HVAC OPTIONS FOR LOW ENERGY RESIDENTIAL BUILDINGS

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Abstract: Data now available from prototype low energy residences confirm that peak heating and cooling loads can be reduced by factors of 3 to 5, using currently available technologies. This work explores the range of candidate HVAC systems capable of dealing with such fundamentally different load profiles as 8°C balance points, spatially non-uniform loads, and non-negligible thermal capacitance. This paper quantifies peak heating and cooling load reductions in a North American climate, investigates the necessity of ductwork, and examines the role of thermal mass effects in low energy residences.

Results reveal surprisingly small differences in overall efficiency among well-designed centralized (ducted) and decentralized vapor-compression heat pump systems employing either mechanical or desiccant dehumidification. Centralized systems offer opportunities for integration of domestic hot water heating, and some types of decentralized systems offer the long-term possibility of being integrated into wall panels or other structural elements. Decentralized system options included DX and secondary loops. Since energy efficiencies are comparable, system selection is therefore likely to be driven by such factors as initial costs, complexity and reliability.

Key Words: ***heat pumps, integrated energy systems, low energy house.***

1 INTRODUCTION

Modern manufacturing and construction technologies offer the potential for mass producing components of building thermal envelopes, and then assembling them onsite in a way that minimizes losses through thermal bridges and nearly eliminates infiltration of outdoor air. At the same time, modern control technologies offer the possibility of operating ventilation, heating, cooling and dehumidification systems in ways that minimize the annual energy use of integrated systems.

Wall (2006) investigated the performance of low energy row houses near Gothenburg, Sweden, which realized a 40% reduction in purchased energy when compared to a home built to normal Swedish standards. Measured infiltration rate was 0.05 air changes per hour (ach) compared to 0.13 ach for standard Swedish construction. Efficient windows balanced winter conduction losses with solar gains.

Most data from existing low energy buildings is from relatively dry climates dominated by heating loads, notably in Austria, Switzerland and Scandanavia. In cool and dry climates it is possible to meet heating loads by conditioning only the ventilation air, and/or to meet modest cooling loads with sensible-only wall and ceiling panels operating above the indoor dewpoint.

This paper addresses hot and humid climates as well, considering desiccant dehumidification systems as well as mechanical. Recently Tsay et al. (2006) addressed this issue experimentally by pairing a desiccant dehumidifier with a CO_2 heat pump and using the heat rejected in cooling mode to regenerate the desiccant dehumidifier. Desiccant performance was related to the temperature used for regeneration, and 63°C waste heat from a transcritical CO_2 system was found adequate to regenerate the desiccant in a hot and humid climate (~32°C at 65% relative humidity).

Thermal storage, specifically the intelligent use of building thermal mass, can play a very significant role in very efficient buildings. Keeney and Braun (1997) investigated building precooling as a way to reduce peak loads in large office buildings. Less work has been done to quantify thermal mass in single family residences, especially energy-efficient ones. However, Reddy et al. (1991) modeled a residence as a simple one resistor, one capacitor network and was able to predict energy savings within 10%.

Wall and ceiling panels have been used for heating and sensible cooling in dry climates, and remain an active research topic for humid climates. Kilkis (2006) analytically optimized a hybrid wall panel system. These hybrid panels could meet part of the sensible loads by radiative heat transfer and latent loads by utilizing liquid desiccant dehumidification. Results suggested that these hybrid wall panels would require approximately 30% of a room's total wall area. Imanari et al. (1999) investigated more traditional chilled water radiant ceiling panels, finding that convection currents generated by the cool ceiling panels led to a room temperature distribution with slightly cooler air at eye level and slightly warmer air near the floor. His test subjects found this "cool head, warm feet" distribution to be more comfortable than that of a traditional HVAC system.

Worldwide attention to the climate change issue, and the need to reduce global carbon emissions nearly 80% by 2050 has spurred extensive cooperative attempts to conduct research in a much more organized manner. One of the most ambitious is the International Energy Agency Heat Pump Programme's (2008) 3-year project aimed at documenting current and proposed building codes in 9 countries, developing consistent performance measures for the wide range of heat pumping technologies that might be suitable for such applications. Many of these technologies involve integration of domestic water heating systems with space conditioning technologies, and exploiting opportunities for tapping renewable energy sources through solar assist, or by ground-coupling of the ventilation system and/or the heat sources and sinks.

