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THE USE OF THERMOSYPHON HEAT PIPES TO IMPROVE THE PERFORMANCE OF A CARBON-AMMONIA ADSORPTION REFRIGERATOR

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Previous work at Warwick has established that heat transfer within an active carbon adsorbent bed can be improved dramatically by the use of a monolithic carbon—aluminium laminate. Experimental results are presented which illustrate the excellent effective conductivity. However, the system tested did not achieve its full potential due to heat transfer problems external to the bed. Heating is achieved by condensing steam on the generators at up to 140°C and cooling by boiling water down to 40°C. This should have achieved rapid heating and cooling but two problems were encountered. Firstly, the complete system was physically complex with many valves, joints etc. and it proved difficult to prevent air leaking in to the sub-atmospheric section where it degraded the cooling performance. Secondly, the thermal mass of the shell and flanges surrounding the generators (needed to contain steam at 10 bar) presented an unacceptably high thermal load that had to be taken through the full temperature swing of each cycle. The external heating and cooling system is being completely redesigned to eliminate these problems. Rather than using the same working fluid to both heat and cool the generators, two separate thermosyphon heat pipe fluids are used in heating and cooling. This reduces the number of switching valves from ten vacuum-sealed ball valves costing £200 each to four solenoid valves costing £5 each. The massive flanges are no longer needed, since the thermosyphon heat pipes are simple tubes. Water will be used in the heating thermosyphons and pentane in the cooling thermosyphons.

INTRODUCTION

Adsorption cycles can be used in heat-driven refrigerators, air conditioners or heat pumps in which the energy source is a burning fuel or waste heat. Examples include the use of waste heat to provide cooling, heat pumping, industrial refrigeration, vehicle air conditioning and the refrigeration of food in parts of the developing world where there is no electricity supply to power conventional vapour compression

Regarding the developing country application, which is one of our major interests, there is a well-documented need for food refrigeration in areas that do not have access to grid electricity. Spoilage of many products, particularly fish, can be as high as 50%. "It is recognized that much of the fish caught is wasted or degraded through poor post-harvest handling; this, plus the importance of maintaining fish stocks, has led to various international agreements, including the Kyoto 'Declaration and Plan of Action on the Sustainable Contribution of Fisheries to Food Security.'"[1] There is a need for systems that will operate without grid electricity, that can be built and maintained in the country of use, and that are low enough in cost to be affordable by local farmers or fishermen. Our suggested solution is a thermal icemaker.

Basic Adsorption Cycle

Adsorption refrigeration and heat pump cycles rely on the adsorption of a refrigerant gas into an adsorbent at low pressure and subsequent desorption by heating. In its simplest form an adsorption refrigerator consists of two linked vessels, one of which contains adsorbent and both of which contain refrigerant as shown in Figure 1. Our preferred adsorbent and refrigerant are active carbon and ammonia. Ammonia is a low cost, environmentally friendly refrigerant. Although toxic if inhaled, if it is released to the environment it is quickly absorbed and neutralised without harm. It has zero ozone depletion potential and zero greenhouse warming effect and is still the refrigerant of choice in many industrial applications.

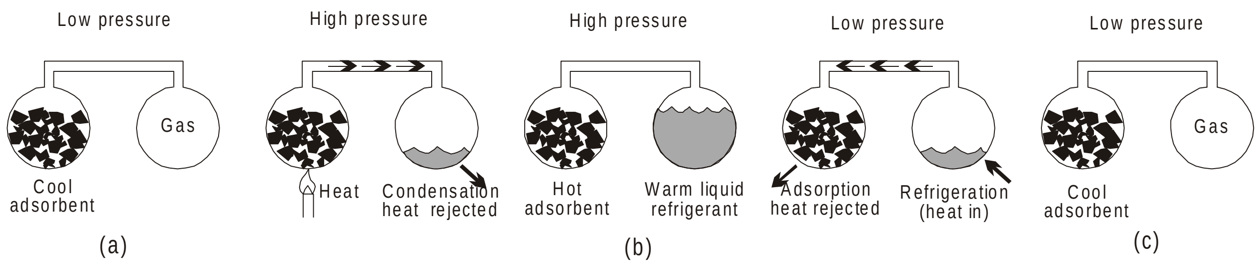

Figure 1. Basic adsorption cycle.

