Enhanced Thermal Storage that Saves Energy

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The majority of the HVAC industry has the opinion that thermal storage systems must consume more energy than and require more capital investment than conventional cooling systems. This is a gross misconception based on speculation, rationalization, or simple misunderstanding rather than analysis. Application of a design approach referred to as "Enhanced Thermal Storage" clearly saves significant amounts of energy directly through system kilowatt hour savings and indirectly through conservation of construction materials.

Introduction "the Problems"

Late 20th century problems have kindled new interest in an old idea. These problems include escalating energy and demand costs, a short fall in electrical generation capacity world-wide, escalating costs of building new power plants, the unwillingness of public utility commissions to raise rates, ozone depletion, the accelerated phase out of Chlorofluorocarbons (CFC's), and global warming. A design concept that appears to be a cure for these problems is Enhanced Thermal Storage and associated technologies.

An Old Idea Reborn

Thermal storage is not an experimental idea, but rather an adaptation of a 19th century concept of using ice for cooling. Short-peak applications have been in use since 1938.

Commercially, thermal storage has been in revival since 1979 in California, Texas, Arizona, Florida, and several New England states. These geographic areas have seen phenomenal growth in the past decade. As generation capacity dwindled, the electric utilities could not build new power plants fast enough to meet demand. Estimates by the Electric Power Research Institute, Rocky Mountain Institute, and various electric utility demand planning groups indicate projected demand will exceed current capacity by 1995. To compensate, utilities began looking towards energy conservation and thermal storage. The simplest definition of thermal storage is that it generates and stores cooling at night when electrical demand is low to be used during the day when demand is high. In theory, if enough buildings utilized or changed to thermal storage, new power plants could be deferred for decades.

Utility Interest

Electric utilities found that by offering incentives based on the amount of Kilowatts shifted to off-peak, they could encourage the use of thermal storage. Soon after the implementation of such incentive programs the power companies discovered that paying customers to shift and conserve power was extremely cost effective. The average cost of bringing on line one Kilowatt (Kw) of power is approximately $3000 (Block, "Global Warming and TES" 1991, ASHRAE Far East Conference on Environmental Issues, Hong Kong). Nationally incentives range from $139 to $900 per Kw shifted to off-peak (Block, "Cost Effective Thermal Storage" 1989, ASHRAE Far East Conference on Air Conditioning in Hot Climates, Kuala Lumpur, Malaysia). Incentives like these have created a whole new interest in thermal storage.

Classic Savings

When thermal storage was reborn in the late 70's, operational cost savings were in the 5% to 10% range due to demand reduction. Classic design approaches simply coupled a thermal storage central plant with conventional air and water distribution. Demand savings were considered to be the only source of cost savings. These design approaches caused paybacks to be (10) ten to (20) twenty years. Interest was sporadic and limited at best. (Block, "Thermal Storage Misconceptions" 1987, ASHRAE Annual Meeting, New York, New York).

The Enhanced Approach

An "Enhance Thermal Storage" system uses high efficiency state-of-the-art ice chiller equipment, cold air distribution, and cold water distribution, and innovative design techniques to reduce overall capital cost and produce system Kwh savings. The rate structures of today coupled with the "Enhanced" approach are producing operational cost savings as high as 70% (Block, 1991, "Global Warming and TES" ASHRAE Far East Conference on Environmental Issues). These same electrical rates, innovative designs, and substantial utility incentives are producing retrofit projects which are recognizing paybacks in well under (3) three years with gross return-on-investments (ROI's) in the 40% plus range (Block, 1988, "Retrofit Opportunities for Thermal Storage", Second Annual DOE Conference, Washington, D.C.). Many new facilities utilizing thermal storage technologies in the original design have documented capital cost savings as high as 25% and estimated operational savings of 70% (Block, 1991, "Global Warming and TES" ASHRAE Far East Conference on Environmental Issues). A large part of the capital and energy cost savings is due to the utilization of high delta T air and water distribution systems. These design techniques substantially reduce the physical size and capital cost of both air and water systems (EPRI, 1988, "Cold Air Distribution Manual"). Operational savings are due in part from saving high demand charges, but also from KWH savings. These energy savings can reach the 40% to 50% levels and beyond through the use of state-of-the-art ice chilling equipment and innovative design and application strategies (Block, 1991, "Global Warming and TES" ASHRAE Far East Conference on Environmental Issues).

