

Simulation of Source Energy Utilization and Emissions for HVAC Systems

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ABSTRACT

An office building and a school building were used to investigate the source energy requirements and environmental emissions attributable to the operation of space conditioning systems serving facility cooling requirements. The hourly electrical and gas demands for a range of space conditioning system types (conventional electric chiller, chilled water storage, ice storage, ice storage with low temperature air, absorption) were traced back to the point where fuel is extracted from the earth (i.e., the source). All electrical consumption was assumed to be generated and delivered by the last utility plant (marginal) dispatched to meet the aggregate electrical demand in two different utility service territories. The two utilities, with different generation mixes and marginal dispatch plans, formed the basis to assess the source energy requirements and emissions associated with electrical usage in the present analysis. Marginal unit dispatch was determined on a lowest operational cost basis, provided with the utility data.

On an annual basis, thermal energy storage systems had lower source energy requirements and resulting environmental emissions than a base case of an electric direct chilling system. Emission results in all categories were lower than the base. Chilled water systems tended to have the best performance, but ice storage with cold air also performed very well. Absorption systems tended to have lower NO_x and SO_2 emissions; however, they had the highest source energy consumption, the highest CO_2 emissions, and the highest Global Warming Index (GWI) compared with all other options.

BACKGROUND

Internationally, as well as domestically, more attention is being focused on the environmental impact of systems that

generate electricity as well as those that consume the electricity. In the United States, the Clean Air Act required reductions in power plant emissions that are contributing to potential global climate changes and acid rain, e.g., CO_2 , CH_4 , SO_2 , etc. In addition, the Montreal Protocol dictated an aggressive phase-out plan for chlorinated fluorocarbons (which directly affect the vast majority of installed cooling capacity throughout the world). Through the Kyoto Protocol, the United States has committed to achieving a 6% reduction in its production of 1991 levels of greenhouse gas emissions by 2005. Such environmental legislation could have repercussions on both the electric supply side and electric demand side.

ASHRAE has made a commitment to understanding how space conditioning technologies impact our environment.¹ To that end, ASHRAE sponsored RP-991 to gain a better understanding of the relative energy and environmental impacts of alternative space conditioning technologies. This paper presents the methods used and the findings from the research project.

In the United States, 90% of the commercial cooling is accomplished using electricity as a primary energy source while only 10% of commercial cooling is accomplished using natural gas as the primary energy source (EPRI 1993). In today's energy marketplace, conflicts are arising due to the ever-increasing demand for and consumption of energy (both gas and electricity). Much of the growth in electricity demand has occurred during peak periods (daytime) and is attributable to air-conditioning loads. Further clouding the picture is the continuing evolution of a competitive electric marketplace.

¹. An excerpt from the ASHRAE bylaws states: "To fulfill its role, the Society shall recognize the effect of its technology on the environment and natural resources to protect the welfare of posterity."

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Utilities are searching for energy solutions that provide the ability to meet the customer's needs (such as providing energy to afford good indoor environmental quality), maximize their profits, and meet the emission benchmarks established by the government. How can we meet the environmental emission reduction commitments if our use of energy continues to escalate?

Two space conditioning technologies that have purported to minimize on-peak electrical demands and reduce the resulting environmental emissions are gas-fired absorption chillers and electric chillers using thermal energy storage. Historically, these technologies have benefited both end-users and utilities since absorption chillers and thermal energy storage have the ability to reduce aggregate electric demand during peak periods. Thermal energy storage provides for the potential reduction of first cost and almost certain reduction in operating costs, as compared to a conventional system without storage meeting the same building cooling loads; however, some have criticized the technology, claiming that it does not reduce source energy consumption. Absorption chillers will have lower electrical energy consumption (compared to electric chillers), but the total energy requirements (electric and thermal) and emissions produced have not been clearly quantified.

INTRODUCTION

Thermal energy storage systems typically produce cooling capacity during the nighttime hours by charging a storage medium (ice or chilled water). During the daytime, the stored energy is available to be discharged on demand to meet the building's cooling loads. As such, a portion (and in some cases, the entirety) of the building's cooling load, and thus the building's energy consumption attributable to the refrigeration portion of the cooling, can be shifted from daytime to nighttime hours. In contrast to direct-chilling systems that often operate at poor part-load efficiencies during a majority of the year, storage systems have the advantage of operating at peak efficiency during the charge cycle. Ice storage systems have an added penalty of operating at the low temperatures needed to make ice; however, application of low temperature air distribution in conjunction with ice thermal energy storage systems mitigates the energy penalty for producing ice with reduced fan energy.

The environmental impact attributable to electricity consumed for cooling is linked to the output of the generating utility. The emissions and fuel requirements of an electric utility will vary over the course of the day and over the time of the year, based on the characteristics (fuel type, heat rate, part-load operation, et al.) of the various types of generating plants dispatched to meet the aggregate system electric demand. Utilities typically dispatch generation to meet the system load in the most economical manner possible; that is, they run the least cost (which are generally the most energy efficient) plants first and then successively add higher cost (less efficient) plants as the aggregate demand warrants. During the

day, a utility will likely have to bring on less efficient equipment in order to meet their peak demand. At night, a utility will be able to meet the aggregate electric demand by running their more efficient "baseload" plants. Another ancillary environmental impact attributable to building electric consumption is the energy losses due to electric transmission and distribution. These losses are typically proportional to utility grid load (TU 1993), resulting in higher losses during the daytime and reduced losses during the nighttime hours.

