Efficient Low-Lift Cooling with Radiant Distribution, Thermal Storage, and Variable-Speed Chiller Controls— Part II: Annual Energy Use and Savings

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This paper evaluates the cooling efficiency improvements that can be achieved by integrating radiant cooling, cool storage, and variable-speed compressor and transport motor controls. Performance estimates of a baseline system and seven useful combinations of these three efficient low-lift inspired cooling technologies are reported. The technology configurations are simulated in a prototypical office building with three levels of envelope and balance-of-plant performance: standard-, mid- and high-performance, and in five climates. The standard performance level corresponds to ANSI/ASHRAE/IESNA Standard 90.1-2004 Energy Standard for Buildings Except Low-Rise Residential Buildings (ASHRAE 2004a). From the savings estimates for an office building prototype in five representative climates, estimates of national energy saving technical potential are developed. Component and subsystem models used in the energy simulations are developed in a companion paper.

INTRODUCTION

High performance can best be achieved by improving the efficiencies of all of the energy features of a given building in a vigorous, but balanced, manner. Use of high-performance envelopes can reduce heating energy in most United States' locations to little more than what is needed to heat the ventilation air plus what is needed for night setback recovery heating on a few mornings in the coldest locations. Ventilation air conditioning (heating, cooling, and [de]humidifying) can be largely satisfied by enthalpy exchange with exhaust air. Heating aside, electricity use in well-designed contemporary buildings is roughly equal parts lighting, cooling and air movement, and equipment operated by occupants. Lighting design, fixture efficacy, daylighting, and controls have reduced lighting power densities substantially and progress continues. Office equipment efficiencies have shown similar progress.

Reduction of cooling loads is a straightforward, albeit first-cost intensive, matter of improving window, window-shading, and envelope performance; of recovering ventilation enthalpy; of improving lighting efficiencies; and of reducing end-user equipment loads. To achieve balanced (equal marginal returns) energy efficiency investments, the high-performance building designer

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must aggressively address both cooling load sources and cooling system efficiency. Cooling system efficiency, in turn, requires attention to both thermodynamic performance and fluid transport performance. That efficient cooling is a significant technical challenge should not deter us. Potential impacts in the world's fastest growing economies are tremendous because growth is projected largely for tropical, subtropical, and desert regions where cooling is a basic need.

This paper will show that substantial improvements in cooling system efficiency can be achieved by integrating low-lift cooling technologies: variable-speed compressor and transport motor controls, radiant cooling with dedicated ventilation air transport and dehumidification, and cool storage. Estimates of cooling energy use and savings potential are developed for a baseline and all feasible combinations of the low-lift cooling technologies.

The three low-lift technologies or technology groups that can be implemented independently are described, and a brief description of why each is of interest is provided.

Radiant Cooling Subsystems (RCSs). RCSs save energy in three ways. First, by achieving a given level of comfort at a higher air temperature, the envelope load is reduced. Second, transport energy is reduced when cooling capacity is delivered by water instead of air. Finally, chilled water temperatures of 15°C or higher, instead of the 10°C needed by all-air systems for humidity control, result in higher chiller efficiency. A dedicated outdoor air system (DOAS) is required to provide fresh, dry air because RCSs cannot perform these functions. Splitting the latent and sensible cooling functions in this way results in better control of zone temperature, humidity and air quality, as well as in better cooling efficiency. The variable-air-volume (VAV) distribution system, being the most efficient all-air system used widely in the United States, serves as the baseline technology that RCSs and DOASs replace.

Thermal Energy Storage (TES). TES (intrinsic or discrete) saves energy by shifting cooling load to night and by spreading the load over time. Shifting the load to night reduces the condensing temperature regardless of what kind of chiller is used. Spreading the load over time can further reduce the average difference between chiller condensing and evaporating temperatures ($T_c - T_e$) if the refrigerant flow rate can be efficiently modulated. Along with foregoing savings mechanisms comes potential energy penalties in the form of storage heat-exchanger penalties that can lead to lower evaporating temperatures and transport energy penalties associated with discrete storage, which involve essentially doubling the transport costs for each charge-discharge cycle. If sufficient thermal capacitance exists in the building fabric and contents, load shifting can be achieved while avoiding both the heat-exchanger and double transport penalties. When thermal energy storage and load-shifting controls are absent (baseline case), the chiller must exactly satisfy all hourly cooling loads.

Variable-Speed (VS) Compressor and VS Transport. VS compressor and VS transport motor operation save energy by striking a balance between condensing-evaporating temperature difference and transport energy. While transport energy is directly tied to the flow-pressure characteristic of each heat transfer fluid circuit and is a very strong function of pump or fan speed, the trade-off between transport and compressor savings, governed by the condenser and evaporator approach temperature relations, is also very sensitive. The theoretical compressor energy savings from reduced condensing-evaporating temperature difference are significant but can easily be wiped out if per-unit compressor and motor losses increase as the flow rate is reduced. The sensitivity and nonlinearity of these trade-offs become more problematic, and therefore require more rigorous modeling, when the chiller system is modulated over a wide range of capacity. The baseline chiller capacity modulation technology chosen for this assessment is the two-speed chiller, which is roughly equivalent to a chiller with two equally sized compressors.

Low-lift savings are achieved through two distinct mechanisms: improved transport efficiency and improved thermodynamic efficiency. Thermodynamic efficiency is mainly a function of condensing and evaporating temperatures, determined in turn by condenser and evaporator approach temperatures and source (T_z) and sink (T_x) temperatures. The potential reduction in compressor work per unit cooling capacity is illustrated in the temperature-entropy (T-s) diagram of Figure 1. The areas of the cycle polygons represent compressor work and the areas under the polygons (extending beyond the plot viewport to absolute zero) represent cooling capacity. The large polygon is determined by condensing and evaporating temperatures typical of a conventional chiller and distribution system, while the smaller polygon is determined by condensing and evaporating temperatures typical (for the same daily total load) of the proposed low-lift system. The T-s diagram illustrates not only that compressor work is much smaller for the low-lift system, but that the cooling capacity of the low-lift system is about 10% larger than for the corresponding conventional system with identical compressor, condenser, and evaporator components.

The paper begins with an overview of the efficient cooling literature. The assessment applies three building performance levels to the same prototypical office building plan. Performance level parameters, including envelope parameters that are indexed to the five climates used in the assessment, are described. The seven combinations of the three low-lift cooling system elements used in the assessment are enumerated and performance maps, based on the baseline and low-lift chiller models developed in the companion paper (Armstrong et al. 2009), are presented. Results of the simulations are reported and discussed. The impact of peak-shifting (i.e., pre-cooling) controls on annual cooling load distribution are presented; these load distributions can be used by chiller manufacturers to guide and optimize designs for variable-speed low-lift equipment. A methodology for estimating potential national energy savings is described, and results of the assessment are presented. The analysis does not attempt to estimate cost-effective potential savings because costs of mass-produced and mass-marketed low-lift components are not yet known. The main contributions of this paper are the thorough exploration of parameter space (i.e., climate, HVAC[1] configuration, balance of system performance), the use of cooling equipment models that are consistent

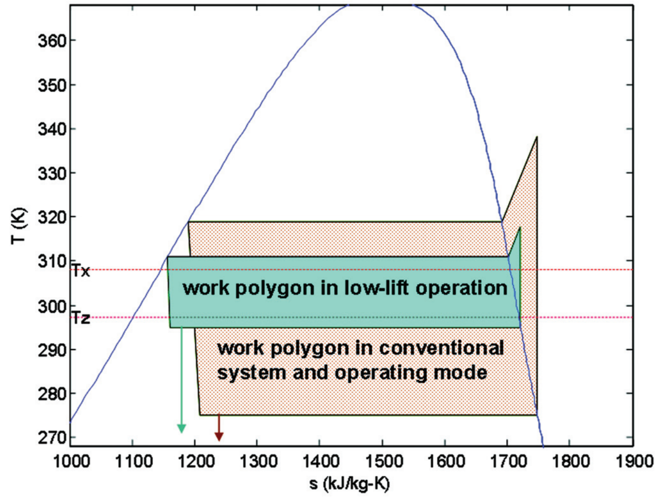

Figure 1. Vapor-compression cycles depicted for typical operating conditions of a conventional and a low-lift cooling system under the same daily total load. Low-lift operation has a slightly larger cooling effect, $T_e(s_{suc} - s_{liq})$, and much smaller work polygon, $\iint TsdTsd$.

over a wide range of lift and part-load fraction, and results that indicate a very feasible and very promising approach that can have tremendous impact on global warming potential in temperate as well as hot-climate regions of countries now experiencing rapid economic growth.

LITERATURE REVIEW

A wide-ranging literature review was performed to identify efficient cooling technologies. Cooling by natural ventilation was not addressed because of its climate-limited application range, but an effort was made to find all significant research, analysis, and design guidance germane to the low-lift technologies described above. The lack of a rigorous evaluation of chiller efficiency benefits potentially enabled by hydronic radiant cooling is surprising. No quantitative assessments of annual energy benefits for cooling plants designed to modulate efficiently over a wide range of capacity fraction (substantially greater than 2:1) were found, and no assessment of the three low-lift technology elements in combination was found. A summary of the relevant literature is presented below.

Radiant cooling was of interest 50 to 60 years ago when the evolving air-conditioning industry was still in the process of trying out a variety of systems—a process that eventually resulted in dominance of the all-air approach in North America. Current interest (from about 1990) in radiant cooling, initially motivated by fan energy savings, is also seen as a way to reduce ventilation flow rates and to control humidity better. Recent work by proponents has focused on accurately estimating panel capacity, on modeling the interactions of convection and radiation, and on supervisory control strategies of decoupled dehumidification/ventilation and sensible cooling systems that ensure comfort and acceptable indoor air quality (IAQ), while avoiding condensation on panels under all conditions. At least one paper has made credible estimates of fan-energy- and higher-air-temperature-effected savings. The loss of (i.e., ~80% of the normal) air-side free cooling potential has been noted, but the potential national impact across climates and seasons of this loss has not been addressed. The effectiveness of applying water- or refrigerant-side economizers, in conjunction with radiant cooling, is not well understood. The impact of higher evaporator temperatures on chiller performance has not been quantified. Radiant cooling systems are gaining market share faster in Asia and Europe than in the United States (Adlam 1947; ASHRAE 2004b; Ayoub et al. 2006; Baker 1960; Carpenter and Kokko 1998; Chantrasrisalai et al. 2003; Conroy and Mumma 2001; Dieckmann et al. 2004; Feustel and Stetiu 1995; Feustel 1999; Franta 1983; Jeong et al. 2003; Jeong and Mumma 2006; Jiang et al. 1992; Kallen 1982; Kilkis 1993, 2000; Kilkis et al. 1994, 1995; Kochendorfer 1996; Manley 1954; Mumma 2001a, 2001b; Novoselac and Srebric 2002; Shank and Mumma 2001; Shoemaker 1954; Simmonds 1994; Stetiu 1997, 1999; Zweifel and Koschenz 1993).

Work on active core cooling has addressed thermal occupancy conditions such as strong vertical temperature gradients. The considerable challenge of control during diurnal and shorter load transients has been addressed in a few papers. Potential improvements in system efficiency through active core cooling combined with peak-shifting and efficient low-lift chiller equipment have not been quantitatively assessed. Air-side free cooling possibilities are limited in radiant/DOAS systems. Refrigerant-side free-cooling has been mentioned, but details of design and performance were not found in the literature, probably because this design traditionally has had little attraction when used with all-air systems or systems with discrete storage (Ataer and Kilkis 1994; Athienitis and Shou 1991; Hauser et al. 2000; Kallen 1982; Kochendorfer 1996; Koschenz and Dorer 1999; Meierhans 1993, 1996; Michel and Isoardi 1993; Olesen 1997, 2000; Olesen et al. 2000; Simmonds et al. 2000; Strand and Pedersen 1997).

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[1] Although heating, ventilation, and air-conditioning subsystems are commonly lumped together under the moniker of HVAC equipment, this paper will not address the heating elements.

