Reducing Unwanted Thermal Gains during the Cooling Season for a Solar Heat Pipe System

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ABSTRACT

The heat pipe augmented solar heating system significantly reduces heating loads relative to other conventional passive space heating systems. However, thermal gains from the system during the cooling season increase cooling loads and tend to increase overall space conditioning loads. The effectiveness of several design modifications and control strategies for the heat pipe wall to reduce unwanted gains was investigated. Computer algorithms were used to simulate four different unwanted gains reduction mechanisms: shading, an opaque cover, a mechanical valve, and switching the elevations of the evaporator and condenser sections to provide heat transfer *out* of the room during the cooling season. For each mechanism, three different control strategies were evaluated: 1) *Seasonal control*, for which the prescribed mechanism was deployed at the beginning and removed at the end of the cooling season, 2) *ambient temperature-based control*, for which the mechanism was deployed if the forecast for the next hour was greater than 18.3°C (65°F), and 3) *room temperature-based control*, for which the mechanism was deployed if auxiliary cooling was required for the previous hour. For the seasonal strategy, a season determination (SD) parameter was defined and results suggested that SD may be a ‘universal’ parameter that can be applied across a range of climates for quick cooling season assessment. The heat pipe system performed best with ambient temperature-based control strategy, and the cover and valve were the best single mechanisms. Experiments on a full-scale heat pipe wall prototype using valve and cover control mechanisms confirmed significant reductions in unwanted thermal gains with little to no increase in room temperature during the testing periods.

NOMENCLATURE

Variables

A_c	heat pipe system absorber area [m ²]		simulations (see Table 1 for more detail) [W/m ² K]
A_{cndsr}	surface area of the condenser [m ²]		
A_s	total heat transfer surface area of the storage tank walls [m ²]	k_{41}	conductance between nodes 4 and 1 in the thermal network used in computer simulations (see Table 1 for more detail) [W/m ² K]
c_p	specific heat for water used for thermal storage [4186 J/kg·K]		
cf	correction factor from bench-scale experiments used to correlate radiation measured at each corner of the collector to mean radiation across the collector	k_{45}	conductance between nodes 4 and 5 in the thermal network used in computer simulations (see Table 1 for more detail) [W/m ² K]
D	outside diameter of the heat pipes [0.029 m]	k_{56}	conductance between nodes 5 and 6 in the thermal network used in computer simulations (see Table 1 for more detail) [W/m ² K]
g	acceleration due to gravity [9.81 m/s ²]		
h_{cndsr}	convection coefficient for the exposed condenser to room air [W/m ² K]	k_{67}	conductance between nodes 6 and 7 in the thermal network used in computer simulations (see Table 1 for more detail) [W/m ² K]
h_{rad}	heat transfer coefficient for radiation between the storage tank wall and room envelope [W/m ² K]		
$h_{rad,cndsr}$	heat transfer coefficient for radiation between the exposed condenser and room envelope [W/m ² K]	k_{71}	conductance between nodes 7 and 1 in the thermal network used in computer simulations (see Table 1 for more detail) [W/m ² K]
$h_{s/r}$	convection coefficient for water (h_s) or room air (h_r), respectively [W/m ² K]	k_{ij}	heat transfer coefficient between nodes i and j within a thermal network [W/m ² K]
I	incident solar radiation on the collector [W/m ²]	k_{cndsr}	total thermal heat transfer from the exposed condenser to room air [W/K]
k_{12}	conductance between nodes 1 and 2 in the thermal network used in computer simulations (see Table 1 for more detail) [W/m ² K]	k_{cond}	conductance through the storage tank wall [W/K]
k_{23}	conductance between nodes 2 and 3 in the thermal network used in computer simulations (see Table 3 for more detail) [W/m ² K]	$k_{conv,cndsr}$	convection between the exposed condenser and the room air [W/K]
k_{23*}	conductance between nodes 2 and 3* in the thermal network used in computer simulations (see Table 1 for more detail) [W/m ² K]	$k_{r,conv}$	convection between the outside of the tank wall and the room air [W/K]
k_{3*6}	conductance between nodes 3* and 6 in the thermal network used in computer simulations (see Table 1 for more detail) [W/m ² K]	k_{rad}	radiative heat transfer between the outside of the tank wall and the room envelope [W/K]
k_{34}	conductance between nodes 3 and 4 in the thermal network used in computer	$k_{rad,cndsr}$	radiative heat transfer between the exposed condenser and the room envelope [W/K]
		$k_{s,conv}$	convection between storage water and the inside tank wall [K/W]
		k_{sr}	total heat transfer from the thermal storage to room air [K/W]
		L	storage tank wall height [1.52 m]

m_i	mass of node i within a thermal network [J/K]	$\dot{Q}_u$	useful thermal gains for the system; transferred from the heat pipes to the thermal storage and room [W]
m_{bs}	mass of the water in the thermal storage tank for the bench-scale experiments [kg]	R_b	ratio of beam irradiation on a tilted vs. horizontal surface
m_{fs}	mass of the water in the thermal storage tanks for the full-scale prototype [kg]	Ra_D	Rayleigh number for free convection from a heated tube
Nu_D	Nusselt number for free convection from a heated tube	$Ra_{s/r}$	Rayleigh number for water (Ra_s) or air (Ra_r)
$Nu_{s/r}$	Nusselt number for water (Nu_s) or room air (Nu_r), respectively	S	absorbed insolation received by the collector [W/m ²]
$Pr_{s/r}$	Prandtl number for water (Pr_s) or room air (Pr_r), respectively	S_i	absorbed insolation received by node i within a thermal network [W/m ²]
$q_{a,aux}$	simulated annual auxiliary heating load per unit collector area [kWh/m ²]	S_{io}	absorbed insolation received by node i , for the previous time step, within a thermal network [W/m ²]
$q_{a,cl}$	simulated annual space cooling load per unit collector area [kWh/m ²]	SD	season determination ratio
$q_{a,hl}$	simulated annual space heating load per unit collector area [kWh/m ²]	th	storage tank wall thickness [3.18 mm]
$q_{a,sw}$	simulated annual heat transfer out of the heat pipe system per unit collector area [kWh/m ²]	T_1	temperature of node 1 for the thermal network of the heat pipe used for computer simulations [K]
$q_{a,uwg}$	simulated annual unwanted thermal gains to storage and room, per unit collector area [kWh/m ²]	T_2	temperature of node 2 for the thermal network of the heat pipe used for computer simulations [K]
$q_{m,hl}$	simulated monthly space heating load per unit collector area [kWh/m ²]	T_3	temperature of node 3 for the thermal network of the heat pipe used for computer simulations [K]
$q_{m,uwg}$	simulated monthly unwanted thermal gains to storage and room, per unit collector area [kWh/m ²]	T_{3*}	temperature of node 3* for the thermal network of the heat pipe used for computer simulations [K]
q_{total}	simulated annual (total) space conditioning load per unit collector area [kWh/m ²]	T_4	temperature of node 4 for the thermal network of the heat pipe used for computer simulations [K]
$\dot{q}_{ij}$	heat transfer rate between arbitrary nodes i and j within a thermal network [W/m ²]	T_5	temperature of node 5 for the thermal network of the heat pipe used for computer simulations [K]
$\dot{Q}_{cndsr}$	rate of energy (power) transferred from the exposed condenser to the room [W]	T_6	temperature of node 6 for the thermal network of the heat pipe used for computer simulations [K]
$\dot{Q}_s$	rate of increase of energy (power) gained by the system thermal mass storage (water) over a one-hour interval [W]	T_7	temperature of node 7 for the thermal network of the heat pipe used for computer simulations [K]
$\dot{Q}_{sr}$	rate of energy (power) transferred from the storage tanks to the room [W]	T_a	ambient temperature [°C, K]
		T_b	base temperature used for calculating annual heating and cooling loads in computer simulations [65°F]
		T_{cndsr}	temperature of the exposed condenser [°C, K]

