Experimental Study on Thermal Performance of Liquid Desiccant Cooling System

Dissertation-II

Submitted in partial fulfilment of requirement for the award of the degree of

Master of Technology in Thermal Engineering

by

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CERTIFICATE

I hereby certify that the work being presented in the dissertation entitled **"Experimental study on thermal performance of liquid desiccant cooling system"** in partial fulfilment of the requirement for the award of Degree of **Master of Technology** and submitted in Department of Mechanical Engineering, Lovely Professional University, Phagwara, is an authentic record of my own work carried out under the supervision of **Mr. Sanjeev Kumar**, **Assistant Professor**, Department of Mechanical Engineering, Lovely Professional University. The matter embodied in this dissertation has not been submitted in part or full to any other University or Institute for the award of any degree.

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Signature of Examiner

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Abstract

Liquid desiccant dehumidification system (LDDS) integrated with vapour compression refrigeration system (VCRS) has been experimentally investigated. A honeycomb PVC fills are used for structural packing of dehumidifier/regenerator. It provides the sufficient surface area for proper mixing of desiccant solution and process air for optimum moisture exchange. The moisture present in the air is removed by liquid desiccant in dehumidifier unit while the diluted desiccant solution is regenerated by transferring the moisture absorbed by the desiccant solution to the surrounding air in the regenerator. Calcium chloride ($CaCl_2$) is used as a desiccant to perform the operation of LDDS and R134a is used as a refrigerant in the VCRS to cool the hot and dehumidified air. LDDS is used to improve the outdoor air quality by eliminating moisture present in the process air and VCRS deals with sensible cooling of air. In this study, moisture removal rate (MRR) and dehumidification effectiveness (ε_{deh}) are calculated to evaluate the performance of dehumidifier. Also the coefficient of performance (COP) of system and COP of VCRS without LDDS are calculated. These parameters are studied at different air temperatures and velocity. MRR decreases from 0.92 to 0.72g/s, dehumidification effectiveness decreases from 0.135 to 0.088, COP of system increases from 2.7 to 4.86, and COP of VCRS without LDDS increases from 2 to 4.7, when the unsaturated air temperature varies from 30 to 40˚C at constant air velocity of 4.0m/s. MRR increases from 0.884 to1.069g/s, dehumidification effectiveness decreases from 0.149 to 0.120, COP of system increases from 3.64 to 4.82, and COP of VCRS without LDDS increases from 3.4 to 3.92, when the unsaturated air velocity varies from 3 to 4.5m/s by keeping constant air temperature of 35˚C. MRR decreases from 1.058 to 0.788g/s, dehumidification effectiveness decreases from 0.148 to 0.061, COP of system increases from 2.75 to 5.84, and COP of VCRS without LDDS increases from 2.41 to 5.27, when the saturated air temperature varies from 30 to 40˚C at constant air velocity of 4.0 m/s. MRR increases from 0.871 to 1.104g/s, dehumidification effectiveness decreases from 0.133 to 0.098, COP of system increases from 3.59 to 4.54, and COP of VCRS without LDDS increases from 3.36 to 4.20, when the saturated air velocity varies from 3 to 4.5m/s by keeping constant air temperature of 35˚C.

Nomenclature

m_a	mass flow rate of air (kg/s)
w_a	specific humidity of air (g/kg of dry air)
A	cross sectional area of duct at inlet of dehumidifier (m^2)
v	velocity of air (m/s)
P	atmospheric pressure (kpa)
R_{spec}	specific gas constant for air (J/kg K)
T	temperature of air (˚C)
P_d	partial pressure of dry air (kpa)
P_v	partial pressure of water vapour (kpa)
M_d	molar mass of dry air (kg/mol)
M_v	molar mass of water vapour (kg/mol)
R	universal gas constant (J/K.mol)
P_{sat}	partial pressure of saturated air (kpa)
P_{wo}	saturation pressure of pure water (i.e. x = 0) at the desiccant solution temperature (kpa)
P_{wx}	saturation pressure of the desiccant solution of concentration x at the desiccant solution temperature (kpa)
w_{eq}	humidity ratio of air in equilibrium
x	desiccant solution concentration (wt %)
h_a	specific enthalpy of air (kJ/kg)
W_{in}	work input (KW)

Greek letters

ε_{deh}	dehumidification effectiveness
ρ	density (kg/m^3)

ϕ	relative humidity

Subscripts

a	air
eq	equilibrium
deh	dehumidifier
i	inlet
o	outlet
in	input
sat	saturated
spec	specific

Abbreviations

LDDS	liquid desiccant dehumidification system
MRR	moisture removal rate
DE	dehumidifier effectiveness
VCRS	vapour compression refrigeration system
ABS	acrylonitrile butadiene styrene

CHAPTER 1

1. Introduction

Energy deficiency is observing a major problem of increasing anxiety in the world, along with the possibility of trouble toward the atmosphere produced by the traditional refrigerants. Therefore the requirement to create the novel substitute energy preserving along with eco-friendly air-conditioning systems has become more significant. The liquid desiccant cooling systems (LDCS) consumed low-quality heat sources can sufficiently accomplish those requirements and concurrently the unwanted heat liberated from the plants can find application in these system where the utilization of brine solution as an absorbent poses no hazard to the environment.

Desiccants have the property of absorbing and holding moisture from the surrounding air when brought into contact with it because they have high attraction towards water vapour. Liquid desiccant is used to extract water vapour from the surrounding air for dehumidification to occur. Dehumidification of the surrounding air by desiccant takes place due to the difference of pressure between water vapour of surrounding air and liquid desiccant. So, the water vapour transfers from greater vapour pressure to lesser vapour pressure and the desiccant can hold the moisture till the partial pressure of water vapour on both sides are equalized. This ability of desiccant to remove moisture from surrounding air leads to fluctuating the latent load from refrigerating cycle to desiccant cycle and to energy saving by decreasing latent load. Liquid desiccant cooling system (LDCS) is supposed to be effective when latent heat ratio is high and sensible heat ratio is low.

There are two types of air conditioning loads such as sensible load and latent load (about 40 %) these loads are compensate by an air conditioner to sustain the wanted inside conditions. So as to eliminate the latent load, the conventional vapour compression refrigeration system (VCRS) or vapour absorption refrigeration system (VARS), cools the surrounding air lower than its dew point for condensing out the water vapour present in the surrounding air and afterward the dehumidified air is reheated to reach the necessary indoor temperature conditions.

Moreover, vapour compression refrigeration systems are driven by electricity which is generated by the high grade heat source functioned power plant with the subsequent releases of CO2 and other dangerous gases into the environment. In conclusion, the refrigerants consumed in this air conditioning system are high or low CFCs based which are responsible for depletion of ozone layer. Hence the liquid desiccant cooling system can be a perfect complement to the conventional vapour compression air conditioning system to beat the properties of its disadvantages. It is very significant that when the low grade energy sources such as solar energy and waste heat that can be used for powering, it can extremely diminishes the functioning prices and rises substantially the approachability to the air conditioning for the people alive in inaccessible zones, mainly in growing nations.

1.1. Principle of desiccant cooling

The desiccant cooling involves in dehumidifying the entering air stream by pushing it over a desiccant material and after that drying the air to the wanted indoor temperature. In order to prepare the system functioning repeatedly, the absorbed moisture need to be pushed out from the desiccant solution to adsorb the moisture in the succeeding cycle, which is carried out via reheating the desiccant solution up to its regeneration temperature. Sensible heat ratio (SHR) play a very important role because the efficiency of a desiccant cooling system mainly depends on it. Principles of desiccant cooling as shown in Figure 1 that includes three components, viz. desiccant dehumidifier, regeneration heat source, and cooling unit as defined in the following.

1.1.1. Desiccant dehumidifier

The desiccant dehumidifier is commonly a gradually revolving desiccant wheel or a repeatedly regenerated adsorbent bed in case of solid desiccant. In case of liquid desiccant, the dehumidifier/absorber is an arrangement in which the liquid desiccant mixes with process air stream. The probable arrangements of dehumidifier contains finned-tube surface, absorber, spray tower, packed tower and coil-type. Packing style of packed towers may be either regular or irregular.

1.1.2. Cooling unit

The cooling unit can be an evaporative cooler or a cold coil and the evaporator of a traditional air conditioner. The desiccant is used to remove the latent load while the cooling unit is used to control the sensible load.

1.1.3. Regeneration heat source

Heating sources provides the thermal energy essential for removing the moisture from desiccant solution after absorbing the moisture in the dehumidifier. Solar energy, waste heat, natural gas etc. are the important sources of the thermal energy. The heat available for regeneration is used to increase the temperature of diluted desiccant solution within the arrangement of a regenerator in which the process air simultaneously blown to move away the water vapour from desiccant solution.

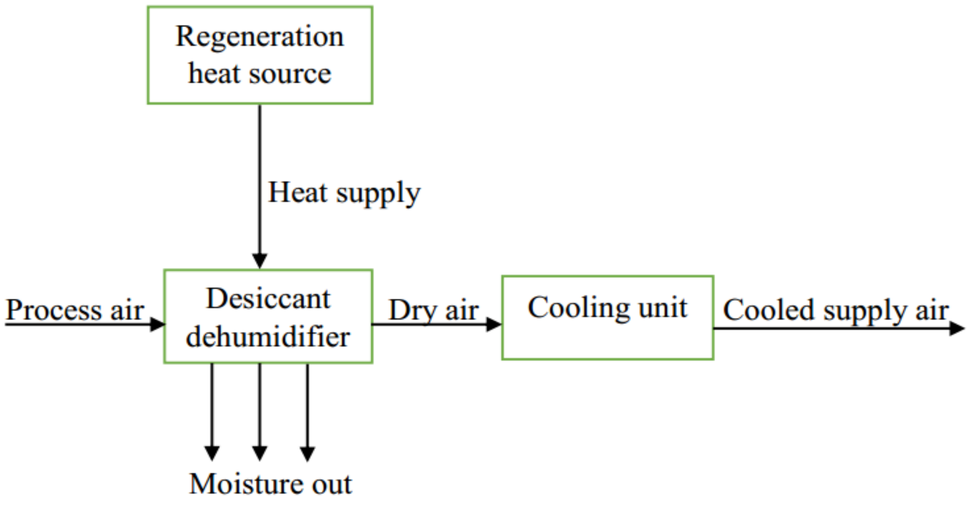

Figure 1: Principle of desiccant cooling

1.2. Liquid desiccant cooling system (LDCS)

Dehumidifier and regenerator are main component of a simple liquid desiccant cooling system as shown in Figure 2. The water vapour present in the process air is absorbed by liquid desiccant in the dehumidifier unit.

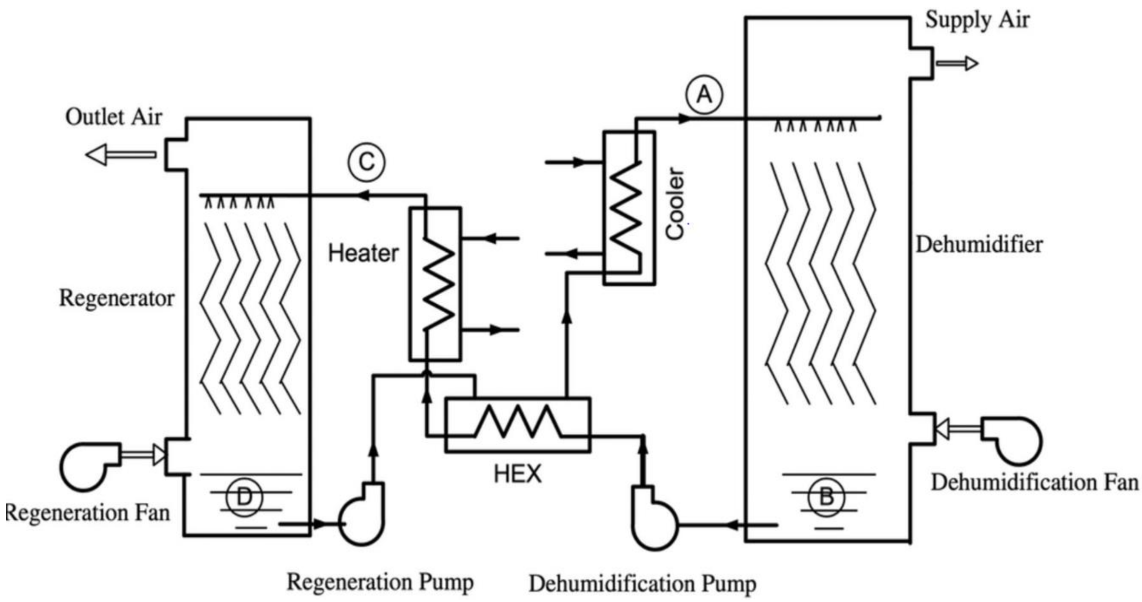

Figure 2: Schematic diagram of liquid desiccant cooling system [34]

The water vapour present in the process air transfers to the desiccant solution due to vapour pressure difference. Heat is released during condensation of water and heat transfer takes place due to mixing of desiccant and moist air. The air is imported to the conditioned space after dehumidification while the diluted or weak desiccant solution is fed to the regenerator with the help of pump. The diluted desiccant solution primarily passed through a heat exchanger before fed to the regenerator and after that passed through a heating coil to increase its temperature. The hot diluted desiccant solution is brought to process air in the regenerator where the water vapour shifted from the diluted desiccant solution to the process air due to vapour pressure difference. After removal of water vapour from diluted desiccant solution, the concentrated desiccant solution before enters to the dehumidifier unit, passes through heat exchanger and cooling coil. The pre-heating of the diluted desiccant solution and pre-cooling of the concentrated desiccant solution is done by using the liquid–liquid heat exchanger.

Variations of vapour pressure of the desiccant solution during the dehumidification and regeneration process as shown in Figure 3. The liquid desiccant solution enters into the dehumidifier in state A, when it has higher concentration and lesser vapour pressure as compared to moist air.

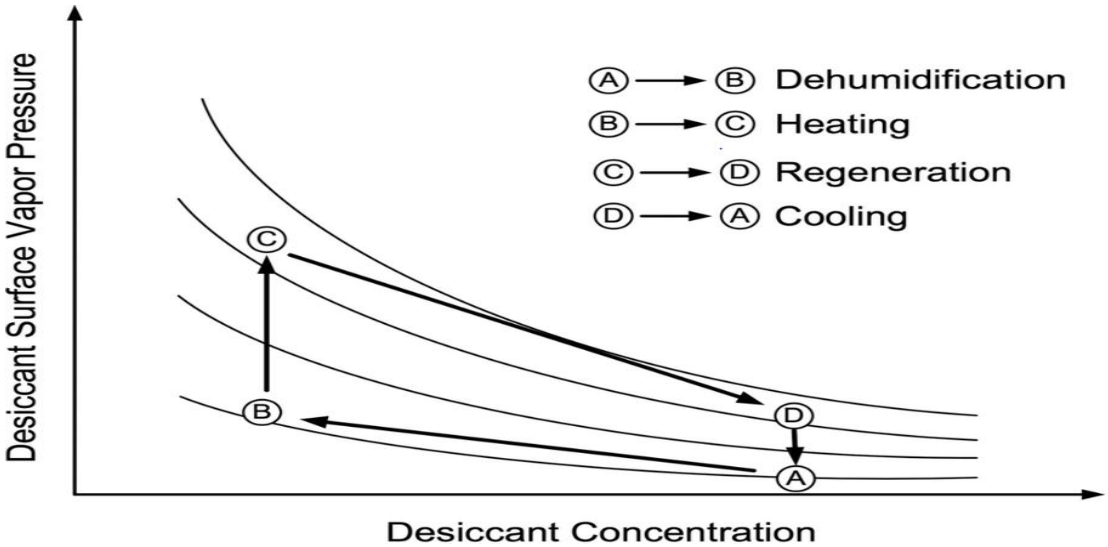

Figure 3: Vapour pressure variations in the desiccant cooling system [34]

Desiccant solution absorbs the moisture from the moist air during dehumidification process and reaches to State B with lesser concentration and greater vapour pressure which is changed further by heating the desiccant solution before it goes into regenerator in state C.

During this state, vapour pressure of desiccant solution is greater than moist air due to which it gives the absorbed water vapour to the moist air, therefore its vapour pressure decreases and concentration rises and it reaches to state D. After this state it is cooled to decrease its vapour pressure further.

1.2.1. Dehumidifier unit

Heat and mass transfer process takes place in the dehumidifier from inlet moist air to liquid desiccant, whereas the heat exchange amongst moist air and desiccant solution takes place due to temperature difference. Mass transfer amongst moist air and desiccant solution takes place due to vapour pressure difference. The process of mass transfer is explained by various theories such as film theory, surface renewal theory, and penetration theory. The most commonly used dehumidifier units consist of coil-type absorber, finned-tube surface, packed tower, and spray tower.

Dehumidifiers must have the following features for better performance:

I. It should have the high rates of heat and mass transfer.
II. There will be a lesser pressure drop between moist air and desiccant solution when passing through the dehumidifier.
III. It must have the huge contact surface area per unit volume.
IV. Dehumidifier should be made of a material suitable for the liquid desiccant to avoid the corrosion.
V. Dehumidifier material should be cheap.
VI. No carryover of liquid desiccant with the moist air.

1.2.2. Types of dehumidifier

I. Direct-contact liquid desiccant dehumidifier
II. Indirect-contact liquid desiccant dehumidifier
III. Adiabatic dehumidifier
IV. Internally cooled dehumidifier

I. Direct-contact liquid desiccant dehumidifier

In direct-contact liquid desiccant dehumidification systems, two packed beds are employed as a dehumidifier and a regenerator. A liquid desiccant solution is sprayed through the structural packing of bed and comes in direct contact with the process air, which is blown through the bed. During the dehumidification process, the warm and humid outdoor air enters the dehumidifier and mixes after direct contact with the sprayed desiccant solution onto bed surface. The system efficiency depends on various parameters like packing density, bed length, air inlet temperature, desiccant solution inlet temperature, desiccant solution type, desiccant solution concentration, air and desiccant solution flow rates etc. Cellulose packing is commonly used in packed bed direct-contact dehumidifier system as shown in Figure 4.

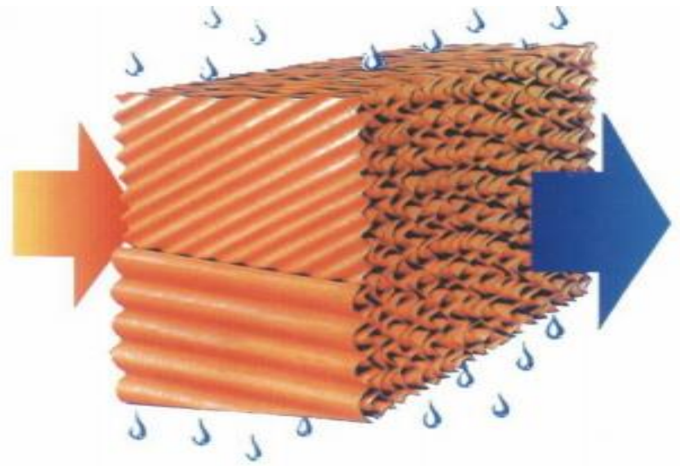

Figure 4: Cellulose packing

The advantages of packed bed are follows:

- Expose large surface area of the desiccant solution to the air
- Compactness
- High efficiency
- Large contact area
- Large contact time

It should be mentioned that a main issue about direct contact liquid desiccant systems is the carry-over problem where some desiccant solution droplets are carried out by the process air, which can be harmful for people's health and can cause corrosion in the ducting system. However, the use of drift eliminator can prove to be effective to some extent for the carry-over problem. Furthermore, high air pressure drops occurs in these system which increase the operating costs. Liquid to-air membrane energy exchangers (LAMEEs), which use a semipermeable membrane to separate the process air and the desiccant solution streams, eliminate the problem of desiccant droplet carry-over associated with direct-contact liquid desiccant energy exchangers. In LAMEE, desiccant solution and air stream remain separated from each other with the help of membrane and moisture transfer takes place through the pores available in membrane. Figure 5 shows the air and desiccant flows through the structure of LAMEE.

