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Performance evaluation of a waste-heat driven adsorption system for automotive air-conditioning: Part II - Performance optimization under different real driving conditions



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ABSTRACT

In this part, Part II, of a two-part study, the validated model of part I is integrated into a general vehicle model in order to predict the performance of the system under real driving conditions. The overall model takes into account all the system components to simulate the dynamic performance of the entire system and predict the cabin temperature at the available waste heat. The system was implemented in a Fiat Grande Punto vehicle and the experimental tests were performed at the Centro Ricerche Fiat (CRF), Italy laboratories. Different design configurations were investigated to explore further improvements of the performance. Results showed that the model was able to well predict the transient performance of the system under different start-up and ambient conditions as well as the normal operating conditions. Using two radiators instead of one radiator increases the cooling capacity by 7.0% and decreases the cabin temperature by 9.1%. At the warming up period, the adsorption system faces serious difficulties to start producing the required cooling. Possible strategies to avoid this problem were studied and compared. In general, it has been proved that the amount of engine waste heat available is sufficient to produce enough cooling to keep reasonably comfortable temperatures in the cabin.

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1. Introduction

The global energy demand for refrigeration and air conditioning systems is rapidly increasing, especially in developed countries. This increase is responsible for the fossil fuels depletion and the greenhouse gas emissions due to the use of electrically driven vapor compression machines (VCRSs) [1].

At present, nearly all automotive air conditioners use traditional VCRS driven mechanically by the car engine. For a 1200 kg car running at 56 kmh⁻¹, the traditional air conditioning system can add up to 5–6 kW power drawn from the engine [2]. Moreover, the fuel consumption can increase by up to 70% for a B class car on an urban cycle under severe ambient conditions (35 °C and 60% RH) [3]. These systems reduce the automotive driving performance and increase the fuel consumption and exhaust gas emissions.

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Therefore, the searching on energy efficient, renewable, and cheap cooling technology is an important goal in the transportation industry.

In automobiles applications, methods of utilizing the waste energy from the automotive engine to drive the car air conditioning system are worthy of investigation. Overall, the efficiency of a diesel engine is about 35-40% and the rest of the energy is lost mainly to the engine coolant loop and the engine exhaust gasses [4]. The temperature of the engine coolant water is about 90-95 °C for cars and 80–90 °C for trucks [5]; while the temperature of the exhaust gas discharged from the tailpipe comes within 150-450 °C [6].

In recent years, thermally driven adsorption A/C systems have witnessed and increasing interest in automobile applications due to the fact that this system is quit, long lasting, cheap to maintain, and environmentally friendly [7-9]. In addition, it can be driven by a low-grade heat source (80-150 °C) like waste heat from automotive engines [10]. In this way, it reduces fuel consumption and exhaust gas emissions. Moreover, the Montreal Protocol, Kyoto Protocol, and Vienna Convention had agreed to phase out HCFCs and CFCs.

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However, still there is no working adsorption A/C systems available at the market for automobiles at present, because of its low efficiency and large weight [11,12]. The need for efficiency improvement promotes scientists to develop new system designs, working pairs, and mathematical modeling [13]. Most of the works are carried out on adsorption and physicochemical properties of different adsorbent-adsorbate pairs [14], different types of adsorption-desorption process, such as heat and mass recovery operations [15,16]. Moreover, many studies have been carried out to improve and optimize the design and operating parameters that affect system performance [17–20].

The application of adsorption cooling systems for automotive air conditioning has attracted significant research work. In 1993, Suzuki [21] made a preliminary study to explain the technological limits in the application of an adsorption system to passenger cars' air conditioning. He concluded that the adsorbent material and the adsorber design to improve heat transfer characteristics are the most important issues for the application of adsorption cooling systems in automobiles.

A number of researchers had investigated the adsorption cooling system for automotive application and prototypes had been built and tested with different working pairs and bed configurations. Vasta et al. [22] developed and tested an adsorption A/C system for a truck cabin. The system powered by the thermal energy coming from the engine coolant loop of the truck. Results showed that the system was able to deliver an average cooling power of 2 kW and keep the truck cabin temperature around 24 °C. Verde et al. [5] constructed and tested a zeolite-water pair adsorption A/C system for a truck cabin. The overall weight and volume of the prototype were about 60 kg and 170 dm³ respectively. A dynamic model was proposed to simulate the engine operation through a standard driving cycle to estimate the amount of waste heat available at the exit of engine hydraulic loop, the cooling capacity and the temperature and humidity of the air in the truck cabin. The proposed system can produce an average cooling power of about 2-3 kW. Tamainot-Telto et al. [23] designed an adsorption prototype with activated carbon-ammonia pair. The system was powered by the waste heat (at 90 °C) from the engine coolant water loop. An average cooling power of about 1.6 kW with an average COP of 0.22 were obtained. Wang et al. [24] presented a zeolite-water adsorption A/C system for locomotive operator cabin. The system can deliver cooling effect of 5 kW and COP of 0.25. The chilled water produced was about 8-12 °C for the fan coil in the locomotive cabin. Jiangzhou et al. [25] developed an adsorption prototype for a locomotive cabin air conditioning employing zeolite-water pair. The weight of the prototype was about 300 kg. The system can produce a cooling power of about 4.5 kW with COP of 0.25. Zhang [4] designed and tested an automobile adsorption chiller using zeolite-water pair driven by the waste heat of a diesel engine. A specific cooling capacity of 25.7 W kg⁻¹ at COP of 0.38 was obtained.

At present, mathematical modeling is the most commonly used technique to predict the performance of an adsorption cooling system. There are several mathematical modeling techniques including the lumped-parameter simulation, distributed-parameter simulation and dynamic simulation [26]. However, the modern developments are focusing on dynamic models which give a more clear idea about the dynamic behavior of the transient heat and mass transfer processes occurring in the beds [27–29]. However, further studies and improvement on these systems are needed in order to enhance and optimize their performance, especially at the strongly dynamic conditions of automotive application with a huge variability of the ambient conditions, and waste heat.

