

## LIMITS TO THE THERMODYNAMIC PERFORMANCE OF A THERMAL WAVE ADSORPTION COOLING CYCLE

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### ABSTRACT

A thermal wave cycle is one method to achieve thermal regeneration in an adsorption cooling cycle. A thermodynamic model is developed to define the performance limits to this cycle. A zeolite-water adsorbent-refrigerant pair with maximum bed temperatures ranging from 110 to 200°C is predicted to have a limiting COP of 1.7 to 3.6. Relative to an ideal simple adsorption cycle with no thermal regeneration, these results translate into the potential to reduce the driving heat transfer by ~ 70%. These reductions in the driving heat transfer are similar to those for a system with an infinite number of beds following the simple cycle with maximum thermal regeneration.

### 1. INTRODUCTION

Air-conditioning represents a large and/or rapidly growing part of the total energy demand in many countries (e.g., [1]). The negative economic and environmental consequences associated with conventional cooling technologies are well documented (e.g., [1] [2]). Adsorption cooling cycles can potentially reduce many of these negative consequences, since the primary energy input can be waste heat or solar thermal energy and environmentally benign adsorbent-refrigerant pairs such as natural zeolite-water can be used [2]. Many reviews of this technology exist (e.g., [2]-[4]).

A potential barrier to commercialization of a solar thermal powered system is large collector costs [5]. Collector costs can be reduced by increasing the adsorption cycle's COP through thermal regeneration [3][4][6]. Models to predict the thermodynamic performance limits of a simple cycle without thermal regeneration and with thermal regeneration using an infinite number of beds have been developed [5][6]. An alternate manner in which to achieve thermal regeneration is through a thermal wave cycle [3]. The objectives for this research are as follows: (1) develop a model to predict the thermodynamic limits to the performance of a thermal-wave adsorption cooling cycle, (2) explore trends in performance as the maximum bed temperature is varied, and (3) compare this limiting performance to that of other adsorption cycles.

### NOMENCLATURE

#### Abbreviations

*HTF* Heat transfer fluid  
*TR* Temperature reservoir

#### Variables

*c* Specific heat of incompressible substance ( $\text{kJ kg}^{-1} \text{K}^{-1}$ )  
*C<sub>coll</sub>* Dimensionless collector cost  
*c<sub>p</sub>* Ideal gas specific heat at constant pressure ( $\text{kJ kg}^{-1} \text{K}^{-1}$ )  
*f<sub>d</sub>* Internal energy factor for dead mass (dimensionless)  
*f<sub>htf</sub>* Mismatch of heat transfer fluid flow (dimensionless)  
*h* Enthalpy per unit mass of refrigerant ( $\text{kJ kg}^{-1}$ )  
*Δh<sub>ads</sub>* Heat of adsorption,  $u_{\text{ads}}(T) - h_v(T)$  ( $\text{kJ kg}^{-1}$ )  
*m* Mass (kg)  
*P* Pressure (kPa)  
*q* Heat transfer per unit mass of adsorbent ( $\text{kJ kg}^{-1}$ )  
*T* Temperature (K)  
*T<sub>o</sub>* Environment temperature (K)  
*T<sub>w</sub>* Wave temperature,  $(T_b + T_f)/2$  (K)  
*u* Internal energy per unit mass of refrigerant ( $\text{kJ kg}^{-1}$ )  
*X* Adsorption capacity ( $\text{kg}_{\text{ads}}/\text{kg}_{\alpha}$ )  
*y<sub>w</sub>* Dimensionless wave position

#### Subscripts

*ads* Adsorbed or adsorption  
*b* Bed volume behind wave  
*clr* Cooler  
*cold* Cold temperature reservoir  
*cond* Condenser  
*e* Exit flow  
*evap* Evaporator  
*f* Bed volume in front of wave  
*F* Fuel  
*htf* Heat transfer fluid  
*htr* Heater  
*hot* Hot temperature reservoir  
*i* Inlet flow  
*l* Saturated liquid  
*P* Product  
*sat* Saturation  
*v* Vapor (saturated and/or ideal gas)

#### Symbols

$\alpha$  Adsorbent  
 $\beta$  Adsorbent bed shell material

## 2. THERMODYNAMIC MODELS OF IDEAL ADSORPTION CYCLES

A conceptual overview of previously modeled adsorption cycles is presented followed by a more detailed presentation of the thermal wave cycle.

