Design and construction of a low cost solar chiller, with calorimetric assessment of the adsorbent bed

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ABSTRACT

This paper discusses the design and construction of a prototype refrigeration system that uses solar energy as a heat source to drive an adsorption refrigeration cycle. The system is designed to be low cost and provide cooling for food preservation in rural, off-grid locales.

Materials and construction methods were selected for their low cost, availability and compatibility with the adsorbent and refrigerant. The adsorbent bed was designed to improve heat transfer, allowing the mass of adsorbent in the system to be reduced to less than one fifth of the mass in many "standard" solar refrigerators. This allows shorter cycle times, and makes the system more compact. To further reduce size, the solar absorber was also incorporated into the adsorbent section, allowing heat to be directly transferred to the adsorbent by thermal conduction.

This paper also makes estimates of system performance based on data obtained using a previously developed calorimeter.

1. INTRODUCTION

Estimates of postharvest losses of commodities in the United States vary from 2% to 25%, depending on the type of food, while the losses in developing nations, where refrigeration is less prevalent, may be 50% or higher (1).

This research intends to design, build and test a low cost refrigeration system suitable for chilling of perishable food in such locations. These areas may not have access to a local or national electricity network, but may have plentiful sunlight; solar powered refrigeration presents the opportunity to reduce food spoilage in such locations.

Previous research in this group has focussed on such topics as designing and building a heat-powered adsorption refrigeration system using zeolite and methanol as the working pair, and the creation and use of a calorimeter for the assessment of adsorbent-refrigerant interactions, with particular emphasis on type A silica gel and water (2, 3, 4, 5).

This research intends to design, create and test a solar powered chiller with a reduced adsorbent mass and improved heat transfer characteristics, giving a relatively short cycle time. Ensuring the system has a short cycle allows the adsorbent to be cycled several times per day, rather than on a diurnal cycle. The reduction in quantity of adsorbent allows for the cost, mass and volume of the system to be reduced. A relatively small scale system was initially envisaged for this research compared to commercial systems currently available.

This paper describes progress on the design and construction of the system and results of preliminary measurements of the adsorbent dynamics measured using a calorimeter previously developed within this group.

2. SYSTEM DESIGN AND MATERIALS SELECTION

The parts and materials that comprise the system and aspects of the design are described below.
2.1 Adsorbent-refrigerant pair selection

An alternative to the adsorbent-refrigerant pairs previously used by this group was considered for this research.

The inflexible, brittle nature of granular adsorbents, and those that are available in bead, pellet or powdered form, including zeolites and silica gels, mean that they are usually adhered to heat transfer surfaces rather than press-fitted between them. When adhered to heat transfer surfaces, the layer thickness is often minimised, as adsorbents generally have poor thermal conductivity; Thicker layers would lead to longer adsorption and desorption times, reducing specific cooling power (SCP).

Generator sizes can be large in systems that use thin layers of adsorbent, however, and the resulting coefficient of performance (COP) can be lower, as thinner layers mean that a higher proportion of (inert) material is required in the system, to support the adsorbent and conduct heat into and out it.

The requirement for creating a relatively dense but thermally conductive adsorbent has led to the development of consolidated or composite adsorbent beds, where the adsorbent combined with materials of higher thermal conductivity and amalgamated into blocks (or similar forms), using pressure and chemical binders (6).

An activated carbon was chosen for use in this research, due to their high rates of adsorption. This is especially the case with activated carbon fibres and cloths, compared to other forms of activated carbon (7). An activated carbon cloth (ACC) was selected for this research (Zorflex FM50K, Chemviron Carbon, Lancashire, UK). This material also has the benefit that it can be cut to shape and pressed between heat transfer surfaces without being damaged. Pressing the ACC between heat transfer surfaces should reduce the thermal resistance between the ACC and these surfaces, and allow for a more compact generator. It also alleviates the requirements for adhesives or binders between the ACC and heat transfer surfaces, which can be incompatible with refrigerants and can soak into the ACC, potentially blocking pores in these parts, rendering parts of the cloth inert.

The choice of refrigerant, and the adsorbent to some degree, will determine the other materials that can be used in such a refrigeration system; incompatibilities may exist between refrigerants and the structural, sealing or bonding materials incorporated into a system.

While water has the advantages of being environmentally and toxicologically benign, its freezing point limits the working temperature that a chiller using water can cool to. Water also has a very low vapour pressure, which can increase ingress of air (if the system is not perfectly sealed), and a high specific volume at suitable working temperatures, which can lead to mass transfer issues. Pure water is compatible with a wide variety of structural materials, though, and the silica gel-water pair has been used most widely in commercial systems.

