Development of
Effective Solution Heat Exchanger

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Abstract

This paper relates to the development of effective solution heat exchanger/s to enable liquid desiccant based dehumidification system achieve high COP. The solution heat exchangers were fabricated using a proprietary aluminium extrusion developed at Heat Pump Laboratory of IIT Bombay. The extrusion has wavy profile which increases the heat transfer area by π/2 while keeping the flow cross section area same. This reduces the hydraulic diameter, which results in higher heat transfer coefficients. It was tested with water and liquid desiccant at 60%, 65% and 70% concentration. Effectiveness in the range of 84 to 92% was demonstrated. Pressure drop as low as 13 kPa was possible by operating in laminar flow regime and decoupling heat transfer and pressure drop. High effectiveness of the solution heat exchangers enables the two stage liquid desiccant fresh air dehumidification cycle to achieve COPs in the range of 1.6 to 2 depending upon the ambient weather conditions.

Introduction

A two stage liquid-desiccant system has higher regeneration COP than a single stage liquid desiccant system. The advantage is quite dramatic when the liquid-desiccant system uses an effective solution heat exchanger. The strong liquid desiccant leaving the regenerator at high temperature needs to be cooled down to ambient temperature before it enters the dehumidifier. Similarly, the weak liquid desiccant is preheated in order to reduce the energy requirement at the regenerator. This is achieved using a Solution Heat Exchanger, SHE.

Rane and Mehta (2013) developed a Potassium Formate, KCOOH, based liquid desiccant system with two stage regeneration. Multi-pass tubes in tube heat exchangers with effectiveness between 38 to 58% were used. COP of 1.03 has been demonstrated. Theoretical analysis of the two stage liquid desiccant dehumidification system shows that COP higher than 1.6 can be achieved if solution heat exchanger with effectiveness higher than 80% could be developed.

Kaynakli and Kilic, (2007), performed detailed thermodynamic analysis of the water/lithium bromide, H_2O/LiBr, absorption refrigeration cycle. Investigations on the influences of effectiveness of SHE on the coefficients of performance, COP, revealed that improvement in effectiveness of the SHE can increases the COP by 44%.

Altener (2002) studied liquid desiccant air conditioning system with a regenerative heat exchanger. In their system, weak liquid desiccant enters the gas-fired boiler. The desiccant boils within the heat exchanger and a mixture of steam and strong desiccant leave the heat exchanger and enters a spin separator. Concentrated desiccant leaves at the bottom of the spin separator and steam exits from the top. The steam becomes the thermal source for additional regeneration in a conventional scavenging-air regenerator. Thus, each unit of energy supplied to the regenerator does double duty.

The solution heat exchanger, SHE, developed in the present work is part of the Two Stage Liquid Desiccant based Dehumidification System. This systems consists of a high temperature regenerator, HTR, a low temperature regenerator, LTR, fresh air dehumidifier, FAD, high temperature and low temperature solution heat exchangers, HT_SHE and LT_SHE, as the major components.

The HT_SHE recovers heat from the intermediate concentration liquid desiccant and preheats the weak liquid desiccant coming from the LT_SHE. The LT_SHE preheats the weak liquid desiccant from the liquid desiccant tank and cools down the strong liquid desiccant from low temperature regenerator before it enters the fresh air dehumidifier to absorb moisture. Preheating of the weak liquid desiccant helps reduce the energy required to boil the liquid desiccant in the high temperature regenerator while the cooling of strong liquid desiccant helps increase the dehumidification efficiency.

Description of Solution Heat Exchanger

Rane and Chavan (2013) discloses enhancement of flow passages in a multi-wall sheet. Novel crests and troughs as shown in *Figure 1* are extruded on the outer and inner surfaces of the multi-wall sheet. These crests and troughs increase the heat exchange area along with rate of heat transfer coefficient enabled by increased mixing of fluid.

Proprietary aluminum extrusion was used in fabrication of both high temperature and low temperature solution heat exchangers. The profile of these heat exchangers is wavy or corrugated. The key parameters that affect the performance of heat exchangers with corrugated geometry are wave pitch, corrugation angle and channel spacing. The SolidWorks model of the HT_SHE and LT_SHE made from the patented extrusion profile in aluminum is shown in *Figure 2*.