### 2.1. Simple Cycle with No Thermal Regeneration

A schematic for this cycle is shown in Fig. 1 (a). The cycle operates between three temperature reservoirs (TR): (1) hot; (2) environment; and, (3) cold. The cycle receives a *fuel* heat transfer  $q_F$  from the hot TR at  $T_{hot}$  and a *product* heat transfer  $q_P$  from the cold TR at  $T_{cold}$ , and rejects a *loss* heat transfer ( $q_{bed} + q_{cond}$ ) to the environment at  $T_o$ . The condenser contains saturated refrigerant at  $T_o$  and  $P_{cond} = P_{sat}(T_o)$ . The evaporator contains saturated refrigerant at  $T_{cold}$  and  $P_{cond} = P_{sat}(T_{cold})$ . The adsorbent bed contains adsorbed and vapor refrigerant and adsorbent. The dimensionless equilibrium adsorption capacity of the adsorbent ( $X \equiv$  mass of adsorbed refrigerant/mass of adsorbent  $= m_{ads}/m_a$ ) decreases with the adsorbent temperature ( $T_a$ ) and increases with refrigerant vapor pressure ( $P_v$ ); i.e.,  $X = X(T_a, P_v)$ . By alternately bringing the bed into thermal communication with the hot TR and the environment, and mechanical communication with the evaporator and condenser by opening and closing valves, refrigerant is adsorbed from the evaporator at  $P_{evap}$  and desorbed to the condenser at  $P_{cond} > P_{evap}$  as shown in the Clapeyron diagram in Fig. 1 (b). Thus a thermally driven compression process is achieved that replaces the mechanically driven compression process in a typical vapor-compression refrigeration cycle. The total mass of refrigerant evaporated in the evaporator is  $m_a \Delta X_{cycle}$ , where  $\Delta X_{cycle} = X(T_o, P_{evap}) - X(T_{hot}, P_{cond})$ . The cooling provided by one cycle per unit mass of adsorbent is

$$q_P = \frac{Q_P}{m_a} = \Delta X_{cycle} [h_v(T_{evap}) - h_l(T_{cond})] \quad (1)$$

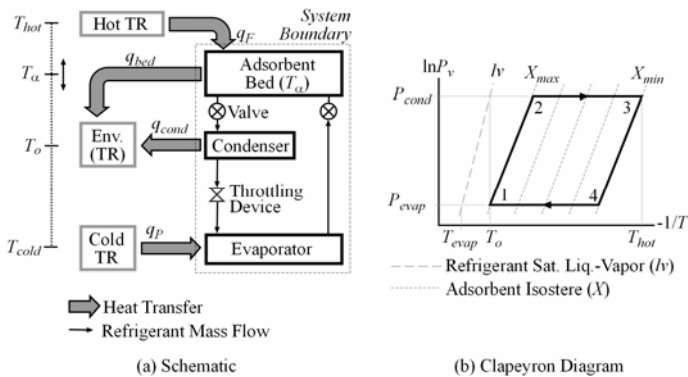


Fig. 1: Schematic and Clapeyron Diagram for an Ideal Simple Adsorption Cycle:  $lv$  = saturated liquid-vapor refrigerant

where  $h_v$  and  $h_l$  are the enthalpies of the saturated vapor and liquid refrigerant.

The thermodynamic performance of this cycle is quantified as  $COP = q_P/q_F$ . Models to predict the thermodynamic limits to the performance of this simple cycle are presented in [5][6].

