HYBRID M-CYCLE COOLER

A Final Year Project Report

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Bachelors of Mechanical Engineering

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ABSTRACT

Vapor Compression Cooling systems are a source of lot of problems for mankind which includes distortion of ozone layer and global warming due to emission of greenhouse gases. Moreover, they have considerably high energy requirements and are also rendered inefficient in humid conditions. Present time requires a cheap, environment friendly alternative which uses recyclable energy. Therefore, a new design of desiccant based evaporative cooling is proposed with aim to decrease temperature as well as humidity. The system runs primarily on electrical energy, which is required for an ordinary amount. The use of desiccant system would dehumidify the air and air heater would be used for regeneration of desiccant system. This dehumidified air can be effectively cooled using a setup of evaporative cooler which only uses water and air as their working fluids. The evaporation used is such that it is indirect and has no direct mixing of water with air. The overall objective is to develop and analyze the M-cycle cooler prototype. It includes testing the proposed design as well as improvements to find out the best potential of Hybrid M-Cycle based cooling system in the optimal dehumidification method. A vital task of this project is to make the whole system 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 greener environment and eco-friendly cooling system that has very low electrical input or can be run entirely on solar energy.

ACKNOWLEDGMENTS

We are foremost grateful to Almighty Allah, Who blessed us with the energy and strength to accomplish all the tasks successfully.

The conclusion of this major project was not feasible without the motivation and guidance of Dr. Emad Uddin at every stage and time.

Consistent prayers, sentimental encouragements and being the moral supporters, we will remain indebted to our parents.

We have placed our whole hard work out to make sure that the system is comprehended and explained easily.

ORIGINALITY REPORT

We hereby declare that no part of the work or report is plagiarized and the workings as well as the findings produced are wholly original. The project has been done under the generous supervision of Dr. Emad Uddin and has not been a subsidized part of any other similar project leading to similar degree's requirement from any institute. Any reference used has been clearly mentioned and we take the utter responsibility if found otherwise.

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ABBREVIATIONS

DEC	Direct Evaporative cooling
AC	Air Conditioner
FDM	Finite Difference Method
RPH	Rotation Per Hour
CMH	Cubic Meter Hour
IDEC	Indirect Evaporative Cooling
EER	Energy Efficiency Ratio
NTU	Number of Transfer Units
COP	Coefficient of Performance
CAD	Computer Aided Design
M-Cycle	Maisotsenko Cycle

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
Cp	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
Ya	Absolute humidity ratio of air stream	-
Yd	Humidity ratio in equilibrium with desiccant	-
De	Desiccant effective diffusivity	m^2/s
Ky	Gas-side mass transfer coefficient	kg/m^2 s
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	-
ν	Kinematic viscosity	m^2/s
τ	Transmissivity	-
α	Absorptivity	-
μ	Dynamic viscosity	kg/m.s
ρ	Density	kg/m^3
η	Efficiency	-

Subscripts

i	Insulation
w	Water
a	Air
m	Mean

CHAPTER 1: INTRODUCTION

With every passing year, the average temperature of earth tends to increase due to increment in global warming which has led to glaciers melt down, shrinkage of rivers and lakes, and exploitation of fossil fuels. According to current survey, more than 95% cooling systems are based on vapor compression system. Conventional vapor compression air-conditioning systems are locally used to enhance the living environment provided that the heat ratio is higher than 75%. And if the heat ratio is decreased below 75% then it consumes more electrical power and due to that, more greenhouse gases are emitted which has devastating results on our environment.

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

The cooling load can be named as sensible and latent part. The conventional system has difficulty overcoming the latent load. In order to reduce the moisture, it cools the load below the dew point and then reheats it which consumes a lot of energy. The systems which incorporates overcooling and then reheating, utilizes excessive amount of electrical energy and emits CO_2.

Problem Definition

Domestic usage of cooling system consumes about 20 – 40 % of global energy production. And out of which 80% energy is produced through coal and other hydrocarbons. This is not only affecting the natural fossil fuel reserves but also becoming the major cause of environmental issues like smog and temperature deviations. The cooling system uses refrigerants which are actually CFCs. Due to these refrigerants, UV rays from sun get a gateway to enter the earth surface. These affect the mankind by directly targeting the central nervous system as well as cause tremendous chronic skin diseases and malignant carcinoma, which is the deadliest form of cells distortion or cancer.

Objective

These problems of air-conditioners can be solved by developing alternative air conditioning system which not only reduces the energy consumption but also caters for the greenhouse gases emission. And this issue can be resolved by a least focused innovative solution which is known as Hybrid M-Cycle Cooler. It can be considered as low energy consuming air conditioning system which uses evaporative latent heat of water. This evaporative cooling system consumes quarter the amount of energy compared to compression cooling system. It can be also named as an improvised from of indirect evaporation cooling.

In direct evaporation cooling system, hot air enters the system mixes up with water and causes the water to evaporate in form of vapors. It is a cooling process due to absorption of energy from hot air by water. Thus the air is cooled. But keeping the mixing of vapors in view with air, the resultant outlet air is more humid. Therefore, in case of these systems cross ventilation is necessary and if kept otherwise the room gets uncomfortable due to increased moisture in the room. Whereas in indirect evaporation cooling system, we can name two channels as the dry channel and the wet channel. Air is brought in through dry channel and passes through heat exchanger. And water is passed, more specifically sprayed from top (Wet channel) over the heat exchanger. Both the air and water have an energy exchange through convection so there is no mixing of water and air taking place. And then air passes through that heat exchanger in cooled form by transferring energy to water to evaporate which causes cooling effect.

The effectiveness of direct cooling is more than the indirect cooling. But it verily depends upon the environmental condition in which it is used. For humid weather, indirect cooling is preferred due to it non-interference of water with air. Numerically, the effectiveness of direct cooling in terms of temperature depression is about 75 to 95% whereas for indirect is 40 to 60%.

In Maisotsenko cooler, which is a further development of indirect cooling system, inlet air enters from the dry channel which possesses low dry and wet bulb temperature and this characteristic is utilized to improve the effectiveness of cooling. In other words, part of air which has passed through the heat exchanger is reused to further cool down the upcoming air. Humidity can be a problem in humid climates so it is necessary to integrate with a dehumidifier in order to absorb humidity from the outlet air.

The dehumidifier used can be of any form but here a desiccant wheel which is designed in such a way that it doesn't affect the flow rate of air, is used. One half of the wheel is used for dehumidification while the other half is used for regeneration of silica gel. The rotation speed of this wheel is adjusted in such a way that time take by the wheel to rotate 180 degrees is equivalent to the time taken to remove the moisture and passed through exhaust system planted above the cooler.

CHAPTER 2: LITERATURE REVIEW

Due to increase in the energy consumption of the world, there is a need for the replacement and revision of the existing energy cycles. International Energy Outlook (2017) has projected that use of electricity is growing 2% annually. It is also noted that the building sector accounts for almost 20 – 40 % of the world energy consumption. The major energy consumer in this sector is air conditioning systems as it consumes about half of the energy that is being consumed in the buildings. So, there is a need for the efficient air cooling systems that are also environment friendly. In the past, air conditioning has been done only by compression systems (that include condenser, evaporator, and compressor) that use refrigerant gases like R-22 and R-134a etc. [1]

Direct Evaporative Cooling

One alternative cooling process is evaporative cooling method. The idea of Evaporative cooling is old. Ancient civilizations used these phenomena predominantly in Egypt. They used porous utensils covered with liquid submerged clothes for food preservation during the extreme hot Egyptian weather. The use of direct evaporative coolers rather than compressor based may be an answer, as they do not require electrical power for cooling but only for air circulation, because cooling is done by evaporation of water into the air. Direct evaporative cooling mechanism is shown in figure 1:

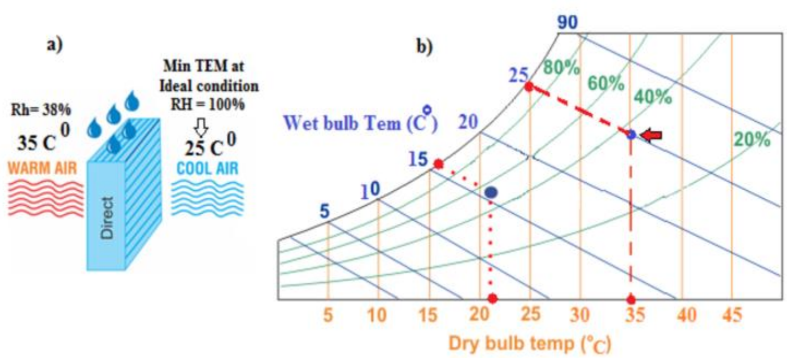

Figure 1: Direct Contact Cooler Mechanism

They may use less electric power than compressor-based coolers, but they have many disadvantages as well. Some of the disadvantages of direct contact air cooler are given below:

1. According to the previous studies, the minimum temperature that can be achieved using the M-Cycle cooler is the wet bulb temperature of the incoming air as shown in Fig 1
2. In DEC, due to the direct contact of air with the water fluid, moisture content of air increases to such levels that it becomes uncomfortable for the people.
3. DEC uses water mixture with air for the cooling effect and due to this usage of unclean water droplets, health issues may arise.
4. DEC are not suitable for tropical and sub-tropical regions where there is high humidity in the air.

Indirect Evaporative Cooling

Due to the problems in both types of cooling systems, the researchers began to research on some other methods. One of which is indirect evaporative cooler. Initially the indirect evaporative cooler combines a DEC and a heat exchanger. This system does not add humidity to the air, so one of the disadvantages of evaporative cooler is resolved but one major disadvantage occurs i.e. its performance decreases.

Theoretically, inlet air wet-bulb temperature can be achieved at the outlet of cooler. But these ideal conditions require infinite surface area and pure counter flow arrangement. But, in reality, the dry side temperature only reaches point “c” as in fig 2. This is the main reason for not commercializing the indirect evaporative cooler. Also, its manufacturing cost and poor rate of heat transfer cannot be justified.

