6.4 MECHANCAL DEPTH: DESIGN PROCEDURE

Dorgan and Elleson's (1993) *Design Guide for Cool Thermal Storage* and Kirkpatrick and Elleson's (1996) *Cold Air Distribution System Design Guide* provided a basis on the design procedure needed to evaluate the mechanical redesign. The procedure included the following steps:

- ❖ Screening initial economics
- ❖ Calculating load profiles
- ❖ Selecting storage type
- ❖ Selecting operating strategy
- ❖ Sizing cooling plant and storage
- ❖ Determining chiller and equipment parameters
- ❖ Sizing cooling coils for cold-air distribution
- ❖ Laying schematics
- ❖ Evaluating economics both first and life cycle cost
- ❖ Finalizing design

- Screening initial economics

The initial economics was already explored in the previous section to identify if thermal storage was applicable. Since it was determined that thermal storage would be a viable option the proceeding step was to determine the building's load profile. Determining the load profiles first, required the design weather conditions and thermal properties of the building, found in the previous step. Additionally, the building occupancy needed to be known. The building is primarily occupied between the hours of 6 a. m. and 5 p.m., Monday through Friday. There are numerous summer school activities which makes the need for cooling all year round.

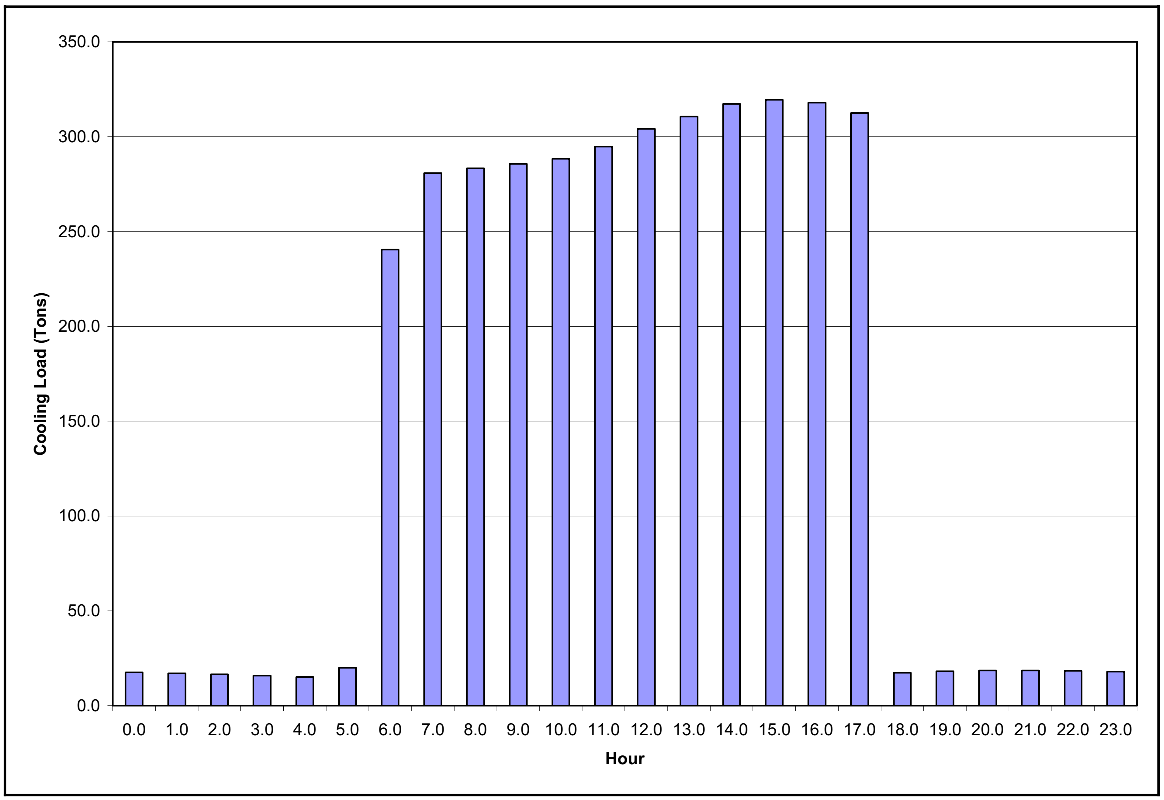
- Calculating load profiles

Then with the aid of a computer energy analysis program, in this case, Carrier's Hourly Analysis Program (HAP) version 4.20, an 8760-hour load profile was obtained. In order to determine the size of the system, HAP was able to generate a 24-hour load profile for the design day (in this case July 23), which is the day with the greatest cooling load, shown, on the following page, in Figure 8. The figure illustrates that there is a relatively low base load compared to the peak load. As stated previously, this scenario favors thermal storage as the peak load can be shifted completely or partially to off peak hours.

- Selecting storage type

After determining the 24-hour load profile the next step was to determine the storage type. Typically, in thermal storage there are two storage types either chilled water or ice. Chilled water storage uses the sensible heat capacity of water, 1 Btu per pound per °F, to store cooling. Meanwhile, ice thermal storage uses the latent heat of water, 144 Btu/lb. One of the major differences between the two is the tank size needed for storage. Since chilled water storage is stored at a warmer temperature, between 39 and 42 °F, versus ice storage, between 22 and 26 °F, the tank size needed is also greater. Tank volume size for a chilled water storage system can range from 11 to 21 ft^3/ton-hr where ice tank volume sizes range from 2.4 to 2.8 ft^3/ton-hr (Dorgan and Elleson, 1993). The size of tanks needed for chilled water is at least four times greater than that of ice storage. Another factor to why ice was chosen was space. As stated previously, the building's site is in the heart of the criminal justice section in Bronx, NY which led to limited space for storage tanks. Considering both first cost and available space, ice was chosen as the storage media.