### 2.2. Simple Cycle with Maximum Thermal Regeneration

Since in the simple cycle presented above the bed is first heated from  $T_o$  to  $T_{hot}$  using  $q_F$  and then cooled from  $T_{hot}$  to  $T_o$  by  $q_{bed}$ , an opportunity to couple these heat transfers in a thermally regenerative process exists. For example, if two beds spaced  $180^\circ$  apart with respect to cycle are used, when one bed is at  $T_o$  and needs to be heated the second is at  $T_{hot}$  and needs to be cooled. If these two beds are brought into thermal contact until they reach thermal equilibrium at some intermediate temperature  $T'$ , the initially cold bed is heated from  $T_o$  to  $T'$  without requiring any  $q_F$ . Consequently  $q_F$  is only required to heat the initially cold bed from  $T'$  to  $T_{hot}$ . Similarly,  $q_{bed}$  is only required to cool the initially hot bed from  $T'$  to  $T_o$ . The maximum temperature to which a bed can be heated using only thermal regeneration increases as the number of thermally interacting beds spaced equally throughout the cycle increases, which in turn decreases  $q_F$  and increases  $COP$ . As the number of beds goes to infinity, the theoretical minimum  $q_F$  and theoretical maximum  $COP$  are defined. Baker and Kaftanoğlu developed models to predict the limiting thermodynamic performance for this infinite bed case [6]. They found that the fraction of  $q_{bed}$  that can be regenerated is very dependent on how well the temperature ranges for desorption ( $T_2$  to  $T_{hot}$ ) and adsorption ( $T_4$  to  $T_o$ ) overlap, which in turn is very dependent on the adsorbent-refrigerant pair choice.

### 2.3. Reversible Cycle

The simple cycle with maximum thermal regeneration still has irreversibilities due to heat transfers across finite temperature differences and throttling. Theoretically these irreversibilities can be fully eliminated through the use of heat pumps, heat engines and turbines. The resulting cycle is thermodynamically equivalent to a Carnot heat engine operating between  $T_{hot}$  and  $T_o$  powering a Carnot refrigeration cycle operating between  $T_o$  and  $T_{cold}$ . The COP for this cycle is  $(1 - T_o/T_{hot}) / (T_o/T_{cold} - 1)$ .

### 2.4. Thermal Wave Cycle

Thermal regeneration can also be achieved using a thermal wave cycle, which is shown schematically in Fig. 2 (a). The condenser, throttling device and evaporator are the same as in the simple cycle, but the adsorbent beds do not follow the simple adsorption cycle. The two beds are thermally connected by a heat transfer fluid (HTF) flowing in a closed loop. This HTF is heated to  $T_{hot}$  in a heater and cooled to  $T_o$  in a cooler.

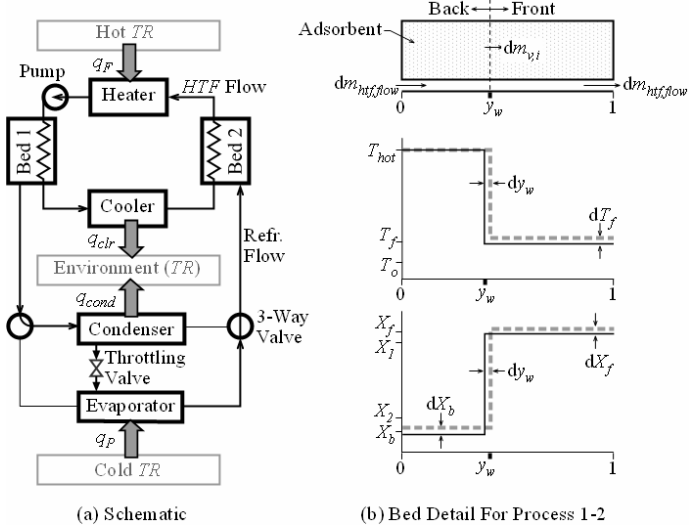


Fig. 2: Ideal Thermal Wave Cycle (a) Schematic and (b) Top-half bed detail

The *HTF* heats and cools the beds via a temperature front that progresses through the bed (i.e., the *thermal wave*). When one bed is heated the other is cooled. The direction of the *HTF* flow is reversible such that the *HTF* can heat or cool either bed.

Conceptually, the adsorbent bed can be imagined as a long concentric-tube heat exchanger with the *HTF* flowing axially through its core and the refrigerant (vapor and adsorbed) and adsorbent in the annulus. The top-half of the bed is shown in Fig. 2(b). A thermodynamically “perfect” system is considered in which the only temperature and pressure gradients are those imposed by the conceptual design of the system. The annulus is completely isobaric at pressure  $P_v$  and is isothermal except for a single step temperature front due to the thermal wave. For a dimensionless bed length of 1, the position of the wave is given by  $0 \leq y_w \leq 1$ . The thermal wave divides the bed into two isothermal volumes, the first behind the wave at  $T_b$  and the second in front of the wave at  $T_f$ . The temperature of the *HTF* entering the bed at  $T_{hf,i}$  fixes  $T_b$  while  $T_f$  fixes the temperature of the *HTF* exiting the bed; i.e.,  $T_b = T_{hf,i}$  and  $T_{hf,e} = T_f$ . The thermal wave results in an adsorption wave that divides the bed into two isosteric volumes at  $X_b$  and  $X_f$ . The masses of adsorbent and adsorbed refrigerant in the back and front systems are

