MATHEMATICAL MODELLING
OF
DESICCANT DEHUMIDIFIER
FOR AIR CONDITIONING SYSTEM

Dissertation submitted in the partial fulfillment of requirements for the award of degree

Of

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in

Thermal Engineering

Submitted by

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CERTIFICATE AND DECLARATION

I hereby declare that the thesis report entitled "**Mathematical Modeling of Desiccant Dehumidifier for Air Conditioning System"** is an authentic record of my study carried out as requirement for the award of degree of Master of Engineering in Thermal Engineering at Thapar University, Patiala, under the supervision of **Dr. Madhup Kumar Mittal, Assistant Professor**, Mechanical Engineering Department (MED). The matter presented for dissertation has not been submitted in any other University / Institute for the award of degree.

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It is certified that the above statement made by the student is correct to the best of my knowledge and belief.

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NOMENCLATURE

h	enthalpy (J/kg)	h_{mass}	mass transfer coefficient(m/s)
t	temperature (K)	Le	Lewis number
ω	humidity ratio	h_s	heat transfer coefficient(W/m^2. ℃)
C_P	specific ratio (J/kg/K)	U	overall transfer coefficient
X	desiccant solution conc(kg/kg soln.)	NTU	Number of Transfer Units
$\emptyset$	reduced temperature	C	Capacity ratio
m	mass flow rate (kg/s)		
h_{fg}	enthalpy of vaporization(J/kg)		**subscripts**
p_v	vapour pressure(Pa)	w	water
D_{AB}	binary diffusion coefficient(m^2/s)	a	air
μ	dynamic viscosity(N.s/m^2)	s	desiccant solution
ρ	density(kg/m^3)	fg	vaporization
ϑ	kinematic viscosity(m^2/s)	1	inlet
Sc	Schmidt number	2	outlet
Sh	Sherwood number	cond.	condensate
Re	Reynolds number	w	wetting
β	equivalent particle diameter(m)	t	total
v	superficial velocity(m/s)	c	critical
ε	packed bed voidage	equ	equilibrium
a	area(m^2)		
σ	surface tension(N/m)		
g	acceleration due to gravity(m/s^2)		
L	characteristic length(m)		
dp	diameter of packing(m)		

ABSTRACT

Air conditioning has become one of the necessities to symbolize the improved living standard. Moreover, the increase in demand for comfort and over occupation of public places put undue pressure on air conditioning system and hence pressure on energy sources. Therefore, the need of the hour is to develop such system which should be more energy efficient.

The major load on the conventional air condition systems is the latent heat load i.e. the removal of the moisture from the outdoor air to improve the standard of the supply air. This load can be separated from the air-conditioning system by using Desiccant Technology in combination with conventional air-conditioning system. This leads to the development of Desiccant-Vapour Compression Hybrid Air Conditioning System.

In the present study a methodology has been presented for the study of the parameters which affect the working of the heart and soul of desiccant system i.e. the dehumidifier. An analytical mathematical model has been developed for the dehumidifier by using various property and conservation equations. In this study Lithium Chloride solution has been used as the desiccant and the model shows good agreement with the experimental results obtained by the previous researchers.

The developed model has been used to find the variation of condensate mass flow rate with the change in inlet air temperature. These results have been compared with the results obtained from the experimental results by Fumo et al.[8]. The modeled results are found to be within 10.62% deviation from the modeled results. Further the model has been used to study the effect of various inlet parameters like inlet desiccant temperature, inler air mass flow rate, inlet desiccant concentration and inlet air humidity on the performance of the dehumidifier. . It has been found that as the inlet desiccant temperature is decreased from 35℃ to15℃ , the reduction in humidity ratio of process air is increased by 370%.

CHAPTER – 1 INTRODUCTION

Due to the improvement in the living standards of people the cooling of the residential and work places is becoming more common. Another reason for increasing air conditioning load is building architectural characteristic and trend, like an increasing ratio of transparent to opaque surfaces in the building envelope to even the popular glass buildings. This has put undue pressure on the energy resources. HVAC system consume approximately 50% of total household energy consumption,[1]. Hence the need of the hour is to develop such systems for air conditioning that can save the energy consumption. Recently several companies started development of water chillers in the power range below 50 kW down to 5 kW [2].

Keeping in view the above problems more economical system for the cooling is to be adopted. Keeping other things constant the building cooling load is directly proportional to the ventilation flow [3].

Past few decades have seen a large growth of the active desiccant cooling components as an important part of the air conditioning systems. Low grade heat is used as the primary energy source in this system with a small contribution of the electrical. Hence such a system will reduce a large portion of the load on the electricity grids. Moreover as low grade heat is used as the primary energy source, waste heat from any industrial plant or even from the condenser of a conventional vapour compression system can be used. This results in the improved efficiency of the system. Since the desiccant cooling system alone cannot bear the complete cooling load therefore it has to be used in conjunction with the conventional system. This results in the development of Desiccant Cooling – vapour compression Hybrid cooling system.

1.1 DESICCANTS

Desiccants are the substances that have high affinity for moisture. Hence, these substances can be used to control the humidity of a space. Most common application of desiccants is in the packaging industry where these help to prevent the deterioration of a product. The desiccants work by either adsorbing or absorbing the moisture. After these get saturated by the moisture, they can be regenerated by the use of hot air. Desiccants can either be solid e.g. silica gel or liquid e.g. hygroscopic salt solutions.

1.2 DESICCANTS IN AIR-CONDITIONING

Humidity control is an important aspect of air conditioning. High humidity level acts as a trigger for the mould growth which is directly linked to respiratory discomfort and allergies [4]. Since 1980’s the desiccant based air conditioning systems have been used in both industrial and residential applications. These systems have proved to be great cost reducers by lowering the latent heat load of the air conditioning equipment

Desiccant in solid or liquid state has great affinity for moisture. The moisture is either adsorbed or absorbed by the solid and liquid desiccant respectively. When the moisture is removed by the desiccant material, a good amount of heat known as heat of adsorption or heat of absorption is released. Hence the latent heat load is converted to the sensible heat load, which can then be removed by the conventional vapour compression system without requirement of lowering the temperature to the dew point temperature. Hence, the combined system of desiccant and conventional cooling i.e. hybrid desiccant-vapour compression air conditioning leads to saving of good deal of energy consumption.

After sometime the desiccant material gets saturated with moisture and needs to be regenerated. This can be achieved effectively by using heated dry air to pass through the desiccant material. This heated air can either be the indoor air or the outside air, which can be heated by using the heat source from some low grade source such as diesel engine or solar energy. Khalid [3] provided some of the low grade heat sources which could be used in the regeneration of the desiccant material as given in Table 1.1. Hence, this low grade heat could be used, which further improves the overall efficiency of the air conditioning system.

Table 1.1 : Various sources of low grade heat for the regeneration air [3]

Sr. No.	Source of heat	Temperature Range (℃)	Equipment needed for using heat	Humidity drop possible (g/kg dry air)
1	Waste water from Textile process mill	80-90	Air-liquid heat exchanger	5 – 6.5
2	Industrial diesel power generating sets	Jacket water (80 - 90)	Air liquid heat exchanger	5 – 6
		Exhaust gases (160 - 180)	Air air heat exchanger	8 – 9.5
3	Fuel cell	80 – 95	Air liquid heat exchanger	5 – 6.5
4	Heat rejected from Condenser	55 - 70	Air-air or air-liquid	3 - 4

1.3 TYPES OF DESICCANT SYSTEMS

Both solid and liquid desiccants have found their use in the air conditioning sector. Each of the desiccant system require its own configuration for proper and sustainable use. Various types of desiccant systems with different configurations have been proposed till date [5]. Each system has its own advantages and disadvantages. These configurations along with respective advantages and disadvantages are discussed below:

1.3.1 Liquid Spray Towers: The schematic of liquid spray tower configuration has been shown in fig. 1.1. In this configuration the process air enters the conditioner module from bottom and passes through a spray of liquid desiccant spray.

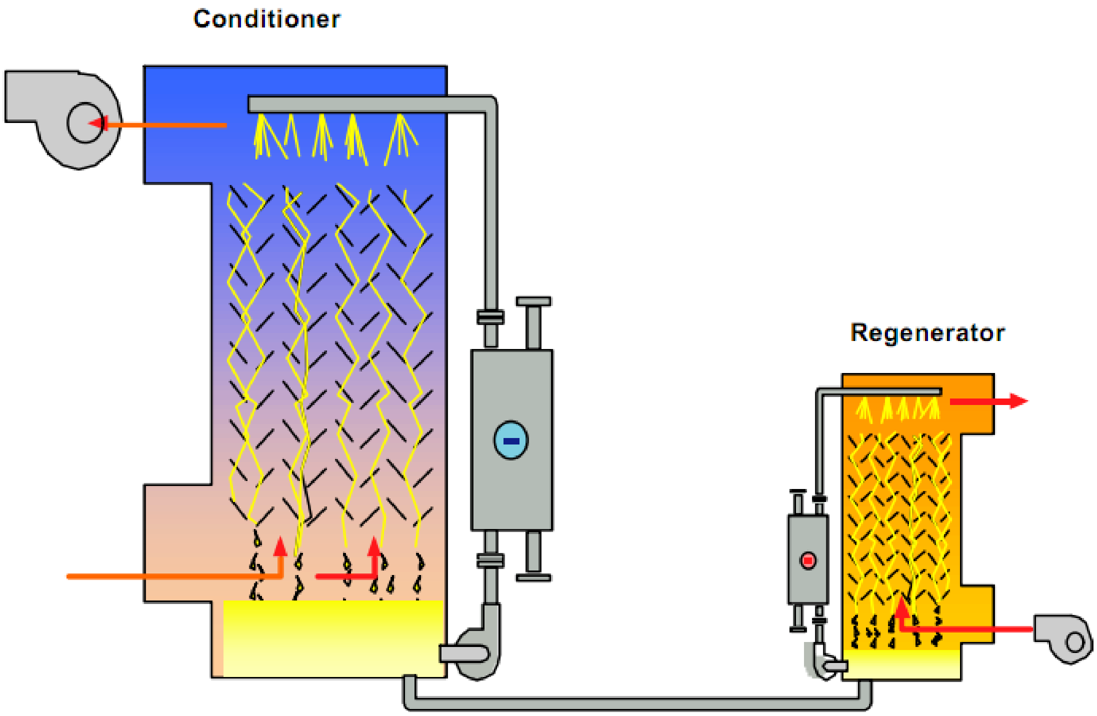


Fig. 1.1: Desiccant cooling system with liquid spray towers [5]

The humidified air is taken out from top. The saturated desiccant liquid is passed to the regenerator module where it is regenerated by the outside air.

