Control with Building Mass—Part II: Simulation

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ABSTRACT

Reductions in building peak electrical demand can be achieved by incorporating building-specific models of thermal dynamics into controllers that will implement short-term peak-period curtailment of HVAC capacity or pre-cool the building prior to peak-period cutbacks to increase the magnitude and duration of the load reduction. The same building-specific model can be used to effect energy savings by providing optimal-start control or optimal-pre-cooling control during unoccupied hours. Control logic was developed for pre-cooling with a central HVAC plant equipped with an air-side economizer. Measurement-based estimates were made of chiller performance and internal-gains schedules. The general transient-thermal-response model of the companion paper (Armstrong et al. 2006) was then used to determine building-specific thermal response and estimate the seasonal benefits of several peak-shifting and night-cooling strategies in the office building. Simulations showed a 30% to 60% reduction in seasonal mechanical cooling loads in the office building due to night cooling.

INTRODUCTION

Models of transient building thermal response, tuned to match specific buildings, have a variety of uses—predicting the change in indoor temperature and potential thermal discomfort associated with peak-period HVAC capacity curtailment, predicting the trajectory of indoor temperatures when load curtailment is preceded by a pre-cooling period, estimating when HVAC equipment should be started to restore comfortable indoor temperatures after reduced night conditioning, and model-based feedback control that better regulates temperatures by accounting for disturbances. This paper, companion to a paper that presents a general, data-driven model for building thermal dynamics (Armstrong et al. 2006), focuses on applications of such models. Relevant literature is reviewed, followed by a discussion of applications of transient-response models. A test building and its monitoring equipment are described. The monitored data are used to characterize actual operation and thermal response of the building. The simulation is described in detail, including parameters for the thermal-response model, estimates from measurements of chiller performance and internal-gains profiles, and control logic for night cooling in plants with air-side economizers. Estimates of the annual benefits of several load-reduction strategies, derived from the tuned simulation model, are then presented.

PREVIOUS WORK

Thermal response to weather, internal loads, and HVAC inputs are unique to each building and difficult to characterize empirically (Braun and Chaturvedi 2002; Norford et al. 1985; Pryor and Winn 1982; Rabl and Norford 1991). The development of a general model and model identification procedure sufficiently robust to be automated is, therefore, a central and challenging prerequisite to widespread adoption of effective curtailment or peak-shifting (Kintner-Meyer et al. 2003; Stoecker et al. 1981), pre-cooling (Keeney and Braun 1997), and optimal start (Seem et al. 1989; Armstrong et al. 1992) control strategies. Verification of curtailment for allocating incentives would, ideally, be based on the same model that is used for curtailment control (Goldman et al. 2002).

The analytical framework for dealing with different climates, rate structures, and occupancies is well established (Braun 1990; Kintner-Meyer and Emery 1995; Morris et al. 1994; Norford et al. 1985; Rabl and Norford 1991; Xing 2004). Reliable one-day-ahead weather forecasts can be downloaded as often as they are updated (daily or even

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hourly). Internal gains, predominantly light and plug loads, can be disaggregated from the building total (Leeb 1992; Laughman et al. 2003) and future loads can be similarly extrapolated from the accumulated history (Seem and Braun 1991; Forrester and Wepfer 1984).

The literature dealing with optimal control of discrete storage (e.g., a tank of ice or cold water) is exemplified by the very thorough work of Henze et al. (1997) and Kintner-Meyer and Emery (1995). Of particular interest to both discrete storage control and building mass control is an assessment of the impact of forecast uncertainties (Henze and Krarti 1999). The literature dealing with pre-cooling of building mass is rich but not consistent in its conclusions—in part because of the uncertainties in building thermal response and in part because of the severe capacity and storage limitation constraints (Andresen and Brandemuehl 1992; Brandemuehl et al. 1990; Braun 1990; Braun et al. 2001; Braun and Chaturvedi 2002; Conniff 1991; Eto 1984; Keeney and Braun 1996, 1997; Morris et al. 1994; Rabl and Norford 1991; Ruud et al. 1990). Night pre-cooling studies have used simplified analyses, as well as forward simulation, to show annual savings in mechanical cooling energy input of up to 50%. There have been few successful demonstrations of significant pre-cooling benefits in real buildings (Keeney and Braun 1997; Braun et al. 2001; Ruud et al. 1990).

Curtailment of cooling plant operation involves the same building-specific thermal response model that is used for night pre-cooling. Curtailment control is addressed in a number of papers. Questions of incentives and dispatch have dominated the discussion; building transient thermal response has not been handled very rigorously except in a few careful simulation studies (Stoecker et al. 1981; Haves and Gu 2001; Goldman et al. 2002; Kintner-Meyer et al. 2003; Xing 2004). Optimal start also requires a transient thermal response model. HVAC controls have traditionally used very simplistic models (Seem et al. 1989). More reliable control can be obtained with more realistic models (Seem et al. 1989; Armstrong et al. 1992).

CONTROL FUNCTION DESCRIPTIONS

Four main control functions require a building-specific transient response model of the conditioned space: curtailment, pre-cooling, optimal start, and setpoint control under large disturbances (model-based control).

