HYBRID LIQUID DESICCANT SYSTEM: DESIGN AND SIMULATION MODELS AND EXPERIMENTAL VALIDATION

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ABSTRACT

The treatment of humidity on HVAC systems is crucial when a satisfactory indoor air quality needs to be reached. Traditional HVAC systems meet the latent cooling load by reducing the air temperature until its dew point, heating subsequently the air in order to reach the supply temperature for user comfort, with the energy waste this entails. On the other hand desiccant wheels requires normally an excessive post-cooling because of employed regeneration temperatures of around 70-80 ºC.

In this paper the design and simulation models and testing results at laboratory scale of a hybrid liquid desiccant system (HLDS), developed in the frame of the EU project nanoCOOl, are presented. The HLDS is especially suited for applications with a low SHR (Sensible Heat ratio) and high ventilation requirements in tropical or subtropical climates.

The aim of the project is to validate the developed technology for a good indoor environment quality, achieving the required ventilation needs, a good occupant comfort by the treatment of temperature and humidity to reach comfort conditions, avoiding the generation of moulds and microbial growth due to the antimicrobial properties of the LiCl.

Detailed models of the HLDS components have been implemented in Engineering Equation Solver (EES) [1], and the whole model of the prototype has been developed, as well. The key parameters for the simulated HLDS, H&MTC (Heat&Mass transfer coefficients) have been experimentally obtained, testing the proof of concept absorber /regenerator in a test bench specially developed at laboratory scale[2]. The obtained values are in agreement with the correlations proposed by Bykov [3] (HTC) and Queiroz [4] (MTC).

Keywords:: Liquid desiccant, HVAC, Falling film,.

1. Introduction

Hybrid liquid desiccant systems (HLDS) combine the liquid desiccant technology for dehumidification of air with conventional compression cycle technology for cooling. The liquid desiccant cycle is designed for meeting the latent cooling load (dehumidification) and possibly part of the sensible cooling load, and the vapour compression cycle provides sensible cooling. They are suitable for diverse applications, such as air conditioning in highly humid climates, like tropical or sub-tropical humid climates [5]. Hybrid air conditioning systems have some advantages. In these systems the regeneration temperature of the liquid desiccant can be lowered from 70-80 ºC to 50-60 ºC, according to literature [6]. Moreover, the size of the cooling coil can be reduced, ideally providing only sensible cooling, having the air dehumidified to the required level in the liquid desiccant system (LDS).

In this paper the design and simulation of a hybrid liquid desiccant system for a case study in Taiwan is presented. The designed system will be constructed and tested first at laboratory scale and after as a demonstrator in Taiwan Building Technology Center in Taipei, in the scope of the Nanocool project.

The demo site comprises two locker rooms in a swimming pool of the National Taiwan University of Science and Technology, with high internal humidity generation, low sensible heat ratio, and high external humidity levels due to sub-tropical humid climate present in Taiwan.

The designed hybrid liquid desiccant system (HLDS) is comprised by a liquid desiccant system (LDS) whose main components are the absorber, regenerator and liquid-liquid heat exchanger; and a conventional Air Handling Unit (AHU) with a cooling coil and a cross-plate heat exchanger for ventilation heat recovery. A polyvalent unit able to simultaneously provide cooling and heating, feeds the absorber and the cooling coil with cold water at 15ºC, and the regenerator with hot water at 55ºC.

First of all, the load calculation for the locker rooms has been performed, in order to size the components of the HLDS. The overall architecture of the HLDS has been defined, and simulation models for the individual components and a global model have been developed; for the sizing of components, selection of commercial components and design of absorber and regenerator. Sensitivity analyses have been carried out in order to define the optimal working conditions of the HLDS.