2 LOADS AND VENTILATION

Since most of the published studies have dealt with low energy residences in cool and/or dry climates, the present analysis begins by focusing on a baseline one storey, 200m^2 house designed for a mid-North American climate (St. Louis) that experiences hot humid summers as well as cold winters. With thermal envelope heat transfer resistances and infiltration rate taken from the prototypical house detailed in Chapter 29 of the ASHRAE Handbook of Fundamentals (2005), this baseline house would likely use less energy than most houses being constructed today in the US. A second "low energy" residence was also characterized, having identical envelope dimensions but a more efficient thermal envelope with U-values and infiltration rate reported by Wall (2006) for energy efficient row houses in Sweden.

2.1 Modeling assumptions

Peak heating and cooling loads were then calculated for both houses. Common characteristics are summarized in Table1, and differences in Table 2. Design loads were determined using 99% ASHRAE climate data for St. Louis, Missouri, which experiences seasonal temperature extremes (-13 to 34°C) that produce peak heating and cooling loads of comparable absolute value. Heat pumps installed in more northerly or southerly locations would be sized for peak heating or cooling loads, respectively. Indoor temperature and relative humidity setpoints were 20°C/30% for heating and 24°C/50% for heating and cooling respectively. Ventilation was calculated for both houses using the standard ASHRAE formula, as a function of floor area and number of occupants. Infiltration rate for the baseline house was calculated using ASHRAE formulae assuming “tight construction”. Other modeling assumptions and methods are detailed by Yannayon and Bullard (2007) and summarized only briefly in this paper.

Table 1- Characteristics of baseline and low energy house

General building parameters	
Length (N. & S. sides)	20 m
Width	10 m
Height	2.5m
Roof pitch	0
Number of doors	2
Door area	2.2 m^2
Number of occupants	4
Number of bedrooms	3
% time house occupied	100%
Construction	Slab on grade
Window area [% of wall]	15%
Window height	1 m
% Fixed	50%
% Operable	50%

Table 2- Differences between baseline and low energy house

	Baseline House	Low Energy House
	U [W/m^2-K]	
Roof/ceiling	0.18	0.08
Exterior walls	0.51	0.10
Doors	2.30	0.80
Floor	0.21	0.11
Windows	2.84	0.85
	Other	
Solar heat gain coefficient (SHGC)	0.67	0.50
Infiltration [ach]	0.28	0.05
Energy wheel	no	yes

2.1 Latent loads

Like ventilation loads, latent loads due to breathing, cooking, bathing etc. are estimated for both residences using standard formulae from the ASHRAE Handbook of Fundamentals. And like ventilation loads they are attenuated by 80% in the low energy house due to the assumed presence of an enthalpy wheel (Dieckmann et al. 2003). Combined with the low infiltration rate, the enthalpy wheel reduces peak latent loads to zero (internal loads are sufficient to maintain humidity), while in the baseline house humidification energy accounts for 6% of peak heating load. During the cooling season latent loads in the baseline house account for 28% of the peak load, and only 15% in the low energy house due primarily to the presence of the energy recovery wheel.

2.2 Sensible loads

The components of sensible load are shown in Figure 1 for both residences. Modeling was straightforward. Solar gain and conduction through windows were calculated from the respective buildings' window properties, with shading coefficients and other parameters taken from the ASHRAE Handbook's prototype and applied to each house. Average appliance loads of 470W were obtained from the same source.