Two such simple systems can be operated out of phase to provide continuous cooling. Such an arrangement has a comparatively low Coefficient of Performance (Refrigeration COP = Cooling / Heat Input and Heat pump COP = Heat Output / Heat Input). Also, since the thermal conductivity of the bed is generally poor, the time taken for a cycle could be an hour or more and the cooling power per mass of adsorbent could be less than 100 W/kg. Greatly improved heat transfer is required to reduce the cycle time to a few minutes and thereby increase the power density of the adsorbent to the order of 1 kW/kg. It can also improve the COP by maximising the quantity of heat regenerated. The heat rejected by one bed when adsorbing can provide a large part of the heat required for desorption in other bed.

The use of monolithic carbon - aluminium laminates has been investigated at Warwick. Monolithic activated carbons can be manufactured in almost any size or shape as a solid block, freeing designs from the constraint of using granular beds of carbon. Our industrial partner, Sutcliffe Carbons Ltd., developed the material. We used it in a form that integrates the carbon bed with aluminium fins that enhance the heat transfer. The carbon shapes and aluminium fins are formed together within the generator shell, ensuring excellent heat transfer.

Properties of Monolithic Carbon

The thermal conductivity of two monolithic carbons, designated LM127 and LM 128 have been measured and are presented Table I. The accuracy is about 13%. The conductivity of monolithic carbon LM127 is nearly four times higher than with a granular carbon bed (0.16 $Wm^{-1}K^{-1}$) because of the absence of large voids and the existence of an unbroken conduction path. The heat transfer coefficient between the carbon and a close fitting plate is also measured.[2]

Table I
Thermal Conductivity of Monolithic Carbons

Sample	Conductivity ($Wm^{-1}K^{-1}$)	Heat transfer coefficient ($Wm^{-2}K^{-1}$)
LM127	0.6194 - 0.0008*T	350
LM128	0.3885 - 0.0003*T	800

T: Temperature (^oC)

The axial and radial permeabilities of samples were measured by using a specially designed test rig. Since the samples tested are porous media with very low gas velocities, the Ergun model[3, 4] is applicable:

$$-\frac{dP}{dz} = \frac{\mu}{K_a} v_a + B_a \rho v_a^2 \quad \text{(axial flow)} \tag{1}$$

$$-\frac{dP}{dr} = \frac{\mu}{K_r} v_r + B_r \rho v_r^2 \quad \text{(radial flow)} \tag{2}$$

Table II
Permeability K and Shape Factor B

Sample	Gas	T_{am}(°C)	T_g(°C)	μ x 10^{-5} (Pa.s)	K x 10^{-14} (m^2)	B x 10^8 (m^{-1})
Axial						
LM127	Air	16.3	16.4	1.7838	**3.6065**	**6.6152**
	Argon	18.1	18.0	2.2088	**3.6486**	**5.2962**
	-	-	-	-	3.63 (*)	5.95 (*)
LM128	Air	19.4	19.0	1.7971	**0.5729**	**58.095**
	Argon	19.1	19.1	2.2150	**0.5658**	**85.815**
	-	-	-	-	0.57 (*)	71.96 (*)
Radial (Converging)						
LM127	Air	18.6	18.6	1.7949	**34.9080**	**0.4863**
	Argon	15.9	15.9	2.1975	**36.4660**	**0.5004**
	-	-	-	-	35.69 (*)	0.49 (*)
LM128	Air	17.2	17.2	1.7882	**1.2584**	**6.5552**
	Argon	18.5	18.4	2.2110	**1.2901**	**8.1518**
	-	-	-	-	1.27 (*)	7.35 (*)
Radial (Diverging)						
LM127	Air	18.9	18.9	1.7967	**34.4550**	**0.4421**
	Argon	16.1	16.2	2.1990	**36.3290**	**0.4357**
	-	-	-	-	35.39 (*)	0.44 (*)
LM128	Air	18.7	18.7	1.7953	**1.3065**	**5.5126**
	Argon	18.8	18.6	2.2124	**1.3671**	**5.3247**
	-	-	-	-	1.34 (*)	5.42 (*)