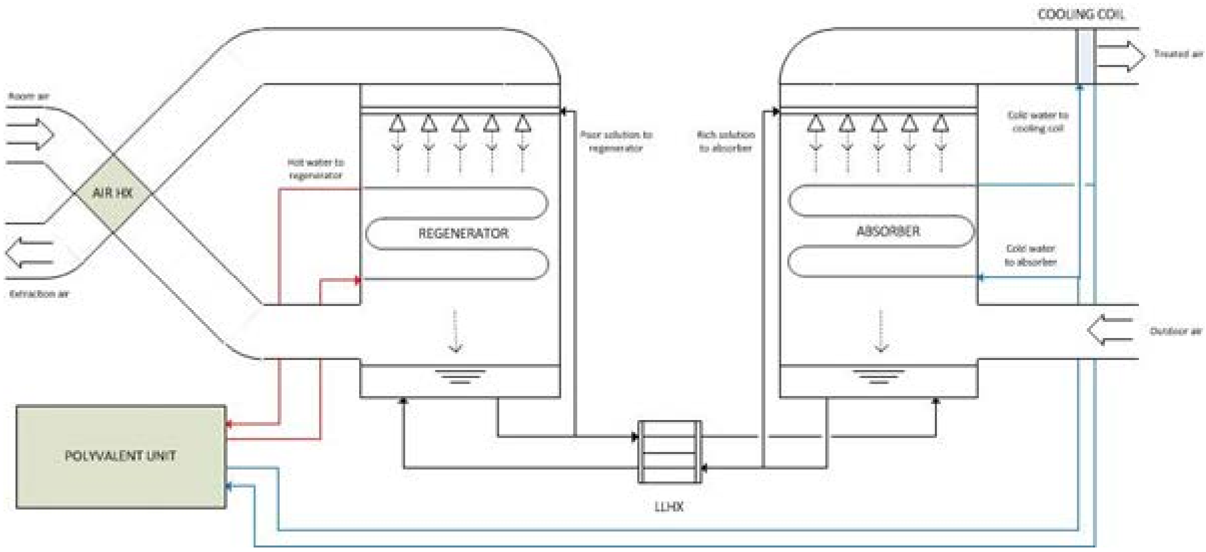

Figure 1.Hybrid liquid desiccant cooling system.

2. Case study

The hybrid liquid desiccant system (HLDS) is designed for air treatment in the locker rooms of a swimming pool of the Taiwan Building Technology Center in Taipei. The case study characterizes for having a high internal latent heat generation during use and much lower sensible load, leading to a low sensible heat ratio. The high external humidity levels impede the use of external air to lower humidity levels in the rooms, making use of the so-called free cooling effect.

Based on the design conditions, the internal sensible and latent heat generation, and the ventilation requirements according to international standards; the cooling and dehumidification loads have been calculated.

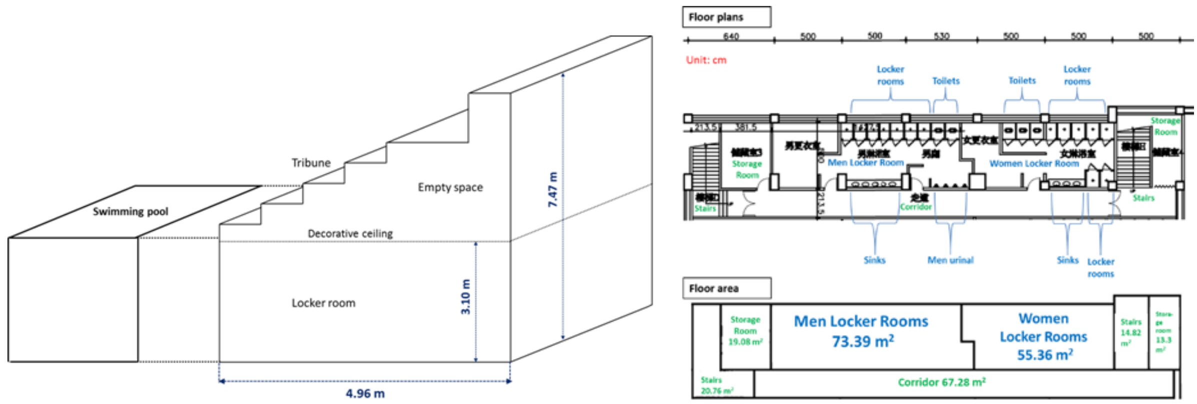

Figure 2. Location and distribution of the locker rooms of Taiwan Building Technology Center in Taipei

The design conditions and the cooling and dehumidification loads are presented below:

Table 1: Design conditions and loads.

