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Chhatrapati Shahu Institute of Business Education & Research (CSIBER)

(An Autonomous Institute)

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# South Asian Journal of Management Research (SAJMR)

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**South Asian Journal of Management Research**
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**Design and Study of Integrated Desiccant Dehumidification and Vapour Compression for Energy-Efficient Air Conditioning System**

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**Abstract**

Energy efficiency and energy conservation are the two significant aspects considered while designing an air conditioning system. Conventional vapour compression air conditioning systems (VCAs) are considered energy-intensive devices, which motivates to stimulate the research towards the energy-efficient air conditioning system, especially for hot and humid climatic conditions. The temperature-independent humidity control through liquid desiccant material provides a substantial impact on minimizing energy shortages and reduces environmental problems without compromising human thermal comfort. In this perspective, efforts are devoted in the present study to integrate the two technologies of desiccant dehumidification and vapour compression air conditioning system. A mathematical model is developed for the dehumidifier/ regenerator module in 'Aspen Plus® V8.8' software with governing equations of energy, mass and species balance. The model predictions are compared with the available published experimental datasets, and noticed a good agreement exists between them with maximum errors of ± 4.9% and ±8.9% for condensation and evaporation rate, respectively. Subsequently, the dehumidification system is integrated with a VCAs of small capacity to investigate the overall system performance, typically for hot and humid climatic conditions. A detailed parametric analysis is carried out to investigate the influence of design parameters (air temperature, desiccant temperature, desiccant concentration and solution and air flow rates) on performance parameters (energetic coefficient of performance, humidity removal rate and system effectiveness of dehumidifier and evaporation rate of regenerator). Further, the performance comparison of such an integrated system with the standalone VCAs is also studied in detail. The comparative results revealed that the overall COP of the integrated system is improved compared to the standalone VCAs at which a significant amount of the total latent heat load is being controlled by the dehumidifier.

**Keywords:** Liquid desiccant dehumidification, Hybrid system, ASPEN Plus, simulation, effectivity, COP, humidity, VCACs, MRR, Evaporation rate,

- **Introduction**

The idea of hybrid system consisting dehumidification system along with vapor compression Air Conditioning system (VCACs) system was conceived long ago due to its various energy saving capabilities. This called for extensive research on making it feasible for industrial or commercial applications. We know VCAC system alone has high and efficient cooling capacity; however, the high energy consumption for sensible and latent cooling has always been an issue. The dehumidification system also has wide industrial role and they too tend to consume high electrical power for low evaporating temperatures. The Hybrid VCAC system with dehumidification can achieve the energy saving we need by utilizing low temperature heat. The desiccant dehumidification system has its own charm for its additional cooling effect on the process air. This also reduces a cooling load on VCAC system.

- **Review of Previous Literatures**

Several research works have been done to perfect this hybrid technology since late 1990s. Howell and Peterson [3] made earliest research on hybrid system and its advantage due to of latent heat removal of dehumidification with high efficiency and to use to save sensible cooling load of vapor compression system. No experimental work was done by them but theoretical possibility of economic and energy saving advantages were suggested. Also predicted energy saving and reduced electric amperage consumption air conditioner unit for same outlet air temperature. Dai et al. [2] actually developed and analysed a hybrid system with VCACs with dehumidification technology. A system with two stage dehumidification system and VCAC system along with a desiccant solution regeneration system was established. The Coefficient of Performance COP) of refrigeration unit of this hybrid system (along with evaporative cooling) was compared with a standalone VCAC system at ARI conditions. The results were impressive 80-90 % improvement in COP was obtained and a significant 20-30%

more cooling effect. A decrease in electric power consumption and reduced size of VCAC system was evident. All these effects bar COP increase was due to evaporative cooling in dehumidifiers and dehumidification was also beneficial for VCACs as water condensation was lowered. Liu et al. [4] Developed a theoretical model to simulate heat transfer and mass transfer coefficients in a crossflow and counterflow dehumidifiers. Both the columns used structural packing and mass transfer coefficients as specified inputs were used for simulation. A temperature profile of column stage was done proving that parameters like desiccant temperature, inlet air humidity and temperature were important parameters which influence the equilibrium and rate of mass transfer in between streams. Yang Li et al. [9] constructed a humidification dehumidification (HDH) system and a mathematical model was developed for this system. A theoretical and experimental investigation of operating parameters like inlet stream temperatures and solution concentration on objective functions which are evaporation rate and specific steam consumption was done. The effect of operating parameters was done on each objective function which suggested a high evaporation rate and low SSC are most favourable. Huang et al. [10] In their research, the electricity consumption model and pressure drop model of humidification-dehumidification (HDH) system was developed. Experimental study was carried out to investigate the effects of operating parameters on the pressure drop of HDH system. The adjustment of solution mass flow rate and solution inlet temperature in humidifier is a priority for getting better results. Several experimental setups were investigated by Fumo and Goswami [2000] and Koronaki [2012] which were used in this study with detail for validation purposes. Researchers like Chung and Ghosh (1996), Oberg and Goswami (1997) and Qi et al. (2019), have developed empirical mass transfer correlations from experimental data which best fit for different systems and it was observed that the empirical formula changes with change in any one of the following parameters in dehumidifier systems e.g., desiccant solute and solvent material, Tower packing choice, operation temperature range and flow direction for the two streams (cross, counter or co-current).

A modified hybrid system small scale system developed by Mansuria et al. (2020) [2] developed another working small-scale hybrid dehumidification-VCACs model and performed using common refrigerant R-32a and Calcium Chloride solution as desiccant solution. The study compared the COP increase with a standalone VCAC system and it was found that a 27.54 % of maximum improvement was recorded. Experimental investigation of influence of parameters (input variables) on improved COP of hybrid system and change in humidity ratio of process air was done. Hybrid systems with absorption refrigeration system studied the for a more efficient lithium salt solution by Su et al. (Jan. 2017) [15] proposed a two-stage liquid desiccant dehumidification system with cascaded utilization of low temperature heat. A sensitivity analysis study was done to check impact of desiccant salt concentration in solution, second stage dehumidifier temperature and mass flow ratio of desiccant solution to inlet air, on supply air humidity ratio and COP of the cascade system. Obviously, optimal values obtained coincided with the theoretical predictions in literature.

- **Research Objective**

All the previous research prompted us to select a lithium Chloride salt for desiccant solution for dehumidification. The goal of this study is to develop a numerically robust dehumidifier and regeneration model and integrate with a VCA system, an prove the workability and efficient desiccation property and feasibility in using it with a VCAs model first and then study the parametric sensitivity of its one or more performance index. The parametric analysis reveals to its optimum operating conditions and behaviour which will be used for later research.

- **Liquid Desiccant Dehumidification System**

The dehumidification of air using liquid desiccant solution is done is packed towers mostly with either co-current, cross current or counter current mode, but this work has chosen counter current mode of operation because of its superior performance. The dehumidification has to be coupled with regeneration column to complete the liquid desiccant cycle and then it can be used in continuous operation.

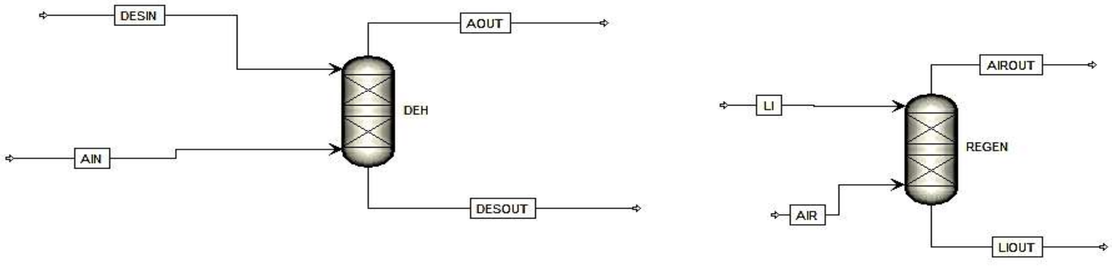



**Figure 1.** Dehumidification Column and Regeneration Column Modelled in ASPEN Plus

The two columns shown in Fig. 1 are modelled in ASPEN Plus V8.8 version to act as packed bed towers for dehumidification using liquid salt solution (Lithium Chloride) as desiccant. The desiccant solution stream is sent to the top stage of the column and ambient air stream is sent to bottom stage. The outlet air streams from both the columns are from the topmost stage and desiccant solution streams are from bottom stage. The liquid desiccant chosen is the Lithium Chloride (LiCl) in water solution which mostly is between 30% to 42% solution in mass percentage. So, by H. E Moran (1956) [3] LiCl.3H2O best fits the modelling. To get desired concentration of in between 30-40% at any point is adjusted by adding calculated moles of water to it.

- **Governing Equations**

To design a dehumidification system, we must be able to model a packed tower into a dehumidification tower. However, in ASPEN Plus version V8.8 does not provide a special dehumidifier, so we model our column using a packed bed absorber (Radfrac) column. To successfully design first we must prove the hypothesis that equations for design of absorber column also work for a dehumidification process.

The basic equations for design of a packed tower are same for both dehumidification tower or absorber, starting from differential mass balance eqn. [Perry's]

For dehumidification process the overall mass balance equations are [Treybal] on converting all the important equations from mass fraction to humidity ratio equation we get equation from which proves the hypothesis that absorber equations work well enough for dehumidification operations.

The Transport equation for design basis from mass balance equations and mass transfer coefficient which is calculated by mass transfer correlation equations. The governing equations are solved in ASPEN Plus compiler by converging method of Newton's method procedure as generally is used in Radfrac.

- **Mass Transfer Correlations**

The mass transfer correlations for dehumidification process depends on every factor od column design. The correlations change for each process if different Desiccant solution, packing material, packing factor, tower dimensions any one or all of these factors change. Therefore, a careful selection of mass transfer correlations is necessary to model correctly working column process, whether be it absorbing or dehumidification or regeneration.

While designing Radfrac tower ASPEN Plus, general equations of mass transfer correlations format is present where the user has to model their correlations according to the rigid equation skeleton. These equations are given as

$Sh_L = c_1 \cdot Re^a_V \cdot Re_L^b \cdot Sc_L^c \cdot Fr_L^d \cdot \varepsilon^e \cdot (1-\varepsilon)^f$ …………………. eqn. (1)

$Sh_V = c_2\, Re_V^i \cdot Re_L^j \cdot Sc_L^k \cdot Fr_L^l \cdot \varepsilon^m \cdot (1-\varepsilon)^n$ …………………..eqn. (2)

Where Sh = Sherwood Number for vapor $Sh_V$ and liquid phase $Sh_L$

Re = Reynold's Number for vapor $Re_V$ and liquid phase $Re_L$

Sc = Schmidt's Number for vapor and liquid phase.

$Fr_L$ = Froude's Number for liquid phase

$\varepsilon$ = Void fraction of Packing material, which is again, calculated by equation (3) [Onda et al.]

$a_t D_p = 6(1-\varepsilon)$ ……………. Eqn. (3)

Where, $a_t$ = Total specific surface area of packing [$m^2/m^3$] (Total surface area per volume)

And $D_p$ is nominal size of packing [m]

- **Experimental Validation of Model**

For a mathematically developed model, it is important to validate the numerically simulated data with experimental data, to check its nearness to real process. The experimental validation does not need to be absolutely accurate but, the deviation in experimental and simulated data must be within an acceptable range.

The dehumidifier and regenerator model was developed according to the design data provided by of Fumo and Goswami (2000) and then the model was validated with experimental data provided by them. The Table 1. lists the packed bed design parameters for the dehumidifier and regenerator column both.

**Table 1. Design of Packed Tower Fumo and Goswami**

| Design Rating of packed Tower | Value | Remarks |
|---|---|---|
| Internal diameter | 24 cm | - |
| Total diameter | 25.4 cm | - |
| Height of packing | 60 cm | - |
| Type of packing | Rasuchert Hiflow | Pall ring is selected. |
| Specific surface area of packing | 210 $m^2/m^3$ | - |
| Size or dimension of packing | 2.54 cm | - |
| Packing material | Polypropylene | Specified as plastic in ASPEN |

The constructed column was used for LiCl solution as desiccant and mentioned the mass transfer correlations by Onda et al. (1968) was best fitting for the column hence the ASPEN developed model also used the same correlations also provided by Onda et al (1968)

The equations are then modified to fit in the column rating design in ASPEN Plus equation format and design for both dehumidifier and regenerator operation into eqn. (1) and eqn. (2). and run with the specified stream input data from the Table 2. And Table 3. The input parameters are air superficial flowrate i.e mass flowrate per cross sectional area, L as superficial solution flowrate, Solution Temperature TL, air temperature Ta, LiCl mass concentration in Solution X in percentage and inlet air humidity ratio Y in (kg water/kg dry air). Air and solution flowrates G and L outlet experimental data were not available hence were not considered for simulation and change in LiCl concentration was too little hence its comparison was ignored.