Curtailment

Curtailment is the response to a utility contractual incentive that results in a reduction in load at the electrical service entrance. Short-term curtailment of plug and lighting loads is generally the most attractive measure for a building manager because it produces additional cooling capacity reductions without occupant discomfort. Dispatchable loads may include such equipment as copiers and printers, two-level lighting, general lighting (if task lighting or centrally controlled dimmable ballasts are available), or even task lighting (if sufficient daylighting is available) and may be extended to comput-ers of workers who are able to indulge in non-computer-related work without undue disruption of their productivity.

A more aggressive curtailment measure involves further reduction in cooling capacity, either by direct chiller control or by raising room temperature (and humidity, if implemented) setpoints. The setpoint changes can be abrupt or gradual, depending on the curtailment response desired. The occupants will experience some loss of comfort in either case, and the magnitude and duration of capacity reduction will be limited by occupants’ tolerances for elevated temperature and humidity levels.

Pre-Cooling

The most difficult, but potentially most effective, curtailment measure requires control functions that anticipate curtailment by anywhere from 1 to 16 hours and increase cooling capacity modestly during this pre-curtailment period. Such control may, for example, achieve zonal conditions that are as cold and dry as can be tolerated by the most sensitive occupant immediately before the curtailment period; the thermal masses of building structure and contents are at or even below this minimum tolerable temperature. With such a favorable initial state, the reduction in cooling capacity and its duration can be made significantly larger—i.e., about double what might be achieved without pre-cooling.

We define three distinct schemes for implementing pre-cooling:

1. A short period of reduced setpoint before the anticipated chiller curtailment time.
2. An extended period of reduced setpoint (starting the day with a reduced setpoint or ramping it down gradually during the entire pre-curtailment occupied period).
3. Night cooling (outside air and/or chiller operation).

Night cooling is a peak-shifting strategy that is well suited to office buildings and similar air-conditioned buildings in climates where cool nights and significant daytime cooling loads coincide through much of the year. Night cooling can save energy by using ambient conditions (cool night air) to remove heat from the building structure and contents or by using forms of mechanical cooling that are particularly efficient under night cooling (low lift, small load) conditions. The maximum usable cooling capacity stored on a given day is proportional to the normal zone cooling setpoint minus the thermal-capacitance-weighted-average of contents and structure temperatures at the start of occupancy. The actual capacity is roughly (but not exactly, because the envelope temperature field is affected by outdoor, as well as indoor, conditions) proportional to the change in the thermal-capacitance-weighted average of contents and structure temperatures between start and end of occupancy or between start and the point when zone and mass temperatures reach a maximum.
Thermal-storage capacity associated with building mass is limited in four respects: total thermal capacitance, rate of charge/discharge, night temperature, and comfort constraints. Within the framework of these basic limitations, there are a number of control strategies that can be used to implement night cooling.

**Constant Volume (CV) Pre-Cooling.** The simplest strategy is to cool the building at night whenever outdoor temperature is below return-air temperature, and, if outdoor temperature is sufficiently low, to modulate the cooling rate to just maintain the minimum comfortable temperature (MCT) until occupancy. On warm nights there may be no free cooling available, while on mild nights the minimum comfortable temperature may not be reached. This strategy is suboptimal for two reasons: no part of the mass can ever be cooled below the MCT and the cost of fan energy is not considered.

**CV Delayed Start.** The simple strategy can be improved by running the fan fewer hours at night. The objective in this strategy is to reach the MCT just before occupancy and, on days with little or no cooling load, to not reach MCT. Implementation is relatively simple because there is only one variable, start time, to be determined each night.

**CV Pre-Cooling/Tempering.** Delayed start is still not optimal because the potential to cool part of the mass below MCT is not exploited. A third CV strategy, therefore, adds another variable, the tempering time. On nights when it is advantageous and feasible to cool a zone below MCT, the tempering time is made just long enough for the zone to return to MCT without additional heating or cooling (i.e., to coast there). With this strategy, the optimal night cooling start and stop times are uniquely determined for each diurnal cycle.

**CV Objective Function.** The foregoing strategies have been described in terms of the thermal state of mass at the start of occupancy. However, this state is not observable and, in any case, is only indirectly related to the objectives of minimizing plant demand and energy costs. An objective function that represents daily electricity cost will therefore be used. To achieve the most effective CV strategy, daily cost must be minimized by enumerating all possible start times and tempering times and finding the combination that minimizes total daily cost.1

**Variable Volume Pre-Cooling.** The fan energy required on any given day can be further reduced by modulating fan speed during the night cooling phase. This is a much more difficult optimization problem because in a given hour, any sense in which daily cost is monotonic with fan speed is lost. A fairly rigorous optimization would forecast whole building operating cost 24 hours into the future and find the room temperature trajectory (e.g., 24 hourly values) that minimizes that cost for each day of the year. Two approaches can be used to make the problem more tractable: discretization of fan speed and application of a fan modulating function in which gain is determined daily by optimizations analogous to the CV-delayed-start and CV-tempering strategies.

**Recovery/Optimal Start**

Temperature recovery refers to the restoration of comfort conditions just prior to the start of each occupied period. In heating mode, maximum energy savings are usually achieved by starting recovery at the last possible moment. Controls that approach or accomplish this are often referred to as optimal start. It is generally desirable, for reasons of energy efficiency, to set zone temperatures back during unoccupied periods. Yet because first-cost and part-load efficiency penalize oversizing, equipment specified on the basis of steady-state design load may not have sufficient excess capacity to effect timely recovery of zone temperature on the coldest or hottest mornings of the year.

