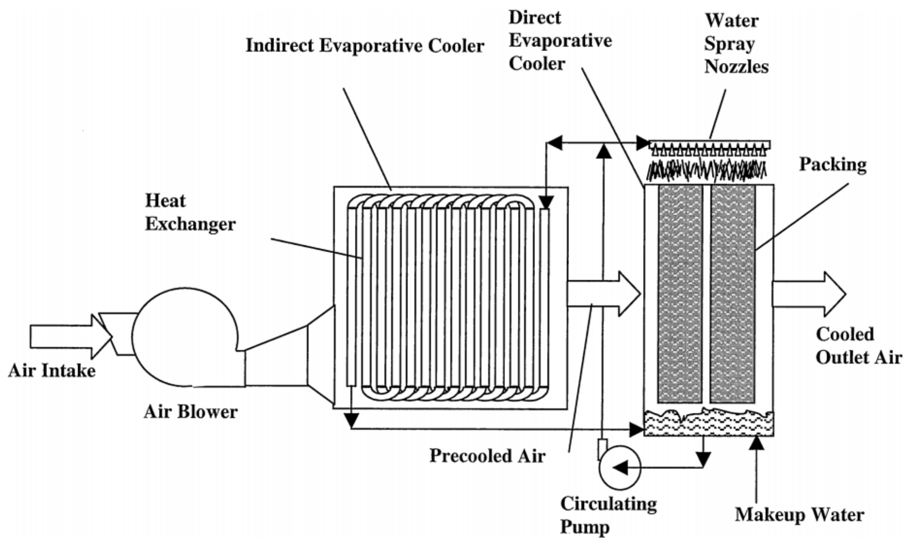

Figure 8 Schematic of combination of indirect and direct evaporative coolers [7]

Direct Evaporative Cooler

For the evaporation of 1 kilogram of water into the general atmosphere, energy of 680W is required. The energy requirement is fulfilled by taking energy form the heat of the air, which leads to lowering of the temperature. This leads to the cooling of 680W for evaporation of each 1kg of water. [8]

In a direct evaporative cooler, the outside air is passed through a water saturated medium, which in turn cools down the air due to effects of evaporation. [9]

As the air passed through the wet medium, the sensible heat is lost and in turn latent heat is gained.

- It's important to note that that web bulb temperature will remain the same while the dry bulb temperature will be reduced.
- We know that total enthalpy of the air before and after passing the wet medium remains the same. There is just an addition of moisture and a reduction of the sensible heat

Humidity of the incoming air is raised as it passes through the wet and in turn lo it’s lowering its temperature. Room is supplied this treated air and to maintain lower levels of humidity the already present air is mostly exhausted rather than recirculated.

This cooling effect is fully felt by the target area and this is a great advantage of fitting the evaporative cooler in a duct. The parameters that are temperature and humidity of the supply air are a huge factor in determining how much cooling can be achieved using this method, this this dependability is a downside to this strategy. Air that has a high level of humidity already will not be able to absorb much moisture, so will not be subject to the cooling effect of the humidifier. Already humid/ saturated air has a lower capability of absorbing moisture thus humidifier’s cooling effect has a lower effect on it. [8]

In the areas and rooms that lack a conventional air conditioning system (central), direct air evaporative cooling can be used a viable alternative to reduce the temperature economically. Water is added to the air after it passed through a nozzle which converts it into very fine droplets. These nozzles are usually coupled with large fans. By placing these coolers in strategic locations open air areas can be cooled at fraction of the cost of a traditional direct expansion cooler. [8]

Types of evaporative coolers based on the media used

The thickness and the material of the evaporative media, the inlet conditions of the air and its flow rate dictate the performance of DEC coolers.

A small pump is used in direct evaporative coolers to spread water over the evaporative medium. The power requirement/ horsepower for such a pump are minimal. Only part of the surface is to be wetted as the reset of the water spreading is carried out by gravity and capillary action.

Cellulose evaporative media like CELdek and GLASdek have a higher efficiency than Aspen pads generally available. Aspen pads have a short life span and have to be replaced after every season.

To achieve a greater surface area per m3 of media the use of rigid media instead of Aspen pads is preferred. The rigid media GLCIER-COR provides 400 m^2 of evaporative surface area per 1 m^3 volume of media. As the media is rigid and does not sag it provides consistent performance. In different applications pads ranging from 2inch to 24inches are used. The manufacturers of these media rate their effectiveness at 75% to 95%. The effectiveness depends on the thickness of the cooling pad and the inlet flow rate of air. This media can last between 7-10 years with regular maintenance.

System type	Evaporative media	Effectiveness	Features
Random media	Excelsior or plastic fiber/foam supported by plastic frame.	>80%	Low effectiveness Short life-time. Hard to clean.
Rigid media	Blocks of corrugated materials: Cellulose, plastic, fiberglass.	75-95%	High initial cost. Longer life-time. Cleaner air.
Remote pad	Random or rigid Pads mounted on wall or roof of building	75-95%	Higher power consumption Bacteria growth

Figure 9 Features of different cooling pads [9]

Some of the studies regarding IDEC cooling and DEC cooling are mentioned below.

Hindoliya and Parmar [10] studied the prospects of evaporative cooling system based of desiccant dehumidification in hot and humid environment. The COP of the system was concluded to being highly dependent on the humidity of an area when calculated at different environmental conditions. As the humidity increases the COP reduces.

Davis RA [11] concluded that for similar cooling load, a greater mass flow rate is needed by the evaporative cooler as compared to the conventional air conditioning system. This is due to the minor enthalpy difference. In evaporative coolers the humidity of the supply air is greater than in conventional ACs.

Cross flow indirect evaporative cooler was utilized by Pescod [12] in the desiccant system. The system was designed in a wat that the water was flowing in the wet channel while the air in the secondary dry channel.

Suryawanshi et al. [13] reached a conclusion that a two stage evaporative cooler is nearly 4.5 times more effective than a conventional air conditioner in hot a dry climate.

Kim and Jeong [14] concluded that by utilizing both DEC and IEC in the cooling system, 74-77% of the energy is saved when it is compared the generally available conventional cooling apparatuses.

Solar Regeneration

Jani and Manish [15] used solar collector which transfers heat to working fluid using a parallel flow heat exchanger and that heated fluid is used to heat up air for regeneration of desiccant wheel. The working fluid is a 50% mixture of propylene glycol and water. The assembly can be seen in Fig. 10.

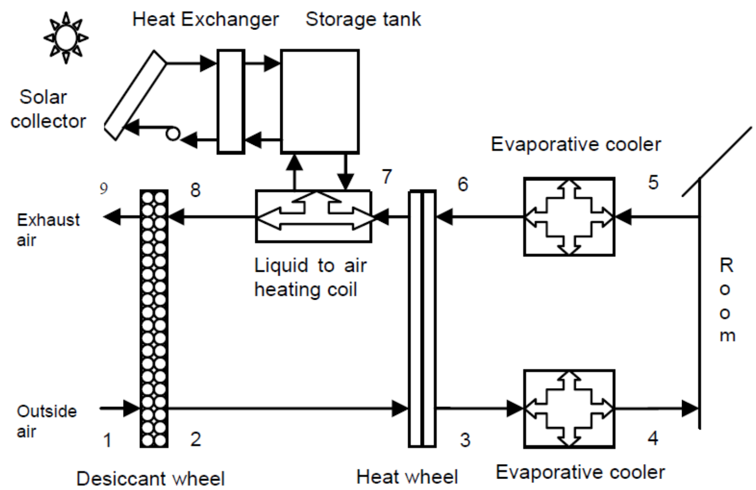

Figure 10 Using Solar Collector for Regeneration of Desiccant [15]

Solar energy may be utilized for regeneration of desiccant wheel. A solar air heater [16] can supply the hot air at regeneration temperature to desiccant wheel. The basic configuration of solar assisted desiccant cooling system is depicted in Fig. 11.

