Reducing Energy Costs and Peak Electrical Demand through Optimal Control of Building Thermal Storage

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ABSTRACT

This paper describes an investigation into the use of building thermal capacitance as a means of reducing the operating costs associated with maintaining adequate comfort conditions in buildings (termed "dynamic building control"). The state of the building thermal storage can be controlled through variations of the zone temperatures over time within the thermal comfort region. The primary opportunities in varying zone setpoints in an optimal fashion are associated with shifting cooling loads from daytime to nighttime to (1) reduce peak electrical demands, (2) take advantage of low nighttime electrical rates, (3) offset mechanical cooling with "free" cooling at night, and (4) enhance equipment operation at more favorable part-load conditions. The approach utilized in this study was to apply dynamic optimization techniques to computer simulations of buildings and their associated cooling systems for a range of conditions in order to determine the maximum possible savings. Results indicate that both energy costs and peak electrical use can be significantly reduced through optimal control of the intrinsic thermal storage within building structures. However, the cost savings depend strongly on several factors including 1) utility rate structure, 2) part-load characteristics of the cooling plant and air handling system, 3) weather,4) the occupancy schedule, and 5) building thermal capacitance.

INTRODUCTION

The potential for storing thermal energy within the structure and furnishings of conventional commercial buildings is significant when compared to the load requirements. Typically, internal gains are on the order of 3 - 7 W per square foot of floor space. A large percentage of these internal gains (e.g., 70%) is in the form of electromagnetic radiation and is absorbed directly by internal surfaces prior to being convected to the air space. The thermal capacity for typical concrete building structures is on the order of 2 - 4 W-h/°F per square foot of floor area. These factors suggest that the load requirements associated with maintaining the conditions of the air space within the comfort zone may be shifted significantly through management of a building's thermal storage with relatively small temperature swings. Practically, this load shifting is accomplished by proper adjustment of space temperature setpoints throughout the course of the day.

In conventional control strategies, the thermal storage of a building is not utilized for reducing operating costs. Optimal start algorithms determine the times for turning equipment on so that the building zones reach the desired conditions at a time when the building becomes occupied. The goal of these algorithms is to minimize the precool (or preheat) time. During occupied hours, the zone conditions are typically maintained at constant setpoints. For these conventional strategies, the assumption is that building mass works to increase operating costs. A massless building would require no time for precooling (or preheating) and would have lower overall cooling (or heating) loads than actual buildings. However, under proper circumstances, use of a building's thermal storage for load shifting can significantly reduce operational costs, even though the total zone loads may increase.

The dynamic adjustment of the space temperature setpoints in order to minimize overall operating costs is termed "dynamic building control." The most significant opportunities for dynamic building control involve cooling of a building rather than heating. In most cases, both the per unit energy cost and energy efficiency of heating systems (e.g., natural gas, oil) do not vary with time. As a result, the optimal control of a building for heating is relatively straightforward. Conventional controls with night setback and an optimal start algorithm that minimizes the time required to return the zone to the minimum comfortable setpoint at occupancy are sufficient. Cooling systems, on the other hand, are typically powered by electricity and are coupled strongly to the ambient conditions. As a result, the cost of operating a system per unit of cooling delivered may vary significantly over the course of a day. The primary opportunities for operational cost savings associated with shifting cooling loads are (1)reducing peak electrical demands, (2) taking advantage of low nighttime electrical rates, (3) utilizing "free" cooling at night to precool the building, and (4) improved part-load performance of equipment.

Figure 1 shows an example of the space temperature variation throughout a day for a zone under both dynamic building control and night setback control. At the onset of occupancy (or high electric rates) for dynamic building control, the zone temperature is at or near the lower limit of the comfort zone due to precooling throughout the previous night and morning. Over the course of the occupied period, the space temperature setpoint is adjusted upward to reach the upper comfort limit prior to the end of occupancy. At the end of the occupied cycle, the equipment turns off and the zone temperature floats above the upper comfort limit. At some point during the night (depending on ambient conditions, plant characteristics, and utility rate structure), the equipment (either mechanical or "free" cooling) turns on to precool the building. Conversely, with a conventional night setback control strategy, the zone temperature is commonly maintained at the upper limit of the comfort zone during the occupied period and floats freely with the equipment off during unoccupied times until the last possible time when the equipment can bring the zone to the upper setpoint at occupancy.

At any given time, the cooling requirement for a space is due to convection from internal gains (lights, equipment, and people)and interior surfaces. Since a significant fraction of the internal gains is radiated to interior surfaces, the state of a building's thermal storage and the convective coupling dictates the cooling requirement. Precooling of the building reduces the overall convection from exposed surfaces during the occupied period as compared with night setback control and can significantly reduce daytime operating costs.

The potential cost savings and comfort effects associated with the use of dynamic building control strategies have not been well documented in the literature. Hartmann (1980) suggested the concept of dynamic building control for HVAC systems. His strategies are based upon the use of night "free" cooling of buildings as a means of reducing energy costs. Shapiro et al.(1988) performed a test for a control strategy designed to utilize night "free" cooling. Although the control strategy worked well in maintaining adequate comfort conditions, the cost savings associated with this strategy were not documented. Stoecker et al. (1981) showed that the peak cooling requirements could be significantly reduced by using a large throttling range for proportional control, resulting in a large temperature swing over the course of the day. Recently, Spratt (1989) compared the operation of a building under dynamic and conventional control strategies. The operating cost savings associated with the dynamic control strategy were not quantified. However, a survey of occupants indicated that comfort conditions were improved under dynamic control.

The purpose of the study described in this paper was to identify the conditions for and magnitude of cost savings associated with the use of dynamic building control. The potential savings for dynamic building control depend upon several factors, including (1) the building's thermal capacitance, (2) the thermal coupling between the air and the mass, (3) the part-load characteristics of the cooling plant and air-handling system, (4) the ambient conditions, and (5) the utility rate structure (e.g., time-of-day rates or demand charges). It is not practical to use actual buildings and cooling systems in order to study these factors in any detail. The approach utilized in this study was to apply dynamic optimization techniques to computer simulations of buildings and equipment. The use of computer simulations allowed for a systematic study of the primary factors affecting the use of dynamic building control. The dynamic optimization applied to the system model was used to determine the "best" possible system performance (i.e., minimum operating costs) assuming that future ambient conditions and internal gain inputs were known (i.e., perfect forecasts). The "true" optimal performance results provided a basis for identifying the potential savings as compared with conventional control strategies.

Figure 2 shows a schematic of the cooling systems considered in this study. The building is cooled with forced-air convection using a chilled-water system that rejects its heat to the ambient through the use of cooling towers. For future reference, the plant is considered to include the chillers, pumps (chilled and condenser water), and the heat exchangers (cooling towers and coils), while the air-handling system includes components that are related to air distribution to the zones (ducts, fans, dampers). The buildings, cooling plants, utility rate structures, and weather patterns considered in this study cover a range of cases that were thought to be representative of commercial multi-story buildings.

METHODOLOGY FOR DYNAMIC OPTIMIZATION

Figure 2 shows the cooling system considered in this study. Optimal control of such a system that takes advantage of the thermal capacitance of the building involves minimizing an integral of operating costs over a specified period of time(e.g., a day) while satisfying required constraints. The operating cost at any given point in time is equal to the product of the power consumption of equipment and the cost of electricity. The constraints include both required comfort conditions for the building (i.e., temperature and humidity) and limits on the operation of equipment (e.g., capacity, safety, etc.). The optimal solution is a trajectory of controls throughout the specified optimization period. The control variables include zone setpoints, coil discharge air temperatures, and all plant controls (e.g., chilled-water temperature, pump and fan speeds).

For a building, the dynamic optimization problem is complicated by the fact that there are discontinuities associated with the different possible modes of operation. The operational modes include (1) mechanical cooling with minimum outside air, (2) mechanical cooling with 100% outside air, (3) "free" cooling with 100% outside air and no mechanical cooling (i.e., economizer), (4) no cooling (i.e., floating zone conditions), and (5)heating. As a result of the discontinuities associated with these control modes, it is not possible to determine continuous open- or closed-loop functions for the optimal control trajectories. The approach utilized in this study involves discretizing the cost function in time and applying a nonsmooth optimization algorithm to determine the set of controls that minimize the sum of costs over the specified time. The computing requirements necessary for solving this problem are enormous. However, the problem may be simplified considerably by decoupling the plant and building analyses.

The dynamics of a cooling plant occur on a relatively smalltime scale and are neglected for the purposes of this study. As a result, the important dynamics in terms of energy storage occur within the structure of the building. In the appendix, a comprehensive transfer function model is presented for modeling the thermal behavior of a thermal zone. The sensible cooling (or heating) required to maintain a specified zone temperature is computed as a function of a set of current and previous uncontrolled variables (e.g., internal gains, solar, ambient temperature),current and previous zone setpoints, and previous sensible cooling requirements. For given cooling requirements and zone setpoints, the optimal plant supervisory controls may be determined using steady-state optimization methods. Furthermore, the performance associated with optimal control of a given plant may be correlated and used within a look-up table during the dynamic optimization of the system as a whole. The decoupling of the plant optimization from the optimal control of the building thermal storage provides the basis for the methodology used in this study.

In order to completely specify the optimization problem, it is necessary to choose an optimization period (i.e., length for summation of costs), a stage interval (i.e., timestep for discretization), and a set of initial and final thermal conditions for the building. The logical choice for an optimization period is 24 hours. A day is the natural cycle for both internal gains (i.e., people, lights, equipment) and ambient conditions (e.g., temperature and solar radiation). A stage interval of one hour is adequate in terms of modeling the dynamics of the building. For meaningful results, it is necessary to introduce the constraint that the energy content of the building at the beginning and end of day are equal. The solution to this constrained dynamic optimization problem is termed the "steady-periodic" solution.

Mathematically, the optimization problem is given as:

Minimize

$$J = \sum_{k=1}^{K} R(k) \times P^*(\mathbf{T}_z(k), \mathbf{f}(k)) \tag{1}$$

with respect to $\mathbf{T}_z(k)$ (k = 1, K) subject to

$$\mathbf{T}_{z,min}(k) \leq \mathbf{T}_z(k) \leq \mathbf{T}_{z,max}(k)$$

where

k	=	stage of the day (e.g., 1 to 24 for a one-hour interval),
K	=	number of stages in the day (e.g., 24 for a one-hour interval),
$R(k)$	=	cost of electricity at stage k,
$\mathbf{T}_z(k)$	=	vector of zone temperature setpoints at stage k,
$\mathbf{T}_{z,min}(k)$	=	vector of minimum allowable zone temperatures at stage k,
$\mathbf{T}_{z,max}(k)$	=	vector of maximum allowable zone temperatures at stage k,
$\mathbf{f}(k)$	=	vector of uncontrolled variables that affect plant power consumption at stage k (e.g., weather and internal gains),
$P^*(k)$	=	minimum plant power consumption at stage k associated with

maintaining zone setpoints of $\mathbf{T}_z(k)$.

Assuming 24-hour plant operation, there is a set of 24 temperature setpoints for each zone that minimizes the total daily cost of operation. For a given trajectory (i.e., variation over the day) of zone setpoints, the steady-periodic sensible zone loads are determined. Since the zone loads depend upon the history of zone loads through a transfer function relationship, it is necessary to determine the steady-periodic loads through iteration on the initial history. For the set of zone load requirements (setpoints and sensible loads) over the day, the plant performance is estimated. This involves determination of the minimum plant power consumption (optimal mode of operation and setpoints) at each stage that satisfies the zone requirements. The performance of the plant affects the zone humidity conditions through the state of the supply air to the zones. The dynamics associated with the zone humidity states are also forced to reach a steady-periodic condition through iteration on the initial history. For each stage, the optimal plant performance and zone humidities are evaluated as outlined in the appendix. Operation of the plant (both mode and discharge air temperature) is constrained to keep zone humidities within limits defined by the ASHRAE (1989) comfort bounds.

The optimization problem defined by Equation 1 is nonsmooth as a result of the discrete modes of operation of the system. Changes in zone setpoints can cause discontinuous changes in operating costs due to the different modes. To solve this problem, a direct search complex method (Gill et al. 1981) was employed. This method is based on function comparison only and no smoothness is assumed.

