Jessica R. Baker

The Montgomery County Conference Center and Hotel (MCCCH), Rockville, MD

5.0 Mechanical Depth

PENNSTATE

5.0 Mechanical Depth:

Mechanical Existing Conditions:

Air-Side Systems

The airside mechanical system for the Montgomery County Conference Center and Hotel consists of a combination of variable air volume and constant volume systems served by eleven different air handling units throughout the building. Eight of the air handling units are located in mechanical rooms throughout the building, two rest on the hotel roof, and one is ceiling mounted in a stairway area (See Sketch 1). Variable air volume boxes with electric reheat distribute air from the air handling units to all spaces in the two-story conference center, as well as, the restaurant and hotel first floor. Constant volume systems are used in the kitchen, exercise room, pool, and guest corridors/elevator areas on each hotel level. (Two of the three air handling units serving the kitchen area provide makeup air to the kitchen hoods, supplying 100% outdoor air.) The following is a list of MCCCH's air handling units/makeup air units and the components of their design.

Air Handling Units 1 and 2: are located in a mezzanine level mechanical room and serve all spaces in the conference center (Variable Volume), see Sketch 1, Section A. The contents of each AHU include min/max OA dampers, re-circulated air with a variable frequency drive fan, economizer, mixing box, filter (30% ASHRAE efficiency), pre-heat coil, cooling coil, heating coil, and a variable frequency drive supply fan all mounted on a 4" high housekeeping pad with ¼" thick neoprene pads. Both air handling units are balanced at a supply air cfm of 50,000, a max OA cfm of 50,000, and a minimum OA cfm of 30,000.

Air Handling Unit 3: is also located in the mezzanine level mechanical room. It serves the restaurant area on the main level between the conference center and hotel (Variable Volume), see Sketch 1, Section B. The contents of the AHU include min/max OA dampers, re-circulated air with a variable frequency drive fan, economizer, mixing box, filter (30% ASHRAE efficiency), cooling coil, heating coil, and a variable frequency drive supply fan all mounted on a 4" high housekeeping pad with ¼" thick neoprene pads. Air handling unit 3 is balanced at a supply air cfm of 12,500, a max OA cfm of 12,500, and a minimum OA cfm of 5,000.

Air Handling Unit 4: is located in the mezzanine level mechanical room. It serves the kitchen area on the main level between the conference center and hotel (Constant Volume), see Sketch 1, Section C. The contents of the AHU include min/max OA dampers, re-circulated air, economizer, mixing box, filter (30% ASHRAE efficiency), cooling coil, heating coil, and supply fan all mounted on a 4" high housekeeping pad with ¼" thick neoprene pads. Air handling unit 4 is balanced at a supply air cfm of 9,000, a max OA cfm of 9,000, and a minimum OA cfm of 2,250.

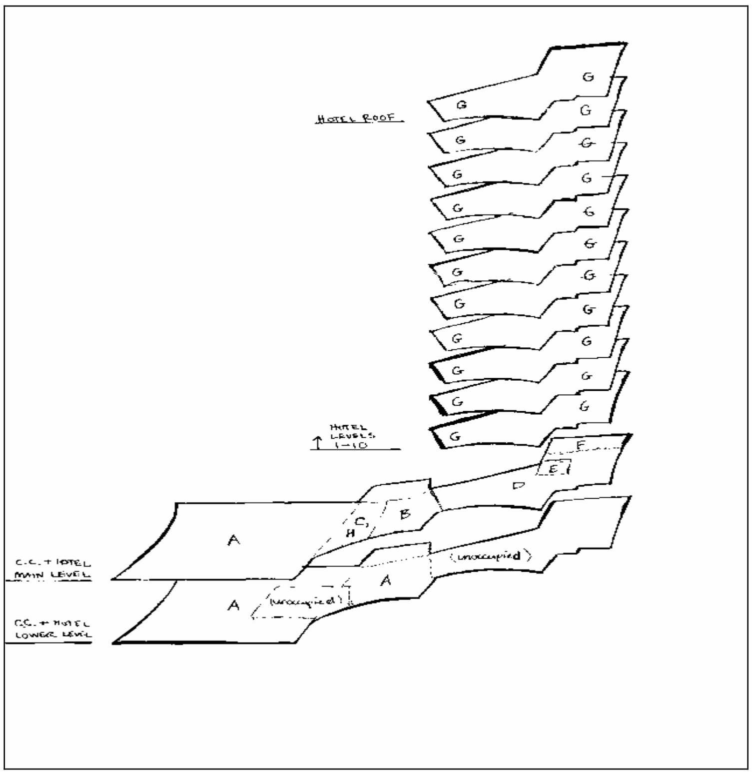
Air Handling Unit 5: is located in a mechanical room on the main hotel level, farthest from the conference center. It serves the hotel's first floor areas (Variable Volume), see Sketch 1, Section D. The contents of the AHU include min/max OA dampers, re-circulated air with a variable frequency drive fan, economizer, mixing box, filter (30% ASHRAE efficiency), cooling coil, heating coil, and a variable frequency drive supply fan all mounted on a 4" high housekeeping pad with ¼" thick neoprene pads. Air handling unit 5 is balanced at a supply air cfm of 9,000, a max OA cfm of 9,000, and a minimum OA cfm of 2,250.