In this part, Part II of a two-part study, performance, and optimization of a silica gel-water on-board adsorption system are investigated based on the validated dynamic model proposed in the first part of this paper, Part I [30]. The overall model was used to simulate the performance of the entire system (engine, adsorption system, heating and cooling circuits, chilled water circuit and cabin) under real driving conditions of a car. The model is able also to estimate the waste heat available at the engine cooling loop. calculate the cooling capacity and monitor the temperature and humidity of the air at the cabin, as a function of the vehicle velocity. ambient temperature and sun radiation. The system performance. as well as the cabin temperature, was analyzed under real driving conditions of start-up strategies and different ambient conditions. In addition, some parametric studies had been carried out to evaluate the system performance under different design (layout) configuration and operation strategies in order to explore further improvements of the system performance.

2. Description of the on-board adsorption A/C system

Fig. 1(a) shows a picture of the on-board adsorption A/C system in the laboratory and Fig. 1(b) shows the system fitted in a passenger vehicle. Fiat Grande Punto vehicle with a 1.9 JTD engine was selected as a reference car. The choice for this diesel engine is justified as it represents a worst case situation, because this engine is very efficient and therefore produces relatively low amount of waste heat. The system was designed and tested under the framework of Topmacs project for automobile air conditioning

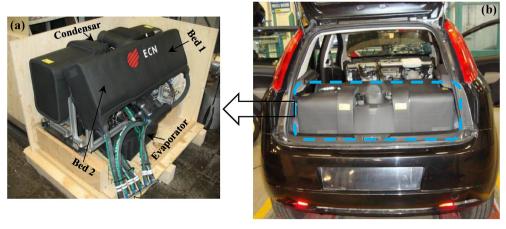


Fig. 1. Picture of the proposed on-board adsorption A/C chiller: (a) in the laboratory and (b) installed in the vehicle.

applications [31]. The description, components, and working principle of the adsorption chiller has been described in detail in the first part of this paper [30].

A schematic diagram of the overall adsorption A/C system implemented in the car is shown in Fig. 2(a), while the adsorption system layout with the main vehicle components is shown in Fig. 2(b).

The system operation is cyclic. The heat from engine waste heat is supplied to one of the beds to produce the desorption of the water vapor, at the same time the cooling water is provided to the other bed by the auxiliary cooler (radiator) using a hydraulic loop, which dissipates the heat to the environment. When Bed 2 is heated, the adsorbent starts to desorb the water vapor inside the

bed. The pressure increases in the bed due to the water vapor and the valve between the adsorbent bed and the condenser opens (the communication valve with the evaporator is kept closed). The water vapor condenses in the condenser, releasing its latent heat of condensation to the cooling water circuit. Liquid water passes from the condenser to the evaporator through an expansion valve. At the same time, the other bed (Bed 1) is being cooled down and starts to adsorb the water vapor. The pressure decreases in the bed, valves are all closed. When the pressure becomes less than the evaporator pressure, the valve between the bed and the evaporator opens, and the bed begins to adsorb the water vapor coming from the evaporator, producing the evaporation of the water in it. During the evaporation stage, the water absorbs heat from the cabin or space

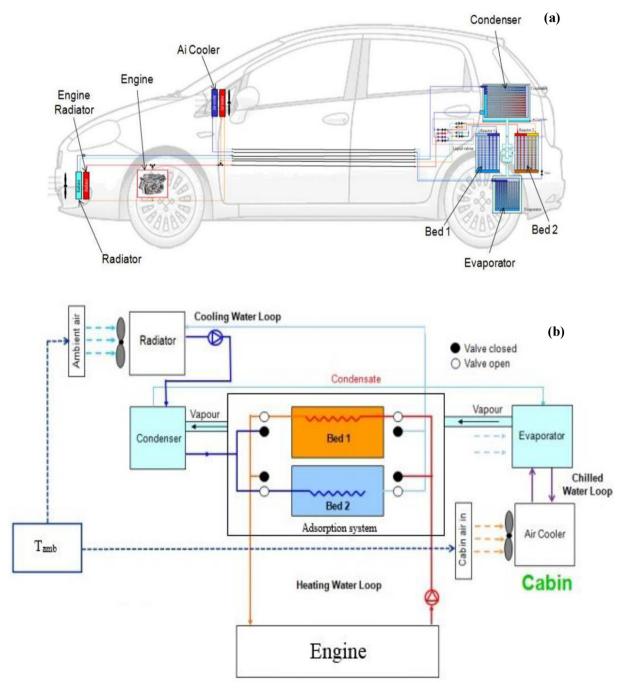


Fig. 2. a) Scheme of the on-board adsorption air conditioning system implemented in the car and b) layout of the on-board adsorption A/C system.

to be conditioned, producing a cooling effect which flows through the cabin cooler (Air Cooler). Then, the water from the auxiliary cooler element is conducted to Bed 2, which is dry and hot, cooling it down, decreasing its pressure and activating the adsorption from the evaporator. On the other hand, Bed 1, which is full and cooled, is heated up to raise its pressure and activating the desorption process.

The secondary water flow rate available to activate the system is always assumed as the same that normally goes through the Engine radiator to cool down the engine. The engine needs to heat up as fast as possible in order to minimize the emissions and increase the efficiency. During the warming up period, there is no waste heat available to the system. When the engine water temperature attains an adequate level, the thermostat opens the cooling circuit to the engine radiator. The inlet hot water is then sent to activate the adsorption cooling system. This means that during the engine warm up no desorption can be carried out until the engine is sufficiently warm. Nonetheless, cooling the cabin is still possible provided that one or two beds have been kept dry so that it can absorb vapor from the evaporator and produce a cooling effect from the beginning of a driving cycle.

The engine coolant system includes a thermostat which controls the engine temperature in order to first allow for the fastest warming up of the engine, and then dissipate the excess heat on the engine radiator, therefore keeping constant the engine operating temperature. The engine radiator will be employed to dissipate the excess heat in the case that the adsorption system is not able to employ all the available waste heat in order not to overheat the engine. Additionally, a way to monitor the water temperature at the outlet of the adsorption system, and measures to ensure that it returns to the engine at an appropriate temperature have been implemented in the overall model. This corresponds to the constraints of engine design, which dictate a maximum acceptable temperature drop through the adsorption system in order not to decrease the engine operation temperature. This would negatively affect the engine efficiency as well as the emissions.

