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## Compact High Efficiency Adsorption Heat Pump

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### ABSTRACT

An innovative adsorption cycle heat pump technology is presented that is compact and capable of achieving high energy efficiency for integrated space heating, air conditioning, and water heating. High energy efficiency is accomplished by effectively recuperating heat within the system to minimize energy consumption. This substantially reduces the thermodynamic losses that occur when the sorbent beds are thermally cycled without effective heat recuperation. Furthermore, equipment cost is reduced by thermally cycling the sorbent beds very rapidly using embedded microchannel heat exchangers, which reduces size and cost of the beds. Performance of the cycle is assessed for ammonia refrigerant and carbon sorbent using two models to simulate a sorption compressor, a simplified lumped-parameter model and detailed finite element analysis. Results from the two models are compared for validation and also used to explore the effects of system configuration, bed geometry, and operating conditions. Primary energy coefficients of performance (COP) as high as 1.03 are predicted for cooling and 1.68 for heating at AHRI standard test conditions, assuming 90% fuel utilization and 3% parasitic power. Furthermore, a heating COP of 1.24 is feasible at -25°C outdoor temperature with no more than 50% reduction in heating capacity.

### 1. INTRODUCTION

Adsorption heat pumps represent an alternative technology for providing space heating and cooling, as well as for other applications such as refrigeration and water heating. The significant difference from conventional technologies is that heat is used to drive a thermodynamic cycle instead of a compressor that consumes electric power. This enables the use of alternative energy sources such as natural gas, exhaust heat, or solar (Meunier 1994; Li and Sumathy 1999), and creates opportunities for off-loading power demand from the grid as well as reducing carbon emissions by using renewable energy sources. While absorption heat pumps using ammonia-water and lithium bromide-water solutions as working fluids are commercially available for building HVAC, solid-based adsorption systems have had difficulty achieving comparable energy efficiencies. This can be attributed to the temperature gaps between the cycle and the external sources and sinks (Meunier, Poyelle et al. 1997) and due to the thermal mass of the sorbent media and the hardware that must be thermally cycled. Sorbent capacity and ratio of hardware mass to sorbent mass are key factors affecting efficiency (Tamainot-Telto, Metcalf et al. 2009a; Tamainot-Telto, Metcalf et al. 2009b). Among the approaches that have been developed to improve efficiency are multi-bed regenerative systems (Critoph 2002), thermal wave (Miles and Shelton 1996; Critoph 1999), and cascading cycles (Douss and Meunier 1989). A second challenge with adsorption heat pumps is the size of the sorbent beds, which is driven by cycle time that is typically limited by heat transfer in the sorbent media. Advancements have been made in improving thermal conductivity of monolith sorbent media (Tamainot-Telto and Critoph 2001) and adding high conductivity fillers (Guilleminot, Chalfen et al. 1994; Aghbalou, Mimet et al. 2004), as well as through advanced bed designs that reduce overall thermal resistance (Critoph and Metcalf 2004).