Air Handling Unit 6: is also located in the mechanical room on the main hotel level, farthest from the conference center. It serves the exercise area on the hotel's main level (Constant Volume), see Sketch 1, Section E. The contents of the AHU include min/max OA dampers, re-circulated air, economizer, mixing box, filter (30% ASHRAE efficiency), cooling coil, heating coil, and supply fan all mounted on a 4" high housekeeping pad with ¼" thick neoprene pads. Air handling unit 6 is balanced at a supply air cfm of 1,400, a max OA cfm of 1,400, and a minimum OA cfm of 350.

Air Handling Unit 7: is ceiling mounted above a stairway near the pool area on the hotel's main level. It exists solely to provide air to the pool area (Constant Volume), see Sketch 1, Section F. The contents of the AHU include min/max OA dampers, re-circulated air, economizer, mixing box, filter (30% ASHRAE efficiency), preheat coil, cooling coil, heating coil, and supply fan. Additionally, the coils inside AHU 7 are coated with heresite to prevent corrosion. Air handling unit 7 is balanced at a supply air cfm of 3,200, a max OA cfm of 3,200, and a minimum OA cfm of 1,000.

Air Handling Units 8 and 9: are both located on the roof of the hotel tower. Each provide ventilation (100% OA) to the guest corridors and elevator lobbies on hotel levels 2 through 10 (Constant Volume), see Sketch 1, Section G. The contents of each AHU include an economizer, filter (30% ASHRAE efficiency), preheat coil, cooling coil, heating coil, and supply fan all mounted on a factory finished 18" high roof curb. Air handling unit 8 is balanced at a supply air cfm of 4,000, a max OA cfm of 4,000, and a minimum OA cfm of 4,000. Air handling unit 9 is balanced at a supply air cfm of 5,000, a max OA cfm of 5,000, and a minimum OA cfm of 5,000. (There are no AHUs that serve the individual hotel guestrooms. Each of these areas is provided for by an individual vertical fan coil unit.)

Makeup Air Units 10 and 11: are located in the mezzanine level mechanical room. They provide 100% outdoor air to the kitchen area in order to offset the kitchen exhaust hoods (Constant Volume), see Sketch 1, Section H. The contents of each AHU include an economizer, filter (30% ASHRAE efficiency), cooling coil, heating coil with face and bypass dampers, and supply fan all mounted on a 4” high housekeeping pad with ¼” thick neoprene pads. Makeup air unit 10 is balanced at a supply air cfm of 5,600, a max OA cfm of 5,600, and a minimum OA cfm of 5,600. Makeup air unit 11 is balanced at a supply air cfm of 6,025, a max OA cfm of 6,025, and a minimum OA cfm of 6,025.



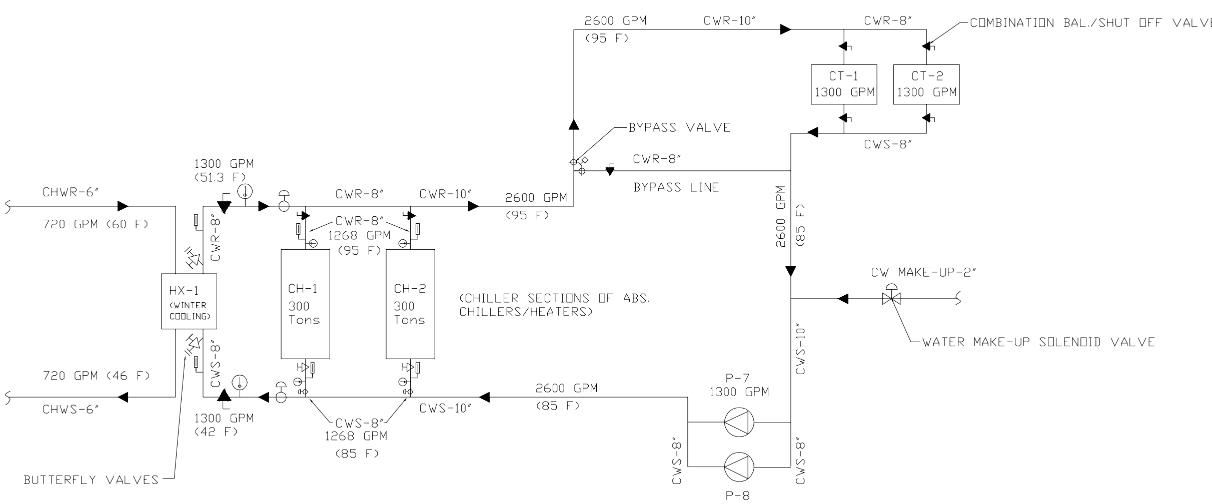
Sketch 1: 3-D Sketch of AHU Serving Areas (Not to Scale)