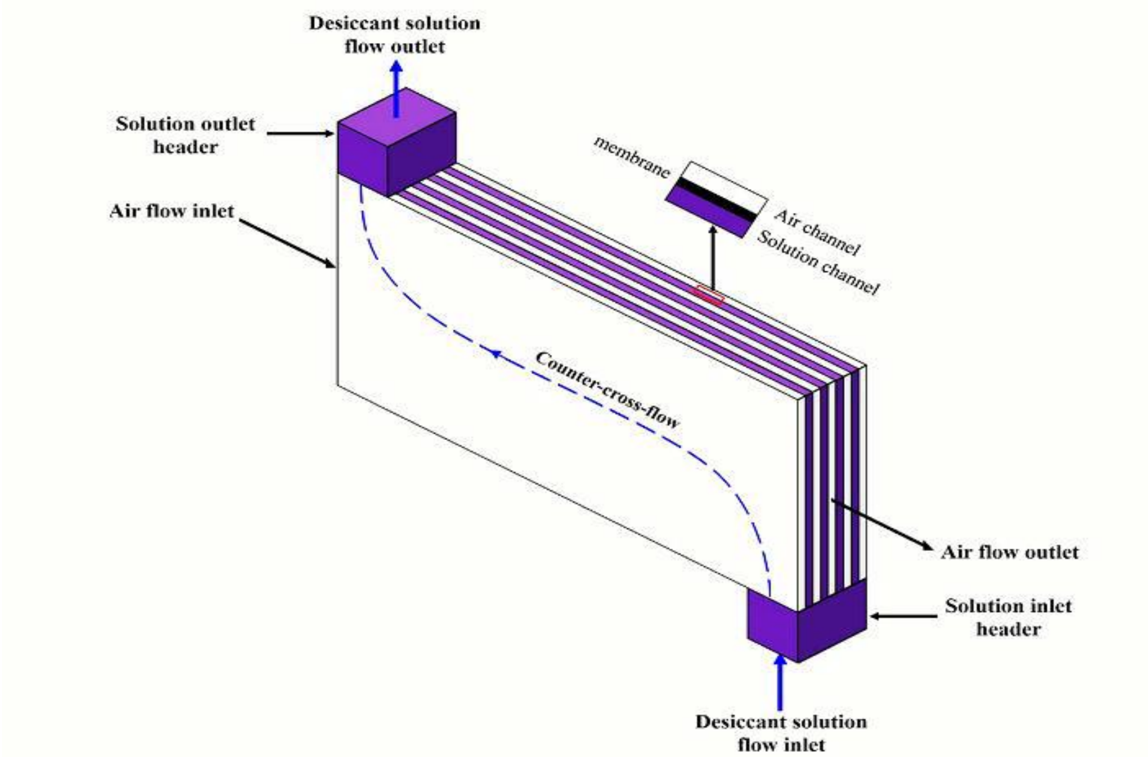

Figure 5: Structure of flat plate LAMEE [35]

II. Indirect-contact liquid desiccant dehumidifier

In indirect contact liquid desiccant dehumidifier, the moist air and desiccant solution will not mix with each other. The desiccant solution is made to pass through membrane or pads and moist air from the surrounding.

III. Adiabatic dehumidifier

The adiabatic dehumidifier without internal cooling unit as shown in Figure 6. It is a simple unit in which water vapour present in the moist air transfer to the desiccant solution while heat transfer takes place due to latent heat of condensation of water vapour and temperature difference. The dehumidification efficiency decreases due to release of heat. The solution that is used to control the temperature in dehumidifier also increases the flow rate that leads to rise the probability of carry-over. The greater flow rates in dehumidifier are tracked by greater flow rates in regenerator that leads to decreases the thermal coefficient of performance of the system.

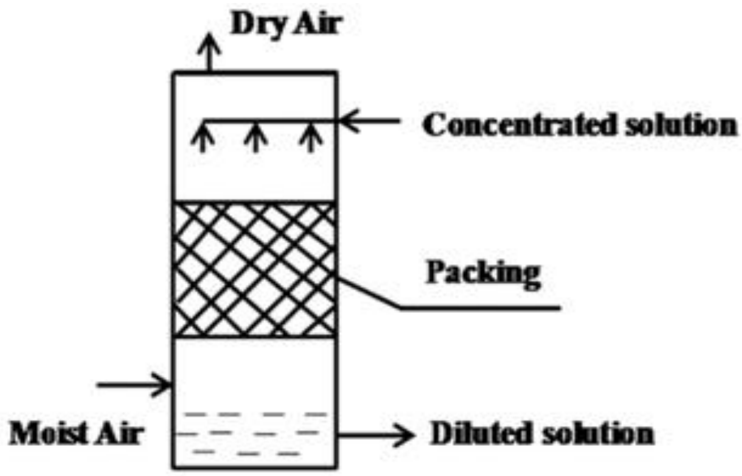

Figure 6: Adiabatic dehumidifier [36]

IV. Internally cooled dehumidifier

An internally cooled dehumidifier with cooling unit as shown in Figure 7. The internally cooled dehumidifier contains an embedded cooling coil that offers the chilled water or air to reject the heat generated throughout the process of dehumidification. The temperature of the desiccant solution and moist air is controlled by cooling unit that leads to increasing the efficiency of the system. The key benefit of cooing coil is that it permits for low flow rates which increases the performance of system and decreases the chance of carry-over.

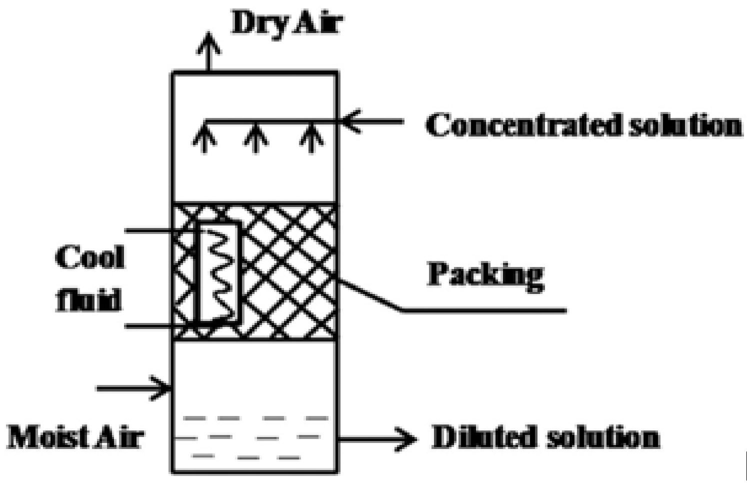

Figure 7: Internally cooled dehumidifier [36]

1.2.3. Flow patterns inside the dehumidifier

The flow patterns of the moist air and desiccant solution as shown in Figure 8. There are three types of flow patterns in the dehumidifier:

I. **Parallel flow:** In parallel flow, desiccant solution and air flows in same direction.

II. **Counter flow:** In counter flow, desiccant solution and air flows in opposite direction.

III. **Cross flow:** In cross flow, desiccant solution and air flows perpendicular to each other.

The contact surface and the process of interaction between desiccant solution and inlet process air are decided by flow patterns. Mathematical model that can be used for an individual liquid desiccant cooling system is also decided by flow patterns. Counter flow is broadly used flow pattern for design purposes of dehumidifier.

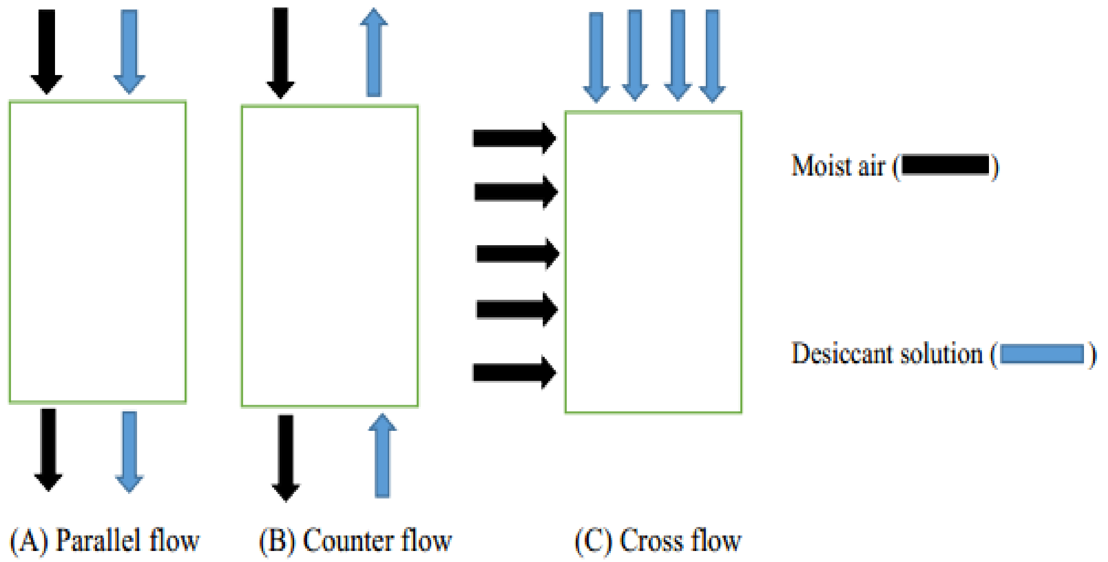

Figure 8: Flow patterns of moist air and desiccant solution in dehumidifier unit

1.2.4. Regenerator unit

The diluted or weak desiccant solution is converted into concentrated or strong solution in the regenerator as shown in Figure 2. Both the regenerator and dehumidifier units are same but the basic operation and process of these two units are reverse to each other. One more dissimilarity is that a dehumidifier unit has an extra coating of isolation to decrease the heat and mass transmission from atmosphere.

In general, desiccant solution is pre-heated before entering to regenerator. The moisture is moved from warm desiccant solution to entering process air at an average temperature that's why a few desiccant regenerators also uses the pre-heated air to regenerate the liquid desiccant solution in case of solid desiccant rather than liquid desiccant regenerator.

1.2.5. Advantages and disadvantages of liquid desiccant cooling system (LDCS)

A liquid desiccant cooling system has the following advantages and disadvantages:

Advantages

I. It has large moisture elimination ability with small regeneration temperature requirement.
II. The desiccant materials are very efficient antifreeze, hence the conditioners can carry on its process without icing or freezing difficulties.
III. It has lower pressure drop in the process air stream.
IV. It utilizes low free energy sources for regeneration process.
V. The entire diluted desiccant solution need not to be circulated for regeneration, only a lesser quantity of desiccant solution can be over-concentrated and mixed with diluted desiccant solution.
VI. The liquid desiccants used in this system have the capability to clean bacteria, viruses, dirt, and dust from the moist air stream which can disturb the human health.
VII. Liquid desiccant conditioners generally have great contact efficiency. Thus, air retains approximately the similar temperature and humidity ratio from inlet to outlet.
VIII. It has greater thermal coefficient of performance (COP) as compare to the solid desiccant cooling systems.

Disadvantages

I. A huge volume of desiccant solution circulated in the system needs a large pumps with large power.
II. To cool the desiccant solution before supplied into the dehumidifier, a separate heat exchanger is needed.
III. Carryover of desiccant solution droplet with the process air may be harmful for human health.

1.3. Desiccant

Desiccant is a material used for removing the moisture from the process air. Desiccants have the property of absorbing and holding moisture when brought into contact with the process air because they have high attraction towards water vapour. The moisture transfer from process air to the desiccant due to vapour pressure difference.

1.3.1. Types of desiccants

There are two types of desiccants:

I. Solid desiccant
II. Liquid desiccant

I. **Solid Desiccant:** The commonly available solid desiccants are silica gel, zeolite, activated alumina and carbon. Generally, solid desiccants can hold extra water vapour and stands high drying capacity as compare to the liquid desiccants. The solid desiccant are compact and it has less corrosion and carryover problem. Gradually revolving wheel layered by desiccant is used for the arrangement of solid desiccant system in which a portion of it capturing the entering process air stream whereas the remaining part of it is being regenerated.

II. **Liquid Desiccant:** Most commonly available liquid desiccants are LiCl, LiBr, CaCl2, and TEG. The other liquid desiccants comprise KCOOH, glycols like MEG (monoethylene glycol) and DEG (diethylene glycol), propylene glycol and combinations of desiccants. The liquid desiccants are sprayed into process air streams or onto the wetted contact surfaces to absorb water vapour from the incoming air and after that they need to be regenerated in a regenerator where water vapour previously absorbed by diluted desiccant is evaporated out with process air by heating.

1.3.2. Properties of liquid desiccants

The liquid desiccant play a very vital role in a desiccant cooling system due to its properties such as dynamic viscosity, conductivity, specific heat capacity, density as well as its operating constraints like regeneration temperature, boiling point elevation and energy storage density that explain its potential for usage as liquid desiccant. The surface vapour pressure is one of the greatest important factors amongst all the properties that leads to heat and mass transmission in the dehumidifier. There are various types of liquid desiccants such as lithium chloride, lithium bromide, and calcium chloride commonly used in liquid desiccant dehumidification system due to the lower vapour pressure. Out of all the three liquid desiccants, calcium chloride ($CaCl_2$) has the least absorption capability, though $CaCl_2$ is most common because it is inexpensive and easily accessible. Moreover, LiCl has a very lower vapour pressure and more stable as compared to the other desiccants.

A good liquid desiccant must have the following properties:

I. It can hold large quantity of water vapour
II. Large permeation absorption capability
III. Low regeneration temperature
IV. Low Viscosity
V. Low vapour pressure
VI. Low crystallization point
VII. High density
VIII. High durability
IX. Higher heat transfer rate
X. Non-volatile
XI. Non-corrosive
XII. Non-toxic
XIII. Non-flammable,
XIV. Odorless and stable
XV. Low-cost

1.3.3. Advantages and disadvantages of liquid desiccant

Advantages and disadvantages of liquid desiccants are follows:

Advantages

I. Liquid desiccants have the lower drop of pressure across the system that makes them applicable to use with low regeneration temperatures.
II. Liquid desiccants are easily pumped that makes the whole unit small and compact.

III. The collected liquid desiccant can be used in the absence of heat source for regeneration.

IV. Ease of manipulation and mobility.

V. It has a capability to remove dirt, dust, bacteria coming with the process air.

Disadvantages

I. All the liquid desiccants such as LiCl, LiBr, $CaCl_2$ and others are corrosive in nature and can damage the system.

II. The problem of carryover of desiccant with the process air can cause major damage on the health of inhabitants.

III. Huge pumps with large power are necessary to control huge capacity of liquid desiccant.

IV. The problem of crystallization also faced by liquid desiccants.

CHAPTER 2

2. Scope of the study

Liquid desiccant cooling systems are eco-friendly and utilizes low grade energy for its operations as compare to the vapour compression refrigeration system (VCRS) and vapour absorption refrigeration system (VARS). The liquid desiccant used in this system does not deplete the ozone layer. The liquid desiccant properties play a very important role on the performance of the system.

So, the scope of our work are following:

I. A low grade energy driven liquid desiccant air conditioning system may be an attractive substitute to the compressor-based technology used in most HVAC applications.

II. A liquid desiccant cooling system can be used for large humidity requirements.

III. It provides better quality of dry and cool air for human comfort and it does not have any harmful effects on the human health.

IV. The proposed LDDS combined with VCRS can also be used for small capacity cooling purpose.

CHAPTER 3

3. Objectives of the study

Due to effective usage of low grade energy, liquid desiccant dehumidification technology is becoming day by day attractive. The necessity is to make liquid desiccant dehumidification system (LDDS) more energy proficient and cost effective than conventional vapour compression refrigeration system (VCRS) and vapour absorption refrigeration systems (VARS). So, the objectives of our work is to:

I. Study the performance of structured packed LDDS integrated with VCRS.

II. Calculation of parameters like moisture removal rate (MRR) and dehumidification effectiveness (ε_{deh}) to evaluate performance of dehumidifier.

III. Calculation of coefficient of performance (COP) of system and COP of VCRS without LDCS.

IV. Comparison of COP of system and COP of VCRS without LDDS.

V. Study the variation of moisture removal rate (MRR), dehumidification effectiveness (ε_{deh}), coefficient of performance COP of system and COP of VCRS without LDDS with respect to different inlet parameters such as air temperature and velocity.

CHAPTER 4

4. Literature review

This chapter is related to review of previous work, done by various researchers in the past, related to liquid desiccant based cooling system. The literature review is categories in three sections such as dehumidifier/regenerator, hybrid liquid desiccant cooling system and properties of liquid desiccant. The details of literature review are discussed as follows.

4.1. Dehumidifier/Regenerator

[1] Jain et al. (2000) experimentally investigated the dehumidifier and regenerator by using LiBr-water solution as a desiccant in a liquid desiccant cooling system. They experimentally examined the falling film tubular absorber acts as a dehumidifier and a falling film plate acts as a regenerator. They calculated two wetness factors F_w and F_h to account for the inappropriate wetting of the available area. They concluded that the higher values of F_w and F_h provides the more uniform wetting which enhances the performance of the system.

[2] Sultan et al. (2002) studied the performance of packed tower regenerator by using $CaCl_2$ as a liquid desiccant and presented the hypothetical model that stating the influence of the system constraints. To study the influence of air and liquid desiccant constraints on the output variables, the experimental results were plotted. The inlet air conditions such as temperature, humidity and flow rate play a very significant role in the regeneration process. They concluded that moisture transfer rate from the desiccant was improved by rise in inlet air flow rate and inlet solution temperature while the regeneration rate was reduced by rise in inlet air humidity ratio and inlet solution concentration.

[3] Fumo and Goswami (2002) investigated the performance of a packed tower dehumidifier and regenerator by using an aqueous lithium chloride as a liquid desiccant. Effectiveness of the dehumidification and regeneration processes as well as the rates of dehumidification and regeneration were determined under the effects of variables such as air temperature and humidity, desiccant temperature and concentration, and air and desiccant flow rates. They concluded that the mass flow ratio of air to the desiccant solution varied from 0.15 to 0.25 for air dehumidification as well as desiccant regeneration which was lesser than the MR values of 1.3 to 3.3 used in most other studies.

[4] Elsarrag et al. (2005) investigated the guidelines for designing as well as performance study of a structured packing (cellulose rigid pad) liquid desiccant cooling system by using TEG as a desiccant. For instantaneous heat and mass transfer amongst process air and desiccant solution in a dehumidifier were conducted by experimental and theoretical evaluations. They concluded that high liquid flow rates do not affect the performance of the system if the value of ratio of liquid to airflow ratio does not exceed 2, and reduction in wet-bulb temperature in the dehumidifier was achieved in the range of 4.5˚C to 9˚C.

[5] Yin et al. (2007) experimentally investigated the dehumidifier and regenerator by using LiCl as a desiccant for liquid desiccant air conditioning system. Dehumidification and regeneration rates were examined under the effects of air temperature and humidity, heating source temperature, desiccant solution concentration and desiccant solution temperature. Maximum tower efficiency was found with the sufficient inlet humidity of the indoor air during the experiments of dehumidification. They suggested that the average mass transfer coefficient of the packing regenerator was 4 g/(m^2 s) on the basis of experimental results. They also concluded that the maximum mass transfer coefficient 7.5 g/(m^2 s) was achieved with the 20% of desiccant solution mass concentration and 77.5˚C of heating temperature.

[6] Jain and Bansal (2007) carried out an experimental study on packed bed dehumidifier with three frequently used liquid desiccant such as TEG, LiCl and $CaCl_2$ by means of experimental relationships for dehumidification effectiveness from literature survey. Experimental comparison of performance of dehumidifiers discloses the suitable choice of kind of column with its functioning constraints comprising solution to air flow rates, inlet concentration of desiccant solution and packing dimensions leads to great dehumidification effectiveness of around 0.9 or more. They concluded that large deviations in dehumidification effectiveness values that varied from 10% to 50% or more when larger variations happening for lesser ratios of liquid to gas flow rates.