**Table 2.** Experimental Validation of Dehumidification data (expt. No. 5) [Fumo Goswami(2000)].

| | G (kg/($m^2$s)) | $T_a$ (℃) | Y (kg/kg) | L (kg/($m^2$s)) | $T_L$ (℃) | X (%) |
|---|---|---|---|---|---|---|
| **Experimental input data** | 1.183 | 40.10 | 0.018 | 6.287 | 30.50 | 34.40 |
| **Experimental output data** | - | 33.10 | 0.0115 | - | 32.900 | 34.3 |
| **Simulation output data** | - | 33.75684 | 0.011128 | - | 31.91966 | 34.3 |
| **Error** | - | 0.65684 | 0.00037 | - | -0.98034 | - |
| **Error Percentage** | - | 1.98 | 3.230549 | - | 2.9 | - |

**Table 3. Experimental Validation of Regeneration data (Expt. No. 3) [Fumo and Goswami (2000)].**

| | G (m/s) | $T_a$ (℃) | Y (kg/kg) | L (m/s) | $T_L$ (℃) | X (%) |
|---|---|---|---|---|---|---|
| **Experimental input data** | 1.4380 | 29.80 | 0.0177 | 6.4790 | 65.10 | 34.50 |
| **Experimental output data** | - | 57.50 | 0.0488 | - | 56.60 | 35.20 |
| **Simulation output data** | - | - | 0.050239 | - | - | 35.19 |
| **Error** | - | - | 0.0014 | - | - | - |
| **Error Percentage** | - | - | 2.949318 | - | - | - |

The data from simulation for both regeneration process and dehumidification process are validated against experimental data from Fumo (2000), and thus the error or deviation from experimental data is recorded. The tables 2 and 3 are one such experimental data sample from the literature whose error percentages in output data and input values are given. The error percentages in the sample data from table 2 do not exceed 3.5 % and regeneration 3% which are acceptable range. The maximum error percent in dehumidifier simulation data and experimental data is 4.68% and it was found that regeneration data error percent is 8.65% for humidity ratio in outlet air stream.

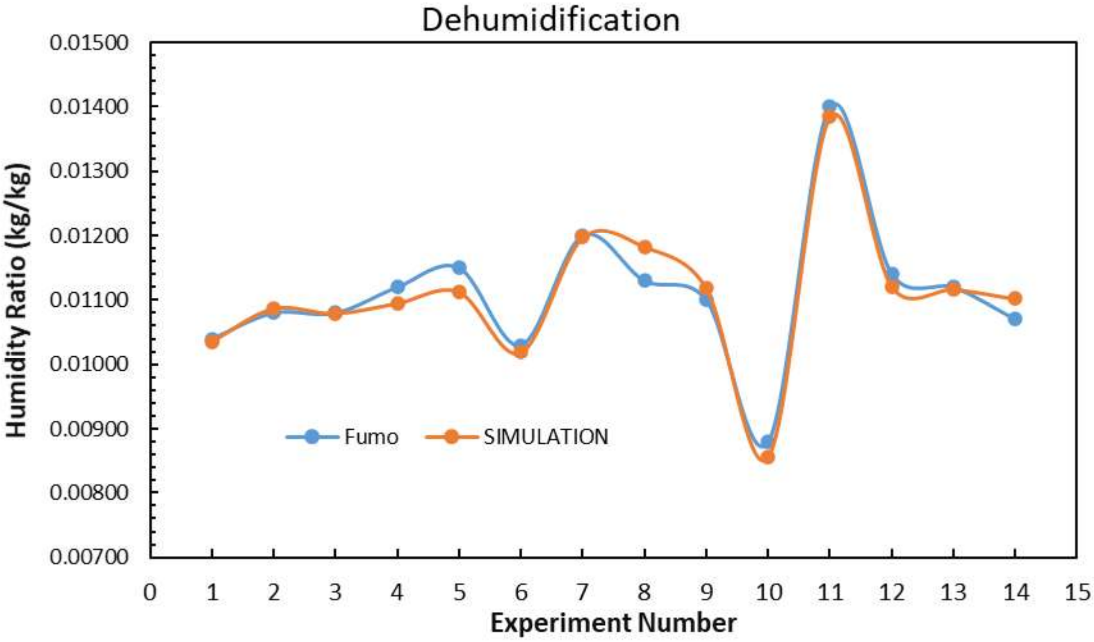



Figure 2. Experimental Validation of Absolute humidity for Dehumidifier Data (Fumo, Goswami (2000))

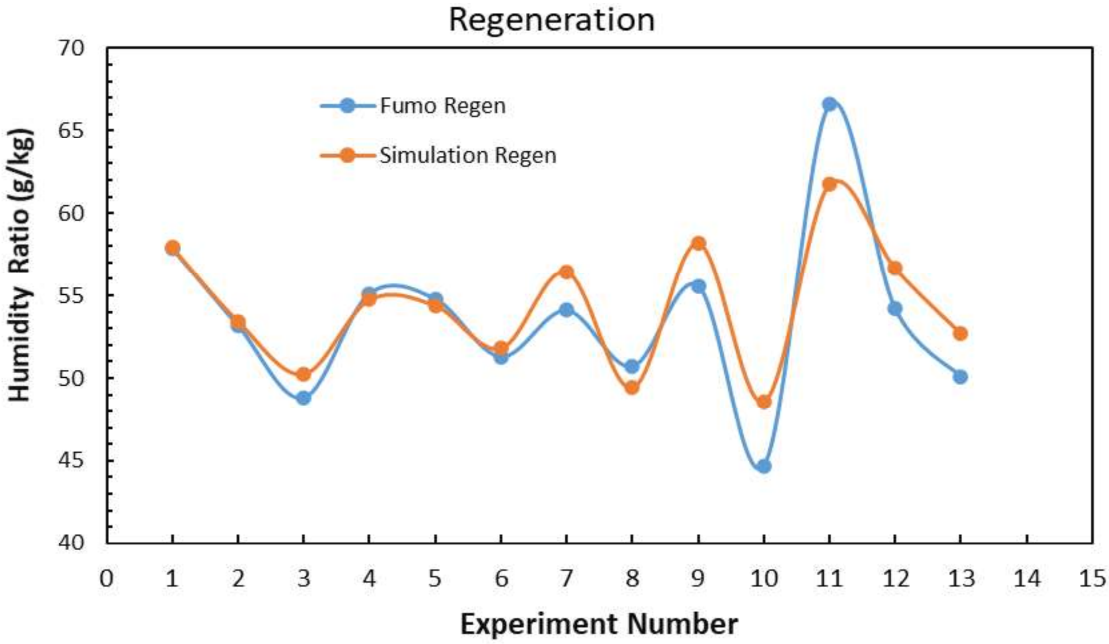



Figure 3. Experimental Validation of Absolute Humidity for Regenerator Data (Fumo, Goswami (2000))

The above two graphs of fig. 2 and fig. 3 are the direct comparison of humidity ratio of simulation and experimental data for corresponding experiment number as mentioned in literature Fumo (2000). We can see that the data are better and closer for dehumidification modelling than regenerator data however still the maximum error percentages are in acceptable range for both models.

To check the robustness, we have again validated the simulation model with experimental data from Koronaki (2012) which also simulated their data with the mass transfer correlations from Chung and Goswami (1996). The dehumidifier model was again developed according to the design data provided by of Koronaki et al. (2012) and then the model was validated with experimental data provided by them. The Table 1. lists the packed bed design parameters for the dehumidifier column.

**Table 4. Design of Packed Tower Koronaki et al. (2012)**

| **Design Rating of packed Tower** | **Value** | **Remarks** |
|---|---|---|
| Internal diameter | 15.24 cm | - |
| Height of packing | 41 cm | - |
| Type of packing | PVC Structured | Generic Structured used |
| Specific surface area of packing | 223 $m^2/m^3$ | - |
| Packing material | Polypropylene | Specified as plastic in ASPEN |

The constructed column was used for LiCl solution as desiccant and mentioned the mass transfer correlations by Chung and Ghosh (1996) was used for their mathematical design column hence in this research ASPEN developed model also used the same correlations also provided by Chung and Ghosh (1996).

This equation is solved to fit into eqn. (2) to be modelled inside ASPEN dehumidifier model and using same input parameter values from experimental data, simulation is run to get output dataset of stream parameters. The stream parameters e. g. outlet air humidity ratio Wa, solution and air temperature (Ts and Ta respectively) are compared with simulation data from developed ASPEN Plus model. Table 4. mentioned the input mass flowrates of air and solution, ma and ms with other parameters. The experimental data for outlet parameters of X, ma and ms are not provided.

**Table 5. Experimental Validation of Regeneration Data (Expt. No. 5) [Koronaki (2012)].**

| | **$m_a$ (kg/s)** | **$m_s$ (kg/s)** | **$T_a$ (℃)** | **$W_a$ (kg/kg)** | **$T_s$** | **X (%)** |
|---|---|---|---|---|---|---|
| **Experimental input data** | 0.0318 | 0.1919 | 24.1. | 0.0149 | 16.90. | 31.0 |
| **Experimental output data** | - | - | 19.1. | 0.008 | 18.30. | - |
| **Simulation output data** | - | - | 18.498 | 0.007803 | 19.14721 | - |
| **Error** | - | - | -0.602 | -0.000197 | 0.8472104 | - |
| **Error Percentage** | - | - | -3.15 | -2.467 | 4.63 | - |

After the simulation and validation of data we have table 4, in which one such experimental data sample from the literature whose error percentages in output data and input values are given. The error percentages in the sample data from table 4 do not exceed 4.65 % and which is acceptable.

The maximum error percent in dehumidifier simulation data and experimental data for humidity ratio value was found to be 8.9% which is acceptable but is still higher. For outlet air temperature, the data has maximum error of 3.15% was observed which is very promising. But for solution outlet temperature, the simulation data deviates by almost 10% which raises concern.

