Modelling of a waste heat driven silica gel/water adsorption cooling system comparison with experimental results

M. Verde[1]

J.M. Corberan[1]

Robert de Boer

Simon Smeding

[1] Instituto de Ingeniería Energética, UPV

This paper was presented at the ISHPC conference, Padua, Italy, 7-8 April 2011

ECN-M--11-060 APRIL 2011

MODELLING OF A WASTE HEAT DRIVEN SILICA GEL / WATER ADSORPTION COOLING SYSTEM COMPARISON WITH EXPERIMENTAL RESULTS

M. Verde[(a)], J.M. Corberan[(a)], R. de Boer[(b)], S. Smeding[(b)]

[(a)] Instituto de Ingeniería Energética, UPV, Camino de Vera s/n, 46022 Valencia, Spain
[(b)] Energy research Centre of the Netherlands, ECN, PO Box 1, 1755 ZG, Petten, The Netherlands

ABSTRACT

In this paper a mathematical model is developed to investigate the performance of a silica gel-water adsorption cooling system. The model is completely dynamic and it is able to calculate the sequential operation of a thermal compressor, evaporator and condenser. The thermal compressor comprises of two identical adsorbent beds operating out of phase in order to achieve a continuous cold production. A single adsorbent bed consisted of three plate-fin heat exchangers in which dry silica gel (Sorbil A) grains were accommodated between the fins. The model was validated by experimental data. The experimental tests were performed in a lab-scale adsorption chiller prototype specifically designed and realized to be driven by low grade waste heat (80-90°C) with a cooling source at 33°C for automobile air-conditioning purposes. The experimental tests were conducted using two different operation system configurations. In the first one, an auxiliary heat recovery circuit is included. In the second system configuration, the auxiliary heat recovery circuit is omitted. In both cases, the mathematical model was able to simulate the dynamic behaviour of the system. The model prediction showed very good agreement with experimental data. The validation of the mathematical model promotes the idea of it being able to simulate a variety of similar prototypes in the future. The heat recovery system was found to have a strong positive effect on the chiller's COP, and a slight positive effect on the cooling power.

1. INTRODUCTION

Nowadays, innovative cooling systems are under development since the traditional air conditioners require high energy consumption and are responsible for emission of ozone depleting gases, such as CFCs, HCFCs. In addition, the recent European Union directive on air conditioning phases out systems using HFC-134a as refrigerant for new cars sold in the EU market from 2008 onwards. The end date of the phase-out period is proposed to be 2012.

Adsorption cooling systems are a promising alternative to conventional vapour-compression systems, since they are more energy efficient, and the environmental problems caused by CFC can be eliminated. This happens because they use water instead of CFCs as refrigerants, and they can be driven either by waste heat sources or

by renewable energy sources. Solid adsorption cooling systems have high potential for application in automotive air conditioning (Suzuki, 1993 and Tchernev, 1999). Most research on this kind of system is related to the development of models in order to predict the behaviour of the adsorption cooling system, and to improve the system's performance, by optimizing the operating conditions, such as temperature levels, mass flow rate and cycle time [3-7].

Part of the work presented here has been carried out in the frame of a R&D project called "Thermally Operated Mobile Air Conditioning Systems -TOPMACS" financially supported by the EC under the FP6 program [8]. The project aims at the design and development of solid adsorption cooling systems driven by low temperature energy coming from the car engine coolant loop for automotive air conditioning applications. The experimental tests were carried out on a lab scale adsorption chiller prototype realized at the ECN laboratories. The machine comprises of a condenser, an evaporator and two identical sorption beds, operating out of phase in order to enable a continuous cold production.The system uses silica gel as adsorbent and water as refrigerant. Such adsorbent material can be efficiently used with a maximum desorption temperature of 80-90ºC, which is suitable for adsorption chillers driven by low temperature heat sources. In this paper, a mathematical model for this system is described. The main purpose of developing the model is to optimize the sorption bed design and simulate the system´s performance, by estimating the cooling capacity and thermal efficiency. The model is completely dynamic and it is able to calculate the sequential operation of a double sorption chiller, calculate the condensation of the vapour at the condenser and the cooling effect produced at the evaporator. One difference between the presented model and models appearing in the literature is that the flow in between the components is based on the pressure difference between them. Also the pressure in the bed is based on state equation as well as mass conversation. This makes the model able to follow the full dynamics of the system. Moreover, different valve operation strategies or automatic operation (reed valves) could be analysed with the employed formulation model. The model is validated by experimental results of the system tested under non-steady flow conditions, showing the excellent capabilities of the model to predict the dynamic behaviour of the system. The validation of the mathematical model promotes the idea of it being able to simulate a variety of similar prototypes in the future.