Figure 8: Nonstorage System Design Day Load Profile



However, the storage type selection is still incomplete. There are three main types of ice thermal storage that was considered including: ice harvesting, external melt and internal melt ice-on-coil. The ice harvesting refrigeration plant generates and releases sheets or tubes of ice with a specially designed evaporator section. Water is pumped out of the storage tank and is distributed over the evaporator surfaces and is either chilled or frozen. The ice harvesting plant is capable of operating as both an ice maker and as a water chiller. Unfortunately, because this chiller has to be specially designed the first cost is extremely high, around $1000 to $2000 per ice making ton (Dorgan and Elleson, 1993). The first cost for an ice harvesting system relative to the size of project could not be cost justified.

An external melt ice-on-coil storage system builds and stores ice on the exterior surfaces of a heat exchange coil submerged in a non-pressurized water tank. In order to charge the storage system, typically a secondary coolant, such as a glycol solution, is circulated inside the heat exchange tubes, causing ice to form on the outside of the tubes. Discharging the stored cooling, the ice on the tubes is melted by warm return water which circulates through the tank. The leaving water is chilled and used to meet the building load. External melt ice storage systems normally build ice to a thickness of 1.5 to 2.5 inches on the pipe. The greater the thickness requires lower charging temperatures around 10-15°F. This means the chiller capacity has to be increased significantly to be able to produce that low of temperature water. This is due to the chiller efficiency dropping for every temperature degree below its rated leaving water temperature, because it requires more energy to lower the temperature (Dorgan and Elleson, 1993). This chiller becomes “de-rated” meaning the actual capacity of the chiller is significantly less than the rated nominal capacity. For example, if a 100-ton chiller is de-rated 30 percent than the chiller actually only produces 70 tons and not the full 100. Also external melt systems require a separate charge and discharge circuit which would add to first cost in piping and in pumps.

Internal melt ice systems work similarly to the external system except instead of using a separate discharging circuit it uses the same circuit for both charging and discharging. In discharge mode, warm coolant flows through the tubes, melting the ice from the inside out and reducing the coolant temperature for use in meeting the cooling load. Building ice, works exactly like same way as external melt only the coolant temperature is warmer, 22-26°F. When comparing between internal and external melt systems the chillers for internal are generally smaller because of the warmer water meaning the chiller is less de-rated as compared to external melt. Another advantage is internal melt only require one circuit for charging and discharging whereas, external requires two separate lines. Therefore, on these principles the internal melt system was chosen.