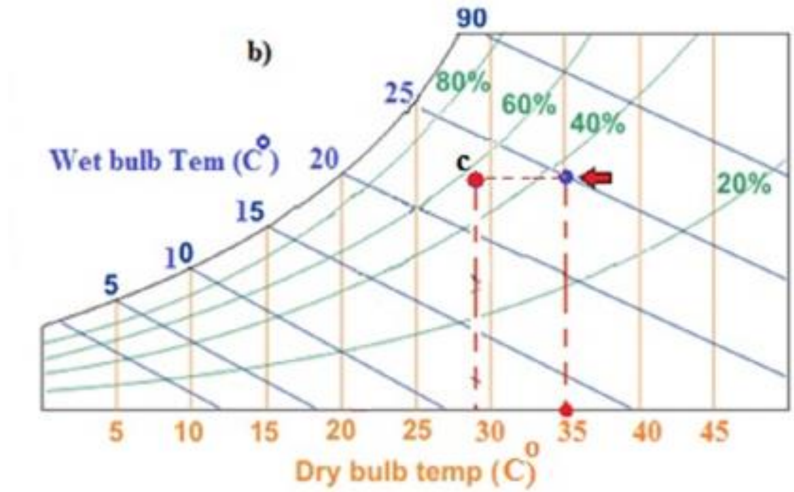

Figure 2: Indirect Evaporative cooling system

All these problems from compressor cooling systems and direct evaporative cooling systems are eliminated by the novel design presented by Maisotsenko. This novel design has been shown in the picture given on the next page. Look at how it separates the dry channel from the wet channel.

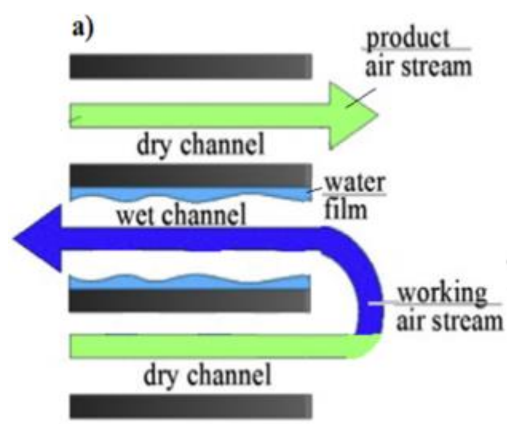

Figure 3: Indirect Cooling System Mechanism

In light of Maisotsenko cycle, the wet side air liquid is pre-cooled before entering the wet channel. Be that as it may, this precooling procedure can be happened by means of various systems. Fig.3 represents diverse counter stream arrangements of precooling procedure of wet side air current by possess wet channel. In any case, in Fig.4, a portion of the primary dry channel air-liquid is returned into the wet channel, which is called regenerative heat exchanger and mass exchanger.

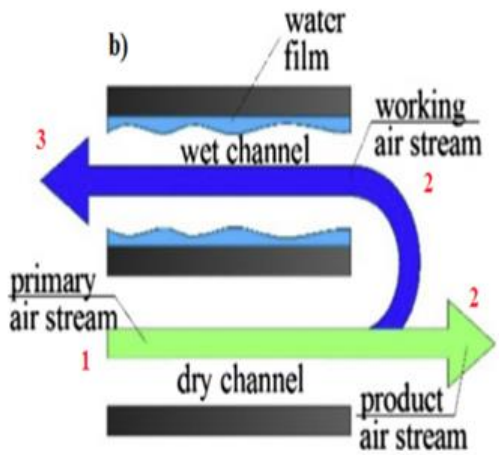

Figure 4: Maisotsenko Cooling System Mechanism

Maisotsenko innovation counters for every disadvantage of the direct evaporative cooling systems. It caters to the additional moisture added to the system and it also caters to safety issue that might arise due to the use of unhealthy water droplets in the air stream. It also reduces the temperature to the dew point temperature, which was not possible with direct evaporative cooler as it was able to reduce it to only the wet bulb temperature of the air stream entering in the cooler. Psychometric graph identified with Fig. 4 is shown in Fig. 5. Even though M-Cycle cooler has many advantages over the direct evaporative cooler, it hasn't been marketed for the general public. Without a doubt, pure counter-stream arrangement in a plate heat exchanger can't be manufactured in light of the geometry of the plates with air entering and leaving from same direction. This working principle of Maisotsenko-Cycle Indirect Evaporative Cooler theoretically overcomes all the disadvantages of direct evaporative cooler.

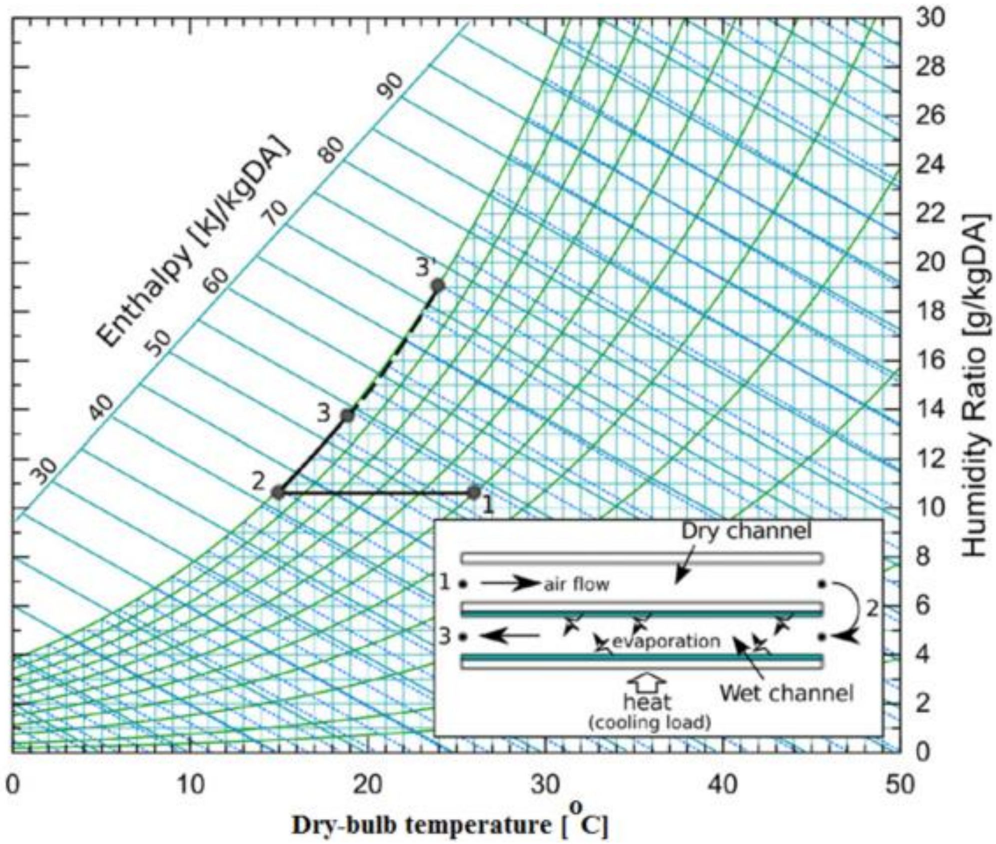

Figure 5: Psychometric Evaluation of Maisotsenko

Advantages of M-Cycle

The main advantages of M-Cycle cooler over others are as following:

1. Temperature can be reduced to dew point in M-cycle cooler, and they consume very less electrical power in comparison to other cooling systems. Dew point cannot be achieved in direct evaporative cooler.
2. Water is being used as a cooling mechanism in both DEC coolers and M-cycle coolers. But M-cycle coolers don't add any moisture to the air stream like compressor based coolers but they use CFC for this purpose.
3. As there is no direct contact of water in M-cycle cooler, health issues that arise due to contaminated water are none.
4. As inlet air temperature increases, its cooling capacity also increases.

5. Its initial and operating cost is very competitive.
6. It incorporates 100% fresh air, so its indoor atmosphere is healthy.

M-cycle is not suitable for tropical regions due to their humid climate. In order to get it work in these conditions, it should be integrated with desiccant systems to get maximum efficiency. Hence integration of liquid-desiccant or solid desiccant with M-cycle has been recently argued. These frameworks include 2 procedures:

- Dehumidifier removes the moisture
- M-cycle removes the sensible heat

Clearly, the viability of the primary stage intrigues on the working nature of the subsequent stage. Power required to drive desiccant system may be in the form of solar energy some electric motor. Though the primary thought of M-cycle evaporative cooler created around 1980, but it is commercialized in this century.

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, solar air heater or waste heat (e.g. solar). Many researchers used solar energy to regenerate the desiccant. As regeneration can be done effectively using solar energy. We are using a solar air heater for this process.

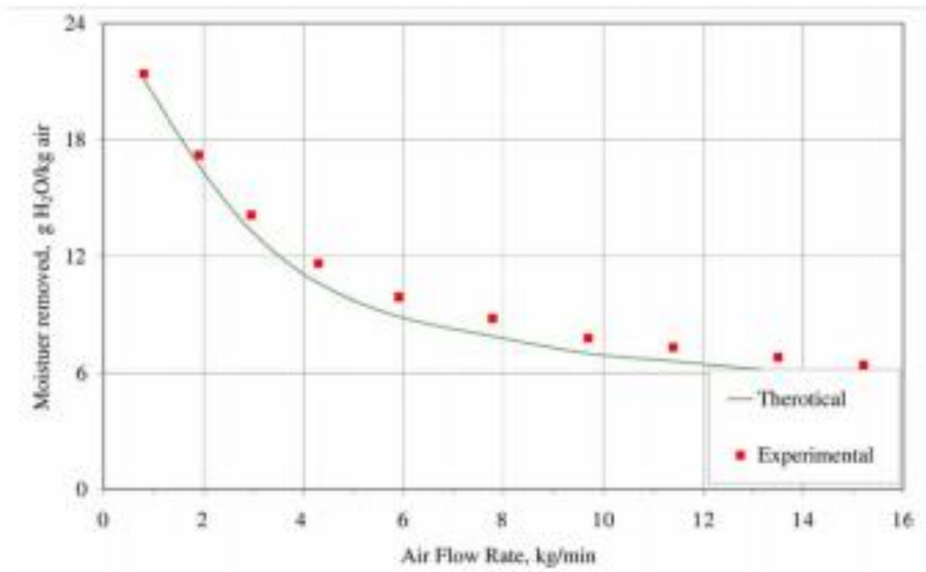

Figure 6: Graph relating mass flow rate of air to moisture removal capacity

Figure 6 [2] relates air flow rate to moisture adsorption in process air side. Increasing air flow rate reduces the moisture absorbed. It is evident from the experimental data as well because at higher flow of air, there would be less time for the air to grab the moisture. Hence moisture adsorption decreases.

