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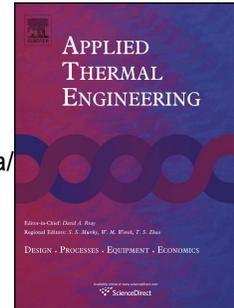
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MODELLING OF AN ADSORPTION SYSTEM DRIVEN BY ENGINE WASTE HEAT FOR TRUCK CABIN A/C. PERFORMANCE ESTIMATION FOR A STANDARD DRIVING CYCLE.

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Abstract: This paper presents the main characteristics of an innovative cooling system for the air-conditioning of a truck cabin, as well as a first estimation of its performance during a standard driving cycle, obtained with a specifically developed vehicle-engine-cooling system overall model. The innovative cooling system consists of a water-zeolite adsorption-desorption system, which employs the waste heat from the engine to produce the cooling of the vehicle cabin. The developed global model is completely dynamic and is able to: reproduce the operation of the engine through a standard driving cycle, evaluate the waste heat available at the engine hydraulic loop; calculate the sequential operation of an adsorption-desorption system, calculate the condensed water per cycle, the cooling effect produced at the evaporator, and finally, the temperature and humidity evolution of the air inside the cabin. 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 the low grade waste heat (80-90°C) from the engine coolant loop of a truck. The experimental activity carried out showed that the chiller is able to generate up to 5 kW of peak cooling power at 10°C (35°C of condensation temperature) with a COP of 0.6. The obtained results show that the system could be able to provide a significant amount of the required cooling.

Keywords: Adsorption cooling; zeolite; truck cabin A/C

1. INTRODUCTION

Europe is making a huge effort to reduce Greenhouse gas emissions. The recent European Union directive on mobile air conditioning (MAC) phases out systems using HFC-134a as refrigerant for new cars placed on the EU market from 2008 onwards. The end date of the phase-out period is proposed to be 2012. Thus new solutions must be found, one direction is the use of CO₂ as refrigerant, while a medium term solution could be the use of adsorption systems. Compared to traditional air conditioners, an adsorption machine can utilize water as refrigerant and is driven by the waste heat from the engine without losing mechanical energy. The adsorption machine can be operated by the low temperature energy coming from the engine coolant loop, which limits the maximum temperatures available for desorption of the adsorbent material. For cars the maximum temperature is usually limited to 90-95 °C while for trucks the maximum temperature is usually limited to lower values 80-85 °C in order to increase the life of the engine.

The potential use of adsorption systems for automotive applications has been already investigated.

See for instance references [1, 2]. Zhang et al. [3,4] already presented a dynamic model of a possible reactor and also presented a comparison between the experimental results of a reactor prototype and the calculated results. Their work was focused on the development of a system driven by the exhaust gases for automotive applications. Wang et al. [5,6,7] presented a detailed dynamic model of an adsorption system and experimental results of a prototype for cabin air-conditioning of a locomotive. The system was also operated by the waste heat of the exhaust gases.

In this paper, an overall vehicle-engine-chiller model and a first estimation of the performance of the system for truck cabin A/C is presented. The model was previously validated against experimental results carried out on a lab-scale adsorption chiller prototype realized in CNR-ITAE laboratory. 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 the development of an innovative cooling system for automotive applications, consisting of a water-zeolite adsorption-desorption system, which by employing the waste heat from the engine is able to cool the cabin. The project targets the design and development of an adequate innovative prototype of adsorption air conditioner for a truck as well as their experimental evaluation.

In contrast with previous works, the present system is based on the utilization of the waste heat contained in the engine coolant instead of the exhaust gases. A previous study allowed concluding that the potential of heat recovery from the engine coolant was much higher than that from the exhaust gases and the size of the reactors could be much smaller. The exhaust gases could be employed in the system to further increase the temperature of the circulating water. However, the analysis of the advantages of this option showed that they were little in comparison with the extra cost and complication of adding a gas to water heat exchanger. Therefore the design of the reactors, the evaporator and condenser is completely original and had to be done to satisfy the difficult operating conditions characteristic of this kind of application, keeping always in mind that a high compactness was an essential requirement.

The prototype consisted of an advanced double-bed adsorber, a condenser and an evaporator realized with finned tube coils with wide exchanging surface and high efficiency. The two beds were realized using a new sorbent called AQSOA-Functional Adsorbent Material – AQSOA-FAM, specially produced by Mitsubishi Chemical [9, 10]. Such adsorbent material was specifically designed for adsorption machines driven by low temperature heat sources (e.g. solar energy and waste heat from the internal combustion engines coolant loop). Indeed, a previous work demonstrated that the material FAM-Z02 can be efficiently used with a maximum desorption temperature of temperatures of 80-90°C [11], which is suitable for utilization of the heat coming from the engine coolant loop. The FAM-Z02 adsorbent was embedded into a lightweight finned flat-tube heat exchanger specially manufactured by the TOPMACS's partner Valeo Thermal systems.

Under real driving conditions the available waste heat from a truck engine is very variable. Additionally the good operation of the engine requires, first not using the heat before the engine has being warmed up, and after the warm up period, then sending the water back to the engine neither too cool nor too hot. All these requirements make the input temperature and mass flow rate to the adsorption cooling quite variable. On the other hand the conditions at the cabin are neither constant. The initial temperature is quite high and it drops as the air conditioning starts to operate. This leads to a considerable variation of the temperature and humidity at the cabin until the comfort conditions are attained, this variation is transmitted from the cabin to the evaporator and then to the reactors. In such variable operating conditions, the optimization of the design and of the operation of the system becomes a considerable difficult task.

In order to assist the design of the prototype and estimate the performance of the system under driving conditions, a global mathematical model of the overall system (engine, sorption cooling system and cabin) has been developed under MATLAB[®] Simulink[®] programming environment. The global model has been employed to carry out a first estimation of the innovative cooling system performance, the results of which are presented in this paper. Also a comparison with the experimental results of the system tested at the lab under steady flow conditions is included showing the excellent capabilities of the model to predict the dynamic behaviour of the system.

2. DESCRIPTION OF THE SORPTION COOLING SYSTEM

Figure 1 shows the layout of one of the possible implementations of the adsorption machine in the overall air conditioning system of a vehicle cabin.

Solid adsorption cooling systems have high potential for application in automotive air conditioning. The main source of waste heat from the engine is the engine cooling water. The temperature at the outlet of the engine normally lies between 90 – 95 °C for cars and 80-90 °C for trucks. The water temperature can then be increased by heat recovery from the exhaust gases in the heat exchanger shown in the figure. However this option was not considered in the final design of the system because the inherent increase in cost and complication of the system.

The water flow rate available to activate the cooling system is the same that normally goes through the radiator to cool down the engine. The engine needs to warm up as fast as possible, so that during the warming up period there is no waste heat available. When the engine water temperature reaches the adequate level, the thermostat opens the cooling circuit throughout the radiator. Alternatively the hot water is sent to activate the sorption system. This means that during the warm up of the engine no desorption can be carried out until the engine is efficiently warm. Anyhow, cooling the cabin is still possible provided that one or two reactors have been kept dry, so that they can absorb vapor from the evaporator and produce cooling effect since the beginning.

The hot water from the outlet of the engine is then directed towards one of the reactors (adsorber 1 in figure 1) and circulates throughout it producing then desorption of the water inside the reactor. The bed pressure increases due to the vapor desorption and the valve between the bed and the condenser opens (the communication valve with the evaporator is kept closed). The vapor condenses in the condenser. Liquid water passes from the condenser to the evaporator through the expansion device. At the same time, the other reactor (adsorber 2) is being cooled thanks to an auxiliary hydraulic loop which dissipates the heat to the ambient. The reactor starts to adsorb the vapor, and then the pressure decreases because the reactor valves are all closed. When the bed pressure becomes lower than the one in the evaporator, the communication valve opens and the reactor starts to adsorb the vapor coming from the evaporator, producing the evaporation of the water in it and the cooling effect over the air of the cabin which flows through the external part of the evaporator. Then, the water from the auxiliary cooler is conducted to adsorber 1, which is dry and hot, cooling it down, decreasing its pressure and activating the adsorption from the evaporator. At the same time, adsorber 2, which is full and cooled, is heated up to increase its pressure and activate the desorption process.