2.1. Testing facility

In order to provide experimental data that will be used to feed the dynamic model of the overall system, several tests were performed on the on-board A/C system implemented in Fiat Grande Punto vehicle at the Centro Ricerche Fiat (CRF), Italy laboratories. Experimental tests were performed in a climatic chamber equipped with a rolling bench as shown schematically in Fig. 3 to simulate the driving cycle under different start-up and ambient conditions. By performing these tests it was possible to define the amount of waste heat in the engine cooling loop that can be recovered and transformed into cooling capacity by the adsorption system as well as to adjust the model of the cabin.

The assessment test follows the same routine as the usual A/C characterization tests performed for cars:

- The vehicle is run on a roll test bench following a programmed standard driving cycle.
- The air is blown at the front of the vehicle in order to simulate ambient air around the vehicle on the road.
- Sun radiation is simulated with artificial lamps.
- Ambient temperature is kept at the desired condition.
- The A/C system is in operation following a standard procedure.
 - The temperature of the air provided by the A/C system and temperatures at several positions of the cabin air are recorded during the tests. These measurements allow characterizing the thermal performance of the cabin and the A/C system.

2.2. Testing cycle

Fig. 4 shows the Normal European Driving Cycle (NEDC cycle) used as the testing cycle through the present study. In Europe, passenger vehicles are normally qualified in terms of fuel consumption using this particular type of cycle. The employed driving cycle is basically composed of two series of four repetitions of elementary urban cycles at low speed (ECE) separated by a higher speed extra-urban cycle (EUDC), as it is shown in the figure.

Under real driving conditions, the available waste heat from the engine varies greatly. In addition, at the start-up (warming-up), no waste heat is available to power the adsorption system until the engine has been warmed up, so the water is sent back to the engine. The engine operation requirements make the temperatures and mass flow rates to the adsorption system quite variable and the conditions at the cabin are not constant either, see Fig. 4. The initial temperature is quite high and it drops as the cooling system starts

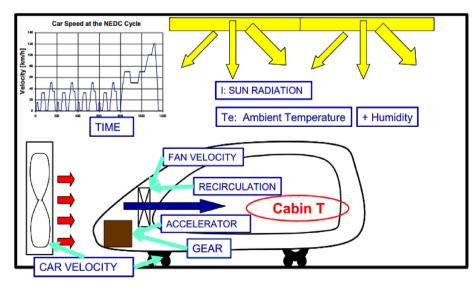


Fig. 3. Scheme of a climatic chamber with rolling bench suitable to perform the A/C assessment tests.

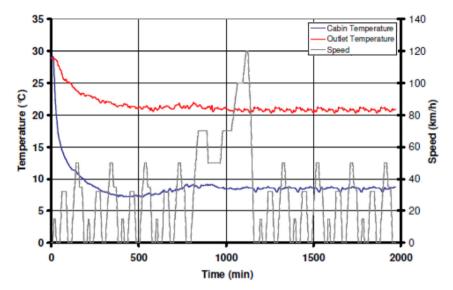


Fig. 4. NEDC-2ECE cycle.

to operate. This leads to a significant variation in the temperature and humidity at the cabin space until the comfort conditions are reached. The variations then transmitted from the cabin to the evaporator and then to the beds.

2.3. Start-up and ambient conditions

During the work, different start-up and ambient conditions have been tested in order to assess and optimize the performance of the proposed adsorption chiller. Table 1 shows the start-up conditions, while Table 2 presents the different ambient conditions which have been considered for the experimental and simulation tests.

Table 1 Start-up conditions.

3. Mathematical modeling and system performance

The non-equilibrium lumped parameter model and its validation procedures for the Sorbil A/water adsorption chiller prototype have been described in detail in the first part of this article [27]. The proposed model is used here to simulate the dynamic performance of the entire system under different real driving conditions as shown in Table 1 and Table 2. The radiator and air cooler were modeled in a similar way as the condenser and evaporator presented in the first part of the paper [30]. Where the heat exchangers were characterized by their overall thermal conductance (UA). A detailed analysis has been carried out and adequate values of UA for

Start-up condition	Assumptions	Remarks
Saturated beds	both beds are saturated	 All the valves of the system remain opened. The system is in equilibrium with the water stored at ambient temperature. The pressure is saturated and the uptake is at the equilibrium at that pressure and temperature. This strategy is the worst case scenario because it does not allow the system to produce cooling effect at the driving cycle start.
One-dry bed	one of the beds is dry	 All the valves of the system are closed and isolated from the rest of the system. One bed remains with the maximum uptake and fully loaded with water, while the other adsorbent bed remains with the minimum uptake and dry, ready to adsorb water vapor when the valve connecting the evaporator and the bed opens. This could only be possible if the valves communicating the beds with the condenser and evaporator are able to practically seal the beds when closed.
Two-dry beds	both beds are dry	 All the valves of the system remain closed, therefore it is capable to maintain both beds completely dry until the following driving cycle. 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. This is possible if the valves that connect the beds with the condenser and evaporator are able to practically seal the beds when closed.

Table 2Thermal testing (ambient) conditions.

Test	Ambient conditions		Driving cycle	A/C way of use	Remarks	
	Temp. [°C]	R.H. [%]		.,		
Test No. 1: Equivalent European summer conditions	28	50	NEDC cycle	Auto, with 20 °C internal set point	Classify the A/C systems with regards to fuel consumption and thermal comfort.	
Test No. 2: Severe summer conditions	35	60	NEDC cycle	Auto, with 23 °C internal set point	Considered for non-European countries, where the thermal load is normally higher.	
Test No. 3: Cool-down test	43	35	NEDC cycle	Auto, with 25 °C internal set point	Severe European climate conditions during the day. Cabin soaking temperature: 63 °C.	

the radiator and air cooler, depending on water and air mass flow rates, have been estimated from experimental results provided by the (CRF).

The main structure of the overall system model is shown in Fig. 5. Each model exchanges information through flows of different fluids and control parameters. Models are additionally composed of sub-models, representing the different subsystems included each.

The different component models with the input data from Topmacs project [31] have been combined in the MATLAB Simulink environment and have been integrated into the overall model. The input data to the overall model are the same as those for a conventional A/C system during the assessment tests. Data consists of ambient temperature and humidity, air conditioning fan velocity, instantaneous power, engine speed, and gear and accelerator position.

The set operating conditions of heat transfer fluid temperatures and flow rates are listed in Table 3. The air flow rate values passing through the radiator and cabin cooler are taken from the experimental data given by CRF.