The advantages of this configuration include large air flow rate, easy monitor of the desiccant quality and zero probability of process and regeneration air mixing.

The disadvantage is that at low sensible loads it is difficult to maintain RH below 10% with this configuration.

1.3.2 Solid Packed Tower: In this configuration solid desiccants like silica are used, which are impregnated on a suitable base to form a dehumidifier packed tower. Fig. 1.2 depicts the schematic of the configuration for solid packed towers.

In this configuration both process air and regeneration air pass through the solid desiccant packed bed.

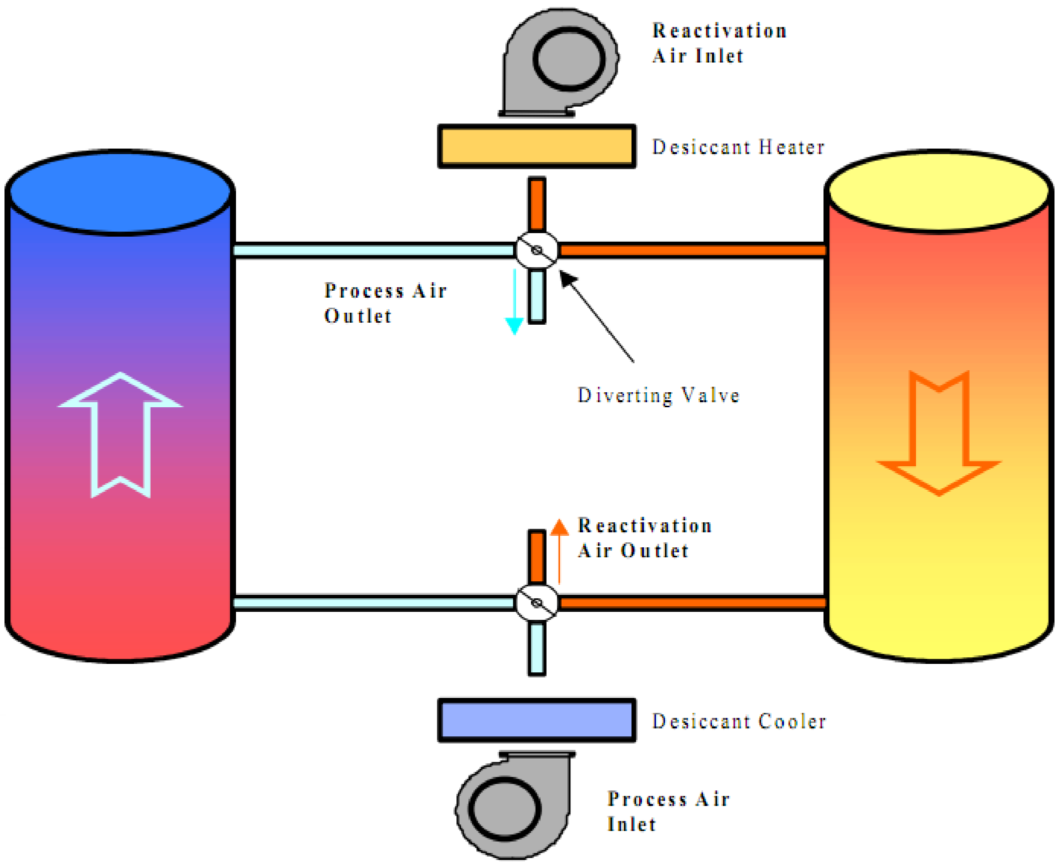


Fig. 1.2: Desiccant cooling system with solid packed desiccant tower [5]

The path of the air streams in diverted alternatively to the two cylinders simultaneously using a diverting valve.

The advantage of this system is that very low dew point temperatures can be achieved with a disadvantage that a critical air velocity is to be maintained for optimal performance.

1.3.3 Rotating Horizontal Bed

As shown in fig. 1.3, in this configuration the desiccant is retained by the horizontal rotating beds, through which the process and the regeneration air passes.

Advantages include simple design, constant outlet moisture content and low first cost. More over the dehumidification and the regeneration can take place side by side leading to a compact system.

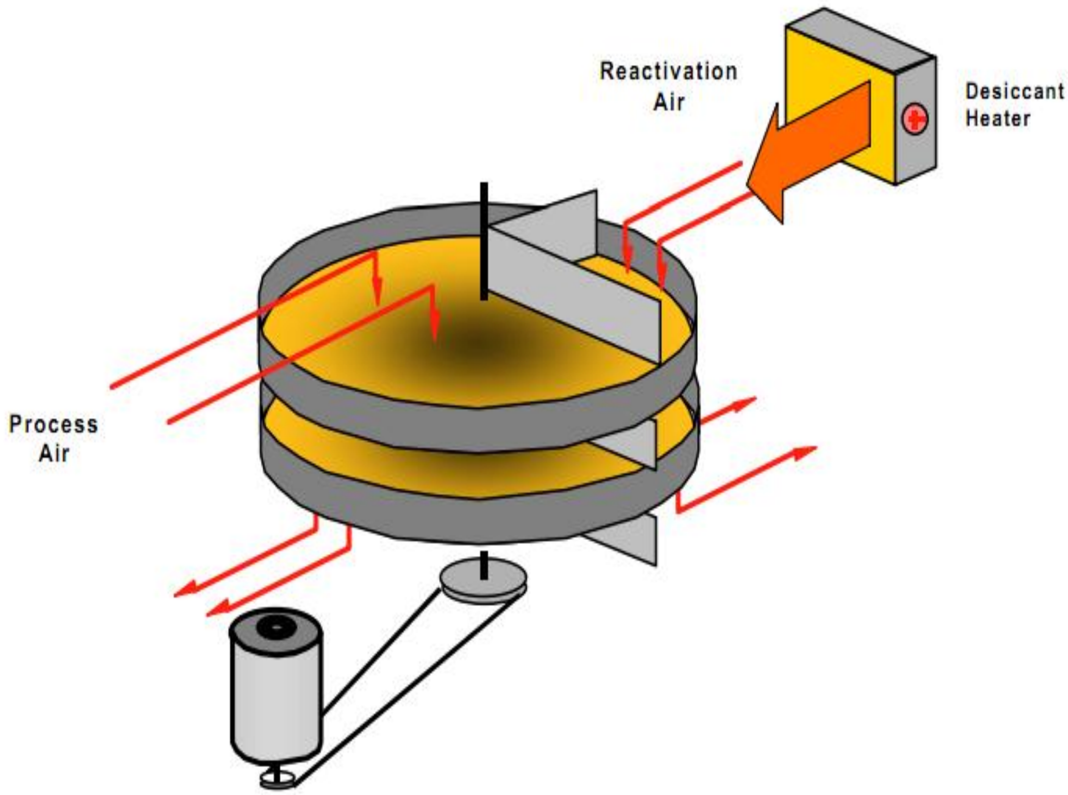


Fig. 1.3: Desiccant cooling system with rotating horizontal bed desiccant [5]

But there are large chances of leakage of regeneration air into the process air and vice-verse which reduces the efficiency of such systems.

Any of the above systems can be chosen based on the desiccant used and the system structural and parametrical design. There can be more number of configurations possible based on the need for particular application and the space availability.

1.4 ADVANTAGES OF DESICCANT SYSTEM

Desiccant based air conditioning systems have made a place in the air conditioning market in a very short time.

Some of the major advantages of using hybrid desiccant – vapour compression air conditioning systems are discussed below.

1) Since this system separates latent cooling from sensible cooling this system is simpler in construction.
2) The load is shifted from the high grade electrical energy to the low grade heat energy.
3) The major refrigerant used is air in this hybrid system which is environment friendly and non-toxic.
4) The desiccant dehumidification system operates at atmospheric pressure hence no need of robust piping and high strength joints.
5) This system helps to remove the drawback of the simple vapour compression cycle. In simple vapour compression cycle first the air is first super cooled to remove moisture and then heated to the supply temperature, which leads to the wastage of energy.
6) Since presence of high humidity and dust causes bacterial and fungal growth in the cooling ducts, hybrid desiccant cooling systems overcome this problem by removing moisture prior to the cooling coil.

1.5 APPLICATION AREAS OF DESICCANT SYSTEM

Desiccant cooling technology finds most of its application in the environment where very low humidity conditions are required, which would be otherwise expensive to be achieved by conventional systems. Moreover the areas where the latent heat load is much larger than the sensible heat load. The works where it is required to continuously supply subfreezing temperature air, the use of desiccant technology saves from the problem of defrosting the frosted cooling coils.

Some of the Application areas include:

1.5.1 Super Markets: Super markets need to maintain high ventilation rates as there is an increase in number of visitors in these markets now a days. More over there is a great need to maintain optimum humidity levels from saving the products from deterioration. These systems could also be used to maintain sub-zero temperature for freezing products.

1.5.2 Theatres: These days there is an increasing trend of watching theatres and cinemas. The competition among them has led to a race for providing better visitor facilities. Fresh and cool air is one of the basic demands of customers. Desiccant based cooling technology is the best possible way of achieving it.