The term "source energy" can be defined in a number of different ways. The term is certainly different than "site energy," which is commonly understood to be the energy actually consumed at the building. In the context of this paper, we define source energy as the energy extracted from the earth (fuel) to accomplish the intended function—comfort cooling. To determine source energy requirements for comfort cooling, we track and account for the energy consumed from the point where fuel is extracted from the earth (the source) through all the intermediate steps to the site where energy is ultimately converted to cooling.

One of the first attempts at establishing a methodology for characterizing source energy associated with various processes was performed by Melcher et al. (1976). The study developed "energy trajectories" for various fossil fuel energy systems. The energy trajectories were part of a net energy analysis. This type of analysis attempted to encompass the fossil fuel steps of extraction, processing, and delivery, as well as to estimate the energy flows attributable to providing the materials used in constructing the equipment as required in each step. The Melcher et al. analysis did not take into account any seasonal effects in distribution efficiency or additional conversion processes, i.e., cooling.

Reindl et al. (1995) looked at the source energy consumption for an office building being served by a single electric utility. The analysis showed that thermal energy storage technologies resulted in lower source energy consumption as well as lower carbon dioxide emissions compared to non-storage systems.

A recent study funded by the California Energy Commission (CEC) (Tabors Caramanis & Associates 1996) specifically looked at the source energy impacts of thermal storage. The study attempted to determine the potential source energy impacts attributable to a large number of thermal storage systems operating within two California utility service territories. As a part of their study, the investigators considered two different methods of assessing source energy and emissions—an incremental method and a marginal method. Their incremental approach utilized a modified average heat rate and emissions characteristic for a utility mix. The marginal method used a "system lambda" to account for the marginal cost per kWh to serve an increased or decreased load.

The CEC study analyzed three different building types—a hospital, a large office, and a small office. The study considered full-shift thermal storage systems, and they further presumed that there would be no net site electrical energy

savings from thermal storage systems compared to non-storage systems. The study indicated source energy savings of 30% using their incremental method and savings of 10% using the marginal method.

METHODOLOGY

The source energy and emissions associated with comfort cooling depend, in part, on the magnitude and timing of energy use at the site. It will also depend on the energy losses due to transmission and distribution, as well as the type of electrical generating equipment online at a given time. As such, this site energy is traced back to the origin.

Several alternative methodologies for quantifying source energy requirements exist. One approach is to determine the mix of electric generating plants operating during a given hour. The site energy consumed could be expressed as a fraction of the composite utility generation operation. The source energy and emissions associated with that site's energy use could be calculated based on that fraction and knowing the composite energy consumed and emissions produced from all plants in the entire utility service territory. The analysis conducted during this project considered an alternative approach. The building space conditioning system's electrical demand is assigned to the last unit of generating equipment dispatched, i.e., the marginal plant. All of the source energy and emissions were considered to be produced by that marginal plant.

In order to identify the marginal plant for a given hour, it is necessary to know what plants are online during that hour. Utilities typically operate their least cost (more efficient) generating equipment to meet their base load, and then progressively higher cost (less efficient) units are dispatched as the load increases. Actual details of hourly dispatch information were not available from either of the utilities employed in this study; however, operation cost information, heat rates, fuel source, and plant size for each plant in the generation mix were available. With this information in hand, dispatches are assumed to occur on the basis of operating cost. Generating plants are dispatched, beginning with the least cost unit, until the system load is met. The last unit dispatched is considered the marginal unit for that hour.

Data were obtained from a number of utilities providing electricity in Wisconsin. Different generation mixes characterized these two utilities. By combining this information with annual utility system load data on an hourly basis, a marginal plant for every hour could be assigned.

The raw data for the two utilities were not in the same format. Plant emissions were expressed in terms of pounds of pollutant per MWh delivered to the plant boundary. Utility #1 had heat rate information at three different levels (low, medium, and high loads) for each plant; the other, Utility #2, only had an average heat rate for each plant. For Utility #1 data, a weighted average heat rate for each plant was determined. The average heat rate was compared to the reported heat rates at the given load levels. Since the plants had data

reported at various low and medium levels, distributions of heat rates as a function of part-load ratio could be developed. These heat rates were then expressed as a fraction of the average heat rate and as a function of the plant's part-load ratio. This procedure was applied for both types (coal and gas) of marginal power plants. Part-load operating curves were established for each type of generation used in the two utility service territories.

ENERGY TRAJECTORIES

The energy trajectories developed by Reindl et al. (1995) were incorporated into the present analysis for the different types of generation and distribution.

Characterizing the transmission and distribution losses on an hourly basis is quite difficult. A previous study by the authors (Reindl et al. 1995) used information from a utility (TU 1993) as typical for the present utility service territories. Expressed in terms of percent, the losses are characterized as follows:

$$\text{Losses} = 6.5 + 1.75X^2 \quad (1)$$

$$X = kW/kW_{peak} \quad (2)$$

where kW is the hourly electrical demand and kW_{peak} is the peak electrical demand over the year.