T_i	temperature of node i within a thermal network [K]	μ	uncertainty for error propagation
T_{io}	temperature of node i , for the previous time step, within a thermal network [K]	ν	kinematic viscosity [m ² /s]
T_j	temperature of node j , for the previous time step, within a thermal network [K]	σ	Stefan-Boltzmann constant [5.67x10 ⁻⁸ W/m ² K ⁴]
T_{jo}	temperature of node j within a thermal network [K]		<i>Superscript</i>
T_r	hourly average temperature of the room [°C, K]	+	only positive values of the argument in parenthesis are summed
T_s	hourly average temperature of the storage tank water [°C, K]		
T_w	storage tank wall temperature [°C, K]		
α	thermal diffusivity [m ² /s]		
β	volumetric thermal expansion coefficient [K ⁻¹]		
ε	radiative emissivity of the storage tank wall [0.95]		
ε_{cndsr}	radiative emissivity of the copper condenser [0.02]		
ΔT_{cndsr}	temperature difference between exposed condenser and room [°C, K]		
ΔT_s	storage water temperature change used for error propagation [°C, K]		
$\Delta T_{s,bs}$	storage water temperature change, for the bench-scale experiment, over a certain time interval [°C, K]		
$\Delta T_{s,fs}$	storage water temperature change, for the full-scale prototype, over a one-hour interval [°C, K]		
ΔT_{sr}	temperature difference between storage room [°C, K]		
ΔT_{sw}	temperature difference between storage water and tank wall [°C, K]		
ΔT_{wr}	temperature difference between tank wall and room air [°C, K]		
Δt	time interval		
η	thermal efficiency for the heat pipe system		
η_{sim}	simulated thermal efficiency for the heat pipe system		
$K_{\sigma/r}$	thermal conductivity for water (K_s) or room air (K_r), respectively [W/m*K]		
K_w	thermal conductivity of the storage tank wall [0.5 W/m*K]		

1. INTRODUCTION

The heat pipe augmented passive solar space heating system greatly improves heating season performance relative to conventional passive space heating systems [Albanese et al. 2012]. The heat pipes consist of three main components (Figure 1): the evaporator, adiabatic section (shown as the “vapor” section in Fig. 1), and condenser. The evaporator is located on the building exterior and is connected to an absorber coated with a selective surface. As the evaporator heats up during insolation, a two-phase fluid within the evaporator boils and rises through the adiabatic section of the heat pipe, which transports the hot, gaseous fluid through the insulated building wall. The fluid next enters the cooler condenser, located in a thermal storage tank within the building interior. Heat is transferred to thermal storage, condensing the two-phase fluid back into liquid, which returns to the evaporator via gravity assistance. During periods of little to no insolation, when the evaporator is cooler than the condenser, the liquid fluid remains in the evaporator and no heat transfer takes place within the heat pipe. Utilizing this thermal diode effect results in high levels of heat transfer into the space and little heat transfer out of the space. Additionally, two-phase heat transfer allows the heat pipe to operate with small temperature gradients throughout and enables the system to operate with small elevation differences between the evaporator and condenser sections so that the entire system can be installed on a south-facing wall.

Figure 1. Schematic of heat pipe operation.

In addition to simulating annual performance of heat pipe systems, including solar fraction and heating loads for various U.S. locations, Albanese et al. [2012] also constructed an experimental bench-scale model to conduct a parametric performance study and determine thermal efficiency of the system (under ideal conditions). This single unit achieved an average system thermal efficiency of 85.1%. A full-scale prototype was subsequently constructed, retrofitted onto a classroom wall on campus, and tested (under less-than-ideal conditions) by Robinson et al. [2013]. The system consisting of five heat pipe units achieved a maximum daily peak thermal efficiency of 83.7% and an average daily peak thermal efficiency of 63.4%, while thermal storage delivered heat to the room throughout the testing period, including a sequence of four consecutive cloudy days.

Robinson & Sharp [2014] implemented design modifications to the Robinson et al. model and constructed a new prototype (Figure 2). Key features of the new design included 1) a copper absorber with reduced thermal resistance between the absorber and evaporator compared to the previous model that used aluminum absorbers epoxy-bonded to copper evaporators, 2) thicker

solar wall insulation and a rubber adiabatic section that reduced thermal losses from storage to outdoors, and 3) one condenser exposed directly to room air that significantly improved early-morning heating of the room and overall solar fraction. This new model and the previous model were installed in identical 12' by 12' rooms of a passive solar test facility (PSTF), and comparative testing was performed during January and February of 2013. Average daily peak efficiency for the previous system under the improved conditions provided by the PSTF was 80.7%, while an efficiency of 85.0% was achieved by the new system. During the testing period, the new model also achieved average storage and room temperatures that were 13.4% and 24.6%, respectively, higher than those of the previous model.

Figure 2. Schematic of the solar heat pipe system highlighting design modifications relative to the previous model.

While the overwhelming focus has been on performance during the heating season, previous investigators have recognized the potential for passive systems to overheat during the cooling season, which impacts the total annual energy required to both heat and cool the building

[Zaheer-Uddin 1989, Athienitis & Ramadan 1999, Kummert et al. 2000, Raicu et al. 2002]. Features to provide cooling from systems designed primarily for heating have been studied [Ghrab-Morcos et al. 1993, Gan 1998, Mihalakakou 2002, Bataineh & Fayez 2011]. Similar to other passive systems, the solar heat pipe wall can generate thermal gains during the cooling season that may increase the cooling load, thus offsetting energy savings achieved during the heating season. The objective of this study was to compare, using computer simulations, the effectiveness of four different mechanisms to reduce summertime solar gains from the heat pipe system, as well as three control strategies defining when the mechanisms are employed. Furthermore, mechanisms that simulations suggested were effective options for cooling season reductions were implemented into the new prototype installed in the PSTF [Robinson & Sharp 2014] and the system was tested during the cooling season (August and September, 2013) in Louisville, KY.

2. METHODS

2.1 Computer Modeling

Simulations were conducted using a thermal network model for the heat pipe system adapted and revised from previous investigators [Corliss 1979, Susheela & Sharp 2001, Albanese et al. 2012, Robinson & Sharp 2014]. The thermal network for the new heat pipe design that was tested for this study is shown in Figure 3. Descriptions for each heat transfer coefficient in Figure 3 are shown in Table 1.

Figure 3. Thermal network for the heat pipe assisted solar wall.

Table 1. Description of thermal heat transfer coefficient values used for heat pipe wall thermal network models.