1.5.3 Hospitals: The fast way of life has led to an increase in health issues in present era. There is also an improvement in medical facilities. Hospitals always need bulk of fresh air to supply so as to reduce the effect of contagious bacteria inside the premises. Moreover, the preservation of some of the high cost medicines needs very precise moisture and temperature control. This could be achieved by the use of hybrid cooling technology.

1.5.4 Hotels: The central cooling and air conditioning system of hotels need a good control over the moisture and the temperature of circulating air, which can be done by the use of desiccant cooling systems.

1.5.5 Office Buildings: With the increase in the use of glass for the construction of office buildings and the increase in number of occupants, both the latent and the sensible heat loads on the cooling systems increase. Hence the use of desiccant systems can provide the best working conditions.

1.6 LIQUID DESICCANTS

Liquid desiccants are chemicals used to dry as well as purify the air either for human or industrial purposes. They have vapour pressure lower than the water and hence the air which passes through it gets de-humidified. Liquid desiccants generally provide better drying capability than there solid counterparts. The dehumidifying properties of a desiccant are greatly dependent upon the temperature and the concentration of the desiccant solution. Increasing the desiccant solution temperature or decreasing the concentration of solution increases the vapour pressure on the solution leading to reduction in the drying capability of the desiccant solution. Hence the parameters should be controlled properly to maintain the effectiveness of the system.

1.7 TYPES OF LIQUID DESICCANTS

There are two major types of liquid desiccants depending upon the chemical properties and the functional requirements.

1) Hygroscopic salts: Salts such as lithium chloride are added to water to obtain such desiccants. LiCl is one of the mostly used hygroscopic salt because of its highly effective drying properties and very few corrosive properties.

2) Glycols: Glycols are similar to hygroscopic salts in function but require higher concentration than them. Generally about twice concentration than hygroscopic salt is required for glycol for similar effectiveness. This leads to an increase in the quantity of working fluid. Due to greater evaporation potential glycols need to be replaced at regular time intervals and are used generally for low temperature applications.

1.8 ALKALI HALIDES

Alkali halides have number of characteristics which are useful in refrigeration and air conditioning applications. Earlier they were used to depress the freezing point in refrigeration applications but now their property to absorb moisture has been used for the dehumidification (drying) purposes particularly of air, either for industrial use or for human comfort. The important point to be stressed here is that the air need not be cooled below the dew point temperature, as is required in conventional systems. This leads to the reduction in energy consumption. Moreover they do not have harmful impacts on the environment, as done by the conventionally used refrigerants (C.F.Cs).

1.9 OBJECTIVE OF PRESENT WORK

The present work has been carried out to achieve the following objectives:

1) To develop a mathematical model to predict the performance of desiccant dehumidifier for air- conditioning system.

2) To validate the mathematical model with the experimental results of the previous researchers.

3) To study the effect of various inlet parameters such as inlet air temperature, inlet desiccant temperature, inlet air mass flow rate, inlet desiccant concentration, inlet air humidity on the performance of the dehumidifier.

CHAPTER-2 LITERATURE REVIEW

Different researchers have proposed different configurations of the desiccant-vapour compression hybrid cooling system resulting in different COPs and effectiveness of the system. However active research in this field did not begin until mid-1970s [6].Various models have also been presented from time to time to understand the system well. It is very important to understand the previous work done in the field. Following is a brief of the work done by different researchers in the past. Though exhaustive, this list is not complete.

Hammad *et al.* [7] studied the effect of various parameters using uni-shell system consisting of a heat exchanger, dehumidifier and a regenerator in a shell. Shell consisted of staggered tube configuration for better transfer characteristics. Equations were developed using combined heat and mass transfer calculations at steady state. Further dimensionless forms of the governing equations were solved using the Range-Kutta method. 25% desiccant solution concentration was used for calculations and the results of the numerical study were found to be in good confirmation with the experimental results. The approximation of numerical results varied from 4.35% to 13.3% of the experimental results with the variation ratio of inlet desiccant temperature to air temperature.

Fumo *et al.* [8] constructed a mathematical model for packed bed absorber and then validated it through an experiment. Copper- constantan thermocouples were used to measure temperature values. Finite difference model was constructed by dividing the height ‘z’ of the dehumidifier into differential heights ‘dz’. The outlet conditions were first guessed and iterations were made to get the known inlet conditions in the dehumidifier. Study was conducted for both (Tri-Ethylene Glycol (TEG) and Lithium Chloride (LiCl) desiccant solutions. The non-uniform liquid distribution in the dehumidifier was also considered. The model was solved both for the dehumidifier and the regenerator, and confirmed to the experimental results.

Lychnos *et al.* [9] studied a solar powered desiccant cooling system by using $MgCl_2$ desiccant solution for green house cooling, both experimentally and by a mathematical model. Two components the absorber and the regenerator were studied by the numerical model of the system. The model was used to predict the out let parameters at various places including Mumbai, Muscat etc. They found that the desiccant cooling system lowered the average daily temperature by about 6 ℃ as compaired to conventional evaporative cooler. Studies were conducted for the un-interruptible growth of tomatoes, soya bean and lettuce throughout the year in hot climate. Mass and energy balances were conducted on the nodes connecting different components, which were further coded and solved using processing software. Pseudo-steady state model was assumed to solve the equations.

Afshin [10] investigated a number of possible liquid desiccants which could be used for the cooling system. These included LiCl, LiBr, $MgCl_2$ and $CaCl_2$. Numerical simulations were carried out and it was concluded that the effectiveness of the system changed by just 0.5 % by using different liquid desiccants. It was also found that pumping energy required by a $MgCl_2$ system was 3.5 times of that required by LiBr system. More over the study was made to consider the chances of crystallization of different salts. The study was made using different conservation equations.

Jaradat *et al.* [11] analyzed the system of dehumidifier for heat and mass transfer calculations. Plate type exchanger consisting of polycarbonate plates was used for study. The experimental study was conducted on the system constructed for drying of agricultural products. Cross flow exchangers were used in which the liquid desiccant and air exchanged heat and mass. Desiccant solution used was LiCl solution. For the air dehumidification, the system was run in adiabatic mode and for the regeneration non adiabatic run was made. The experimental results were compaired with a finite difference model assuming laminar flow conditions and uniform air and desiccant distribution. Air relative humidity change was studied with and without auxiliary heating unit and a comparison was made between them. The regenerator used was tube type with vertical tubes for the flow of desiccant and regeneration air.

Kumar *et al.* [12] conducted an experimental study of the effect of inlet parameters on the outlet conditions. Calcium chloride has been used as the desiccant solution for the experimental studies. Inlet air velocity was varied by using an electric blower to study the effect of inlet air flow rate. The dehumidifier consisted of packed bed type with plates oriented at 90° to each other. Effects of solution flow rate and solution inlet temperature was studied.

Salarian *et al.* [13] performed an experiment using packed bed dehumidifier to study the effect of ratio of inlet air to desiccant mass flow rate on the system performance. LiCl desiccant solution was used for the study by using axial extract fans at the top of the dehumidifier to extract the room air from the bottom through the packed bed. It was concluded that there is an optimum air to desiccant flow ratio for which the system performance in maximum. From analytical solution this optimum ratio can be judged.

Babakhani *et al.* [14] constructed an analytical solution model for liquid desiccant dehumidification in a packed bed dehumidifier. Assuming high desiccant flow rate the effect of various parameters was studied. Considering a differential volume simultaneous heat and mass transfer equations were solved. A FORTRAN computer program was developed by dividing the whole height 'z' of the dehumidifier into 1000 differential heights. The calculations were made starting from the bottom of the dehumidifier and moving towards the top and the study of all the inlet parameters was conducted.

Bassuoni *et al.*[15] worked on the development of analytical model for the study of parametric effect on the performance of desiccant cooling system. The results found are within 6% deviation. The liquid desiccant cross flow air dehumidifier was modeled with the conservation equations. The model describes the coupled heat and mass transfer calculations which take place simultaneously inside the dehumidifier.$CaCl_2$ is used for the modeling purpose and the empirical correlation developed by Moon et al. is used to get the dehumidification effectiveness of the mass exchanger.

Kumar *et al.* [16] presented a simplified model of desiccant dehumidifier for a structured packed bed dehumidifier. A honeycomb celdek packing has been used in the study with liquid desiccant flowing downward and air flowing upward i.e. a cross flow mass exchanger. Wetness fraction as described by Jain et al. has been used in this model. By developing the differential equations these non-linear coupled heat and mass equations have been solved. Further numerical solution has been conducted for the control volume by dividing into differential control volumes.

Sabek *et al.* [17] used an open cycle absorption system to formulate the analytical mathematical model. The study was mainly concerned with the study of performance of desiccant cooling system under Tunisian climatic conditions. The consumption of electrical energy in the working of the system was also studied along with the parametric effect on the performance.

Gao *et al.* [18] undertook an experimental investigation of liquid desiccant dehumidification in combination with the evaporative cooling equipment. The heat and mass transfer effectiveness of the combined system was indicated by the SHR (sensible heat ratio) , the moisture and the temperature reductions. It was proved in the results that the effectiveness of the dehumidification in the 1st half of the process affected the overall efficiency of the system to a great extent. Lithium bromide desiccant solution was used in this experimental study.

CHAPTER-3 MATHEMATICAL MODELLING

3.1 ASSUMPTIONS

The following assumptions have been made for the development of mathematical model for desiccant dehumidifier of air conditioning system:

1. The system is assumed to work at adiabatic conditions i.e. no heat loss to surrounding.

2. The given system works at steady state i.e. sufficient time has elapsed since the system is working.

3. In interface area i.e. area between both the phases, the properties of both air and desiccant are calculated at same conditions.

4. No desiccant gets carried away by the process air or regeneration air i.e. the quantity of desiccant in the desiccant solution remains constant.

5. Heat transfer resistance is negligible in the desiccant solution as compared to that in the air.

6. Transfer surface areas in both the interfaces i.e. liquid desiccant and the air are equal.

3.2 BASIC GOVERNING EQUATIONS & METHODOLOGY

For the development of mathematical model the basic conservation and governing equations are written for the dehumidifier system.