T_{am}: Ambient temperature (°C) ; T_g: Gas temperature (°C) ; (*) : Combined value = Mean value

The carbon LM127 (coarse power) has an axial permeability (3.6 x 10^{-14} m^2) that is about six times higher than the carbon LM128 (fine powder) (0.6 x 10^{-14} m^2) as shown in Table II. However the radial permeability of the sample LM127 (36 x 10^{-14} m^2) is about twenty-five times higher than the sample LM128 (1.3 x 10^{-14} m^2). For the same sample, the radial permeability is about three (for LM128) to ten (for LM127) times higher than the axial permeability. Regarding the radial tests, there is no significant difference between the converging and diverging permeability measurements (less than 5 % - order of magnitude of measurement errors). The anisotropic nature of the permeability is assumed to be caused by the manufacturing process.

The generator made with carbon LM127 offers the best performance. The radial pressure drop is about 50 mbar (central hole diameter 5 mm, outer diameter 50 mm) when cooling at 1 kW/kg carbon, compared with LM128 generator which has a pressure drop of about 1500 mbar with the same hole diameter.

Ammonia Porosity

The adsorption refrigeration system under development uses ammonia as a refrigerant and so a detailed knowledge of the porosity characteristics of the carbons with ammonia is required. The ammonia concentration of the carbon samples was investigated by using the porosimeter developed at Warwick by Critoph.[5] The variation of the concentration x using a modified Dubinin-Radushkevich equation[5] is:

$$x = x_o \exp\left[-K\left(\frac{T}{T_{sat}} - 1\right)^n\right] \tag{3}$$

The values of x_o, K and n are calculated from experimental data by minimizing the sum of the squares of the differences in concentration predicted by Dubinin-Radushkevich equation (Equation 3) and those measured. The results for the two sample of carbon (LM127 and LM128) are given in Table III. The two samples present nearly similar Dubinin coefficients because of the similarity of both base precursors. However, the concentration with monolithic carbon is about 30 % higher than with ordinary granular carbon from the same precursors (208C): the maximum concentration is about 0.36 kg/kg with monolithic carbon (LM127) and is about 0.29 kg/kg granular carbon.[5]

Table III
Dubinin Coefficients (x_o, K and n) with Carbon-ammonia Pair

Sample	x_o	K	n	SEE
LM127	0.3629	3.6571	0.9400	0.0019
LM128	0.3333	3.6962	0.9900	0.0028

SEE: *Standard Estimated Error.*

Effective Specific Heat

The specific heat of the carbon without adsorbate is given in Table IV. The measurements were carried out using a scanning differential calorimeter by Dr. G. Restuccia at CNR-TAE [6]. On the basis of the concentration test and the similarity of the specific heat of each sample (LM127 and LM128), as we expect, the effective specific heat of the two samples is very close.

Table IV
Specific Heat as a Function of Temperature

Sample	Specific heat (J $kg^{-1}K^{-1}$)
LM127	802.51+2.811*T
LM128	775.62+2.826*T

T: Temperature (^oC)

ORIGINAL SYSTEM DESIGN

The form of generator used is displayed in Figure 2, which shows the dimensions finally used. The generator (made with 1m stainless steel tube) is heated and cooled externally by condensing or evaporating water. The interior is a laminate of carbon discs and aluminum fins formed in situ. The full generator is 1 m long and contains about 455 carbon-aluminum layers. Its total weigh is about 4.1 kg. A sectional photograph is shown in Figure 3.