The dimensionless response of room temperature to a step change in zone heat input rate, \( Q \), is defined as

\[
\Psi(t) = \frac{T(t) - T(0)}{T(\infty) - T(0)}
\]

where

- \( T(t) \) = room temperature at \( t \) time from start of recovery,
- \( T(0) \) = initial room temperature at \( t = 0 \),
- \( T(\infty) \) = limit to which room temperature tends as \( t \to \infty \).

Note that \( T(\infty) \) depends on the boundary conditions including the ambient temperature \( T_w \), (determined by weather and buffer space conditions) and heating or cooling effect \( Q \) (determined by internal gains and plant operation). Thus, room temperature trajectory during recovery, \( T(t) \), depends on conditions (initial room temperature, outside temperature, heating capacity, and internal gains) in the expected way even though the dimensionless step response, \( \Psi(t) \), is determined completely and uniquely by the thermal parameters of the building envelope and its contents (Armstrong et al. 1992). The dimensionless temperature response to a step change in heating or cooling input, \( Q \), has a form consistent with Equation 8 in the companion paper:

\[
\psi(t) = 1 - (\gamma_1 \exp(\lambda_1 t) + \gamma_2 \exp(\lambda_2 t) + \cdots + \gamma_6 \exp(\lambda_6 t))
\]

where the \( \lambda \) variables are the eigenvalues (poles) of the system or a reduced-order approximation of the system, as can be identified by fitting the iCRTF model (Armstrong et al. 2006) to excitation and response time series data.

The step response is a reasonable basis (always conservative, albeit moderately so in most practical cases) for sizing plant and distribution equipment. However, it may be too conservative as a basis for optimal start control (Seem et al. 1989). An on-line model is ideal because it safely allows the latest possible plant start times on mornings when temperature recovery is needed.

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1. In practice, a one-hour timestep is convenient and reasonable in terms of control resolution. Also, complete enumeration is found to be unnecessary. One starts with the earliest start time and shortest (zero) tempering time. Tempering time is increased until daily cost stops decreasing. The process is repeated with progressively later start times until the daily cost stops decreasing with lateness of start time.
**Model-Based Control**

Room temperature is usually controlled by a thermostat and terminal unit, acting together as a local control loop. However, room temperature is not determined by actions of the terminal unit alone; it is also responsive to other highly variable conditions—weather, adjacent zone temperatures, solar and internal gains—that are generally treated as disturbances. A long-recognized way to improve control is to measure the important disturbances and estimate their effect on the response, as shown in Figure 1. The level of potential improvement depends on the relative importance of the measurable disturbances and on the accuracy of the plant model. Accuracy, in many cases, hinges on the ability of a model to adapt to changing plant parameters.

Variations of model-based control may include such additional features as predictive control. Ways may be provided to deal with loss of sensor(s) or loss of internet weather forecast inputs. Also note that the traditional compensator can be tuned for model-based control or stand-alone control. (In the latter case, near-optimal compensator gains are obtained with knowledge of the plant model.) Both sets of gains can be stored so that reasonable control can be maintained if the plant model loses key sensor inputs.

**DESCRIPTION OF EXPERIMENTAL BUILDING**

To demonstrate control applications of data-driven models, we simulate an existing building using submodels developed not from engineering descriptions but from the measured behaviors and responses of the building, its HVAC plant, and other energy subsystems. The test building is a 70,000 ft² (6,506 m²) municipal office in East Los Angeles, built in 1973. It is built on a 24 ft (7.3 m) square structural grid; the “south” wall normal actually points ~30° east of south) and ten bays wide (the long side running almost east to west; the built in 1973. It is built on a 24 ft (7.3 m) square structural grid, 17,300 ft² (1606 m²) footprint. The basement is essentially deep (floors 1 and 2 extend one bay farther north), giving it a height is 14 ft (4.3 m). The windows are operable but usually windowless. Windows on the main floors (1 and 2) are placed every 4 ft (1.2 m) to fit the 24 ft (7.3 m) grid; the floor-to-floor height is 14 ft (4.3 m). The windows are operable but usually remain closed. The 4,300 ft² (400 m²) third floor (penthouse) has no windows except on the 50 ft (15.2 m) east wall, looking onto a rooftop terrace, which is essentially all glass with a 20 in. (0.51 m) high transom strip of fixed glazing above an 80 in. (2.0 m) main span of sliding doors and fixed glazing.

The building is conditioned by a constant-volume dual-duct HVAC system. The plant consists of two 1.1 MBtuh input (322 kW) gas-fired boilers, two four-stage 90 ton (316 kW) reciprocating chillers, two 7.5 hp (5.6 kW) chilled water pumps, two 10 hp (7.5 kW) condenser water pumps, two 1.5 hp (1.1 kW) hot water pumps, and two 5 hp (3.7 kW) cooling tower fans. One of each pump pair runs as needed and the tower fans sequence on condenser water temperature. The air-distribution system has one 60 hp (44.8 kW) supply fan and one 20 hp (14.9 kW) return fan.