Heat Transfer Coefficient	Description	Value (W/m ² K) [minimum/maximum values given for temperature- dependent coefficients]
k_{71}	Overall collector loss coefficient	0.45 / 3.72
k_{12}	Absorber fin conduction and convection to the working fluid	116.77
k_{23} (heating mode)	Two-phase heat transfer from evaporator to condenser when $T_1 > T_4$ for condensers in storage; continuous conduction along heat pipe wall	2.43×10^{-4} / 394.44
k_{23*} (heating mode)	Two-phase heat transfer from evaporator to condenser when $T_1 > T_6$ for condenser in room air; continuous conduction along heat pipe wall	6.07×10^{-5} / 61.64
k_{23} (switched mode)	Two-phase heat transfer from condenser to evaporator when $T_4 > T_1$ for condensers in storage; continuous conduction along heat pipe wall	2.43×10^{-4} / 394.44
k_{23*} (switched mode)	Two-phase heat transfer from condenser to evaporator when $T_6 > T_1$ for condenser in room air; continuous conduction along heat pipe wall	6.07×10^{-5} / 61.64
k_{41}	Conduction through insulated wall	0.16
k_{34}	Convection between condenser and	2.53 / 111.96

	storage water	
k_{3*6}	Convection between condenser and room air	15.73 / 75.76
k_{45}	Conduction through water tank wall	286.83
k_{56}	Convection between tank wall and room	10.29 / 15.62
k_{67}	Building Load-to-Collector Ratio (LCR)	10

Between nodes i and j , the heat transfer rate $\dot{q}_{ij}$ per unit of collector area is given by

$$\dot{q}_{ij} = k_{ij} (T_j - T_i) \quad (1)$$

where k_{ij} is the heat transfer coefficient between nodal temperatures T_j and T_i , respectively. For the i^{th} node, the energy balance equation is

$$m_i c_{p,i} \frac{dT_i}{dt} = \sum_j [k_{ij} (T_j - T_i)] + S_i \quad (2)$$

where m_i is the mass (per unit collector area), $c_{p,i}$ is the specific heat, S_i is the solar power received per unit collector area (node 1 from Fig. 3), and t is time. Using a central difference discretization scheme over time step Δt , equation 2 becomes

$$\left(\frac{2m_i c_{p,i}}{\Delta t} + \sum_j k_{ij} \right) T_i - \sum_j (k_{ij} T_j) = \frac{2m_i c_{p,i} T_{i0}}{\Delta t} + \sum_j [k_{ij} (T_{j0} - T_{i0})] + (S_i + S_{i0}) \quad (3)$$

where the previous time step is denoted by the zero subscript. Using MATLAB software and Typical Meteorological Year weather data (TMY3), all nodes were simultaneously solved as a function of time from a set of initial temperatures. Iterations were used for heat transfer coefficients dependent on nodal temperatures. Key baseline parameters included a Load to Collector Ratio (the ratio of the UA value for the nonsolar part of the building to the collector area; heat transfer coefficient k_{67} in Fig. 3 and Table 1) $LCR = 10 \text{ W/m}^2\text{K}$, and a defined room comfort range of 18.3 to 23.9°C (65 to 75°F). To simulate auxiliary heating and cooling, room

nodal temperatures were restricted within the room comfort range. Two additional room comfort ranges – 20.0 to 22.2°C (68 to 72°F) and 20.6 to 21.7°C (69 to 71°F) – were tested to study the effects that room comfort limits have on annual space conditioning loads. To represent a range of climates, including available insolation and seasonal temperatures, simulations were conducted for four different locations: Albuquerque, NM (35.04°N, 106.62°W), Louisville, KY (38.18°N, 85.73°W), Rock Springs, WY (41.46°N, 109.44°W), and Madison, WI (43.13°N, 89.33°W).

Four different unwanted gains reduction mechanisms were simulated:

- Shading – Beam insolation is eliminated, simulating an overhang above the collector (Fig. 4). Diffuse and ground reflected insolation is still received by the collector.
- Cover – All insolation is eliminated, simulating an opaque cover over the collector (Fig. 5).
- Valve – Phase change heat transfer in the heat pipe is turned ‘on’ or ‘off,’ simulating a valve in the adiabatic section of the heat pipe (Fig. 6).
- Switching – The direction of heat transfer in the heat pipe is reversed, simulating the switching of the elevations of the evaporator and condenser sections of the heat pipe (Fig. 7). Heat is thus transferred *out* of thermal storage when the absorber is cooler than storage, which tends to occur only during nighttime and cloudy days. A system design in which the evaporator and condenser sections of the heat pipe are leveled would make this mechanism more convenient to employ, thus a bench scale experiment with leveled evaporator and condenser sections was performed to examine the effects on heat pipe performance (Section 2.2).

Figure 4. Shading (blocking of beam radiation) to reduce unwanted gains.

Figure 5. Opaque cover (blocking all solar radiation) to reduce unwanted gains.

Figure 6. Valve to reduce unwanted gains.

Figure 7. Switching to reduce unwanted gains.

For each mechanism, three different control strategies were evaluated:

- Seasonal Control – The mechanism is employed at the beginning of the cooling season and removed at the end of the cooling season. For the seasonal strategy, a parameter was derived for determining the optimal months in which the unwanted gains reduction mechanism should be deployed to minimize overall space conditioning loads.
- Ambient Temperature-Based Control – The mechanism is employed if the forecast for the *next* hour (based on TMY3 weather data) is greater than 18.3°C (65°F).
- Room Temperature-Based Control – The mechanism is employed if the room temperature for the *previous* hour reached the upper comfort limit of (23.9°C / 75°F). At the beginning of the current hour, control mechanism employment is dependent on the room temperature at the conclusion of the previous hour, thus this strategy is similar to a typical thermostat.

Solar energy absorbed by the collector is given by [Duffie and Beckman, 2013]

$$S = (I_b + I_d A_i) R_b (\tau\alpha)_b + I_d (1 - A_i) (\tau\alpha)_d \left(\frac{1 + \cos\beta}{2} \right) \left[1 + f \sin^3 \left(\frac{\beta}{2} \right) \right] + I \rho_g (\tau\alpha)_g \left(\frac{1 - \cos\beta}{2} \right) \quad (4)$$

where the first group of terms on the right hand side of the equation represents the direct (beam) component of solar irradiation, the next group of terms represents the diffuse component, and the final group of terms represents the ground reflected component. Unwanted gain reduction mechanisms were simulated as follows:

- Shading – the beam component of Equation 4 was removed.
- Cover – Equation 4 was set equal to zero.
- Valve – Heat transfer coefficients k_{23} and k_{23*} were reduced to only conduction along the pipe wall, regardless of temperature.
- Switching – k_{23} was set equal to the thermal conductance along the heat pipe wall when the

outdoor end of the heat pipe (the evaporator in the normal mode) is hotter than the indoor end, $T_2 > T_3$. When $T_3 > T_2$, k_{23} the heat transfer coefficient for two-phase flow within the pipe was added. k_{23*} was set similarly, except that the condenser temperature is T_{3*} rather than T_3 .

Simulations calculated annual heating and cooling load and the total annual space conditioning load resulting from employing each individual mechanism, and feasible combinations of each, using each respective control strategy.