Methodology "a Sample Project"

To demonstrate the source of these potential windfall savings we will examine a specific project that has been constructed. The analysis shall include a classic ASHRAE "High Efficiency" design that meets all the parameters set by ASHRAE Standard 90.1 (1992) which shall be referred to as conventional, a water source heat pump system that meets low cost economic needs, and Enhanced Thermal Storage that goes well beyond both those requirements from a systems approach.

Our sample project is a 200,000 square foot office building located in New England. The owner is a speculative developer so capital cost is most important to him. The tenant, however, is a large computer manufacturer whose main interest is long term energy cost savings. Often these two points of view are envisioned as mutually exclusive, but in this case a compromise was reached after extensive analysis of three system types.

Based upon the tenants needs the project cooling load calculation parameters are as follows;

1.4 Watts/SF Lighting
4 Watts/SF Equipment Load
Envelope Overall Thermal Transmittance Value (OTTV) < 25
Infiltration .1 CFM/SF of Exterior Surface AREA
Outside Air 20 CFM/Person
Occupancy Maximum 2000 Persons

A computer simulation of project cooling load resulted in the following:

Peak HVAC Load 650 Tons
Daily Load Profile 4500 Ton-Hrs
Weekly Load Profile 22,500 Ton-Hrs
Electric Utility On-Peak Demand Window 8 Hrs/Day
40 Hrs/Week

System Costing

The local contractor community shared an opinion that thermal storage systems required that a substantial contingency or "fear factor" be applied in the bidding process to allow for system failure. To minimize this effect on the accuracy of the costing data, three completely different sets of costing documents were prepared. The documents were broken down into four major categories, central plant, water side, air side, and controls. These documents were distributed directly to the subcontractors by the design team. Since no one contractor or subcontractor bid a complete set of documents and was not responsible for a complete and working system the "fear factor" based upon system liability was minimized. The results are indicated in Table 1.

The level of detail in Table 1 is necessary to indicate in which system components costs savings may be achieved. The cost premium of the thermal storage central plant is some 43%. However, this premium is negated by the significant material savings in the cooling delivery systems (i.e. the air and water distribution). This is an important concept because construction material must be manufactured through the use of energy intensive methods. Significant material savings such as galvanized sheetmetal, piping, valves, and other accessories indirectly saves significant amounts of energy!

System Efficiencies

System efficiency is perhaps the most hotly debated issue within the engineering community. Thermal storage has traditionally been portrayed as an energy inefficient system. In laboratory tests because of the lower evaporator temperatures there is no doubt thermal storage chillers

Table 1.

Part 1 Central Plant

Heat Pump System			Conventional			Thermal Storage		
Type	Size	Cost	Type	Size	Cost	Type	Size	Cost
Cooling Tower	980 Tons	$37,240	Centrifugal	650 Tons	$125,000	Ice Harvester	180 Ton	$180,000
			Cooling Tower	650 Tons	$25,000	EVAP Condenser	230 Ton	$46,000
			Condenser Pumps	1950 GPM	$15,000	N/A		
			Condenser Piping 100'	(10")	$8,000	N/A		
			Electrical	600 Amps	$50,000	Electrical	281 Amps	$15,000
						Tank	34,000	$78,000
Total		$37,240	Total		$223,000	Total		$319,000
Part 2 Waterside								
CHWS	Avg 8"	$290,000	CHWS&R	Avg 6"	$219,000	CHWS&R	AVG 4"	$118,000
Pipe Insulation	Avg 8"	$36,000	Pipe Insulation	Avg 6"	$11,800	Pipe Insulation	Avg 4"	$9,700
CWS Pumps	2940 GPM	$29,000	CHWS Pumps	1300 GPM	$10,000	CHWS Pumps	650 GPM	$12,000
Valves	Various	$27,000	Valves	Various	$54,000	Valves	Various	$22,000
Electrical	200 Amps	$14,000	Electrical	60 Amps	$5,000	Electrical	60 Amps	$5,000
						Heat Exchanger	700 Ton	$25,000
Total		$396,000	Total		$299,800	Total		$191,700
Part 3 Air Side								
Heat Pump Units	(326)@3	$537,900	AHU's	(10)@27K	$199,500	AHU's	(10)@18K	$133,500
Ductwork	69K lbs	$207,000	Ductwork	200K lbs	$600,000	Ductwork	89K lbs	$267,000
Insulation	128K SF	$64,000	Insulation	400K SF	$200,800	Insulation	178K SF	$89,000
			Vav Boxes	266	$133,000	Fan Powered Boxes	178	$178,000
			Diffusers	1064	$53,000	Diffusers	712	$35,600
Electrial	326	$163,000	Electrical	266	$133,000	Electrical		$89,000
Total		$971,900	Total		$1,319,300	Total		$792,100
Part 4 Controls								
Central Plant	20	$10,000	Central Plant	40	$20,000	Central Plant	40	$20,000
Airside	0		Airside	306	$68,850	Airside	218	$49,500
Processor	1	$15,000	Processor	1	$33,000	Processor	1	$33,000
Total		$25,000	Total		$121,850	Total		$102,500
System Totals								
Central Plant	$37,240	Central Plant	$223,000	Central Plant	$319,000			
Waterside	$396,000	Waterside	$299,800	Waterside	$191,700			
Airside	$971,900	Air side	$1,319,300	Airside	$792,100			
Controls	$25,000	Controls	$121,850	Controls	$102,500			
Totals	$1,430,140	Totals	$1,963,950	Totals	$1,405,300			
System Premium	$24,800	System Premium	$492,650		$0			
Cost Per SF	$7.15	Cost Per SF	$9.81		$7.02			

