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Cold Air Distribution in Office Buildings: Technology Assessment for California

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COLD AIR DISTRIBUTION IN OFFICE BUILDINGS: TECHNOLOGY ASSESSMENT FOR CALIFORNIA

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ABSTRACT

This paper presents the results of a study to assess the current state of practice, and energy and operating cost implications of cold air distribution in California, and to identify the key research needs for the continued development of this technology in new commercial buildings in the state. Whole-building energy simulations were made to compare the energy performance of a prototypical office building in three California climates using conventional and cold air distribution, with and without ice storage, to show the impacts of load shifting, energy use, and utility costs for three typical utility rate structures. The merits of economizers and fan-powered mixing boxes were also studied when used in conjunction with cold air delivery. A survey was conducted to assess the perceived strengths and limitations of this technology, perceived barriers to its widespread use, and user experience. The survey was based on interviews with consulting engineers, equipment manufacturers, researchers, utility representatives, and other users of cold air distribution technology. Selected findings from the industry survey are also discussed.

Cold air distribution (CoAD) is found to always reduce fan energy use in comparison to conventional 55°F (13°C) air distribution systems, when conditioned air is delivered directly to the space (no fan-powered mixing boxes). Total building energy use for ice storage/CoAD systems was always higher than a well-designed conventional system, but significantly lower than a commonly-installed packaged system. When a favorable utility rate structure was applied, the load-shifting benefits of ice storage/CoAD systems produced the lowest annual operating costs of all system-plant configurations studied.

INTRODUCTION

Space cooling in commercial buildings represents the largest single category of electricity demand occurring during a utility's summer on-peak hours. Ice storage is one form of thermal energy storage (TES), or off-peak air-conditioning, that allows most of a building's cooling energy requirements to be shifted to off-peak periods, thus reducing the need for the

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utility to increase generating capacity. To encourage building owners and developers to adopt TES systems in their designs, many utilities have introduced financial incentive programs. Cold air distribution (CoAD) technology in commercial buildings has arisen because of the building space and cost savings that can be realized by combining it with ice storage. By distributing lower temperature air (40°F to 50°F [4.4°C to 10°C]) throughout the building, a CoAD system can take greater advantage of the chilled water (typically 34°F to 36°F [1°C to 2°C]) produced by the ice storage system. The colder temperature allows primary supply air volumes to be reduced compared to a conventional 55°F (13°C) supply air design, while still satisfying the building's cooling load. Consequently, fans and ducts can be downsized, reducing first costs and operating costs, and often saving valuable floor area and vertical height. Since the reduction in fan energy use also occurs primarily during on-peak hours, this further reduces peak electricity demand.

To date the Electric Power Research Institute (EPRI) has sponsored a considerable amount of research on the topics of cool storage and cold air distribution. Dorgan and Elleson (1988) present a comprehensive design guide on CoAD systems based on current practice and available research at that time. Interest in obtaining performance data on operational TES/CoAD systems has led to a few field studies reported in the literature [Dorgan and Elleson 1987, Merten et al. 1989, Dorgan et al. 1990, Landry and Noble 1991]. Whole-building energy simulation studies have been completed to investigate energy use and operating costs for an ice TES/CoAD system in comparison to a conventional system for six U.S. climates [Hittle and Bhansali 1990], and to analyze the energy penalties associated with reduced economizer use with a CoAD system [Catanese 1991]. The importance of utility rate structures in motivating the application of thermal energy storage for off-peak cooling has been recognized by many members of the building industry, and has been discussed by Knebel (1990) and MacCracken (1990). In recent years, several workshops and seminars have been held to disseminate information on TES/CoAD technology to a larger audience [EPRI 1987, EPRI 1990a, ASHRAE 1990a, EPRI 1991a]. Other available publications provide updated summaries of important issues in TES/CoAD technology [EPRI 1990b, EPRI 1991b, Elleson 1991].

Recent research on cold air distribution has focused on the room air diffusion problem, in terms of maintaining both acceptable thermal comfort and indoor air quality. Berglund (1991) reports that the reduced humidity levels occurring in buildings with cold air distribution, can actually provide improved perceptions of comfort and air freshness compared to those experienced at the same temperature in a space conditioned with conventional 55°F air. He quantified the benefits of this relationship by recommending that zone drybulb temperature be adjusted upward by 1°F for each 10°F reduction in supply air dewpoint temperature.

Traditionally, for CoAD cooling applications, fan-powered mixing boxes (FPMB) have been used to raise the supply temperature and flow rate, and to ensure adequate diffuser performance. However, more recent research and development have focused on supplying cold air directly to the zones, eliminating the electricity use and capital and maintenance costs of FPMBs. Concern over the performance of diffusers supplying air directly to the space under low-temperature, low-flow conditions has prompted a number of experimental and numerical studies. Gadgil et al. (1991) describe detailed laboratory tests of a commercially-available linear diffuser demonstrating acceptable performance with a 30°F (17°C) supply/room temperature difference and a supply volume of 0.3 cfm/ft^2 (1.5 $L/s \cdot m^2$). Anderson et al. (1991) describe the development of an innovative experimental technique for visualizing the airflow from cold air diffusers. Kirkpatrick et al. (1991) present a simple analytic model of cold air jet performance. Miller (1991) presents a design methodology for selected cold air diffusers based on previously completed laboratory tests and the ASHRAE Air Diffusion Performance Index [ASHRAE 1990b].

The purpose of this paper is to present the results of a study to assess the current state of practice, and energy and operating cost implications of cold air distribution in California, and to identify the key research needs for the continued adoption of this technology in new commercial buildings in the state. This report comprises the results of two major related efforts.

1. Whole-building energy simulations using the DOE-2.1E computer program were performed to investigate the energy performance and operating costs of a prototypical new California office building using cold air distribution (42°F [5.6°C]) in comparison to the same building with two different conventional 55°F (13°C) air distribution systems (packaged system and component-assembled system). To emphasize the energy saving potential of cold air distribution, with the exception of a separate series of simulations investigating fan-powered mixing boxes, all simulations assumed the direct supply of conditioned air (42°F or 55°F) to the space, without the use of fan-powered mixing boxes. Simulations explored the energy use and operating costs for cold air distribution (1) with four different ice storage capacities (one without storage, one with partial ice storage, one with full ice storage, and one with weekly ice storage), (2) in comparison with economizer use, (3) for different fan-powered mixing box designs and control strategies for direct supply of cold air, and (4) for three different utility rate structures. Most of the simulations were repeated for three California climates, representing areas of potentially rapid growth in new office construction: San Jose, Fresno, and San Bernardino.

2. A survey was conducted by interviewing consulting engineers, equipment manufacturers, researchers, utility representatives, and other users of cold air distribution technology. The information gathered through the survey was used to assess the current state of practice in California by producing a list of current California projects involving cold air distribution and a discussion of the factors influencing the future development of cold air distribution.

The results of the DOE-2 simulations and industry survey are presented in the next two sections. The report concludes with brief discussions of the implications for utility support for CoAD technology, and future trends and important research needs for the continued adoption of cold air distribution technology in new commercial buildings in California.

DOE-2 SIMULATIONS

Description of the Prototype Building

Huang et al. (1991) have investigated the characteristics of commercial office buildings in California and have separated them into two categories: old and new. This study found that recently built buildings in mild California climates such as Los Angeles frequently use 76 - 80 kBtu/ft^2 (870 - 910 MJ/m^2) per year in fuel and electricity, an improvement over many buildings of the older stock which use roughly 130 kBtu/ft^2 (1480 MJ/m^2). Much of the description of the building shell, scheduling and internal energy use in the definition of the prototypical new California commercial building used as input to DOE-2.1E in this study is based upon their work.

After discussions with members of the Simulation Research Group at Lawrence Berkeley Laboratory and the Department of Architecture of the UC Berkeley campus, some of the building shell characteristics and cooling systems were modified to improve energy efficiency and reflect some of the recent construction trends.

The prototype building is defined to have three floors of 60,000 ft^2 (5,570 m^2) each and has steel and spandrel glass R7.5 walls. Thirty percent of the gross wall area is double-paned glass, having a normal transmittance of sixty two percent. The roof is steel and metal decking under tar and gravel and is insulated to R15.8. The interior floors are carpeted four inch concrete and interior walls are steel stud and gypsum board.

Each floor for simulation purposes is divided into the usual single core zone and four perimeter zones consistent with the assumed uniformity of building use. Between floors and below the roof are three foot high plenums for utilities with bases of lay-in acoustical tiles, producing an 8.5-ft (2.6-m) floor-to-ceiling height. Thermal transfer takes place between core and perimeter zones and between zones and plenums above.

Scheduled use of the building is the standard five day work week with hours of 8 am to 5 pm. Limited use is assumed during the weekends and evenings. Full occupancy is assumed on work days with one occupant for every two hundred square feet of gross floor area. The lighting load at full occupancy is 1.67 W/ft^2 and equipment loads are 0.8 W/ft^2. In addition maximum domestic hot water use of 73,000 Btu/hr (21 kW) and elevators which draw 195 Btu/hr (57 kW) when in full use are assumed. Elevator use coincides closely with occupancy of the building.

Infiltration is assumed to be approximately 0.35 ACH in the perimeter zones and 0.25 ACH in the core zones when the building is unoccupied and therefore not pressurized.

To provide a broader basis for comparison, the simulations modeled two different design approaches to a conventional 55°F (13°C) variable-air-volume (VAV) system. In one case, two packaged VAV units provide cooling by direct expansion and provide heating by hot water to terminal reheat coils. One unit serves primarily the core zones and the other primarily perimeter zones. They are equipped with economizer option and are air-cooled.

The second conventional 55°F configuration as well as all cold air (42°F [5.6°C]) distribution systems studied use two simple VAV systems with terminal reheat. The cooling coil is served by chilled water or a 25% ethylene glycol water solution, and both primary heating and terminal coils are served by hot water. One system serves primarily perimeter zones and the other primarily the core regions of the building. Supply and return fan sizes are determined from building cooling loads and the minimum supply air temperature.

Selection of 42°F supply air temperature for the cold air distribution systems simulated in this study was based on the results of our industry survey; 42°F represented the lowest design temperature in use by consulting engineers practicing in California. It should be noted that 44°F to 46°F supply air temperature may be a more economic choice, depending on climate and building characteristics.

Primary supply air static pressure is 4.0 inches of water and the return fan static pressure is 1.5 inches of water. Pressure is controlled by fan speed. Cooling set points for zone thermostat are 76.2°F (24.6°C) whenever the minimum supply air temperature is 55°F and adjusted upward to 77.1°F (25.1°C) if 42°F minimum supply air is specified. This upward adjustment of the thermostat cooling set point is a conservative estimate of the occupant comfort benefits associated with the reduced humidities obtained with 42°F supply air temperature [Berglund 1991]. The heating setpoint is 72.0°F (22.2°C) in all simulations in this study.

The outside air intake dampers are fully closed when the building fans are off and during warm-up or summer pull-down periods. The minimum supply air rate is 0.30 cfm/ft^2 which is lower than normal practice, but is found to provide adequate diffuser performance even at low supply air temperatures [Gadgil et al. 1991]. The outside air supply during occupancy is at least 19 cfm (9 L/s) per person. The VAV boxes can be throttled to 20 percent in the core zones and to 30 percent in the perimeter zones. Enthalpy controlled economizer operation is permitted whenever ambient air enthalpy is less than that of the return air.

One preassembled (packaged) VAV system-plant configuration and five component-assembled VAV system-plant configurations were chosen for comparison and investigation. The primary heating equipment for all component-assembled configurations includes a gas-fired hot water boiler serving the main heating and terminal coils, and a gas-fired domestic water heater. In the absence of an evaporative condenser model in DOE-2, an oversized cooling tower with two-speed fans is used to approximate its performance. All simulations of component-assembled configurations use the same chiller performance parameters, which approximate an efficient positive displacement machine. Efforts were made to obtain industrial-rated screw chiller performance maps from manufacturers, however full curves were not made available. Therefore, these machines are simulated at this juncture only in an approximate manner, but the approximations match the manufacturer's measurement data, whenever available. Positive displacement machines, such as screw chillers, are more tolerant of the higher differential pressures encountered during ice-making. At this time we are aware of only a few centrifugal machines which are recommended for ice-making applications (this is confirmed by our industry survey results). For the component-assembled configurations, the circulation pumps are variable speed and operate against the same head pressures in the two non-storage configurations and against an increased head in the case of the storage configurations. To decrease pumping costs of cool storage and retrieval, we use the seven-foot pressure drop characteristic of the commercially-available ice-ball system (manufacturer's literature). All chiller capacities are chosen to just meet the peak cooling demand (i.e., no reserve cooling margin) with the intent of reducing electrical costs. Some configurations show slight overloading during peak demand periods, which in the case of the no storage configurations may occur during early morning pull-down periods. In no case is the cooling system overloaded more than 11 hours during the year.

Elaboration of each configuration follows.

<u>Case 1 - 55°F Packaged</u>: A configuration that represents approximately two-thirds of installed cooling capacity in the U.S. [Piepsch 1991] and is characterized by relatively poor energy performance is the packaged, or unitary, HVAC unit. In the present study, this "rooftop" unit is simulated to have economizer capability, using reciprocal compressors, direct expansion cooling, and dry cooling towers to provide 55°F supply air. No attempt is made to refine the performance of any component for energy efficiency.

<u>Case 2 - 55°F Base Case</u>: In this, the base case, two chillers are simulated to provide 55°F supply air. One chiller is sized to carry one third and the second two thirds of design loads and are operated throughout the year as close to capacity as possible by hourly scheduling. This configuration was suggested by one of our utility contacts and achieves lower electricity use than the often used installation of two chillers of equal capacity.

<u>Case 3 - 42°F Without Storage</u>: Again, two chillers are simulated, sized as above to provide 42°F supply air on demand with direct injection diffusers.

<u>Case 4 - 42°F With 1/2 Storage</u>: Ice storage is added which is sized to meet half the cooling design day load and to be charged during off-peak hours by a dedicated chiller. The storage tank is assumed to be insulated and buried to reduce thermal losses to approximately 2.5% of capacity per 24-hour period. Measured thermal storage losses have exceeded 10% of capacity per 24-hour period, resulting in significant penalties in overall annual performance [industry survey results, Merten et al. 1989].

The discharge rate of the storage is driven by demand up to a maximum that would deplete storage by the end of the business day. If additional cooling is needed a supplemental chiller placed upstream of storage precools returning solution to storage. Upstream placement enables the chiller to operate at a higher suction temperature and thus be more efficient [Peters et al. 1986]. It is realized that in practice one may require a single machine to provide both ice generation and daytime assistance. However, for purposes of tabulating chiller performance it is more convenient to simulate two machines and separate their functions. Supply air is provided through direct injection diffusers.

Various other arrangements were evaluated. Included among those investigated were (1) operating the chiller first at low suction temperatures to provide load to capacity and calling upon storage to assist during periods of higher demands; (2) storage is used first and the chiller operates at low suction temperatures when load exceeds maximum storage discharge rate; and (3) storage and chiller are operated in parallel whenever a load exists. These control options were not cost competitive in part due to the higher daytime demand charges used by the utility rate structures selected.

Case 5 - 42°F With Full Storage: Ice storage is sized to meet all the needs of the cooling design day. The tank is assumed to be insulated such that thermal losses are approximately 2.5% of total capacity per 24-hour period. The storage is charged during off-peak hours Sunday through Thursday, with any carryover from Friday contributing to additional storage losses. Supply air is provided through direct injection diffusers.

Case 6 - 42°F With Weekly Storage: Ice storage is sized to meet all the needs of the most demanding five-day work week with charging permitted weekday evenings and weekends if necessary to bring storage to capacity by business hours on Monday mornings. Storage loss rates are the same as those assumed for Cases 4 and 5. No supplemental chiller assists cooling demands during occupied hours throughout the week, reducing utility demand charges. Although not used in this case, some strategies schedule the chiller to operate 24 hours per day, allowing smaller chiller and storage capacities to be installed. Supply air is provided through direct injection diffusers.

Several other configurations for study were considered and included, augmenting the original scope of the study, as discussed briefly below.

During the industry survey, it was learned that most cold air delivery systems use fan-powered mixing boxes to prevent cold drafts in the occupied regions of the zone. After laboratory measurements showed that direct supply of low temperature air could also be successful [Gadgil et al. 1991], we decided to perform four simulations using fan-powered mixing boxes to investigate their energy use and operation costs, depending on their design and operational control strategy.

We found during the industry survey that the use of air-side economizers may be omitted in new high-rise construction primarily because of cost and the reduction in revenue-producing floor area required by the additional mechanical equipment. Apparently it is possible to satisfy California building performance standards without this measure [California Energy Commission 1992]. Six simulations were performed covering the three climates to investigate the energy and cost penalties without economizer use for both 55°F and 42°F supply air.

A configuration not investigated in this study is the use of ducting designed for 55°F supply air, but which would be used to distribute 42°F air to realize the benefits of lower supply fan static pressures. There are examples of very successful retrofits to cold air delivery using ducting originally designed for 55°F supply air (Brady 1986). This configuration was not investigated as we chose to focus on new construction designs that take advantage of the savings in building envelope size and use reduced duct sizes to keep supply fan pressures at normal values.

Another possible configuration was to avoid the use and benefits of low temperature air by supplying 55°F air from ice storage. In most cases with this system, very poor use is made of the investment to produce the cool storage. However, preliminary simulations were performed with this configuration for the three selected California climates in view of the high percentage of time economizer operation is possible during normal occupancy. This has been noted earlier by Hittle and Bhansali (1990).

Other configurations were discussed which involved portions of one or more of the features enumerated above. The configurations presented here are believed to most directly explore the advantages and disadvantages of using TES/CoAD.

Climates

Three regions within California were chosen to study the options described above: (1) San Bernardino, Riverside and environs, (2) the Central Valley in and around Fresno, and (3) the regions to the south and east of San Francisco Bay including San Jose and Livermore. Two reasons for choosing these regions for study are that they represent areas of rapid population growth where significant new office construction is taking place and at the same time present important though not the extreme climate differences found in California.

Utility Rate Structures

Table 1 shows details of three utility rate structures, "A", "B", and "C", that were adapted from those existing in California and modified for this study. Of the three, "A" is more favorable toward load shifting efforts because demand charges from 10 pm to 6 am are 17% of peak-period demand charges, which apply between the hours of 10 am and 5 pm during the summer months. Even during the winter months off-peak demand charges are 72% of peak. Utility "B" summer demand charges from 9 pm to 8 am are approximately 28% of the peak demand charges which apply between 11 am and 6 pm. For comparable periods, Utility "C" shows off-peak to be 36% of peak demand charges. Neither "B" nor "C" offer lower demand charges during winter months, when cooling is often needed and thus do not benefit ice storage systems. Energy costs for Utility "A" during the summer off-peak period are 49% of on-peak rates, those for "B" and "C" are 46% and 52%, respectively, of on-peak rates. Off-peak winter energy charges for Utility "A" are 52% of peak, and those of Utilities "B" and "C" are both 87% of peak. After careful analysis of some simulation results it was found that rate structure "B" caused the monthly utility charge limit of $0.16/kWh to be applied during several summer months.

Ice making is confined to low demand charge periods which begin weekday evenings at 10 pm, and is permitted, if needed, to continue to 8 am. Ice making schedules remain the same throughout the year as the three climates investigated have significant cooling requirements during the winter months. It was found that slight changes in rate structures may alter the attractiveness of TES/CoAD systems significantly.

Results

The simulation results are discussed below for each of the three climate regions and six system configurations. The primary emphasis for comparison purposes is that of ventilating and cooling system electricity use and its effect on operating costs. Capital costs are not quantified as a part of this study; their influence is reflected in the industry survey summarized in the subsequent section.

First, the six system configurations for each of the three climates are discussed. This is followed by results for economizer use, and finally, fan-powered mixing boxes (FPMB). Since many of the points of interest occur repeatedly for the three climates, three figures are introduced at this time. Figure 1 presents annual cooling and fan electricity use for each of the three climates. Figure 2 presents a comparison of the annual total building operating costs for the three selected utility rate structures for each of the three climates. Figure 3 summarizes the installed chiller and storage capacities for all cases except the packaged system for San Jose, Fresno, and San Bernardino. For Case 4, the partial storage configuration, only the ice-making chiller capacity is shown. All stated chiller sizes are rated for standard ARI conditions (i.e., 44°F (6.7°C) leaving water temperature, 85°F (29.4°C) entering condensing water temperature).