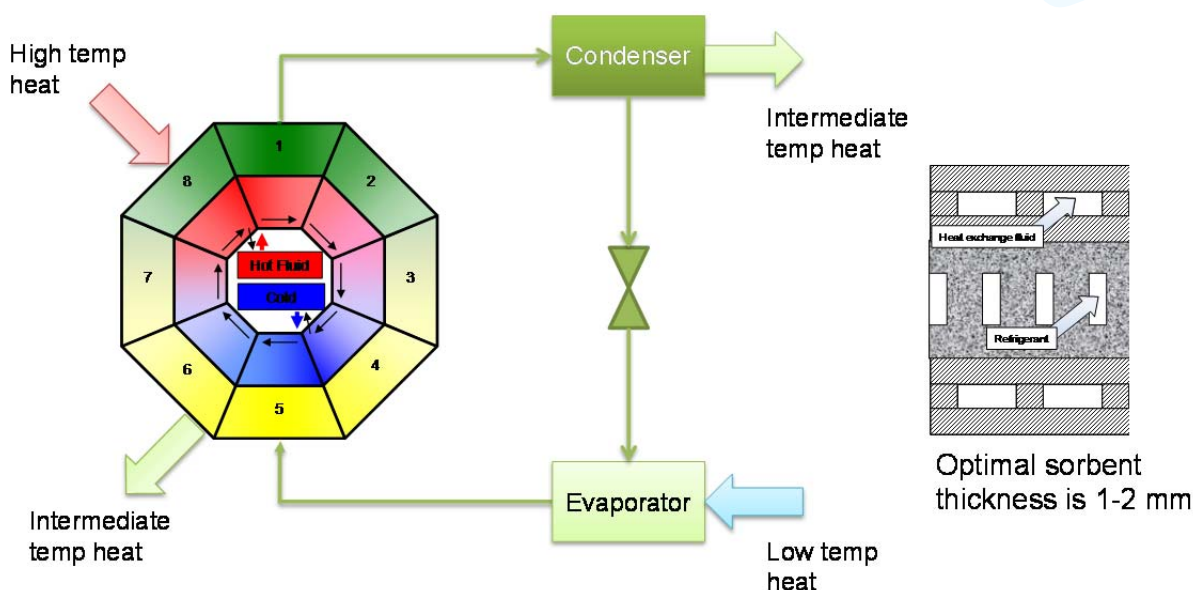
Here, a patented, novel concept is explored that promises high energy efficiency and rapid cycling of the sorbent beds (Wegeng, Rassat et al. 2003; Wegeng, Rassat et al. 2004; Wegeng, Rassat et al. 2005). High energy efficiency

is obtained with multi-beds that are configured to effectively recuperate heat from beds being cooled to beds being heated. A similar concept was developed by Tamainot-Telto et al. (2009). The concept used here was originally developed for compressing CO<sub>2</sub> from the Martian atmosphere as a raw material for in situ propellant production (Brooks, Rassat et al. 2005). Performance is assessed for an ammonia adsorption system using ATMI monolithic carbon and a stacked-plate bed architecture (Tamainot-Telto, Metcalf et al. 2009a).

## 2. CONCEPT

A schematic for a high-efficiency adsorption heat pump is shown in Figure 1. The condenser, throttle valve, and evaporator are equivalent to devices found in vapor compression cooling systems, but the mechanical compressor is replaced with an adsorption compressor, which uses high temperature heat and rejects intermediate temperature heat to create compression. Consequently, the principle energy source is heat instead of electric power, although power is still required for moving heat transfer fluids and other parasitic loads. High temperature heat has multiple possible sources including burning fuel, such as natural gas, solar thermal heat, or a waste heat source. When operated in cooling mode, such as for air conditioning or refrigeration, the evaporator provides the cooling duty while intermediate heat from the condenser and compressor is rejected. The intermediate heat is used in heating mode, such as for space or water heating, and energy efficiency gains are obtained by the evaporator extracting heat from a cold source.

The key for attaining high energy efficiency is the multi-bed adsorption compressor depicted in Figure 1. Each bed in the system is thermally cycled as it virtually rotates counter-clockwise. Refrigerant is adsorbed at low pressure when a low-temperature bed is at position 5, and then desorbed at high pressure as the bed is heated in position 1. The beds do not physically rotate, but are 'moved' using a system of valves to change inlets and outlets. The beds are heated and cooled using a heat transfer fluid that flows through the beds sequentially in the clockwise direction. Heat is added to the heat transfer fluid prior to entering the bed at position 1 and removed prior to position 5. Using a heat transfer fluid to heat and cool beds effectively recuperates heat from beds being cooled to beds being heated. The concept is illustrated in Figure 1 with system of 8 beds, but modeling results have shown that there are marginal efficiency gains with more than 4 beds (TeGrotenhuis, Humble et al. 2012), and a minimum of 6 beds is needed to maintain continuous refrigerant flow.