$$m_{\alpha,b} = y_w m_{\alpha} \quad (2)$$

$$m_{\alpha,f} = (1 - y_w) m_{\alpha} \quad (3)$$

$$m_{ads,b} = y_w m_{\alpha} X_b \quad (4)$$

$$m_{ads,f} = (1 - y_w) m_{\alpha} X_f \quad (5)$$

The thermal capacitance of the bed is due to both the *live* refrigerant mass that has both sensible and latent loads, and the *dead* adsorbent, *HTF*, and bed shell material masses that only

have sensible loads. An internal energy factor  $f_d$  for the dead masses is defined as

$$f_d = 1 + \frac{m_{\beta} u_{\beta} + m_{hf} u_{hf}}{m_{\alpha} u_{\alpha}} \quad (6)$$

such that the internal energy of the dead masses is  $f_d m_{\alpha} u_{\alpha}$ . The variables  $m_{\alpha}$  and  $u_{\alpha}$  are fixed by thermodynamic constraints while  $f_d$  is fixed by structural and heat transfer considerations beyond the scope of this research. In the ideal case  $f_d \rightarrow 1$  while in any real case  $f_d > 1$ . The following assumptions are made: (1) all substances in the bed are either incompressible or ideal gases; (2) constant specific heats, which also results in  $f_d$  being constant; (3) the mass of refrigerant in the vapor phase is negligible; and, (4) refrigerant desorbs at  $T_w = (T_b + T_f)/2$ .

The mathematical model is developed for a single bed undergoing a cycle. At state 1  $y_{w,1} = 0$ ,  $T_{b,1} = T_{f,1} = T_o$  and  $P_{v,1} = P_{evap}$  and all valves are closed.

**Process 1→2:** *HTF* at  $T_{hot}$  is circulated through the bed causing the thermal wave to begin advancing. The increase in  $T_{\alpha}$  at the wave causes, in series, refrigerant to desorb,  $P_v$  to increase, and  $X_b$  and  $X_f$  to increase slightly. A mass of the desorbed refrigerant that is sufficient to cause some finite increase in vapor pressure but is negligible relative to the total amount of adsorbed refrigerant in the bed remains in the vapor phase, while the remaining desorbed refrigerant is reabsorbed both behind and in front of the wave. According to the thermally perfect bed model,  $T_b = T_{hot}$ ,  $dT_b = 0$ , the *HTF* exits at  $T_{hf,e} = T_f$  and the heat of adsorption released causes  $dT_f > 0$ . A differential energy analysis is first performed for some unspecified but fixed length of the bed in front of the wave extending from  $y_i$  to 1 (i.e.,  $y_w < y_i < 1$ ). A portion of the vapor desorbed refrigerant enters this part of the bed ( $dm_{v,i}$ ) and becomes adsorbed ( $dm_{ads,i} = dm_{ads,i}$ ). An energy balance yields

$$h_{v,i} dm_{v,i} = f_d d(m_{\alpha,i} u_{\alpha,i}) + d(m_{ads,i} u_{ads,i}) \quad (7)$$

where the *HTF* passes isothermally through the system. This part of the bed is spatially isothermal at  $T_i = T_b$ , and therefore  $u_{\alpha,i} = u_{\alpha,f}$ ,  $u_{ads,i} = u_{ads,f}$  and  $X_i = X_f$ . The masses of adsorbent and adsorbed refrigerant are  $m_{\alpha}(1 - y_i)$  and  $m_{\alpha}(1 - y_i)X_f$ . Both  $y_i$  and  $m_{\alpha}$  are constant,  $X_f$  increases due to adsorption and  $u_{\alpha,f}$  and  $u_{ads,f}$  increase with  $T_f$ . For constant specific heats and noting  $dm_{v,i} = dm_{ads,i} = (1 - y_i)m_{\alpha}dX_f$ , Eqn (7) can be rewritten as

$$\begin{aligned} (f_d c_{\alpha} + X_f c_{ads}) dT_f \\ + [u_{ads}(T_f) - h_v(T_f) + c_p(T_f - T_i)] dX_f = 0 \end{aligned} \quad (8)$$