As mentioned previously, the Melcher et al. (1976) analysis did not take into account variations in the natural gas pipeline distribution efficiency throughout its seasonal operation. The delivered volume of gas is typically higher in the winter than in the summer. Therefore, it is reasonable to expect that the overall process efficiency would vary over the course of the year.

The Energy Information Administration (1992) data were used in order to characterize the efficiency of the gas distribution process. These data contained national values collected from the years 1989-1991. An average value was generated for each month of the year, and the subsequent average curve was fitted to the following functional form:

$$\text{Efficiency} = 90.12 + 2.065 \cos(2\pi Z) \quad (3)$$

$$Z = (M - 1)/12, \quad (4)$$

where M represents the month ($M = 1$ for January, $M = 2$ for February, ...).

Natural gas is used for two different purposes in this investigation. It is used as a fuel source for direct-fired gas cooling equipment at the site and also for electrical power generation by the utility.

Figure 1 shows the trajectories of energy and emissions associated with production of electricity by natural gas and coal-fired plants. The double lines connecting the various stages of the trajectories depict the "principal energy" flows. Above and below each step are shown the external energy and resulting emissions associated with that step. These baseline energy flows and emissions were obtained from

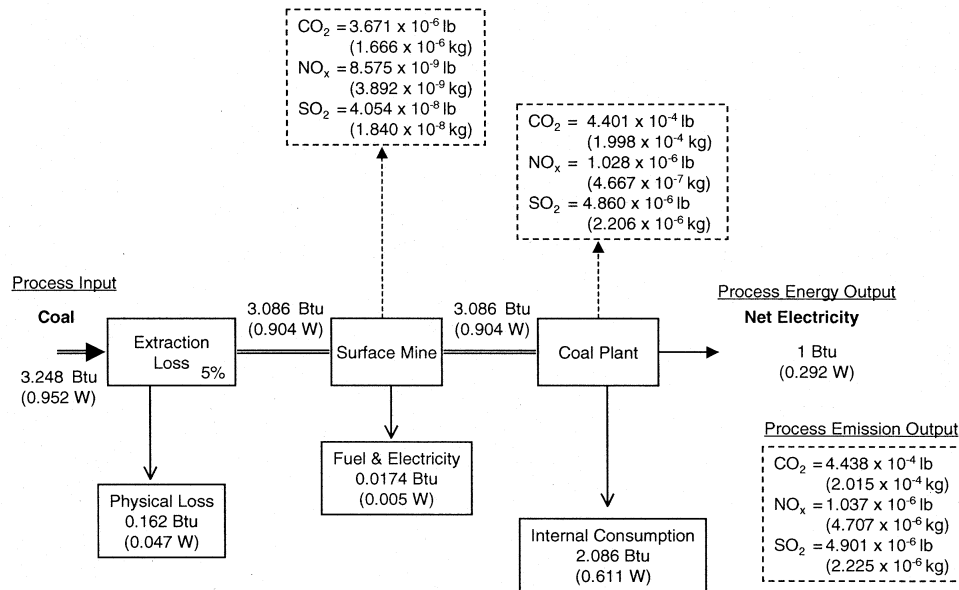


Figure 1 Energy and emission trajectory for a natural gas power plant.

Melcher et al. (1976) but varied on an hourly basis for the present investigation.

The processes involved in delivering natural gas begin at the wellhead of an oil or natural gas well. The losses at the wellhead consist of spills, leaks, flaring, etc. In the gathering process, the gas is collected from the various wells in a well field and piped to the processing plant. At the processing plant, the gas is dehydrated and prepared for introduction into the pipeline. The largest losses occur during the distribution phase.

When natural gas is used for electrical power generation, an extra process step of generation needs to be added to the trajectory. The heat rate and emission data will depend upon the marginal power plant during the given hour. Figure 1 illustrates the natural gas trajectory with the electrical power production, as well as the transmission and distribution. In this case, 3.644 Btu (1.07 W) of natural gas is required to produce 1 Btu (0.292 W) of electricity to the site. The power plant in this example has a heat rate of 11,887 Btu/kWh.

Coal is the other fuel source that supplies power plants on the margin. For this particular example power plant, 3.248 Btu (0.95 W) of natural gas is required to produce 1 Btu (0.292 W) of electricity supplied; the power plant had a heat rate of 10,529 Btu/kWh (3.085 kW/kW). Figure 2 illustrates the coal trajectory.

BUILDING MODEL

Two distinct building types were considered in this analysis: a commercial office building and a school. The commercial office building was an 11-story 506,000 ft² (47,009 m²) building with a peak cooling load of 980 tons (3446 kW). The school was a 110,000 ft² (10,219 m²) single-story building with a peak cooling load of 200 tons (703 kW). The annual

hourly building loads were determined with a commercially available annual hourly simulation program. The weather data used for the building load determination was TMY2 (Marion and Urban 1995) data for Madison, Wisc.