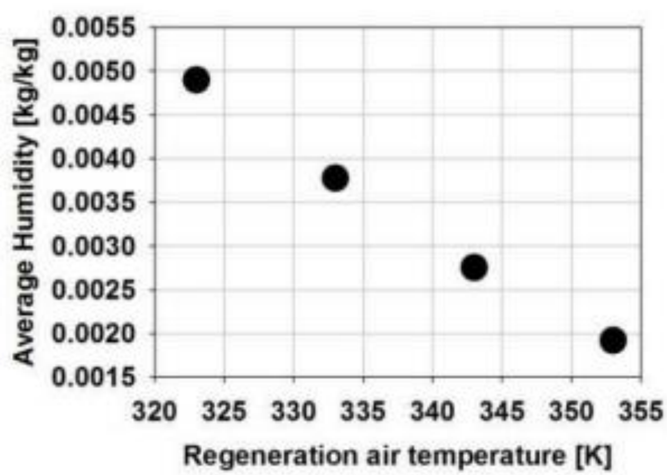

Figure 7: Graph showing effect of regeneration air temperature on outlet humidity

Figure 7 [3] relates regeneration temperature with average humidity of process air after passing through desiccant wheel. For higher regeneration temperatures, the humidity is better controlled.

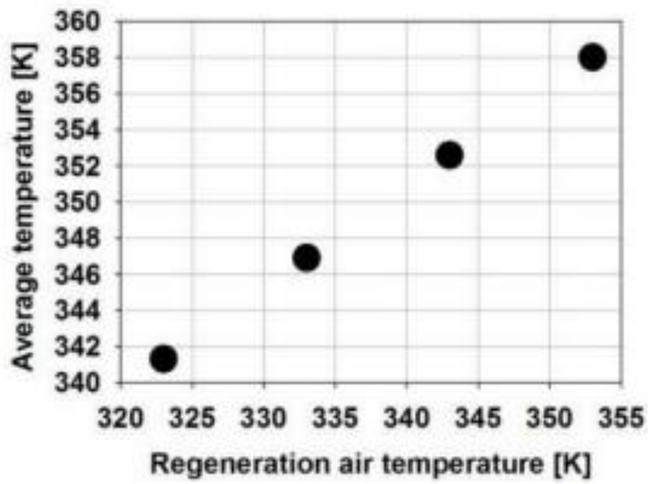

Figure 7: Graph showing effect of regeneration air temperature on outlet temperature

Figure 8 [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. It is due to the reason that when silica gel absorbs moisture, exothermic reaction occurs which increases temperature of dry air, which is then exited from the wheel.

Desiccant Material

Different desiccant material has different tendency to absorb the moisture. There are two different types of desiccant materials i.e. solid and liquid. Solid desiccants 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. Some of the various advantages and disadvantages are listed below in the table. The factors incorporated includes the temperature range of optimized regeneration, the portability and the cost effectiveness of the system.

Table 1: Comparison between Solid and Liquid Desiccant Properties

Properties Regeneration	**Solid Desiccant**	**Liquid Desiccant**
Regeneration Temperature	Medium	Low
Carry over	Low	High
Compactness	High	Low
Cost	Low	High

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 [4]

At lower regeneration temperature, the performance of silica and polymer are comparable whereas at higher regeneration temperatures, silica outperforms polymer and zeolite in

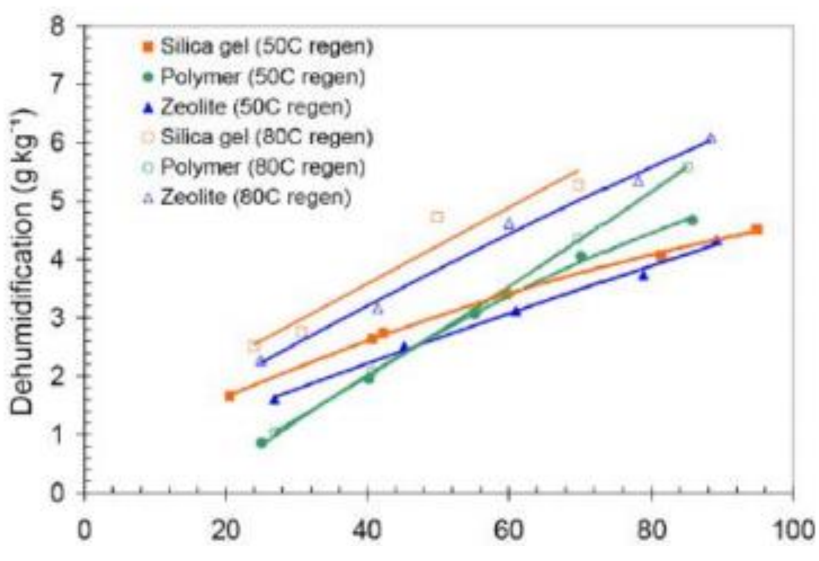

Figure 8: Graph showing effect of inlet RH on dehumidification capacity of various desiccants

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 the desiccant wheel, air temperature becomes quite high. As the cooling capacity of DEC goes as far as the wet bulb temperature of the supply air stream, this supply air needs to be cooled before entering the cooler. 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 3 desiccant air-conditioning systems coupled with different evaporative coolers and compare their results in their studies:

(1) the cross flow M-cycle HMX
(2) the regenerative counter flow M-Cycle HMX and
(3) the standard cross flow evaporative air cooler.

Previous Studies

- Hindoliya and Parmar [6], studied the prospects of evaporative cooling system based on 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.
- Hindoliya and Parmar [6] studied the evaporative cooling system that is based on desiccant dehumidification at different environmental conditions and COP of the system is calculated. The relation between the COP of system and humidity of an area is founded to be inversely proportional as greater humidity results in lesser COP of system.
- Davis RA [7] 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 stock air is more prominent than in conventional ACs.
- Cross flow indirect evaporative cooler was utilized by Pescod [8] in the desiccant system. The system was designed in a way that the water was flowing in the wet channel while the air in the secondary dry channel.
- Suryawanshi et al. [9] 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 [10] concluded that by utilizing both DEC and IEC in the cooling system, 74- 77% of the energy is saved when it is compared with compression cooling system.

Solar Regeneration

A solar air heater will be used to regenerate the silica gel so that it can be reused again and again after it has been consumed by the humidity of the air. The working fluid is a 50% mixture of propylene glycol and water. [11]

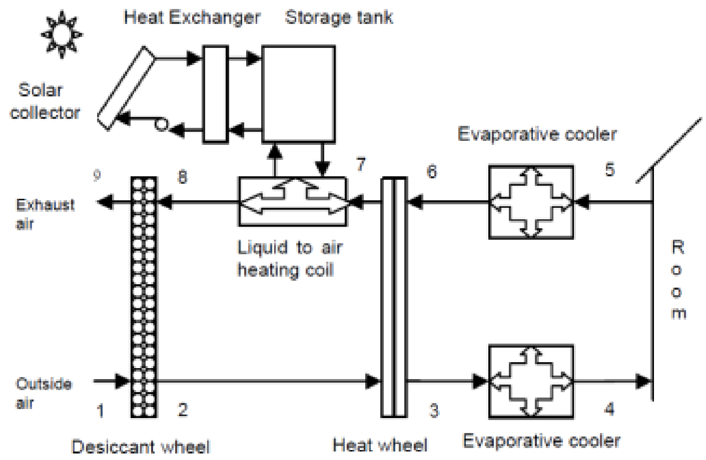

Figure 9: Solar Collector for Regeneration of Desiccant

This solar energy is used for the regeneration of the desiccant wheel. Such a type of solar air heater can supply the hot air which can be used to achieve the optimum temperature required to regenerate the desiccant material.

The basic configuration of such a cooling system is shown below [12]:

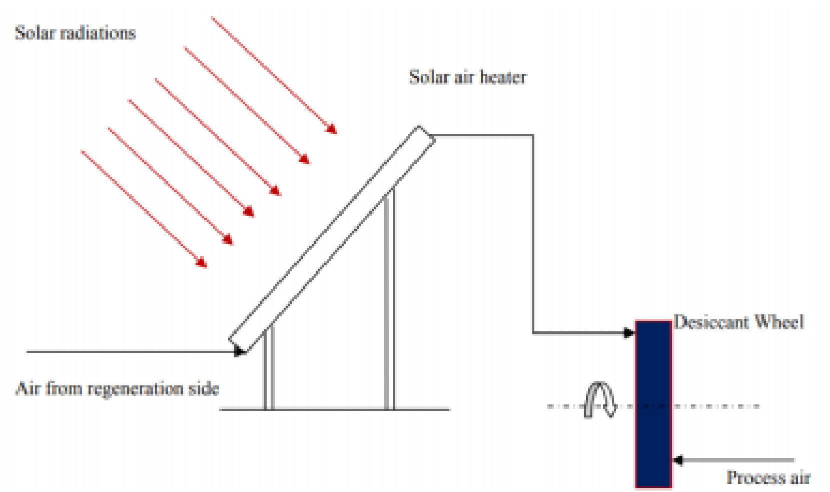

Figure 10: Using Solar Air Heater for Regeneration of Desiccant

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 15 solar collectors, 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.

CHAPTER 3: METHODOLOGY

Analysis of M-Cycle Heat Exchanger

Different parameters can be used for the calculations of the M-Cycle Heat Exchanger. The parameters that we are using are the flowrate of hot air entering, the flowrate of the water inlet, the humidity of the inlet air and the temperature of inlet air, temperature of outlet air and the temperature of water inlet. Area of inlet and outlet are also assumed according to the requirement. Using these parameters, the area of heat exchanger and the cooling capacity is calculated.