2. MODEL DEVELOPMENT

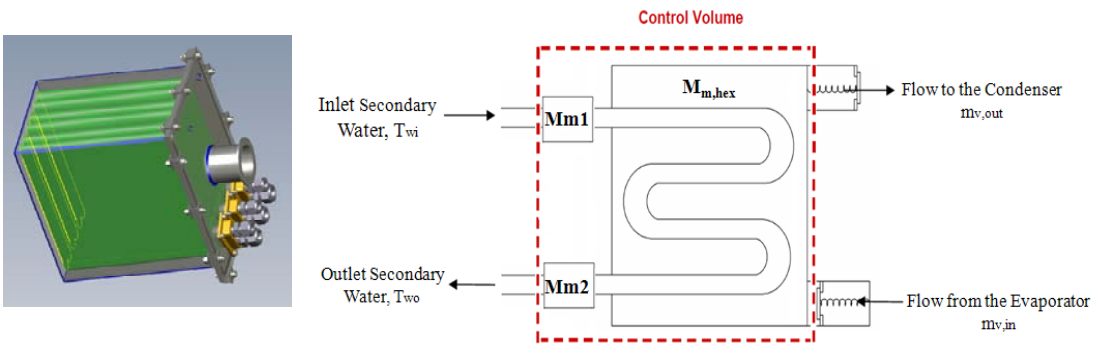


Fig. 1– (a) Adsorbent bed. (b) Single adsorbent heat exchanger, with control volume (dashed line).

A schematic of the adsorbent bed is shown in Fig. 1(a). An adsorbent bed consists of three plate fin heat exchangers packed with silica gel grains between the fins. The modeling of the adsorbent bed involves heat and mass conservation equations for the main body of the heat exchanger (metal fins and tubes), the adsorbent/adsorbate/inter-particle vapour, and the secondary fluid (water) which flows through the bed. The intake and outtake tubes used to provide the system with the secondary fluid are also taken into account. The control volume for which the energy balance applies is shown in Fig. 1(b) by the dashed line, and it represents a single adsorbent heat exchanger, plus the intake and outtake water tubes; the two metal masses, M_{m1} and M_{m2}, represent the intake and outtake water tube masses, respectively, and $M_{m,hex}$ represents the metal mass of the adsorbent heat exchanger (metal fins and tubes).

For the modelling, the following main assumptions for the adsorbent beds have been considered:

- Non equilibrium conditions with a simple kinetic model are considered at the adsorption desorption beds, all along the whole operating cycle.
- The general modelling approach is the use of zero-dimensional models (uniform temperature distribution in each operating unit at any instant).
- It is assumed that there is an empty space in the adsorbent heat exchanger (due to particle porosity, clearance volume, etc.) which is partially filled with water vapour. Consequently, the pressure at the reactors depends on the instantaneous mass of vapour contained inside them.
- The flow of water vapour among the beds, the condenser and the evaporator is governed by the pressure difference between these elements and the position of the valves.
- Thermal losses to the inert masses (intake/outtake tubes, heat exchanger fins and tubes) during the heating and cooling of the beds are taken into consideration.
- For the adsorbent heat exchanger, a detailed analytical study has been carried out in order to estimate an adequate global UA value (W/K) depending on the sorbent thermal properties as well as the geometrical characteristics of the bed.

The basic governing equations employed for the adsorbent bed modelling are described below. The governing equations for the condenser and evaporator are presented in the literature (Verde et al., 2010).

2.1 Adsorption rate

Non-equilibrium conditions of the adsorbent material have been considered, so the adsorption rate depends on the difference between the instantaneous uptake at the bed and the one that would be obtained at the equilibrium conditions. The uptake w_b is the instantaneous uptake in kg adsorbate/kg of dry adsorbent and is given by the following equation (Sakoda[a] et al. 1984):

$$\frac{dw_b}{dt} = \left(k_1 \cdot e^{-k_2/T_b} \left(w_{eq} - w_b \right) \right) \quad \textbf{(1)}$$

where, k_1 and k_2 are constant values taken from the literature (Sakoda[b] et al. 1984); w_{eq} is the equilibrium uptake in kg adsorbate/kg of dry adsorbent and can be expressed by the equation shown below (Freni et al., 2002),

$$\ln P_b = a\left(w_{eq}\right) + \frac{b\left(w_{eq}\right)}{T_b} \quad \textbf{(2)} \qquad a(w) = a_0 + a_1 \cdot w + a_2 \cdot w^2 + a_3 \cdot w^3 \quad \textbf{(3)} \qquad b(w) = b_0 + b_1 \cdot w + b_2 \cdot w^2 + b_3 \cdot w^3 \quad \textbf{(4)}$$

where, P_b is the bed pressure in mbar and T_b bed temperature in K. The numerical values of the constants a and b (i = 0, 1, 2, 3) are obtained experimentally and depend on the adsorbent/ water pair used. The values of the coefficients for the silica gel-water pair were taken from the literature (Restuccia et al., 1999).

2.2 Energy and mass balance equations

The energy balance on the control volume represented in Fig.1 (b) can be written as:

$$C_b \frac{dT_b}{dt} = Q_{heat/cool} + M_s \cdot \frac{dw_b}{dt} \cdot \Delta H - \left(\frac{\dot{m}_{v,in}}{N_g} \right) \cdot cp_v \cdot \left(T_b - T_{evap} \right) \quad \textbf{(5)}$$

where C_b is the thermal mass of the adsorbent heat exchanger (adsorbent + adsorbate + metal); N_g is the number of adsorbent heat exchangers in the bed; M_s is the mass of dry adsorbent per heat exchanger; $m_{v,in}$ is the refrigerant mass flow entering the adsorbent bed; w_b is the instantaneous uptake; ΔH is the heat of ads/desorption; T_b is the bed temperature; T_{evap} the evaporation temperature; $Q_{heat/cool}$ is the heat transferred between the secondary fluid (which can be heating water or cooling water), and the adsorbent. This heat input or output depends on the operation mode of the bed (cooling-adsorbing/heating-desorbing).The Eq. (5) is valid for the bed in adsorption mode as well in desorption mode. The only difference in the energy equation is that in desorption mode (heating the bed), the term $\dot{m}_{v,in} \cdot cp_{w,v} \cdot \left(T_b - T_{evap} \right)$ is neglected. This term corresponds to the sensible heat required to heat up the vapor from the evaporation temperature up to the bed temperature during the adsorption process.