use more energy. These laboratory tests only compare the tow different operating set points under the same conditions in the same piece of chiller equipment. However, few air conditioning applications exist or operate under these lab conditions. Real world applications indicate that when conventional "high efficiency systems" operating under actual conditions and application are compared to thermal storage systems operating under actual conditions and applications, the disparity narrows. If all energy saving attributes associated and possible with each system are analyzed thermal storage is clearly more energy efficient.

The key wording here is design attributes. Referring to our sample project consider this comparison of design strategies;

Heat Pump	Conventional		Thermal Storage
(326) Units @ 3 Ton ea	Chiller	Peak	Weekly Load Shift
CWS Pumps 2940 GPM @ 100'HD	CHWS Pumps	650 GPM ea @ 100'Head	325 GPM ea @ 40'Head
	Fans	27,000 Cfm ea 4.0" SP	18,000 GPM @ 1" SP
	Terminal Units	Fan Powered VAV/VAV	Series Fan Powered VAT

The design strategies differ because of limitations imposed by either economics, equipment operating parameters, packaged unit size, and in the case of the heat pumps sensible heat capacity.

To illustrate these limitations consider first the heat pump system. The sensible heat ratio for the building is 95%. Yet the best sensible heat capacity ratio available in small (3) ton heat pumps is 75%. Due to this disparity in capacities and the mismatch of "packaged" capacities with zone requirements, the heat pump system application requires significantly more units (326 @ 3 tons or 978 total tons) to satisfy a calculated 650 ton peak. This requires a larger cooling tower (978 tons vs 650 tons), larger pumps, piping, control valves, and other accessories.

Now consider the chilled water systems for both the conventional and enhanced thermal storage systems. The conventional chilled water system is designed based upon 45°F leaving water temperature (LWT) and 55°F entering water temperature (EWT) at the chiller. This ten degree temperature delta causes large volumes of chilled water to be required. This large volume translates into very large piping sizes. To minimize capital cost designers select pipe sizes at a maximum of 10 feet per second velocity. This volume and velocity of water dictate a system pressure loss of 100 in the sample project. The enhanced thermal storage chilled water system design takes a different approach. The chilled water system is sized based upon 36°F LWT and 60°F EWT at the chiller and a maximum of 5 FPS when sizing piping. The lower volume of water caused by the twenty-four degree temperature delta significantly reduces pipe size and therefore capital cost. The enhanced thermal storage approach is to use a lower velocity to reduce the system head pressure and "give back" some of the capital cost savings. This becomes operationally attractive because of vastly smaller pump sizes coupled with moderately smaller piping. The overall effect is a 25% smaller chiller water distribution system with 75% smaller pumps.