Instrumentation was used to record environmental conditions and building loads. Two nonintrusive load monitors (Laughman et al. 2003) were installed to measure electrical loads, one at the service entrance and one at the central fan/chiller motor control panel. The meter at the motor-control panel was trained to recognize the major HVAC loads. Three micro-loggers were placed on each of the two main floors and one each in the basement and third-floor open office areas to monitor room temperature. Additional thermal measurements included temperature and humidity of return, mixed, hot deck, and cold deck air. Fan inlet pressure taps measured flow rates at the supply and return fans while thermal anemometers measured mass flow rate² to determine the damper-controlled division of supply air between hot and cold decks.

**SIMULATION OF PEAK SHIFTING CONTROLS**

To compare cooling control strategies we must estimate annual energy use and cost for each strategy while all other aspects of building operation are held constant. These “constants” include fan and chiller performance curves; fan, chiller, and auxiliary equipment control sequences; weekly and holiday schedules that describe internal gains, minimum outside air, and fan static pressure; zone temperature setpoint schedule and related control parameters; utility rate schedule; and weather.

The information flow and main functional elements of the simulation are shown in Figure 2. The thermal response model of the companion paper (Armstrong et al. 2006) is integrated with economizer operation. This model is general in form and is made building-specific by plugging in the model order, number of walls, and coefficient values. Other functional elements are described in this paper.

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² Thermal anemometers continuously measure mass velocity, \( \rho V_{\text{point}} \). A traverse of each deck provides one-time measurement of the area-weighted velocity profile, \( [A_iV_i] \), and an effective area, \( A_{\text{eff}} = \left( \Sigma A_iV_i \right) / V_{\text{point}} \), used to estimate mass flow rate at each timestep, \( \dot{m}_r = A_{\text{eff}}(\rho V)_r \).
Comprehensive Room Transfer Function (CRTF) for Simulation

From the companion paper, the discrete-time model can be evaluated recursively for the conduction component of cooling load, $Q$, or zone temperature, $T$:

$$Q = B^{1:n}(\phi, Q) + B^{0:n}(0_\theta, T) - B^{0:n}(0, T)$$  \hspace{1cm} (3)

$$0_\theta T = B^{0:n}(\phi, Q) + B^{0:n}(0_\theta, T) - B^{1:n}(0, T)$$  \hspace{1cm} (4)

If we consider the cooling effect delivered to the zone to be a negative heat rate, $QC$, and heating to be a positive heat rate, $QH$, the zone heat balance is

$$QHC + Q + Qig + Qsg = 0$$  \hspace{1cm} (5)

where

- $QHC = QH + QC$ = the net heating (cooling) effect,
- $Qig$ = internal loads including lights, equipment, and metabolic heat, and
- $Qsg$ = undelayed solar gains.

When outside air is introduced at temperature $T_{OA}$ and mass flow rate $F$, the zone heat balance is

$$QHC + FC_p(T_{OA} - T) + Q + Qig + Qsg = 0.$$  \hspace{1cm} (6)

It is convenient to define a variable, $QP$ that collects the terms of Equations 3 and 4 that have been evaluated in previous simulation timesteps:

$$QP = B^{1:n}(\phi, Q) + B^{1:n}(0_\theta, T) - B^{1:n}(0, T)$$  \hspace{1cm} (7)

We can combine Equations 3, 6, and 7 to obtain an expanded expression for the zone heat balance:

$$FC_p(T_{OA} - T) + \theta_\theta T_w - 0_\theta T + QHC + QP + Qig + Qsg = 0.$$  \hspace{1cm} (8)

A corresponding expression for zone temperature response is

$$T = \frac{FC_pT_{OA} + \theta_\theta T_w + QHC + QP + Qig + Qsg}{0_0 + FC_p}.$$  \hspace{1cm} (9)

### Table 1. Coefficients of CRTF Model Identified for the Test Building

<table>
<thead>
<tr>
<th>Term*</th>
<th>Coefficient</th>
<th>CI/</th>
<th>coef</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$Q(t-1)$</td>
<td>0.7751</td>
<td>212.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$T_1(t-0)$</td>
<td>-0.2249</td>
<td>0.745</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$T_1(t-1)$</td>
<td>-3.456</td>
<td>11.3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$T(t-0)$</td>
<td>194.4</td>
<td>109.3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$T(t-1)$</td>
<td>-198.1</td>
<td>109.3</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

* $Q$ is in kBtuh and $T$ is in °F.

A thermal response model was obtained from data collected during three weeks in September 2002 in the previously described test building. Hourly observations were made of outdoor temperature, average indoor temperature, solar radiation, windspeed, and net rate of heat removal from the conditioned space. A single sol-air temperature, used to represent the exogenous temperature excitation of the system, was evaluated using a wind-speed-dependent heat transfer coefficient (ASHRAE 2005). The net rate of conduction gains was estimated from the other terms of the heat balance (Equation 5). The coefficients of the CRTF model (Equation 3) and their confidence intervals, obtained by constrained least squares, are listed in Table 1. The responses produced by the model for the test building are compared to the measured responses in Figures 3 and 4.

### Internal Gain and Occupancy Schedules

Weekday and weekend profiles of electrical loads within the conditioned envelope were generally found to be distinct and repeatable. The office building was supposed to be operated in weekend mode on Fridays, but some people worked and the resulting lighting and plug loads were variable. The 24-hour weekday and weekend profiles shown in Figure 5 were obtained for use in the simulations by averaging the hourly measured kW numbers.

