CONTINUOUS MULTIPLE-BED REGENERATIVE ADSORPTION CYCLE REFRIGERATOR / HEAT PUMP

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Abstract

A refrigeration / heat-pump system based on a number of simple tubular adsorption modules is described. A single module is comprised of a generator and a receiver/condenser/evaporator. A single generator consisting of a 12.7 mm stainless steel tube lined with 3 mm of monolithic active carbon has been manufactured. A complete module has been tested in a simple rig, which subjects it to alternating hot and cold airstreams, desorbing and adsorbing ammonia. A complete system, consisting of 32 modules has been modelled in detail and its predicted performance is presented. Key parameters have been varied and their effect on the performance discussed.

KEYWORDS

Simulation, adsorption, carbon, ammonia, heat-pump, refrigeration, regenerative

INTRODUCTION

It is well known that the major challenges for adsorption cycles are:

- To improve heat transfer in adsorbate beds, thereby reducing size and cost.
- To improve regenerative heat transfer between beds, thereby increasing COP.

Current progress is reviewed by Meunier [1]. Bed conductivities have been improved from 0.1 W m^{-1} K^{-1} [2,3] to 0.4 W m^{-1} K^{-1} for monolithic carbons [4] and 5 W m^{-1} K^{-1} for graphite composites [5]. Cooling and heating COP and the SCP of a range of sorption machines are given in [6]. It is not possible to generalise about them, since many different applications were studied, but it is probably true to say that in air conditioning applications adsorption machines need to be improved if they are to compete with multiple-effect Lithium Bromide absorption systems.The ideas presented below aim at both improving the efficiency and reducing the cost of adsorption refrigeration and heat pumping.
They are the subject of a patent [7].

CONCEPT

In order to achieve good regeneration between adsorption and desorption it is necessary to either employ some form of thermal wave [8,9] or many beds [10]. This study examines the possibility of using many simple modular beds in an arrangement that allows effective heat transfer between them. A single module is shown schematically in Figure 1. It is a tube having a sorption generator at one end and a combined evaporator and condenser at the other. The first such module that we have made has a stainless steel tube 12.7 mm in diameter and 500 mm long containing the generator. It is lined with a 3 mm layer of monolithic carbon provided by Sutcliffe Carbons Ltd. The carbon shape is formed within the tube and so the heat transfer between steel and carbon is very high. The other end of the module is a receiver for the liquid adsorbate; ammonia in our work. The optional wire gauze acts as a wick to keep the liquid in contact with the wall. The adiabatic section separates the generator and receiver, reducing longitudinal conduction between them. In the current design with 0.25 mm wall thickness tube, a 20mm 'adiabatic' length reduces mean longitudinal conduction to less than 1W. This could be further reduced if necessary.

Heat may be transferred to or from the generator and receiver by passing air or any other fluid across them. Fins can enhance this if necessary. In desorption, the generator is heated and desorbed refrigerant condenses in the receiver, which is cooled. In adsorption the generator is cooled and heat is provided to boil the refrigerant in the receiver. The whole module is the very simplest type of adsorption refrigerator or heat pump.