3.2.1 ENERGY BALANCE ACROSS DEHUMIDIFIER

3.2.1.1 <u>Enthalpy of moist air</u>

The enthalpy of dry air in kJ/kg at any temperature between 0℃ and 60 ℃ is given approximately by:

$$\boldsymbol{h = 1.007 \times t - 0.026} \tag{3.1}$$

Where t is in ℃.

The enthalpy of water vapour in kJ/kg at any temperature is given by:

$$\boldsymbol{h_w = 2501 + 1.88 \times t} \tag{3.2}$$

Therefore, total enthalpy of moist air ,in kJ/kg is:

$$\boldsymbol{h_a = (1.007 \times t - 0.026) + \omega_a \times (2501 + 1.88 \times t)} \tag{3.3}$$

3.2.1.2 <u>Enthalpy of desiccant solution</u>

Enthalpy of desiccant solution is given by:

$$\boldsymbol{h_s = C_{p,s} \times t_s}$$

Conde [19] provided an empirical relation for the calculation of specific heat for LiCl desiccant solution based on the solute concentration and the temperature.

$$\boldsymbol{C_{p,s}(t, X) = C_{p,H2O}(t) \times [1 - f1(X) + f2(t)]} \tag{3.5}$$

Where, $$\boldsymbol{C_{p,H2O}(\emptyset) = A + B\emptyset^{0.02} + C\emptyset^{0.04} + D\emptyset^{0.06} + E\emptyset^{1.8} + F\emptyset^{8}} \tag{3.6}$$

$\emptyset = (t/288) - 1$, where t is in K.

$f1(X) = (A \times X) + (B \times X^2) + (C \times X^3)$

$f2(t) = F \times \emptyset^{0.02} + G\emptyset^{0.04} + H\emptyset^{0.06}$

The values of the constants used in the above equations are provided in the table 3.1.

Table 3.1 Parameters to be used in empirical equation of specific heat

	A	B	C	D	E	F	G	H
$LiCl - H_2O$	1.439	-1.243	-0.120	0.128	0.629	58.522	-105.63	47.794
$CaCl_2 - H_2O$	1.637	-1.690	1.0512	0.0	0.0	58.522	-105.63	47.794

Overall energy balance equation for the dehumidifier can be written as:

$$\boldsymbol{m_a \times (h_{a1} - h_{a2}) = m_s \times (h_{s1} - h_{s2}) + m_a \times (\omega_1 - \omega_2) \times h_{fg}} \quad \textbf{(3.7)}$$

i.e.

$$[\text{Total change in enthalpy of the process air}] = [\text{sensible enthalpy change of liquid desiccant}] + [\text{Latent heat change of the process air}]$$

3.2.2 MASS BALANCE OF DESICCANT ACROSS DEHUMIDIFIER

Since it is assumed that no desiccant gets carried away by the process air, therefore the quantity of desiccant remains same at entry and exit of the dehumidifier. Hence,

$$\boldsymbol{m_{s1} \times X_1 = m_{s2} \times X_2} \quad \textbf{(3.8)}$$

3.2.3 MASS BALANCE OF AIR-WATER VAPOUR

Amount by which desiccant solution is diluted is given by

$$\boldsymbol{\dot{m}_{cond} = \dot{m}_a \times (\omega_1 - \omega_2)} \quad \textbf{(3.9)}$$

Also, $m_{s2} = m_{s1} + m_{cond}$

Therefore, $m_{s1} \times X_1 = [m_{s1} + m_a \times (\omega_1 - \omega_2)] \times X_2$

Hence, $\boldsymbol{X_2} = [\mathbf{1}/\{\mathbf{1} + (\boldsymbol{m_a}/\boldsymbol{m_{s1}}) \times \Delta\boldsymbol{\omega}\}] \times \boldsymbol{X_1}$ **(3.10)**

3.2.4 VAPOUR PRESSURE OF DESICCANT SOLUTION

The air humidity ratio at the liquid desiccant interface is determined from the vapour pressure. Vapour pressure can be determined from the curve fit given by [8]

$$\boldsymbol{P_v} = (\boldsymbol{a_0} + \boldsymbol{a_1} \times \boldsymbol{t} + \boldsymbol{a_2} \times \boldsymbol{t^2}) + (\boldsymbol{b_0} + \boldsymbol{b_1} \times \boldsymbol{t} + \boldsymbol{b_2} \times \boldsymbol{t^2})\boldsymbol{X}$$

$$+ (\boldsymbol{c_0} + \boldsymbol{c_1} \times \boldsymbol{t} + \boldsymbol{c_2} \times \boldsymbol{t^2})\boldsymbol{X^2} \qquad \textbf{(3.11)}$$

Where, t is in °C .

Various coefficients used in Eq. (3.11) can be used as provided in Table 3.2

Table 3.2 Coefficients to be used for vapour pressure calculations

a_0	4.58208	b_1	-18.3816	c_0	21.312
a_1	-0.159174	b_2	0.5661	c_1	0.666
a_2	0.0072594	b_3	-0.019314	c_2	-0.01332

3.2.5 DIFFUSION COEFFICIENT FOR DEICCANT SOLUTION

The value of diffusion coefficient for the desiccant solution changes with the change in the desiccant concentration. The values of diffusion coefficient for various desiccant solution concentrations at a temperature of 30°C have been taken from the published work of Afshin [10]. These values are given in table 3.3.

Table 3.3 Diffusion coefficient for LiCl solution at various concentrations

% desiccant concentration	D_{AB} (m^2/s)
29.9 %	0.924 E-9
30.9%	0.8765 E-9
31.6%	0.8446 E-9
32.4%	0.7968 E-9
33.5%	0.7570 E-9

3.2.6 DYNAMIC VISCOSITY FOR DESICCANT

Conde [19] provided the empirical correlation for the dynamic viscosity of the desiccant solution as given below:

$$\boldsymbol{\mu_s = \mu_{H_2O}}(\emptyset) \times \boldsymbol{exp}\left(\boldsymbol{\mu_1 \times \gamma^{3.6} + \mu_2 \times \gamma + \mu_3 \times \frac{\gamma}{\emptyset} + \mu_4 \times \gamma^2}\right) \quad \textbf{(3.12)}$$

where,

$$\gamma = X/(1-X)^{1/0.6}$$

$$\boldsymbol{\mu_{H_2O}}(\emptyset) = \boldsymbol{\mu_{H_2O,0}} \times \left(+\boldsymbol{B}\emptyset^{\mathbf{0.02}} + \boldsymbol{C}\emptyset^{\mathbf{0.04}} + \boldsymbol{D}\emptyset^{\mathbf{0.08}} + \boldsymbol{E}\emptyset^{\mathbf{2.85}} + \boldsymbol{F}\emptyset^{\mathbf{8}}\right) \quad \textbf{(3.13)}$$

Where, $\mu_{H_2O,0}$ is the dynamic viscosity of water at 0°C and its value is 0.001787 $\mathrm{N.s/m^2}$. and

$\emptyset = \frac{T}{228} - 1$, where T is temperature in K.

The values of various constants used to determine $\boldsymbol{\mu_{H_2O}}(\emptyset)$ in Eq.(3.12) are given in table 3.4.

Table 3.4 Values of coefficients to determine $\boldsymbol{\mu_{H_2O}}(\emptyset)$

A	B	C	D	E	F
1.026	12481.702	-19510.923	7065.286	-395.561	143922.996

3.2.7 DENSITY

3.2.7.1 DENSITY OF DESICCANT SOLUTION

Conde [19] provided following empirical correlation for the calculation of the density of the desiccant solution.

$$\boldsymbol{\rho_s = \rho_{H_2O}\,(t) \times \sum_{i=0}^{3}[\rho_i \times \{X - (1 - X)\}^i]} \quad \textbf{(3.14)}$$

Where,

$$\boldsymbol{\rho_{H_2O}(t) = \rho_{c,H_2O} \times \left(1 + B_0\delta^{1/3} + B_1\delta^{2/3} + B_2\delta^{5/3} + B_3\delta^{16/3} + B_4\delta^{43/3} + B_5\delta^{110/3}\right)} \quad \textbf{(3.15)}$$

Here, ρ_{c,H_2O} = value of density of water at critical temperature

and the values of coefficients B_0, B_1, B_2, B_3, B_4, B_5 are given in table 3.5

Table 3.5 Values of B_i to be used for water density calculations

B_0	1.994
B_1	1.099
B_2	-0.509
B_3	-1.762
B_4	-45.9
B_5	-723692

$\delta = 1 - \emptyset$ and $\emptyset = t/t_{critical}$

't' in kelvin.

And the values of various constants used in Eq.(3.14) are given in table 3.6.

Table 3.6 Parameters for lithium chloride and calcium chloride for density

	LiCl – H2O	$CaCl_2$ – H2O
ρ_0	1	1
ρ_1	0.541	0.836
ρ_2	-0.304	-0.436
ρ_3	0.101	0.105

3.2.7.2 MOIST AIR

Density of moist air is calculated by using the correlation provided by Picard [21]:

$$\boldsymbol{\rho_{air} = (\rho \times M_a)/(Z \times R \times T) \times [1 - \omega_1 \times (1 - M_v/M_a)]} \quad \textbf{(3.16)}$$

Where Z = coefficient of compressibility, R = gas constant, T = temperature in K.

3.2.8 SCHMIDT NUMBER

Schmidt number (Sc) plays an important role in mass transfer problems as is played by Prandtl number in heat transfer calculations. In gases both the mass and momentum are transported by similar means, hence the value of Sc is close to unity. But in liquids since the molecules are closely packed, therefore the transport of momentum is much greater than that of mass. Therefore, value of Sc came out to be much greater than unity.