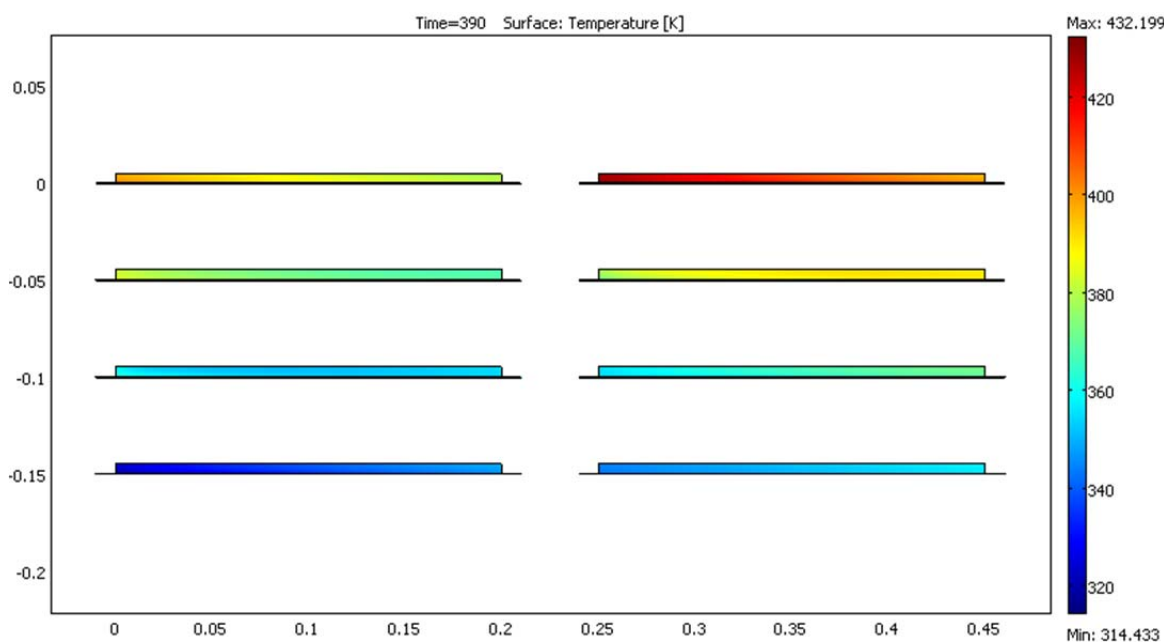
**Figure 1:** Schematics of the multi-bed adsorption heat pump concept and stacked plate bed design.

The second important aspect of the concept is superior bed productivity, which is necessary to reduce the size and cost of the sorbent beds. High productivity is achieved by rapid cycling which is made possible by high heat and mass transfer rates. This is achievable with a stacked-plate bed structure like the one shown in cross-section in Figure 1, which has monolithic sorbent plates interleaved with heat transfer structures. The geometry is very similar

to the plate-type sorption generator that was previously shown to have high bed productivity (Critoph and Metcalf 2004). Refrigerant flows into and out of the beds through microchannels to reduce pressure drop, while heat transfer fluid flows that microchannels in alternating layers for heating and cooling.

### 3. MODELING

Performance of the multi-bed adsorption heat pump has been predicted using two different computational models. The first is a dynamic, 2-dimensional finite element model (FEA) solved using COMSOL<sup>®</sup> Multiphysics software. Figure 2 shows a snapshot of temperature profiles from a simulation of an 8-bed system. The model for each bed has two domains, the sorbent bed and the heat transfer channel, which are coupled through continuity temperature and heat flux boundary conditions. In Figure 2, the temperature profiles of the bed domains are visible, while the smaller heat transfer domains only appear as lines below the beds. The porous sorbent is modeled as a slab with the top surface open for refrigerant flow into or out of the media, and flow is described by Darcy's law with an ideal gas. The sorbent has two phases, free and sorbed gas, and adsorption rate is modeled using a linear driving force model—the rate of sorption is proportional to the difference between the equilibrium loading at the local temperature and pressure and the local adsorbed concentration. Half of a heat transfer channel is modeled with a symmetry boundary condition on the bottom, a free thermal boundary condition coupled to the sorbent media on the top, and an imposed laminar flow profile. Heat transfer fluid enters at one end at a uniform temperature and exits the other end. Convective-diffusive heat balance equations with dynamic accumulation terms are solved for both the sorption bed and the heat transfer channel, and the sorbent heat balance equation includes a heat of adsorption term. The beds are coupled through the heat transfer fluid inlet temperatures. The upper right and lower left beds in Figure 2 have fixed inlet temperatures simulating heat input into and heat rejection from the compressor, respectively. The heat transfer fluid inlet temperatures of the other beds are coupled to the outlet temperatures of the previous bed in the system.