The ability of radiant ceiling panels (RCPs) to cool the building fabric and contents at night will be even more problematic, in terms of capacity, transport energy, and control, than for active core systems. Although no papers or reports were found to directly address the RCP precooling question, there is a body of residential-scale night cooling literature in which the small driving temperature differences involved are an oft-cited problem. Effective residential-scale ambient-coupled cool storage systems have been demonstrated in climates where ambient night cooling is feasible. Systems based on water, packed bed, intrinsic building thermal capacitance, and a variety of phase change materials with transition points above room temperature (heating) and below room temperature (cooling) have been developed. Storage schemes developed for residential ambient night cooling such as building mass, stratified water tanks, water tubes, phase change material (PCM) building materials, PCM-packed beds, and rock beds all have potential. Although it may have benefits in low-lift dehumidification, for low-lift sensible cooling ice storage is anathema (Balcomb 1983; Buddhi 2000; Hay and Yellott 1969; Karaki 1978; Peck et al. 1979; Perkins 1984; Stoecker et al. 1981; Telkes and Raymond 1949; Turner and Chen 1987).

In commercial buildings, energy savings (not just demand savings) with discrete or intrinsic thermal energy storage and off-the-shelf (in some cases with modified controls) vapor-compression cooling equipment have been successfully demonstrated in a few large (>100,000 ft^2) buildings. Large buildings have the advantage that fan speed and static pressure can typically be adjusted by the control system to cool a building at night by ambient air or with a chiller running near its maximum efficiency point. Results of night precooling have been less successful where plant part-load and low-ambient efficiencies are less remarkable. Most existing small buildings are in this category, because night operation of fans results in significant additional energy use. The system efficiency of a plant designed for 50°F (10°C) supply air does not improve much as ambient temperature drops, even at the optimal part-load point (Andresen and Brandemuehl 1992; Armstrong et al. 2006; Aynur et al. 2006; Brandemuehl et al. 1990; Braun 1990, 2003, 2006; Braun et al. 2001; Braun et al. 2002; Braun and Chaturvedi 2002; Braun and Mercer 2003; Braun and Zhong 2005; Calobrisi et al. 1980; CEC 1996; Chiu and Zaloudek 1987; Conniff 1991; Eto 1984; Henze et al. 1997; Kallen 1982; Keeney and Braun 1996, 1997; Morris et al. 1994; Perkins 1984; Rabl and Norford 1988; Ruud et al. 1990; Stoecker et al. 1981; Sun et al. 2006).

Work on DOASs have focused on proper control over the wide range of outdoor conditions that such systems face, and on performance-cost-pressure-drop considerations—trade-offs to which all-air system A/C plants are particularly sensitive. Mention of design integration problems is conspicuously absent from the literature, as one might expect, because DOASs have been treated as providing a predominantly stand-alone function. Supervisory control of reheat and determination of the optimal DOAS supply air temperature are areas of interaction with radiant cooling that may need further study (Conroy and Mumma 2001; Shank and Mumma 2001; Zweifel and Koschenz 1993). Note that Adlam (1947) documents two working radiant cooling/DOAS systems, based on technologies available at the time, including both a solid desiccant wheel and liquid-desiccant dehumidification system.

A Department of Energy (DOE) Commercial Unitary Air Conditioner analysis (CUAC 2004) found that the best path for improving package equipment efficiencies from EER-10 to EER-12 was to increase evaporator and condenser size, thus reducing the condensing-evaporating temperature difference. Reducing compressor speed is equivalent to increasing evaporator and condenser size. Most variable-speed centrifugal compressors still require inlet prerotation vanes—and suffer the associated losses—to modulate below about 50% capacity fraction. Various other methods of capacity modulation, all less efficient than state-of-the-art variable-speed motors/drives, have been extensively researched. However, the modeling literature has little to say about variable-speed performance of positive displacement compressors. Although increased evaporating temperature is often mentioned as an advantage of radiant cooling, its impact on chiller efficiency in conjunction with other improvements has not been analyzed. Similarly, the potential for temperature glide (i.e, the Lorentz cycle) to reduce pressure ratio and transport flow requirements simultaneously is known, but system-level assessments are lacking (ADL 2000; Aynur et al. 2006; Bendapudi and Braun 2002; Brasz 2006; Bullard and Radermacher 1994; Conry et al. 2002; CUAC 2004; Garland 1980; Hiller 1976; Hu and Yang 2005; IEA 1996; IIR/Kruse 1992; Jahnig et al. 2000; Jin 2002; Lewis 1990; Stoecker et al. 1986; Takebayashi et al. 1994; TIAX 2002; Tassou and Qureshi 1994; Yilmaz 2003).

Variable-speed compressor, fan, and pump operation has the potential to substantially improve low-ambient and part-load cooling performance. Optimal control based on component performance models, heretofore applied mainly to centrifugal chillers, has been referred to as *[optimal] static chiller control.* Static chiller control has been applied to large plants, both with and without discrete cool storage. Large plants are predominantly centrifugal chiller plants for which the efficient modulating range is limited to about 2.5:1. Reciprocating and rolling-piston compressors are generally better suited for efficient, wide-range variable-speed operation than centrifugal compressors and may also be better suited for an efficient, wide range of pressure ratio operation than screw and scroll compressors. The energy savings potential of combining radiant cooling with a chiller that can modulate efficiently down to low-capacity fractions and is also optimized for low-pressure ratio conditions has not been addressed. Guidelines for variable-speed chilled water pump and supply fan operation abound but primarily in the context of large, centrifugal-chiller-based plants (Aynur et al. 2006; Bahnfleth and Peyer 2004; Brasz 2006; Braun et al. 1987a, 1987b, 1989a, 1989b, 1989c; Braun 1990; Coad 1985; Englander and Norford 1992; Garland 1980; Gordon and Ng 2000; Hardaway 1982; Hartman 2001; Henze et al. 1997; Hydeman et al. 2003; King and Potter 1998; Kintner-Meyer and Emery 1995; Tassou and Qureshi 1994).

It is difficult to predict what low-lift cooling would cost in a commodity market—or even in substantial niche markets—because DOASs and radiant distribution equipment are not currently produced in volume. However, high-end residential and small commercial cooling and heat-pump markets in Japan are now dominated by variable-refrigerant-volume equipment that does not cost much more than comparable two-speed equipment. There have also been a number of market assessments for TES systems and efficient cooling equipment that generally show a good tradeoff between capital and operating costs (Brown and Spanner 1988; Emmerich and McDowell 2005; Hattrup 1988; Itron 2006; Katipamula and Gaines 2003, Reed et al. 2004; Sodec 1999; SWA 2004; Weijo and Brown 1988).

In summary, the literature reveals a great deal of effort and progress toward characterizing, developing, and demonstrating individual efficient cooling technologies. However, there are some important gaps. It is almost impossible to find compressor and chiller performance data and models for low speed (less than 50% of nominal) and low pressure ratio (less than 1.6) operation. And there is a need to examine multiple efficient cooling technologies—especially technologies that appear to be highly complementary—by system-level simulations to determine which combinations work best in various building designs and climates.

BUILDING DESCRIPTION

To assess low-lift benefits, a baseline cooling system, each low-lift technology alone, and each possible combination were simulated in a prototypical building. Two sets of office building prototypes were reviewed for the current work: the Lawrence Berkeley National Laboratory's (LBNL's) prototypes (Huang and Franconi 1999) and the Advanced Energy Design Guide (AEDG) small and medium office prototypes (Jarnagin et al. 2006). The AEDG prototypes were selected for this analysis because the DOE-2 building models were subject to an extensive consensus process within ASHRAE Special Projects Working Group 102, which resulted in building descriptions with configuration and energy performance features typical of new construction. The SP102 working group developed two baseline office building prototypes—a 5000 ft^2 light-frame building and a 20,000 ft^2 two-story skeleton-frame building—for the AEDG work, as shown in Figure 2.

The 20,000 ft^2 building, referred to as the *medium office*, was selected for this assessment. The medium office prototype is a two-story building with one core and four perimeter zones representative of a broad class of commercial construction. Because zone load is primarily a function of exterior wall orientation, each zone is modeled to include both the first and second floor spaces. Each zone is served by a packaged rooftop unit.

The original AEDG prototypes were designed to be in compliance with *ANSI/ASHRAE/IESNA Standard 90.1-1999, Energy Standard for Buildings Except Low-Rise Residential Buildings* (ASHRAE 1999). The medium office prototype was updated to comply with ANSI/ASHRAE/IESNA Standard 90.1-2004 (ASHRAE 2004a), and the analysis grid presented in Table 1 was developed. Three component performance levels—standard, mid, and high—were thus defined in each of the following analysis grid dimensions:

- wall and roof U-factors
- window performance (U-factor and solar heat gain coefficient [SHGC])
- window-wall ratio and shading
- light and plug load power density
- specific fan power

The medium office prototype was simulated using DOE-2.2 to produce cooling load time series for five representative climate zones. Standard-, mid-, and high-performance versions of the medium office prototype were simulated resulting in 15 combinations (3 × 5) of building and climate for the analysis of annual HVAC energy consumption.

The selected five climate locations—Baltimore, Chicago, Houston, Los Angeles, and Memphis—represent the five key climate zones used by the DOE in its building energy codes development work (Briggs et al. 2002). About three-fourths of the United States' population is covered by the five DOE climate zones. The five representative cities capture significant variability of both outside-air temperature and humidity, and of day length and sun-sky conditions. Chicago (N41.8°) has cold and dry winters and represents the single most populous climate zone. Los Angeles (N33.9°) is warm and dry during most of the year and represents the southwestern U.S. maritime

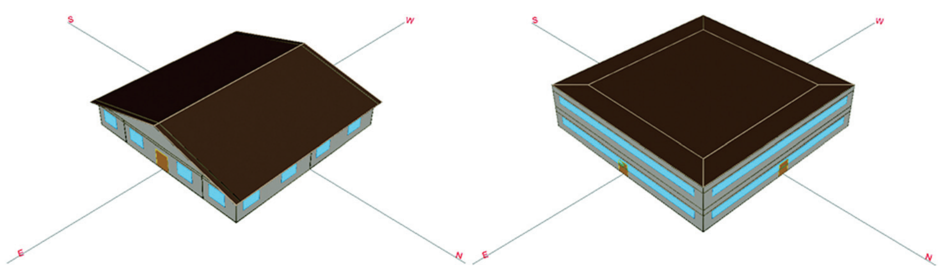

Figure 2. Basic geometry of AEDG small and medium office prototypes (not to scale).

region. Memphis (N35.1°) and Baltimore (N39.2°) have mild weather and represent the middle latitudes of the country. Houston (N30.0°) represents the hot and humid climates found along the Gulf and southern Atlantic coastal regions.

CHILLER, DISTRIBUTION, AND TES CONFIGURATIONS

Performance map models of the key components—chiller, DOASs, and radiant panels—were developed for use with hourly building (medium office) cooling loads simulated by DOE-2.2. The modeling and simulation experimental plan is described below. Modeling of chiller and distribution system energy is done through post-processing of the hourly building cooling loads. Chiller and distribution subsystem models are described in the companion paper. The use of purpose-built performance models ensures that calculated performance differences are the result of the intended configuration differences alone; other cooling and ventilation components are identical across all system configurations.

The base case is assumed to use a VAV air-handling unit (AHU) because this is generally the most energy efficient type of all-air distribution system. Thus, the base HVAC system is modeled as a VAV no-reheat system fed by a small two-speed chiller to condition the occupied spaces of the building with no load shifting. In addition to the base system, seven alternative HVAC system configurations are formed by taking all possible combinations of the three low-lift technology elements described in the introduction:

Case 1: two-speed chiller with VAV AHU—the base-case HVAC configuration

Case 2: variable-speed chiller and VAV AHU—this configuration uses the base-case VAV AHU but with variable-speed chiller (variable-speed compressor, pump, and fan) equipment

Case 3: two-speed chiller with RCP/DOAS—this configuration assumes the base case two-speed chiller but with a hydronic distribution system serving radiant cooling/heating panels and a DOAS to condition ventilation air

Case 4: two-speed chiller with VAV AHU and TES—this is the base case system modified to use an idealized TES for load shifting

Case 5: variable-speed chiller with RCP/DOAS—combines the alternatives provided separately in Case 2 and Case 3 (low-lift variable-speed chiller and RCP/DOAS)

Case 6: variable-speed chiller, VAV AHU and TES—this is the Case 2 system modified to use an idealized TES for load shifting

Table 1. Analysis Grid for Simulating 8,760-Hour Cooling Load Sequences

	Component Performance Levels to be Analyzed		
Component	**Standard**	**Mid Performance**	**High Performance**
Wall-roof U-factor	90.1-2004[a]	2/3 of 90.1-2004	4/9 of 90.1-2004
Window U-factor and SHGC	90.1-2004[a]	2/3 of 90.1-2004	4/9 of 90.1-2004
Window-to-wall ratio	40%	20%	20%+Shading[b]
Light and plug loads[c] (W/sf)	1.3 + 0.63	0.87 + 0.42	0.58 + 0.21
Fan power (W/scfm)[d]	0.8	0.533	0.356

(a) Values change with climate zones.
(b) Completely shade all windows from solar direct beam.
(c) Power density during hours of the highest loads defined in the DOE-2.2 weekly load schedules.
(d) Total HVAC fan power divided by total HVAC fan flow rate.

