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SIMULATION AS A DESIGN TOOL FOR SOLAR COOLING SYSTEMS IN HUMID, TROPICAL CLIMATES

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ABSTRACT

This paper presents an optimised solar cooling system design for residential scale buildings in humid climates. A full simulation model was developed to study the key parameters for a 5 kW ammonia-water absorption cooling system. The system was modelled specifically to take into account the humidity of the air and its effects on system efficiency. In the field of solar cooling, humidity is not frequently studied, but it quite *significantly* affects system performance. In this study, it was found that if moisture is taken into account, 20 m of collectors and 0.6 m^3 of both hot and cold-water tank storage are needed. Thus, a larger collector array is required than reported in frequent studies which rarely consider humidity.

INTRODUCTION

Energy consumption has increased in recent years with the development of the global economy, increasing population, and the constantly growing demand for comfort in buildings. The energy consumed by buildings accounts for 40% of total primary energy consumption in IEA countries (Wang et al. 2009). This amount of energy is mainly used for air conditioning indoor spaces, heating water, electrical appliances and lighting (European Commission 2010). The continued rise in the living and working comfort conditions combined with reduced air conditioning unit price, has led to fast proliferation of these systems. In fact, energy demand for heating is projected to increase until at least 2030. The growth is likely to be even more dramatic for cooling – an area which is may increase rapidly over the current century due to climate warming. For example, the number of installed air-conditioning systems in Europe with cooling capacity over 12kW has increased by a factor of 5 in the last 20 years (Peel, Finlayson, and Mc Mahon 2007).

Tropical climates may be an even larger source of growth where population and the mean annual temperature is high (Fong et al. 2010). This makes the development of new functional technologies for those climates particularly interesting because of the huge potential market. Solar cooling, in particular, provides an interesting engineering and economic opportunity for these regions. Tropical climates are some of the most favourable in the world for solar cooling since they are blessed with excellent solar resources (National Renewable Energy Laboratory 2011).

Presently, the most advanced solar cooling thermal technology (in terms of global price and general performance) is absorption refrigeration. Single effect absorption systems have a *thermal* Coefficient of Performance (COP) near 0.7 as compared to the 0.4 that is achievable with adsorption technology (Lecuona et al. 2009). Thus, absorption cooling was chosen for this study.

BACKGROUND

Many studies have been carried out for solar cooling in continental climates and sub-tropical climates, but much less work has been dedicated to tropical climates. Climates are generally separated into several groups by the "Köppen-Geiger classification" (Peel et al, 2007). The main categories are Tropical, Arid, Temperate, Cold, and Polar. These categories are differentiated by their coldest and warmest month mean temperatures. Each main category is also sub-divided into groups defined by precipitation as shown in table 1.

Table 1: Characteristics of tropical climates

CLIMATE	CHARACTERISTICS
Rainforest (Af)	Precipitation of the driest month superior to 60 millimetres
Monsoon (Am)	Precipitation of the driest month superior to "100 minus mean annual precipitation divided by 25" millimetres, it has not to be a rainforest.
Savannah (Aw)	Precipitation of the driest month inferior to "100 minus mean annual precipitation divided by 25" millimetres, it has not to be a rainforest.

The three climates shown in the table are typical of the equatorial, tropical zones. We chose to focus on South Asia and North Australia, selecting three populated areas (cities) for market considerations.

- Darwin: 100 000 inhabitants, Savannah
- Cairns area: 100 000 inhabitants, Monsoon

- Singapore: 26 000 000 inhabitants, Rainforest

For these cities, figures 1 and 2 show beam and diffuse radiation and relative humidity and temperature evolution through one year. The curves result from a moving average of four days plus a sixth degree polynomial interpolation based on TMY-2 files used on TRNSYS. Original data provided by Solar Radiation Data Base files.