Water-Side Systems

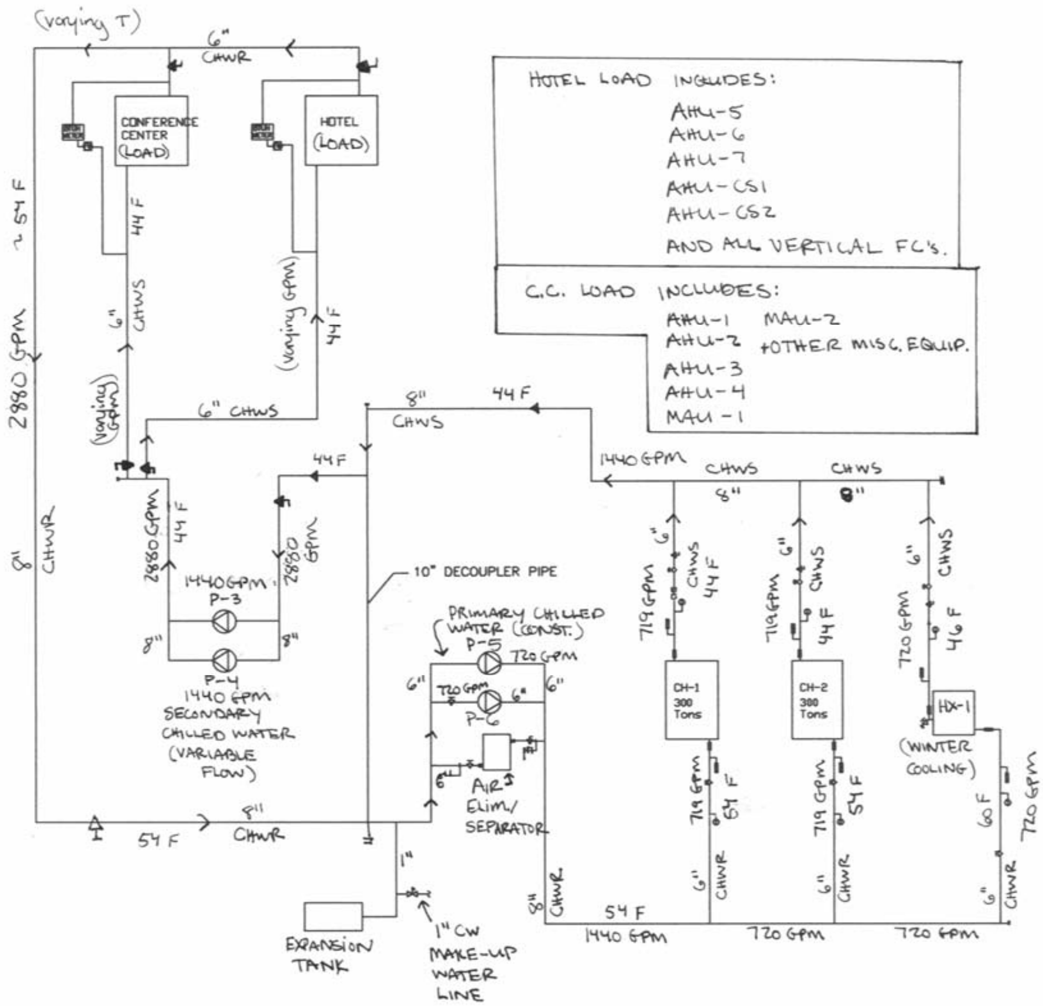
Two 300 ton/5,000 MBH direct fired absorption chillers with dual fuel natural gas/No. 2 oil burners provide chilled and hot water to the conference center and hotel. Each possesses a COP of 1.25 (at IPLV) and a heating efficiency of 85%. The absorption cycle in these machines, which takes the place of the compressor in vapor compression chiller equipment, is based on a water and lithium bromide solution. The two chillers/heaters are located along with the building's end suction pumps in a mechanical room on the lower level of the conference center. Two primary (one is stand-by), constant flow and two secondary, variable flow end-suction pumps distribute the building's chilled water supply. Hot water is delivered via two (one is stand-by), constant flow end suction pumps. Two 1300 gpm cooling towers assist the absorption chillers and are located on the roof of the hotel. This condenser water loop is driven by two constant volume end suction pumps. In the winter months, a 1300/720 gpm plate and frame heat exchanger operates within the condenser water loop to provide "free cooling." Schematics of this overall system can be found in the next section of this report.