Hence, the Schmidt number which is the ratio of momentum diffusivity to mass diffusivity can be found as:

$$\boldsymbol{Sc = \frac{Ability\ of\ the\ fluid\ to\ transfer\ momentum\ by\ molecular\ means}{Ability\ of\ the\ fluid\ to\ transfer\ mass\ by\ molecular\ means}}$$

$$\boldsymbol{= \vartheta/D_{AB}} \quad \textbf{(3.17)}$$

3.2.9 SHERWOOD NUMBER

The physical properties of the fluid, the flow geometry and the average velocity affect the mass transfer coefficient to a great extent. This dependence can be easily depicted in a dimensionless form. The dimensionless form of mass transfer coefficient can be formed by the **Sherwood number (Sh)**. Davies [20] provided with an empirical relation to find out the Sherwood number as follows:

$$\boldsymbol{Sh = \left[2.25 \times 10^{-4} \times (1 - X)^{-0.75} \times (1/L)^{0.1} \times Sc^{0.333} \times Re^{1.0}\right]} \qquad \textbf{(3.18)}$$

Where, Re = Reynold's number for fluid flow in a packed bed = $\frac{(\boldsymbol{\rho} \times \boldsymbol{v} \times \boldsymbol{\beta})}{\boldsymbol{\mu}(\mathbf{1}-\boldsymbol{\varepsilon})}$ **(3.19)**

β = spherical equivalent particle diameter for packing.

$$\beta = 6 \times \frac{Volume\ of\ particle}{surface\ area}$$

v = superficial velocity of the desiccant solution in the dehumidifier (m/s).

ε = packed bed voidage $= \frac{empty\ volume\ in\ the\ packed\ bed}{total\ volume\ (\,i.e.\ empty\ volume + solid\ volume)}$

Industrial literature on tower packings [22] provided the data about the packed bed voidage for plastic packings as given in table 3.7.

Table 3.7 Packing voidage values for plastic packing

SIZE	BULK DENSITY (kg/m^3)	SURFACE AREA (m^2/m^3)	VOID FRACTION (%)
15-7	80	313	91
25-7	85	214	91
38-1	51	150	94
50-0	50	110	94
50-3	52	95	94
50-6	46	90	94

3.2.10 CHARACTERISTIC LENGTH

The procedure to determine the characteristic length is explained below:

3.2.10.1 WETTED SURFACE AREA

Since the surface tension of desiccant solution is high, therefore, the wetted area is actually less than the total surface area. An empirical correlation for ratio of specific wetted surface area to total specific area is provided by Goswami [23] .

$$\frac{a_w}{a_t} = \mathbf{1} - \boldsymbol{exp}\left[-\mathbf{1.45} \times (\boldsymbol{\sigma_c}/\boldsymbol{\sigma_s})^{\mathbf{0.75}} \times (\boldsymbol{L}/\boldsymbol{a_t} \times \boldsymbol{\mu_s})^{\mathbf{0.1}} \times (\boldsymbol{L}^{\mathbf{2}} \times \boldsymbol{a_t}/\boldsymbol{\rho_s}^{\mathbf{2}} \times \boldsymbol{g})^{-\mathbf{0.05}} \times (\boldsymbol{L}^{\mathbf{2}}/(\boldsymbol{\rho_s} \times \boldsymbol{\sigma_s} \times \boldsymbol{a_t})^{\mathbf{0.2}})\right] \quad \textbf{(3.20)}$$

Where, σ_c = critical surface tension of the solid surface

& σ_s = surface tension of the desiccant at the given concentration and temperature .

3.2.10.2 SURFACE TENSION

Surface tension of desiccant solution is different from that of water. Empirical correlation used for the calculation of surface tension of the desiccant is provided by Conde [19]:

$$\boldsymbol{\sigma_s}(\boldsymbol{X}, \emptyset) = \boldsymbol{\sigma_{water}}(\emptyset) \times (\mathbf{1} + \boldsymbol{\sigma_1} \times \boldsymbol{X} + \boldsymbol{\sigma_2} \times \boldsymbol{X} \times \emptyset + \boldsymbol{\sigma_3} \times \boldsymbol{X} \times \emptyset^{\mathbf{2}} + \boldsymbol{\sigma_4} \times \boldsymbol{X}^{\mathbf{2}} + \boldsymbol{\sigma_5} \times \boldsymbol{X}^{\mathbf{3}}) \quad \textbf{(3.21)}$$

Where , Θ = t/t critical , $\emptyset = t/t_{critical}$

$$\boldsymbol{\sigma_{water}}(\emptyset) = \boldsymbol{\sigma_0} \times [\mathbf{1} - \boldsymbol{C}(\mathbf{1} - \emptyset)] \times (\mathbf{1} - \emptyset)^{\boldsymbol{\eta}} \quad \textbf{(3.22)}$$

where, C = 0.625 , η = 1.256

The values of the coefficients for surface tension Eq.(3.21) are given in table 3.8.

Table 3.8 Values of coefficients for surface tension calculation

σ_0	235.8
σ_1	-2.76
σ_2	-12.01
σ_3	14.75
σ_4	2.443
σ_5	-3.147

Then by using the value of a_w , characteristic length can be found as:

$$L_C = a_w/d_p \qquad \textbf{(3.23)}$$

Where, d_p= diameter of packing.

By the definition of Sherwood number,

$$Sh = \frac{h_{mass} \times L_C}{D_{AB}} \qquad \textbf{(3.24)}$$

Therefore, mass transfer coefficient, h_{mass} can be calculated as given below:

$$h_{mass} = \frac{Sh \times D_{AB}}{L_C} \qquad \textbf{(3.25)}$$

3.2.11 LEWIS NUMBER

Now using the analogy between heat and mass transfer , and the dimensionless number , **Lewis number (Le)** we can find the value of heat transfer coefficient from the value of mass transfer coefficient as follows :-

***Le* = Thermal diffusivity / Mass diffusivity**

$$= h_s/h_{mass} \times C_{p,s} \qquad \textbf{(3.26)}$$

3.2.12 TRANSFER COEFFICIENT

Therefore, heat transfer coefficient in desiccant solution,

$$\boldsymbol{h_s = (Le \times h_{mass})/C_{p,s}} \tag{3.27}$$

Literature on liquid desiccant provides that the best results are obtained at Lewis number **Le = 1.1**. [24]

Using the same approach, mass transfer coefficient ($h_{mass,air}$) and heat transfer coefficient (ha) for air can be found out.

We know that overall heat transfer coefficient can be found as:

$$\boldsymbol{\frac{1}{U} = \frac{1}{h_s} + \frac{1}{h_a}} \tag{3.28}$$

It has been assumed that the heat transfer resistance in the desiccant is very low. Hence, $\frac{1}{h_s}$ becomes negligible in the above equation and can be neglected.

Similarly, over all mass transfer coefficient (U_m) can be calculated from individual mass transfer coefficients.

3.2.13 NUMBER OF TRANSFER UNITS (N.T.U)

Babakhani [24] provides the formula to find N.T.U. as:

$$\boldsymbol{NTU_{mass} = (U_m \times a_w \times L_p \times A_t)/m_a} \tag{3.29}$$

Where, U_m is the overall mass transfer coefficient.

A_t= total area of packing , a_w= wetted area of the packing , m_a = mass flow rate of air ,Lp =total length of packing used for mass transfer.

Similarly,

$$\boldsymbol{NTU_{heat} = U \times A_s/C_{min}} \tag{3.30}$$

3.2.14 EFFECTIVENESS

Barlow [6] gives relation for effectiveness to be used for heat and mass exchangers.

$$\boldsymbol{\epsilon} = \frac{\mathbf{1} - \boldsymbol{exp}(-\boldsymbol{NTU}(\mathbf{1} - \boldsymbol{C}))}{\mathbf{1} - \boldsymbol{C} \times \boldsymbol{exp}(-\boldsymbol{NTU}(\mathbf{1} - \boldsymbol{C}))} \qquad \textbf{(3.31)}$$

where,

$C = C_{min}/C_{max}$

ϵ = effectiveness of the dehumidifier.

[6] gives the expressions for calculation of C_{min} and C_{max} . For mass transfer calculations these are the mass flow rates of air and desiccant respectively i.e. mass capacities. For heat transfer problems these are the heat capacities for air and desiccant solution respectively.

Also,

$$\boldsymbol{\epsilon} = \frac{(\boldsymbol{\omega_1} - \boldsymbol{\omega_2})}{(\boldsymbol{\omega_1} - \boldsymbol{\omega_{equ}})} \qquad \textbf{(3.32)}$$

Where, ω_{equ} is the humidity ratio of the air in equilibrium with the desiccant solution.

3.2.15 EQUILIBRIUM HUMIDITY RATIO

Equilibrium humidity ratio is the humidity ratio at the interface between the desiccant and the air.

ω_{equ} can be found from the value of partial vapour pressure of desiccant solution at interface by using:

$$\boldsymbol{\omega_{equ}} = \frac{\mathbf{0.662} \times \boldsymbol{p_v}}{\boldsymbol{p_a} - \boldsymbol{p_v}} \qquad (\mathbf{3.33})$$

Where p_a = atmospheric pressure.

From this ω_2 i.e. the outlet humidity of air can be found.

Thereafter, the mass rate of condensate can be calculated by, $\boldsymbol{m_{cond}} = \boldsymbol{m_a} \times (\boldsymbol{\omega_1} - \boldsymbol{\omega_2})$.

3.2.16 DIFFERENTIAL DILUTION ENTHALPY

When water vapour is absorbed by the salt solution, heat is released i.e. it is an exothermic process. This released heat is more than the energy released on condensation of pure water. This difference is called differential dilution enthalpy. Conde [19] provided an empirical relation for this enthalpy.

$$\Delta \boldsymbol{h} = \Delta \boldsymbol{h_0}(\mathbf{1} + (\boldsymbol{\tau}/\boldsymbol{A_1})^{\boldsymbol{A_2}})^{\boldsymbol{A_3}} \qquad \textbf{(3.34)}$$

Where, $\tau = salt\ mass\ fraction = X/(A_4 - X)$

$$\Delta \boldsymbol{h_0} = \boldsymbol{A_5} + \boldsymbol{A_6}.\boldsymbol{\theta}$$

The value of various constants for LiCl used is provided in table 3.9.