Annual heating load per unit collector area is

$$q_{a,hl} = \sum_{year} k_{67} (T_b - T_7)^+ \Delta t \quad (5)$$

and annual cooling load per unit collector area is

$$q_{a,cl} = \sum_{year} k_{67} (T_7 - T_b)^+ \Delta t \quad (6)$$

where T_7 is ambient temperature (from TMY3 data) and T_b was the commonly used base temperature of 18.33°C (65°F). $q_{a,cl}$ is equivalent to annual auxiliary cooling. Annual auxiliary heating load per unit collector area is

$$q_{a,aux} = \sum_{year} [k_{67} (T_6 - T_7) - k_{56} (T_5 - T_6) - k_{3*6} (T_{3*} - T_6)]^+ \Delta t \quad (7)$$

whenever ambient temperature was less than room temperature ($T_7 < T_6$), and room temperature dropped to the lower comfort limit ($T_6 = 18.33^\circ\text{C}$). The second and third term within the summation represents the contribution from the thermal storage and exposed condenser, respectively, to serving the heating load. The first term is different from the heating load (equation 5) in that room temperature T_6 may exceed T_b .

For assessment of the cooling capabilities of the heat pipe system using the switching mechanism, annual heat transfer *out* of the system, per unit collector area, is

$$q_{a,sw} = \sum_{year} \left[k_{34} (T_4 - T_3) + k_{3*6} (T_6 - T_{3*}) \right]^+ \Delta t \quad (8)$$

whenever the system is in cooling mode (all summer for seasonal control, when ambient temperature is greater than 18.3°C (65°F) for ambient control, and when room temp is limited to 23.9°C (75°F) for room control.) Annual unwanted thermal gains per unit collector area are

$$q_{a,uwg} = \sum_{year} \left[k_{56} (T_5 - T_6) + k_{3*6} (T_{3*} - T_6) \right]^+ \Delta t \quad (9)$$

whenever room temperature reached the upper comfort limit, and ambient temperature is greater than room temperature.

For the seasonal strategy, a parameter that defines the cooling season was also developed. The first step of this method involved conducting simulations in which an opaque cover was added to the heat pipe system for every feasible sequence of months that could represent the cooling season. The best cooling season definition was that which minimizes total annual auxiliary energy use

$$q_{total} = q_{a,aux} + q_{a,cl} + q_{a,uwg} \quad (10)$$

The optimal cooling season was then compared to a new parameter, the season determination (*SD*) ratio

$$SD = \frac{q_{m,uwg}}{q_{m,hl}} \quad (11)$$

where the calculated unwanted gains for a respective month is

$$q_{m,uwg} = \sum_{month} [k_{56}(T_5 - T_6) + k_{3*6}(T_{3*} - T_6)]^+ \Delta t \quad (12)$$

and heating load for a month is

$$q_{m,hl} = \sum_{month} k_{67}(T_b - T_7)^+ \Delta t \quad (13)$$

SD was calculated for each month of the year for the four initial locations used in simulations (Albuquerque, Louisville, Rock Springs and Madison), and four additional locations: Boston, MA, Chicago, IL, Denver, CO, and Seattle, WA. These locations were added to evaluate repeatability and validity of *SD* for identifying the optimal cooling season months, thereby reducing the number of simulations required to predict performance for systems using the seasonal control strategy.

Finally, algorithms were created to extract the system thermal efficiency for comparison with measured thermal efficiencies obtained for the full-scale prototype [Robinson & Sharp, 2014]. In the same manner as the experimental analysis conducted for the prototype, efficiency values were calculated from data collected for each hour of every day for January and February. Any days (during this period) in which simulated efficiencies were negative between 9am and 5pm signified cloudy conditions and were discarded from the analysis. Simulated efficiency was calculated using

$$\eta_{sim} = \frac{\frac{mc_p}{\Delta t} (T_4 - T_{4,0}) + k_{56}(T_5 - T_6) + k_{3*6}(T_{3*} - T_6)}{I} \quad (14)$$

where $T_{4,0}$ is the water temperature for the previous time step and I is the incident radiation on the vertical collector per unit of aperture area.

2.2 Experimental Setup

2.2.1 Bench-scale Experiments

The bench-scale model (Fig. 8) was identical to the one used by Albanese et al. [2012], with one key exception: thicker insulation was added around the storage tank to reduce thermal losses from the tank to ambient air.

Figure 8. Schematic of model used for bench-scale experiments.

Important features of the bench-scale experiment(s) included:

- Three metal halide lamps, that approximate the solar spectrum, to simulate insolation on the collector
- One 1/8 inch thick low-iron glass cover with an aperture area of approximately 0.45 m^2
- One inch diameter copper heat pipe – evaporator end attached to absorber, condenser end immersed in water, insulated copper adiabatic section
- Two copper absorber plate fins, resulting in a total absorber area of approximately 0.31 m^2 , each soldered to opposite center-lines of the copper evaporator pipe; both fins and the exposed half of the evaporator were plated with a black chrome selective surface

- Insulated wall, constructed of 0.20 m (8 inches) thick R-20 insulation framed by 0.01m-thick (1/2 inch) plywood, between the absorber/evaporator and tank
- Heat pipe working fluid was R124 refrigerant filled to 120% of the evaporator volume
- A 189.3 L (50-gallon) plastic thermal storage water tank

Experimental configurations included baseline experiments with each section of the heat pipe sloped at 5 degrees and experiments with leveled evaporator and condenser sections (for accommodating the switching mechanism). A minimum of three separate runs for each case were conducted to ensure repeatability.

2.2.2 Full-scale Prototype

The new model installed in the south-facing wall of the PSTF [Robinson & Sharp 2014] (Figures 9 and 10) was used to test the effectiveness of control mechanisms in reducing unwanted thermal gains to storage and to the room. The model consisted of one module with five individual one-inch nominal diameter heat pipes. Absorbers for the collector were coated with a black chrome selective surface, and collector dimensions consisted of 2.6 m² of aperture area and 2.4 m² of absorber area. Four of the five heat pipe condensers were located in 64.4 L (17 gallons) thermal storage tanks, while the fifth condenser was exposed directly to room air. The PSTF was made of structural insulated panels (SIPs) consisting of 12" thick panels with an R-value of 7.93 m²*K/W (45 ft²F*hr/Btu) for the floor and walls, and 16" for the roof with an insulating value of 11.10 m²*K/W (R-63). The building incorporated a clerestory section and was constructed so that the south-facing wall and clerestory were oriented due south. During experimentation, the clerestory windows were covered with an opaque material to eliminate solar gains, so that the heat pipe system was the only source of heat added to the room. No auxiliary heating or cooling was used.

Figure 9. External view of the new model (right side) for the heat pipe augmented passive solar space heating prototype.

Figure 10. Internal view of the new model for the heat pipe augmented passive solar space heating prototype.

The cover mechanism consisted of an opaque white vinyl cover over the collector. For the valve mechanism, a valve was installed in each heat pipe between the adiabatic and condenser sections. Closing the valve isolated the adiabatic section from the condenser section. Experiments using individual mechanisms and both simultaneously were conducted during August 15 -24, August 26 – September 2, and September 4 -11 of 2013. Thermal gains to storage and room were then compared to gains that would have resulted using no control mechanisms, estimated using a procedure described in Section 2.4.2.