Schematic Drawings of the Mechanical Existing Conditions

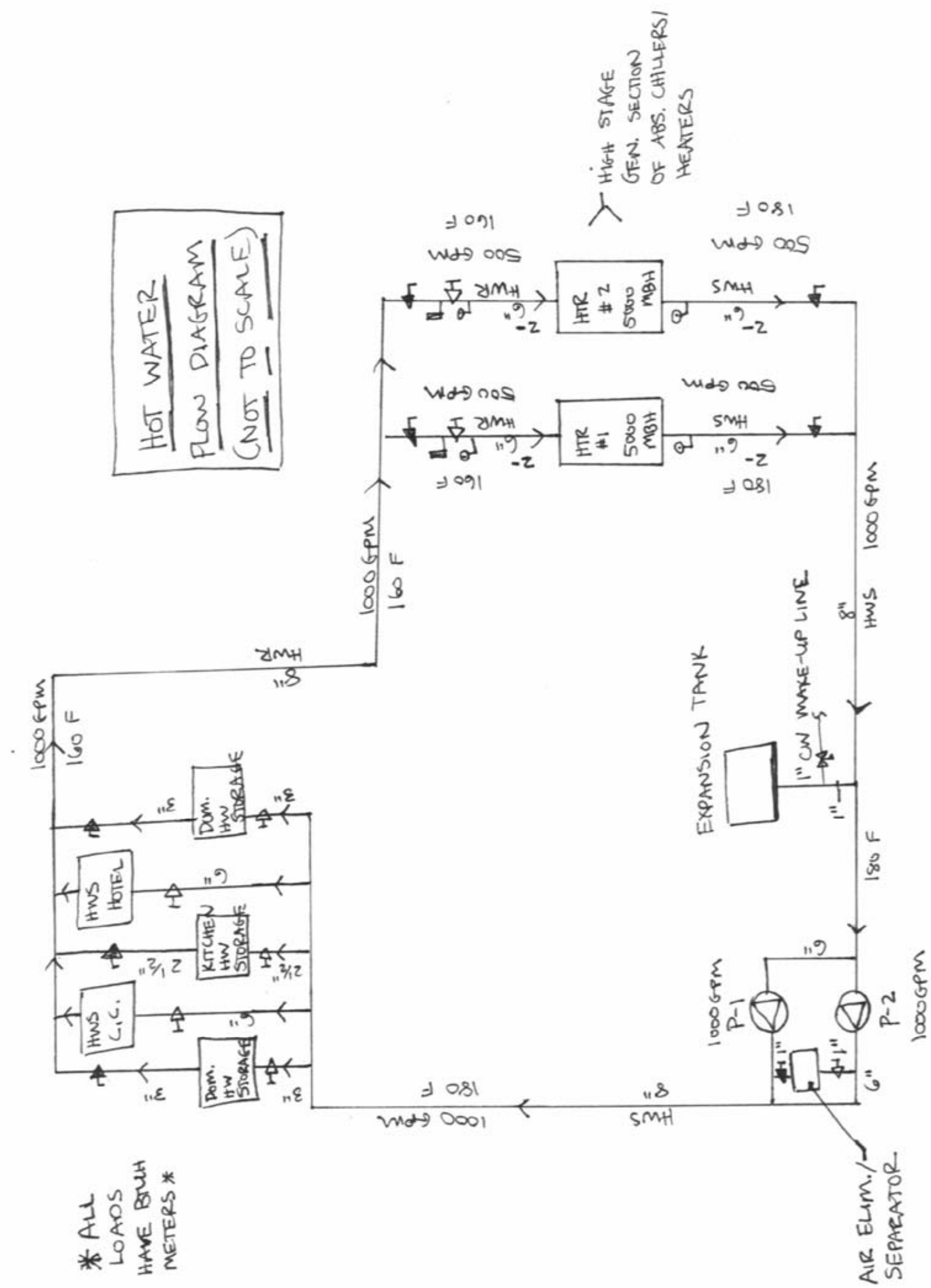
The following are simplified schematic drawings for MCCCH's condenser water, chilled water, hot water (heating and domestic), and air systems (AHUs).