Eqn (8) is independent of  $y_i$  and therefore applies to the entire front part of the bed. For a system boundary drawn close to the wave, the refrigerant vapor enters at  $T_w$ . Defining an isothermal heat of adsorption as  $\Delta h_{ads} = u_{ads} - h_v$  yields

$$(f_d c_{\alpha} + X_f c_{ads}) dT_f + [\Delta h_{ads} + c_p(T_f - T_w)] dX_f = 0 \quad (9)$$

For any  $P_v$  there is a unique value of  $T_f$  that satisfies Eqn (9).

Since no adsorbent leaves the bed during this process, conservation of refrigerant mass requires

$$y_w m_\alpha X(T_{hot}, P_v) + (1 - y_w) m_\alpha X(T_f, P_v) = m_\alpha X(T_o, P_{evap}) \quad (10)$$

which fixes  $y_w$  once  $T_f$  and  $P_v$  are known.

An energy balance on the entire bed for some differential mass of HTF  $dm_{htf,flow}$  flowing through the bed requires

$$(h_{htf,i} - h_{htf,e}) dm_{htf,flow} = f_d d(m_{\alpha,b} u_{\alpha,b} + m_{\alpha,f} u_{\alpha,f}) + d(m_{ads,b} u_{ads,b} + m_{ads,f} u_{ads,f}) \quad (11)$$

Recall  $T_b = T_{hot} = \text{constant}$  and therefore  $du_{\alpha,b} = du_{ads,b} = 0$ . Also,  $dT_f > 0$  and  $dP_v > 0$ . Due to the movement of the wave and adsorption,  $dm_{\alpha,b} \neq 0$ ,  $dm_{\alpha,f} \neq 0$ ,  $dm_{ads,b} \neq 0$ ,  $dm_{ads,f} \neq 0$ . Conservation of mass requires  $dm_{\alpha,b} + dm_{\alpha,f} = 0$  and  $dm_{ads,b} + dm_{ads,f} = 0$ . Using Eqns (2) through (5)

$$c_{htf} (T_{hot} - T_f) \frac{dm_{htf,flow}}{m_\alpha} = (1 - y_w) (f_d c_\alpha + X_f c_{ads}) dT_f + (T_{hot} - T_f) [(f_d c_\alpha + c_{ads} X_b) dy_w + c_{ads} y_w dX_b] \quad (12)$$

Eqn (12) can be solved for  $dm_{htf,flow}$  once  $P_v$ ,  $T_f$  and  $y_w$  are fixed.

**Process 2→3:** The valve connecting the bed to the condenser is opened. As the wave progresses the desorbed refrigerant leaves the bed and is condensed in the condenser, and the process proceeds isobarically at  $P_{cond}$ . An energy balance on the entire bed requires

$$(h_{htf,i} - h_{htf,e}) dm_{htf,flow} - h_{v,e} dm_{v,e} = f_d d(m_{\alpha,b} u_{\alpha,b} + m_{\alpha,f} u_{\alpha,f}) + d(m_{ads,b} u_{ads,b} + m_{ads,f} u_{ads,f}) \quad (13)$$

Since no refrigerant is being adsorbed behind or in front of the wave,  $dT_f = dT_b = 0$ . Conservation of mass for the adsorbent requires  $dm_{\alpha,b} = -dm_{\alpha,f} = m_\alpha dy_w$ , and conservation of mass for the refrigerant requires  $dm_{v,e} = -dm_{ads,b} - dm_{ads,f}$ , where  $m_{v,e}$  is the refrigerant vapor exiting the bed. The refrigerant is assumed to leave the bed at  $T_w$ , applying Eqns (4) and (5), and integrating yields

$$\frac{c_{htf} (T_{hot} - T_{f,2})}{1 - y_{w,2}} \frac{m_{htf,flow,2-3}}{m_\alpha} = \Delta h_{ads} (X_{b,2} - X_{f,2}) + \left( f_d c_\alpha + \frac{X_{b,2} + X_{f,2}}{2} c_p \right) (T_{hot} - T_{f,2}) \quad (14)$$

which fixes  $m_{htf,flow,2-3}$ . Process 2→3 ends when  $y_w = 1$ , at which point the entire bed is at  $T_{hot}$  and  $P_{cond}$ . Since no refrigerant leaves the bed during  $1 \rightarrow 2$ ,  $\Delta X_{cycle} = X_2 - X_3$ .