4. System performance

The transient model is able to calculate the amounts of heat exchanged in each component, the performance of the system, and consequently the air temperature inside the cabin. The cooling capacity of the system for the evaporator:

$$\dot{Q}_{chill} = \dot{m}_{w,chill} C p_w \int\limits_{0}^{t_{cycle}} (T_{chill,i} - T_{chill,o}) dt \tag{1}$$

The condenser capacity:

$$\dot{Q}_{cond} = -\dot{m}_{w,sec}Cp_{w}\int\limits_{0}^{t_{cycle}}(T_{sec,i} - T_{sec,o})dt \tag{2}$$

The heating capacity of the bed in desorption mode:

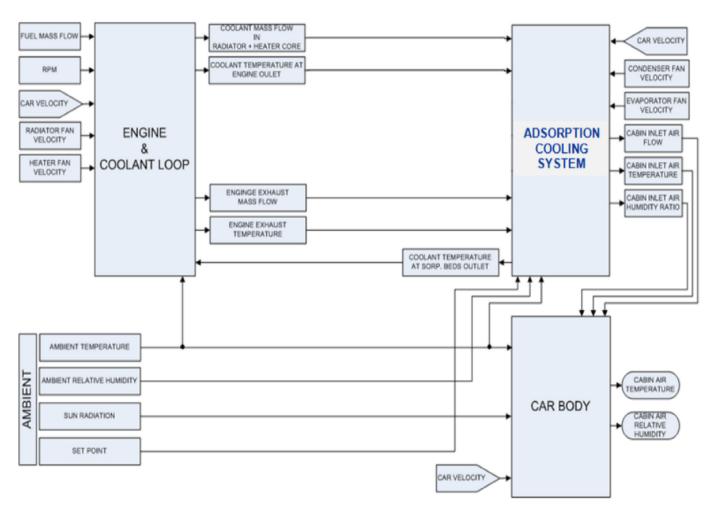


Fig. 5. Overall model structure.

Table 3The nominal operating condition for the on-board adsorption system.

Hot water inlet		Cooling water inlet	(cond. + adsorber)	Chilled water inlet		
Temp. [°C]	Flow rate [kgs ⁻¹]	Temp. [°C]	Flow rate [kgs ⁻¹]	Temp. [°C]	Flow rate [kgs ⁻¹]	
90	0.20	33	0.20	15	0.13	

$$\dot{Q}_{heat} = \dot{m}_{hw,b} C p_w \int\limits_0^{t_{cycle}} (T_{hw,i} - T_{hw,o}) dt \tag{3}$$

The cooling capacity of the bed in adsorption mode:

$$\dot{Q}_{cool} = -\dot{m}_{cw,b}Cp_{w}\int_{0}^{t_{cycle}} (T_{cw,i} - T_{cw,o})dt$$
 (4)

The COP of the system is defined as the ratio between the heat extracted by the evaporator and the heat source input to the bed.

$$COP = \frac{\dot{Q}_{chill}}{\dot{Q}_{heat}} = \frac{(\dot{m}Cp)_{chill} \int_{0}^{t_{cycle}} (T_{chill,in} - T_{chill,o}) dt}{(\dot{m}Cp)_{des} \int_{0}^{t_{cycle}} (T_{hw,in} - T_{hw,o}) dt}$$
(5)

The performance of the adsorption chiller is mainly characterized by the cooling capacity of the system (\dot{Q}_{chill}) and by the coefficient of performance (COP).

5. Results and discussion

5.1. System optimization at constant operating conditions

Optimizing the design and operating parameters of an adsorption system which has variable operating conditions becomes a considerably difficult task. Fig. 6(a,b) show the simulation results of the effect of adsorption/desorption cycle time variation on the average cooling capacity and COP at different ambient temperatures and constant engine coolant water value at 90 °C. The ambient temperatures values are assumed to correspond to that considered by CRF for the assessment tests shown in Table 2. The average cooling capacity and COP are calculated by the model as the average for the last cycle, once the system is operating in steady-state conditions and the initialization period does not have an effect on the performance anymore.

As it can be seen from Fig. 6(a), the highest values of the mean cooling capacity are obtained between 600 and 800 s cycle times. When cycle times are lower than 400 s the cooling effect decreases suddenly, and this is due to the fact that there is not enough time for adsorption or desorption to occur satisfactorily. On the other

hand, when cycle times are longer than 800 s, the cooling capacity decreases gradually since adsorption tends to saturate as the adsorbent approaches to its equilibrium condition regardless of the cycle time employed. On the other hand, the COP values increase uniformly as the cycle time increases, this is because the final part of the adsorption/desorption process increases the cooling produced but requires a longer time, and therefore the cooling capacity decreases.

In addition, it can be observed that the cooling capacity increases with decreasing ambient temperature. When the outdoor air temperatures are high (such as 43 °C), the cooling down of the bed is taking place at a higher temperature. This has a negative effect on the cooling capacity since a less amount of water vapor is adsorbed in the adsorbent bed. It can be said that the refrigeration effect will be improved when the system is operating at lower ambient temperatures since the cooling process of the beds will be more efficient, allowing faster cycling times. In contrast, the COP decreases with decreasing ambient temperatures. This occurs because the system requires more engine waste energy to heat the adsorbent bed when working at lower ambient temperatures since it will be more difficult to heat the bed after it has been cooled down to lower cooling temperatures. However, the impact of the cycle time on the COP is more significant than the ambient temperatures. When the ambient temperature drops from 43 °C to 28 °C, the cooling capacity increases from 1.29 kW to 1.46 kW and the COP decreases from 0.39 to 0.34. On the other hand, the optimum cycle time maximizing the capacity increases from 656 to 812 s when the environmental temperature increases from 28 to 43 °C as shown in Table 4.

The results obtained provide a guideline for the design and optimization of a control system aimed to provide the maximum cooling output at different operating conditions, it is apparent that an optimum value around 700 s for the cycle time (i.e. an ads/desorption time of 350 s) could make the system achieve a good performance for the whole range of ambient temperatures studied (from $28\,^{\circ}\text{C}$ to $43\,^{\circ}\text{C}$).

The results obtained provide a guideline for the design and optimization of a control system aimed to provide the maximum cooling output at different operating conditions. In addition, it is apparent that an optimum cycle time value of about 700 s (i.e. an adsorption/desorption time of 350 s) could make the system achieve a good performance for the whole range of studied ambient temperatures (from 28 °C to 43 °C).