[7] Liu et al. (2007) done an experimental study on heat and mass transmission between air and liquid desiccant by using LiBr as a desiccant in a cross-flow regenerator with celdek structured packing. Mass transfer performance of the regenerator was defined by considering the moisture removal rate and regenerator effectiveness. For decent agreement amongst expected values and investigational data with correlation coefficient of 0.962, a dimensionless mass transfer relationship was anticipated. They suggested that the water

vapour elimination rate rises with growing air flow rate, desiccant flow rate and inlet desiccant temperature, reduces with humidity ratio of inlet air and inlet desiccant concentration, and changes minute with inlet air temperature. They also suggested that the regenerator effectiveness rises with desiccant flow rate and inlet concentration, reduces with air flow rate and desiccant inlet temperature, and changed minute with inlet air temperature and humidity ratio.

[8] Zhang et al. (2010) studied the mass-transfer features of a dehumidifier/regenerator with structured packing by using lithium chloride as a desiccant. Air dehumidification and desiccant regeneration were carried out experimentally in a typical working ranges of air conditioning applications. They suggested that with the increase in air velocity from 0.5 to 1.5 m/s, mass transfer coefficient in the dehumidifier and regenerator was varied from 4.0 to 8.5 g/m^2s and from 2.0 to 4.5 g/m^2s, respectively. Overall mass-transfer coefficients decreases due to higher solution temperature. The variations between predicted values and experimental values were in the range of ±20%.

[9] Bansal et al. (2011) experimentally compared the performances of an adiabatic and an internally cooled dehumidifier with structured packing by using $CaCl_2$ as a desiccant with a cross-flow arrangement. Moisture removal rate, dehumidifier effectiveness and mass transfer coefficients were evaluated to compare the performances of the dehumidifier with and without internal cooling. They suggested that the maximum effectiveness and moisture removal rate were achieved within internal cooling.

[10] Bassuoni (2011) carried out an experimental study on structured packing dehumidifier/regenerator having a density of 390kg/m^3, corrugation angle of 60° and void fraction of 0.88 by using $CaCl_2$ as a liquid desiccant. Performance of the system was evaluated in terms of moisture removal rate, effectiveness of dehumidifier, mass transfer coefficients and COP. They concluded that the increase of mass transfer coefficient and MRR for both dehumidifier/regenerator was established by increasing both air and solution flow rates. They also suggested that payback period (PP) of DDS was found to be 11 months with yearly running cost savings of around 31.24% as compared to the vapour compression system (VCS).

[11] Yin et al. (2011) carried out the performance analysis of a regenerator with desiccant solution (LiCl-H_2O) by utilizing hot air mathematically. Hot air temperature was nearby 65˚C at lowest inlet solution temperature. The important effects of operational parameters were investigated to evaluate the thermal efficiency of regeneration and rate of regeneration. They suggested that thermal efficiency of regeneration would be optimum when the ratio of air to desiccant flow rate was around 8.

[12] Gao et al. (2013) experimentally investigated a partially internally cooled dehumidifier by evaluating dehumidification effectiveness and moisture removal rate and compared with adiabatic one. Cooling water was supplied only to the latter part of dehumidifier for energy saving which considerably enhanced its performance. They calculated a dimensionless Sherwood number (S_h) which was interrelated by Schmidt number, Reynolds number, water content of the desiccant, and flow rate ratio of the desiccant over air. They concluded that the dehumidification effectiveness and water vapour elimination rate within an internally cooled dehumidifier were increased due to lower cooling water temperature. They also suggested that the deviations between expected value and investigational data were in the range of ±20%.

[13] Vias and Thakur (2014) studied the performance of a PVC frill structured packing regenerator with cross-flow arrangement by using TEG as a liquid desiccant. Various parameters such as moisture removal rate, concentration change and regenerator effectiveness were calculated to evaluate the performance of the system. They suggested that moisture removal rate was increased due to increase in mass flow rate of air while the moisture removal rate was decreased due to rise in inlet air temperature.

[14] Kim et al. (2015) investigated a basic model for packed-bed tower regenerator by using lithium chloride (LiCl) as a liquid desiccant. It was found that greater temperature and lesser concentration of the desiccant solution leads to greater vapour pressure and greater equilibrium humidity ratio, resulting in high regeneration rates. They concluded that adequate regeneration can be achieved at high inlet air temperature, low concentration, and high solution mass flow rate when the humidity ratio of inlet air is lesser than a definite value.

[15] Liu et al. (2015) carried out experimental investigation on internally-cooled dehumidifier made of thermally conductive plastic that can sustain superior corrosion resistance by using lithium bromide (LiBr) as a liquid desiccant. Experimental as well as simulation investigation were performed to calculate the system performance. Water vapour elimination rate, volumetric mass transfer coefficient and dehumidification/regeneration efficiency were studied in conjunction with the inlet constraints to define the performance of the system. They suggested that heat and mass transfer coefficient was enhanced by increasing the flow rate.

[16] Xian Li et al. (2016) studied a dynamic modeling of dehumidifier based on the heat and mass transfer principles in which the spatial differentials of fluid properties were approximated by discretization and dynamical interactions along the flow direction. Firstly all the unknown model constraints were calculated from static experimental data by Levenberg-Marquardt algorithm and after that it refined from dynamic experimental data by extended Kalman filter. They concluded that the proposed model performs well in the experimental validation and it was applied in future control design and fault diagnosis application.

[17] Wang et al. (2016) carried out an experimental study on the performance of a structured packed liquid desiccant dehumidifier (high specific surface area ($650m^2/m^3$) with counter flow arrangement by using LiCl as a liquid desiccant. The proposed experimental equations relating the moisture effectiveness and enthalpy effectiveness with critical inlet constraints were established to calculate the performance of alike dehumidifier. Proposed experimental relationships were authenticated by using the experimental data of the current study, and matched with experimental data informed by other researcher. They suggested that variations were within ±10% for the earlier and within ±15% for the latter. Effects of the inlet conditions of the air and the desiccant along with the packing height on the dehumidification performance were also examined and compared with the results informed in earlier studies.

[18] Chen et al. (2016) investigated the mass transfer performance of Z-type packing in the dehumidifier by using LiCl as a liquid desiccant. Dehumidification and regeneration performance of the planned packing were tested experimentally during summer conditions. They concluded that Z-type packing has outstanding dehumidification performance with

the mass transfer coefficient ranging from 6.5 to 14.2 g/m^2 s, around 70% greater than the corrugated cellulose packing and the plant fiber packing on average. The regeneration performance of the Z-type packing was also comparable with the plant fiber packing. The economic study was further accompanied and it validates the low cost of the Z-type packing under the identical moisture elimination/evaporation rate.

4.2. Hybrid liquid desiccant cooling system

[19] Dai et al. (2001) investigated the enhancement in the performance of vapour compression air-conditioning with the help of liquid desiccant cooling system. The cooling effect and COP of the novel hybrid system was improved significantly as compared to vapour compression system alone. They were carried out the psychrometric study under different cases at constant outlet temperature and humidity and they also studied the effects of dehumidification and evaporative cooling. They concluded that the advantages of the system were presented by lower electricity consumption, higher COP of system and reduced size of VCS, etc.

[20] Mohan et al. (2008) studied the performance of a liquid desiccant columns for a hybrid air-conditioner. They studied the heat and mass transfer analysis for the dehumidifier and regenerator columns in counter flow configuration by using psychrometric equations and liquid desiccant property data. They concluded that high dehumidification in the absorber was achieved at high humidity ratio and low temperature of the inlet air. Likewise, the regeneration was improved by enhancing the temperature and reducing the humidity ratio of the inlet air to the regenerator.

[21] Zhang et al. (2010) investigated the performance analysis of hybrid air conditioner integrated with liquid desiccant dehumidification in summer and winter modes in which sensible load was handled by a vapour compression heat pump and latent load was handled by liquid desiccant dehumidification system using lithium chloride as a desiccant. They suggested that the COP of the hybrid system was enhanced by 20% and 100% in summer and winter, respectively as compared to traditional air conditioning system.

[22] Bergero and Chiari (2011) examined the performances of a hybrid air-conditioning system in which a vapour compression inverse cycle is integrated with liquid desiccant

dehumidification system working with hydrophobic membrane by using LiCl as a desiccant. The LiCl solution was cooled with the help of a vapour-compression inverse cycle by using KLEA 407C as a refrigerant, and the solution was regenerated in another membrane contactor by utilizing the heat rejected by the condenser. A SIMULINK calculation programme was designed for simulating the system under examination in steady-state conditions. They concluded that the simulations results shows that a significant energy savings more than 50% at high latent load in the conditioned environment.

[23] Yamaguchi et al. (2011) studied hybrid system comprises of liquid desiccant system and vapour compression heat pump in which LiCl was used as a desiccant and R407C was used as a refrigerant. In this study, the key features of the hybrid system was that the absorber and regenerator were incorporated with the evaporator and condenser respectively. The suggested that the humidity difference of 5.9 g/kg was dehumidified by the system under summer conditions, and the calculation results proved that COP could become higher by enhancing the isentropic efficiency of compressor and the temperature efficiency of solution heat exchanger.

[24] Mohan et al. (2014) investigated the performance of liquid desiccant-vapour compression hybrid air conditioner by using LiBr as a liquid desiccant. They studied the performance of the hybrid system by varying room air temperature and humidity ratio, and the innovation of the study was that the solution to air flow (S/A) ratio was considered to be very low (0.01). They concluded that the dehumidification of air and regeneration of liquid desiccant was reduced with an increase in room temperature and dehumidification of air along with regeneration of the liquid desiccant was increased with an increase in room humidity ratio. They also suggested that the dehumidification was improved up to 2 g/kg by liquid desiccant loop by absorbing moisture in the absorber.

[25] Ortegon et al. (2016) performed the modelling and dynamic simulation of hybrid liquid desiccant system by using TRNSYS in which LiCl was used as a liquid desiccant. In this study, variables that have the extreme influence on the system were recognized by a sensitivity analysis that evaluated the seasonal performance by means of dynamic long-term simulations and with the help of these results the control strategy of the system can be enhanced. They concluded that the $SCOP_{LDS}$ was increased about 11% with the rise in

regeneration temperature from 60 to 70˚C and $SCOP_{GLOBAL}$ was increased about 21.7% by decreasing the LiCl mass fraction from 0.440 to 0.385.

[26] Bouzenada et al. (2016) investigated the performance of a liquid desiccant air-conditioner driven by evacuated tube, flat-plate, or hybrid solar thermal arrays by using a TRNSYS model that was adapt to show the variations in LDAC effectiveness with different conditions. The model was used to evaluate the alternatives for the collector array design including the use of flat-plate collectors (FPC), ETCs and hybrid arrays including of both FPC and ETCs. They suggested that the simulations were organized for three cities consisting of Toronto (Canada), Tunis (Tunisia), and Calcutta (India) to attain approximate solar fractions to ETC arrays which was found that FPC arrays needed nearly 15 m^2 more area than the ETC arrays in Toronto and Tunis, and 30 m^2 more in Calcutta. They also concluded that Tunis presented the best results for a hybrid system, a ratio of 30% FPCs and 70% ETCs attained the same solar fraction (i.e., 0.56) as the array including of only ETCs.

4.3. Properties of desiccant

[27] Ertas et al. (1992) studied the properties of a liquid desiccant solution by mixing lithium chloride and calcium chloride. They measured the physical properties for different combination of this mixture such as density, viscosity and vapour pressure to evaluate the heat and mass transfer in a desiccant-air contact system. They concluded that the new Cost-effective liquid desiccant (CSLD) containing of 50% each of LiCl and $CaCl_2$ showed superior in all aspects.

[28] Conde (2004) investigated the properties of an aqueous solutions of LiCl and $CaCl_2$. They developed a calculation model for thermophysical properties like solubility boundary, surface tension, vapour pressure, density, dynamic viscosity, thermal conductivity, specific thermal capacity and differential enthalpy of dilution. They suggested that these constraints were play a very vital role in designing of air conditioning equipment.

[29] Hassan and Hassan (2008) proposed a new liquid desiccant by mixing calcium nitrate with calcium chloride in different weight combination. They were calculated the physical properties of a suggested liquid desiccant like vapour pressure, density, viscosity.

Examination of heat and mass transfer amongst a thin liquid layer of the suggested desiccant and the air flowing over a rectangular channel and various reasons that disturbing the dehumidification process of air were studied. They suggested that the anticipated desiccant with a combination of 50% of the weight of water calcium chloride and 20% calcium nitrate provide a substantial rise in vapor pressure depression as compared to the other solution within the concentrations tested.

[30] Liu et al. (2010) studied the mass transfer performance comparison between lithium bromide and lithium chloride aqueous solutions. Heat capacity ratio of air to desiccant and mass transfer unit NTUm were the key performance influencing factors according to the analytical solutions of heat and mass transfer process. They suggested that dehumidification performance of LiCl solution was superior and regeneration performance of LiBr solution was a little better or almost the same as that of LiCl solution for same desiccant mass flow rate.

[31] Longo and Gasparella (2016) executed the experimental measurement of thermophysical properties of potassium formate (H2O/KCOOH) that contains salt concentration from 60 to 80% within temperature range of 1 to 80˚C. They concluded that the H2O/KCOOH desiccant reveals a thermal conductivity 23 - 33% lower than water and the dynamic viscosity 4 - 30 times higher than water.

[32] Yao et al. (2016) carried out an experimental examinations on vapor pressure models for combination of LiCl and $CaCl_2$ desiccant solutions by using NRTL and SMR model. They concluded that SMR model has decent precision in calculating the surface vapor pressure of mixture (LiCl-$CaCl_2$) solution under the mass concentration lower than 10% and it has poor precision as the mixture solution concentration surpasses 20%, while NRTL model was more useful for calculating the surface vapor pressure of mixture solution with a mass concentration greater than 30%.

[33] Zhao et al. (2016) investigated the selection of mixed liquid desiccant and performance examination of a liquid desiccant cooling system. Vapor pressure of mixed liquid desiccants was calculated by NRTL equation. They studied the six absorbent solutions (LiBr/LiCl + $CaCl_2$/$MgCl_2$ + water/methanol) and systems using them. The obtained results shows that the dehumidification efficiency of mixed liquid desiccant was greater as

compared to the single solution. They concluded that LiBr-$CaCl_2$-water system can achieve greater COP as compared to the other systems under the condition of the similar dehumidification performance during summer, and LiCl-methanol-water may be a probable absorbent for energy saving. They suggested that the obtained experimental results were play a very significant role for the selection of appropriate liquid desiccants and engineering design.

4.4. Research gap

Many researchers had experimentally investigated the liquid desiccant cooling system with different kind of structural packing in dehumidifier. But no research work has been done on the packing of dehumidifier with PVC fills and LDDS integrated with VCRS.

So, the experimental analysis of dehumidifier packing with PVC fills for optimum moisture removal from the process air and the performance evaluation of LDDS with VCRS will definitely be the novelty of this project.

CHAPTER 5

5. Experimental set-up

This chapter is related to the experimental work that is performed to investigate the performance of liquid desiccant cooling system (LDCS) integrated with vapour compression refrigeration system (VCRS). The experiments have been performed at Lovely Professional University (LPU), Jalandhar, India [$31^{\circ}15'17.98''$ (latitude) North and $75^{\circ}42'56.382''$ (longitude) East] at 55 block.

5.1. Components of experimental set-up

The experimental set-up consists of two sub-systems, namely LDCS and VCRS. Desiccant system reduces the humidity of the air and VCRS system reduces the temperature for better cooling. The major components used in the experimental set-up are dehumidifier, regenerator, liquid desiccant, solution heating chamber, solution cooling chamber, air heater, spray nozzle pump, humidification chamber, solution pump, axial fan, connection pipes and VCRS. The photographic view of experimental set-up as shown in Figure 9.

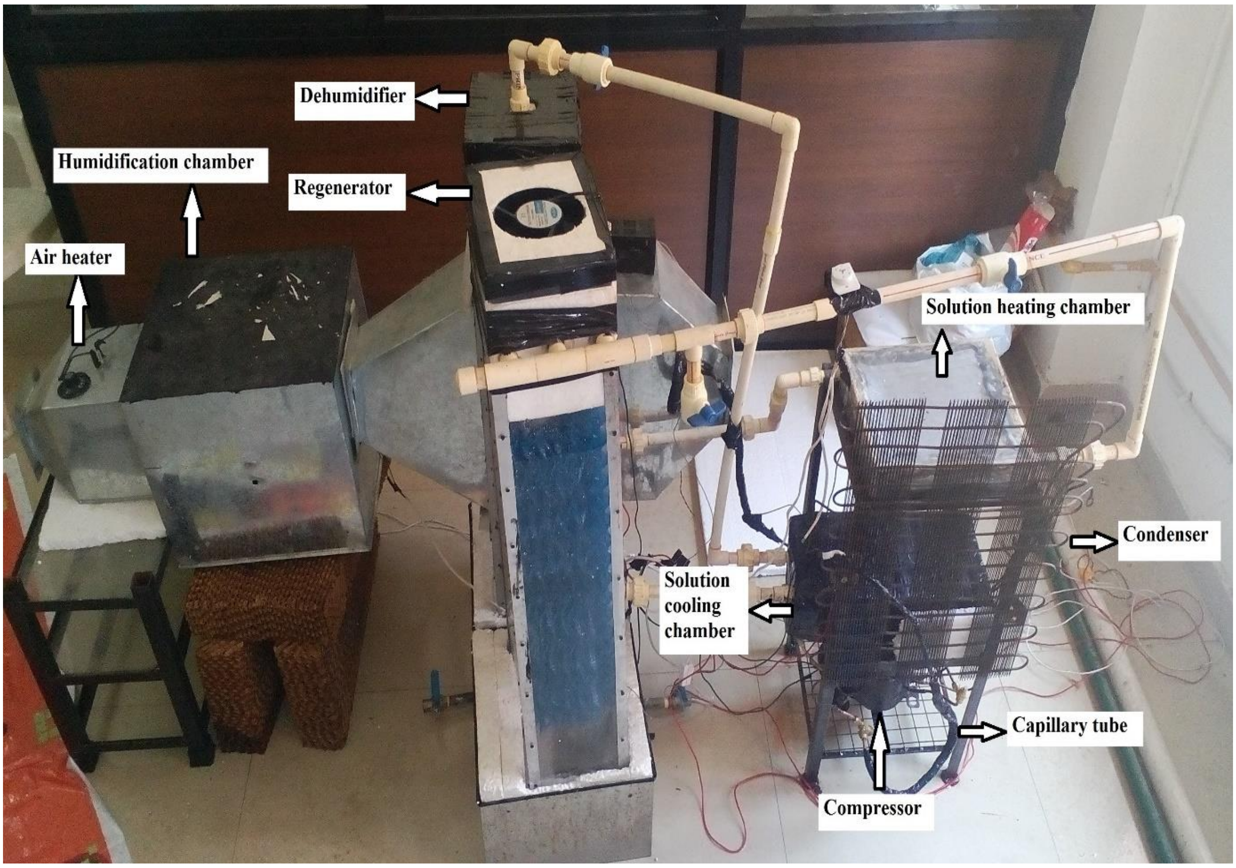

Figure 9: Photograph of experimental set-up

5.1.1. Dehumidifier

It dehumidifies the air passing through it by the action of liquid desiccant. It consists of honeycomb structure PVC fills over which liquid desiccant is made to fall by the virtue of gravity through shower at the top of the dehumidifier. The air to be dehumidified is passed through a duct in a cross flow direction. Mixing of air with desiccant solution occurs in the PVC fills that provides the sufficient surface area for moisture exchange. The hygroscopic liquid desiccant absorbs the moisture present in the process air, dehumidified air leaving at outlet and diluted desiccant solution collected in the bottom tank of dehumidifier. Specifications of components used in dehumidifier are listed in Table 1. The photograph of dehumidifier as shown in Figure 10.

PVC fills of honeycomb structure are used for the structural packing of dehumidifier. It provides the sufficient surface area for proper mixing of process air and liquid desiccant solution that results in better moisture removal. Specifications of PVC fills are listed in the Table 2. The photograph PVC fills is shown in Figure 11. Ducts are used for inlet and outlet air flow made up of GI sheets. Dimensions of the duct are shown in Figure 12.