2.3 Instrumentation

2.3.1 Bench-scale Experiments

Instrumentation for the bench-scale experiments included 24 calibrated T-type thermocouples, with eight located on the absorber, three on the evaporator section, two on the adiabatic section, three on the condenser section, and eight placed within the tank. Data acquisition was accomplished using a National Instruments chassis in conjunction with an SCXI 1600 16 bit digitizer and an SCXI 1102B isolation amplifier with an SCXI 1303 thermocouple module, and LabVIEW software was used to log the data at a sampling rate of 0.01667 Hz (once per minute). Additionally, four pyranometers were mounted, one at each corner of the collector, to measure modeled solar input from the metal halide lamps.

Prior to each experiment, the metal halide lamps were powered on for approximately one hour to ensure stabilization. Calibration of the light distribution across the collector aperture was accomplished by using pyranometers to measure incident radiation across the collector at 15 different equidistant locations. Using combinations of mirrored reflectors and aluminum foil blinders, the lamps were placed in an orientation that yielded a light distribution with a maximum standard deviation normalized by mean radiation value of 5% in conjunction with a minimum mean radiation value of 750 W/m^2 . Also, a correction factor, cf , correlating radiation measured at the four corners to mean radiation across the entire absorber was calculated.

2.3.2 Full-scale Prototype

A data acquisition system and accompanying software identical to that used for the bench-scale experiments were also used for the prototype experiments. Two pyranometers were mounted on the center of the south wall of the PSTF, one aligned with the top and the other with the bottom of the collectors. Four T-type thermocouples were placed on each central absorber,

three on each central evaporator, and two inside each individual storage tank. Additional placements of thermocouples included one attached to the tank wall to measure tank surface temperature, one placed in the room to measure room temperature, two on the exposed condenser, and two placed outside to measure ambient air temperature.

2.4 Data Analysis

2.4.1 Bench-scale Experiments

Bench-scale system thermal efficiency, η , was given by

$$\eta = \frac{\dot{Q}_u}{IA_c} \quad (15)$$

where I is the measured incident radiation on the collector and A_c is the collector (absorber) area.

Useful power gain, $\dot{Q}_u$, was based on the rate of temperature increase in the tank given by

$$\dot{Q}_u = \frac{M_{bs}c_p\Delta T_{s,bs}}{\Delta t} \quad (16)$$

where M_{bs} is the mass of the water in the bench-scale experiment tank, c_p is the specific heat of water, and $\Delta T_{s,bs} / \Delta t$ is the rate of change of bench-scale tank temperature with time, which was found by fitting a line to experimental results over a 3°C storage temperature rise from 25 to 28°C. Due to the limited range of operating conditions, these data were insufficient to determine an energy collection efficiency curve (such as Figure 13 for the simulations and the full-scale system).

2.4.2 Full-scale Prototype

All measured temperatures and insolation for the full-scale prototype were averaged over hourly intervals and used to calculate system efficiency by equation 15. With mechanisms

deployed to reduce unwanted gains (for these experiments, a cover or a valve in each heat pipe), the objective was to minimize power delivered by the heat pipes to storage and to the room (which are unwanted thermal gains during the cooling season)

$$\dot{Q}_u = \dot{Q}_s + \dot{Q}_{sr} + \dot{Q}_{cnds r} \quad (17)$$

where $\dot{Q}_s$ represents the net power retained in all four storage tanks

$$\dot{Q}_s = \frac{M_{fs} c_p \Delta T_{s,fs}}{\Delta t} \quad (18)$$

where M_{fs} is the mass of the water in the full-scale prototype storage tanks and $\Delta T_{s,fs}$ is the full-scale water temperature change during the one-hour time interval Δt . $\dot{Q}_{sr}$ is the power transfer from the storage tanks to the room

$$\dot{Q}_{sr} = k_{sr} (T_s - T_r) \quad (19)$$

where T_s and T_r are the measured temperatures for the water and room, respectively, and k_{sr} is the total heat transfer from the thermal mass to the room air, and $\dot{Q}_{cnds r}$ is the power transfer from the exposed condenser to the room. A thermal resistance network for k_{sr} is shown in Figure 11, and a description for each thermal resistance from Figure 11 is shown in Table 2.

Figure 11. Thermal network used to calculate $\dot{Q}_{sr}$ for heating units with a thermal storage tank.

The network included convective heat transfer to the inside surface of the tank wall, $k_{s,conv}$, conduction through the tank wall, k_{cond} , and parallel radiation, k_{rad} , and convection, $k_{r,conv}$, from the wall of the tank to the room. Applying the thermal network shown in Figure 11, k_{sr} is

$$k_{sr} = \left[\frac{1}{k_{s,conv}} + \frac{1}{k_{cond}} + (k_{rad} + k_{r,conv})^{-1} \right]^{-1} \quad (20)$$

The convection coefficient for water or air is

$$k_{s/r,conv} = \frac{\kappa_{s/r} Nu_{s/r} A_s}{L} \quad (21)$$

where $\kappa_{s/r}$ is the thermal conductivity for water or air, L is the tank wall height, and the Nusselt number, $Nu_{s/r}$, for water or air is dependent on whether convection conditions are laminar or turbulent. For laminar flow, the Nusselt number for free convection from the vertical sides of the tanks is [Incropera & DeWitt, 2011]

$$Nu_{s/r} = 0.68 + \frac{0.67 Ra_{s/r}^{1/4}}{\left(1 + (0.492 / Pr_{s/r})^{9/16}\right)^{4/9}} \quad (22)$$

where $Pr_{s/r}$ is the Prandtl number. For turbulent conditions, the Nusselt number for the vertical sides is

$$Nu_{s/r} = 0.825 + \frac{0.387 Ra_{s/r}^{1/6}}{\left(1 + (0.492 / Pr_{s/r})^{9/16}\right)^{8/27}} \quad (23)$$

Additionally, convective heat transfer takes place between the room and the top surface of the top storage tank and the bottom surface of the bottom storage tank, respectively. Convective heat transfer between the storage water and top surface of the top storage tank was found using [Incropera & DeWitt, 2011]

$$Nu_s = 0.15 Ra_s^{1/3} \quad (24)$$

and convective heat transfer between the top surface of the top tank and the room was found using [Incropera & DeWitt, 2011]

$$Nu_s = 0.54Ra_s^{1/4} \quad (25)$$

Convective heat transfer from the storage water to the bottom surface of the bottom tank, and between this surface and the room was found using [Incropera & DeWitt, 2011]

$$Nu_{s/r} = 0.52Ra_{s/r}^{1/5} \quad (26)$$

Laminar to turbulent transition depended on the Rayleigh number

$$Ra_{s/r} = \frac{g\beta_{s/r}\Delta T_{sw/wr}L^3}{\nu_{s/r}\alpha_{s/r}} \quad (27)$$

where g is the acceleration due to gravity, $\beta_{s/r}$ is the coefficient of volumetric thermal expansion for the water or air, $\nu_{s/r}$ is the kinematic viscosity for water or air, and $\alpha_{s/r}$ is the thermal diffusivity for water or air. The temperature difference between mediums is

$$\Delta T_{sw} = T_s - T_w \quad (28)$$

or

$$\Delta T_{wr} = T_w - T_r \quad (29)$$

where T_w is the tank wall temperature. For $Ra_{s/r} < 10^9$, free stream conditions are laminar, and when $Ra_{s/r} > 10^9$, they are turbulent.