Case 7: two-speed chiller with RCPs/DOASs and TES—this is the Case 3 system modified to use an idealized TES for load shifting

Case 8: low-lift variable-speed chiller with RCPs/DOASs and TES—the complete low-lift technology incorporating variable-speed chiller, RCPs/DOASs and idealized TES

Cases 2, 5, 6, and 8 use advanced variable-speed compressor and transport (fan and pump) controls to optimize the instantaneous hourly operation of the chiller and distribution systems. Low-lift is achieved by closer (on average) evaporator and condenser approach temperatures.

Cases 3, 5, 7, and 8 use RCPs/DOASs to raise the average chiller refrigerant evaporating temperature. Higher average evaporating temperature is a key element of low-lift operation.

Cases 4, 6, 7, and 8 implement a 24-hour look-ahead algorithm to optimize charging of the TES. Low-lift operation is achieved partly by lower average condensing temperature and partly by closer approach temperatures in the evaporator and condenser at lower average capacity fraction.

Note that Cases 2, 3, and 4 each employ one of the three low-lift technology elements, Cases 5, 6, and 7 each combine two of the three technology elements, while Case 8 combines all three.

The low-lift technology elements were assessed individually as well as in all possible combinations to demonstrate the synergisms of the technology elements, and thus to illustrate the importance of careful systems integration in achieving exceptional energy performance.

Three versions of the chiller model were developed to produce the two chiller performance maps shown in Figure 3. The VAV system uses an air-side economizer so its chiller performance map does not extend to air-side economizer conditions. The map has three regions computed by the same chiller model but corresponding to the three chilled water supply temperature scheduled outdoor temperature ranges defined in Appendix G of ANSI/ASHRAE/IESNA Standard 90.1-2004 (ASHRAE 2004a). The VAV system capacity fraction is given in terms of rated capacity (ARI 2003). The second performance map, for the RCP system, includes both compressor and refrigerant-side economizer operation. The chiller model for economizer operation uses the same components as the chiller for compressor operation, except that the compressor is replaced by the flow-pressure characteristic of a bypass valve during economizer operation. At each performance evaluation, the two RCP system maps are evaluated and the mode of operation (compressor or

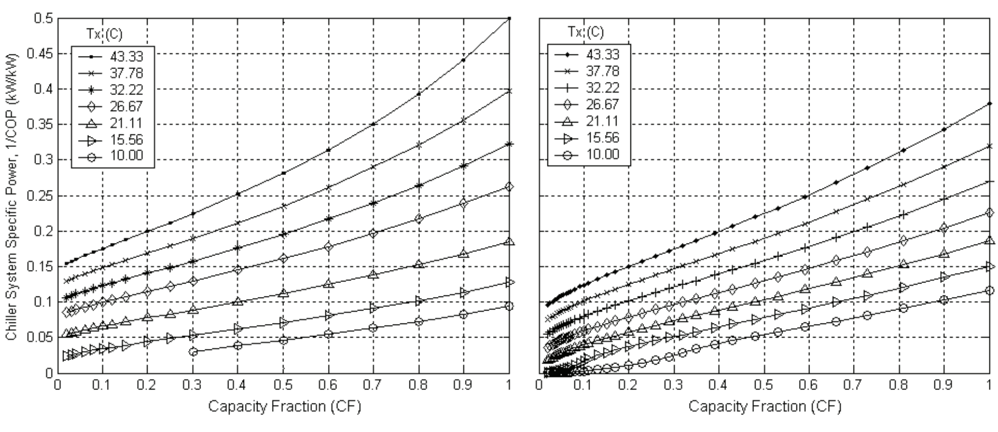

Figure 3. Chiller system performance maps for VAV Cases 1, 2, 4, and 6 (left), and for RCP Cases 3, 5, 7, and 8 (right). RCP-chiller transition to refrigerant-side economizer operation is evident at $CF \sim 0.15$ on the 15.56°C line and at $CF \sim 0.35$ on the 10.00°C line.

economizer) is determined by which map evaluation returns the lower *1/*COP (kW/kW$_{th}$ or kW/ton). The RCP system capacity fraction is given in terms of full-speed capacity at outdoor temperature T_x = 35°C (95°F) and indoor temperature T_z = 22.2°C (72°F).

Two-speed operation of the compressor, condenser fan, and chilled water pump is simulated by performance curves derived from the corresponding variable-speed performance map. The low- and high-speed-specific power curves—functions of outdoor temperature only—are obtained by evaluating the corresponding variable-speed performance map at capacity fraction equals 0.5 and 1.0.

Building sensible heat ratio (SHR) is determined by the standard DOE-2.2 cooling coil model. Normally, SHR would differ slightly among the baseline and seven low-lift configurations to the extent that inlet and coil surface temperatures differ. However, for this study the DOE-2.2 time-series of hourly SHR were applied to all cases so that differences in efficiency with SHR would not affect comparative results. Enthalpy recovery ventilation was also modeled by DOE-2.2 using a constant-effectiveness model. The latent load that remained after enthalpy recovery was satisfied by a DX dehumidifier modeled as two subsystems: the wetted evaporator coil and a scaled-down version of the variable-speed chiller with heat rejection to the ventilation supply air. Resulting sensible load was added to the building sensible load and could therefore be treated as peak-shiftable load. Airflow and fan power were determined by ventilation demand, while compressor power was determined by evaporator inlet conditions and the latent load remaining after enthalpy recovery.

The annual energy simulations use DOE-2.2-generated load sequences to which DOAS reheat has been added for the cases that use DOASs. This and two other aspects of the RCPs to VAV comparison present unavoidable difficulties. The annual cooling load presented to the chiller is less for VAV cases to the extent of air-side economizer operation. Conversely, loads are less for the RCP/DOAS cases to the extent that enthalpy recovery reduces ventilation loads. Annual cooling loads for RCP cases thus differ by up to ±15% from the corresponding VAV loads.

For systems without TES, the appropriate (VAV or RCPs) chiller map is applied directly to the corresponding baseline load sequence for the climate and building performance level of interest. For systems with TES, the idealized storage described in the companion paper with no losses, no storage carryover from one day to the next, and capacity sufficient for the peak cooling day is assumed. The problem is further simplified by using energy, not energy cost, as the objective function. This approach is generally most sensitive to equipment part-load performance. It is also in keeping with the net-zero-energy project motivation. Impacts of realistic deviations from the ideal storage model are being addressed in ongoing work.

Given a reliable and complete chiller performance map, COP $= f(T_x, T_Z, Q_{Load})$, there is a sequence of 24-hourly chiller cooling rates $Q(t)$, which minimizes the energy input for one day of operation:

$$\text{Minimize} \quad J = \sum_{t=1}^{24} \frac{Q(t)}{\mathrm{COP}(t)} \tag{1}$$

subject to just satisfying the daily load requirement:

$$\sum_{t=1}^{24} Q_{Load}(t) = \sum_{t=1}^{24} Q(t) \tag{2}$$

and to the capacity constraints:

$$0 \le Q(t) \le Q_{Cap}(T_x(t), T_z(t)) \qquad t = 1{:}24$$

where

COP	$= f(T_x, T_Z, Q)$ = chiller coefficient of performance ($kW_{cooling}/kW_e$)
T_x	= outdoor dry-bulb temperature
T_z	= zone temperature
Q	= evaporator heat rate—positive for cooling (Btu/h, ton, or kW_{th})
Q_{Load}	= building cooling load with no peak-shifting
Q_{Cap}	$= f(T_x, T_z)$ = chiller cooling capacity at full-speed operation

Note that the idealized storage model involves no interaction between hourly cooling rate and indoor temperature; the indoor temperature trajectory for the TES cases is assumed to be the same as for the baseline (no TES) case. The idealized TES model eliminates the need to model a transient room temperature trajectory but requires the daily total load constraint instead.

Given a building's hour-by-hour cooling load trajectory and the outdoor dry-bulb[2] temperature trajectory, the optimization through the whole year in 365 one-day blocks can be run to find the hourly chiller cooling rates (i.e., what we call the *peak-shifted load trajectory*) that minimize ventilation, cooling, and transport input energy for the year.

Equipment sizing is specific to each building performance level and climate. The performance map models are linearly scaled so that scaled capacity under design conditions satisfies the peak hourly load obtained from the corresponding DOE-2.2 annual simulation. The same scaling factor is used for the TES and non-TES cases, even though the full capacity may not be used in the TES cases.

ENERGY SAVINGS POTENTIAL

Energy use estimates for the eight low-lift technology configurations and the three building performance levels are presented in this section. Simulation results are summarized, in terms of annual energy required to operate the HVAC equipment, in Figures 4 through 6 and Table 2.

The percent energy savings (i.e., percent of HVAC energy) for seven low-lift technology configurations (Cases 2 through 8) with respect to the base case (Case 1) are shown in Table 3 for the three building performance levels. For each row, percent savings are computed with reference to the corresponding Case 1 energy consumption. Note that performance of the technology configurations generally improves in the order that the cases were originally presented. Notable exceptions are that Cases 4 and 5 are consistently transposed and the situation for the mild Los Angeles climate is quite mixed.

Results for the baseline building (Figure 4) show that the annual energy savings for the radiant cooling panel system with variable-speed chiller and ideal thermal storage compared to the VAV system with two-speed chiller range from 74% for a hot climate (represented by Houston) to 70% for milder cooling climates (represented by Los Angeles and Chicago). Note that the savings for the full low-lift technology compared to the next best partial low-lift technology—in which the chiller operates at two speeds instead of in full variable-speed mode—are significant, ranging from 27% (Houston) to more than 32% (Los Angeles). Note also that RCPs/DOASs perform the best out of all partial low-lift technology systems involving one element, and TES with RCPs/DOASs perform the best out of all systems involving two elements.

Results for the building with a mid-performance building design (Figure 5) fall between those of the baseline and high-performance building and are similar to the percent savings

[2] One could use a linear combination of wet- and dry-bulb temperatures and an appropriate chiller-with-cooling-tower performance map to estimate energy use for low-lift system configurations that use a water-cooled chiller.

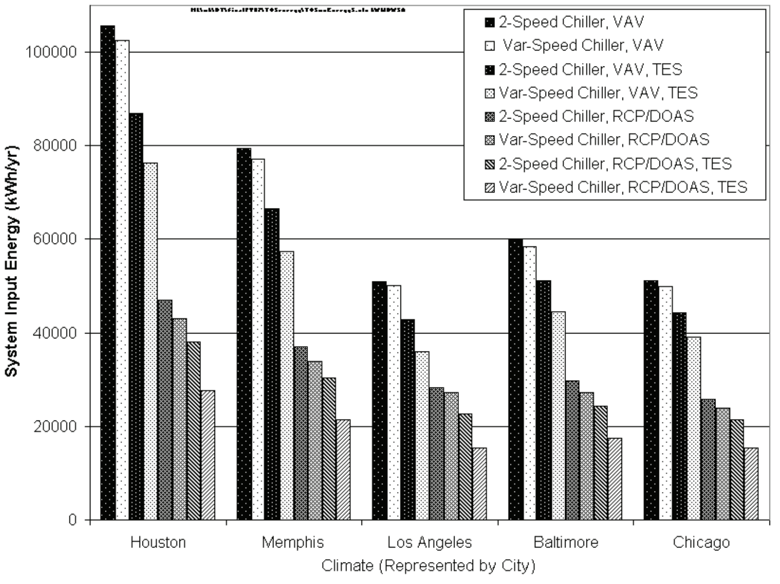

Figure 4. Results of standard-performance building annual energy simulations for different chiller-distribution-thermal-storage system configurations in five climates.