An alternative expression to the standard NTU formulation has been developed in order to determine the heat exchanged between the secondary fluid and the adsorbent under non-steady conditions. The correlation takes into account the effect of both inlet and outlet temperature differences on the heat flux. Since there is no clear rule to estimate the correct weighting factors, a combination of 80% (inlet temperature difference) and 20% (outlet temperature difference) has been considered in the model.

The heat transfer $\dot{Q}_{heat/cool}$ is represented by the following correlation:

$$\dot{Q}_{heat/cool} = \varepsilon_b \cdot \dot{m}_{w,g} \cdot cp_w \cdot \left[0.8\left(T_{wgi} - T_b \right) + 0.20\left(\frac{T_{wgo} - T_b}{1 - \varepsilon_b} \right) \right] \quad \textbf{(6)}$$

where ε_b is the heat exchanger effectiveness; $m_{w,g}$ is the secondary fluid mass flow (heating or cooling water) circulating through the adsorbent heat exchanger; T_{wgi} and T_{wgo} are the temperatures of the secondary water at the inlet and outlet of the adsorbent heat exchanger, respectively.

The thermal losses to the inert masses due to heating and cooling the bed have been taken in consideration in the model. The following differential equations have been developed in order to calculate the temperature of the intake metal tube (T_{m1}), the temperature of the outtake metal tube (T_{m2}), the temperature of the secondary water at the inlet of the heat exchanger (T_{wgi}), the temperature of the secondary water at the outlet of the heat exchanger (T_{wgo}) and the temperature of the secondary water leaving the bed (T_{wo}).

The heat transfer from the inlet secondary water to the intake metal tube $\dot{Q}_{losses\_1}$ is given by the following equation:

$$\dot{Q}_{losses\_1} = \dot{m}_{w,g} \cdot Cp_w \cdot \varepsilon_{m1} \left[0.8 \cdot (T_{wi} - T_{m1}) + 0.20 \cdot \frac{T_{wgi} - T_{m1}}{1 - \varepsilon_{m1}} \right] \quad \textbf{(7)}$$

where, ε_{m1} is the surface effectiveness of the intake tube.

The heat transfer from the outlet secondary water to the outtake metal tube $\dot{Q}_{losses\_2}$ is given by the following equation:

$$\dot{Q}_{losses\_2} = \dot{m}_{w,g} \cdot Cp_w \cdot \varepsilon_{m2} \left[0.8 \cdot (T_{wgo} - T_{m2}) + 0.20 \cdot \left(\frac{T_{wo} - T_{m2}}{1 - \varepsilon_2} \right) \right] \quad \textbf{(8)}$$

where, ε_{m2} is the surface effectiveness of the outtake tube.

The temperature of the intake metal tube T_{m1} and outtake metal tube T_{m2} is given by the following equations:

$$C_{m1} \frac{dT_{m1}}{dt} = \dot{Q}_{losses\_1} \quad \textbf{(9)}$$

$$C_{m1} \frac{dT_{m2}}{dt} = \dot{Q}_{losses\_2} \quad \textbf{(10)}$$

where, C_{m1} and C_{m2} is the thermal capacity of the intake and outtake tube (metal), respectively.

The temperatures of the secondary water at the inlet of the heat exchanger T_{wgi} and at the outlet of the heat exchanger T_{wgo} are given by the following equations:

$$C_{w,m1} \frac{dT_{wgi}}{dt} = \dot{m}_{w,g} \cdot cp_w \cdot (T_{wi} - T_{wgi}) - \dot{Q}_{losses\_1} \quad \textbf{(11)}$$

where, $C_{w,m1}$ is the thermal capacity of the secondary water inside the intake tube; T_{wi} is the secondary water at the inlet of the bed (which means in the intake tube).

$$C_{w,g} \frac{dT_{wgo}}{dt} = \dot{m}_{w,g} \cdot cp_w \cdot (T_{wgi} - T_{wgo}) - \dot{Q}_{heat/cool} \quad \textbf{(12)}$$

where, $C_{w,g}$ is the thermal capacity of the secondary water inside the adsorbent heat exchanger.

Finally, the temperature of the secondary water leaving the bed T_{wo} (in the outtake tube) is given by the following equation:

$$C_{w,m2} \frac{dT_{wo}}{dt} = \dot{m}_{w,g} \cdot cp_w \cdot (T_{wgo} - T_{wo}) - \dot{Q}_{losses\_2} \quad \textbf{(13)}$$

where, $C_{w,m2}$ is the thermal capacity of the secondary water inside the outtake tube.