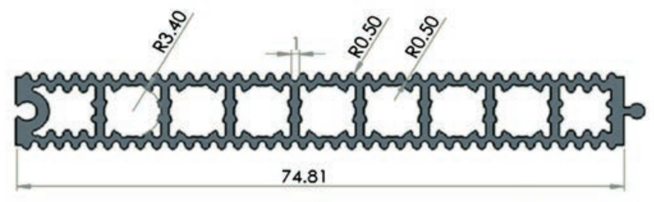


Figure 1: Cross section view of multi-wall sheet (Rane and Chavan, 2014)

Figure 2: High and low temperature solution heat exchangers

The dimensions and internal details of the aluminum extrusion used for fabrication of the High Temperature SHE, HT_SHE, and the Low Temperature SHE, LT_SHE, are given in *Table 1*.

Table 1: Internal details for HT_SHE & LT_SHE

Internal Details for extrusion passages		A	B	C	Extrusion Length (A+7B+C) 1 m	Extrusion Length 1.219 m	Total Number of Extrusions 6	Total Number of Cores 2
peri.$_{\text{ext.p.i}}$	mm	33.2	32.77	38.2	300.8	300.8	1804.7	3609.5
A.cs.$_{\text{ext.p.i,}}$	cm^2	0.36	0.44	0.38	3.82	3.82	22.9	45.8
vol.$_{\text{ext.p.i}}$	Cc	36.8	44.2	38.8	385	469.3	2815.9	5631.8
A.he.$_{\text{ext.p.i}}$	m^2	0.0332	0.0328	0.0383	0.301	0.367	2.2	4.4
weight/A.$_{\text{he.i}}$	kg/m^2				2.581	2.581	2.58	2.58

Table 2 shows the external details of the aluminium extrusion used in the fabrication of the High Temperature SHE, HT_SHE, and the Low Temperature SHE, LT_SHE. Six extrusions having a length of 1.219 m each were welded to 2 mm wide spacers. The spacers create the gap between adjacent extrusions, which in turn defines the shell side of the solution heat exchanger. Although the strong solution flow rates and heat duties are slightly different, the heat exchangers were kept identical for convenience of fabrication and integration. V shaped headers help distribute the liquid desiccant better through all the parallel passages of the solution heat exchanger. *Figure 2* shows the 3D SolidWorks model of the headers on the high and low temperature solution heat exchangers.

Table 2: External details for HT_SHE and LT_SHE

External Details of Extrusion		Channel Length 1 m	Channel Length 1.219 m	Total Number of Channels in a Core 6	Total Number of Cores 2
peri$_{\text{.ext.p.e}}$	mm	232.8	232.8	1396.9	2793.8
A.$_{\text{cs.ext.p.e}}$	cm^2	2.877	2.877	17.3	34.5
vol.$_{\text{ext.p.e}}$	Cc	287.7	350.6	2103.9	4207.7
A$_{\text{.he.ext.p.e}}$	m^2	0.233	0.284	1.7	3.4
weight	Kg	0.777	0.947	5.7	11.4
weight/A$_{\text{.he.ext.p.e}}$	kg/m^2	3.338	3.338	3.338	3.338

Theoretical Estimation of SHE Effectiveness using NTU Method

The theoretical estimation of solution heat exchanger effectiveness was performed based on the liquid desiccant flow rate and expected temperature at different sections of the two stage liquid desiccant based dehumidifier. In the 3 TR dehumidification system, cold liquid desiccant at a mass flow rate of 78.6 kg/h gets pumped to the low temperature regenerator through low temperature and high temperature solution heat exchanger where it gets preheated. Liquid desiccant boils in the high temperature regenerator. Steam and intermediate concentration liquid desiccant get separated in steam liquid desiccant separator. Intermediate concentration liquid desiccant at a mass flow rate of 75.7 kg/h enters high temperature solution heat exchanger.

