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Optimisation of a Regenerator for Open Cycle Liquid Desiccant Air Conditioning Systems

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Abstract

Solar assisted air conditioning systems using liquid desiccants represent a promising option to decrease the high summer energy demand caused by electrically driven vapour compression machines. However, for these systems high efficient and reliable components need to be developed and the design has to be adjusted to each respective building design, location, and user behaviour. The paper presents the design of a prototype for one of the main components of liquid desiccant systems and reports on its experimental performance. A numerical model has been developed for the simulation environment TRNSYS and validated using the experimental data. Using this, a system model of a single family building for heating, cooling and domestic hot water for two Australian locations has been developed. Combining this with a Powel Algorithm, the control of the cooling and dehumidification system has been optimised.

Keywords: Air conditioning, solar thermal energy, desiccant system, simulation, optimisation

1. Introduction

The energy demand for air-conditioning to provide temperature and humidity control has increased continuously throughout the last decades and is still rising. This increase is found both in commercial and residential buildings and is largely caused by increased thermal loads, residents' comfort demands, and architectural trends. Since by far most of the air-conditioning systems are electrically driven vapour compression machines, the increase is responsible for a large rise in electricity demand and especially high peak loads. The substitution of these compression machines by thermally driven cooling systems using renewable energy or waste heat is a promising alternative. In particular, due to a high correlation between solar irradiation and cooling demand for most buildings, the application of solar energy is very attractive.

The use of liquid desiccants in an open cycle system is a promising solution for solar assisted air-conditioning in humid climates or for buildings with high humidity loads. A possible system concept for this is shown in Figure 1. The main components of these air conditioning systems are the absorber, regenerator, indirect and direct evaporative cooling units for the dehumidified fresh air and heat recovery stages for both, the desiccant solution and the regeneration air. In the absorber, a hygroscopic solution, e.g. LiCl or $CaCl_2$, is directly brought in contact with fresh air, which it dehumidifies. Since during the absorption process heat is released, cooling of the process is necessary. This cooling effect can be provided either using cooling water [1], e.g. from a cooling tower, or with indirect evaporative cooling using exhaust air from the building [2]. Additional cooling of the fresh air can be achieved with direct/indirect evaporative cooling.

While absorbing the moisture, the concentration of the hygroscopic solution and thus, its capability to absorb water, decreases. This requires drying of the solution, which can be done in a regenerator, where solar or other low grade thermal energy is used to drive the process. Solution tanks for concentrated and diluted solutions offer the option to operate the system even at times when no solar energy is available. Heat recovery from the air and desiccant is necessary for the regeneration process to achieve a high coefficient of performance.

One advantage of solar thermal and especially liquid desiccant air conditioning is that the system can be integrated into a solar thermal system for domestic hot water and space heating. This integration would improve the economics for both the air conditioning part and the conventional solar thermal system. One option for this would be to use the regenerator in winter as a water-to-air heat exchanger for space heating. In this case, the absorber could provide heat recovery. Domestic hot water would be provided all year around using the solar combi-storage tank. However, for such an integration all system components and their sizes as well as the control system and flow rates have to be adjusted to each other in an optimal way and depend on the location, building designs and user behaviour.