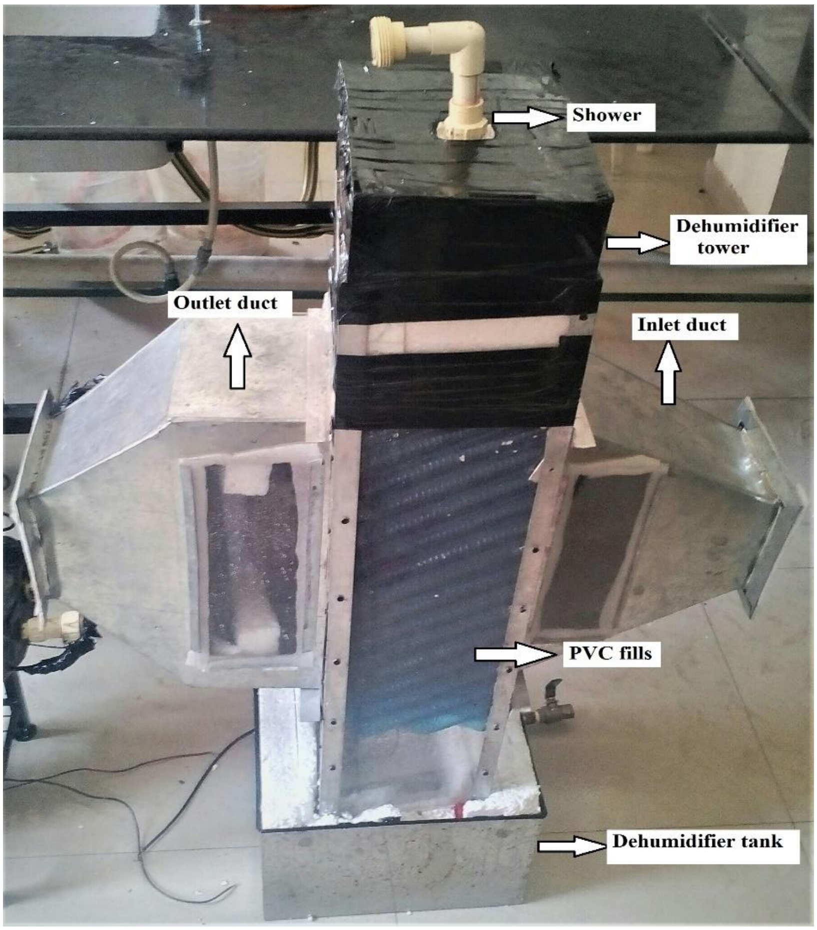

Figure 10: Photograph of dehumidifier

Table 1: Specifications of components used in dehumidifier

S. No.	Components	Dimensions (Inches)	Material
1.	Shower	6 × 6	Stainless steel
2.	Dehumidifier tower	12 × 8 × 39	Acrylic sheet (2mm thick)
3.	Dehumidifier tank	10 × 7 × 7	Stainless steel
4.	Structural packing	12 × 8 × 24	PVC fills

Figure 11: Photograph of PVC fills

Table 2: Specifications of PVC fills

S. No.	Parameters	Value
1.	Dimension of standard type of PVC fills	600 mm x 300 mm x 150 mm
2.	PVC honey combs	25 mm {12. 5 + 12. 5}
3.	Corrugation pitch	2.8 mm
4.	Corrugation depth	3 mm
5.	Spiral angles	21
6.	Number of PVC sheets per 150 mm width	12
7.	Heat transfer area/cu. Ft. of fills volume	1cu. Ft.
8.	Thickness of finished fills.	0.30 mm
9.	Heat distortion temperature for normal PVC	60 ˚C
10.	Weight of std. Fills.	800 gm to 1 kg

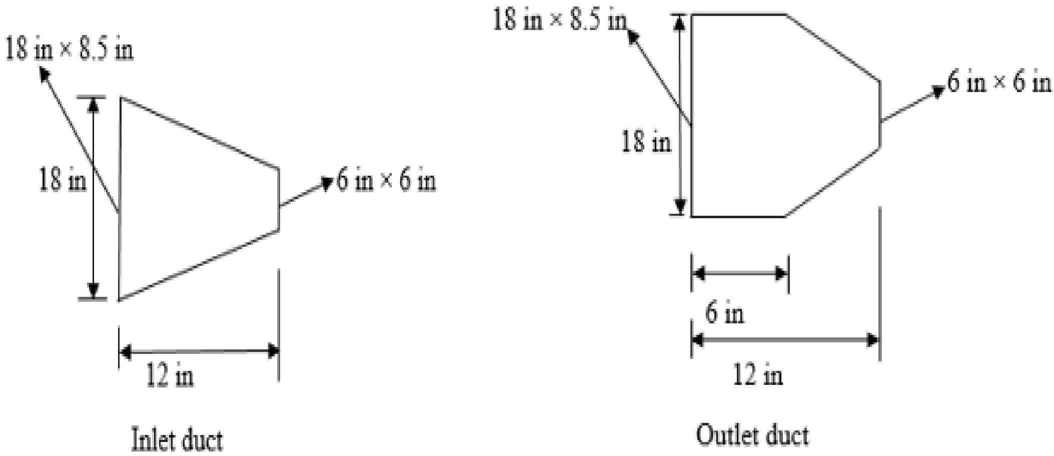

Figure 12: Dimensions of inlet and outlet duct

5.1.2. Regenerator

It is used to regenerate the liquid desiccant from its diluted state after the absorption of moisture from the process air in the dehumidifier. Hot diluted liquid desiccant solution is made to fall through sprayer over the PVC fills by the action of gravity at the top and ambient air is passed from the bottom of the regenerator unit in a counter flow direction. As the hot liquid desiccant mixes with the process air in the PVC fills, it transfers the moisture to the air and collects in the regenerator tank in the concentrated form. The air carrying moisture is expelled out through the exhaust at the top of the regenerator. Specifications of components used in regenerator are listed in Table 3. The photograph of regenerator tower is shown in Figure 13. The dimensions of the sprayer as shown in Figure 14.

Figure 13: Photograph of regenerator

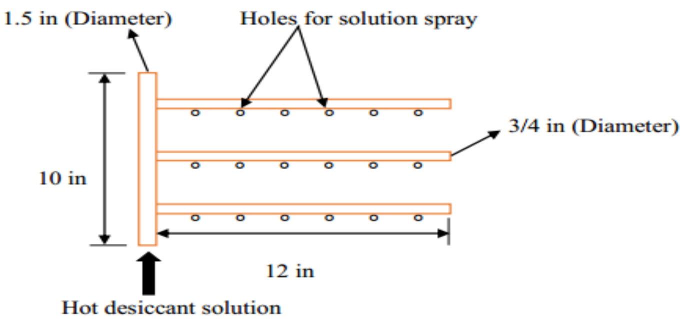

Figure 14: Dimensions of spray arrangement

Table 3: Specifications of components used in the regenerator

S. No.	Components	Dimensions (Inches)	Material
1.	Shower	6 × 6	Stainless steel
2.	Regenerator tower	12 × 8 × 39	Acrylic sheet (2mm thick)
3.	Regenerator tank	10 × 7 × 7	Stainless steel
4.	Structural packing	12 × 8 × 24	PVC fills

5.1.3. Liquid desiccant

Calcium chloride ($CaCl_2$) is used as a liquid desiccant in the system. Low cost, easy availability and good properties as a desiccant makes it right choice to be used in the system. Heat is evolved as it dissolves in the water due to exothermic reaction. The solution of liquid desiccant used in the system is 30% wt concentration which is made by dissolving 8 kg of $CaCl_2$ in 26.5 liters of water. The specification of calcium chloride are listed in Table 4. The photograph of calcium chloride (fused) used to perform the system operation is shown in Figure15.

Table 4: Properties of calcium chloride

S. No.	Parameters	Value
1.	Molar mass	110.98 $g.mol^{-1}$
2.	Appearance	White powder, hygroscopic
3.	Odor	Odorless
4.	Density	2.15 g/cm^3
5.	Melting point	772-775 ˚C
6.	Low freezing temperature of solution	-50.5 ˚C
7.	Low vapour pressure of solution	43 mm Hg at 32 ˚C

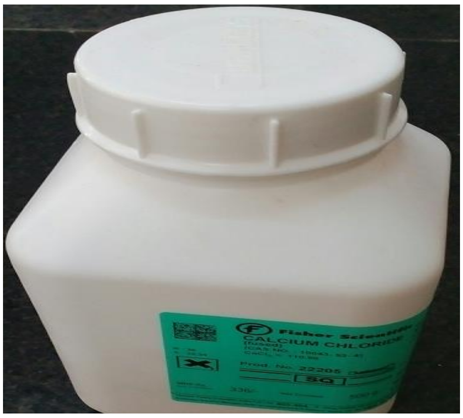

Figure 15: Photograph of calcium chloride

5.1.4. Desiccant solution heating chamber

Solution cooling chamber is a unit in which heater is used to heat the diluted liquid desiccant solution before entering the regenerator. The heater increases the temperature of the solution up to 50˚C so that the vapour pressure of desiccant solution increases more than the vapour pressure of the process air due to which it loses the moisture to the process air. Circular copper coil of capacity 1 KW is used as a heater with the thermostat as shown in Figure16. It is attached at the base of the chamber in which desiccant solution is being heated as shown in Figure 17. Specifications of solution heating chamber are listed in Table 5.

Table 5: Specifications of desiccant solution heating chamber

S. No.	Components	Dimensions (Inches)	Material
1.	Heating chamber	12 × 12 × 3	GI sheet
2,	Heating chamber stand	13 × 13 × 31	Iron angles

Figure 16: Photograph of heater coil

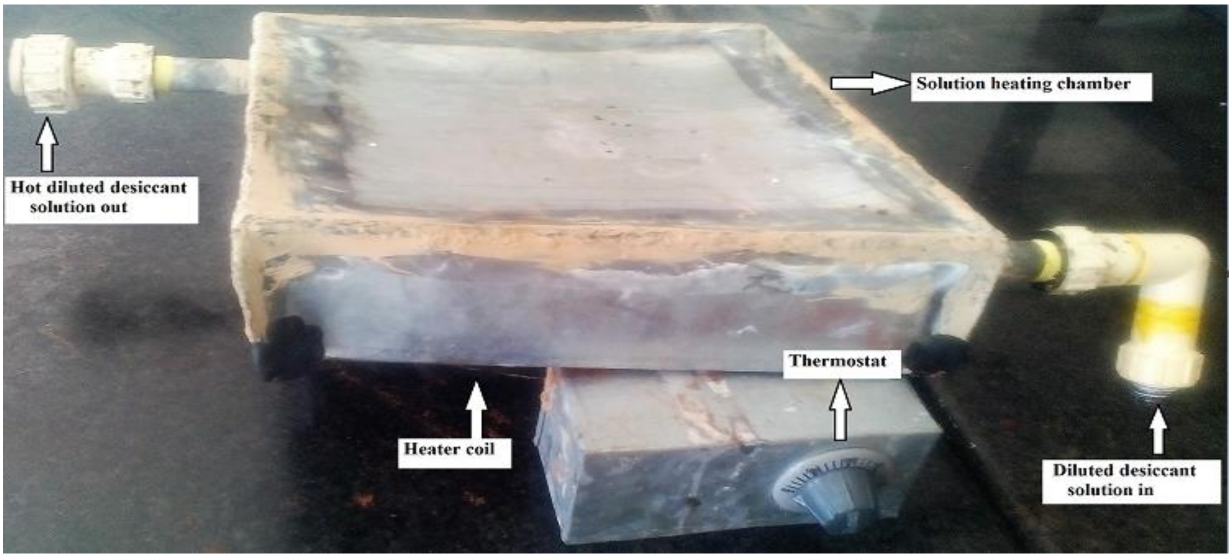

Figure 17: Photograph of desiccant solution heating chamber

5.1.5. Desiccant solution cooling chamber

The desiccant solution cooling chamber coupled with copper coil connected to evaporator coil and compressor, is used to cool the hot concentrated liquid desiccant coming from regenerator tank with the help of refrigerant (R134a) coming from the evaporator coil before entering to the dehumidifier. This chamber is made up of GI sheet with the dimensions as length 13 inches, breadth 6 inches and height 2 inches as shown in Figure 18.

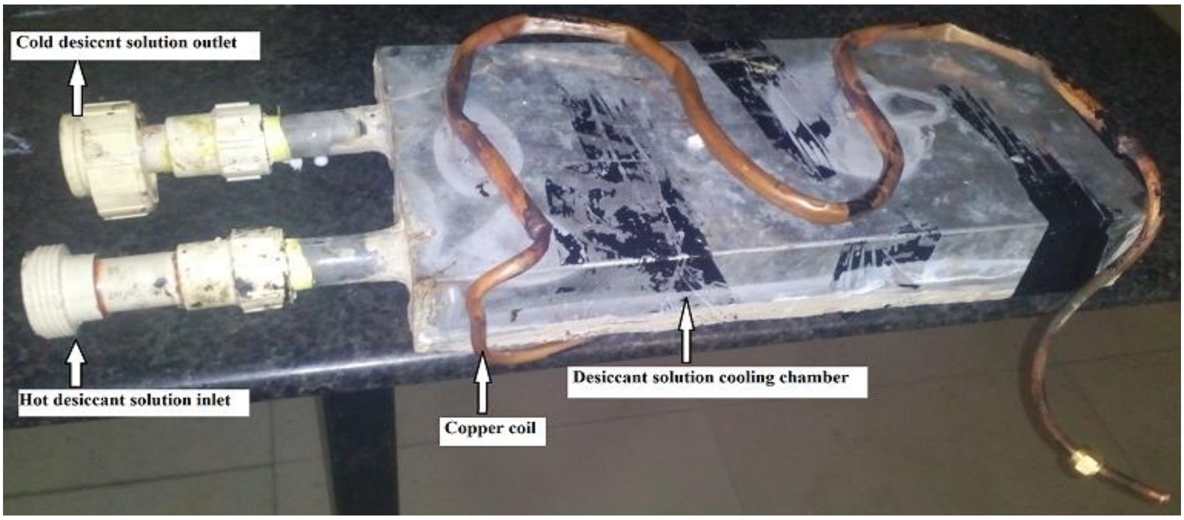

Figure 18: Photograph of desiccant solution cooling chamber

5.1.6. Solution pump

Two submersible pumps are used for pumping liquid desiccant solution and maintaining circulation in the system. One pump is used to pump the diluted liquid desiccant solution from dehumidifier tank to the solution heating chamber and the other pump is used to pump hot liquid desiccant solution from regenerator tank to the solution cooling chamber. The specifications of pump are given in Table 6. The photograph of the pump is shown in Figure 19.

Figure 19: Photograph of pump

Table 6: Specifications of pump

S. No.	Parameters	Value
1.	Material	ABS
2.	Voltage	165-220/50 Hz
3.	Frequency	50 Hz
4.	Power	40 W
5.	H- max	2.8 m
6.	Maximum head	2800 mm
6.	Output	3800 L/H

5.1.7. Axial fan

Two axial fans are used in the system for air circulation. The first axial fan is located inside the inlet duct to draw the surrounding air into the dehumidifier and the second axial fan is located at the top of the regenerator to exhaust the hot air carrying the moisture from the desiccant solution. Specifications of axial fan are listed in Table 7. Photograph of the axial fan is shown in the Figure 20.

Table 7: Specifications of axial fan

S. No.	Parameters	Value
1.	Model	FM17051A2HSL
2.	Voltage	220V/240V
3.	Frequency	50/60 Hz
4.	Current	0.25 amp
5.	Size	$170 \times 170 \times 51$ mm

Figure 20: Photograph of axial fan

5.1.8. Air heater

Air heater is used to increase the temperature of the inlet air by using circular copper coil electric heater runs on AC single phase 220 volts having 1 KW power. When the electric current passes through the resistance element, it results heating of element which ultimately heats the air on coming contact with the coil. A thermostat is used for controlling the different temperature variation. The air heater duct is attached to the inlet of humidification chamber as shown in Figure 21. The dimensions of the air heater duct as shown in Figure 22.

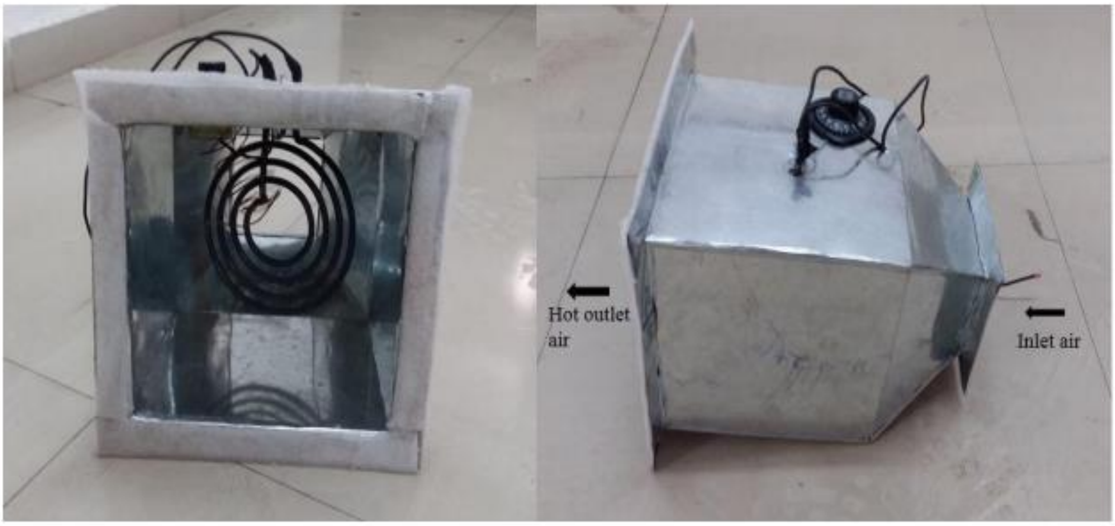

Figure 21: Photograph of air heater

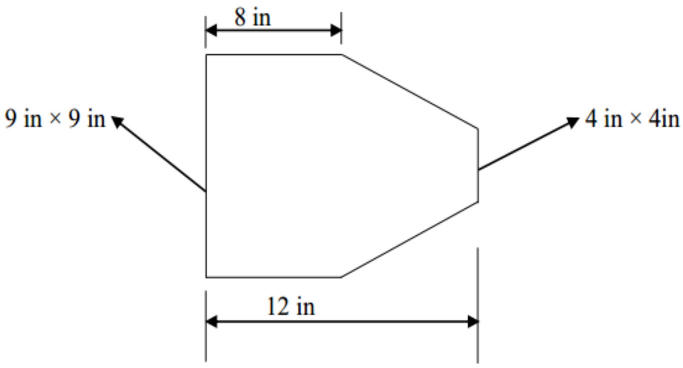

Figure 22: Dimensions of air heater duct

5.1.9. Humidification chamber

Humidification chamber is used to create different levels of humid conditions to study the performance of the system. Humidity is created by spraying hot water through the nozzle of spray pump in the chamber. After air heater, hot air enters into the humidification chamber where it gets humidified and then hot humidified air enters into the dehumidifier through duct. This chamber is made up of GI sheet with the dimensions of 16in x 16in x

16in in which the inlet passage provided for hot air with dimensions of 9in x 9in and the outlet passage provided for hot dehumidified air with dimensions of 6in x 6in. The photograph of humidification chamber as shown in Figure 23 and spray pump as shown in Figure 24.

Figure 23: Photograph of humidification chamber

Figure 24: Photograph of spray pump

5.1.10. Connection pipes

The pipes made up of CPVC is used for the flow of liquid desiccant. The diameter of pipes used for connection is 0.75 in. Elbows, unions and flow control valve made up of CPVC are used for the proper flow and fitting of the connections.

5.1.11. Vapour compression refrigeration system

A VCRS is integrated with LDDS for cooling of air in the experimental set-up. It consist of compressor, condenser, evaporator and capillary tube. A compressor of capacity 220 Watt and a refrigerator condenser is used in VCRS. The evaporator coil is fitted with the outlet duct of dehumidifier to cool the hot and dehumidified air with the help of refrigerant as shown in Figure 25. R134a is used as refrigerant in the VCRS.

Figure 25: Photograph of arrangement of evaporator coil in the outlet duct of dehumidifier

5.2. Measuring Instruments

Different types of measuring instruments are used for measuring the different parameters such as relative humidity of air, dry bulb temperature and wet bulb temperature of inlet and outlet air, temperature of desiccant solution at various points and inlet air velocity.