Concentration of liquid desiccant is 65% at the pump inlet, and 67.5% at the high temperature regenerator, HTR, outlet and 70% at LTR outlet. The strongest liquid desiccant at a mass flow rate of 73.4 kg/h is having 70% concentration. In low temperature solution heat exchanger, weak liquid desiccant is exchanging heat with strong liquid desiccant from low temperature regenerator,

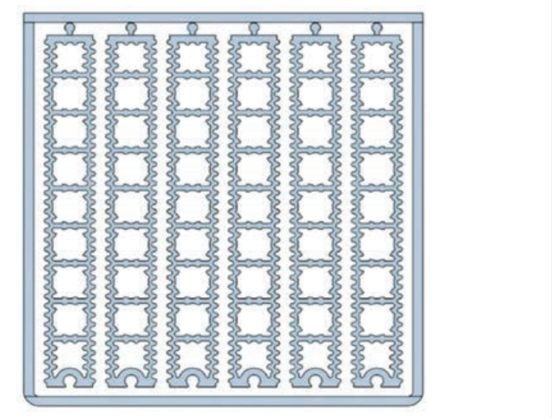


Figure 3: Cross sectional view of solution heat exchanger

LTR. Moderate temperature weak liquid desiccant exchanges heat with hot intermediate concertation liquid desiccant in the high temperature solution heat exchanger.

Cross sectional view of the solution heat exchanger is given in *Figure 3*. The area of cross section and area available for heat transfer can be calculated from the internal and external details of the solution heat exchanger given in *Table 1* and *Table 2*.

The cross sectional area of the tube side is the sum of the cross section of the nine channels of the single aluminum extrusion multiplied by the six extrusions in the shell.

$$A_{.cs.tube} = 6(A_{.cs.ext.p.i.A} + 7\ A_{.cs.ext.p.i.B} + A_{.cs.ext.p.i.C})$$

$$A_{.cs.tube} = 6(0.36\ cm^2 + 7\ .\ 0.44\ cm^2 + 0.38\ cm^2) = 22.9\ cm^2 \tag{1}$$

Similarly, the perimeter of the tube side of the solution heat exchanger is obtained from *Equation 2*.

$$peri_{.tube} = 6(peri_{.cs.ext.p.i.A} + 7\ peri_{.cs.ext.p.i.B} + peri_{.cs.ext.p.i.C})$$

$$peri_{.tube} = 6(33.2\ mm + 7\ .\ 32.77 + 38.2\ mm) = 1804.7\ mm \qquad 2)$$

Shell side cross sectional area and perimeter can be directly taken from *Table 2* for six extrusions.

$$A_{.cs.shell} = 17.3\ cm^2$$

$$peri_{.shell} = 1396.9\ mm$$

Sample calculations for solution heat exchanger effectiveness is presented as follows. Since the design of both low temperature and high temperature heat exchangers is the same, it can be simplified to a single heat exchanger with length L equal to twice of the low temperature solution heat exchanger, LT_SHE, and high temperature solution heat exchanger, HT_SHE. Fluid temperature at the inlet of the low temperature solution heat exchanger and high temperature solution heat exchanger is known. The highest temperature is 95°C, which is slightly lower that the boiling temperature of water. The lowest temperature, 33°C, is at the cold side inlet of low temperature solution heat exchanger.

The overall heat transfer coefficients of solution heat exchangers are obtained using the following assumptions:

i. Internal and external surface of the aluminum extrusion is equivalent to that of similar cross section rectangular passage
ii. Fluid flow inside the passage and on the surface of the extrusion is assumed to be hydrodynamically fully developed
iii. Heat flux is uniform at the solution heat exchanger wall
iv. Steady state operation
v. Fluid properties are calculated at the mean temperature

The mean fluid temperature is the average of cold fluid at low temperature solution heat exchanger inlet and hot fluid at the high temperature solution heat exchanger inlet.

$$t_m = \frac{t_{.h.w.htshe.i} + t_{.c.w.ltshe.i}}{2}$$

$$t_m = \frac{(95^\circ C + 33^\circ C)}{2} = 64^\circ C \tag{3}$$

Fluid properties at 64°C can be obtained from the property table provided by Engineering Toolbox. Density, ρ = 980.5 kg/m^3, thermal conductivity, k = 0.659 W/m.K, dynamic viscosity, μ = 0.0004332 Pa.s, specific capacity, c_p = 4.187 kJ/kg.K, and Prandtl number, Pr = 2.75.

