Engineering Research Journal 116 (April 2008) M1-M28

EXPERIMENTAL STUDY FOR COMPACT LIQUID DESICCANT DEHUMIDIFIER/REGENERATOR SYSTEM

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Abstract

Liquid desiccant systems find many applications in a very large variety of industrial and daily usage products including the new HVAC systems. The present study introduces a new design for compact uni-shell dehumidifier/ regenerator liquid desiccant system. The latest trends in this area suggest that hybrid systems are of current interest to HVAC industry not only for high latent heat load applications but also for improving indoor air quality. The present study aim to evaluate the performance of the proposed liquid desiccant air dehumidifier system. The performance parameters for the air dehumidifier are the reduction ratio of air humidity ratio and the dehumidifier effectiveness.

Several experiments were carried out under the following operating conditions; cooling water inlet temperature (10^oC - 18^oC), desiccant solution inlet temperature (26^oC- 33^oC), air flow rate (3.4 l/s - 6 l/s), air inlet temperature (38^oC- 51^oC), air inlet humidity ratio (21-25g_w/kg_{da}), desiccant solution mass flow rate (0.04 kg/s - 0.13 kg/s), desiccant solution to air mass flow rate ratio (10-26), heating water temperature (42^oC-51^oC), and desiccant solution concentration (10% to 45%).

Experimental setup was designed and constructed to achieve the operating conditions mentioned before. The experimental results showed that both the reduction ratio of air humidity ratio and effectiveness of the dehumidifier increase as desiccant solution mass flow rate, desiccant solution to air mass flow rate ratio, the desiccant concentration, and the heating water temperature increase. Also, both the reduction ratio of air humidity ratio and effectiveness of the dehumidifier decrease when the cooling water inlet temperature, the desiccant solution temperature and the air flow rate increase. The performance parameters were almost unaffected with the inlet air temperature. The dehumidifier reduction ratio of air humidity ratio slightly increases while its effectiveness remains constant when the air inlet humidity ratio increases. Empirical correlations based on the present experimental results for the performance parameters in terms of the different operating conditions were developed.

Key words: liquid desiccant–dehumidifier–regenerator–heat and mass transfer.

1. Introduction

Dehumidification is the process of water vapor removal from moist air. It can be achieved by either increasing the pressure of air or by absorption/adsorption of moisture by a solid/liquid material called a desiccant. Desiccants are materials that have high affinity for water vapor. The removal of moisture from the air depends on the difference in water vapor pressure present in the moist air and that hold by the desiccant. Sorption always generates heat that includes the latent heat of water vapor sorbed by the desiccant and an additional heat of sorption varying between 5% and 25% of the latent heat [1]. Absorption leads to a chemical change of the desiccant, while adsorption takes place on the surface of the desiccant without any chemical change. Most of the absorbents are liquids, while more of the adsorbents are solids, [1]. Both solid and liquid desiccants are extensively used for dehumidification and cooling. Some of the merits of liquid desiccant systems include improved indoor air quality, acting as disinfectant, single regenerator for multiple conditioners and flexibility in its location. But, common problems involve carryover of solutions into air stream, crystallization of salts and corrosion. Liquid desiccant cooling systems have been proposed to replace the conventional vapor compression cooling systems in hot and humid areas.

Many researchers [2-5] have shown that the liquid desiccant cooling systems can reduce the overall energy consumption as well as shift the energy use away from the electricity and toward renewable energy such as solar energy and waste heat since Lof [6] developed the first liquid desiccant system.

Recently Fumo and Goswami [7] gave a brief summary on the liquid desiccant systems, and Younus et al. [8] presented a thermodynamic analysis of different liquid desiccants. Dai and Zhang [9] investigated numerically heat and mass transfer processes in a cross flow liquid desiccant dehumidifier in which wet durable honeycomb paper constituting the packing material. They developed a mathematical model able to determine the heat and mass transfer coefficients between the air and falling film of liquid desiccant. Jain and Bansal [10] presented an overview of liquid desiccant technology along with a complication of experimental performance data of liquid desiccant dehumidifiers, empirical dehumidification effects and mass transfer correlations in a useful and easy to read in a tabular format. The latest trends in this area suggest that hybrid systems are of current interest to HVAC industry.

A simple model for the preliminary design of an air dehumidification process occurring in a packed bed using liquid desiccant through dimensionless vapor pressure and temperature ratios is developed by Gandhidasan [11]. Elsayed et al. [12] used a finite difference model to calculate the effectiveness of heat and mass transfer in packed beds during air dehumidification and regeneration of liquid desiccant solution, rasching rings of 25 mm are used as packing materials in beds. Radhwan et al. [13] used one dimensional model to simulate the process

occurring in a counter flow air calcium chloride desiccant packed bed dehumidifier and predict the performance of the bed at different air and liquid desiccant inlet conditions, air and liquid flow rates and bed length. Lui et al. [14] presented analytical solutions of the air and desiccant parameters inside a parallel flow, counter flow and cross flow dehumidifier/regenerator under some reasonable assumptions based on the available heat and mass transfer models. Good agreement of these solutions with both available numerical and experimental data was observed. Linear approximation was made to find out the dependence of equilibrium humidity ratio on solution temperature in a simplified analysis of packed bed liquid desiccant dehumidifier/regenerator was proposed by Chengqin et al. [15]. New parameters were defined and original equations were rearranged to obtain two coupled ordinary differential equations. Also, Chengqin et al. [16] presented a theoretical study on the analysis of adiabatic liquid desiccant dehumidification/regeneration process with slug flow assumption. They developed a controlling equation for the quasi-equilibrium processes where the two fluid streams are in contact in quasi-equilibrium conditions. Results from this equation with numerical integration for the solution are presented as a process curves on a psychrometeric chart. Two of these curves are found to be characteristic of typical types of adiabatic dehumidification/regeneration process. One for small enthalpy change of air and low mass flow rate of solution and the other with small concentration change at high mass flow rate of solution.
Pietruschka et al. [17] presented new desiccant cooling cycles to be integrated in residential mechanical ventilation systems. The process shifts the air treatment completely to the return air side, so that the supply air can be cooled by a heat exchanger. Purely sensible cooling is an essential requirement for residential buildings with no maintenance guarantee for supply air dehumidifiers.
This paper introduces an experimental investigation to evaluate the performance of compact uni-shell liquid desiccant dehumidifier/regenerator system under different operating conditions.

2. Experimental setup, instrumentation, procedures and data analysis

An experimental investigation of the dehumidifier-regenerator liquid desiccant uni-shell system performance was carried out. For this purpose, a test apparatus was designed and constructed.

The Experimental set up is shown schematically in Fig. (1). The apparatus consists mainly of the following sub-systems;

1- Uni-shell block.
2- Cooling water circuit.
3- Heating water circuit.
4- Process air supply.

2.1 Uni-shell block

The uni-shell block includes dehumidifier-heat exchanger-regenerator uni-shell in which cold water pipe block is placed in the dehumidifier section, hot water pipe block is placed in the regenerator section and the heat exchanger tubes are placed in the heat exchanger section. Process air is supplied to the dehumidifier by an air distributor whereas the regenerator is supplied by atmospheric air through another similar air distributor. Calcium chloride solution was utilized to be the liquid desiccant with different concentrations which absorbs the water vapor from process air in dehumidifier side and rejects it in regeneration side. The dehumidifier-heat exchanger-regenerator uni-shell consists of a cylinder with 700 mm inner diameter, 3 mm thickness and 1200 mm length. This cylinder contains three partitions and two flanges were welded in two sides, one as inner flange and another as outer flange. Figure (2) shows the details of uni-shell. The cylinder (uni-shell) was drilled in two locations with diameter of 50 mm, one in the dehumidifier zone in middle of the upper side, the second in the regenerator zone in middle of the upper side also. The two tubes having 50 mm outer diameter and 100 mm length were welded at these two holes for exit air from both dehumidifier and regenerator.

Two PVC Tee connection were welded at both tube exits from both dehumidifier and regenerator, the horizontal T-branch was welded with the air duct while the vertical T-branch closed by threaded cap which used to fill uni-shell with desiccant solution.

The air duct, manufactured from a 50 mm diameter PVC tube, contains two measurement sections to measure both dry and wet bulb temperatures at the exit from each dehumidifier and regenerator.

The outer fixed flange which welded in one end of cylinder was fabricated from steel with 3 mm thickness and 760 mm outer diameter, 700 mm inner diameter. The two plates (685x45 mm) were welded in the two partitions as illustrated in Fig. (3). The outer fixed flange and the two plates were drilled, the diameter of each hole is 9 mm and the distance between the center of any hole and the center of the next hole is 50 mm. M8 bolts with 20 mm length were welded in holes where the bolt head in the inside direction and the bolt thread in the outside direction.

The inner fixed flange which is welded in the other end of cylinder was fabricated from steel with 3 mm thickness and 700 mm outer diameter, 640 mm inner diameter. One plate have dimension of (640x250 mm) was welded in the three partitions as depicted in Fig. (4). The inner fixed flange and the plate were drilled, the diameter of each hole 9 mm and the distance between the center of any hole and the center of the next hole is 50 mm.

For both dehumidifier and regenerator, air charged in desiccant solution as bubbles by two stainless steel tubes, (air distributors), each tube has 60 mm outer diameter and 1200 mm length. The upper surface has several rectangular notches as groups. The two stainless steel tubes are four burner tubes; each burner tube has 60 mm outer diameter and 600 mm length. Both two burner tubes were welded together to produce the air distributor.

Two movable front flanges were made from a circular steel plate with 3 mm thickness and 760 mm diameter, from vertical center line with 82 mm in both left and right sides of the circular plate was cut to create the identical two pieces as shown in Fig. (5) Each piece was drilled at circumference, the diameter of each hole 9 mm and the distance between the center of any hole and the center of the next hole is 50 mm.

Also, two movable back flanges were made from circular steel plate with 3 mm thickness and 694 mm diameter, from vertical center line with 105 mm in both left and right sides of the circular plate was cut to create the identical two pieces as illustrated in Fig. (6). Each piece was drilled at circumference, the diameter of each hole is 9 mm and the distance between the center of any hole and the center of the next hole is 50 mm. M8 bolts with 20 mm length were welded in holes where the bolt head in the inside direction and the bolt thread in the outside direction.

Both movable front and back flanges were drilled. The holes were arranged in a staggered configuration, as shown in Fig. (7), the hole diameter 16.5 mm and the horizontal and vertical pitches are 40 mm.

A block of twenty copper tubes with 1220 mm length, 16 mm outer diameter and 1 mm thickness were welded in each front and back movable flanges. This arrangement was used for cooling the desiccant solution in the dehumidifier part by passing cold water inside the copper tubes. Another similar block of copper tubes were also welded in each front and back movable flanges. This arrangement was used for heating the desiccant solution in the regenerator part by passing hot water inside the copper tubes. Four water boxes were made from steel with 3 mm thickness as shown in Fig. (8), these boxes collect water from each inlet and outlet sides for both cold and hot water

The heat exchanger occupied the space between the regenerator and the dehumidifier. Heat exchanger was made of copper tubes with 12 mm outer diameter and 1 mm thickness. The diluted solution is pumped from dehumidifier top to the heat exchanger left side and then heated by concentrated solution which passes from regenerator top around heat exchanger tubes and then flows to the bottom of the dehumidifier through a tube; see Fig. (1).