San Jose

Results are presented first for climate zone 4 which includes such areas as Livermore, San Jose, and Sunnyvale.

Cooling and fan energy use for the San Jose climate among the six system configurations are summarized by Figure 1a. It is apparent that the 55°F packaged system, case 1, is the most energy intensive among the six configurations, followed by the 42°F supply air system without storage. Fan electrical use for case 1 is nearly identical to that for the base case, however, the electrical energy used for the compressors and cooling tower fans exceed that of the base case by 80%. These factors are primarily responsible for increasing the percentage of total building electricity used for cooling to approximately 22% versus 17% for the base case. Comparing the 42°F supply air case without storage to the base case, even though the total fan electrical use decreases by 36%, the chiller use increases by more than 113% (compared to the base case), which overshadows savings in fan energy use. Even though the fraction of total electricity use attributable to building fans decreases to 4.4%, down from the base case of 7.6%, the electricity use for cooling (chiller, pumps, condenser fans) increases to about 17% of the total building electricity use, compared to 9% in the base case.

The half storage configuration shows a 60% increase in plant cooling load, using 72% more electricity for cooling than the 55°F base case. Overall cooling system performance as measured by COP shows a decrease of 12% compared to the 55°F base case. Frequent part loading and higher daytime condenser temperatures of the supplemental chiller decrease its efficiency significantly. Energy use by the condenser increases roughly 40% over that of the base case, in part because the cooling tower is sized to meet the requirements of both simulated chillers, resulting in the simulation of a somewhat oversized fan.

Of the total energy stored in the form of ice, approximately 3.3% is lost through thermal gain over the year, 96.2% is recovered for building cooling and the remainder is carried over to the next year on the last day of the simulation. Even though the 22 MBtu (1,830 ton-hr, 6.4 MWh) storage is sized to meet half the design day cooling needs, about 83% of the total annual cooling load is met by storage. Storage is capable of meeting nearly all cooling needs during the winter periods as hourly reports show no supplemental chiller use during the months of December and January.

Even though the full storage case causes the plant cooling load to increase by 65% over the base case and chiller electricity use to increase by 104%, the lower off-hour rates and a 30% reduction of peak demand actually lower the annual electricity costs. Two dedicated ice-making chillers of 5.0 MBtu/hr (417 tons, 1.5 MW) total capacity are necessary to meet the larger loads. It is also worth mentioning that the lower nighttime ambient temperatures reduce condenser fan kWh for the full storage case to 12% below the base case use.

Case six, using five-day storage, compared with the base case uses 34% less fan energy, but a 75% increase in plant cooling load increases total electricity used for cooling by 85%, with chiller use alone increasing by 115%. A storage capacity of 105 MBtu (8,750 ton-hrs, 30.8 MWh) is required to satisfy 100% of the cooling demands, but loses approximately 9% of generated cooling capacity annually.

Significance of utility rate structures to offer encouragement to shifting building electricity use to off-peak periods, and reducing daytime peak demand, is seen in Figure 2a. In this figure, the annual electricity costs are shown for three rates if applied to the six configurations in the San Jose climate. Using utility "A" rate structure, the half storage case results in a 8.7% electricity cost decrease, full storage a 13.5% decrease, and weekly storage a 13% decrease below the base case, while the packaged system raises electricity costs by 11.6%, and the 42°F no storage case by 5.3% over those of the base case. Utility rate "B" produces a 2% increase for half storage, and a 1% decrease for both full and weekly storage. The packaged unit raises electricity costs 7.5%, and the 42°F no storage 6.5% above those of the base case. Electricity costs based on utility rate "C" show similar trends to those for utility rate "B".

Fresno

California climate zone 13 represents Fresno, an inland region of greater dry bulb temperature swings and generally lower relative humidities.

Figure 1b presents a summary of the cooling and fan energy use for the six cases for the Fresno climate. As in the coastal climate, the base case uses the least electricity and the packaged unit the most. The penalty for using dry coolers appears to be convincing as the packaged unit uses 97% more electricity for cooling than the base case. The second highest electricity use is the 42°F no-storage case. Annual total building electricity use by the packaged system is 12.1% above the base case, 42°F no storage is 4.4% above, half storage is 1.6% above, full storage is 2.3% above, and weekly storage is 2.9% above.

Figure 2b is a summary of the utility electricity costs using the three rate schedules when applied to the simulation results for Fresno climate. The packaged unit increases peak demand by approximately 27%. The half-storage case shaves peak demand by 17% and the full and weekly storage by about 33%. When the advantages of using off-peak energy are included, the utility rate schedule "A" produces a cost savings of 11% for the half storage and 17% for both the full and weekly storage cases. The rate structure of utility "B" changes operating costs of the storage options only slightly from the base case. Utility rate "C" offers a 5% cost savings for the half storage and a 10% savings for full and weekly storage. The packaged system causes cost increases of 19% with Utility "A" and 14% for both Utilities "B" and "C".

Figure 4 displays chiller monthly COP in response to varying operating conditions. Chiller COP excludes the auxiliary energy consumption of cooling tower fans, condenser water pumps, and cold loop pumps. Chiller COP for the packaged units is not included. In the figure, six chiller COPs are shown, as the performances of two chillers are modeled separately for the half storage configuration. As discussed previously, one is the main chiller, dedicated to ice making (42°F 1/2 Sto Main), and the other is the supplemental chiller, used for producing chilled water (42°F 1/2 Sto Supp). The same trends are observed (and not shown for brevity) for the five configurations in the other two climates studied. As mentioned previously for the half storage configuration, frequent part loading of the supplemental chiller reduces its performance to nearly that of ice-making levels. During the months of January and December the supplemental chiller does not operate, and during February and November only infrequent and light loads are encountered. This is one of the primary reasons why overall cooling COP for the half-storage configuration is only slightly better than the full storage case. The half-storage ice-making chiller is simulated to have a higher average COP than its counterparts in the full and weekly storage cases because heat can be rejected from a cooling tower which is sized for both the supplemental and ice-making machines. The advantage of only nighttime chiller operation, as in the case of full storage, reduces the condenser fan energy use below that of the base case and together with the cooler condenser temperatures reduces chiller head pressures. Chiller performance for the weekly storage case is very similar to that of the full storage case. The ratio of the total heat dissipated by the condenser to the total cooling load is greatest in the case of the 42°F delivery air without storage. This is indicated by the consistently lower COP throughout the year with noticeable part loading during January and December. This effect would be more severe if two chillers of equal capacity were installed.

Figure 5 shows the fraction of stored energy recovered, by the month, for well insulated and properly sized storage in the Fresno climate for the three storage configurations. The low fraction for the month of January is a simulation artifact which has been brought to the attention of DOE 2.1E authors. It is noted that half storage yields roughly 5% greater fractions of recoverable energy than full storage during the cooler months as a result of greater drawdown. As demand for cooling increases during the warmer months these differences become much less as both are used to near design capacity. Weekly storage losses are higher because of the larger surface area and low fractional drawdown during winter months. Half storage provides an annual storage energy recovery ratio of 95%, full storage provides a ratio of 92%, and weekly storage provides 89%. Half storage is able to satisfy 76% of annual cooling load requirements, and full and weekly storage satisfy 100%. It can also be inferred that multiple storage modules would even the profiles considerably and reduce lost chiller work. At the present time simulation of more than one cold storage tank is not possible.

San Bernardino

Climate zone 10 includes San Bernardino and Riverside and is influenced less frequently than Los Angeles by marine air masses. It has a higher sustained air conditioning requirement than Fresno which has some longer periods of valley fog during the winter months.

Differences between predictions for the six configurations in San Bernardino weather are similar to those described earlier for Fresno weather. Cooling and fan energy use are summarized in Figure 1c and utility cost comparisons are presented in Figure 2c. Total building annual electricity use for the packaged system is 11% above the base case. Total annual electricity use exceeds that for the base case by the following amounts: 42°F no storage by 4.5%, half storage by 1.6%, full storage by 2.1%, and weekly storage by 3.0%.

Economizers

The economic benefit of an air-side economizer is expected to be sensitive to climate and the cooling coil temperature. Additional simulations were run with economizer use enabled and disabled for the no storage cases of 55°F and 42°F supply air, so benefits could be more easily compared. All economizer simulations are with enthalpy control, where use of outside air is rejected if enthalpy is greater than that of the return air. Because of reduced building air humidity ratios when low temperature supply air is used, the ambient enthalpies which are greater than about 25 Btu/lbm (58 kJ/kg) dry air must be rejected when 42°F supply air is used, and ambient enthalpies which are greater than about 28.8 Btu/lbm (67 kJ/kg) dry air must be rejected when 55°F supply air is used.

A series of simulations were performed in each of the three climates studied, with the economizer use controlled only by dry bulb temperature in the case of 42°F supply air. Energy consumption is significantly reduced with economizer use based on the following two strategies. (1) In coastal climates ambient air is used only when the dry bulb temperature is more than 10°F below the building return temperature. (2) In the drier inland climates ambient air is used only when the dry bulb temperature is more than 5°F below the return air temperature. Admittance of higher ambient temperatures for economizer use commonly causes much greater latent loads. When economizer operation is enthalpy controlled, energy consumption is reduced further due to improved control of both latent and sensible loads imposed on the cooling coil. It is recognized that enthalpy controlled systems need more maintenance than dry bulb controllers.

Figure 6 is a comparison of important components of the cooling energy use for the three climates, San Jose, Fresno, and San Bernardino, using 55°F and 42°F supply air with and without economizer use. Figure 6a shows that failure to use economizers in San Jose with 55°F supply air increases the plant cooling load by 78%, the total electricity used for cooling by 65%, and fan electricity by 6%. The slight reduction in fan energy use with economizer results from a lower load on the cooling coil and a reduced supply temperature permitted by an approximate four degree throttling range of the mixed air temperature controller. The annual building electricity use increases by 7% as a result of disabling economizers, causing utility electrical costs to increase by 5% for "A" and 6% for utilities "B" and "C".

If 55°F air rather than 42°F air is supplied from ice storage in the coastal climate of San Jose, opportunity to use economizers is greatly increased. The extent of economizer use affects the number of hours the night scheduled chillers must operate to supply ice. A simulation was made to investigate this configuration, and in this preliminary analysis, the energy used for cooling is about 20% greater than that of the base case. If the fan energy required is added to the total cooling energy, the total annual electricity use for ventilation and cooling is lower than any of the four 42°F supply air configurations. The total building electricity use increases only about 1.8% over that of the base case, the peak demand decreases by 24%, and the electrical operating costs decrease about 11% for utility "A", are unchanged for utility "B", and decrease nearly 6% for utility "C".

Penalties are less severe for non use of economizer in configurations with 42°F supply air. In the California climates studied, the hours of availability of sufficiently low ambient enthalpies (less than about 25 Btu/lb dry air) during periods requiring building cooling are greatly diminished from the case of 55°F supply air (which requires ambient enthalpies to be less than about 28.8 Btu/lb dry air). Thus, for the San Jose region for the 42°F supply air configuration, the increase in total cooling load and in total electricity use for cooling owing to non-use of economizer is 17%. Building fan electricity consumption increases by 4.5% having little impact on the total building electricity use, and operating costs increase by approximately 2% for all utilities.

Figure 7 shows the frequency distribution of ambient enthalpy for the San Jose climate. From the figure, one can deduce the maximum number of hours during the year economizer use is possible for the two cases of 42°F supply air and 55°F supply air for the San Jose area. Since weather patterns are not significantly affected by the day of the week, the ratios of maximum economizer availability (presented inside the box in Figure 7) are calculated by dividing the number of business hours for which economizer use is possible, by the total business hours of the year, using all 7 days per week. The ratios do include hours during which the building does not need cooling. Between the horizontal dashed lines at 28.8 and 25.0 Btu/lbm dry air are the additional number of business hours throughout the year that permit use of economizers when 55°F supply air is used instead of 42°F supply air. For the San Jose area this implies economizer use is possible 93% of the normal occupancy periods if 55°F supply air is specified, and 58% of the same periods if 42°F supply air is used.

Fresno's climate shows a smaller cost penalty for not using the economizer than does the cooler coastal San Jose area. Figure 6b shows the effects of using an economizer on annual and fan energy use in Fresno. For the 55°F case without economizer, fan energy use remains nearly constant, the total plant cooling load increases by 33%, and total electricity use for cooling increases by 32%. Total building kWh increases by 4.3% causing electricity costs to rise by 2.4% for utility "A", 3.3% for utility "B", and 3.1% for utility "C". Use of 42°F supply air lowers building relative humidity levels to roughly 30 percent, and, coupled with the usually much higher daytime temperatures of the region, reduces the penalties for operation without economizer. The total fan electricity use decreases 7.6% and the total cooling load and electricity use for cooling increase by about 13%. The total building electricity use increases by 2.6% and utility electrical costs by 1.3% for utility "A" to 1.8% for both utilities "B" and "C". The preliminary simulation with 55°F air from full ice storage, in place of the standard 42°F air, results in total cooling and fan electricity use which is approximately 9.6% greater than the 55°F base case. However, the operating costs are 13% lower than the base case for utility "A", almost identical with base case costs with utility "B", and 7% lower than base case costs for utility "C". Figure 8 shows the fraction of business hours that economizer use is possible using 55°F air is 79% and with 42°F air is 58%.

Similarities are evident between the trends in predictions for both Fresno and San Bernardino climates for both the 55°F supply air and the 42°F supply air. Figure 6c shows the effects of using an economizer on annual cooling and fan energy use in San Bernardino. The predictions for 55°F supply cases show roughly a 36% increase in total plant cooling load and a 31% increase in electricity use for cooling resulting from not using the economizer, producing a 4.6% increase in total building electrical use. Electric utility costs increase 2.6% for "A", 3.4% for "B", and 3.5% for "C". When 55°F air is obtained from the configuration with full ice storage, the changes in prediction results for San Bernardino very nearly duplicate those noted for Fresno.

The 42°F supply cases without economizer use for San Bernardino showed an increase in total building cooling load and electrical use for cooling of 13%, and an increase of 2.7% in electricity use overall, resulting in cost increases of 1.5 to 1.8%. It is interesting to compare performance differences for San Bernardino between the full storage low temperature case and the 55°F no storage case when neither use economizers. The total building kWh decreases by a negligible 0.2% for the storage case, but the peak demand decreases by 31%, the annual electricity costs decrease by 18% for "A", 6.8% for "B", and 12.4% for "C". If similar comparisons are made for the coastal climates, the penalty for not using the economizer with 55°F air is even greater. Since our industry survey showed significant reluctance to use economizers in large commercial buildings, the increased operational costs resulting from foregoing economizer use with cold air distribution are to some degree hypothetical for this class of buildings. Figure 9 shows that economizer use in San Bernardino could represent 84% of business hour operation when 55°F supply air is used, and 58% of the time when 42°F air is used.

Fan-Powered Mixing Boxes (FPMB)

Four simulations were conducted in which system terminal fan-powered mixing boxes were added to investigate their energy use and explore control strategies when they are used with cold air delivery systems. FPMBs are helpful in maintaining air flow in conditioned zones especially when the system fans are operating at or near minimum flows. These units induce plenum air which may be return air from other zones in the building and reintroduce it to the zones they serve along with whatever primary air is supplied. Because fractional horsepower motors are used, these units are inefficient compared to the main supply motor-fan combinations. They can provide an effective first step in meeting heating demands as the plenum air is frequently several degrees warmer than the conditioned space below and the added heat from the unit motor itself elevates the reintroduced air temperature. Concern over higher operating costs is balanced by the designer's greater confidence of occupant comfort when FPMBs are installed in buildings using cold air. This provided the justification for exploring this dimension of practice with four simulations.

Because many designers use these units as air "blenders", with installations serving essentially all conditioned zones rather than only the perimeter zones, the building zones were, for simulation purposes, reconfigured so that all zones are equipped with FPMBs. The simulations were based on San Jose climate with full ice storage configuration. Thus, no cooling equipment is in use during FPMB operation except for circulation pumps supplying the cooling coil.

Four graduated control strategies of FPMB use were explored for their impact on total building fan energy. The first 3 simulations used a parallel FPMB configuration which does not handle primary air flows but supplements primary air to diffusers with plenum air which may or may not be passed across an active heating coil. All parallel units were sized to handle 80 percent of the maximum primary air flow to the zone served. The zone temperature served by each unit usually controls the fan motor.

In the first simulation, the FPMB units were activated if the zone temperature fell below the deadband midpoint to assist in heating the zone, at first, by merely reintroducing warmer plenum air (which also has the fan motor heat added). If the temperature continued to fall the terminal coil would carry hot water for active heating of the zone. This was approximately equivalent to operating the mixing fan only as a boost to supply airflows during the heating mode. This FPMB control strategy enhances diffuser performance only under heating conditions.

In the second simulation, the FPMB units were activated near the top of the deadband to assist air motion within the zone when the primary flow rates to the zone are near or at their minimums. As the zone temperature falls (i.e., in the heating mode), its operating strategy duplicated that of the first simulation. This control setting did not affect peak demand charges which are generated at times of maximum cooling loads. A slight reduction in heating energy was observed as expected, although annual heating costs did not exceed $5,000 in any of the simulations.

For the third simulation, the same FPMB units were activated slightly above the temperature at which active cooling begins. This strategy ensures the mixing of warmer room air with the cold supply air whenever the supply reaches small and medium flow rates. At these flow rates it is often suspected that the room air may not mix well with supply air, and the room occupants may be subjected to uncomfortable drafts at times when the diffusers would still be at less than design flows. With this control strategy, electricity use by the chiller decreased by approximately 18%. Total fan energy remained very close to the non-FPMB case. This intriguing simulation result from DOE-2.1E seems to indicate that zones requiring cooling (interior zones) may benefit from the use of return air from those zones requiring heating (exterior zones). This results in a reduction in primary fan delivery as well as the decrease in cooling energy use. While we believe this energy conserving effect is real, we are currently unable to verify the magnitude of the predicted cooling energy reductions. Authors of DOE-2.1E have been notified and this effect is under investigation. During our industry survey we did not encounter any of the above FPMB control strategies in use.

The fourth simulation used a series FPMB which operated whenever the main building fans were on (100% of the time). Series units must handle primary air as well as the induced air and for this preliminary investigation were sized to handle 110 percent of the primary air flow. Although this is not an energy efficient strategy, a significant number of designers working with cold air distribution use continuously operating series or parallel mixing boxes to ensure successful diffuser performance at all times.

A manufacturer's performance characteristics for an efficient fan and motor were used for the simulations. Figure 10 presents the simulation results in terms of fan energy (MWh) and fan-attributable fraction of total building electricity consumption. Figure 11 presents the effects of FPMB operation on annual total building electricity costs, in terms of the percentage change from the case with no FPMB. The lowest temperature setpoint simulation shows the least impact, especially in California climates which do not have the extended heating periods other parts of the country do. The total fan energy use increased 1.6% raising the total building electrical use and utility costs by less than 0.15% with no noticeable impact on building peak demand since these units would not be operating during peak demand periods. Results from the second simulation show that building fan use increases by 12.4% over the no FPMB case, total building electricity use and utility costs by about 0.5% without raising the peak demand. This indicates that the majority of time the building is conditioned, zone temperatures are much more likely to be on the high side of the deadband midpoint. When FPMB operation extends into the active cooling temperature ranges as in the third simulation, total fan energy remained at the same level as the previous case, with a negligible increase in peak demand over the no FPMB case. For this case, utility "A" costs were essentially unchanged, and utilities "B" and "C" declined by 0.9%. The cost reduction originates primarily in the reduced chiller energy which as mentioned declined about 18%. If series units are operated whenever the primary fan is turned on, the fan energy increases by roughly 150%, the peak demand increases about 4.2%, and the utility costs for all three rates increase approximately 10 to 11%. In terms of fan electricity consumption as a fraction of total building electrical use, the heating only strategy (first simulation in this series) affected fan energy percentage negligibly, the air motion assist raised the fraction from about 4.5 to 5.0%, the limited cooling range to 5.1%, and the series application to 10.3%.