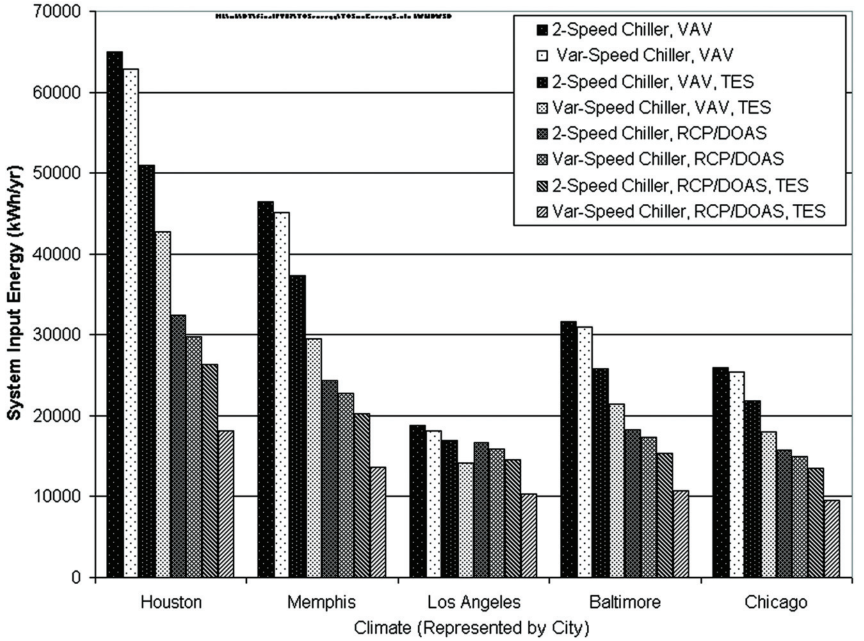

Figure 5. Results of mid-performance building annual energy simulations for different chiller-distribution-thermal-storage system configurations in five climates.

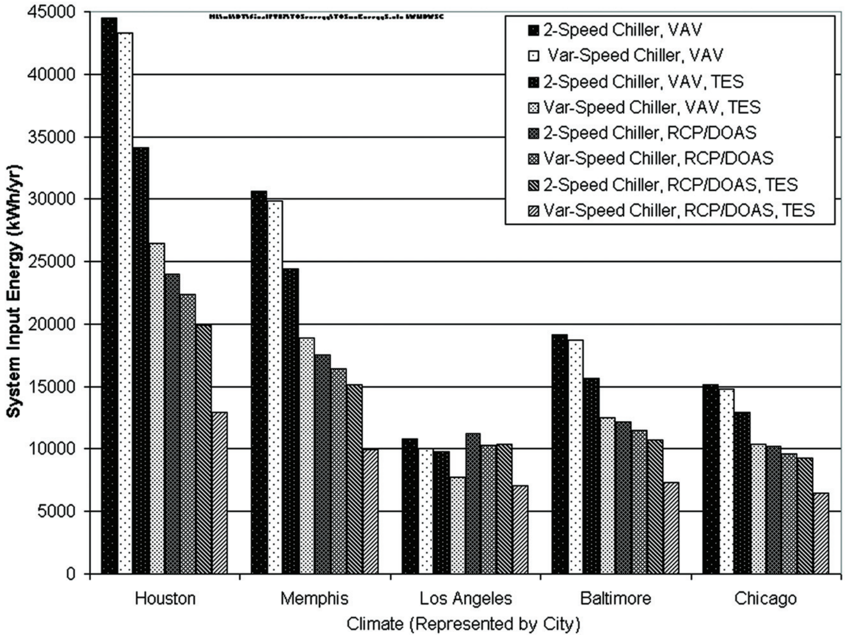

Figure 6. Results of high-performance building annual energy simulations for different chiller-distribution-thermal-storage system configurations in five climates.

numbers for the baseline building, except in the mild Los Angeles climate. The energy saved by the full low-lift technology is 73% for Houston and 63% for Chicago. However, the corresponding energy saving for Los Angeles is only 45.5%. This reflects two things: 1) that in a mild climate, HVAC energy is strongly affected by economizer operation; and 2) that for the reduced specific-fan-power design of the mid-performance building, the air-side economizer (VAV) cases benefit from a substantial reduction in transport energy, whereas in the refrigerant-side economizer (RCPs/DOASs) cases, performance is unchanged.

The mid-performance and base building rankings with respect to equipment configuration are quite similar. The savings for the full, low-lift HVAC configuration compared to the next best partial configuration—in which the chiller operates at two-speeds instead of full variable-speed mode—are actually a bit larger than for the base building, ranging from 29.5% (Chicago) to 32.6% (Memphis). The RCPs/DOASs configuration still performs best of the partial low-lift configurations involving one element, and TES with RCPs/DOASs still perform the best of all of the systems involving two elements.

Results for the building with the highest level of envelope, lighting, and office equipment performance (Figure 6) are similar to the mid-performance building results, except with a further closing of the ranks for the economizer-compatible climates—Los Angeles, Baltimore, and Chicago. The savings for the full low-lift configuration are 71% for Houston, 57% for Chicago, and 34.5% for Los Angeles.

The percent savings for the full low-lift configuration compared to the next best partial configuration are significantly better than those of the baseline and mid-performance buildings, ranging from 30% (Chicago) to 35% (Houston). The RCP/DOAS configuration again performs best of all of the partial low-lift systems involving one element, and TES with RCP/DOAS still

Table 2. Annual Energy Use (kWh) for Eight Cooling/Ventilation System Configurations and Three Prototypical Office Building Balance-of-System Performance Levels

Building Performance Level	Climate	Case 1	Case 2	Case 3	Case 5	Case 4	Case 6	Case 7	Case 8
Standard performance (ASHRAE 90.1-2004)	Houston	105,555	102,454	86,958	76,325	47,055	42,993	37,973	27,583
	Memphis	79,488	77,099	66,574	57,312	37,012	33,839	30,295	21,323
	Los Angeles	50,915	50,142	42,811	35,934	28,310	27,141	22,722	15,348
	Baltimore	59,933	58,344	51,148	44,564	29,638	27,268	24,233	17,404
	Chicago	51,213	49,822	44,333	39,097	25,829	23,855	21,315	15,361
Mid performance	Houston	64,973	62,870	50,949	42,693	32,363	29,729	26,331	18,158
	Memphis	46,467	45,169	37,278	29,554	24,379	22,696	20,306	13,681
	Los Angeles	18,844	18,078	16,967	14,145	16,683	15,886	14,572	10,264
	Baltimore	31,677	30,911	25,781	21,465	18,243	17,274	15,353	10,663
	Chicago	26,000	25,387	21,788	18,037	15,697	14,889	13,478	9509
High performance	Houston	44,525	43,305	34,129	26,484	23,965	22,369	19,902	12,950
	Memphis	30,621	29,878	24,392	18,843	17,498	16,390	15,159	9959
	Los Angeles	10,790	10,064	9784	7772	11,215	10,326	10,355	7071
	Baltimore	19,143	18,711	15,633	12,518	12,130	11,459	10,745	7347
	Chicago	15,146	14,791	12,915	10,341	10,216	9640	9312	6502

performs the best of all of the systems involving two elements. For Los Angeles, however, VAV is retained in the best-performing one- and two-element configurations. This reflects the very low value of specific fan power assumed for the high-performance building, which benefits the air-side economizer (VAV cases), whereas the refrigerant-side economizer (RCP/DOAS) performance is again unchanged. Thus, the best partial configuration involving one element in Los Angeles is the TES configuration. Although the idealized TES model makes no distinction between storage types, the effectiveness of intrinsic storage in the Los Angeles high-performance building, together with its very low cost, make the intrinsic storage approach attractive for high-performance buildings in this climate.

CHILLER PERFORMANCE AND LOAD DISTRIBUTION WITH PEAK SHIFTING

Thermal energy storage can be used to shift a significant portion of the sensible cooling load to night-time hours. The benefit of shifting some of the chiller load into night-time hours is to present an altered cooling load distribution so that the chiller can operate at a lower average capacity fraction and can also provide a larger portion of the daily total load at cooler night and early morning temperatures. This is in stark contrast to the traditional chiller or package A/C mode of operation, where cooling loads are concentrated around times of peak outdoor-air temperature.

Figure 7 shows typical chiller output trajectories for Chicago. The no-TES variable-speed VAV and RCPs trajectories have been omitted for clarity. Compared to the baseline load trajectory for a typical VAV system, the optimally peak-shifted load trajectory for a variable-speed chiller and radiant panel cooling system is flattened by shifting part of the daytime cooling load to night.

Table 3. Percent Energy Savings with Respect to Case 1 for Seven Cooling/Ventilation System Configurations and Three Building Performance Levels

Building Performance Level	Climate	Case 2	Case 3	Case 5	Case 4	Case 6	Case 7	Case 8
Standard performance (ASHRAE 90.1-2004)	Houston	3%	18%	28%	55%	59%	64%	74%
	Memphis	3%	16%	28%	53%	57%	62%	73%
	Los Angeles	2%	16%	29%	44%	47%	55%	70%
	Baltimore	3%	15%	26%	51%	55%	60%	71%
	Chicago	3%	13%	24%	50%	53%	58%	70%
Mid performance	Houston	3%	22%	34%	50%	54%	59%	72%
	Memphis	3%	20%	36%	48%	51%	56%	71%
	Los Angeles	4%	10%	25%	11%	16%	23%	46%
	Baltimore	2%	19%	32%	42%	45%	52%	66%
	Chicago	2%	16%	31%	40%	43%	48%	63%
High performance	Houston	3%	23%	41%	46%	50%	55%	71%
	Memphis	2%	20%	38%	43%	46%	50%	67%
	Los Angeles	7%	9%	28%	-4%	4%	4%	34%
	Baltimore	2%	18%	35%	37%	40%	44%	62%
	Chicago	2%	15%	32%	33%	36%	39%	57%

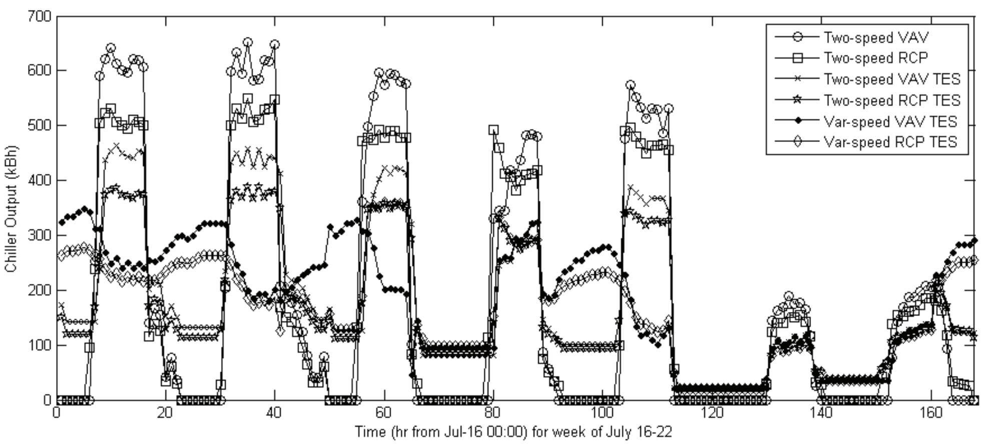

Figure 7. Typical summer cooling load trajectories for week of July 16–22 in Chicago.

Some aspects of the performance trade-off between free cooling and low-lift operation are also illustrated in Figure 7. On the Thursday morning shown, RCPs systems don't perform as well as VAV systems, even when they don't have TES; this reflects the coincidence of cooling load and free cooling potential. The interaction of night free cooling potential and daytime cooling load is evident in the Wednesday–Friday chiller output trajectories for the last two TES cases, variable-speed VAV with TES, and variable-speed RCPs with TES.

Figure 8 and Table 4 show the baseline cooling load distribution for the medium office at different temperature bins and part-load ratios for Chicago. Figure 9 and Table 5 show the optimally peak-shifted cooling load distribution for the medium office using a variable-speed chiller, RCP/DOAS system, and idealized TES. These two surface plots illustrate the distribution of annual cooling load, expressed in full-load-equivalent operating hours (FLEOH), as a function of outdoor dry-bulb temperature and part-load ratio.

Compared to the baseline load distribution, the operation hours for the full low-lift configuration (variable-speed chiller, RCPs/DOASs, and TES) system are significantly shifted in three respects. First, the bin in which the FLEOH peak occurs is typically 15°F (8.3°C) lower than in the baseline chiller load distribution; second, although the chiller does continue to operate at high outdoor temperatures, it does so at much lower part-load ratios; and third, the FLEOHs of operation at low outdoor temperature and low part-load ratio are significantly increased.