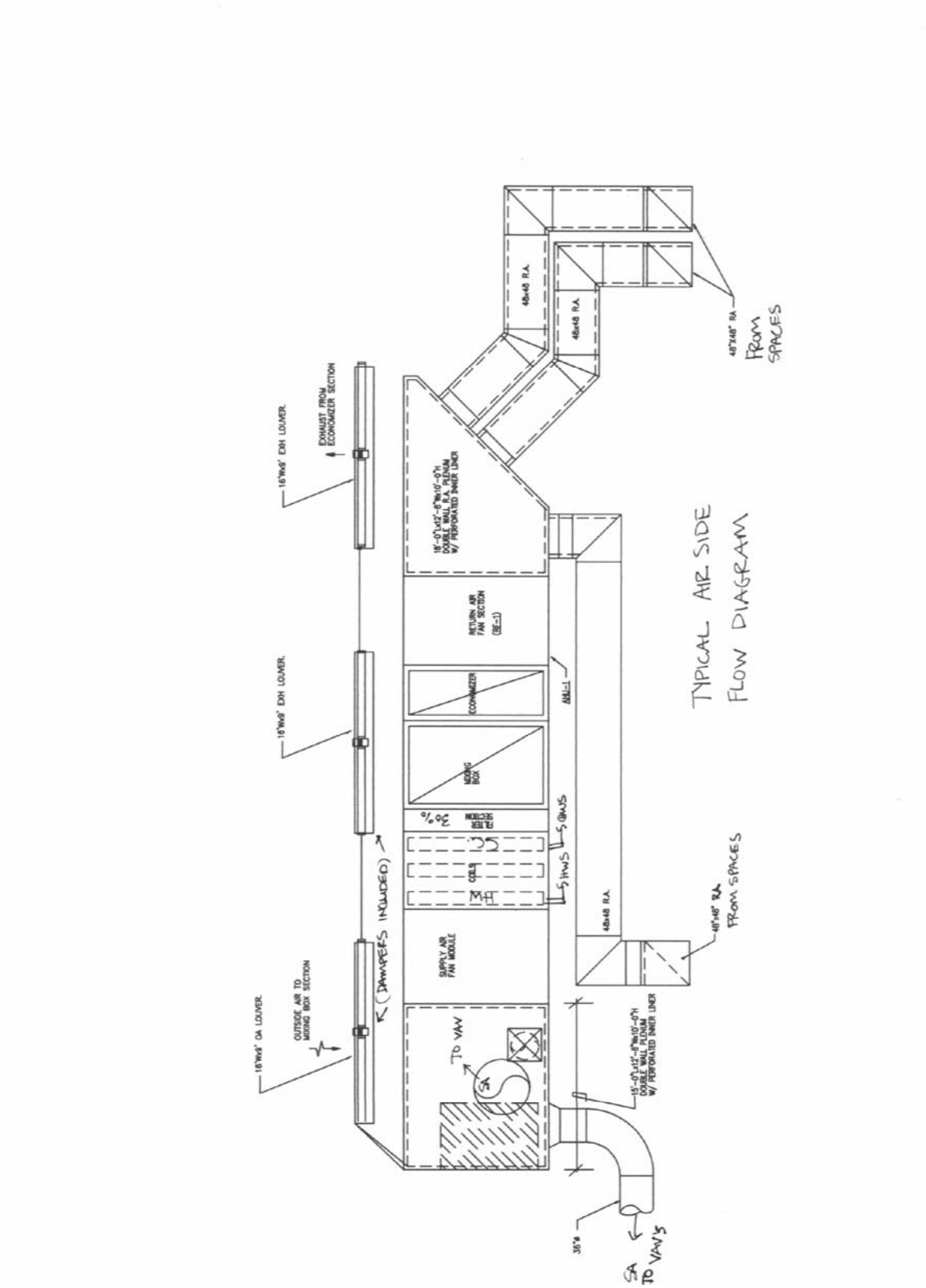


CONDENSER WATER FLOW DIAGRAM
(NOT TO SCALE)



CHILLED WATER FLOW DIAGRAM (NOT TO SCALE)


HOT WATER
FLOW DIAGRAM
(NOT TO SCALE)
HIGH STAGE
GEN. SECTION
OF ABS. CHILLERS/
HEATERS
HTR #1 5000 MBH
HTR #2 5000 MBH
EXPANSION TANK
1" CW MAKE-UP LINE
AIR ELIM./
SEPARATOR
* ALL
LOADS
HAVE BTUH
METERS *


TYPICAL AIR SIDE
FLOW DIAGRAM
OUTSIDE AIR TO MIXING BOX SECTION
EXHAUST FROM ECONOMIZER SECTION
(DAMPERS INCLUDED)
SUPPLY AIR FAN MODULE
COILS
FILTER SECTION
30%
MIXING BOX
ECONOMIZER
RETURN AIR FAN SECTION
48x48 R.A.
FROM SPACES
TO VAV
SA TO VAV'S

Schedule of Operation

The hours of operation for MCCCH were assumed to be different between the hotel and conference center portions of the building. The hotel was designed to be operating 24-hours/day, 365 days/year. The conference center was assumed to have the following operation schedule:

Monday-Thursday	7:00AM to 7:00PM
Friday-Sunday	6:00AM to 12:00PM

It was also assumed that both the hotel and conference center would be open for events on holidays, etc.

Outdoor Design Conditions

Climatic Design Conditions for 1,459 locations throughout the United States, Canada, and around the world are listed in Chapter 27 of the ASHRAE Fundamentals. This chapter gives a table containing design information pertaining to heating and cooling in buildings. For the MCCCH project, the design condition data from the Washington, District of Columbia location was used and is as follows:

Summer Conditions (ASHRAE 1%) at Atmospheric Pressure in D.C.

Dry Bulb Temperature: 95 ^{o}F

Coincident Wet Bulb Temperature: 76 ^{o}F

Summer Daily Range: 16.6 ^{o}F

Winter Conditions (ASHRAE 99%) at Atmospheric Pressure in D.C.

Dry Bulb Temperature: 15 ^{o}F

Coincident Wet Bulb Temperature: 12.2 ^{o}F

Indoor Design Conditions

The indoor design conditions were specified in MCCCH's project documents / specifications and are the following:

Dry Bulb Temperature: 72 ^{o}F

Relative Humidity: 50%

(MCCCH's spaces were to be kept within an approximate 1.5 ^{o}F range of this set point.)

Air Handling Unit Operation

Description

The exact specifications and areas served by the different air handling units throughout MCCCH have already been mentioned above. However, the way in which the AHUs operate has not yet been analyzed.

Most of the air handling units throughout the building supply 50-52 degree F air to the different spaces. All units, except the MAUs in the kitchen area, have outdoor air (OA) dampers running on an economizer cycle between 100% OA and the minimum calculated ventilation air rate. The units also contain centrifugal supply and return fans that slightly pressurize the building spaces. This pressurization was done in such a way that a special smoke containment system could be installed throughout the conference center portion of the building. Smoke zones and fan shut-off and start-up sequences were completely simulated and implemented. Variable speed controls were used on all the different units as well (see AHU descriptions above). Most units do not incorporate the use of ducted returns.