Load profiles for both high temperature (55°F/13°C) supply air and low temperature (44°F/6.7°C) supply air distribution were calculated for both the office and school building. The air distribution system was a variable air volume (VAV) system with a supply fan static pressure drop of 4.0 in. water gauge (995 Pa) and 4.25 in. water gauge (1060 Pa) for high and low temperature air, respectively. Both high and low temperature air systems had a return fan with a static pressure drop of 0.5 in. water gauge (124 Pa). The thermostat setting was 74°F (23°C) during occupied hours and 82°F (28°C) during unoccupied hours. The occupied setpoint was imposed one hour prior to occupancy. The fans were “on” during occupied hours and allowed to cycle during the unoccupied hours to maintain the setup thermostat setting. Outside air ventilation was provided at 15 cfm/person (7.1 L/s per person) peak occupancy during all occupied hours.

CENTRAL PLANTS

Five different central plant options were investigated: direct cooling electric chiller, chilled water storage, ice storage, ice storage with cold air distribution, and double-effect direct-fired absorption chiller (Table 1). Commercial simulation package routines provided the fan energy usage. The thermal energy storage systems were operated as a single chiller combined with storage. The chiller would serve off-peak loads as well as recharge storage. The intent of the central plant system design was to minimize chiller and storage size. The sizing of these systems was dependent upon the timing and

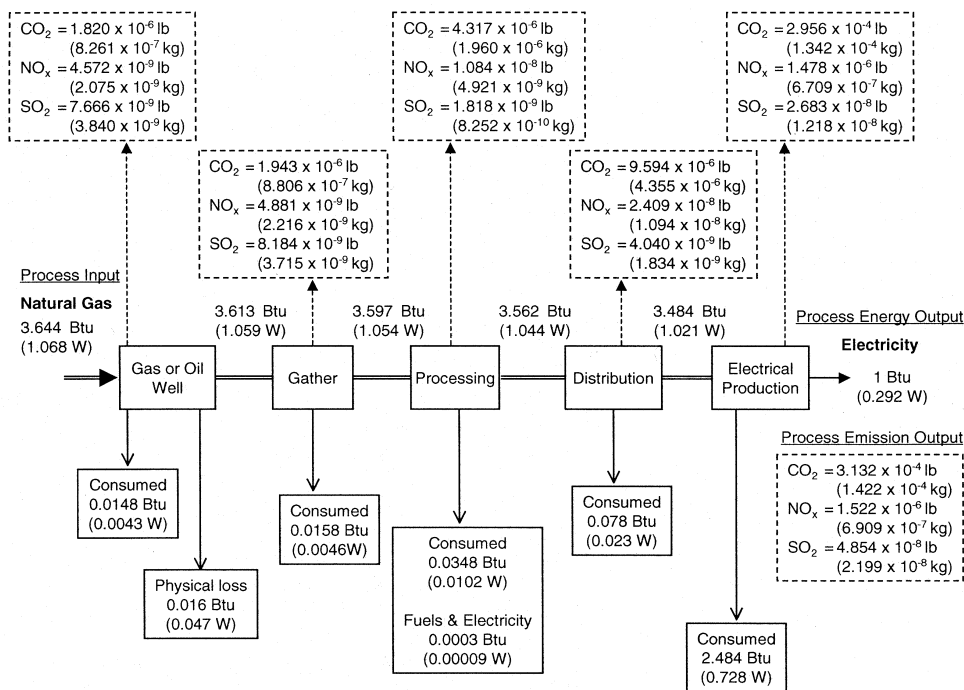


Figure 2 Energy and emission trajectory for a coal power plant.

duration of the on-peak period. Two different on-peak periods were investigated.

The electric chillers were modeled as water-cooled with a cooling tower (for the office building and the school) and air-cooled (for the school only). The compressor capacity and power requirements were based upon a polynomial curve fit of manufacturer data. The performance of the entire chiller plant—compressor, condenser, and evaporator—is calculated hourly by balancing all components based upon current load and ambient conditions. The part-load performance of the chiller is modeled by typical unloading characteristics of a

positive displacement compressor. The same part-load performance is used for all electric chillers.

OPERATING STRATEGY

In direct chilling mode, there is only one chiller sized to meet the peak load for each building type. The chiller runs at part load for most hours of the year. For each building type, there were some hours with very small loads compared to the maximum load, which would result in very inefficient operation. To avoid having these hours of very inefficient operation

TABLE 1
Central Plant Specifics

	Cooling Coil Design Approach, °F (°C)	Cooling Coil Water, ΔT, °F (°C)	Supply Air Temperature, °F (°C)	Operating Strategy
Direct Chilling	8.1 (4.5)	10 (5.6)	55 (12.8)	Direct
Chilled Water Storage	8.1 (4.5)	20 (11.1)	55 (12.8)	Shift 12 p.m. - 8 p.m. Shift 8 a.m. - 5 p.m. Load Level
Ice Storage	9.0 (5.0)	16 (8.9)	55 (12.8)	Shift 12 p.m. - 8 p.m. Shift 8 a.m. - 5 p.m. Load Level
Ice Storage with Low-Temp. Air	6.0 (3.3)	16 (8.9)	44 (24.4)	Shift 12 p.m. - 8 p.m. Shift 8 a.m. - 5 p.m. Load Level
Absorption Chilling	6.0 (3.3)	10 (5.6)	55 (12.8)	Direct

skew the results, the chiller was controlled to let the cooling load carry over until the integrated load was at least 10% of its capacity.