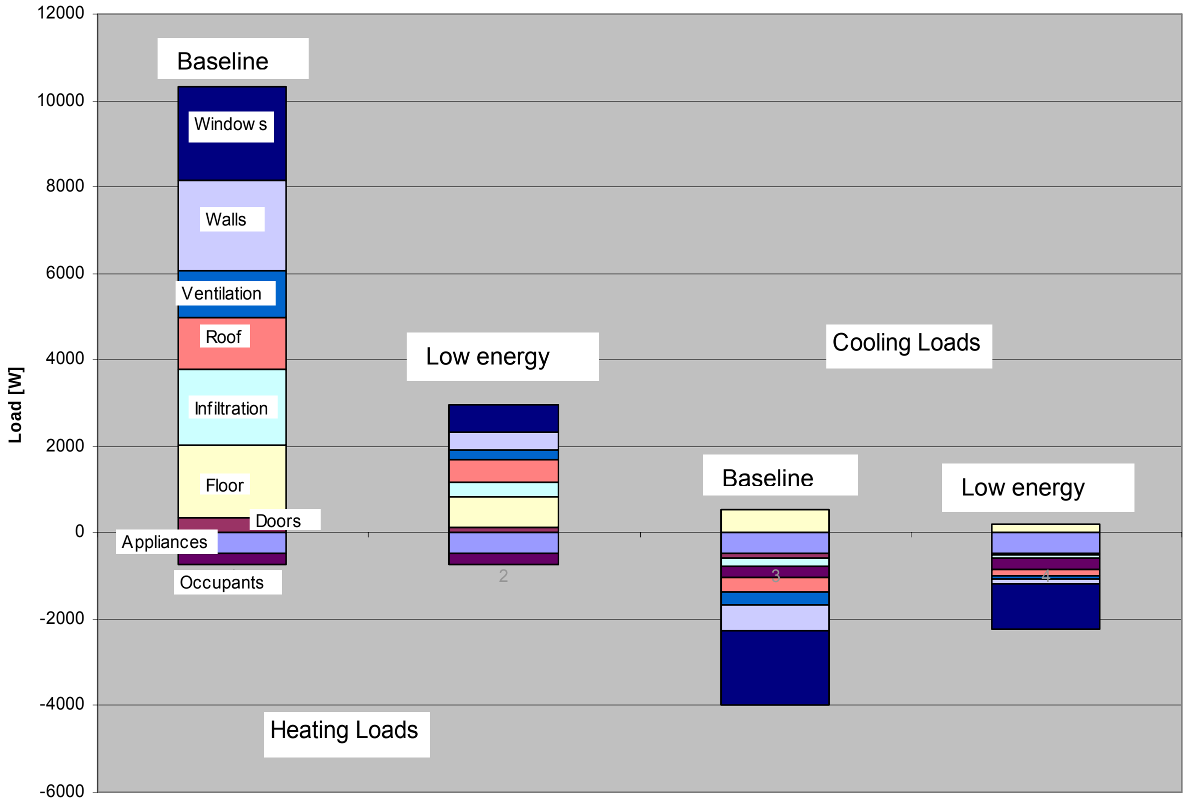

Figure 1. Components of sensible loads

Note in Figure 1 that the fenestration properties taken from Wall's demonstration project in Sweden result in a net heat loss from the low energy house when located in the St. Louis climate. More recent analyses by Arasteh et al. (2006) show how proper selection of fenestration can produce a net solar gain during the heating season throughout the US, large enough to offset the window's cooling load during summer. Thus the loads shown here for the low energy house are conservative.

2.3 Nonuniform solar loads

Ventilation systems must be designed to provide enough circulation to offset nonuniform thermal loading. The peak summer solar (direct plus diffuse) load on the baseline (south-facing) house is 1082 W, and 853 W (or 20% less) for the efficient house due to its lower solar heat gain coefficient. However the greatest nonuniformity occurs when sun angles are low during morning and afternoon.

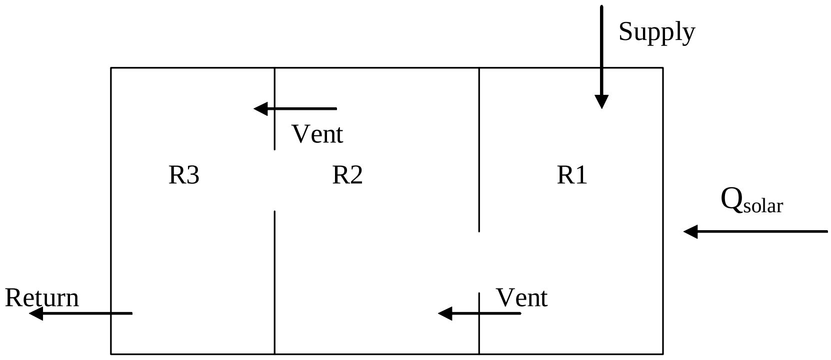

Figure 2. Centralized system with asymmetric load

For simplicity the house is modelled as having three rooms separated by two doorways, to determine whether natural convective coupling and conduction through interior walls would overcome temperature maldistribution caused by the worst-case asymmetric thermal loads (e.g. solar; kitchen) or a central ventilation system discharging air into only one room. Of course a set of smaller individual heat pumps located in each room could offset the nonuniformity, but their aggregate capacities may exceed the peak load on the house. This simplification does not necessarily imply that the house has only three rooms, only that no more than two doorways separate the supply and return air ducts.