NATIONAL ENERGY SAVINGS

To estimate the national energy savings potential requires a translation from savings per building to savings across all (or some representative defined set of) buildings. A simplified process was developed to scale the percent savings computed in the previous section. This process consists of the following: 1) mapping the 16 climate locations typically used in DOE technical analyses to the five climate zones used in the current analysis, and 2) identifying building types that are both suitable for application of low-lift cooling technologies and for

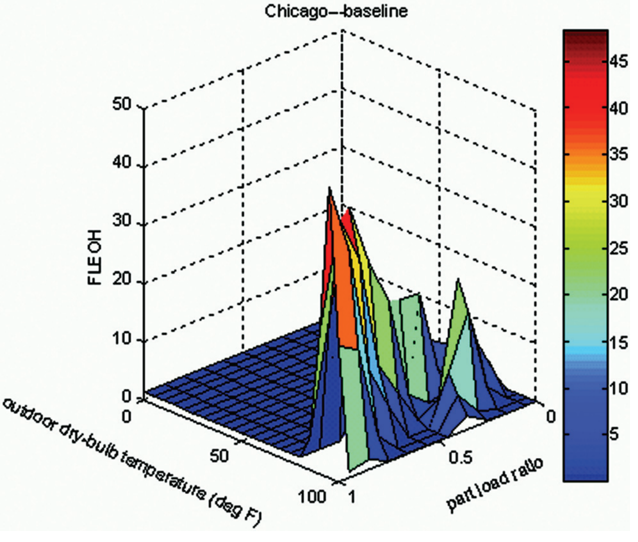

Figure 8. Baseline building sensible cooling load distribution for Chicago.

Table 4. Annual FLEOHs by Capacity Fraction (across) and Outdoor Dry-Bulb Temperature (down)

T_{ODB} (°F) bin upper bound	Σ over CF	Capacity Fraction (CF) 0.05	0.15	0.25	0.35	0.45	0.55	0.65	0.75	0.85	0.95
Σ over T_{ODB}	**→↓→**	**16.75**	**42.19**	**77.5**	**56.72**	**26.53**	**44.2**	**55.57**	**91.34**	**121.01**	**98.11**
35	0	0	0	0	0	0	0	0	0	0	0
40	0.25	0.25	0	0	0	0	0	0	0	0	0
45	0.53	0.27	0.26	0	0	0	0	0	0	0	0
50	8.61	0.73	3.64	2	2.24	0	0	0	0	0	0
55	13.62	0.6	5.05	6	1.97	0	0	0	0	0	0
60	19.21	1.06	3.81	7.22	7.12	0	0	0	0	0	0
65	6.09	1.44	1.12	2.59	0.94	0	0	0	0	0	0
70	26.02	4.22	7.67	9.97	3.34	0.82	0	0	0	0	0
75	68.47	0.36	1.29	5.65	19.9	20.2	15.3	5.77	0	0	0
80	139.4	4.96	5.92	2.68	1.77	4.6	22.1	29.9	40.9	22.9	3.7
85	155.7	2.86	10.1	22.5	3.11	0.91	6.8	13.2	32.4	40.5	23.3
90	134.8	0	3.33	17.5	8.47	0	0	6.7	14.4	36	48.4
95	51.86	0	0	1.14	5.38	0	0	0	3.64	19.9	21.8
100	5.35	0	0	0.25	2.48	0	0	0	0	1.71	0.91

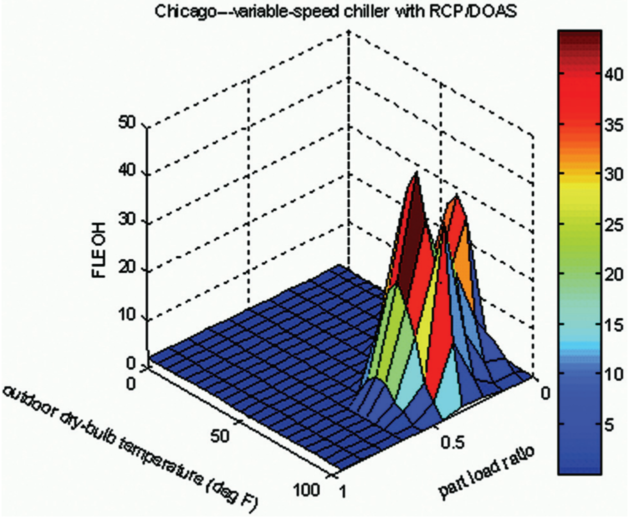

Figure 9. Building peak-shifted cooling load distribution for variable-speed chiller with RCP/DOAS system for Chicago.

Table 5. Annual FLEOHs by Capacity Fraction (across) and Outdoor Dry-Bulb Temperature (down) for the Variable-Speed Chiller with RCPs/DOASs and TES in Chicago

T_{ODB} (°F) bin upper bound	Σ over CF	Capacity Fraction (CF) 0.05	0.15	0.25	0.35	0.45	0.55	0.65	0.75	0.85	0.95
Σ over T_{ODB}	→↓→	**94.7**	**169.1**	**140.1**	**238.0**	**98.4**	**18.9**	**0**	**0**	**0**	**0**
10	0.02	0.02	0	0	0	0	0	0	0	0	0
15	0.07	0.07	0	0	0	0	0	0	0	0	0
20	0.16	0.16	0	0	0	0	0	0	0	0	0
25	0.12	0.12	0	0	0	0	0	0	0	0	0
30	0.66	0.66	0	0	0	0	0	0	0	0	0
35	1.68	1.68	0	0	0	0	0	0	0	0	0
40	4.28	4.13	0.15	0	0	0	0	0	0	0	0
45	10.74	7.27	2.11	1.36	0	0	0	0	0	0	0
50	17.84	8.12	4.48	5.24	0	0	0	0	0	0	0
55	27.17	10.5	10.6	4.81	1.26	0	0	0	0	0	0
60	53.38	15.7	16.4	12	8	1.28	0	0	0	0	0
65	90.90	12.9	19.3	16.8	30.5	11.4	0	0	0	0	0
70	127.2	11	32.6	20	39.7	20.1	3.75	0	0	0	0
75	137.6	8.5	35.4	20	44.4	23.6	5.73	0	0	0	0
80	117.8	5.76	31.7	20.3	35.4	19.5	5.11	0	0	0	0
85	83.73	4.49	11.7	23.4	27.5	14	2.64	0	0	0	0
90	62.67	2.69	3.83	11.8	37	5.72	1.63	0	0	0	0
95	20.78	0.72	0.6	3.41	13.2	2.85	0	0	0	0	0
100	2.39	0.18	0.25	0.95	1.01	0	0	0	0	0	0

which it is reasonable to estimate the potential total energy savings based on cooling system percent energy savings estimates. National savings potential will be estimated only for the new[3] commercial building types that satisfy both.

The simplified scaling process requires knowing the distribution of new commercial building floor area by climate zone. The area weighting factors used in the process were obtained from a National Renewable Energy Laboratory (NREL) study (Long 2007), which evaluated the proposed new ASHRAE Standard 189 (ASHRAE 2007) first public review draft. In the NREL study, energy use for 15 commercial building prototypes was simulated for 16 different U.S. climate zones. The building definitions used in this study were drawn from a draft set of buildings developed under a separate DOE/NREL research project being done to create benchmark EnergyPlus models for typical new construction. The set of climate zones used in the DOE/NREL analysis was developed for that early effort.

[3] Retrofit of existing buildings is technically feasible but not, in general, as life-cycle cost effective.

Table 6. Benchmark Building Prototype Areas

Building Type	Conditioned Floor Area, ft^2 (m^2)
Fast food	5046 (469)
Grocery	31,495 (2927)
Hospital	661,912, (61,516)
Hotel	292,780 (27,210)
Large office	673,167 (62,562)
Medium office	61,773 (5741)
Motel	39,500 (3671)
Outpatient health care	42,793 (3977)
Primary education	73,577 (6838)
Restaurant	17,732 (1648)
Retail	86,586 (8047)
Secondary education	166,134 (15,440)
Small office	21,025 (1954)
Strip mall	1,125,335 (104,585)
Warehouse	189,290 (17,592)

Table 6 lists the conditioned floor areas used by the NREL in defining the 15 prototype commercial buildings. Note that the NREL benchmark small office floor area of 21,025 ft^2 (1954 m^2) is comparable to the medium office floor area of 20,000 ft^2 (1859 m^2) used in the AEDGs and in this report.

As part of the benchmark research, the NREL developed floor area estimates based on value of new construction from the 2002 Economic Census, square foot cost models from R.S. Means, 2003 CBECS (EIA 2003), and 2002 to 2003 population growth data. The economic data were mapped to climate zones using geographical information systems (GIS) tools. The total square footage estimates for each building type and climate zone are shown in Tables 7 and 8. These floor area weighting factor estimates are used for the national energy savings calculation.

Climate Zone Mapping. Figure 10 illustrates how the 16 climate zones in the NREL's study are mapped to the five climate zones in this study. Climate conditions such as cooling degree days and humidity conditions were used to determine what climate zone to map to. Among the 16 climate zones used to cover the entire country, nine climate zones mapped to the five climate zones in this study. These nine climate zones cover 91% of total new construction of commercial building stock. Phoenix (Zone 2B) and Las Vegas (Zone 3B-other) were not included because these two climate zones are hot and dry climates that cannot be represented by any of the five climate zones used for the current study. Duluth (Zone 7) and Fairbanks (Zone 8) were not included because buildings in such cold climates do not have significant cooling that cannot be satisfied by outdoor-air economizer equipment. San Francisco (Zone 3C), Seattle (Zone 4C), and Helena (Zone 6B) are all cool climates that weren't included because well-designed buildings in these climates also get most of their cooling loads satisfied by application of outdoor air economizers.

Building Type Mapping. Table 9 lists the 15 building types used in NREL's study. Although some low-lift elements can be applied to all the building types, the energy savings estimation is

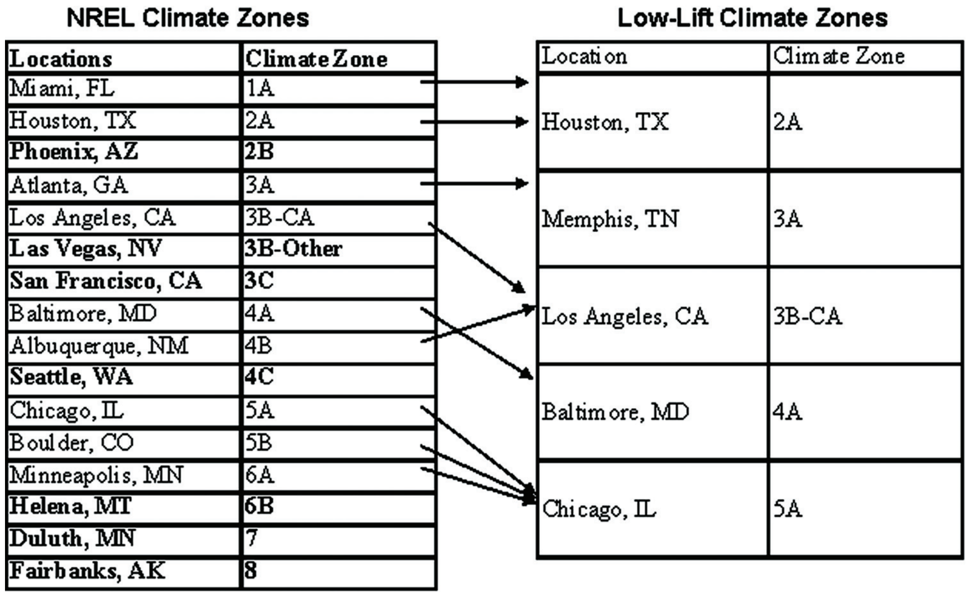

Figure 10. Mapping between NREL climate zones and low-lift climate zones (bolded climate zones were not included).

an office building can be reliably applied to all other building types. We therefore included only building types that satisfy both of the following requirements:

1. The full low-lift technology suite must be applicable to the building type (e.g., 24-hour hospital operations have little load-shifting potential).
2. The use schedules, internal gains, and envelope characteristics should be reasonably similar to those of the medium-office building prototype.

Table 9 shows whether a particular building type is suitable for the full low-lift application and whether the potential savings from the application can be reasonably represented using the results from the medium-office building modeled. Among these 15 building types, eight types—representing about 75% of the new construction commercial building stock by floor area—were included in the national energy savings estimation. Fast food and restaurant buildings were not included because these two building types often have very peaky loads at certain time periods and would require the system to respond quickly to large load changes. Therefore, these types are usually not good applications for building-intrinsic thermal storage, radiant cooling, or DOASs. The energy use for grocery buildings is usually dominated by the refrigeration systems, so the application of a percentage savings for office buildings to grocery stores would not be accurate. Further, grocery store cooling energy requirements are often low as a result of the presence of refrigeration equipment continuously removing load from the space.