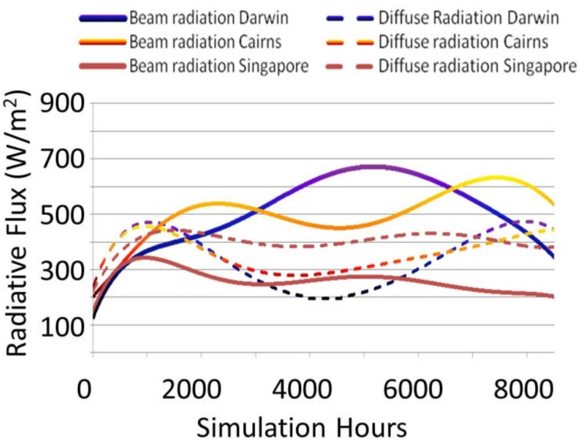

Fig. 1 - Solar radiation in three different locations

Figure 1 shows that diffuse radiation is in the range of 200 to 700 kJ/hr.m throughout the year in all three cities. In Singapore, diffuse radiation is even greater than beam radiation. In Cairns, the ratio beam/diffuse radiation is usually greater than unity, but the beam radiation drops at the mid of the year. In Darwin, the beam radiation is increasing constantly until mid-year, whereas the diffuse radiation is decreasing. The amount of diffuse radiation compared to beam radiation is obviously correlated to the relative humidity due to cloud cover (Thornton, Hasenauer, and White 2000).

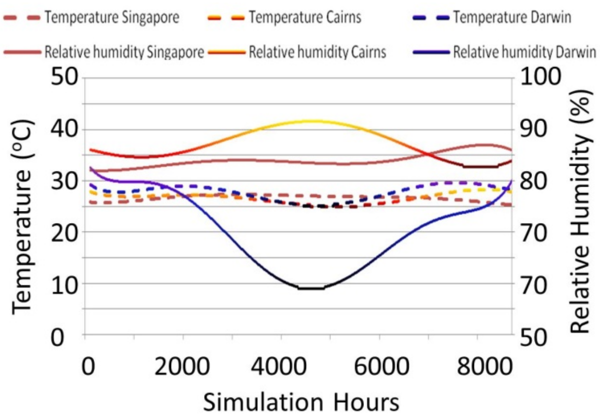

Fig. 2 - Weather conditions evolution through the year in three different locations

To achieve comfort in high humidity dryer systems or significant overcooling is required. Humidity will generally affect all air/water heat exchangers with the appearance of condensation – which requires consideration of the latent heat.

Several solar cooling studies have used TRNSYS as a simulation tool to define optimum systems (Ghaddar, Shihab, and Bdeir 1997), (Hammad and Zurigat 1998), (Florides et al. 2002), (Assilzadeh et al. 2005), (A. H. H. Ali, Noeres, and Pollerberg 2008), (Eicker and Pietruschka 2009). Most of these studies, however, did not account for the added complexity presented by humid climates. In general, these studies have found that solar cooling systems are marginally competitive only when combined with domestic water heating or when the cooling demand is large during periods where solar radiation is high.

From this, we infer that the acceptable performance of a solar cooling plant only results from optimization of various parameters: cooling technology, type and area of solar panels, heat and cold storage capacity, control strategy for heating and cooling loops, and when the system is well-matched with cooling (and possibly) heating demand.

SIMULATION

The aim of this work is to model a complete, small-scale system in a humid climate (comprised of solar collectors, storage tanks, and a novel 5kW ammonia/water absorption chiller developed and modeled by the French National Institute for Solar Energy [INES] laboratory). The system was designed to cover a typical house load defined by the Task 32 of IEA during the whole year.

TRNSYS was used to model the system, together with the weather values of a typical meteorological year (TMY) file for each cities encountered. As a basis, we will use the general system configuration presented by Ghiaus and Jabbour (2011). Figure 3 shows the general schematic of our design which we will optimize for the following: to reduce the global consumption of our system, and to meet the humidity requirements of tropical climates. We chose not to use any auxiliary heater and to implement a comfort-driven control strategy.

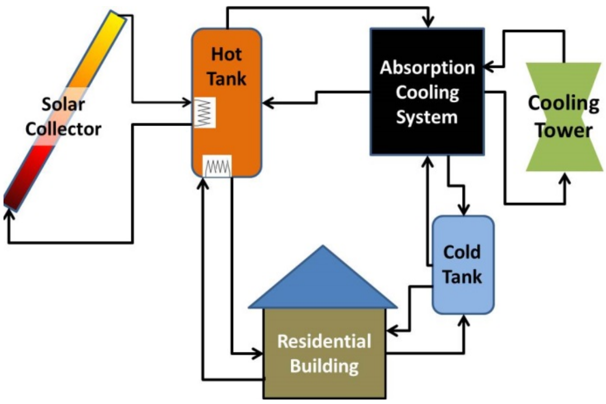

Fig. 3 – Simplified solar cooling system schematic

The Cooling Load

The International Energy Agency has created standards in order to compare energetic systems. Heimrath (2007) among the Task 32 of IEA as published one which can be used for HVAC systems that we use as a basis to define the case study house. The principal characteristics of the building are summarized in table 2.