Figure 2 illustrates the case of a central system sized for peak load, discharging into Room 1 of the low energy house to offset 526 W solar load on that room. Figure 3 shows how the temperature differences between the rooms decrease as the central system's air flow rate increases above the 1.5 m^3/min (cmm) minimum fresh air requirement for the house. Results depict the effect of ventilation alone, when outdoor ambient temperature near the balance point of 8°C. The levelizing effect of internal loads (occupants; appliances) is apparent from the second graph, suggesting that extensive ductwork and high air flow rates are not required for worst-case solar load asymmetry.

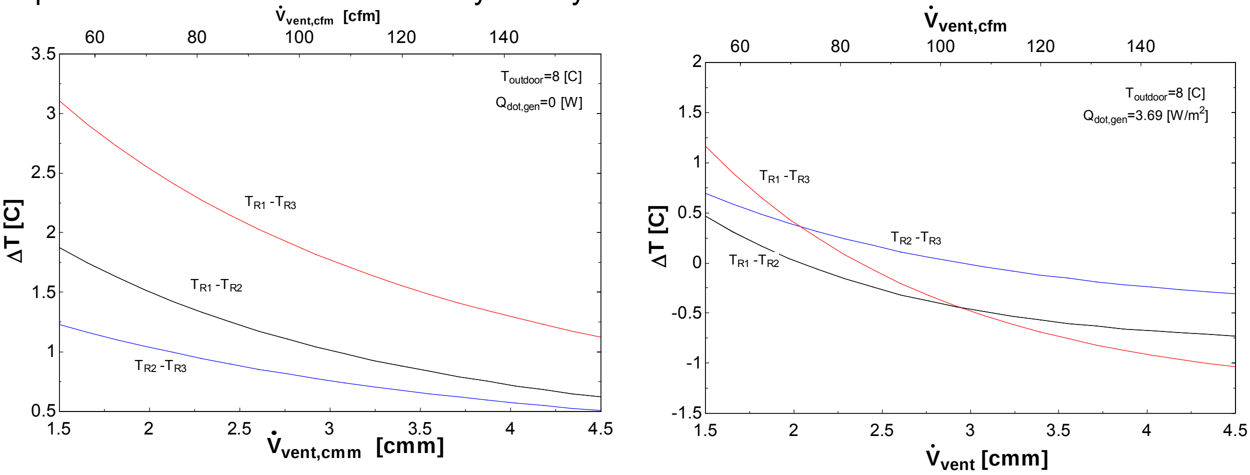

Figure 3. Effect of internal loads on room-to-room temperature differences

The analyses by Yannayon and Bullard (2007) demonstrated that load asymmetry is greatest when T_{amb} is near the balance point; i.e. where conduction through the envelope is negligible. At other outdoor temperatures the sensible loads acting on all three rooms are relatively uniform and have little effect on the ΔT's between rooms. They simply add to the overall heating or cooling requirement. The most extreme asymmetric loads (e.g. a 2 kW oven) tend to be temporary, and can be handled by opening windows or by a kitchen exhaust fan, which during summer would impose a small energy penalty by increasing sensible and latent loads.

3 CENTRALIZED A/C SYSTEM

The preceding analysis showed how a small ventilation system distributing the minimum 1.5 cmm of fresh air to living area, flowing through rooms before returning via kitchen and bathrooms, can carry enough additional energy through open doors to keep room-to-room temperature differences around 1°C, even in the presence of the worst-case asymmetric solar loading. Interestingly, the greatest asymmetric loading may be imposed by the heating system. Consider for example a dedicated outside air system (DOAS) in which the peak heating load is met by warming only the 1.5 cmm ventilation air to 60°C. In that case the room receiving the supply air will be 3°C warmer than the opposite end of the house where the return air duct is located. Off-design ΔT's will be lower, due to cooler supply air.

As ambient air temperature rises above the balance point, thermal comfort can be maintained naturally by opening windows, or mechanically by increasing the ventilation rate above 1.5 cmm. In the latter case at T_{amb} ~20°C about 10 cmm would be required to hold indoor air temperature below 25°C. On warmer days air conditioning would be required.