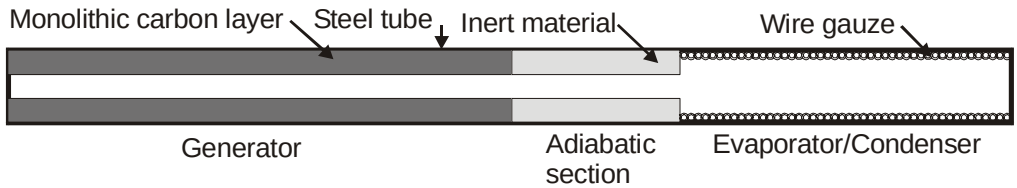

Fig. 1 Sorption Module

It is possible to combine many such modules in an assembly that allows regenerative heating between them. Figures 2 and 3 show one possible way of doing this in schematic form. In this example the machine is an air-to-air heat pump or air conditioner. 16 modules are shown, arranged in a cylindrical shell. All rotate about the central axis (X in Figure 2) typically completing one revolution in ten minutes. Air is blown over the tubes, counterflow to their direction of motion and exchanging heat with them. Seals prevent the air from travelling directly between the adsorbing and desorbing zones but allow the tubes to pass through when necessary. Consider the path of a single tube beginning at position 1 in Figure 3. The carbon is at its coldest, perhaps 50ºC and has maximum concentration. As it moves clockwise through the annular duct it is heated by air flowing in the opposite direction. If the 'thermal mass flow rate' of the generator is approximately the same as that of the air, then the desorption section acts like a counterflow heat exchanger with capacity ratio close to unity. In reality, the effective specific heat of the carbon varies during a cycle but it is still possible to balance the flow rates quite well. The result is that by the time the module reaches the end of the desorption section it is perhaps at 200ºC. Whilst the carbon is heated it desorbs ammonia which condenses in the receiver section of the module. In an air conditioning application, a stream of ambient temperature air cools the receiver section. A similar process occurs in the adsorbing section, but with evaporation occurring in the receiver, which cools the airstream passing over it. In the course of cooling down, the generator tubes heat the stream of ambient air induced at position 2. This pre-heated air is removed from the annular duct at position 3 and heated by an external high grade heat source before being re-introduced to the duct at position 4.
The greater the number of modules, the better the approximation to a continuous process with counterflow regeneration of heat.

The advantages of the system are the absence of refrigerant valves and complex or expensive heat exchangers. Potential problems might include the sealing mechanism to direct the air flows and possible degradation of the carbon due to thermal shock.

INITIAL EXPERIMENTATION

Previous experience with monolithic carbon [11] suggested that monolithic carbon should be proof against the thermal cycling and could be made to adhere strongly to the tube wall. A manufacturing technique was devised and a 500 mm length of 12.7 mm outside diameter 0.9 mm wall thickness stainless steel tube was lined with 3 mm of active carbon, as shown in Figure 4. A simple test rig was built which subjected the generator to alternating flows of hot (150ºC) and ambient air whist the receiver was kept in a flow of ambient air. An optional glass receiver allowed observation of the quantity of liquid ammonia building up or boiling. Repeated heating and cooling for tens of cycles showed no tendency for the carbon to break up and drop into the receiver. This was encouraging, but obviously in a real system there will be many thousands of cycles and further testing is necessary.