The significant finding illustrated in Figures 10 and 11 is the following. If the FPMBs are operated continuously (strategy 4), they lead to a substantial increase in fan energy use and on-peak demand compared to the cases of direct supply of cold air (no FPMB), or when parallel FPMBs are operated under various zone control strategies. Strategies 1, 2, and 3 avoid or have little impact on peak demand, and elevate fan energy marginally by operating the FPMBs only when the cooling loads are small or medium. Thus, they do not lead to larger energy or peak-demand penalties, while at the same time ensuring adequate diffuser performance at low supply rates of cold air. Simulation results indicate that chiller use may actually be decreased when FPMBs are operated within the cooling throttling range (strategy 3).

If sufficient data from further research can identify optimum control strategies for FPMBs, these results indicate that a satisfactory compromise can be reached between the designer's reluctance to specify only direct cold air supply, and the unwarranted on-peak loads and energy use caused by continuously operated FPMBs.

INDUSTRY PERSPECTIVE

A survey was conducted during the months of May through November 1991 by contacting several practicing engineers, equipment manufacturer representatives, researchers, utility and energy commission representatives, and other users of cold air distribution technology. The contact list was started through a few well-known consulting engineers and a recent EPRI-sponsored workshop on cold air distribution (EPRI 1991a). The list grew largely through word-of-mouth as we attempted to identify all major cold air distribution projects and associated users of CoAD technology in California. A complete list of contributors to the content of this report appears in the *Acknowledgments*. The purpose of the survey was to assess the current state of practice and future directions and needs of this technology, with an emphasis on conditions in the state of California. The results of the survey had three major uses.

1. Support for DOE-2 Simulations.
   Performance data from equipment manufacturers helped to more accurately specify the cooling plant and HVAC equipment. Recommendations and design approaches used by practicing engineers provided additional guidance for identifying a system configuration and operating strategy that represented an energy-efficient design in the current building market.

2. List of California Cold Air Distribution (CoAD) Projects.
   One of the best ways to obtain information on cold air distribution from our contacts was to discuss their experiences with completed or ongoing projects. Based on the information made available to us, we have compiled a list of current California projects involving cold air distribution.

3. Factors Influencing the Development of Cold Air Distribution in California.
   The information gathered from our contacts provided a firsthand look at the reasons behind their willingness or reluctance to consider cold air distribution in their building designs. An assessment is made of the advantages, disadvantages, and future trends of CoAD technology in California based on the gathered information.

The ways that the survey results were used to develop a realistic plant and HVAC system configuration have been discussed in the previous section on DOE-2 simulation results. In this section we present the compiled list of cold air distribution and related projects in California, and discuss in detail the factors influencing the development of this technology in the state.

List of California Cold Air Distribution Projects

For each case in the following list of California projects, we present a brief description of the building, cooling plant, and air distribution system. Information on other operational CoAD systems has been previously reported by Dorgan and Elleson (1987), Dorgan et al. (1990), and Landry and Noble (1991). We have chosen not to identify buildings by name or associated design engineers. The focus of the presented information is to assess the range of equipment configurations and design approaches being used by currently practicing engineers who include cold air distribution in their designs. The projects are grouped into three categories: (1) TES/CoAD - buildings that combine thermal energy ice storage with cold air distribution; (2) CoAD only - buildings using cold air distribution without thermal energy storage; and (3) TES only - buildings using thermal energy ice storage without cold air distribution.

The list is based on information made available to us during the relatively short period of time allowed for the survey. The list is not long, indicating that cold air distribution is still a young technology in California. In addition to those listed below, Pacific Gas & Electric (PG&E) and Southern California Edison (SCE) are currently developing demonstration installations of ice storage. Both will have the capability to provide low temperature air directly to the space, although, at first, only the SCE facility will be set up to operate this way by including specially-designed diffusers for cold air applications.

TES/CoAD Projects

Building A

This 12-story office tower located in southern California was completed in 1987. The performances of both the ice storage system and cold air distribution system were the subjects of previous detailed field studies [Merten et al. 1989, Dorgan et al. 1990]. Each floor is 21,000 ft^2 for a total for the building of 265,000 ft^2. A cold air distribution system was chosen for this building because it permitted a reduced floor-to-floor height. The duct sizes necessary for a conventional air temperature system were too large for the space available.

The cooling plant is an ice harvester designed for a weekly cycle. The storage tank is partially charged during the weekday off-peak periods and is brought to full charge during the weekend off-peak period. Ice water is circulated from the storage tank at 33°F through a plate frame heat exchanger to cool the building cooling loop water to 36°F. The design cooling load is 600 tons. The ice harvester capacity is 245 tons for making ice, or 357 tons for chilling water. The overall design day load is 3,600 ton-hours. The storage tank has a capacity of about 7,900 ton-hours.

Variable-volume air handling units located on each floor of the building have a design capacity of 14,500 cfm. A constant design ventilation rate of 9% is supplied to each floor through a central air shaft. Primary air at 45°F is supplied to parallel fan-powered mixing boxes. These mixing boxes operate continuously during occupied periods to combine primary air with room return air, producing a diffuser outlet temperature of 55°F to 60°F.

Building B

This 20-story, 350,000 ft^2 office building in northern California combines thermal storage with cold air distribution. The local building code placed height limitations on all high-rise buildings in the area. By using cold air distribution to reduce duct sizes and additional design adjustments by the structural engineer, the floor-to-floor height was reduced to 12 feet, while still maintaining an 8 ft 10 in. floor-to-ceiling height. This allowed one additional floor to be added.

The cooling plant consists of two Carrier 300-ton (for water chilling) centrifugal chillers combined with a FAFCO ice storage system designed for partial storage operation. The ice storage is contained in 17 storage tanks located in the building's below-grade garage and has a capacity of 3,000 ton-hours. Significant usable parking space was conserved by locating the storage tanks under the sidewalks in areas not being used by the local authorities. At the current occupancy of only four floors, the ice storage is able to meet the entire daily cooling load, so the system is presently operated as a full storage system. When fully occupied the building's peak cooling load is designed to be 750 tons, and the total design day cooling load is 11,000 ton-hours. Ice storage leaving water temperature is set at 38°F.

The air distribution system is a low temperature floor-by-floor system. A typical office floor has one air handling unit with a design capacity range of 8,000 to 12,000 cfm. The coil leaving temperature is 44°F to 45°F. Enviro-Tec series fan-powered mixing boxes mix primary air at 46°F to 47°F with plenum air to produce conventional diffuser outlet temperatures through either perforated or linear slot diffusers.

Building C

This 35-story, 585,000 ft^2 building is located in southern California. Local building ordinances dictated a 500-foot height limit for the building. By using cold air distribution to reduce duct sizes and additional design adjustments by the structural engineer, the floor-to-floor height was reduced to 12 ft 2 in. Occupancy is expected in early 1992.

The cooling plant consists of two Trane centrifugal chillers and a Reaction Ice storage system designed for load-leveling partial storage operation. The storage system is made up of 6 steel tanks packed with "ice lenses" (durable plastic water containers) and located at the bottom of the elevator shafts in the below-grade parking garage. Expansion and contraction of the ice lenses in response to charging and discharging of the storage provides a unique system for monitoring the current inventory of ice storage capacity. The ice storage has a capacity of 10,000 ton-hours, with a maximum output of 1,650 tons. Each chiller is rated at 1,140 tons for water chilling, or 750 tons for ice making, allowing the storage to be completely charged during the daily off-peak period of 10 pm to 6 am. The building's peak design load is 2,200 tons. 41°F chilled water is provided to the cooling coils of the floor-by-floor air distribution system.

Each floor has its own variable-volume air handling unit. No outside-air economizer is used. 46°F primary supply air is provided to Trane parallel fan-powered mixing boxes, which are operated continuously during occupied periods. This allows conventional temperature air (55°F to 60°F) to be supplied to the conditioned spaces through standard slot or perforated diffusers.

Building D

This 24-story, 276,000 ft^2 office building is located in southern California. If the ducts had been downsized to match the cold air distribution design, the floor-to-floor height could have been reduced by 3 inches. However, this was not done as the ducts were sized for 51°F supply air. The building is scheduled to be occupied in April 1992.

The cooling plant consists of two 587-ton (for water chilling) Trane centrifugal chillers with a storage system of Cryogel Ice Balls in an open concrete tank located under the parking ramp. The system is designed for full storage operation with an ice storage capacity of 9,600 ton-hours. The building's peak cooling load is 1,200 tons. The ice storage is fully charged during the off-peak hours between 10 pm and 7 am. The secondary chilled water loop operates with a 41°F supply temperature and a 54°F return temperature. The primary loop is 25% ethylene glycol with a 38°F/51°F split. A plate frame heat exchanger separates the primary and secondary loops.

A floor-by-floor low temperature air distribution system is used. Each floor has a Mammoth V-Cube air handling unit with a design capacity of 15,000 cfm and a substantially increased cooling coil size. The coil leaving temperature is 46°F, producing a minimum diffuser outlet temperature of 49°F under peak cooling conditions. The supply air temperature is reset up to 60°F under minimum load conditions. Environmental Technologies parallel VAV fan-powered mixing boxes are scheduled to be used only in the perimeter zones for heating. Supply diffusers have not yet been selected, but will be either a slot/troffer design, or an improved small-jet diffuser designed specifically for cold air applications.

Building E

This 38,000 ft^2 office building, located in southern California, has been occupied since early 1991. A rebate from SDG&E covered 40-50% of the installation costs for the TES system. The cooling plant consists of McQuay air-cooled reciprocating chillers (total capacity of 187 tons to make 44°F water) with a 950 ton-hour Cryogel ice storage system. The building's design day cooling load is 145 ton-hours and peak cooling demand is 190 tons.

The cold air distribution system has a 38°F coil leaving air temperature. Both series and parallel Enviro-Tec fan-powered mixing boxes are used. Under peak cooling load conditions, the minimum diffuser outlet temperature is 44°F.

Building F

This 20-story, 200,000 ft^2 office building located in northern California was completed in 1981, making it the earliest California cold air distribution project identified in our survey. The cooling plant is designed for partial storage operation and consists of two Carrier reciprocating compressor/water-cooled condensing units, each having a 100-ton capacity for making ice. Ice-on-coil Girton ice builders comprise the 2,000 ton-hour ice storage system.

A floor-by-floor air distribution system is straight variable-air-volume with hot water reheat, including mono-flow risers. A water-side economizer is also used. By using diffuser outlet temperatures (49.5°F to 50°F) at the high end of the range associated with cold temperature applications, fan-powered mixing boxes were not necessary and standard Tempmaster slot diffusers were used throughout the building without any performance problems.

Building G

This 3-story, 164,000 ft^2 office building is located in southern California. The building owner decided to install a cold air distribution system to reduce the first costs associated with the ice storage system. The CoAD system does eliminate the use of a full economizer option. The building has been occupied for two years.

The cooling plant is a Mueller ice harvester system having a 176-ton nominal capacity for ice making and a 270-ton nominal capacity for water chilling. The ice storage capacity is approximately 2,800 ton-hours, and was designed to operate as a full storage system that is completely recharged on a weekly basis. The building's design cooling load is 450 tons.

The cold air distribution system is variable volume with a 42°F coil leaving air temperature. Reddi-Heat parallel fan-powered mixing boxes are operated continuously during occupied periods. The side-pocket design of these mixing boxes improves the blending of plenum air with primary air before delivering 55°F air into the space through lay-in perforated diffusers.

CoAD Only Projects

Building H

This 7-story 140,000 ft^2 building, located in southern California, has been occupied for approximately one year utilizing a cold air distribution system. These systems are commonly linked to cool storage systems in the form of either chilled water or ice storage. In this system, however, thermal energy storage was not utilized. The advantages pursued were lower floor-to-floor heights resulting from smaller ducts, and a smaller penthouse resulting from smaller equipment. These advantages led to reduced general construction costs.

The HVAC system is designed around a customized roof-top unit. The unit consists of dual screw compressors with pumped liquid overfeed coils. The local utility provided a rebate because of the high efficiency of the system design. The system capacity is 300 tons. The cooling coil leaving air temperature is 40°F. Heating is supplied in two modes. The morning warm-up is provided by electric duct heaters, and radiant panels are used for spot heating during the day.

The central air handling unit distributes cooled air through Enviro-Tec VAV terminal units, providing a diffuser outlet temperature of approximately 42°F. The low temperature air is supplied directly to the occupied space through diffusers designed specifically for cold air applications. The diffuser is fabricated from plastic components to reduce any condensation problems and features a series of small air jets to provide rapid mixing with the room air.

Building I

This 21-story, 430,000 ft^2 (25,000 ft^2/floor) office building is located in southern California. Due to anticipated extremely high internal loads (as much as 5 W/ft^2 for equipment load alone resulting from high density of computer-related hardware), the design decision was made to go with cold air supply, even though cool storage was not used. This allowed required duct sizes to be reduced. The building was occupied in September 1991.

The cooling plant consists of two 600-ton York centrifugal chillers and one 275-ton York screw chiller. Normal occupancy is 8 am to 6 pm. The smaller screw chiller is used to handle nighttime and low weekend cooling load needs. The chilled water loop operates with a 5% glycol solution having a 38°F supply temperature and 52°F return temperature.

The air distribution system is low temperature and floor-by-floor, with a central outside air supply system. Each floor has a Mammoth V-Cube air handling unit with a capacity of 12,000 cfm. The total maximum outside air flow rate is 9,000 cfm. 45°F temperature air leaves the cooling coils, and, after passing through Environmental Technologies series fan-powered mixing boxes, is supplied through 2 ft by 2 ft perforated diffusers at a standard temperature of 55°F.

TES Only Projects

The following projects, although not incorporating cold air distribution, are included in the survey because they all use ice storage, and it is instructive to learn why a CoAD system was considered and rejected as a viable option for each particular building. This list is not intended to represent a complete list of TES only projects in California.

Building J

This 50,000 ft^2, low-rise office building located in southern California uses thermal energy storage, TES, to shift the midday peak load to the off peak hours. This has reduced the energy costs for the building owner.

The cooling plant uses two McQuay air-cooled, reciprocating chillers with a Cryogel ice storage system. The Ice-Ball storage has a capacity of 1,330 ton-hours, and is contained in tanks located adjacent to the building. The total cooling plant capacity is 187 tons at 95°F condensing temperature. The chillers are operated in one of two modes depending on whether they are being used for ice making or for building cooling. During ice making they supply 22°F water to the ice storage tanks. During building cooling the chillers supply 42°F water to the cooling coils. The building's design day cooling load is 160 ton-hours and peak cooling demand is 195 tons.

Cold air distribution was not used because of planned building use. The building is classified as mixed occupancy, with a majority of its floor space dedicated to clean rooms and laboratories, and a small amount of office space. There was concern that the CoAD system could not provide the required high rates of airflow for clean room and laboratory applications.

Building K

This 765,000 ft^2 building complex in southern California features a 30-story office tower and a 27-story hotel. The original engineering analysis demonstrated that the use of cold air distribution could reduce the floor-to-floor height by as much as six inches. This reduction in building height altered the architectural appearance of the two-tower complex so significantly that it was rejected in the final design.

The cooling plant is designed for full storage operation and consists of two Carrier centrifugal chillers, each having a 1,020-ton capacity for making ice. The Reaction Ice storage system has a capacity of 12,000 plus ton-hours.

Each building uses a floor-by-floor, straight, variable-air-volume air distribution system. The office tower uses electric resistance reheat, and the hotel uses water-based, fan-coil units. The diffuser outlet temperature is approximately 52°F.

Building L

This 2.5 million ft^2 building complex in southern California features two 14-story detention facilities, a 10-story hospital, and several other smaller support buildings. About half of the total floor area (1.2 million ft^2) is under construction as an expansion to the existing 1.3 million ft^2 of occupied space. Occupancy of the new buildings is expected within 1 1/2 years.

When completed, the new cooling plant will handle the entire building complex, including the existing structures, making it the largest ice-based thermal energy storage plant in the country. Six York screw chillers and a Cryogel Ice Ball storage system are designed for full storage operation during the hours of 11 am to 7 pm. Three of the chillers, with a total capacity of 3,000 tons for ice making, are used to charge the 48,000 ton-hour ice storage system. The other three chillers, with a capacity of 3,750 tons for water chilling, are available to handle instantaneous loads since the detention facilities are occupied on a 24-hour basis. For most of the year the ice-making chillers are able to operate at an electric input ratio close to 0.6 kW/ton, due to the availability of very cold cooling tower water.

Cold air distribution was not used in the building due to the high ventilation rates required to offset problems associated with odors and air quality. A combination of constant and variable volume control, along with coil temperature reset, provides 50°F to 65°F diffuser outlet temperatures. The air distribution system is dual duct with VAV terminal units; no fan-powered mixing boxes are used.

Factors Influencing the Development of Cold Air Distribution in California

The objective of our survey was to assess the current state-of-the-art of cold air distribution technology in California. The discussion in this section is based on the knowledge, experiences, and opinions of those contacted during the survey. Since our survey focused on users of CoAD technology, the results are representative of a sophisticated, highly knowledgeable, but small segment of the currently practicing building/HVAC industry as a whole.

First-Cost Considerations

In its current state of development, cold air distribution can provide important economic benefits to ice storage systems. While the desirability of thermal energy storage is primarily driven by utilities, who provide incentives to commercial customers for shifting energy demand from peak to off-peak hours, for most commercial buildings with ice storage the first-cost savings of including cold air distribution are indisputable. For the majority of buildings surveyed, the combination of CoAD with TES produced a design that was much more competitive in first cost than a comparable TES system alone was with conventional HVAC design. With further research and development, cold air distribution installations will provide even greater economic advantages for utilities and their customers.

Almost every high-rise CoAD building surveyed listed space savings and the resulting cost savings as a major consideration for the inclusion of the CoAD system. Smaller-sized ducts (typical ducts for a CoAD system are 25% to 40% smaller than for a conventional 55°F system) allowed designers to offer a reduction in floor-to-floor height to the building owners. In at least two cases, this allowed one additional floor to be added to the building while staying below the maximum building height limitation imposed by the local building code. In one of these buildings, by combining smaller ductwork with structural modifications to allow extra space for ducts to pass major support beams, the floor-to-floor height was reduced by nearly eight inches. For this building owner, the rental revenue alone from the one additional floor far outweighed the extra construction costs, and was the major economic incentive for installing a TES/CoAD system.

Despite the potential advantages of reducing floor-to-floor height in a building, there are other considerations that can influence a final design. One building owner with an ice storage system did not choose to install a CoAD system because the reduced building height would disrupt the overall architectural balance of the two-towered building complex.

First cost savings were also acknowledged in the survey results. Because of the reduced volume of air typically required in a CoAD system, the smaller ductwork and air distribution equipment is less expensive, and can increase the available rentable area on each floor.

Room Air Distribution

In a CoAD system, after distributing the low temperature primary air throughout the building, a critical consideration reported by all survey respondents was to determine which method would be used to deliver air into the conditioned space. As in any air distribution system, it is essential that, for both cooling and heating modes, satisfactory room air diffusion be provided to ensure acceptable thermal comfort conditions and indoor air quality for the building occupants. In light of the increased public awareness of building comfort and air quality issues, consulting engineers are not willing to take any substantial risks in this area.

The direct supply of 40°F to 50°F temperature air for space cooling, while using the least amount of fan energy, increases the chances of poor diffuser performance. Practicing engineers and researchers alike recognize that the higher buoyancy forces associated with larger room/supply temperature differences combined with lower inertial forces from reduced diffuser outlet velocities are driving the fundamentals of the flow problem in the direction of diffuser failure. Not surprisingly, most of the CoAD projects surveyed used fan-powered mixing boxes to raise the air temperature and flow rate before delivering it to the space. Only two CoAD projects supplied cold air directly to the space: one utilized new diffusers designed specifically for cold air applications; and the other used a 50°F supply temperature, at the high end of the range associated with low temperature design.

