A PROTOTYPE ADSORPTION HEAT PUMP/CHILLER USING FORCED CONVECTION HEAT TRANSFER AND THE ACTIVE CARBON/AMMONIA PAIR.

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ABSTRACT
A solid adsorption heat pump has been built using the refrigerant gas itself as a heat transfer medium. A moving temperature front or 'thermal wave' is created within the bed of granular active carbon to facilitate a highly regenerative cycle. A novel and compact geometry has been used to simplify manufacture and to reduce the volume of the machine.

KEYWORDS
carbon, ammonia, heat pump, forced convection, regenerative.

INTRODUCTION
For simple sorption cycles an essential requirement to achieve high efficiency is that some of the heat rejected by the sorbant during the sorption process should be used in the desorption process. This is known as thermal regeneration

In liquid sorption systems thermal regeneration is achieved by the use of counterflow heat exchangers. With solid sorption systems, this is more difficult to achieve, although moving beds are possible. The thermal wave recuperator however can be seen as an analogue of the counterflow heat exchanger and has provided a useful technique for regenerative adsorption cycles.[1] In the conventional recuperator, as used in the steel industry, heat is stored in a solid matrix through which a hot gas is passed. This sensible heat is then used to heat a gas stream, usually in the opposite direction to the initial gas flow. A high effectiveness can be achieved if the heat transfer is good and the flow orderly enough (little axial mixing) to sustain a thermal wave. Typical temperature profiles generated during the heating and cooling phases in adsorbent beds for this application are shown in Figure 1.

The characteristic which largely determines effectiveness of a recuperator is the ratio of heat exchange capacity to heat capacity flowrate known as the Number of Transfer Units (NTU).

$$NTU = \frac{\alpha\ A}{\dot{m}\ Cp}$$

Whilst a counterflow heat exchanger may be able to achieve an effectiveness of 0.9 with NTU of 9 a similar performance for a recuperator requires a NTU of 20 or more, dependent on the ratio of solid heat capacity to total gas capacity flow.[2]

In order to achieve this high NTU value one of the areas of research at the University of Warwick is the convective thermal wave [3,4,5]. In this system the refrigerant gas itself is used as the heat transfer medium. This approach exploits the high surface area of the granular solid to get good heat transfer, but brings some restrictions. Since the two beds are at different pressures, it is no longer possible to transfer heat directly from the bed being cooled to that being heated. Instead, in this system the heat is stored in a packed bed of some inert material, such as steel or ceramic spheres. This heat is then recovered when the gas flow is reversed. A diagram of this type of machine is shown in Figure 2. In this diagram, the active bed on the right (carbon bed 1) is being cooled and the heat rejected is being stored in inert bed 1. Carbon bed 2 is being heated using gas that has been pre-heated by passing through the inert bed 2 before entering the heater.

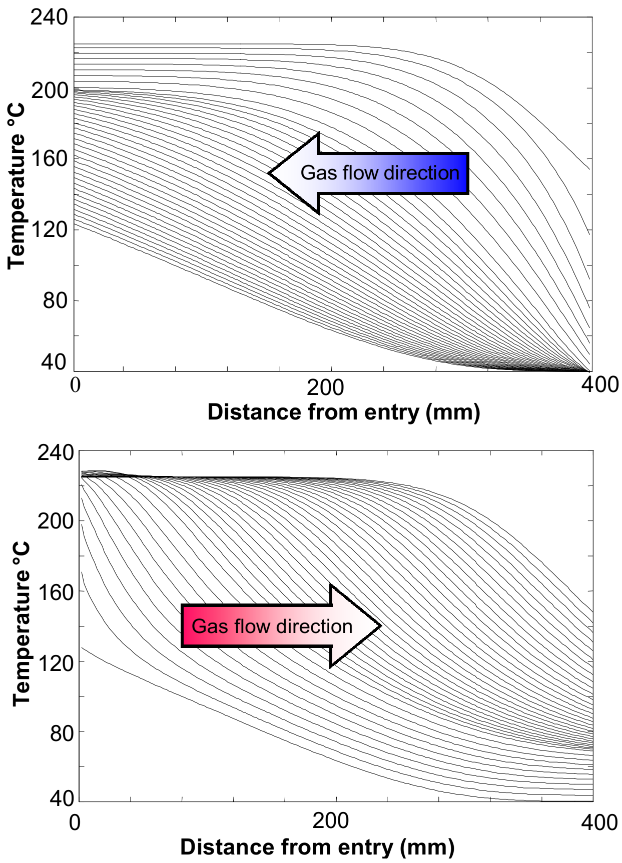

Figure 1 Temperatures in the active bed during the cold blow (top) and the hot blow (bottom).