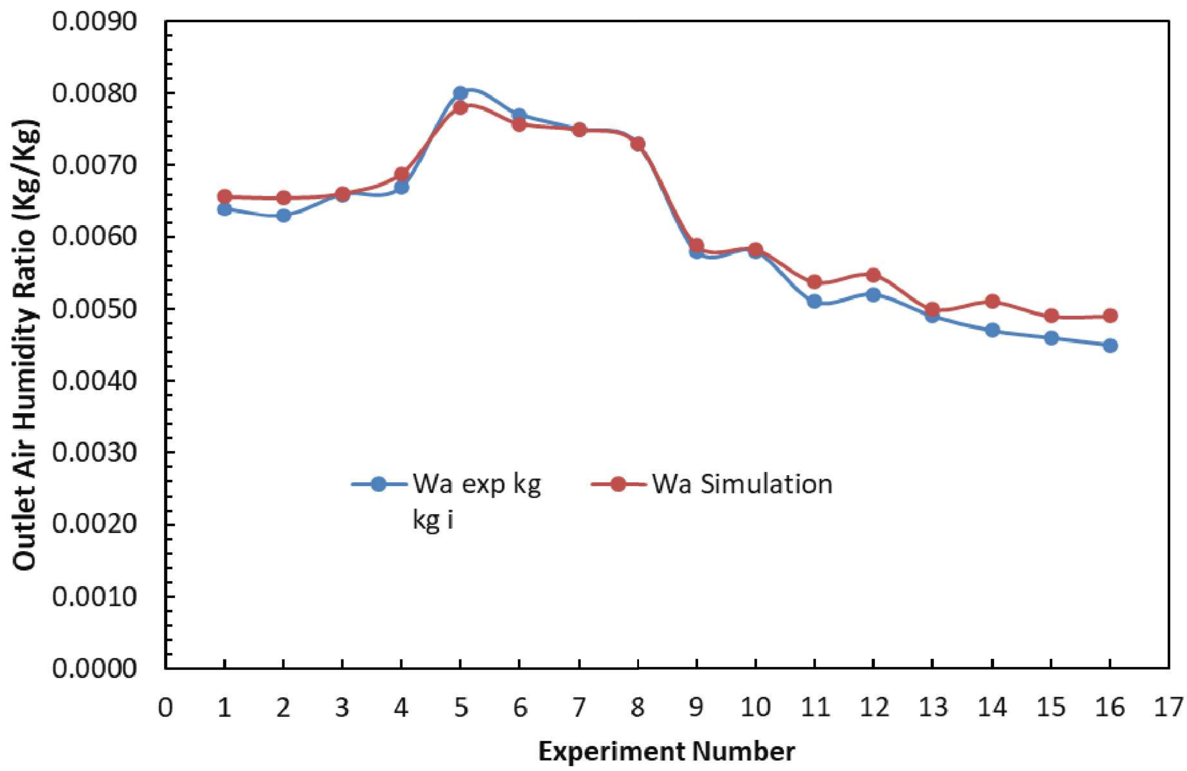



**Figure 4.** Experimental data comparison with simulation data Koronaki (2012).

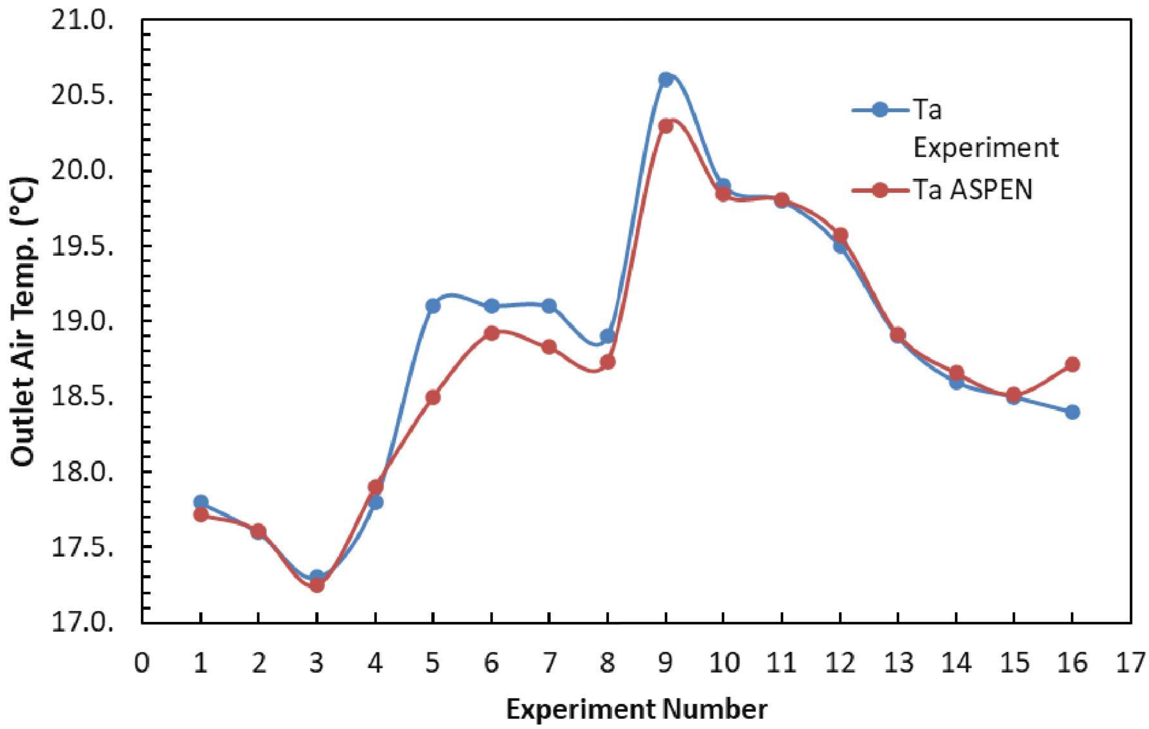



**Figure 5.** Validation of simulation data with Experimental Koronaki (2012)

The validation of air temperature and air humidity ratio graphs in fig. 4 and fig. 5 are the proof of successful modelling of dehumidifier packed tower using the transfer correlations from Chung and Ghosh (1996) for gas side mass transfer. In figure 4 the maximum error of 8.9% is observed on experiment number 16 which is apparent on graph. The experiment number 5 in fig.5 is point of maximum error which is 3.15%. The solution temperature has a larger deviation up to 11% hence the conclusion can be drawn that numerical modelling of random packings are better to model in ASPEN Plus, however if properly defined structured packing can also work well.

These validation graphs are proof of robust system modelling and prompts us to use these correlations equations for dehumidifier column design in later stages of the research.

- **System Description of Hybrid Desiccant dehumidification integrated with VCAC system.**

The system shown in Fig.6 shows the schematic description flowchart of the proposed system that combines a VCAC system and a liquid desiccant dehumidification (LDDh) system. This consists two complete loops; outer cycle is desiccant dehumidification system along with desiccant regeneration system, inner cycle is a standard VCACs driven by a standard refrigerant R-32a.

In LDDh cycle the humid warm air is let inside a dehumidifier column which lowers the humidity ratio of the inlet air and also reduced temperature alongside due to evaporative cooling. The mass transfer of water vapor from air to desiccant solution is due to colder solution temperature the humidity of gas is higher than gas-liquid interface which the moisture transfer from air to solution. then the dried air is now sent to a solution heat exchanger which serves as evaporator for refrigeration cycle. The air is now cooled to suitable temperature for supply. The desiccant solution chosen is Lithium Chloride solution for its availability and better efficiency. A colder desiccant solution is passed through dehumidifier column and is diluted in the process. The diluted solution is now needing to be regenerated in order to reverse the mass transfer, the solution is heated in condenser unit of refrigeration process. This increase in temperature shifts the equilibrium concentration so that the solution now loses moisture when solution comes in contact with air in Regenerator unit after it exits the condenser and is heated. A bone-dry air stream or ambient stream is fed to regenerator to absorb the moisture from hot dilute desiccant solution. The solution now again concentrated undergoes cooling either in an air-cooled fin and tube heat exchanger or a water-cooled solution heat exchanger. By cooling the solution becomes ready for dehumidification operation and this completes the desiccant solution cycle.

The inner cycle is a standard VCAs cycle, consisting of compressor to pressurize the refrigerant. Hot and compressed refrigerant is passed through a condenser to liquify the refrigerant rejecting heat to the desiccant solution. The liquified refrigerant is now sent to evaporator after being passed through throttle valve to depressurize. In evaporator the real work is done by the refrigerant where it cools the air stream. This cooling effect is the useful work load needed to calculate coefficient of performance (COP). The exit stream from evaporator is again channelled into compressor completing the refrigeration cycle.

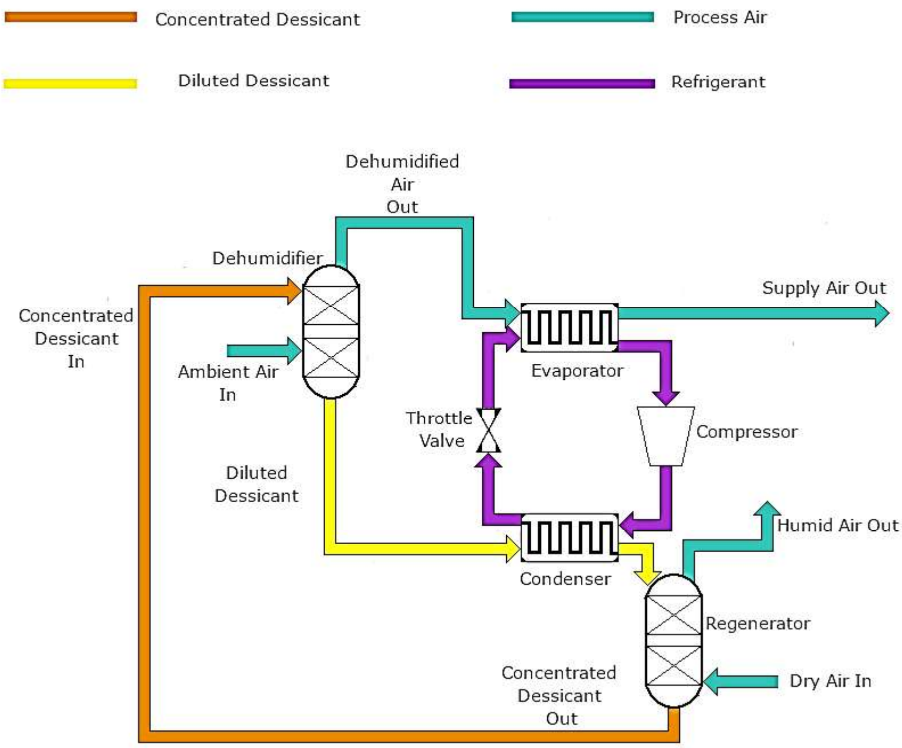



Figure 6. Schematic Configuration of a Hybrid System of Liquid Desiccant Dehumidification and VCAC Cycle

- **System Simulation and Modelling**

The hybrid model is developed in ASPEN Plus to simulate the working conditions of a similar lab scale experimental setup validating the objective parameters and working conditions. The following sections explain the approached designing.

- **Thermodynamic Property Methods**

The simulation consists of different compound and mixture streams which all behave different under real conditions (pressure and temperature conditions), so any one model is not suitable for all streams as highly condensed refrigerant stream and hot air stream have very different thermodynamic property equations.

ASPEN Plus has built in models for simulation of hybrid system dehumidification and VCAC system. For desiccant solution electrolytic non-random two liquids (ELECNRTL) method in ASPEN PLUS is most suitable with all binary interactions specified. For air streams ELECNRTL both best because of its direct interaction with desiccant solution as well as with minimal deviations from ideal behaviour as the mass flowrate are very low. The solution refrigerant has REFPROP as best suited model for thermodynamic calculations.

- **ASPEN Model**

Two models – one Hybrid LDDh with VCAC system and a Standalone VCACs were developed in ASPEN Plus software for simulation and comparison with each other. All parameters of VCAC cycle were kept same in standalone VCACs model and Hybrid Model. Fig. 7. shows the complete Hybrid Model flowsheet and Fig. 8. is the standalone VCACs flowsheet.

The hybrid LDDh system Fig.7 has the same flowsheet and component specification as Fig.6 but with an extra Flash Drum. This flash drum is to flash evaporate any air dissolved in desiccant solution in simulation. This dissolution is a common phenomenon in ASPEN RadFrac column operation, hence we introduce a flash drum only for the sake of it, to minimize air content in solution.

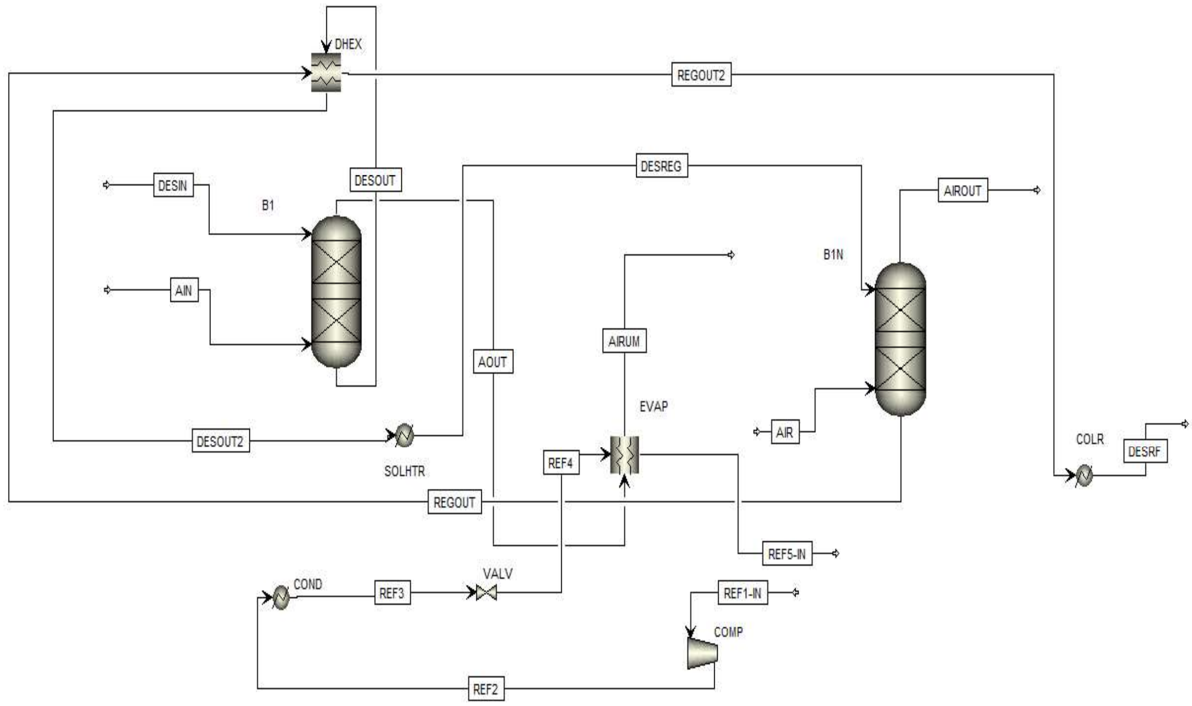



**Figure 7.** Model and Material Streams of Hybrid System in ASPEN Plus

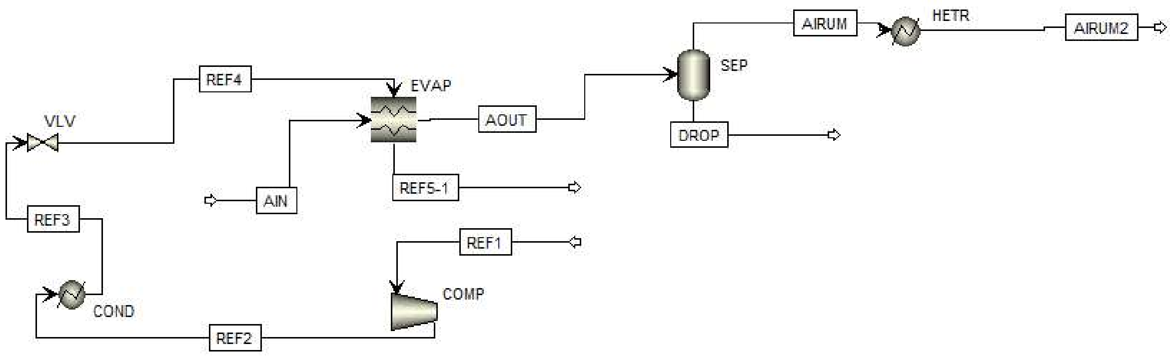



**Figure 8.** ASPEN Flowsheet of Standalone Vcacs Model for Reference

- **Parametric specifications and Assumptions**

The Table 6 demonstrates all initial parametric and design specifications for base-case to simulate the system which fulfils all required objectives.

**Table 6.** Parameter Specifications of the Hybrid System

| *Component* | *Parameter* | *Value* | *Unit/Remarks* |
|---|---|---|---|
| ***LiCl solution*** | Temperature | 29 | ℃ |
| | Salt concentration | 0.39 | Mass Fraction |
| | Mass flow rate | 0.67 | kg/s |
| | Inlet pressure | 1.0 | Atm |
| ***Ambient air*** | Dry Bulb Temp. | 33 | ℃ |
| | Humidity Ratio | 21.5 | gm/kg |
| | Mass flow inlet<br>Air velocity | 0.086<br>2 | kg/s<br>m/s |
| | Inlet pressure | 1.0 | Atm |
| | REGN inlet pressure | 1.0 | Atm |
| ***Compressor*** | Efficiency | 95 | Percent |
| | Outlet pressure | 5.3 | Bar |
| | Pressure range | 3.7-5.3 | bar |
| ***Valve*** | Outlet pressure | 3.7 | Bar |
| ***Dehumidifier*** | Stages | 2 | |
| ***Regenerator*** | Stages | 2 | |

- **Component Modelling and Working**

ASPEN Plus components are used with predefined standard equations for calculations of outlet streams conditions. Following subsections are details of the component blocks used in the simulation.