The full, low-lift technology configuration can be applied to portions of hospital and hotel building types; however, it is difficult to estimate the potential savings in these two building types from the office building results because their 24/7 operation schedule is very different from the assumed office occupancy schedule. The motel buildings often use packaged terminal air conditioners that must respond quickly to unpredictable occupancy changes, so this building

Table 7. New Building Floor Area (ksf) Built Each Year by Type in Zones 1, 2, and 3

Building Type	Climate Zone						
	1A	2A	2B	3A	3B-CA	3B-O	3C
Fast food	144	1019	225	1159	1084	173	50
Grocery	431	3058	674	3477	3253	520	148
Hospital	993	8737	1655	14,099	8208	1191	397
Hotel	2342	9750	1962	8198	8725	2928	410
Large office	2154	18,849	4241	21,137	24,099	2693	1077
Medium office	2261	19,792	4466	22,207	25,315	2817	1161
Motel	1323	5487	1098	4614	4910	1651	225
Outpatient	655	5811	1091	9380	5452	800	248
Primary edu	3377	22,993	4312	24,501	21,779	3326	1001
Restaurant	647	4585	1011	5215	4880	780	223
Retail	2017	14,269	3143	16,226	15,187	2424	693
Secondary edu	2758	18,806	3522	20,052	17,810	2725	814
Small office	1104	9655	2178	10,834	12,348	1375	566
Strip mall	3939	28,021	6189	31,847	29,822	4726	1350
Warehouse	3275	16,941	3029	19	26,576	5376	1211
Total	27,420	187,773	38,795	192,966	209,447	33,505	9575

type was also excluded. Warehouse buildings were not included because only the office portions of most warehouses are cooled.

In summary, the energy use calculated for each of the cases from simulations can be applied to nine climate zones and eight building prototypes to estimate the national technical energy savings representing approximately 69% of floor area of total new commercial building stock.

National Energy Results. The annual national energy savings potential (cooling, fan, and pump) was estimated by applying the previously described methodology to the energy savings estimated for the medium-office building for each building performance level and for the five selected climate locations, which were described in the case study section. Table 11 summarizes the national energy savings for the full low-lift technology suite (Case 8), compared to the conventional VAV system with two-speed chiller (Case 1). Note that these annual estimates are for new construction and building types and climate locations for which the full low-lift technology suite is applicable (see previous section). Although it is likely that parts of the low-lift technology suite are applicable for a large portion of the existing commercial building stock and the full low-lift technology suite may be applicable to a substantial fraction of the existing building stock, that potential wasn't estimated in this study, because the primary market—as with most advanced building systems—is new construction. In this sense, the technical potential presented here is conservative. In addition, the savings estimates are for cooling systems (chiller, fan, and pumps) only. If the heating systems savings were to be included, the total savings estimates would again be higher. The savings are also expressed as a percentage of baseline cooling energy use for each building performance level.

For baseline buildings that are compliant with ANSI/ASHRAE/IESNA Standard 90.1-2004 (ASHRAE 2004a), the full low-lift technology suite saves about 0.0098 quads of site electricity

Table 8. New Building Floor Area (ksf) Built Each Year by Type in Zones 4 through 8

Building Type	Climate Zone								
	4A	4B	4C	5A	5B	6A	6B	7	8
Fast food	2031	64	232	2044	516	575	9	81	5
Grocery	6091	192	696	6132	1546	1726	25	243	0
Hospital	23,961	463	1920	22,439	3971	5891	728	530	0
Hotel	13,878	410	1698	14,844	4831	3221	732	439	0
Large office	47,862	1212	5991	42,477	9559	12,050	1414	1683	0
Medium office	50,339	1260	6332	44,643	10,057	12,701	1507	1736	62
Motel	7797	229	964	8358	2718	1809	403	241	0
Outpatient	15,983	304	1288	14,973	2653	3920	501	372	0
Primary edu	53,660	1464	5783	62,857	9697	10,794	1405	2391	221
Restaurant	9137	289	1043	9199	2319	2587	39	364	18
Retail	28,426	900	3238	28,617	7221	8044	121	1134	0
Secondary edu	43,909	1196	4735	51,435	7941	8838	1146	1960	166
Small office	24,557	614	3089	21,778	4907	6194	734	845	42
Strip mall	55,817	1801	6414	56,267	14,179	15,755	225	2251	0
Warehouse	36,325	1003	5149	38,085	11,887	10,638	1798	1495	0
Total	419,773	11,402	48,571	424,148	94,004	104,741	10,787	15,765	513

Table 9. Building Types Included in National Energy Savings Analysis

Building Type	Included	Reasons for Not Including
Fast food	No	Not suitable for TES application.
Grocery	No	Load profile not well represented by office buildings and relatively low space cooling energy use.
Hospital	No	Load profile not well represented by office buildings.
Hotel	No	
Large office	Yes	
Medium office	Yes	
Motel	No	Not suitable for TES application.
Outpatient health care	Yes	
Primary education	Yes	
Restaurant	No	Not suitable for TES application.
Retail	Yes	
Secondary education	Yes	
Small office	Yes	
Strip mall	Yes	
Warehouse	No	Low space cooling energy use.

Table 10. Summary of National Technical Site Electricity Savings Potential for the Year 2007 for Low-Lift Cooling (Variable-Speed, RCPs/DOASs) with TES Assuming 100% Penetration

Building Performance Level	National Electricity Savings for Operation of Cooling Equipment	
	Quad/Year	Percentage
90.1-2004 compliant building	0.0098	71%
Mid-performance building	0.0048	64%
High-performance building	0.0028	58%

Table 11. Summary of Total National Potential Site Electricity Savings and Potential Source Energy Savings in 2020 for Low-Lift Cooling with TES Assuming 100% Penetration (Quad/Year)

Building Performance Level	Site	Source
Baseline building	0.146	0.44
Mid-performance building	0.072	0.22
High-performance building	0.042	0.13

use in one year of new construction, with the technology being applied to approximately 69% of floor area of total 2007 U.S. new commercial building stock (assuming 100% penetration). The annual site electricity savings are about 0.0048 quads for mid-performance buildings and 0.0028 quads for high-performance buildings.

The annual national technical energy savings for different system configurations compared to the conventional VAV system with two-speed chiller (Case 1) are shown in Figure 11. For baseline buildings, the savings range from 0.0004 quads for variable-speed chiller system configured with conventional VAV distribution to 0.0098 quads for the full technology suite.

Assuming the new construction growth rates (1%) remain the same for the next 14 years (through the year 2020), the total national technical site energy savings potential (again assuming 100% penetration) for the baseline building would be 0.146 quads (Table 11). To reiterate, all of these savings are in site energy terms. To calculate source energy savings at the power plant, using average fossil-steam heat rates, the previous estimates should be multiplied by 3.[4] The total savings potential—relative to the baseline building—is therefore 0.44 quads.[5]

The national technical energy savings in 2020 for different system configurations compared to a conventional VAV system with two-speed chiller are shown in Figure 12. For baseline buildings, the savings range from 0.006 quads for variable-speed chiller system configured with conventional VAV distribution (Case 1) to 0.146 quads for the full technology suite (Case 8).

CONCLUSION

Electrical power for HVAC, which in most buildings translates to electrical power for cooling (compressors and package equipment) and transport (pumps and fans), may be treated as the

[4] Per the *2007 Buildings Energy Databook* (DOE 2007), the stock average fossil fuel steam heat rate (Btu/kW_e) will be 10,181 in 2020—see http://buildingsdatabook.eren.doe.gov/docs/6.2.5.pdf. This compares to the energy units relation of 3412 Btu/kW_{th}, about a factor of three difference.

[5] For reference, 10^{15} Btu/yr (1 quad or 1.05 x 10^{18} Joule/yr) is equivalent to the output of 47 gigawatts of coal-fired power plant capacity at current heat rates and capacity factors.

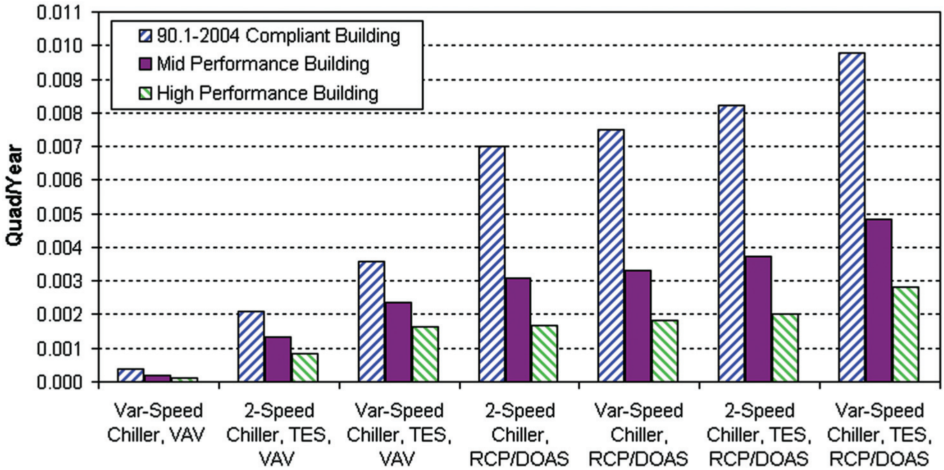

Figure 11. National technical site electricity savings over the conventional VAV system with two-speed chiller and no TES (Case 1) for different system configurations in the year 2007 assuming 100% penetration of one year's new construction.

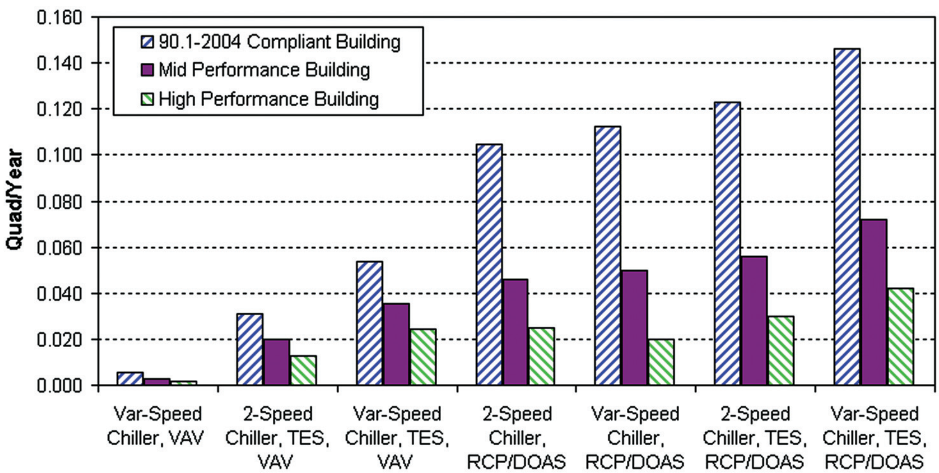

Figure 12. National technical site electricity savings in 2020 over the conventional VAV system with two-speed chiller and no TES (Case 1) for different system configurations assuming 100% penetration of new construction over a 14-year period.

understood as a matter of improving window, window-shading, and envelope performance; of recovering ventilation enthalpy and better controlling ventilation rates; of improving lighting efficiencies; and of reducing end-user equipment loads.

This analysis shows that significant cooling system efficiency gains can be achieved by integrating low-lift cooling technologies: variable-speed compressor and transport motor controls, radiant cooling with dedicated ventilation air transport and dehumidification, and cool storage.

The cooling energy savings for a standard-performance building range from 70% to 74% and, for a very high-performance building, from 34% to 71%.

Cooling plant savings result from efficient compressor operation at low-pressure ratios and over a wide speed range. Compressor and chiller performance in these regions has not been given much attention. The chiller and DX-dehumidifier equipment modeled in the analysis exhibit performance typical of existing package equipment at typical design conditions but represent a significant improvement in performance under part-load and low-lift conditions because compressor and transport motor speeds were independently controlled for optimal performance.

Low-lift operation does not benefit much from two well known, but costly and complex, measures: multistage compression and liquid recycle or other form of intercooling. Low discharge temperature is achieved instead by low suction superheat, low internal pressure drops, large heat-transfer capacity per unit refrigerant mass flow, and the external design factors—RCPs, night precooling, and VS compressor operation—that result in low pressure ratios.

The three low-lift technologies, when combined, result in consistently large savings in spite of wide variations in savings when applied one at a time. For example, the RCPs/DOASs element alone results in savings of between 9% and 23%. The VS chiller alone results in savings of only 2% to 7%. But when a VS chiller is added to HVAC configurations that already include RCPs/DOASs and/or TES, the incremental savings range (except in mild Los Angeles) from 20% to more than 50%.