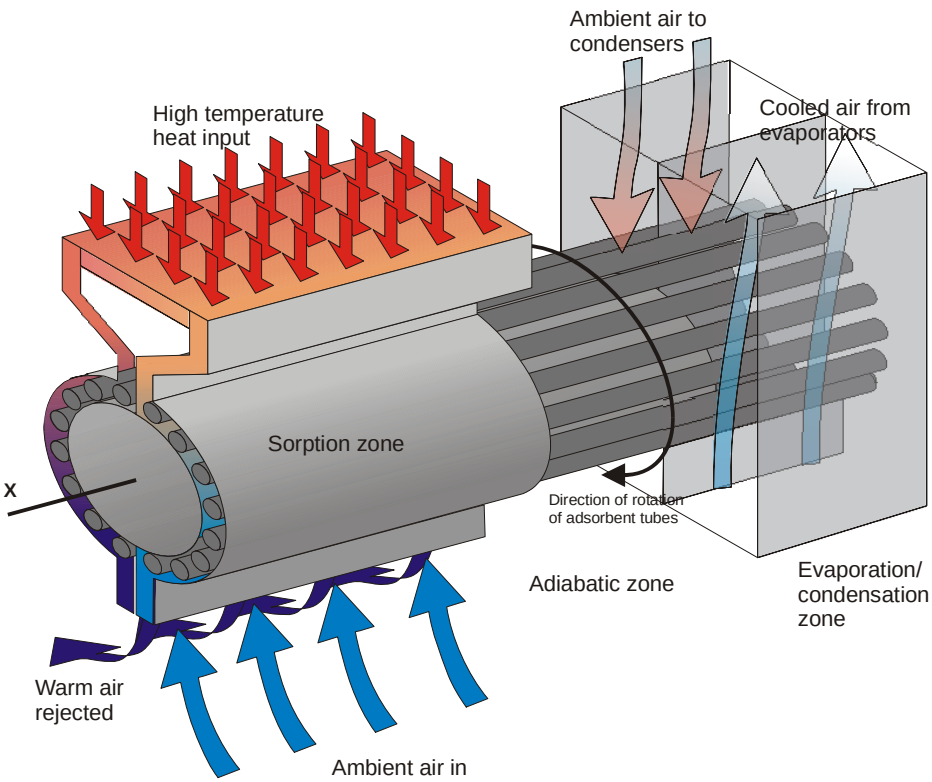

Fig. 2 Schematic air to air heat pump / chiller

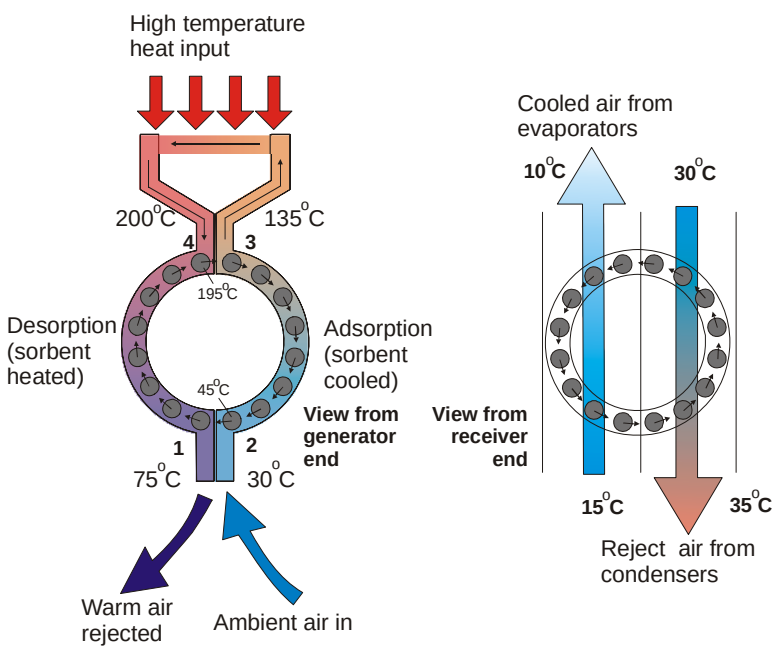

Fig. 3 Section through the generators.

SIMULATION MODEL

A simulation model has been written in Matlab™ which is descibed fully in [12]. The example presented is one that gives reasonable performance but is not optimised in many respects. There are many design parameters and many different functions (COP, SCP, economics etc.) that may be optimised and so the concept of a single optimum design is not necessarily useful.
The key design parameters and operating conditions chosen are in Table 1.

Fig. 4 Generator Section

Table 1 Design and operating parameters of base case example

Parameter	Value
Number of modules in desorption zone	16
Number of modules in adsorption zone	16
Number of receivers adiabatic during desorption	3
Number of receivers adiabatic during adsorption	3
Generator length	1 m
Generator diameter	12.7 mm
Generator finning	Aluminium, 3 mm pitch, 25.4 mm square.
Generator duct air velocity	1.0 m s^{-1}
Air heater outlet temperature	200°C
Generator air inlet temperature	30°C
Condenser air inlet temperature	30°C
Evaporator air inlet temperature	15°C
Time for one complete revolution or cycle	864 s

After four revolutions the COP in cooling is 0.60 and the COP in heating is 1.53 rather than 1 + Cooling COP The disparity is due to the fact that conditions at the start of the final revolution are not precisely those at the end. The major performance indicators are given in Table 2.

Table 2 Base case performance indicators

Performance Indicator	Value
COP cooling	0.60
COP heating	1.52
Cooling power	291 W
SCP	142 W kg^{-1}
Mean temperature of air entering heater	170 °C
Mean temperature of air leaving desorption zone	51 °C
Mean condensing temperature	38 °C
Mean evaporating temperature	8.9 °C

The behaviour of the system is shown in Figure 5, which shows the mean carbon temperature of 4 of the 32 modules (Nos. 1,9,17,25) with time over four complete cycles of 864 s. It shows how the starting transients develop and then damp towards regular cyclic behaviour. The transient form is a result of the constant heater outlet temperature – a constant power heater would produce a different transient.
Figures 6 shows the heat input or output of an individual module during the final revolution of 864 s. It is close to 1000 W kg^{-1} for most of the cycle.

KEY PARAMETERS AND THEIR EFFECT ON PERFORMANCE

There are over one hundred parameters input to the simulation but the effect of five major ones are illustrated here: Thermal capacity ratio, number of modules, temperature of air leaving the heater, generator heat transfer coefficient and the evaporator inlet temperature. The results are shown in Table 3.