2.2 Cooling water circuit

The cooling water circuit is depicted in Fig. (1), where the cold water was used to cool the desiccant solution and the process air. The cold water temperature ranged from 10^{o}C to 30^{o}C. Therefore, to reach a specified temperature for cold water, a cooling water circuit was installed. The main components of the cooling circuit are the refrigerating machine, circulating pump and cold water tank.

The refrigeration machine is 2 HP cooling capacity, but the evaporator was replaced by copper tube coil with outer diameter of 500 mm and 850 mm height, and the outer copper tube diameter is 16 mm.

A centrifugal pump of 0.5 HP and maximum flow rate of 28 l/min was used to circulate the cold water through cooling water circuit which includes the cold water tank, the cold water pipes of the dehumidifier and cold water tank.

The cold water tank has the dimensions of 560 mm diameter and 980 mm height. This tank was supplied with fresh water which comes from the water supply at its top end. The copper tube coil was placed concentrically in the tank. At 50 mm from the tank bottom, steel pipe was welded to the tank from one end; the other was connected to the circulating pump. At the cross section of the tank and the steel pipe connection, a strainer was placed to prevent impurities to pass through the steel pipe.

The operation of the refrigerating machine was carried out both manually and automatically. There is a thermostat which has a resolution of 0.1^{o}C is connected to the outlet pipe of the cold water tank as shown in Fig. (1), the temperature in the circuit can be remained constant within 1^{o}C. In water tubes, there was a hole where a thermocouple was inserted through it for measuring the circulating water temperature at inlet and outlet.

2.3 Heating water circuit

The heating water circuit is illustrated in Fig. (1), where the hot water was used for heating the desiccant solution and the regeneration air. The hot water temperature ranged from 35^{o}C to 60^{o}C. Therefore, to reach a specified temperature range for hot water, a heating water circuit was installed.

The main components of the heating circuit are water heating tank and circulating pump. The water heating tank has 420 mm outer diameter, and 800 mm height. At the bottom of the tank, a steel pipe was welded. This pipe was connected to the circulating pump. At the top of the tank, another steel pipe was welded. This pipe was attached to the regenerator intake. The tank surface was drilled at two locations for two electric heating resistances. The two electric heating resistances were connected in parallel, one of 1.2 kW and another 2 kW capacities. The operation of the electric heating resistances was carried out both manually and automatically. There is a thermostat which has a resolution of 0.1^{o}C is connected to the outlet pipe of the heating tank as illustrated in Fig. (1). The temperature in the circuit can be hold constant within 1^{o}C, there was a hole

where a thermocouple was inserted through it for measuring circulating water temperature at inlet and outlet.
A similar pump to that used in the cooling water circuit was used to circulate the hot water through the hot water tank, the heating tank, the hot water pipes of the regenerator and finally to the hot water tank again.

2.4 Air supply system

Air was supplied to the dehumidifier at variable rates, temperatures, and humidities from an air supply system which consists of a centrifugal blower, air heater, humidifier and air distributor. The process air supply was designed to provide air flow rater ranged from 3.4 to 6 l/s at conditions within the envelope bounded by (28 – 33) ^{o}C wet-bulb temperature, (21 - 25) g_w/kg_{da} humidity ratio, and (38 – 51) ^{o}C dry-bulb temperature. To simulate the latent and sensible heat loads on the conditioner unit, provisions were made to artificially create the required humidity and temperature level of the process inlet air in an open loop configuration, i.e., the air exiting the test unit was being discharged to the atmosphere.

2.5 Instrumentation and control

Six readymade chromel-alumel thermocouples of 2 mm diameter to measure the inlet and outlet temperature of cold water, hot water and desiccant solution inlet and outlet of the heat exchanger are used. Also, four pairs of thermocouples were used to measure the dry and wet bulb temperature of air inlet and outlet for both dehumidifier and regenerator. A wet cotton wick was provided for the wet bulb temperature thermocouple. Relative humidity of ambient air in the laboratory was measured by the hygrometer fixed near the air blower. These temperature readings have an accuracy of 0.1^{o}C.
The flow rates of the circulating cooling water, the circulating heating water and the circulating desiccant solution were measured by three calibrated orifice meter. Also, the flow rates of the process air for both the dehumidifier and the regenerator were measured by two orifice meters with the same design of liquid flow rate orifice meter.
To obtain the required level of process air dry bulb temperature, one heater with proportional temperature controller were installed in the process air duct; humidity ratio could be controlled by the combination of moisture from steam generator spray into the air supply duct.
The calcium chloride solution level reading is determined from level indicator scale which is attached to the level indicator at the dehumidifier and the regenerator and this reading was remaining constant for all test runs.

2.6 Experimental Work Procedures:

The steps of preparation and charging of the setup were conducted after check all thermocouples readings, check all connections and test the pumps, the air blower and the refrigeration machine. These steps are as follows; before starting the actual experiments, the uni-shell system was thoroughly cleaned by repeated washing with distilled water, after the system had been perfectly dried, the Ca Cl_2 solution with any prescribed concentration was charged.

In order to implement any experimental run, the following procedures are followed; the hot water thermostat is adjusted at desirable temperature and the two heaters in the hot water tank are switched on, the cold water thermostat is adjusted at desirable temperature and the refrigerating machine is switched on, after the temperatures in both hot and cold water temperatures reach the adjustable temperatures, both circulating pumps of cold water and heating water circuits are switched on, the flow rate for each cold and hot water is adjusted by the valve, the desiccant solution pump is also switched on, then the air blower is switched on and the flow rates for both dehumidifier and regenerator is adjusted, finally, the reading are then taken

2.7 Experimental Data Analysis:

Humidity ratio reduction in dehumidifier was calculated as follows:

$$\Delta W = W_i - W_o \tag{1}$$

Equilibrium condition between air and desiccant solution:

Using data from ref. [24] for Ca Cl_2, the following correlation is developed for P_{wz}

$$\frac{P_{wz}}{P_{ws}} = 1.146 - 1.76x \tag{2}$$

for $0.2 \le x \le 0.55$ and $0 \le T_s \le 80\ ^{o}C$

The value of P_{ws} is predicted in terms of temperature using the following correlation

$$P_{ws} = 0.0159 T_s^{\,1.6438} \tag{3}$$

The air equilibrium humidity ratio then becomes:

$$W_e = 0.62185 \frac{P_{wz}}{P - P_{wz}} \tag{4}$$

Where, P is the total pressure of air on the desiccant solution (101.325 KPa).

Dehumidifier effectiveness was calculated from the following relation:

$$\varepsilon = \frac{W_i - W_o}{W_i - W_e} \tag{5}$$

3. Experimental results and discussions

Various parameters namely; cooling water inlet temperature, desiccant solution inlet temperature, air inlet temperature, air inlet humidity ratio, air flow rate, desiccant solution mass flow rate, heating water inlet temperature, desiccant solution to air mass flow rate ratio and desiccant concentration. Surely influence the dehumidifier performance is represented by humidity ratio reduction and the dehumidifier effectiveness. The effect of each parameter is discussed in the following sections:

3.1 Effect of cooling water temperature

The effect of cooling water temperature on the dehumidifier air humidity ratio reduction and effectiveness is illustrated in Figs. (9) and (10) respectively. It is noticed that the dehumidifier air humidity ratio reduction and dehumidifier effectiveness decrease remarkably as cooling water inlet temperature increases. This trend may be due to the increase in the desiccant solution temperature. Which causes the surface vapor pressure of the desiccant solution to increase as a result, the average water vapor pressure difference between the air and desiccant solution in the dehumidifier decrease, this leads higher air outlet humidity ratio thereby, the humidity reduction and dehumidifier effectiveness decrease when the cooling water inlet temperature increases.

3.2 Effect of desiccant solution temperature

The influence of desiccant solution temperature on the dehumidifier air humidity ratio reduction and effectiveness is depicted in Figs. (11) and (12). It is observed from these figures that the air humidity ratio reduction and dehumidifier effectiveness are inversely proportional to the desiccant solution inlet temperature. It may be explained as follow: increasing the solution temperature increases, the surface vapor pressure of the desiccant increased and so, the average water vapor pressure difference between the air and desiccant in the dehumidifier decreases, which led to higher air outlet humidity ratio and, hence, the humidity reduction and dehumidifier effectiveness finally decreased with the desiccant temperature increase.

3.3 Effect of atmospheric conditions

3.3.1 Air inlet temperature

Figures (13) and (14) illustrate the effect of air inlet temperature on dehumidifier air humidity ratio reduction and effectiveness. It can be seen from the figures that the dehumidifier air humidity ratio reduction is affected slightly by the air inlet temperature. Increasing in the air inlet temperature may increase the sensible heat transfer rate thereby increasing the solution temperature. This results in an increase in the vapor pressure of the desiccant solution and hence lower mass transfer rate. In fact, at higher air inlet temperatures, the sensible heat transfer and the resulting increase in solution vapor pressure may cause mass transfer to take place in the opposite direction, i. e., from desiccant solution to air stream. Lower air inlet temperatures are preferred to achieve high dehumidification rates. However, in the present experiments, the desiccant flow rate and, hence, its thermal capacity was so that the mentioned effect was fairly weak. Therefore, the dehumidifier air humidity reduction was almost unaffected by the air inlet temperature.

Also, it can be seen from the figures that the dehumidifier effectiveness is affected slightly by the air inlet temperature. Therefore, the dehumidifier effectiveness were almost unaffected by the air inlet temperature.

3.3.2 Air inlet humidity ratio:

The effect of the air inlet humidity ratio on the humidity ratio reduction and dehumidifier effectiveness is illustrated in Figs. (15) and (16), respectively. The mentioned figures show that the humidifier air humidity ratio reduction and increases with inlet air humidity ratio increases. The effect on the air humidity ratio reduction was caused by increased average water vapor pressure difference between the air and the desiccant solution with increasing air inlet humidity ratio. The mentioned figures show that the dehumidifier effectiveness is almost unchanged with inlet air humidity ratio. The effect on the dehumidifier effectiveness lay in increasing the air inlet humidity ratio.

3.4 Effect of air flow rate

The effect of air flow rate on the dehumidifier humidity ratio reduction and dehumidifier effectiveness is illustrated in Figs. (17) and (18). It is noticed from these figures that, the air humidity ratio reduction and dehumidifier effectiveness decrease with increasing air flow rate. The effects can be explained as follows: when the air flow rate increases, the outlet air humidity ratio and dehumidifier effectiveness decreases due to the reduced residence time for the air in the dehumidifier.

3.5 Effect of desiccant solution mass flow rate

The effect of desiccant solution mass flow rate on the dehumidifier air humidity ratio reduction and dehumidifier effectiveness is illustrated in Figs. (19) and (20). It is noticed from these figures that, the air humidity ratio reduction and dehumidifier effectiveness increases with increasing desiccant solution mass flow rate. The effects can be explained as follows: with increasing the desiccant solution mass flow rate, the variation of the desiccant solution concentration through the dehumidifier decreases and the variation of the surface vapor pressure decreases, and hence increasing the average water vapor pressure difference between the desiccant solution and the air in the dehumidifier.

Increasing the desiccant solution flow rate also increases the mass transfer coefficient between the desiccant solution and the air in the dehumidifier.