No heat/mass recovery has been considered in the system because it would increase the weight and complexity (more piping, valves, and more complicated management) and additionally, the duration of the cycle time would become longer and hence cooling capacity lower.

3. OVERALL MODEL

The main purpose of developing the Overall Model is to simulate the performance of the entire system (engine, sorption cooling system and cabin). Therefore, the Overall Model was developed to take into account all the systems and subsystems, from the engine to the air cabin, in a way that the entire A/C system can be virtually assessed similarly to the real system during typical standard characterization tests. The model has been developed under MATLAB[®] Simulink[®] mathematical programming environment and includes models for each of the mentioned systems. The driving cycle assessment is performed under Simulink[®].

The global model is completely dynamic and is able to reproduce the operation of the engine throughout the driving cycle, evaluate the waste heat available at the engine cooling hydraulic loop and at the exhaust gases, calculate the sequential operation of a double bed adsorption chiller,

calculate the condensation of the vapor at the condenser and the cooling effect produced at the evaporator, and finally, the temperature and humidity evolution of the air at the cabin, as a function of the external sun radiation, ambient temperature and vehicle velocity.

The nature of the Overall Model is completely non-steady. The inputs to the model are the same as the inputs for a conventional A/C system during a typical vehicle characterization test which follows a standard driving cycle: ambient temperature and humidity, A/C fan velocity, instantaneous power and engine speed (rpm), and finally, gear and accelerator position.

Of course all the geometrical and operational characteristics of the engine, hydraulic systems, reactors, coolers, and condenser and evaporator must also be provided.

A convenient lumped parameter model has been developed for each subsystem of the global system in such a way that they provide for a reasonable accuracy in the estimation of their performance as well as a moderate calculation time. One basic 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 reactor is based on the state equation as well as on the mass conservation. 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 analyzed with the employed formulation model.

3.1 REACTOR MODEL

In the modelling phase, and as a result of the research activity carried out by CNR-ITAE, the next assumptions have been considered:

- Non equilibrium conditions with a simple kinetic model is considered at the adsorption-desorption bed reactors, 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 bed (due to particle porosity, clearance volume, etc.) which is partially filled with water vapour. So that, the pressure at the reactors depends on the instantaneous mass of vapour contained inside.
- The flow of water vapour among the bed reactors, the condenser and evaporator is governed by the pressure difference in between them and the position of the valves.
- The heat exchangers are characterised by their global UA value (W/K). For the bed reactors, a detailed analytical study has been carried out in order to estimate an adequate UA value depending on the sorbent thermal properties as well as the geometrical characteristics of the reactor.

In the following, the basic governing equations employed for the double-bed reactor modeling are described:

Conservation equations during the desorption phase (assumed to be bed reactor 1):

The energy equation for the bed reactor leads to the following two ODEs for the bed temperature and the outlet temperature of the hot water leaving the reactor:

$$C_{b1} \frac{dT_{b1}}{dt} = M_s \cdot \frac{dw}{dt} \Big|_{b1} \cdot \Delta H(T_{b1}) + \varepsilon_{b1} \cdot \dot{m} C p_{water}^{engine \text{ coolant}} \cdot (T_{water, inlet}^{engine \text{ coolant}} - T_{b1}) \quad (1)$$

$$C_{water}^{engine \text{ coolant}} \frac{dT_{water, outlet}^{engine \text{ coolant}}}{dt} = \dot{m} C p_{water}^{engine \text{ coolant}} \cdot (T_{water, inlet}^{engine \text{ coolant}} - T_{water, outlet}^{engine \text{ coolant}}) - \varepsilon_{b1} \cdot \dot{m} C p_{water}^{engine \text{ coolant}} \cdot (T_{water, inlet}^{engine \text{ coolant}} - T_{b1}) \quad (2)$$

The uptake w is the instantaneous uptake at the bed reactor. It is assumed that the rate of desorption depends on the difference between the instantaneous uptake at the reactor and the one that would be obtained at equilibrium w_{eq} [7]:

$$\frac{dw_b}{dt} = \left(k_1 \cdot e^{-k_2/T_b} (w_{eq} - w_b) \right) \quad (3)$$

The equilibrium uptake is mainly a characteristic of the sorption material and grain size and must be determined experimentally. The following equation is employed to characterize the solid/water equilibrium [12]:

$$\ln P_b = A(w_{eq}) + \frac{B(w_{eq})}{T_b} \quad (4)$$

Typically, 3rd degree polynomials are employed for A and B to correlate the experimental results. The employed material has been experimentally characterized by CNR ITAE [12, 13].

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

$$P_b = P_a + P_v = \frac{RT_b}{V_{bed}} \left(\frac{m_a}{M_a} + \frac{m_v}{M_v} \right) = \frac{RT_b}{V_{bed}} \frac{m_v}{M_v} \quad (5)$$

Where P_a is the pressure due to the non-condensable gases inside and P_v is the pressure due to the water vapor. In the following it is assumed that non-condensable gases have been totally removed. The ODE for the pressure at the reactor then results:

$$\frac{dP_b}{P_b} = \frac{dm_v}{m_v} + \frac{dT_b}{T_b} \rightarrow \frac{dP_b}{dt} = P_b \left(\frac{1}{m_v} \frac{dm_v}{dt} + \frac{1}{T_b} \frac{dT_b}{dt} \right) \quad (6)$$

Finally, the continuity equation at the reactor provides the necessary link between the uptake variation and the vapor flow rates leaving or entering the reactor.

$$\frac{dm_v}{dt} = -M_s \frac{dw}{dt} - \dot{m}_{v, out} + \dot{m}_{v, in} = -M_s \frac{dw}{dt} - \dot{m}_{v, des} + \dot{m}_{v, ad} \quad (7)$$

This set of equations constitutes a system of 4 ODEs for T_b , P_b , w_b and m_v

Conservation equations during the adsorption phase (assumed to be bed reactor 2):

The energy equation for the bed reactor leads to the following two ODEs for the bed temperature and the outlet temperature of the cooling water leaving the reactor. The only difference with the corresponding equation for the desorption phase is that now the energy equation must include the term corresponding to the sensible heat required to heating up the vapor from the evaporation temperature up to the value corresponding to the adsorbing reactor.

$$C_{b2} \frac{dT_{b2}}{dt} = M_s \cdot \frac{dw}{dt} \Big|_{b2} \cdot (\Delta H(T_{b2}) - C_{p_v}(T_{b2} - T_{evap})) - \varepsilon_{b2} \cdot \dot{m} C_{p_{water}}^{air\ cooler} \cdot (T_{b2} - T_{water, inlet}^{air\ cooler}) \quad (8)$$

$$C_{water, air\ cooler} \frac{dT_{water, outlet}^{air\ cooler}}{dt} = -\dot{m} C_{p_{water}}^{air\ cooler} \cdot (T_{water, outlet}^{air\ cooler} - T_{water, inlet}^{air\ cooler}) + \varepsilon_{b2} \cdot \dot{m} C_{p_{water}}^{air\ cooler} \cdot (T_{b2} - T_{water, inlet}^{air\ cooler}) \quad (9)$$

The rest of the equations for w_b , p_b , and m_v are the same as (2), (5) and (6).