Since equilibrium is not assumed it is necessary to incorporate an equation for the pressure in the bed. This equation comes from the equation of state for the water vapour inside the bed:

$$P_b = P_a + P_v = \frac{RT_b}{V_b} \left(\frac{m_a}{M_a} + \frac{m_{v,b}}{M_v} \right) = \frac{RT_b}{V_b} \frac{m_{v,b}}{M_v} \quad \textbf{(14)}$$

where P_a is the pressure due to the non-condensable gases inside and P_v is the pressure due to the water vapour in the bed. It is assumed that non-condensable gases have been totally removed. The ODE for the pressure at the bed P_b then results:

$$\frac{dP_b}{P_b} = \frac{dm_{v,b}}{m_{v,b}} + \frac{dT_b}{T_b} \rightarrow \frac{dP_b}{dt} = P_b \left(\frac{1}{m_{v,b}} \frac{dm_{v,b}}{dt} + \frac{1}{T_b} \frac{dT_b}{dt} \right) \quad \textbf{(15)}$$

where, $m_{v,b}$ is the total mass of vapour in the bed.

Finally, the continuity equation at the reactor provides the necessary link between the uptake variation and the vapour flow rates leaving or entering the bed. The mass balance for the bed can be written as:

$$\frac{dm_{v,b}}{dt} = -M_s \cdot N_g \cdot \frac{dw_b}{dt} + \dot{m}_{v,in} - \dot{m}_{v,out} = -M_s \cdot N_g \cdot \frac{dw_b}{dt} + \dot{m}_{v,ad} - \dot{m}_{v,des} \quad \textbf{(16)}$$

where $m_{v,b}$ is the total mass of vapour in the bed; $m_{v,in}$ and $m_{v,out}$ are the refrigerant (water vapour) flow rates entering and leaving the bed, respectively. In order to calculate the flow rate of water vapour between the beds, the condenser, and the evaporator through the interconnecting pipes and valves, the following assumptions have been considered:

- The valves are considered fully opened or fully closed, depending on the pressure difference. They are assumed to react instantaneously.
- The valve to the condenser is only open when the pressure upstream is higher than the one at the condenser. Otherwise it remains closed.
- The valve at the evaporator is only open when the pressure downstream is lower than in the evaporator. Otherwise it remains closed.

Consequently, the instantaneous flow rates, $m_{v,in}$ and $m_{v,out,}$, can be calculated as follows:

$$\dot{m}_{v,out} = \begin{cases} = 0 & If\ P_b < P_{cond} \\ = A_{cond}\sqrt{2 \cdot \rho_v(T_b, P_b) \cdot (P_b - P_{cond})} & If\ P_b \geq P_{cond} \end{cases} \quad \textbf{(17)}$$

$$\dot{m}_{v,in} = \begin{cases} = 0 & If\ P_b > P_{evap} \\ = A_{evap}\sqrt{2 \cdot \rho_v(T_{evap}, P_{evap}) \cdot (P_{evap} - P_b)} & If\ P_b \leq P_{evap} \end{cases} \quad \textbf{(18)}$$

This set of equations constitutes a system of 9 ODEs for T_b, T_{m1}, T_{m2},T_{wgi},T_{wgo},T_{wo},P_b,w_b and $m_{v,b}$ for each bed.

3. PROTOTYPE ADSORPTION CHILLER TEST FACILITY

3.1 *Adsorption chiller prototype description*

3.2

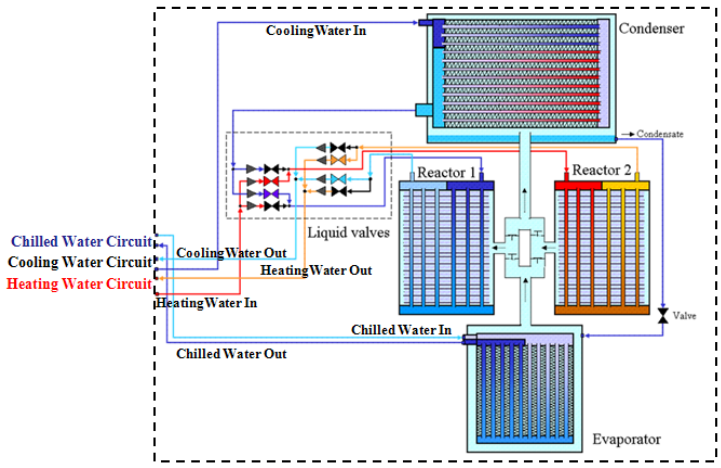


Fig. 2 – Schematic of the considered adsorption cooling system, with boundary conditions for the system's performance calculation (dashed line).

The flow diagram of the adsorption chiller analysed in this paper is shown in Fig. 2.The adsorption chiller is connected to three secondary water circuits for heating, cooling, and chilled water. The hot water is sent to one of the two beds, depending on the position of the valves in the liquid circuit. The cooling water is first sent to the condenser and then through the valves to the other reactor. The chilled water circuit is directly fed to the evaporator. The adsorbent beds operate in counter-phase in order to allow a continuous useful effect: one bed is in cooling mode while the other is in regeneration mode.