A laboratory prototype system built capable is designed to produce up to 3 kW of cooling power with a heating power input of 10 kW. A schematic of the whole system is shown in Figure 4. There are two generators (Gen. 1 and Gen. 2). Heating and cooling of both is by a common thermosyphon heat pipe using water as the working fluid. The prototype has a large number of individual valves controlled through a computer with Workbench software. Steam from the boiler is diverted to one generator whilst the other is cooled by low pressure boiling water that rejects heat in the condenser. The layout was chosen for maximum flexibility in the laboratory.

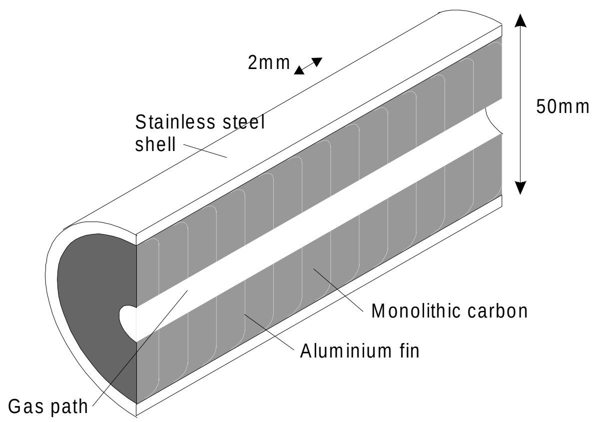

Figure 2. Generator geometry.

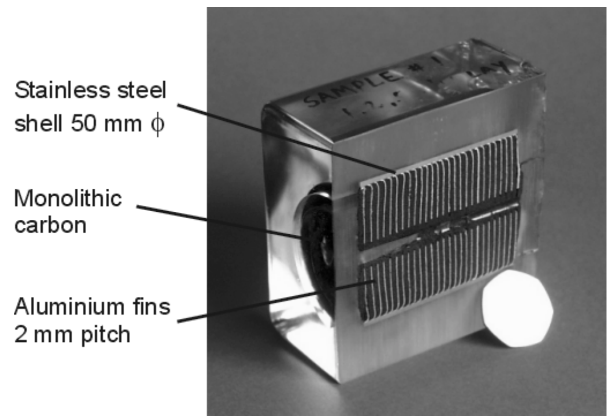

Figure 3. Generator section (total generator length is 1m).

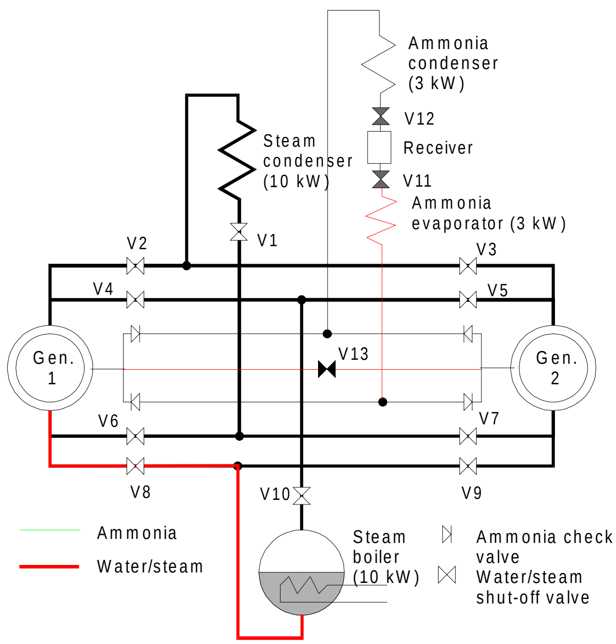

Figure 4. Laboratory prototype of adsorption refrigerator.

