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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 the 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 a 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.

A considerable amount of research has been done 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 (Dorgan and Elleson 1987; Merten et al. 1989; Dorgan et al. 1990; Landry and Noble 1991). Whole-building energy simulation studies have investigated 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, 1990a, 1991a; ASHRAE 1990a). Other available publications provide updated summaries of important issues in TES/CoAD technology (EPRI 1990b, 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

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(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 dry-bulb 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·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 analytical model of cold air jet performance. Miller (1991) presents a design methodology for selecting 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 the 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.

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.

In a second part of the study, 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 whole-building energy simulations are presented and discussed in detail below. Only selected findings from the industry survey are included in this paper. For a full discussion of the survey, refer to Bauman et al. (1992).

DESCRIPTION OF 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 constructed buildings in mild California climates such as Los Angeles frequently use 76 to 80 kBtu/ft^2 (870 to 910 MJ/m^2) per year in fuel and electricity, an improvement over many buildings of the older stock that 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.

The prototype building is defined as having 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 62%. 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.

For simulation purposes, each floor 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 a.m. to 5 p.m. Limited use is assumed during the weekends and evenings. Full occupancy is assumed on work days with one occupant for every 200 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/h (21 kW) and elevators that draw 195 Btu/h (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 the perimeter zones. They are equipped with an 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 economical 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 setpoint for zone thermostats is 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 setpoint 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% in the core zones and to 30% 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 that 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.

Case 1—55°F Packaged: 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.

Case 2—55°F Base Case: 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.

Case 3—42°F without Storage: Again, two chillers are simulated, sized as above to provide 42°F supply air on demand with direct injection diffusers.

Case 4—42°F with Half Storage: 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.

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 (Bauman et al. 1992), 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.

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).

The configurations presented here are believed to most directly explore the advantages and disadvantages of using TES/CoAD.

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 p.m. to 6 a.m. are 17% of peak-period demand charges, which apply between the hours of 10 a.m. and 5 p.m. during the summer months. Even during the winter months off-peak demand charges are 72% of peak. Utility B summer demand charges from 9 p.m. to 8 a.m. are approximately 28% of the peak demand charges, which apply between 11 a.m. and 6 p.m. 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 periods of low demand charge which begin weekday evenings at 10 p.m., and is permitted, if needed, to continue to 8 a.m. 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 the six system configurations and two of the three climatic regions (San Jose and Fresno). The simulation results for San

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

Bernardino are not presented here due to similarities with results for Fresno. Refer to Bauman et al. (1992) for a complete set of simulation results. 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 but may be inferred from equipment sizes.

First, the six system configurations 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 during the discussion, three figures are introduced at this time. Figure 1 presents annual cooling and fan electricity use for the two climates, San Jose and Fresno. Figure 2 presents a comparison of the annual total building operating costs for the three selected utility rate structures for the two climates. Figure 3 summarizes the installed chiller and storage capacities for all cases except the packaged system for San Jose and Fresno. 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

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 exceeds 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-h, 6.4 MWh) storage is sized to meet half the design-day cooling