In order to close the reactor model, it is now necessary to include equations for the calculation of the flow rate of vapor in between the reactors, the condenser, and the evaporator, through the interconnecting pipes and valves. 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 can be calculated as follows:

$$\dot{m}_{v, out} = \begin{cases} = 0 & \text{if } P_b < P_{cond} \\ = A_{cond} \sqrt{2 \cdot \rho_v(T_b, P_b) \cdot (P_b - P_{cond})} & \text{if } P_b \geq P_{cond} \end{cases} \quad (10)$$

$$\dot{m}_{v, in} = \begin{cases} = 0 & \text{if } P_b > P_{evap} \\ = A_{evap} \sqrt{2 \cdot \rho_v(T_{evap}, P_{evap}) \cdot (P_{evap} - P_b)} & \text{if } P_b \leq P_{evap} \end{cases} \quad (11)$$

Finally, adequate control programming simulating the operation of the valves to switch the alternatively heating/cooling of each reactor has been implemented.

3.2 CONDENSER AND EVAPORATOR MODEL

The following general assumptions for condenser and evaporator have been considered:

- The water vapour is assumed to be a perfect gas

- Fluid is considered always in thermodynamic equilibrium corresponding to saturation conditions.
- Pressure is dependent on the amount of vapour existing in the top of the condenser/evaporator. It is then assumed that temperature of the liquid condensing around the cold surfaces reacts instantaneously to the variation of pressure keeping under saturation. This assumption is perfectly reasonable at the condenser since the water liquid film around the tube is very thin. It is also assumed that the amount of water liquid stored in the condenser is small. Therefore, temperature is calculated as the saturation temperature corresponding to the instantaneous pressure of condenser/evaporator. At the evaporator this hypothesis is less accurate since the amount of liquid is much higher, but in any case the simplification is justified because variations in pressure during operation are rather small.
- It is assumed that the entire water flow rate coming from the condenser is instantly evaporated. In other words, it is assumed that there is an expansion device which keeps constant the liquid level at the evaporator, so that the amount of incoming liquid exactly balances the evaporation rate. This means that the outlet water flow rate of the condenser is controlled by the evaporation rate.

In the following, the ODEs for the condenser are presented and explained. The equations for the evaporator are very similar and based on the same physical hypothesis.

CONDENSER

The energy conservation equation for the vapour region surrounding the condenser tubes provides a way to evaluate the condensation rate:

$$\dot{m}_{cond} = \frac{\dot{m}_{sec} C_{p_{sec}} \cdot \epsilon_{cond} \cdot (T_{sec,out} - T_{sec,in})}{C_{p_{w,vapour}} (T_{cond,in} - T_{cond}) + h_{fg}(P_{cond})} \quad (12)$$

Now, the equation of state for the vapour provides a way to evaluate the variation of the pressure as a function of the variation of the vapour mass, volume and temperature:

$$P_{cond} = \frac{RT_{cond}}{V_{cond}} \frac{m_{v,cond}}{M_v} \rightarrow \frac{dP_{cond}}{P_{cond}} = \frac{dm_{v,cond}}{m_{v,cond}} + \frac{dT_{cond}}{T_{cond}} \quad (13)$$

The variation of the vapour volume comes from the variation of the liquid level in the condenser vessel:

$$dV_{v,cond} = -dV_{liquid} \quad (14)$$

Finally, the assumption of saturation provides a way to relate the temperature variation on the above expression with the pressure variation.

$$T_{cond} = T_{sat}(P_{cond}) \rightarrow dT_{cond} = \frac{dT_{sat}(P_{cond})}{dP} \cdot dP \quad (15)$$

Combining equations (13) and (15), the following ODE for the condensing pressure is obtained:

$$\frac{dP_{cond}}{dt} = \frac{\left(\frac{P_{cond}}{m_v}\right) \times \left(\frac{dm_v}{dt}\right)}{1 - \frac{P_{cond}}{T_{cond}} \frac{dT_{sat}}{dP}} \quad (16)$$

$$\frac{dm_v}{dt} = \dot{m}_{cond,in} - \dot{m}_{cond} \quad (17)$$

The mass conservation equation for the vapour (17) links the variation of vapour mass in the condenser with the incoming mass flow rate from the reactor and the condensation rate.

The obtained system of ODEs (equations 13 and 17) allows for the calculation of the instantaneous variation of the mass of vapour and of the condensing pressure. Equation (12) allows for the estimation of the condensation rate, and the saturation hypothesis, for the evaluation of the condensation temperature.

The energy conservation equation leading to Equation (12) does not include the energy required for the variation of temperature for the condensed liquid, and it is also assumed that the liquid follows the condensing temperature variations. A trial to include such a term into the equations was done but then a new ODE is necessary with strong coupling with the other equations, and no available solver was able to find the solution of the resulting system of equations. In any case, the neglected term is very small.

Finally, it is possible to write the mass conservation equation for the liquid in the condenser and therefore providing the link between the liquid volume and the vapour volume:

$$\frac{dm_l}{dt} = \dot{m}_{cond} - \dot{m}_{evap} \quad (18)$$

$$dV_{liquid} = \frac{dm_l}{\rho_l} \quad (19)$$

CONDENSER SECONDARY FLUID

The energy conservation equation for the secondary fluid becomes:

$$C_{cond} \frac{dT_{sec,out}^{cond}}{dt} = \dot{m}_{sec} \cdot Cp_{sec} \cdot (T_{sec,in} - T_{sec,out}) - \epsilon_{cond} \cdot \dot{m}_{sec} \cdot Cp_{sec} \left(0.8 \cdot (T_{sec,in} - T_{cond}) + 0.2 \cdot \frac{T_{sec,out} - T_{cond}}{(1 - \epsilon_{cond})} \right) \quad (20)$$

4. DESCRIPTION OF THE LAB SCALE ADSORPTION CHILLER PROTOTYPE

The system consists of a double bed adsorber, working in counter phase to get a quasi-continuous cooling effect, connected with an evaporator and a condenser. The adsorbent bed was designed to be light, thermally efficient and suitable to work with a low temperature heat source.

The adsorbent material used belongs to a novel family of materials, called AQSOA-Functional Adsorbent Materials (FAMs) developed by Mitsubishi Chemical for applications where low temperature heat sources are available, while water was used as refrigerant. Indeed, in a previous work [11] was demonstrated that material FAM-Z02 can be efficiently used with a maximum desorption temperature of 80-90°C. Loose grains of adsorbent (with particle size of 0.250 – 0.450 mm) were embedded into an aluminium finned flat tube heat exchanger (Figure 2) and the single module adsorber realized was tested in a specific lab scale test previously installed in the CNR-ITAE laboratories. Figure 2 shows a single heat exchanger (or “module”) used and a detailed view of the embedded zeolite grains. Table 1 reports the main properties of a single module, which allows rather low mass metal/mass adsorbent ratio.

The obtained results allowed to properly conceive the full-scale double bed adsorber, which was designed to provide a continuous cooling power in the range of 2 – 5 kW depending on the operating conditions.

The Figure 3 shows the full scale laboratory adsorption chiller which consists of a twin-beds adsorber, a single evaporator and condenser. The vacuum chambers containing the double adsorber have been specifically designed to fit the adsorbent bed and present various flanges that allow the connection with the other components of the prototype and the installation of the measurement devices (pressure gauges, temperature sensors), necessary to control and manage the system during testing. The twin-beds adsorber consists of 6 + 6 single modules holding the zeolite grains by a light metallic net. The evaporator and the condenser are “two phases” exchangers, in which both liquid and vapour phases are present. A copper finned tube coil as inner exchanger allows large heat exchange surface and high efficiency. The bottom side of the exchanger cools down/heats up the liquid phase, while the upper part exchanges thermal power with the steam. This feature allows high evaporation/condensation from the liquid phase inside the evaporator/condenser.

The system realized is able to work in different operating conditions. Table 2 reports the settable temperatures and flow rates for the heating source (T_h , F_h), the evaporator (T_{ev} , F_{ev}) and the condenser (T_{con} , F_{con}).

In order to interface the external heat source/sink with the adsorption chiller a test bench was realized. The test bench allows to perform the different phases of the cooling cycle and to measure the relevant parameters.