Methanol has been widely studied as a refrigerant. It has lower freezing point than water, allowing it to be used in ice makers, and has a higher vapour pressure at working temperatures. As has been found in previous research by this group, however, it is less compatible with many adhesives and sealants, and can corrode metals over time. Methanol has also been shown to be decompose into various other unwanted compounds in the presence of metallic materials typically found in solar refrigeration systems (especially aluminium alloys) and as trace materials in activated carbon adsorbents (9). The decomposition is accelerated at elevated temperatures, but was found to occur at temperatures of 110°C, which are easily reached in solar collectors during the day.

Ethanol has similar properties to methanol, with a lower freezing point and higher vapour pressure than water. Although the vapour pressure ethanol is somewhat lower than that of methanol, ethanol is far less toxic and is more compatible with many of the materials used in these types of refrigeration systems. For these reasons, ethanol was selected as the refrigerant for this system.

2.2 Construction materials

Refrigeration quality seamless copper tubing to BS EN 1057 was used throughout the main parts of the refrigeration circuit, due to its low cost, workability and high thermal conductivity.

A section of Perfluoroalkoxy (PFA) plastic tubing was used as a condensate receiver at the outlet of the condenser. The tubing is translucent, allowing the volume of condensate to be assessed just before the inlet to the evaporator.

Valves and fittings for tubing connection and branching were sourced for their ease of use and sealing quality under vacuum. Pressure transducer bodies were available only in stainless steel, so the reduced-bore branches to the pressure transducers were also specified in stainless steel. Otherwise all connectors, reducers and tees were specified in brass to reduce the potential for galvanic corrosion. The PFA tubing and the valves, fittings and pressure transducers used in the main refrigeration circuit were obtained from Swagelok (Bristol, UK).
Care was taken to minimise the number of joints and fittings, and use high quality fittings to minimise leak sources. Where removable parts were required, at the pressure transducers and the glass generator envelope, o-ring sealed fittings were used rather than screw-in fittings.

Standard (plumbing-type) copper tubing and brass fittings were used for the other parts of the system, including the central generator heat exchanger and the tubing to the heat exchanger, and aluminium fins were used in the generator section, fitted to the central generator tube, as described below.

![Diagram of the system layout](image)


A system diagram is shown in Fig 1, and a photograph of the system from the front is shown in Fig 2. The generator section in Fig 1 is shown rotated from its actual position by 90° (about its longest axis) in order to display the pipework that feeds into the generator more clearly. All of the valves are shown rotated so that they are viewed side-on, also to enhance clarity. The copper tubes that run down vertically from the central generator heat exchanger, and allow cooling or heating fluids to be passed through it, can be seen below the generator in Fig 2. These have been omitted from Fig 1 to show the main refrigeration circuit more clearly. Sturdy polypropylene (PP) containers were used to make the water tanks that the condenser and evaporator coils sit within, as shown in Fig. 1 (shown semi-transparent to show coils within) and Fig. 2. These are described in more detail below.

2.4 Part sizes, specifications and manufacture

To reduce pressure drops as vapour passed around the circuit, 3/4 inch (19.05 mm) tube was used in the vapour flow sections. This was the largest size of tubing that was still relatively easy to work with, allowing the bending and coil manufacture to be carried out by a single individual. Straight 3 m sections of 18 SWG (approximately 0.56 kg/m) copper tubes were used to make the straight sections of the circuit and 15 m coils of 19 SWG (approximately 0.49 kg/m) were used to create the condenser and evaporator coil. The straight and coiled tube was bent by hand or using pipe benders where required. The PFA tubing was also specified as being 3/4 inch outer diameter and 0.062 inch (1.57 mm) wall thickness.

![Photograph of the system](image)

Fig. 2: Photograph of system.
1/2 inch (12.70 mm) 20 SWG copper tube of the same quality was used either side of the three 1/2 inch needle valves used at the refrigerant input/evacuation port, the refrigerant drain port and the condensate flow control valve.

The 3/4 inch tube was reduced to 1/2 inch tube before the needle valve positioned before the entry to the evaporator coil, and expanded back to 3/4 at the evaporator coil entry. At this point the fluid should be mainly liquid, so full (3/4 inch) bore tubing was not required.

The system was assembled within the confines of a flexible industrial racking unit. The racking unit was selected for size, both for ease of movement to different locations and for a suitable volume within which to build the refrigeration system. A sturdy base with castors was fitted below the racking to aid movement of the whole system. The racking, with a height, width and depth of 1780 x 1200 x 600 mm, was divided up into three main levels with the generator on the top level, condenser on the middle level and evaporator at the bottom, with tubes interconnecting these main parts (see Fig. 1 and Fig. 2).