Conduction through the tank wall is

$$k_{cond} = \frac{\kappa_w A_s}{th} \quad (30)$$

where th is the tank wall thickness and κ_w is the thermal conductivity of the tank wall. The radiative heat transfer coefficient was found using [Incropera & DeWitt, 2011]

$$k_{rad} = \varepsilon \sigma A_s (T_w + T_r)(T_w^2 + T_r^2) \quad (31)$$

where ε is the radiative emissivity of the water tank wall and σ is the Stefan-Boltzmann constant.

For the condenser exposed directly to room air, $\dot{Q}_{cndsr}$ is

$$\dot{Q}_{cndsr} = k_{cndsr} (T_{cndsr} - T_r) \quad (32)$$

where T_{cndsr} is the temperature of the exposed condenser and the thermal network for k_{cndsr} is shown in Figure 12.

Figure 12. Thermal resistance network used to calculate q_{out} for a heating unit with the condenser exposed directly to room air.

Applying the thermal network shown in Figure 12, k_{cndsr} is

$$k_{cndsr} = k_{rad,cndsr} + k_{conv,cndsr} \quad (33)$$

where $k_{conv,cndsr}$ is the conductance for convection from the directly exposed condenser

$$k_{conv,cndsr} = \frac{\kappa_r Nu_D A_{cndsr}}{D} \quad (34)$$

where A_{cndsr} is the surface area of the condenser, D is the outside diameter of the condenser and the Nusselt number is [Incropera & Dewitt, 2011]

$$Nu_D = \left[0.60 + \frac{0.387 Ra_D^{1/6}}{\left(1 + (0.559/Pr)^{9/16} \right)^{8/27}} \right]^2 \quad (35)$$

and the Rayleigh number is [Incropera & Dewitt, 2011]

$$Ra_D = \frac{g\beta(T_3 - T_6)D^3}{\nu\alpha} \quad (36)$$

The radiative heat transfer from the condenser is

$$k_{rad, cndsr} = \varepsilon_{cndsr} \sigma A_{cndsr} (T_{cndsr} + T_r)(T_{cndsr}^2 + T_r^2) \quad (37)$$

where ε_{cndsr} is the radiative emissivity of the copper condenser.

The properties of the water in the tank and the air in the room were determined by interpolation from thermophysical property tables [Incropera & DeWitt 2011] at the average tank and room temperature over the course of that hour.

Useful gains that would have occurred in the full-scale prototype had mechanisms not been employed to reduce gains were estimated by characterizing system efficiency versus loss potential to insolation ratio (Figure 13). Experimental energy collection efficiency data in Figure 13 were acquired during normal operation of the prototype during January and February of 2013 [Robinson & Sharp 2014]. Data points were selected around solar noon in accordance with ASHRAE standard 93 [2010] as described by Robinson et al. [2013]. Using the linear fit equation from Figure 13, system efficiency, without utilizing control mechanisms, was approximated as

$$\eta = 0.793 - 3.647 \frac{T_s - T_a}{I} \quad (38)$$

where T_a is the ambient temperature. Also shown on Figure 13 are data for the simulated system sampled during the same period of TMY3 weather for Louisville, selected according to the same ASHRAE standard. The intercepts for the two systems are in good agreement, but the slope for the experimental system is greater, perhaps indicating greater losses than modeled in the simulations. However, the experimental data set was limited to a narrow range of the collection efficiency parameter (on the abscissa) compared to simulations, thus this interpretation is

tentative. Over the range of the experimental data, the performance of the two systems is reasonably close.

Figure 13. System energy collection efficiency for the full-scale prototype (solid line) using data collected during normal operation January & February 2013 versus simulations (dashed line) using TMY3 weather.

2.5 Error Propagation

2.5.1 Bench-scale Experiments

Bench-scale thermal efficiency (Equation 15) depends on measurements of radiation received by the collector and storage temperatures (Equation 16). Each of these measurements was subject to digitization error in addition to rated uncertainty for the thermocouples and pyranometers. Therefore, the uncertainty in calculated system efficiency, μ_η , was found using

$$\mu_{\eta} = \sqrt{\left(\frac{\partial \eta}{\partial \Delta T_s} \mu_{\Delta T_s}\right)^2 + \left(\frac{\partial \eta}{\partial I} \mu_I\right)^2} \quad (39)$$

where ΔT_s is the storage temperature difference from Equation 16 and μ represents the uncertainty in temperature and radiation measurements, respectively.

For any arbitrary value of $\Delta T = T_1 - T_2$, we have

$$\frac{\partial \Delta T}{\partial T_1} = \frac{\partial \Delta T}{\partial T_2} = 1 \quad (40)$$

therefore the digitization uncertainty for any ΔT is

$$\mu_{\Delta T, digit / tc} = \sqrt{\left(\frac{\partial \Delta T}{\partial T_1} \mu_{T_1}\right)^2 + \left(\frac{\partial \Delta T}{\partial T_2} \mu_{T_2}\right)^2} = \sqrt{2\mu_T^2} = \sqrt{2}\mu_{digit / tc} \quad (41)$$

where μ_{digit} is the digitization error in each temperature measurement, which for the SCXI-1600 and SCXI-1102 modules was ± 0.027 K/level, and thermocouple uncertainty, μ_{tc} , is roughly 0.1 K up to approximately 100°C [Ripple et al., 1994]. Thus the calculated value for $\mu_{\Delta T, digit}$ was 0.038°C, and the calculated value for $\mu_{\Delta T, tc}$ was 0.141°C. Accordingly, the overall uncertainty in storage temperature measurement was

$$\mu_{\Delta T_s} = \sqrt{\mu_{\Delta T, tc}^2 + \mu_{\Delta T, digit}^2} \quad (42)$$

resulting in an overall uncertainty in temperature measurement of 0.146 K.

The pyranometers have an overall error of $\pm 3\%$ of the measured value, as specified by the manufacturer Kipp and Zonen. Accordingly, the maximum μ_{η} occurred at the maximum value of radiation measured during the testing period - equal to 825.3 W/m² and resulting in a maximum μ_s of 24.76 W/m². Values for each derivative from Equation 39 are shown in Table 2. The derivative associated with change in storage temperature represented the greatest uncertainty while the derivative associated with radiation measurement represented the least uncertainty.

Uncertainty for water mass was $< 0.5\%$, so was not included. The estimated maximum uncertainty in system efficiency from equation 39 was $\pm 4.57\%$.

Table 2. Value for each derivative associated with overall uncertainty in calculated system efficiency for the bench-scale experiments.