- **Dehumidifier Column**

Denoted as DEH in flowsheet, the dehumidifier column is modelled as RadFrac column in ASPEN plus with packing material as Pall ring and specified number of stages to ensure a observable change in air humidity. The air stream is sent at bottommost stage and desiccant is dropped at topmost stage known as condenser stage in ASPEN. The mass transfer occurs due to vapor pressure difference in both streams. The packing material, packing factor and height equivalent theoretical plate HETP were defined accordingly.

- **Regenerator Column**

This is denoted as REGN in ASPEN flowsheet. Its modelling and design are similar to dehumidifier column, even though its purpose is polar opposite. For convenience the equipment designing parameters like packing characteristics and stages are kept same. The input streams are a dilute desiccant at top stage and a drier air stream at bottom stage. The mass transfer direction is reversed because of the water vapor pressure difference of desiccant is air stream.

- **Evaporator**

Evaporator component is modelled as solution heat exchanger blocks in ASPEN and denoted in flowsheet as EVAP. The refrigerant and air stream are input for evaporator so Both REFPROP and ELECNRTL are used as thermophysical methods respectively. Flow pattern in the heat exchanger is specified as counter current flow. The vapor quality of refrigerant outlet stream is defined as one which in turn determines air stream outlet temperature. This is only one major input which specifies that latent heat is added to refrigerant stream. The air stream is cooled to comfortable temperature and humidity and is now ready for supply.

- **Condenser**

In ASPEN plus model condenser has both Refrigerant stream and desiccant stream for each stream thermodynamic method needs to be clearly specified. Denoted as COND in flowsheet is a heat exchanger model which calculates the net duty and heat duty by given flash type output parameters. The liquid fraction of outlet refrigerant is set as one. However, in condenser both sensible and latent heat is removed from refrigerant to heat the desiccant solution.

- **Pressure Changers**

An isentropic compressor model denoted as COMP in both flowsheets are used to increase the refrigerant pressure to ensure that when the refrigerant is throttled using a valve (VLV in flowsheet fig. 7 fig. 8) the refrigerant reverts back to its original state after doing its work in evaporator and condenser.

- **Performance Indices**

According to various literature sources we considered two performance parameters or indices of major significance in designing our hybrid system.

- **Coefficient of Performance of Hybrid System (COPH)**

Generally defined as direct cooling effect produced by an equipment, however our cooling effect is also dependent on dehumidification process as it changes air quality. According to many literatures referenced e.g. Su. Et al. [15] direct cooling effect can be calculated by following equation.

$$COP_H = \frac{Qdeh - Qvcr}{\sum W} \quad \text{………………. eqn. (2)}$$

where, $Q_{deh} = H_{in} - H_{O1} = [(m_{air} \cdot h_{in}) - (m_{air2} \cdot h_{O1})]$…………… eqn. (2.1)

$Q_{VCR} = H_{O1} - H_O = \{(m_{air2} \cdot h_{O1}) - (m_{air3} \cdot h_O)\}$……………….. eqn. (2.2)

Where, *Hin* = Total Mass Enthalpy of inlet air which is also calculated as ($m_{air} \cdot h_{in}$), *Ho* is Total mass enthalpy of supply or outlet air from evaporator and $H_{O1}$ is the Total mass enthalpy of air stream coming out of dehumidifier column.

The heat removal loads are divided by summation of net-work ($\sum W$) and heat duty required for all components like compressor, condenser, solution heater etc. Qdeh is the dehumidifier heat load removal from the air stream in the process and Qvcr is the heat load removal in evaporator of VCAs system.

$\sum W$ is given as sum of $Q_{sol}$, $W_{comp}$ and $Q_{cond}$, mainly, for given calculations in this work where $Q_{sol}$ is net heat duty required by solution heater for regeneration, $W_{comp}$ for compressor net power input and $Q_{cond}$ is the condenser net duty of heat taken out from refrigerant.

- **Change in Humidity Ratio of Air Stream (Δω)**

Humidity ratio calculated in ASPEN by grams of moisture per kg of dry air. Extent of change in this humidity depicts effectiveness of dehumidifier and the point where this parameter changes from positive value to negative value holds a very important role in designing of this system.

$\Delta\omega = \omega_1 - \omega_2$ eqn. (3)

Where, Δω is the change in humidity ratio of process air, ω1 is the humidity ratio of input ambient air stream to dehumidifier and ω2 is humidity ratio of drier air stream out of dehumidifier.

- **Dehumidifier Effectiveness (εm)**

The dehumidification or dehumidifier/column effectiveness is the measure of performance of the dehumidifier. It is defined as the ratio of actual change in absolute humidity ratio between inlet and outlet air conditions of dehumidifier to the maximum possible change in absolute humidity. It can be expressed as below equation (3).

$\varepsilon_m = \frac{\omega 1 - \omega 2}{\omega 1 - \omega e}$ ……………… eqn. (4). or $\varepsilon_m = \Delta\omega / (\omega_1 - \omega_e)$ ………… eqn. (5)

where ω1 is the absolute humidity of ambient air at inlet of dehumidifier and ω2 is the absolute humidity of outlet air stream of dehumidifier column. The ωe is absolute humidity of air which is at equilibrium with LiCl solution at inlet concentration and temperature.

- **Moisture Removal Rate or Condensation Rate (MRR).**

It is the performance measure of a dehumidifier column which is defined as the total amount of moisture removed from the air stream in the column. It is expressed mathematically as equation given below

$MRR = \Delta\omega \cdot m_{air}$ or $MRR = (\omega_1 - \omega_2) \cdot m_{air}$ ………….. eqn. (6)

Where ω1, ω2 and Δω has the same meaning in eqn. (3) and eqn. (5), and mair is the mass flow rate of air stream out of the dehumidifier column.

- **Evaporation rate (EVR) for a Regenerator**

This is similar performance parameter as MRR but for Regenerator column. The EVR is defined as the total moisture added to the air stream from hot and dilute desiccant stream. Mathematically it is defined as

$MRR = \Delta\omega_r \cdot m^r_{air}$ or $EVR = (\omega_3 - \omega_2) \cdot m^r_{air}$ ………….. eqn. (7)

Where ω2 is the same as in eqn. (6) or the humidity ratio of air stream entering the regenerator column, ω3 is the humidity ratio of air stream exiting regenerator which is more humid and Δωr is the difference between the two. The mrair is mass flow rate of air stream out of the regenerator column.

- **Results and Discussion**

We conduct both simulation and parametric analysis in ASPEN plus. For simulation a base case is set with input parameters in Table 1. Then the sensitivity analysis of each performance index is done in order to get an idea of optimal performance.

- **Mass Balance for Model Verification**

Mass balance of cycle end streams in ASPEN Flowsheet prove the feasibility of cyclic process of desiccant and Refrigerant cycle.

**Table 8.** Mass Balance Verification in Hybrid System (fig.3)

| Stream Name (ASPEN flowsheet) | DESIN | DREG | REFIN | REFCOMP |
|---|---|---|---|---|
| Total Desiccant Mass flow (kg/sec) | 0.70 | 0.699976 | | |
| LiCl mass flow (kg/sec) | 0.273 | 0.27299 | | |
| Water mass flow (kg/sec) | 0.427 | 0.4270004 | | |
| Refrigerant Mass flow (kg/hr) | | | 7.0 | 7.0 |

- **State Point Results of Proposed System**

ASPEN generated state points refer to each stream state in cyclic process. Table 3. depicts the stream states denoted by names in ASPEN Flowsheet. These are a base case simulation values of both input and results before changes are made in parametric analysis.

**Table 9.** Streams State Point Results (States Refer To Fig.3)

| Label Number | State | Mass flow rate (kg/sec) | Temperature (℃) | Pressure (bar) | Vapor Fraction | |
|---|---|---|---|---|---|---|
| Dehumidifier inlet air | | | | | | Humid. Ratio |
| 1 | AIN | 0.086 | 33 | 1.01 | 1 | |
| 2 | AOUT | 0.085 | 29.4646 | 1.01 | 1 | |
| 3 | AIRRUM | 0.084593 | 21.923 | 1.01 | 1 | |
| Regenerator inlet air | | | | | | |
| 11 | AIR | 0.086 | 33 | 1.01 | 1 | |
| 12 | AIROUT | 0.0869601 | 37.34352 | 1.01 | 1 | |
| Desiccant Loop | | | | | | |
| 4 | DESIN | 0.7 | 30 | 1.01 | 0 | |
| 5 | DESOUT | 0.7014069 | 33.34803 | 1.01 | 0 | |
| 6 | DESOUT2 | 0.7014069 | 44.27037 | 1.01 | 0 | |
| 7 | DESREG | 0.7014069 | 49.52 | 1.01 | 0 | |
| 8 | REGOUT | 0.7004 | 45.46677 | 1.01 | 0 | |
| 9 | REGOUT2 | 0.7004 | 34.5 | 1.01 | 0 | |
| 10 | DESRF | | 29 | | | |
| Refrigerant Loop | | | | | | |
| 13 | REF1-IN | 0.00195 | 12 | 3.7 | 1 | |
| 14 | REF2 | 0.00195 | 39.78799 | 5.3 | 1 | |
| 15 | REF3 | 0.00195 | -12.69909 | 5.3 | 0 | |
| 16 | REF4 | 0.00195 | -22.41809 | 3.7 | 0.0468382 | |
| 17 | REF5 | 0.00195 | -22.41809 | 3.7 | 1 | |

**Table 10.** Stream State Point Results for Standalone VCAC System (Refer Fig.4)

| | State | Mass flow rate (kg/sec) | Temperature (℃) | Pressure (bar) | Vapor Fraction |
|---|---|---|---|---|---|
| Refrigerant Loop | | | | | |
| 5 | REF1 | 0.0127 | 12 | 3.3 | 1 |
| 6 | REF2 | 0.0127 | 54.72995 | 5.7 | 1 |
| 7 | REF3 | 0.0127 | -10.63026 | 5.7 | 0 |
| 8 | REF4 | 0.0127 | -25.34751 | 3.3 | 0.0937756 |
| 9 | REF5 | 0.0127 | 12 | 3.3 | 1 |
| Air Stream | | | | | |
| 1 | AIR1 | 0.086 | 33 | 1.0135 | 1 |
| 2 | AIR2 | 0.086 | 13.6 | 1.0135 | 0.9817526 |
| 3 | AIR3 | 0.0850154 | 13.6 | 1.0135 | 1 |
| 4 | AIR4 | 0.0850154 | 22 | 1.0135 | 1 |

- **Parametric Analysis of Performance Indices**

A parametric sensitivity demonstrates the influence of design parameters on performance of the system. This sensitivity analysis provides a specific insight to the system when applied through various operating conditions. Our main performance indices COPH and Moisture effectiveness (εm) moisture removal rate MRR and evaporation rate EVR are the design specifications whose response with respect to changing design specification parameters considered, e.g., Desiccant solution Temperature (Td), air mass flow rate (mair) and (mrair), Desiccant Concentration (Xin) to dehumidifier and regenerator, Air Temperature (Tair) and Desiccant flowrate (mdes). When one parameter is changes the others are set constant and the response of each performance index is calculated in ASPEN and plotted. However, for calculation of COP the desiccant cycle was set to complete the loop by adjusting input parameters.

Fig. 9 below shows the COPH value calculated against varying desiccant temperature (Td) The decreasing trend in COPH response is as expected. The decreasing trend of COPH is due to reduced dehumidification duty which is direct result of lesser dehumidification because of increased vapor pressure of solution reducing the driving force of dehumidification process. Minimum value of Td taken is 25℃ which is low enough for a solution to be stored in a tropical climate condition without additional cooling requirement, and the highest value of Td taken is 38℃ because above this temperature the no satisfactory dehumidification occurs and COPH value falls too low (rendering the purpose of dehumidifier null).

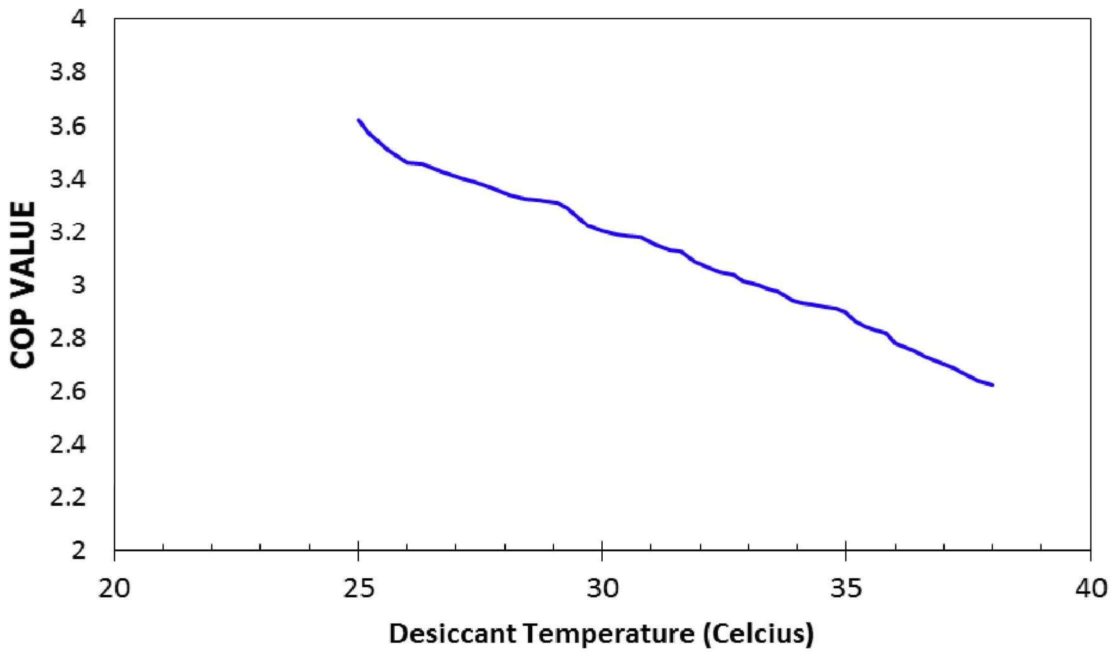



**Figure 9.** COPH vs Desiccant Temperature Td

The fig. 10 is sensitivity of COPH with respect to mair. The trend is increasing throughout the graph but the slope of the curve decreases after around 0.09 kg/s. This is because the dehumidifier heat load is increased up to a certain point as desiccant solution flowrate is fixed and ratio of solution to air mass flowrate has reached optimal level at 0.09 kg/sec. As the air flowrate crosses optimal point the dehumidifier load starts decreasing but the flowrate increase dominated the COP calculation hence, the slope of the graph after this point is less.