All copper and PFA tubing was fixed in position so that it had a minimum incline, to ensure that any liquid within the system (except that in the evaporator) drains into the PFA tubing under the action of gravity, removing the need for any pumping, and ensuring accurate volumetric measurement of the condensate.

The condenser and evaporator were sized to accommodate the estimated heat flows expected in these parts, and the PFA tubing was sized to accommodate sufficient volume of condensate. Where possible, the tubing was made larger than necessary, within the confines of the containers and the racking unit.

The water tanks in which the coils are immersed (described above) act as water baths. Here, the high surface heat transfer coefficient (SHTC) on the outside of the coil (relative to the SHTC that would be expected of, say, air under natural or forced convection) allows a high level of heat transfer between the condensing or evaporating refrigerant within the coils and the water surrounding the coils. This allows a shorter coil to be employed, without the need for external finning to increase surface area, keeping the coils short and the containers compact.

The condenser tank was sized to contain a relatively large quantity of water, and will contain 66.5 l of water with the tank fully filled to cover the condenser coil. The large volume of water provides a high heat capacity, allowing the water surrounding the condenser to effectively condense the refrigerant during a desorption phase without its temperature deviating excessively. The condenser coil was formed into an open downward helix, the main section of which has an external diameter of 360 mm, with a spacing of approximately 18 mm between coil turns, to allow good water circulation around the coil surfaces, and a total coil length of 7.1 m.

The base and walls of the evaporator tank were internally lined with polystyrene board, with a thickness of 25 mm and thermal conductivity of 0.038 W/m.K, in order to insulate the liquid within. The inside surface of the boards was lined with thick polyethylene sheeting to waterproof and protect and boards, giving an insulated and waterproof liquid bath around the evaporator. The evaporator coil was fitted so that it rested upon four feet, which in turn rested on a rigid polyethylene board in order to distribute the weight and protect the polyethylene sheet from puncture. The coil was formed into a flat spiral, with the refrigerant flow from the inside to the outside of the spiral. The coil has internal and external diameters of 235 and 480 mm, respectively, with approximately 2 to 4 mm between coil turns, again allowing good circulation around the coil surfaces. Using a flat spiral allowed for sufficient coil length (and therefore surface area) to be fitted into a minimum volume of water; the coil can be covered with using approximately 8.7 l of water. This gives the water in the evaporator tank a lower heat capacity, allowing the quantity of refrigerant evaporated during an adsorption phase to be inferred from the temperature change, and allowing the temperature to be reduced to the working temperature the in only a few cycles. The horizontal spiral also maximises the surface area over which the refrigerant can boil in the evaporator. The flat spiral part of the coil is 6.5 m in length, and the total length of the evaporator section (in the evaporator container) is 7.3 m.

The 3/4 inch PFA tubing section between the condenser and the evaporator is 2.4 m in length, with a wall thickness of 1.6 mm.

2.3 Specialised construction and parts

The adsorbent bed for the generator section of the system was manufactured using a modification of an existing process used to make domestic and commercial heater (Turnbull & Scott (Engineers) Ltd., Hawick, Scotland). In this process, aluminium fins are stamped out give 75 mm x 75 mm, 0.38 mm thick aluminium fins with 0.35 mm high stiffening ridges and a central elongated collar, as shown in Fig. 3. These fins are usually fitted over a central 28 mm copper tube and the tube expanded to press-fit the fins onto the tube, with the collar on the aluminium fins giving a good fiction fit and providing an extended heat transfer surface between the copper tube and fins.
Before the fins were fitted to the central tube, 5 mm of each fin was folded over by 90°, as shown in Fig. 3. This provided a second extended heat transfer surface onto which a solar selective material, consisting of a 75 x 500 x 0.5 mm aluminium sheet, with a selective absorber coated onto one outer surface (Alanod-Solar GmbH & Co. KG, Germany) could be adhered.

ACC pieces, as shown in Fig. 3, were cut from a large sheet of ACC so that they fitted into the space between the pre-folded fins. Due to the large number of pieces required, the pieces were cut using a computer numerical control (CNC) cutter, fitted with a vacuum bed to hold the cloth in place while cutting. This machine is usually used for cutting glass and carbon fibre composite materials for the aerospace industry.

102 pre-folded fins were threaded onto the central copper tube, with 9 layers of ACC between each of the fin pairs, giving a total ACC mass of approximately 870 g, between 750 g of aluminium fins. The ACC had been exposed to the atmosphere before its it was weighed. Previous measurements estimate that the carbon cloth would contain approximately 28% by mass of atmospheric moisture under these circumstances, and so the dry mass of the carbon cloth would be approximately 630 g.

