OFF-PEAK COOLING USING PHASE CHANGE MATERIAL

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The electric utilities in the United States are faced with continued rapid growth in electrical demand. The traditional response to growth in demand has been the expansion of generating capacity. However, economic, environmental, and scheduling constraints will combine to make adequate expansion difficult, if not impossible, in the 1980's. This thesis examines load management as an alternative to the proliferation of generating plants. This path is illustrated by the development of an air conditioning system designed to displace power consumption in commercial buildings from peak to off-peak periods.

The urban domestic utilities face their peak loads during the summer air conditioning season. The displacement of daytime air conditioning will therefore reduce the utilities' annual peak load. The proposed air conditioning system uses off-peak power, and conventional mechanical equipment, to recharge a thermal energy storage system. The thermal storage medium is a sodium sulfate-based phase change material (PCM) enclosed in small thin bags. These bags are distributed throughout the ceiling plane of the building; supported by special ceiling tiles. At night, the PCM is charged by chilling the plenum space; during the day, the PCM, in direct contact with the occupied space, removes sensible heat gains by melting. System components are described by their programmatic requirements.

The performance of the proposed off-peak cooling system was simulated under a variety of conditions involving interior zone commercial office spaces. It was found, using thermal comfort criteria, that the system performed well using a PCM set point temperature of 67°F. Under the operating parameters established, the simulations indicated that the energy flow rates were adequate for successful system performance.

Finally, it was determined that the system can provide immediate savings through customer demand leveling and increased chiller efficiency. However, the major potential for savings will relate to anticipated utility incentives for off-peak electrical use.

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Title:	Research Associate

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This thesis is dedicated to Claudia with love.

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1.

Introduction

The electric utilities face major problems. They must provide electrical energy to meet the exponentially increasing energy demands of our society. Accompanying the general increase in electrical energy utilization is a steep rise in peak load demand. For the majority of electric utilties, this peak load has shifted from winter to summer in response to the acceptance of air conditioning. In general, the utilities cannot store generated energy. They must generate in a pattern that matches the demand placed on the system. This shifting demand pattern, with peaks and valleys, requires that a mix of generating plants be available to meet the changing load levels. Peak loads are commonly met by operating the system's least efficient generators. The utilities must provide enough capacity to meet their yearly peak load. The rising summertime peak loads have made it necessary for the utilities to double their power system capacity approximately every ten years. This represents a substantial investment of capital. Approximately $125 billion will be required nationwide in the decade between 1970 and 1980.

This thesis begins with the analysis of alternatives to the rapid expansion of generating capacity. The situation faced by the utilities is examined in depth and the benefits of load management are examined. The alternative offered by load management is examined in detail through the development of a new off-peak air conditioning system. The idea underlying off-peak air conditioning is to store coolness at night and use it to meet cooling loads during the day. This system, in effect, displaces the consumption of electricity from peak periods to nighttime off-peak periods. This redistribution of the power load improves the utility system load factor. The improved load factor results in less fuel consumption and environmental impact. Another important benefit is a reduction in the amount of capital required.

The off-peak air conditioning system proposed in this thesis is developed through a conceptual stage. The system performance is predicted through numerical simulation of the thermal energy flows. Investigation of system performance is limited to implementation in the interior zones of commercial office structures. Finally, the economic benefits of the proposed system are examined under the existing utility rate structure. The off-peak cooling system is but one of the many innovations necessary for our nation's vitality in this new age of valuable energy. It offers a potentially important change in our patterns of energy use.

2.

An Electric Utility Perspective

It was a century ago that Thomas Alva Edison invented the electric light bulb. That invention heralded the birth of an industry that has grown to be an integral part of every life in the United States. In the past century the electric utilities have experienced a growth rate paralleled by few other enterprises. Today they collectively form the nation's largest industry. But the winds of change are upon us. The utilities are now faced with unprecedented obstacles to their continued rapid growth. Proposed growth rates for generating capacity are at a scale that threatens the financial stability of the system. The public, meanwhile, is increasingly aware of the social and environmental consequences associated with unabated expansion. Having long taken a limitless supply of inexpensive electricity for granted, we have developed extravagant and wasteful habits. At this time we must carefully re-examine our electric needs. Future industry policy could range from the continuation of historic growth rates to a program of no growth. The path ahead lies between these extremes in establishing a growth pattern that achieves a rational balance between the benefits and costs of large scale electrical power generation.

THE GOLDEN AGE

The challenges currently facing the electric utility industry are better understood with a little historical perspective. In the earliest days of the industry, electricity was provided during the hours of darkness only and the generators would be shut down during the day. This idle investment during the daylight hours led to the first utility promotion of off-peak use , the introduction and support of electric irons and washing machines. This soon led to plant operation full time on Mondays and Tuesdays.[1] The pattern thus established, expansion and promotion, continued to fuel rapid growth for 80 years. During this period electrical energy production never failed to double every 9 or 10 years. There were, however, several changes in the structure of the industry as its impact on American life increased.

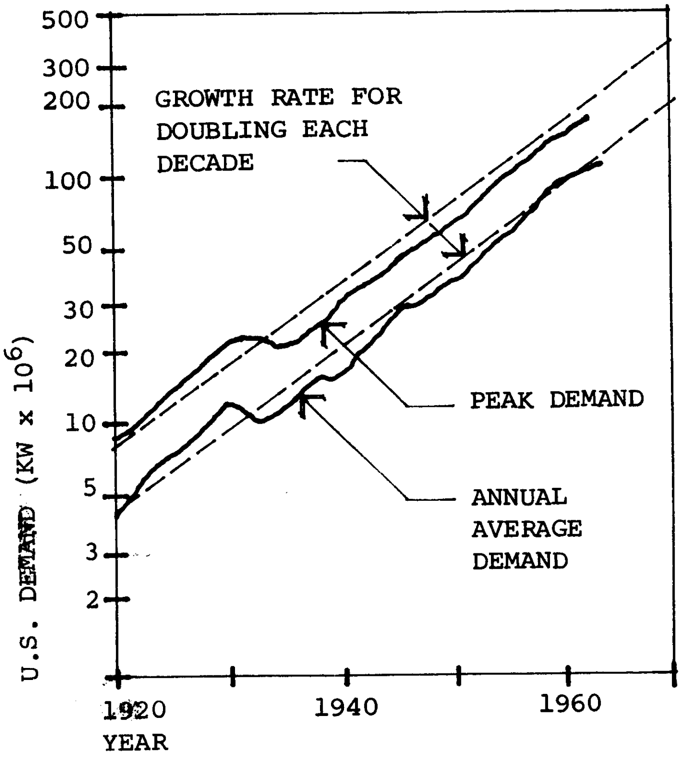


FIGURE 1 Electric Power Demand During the Past Half-Century[2]

In the first quarter of this century, electricity was produced by small companies serving localized areas, often single customers. In the 1920's many of these companies were consolidated, by way of the holding company, in order to realize advantages of economy-of-scale.[3] The holding companies could afford the capital and skills required to expand at a rapid pace. The result was the construction of high capacity facilities with increased efficiencies.In the late 1920's, the Federal Trade Commission began a massive investigation of holding companies. This investigation ultimately spawned the Federal Power Act which provided for federal regulation of the wholesale transactions of investor owned utilities. At the same time, the courts established the state's prerogative to regulate retail power sales. The electricity utility industry became an officially recognized monopoly under regulatory control. This control was exercised to limit profits to a "reasonable amount". As profits were limited the utilities channeled prosperity into growth rather than profit. The utilities became a capital intensive industry with growth rates that required an investment to earnings ratio of 5 to 1.[4] By comparison, the ratio for the steel industry is 1 to 1. The American economy had little trouble absorbing the increased energy output.

The industry fared well in the years between World War II and the mid-1960s. This period was characterized by continued growth, and stability in all of the factors affecting industry well being. Load characteristics, both peak load and total consumption, were well understood and varied little from historic growth.[5] Plant expansion programs were designed to provide for this growth with new, large, efficient baseload plants. As the older inefficient plants were retired, or moved to peaking duty, the cost of electricity decreased. Utility revenues increased in the same period with a steady growth in new customers and electric power consumption per customer. As engineering methods improved, the cost of constructing additional capacity decreased on a per kilowatt basis. Even the cost of crude oil showed a generally declining trend during this period. Stable rate structures required little interaction between the utilities and government regulatory agencies. Finally, capital sources were abundant with a large proportion of capacity financing handled in house. This was the golden age of the utilities.

This golden age also saw the advent of the most significant development in the United States' utility picture, the introduction of comfort air conditioning. In the late 1940s, air conditioning was introduced to the public in commercial spaces and quickly became popular. The acceptance of air conditioning began a trend in which urban utilities experienced a shift in yearly peak load from winter to summer. Yearly peak load denotes the greatest instantaneous demand placed on a system during the year. This is a significant value for it determines the amount of generating capacity that a system must have available. The summer peak started on the Gulf Coast, then swept up the Mississippi Valley and across the nation.[6] During this period, the utilities actively promoted air conditioning, directly and through the advocation of higher artificial lighting levels. They were very successful.

NEW CHALLENGES

In the mid-1960's, the stable world of the utilities began to falter. One set of problems came with an increase in environmental awareness. The public that had lived through the 1930's depression and had come to equate industry with jobs and prosperity was giving way to a younger generation. This new public questioned industrial abuse of the environment and successfully promoted legislation to protect various ecosystems. The utilities were affected in several ways. First, additional review and approval processes for generation plant siting caused long delays in obtaining approval for construction of plants. Construction delays resulted in higher cost for materials and financing. Second, the tighter regulations concerning sulfur content in fuels resulted in sharp price increases for low sulfur fuel. These price increases were just the beginning. The closing of the Suez Canal in 1960 signalled an increase in tanker fees and the beginning of price increases by OPEC. Gas and coal prices followed the escalation of oil with fuel costs rising over 250% between 1965 and 1975.[7] Production costs for the utilities were following suite.

As production costs for the utilities escalated at a rapid pace, capital funds for plant expansion became a major problem. For the first time, the lag between submission of utility requests for higher rates and regulatory approval became significant. Utilities were often caught selling at lower profit margins because of this lag. The resulting drop in revenues affected acquisition of capital funds by lowering the performance of utility stocks. The utilities' expansion plans were closely tied to forecasts of continued growth in peak loads. The projected growth of peak load is the major factor which compels the construction of new plants.[8] With forecasts calling for a doubling of peak load demand within 10 years, the need for construction capital was enormous. This demand grew even larger as inflation affected construction and equipment costs. For example, between 1965 and 1975 the cost for construction of coal fired capacity in Colorado rose from $148/kw to over $500/kw.[9] In addition, interest rates were increasing rapidly, reaching "astronomical proportions by 1973".[10] The uncertainty associated with revenues had the effect of raising the proportions of external funds used for construction,furthering a drop in the utilities' financial rating. In short, the utilities were faced with an enormous investment to finance in the face of lowered revenues, escalating prices, lack of internal funds, a failing financial rating, and sharply inflated interest rates.

As the United States witnessed major changes affecting utility well being, another subtle and important change began to occur. Energy use patterns experienced a shift from the path forecast by the utilities. Under the influences of conservation appeals (induced by the OPEC oil embargo) and the burden of higher prices, Americans began to conserve energy. An immediate effect of this conservation was the further decrease of utility revenues. The utilities, hard pressed for capital, responded by requesting rate hikes to cover the shortfall.[11] Faced with higher rates as a consequence of conservation, an angry public became earnestly involved in the regulatory process for the first time since the 1920's. The public's

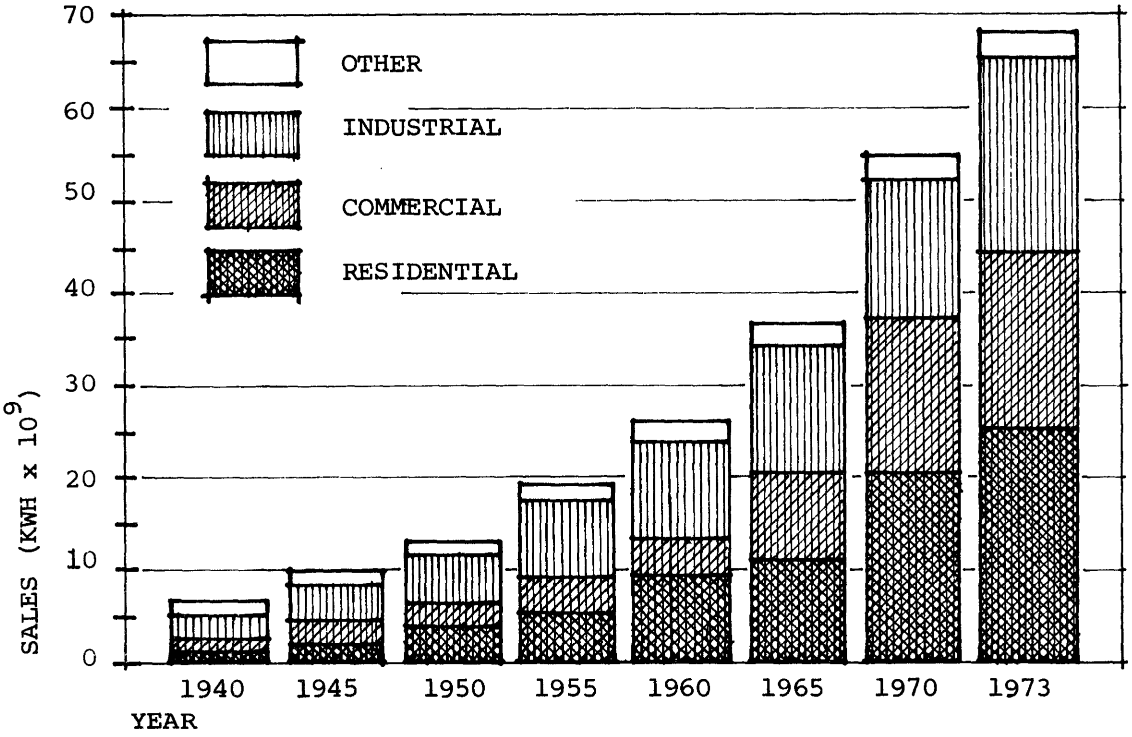


FIGURE 2. Electrical Energy Sales in New England, 1940 to 1973 #12

involvement prolonged regulatory procedures,further delaying capital expansion. The utilities began to cut construction programs despite projected need. The New York Times of August 11, 1974 noted that "shelved construction programs could add up to future shock in the utility business... since the beginning of the year, investor owned utilities have reported reductions in 1974 spending for plant and equipment totalling more than $2 billion." It was becoming evident that the projected growth in demand could not be supported. At the same time, reductions in the growth rate of energy use case doubt on the projections themselves.

I should examine one final factor that contributes to uncertainty about adequate electrical capacity in the future. The utilities have embarked on an ambitious program to bring large scale (1,000 mw plus) nuclear plants on line for baseline service. Nuclear plants promised further economies-of-scale through lower fuel and production costs. This promise has not been realized. When operating, nuclear plants do produce electricity inexpensively. However, experience has shown that these large complex plants spend high percentages of time off-line due to forced outages and scheduled maintenance.[13] This point is well illustrated by a March 1979 order from the Nuclear Regulatory Commission closing five Northeast nuclear plants for emergency system inspection. Excessive time off-line causes higher production costs that eliminate the desired economies-of-scale. In addition, as the size of an individual plant increases, so must provisions for reserve capacity. Nuclear plants are also characterized by lengthy approval and construction times. Lead time for construction of a large plant is now taken to be in excess of 12 years.[14] Because of this long lead time, expansion programs must be based on long range forecasts, leaving little flexibility in the event of error. Beyond these problems, mounting public concern over safety, ethical, and environmental issues is lending an uncertain air to the future of the nuclear program.

In summary, the utilities today face severe challenges. While revenues and load growth have stabilized since the oil embargo, the experience of the early 1970's has left an impression of vulnerability. Reserve operating capacities have been reduced. Construction programs have been delayed by a myriad of problems, while costs have skyrocketed. New plants have performed at lower than expected capacity factors. Environmental and regulatory procedures have claimed time and capital resource. These problems plus the long lead time for new nuclear baseload plants, combine to create a situation where "the probability of a shortfall in electric generating capacity may increase significantly in the early 1980s."[15] This potential shortfall has implications beyond the shortage of electricity. Short term attempts to fill this shortfall would involve expansion of fuel intensive peaking units due to their short construction time. These plants would compete for scarce oil and gas, putting heavy pressure on these markets and heightening our dependence on unreliable foreign supplies.

AN ALTERNATIVE TO SHORTFALL

The gloomy prognosis I have painted for the utilities can be brightened a good bit by reexamining some basic assumptions. The first assumption, one used by utilities in forecasting future loads, is that electrical demand will continue to follow patterns established during earlier, less complicated times. These forecasts are the cornerstone of utility expansion programs, establishing the amount a utility must attempt to invest in capacity. Projections by the Electric Power Research Institute use an approach that "emphasizes how people actually do use energy rather than how they could or should use it... (making no) allowance for such non-price factors as an energy conservation ethic... So far, this assumption is consistent with postembargo consumption experience."[16] In effect historical trends are extrapolated to yield future needs. There is no attempt to establish capacity requirements based on need, no recognition of excessive waste in current energy use patterns. It is in this fact, according to Richard Stien, that our salvation may lie.[17] If the projections by the EPRI do not recognize an energy conservation ethic, this is a circumstance caused by retail energy pricing structures, not lack of conservation opportunity. Beyond gains from the reduction of energy consumption per se, there are opportunities to examine and shift the timing of consumption. A shift of consumption to periods of excess generating capacity can result in better utilization of existing facilities.Therein lies the tale of this thesis.

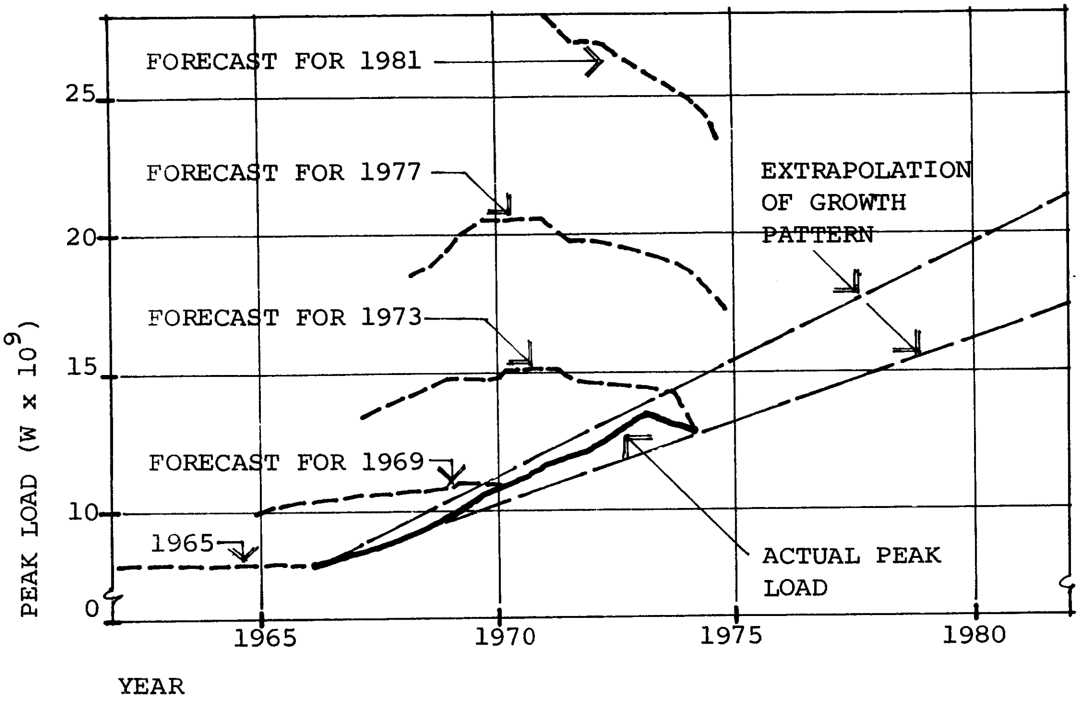


FIGURE 3. New England Peak Load Forecasts vs. Actual Growth[18]
(Forecasts vary with date on abscissa)

The term energy management encompasses all paths available to the utilities for improving their efficiency. This efficiency is measured by several indicators as described below:

LOAD FACTOR is the ratio of average demand to peak demand for a given time period, usually a day or year. This ratio can be calculated for either individual customers or for an entire utility system. Yearly load factors are an indicator of system utilization.

CAPACITY FACTOR is the ratio, for a given time period, of a system's total output in kwh to the system's maximum potential output (all generators running full time). This is a better indicator of system utilization because it reveals the magnitude of reserve equipment, forced outages, and scheduled maintenance.

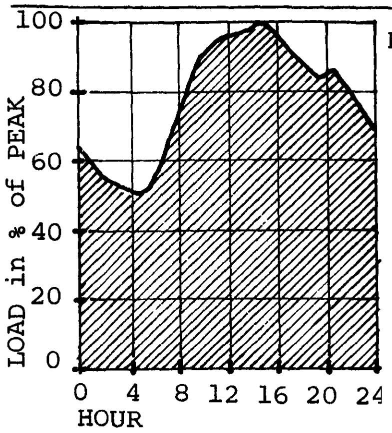


FIGURE 4

DAILY LOAD CURVE is a graphical representation of electrical demand vs. time over a 24 hour cycle. This demand may be the needs of a customer or the load placed on a utility system. The highest point on the graph represents the time and magnitude of the day's peak load. The illustration represents a summer peak day for Boston Edison.[19]

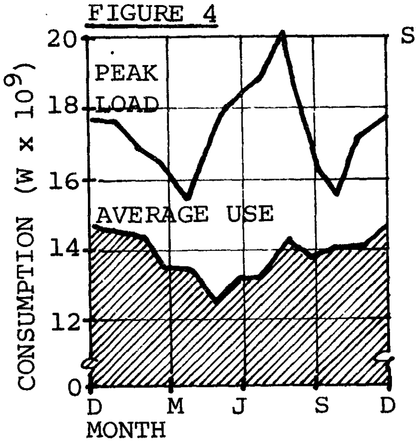


FIGURE 5

SEASONAL LOAD CURVE is a graph showing demand vs. time on a yearly basis. Daily peak loads and average loads are plotted. Graph peak represents the day and magnitude of the annual peak load. The illustration represents 1978 data for Boston Edison.[20]

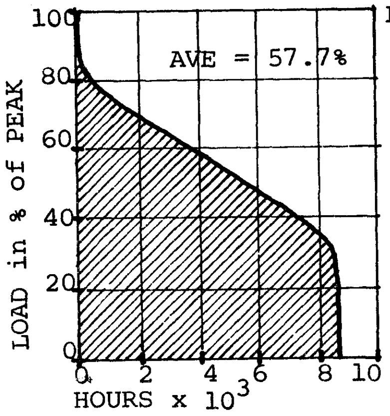


FIGURE 6

LOAD DURATION CURVE portrays the same information contained in the previous load curves except for the timing of demand. Demand values are plotted in descending order with 24 hour values for a daily curve and 8760 hourly values for a yearly curve. This graph is normally constructed for utility systems and is useful for visualizing the character of peakload vs. baseload. This curve represents Boston Edison in 1978.[21]

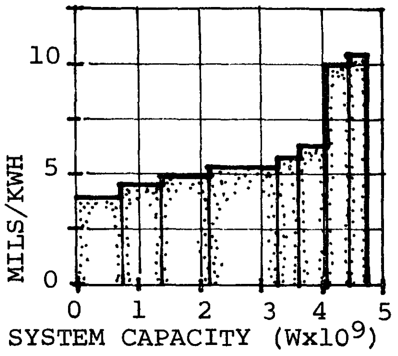


FIGURE 7

ORDER OF MERIT is a listing of power stations in an electrical system in the order of running costs. The cheapest stations are run as baseload stations, the most expensive only for infrequent peak loads, and the rest in between according to their merit order. This figure illustrates the order of merit for the Philadelphia area.[22]

In 1975, Frank Zarb, Administrator of the Federal Energy Administration announced the following goals for the nation's electrical industry.[23] Increase the overall load factor from an existing 62% to 69%. Improve existing capacity factor from 49% to 57%. This would allow an increase in baseload generation from 45% of total load to 55%. If these goals were met in combination with a 10% increase in end-use efficiency, foreign oil imports could be reduced by 1.3 million barrels a day. The industry's existing utilization of less than one half of its installed generating capacity leaves ample margin for improvement.

MANAGEMENT IN LIEU OF EXPANSION

Utility programs for energy management can fall under three major categories: supply management, end-use conservation and load management. Supply management includes all measures involving manipulation of a system's generating capacity. One of these measures, pooling, has been responsible for a reduction in nationwide reserve capacity.[24] Pooling is the interconnection of neighboring utilities into a regional grid such as NEPOOL in New England. Member utilities can aid each other in the event of forced outages, or load imbalances. Supply management also includes programs for storing energy at the utility scale. The only feasible method to date for utility scale storage is the use of pumped hydro storage. Specialized site requirements will prevent widespread adoption of this method. Supply management decisions include the selection of capacity mix for a given load situation and the planning of capacity additions. The utilities have traditionally been involved in supply management, shaping their systems to best respond to an "independent load curve". However, it is apparent that the utilities own cost and pricing patterns, functions of supply management, are very influential on the nature of this "independent load curve".[25] Load management techniques call for a more direct interaction.

The seasonal load factor is a good measure of utility system efficiency. Maximum theoretical efficiency in a utility would occur in a system that experienced a steady load at all times,with customers willing to interrupt their service for utility maintenance and forced outages. This peakless condition would result in a load factor of 1 and, if the equipment required infrequent maintenance, the capacity factor could be above .85. In reality, these conditions never occur, utilities serve loads that vary with time and customers that require continuous service. Generating units required for peak load conditions stand idle for long periods and the utilities must maintain reserve capacity for eventual outages. Load management techniques seek to increase system efficiency by manipulating the nature of the loads served by the system.

The use of load management techniques to improve system efficiency is not a new development, or one unique to the electric power industry. The utilities have long sought to improve load factors by promoting electrical use during times of excess generating capacity. For example, as mechanical cooling gained acceptance, utility sales organizations have selectively promoted electric heating to raise winter consumption rates, and thereby, the seasonal load factor. There is now a new, and hopeful connotation for the term load management. As used today, load management still seeks to improve system load factors, not by soliciting new consumption, but by shifting consumption from peak to off-peak periods. This is an important distinction, for in addition to improving system utilization, load management now aims to reduce peak capacity requirements. Industries with large capital investments have long recognized the benefits of load management. The Bell System uses load management, in the form of time of day rates, to shift personal calls from crowded daytime business hours to times of excess system capacity. This results in less equipment required and a lower percentage of idle equipment at night and on weekends. The airlines offer similar rate incentives for late night air travel. During the 1950s and 1960s, scale economies and technical advances in electrical generation made load control unnecessary.[26] This no longer holds true in the 1970s.

An implementation of load management measures could simultaneously further the goals of utility, consumer, environmental, and government sectors. The utilities would benefit in short and long term categories. In the short term, improved utilization of existing facilities would reduce pressure for generating capacity expansion. Load management techniques could reduce the need for new installed capacity by 33% in the next ten years.[27] This represents a potential savings in excess of 50 billion 1974 dollars. Reductions would also be made in the expansion of transmission and distribution facilities. A reduction in today's tremendous pressure for expansion would give us time to reaccess the goals and consequences of our current national energy policy.

The utilities would also benefit by a long term improvement in generating efficiency. Because there is no economical way to store electricity at a large scale, the utilities must generate electricity to match demand as it occurs. This production is provided by a mix of generating plants. As load varies, the proportion of different types of plants on line will vary. The most efficient type of generating plant is the baseload plant. Baseload facilities, usually new, large plants with nuclear or fossil fuel drive, must be run continuously. This limitation restricts their use to a level of loading that occurs throughout the day. The remaining load, as it varies during the day, is carried by older, less efficient plants called cyclers. The extreme peak portion of a systems' load is assigned to inefficient plants known as peakers. Fuel cost for peakers can run three times the cost of fossil fueled baseload plants and ten times the fuel cost of nuclear plants. Load management, by improving a systems load factor, can increase the share of load assigned to the more efficient baseload plants. The end result is lower fuel cost per kwh produced.

Load management also benefits the consumer. Savings realized by the utilities from decreased fuel use would be passed on via a lowered fuel adjustment charge. Reduced pressures for capacity expansion could lower the cost of capital acquisition by raising the contribution of internal funds. The utilities would not be forced to burden the consumer with charges for construction work in progress. Beyond this sharing of utility benefits, customers will be in a position to benefit from personal decisions based on the rate incentives essential to load management. By carefully considering the timing and nature of his energy requirements, the consumer can program his energy use for maximum benefit to the utility. In return, the utility will share these benefits with the consumer in the form of lower rates. The consumers who are unable, or unwilling, to modify their use patterns will be faced with charges that accurately portray the burden they place on the system.

Environmentalists have long urged the adoption of load management techniques as an alternative to the present rate of plant proliferation.[28] The utilities are experiencing increasing opposition to plant and transmission expansion from groups concerned about environmental and aesthetic issues. In addition, sites well suited to hydro facilities, and those offering appropriate thermal sinks for large baseload plants are becoming rare.[29] The advent of load management can offer an extension in the time available to evaluate the complex and vital environmental issues involved in generating capacity expansion. The government would benefit from a reduction in the consumption of fossil fuels by peaking plants. We are faced with a dependence on foreign oil that weakens not only the national economy, but our security as well. In response to the OPEC oil embargo, the Office of Emergency Preparedness published goals for short, medium, and long term periods. Manipulation of the daily electrical system load (i.e., increasing the load factor) was the only measure recognized in all three categories. Reduction in the use of liquid fossil fuels improves our balance of payments and preserves this resource for more appropriate use.

LOAD MANAGEMENT TECHNIQUES

To realize the advantages of load management, utilities must design and implement programs encouraging the displacement of peak energy use. There are several approaches to this end, comprising two major categories. The first category, hard load control, includes methods that allow a utility to command physical control of a customer's electrical demand. Thirty years ago, Detroit Electric Company instituted a hard control load management scheme for electric hot water heaters. Timers were used to limit the operation of 200,000 electric hot water heaters to off-peak hours. Although this program was allowed to lapse during the 1950s and 1960s, it was considered successful and has recently been revived. Technology has provided promising devices for hard control. With one of these, ripple control, the utilities can control the status of customers' equipment with one way signals transmitted through the power lines. Much work is now being done on two way devices. With physical control over the load imposed by specific end uses and/or specific customers, a utility dispatcher can carefully orchestrate system generation for efficient fuel consumption. The customer's electrical rate is modified in compensation for his modification of end use hardware and patterns. Utilities are attracted by the degree of control provided with this type of system. The dispatcher need restrict electrical use only during the exact periods that threaten a balanced load profile. Historically, the utilities have planned as though the cost of supply inadequacy was infinite.[30] The advent of hard control proposals may lead to a reevaluation of this premise to the mutual advantage of utility and consumer.

Load management goals can also be achieved through the implementation of soft load control measures . Measures in this category rely on consumer response to economic incentives, usually in the form of off-peak or time of day rate structures. This type of load management has been practiced successfully in many foreign countries including West Germany, Britain, France, New Zealand, and Australia.[31] Customers in these countries, responding to electrical rate structures, have shifted certain types of energy use from peak periods to off-peak periods. With less air conditioning equipment than the United States, Europe experiences peak demands during the winter heating season. Both Britain and West Germany have approximately 150,000 MWh of off-peak heating storage. In Germany, this storage represents a 40% reduction in system peak load.[32] Britain has a program combining tariffs for cheaper off-peak electricity and the development of special electric products tailored to customer needs.Public acceptance has been achieved through wide ranging promotion reminiscent of our domestic utility sales promotion during the 1960s. Market research indicates a 95% satisfaction rate with Britain's off-peak storage heating systems.[33] While major differences exist between the electrical systems in Europe and the United States, the success of soft control measures in Europe supports their potential in America.