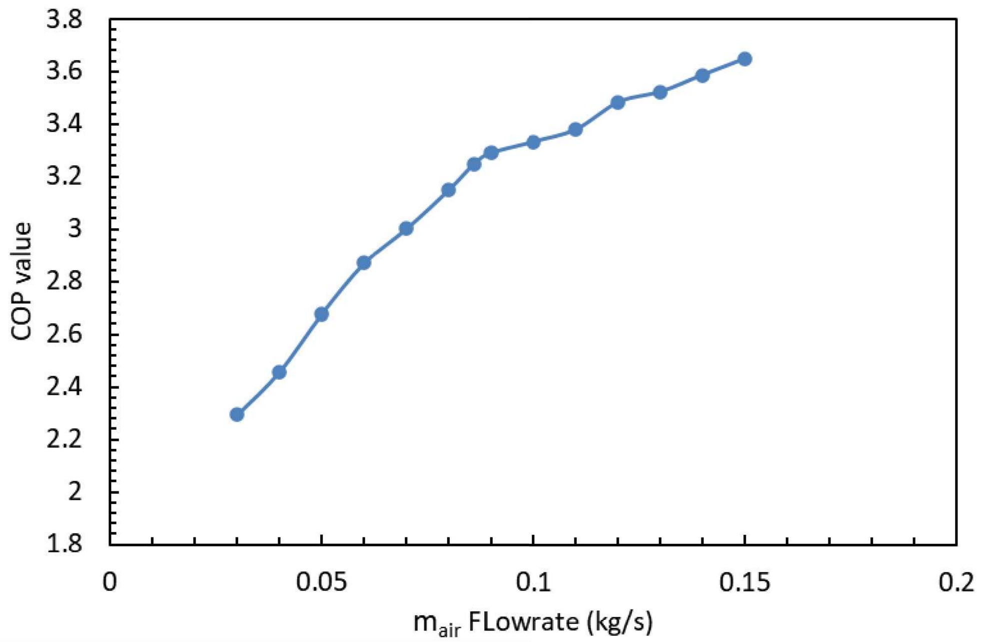



**Figure 10.** COPH Vs Mair Air Mass Flowrate

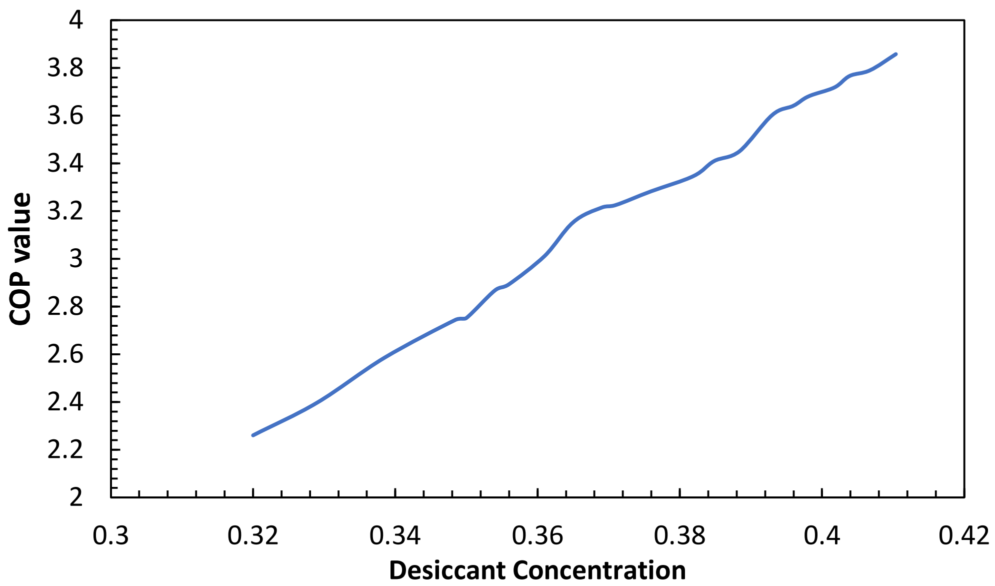



**Figure 11.** COPH Vs Desiccant Concentration in Solution

The $COP_H$ variation with respect to concentration of LiCl in solution is shown in fig. 11. Evidently the $COP_H$ value is increasing with increasing concentration of LiCl in solution. This trend is increasing because the higher concentration causes higher dehumidification of air thereby increased dehumidifier heat load removal from air. The concentration varies from 0.32 to 0.41 of LiCl concentration because below 0.32 the dehumidification does not occur satisfactorily to have any impact on COP and above the 0.41 mass concentration the LiCl solution tend to crystallize in real life at room temperature (28℃-35℃).

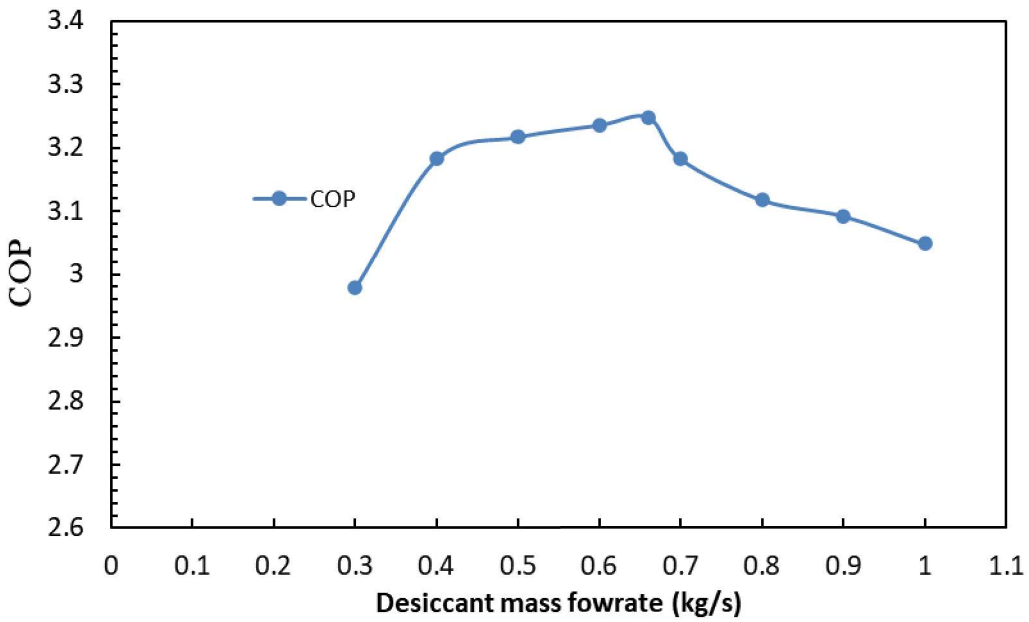



**Figure 12.** COPH vs Desiccant Mass Flowrate.

The figure 12 above depicts COPH variation with Desiccant solution mass flowrate has an increasing trend up to a certain point and then the trend decreases with increasing flowrate. The increasing trend is due to higher dehumidification with increased desiccant flowrate thereby increasing dehumidification load removal. But after a certain point the dehumidification is not as pronounced and the heat required by solution heater to regenerate the solution which causes the COPH to decrease if the solution flowrate is increased into dehumidifier column. The solution mass flowrate has to be at least three times the air mass flowrate to show any significant dehumidification hence 0.3 kg/s is lowest point on the plot.

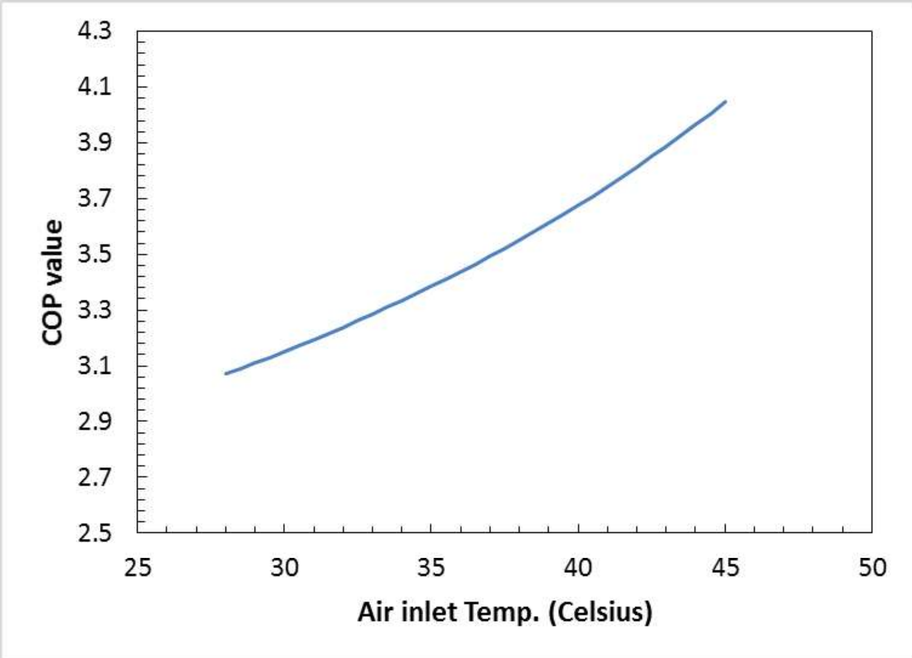



**Figure 13.** COPH vs Temperature of Inlet Air

The inlet air at ambient conditions into dehumidifier also affects COPH if its temperature changes, as depicted in fig. 12. COPH is apparently increasing with increased air temperature. This increasing trend is due to two main reasons, firstly the dehumidifier column has greater dehumidification heat removal for lower air temperature and for higher temperature sensible cooling in packed bed column also occurs because the solution temperature is fixed at 29°C. Another reason is that the solution when comes in contact with the hotter air the outlet temperature of air is also increased hence, the heat duty required by solution heater for regeneration decreases which gives higher COPH value. The ambient air temperature is chosen to vary between 28°C to 45°C which is the temperature range of a tropical climate condition year-round, as the hybrid dehumidification VCAs works best for warm and humid climates.

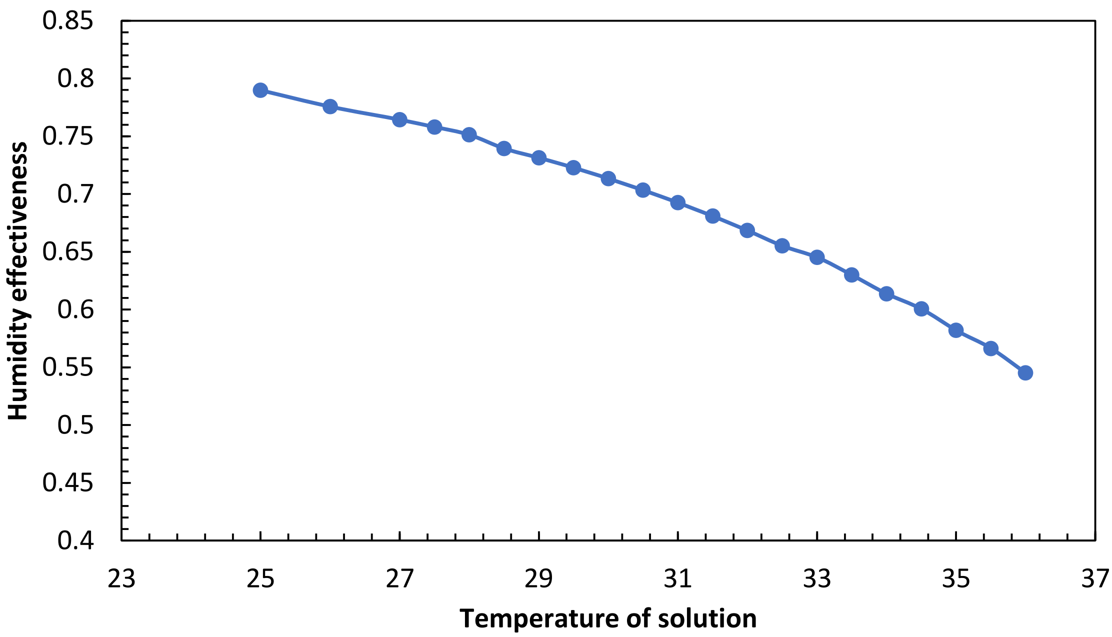



**Figure 14.** Dehumidification Effectiveness vs Solution Temperature

Dehumidifier effectiveness decreases of a solution with increasing temperature as evident from fig. 14 plot. The temperature of solution increases the vapor pressure of the solution which in turn increases the equilibrium humidity ratio of the system but the actual humidity ratio of outlet air stream increases more steeply, hence the effectiveness is decreased at higher temperature. The calculation results from ASPEN also show that $\Delta\omega$

decrease with increasing temperature which is obvious. The temperature range in plot of fig. 14 is chosen to be 25°C to 36°C as for a tropical warm climate condition the lowest room temperature for liquid storage would not dip below 25°C with no additional cooling and solution temperature higher than 36°C would not give good dehumidification results.