Table 2: Principal characteristics of our cooling load

CHARACTERISTIC	VALUES
House's volume	500 m^3
House's construction materials	Models SFH 30, 60, 100 (Heimrath, 2007)
Inhabitant's gain	100 W/person (ISO 7730 standard, "seated at rest") (Anon. 2005)
Natural air change rate	0.4 air changes per hour (leaks)
Control-driven ventilation	1.5 air changes per hour from 7a.m. to 8p.m. / 3 air changes per hour from 8p.m. to 7a.m.

It should be noted that we concentrate on medium range energy efficiency buildings. Low energy buildings do not match the construction standard of the chosen geographical zone. Highly efficient buildings (with losses of no more than 100 kWh/m A) are only economically viable in extreme conditions. The occupancy profile is defined hour per hour with a maximum of 4 people from 8 p.m. to 6 a.m. and a minimum of 1 person at midday. Internal electric gains fluctuat during the day (maximum at evening) and are different during weekends. The ventilation system will re-use the internal air without introducing fresh air since the natural air change rate is sufficient for 4 people. Increasing the air flow rate during the night will increase the noise produced by the system but will improve performances as their will be no saturation of the system – saturation can appear with a blown volume of 1.5 units per hour.

The Solar Loop

On account of the mixed highly diffuse solar resource, we choose glazed flat plate solar collectors whose principal characteristics are given in table 3. It should be noted that these work well in locations with large ratios of diffuse light. A first water loop is heated by the collectors then the heat is exchanged in the hot-water tank storage. Pipes linking the hot water from the heat exchanger to the collector are considered in accordance with the Task 32. A differential controller is used to stop the pump when temperature into the collector is below the storage tank in order preserve thermal energy. Storage is considered with variable inlet temperatures and uniform losses. This means that the heat source flow and the cold side flow are entering the tank at the nodes closest in temperature to their respective flows. The coldest water is sent to the building whereas the warmest water is sent to the absorption chiller.

Table 3: Principal characteristics of the solar loop

COMPONENT	COEFFICIENTS	VALUES
Glazed Flat plate collector (Dynamic Efficiency ASHRAE defined) - Type 1290	$F'(\tau\alpha)_{en}$	0.80
	c_1	3.50
	c_2	0.015
	c_3	0
	c_4	0
	c_5	6.5
	c_6	0
	$K_{\theta d}$	0.90
Collector pipes – Type 31	length	15m
Heat Exchanger Water/water – Type 5 (dived into the hot water storage)	overall heat transfer coefficient	15000 kJ/hr.K
Storage tank – Type 4	number of nodes	6

Table 4: Principal characteristics of the cooling and the domestic hot water loop

COMPONENT	PARAMETER	VALUE
Absorber Condenser Pump	Flow rate	1900 kg/hr
Water/DHW heat exchanger	Overall heat transfer coefficient	10000 kJ/hr.K
Water/Chilled air heat exchanger	Overall heat transfer coefficient	1500 kJ/hr.K
Water/External air heat exchanger	Overall heat transfer coefficient	3000 kJ/hr.K
DHW Storage – Type 4	Volume	300 l
	Number of nodes	6
Input water pump	Temperature of water	20°C
	Flow rate	Equal to the flow rate to the load

The Cooling Loop

Three heat exchangers are used to cool the water from the absorption chiller. The first one is linked to the domestic hot water loop and allows warming of the DHW; the second is linked to the subcooled air provided by the absorption chiller to warm it up. The third one is an air/water heat exchanger with forced convection driven by fans. The flow rate of the absorber condenser pump corresponds to the conditions in which the absorption chiller has been tested (Boudéhenn et al. 2007).