Next consider the air distribution systems. Conventional variable air volume (VAV) air systems are designed for a cooling coil discharge air temperature of 55°F at a room temperature setpoint of 75°F. Because of the amount of air required to cool the space by this design (a 20° delta "T"), the limited ceiling space imposed by the shortest possible floor-to-floor height of the project (an architectural capital cost savings technique), and the tremendous increase in interior cooling loads due to computerization, large and heavy rectangular duct with a poor aspect ratio is required. This translates into the single most costly item in the HVAC system. Duct friction losses are high because of the high velocities and poor aspect ratios needed to fit the duct above the ceiling. To limit mechanical room space (a architectural requirement to maximize net rentable space) and equipment capital cost, the air handling units (AHU's) are selected at 550 FPM velocities across the coils and filters. Overall this equates to some 4.0" of total static pressure on the fan and some 325 horsepower of fan motors for the sample project. Additionally, recent concern for improved indoor air quality are pushing designers into using constant volume fan powered terminal units (Series Fan Powered VAV boxes) in conventional VAV air distribution systems to eliminate stagnant air problems associated with early VAV applications. This adds capital cost and 172 HP of fractional, inefficient fan motors to the conventional system.

Conversely, the enhanced thermal storage system is designed to deliver 45°F air to the space, but only at peak the time of cooling load. This method actually improves indoor air quality and the comfort level of occupants (Scofield, C.M. 1991, "Low-temperature air with High IAQ", ASHRAE Far East Conference on Environmental Issues). The primary duct system (duct between the AHU and the fan powered box) is sized for a 35°F temperature delta at the time of peak cooling load requirements. During part load conditions the cooling coil leaving air temperature is reset upward instead of varying the system volume. Constant volume fan powered terminals are used to mix the primary supply air (at a temperature which ranges from 40F to 60°F) with recirculated plenum air to maintain the space temperature setpoint. This design approach reduces the primary and secondary (duct between the fan powered terminal unit and the space) air volume to some 50% of the conventional system. Selecting the maximum amount of air at the lowest temperature at the load peak allows for much smaller round duct to be used. Round duct is much more cost effective and has less friction loss due to the perfect aspect ratio. The difference in air volume combined with the perfect aspect ratio translates into a duct system savings in sheetmetal poundage, associated insulation, and other air systems devices of some 70% (SMACNA "Sheetmetal Design Guide" 1977). The enhanced approach is to again "give back" some of these capital cost savings in order to reduce energy consumption. The duct system is sized with lower velocities in order to significantly reduce total static pressure. Additionally, because the AHU's are 50% smaller, they may be economically changed to face velocities of less than 300 FPM. Overall this equates to a system with less than 1" total static pressure including the series fan powered terminal units. Air handling unit fan motor size for the entire project is reduced to 30 HP and fan powered boxes use 87.5 HP.

A comparison of cool generation systems between the heat pump, conventional, and enhanced thermal storage (ETS) finds they operate during completely different times. The conventional chiller and heat pump system operates during the heat of the day, and the ETS unit operates during the cool of the night. This is not simply an observation, but when analyzed a distinct design advantage for the thermal storage system. Although the thermal storage chiller is theoretically less efficient because of the lower suction temperatures required, it has vastly larger evaporator and condenser sections and operates in ambient temperatures that are much lower. Furthermore, the ETS system can take better advantage of these lower ambient. Consider that to maintain compressor head pressure for adequate oil and lift a conventional chiller requires a pressure differential equivalent to approximately 27°F above the suction temperature. At a suction temperature of 35° (to maintain 45° LWT) the minimum setpoint would be 62°F, however, most controls contractors set this up to 80°F for a safety factor. An ice chiller operates at 22°F (to make ice at 32°F), therefore, the minimum operating temperature is approximately 49°F. These factors combine to redefine the operating efficiency algorithm. This redefinition considerably closes the gap in efficiency. The application of these design techniques results in computerized energy use simulations as follows:

	Peakload	Peakload	Peakload
Part 1 Central Plant Chiller	N/A	.665 (R-123)	.75 @ Ice Making & .55 in Chiller Mode (R-22)
Tower	.22	.16	.08
Pumps		.2	N/A
Part 2 Waterside			
Pumps	.11	.08	.04
Part 3 Airside			
AHU's	1.8	.5	.046
FP Boxes		.266	.135
Part 4 Controls			
CPU	.0004	.0004	.0004
Totals	2.134	1.8714	1.0504
On-Peak Totals	2.134	1.8714	0.2214
Annual KWH Based on 1500 FLH	2,080,650	1,824,615	1,023,750
KWH Premium	1,056,900	800,865	0
Percent Savings	0	13%	61%
Annual Operating Cost	$178,103	$156,186	$39,360
Percent Savings	0	13%	78%
Energy Savings @ $0.0256/KWH	$0	$6,554	$27,056
Demands Savings @$10/KW	$0	$15,363	$111,677
Water Usage	139,000 Gals	108,000 Gals/Year	35,640 Gals/Year
Water Cost Per Year	$6,950	$5,400	$1,782
Savings	$0	$1,550	$5,168
Total Annual Savings	$0	$23,467	$143,901
20 Year Life Cycle Savings	$0	$488,163	$2,993,140