The different types of measuring instruments used in the experimental setup are RTD-PT100 thermocouple, Anemometer, Sling psychrometer, and DBT-WBT thermometer.

5.2.1. RTD-PT100 thermocouple

Resistance temperature detector (RTD) is used to show the temperature on display and measure temperature by correlating the resistance of RTD element with temperature. PT100 thermocouples are connected with a digital temperature indicator that shows the temperature with a resolution of 0.1˚C. PT100 thermocouple also known as platinum resistance thermocouple which is used to determine the temperature. In platinum sensor, change in temperature of 1˚C will cause a change of 0.384 ohm in resistance. Mostly used type of PT100 has a resistance of 100 ohms and 138.4 ohms at and 0˚C and 100˚C respectively. It is used to measure the temperature of liquid desiccant at various points in the system. The specifications of RTD-PT100 are listed in Table 8. The photograph of RTD-PT100 used in the system is shown in Figure 26.

Figure 26: Photograph of RTD-PT100 thermocouple

Table 8: Specifications of RTD-PT100 thermocouple

S. No.	Parameters	Value
1.	Supply	AC 230 Volt
2.	Model	Nutronics DTL-200M
3.	Range	Max 800
4.	Sensor	Platinum

5.2.2. Anemometer

Anemometer is used to measure the velocity of air by the rotating fan. Air flow strikes on fan blades resulting in rotation of blades and tiny generator to which it is connected. The generator is connected to electronic circuit that shows the wind speed on a digital display. Mass flow rate of inlet air to dehumidifier is calculated with the help of air velocity. The specifications of Anemometer are listed in Table 9. The photograph of anemometer used for measurement of air velocity is shown in Figure 27.

Figure 27: Photograph of Anemometer

Table 9: Specifications of Anemometer

S. No.	Parameters	Value
1.	Wind velocity	0.0 - 45.0 m/s
2.	Resolution	0.001
3.	Lowest point of start value	0.3 m/s
4.	Accuracy	±3% ± 0.1

5.2.3. Sling psychrometer

Sling psychrometer is used for measuring the dry bulb temperature and wet bulb temperatures. It is manually operated measuring instrument and having a temperature range of 0˚C to 50˚C. It contains two thermometers, one is dry bulb temperature (DBT) and other is wet bulb temperature (WBT). The WBT measurement is provided with a wet cotton piece on tip of thermometer. Rotating screws are provided for the turning of the instrument. Each reading is taken by rotating the instrument 10 to 15 times. The photograph of the sling psychrometer used for measurement of DBT and WBT of air is shown in Figure 28.

Figure 28: Photograph of Sling psychrometer

5.2.4. Wet and dry thermometer

It gives the wet and dry bulb temperature of the air. Dry bulb temperature denotes the normal temperature of the air when it is not affected by the moisture in the air while the wet bulb temperature denotes the temperature when the bulb is covered by wet cloth exposed to the air. The temperature range for the thermometer is from -10˚C to 50˚C with the resolution of 0.1˚C. The photograph of the wet and dry thermometer is shown in Figure 29.

Figure 29: Photograph of wet and dry thermometer

CHAPTER 6

6. Research methodology

The problem at the hand involves the performance evaluation of structured packed LDDS integrated with VCRS.

PVC fills are used for the structural packing of dehumidifier/regenerator. It provides the sufficient surface are for mixing of desiccant solution and the process air for moisture exchange.

Calcium chloride is used as a liquid desiccant to perform the operation of the LDDS. In VCRS, R134a is used as a refrigerant that takes the heat from the hot air coming from the dehumidifier to cool the air.

The experiments are conducted by varying inlet air conditions into dehumidifier as follows:

I. The unsaturated air temperature varies from 30˚C to 40˚C by air heater with the help of thermostat at constant air velocity of 4.0 m/s.

II. The unsaturated air velocity varies from 3 to 4.5 m/s with the help of regulator of fan by keeping constant air temperature of 35˚C.

III. The saturation of air is done by spraying the water with the help of spray pump in the humidification chamber. During saturation of air, the air temperature varies from 30˚C to 40˚C by keeping constant air velocity of 4.0 m/s, and the air velocity varies from 3 to 4.5 m/s by keeping constant air temperature of 35˚C.

The performance of structural packed LDDS is estimated on the basis of moisture removal rate and dehumidification effectiveness. Also the COP of system and COP of VCRS without LDDS are calculated.

CHAPTER 7

7. System operation

The proposed system is a combination of liquid desiccant dehumidification system (LDDS) and vapour compression refrigeration system (VCRS) in which dehumidification of air is achieved by LDDS and cooling of air is achieved by VCRS. The schematic diagram of the proposed system as shown in Figure 30.

In dehumidifier, the liquid desiccant absorbs the moisture from the incoming hot and humid air, and after dehumidification process the dehumidified air at the outlet of dehumidifier is passed to the evaporator where the heat of air taken away by refrigerant (R134a) to cool the air. After absorbing the moisture, liquid desiccant gets diluted and collected in the dehumidifier tank. In order to get the original state of the diluted liquid desiccant solution, it is regenerated in the regenerator. The diluted liquid desiccant solution is pumped to the solution heating chamber , where its temperature is raised around 50˚C and after that the hot desiccant solution is sent to regenerator in which it losses the moisture to the surrounding air passing through it and regenerated solution (concentrated form) is collected into the regenerator tank.

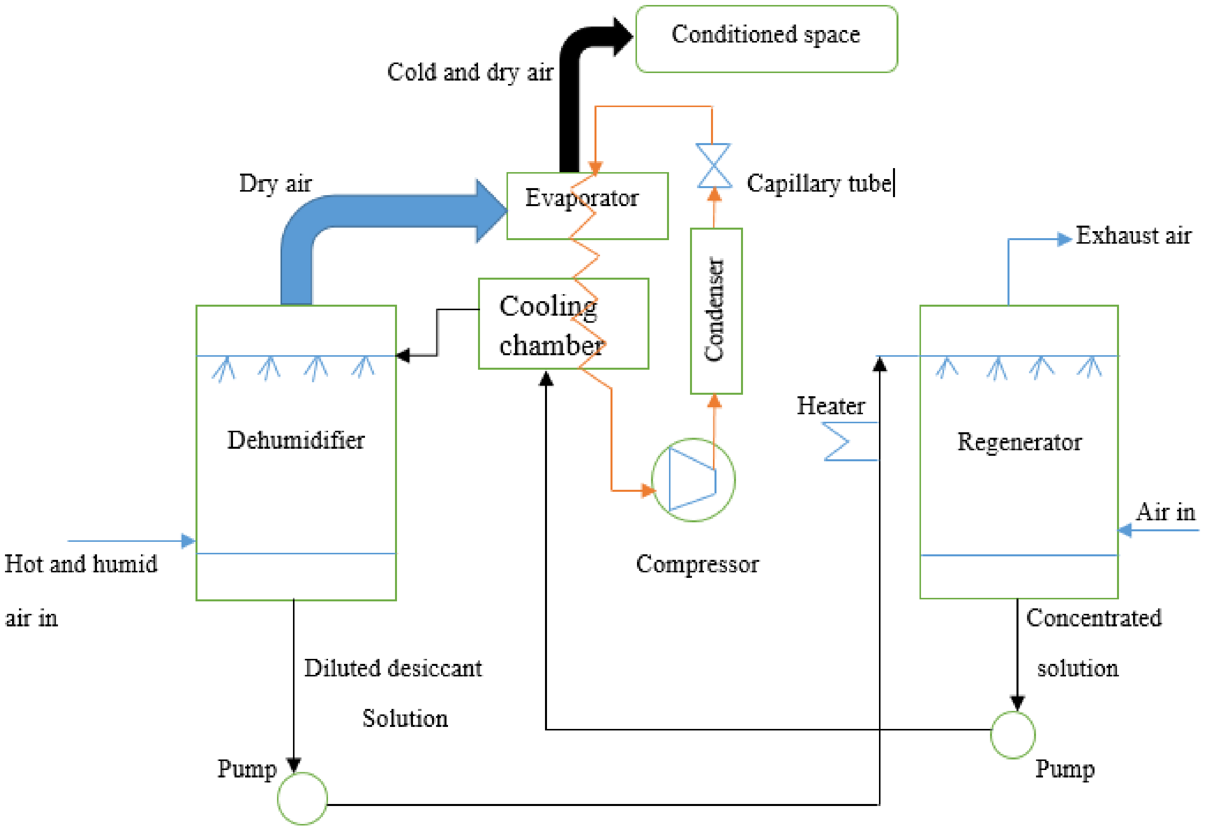

Figure 30: Schematic diagram of a proposed system

The concentrated solution is pumped to the desiccant solution cooling chamber where it gets cooled by refrigerant coming from the evaporator coil and after that it is pass to the dehumidifier, thus cycle continues.

The performance of system is investigated under different air inlet conditions like air temperature and velocity. For different air temperature, air heater is used in a duct before entering of air in dehumidifier and to meet different humidity levels, humidification chamber is used after air heater.

CHAPTER 8

8. Performance analysis of experimental data

The performance analysis of a LDDS integrated with VCRS is evaluated on the basis of moisture removal rate (MRR), dehumidification effectiveness (ε_{deh}), COP of the system and COP of VCRS without LDDS.

8.1. Moisture removal rate (MRR)

It is defined as the rate at which moisture is absorbed from inlet air. It is calculated from equation (1) given by Mehla and Yadav [37] as

$$MRR = m_{a,i}(w_{a,i} - w_{a,i}) \quad \ldots\ldots (1)$$

Where, $w_{a,i}$ = Humidity ratio of inlet air (g/kg of dry air)

$w_{a,i}$ = Humidity ratio of outlet air (g/kg of dry air)

$m_{a,i}$ = Mass flow rate of inlet air (kg/s)

The mass flow rate of the inlet air is calculated from equation (2) as

$$m_{a,i} = \rho A v \quad \ldots\ldots (2)$$

Where, ρ = Density of the air (kg/m^3)

A = Area of the column (m^2)

v = Velocity of air (m/s)

The density of air varies with respect to pressure and temperature, so it is calculated by using ideal gas equation (3) as

$$\rho = \frac{P}{R_{spec}\, T} \quad \ldots\ldots (3)$$

Where, $P = 101.325 \times 10^3$ Pa (Atmospheric pressure)

R_{spec} = 287.058 J/kg.K (Specific gas constant for air)

T = Temperature of air in K

Also, the density of air varies with the addition of water vapour in it, so the density of humid air is calculated by equation (4) as [38]

$$\rho_{humid\ air} = \frac{P_d M_d + P_v M_v}{RT} \quad \ldots\ldots (4)$$

Where, P_d = Partial pressure of dry air (kpa)

P_v = Partial pressure of water vapour (kpa)

$M_d =$ 0.028964 kg/mol (Molar mass of dry air)

M_v = 0.018016 kg/mol (Molar mass of water vapour)

$R =$ 8.314 J/K.mol (Universal gas constant)

T = Temperature in K

Partial pressure of water vapour is calculated by equation (5) as

$$P_v = \varphi P_{sat} \quad \ldots\ldots (5)$$

Where, φ = Relative humidity

P_{sat} = Partial pressure of saturated air (kpa)

Partial pressure of saturated air is calculated by equation (6) as [39]

$$P_{sat} = 6.1078 \times 10^{\frac{7.5T}{T+237.3}} \quad \ldots\ldots (6)$$

Partial pressure of dry air is calculated by equation (7) as

$$P_d = P - P_v \quad \ldots\ldots (7)$$

Where, P = 101325 pa (Atmospheric pressure)

8.2. Dehumidification effectiveness (ε_{deh})

It is defined as the actual drop in humidity ratio of inlet air to the maximum possible drop. It is calculated from equation (8) given by Mehla and Yadav [37] as

$$\varepsilon_{deh} = \frac{(w_{a,i} - w_{a,o})}{(w_{a,i} - w_{eq})} \quad \ldots\ldots (8)$$

Where, w_{eq} is the humidity ratio of air in equilibrium with desiccant solution (CaCl2), which is calculated by following equations [40]

$$P_{wo} = 0.0159 T_s^{1.6438} \quad \ldots\ldots (9)$$

$$P_{wx} = P_{wo}(1.146 - 1.76x) \quad \ldots\ldots (10)$$

$$w_{eq} = 0.62185 \frac{P_{wx}}{P - P_{wx}} \quad \ldots\ldots (11)$$

Where, P_{wo} is the saturation pressure of pure water (i.e. x = 0) at the desiccant solution temperature and P_{wx} is the saturation pressure of the desiccant solution of concentration x at the desiccant solution temperature.

8.3. Coefficient of performance (COP)

COP is defined as the ratio of refrigeration effect to the work input.

The performance of the hybrid liquid desiccant air conditioning system is evaluated by the two types of COP such as $(\mathrm{COP})_{\mathrm{system}}$ and $(\mathrm{COP})_{\mathrm{VCRS\ without\ LDCS}}$.

The $(\mathrm{COP})_{\mathrm{system}}$ is calculated by equation (11) as

$$(\mathrm{COP})_{\mathrm{system}} = \frac{(h_{a,i} - h_{a.o})}{(W_{in})} \quad \text{...} (11)$$

Where, $h_{a,i}$ = Enthalpy of inlet air

$h_{a.o}$ = Enthalpy of outlet air

$W_{in} = W_{heater} + W_{axial\ fan} + W_{pump}$

The $(\mathrm{COP})_{\mathrm{VCRS\ without\ LDCS}}$ is calculated by equation (12) as

$$(\mathrm{COP})_{\mathrm{VCRS\ without\ LDCS}} = \frac{(h_{a,i} - h_{a.o})}{(W_{in})} \quad \text{..} (12)$$

Where, $W_{in} = W_{heater} + W_{axial\ fan}$

CHAPTER 9

9. Experimental results and discussion

In this chapter, the obtained results from the experiments are conceded. The experiments are carried out under different conditions. The performances of the system are investigated by calculating parameters like moisture removal rate (MRR), dehumidification effectiveness (ε_{deh}), COP of system and COP of VCRS without LDDS. During the experimentation, the average temperature of desiccant solution at different points in the system are listed in Table 10.

Table 10: Temperature of desiccant solution at different point in the system

S. No.	Positions	Temperature (˚C)
1.	Dehumidifier inlet	29
2.	Solution cooling chamber inlet	35
3.	Solution cooling chamber outlet	30
4.	Regenerator inlet	50

9.1. Experimental results

The experiments are carried under four cases to evaluate the performance of the system.

Case 1: Unsaturated air with different temperature at constant air velocity

Case 2: Unsaturated air with different velocity at constant air temperature

Case 3: Saturated air with different temperature at constant air velocity

Case 4: Saturated air with different velocity at constant air temperature

9.1.1. Case 1: Unsaturated air with different temperature at constant air velocity

The unsaturated air means the air that contains less amount of moisture content. The temperature of unsaturated air varies with the help of heater whose temperature is controlled by thermostat. In this case the air temperature varies from 30 to 40˚C by keeping constant air velocity of 4.0 m/s. The performance parameters like moisture removal rate (MRR), dehumidification effectiveness, COP of system, and COP of VCRS without LDDS are calculated at various temperatures.

(i) Variation of moisture removal rate (MRR) and dehumidification effectiveness (DE) at different air temperature

Figure 31 shows the variation of moisture removal rate (MRR) and dehumidification effectiveness (DE) with respect to unsaturated air temperature. The air temperature varies from 30 to 40˚C by keeping constant air velocity of 4.0 m/s. MRR decreases with the increase in unsaturated air temperature. The reason for this is that the increase in air temperature causes the lower vapour pressure difference between desiccant solution and process air. Thus, at higher temperature, driving potential for moisture transfer reduces which results in lower absorption capacity for desiccant solution. Hence MRR decreases from 0.92 to 0.72g/s as air temperature increases. The dehumidification effectiveness decreases with the increase in unsaturated air temperature. In fact, an increase in air temperature causes the decrease in difference of specific humidity of air due to decrease in vapour pressure difference between desiccant solution and process air, and hence dehumidification effectiveness decreases from 0.135 to 0.088 as the temperature of air increases.

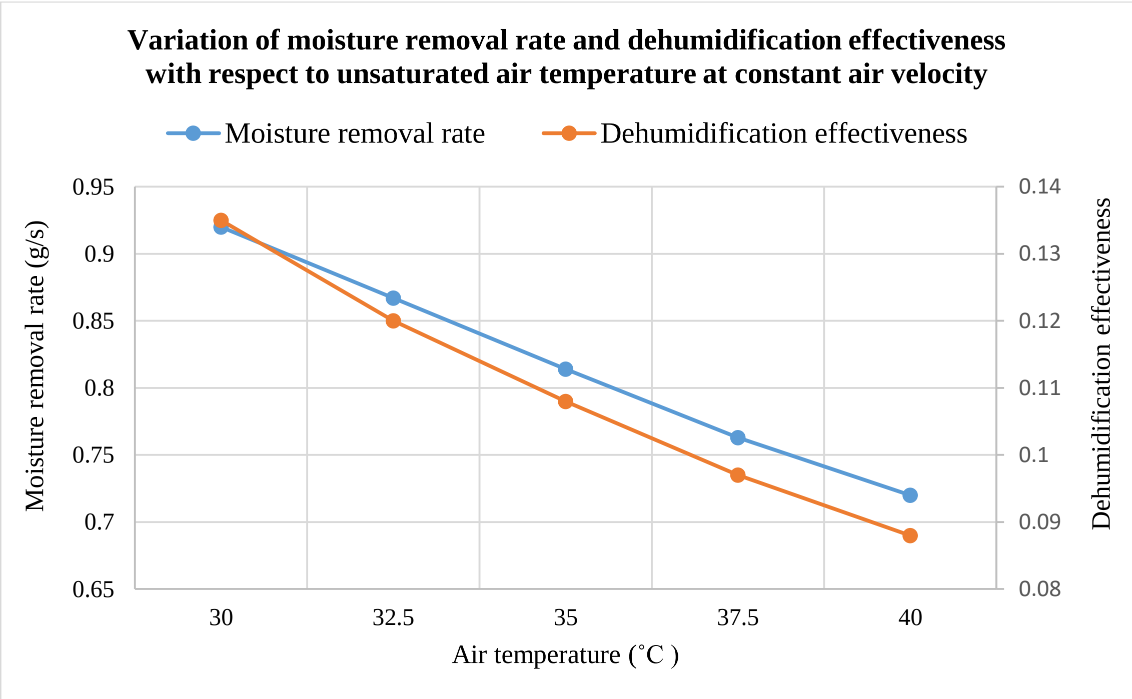

Figure 31: Variation of moisture removal rate and dehumidification effectiveness with unsaturated air temperature

(ii) Variation of COP of system at different air temperature

Figure 32 illustrate the variation of COP of system with respect to unsaturated air temperature. The air temperature varies from 30 to 40˚C by keeping constant air velocity at 4.0 m/s. COP of system increases with the increase in unsaturated air temperature. The reason for this is that the enthalpy difference of air increases with the increase in air temperature that will leads to increase in refrigeration effect results in increase in COP of system. Also for the COP of system, the total load (sensible and latent load) of air is covered by both LDDS and VCRS. Hence COP of system increases from 2.7 to 4.86 as the air temperature increases.