The optimization process outlined thus far assumes 24-hour operation of the plant. However, it is also necessary to determine the optimal hour at which the equipment should turn on to precool the building. This is accomplished by successive application of the dynamic optimization process. Initially, a precool period of one hour prior to occupancy is assumed. A good initial guess for the optimal solution for zone temperatures in this situation is the conventional strategy of setpoints at the upper limit of the comfort zone. The dynamic optimization is repeated for increasing hour increments of the precool period up to a limit of 24-hour operation of the equipment. The solution for the previous increment is used as an initial guess for the next increment. At the conclusion, the precool period giving the lowest daily cost is optimal.

Another optimization problem that arises in this study is one of minimizing the peak electrical demand over a day. In this case, the cost function that is minimized is the maximum total building electrical use for the day. Mathematically, the optimization problem is stated as:

Minimize

$$J = \text{Maximum}\left[P_{bldg}(k) + P^*(\mathbf{T}_z(k), \mathbf{f}(k))\right] \quad \text{for all k} \tag{2}$$

with respect to $\mathbf{T}_z(k)$ (k = 1, K) subject to

$$\mathbf{T}_{z,min}(k) \leq \mathbf{T}_z(k) \leq \mathbf{T}_{z,max}(k)$$

where $P_{bldg}(k)$ is the noncooling building electrical use at stage k.

MODELING APPROACH

The appendix describes models for representing the building, cooling system, and weather utilized within this study. The building and plant models represent simplifications of more detailed modeling approaches that significantly reduce the computing requirements associated with determining optimal dynamic control. System identification techniques are applied to a detailed model and physical description of the building in order to create a simplified model. This is commonly referred to in the literature as the "inverse" problem. The plant modeling also begins with detailed models and descriptions. The optimal plant performance is estimated by applying optimization techniquesto a simulation of the plant. The optimal plant performance is then correlated in terms of the primary variables that affectits performance. The models of the building and plant are combined with a model of the air-handling system in order to create a model for estimating overall system performance. The dynamic optimizer determines zone temperature setpoints. For given zone requirements, an additional optimization of the air handler is necessary to determine the best mode of operation (e.g., "free" cooling, mechanical, etc.) and the optimal discharge air temperature. Rather than utilize real weather data, a synthetic weather generator was developed that produces diurnal variations in ambient dry-bulb, wet-bulb, and solar radiation based upon daily statistics. The advantage of this approach over the use of real weather data is that it allows for systematic parametric studies of the effects of weather on dynamic building control in terms of simple statistical parameters. The variables that characterize the diurnal variations in ambient temperature, humidity, and solar radiation are the daily average temperature and solar clearness index. The daily solar clearness index is defined as the ratio of the daily horizontal radiation to the extraterrestrial value.

CHARACTERISTICS OF THE SYSTEMS STUDIED

The most important factors affecting the use of dynamic building control are (1) the design and use of the building (e.g., thermal capacitance, coupling between the surfaces and air, internal gain schedule), (2) the performance characteristics of the cooling plant and air-handling equipment, and (3) the utility rate structure. In this section, the characteristics utilized in this study for considering each of these factors, using the models described in the appendix, are presented.

Building Zones

For the purposes of this study, only simulations of a single thermal zone representing a single floor within a high-rise building (identical floors) were considered. The zone was rectangular with south- and north-facing exteriors 300 ft wide by 10 ft high and east and west exteriors 150 ft wide by 10 ft high. Half of the north and south surface area was windows, while there were no windows on the east and west surfaces. There was an additional 7000 ft^2 of wall surface area associated with interior partitions. The thermal capacitance of a building also includes furnishings, such as desks, file cabinets, and bookshelves. These additional surfaces were not considered in this study, so the results maybe slightly conservative.

Two different sets of building construction materials were considered comprising zones termed "heavy" and "light." Tables 1 and 2summarize the wall and window constructions for these zones.

The inside convection coefficient for walls and windows was assumed to be a constant value of 1 Btu/h-ft^2-°F, while the outside convection coefficient was a constant3 Btu/h-ft^2-°F.

The "heavy" zone has approximately twice the thermal capacitance of the "light" zone. However, the ability of the "light" zone to store thermal energy is significant. Figure 3 shows the response of both the "heavy" and "light" zones to a unit step change in the zone temperature. The "light" zone requires approximately24 hours to approach steady state, while the "heavy" zone nears steady state in about 48 hours.

Two different schedules of occupancy were considered, corresponding to buildings with 12-hour and 24-hour occupancy. The occupied period is considered to be the time interval during which ventilation air is required and zone conditions are maintained between specified comfort limits. Commercial buildings would typically have occupied periods of about 12 hours, while a hospital or apartment complex are examples of 24-hour occupied buildings. In this study, the12-hour occupied period was assumed to occur from 6 a.m. to 6p.m. During occupied periods, the lower and upper limits on zone temperature were 68°F and 76°F, while the lower and upper limits on humidity ratio were 0.004 and 0.012. The ventilation requirement for the zones during occupancy was assumed to be a constant rate equal to 10% of the design air-handler flow. During unoccupied periods, the limits on the zone temperature were expanded to 55°F and90°F. The effect of the lower temperature limit for unoccupied periods on the peak power consumption was also considered.

The internal gains for a building vary throughout the day primarily due to changes in occupancy. For the 12-hour occupancy considered in this study, internal gains as a percentage of peak internal gains (i.e., full occupancy) were as follows: 100% from 8 a.m. until 4 p.m., 50% from 7 to 8 a.m. and from 4 to 5 p.m., 25%from 6 to 7 a.m. and 5 to 6 p.m., and 10% from 6 p.m. to 6a.m. Two 24-hour occupancy schedules were also utilized: full occupancy for 24 hours (i.e., constant internal gains) and full occupancy for 12 hours (6 a.m. - 6 p.m.) and one-third occupancy for the rest of the day (6 p.m. - 6 a.m.). Internal gains related to evaporation of moisture were considered to be 15% of the total internal gains, while the percentage of the sensible gains that are radiated directly to surfaces was about 77%.

Cooling Plants and Air Handlers

Three different cooling plants were considered, having "good," "flat," and "poor" part-load characteristics. Figures 4 -6 show the overall plant coefficient of performance (COP) for the three cooling plants considered as a function of the part-load ratio and the difference between the discharge air and ambient wet-bulb temperatures. Each plant has the same COP at design conditions but has different sensitivities to the load and temperature differential. The performance of "real" systems would typically fall somewhere between those of the "good" and "flat" characteristics exhibited in Figures 4 and 5.

The plant performance shown in Figure 4 has a very favorable part-load characteristic. This plant utilizes variable-speed motors for all equipment including the chillers, pumps, and fans. As a result, the efficiency of the system improves at part-load conditions where smaller temperature differences across heat exchangers occur and lower flow rates are required. Depending upon the temperature differential, the best plant efficiency occurs between about 30% and 50% of the design load. At very low loads, inefficiencies associated with compressor operation offset other performance improvements, resulting in an overall degradation in system performance.

For the plant with the "flat" part-load characteristic shown in Figure 5, the efficiency of the plant is assumed to be constant with respect to changes in load. This hypothetical system might be representative of a plant with several stages of chillers, pumps, and towers that are in parallel and sequenced according to the load.

Figure 6 shows the plant performance for a plant with a very poor part-load characteristic. The "poor" plant utilizes all fixed-speed equipment with only a single stage of operation for each device. As a result, the plant efficiency falls off considerably at lower loads. In contrast to the "good" part-load plant, the efficiency of this "poor" part-load plant is maximum at the design load.

Only variable-air-volume (VAV) systems were considered within this study with two possibilities for modulating the airflow: variable-speed fans and variable-pitch fan blades. With a variable-speed fan, the speed of the fan is adjusted to give the required airflow. In this case, the power consumption was assumed to obey the fan laws. A more common method for modulating supply airflow utilizes variable-pitch fan blades with fixed-speed motors. In this case, more power is generally required to deliver the same airflow. Figure 7 shows the part-load characteristics for air-handler fans under variable-speed and variable-pitch control. At part-load conditions, the power consumption is significantly greater for fixed-speed, variable-pitch than for variable-speed control.

In the sections that follow, the terms "good," "flat," and "poor" part-load plant refer to the following combinations of plant and VAV air-handler fan characteristics presented in this section.

1. "good" part-load plant: plant characteristic of Figure 4 with variable-speed air-handler fan control

2. "flat" part-load plant: plant characteristic of Figure 5 with fixed-speed, variable-pitch air-handler fan control

3. "poor" part-load plant: plant characteristic of Figure 6 with fixed-speed, variable-pitch air-handler fan control

OPTIMAL VS. CONVENTIONAL CONTROL

With the night setback control strategy utilized in this study, the zone temperature is maintained at the upper limit of the comfort zone during the occupied period and floats freely with the equipment off during unoccupied times until the last possible time when the equipment can bring the zone to the upper setpoint at the time of occupancy. In this section, the costs associated with the optimal dynamic building control are compared with this conventional night setback control for a variety of conditions.

Minimum Energy Costs with No Time-Of-Day Rates

In the absence of any special electric time-of-day rates and demand charges, it is still possible to realize significant operating cost savings by using dynamic building control under the right circumstances. The most obvious opportunity involves the use of "free" nighttime precooling. A significant fraction of the daytime load that would normally be met by mechanical cooling can be satisfied through the use of cool nighttime ambient air.

Figure 8 shows zone temperature variations for both optimal dynamic and night setback control for a day with significant opportunities for nighttime free cooling. Also shown on this plot is the diurnal variation in ambient temperature. The optimal strategy for this particular system and day involves maintaining significantly lower setpoints than for conventional control. The building is precooled beginning shortly after the end of the occupied period to a minimum temperature of 70°F. As internal gains increase due to occupancy, the space temperatures are increased to an upper limit of 76°F. The zone space temperature remains at the upper comfort limit for a large portion of the occupied period. This tends to minimize gains from the interior surfaces during this time. In contrast, the conventional strategy maintains 76°F throughout the occupied period, while the space temperatures float freely during unoccupied times.

Figure9 shows both mechanical and "free" cooling (ventilation) energy associated with the control strategies and conditions of Figure8. The optimal strategy continuously cools the space throughout the day using either "free" or mechanical cooling. On the other hand, the night setback controller cools the building from about an hour before occupancy until the end of the occupancy period. The amount of (and peak) mechanical cooling required for the system is significantly less with dynamic control as compared with the conventional strategy. This reduction is achieved primarily through the use of "free" precooling of the building during the nighttime hours. The time variation in the "free" cooling energy approximately mirrors the variation in ambient temperature. The maximum "free" cooling occurs near the point of minimum ambient temperature, while minimum use of "free" cooling is near the maximum ambient temperature. "Free" cooling for the conventional strategy peaks when the system turns on to bring the space temperature to the occupancy setpoint. The system operates without mechanical cooling until "free" cooling alone can no longer maintain setpoint.

For the single day results represented in Figures 8 and 9, the operating costs associated with dynamic control are approximately35% less than those for night setback control. For this situation, the savings are primarily attributable to the use of "free" nighttime cooling. However, there are also significant opportunities for higher temperature days. Figure 10 shows normalized daily costs for optimal and night setback control as a function of average daily ambient temperature. The costs for dynamic control are significantly less than for conventional control over the entire range. At average temperatures greater than about 75°F, minimal opportunities exist for "free" cooling. For higher ambient temperatures, the reductions in operating costs associated with dynamic control are due to improved plant performance. The profile of cooling loads over the day is generally flatter for dynamic control as compared with night setback control. This may benefit the plant performance in two ways: (1) operation closer to peak efficiency for "good" part-load characteristics and (2) better efficiency resulting from plant operation during low-temperature ambient conditions.

In addition to the average daily temperature, the other important daily weather statistic is the clearness index. The daily clearness index is the ratio of total horizontal radiation to the extraterrestrial radiation. In addition to defining the total daily solar radiation, the clearness index affects the variation in ambient temperature and humidity ratio. The clearer the day, the greater the ambient temperature and humidity swing over the course of the day. Figure11 shows the effect of the daily clearness index on the normalized daily costs for both optimal dynamic and conventional night setback control.

For small values (i.e., less than 0.5), the costs increase with an increasing clearness index. This is primarily due to the increased solar energy loads. At larger values (greater than 0.5), costs decrease with the clearness index due to greater opportunities for "free" cooling associated with the larger ambient temperature swings. It is interesting to note that the savings associated with optimal dynamic control, as compared with night setback control, are only slightly affected by the clearness index. Much of the benefit for low ambient temperatures with large temperature swings is realized with conventional control during the early occupied hours.