Since the shape of the channels is assumed to be rectangular, the hydraulic diameter is calculated using *Equation 4*.

$$d_{h.tube} = \frac{4\ A_{.cs.tube}}{peri_{.tube}}$$

$$d_{h.tube} = \frac{4 \times 0.00229\ m^2}{1.8047\ m} = 0.00508\ m \tag{4}$$

The hydraulic diameter for the shell side of the solution heat exchanger can be obtained using *Equation 5*.

$$d_{h.shell} = \frac{4\ A_{cs.shell}}{peri_{shell}}$$

$$d_{h.shell} = \frac{4 \times 17.3\ cm^2}{139.69\ cm} = 0.00495\ m \tag{5}$$

Based on the mass flow rate of water, density and cross section area, average velocity of the fluid is determined. Tube side and shell side fluid velocities are calculated as follows:

The mass flow rate of water, mf.w = 0.02028 kg/s, is the equivalent of the mass flow rate of weak liquid desiccant corresponding to 1 TR dehumidification capacity.

$$v_{av.tube} = \frac{mf_w}{\rho_w\ A_{.cs.tube}}$$

$$v_{av.tube} = \frac{0.02028\ kg/s}{980.5\ \frac{kg}{m^3} \times 0.00229\ m^2} = 0.00903\ m/s \tag{6}$$

Shell Side,

$$v_{av.shell} = \frac{mf_w}{\rho_w\ A_{cs.shell}}$$

$$v_{av.shell} = \frac{0.02028\ kg/s}{980.5\ kg/m^3 \times 1.73 \times 10^{-3}\ m^2} = 0.012\ m/s \tag{7}$$

Reynold's number is an indicator of the type of flow to be expected, laminar or turbulent.

Passage side Reynold's Number,

$$Re_{tube} = \frac{\rho_w\ v_{av.tube}\ d_{.h.tube}}{\mu}$$

$$Re_{tube} = \frac{980.5\ \frac{kg}{m^3} \times 0.00903\ \frac{m}{s} \times 0.00508\ m}{0.0004332\ \frac{kg}{m.s}} = 104 \tag{8}$$

Shell side Reynold's Number,

$$Re_{shell} = \frac{(\rho_w\ v_{av.shell}\ d_{h.shell})}{\mu}$$

$$Re_{shell} = \frac{(980.5\ kg/m^3 \times 0.012\ \frac{m}{s} \times 0.00495\ m)}{0.0004332\ kg\ /m.s} = 134 \tag{9}$$

The Reynolds number was much lower than 2,300 both on the shell side and inside the passages. So the flows were laminar on both the sides.

Thermal entry length was calculated based on the empirical relation for laminar flow. (Fundamentals of Heat and Mass Transfer, Incropera F and DeWitt D).

$$L_t / d_{h.tube} = 0.05 \ Re_{tube} \ Pr$$

$$L_t = 0.05 \times 103.8 \times 2.75 \times 0.00508 = 0.073 \text{ m} \qquad (10)$$

Total length of each solution heat exchanger is 1.219 m, which was significantly larger than the thermal entry length. Hence, the assumption of neglecting the entrance effect and considering the flow to be thermally fully developed was reasonable.

In a square channel a/b =1, and form uniform heat flux the Nusselt number is 3.61 (*Table 3*, Incropra and DeWitt D). The calculated heat transfer coefficients for the passage side and shell side were 468.3 W/m².K and 1095.7 W/m².K respectively.

$$h_{tube} = \frac{Nu_{.tube} \ k}{d_{.h.tube}}$$

$$h_{tube} = \frac{3.61 \times 0.659 \text{ W/m.K}}{0.00508 \text{ m}} = 468.3 \text{ W/m}^2\text{K} \qquad (11)$$