3.1 Configuration

Figure 4 shows the placement of the [sensible and latent] energy recovery device in the ventilation air stream. It affects not only the magnitude of the loads but also alters the sensible heat ratio. Also shown is the recirculation loop, which is sized to carry the required air flow across the indoor heat exchanger as described in the following subsection.

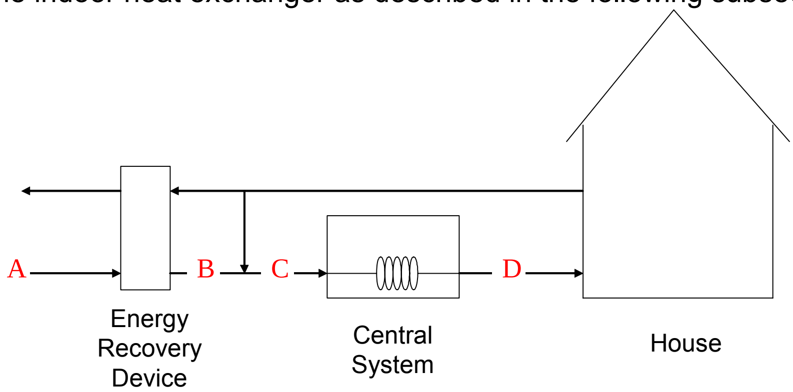

Figure 4. Centralized system configuration (Labels A, B, C, and D refer to possible desiccant dehumidifier locations as discussed in section 4.1)

3.2 Meeting sensible and latent loads

In air-conditioning (a/c) mode, in order for an evaporator to meet peak sensible and latent cooling loads, the refrigerant flow rate (compressor speed) must be sufficient to meet the total load, and the air flow rate must be adjusted to produce an evaporator surface temperature low enough to remove the latent load. The values of these two variables are uniquely determined by the magnitudes of sensible and latent loads. At the summer peak condition required air flow rate is 13 cmm for the low energy building, far greater than the minimum fresh air ventilation requirement. Therefore more extensive recirculation ductwork will be required, and room-to-room temperature maldistribution will not be an issue.

At the winter design condition supply air is usually delivered around 40°C, in the interest of maximizing heat pump efficiency without sacrificing comfort. However it is possible to meet the building's heating load by heating only the 1.5 cmm ventilation air stream to a 60°C supply air temperature instead of the "normal" 40°C. Thus in a cold climate where air conditioning is not required, a dedicated outdoor air system (DOAS) with the minimal air duct system may suffice. However the asymmetric configuration of the simple duct layout shown in Figure 2 would result in a 3°C temperature difference between the warmest and coldest room on the coldest winter night.

The same DOAS system at the summer peak condition would be very inefficient, because of the requirement to bring 13 cmm of fresh air into the building, compared to the 1.5 cmm required for ventilation (Table 3). Even with the energy recovery device operating, this would increase sensible load by 20% and latent load by about a factor of 3. Thus a recirculation air loop is required in warm humid climates, and the 13 cmm flow rate through the house is sufficient to offset large asymmetric loads and equalize room-to-room temperature differences during all seasons. Note that the total air flow rate, 13 cmm, is still less than half the air flow rate needed in the baseline house.

Table 3. Dedicated outside air system at sumer peak load

		Pure DOAS	Central system w/recirculation	units
Evaporator area	A_evap	11	10.4	[m^2]
Cycle COP	COP_cyc	5.4	5.5	[-]
System COP	COP_sys	4.1	4.5	[-]
Sensible capacity	Q_dot_sens	2.5	2.1	[kW]
Latent capacity	Q_dot_lat	1.3	0.43	[kW]
Vent air flow	V_dot_fresh	13	1.6	[cmm]
Recirculation	V_dot_recirc	0	11	[cmm]
Total air flow	V_dot_total	13	12.6	[cmm]
Blower power	W_dot_blower	0.144	0.057	[kW]
Fan power	W_dot_fan	0.048	0.033	[kW]
Compressor power	W_dot_comp	0.692	0.464	[kW]
System power	W_dot_total	0.919	0.566	[kW]

Thus in a central system with a single evaporator meeting simultaneously the sensible and latent a/c loads, the evaporator must operate below the dew point of the indoor air (~15°C) at all times. When the sensible load ratio is low (e.g. <<50% as on hot humid summer nights when ambient temperature approaches the dew point) the evaporating temperature may need to be quite low in order to meet the latent load. On the other hand when sensible loads exceed 80% the system can operate only a few degrees below 15°C. A variable speed compressor, blower and fan are necessary to maximize system efficiency while maintaining comfort across this entire range of sensible and latent load ratios.