For the thermal storage systems, the operation was based upon three different strategies. Two were full shift strategies corresponding to two different on-peak periods. One scenario was to shift all of the space cooling loads from 12 p.m. through 8 p.m.; the other was to shift all cooling loads from 8 a.m. through 5 p.m. During these hours, only the hydronic system (including storage) and fan systems operate to meet cooling loads. During all other hours, the chiller is allowed to run in order to meet cooling loads and to charge the storage.

The third strategy was a load-leveling chiller priority. In this strategy, the chiller always operates in order to meet the load. Storage would be called upon only when the cooling load exceeds the available capacity of the chiller.

DIRECT CHILLING

The cooling coil was modeled to operate with a fixed approach of 8.1°F (4.5°C) between the water and the air. Two-way valves operated to maintain a 10°F (5.6°C) water-side temperature differential across the coil. The water-side flow rate to individual coils varied in direct proportion to the hourly cooling load requirements. The chiller-side (primary) pump operated with a head of 25 ft (74,644 Pa) and a pump efficiency of 65%. The load-side (secondary) pump operated with a minimum head of 10 ft (29,858 Pa), but this value was dependent on the flow rate to the load. The nominal pump efficiency was 65% as well.

CHILLED WATER STORAGE

The chilled water storage system functioned similarly to the non-storage system. An electrically driven chiller was operated in conjunction with the chilled water storage tank to meet the space cooling loads. The storage tank was a stratified chilled water tank with a design temperature differential of 20°F (11.1°C) - 40°F (4.4°C) supply, 60°F return (15.6°C).

The chiller operates to charge the tank during off-peak hours. A deadband of 100 ton-h (352 kWh) was applied to chiller operation for charging. During on-peak hours, storage operates in conjunction with the chiller to meet the cooling load. The control strategy was chiller priority—the chiller ran to meet the load and then storage was utilized if necessary. For the full-shift systems, storage would meet the load exclusively during the on-peak hours. For the load-leveling systems, storage would supply the supplemental cooling that could not be met by the chiller directly.

ICE STORAGE

The static ice systems used an arrangement that was similar to the chilled water systems. The major difference was that the working fluid was a secondary coolant consisting of a 25% (by weight) mixture of ethylene glycol. The storage tanks were based on a 190 nominal ton-h internal-melt ice-on-coil tank.

For charging, the chiller operated by delivering the coolant to the storage tanks at a temperature below the freezing point of water (32°F/0°C). The performance of the ice storage tanks was calculated using the Jekel (1991) effectiveness model. This model determines the effectiveness of charging and discharging as a function of the flow rate through the tank, the flow inlet temperature, and the tank's state of charge. During charging, the brine temperature leaving the chiller was set downward to 24°F (-4.4°C). If a cooling load occurred off-peak, the chiller would try to meet both the load and charge storage. If the chiller's entering brine temperature rose to cause the chiller's leaving brine temperature to rise above 32°F (0°C), the storage tanks would be bypassed and the chiller would reset to meet the load solely. If cooling load exceeded the chiller capacity, the additional load was met by utilizing the storage (chiller-priority partial storage).

The chiller operated to charge the tanks during off-peak hours. During on-peak hours, storage would operate in conjunction with the chiller to meet the cooling load. For the full-shift systems, storage would meet the load exclusively during the on-peak hours. For the load-leveling systems, storage would supply the supplemental cooling that could not be met directly by the chiller (chiller-priority control).

ABSORPTION CHILLING

The absorption system was modeled based upon a water-cooled direct-fired double-effect absorption chiller. The full- and part-load performance of the absorption system was developed from manufacturer data. The control strategy for the absorption chiller was identical to that of the conventional electric chiller. The chiller is meant to only operate to meet the instantaneous space cooling load. The chiller plant was sized to meet the maximum hourly cooling load.

The cooling coil was modeled to operate with a fixed approach of 6°F (3.3°C) between the water and the air. There was water-side temperature difference of 10°F (5.6°C) across the coil. The flow rate to the coil was dependent on the load requirements. The load side pump operated with a minimum head of 10 ft (29,858 Pa), but this value was dependent on the flow rate to the load. The condenser pump operated with 40 ft (11,943 Pa) of head. The pump efficiency for both of these pumps was 65% as well.

AUXILIARIES

The water-cooled condenser for the electrical chillers was based on a curve fit of manufacturer data. The model calculated an approach temperature (i.e., the difference between the cooling tower leaving water temperature and the outside air wet-bulb temperature) based upon the outside air wet-bulb temperature and the range. The range was based upon the difference between the cooling tower entering water temperature and the cooling tower leaving water temperature. Electrical energy usage for the pumps and cooling tower fans at full load was estimated to be 0.0667 kW/ton (0.0189 kW/kW) of heat rejected.

For the absorption system, a similar water-cooled condensing model was used. Electrical energy usage for the pumps, cooling tower fans, and the absorption machine auxiliary at full load was estimated to be 0.110 kW/ton (0.0312 kW/kW_e) of heat rejected.

The air-cooled condensing unit was modeled with a 15°F (8.3°C) temperature difference between saturated condensing temperature and outside air dry-bulb temperature. The energy consumption for the condensing unit at full load was estimated to be 0.0746 kW/ton (0.0212 kW/kW_e) of heat rejected.