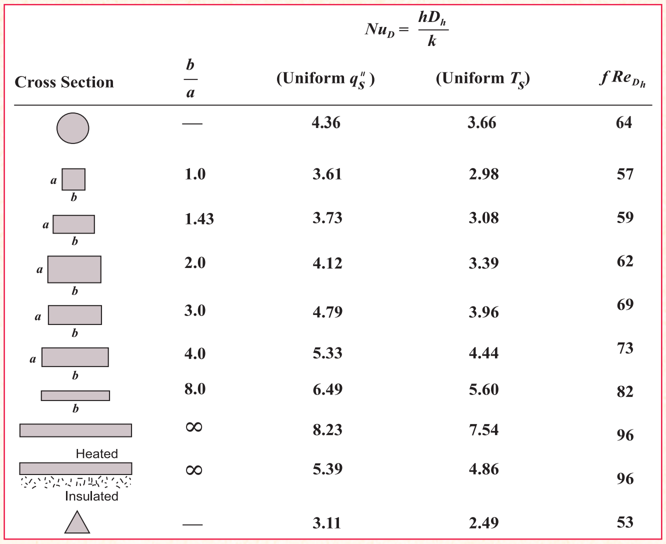
On the shell side a/b is very large, >50. Nusselt number of 8.23 was considered, for a/b = infinity and uniform heat flux on both surfaces.

$$h_{shell} = \frac{Nu_{shell} \ k}{d_{h.shell}}$$

$$h_{shell} = \frac{8.23 \times 0.659 \text{ W/m.K}}{0.00495 \text{ m}}$$

$$h_{shell} = 1095.7 \ \text{W/m}^2\text{.K} \qquad (12)$$

Table 3: Cross section and Nusselt Number (Incropera and DeWitt, 2000)



		$Nu_D = \frac{hD_h}{k}$		
Cross Section	$\frac{b}{a}$	(Uniform q''_S)	(Uniform T_S)	$f\,Re_{D_h}$
	—	4.36	3.66	64
	1.0	3.61	2.98	57
	1.43	3.73	3.08	59
	2.0	4.12	3.39	62
	3.0	4.79	3.96	69
	4.0	5.33	4.44	73
	8.0	6.49	5.60	82
	∞	8.23	7.54	96
Heated Insulated	∞	5.39	4.86	96
	—	3.11	2.49	53

After obtaining the heat transfer coefficients for the passage side and shell side of the solution heat exchanger, the overall heat transfer coefficient was calculated. Thickness of aluminum wall, thkAl.w, is 0.0015 m. Thermal conductivity of aluminum, kAl, at 65°C is 234 W/m.K.

Total heat transfer surface area of internal flow or tube can be calculated by multiplying the passage side perimeter by the length of the heat exchanger,

$$A_{.he.tube} = peri_{\cdot tube} \ l_{extru}$$

$$A_{.he.int.f} = 1.8047 \text{ m} \times 2 \times 1.219 \text{ m} = 4.4 \text{ m}^2 \qquad (13)$$

Similarly, total heat transfer surface area of external or shell side flow,

$$A_{.he.shell} = peri_{\cdot shell} \cdot l_{extru} \qquad (14)$$

$$A_{.he.shell} = 1.4 \text{ m} \times 2 \ . \ 1.219 \text{ m} = 3.4 \text{ m}^2$$

$$\frac{1}{U_{shell} A_{.he.shell}} = \frac{1}{h_{tube} A_{.he.tube}} + \frac{1}{h_{.shell} A_{.he.shell}} + \frac{thk_{Al.wall}}{k_{Al} A_{.he.shell}} \qquad (15)$$

Effectiveness of solution heat exchanger based on NTU method for heat capacity ratio equal to 1 is given by *Equation 17*.

$$C_r = \frac{C_{min}}{C_{max}} = \frac{(c_p \ mf)_{min}}{(c_p \ mf)_{max}} = 1 \qquad (16)$$

NTU-Effectiveness,

$$\varepsilon = \frac{NTU}{1 + NTU}, \ \text{for } C_r = 1 \qquad (17)$$

Mass flow rate is the same in both the tubes and shells plus cp is constant. Hence Cr is unity.

$$NTU = \frac{UA}{C_{min}} = \frac{UA}{c_p \ . \ mf_w}$$

$$NTU = \frac{1323.4 \text{ W/K}}{4187 \text{ J/kg. K} \times 0.02028 \text{ kg/s}} = 15.6$$

$$\varepsilon_{she} = \frac{15.6}{1 + 15.6} = 94\% \qquad (18)$$

The theoretical analysis of the solution heat exchangers using NTU Method shows the effectiveness of SHE can be 94%.