The energy savings associated with dynamic building control are very dependent upon the building and plant characteristics, along with the weather conditions. Figure 12 shows percent daily savings for dynamic control as compared with conventional control for four systems over a range of ambient conditions. As a percent of daily costs, the savings are greatest at low ambient temperatures where the greatest "free" cooling opportunities exist. The system with the "good" part-load performance has opportunities for significant savings over the entire range of weather conditions for both light and heavy building construction. The opportunities are much less significant for systems with less favorable plant part-load characteristics. At high temperatures, the savings approach zero for the "bad" and "flat" part-load characteristics. In the absence of "free" cooling at high temperatures, there is little potential for improved plant performance with "flattening" the load profile through dynamic control for these systems. Even at low ambient temperatures, the "free" cooling opportunities are smaller for these systems than for the "good" part-load plant because of the use of variable-pitch rather than variable-speed air-handler fans.

Minimum Energy Costs with Time-Of-Day Rates

The opportunities associated with dynamic building control are greater for situations where electric rates vary with time of the day. Figure 13 shows comparisons of daily costs for systems and conditions of Figure 12, except that there is an on-peak period from 8 a.m. to 8 p.m. where electric rates are twice the off-peak values. Optimal dynamic building control results insignificant savings for almost all systems and ambient conditions considered. Again, the greatest opportunities exist for systems with "good" part-load characteristics.

Utilities differ in terms of the incentives they offer for off-peak electric use. Both on-peak hours and the ratio of on-peak to off-peak electric rates may vary considerably between different utilities. The opportunities for dynamic building control depend significantly upon these rate factors. Figure 14 shows the effect of the number of hours for on-peak electric rates (centered about noon) and the ratio of on-peak to off-peak rates. In general, better opportunities exist for higher ratios of on-peak to off-peak rates and longer on-peak hours. However, the savings are more sensitive to the ratio of on-peak to off-peak rates than to the length of the on-peak period. It is interesting to note that the savings approach a maximum with increasing on-peak period. In the limit, the percent savings for a zero-length on-peak period equal those for a 24-hour period.

Many buildings, such as hospitals or apartments, have 24-houroccupancy periods. A 24-hour occupancy reduces the opportunities for utilizing thermal storage within the building due to a reduced allowable temperature swing and "flatter" internal gain profile. Figure 15 shows results similar to Figure 13 for a building with full occupancy for 24 hours (i.e., constant internal gains) and a building with full occupancy for 12 hours (6 a.m. - 6 p.m.) and one-third occupancy for the rest of the day (6 p.m. - 6 a.m.). The savings associated with optimal dynamic control are significantly less for buildings with 24-hour occupancy (Figure 15) as compared with 12-hour occupancy (Figure 13). One of the primary reasons for the reduced savings is due to the fact that with 24-hourequipment operation under the conventional strategy, the building's thermal storage naturally acts to shift part of the internal loads to the night. However, with low ambient temperatures and reduced nighttime occupancy, there are sufficient "free" cooling opportunities to justify the use of dynamic building control for this system with 24-hour occupancy.

Minimizing Peak Electrical Demands

One of the major benefits of dynamic building control can be in reducing the peak cooling loads, thereby reducing peak electrical use as was illustrated in Figure 9. Minimizing the total daily energy costs automatically shifts a significant portion of the cooling load to off-peak hours and reduces peak use. However, peaks can be further reduced by controlling the building in an optimal manner with the goal of minimizing the peak electrical demand. This is only an important consideration on certain days of the year, when the peak daily electrical use approaches historical levels.

Figure 16 shows the potential for reducing the peak daily electrical use for a building with an optimal strategy as compared with conventional night setback control. These results were developed by optimizing the trajectory of zone setpoints in order to minimize the peak use using the same systems and conditions as for the results of Figure 13. For all systems considered, the reduction in peak electrical use associated with optimal control is very significant. Again, the greatest potential savings exist for heavy zones and with cooling plants having "good" part-load characteristics. For the most part, the reductions in peak demand increase with increasing ambient temperature as the electrical energy associated with cooling requirements becomes a larger portion of the total building electrical use. The only exception occurs at low ambient t part-loads during the day where its performance is very poor.

The optimal control strategies for all cases considered for Figure 16 involved cooling the structure to the minimum allowable temperature during unoccupied times and heating the space to the minimum comfort limit, if necessary, at the time of occupancy. The minimum unoccupied setpoint was assumed to 55°F, while the occupied lower comfort limit was 68°F. Both the peak daily electrical use and the energy costs associated with the optimal control for minimum peak consumption are sensitive to the minimum unoccupied temperature.

Figure 17 illustrates the effect of the minimum unoccupied space temperature on the reduction in both peak electrical consumption and energy costs for optimal as compared with night setback control. Two optimal strategies are shown: one that minimizes the peak electrical consumption for the day (i.e., maximum peak reduction)and another that minimizes the total energy cost (i.e., maximum energy cost savings). The maximum peak reduction associated with optimal control increases with decreasing minimum unoccupied temperature, but at the expense of significantly increased energy costs. Part of the reason for the energy cost penalty is associated with requirements for heating at the time of occupancy. Below a minimum temperature of about 58°F, the energy costs are greater than those for conventional control (i.e., negative savings). The maximum energy cost savings and associated peak reduction are not affected by the minimum unoccupied zone temperature for this system, because the optimal control does not bring the precooled space temperature below65°F. It is interesting to note that the peak reduction associated with the minimum cost strategy is not significantly less than the maximum possible at a minimum temperature of about 65°F. This was also found to be the case for other systems considered.

A 24-hour occupancy affects the opportunities for peak electrical demand reduction through dynamic building control. Figure 18shows results similar to Figure 16 for a building with full occupancy for 24 hours (i.e., constant internal gains) and a building with full occupancy for 12 hours (6 a.m. - 6 p.m.) and one-third occupancy for the rest of the day (6 p.m. - 6 a.m.). The peak reductions associated with optimal dynamic control are significantly less for buildings with 24-hour occupancy (Figure 18) as compared with 12-hour occupancy (Figure 16). However, with reduced nighttime occupancy, there is sufficient potential to justify the use of dynamic building control for this system with 24-hour occupancy.

CONCLUSIONS AND RECOMMENDATIONS

In this study, optimization routines were applied to computer simulations of buildings and their associated cooling systems in order to investigate the use of a building's thermal capacitance as means of reducing operating costs (dynamic building control). Specifically, these simulations were used to identify the conditions for and magnitude of cost savings associated with the use of dynamic building control.

Results of this study showed that both energy costs and peak electrical use can be significantly reduced through proper control of the building's thermal storage. However, the cost savings associated with the use of dynamic building control depend upon several factors, including (1) utility rate structure, (2) part-load characteristics of the cooling plant and air handling system, (3) weather, (4) the occupancy schedule, and (5) building thermal capacitance. Specifically, the following conclusions result from this study:

1. Energy costs can be significantly reduced for buildings in the absence of time-of-day rates if the cooling plant and air handler have favorable part-load characteristics, regardless of the ambient conditions. However, the savings are most significant at low ambient temperatures where "free" nighttime precooling opportunities exist.

2. For all systems considered, energy costs can be significantly reduced (e.g., 10 - 50%) in the presence of time-of-day rates, even for plants with unfavorable part-load characteristics. Again, the savings are most significant at low ambient temperatures when "free" precooling is possible.

3. For all systems considered, proper management of the building's thermal storage resulted in significant reductions (10 - 35%)in the peak electrical use.

4. In general, more significant energy cost savings associated with the use of dynamic building control exist for higher ratios of on-peak to off-peak rates and longer on-peak hours.

5. Energy cost and peak electrical use reductions are much less significant for 24-hour than for 12-hour occupied buildings.

6. In terms of energy cost savings, there appears to be little advantage in precooling the building below the lower limit of the comfort zone.

7. For maximum peak reduction, the best strategy is to precool the building to a lower temperature than for minimum energy cost. However, there can be a significant energy cost penalty associated with minimizing the peak due to increased thermal gains and possible heating requirements at occupancy. When operating to minimize peaks, a lower unoccupied temperature limit of about five degrees below the occupied comfort limit appears to be a good compromise between energy costs and peak reduction potential.

The cost savings presented within this study for dynamic building control represent an upper bound for the systems considered, assuming optimal control with perfect forecasts of future conditions. In practice, the opportunities could be significantly less due to imperfect knowledge of the system and forecasting. These results could also be affected by the assumption of a large radiative component of internal gains (i.e., 77%). If internal gains were primarily convective, then greater space temperature variations would be required to drive the building structure through the same temperature swings.

In any case, the opportunities associated with proper control of the thermal storage in buildings appear to be significant enough to warrant further study with a goal toward implementation of control strategies within commercial buildings. However, there are several issues to be resolved prior to implementation of dynamic building control on a wide scale. Areas of future work include:

1. Control Algorithms: Further work is necessary to develop simplified control strategies for dynamic building control.

2. Annual Savings: The annual cost savings associated with implementation of dynamic building control strategies should be estimated through the use of simulation. Ideally, building and plant modeling characteristics would be determined using measurements from a test-site application.

3. Occupant Comfort: It is necessary to judge the impact of dynamic building control strategies on occupant comfort requirements. Precooling of the building produces colder wall and other internal surface temperatures, which could raise the space temperature setpoints desired by occupants. This is beneficial in terms of discharging the thermal storage. The effect of "free" occupant setpoint adjustment during the discharge period, as compared with optimal setpoint adjustment, should be studied through the use of both simulations and actual field tests.

4. Impact of Building Type: The cost savings associated with dynamic building control for a single-story building with significant ground or ambient-air coupling would probably be less. The effect of the building design on the potential for dynamic building control should be studied in greater detail.

5. Electrically Driven Heating Systems: Preheating of a building could be beneficial in the presence of time-of-day electric rates when utilizing electric resistance heat or an electric heat pump. The potential savings and algorithms for optimal preheating should be studied through the use of simulations.

ACKNOWLEDGMENTS

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APPENDIX: SIMULATION MODELS

Building Model

Several simulation programs exist for estimating the heating or cooling requirements of buildings. Many of these packages utilize transfer function representations for the dynamic behavior of a building's massive elements. One such program, TRNSYS (1988), incorporates a building model (TYPE 56) that utilizes transfer function representations for walls along with an energy balance on the zone air to estimate the heat transfer requirements. This program estimates the wall transfer function equations based upon a physical description of the layers that comprise the wall. Additionally, energy balances on each zone air space require specification of the zone geometries, zone controls, and schedules of internal gains (e.g., occupancy).

Recently, Seem et al. (1988) presented a method for estimating the cooling or heating requirements for a zone using a single comprehensive transfer function equation. The coefficients of the transfer function may be calculated from the properties and geometry of the walls and zone. Alternatively, transfer function coefficients could be estimated using regression techniques applied to measurements on a zone or to results of a detailed simulation of the zone.

In this study, a simplified form of Seem's comprehensive transfer function equation was utilized for modeling a zone. This approach results in a significant reduction in the computing requirements associated with the dynamic optimization as compared with incorporating the more detailed analysis within the optimization.

For each thermal zone, the following form for a comprehensive transfer function is utilized for estimating the zone sensible cooling load required to maintain a specified zone temperature at any stage k.

$$Q_{z,k} = \sum_{i=0}^{N} \left[a_i T_{a,k-i} + b_i T_{z,k-i} + c_i Q_{g,s,k-i} + d_i Q_{sol,k-i}\right] + \sum_{i=1}^{M} e_i Q_{z,k-i} \qquad (3)$$

where the a's, b's, c's, d's, and e's are the transfer function coefficients and

$Q_{z,k}$ = sensible cooling requirement for stage k,

$T_{a,k}$ = ambient temperature for stage k,
$T_{z,k}$ = zone temperature setpoint for stage k,
$Q_{g,s,k}$ = total sensible internal gains (e.g., lights, people, equipment)for stage k,
$Q_{sol,k}$ = total incident solar radiation on all exterior zone surfaces for stage k.