Components installed (plate-type heat exchangers, temperature and pressure sensors, flow meters, electro-valves, and hydraulic pumps) are electronically managed. In fact, a data acquisition and control system was realized by specific software implemented by the LABVIEW language. It allows the fully-automatic operation of the system and the management of different control strategies, including single bed and double bed operation mode. The high temperature water was produced by a

boiler in order to simulate the waste heat from the engine. As well, a heat sink and a specific hydraulic loop allow the cooling of the adsorbers and the condenser. The data acquisition and control system of the testing bench is integrated with the management of the adsorption chiller. In particular, the four steps of the thermodynamic cycle can be “time-controlled” or “p,T-controlled”. It means that the phases can be performed until reaching a fixed duration or fixed values of pressure and temperature of the adsorbent beds.

5.

EXPERIMENTAL AND MODEL RESULTS

In this work, more than 2000 tests (each test made of 30 runs) were performed at different operating conditions. In the graphs below, a sample of the experimental results obtained with water inlet temperature of 90°C, condensing of 35°C and evaporating temperature of 10°C are presented in comparison with the model results. Based on experimental results the average cooling power ranges from 2 up to 3 kW, with a maximum peak of about 5 kW and a COP of 0.6. The useful effect is continuous, which means that the design of the double bed system was correct.

Figure 4 shows the comparison between bed pressure and outlet temperature predicted by the model and experimental results for the first 3000 seconds at the above mentioned operating conditions. The experimental results show some differences between the temperature profiles of the reactors. They are clearly distinguishable for the pressure at the lowest values which correspond to the adsorption phase. This difference could be caused by different pressure losses in the flow from the evaporator to the reactor across the corresponding valves, or simply due to the uncertainty of measurements. However, also the reactor temperature at the outlet of the reactor is not symmetric, showing slightly differences between both reactors. This can be due to a different permeability and compactness of the adsorbent material in the reactor, leading to the observed differences in behavior. The model is not able to reproduce those differences, since it assumes that both reactors are identical.

Figure 5 shows the comparison between calculated and measured results for the pressure at the evaporator and condenser. As can be observed, the adjustment between calculated and measured results is very good in phase and amplitude regardless the series of hypothesis assumed for their modeling, especially the assumption of saturated conditions.

Figure 6 shows the comparison between the calculated and measured useful cooling power. The adjustment, in general, is remarkably good taking into account that the useful cooling is the final result of the system and the approximate nature of the model. It should be also pointed out that experimental uncertainty in the evaluation of the cooling capacity is quite high, since the average temperature difference of the water across the evaporator is just 1.4 °C.

All in all, the obtained adjustment is very good, so 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. This feature is of great importance for the developed model, since under real driving operation at the vehicle, the conditions are fully transient.

Finally, Figure 7 shows the equilibrium value of the uptake calculated from the reactor pressure and temperature, for both measurements and calculations. The dashed line in the figure corresponds to the calculated values of the instantaneous uptake. As can be observed, the calculated and measured equilibrium uptake is very similar. There are only a few peaks on the experimental results which are not explained by the model. Anyhow, the equilibrium uptake for the experimental results has been evaluated from the instantaneous recordings of bed pressure and temperature, and the temperature inside the reactor is difficult to measure, especially at the switching moments in which these peaks appear. On the other hand, the observed differences between the equilibrium and non equilibrium uptake, and the good agreement obtained between experimental and measured results, seems to indicate that the kinetics of the adsorption/desorption plays an important role in the dynamics and performance of the system, and that the employed model for the kinetics is enough accurate to describe the process.

This agreement between measured and experimental results was checked to be good for all the tested conditions of the prototype at the laboratory, so the model can be effectively used to estimate the possible performance of the system on the vehicle.

6. ESTIMATION OF THE SYSTEM PERFORMANCE ON VEHICLE

The overall model is capable of simulating the whole A/C characterization test for the truck under a standard driving cycle. The driving cycle used is the Normal European Driving Cycle (NEDC) and is constituted of four repetitions of elementary urban cycles at low speed (ECE) and one higher speed extra urban cycle (EUDC). It incorporates most of the details of the adsorption-desorption system, the engine and the cabin. Different conditions have been simulated in order to assess the performance of the whole system (engine, sorption cooling system and cabin). The test has been performed at ambient temperature of 28°C and relative humidity of 50%, and a cool-down test at severe conditions has been also considered.

These assessment tests have been carried out assuming three different start-up conditions. The first one is assuming that both beds are saturated. This means that all the valves of the sorption system are open, so that all the system is in equilibrium with the water stored in the system at ambient temperature. Under those conditions, the reactors and the water are at ambient temperature; the pressure is the one corresponding to saturation, and the uptake corresponds to the equilibrium value. This strategy is the worst case scenario because it does not allow the system to produce cooling effect since the beginning. The second start-up condition is considering one of the beds dry, which is possible by keeping the valves closed and the sorption bed isolated from the rest of the system. In this way, one sorption bed remains with the maximum uptake and fully loaded with water, while the other sorption bed remains with the minimum uptake and dry, ready to adsorb water vapour when the valve connecting the evaporator and the sorption bed opens. Finally, the third start-up condition is keeping both beds dry by using the same strategy. In real operation, the beds could be initially dry if, after the use of the system during operation, the beds are dried with the remaining waste heat stored in the engine, once the engine has been switched off. As mentioned above, this could only be

possible if additionally, the valves communicating the beds with the condenser and evaporator are able to practically seal the beds when closed.

The optimization study has covered the whole range of variation of ambient conditions at the different possible start conditions in order to evaluate their influence on the obtained performance.

Figure 8 shows the simulating results obtained along the test at ambient temperature of 28°C and relative humidity of 50% for the case of most favorable start-up condition: both sorption beds are dry.

Figure 8 (a) shows the temperature and pressure in the reactor as a function of time. The behavior of the system during the initial period is influenced by the fact that there is no available waste energy at the engine to activate the system. In the first approximately 200 seconds the engine temperature is under 75°C, so it is not possible to use the engine coolant as a heat source. When the engine temperature goes above that threshold which insures high engine efficiency, the system is able to use the engine coolant as a heat source. Given that a truck operates in quite constant driving conditions, after the indicated initial conditions, the engine is able to provide a considerable amount of hot water at almost constant temperature along the rest of the driving cycle. Therefore, the adsorption-desorption cycles become quite regular.

Figure 8 (b) shows the uptake evolution. As it can be seen the bed 1, which is being heated, is kept dry (with the minimum uptake) until the other bed reaches the equilibrium conditions, this happened approximately 600 s after the engine start-up. At the same time, the bed 2, which is being cooled, is adsorbing water vapour from the evaporator until the cycle is reversed. After 600 s (10 min), the cycle is reversed, then bed 2 starts to adsorb while bed 1 starts desorbing. Uptake variation ranges from 12% to 24%.

Figure 8 (c) shows the produced cooling power as a function of time. As it can be observed, the system is able to produce cooling effect since the very beginning given that the beds are assumed to be initially dry. As can be seen, the system is able to produce an average refrigeration capacity of approximately 1400 W.

Figure 8 (d) shows the cabin temperature as a function of time. The temperature decreases during each cooling cycle. Then, during the reversing period, it slightly increases due to the heat transferred from the body of the cabin to the inner air. As can be observed, the system is able to cool down the air quite fast, reaching a quite low temperature. The final cabin temperature depends on the test. In this case, the system is able to decrease the cabin temperature up to 14 °C. In real use, the cabin thermostat would switch off the system once the comfort temperature has been reached. The air temperature control has not been employed in the assessment tests in order to facilitate the comparison among the different situations analyzed.

Figure 9 shows the simulating results obtained along the test at ambient temperature 28°C and relative humidity 50% for the different considered start-up conditions: 2 beds saturated, 1 bed dry and 2 beds dry. The system with two beds dry is able to produce a mean cooling power of 1400 W, with one bed dry is able to produce a mean cooling power of 1295 W, and with both beds saturated is

able to produce a mean cooling power of 974W. In all start-up conditions the system is able to reach a cabin temperature lower than 20 °C which is the comfort temperature.