The first experimental machine
In order to investigate the behaviour of such a system a laboratory demonstration machine was constructed. To economise and retain flexibility a system using only one active and one inert bed was made. The layout of this

machine is shown in figure 2. The gas was circulated through the system by a Roots type blower driven by an inverter controlled motor. The heat exchangers were of plate and shell design with the ammonia on the shell side. This method of construction gave good heat transfer, but more importantly had better resistance to the periodically changing pressure loads. Bypass loops enabled the heat exchangers to be put out of circuit. A branch from the main loop lead to a water-cooled condenser and the condensate was piped to an evaporator/receiver. This receiver was suspended by springs, allowing the mass of ammonia condensed to be measured using a linear displacement variable transformer (LDVT). This equipment was run for a range of conditions and operating regimes. A cooling COP of 0.8 was achieved with an evaporator temperature of 0 C, a condenser temperature of 35 C, heat input at 225 C and heat rejected at 40 C. [6]

The prototype adsorption machine

Although the laboratory demonstration machine has been run successfully it had a number of shortcomings.

The machine is large and, although composed of standard components is costly to build.

There is a large dead volume. This has the effect of reducing the amount of cooling that can be provided since the change in the mass of gas in this volume does not do any useful cooling.

The heat exchangers are costly and bulky.

Some condensation took place in the cooling heat exchanger and other cold parts of the machine.

In order to address these problems a new design is under construction. This machine will use two pressure vessels, each containing a heating and a cooling heat exchanger, condenser, circulating fan, active and inert bed. The heat exchangers in this machine will be a fin and tube design, which is well able to withstand the periodic pressure loads. A schematic for this configuration is shown in Figure 3. In this machine careful design and sizing of the heat exchangers is expected to keep the pumping power down to 350W for a machine designed for 12kW cooling capacity.

General design considerations.

Sizing of the beds and heat exchangers is an important part of the design process. The bed grain size, overall diameter and length must be selected carefully. A balance must be struck between the requirements of high regenerator effectiveness and low pressure loss. Data from experiments performed with the proof of concept rig have enabled this design balance to be made. Data from pressure drop experiments are shown in Figure 4.

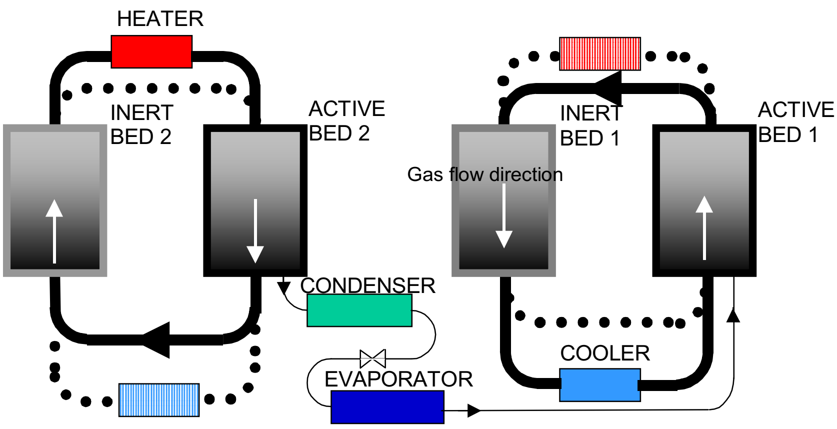

Figure 2. a simple adsorption heat pump using forced convection.

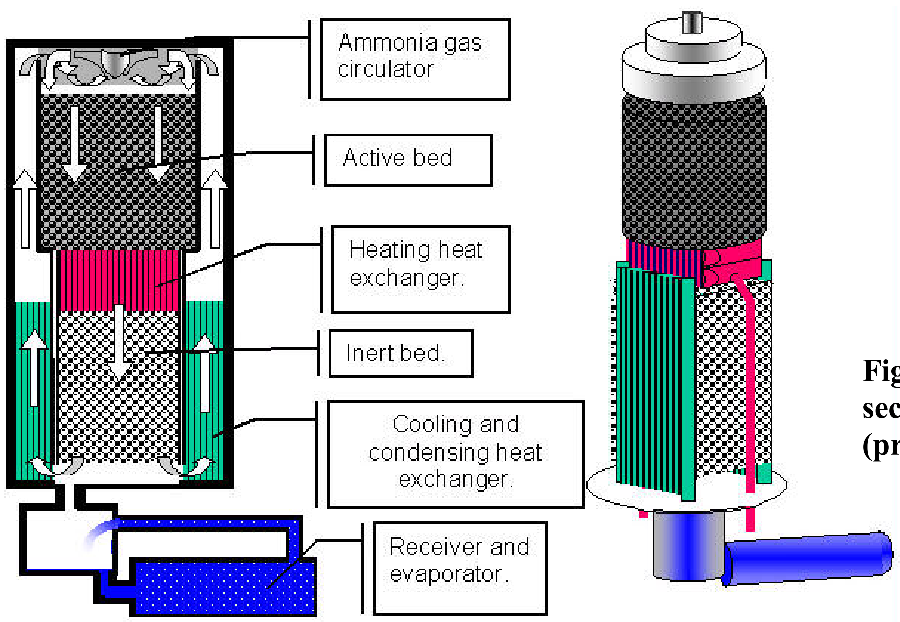

Figure 3. Prototype heat pump section view and general view (pressure vessel removed).