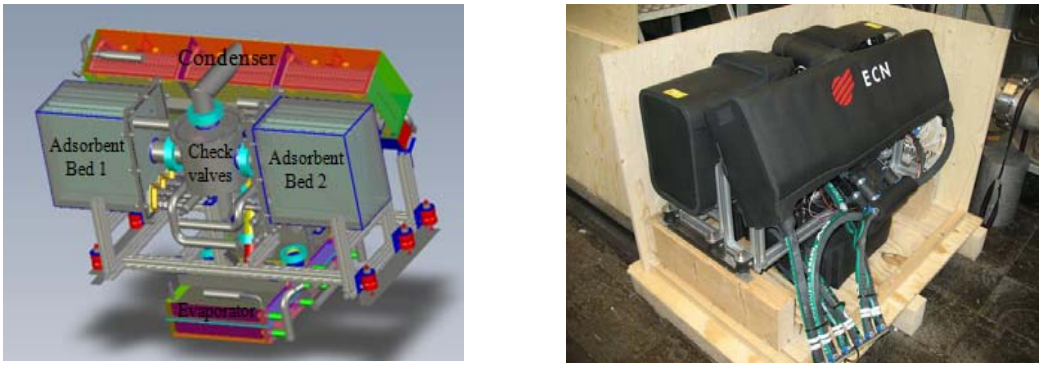
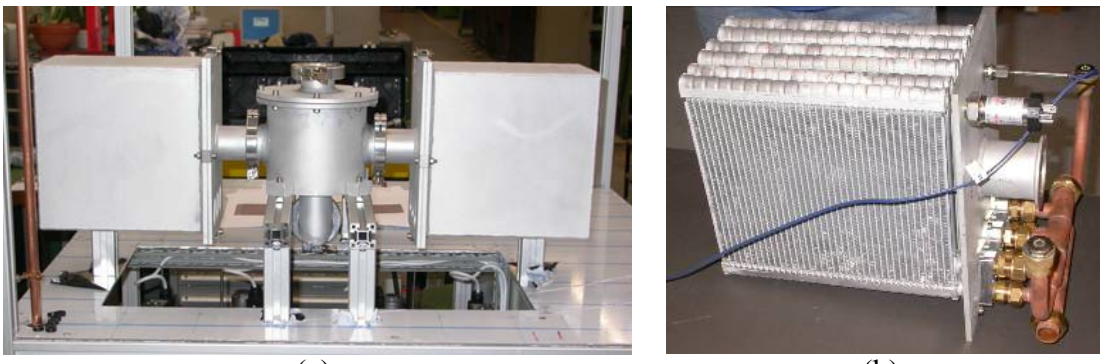


Fig.2 - Design drawing and picture of the prototype adsorption cooling system.

The main components of the adsorption chiller prototype are a water cooled condenser, an evaporator, two adsorbent beds, check valves to direct the refrigerant vapour flow, a condensate valve connected to a liquid level control in the evaporator and four three way valves to direct the heating and cooling water circuits alternately to both adsorbent beds. A design drawing of the prototype system and a picture of it are shown in Fig.3. Temperature sensors are installed inside the beds, the evaporator and the condenser and pressure sensors are mounted on the condenser and evaporator and on both beds. Temperature sensors are also installed to measure the inlet and outlets of the external (secondary) water circuits of the system.



(a) (b)

Fig.3– (a) Two sorption beds connected to the central housing that contains the refrigerant check valves. (b) One sorption bed assembly.

The thermal compressor section of the adsorption system consists of two beds, see Fig. 3(a). Each reactor has three tube-fin heat exchangers connected in parallel, see Fig.3 (b). The weight of one aluminium heat exchanger is 1 kg and the fin side of each heat exchanger is filled with 1 kg of silica gel grains. The beds are connected to a housing that contains the gravity operated refrigerant check valves. The condenser has three heat exchangers (automotive evaporators) connected in parallel and contained in a stainless

steel envelope. It has internal reinforcements to withstand the forces of the internal vacuum. The evaporator has four tube-fin heat exchangers (automotive heater cores) that lie horizontally in 2 sections on top of each other. Each heat exchanger has a water layer at the lower part of the fin side.
The overall weight of the prototype system is 86 kg, not including the weight of the water in the circuits for heating, cooling and chilling, the refrigerant water in the evaporator, and excluding the thermal insulation.

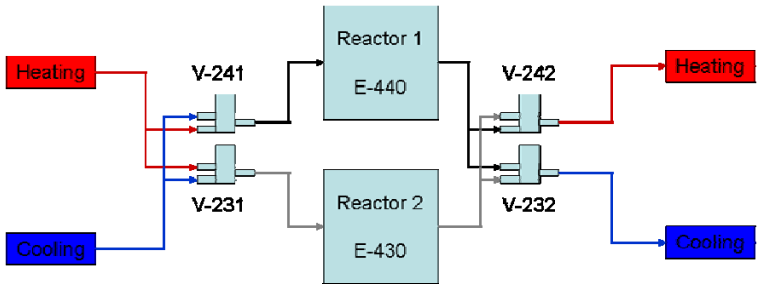


Fig.4 - Lay-out of the heating and cooling circuits for the two silica gel beds.