The experimental tests were carried out at various working conditions ($-10^oC<T_E<15^oC$ - evaporation; $30^oC<T_C<45^oC$ - condensation; $100^oC<T_G<150^oC$ - generation and $200s<\Delta t<1000s$ - half cycle time). The refrigerator performance (cooling power P_c, specific cooling power P_{sc} and COP) at the evaporating temperature limits ($T_E = -10^oC$ and $T_E=15^oC$) with cycle time Δt = 920s are show in Table V. The performance is relatively low mainly because of the high cycle time. Further tests have proved that the cooling performance increases whilst the cycle time decreases as shown in Table VI. At the limiting cycle time due to the full generator thermal mass, the refrigerator could provide a specific cooling power up 0.218 kW/kg carbon with a COP of about 0.50.

Table V
Refrigerator Performance at Evaporating Temperature Limits

T_E (^oC)	P_c (kW)	P_{sc} (kW/kg)	COP
-10	0.110	0.040	0.23
15	0.380	0.136	0.52

($T_C = 31^oC$, $T_G = 125^oC$ and $\Delta t = 920s$)

Table VI
Refrigerator Performance at Half Cycle Time Limits

Δt (s)	P_c (kW)	P_{sc} (kW/kg)	COP
310	0.570	0.203	0.50
920	0.310	0.110	0.45

($T_E = 10^oC$, $T_C = 31^oC$ and $T_G = 125^oC$)

The full description of the laboratory prototype of the adsorption refrigerator built and the analysis of experimental results of the performance of the machine at various working conditions are well documented in Tamainot-Telto and Critoph[6] and Critoph.[7]

ACHIEVEMENTS AND REMAINING PROBLEMS

The use of monolithic carbons would appear to be a good solution but there are unsolved technical and design problems. The advantages are that manufacture is possible in the country of use (although the carbon material will probably be imported) and that the system requires no complex moving parts and little or no maintenance. Internal heat transfer problems have been eliminated, but efficient heat transfer to and from the shell is a limitation. The system was made to work in the laboratory, but had some disadvantages, particularly when considering the requirements of simplicity and reliability.

The problems were:

1. A large number of high cost valves were needed to divert the steam or water flows to or from the appropriate generator.
2. The use of sub-atmospheric pressure water in such a complex system made the elimination of inward air leaks almost impossible over a long period of time. Any inward leak of air leads to degradation and final failure of the cooling loop as air builds up in the condenser.
3. Each generator was surrounded by a flanged shell, capable of containing steam at 10 bar. Whilst the thermal mass was taken into account in the design, it would still be desirable to reduce the mass, thereby increasing both the power and COP.
4. When switching from heating to cooling, care must be taken to ensure that there is an adequate supply of liquid remaining in the generator. This required a degree of precision in control that was achievable in the laboratory but problematic in a commercial low-cost product.

These considerations have lead us to search for new solutions suited to applications in developing countries It is important to note that energy efficiency, whilst desirable, is not the most important criterion. If making ice for food preservation in developing countries, a likely heat source is burning agricultural waste that will raise steam to drive the refrigerator. In many cases the cost of the energy source will be quite small compared to the benefits of the cooling and capital cost and reliability are the most important factors. For these reasons we do not consider the use of more efficient but complex regenerative cycles to be appropriate in these applications.

PROPOSED TECHNICAL SOLUTION

We propose keeping the advantages of boiling and condensing heat transfer but removing the complexity by using separate heating and cooling circuits, as shown in Figure 5. We also need a new low cost and effective means of switching the heat inputs and outputs. The use of switchable thermosyphon heat pipes will remove the need for complex or expensive valves. To our knowledge, the only other group to try to use this approach is Vasiliev's team in Minsk.[8]