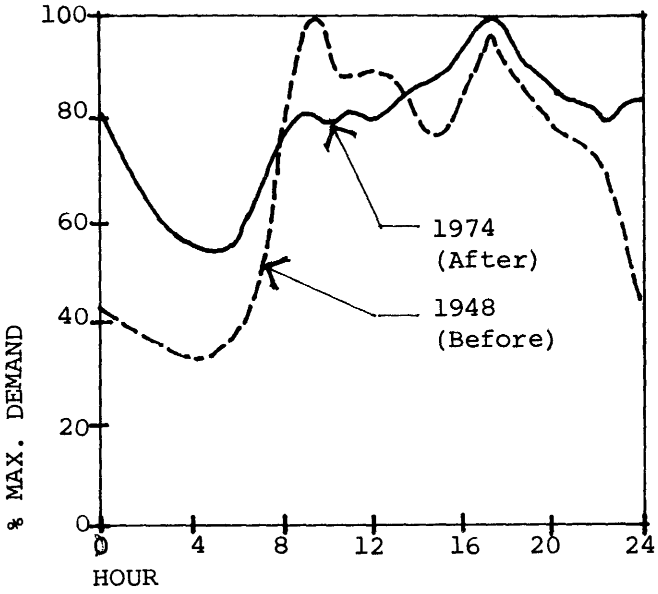


FIGURE 8. Comparative shape of British Load Curve before and after Load Management.[34]

EXISTING RATE STRUCTURES

The introduction of soft load controls in the United States will require a departure from the pricing structures now used by the industry. Current rate structure is a complex issue involving historic precedent, the monopolistic nature of the industry, and regulatory involvement. Existing rates are not designed to maximize profit, which is limited by regulation, nor do they reflect the true costs of service. The existing rates are designed in a manner that encourages the growth of electrical use over time, particularly in time periods that smooth an individusl customers load curve. These rates were established in a historical context reflecting decades of energy growth with declining costs. While these tariffs are periodically raised to cover the effects of inflation, their survival is testament to the conservative nature of the industry.

The rate charged per unit of electrical power (kwh) consumed will depend on who the customer is, the amount he consumes, and in some cases, how much he consumes at one time. Residential rates, for example Boston Edison Company's Residential Rate B, charge the customer for energy consumed on a declining block basis. Based on research suggesting that the price is the most important determinant of the quantity of electricity demanded, the declining block rate can be considered to encourage additional consumption. The residential charge includes no specific component for contribution to the system's peak load and the expenses of answering same. Residential customers are also charged a fuel adjustment charge based on the average cost of fuel used throughout the preceding month. Again,this charge does not recognize peak period consumer demands that result in inefficient use of base fuels.[35]

Existing commercial rates also fail to assign the costs associated with demands placed during peak load periods. These rates, for example Massachusetts Electric Company Rate H, include several different charges. Commercial customers are charged for energy consumed using a declining block rate based on maximum demand increments. In addition, customers must pay a demand charge for the maximum rate of consumption during the billing period. A "ratchet" is placed on the year's maximum demand extending its effect through the year. If a customer's peak demand occurs during system peak periods it necessitates expensive generating capacity. If the customer's peak demand occurs during off-peak conditions, the system load factor is, in fact, improved and no additional capacity is required. Either way, the customer pays the same demand charge.This type of demand accounting provides the customer with incentive to level his own load profile but there is no mechanism to insure that this will be timed to the benefit of the utility. Commercial rates also include the same fuel adjustment charge levied on residential customers. Finally, there is usually a small,fixed administrative charge. Several utilities have recently added a controversial charge for construction work in progress. Again, this charge is not directed toward the specific customers whose demand is responsible for the expansion program. The existing rate structures spread the cost of capacity expansion among all customers whether they contribute to the need or not.

RATE REFORM

As we enter the 1980s, we will continue to desire the amenities of electric power, and we will feel the need for even greater quantities of it. Yet, while we appreciate its value, we must also recognize its cost.

A wide range of interests share the opinion that rate reform offers the vehicle for making the cost of electricity explicit. Quoting the National Association of Regulatory Utility Commissioners: "A major fault with current rate structures is that they ignore peak load costs... We recommend that consideration be given to peak load pricing as a way to relieve some of the financial and operating stress on the system and to assure that the incidence of costs falls on the appropriate user."[36] The Environmental Defense Fund expresses a similar view suggesting that utility growth must be paid for by those who occasion it and benefit therefrom: "Let growth go forward when the consumer is willing to bear full cost."[37] New pricing must serve the objective of improving resource allocation by accurately portraying to customers the cost of serving periodic demand. This is a tall order.

Translating utility costs into a viable rate structure represents quite a challenge. Utility costs include the variable cost of fuel, maintenance, and labor plus the fixed cost of capital investment carrying charges. In addition, production involves administrative costs and cost issues related to outages, environmental impact, etc. Economic theory provides several approaches for quantifying the costs of growth in utility service. In a competitive marketplace, prices would be based on short term marginal cost (STMC), assigning the full cost of expansion to the customer who requires the same. This scheme fails in the monopolistic context of the utilities because it produces extreme variation in price over short time spans and ultimately results in unpredictable revenues. There is agreement among economists that the concept of long term incremental cost (LTIC) is more appropriate for the electric industry.[38] This pricing mechanism distributes marginal costs over large incremental blocks of sale, dampening short term price fluctuations. While the theories for price revision are rigorous, rate designs must make simplifying assumptions and approximations. These simplifications are related to the limitations of contemporary metering technology. There are currently several proposals for rate structures that represent compromise between economic theory and existing technology.

The various rate structures proposed for soft load control typically include charges for energy and demand that vary with time. These rates are commonly labelled as time-of-day, peak pricing, and marginal pricing rates. The prices may vary in discrete periods over diurnal and seasonal cycles. Present metering technology limits the number and frequency of these periods. Time-of-day rates, when compared with existing rates, should provide incentives for off-peak energy use without encouraging growth in total consumption. To do this, the customer should be faced with economic disincentives for energy demand during peak periods, with flat or increasing rates for incremental blocks of use. Rates that do not provide the disincentives will, like their predecessors, encourage growth in total consumption. This situation exists when time-of-day rates are offered as an option. The utilities have accrued a vast amount or experience in sales campaigns directed toward increasing energy use during off-peak periods. On New York's Roosevelt Island, Con Edison recently paid a $500.00 cash premium for each unit constructed with electric heating.[39] The utilities have also gained successful experience with time-of-day rates offered to large commercial and industrial users.[40] The European utilities offer encouraging precedent for the introduction of these rates in domestic residential and commercial categories. Yet, the utilities seem reluctant to initiate rate reform programs on their own initiative. This reluctance is currently being examined in regulatory meetings nationwide.

Electric industry's reluctance to institute time-of-day pricing reflects the conservative nature of the industry. The existing rates have served them well for decades, supplying revenue and promoting rapid system growth. This historic rate of growth, with billions of dollars invested in construction programs, has vested substantial political power to those who have a financial stake in the conventional electric power supply scenario. The industry often argues that "load management seems to mean a basic change in the philosophy of service, calling for the customer's electricity demand to be shaped to fit the utilities generating capacity, rather than the other way around."[41] The change in philosophy, in fact, concerns only the reasons for shaping customer demand.

Utility sales programs have long sought to influence customer demand. The emphasis now is on the promotion of efficiency rather than growth. The consumer benefits by being able to decide, based on true cost, the importance of system growth. The hesitation to influence major changes in consumer energy use habits is difficult to understand in light of the fundamental problems involved in expansion to support these habits. The utilities also express concern that the introduction of new and different rates could jeopardize essential revenues. This is a valid concern that must be addressed by the rate designers and regulatory boards. The various regulatory agencies have taken measures to promote load management techniques. Tn January 1978, the Massachusetts Department of Public Utilities directed the State's utilities to develop plans for adopting time-of-day pricing. In response, Boston Edison Company filed optional experimental rates, including Commercial Rate T.1. This rate was designed as an extrapolation of existing energy use patterns without direct relationship to utility costs. This reduces the effectiveness of the experiment, as does its optional status.

The utilities are also investigating other methods of load control. These possibilities include load management contract rates, peak period demand charges, simple monthly credits for load limitation, and utility ownership of load management hardware. While these mentods have their merit, rate reform on the basis of marginal cost holds the greatest promise because of its direct relationship with cost. The next challenge is the identification of end use categories suitable for the use of off-peak energy.

REFERENCES

1. Shannahan, John H.K., "Load Management--Historical Perspective", Proceedings of the Conference on the Challange of Load Management (Springfield, VA: National Technical Information Service, 1975), p. 5.

2. Philadelphia Electric Company, A Study of Electric Power Generation (Philadelphia, Penn: Franklin Institute Research Laboratories, 1973), p. 2-3.

3. Shannahan, p.6.

4. Saaty, Thomas L. et al., System and Economics Analysis for Off-Peak Air Conditioning (Springfield, VA: National Technical Information Service, 1972), p. 46.

5. New England Regional Commission, New England Power Pool (Boston, MA, 1976), p.2-4.

6. Shannahan, p.6.

7. "Load Management", EPRI Journal, May 1977, p. 7.

8. Habriet, Ernest R., "The Energy Puzzle: Electricity Rates, Space Conditioning, and Load Management", Proceedings of the Conference on the Challenge of Load Management (Springfield, VA: National Technical Information Service, 1975), p. 85.

9. Mills, Gary, "Demand Electric Rates", ASHRAE Journal, January, 1977, p. 44.

10. New England Power Pool, p. 4.

11. Zarb, Frank G., "Keynote Address", Proceedings of the Conference on the Challenge of Load Management (Springfield, VA: National Technical Information Information Service, 1974), p. 2.

12. New England Regional Commission, p. 37.

13. Berlin, Edward, et al., Perspective on Power (Cambridge, MA: Ballinger Publishing Co., 1974), p. 8.

14. "Load Forecasting-- Modeling with Judgement", EPRI Journal, August 1977, p. 12.

15. Federal Power Commission, *The Adequacy of Future Electric Power Supply: Problems and Policies*, (Washington, D.C.: U.S. Government Printing Office, 1976),p. 41.

16. "Demand '77", EPRI Journal, December 1977, p. 21.

17. Stein, Richard G., *Architecture and Energy* (Garden City, New York: Anchor Press/Doubleday, 1977), passim.

18. New England Regional Commission, *Electric Power Demand and Supply in New England* (Boston, MA: 1975), p. 58.

19. Boston Edison Company, *Long Range Forecast of Electric Power Needs and Requirements* (Boston, MA: 1979), p. II-138.

20. Boston Edison Company, personal correspondence, 1979.

21. Boston Edison Company, p. II-132.

22. Altman, A., *Conservation and Better Utilization of Electric Power by Means of Thermal Energy Storage and Solar Heating* (Philadelphia, PA: Towne School of Civil and Mechanical Engineering, 1971), p. 103.

23. Zarb, p. 3.

24. Federal Power Commission, p. LV.

25. "Load Management", p. 9.

26. "Load Management", p. 7.

27. Zarb, p. 3.

28. Berlin, p. 29.

29. Yarosh, M.M., "The Crisis in Power Plant Siting", Mechanical Engineering, June 1971, p. 11.

30. Federal Power Commission, p. 58.

31. "Load Management", p. 8.

32. Ashbury, J.G., et al., Commercial Feasibility of Thermal Storage in Buildings for Utility Load Leveling (Argonne, Illinois: Argonne National Laboratory, 1977), p. 2.

33. Hughes, Geoffrey J., "Some Aspects of the Impact of Electricity Marketing on the System Load Pattern", Proceedings of the Conference on the Challenge of Load Management (Springfield, VA: National Technical Information Service, 1975), p. 59.

34. Hughes, p. 62.

35. Habriet, p. 85.

36. "Should Utility Rates be Redesigned?" EPRI Journal, March 1976, p. 13.

37. Habriet, p. 93.

38. Berlin, p. 24.

39. Stein, p. 200.

40. "Load Management", p.7.

41. "Load Management", p.8.

3.

Load Management Using Thermal Storage

Before treating the specific areas amenable to off-peak pricing, it is useful to examine, briefly, our national patterns of energy consumption. Energy use in the United States can be separated into the major categories of transportation (25%), industry (30%), direct use of fossil fuel for space and water conditioning (20%), and electrical generation (25%). The fuel used in electrical generation produces the useful electricity delivered to consumers and, in addition, a large component of wasted energy. In 1970, for example, the utilities consumed raw fuel worth approximately 17 quads to generate approximately 5.6 quads of electricity.[1] Another 10% was lost in transmission and distribution. Thus, while the utilities consumed 25% of the nation's raw fuel in 1970, they served only 9% of the nation's end use energy demand. Put another way, every BTU consumed (or saved) on the customer's side of the meter represents 3.4 BTU of raw fuel at the plant. The energy that does survive as electricity is sold for industrial (44%), residential (30%), and commercial (26%), end use.[2] The industrial end use component is largely used for direct drive. Residential uses include space

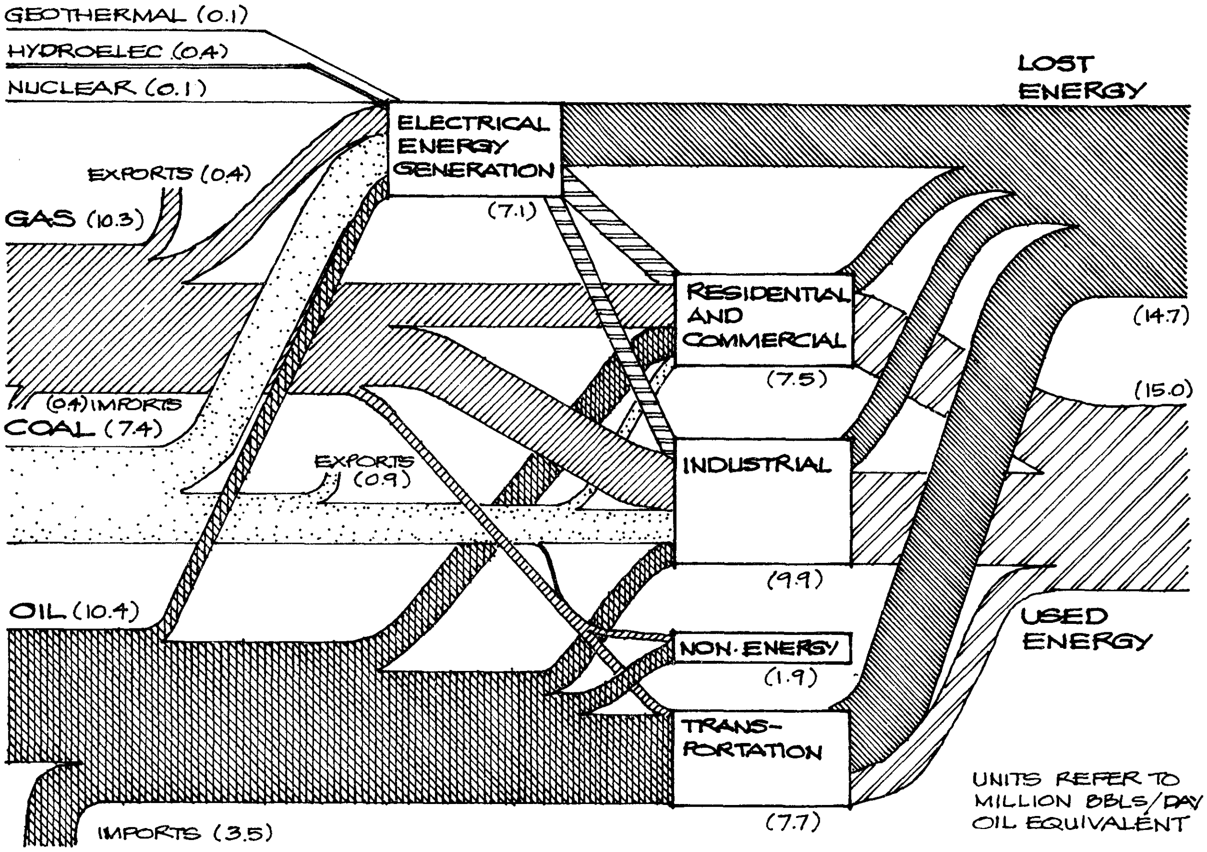


Figure 9. United States Energy Flow Pattern - 1970.[3]

heating, hot water heating, air conditioning, refrigeration. The various scenarios that call for a growth in national energy consumption also call for an increase in the proportion of these needs met by electricity.

As a major energy use category, electrical generation is unique in the fact that raw fuels are consumed in a centralized location, physically remote from the ultimate end uses. The patterns of raw fuel consumption are dependent on the nature and duration of loads placed on the system. Historically, electricity has been generated in quantities that match customers' total demand at any given time. Then, as now, there was no feasible technology for storing electricity at a plant site. The implementation of off-peak pricing will present consumers with a choice between higher prices for conventional peak period use patterns and lower prices for consumption during off-peak periods. This shift in consumption can be accomplished by implementing one or both of the following strategies. The first strategy is simply a shift in the timing of electrical end use. For example, a factory might shift production involving electricity intensive processes into off-peak periods. This involves an additional cost for nighttime labor and is clearly not applicable to all situations. This strategy would prove particularly unwieldy in the commercial and residential categories. Asking the secretarial staff to work during the early morning off-peak period would prove expensive and inconvenient. There is, however, another,more attractive prospect. This strategy involves storing energy generated during off-peak periods to meet loads incurred during peak hours. While the technology for storing electricity at a large scale does not exist at this time, we do have methods for storing energy in other forms. The most promising of these is thermal energy storage.

THERMAL ENERGY STORAGE

Thermal energy storage (TES) is a method particularly appropriate for electrical end use categories that involve thermal energy as an end product. With provisions for TES, a system can convert electrical energy to thermal energy during off-peak periods. The thermal energy is stored and later used to meet loads, in lieu of conventional electrical equipment, during peak load periods. There are quite a few advantages to customer side TES as opposed to centralized storage. This type of system can have a higher round trip efficiency (almost 100%) than central utility storage. Storage on the customer side of the meter results in reduced transmission facility requirements. While the utility scale storage in the form of pumped hydro would be subject to environmental and geological plant siting constraints, neither of these would apply to customer side TES. On the other hand, the utilities would not be able to exercise physical control over customer side TES without investment in a sophisticated control system. However, soft control via time of day pricing, can achieve the goal of reducing demand during peak load periods, and in addition, improve load factors. Customer side TES is a system well adapted to the goals of load management.

There are a number of electrical end use categories well suited to customer side TES. The major candidates, space heating, water heating and air conditioning represented 17% of the electricity sold nationally in 1973.[4] Today the utilities are actively promoting TES for domestic hot water. This system is easily realized because the storage medium, hot water, is the end product. A combination of rate incentives by the the utilities, code requirements for conservation, and good mechanical performance has led to rapid consumer acceptance. Storage systems for space heating are at various stages of development with primary emphasis on the integration of storage and solar energy. Charging this storage with off-peak electric heating can easily be included in this type of scheme. The greatest potential lies in the application of TES to air conditioning systems. Air conditioning is a major contributor to the summertime annual peak load experienced by most urban utilities. This is the type of load, one that contributes to the maximum annual peak, that must be displaced to off-peak periods. Thus, the winter peaking utilities of Europe have concentrated on TES for space heating. The United States must forward research and development in TES systems for space cooling.

TES AND AIR CONDITIONING

It is quite a technical achievement that air conditioning has become such an integral part of our architecture, and our lives, during the short period since the end of World War II. The impact has been tremendous. Commercial buildings designed during the last two decades are totally dependent on mechanical ventilation and air conditioning systems. Mechanical failure or power outage, in effect, renders these buildings uninhabitable. The form and fenestration of contemporary structures relate to abstract stylistic principles or first cost economic criteria, while mechanical systems, and an abundant supply of energy, hold the responsibility for thermal comfort. Modern commercial structures are characterized by large interior zones that require 100% artificial light. As recommendations for lighting levels increased over the years, these interior zones required cooling year round to maintain comfort conditions. The adoption of air conditioning in commercial spaces led to the acceptance and proliferation of residential air conditioning systems. Americans became acclimated to mechanical cooling, with air conditioning becoming a common feature even in automobiles. The acceptance of air conditioning resulted in an increasing use of energy in buildings. The advent of high energy prices is leading to a reexamination of the design decisions responsible for high air conditioning loads. One can assume that future commercial buildings will still require air conditioning, albeit more efficient systems. Because of the characteristically long life of buildings, new, more efficient systems are also required for retrofit into the existing stock of buildings.[5] Thus we must simultaneously strive to limit our dependence on mechanical cooling and increase the efficiency of the systems that are required.

The introduction of thermal storage capacity into an otherwise conventional HVAC system can provide an important degree of operational flexibility. This flexibility allows system operation to be scheduled in a manner that forwards two major goals. First, the operation of major mechanical components can be scheduled during off-peak periods accruing benefit to consumer and utility. Second, the efficiency of various components, especially the chiller, can be improved by running the system at design capacity during the off-peak charging period. The operational strategy for off-peak cooling is straightforward. In response to utility incentives, the bulk of chiller operation would be shifted to designated off-peak hours. During this off-peak period, the chillers would be used to charge cool storage. In all likelihood, the chillers would be run at maximum capacity until the thermal storage had reached full capacity. The charged storage would be used to meet part, or all, of the heat loads that occur during the next peak period. To be of real benefit to the utility, the storage system must be consistently capable of displacing a building's air conditioning load from the peak load periods. The typical profile of a summer peaking utility has a peak load that is not pointed but rather shaped like a mesa. This type of profile mandates that off-peak systems be able to displace a peak load that occurs throughout a design day. Thus storage capacity should be sufficient to handle a full design day cooling load.

The consumer benefits derived from the use of cool storage are not totally dependent of the adoption of time-of-day rates. Under existing tariff systems, based on the Hopkinson rate structure, an individual customer still accrues substantial benefit by improving his individual load profile. The Hopkinson rate includes direct and implicit charges for maximum demand placed on the system during a billing period. By scheduling chiller operation during nighttime hours,a customer can remove the chiller component from his daytime peak with a corresponding reduction in demand charges. In addition to savings found within the various rate structures, a customer will benefit from improved system efficiency. Chiller operation during nighttime periods allows the system to reject heat to more receptive, cooler, nighttime atmospheric conditions. The most dramatic realization of this improvement would occur when climatic conditions permit the direct use of cool outside air to charge the storage. The natural diurnal cycle would replace a mechanical chiller as a cooling source. Thermally massive walls and floors have long been used in hot desert regions to moderate daytime heat gain by retaining nighttime radiant loss.[6] In hot arid areas today, large diurnal temperature swings and effective thermal storage systems can combine to eliminate, rather than displace, mechanical chiller operation. However, the majority of the country must contend with a less favorable climate. These areas will still realize an increase in efficiency by displacing chiller operation from the warmest part of the day to more favorable, cooler hours.

A conventional air conditioning system is designed to provide comfort conditions during the warmest days of the year. During these brief spells of very hot weather, a system's chiller operates at its designed maximum chilling capacity. Large chillers operate at their greatest efficiency when run at full load, but as the load is reduced their efficiency drops. Without cool storage a system must respond to cooling loads as they occur and at the rate in which they occur. Therefore, systems designed for maximum load conditions must be run at partial capacity to meet an average day's load, with a consequential reduction in overall mechanical efficiency. Using cool storage, the chillers can be run on a schedule designed for efficient use of electricity. As the nighttime charging period begins, the chillers are run at full capacity until the previous day's cooling load is removed from storage. The main chillers are then shut down until the next charging period. With this scheme, the chillers are always run to their best advantage, producing more tons of cooling for fewer kilowatt hours. Furthermore, because the day's cooling load is spread out over a longer charging period, the chilling capacity of the system can be reduced. In effect, a smaller chiller is run at full load for a longer period rather than a large chiller run at partial load for shorter periods. The addition of cool storage capacity to an existing HVAC system can expand the overall chilling capacity of the existing equipment. This promises a savings when existing buildings are expanded. All of these benefits combine to lower operation expenses for the consumer.

POTENTIAL MARKETS

To this point, air conditioning has been treated as a homogeneous electrical end use category. In the fact that air conditioning is a dominant force in the timing and shape of summer utility peaks, it is homogeneous. Air conditioning units found in a range of buildings will receive heavy use in response to a hot day. However, the implementation and physical nature of air conditioning covers a wide range of mechanical systems in many types of buildings. Air conditioning can vary from the sophisticated and complex systems found in high rise commercial structures to the inefficient and simple window units found in a bedroom window. The potential for cool storage, and effective load management, will vary with building type. Each building category will have its own mechanical vocabulary and characteristic economic constraints. These economic constraints reflect the goals of the building's owner and, to a greater degree, the regulation, taxation and capital policies imposed by government and lenders. At this time, taxes are structured in a manner that encourages reduction of first cost spending at the expense of higher operating costs. Capital investment is accredited for tax purposes over many years in a depreciation schedule, while operating expenses are 100% deductable for income tax purposes as they occur. In addition, provisions for accelerated depreciation discourage long term ownership of some building types, further discounting the importance of operating costs. Therefore, at this time, certain types of building construction will be more amenable to off-peak cooling systems than others.

There are several major building type categories including high rise commercial, institutional, apartment, low rise commercial, and single family residential. High rise commercial buildings offer an attractive market for off-peak cooling systems. These buildings are characterized by inoperable windows and large internal gains,features that result in year round cooling requirements. The mechanical system usually involves large central centrifugal chilled water plants with air handling equipment distributed throughout the building. While the first cost issues are important, operating costs also hold large significance with owners contemplating long term ownership. This type of building has the highest rent per square foot, allowing the incremental costs of cool storage to be passed on to renters during the early years. Air conditioning in high rise commercial structures represents a major portion of summer peak load. By contrast, low rise commercial buildings are often developed as speculative ventures involving very short term ownership. This type of construction is extremely sensitive to first cost issues, commonly requiring a one year payout on "nonessential" items. Mechanical systems usually consist of inexpensive, roof mounted, direct expansion units. Without change in existing tax structures, it would be difficult to implement cool storage in this type of structure. As raw fuel becomes more precious, perhaps the scenario for this type of cheap construction will change. The same capital limitations apply to apartment buildings, motels, and hotels. However, with these buildings, operating costs become more important. The trend for mechanical systems in apartment buildings has been toward individual systems for each apartment unit. This eliminates the need for central ductwork and provides maintenance flexibility. It also provides flexibility for the later installation of cool storage systems. While this is also a speculative market, increasing fuel costs may provide incentives for such a retrofit.

In contrast to the low rise commercial category, institutional buildings are normally designed for long term ownership. The facilities of churches, universities, government and other non-profit institutions compose a large category of existing and ongoing construction. Long term ownership produces a commitment to higher construction quality and lower operating costs. Like high rise commercial buildings, the institutional facility will be the product of a professional design team, including architects and engineers. This is often not the case for houses, apartments and low rise commercial. While institutional clients are concerned with first cost issues, their desire for rational operating costs could lead to acceptance of off-peak cooling. Institutional buildings cover the whole spectrum of construction from garage to high rise. The final major building type category, single family residential, offers a wide range of values and priorities. As demonstrated by recent solar developments, some homeowners are very involved in the way their buildings use energy. Other homeowners seem hardly aware of the issue. A survey conducted by the National Center for Energy Management and Power concluded that residential consumers would have significant skepticism about purchasing cool storage systems.[7] The same study concluded that this skepticism would be readily dissipated by demonstrating the concept in public spaces, for instance, high rise and institutional buildings. This is the same process that led to the original acceptance of air conditioning proper. In conclusion, high rise commercial and institutional buildings offer the greatest possibilities for off-peak cooling systems today. They are relatively expensive building types minimizing the relative effect of incremental cool storage costs. The owners will benefit from reduced operating costs. These building types are significant because they are considered a key to acceptance in other markets. In addition, they have mechanical systems that are amenable to cool storage modifications.

COOL STORAGE STRATEGIES

The last decade has seen the introduction of several prototypical cool storage systems. Before reviewing these in detail, it might be useful to establish a set of generalized criteria for system performance. A primary qualification for a successful system is low initial cost. As cost is lowered, payback time is shortened and potential market acceptance increased. Simplicity can be a factor in achieving low initial cost; complicated systems certainly tend to be more expensive. A second major requirement is reliability. The storage system must work every day that the building requires chiller operation. Failure for only one day can result in increased peak demand on the utility or loss of thermal comfort conditions. This system must also be able to achieve a long life span with normal levels of maintenance. As part of the mechanical system performance should remain consistant for a period of decades. Third, a system must meet the safety standards normally demanded of other building components. A system should not impose penalties on the structural system, or in other words, it should be lightweight. A system should be applicable to retrofit jobs. An important consideration in retrofit and new construction applications is loss of space. A system should be reasonably compact. Finally, the storage components of a system should meet basic performance criteria without reducing the efficiency of other mechanical components. One important example is the prevention of condensation within the building volume. Above all, the primary goal of a thermal storage HVAC system, or for any HVAC system,is the maintenance of environmental conditions conducive to thermal comfort.

The implementation of cool storage systems can follow several general design strategies. The two major issues determining the nature of a system are the choice of storage medium and the method of its distribution. Thermal energy can be stored as sensible heat, latent heat, or a combination of the two. It is confusing but accurate, to describe a cooling system in terms of stored heat. The storage medium is used to remove heat from a building by storing it during the daytime. This stored heat is then removed by mechanically cooling the storage during off-peak hours. Sensible heat storage is achieved by raising the temperature of a mass. The energy stored is directly proportional to the storage temperature swing for a given mass. Sensible storage materials include liquids, commonly water, and solids, often masonry or concrete. Thermal storage can also involve the latent heat associated with a change of state in the storage material. For example, the conversion of water at 32°F to ice at 32°F liberates much more heat than lowering the temperature of 33°F water by 1°F. Phase change storage materials include water, various salt mixtures, and parafins. Equally as important as the choice of storage material, is the manner in which it is incorporated into the building. Storage material can be centralized in a single compact storage location or, by contrast, it can be distributed in a low density throughout the building. Decisions regarding the method and materials of storage determine the character and effectiveness of a given system.

Even though cool storage is a relatively new development, there have been several systems designed and implemented during the last decade. With systems installed in commercial and residential buildings, the experience gained warrants a quick review. A large number of these systems store heat sensibly in centralized water storage tanks. In Minnesota, a 32,000 SF multistory office building is cooled by energy stored in two 40,000 gallon, underground tanks.[8] This system shaves 100 kw from the building's peak load at an incremental construction cost of $115,000. The chilled water is stored at 45°F. A similar installation involving 300,000 gallons of storage is planned for a Toronto office building.[9] Studies based on this system indicate a cost of $270 per stored ton of cooling capacity in a large system and $392 per stored ton in a small system. This price, in 1975 dollars, includes interconnecting hardware. Dudley and Freedman designed a residential scale chilled water storage system with 2 1/3 ton capacity. This system stored 600 gallons of water at 35°F with system recharge accomplished with a compressor one half the size of a normal system.

Central water storage systems are subject to several problems. In order to chill water to the low temperatures required (35°F), the temperature at the evaporator must be lowered from the conventional setting in the 45°F to 55°F range. If the system is implemented in a simple configuration, chilled water is circulated to the space, where it picks up heat, and is then returned to the storage tank. The temperature of the tank will rise as the day progresses, leaving less cooling potential available for maximum afternoon loads. This problem, called blending, can be prevented by the installation of baffles or by using multiple tanks. This added complexity requires sophisticated controls resulting in higher cost and reduced dependability. With sensible storage using water, the high temperature is set for humidity control at 55°F and the low temperature is fixed at the freezing point. This limited Δt mandates large volumes of water with associated structural and container expenses. The design of a chilled water storage system becomes more complex as various subtleties emerge. Unfortunately, efforts to simplify system design can result in compromised performance.

Cool stroage can also be accomplished using water as a phase change material. Wisconsin Electric Company has built an experimental system that uses off-peak electricity to build ice on a 200' long evaporator.[11] This evaporator is located in a 180 gallon water tank. Space cooling is provided by circulating water from the ice tank to air handling units. While this type of system overcomes the disadvantages of blending, it does encounter several new problems. The ice build-up must not freeze the tank solid or it will interupt the water flow. In addition, this system will consuume unnecessary electricity because the evaporator must operate below 32°F to produce ice. The evaporator temperature is depressed even further, 10°F to 20°F, as ice build-up on the coils adds thermal resistance to the system. The study concluded that it is an unalterable fact that ice making will consume more energy than conventional systems per BTU delivered. A larger scale system is being tested at Oak Ridge National Laboratory.[12] This set up includes a 21,870 gallon basement tank that is charged throughout the winter. The large ice build-up is then used to meet summertime heat loads. This is an annual cycle energy system. This experiment required an estimated 24,300 linear feet of evaporator tube. In addition, the system incorporates 6 heat exchangers, several liquid pumps, and 4 three-way valves. Storage over the annual cycle proves to be very complex, and expensive. In summary, ice systems suffer a loss of efficiency due to depressed evaporator temperatures. This loss in the efficiency and the complicated equipment required are major drawbacks.

There are alternatives to water as a phase-change material. Latent heat can be stored in a variety of materials including a series of salt mixtures. The immediate advantage of using a salt mixture is the elevation of freezing temperature above the 32°F of water, allowing higher, more efficient evaporator temperatures. The University of Delaware has a cool storage experiment that uses a mixture of sodium sulphate, sodium chloride, ammonium chloride, and water. This salt is stored in 620 round plastic tubes, 1.25 inches in diameter, spaced 3" o.c. in a box 6' on each side. This is a 14% density of phase change material. The system, charged by a heat pump, is described as supplying a "large component of the cooling demand".[13] While we are left to speculate about the performance of the system, it was found that crystallization at the heat exchange surface caused a steady increase in discharge temperature as air is forced through the storage chamber. While the storage mixture is relatively inexpensive, it is subject to two problems that historically plague phase-change materials. When freezing, the mixtures tend to supercool, or chill beyond their freezing point without changing state. Also, after several hundred freeze-thaw cycles, the salt components tend to separate resulting in significant loss of heat content. This tendency is called pooling or stratification. The University of Delaware project undoubtedly included provisions to minimize supercooling and stratification; however, we do not know how effective these measures were.