Outdoor design conditions	30ºC / 21,5 g/kg dry air
Comfort design conditions	24ºC / 11,24 g/kg dry air
Ventilation rate	2500 m^3/h
Internal sensible heat load	3,53 kW
Ventilation sensible heat load	4,99 kW
Carga Sensible Total	8,53 kW
Cargainternalatente	8,79 kW
Cargalatente de ventilación	21,26 kW
Cargalatente Total	30,05 kW

3. Design and Simulation

Models of the individual components of the Liquid Desiccant System (absorber, regenerator and liquid-liquid heat exchanger) and the Air Handling Unit (cooling coil, air-air plate heat exchanger) have been implemented in EES (Engineering Equation Solver) [1]. By combining the separate models of each component, a versatile mathematical model has been developed, which can be used for both the design and simulation of the HLDS.

3.1. Liquid Desiccant System

The absorber and the regenerator are falling film type, internally cooled and heated respectively. They are comprised by a polypropylene tube bundle, a liquid distributor system with spray nozzles, and a demister inside a fiber glass tower. The tube bundles are formed of individual modules of 98 tubes. The modules are linked horizontally in threes, and then vertically to form 14passes, with a total area of 53.8 m^2. A good wettability of the tubes is key for obtaining good performance in the liquid desiccant cycle, for that reason the polypropylene tubes have received a plasma treatment in order to improve their wettability [7].

The air flows from bottom to top getting in contact with the descendent lithium chloride solution, which forms a falling film outside the tubes. The rich LiCl solution absorbs humidity from air in the absorber, and the poor LiCl solution desorbs humidity enriching the solution in the regenerator. Cold water (15ºC) and hot water (55ºC) flows inside the tubes, cooling the solution and the air in the absorber, and heating the solution and the air in the regenerator.

Models for the absorber and regenerator are based on the theoretical model proposed by Hellmann and Grossmann [8] and recently revised by Gommed and Grossman [9]. Water and salt mass balances, energy balances, thermodynamic equilibrium and heat and mass transfer equations are included in the model. The overall architecture of the subroutine of the air-solution contactors (Absorber/Regenerator) which are the most critical components in the LDC is presented in the following flow diagram:

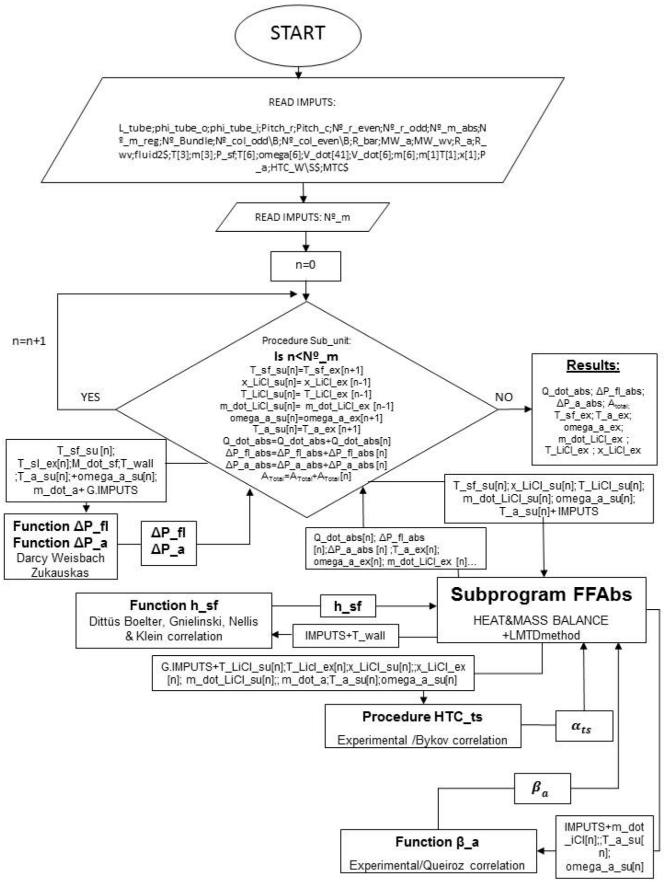

Figure 3.Architecture of the LDC model.