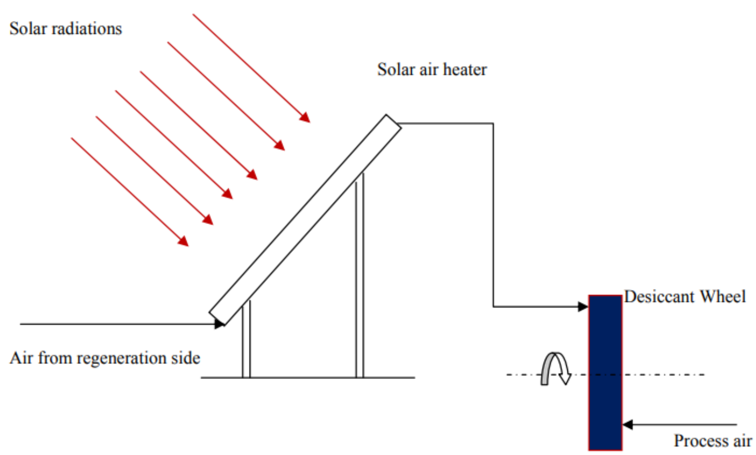

Figure 11 Using Solar Air Heater for Regeneration of Desiccant [16]

Solar air heaters have lower production, running and maintenance costs as compared to solar collectors and have lower losses because solar radiations directly heat up air whereas in case of solar collector, solar radiations warm up working fluid which in turn is used to warm up the air. Solar collectors although come with advantage of thermal storage. Solar air heaters have numerous preferences over liquid heaters regarding the issues of corrosion, boiling, freezing and leaks.

Amit and Avinash [17] used a point concentrator type solar collector, which focuses all the direct and diffuse solar rays that strike on the spherical dish of the reflector to the compact region of the collector. The temperature of the absorber is elevated up to 500°C. This elevated temperature of the absorber is due its black surface and it is appropriate to regenerate the desiccant material without utilization of any high grade power. The desiccant is regenerated only due to the elevated temperature without the passage of hot air through them. The solid desiccant is used to dehumidify humid air.

Neeraj [18] used evacuated solar tube with PCM to increase the temperature of air to be used as regeneration air.

The evacuated tube assembled up of two co-hub glass tubes, one is an external glass tube and the second one is an absorber tube. The water passes through the annular clearance between two rectangular boxes while the annular clearance between the small rectangular box and round tube is packed with Acetamide (PCM). The air is made to pass through the middle hollow round tube.

Solar irradiance heats up the filled water inside evacuated tube. This heat energy is transmitted to the PCM. It starts to accumulate the heat and simultaneously transmits the heat to the air passage through the hollow round tube during the day. Air is heated by the stored energy of PCM at night. This hot air is utilized to regenerate the desiccant wheel. The assembly is shown in Fig. 12.

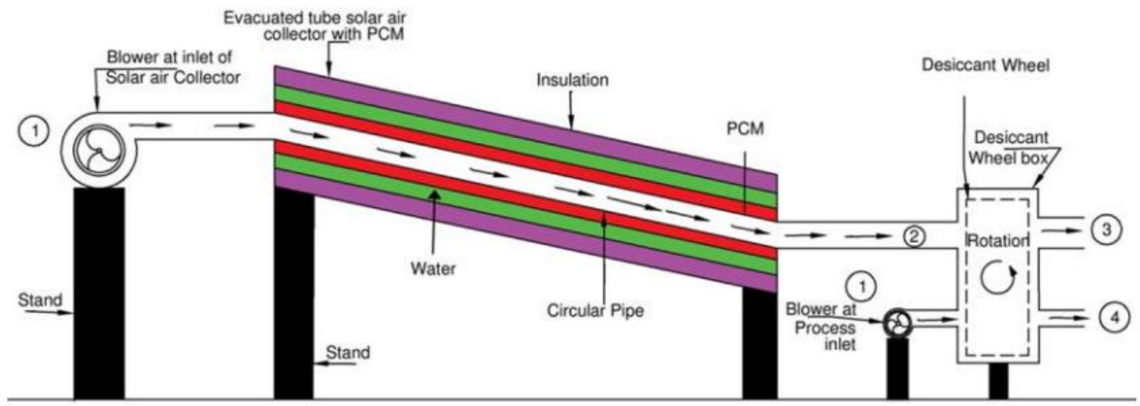

Figure 12 Using Evacuated Solar Tube with PCM for regeneration of Desiccant [18]

Simple Flat Plate Solar Air heater

Gargetal. [19] explored the outcome of the enhanced heat transfer area on a typical SAH. Rectangular fins were used for this purpose. Performance curves were established for different collector arrangements, and the efficiency was noted to be higher for a larger number of fins internal to the duct.

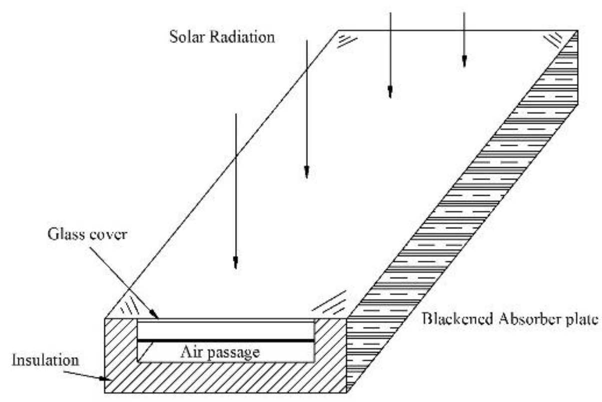

Figure 13 Schematic of a simple Solar Air Heater [19]

Sorour and Mottaleb [20] investigated about corrugated, channel-type SAH and its working parameters. These design variables (i.e., different cross sections) were examined for varying flow rates, numbers of glass covers and the T_{in}. It was inferred that a corrugated absorber plate enhances the efficiency and double glass coating gives leading performances for all the flow configurations and corrugated absorber plates.

Gargetal. [21] formulated a theory for a four-pass SAH and evaluated performance for different passes (i.e., one pass to four-pass).The effect of number of glass covers on one to multi-pass was explored in a detailed manner. Low mass flow rates and large plate lengths give good results with two or three pass SAH. The performance of two or three pass SAHs showed not much change with inclusion of glass covers for smaller plate lengths and higher mass flow rates.

Bhargavaetal. [22] redesigned a typical SAH in which two metal plates and one glass cover were utilized, the air was forced in the bottom channel, while the stagnant air between the plate and glass cover prevents heat loss. The efficiency of the top channel was observed to be quite high, whereas the decrease in the bottom channel efficiency was not that much. It was depicted that, if the top channel can be kept entirely open, the efficiency loss in the bottom channel due to the air passage in the top channel was under 5%.

CHAPTER 3: METHODOLOGY

Desiccant Wheel

Desiccant wheels have honeycomb (matrix) structure. This matrix contains the dehumidifying (desiccant) agent and supporting material. The matrix forms numerous channels for air flow where the desiccant absorbs moistures. The cross section of desiccant wheel is divided into portions; one is process air side and other is regeneration air side. Adsorption i.e. removal of moisture from air occurs in the process air side whereas desorption i.e. removal of moisture from desiccant occurs in the regeneration air side.