Following are the assumed values for the parameters:

$$T_{in\text{-}air} = 45° \text{ C} = 318\text{K}$$

$$T_{out\text{-}air} = 28° \text{ C} = 301\text{K}$$

$$T_{in\text{-}water} = 25° \text{ C} = 298\text{K}$$

$$RH_{in\text{-}air} = 28\%$$

$$RH_{out\text{-}air} = 40\%$$

$$\text{Volume Flowrate}_{air\text{-}in} = 0.45 \text{ m}^3/\text{s}$$

$$\text{Volume Flowrate}_{water\text{-}in} = 0.0000695 \text{ m}^3/\text{s}$$

As we don't know the outlet temperature of the water after the heat is exchanged, we will be using the NTU method here to find the required area and the cooling capacity. Number of transfer units of a cross flow heat exchanger can be calculated using the formula given below;

$$NTU = -\ln(1 + \frac{\ln(1 - c\epsilon)}{c})$$

Whereas ϵ is effectiveness and it can be found out as:

$$\epsilon = \frac{C_{max} \times \Delta T_{min}}{C_{min} \times \Delta T_{max}}$$

and

$$c = \frac{C(Min)}{C(Max)}$$

We know that the values of Q_{actual} and Q_{max} can be found as:

$$\Delta Tmin = (T_{H\text{-}in} - T_{H\text{-}out})$$

$$\Delta Tmax = (T_{H\text{-}in} - T_{c\text{-}out})$$

From the volume flowrates, we can find the mass flowrates of the parameters by multiplying those with the density.

$$Density\ of\ Water = 998\ kg/m^3$$

$$Density\ of\ Air = 1.225\ kg/m^3$$

The mass flowrates of inlet air and inlet water turn out to be:

$$\dot{m}(air\ inlet) = 0.55125\ kg/s$$

$$\dot{m}(water\ inlet) = 0.069361\ kg/s$$

Thus, the values of C_{min} and C_{max} are found by multiplying the mass flowrates with the respective specific heat of water and air:

$$C_{min} = C_{water} = 306.436898\ J/K$$

$$C_{max} = C_{air} = 554.00625\ J/K$$

The value of C and effectiveness can be found out from here:

$$C = 0.5531$$

$$\epsilon = 0.6507$$

Thus, the value of NTU can be found as:

$$NTU = -\ln\left(1 + \frac{ln(1 - c\epsilon)}{c}\right) = 1.643409$$

Now the required heat exchanger area can be found as:

$$A_s = \frac{\text{NTU} \times \text{C(min)}}{\text{U}}$$

Whereas, U is the overall heat transfer coefficient which is found from:

$$\frac{1}{U} = \frac{1}{h(air)} + \frac{L}{k} + \frac{1}{h\ (water)}$$

The value of U comes out to be 250 W/m^2. K.

Hence, the area required comes out to be

$$A = 2.518006994\ m^2$$

Analysis of Different Dehumidification Methods

At constant dry bulb temperature, the removal of water content from air is termed as dehumidification. Following are a few types of dehumidification methods which are used in different fields;

Whole House Ventilation Dehumidifiers

It is cheaper than any plug-in dehumidifier. It is a very simple and easy method that is used for the dehumidification of air. In this method cold outside air is forced and hot inside air is exhausted. This method of dehumidification cannot be combined with other cooling systems.

Condensation Dehumidification

In condensation dehumidification, air is cooled below the dew point by removing moisture form air. Main elements used in condensation air dehumidifiers are fan, compressor, condenser, and evaporator. This type of dehumidifiers can’t work below 5-degree temperature due to their working nature.

Adsorption Dehumidification

Different desiccant materials have different tendency to absorb the moisture. There are two different types of desiccant as given on the next page.

- Solid Desiccant
- Liquid Desiccant

Both solid and liquid desiccant have pros and cons. Selection of desiccant material depends on the operating conditions. This selection criterion is defined in the table given on the next page.

Table 2: Comparison between Desiccant and Vapor Compression System

Parameter	**Vapor**	**Desiccant**
Indoor Air Quality	Average	High
Energy source	Electrical and Natural Gas	Waste heat, solar energy or any low-grade
Moisture Removal Capacity	Average	High
Operational Cost	High	Minimum
Energy Storage Capacity	Low	High
Solutions	HFC, CFC, HCFC	LiCl, LiBr, HCOOK
Effect on Environment	Harmful	Comparatively Eco-Friendly

Liquid Desiccant Systems (LDS)

The main working principle of LDS is vapor pressure difference. When moist air passes through absorber unit, water content from air goes to desiccant solution. This is cooled by indirect evaporative cooler and diluted solution is pumped for regeneration. Regeneration is done by hot stream of air. Hot air takes away moisture from solution. Lithium Chloride (liCl), lithium Bromide (LiBr), Calcium chloride and potassium-formate are usually used as liquid desiccant.

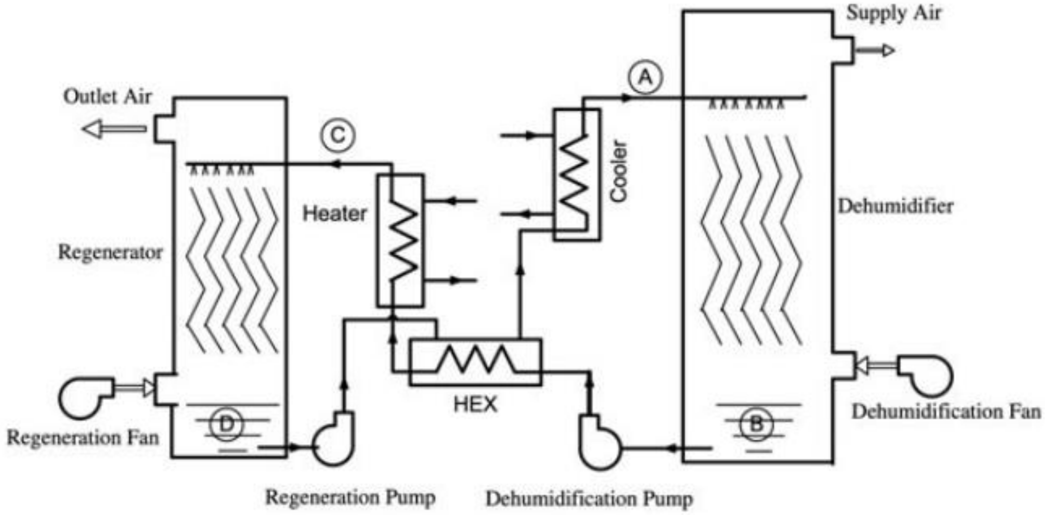

Figure 11: Schematic of an LDS

Disadvantages

(1) They are corrosive in nature.

(2) Liquid desiccants can contaminate and add odor in air which can be harmful for consumer.

(3) An additional pump is used to for liquid desiccant which increases its power consumption.

(4) A layer of desiccant material settles in cooling unit and decrease its cooling efficiency.

Solid Desiccant System

The working principle of solid desiccant is that they absorb water from incoming air and are regenerated by hot air. Some solids change their color due to change in the water of crystallization.

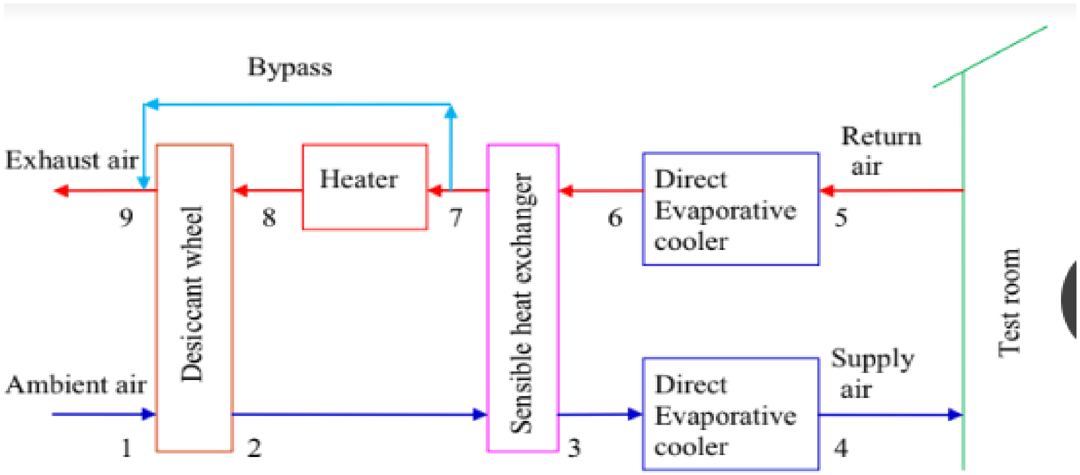

Figure 12: Schematic of a Solid Desiccant Cooling System

Solid desiccants 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, which are listed below in the table. The factors incorporated include the temperature range of optimized regeneration, the portability and the cost effectiveness of the system.