3.3 Separating sensible and latent loads

Instead of dealing with sensible and latent loads simultaneously, it is possible to handle them separately and operate with greater efficiency. One option is to install a small a/c system to dehumidify only the ventilation air stream downstream of the energy recovery device before it mixes with the recirculated air. Another is to use a desiccant to dehumidify the ventilation air stream. In both cases this would allow for sensible-only cooling of air at the central system's evaporator, which would operate above the indoor air dewpoint, thus increasing the cycle COP. However the increased blower power needed to move more air could offset most or all of the compressor power savings, resulting in little or no net gain in system COP.

These tradeoffs will be addressed in the next section, which quantifies the potential for using decentralized heat pumps instead of a single central system. By eliminating most duct losses, the separation of sensible and latent loads becomes a more attractive option due to the larger evaporator air flow rates required when operating above the dewpoint.

4 COMPARISON WITH DECENTRALIZED A/C SYSTEM

The greatest advantage of ductless (mini-split) systems is obvious from their name. Asymmetric thermal loads can be handled by modulating capacities of the respective units instead of relying on extensive ductwork (and associated blower power requirement) to equalize temperatures between rooms. In poorly insulated buildings there is another substantial benefit, namely the energy savings realized by heating or cooling only the occupied rooms (zones). However such savings only materialize if large room-to-room temperature differences can be maintained within the building. As the foregoing analysis has shown, temperatures quickly equalize in low energy houses because of their lower envelope loads – heat transfer through a room's interior walls and door is the same order of magnitude as that through the exterior walls.

Ductless split systems by themselves are significantly more efficient than their centralized counterparts, because duct losses are limited to the ventilation system with its very low flow rate.. Table 3 shows that for a centralized system the blower accounts for about 10% of total system power; about half of that is dissipated in the ducts. All else being equal, decentralized systems therefore have about a 5% efficiency advantage.

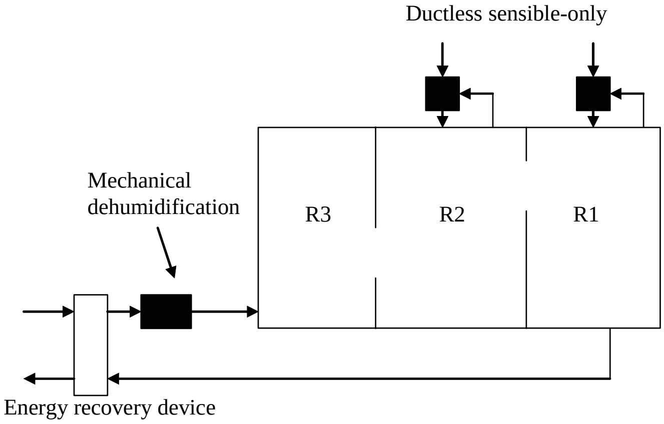

Figure 5. Decentralized system components

Maximum capacities of the ductless split systems can be selected and located in a manner that accommodates anticipated load nonuniformities, and takes advantage of the ventilation system layout and the convective couplings through doorways to serve more than one room. Figure 5 shows a typical decentralized system configuration, one with a mechanical dehumidifier handling all the latent loads in the ventilation air stream and at the same time providing sensible cooling to Room 3. This allows the other two units to operate above the dewpoint in sensible-only mode, supplying a greater volume of cool air at a higher discharge temperature.

4.1 Desiccant dehumidification

Desiccant dehumidification is an option for both central and decentralized systems. The latent heat released during adsorption, from the larger system's perspective, simply converts latent to sensible load by heating the air. Theoretically the energy needed to regenerate the desiccant, by rejecting water vapor to the exhaust air stream, can be supplied by the condenser. The required temperature of the regeneration heat supply differs among various solid desiccant materials, and the availability of sufficient high-temperature heat depends on the type of heat pump. For example a transcritical CO_2 heat pump will reject much of its heat above 40°C, even at moderate outdoor ambient temperatures, while a subcritical system will reject only the desuperheating fraction of its condenser heat at such a high temperature. Adding suction line heat exchange would increase the amount high temperature heat rejected from either system.