Heat and mass transfer coefficients involved in the modelling are the global heat transfer coefficient between external medium and the solution (U), the heat and mass transfer coefficients between air-solution interface and the air stream (α_a, β_a), and the heat and mass transfer coefficients between air-solution interface and the solution stream (α_{IS}, β_{IS}).

The heat transfer coefficient between external medium and the solution stream is calculated as:

$$U_{Abs} = \frac{1}{\frac{\phi_{tube\_o}}{\phi_{tube\_i}} \cdot \frac{1}{\alpha_{sf}} + \frac{\left(\frac{\phi_{tube\_o}}{2} \cdot Ln\left(\frac{\phi_{tube\_o}}{\phi_{tube\_i}}\right)\right)}{\lambda_{tube}} + \frac{1}{\alpha_{ts}}}$$

The water-tube heat transfer coefficient (α_{sf}) has been calculated based on Nusselt number, depending on the flow regime by using correlations described by Nellis& Klein [10]. For laminar region Re<2300,

$$Nu = 4,36 + \frac{(0,1156 \; + 0,08569/Pr^{0,4}) \cdot Gz}{1 + 0,1158 \cdot Gz^{0,6}}$$

The Gnielinski equation for fully developed flow for 2300<Re<10000, using the friction factor from ref. [8],

$$ff = \left(-\frac{0,001570232}{\ln(Re_{sf})} + \frac{0,394203137}{\ln(Re_{sf})^2} + \frac{2,534153311}{\ln(Re_{sf})^3}\right) \cdot 4 \qquad Nu = \frac{\left(\frac{ff}{8}\right) \cdot (Re_{sf} - 1000) \cdot Pr_{sf}}{1 + 12,7 \cdot \sqrt{\frac{ff}{8}} \cdot \left(Pr_{sf}{}^{\left(\frac{2}{3}\right)} - 1\right)} \cdot \left(\frac{\mu_{sf}}{\mu_{sf_{wall}}}\right)^{0,14}$$

And the Dittus-Boelter equation for Re>10000,

$$Nu = 0,023 \cdot Re_{sf}{}^{0,8} \cdot Pr_{sf}{}^{n} \cdot \left(\frac{\mu_{sf}}{\mu_{sf_{wall}}}\right)^{0,14}$$

$n = 0,4$ for heating of the fluid (absorber)

$n = 0,3$ for cooling of the fluid (regenerator)

The tube-solution heat transfer coefficient (α_{ts}) and the mass transfer coefficient between air-solution interface and the air stream (β_a) have been obtained from experimental results of the field tests developed in Technion (Israel Institute of Technology) in Haifa [2]. In the field tests a Liquid Desiccant System was experimentally tested using different types of tubes in absorber mode. The performance with plastic tubes, which showed poor wettability, was lower than with the titanium tubes, which presented good wettability. The designed HLDS will be using polypropylene tubes which have received a plasma treatment in order to increase their wettability [7]. Therefore, the experimental values used for the modelling correspond to the experiments carried out with titanium tubes. In addition, in order to have a model available to represent the performance of the Absorption/desorption processes out of the working conditions of the experiments carried out on Ref. [2] the model has been improved including the correlations proposed by Bykov [3] (α_{ts}) and Queiroz [4] (β_a) that faithfullyrepresent the performance of the Absorber/Regenerator and the obtained values are in agreement with the experimentally obtained α_{ts} and β_a values.

The implementedHTC correlationproposed by Bykovet al. [3] depends on the air Reynolds number and solution Prandtl and Reynolds numbers.