Sequence of Operation

The air handling units in the hotel portion of the building operate almost constantly throughout the year while the conference center units operate on an occupied/unoccupied basis. Normally, on days that the conference center will be occupied, the AHUs that serve the conference center will begin a warm-up approximately one hour prior to the arrival of people. Most of the time, the building's OA dampers will be closed throughout this warm-up and the building will re-circulate its own air (while cooling or heating if the need be) until the return air reaches a "cool enough" or "warm enough" set point, (the exact set point was unknown for this report). At the time that people begin to arrive at the conference center, the exhaust fans will start relieving some of the building's indoor air while the OA dampers open to provide proper ventilation. At this point, the economizer cycle will be in full use and the OA will be used to cool when its temperature is less than the “cool enough” set point. At higher OA temperatures, the OA dampers will again close and the building’s AHUs will take care of the building’s cooling load. Temperature sensors and air mixing sensors control the economizer cycle and OA dampers. The mechanical cooling will provide 50-52 degree F air at the AHUs. Once this air reaches the VAV boxes or the supply diffusers, it is normally somewhere around 55 degrees F. When a space becomes unoccupied, terminal reheat is used throughout the conference center spaces until certain systems shut off.

Safeties that exist for the building’s air-side system include pre-heating and heating coil bypass and recirculation pumps in order to avoid the freezing-up of any AHU coils. Smoke detectors and sensors/fire dampers were installed throughout the ductwork and major air-side equipment in the building. These devices play into the smoke containment system, another safety, which was mentioned earlier in the report.

Direct-Fired Absorption Chiller Operation

Description

The Montgomery County Conference Center and Hotel’s two direct-fired absorption chillers run mainly on natural gas but contain duel fuel burners for back-up use with No. 2 fuel oil. Each possesses a COP of 1.25 (at IPLV) and a heating efficiency of 85%. The absorption cycle in these machines, which takes the place of the compressor in traditional chiller equipment, is based on a water and lithium bromide solution.

Sequence of Operation

With a hotel, the demand for hot water is almost always constant. Therefore, normally, at least one absorption chiller in MCCCH is operating at all times. (During the day, people in the hotel need hot water along with the hotel’s kitchen and laundry areas. At night, hotel guests demand even more hot water while the laundry also continues.) However, this is ok due to the fact that the hotel also needs cooling around the clock. With at least one absorption chiller running at all times, both heating and cooling will be produced at a constant rate. Most of these energies are used up by just the hotel. As the conference center's occupancy changes, MCCCH's second absorption chiller will turn on and off to provide the necessary treatment in that area of the building.

When the heating and cooling loads/demands do not completely match between the building spaces and absorption machines, thermal storage is implemented. The usual case is that the building demands more cooling/chilled water than hot water. For this situation, MCCCH's design engineers installed hot water holding tanks for the different areas of the building that are likely to demand a lot of hot water all at once (kitchen, hotel, and laundry).

When chilled or hot water is needed in either portion of the building, the respective pumps will turn on and begin pumping water to the particular area. One of the absorption chillers/heaters is likely to already be running and will begin supplying these pumps. (Chiller/Heater supply water temperatures would be based on (OA) controls set throughout the building's AHUs and other mechanical equipment.) Flow sensors/meters will monitor all flows through the chillers/heaters and pumps. If any problems occur, valves will be shut and the pumps turned off prior to any damage. Normally, the chillers/heaters are kept running as they don't like to be quickly shut down and then turned on again. If a severe problem would exist with the chillers/heaters, they would be shut down immediately. In an additional effort to avoid the need for a quick shut down, several stand-by pumps were also installed and wired to start-up at the failure of any lead pumps.

Mechanical System Redesign:

Introduction / Redesign Criteria and Objectives

The idea behind the mechanical system redesign of MCCCH was to improve the system's overall performance while also lowering its life-cycle cost. The redesign process followed a general outline put forth by the Pennsylvania State University's Architectural Engineering Department. "All modifications and changes related to the original building design and construction methodology were solely for the purpose of academic development. Changes and discrepancies in no way imply that the original design contained errors or was flawed. Different assumptions, code references, requirements, and methodologies were incorporated into the redesign; therefore, investigation results varied from the original design" (P.S.U. A.E. Thesis E-studio Disclaimer).

One major goal of the redesign involved improving matters that directly affected the building's owner and surrounding environment. As stated before, the main goal behind the redesign was to decrease the building mechanical and related systems' life-cycle costs. (Lower building system first costs were not primary objectives of the redesign as they were not considered driving factors for success. However, in the end, the overall redesign did reduce most system first costs.) One way in which all of this was attempted entailed decreasing the building's annual energy consumption. By doing this, the monthly/annual bills for the facility/building owner were decreased. This also resulted in a more environmentally friendly building. Less energy usage converted directly into less fossil fuel use, and less fossil fuel usage meant fewer emissions by both the building's electricity supplier and the building itself. In an area like Washington D.C. and Baltimore, such savings would be very appealing to building owners as utility rates can be extremely lofty. The fact that the redesign proved beneficial to the environment created even further encouragement.

Another goal for the successful redesign of MCCCH's mechanical system involved providing an improved, clear, and concise control/operation strategy for any altered or new building systems. This acted as a check for successful system implementation while it also highlighted additional benefits to the owner. The clearer the control and operation of the building systems, the easier the redesign would be to maintain for building operators.