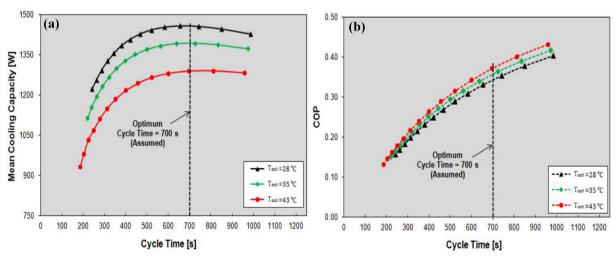


Fig. 6. Effect of adsorption/desorption cycle time on the system (a) Cooling capacity and (b) COP.

Table 4Results of the cycle time optimization study.

T _{amb.} [°C]	Optimum cycle time to maximize cooling capacity [s]	Optimum Ad/desorption time [s]	Maximum cooling capacity [kW]	СОР
28	656	328	1.46	0.34
35	726	363	1.39	0.36
43	812	406	1.29	0.39

5.2. System performance evaluation under different real driving conditions

Different start-up and ambient conditions such, as shown Tables 1 and 2in, have been simulated under real driving conditions in order to assess the performance of the entire adsorption A/C system. During all tests it was assumed that the A/C system is working in recirculation mode. This means that the inlet air temperature to the cabin cooler is at the cabin temperature, which will be decreasing with the time. This is the usual mode of operation for the A/C system in summer conditions.

The control strategy of the beds implemented in the overall model was based on a fixed adsorption/desorption cycle time of 350 s corresponding to the optimum value obtained from the parametric studies presented in Sub-section (5.1). This means that the adsorption/desorption cycle will be reversed when it reaches a set value which is 350 s. However, the simulation results showed a poor performance when a fixed cycle time was implemented in the model. This happened because the bed could not be heated up efficiently once the bed temperature at the end of the cycle was not

close to 90 °C, and as a consequence, the system could not adsorb enough vapor. This fact negatively affected the system performance. Therefore, a different bed control strategy was implemented which was not only based on the cycle time but also on the maximum temperature of the bed. This strategy is based on a double condition: the cycle will be reversed when the bed in heating mode reaches at least 85 °C, and the cycle time reaches at least its optimum value (350 s). The results obtained showed a good operation of the system so this double condition was implemented in the overall model for all assessment tests.

5.2.1. Test No. 1: equivalent European summer conditions

Fig. 7(a—d) show the simulation results obtained at the equivalent European summer conditions (Test No. 1) and start-up conditions for both beds being completely dry (Two-dry beds).

Fig. 7(a) shows the variation in temperature and pressure profiles in the adsorbent beds with the time (0:2000 s). As discussed before, the available waste heat from the engine under real driving conditions is very variable, especially during the engine warming up period. Approximately, in the first 660 s after engine start-up, the

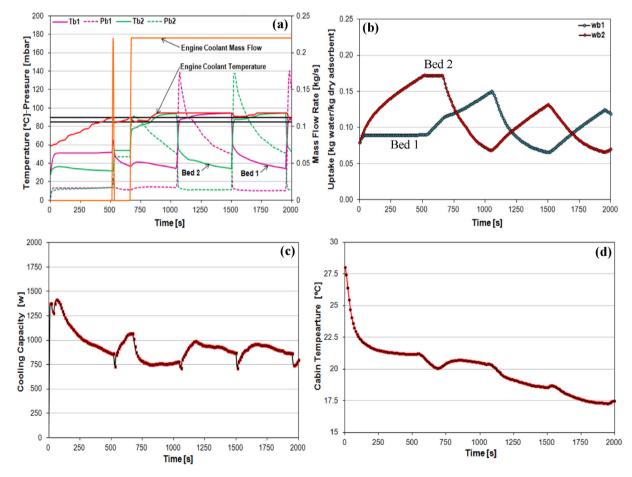


Fig. 7. Equivalent European summer conditions with **Two-dry beds**: (a) Temperature and pressure profiles of bed 1 and bed 2, engine coolant temperature and mass flow rate, (b) water content (uptake) in two beds, (c) cooling capacity, and (d) cabin temperature.

outlet coolant temperature of the engine is below 90 °C, so it is not possible to use the engine coolant water as a heat source for the adsorption system. At 550 s there is a first activation peak and the engine coolant temperature reach 90 °C for the first time. However, the bed is very cold and absorbs a lot of heat, producing a sudden drop in the engine coolant temperature. Consequently, the thermostat which controls the engine coolant flow to be used by the system is switched off, causing a deactivation of the adsorption system. After 660 s, the adsorption system is activated again and is able to use the waste energy from the engine coolant at a temperature between 85 and 90 °C, until the time reaches 850 s. After the 850 s, the engine outlet coolant temperature goes above 90 °C, ensuring high engine efficiency. The adsorption system is then able to use the engine coolant as a higher temperature heat source, which lies between 90 and 95 °C. After the car operates in full constant driving conditions, the engine was able to provide the amount of hot water at almost constant flow rate along the rest of the driving cycle and the adsorption-desorption cycles become steady.

The water content of the adsorbent beds is shown Fig. 7(b). During the first half cycle time (360 s), bed 1 being heated up, the adsorbent in bed 1 was in desorption mode and the water content decreases to about 0.07 kg/kg of Silica-gel. At the same time, bed 2 was in the cooling mode, the adsorbent in bed 2 was in the adsorption mode and the water content of the adsorbent was increased (adsorbing water vapor from the evaporator) to about 0.14 kg/kg of Silica-gel. The adsorbents exchange their roles in the next 360 s in the third cycle. When bed 2 almost reaches equilibrium conditions, at 520 s after the engine start-up, the cycle was reversed, bed 1 then starts to adsorb since it was dry and empty of water, and bed 2 starts to desorb as it is fully loaded with water. For the last stable cycle (from 1500 s to 1950 s) the uptake variation ranges from 6.5% to 13%.

Simulation results of the produced cooling capacity with time is shown in Fig. 7(c). It is clear that the cooling capacity changes with time and the system are able to produce a cooling effect from the beginning. In this case, the heat from the engine was not needed to start producing cooling since the beds were assumed to be initially dry. In this way, once one of them was cooled down and connected to the evaporator, it starts to adsorb and produced a cooling effect immediately. This start-up strategy was the best scenario since it allowed the production of cooling since the beginning of the driving cycle without the necessity of the engine coolant temperature to reach 90 °C. At steady state conditions, the adsorption

system was able to produce an average cooling capacity of about 925 W.