A sodium sulphate based phase-change material has also been used in a scheme developed by General Electric.[14] GE has designed a system consisting of a horizontal cylinder filled with sodium sulphate decahydrate. This drum incorporates a nucleating device to prevent supercooling and, in addition, rotates to mechanically agitate the salts. This agitation prevents stratification. While this storage technique has been proven effective in laboratory scale experiments, the implementation of this system would prove awkward. The system does consume energy to rotate the drums at approximately 3 rpm. In addition, the moving drum defies the simple connection of heat exchange mechanisms. A successful system using a salt as a phase-change material must adequately address the following problems. The salt mixture must be properly packaged to prevent loss of water content. The design of the packaging system must recongnize a tendency toward reduction of surface heat transfer due to crystal build-up. The mixture must have proper nucleation to prevent supercooling. Seperation of constituents due to incongruent melting must be avoided, either through thickeners or mechanical agitation. If these problems are properly solved, the salt mixtures offer an inexpensive and effective heat storage material.

Several cool storage systems involve the use of solid materials for sensible heat storage. Solid storage, in the form of centralized rock beds, has been implemented in a range of scales. One interesting example of commercial scale rock storage is the system proposed for the SITE-1 office building in Sacremento, California.[15] The installation has two large rock beds, with each bed containing 14,000 cubic feet of 1" diameter rocks. These rocks are cooled at night by blowing chilled air through the bed. During the day, cooling loads are met by circulating room air through the storage. These beds will be used to reduce the chiller component of the summer peak load from 195 kw to 46 kw. In total, these beds will carry 75% of the building's annual cooling load. However, rock beds are subject to the same blending problems that plague chilled water storage. In addition, rock bed storage temperatures are usually too warm to provide humidity control.[16] This will limit the effectiveness of rock storage in humid areas. Because rock bed storage, like chilled water storage, is centralized, air handling equipment must be run throughout the day to deliver cooling from storage to the occupied spaces. The air handling equipment makes an undesirable contribution to daytime peak load. This contribution to peak load can be avoided by decentralizing the cool storage system.

One method of implementing a decentralized cool storage system is using the structural mass of a building for storage. This is the type of system designed for the Department of Justice Building in Sacremento, California.[17] At night, cool night air is circulated through the building. The cool air comes in close contact with the structural slabs, cooling the concrete structure from 4°-8°. This system is limited by the heat transfer rate between the air stream and the concrete surface, and by the surface area of structure available for storage. Another important issue is the thermal relationship between the cooled concrete mass and the occupied space that requires air conditioning. If these relationships can be balanced then several benefits accrue. The cost of the thermal storage medium can be totally, or at worst partially, assigned to the structural system. The cooled slabs will be in direct contact with the occupied spaces, thereby eliminating the need for the use of major air handling equipment during peak periods. There are no heat exchangers between the cool storage and the spaces served. Unfortunately, there are also some disadvantages. This type of system is not effective for humidity control. If, for instance, the space required heating in response to a change in the weather, the cooled slabs would hinder this changeover. Despite the disadvantages, decentralized storage offers some very attractive features.

Several basic working assumptions can be based on experience gained from these early cool storage experiments. The first assumption is the preference of phase-change storage over sensible heat storage. This assumption is based on the limited temperature swing available for sensible storage and the large mass therefore required. This conclusion was also reached by an Argonne National Laboratory study assessing energy storage technology and systems. They concluded "the operating temperature range available for storage of sensible heat precludes the use of compact, low cost storage devices. Given the large potential benefits in storage air conditioning applications, further research and development on phase-change materials appears to be justified."[18] In addition, an exhaustive cool storage assessment study by GE identified "phase change storage as the probable technology for cool storage with air conditioning."[19] Another basic assumption is that low density, decentralized storage is preferrable to a centralized storage system. This is based on peak load reductions in air handling equipment operation that would otherwise be required. It should be noted that decentralized sensible storage in structural materials has more potential than a centralized sensible storage scheme. Decentralized storage must still deal with the issues of humidity control and balanced energy flows. It is from this perspective that schematic design of the proposed cool storage system began.

REFERENCES

1. Stein, Richard G., Architecture and Energy (Garden City, NY: Anchor Press/Doubleday, 1977), p. 12.

2. Stein, p. 13.

3. _______________, The U.S. Energy Equation and the Role of Electricity (New York: IEEE, 1976), p. 8.

4. American Institute of Architects, Energy Conservation in Building Design (Washington, D.C.: AIA, 1975), p. 117.

5. Energy Research and Development Administration, A National Plan for Energy Research, Development, and Demonstration (Washington, D.C.: U.S. Government Printing Office, 1977), p. 14.

6. General Electric, Cool Storage Assessment Study (Springfield, VA: National Technical Information Service, 1977), p. 2-12.

7. Wittreich, Warren J., Product Acceptance of Off-Peak Air Conditioners (Springfield, VA: National Technical Information Service, 1972), p. 9.

8. "Office Uses Off-Peak Thermal Storage", Electrical World, November 1975, p. 71.

9. Lilleleht, L.U. ED., Proceedings of Workshop on Solar Energy Storage Subsystems (Charlottesville, VA: ASHRAE, 1975), p. 58.

10. General Electric, p. 2.2.

11. General Electric, p. 2.3.

12. General Electric, p. 2.5.

13. Argonne National Laboratory, Assessment of Energy Storage Technologies and Systems (Illinois: Argonne National Laboratory, 1976), p. 46.

14. General Electric, p. 2.7

15. Corson, Bruce A., "An Energy Efficient Office Building for the State of California", <u>Passive Solar-State of the Art</u> (Newark, Delaware: ISES, 1978), p. 235.

16. Lillelheht, p. 42.

17. Telephone Conversation with Dubin-Bloome, Inc., May 1979.

18. Argonne National Laboratory, p. 11.

19. General Electric, p. 2.7.

4.

A Proposed Off-Peak Cooling System

This section will describe a new design for an off-peak air conditioning system. Any air conditioning system, whether conventional or off-peak is designed to provide ambient conditions conducive to human comfort in the space it serves. The majority of these comfort criteria relate to thermal comfort. Indoor air dry bulb temperature must be held within a range considered appropriate for the season. Relative humidity must be controlled, primarily for human comfort, and also to prevent condensation on cold surfaces. Air conditioning systems are used to control the temperatures of surrounding surfaces, providing, with the help of insulation, a pleasant mean radiant temperature. In addition, indoor air velocities must be maintained within a comfortable zone between stagnant and drafty conditions. Other issues involving human comfort include the recirculation and filtering of air to minimize odor and dust problems. Indoor air from areas with characteristic odor problems must be exhausted entirely and replaced by tempered, filtered outdoor air. In many buildings, the mechanical system must also supply air for biological needs. In recent history, the demands placed on air conditioning systems have expanded, requiring larger and more exotic solutions, with the common denominator often being increased fuel use. This is a trend that must be reversed. However, the basic goal of providing comfort for building occupants must remain as the prime determinant in system design.

Before describing the specifics of the new proposed system, it is important to remember than an air conditioning system never really stands alone. It is always a part of, and designed for, the building it serves. In this context, it is largely the design of the building that determines the nature and magnitude of the various demands an air conditioning system must face. Richard Stein conducted a survey that found a ratio of 6 to 1 in energy requirements among a group of office buildings otherwise similar in age, location, purpose, etc.[1] A prerequisite for the realization of meaningful mechanical system efficiency is the sound design of the building it serves. The commercial structure category has been identified earlier as the first market likely to accept the concept of off-peak cooling. The off-peak cooling system proposed in this thesis is designed primarily for commercial office structures, although it is certainly not limited to this application. It is expected that applications of this system,whether new construction or retrofit, would occur in structures that also include other energy conservation features. Thus, the system's design, and performance, will be gauged in the context of an energy conscious, contemporary structure. This implies moderate cooling loads. Building heat gains can be separated into sensible and latent categories. Sensible heat gains are experienced through an increase in dry bulb temperature. Sensible heat gain sources typically include solar radiation, lighting, occupants, ventilation, etc. Latent heat involves the addition of moisture to the interior air raising the wet bulb temperature. Latent heat can come from people, appliances, and humid outdoor air. The off-peak cooling system must remove these loads while minimizing the use of peak period electricity.

SYSTEM CONFIGURATION

This thesis proposes an off-peak cooling system based on a decentralized geometry. Primary thermal storage for the system is provided by phase-change material placed in the ceiling plane of the conditioned spaces. This phase-change material, a modified sodium sulphate mixture, is contained in heat sealed bags. The bags are supported by a special ceiling tile designed to provide good thermal contact between the salt mixture and the conditioned space. The salt bags are also exposed to the interstitial space between the building structure and the suspended ceiling tiles. This interstitial space serves as a plenum for circulating chilled air. At night, off-peak electricity powers mechanical equipment to chill air and circulate it through the plenum. This chilled air circulation freezes the salt mixture at the ceiling plane and cools the building's exposed structural mass as well. During the day, as the building experiences heat gain, the salts slowly melt, and aided by the cooled structural mass, they store the sensible component of the day's heat gain. The sensible heat gain is thus removed without the need for peak period operation of chillers or air handling units. This storage system cannot, however, remove latent heat gains. There is also a need for a minimal amount of ventilation air required by building code. Thus, a small mechanical system must be operated during the day to remove these loads. However, the majority of the day's heat load, the sensible component, is stored during the day and later removed using nighttime off-peak electricity.

PHASE-CHANGE MATERIAL

The salt used in the proposed storage system, sodium sulphate decahydrate ($Na_2SO_4 \cdot 10H_2O$) has historically been subject to the shortcomings normally associated with phase-change materials. In review, these shortcomings included two major problems, stratification and supercooling. These problems were successfully overcome in a similar salt mixture used in MIT's Solar Building 5.[2] This project, directed by Timothy E. Johnson, used a thermal storage system consisting of modified sodium sulphate decahydrate encased in a polymer concrete

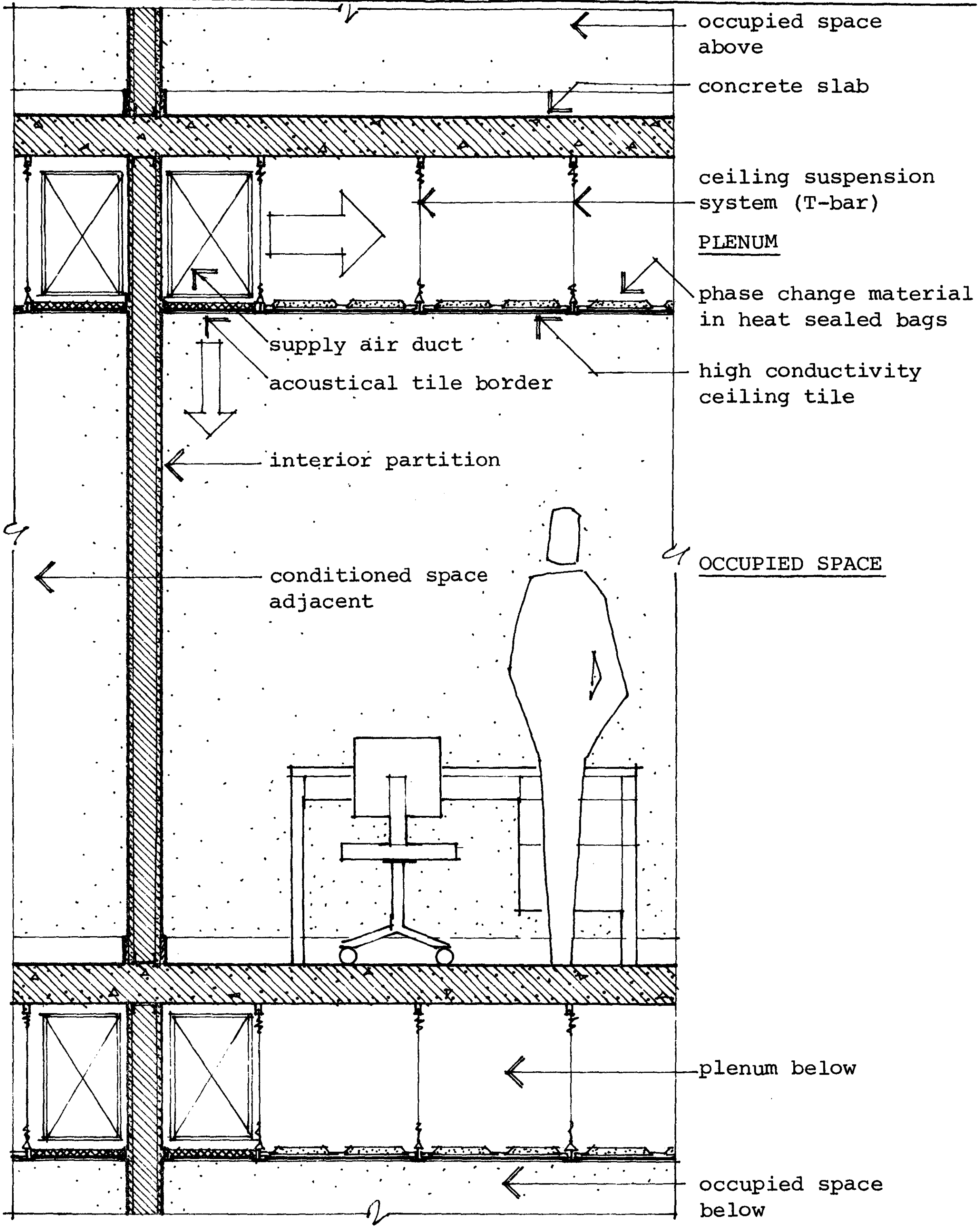


FIGURE 10 Vertical Section Through Interior Zone Space Showing System Components.

ceiling tile. The modifications to the basic sodium sulphate decahydrate mixture were required for several reasons. First, it was necessary to depress the salt's melting point to the desired 73°F . In addition, modifications were necessary to prevent the occurance of stratification and supercooling. In an unmodified state, sodium sulphate decahydrate has a melting point of approximately 88°F. This melting point can be lowered by the addition of a eutectic agent, in the case of Solar Building 5, sodium chloride. The problem of supercooling was minimized by the addition of a nucleating agent, Borax. The prevention of stratification, or pooling, due to incongruent melting required a two prong solution. First, a thixotropic thickener, fumed silica, was added to the solution to retard component migration during the melt phase. Second, the salt mixture was packaged in thin (3/8") horizontal layers to minimize the gravitational pressure heat while still allowing crystal growth by diffusion. This combination of measures has prevented appreciable stratification in the Solar Building 5 mixture after more than 4500 freeze-thaw cycles. The Solar Building 5 project represents a major advance in practical latent heat storage using a sodium sulphate decahydrate based material. These same features can be incorporated into a salt mixture designed for cooling applications.

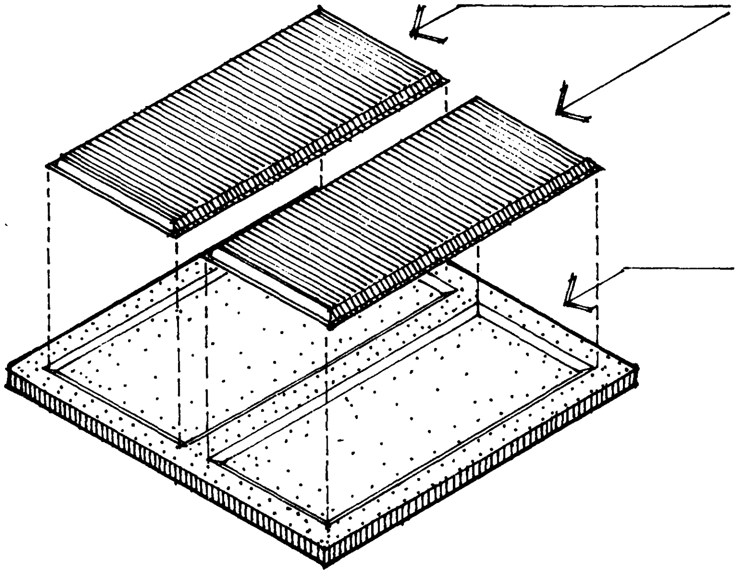
Experience gained in the work on Solar Building 5, can be directly applied to the design of the cool storage system. This new system follows the same geometry used in Solar Building 5, with phase change material distributed in thin horizontal layers at the ceiling plane. By following this precedented geometry, the cool storage system should be able to use the same solutions to supercooling and stratification problems. The proposed system will differ from Solar Building 5's storage scheme in three minor points. First, the salt packaging system must be revised to meet new economic and performance criteria. Second, a thermal storage system for cooling applications will require a set point temperature lower than the 73°F previously used. Finally, the cool storage system will be charged in a different manner, requiring a careful study of the system's heat transfer properties. These changes are all based on a solid foundation established by the system at Solar Building 5.

The phase material used in Solar Building 5 is first packaged in thin, heat sealed envelopes, then cast as the core of polymer concrete tiles. The proposed cooling system will use a "stand alone" version of the heat sealed bag in order to reduce overall system cost. This bag will be placed exposed, on top of a new type of ceiling tile. The existing heat sealed bag is composed of 1 mil aluminum foil laminated on both sides with 2 mil polyethylene. The foil is necessary to prevent water permeation over a period of years. The polyethylene protects the aluminum from corrosion, provides additional strength, and allows the material to be heat sealed. The salt bags are nominally 1' wide by 2' long by 3/4" thick. Each bag is separated into horizontal compartments by an internal 2 mil polyethylene sheet providing two compartments, each 3/8" thick. Without additional protection, the bag is susceptable to puncture by the sharp crystals formed during the salt's solid phase. In order for this bag to stand alone, its walls must be thickened. The most important membrane to protect will be the foil layer. This may remain thin if protected by a stronger interior of fibrous polyethylene laminate. Otherwise, the basic construction and geometry of the bag can remain unchanged.

It is intuitively clear that the melting point of a latent heat storage material used in a cooling system must fall below the desired air temperature off the room. The room air temperature in Solar Building 5, a heating application, remains approximately 10°F below the temperature of the storage tiles. An initial working assumption is that this 10°F Δt would also apply when the storage is cooler than the adjacent air. Another preliminary assumption is that the mean interior dry bulb temperature for summer conditions should be 74°F. This would establish the desired phase change temperature at 64°F. Several experiments were conducted to test the effectiveness of various compounds in lowering the melting point of sodium sulphate decahydrate from the normal 88°F to the desired 64°F. The results of these experiments are summarized in Appendix A. The most promising candidates were sodium chloride, ammonium chloride, and lithium carbonate. The experiments were run to establish the quantity of each eutectoid required to lower the set point and to find the latent heat content of the resulting mixtures. Subsequent research by the Cab-O-Sil Division of the Cabot Corporation has identified ammonium chloride as the most promising candidate. Experimental results indicate that the set point temperatures can readily be lowered to the desired region with little or no loss in latent or sensible heat content relative to the mixture used in Solar Building 5. This heat content is lower than the theoretical capacity of sodium sulphate decahydrate, but is consistent with practical applications of the material in the past. In addition, preliminary experiments indicate that the change in eutectic agent will not compromise the measures taken to minimize pooling and stratification. There is no anticipated obstacle to the economical production of a lower set point mixture. The Cabot Corporation, producers of the Solar Building 5 mixture, is actively involved in the formulation and testing of new salt mixtures, and packaging, appropriate for cool storage.

CEILING TILES

As described earlier, the packaged phase change material will be supported by ceiling tiles distributed throughout the conditioned space. This ceiling system will be similar to the standard suspended ceiling systems common to commercial spaces, with the exception of the ceiling tile itself. The tile must be able to support the 4 to 6 pounds per square foot load of the salt storage with less than 1/240 deflection. The tile itself should be as light as possible in order to reduce the structural loads imposed on the suspension system . In addition, the tile material must impose very little thermal resistance between the cool storage it supports and the occupied space below it. This requirement would indicate a material with high thermal conductivity implemented in as thin a section as possible. Thermal conductance should exceed 16 BTU/Hr°FSF. This figure will result in a tile resistance that is approximately 1/10 the resistance of the air film between the tile and the room. Such a relationship insures that the surface temperature of the tile will remain within 1°F of the PCM temperature when the assembly is subjected to the anticipated 10°F Δt. The tile must also have a very close fit to the phase change material envelope because the presence of air gaps between tile and bag would increase thermal resistance. This tile will be suspended in a standard t-bar ceiling suspension system with a maximum span of 2'0". The tile edges must exhibit a close fit to the suspension system in order to minimize the infiltration of air from the plenum to the occupied space while the salts are being charges. This tight fit is also necessary to reduce the migration of latent heat gains into the plenum. One final criteria related to thermal performance is the specification of high absorptance for long wave radiation (above 10 microns). This requirement is met by most building materials.



PHASE CHANGE MATERIAL
HEAT SEALED IN FOIL AND
POLYETHENE BAGS.
1' x 2' x 3/4" NOMINAL.

CEILING TILE PROVIDES
TRAY-LIKE SUPPORT WITH
MAXIMUM HEAT CONDUCTION.
2' SQUARE NOMINAL FOR
STANDARD T-BAR SUSPENSION
SYSTEM

FIGURE 11. Sketch of PCM Bags and Support Tile.

In addition to the characteristics required for successful thermal and structural performance, the ceiling tile must also fulfill the role of a finish ceiling material. This will entail features required for finish, acoustical, lighting, and economic considerations. As an architectural finish the tile must be compatible with other common finish materials, specifically acoustic ceiling tile, because the two will likely be intermixed. The primary thermal considerations will probably result in a dense, hard finish surface that will be excellent for thermal conductance, but perform poorly as an acoustical absorber. The inclusion of acoustical treatment in the tile is highly desirable, but only if it does not interfer with thermal performance. Design implementation of the cool storage system must include explicit treatment of acoustical issues, whether in the tile design itself or in additional, external measures. This ceiling system will perform at its maximum potential with the implementation of an indirect lighting scheme. Although ceiling mounted fixtures are possible, being placed in the grid and buffered by acoustical tile, the conflict between lights as a point heat source and adjacent cool storage would be minimized by using indirect lighting. Therefore, the new tile material, like standard ceilings today, should exhibit a high reflectance coefficient for visible light. Finally, the proposed tile should be economical, eliminating the use of exotic and expensive materials. This is quite a list of demands for an inexpensive, mass produced ceiling product. Owens Corning-Fiberglass, a leading producer of finish ceiling materials is currently involved in the design of this new tile material. Happily, several promising materials have already been identified.

MECHANICAL SYSTEM

The mechanical system used to charge the cool storage material will be conventional in almost every aspect, using standard mechanical components. Chilled air is circulated through the plenum at night to freeze the salt storage and to cool exposed structural mass. The source of this chilled air could be an air handling unit connected to a central water chiller, a direct expansion refrigeration unit, or even outside air if environmental conditions permit. The frozen salts, at a melting temperature of approximately 64°F, are able to store the day's sensible cooling load with the aid of the cooled structural mass. However, the 64°F set point temperature is too high to remove moisture, or latent heat gains from the air. This load, the latent heat gain from outside air, occupants, and appliances, will require the daytime operation of a much smaller air conditioning system. If the air handling units for this system use two speed fans, the same units can charge the plenum at night and carry latent loads during the day. The air circulation through the plenum at night is a closed loop, with no outside ventilation required. During the day, a minimal amount of outside air is required for ventilation with this intake balanced by exhaust from odor prone areas. The air handling equipment must handle a much larger quantity of air during the nighttime charging period. The daytime air conditioning equipment must be designed to handle only the latent component of the building load.

We have grown accustomed over the last 30 years, to a fine degree of control over our interior environments. Conventional mechanical systems are capable of maintaining interior conditions within 2°F of a specified thermostat setting. The success of the proposed off-peak cooling system will be determined not only by its economic advantages, but also by the effectiveness it demonstrates in maintaining thermal comfort conditions. without the amenity of thermostatic feedback. There are several issues that affect the ability of the system to accomplish this goal. The isothermal nature of latent heat storage is a great advantage toward this end. The salt mixture melts at a rate that responds to the thermal load on the space. In a situation with very little heat gain, the

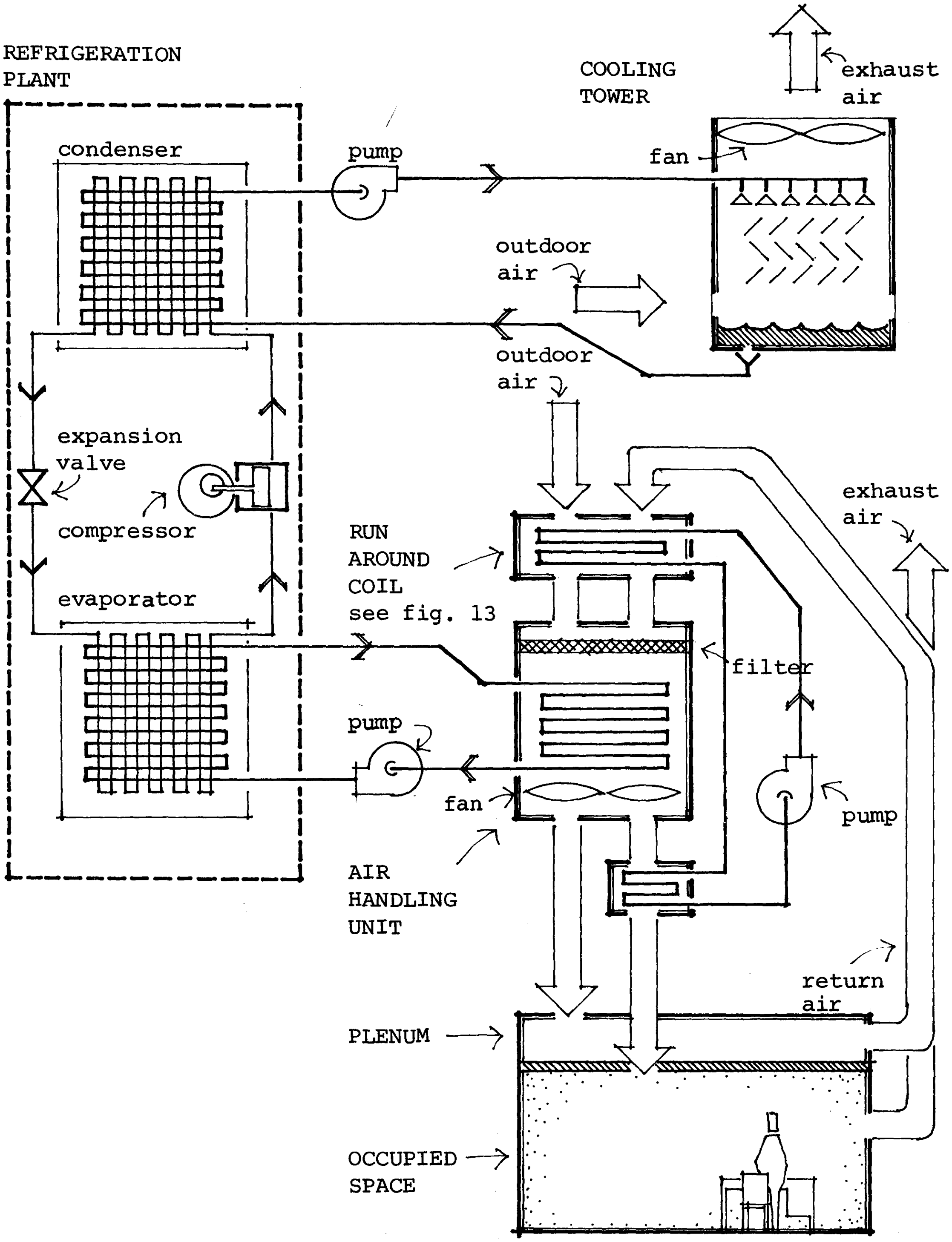


FIGURE 12 Mechanical System Schematic

the salts melt slowly. As heat gain increases, so does the melting rate, preventing a rapid increase in interior dry bulb temperature. Of course, to realize this form of self regulation the storage system must be properly sized to handle the sensible heat loads encountered in the space. Likewise the daytime mechanical system must be adequate to meet the latent heat loads. Finally, beyond the issue of storage capacity, the system must be designed to allow adequate energy flow to meet the loads encountered. The heat transfer between the charging air and the thermal storage material must be adequate to handle a maximum day's load within the allotted charging period. Thermal transfer between the charged storage and the occupied space must also be adequate to handle cooling loads at the rate in which they occur. These issues can be investigated by simulating the performance of the system under various operating conditions. Feedback from these simulations can be used to further refine the design and juxtaposition of system components.

REFERENCES

1. Stein, Richard G., Architecture and Energy (Garden City, NY: Anchor Press/Doubleday, 1977), p. 11.

2. Johnson, Timothy E., et al., MIT Solar Building 5--Initial Performance (Cambridge,MA: 1979), p. 18.

5.

Simulation of System Performance

Simulation of the proposed off-peak cooling system's thermal performance is accomplished through the use of mathematical models. These models evaluate thermal traffic through the system caused by the incidence of cooling loads and the operation of cooling equipment. To do this, the thermal relationships of the various system components are summarized in a thermal network. This network is then cycled through hourly inputs representing the operational patterns of the space, and situation, being modelled. These inputs include the introduction of cooling loads from occupants and weather as well as the introduction of cool air from mechanical equipment. The model provides an hourly accounting of important parameters including air temperatures, surface temperatures, and thermal transfer. These outputs can be evaluated for system performance in thermal comfort, energy consumption, the adequacy of system components, etc. This simulation process requires a sharp definition of the thermal characteristics of the proposed system. In addition, the environmental context faced by the system must be defined in terms of the magnitude and timing of cooling loads.

Once the system and the cooling loads are amply described, the simulation process provides valuable feedback on performance issues.

In developing the initial system configurations, several limiting assumptions have been made. First, as previously stated, the buildings served by the system are assumed to have implemented a range of energy conservation measures. In effect, this limits the loads encountered by the system to a moderate level, avoiding the extreme loads of wasteful buildings. Second, the system is considered applied to a building's interior zone only. This is not to suggest that cool storage is not appropriate for exterior zones; however, this zone assumption simplifies heat gain patterns for preliminary analysis. There are interesting challenges associated with cool storage for exterior zones. For example, the exterior zone of a building is much more sensitive to weather loads, tending to cycle from heating load to cooling load. The presence of charged cool storage in a space experiencing heating requirements must be avoided. Another assumption is that hypothetical building loads are for a Boston, Massachusetts location. Background and building load data is available for a 750,000 SF building located in the Boston area. This will allow comparison of hypothetical building load data to empirical figures from an existing structure. Finally, it is assumed that the building heat gains in the interior zone are evenly distributed. This is normally the case, being particularly true when the office layout incorporates open planning and indirect lighting. These assumptions are used in the initial system performance simulations. Variations and their effect on performance will be discussed later.

COOLING LOADS AND HEAT GAIN

Commercial buildings are subject to heat gain from a number of different sources. Each component of the cooling load has its own character, determined by its timing, distribution, magnitude, and latent heat content. There is a distinction between a building's heat gain and its cooling load. Heat gain, as the name implies, is the amount of heat introduced into the building at any given time. This impact, coming from within and without the building, raises the indoor air temperature and the temperature of the building's mass. The energy stored in the building mass, however slight, does not contribute to load at that specific time. Cooling load, as reflected by chiller operation in a conventional system, is the amount of cooling necessary to keep interior air conditions in the comfort range. Of course, the heat gain stored in the building mass ultimately contributes to the building's load; however, its impact is shifted to a later time. In a conventional building, storage capacity is usually exhausted very quickly though it often dampens morning cooling loads. The use of specific cool storage, in effect, exaggerates this dampening by displacing cooling loads until off-peak periods. A cool storage system must be designed to effectively handle the types of heat gain incident on the system.

Heat gain sources can be categorized by their relationship with either the exterior weather conditions or the activities associated with building occupancy. The impact of weather related heat gain falls largely on the exterior zones of a building. These zones are usually defined as spaces within 15 feet of an exterior wall. Building spaces below roof areas are also subject to weather related loads. These loads include the impact of solar radiation on building surfaces and through glazed areas. The exterior walls and roof are the source of conductive heat gains. The exterior wall and its associated openings represent a source of heat gain via infiltrations. However, infiltration gains in commercial structures are usually minimal, or non existant, due to the practice of maintaining a positive building pressure. A final major weather related load is the introduction of outside ventilation air into the building's exterior and interior zones. This ventilation air, replacing exhausted air, must be tempered and represents a combination of latent and sensible heat gain. These weather related loads will, of course, vary with season, in fact, shifting between heating and cooling loads. The magnitude of these loads during any given season will vary with daily climatic conditions, notable solar radiation and exterior ambient temperature. Solar radiation will vary with cloud cover and its directional characteristics relative to various building surfaces. This, in effect, establishes several daily load cycles depending on the zone of the building in question. Ambient temperature, related to solar radiation, will effect conductive gains, infiltration loads and the heat gain from ventilation requirements. This influence will vary over the diurnal cycle with maximum impact in the afternoon. Weather related loads, with the exception of ventilation, are unrelated to the operational status of the building.