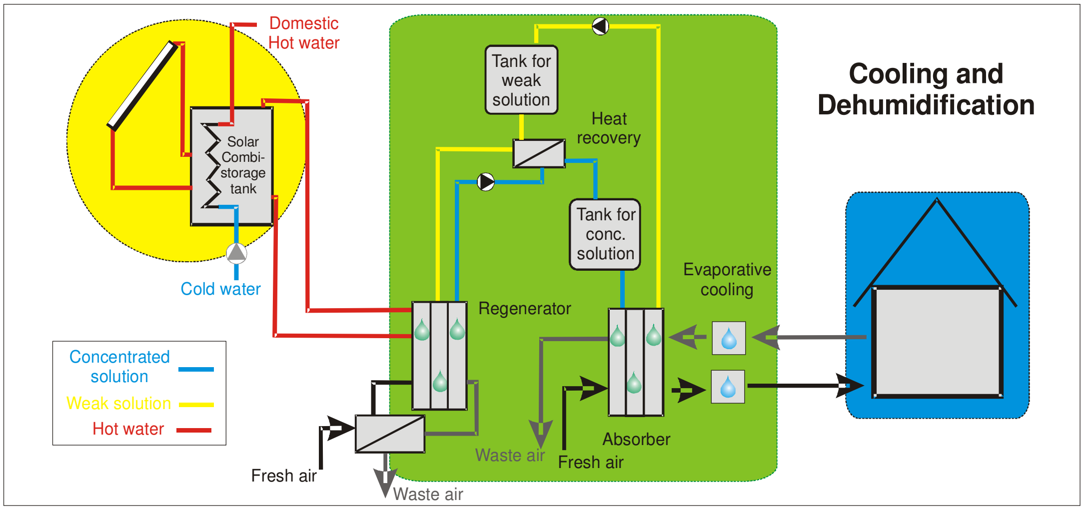

Figure 1: Design of a solar thermal system for liquid desiccant air-conditioning and the provision of domestic hot water. For additional space heating, the regenerator could be used as a water-to-air heat exchanger and the absorber could provide heat recovery. In this case, the desiccant circuit and the evaporative cooling stages would not be in operation.

2. Regenerator design

2.1 Prototype

In order to use water or water/glycol based solar collectors as the thermal heat source, a regenerator using hot water for the process is required. At present, mainly three different water driven regenerator designs have been investigated:

- Single or multi effect boiler
- Packed bed regenerators
- Plate heat exchanger designs

All these designs have incorporated the principle of vaporising water out of the solution, which is then taken away by the regeneration air. For this, either the solution or the regeneration air has to be heated to increase the vapour pressure of the solution. From the listed designs, boilers represent the simplest concept. However, they require high heating temperatures, which cannot be efficiently delivered by common flat plate collectors. Packed bed regenerators are easy to manufacture but require high desiccant and regeneration air flow rates. Such high flow rates increase the possibility of carry over. In contrast, low flow rates can be used in plate heat exchanger designs. However, since salt solutions are highly corrosive, common metal heat exchanger designs cannot be used.

Thus, a plastic regenerator prototype based on a plate heat exchanger design has been developed and tested. Figure 2 shows a photograph of the regenerator prototype, which is described in [3] in more details. Similar designs have been investigated before for water-cooled absorber constructions [1] and a second prototype is currently under construction.

Figure 2: Photograph of the regenerator prototype. Air flows from the bottom to the top in-between the plates of the regenerator, desiccant solution wets the cotton layers, introduced by a desiccant distribution system on top of each plate and hot water flows through the channels of the twin walls from the bottom of the front manifold to the top of the rear manifold in a counter/cross flow arrangement.

2.2 Model development

A numerical model for the regenerator has been developed for TRNSYS as a two-dimensional model. For this, the enthalpy balances for the hot water flow, the desiccant film and one air stream are determined for each of the elements along the channels. For each element, heat and mass transfer from the desiccant to the air stream as well as heat transfer through the heat exchanger plates are determined simultaneously. The single enthalpy equations follow approaches from [4]. The heat transfer coefficients are determined using Reynolds and Nusselt number relationships. With these coefficients, the mass transfer coefficients follow from the Lewis analogy. The same model can be used for water-cooled absorbers. For evaporative cooled absorbers a similar model was developed which is described in [5].

2.3 Comparison of model and experiments

Figure 3 shows a comparison between simulation results carried out using the TRNSYS model and experimental results of the prototype. Since the numerical model is a static model, only quasi steady state conditions can be used for comparison. From the first testing period, one data set can be used, from the second period three more sets are available. It can be seen that simulated and measured air and water temperatures show a good agreement at these points. However, the heat transfer through the heat exchanger plates is lower than expected and the parameter has been adjusted to the experimental data. This low UA-value can be explained by uneven flow distributions of air and especially water inside the regenerator. In addition, the comparison of the humidity shows that the mass transfer is overestimated in the simulations. The main reason for this would again be uneven flow distribution of the LiCl solution on the plate heat and mass exchanger plates. This has not been considered in the simulations.