Fig. 7(d) shows the change in cabin temperature with the time. As shown in the figure, the proposed system was able to cool down the cabin very fast, achieving a quite low temperature in the cabin. The cabin temperatures decrease during each cooling cycle and slightly increases during the reversing period, this is due to the heat transferred from the cabin space to the inner air. However, the system was able to decrease the cabin temperature from 28 °C to 17.5 °C during 2000 s, depending on the test. In the real case, the cabin thermostat would switch off the system once the comfort temperature (20 °C) has been reached.

Fig. 8(a—b) show the simulation results for the tests at the equivalent European summer conditions (Test No. 1) and different start-up conditions: one-dry bed, Two-dry beds, and saturated beds.

As shown in Fig. 8(a) at the beginning, the system with two beds saturated takes a long time to start producing a cooling effect (around 1100 s), while with one bed dry it was possible to have cooling production since the beginning of the driving cycle with about 500 W. In the case of both beds initially dry, the system was able to produce a higher cooling effect, and therefore a faster decrease of the cabin temperature. Nevertheless, the difference between having one or two beds dry is not so important. This indicates that in order to have a good performance it would only be necessary that, one of the beds is dried and sealed after the engine is switched off. The system with one dry bed was able to produce a mean cooling capacity of 700 W, while with two dry beds was able to produce a mean cooling capacity of 925 W, and with both beds saturated was able to produce 300 W. In one-dry bed and two-dry beds start-up conditions the system was able to reach a cabin temperature lower than 20 °C which is the comfort temperature, Fig. 8(b). Keeping both beds dry results in more efficiency for the system performance, and also the comfort conditions are reached quicker.

5.2.2. Test No. 3: cool-down test

Fig. 9(a–d) show the results obtained for the cool-down test (Test No. 3) with the better start-up condition case (Two-dry beds) with the time. The cool-down test was the most severe test of all studied tests, the ambient temperature was 43 $^{\circ}$ C and 35% RH. They have highly severe conditions for the thermal compressor, where the initial cabin temperature was higher due to soaking (46 $^{\circ}$ C). Therefore, the performance of the adsorption system drastically deteriorates.

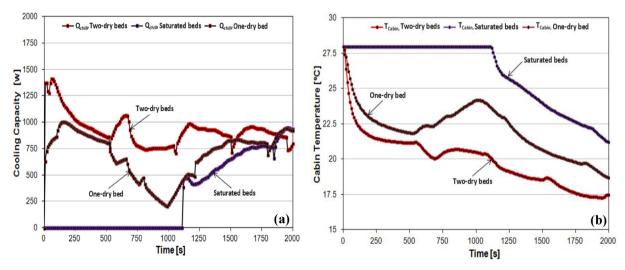


Fig. 8. Equivalent European summer conditions (Test No. 1) and different start-up strategies: (a) Cooling capacity and (b) Cabin temperature.

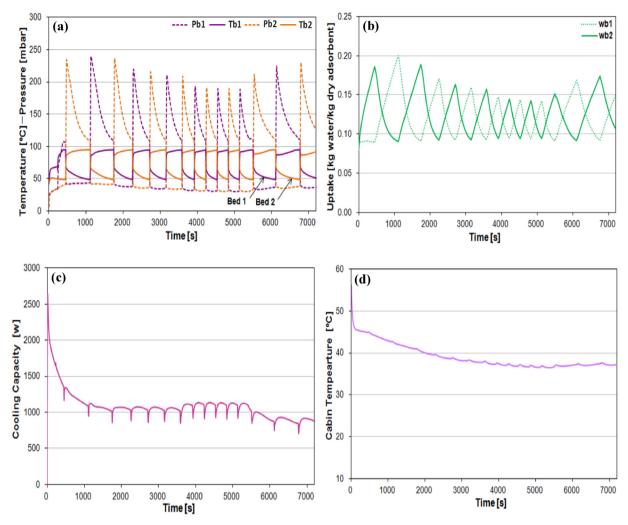


Fig. 9. Cool-down (Test No. 3) with Two-dry beds start-up conditions: (a) Temperature and pressure profiles of bed 1 and bed 2, (b) water content in two beds, (c) cooling capacity, and (d) cabin temperature.

As can be seen from Fig. 9(a), due to the characteristics of the cool-down test, the adsorption-desorption cycles were more stable in comparison with the results obtained previously for Test No. 1. At the cool-down test, the car is in highway driving mode, which guarantees that a more constant engine coolant mass flow rate feed the adsorption system. Furthermore, the cycle time was shortened since the system performs at a higher cooling temperature so that it tends to follow quite fast adsorption-desorption cycles, and the range of uptake variation becomes very small as shown in Fig. 9(b). As a consequence, the number of cycles per driving cycle and their frequency increase, which leads to a slight increase of the cooling capacity. Thus, the system was able to produce a mean cooling capacity of approximately 1.1 kW, Fig. 9(c).

Nevertheless, due to the high severe conditions, the cooling effect produced by the system was not enough to carry the cabin temperature to comfort conditions (25 $^{\circ}$ C). The cabin temperature was about 37 $^{\circ}$ C as shown in Fig. 9(d). However, increasing the cycle time and modifying the switching criterion can improve the operation.

5.3. System performance comparison at different test conditions

Table 5 summarizes the system performance obtained at different operating conditions for the best start-up procedure (Two-dry beds). The table shows the different ambient conditions (T_{amb}) with comfort temperatures (T_{comf}) , cabin temperatures

 (T_{cab}) , cooling capacity (\dot{Q}_{chill}) and corresponding COP.

As can be seen, for the Equivalent European Summer Conditions (Test No. 1), the adsorption cooling system was able to deliver a refrigeration capacity of 925 W with a COP of 0.40. The obtained final cabin temperature was 17.5 °C, which means that comfort conditions (20 °C) can be perfectly achieved. For Test No. 2 (Severe Summer Conditions), a mean cooling capacity of 930 W with COP of 0.42 was achieved. Under these conditions, the system was able to produce a reasonable cooling effect but not enough to carry the cabin temperature to comfort conditions (target value 23 °C) and was only able to keep the cabin at around 24.4 °C. For the most severe of all assessment tests (Test No. 3), the system was able to produce a mean cooling capacity of 1100 W with a COP of 0.44. The cooling effect produced by the system was not enough to carry the cabin temperature (37 °C) to comfort conditions (target value 25 °C). It can be observed that under severe conditions the COP of the system tends to increase. This is simply due to the fact that the system performs at a higher cabin temperature, which leads to an increase of the evaporation temperature.