There is a group of cooling loads that occur in patterns related to building occupancy. These loads are imposed by the services provided for the occupants and by the occupants themselves. A major category of occupancy related load is artificial lighting. Lighting equipment generates a large amount of heat in addition to light. Indeed, following the law of thermodynamics establishing conservation of energy, all energy devoted to lighting in interior zones eventually contributes to the cooling load. This lighting load is particularly acute due to the existence of large interior zones devoid of natural light.

In addition, the recent historical tendancy has been toward higher and higher lighting levels, a trend made possible by air conditioning. Other electrical uses contribute to heat gain in a similar manner. These include office machinery,elevators, fans, pumps, etc. Computers also contribute to heat gain; however, their specialized environmental requirements mandate separate mechanical systems. Other areas such as kitchens, washrooms, and mechanical rooms also receive specialized mechanical treatment. A final category of occupancy relted gain is the heat from people. Unlike heat from electrical use which is sensible heat only, people contribute heat through both sensible and latent mechanisms. Occupancy related heat gains affect both interior and exterior building zones. They comprise the majority of interior zone heat gain, with the only weather related gain for this zone being ventilation air. This results in a very steady pattern of heat gain that follows the cycle of occupancy in interior zones.

HEAT GAINS FOR SIMULATION

The first step in simulating the performance of the system is to establish the context of the installation and the loads that it must face. The first simulations are based on a suburban office building located in the metropolitan Boston area. This building has 757,000 SF of conditioned space on six floors. The building's occupancy represents a typical range of office tenants, from insurance agencies to law offices. The building is arranged about a large central court. Exterior walls have a moderate ratio of glazed area to solid area, with fenestration minimal on the southern exposure. Building management has actively pursued programs aimed at reducing energy waste, with successful results. This building was selected for this study because it typifies the type of project well suited to off-peak cooling.

Data is available regarding mechanical equipment operation, electrical power consumption, and local weather patterns. The building remains unnamed at the request of the owner. Unlike applications of thermal storage for heating applications, which are sized to meet average conditions, an off-peak cooling system must be designed to meet worst case conditions. This is not to say that interior conditions during the hottest day of the century must remain well within the comfort zone. Off-peak cooling systems, like conventional HVAC systems, will be designed to provide comfort during all but 2 1/2% of the cooling season. Under extraordinary heat loads the system will be unable to prevent minor overheating. This situation must not prompt the use of the night-time charging chillers to meet daytime overload. This would increase the peak load of the utility system and eliminate much of the advantage and financial incentive of the off-peak cooling scheme.

A cooling load profile for the first simulation was derived from data taken at the test building during the week that it experienced its 1978 peak electrical demand. This year's peak demand, as recorded by Massachusetts Electric Company's magnetic recording meter, was 4800 kw occurring at 11:45 AM on Thursday 17 August 1978. Cooling loads for this week were determined on an hourly basis by examining the building's chiller log which recorded pressure and temperature drops across the chiller. The 1560 ton York chiller was designed to operate at 3120 gpm over a 19.56 pound pressure drop. The flow rate for a given pressure drop is found with the following expression.

$$\text{flow rate (gpm)} = 3120 \text{ gpm } (\Delta \text{ pressure}/19.56 \text{ pounds})^{1/2}$$

This flow rate can then be used with the temperature drop to find cooling output in the following expression.

$$\text{Cooling output (in BTU/hr)} = \text{flow rate (gpm)} \times 60 \text{ (min/hr)} \times \Delta t(^\circ\text{F}) \times .1337 \text{ (gal/cu.ft)} \times 62.4 \text{ (BTU/cu.ft}^\circ\text{F)}$$

This hourly cooling output was devalued by 10% for transmission losses between the chiller and occupied spaces. The net result was the total cooling load of the building. The gross cooling load was divided by the building's square footage to yield a cooling load in BTU/hrSF. Of course this load was not actually distributed evenly throughout the building; however, this assumption was made for the early simulation calculations. This assumption has a conservative effect for it averages the weather related building loads throughout the building. The maximum cooling load calculated by this method is 24.7 BTU/hrSF.

The peak load of the building can also be calculated by estimating the contribution of the various heat gain components. The groundwork for this procedure was established in an energy audit conducted for the building by a Boston firm specializing in such studies. This audit established various heat gain parameters for the building. For instance, the building's total conductance gain is given as 169,000 BTU/hr°F. This conductance times a 14°F temperature difference during peak load conditions yields a load of 3.12 BTU/hrSF. Building occupancy for office structures is estimated at 1 person per 100 SF. Each person contributes approximately 250 BTU/hr sensible gain and 200 BTU/hr latent gain yielding a total occupant gain of 3.2 BTU/hrSF. Heat gain from ventilation air is assumed to be the load imposed by cooling air from exterior design conditions of 85°F, 60% RH to interior conditions of 75°F, 50% RH. Typical office ventilation rates are in the range of .2 cfm of outdoor air per square foot yielding a building load of 8.5 BTU/hrSF which must all be assigned to the daytime chiller. The electrical load imposed on the building for artificial lights, equipment, and air handling equipment amounted to 1832 kw or 8.25 BTU/hrSF. Finally, the energy audit estimated building solar gains to be approximately 2 BTU/hrSF. This low figure can be attributed to the lack of south facing glazing as well as the moderate glazing to solid ratio. These heat gain figures are summarized in Table No. 1.

TABLE NO. 1 COMPUTED HEAT GAIN SUMMARY

Heat Gain Component	Contribution in BTU/hrSF Under Design Conditions (i.e., maximum load)
Conductance	3.12
Occupants (sensible)	1.8
Occupants (latent)	1.4
Ventilation	8.5
Electrical	8.25
Solar	2.0
Total	25.17

This total of 25.17 BTU/hrSF peak load is in close agreement with the 24.7 BTU/hrSF peak load calculated using the building's chiller log. Of this 25 BTU/hrSF the contributions of occupant latent loads and outdoor ventilation air must be assigned to the daytime chiller. This is a total of 9.9 BTU/hrSF.

One goal in the design of an off-peak cooling system is to minimize the loads carried by the daytime chiller. The major portion of daytime chiller operation is required to provide ventilation air to the occupied spaces. Regulatory agencies are currently reviewing code requirements for ventilation rates. The National Bureau of Standards has suggested an office ventilation rate of 5 cfm/person.[1] This value has also been suggested by Fred Dubin in his Energy Conservation Manual. At this time, however, the current building regulations must be followed. Ventilation requirements for office space in Massachusetts are specified under the Massachusetts State Building Code to follow ASHRAE Standard 63-77. This standard requires a minimum of 10 cfm per office worker based on one worker per 100 SF. This value of 0.1 cfm/SF is one half the load assumed in the earlier heat gain calculations. Adoption of this ventilation rate would lower the load on the daytime chiller from 9.9 BTU/hrSF to 4.7 BTU/hrSF. It might also be assumed that the 2 BTU/hrSF solar load would be carried by the daytime chiller. These loads might occur at a rate that is too localized and too fast for the tiles to absorb. This brings the daytime chiller load to 6.7 BTU/hrSF or approximately a 400 ton chiller. This is one quarter the current chiller requirement.

The ventilation load on the daytime chiller is described as the amount of energy required to lower outside air at design conditions to indoor air conditions. This is a challenging task because latent heat loads are involved in addition to the sensible heat component. The only practical method of removing latent heat, or moisture, from the air is to chill the air to the dewpoint of the interior design conditions. In conventional HVAC systems, outdoor air is mixed with return air and then cooled to about 55°F and 100% RH. This cooling removes moisture from the air because the 50°F fin tube surfaces are colder than the 68°F saturation temperature of the outdoor air. This process produces more sensible cooling than is required to temper the outdoor air, the extra component being the energy required to lower the drybulb temperature from 75°F to 55°F. This additional sensible cooling is normally supplied to the space, in the form of 55°F air, to remove sensible heat gains. In some cases, for example a terminal reheat system, the 55°F air is actually reheated before it is supplied to the occupied space. However, this additional sensible cooling is not desirable in the context of the proposed off-peak cooling system, because it increases the load carried by the daytime chiller. This problem is circumvented by reclaiming the additional sensible cooling after the outdoor air is cooled to the proper wet bulb temperature. This is accomplished by using a run around coil heat exchanger. This heat exchanger has fin tube coils that chill incoming outdoor ventilation air. These coils are connected by piping through which a heat exchange fluid, ethylene glycol and water, is circulated by a pump. Sensible heat is reclaimed by the down stream coil and used to precool incoming air at the upstream coil. The net effect is a reduction in the load carried by the daytime chiller with the displaced sensible load carried by the storage system.

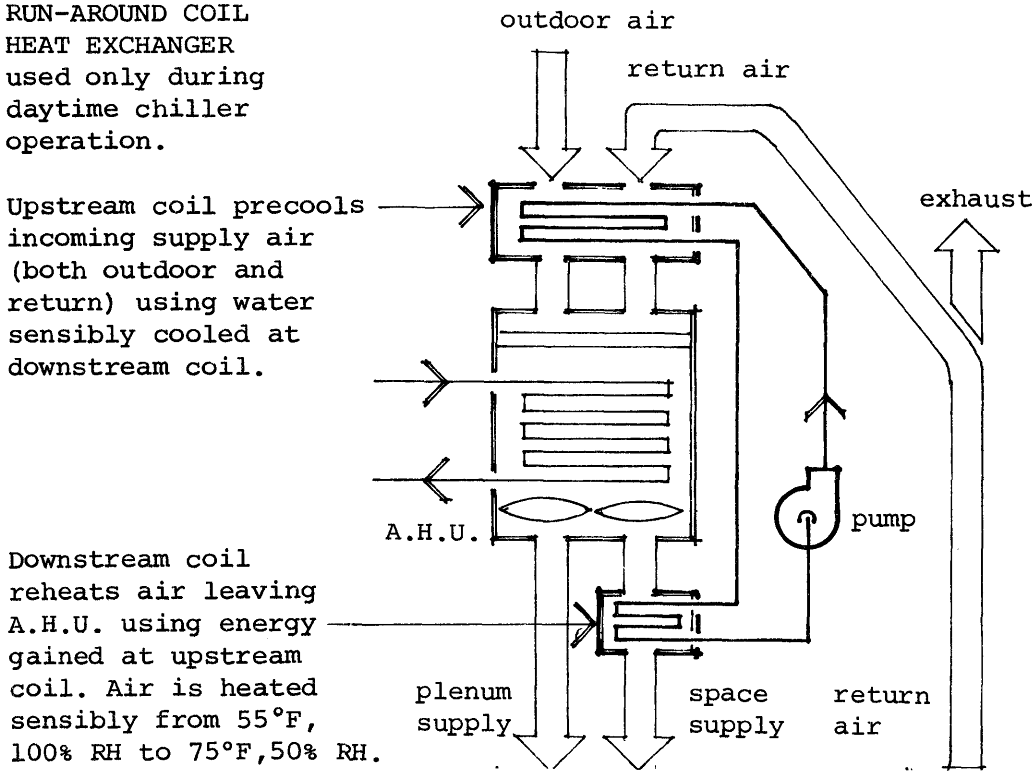


FIGURE 13. Run-Around Coil Heat Exchanger Schematic

The cooling loads used in the initial simulation were synthesized from the loads experienced by the test building during the week of its 1978 peak load. These loads, calculated on an hourly basis from chiller log data, are presented in Figure number 14. This data profiles the building load, on a per square foot basis, for three days culminating in the annual peak load. This graph also includes a shaded area that represents the contribution of the daytime chiller. This 400 ton chiller providing 6.3 BTU/hr SF, is run for 10 hours per day during the occupied period. This 10 hour period, from 9 AM to 7 PM, corresponds to the timing of the heat gains the daytime chiller must carry; latent heat from people and ventilation air. The remainder of the building's cooling load is carried by the cool storage system. The thermal storage material recharged period will depend on occupancy patterns and the pricing periods specified by the utilities in future time of day rates. The data from the test building was combined with cooling load calculations to produce a cooling load profile representing one typical design load

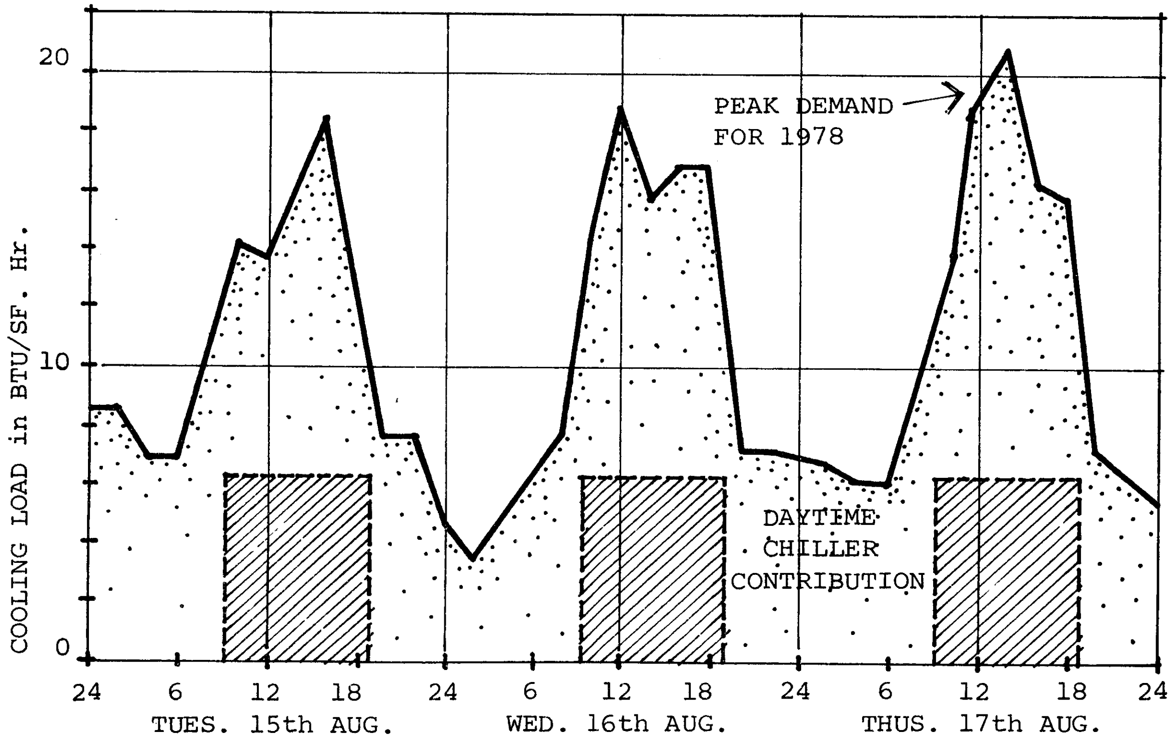
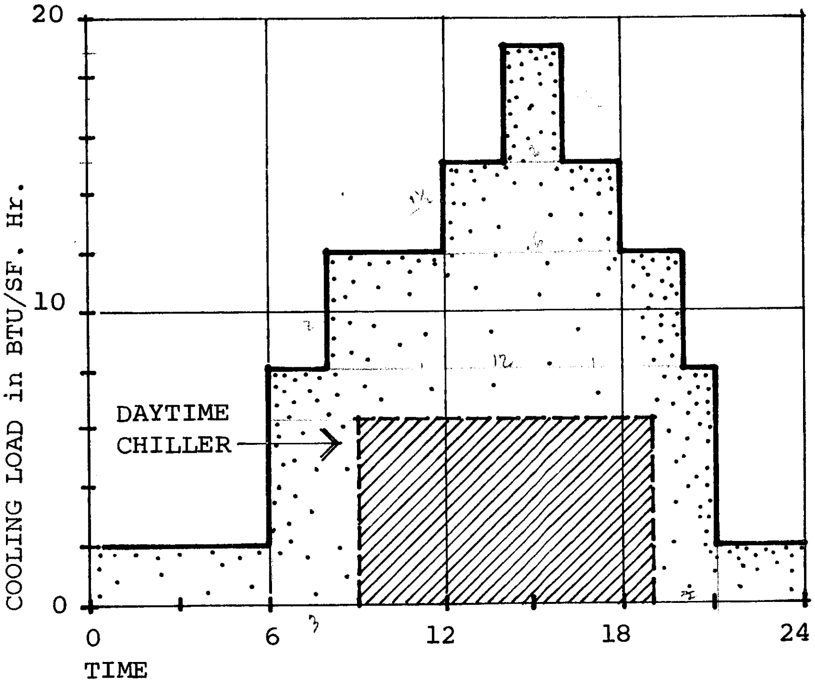


FIGURE 14. Actual Cooling Loads from Monitored Building, August 1978.

day as shown in Figure number 15. This model represents the design load experienced in an interior zone space. The solar impact component is not included in the profile because this load is peculiar to exterior zones. This profile also differs in the magnitude of cooling load experienced during the largely unoccupied off-peak hours between 9 PM and 6 AM. The building's highest actual off-peak loads are due to an anomaly in the building's occupancy, a 24 hour data processing operation. The value of 2 BTU/hrSF is considered more appropriate. The load profile given in Figure number 15 is the pattern used for the initial simulations.



NOTE:
This cooling load profile represents a synthesis of data from peak load periods at a monitored building and calculated heat gains for a typical interior zone office space.

FIGURE 15. Synthesized Load Profile used in Initial Simulations.

THERMAL NETWORK

Simulations for this study were run on a seven node thermal network similar to the TEANET program written by Joseph T. Kohler and marketed by Total Environmental Action, Inc. This program uses an implicit finite difference technique to solve transient heat conduction problems.[2] The program is run on a Texas Instruments TI-59 hand held programmable calculator equipped with a PC-100A printer.

A thermal network of 7 nodes, the maximum allowed by the program, is constructed to model the simulated space. Each node is characterized by a capacitance, or mass heat capacity. Each node may be connected to any other node by a conductance with the assumption that conductance between nodes is linear. There is a provision for energy input from without the network as in the case of a one hour time step. At each hour, node temperatures are revised to reflect the thermal traffic during the preceding hour. This is an effective simulation technique that models the complex interaction of thermal conductances, material capacitance, auxiliary energy inputs, and the variation of heat gains. In order to apply this tool, these various parameters must be specified within the framework of a thermal network.

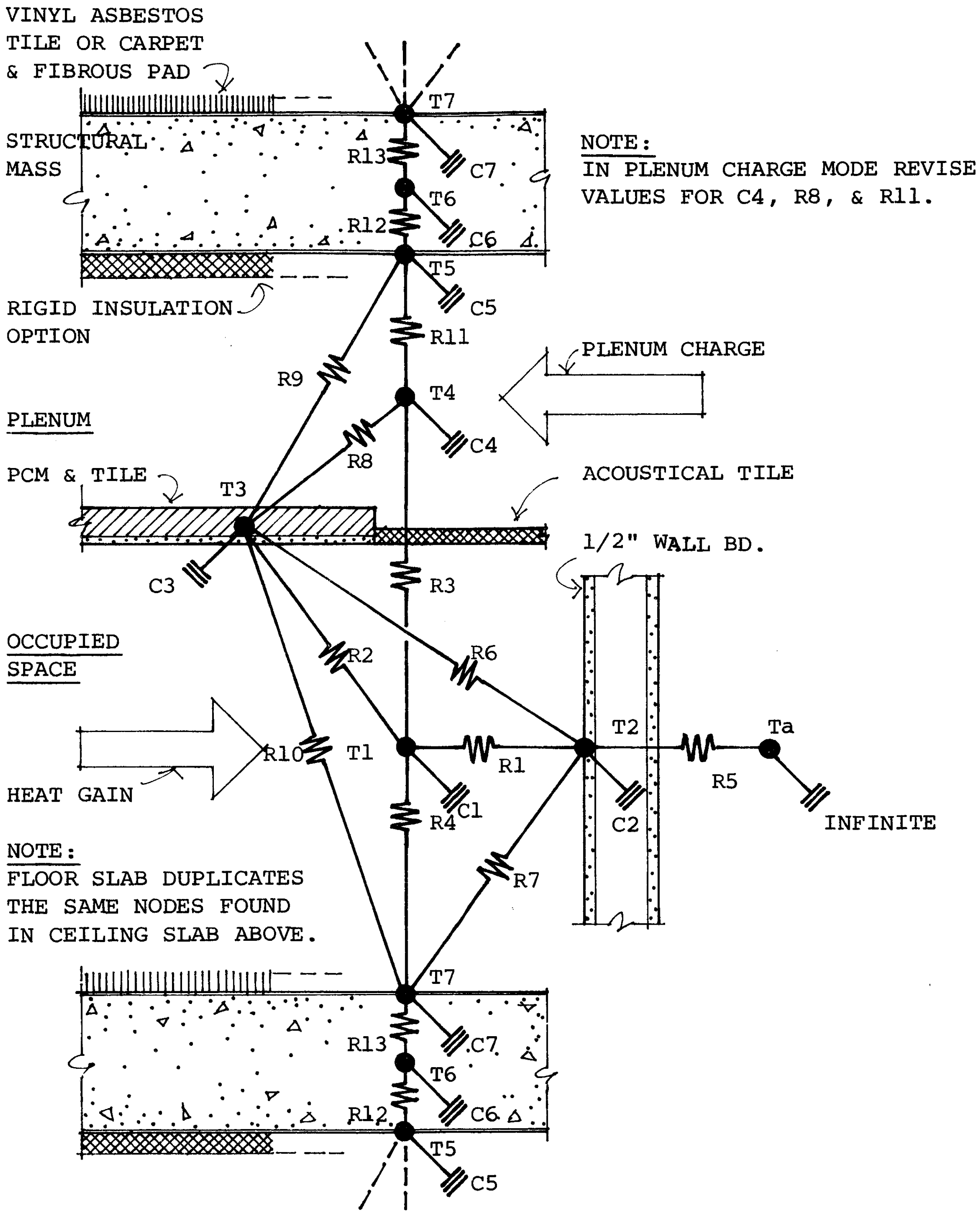
The thermal network used in the initial simulation is designed to model an interior office space. It is assumed that this office space is 500 SF and surrounded on all sides, above, and below by conditioned spaces. The occupied space has a suspended ceiling, with phase-change cool storage, separating it from the plenum above. The building structure includes 5 inch thick concrete slabs with finish floor to finish floor dimensions of 11'0". Node locations are summarized in Table No. 2.

TABLE NO. 2

Node No.	Location
1	Room air space
2.	Interior wall finish (all walls)
3.	Phase-change storage material and tile
4.	Plenum air space
5.	Surface of concrete slab facing plenum
6.	Concrete slab
7	Surface of concrete slab facing room

This thermal network is diagrammed in Figure No. 16. Each node must be assigned a capacitance and the relationship of nodes must be defined through conductances. The values designated for the nodes related to the plenum will vary depending on whether the plenum air is being circulated as during the storage charge period, or whether the air is still, as it is during the peak periods. Aside from these variations, and the changing patterns of heat gain throughout the day, the values for the network remain unchanged during a run. It should be noted that node 7, the concrete surface at the room, is in thermal contact with the concrete slab and the room air. In effect, this network continues through the vertical building section into the occupied spaces above and below. The next task is assigning node capacitances.

In the first situation modelled, the concrete slab is exposed to both plenum and occupied space. The capacitance for node 1, the room air space, is rated at 353 BTU/°F. This includes the negligible capacitance of the air in addition to room furnishings. Node 2 includes the mass of interior partitions. The 900 SF of 5/8" thick wall board represents 47 cubic feet of gypsum with a capacitance of 947 BTU/°F. For capacitance at node 3, the phase-change storage material is assumed, as a starting point, to cover 65% of the 500 SF ceiling area. The phase-change storage has a large heat capacitance with the majority of the heat



NOTE:
IN PLENUM CHARGE MODE REVISE VALUES FOR C4, R8, & R11.

NOTE:
FLOOR SLAB DUPLICATES THE SAME NODES FOUND IN CEILING SLAB ABOVE.

NOTE:
NODES NO. 7 & 5 ARE MASSLESS SURFACE NODES WHICH MAY OR MAY NOT INCLUDE RESISTANCE FROM CARPET OR INSULATION IN ADDITION TO AIR FILM RESISTANCE. ALL CONCRETE CAPACITANCE IS AT NODE 6.

FIGURE 16 Thermal Network Used for Initial Simulations stored in the process of melting the salts.This heat capacitance cannot literally be described in terms of BTU/°F because of the latent heat storage mechanism. The simulation method has no provision for latent heat storage, therefore, the heat capacitance must be modelled as a high sensible heat capacitance over a limited temperature swing. The tiles and the 5 pounds per SF of phase-change material they support, provide 150 BTU of storage over a 5°F temperature swing. For 325 SF of coverage this is modelled as a 10,000 BTU/°F capacitance over the limited 5°F temperature swing. Node 4, the plenum air space, must be modelled two different ways. When the air space is static, during the daytime peak periods, the capacitance is that of the air or 18 BTU/°F. However, when air is circulated through the plenum to charge the tiles, this node has a higher effective capacitance due to the mechanical cooling of the circulated air. This capacitance is taken to be the BTU supplied per hour by the nighttime chillers. During the simulation run, the temperature of node 4 is lowered by 1°F during each hour the plenum is being charged. This, in effect, introduces the specified BTU/hr from the chiller into the plenum. The mechanical cooling of the plenum space during charge periods would limit temperature excursions within this small range. Nodes 5 and 7 represent the surfaces of the concrete slab. Both of these nodes are modelled with zero mass, thus capacitance is 0 BTU/°F. The 5 inch thick concrete slab covers 500 SF providing 208 cubic feet of sensible storage. At 31.7 BTU/cuft°F this gives node 6 a capacitance of 6603 BTU/°F. These capacitances are summarized later in Table No. 3.

The next stage in simulating system performance is the definition of the thermal relationships that occur between the nodes. Heat can be exchanged from node to node by radiation, convection, conduction, a combination of these mechanisms, or not at all. The thermal relationship between nodes is defined by conductance which is a product of U value and the surface area of the nodes in question. These relationships set the stage for the analysis of crucial energy flows and are summarized in Table no. 3. For example, the room air temperature as modelled by node number 1,is affected by four different conductances in addition to hourly heat gain. Room air transfers heat by convection with nodes 2,3, and 7 and transfers heat by conduction with node 4. The proper definition of conductance values is critical to the integrity of the simulation. The values assumed for the initial simulation run will be quickly outlined:

R1. The heat transfer between room air and the interior wall surfaces is via convection, with a heat transfer coefficient of .60 BTU/SF°Fhr and wall surface area of 900 SF this yields a total thermal conductance of 540 BTU/°Fhr.

R2. The heat transfer between room air and the phase change material (PCM) storage tile surface is via convection. The heat transfer coefficient is taken at .75 BTU/SF°Fhr (heat flow up) and tile surface area is 325 SF yielding a total thermal conductance of 244 BTU/°Fhr.

R3. The heat transfer between room air and plenum air involves conductance through the 175 SF of acoustical ceiling tile that finishes the balance of the ceiling. The tile has an R value of 1.42 and air film resistances total 0.73 yielding a total resistance of 3.15 or a heat transfer rate of .32 BTU/SF°Fhr. This gives a total thermal conductance of 56 BTU/°Fhr.

R4. The heat transfer between the bare concrete floor and room air is via convection. For heat flow down, this transfer coefficient is .30 BTU/SF°Fhr over a 500 SF floor area for total thermal conductance of 150 BTU/°Fhr.

R5. The heat transfer between the interior wall and the room beyond is actually negligible due to the very low Δt. Nevertheless, the walls have a U value of .28 BTU/SF°Fhr and an area of 900 SF for a total thermal conductance of 252 BTU/°Fhr.

R6. The heat transfer between the interior wall and the PCM tiles is via radiation. The heat transfer rate for all radiant transfer in this temperature range (50° to 100°F) is considered to be .90 BTU/SF°Fhr.[3] The ceiling tiles,with an area of 325 SF,are estimated to "see" the 900 SF of wall for 98 SF of transfer area. This method involves making an estimate of the amount of radiant transfer between the ceiling and walls relative to that between the ceiling and floor. The maximum sum of the two is limited by the 325 SF area of the tile surface. Transfer between tile and wall finish is .90 BTU/SF°Fhr over 98 SF for a total thermal conductance of 88 BTU/°Fhr.

R7. The heat transfer between the interior wall finish and the concrete floor is also via radiation. Transfer rate is .90 BTU/SF°Fhr over an estimated area of 500 SF for a total thermal conductance of 450 BTU/°Fhr.

R8. The heat transfer between the PCM storage tiles and the plenum air will vary depending on the status of air circulation in the plenum. During the static daytime period, the heat transfer rate is .30 BTU/SF°Fhr with a tile area of 325 SF for total thermal conductance of 97 BTU/°Fhr.

During the nighttime charging period, the heat transfer coefficient will vary with flow turbulence and air velocity. The determination of this convective heat transfer coefficient is a matter of some confusion. A literature search summarized in Appendix B found a dozen methods for calculating this coefficient with various methods yielding a wide range of values. This simulation will use a value of 1.3 BTU/SF°Fhr. There is flexibility in the system for compensation should this value prove to be too optimistic. Total heat transfer could be increased by extending the PCM surface area or by increasing the plenum air velocity. Using the value of 1.3 BTU/SF°Fhr with an area of 325 SF gives a total thermal conductance, during charging periods, of 422 BTU/°Fhr.

R9. Thermal contact between the PCM tiles and the plenum surface of the concrete slab is via radiation. The PCM area of 325 SF and radiant transfer coefficient of .90 BTU/SF°Fhr yields a total thermal conductance of 292 BTU/°Fhr.

R10. The PCM tiles are in radiant contact with the room side surface of the concrete floor. This contact is assigned an area of 227 SF representing the balance of the PCM surface area after allowing for radiant contact with the room wall surfaces (R6). With a radiant transfer coefficient of .90 BTU/SF°Fhr this yields a total thermal conductance of 204 BTU/°Fhr.

R11. The heat transfer between the plenum surface of the concrete slab and the plenum air is also governed by the air circulating status of the plenum. During static periods (daytime) the convective transfer coefficient is .73 BTU/SF°Fhr over 500 SF for a total thermal conductance of 365 BTU/°Fhr.

During the nighttime charge period the aforementioned convective transfer coefficient of 1.3 BTU/SF°Fhr governs yielding a total thermal conductance of 650 BTU/°Fhr.

R12. Heat transfer between the plenum surface of the concrete and the concrete mass node is via conduction. The concrete mass node is located at the center line of the concrete slab or 2.5° from either surface. Concrete has a resistance of .13 BTU/SF°Fhr per inch thickness giving a total transfer coefficient of 3.0 BTU/SF°Fhr. Slab area is 500 SF giving a total thermal conductance of 1500 BTU/°Fhr.

R13. Heat transfer between the concrete mass node and the occupied space surface of the concrete is via conduction. This value is identical to R12 at 1500 BTU/°Fhr.

TABLE NO. 3. NODE SUMMARY FOR RUN NUMBER 1.

From Node No.	To Node No.	Mark	Conductance in BTU/SF°Fhr	SF	UXA BTU/hr°F	Capacitance BTU/°F
1	2	R1	.60	900	540	353
	3	R2	.75	325	244	
	4	R3	.32	175	56	
	7	R4	.30	500	150	
2	Ta	R5	.28	900	252	947
	3	R6	.90	98	88	
	7	R7	.90	500	450	
3	4	R8	.30	325	97	9999
	(4)	(R8)	(1.30)	(325)	(422)	
	5	R9	.90	325	292	
	7	R10	.90	227	204	
4	5	R11	.73	500	365	18
	(5)	(R11)	(1.30)	(500)	(650)	(8750)
5	6	R12	3.0	500	1500	0
6	7	R13	3.0	500	1500	6603
7						0

Note: Use values in brackets during charge modes

INITIAL SIZING OF SYSTEM COMPONENTS

It is useful to review the thermal characteristics necessary for a successful off-peak cooling system at this stage. The system must be carefully designed to provide thermal comfort under a wide range of cooling loads, including design day conditions. The design day, or maximum load, conditions will govern in establishing the heat capacity and heat transfer characteristics of the system components. Several major thermal relationships will be quickly examined as a check of system viability.

First, the thermal storage capacity must be capable of meeting the design day cooling load without the input of additional cooling, beyond the daytime chiller. The mechanical chilling equipment must be sized to recycle the thermal storage system within the time constraints of the overnight charging period. Beyond the mechanical requirements for this storage recharge, the heat transfer properties of the charging system must be adequate. The heat transfer from plenum air to thermal storage must keep pace with the amount of cooling provided by the recharge chillers. Finally, the heat transfer between the charged storage and the occupied space, during the daytime peak periods, must be capable of meeting peak cooling loads as they occur.