The order of the model (i.e., the values of N and M) and the transfer function coefficients were estimated in this study by applying nonlinear regression to the results of hourly simulations determined with the TRNSYS building model. Hourly sensible zone loads were generated for a year with Madison TMY data for all zones considered. An adequate model order for the comprehensive transfer function of Equation 3 for the zones considered is N= 4 and M = 1. In general, this model estimates hourly cooling(or heating) loads to within about 1% of the more detailed modeling approach.

For the case of a floating zone temperature (e.g., equipment off or specified zone cooling), the zone temperature is estimated by solving Equation 3 for $T_{z,k}$ as

$$T_{z,k} = \frac{1}{b_0}\left[Q_{z,k} - \sum_{i=1}^{M} e_i Q_{z,k-i} - \sum_{i=0}^{N}\left[a_i T_{a,k-i} + c_i Q_{g,s,k-i} + d_i Q_{sol,k-i}\right] - \sum_{i=1}^{N} b_i T_{z,k-i}\right] \quad (4)$$

The zone humidity ratio is estimated through a moisture balance on the space, assuming a lumped capacitance for the moisture storage components (e.g., air, walls, furnishings, books). At any given instant in time, the differential equation describing the rate of change of the zone humidity ratio is

$$\frac{d\omega_z}{dt} = \frac{1}{C_H h_{fg}}\left[Q_{g,l} - Q_{z,l}\right] \quad (5)$$

where

ω_z = zone humidity ratio (mass of water vapor per mass of air),
$Q_{g,l}$ = latent energy gains to the space (e.g., people, cooking),
$Q_{z,l}$ = latent energy removal from the space due to the cooling system,
C_H = effective moisture capacitance (units of mass, typically10 times the mass of the zone air),
h_{fg} = heat of vaporization of water.

Equation 5 is solved numerically at each simulation stage. The latent energy gain to the zone is an input, while the latent removal rate is an output from the combined plant and air-handler model and depends upon the current control mode and zone sensible load requirements. Humidity control of the zone was not considered in this study but was allowed to float freely within the comfort limits.

Plant Model

As previously described, the optimization of the plant and building may be decoupled. In the absence of thermal storage, the dynamics associated with a cooling plant are small in comparison with the building. Neglecting these dynamics, optimal control of a cooling plant involves minimizing the instantaneous power consumption of the chillers, cooling tower fans, condenser water pumps, chilled-water pumps, and the air-handling fans while providing the cooling required to maintain the specified zone setpoints.

For a given load requirement and ambient conditions, the optimal control and minimum power consumption may be determined using a nonlinear optimization applied to a model of the plant. Furthermore, an optimal performance map may be established for the plant in terms of the primary uncontrolled variables that affect its performance. Braun (1988, 1989) found that for mechanical cooling with minimum outside air, the optimal plant control and power consumption correlate as a function of the total chilled-water load and the ambient wet-bulb temperature. However, in this study, it is necessary to also consider mechanical cooling with 100% outside air and free cooling (100 % outside air with no mechanical cooling) as possible modes of operation.

The methodology utilized in this study for the cooling plant involved developing an optimal performance map for the plant, not including the air-handler fan control or power consumption. The mathematical equipment models and optimization algorithm utilized to map the cooling plant performance are described by Braun (1988, 1989). In this study, it was found that the power consumption of the plant, excluding the air-handler fans, correlates as a function of two variables: (1) the total chilled-water load and (2) the temperature difference between the ambient wet-bulb and supply air temperature exiting the cooling coils. Furthermore, an adequate correlating function for the part-load factor is

$$\begin{aligned} PLF = \; & p_0 + p_1\, PLR + p_2\, PLR\, PLR \\ & + p_3 (T_{wb}-T_{as}) + p_4 (T_{wb}-T_{as})^2 + p_5 * PLR * (T_{wb}-T_{as}) \end{aligned} \tag{6}$$

where a_0, a_1, a_2, a_3, a_4, and a_5 are empirical coefficients specific to the cooling plant and

PLF = plant power relative to the design power consumption,
PLR = chilled-water load relative to the design load,
T_{wb} = ambient wet-bulb temperature,
T_{as} = coil discharge air temperature.

Over a wide range of conditions, the above correlation was found to be accurate to within about 2%.

In order to evaluate the total chilled-water load for a given sensible zone requirement and mode of operation, it is necessary to have a method for estimating the sensible-to-total load ratio of the cooling coils. The following function approximately correlates the sensible load ratio in terms of coil part-load ratio and entering and exiting conditions:

$$SHR = s_0 + s_1\, SPLRSHR_{des} + s_2\, (SPLRSHR_{des})^2 + s_3\, (T_{dp,i} - T_{as}) + s_4\, (T_{dp,i} - T_{as})^2 + s_5 SPLR\, SHR_{des}\, (T_{dp,i} - T_{as}) \qquad (7)$$

where s_0, s_1, s_2, s_3, s_4, ands_5 are empirical coefficients specific to the cooling coil and method of control and

SHR = sensible load ratio defined as the sensible cooling load relative to total cooling load,
SPLR = sensible part-load ratio defined as the ratio of the coil sensible cooling to the sensible load at design conditions,
SHR_{des} = sensible load ratio at design conditions,
$T_{dp,i}$ = dew-point temperature of entering air.

The result of Equation 7 must be constrained to be less than or equal to 1. In validating the form of Equation 7, the chilled-water temperature of the plant was considered to be optimally controlled. The accuracy of Equation 7 in mapping the coil sensible-to-total load ratio for a cooling plant under optimal control was tested for a wide range of conditions and was found to be adequate for the purposes of this study. In general, the sensible-to-total load ratio was between about 0.6 and 1.

Air-Handler Model

The air-handler flow was assumed to be modulated to maintain the prescribed zone setpoints. There are two possibilities for modulating the airflow in a VAV system that were considered in this study. The most efficient method involves the use of variable-speed fan motors, where the speed of the fan is adjusted to give the required airflow. The power consumption for a variable-speed air-handler fan was assumed to obey the fan laws and was computed as

$$P_{ahu} = P_{ahu,des}\left[\frac{\dot{m}_{ahu}}{\dot{m}_{ahu,des}}\right]^3 \qquad (8)$$

where $\dot{m}_{ahu}$ is the supply air flow rate from the air handler and $P_{ahu,des}$is the air-handler fan power at a design flow of $\dot{m}_{ahu,des}$.

A more common method for modulating supply airflow utilizes variable-pitch fan blades with fixed-speed motors. In this study, the power consumption was determined for variable-pitch fan control using a correlation from the BLAST (1981) simulation program.

$$P_{ahu} = P_{ahu,des}\left\{0.517 - 0.784\left[\frac{\dot{m}_{ahu}}{\dot{m}_{ahu,des}}\right] + 1.26\left[\frac{\dot{m}_{ahu}}{\dot{m}_{ahu,des}}\right]^2\right\} \tag{9}$$

At part-load conditions, the power consumption is always greater for fixed-speed, variable-pitch than for variable-speed control.

Overall Cooling System Performance

For a given trajectory of zone setpoints, a unique set of steady-periodic(i.e., identical initial and final energy states) sensible zone loads are determined through iteration on the initial unknown load history. The dynamics associated with the zone humidity states must also reach a steady-periodic condition by iteration on the initial zone humidity. At each stage, the optimal mode of operation (e.g., mechanical cooling, free cooling) to meet the sensible requirements is determined by evaluating the optimal power consumptions for each mode and choosing the one with the minimum. For a specified room state, mode of operation, and discharge air temperature, the power consumption is estimated using the plant and air-handler models defined in the previous sections. In order to evaluate the minimum plant and air-handler power consumption for modes with mechanical cooling, a one-dimensional golden-section optimization is used to estimate the optimal discharge air temperature. The zone humidities are allowed to float freely between limits defined by the ASHRAE (1989) comfort bounds. Operation of the plant (both mode and discharge air temperature)is constrained to keep zone humidities within these limits.

Statistical Weather Generation

The weather data required to estimate the performance of the plant and building are the ambient dry-bulb temperature, ambient humidity ratio, and the solar radiation. It is also necessary to evaluate the incident radiation on the exterior building surfaces. This requires knowledge of components of both beam and diffuse radiation along with surface orientations and ground reflectance properties. In order to study the effect of weather on dynamic building control in a systematic manner, the approach utilized in this study was to generate diurnal variations with statistical correlations in terms of average daily weather variables.

Duffie and Beckman (1980) describe a method for estimating typical hourly variations in horizontal beam and diffuse radiation from average daily horizontal radiation, location, and day of the year. They also present methods for determining radiation on surfaces of any orientation from geometry and horizontal beam and diffuse radiation. Erbs (1984) has shown that the average diurnal variations in ambient temperature and humidity depend primarily upon the average temperature and solar clearness index for the day. The solar clearness index is defined as the ratio of the horizontal radiation to the extraterrestrial value. Erbs developed correlations from statistical analysis of long-term weather data for several locations. The relationships from Duffie and Beckman (1980) and Erbs (1984) for hourly variations in solar radiation, ambient temperature, and humidity were utilized in this study.

TABLE 1

Construction Materials for "Heavy" Zone

Structure	Description
Exterior Walls	ASHRAE exterior wall #24: finished 6 in. heavyweight concrete with 4 in. face brick exterior
Interior Partitions	ASHRAE interior partition #5: plaster finish on 4 in. heavyweight concrete
Floor	ASHRAE interior #39 (except with a rug): rug on 4 in. heavyweight concrete with false ceiling
Ceiling	reverse of floor description
Windows	double-glazed with an overall thermal conductance of0.5 Btu/h-ft^2-°F and a constant solar transmittance of 0.8

TABLE 2

Construction Materials for "Light" Zone

Structure	Description
Exterior Walls	ASHRAE exterior wall #2: finished 4 in. lightweight concrete
Interior Partitions	ASHRAE interior partition #24: wood partitions
Floor	ASHRAE interior #38 (except with a rug): rug on 2 in. heavyweight concrete with false ceiling
Ceiling	reverse of floor description
Windows	double-glazed with an overall thermal conductance of0.5 Btu/h-ft^2-°F and a constant solar transmittance of 0.8

FIGURE CAPTIONS

Figure 1. Example Space Temperature Variation for Dynamic Building Control

Figure 2. Chilled Water Plant Schematic

Figure 3. Response to a Step Change (1°F) in Zone Temperature

Figure 4. COP for Plant with "Good" Part-Load Characteristic

Figure 5. COP for Plant with “Flat" Part-Load Characteristic

Figure 6. COP for Plant with "Poor" Part-Load Characteristic

Figure 7. Air Handler Part-Load Characteristics

Figure 8. Space Temperature Variations for Optimal and Night Setback Control (no time-of-day rates, heavy zone with "good" part-load plant, clearness index= 0.6, average ambient temperature = 65°F)

Figure 9. Mechanical and "Free" Cooing Energy for Optimal and Night Setback Control (no time-of-day rates, heavy zone with "good" part-load, clearness index = 0.6, average ambient temperature = 65°F)

Figure 10. Daily Energy Costs for Optimal and Night Setback Control(no time-of-day rates, heavy zone with "good" part-load, clearness index = 0.6)

Figure 11. Effect of Clearness Index on Daily Energy Costs for Optimal and Night Setback Control(no time-of-day rates, heavy zone with "good" part-load, average ambient temperature = 70°F)

Figure 12. Daily Energy Cost Savings for Optimal vs Night Setback Control (no time-of-day rates, clearness index = 0.6)

Figure 13. Daily Energy Cost Savings for Optimal vs Night Setback Control(on-peak period: 8 a.m.- 8 p.m., 2-to-1 on-to-off peak rates, clearness index = 0.6)