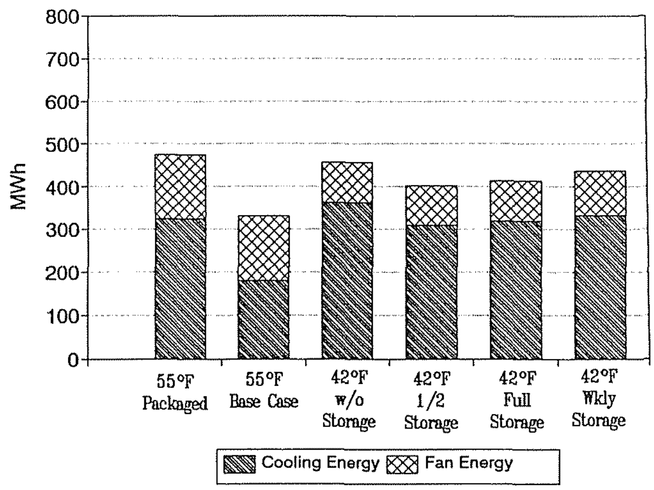

Figure 1a *Annual cooling and fan electricity use: San Jose.*

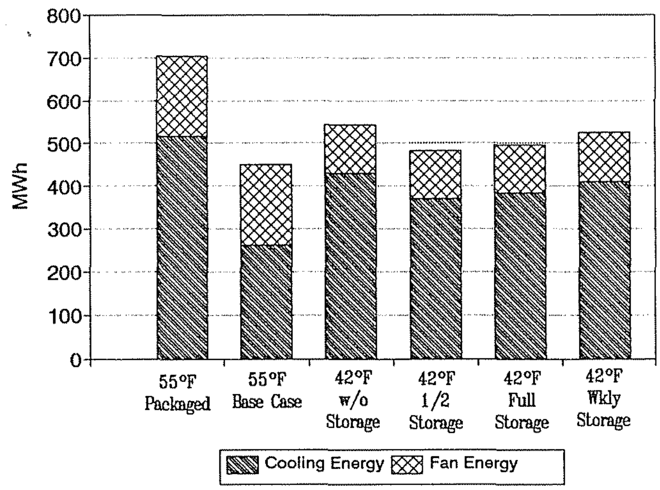

Figure 1b *Annual cooling and fan electricity use: Fresno.*

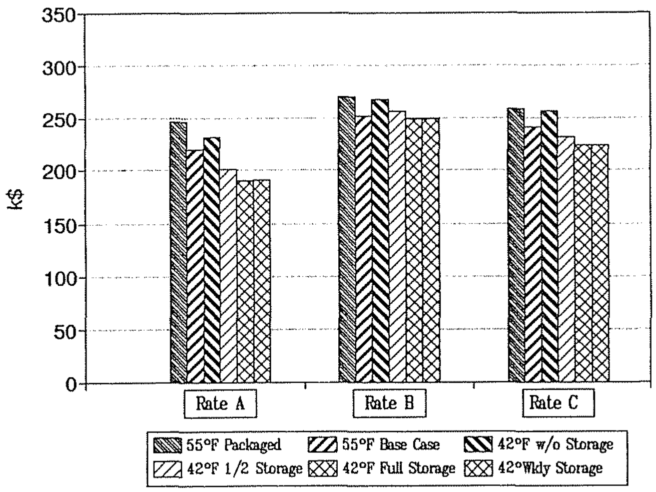

Figure 2a *Annual total building electricity cost: San Jose.*

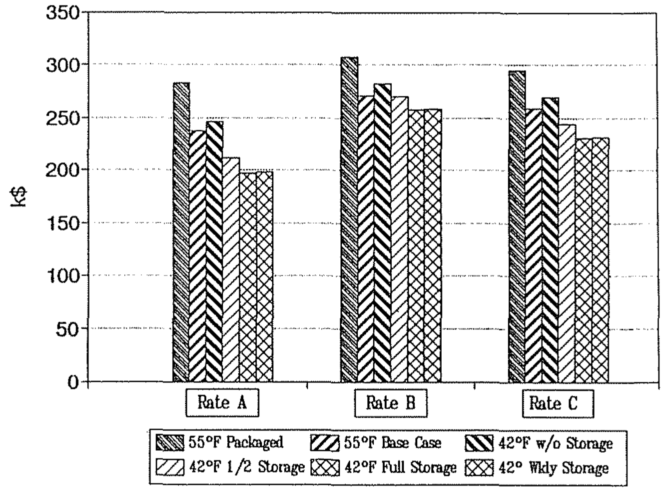

Figure 2b *Annual total building electricity cost: Fresno.*

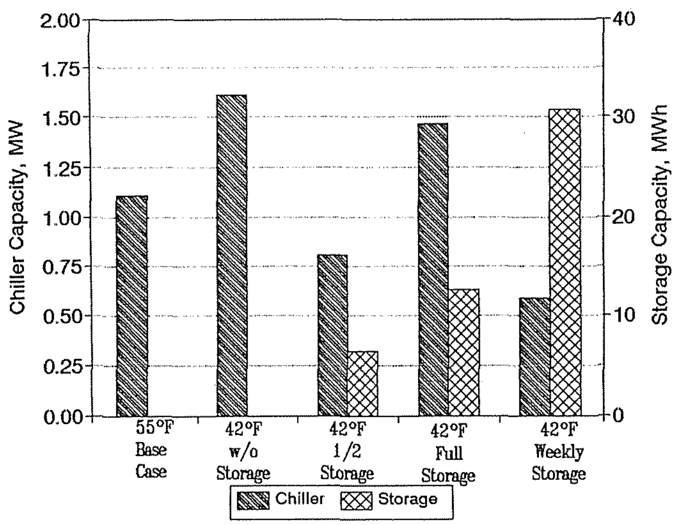

Figure 3a *Installed chiller and storage capacities: San Jose.*

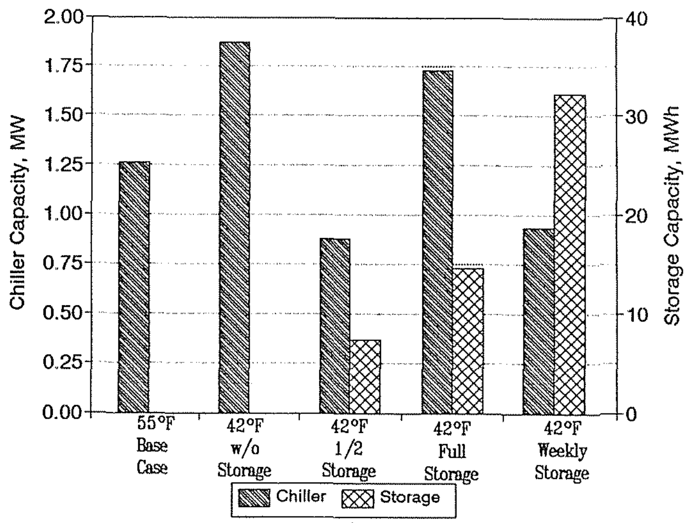

Figure 3b *Installed chiller and storage capacities: Fresno.*

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/h (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-hr, 30.8 MWh) is required to satisfy 100% of the cooling demands but loses approximately 9% of generated cooling capacity annually.

The significance of utility rate structures to offer encouragement to shifting building electricity use to off-peak periods-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 an 8.7% decrease in electricity cost, full