Figure 4 shows four possible locations for installing a desiccant dehumidifier in the supply air duct of a central system. Simulations showed clearly that location A would be most efficient for two reasons. First by releasing the heat of adsorption into the ambient air upstream of the energy recovery device, that additional heat can be rejected immediately into the exhaust air stream. In this implementation the energy recovery device is sensible-only, to avoid conflict with the desiccant. Placing the desiccant at location B would also degrade performance of the energy recovery device. The second reason for selecting location A instead of C or D is to avoid the pressure drop penalty caused by recirculating 13 cmm through the desiccant instead of only the 1.5 cmm ventilation air.

For a decentralized system such as that shown in Figure 5, the desiccant would be placed upstream of a sensible-only energy recovery device for the reason stated above, replacing the mechanical dehumidifier. However with mini-split systems, it would be more difficult to recover free condenser heat to regenerate the desiccant. In that case it may make economic sense to use a multi-split system with a single condensing unit supplying multiple evaporators.

4.2 Peak cooling performance comparison

Table 4 summarizes simulation results for our types of cooling systems for a cooling design day in St. Louis (T_{amb}=34 °C). These include centralized and decentralized systems with mechanical and desiccant dehumidification.

Table 4. Summary of cooling system performance

		Mechanical Dehumidification		Desiccant Dehumidification		
		Centralized System	Decentralized System	Centralized System	Decentralized System	units
Evaporator area	A_evap	10.4	13.3	11.01	20.2	[m^2]
Cycle COP	COP_cyc	5.5	5.5	5.5	6.7	[-]
System COP	COP_sys	4.5	4.6	4.2	5.1	[-]
Sensible capacity	Q_dot_sens	2.1	2.1	2.5	2.5	[kW]
Latent capacity	Q_dot_lat	0.43	0.41	0.43	0.39	[kW]
Air circulation rate	V_dot	13	21	13	36	[cmm]
Blower power	W_dot_blower	0.055	0.042	0.081	0.073	[kW]
Fan power	W_dot_fan	0.033	0.031	0.029	0.032	[kW]
Compressor power	W_dot_comp	0.464	0.450	0.402	0.372	[kW]
Total power	W_dot_sys	0.563	0.530	0.531	0.488	[kW]

All four configurations require similar amounts of total system power at the design cooling condition. This is not surprising since all sensible cooling, regardless of dehumidification method, is accomplished using a vapor-compression cycle. The small advantage found for the desiccant-based systems is based on the optimistic assumption that desiccant regeneration can be accomplished by recovering at least 39 W of waste heat from the outdoor units. In practice this may be difficult, diminishing the differences between the two dehumidification approaches.

Given a particular method of dehumidification, decentralized systems perform somewhat better than well-designed centralized counterparts, although total energy savings are on the order of 4-6%. Using mechanical dehumidification, this savings comes from the blower, due to the absence of interior ductwork, and from the compressors in the room a/c units, because they can operate at a higher suction pressure when performing sensible-only cooling. Desiccant dehumidification in a decentralized system requires even less power, mainly because the desiccant has converted 39 W of latent load into sensible load that can be removed by compressors operating at higher suction pressures in sensible-only mode. The compressor power savings are more than sufficient to offset the increased blower power, which is detailed in Table 5.

Table 5. Components of total blower power

	Mechanical Dehumidification		Desiccant Dehumidification		
	Centralized system	Decentralized system	Centralized system	Decentralized system	units
Evaporator coils	25	41	26	70	[W]
Energy wheel	4	4	4	4	[W]
Ducts	28	0	32	0	[W]
Desiccant	0	0	3	3	[W]
Total blower power	57	45	65	76	[W]

Two other types of systems were also investigated; both would handle latent loads in the ventilation air stream and employ sensible-only systems inside the building. The first option envisioned distributed wall panels with a small compressor inside, and the indoor and outdoor surfaces serving as natural convection heat exchangers. In order to achieve a significant (~10%) energy savings compared to mini-split systems, the evaporating temperature would need to be maintained only 1°C below the dewpoint-constrained wall surface temperature of 18°C and the small compressor would need to be as efficient as those in larger systems. The second type of system envisioned a secondary water loop running through ceiling or wall panels. However the pumping energy requirement, and even a 1°C temperature difference penalty between the primary and secondary loops would make it very difficult to operate more efficiently than decentralized split systems.