In order to establish a proper mathematical model for desiccant wheel, following assumptions are made [23], [24]:

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.

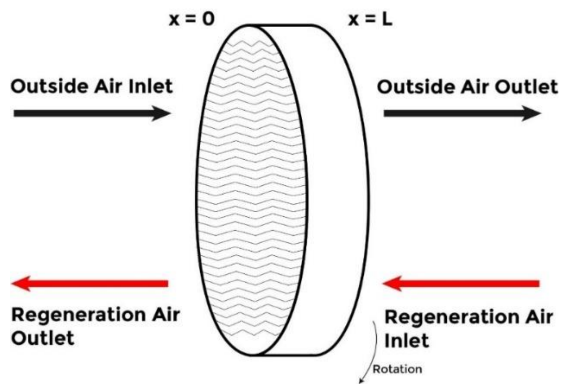

Figure 14 Simple model of Desiccant Wheel [25]

Governing Equation

There are four main governing equations, the mass and energy conservation equation for both the desiccant material and air [25]. 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.

Conservation of Moisture in Air

$$\boldsymbol{d_e \rho_a \left(\frac{\partial Y_a}{\partial t} + u\frac{\partial Y_a}{\partial z}\right) = K_y (Y_d - Y_a)} \quad (1)$$

The main term on the left-hand side is the moisture capacity term. The second term on the left-hand side speaks to the rate of dampness variation in the direction of the air flow. The primary term on the right-hand side communicates the rate of dampness variation brought about by the convective mass exchange.

Conservation of Energy in Air

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

The main term on the left-hand side represents energy storage in the damped air. The second term on the left-hand side speaks to the rate of energy variation in the air in the axial direction.

The third term on the left-hand side is the heat conduction term in the air. The main term on the right-hand side communicates the convective heat transfer between the desiccant and air. The second term on the right-hand side indicates the sensible heat transfer between the desiccant and air.

Conservation of Moisture in Desiccant

$$\boldsymbol{\delta\rho_d\left(\frac{\partial W}{\partial t}+D_e\frac{\partial^2 W}{\partial z^2}\right)=K_y(Y_d-Y_a)} \tag{3}$$

The main term on the left-hand side is the moisture capacity term for the desiccant. The second term on the left-hand side represents mass diffusion term in axial direction within the solid desiccant. The main term on the right-hand side represents the convective mass transfer between the desiccant and air.

Conservation of Energy in Desiccant

$$\boldsymbol{\delta c_{pd}\rho_d\left(\frac{\partial T_d}{\partial t}-\frac{k_d}{c_{pd}\rho_d}\frac{\partial^2 T_d}{\partial z^2}\right)}$$

$$\boldsymbol{=h(T_a-T_d)+}q_{st}\boldsymbol{K_y(Y_a-Y_d)+}c_{pv}\boldsymbol{K_y(Y_a-Y_d)(T_a-T_d)} \tag{4}$$

The main term on the left-hand side is the energy capacity term of desiccant. The second speaks to the transfer of heat by heat conduction within desiccant. The main term on the right-hand side determines the convective heat transfer between the air and desiccant. The second term on the right-hand side communicates the effect of heat due to adsorption. The third term on the right-hand side determines the sensible heat transfer between the air and desiccant.

Mathematical Solution

The governing equations were solved using finite difference method (FDM). Due to nonlinearity and complexity of the above model, it is difficult to obtain analytical solution. Hence an alternate approach was used.

Alternate Approach

The honeycomb (matrix) structure results in high pressure drop across the desiccant wheel. A novel design of desiccant wheel with radial blades showed good results in experimentation done in paper [26], [27] with insignificant pressure drop across the desiccant wheel. The novel design consisted of radial blades originating from center to the circumference of desiccant wheel instead of the honeycomb (matrix) structure. The experimental results of this design were compared with the results of simulator created by NOVEL Aire Technology to solve the honeycomb structured desiccant wheel. The simulator interpolates the given conditions and displays the outlet conditions.

The experimental inlet conditions were:

Table 2 Experiment Inlet Conditions

Air Volume Flow Rate	180 CMH
Rotation Speed	35 RPH
Regeneration Temperature	66 ℃

The process and regeneration air sides were 50% percent of the desiccant wheel. The desiccant material used was silica gel particles. The results obtained from experimentation were as follows.

Table 3 Temperature and humidity results before and after desiccant wheel

Inlet Air Channel			
Before Wheel		After Wheel	
Humidity [%rH]	Temp. [°C]	Humidity [%rH]	Temp. [°C]
41.70	26.70	26.20	28.10
42.33	26.57	26.05	28.09
42.02	26.00	26.51	28.66
79.77	23.48	29.45	29.20
81.78	22.45	29.15	29.65
82.47	21.92	28.49	29.83
83.43	21.37	27.84	30.00
89.13	20.80	27.65	30.11
93.35	20.40	27.82	30.00
93.54	20.33	28.71	30.09
91.28	20.11	27.46	30.37

The pressure drop across desiccant wheel was found to be 2 Pa [27].

To compare the results with the results of the simulator created by NOVEL Aire Technology, similar conditions of volume flow rate, rotation speed and regeneration temperature were given to the simulator for desiccant wheel model: WSG 250x200 which uses silica gel as desiccant material.

At 23^{o} C dry bulb temp and 80% relative humidity of inlet process air, the results of the simulator showed 30^{o} C rise in temperature and moisture removal of about 10 g/kg of dry air. The pressure drop across desiccant wheel is found to be around 200 Pa. The results are shown in figure.

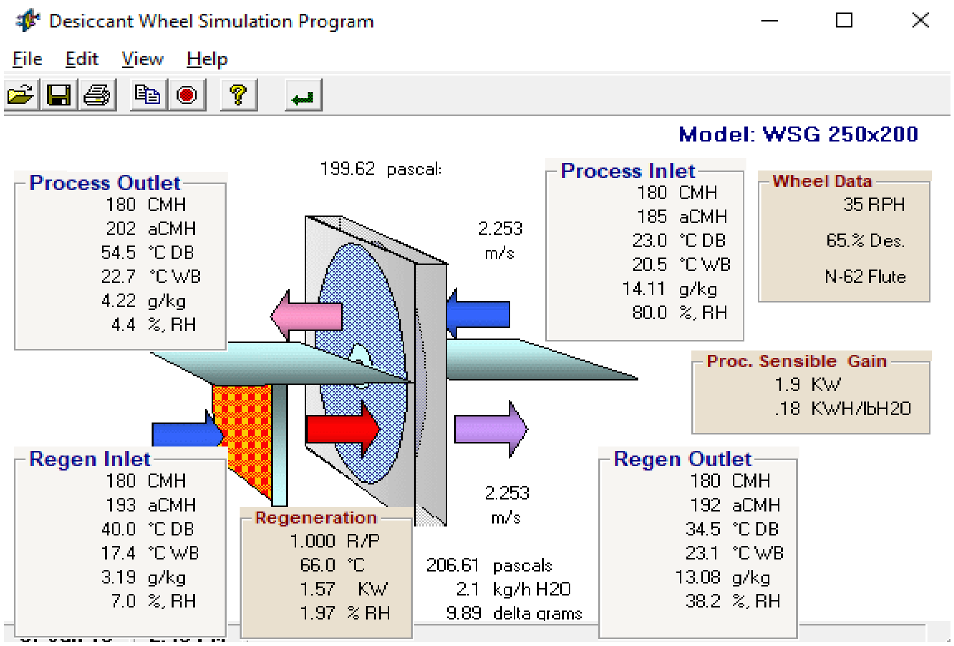

Figure 15 Input conditions on NOVEL Aire Program

Comparing it with the experimental results in table, the moisture removal in the experiment with the radial blades desiccant wheel was 8g/kg of dry air at 23^{o} C dry bulb temp and 80% relative humidity of inlet process air, which is comparable to that provided by the simulator. Moreover, the pressure drop across the radial blades desiccant wheel is 2 Pa, which is significantly very low compared to the simulator result. This also reduces pumping power requirement for the system.