storage in a 13.5% decrease, and weekly storage in a 13% decrease below the base case. The packaged system raises electricity costs by 11.6% and the 42°F no-storage case raises them 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 raises them 6.5% above those of the base case. Electricity costs based on utility rate C show trends similar to those for utility rate B.

Fresno

Fresno represents an inland region of greater dry-bulb temperature swings and generally lower relative humidities than San Jose. 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, the 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 the base case.

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

Figure 4 displays monthly chiller 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; 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

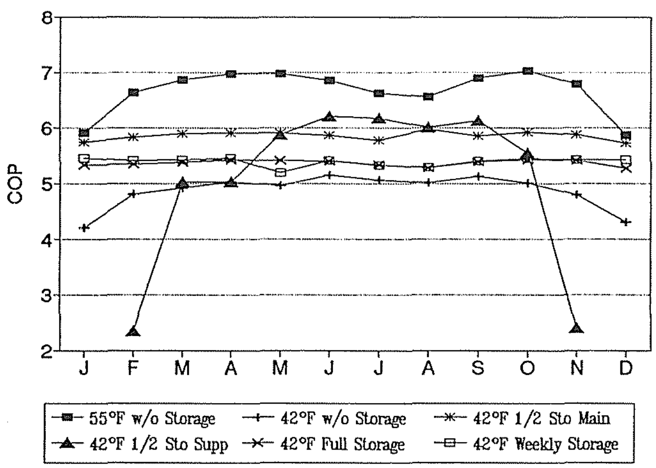

Figure 4 Chiller monthly COP: Fresno.

cases because heat can be rejected from a cooling tower that 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 that 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.