RESULTS

The sizing of the equipment was done to meet the cooling loads on the peak day. The peak day profiles for the cold air systems were different from those for the other systems. In terms of the storage scenarios, the systems were sized to minimize both chiller and storage size. The sizing criteria were such that the available storage capacity never went below 20% of the nominal capacity. The sizing results are shown in Tables 2–4. The energy and emission results are shown in Tables 5–10.

Using the conventional system (electric chiller) as a base, all of the thermal storage technologies resulted in lower source energy requirements. This was true for both utility generation mixes. While thermal storage could result in higher site energy consumption, source energy requirements were always lower. This was also true for the emissions associated with energy use. Thermal storage technologies always had lower emissions than the base case.

The gas absorption system had considerably higher source energy requirements than the base case. However, it did produce the benefit of lower emissions in most of the categories (SO₂, N₂O, NO_x, CH₄, and particulates). In fact, it had the lowest level in all of those categories of any of the technologies. However, it did produce the highest CO₂ emissions. Gas absorption systems performed better against the air-cooled electric systems for the school. Energy and emission results were higher for the air-cooled electric systems than their water-cooled counterparts in the school.

A global warming index was evaluated based upon the reported emission types. Global warming potential pertains to the radiative forcing by a chemical, its ability to reflect and absorb heat from the earth. The direct global warming index (GWI), relative to that of carbon dioxide, for the various emission types are CO₂ (1), SO₂ (0), N₂O (270), NO_x (0), CH₄ (11), and particulates (0). These values are for a 100-year horizon, as taken from Isaksen et al. (1992). This type of metric puts the emission constituents on a common basis for comparison. Note that we only included the direct effect of the emissions. There can also be an indirect effect that pertains to how the presence of a chemical affects reactions within the atmo-

sphere. However, the evaluation of that indirect effect is much more difficult to quantify.

CONCLUSIONS

Thermal energy storage systems should be promoted as an environmentally beneficial technology. These systems have been historically touted as beneficial in terms of operation cost. This study suggests that the economic benefits can be accompanied by environmental ones. Thermal energy storage systems typically resulted in appreciable savings of source energy as well as reduced emissions along the entire path from the source to the power plant. The nature of the operation of thermal energy storage systems allows them to take advantage of the more efficient nighttime generating capacity.

Scenarios using thermal storage technology always (with one exception) showed a source energy benefit when compared to a conventional electric chilling base case. Source energy reductions were generally on the order of 10%. Thermal energy storage technologies also showed significant reductions in CO₂, SO₂, N₂O, NO_x. Thermal storage did sometimes suffer in terms of CH₄ and particulates; however, global warming impact reductions were also on the order of 10%.

REFERENCES

- EIA (Energy Information Administration). 1992. *Electric power annual—1990*.
- EPRI (Electric Power Research Institute). 1993. *Space conditioning system selection guide*. Palo Alto, Calif.
- Isaksen, I.S.A., V. Ramaswamy, H. Rodhe, and T.M.L. Wigley. 1992. Radiative forcing of climate. *Climate change 1992*.
- Jekel, T.B. 1991. Modeling of ice-storage systems. Masters Thesis, University of Wisconsin.
- Marion, W. and K. Urban. 1995. User's manual for TMY2s, typical meteorological years derived from the 1961-1990 national solar radiation database. National Renewable Energy Laboratory.
- Melcher, A.G., K. Maddox, C. Prien, T. Nevens, P. Yesavage, P. Dickson, J. Fuller, W. Loehr, R. Baldwin, and R. Bain. 1976. *Net energy analysis: An energy balance study of fossil fuel resources*. The Colorado Energy Research Institute.
- Reindl, D.T., D.E. Knebel, and R.A. Gansler. 1995. Characterizing the marginal basis source energy and emissions associated with comfort cooling systems. *ASHRAE Transactions* 101 (1): 1353-1363.
- Tabors Caramanis & Associates. 1996. *Source energy and environmental impacts of thermal energy storage*. CEC (California Energy Commission).
- TU Electric. 1993. *Allocation of 1990 electrical losses*.

TABLE 2
System Sizes for Office Building

System	Nominal Chiller Capacity, ton (kW _l)	Nominal Chiller Efficiency COP	Icemaking Chiller Capacity, ton (kW _l)	Icemaking Chiller Capacity COP	Nominal Storage Capacity, ton-h (kWh)
Electric Chiller	1100 (3868)	6.20	—	—	—
Absorption	1100 (3868)	1.03	—	—	—
On-Peak 12 p.m. - 8 p.m.					
Chilled Water Storage	760 (2672)	6.20	—	—	7400 (26,018)
Ice Storage	1160 (4079)	6.20	800 (2813)	4.70	13,300 (46,762)
Ice Storage with Cold Air	1197 (4209)	6.20	825 (2901)	4.70	14,440 (50,630)
On-Peak 8 a.m. - 5 p.m.					
Chilled Water Storage	900 (3164)	6.20	—	—	9900 (34,808)
Ice Storage	1451 (5102)	6.20	1000 (3516)	4.70	15,200 (53,443)
Ice Storage with Cold Air	1451 (5102)	6.20	1000 (3516)	4.70	17,480 (61,459)
Load Level					
Chilled Water Storage	525 (1846)	6.20	—	—	5000 (17,580)
Ice Storage	725 (2549)	6.20	500 (1758)	4.70	2090 (7348)
Ice Storage with Cold Air	761 (2676)	6.20	525 (1846)	4.70	4750 (16,701)