The variable-speed savings, when added after TES, are largest because the load-shifting process results in almost all the load being shifted from a high to a low part-load operating range, where a variable-speed reciprocating chiller becomes very efficient. Even the best variable-speed centrifugal chillers start to lose efficiency below about 35% rated capacity (Conry et al. 2002). Load distributions of the type presented in Tables 4 and 5 contain the application-specific information needed to guide and optimize designs for variable-speed low-lift equipment.

The proper design and integration of low-lift technologies requires careful attention to controls. Controls, in turn, can become a maintenance issue with associated loss, over time, of system efficiency. Integrated delivery of the low-lift system, similar to the approach used for VS DX cooling equipment, is one possible way to address both of these issues. However, for broadest market penetration, it would be preferable for manufacturers to supply integrated controls with less of a black box approach. A controls package with options that permit flexibility in terms of hydronic distribution (e.g., active-core, ceiling panels, or the two combined), and in the coordination of RCPs and DOASs, would be extremely desirable.

The foregoing analysis is based on the use of vapor-compression equipment for both the sensible and latent cooling loads. Similar low-lift benefits can be expected with absorption cooling plants and thermally regenerated desiccant dehumidification equipment. The role of TES will generally be diminished in solar-powered cooling applications. It would be interesting, nevertheless, to compare the solar aperture area needed for a state-of-the-art solar-thermal-powered absorption and desiccant cooling system to the apertures needed by state-of-the-art photovoltaic-powered and state-of-the-art solar-thermal-turbine-powered vapor-compression systems for the standard-, mid-, and high-performance building prototypes simulated in a few desert and sun-belt climates.

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REFERENCES

Adlam, T.N. 1947. *Radiant Heating; A Practical Treatise on American and European Practices in the Design and Installation of Systems for Radiant, Panel, or Infra-Red Heating, Snow Melting and Radiant Cooling.* New York: The Industrial Press.

ADL. 2000. Energy efficient rooftop air conditioner: continuation application technical progress statement. Prj.#: DE-FC26-99FT40640, Arthur D. Little, Inc., Cambridge, MA.

Andresen I., and M.J. Brandemuehl. 1992. Heat storage in building thermal mass—a parametric study. *ASHRAE Transactions* 98(1):910–18.

ARI. 2003. *Standard 550/590: Performance Rating of Water Chilling Packages Using the Vapor Compression Cycle.* Arlington, VA: Air Conditioning and Refrigeration Institute.

Armstrong, P.R., W. Jiang, D. Winiarski, S. Katipamula, L.K. Norford, and R.A. Willingham. 2009. Efficient low-lift cooling with radiant distribution, thermal storage and variable-speed chiller controls—Part I: Component and subsystem models. *HVAC&R Research* 15(2):366–401.

Armstrong, P.R., L.K. Norford, and S.B. Leeb. 2006. Control with building mass—Part I: Thermal response model identification and Part II: Simulation. *ASHRAE Transactions* 112(1):449–61.

ARTI. 2004. Research Roadmap, Air-conditioning and Refrigeration Technology Institute. www.arti-21cr.org/documents/roadmap.pdf.

ASHRAE. 1999. *ANSI/ASHRAE/IESNA Standard 90.1-1999, Energy Standard for Buildings Except Low-Rise Residential Buildings*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 2004a. *ANSI/ASHRAE/IESNA Standard 90.1-2004, Energy Standard for Buildings Except Low-Rise Residential Buildings*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 2004b. *BSR/ASHRAE Standard 138P, Method of Testing for Rating Ceiling Panels for Sensible Heating and Cooling* (third public review). Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 2007. *BSR/ASHRAE Standard 189P: Standard for the Design of High-Performance Green Buildings Except Low-Rise Residential Buildings* (first public review draft). Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Ataer, A.E., and B.I. Kilkis. 1994. An analysis of the solar absorption cycle when coupled with in-slab radiant cooling panels. *Proceedings of the American Society of Mechanical Engineers IAHP Conference, AES, New Orleans, LA,* pp. 385–91.

Athienitis, A.K., and J.G. Shou. 1991. Control of radiant heating based on the operative temperature. *ASHRAE Transactions* 97(2):787–94.

Aynur, T.N., Y. Hwang, and R. Radermacher. 2006. Experimental performance measurements for VRV AC/HP systems. International Refrigeration and Air Conditioning Conference, July 17–20, Purdue University, West Lafayette, IN.

Ayoub M., N. Ghaddar, and K. Ghali. 2006. Simplified thermal model of spaces cooled with combined chilled ceiling and positive displacement ventilation system. *HVAC&R Research* 12(4):1005–30.

Bahnfleth, W.P., and E. Peyer. 2004. Variable primary flow chilled water systems: Potential benefits and application issues, Volume I. ARTI-21CR/611-20070-01, American Refrigeration Institute, Arlington, VA.

Baker, M. 1960. Improved comfort through radiant heating and cooling. *ASHRAE J.* 2(2).

Balcomb, J.D. 1983. Thermal network reduction. *Proceedings of* the *Annual American Solar Energy Society Conference, Minneapolis, MN.* See also: LA-UR-83-869, LA-9694-MS.

Bendapudi, S., and J.E. Braun. 2002. ASHRAE 1043-RP: A review of literature on dynamic models of vapor-compression equipment. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Brandemuehl, M.J., M.J. Lepoer, and J.F. Kreider. 1990. Modeling and testing the interaction of conditioned air with building thermal mass. *ASHRAE Transactions* 96(2):871–75.

Brasz, J.J. 2006. Comparison of part-load efficiency characteristics of screw and centrifugal compressors. International Compressor Conference, July 17–20, Purdue University, West Lafayette, IN.

Braun, J.E. 1990. Reducing energy costs and peak electrical demand through optimal control of building thermal storage. *ASHRAE Transactions* 96(2):876–88.

Braun, J.E. 2003. Load control using building thermal mass. *ASME JSEE* 125:292–301.

Braun, J.E. 2006. ASHRAE 1252-RP: Interaction between dynamic electric rates and thermal energy storage control—Phase II. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Braun, J.E., and N. Chaturvedi. 2002. An inverse grey-box model for transient building load prediction. *HVAC&R Research* 8(1):73–99.

Braun, J.E., and K. Mercer. 2003. VSAT: Ventilation Strategy Assessment Tool, CEC-500-2005-011. www.energy.ca.gov/2005publications.

Braun, J.E., and Z. Zhong. 2005. Development and evaluation of a night ventilation precooling algorithm. *HVAC&R Research* 11(3):433–58. Also see www.archenergy.com/cec-eeb/P3-LoadControls.

Braun, J.E., J.W. Mitchell, and S.A. Klein. 1987a. Models for variable-speed centrifugal chillers. *ASHRAE Transactions* 93(1): 1794-1813.

Braun, J.E., J.W. Mitchell, and S.A. Klein. 1987b. Performance and control characteristics of a large cooling system. *ASHRAE Transactions* 93(1):1830–52.

Braun, J.E., S.A. Klein, W.A. Beckman, and J.W. Mitchell. 1989a. Methodologies for optimal control of chilled water systems without storage. *ASHRAE Transactions* 95(1):652–62.

Braun, J.E., S.A. Klein, J.W. Mitchell, and W.A. Beckman. 1989b. Applications of optimal control to chilled water systems without storage. *ASHRAE Transactions* 95(1):663–75.

Braun, J, S.A. Klein, and J.W. Mitchell. 1989c. Effectiveness models for cooling towers and cooling coils. *ASHRAE Transactions* 95(2):164–74.

Braun, J.E., K.W. Montgomery, and N. Chaturvedi. 2001. Evaluating the performance of building thermal mass control strategies. *HVAC&R Research* 7(4):403–28.

Braun, J.E., T.M. Lawrence, C.J. Klaassen, and J.M. House. 2002. Demonstration of load shifting and peak load reduction with control of building thermal mass. *Proceedings of the 2002 American Council for an Energy-Efficient Economy Summer Study on Energy Efficiency in Buildings, Washington, DC,* pp. 55–68.

Briggs, R.S., R.G. Lucas, and Z.T. Taylor. 2002. Climate Classification for Building Energy Codes and Standards, Pacific Northwest National Laboratory. www.energycodes.gov/implement/climatezones_04_faq.stm.

Brown, D.R., and G.E. Spanner. 1988. Current cost and performance requirements for residential cool storage systems. PNL-6647, Pacific Northwest Laboratory, Richland, WA.

Buddhi, D. 2000. Selected references (1900 - 1999) on phase change materials and latent heat energy storage systems. www.fskab.com/annex17/Bibliography.pdf.

Bullard, C.W., and R. Radermacher. 1994. New technologies for air conditioning and refrigeration. *Annual Rev. Energy Environ*. 19:113–52.

Calobrisi, G., J.M. Henderson, H.L. Brown, and B.B. Hamel. 1980. Comparison of performance and computer simulation of a thermal storage system for commercial buildings. *ASHRAE Transactions* 86(1):336–50.

CEC. 1996. Source energy and environmental impacts of thermal energy storage, California Energy Commission. www.energy.ca.gov/reports/500-95-005_TES-report.pdf.

Carpenter, S.C., and J.P. Kokko. 1998. Radiant heating and cooling, displacement ventilation with heat recovery and storm water cooling HVAC system. *ASHRAE Transactions* 104(2):1321–26.

Chantrasrisalai, C., V. Ghatti, D.E. Fisher, and D.G. Scheatzle. 2003. Experimental validation of the EnergyPlus low-temperature radiant simulation. *ASHRAE Transactions* 109(2):614–23.

Chen, Y., E.A. Groll, and J.E. Braun. 2004b. Modeling of hermetic scroll compressors: Model validation and application. *HVAC&R Research* 10(3):307–29.

Chiu, S.A., and F.R. Zaloudek. 1987. R&D opportunities for commercial HVAC equipment. PNL-6079, Pacific Northwest Laboratory, Richland, WA.

Coad, W.J. 1985. Variable flow in hydronic systems for improved stability, simplicity and energy economics. *ASHRAE Transactions* 91(1B):224–37.

Conniff, J.P. 1991. Strategies for reducing peak air conditioning loads by using heat storage in the building structure. *ASHRAE Transactions* 97(1):704–709.

Conroy, C.L., and S.A. Mumma. 2001. Ceiling radiant cooling panels as a viable distributed parallel sensible cooling technology integrated with dedicated outdoor-air systems. *ASHRAE Transactions* 107(1):578–85.

Conry, R., L. Whelan, and J. Ostman. 2002. Magnetic bearings, centrifugal compressor and digital controls applied in a small tonnage refrigeration compressor design. *ICEC*, July 16–19, Purdue University, West Lafayette, IN.

CUAC. 2004. Technical support document: Energy efficiency program for commercial and industrial equipment. Commercial Unitary Air Conditioners And Heat Pumps, U.S. Department of Energy, Assistant Secretary, Office of Energy Efficiency and Renewable Energy, Building Technologies Program, Appliances and Commercial Equipment Standards, Washington, DC.

Dieckmann, J., K.W. Roth, and J. Brodrick. 2004. Radiant ceiling cooling, emerging technology report. *ASHRAE J.* 46(6):42–43.

DOE. 2007. 2007 Buildings Energy Databook. http://buildingsdatabook.eren.doe.gov/docs/6.2.5.pdf.

EIA. 2003. 2003 commercial building energy consumption and expenditures (CBECS), Energy Information Administration, public use data, micro-data files. www.eia.doe.gov/emeu/cbecs/.

Emmerich, S.J., and T. McDowell. 2005. Initial evaluation of displacement ventilation and dedicated outdoor air systems for U.S. commercial buildings. NISTIR 7244, Gaithersburg, MD.

Englander S.L., and L.K. Norford. 1992. Saving fan energy in VAV systems—Part 2. Supply fan control for static pressure minimization using DDC zone feedback. *ASHRAE Transactions* 98(1):3–18.

Eto, J.H. 1984. Cooling strategies based on indicators of thermal storage in commercial building mass. Annual Symposium on Improving Building Energy Efficiency in Hot and Humid Climates, ESL, Texas A&M University, College Station, TX.

Feustel, H.E., ed. 1999. Special issue on hydronic radiant cooling. *Energy and Buildings*, 33(2):117–210.

Feustel, H.E., and C. Stetiu. 1995. Hydronic radiant cooling—preliminary assessment. *Energy and Buildings* 22(3):193–205.

Franta, G. 1983. Personal communication *in re* Reid Dennis house, Sun Valley, ID.