**Figure 2:** Example temperature profiles of an 8-bed adsorption compressor obtained from 2-D FEA modeling.

In addition to predicting detailed thermal and concentration profiles within the bed, the FEA model is used to predict overall performance metrics including coefficient-of-performance and cooling duty for the heat pump system depicted in Figure 1. Furthermore, overall heat and mass transfer coefficients can be calculated from bed-averaged properties and fluxes.

The FEA model is computationally intensive requiring hours of computer time to solve a single case, which makes optimizing operating parameters onerous and impractical. Therefore, a second lumped-parameter (LP) model using

transport coefficients validated by the FEA model was used for parametric and optimization studies. The five time-dependent variables for each bed are pressure, uniform bed temperature, total bed loading, heat exchanger fluid outlet temperature, and ammonia flow rate into or out of the bed. Single-bed models are coupled together in the same manner as in the FEA model. The simulation proceeds for one cycle step where the beds change position, so the final bed temperature, pressure, and loading must be equal to the starting conditions of the next bed position. This creates a boundary value problem that is solved with a successive substitution shooting method implemented in MATLAB<sup>®</sup>. Further details of the lumped-parameter model are provided by TeGrotenhuis et al (2012).

### 3. RESULTS

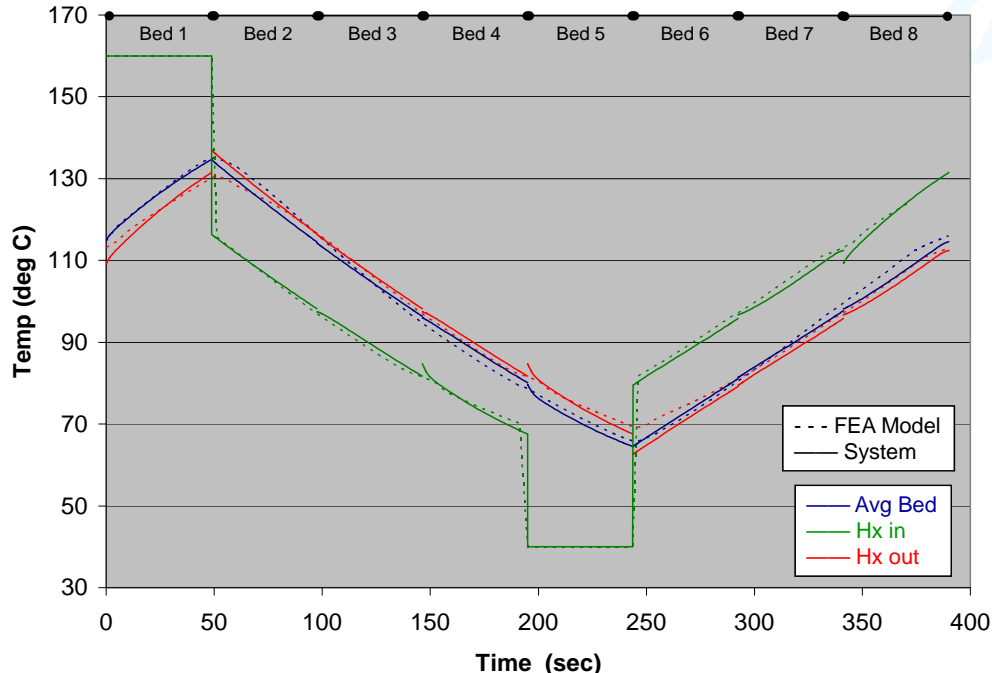
#### 3.1 Model Comparison

Calculations were performed using a monolithic carbon material developed by ATMI that is similar to SDS monolithic carbon previously investigated (Tamainot-Telto, Metcalf et al. 2009b) but having increased working capacity. The parameters for the Dubinin-Astakhov equation,

$$x = x_0 \exp \left( K \left( \frac{T}{T_{sat}} - 1 \right)^n \right) \quad (1)$$

are  $x_0=0.2599$  g NH<sub>3</sub>/g C,  $K=5.8413$ , and  $n=1.8$ . The heat of adsorption of ammonia is 33.73 kJ/mol which is assumed to be constant. The carbon bulk density is 0.894 g/cm<sup>3</sup>, internal void fraction is 10%, specific heat is 1.14 J/gK, thermal conductivity is 1.77 W/mK, and permeability is  $28 \times 10^{-14}$  m<sup>2</sup>. The heat transfer fluid is assumed to be Paratherm with a heat capacity of 2.2 J/gK and thermal conductivity equal to 0.12 W/mK.