Testing of Solution Heat Exchanger

High temperature and low temperature solution heat exchangers were tested to determine their effectiveness and pressure drop. The experimental set up for testing the solution heat exchangers is shown in *Figure 4*. It consists of high temperature and low temperature solution heat exchangers, liquid desiccant tank, indirect evaporative cooler, electric heater, rotameter and a U-tube manometer.

Cold liquid desiccant at around 33°C from the liquid desiccant solution tank was pumped to the low temperature solution heat exchanger through a rotameter calibrated for LD, potassium formate (KCOOH) solution. Cold liquid desiccant was pre-heated in the LT_SHE by the intermediate temperature liquid desiccant from the HT_SHE. Then it was further preheated in the HT_SHE before it entered the electrical heater. In the electrical heater, liquid desiccant was heated up to 95°C. Hot liquid desiccant then exchanges heat with intermediate temperature liquid desiccant inside the HT_SHE.

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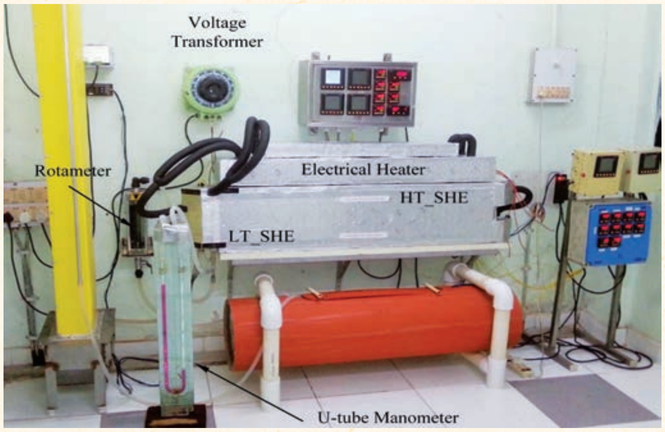
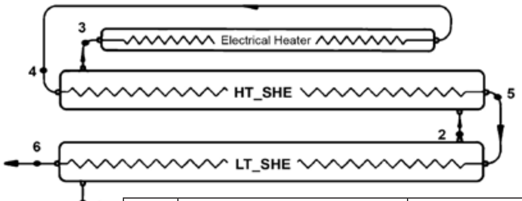


Figure 4: Solution heat exchanger experimental setup



State	Description	Nomenclature	Temp. °C
1	Cold LD at ltshe inlet	$t_{c.ld.ltshe.i}$	33.0
2	Cold LD at ltshe outlet	$t_{c.ld.ltshe.o}$	47.0
3	Intermediate LD at htshe outlet	$t_{int.ld.htshe.o}$	85.0
4	Hot LD at htshe inlet	$t_{h.ld.\ htshe.i}$	95.4
5	Hot LD at htshe outlet	$t_{h.ld.\ htshe.o}$	56.5
6	Warm LD at ltshe outlet	$t_{w.ld.ltshe.o}$	41.0

Figure 5: State points of liquid desiccant for 85% SHE effectiveness

Liquid desiccant at the outlet of LT_SHE was expected to be at 44°C for solution heat exchanger effectiveness of 80%. This liquid desiccant needed to be cooled back to 33°C in order to maintain steady state during the test. This cooling was achieved using an indirect evaporative cooler designed for this purpose.

At the design flow rate of 1.5 lpm, around 1.2 kW energy was needed to be removed to maintain steady state. Indirect evaporative cooler with a capacity of 2 kW energy was developed for this purpose. *Figure 6* shows the schematic diagram of the experimental setup along with indirect evaporative cooler, IEC.

Figure 5 shows the temperature of liquid desiccant for a mass flow rate of 1.5 lpm and 85% solution heat exchanger effectiveness. In this particular case, the indirect evaporative cooler was expected to cool the liquid desiccant at state 6, which was 41°C, to 33°C.