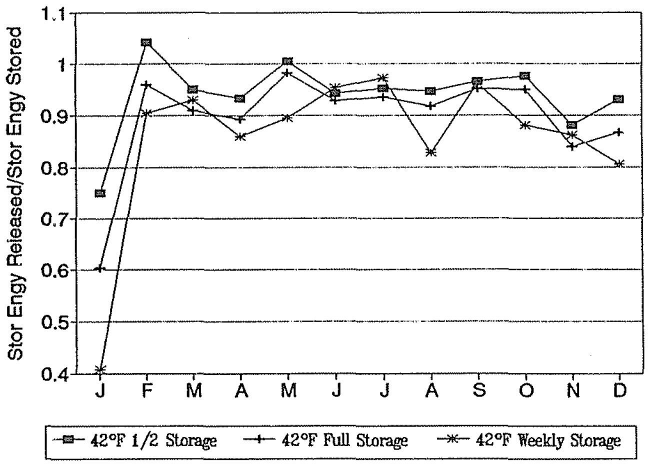

Figure 5 *Storage energy recovery ratio, monthly: Fresno.*

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 that 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 that 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 further reduced 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 two climates, San Jose and Fresno, 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 an 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 utility 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, the 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

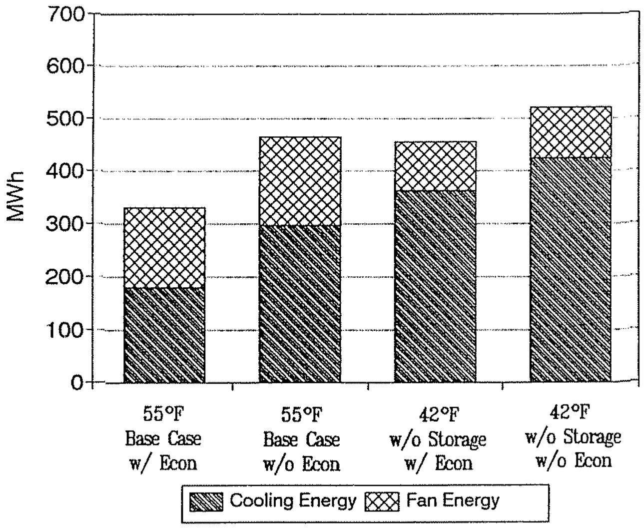

Figure 6a *Effect of economizer operation on annual cooling and fan electricity use: San Jose.*

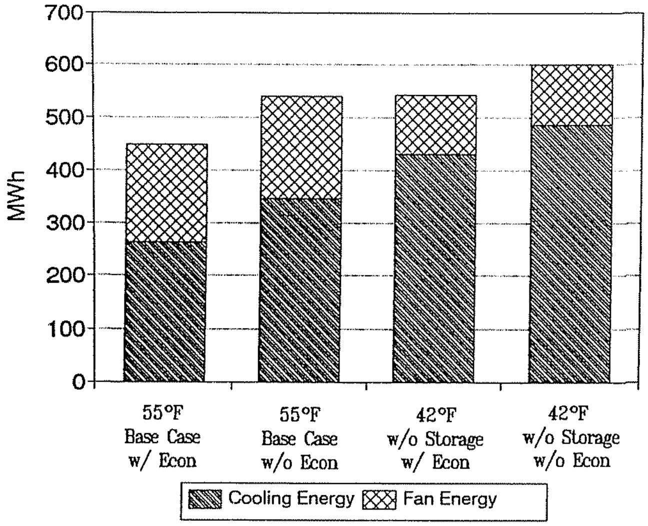

Figure 6b *Effect of economizer operation on annual cooling and fan electricity use: Fresno.*

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 not using an 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 not using a 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. 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 seven days per week. The ratios do include hours during which the building does not need cooling. Between the horizontal 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 in 93% of the normal occupancy periods if 55°F supply air is specified and in 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% 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 that 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 that the fraction of business hours when economizer use is possible using 55°F air is 79% and with 42°F air is 58%.