Table 3: Comparison between Solid and Liquid Desiccant Properties

Properties Regeneration	**Solid Desiccant**	**Liquid Desiccant**
Regeneration Temperature	Medium	Low
Carry over	Low	High
Compactness	High	Low
Cost	Low	High

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 is given in the picture;

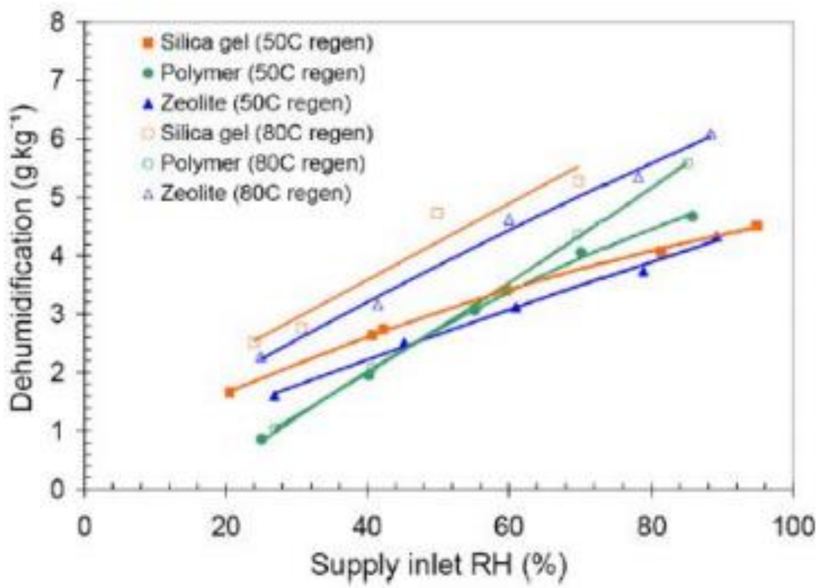

Figure 13: Graph showing effect of inlet RH on dehumidification capacity of various desiccants

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

Analysis of Desiccant Wheel

Traditionally, 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 [13] [14];

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 conditions 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 in this section. First of all, a picture of the wheel is shown in the next page [15];

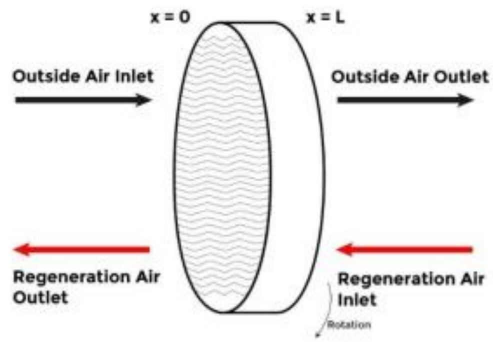

Figure 14: Simple model of Desiccant Wheel

Governing Equation

There are four main governing equations, the mass and energy conservation 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.

Conservation of Moisture in Air

The terms on the left side of the equation are the moisture capacity and the rate of dampness variation in the direction of the air flow. The term on the right side communicates the rate of dampness variation brought about by the convective mass exchange.

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

Conservation of Energy in Air

The left-hand side represents energy storage in the damped air times the rate of energy variation and the heat conduction in the air. The other side of the equation communicates heat transfer through convection and through sensible heat between the desiccant and air.

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

Conservation of Moisture in Desiccant

The left side of the equation determined the moisture capacity of the desiccant times the mass diffusion in axial direction within the solid desiccant. The other side of the equation is the convective mass transfer between the desiccant and air.

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

Conservation of Energy in Desiccant

The left side tells the energy capacity of desiccant into heat transfer within the desiccant through conduction while the right side tell us about the convective heat transfer between the air and desiccant, effect of heat due to adsorption and sensible heat transfer between the air and desiccant.

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

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 a research paper [16] [17] 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 Air 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 4: Experiment Inlet Conditions

Air Volume Flow Rate	**180 CMH**
Rotation Speed	35 RPH
Regeneration Temperature	60 ℃

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 5: 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 for the calculations provided above [17]

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 370x200 which uses silica gel as desiccant material.

At 26 C dry bulb temp and 41.7% relative humidity of inlet process air, the results of the simulator showed 2 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.

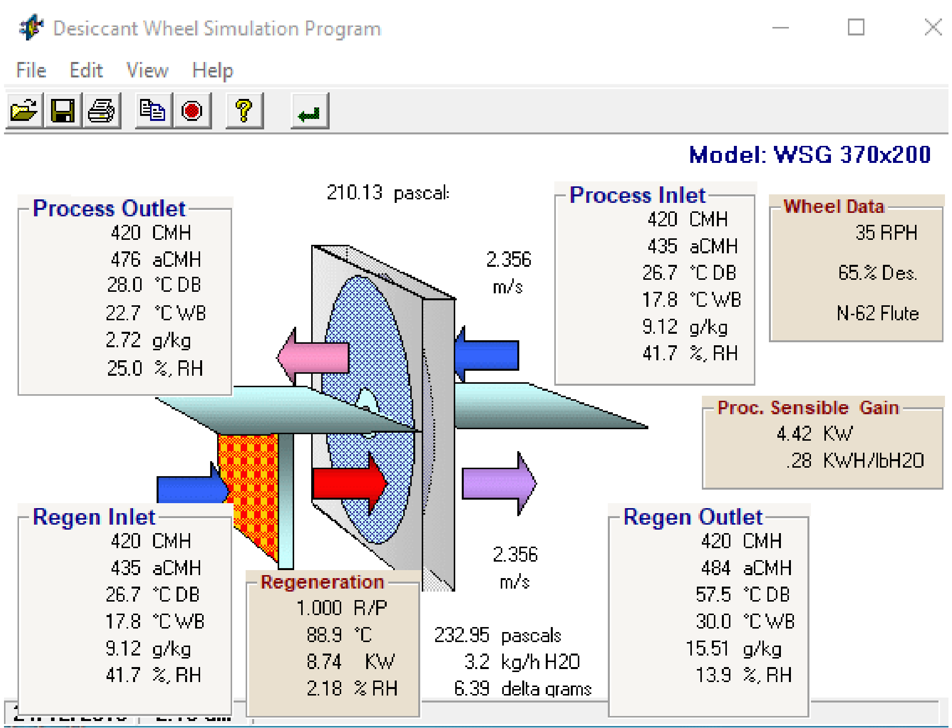

Figure 15: Simulation analysis of Desiccant Wheel

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 22.7 C dry bulb temp and

25% relative humidity of outlet 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.

Thus this simulator can be used to find the values for our system as well. Giving it an input value of 28 C and 73.3% relative humidity, we found out that the processed air at the outlet would be of 30.1 C with a relative humidity of 34%.

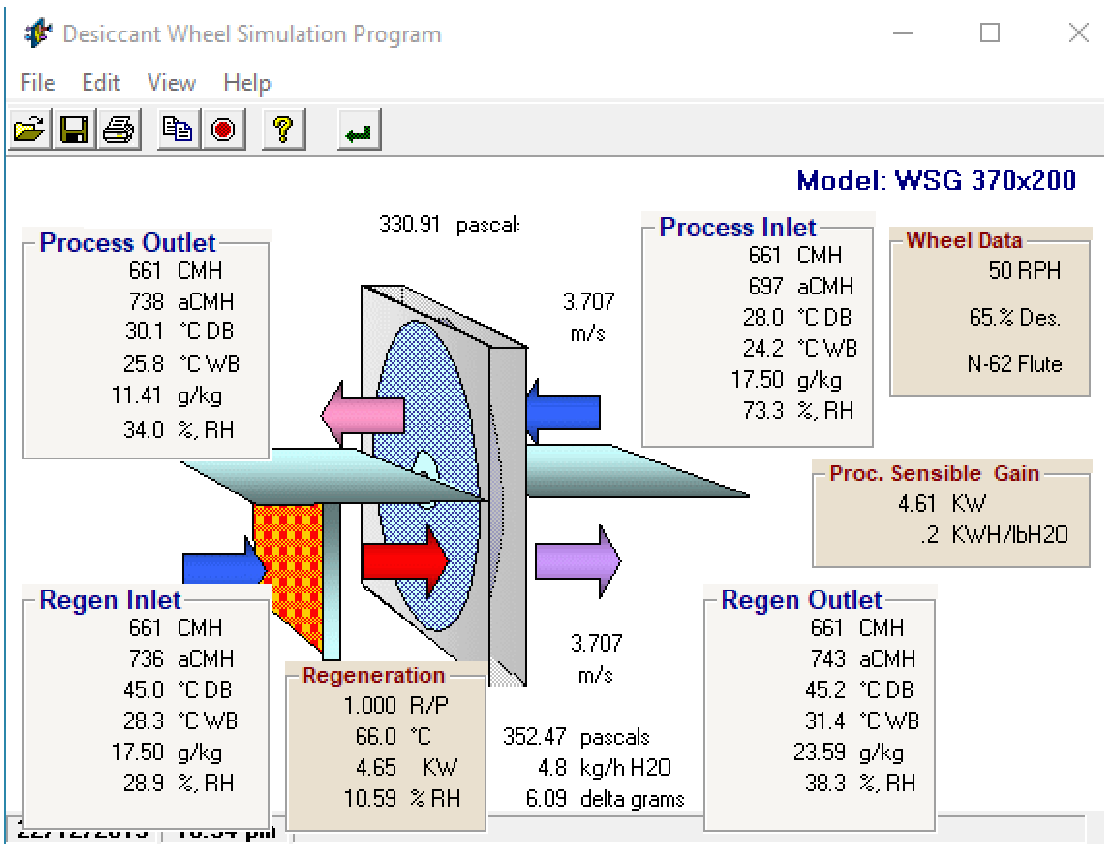

Figure 16: Input conditions on NOVEL Air Program

Solar Air Heater Analysis

A solar heater is being used here to regenerate the desiccant material that we are using. We know that solar radiations are not completely used by the solar heater and only the energy transferred by the solar cells is the energy which we can use whereas the energy that is absorbed or reflected by the solar plates counts towards wasted energy.

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

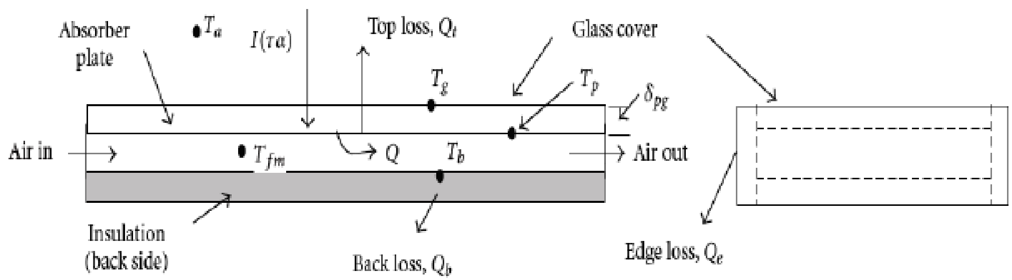

Figure 17: Losses in a Solar Air Heater

There are many types of losses that we face in the solar air heater and those losses are discussed below. To reduce those losses, we need to use a glass cover of high transmissivity which can transfer more energy and wastes less energy. After that, this transferred energy gets to the absorber plate which is then used.

$$\text{Input Energy} = \text{Transmissivity of Glass Cover} \times \text{Absorptivity of Absorber Plate} \times \text{Solar Radiation}$$

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

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

Top Losses

Heat transfer from the absorber plate, which is at the mean temperature T_{p}, to the internal glass cover surface, which is at temperature T_{gi}, takes place by radiation and convection.

$$Q_{\mathrm{tpg}} = A\ [\sigma\ (\mathrm{T_p^4} - \mathrm{T_{gi}^4})\ (\frac{1}{\varepsilon \mathrm{p}} + \frac{1}{\varepsilon \mathrm{g}} - 1)^{-1} + h_{\mathrm{pg}}\ (T_{\mathrm{p}} - T_{\mathrm{gi}})]$$

εp & εg here are the emissivities of plate and glass respectively while h_{pg} is the convective heat transfer coefficient between glass and plate.