TABLE 3
System Sizes for School Building—Water Cooled

System	Nominal Chiller Capacity, ton (kW _l)	Nominal Chiller Efficiency COP	Icemaking Chiller Capacity, ton (kW _l)	Icemaking Chiller Capacity COP	Nominal Storage Capacity, ton-h (kWh)
Electric Chiller	200 (703)	6.20	—	—	—
Absorption	200 (703)	1.03	—	—	—
On-Peak 12 p.m. - 8 p.m.					
Chilled Water Storage	140 (492)	6.20	—	—	1200 (4219)
Ice Storage	181 (636)	6.20	125 (439)	4.70	1520 (5344)
Ice Storage with Cold Air	203 (714)	6.20	140 (492)	4.70	1900 (6680)
On-Peak 8 a.m. - 5 p.m.					
Chilled Water Storage	150 (527)	6.20	—	—	1100 (3868)
Ice Storage	232 (816)	6.20	160 (563)	4.70	2850 (10,021)
Ice Storage with Cold Air	247 (868)	6.20	170 (598)	4.70	2850 (10,021)
Load Level					
Chilled Water Storage	90 (316)	6.20	—	—	1000 (3516)
Ice Storage	130 (457)	6.20	80 (281)	4.70	380 (1336)
Ice Storage with Cold Air	138 (485)	6.20	95 (334)	4.70	950 (3340)

TABLE 4
System Sizes for School Building—Air Cooled

System	Nominal Chiller Capacity, ton (kW _e)	Nominal Chiller Efficiency COP	Icemaking Chiller Capacity, ton (kW _e)	Icemaking Chiller Capacity COP	Nominal Storage Capacity, ton-h (kWh)
Electric Chiller	200 (703)	4.50	—	—	—
Absorption	200 (703)	1.03	—	—	—
On-Peak 12 p.m. - 8 p.m.					
Chilled Water Storage	140 (492)	4.50	—	—	1200 (4219)
Ice Storage	181 (636)	4.50	125 (439)	3.10	1520 (5344)
Ice Storage with Cold Air	203 (714)	4.50	140 (492)	3.10	1900 (6680)
On-Peak 8 a.m. - 5 p.m.					
Chilled Water Storage	150 (527)	4.50	—	—	1100 (3868)
Ice Storage	232 (816)	4.50	160 (563)	3.10	2,850 (10,021)
Ice Storage with Cold Air	247 (868)	4.50	170 (598)	3.10	2850 (10,021)
Load Level					
Chilled Water Storage	90 (316)	4.50	—	—	1000 (3516)
Ice Storage	130 (457)	4.50	80 (281)	3.10	380 (1336)
Ice Storage with Cold Air	138 (485)	4.50	95 (334)	3.10	950 (3340)

TABLE 5
Office Source Energy Usage

System	Utility #1			Utility #2		
	Site Electricity (%)	Source Electricity (%)	Source Energy (%)	Site Electricity (%)	Source Electricity (%)	Source Energy (%)
Electric Chiller Base	100	100	100	100	100	100
Absorption	57	57	133	57	57	127
On-Peak 12 p.m. - 8 p.m.						
Chilled Water Storage	86	82	82	86	85	85
Ice Storage	98	93	93	98	96	96
Ice Storage with Cold Air	94	89	89	94	92	92
On-Peak 8 a.m. - 5 p.m.						
Chilled Water Storage	90	85	85	90	89	89
Ice Storage	103	97	97	103	101	101
Ice Storage with Cold Air	96	90	90	96	94	94
Load Level						
Chilled Water Storage	83	82	82	83	83	83
Ice Storage	86	86	86	86	86	86
Ice Storage with Cold Air	82	82	82	82	82	82

TABLE 6
School Source Energy Usage—Water Cooled

System	Utility #1			Utility #2		
	Site Electricity (%)	Source Electricity (%)	Source Energy (%)	Site Electricity (%)	Source Electricity (%)	Source Energy (%)
Electric Chiller Base	100	100	100	100	100	100
Absorption	63	62	127	63	62	145
On-Peak 12 p.m. - 8 p.m.						
Chilled Water Storage	87	84	84	87	85	85
Ice Storage	92	88	88	92	91	91
Ice Storage with Cold Air	93	88	88	93	92	92
On-Peak 8 a.m. - 5 p.m.						
Chilled Water Storage	91	87	87	91	89	89
Ice Storage	91	88	88	91	90	90
Ice Storage with Cold Air	91	88	88	91	90	90
Load Level						
Chilled Water Storage	87	86	86	87	87	87
Ice Storage	88	88	88	88	88	88
Ice Storage with Cold Air	86	86	86	86	86	86