As can be seen, when the two beds are saturated, the system takes a long time to start producing refrigeration effect (around 400 s), while if at least one bed dry it is possible to have refrigeration effect since the beginning. With two beds initially dry the system is logically able to produce a higher capacity and therefore to decrease faster the temperature of the cabin. Nevertheless, the difference in between having two or one bed dry is not so important. This indicates that in order to have good performance it would only be necessary that after the engine is switched off, one of the reactors is dried and then sealed.

Figure 10 shows the simulating results obtained along the cool-down test, again for the best start-up condition: 2 beds dry. The cool-down test is the most severe of all assessment tests. The ambient temperature is set at 40°C, however, the temperature of the cabin is higher due to the soaking (46°C). These are highly severe conditions for the thermal compressor and as a consequence the performance of the system drastically deteriorates, the sorption system is able to produce cooling effect but not enough to carry the cabin temperature to comfort conditions, and is only able to keep the cabin at around 39 °C. The system is obviously undersized if it has to cope with these severe conditions. However, the system is still able to produce a refrigeration capacity of approximately 1250 W. For the worst scenario (two beds saturated) the system is only able to produce a refrigeration capacity of approximately 1000 W.

The reason for the deterioration of the performance of the system is mainly the increase of the ambient temperature for the rejection of the heat, so that the minimum bed temperature is 44 °C. As can be seen in figure 10 (b) the range of variation of the uptake then becomes very small and the cycles become very short. The operation could be improved by modifying the switching criterion and increasing the cycle time.

A prototype of adsorption chiller for a truck was finally realized by CNR-ITAE in the framework of the EC project TOPMACS. The overall size of the prototype was 170 dm³ and its weight 60 kg so it was suitable for mobile applications. The specific cooling power at the laboratory was around 300-600 W/kg of adsorbent. More details of the prototype can be found in [14].

7.

CONCLUSIONS

A model of a possible sorption cooling system for automotive applications driven by the engine waste heat has been developed. The model is completely dynamic and is able to reproduce the operation of the engine through a standard driving cycle, evaluating the waste heat available at the engine hydraulic loop and at the exhaust gases, calculating the sequential operation of a two-reactors adsorption-desorption system, calculating the condensation of the vapor and the cooling effect produced at the evaporator, and finally, the temperature and humidity evolution of the air at the cabin.

The model of the sorption-desorption system has been compared with experimental results of a double-bed zeolite-water vapor system. The experimental results allowed for the adjustment of the constant of the kinetics. Once adjusted, the calculated results are in very good agreement with the experiments, proving the good capabilities of the model to predict the system performance. Then, the overall model has been able to reproduce the behavior of the engine and the available waste heat through a standard driving cycle and has been employed for a first assessment of the cooling system with a two beds cooling system.

The first assessment of results shows that the system could be able to provide for a significant amount of the required cooling. A design and optimization study will follow now to determine the maximum cooling effect that it is possible to get from the available waste heat all along the driving cycle, depending on the employed sorption material, the size of the reactors, the size of the auxiliary components: evaporator, condenser and cooler, and the operating parameters.

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NOMENCLATURE

| | |
|------------|--|
| A | effective flow area |
| C | thermal capacity |
| C_p | specific heat |
| h_{fg} | water phase change enthalpy |
| ΔH | sorption heat |
| k_n | constants of the Kinetics equation (n=1,2) |
| m | mass |
| \dot{m} | mass flow rate |
| M | mass (also molecular mass) |
| P | pressure |
| r | Universal gas constant |
| t | time |
| T | temperature |
| V | volume |
| w | uptake |
| ρ | density |

ε heat exchanger effectiveness

SUBSCRIPTS

a air
ad adsorption
b bed
cond condenser
des desorption
eq equilibrium conditions
evap evaporator
h hot water
in inlet
out outlet
s sorbent
sec secondary fluid
v vapor

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TABLES

Table 1: Main properties of a single adsorber module.

| | |
|---|---------------|
| <i>Dimension (l×h×t) [mm]</i> | 257×170×27 |
| <i>Weight [g]</i> | 636 |
| <i>Overall Volume [l]</i> | 1.1 |
| <i>Adsorbent material</i> | AQSOA-FAM-Z02 |
| <i>Particle size of adsorbent [μm]</i> | 250-400 |
| <i>Adsorbent mass [g]</i> | 480 |
| <i>Metallic mass/Adsorbent mass</i> | 1.3 |

Table 2: Settable operating condition.

| | | | |
|-------------------------------------|------------|-------------------------------------|----------|
| <i>T_h [°C]</i> | up to 120 | <i>F_h [LPM]</i> | up to 45 |
| <i>T_{evap} [°C]</i> | 5 – 25 °C | <i>F_{ev} [LPM]</i> | 10 – 25 |
| <i>T_{cond} [°C]</i> | 20 – 45 °C | <i>F_{con} [LPM]</i> | 10 – 25 |

FIGURES

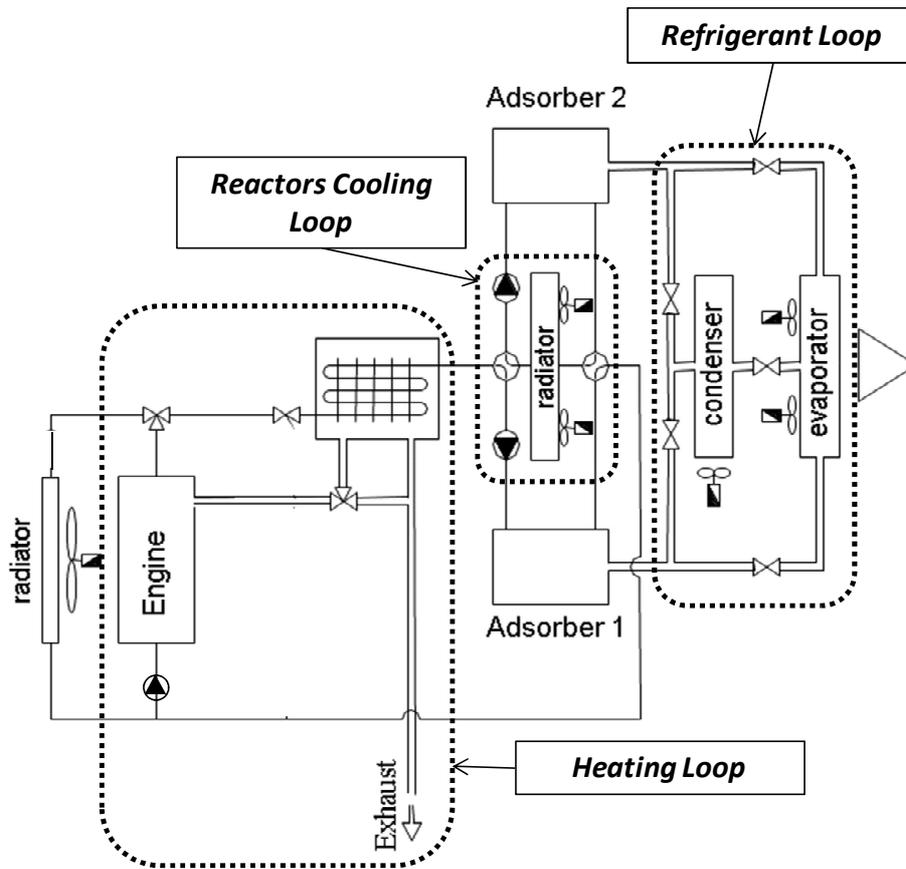


Figure 1: Adsorption-desorption cooling system layout.