The major reasons (offered unanimously by the engineers interviewed) for avoiding the direct supply of low temperature air (40°F to 50°F) into the conditioned space of the building were: (1) lack of availability of supply diffusers with the manufacturer's backing from performance testing under low supply air temperature conditions, and (2) lack of confidence in the capability of a low-volume cold air distribution system to provide effective ventilation throughout the conditioned space of the building.

Recent research [Gadgil et al. 1991, Miller 1991] has shown that currently available diffusers can provide acceptable room air diffusion with cold supply air temperatures under the right operating conditions. A few engineers expressed a willingness to use diffuser outlet temperatures as low as 45°F when providing maximum cooling. They were confident that the higher supply air volume under peak load conditions was sufficient to ensure satisfactory diffuser performance. On the other hand, some engineers avoided anything but conventional supply air temperatures by specifying continuously operating fan-powered mixing boxes in all of their designs. At the reduced supply volumes characteristic of part load conditions, however, almost all engineers were reluctant to continue to use 45°F diffuser outlet temperatures. Given the currently available diffusers on the market, the most common solutions offered for part-load diffuser performance problems were: (1) use of fan-powered mixing boxes, and (2) implementation of a control strategy to reset supply air temperature at the cooling coil.

Concern over poor diffuser performance is not unwarranted. Several survey respondents recalled from previous experience that they had encountered diffusers that were dumping at low velocity and temperature conditions. Some of these failures, in fact, occurred in conventional temperature systems. As has been previously recommended by Dorgan et al. (1990), EPRI (1990a), and EPRI (1991a), our survey reports a consensus that there is a strong need for research, development, and testing, leading to the introduction on the market of new diffusers for cold air distribution. In addition, research can also provide the designers with reliable data for "setpoints" for intelligent control strategies that use FPMBs only for off-peak (and part-load) cooling with cold air supply.

In most California climates, space heating is not a significant issue in commercial buildings. None of the survey respondents reported any diffuser performance problems in the heating mode for CoAD system designs. To ensure that discharge velocities are high enough to offset the buoyancy effects of warm air, thus maintaining acceptable comfort conditions throughout the heated zone of the building, fan-powered mixing boxes are commonly installed in the perimeter zones. The relatively small fraction of the time that these fan units must be operated produces an almost insignificant increase in total fan energy use (see Figure 10), although first costs will be slightly higher.

Fan Energy

As shown previously in the results of the DOE-2 simulations, a CoAD system, with or without ice storage, can produce significant fan energy reductions compared to an air distribution system using conventional 55°F supply air. Since fan operation in commercial buildings occurs primarily during daytime hours when utilities charge their highest rates, the operating cost savings become even more important.

Energy use of fan-powered mixing boxes remains a concern for many consulting engineers because of the long hours of annual use of these devices. The electricity used by mixing box fans, particularly series fan-powered boxes, is often significantly higher than the central fan energy use. The reasons for this increase are that the smaller mixing box fans and motors are relatively inefficient, and, while central fans are most commonly operated on a variable speed/volume basis, mixing box fans are not. They often operate at full volume during the entire occupied period. Despite these considerations, if faced with designing a cold air distribution system, nearly every design engineer interviewed would choose to specify parallel or series fan-powered mixing boxes to ensure that conventional (55°F to 65°F) diffuser outlet temperatures and adequate room air circulation are maintained. In their opinion, the energy penalty and increased installation, operation, and maintenance costs associated with the use of mixing boxes is much more preferable than the perceived alternative scenario: the potential for problems and occupant complaints about air quality (e.g., poor air circulation) and comfort (e.g., diffusers dumping cold air into the occupied zone of the building).

The most energy intensive solution to the diffuser performance problem is the use of continuously operating series fan-powered mixing boxes. In comparison, parallel fan-powered mixing boxes use less energy, although some engineers prefer to operate them continuously with the central fan. A more sophisticated energy-saving control strategy used by some engineers is to only turn on the parallel fan-powered boxes when the required supply air volume drops below 50 to 60% of peak demand, used in conjunction with supply air temperature reset to avoid the use of the fan-powered mixing box for the majority of the year. Of all the options, supply air temperature reset can be the most energy efficient. A typical reset schedule is to raise the supply air temperature in response to reduced demand for cooling, provided the maximum individual zone cooling load is satisfied. This strategy will use slightly more fan energy, but not significantly more, since the supply temperature is raised only under low load conditions or during cooler weather. When properly instituted, supply air temperature reset can prevent the delivery of low-temperature air at a volume below the rated performance of the diffuser, without the additional energy use of fan-powered mixing boxes. Therefore, reliable research data are needed on the "setpoints" (above which diffuser performance with direct cold air supply is acceptable, and) below which FPMBs can be switched on using an intelligent control strategy.

As an example of the importance of mixing box fan energy use, Dorgan et al. (1990) monitored the energy use of one of the operational California CoAD projects and compared it with the computer-predicted energy use of a conventional 55°F air distribution system. Although the central supply fan energy use for the CoAD system was 34% less than that of the conventional system, parallel fan-powered mixing box operation offset these savings, producing a 9% increase in the total fan energy used by the CoAD system over the conventional system. The reason for this inefficiency was that while the mixing box fans were assumed to turn on in the computer simulations only when the primary airflow dropped below 50% of maximum (a reasonable control strategy), in the real building the mixing boxes ran continuously during occupied hours.

Equipment for Cold Air Distribution

In discussions with our contacts, several examples were cited, including two in California, in which currently available high induction diffusers were used successfully with direct supply of low temperature air (e.g., 45°F air at 0.3 to 0.4 cfm/ft^2). One of the more widely applied of these existing diffusers is a linear diffuser having an exceptionally narrow inlet slot. This design produces higher supply velocities, and improved room air movement and mixing. During tests of this diffuser conducted in the Controlled Environment Chamber at UC Berkeley during Phase I of this project, the diffuser was found to perform well at a supply temperature of 45°F and supply volume less than 0.3 cfm/ft^2 [Gadgil et al. 1991].

Despite these accomplishments, most of the knowledgeable survey respondents (let alone the large majority of consulting engineers in today's HVAC market) would be reluctant to go with direct cold air supply on one of their jobs. In California, concern over diffuser performance may be even greater due to low lighting design power levels dictated by the Title-24 Non-Residential Standards [California Energy Commission 1992], leading to even lower airflow requirements (as low as 0.15 cfm/ft^2) under cold air conditions. In response to this clearly identified need of CoAD technology, the design and development of diffusers suitable for the direct supply of 40°F to 50°F is attracting a great deal of attention among forward-looking air distribution equipment manufacturers.

At a recent EPRI workshop [EPRI 1991a], several equipment representatives expressed a commitment by their companies to provide new products and more extensive performance data to support claims of satisfactory operation with low temperature air. One of the few California buildings with direct supply of cold air uses a relatively new diffuser design that has demonstrated exceptional performance under low temperature, low flow conditions. The diffuser delivers air to the space through a series of small-diameter (1/2 inch) openings; the supply air jets induce rapid mixing with the room air. The diffuser is also fabricated from plastic components whose insulating properties help to eliminate sweating problems. In addition to the diffuser manufacturers, several of the consulting engineers we spoke with were aware of this diffuser. Laboratory tests have shown this diffuser to perform satisfactorily (no dumping) at supply temperatures of 40°F and below and airflows as low as 0.15 cfm/ft^2. Measured room air temperatures within one foot of the inlet were reported to be 60°F to 65°F.

It is somewhat surprising that this diffuser has not been more extensively used in cold air installations. One criticism deals with its appearance. Interior designers may resist using it because the diffusing elements protrude below the ceiling. In recent years, however, the diffuser has been used successfully under humid conditions in Florida on a number of cold air jobs. Two diffuser manufacturers that we spoke with claimed to be ready to introduce on the market in early 1992 updated versions of this same diffuser.

Another type of diffuser having potential for cold air applications utilizes a swirl or twist technology to promote high induction rates with the room air. The diffuser has been marketed predominantly in Europe, however, more performance data are needed under cold air conditions.

There has been interest in improving the energy efficiency of mixing boxes. Non-powered induction boxes eliminate mixing fans altogether and use the central fan to induce mixing of primary and plenum air before delivering it to the room. One such product on the market is currently being tested with 45°F supply air. Total pressure drop through the unit is less than 0.5" H_2O. Future developments are also anticipated to improve the efficiency of mixing box fans and motors. Electrically commutated motors, for example, are not too expensive, are more efficient, and can be operated on a variable-speed basis.

Moisture/Condensation

In an ice storage and cold air distribution system, greater care is needed during installation to avoid condensation problems. Failure to ensure that all cold water pipes and cold air ducts are adequately insulated and sealed can cause some headaches, as demonstrated in one of the California CoAD projects. Apparently, a large amount of the duct vapor seal and insulation was damaged during construction. As a result, a significant number of ceiling tiles received moisture damage and had to be replaced. Since the insulation problem was corrected, the CoAD system has worked well.

Despite the above example, the large majority of survey respondents had witnessed very few condensation problems in their experiences with cold air distribution systems. Because techniques to control potential moisture problems are well understood, with proper attention it is relatively straight-forward to complete a successful installation. Condensation becomes a greater concern in humid climates, such as Florida, but even there several CoAD projects are currently operating with no troubles from moisture. Clearly, in California with its lower humidity levels, condensation is even less of a major concern.

A few rules of thumb are still in order. The most likely location for moisture buildup occurs downstream of the main cooling coil. Specification of external duct insulation is preferred here (as opposed to the common practice of internal insulation), and the use of a non-fibrous insulating material, would also be beneficial. If not controlled, moisture buildup, particularly in fibrous insulating materials, can promote the growth and spread of undesirable bacteria, including Legionella. During the initial building startup and other startups occurring when high humidities exist in the building (e.g., Monday mornings), some care must be taken to gradually step down the supply air temperature based on measured room dew point temperature.

System Design Engineering Approaches

Ice storage and cold air distribution systems are relatively new technologies that are not familiar to many practicing design engineers. In the opinions of several of our contacts, more often than not, a typical engineer takes a rather lazy approach to system design, relying on previous successes and rules of thumb, while avoiding novel approaches. Clearly, the economics of "dusting off" an old design play an important role in this pattern of non-innovative design solutions. Historically, engineers have also generally not received much credit for incorporating energy saving features into their designs. To encourage the engineering community to take on new approaches that reduce energy consumption and shift energy demand to off-peak hours, financial incentives provided by utility companies are a critical factor.

The design, installation, and startup of a TES/CoAD system will generally take longer and be more costly to complete than a conventional system, not only because of its novelty, but also because there are more critical decisions to be made regarding design and operation. For example, the penalties for running out of storage prematurely are severe. A high utility demand charge may result from having to turn on a supplemental chiller during on-peak hours, and in the worst case, the downsized supplemental chiller does not have enough capacity to meet the building load. With space savings being an important selling point for CoAD systems, a higher level of coordination between HVAC and structural engineers is required. Finding the necessary space for the storage system is also a significant consideration. In one of the installed California CoAD systems, TES was not used, in part, because of a lack of available space.

A hidden benefit of the advanced engineering approaches needed for TES/CoAD systems is that successful designs will be operated and maintained with greater care, and will often include additional improvements over conventional system designs. Chiller optimization is one area where more sophisticated engineering decisions can have a positive impact on the building's energy consumption. One consulting engineer uses the following approach: (1) substantially increases coil size, (2) increases water split to 20°F to 25°F, (3) reduces chilled water pumping power by about 50%, and (4) significantly lowers cooling tower return temperatures. With this methodology, he is able to obtain chiller efficiencies on the order of 0.5 kW/ton for making ice, while compressor efficiencies for water chilling in many conventional designs going in today are no better than 0.6 kW/ton.

High-rise HVAC design practice has also changed over the past five to ten years in response to changing tenant needs. Earlier designs most commonly incorporated a centralized air distribution system because of the higher overall efficiencies associated with large air handling units (AHU) and simpler system configurations. These systems required large vertical air shafts to distribute ventilation and conditioned air to each floor. As multi-tenant office buildings are now the norm, most design engineers are installing floor-by-floor AHUs on all of their high-rise jobs. This approach allows individual tenant needs to be more effectively satisfied (e.g., weekend occupancy of a few floors of the building, or specialized cooling requirements of individual zones containing computer equipment). Computerized energy monitoring systems allow individual metering of tenant energy use, although since these operating costs are typically passed on to the tenant, unless the building is owner occupied, there is little incentive for the owner to conserve energy.

In a floor-by-floor air distribution system, the inclusion of an air-side economizer can create big headaches for the owner and design team. If a centralized economizer is used, the same large space-consuming vertical shafts that could be eliminated with the floor-by-floor design become necessary. On the other hand, if fresh air intakes and exhaust vents are provided for each AHU, the appearance of the building's facade can be severely impacted, and most probably rejected by the architect. The implications of these considerations are that the majority of high-rise buildings going in today do not have air-side economizers. In terms of energy consumption, this situation flies in the face of the well-recognized significant cooling energy benefits associated with economizer use in California (see Figure 6). Although Title-24 is written to encourage economizer use, in its current version, engineers are able to satisfy the performance compliance in most cases without including an air-side economizer. The new revision of Title-24 [California Energy Commission 1992], addresses the economizer issue more specifically.

Until current design practices concerning economizers are changed, however, the energy performance of a cold air distribution system looks significantly better in comparison. As discussed previously in the DOE-2 simulation results, if economizers are eliminated, the total energy use of a TES/CoAD building with 42°F supply air is equal to or less than that for a conventionally designed building with 55°F supply air.

The use of economizers is also driven by concern for the indoor air quality issue. If a building's ventilation system is sized only to accommodate the minimum outside air requirements, there will be no capacity for providing fresh air at a higher rate (flushing the building) in the event that an air quality problem develops. As a result, some engineers are installing partial outside-air economizers.

Controls and Operation

Most of the engineers we talked with stressed the critical importance of controls to a successful TES/CoAD system. "Controls cause 90% of the problems with TES/CoAD jobs" was a common phrase offered. As control strategies tend to be more complex for effectively operated systems, direct digital control (DDC) systems were considered essential, and some controls experts recommended the use of a more expensive industrial-grade controller on all TES projects. The cost of one major breakdown can easily be more than the difference in cost between a standard and higher quality control system. One survey respondent noted that because buildings in the U.S. are built on the basis of $\$/ft^2$, less expensive control systems are often selected. By comparison, in Europe where buildings are constructed by design/build contractors, the buildings are built to last, and higher quality control systems are more commonly specified and installed.

Building operators also need to be educated about TES/CoAD technology. Several of our contacts acknowledged that their TES/CoAD systems were working well due, in part, to knowledgeable operators. However, a number of examples were given of system failures that were due to operational breakdowns. In one building, the operator did not understand the system and changed the control software from its original storage mode to a conventional non-storage mode of operation. Another project was being operated as a full storage system instead of a partial storage system.

As described previously, in CoAD systems fan-powered mixing box control strategies can impact energy use. If mixing boxes are eliminated from CoAD system designs and low temperature air is delivered directly to the space, the accurate control of supply air volume and temperature will become even more critical to ensure acceptable room air distribution. Supply air conditions are typically controlled by a zone thermostat. Under low load conditions, however, some survey respondents felt that it could be possible for the zone load to be satisfied, thereby reducing supply volume to a minimum, while effective room air movement to maintain air quality and "freshness" could be compromised. Ideally, some kind of sensor to measure stuffiness/freshness could be utilized to assist in the complete control of supply air conditions. Carbon dioxide sensors can be used as one approach to provide this type of information, although carbon dioxide levels are not the only determinant of acceptable room air distribution. Dorgan and Elleson (1988) provide a good summary of many of the recommended control sequences for cold air distribution systems.

Most of the consulting engineers we spoke with emphasized the significance of building commissioning. While commissioning is important to the successful operation of any air-conditioning system, since many building designers, contractors, and operators are relatively unfamiliar with TES/CoAD technology, the commissioning process becomes even more critical for these systems. If a system is commissioned, its operation will more likely be optimized in terms of reduced costs, reduced energy use, and fewer number of occupant complaints. The selection of an experienced controls contractor plays an important role in the success of building commissioning. After construction is completed, the contractor will have primary responsibility for working closely with the building operator during building startup and the initial charge of the storage system.

Building Codes and Standards

The design and operation of air distribution systems in the U.S. are guided predominantly by the performance criteria described in two ASHRAE standards: (1) ASHRAE Standard 55-81, "Thermal Environmental Conditions for Human Occupancy" [ASHRAE 1981]; and (2) ASHRAE Standard 62-89, "Ventilation for Acceptable Indoor Air Quality" [ASHRAE 1989]. In California, the California Energy Commission Energy Efficiency Standards for Non-Residential Buildings (Title-24) places limitations on the total annual energy use of the building [California Energy Commission 1992].

Because Title-24 focuses on total building kWh and not demand load shifting, TES/CoAD systems are not actively promoted in the current version of the standard. In the performance approach to compliance, available simulation techniques do not incorporate thermal energy storage models. The only incentive provided by Title-24 is to allow the exclusion of any additional energy use (above that of a conventional non-storage system) required to operate the system. This would include such items as additional cooling energy due to lower chiller suction temperatures, pumping energy to operate the storage, and storage energy losses.

Title-24 limits lighting power densities in new construction. These lower internal heat loads have the effect of further reducing the required primary airflows. In a CoAD system, required airflows could approach 0.1 to 0.2 cfm/ft^2, heightening concern over diffuser performance with direct supply of low temperature air.

The energy benefits of using an air-side economizer in most California climates are well-recognized and encouraged by Title-24. However, in its current version, engineers are able to satisfy the performance compliance in many cases without including an economizer. In the new revision of Title-24, designers have less freedom to assign improved efficiencies to their system components while not incorporating these same improvements into the reference building's energy budget. These restrictions are intended to make it more difficult to avoid economizer use.

Local building codes can often be in direct conflict with the intentions of Title-24. For example, by placing restrictions on the use of roof areas and other space limitations, local codes can preclude the placement of economizer inlet/exhaust grilles and fans.

UTILITY IMPLICATIONS

The application of TES/CoAD technology to commercial buildings has been shown to have significant load management benefits for utility companies. In this report we have demonstrated that the implementation of a utility rate structure that substantially reduces electricity rates during off-peak hours can make the difference between no incentive and a strong incentive to realize lower operating costs by installing a TES/CoAD system (see Figure 2). While this time-of-use rate structure primarily addresses the load shifting benefits of a cool storage system, the inclusion of cold air distribution using direct supply of cold air or intelligently-controlled parallel fan-powered mixing boxes always reduces energy consumption and makes the economics of an ice storage system more cost competitive.

Several of the surveyed design engineers acknowledged the importance of other utility financial incentive programs to the promotion of TES technology. Up-front cash rebates for reducing peak demand were very helpful to offset higher first costs associated with storage systems. In the past, some utilities have restricted the amount of available funds to a level that allowed relatively few participants in such a rebate program. More money will need to be made available for rebates to be effective on a larger scale in the future.

Although the design engineers we spoke with represent a relatively well-informed segment of the HVAC industry, they agreed that a well-designed TES/CoAD system required additional analysis time during the design phase. For engineers unfamiliar with these technologies, the extra design time will be even greater. To assist design firms to cover the additional costs necessary to carry out feasibility studies of TES/CoAD designs, utilities often make limited funds available up front.

The attractiveness of thermal energy storage technology will continue to be driven largely by intelligent utility financial incentive programs. The inclusion of cold air distribution with ice storage designs can serve as an additional strong incentive for installing TES/CoAD systems. Despite compelling energy and cost benefits, however, the large majority of design engineers will hesitate to take full advantage of CoAD technology until they have full confidence in all aspects cold air distribution system performance. For these reasons, utilities can further promote load management technology by supporting research addressing the major needs of cold air distribution.