- 690 $\leq Re_{air} < 3000$

$$Nu_s = 3{,}3 \cdot 10^{(-3)} \cdot \mathrm{Re_s}^{0{,}3} \cdot \mathrm{Re}_a{}^{0{,}15} \cdot Pr_{\mathrm{s}}{}^{0{,}61}$$

- 3000 ≤R_{eair}<6900

$$Nu_s = 1{,}1 \cdot 10^{(-2)} \cdot \mathrm{Re_s}^{0{,}3} \cdot Pr_{\mathrm{s}}{}^{0{,}62}$$

- Re_{air} ≥6900

$$\alpha_{ts} = \frac{\mathrm{Nu_s}}{(\bar{\nu}_{\mathrm{s}}{}^2/g)^{(1/3)}} \cdot \bar{\lambda}_{\mathrm{s}}$$

$$Nu_s = 0{,}24 \cdot \mathrm{Re_s}^{0{,}3} \cdot \mathrm{Re}_a{}^{(-0{,}36)} \cdot Pr_{\mathrm{s}}{}^{0{,}66}$$

In addition, a procedure has been incorporated in the developed EES model in order to calculate the MTC in the air solution interfaceeither using the experimentally obtained value [2] or using the correlation developed by Queiroz et al.[4]:

$$\beta_a = \frac{\left(1{,}53 \cdot \left(\frac{\dot{m}_a}{A_{pass\ min}}\right)^{(1{,}33)} \cdot \left(\frac{\dot{m}_{LiCl_{ex}}}{\dot{m}_a}\right)^{(0{,}564)}\right)}{Specific_{Area}}$$

The values obtained experimentally in Ref. [2] have been compared against the values obtained from the implemented correlations for the same working conditions in the absorber, the results of the mentioned comparison are presented in the following table:

Table 2: Calculated/Experimental H&MTC, Absorber.

Item	Calculated Ref. [3]; Ref. [4]	Experimental
Resulting solution-tube thermal resistance [m^2 K/W]	1.54-03	1.48E-03
Resulting solution-tube Heat transfer coefficient [W/ m^2 K]	650	675.67
Mass transfer coefficient between air-solution interface and the air stream (kg/m^2s).	0.06	0.05

Assumption of a Lewis factor equal to one is considered, which is applicable for air.

$$Le = \alpha_a / \beta_a \cdot c_{p_{air}}$$

The liquid-liquid heat exchanger is a plate heat exchanger made of a polymeric matrix composite, including graphene nanoparticles, from SGL Company. It is used to precool the solution going to the absorber, and to preheat the solution going to the regenerator. Its model is based on the ε-NTU method with correlations for heat transfer in plate heat exchangers described by H. Martin [11].

The model has been contrasted with the design data from SGL. Heat exchanger effectiveness for design working conditions is 0.85.

$$Nu_s = 1{,}615 \cdot \left(ff_s \cdot \frac{Re_s}{64} \cdot Re_s \cdot Pr_s \cdot \frac{D_h}{L} \right)^{1/3} \cdot \left(\frac{\mu_s}{\mu_{s_{wall}}} \right)^{\frac{1}{6}}$$

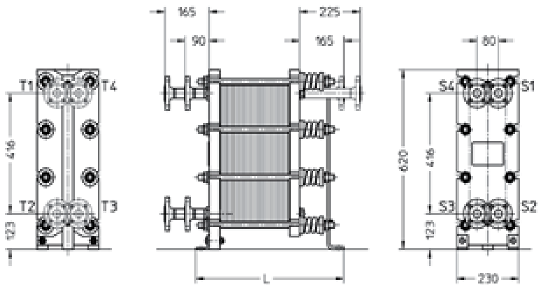

Figure 4: Liquid-liquid heat exchanger. SGL

3.2. Air Handling Unit and polyvalent unit

The air handling unit comprises the air plate heat exchanger for heat recovery of ventilation, the cooling coil, and the corresponding fans, dampers and filters for the system. The casing and the arrangement of the different elements will be specially designed for its optimal connection with the LDS.

The air heat exchanger is a compact plate heat exchanger, with cross-flow configuration, made of aluminum and with internal fins to increase the heat transfer between both air streams. The use of such equipment enables considerable savings to be achieved in the operating costs of air-conditioning plants, and thus the saving of energy that would otherwise be lost. In the HLDS it is used for preheating the air entering the regenerator. It is modelled by using the ε-NTU method.