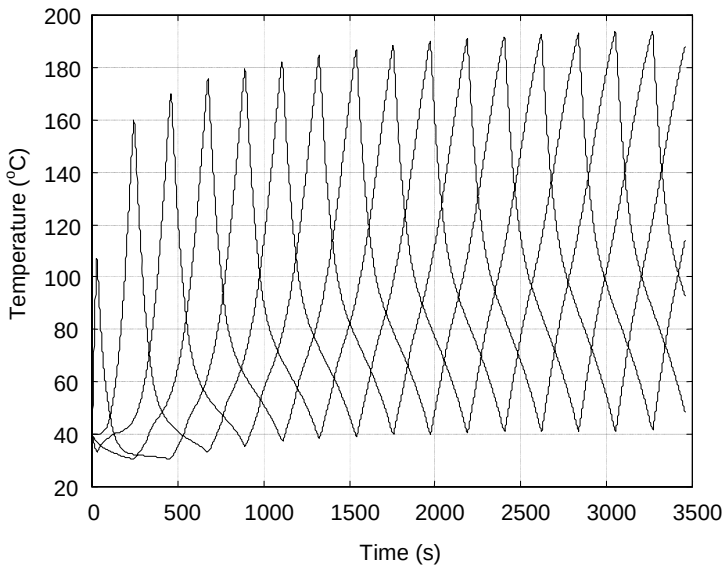

Fig. 5 Module temperatures v. time

The capacity ratio effect is illustrated by varying the generator air duct velocity. In the base case it is 1.0 m s^{-1} and alternative cases at 1.1 and 0.9 m s^{-1} have been simulated. The reduced COP of both cases is shown in Table 3. Similar effects are observed if the rotational speed of the modules is varied.

The variation in COP and SCP with the heater exit temperature is shown in Figure 7. It is noticeable that the COP is comparatively insensitive to heater temperature in the range chosen, although it does have a peak at around 175ºC. At higher temperatures, although a little more ammonia can be desorbed, not all of the sensible heat load can be regenerated and COP drops. However, the cooling power rises with increasing heater temperature in a linear fashion. The minimum mean concentration for a heater temperature of 250ºC is 3% whereas for 150ºC it is 10%. The variation of performance with heat transfer coefficient over the generator fins is evident from Table 3. At high values of $h_{generator}$ the effectiveness of a single tube fin is so close to unity that further gains are minor. In reality $h_{generator}$ is a function of the air speed but is treated as an independent variable here in order to separate the effects on heat transfer and thermal capacity.

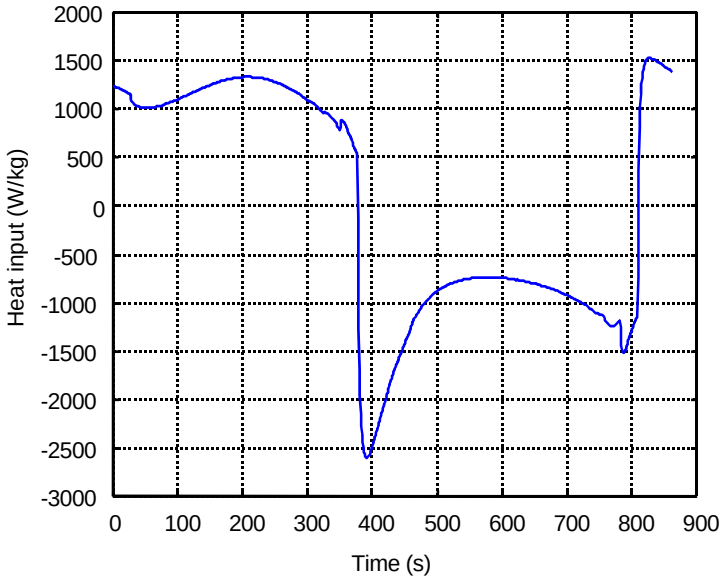

Fig. 6 Specific heat input v. time

The evaporating and condensing temperatures are determined by the air inlet temperatures to the separate evaporators and condensers. The variation is illustrated in Table 3;

COP varies linearly from 0.52 to 0.69 as the inlet temperature to the evaporator increases from 10ºC to 20ºC.