CONCLUSIONS AND FUTURE TECHNOLOGY NEEDS

A study was completed to provide a current assessment of cold air distribution technology in California. A series of energy simulations of a 3-story prototypical office building were completed using the DOE-2.1E computer program. These simulations were used to examine the energy use and operating costs for six system configurations: (1) packaged system using conventional 55°F (13°C) supply air with no energy conservation strategies, (2) component-assembled conventional 55°F (13°C) air distribution system without storage, (3) 42°F (5.6°C) supply air with conventional chiller without storage, (4) 42°F (5.6°C) supply air with partial (half) ice storage system, (5) 42°F (5.6°C) supply air with full ice storage system, and (6) 42°F (5.6°C) supply air with weekly storage system. All of the above simulations assumed the direct supply of conditioned air (42°F or 55°F) to the space, without the use of fan-powered mixing boxes. Simulations were repeated for three California climates, representing areas of potentially rapid growth in new office construction: San Jose, Fresno, and San Bernardino. Additional simulations were performed to explore energy use and operating cost implications of restricted economizer use and different control strategies for the use of fan-powered mixing boxes.

A survey was completed of consulting engineers, equipment manufacturers, researchers, utility representatives, and other users of cold air distribution to assess the current state of practice related to CoAD technology in California. A list of current California projects involving cold air distribution was compiled, and factors influencing the future development of cold air distribution were identified and discussed. Collected equipment performance data and recommended design approaches were also used to guide the specification of the cooling plant and air distribution system for the DOE-2 simulations.

The major conclusions from the DOE-2 simulations were as follows:

1. In all three climates, annual cooling energy use for the four cases involving cold air distribution was always greater than the base case, a component-assembled conventional 55°F air distribution system without storage. The most energy intensive of the six cases studied was the packaged system (configuration 1 above) followed by the 42°F without storage case (configuration 3 above). Annual cooling energy use for these two cases nearly doubled in comparison to the base case.
2. Fan energy use for the four cases involving cold air distribution always decreased in comparison to the base case. These savings helped, but did not completely offset the cooling energy increases.
3. Compared to the system configuration using cold air distribution without storage, all three combinations of ice storage with 42°F supply air always reduced cooling and total building energy use.
4. The base case configuration always produced the lowest total building energy use. However, with the fairly efficient cold air system designs used in this study, the largest increase in predicted total building annual energy use was only 6.3% over the base case for 42°F without storage in San Jose, and the largest increase in total building energy use for an ice storage/CoAD system was only 4.8% over the base case for 42°F with weekly storage, also for San Jose.

5. The reduction in peak electrical demand for the three ice storage/CoAD systems (approximately -15% for half storage and -30% for full and weekly storage) contributed to lower annual operating costs in comparison to the base case, when a favorable utility rate structure was applied (-8% to -11% for half storage, and -13% to -17% for full and weekly storage). The highest annual operating costs were consistently obtained for the packaged system in all climates and for all utility rates. Since cold air distribution without storage provided only minimal, if any, peak demand reductions, operating costs were always higher than the cold air systems and the 55°F base case.
6. Economizer use played an important role in energy savings, particularly for mild marine-influenced California climates. In San Jose, failure to use an economizer with 55°F supply air increased the total building annual electrical use by nearly 7% and operating costs by 5% to 6%. The economizer penalty was so severe that, if it was not included in the base case 55°F supply air system (a surprisingly common practice in hi-rise construction in California, as discovered in the survey), the comparative energy picture for TES/CoAD systems was significantly improved. A sample simulation found that operating without an economizer in San Bernardino using full storage and 42°F supply air used essentially the same amount of energy annually (0.2% decrease) compared to the 55°F supply air case without storage and also without an economizer.
7. Use of fan-powered mixing boxes (FPMBs) increased distribution energy consumption and peak demand over a wide range (from 0% to 150%), depending primarily on the type of FPMB and mode of operation.

The industry survey demonstrated that cold air distribution (and ice storage) systems are still not being applied on a widespread basis in California. The number of ongoing or completed projects was rather limited. The variety of ice-making equipment reported for the listed TES/CoAD projects indicates that the market is still relatively wide open. A strong preference for a few brands of ice-makers has not yet developed among practicing engineers. Although CoAD systems are most effectively matched with ice storage installations, cold air distribution without storage is also a viable option, particularly as a retrofit in a building that has experienced a significant increase in heat loads.

In most California climates, space heating is not a significant issue in commercial buildings. There were no reports of diffuser performance problems in the heating mode with CoAD system designs. A recommended strategy to handle heating conditions is to install fan-powered mixing boxes in perimeter zones, and operate them only during heating mode to increase mixing of the warm supply air with the room air. The relatively small fraction of the time that these fan units must be operated in California climates produces an almost insignificant increase in total fan energy use.

As more designers and building owners become familiar with using ice storage systems for load management, largely in response to utility incentive programs, cold air distribution will also be considered as an increasingly attractive option. In its current state of development, however, significant energy-saving features of CoAD technology are not being effectively utilized in installed systems. This situation stems from a lack of confidence on the part of consulting engineers in the ability of a cold air distribution system to provide acceptable room air distribution, both in terms of comfort and indoor air quality, without the use of fan-powered mixing boxes. It is recommended that additional research be completed to address the major needs of CoAD technology, and to support the development of new products and energy-efficient designs for systems using ice storage with cold air distribution.

Looking toward the future of cold air distribution technology in California, the DOE-2 simulations and survey results identified a number of key issues where further study would be desirable. With advancements in these areas, a reasonable goal for a well-engineered ice storage/cold air distribution system would be to use the same or less total energy compared to a conventional system design. Issues for further research are listed below.

1. Develop improved methods for supplying low temperature air to occupied spaces. These methods must be capable of maintaining acceptable thermal comfort throughout the occupied zone of the building (e.g., avoidance of cold drafts), and providing effective ventilation (e.g., maintain adequate air circulation).
2. New products and operating strategies are needed for the following three categories:
   (a) Fan-powered mixing boxes, including improved control strategies and more efficient motors and fans;
   (b) System-powered induction boxes; and
   (c) Direct supply of low temperature air through high induction diffusers.
3. Research required to support the above equipment developments include (1) independent laboratory testing of cold air delivery products, (2) field monitoring of operational CoAD systems, and (3) detailed numerical modeling to improve our understanding of the fundamentals of room air motion and air quality resulting from the use of CoAD systems.
4. System energy use and operating costs are highly sensitive to chiller configuration and operating conditions. Chiller optimization studies could provide significant energy and cost benefits.
5. Implementation of effective control strategies can make the difference between a successful TES/CoAD installation and an unsuccessful one. Development of new and advanced control strategies can help to further optimize the overall system performance. For example, a modular storage design can be utilized more efficiently with a knowledge-based control system that predicts in advance daily energy requirements and adjusts the extent of storage charge accordingly. Supply air temperature reset strategies raise the supply temperature in response to reduced demand for cooling, provided the maximum individual zone cooling load is satisfied. In addition to reducing the risk of poor diffuser performance under low temperature, low airflow conditions in a CoAD system supplying cold air directly to the occupied space, this strategy can increase the availability of economizer use in California climates.

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LIST OF TABLES AND FIGURES

Table 1: Utility Rate Structures

	Utility A		Utility B		Utility C	
	Winter Oct 1 - Mar 31	Summer Apr 1 - Sept 30	Winter Nov 1 - Apr 30	Summer May 1 - Oct 31	Winter Nov 1 - Apr 30	Summer May 1 - Oct 31
On-peak demand ($/kW)	3.94	16.92	4.20	16.00	4.15	11.60
Off-peak demand ($/kW)	2.85	2.85	4.20	4.50	4.15	4.15
On-peak rate ($/kWh)	0.06876	0.07671	0.0700	0.1400	0.06544	0.11277
Partial-peak rate ($/kWh)	0.04305	0.05050	0.0700	0.1000	0.06544	0.07654
Off-peak rate ($/kWh)	0.03550	0.03753	0.0610	0.0650	0.05669	0.05843
Monthly ave. rate limiter ($/kWh)	0.35	0.35	0.16	0.16	0.15379	0.15379
Ratchet	none	none	none	none	none	none
Week-day on-peak hours	5 pm - 8 pm	10 am - 5 pm	none	11 am - 6 pm	none	11am - 6 pm
Week-day partial-peak hours	6 am - 5 pm, 8 pm - 10 pm	6 am - 10 am, 5 pm - 10 pm	8 am - 10 pm	8 am - 11 am, 6 pm - 9 pm	8 am - 10 pm	8 am - 11 am 6 pm - 9 pm
Off-peak hours	10 pm - 6 am*	10 pm - 6 am*	10 pm - 8 am*	9 pm - 8 am*	10 pm - 8 am*	9 pm - 8 am*