Figure 3 shows a comparison between the FEA and LP models of the average bed and heat exchanger fluid inlet and outlet temperatures through a complete cycle of an 8-bed system. There are discrepancies in the details of the thermal profiles, most notably at the transitions between bed positions.



**Figure 3:** Comparison of temperatures between FEA and lumped-parameter (System) models.

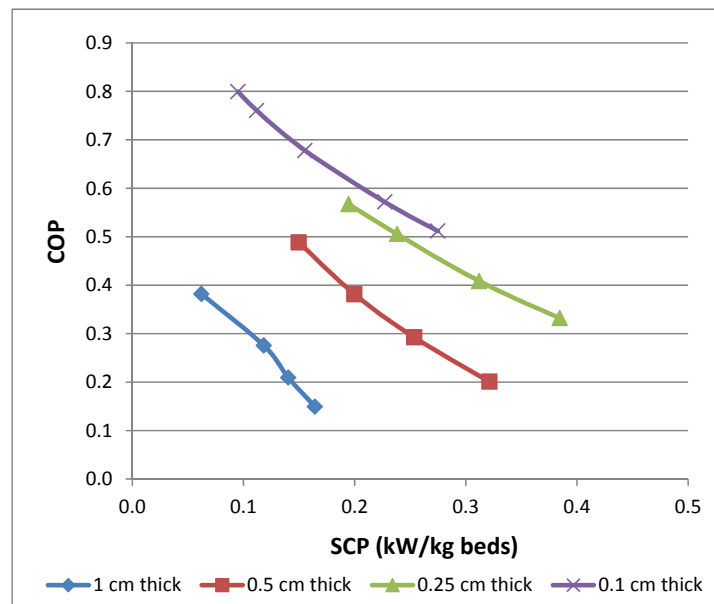
A more important comparison is the heat pump performance metrics of energy efficiency and sorbent bed productivity. Energy efficiency is indicated by the coefficient of performance (COP), defined as the cooling capacity divided by the heat input, where the time-averaged duty from the high temperature heat source is the heat

input. Cooling capacity is calculated by simulating an ammonia refrigeration cycle to determine the evaporator duty per mass of ammonia flow. A state point is chosen for the ammonia leaving the condenser, which is expanded adiabatically from the high pressure to low pressure for the cycle, and the remaining latent heat divided by the total ammonia mass flow is the cooling capacity. Bed size is assessed by the specific cooling power (SCP), defined here as the cooling capacity divided by the total bed mass including sorbent and hardware mass. For the case shown in Figure 3, the cooling capacity predicted by the FEA and LP models are  $53.1 \text{ kW/m}^3$  and  $47.2 \text{ kW/m}^3$ , respectively, so the LP is 11.2% lower than the FEA model. The COP values are 0.40 and 0.38  $\text{kW/m}^3$  for the FEA and LP models, respectively, a 4.6% discrepancy. Bed productivities are the closest with the FEA model predicting  $0.047 \text{ kW/kg}$ , while the LP models predicts 2.7% lower specific cooling capacity. While this case indicates the LP model under predicts performance compared to the FEA model, it is unclear whether this is a consistent outcome.

### 3.2 Heat Pump Performance

The computationally efficient lumped-parameter model facilitates parametric studies to ascertain optimal performance for a given bed design and operating conditions. For each case, the heat transfer fluid flow rate and cycle time are varied to obtain an optimal operating curve that represent the trade-off between energy efficiency and bed productivity. Figure 4 shows operating curves derived for four bed thicknesses.

Sorbent bed thickness is a design parameter that illustrates a performance trade-off. Bed productivity increases as the sorbent plates become thinner, because heat transfer dynamics become faster as the plates become thinner and the heat transfer area increases. On the other hand, the increase in heat transfer area means more structural mass that must also be thermally cycled which diminishes energy efficiency. Which effect dominates is elucidated by calculating operating curves as shown in Figure 4. A standard heat exchanger design was used for each plate thickness and the additional structural mass including headers, vessel walls, and fittings was assumed to be 50% of the heat exchanger mass. As a result, the ratio of structural mass to sorbent mass was only 0.25 for a plate thickness of 1 cm, but increased to 0.52, 1.03, and 2.6 for 0.5 cm, 0.25, and 0.1 cm thicknesses, respectively. Despite the steep increase in structural mass, the operating curve continues to improve even with the thinnest plates. However, the other consideration is bed manufacturing cost which obviously will increase with thinner plates, so the economic optimum is unlikely to correspond to the optimum operating curve.