The density of thermal storage capacity will be determined by the maximum cooling loads experienced in the space served by the system. The design day cooling load has been established for the 500 SF interior zone test space. The load profile established for this space, see Figure No. 15, has a total daily load of 229 BTU/DaySF of floor area. The operation of the daytime chiller, 6.3 BTU/SFhr for 10 hours a day, will reduce this load to 166 BTU/DaySF. Of this 166 BTU/Day SF approximately 64 BTU/DaySF will occur during the off-peak charging period. This component does not require thermal storage capacity. The remaining load of 102 BTU/DaySF of floor area must be carried by the thermal storage system. As an initial assumption, the thermal storage ceiling tiles will cover 65% of the ceiling area. Each square foot of these ceiling tiles will support approximately 5 pounds of PCM with a heat capacity of 30 BTU/pound. Thus, for each square foot of floor area, there will be .65 square feet of PCM storage providing a capacity of 98 BTU/DaySF of floor area. In addition to the PCM tiles, the 5 inch thick concrete slab provides 46 BTU/DaySF of floor area when cycled through an anticipated 3.5° temperature swing. The contribution of this sensible storage will depend on the nature of its thermal contact with the occupied space. The participation of this sensible storage will decrease the load on the PCM storage and therefore its temperature swing in the model. Thermal storage capacity appears adequate on the basis of these rough calculations. Successful performance will depend on the system's ability to cycle this storage.

For the purposes of this study, the off-peak recharging period is assumed to be 14 hours long. The mechanical chilling equipment must be sized to recharge the thermal storage system within this time constraint. Of the total daily load of 229 BTU/DaySF, the nighttime chillers must carry 166 BTU/DaySF; 64 BTU in direct loads and 102 BTU for recharging storage. Over the 14 hour charging period this requires chilling capacity of 12 BTU/SHhr for the 757,000 SF test building. This would yield 750 tons of chilling capacity were the whole building subject to the load pattern established for the interior zone. Actually, additional exterior zone loads would require an additional 150 tons of cooling bringing the total nighttime chiller load to 900 tons. This load could be met by running an additional 500 ton chiller in conjunction with the 400 ton chiller required for daytime loads. The 900 tons of chilling capacity can be compared to the 1560 tons required by the existing conventional system. Of course, the off-peak system, like the existing mechanical system, will require additional stand-by chillers to allow for scheduled maintenance and equipment failure. This stand-by capacity is also reduced relative to that required by a conventional system.

A major limiting factor in the flow of energy through this system is the rate of heat transfer between the chilled plenum air and the thermal storage it is charging. This heat transfer rate is determined by the surface area of storage exposed, the convective heat transfer coefficient, and the temperature difference between air and storage. The relationships are summarized by:

$$\text{Heat Transfer (BTU/hr)} = \text{Convective Transfer Coefficient (BTU/hrSF°F)} \times \text{area (SF)} \times \Delta t\ (°\text{F})$$

The phase-change material has a thermal storage capacity of 150 BTU/SF. This capacity must be charged in the 14 hour charging period, requiring a heat transfer rate of 10.7 BTU/hrSF. Assuming a convective transfer coefficient of 1.3 BTU/hrSF°F, as discussed earlier and in Appendix B, this would require a Δt of 8.2°F. The Δt in charging the phase-change storage can be defined as:

$$\Delta t_{\text{charge}} = t_{\text{set point}} - \left(\frac{\text{T plenum supply} + \text{T plenum return}}{2}\right)$$

With an assumed PCM set point temperature of 64°F and a Δt charge of 8.2°F, the average plenum air temperature, during the charge cycle, must be 55.8°F. Sensible heat storage in the exposed concrete slab is assumed to be 46 BTU/DaySF. With the same 14 hour charging period and a Δt of 8.2°F, the convective transfer coefficient for this surface need only be .4 BTU/hrSF. This should be easily obtained even if the concrete is isolated from the mainstream of the airflow.

It is desirable to run a quick check on the required supply air temperature for the plenum charge cycle. The temperature swing of the charging air can be calculated with system characteristics determined so far. Design day load on the test space is 229 BTU/DaySF of floor area of which 166 BTU/DaySF must be supplied during the nighttime charging period. This amounts to 11.8 BTU/hrSF of floor area over the 14 hour charging perior. A working assumption is made at this point that the plenum charging air will be supplied at a rate of 2 cfm/SF of floor area.

This figure is about twice the air handling capacity required by conventional HVAC systems. However, this higher flow rate will provide higher convective transfer coefficients and a lower Δt for the charging air. For 2 cfm/SF of air to provide 12 BTU/hrSF of cooling, the air must cycle through a temperature swing defined by

$$\Delta t_{\text{plenum air}} = \frac{12 \text{ BTU/hrSF}}{2 \text{ cfm/SF} \times 60 \text{ min/hr} \times .018 \text{ BTU/cf}^\circ\text{F}} = 5.5^\circ\text{F}$$

Previous calculations gave an average plenum air temperature of 55.8°F. This average temperature minus one half of the temperature swing gives a supply air temperature of 53°F. This is very near the 55°F chilled air temperature supplied by conventional mechanical system components. The plenum charging cycle requires air at the same temperature a conventional system would provide, yet it has twice the airflow and operates over a longer period. At first glance, it would appear that the off-peak system is imposing a considerably larger cooling load. In fact, the loads are the same. The disparity is explained by examining the air flows in the system. In the closed loop charging mode, the air to be chilled is 100% return air at the relatively low temperature of 58.5°F. Because of this low return temperature the large volume of air chilled does not represent a larger cooling load.

The heat flow between the cool storage tile and the occupied space is a final parameter that must be checked for adequacy. The load profile established for the interior zone test space has a maximum load of 19 BTU/hrSF of floor space. Of this maximum load, the daytime chiller will provide 6.3 BTU/hrSF leaving a load of 12.7 BTU/hrSF of floor area for the cool storage system to carry. The PCM tiles have a heat transfer coefficient, heat flow up, of 1.64 BTU/hr°SF and a Δt of approximately 10°F. This provides a heat transfer rate of 16.4 BTU/hrSF for each SF of PCM tile surface. With a 65% coverage of PCM tile in the ceiling, this will provide 10.6 BTU/hrSF of floor area. Thus, the tiles will be capable of absorbing 85% of the maximum daytime load that is not removed by the daytime chiller. This slight shortcoming in heat flow is allowable in light of the short duration of maximum load conditions. In addition, the concrete slab can be expected to contribute to the remaining 15% of peak load. The magnitude of this contribution will depend on the time lag effects of the concrete and the finish floor material in the room.

On the basis of the preceding figures concerning heat capacitances and heat transfer within the proposed system one can conclude that the proposed off-peak cooling system can conceptually function. Successful system performance will depend on the realization of the assumed thermal characteristics presented in this section. Having established the thermal parameters of the system based on the summary energy flows, the next step is a detailed examination of system performance on an hour by hour basis. The simulation program TEANET is used to provide this information, providing an hourly accounting of thermal transfer within the system. Simulation with TEANET will begin with a basic performance run based on the thermal characteristics already established. Further runs will include several simulations to establish the sensitivity of various parameters. However, before presenting these simulations, one final nuance must be explained.

During the plenum charge cycle, the plenum air is constantly recirculated through the air handling system. At each pass, the air temperature is reduced 5.5°F by mechanical chilling at the air handling unit. The 500 SF test space requires 1000 cfm following the assumed circulation rate of 2 cfm/SF. WIth a plenum volume of approximately 1000 cubic feet, the system will experience an air change once every minute. Clearly the thermal traffic in this process occurs in a faster time frame than the one hour time step used in TEANET. If the plenum air were represented during its charge cycle by its static capacitance value of 18°F BTU/°F, exaggerated and incorrect temperature values would occur. This is unacceptable because heat transfer from plenum air to thermal storage is a function of plenum air temperature. In addition, the plenum air temperature will vary during the overnight charge cycle. This variation rules out a fixed plenum temperature approach. The input of cooling at a constant rate (i.e., 100% output of nighttime chillers) will establish a constant Δt between charging air and thermal storage. It follows that the plenum air temperature will vary in step with the thermal storage temperature. Preliminary indications place this temperature variation in the 3°F to 3.5°F range during the overnight charging cycle. In order to accurately model the plenum air temperature during the charge period, node 4 must exhibit a very small hourly temperature swing. This characteristic is common to nodes associated with high thermal mass.

The node describing the plenum air space can be changed to reflect the process of recirculating and cooling the planum air. During the charge period, the capacitance of node 4 is revised from 18 BTU/°F, representing the actual 1,000 ft^3 of air, to a much higher value. This is as though all the air circulated through the plenum during the hour was present in one large mass. One consequence of modelling node 4 with a high capacitance is an apparent, though non-existent, loss of cooling energy during the charging process. The plenum air, now characterized by a high thermal mass, cools in response to reductions in thermal storage temperature. This cooling of the mass used to model the air does not actually occur. To compensate, the amount of cooling supplied to the plenum, in the simulation only, must be increased to cover this non-existent **loss.** For the TEANET simulations, the plenum air capitance will be modelled as 8750 BTU during the charging period. At each hour in the charging period, the plenum air temperature will be reduced by 1°F. This temperature reduction is the equivalent of providing 8750 BTU/hr of cooling to the plenum. A portion of this cooling, however, is lost as the high mass plenum air model is cooled. This loss lowers the actual cooling provided to the simulation from 8750 BTU/hr to approximately 6,500 BTU/hr. These figures are derived below:

Apparent cooling supplied	8750 BTU/hrx14 hrs = 122,500 BTU
Cooling of high mass plenum air model	8750 BTU x 3.5°FΔt = 31,000 BTU
Actual cooling supplied	91,500 BTU

This actual cooling of 91,500 BTU over 14 hours is 6545 BTU/hr or approximately 13 BTU/hrSF. This is close enough to initial assumption of 12 BTU/hrSF. This modelling method can be checked for validity by monitoring the Δt between the plenum air and thermal storage. If the Δt remains constant, a constant amount of cooling is provided and the model is successful.

FIRST SIMULATION RUN

The basic simulation, labeled run number 1, is shown in Figures no. 17, 18, and 19. These graphs represent the system's thermal behavior over three consecutive 24 hour periods. The simulation is started at 9AM in the first day. Initial node temperatures, for each of the seven nodes were estimated. The simulation was run using the heat gain profile developed for design day conditions in an interior zone space located in Boston. This load profile was summarized in Figure no. 15. This simulation is based on the thermal network, and its capacitance and conductance

FIGURE 17

Simulation Run No. 1
Day No. 1

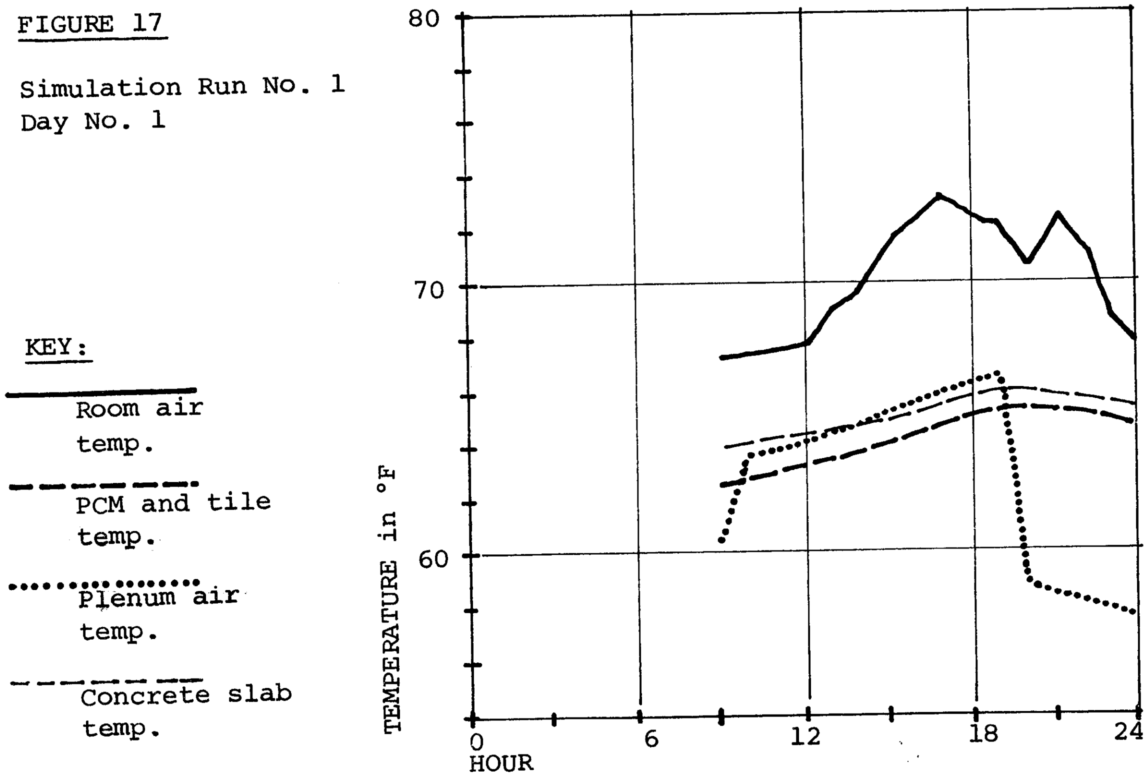


FIGURE 18

Simulation Run No. 1
Day No. 2

KEY:

——— Room air temp.

— — — PCM and tile temp.

•••••••• Plenum air temp.

— — — Concrete slab temp.

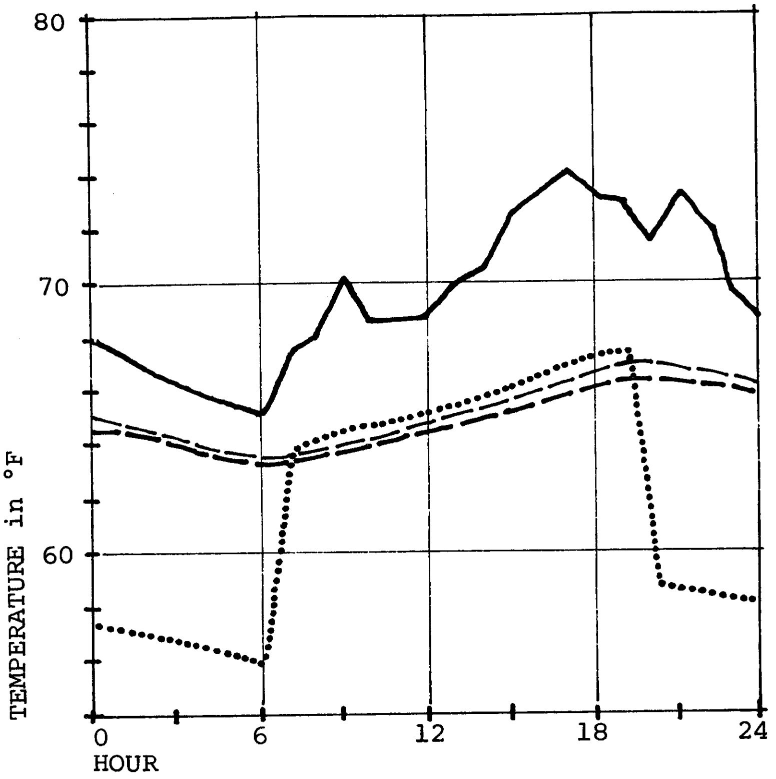


FIGURE 19

Simulation Run No. 1
Day No. 3

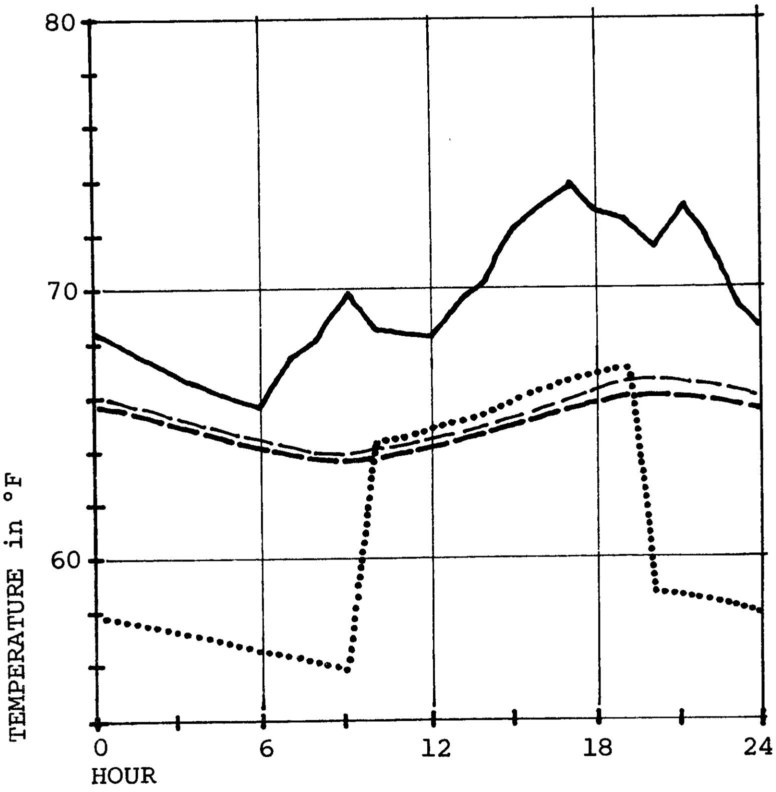
KEY:

——— Room air temp.

— — — PCM and tile temp.

•••••••• Plenum air temp.

— — — Concrete slab temp.



characteristics that were established earlier in the paper. The network is shown in Figure no. 16, and the thermal values are summarized in Table no. 3. The daily operation cycle is divided into a 10 hour peak period and a 14 hour off-peak period. The peak period, from 9 AM to 7 PM, is the operating period for the daytime chiller only. The off-peak period, from 7 PM to 9 AM is the charging period for the thermal storage system. Note that in day number 1 the charging period was shortened to prevent excessive space cooling. The system's performance for day three indicates that the system has reached a state of near equilibrium. The temperature of each node at the beginning of the period is nearly equal to the temperature at the end of the period. Hourly values for node temperature during this third 24 hour period are given in Table no. 4. This initial simulation run can be the basis for several major conclusions.

A quick review of the system can be made by comparing the results of simulation run number 1 to some of the predictions and assumptions made in the rough energy balance calculations. These calculations predicted a Δt between plenum charging air and thermal storage of 8.2°F. This Δt was necessary to achieve adequate heat transfer. The figures from day 3 of simulation run number 1 indicate that this value does occur. In addition, the 55.8°F deemed necessary as the average plenum air temperature during the charge cycle is evidenced at the end of the charging period. At the end of the charging cycle, the PCM storage is at its 64°F set point rather than the warmer temperatures experienced at other times. In other words, the predicted value of 55.8°F for average plenum air holds true. The Δt between the PCM storage and room air during daytime peak heat gain conditions falls short of the assumed 10°F. This 2°F shortfall reduces the heat transfer from the room, and is partially responsible for the 4°F air temperature rise during this period. The PCM storage cycles through a Δt of 2.6°F rather than the 5°F described earlier. The concrete slab mass undergoes a 3.2°F temperature excursion which is very near the assumed 3.5°F swing. The discrepancy in PCM temperature swing can be explained by the participation of the concrete slab's thermal storage. The total storage in both PCM and concrete slab

TABLE NO. 4: NODE TEMPERATURES FOR THE THIRD 24 HOUR PERIOD OF SIMULATION RUN NUMBER 1

	Nodes: Temperature in°F*						
	1	2	3	4	5	6	7
Hour	Room Air	Room Finish	PCM Tile	Plenum Air	Conc. Surface	Conc. Mass	Conc. Surface
0	68.5	69.9	66.0	58.0	64.0	66.0	67.0
1	68.0	65.5	58.0	64.0	64.0	66.0	65.5
2	67.0	67.5	65.0	57.5	63.5	65.5	66.0
3	67.0	67.0	65.0	57.0	63.5	65.0	65.5
4	66.5	66.5	64.5	57.0	63.0	65.0	65.5
5	66.0	66.0	64.5	56.5	62.5	64.5	65.0
6	65.5	65.5	64.0	56.5	62.5	64.0	65.0
7	67.5	66.0	63.0	56.0	62.0	64.0	65.0
8	68.0	66.0	64.0	56.0	62.0	64.0	64.5
9	70.0	67.0	63.5	56.0	62.0	63.5	65.0
10	68.0	67.0	64.0	64.0	64.0	64.0	65.0
11	68.5	67.0	64.0	64.5	64.0	64.0	65.0
12	68.5	67.0	64.0	65.0	64.5	64.5	65.5
13	69.5	67.5	64.5	65.0	64.5	64.5	65.5
14	70.0	68.0	64.5	65.5	65.0	65.0	66.0
15	72.0	68.5	65.0	66.0	65.0	65.0	66.5
16	73.0	69.5	65.0	66.5	65.5	65.5	67.0
17	73.5	70.5	65.5	66.5	66.0	66.0	67.5
18	73.0	70.5	66.0	67.0	66.5	66.5	68.0
19	72.5	70.5	66.0	67.0	66.5	66.5	68.0
20	71.0	70.5	66.0	58.5	64.5	66.5	67.5
21	73.0	70.5	66.0	58.5	64.5	66.5	68.0
22	72.0	70.0	66.0	58.5	64.5	66.5	67.5
23	69.0	69.5	65.5	58.0	64.0	66.0	67.0
24	68.0	68.5	65.5	58.0	64.0	66.0	66.5

*Note: all figures are rounded to the nearest 1/2°F is approximately 96.7 BTU/SF of floor area. This is very close to the 102 BTU/SF calculated need. Additional thermal storage in the room finishes, node 2, increases the total by 12 BTU/SF. Discussion of the PCM temperature cycle, and its ramifications on required storage capacity, will follow after. Finally, the plenum air temperature did cycle through 3.0°F of the 3.5°F swing predicted in the discussion of the non-existent storage capacity problem.

THERMAL COMFORT

As its primary goal, air conditioning is charged with maintaining a thermal environment that is conducive to human thermal comfort. Before the results of the first simulation run can be evaluated for this criteria, the conditions necessary for thermal comfort must be established. The human body is incredibly versatile, physiologically capable of surviving in a wide range of environmental conditions. However, the preception of thermal comfort is limited to a much narrower range of conditions. Comfort is achieved only if the variables of activity, clothing, and environmental conditions are balanced. The human body controls heat loss through the manipulation of skin temperature and the production of sweat. The purpose of the body's thermoregulatory system is to maintain an essentially constant internal body temperature. To achieve this constant body temperature, the body as a system must experience no net heat gain or loss. Therefore, a heat balance must be established in which heat production in the body equals heat loss from the body. The heat produced by the body will vary with an occupant's activity level and physical condition. The mechanisms by which the body loses heat are radiation, convection, conduction, and evaporation. The quantities of heat rejected via these mechanisms will depend on the body's regulation of skin temperature and sweat as well as the environmental variables of air temperature, relative humidity, air velocity and mean radiant temperature. Finally, the rate at which the body loses heat will be influenced by the amount of clothing worn. These major variables can be juxtaposed to produce a wide range of situations in which human thermal comfort would be achieved. However, in the context of an office environment this range is limited by the predictable values of activity and clothing variables.

While the long term maintenance of a body's heat balance is considered mandatory for biological survival, it does not guarantee thermal comfort. Empirical studies have concluded that skin temperature and sweat production values must fall within a narrow range for thermal comfort. The specific values required for thermal comfort will vary with activity level. For sedentary persons, typical thermal comfort conditions require a mean skin temperature of 93.2 and the absence of sweat secretion.[4] At higher activity levels the skin temperature necessary for comfort falls, and moderate sweating takes place. P. O. Fanger has developed a complex equation that models all of the variables affecting thermal comfort.[5] This equation, based on heat transfer theory and empirical studies, was used by Fanger to perform a series of sensitivity analyses on the major comfort variables. The result is a series of charts that summarize the environmental conditions necessary for thermal comfort over a range of clothing and activity levels. The four charts applicable to the interior office environment are shown in Figure no. 20.[6] These charts contain evaluations for activities in the sedentary range (50 kcal/hrm^2) and medium range (100 kcal/hrm^2) for light exercise. An office worker is expected to produce 60 kcal/hrm^2 typing and 100 kcal/hrm^2 when walking slowly. A median value of 75 kcal/hrm^2 is taken as a typical level for further analysis. Clothing levels during the summer would range from 0.5 clo for short sleeves and open collar to 1.0 clo for a business suit. A median value of .75 clo will be used to represent the typical outfit. Thus, the thermal comfort criteria from Fanger's charts are interpolated to produce a graph appropriate for the office environment, Figure no. 21. These charts make the acceptable assumption of 50% relative humidity. Room air velocity represents another variable in comfort analysis. Room air velocities below 15 fpm lead to complaints of stagnancy; velocities in excess of 50 fpm are associated with draftiness.[7] Therefore, the composite chart in Figure no. 21 is based on an air speed of 40 fpm (.2 m/s). This composite chart will be the basis for evaluating the thermal environments predicted by the cooling system simulations.

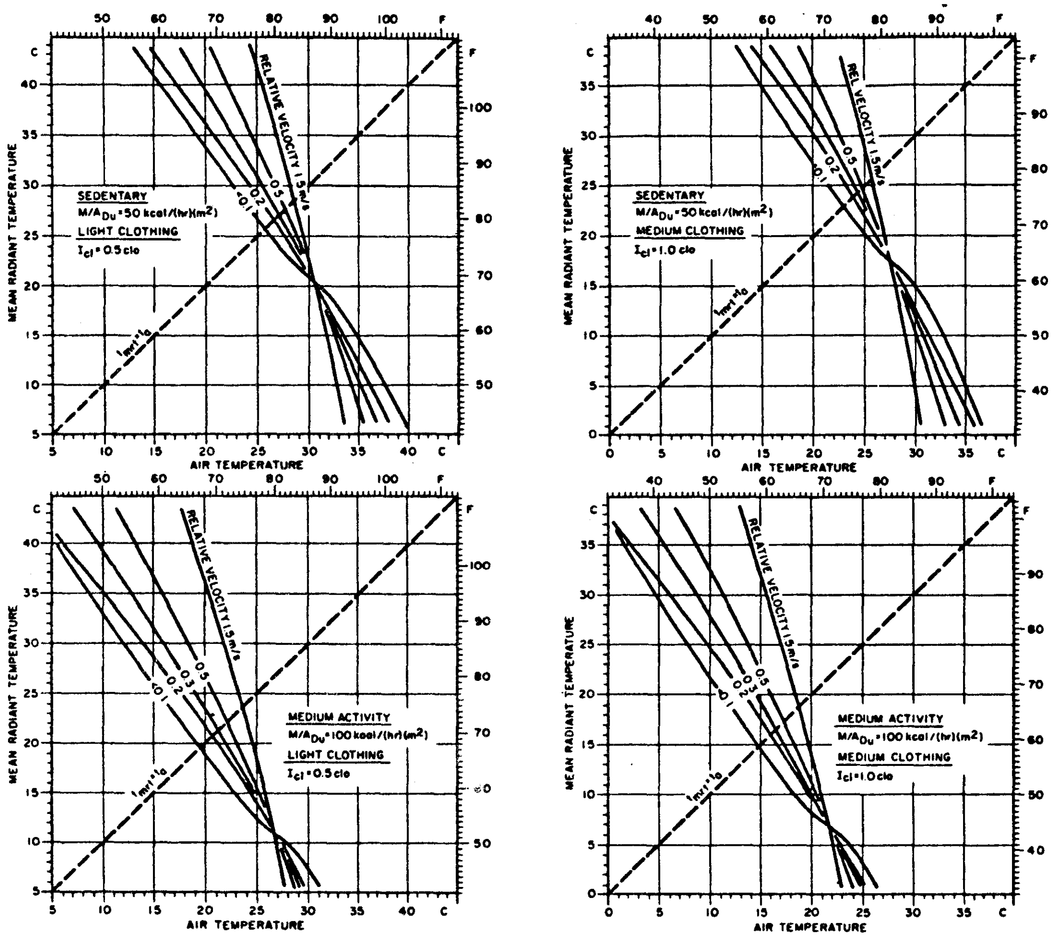


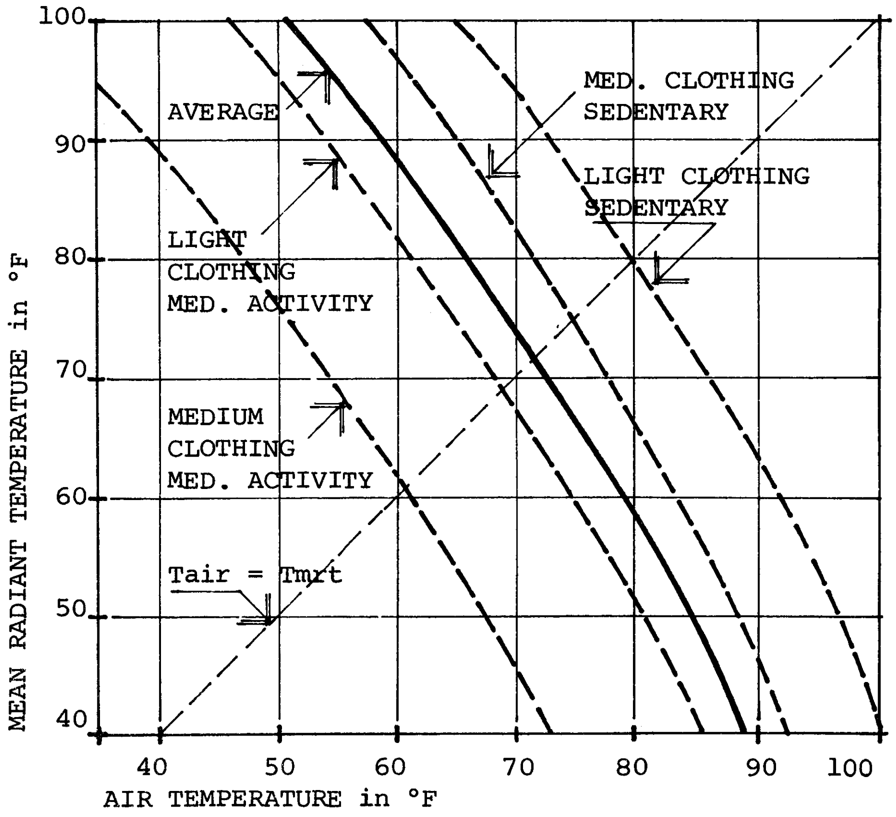
FIGURE 20 Combined Influence of Mean Radiant Temperature and Air Temperature.[6]

The evaluation of thermal comfort in the case described by simulation number 1 involves the two remaining major variables, room air temperature and mean radiant temperature. Room air temperatures are listed under node 1. Mean radiant temperatures are the average of room finishes, node 2, and the PCM storage, node 3. Relative humidity is assumed to remain near 50% with daytime chiller operation removing latent heat loads. Air speed has been established as 40 fps, the clo value at 0.75, and activity level as 75 kcal/hrm^2. The air temperatures experienced in the room range from 65.6°F at 6 AM to 73.5°F at 5 PM. This 8°F temperature swing is greater than the Δt that would occur with a conventional mechanical system. However, this temperature swing is advantageous in the effect it has on heat transfer. As the heat gains in the space increase, the room air temperature rises. This increases the Δt between the room air and the storage, resulting in greater heat transfer between the two. The cooling effect if the storage therefore increases in response to higher heat gains. The storage system can provide cooling for peak loads at twice the rate it provides for marginal load periods. This occurs only if the occupied space is cycled through this typical 8°F temperature swing.

FIGURE 21

Comfort
Criteria
from P.O.
Fanger.