*plus all 24 hours weekends and holidays

Figure 1a

Annual Cooling & Fan Electricity Use
San Jose

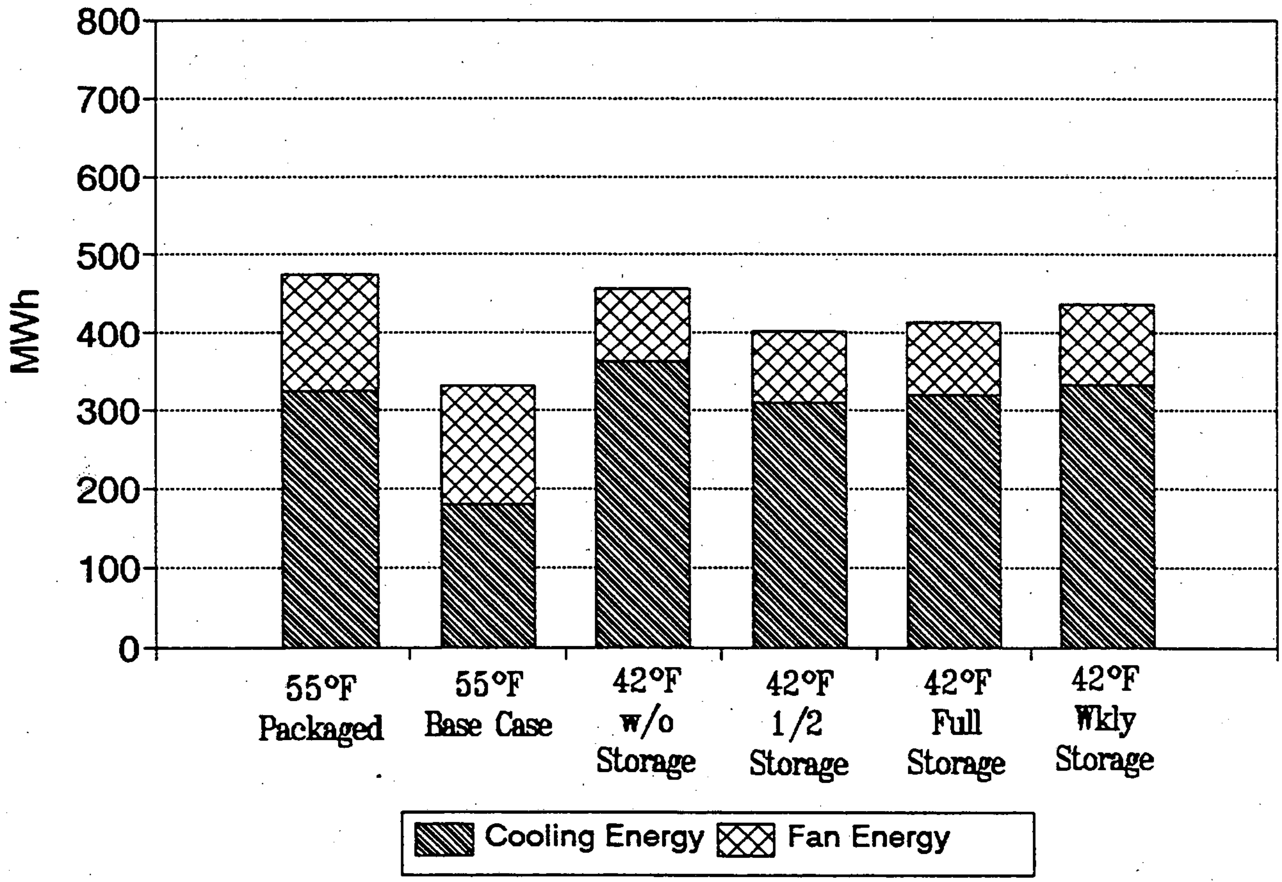


Figure 1b
Annual Cooling & Fan Electricity Use
Fresno

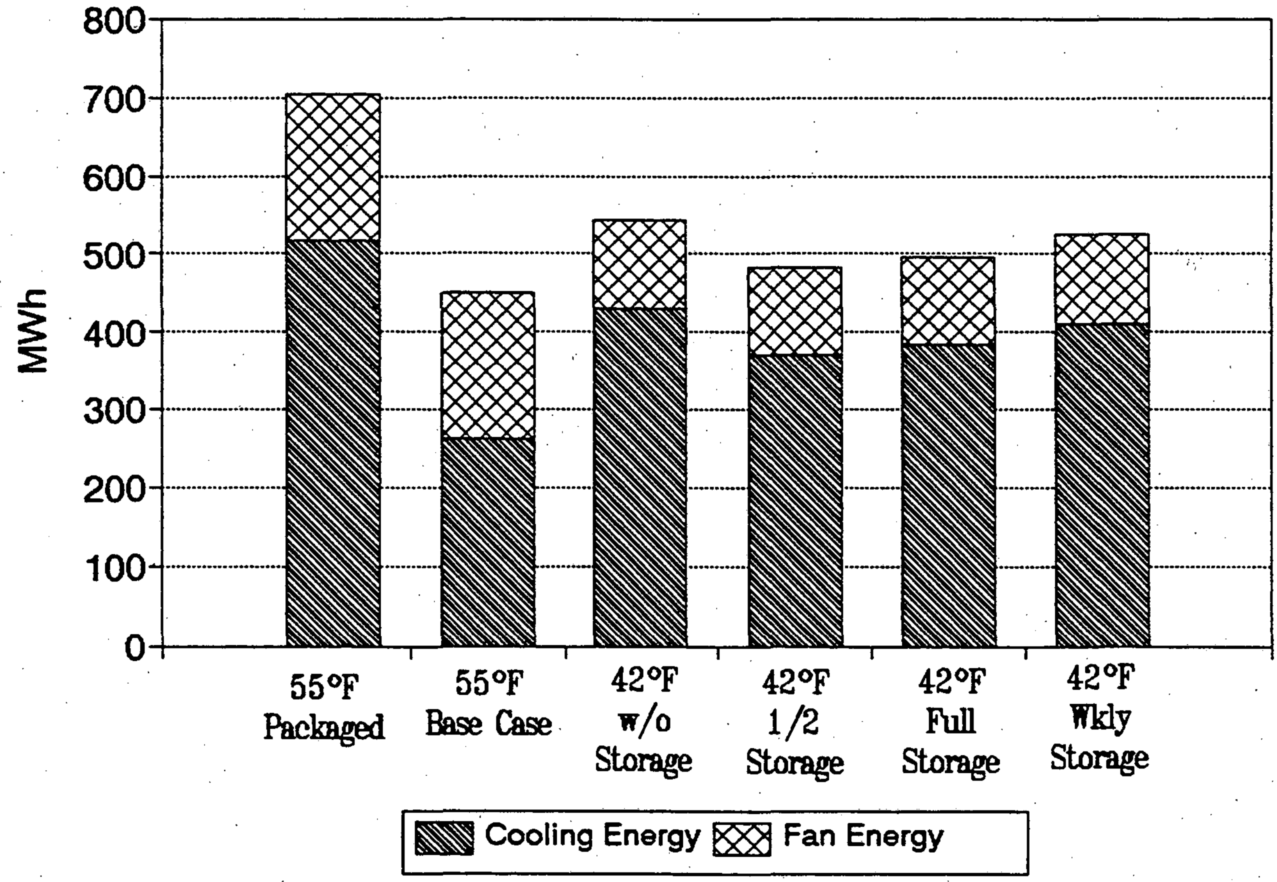


Figure 1c

Annual Cooling & Fan Electricity Use
San Bernardino

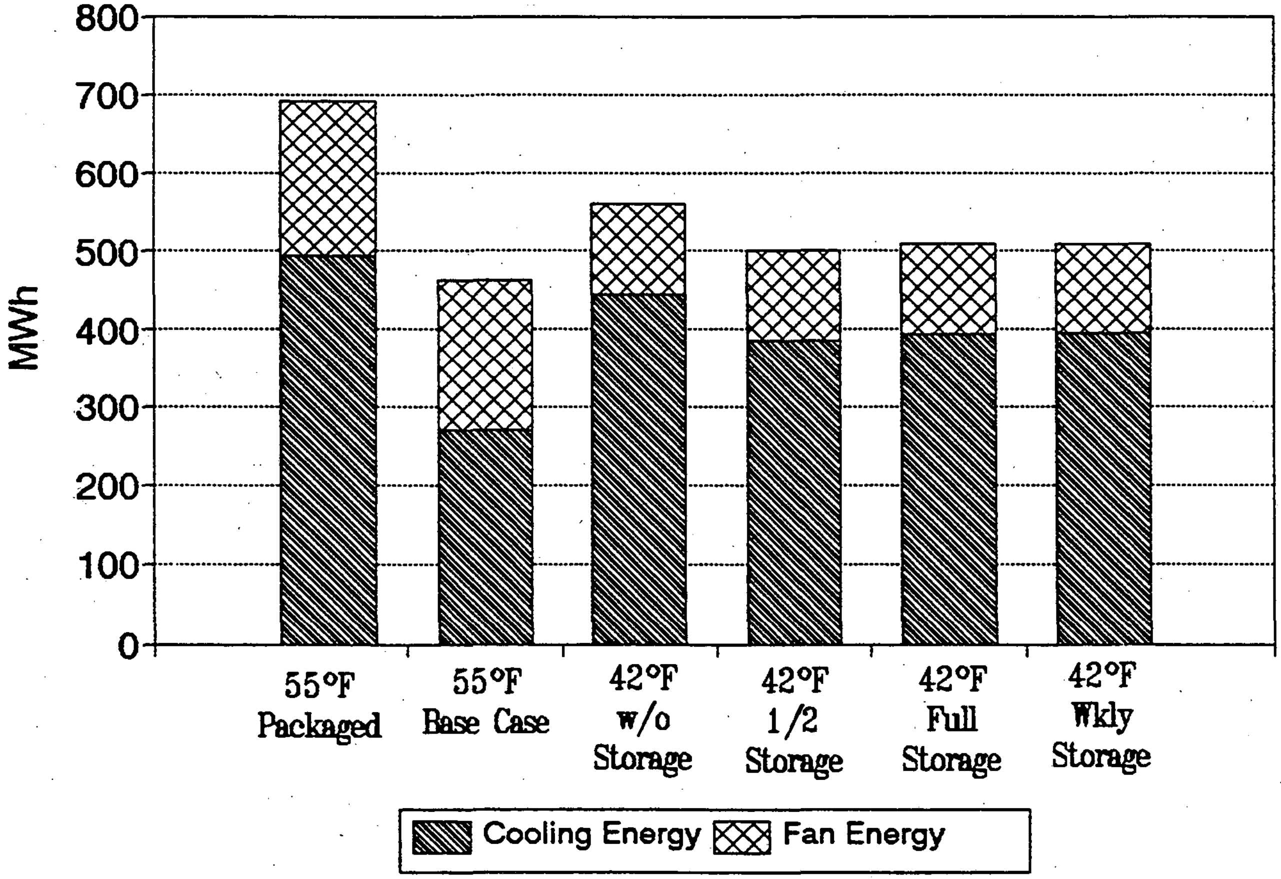


Figure 2a

Annual Total Building Electricity Cost
San Jose

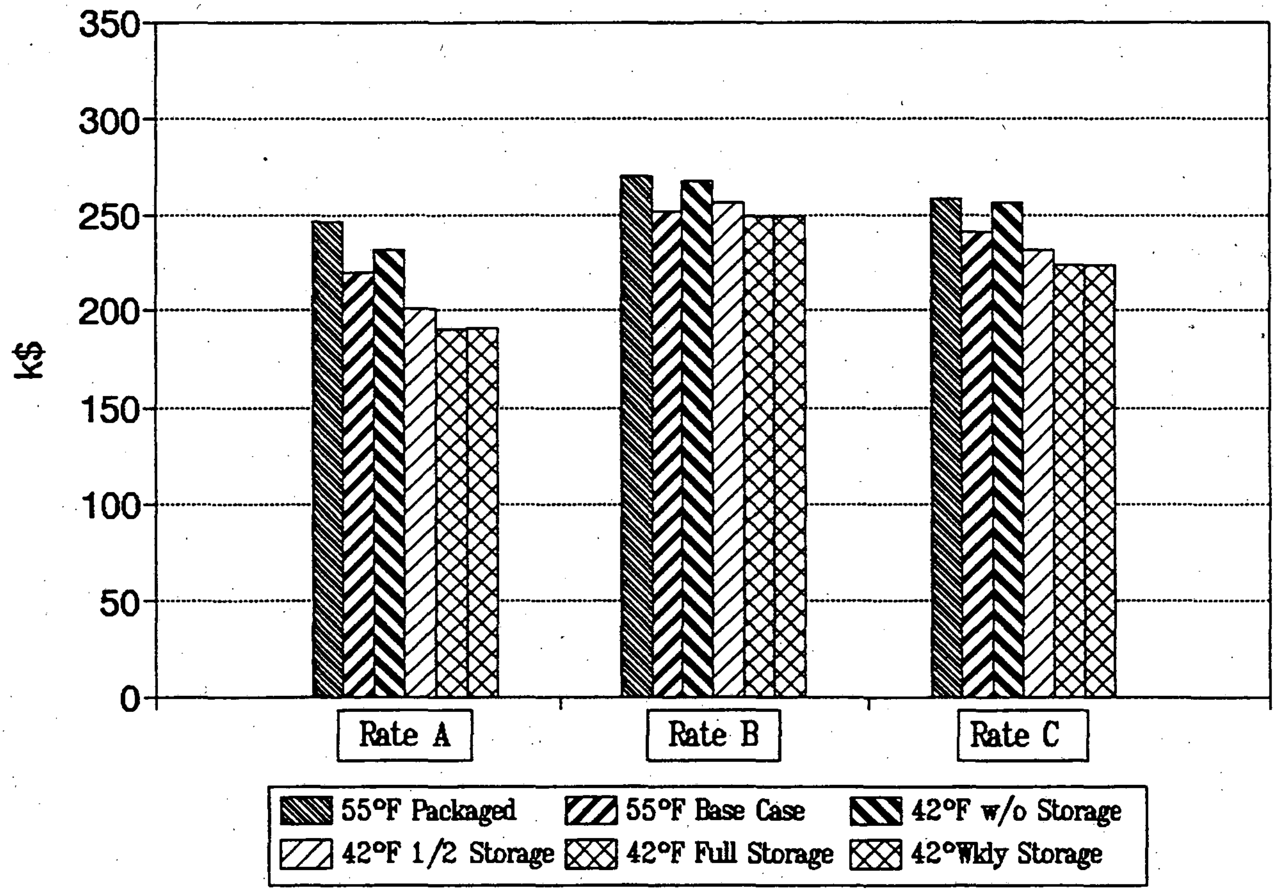


Figure 2b

Annual Total Building Electricity Cost
Fresno

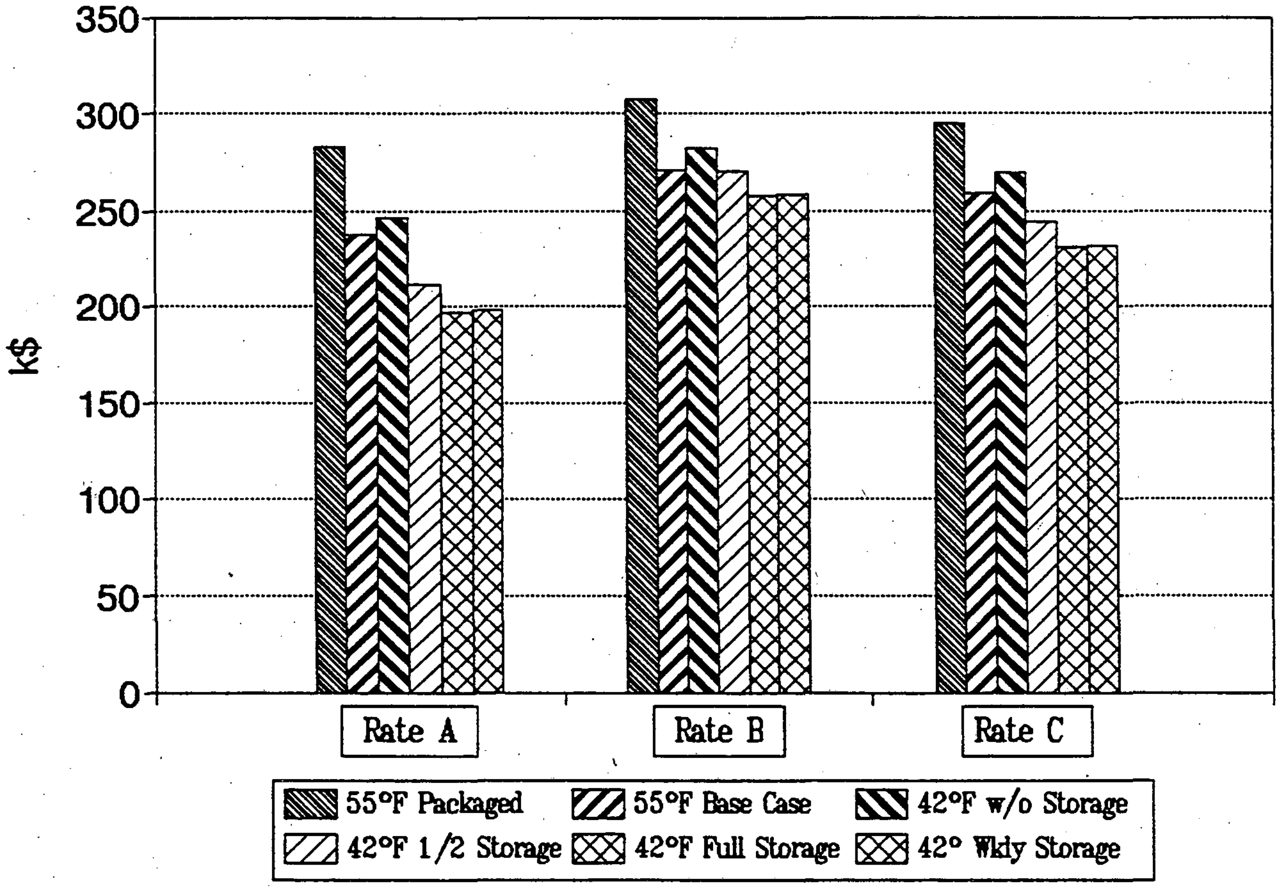


Figure 2c

Annual Total Building Electricity Cost
San Bernardino

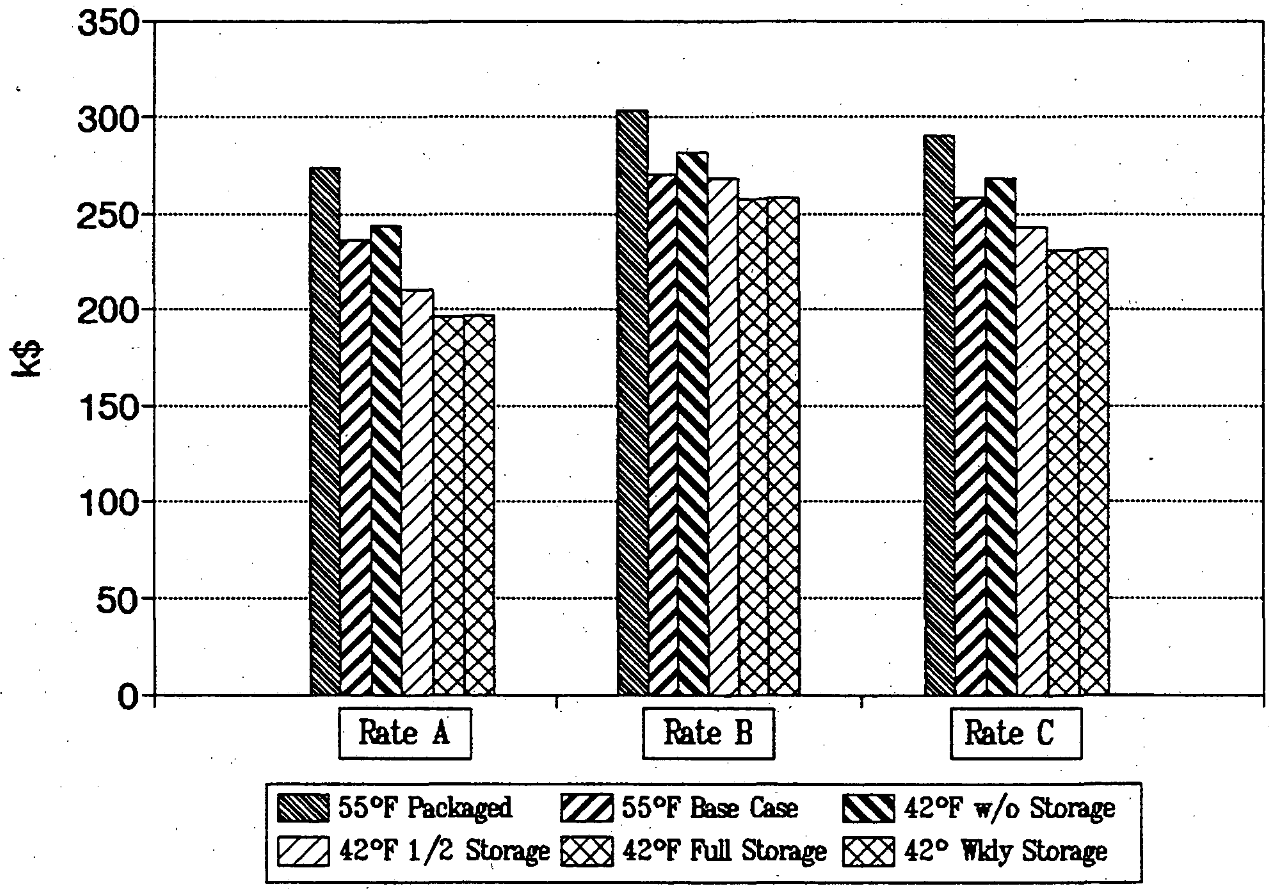


Figure 3a

Installed Chiller & Storage Capacities
San Jose

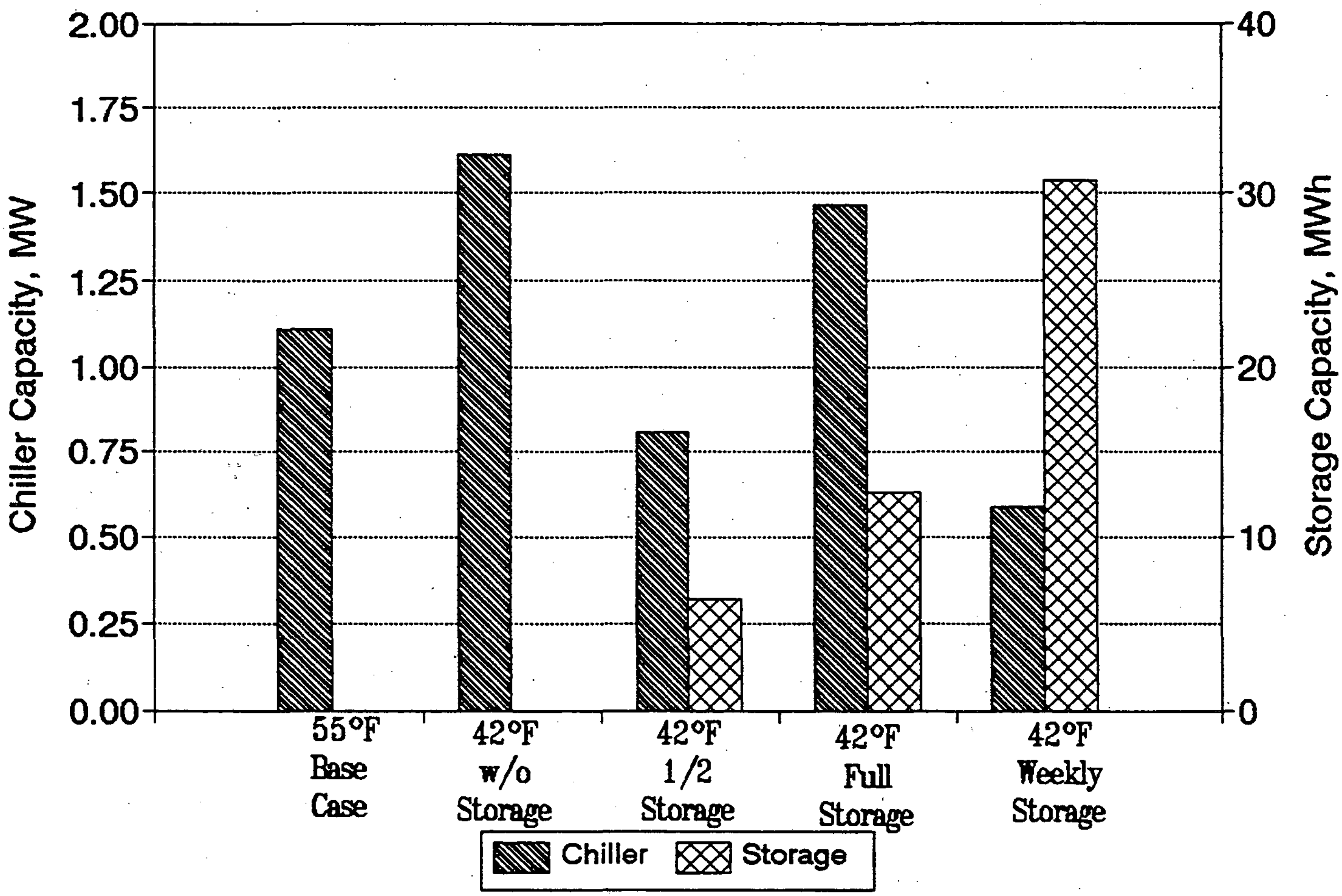