**Process 3→4:** The valve connecting the bed to the condenser is closed and the direction of the HTF is reversed such that now the HTF is entering at  $T_o$ . The process is

essentially the reverse of process  $1 \rightarrow 2$ , with refrigerant being desorbed behind and in front of the wave and re-adsorbed at the wave,  $T_b = T_o$ , and described by a set of Eqns parallel to Eqns (7) through (12).

**Process 4→1:** The valve connecting the bed to the evaporator is opened and the wave continues propagating through the bed except now the adsorbed refrigerant is taken from the evaporator. The process is essentially the reverse of process  $2 \rightarrow 3$  and is described by a set of Eqns parallel to Eqns (13) through (14).

**$q_F$  and  $q_{clr}$ :** The total mass of HTF that must flow through the bed for process  $1 \rightarrow 2 \rightarrow 3$  is

$$\frac{m_{htf,flow,1-3}}{m_\alpha} = \int_1^2 \frac{dm_{htf,flow}}{m_\alpha} + \frac{m_{htf,flow,2-3}}{m_\alpha} \quad (15)$$

while that for process  $3 \rightarrow 4 \rightarrow 1$  is

$$\frac{m_{htf,flow,3-1}}{m_\alpha} = \int_3^4 \frac{dm_{htf,flow}}{m_\alpha} + \frac{m_{htf,flow,2-3}}{m_\alpha} \quad (16)$$

For the conditions investigated  $m_{htf,flow,1-3} < m_{htf,flow,3-1}$  due primarily to differences in the way that the adsorption capacity changes for process  $1 \rightarrow 2$  versus  $3 \rightarrow 4$ . This difference in HTF mass flows is termed the *HTF mismatch* and is quantified as

$$f_{htf} = \frac{m_{htf,flow,3-1}}{m_{htf,flow,1-3}} \quad (17)$$

Two HTF circulation models are considered to investigate the effect of  $f_{htf}$  on the cycle's thermodynamic performance limits. The first model is referred to as the *thermal wave cycle with bypass* in which once the HTF mass  $m_{htf,flow,1-3}$  has passed through the heater, the remaining mass  $m_{htf,flow,1-3}(f_{htf} - 1)$  is passed directly from the bed being cooled to the cooler, bypassing the heater and the bed that has been heated. Therefore  $m_{htf,flow,1-3}(f_{htf} - 1)$  is circulated only between the bed being cooled and the cooler and only  $m_{htf,flow,1-3}$  must be heated from  $T_{f,3-4-1}$  to  $T_{hot}$ . The total heat transfer to the heater is then

$$\frac{m_\alpha q_{htf}}{c_{htf}} = \int_3^4 (T_{hot} - T_f) dm_{htf,flow} + (m_{htf,flow,1-3} - m_{htf,flow,3-4})(T_{hot} - T_{f,4}) \quad (18)$$

The total heat transfer from the cooler is

$$\frac{m_\alpha q_{clr}}{c_{htf}} = \int_1^2 (T_f - T_o) dm_{htf,flow} + m_{htf,flow,2-3} (T_{f,2} - T_o) + m_{htf,flow,1-3} (f_{htf} - 1) (T_{f,4} - T_o) \quad (19)$$

The second model is referred to as the *thermal wave cycle without bypass*. In this case, all  $m_{htf,flow,3-1}$  passes through both the heater and the cooler. The total heat transfer to the heater is

$$\frac{m_a q_{hr}}{c_{hf}} = \int_3^4 (T_{hot} - T_f) dm_{hf,flow} + m_{hf,flow,4-1} (T_{hot} - T_{f,4}) \quad (20)$$

After the *HTF* mass  $m_{hf,flow,1-3}(f_{hf} - 1)$  is heated to  $T_{hot}$  in the heater, it passes isothermally through the bed that has already been heated to  $T_{hot}$  before being cooled in the cooler to  $T_o$ . The total heat transfer from the cooler is

$$\frac{m_a q_{ctr}}{c_{hf}} = \int_1^2 (T_f - T_o) dm_{hf,flow} + m_{hf,flow,2-3} (T_{f,2} - T_o) + m_{hf,flow,1-3} (f_{hf} - 1) (T_{hot} - T_o) \quad (21)$$