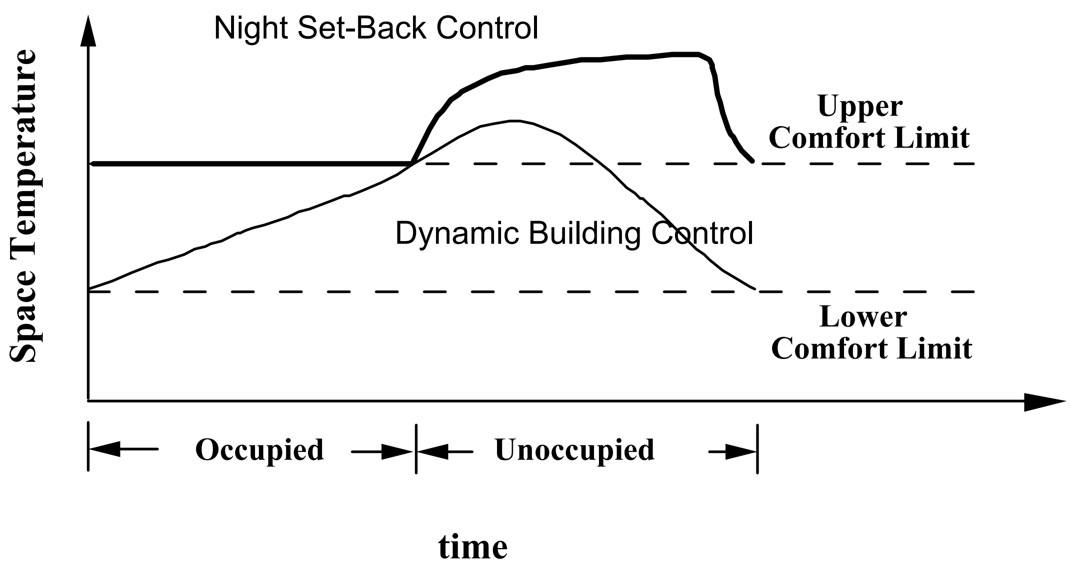
Figure 14. Effect of On-Peak Period and Rates on Daily Energy Cost Savings (on-peak period centered about noon, heavy zone with "flat" part-load, clearness index = 0.6, average ambient temperature= 80°F)

Figure 15. Effect of Occupancy Period on Daily Energy Cost Savings (on-peak period: 8 a.m.-8 p.m., 2-to-1 on-to-off peak rates, heavy zone with "good" part-load, clearness index = 0.6)

Figure 16. Daily Peak Electrical Usage Reduction for Optimal vs Night Setback Control (minimum unoccupied setpoint = 55°F, clearness index = 0.6)

Figure 17. Effect of Minimum Unoccupied Zone Temperature on Daily Peak Reduction and Energy Cost Savings (on-peak period: 8 a.m.- 8p.m., 2-to-1 on-to-off peak rates, light zone with "flat" part-load, average ambient temperature = 85°F, clearness index = 0.6)

Figure 18. Effect of Occupancy Period on Daily Peak Reduction (heavy zone with "good" part-load, clearness index = 0.6)



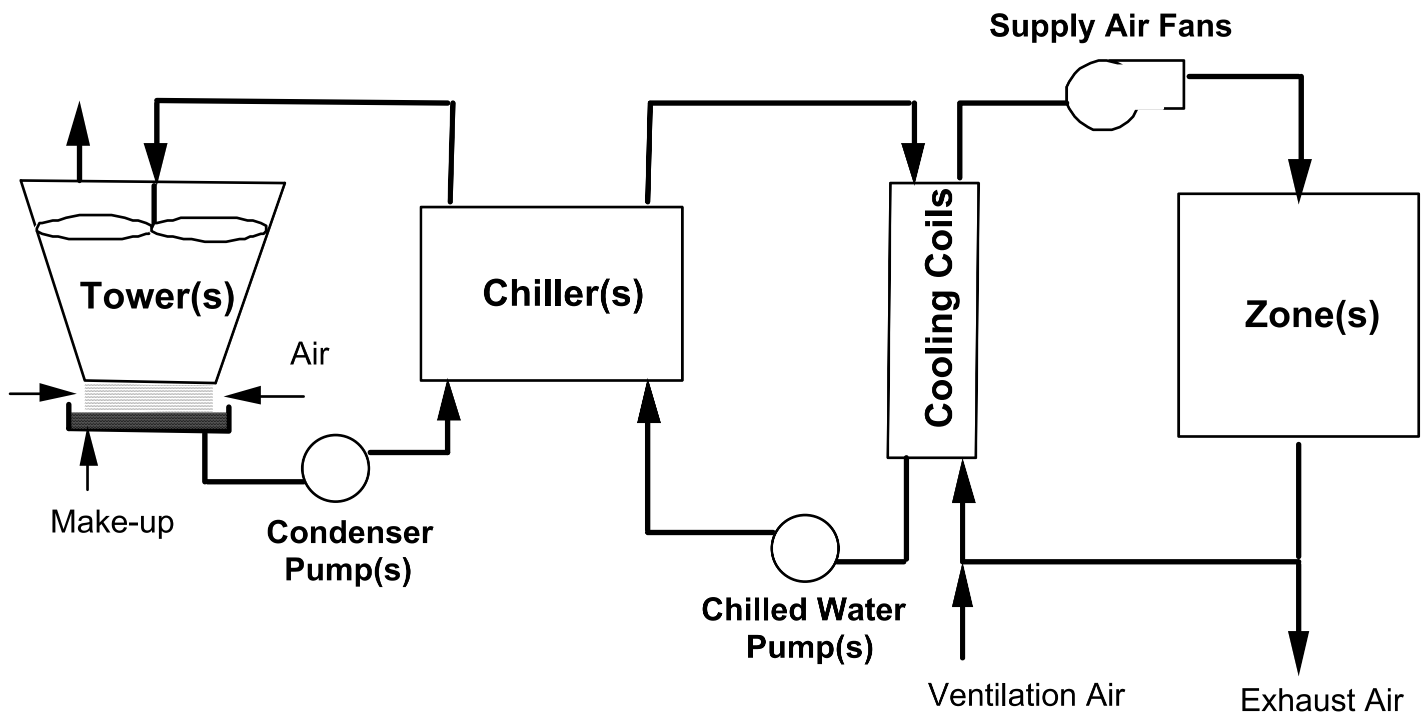
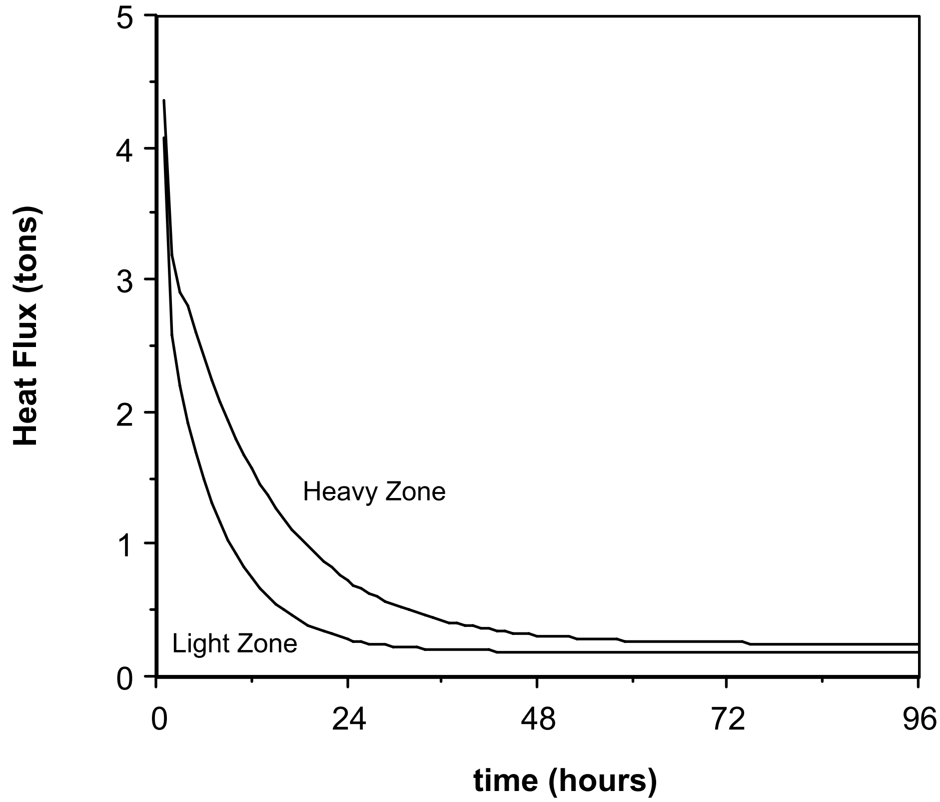
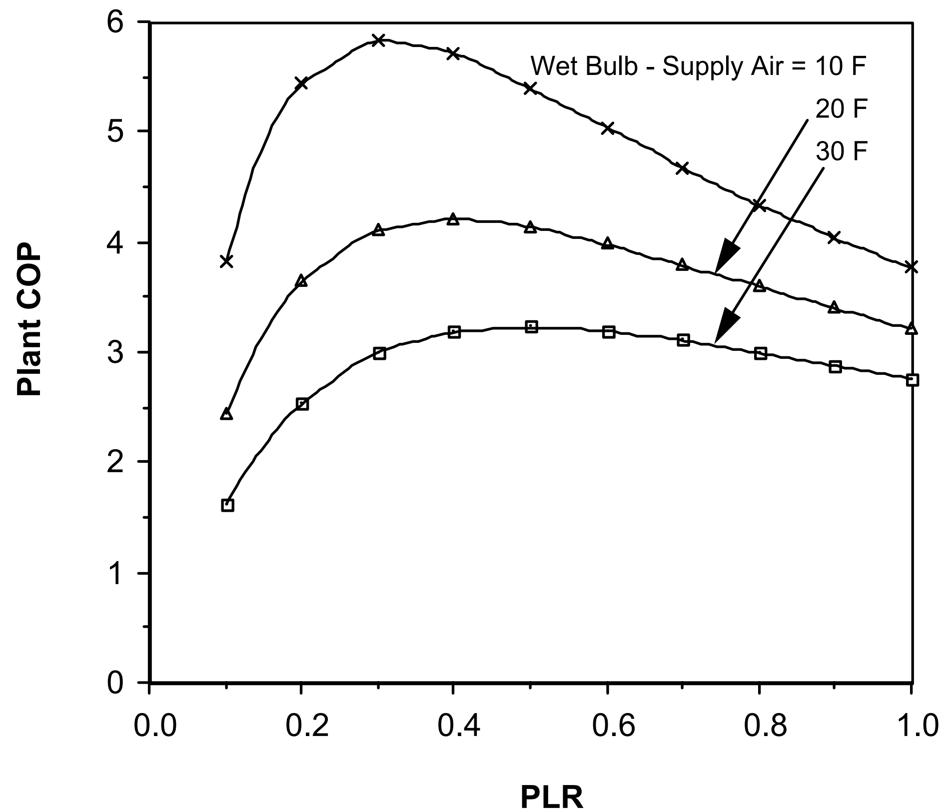
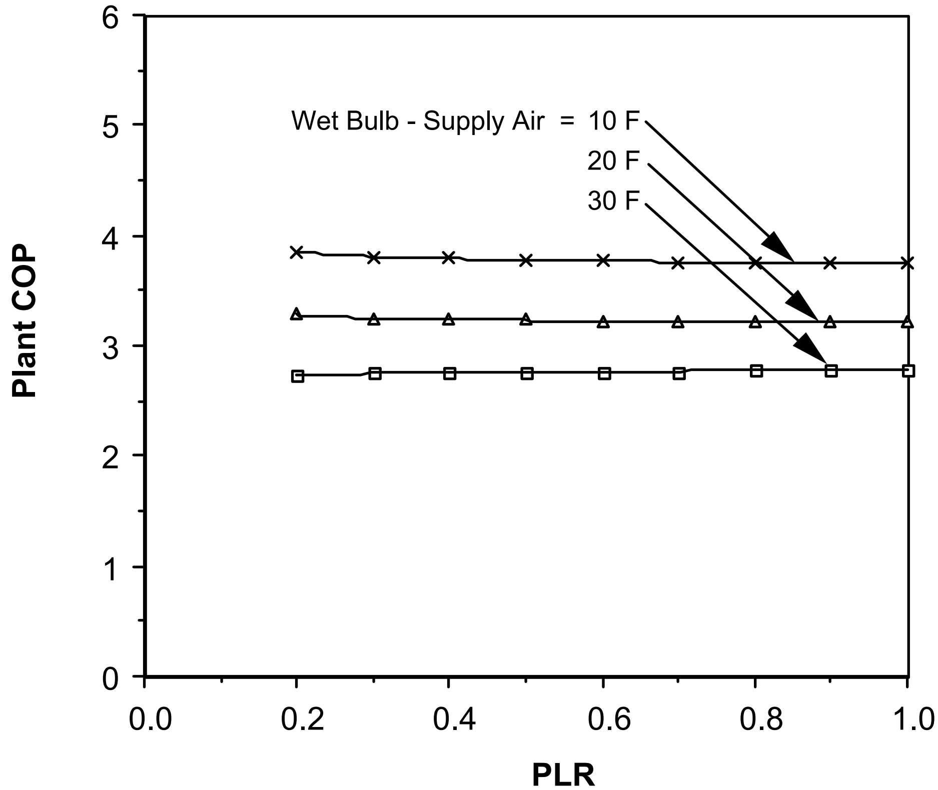