5 THERMAL CAPACITANCE

Both the building and its contents are capable of storing and releasing energy, thereby shifting and reducing peak building loads. The effect may be minimal in the case of the baseline house with its peak heating and cooling loads of about 9 kW and 5 kW, respectively. On the other hand with a low energy house, with peak sensible loads on the order of 2 kW, the effect may be substantial.

Table 6. Estimated thermal capacitance

	Walls	Contents
$m_{active\ thermal}$	6024 kg	1400 kg
c	1.1 kJ/kg-K	1.8 kJ/kg-K
ΔT_{swing}	3°C	3°C
Q_{store}	5.4 kWh	2.0 kWh
h_{avg}	2.0 W/m^2-K	1.6 W/m^2-K
$A_{surface}$	628 m^2	172 m^2
$T-T_{indoor}$	1.5°C	1.5°C
Q_{disch}	1.3 kW	0.4 kW

By making a few simple assumptions about the dimensions and thermal capacitance of gypsum board walls and ceilings, and the height and surface/volume ratio of its contents, it is possible to roughly quantify the contribution of thermal capacitance to meeting the ~2 kW peak heating load in the low energy house. Results in Table 6 for a 3°C temperature swing, show that both walls and contents can discharge their stored energy in about four hours. The implications for the heating and cooling seasons are discussed next.

Peak heating loads experienced by the low energy house, 2.2 kW, are smaller than peak domestic water heating load: ~12 kWh/day, recharging in a 4-hour period. Continued improvements in water-conserving plumbing fixtures and appliances may reduce this requirement, but taking it as an upper bound it is possible to envision a moderately oversized (4 kW) compressor in a central system providing both heat and hot water without sacrificing comfort on the winter design day. Since the thermal capacitance of the walls alone can provide 1.3 kW for 4 hours, the heat pump need only supply 0.9 kW for space heating while dedicating the remaining 3.1 kW capacity to recharging the hot water tank. Of course this is the worst case where the water storage is depleted at the peak heating hour. More commonly, the water storage could be replenished at frequent, shorter intervals during off-peak hours, diminishing the need for a substantially oversized compressor.

The same analysis applies to space heating with a DOAS of the kind described in Section 2; moderately oversizing the compressor would allow a single system to meet peak space and water heating demands.

In cooling mode, an integrated appliance such as this might be described as a water heater that provides free air conditioning for 4 hours per day, while the central a/c system's condenser rejects about 3 kW to the water storage tank. It is not completely free, however, since any subcritical (e.g.R410A) system will need to reject heat at an elevated condensing temperature in order to heat water to the desired temperature of 60°C.

6 CONCLUSIONS

The energy efficiencies of potentially promising HVAC technologies for low energy residences have been compared using simple simulation analyses based on transparent and easily modified assumptions. The unexpected result was that it is difficult to eliminate any of the candidate systems based on energy considerations alone. All were so efficient that differences in energy costs will likely be dominated by other factors such as cost and complexity.

Decentralized systems can offer a 4-6% energy savings over their centralized counterparts. A centralized system, however, may be easily integrated with the domestic hot water system. Unless future hot water demands are reduced through conservation, the compressor must be oversized to 4 kW to meet the 4-hr recharge constraint, and must operate in tandem with space heating during winter.

It was found that the use of desiccant dehumidification can reduce total system power by 8-10% over the same system utilizing mechanical dehumidification. However, this energy savings relies on the optimistic assumption that condenser waste heat may be used to accomplish desiccant regeneration. This may be difficult to accomplish in practice, especially on off-design days when condensers reject heat at cooler temperatures. If that is the case, the efficiencies of mechanical and dessicant dehumidification may be comparable, with the mechanical option favored on the basis of minimal maintenance cost.

Secondary loop systems and wall panels with integrated heat pumps were also investigated, and were found to yield significant energy savings only under the most optimistic assumptions. With wall surfaces maintained warmer than the indoor dewpoint, the driving temperature differential produces relatively low natural convection heat transfer coefficients. Therefore wall panel systems require separate dehumidification systems that are sufficiently robust to tolerate the nonuniformity of latent load generation within the building.

Since this analysis revealed little difference in energy efficiency among the air-source vapor-compression systems examined here, it may be useful to expand the range of systems considered, using the same simple modeling approach to examine systems employing various degrees of ground-coupling or solar assist.

7 ACKNOWLEDGEMENT

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