5.4. Design modification of the on-board system layout using two radiators

In order to further increase the system performance, a new system layout with two independent cooling loops (two radiators)

Table 5System performance at different operating conditions and **Two-dry beds** start-up condition.

Test	Ambient conditions		T _{comf} [°C]	T _{cab} [°C]	Qchill [W]	COP
	T _{amb} [°C]	R.H [%]				
Test No. 1: Equivalent European Summer Conditions	28	50	20	17.5	925	0.40
Test No. 2: Severe Summer Conditions	35	60	23	24.4	930	0.42
Test No. 3: Cool-down	43	35	25	37	1100	0.44

has been investigated as an alternative to the reference system shown Fig. 2(b). The only difference between this system layout and the previously analyzed is the number of radiators. Instead of using one radiator to cool in sequence the condenser and then the adsorbent bed, the new proposed system comprises of two radiators working in independent cooling loops. This implies that one radiator is used to cool down the bed and another is used to cool down the condenser as shown in Fig. 10.

5.4.1. Performance evaluation of the alternative system layout

Fig. 11(a, b) present the simulation results of the proposed alternative layout system compared with the results of the reference system. The comparison is based on a constant engine coolant water temperature at 90 °C and Equivalent European summer conditions ($T_{am}=28$ °C and 50% R.H).

The simulation results showed that the alternative layout system has logically a better system performance. The maximum cooling capacity increased by 12.3% (from 1.46 to 1.64 kW) and the optimum cycle time decreased by 28.4% (from 656 to 511 s) compared to the reference system layout. The cooling of the beds for the alternative layout system takes place at a lower

temperature, allowing faster cycling times and better performance. In contrast, in the original system layout, the cooling water enters the bed at higher temperatures since it has previously absorbed heat from the condenser, consequently, less amount of water vapor will be adsorbed by the bed. This negatively affects the performance of the system.

Interestingly, COP is slightly decreased by 9.1% with the alternative layout. This occurs because the system requires more engine waste energy to heat the sorption bed when working at lower cooling temperatures. This happens as it will be more difficult to heat the bed after it has been cooled down to lower cooling temperatures in the cycle time which maximizes the cooling capacity.

Fig. 12(a, b) shows the variation of the bed inlet $(T_{hw,i})$ and outlet heating $(T_{hw,o})$ water temperatures and inlet $(T_{cw,i})$ and outlet $(T_{cw,o})$ cooling water temperatures with cycle time for the original and alternative system layouts. From the results obtained, when using the proposed alternative layout system, the inlet cooling water temperature slightly decreases from 37.6 to 36 °C (at the optimum cycle time). The cooling water entering the bed in the original layout absorbs first the heat from the condenser, coming back to the bed at a higher temperature. In addition, the difference

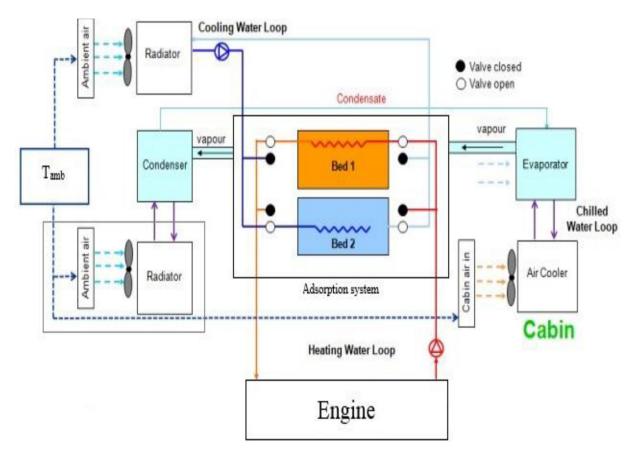


Fig. 10. Scheme layout of the proposed alternative system configuration of the onboard adsorption A/C system.

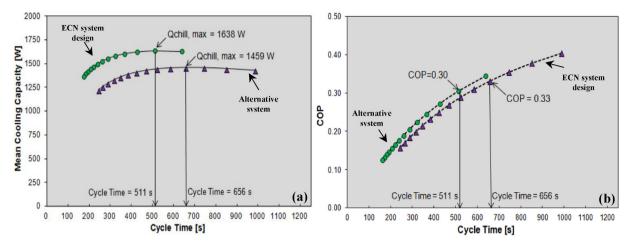


Fig. 11. Effect of cycle time on the system performance for different system layouts: (a) Cooling Capacity and (b) COP.

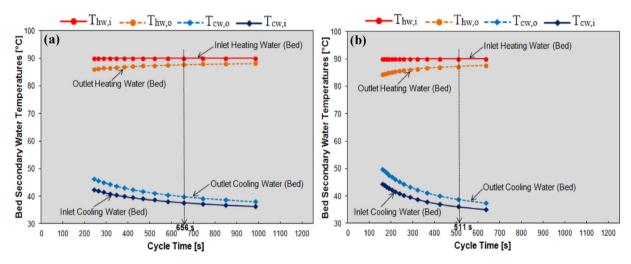


Fig. 12. Variation of the bed heating and cooling water temperatures with cycle time for different system layouts: (a) Original system design and (b) Alternative system design.

between the inlet and outlet cooling water temperature was larger, which means that the bed was being cooled more efficiently and it takes a shorter time to cool down. Consequently, the cycle time was shorter in comparison with the original layout.

Fig. 13(a, b) show the variation of the condenser, evaporator,

chiller and secondary fluids temperatures with the cycle time for both the original and the alternative layouts.