The Domestic Hot Water Loop

The hot water profile assumes a daily hot water consumption of 120 litters (S.A. Kalogirou and Tripanagnostopoulos 2006). The warmest water goes to the load whereas the coldest goes to the first recovery heat exchanger into the cooling loop. A diverter, a mixer, a second heat exchanger and a PID controller permit recovering heat from the hot water storage tank. A PID controller keeps the temperature sent to the load constant allowing additional heat recovery from the solar hot water tank.

The Absorption Chiller Loop

This loop contains the absorption chiller, the chilled water storage tank, and the air cooling loop to the building. Chilled water passes through an air/water heat exchanger in order to cool the blown air. The system also has to control humidity ratio of indoor air. Subcooling is needed to ensure comfort in the absence of other control measures.

The absorption machine created by CEA-INES researchers cannot be described entirely here. The 'delta-delta T' temperature model of Ziegler et al. 1999 is used to study this collector:

$$\Delta\Delta t = t_G - (1+R)\cdot t_{AC} + R\cdot t_E \qquad (1)$$

In this expression, t_G, t_{AC} and t_E are external arithmetic mean temperatures at the generator, absorber-condenser, and evaporator, respectively. R is a factor equal to ~ 1.59. The different calorific powers are expressed as linear function of the characteristic temperature function:

$$P_x = A_x \cdot \Delta\Delta t + B_x \qquad (2)$$

A_x and B_x are coefficients defined with experimental results in steady state conditions for each calorific power P_x of the chiller (Boudéhenn et al. 2007). A differential controller stops the chiller's operation when the generator's temperature is below a particular set temperature, which we will study further.

In order model the air/water heat exchanger we take into account the seasonal humidity of the air. We assumed that since the water flow rate is much larger than the airflow rate, heat exchanges are not affected by water temperature changes. Moreover, psychometrics properties of moist air are following polynomial interpolations defined further and heat capacities are constant during the exchange. Every relation used to model the heat exchanger is listed in table 5. Pressure and enthalpy under saturation relations are second-degree polynomial interpolations from discrete values and the temperature under saturation relation solves a second-degree equation.

Table 5: Humid air equations

PROPERTIES OF MOIST AIR	RELATION
Absolute humidity under saturation	$x_{sat} = d\cdot\frac{P_{sat}}{P_m - P_{sat}}$ $with\ d = \frac{M_v}{M_{as}}$ (3)
Relative humidity	$e = \frac{p_v}{p_{sat}} \sim \frac{x}{x_{sat}}$ (4)
Enthalpy	$h = (c_{p,AS}\cdot T) + x\cdot(c_{p,V}\cdot T + L_{v,0})$ (5)
Pressure under saturation	$P_{sat} = 6{,}73\cdot 10^{-2}\cdot T^3 + 1{,}49\cdot 10^{-1}\cdot T^2 + 56{,}4\cdot T + 595{,}81$ (6)
Enthalpy under saturation	$h_{sat} = 0{,}0011\cdot T_{sat}^3 + 0{,}0055\cdot T_{sat}^2 + 1{,}8405\cdot T_{sat} + 9{,}3044$ (7)
Temperature under saturation	$T_{sat} = \frac{-b+\sqrt{\Delta}}{2a}$ (8) $with\ a = 0{,}0657/\ b = 1{,}011/c = 11{,}067\ and\ \Delta = b^2 - 4a(c - h_{sat})$

Figure 4 shows the potential routes of achieving thermal comfort in the building. Dot A represents the input state of the blown air, dot B the saturation limit of the air at T_A, Dot C' is for "perfect heat exchanger output", achievable with perfect heat exchanger efficiency, dot C is the real "cooled air output" and dot D is for the "blowing point" warmed up by the condenser/absorber.

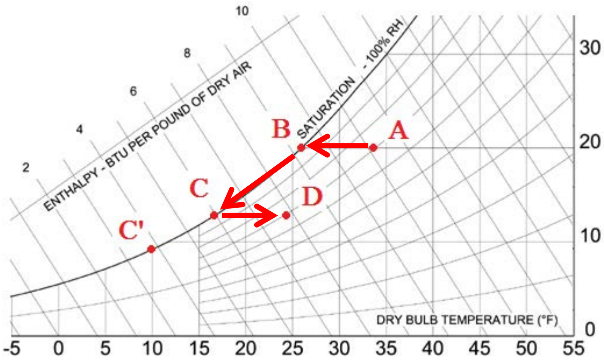

Fig. 4 - Psychometric chart relative to the heat exchanger air/chilled water in case of condensation

Tracing this path on the psychometric allows us to find $h_{C_{cond}}$, the enthalpy lost via condensation.