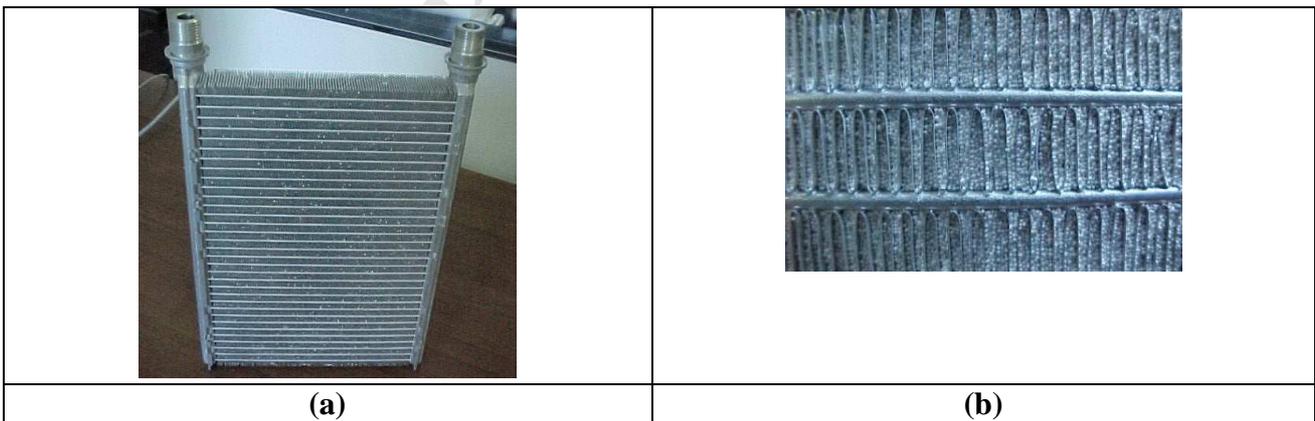


Figure 2: Heat exchanger used for the adsorber manufacture: (a) General (b) Detailed view.



(a)



(b)

Figure 3: (a) The adsorption chiller realized. (b) The full scale laboratory adsorption chiller connected to the test-bench.

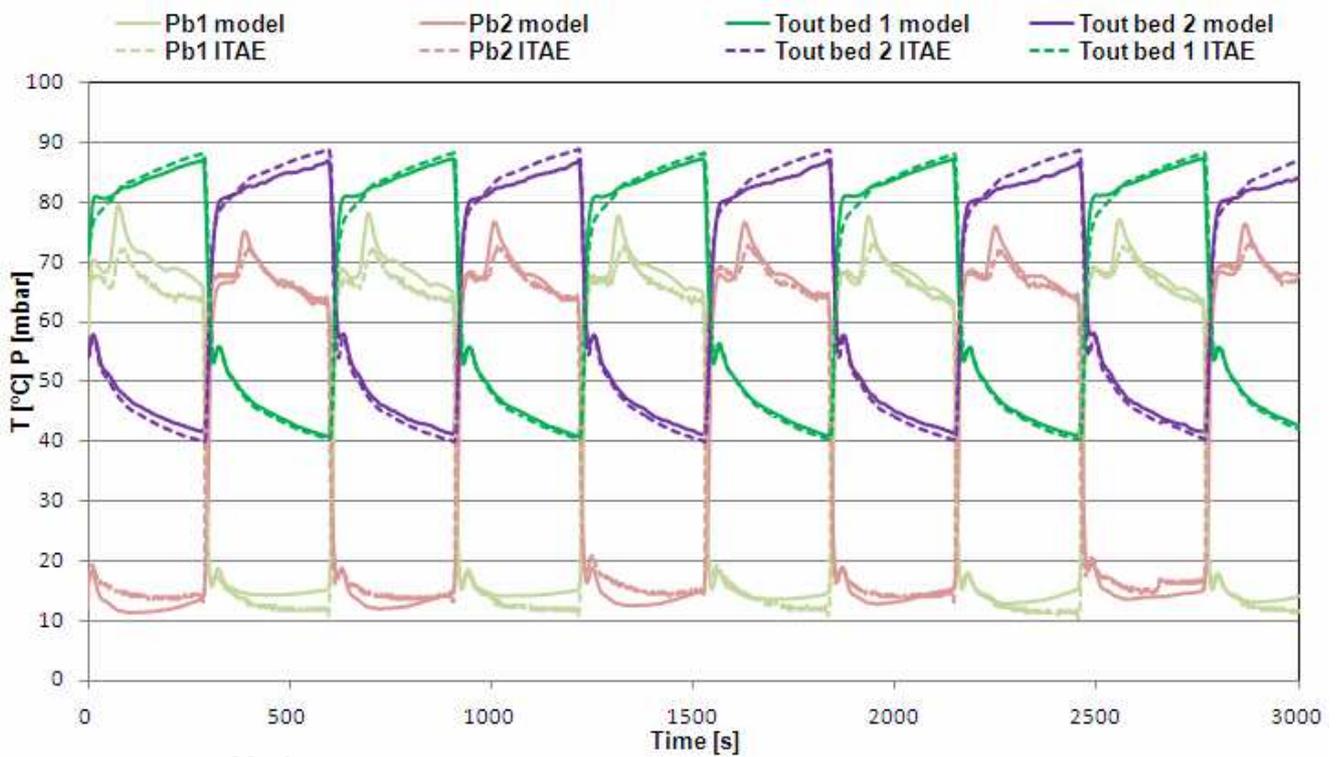


Figure 4: Comparison between calculated and measured results: bed 1 and bed 2 pressures and outlet temperatures.

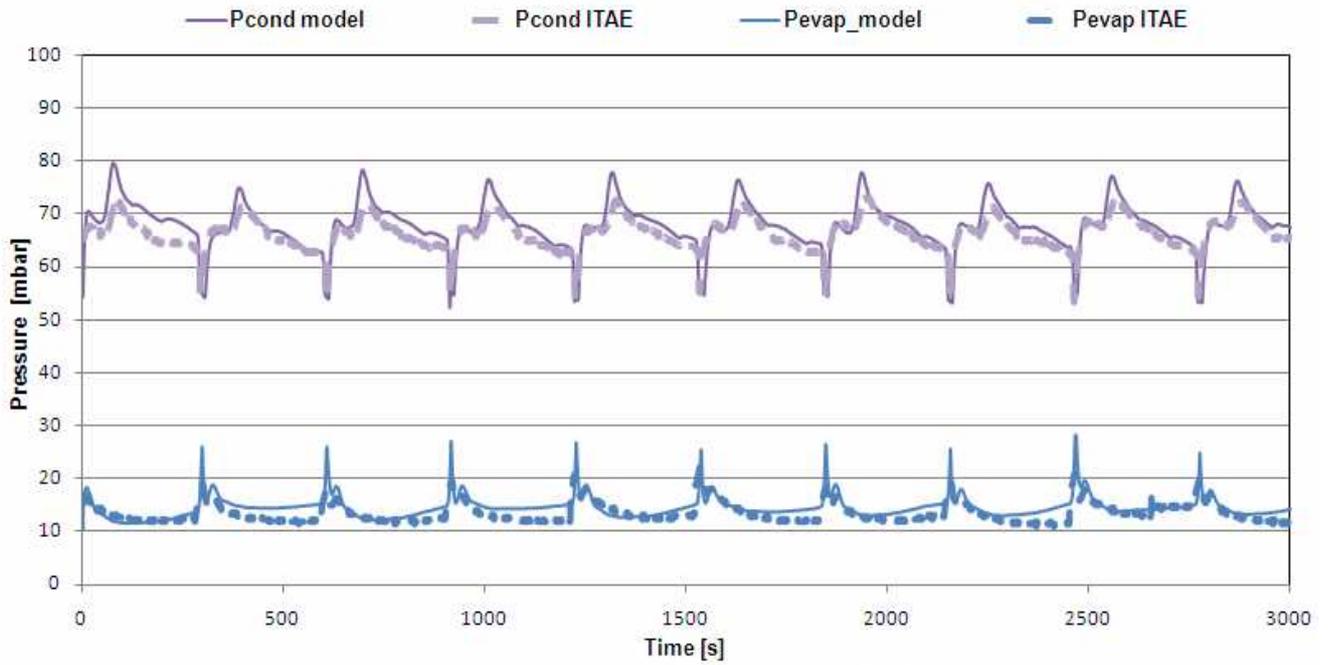


Figure 5: Comparison between calculated and measured results: pressure at evaporator and condenser.