Figure 3b

Installed Chiller & Storage Capacities
Fresno

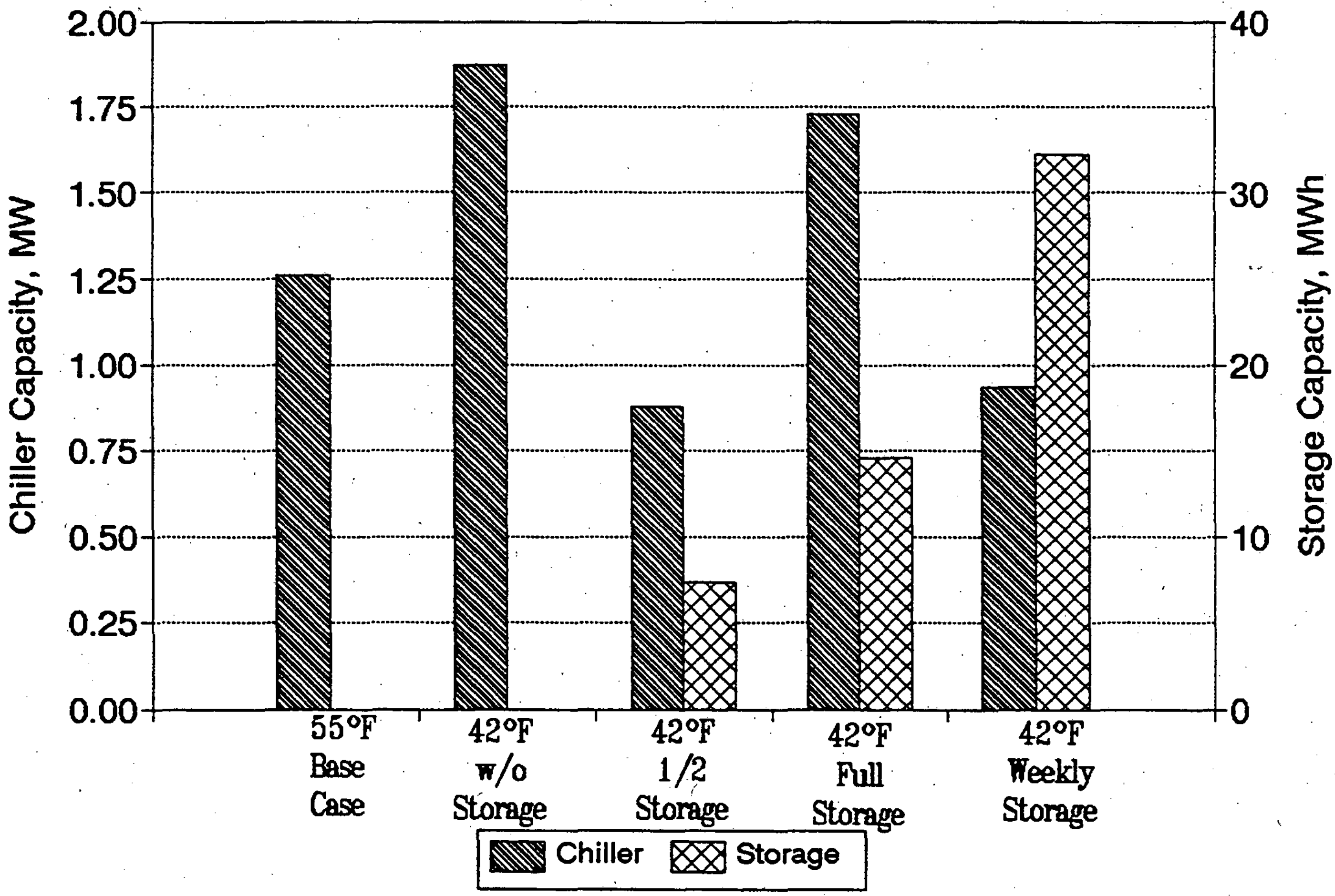


Figure 3c

Installed Chiller & Storage Capacities
San Bernardino

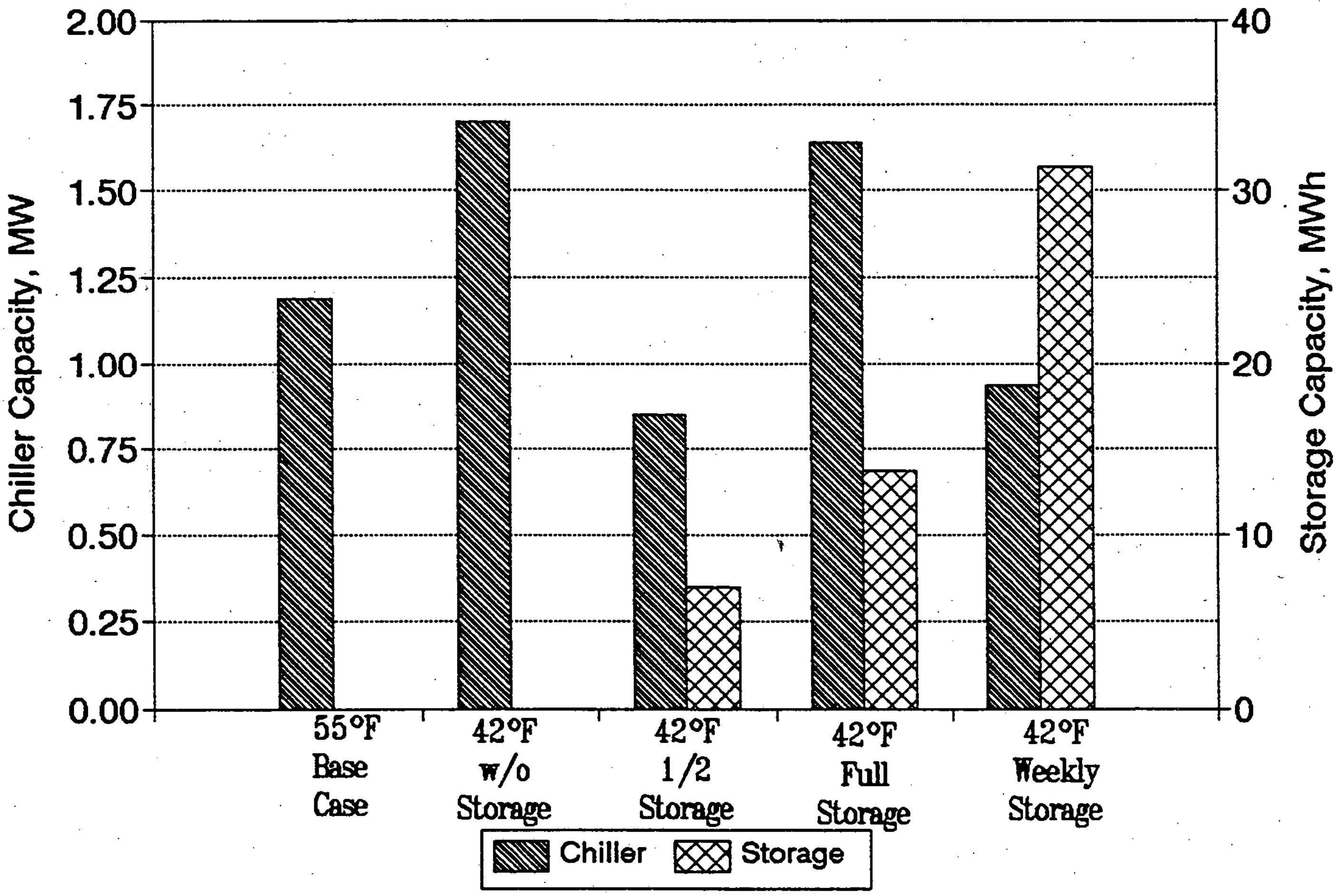


Figure 4
Chiller Monthly COP
Fresno

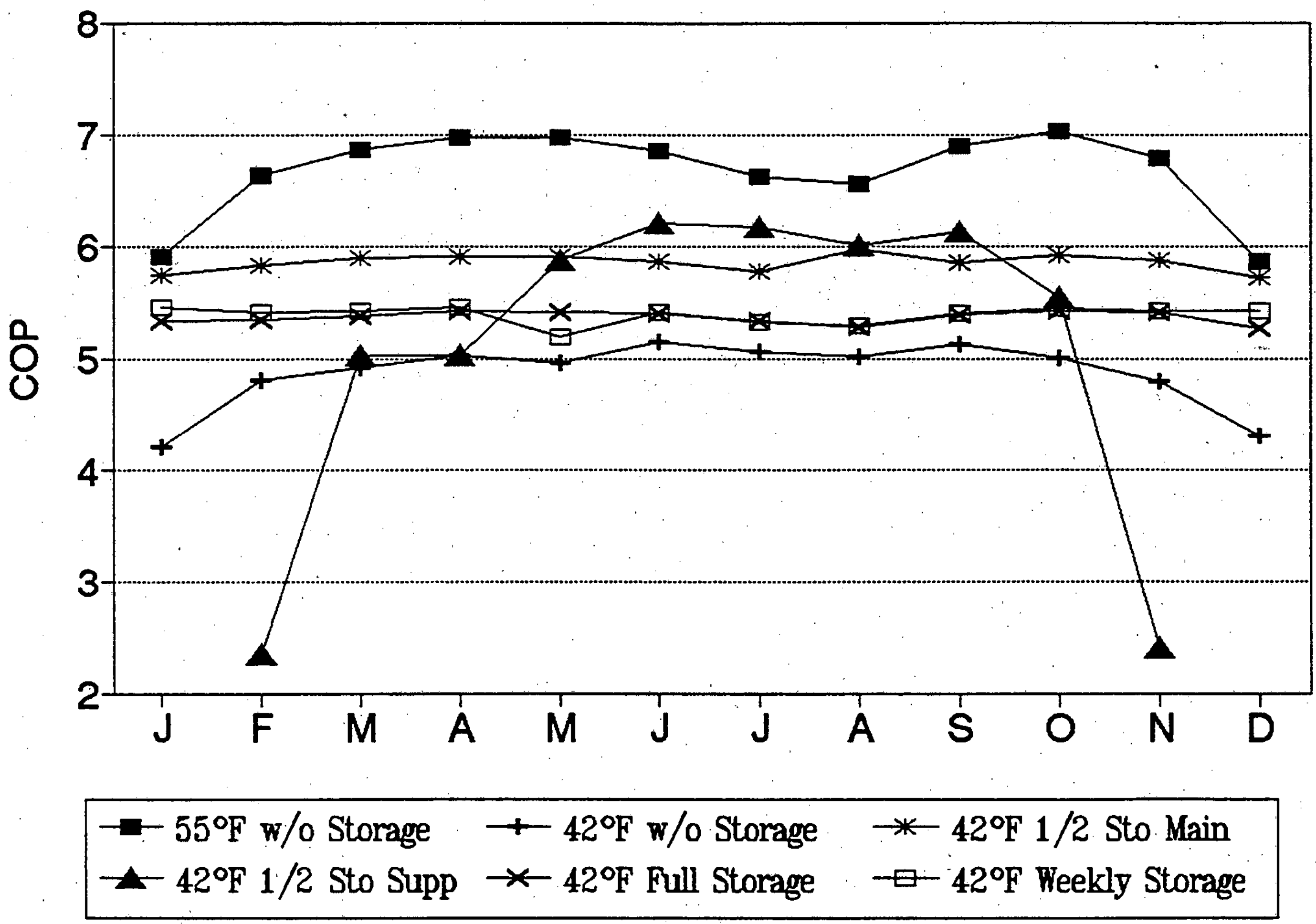


Figure 5

Storage Energy Recovery Ratio, Monthly
Fresno

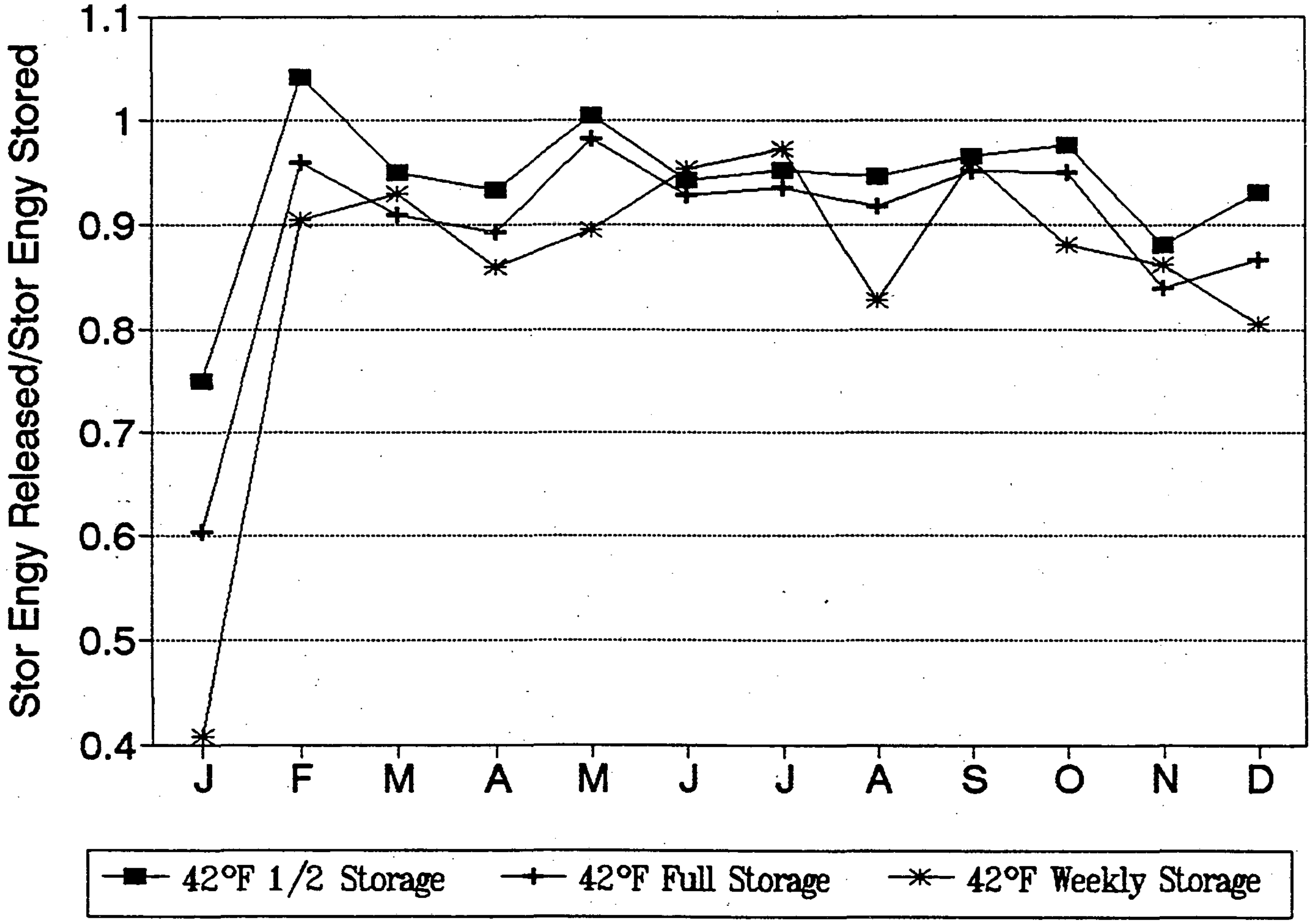


Figure 6a

Effect of Economizer Operation on Annual Cooling and Fan Electricity Use

San Jose

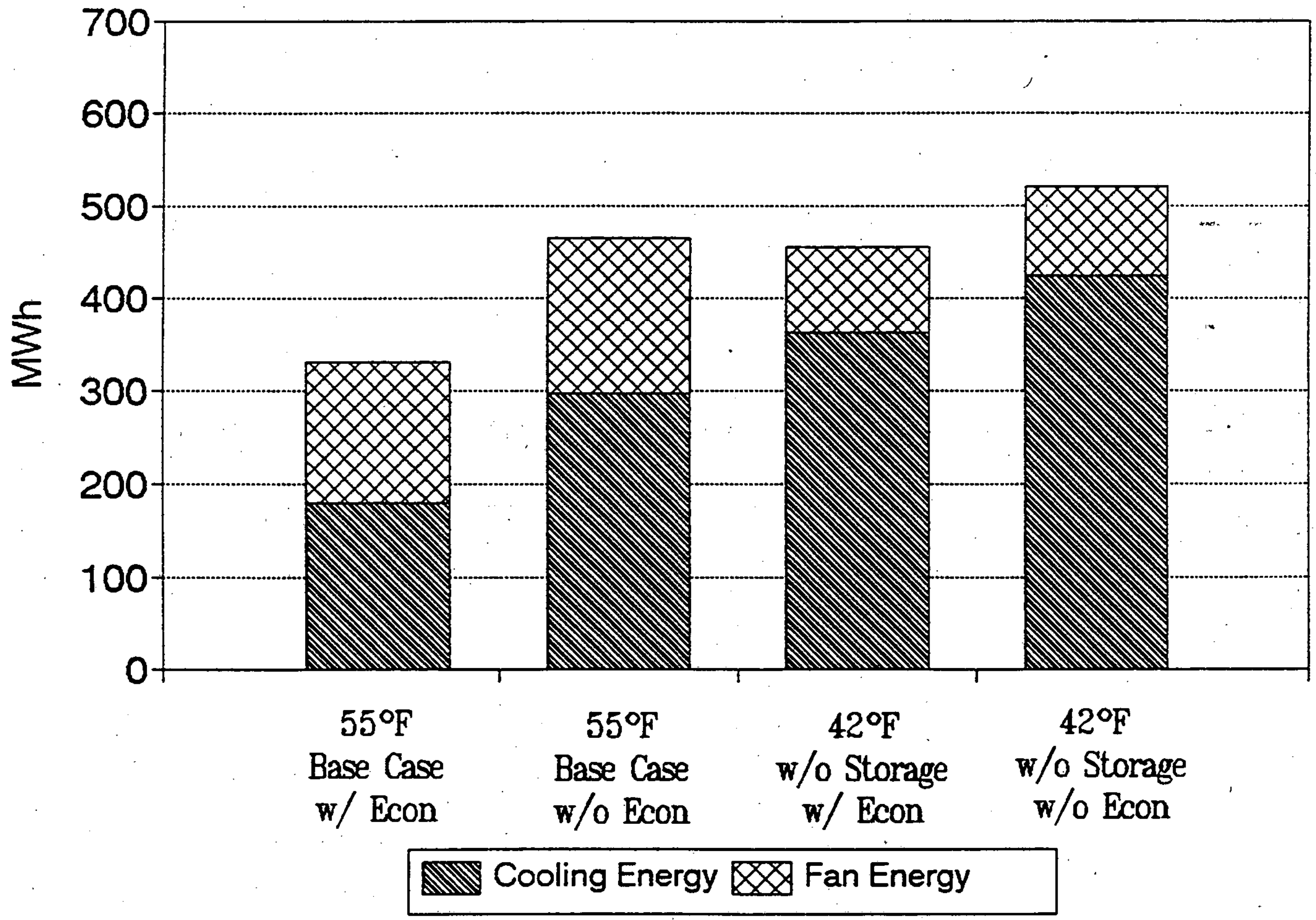


Figure 6b
Effect of Economizer Operation on
Annual Cooling and Fan Electricity Use

Fresno

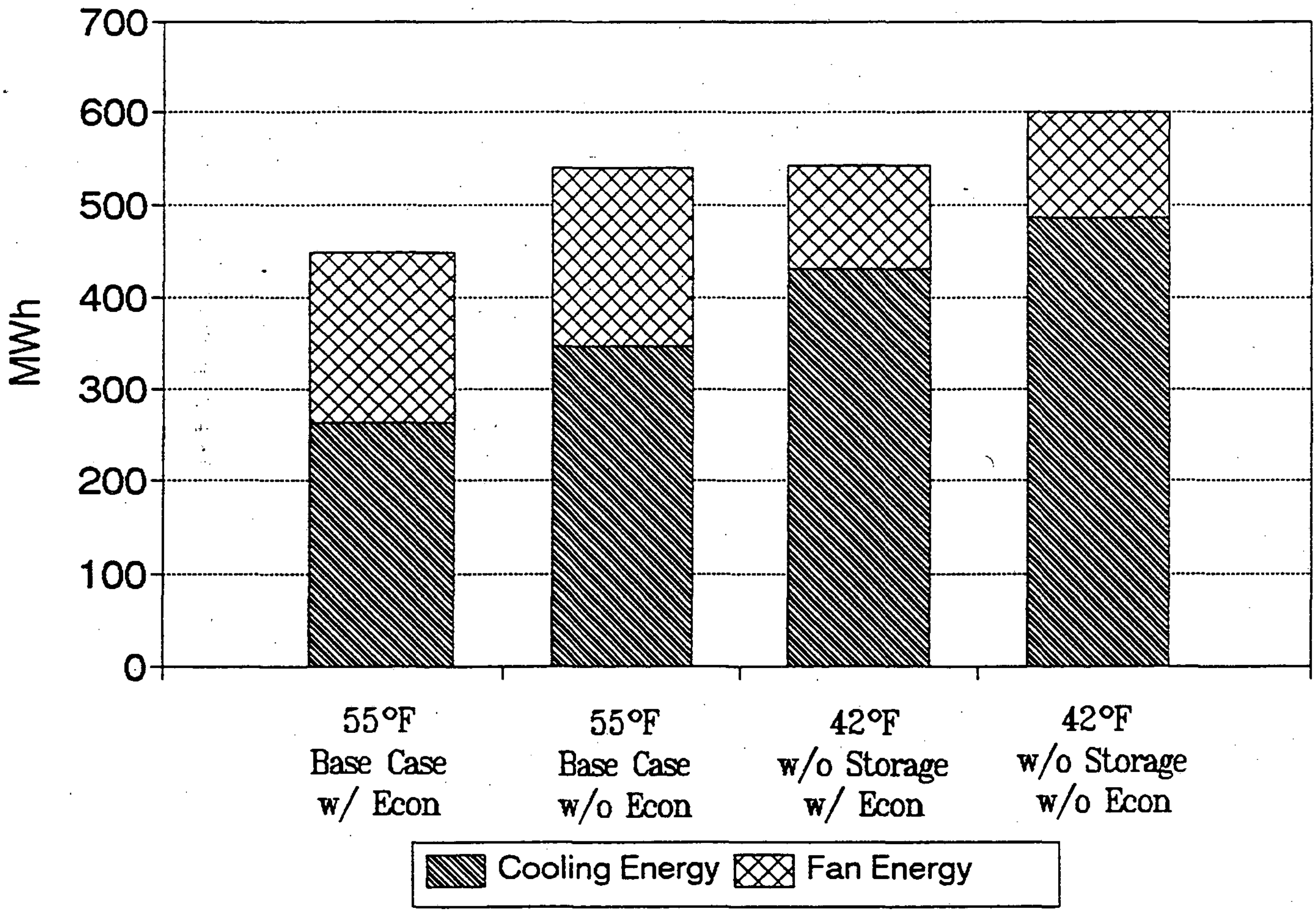


Figure 6c
Effect of Economizer Operation on
Annual Cooling and Fan Electricity Use
San Bernardino