A PLC system controls the operation of the liquid circuit valves. The repeated heating and cooling of the reactors is controlled by these valves, according to the scheme in Fig.4. The system can be operated either on fixed times for heating and cooling of the beds, or on flexible timing, where the switching of the valves is controlled by the actual temperature differences between the inlet and outlet temperature of the heating circuit and the cooling circuit. As an example, when both of these differences between inlet and outlet are less than 1K, the valves at the inlet (V241, V231) are switched. The valves at the outlet (V242, V232) are switched with a time delay with respect to the inlet valves. This delay prevents hot water at high temperature to be sent directly to the cooling circuit and cooling water to be sent to the heating circuit. The delay time is set to such a value that the outlet temperatures are almost identical, and in the range of the average temperature of the heating and cooling water circuit.

3.3 Laboratory tests

To evaluate the performance of the prototype the three water circuits are connected to a heating, a cooling and a chilled water circuit. All secondary water circuits contain a temperature controlled water storage and a pump. Each circuit has a flow measurement device (Burkert), temperature sensors (PT100) at the inlet and return and measurement of the pressure drop. From the secondary water flow rate, and the temperature drop in the heating and chilled water circuit, the cooling power and COP were estimated at a fixed cycle time of 6 minutes. The assumed boundary condition for the cooling power and COP calculation is presented in Fig.2. For the model and experimental tests, the cooling power and COP were calculated from time-independent average values at the above mentioned conditions.

4. EXPERIMENTAL AND MODEL RESULTS

The experimental tests have been performed at different conditions in order to assess the performance of the system. In the graphs below, a sample of the experimental results obtained with water inlet temperature of 90ºC, condensing temperature of 33ºC and evaporating temperature of 15ºC are presented in comparison with the model results. The graphs show four peaks, each of them corresponding to half a cycle. Accordingly,

two complete cycles are shown with a fixed cycle time of 6 minutes. These experimental tests have been carried out assuming two different operation system configurations. The first one is assuming that an auxiliary heat recovery circuit is omitted. This means that the hot water and cooling water leaving the beds returns to the heating and cooling circuit, respectively. The second system configuration is considering that an auxiliary heat recovery circuit is included. This means that the hot water leaving one of the beds is directly sent to the cooling circuit, and cooling water leaving the other bed is sent to the heating circuit until the outlet water temperatures are almost identical. At this moment, the valves at the outlet are switched, and the outlet water is sent to the secondary water circuit as in the first case (i.e. outlet hot water is sent to the heating circuit and outlet cooling water is sent to the cooling circuit).

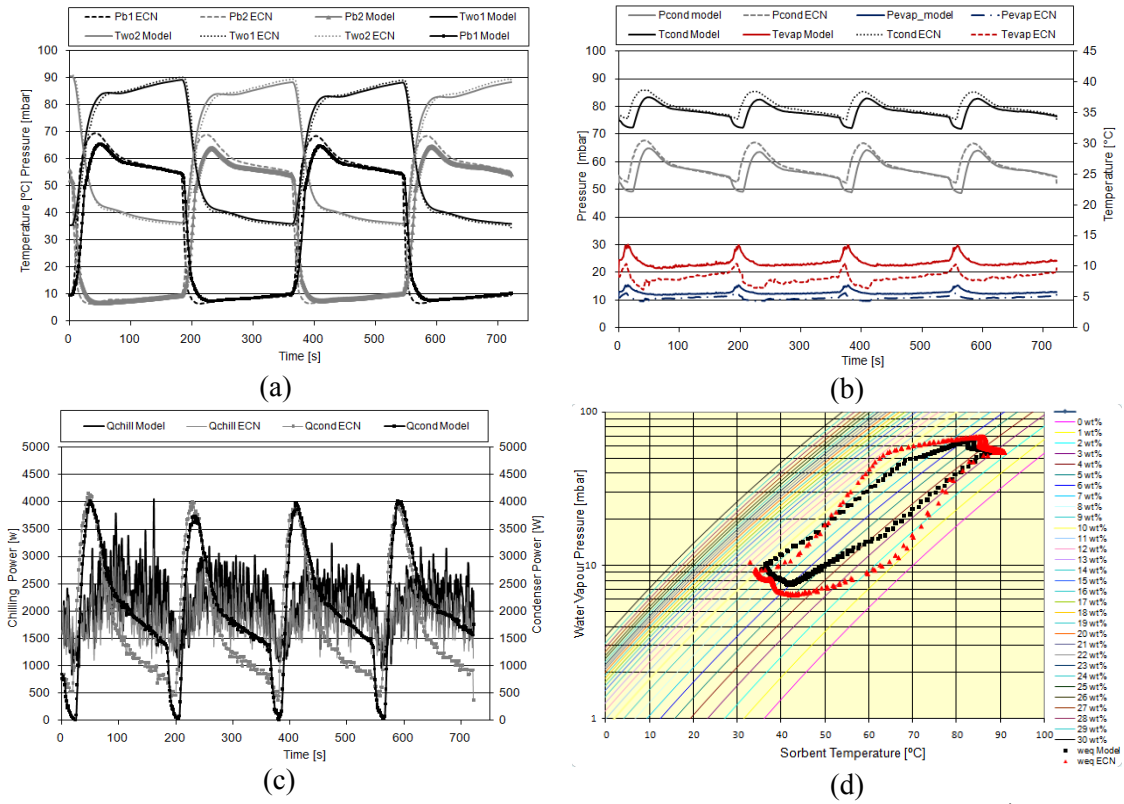


Fig. 5 – *Comparison between calculated and experimental results for the* ***1st System Configuration****: (a) Bed 1 and Bed 2 pressures and outlet water temperatures. (b) Temperatures and pressures at condenser and evaporator. (c) Chilling and condenser capacity. (d) Equilibrium uptake evolution.*