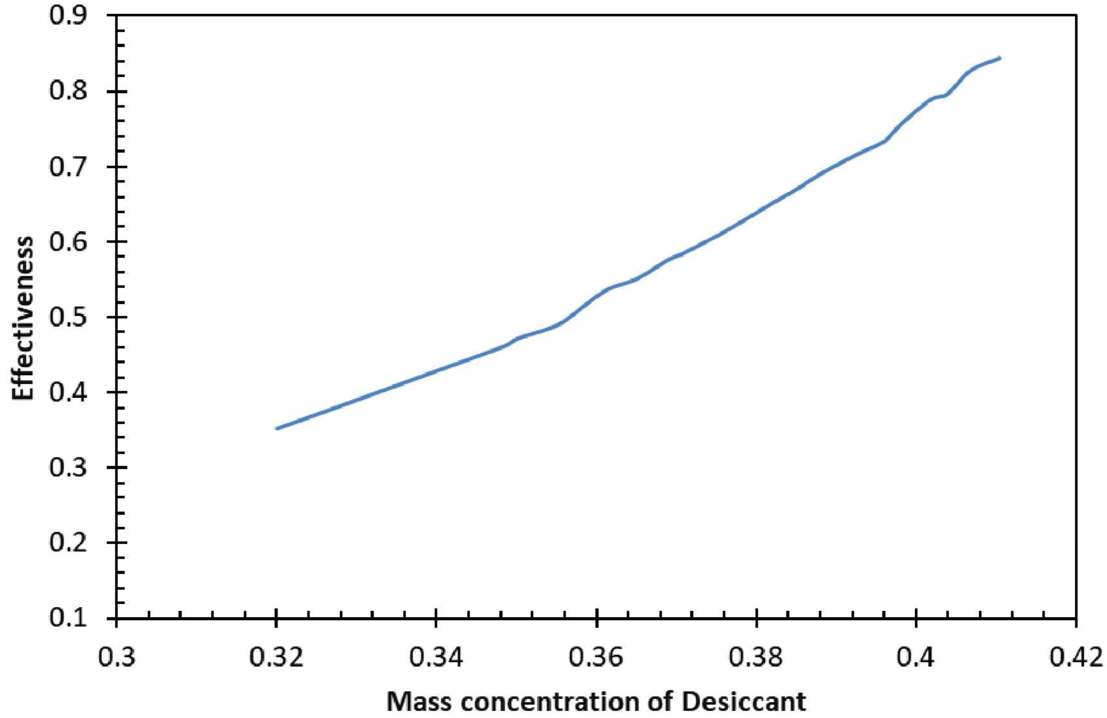



**Figure 15.** Dehumidifier Effectiveness vs Mass Concentration of Desiccant Solution

Mass concentration of LiCl in desiccant solution is directly proportional to dehumidifier effectiveness as evident from fig. 15 plot of an increasing curve of $\varepsilon_m$ vs concentration graph. The LiCl of solution decreases the vapor pressure of the solution which in turn decreases the equilibrium humidity ratio of the system but the actual humidity ratio of outlet air stream also decreases, hence the effectiveness is increased if concentration of LiCl increases. The lowest mass concentration point is 0.32 because the dehumidification of air stream is not pronounced if LiCl concentration falls below this point and crystallization occurs for LiCl solution above concentration of 0.41 at room temperature.

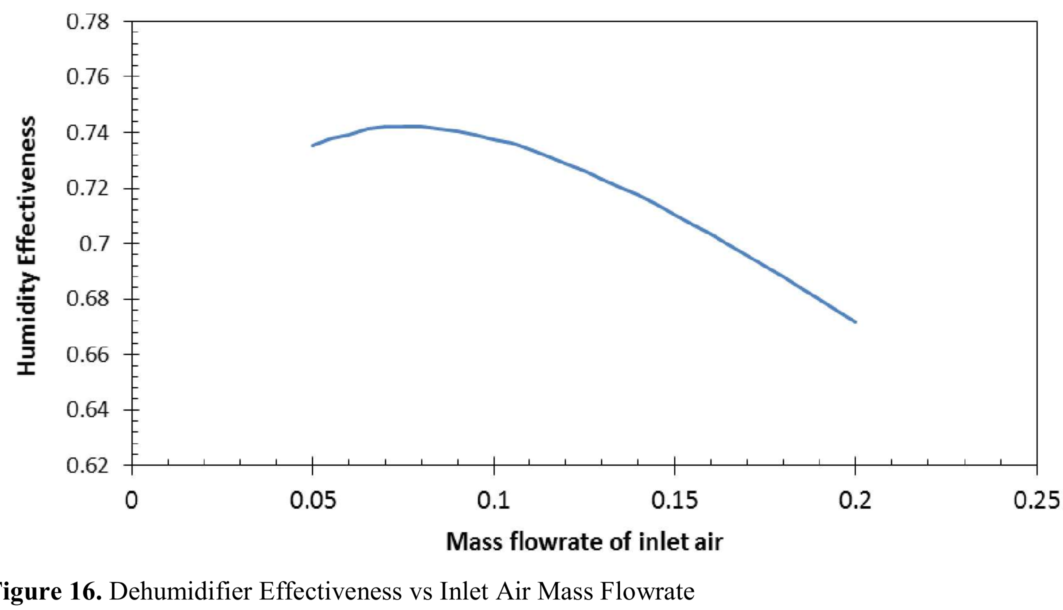



**Figure 16.** Dehumidifier Effectiveness vs Inlet Air Mass Flowrate

The plot of fig.16 shows the curve of changing dehumidification effectiveness with increasing mass flowrate of air. The graph first increases up to a maximum point (εm = 0.742189), it is the point where optimal solution to air mass ratio is reached (at about 8-8.5) then the dehumidification effectiveness decreases as lesser dehumidification is observed at higher air flowrate, even though equilibrium humidity ratio remains unchanged. The mass flowrate of air is varied between 0.05 kg/s to 2 kg/sec to make the increasing and decreasing trend of effectiveness curve apparent.

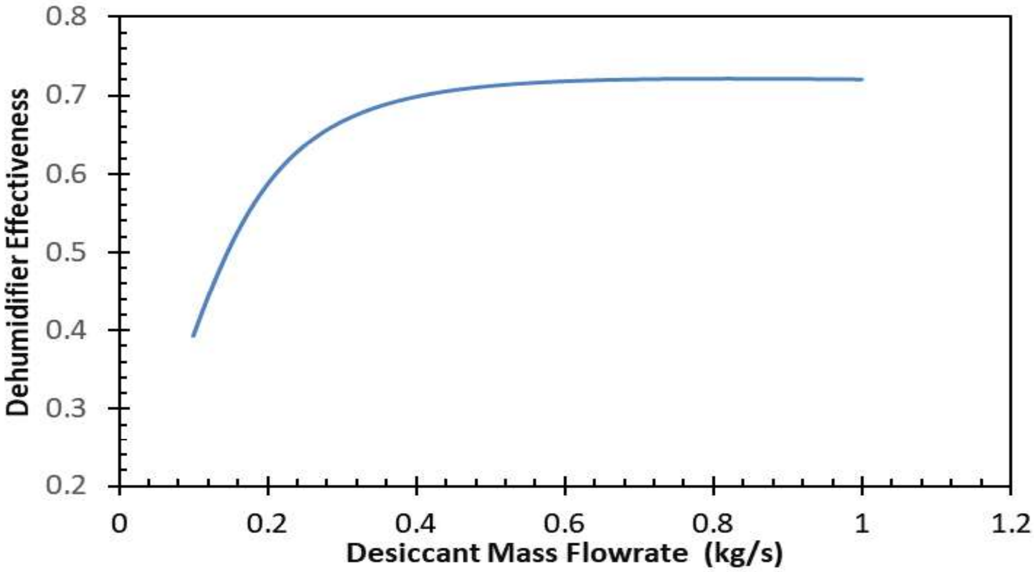



**Figure 16** .Dehumidifier Effectiveness vs Mass Flowrate of Desiccant Solution

The plot of dehumidifier effectiveness of fig. 16 shows the asymptotic curve whose maximum value for this limit is 0.722. The curve did not reach a maximum value till the end of the curve because desiccant flowrate if increased still gives slightly better dehumidification at higher flowrate even after crossing the optimal solution to air mass ratio. This however is not feasible for system design and COPH calculation. The increasing curve is expected to rise quickly from lower value of 0.1 to 0.7 kg/s which was evident from graph and after 0.7 kg/s

**Moisture Removal Rate (MRR)**

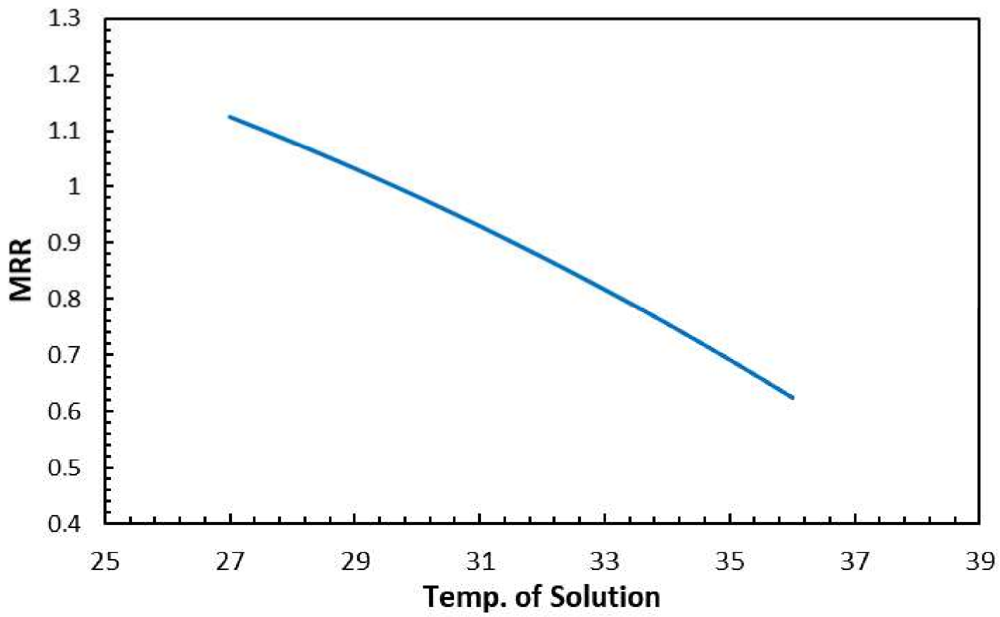
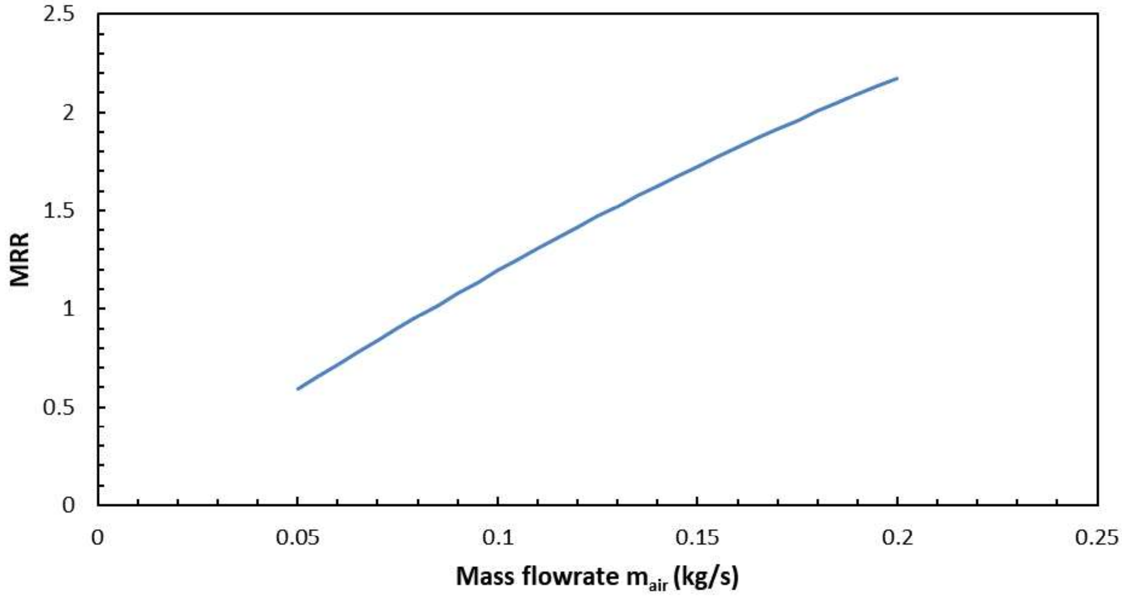



**Figure 17.** Moisture Removal Rate vs Temperature of Solution

The MRR of dehumidifier is dependent of temperature of solution as depicted in fig.17.As we have seen earlier εm and Δω both decrease with increased temperature so it is logical that MRR will also decrease. The temperature range is already justified earlier.