3.6 Effect of desiccant to air mass flow rate ratio (L/G)

The effect of desiccant solution to air mass flow rate ratio on the dehumidifier air humidity reduction ratio and dehumidifier effectiveness is illustrated in Figs. (12) and (13). It is noticed from these figures that, the air humidity reduction ratio increases with increasing desiccant solution to air mass flow rate ratio. The effects can be explained as follows: with the desiccant solution flow rate increasing, the variation of the desiccant solution concentration through the dehumidifier decreases and the variation of the surface vapor pressure decreases, and hence increasing the average water vapor pressure difference between the desiccant solution and the air in the dehumidifier. Increasing the desiccant solution flow rate also increased the mass transfer coefficient between the desiccant solution and the air in the dehumidifier. By increasing desiccant solution to air mass flow rate ratio leads to an increase of the air humidity reduction ratio and dehumidifier effectiveness

3.7 Effect of desiccant solution concentration

The effect of desiccant solution concentration on the dehumidifier air humidity ratio reduction and dehumidifier effectiveness is illustrated in the Figs. (9) to (12) and Figs. (17, 18, 21, 22). The figures indicate that the air humidity ratio reduction and dehumidifier effectiveness increase with the increase of desiccant solution concentration. That is, as the desiccant solution concentration increases, the ability of the desiccant solution to absorb moisture increases. This is because as he desiccant solution concentration increases, the vapor pressure of the desiccant solution decreases and therefore higher driving force between the phases for mass transfer results. So, higher desiccant solution concentrations favor increased dehumidification.

4. Validations of the experimental results

To validate the present experimental results, comparisons of these results are made with those available in the published literature. The effect of desiccant solution mass flow rate on the air humidity ratio reduction is illustrated in Fig. (23). It is seen from the figure that the air humidity ratio reduction increases with the increasing the desiccant solution mass flow rate. The results of Liu et al. [18] and Khan et al. [19] are represented in this diagram for comparison. The trend of the present results is in agreement with that of [18], [19]. Also, it is observed that the slope of the present results trend is higher than that of previous results of [18], [19] which mean higher performance of the present design.

The effect of desiccant solution mass flow rate on the dehumidifier effectiveness is illustrated in Fig. (24). It is seen from the present results that the dehumidifier effectiveness increases with the increasing the desiccant solution mass flow rate. The results of Khan et al. [19] and Liu et al. [23] are represented in this diagram for comparison. These results are in agreement with the present work for the parameters studied.
The effect of desiccant solution temperature on the air humidity ratio reduction is illustrated in Fig. (25). It is noticed from the figure that the air humidity ratio reduction decreases with the increasing the desiccant solution temperature. Good agreement between the present results and those obtained by Liu et al. [18], Khan et al. [19] and Longo and Gasparella [21].
The effect of desiccant solution/air mass flow rate on the dehumidifier effectiveness is illustrated in Fig. (26). It is observed from the figure that the dehumidifier effectiveness increases with the increasing the desiccant solution/air mass flow rate ratio. The results of Lowenstein and Gabruk [22] are represented in this diagram for comparison. These results are agreement with the present work for the parameters studied.

The effect of desiccant solution/air mass flow rate ratio on air humidity ratio reduction is illustrated in Fig. (28). It is seen from the present results that humidity ratio reduction increases with the increasing the desiccant solution/air mass flow rate ratio. The trend of the present results is in fair agreement with that of Lowenstein and Gabruk [22].

5. Correlations for the performance indices

The selected performance indices for the air dehumidifier in the present investigation are the air humidity reduction and the dehumidifier effectiveness. These indices are correlated utilizing the present experimental data by least square method as a function of dimensionless cooling water temperature, desiccant solution temperature, liquid to gas flow rate ratio and desiccant solution concentration. The correlations equations (6, 7) represent the experimental data with a maximum error of ±15%. The correlation equations for the air humidity reduction ratio and dehumidifier effectiveness, the experimental data, as well as the upper and lower limits are illustrated in Figs. (28-29) respectively.

$$\frac{\Delta w}{w_i} = 0.0457\left(\frac{T_c}{T_{a,i}}\right)^{-1.0965}\left(\frac{T_s}{T_{a,i}}\right)^{-0.777}\left(\frac{L}{G}\right)^{0.1928}(x)^{0.1665} \tag{6}$$

$$\varepsilon = 0.1069\left(\frac{T_c}{T_{a,i}}\right)^{-1.8689}\left(\frac{T_s}{T_{a,i}}\right)^{-0.0614}\left(\frac{L}{G}\right)^{0.1503}(x)^{0.1554} \tag{7}$$

where; $0.3 \le (T_{cw}/T_{a,i}) \le 0.45$, $0.675 \le (T_{si}/T_{a,i}) \le 0.7975$, $2.466 \le (L/G) \le 17.747$, $0.1 \le x \le 0.45$

6. Conclusions

In view of what has been introduced the following conclusions can be drawn:

1. Both the humidity reduction ratio and the dehumidifier effectiveness decreases with the increase of cooling water temperature and liquid desiccant solution temperature.
2. The inlet air temperature (ambient air temperature) has insignificant effect on the dehumidifier performance indices. While the increase of inlet air humidity ratio (ambient air humidity ratio) the humidity reduction ratio increases.
3. The increase of air flow rate leads to decrease in the two performance indices of the dehumidifier.
4. The increase of liquid desiccant solution flow rate leads to an increase of the dehumidifier performance indices.
5. As the solution to air mass flow rate ratio increases the performance indices increases.
6. Both the performance indices increase with the increase of the liquid desiccant solution concentration.
7. Two empirical correlations equations for humidity reduction ratio and dehumidifier effectiveness are deduced as:

$$\frac{\Delta w}{w_i} = 0.0457\left(\frac{T_c}{T_{a,i}}\right)^{-1.0965}\left(\frac{T_s}{T_{a,i}}\right)^{-0.777}\left(\frac{L}{G}\right)^{0.1928}(x)^{0.1665}$$

$$\varepsilon = 0.1069\left(\frac{T_c}{T_{a,i}}\right)^{-1.8689}\left(\frac{T_s}{T_{a,i}}\right)^{-0.0614}\left(\frac{L}{G}\right)^{0.1503}(x)^{0.1554};$$

   With maximum error of ±15%, $0.3 \le (T_{cw}/T_{a,i}) \le 0.45$, $0.675 \le (T_{si}/T_{a,i}) \le 0.7975$, $2.466 \le (L/G) \le 17.747$, and $0.1 \le x \le 0.45$

8. The present experimental work shows new compact design-high performance for air dehumidifier-heat exchanger-regenerator liquid desiccant system and validated by comparing its results with previous published ones.

Nomenclature

Symbols	Description	Units
m	mass flow rate	kg/s
L/G	liquid to gas mass flow rate ratio	----
P_{wz}	saturation pressure of desiccant solution at concentration x and temperature T_s.	kPa
P_{ws}	saturation pressure of pure water at temperature T_s	kPa
P	total pressure of air on the desiccant solution =101.325	kPa
T	temperature	^oC
V_a	volume flow rate	l/s

W	humidity ratio of the air	$g_w/kg_{d.a}$
x	desiccant solution concentration	$kg_{salt}/kg_{solution}$

Subscripts

a	air	min	minimum
cw	cold water	o	outlet
hw	hot water	*s*	solution
i	inlet	w	water

Greek Symbols

Δ	reduction
ε	dehumidifier effectiveness

ABBREVIATIONS

HVAC	Heating, ventilating and air conditioning
PVC	Ploy-vinyl chloride

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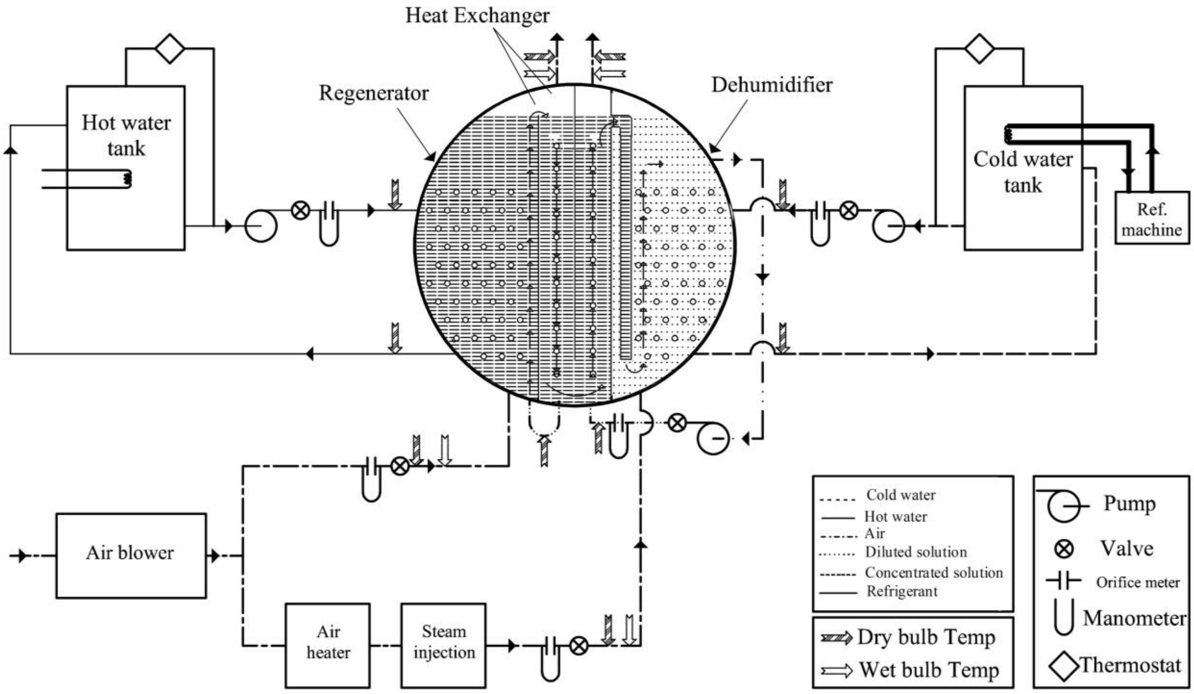


Fig. (1): A schematic diagram of the experimental setup

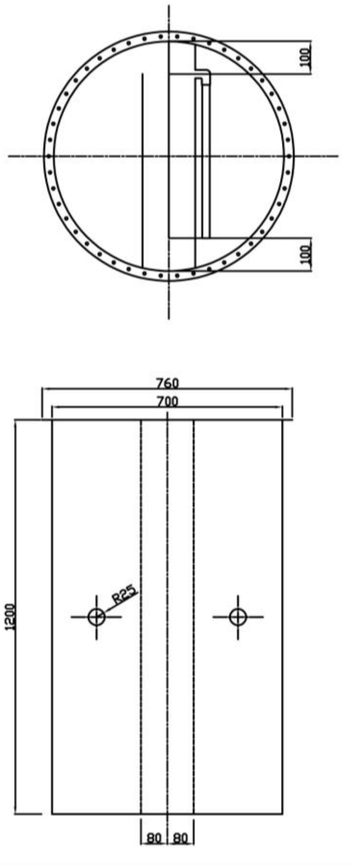


Fig. (2): Schematic diagram for dehumidifier-heat exchanger-regenerator uni-shell