3. System Simulations and Optimisations

Using the regenerator model, a TRNSYS system model for a typical Australian single-family house (ground floor: 170 m²) has been developed. The system has a solar thermal system, which provides heat for space heating, domestic hot water and for regenerating the desiccant solution in the air conditioning subsystem. To investigate the system performance in different climates, two locations have been chosen. Adelaide has a mild climate in winter and a hot and mostly dry climate in summer whereas Brisbane is also mild in winter but hot and humid in summer. A fresh air supply of 1.8 air changes per hour has been considered. Depending on whether cooling and dehumidification is required, additional recirculation air changes per hour have been introduced.

The comfort conditions have been set to 30-65 % relative humidity with a maximum absolute humidity of 12g moisture/kg dry air. The temperature has been set to 20-26°C. In case the ambient temperature exceeds 29°C, with every 3 K increase, the indoor temperature is allowed to increase 1 K [6].

In order to optimise the design and operation of the system, the TRNSYS simulation has been coupled with an optimisation tool, which provides classical and evolutionary algorithms [7]. To avoid unnecessary long optimisation times, three weeks in January, which cover cooling and dehumidification peak load periods for both locations, have been selected for the investigations. Thus, heating periods have not been considered in these investigations.

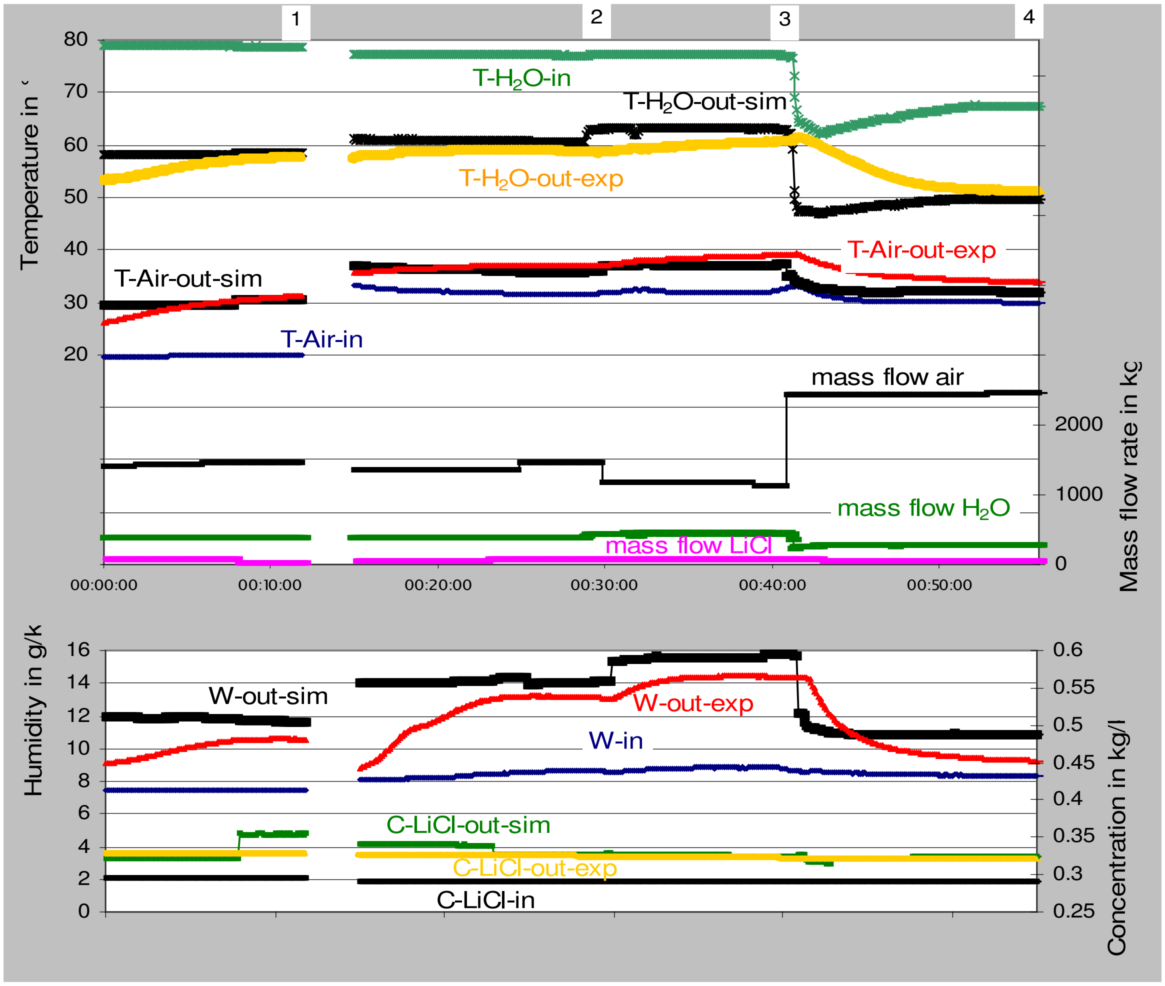

Figure 3: Comparison between simulation and experimental results of the regenerator. The top diagram shows temperatures and flow rates, the bottom diagram air humidity and solution mass concentration. The comparison consists of two testing periods, divided by the blank area in the diagram. The times to be considered for the comparison are indicated by the numbers 1 to 4 at the top of the diagram.

Since no reliable costs functions for liquid desiccant air conditioning systems were available, no economic optimisation could be performed. Thus, the sizes of all components including collector, storage tanks, heat exchangers, absorber and regenerator are kept constant. In contrast to this, all relevant air, water and desiccant flow rates through the absorber and the regenerator have been chosen as optimisation parameters with the following objective function:

$$\zeta = Q_{Aux,Reg} + \frac{1}{\eta_{Power\ \ plant}} \cdot Q_{Electric} + F_{Penalty} \cdot Q_{Aux,Comfort} \qquad (1)$$