The complete composite bed measures 518 x 75 x 70 mm. It provides an effective means for removal of heat from the ACC during adsorption by conduction along the fins to the central tube, where heat can be carried away by a fluid circulating within the tube, and addition of heat during desorption by absorption of solar radiation on the solar selective surface and conduction from there along the fins to the ACC.

The central tube was brazed to a brass base plate, so that it penetrated the base plate, and a leak-tight seal was made between the tube and base plate. Similar joins were made for two additional copper tubes of 3/4 inch diameter. These tubes were used for connecting the rest of the refrigeration circuit to the generator and for passing temperature measurement devices into the generator section.

The top of the central tube was capped, and a 15 mm diameter copper tube was passed up the centre to approximate just short of the top cap. This was fused to a tee piece at the base of the central 28 mm tube and a second 15 mm tube was fitted to the tee. These parts, when brazed together, allowed a cooling fluid to be passed up inside the space between the 28 and 15 mm tubes and back down the central 15 mm tube, providing a coaxial central heat exchanger for the adsorbent bed, with only one penetration into the generator, and with the fins and ACC mounted directly onto the outer surface of the heat exchanger.

Fig. 3: Construction of the adsorbent bed, dimensions in mm.

The central heat exchanger also provided a method by which the bed could be heated if required. This has been successfully tested using steam as the heat transfer fluid in order to desorb the atmospheric water adsorbed onto the ACC before the system is filled with refrigerant.

The brass base plate also featured an o-ring seal to seal the generator against a glass dome that was specially made to fit over the adsorbent bed, sealing the generator section while allowing solar radiation to be admitted to the solar selective surfaces within.

3. ADSORBENT CALORIMETRY

A recently developed calorimeter was used to obtain some initial adsorption kinetics and equilibrium data for the adsorbent, assisting the design process.

The calorimeter design allows the adsorption kinetics to be measured, not with the adsorbent in isolation, but together with the heat transfer that occurs in the adsorbent bed,
while maintaining the sample at typical system conditions. The calorimeter is discussed in a previous publication (5). More tests using this system are planned.

Approximately 6.50 g of ACC was used in the calorimeter. This had been exposed to the atmosphere before insertion, and so, again, would have contained approximately 28% by mass of atmospheric moisture. The actual mass of (dry) ACC was therefore approximately 4.68 g.

The ACC adsorbent was mounted in a finned arrangement with similar properties to the adsorbent bed throughout the adsorption tests. Before each test, the ethanol was desorbed from the ACC by heating the ACC to 70°C for approximately one hour, while maintaining a vacuum on the adsorbent section.

During the tests, the temperature of the finned substrate was controlled to 30°C, while the evaporator section below was maintained at 20, 10, 5, 0 or -5°C. Opening the valve between the adsorbent and evaporator sections allowed adsorption to progress while the heat flows were estimated, and the temperatures were maintained.

Adsorption demonstrated in all cases and the change in adsorbent loading was estimated to be over 30%. The duration of the adsorption phase varied with evaporator temperature, with a minimum time of approximately 250-300 s.

If adsorption and desorption times of this order are achievable in the full system, it will allow for multiple cycles per day, improving SCP.

4. DISCUSSION AND CONCLUSIONS

A prototype refrigeration system that uses solar energy as a heat source to drive an adsorption refrigeration cycle, intended for food preservation in rural, off-grid locales, was discussed.

This system is designed to be relatively compact and use low cost, easily available materials and construction methods, while allowing solar powered adsorption refrigeration cycles to be studied.

The system was designed to have relatively short cycle times. It has been shown previously with similar adsorption pairs, that the adsorbent can heat up enough to significantly reduce adsorption rates, if a thick adsorbent bed is used without including pathways for the rapid dissipation of heat (10). While a compact bed can increase the mass of refrigerant adsorbed per unit volume, it is also important to include highly thermally conductive pathways to ensure rapid adsorption and short cycle times. The design of the generator, with a composite adsorbent bed that contains closely spaced aluminium fins, should allow rapid heat transfer during adsorption.

Some of the parts, especially the fittings used, are more costly than may be used in a production system. This system is intended for use as a research tool, however and so there is a requirement for a build that allows changes to be made if necessary.

On completion of a final design, considerable cost savings could be made by replacing these fittings with cheaper alternatives and making direct tube-tube connections where possible.

This system should allow the interaction between the construction materials and the adsorbent and refrigerant to be assessed over time, and recommendation to be made for future designs. It may be possible to test other refrigerants in the system, and if the generator section were exchanged, other adsorbents could also be tested.

Future research intends to investigate alternatives to the relatively costly sensors and system valves that are required for monitoring the system and changing between cycle phases. Automation of these processes, if it is possible at low cost would be a considerable advantage.

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6. REFERENCES