System Efficiencies

Demand and KWH savings such as these will help defer power plant construction, electrical distribution grid upgrades, and ultimately electrical rate increases. This should make professional and lay conservationists happy because fewer power plants mean less "greenhouse" emissions. Additionally, more power used at night causes many power plants to run more efficiently, less CO_2 and SO_2 (Block, 1990, "Global Warming and Thermal Storage", New England Environmental Expo, Boston, Massachusetts). As indicated previously thermal storage systems operate at night when temperatures are lower. These lower temperatures mean greater chiller efficiency and less water loss for water cooled units. Less cooling tower evaporation means less chemical usage. This should also equate to lower maintenance and operational costs due to less chemical use, which equates less toxic waste.

Chlorofluorocarbons (CFC's)

Thermal storage impacts the CFC issue in several ways. First, as existing refrigerants are phased out, the replacements are less efficient. For example, replacing a CFC (R-11) with a non-CFC (R-123) would cause a tonnage capacity drop of some 10% to 25%. Because the replacement is thermally less efficient, the power usage of the chiller would also go up from 15% to 22% (Fischer, S.K., Creswick, F.A.,1989. "Energy-use impact of chlorofluorocarbon alternatives." ORNL/CON-273.). Facilities will have an immediate shortage of capacity and higher energy bills. The use of thermal storage would eliminate this shortage while shifting the now less efficient chiller to the lower night time electrical rates. The shift to night time rates (in areas where the on-peak, off-peak demand differential cost is $5.00/KW) can equate to a positive cash flow in less than (3) years. Without thermal storage positive cash flow is unlikely.

Another impact is more direct, several modern thermal storage chillers use Hydrochlorofluorocarbons (HCFC's). HCFC's have an ozone depletion factor less than one tenth that of CFC's. Although HCFC's have a green house gas effect and long half life (up to 680 years for R-22) the use of the thermal storage approach reduces chiller size (CFC Update, The Trane Company, Volume 10, January 1992). In some sizing and operating strategies chiller size is reduced by some 80%. Similarly refrigerant use would decline by an equal amount. Many thermal storage chillers can also utilize R-717 (ammonia). Ammonia has no ozone depletion factor at all. The current replacement refrigerants have some ozone depletion factors, however, the biggest problem may be toxicity. Many of these replacements are showing indication of being carcinogens (The Air Conditioning, Heating and Refrigeration News, July 1, 1991).

Retrofit Projects

Retrofit applications of thermal storage technologies need not be limited to facilities with chilled water plants. Facilities utilizing an HVAC system with an EER of (13) thirteen or less are a viable candidate. This includes; rooftop systems, split-system DX, air-to-air heat pumps, water source heat pumps, air cooled chillers, and many other configurations of cooling systems. Prospective candidates need not have a narrow occupied time period, (24) twenty-four hour facilities qualify. Size is also not an issue (Block, 1987, "Thermal Storage Misconceptions", ASHRAE Annual Meeting, New York, New York). Projects as small as 1000 SF (small churches in the south have used an ice-on-pipe design for some 30 years sized to build ice over 164 hours to be used on Sunday mornings) and as large as an entire city can cost effectively utilize thermal storage technologies.

Summary

In review of the benefits of "Enhanced Thermal Storage" we find it is cost effective to the electric utilities through the shifting of electrical load to a time when generation plants are under utilized and therefore, also helps maintain lower electrical rates. Capital cost is comparable with very cost effective conventional systems. Enhanced thermal storage can stop the tonnage shortfall as CFC's are phased out and some systems use no CFC's at all. More importantly ETS saves energy both directly and indirectly. The indirect energy savings are significant because an enhanced thermal storage system using cold air and cold water design techniques saves large quantities sheetmetal, piping, insulation, and other construction materials that require energy to manufacture. The direct savings are potentially in the 61% range as compared to mainstream "high efficiency" conventional systems. Enhanced thermal storage is applicable to all size and occupancy facilities. Although this all sounds too good to be true, enhanced thermal storage is indeed a truth and these benefits are beginning to be more completely understood and hopefully utilized in the near future.