Based on above comparison approach, the novel design of desiccant wheel with radial blades is designed for our solar desiccant evaporative cooling.

Direct Evaporative Cooler

$$Nu = 0.10 \left(\frac{l_e}{l}\right)^{0.12} Re^{0.8} Pr^{0.333} \tag{5}$$

$$l_e = V/A \tag{6}$$

The value of l_e is given by the manufacturer, for instance the media (used in evaporative cooler) GLACIER-COR provides 400 m^2 of evaporative surface area per 1 m^3 volume of media.

Hence, l_e= 1/400 = 0.0025m

$$\boldsymbol{h_s = N_u k / l_e} \tag{7}$$

$$\frac{\mathbf{T_1 - T_2}}{\mathbf{T_1 - T_s}} = \mathbf{1 - exp}\left(-\frac{\boldsymbol{h_s A}}{\mathbf{m_a}\boldsymbol{c_{pu}}}\right) \tag{8}$$

In the above equation both sides are equal to the effectiveness and it is evident that for a higher effectiveness, following are the governing parameters. Large area of heat transfer that is dictated by the medium of evaporation used, a high convective heat transfer coefficient and low inlet flow rate or more generally lower mass flow rate.

Solar Air Heater

Solar radiations fall on the glass cover but not all energy is transferred ahead. Only the energy transmitted through glass cover is useful energy, while the energy absorbed or reflected is wasted.

$$\text{Transmissivity} + \text{Absorptivity} + \text{Reflectivity} = 1$$

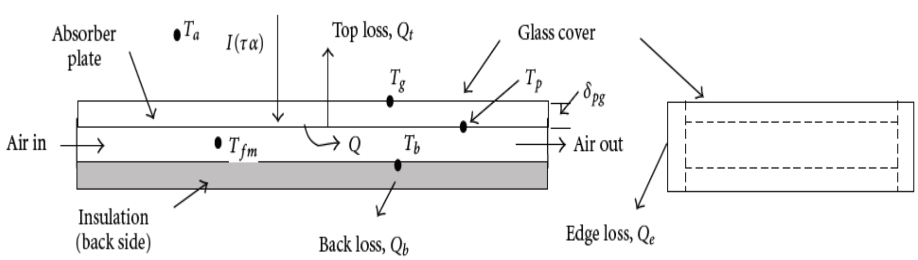

Figure 16 Losses of a conventional SAH [22]

So a glass cover of high transmissivity is required to cut down on the losses. This transmitted energy is passed down to the absorber plate. Here only the absorbed energy is useful while the energy transmitted or reflected is wasted.

$$Input\ Energy = \text{Transmissivity of glass cover} \times \text{ Absorptivity of absorber plate } \times \text{ Solar Radiation}$$

The overall heat loss from a solar air heater is the sum of heat loss from top, back and edge of collector.

$$Output\ Energy = Input\ Energy - Losses$$

Losses

Top Losses

Heat transfer from the absorber plate at mean temperature Tp to the internal surface of the glass cover at temperature Tgi takes place by radiation and convection.

$$Q_{tpg} = A[\sigma\left(T^4{}_p - T^4{}_{gi}\right)\left(\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_g} - 1\right)^{-1} + h_{pg}(T_p - T_{gi})] \tag{9}$$

$$T_{gi}\ is\ glass\ inner\ temperature$$

Heat transfer through glass cover of thickness d_g by conduction is

$$Q_{tg} = \frac{k_g A(T_{gi} - T_{go})}{d_g} \tag{10}$$

kg is thermal conductivity of glass Tgo is temperature of outer surface of glass cover

From the external surface of the glass cover, the heat is lost by radiation to the sky at temperature Ts and by convection to the ambient air.

$$Q_{tgo} = A[\sigma\varepsilon_g\left(T^4{}_{go} - T^4{}_s\right) + h_w(T_{go} - T_a)] \tag{11}$$

The convective heat transfer coefficient h_w for air flowing over the external surface of the glass cover depends mainly on the wind velocity V_{wind}. McAdams [28] obtained an experimental result as:

$$h_w = 5.7 + 3.8\, V_{wind} \tag{12}$$

$$V_{wind}\ is\ speed\ of\ wind\ over\ glass\ , h_w\ is\ convective\ heat\ transfer$$

$$T_s = 0.0552 T_a{}^{1.5}\ [29] \tag{13}$$

T_a is ambient temperature

For the approximation of the convective heat transfer coefficient between the absorber plate and glass cover h_{pg} the correlation of Buchberg et al. [30] has been used:

$$Nu = 1 + 1.446\left(1 - \frac{1708}{Ra}\right) for\ 1708 < Ra < 5900$$

$$Nu = 0.229(Ra)^{0.252}\ for\ 5900 < Ra < 9.23\ \times 10^4$$

$$Nu = 0.157(Ra)^{0.285}\ for\ 9.23\ \times 10^4 < Ra < 10^6$$

$$Ra = \left[\frac{g(T_p - T_{gi}) d_{pg}{}^3}{T_{mpg}\, v_{mpg}{}^2}\right] Pr \tag{14}$$

$$v_{mpg}\ is\ viscosity\ of\ air$$

Back Losses

Heat loss from back

$$Q_b = \frac{A(T_b - T_a)}{\frac{d_i}{k_i} + \frac{1}{h_w}} \tag{15}$$

d_i and k_i are thickness and conductivity of insulation

Edge Losses

For the edge loss estimate, the following equation suggested by Klein [31] has been used

$$Q_e = 0.5A_e(T_p - T_a) \tag{16}$$

The thermo-physical properties of the air have been calculated at the corresponding mean temperature T_m. The following relations of the thermo-physical properties, obtained by correlating data from Holman [32], have been used:

$$C_p = 1006(\frac{T_m}{293})^{0.0155} \tag{17}$$

$$k = 0.0257(\frac{T_m}{293})^{0.86} \tag{18}$$

$$\mu = 1.81 \times 10^{-5}(\frac{T_m}{293})^{0.735} \tag{19}$$

$$\rho = 1.204(\frac{293}{T_m}) \tag{20}$$

$$Pr = \frac{\mu C_p}{k} \tag{21}$$

$$Q_L = Q_{tpg} + Q_{tg} + Q_{tgo} + Q_b + Q_e \tag{22}$$

$$Q_L = AU_L(T_p - T_A) \tag{23}$$

$$T_p\ is\ mean\ absorber\ plate\ temperature\ T_A\ is\ ambient\ temperature$$

$$Q = AI(\tau\alpha) - Q_L \tag{24}$$

$$\eta = \frac{Q}{AI(\tau\alpha)} = \frac{\dot{m}C_p(T_o - T_i)}{AI(\tau\alpha)} \tag{25}$$