Heat transfer through glass cover of thickness dg by conduction is

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

Whereas,

K_g = Thermal Conductivity of Glass

Tgo = Outer temperature of glass cover surface

From the external glass cover surface, the heat is lost either to the sky at T_s temperature by method of radiation or to the ambient air through method of convection.

$$Q_{t\text{-}go} = A\, [\sigma\, \varepsilon_g\, (T_{go}^4 - T_s^4) + h_w\, (T_{go} - T_a)]$$

Where

h_w = Convective heat transfer coefficient of air

σ = Stefan Boltzmann's Constant

The h_w for air flowing over the external glass surface cover depends mainly on the wind velocity V_{wind}. McAdams [18] obtained an experimental result and found out the values of h_w and T_s as:

$$h_w = 5.7 + 3.8 * V_{wind}$$

$$T_s = 0.0552\, T_a^{0.5}$$

Here,

T_a is the ambient temperature [19]

For the approximation of the convective heat transfer coefficient between the absorber plate and glass cover h_{pg} the correlation of Buchberg et al. [20] has been used:

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

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

$$Nu = 0.157(Ra)^{0.285}\ for\ 9.23 \times 10^4 < Ra < 106$$

$$Ra = \frac{\mathrm{g(T_p - T_{gi})d_{pg}^3}}{\mathrm{T_{mpg}\, v_{mpg}^2}}\ .\ \mathrm{Pr}$$

Here,

T_{mpg} = Mean temperature between plate & glass

d_{pg} = Space between plate & glass

ν_{mpg} = Kinematic viscosity of air at T_{mpg}

Back Losses

Heat loss from back

$$Q_\mathrm{b} = \frac{\mathrm{A\ (T_b - T_a)}}{\frac{\mathrm{d_i}}{\mathrm{k_i}} + \frac{1}{\mathrm{h_w}}}$$

Here,

d_i = Thickness of Insulation

k_i = Conductivity of Insulation

Edge Losses

For the edge loss estimate, the following equation suggested by Klein [21] has been used;

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

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 [22], have been used:

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

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

$$\mu = 1.81 \times 10^{-5} (\frac{T_m}{293})^{.0735}$$

$$\rho = 1.204 (\frac{293}{T_m})^{.0155}$$

$$\mathrm{Pr} = \frac{\mu\, c_p}{k}$$

Hence, the overall losses can be found as;

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

$$Q_L = AU_L\,(T_p - T_A)$$

The final output energy and the efficiency can be found, using the following formulas;

$$Q = I(\tau\alpha) - Q_L$$

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

Calculations

To achieve a temperature rise of 20 K, we get the following values for the temperatures,

$$T_i = 303\ \mathrm{K}$$

$$T_o = 323\ \mathrm{K}$$

$$T_a = 303\ \mathrm{K}$$

$$T_m = 313\mathrm{K}$$

Using these values of the temperatures, we can find out the following values, using the formulas given above;

$$\text{Prandtl Number} = 0.6574$$

$$\text{Rayleigh Number} = 19205$$

$$\text{Nusselt Number} = 2.7495$$

The values of h_w & h_{pg} can also be calculated as;

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

$$h_w = 13.3\ W/m^2K$$

Thus, the final values of losses per unit area come out to be;

$$Q_{tpg}/A = 3.6947\ W/m^2$$

$$Q_{tg}/A = 150.3\ W/m^2$$

$$Q_{tgo}/A = 187.4252\ W/m^2$$

$$Q_b/A = 33.2821\ W/m^2$$

$$Q_e/A = 14.75\ W/m^2$$

The total values of losses here can be found by adding all the individual losses;

$$Q_L/A = Q_e/A + Q_b/A + Q_{tgo}/A + Q_{tg}/A + Q_{tpg}/A$$

$$Q_L/A = 389.1525\ W/m^2$$

Hence, the total energy absorbed and the efficiency of the solar AH can be found out using;

$$Q = I(\tau\alpha) - Q_L$$

$$Q/A = 1000 \times (0.8) - 389.1525 = 410.8480\ W/m^2$$

$$\eta = \frac{Q}{AI(\tau\alpha)} = 51.36\ \%$$

Thus, the final air flow can be found as;

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

$$\dot{m}/A = 0.0204\ kg/s\text{-}m^2$$

Hence, we can deduct from here that for a flow rate of 0.0204 kg/s, we can find the area of the solar air heater plate to be 1 m^2. Additional fins on the absorber plate can also be used to increase the efficiency of the plate.

Cost Breakdown

This section of the report depicts the construction and development materials used for the product. The main target was to utilize the local market and general standard material that helps in better commercialization of the cooler. The market survey included the cost analysis for the construction, maintenance and other contingency finances. Overall the cost of one complete operational cooler was found to be around 44,000 PKR. The cost may fluctuate and is highly time dependent. The initial cost of the components of Hybrid M-Cycle Cooler are given in the table below;

Table 6: Initial Cost of Components of Cooler

Component	Quantity (Units)	Cost (PKR)
Outer Frame of Aluminum Sheets	1	5,000
Heat Exchanger	1	15,000
Dehumidification Wheel	1	14,450
Silica Gel	1 kg	600
Inlet Fan	1	5,000
Ventilation Fans	2	1,000
Water Pump	2	1,200
Iron Frame Structure for Dehumidifier Frame	1	1,000
Holding Bearings	4	500
GMSA Glue	5	250

The above mentioned cost includes the cost of wages while manufacturing and assembling the components as well.

The running cost of the cooler includes the electricity cost and the maintenance cost. It does not require maintenance so it can be neglected and about the electricity cost, that can be compared with that of a compressor based air conditioning system in terms of power consumed. The Hybrid M-Cycle Cooler requires power of only 250 Watts while a compressor based air conditioning system for 1 tonne of cooling requires roughly 1300 watts. That is a roughly 80% decrease in the electrical power consumption while giving the same outputs of dehumidified cool air.

CHAPTER 4: RESULTS AND DISCUSSION

Final Design

The final design of the Hybrid M-Cycle Cooler is shown below. Humid ambient air enters through the inlet fan and passes through heat exchanger with a greater flow rate induced by it. Heat exchanger absorbs the heat and this function is enhanced by the presence of venting fan present at the top of the assembly. Then the cool humid air passes through the desiccant wheel that is integrated at the end of cooler, which is discussed in the next section.

Following is the final design of the Hybrid M-Cycle Cooler:

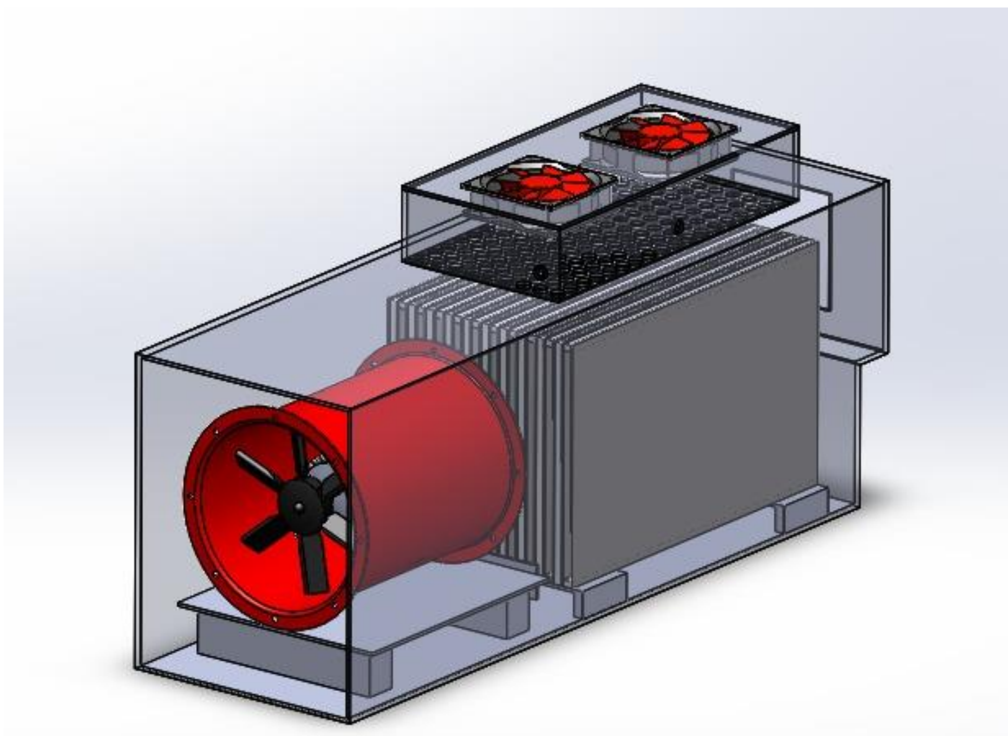

Figure 18: Final Design of Hybrid M-Cycle Cooler

Now by observing the greyscale model, we can see the addition of desiccant wheel at the outlet end. The desiccant wheel is divided into two parts with a slid dividing them in a latitude orientation. One part of the wheel helps absorbing moisture to the level of human comfort whereas the other part is the regenerative part which is attached to a heating element which dissipates the moisture absorbed by the desiccant material.