Fig.5 shows the comparison between the simulated and experimental results for the first system configuration, at the above mentioned conditions. Fig. 5(a) shows the comparison between bed pressure and outlet temperature predicted by the model and experimental results. The experimental results show some differences between the temperature profiles of the beds. They are clearly distinguishable for pressures at the highest values range, which correspond to the desorption phase. This difference could be caused by different pressure losses in the flow from the condenser to the bed across the corresponding valves, or simply due to the uncertainty of measurements. Furthermore, the water temperature at the outlet of the bed is asymmetric, showing slight differences between both beds. This can be due to a different permeability and compactness of the adsorbent material in the bed, leading to the observed differences in behavior. The model is not able to reproduce those differences, since it assumes that

both beds are identical. Fig. 5(b) shows the comparison between calculated and measured results for the pressure and temperature at the evaporator and condenser. As it can be observed, the adjustment between calculated and measured results is very good in phase and amplitude, regardless of the hypothesis assumed for the modeling. Fig. 5(c) shows the comparison between the calculated and measured useful cooling power and condenser power. The adjustment is remarkably good, taking into account the approximate nature of the model. Finally, Fig. 5(d) shows the equilibrium value of the uptake, calculated from the reactor pressure and temperature, for both measurements and calculations. The equilibrium uptake for the experimental results has been evaluated from the instantaneous recordings of bed pressure and temperature. Probably due to the difficulty of measuring the temperature inside the bed, the calculated and measured equilibrium uptake is not very similar.

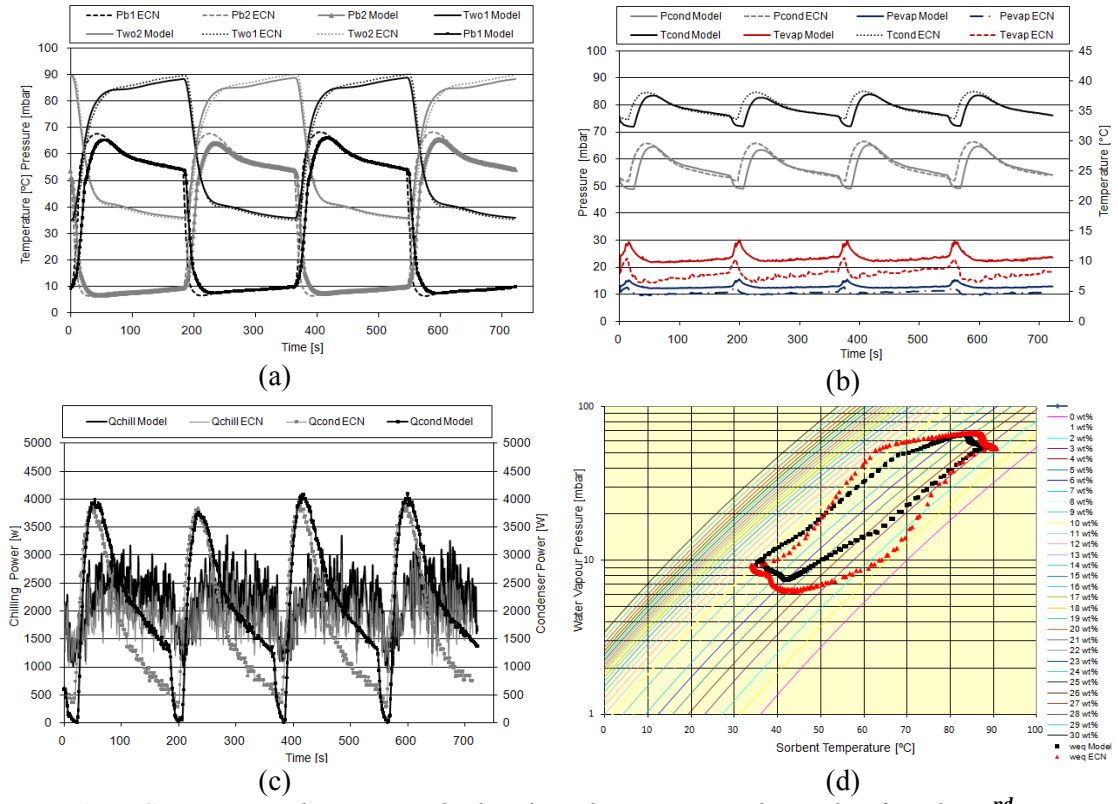


Fig. 6 – Comparison between calculated and experimental results for the ***2nd System Configuration****: (a) Bed 1 and Bed 2 pressures and outlet water temperatures. (b) Temperatures and pressures at condenser and evaporator. (c) Chilling and condenser capacity. (d) Equilibrium uptake evolution..*

Fig.6 shows the comparison between the simulated and experimental results for the second system configuration, at the above mentioned conditions. This agreement between measured and experimental results was checked to be good for both system configurations, so the model can be effectively used to estimate the possible performance of the system at different operation types. All in all, the obtained adjustment is very good, and it clearly indicates that the model has good prediction capability for transient conditions and is able to capture most of the dynamics of the system.