CHAPTER 4: RESULTS AND DISCUSSIONS

Final Design

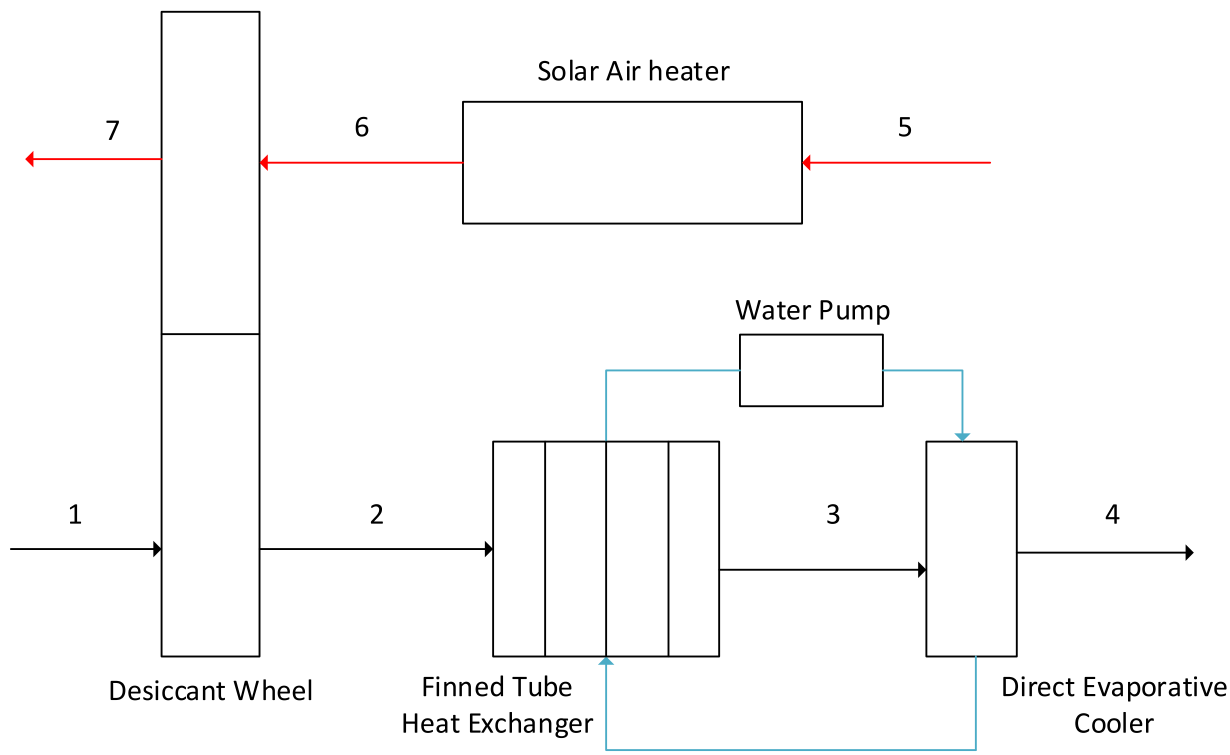

Figure 17 Schematic representation of final design

Humid Ambient air at 1 enters the processing side of desiccant wheel. After dehumidification, temperature of process air increases due to adsorption. Process air is sensibly cooled between state point 2 and 3 by finned tube heat exchanger. It is further cooled between state point 3 and 4 by evaporative cooler before entering to the room. The water from evaporative cooler is first pumped to the heat exchanger and then passes onto the cooling pads. This is done because water after evaporation is cooler. Ambient air at 5 enters the solar air heater where its temperature is raised using solar radiations. Between points 6 to 7, regeneration air removes moisture from dehumidifier by desorption process before leaving to atmosphere.

DW CAD Model

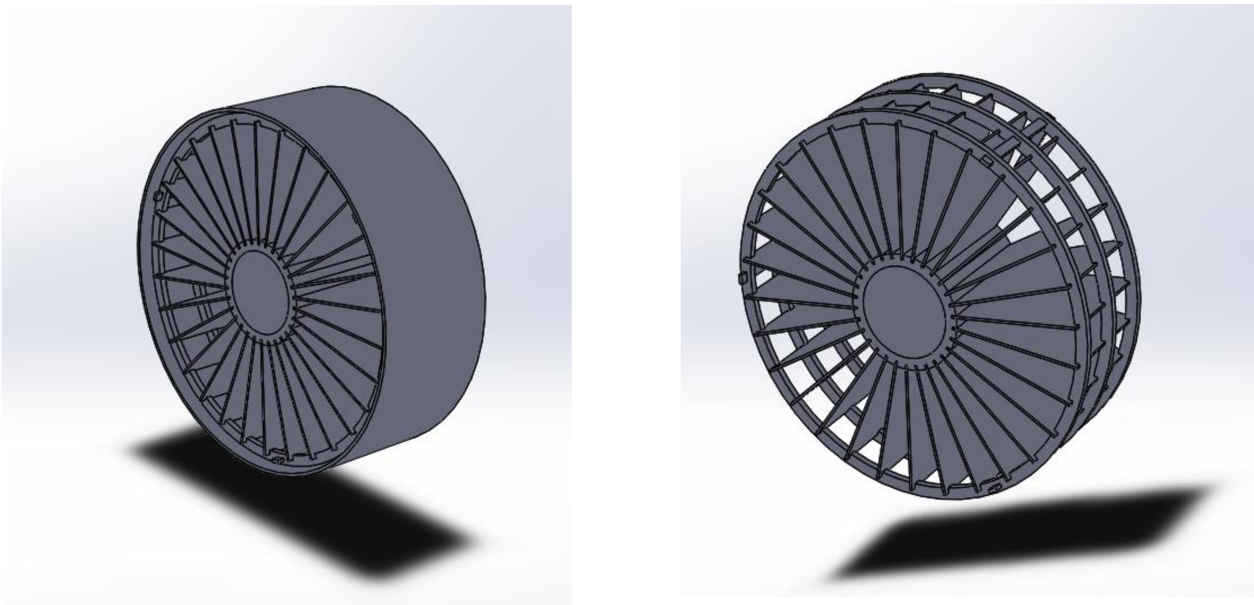

Figure 18 Final design of desiccant wheel

Heat Exchanger

As the values of air temperature at inlet is known and at outlet it can be assumed to be 10℃ higher than the inlet and temperature of water at inlet is also known therefore to calculate the required area of heat exchanger NTU effectiveness method was used.

Number of transfer units of a cross flow heat exchanger can be calculated using heat capacity ratio and effectiveness of heat exchanger.

$$\mathrm{NTU} = -\frac{1}{\sqrt{1+c^2}} \ln\left(\frac{\frac{2}{\varepsilon}-1-c-\sqrt{1+c^2}}{\frac{2}{\varepsilon}-1-c+\sqrt{1+c^2}}\right) \text{[33]} \quad (26)$$

Where $c = \frac{C_{min}}{C_{max}}$ and $\varepsilon = \frac{Q_{actual}}{Q_{max}}$ is effectiveness of radiator

Required area of radiator is calculated using $A_s = \frac{NTU * C_{min}}{U}$

Mass flow rate, inlet and outlet temperature and specific heat at constant pressure of air is taken as $\dot{m} = 0.21\ kg/s$, $T_{in} = 50^o\ C$, $T_{out} = 40^o\ C$ and $C_p = 1.005\ kJ/kg - K$ respectively.

Actual heat transfer from the heat exchanger can be calculated as

$$Q_{actual} = C_{air} * (T_{in} - T_{out}) = 2.11\ kW$$

Where $C_{air} = \dot{m} * C_p$

Mass flow rate, inlet temperature and specific heat at constant pressure of water are taken as $\dot{m} = 0.04\ kg/s$, $T_{in} = 25^o\ C$ and $C_p = 4.18\ kJ/kg - K$.