Parameter	Base Case Value	New Value	COP (cooling)	SCP (W kg^{-1})	$T_{ads\ out}$ (ºC)	$T_{des\ out}$ (ºC)	$T_{ev\ out}$ (ºC)	$T_{con\ out}$ (ºC)	$T_{ev\ mean}$ (ºC)	$T_{con\ mean}$ (ºC)
Air speed (m s^{-1})	1.0	1.1	0.56	176	164	59	9.7	36.4	8.0	38.5
		0.9	0.57	110	173	48	10.8	36.1	9.2	38.1
Heater temp. (ºC)	200	250	0.54	175	209	60	9.8	37.3	8.0	39.4
		225	0.58	160	190	55	9.8	36.6	8.1	38.3
		175	0.61	121	150	48	10.7	35.8	9.3	37.6
		150	0.60	101	129	45	11.5	34.7	10.2	36.2
$h_{generator}$	120	60	0.54	139	167	55	10.5	36.0	9.0	37.8
		180	0.61	142	171	50	10.4	36.3	8.8	38.1
Evaporator air in (ºC)	15	20	0.69	159	171	50	15.2	36.9	13.7	38.8
		10	0.52	127	169	52	5.6	35.7	4.1	37.3
Number of modules	32	24	0.44	158	166	56	9.5	36.9	7.4	39.0
		40	0.71	125	173	48	11.1	35.7	9.9	37.2
		50	0.82	107	174	45	11.8	35.0	10.8	36.4
		64	0.91	89	176	43	12.3	34.1	12.3	34.2
Base Case			0.60	142	170	51	10.4	36.2	8.9	38.1

Table 3 Parametric study results

The variation in COP and SCP with the number of tube modules is non-linear over the range chosen. In the limit of one tube there is no regeneration and for an infinite number of tubes there will be maximum regeneration. Assuming the 'mass flow rate' of modules to be constant the SCP will decrease with an increasing number of tubes. Figure 8 shows the variation for 24 to 64 tubes. The improvement in COP at high module numbers is due mainly to the improved regeneration, but also to the reduced temperature lift resulting from the reduced evaporator and condenser load.

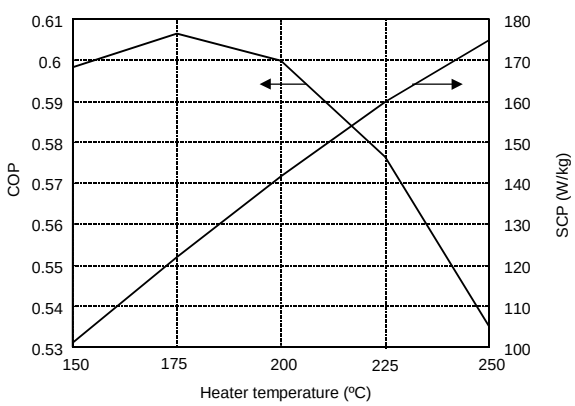

Fig. 7 Performance v. Heater Temperature

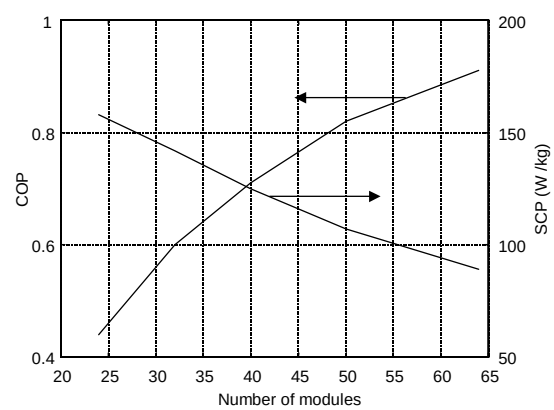

Fig. 8 Performance v. Number of modules

CONCLUSIONS AND FUTURE WORK

A new continuous adsorption refrigeration system has been described which uses a number of simple tubular modules. A single module has been made and tested and a programme written to simulate many modules in a real system. The design has not yet been optimised but a parametric study has revealed the way in which key parameters affect COP and SCP. It is hoped to build a small prototype system within a year. Future work will also attempt a Second Law analysis of the system in order to discover the ideal limitations of the concept.

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Nomenclature

COP-Coefficient of performance (-); $h_{generator}$-Heat transfer coefficient of generator tube/fin (W $m^{-2}K^{-1}$);SCP Specific Cooling Power (W kg^{-1} adsorbent); T Temperature (K);$T_{ads\ out}$ Mean temperature of air leaving adsorption section over complete cycle (ºC); $T_{con\ mean}$ Mean condensing temperature over complete cycle (ºC); $T_{con\ out}$ Mean temperature of air leaving condensing section over complete cycle (ºC);$T_{des\ out}$ Mean temperature of air leaving desorption section over complete cycle (ºC); $T_{ev\ mean}$ Mean evaporating temperature over complete cycle (ºC); $T_{ev\ out}$ Mean temperature of air leaving evaporating section over complete cycle (ºC); T_{sat} Saturation temperature (K)

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