The cooling coil has been modelled by using the model described in [12]. This model is based on epsilon-NTU and LMTD and LMHD equations and uses Braun's hypothesis to simulate the behavior of the cooling coil in an AHU (air handling unit). Normally, cooling coils in Air Handling Units deal with sensible and latent cooling loads, condensing water from air in the surface of the tubes. In this case, the cooling coil serves for dealing only with sensible cooling loads; no condensation will occur in the surface of the tubes, so in theory the dry cooling coil approach could be used for the modelling. However, the model includes also the wet coil approach, in case the cooling coil may be used for dehumidification as well. In that case, the cooling coil should be oversized respect to the actual size.

For the air side the correlation proposed by Wang et al.[13] is selected and the HTC validated against a reference HX calculation software; on the other hand for the calculation of the tube side coefficient the correlation proposed by Gnielinski [14] for turbulent region and the described on Nellis& Klein [6] for the laminar one as in the case of the absorber and regenerator have been used.

The polyvalent unit is a heat pump able to provide heating and cooling simultaneously, by recovering heat from a water condenser when the machine is working on dual mode, and by condensing with air when the machine is working on cooling mode. Therefore, it will be used in cooling mode when the LDS is only dehumidifying and cooling (regenerator off), or in dual mode when the system is regenerating LiCl solution as well, with no need from another external source of heat.

3.3. Whole cycle modelling and results

The completed individual models of the LDS (absorber, regenerator and liquid-liquid heat exchanger),

and the AHU (air heat exchanger and cooling coil), have been combined to develop the complete model of the HLDS. A split system is included at the solution outlet of the absorber and regenerator, so that the amount of solution which is recirculated to the absorber and regenerator from the bottom of each tower is set to 0,9 of the outlet solution from each component; sending 10% of the solution through the liquid-liquid heat exchanger. The figure below shows the results from the modelling of the whole HLDS.

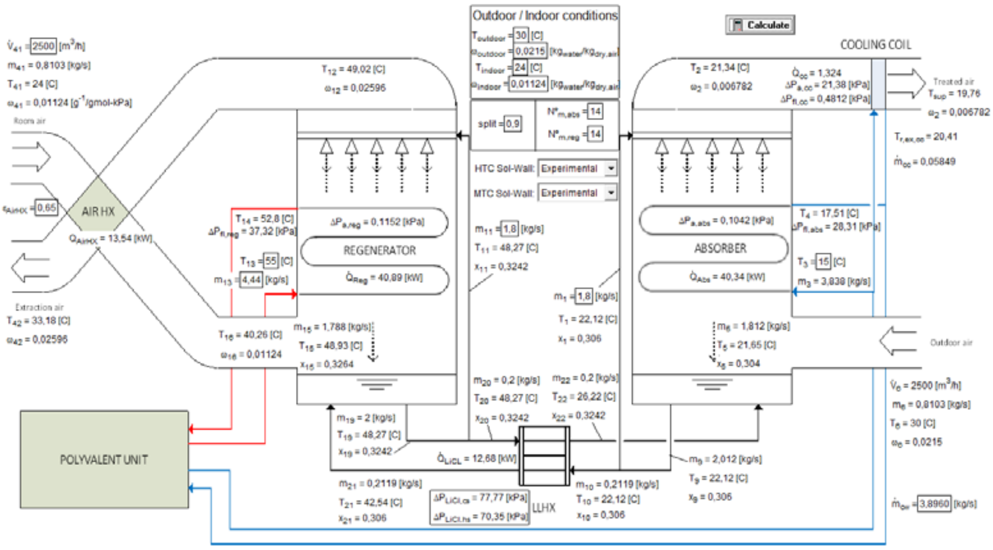

Figure 5: Results from simulation of the HLDS.