Garland, M.W. 1980. Compressor capacity control for air-conditioning system partial load operation. *ASHRAE Transactions* 88(1):477–84.

Gordon, J.M., and K.C. Ng. 2000. *Cool Thermodynamics*. Norwood, MA: Cambridge Int'l Science Publishing.

Griffith, B., et al. 2007. Assessment of the technical potential for achieving net zero-energy buildings in the commercial sector. TP-550-41957, National Renewable Energy Laboratory, Golden, CO.

Hardaway, L.R. 1982. Effective chilled water coil control to reduce pump energy and flow demand. *ASHRAE Transactions* 88(1):331–41.

Hartman, T. 2001. Ultra-efficient cooling with demand-based control. *HPAC* 73(12):29–35.

Hattrup, M.P. 1988. Literature review of market studies of thermal energy storage. PNL-6457, Pacific Northwest Laboratory, Richland, WA.

Hauser, G., C. Kempkes, B.W. Olesen, and D.F. Liedelt. 2000. Computer simulation of the performance of a hydronic heating and cooling system with pipes embedded into the concrete slab between each floor. *ASHRAE Transactions* 106(1):702–10.

Hay, H.R., and J.I. Yellott. 1969. International aspects of air conditioning with movable insulation. *Solar Energy* 12(4):427–30.

Henze, G.P., R.H. Dodier, and M. Krarti. 1997. Development of a predictive optimal controller for thermal energy storage systems. *HVAC&R Research* 3(3):233–64.

Hiller, C.C. 1976. Improving heat pump performance via compressor capacity control. PhD thesis, Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, MA.

Hu, S-C, and R-H Yang. 2005. Development and testing of a multi-type air conditioner without using AC inverters. *Energy Conversion and Management* 46(3):373–83.

Huang, J., and E. Franconi. 1999. Commercial heating and cooling loads component analysis. LBL-37208, Lawrence Berkeley National Laboratory, Berkeley, CA.

Hydeman, M., et al. 2003. Advanced VAV system design guide. 500-03-082-A11, California Energy Commission, Sacramento, CA.

IEA. 1996. *Annex 28—Low Energy Cooling, Subtask 2: Reference Building Description*. Zürich, Switzerland: International Energy Agency.

IIR/Kruse. 1992. *Compression Cycles for Environmentally Acceptable Refrigeration, Air Conditioning and Heat Pump Systems*, (pp. 10–13). Paris: Int'l Inst. of Refrigeration.

Itron. 2006. *California end-use survey*. www.energy.ca.gov/ceus/index.html.

Jahnig, D.I., D.T. Reindl, and S.A. Klein. 2000. A semi-empirical method for representing domestic refrigerator/freezer compressor calorimeter test data. *ASHRAE Transactions* 106(2):122–30. See also http://minds.wisconsin.edu/handle/1793/7683.

Jarnagin, R.E., B. Liu, D.W. Winiarski, M.F. McBride, L. Suharli, and D. Walden. 2006. Technical support document: Development of the advanced energy design guide for small office buildings. PNNL-16250, Pacific Northwest National Laboratory, Richland, WA.

Jeong, J., S.A. Mumma, and W.P. Bahnfleth, Jr. 2003. Energy conservation benefits of a dedicated outdoor air system with parallel sensible cooling by ceiling radiant panels. *ASHRAE Transactions* 109(2):627–36.

Jeong, J.W., and S.A. Mumma. 2006. Eight simple steps for designing a DOAS with ceiling radiant cooling panels. *ASHRAE Journal* 48(10):56–66.

Jiang, Z., Q. Chen, and A. Mosere. 1992. Indoor airflow with cooling panel and radiative/convective heat source. *ASHRAE Transactions* 98(1):33–42.

Jin, Hui. 2002. Parameter estimation based models of water source heat pumps. PhD thesis, Department of Mechanical Engineering, Oklahoma State University, Stillwater, OK.

Kallen, H.P. 1982. Analysis–off peak cooling methods to reduce energy consumption. *ASHRAE Journal* 24(12):30–33.

Karaki, S. 1978. Air conditioning by nocturnal evaporative cooling of a pebble-bed. DOE Solar Cooling Conference, February 15–17, San Francisco, CA.

Katipamula, S., and S. Gaines. 2003. Characterization of building controls and energy efficiency options using CBECS. PNWD-3247, Pacific Northwest National Laboratory, Richland, WA.

Keeney, K., and J. Braun. 1996. A simplified method for determining optimal cooling control strategies for thermal storage in building mass. *HVAC&R Research* 2(1):59–78.

Keeney, K.R., and J.E. Braun. 1997. Application of building pre-cooling to reduce peak cooling requirements. *ASHRAE Transactions* 103(1):463–69.

Kilkis, B.I. 1993. Advantages of combining heat pumps with radiant panel and cooling systems. *IEA Heat Pump Centre Newsletter* 11(4):28–31.

Kilkis, B.I. 2000. Rationalization and optimization of heating systems coupled to ground-source heat pumps, *ASHRAE Transactions* 106(2):817–22.

Kilkis, B.I., S.S. Sager, and M.A. Uludag. 1994. A simplified model for radiant heating and cooling panels. *Simulation Practice and Theory* 2(2):61–76.

Kilkis, B.I., S.R. Suntur, and M. Sapci. 1995. Hybrid HVAC systems. *ASHRAE Journal* 37(12):223–28.

King, D.J., and R.A. Potter. 1998. Description of a steady-state cooling plant model developed for use in evaluating optimal control of ice thermal energy storage systems. *ASHRAE Transactions* 104(1A): 42–53.

Kintner-Meyer, M., and A.F. Emery. 1995. Cost optimal analysis and load shifting potentials of cold storage equipment. *ASHRAE Transactions* 101(2):539–48.

Kochendorfer, C. 1996. Standard testing of cooling panels and their use in system planning. *ASHRAE Transactions* 102(1):651–58.

Koschenz, M., and V. Dorer. 1999. Interaction of an air system with concrete core conditioning. *Energy and Buildings* 30(2):139–45.

Lewis, M.A. 1990. Microprocessor control of centrifugal chillers-new choices. *ASHRAE Transactions* 96(2):800–805.

Mackensen, A., S.A. Klein, and D.T. Reindl. 2002. Characterization of refrigerant system compressor performance. IRAC Conference, Purdue University, West Lafayette, IN.

Manley, J.K., Editor. 1954. *Radiant heating, radiant cooling*. School of Architecture, Pratt Institute, New York.

Marques, M.E., and P.A. Domanski. 1998. Potential coefficient of performance improvements due to glide matching with R-470C. *IRAC Conf.* Purdue, West Lafayette, IN.

Meierhans, R.A. 1993. Slab cooling and earth coupling. *ASHRAE Transactions* 99(2):511–18.

Meierhans, R.A. 1996. Room air-conditioning by means of overnight cooling of the concrete ceiling. *ASHRAE Transactions* 102(1):693–97.

Morris, F B., J.E. Braun, and S.J. Treado. 1994. Experimental and simulated performance of optimal control of building thermal performance. *ASHRAE Transactions* 100(1):402–14.

Mumma, S.A. 2001a. Ceiling panel cooling systems. *ASHRAE J.* 43(11):28–32.

Mumma, S.A. 2001b. Designing dedicated outdoor air systems. *ASHRAE J.* 43(5):28–31.

Novoselac, A., and J. Srebric. 2002. A critical review on the performance and design of combined cooled ceiling and displacement ventilation systems. *Energy and Buildings* 34(5):497–509.

Olesen, B.W. 1997. Possibilities and limitations of radiant floor cooling. *ASHRAE Transactions* 103(1):42–48.

Olesen, B.W. 2000. Hydronic radiant heating and cooling of buildings using pipes embedded in the building structure. 41 AICARR Conference, Milano, Italy.

Olesen, B.W., D.F. Liedelt, E. Michel, F. Bonnefoi, and M. De Carlie. 2000. Heat exchange coefficient between floor surface and space by floor cooling: theory or a question of definition. *ASHRAE Transactions* 106(1).

Peck, J.F., et al. 1979. Two-stage evaporative cooling using a rockbed associated with the active clearview solar collector. *Proceedings of the International Solar Energy Society*, *Orlando, FL,* pp. 697–701.

Perkins, D. 1984. Heat balance studies for optimising passive cooling with ventilation air. *ASHRAE Journal* 26(2):27–29.

Rabl, A., and L.K. Norford. 1988. Peak load reduction by preconditioning buildings at night. Fifth Annual Symposium in Improving Building Efficiency in Hot and Humid Climates, Texas A&M, College Station, TX. See also *J Energy Res*. 15:781–98.

Reed, J., K. Johnson, J. Riggert, and J. Dion. 2004. The structure, ownership, and energy use characteristics of the retail submarket of U.S. commercial building market. American Council for an Energy-Efficient Economy Conference, Washington, DC.

Ruud, M.D., J.W. Mitchell, and S.A. Klein. 1990. Use of building thermal mass to offset cooling loads. *ASHRAE Transactions* 96(2):820–29.

Shank, K.M., and S.A. Mumma. 2001. Selecting supply air conditions for a dedicated outdoor air system working in parallel with distributed sensible cooling terminal equipment. *ASHRAE Transactions* 107(1):562–71.

Shoemaker, R.W. 1954. *Radiant Heating, Including Cooling And Heat-Pump Applications*, 2d ed. New York: McGraw-Hill.

Simmonds, P. 1994. Control strategies for combined heating and cooling radiant systems. *ASHRAE Transactions* 100(1):1031–39.

Simmonds, P., S. Holst, S. Reuss, and W. Gaw. 2000. Using radiant cooled floors to condition large spaces and maintain comfort conditions. *ASHRAE Transactions* 106(1):695–701.

Sodec, F. 1999. Economic viability of cooling ceiling systems in U.S. commercial buildings. *Energy and Buildings* 33(2):195–201.

SWA, Inc. 2004. *GSA/LEED® Cost Study*. Norwalk, CT: Steven Winter Associates, Inc.

Stoecker, W.F., R.R. Crawford, S. Ikeda, W.H. Dolan, and D.J. Leverenz. 1981. Reducing the peak of internal air-conditioning loads by use of thermal swings. *ASHRAE Transactions* 87(2):599–608.

Stoecker, W.F., H. Kruse, D. Didion. 1986. Recent advances in refrigeration machinery: Design, manufacture and application. U.S. National Committee of the IIR Short Course, Purdue University, West Lafayette, IN.

Stetiu, C. 1997. Radiant cooling in U.S. office buildings. thesis, University of California, Berkeley, LBNL-41275.

Stetiu, G. 1999. Energy and peak power savings potential of radiant cooling systems in U.S. commercial buildings. *Energy and Buildings* 33(2):127–38.

Strand, R.K., and C.O. Pedersen. 1997. Implementation of a radiant heating and cooling model into an integrated building energy analysis program. *ASHRAE Transactions* 103(1):949–58.

Sun, C., T. Rossi, and K. Temple. 2006. Interaction between dynamic electric rates and thermal energy storage control—Phase I. ASHRAE RP-1252, Atlanta, GA.

Takebayashi, M., H. Kohsokabe, K. Sekigami, K. Suefuji, I. Tsubono, and K. Inaba. 1994. Performance improvement of a variable-speed controlled scroll compressor for household air conditioners. *ASHRAE Transactions* 100(1):471–75.

Tassou, S.A., and T.Q. Qureshi. 1994. Investigation into alternative compressor technologies for variable speed refrigeration applications. Int'l Compressor Engrg. Conference, Purdue University, West Lafayette, IN.

Telkes, M., and E. Raymond. 1949. Storing solar heat in chemicals. *Heating and Ventilating* 46(11):80–86.

TIAX. 2002. Energy consumption characteristics of commercial building HVAC systems, volume III: Energy savings potential. TIAX, Cambridge, MA. www.tiax.biz/aboutus/pdfs/HVAC3-FinalReport.pdf.

Turner, R.H., and F.C. Chen. 1987. Research requirements in the evaporative cooling field. *ASHRAE Transactions* 93(1):185–96.

Weijo, R.O., and D.R. Brown. 1988. Estimating the market penetration of residential cool storage technology using economic cost modeling. PNL-6571, Pacific Northwest Laboratory, Richland, WA.

Yilmaz, M. 2003. Performance analysis of a vapor compression heat pump using zeotropic refrigerant mixtures. *Energy Conversion and Management* 44(22):267–82.

Zweifel, G., and M. Koschenz. 1993. Simulation of displacement ventilation and radiant cooling with DOE-2. *ASHRAE Transactions* 99(2):548–55.