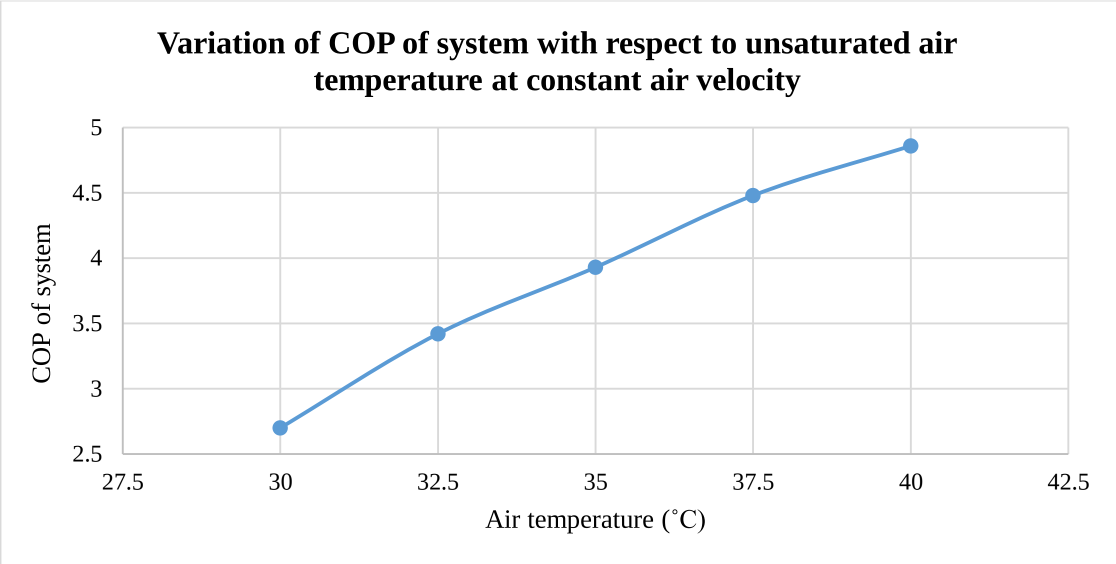

Figure 32: Variation of COP of system with unsaturated air temperature

(iii) Variation of COP of VCRS without LDDS at different air temperature

Figure 33 shows the variation of COP of VCRS without LDDS with respect to unsaturated air temperature. The air temperature varies from 30 to 40˚C at constant air velocity of 4.0 m/s.

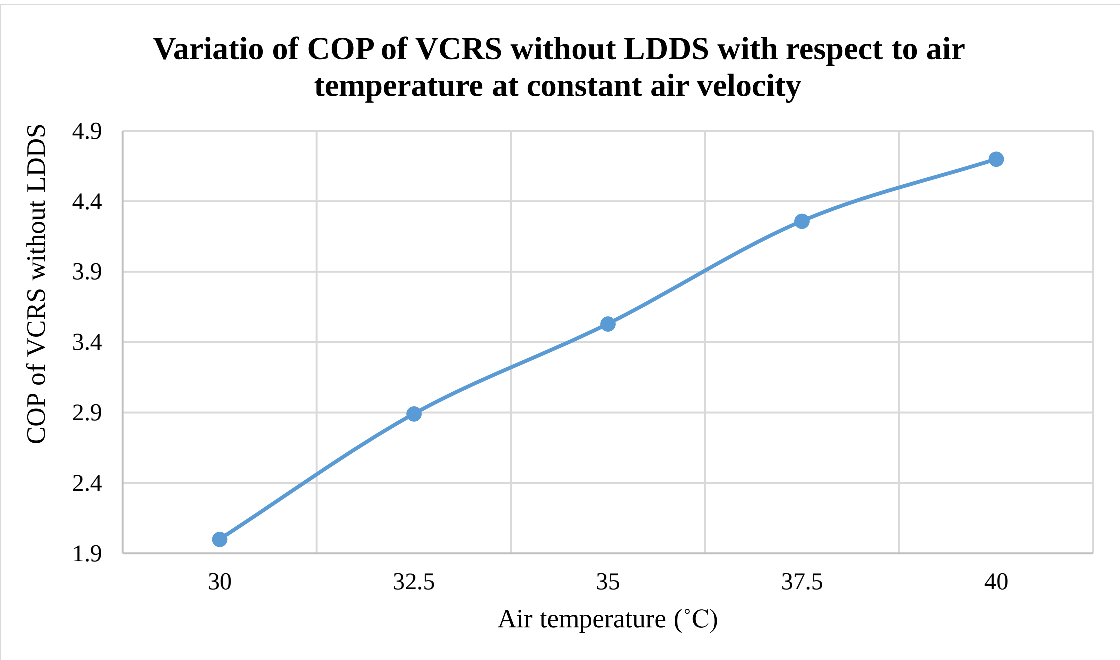

Figure 33: Variation of COP of VCRS without LDDS with air temperature

COP of VCRS without LDDS increases with the increase in unsaturated air temperature. The reason for this is that the sensible load increases due to increase in air temperature. The increase in sensible load causes the increase in enthalpy difference of air that will leads to increase the refrigeration effect, hence the COP of VCRS increases from 2 to 4.7 as air temperature increases.

(iv) Comparison of COP of system and COP of VCRS without LDDS at different air temperature

Figure 34 shows the comparison of COP of system and COP of VCRS without LDDS with respect to unsaturated air temperature. The air temperature varies from 30 to 40˚C by keeping constant air velocity of 4.0 m/s.

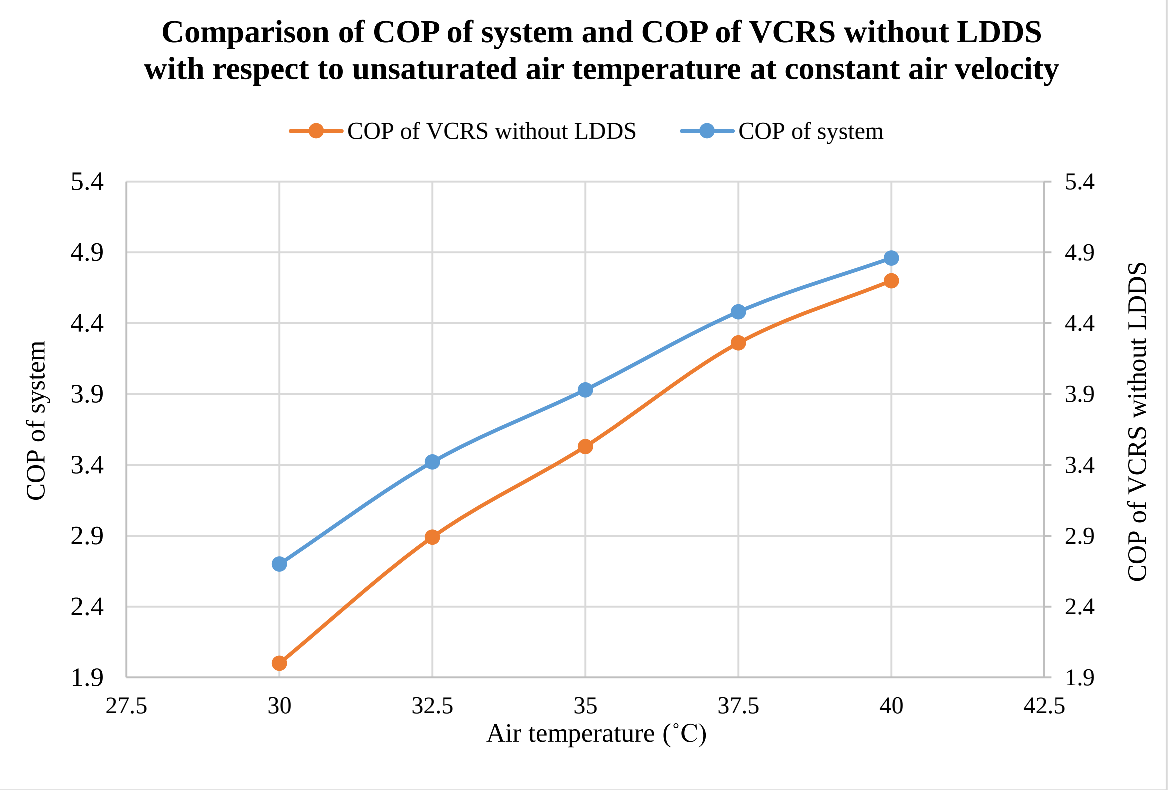

Figure 34: Comparison of COP of system and COP of VCRS without LDDS with the unsaturated air temperature

It is clear from the figure that the COP of system increases more as compare to COP of VCRS without LDDS as the unsaturated air temperature increases. In fact, with the increases in air temperature, the enthalpy difference of air is more within the system as compare to the VCRS, and hence the refrigeration effect within the system is more than

VCRS. Also for the COP of system, the total load (sensible and latent load) of air is handled by both LDDS and VCRS, while for the COP of VCRS without LDDS, the total load of air is handled by VCRS only. So, the increment in COP of system is more than the COP of VCRS as the air temperature increases.

9.1.2. Case 2: Unsaturated air with different velocity at constant air temperature

The unsaturated air velocity varies with the help of regulator of fan. In this case the air velocity varies from 3 to 4.5m/s at constant air temperature of 35˚C. The performance parameters like moisture removal rate (MRR), dehumidification effectiveness, COP of system, and COP of VCRS without LDDS are calculated at various air velocity.

(i) Variation of moisture removal rate (MRR) dehumidification effectiveness (DE) at different air velocity

Figure 35 illustrate the variation of moisture removal rate and dehumidification effectiveness with respect to unsaturated air velocity. The air velocity varies from 3 to 4.5m/s by keeping constant air temperature of 35˚C.

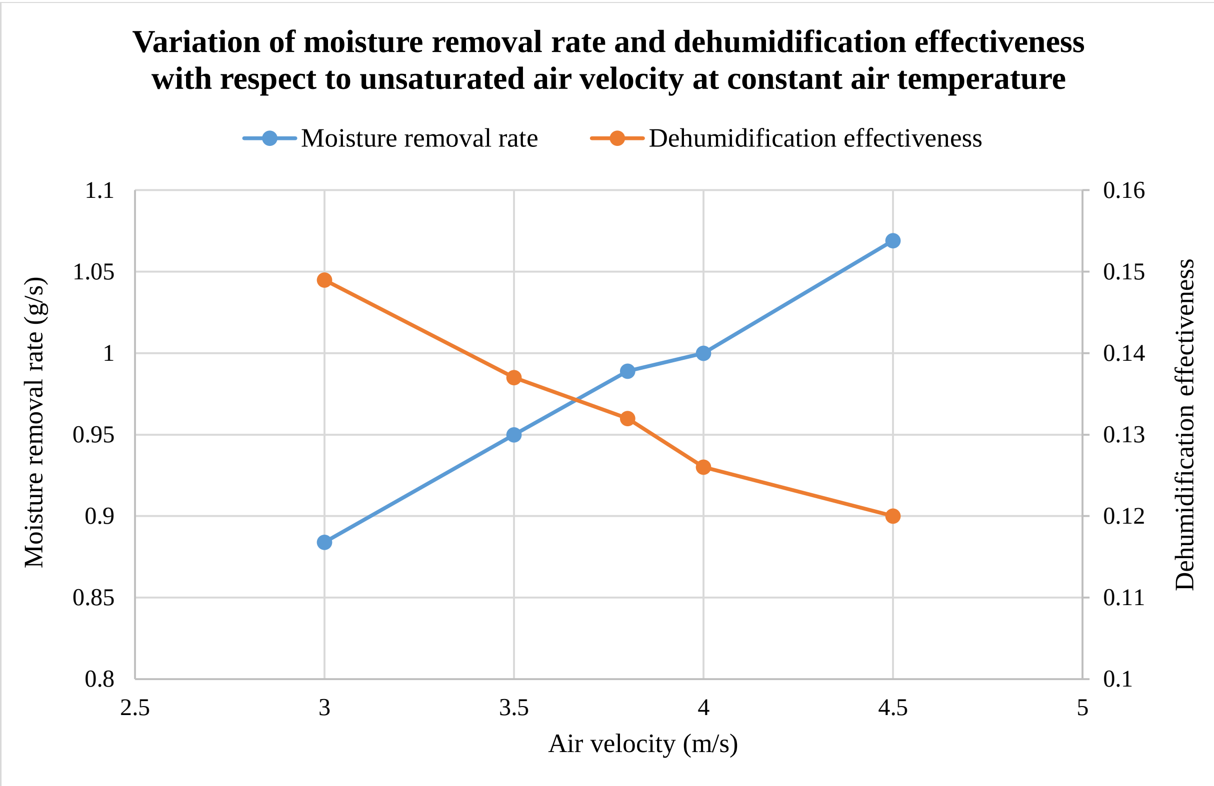

Figure 35: Variation of moisture removal rate and dehumidification effectiveness with unsaturated air velocity

MRR increases with the increase in unsaturated air velocity. It is due to the fact that an increase in air velocity increases the mass transfer coefficient between desiccant solution and air flow, and hence MRR increases from0.884 to 1.069g/s as the air velocity increases. Dehumidifier effectiveness decreases with the increase in unsaturated air velocity. When the air velocity increases, the outlet specific humidity of air increases due to reduced contact time for air with the desiccant solution in the dehumidifier. So, the dehumidification effectiveness decreases from 0.149 to 0.120 as the air velocity increases.

(ii) Variation of COP of system at different air velocity

Figure 36 shows the variation of COP of system with respect to unsaturated air velocity. The air velocity varies from 3 to 4.5 m/s at constant air temperature of 35˚C.

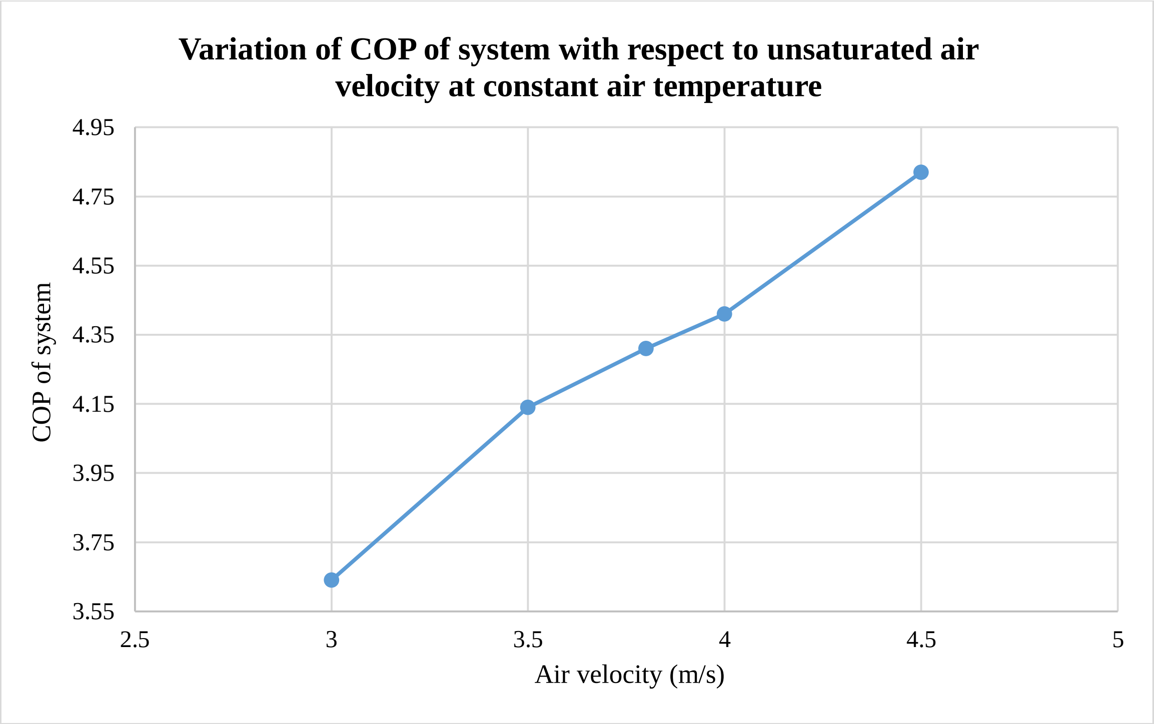

Figure 36: Variation of COP of system with unsaturated air velocity

For the COP of system, the latent load is covered by LDDS and sensible load is covered by VCRS. COP of system increases with the increase in unsaturated air velocity. The reason for this is that when the air velocity increases, mass flow rate of air increases that causes the increases in the refrigeration effect, and hence COP of system increases from 3.64 to 4.82 as the air velocity increases.

(iii) Variation of COP of VCRS without LDDS at different air velocity

Figure 37 illustrate the variation of COP of VCRS without LDDS with respect to unsaturated air velocity. The air velocity varies from 3 to 4.5 m/s by keeping constant air velocity of 35˚C.

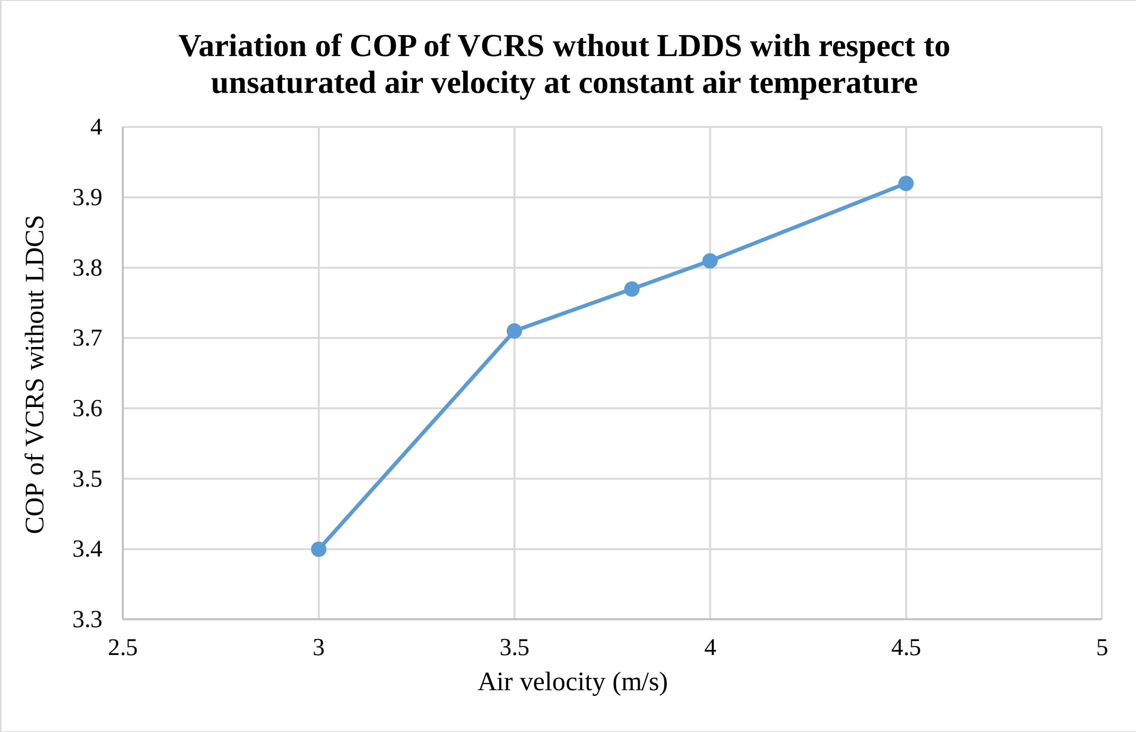

Figure 37: Variation of COP of VCRS without LDDS with unsaturated air velocity

The COP of VCRS without LDDS increases with the increase in unsaturated air velocity. In fact, an increase in air velocity increases the mass flow rate of air that will leads to increase in refrigeration effect, and hence COP of VCRS without LDDS increases from 3.40 to 3.92 as air velocity increases.

(iv) Comparison of COP of system and COP of VCRS without LDDS at different air velocity

Figure 38 shows the comparison of COP of system and COP of VCRS with respect to unsaturated air velocity. The air velocity varies from 3 to 4.5 m/s by keeping constant air temperature of 35˚C. The figure shows that the COP of system increases more as compare to the COP of VCRS without LDDS as the unsaturated air velocity increases. In fact, with an increase in air velocity, the enthalpy difference of air within the system is more than VCRS due to which refrigeration effect is more within the system than VCRS. Also for the

COP of system, the total load (sensible and latent load) of air is covered by both LDDS and VCRS, while for the COP of VCRS without LDDS, the total load of air is covered by VCRS only. So, the increment in COP of system is more than the COP of VCRS without LDDS as the air velocity increases.