- Selecting Operating Strategy

The next step after determining the storage type was to decide on the operating strategy either: partial or full storage. Both partial and full storage systems have their distinct advantages. A partial storage system is able to meet a portion of the on-peak cooling load from storage, with the remainder of the load met by the operating chiller equipment. Typically, in a partial storage system the chiller is smaller than compared to a nominal chiller. Partial storage systems can be operated in two control strategies: load-leveling and demand-limiting operation. A load-leveling system generally operates at full capacity continuously throughout the day, charging during off-peak and directly cooling during on-peak, while a demand-limiting system operates at a reduced capacity during on-peak hours. Although, a full storage system shifts the entire on-peak cooling load to off-peak periods. The downfall with full and demand-limiting storage strategies is that they require more storage capacity and larger chiller sizes as compared to the load-leveling strategy (Dorgan and Elleson, 1993). This analysis explored the options of both load-leveling partial storage and full storage. The same 24-hour load profile that was generated for the nonstorage system was used to determine the required storage capacity needed for both full and partial storage, shown in Figures 9 and 10. The *Design Guide for Cool Thermal Storage* was again used to determine the chiller and storage capacities needed.

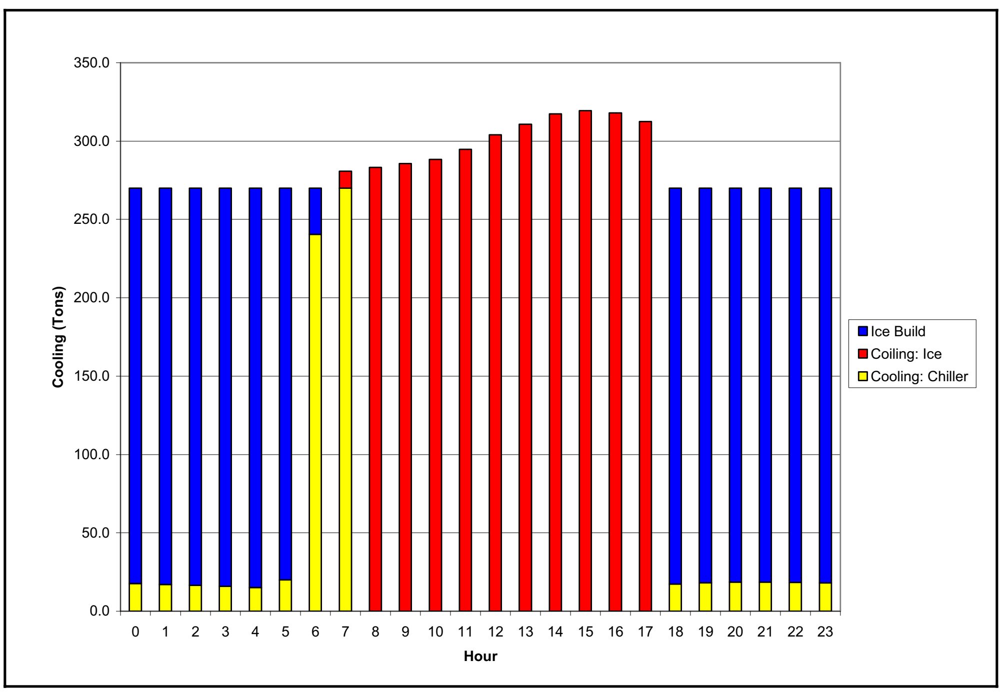
- Sizing cooling plant and storage

From Figures 9 and 10, it was originally calculated for chillers sizes for full storage to be 270 tons and partial storage to be 160 tons. However because the leaving water temperature is between 22-26°F the chiller capacity is reduced approximately 33 percent. Therefore, a larger capacity chiller needed to be selected to meet the required loads. The following components was selected and sized:

- Full Storage

  - ❖ (2) 410 ton propylene glycol chillers
  - ❖ (2) 1400 gpm cooling towers
  - ❖ (6) 486 ton-hour ice storage tanks
  - ❖ (1) 162 ton-hour ice storage tanks