Table 3.9 Values of coefficients for differential dilution enthalpy

A_1	A_2	A_3	$(A_4$	$\boldsymbol{A_5}$	$\boldsymbol{A_6}$
0.9	-1.96	-2.3	0.6	169.1	457.8

CHAPTER-4 RESULT AND DISCUSSION

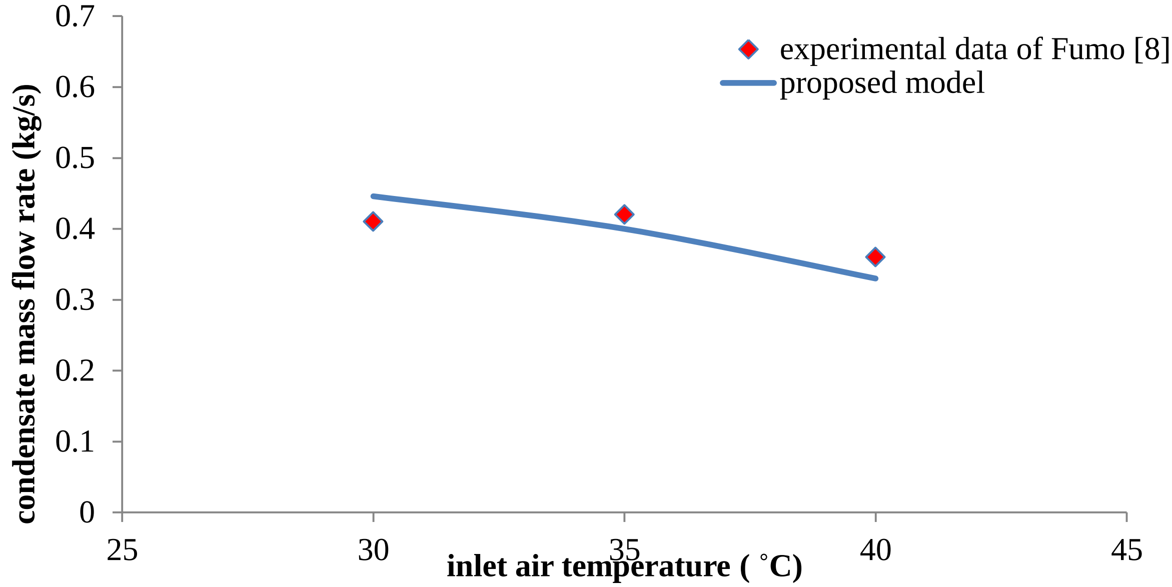
4.1 Validation of mathematical model

In order to validate the mathematical model for desiccant dehumidification the condensate mass flow rates at different inlet air temperatures and at different inlet air humidity measured by Fumo [8] are compared with those predicted by the model. The details of the experimental parameters used by Fumo [8] are given in table 4.1.

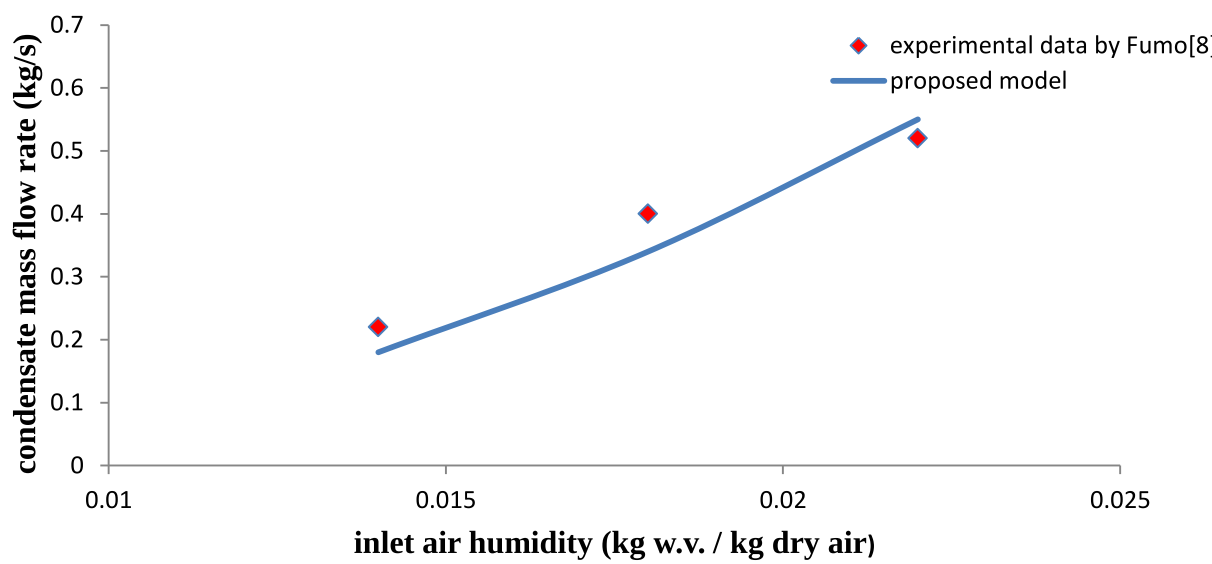
Table 4.1 Experimental parameters used by Fumo [8]

PACKING TYPE	RAUSCHERT HIFLOW RINGS
MATERIAL OF PACKING	POLYPROPYLENE(PP)
SIZE OF PACKING	2.54 cm
SPECIFIC SURFACE AREA	210 m^2/m^3
INTERNAL DIA OF DEHUMIDIFIER	0.24 m
HEIGHT OF PACKING	0.60 m
INLET AIR TEMPERATURE	30
INLET DESICCANT TEMPERATURE	30
SUPERFICIAL AIR FLOW RATE	1.192 kg/ /s
SUPERFICIAL DESICCANT FLOW RATE	6.26 kg/ /s
INLET AIR HUMIDITY RATIO	0.0179
INLET DESICCANT CONCENTRATION	33.8
DESICCANT SOLUTION	LITHIUM CHLORIDE

Now using the same experimental parameters as given in table 4.1, the condensate mass flow rates at different inlet air temperatures and different air humidity were obtained from the previous mathematical model and they are compared with the experimental results of Fumo [8] as shown in fig. 4.1.



(a)



(b)

Fig.4.1 Validation of mathematical model with experimental results of Fumo[8]

Figure 4.1 shows that the results of the mathematical model predict the experimental data of Fumo [8] with mean deviation of 10.62 percent. Therefore, it can be concluded that the results obtained from mathematical model are in good agreement with the experimental results of Fumo [8].

Hence the above desiccant dehumidifier model can be used to study the effect of other inlet parameters on the performance of the dehumidifier.

4.2 Parametric Study of dehumidifier performance

The same design parameters as given in table 4.1 have been used in the present model to study the effect of various inlet parameters on the properties of dehumidifier.

4.2.1 Effect of inlet desiccant temperature on humidity ratio of process air

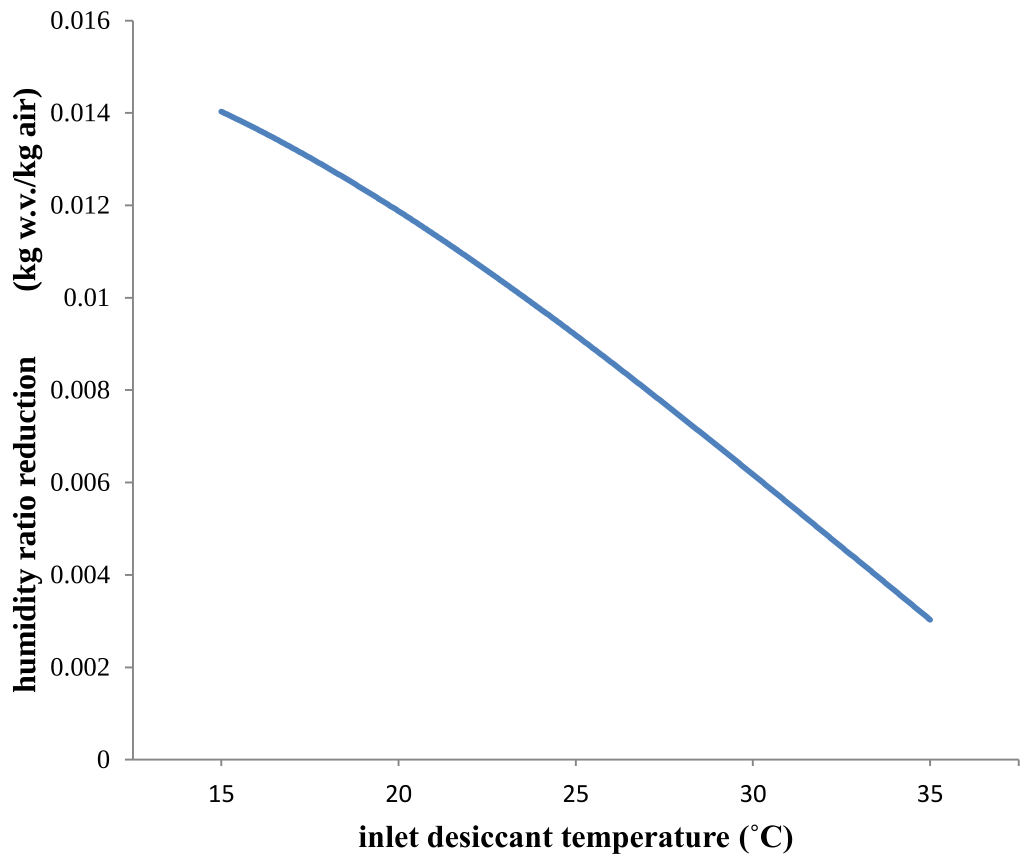


Fig. 4.2 Variation of humidity change with desiccant inlet temperature

From fig. 4.2 it is clear that with increase in the desiccant inlet temperature, there is decrease in the humidity ratio reduction of process air. We know that ability of desiccant solution to absorb moisture, greatly depends on its temperature. With increase in desiccant inlet temperature, its moisture absorbing property decreases and hence the reduction in humidity ratio of process air decreases and thus decreases the effectiveness of the dehumidifier. Hence in order to get maximum effectiveness from the dehumidifier the desiccant temperature should be as low as possible.