Calculated impulsion air conditions are 19.75ºC and a humidity ratio of 6.8 g/kg dry air. In order to achieve these conditions, the required capacity of the absorber and regenerator is 39.78 kW and 40.75 kW respectively, with a 12 kW capacity liquid-liquid heat exchanger. The air plate heat exchanger has a capacity of 13.4 kW. The cooling coil needs to provide 1.85 kW in the design conditions, however for the construction of the prototype it will be oversized in order to be able to deliver the necessary cooling to meet the sensible cooling load, which leads to a 9 kW cooling coil. Considering the electrical consumption of the polyvalent unit, the HLDS has a COP of 2.7.

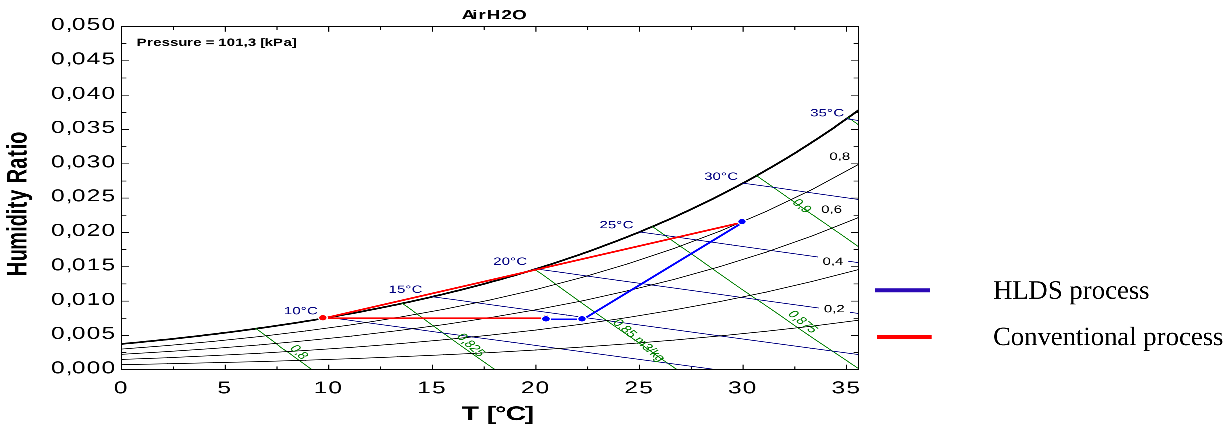

Figure 6: Air treatment processes in the HLDS and in a conventional cooling system.

A representation of the HLDS process in a psychometric chart is shown below, as well as the corresponding process to the treatment of air with a conventional system. In the conventional system, the air needs to be cooled down to a high extent in the cooling coil to reach the required dehumidification. In order to achieve comfort indoor conditions, the air needs to be reheated in a reheating coil. With the HLDS process, the need for reheating is eliminated, which leads to important energy savings. Taking into account the reheating consumption in the conventional system, an improvement of around 45% in COP is achieved.

COP of the HLDS and conventional system are calculated as:

$$COP_{HLDS} = \frac{Cooling\ effect}{Energy\ consumed\ by\ polyvalent\ unit}$$

$$COP_{conv} = \frac{Cooling\ effect}{\frac{Q_{cc}}{COP_{heat\ pump}} + Q_{reheat}}$$

4. Future developments

Based on the cooling loads calculation, a hybrid liquid desiccant system has been designed and simulated for a case study in Taiwan, sizing every component to match the sensible and latent cooling loads. At this stage of the project, the whole prototype has been alreadymanufactured according to the 3D CAD developed and the control algorithm is already implemented in the PLC. An image of the of the whole HLDS placed in the laboratory is shown below.