**Collector Cost Estimate:** Recall that the primary motivation in this line of research is to explore possible cycle enhancements to reduce collector costs through thermal regeneration. To investigate basic trends in collector costs, an ambient temperature of 32 °C and 935 W/m<sup>2</sup> of solar radiation ( $\dot{Q}_s/A$ ) are assumed [5]. The collector area to cooling power ( $\dot{Q}_p$ ) is estimated as

$$\frac{A_{coll}}{\dot{Q}_p} = \frac{1}{(\dot{Q}_s/A) \eta_{coll} COP} \quad (22)$$

The cost and performance of the evacuated tube collector with heat pipe identified as EH-AN2 in [5] are used in this analysis to yield a collector cost estimate per unit cooling power:

$$\frac{Collector\ Cost}{\dot{Q}_p} = \frac{A_{coll}}{\dot{Q}_p} \times \frac{Collector\ Cost}{Area} \quad (23)$$

### 3. ANALYSIS

A centered-averaged finite-difference solution scheme is used to approximate process 1→2 and 3→4. A fixed vapor pressure step size  $\Delta P_v$  is assumed such that  $P'_v = P_v + \Delta P_v$  and  $T_f$ ,  $y_w$ ,  $X_b$ , and  $X_f = T_f(P'_v)$  are estimated at each step using the initial condition  $T_f = T_o$  and  $y_w = 0$ .

Data for zeolite NaX are used for which  $c_a = 0.836$  kJ/kg·K and  $\Delta h_{ads} = -3200$  kJ/kg [7]. A constant heat of adsorption is assumed and since the vapor is modeled as an ideal gas, the definition  $\Delta h_{ads} \equiv u_{ads} - h_v$  requires  $c_{ads} = c_p$  for thermodynamic consistency [8]. The temperatures  $T_{cold}$  and  $T_o$  are fixed at 10 and 30°C and simulations are run for  $T_{hot}$  ranging from 110°C to 200°C. Using previously developed models [5][6], simulations were also run using these inputs for the simple cycle without thermal regeneration and maximum thermal regeneration (infinite beds), and a reversible cycle.

A global energy balance for a system boundary drawn around the heater, cooler, and two beds is used to test the overall thermodynamic consistency of the model with respect to energy conservation. The energy error is defined as

$$Error = \frac{q_F - q_{ctr} - \Delta X_{cycle} c_p (T_{w,2} - T_{evap})}{q_F} \quad (24)$$

### 4. RESULTS

The absolute error in the global energy balance for all conditions explored was < 0.1%, which is considered sufficiently small to give confidence to the solution methodology and results. The *HTF* mismatch  $f_d$  is relatively constant at ~1.10, decreasing non-linearly with  $T_{hot}$  as 1.17 at 110 °C, 1.10 at 150 °C and 1.06 at 200 °C.

The *COP* for the limiting case of  $f_d = 1$  for the thermal wave cycles with and without bypass, the simple cycle without thermal regeneration and maximum thermal regeneration, and the reversible cycle are presented as a function of  $T_{hot}$  in Fig. 3. Significantly, for the conditions explored the thermal wave cycle has a higher limiting *COP* than the simple cycle with maximum thermal regeneration. The reason is that, relative to the simple cycle, the processes 1→2 and 3→4 condition the adsorbent in front of the wave such that  $T_{2,wave} < T_{2,simple}$  and  $T_{4,wave} > T_{4,simple}$ . In [9] for the simple cycle with maximum thermal regeneration, Baker and Kaftanoğlu defined the temperature interval  $T_2$  to  $T_4$  over which both the heated and cooled beds have sensible loads as a *paired sorption regime* and demonstrated that the size of this regime strongly impacts the sensible loads that can be thermally regenerated. Thus the thermal wave cycle has a *larger paired sorption regime* for this adsorbent refrigerant pair, which leads to a larger opportunity for thermal regeneration.

The collector costs per unit power are presented as a function of  $T_{hot}$  in Fig. 4. The competing effects of *COP* increasing with  $T_{hot}$  and the collector efficiency decreasing with  $T_{hot}$  tend to offset one another for the regenerative cycles. The differences in the collector costs for the three thermally regenerative ideal cycles are most likely small relative to the deviation of any actual cycle's performance from these thermodynamic limits.