III. SYSTEM OPTIMISATION

A number of runs are carried out in order to optimise the various factors affecting the performance of the system. Each parameter was studied separately while holding everything else constant. The residential building is a 100kWh/m A family house with 50% of its north façade glazed, 20% of its east and west façade, and 12% of its south façade. The house is situated in Darwin, with 200 m of internal wall, and the external since that is external insulation. To define the performance of the system, we will look at the general cooling energy provided during one year, the overall COP of the system, and the comfort conditions concerning temperature and humidity.

Table 6: Parameters range and optimal values

PARAMETERS	RANGE OR OPTIMAL VALUE
Azimuth	300° (Darwin)
Slope of surface	20° (Darwin)
Collector's area	[5 to 25m]
Solar hot water tank's volume	[0.1 to 1 m^3]
Collector's water flow rate	250 kg/hr
By-Pass' temperature	[50 to 70°C]
Cold water tank's volume	[0.2 to 1 m^3]
Distributed temperature	[10 to 18°C]

A preliminary study run with TRNSYS gives us the optimal orientation (i.e. the azimuth and the slope of surface) relative to Darwin and an optimal value of collector's water flow rate – 300°, 20°, and 250 kg/hr, respectively.

Parameters characterization

The study's methodology is first to reduce the number of parameters studying each one separately and observing their level of influence, then to study more precisely the most influent and their effects on each other. The study's range of parameters is summarized in table 6.

Collector area. All parameters are fixed to their optimal value or to the mid of their range. A first obvious observation looking at figure 5 is that the system's performance is highly related to the collector's area. We can assume that those curves are function of other parameters and that we will need to study the collector's area coupled to other parameters. We can also see that the humidity comfort condition is harder to reach than the temperature comfort condition.

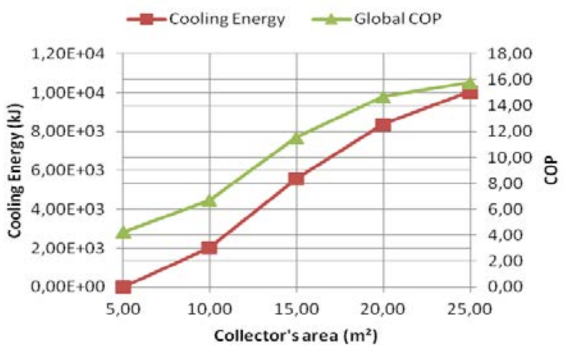

Fig. 5 - Energy performance of the system as a function of collector area

Solar hot water tank's volume. The energy performance of the system is shown in figure 6. When the tank's volume increases to 0.6 m^3, the COP and the cooling energy provided by the absorption chiller also increase. After the max value, these begin decreasing while the collected solar energy continues to increase. The maximum associated with the tank's volume can be explained by the fact that as the tank becomes to large fewer changes are possible from a relatively small daily solar irradiance. Comfort does not change much after 0.6 m^3 so over-sizing the tank will improve annual performance. (Note: It will, however, hurt payback time).

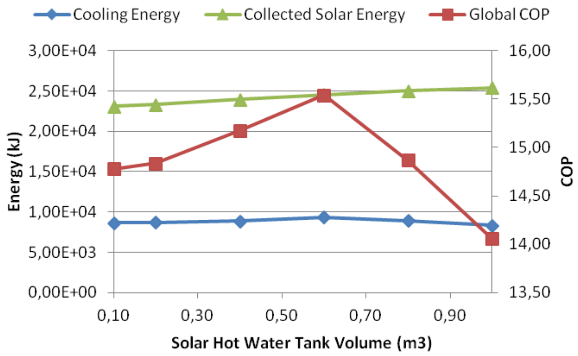

Fig. 6 - Energy performance of the system as a function of the hot water tank volume

Bypass temperature. The COP of the absorption chiller is dependent on the temperature at the generator. We assume that energy savings can be achieved by charging the hot tank only when the solar collector is hot. If the collector outlet temperature is below this point, it bypasses the hot tank. This point is called the "Bypass temperature". Figure 7 shows that the optimal bypass temperature for humidity and temperature comfort is 55°C. If this temperature is too high, comfort drops significantly. The initial increase of performance is explained by the fact that the absorption chiller performs better with high temperature at the generator. If the bypass temperature is too high, however, less hot water makes it to the generator throughout the year.