Listed temperatures are calculated after accounting for the lower flow rate of the hot and concentrated LD stream as compared to the higher flow rate of weak LD stream as in an actual two stage LD based dehumidification system

SHE Experimental Results

First, the test setup was run with water at different mass flow rates. Then, the solution heat exchangers were tested with liquid desiccant at 70% concentration. Required quantity of water was added to reduce the concentration of Potassium Formate, KCOOH, to 65% which was used as liquid desiccant. Concentration was determined by measuring the specific gravity. At this concentration, the solution heat exchangers were tested for effectiveness and pressure drop by varying the mass flow rate from 1 lpm to 1.5 lpm with an increment of 0.1 lpm at a time. Similarly, tests were conducted with liquid desiccant, KCOOH, at 60% of concentration.

The objective of testing the solution heat exchanger with liquid desiccant at different concentrations was to understand the effect of concentration of KCOOH on the effectiveness and pressure drop inside the solution heat exchangers. The results revealed that effectiveness did not change appreciably with concentration being changed in the range from 60% to 70%. However, the pressure drop was found to increase with the increase in concentration due to viscosity increase. Pure water had the lowest pressure drop and liquid desiccant with 70% concentration offered the highest pressure drop. *Figure 7* shows the variation of effectiveness and pressure drops for various solution concentrations and flow rates. The effectiveness of the solution heat exchanger is in the range of 85% to 92% for flow

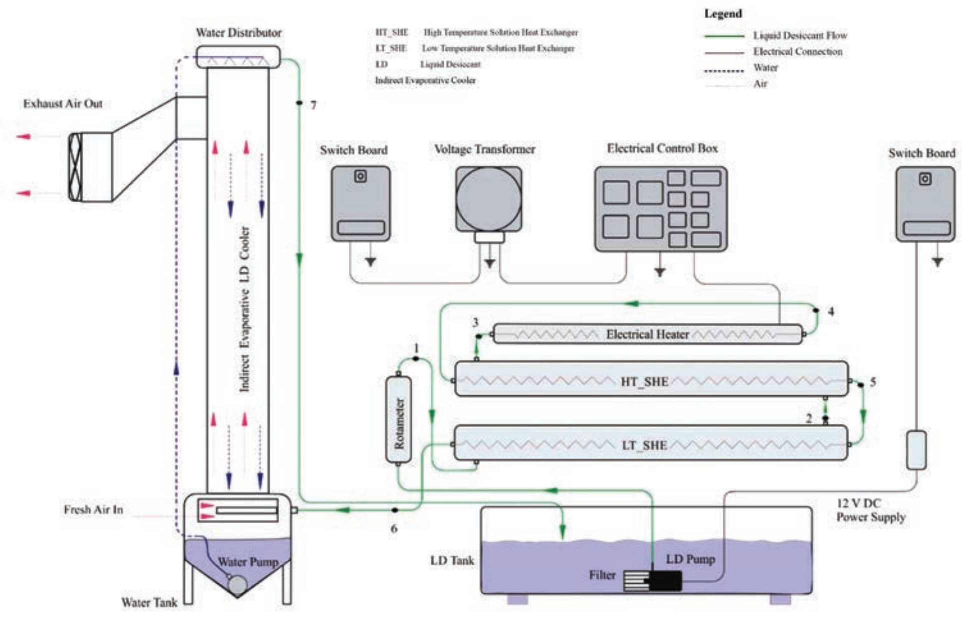


Figure 6: Schematic diagram of experimental setup for solution heat exchanger testing

rate of 1 lpm to 1.5 lpm. The pressure drop is in the range of 6 kPa to 13 kPa. These values of effectiveness are significantly better than those reported earlier and the pressure drops are very low. Low pressure drop will help design the two stage liquid desiccant system with a single solution pump.

Error in measurement of temperature when using K-Type thermocouple is the greater of ± 0.5°C or ± 0.5% of reading.