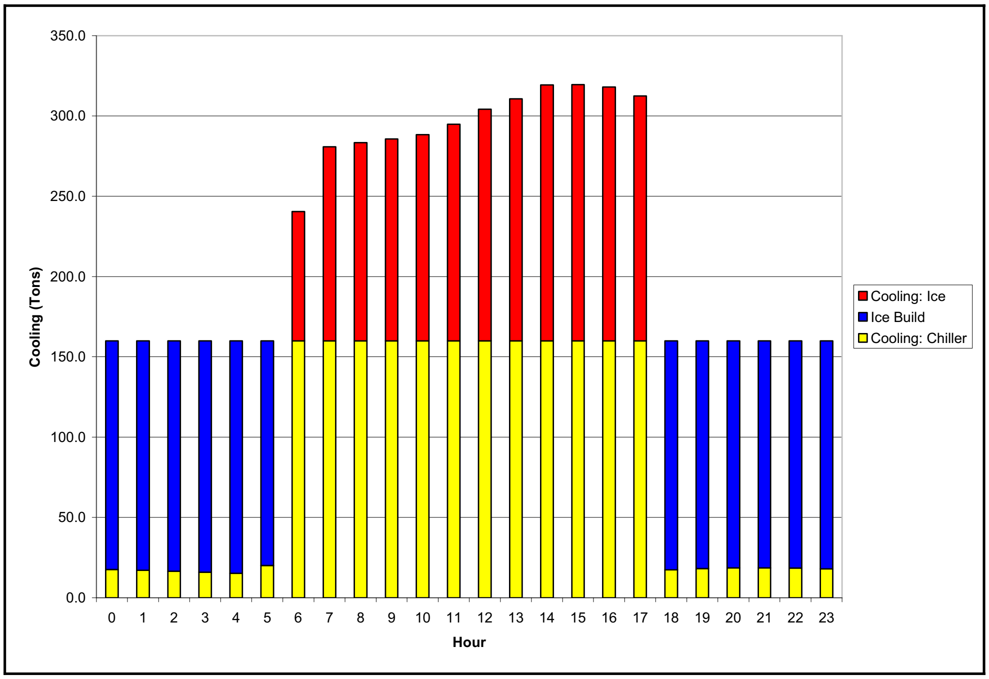
Figure 9: Full Storage System Design Day Load Profile



- Partial Storage

  - ❖ (2) 240 ton propylene glycol chillers
  - ❖ (2) 800 gpm cooling towers
  - ❖ (3) 486 ton-hour ice storage tanks
  - ❖ (2) 162 ton-hour ice storage tanks

Figure 10: Partial Storage System Design Day Load Profile



- Determining chiller and equipment parameters

The chillers were selected for peak load conditions and designed with the "n+1" rule of thumb. The n+1 rule of thumb is designing one chiller or a combination of chillers and adding one of equal capacity. This is done for redundancy, incase a chiller would fail or is in need of maintenance the redundant chiller would be able to operate to serve the cooling load. Adding an "extra" chiller does add significant first cost but the tradeoff is that it also provides a fail-safe and flexibility to the system.

Another option during the process of chiller selection was to incorporate a "base" chiller. A base load chiller is used to serve the average load during off-peak times. A base load chiller can be advantageous when the average load during off-peak hours remains constant. In this case, the base load does remain relatively constant however; the average load was only around 20 tons. The option of installing a 20 ton base chiller was explored however; was not implemented. There were several reasons as to why a base chiller was not selected. Having a chiller with an extremely small capacity compared to the capacity needed for the peak loads was not economically cost effective. Aside from the actual chiller having a high first cost additional pumps and controls would need to be added which also increase first cost. To meet the demands of the off-peak loads, the resolution was to add a three-way valve and additional piping and during the charging mode a portion of the cold water, 26°F, would be diverted and mixed with the return, 51°F, to meet the required cooling coil entering water temperature at 39°F.