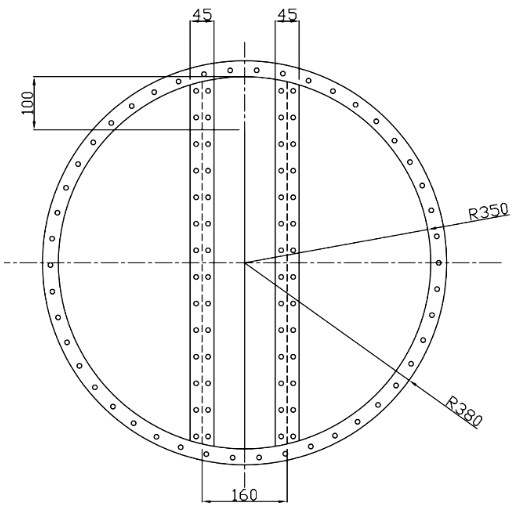


Fig. (3): The fixed front flange

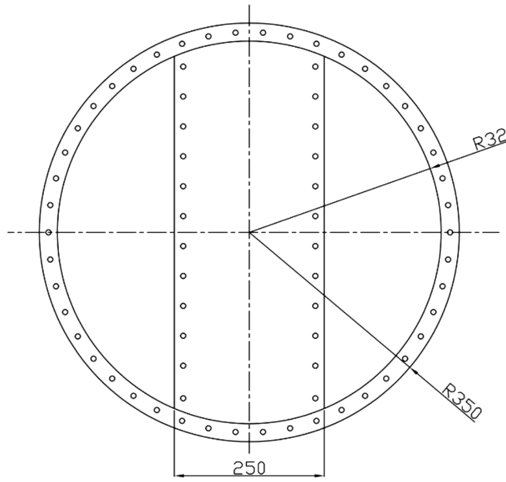


Fig. (4): The fixed back flange

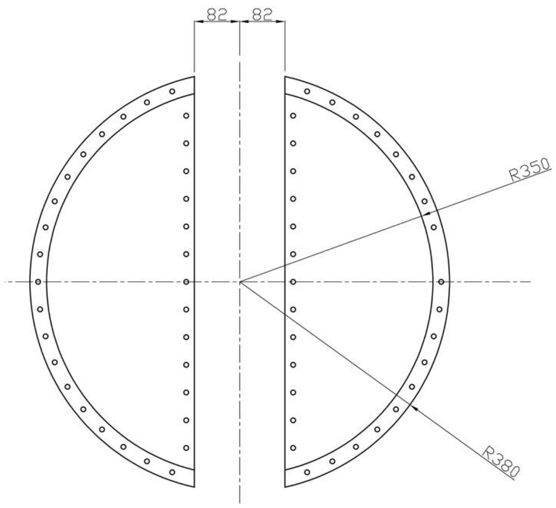


Fig. (5): The movable front flange

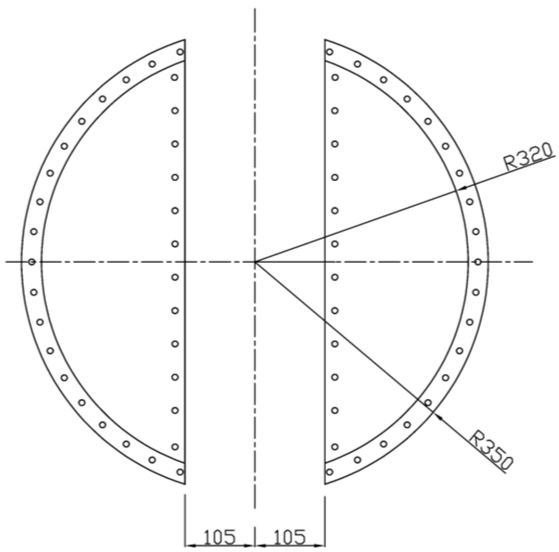
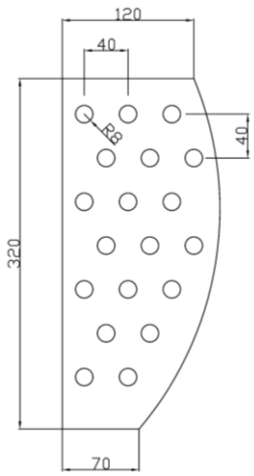


Fig.(6): The movable back flange



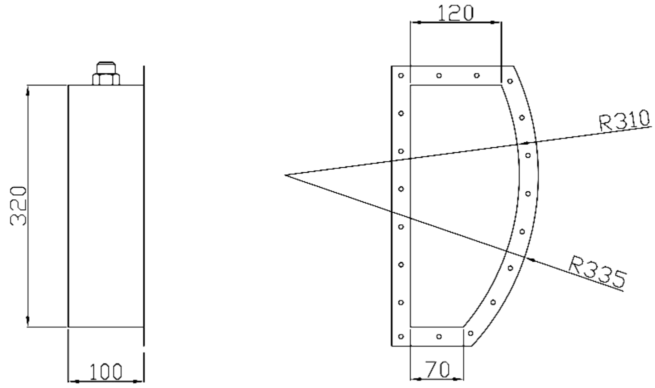


Fig. (7): The hot/cold water tubes arrangement

Fig. (8): Water box details

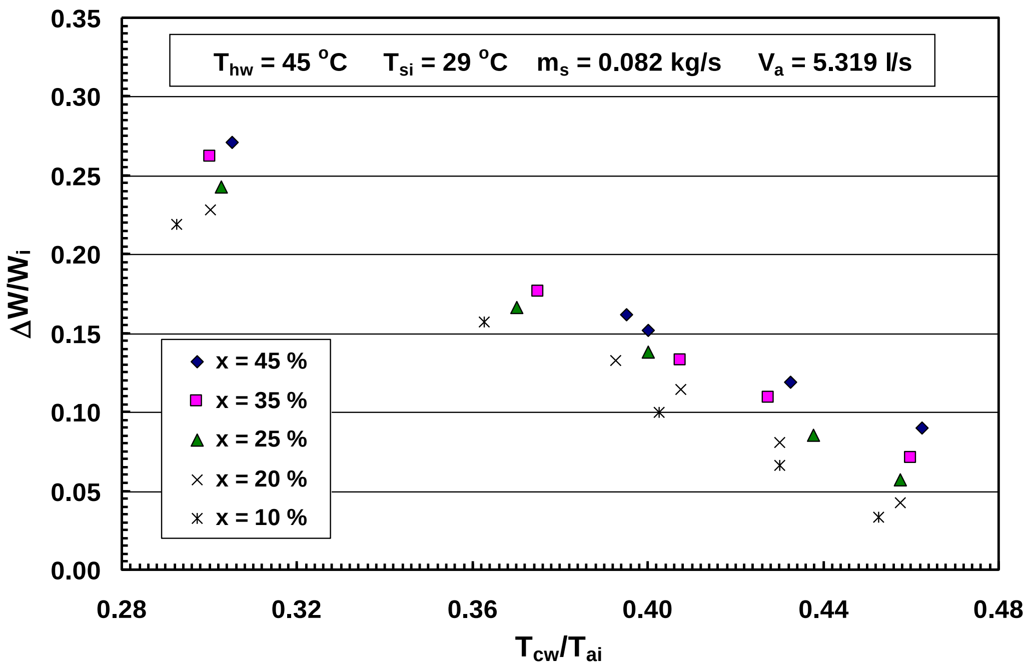


Fig. (9): Influence of cooling water inlet temperature on air humidity ratio reduction at different desiccant solution concentrations

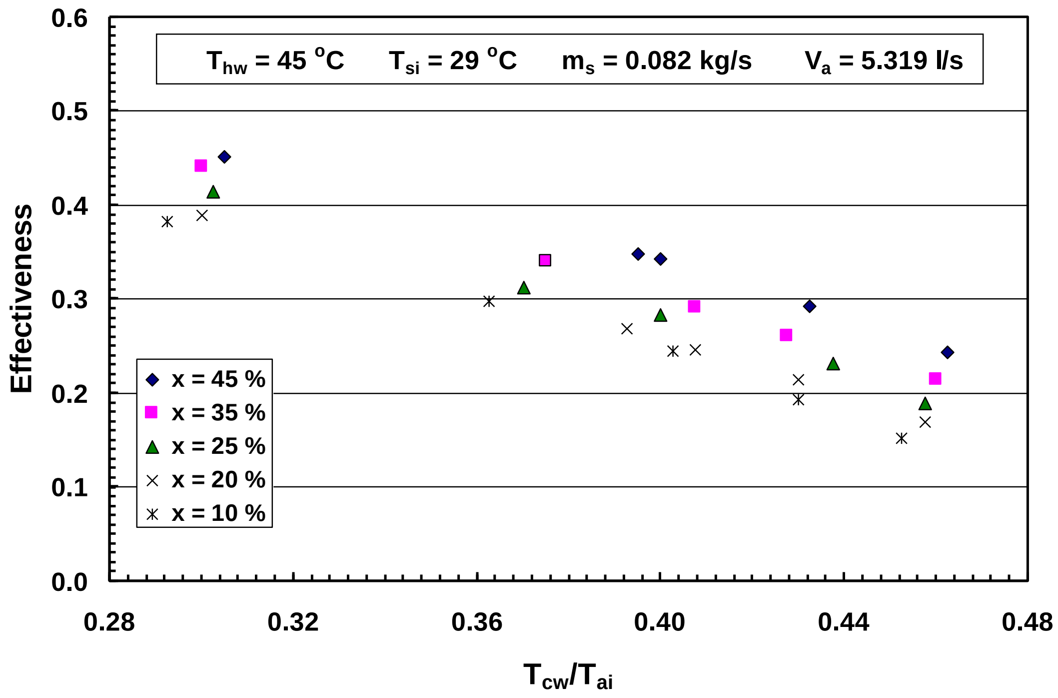


Fig. (10): Influence of cooling water inlet temperature on dehumidifier effectiveness at different desiccant solution concentrations

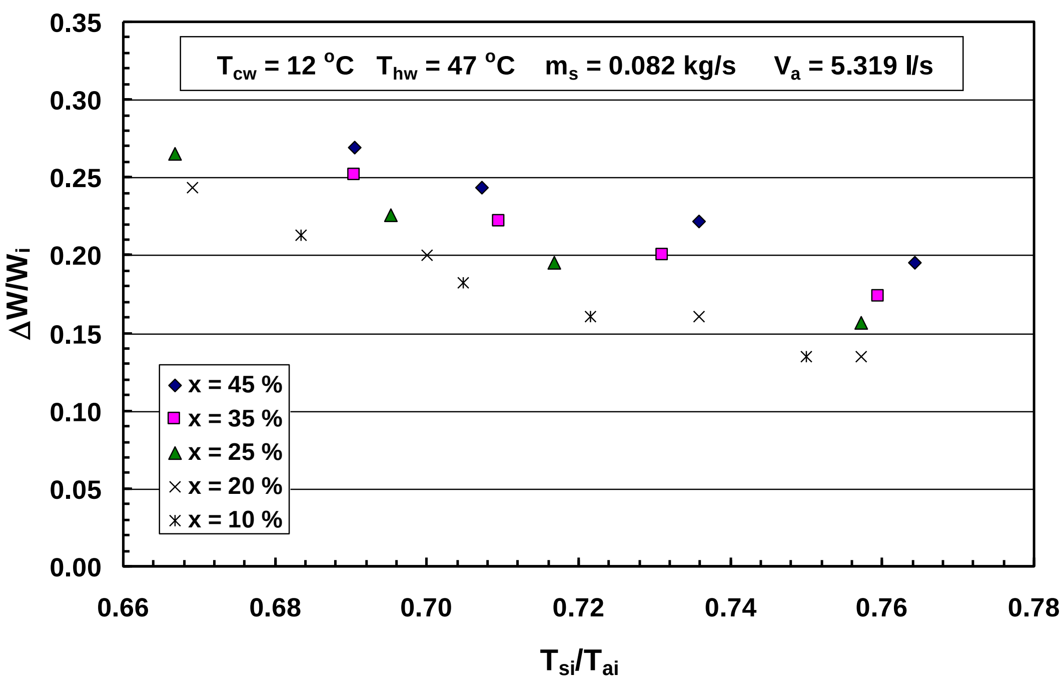


Fig. (11): Influence of desiccant solution inlet temperature on air humidity ratio reduction at different desiccant solution concentrations