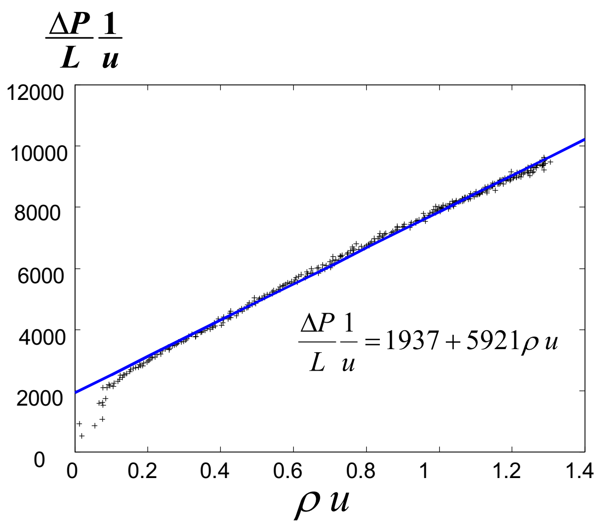

Figure 4. Pressure loss test results for the carbon bed with ammonia.

The pressure loss through a packed bed is considered to be due to a combination of viscous losses related to the hydraulic diameter of the interstitial passages and inertia forces caused by the changes of flow direction within the bed. The most commonly used relationship is that formulated by Ergun [7] where those two components compose the right hand part of the equation..

$$\frac{\Delta P}{L} = c\frac{(1-\psi)^2}{\psi^3}\frac{\mu\, u}{d^2} + m\frac{(1-\psi)}{\psi^3}\frac{\rho\, u^2}{d}$$

For the carbons used in the machines developed at Warwick with ammonia this equation reduces to the following relation when viscosity, void ratio and grain diameter are grouped together.

$$\frac{\Delta P}{L} \approx 1900u + 6000\rho\, u^2$$

For the range of gas velocities under consideration inertial effects dominate.

The large change in gas density between the condensing pressure and evaporating pressure has a strong effect on the volume flowrate and hence work required to circulate the gas.

At high pressure (20bar) and low temperature (60C) the interstitial gas velocity in the bed is 0.24 m/s. At the low pressure (5bar) and high temperature (200C) this rises to 1.7 m/s. This large velocity variation is caused by the change in density of the gas.

The density of the gas, or more specifically the heat capacity per unit volume is an important factor to be considered when assessing the suitability of heat transfer by forced convection of the refrigerant gas tocycles using particular refrigerants. This technique is unsuitable for water or methanol. The use of high pressure gases such as carbon dioxide is certainly of interest although the change in mass of refrigerant filling the void volume between the high and low pressure phases becomes large.

Almost all components of the new machine are conventional except for the ammonia circulating fan. This unit has been the principal focus of development over the last year. The conditions of volume flow, density and pressure difference led to the selection if a radial flow fan of 128mm diameter, operating at a maximum speed of 10,000 RPM. the impeller and stator are shown in Figures 5 and 6. The fan is driven by a switched reluctance motor through a set of magnetic couplings. The flow reversal is achieved by 16 cylindrical conduits with helical grooves arranged around the perimeter of the fan.

The heat exchangers within the pressure vessel are of the tube-fin type. The heating heat exchanger has a thermosyphon with condensing water on the tube side. This gives a high internal heat transfer characteristic and avoids the need for a high temperature fluid pump. The heat transfer area available on the gas side is 6 m^2. The cooling and condensing heat exchangers are connected by a water loop to a fan–coil assembly rejecting heat to air. The heat transfer area available on the gas side is 8 m^2.

The evaporator is of a plate type, cooling a water/glycol loop. Solenoid valves will be used to isolate the evaporator from the rest of the system during the condensing phase.

Figure 5. The Impeller

Figure 6. The stator ring.

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Nomenclature

A	Heat transfer area in the bed per unit volume (m^2/m^3)
c	Empirical constant used in the Ergun equation
Cp	Heat capacity of the gas (J/kg K)
d	Grain diameter (m)
L	Bed length (m)
m	Empirical constant used in the Ergun equation
$\dot{m}$	Massflow of gas per unit cross section area of the bed (kg/m^2s)
P	Pressure (Pa)
u	Approach velocity of the gas (m/s)
α	Heat transfer coefficient (W/m^2K)
Ψ	Ratio of void volume to bulk volume
μ	Viscosity (Pa.s)