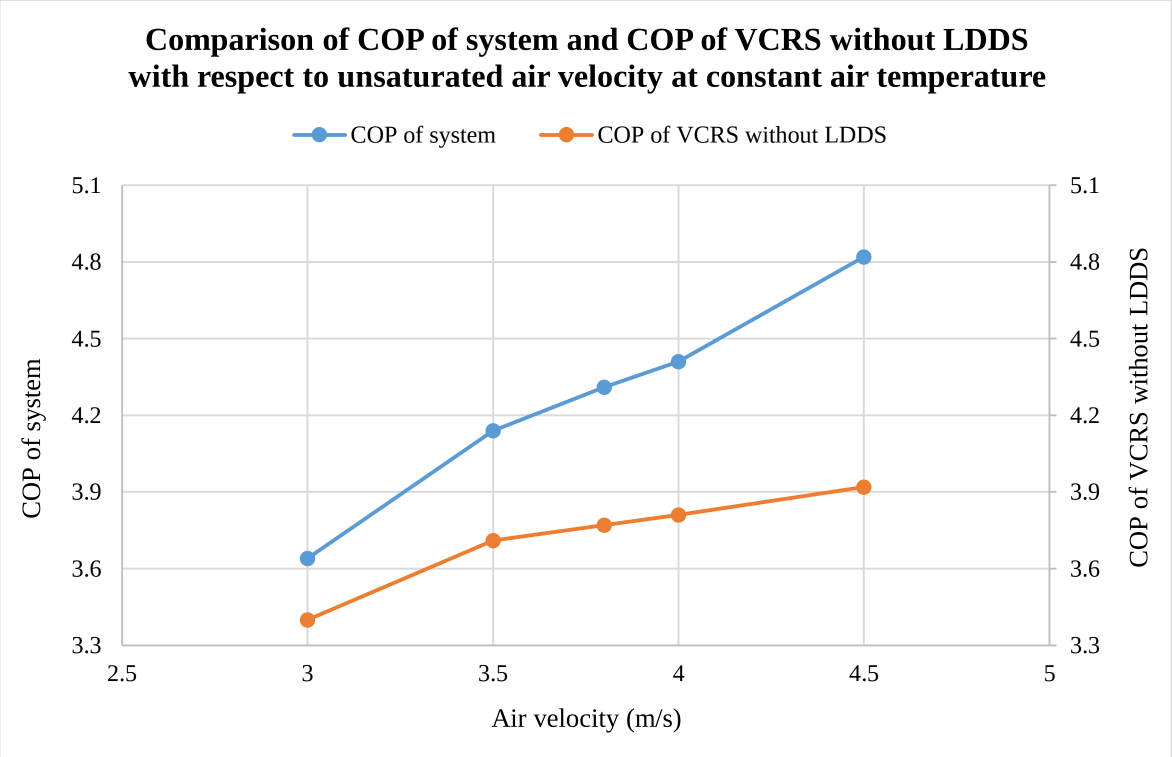

Figure 38: Comparison of COP of system and COP of VCRS without LDDS with unsaturated air velocity

9.1.3. Case 3: Saturated air with different temperature at constant air velocity

Saturated air means the air that contains maximum amount of water vapour. The saturation of air is done by spraying water with the help of spray pump in the humidification chamber. In this case the saturated air temperature varies from 30 to 40˚C with the help of air heater by keeping the constant air velocity of 4.0 m/s. The performance parameters like moisture removal rate (MRR), dehumidification effectiveness, COP of system, and COP of VCRS without LDDS are calculated at various air temperature.

(i) Variation of moisture removal rate (MRR) dehumidification effectiveness (DE) at different air temperature

Figure 39 shows the variation of moisture removal rate and dehumidification effectiveness with respect to saturated air temperature. The air temperature varies from 30 to 40˚C by keeping constant air velocity of 4.0 m/s. MRR decreases with the increase in air saturated temperature. The reason for this that the increase in air temperature decreases the vapour pressure difference between the desiccant solution and the process air. Due to decrease in difference of vapour, the driving force for mass transfer reduces, hence MRR decrease from 1.058 to 0.788g/s when the air temperature increases.

Dehumidification effectiveness decreases with the increase in saturated air temperature. In fact, the increase in air temperature causes the decrease in difference of specific humidity ratio of air due to decrease in difference of vapour pressure between desiccant solution and air in the dehumidifier. Hence dehumidification effectiveness decreases from 0.148 to 0.061 with the increase in air temperature.

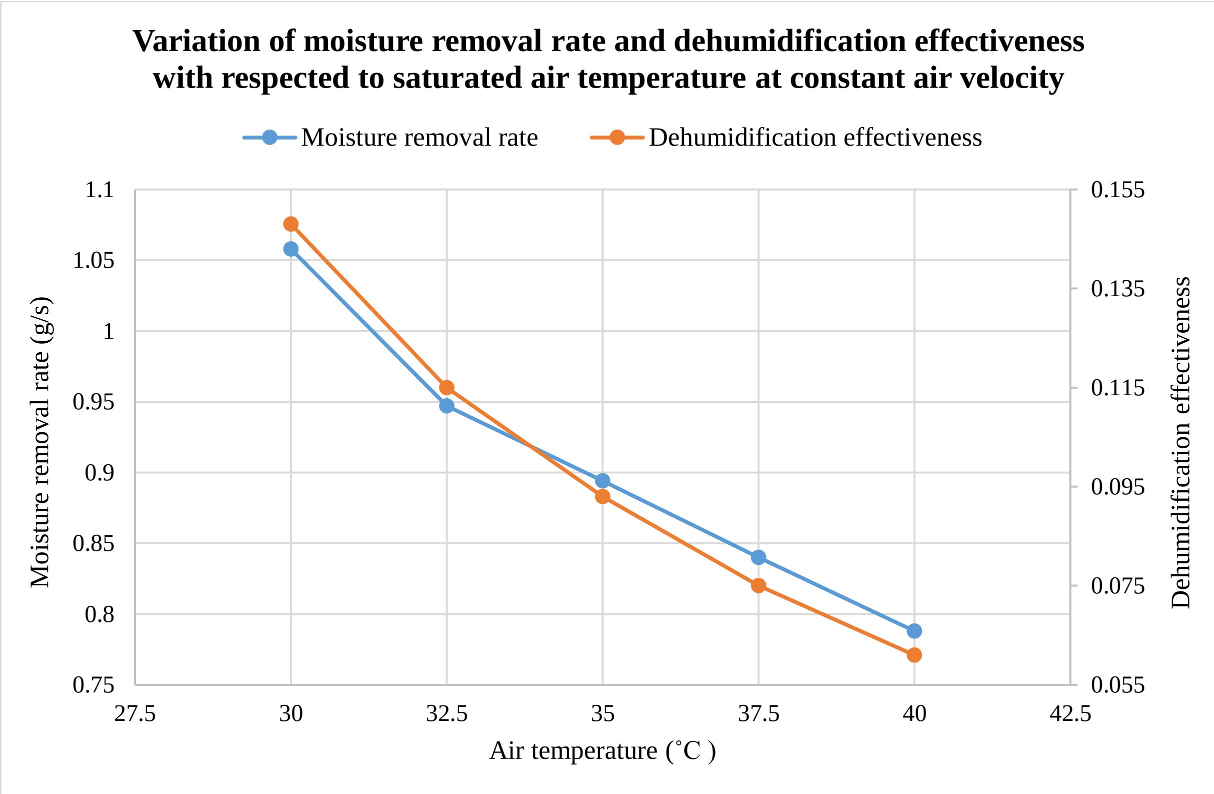

Figure 39: Variation of moisture removal rate and dehumidification effectiveness with saturated air temperature

(ii) Variation of COP of system at different air temperature

Figure 40 illustrate the variation of COP of system with respect to saturated air temperature. The air temperature varies from 30 to 40˚C by keeping constant air velocity of 4.0 m/s. The COP of system increases with the increase in saturated air temperature. The reason for this is that the increase in air temperature causes the increases the enthalpy difference of air that will leads to increase in refrigeration effect. So the COP of system increases from 2.75 to 5.84 with the increase in air temperature.

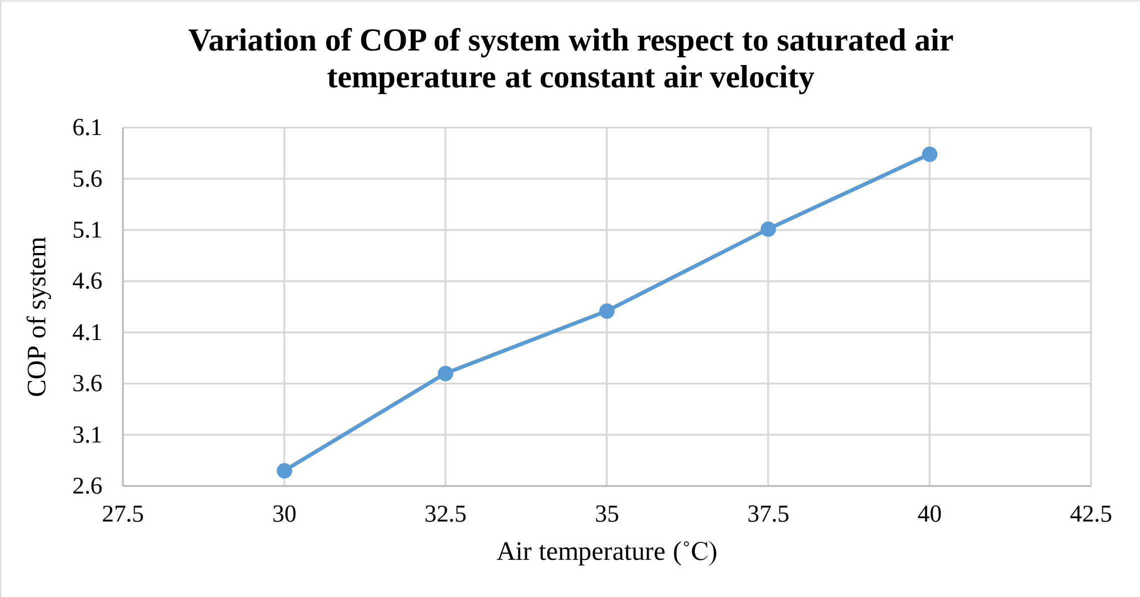

Figure 40: Variation of COP of system with saturated air temperature

(iii) Variation of COP of VCRS without LDDS at different air temperature

Figure 41 illustrate the variation of COP of VCRS without LDDS with respect to saturated air temperature. The air temperature increases from 30 to 40˚C at constant air velocity of 4.0 m/s.

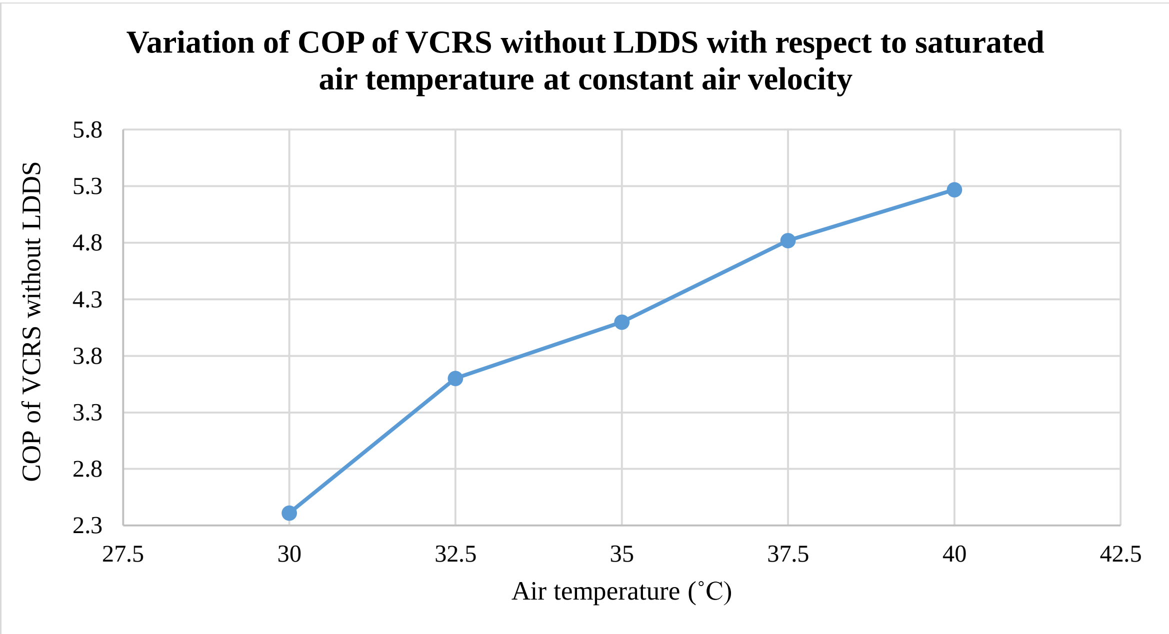

Figure 41: Variation of COP of VCRS without LDDS with saturated air temperature

COP of VCRS without LDDS increases with the increase in saturated air temperature. When the air temperature increases, enthalpy difference of air increases due to increase in sensible load of air that will leads to increase in cooling effect, and hence COP of VCRS without LDDS increases from 2.41 to 5.27 with the increase in air temperature.

(iv) Comparison of COP of system and COP of VCRS without LDDS at different air temperature

Figure 42 illustrate the comparison of COP of system and COP of VCRS without LDDS with respect to saturated air temperature. The air temperature varies from 30 to 40˚C at constant air velocity of 4.0 m/s.

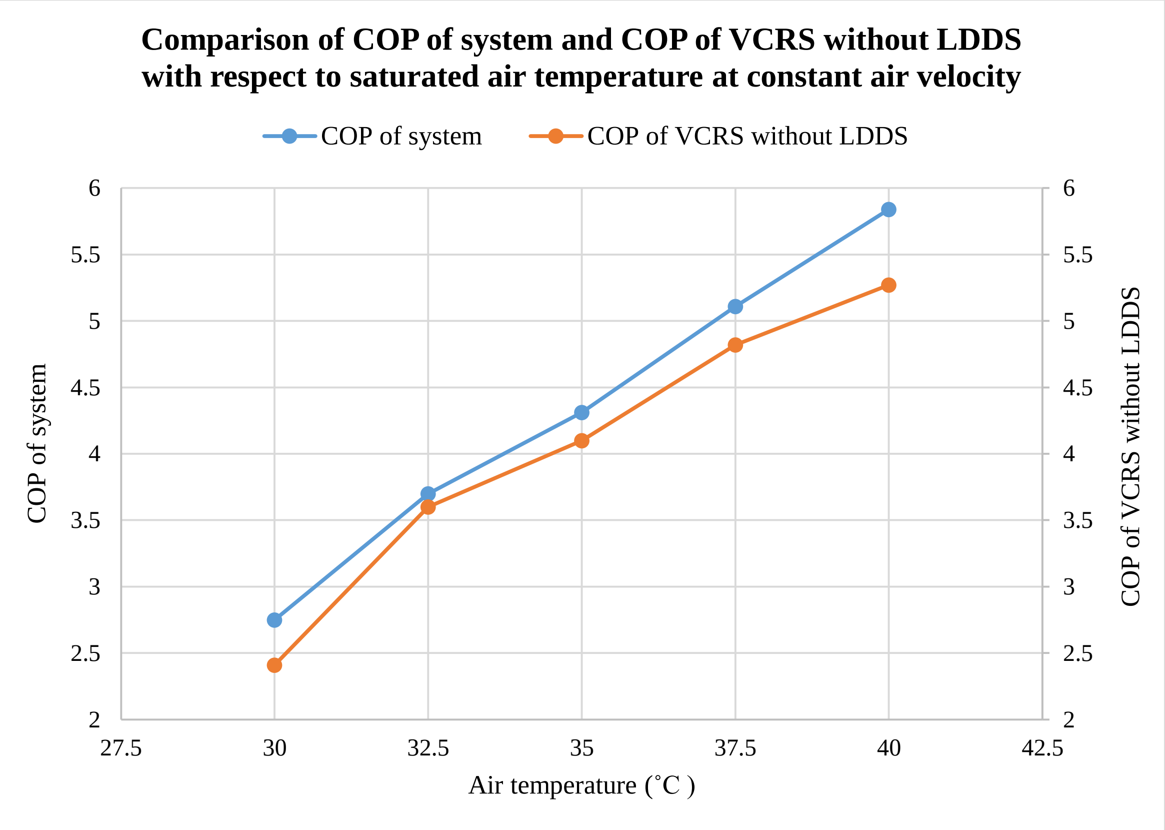

Figure 42: Comparison of COP of system and COP of VCRS without LDDS with saturated air temperature

It is clear from the figure that the increment in COP of system is more as compare to COP of VCRS without LDDS with the increase in saturated air temperature. The reason for this is that with an increase in in air temperature, the enthalpy difference of air within the system is more as compare to the VCRS. Also for the COP of system, the total load (sensible and latent load) of air is covered by both LDDS and VCRS but for the COP of VCRS without

LDDS, the total load of air is covered by VCRS only. Hence the COP of system increases more as compare to COP of VCRS without LDDS when the air temperature increases.

9.1.4. Case 4: Saturated air with different velocity at constant air temperature

The saturation of air is done by spraying water with the help of spray pump in the humidification chamber. In this case the saturated air velocity varies from 3 to 4.5 m/s with the help of fan of regulator by keeping constant air temperature of 35˚C. The performance parameters like moisture removal rate (MRR), dehumidification effectiveness, COP of system, and COP of VCRS without LDDS are calculated at various air temperature.

(i) Variation of moisture removal rate (MRR) and dehumidification effectiveness (DE) at different air velocity

Figure 43 shows the variation of moisture removal rate and dehumidification effectiveness with respect to saturated air velocity. The air velocity varies from 3.0 to 4.5 m/s by keeping constant air temperature of 35˚C.

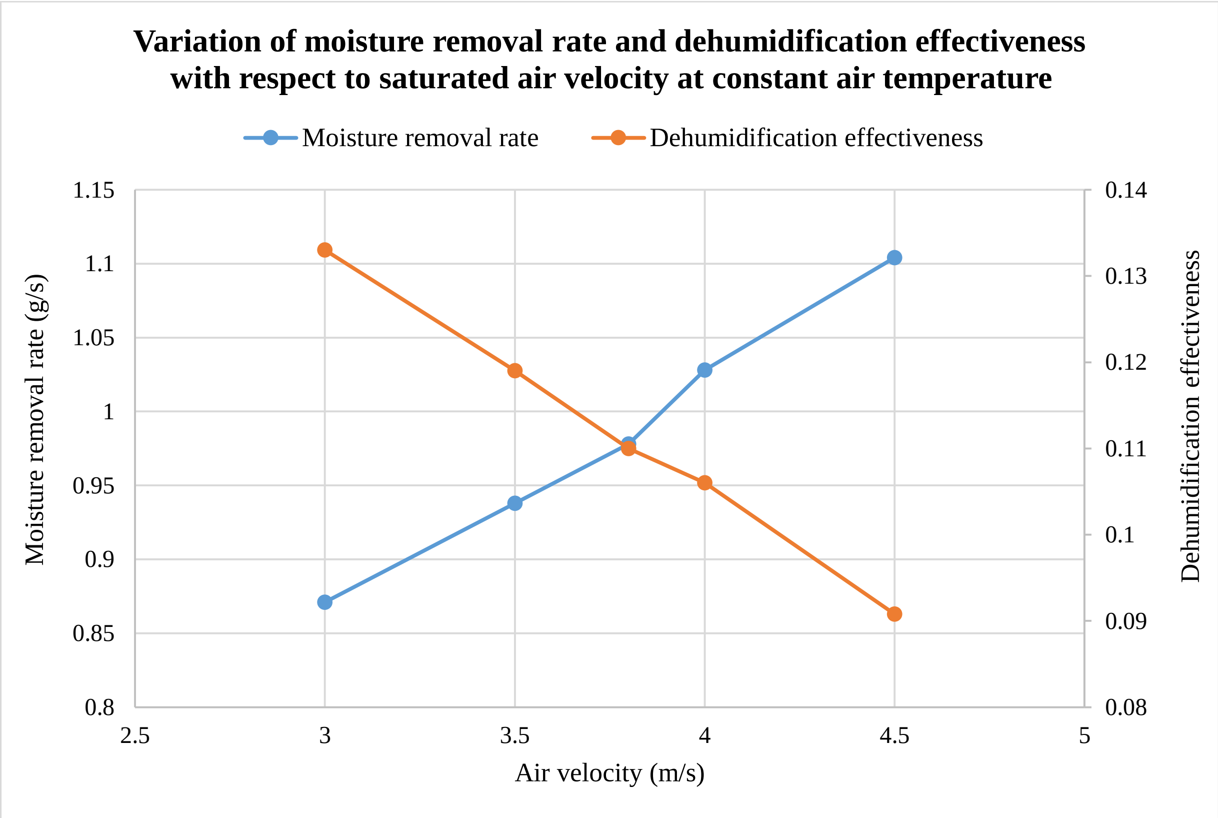

Figure 43: Variation of moisture removal rate and dehumidification effectiveness with saturated air velocity

MRR increases with the increase in saturated air velocity. In fact, the increase in air velocity increases the air flow rate that causes the increase in mass transfer coefficient between desiccant solution and air, hence MRR increases from 0.871 to 1.104g/s as the air velocity increases.