The first term in equation (1) represents the auxiliary heat demand for the domestic hot water storage, the second term considers the electricity consumption of the pumps and fans based on estimated pressure losses in the system, and the third term stands for the remaining energy demand required to meet the desired comfort conditions. The latter has been weighted with a penalty factor. Since the objective function was expected to be smooth with the parameter variations, a Powell-Algorithm [8] has been used for the optimisations.

Table 1 shows some of the optimised parameters as well as the related energies for two different $F_{Penalty}$ -factors for Brisbane. It can be seen that depending on the necessity to strictly maintain comfort conditions, the system needs to be operated with different flow rates. Table 2 indicates that especially an increase of the air recirculation flow rate through the absorber helps to maintain the indoor temperature within the limits for most of the times. The fractions of times when these conditions cannot be achieved are reduced to 9.2 % basing on a $F_{Penalty}$ -factor of 3. However, this leads to a big increase of the required dehumidification demand due to more evaporative cooling and thus a much higher overall energy demand. Because of this, the solar fraction related to only the thermal auxiliary energy demand decreases from 82 % for a $F_{Penalty}$ -factor of 3 to 60 % for a factor of 30. In addition, for a high penalty factor, the absorber desiccant flow rate needs to be much higher. Together with a much lower desiccant flow rate in the regenerator the average operating desiccant concentration level increases within the whole system by about 3 %(weight).

$F_{Penalty}$	3	30
$F_{Cooling}$ in 1/h	0.2	0.4
$\dot{m}_{Des,absorber}$ in kg/h	44.2	77.0
$\dot{m}_{Des,regenerator}$ in kg/h	117.4	50.3
$Q_{Collector}$ in MJ	7020	6592
$Q_{Regenerator}$ in MJ	6771	9096
$Q_{Aux,tank}$ in MJ	1220	4289
$Q_{Aux,Comfort,sensible}$ in MJ	475	120
$Q_{Aux,Comfort,latent}$ in MJ	85	99
Q_{Elec} in MJ	464	547

Table 1: Optimised parameters and energy values for two different penalty factors

$F_{Penalty}$	3	30
Max Delta T	3.5 K	2.5 K
Fraction of no comfort	29.0 %	9.2 %
Max absolute Humidity	11.8 g/kg	15.7 g/kg
Max Relative Humidity	74 %	78 %
Fraction due to Humidity	0 %	1.3 %
Fract. due to rel. Humidity	0.4 %	1.3 %
Fract. due to Temperature	28.6 %	7.7 %

Table 2: Deviations from comfort conditions and fractions, when comfort conditions cannot be achieved.