For the range of investigated operating parameters, as the inlet desiccant temperature is decreased from 35℃ to 15℃, the reduction in humidity ratio of process air is increased by 370 %.

4.2.2 Effect of inlet air mass flow rate on humidity ratio of process air

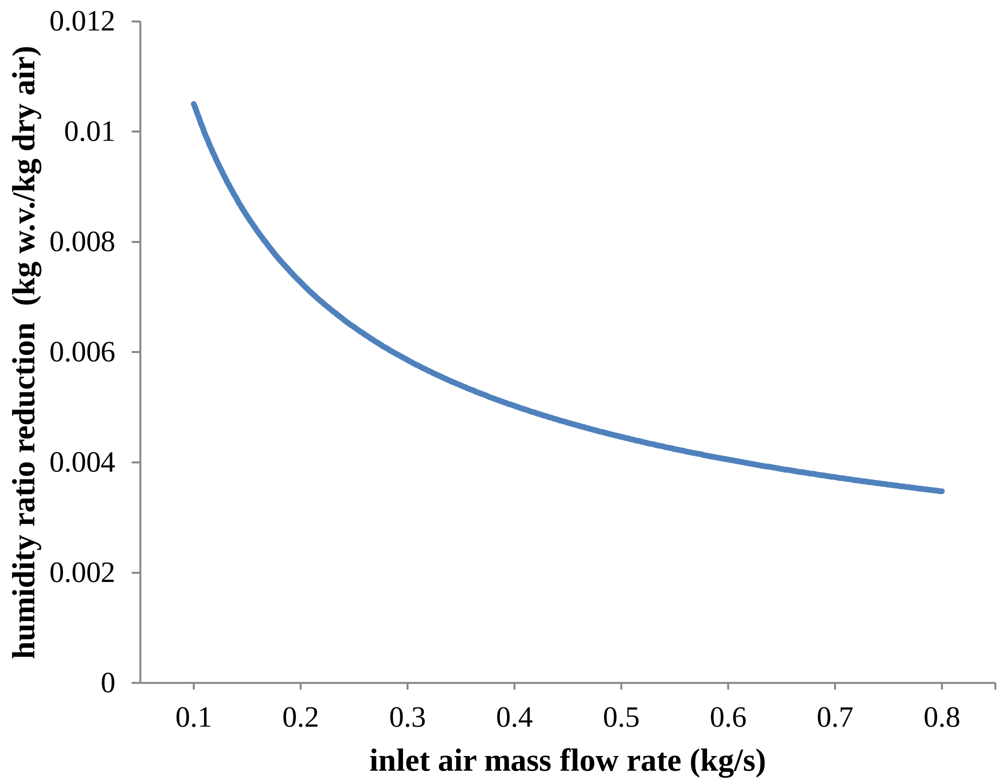


Fig.4.3 Variation of humidity ratio reduction of air with inlet air mass flow rate

From figure 4.3 it is clear that on increasing the inlet air mass flow rate there is decrease in humidity change. This can be attributed to the fact that with increase in the inlet air mass flow rate the incoming air gets less time for moisture diffusion to the desiccant solution. Hence, the dehumidification could not be that effective, which reduces the change in humidity ratio and ultimately sends out more humid air. Hence the inlet air mass flow rate should be a compromise between quantity and quality of air requirement. Increase in one leads to decrease in other.

For the range of investigated operating parameters, as the inlet air flow rate decreases from 0.8 kg/s to 0.1 kg/s, the reduction in humidity ratio of process air increases by 224.3%.

4.2.3 Effect of inlet air mass flow rate on condensate mass flow rate

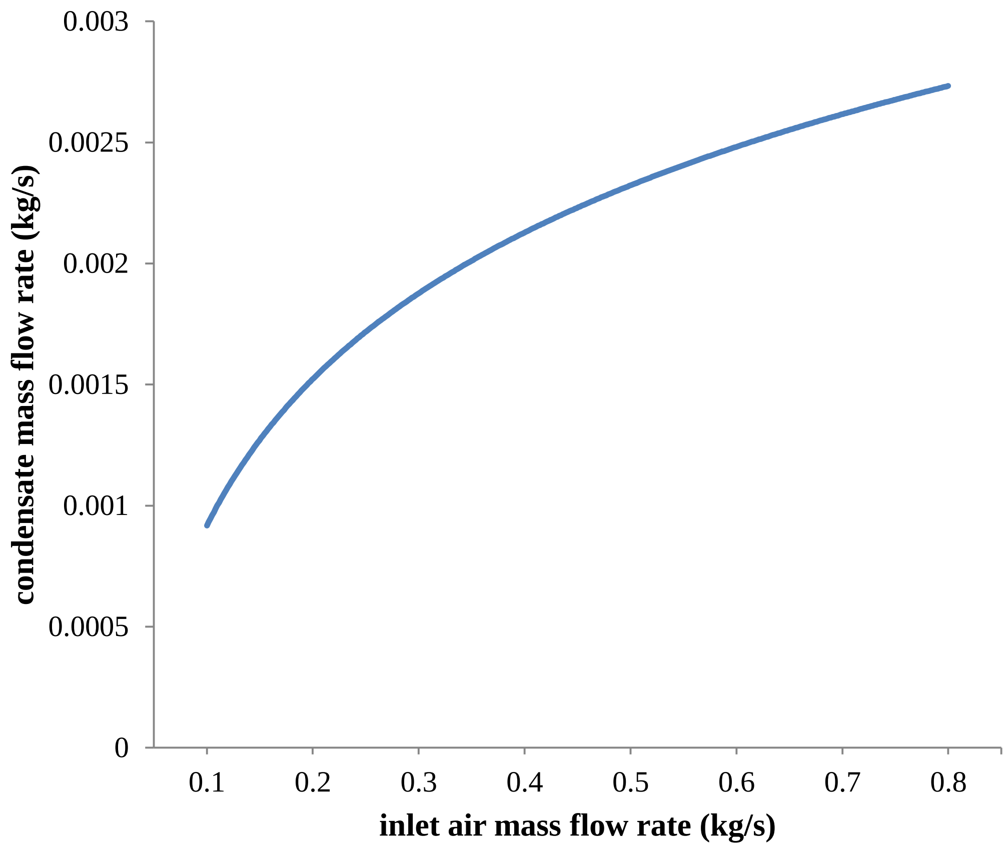


Fig. 4.4 Effect of inlet air mass flow rate on condensate mass flow rate

Mass of condensate increases with the increase in inlet air flow rate as can be seen in figure 4.4 .The rate of increase is steep at starting and decreases with further increase in inlet air mass flow rate. Though with increase in air mass flow rate there is increase in condensate mass flow rate but at higher air flow rates the rate of increase decreases. This can be explained by the fact that with increase in air flow rate there is less time for contact between air particle and the desiccant. Due to which there is less diffusion of moisture from air to the desiccant solution.

For the range of investigated operating parameters, as the inlet air mass flow rate is increased from 0.1 kg/s to 0.8 kg/s, the condensate mass flow rate increases by 184%.

4.2.4 Effect of inlet desiccant concentration on change in air humidity

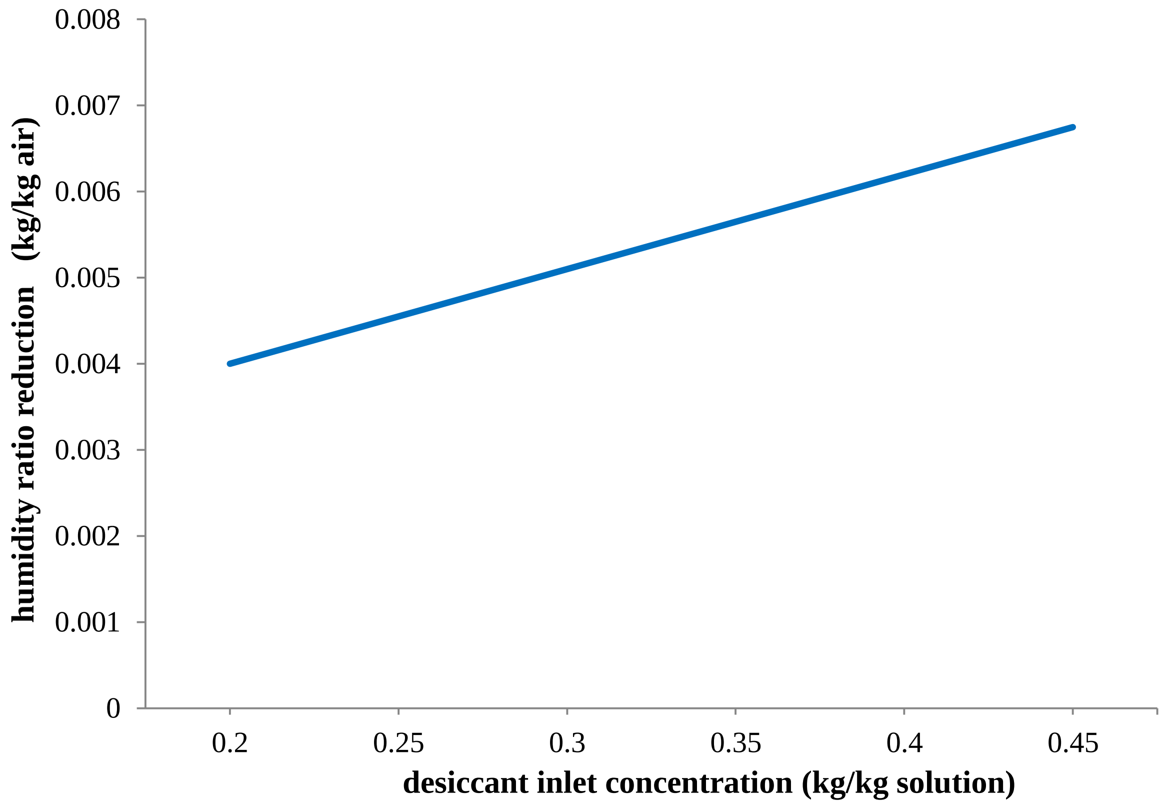


Fig. 4.5 Effect of inlet desiccant concentration on humidity ratio reduction

From fig. 4.5 it can be noted that with increase in desiccant concentration in the solution the humidity ratio change increases. With the increase in the concentration of desiccant in solution the amount of desiccant in solution increases which decreases the vapour pressure on the solution. This reduction in vapour pressure acts as the driving force for the absorption of moisture. Hence the desiccant solution absorbs more moisture in concentrated form. Therefore, the change in humidity ratio increases with increase in desiccant concentration.