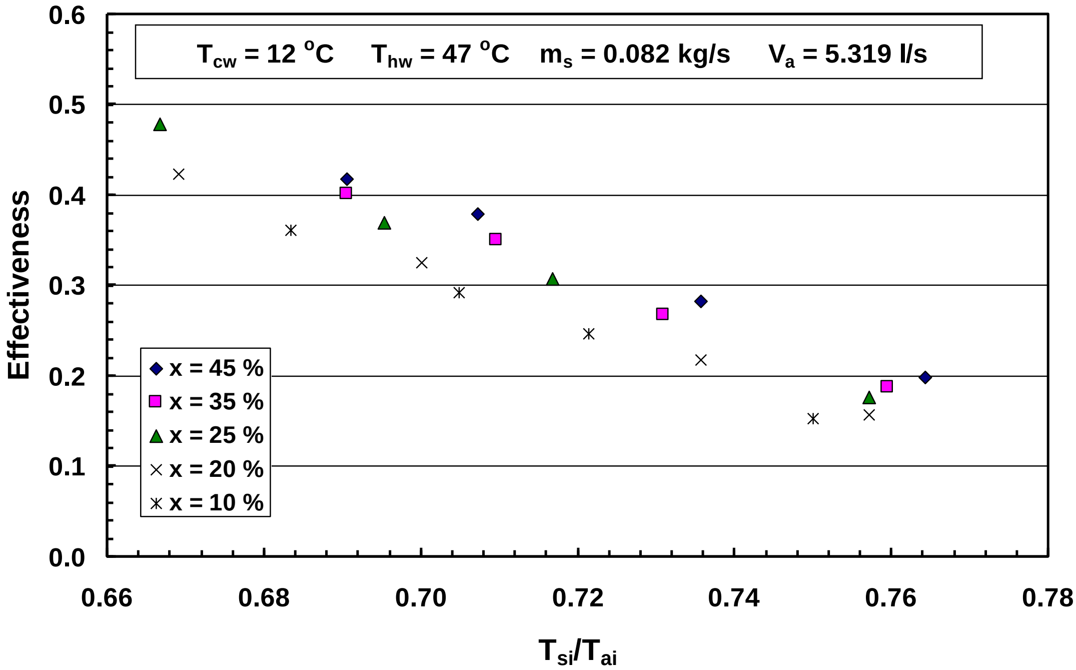


Fig. (12): Influence of desiccant solution inlet temperature on dehumidifier effectiveness at different desiccant solution concentrations