The K-Type thermocouples used for temperature measurements are off the standard type.

$$\Delta\varepsilon = 0.65\sqrt{\left(\frac{0.5}{47.9}\right)^2 + \left(\frac{0.5}{33.4}\right)^2 + \left(\frac{0.5}{55.7}\right)^2 + \left(\frac{0.5}{47.9}\right)^2}$$

$$\Delta\varepsilon = 0.014$$

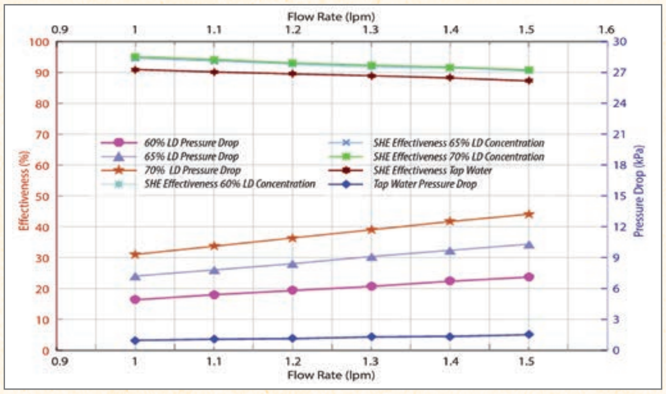


Figure 7: Effectiveness and pressure drop at different volumetric flow rates

Hence, the effectiveness of the solution heat exchanger for the volumetric flow rate between 1 lpm and 1.5 lpm is in the range of 85%±1.4 to 92±1.4%.

Conclusion

COP of a two stage liquid desiccant dehumidification system is greatly influenced by the performance of solution heat exchangers. Improvements in the solution heat exchanger effectiveness, its size, weight and cost were identified as the most important tasks in the development of 1 to 3 TR solar fresh air dehumidifier.

Development of solution heat exchanger with high effectiveness and low pressure drop is reported in this paper. Effectiveness in the range of 84 to 92% is experimentally demonstrated. Pressure drop lower than 15 kPa is realized for strong liquid desiccant concentration of 70%. This development can enable the two stage liquid desiccant based fresh air dehumidification cycles to achieve dehumidification COPs in the range of 1.6 to 2, depending upon ambient weather conditions.

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Nomenclature

A	area, m^2
Al	aluminium
C	heat capacity, kJ/K
c	specific heat capacity, kJ/(kg.K)
COP	coefficient of performance
d	diameter, m
HTR	high temperature regenerator
HT_SHE	high temperature solution heat exchanger
k	thermal conductivity, W/(m.K)
KCOOH	potassium formate
LTR	low temperature regenerator
LT_SHE	low temperature solution heat exchanger
mf	mass flow rate, kg/s
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
U	overall heat transfer coefficient, W/(m2.K)

Greek

ε	effectiveness
ρ	density
μ	dynamic viscosity

Subscript

av	average
cw	cold water
cs	cross section
e	external
l	length
ext	extrusion
i	internal
he	heat exchanger

References

1. Altener Project (2002). "Promoting solar air conditioning: Technical overview of active techniques". 4.1030/Z/02 121/2002.
2. Incropera F P and DeWitt D P, "Fundamentals of Heat and Mass Transfer", Wiley Publications Fourth Edition 2000.
3. Kaynakli O and Kilic M (2007), "Theoretical Study on the Effect of Operating Conditions on Performance of Absorption Refrigeration System" Energy Conversion and Management, vol 48 (2), pp 599-607.
4. Mehta J R (2013), "Liquid Desiccant based Solar Air Conditioning with Cogeneration of Hot Water and Potable Water", PhD Thesis, Indian Institute of Technology, Mumbai, pp 50-192.
5. Rane M V and Chavan S A (2013), "Heat and Mass Exchanger", Indian Patent Application, 1349/MUM/2013.
6. Rane M V and Chavan S A (2014), "Indirect Evaporative Precooling of Fresh Air using Heat Exchangers with Enhanced Flow Passages", International Conference on Energy and Indoor Environment for Hot Climate, Doha, Qatar, 12602.
7. Rane M V and Mehta J R (2014), "Two Stage Regeneration of Liquid Desiccant using Solar Energy", International Sorption Heat Pump Conference (ISHPC 2014), Maryland, USA, 111.