**Figure 4.** COP versus SCP for an 8-bed system operating with  $160^\circ\text{C}$  heat source,  $40^\circ\text{C}$  cold sink, and  $9^\circ\text{C}$  chill with 0.1 cm ( $\times$ ), 0.25 cm ( $+$ ), 0.5 cm ( $\blacksquare$ ), and 1 cm ( $\blacklozenge$ ) thick carbon plates.



### 3.2 Space Cooling with Natural Gas

Sorption heat pumps running primarily using heat obtained from burning fuels, solar heat, or waste heat sources have lower coefficients of performance than electric heat pumps because of the lower exergy content of the energy source. However, they can be compared on a common basis if the power consumed is related back to a primary energy source used for generation. For an electric heat pump, the primary-energy COP is

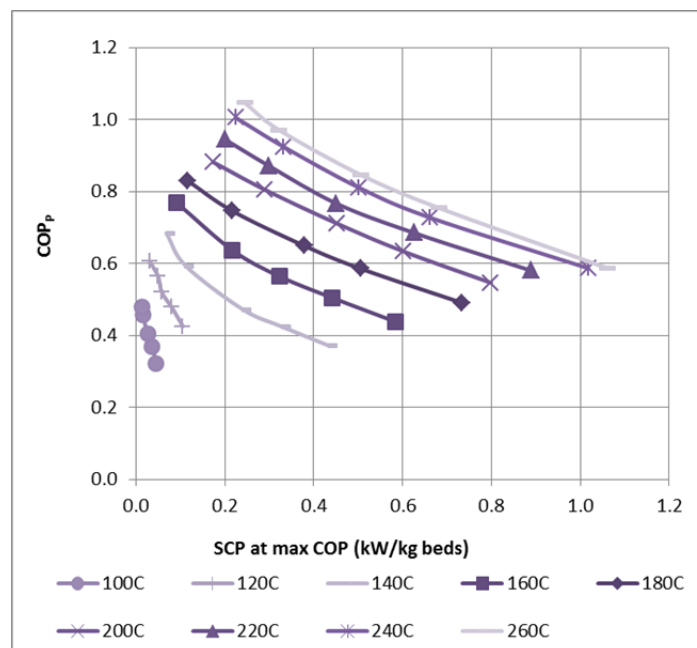
$$COP_P = \eta_E COP_E \quad (2)$$

Where  $\eta_E$  is the overall efficiency of delivering power to the consumer, and  $COP_E$  is the coefficient of performance of the system itself, including parasitic power for pumps and fans. The U.S. national average thermal efficiency of all fossil fuel power plants has been trending up from about 32.5% in 1999 (DOE and EPA 2000) as older inefficient power plants are retired and newer higher efficiency plants are brought into the fleet. Recent natural gas combined cycle plants can exceed 50% efficiency (NPC 2007). Distribution losses lead to overall efficiency of delivered residential power in the range of 28-32% (Lasseter and Paigi 2004). In January 2006, the U.S. established a seasonal energy efficiency ratio (SEER) of 13 as the minimum efficiency standard for residential air conditioners, and the ENERGY STAR program requires a SEER of 14. The equivalent energy efficiency ratios (EER) are 11.2 and 11.8, respectively (Hendron and Engebrecht 2010). Applying a delivered power efficiency of  $\eta_E=0.3$ , the corresponding  $COP_P$  values are 0.98 and 1.03, respectively, which are reasonable benchmarks for cooling.