	Experimental		Simulation	
System Configuration:	**Qchill [W]**	**COP**	**Qchill [W]**	**COP**
No external heat recovery circuit	1893	0.29	2153	0.35
With an external heat recovery circuit	1912	0.44	2155	0.47

Table 1 – System performance.

Table 1 shows the system performance for the two complete cycles of approx. 6 min for both configurations. Based on the model results, the system is able to produce a mean cooling power of 2153 W and a COP of 0.35, without the heat recovery circuit. If the auxiliary heat recovery circuit is considered, the system is able to produce a mean cooling power of 2155 W and a COP of 0.47. Based on experimental results, the system is able to produce a mean cooling power of 1893 W and a COP of 0.29 for the first system configuration, and a mean cooling power of 1912 W and a COP of 0.44 for the second system configuration. The auxiliary heat recovery circuit was found to have a strong positive effect on the system´s COP, and a slight positive effect on the cooling power.

5. CONCLUSIONS

A model of an adsorption chiller prototype has been developed. The model is completely dynamic and it is able to calculate the sequential operation of a double sorption chiller, the condensation of the vapour at the condenser and the cooling effect produced at the evaporator. The model of the adsorption cooling system has been compared with experimental results of a double-bed silica gel-water system. The experimental tests were conducted using two different configurations. In the first one, an auxiliary heat recovery circuit is included. In the second case, the auxiliary heat recovery circuit is omitted. In both cases, the calculated results are in very good agreement with the experiments, proving the good capabilities of the model to predict the system performance at different configurations. The validation of the mathematical model promotes the idea of it being able to simulate a variety of similar prototypes in the future.

ACKNOWLEDGEMENTS

This work has been partially supported by the European Commission under the 6th Framework-program (Contract Ref. TST4-CT-2005-012471). The authors are very grateful for their support. Moreover, the authors are grateful to FCT-Portugal for funding this work.

NOMENCLATURE

A, effective flow area
C, thermal capacity
Cp, specific heat
ΔH, sorption heat
k_n, constants of the kinetics equation (n=1,2)
M, mass (also molecular mass)
$\dot{m}$, mass flow rate

m, mass
P, pressure
R, Universal gas constant
t, time
T, temperature
V, volume
w, uptake
ρ, density
ε, heat exchanger effectiveness
N, number
Q, Heat flux (Heating or Cooling)

SUBSCRIPTS

a, air
b, bed
cond, condenser
evap, evaporator
eq, equilibrium conditions
i, inlet
o, outlet
in, inlet
out, outlet
ad, adsorption
des, desorption
s, sorbent
w, secondary water
v, water vapour
g, generator
m1, intake metal tube
m2, outtake metal tube

REFERENCES

[1] Suzuki M., 1993, Application of adsorption cooling systems to automobiles, Heat *Recovery Systems & CHP*, 13: 335 - 340.
[2] Tchernev D., 1999, A waste heat driver automotive air conditioning system, *Proc. of International Sorption Heat Pump Conference*, Munich, Germany, 24 - 26 March: 65 - 70.
[3] Zhang L.Z., Wang L.,1997, Performance estimation of an adsorption cooling system for automobile waste heat recovery, *Applied Thermal Engineering* ,17:1127 - 1139.
[4] Zhang L.Z., 2000, Design and testing of an automobile waste heat adsorption cooling system, *Applied Thermal Engineering,* 20:103 - 114.
[5] Jianzhou S., Wang R.Z., Lu Y.Z., Xu Y.X., Wu J.Y., 2002, Experimental investigations on adsorption air-conditioner used in internal-combustion locomotive driver cabin, *Applied Thermal Engineering*, 22:1153 - 1162.
[6] Lu Y.Z., Wang R.Z., Jianzhou S., Xu Y.X., Wu J.Y., 2004, Practical experiments on an adsorption air conditioner powered by exhausted heat from a diesel locomotive, *Applied Thermal Engineering*, 24:1051 - 1059.
[7] Sakoda A., Suzuki M., 1984, Fundamental study on solar powered adsorption cooling system, *Journal of Chemical Engineering of Japan,*17: 52 - 57.
[8] TOPMACS, Thermally Operated Mobile Air Conditioning Systems, EU FP6 R&D Project, Contract Ref. TST4-CT-2005-012471.
[9] Verde M., Cortes L., Corberan J.M., Sapienza A., Vasta S., Restuccia G. ,2010, Modeling of an adsorption cooling system driven by engine waste heat for truck cabin A/C.Performance estimation for a standard driving cycle, *Applied Thermal Engineering*, 30:1511-1522.

[10] Freni A., Tokarev M.M., Restuccia G., Okunev A.G., Aristov Yu I., 2002, Thermal conductivity of selective water sorbents under the working conditions of a sorption chiller, *Applied Thermal Engineering*, 22.
[11] Restuccia G., Aristov Yu. I., Maggio G., Cacciola G., Tokarev M.M., 1999, Performance of sorption systems using new selective water sorbents, *Proc. of International Sorption Heat Pump Conference*, Munich, Germany, 24 - 26 March: 219 - 223.