Dehumidification effectiveness decreases with the increase in saturated air velocity. The reason for this is that the increase in air velocity causes the increase in specific humidity of outlet air due reduced contact time between desiccant solution and air in the dehumidifier, as a result dehumidification effectiveness decreases from 0.149 to 0.120 as the air velocity increases.

(ii) Variation of COP of system at different air velocity

Figure 44 illustrate the COP of system with respect to saturated air velocity. The air velocity varies from 3.0 to 4.5 m/s by keeping constant air temperature of 35˚C.

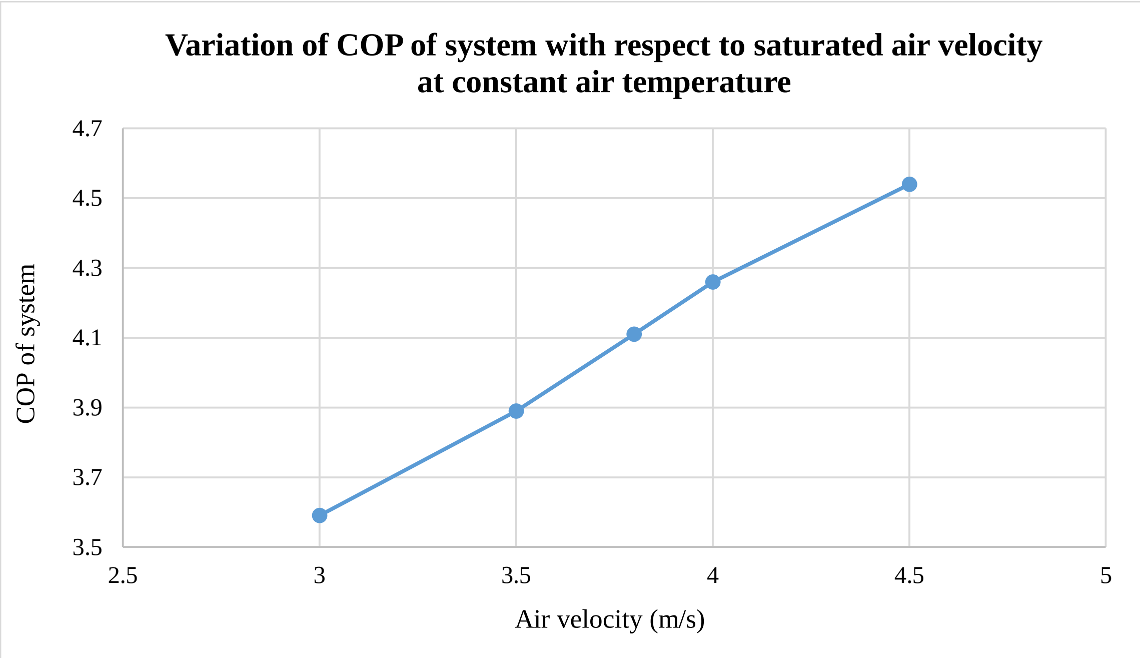

Figure 44: Variation of COP of system with saturated air velocity

For the COP of system, the total load (sensible and latent load) of air is covered by both LDDS and VCRS. The COP of system increases with the increase in saturated air velocity. In fact, the increase in air velocity causes the increase in mass flow rate of air and a very slight decrement in enthalpy difference of air, hence refrigeration effect increases due to which COP of system increases from 3.59 to 4.54 as air velocity increases.

(iii) Variation of COP of VCRS at different air velocity

Figure 45 shows the variation of COP of VCRS without LDDS with respect to saturated air velocity. The air velocity varies from 3.0 to 4.5 m/s by keeping constant air temperature of 35˚C.

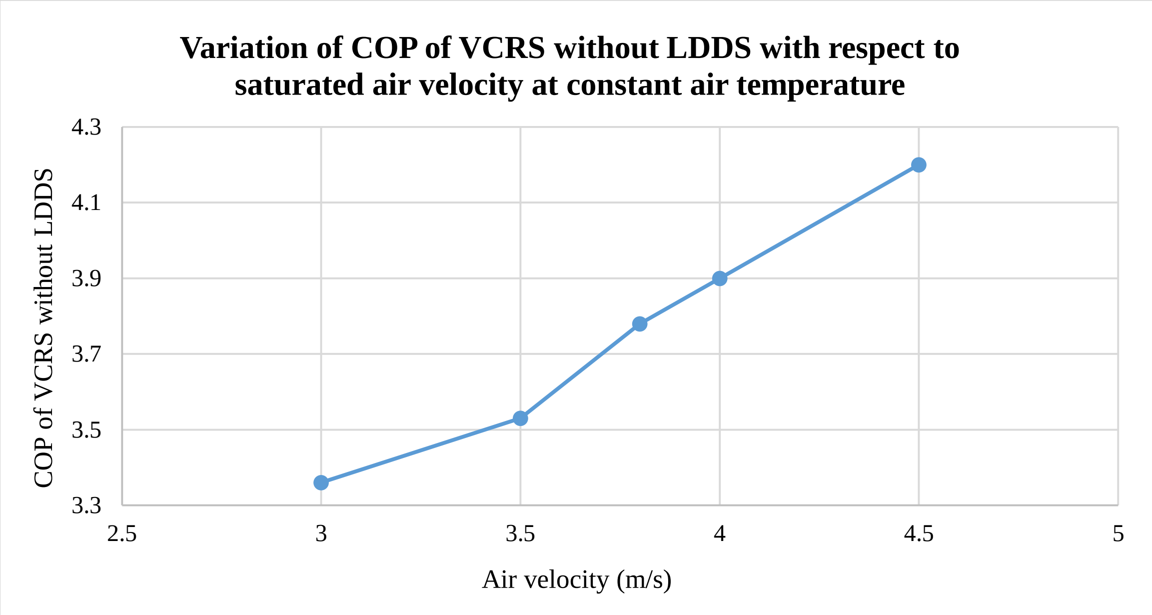

Figure 45: Variation of COP of VCRS without LDDS with saturated air velocity

COP of VCRS without LDDS increases with the increase in saturated air velocity. The reason for this is that with an increase in air velocity causes the increase in mass flow rate of air that will leads to increase in refrigeration effect, and hence the COP of VCRS without LDDS increases from 3.36 to 4.20 as the air velocity increases.

(iv) Comparison of COP of system and COP of VCRS at different air velocity

Figure 46 illustrate the comparison of COP of system and COP of VCRS without LDDS with respect to saturated air velocity. The air velocity varies from 3.0 to 4.5 m/s by keeping constant air temperature of 35˚C. The figure shows that the increment in COP of system is more as compare to the COP of VCRS as the saturated air velocity increases. In fact, due to the increase in air velocity, the enthalpy difference of air within the system is more as compare to the VCRS, hence the refrigeration effect within the system is more as compare to the VCRS. So, the increment in COP of system is more as compare to the COP of VCRS without LDDS as the air velocity increases. Also for the COP of system the total load (sensible and latent load) of air is handled by both LDDS and VCRS while for the COP of VCRS the total load of air is handled by VCRS only.

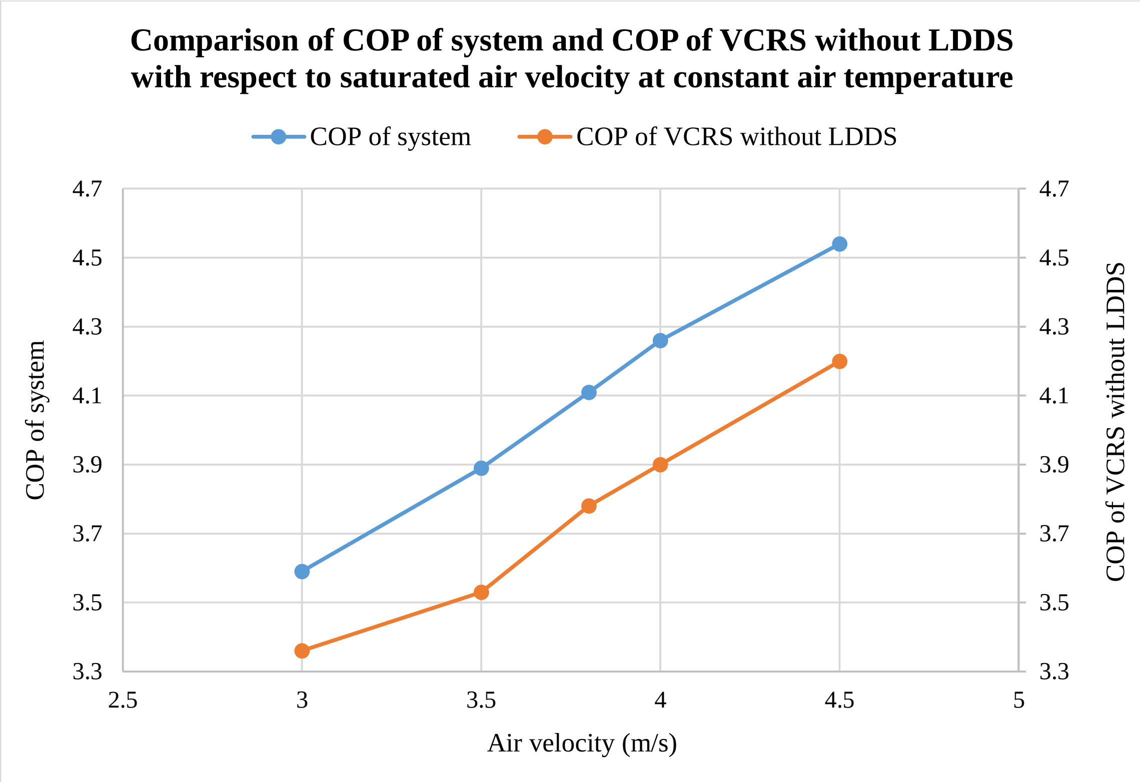

Figure 46: Comparison of COP of system and COP of VCRS without LDDS with saturated air velocity

CHAPTER 10

10. Conclusions and future scope

10.1. Conclusion

In the present study, the performance of a liquid desiccant dehumidification system integrated with vapour compression refrigeration system has been analyzed. The following conclusions have been drawn from the results:

I. MRR decreases from 0.92 to 0.72g/s, dehumidification effectiveness decreases from 0.135 to 0.088, COP of system increases from 2.7 to 4.86, and COP of VCRS without LDDS increases from 2 to 4.7, when the unsaturated air temperature varies from 30 to 40˚C at constant air velocity of 4.0m/s.

II. MRR increases from 0.884 to1.069g/s, dehumidification effectiveness decreases from 0.149 to 0.120, COP of system increases from 3.64 to 4.82, and COP of VCRS without LDDS increases from 3.4 to 3.92, when the unsaturated air velocity varies from 3 to 4.5m/s by keeping constant air temperature of 35˚C.

III. MRR decreases from 1.058 to 0.788g/s, dehumidification effectiveness decreases from 0.148 to 0.061, COP of system increases from 2.75 to 5.84, and COP of VCRS without LDDS increases from 2.41 to 5.27, when the saturated air temperature varies from 30 to 40˚C at constant air velocity of 4.0 m/s.

IV. MRR increases from 0.871 to 1.104g/s, dehumidification effectiveness decreases from 0.133 to 0.098, COP of system increases from 3.59 to 4.54, and COP of VCRS without LDDS increases from 3.36 to 4.20, when the saturated air velocity varies from 3 to 4.5m/s by keeping constant air temperature of 35˚C.

10.2. Future scope

The enhancement in the design of components of hybrid cooling systems for better performance and nominal loses is important to consider. Supplementary changes that can be employed in the proposed experimental set-up to improve the performance of the system are:

I. Internal cooling arrangement can be employed in dehumidifier to counter the heat generation during moisture absorption and better dehumidification.

II. Drift eliminator can be used to reduce the problems of carry-over of desiccant in case of structural packed bed dehumidifier/regenerator.

CHAPTER 11

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APPENDIX

Case 1: Unsaturated air with different temperature at constant air velocity

Readings

CONDITIONS OF AIR AT DEHUMIDIFIER INLET					
Velocity (m/s)	**DBT (˚C)**	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
4.0	30	23	55	14.8	67.97
4.0	32.5	24.4	52	15.9	73.45
4.0	35	25.5	47	16.7	77.97
4.0	37.5	26.6	43	17.5	82.7
4.0	40	27.5	39	18.1	88.67
CONDITIONS OF AIR AT DEHUMIDIFIER OUTLET					
	DBT (˚C)	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
	29.5	21.5	50	12.8	62.36
	31.5	22.9	48	14	67.52
	34	24.15	45	14.9	72.36
	36.4	25.3	41	15.8	77.05
	39	26.4	37	16.5	81.74
CONDITIONS OF AIR AT EVAPORATOR OUTLET (WITH LDDS)					
	DBT (˚C)	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
	26	19	52	10.9	53.84
	27	19.5	50	11.1	55.47
	28	20	48	11.3	57.12
	29	20.5	46	11.6	58.82
	30	21	45	11.9	60.54
CONDITIONS OF AIR AT EVAPORATOR OUTLET (WITHOUT LDDS)					
	DBT (˚C)	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
	28	21.5	57	13.4	62.41
	29	22.3	56	14.2	65.34
	30	23	55	14.8	67.97
	31	23.67	54	15.4	70.56
	32	24.33	53	16	73.18

Results after calculation

MRR (g/s)	ε_{deh}	**COP of system**	**COP of VCRS without LDDS**	**ρ(kg/m^3)**	**m_a (kg/s)**
0.92	0.135	2.7	2	1.165	0.46
0.867	0.120	3.42	2.89	1.156	0.456
0.814	0.108	3.93	3.53	1.146	0.452
0.763	0.097	4.48	4.26	1.137	0.449
0.72	0.088	4.85	4.7	1.128	0.445

Case 2: Unsaturated air with different velocity at constant air temperature

Readings

CONDITIONS OF AIR AT DEHUMIDIFIER INLET					
Velocity (m/s)	**DBT (˚C)**	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
3.0	35	26	49	17.5	80.15
3.5	35	26	49	17.5	80.15
3.8	35	26	49	17.5	80.15
4.0	35	26	49	17.5	80.15
4.5	35	26	49	17.5	80.15
CONDITIONS OF AIR AT DEHUMIDIFIER OUTLET					
	DBT (˚C)	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
	33.5	24	46	14.9	71.78
	33.8	24.2	45	15.1	72.58
	34.4	24.42	44	15.2	73.45
	34.9	24.62	43	15.3	74.25
	35.5	24.89	42	15.4	75.35
CONDITIONS OF AIR AT EVAPORATOR OUTLET (WITH LDDS)					
	DBT (˚C)	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
	27	19.2	48	10.7	54.47
	27.5	19.4	47	10.8	55.12
	28	19.7	46	10.9	56.11
	28.5	19.9	45	11	56.77
	29	20.1	44	11.1	57.43
CONDITIONS OF AIR AT EVAPORATOR OUTLET (WITHOUT LDDS)					
	DBT (˚C)	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
	29	22.83	59	15	67.35
	29.5	23.05	58	15.1	68.18
	30	23.25	57	15.2	68.94
	30.3	23.4	56	15.3	69.52
	30.8	23.6	54	15.4	70.29

Results after calculation

MRR (g/s)	ε_{deh}	**COP of system**	**COP of VCRS without LDDS**	**ρ(kg/m^3)**	**m_a (kg/s)**
0.884	0.149	3.64	3.40	1.146	0.34
0.95	0.137	4.14	3.71		0.396
0.989	0.132	4.31	3.76		0.43
1.0	0.126	4.41	3.81		0.452
1.069	0.120	4.82	3.92		0.509

Case 3: Saturated air with different temperature at constant air velocity

Readings

CONDITIONS OF AIR AT DEHUMIDIFIER INLET					
Velocity (m/s)	**DBT (˚C)**	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
4.0	30	23.5	58	15.6	69.93
4.0	32.5	25.8	59	18.2	79.38
4.0	35	28.5	60	21.5	90.32
4.0	37.5	30.6	61	25.2	102.48
4.0	40	32.8	62	29.5	114.84
CONDITIONS OF AIR AT DEHUMIDIFIER OUTLET					
	DBT (˚C)	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
	29	21.6	55	13.3	63.11
	32	24.4	55	16.1	73.47
	34.5	27	59	19.5	84.67
	37	29.55	60	23.3	69.97
	39.5	31.9	61	27.2	109.6
CONDITIONS OF AIR AT EVAPORATOR OUTLET (WITH LDDS)					
	DBT (˚C)	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
	26	19.5	55	11.5	55.5
	27.5	20.75	55	12.6	59.75
	29	22.8	59	14.9	67.24
	31	24.7	60	17.4	74.76
	33	26.6	61	19.4	82.92
CONDITIONS OF AIR AT EVAPORATOR OUTLET (WITHOUT LDDS)					
	DBT (˚C)	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
	27.5	21.7	60	13.9	63.16
	29	23.5	63	16	69.97
	31	25.4	65	18.5	78.58
	32.5	27.8	70	21.8	88.52
	34.5	30	72	25.3	99.45

Results after calculation

MRR (g/s)	ε_{deh}	**COP of system**	**COP of VCRS without LDDS**	**ρ(kg/m³)**	**m_a (kg/s)**
1.058	0.148	2.75	2.41	1.154	0.456
0.947	0.115	3.70	3.60	1.143	0.451
0.894	0.093	4.31	4.1	1.132	0.447
0.84	0.075	5.11	4.82	1.12	0.442
0.788	0.061	5.84	5.27	1.109	0.438

Case 4: Saturated air with different velocity at constant air temperature

Readings

CONDITIONS OF AIR AT DEHUMIDIFIER INLET					
Velocity (m/s)	**DBT (˚C)**	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
3.0	35	27.2	55	19.6	85.57
3.5	35	27.5	57	20.2	86.97
3.8	35	27.9	59	20.9	88.87
4.0	35	28.3	61	21.7	90.81
4.5	35	28.7	63	22.4	92.77
CONDITIONS OF AIR AT DEHUMIDIFIER OUTLET					
	DBT (˚C)	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
	33.8	25.41	51	17	77.59
	34	25.9	53	17.8	79.57
	34.2	26.4	55	18.6	81.96
	34.6	27	56	19.4	84.67
	35	27.5	57	20.2	86.97
CONDITIONS OF AIR AT EVAPORATOR OUTLET (WITH LDDS)					
	DBT (˚C)	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
	28	20.8	53	12.4	59.9
	28.5	21.7	55	13.5	63.13
	29	22.4	57	14.3	65.17
	29.5	23	58	15	67.99
	30	23.8	60	16	71.12
CONDITIONS OF AIR AT EVAPORATOR OUTLET (WITHOUT LDDS)					
	DBT (˚C)	**WBT (˚C)**	**Relative humidity (%)**	**Humidity ratio (g/kg of dry air)**	**Enthalpy (kJ/kg)**
	30	24.2	62	16.7	72.74
	30.5	24.85	63	17.4	75.41
	31	25.33	64	18.1	77.5
	31.4	25.85	65	18.8	79.65
	31.8	26.4	66	19.6	82.07

Results after calculation

MRR (g/s)	ε_{deh}	COP of system	COP of VCRS without LDDS	$\rho (kg/m^3)$	m_a (kg/s)
0.871	0.133	3.59	3.36	1.133	0.335
0.938	0.119	3.89	3.53	1.132	0.391
0.978	0.110	4.11	3.78	1.132	0.425
1.028	0.106	4.26	3.90	1.131	0.447
1.104	0.098	4.54	4.20	1.131	0.502