Derivative	Value
$\partial \eta / \partial \Delta T_s$	0.2673
$\partial \eta / \partial \Delta I$	-0.0009

2.5.2 Full-scale Prototype

System thermal efficiency (Equation 15) depended on measurements of storage, room and condenser temperatures (Equations 18, 19 and 32) and of insolation. Therefore, the uncertainty in calculated system efficiency, μ_η , was found using

$$\mu_\eta = \sqrt{\left(\frac{\partial \eta}{\partial \Delta T_s} \mu_{\Delta T_s}\right)^2 + \left(\frac{\partial \eta}{\partial \Delta T_{sr}} \mu_{\Delta T_{sr}}\right)^2 + \left(\frac{\partial \eta}{\partial \Delta T_{cnds}} \mu_{\Delta T_{cnds}}\right)^2 + \left(\frac{\partial \eta}{\partial I} \mu_I\right)^2} \quad (43)$$

where ΔT_s is the storage temperature difference from Equation 18, ΔT_{sr} is the temperature difference between storage and room ($T_s - T_r$) from Equation 19, and ΔT_{cnds} is the temperature difference between the exposed condenser and room ($T_{cnds} - T_r$) from Equation 32. Uncertainties for temperature differences and pyranometer measurements are the same as in section 2.5.1. Similarly, uncertainty for water mass was $< 0.5\%$, so was not included. Values for each derivative from Equation 43 are shown in Table 3. The estimated uncertainty in system efficiency was $\pm 5.00\%$.

Table 3. Value for each derivative associated with overall uncertainty in calculated system thermal gains.

Derivative	Value

$\partial \eta / \partial \Delta T_s$	0.3132
$\partial \eta / \partial \Delta T_{sr}$	0.0017
$\partial \eta / \partial \Delta T_{cnds r}$	0.0026
$\partial \eta / \partial \Delta I$	-0.0010

3. RESULTS & DISCUSSION

3.1 Bench-scale Experiments

Modifications to the baseline bench-scale experiment (*i.e.*, adding a thermal blanket around the storage tank to reduce thermal losses) increased system thermal efficiency to 89.1% (up from 85.1% obtained by Albanese et al. [2012]). Bench-scale experiments revealed a decrease in average system thermal efficiency (to 82.2%) of 6.9% when the evaporator and condenser sections were leveled. This decreased efficiency was applied to all simulations involving the switching mechanism by increasing $q_{a,aux}$ by 6.9%.

3.2. Simulations

3.2.1. Baseline Simulations

As curve fit in Figure 13, the efficiency for the simulated system had an intercept of 78.9%, which was quite close to the 79.3% intercept for the full-scale prototype. While the curve fits predict disagreement farther from the ordinate, over the limited range of the experimental data, validation of the simulations is reasonable.

Baseline annual heating, cooling and total loads without the heat pipe wall are shown in Figure 14 for each of the four locations representing different climate types. Madison and Rock Springs have substantially colder winters and higher heating loads, while Louisville and

Albuquerque have warmer summers and higher cooling loads. It is evident that the heating load dominates in all locations.

Figure 14. Annual loads for four locations without the heat pipe wall using a room comfort range of 18.3°C to 23.9°C .

The effect of indoor comfort temperature range is shown in Figure 15, in which simulated annual unwanted gains ($q_{a,uwg}$), auxiliary heating ($q_{a,aux}$) and total energy (q_{total}) loads in Louisville are plotted for a building with the heat pipe wall. Without reduction mechanisms, unwanted gains in Figure 15 are larger than the baseline cooling load (Figure 14), and increase as the comfort temperature range is restricted. With the narrowest comfort range, the total load is larger with the heat pipe system than without (Figure 14). A 1.1°C increase in acceptable range – from 20.6 – 21.7°C to 20.0 – 22.2°C – decreased the annual load by 10.2%. Increasing the range to 18.3 – 23.9°C decreased the annual load an additional 15.4%. Results were similar in the other climates. All of the following results use the larger 18.3 – 23.9°C range.

Figure 15. Annual unwanted gains, auxiliary heating and total energy load for Louisville utilizing the heat pipe wall (without reduction mechanisms) for room comfort ranges of 18.3-23.9°C (65-75°F), 20.0-22.2°C (68-72°F) & 20.6-21.7°C (69-71°F).

Energy use in buildings with heat pipe walls without mechanisms to reduce unwanted gains are shown in Figure 16. All components of energy are normalized by baseline totals without the heat pipe system (Figure 14). The heat pipe wall serves a substantial portion of the heating load, but the cooling load more than doubles in all locations due to unwanted gains from the system during the cooling season. In spite of the increased cooling load, the total load in each location is lower with the heat pipe system than without.

Figure 16. Annual loads for four locations with the heat pipe wall.

3.2.2. Control Strategies

For the seasonal control strategy, simulations showed that any month with $SD \geq 0.6$ should be included as part of the cooling season to minimize auxiliary energy use. This rule worked for all eight locations, suggesting that SD may be a ‘universal’ parameter that can be applied to a wide range of climates for quick assessment of its optimal cooling season. Cooling seasons, based on SD, for each of the eight locations are shown in Table 4.

Table 4. Cooling season for eight locations based on SD ratio.

Location	Cooling Season
Albuquerque, NM	May - September
Boston, MA	June - September
Chicago, IL	June - September
Denver, CO	June - September
Louisville, KY	May - September

Madison, WI	June – August
Rock Springs, WY	July - August
Seattle, WA	July - August

The control strategy based on ambient temperature provided the lowest levels of unwanted gains for all locations and mechanisms (Figure 17). Even though auxiliary heat requirements were higher than for other control strategies in most cases, ambient temperature-based control produced the lowest total loads.

Figure 17. Annual loads for four locations for each control strategy using the cover mechanism.

Results for ambient temperature-based control in Louisville are shown in Figure 18 with each mechanism and their practical combinations. Trends in performance of mechanism combinations for the other three locations were similar to Louisville. The best single mechanism was the valve. The cover mechanism accumulated some unwanted gains during periods in which high ambient temperatures resulted in convective and conductive gains to the absorber. With no valve to stop two-phase heat transfer, small amounts of heat was occasionally transferred to the system even

when insolation was eliminated by the cover. On the other hand, the valve mechanism was simulated as complete elimination of two-phase heat transfer within the heat pipes. Even though conduction along the adiabatic section pipe wall remained, simulated unwanted gains were near zero for the year.

Figure 18. Annual loads in Louisville for unwanted gains reduction mechanisms with ambient temperature based control.

The switching mechanism provided heat transfer out of the system, q_{sw} , that depended on control strategy and location (Figure 19). The seasonal strategy resulted in the largest amount of heat removal from the system, because the switching mechanism is employed during the entire cooling season. Therefore, cool days that occur during the season promote greater convective and conductive heat transfer from the collectors. On the other hand, the ambient-based and room-based strategies allow cooling by the switching mechanism only when ambient air is above the lower and upper room comfort limits, respectively. Thus heat transfer out the system occurs at smaller rates and happens less frequently. In particular, these two strategies do not take

advantage of diurnal temperature swings in dry and high altitude climates. The magnitudes of cooling for all strategies were small, amounting to maximums of 3.5%, 4.9%, 4.8%, and 6.9% of the total cooling load in Louisville, Albuquerque, Madison and Rock Springs, respectively. These values are limited in part because the heat pipe system was designed for heating with features such as low IR emittance and transmittance for the selective absorber and glass cover, respectively. Although the seasonal strategy resulted in the largest amount of cooling from switching, higher unwanted gains with this strategy produced higher total annual loads than the ambient-based control strategy (Fig. 17). Another contributing factor was the reduced heating performance of the leveled evaporator and condenser sections of the heat pipe to accommodate switching, which resulted in higher auxiliary requirement. Together, these factors yielded 2.2% and 3.1% higher annual load for the switching mechanism than the cover and valve mechanisms, respectively. Forecast control based on prediction of cooling needs might better utilize the capability of the switching mechanism, particularly if heat transfer properties of the system were more favorable for cooling. The combinations of mechanisms most effective in reducing annual load were those involving the valve (valve and cover, and valve and shading), but these combinations did not reduce the load significantly below that of the valve alone.