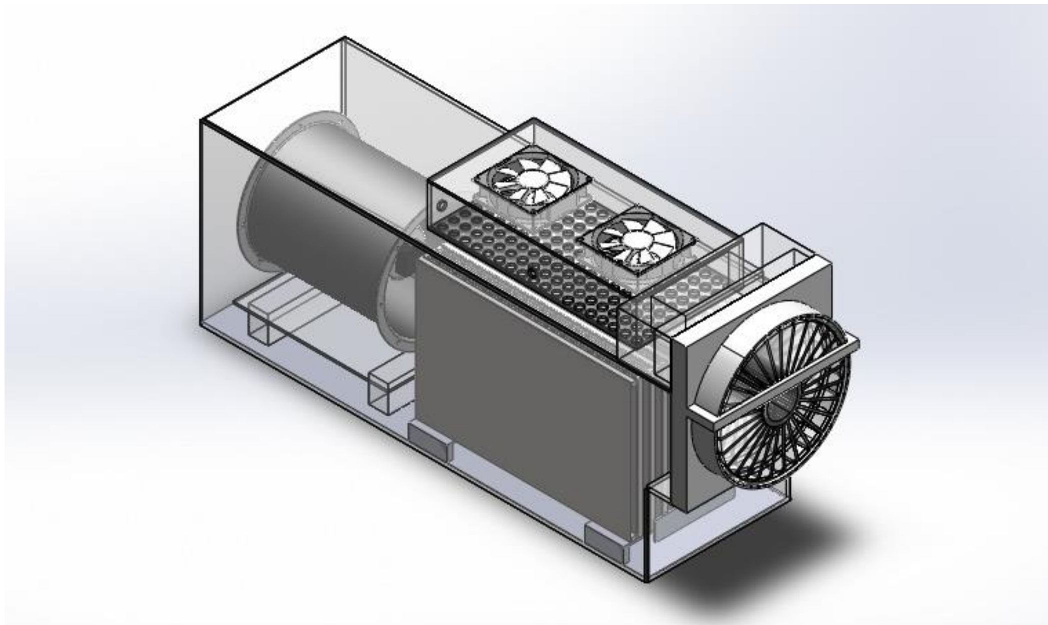

Figure 19: Final Design of Hybrid M-Cycle Cooler (Back View)

Now when the air stream passes through the first portion, the silica gel absorbs the moisture, but as it is an exothermic process so there is a slight increase in temperature, which is shown in the psychometric analysis. But as silica gel absorbs moisture content, it becomes saturated. When the desiccant wheel moves half a rotation then it becomes the regenerative part of the wheel. Solar air heater is used to heat the saturated silica gel, and once it unsaturated, it again acts as a dehumidifier and this process repeats in cycle. The hot air after regenerating the fan is then sucked to the outer atmosphere through the venting fans installed at the top.

The desiccant wheel is a fan shaped material which is made of carbon enforced fiber. Above that, the fan blades are coated with an hygroscopic material to absorb moisture which in result reduces humidity factor of the outlet air. These hygroscopic material have a certain limit to absorb moisture, so it is important to consistently apply the regeneration process so that desiccant wheel provides optimum performance.

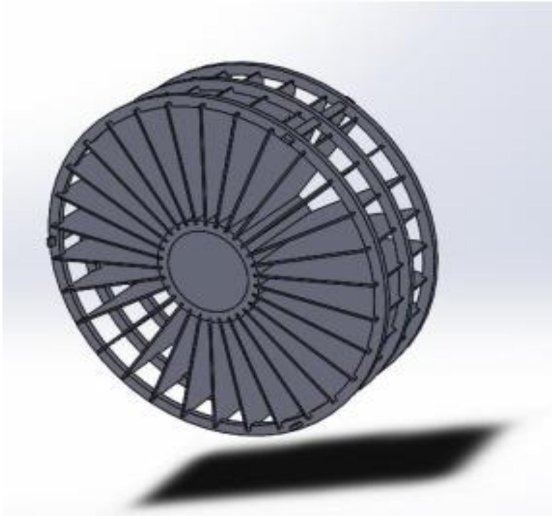

Figure 21 (a): Desiccant Wheel

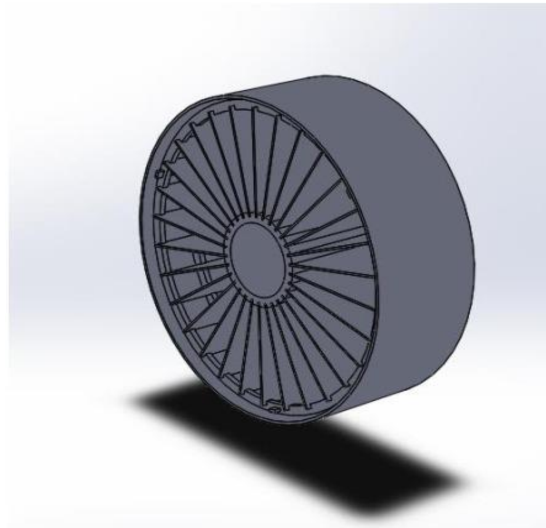

Figure 20 (b): Desiccant Wheel

The major chuck of cost is invested in the construction of desiccant wheel due to its refined development and machining. And the material used is carbon renforced fiber which becomes quite expensive as the factor of strength is added. But it has a huge advantage of long life, light weight and effective wheel. This novel design of this wheel solve problem pressure loss in the air stream. The honey comb structure causes pressure loss more than 200 Pa but our wheel causes pressure loss of less than 5 Pa.

Heat Exchanger

The inlet values of air temperature and water temperature to be used in heat exchanger are known so assuming the output temperature of air that we desire, we can find out the required area of the heat exchanger using the NTU method, as it has been done in the methodology section.

We know that

$$\mathrm{NTU} = -\ln(1 + \frac{\ln(1 - c\epsilon)}{c})$$

Whereas ϵ is effectiveness and it can be found out as:

$$\epsilon = \frac{C_{max} \times \Delta T_{min}}{C_{min} \times \Delta T_{max}}$$

and

$$c = \frac{C(Min)}{C(Max)}$$

Using the following values:

$$T_{in\text{-}air} = 45° \text{ C} = 318\text{K}$$

$$T_{out\text{-}air} = 28° \text{ C} = 301\text{K}$$

$$T_{in\text{-}water} = 25° \text{ C} = 298\text{K}$$

$$RH_{in\text{-}air} = 28\%$$

$$RH_{out\text{-}air} = 40\%$$

$$\text{Volume Flowrate}_{air\text{-}in} = 0.45 \text{ m}^3/\text{s}$$

$$\text{Volume Flowrate}_{water\text{-}in} = 0.0000695 \text{ m}^3/\text{s}$$

$$\text{Density of Water} = 998 \text{ kg/m}^3$$

$$\text{Density of Air} = 1.225 \text{ kg/m}^3$$

We get the final value of NTU;

$$NTU = -\ln\left(1 + \frac{\ln(1 - c\epsilon)}{c}\right) = 1.643409$$

Now the required area of the heat exchanger can be found as:

$$A_s = \frac{\mathrm{NTU} \times C(\min)}{U}$$

Whereas, U is the overall heat transfer coefficient which is found from:

$$\frac{1}{U} = \frac{1}{h(\mathrm{air})} + \frac{L}{k} + \frac{1}{h\ (\mathrm{water})}$$

The value of U comes out to be 200 $W/m^2.K$.

Hence, the area required comes out to be

$$A = 2.518006994\ m^2$$

Table 7: Properties of Heat Exchanger

Properties	**Values**
NTU	1.643409
Area of Heat Exchanger	2.518006994 m^2
Cooling Capacity	3920.5 W

Desiccant Wheel

The analysis of the desiccant wheel was done by using desiccant wheel simulation program by NOVEL technologies. Its credibility was judged by comparing the results of the output of this software after using some input values, with the output results that were found from the experimentation. As those results were comparable, it was known to be a credible source of information. Thus, our results were found out using the input values that we had for the desiccant wheel.

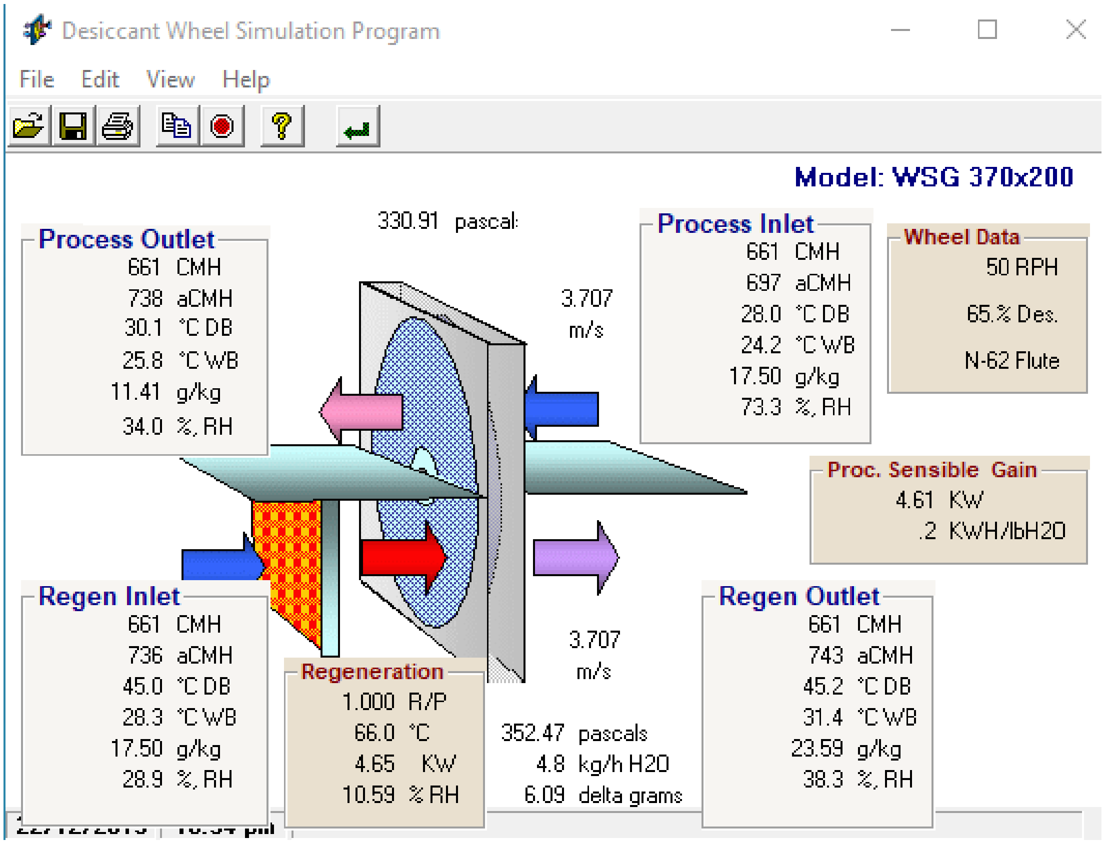

Figure 21: Desiccant Wheel Simulation Results

Following are the inputs and the outputs of this simulation in a tabular form;

Table 8: Results of Desiccant Wheel Simulation

Parameters	**Inputs**	**Outputs**
Temperature	28.0° C	30.1° C
Relative Humidity	73.3%	34.0%

Solar Air Heater

The analysis of Solar Air Heater gave the value of $\dot{m}$/A to be 0.021 kg/s-m^2, with an efficiency of 53.6%. This indicates that for the mass flowrate of 0.021 kg/s, we need to have a solar plate of 1 m^2 area to overcome the losses and deliver the necessary amount of heat required to regenerate the desiccant wheel.