ASSUMES
R.H. = 50%
air speed =
.2m/s

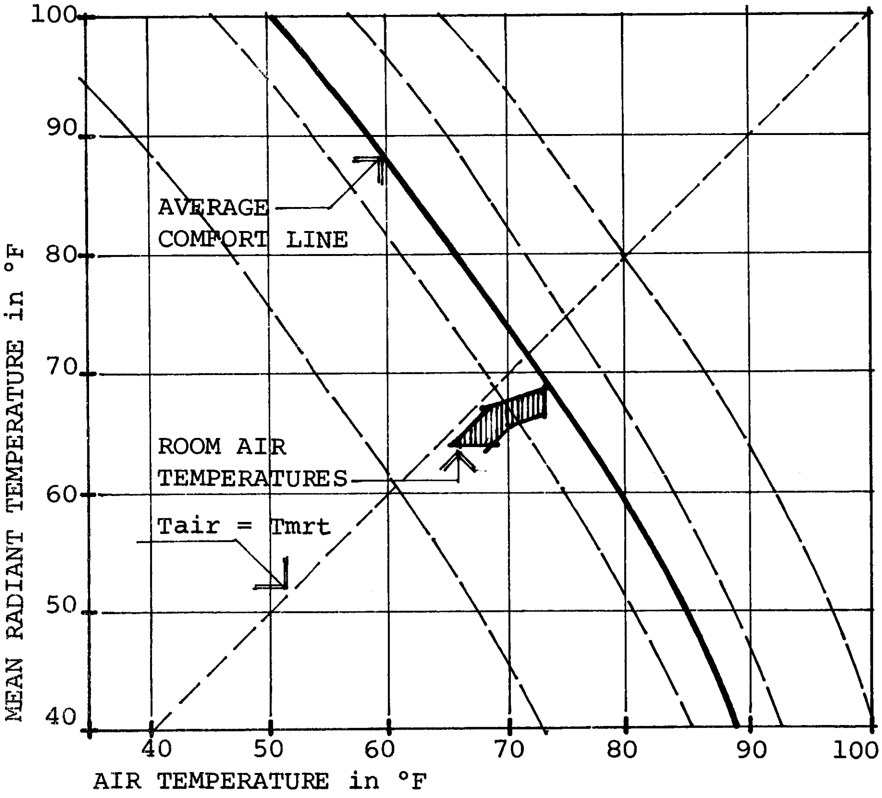


CONCLUSIONS

The off-peak cooling system certainly seems to work throughout the day; in fact it is working too much. The minimum temperature of 65.5°F is intuitively much too cold for thermal comfort. The maximum of 73.5°F seems well below the allowable high temperature, particularly in light of proposed legislation for 80°F thermostat settings. Both of these observations are further reinforced by the presence of even lower mean radiant temperatures. The room conditions are plotted on the thermal comfort graph, Figure no. 22. In examining this graph, one can conclude that the temperature in the system, both radiant and air, can be

FIGURE 22.

Air temperature from run number 3 plotted on comfort chart. 64°F set point temp.



increased by a minimum of 3°F. This increase will establish the temperatures experienced during the occupied portion of the day near the desired thermal comfort criteria. This change can be accomplished by raising the phase-change set point temperature to 67°F. This should raise the minimum air temperature to 69°F and the maximum to 76.5°F. The higher set point temperature will raise plenum temperatures and reduce the threat of condensation. It will also raise the temperature of the plenum supply air from 53°F to 56°F resulting in increased chiller efficiency. Therefore, the first major conclusion from simulation run number 1 is the need for a higher set point temperature.

The timing of the chiller operation and the patterns of heat gain in the space combine to cause air temperature fluctuations through the daily cycle. The lowest air temperature comes at 6 AM, the end of a 9 hour unoccupied period characterized by low heat gains. The air temperature then quickly rises 4°F during the next three hours in response to occupancy, and increased heat gain. The occupants contribute to this temperature rise by using machinery and lights, as well as rejecting body heat. A possibility exists that pre-occupancy heating may be desirable to raise the room air temperature before the workers arrive. This could be accomplished by conventional heating through the air handling unit,or by turning on the lights as an early heat source. Perhaps the early morning hours could be used by the janitorial crews thus promoting early gains during this period. A final possibility is to encourage the earliest occupants to increase their clothing levels for the first hour or so. They could, like an old acquaintance of mine, keep a light sweater of wrap at work for this purpose. The air temperature shows a dip at 9 AM as the daytime chillers begin their 10 hours of operation. This reduction of temperature could be reduced or eliminated by running the chiller at partial output. The purpose of the daytime chiller is to remove latent heat loads and, therefore, it should not have a great affect on room air temperature. The air temperature rises steadily from noon until its peak at 5 PM. This rise is in response to the maximum loads of the day. The magnitude of this temperature change could be reduced by increasing the thermal contact between the phase change material and the room. This would be done by increasing the surface area involved in the heat exchange, however, it does not seem necessary at this point. The air temperature falls slowly following the afternoon peak load, then rises briefly as the daytime chiller ends its operation. The temperature then continues to fall until the minimum occurs again at 6 AM. In summary, beyond the set point temperature issue, the daily temperature cycle would appear to be acceptable with the possible exception of the valley that occurs near 6 AM. Under partial load conditions, careful operation of both daytime and nighttime chillers should result in a profile that is even flatter.

The temperature swings experienced by the various nodes can be examined to determine the total amount of storage achieved during the simulation. This process is summarized in Table no. 5.

TABLE NO. 5. Thermal Storage at Each Node

Node	Location	Capacitance in BTU/°F	Δt °F	Thermal Energy Stored in BTU/Cycle
1	Room air	343	7	2,471
2	Room finishes	947	6	5,682
3	PCM storage	10,000	2.6	26,000
4	Plenum air	NA	0	0
5	Conc.surface	0	0	0
6	Conc.mass	6,603	3.2	21,130
7	Conc.surface	0	0	0
			Total	55,283 BTU

The total thermal energy stored is 55,283 BTU/cycle. In addition to the energy that ends up in storage, the nighttime chiller provided 34,820 BTU/cycle to meet heat gains that occured during this period. The total of these two outputs is 90,103 BTU/cycle. This can be compared to the anticipated demand during this period of 12 BTU/SF of floor area or a total of 84,000 BTU/cycle. The amount found in the simulation is larger because thermal energy stored amounted to 110 BTU/SF of floor rather than the anticipated 102 BTU/SF. This additional storage occurs in node 2, the room finishes. The 90,103 BTU/cycle accounted for during the nighttime portion of the simulation can also be compared to the amount of energy entering the simulation during this period. The charge was modelled at 8750 BTU/hr for 14 hours or a total of 122,500 BTU/cycle. The 32,399 BTU difference between storage and charge figures is the result of the previously discussed manipulation in modelling the plenum charge. This figure is the high air mass non-existent cool storage. With this taken into account thermal energy flow during the charging period is balanced and behaves roughly as predicted.

The phase-change material was modelled at a thermal storage capacity of 150 BTU/SF. This was accomplished by assigning a capacitance of 30 BTU/SF°F over a 5°F temperature range beginning at the 64°F set point temperature. Simulation run number 1 indicates that the PCM experienced a temperature swing of only 2.6°F. This represents a storage of 78 BTU/SF of PCM tile under design day heat gain conditions. However, it has already been demonstrated that the tile should have been charged with 150 BTU/SF (1.3 BTU/hr°F convective transfer coefficient x 8.2°F Δt x 14 hours). The difference between the 150 BTU/SF charge and the 78 BTU/SF stored can be explained by the PCM tile's participation in meeting heat gains that occur during the nighttime charging period. While these gains have been discussed before, no explicit method of removing them has been established. Gains during this period totaled 34,820 BTU/cycle or 107 BTU/SF of PCM tile. It can be assumed that the tile contributed 72 BTU/SF of this load with the remaining gains handled by the concrete slab storage.

As demonstrated earlier, the total storage provided by the PCM, concrete slab and interior finishes was a more than adequate 110 BTU/SF of floor area. It is interesting to note that the temperature of the concrete slab remains very close to the temperature of the PCM. This coupling is not surprising in light of the radiation contact between the two. One can conclude from these storage figures that the quantity of PCM required for each thermal tile could be lowered from 5 pounds/SF to 3 pounds/SF. This would reduce the thermal capacity of each tile from 150 BTU/SF to 90 BTU/SF.

This first simulation run also provides the opportunity to examine the system for potential condensation problems. Condensation is a concern in any air conditioning system due to the possibility of exposing a cool surface to the relatively moist air of the building. However, the surface temperatures encountered in the operation of this cooling system do not present much of a problem. The occupied space is exposed to the revised set point temperature of 67°F as a minimum surface temperature. It is assumed that the exposed metal ceiling suspension system components will equilibriate at the tile temperature due to their intimate thermal contact. The thermal tile surfaces, at 67°F, can be exposed to air at 75°F db and 65% relative humidity without condensation. The occupied periods characterized by high latent heat gains, will concurrently have latent heat removal via the daytime chiller. This chiller will be more than adequate to maintain interior conditions below the moisture content at which condensation will occur. Should water vapor migrate into the plenum during the day, no condensation would occur because the plenum surfaces are warmer than the PCM during this period. During the charge mode, the plenum is exposed to 55°F air. This will conceivably chill the exposed, plenum side, suspension system components below 60°F. However, this occurs during a period of low occupancy and minimum intake of outdoor air; therefore, very little, latent heat. In addition, the plenum air is recirculated, and exposed to a 53°F coil surface, at the rate of one air change per minute. Any condensation during this period would occur on the coil surface. One can conclude that this system will not experience condensation problems.

FURTHER SIMULATION

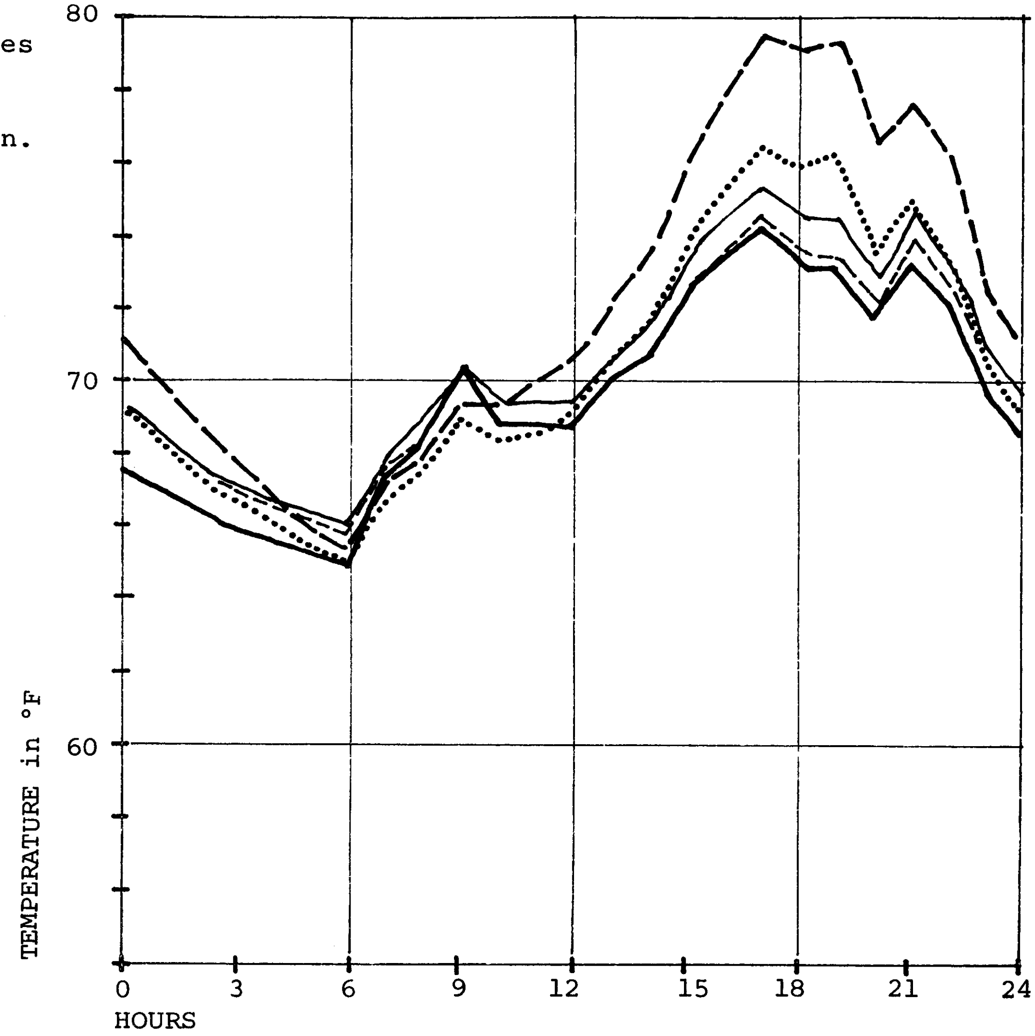
The first simulation run has provided evidence that the system's concept is viable. The off-peak cooling system has behaved largely as expected. When analyzed using the criteria of human thermal comfort, the environmental conditions described by the simulation suggest that the space is running too cool. Consequently, the phase-change set point temperature will be raised 3°F to 67°F. The simulation also led to the conclusion that the thermal capacity of the PCM storage could be lowered by 40% to 90 BTU/SF of thermal tile. This will allow thinner packaging sections and a reduction of ceiling weight. The system was examined for problems with condensation and given a clean bill of health. At this point, new simulations must be run to determine the effects of system operation with the new phase-change temperature and under partial cooling load conditions. However, before these simulations are presented, another set of simulation runs is described. These runs were conducted to test the effectiveness of the present thermal storage configuration. This series is based on the same 500 SF internal zone space using the heat gain profile presented for simulation run number 1. In these runs the thermal storage system was modified as follows:

RUN 2	System run without PCM storage material
RUN 3	Concrete slab insulated on room side by carpet
RUN 4	Concrete slab insulated on both sides
RUN 5	No PCM storage and concrete slab insulated on both sides

These runs are based on the original 64°F set point temperature allowing direct comparison with simulation run number 1. Each run begins at 9 AM of day 1 with the initial temperature of the nodes identical for each run. The graphs presented for these runs represent day number 2 of each respective run. The room air temperatures resulting from each of these changes are compared to run number 1 in Figure no. 23.

In simulation run number 2 the conditions modelled were identical to run number 1 with one exception. The phase-change material was omitted from the ceiling system. The support tile is included, providing node 3 with 341 BTU/°F of sensible heat storage. As shown in Figure no. 24, this simulation run results in afternoon room air temperatures that are 3°F warmer than those encountered when the PCM is included. The minimum temperature remains the same. An audit of storage cycle thermal energy is summarized in Table no. 6.

NOTE: Temperatures are from second day of each run.



KEY:

━━━━━━━━	Run No. 1	Standard configuration PCM, concrete exposed to plenum and room.
━••••••••••••	Run No. 2	System run without PCM storage material.
— — — — —	Run No. 3	PCM, concrete slab insulated on room side by carpet.
————————	Run No. 4	PCM, concrete slab insulated on both sides.
▬ ▬ ▬ ▬ ▬	Run No. 5	System run without PCM, concrete slab insulated on both sides.

FIGURE 23 A Comparison of Room Air Temperatures from the Simulation Runs Using a 64°F PCM Set Point.

TABLE NO. 6 Thermal Storage in Simulation Run No. 2 (No PCM in system)

Node	Location	Capacitance in BTU/°F	Δt °F	Thermal Energy Stored in BTU/Cycle
1	Room air	353	11.6	4,094
2	Room finishes	947	8.6	8,144
3	Support tile	341	10.0	3,410
6	Concrete slab	6,603	5.2	34,355
		Total thermal storage		49,983
		Nighttime heat gain		34,820
		Total		84,803

The total thermal energy accounted for is very close to the anticipated 84,000 BTU/charge cycle (12 BTU/hrSF x 500 SF x 14 hours/cycle). The figures evidence a 10% decrease in total thermal storage while the storage Δt for the concrete slab is almost doubled. The temperature swing of the ceiling tiles is increased fourfold. The higher temperatures of the storage surfaces during the afternoon cause the increase in room air temperature. The plenum charge temperature followed the concrete storage temperature by the expected range of 8°F to 9°F resulting in a charge rate similar to the first run. However, more thermal energy is stored in the concrete. This is partially because the low mass ceiling tiles are cooler, and therefore carry a larger portion of the nighttime heat gains. This allows the charging Δt of the concrete to increase. The situation modelled in run number 2 provides almost the same quantity of thermal storage without the use of phase-change material. However, the increase in afternoon room air temperature, and daytime radiant temperatures, make this system less attractive on the basis of performance.

Simulation run number 3 is distinguished by the addition of wall to wall carpeting and padding. This floor finish introduces an additional thermal resistance of 2.0°FSFhr/BTU between the concrete slab and the occupied space. The thermal storage capacities of all components, including the PCM, are modelled using the values of run number 1. As shown in Figure no. 25 this modification results in a temperature profile similar to the initial run. The morning air temperatures run about 1°F warmer, raising the minimum temperature encountered while the afternoon temperatures are only 1/2°F warmer. The addition of carpet, therefore, provides a little relief to the morning overcooling problem. The thermal energy stored is summarized in Table no. 7.

TABLE NO. 7. Thermal Storage in Simulation Run No. 3 (Concrete slab isolated by carpet)

Node	Location	Capacitance in BTU/°F	Δt °F	Thermal Energy Stored in BTU/Cycle
1	Room air	353	8.2	2,894
2	Room finishes	947	6.3	5,966
3	PCM and tile	10,000	2.7	27,000
6	Concrete slab	6,603	2.0	13,206
		Total thermal storage		49,066
		Nighttime heat gain		34,820
		Total		83,886

The carpet causes the concrete slab to run cooler than the PCM storage. The concrete slab reinforces the PCM storage through a radiant couple; cooling the thermal tiles as they cool the room. The reduced slab temperature improves this reinforcing effect. The concrete is cycled through a smaller temperature swing resulting in less storage quantitatively. However, the effect of this is minimal; a small rise in afternoon room air temperatures. It would seem that the addition of the carpet has had little deleterious effect on system operation. This can be attributed to

FIGURE 24

Simulation Run No. 2,
System Without PCM,
Day No. 2.

KEY:

━━━━━ Room air temp.

━ ━ ━ ━ Support tile only temp.

•••••••••••• Plenum air temp.

– – – – – Concrete slab temp.

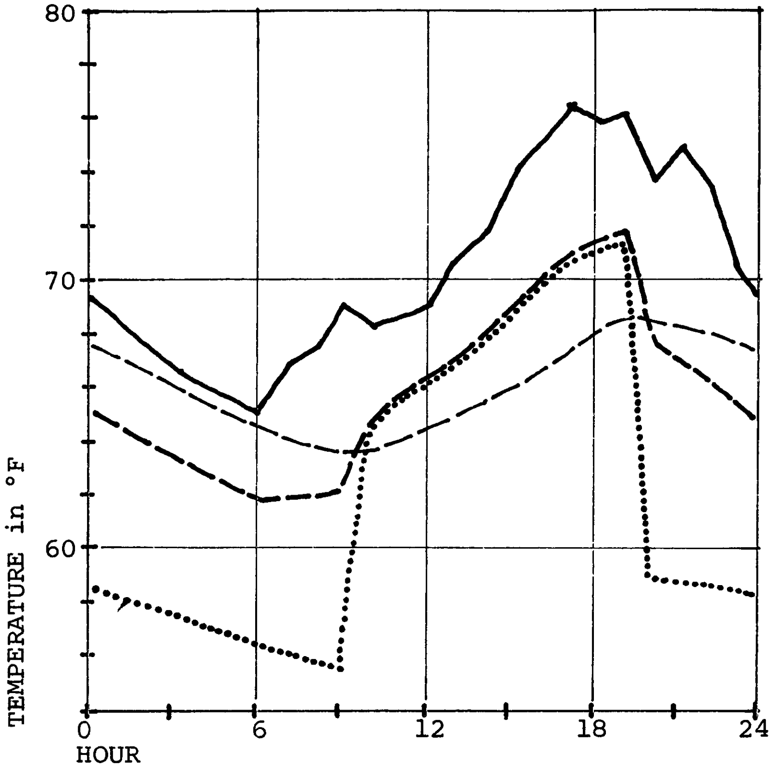


FIGURE 25

Simulation Run No. 3,
Carpet and Pad on
Finish Floor,
Day No. 2.

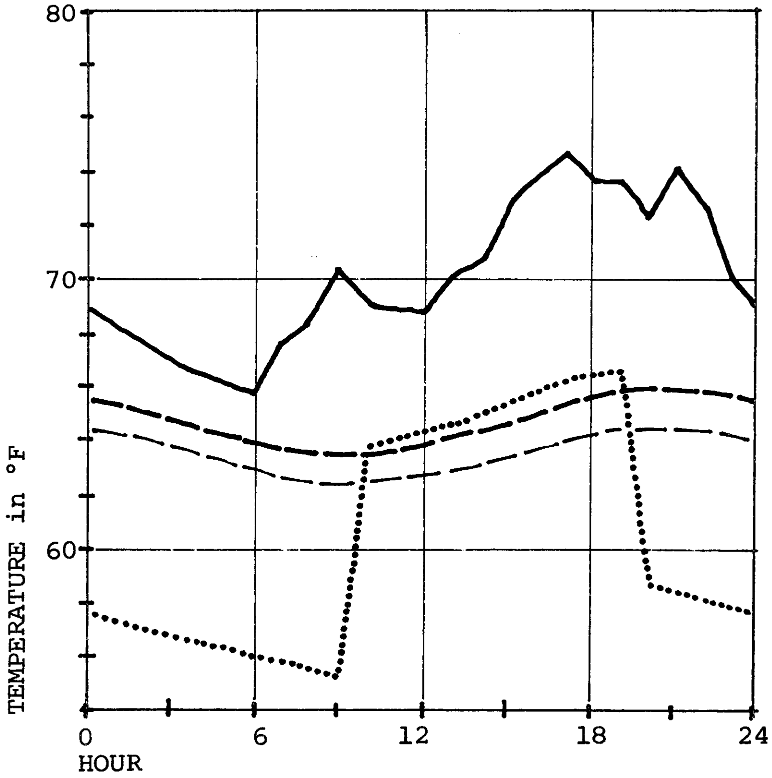
KEY:

━━━━━ Room air temp.

━ ━ ━ ━ PCM and tile temp.

•••••••••••• Plenum air temp.

– – – – – Concrete slab temp.



the relative magnitude of the revised R13 conductance (.46 BTU/hrSF°F) when compared to the existing R4 (.60 BTU/hrSF°F). It can be seen that the R4 value was a constricting point before the carpet addition and that the reduction of heat flow by the carpet is proportionally minor. This reduction was responsible for the decrease in concrete Δt. There are advantages to this situation. The minimum room air temperature was raised. Also, the temperatures at node 7, the finish floor surface, run 1.5°F warmer during occupied periods relative to those experienced during run number 1. This would provide a more comfortable surface temperature for those wearing thin summer shoes.

In simulation run number 4, shown in Figure no. 26, the concrete floor slab is thermally isolated on both sides. As in run number 3, carpet and pad provide thermal resistance on the occupied space side of the slab. In addition, 1" thick, foil faced rigid insulation covers the concrete slab on the plenum side. This effectively isolates the concrete from the charging plenum air , limiting the temperature swing of the concrete. This reduction of available storage,and the surface area available for charging, will reduce the amount of thermal energy that the system can handle during the charging period. This is evidenced by 5.7°F temperature drop in the plenum air mass during the charge cycle. This drop results in a Δt between storage and charging air that exceeds the prescribed 8.2°F. This exaggerated Δt causes an unrealistic charge Δt for the PCM of 3.2°F rather than the characteristic 2.7°F. The correct figure of 2.7°F is shown in the following thermal audit:

TABLE NO. 8. Thermal Storage in Simulation Run No. 4 (Concrete slab isolated on both sides)

Node	Location	Capacitance in BTU/°F	Δt °F	Thermal Energy Stored in BTU/Cycle
1	Room air	353	9.6	3,388
2	Room finishes	947	6.8	6,439
3	PCM storage tiles	10,000	2.7	26,000
6	Concrete slab	6,603	1.5	9,904
		Total thermal storage		45,731
		Nighttime heat gain		34,820
		Total		80,551

This audit shows the lowest thermal storage encountered to date. The PCM, without the reinforcing effect of the slab storage, experiences a 3.2°F swing in the discharge or daytime mode. The plenum charging system would not be able to fully recharge the PCM thermal storage during the 14 hour charging period. The PCM temperature could be expected to increase (in the model) 1.1°F per cycle. The cool storage system could still function in this situation; if the storage capacity is oversized and fully charged at the beginning of each week. The storage would be partially depleted each day and recharged as much as possible during the night. Each successive day would find less potential cooling in the storage system. System temperatures, however, should remain stable through the week due to the isothermal nature of the PCM. The PCM would require a very long charging period during the weekend to compensate for the shortfall in charging during the week. Clearly these problems are academic if a concrete slab is available for auxiliary thermal storage for there would be no reason to insulate it. However, this run does indicate the need for this auxiliary storage, a thermal mass that is exposed to both plenum and PCM.

Simulation run number 5 was undertaken to demonstrate the results of operating the off-peak cooling system without providing adequate storage capacity. This simulation combines the conditions of run number 2, no

PCM storage, and run number 4, an isolated concrete slab. Beyond the storage capacity problem, this run encounters the problem faced by run number 4, an inadequate thermal transfer rate during the off-peak charging period. As shown by Figure number 27, this combination of problems results in an afternoon space temperature near 80°F and a daily room air temperature swing of 14.5°F. The radiant temperatures of the space, as evidenced by node 3, also experience a wide temperature swing, further compounding the thermal comfort problems. While a system with these characteristics would make little sense, this simulation does demonstrate the relative effectiveness of the proposed off-peak cooling system configuration modelled in run number 1.

This set of simulation runs leads to several basic conclusions. The phase-change material is indeed necessary to the system. Its isothermal characteristics are responsible for the relatively small 8°F room air temperature swings in runs number 1, 3 and 4. While the sensible heat analogy used in the simulations is not strictly isothermal, this is acceptable because the surface temperature of the PCM will vary over a limited swing due to variations in the salt's internal resistance. The carpet modelled in run number 2 is acceptable, even desirable, particularly with resistance values in the 1.5 to 2.0 °F/BTUSF range. The lack of a secondary storage mass exposed to plenum and PCM will result in a reduction in the charge rate. This reduction leads to increased storage capacity requirements in the PCM. It also complicates the charging schedule, requiring additional weekend charging periods. An off-peak cooling system as initially defined in simulation run number 1, with the carpet modification from run number 2, is the most promising configuration to date. As discussed earlier, this system can benefit from an increase in set point temperature and a decrease in PCM thermal capacity. These issues, and system performance under partial load conditions, are addressed in the next set of simulations.

FINAL SYSTEM SIMULATIONS

The decision to raise the system's set point temperature will provide several operational benefits. Foremost, the dry bulb temperatures

FIGURE 26

Simulation Run No. 4,
Slab Isolated on Both
Sides,
Day No. 2.

KEY:

Room air temp.

PCM and tile temp.

Plenum air temp.

Concrete slab temp.

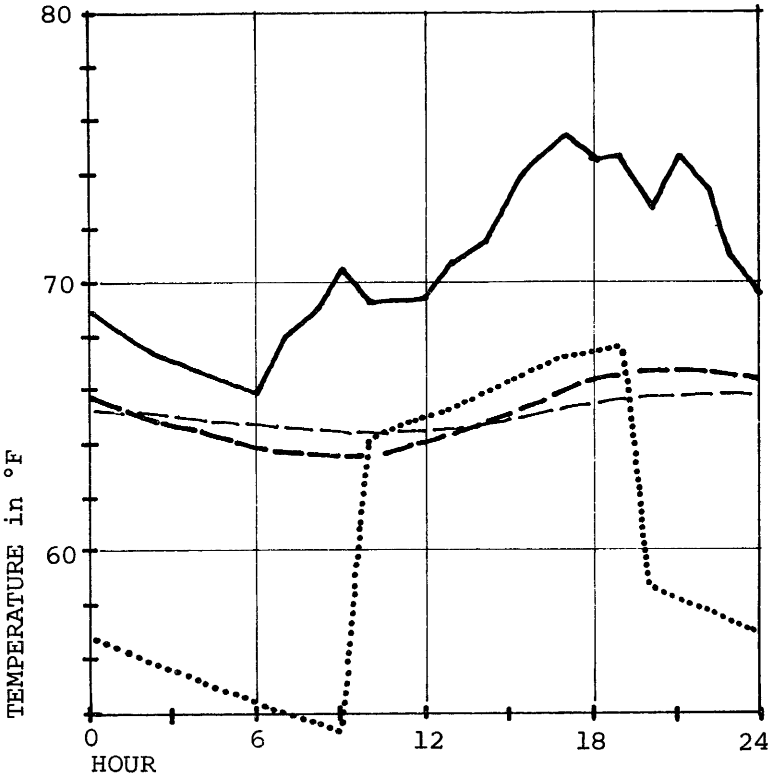


FIGURE 27

Simulation Run No. 5,
System Without PCM,
Slab Isolated on Both
Sides,
Day No. 2.

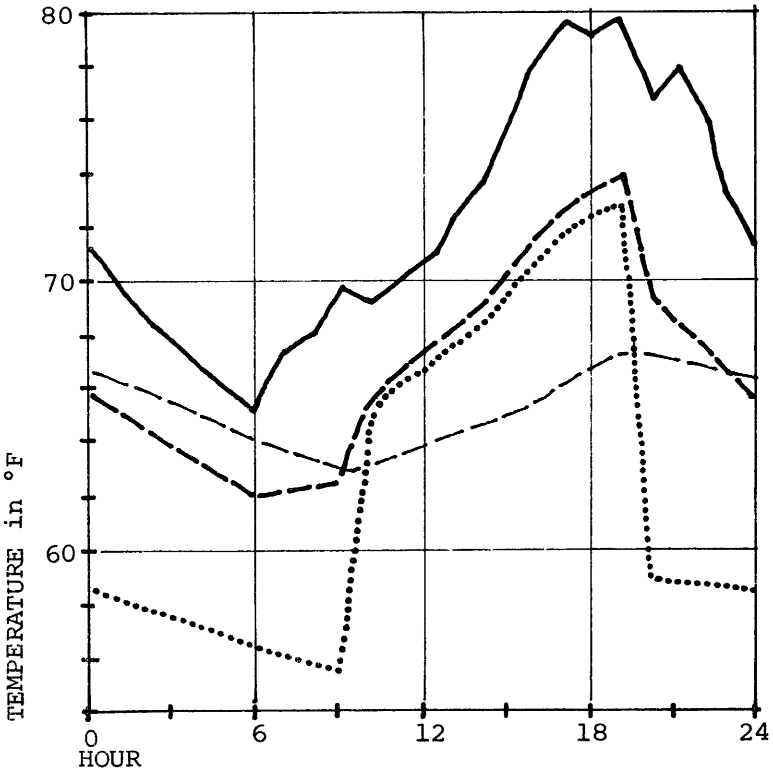
KEY:

Room air temp.

Support tile only temp.

Plenum air temp.

Concrete slab temp.



encountered in the occupied space should fall within the conditions specified for human thermal comfort. The fixed nighttime charging rate establishes a fixed Δt between the salt storage and the charging air. Raising the set point temperature will also raise the temperature of the charging air, the temperature at the evaporator, and the efficiency of the chiller. Also, the warmer operating temperatures will further reduce the possibility of condensation. Simulation run number 6 was conducted to verify the assumption that the 3°F increase in the PCM set point temperature would result in a proportional increase in room air temperature.This run includes a carpet finish on the concrete slab and can be directly compared to run number 3. These two runs differ only in the set point of the PCM. The results of the third 24 hour period of run number 6 are presented in Figure no. 28. The energy storage is summarized as follows:

TABLE NO. 9 Thermal Storage-Simulation Run No. 6

Node	Location	Capacitance in BTU/°F	Δt °F	Thermal Energy Stored in BTU/Cycle
1	Room air	353	9.7	3.424
2	Room finishes	947	6.7	6,344
3	PCM Tile	10,000	2.7	27,000
6	Concrete slab	6,603	2.4	15,847
		Total thermal storage		52,615
		Nighttime heat gain		34,820
		Total		87,435

These figures indicate that the system stores essentially the same quantities of thermal energy encountered in the previous runs that incorporate PCM.

Figure no. 29 compares the phase-change temperatures (node 3) and room air temperature (node 1) encountered in runs 3 and 6. This graph illustrates that the relationship between these two variables remains

FIGURE 28

Simulation Run No. 6, Carpet on Concrete Slab, New Set Point of 67°F., Day No. 2

KEY:

━━━━ Room air temp.

━ ━ ━ ━ PCM and tile temp.

•••••••• Plenum air temp.

──── Concrete slab temp.