Figure 19. Cooling provided by the switching mechanism.

The total annual space-conditioning loads for each of the four locations using the heat pipe wall with the best control strategy (ambient temperature) are shown in Figure 20. This graph presents the clearest comparison of mechanisms based on total annual energy use. It is evident that the valve mechanism, as simulated, produces the lowest total loads in all locations. The cover matches the valve within one percentage point. Switching produces higher loads in all locations except Albuquerque. This was because high performance from the heat pipe system in sunny Albuquerque results in a low $q_{a,aux}$ requirement, thus the reduction in system performance when the evaporators and condensers are leveled doesn't have as much significance on q_{total} as it does in locations with smaller solar fractions. Shading is only competitive with the cover and valve in Rock Springs, the climate with the lowest cooling load (Figure 14).

Figure 20. Total annual loads with different mechanisms with ambient temperature-based control strategy.

3.3 Full-scale Experiments

System efficiency, η , from 9am to 5pm for three consecutive days, is shown in Figures 21 – 23 for the valve mechanism, cover mechanism, and combination of valve and cover, respectively. η would be zero if the mechanisms were perfectly effective, but as can be seen in the figures, some heat transfer to the interior remains. Figures 21 – 23 also show estimated efficiencies had control mechanisms not been used (Equation 38). Unlike the simulations, the valve (Figure 21) was less effective in the experiments, and the cover and valve combination (Figure 23) provided noticeable benefits relative to each single mechanism.

Figure 21. Prototype efficiency with and without the valve mechanism on August 22 - 24, 2013.

Figure 22. Prototype efficiency with and without the cover mechanism on September 9 - 11, 2013.

Figure 23. Prototype efficiency with and without the combination of valve and cover mechanisms on August 28 - 30, 2013.

For each three-day period, system thermal gains were reduced by an average of 83.8% for the valve mechanism (Fig. 21), 93.2% for the cover mechanism (Fig. 22), and 98.1% when combining the two mechanisms (Fig. 23). These experimental values are compared to simulated annual reductions in unwanted thermal gains using the same mechanisms in Figure 24. Because of the difference in time periods and weather data for the experiments and simulations, this comparison is qualitative, but nonetheless provides another indication of the relative effectiveness of the mechanisms as tested. While experimental reductions for the cover and the combination were within the simulated range for all three strategies, experimental reductions using the valve mechanism were lower than reductions reflected in simulations. One possible cause of the smaller reduction is that, although the valve and the adiabatic section were insulated, some heat transfer to the room may have occurred. In particular, each valve was installed between the rubber adiabatic section and the copper condenser section, and two-phase heat transfer likely still occurred between the evaporator and the lower face of the valve. Therefore,

high temperatures at the lower face of the valve (hourly average absorber temperatures were as high as 107.5°C (225.4°F) during this testing period) resulted in additional thermal gains to the room and storage via conduction through the valve and to the condenser.

Figure 24. Simulated annual and experimental three-day unwanted gains reductions in Louisville.

To investigate whether heat was transferred through the valve to the condenser section, temperatures of the condenser exposed to air were compared on similarly sunny days when the valve was closed and open (Figure 25). While it can be seen that the condenser temperature does rise slightly during insolation with the valve closed, the rate of rise is much greater with the valve open. This suggests that thermal gains to the condenser were a result of conduction heat transfer along the copper pipe wall, instead of the much higher two-phase heat transfer when the valve is open. Consequently, it is expected that a system design in which the valve is installed in the middle of the rubber adiabatic section would further reduce unwanted gains since heat

transfer downstream of the valve would be restricted by thermally resistive rubber. The low temperature rise in the condenser with the closed valve also suggests that greater heat may flow to the room from the potentially very hot section of the heat pipe between the absorber and the closed valve.

Figure 25. Prototype exposed condenser temperatures using the valve mechanism on August 22, 2013, and without the valve mechanism on January 3, 2013.

4. CONCLUSIONS

Computer simulations show that maximum reductions in the overall annual space heating and cooling loads are achieved when the heat pipe augmented solar wall utilized a control strategy based on ambient temperature. The best simulated control mechanism was the valve, which essentially eliminated unwanted gains and produced the lowest total annual load. Close behind were the cover and switching, which also nearly eliminated unwanted gains. The cover required

the same auxiliary heating as the valve, but switching required about 2% more due to performance reductions during the heating season. Cooling by the switching mechanism was limited by the optical and thermal design of the collector (to favor heating performance), and heating performance was reduced by leveling the evaporator and condenser sections for more convenient switching. Combining valve or cover with other mechanisms did not significantly improve simulated performance. Shading provided much lower reductions in unwanted gains.

Experimental results confirmed the performance of the cover, but suggested that a small amount of heat was still transferred when the valve was closed. Changes in valve placement and heat pipe design, for instance by adding a section of low conductivity rubber tubing between the valve and the condenser, may reduce, but not entirely eliminate, closed-valve heat transfer.

With the valve and cover providing nearly the same performance, practical implementation of automatic control becomes a consideration. Five valves (one in each heat pipe) would be straight-forward to actuate, would require little additional space and the components would be located indoors and out of the weather. On the other hand, an automated cover is a single, but large, mechanism that must withstand weather conditions. An electrochromic surface could potentially be used to block solar radiation without moving parts. Both cover and valve mechanisms provide opportunities for modulated control, rather than on/off control, which could be advantageous, particularly in the fall and spring when heating and cooling loads can occur within short time periods. Independent control of the heat pipe with exposed condenser might be especially beneficial due to its fast response.

The heat pipe augmented solar wall, analyzed with respect to heating only, exhibits positive returns with increasing solar collection area. However, as collector area increases, so does the potential for unwanted gains during the cooling season, which if reduction mechanisms are not

incorporated, will at some threshold begin to increase overall annual load. The heat pipe system shares this common limitation with other passive solar systems. By applying the mechanisms identified in this study, freedom is returned to use larger collector area without increasing unwanted gains to unacceptable levels, and the overall annual load can be reduced to much lower levels than without the mechanisms. Previous studies have shown that lower values of LCR can serve more than 90% of the annual heating load in many locations throughout the U.S. [Robinson & Sharp, 2014]. Research addressing renewable, passive solutions for combining heating and cooling from alternative sources is the next step towards full realization of net-zero energy space conditioning of buildings.

5. ACKNOWLEDGEMENT

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HIGHLIGHTS

- FOUR MECHANISMS AND THREE CONTROL STRATEGIES FOR REDUCING GAINS WERE SIMULATED
- VALVES TO TURN OFF THE HEAT PIPES AND A COVER REDUCED GAINS MOST
- AN OVERHANG WAS LESS EFFECTIVE
- TESTS OF A FULL-SCALE PROTOTYPE CONFIRMED REDUCTIONS WITH VALVES AND COVER