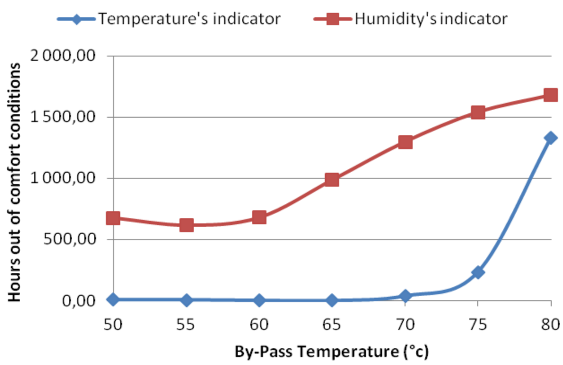

Fig. 7 - Comfort performances of the system as a function of by-pass temperature

Distributed temperature. A PID's controller linked to the cold water tank storage allows us to control the temperature of the water sent to the load. We found that performance and dehumidification are closely related to this temperature. Humidity decreases initially with the set cold temperature, but only up to a point. The initial decrease is due to better condensation in the heat exchanger. The increase that happens later reveals that the system cannot reach the set temperature much throughout the year.

To summarise, we can say that the hot and cold water tank volumes can be set at (a slightly oversized) $1m^3$. The collector water flow rate and bypass temperature can set to 250 kg/hr and 55°C, respectively.

Coupled Study

With the above parameters set, we now can perform a coupled study with collector's area varying between 5 and 42.5 m and distributed temperature varying between 8 and 18°C. Results of this analysis are plotted in figures 8 and 9. It can be seen that the 26°C internal air condition is met after the collector area tops $20m^2$, but the 70% internal relative humidity is only reached when collector area is greater than 35 m . This can be explained by first considering that reducing distributed temperature also reduces the COP of the absorption chiller.

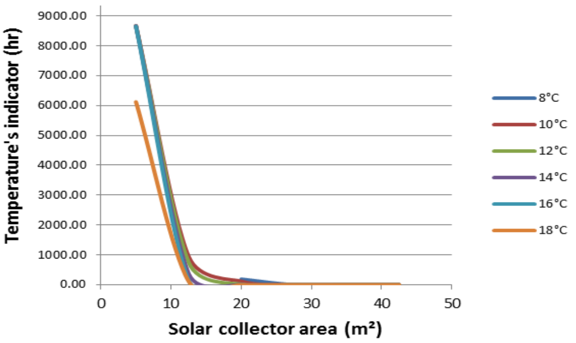

Fig. 8 – Hours spent above 26°C function of chilled water temperature and collector's area

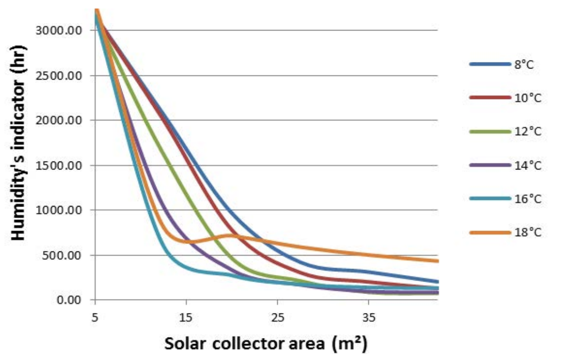

Fig. 9 – Hours spent above 70% relative humidity as a function of collector area

Overall, comfort temperature is easier to reach than comfort relative humidity. Indeed, to get a relative humidity below 70%, we have to cool the air well below 26°C. It can be inferred that if you set the chilled water temperature below 12°C, the absorption chiller's COP is too low and the system will provide very little cooling energy to dry the air. Sizing our system depends on the function we want to focus on.

Final Sizing

If, as is the case in most studies, drying the air is not a priority, 18m of solar collectors with a distributed temperature of 18°C are sufficient to cool the house. This study indicates that it takes almost twice as much solar collector area (at significant expense) to achieve the 'drying function'. The last part of this study focuses on choosing a coupled collector area/distributed temperature taking into account the 'drying function'. In this analysis, each parameter is optimized based on energy and comfort performance - results are shown in table 7.