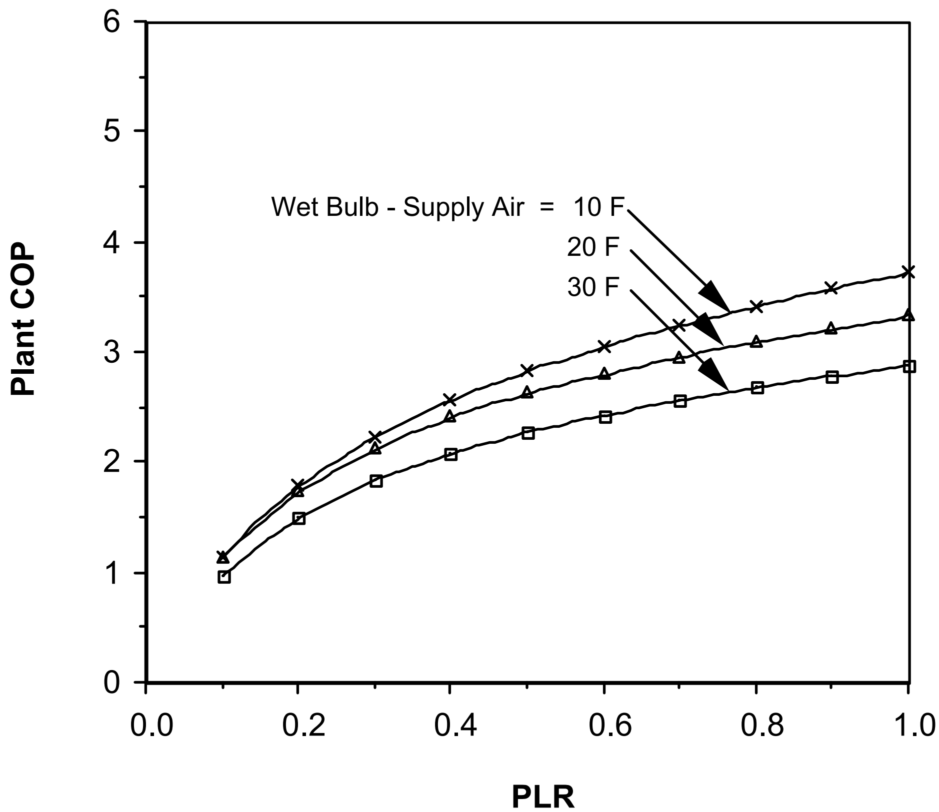
Figure 2. Chilled Water Plant Schematic


Heat Flux (tons)
Heavy Zone
Light Zone
time (hours)
0
24
48
72
96
0
1
2
3
4
5

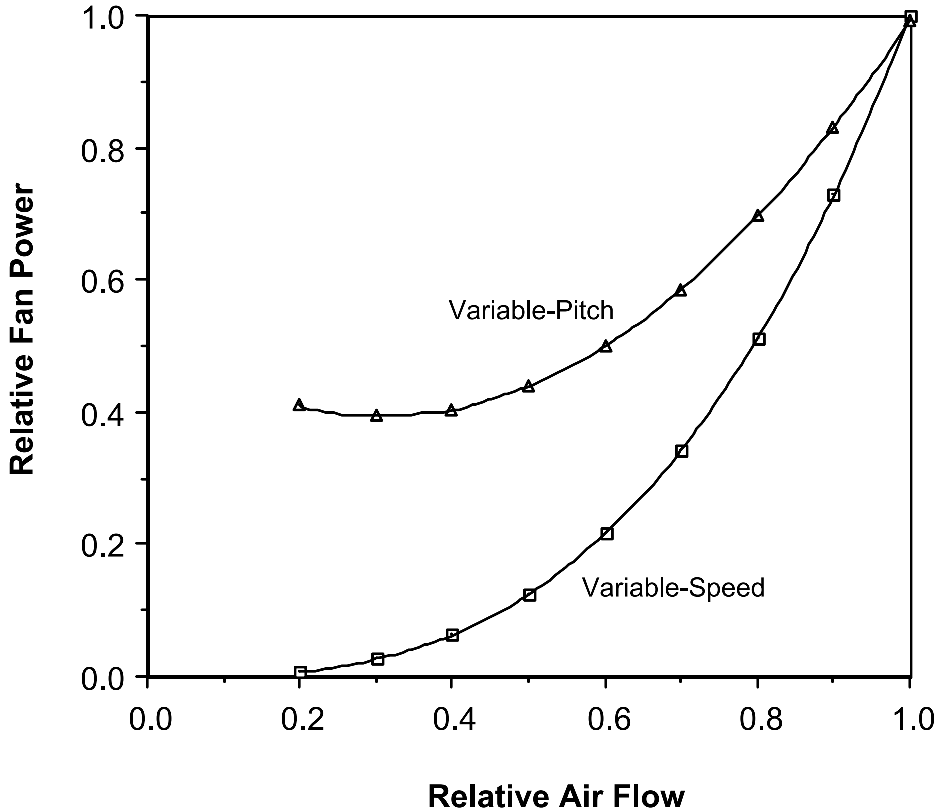

Plant COP
Wet Bulb - Supply Air = 10 F
20 F
30 F
PLR
0.0
0.2
0.4
0.6
0.8
1.0
0
1
2
3
4
5
6

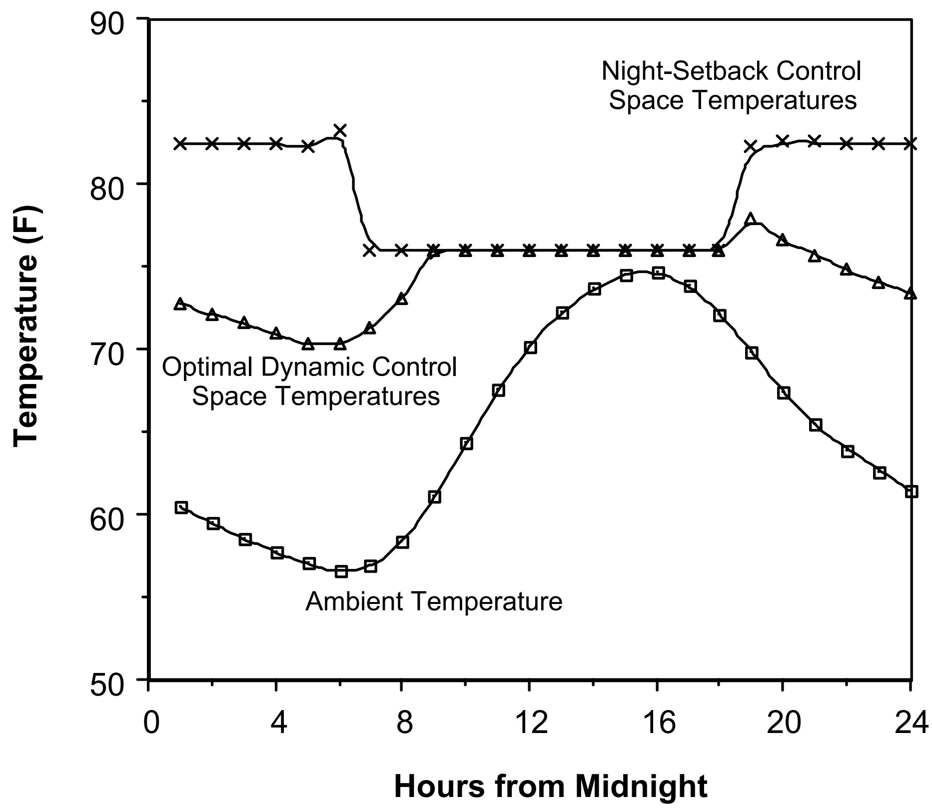


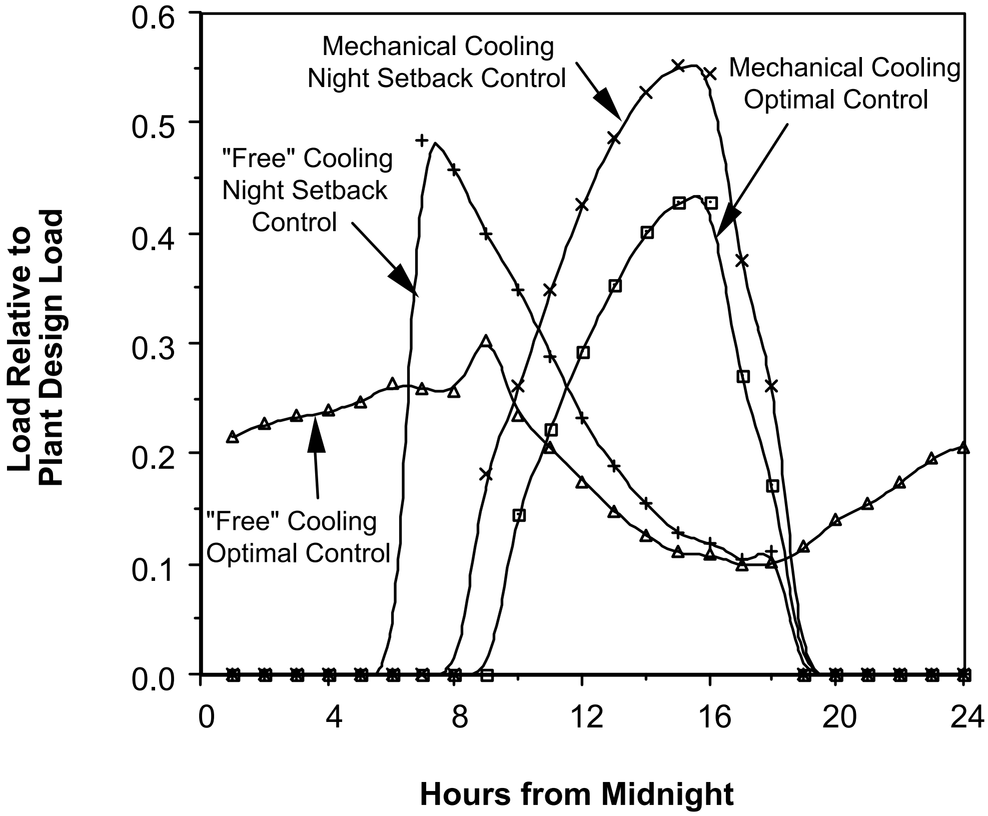
Wet Bulb - Supply Air = 10 F
20 F
30 F
Plant COP
PLR
0
1
2
3
4
5
6
0.0
0.2
0.4
0.6
0.8
1.0