Table 9: Results of Solar Air Heater

Parameters	**Inputs**
Efficiency	51.36%
Total Energy Delivered Per Area	410.8480 W/m^2
Air Flow Rate	0.0204 kg/s-m^2
Area of the Solar Plate	1 m^2

Psychometric Evaluation

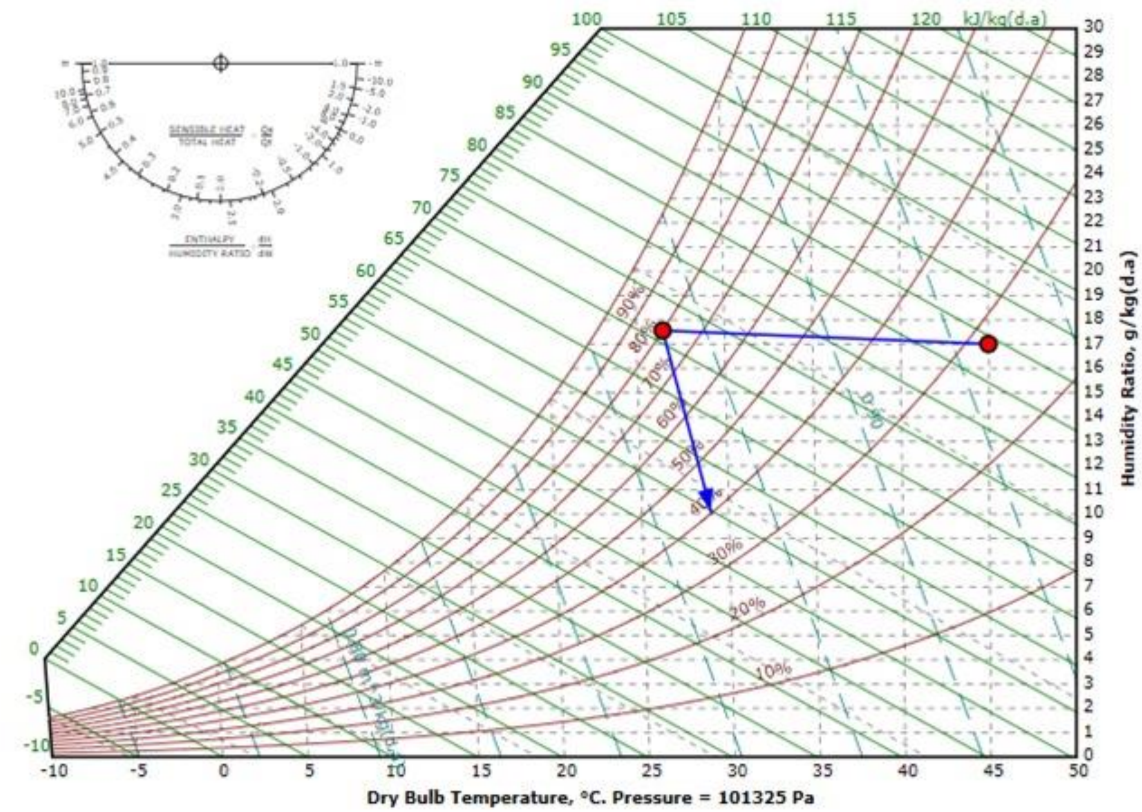

Figure 22: Psychometric Chart

The psychometric chart above shows the functionality of cooler with the relation of temperature and humidity. As we can see that initial temperature is taken to be 45° C for a hot humid climate. When the air passes through the heat exchanger, the temperature is reduced to 27° C, shown by the second red point. Then the 2nd lower line represents the effect of desiccant wheel which in turns lowers the relative humidity to be around 40% and increases the temperature just a little bit to 30.1° C.

Cooling Load

We can also find the cooling capacity of the cooler from the equation:

$$Q = \dot{m}\,(h_{in} - h_{out})$$

The values of enthalpies can be found out from the chart and the value of mass flow rate is half of the air flow rate on the outlet of the heat exchanger because half of it is recycled for better cooling. Hence,

$$h_{in} = 88{,}000 \text{ kJ/kg}$$

$$h_{out} = 57{,}600 \text{ kJ/kg}$$

$$\dot{m} = 0.13278 \text{ kg/s}$$

Thus, the cooling capacity comes out to be 3920.5 W which is equivalent to 1.11 tons of refrigeration.

Discussion

Temperature can be reduced to dew point in M-cycle cooler with low power consumption. Water is the working fluid and it doesn't add moisture to the air stream, so health issues that arise due to contaminated water is none. Moreover, as the inlet air temperature increases, its cooling capacity also increases. With least operating cost, it incorporates 100% fresh and healthy air. The conventional M-cycle is not suitable for tropical regions due to their humid climate. In order to get it to work in these conditions, it should be integrated with desiccant systems to get maximum efficiency, which we have done in our proposed prototype. The desiccant options being considered are the solid and liquid based but we have used the solid desiccant due to its more efficiency and less effect on environment. With this all, solar regeneration method is a better method than the electric heater in terms of power consumption and cost efficiency.

CHAPTER 5: CONCLUSION AND RECOMMENDATIONS

Conclusion

The Hybrid M-cycle cooler that we have developed with the desiccant wheel is designed for a small specific room with a cooling capacity of 1.11 tonne. It consumes 80% less energy than a conventional air conditioning system. This hybrid M-cycle cooling system handles the cooling load of the room with minimal power consumption as compared to the conventional compressor based air conditioning systems. This proposed prototype fulfils its purpose and the up scaling of this prototype which can handle large cooling loads is discussed in the recommendation section.

Achieved temperature and humidity within human comfort zone are;

Temperature = 25° C - 30° C

Humidity = 30 -70%

1.11 Tonne Air conditioning requires 1300 watt power by a conventional compressor based AC and our prototype will consume less than 250 watts

Energy Saved = 80%

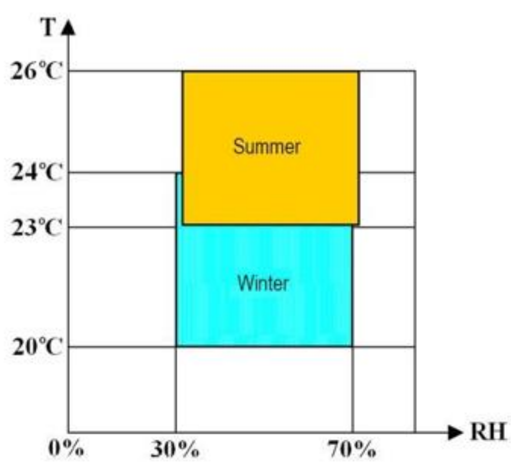

Figure 23: ISO/EN 7730 chart depicting human thermal comfort conditions

Human comfort levels are defined by standards such as ISO/EN 7730 [23]. They define the conditions in which a person can live and is comfortable. The comfort requirements are divided into regions; winters (heating season) and summers (cooling season), as depicted in the picture on the previous page.

Recommendations

Some of the recommendations and limitations of the proposed prototype are discussed in this section and are as following;

- In the future, the shape of the prototype can be altered to make it look more aesthetically pleasing. To make it a successful product on a commercial level, it needs to be made more appealing and aesthetically pleasing for the public.
- One of the problem being faced is the reduced efficiency of the cooling system. It is because of the reason 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.
- We have used solar air heater for regeneration purpose but it works only in the sunlight. To make it more stable and sustainable, we need to couple it with some energy sourcing techniques such as Phase Change Materials (PCM) which not only store the energy for future use but also eradicate the problem of fluctuations.

Future Work

There are some directions in which further research can be done to make the proposed prototype even better and more efficient.

- Improvements can be made in the design of desiccant wheel. In our prototype cooler, we have made the desiccant wheel a separate entity that can be easily integrated with the cooler for better cooling, whenever required. So the combinations of these desiccant wheels can be tested and used in future.
- In the future, the experimentation can be done on desiccant wheel blades to check its effect on dehumidification. We have used the straight blades but airfoil shaped radial blades can be tested for this purpose.
- Our prototype is designed for small rooms and in order to use it for industrial use, its capacity has to be increased to handle larger cooling loads. So, for this purpose, the upscaling of our prototype is required. The scale up of our prototype is rather very simple as just the dimensions of each component has to be increased along with the increase in inlet flow rate, which corresponds to the cooling load of the target area.

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APPENDIX I: SPECIFICATIONS OF CAD MODEL

Material of cooler	Acrylic
Height of cooler	18 cm
Dimension of Heat Exchanger	55.5*35.5*32 cm
Diameter of DW	35 cm
No. of blades of DW	28

APPENDIX II: DATA FOR SOLVING HEAT EXCHANGER

Density of water	998 kg/m^3
Density of air	1.225 kg/m^3
Specific heat of water at constant pressure	4.18 kJ/kgK
Specific heat of air at constant pressure	1.005 kJ/kgK

APPENDIX III: DATA FOR SOLVING 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
Gap between Glass & Plate	10 cm