The mass flowrate of air is directly proportional to MRR given in equation so the trend in fig. 18 is justified. However, Δω decreases after the air flowrate optimal solution to air mass ratio but, the higher mair dominates and the trend remains increasing.

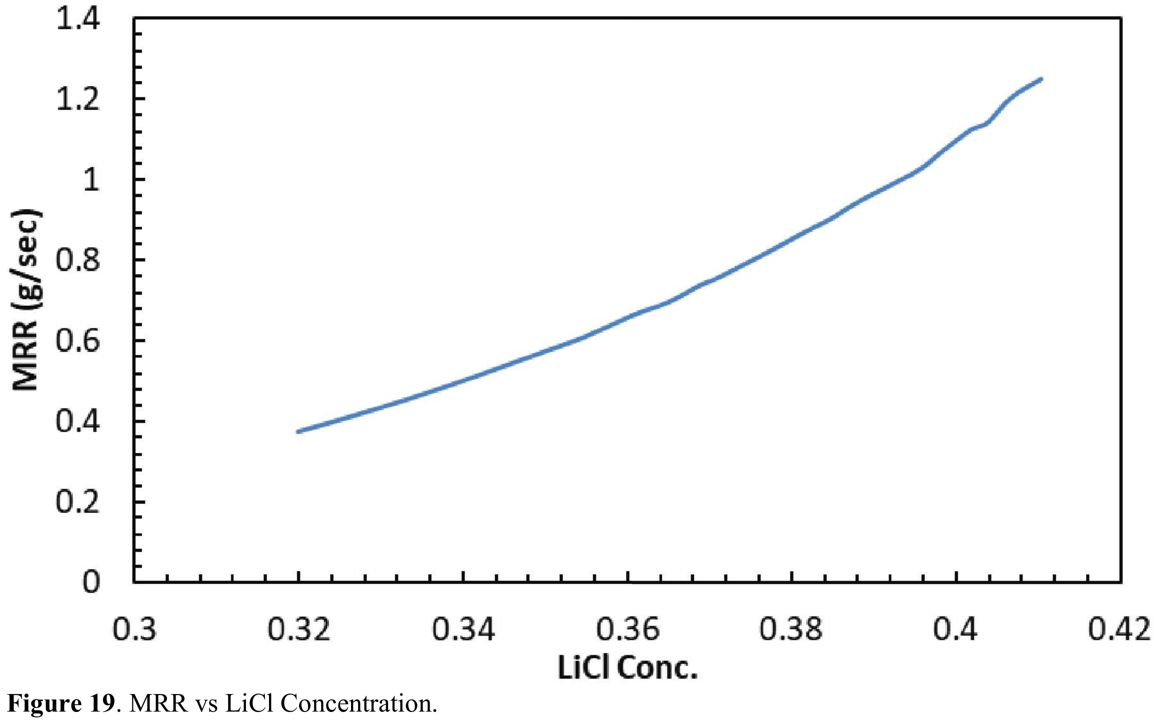



**Figure 19**. MRR vs LiCl Concentration.

Concentration of LiCl in desiccant solution causes increased Δω value which in trun significantly increases the MRR value even if $m_{air}$ value is kept constant. The plot of Fig.19 proves the previous statement true. Important point to not higher concentration can achieve higher MRR value without increased energy in transporting large amount of desiccant solution, but care must be taken no not cross the LiCl concentration above 40% of solution mass.

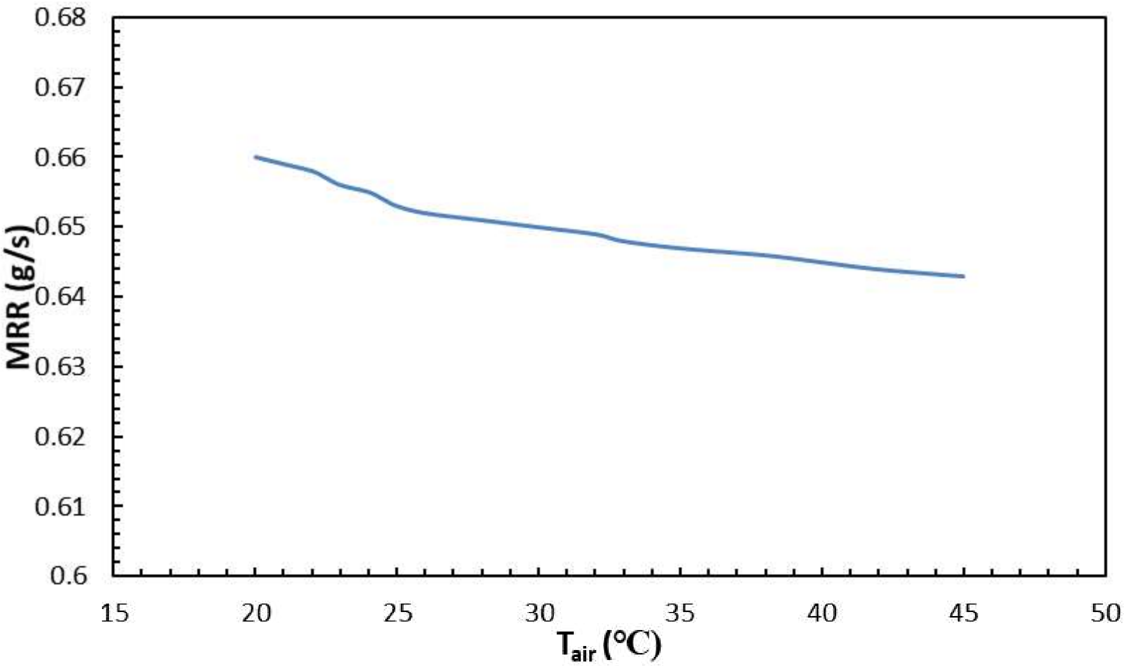



**Figure 20.** MRR vs Inlet air Temperature Tair

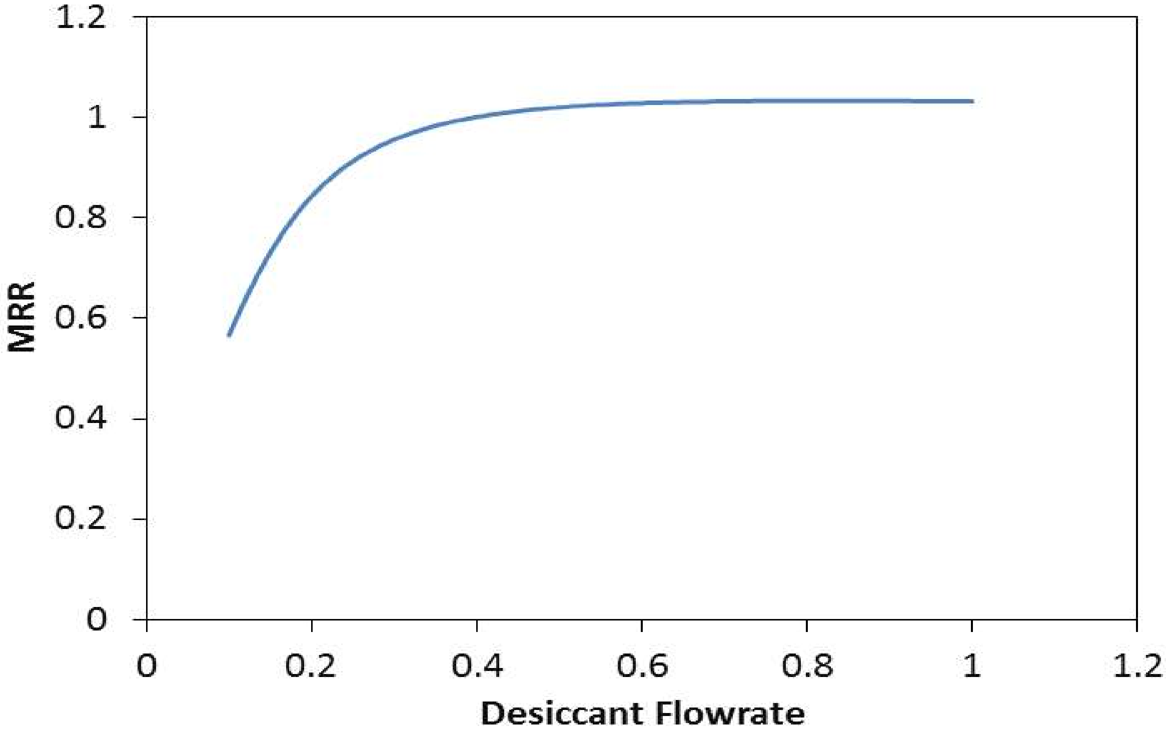



**Figure 21.** MRR vs Mass Flowrate of Desiccant Solution

Sensitivity of MRR with respect to solution flowrate is plotted in fig. 21. The graph shows an increasing trend up to 0.67 kg/s value then the slope of the curve flattens till 1kg/s. The rapid increasing trend is continued till the solution mass flow reaches the optimum solution to air mass ratio and after that the slope quickly flattens to almost a horizontal line after 0.7 kg/sec. This is explained by the very minute increase in $\Delta\omega$ value after 0.67 kg/sec and the constant $m_{air}$ value do not contribute in improving MRR significantly.

**Evaporation Rate (EVR)**

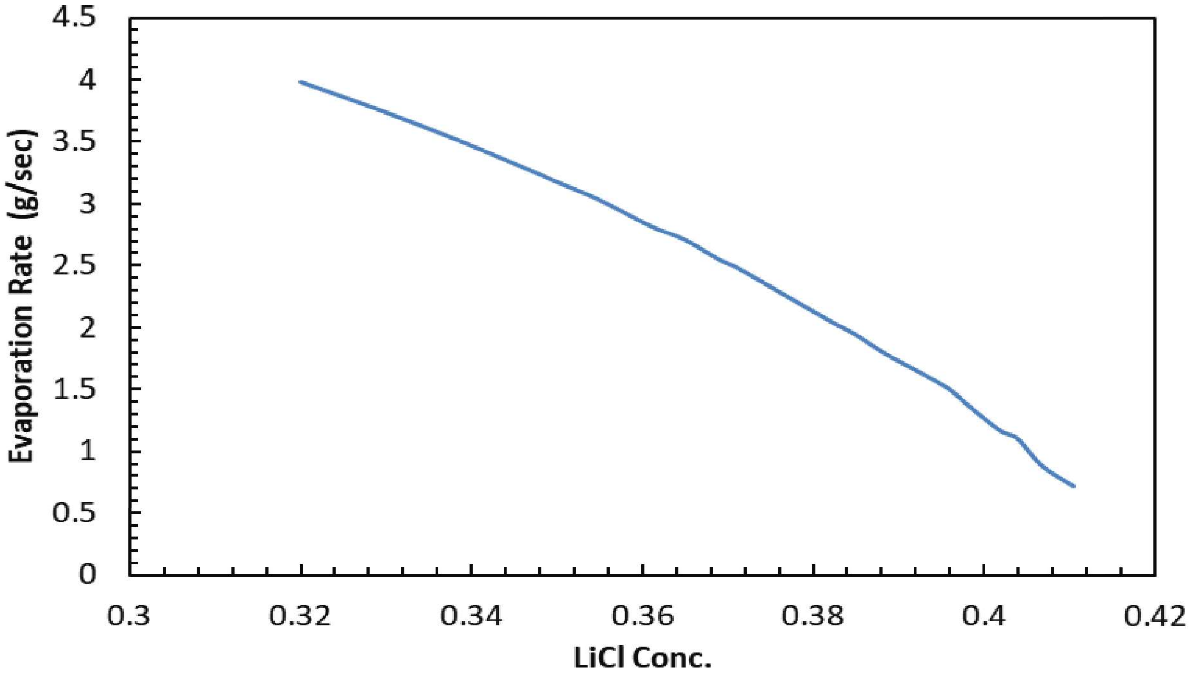



**Figure 22.** Evap. Rate vs Licl Concentration

Evaporation rate of regenerator heavily depends on LiCl concentration in solution as evident in fig. 22. As the LiCl concentration increases the dehumidifier effectiveness increases which is known (fig.10) hence regeneration with hot desiccant solution in regenerator column decreases, this is mainly because the dehumidification driving force is increased in higher concentration of solute. The range of LiCl mass concentration is 0.32 to 0.41, the reason is explained earlier.