Selecting the type of chiller was another issue that had to be addressed. Centrifugal, reciprocating and screw chillers are all capable of producing the necessary chilled water for ice storage. However, the problem with centrifugal chillers selections must be made specific for designed operating conditions while reciprocating and screw chillers are more adaptable to a wider range of leaving temperatures. The YORK® MaxE™ water-cooled screw chillers were selected to provide the necessary chilled water. The technical specifications for these chillers can be seen in Appendix A.

Figure 11: YORK® MaxE™ water-cooled screw chillers

As stated earlier, the original mechanical systems were air cooled DX condensers. The redesign required water cooled chillers which means that cooling towers needed to be added to the system. After selecting the chillers the required condenser flow rates were obtained and Marley's NC Class cooling towers were selected. The technical specifications for these cooling towers can be seen in Appendix A.

Figure 12: Marley NC Class Cooling Towers

The other major equipment that needed to be selected was the ice storage tanks themselves. Calmac's IceBank ice storage tanks were selected as the storage containers. The IceBank model 1190C has the capacity for 162 ton-hrs of storage and the model 1190C, which simply are three model 1190C tanks piped together. Evaluating the tank capacities and capacities needed for cooling, six model 1500C and one model 1190C tanks, totaling 19 tanks and 3078 ton-hrs of cooling, were selected for full storage and three model 1500C and two model 1190C tanks, totaling 11 tanks and 1782 ton-hrs of cooling, were selected for partial storage. More information can be found in Appendix A.

Figure 13: Calmac IceBank Storage Tanks

- Sizing cooling coils for cold-air distribution

Taking advantage of the colder water being supplied from the ice thermal energy system cold air distribution was also applied. An advantage of using cold air distribution was that it would reduce the required airflow. Equation (1) was used to determine the reduced amount of airflow required and still meet the capacity needed for the load. One of the main components that needed to be changed was the cooling coils inside the packaged air handlers. The current cooling coils were sized for entering and leaving water temperatures of 44 and 55°F, with entering and leaving air at 85 and 55°F. In order to handle the new entering and leaving water temperature of 39 and 51°F, and entering and leaving air at 85 and 44°F, the cooling coils needed resized. Obtaining the new cooling coils Carrier's AHUBuilder v. 5.42 was used. Knowing the sensible load required, the new airflow and the new enter and leaving water temperature, AHUBuilder was able to specify the new cooling coils. Table 3 represents the original designed air handler's airflow, flow rate and cooling capacity and Table 4 represents the resized air handlers. The calculated cooling coil data from AHUBuilder can be seen in Appendix A.

Equation 1: $Q_s = 1.08*(CFM)*\Delta T$

Q_s = Required Sensible Load
CFM = Required airflow
ΔT = Change in entering and leaving air temperature

Table 3: Original Design Data

	CFM	GPM	Sensible	Total
AHU-1	48000	404	1614.0	2440.0
AHU-2	19000	164	612.4	903.5
AHU-3	18500	138	578.7	848.3
AHU-4	3400	25	101.3	152.1
AHU-5	12000	119	479.3	728.1
AHU-6	5200	47	178.6	287.5
AHU-7	12000	77	333.3	471.6
AHU-8	6000	53	207.3	323.0
AHU-9	7200	47	202.4	284.9
AHU-10	3100	22	91.0	134.1
Totals:	134400	1096	4398.3	6573.1

Table 4: Redesigned Data

	CFM	GPM	Sensible	Total
AHU-1	37500	336.6	1656.3	2442.8
AHU-2	14500	120.2	621.9	875.3
AHU-3	13500	112.3	580.4	817.6
AHU-4	2400	19.3	103.4	140.6
AHU-5	11200	92.8	480.4	675.6
AHU-6	4200	35.8	183.4	260.9
AHU-7	8000	64.3	335.4	468.2
AHU-8	4800	40.7	208.4	296.0
AHU-9	4800	40.7	208.4	296.0
AHU-10	2400	17.7	95.1	128.9
Totals:	103300	880.4	4473.1	6401.9