So maximum possible heat transfer can be calculated as

$$Q_{max} = C_{min} * \left(T_{in,air} - T_{in,water}\right) = 4.18\ kW$$

Effectiveness of heat exchanger can be calculated as

$$\varepsilon = \frac{Q_{actual}}{Q_{max}} = 0.5$$

Heat capacity ratio was calculated as

$$C = \frac{C_{min}}{C_{max}} = 0.7924$$

From NTU formula NTU was calculated to be 1.0336

Overall heat transfer coefficient between air and water 'U' was taken as 120 kW/m^2-K [34].

Required area of heat exchanger calculated by $A_s = \frac{NTU * C_{min}}{U}$ was found to be 1.44 m^2

Area Requirement for SAH

To achieve a temperature rise of 20^o C.

$$Prandtl\ Number = 0.702$$

$$Rayleigh\ number = 19633$$

$$Nusselt\ Number = 2.62$$

$$h_{pg} = 0.07 \frac{W}{m^2K}$$

$$\frac{Q_{tpg}}{A} = 38.7 \frac{W}{m^2}$$

$$\frac{Q_{tg}}{A} = 150 \frac{W}{m^2}$$

$$h_w = 13.3 \frac{W}{m^2K}$$

$$\frac{Q_{tgo}}{A} = 177.3 \frac{W}{m^2}$$

$$\frac{Q_b}{A} = 3.5 \frac{W}{m^2}$$

$$\frac{Q_e}{A_e} = 1.25 \frac{W}{m^2}$$

$$\frac{Q_L}{A} = 370.75 \frac{W}{m^2}$$

$$U_L = 17.17 \frac{W}{m^2K}$$

$$\frac{Q}{A} = (0.8 \times 1000) - 370.75 = 429.25 \frac{W}{m^2}$$

$$\eta = 53.6\%$$

$$\frac{\dot{m}}{A} = 0.021 \frac{kg}{s\, m^2}$$

Thus 1 m^2 area of absorber plate can provide a mass flow rate of 0.021 kg/s while raising its temperature by 20 degrees. This area can be augmented easily by the using fins on the absorber plate which also increase its efficiency.

Psychrometric Evaluation

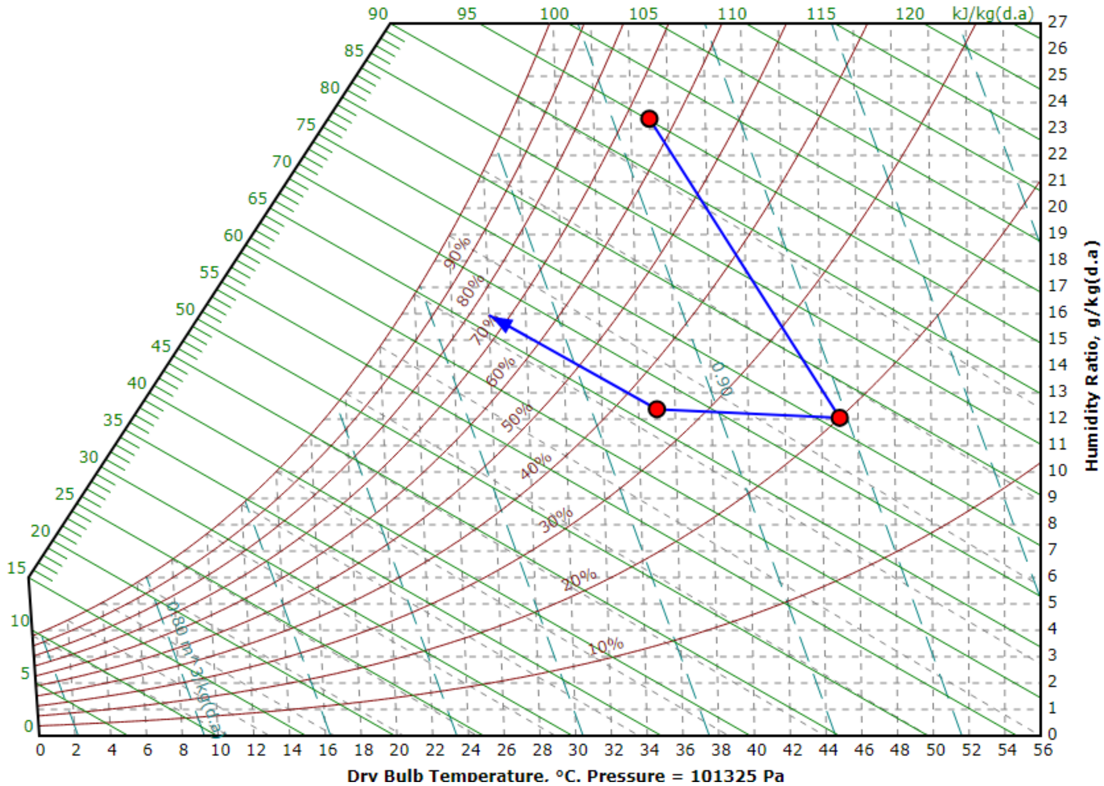

Figure 19 Psychrometric representation of test case

$$Enthalpy\ Difference = 29\ kJ/kg$$

$$mass\ flow\ rate = 0.035\ kg/s$$

$$Cooling\ load = 1.02\ kW = 0.3\ Tonnes\ of\ Refrigeration$$

CHAPTER 5: CONCLUSION AND RECOMMENDATION

Experimental Evaluation

At ambient conditions

Table 4 Experimental results for ambient conditions

Stage	Position	Temperature (C)	Relative Humidity (%)
1	Inlet	34	20
2	After Desiccant Wheel	36	15
3	After HX	30	20
4	Exit	25	60

Cooling Load = 0.382 kW

Flow Rate = 0.06 kg/s

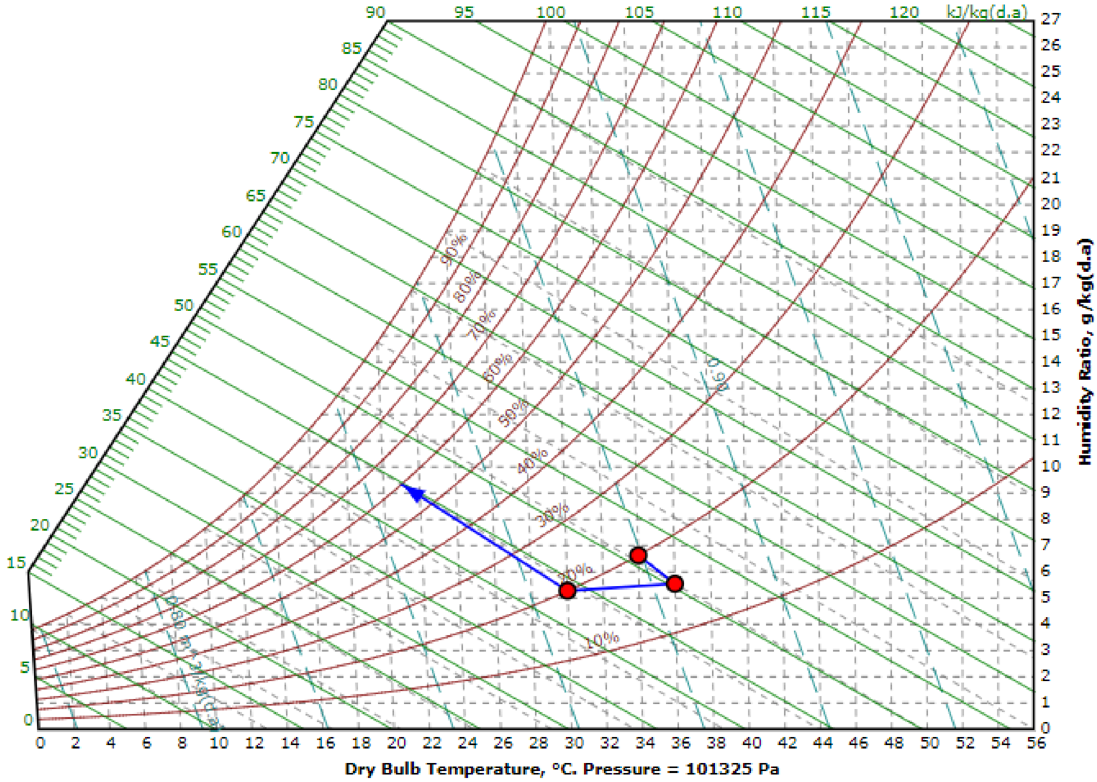

Figure 20 Psychrometric results for ambient conditions

At induced High Humidity

Table 5 Experimental results for high humidity conditions

Stage	Position	Temperature (C)	Relative Humidity (%)
1	Inlet	32	55
2	After Desiccant Wheel	36	30
3	After HX	31	40
4	Exit	25	70