A final objective for the redesign of MCCCH included providing a clear design while not losing any of the beneficial aspects of the building's original systems. Any changes made to the original building systems were checked for negative influences and all processes of review were based upon an "apples to apples" comparison.

Problem Statement / Solution Overview / Design Options and Considerations

The main mechanical system alteration consisted of selecting and designing an optimized chiller plant for MCCCH. The application of latent thermal ice storage was implemented throughout the design process of the optimum chiller plant configuration.

MCCCH's current chilling plant contains two direct-fired (natural gas) absorption chillers that were essentially thrown into the building's mechanical system and never optimized. The original mechanical design incorporated the use of two vapor-compression machines but, the building's absorption chilling machines were offered to the owner at a very low first cost. Because of this, the original design of the building's chiller plant was changed to include the absorption machines and not the vapor-compression machines. To my knowledge, the affects of this modification on the building's operating/life-cycle costs were never really studied. (At the time of design, the building's first costs were of main concern.)

Therefore, for my thesis project, this absorption chilling plant configuration was tested against several other central cooling plant designs in order to determine and select the most favorable chiller plant design for MCCCH. One comparative design arrangement involved vapor-compression (electric), water-cooled chillers coupled with hot water boilers (and later, thermal storage). Another comparison consisted of a hybrid absorption (natural gas)/vapor-compression (electric) plant designed for chilled and heating/domestic hot water production. The optimum chiller plant configuration was selected based upon each working arrangement's energy consumption, life-cycle cost, and simplicity of control.

From there, a preliminary design of the most favorable plant type was sized and created for MCCCH. This plant was then further optimized between four specific chiller plant equipment manufacturers; Carrier, McQuay, Trane, and York. Again, the optimum was selected based upon each arrangement/manufacturer's respective energy consumption, life-cycle cost, and simplicity of control for MCCCH.

Finally, the addition of a cool thermal storage system to the optimum chiller plant design was studied. Several different types of cool thermal storage were considered including daily full - chilled water storage, daily full - glycol ice storage, daily partial/load leveling - chilled water storage, daily partial/load leveling – glycol ice storage, and daily partial/demand limiting – glycol ice storage. Cost comparisons were made and feasibility studied in order to determine the best cool thermal storage design for MCCCH.

The most favorable cool thermal storage plant was then sized/designed and coupled with a downsized version of MCCCH's optimum chiller plant type (determined initially). This upgraded central chilling plant with cool thermal storage was then compared to the building's original optimum plant using respective energy consumption, life-cycle cost, and simplicity of control. Between the two plants, the paramount was selected and recommended as the final design for MCCCH's central chilling plant.

All central chilling plant comparisons utilized “up to date” utility rates for the building’s geographic location as the rates played a major role in selecting the optimum plant for MCCCH. Also, the final design recommendation was fully compared to MCCCH’s original cooling plant design.

Justification

Improved economics and lessened environmental impact were two of the expected outcomes of the redesign for the Montgomery County Conference Center and Hotel. As a result of optimizing the building’s central chilling plant, the building’s life-cycle costs were greatly decreased and less on-site energy was used.

However, to clarify, the reduction in on-site energy usage only occurred due to a switch from natural gas to electric driven chilling machines and will only positively affect the building’s *immediate* environment (as there are no longer direct emissions coming from the building’s natural gas usage). Despite the fact that electric chillers have higher efficiencies than natural gas absorption machines, generation and transmission losses make electricity highly inefficient (only 28% efficient to be exact). Therefore, MCCCH’s final redesign will most likely have a negative affect on the earth’s environment as a whole; given that the amount of indirect emissions from electricity usage far exceeds the amount of emissions produced by natural gas usage (for this particular building application).

In effect, the overall redesign of MCCCH’s central chilling plant can only really take credit for lowering the building’s life-cycle costs.

Coordination and Integration

The greatest coordination issue of the redesign involved the rearranging and addition of major equipment in the building’s central chilling/heating plant. Several pieces of new equipment were added to the building. Other existing equipment was kept and relocated, or simply replaced. Either way, space was an issue. Therefore, throughout the redesign, all areas allocated as mechanical rooms were precisely laid out in order to fit all equipment properly.

Another large integration issue entailed maintaining the building's superficial appearance throughout the redesign of the mechanical system. Several large pieces of equipment ended up being placed on or around the exterior of the building. However, these changes in no way detracted from the building itself. Much special care was taken to maintain MCCCH's architectural integrity.

Selection of Chiller Plant Type / Initial Chiller Plant Optimization and Design

As stated before, MCCCH's original central chilling plant consisted of 2-300 ton, direct-fired absorption chillers that provided the building with all of its chilled and heating/domestic hot water. Both chillers were powered by natural gas and also had the capability of using #2 fuel oil as backup.

Again, as mentioned earlier, these two absorption machines were essentially "thrown into" the building's mechanical design at the last minute. Large amounts of first cost savings for the owner played a major role in the hasty decision to change the design of the building's central chilling plant. However, there was some question in my mind as to whether or not this type of central chilling plant was best for this particular application. Was this the most energy efficient design and were the operating costs of this plant going to be competitive enough with those that could be produced by other types of central chilling plants given MCCCH's geographic area and specific utility rates? Did the owner know the implications of this last minute decision and how it may affect his building operation costs throughout the life of the building despite the very low initial costs? Therefore, my chiller plant optimization process began.