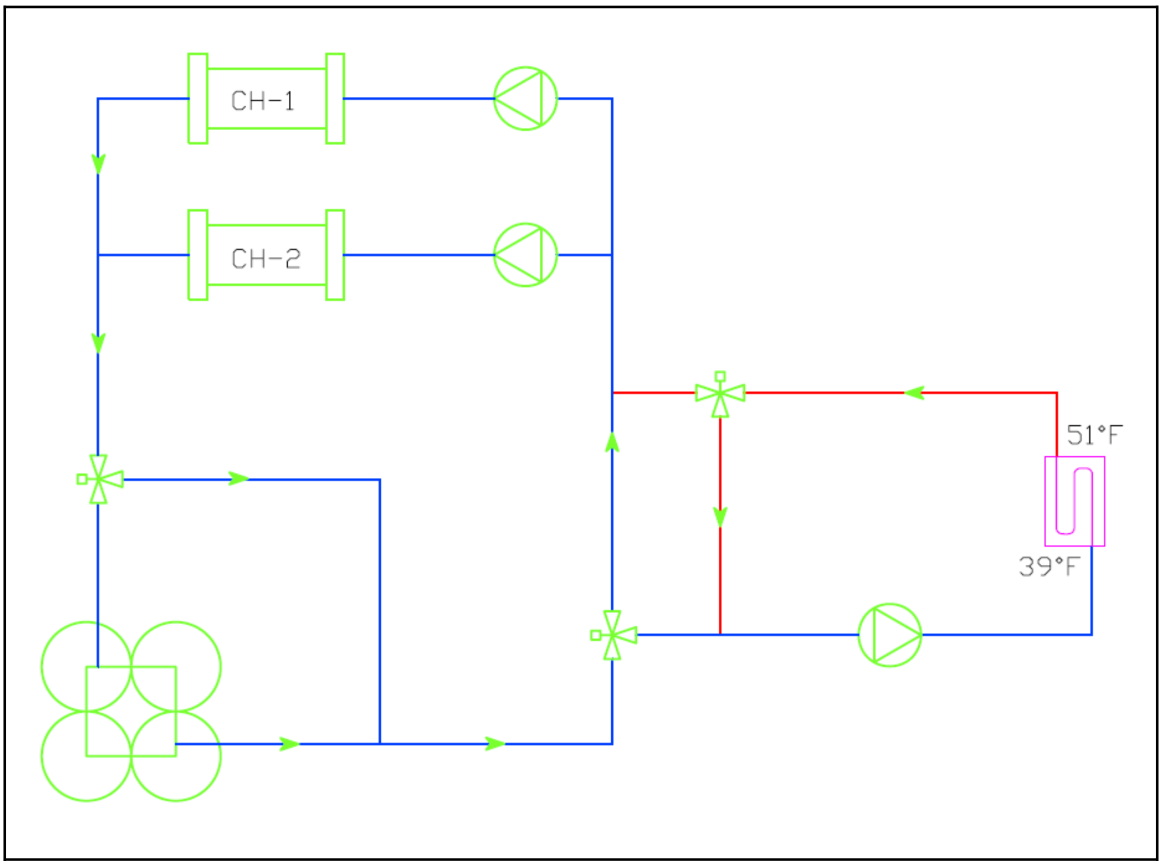
The biggest concern with cold air distribution involves concerns with condensation both through the ductwork and off the supply diffusers. Although not specifically studied in the analysis there are ways to overcome these problems. The easiest way of solving the ductwork issue is assure that all the ducts are properly insulated; improperly insulating the ductwork can lead to condensation problems with any system not just cold air distribution systems. There are two ways to solve to the supply diffuser issue involving either switch the conventional supply diffusers to linear slot diffusers or use fan-powered VAV boxes to blend a portion of the return air with the supply air and typical 55°F air will be delivered to the spaces. The linear diffusers have a higher momentum (mass flow rate × velocity) of cold air increases the throw. This is important because the air will mix faster and will then be able to satisfy thermal comfort. A slight downside to the fan-powered box option is that it will increase fan energy slightly because of the need to operate continuously during use. However, both options are viable and will solve the problem of condensation on the diffusers.