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

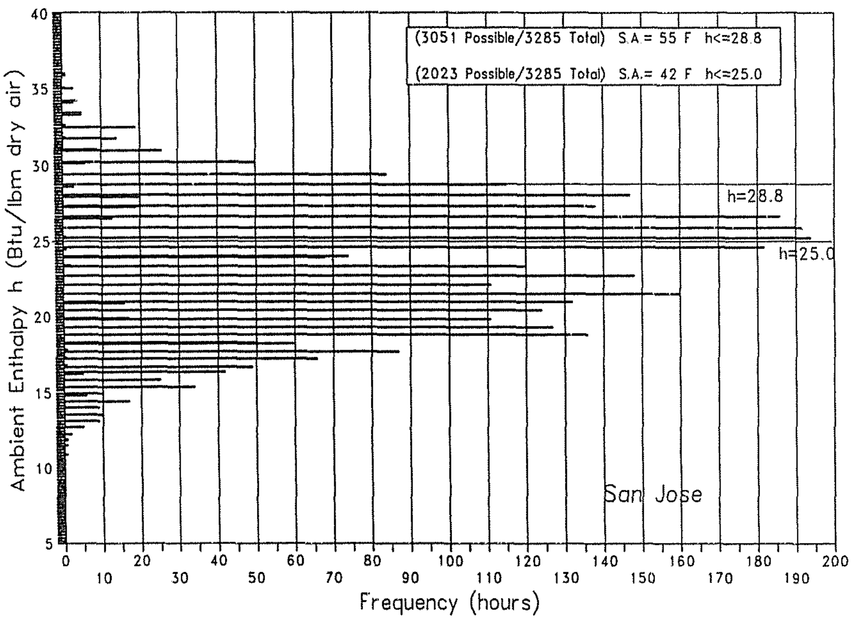

Figure 7 *Annual maximum economizer use, enthalpy controlled: San Jose.*

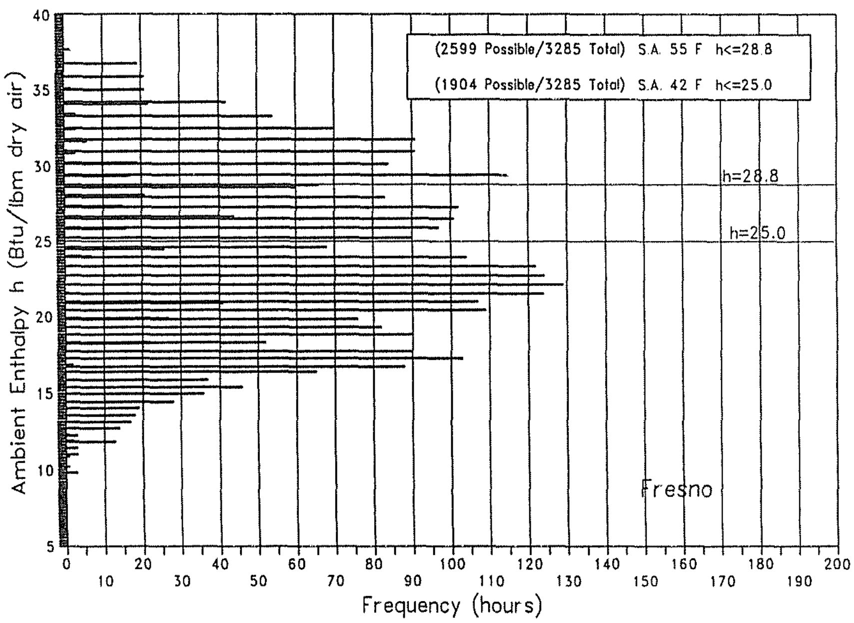

Figure 8 *Annual maximum economizer use, enthalpy controlled: Fresno.*

low-temperature case and the 55°F no-storage case when neither uses economizers. The total building kWh decreases by a negligible 0.2% for the storage case, but the peak demand decreases by 31%, and 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.