Sorption heat pumps operating on natural gas do not fully utilize the higher heating value of the fuel and have parasitic power loads to run pumps and blowers. The COP based on heat input is converted to a primary energy  $COP_P$  using the equation

$$COP_P = \left( \frac{1}{\eta_{FU}} + \frac{f_P}{\eta_E} \right)^{-1} COP \quad (3)$$

where  $\eta_{FU}$  is the fuel utilization efficiency defined as the fraction of higher heating value delivered to the sorption cycle, and  $f_P$  is the ratio of parasitic power to the higher heating value of the fuel consumed. Natural gas condensing furnaces now exceed 90% fuel utilization efficiency, so the value  $\eta_{FU}=0.9$  is used in Equation (3). The parasitic power demand depends on the overall heat pump design for a given duty and application, so a value of 3% is assumed here.



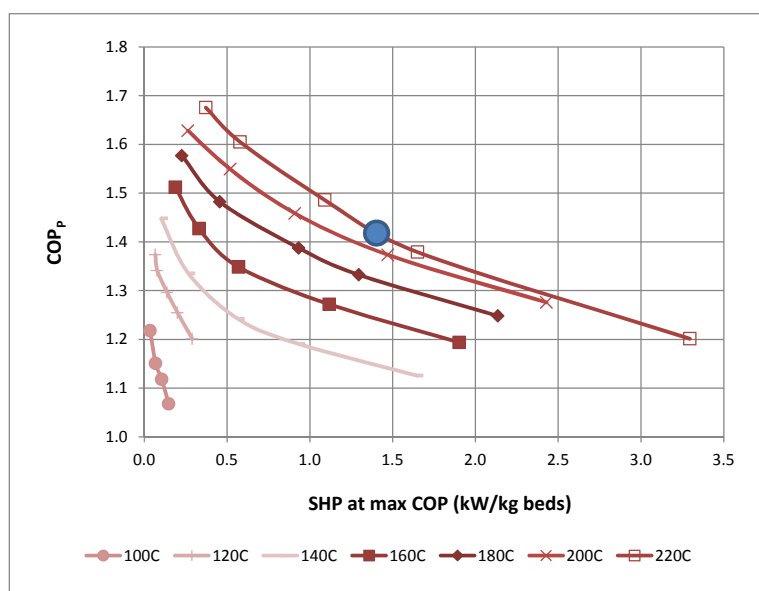
**Figure 5.**  $COP_P$  for cooling based on primary energy input versus SCP for a 6-bed system of 0.25 cm carbon plates operating with 40.6°C cold sink and 7.2°C evaporating temperature at a range of heat delivery temperatures.

Figure 5 shows operating curves calculated using the LP model for a refrigeration heat pump operating with a range of heat source temperatures, a condensing temperature of 40.6°C, and an evaporating temperature of 7.2°C. These temperatures are compatible with the AHRI 2008 Standard test conditions for air conditioners (Air-Conditioning 2008) at 26.7°C indoor temperature of and 35°C outdoor temperature. The heat rejection temperature for the adsorption compressor is also set to 40.6°C. At the highest source temperature of 260°C, which is easily obtained with natural gas combustion, the COP<sub>p</sub> can exceed 1.0, making it competitive with commercial electric air conditioners. Lower temperature heat sources, such as waste heat or solar, below 100°C are more in the range of COP<sub>p</sub>=0.5 and require much larger sorbent beds, but have the advantages of a free energy source and reduced life-cycle carbon emissions.

Similar results are presented in Figure 6 for space heating at a comparable condensing temperature of 40°C and a colder evaporating temperature of 2.8°C. These temperatures are compatible with the AHRI 2008 Standard conditions for space heating with an indoor temperature of 21.1°C and outdoor temperature of 8.3°C. The heat rejection temperature for the adsorption compressor is also set to 40°C. The heating mode COP<sub>p</sub> is equal to one plus the cooling mode COP<sub>p</sub>, because both the reject heat from the compressor and the condenser duty are used for space heating. At moderate outdoor temperatures where heat pumps show the most benefit for space heating, the primary energy COP can approach 1.7, which is double the energy efficiency of natural gas furnaces operating with 0.85 fuel utilization efficiency (FUE).