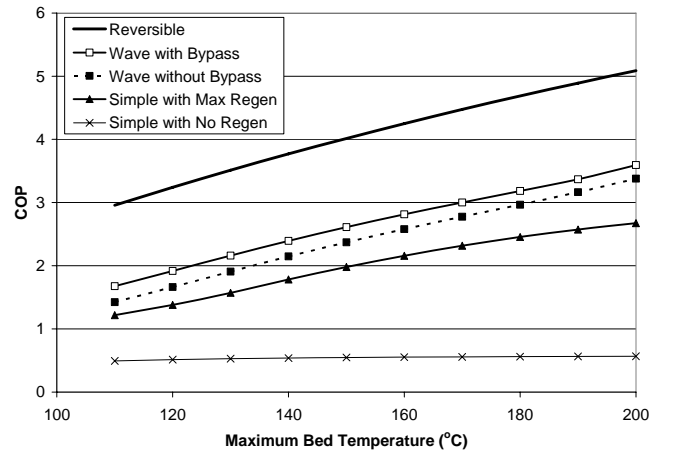


Fig. 3: Limiting *COP* versus maximum bed temperature ( $T_{hot}$ )

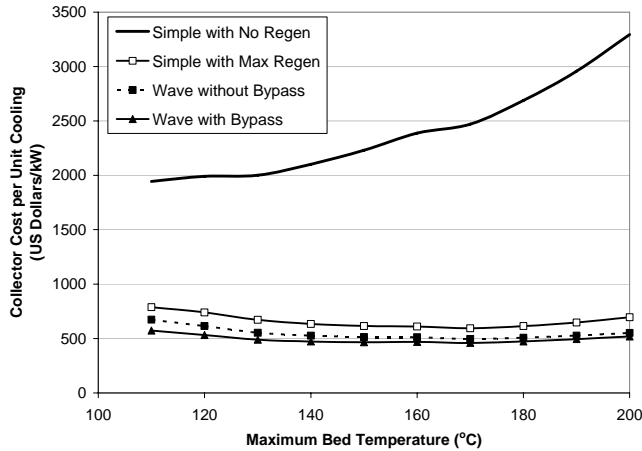


Fig. 4: Collector cost per unit cooling power as a function of maximum bed temperature

## 5. CONCLUSIONS

For the thermodynamic limiting conditions explored, implementing thermal regeneration in an adsorption cooling cycle has the potential to reduce the driving heat transfer per unit cooling by approximately 70%. Reducing the driving heat transfer is particularly significant for solar thermal powered systems as collector costs are predicted to dominate the system's capital costs. The thermodynamic limits to the performance of the thermal wave cycle (COP's 2.5~3.5) are predicted to be higher than that for a system with an infinite number of beds following the simple adsorption cycle and utilizing maximum thermal regeneration between the beds (COP's 1.25~2.75). The minimum required heat transfer fluid mass flow through the bed being cooled is approximately 1.1 times that being heated. Utilizing a by-pass loop to pass the minimum mass of heat transfer fluid through the heater results in ~ 10% increase in COP. Collector costs for the thermally regenerative costs tend to be a weak function of  $T_{hot}$ .

Differences in the limiting COP's for the three regenerative cycles considered are likely to be small relative to the practical barriers to achieve any of these cycles. Therefore choosing the "best" cycle with thermal regeneration among these three will likely be based on heat and mass transfer characteristics that cause the actual performance to deviate from these limits and the cost to manufacture the adsorption cooling system, and not based on thermodynamic performance limits. Specifically, the deviation of the performance for an actual thermal wave cycle from the ideal cycle considered here will depend heavily on several factors including the following: (1) how closely the bed can be made to approximate two spatially isothermal volumes separated by a single, discrete step temperature front (which will depend on the heat transfer characteristics of the bed); (2) how closely the bed can be made to approximate a single isobaric volume, which will depend on the mass transfer characteristics inside the bed.

Future work will focus on moving these ideal cycles with thermal regeneration into designs that are more practical to implement. The effects of transient ambient and building loads will be explored using simulations. Experiments to measure the adsorption characteristics of a natural zeolite found in Turkey have started. The results from these experiments will be used in these models to guide efforts to design a commercially viable solar-thermal powered adsorption cooling system.

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