Cooling Load = 0.829 kW

Flow Rate = 0.06 kg/s

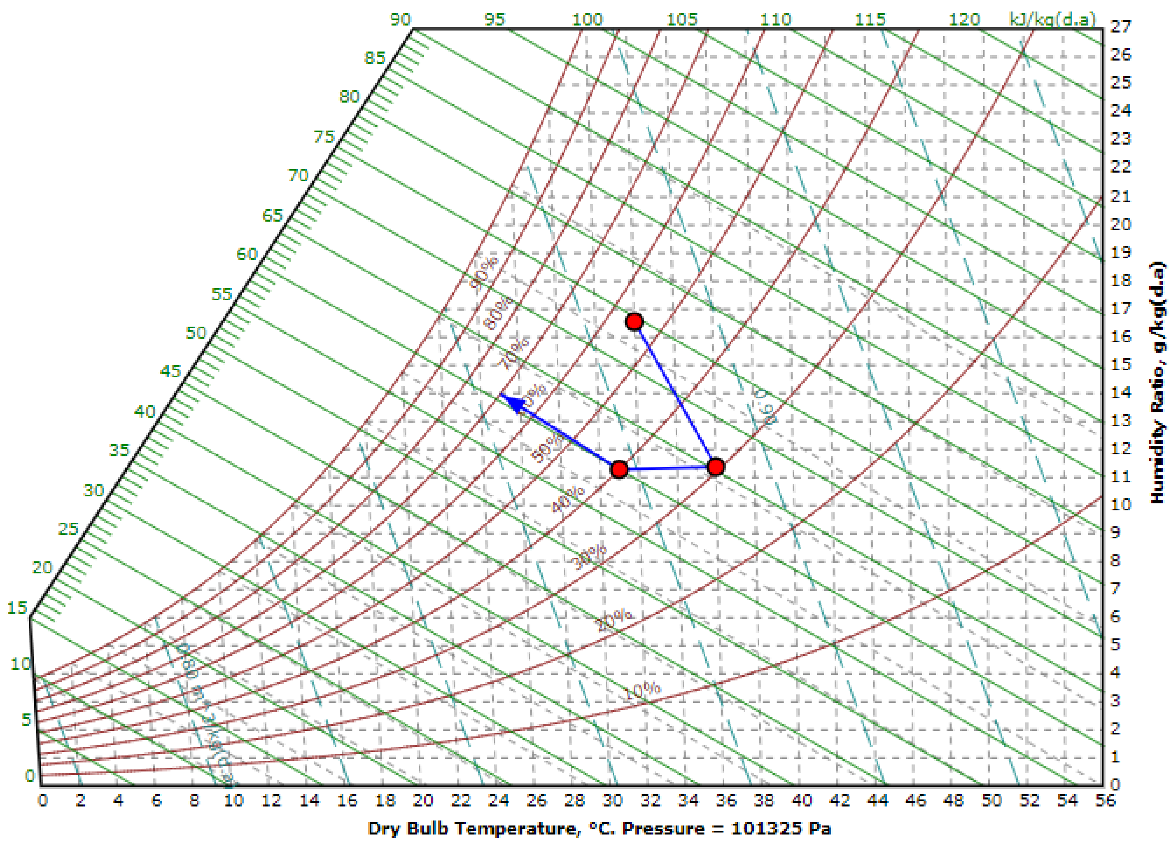

Figure 21 Psychrometric results for high humidity conditions

At induced High Temperature

Table 6 Experimental results for high temperature conditions

Stage	Position	Temperature (C)	Relative Humidity (%)
1	Inlet	44	21
2	After Desiccant Wheel	49	11
3	After HX	37	20
4	Exit	25	65