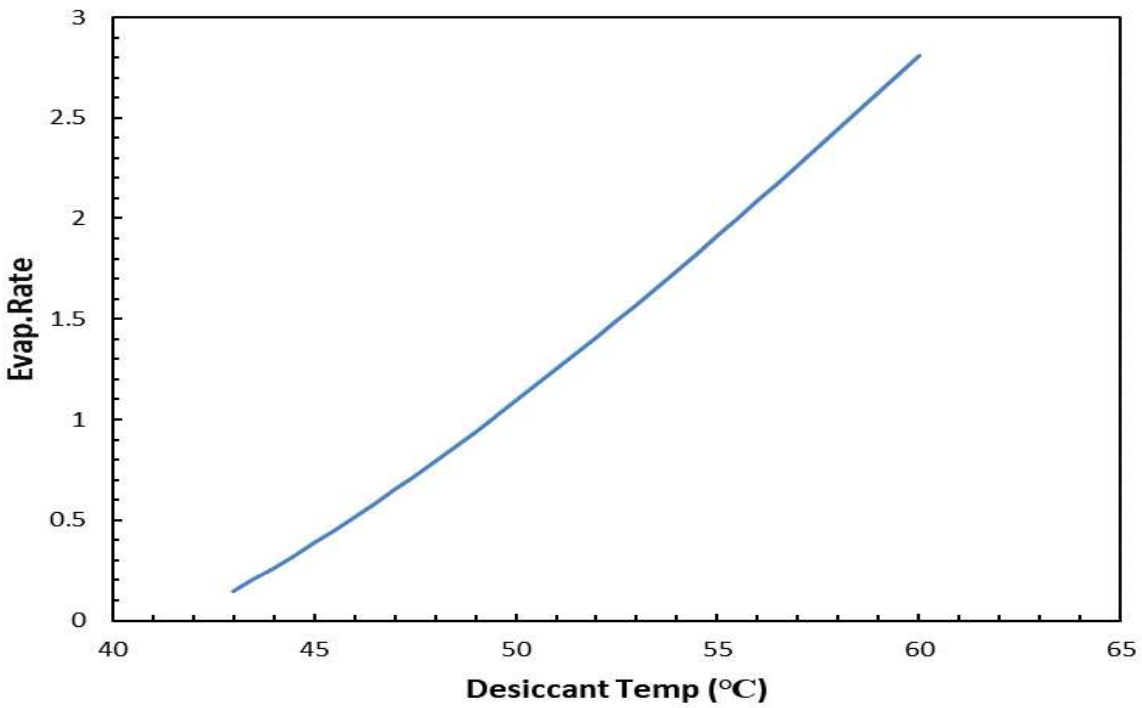



**Figure 23**. Evaporation Rate vs Desiccant Solution Temperature

EVR or evaporation rate increases significantly in fig. 23 with increasing temperature of desiccant stream fed into regenerator column. The regeneration temperature of desiccant solution is at least to be enough to produce some evaporation up to the point where it regenerates same amount of desiccant solution which is fed into dehumidifier. Hence the temperature range taken is 43°C to 60°C as below 42°C no evaporation is observed and above 60°C it is not necessary because column will evaporate too much of water from solution which is not desired.

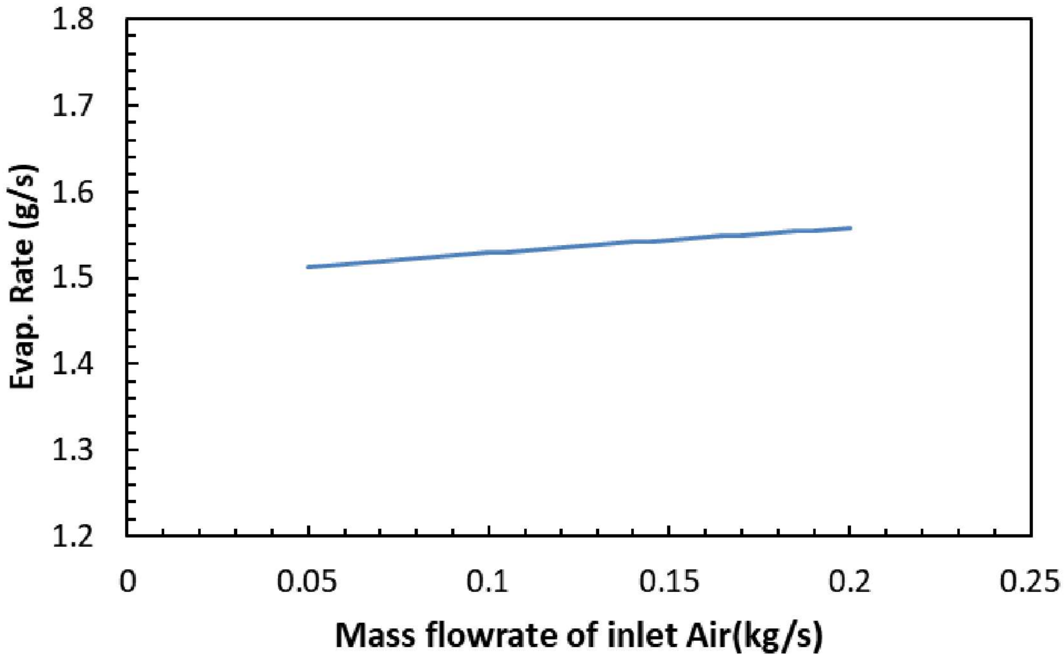



**Figure 24.** Evaporation Rate vs Mass Flowrate If Air Inlet into Dehumidifier Mair

The graph of EVR with respect to changing mair affects very little if mair is increased even four times as seen in fig. 24 but, EVR slightly increases by about 0.04 gm/sec which is noticeable only numerically, physically it has no meaning though. The mair has no direct relation with EVR with (neither mass nor heat transfer effects remain with desiccant when it reaches regenerator column), other than concentration change of solution which is not very pronounced to show effects in regeneration process.

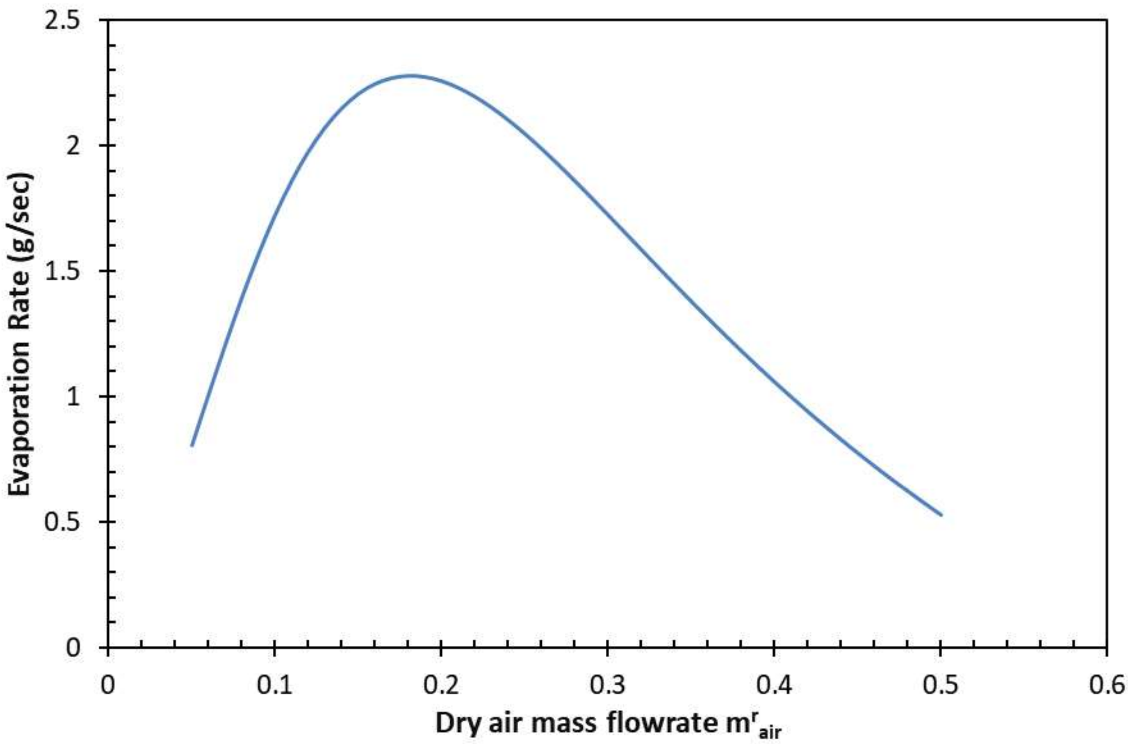



**Figure 2.** Evaporation Rate vs Mass Flowrate of Air into Regenerator $m^r_{air}$

The mass flowrate of air stream into regenerator column at ambient conditions affect the evaporation rate drastically as shown in fig. 25. The curve shows a skewed bell shape with maxima being at 0.18 kg/sec with EVR value being 2.276 g/s after which the EVR shows negative slope. This behaviour can be explained in the same way as dehumidifier which is the optimum solution to air mass point. At this optimum ratio the maximum change in humidity Δωr in air stream is observed (point of maximum mass transfer). After the air flowrate increases beyond 0.18 kg/s we get smaller Δωr value and hence smaller EVR is obtained. The drop in Δωr value is more severe than drop in Δω value for dehumidifier (in case of MRR) which dominated the increasing mrair value hence ultimately giving lesser EVR. The slope of increasing trend is much higher than decreasing trend is because of increasing mrair value comes into play while calculating EVR.

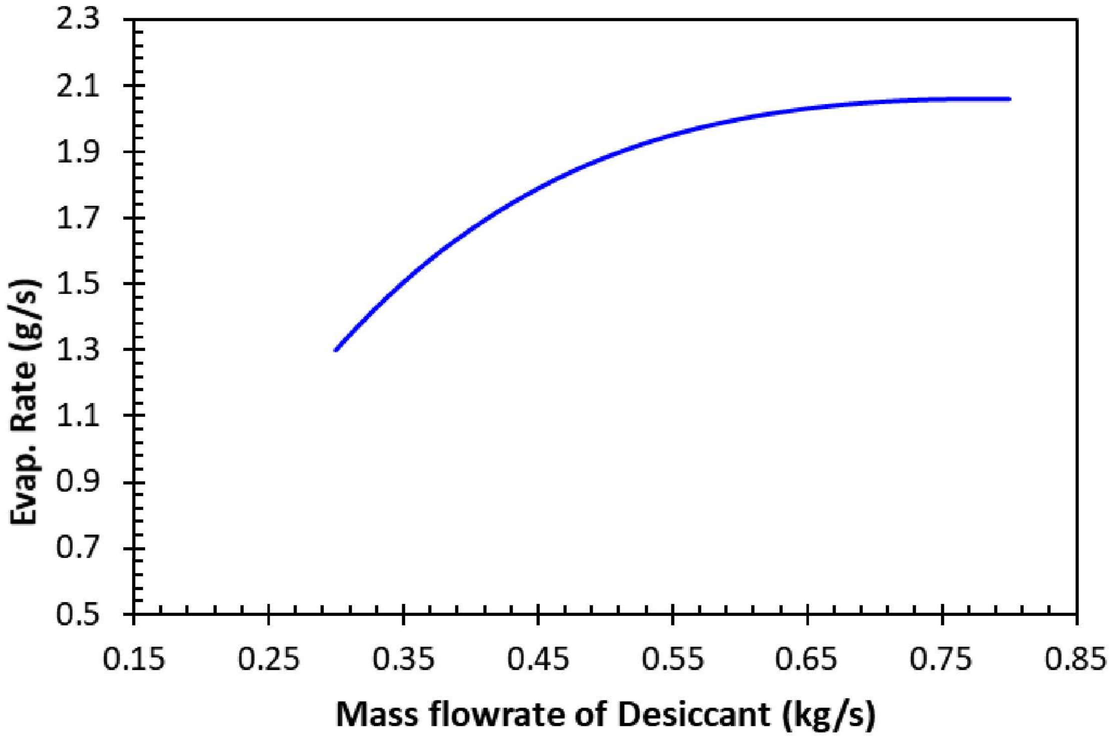



**Figure 26.** Evaporation Rate vs Desiccant Mass Flowrate in Desiccant Cycle

Unlike air flowrate, desiccant solution flowrate only increases the evaporation rate. In fig. 26 we can see the trend; however, the curve flattens at solution mass flowrate higher than 0.65 kg/s. The EVR increases very slowly after this point but it does keep increasing. The mass flowrate of desiccant is varied from 0.3 to 0.85 kg/s which covers the possible behaviour of the curve that it can display.

In regeneration process the best or optimum solution to air mass ratio is found out to be 3.9 from the last two EVR graphs and for dehumidification best solution to air mass ratio is about 8.

- **Conclusions**

This work is focussed on developing a working liquid desiccant dehumidification column and a solution regeneration column using ASPEN Plus software. The model was intensively developed by all the literatures of liquid dehumidification system which and selecting best fitting set of equations for designing the packed bed columns. The built-in packed bed absorber column in ASPEN had to modified in rating-based calculation method, by all the mass transfer correlation equations, heat transfer correlations equations and interfacial area correlation equations of packed bed from best sources Chung Koronaki (1996). The developed model was then validated with experimental data from the same source to check its accuracy in modelling which came out with acceptable error margin of 5-7%. The validation was again done with other experimental data also Fumo [2004] to verify the feasibility of model. After the satisfactory validation with different experimental data the dehumidification and regeneration columns were joined to form a cyclic process and integrated with VCACs system to get a working hybrid system.

The hybrid model once giving desired results was subjected to parametric analysis of four performance indices (i) COPH, (ii) dehumidifier effectiveness εm, (iii) MRR and (iv) EVR with design parameters e.g., Desiccant solution Temperature (Td), air mass flow rate (mair) and (mrair), Desiccant Concentration (Xin) to dehumidifier and regenerator, Air Temperature (Tair) and Desiccant flowrate (mdes). After rigorous parametric analysis of these four performance indices some expected and newer results were observed.

Parametric analysis also revealed the desiccant solution to air mass ratio for regeneration purposes is about 3.7-4.0 by EVR sensitivity because beyond this point evaporation rate and Δωr decreases quickly. So, for a fixed solution flowrate in Dehumidification-Regeneration cycle the air mass flowrate into regeneration should be about twice the amount of air mass flowrate into dehumidifier column.

- **Declaration of Competing Interest**

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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