TABLE 7
School Source Energy Usage—Air Cooled

System	Utility #1			Utility #2		
	Site Electricity (%)	Source Electricity (%)	Source Energy (%)	Site Electricity (%)	Source Electricity (%)	Source Energy (%)
Electric Chiller Base	100	100	100	100	100	100
Absorption	58	57	117	58	57	134
On-Peak 12 p.m. - 8 p.m.						
Chilled Water Storage	79	77	77	79	78	78
Ice Storage	95	91	91	95	94	94
Ice Storage with Cold Air	98	94	94	98	96	96
On-Peak 8 a.m. - 5 p.m.						
Chilled Water Storage	83	80	80	83	81	81
Ice Storage	94	92	92	94	93	93
Ice Storage with Cold Air	97	94	94	97	96	96
Load Level						
Chilled Water Storage	81	81	81	81	81	81
Ice Storage	87	87	87	87	87	87
Ice Storage with Cold Air	90	91	91	90	91	91

TABLE 8
Office Emission Characteristics

System	Utility	CO₂ (%)	SO₂ (%)	N₂O (%)	NO_x (%)	CH₄ (%)	Part (%)	GWI (%)
Electric Chiller	1	100	100	100	100	100	100	100
	2	100	100	100	100	100	100	100
Absorption	1	120	56	56	76	59	58	120
	2	117	59	61	78	64	63	116
On-Peak 12 p.m. - 8 p.m.								
Chilled Water Storage	1	86	71	70	81	115	106	85
	2	85	78	81	81	76	81	85
Ice Storage	1	98	81	80	92	131	120	97
	2	96	89	92	92	87	92	96
Ice Storage with Cold Air	1	93	77	76	88	126	114	93
	2	92	84	87	88	81	88	92
On-Peak 8 a.m. - 5 p.m.								
Chilled Water Storage	1	89	73	71	84	123	115	89
	2	89	81	84	84	79	83	89
Ice Storage	1	101	84	82	96	137	126	101
	2	101	92	95	95	91	95	101
Ice Storage with Cold Air	1	94	78	76	89	129	118	94
	2	94	85	88	89	83	88	94
Load Level								
Chilled Water Storage	1	83	81	80	82	88	87	83
	2	83	82	82	83	84	82	83
Ice Storage	1	86	86	85	86	87	87	86
	2	86	87	87	86	88	86	86
Ice Storage with Cold Air	1	82	83	82	82	81	81	82
	2	82	83	83	83	84	83	82

TABLE 9
School Emission Characteristics—Water Cooled

System	Utility	CO₂ (%)	SO₂ (%)	N₂O (%)	NO_x (%)	CH₄ (%)	Part (%)	GWI (%)
Electric Chiller	1	100	100	100	100	100	100	100
	2	100	100	100	100	100	100	100
Absorption	1	117	59	61	78	64	63	116
	2	121	61	61	81	59	61	121
On-Peak 12 p.m. - 8 p.m.								
Chilled Water Storage	1	86	76	76	84	104	100	86
	2	86	79	80	84	78	84	86
Ice Storage	1	91	78	78	88	116	109	91
	2	91	84	86	88	82	88	91
Ice Storage with Cold Air	1	93	78	75	88	124	117	93
	2	92	84	86	89	83	89	92
On-Peak 8 a.m. - 5 p.m.								
Chilled Water Storage	1	90	81	81	87	106	101	90
	2	90	83	84	88	81	87	90
Ice Storage	1	90	83	83	88	103	99	90
	2	90	83	84	87	85	88	90
Ice Storage with Cold Air	1	90	84	81	88	106	102	90
	2	90	83	84	87	86	88	90
Load Level								
Chilled Water Storage	1	86	87	86	87	85	86	86
	2	87	87	86	88	88	88	87
Ice Storage	1	87	90	90	88	82	84	87
	2	88	89	88	90	90	90	88
Ice Storage with Cold Air	1	86	91	89	87	79	83	86
	2	86	88	86	89	95	90	86

TABLE 10
School Emission Characteristics—Air Cooled

System	Utility	CO₂ (%)	SO₂ (%)	N₂O (%)	NO_x (%)	CH₄ (%)	Part (%)	GWI (%)
Electric Chiller	1	100	100	100	100	100	100	100
	2	100	100	100	100	100	100	100
Absorption	1	107	54	56	72	60	59	107
	2	111	56	56	75	53	56	111
On-Peak 12 p.m. - 8 p.m.								
Chilled Water Storage	1	79	71	71	77	93	90	79
	2	78	72	73	78	72	77	78
Ice Storage	1	94	83	82	91	117	109	94
	2	94	86	88	90	85	91	94
Ice Storage with Cold Air	1	97	85	82	93	125	117	97
	2	97	88	90	93	89	94	97
On-Peak 8 a.m. - 5 p.m.								
Chilled Water Storage	1	82	75	75	80	94	90	82
	2	82	75	76	80	73	80	82
Ice Storage	1	93	89	88	92	102	100	93
	2	93	86	87	91	91	92	93
Ice Storage with Cold Air	1	96	92	89	95	109	107	96
	2	96	88	89	94	92	94	96
Load Level								
Chilled Water Storage	1	80	81	81	81	79	81	80
	2	81	81	81	82	82	82	81
Ice Storage	1	86	90	90	87	81	81	86
	2	88	87	86	91	89	90	88
Ice Storage with Cold Air	1	90	95	92	91	85	88	90
	2	91	94	92	95	97	95	91