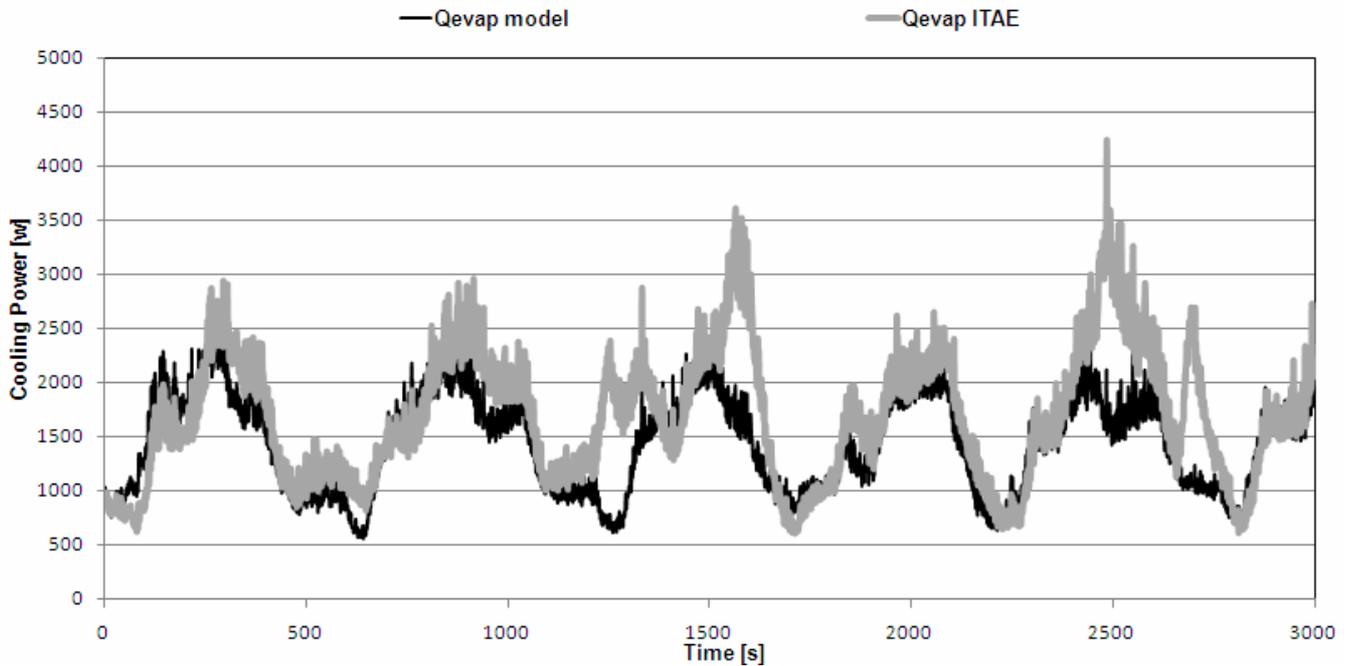


Figure 6: Comparison between calculated and measured results: evaporator cooling capacity.

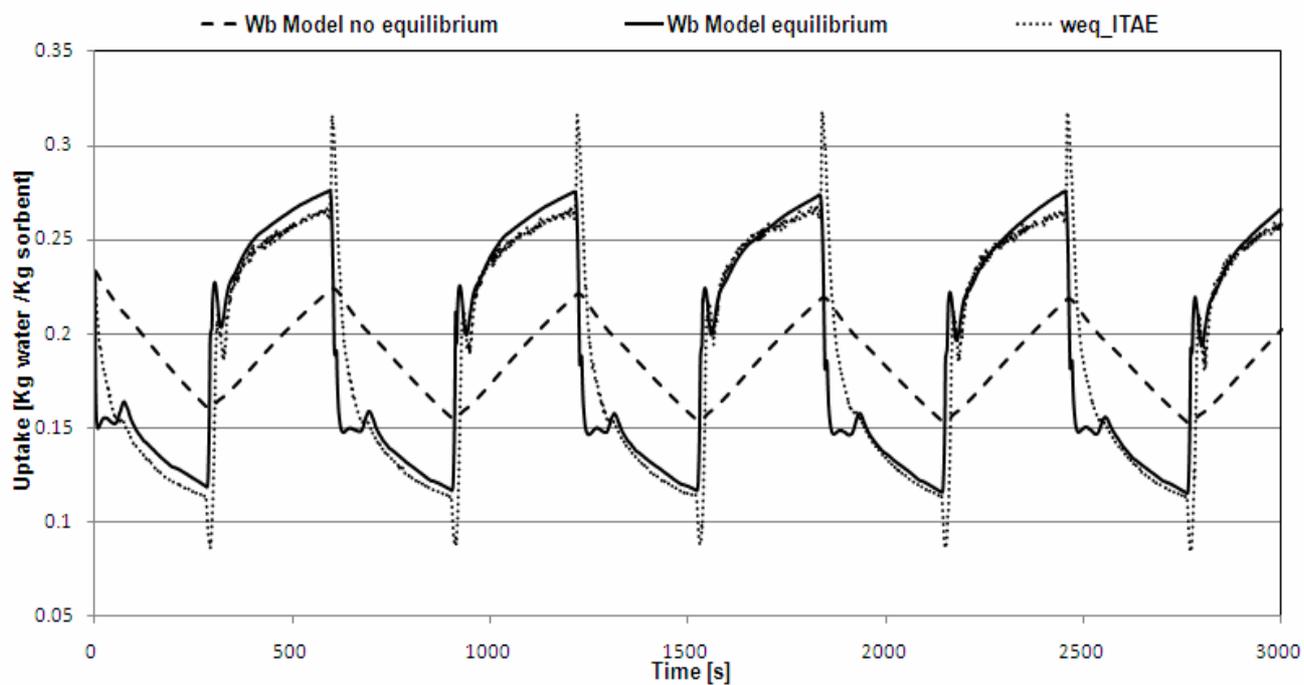


Figure 7: Uptake evolution.