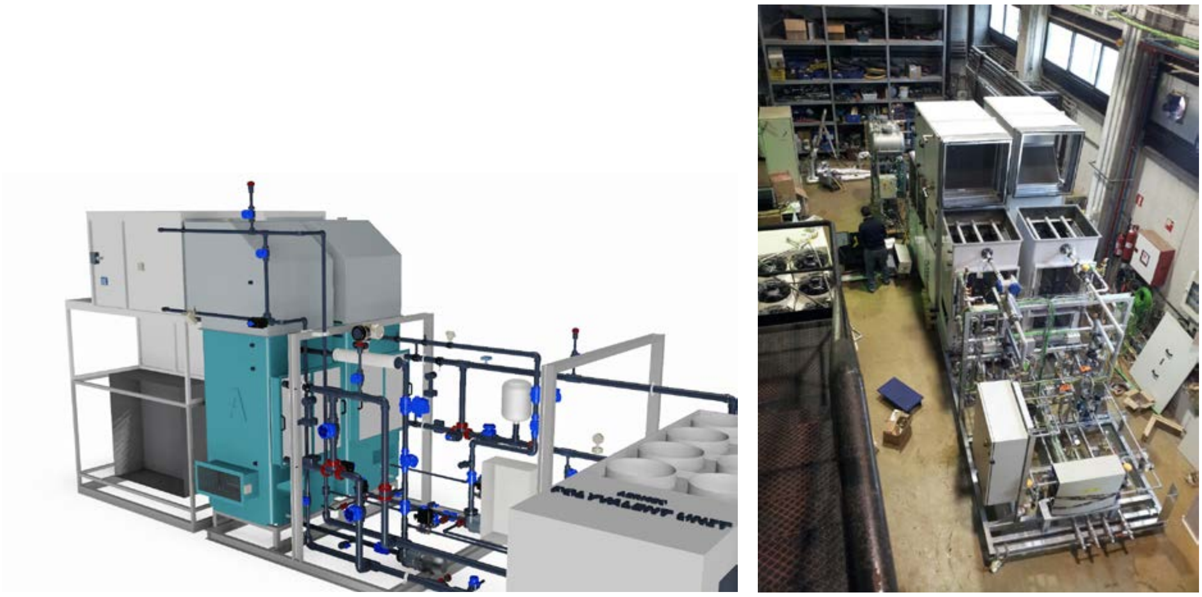

Figure 7:3D CAD design and prototype of the HLDS in the laboratory

The HLDS is already built and it is already in the laboratory after some preliminary test carried out in the different parts of the prototype, i.e.: pressure test ensuring the absence of leakages, double-check of sensors and actuators and flow circulation test. Actually the laboratory scale testing has been already started firstly in a manual way and afterwards checking the implemented control algorithm, for validating its operation, optimizing the design and refining the control strategy. After that, it will be sent to Taipei for its testing in continuous operation under real conditions in the selected demo site, for a period of time of six months.

5. Acknowledgement

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6. Nomenclature

$HLDS$ Hybrid Liquid Desiccant System.

LDS LiquidDesiccantsystem

AHU Air Handling Unit

$H\&MTC$ Heat&Mass transfer coefficients.

HTC Heattransfer coefficient

MTC Mass transfer coefficient

U Global heat transfer coefficient between external medium and solution (W/m^{2}K)

α_a Heat transfer coefficient between air-solution interface and the air stream (W/m^{2}K).

β_a Mass transfer coefficient between air-solution interface and the air stream (kg/m^{2}s).

α_{ts} Tube Wall/Solution streamHeat transfer coefficient (W/m^{2}K).

β_{IS} mass transfer coefficient between air-solution interface and the solution stream (kg/m2s).

α_{sf} Secondary fluid tube heat transfer coefficient (W m^{-2} K^{-1}).

Le Lewis number.

ϕ Diámeter (m).

λ Thermal conductivity (W m^{-1} K^{-1}).

Re Reynolds number.

Nu Nusselt number.

Pr Prandtl number.

Gz Graetz number.

ff Friction factor.

μ Dynamic viscosity (Pa s).

ν Kinematic viscosity (m^2 s^{-1})

g Gravity (m s^{-2}).

ρ Density (kg m^{-3}).

D_h Hydraulic diameter (m)

L Length of plate (LLHX)

ε Effectiveness

NTU Number of Transfer Units

$\dot{Q}$ Power (kW)

COP Coefficient of performance

Subscripts

su Supply

ex Exhaust

a air

s solution

sf Secondary fluid.

IS Interface solution.

i Interface.

Abs Absorber

ts Tube-solution.

o Outer.

i Inside.

cc Coolingcoil

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