Fans were simulated to operate from 6:00 a.m. to 5:00 p.m. Monday-Thursday, matching the observed schedule within 30 minutes.
Weather Data and Sun Angles

For simulation, the TMY2 weather file provides temperature, windspeed, humidity, and the direct and diffuse components of solar flux for each hour of the year. In actual implementation, these weather variables must be obtained from on-line forecasting services and site observations.

The treatment of sun was necessarily simplified in the model-identification phase because only total horizontal radiation was measured. The training data were obtained on a series of clear days in September when the hourly total on a south-facing surface at the latitude of Los Angeles is not very different from the hourly total on the horizontal. An effective aperture area for the south wall, which admits over 90% of the solar beam radiation available to the entire test building at this time of year, was therefore inferred.

For annual simulations, however, it is important to split direct and diffuse radiation and to model the dependence of incident and transmitted solar radiation on windows that contribute significantly to the building’s heat budget. This was
accomplished using the standard relations for direct beam incident angle as a function of solar zenith and azimuth and surface tilt and azimuth (Iqbal 1983).

**Chiller Performance**

For the simulation to estimate operating cost in each timestep we must provide a relation between cooling loads, as calculated by the transfer functions, and chiller power. The test building’s chiller power depends on total load (sensible plus latent), chilled-water temperature, and outdoor wet-bulb temperature. For constant chilled-water temperature, chiller power can be characterized as a function of load over a range of ambient temperatures. If the latent load is nearly a constant fraction of the total, chiller power can be related to sensible load alone. This is useful, because thermal balances as used in this study and by other research on use of thermal mass (e.g., Braun and Chaturvedi 2002) account only for sensible loads.

Office building coil loads were monitored for two weeks, August 29–September 9, 2002, a period that included some very hot weather as well as typical temperate conditions. The total coil load was computed from its sensible, $F_{cp}(T_{cold} - T_{mix})$, and latent, $F_{htg}(w_{mix} - w_{cold})$, air-side components. The latent fraction increased very slightly with load, as expected for an essentially constant air-side flow rate. Chiller-specific power, expressed as chiller power per unit coil load (kW/ton), increased with load. This was also expected. The condenser and evaporator areas are fixed so the approach temperatures increased with load. This was also expected. The condenser and evaporator areas are fixed so the approach temperatures increased with load. This was also expected.

For discrete-time simulation, it is inconvenient and unnecessary to emulate the cycling between stages that occurs with periods shorter than the simulation time step. A power law, relating sensible load to chiller power as shown in Figure 6, can therefore be used to represent chiller performance in the simulation.

![Power law used to simulate part-load chiller performance in the test office building.](image)

**Fan Performance**

Fan performance is based on the fan power loads and airflow rates measured in the test building. Supply fan heat is added to the mixed airstream. Although the test building of this analysis has CV fans, a variable air volume (VAV) system can also be simulated within the simulation framework. Fan power must be specified as a function of fan power and flow rate, where flow rate is a function of cooling load as described in the CRTF/economizer block.

**Room Temperature Control**

We will first simulate several control strategies that make use of acceptable temperature swings but don’t require a thermal response model. We define two such strategies as follows: (1) reduced economizer setpoint with no night cooling and (2) reduced economizer setpoint with unrestricted night cooling.

We then simulate an optimal night pre-cooling strategy based on an objective function in which the thermal response is forecast based on the inverse model. The particular optimal control uses night cooling start time as its decision variable. We call this strategy *optimally delayed night cooling*.

The room temperature setpoint maintained by the chiller is the same for all simulations: it is the time average of room temperatures measured in the test building during occupied periods, 74.5°F (23°C). Conventional economizer control is simulated by admitting enough outside air to maintain this setpoint and, if 100% outside air is not sufficient, by running the chiller to provide sufficient additional capacity. To take further advantage of economizer cooling, we adopt a two-stage cooling strategy wherein a lower cooling setpoint is maintained as long as there is sufficient economizer cooling capacity. This will result in later chiller start times on days when outdoor temperature is lower than return air temperature and when room temperature reaches or exceeds the first-stage cooling setpoint. With this background, the three cooling control strategies may be concisely described as follows.

**Reduced Economizer Setpoint with No Night Cooling.** With reduced economizer setpoint, forced ventilation cooling begins when the zone temperature rises to the economizer setpoint. If economizer capacity is sufficient, the zone temperature will stop rising; if not, zone temperature will continue rising but at a lower rate. When the normal (stage two) setpoint is reached, the chiller will come on but ventilation cooling will continue to contribute to the extent of available economizer capacity. There is no ventilation or chiller cooling at night.

**Reduced Economizer Setpoint with Unrestricted Night Cooling.** Same as above but with ventilation cooling enabled at night, the fans will bring in sufficient outside air to maintain the economizer setpoint. This results in zone temperatures at the start of occupancy that are below the stage-two setpoint on most days, and the result is longer fan run times but lower chiller run times.

**Optimally Delayed Night Cooling.** Same as above but with the onset of night ventilation cooling delayed each night for the amount of time that minimizes operating cost for the next day.
Setpoint and Capacity Simulation

Fan and thermostat setpoints can come from the schedule block (hardwired control) or from the optimizer (setpoint schedules vary from day to day). In either case, discrete thermostat setpoints are used so that the rate of HVAC heating and cooling does not vary continuously with zone temperature. The operating point \((Q, T)\), which must be evaluated at each timestep, cannot, therefore, be solved by a derivative method. We instead used a sequential algorithm to determine at which setpoint, or within which thermostat band, the zone temperature must be. Once this is determined, the heat rate and exact zone temperature can be evaluated directly.