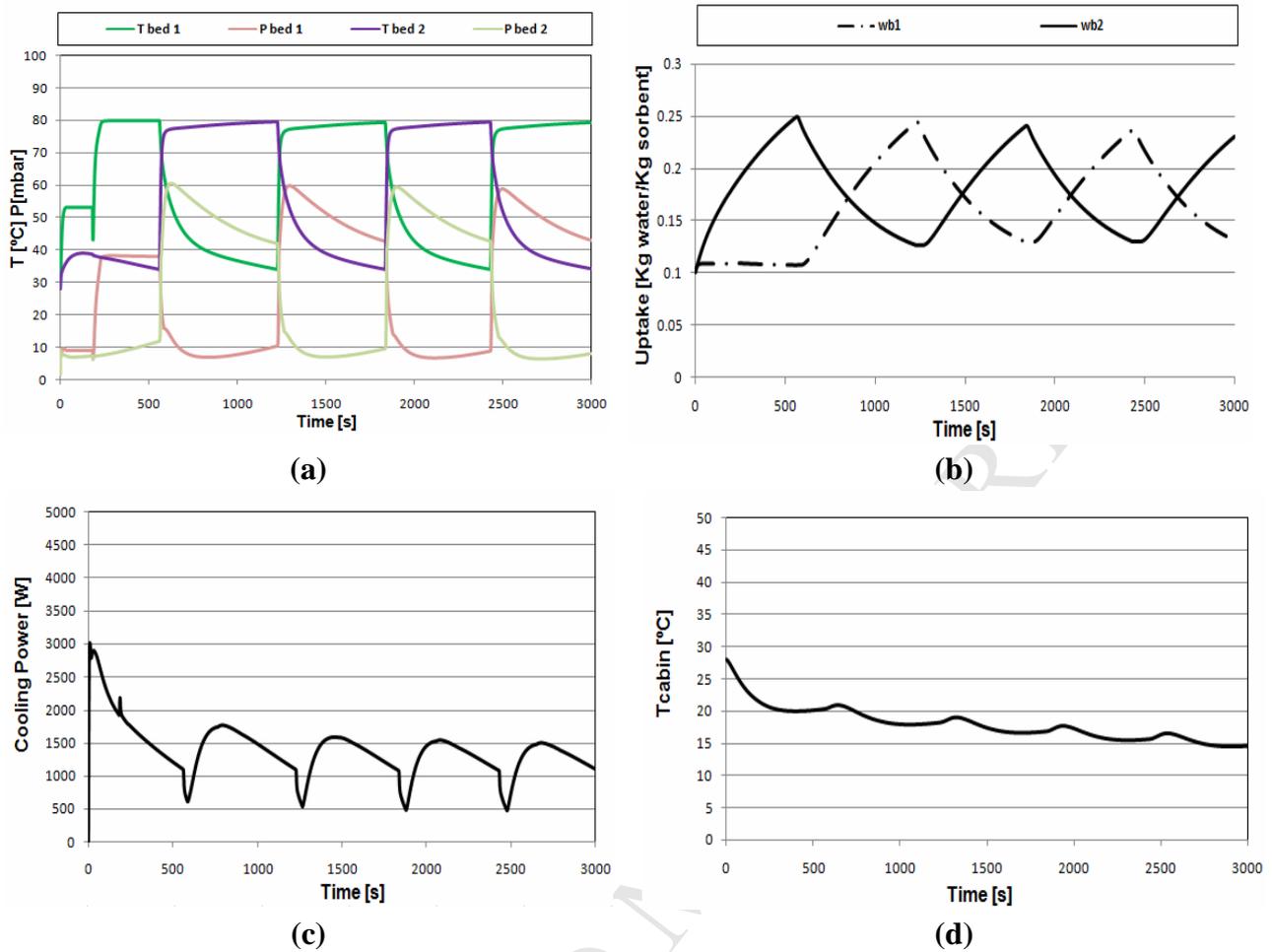
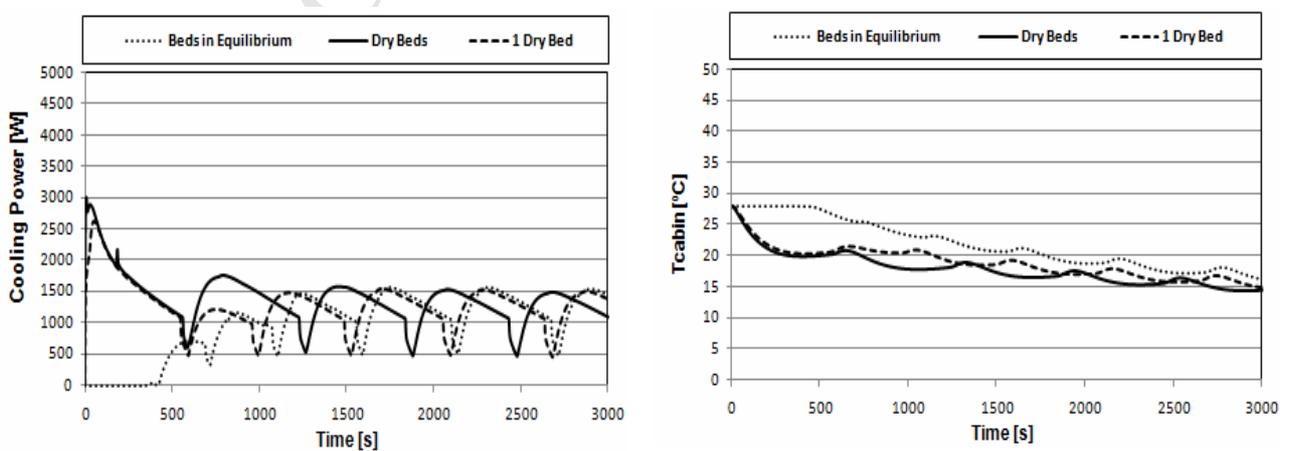


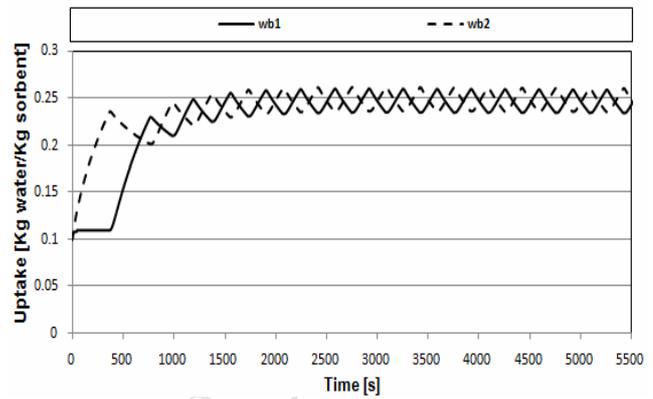
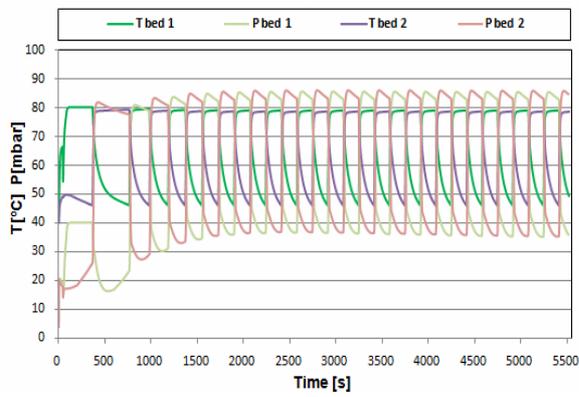
Figure 8: Calculated results for NEDC Test at 28 °C/ 50% RH and start-up strategy - 2 beds dry: (a) Temperature and pressure at the bed reactors; (b) Uptake evolution; (c) Cooling capacity of the sorption system; (d) Cabin temperature evolution.



(a)

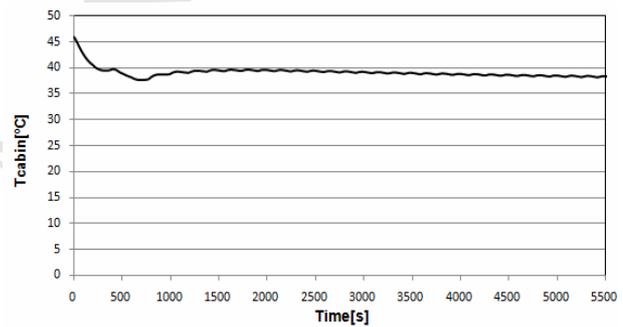
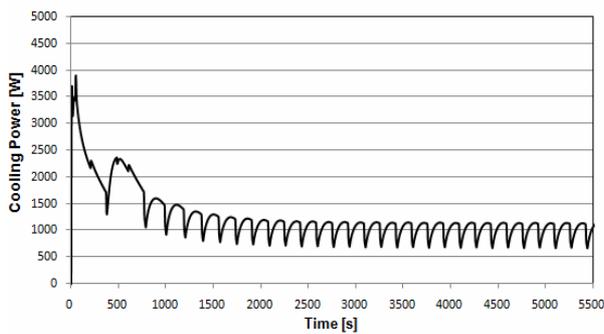
(b)

Figure 9: Calculated results at NEDC Test at 28 °C/ 50% RH and 3 start-up strategies: (a) Cooling capacity of the sorption system; (b) Cabin temperature evolution.



(a)

(b)



(c)

(d)

Figure 10: Calculated results at COOL-DOWN Test and start-up strategy - 2 beds dry: (a) Temperature and pressure at the bed reactors; (b) Uptake evolution; (c) Cooling capacity of the sorption system; (d) Cabin temperature evolution.