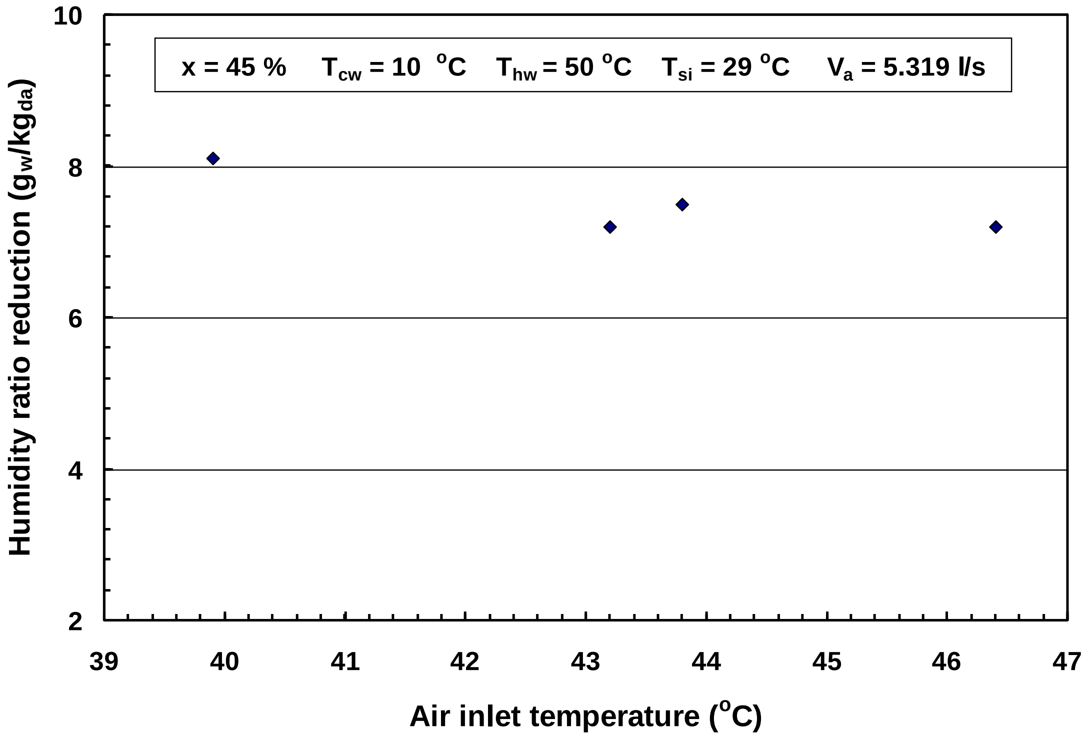


Fig. (13): Influence of air inlet temperature on air humidity ratio reduction

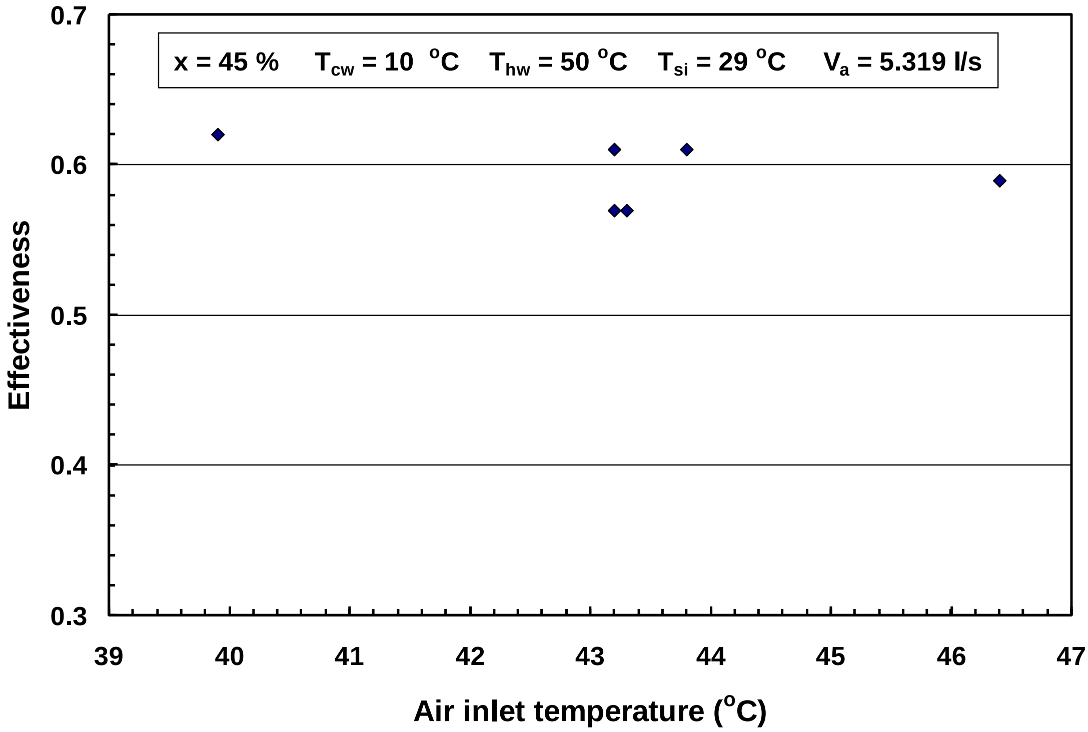


Fig. (14): Influence of air inlet temperature on dehumidifier effectiveness

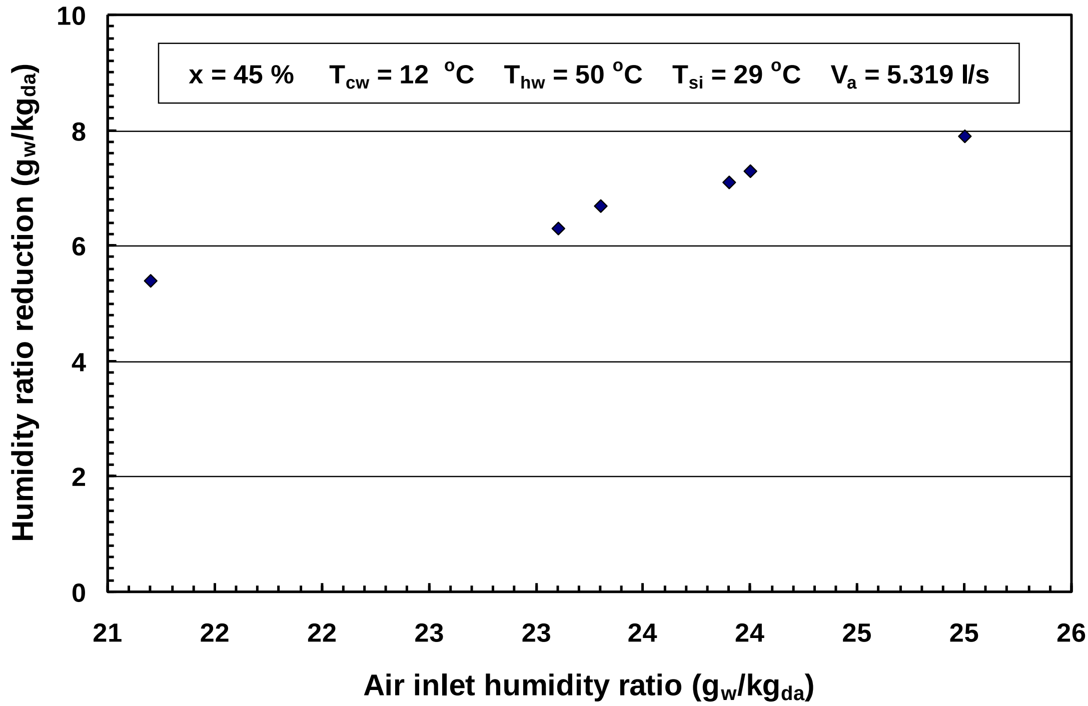


Fig. (15): Influence of air inlet humidity ratio on air humidity ratio reduction

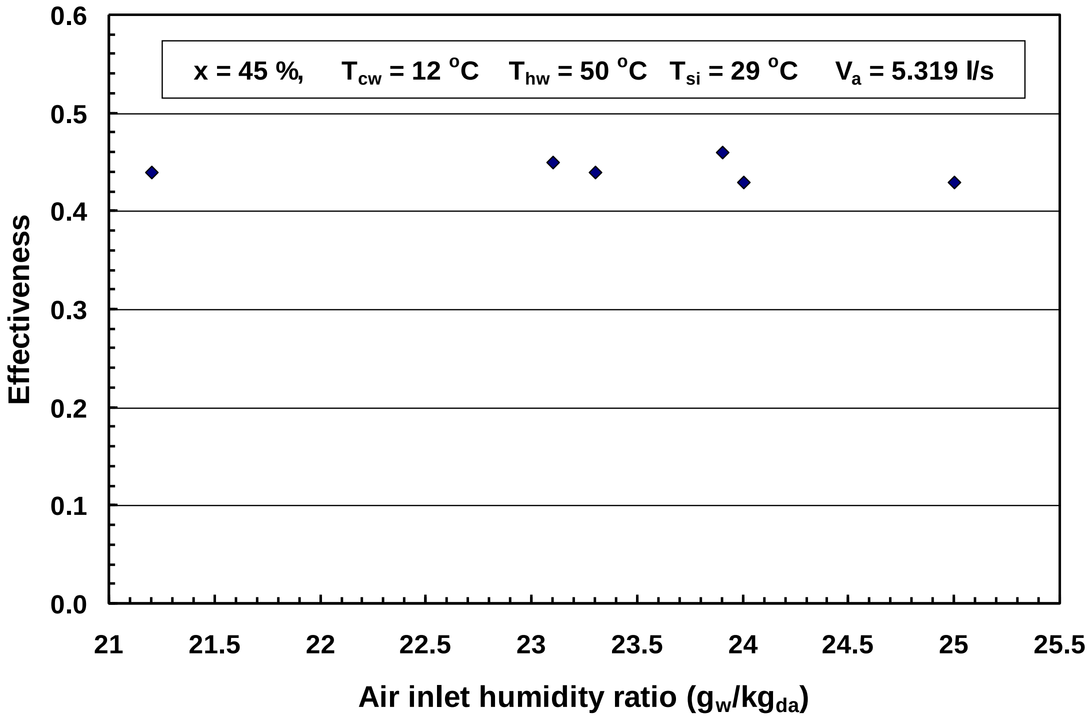


Fig. (16): Influence of air inlet humidity ratio on dehumidifier effectiveness

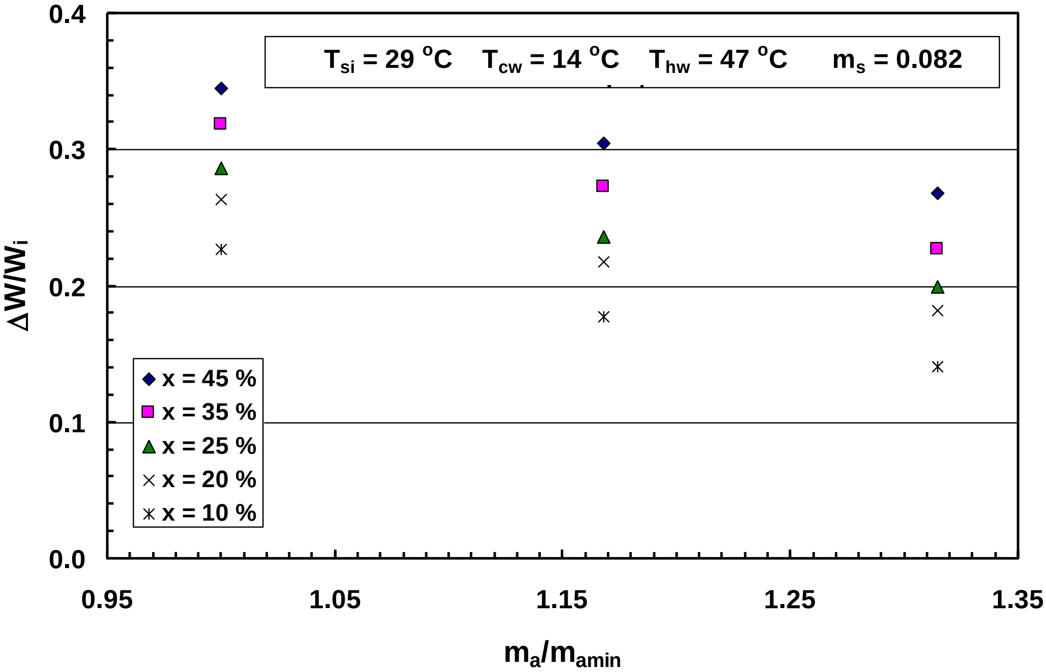


Fig. (17): Influence of air flow rate on air humidity ratio reduction at various concentrations

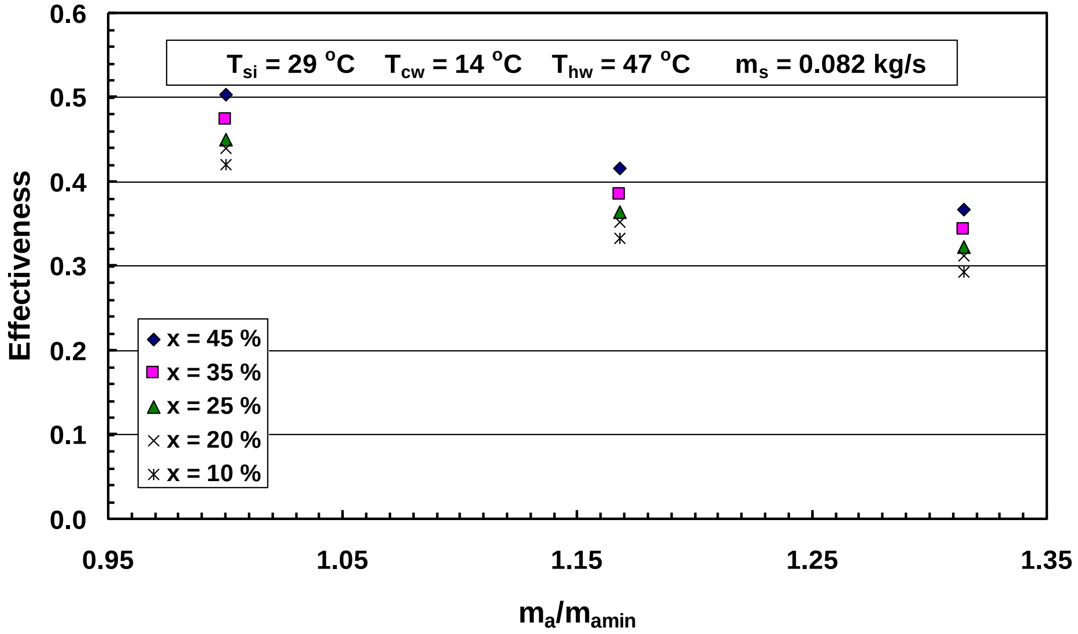


Fig. (18) Influence of air flow rate on dehumidifier effectiveness at various concentrations

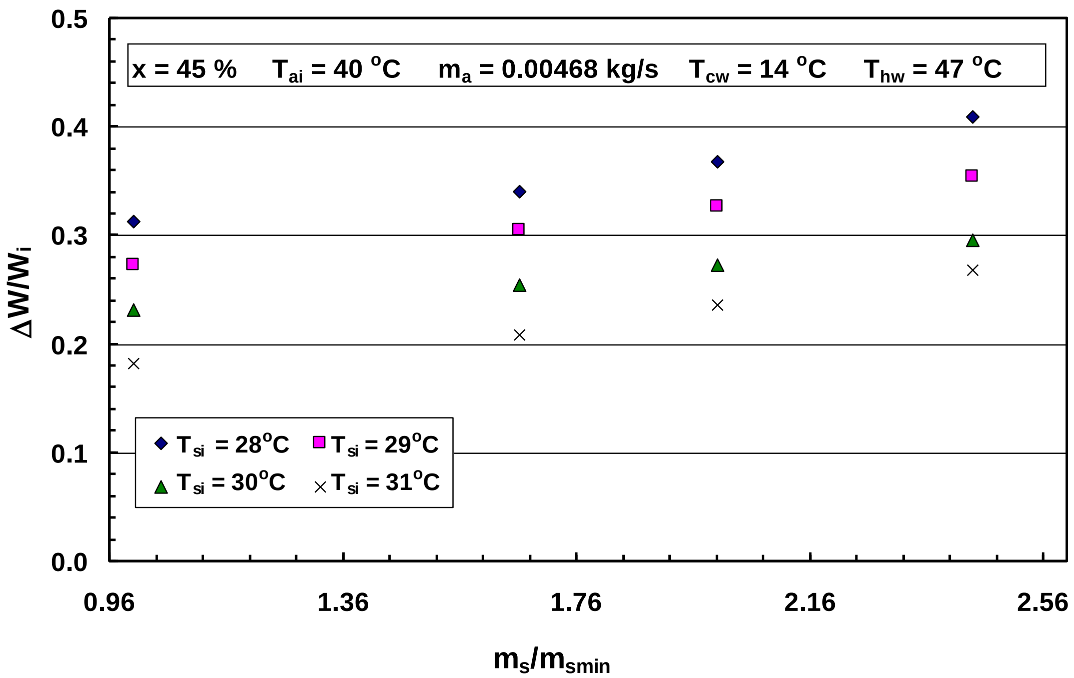


Fig. (19): Influence of desiccant solution mass flow rate on air humidity reduction ratio at various desiccant solution inlet temperatures

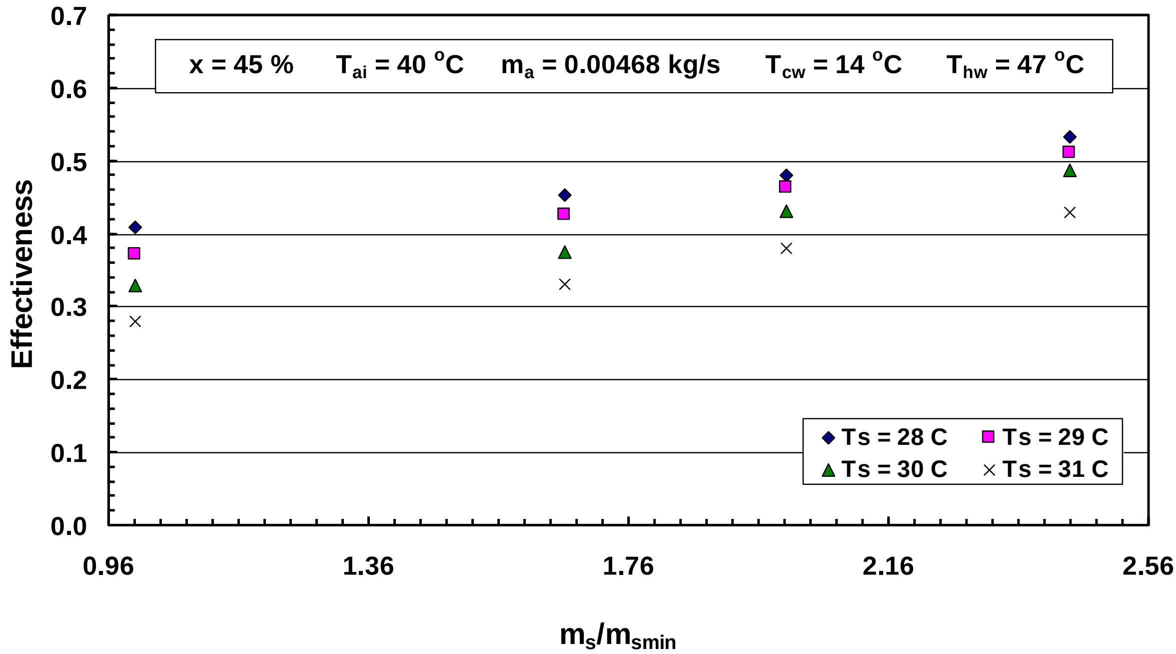


Fig. (20): Influence of desiccant solution mass flow rate on dehumidifier effectiveness at various desiccant solution inlet temperatures