There are three zone setpoints for heating \((T_{H1})\), economizer cooling \((T_{C1})\), and vapor-compression cooling \((T_{C2})\), as shown in Figure 7. These setpoints must satisfy the inequality

\[ T_{H1} < T_{C1} < T_{C2} , \]

and the corresponding plant capacities must satisfy

\[ Q_{H1} > 0 > Q_{C1} > Q_{C2} . \]

Note that economizer cooling capacity at 100% outside air is given by

\[ Q_{C1} = -F_{nom}c_p(T - T_{OA}) \] with \( T > T_{C1} \),

or zero, whichever is less. Cooling capacity \( Q_{C2} \) is the sum of \( Q_{C1} \) and the capacity of the vapor-compression equipment at the highest stage of cooling.

The CRTF heat rate and new zone temperature \((Q, T)\) are evaluated at each time step by applying the plant capacities and corresponding setpoints progressively (Figure 7) starting with \( Q_{C2} \):

\[
\begin{align*}
& \text{if } T < T_{C2}, \quad QC = \text{CRTF}(T_{C2}); T = T_{C2} \\
& \text{if } QC > Q_{C2}, \quad QC = \text{CRTF}(T_{C2}); F = F_{nom} \cdot QC/Q_{C2} \\
& \text{if } T < T_{C1}, \quad QC = \text{CRTF}(T_{C1}); F = F_{nom} \cdot QC/Q_{C1} \\
& \text{if } QC > Q_{C1}, \quad QC = \text{CRTF}(T_{C1}); F = F_{nom} \cdot QC/Q_{C1} \\
& \text{if } T < T_{H1}, \quad Q_H = -\text{CRTF}(T_{H1}); F = F_{nom} \cdot Q_H/Q_{H1} \\
& \text{if } Q_H > Q_{H1}, \quad T = \text{CRTF}(-Q_{H1}; F_{min}) \\
& T = i\text{CRTF}(Q_{C2}; F_{nom}); \quad \text{if } T < T_{C2} \text{, } QC = \text{CRTF}(T_{C2}); T = T_{C2} \text{, } \\
& \text{if } QC > Q_{C2}, \quad QC = \text{CRTF}(T_{C2}); F = F_{nom} \cdot QC/Q_{C2} \text{, } \\
& \text{if } T < T_{C1}, \quad QC = \text{CRTF}(T_{C1}); F = F_{nom} \cdot QC/Q_{C1} \text{, } \\
& \text{if } QC > Q_{C1}, \quad QC = \text{CRTF}(T_{C1}); F = F_{nom} \cdot QC/Q_{C1} \text{, } \\
& \text{if } T < T_{H1}, \quad Q_H = -\text{CRTF}(T_{H1}); F = F_{nom} \cdot Q_H/Q_{H1} \text{, } \\
& \text{if } Q_H > Q_{H1}, \quad T = \text{CRTF}(-Q_{H1}; F_{min}) \text{, }
\end{align*}
\]

As soon as the bracketing setpoints have been found, the algorithm can break out of the progression and can evaluate \( Q_{HC} \) from the heat balance:

\[ Q + Q_{HC} + Q_{ig} + Q_{sg} = 0 \]

where

\[ Q_{ig} = \text{internal loads including lights, equipment, and metabolic heat} \]
\[ Q_{sg} = \text{undelayed solar gains} \]
\[ Q_H = \text{Q}_H \cdot Q_C = 0, \text{ when } Q_{HC} > 0 \]
\[ Q_H = 0, Q_C = Q_{HC}, \text{ when } Q_{HC} < 0 \]

and the net conduction through the envelope is

\[ Q = \theta_{w,o} T_w - \theta_{p} T + Q_p \]

The final evaluation of zone temperature is then given by Equation 9. Note that the simulation does not model simultaneous heating and cooling; this idealization is expected to be conservative, if anything, with respect to assessing benefits of economizer and night cooling control strategies because simultaneous heating and cooling occurs mainly in cool weather when economizer and night cooling are most valuable.

Utility Rate Structure

The rate structure for the simulations was an interruptible rate with on-peak, mid-peak, and off-peak charges for both energy and demand in summer and winter seasons. The charge rates applied to each element of the rate structure are listed in Table 2. The rate included a ratchet charge of $6.60 per current month peak kW or $3.30 per past 11-month peak kW, whichever is greater.

The natural gas rate was taken to be constant at $0.60/therm; a more elaborate rate structure can be plugged in.

Energy and Demand Costs

The average hourly electrical loads are added to obtain the service entrance load for each hour. The utility rate structure is applied to provide the optimizer with hourly and 24-hour inte-
grated costs during its iterative solution process. The cost-processing block returns current day costs to the optimizer during successive iterations. Once a solution has been obtained, the daily cost is added to the annual cost accumulator.