Cooling Load = 1.028 kW

Flow Rate = 0.06 kg/s

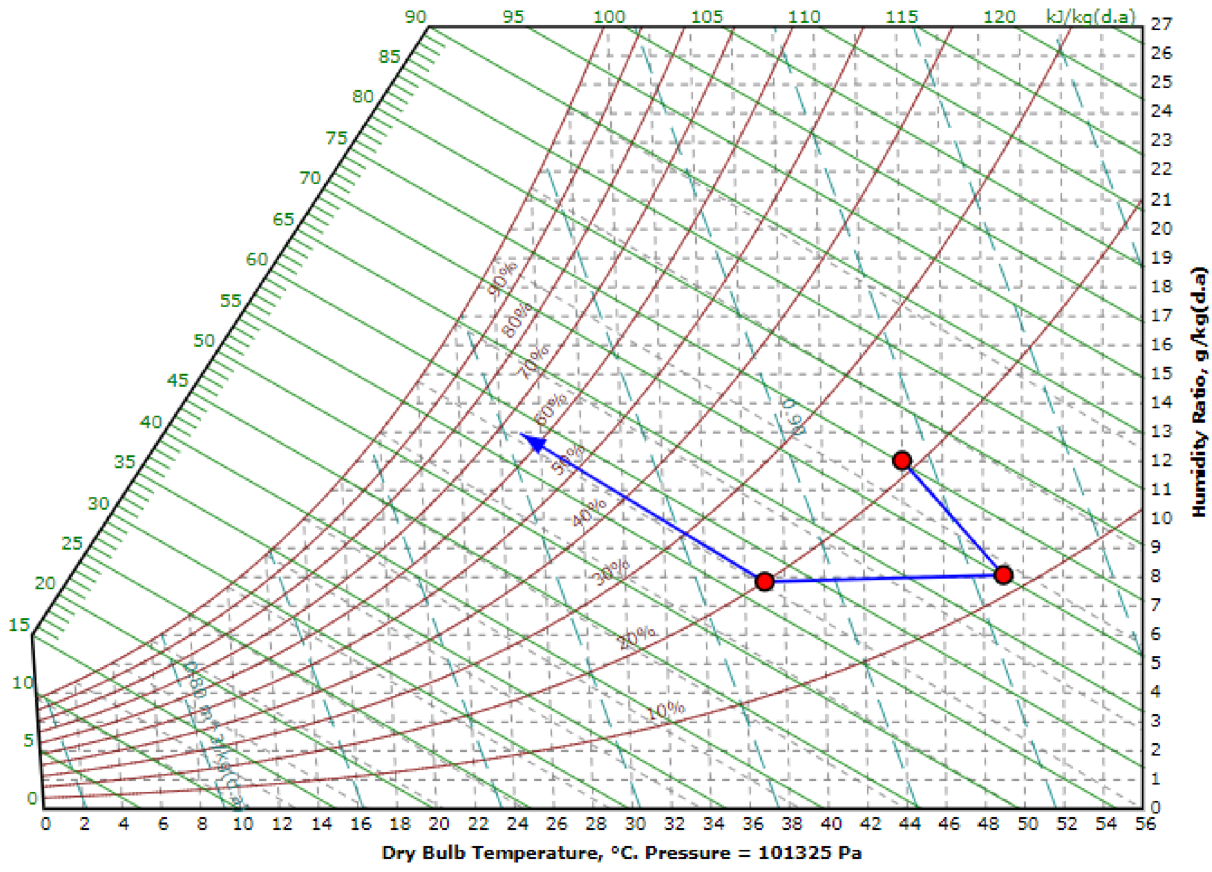

Figure 22 Psychrometric results for high temperature conditions

Solar Air Heater

Table 7 Data of solar air heater

Ambient	33 ℃
Output	68 ℃
At Inlet of Desiccant	55 ℃

Our desiccant based cooling system along with its axillary components are a prototype designed for a small specific room. The system can handle the cooling load of this room with minimal power consumption as compared to conventional air conditioning systems. This being a prototype still fulfils the demands of the application it is meant for. Up scaling of this prototype to handle large cooling loads is discussed in the future works section.

Human comfort levels are defined by standards such as ISO/EN 7730 [35]. They define in what circumstances a human being can live in conditions that are comfortable.

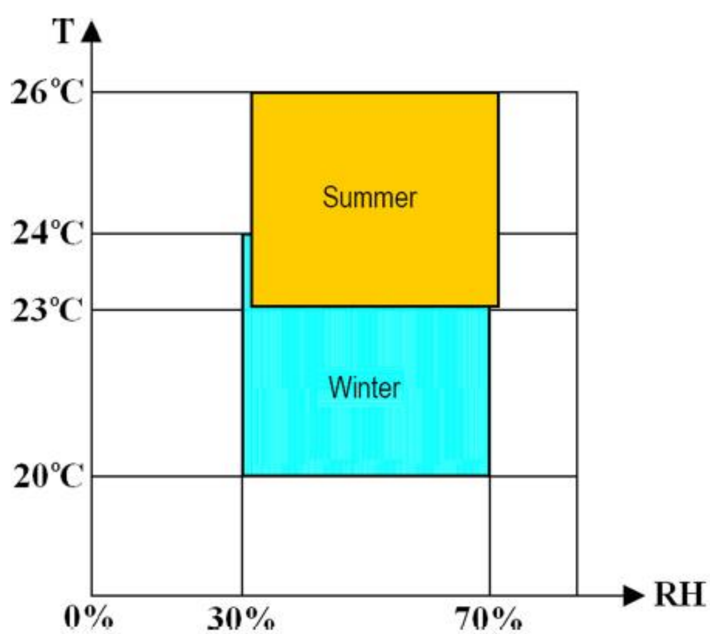

Figure 23 ISO/EN 7730 chart depicting human thermal comfort conditions

Techno-Economic Evaluation

Table 8 Variable costs of component of project

Desiccant Wheel	
Material	Price (Rs)
Nylon	2500
Acrylic	700
End Mill Cutters	1000
Rotor	800
Silica Granules	700
Epoxy	350
Woodworking	700
Rotating Assembly	2000
SUM	8750
Evaporative Coolers	
Material	Price (Rs)
Heat Exchanger	3400
Water Pump	600
PVC Assembly	200
Cooling Pads	500
Pipes	300
Fans	400
SUM	5400
Solar Air Heater (SAH)	
Material	Price (Rs)
Woodworking	1200
Metal Sheet	600
Cutting & Bending	600

Black Spray	350
Glazing	1400
Fans	400
SUM	4550
Duct	
Material	Price (Rs)
Metal Sheet	1300
Cutting & Bending	500
Fasteners	200
Styrofoam Sheet	150
SUM	2150
TOTAL COST	20850

Table 9 Fixed costs of components

Wages	
Marketing Team Members	3
Wage per Person	Rs 40000
Total Marketing Wage	Rs 120000
Sales Team Members	5
Wage per Person	Rs 30000
Total Sales Wage	Rs 150000
Incentives / Month	Rs 15000
Laborers	5
Wage per Person	Rs 35000
Total Laborer Wage	Rs 175000
Rent	
Rent Place	Rs 50000

Logistics Rent	Rs 60000
Operation Bill	Rs 20000
Components	**Cost**
Lighting, Fans & Others	40000
Universal Milling Machine	200000
Hand bending Machine	120000
Wood Cutting Machine	70000
Necessary Tools	40000
Marketing Cost / Sale	6%
Distribution Cost / Sale	7%
Packaging Cost Sale	2%
Total	Rs 3100

Table 10 Break even analysis

Working Hours	**10**
Working Days / Month	20
Units Produced / Month	60
Total Fixed Cost	Rs 6230000
Total Variable Cost	Rs 24000
Sale Cost	Rs 32000
Profit	Rs 8000
Break Even Quantity	780 units
Break Even Time	13 months

Table 11 Comparative analysis

Component	Energy Consumption (W)	Conventional Air Conditioner	
Rotor	11	Company	Haier
Water Pump	25	Type	DC Inverter
Fans	42	Cost	Rs 2000
Total	78	Energy Consumption	330 W
Cooling Load	1000 W	**Cooling Load**	1000 W
Energy Efficiency Ratio	12	Energy Efficiency Ratio	3
Operating Hours / Day	10	Operating Hours / Day	10
Units / Month	24	Units / Month	99
Tariff	8	Tariff	8
Bill / month	192	Bill / Month	792
		Saving	75%

As this is a dual weather system, so it can be utilized throughout the year.

The savings per month will be Rs. 6000 and break even for additional buying cost can be achieved with savings within 20 months.

Future Work

Our prototype can be up scaled in future applications to handle greater cooling loads. This up scaling process is rather simple as just the dimensions of the components in to be increased along with increasing in inlet flow rate that corresponds to the cooling load of the target area.

A problem that during the application which reduces the efficiency of the cooling process is that when the blades move from the regeneration portion to the process side of the operation, some warm air and the process air is mixed that increases the temperature of the process air, thus reducing the overall cooling effect. This effect is more prevalent in honeycomb desiccant wheels and in our design this mixing of air is reduced to a large extent. However, to completely remove this design flaw in future applications special methods can be used such as leaving some space between the regeneration and the process side so that the two air streams do not mix.

Our design of the radial desiccant wheel had straight blades and our purpose is to test the pressure drop and humidity of this novel design. In future work experimentation with airfoil shaped radial blades can also be tested and its effect of the dehumidification can be studied.

Our radial desiccant wheel has a dimension of 110mm depth. In future work this depth can be increased or multiple desiccant wheels in series can be connected and the effects of the aforementioned arrangement can be compared with our arrangement.

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APPENDIX I: DATA TO SOLVE SOLAR AIR HEATER

Thickness of glass cover	4 mm
Thermal conductivity of glass cover	0.6
Thickness of insulation	5 mm
Thermal conductivity of insulation	0.037
Wind Speed	2 m/s

APPENDIX II: SPECIFICATIONS OF DW CAD MODEL

Number of blades	30
Material of blades	Acrylic
Desiccant Wheel Thickness	110 mm
Desiccant Wheel Diameter	300 mm
Weight	3 kg

APPENDIX III: DATA TO SOLVE HEAT EXCHANGER

Overall heat transfer coefficient	120 kW/m^2-K
Specific heat of water at constant pressure	4.18 kJ/kg-K
Specific heat of air at constant pressure	1.005 kJ/kg-K