- Ice storage system schematic

As previously analyzed, in both full and partial storage there will be two chillers specified. The schematic layout for both would be identical with the exception of the size of the equipment. Again as stated previously, there will not be a base load chiller therefore; the load during off-peak times will be met by the chiller directly. The important aspect with this design is the controlling sequence. It is imperative that when cooling directly from the chiller during ice build mode that the water be mixed with the return, enough to warm the water to the rated 39°F. This is important because sending 26°F water directly to the cooling coil may freeze the coil and cause major problems. To assure that the water is mixed properly three-way temperature and diverting valves were added. The valves will modulate to mix the proper flow to obtain the rated temperature to meet the loads. Another control strategy that had to be determined was for partial storage. Figure 13 is the schematic for the new redesigned thermal energy storage system.

During partial storage there were two options to operate either chiller priority or storage priority. Chiller priority results in higher chiller use during on peak times while storage priority more of the on peak cooling is done by the storage. Like most decisions there is a trade-off, chiller priority operates the chiller with better efficiency but cost more during on-peak hours while storage priority is the opposite the chiller operates less efficiency but cost less during on peak times. Also storage priority requires more storage capacity which again adds to the first cost and because the difference between on and off peak rates were not significant chiller priority was selected.

Figure 13: Ice Storage Schematic



- Evaluating first cost

One of the main goals of the entire redesign process was to minimize the first cost. In addition to the cost concerns stated earlier the following must also be taken into account: chilled water pumps, condenser pumps, additional secondary pump, additional valves, piping and glycol.

- Chilled Water Pumps

  The addition of thermal storage and cold air distribution reduced the size of chillers which would also reduce the chilled water flow and correspondingly reduce the chilled water pumps needed.

- Condenser Water Pumps

  Since the original chillers were air cooled packaged chillers there were no condenser pumps. However, in the redesign cooling towers were added to reject the heat from the chillers. This required the addition of condenser pumps to be added to the system.

- Additional Secondary Pump

  Originally, the chilled water was pumped directly to the air handling units in a primary loop system. The addition of a secondary loop required also an additional pump.

- Additional Valves

  Three-way mixing valves that were not needed in the original design was now required. The additions of these valves were needed to properly control the temperature and flow throughout the system.

- Piping

  There was a need for additional piping in the system because of the change from a primary system to a primary/secondary system. However, because the flow is less the pipe sizes were able to be reduced.

- Glycol

  The original system was a fresh water system and did not require the use of glycol. However, to achieve the low temperature required a 25% by volume glycol system was needed. Propylene glycol was selected because it was less toxic compared to its ethylene counterpart.

Results

The first cost, operating cost, life cycle cost, and simple payback period were determined and analyzed for the conventional, partial storage and full storage systems.

Table 5 represents a summary of the first cost comparison; the entire first cost analysis can be seen in APPENDIX B. The first cost analysis used the *2006 RS Means Mechanical Cost Data.*

Table 5: First Cost Comparison

	Conventional		Partial Storage		Full Storage	
	Material	Labor	Material	Labor	Material	Labor
Chillers	$314,000.00	$34,080.00	$378,103.00	$47,003.00	$614,103.00	$52,936.00
Pumps	$9,525.00	$1,695.00	$22,100.00	$1,365.00	$22,100.00	$1,365.00
Air Handling Units	$187,050.00	$13,185.00	$154,750.00	$11,985.00	$154,750.00	$11,985.00
Cooling Coils	$23,828.00	$12,638.00	$27,315.00	$14,537.00	$27,315.00	$14,537.00
Fans	$59,985.00	$16,140.00	$41,135.00	$12,225.00	$41,135.00	$12,225.00
Air Distribution	$396,525.00	$836,663.00	$375,875.00	$740,544.00	$375,875.00	$740,544.00
Pipe	$128,765.00	$157,613.00	$103,012.00	$126,090.00	$103,012.00	$126,090.00
Pipe Insulations	$38,585.00	$29,960.00	$30,868.00	$23,984.00	$30,868.00	$23,984.00
Subtotal	$1,158,263.00	$1,101,974.00	$1,133,158.00	$977,733.00	$1,369,158.00	$983,666.00
Grand Total	$2,260,237.00		$2,110,891.00		$2,352,824.00	

Determining the operating cost required a few steps. First the energy model needed to be created to see the yearly cooling required. Carrier's HAP was once again used to simulate the 8760-hour energy model. Although thermal storage could not be simulated directly from HAP it was however, able to take into account the cold air distribution savings. The two energy models have been recreated in Microsoft's Excel sheets and are included in Appendix B. Table 6 represents the peak loads on design day for each month.