Table 7: Optimal sizing of the system for Darwin

PARAMETER	OPTIMAL VALUE
Collector's area	20 m
Distributed temperature	15 °C
Solar hot water tank's volume	0.6 m^3
Cold water tank's volume	0.6 m^3
Collector's water flow rate	300 kg/hr
By-pass temperature	49 °C

The effect of climate

Applying the optimal sizing determined above, we will study the effect of different climates on the coupled collector's area/chilled water temperature – which are the most relevant coupled parameters. The major difference between different tropical climates is the relative humidity throughout the year. Thus, we will focus on the humidity performances of the system and the impact of climate changes. According to figure 10, our system works better in Savannah climates and barely works in Rainforest climates. For example, if only 20m of solar collectors are needed to spend less than 200 hours above 70% of humidity in Darwin, you need at least 40 m of solar collectors to reach that same level of performance in Singapore. This study indicates that monsoon and savannah climates are both adapted for our system (with a similar sizing) but we cannot use this design rainforest climates.

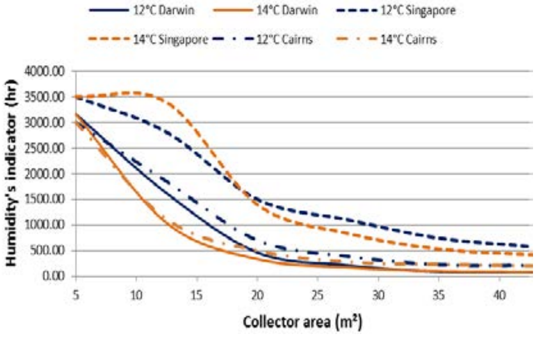

Fig. 10– Comfort performances function of distributed temperature and collectors' area in Darwin, Singapore and Cairns

CONCLUSION AND FUTURE WORK

In this work the optimal design and performance of a solar thermal cooling plant for a residential building, in tropical climates, was analysed. Using the model of a real absorption chiller (conceived and tested by INES), the main goal of this study was to assess the capability of this machine to operate in humid, tropical climates. The system was modelled in the software TRNSYS. We mainly focused on the model of heat exchanges between air and chilled water, taking into account potential condensation. Using the international standard proposed by IEA, the model was tested on a 200 m single family house consuming 60 kWh/m A, located in Darwin. An optimal design found regarding comfort conditions defined as 26°C and 70% internal temperature and relative humidity, respectively. The parametric study showed that to achieve the comfort condition throughout the year, a minimum of 20m of solar collectors and $0.6m^3$ of both hot and cold storage is needed. Other parameters of the control strategy have been determined such as the optimal water flow rate through the solar collectors, the optimal bypass temperature, and the optimal chilled water temperature. Finally our system has been tested with different tropical climates and the rainforest climate was found as the only climate where it cannot work. The parametric study assumes that each parameter was independent from one another and an accurate study of the global behaviour of our system would be interesting for future work. It could also be useful to compare our model with experimental data to prove its accuracy. In our model, only one type of solar collector has been tested and further studies should consider others, specially focusing on recovering the maximum of the diffuse radiation which is predominant in rainforest climates. Finally, an economic study should be conducted in order to define a payback time to determine the economic viability of our system. Lastly, we plan to compare such a system with standard HVAC and to a solar PV connected with a heat pump.

NOMENCLATURE

A_X = Absorption chiller experimental coefficient
B_X = Absorption chiller experimental coefficient
COP = Coefficient of performance
e = Relative humidity
h = Enthalpy
L = Latent energy
$\dot{m}$ = Mass flow rate
p = Partial pressure
P_X = Calorific power of the absorption chiller
Q = Thermal energy
t_a = Surrounding air temperature
V_{HOUSE} = Air volume occupied by the building
x = Absolute humidity
$\Delta\Delta T$ = Characteristic temperature function
θ = Incident angle of the beam irradiance
ε, η = Efficiency

SUBSCRIPTS

$cond$ = with condensation
$/cond$ = without condensation
in = input
m = melt or mean
out = output
sat = saturation
$temp$ = temperature
v = vapour
w = water

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