Investigations for Adelaide show that for the same period of high peak loads and even with a low penalty factor, only at 0.8 % of the times the indoor conditions exceeded the desired comfort conditions. The maximum temperature deviation from the set conditions was 1.1 K. However, simulations without a liquid desiccant system but only indirect and direct evaporative cooling showed that the indoor temperature can be kept at all times below the set conditions. In this case, at 1.5 % of the times, the humidity in the building exceeded the limits with a maximum absolute humidity of 14.2 g/kg and a maximum relative humidity of 77.6 %.

4. Conclusion

In order to optimise liquid desiccant air conditioning systems and their control scheme, the desired level of comfort is one of the crucial factors. Especially in humid climates, there is a basic conflict between humidification and cooling, if direct evaporative cooling is used. To reduce both, humidity and temperature, the required regeneration energy increases significantly. In hot but dry climate like Adelaide, direct and indirect evaporative cooling stages are sufficient to guarantee comfort conditions for almost the whole time. Thus, for residential houses, a liquid desiccant system would not be necessary.

To generalise the results furthermore, a second prototype is currently under construction and will be tested with a higher variation of operating conditions. In order to design the system, the optimisations need to be extended to all component sizes. For this, economic issues need to be considered as well. Based on the economics, it has to be decided, if for instance components like desiccant tanks or heat recovery heat exchanger are essential for a good operation of the system.

5. Acknowledgment

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Nomenclature

W	Air humidity (g moisture/ kg dry air)
C	Solution concentration (kg salt/ kg solution)
$F_{cooling}$	Additional air recirculation flow rate if cooling is required (1/h)
$F_{Penalty}$	Penalty factor (-)
$\dot{m}_{des,absorber}$	Desiccant mass flow rate through the absorber (kg/h)
$Q_{Collector}$	Collector gain (MJ)
$Q_{Regnerator}$	Energy used for the regeneration (MJ)
$Q_{Aux,tank}$	Auxiliary energy delivered to the hot water tank (MJ)
$Q_{Aux,Comfort,sensible}$	Auxiliary energy required to meet the sensible comfort conditions (MJ)
$Q_{Aux,Comfort,latent}$	Auxiliary energy required to meet the latent comfort conditions (MJ)
Q_{Elec}	Electric energy consumption of pumps and fans (MJ)
T	Temperature (°C)
$\dot{m}_{des,regenerator}$	Desiccant mass flow rate through the regenerator (kg/h)
ζ	Objective function (MJ)
$\eta_{Power\ \ plant}$	Efficiency of electricity supply from power plant and grid (-)

References

[1] Lävemann, E., Peltzer, M. (2003): *Solar Liquid Desiccant cooling System Demonstration Plant*, ISES Solar World Congress, Goeteborg, 14.-19.6.2003.

[2] Saman, W. Y., and Alizadeh, S. (2000): *An Experimental Study of a Cross-flow Type Heat Exchanger for Dehumidification/Cooling,* Proc 38th Annual Conference of the Australian and New Zealand Solar Energy Society.829-836.

[3] Krause, M., Saman, W., Vajen, K. (2005): *Open Cycle Liquid Desiccant Air Conditioning Systems – Theoretical and Experimental Investigations,* 43rd ANZSES Conference Solar 2005, 28.11.-30.11.05, Dunedin, New Zealand

[4] Khan, A., Sulsona, F.: *Modelling and parametric analysis of heat and mass transfer performance of refrigerant cooled liquid desiccant absorbers*, International Journal of Energy Research, **22**, 1998.

[5] Krause, M., Saman, W., Vajen, K. (2005): *Regenerator Design for Open Cycle Liquid Desiccant Systems – Theoretical and Experimental Investigations,* 1. International Conference Solar Air Conditioning, 6.-7.10..2005, Staffelstein, Germany

[6] DIN 1946 Part 2 (1994). Raumlufttechnik, Gesundheitstechnische Anforderungen (VDI-Lüftungsregeln), Beuth Verlag GmbH Berlin.

[7] Krause, M., Vajen, K., Wiese, F., Ackermann, H. (2002): *Investigations on Optimizing Large Solar Thermal Systems,* Solar Energy Journal, Vol (73), No. 4, pp. 217-225, 2002

[8] Press, W.H., Teukolsky, S.A., Vetterling, W.T., Flannery, B.P. (1997), *Numerical Recipes in C: The Art of Scientific Computing,* Cambridge University Press, Cambridge