**Optimizer**

The optimizer controls fan and room temperature setpoints based on operating costs estimated over a finite forecast horizon. The optimally delayed pre-cooling strategy requires a simple optimization algorithm that uses the identified thermal response model to forecast the cooling load up to 24 hours in advance. Because the simulation runs with one-hour timesteps and there is only one discrete variable in the optimization, it is easiest to solve this problem by complete enumeration. Taking the 24-hour period beginning at the end of occupancy, one makes \(n\) forecasts, where \(n\) is the number of unoccupied hours, of the cost of electricity to operate the building for the next 24 hours if the fans are started at hour \(i\), where \(i\) takes on values of 1 to \(n\). The objective function is:

\[
J = U(P(1:24))
\]

where

\[
U(x) \text{ is the 24-hour billing function,}
\]

\[
P(1:24) = P_{ch}(1:24) + P_{fan}(1:24) + P_{other}(1:24)
\]

is the vector of hourly service entrance loads,

\[
P_{ch}(1:24) = f(Q_c(1:24))
\]

is chiller plant power, based on Figure 4, plus pump and tower loads, and

\[
Q_c(1:24)
\]

is given in terms of current and past conditions, including \(T(t)\), by Equation 12 as constrained by the room temperature setpoints \(T_{C1}\) and \(T_{C2}\).

To approximate monthly demand billing in a way that can be duplicated in practical applications, we add incremental demand cost on each day when demand exceeds the peak of the preceding partial month.

**Table 2. Rate Structure Used for Simulation of All Control Cases**

<table>
<thead>
<tr>
<th>Rate Component</th>
<th>Units</th>
<th>Off-Peak</th>
<th>Mid-Peak</th>
<th>Peak</th>
</tr>
</thead>
<tbody>
<tr>
<td>Winter Energy</td>
<td>$/kWh</td>
<td>0.08944</td>
<td>0.12141</td>
<td>0.12141</td>
</tr>
<tr>
<td>Winter Demand</td>
<td>$/kW</td>
<td>0.0000</td>
<td>0.0000</td>
<td>0.0000</td>
</tr>
<tr>
<td>Summer Energy</td>
<td>$/kWh</td>
<td>0.08828</td>
<td>0.10917</td>
<td>0.19564</td>
</tr>
<tr>
<td>Summer Demand</td>
<td>$/kW</td>
<td>0.0000</td>
<td>2.7000</td>
<td>17.950</td>
</tr>
</tbody>
</table>

*Mid-peak hours are 9 a.m. to 11 p.m. every weekday except for the on-peak hours of 1 p.m. to 6 p.m. every weekday.

**SIMULATION RESULTS FOR PRE-COOLED OFFICE BUILDING**

The annual performance of night pre-cooling control strategies was evaluated by using TMY2 Los Angeles weather data to drive the office-building thermal response model. The cooling capacities provided by the chiller plant and by outside air were separately integrated over the year for each case simulated. These numbers and the annual electricity (energy and demand) costs together provide a complete picture of alternative control strategies.

The base case, no night cooling with the economizer setpoint equal to the mechanical cooling setpoint (74.5°F [23.6°C]), is shown in Figure 8. Mechanical cooling represents almost 60% of the total annual load. As the economizer setpoint is reduced, the total cooling load is increased but the annual mechanical cooling share is reduced to about 50% of the total with the most extreme economizer setpoint of 68°F (20°C). The cooling shares are not very sensitive to economizer setpoint because there is little daytime economizer cooling potential on days when there would be significant chiller load without economizer cooling.

With optimal nightly delayed-start times, the annual fan energy (hours of fan operation) is greatly reduced, the amount of economizer cooling is moderately reduced, and the amount of mechanical cooling is increased slightly, as shown in Figure 9.
Optimally delayed pre-cooling, as defined here, is not necessarily the lowest-cost cooling strategy that can be devised. For example, one test conducted during the project involved opening the test building’s operable windows on a cool night. The result was a delay in chiller start time of more than two hours (Norford and Armstrong 2003). Although technically feasible in this building, a practical implementation of natural ventilation would require automatic control of operable windows or other large ventilation openings provided for the purpose.

Another strategy that could be attractive in a climate with large diurnal temperature swings is to take the building to a temperature below the stage-one cooling setpoint (an uncomfortably low temperature) but terminate cooling just long enough before start of occupancy for room temperatures to return, by passive release of residual heat deep in the building fabric and contents, to the stage-one cooling setpoint. We call this strategy *optimal delayed pre-cooling with tempering*. The strategy clearly requires a thermal response model of high fidelity and reliability. A whole-building CRTF modeling approach simply may not be adequate; further research into identification of multiple-zone CRTF (or other) transient-response models may be needed.

With variable-speed fans, further savings are possible, but the optimization turns into a larger problem. The most direct formulation solves for 24 hourly room temperatures subject only to $T(h_1:h_2) < T_{C2}$, where $h_1$ and $h_2$ are the first and last hours of occupancy. In each hour, the pump, fan, tower, and compressor speeds and staging are determined by static optimization (ASHRAE 2003). Although this formulation is not completely rigorous (the room temperature in the last occupied hour has a small effect on the operating cost for the next 24 hours), the carryover effect has been shown to be small over the practical range of plant thermal storage capacity (Henze et al. 1997). The objective function (Equation 14) is modified to become

$$ J = U(P(1:24)) $$

(15)

where

$U(x)$ is the 24-hour billing function,

$P(1:24) = P_{\text{plant}}(1:24) + P_{\text{other}}(1:24)$ is the vector of hourly service entrance loads,

$P_{\text{plant}}(t) = f(Q_C(t), T(t), \text{other conditions at time } t)$ is statically optimized chiller plant power, and

$Q_C(t)$ is given in terms of current and past conditions, including $T(t)$, by Equation 12.
One-hour timesteps are reasonable for control of night pre-cooling. For chiller load-shedding response to an electric utility’s call for curtailment, a model with half-hour or smaller timesteps must be identified. The chiller output that would occur without curtailment can be forecast using the CRTF. Curtailment control can then be implemented by evaluating the zone temperature trajectory given by the iCRTF response to a curtailment trajectory given in terms of chiller capacity reductions at each timestep. The resulting setpoint trajectory must be broadcast to all zones.