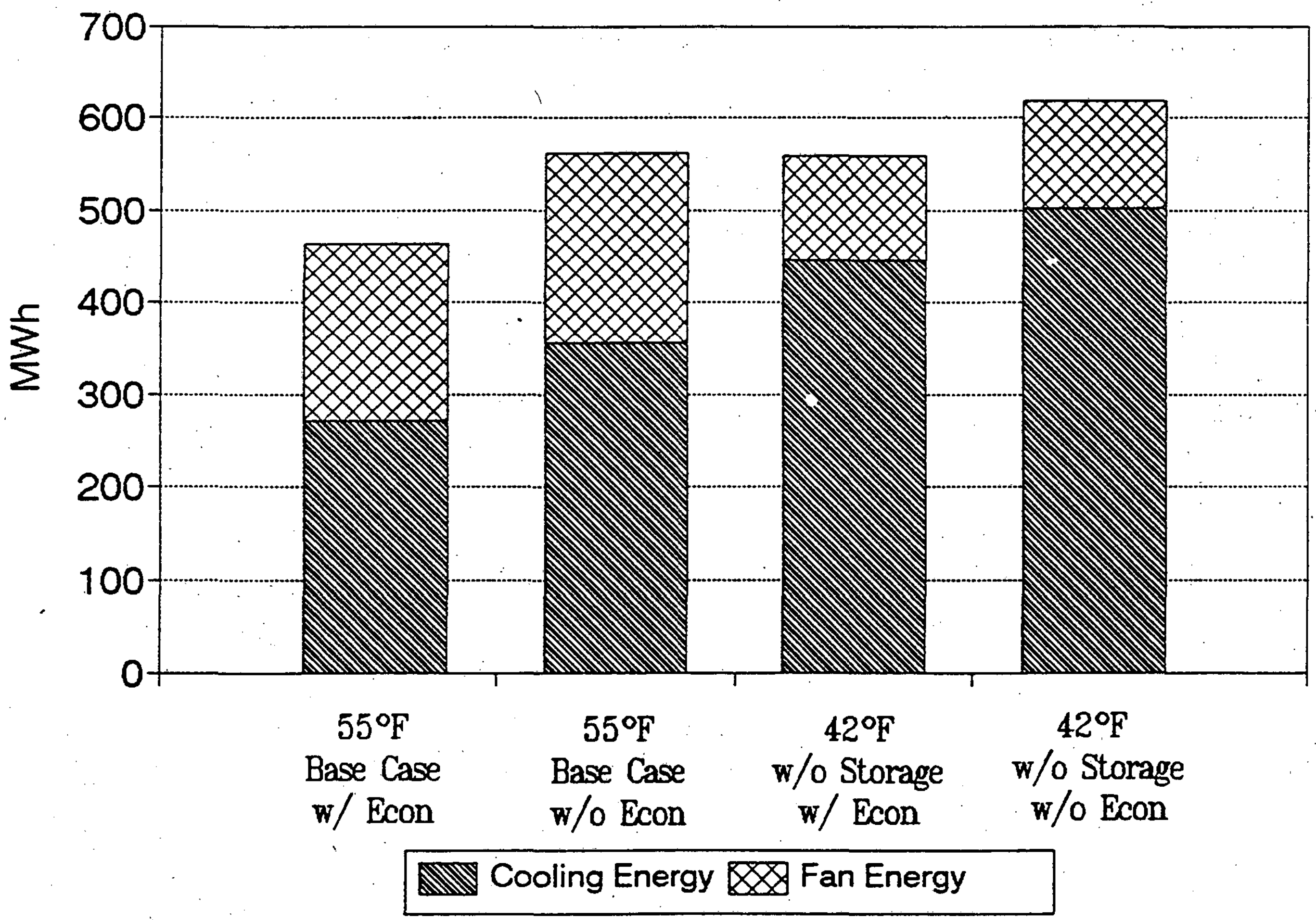


Figure 7 ANNUAL MAXIMUM ECONOMIZER USE
ENTHALPY CONTROLLED

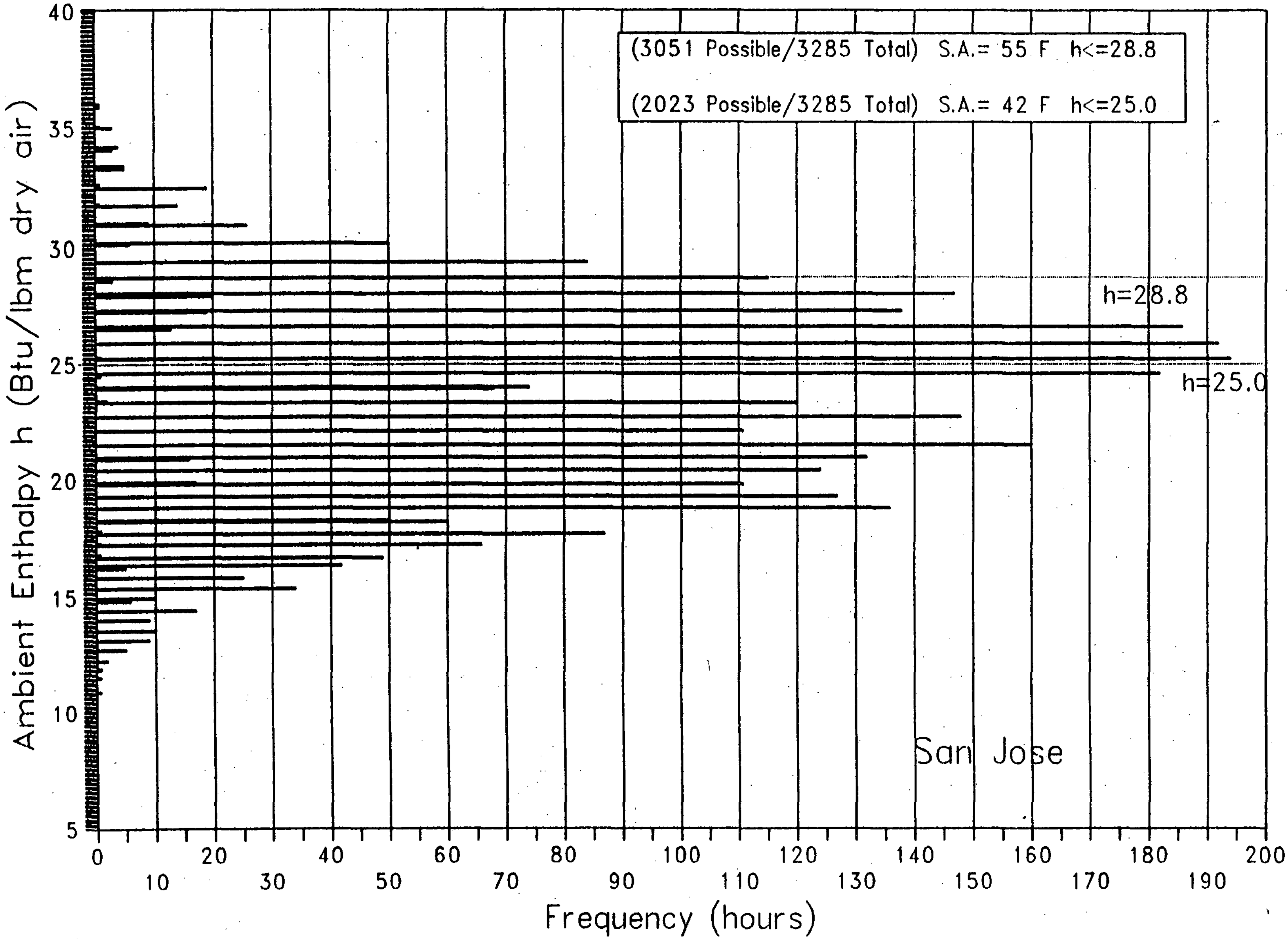


Figure 8 ANNUAL MAXIMUM ECONOMIZER USE
ENTHALPY CONTROLLED

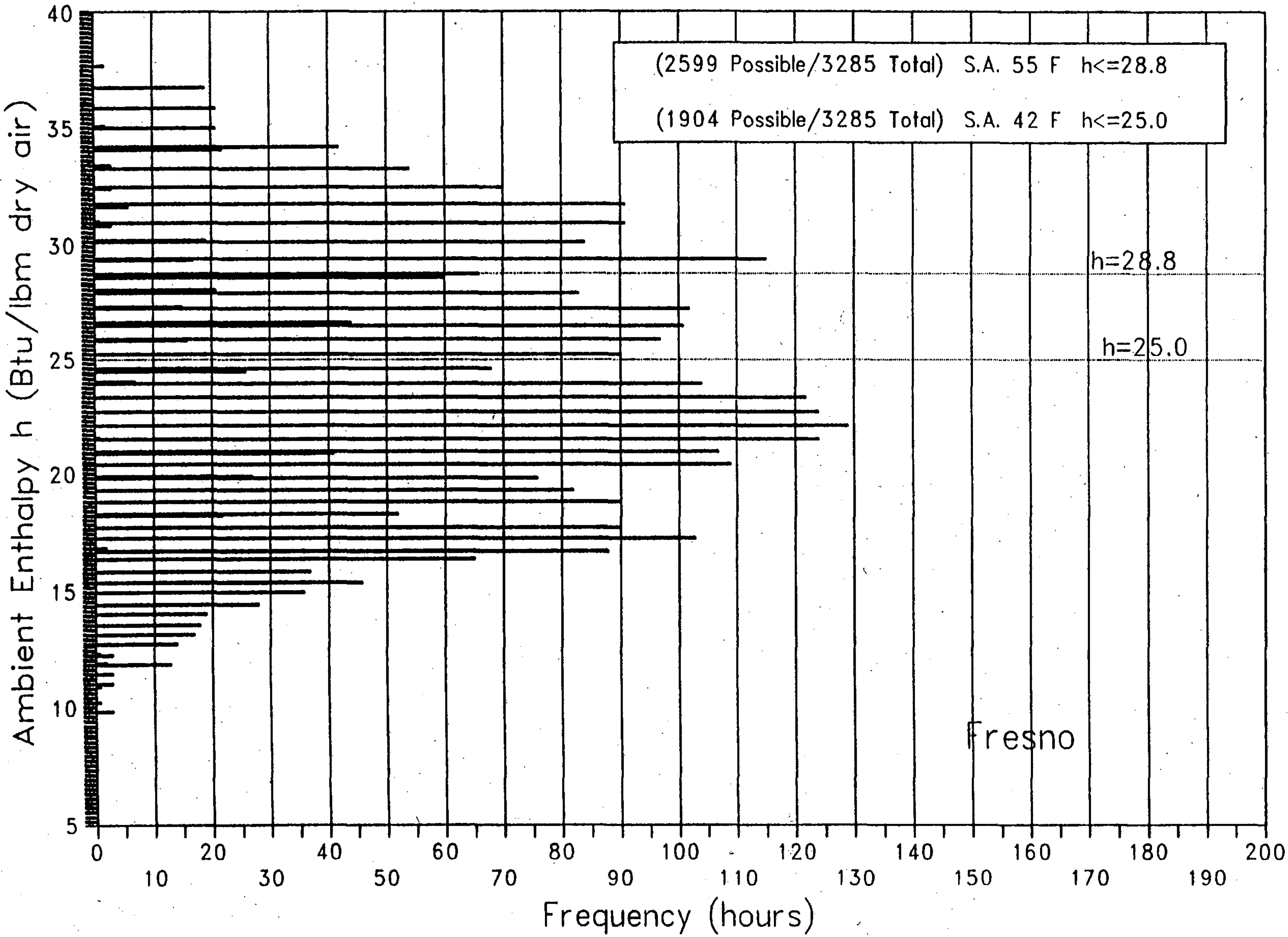


Figure 9 ANNUAL MAXIMUM ECONOMIZER USE
ENTHALPY CONTROLLED

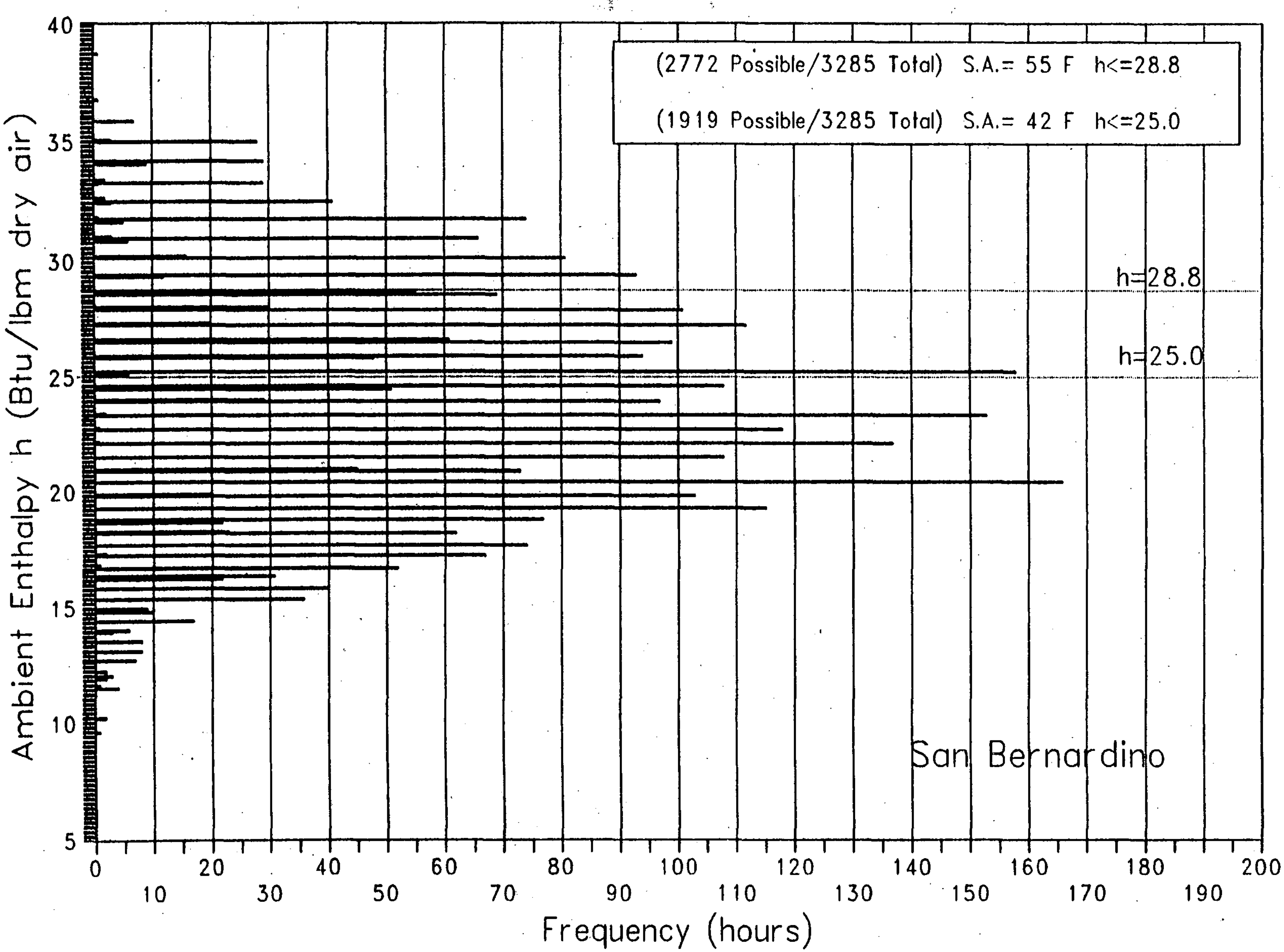


Figure 10

Effects of Fan-Powered Mixing Boxes (FPMB) on Total Fan Electricity Use

San Jose, Full Storage, 42°F

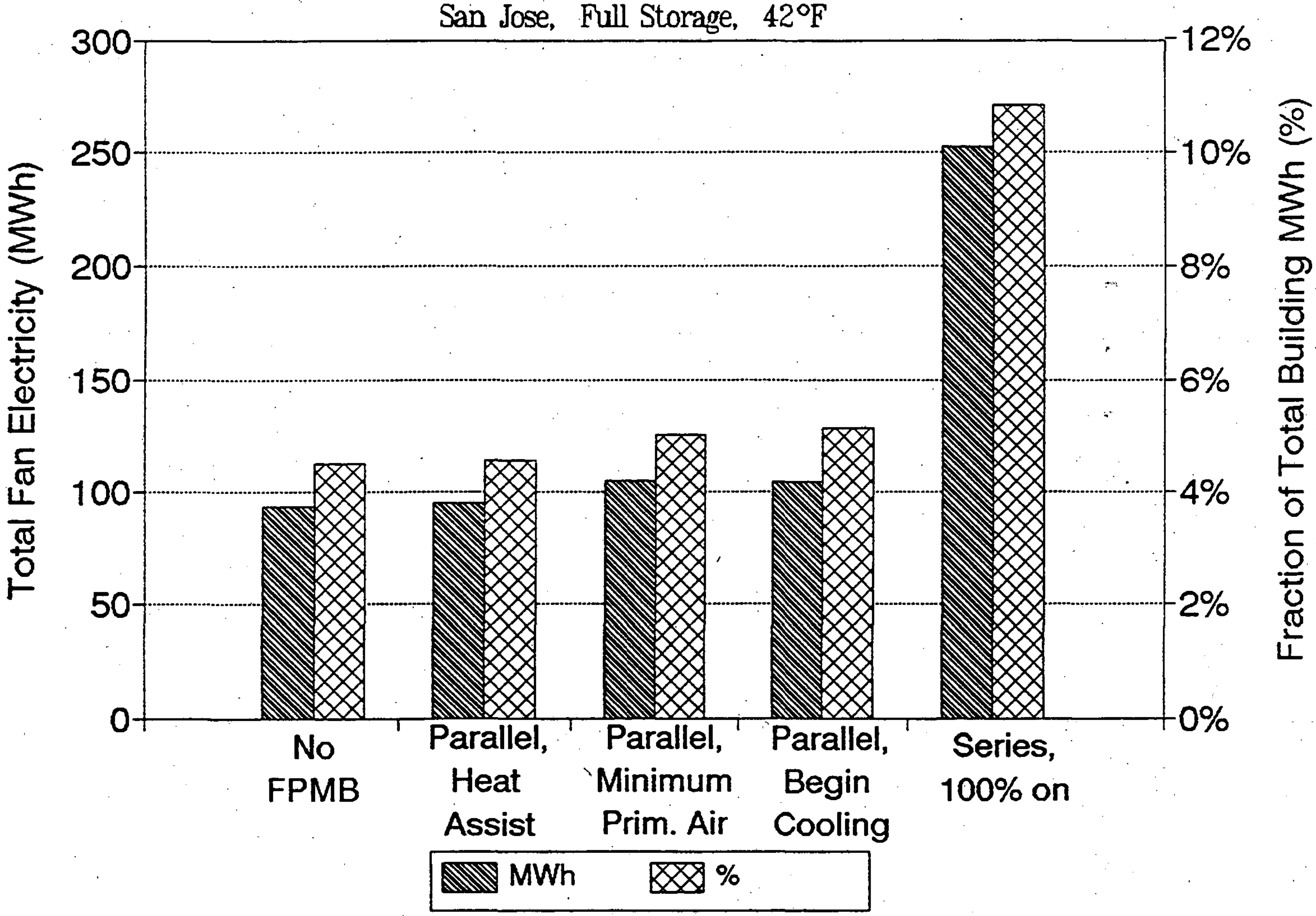
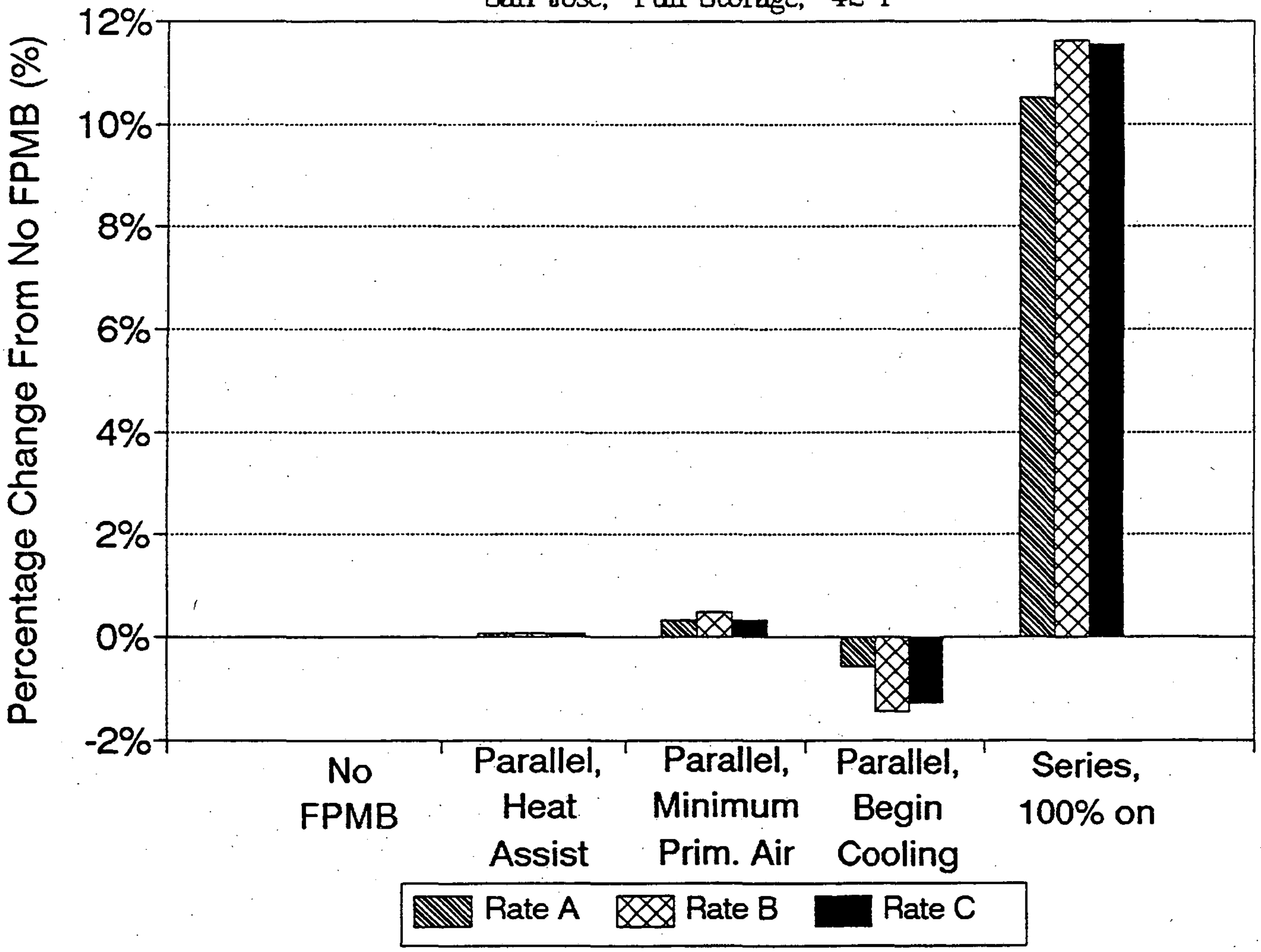


Figure 11

Effects of Fan-Powered Mixing Boxes (FPMB) on Annual Total Building Electricity Cost

San Jose, Full Storage, 42°F



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