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 airflow 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 the 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 three simulations used a parallel FPMB configuration, which does not handle primary airflows but supplements primary air to diffusers with plenum air that may or may not be passed across an active heating coil. All parallel units were sized to handle 80% of the maximum primary airflow 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 that 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% of the primary airflow. 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 9 presents the simulation results in terms of fan energy (MWh) and fan-attributable fraction of total building electricity consumption. Figure 10 presents the effects of FPMB operation on total annual 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, and total building electricity use and utility costs increase 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 utility B and C declined by 0.9%. The cost reduction originates primarily in the reduced chiller energy, which, as men-

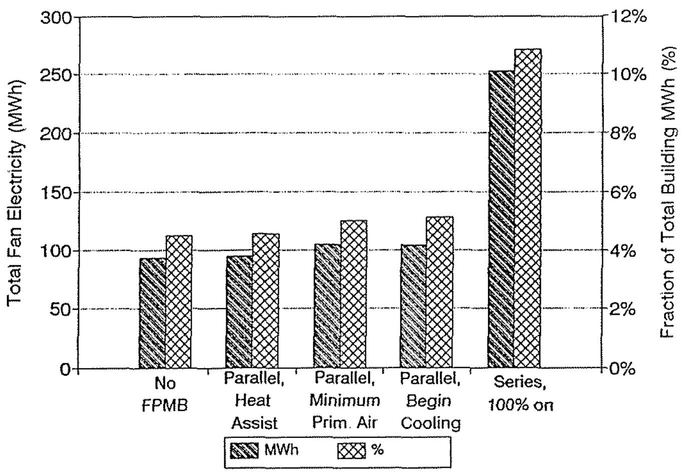

Figure 9 *Effects of fan-powered mixing boxes (FPMB) on total fan electricity use: San Jose, full storage, 42°F.*

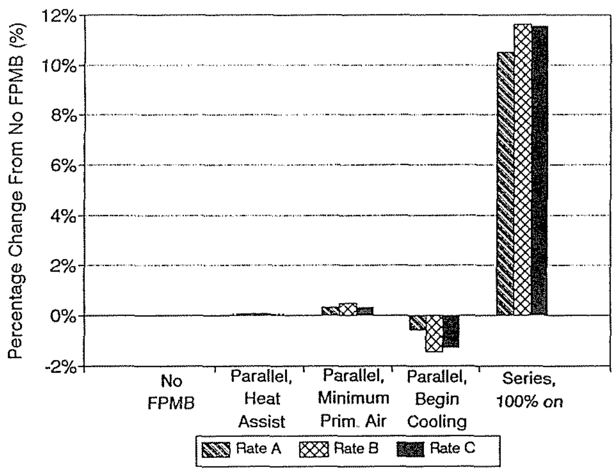

Figure 10 *Effects of fan-powered mixing boxes (FPMB) on annual total building electricity cost: San Jose, full storage, 42°F.*

tioned, 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, and 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 9 and 10 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 interviewing 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 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 survey appears in Bauman et al. (1992). 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 Whole-Building Energy 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 compiled a list of current California projects involving cold air distribution (see Bauman et al. 1992).
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 was 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 simulation results. In the following section, we discuss a few of the important factors influencing the development of this technology in California.

Factors Influencing the Development of Cold Air Distribution 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.

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 (4.4°C to 10°C) 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 (7°C) 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.

Fan Energy 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. 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 [13°C to 18°C]) 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).

Economizers High-rise HVAC design practice has changed over the past five to ten years in response to changing tenant needs. As multi-tenant office buildings are now the norm, most design engineers are installing floor-by-floor air-handling units (AHUs) on all of their high-rise jobs. 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 can 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 will be rejected by the architect. The implications of these considerations are that the majority of high-rise buildings 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).

Controls and Operation Most of the engineers 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.

Most of the consulting engineers with whom we spoke emphasized the significance of building commissioning. Commissioning is important to the successful operation of any air-conditioning system but, 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 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 start-up and the initial charge of the storage system.

In addition to the topics discussed above, the industry survey addressed (1) equipment for cold air distribution, (2) moisture and condensation, (3) system design engineering approaches, (4) building codes and standards, and (5) utility incentives. Refer to Bauman et al. (1992) for a full discussion of the survey findings.

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 three-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. Factors influencing the future development of cold air distribution were identified and discussed.

The major conclusions from the whole-building energy 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 conventional component-assembled 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 other 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 high-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. 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.

Issues for further research are listed below. 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.

1. Develop improved methods for supplying low-temperature air to occupied spaces.
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 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.

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