Table 6: Peak Loads Design Day

	PEAK TON		PEAK TON
JAN	110.7	JUL	319.5
FEB	134.7	AUG	317.5
MAR	188.5	SEP	289.9
APR	227.2	OCT	245.3
MAY	277.5	NOV	190.1
JUN	301.8	DEC	132.1

To determine the annual operating cost the energy models that were created and the initial utility rates were needed. Then knowing the on-peak and off-peak kWh use the operating cost for the conventional and full storage systems could be calculated. However, it is more difficult to determine the annual energy cost for partial storage because of the fluctuation of on-peak loads. Therefore, equation (2) from Pacific Gas & Electric's article *Thermal Energy Storage Strategies for Commercial HVAC Systems* was used to estimate the operating cost. Chiller efficiency had to be factored into the estimations as well. Producing ice is less efficient than producing regular chilled water. Dorgan and Elleson's (1993) *Design Guide for Cool Thermal Storage* had the ice chillers around 1.1 kW/ton and 0.7 kW/ton was used for the conventional chillers. The Table 7 breaks down the estimated annual energy costs.

Equation (2):

$$\text{kWh}_{\text{shifted}} = \#\ \text{tons}_{\text{shifted}} \times (\text{kW/ton})_{\text{chiller performance}} \times \text{on-peak hours} \times \text{load shape factor}$$

"The load shape factor is a needed multiplier because peak cooling load typically is not constant. This factor, used in the above equation, is for the on-peak period only (the time when cooling load will be shifted) and for the peak cooling load for that day. Typical load shape factors are in the range of 60 to 90 percent for a variety of building types and climates." – PG&E

Table 7: Estimated Annual Energy Cost

	Conventional System	Partial Storage	Full Storage
Demand Cost (kW)	26626.01	22318.81	0.00
Operating Cost (kWh)	49140.21	40079.48	55800.05
Total Annual Cost	75766.22	62398.29	55800.05

The Life Cycle cost was estimated at a 25-year period, beginning in 2003 when the actual building was occupied. Electricity escalation factors were found for electricity using the *Energy Price Indices and Discount Factors for Life-Cycle Cos Analysis – April 2006*. Table 8 represents the estimated life-cycle cost.

Table 8: Estimated Life Cycle Cost

	Conventional	Partial Storage	Full Storage
First Cost	2260237.00	2110891.00	2352824.00
25 yr LC Cost	2533622.49	2086598.70	1865953.67
LLC	4793859.49	4197489.70	4218777.67

Finally, the payback period for both partial and full storage were calculated and shown in the Table 9.

Table 9: Simple Payback Period

	Partial Storage	Full Storage
Payback (yrs)	-1.9	10.5

- Finalizing Design

There are many factors involved when considering the feasibility of implementing any new system. Typically, first cost is the main factor as to whether a system is implemented or not regardless of energy optimization. Owners want a quick payback to their investments and disregard the other issues. In analyzing thermal energy storage alone the first cost was significantly more than a conventional system. By adding cold air distribution allowed the first cost for partial storage to be cheaper than that of the conventional system. In deciding between the two operating strategies partial storage seemed the most cost effective and overall the most optimal chose. Space was another issue as to why partial storage was more applicable, full storage required larger chillers, cooling towers and storage capacity. Space is at a premium in New York City therefore, more space needed to be devoted to occupancy use than for building life systems. Considering both first cost and space partial storage was chosen over full storage. As far as comparing thermal storage with the conventional system, not only is the cost of thermal storage with cold air distribution cheaper but it also reduces the energy consumption through the mechanical systems. By not only saving on demand charges but reducing energy consumption and smaller equipment makes thermal storage an applicable solution.