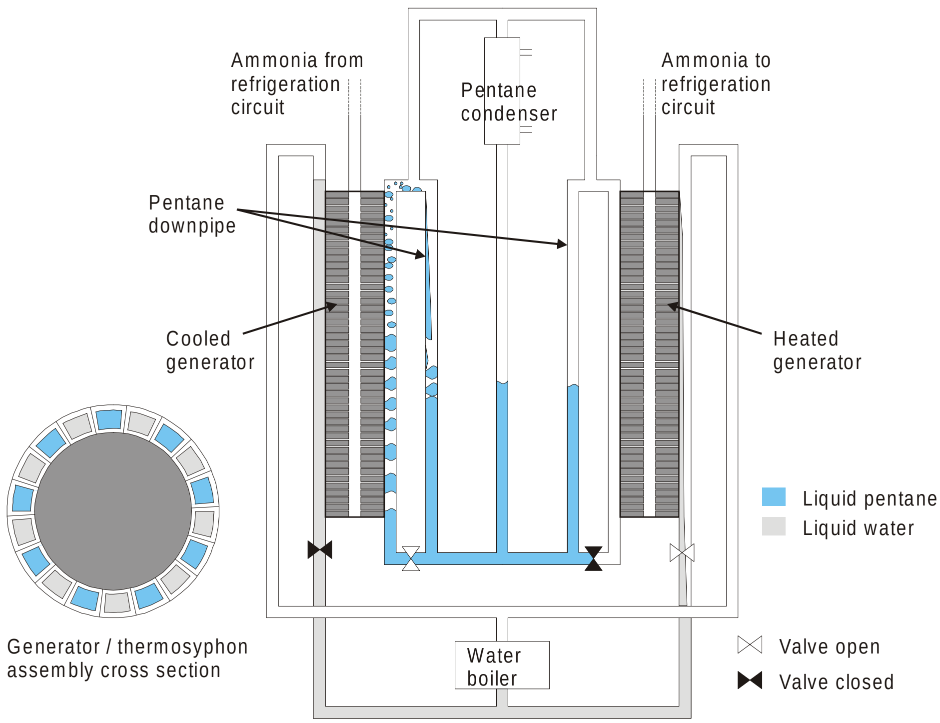

Figure 5. Proposed improved design.

Figure 5 shows the left-hand adsorption generator being cooled by boiling pentane and the right hand generator being heated by condensing water. In reality each generator will probably be surrounded by an alternating series of water condensing tubes and pentane evaporating tubes. One such arrangement is shown in cross section in Figure 5. At present, these two working fluids seem best adapted to the heating and cooling phases respectively, being above atmospheric pressure in their normal working state, with good heat transfer properties, environmentally friendly and low cost.

The valves shown in Figure 5 divert the heating and cooling to each generator as required. Closing a valve to block the return of condensate can disable the heating loop on a generator. The condenser tube fills and heating of that generator ceases. Closing a valve to block the return of condensate to the evaporator can disable the cooling loop on a generator. These valves are simple and low cost since they only need block a liquid flow against a small head. In a UK application such as waste heat utilization they would be simple refrigeration solenoid valves, typically costing £5. For developing country use it would be possible to design a simple thermostatic valve that could operate without an electricity supply. It should be noted that other low-cost valve arrangements are possible and that Figure 5 is only illustrative of one arrangement.

At the time of writing, preliminary research is underway on heat transfer in the pentane thermosyphon heat pipe and funds are being sought to build and test a complete system.

ACKNOWLEDGEMENT

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NOMENCLATURE

B_a and B_r	Material shape coefficients for axial and radial tests respectively
COP	Coefficient of Performance
K	Constant. in Equation (3)
K_a and K_r	Axial and radial permeabilities respectively (m^2)
N	Constant. in Equation (3)
P	Gas pressure (Pa)
P_C	Cooling power (kW)
P_{SC}	Specific cooling power (kW/kg carbon);
R	Sample radius (m); T-Sample temperature (ºC or K as indicated);
T_C	Condensing temperature (ºC);
T_E	Evaporating temperature (ºC);
T_G	Generating temperature (ºC); T_{sat} Saturation temperature (K);
x_o	Concentration under saturation conditions (kg NH_3/kg Carbon);
z	Sample length (m);
μ	Gas viscosity (Pa s);
v_a , v_r	Axial and radial velocities respectively (m s^{-1});
ρ	Density (kg m^{-3})

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