The simulation results showed that with the alternative layout system (two independent cooling loops) the inlet condenser cooling water temperature decreases from 36.86 to 30.59 °C and as a

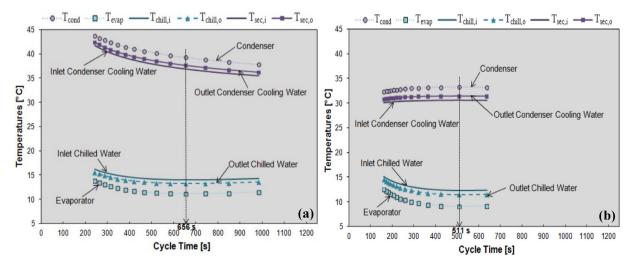


Fig. 13. Variation of the evaporator and condenser secondary water temperatures with cycle time for different system layouts: (a) ECN system design (b) Alternative system layout.

consequence, the condenser temperature decreases from 39.24 to 33.26 °C compared with the ECN system layout. The condenser was obviously able to reject more heat, in turn improving the system performance. In the original layout, the cooling water entering the condenser first absorbed the heat from the bed coming back to the condenser at a higher temperature. Moreover, the temperature of the chilled water entering the evaporator decreased from 14 to 12.3 °C and the evaporator temperature decreased from 11 to 9 °C when having two independent cooling loops instead of one. When having a single cooling water loop, the refrigerant came from the condenser to the evaporator at a higher temperature, which results in an increase of the evaporator temperature. Also, it absorbed less amount of vapor from the evaporator since it was more difficult to cool down the bed. Consequently, the evaporator pressure increased. This negatively affected the overall performance of the system.

5.5. Effect of the alternative layout system on the system performance

The possibility of employing two independent cooling loops with two radiators to enhance the performance of the adsorption A/C system had been considered as an option at the beginning of the TOPMACS project [31]. However, the decision was made not to proceed with this option, since it would increase the complexity of the on-board system in terms of size and cost. As the overall model had been built in such a way that it is permits testing different layouts and therefore it was decided to study this possibility as well.

Fig. 14(a, b) shows a comparison between the ECN system and the alternative system layout in terms of cooling capacity and the produced cabin temperatures. The tests performed at the Equivalent European summer conditions (Test No. 1) for the most favorable start-up condition (two-dry beds).

The results showed that the system with two radiators has a higher performance than that of one radiator. Using the alternative layout system instead to the ECN system layout, the average cooling capacity increased by 7.0% (from 925 to 990 W) and the cabin temperature decreased by 9.1% (from 17.5 to 16 $^{\circ}\text{C}$) but the COP decreased by 25% (from 0.40 to 0.30). This happens because, in the case of using the same radiator to cool the condenser and then the bed in series. Consequently, it becomes more difficult to cool down the beds with water coming from the condenser at a higher

temperature. Therefore, a less amount of water vapor was adsorbed, resulting in a lower refrigeration effect. On the contrary, when using two radiators to cool down the bed and another is used to cool down the condenser, the COP decreases. This happens due to the fact that the system using two radiators performs at a lower evaporation temperature due to a decrease of the cabin temperature. The COP is very sensitive to the evaporation temperature. However, it is clear that using two independent radiators is more efficient, leading to better performance in terms of cooling capacity and cabin temperature. However, the difference is not high, and whether this option is cost effective or not would depend on the increase of the cost to include the second radiator.

6. Conclusions

In Part II of the two-part study, the validated model of the first part (I) was used to predict the transient performance and cabin temperature of the adsorption A/C system when implemented in a car under real driving and different ambient conditions. The system performance is also evaluated and optimized under different operating and design conditions such as; i) cycle time, ii) different real driving conditions (start-up and ambient conditions), and iii) provide a new system layout with two independent cooling loops (two radiators) as an alternative to the reference system with one radiator. A prototype adsorption air conditioning was developed and test derived by waste heat from the engine coolant loop in the laboratory of ECN.

Results showed that after the car operated in full constant driving conditions, the engine was able to provide the amount of hot water at almost constant flow rate along the rest of the driving cycle and the adsorption-desorption cycles become steady. The proposed system is able to deliver an average cooling capacity of about 925 W with COP of 0.40 and a cabin temperature of 17.5 °C at Equivalent European summer conditions. An alternative system layout with two independent cooling loops (two radiators) was proposed in order to increase the system performance. The obtained results showed that the proposed system layout had a higher cooling capacity but of course it demanded an extra radiator and the system complexity increased. In regard to the start of the system, it was clear that in order to have cooling effect from the beginning of the driving, at least one bed should be dried and sealed in a way that it is would be able to adsorb vapor from the very beginning of the cycle. Otherwise, taking into account the

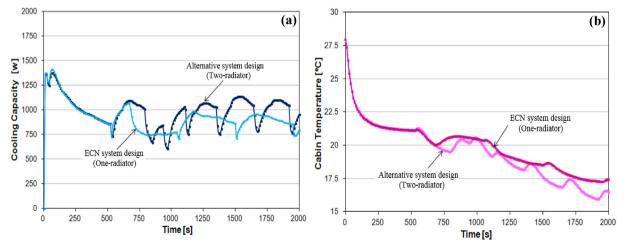


Fig. 14. Performance of the system at Equivalent European summer conditions (Test No. 1) and two-dry beds start-up strategies for different system layouts: (a) Cooling capacity and (b) Cabin temperature.

necessary time for the engine warming up and the variability of the waste energy after that moment, the starting of the cooling would take a considerable time. Keeping both beds dry would, of course, allow a higher cooling capacity. However this would require a longer drying period at the end of the driving cycle, another sealing valve, and a more complex control system when the extra cooling provided was not so much. From the obtained results, one bed dried proved to be best design compromise. At moderate thermal loads, the designed system was able to provide enough cooling capacity and was able to decrease the cabin temperature below the comfort temperature in a reasonable time. Under high thermal load, as for instance, under soak conditions, the system seems to be still undersized although it was still able to significantly decrease the cabin temperature from its very high initial value.

However, the amount of waste heat available in the engine coolant loop was sufficient to drive the adsorption system and to produce enough cold to keep comfortable temperatures in the cabin. The model was able to run under the extreme variable conditions of a real driving cycle and capture the complex dynamics of the whole system.

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Nomenclature

COP: coefficient of performance C_p : specific heat, $J k g^{-1} k^{-1}$ m: mass flow rate, $k g s^{-1}$ T: temperature, k

Subscripts

Am: ambient
B: bed
Chill: chilled water
Cond: condenser
Des: desorber
Evap: evaporator
I: inlet
Out: outlet
Sec: secondary
W: water
W: uptake, kgkg⁻¹