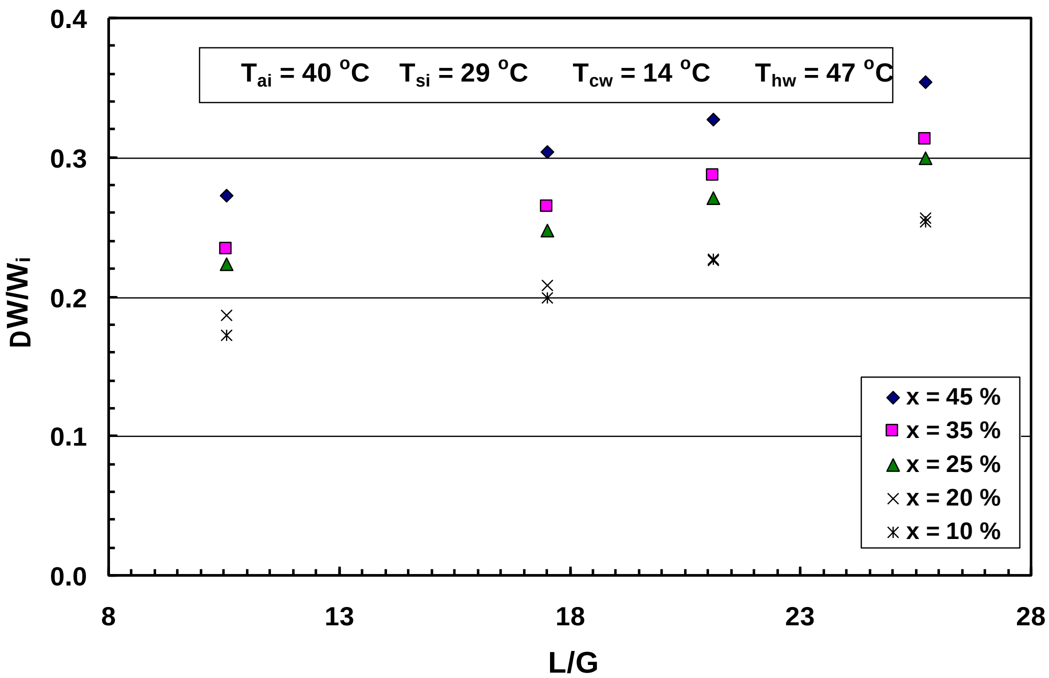


Fig. (21): Influence of liquid/gas flow rate on air humidity reduction ratio at various concentrations

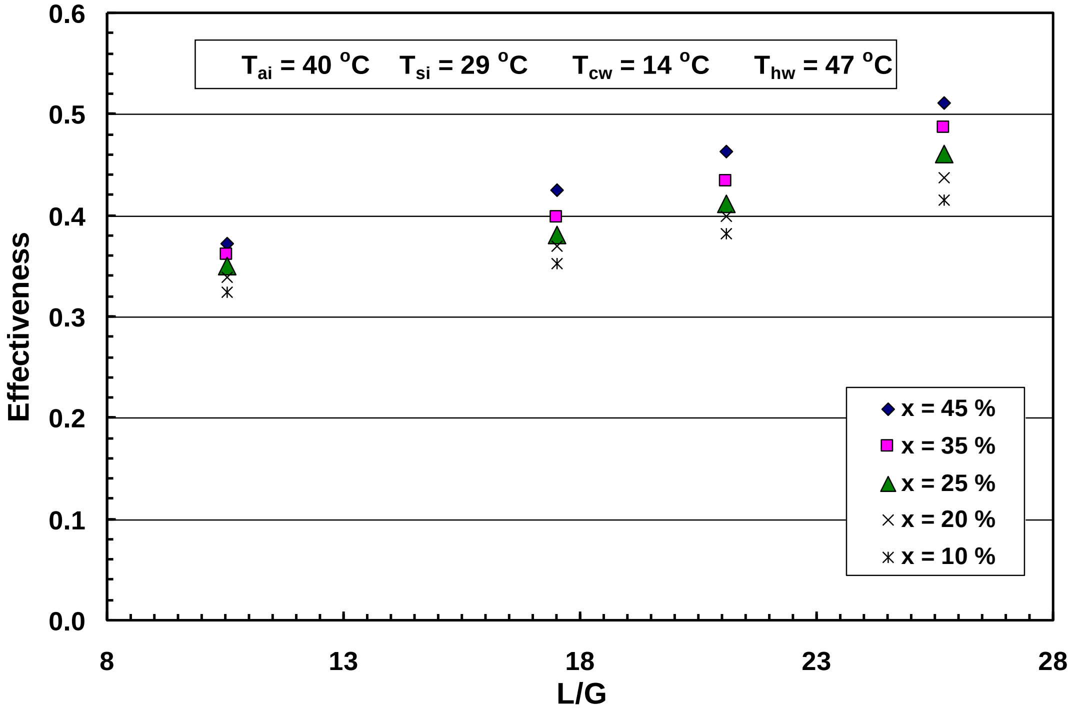


Fig. (22): Influence of liquid/gas flow rate on dehumidifier effectiveness at various concentrations

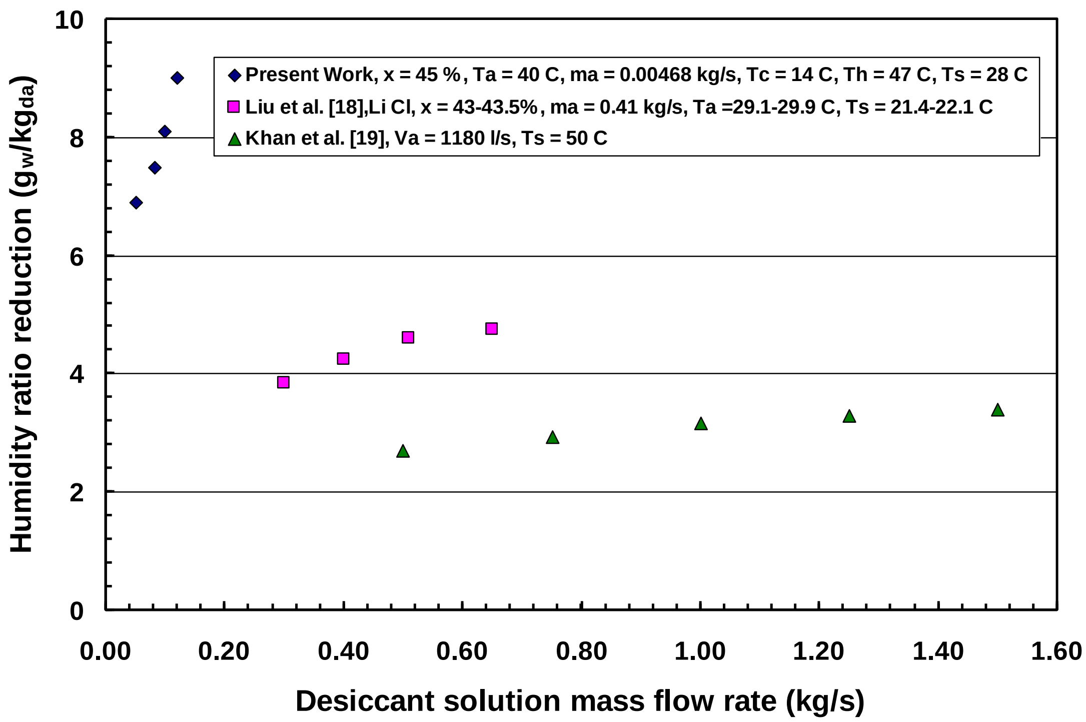


Fig. (23) Influence of desiccant solution mass flow rate on air humidity ratio reduction

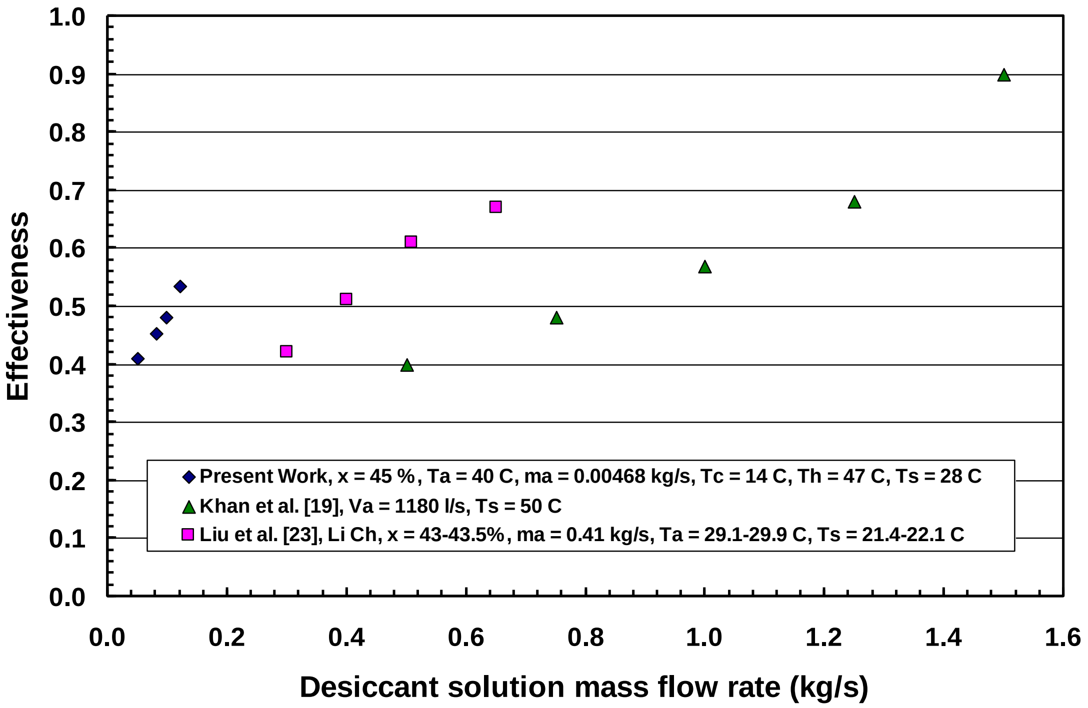


Fig. (24) Influence of desiccant solution mass flow rate on humidifier effectiveness