SUMMARY

A simulation environment was developed for the study of optimal and suboptimal control strategies involving building thermal transient response. Chiller-performance curves, utility rate structures, and building-thermal-response models can be plugged in to perform building-specific analyses. A pre-cooling control strategy with variable subcooling time and variable tempering time was developed and implemented within the simulation environment. The strategy results in significant reduction of mechanical cooling using typical Los Angeles weather, utility rate structure, and office occupancy schedules. Sensitivity of annual savings to first-stage cooling setpoint was characterized with no pre-cooling and with two different pre-cooling strategies.

In the base case (no night cooling with the economizer setpoint equal to the mechanical cooling setpoint), mechanical cooling represents 60% of the total annual cooling load. This was reduced to 50% when the economizer setpoint was reduced by 6°F (3°C). With economizer cooling enabled at night, mechanical cooling was reduced another 30% by eliminating morning pull-down loads and yet another 30% when the economizer setpoint was reduced by 6°F (3°C). With optimal nightly delayed-start times, total annual cooling load numbers are 10%-15% lower and there is substantial reduction in fan hours as well.

Because it is primarily a controls measure and because it does not involve utility coordination,3 the implementation cost for night cooling is potentially quite low. However, packaging is a key issue. The control package must provide reliable, autonomous identification of the plant’s thermal response so that pre-cooling will execute with no special operator intervention and will result in no significant increase in occupant complaints. At the same time, one must provide rigorous and meaningful feedback to the operator about current expected and past documented savings and the accuracy of past response forecasts that have made those savings possible. To realize the potential benefits of pre-cooling it is important to design a chiller plant and associated controls that will provide good part-load efficiency. In the most aggressive implementations, the control system must be able to autonomously adjust zone setpoints at any time.

ACKNOWLEDGMENT

This research was sponsored by the California Energy Commission with additional support from the Grainger Foundation, the US Navy’s ONR Control Challenge, the National Science Foundation, and Talking Lights, LLC. Ron Mohr of Internal Services Division, East LA County, provided untiring support including frequent retrieval of data from widely dispersed room temperature micro-loggers.

NOMENCLATURE

\[ B^n(x, y) = \sum_{k=0}^{n} x^y(t-k) = \text{backshift polynomial} \]
\[ \theta = \text{zone-temperature coefficients that appear in one of the CRTF's backshift polynomials} \]
\[ \theta_W = \text{ambient sol-air temperature coefficients in backshift polynomial} \]
\[ \phi = \text{heat-flux coefficients in backshift polynomial} \]
\[ \Psi(t) = \text{dimensionless step response of zone temperature} \]
\[ Q = \text{zone heat-input rate due to conduction through the enclosure surfaces, Btu/h (W)} \]
\[ Q_H = \text{zone heat-input rate due to heating equipment, Btu/h (W)} \]
\[ Q_C = \text{negative zone heat-input rate due to cooling equipment, Btu/h (W)} \]
\[ Q_{HC} = \text{net zone heating (> 0) or cooling (< 0) provided by HVAC plant and distribution, Btu/h (W)} \]
\[ Q_{ig} = \text{internal loads, Btu/h (W)} \]
\[ Q_{sg} = \text{undelayed solar gains, Btu/h (W)} \]
\[ Q_p = \text{sum of conductive heat fluxes evaluated at past timesteps in an energy balance} \]
\[ T = \text{zone temperature, °F (°C)} \]
\[ T_W = \text{ambient sol-air temperature for a wall of a given orientation, °F (°C)} \]
\[ T_{OA} = \text{outside air dry-bulb temperature, °F (°C)} \]
\[ T_{cold} = \text{cold deck dry-bulb temperature, °F (°C)} \]
\[ T_{mix} = \text{mixed-air dry-bulb temperature, °F (°C)} \]
\[ T_{H1}, T_{H2} = \text{stage-one and stage-two zone heating setpoints, °F (°C)} \]
\[ T_{C1}, T_{C2} = \text{stage-one and stage-two zone cooling setpoints, °F (°C)} \]
\[ W_{cold} = \text{cold-deck humidity ratio} \]
\[ W_{mix} = \text{mixed-air humidity ratio} \]
\[ F = \text{mass flow rate, kg/s} \]
\[ F_{nom} = \text{nominal (maximum) supply fan mass flow rate, lb/h (kg/s)} \]
\[ F_{min} = \text{minimum supply fan mass flow rate, lb/h (kg/s)} \]
\[ C_p = \text{specific heat of air at constant pressure, Btu/lb-°F (J/kg K)} \]
\[ CRTF = \text{comprehensive room transfer function, for zone or whole building conductive heat flux} \]
\[ iCRTF = \text{inverse comprehensive room transfer function, for zone or cross-zone average temperature} \]

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3. Verification becomes a matter of internal evaluation rather than being a contractual issue.
REFERENCES