For the range of investigated operating parameters, as the inlet desiccant concentration increases from 0.2 to 0.45, the reduction in humidity ratio of process air increases by 70%.

4.2.5 Effect of inlet air flow on sensible heat ratio

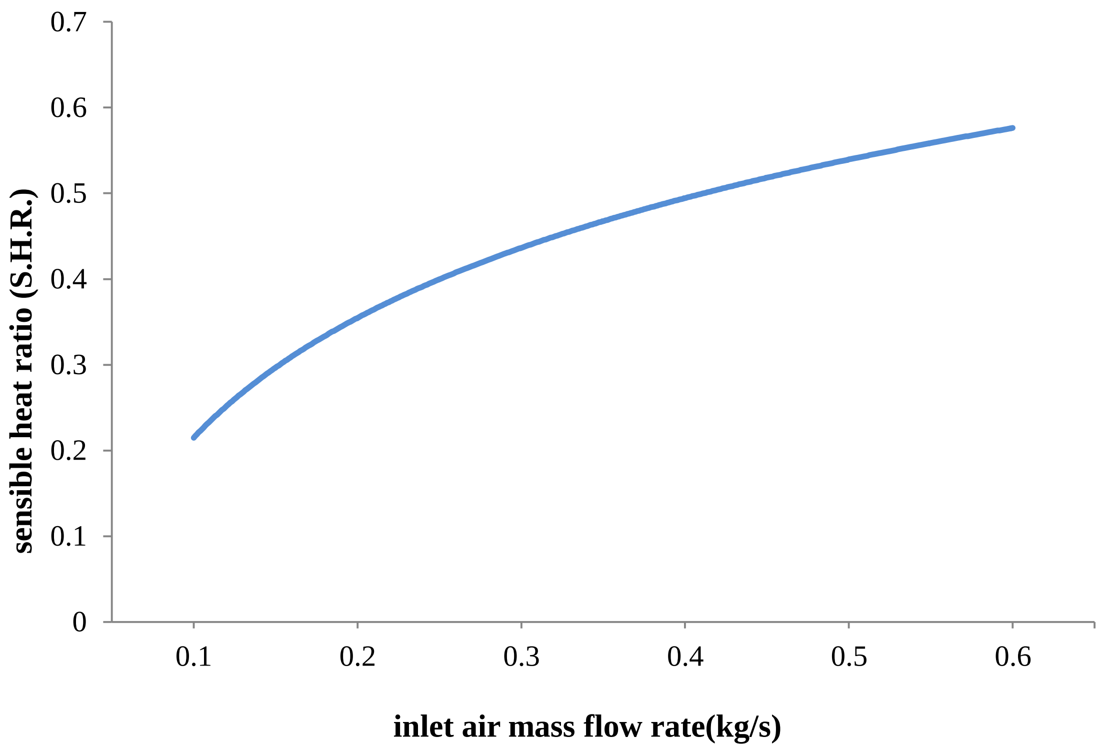


Fig. 4.6: Effect of inlet air mass flow rate on the S.H.R.

Sensible heat ratio is the ratio of sensible heat load to the total heat load on the dehumidifier. From fig. 4.6 it can be clearly noted that with the increase in inlet air mass flow rate the sensible heat ratio increases. That means the sensible heat load on the system increases as compared to the latent heat load. Such a behavior can be explained from the fact that as the inlet air mass flow rate increases, the moisture removed from the air also increases. Hence the heat of adsorption added by the desiccant to the air also increases. Therefore, the temperature of air increases by small amount equal to the heat of adsorption.

For the range of investigated operating parameters, as the inlet air mass flow rate increases from 0.1 kg/s to 0.6 kg/s , the sensible heat ratio increases by 176%.

4.2.6 Effect of air humidity on desiccant outlet temperature

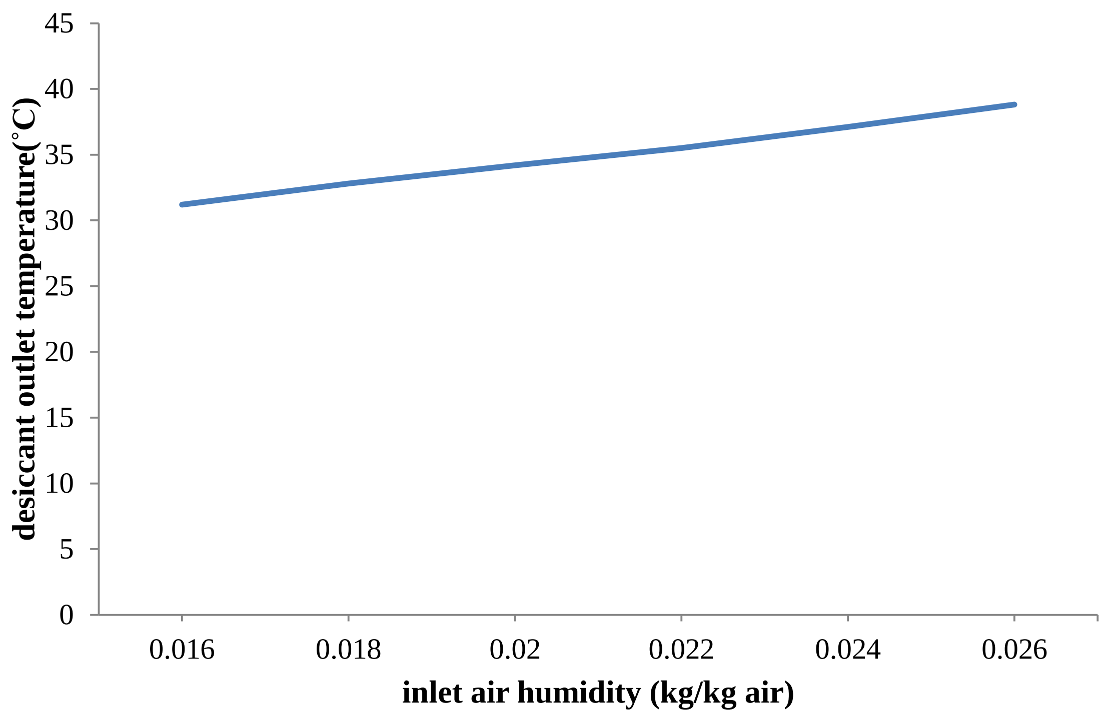


Fig. 4.7 : Effect of air humidity on desiccant outlet temperature

From fig. 4.7 it can be clearly seen that as the inlet air humidity increases, the temperature of desiccant at outlet also increases. More is the humidity of the inlet air, greater will be the adsorption by the desiccant and more will be the heat of absorption given to the air. Hotter will be the air, so more will be the heat transfer to the desiccant solution. Hence more will be the desiccant outlet temperature. This increasing desiccant temperature will adversely affect the dehumidification capacity of the desiccant solution.

For the range of investigating operating parameters, as the inlet air humidity is increased from 0.016 to 0.026, the desiccant outlet temperature increases by 26%.

CHAPTER-5 CONCLUSION & FUTURE SCOPE

5.1 Conclusion

The present work has been done with Lithium Chloride as desiccant in a packed bed desiccant dehumidifier. Following conclusions could be drawn from the parametric study of various input parameters and their effect on the performance of dehumidifier:

1) The reduction in humidity ratio of process air increases almost linearly with the reduction in inlet desiccant temperature. For the range of investigated operating parameters, as the inlet desiccant temperature is decreased from 35℃ to 15℃, the reduction in humidity ratio of process air is increased by 370 %.

2) For the range of investigated operating parameters, as the inlet air flow rate decreases from 0.8 kg/s to 0.1 kg/s, the reduction in humidity ratio of process air increases by 224.3%. More is the quantity of process air flowing through the dehumidifier, lower will be the quality. Hence a balance should be kept between the amount of air and the humidity ratio.

3) For the range of investigated operating parameters, as the inlet air mass flow rate is increased from 0.1 kg/s to 0.8 kg/s, the condensate mass flow rate increases by 184%.

4) For the range of investigated operating parameters, as the inlet desiccant concentration increases from 0.2 to 0.45, the reduction in humidity ratio of process air increases by 70%.

5) For the range of investigated operating parameters, as the inlet air mass flow rate increases from 0.1 kg/s to 0.6 kg/s , the sensible heat ratio increases by 176%.

6) For the range of investigating operating parameters, as the inlet air humidity is increased from 0.016 to 0.026, the desiccant outlet temperature increases by 26%.

5.2 Future scope

Desiccant cooling technology can be seen as the present day solution to save from the harm caused by conventional refrigerants. The work done in this study can be further extended to combine all the components of a hybrid desiccant – vapour compression air conditioning system. Such a model could help us understand the overall performance of the system. The impact of pressure drop and friction in the dehumidifier packing could also be included to get more realistic results. Moreover the study can be extended to compare different desiccant solutions or their mixtures so as to select the best desiccant according to the given working conditions.

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