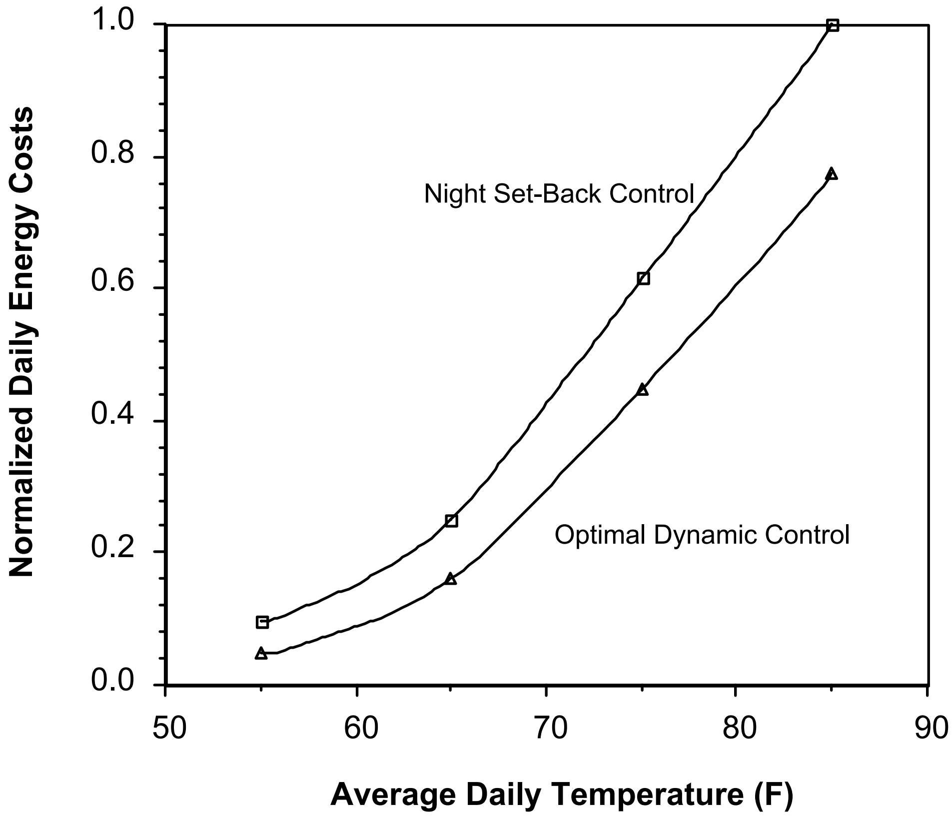
Wet Bulb - Supply Air = 10 F
20 F
30 F
Plant COP
PLR
0
1
2
3
4
5
6
0.0
0.2
0.4
0.6
0.8
1.0


1.0
0.8
0.6
0.4
0.2
0.0
Relative Fan Power
Variable-Pitch
Variable-Speed
0.0
0.2
0.4
0.6
0.8
1.0
Relative Air Flow

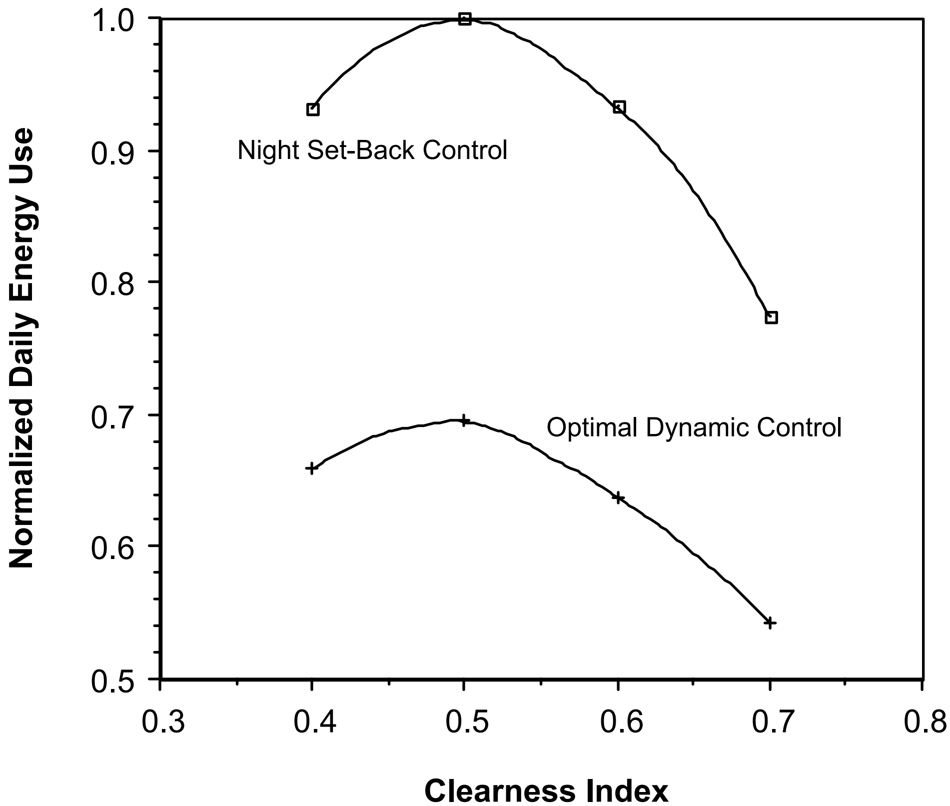

90
80
70
60
50
Temperature (F)
Night-Setback Control
Space Temperatures
Optimal Dynamic Control
Space Temperatures
Ambient Temperature
0
4
8
12
16
20
24
Hours from Midnight



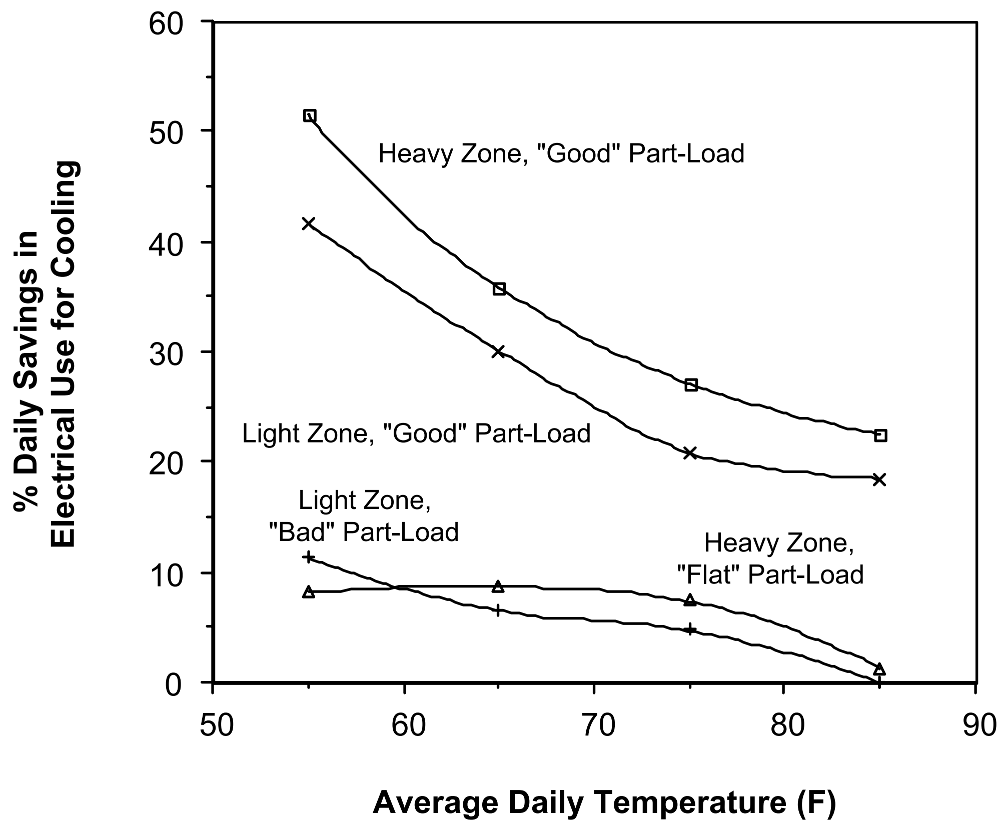
Mechanical Cooling
Night Setback Control
Mechanical Cooling
Optimal Control
"Free" Cooling
Night Setback
Control
"Free" Cooling
Optimal Control
Load Relative to
Plant Design Load
Hours from Midnight
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0
4
8
12
16
20
24



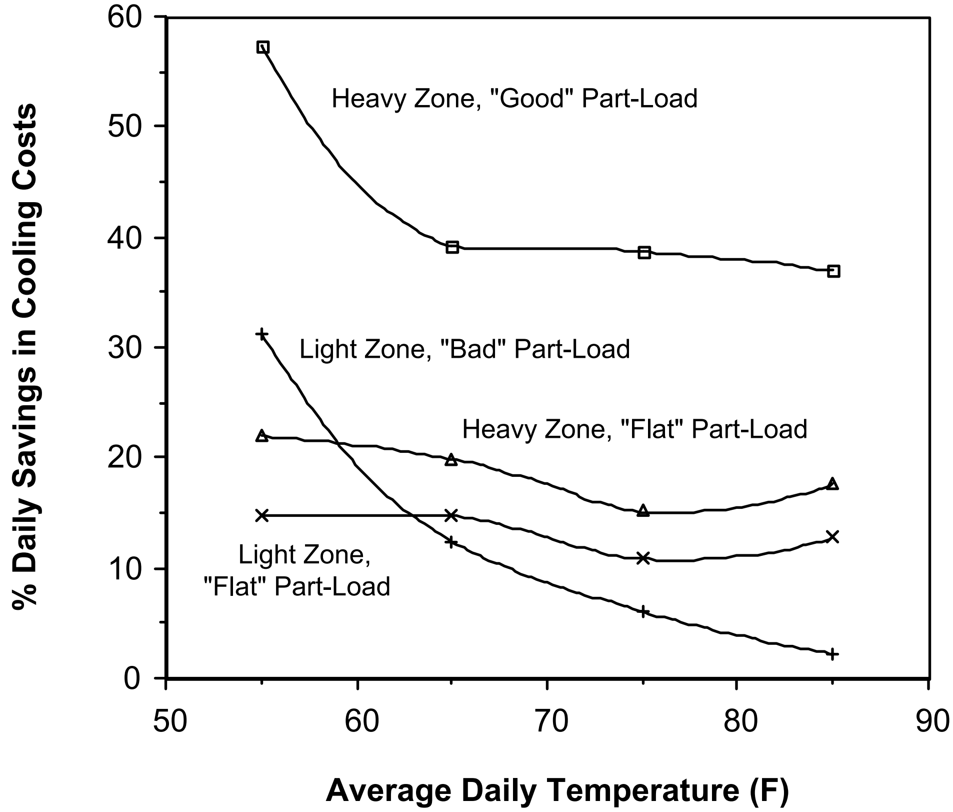
Night Set-Back Control
Optimal Dynamic Control
Normalized Daily Energy Costs
Average Daily Temperature (F)
0.0
0.2
0.4
0.6
0.8
1.0
50
60
70
80
90