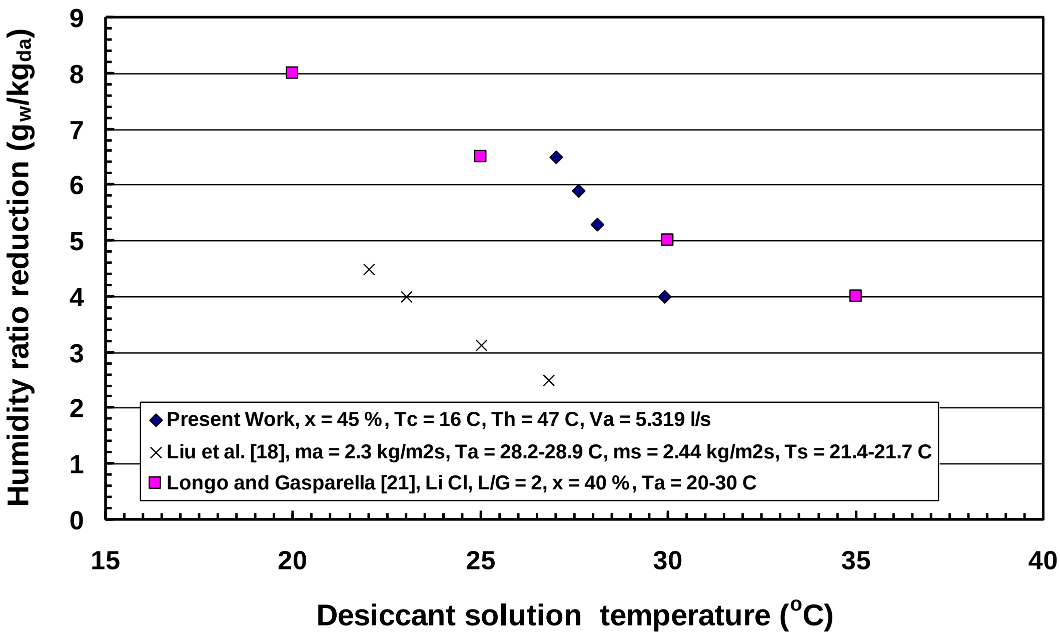


Fig. (25) Influence of desiccant solution temperature on air humidity ratio reduction

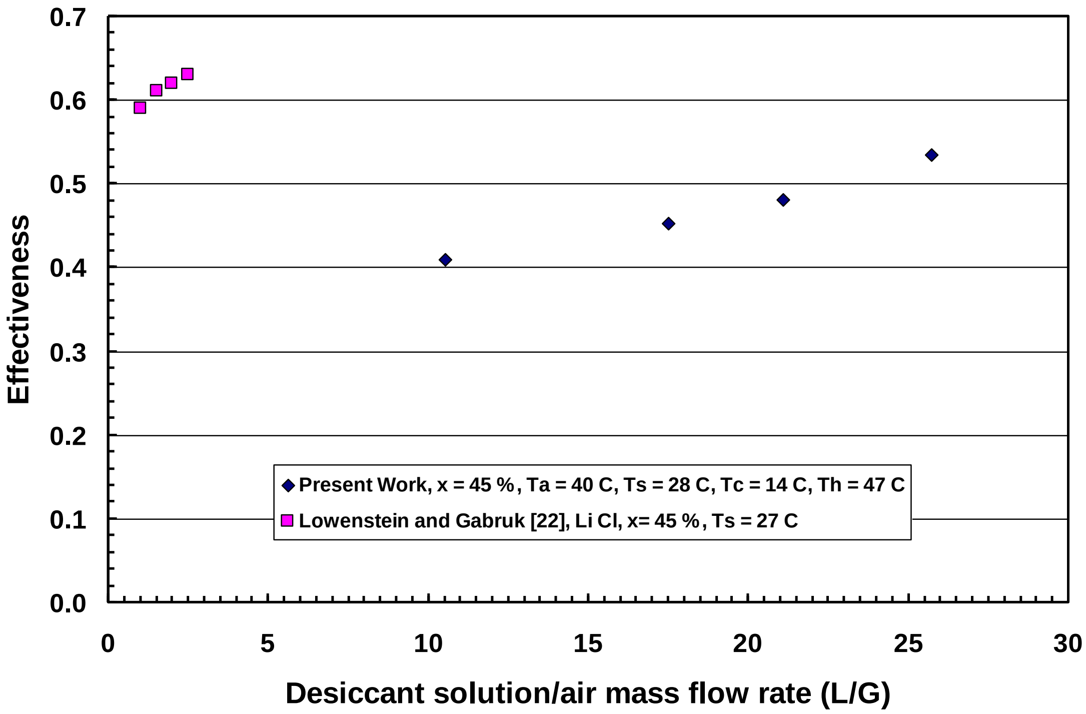


Fig. (26) Influence of desiccant solution /air mass flow rate on humidifier effectiveness

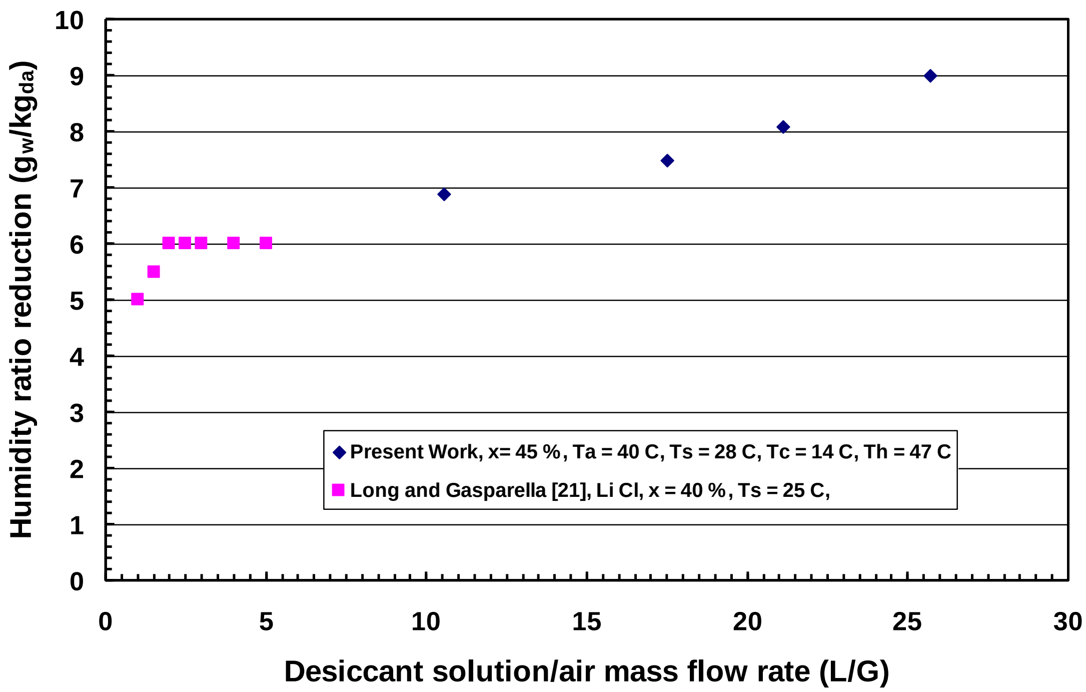


Fig. (27): Influence of desiccant solution /air mass flow rate on air humidity ratio reduction

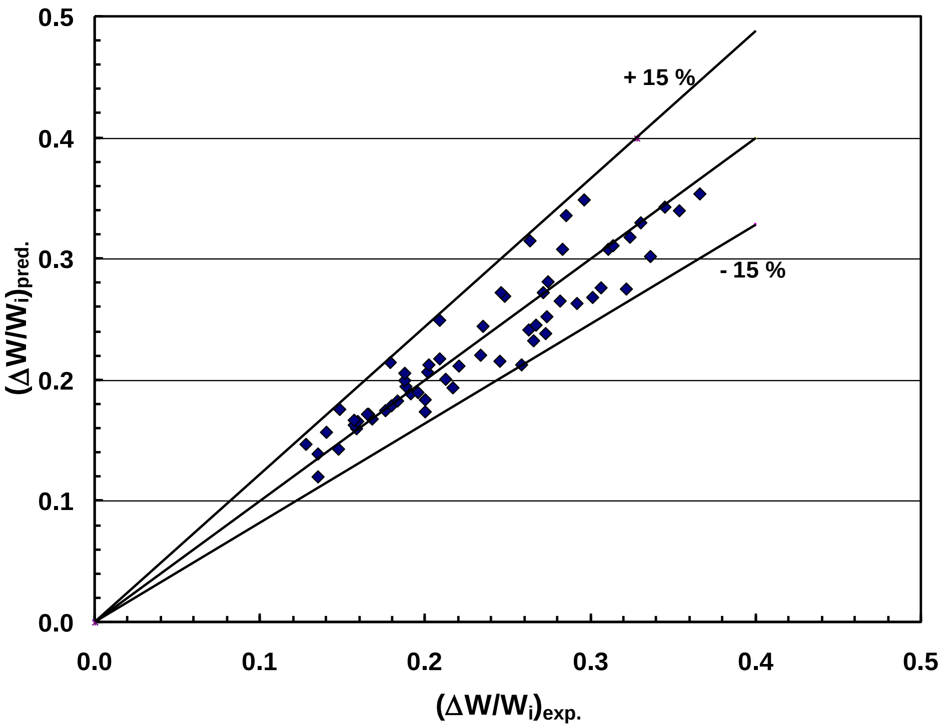


Fig. 28 Comparison between the experimental values and the predicted values of air humidity reduction ratio for experimental runs

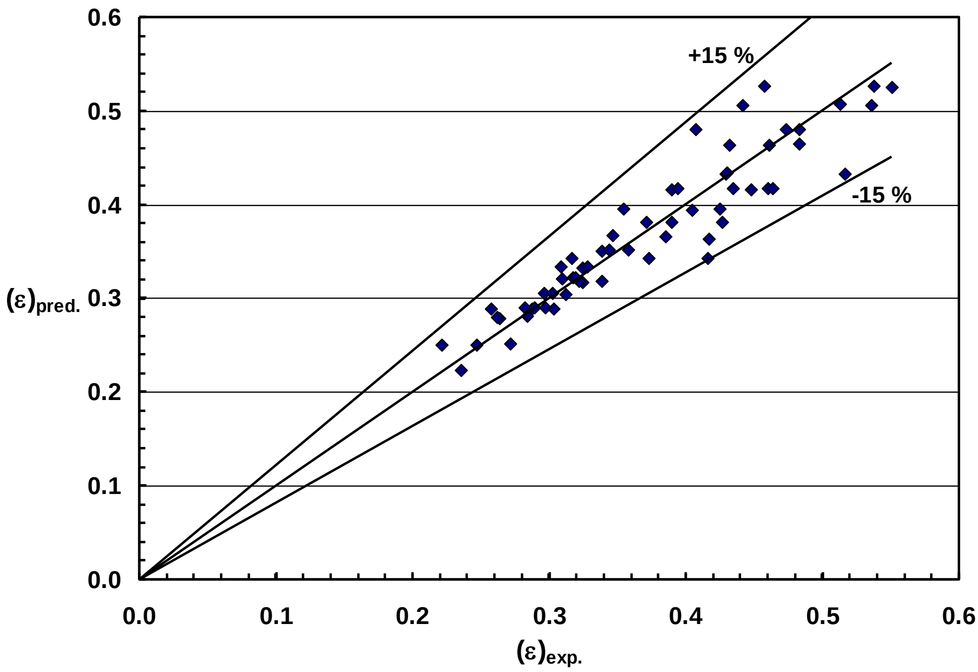


Fig. 29 Comparison between the experimental values and the predicted values of dehumidifier effectiveness for experimental runs