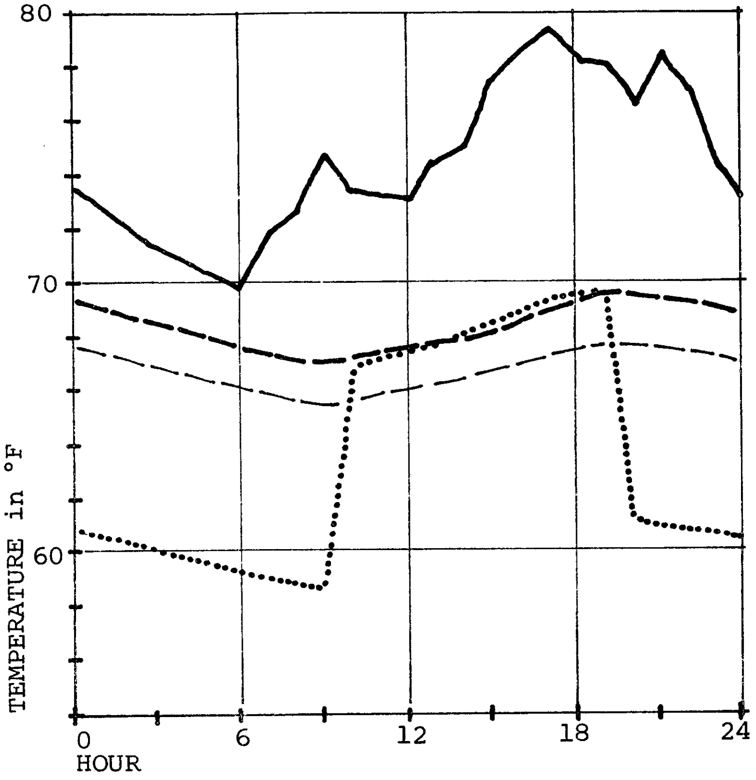


FIGURE 29

Comparison of Run No. 3. (64° Set Pt.) and Run No. 6 (67°F Set Pt.). Carpet on Slab, Day No. 2

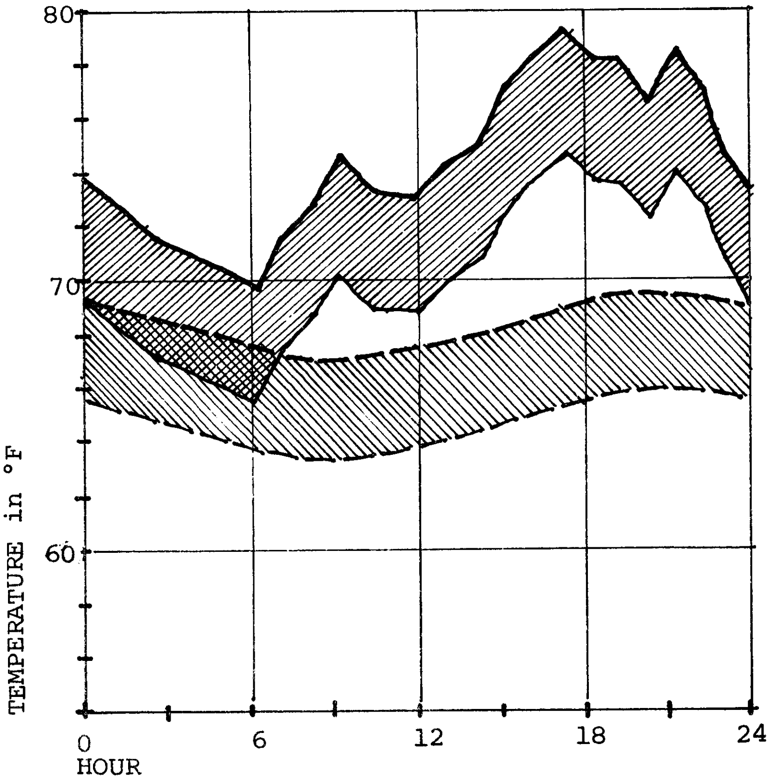
KEY:

──── Room air, Run 3

– – – – PCM and tile Run 3

━━━━ Room air Run 6

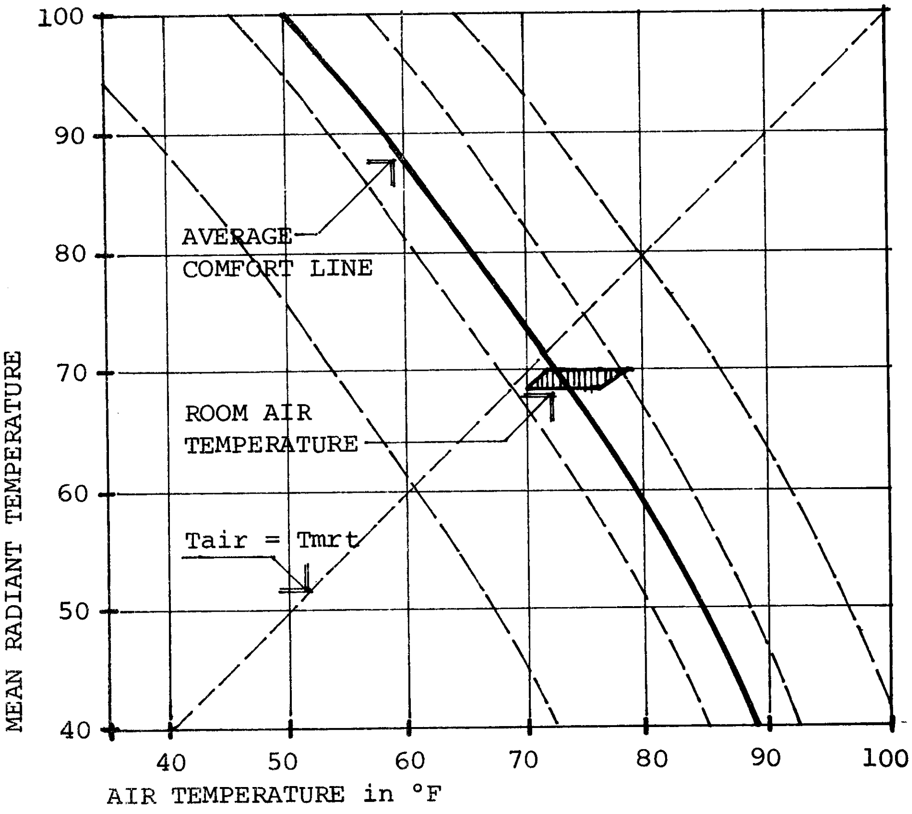
━ ━ ━ ━ PCM and tile Run 6



consistent despite the shift in set point temperature. This is because the temperature difference between the phase-change storage and the room air temperature is largely a function of the cooling loads encountered by the space, a variable that was not changed. The new set point temperature, in the context of the cooling load used in the simulations, provides room air temperatures ranging from 70°F to 79°F. These air temperatures, as plotted on Figure 30, fall much closer to the conditions specified for thermal comfort than previous runs. In conclusion, this run provides evidence that the system can fulfill a prime goal of the program. It provides conditions of thermal comfort under design day conditions with minimum use of peak period electricity.

FIGURE 30

Air Temperature from Run No. 6 Plotted on Comfort Chart 67°F Set Point Temperature.



ADDITIONAL PERFORMANCE ISSUES

Before drawing final conclusions concering the thermal performance of the system, several operational issues must be examined in greater depth. Methods for system control must be evaluated. Actual performance under partial load conditions should be examined for potential overcooling Methods for reducing the early morning overcooling should be discussed along with the implications of spot heat gains. Finally, although beyond the scope of this evaluation, the design challenges presented by exterior zone spaces will be briefly discussed.

The issue of control over the proposed off-peak cooling system becomes complex due to the nature of the scheme. The PCM thermal storage, located in the ceiling, is in direct thermal contact with the occupied space. Therefore, once the system is charged it will continue to cool the occupied space until it is completely discharged or until the space becomes as cool as the storage. Fortunately, the thermal storage system is self regulating. The use of latent heat storage provided by the storage system will vary proportionally with the Δt between the PCM and the space. As the space cools, and the need for additional cooling is reduced, the lowered Δt conveniently provides less cooling. On the other hand, as heat gains increase, the space temperature will moderately increase, raising the Δt and the amount of cooling provided. This self regulation works nicely. In the simulations, a room air increase of 9°F would quadrupule the cooling provided to the space. The interrelationship between cooling load, space temperature, and the amount of cooling provided will never allow the off-peak system to achieve the relatively constant air temperatures associated with conventional mechanical systems. It should be noted that a 9°F temperature swing is by no means extreme, particularly when transition through this range occurs at a slow pace.

There are a number of secondary control issues. The initial design of the storage system, including PCM capacity per unit area and thermal tile coverage per unit area, will establish a room temperature range for given load conditions. This design must be closely coordinated with the anticipated room cooling load. The timing of the storage charging period will affect the room temperatures experienced during early morning occupancy. When storage requires only a partial charge (following a day with light heat gain) the delay of this charge until the end of the charging period will result in warmer room air temperatures in the morning. While the daytime chiller is designed for latent heat gain only, its schedule of operation will influence the building's environment. Daytime chiller operation should be delayed as long as possible during the morning start up to avoid aggrevating the overcooling tendency. Finally, the building's heat gain itself can be modified to enhance system performance. During peak load conditions, ventilation air should be reduced to the minimum amount possible. This will allow the daytime chiller to continue at maximum output contributing to the removal of sensible gains. Building maintenance and operation schedules should be oriented toward reducing the afternoon peak heat gains. The successful performance of the off-peak cooling system will depend on proper system design and cooling load management.

The subject of system performance under partial load conditions should be examined in greater detail. Although the off-peak cooling system is designed to handle the extreme of design day conditions, it will also be operated during periods with lighter cooling loads. For interior zone spaces, the occupant related loads from electrical use, lighting and body heat gain will remain near constant through the year. On the other hand, the weather related loads of ventilation, conductance and solar gain will vary widely during the year. Simulation runs number 7 and 8 were undertaken to examine system performance under occupancy related loads only. For these runs, the exterior environment contributed no latent or sensible heat gain as would be the case in some fall and spring periods.

The resulting heat gain profile is shown in Figure no. 31. Note that unlike earlier runs, this heat gain profile has no gain during the

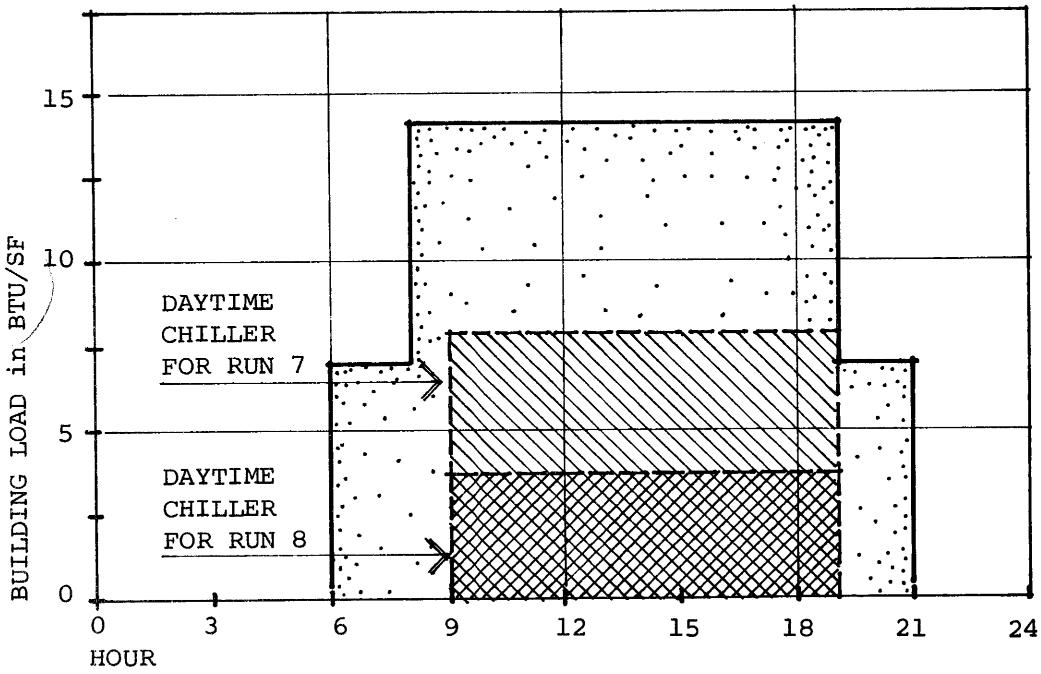


FIGURE 31 Partial Load Profile Used in Runs 7 & 8

9 hours between 9 PM and 6 AM. The daytime chiller is sized to carry all latent heat loads during design day conditions. Under partial load conditions, the daytime chiller can be operated at full capacity, and maximum efficiency to carry a portion of the sensible heat gain in addition to the occupancy related latent heat gain. This situation is modelled in run number 7. It should be noted that this method of daytime chiller operation does not increase the yearly peak demand of the cooling system. Another option is the limitation of the daytime chiller to the remaining latent loads only. This variation is represented by run number 8 in which the daytime chiller is run at one half its rated capacity. Both of these simulation runs are based on the basic system configuration used for simulation run number 1 in which the concrete slab is exposed to the plenum and the concrete space.

Simulation run number 7 demonstrates that when the daytime chiller is fun at full capacity under partial load conditions, room air temperature remains almost constant. As illustrated in Figure 32, room air remains within 1°F of 72°F during the daytime occupied period. This can be explained by the relatively constant nature of the cooling load carried by the thermal storage system. During the unoccupied portion of the day the space is subject to no heat gain. Under these conditions the room air temperature quickly approaches the 67°F storage temperature. This represents an early morning space temperature that is obviously too cool. This problem is compounded by the scheduling of the recharging period. In simulation run number 7, the partially depleted thermal storage is recharged in the early portion of the 14 hour recharging period. This practice hastens the depression of room air temperature toward the storage temperature. One might argue that the presence of no heat gain would imply that there are no occupants and therefore this midnight over cooling is acceptable. Nevertheless, the practice of scheduling the recharge period as late as possible would minimize the amount of time that the space is overcooled.

Simulation run number 8 is illustrated in figure number 33. This run represents the basic system's performance under partial load conditions with the daytime chiller operation at one half capacity. The reduction in the daytime chiller output increases the load carried by the thermal storage and therefore also increases the daytime room air temperature. Room air reaches a maximum of 75°F and the hours between 2 AM and 7 AM witness air temperatures between 68°F and 70°F. Note that the charging cycle was delayed, beginning at midnight and running until 9 AM. While this maneuver was successful in keeping late night temperatures above 70°F, the morning room air still approaches the thermal storage temperature. While this is undesirable, it must be considered unavoidable under zero heat gain conditions. As discussed earlier, the cool room air temperature experienced during the earliest hours of occupancy must be dealt with in some fashion. Hopefully, the use of a quick preheating system can be avoided. Other options include the scheduling of janitorial services with their associated internal gains, during this period; or perhaps the earliest occupants can adjust their clothing levels for the air temperatures encountered. As run number 8 approaches midnight, all nodes begin to equilibriate near 70°F. This discharged state would be appropriate for periods when occupancy was not expected during the next day. The daytime chiller could be used for the loads that did occur and the storage would be recharged in the nighttime period prior to the day of occupancy return.

The simulations run to test the proposed off-peak cooling system have been based on an idealized cooling load profile. In this model, cooling loads for the interior zone of an office building have been assumed to be moderate and uniformly distributed. This is not an unreasonable assumption for interior office spaces; however, there will be exceptions. Some office buildings have twice the 2.5 watts/SF electrical load that is used in the simulation. An inefficient building is not a good candidate for this off-peak cooling system. The cooling loads that the proposed system can handle are limited by the thermal

FIGURE 32

Simulation Run No. 7,
Partial Load Conditions,
Day No. 1.

KEY:

———— Room air temp.

— — — — PCM and tile temp.

•••••••••••• Plenum air temp.

— - — - — Concrete slab temp.

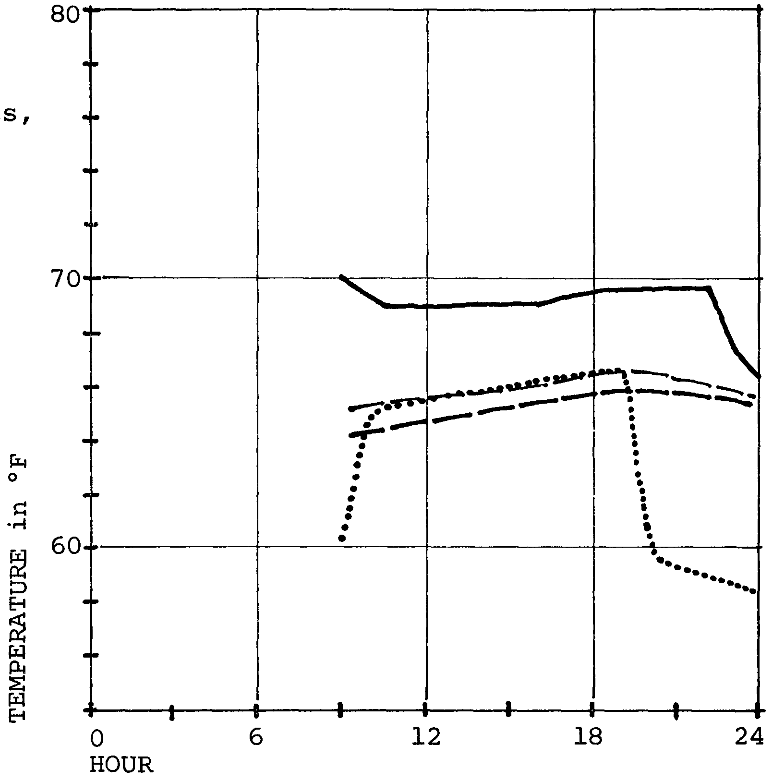


FIGURE 33

Simulation Run No. 8
Partial Load Conditions,
Day No. 2.

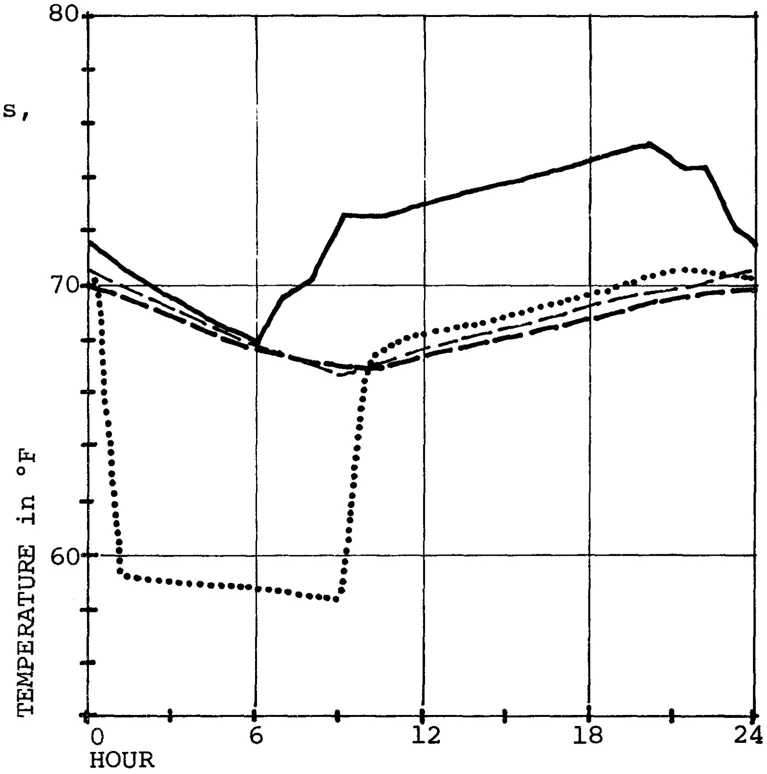
KEY:

———— Room air temp.

— — — — PCM and tile temp.

•••••••••••• Plenum air temp.

— - — - — Concrete slab temp.



transfer between the ceiling surface and the occupied space. This thermal transfer is determined by the thermal transfer coefficient, the Δt between space and storage tile, and the area of transfer surface (i.e., thermal tiles). The thermal transfer coefficient is fixed. To increase the system's Δt, by lowering the PCM set point, would aggrevate the morning overcooling problem. Finally, the 65% coverage of thermal tiles assumed in the simulation runs could theoretically be increased by 50%; however, this would prove impractical in reality. Therefore, the off-peak cooling system is best suited to reasonably efficient buildings, as represented by the Boston test case. A committment to this type of system could easily be paralleled by an effort to insure high building efficiency.

The presence of a uniform load distribution is also necessary to the proposed system's successful performance. The presence of a large "spot" load, for instance a photocopy machine, might exhaust an area's storage capacity. This could cause localized overheating at the end of the day. This type of situation must be evaluated on a case by case basis with remedial action taken in the form of additional thermal storage or explicit mechanical treatment. In this same vein, the thermal storage tiles could not be placed adjacent to flush ceiling mounted light fixtures. These fixtures might exhaust the adjacent storage tile prematurely. This should be an area for further experimentation. If ceiling fixtures are used, they could be buffered by acoustical tile. Suspended light fixtures would be preferable to flush ceiling fixtures. Indirect, furniture mounted lighting would also provide a desirable uniform heat gain from lights. If fixtures are not located in the ceiling, additional area is available for thermal tiles. The heat gains in each space must be evaluated, then the thermal storage and mechanical backup, if required, sized appropriately. The quantity and quality of cooling loads experienced in each space will determine the viability of the proposed off-peak cooling system for that space. The systems that contribute to the cooling load can be designed with requirements of the cooling system in mind.

The detailed evaluation of system thermal performance in exterior building zones is beyond the scope of this thesis. While perimeter zones will represent a more complex application of off-peak cooling, they must be addressed because they will affect the economic merit of the system. Commercial consumers are currently charged on the basis of total electrical demand. Reductions in interior zone peak period demands must be complimented by exterior zone reductions to establish significant demand savings. I would like to briefly discuss the challenges involved in applying the proposed system to exterior zone spaces. An office building will normally contain a series of exterior zones, one for each facade. An exterior zone usually includes the area from a building's exterior wall inward for about fifteen feet. The exterior zones in an office building differ from the interior zone, and from each other, in the nature of the cooling loads they face. They share the occupancy related loads and ventilation loads experienced by the interior zones. Beyond these, the exterior zones are subject to additional heat gain from solar impact and thermal conduction. The magnitude of these additional loads will be strongly affected by the orientation of the zone in question and the design of the weatherskin. Unlike the interior building zone, the exterior areas can require explicit heating in the winter due to heat loss through the exterior wall. Exterior zones are subject to wider load variations in both short and long term time frames.

The implementation of the proposed cool storage system in an exterior zone will involve two major challenges. First, the storage system must be designed to handle both the quantity and rate of the space's daily heat gains. This may prove quite a task when large solar gains are involved. The window and ceiling tile systems can be designed to minimize this type of impact by providing solar rejection at the window and perhaps absorption at the ceiling. A second, and more serious problem is encountered when a space served by the proposed thermal storage system also experiences a heating load. It is not unusual for an exterior office to require cooling to remove solar gains during the day and heating to replace conduction loss at night. This schedule of operation is not amenable to the off-peak cooling system. Obviously, charging the PCM storage at night would exacerbate the heating load concurrently experienced by the space. Fortunately, this dilemma would not occur during the mid-summer period, a time of cooling loads only. This is also the period in which the utilities experience their yearly peak demand. Exterior zone off-peak operation during only these safe summer months would still yield the benefits associated with load management. The thermal storage system should not be used when substantial heating loads are anticipated. The exerior zones will provide a more diverse and challenging area of application for the proposed system. This is definitely a subject that would benefit from further study.

REFERENCES

1. Rahme, Roy W., "Trends in Heating, Ventilating, and Air Conditioning Systems for Maximum Energy Efficiency", ASHRAE Journal, January 1975, p. 70.

2. Kohler, Joseph T. and Sullican, Paul W., Teanet Users Manual (Harrisville, New Hampshire: Total Environmental Action, 1978), passim.

3. Krieth, Frank, Principles of Heat Transfer (Scranton, Penn.: International Textbook Company, 1960), p. 208.

4. ASHRAE Handbook 1977 Fundamentals (New York, NY: ASHRAE, 1977), p. 8.2A.

5. Fanger, P.O., Thermal Comfort, Analysis and Applications in Environmental Engineering (Copenhagen, Denmark: Danish Technical Press, 1970), p. 37.

6. ASHRAE, p. 8.26.

7. Egan, M. David, Concepts in Thermal Comfort (Englewood Cliffs, New Jersey: Prentice-Hall, Inc., 1975), p. 12.

6.

Economic Analysis

Beyond thermal and architectural considerations, the proposed off-peak cooling system must also meet economic criteria. Earlier sections of this thesis described various benefits that load management, through thermal storage, might provide. The government, and our national economy would benefit through decreased dependence of foreign energy sources. The electric utilities would increase efficiency and decrease capital costs through reduced growth in peak demand. Although there is a possibility that the government might provide economic encouragement for the implementation of off-peak systems, it is more likely that the utilities will provide the major incentives. The vehicle for these incentives will probably be time-of-day rates. Current rates do not reflect the burden utilities must carry in the face of rising peak demand. Nor do they reward the beneficial aspects of the proposed off-peak system.

An economic evaluation of the system at this point in time is something of a dilemma. The major anticipated incentives for system implementation, time-of-day, rates have yet to be established. On the other hand, the current scenario faced by the electric utilities will certainly prompt rate revision of some sort. While the proposed system will undoubtedly benefit from future rate revisions, the extent of this benefit is unknown at this time, and therefore future economic performance is difficult to ascertain. The economic evaluation that follows is based on the existing conventional rate structure encountered in Massachusetts. While the proposed system will demonstrate savings, through a generalized load leveling effect, these savings are not the main event. This evaluation is presented as a reflection of the system's economic character under current conditions. Hopefully,the rate structure will soon change, allowing full economic benefit of the system to emerge.

The proposed system was evaluated under current rate structures. This study is based on the same Boston area building used for thermal load calculations in earlier sections This building offered detailed records of mechanical system operation and gross electrical consumption. In review, the building is a commercial office structure in the Boston Metropolital area. This building has 757,000 SF of conditioned space on six floors. The occupancy represents a typical range of speculative office tenants from insurance agencies to computer firms. The building is arranged about a large central court. Exterior walls have a moderate ratio of glazed area to solid wall, with fenestration minimal on the southern exposure. Building management has actively pursued programs aimed at reducing energy waste, with moderately successful results. This building was selected for this study because it typifies the type of project well suited to off-peak cooling.

The building under study is serviced by Massachusetts Electric Company. The billing for power consumed is under commercial rate H, effective 8 June 1979. This rate is outlined as follows:

TABLE NO. 10. Massachusetts Electric Company Commercial Rate H

Demand Charge:

$830.00	for the 1st 500 kw or less ($1.66/kw)
1.57	per kw for the excess

Energy Charge per kwh:

$.02648	for the 1st 50,000 kwh
.02350	for the next 50,000 kwh
.02043	for the next 100,000 kwh
.01927	for excess of 200 H.U. per kw demand
.01464	for excess of 300 H.U. per kw demand
.01362	for excess of 400 H.E. per kw demand
.01300	for excess of 500 H.U. per kw demand

Fuel Adjustment Charge:

$.01858	per kwh

Calculation of
Incremental Energy Costs

The energy charge under Rate H depends upon the number of hours use of the billing demand for the month, or the electrical consumption for the month divided by the billing demand. An examination of recent consumption and demand data indicates that this figure will tend to be close to 400 hours use. Assuming that the value will be above this figure one half the time, the cost would average:

$.01464/kwh + $/01362 ÷ 2 = $.01413/kwh

Adding the fuel adjustment charge the total cost would be:

$.01413/kwh + $.01858/kwh = $.0327/kwh

Calculation of Incremental Demand Cost

An increase in peak demand results in both a direct demand charge and an increase in the energy charge. This increase in the energy charge represents an implicit demand charge which must be accounted for in determining the cost of demand.

If consumption falls between 300 and 400 hours of use of demand, for each added kilowatt of demand, 200 kwh are shifted into the $.01464 block. The result is a cost of:

$$200 \text{ kwh/kw} \times \$.02043/\text{kwh} + 100 \text{ kwh/kw} \times \$.01927/\text{kwh}$$

$$- 300 \text{ kwh/kw} \times \$.01464/\text{kwh} = \$2.40/\text{kw}$$

If consumption falls between 400 and 500 hours use of demand, the cost by similar calculation is $2.80/kw. Using the average of these two figures and adding the direct demand charge results in a total of:

$$\$2.60/\text{kw} + \$1.57/\text{kw} = \$4.17/\text{kw of demand}$$

All savings calculations will be based on the existing Mass. Electric Rate H. This rate is typical of existing rate structures in the United States. During months when monthly peak demand is less than 80% of the yearly peak demand, the 80% of the yearly peak demand is used to calculate demand charges. This is known as a peak "ratchet".

Potential Areas of Savings Under Existing Rates

In examining the application of cool storage with phase-change thermal storage to this type of building, several potential areas of saving are apparent. These are listed below and then discussed in greater detail.

A. Reduced demand charges resulting from the use of off-peak electricity for chiller and air handling unit operation.

B. Increased chiller efficiency resulting from operation under design load conditions.

C. Increased chiller efficiency resulting from more favorable exterior ambient conditions during off-peak periods.

D. Increased use of economizer type operation modes enabling nighttime ambient conditions to meet daytime loads without mechanical chilling.

E. Lower initial equipment cost for mechanical system components including chillers, condensing units, cooling towers, pumps, piping, etc.

In order to explore these potential savings, the following chiller operation strategy is assumed. One 400 ton chiller will be operated during the day to satisfy latent cooling loads and thereby prevent condensation. This same chiller also tempers incoming ventilation air. The air handling units will be run to provide ventilation air only during the day, providing an air flow of .1 cfm per SF to the occupied space. This mode of operation will run approximately 10 hours per day. During the largely unoccupied off-peak periods, the 400 ton chiller will be augmented, as required, with an additional 500 ton chiller to freeze the cool storage. This will be accomplished by cooling the interstitial space between the ceiling plane and the structure to 55°F. This chilling cycle enables the ceiling storage to absorb sensible cooling loads during the night and through the next day. Chillers will be operated during this period only as required to recharge the cool storage. Chiller operation will always be at 100% of the units rated output. Air handling units will be run at approximately 1.8 cfm per SF during this period.

A. Reduced Demand Charges

The use of off-peak cooling enables a commercial structure to reduce peak power consumption levels. Chiller and air handling unit loads represent between 40% and 50% of the peak summer loads. Historically, air conditioning systems have had to generate cooling capacity to meet building cooling loads as they occur. With the use of cool storage, this cooling capacity can be generated during off-peak hours when the power consumed will not contribute to the afternoon peak loads. The exception to this operation strategy is the proposed 400 ton chiller used to meet daytime latent cooling loads. Thus, in lieu of the 1560 ton chiller presently operated during the day, we have only the proposed 400 ton chiller. The reduction of peak demand occurs directly during each of the six months that the chiller is operational. In addition, the peak demand cost is reduced during winter months by eliminating the chiller component of the peak "ratchet". This in effect returns the actual peak demand as the pricing index during the winter. Air handling units at the existing building currently move approximately 1 cfm/SF (.2 cfm of outdoor ventilation air and 0.8 cfm of recirculated air). Peak demand is also lowered by the reduction of this daytime air handling quantities from the present 1 cfm per SF to the proposed .1 cfm per SF.

Savings are calculated by substituting the 336 kw maximum demand of the 400 ton chiller for the peak monthly demand of the 1560 ton chiller. The air handling peak load is reduced from 856 kw to approximately 300 kw. The following table summarizes these changes based on 1977 data and assigns savings based on the aforementioned $4.17 per peak kw saved.

TABLE NO. 11 Savings Due to Peak Load Reduction

Month	Normal Demand Peak kw	Peak Reduction kw	Savings @$4.17/kw $
Jan.	(3776)†	(656)	2735
Feb.	(3776)	(496)	2068
March	4240	0	0
*April	4560	1224	5104
*May	4720	1440	6004
*June	4800	1484	6188
*July	5520	1484	6188
*Aug.	4880	1484	6188
*Sept.	5520	1476	6155
Oct.	4560	0	0
Nov.	(4416)	(336)	1401
Dec.	(4416)	(896)	3736
			$ 45,676 Saved

*Months chiller is operational

†Bracketed values indicate ratchett clause governs

B. Increased Chiller Efficiency Due to Operation at Design Load

The 1560 ton chiller presently used in our test building was designed to cool the building under worst case (2 1/2%) conditions. Because these conditions occur only during a few days of the summer, the chiller must operate at partial capacity the majority of the time. A chiller has its highest efficiency when operating at 100% with efficiency dropping as the load on the chiller is reduced. Figure no. 34 shows the efficiency curve for the existing 1560 ton chiller; maximum efficiency at 100% load is 0.84 kw/ton.

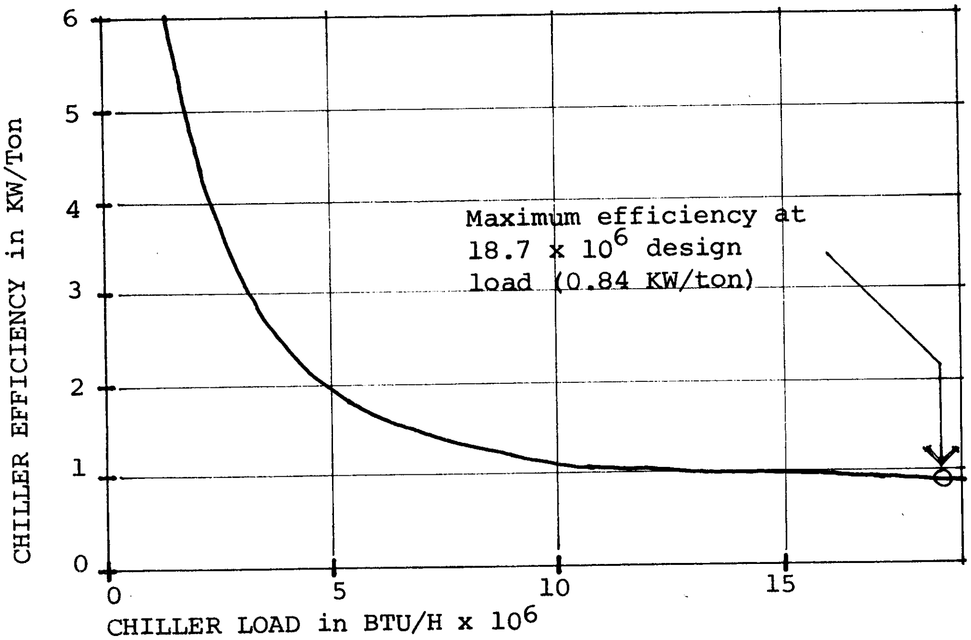


FIGURE 34 Chiller Efficiency vs. Load in Monitored Building

The chiller involved in charging the proposed cool storage system can be run at 100% capacity until the cool storage is fully charged. Thus losses in efficiency caused by partial load conditions do not exist.