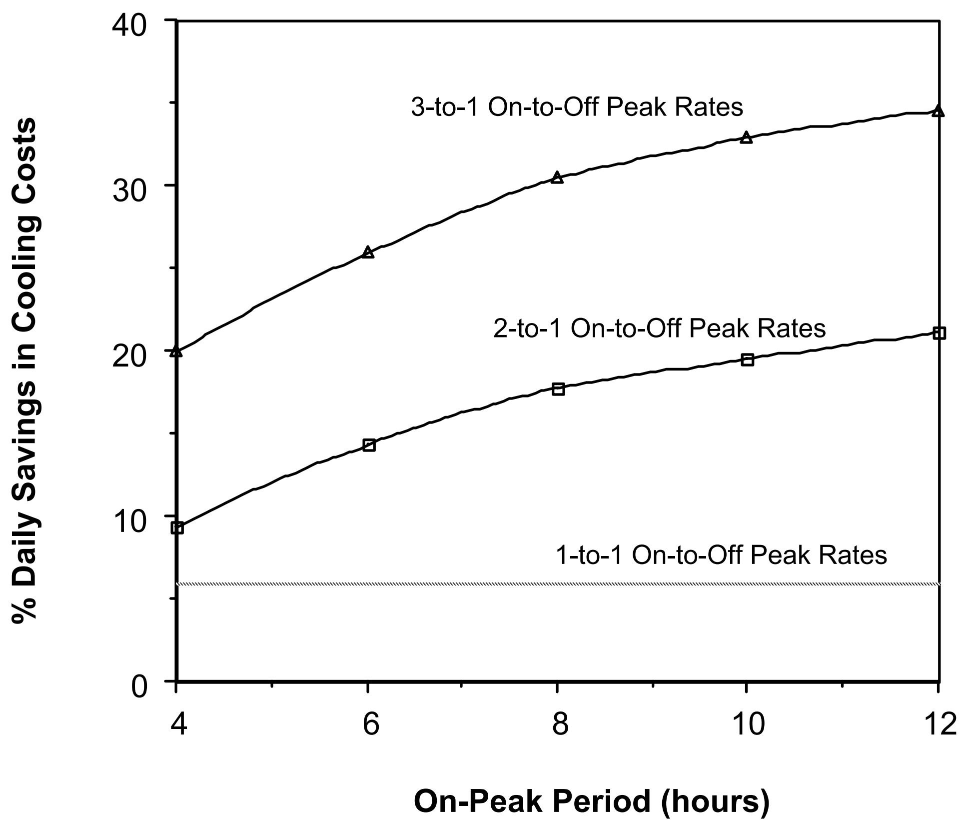


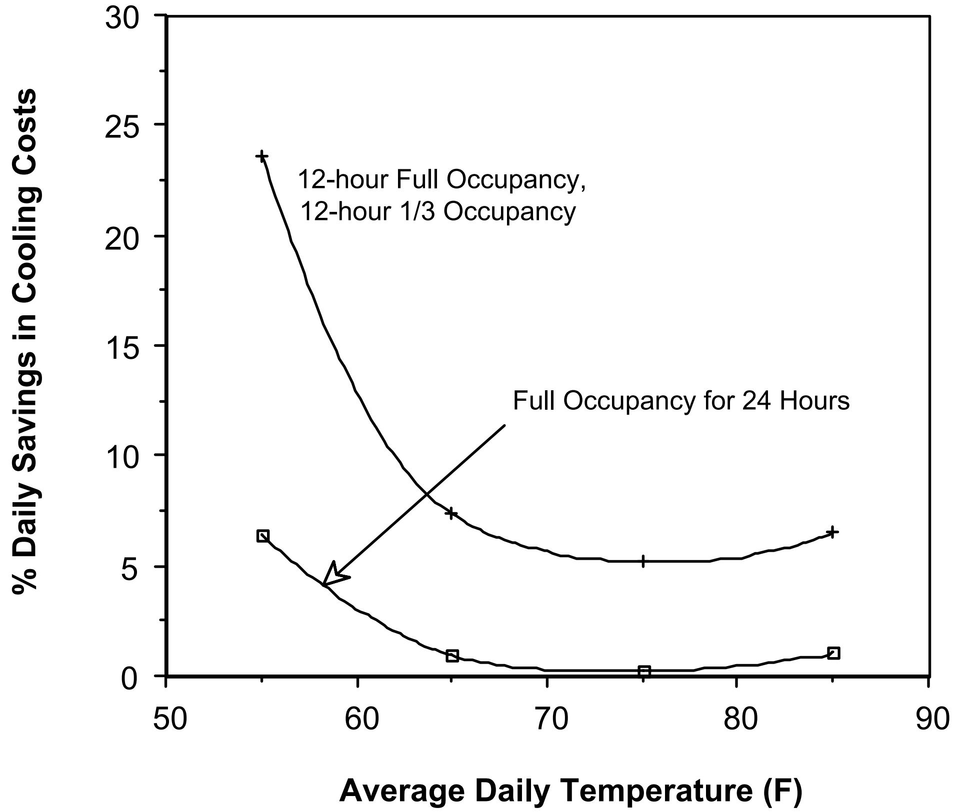
Normalized Daily Energy Use
Night Set-Back Control
Optimal Dynamic Control
Clearness Index
1.0
0.9
0.8
0.7
0.6
0.5
0.3
0.4
0.5
0.6
0.7
0.8



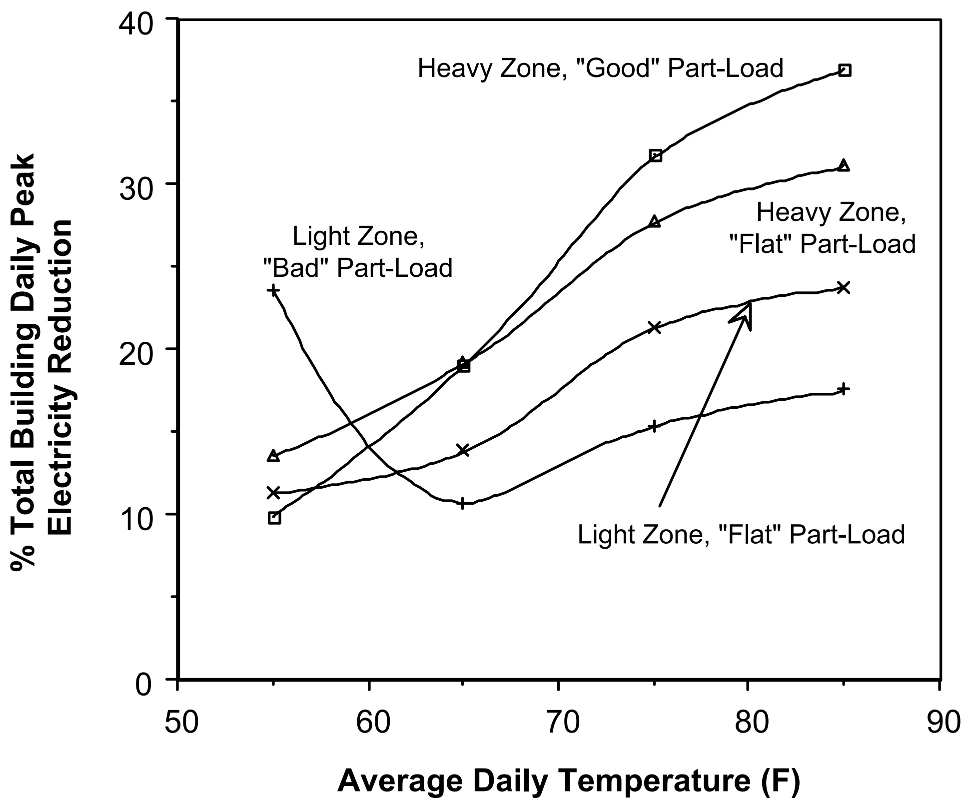
% Daily Savings in
Electrical Use for Cooling
Heavy Zone, "Good" Part-Load
Light Zone, "Good" Part-Load
Light Zone,
"Bad" Part-Load
Heavy Zone,
"Flat" Part-Load
Average Daily Temperature (F)
60
50
40
30
20
10
0
50
60
70
80
90


60
50
40
30
20
10
0
50
60
70
80
90
% Daily Savings in Cooling Costs
Average Daily Temperature (F)
Heavy Zone, "Good" Part-Load
Light Zone, "Bad" Part-Load
Heavy Zone, "Flat" Part-Load
Light Zone,
"Flat" Part-Load

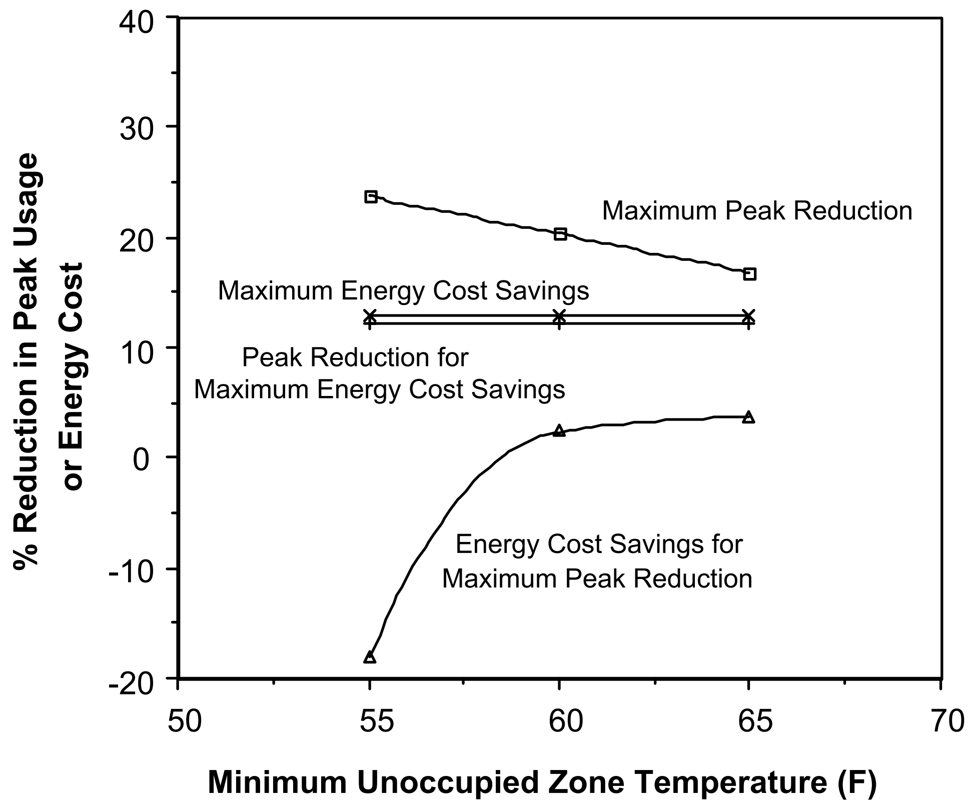

40
30
20
10
0
4
6
8
10
12
% Daily Savings in Cooling Costs
On-Peak Period (hours)
3-to-1 On-to-Off Peak Rates
2-to-1 On-to-Off Peak Rates
1-to-1 On-to-Off Peak Rates

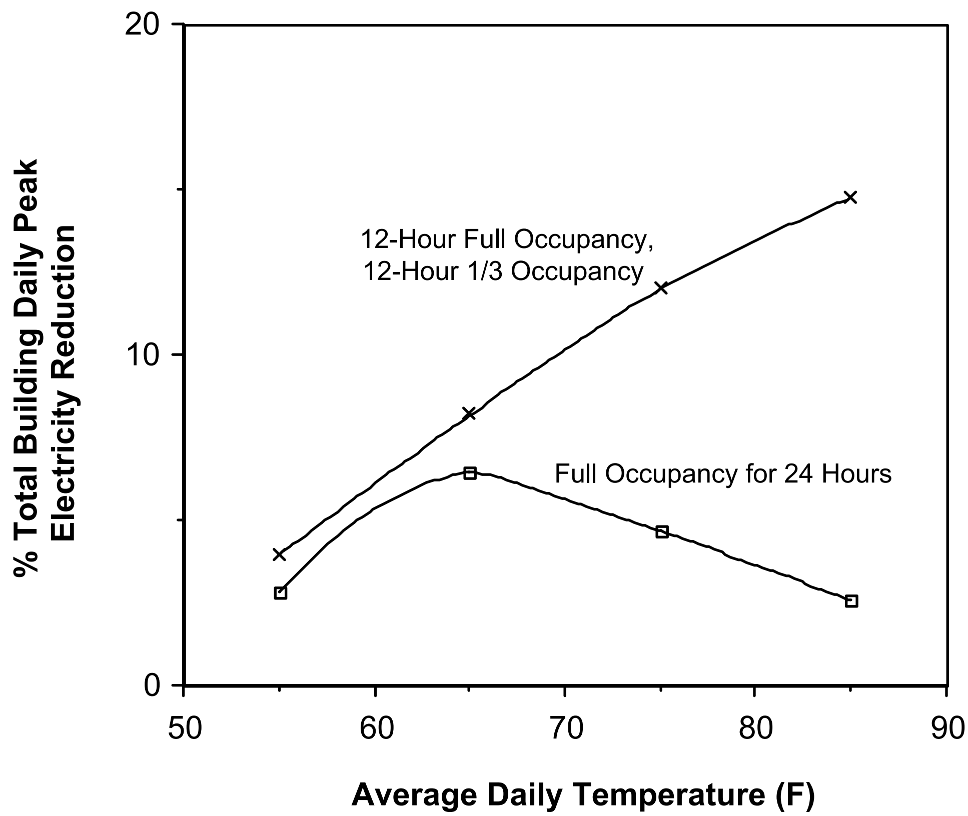


% Daily Savings in Cooling Costs
12-hour Full Occupancy,
12-hour 1/3 Occupancy
Full Occupancy for 24 Hours
Average Daily Temperature (F)



% Total Building Daily Peak
Electricity Reduction
Heavy Zone, "Good" Part-Load
Light Zone,
"Bad" Part-Load
Heavy Zone,
"Flat" Part-Load
Light Zone, "Flat" Part-Load
Average Daily Temperature (F)


40
30
20
10
0
-10
-20
50
55
60
65
70
% Reduction in Peak Usage or Energy Cost
Minimum Unoccupied Zone Temperature (F)
Maximum Peak Reduction
Maximum Energy Cost Savings
Peak Reduction for Maximum Energy Cost Savings
Energy Cost Savings for Maximum Peak Reduction


20
10
0
50
60
70
80
90
% Total Building Daily Peak Electricity Reduction
Average Daily Temperature (F)
12-Hour Full Occupancy, 12-Hour 1/3 Occupancy
Full Occupancy for 24 Hours