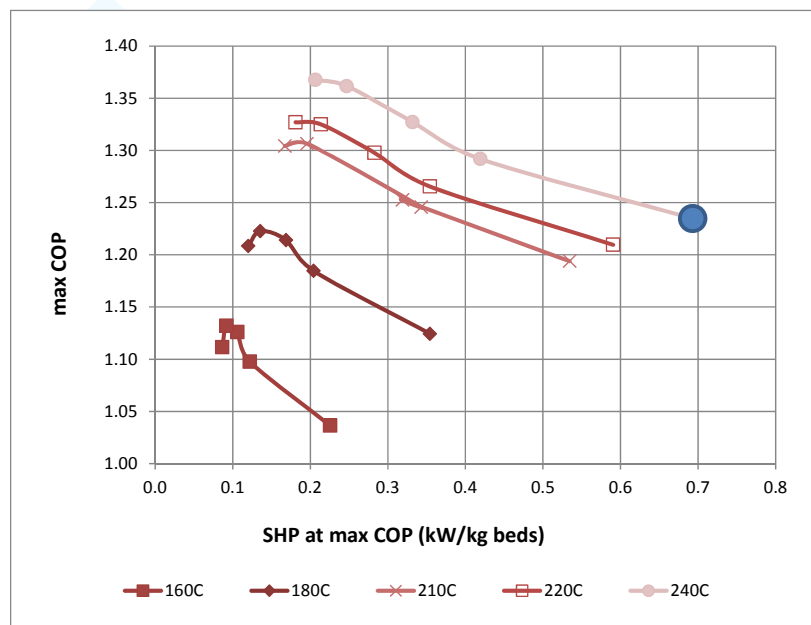
As expected, heat pump performance suffers significantly as the outside temperature drops, both in energy efficiency and in heating capacity. Figure 7 shows LP model results when the evaporator temperature is reduced to -29°C, which would accommodate space heating in cold climates as low as -25°C outside temperature. As expected, the operating curves shift to lower heating COP<sub>p</sub> and specific heating power (SHP) values. Lower specific heating power corresponds to larger sorbent beds and/or reduced heating capacity. Nevertheless, heating COP<sub>p</sub> values exceeding 1.35 are feasible indicating a significant benefit in energy efficiency even in very cold climates.

The blue circles in Figures 6 and 7 illustrate that it is possible to keep the loss in heating capacity to 50% as the outside temperature drops from 8.3°C to -25°C while maintaining high energy efficiency. In Figure 6, the blue circle corresponds to a COP<sub>p</sub> of about 1.42 while the system can produce 1.4 kW/kg of bed mass, which includes the structural mass of the beds but excludes other system components. At 50% capacity, the SHP drops to 0.7 kW/kg, which corresponds to a COP<sub>p</sub> of 1.24 at the -25°C outside temperature condition, which is 46% higher energy efficiency than a 0.85 FUE natural gas furnace.



**Figure 6.** Heating mode COP<sub>p</sub> versus SHP for a 6-bed system of 0.25 cm carbon plates operating with 40°C condensing and 2.8°C evaporating temperatures at indicated heat source temperatures.





**Figure 7.** Heating mode  $COP_p$  versus SHP for a 6-bed system of 0.25 cm carbon plates operating with 40°C cold sink and -25°C chill at indicated heat source temperatures.

#### 4. CONCLUSIONS

A concept has been described for a high efficiency adsorption heat pump, and two different computational models have been presented for predicting performance. Comparison of results from a simplified lumped-parameter model to a detail FEA model indicates errors of less than 10% in using the computational efficient lumped-parameter model. Results have been presented for ammonia refrigerant and monolithic, nano-structured carbon constructed in a stacked-plate architecture with integrated microchannel heat exchangers. Operating curves were predicted for plate thicknesses from 1 mm to 1 cm, and results indicate that performance continues to improve with thinner plates, even in going from 2.5 mm to 1 mm. Performance gains must be assessed in the context of higher fabrication costs.

Calculated results with natural gas combustion as the heat source at standard test conditions for residential cooling and heating indicate energy efficiencies are feasible that are competitive with commercial HVAC systems when compared on a primary energy basis. Primary energy coefficients of performances as high as 1.03 for cooling and 1.68 for heating have been obtained. Simulations performed for cold climates down to -25°C outside temperature also show promising energy efficiency gains over furnaces, with  $COP_p$  as high as 1.37 predicted. A system that could achieve a  $COP_p$  of 1.42 at 8.3°C outside temperature could maintain 50% of heating capacity at a  $COP_p$  of 1.24 when the temperature drops to -25°C.

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