To establish savings in this category, the chiller logs for the 1978 cooling season were examined. Pressure and temperature drops across the chiller were used to establish the cooling capacity provided by the chiller. The actual electrical consumption in providing this cooling capacity was compared, hour by hour, with the projected electrical consumption had the chiller been run at its maximum efficiency (0.84 kw/ton). The difference in these two electrical consumption rates was assigned to savings. This information is summarized below by month.

TABLE NO. 12. Savings Due to Design Load Chiller Efficiency

Month	kw Saved	Cost @ $.0327/kw
April	5,112	167
May	99,433	3,252
June	150,024	4,906
July	228,970	7,487
August	269,587	8,816
September	73,783	2,413
Total	826,909	27,041

Note: The 1978 cooling season was close to statistical normal for Boston

C. Increased Chiller Efficiency Due to More Favorable Exterior Ambient Conditions

In the off-peak cooling scheme the bulk of chiller operation is shifted from daytime periods to nighttime periods. The lower outside wet and dry bulb temperatures at night are more favorable for the condenser circuit cooling mechanism, whether cooling tower or spray pond. The lower condenser water temperature results in increased efficiency. From consultation with sales personnel with Baltimore Air Coil a range of 5% to 10% increase in efficiency is deemed appropriate.

To calculate the savings, the present daytime chiller operation is summed and reduced by a factor of 0.8 to determine nighttime chiller operation under the off-peak scheme. A 7.5% increase in efficiency is applied and this is converted to dollars. This results in a savings of $1,415.

D. Increased Use of Economizer Type Operation

The economizer type cycle is a common feature in large cooling systems today. This type of operation uses outdoor air as a source of cool, in lieu of a chiller, when outdoor conditions permit. Without cool storage the outside conditions must be favorable at the time of the cooling need (i.e., during the day). However, with cool storage, the outdoor air during cooler nighttime periods can be used to meet daytime loads.

To calculate the availability of this option, each day of chiller operation in 1978 was examined for occurrence of outdoor temperatures below 55°F. The chiller operation was eliminated, and savings assigned, if 15 hours of outside temperature below 55°F occurred. If less than 15 hours were available, the chiller operation was reduced proportionally. The results of this analysis are summarized below by month. Savings resulting from daytime operation of the economizer cycle apply to both existing and proposed systems and therefore do not contribute to the incremental off-peak savings.

TABLE NO. 13 Savings Due to Expanded Economizer Operation

Month	kw Saved Economizer	Cost @ $.0327/kw in $
April	7,806	255
May	6,935	267
June	8,428	276
July	0	0
August	1,293	42
September	10,000*	327
Total	34,462	$1,127

*Estimate, data unavailable

E. Lower Initial Equipment Cost

The off-peak cooling scheme would result in savings due to a reduced initial investment in cooling equipment. The off-peak scheme requires 900 tons of cooling capacity rather than the existing figure of 1560 tons.

Reductions in chiller size would also be reflected in smaller cooling towers, chilled water lines, pumps, etc. However, the air handling units would have to be increased to provide the 1.8 cfm/SF air flow during the charging period. In addition, the outside air intakes require a run around heat exchanger. Interviews with sales personnel with York, a manufacturer of cooling system components, established a probable savings in the order of $60,000. If this is amortized over an arbitrary 20 year period the savings per year is $3,000.

SUMMARY OF SAVINGS

A.	Reduced demand charges	$45,676
B.	Increased efficiency 100% load	27,041
C.	Increased efficiency ext. conditions	1,415
D.	Additional economizer operation	1,127
E.	Initial equipment savings	3,000
	Total	$78,259

These savings would be used to amortize the incremental investment required by the cool storage system. This investment includes the cost of the storage material and instrumentation/control equipment. The tile and salt module might be expected to cost $2.40 per SF minus a $.30 credit for the acoustical tile it replaces. Assuming a 60% coverage for the entire building, this cost equals $1.26 per SF of building area or a total investment of $953,820. The resulting straight line payback would be 12.2 years. It should be noted that this payback is conservative due to two factors. First, the savings are calculated using the present electrical rate structure. While this rate favors a "flat" load ratio it is predicted that incentives for off-peak use in future rate structures will be even more attractive. Second, this report examines the economics of an application in Boston, Massachusetts. Savings are expected to be greater in climates with a larger cooling load. While savings will vary with regional climates, building envelope design and regional pricing mechanisms will also have major impacts. This economic analysis indicates moderate savings for the proposed system under existing rates and the less favorable Boston climate. The mature economic character of the proposed system will develop with upcoming utility pricing policies. These will inevitably lead to a much more attractive amortization rate.

7
Conclusion

As an issue the use of energy has grown from obscurity to become a major global concern. The United States is faced with a scenario of rising prices and dwindling supply that mandates a domestic policy of rational energy consumption. A rational energy policy will involve many methods and a wide range of programs. This thesis examined the potential for improving energy utilization in the nation's electric utilities. Through the use of load management techniques, the load profile of a utility can be manipulated. This manipulation provides benefits in several forms. A larger portion of the utilities' load can be met by relatively efficient baseload plants. Generating capacity expansion is reduced providing economic benefits to the utility and reducing harmful environmental impact. The consumer ultimately benefits from reduced utility costs. Load management can be implemented through the vehicle of peak period pricing. This pricing scheme encourages the displacement of peak loads to off-peak periods. In many cases, this displacement will involve storing energy rather than rescheduling the ultimate use of the electricity involved. While storage of electricity is impractical at this time, there are several systems for storing thermal energy. Load management techniques are ideally suited to air conditioning. This load contributes to a utility system's annual peak load and it can be satisfied by stored thermal energy. Off-peak air conditioning, if widely adopted, can provide major benefits to utility, consumer, environmentalists and government concerns.

The prime goal of an off-peak air conditioning system is the maintenance of thermal comfort with a minimal use of peak period power. There are many approaches to the design of an off-peak air conditioning system. Thermal storage can involve sensible or latent heat and appear in centralized or decentralized geometries. The proposed system combines a decentralized geometry with the latent heat storage of a sodium **sulphate** based phase-change material. This system was developed for implementation in a commercial office structure. This building type is appropriate for several reasons. Commercial structures commonly experience cooling loads throughout the year due to high internal gains. Research indicates that public acceptance of off-peak cooling at a residential scale will require precedent in public buildings. Finally, commercial buildings normally require complex mechanical systems and can easily absorb the incremental cost of thermal storage. With these factors in mind, an off-peak cooling system was designed for implementation in commercial office structures.

The proposed cooling system uses off-peak power, and conventional mechanical equipment, to recharge a thermal energy storage system. The thermal storage medium is a sodium sulphate based phase-change material enclosed in small, thin bags. These bags are distributed through the ceiling plane of the building; supported by special ceiling tiles. At night the PCM is charged by chilling the plenum space above the tiles. During the day, the PCM in direct thermal contact with the occupied space, removes sensible heat gain by melting. Latent heat gains are removed with the operation of a small daytime chiller. This system has several strengths relative to other proposals involving centralized storage and sensible heat systems. The isothermal PCM provides a self-regulation feature due to its direct thermal contact with the occupied space. This direct thermal contact also results in a significant reduction in the use of air handling units during peak periods. Latent heat storage is relatively lightweight and as a modular ceiling component the proposed system is easily installed or replaced. The mechanical system used to recharge the thermal storage is composed of conventional equipment. This system is always operated at maximum mechanical efficiency; the chillers operate at design load and have conventional evaporator temperatures. In addition, the mechanical plant is sized smaller than a conventional air conditioning system for the same load. The proposed system offers a relatively simple method of reducing peak power consumption.

Simulation of the performance of the proposed off-peak cooling system provided evidence that the concept was viable. The system fulfills the prime goal of maintaining thermal comfort conditions. The scheduling of air handling units and daytime chiller operation minimizes peak period power consumption. Simulations indicated that the set point of the phase-change material should be set at 67°F. This set point should not cause condensation problems. Laboratory investigation established the groundwork for this new mixture. Room temperatures in the final system configuration cycled from 71°F to 79°F under design load conditions. Partial load conditions produced a reduction in temperature fluctuation. With the exception of the ceiling, a wide range of room finishes are acceptable, including carpeting. This thesis was limited in scope to the simulation of interior zone spaces. Further investigation is warranted regarding system performance under other conditions.

Economic analysis indicated that the system would provide moderate benefit under existing utility commercial rate structures. These benefits are related to improved mechanical efficiency and customer load leveling. Major economic benefit can be expected with the future adoption of time-of day pricing structures in the United States. European precedent indicates that time-of-day pricing is a promising load management technique. In conjunction with this pricing scheme, the proposed off-peak cooling system offers a valuable tool for displacing peak electrical loads. This is one step toward a more rational use of our valuable energy supplies.

APPENDIX A: Preliminary Laboratory Investigation of Sodium Sulphate Compounds for Cool Storage

The off-peak cooling system described in this thesis requires a substantial amount of thermal storage. This storage, located in the ceiling plane of the conditioned space, is slated to be a phase-change material. Latent heat storage in a PCM provides the lightweight, isothermal characteristics necessary to this system. Sodium sulphate decahydrate was selected as the base material. Although the phase-change temperature of Na_2SO_4 is normally 88°F, it can be lowered by adding various compounds. This type of modification produced a 73°F set point mixture that was used in MIT's Solar Building 5. This project also saw the development, and testing of methods that minimized the historical drawbacks of Na_2SO_4 as a thermal storage material. These methods included the addition of a thickener in conjunction with novel packaging techniques to prevent stratification. This Appendix describes preliminary labwork undertaken to:

1. Determine if the set point could actually be reduced to the temperature required for cool storage (initially 64°F, later revised to 67°F).

2. Determine if the mixture's heat content would remain stable over time at a level adequate for system operation. (This heat content should be roughly that of the Solar 5 mixture, 30-35 BTU/lb including the sensible heat of a 5°F temperature swing.)

3. Determine if the Solar 5 methods would again prove successful in minimizing supercooling and stratification. This early laboratory work was an important step in establishing the viability of the proposed system. Obviously, without an adequate storage material, the system could not function.

Experiments were run using three different compounds as eutectic agents. Sodium chloride (NaCl), the compound used for the Solar 5 mixture was tested in a range of quantities. In addition, lithium carbonate (Li_2CO_3) and ammonium chloride (NH_4Cl) were examined. These two candidates were among nine compounds examined in a preliminary review by a student group in MIT's Chemical Engineering Department. Each candidate mixture was

was tested to determine its set point temperature. If this initial test indicated a promising candidate, further tests were conducted to determine heat content.

To investigate set point temperature, a batch of each candidate was mixed and tested using the following procedure:

Step 1: Place 224 grams of water @85°F in container, add 16 grams of Cab•O•Sil M-5 and blend until well mixed.

Step 2: Mix in dry container 12 grams of borax, 176 grams of sodium sulphate, and the eutectoid under investigation. Remove all lumps.

Step 3. Add dry chemicals to Cab•O•Sil solution and mix thoroughly until uniform.

Step 4: Place approximately 80 grams of this mixture in a small paper cup. Embed a dry bulb thermometer in this sample and place in a 51.8°F environment (refrigerator).

Step 5: Monitor and record mixture temperatures at fixed intervals as salt cools.

Step 6: If mixture shows promising plateau, save a portion of the sample in a sealed 12 millileter glass centrifuge vial.

By examining the cooling rate of the salts while exposed to the 51.8°F refrigerator environment, the phase-change plateau becomes evident. Several candidate mixtures are summarized in the following table:

TABLE A1. Potential PCM Candidates

No.	Eutectic Agent	Quantity of Agent in grams	Approximate Phase-Change Plateau
1	NaCl	44.8	68°F
2	NaCl	48.0	66°F
3	NaCl	51.2	65°F
4	NaCl	52.8	64°F
5	NaCl	54.4	63°F
6	NaCl	64.0	None
7	NH_4Cl	44.0	61°F
8	NH_4Cl	51.2	58°F
9	Li_2CO_3	70.0	>86°F

These results show that the desired set point temperature can be achieved with both NaCl and NH_4Cl. The results for the more promising candidates, numbers 2, 3, and 5 are shown in the following graphs.

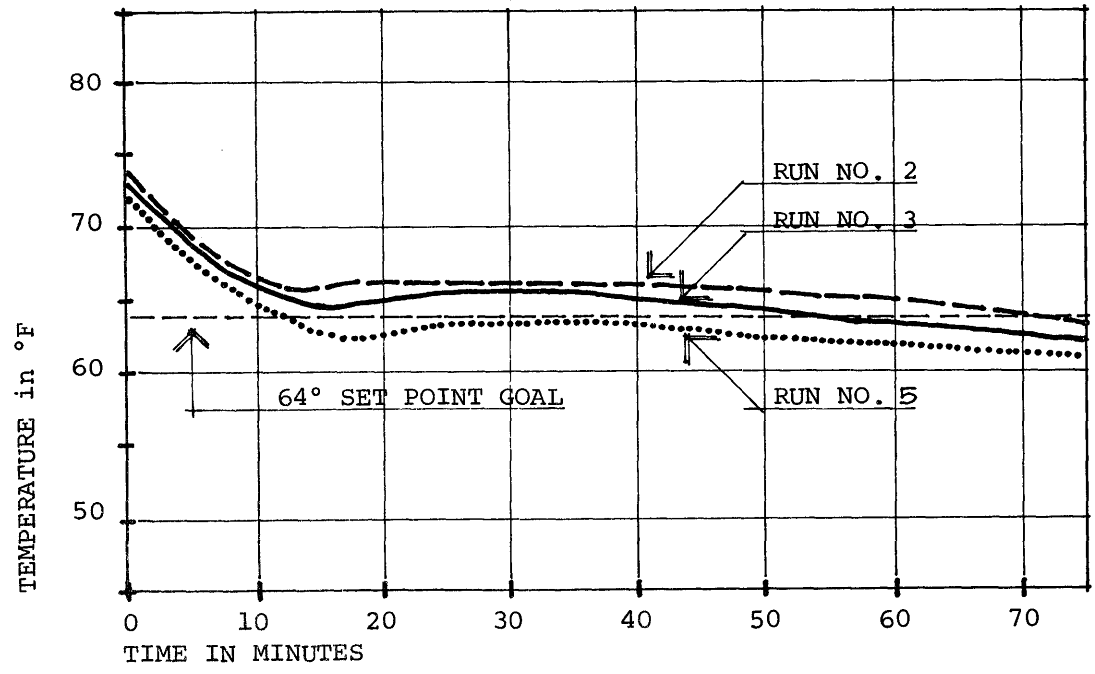


FIGURE 1A Cooling Profile for Samples 2, 3, & 5 Exposed to a 51°Ambient Environment

Figure A1 shows the temperature of each sample as it is cooled in the refrigerator. Each curve shows a characteristic minor supercooling dip before the temperature stabilizes at a set point plateau. The NH_4Cl mixture has a set point below the desired 64°F yet it also has the lowest mass of any compound tested. It is assumed that the set point can be raised by decreasing the proportion of NH_4Cl used in the mixture. Minimizing the mass of additives in the mixture should result in increased heat content. Sample number 4, 52.8 grams of NaCl, is shown in Figure A2. This sample was monitored during the cooling cycle and during the heating cycle to determine if the plateau was consistent between cycles. The results show a separation of approximately 1°F to 2°F indicating a stable relationship.

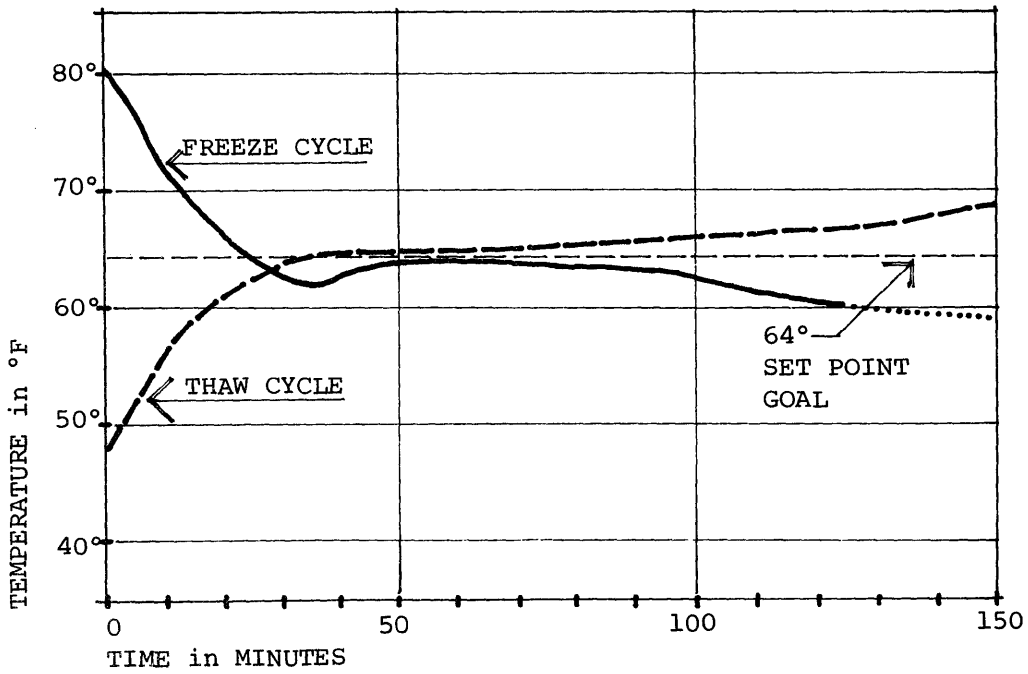


FIGURE A2 Cooling and Warming Profiles for Sample Number 4 Exposed to 55°F and 75°F Ambient Temperatures.

Samples from runs 3, 5, and 7 were examined for heat content using the following procedure.

Step 1: Precool three thermos jugs, bottle stoppers, and enough water to fill the same to approximately 59°F.

Step 2: Warm salt sample in 12 ml vial, sensible control in 12 ml vial and control 12 ml vial to approximately 74°F using a warm water bath. Insure that salt sample is completely thawed.

Step 3: Simultaneously immerse the salt vial, the sensible control vial, and the control vial in separate thermos jugs; fill jugs with chiller water and seal.

Step 4: Mildly agitate jugs at 5 minute intervals until temperature in salt thermos has stabilized.

Step 5: Record the water temperature of each thermos jug. Measure the mass of the water in each jug. Measure the mass of each vial.

Step 6: Cycle samples through 4 freeze-thaw cycles per day for future tests.

Step 7: Calculate the sensible and latent heat stored using the following relationships.

The latent heat content of the salt mixture is determined by calculating the total heat absorbed by the water from the salts and subtracting the heat absorbed by the water from the sensible control vial. This vial is filled with a non phase-change salt with approximately the same mass and specific heat.

$$\text{Latent heat content of salt sample} = (\Delta t_{\text{water salt thermos}} - \Delta t_{\text{water control thermos}})(1.8°\text{F}/°\text{C})$$
$$\text{X(mass of water in grams/grams per pound)(1 BTU/lb°F)}$$

$$\text{Latent heat absorbed per pound salt} = \text{(latent heat content salt sample/grams of salt)(454 grams/lb)}$$

Sensible heat content is calculated by comparing the Δts of the thermos containing the sensible control vial and the thermos containing an empty glass vial (which accounts for the heat content of the glass). Sensible heat was in the range of 7 BTU/lb over a typical 10°F Δt. The results of these tests are summarized below.

TABLE NO. A2 Heat Content of the Most Promisin Mixtures

No.	Eutectic Agent	Grams of Agent	Total Heat Content in BTU/lb After Cycle			
			0	50	100	150
3	NaCl	51.2	N/A	26.8	20.7	20.1
5	NaCl	54.4	N/A	28.2	22.4	21.9
7	NH_4Cl	44.0	30.7	N/A	N/A	N/A

These results indicate a heat content that is lower than that encountered in the Solar 5 mixture. The salts using NaCl as an eutectic agent also demonstrate a reduction in heat content as the samples undergo additional cycling. Intuitively, one might expect the mixtures' performance in heat content to decrease as the proportion of additives, including eutectic agents, increases. However, these tests do indicate that a phase change material with the proper set point is possible. In addition, while the heat content is lower than desired, it is still significant. Latent heat over a 10°F swing is 5 times the sensible heat content. At this stage further investigation is underway by the Cabot Corporation.

APPENDIX B: Convective Heat Transfer in a Low Δt, Low Air Speed Environment

Adequate thermal transfer characteristics are mandatory for the success the proposed off-peak cooling system. The issue of thermal transfer becomes acutely important during the recharge cycle of the PCM storage. This recharge is accomplished by circulating chilled air through the plenum above the PCM bags. Heat transfer during the charge cycle can be defined as follows:

$$Q_{charge} = \text{Area} \times \Delta t \times \text{Time} \times hc \quad (1)$$

where:

Area = The surface area of the PCM storage bags exposed to the plenum in SF. This value has been assumed to be 65% of the floor area of the room involved.

Δt = The difference between the PCM temperature and the average charging air temperature in °F. The PCM set point temperature has been defined at 67°F. The supply air during the charge period will be near 55°F. A decrease in supply temperature might result in lower chiller efficiency and condensation problems.

Time = The charging period in hours. Taken to be 14 hours for this study.

hc = The convective transfer coefficient in BTU/hr°FSF. This value is the subject of this Appendix. Preliminary indications suggest a value near 1.0 is desired.

It can be seen from equation 1 that total heat transfer during the charging period will vary in direct proportion with hc. This makes the value of hc a sensitive issue.

A literature search was conducted to establish a method of predicting the convective transfer coefficient for the recharge cycle. Several methods were found for establishing a forced convection coefficient. The majority of these appeared to be curve fits from empirical data. In addition, another method was found for determining hc when natural convection prevails. Each of these methods was evaluated for a range of air velocities in the context of recharging the plenum space. The variables for these calculations include:

1. Thermal conductivity (k) = .01466 BTU/hrSF°F
2. Fluid density (p) = .07633 lbm/fthr^{3}
3. Dynamic viscosity (μ) = .04339 lbm/ft
4. Prandtl number (N_{pr}) = .712
5. Temperature difference (Δt) = 10°F
6. Fluid velocity (v) = varies ft/sec
7. Characteristic length (L) = 2'0"

These variables apply to 60°F air. The characteristic length of 2'0" is used for a variety of reasons. Dr. Peter Griffith of MIT suggests 2 x plenum height for the characteristic length. For an eighteen inch deep plenum, this would yield an L of 3'0". However, Meinel and Meinel quoted Taber as recommending a maximum value of L = 60 cm ≈ 2'0" for convecting into free space. Each of the following methods is evaluated for a range of velocities. The results of these evaluations are graphed for comparison. The first graph, Figure B1, illustrates the values of hc for V = 0 fps through 50 fps. The second graph, Figure B2, covers the values for V = 0 fps through 2 fps in greater detail. Several of these methods relate a dimensionless Nusselt number to hc through the following relationship:

$$hc = (N_{nu})(K/L)$$

where:

hc = convective transfer coefficient (BTU/hr°FSF)

N_{nu} = Nusselt number

K = thermal conductivity (BTU/hr°Fft)

L = characteristic length (ft)

Calculation Methods

A. From: Meinel and Meinel, Applied Solar Energy, Reading, Massachusetts, Addison-Wesley Publishing Co., 1976, p. 346

This method is attributed to Duffie and Beckman.

$$hc = N_{nu} K/L$$

where: N_{nu} = Nusselt number dimensionless

K = thermal conductivity (.01466 BTU/hrft°F)

L = characteristic length (2 feet)

$$N_{nu} = 0.0158 (N_{RE})^{0.80}$$

B. From: Meinel and Meinel,

This method is attributed to Kreith

$$N_{nu} = 0.023 (N_{RE})^{0.80}$$

C. From: Meinel and Meinel,

This method is attributed to Welty (for RE: = 40-4000)

$$N_{nu} = 0.170 (N_{RE})^{0.47}$$

D. From: Meinel and Meinel

This method is attributed to Welty (for N_{RE} = 4000-40,000)

$N_{nu} = 0.062\ (N_{RE})^{0.62}$

E. From: Chapman, Heat Transfer, The Macmillan Company, New York, 1967, p. 330.

This method is also found in Kreith, Principles of Heat Transfer, 1960.

$N_{nu} = 0.036\ (N_{REL})^{0.80} (N_{PR})^{1/3}$

F. From: McAdams, Heat Transfer, 2nd Ed, McGraw Hill, 1942, p.207

This method is also found in Meinel and Meinel as (hc - 1.0 + 0.304v when v is in mph) attributed to Hottel and Woertz

$hc = 1.00 + 0.33v$

where v is air velocity in miles per hour

G. From: Private conversation with Norman Saunders, PE. This is his method as published in his book Basics.

$hc = 4 + 4v$

where: hc = convective transfer coefficient in watts/sq.meter/°C

v = velocity in meters/second

H. From: Meinel and Meinel, p. 347

This equation is a linear function fitted to empirical data collected for flat plate collectors by Hottel and Woertz

$\underline{hc = 0.000136\ (1 + 0.0068v)}$

where: hc = convective transfer coefficient in cal/cm^{2}sec°C
v = velocity in cm/second

I. From: Kreith, Frank, Principles of Heat Transfer, Scranton, PA, International Textbook Company, 1960, p. 268.

$\underline{N_{nuL} = 0.664\ (N_{REL})^{.5}\ (N_{PR})^{1/3}}$

J. From: Krieder and Krieth, Solar Heating and Cooling, New York, Hemisphere Publishing Corp., p. 225

for situations when v x L is less than 15

$\underline{hc = 0.35(v/L)^{.5}}$

K. From: Krieder and Krieth,

for situations where v x L is greater than 15

$\underline{hc = 0.54(v^4/L)^{1/5}}$

L. From: ASHRAE, Handbook of Fundamentals, New York, 1977, p. 215.

A method for vertical surfaces with v<16 fps at room temperature

$\underline{h = 0.99 + 0.21v}$

M. From: Chapman

for natural convection horizontal surface, heat flow up turbulent. This formula is found in almost all texts

$\underline{h = 0.22(\Delta t)^{1/3}}$

FIGURE B1

Comparative Evaluation of h_c Using Methods Described in this Appendix.

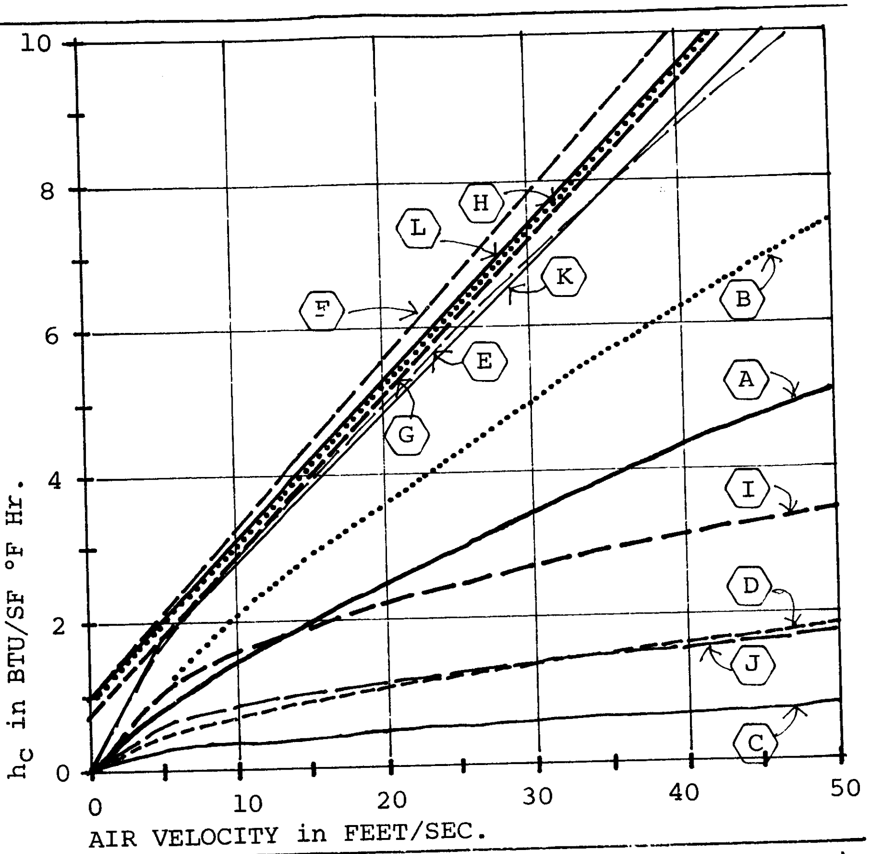
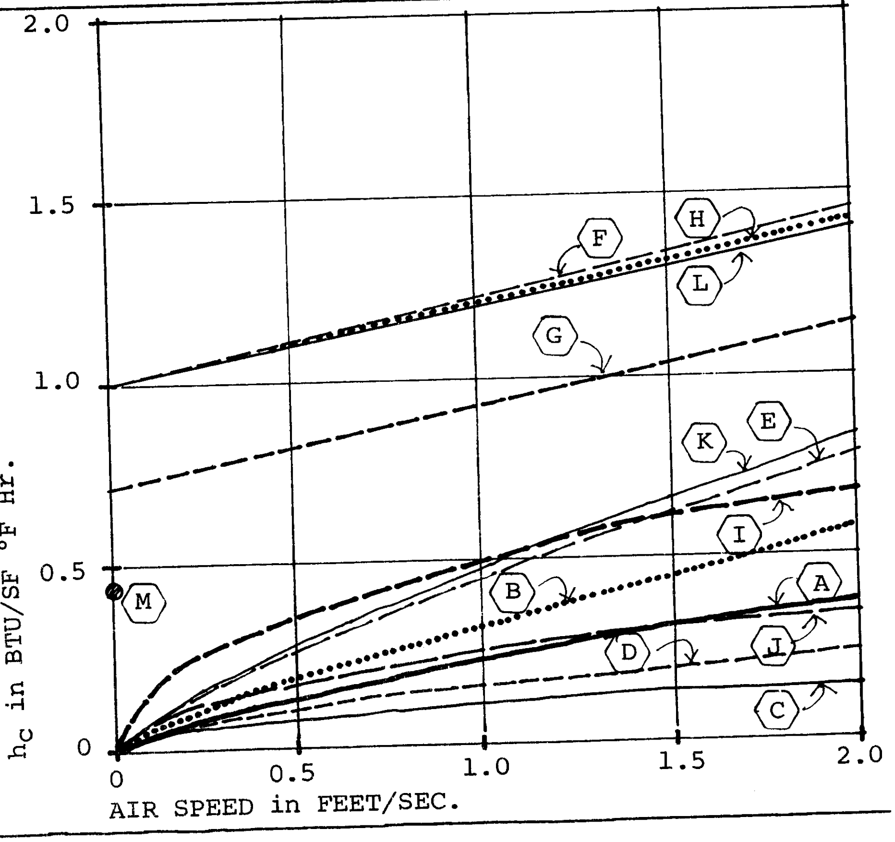


FIGURE B2

Comparative evaluation of h_c at Low Air Speed Using Methods Described in this Appendix.



The results of the literature survey, and the preceding calculations are inconclusive due to the wide range of convective transfer coefficients that result for a given air velocity. The evaluations for hc at velocities above 10 fps, and above the velocities expected in this application, show greater consistancy than the evaluations at lower velocities. Dr. Peter Griffith suggests that the values derived for natural convection (.47 BTU/hr°FSF @ 10°F Δt) should be considered a minimum value. Yet, all but four of the methods evaluated yield a value of hc = 0 for 0 fps velocity. This discrepancy is indicative of the complexity of calculating hc in a low air velocity context. Essentially, the application under investigation falls in a shady area between natural and forced convection. Intuitively, the presence of a forced air flow, albeit slow, will increase hc above the level expected for natural convection alone. The extent of this increase is unknown at this time. Should the value of hc fall short of the value required compensating measures may be taken to increase Q charge. The heat transfer area could be increased by extending the surface of the PCM bag. The Δt might also be increased by lowering the supply air temperature. Finally, the air velocity in the plenum might be increased by reducing the cross sectional area allowed for airflow. Although it is difficult to predict the exact value of hc, the proposed system is flexible enough to accommodate a reasonable range of values.