My first step involved developing a very thorough load/energy analysis/simulation of the Montgomery County Conference Center and Hotel using Carrier's Hourly Analysis Program (HAP) and the building's absorption cooling/heating central plant. This had to be done because no load or energy calculations could be obtained from the building's design engineers. Therefore, this step was crucial as all the modeling had to be extremely specific in order to produce accurate and useable output.

Obtaining and using the correct utility rates (electric and natural gas) for the building was one very important key to properly modeling MCCCH in Carrier's HAP. These rates would not only be used for modeling the absorption plant but for all other plant simulations and comparisons throughout the entire chiller plant optimization process. Both the electric and natural gas rate structures used in the modeling of MCCCH can be found in Appendix A.

Applying the correct building occupancy and thermostat schedules was also very key to the HAP modeling process. However, all of the building schedules utilized for this project will be covered later in the 'cool thermal storage system design' section of this report. (The same schedules were used for all central chiller plant configurations and simulations.)

Thus, the HAP calculations for MCCCH's central absorption chilling/heating plant were performed and the annual operating costs and energy consumption can be seen below in the HAP Table 2. The full HAP output for this plant can be viewed in Appendix C.

Annual Costs

Component	MCCCH_Corrected ($)
Air System Fans	22,314
Cooling	53,857
Heating	10,867
Pumps	16,834
Cooling Tower Fans	17,466
HVAC Sub-Total	**121,339**
Lights	63,996
Electric Equipment	6,004
Misc. Electric	0
Misc. Fuel Use	10,958
Non-HVAC Sub-Total	**80,958**
Grand Total	**202,297**

Annual Energy Consumption

Component	MCCCH_Corrected
HVAC Components	
Electric (kWh)	2,241,431
Natural Gas (therms)	269,646
Fuel Oil (na)	0
Propane (na)	0
Remote HW (na)	0
Remote Steam (na)	0
Remote CW (na)	0
Non-HVAC Components	
Electric (kWh)	2,701,812
Natural Gas (therms)	47,263
Fuel Oil (na)	0
Propane (na)	0
Remote HW (na)	0
Remote Steam (na)	0
Totals	
Electric (kWh)	4,943,243
Natural Gas (therms)	316,909
Fuel Oil (na)	0
Propane (na)	0
Remote HW (na)	0
Remote Steam (na)	0
Remote CW (na)	0

Table 2: HAP Output for MCCCH's Absorption Cooling/Heating Plant

This output was then compared to two different types of central chilling plant designs/simulations for MCCCH.

One simulation consisted of a hybrid (gas absorption and electric vapor-compression) central chilling plant. In order to perform this simulation, one of the building's original absorption machines was coupled with a 'generic' 300 ton, centrifugal vapor-compression machine and one natural gas boiler in order to provide the building its chilled and heating/domestic hot water. One of the building's cooling towers was downsized from 1300 gpm to 900 gpm for the vapor-compression machine. The condenser water pump and piping for this tower was also downsized. The original chilled and hot water flow rates were maintained.

This HAP calculation was run and Table 3 below shows the annual operating costs and energy consumption for MCCCH with a hybrid gas and electric type of central chilling plant. Again, the full results of this simulation can be seen in Appendix C.

Annual Costs

Component	MCCCH_Corrected ($)
Air System Fans	22,504
Cooling	40,105
Heating	11,272
Pumps	15,999
Cooling Tower Fans	13,013
HVAC Sub-Total	**102,894**
Lights	64,495
Electric Equipment	6,051
Misc. Electric	0
Misc. Fuel Use	11,385
Non-HVAC Sub-Total	**81,931**
Grand Total	**184,825**

Annual Energy Consumption

Component	MCCCH_Corrected
HVAC Components	
Electric (kWh)	2,428,347
Natural Gas (therms)	158,245
Fuel Oil (na)	0
Propane (na)	0
Remote HW (na)	0
Remote Steam (na)	0
Remote CW (na)	0
Non-HVAC Components	
Electric (kWh)	2,701,812
Natural Gas (therms)	47,263
Fuel Oil (na)	0
Propane (na)	0
Remote HW (na)	0
Remote Steam (na)	0
Totals	
Electric (kWh)	5,130,159
Natural Gas (therms)	205,508
Fuel Oil (na)	0
Propane (na)	0
Remote HW (na)	0
Remote Steam (na)	0
Remote CW (na)	0

Table 3: HAP Output for MCCCH's Hybrid Cooling/Heating Plant

The other design/simulation consisted of an all electric vapor-compression central chilling plant. In order to perform this simulation, two 'generic' 300 ton centrifugal, vapor-compression machines and two natural gas boilers were used to provide the building with chilled and heating/domestic hot water. Both of the building's cooling towers were downsized from 1300 gpm to 900 gpm for the vapor-compression machines. The condenser water pumps and piping for these towers were also downsized. The original chilled and hot water flow rates were maintained.

This HAP calculation was run and Table 4 below shows the annual operating costs and energy consumption for MCCCH with an all electric type of central chilling plant. Again, the full results of this simulation can be seen in Appendix C.

Annual Costs

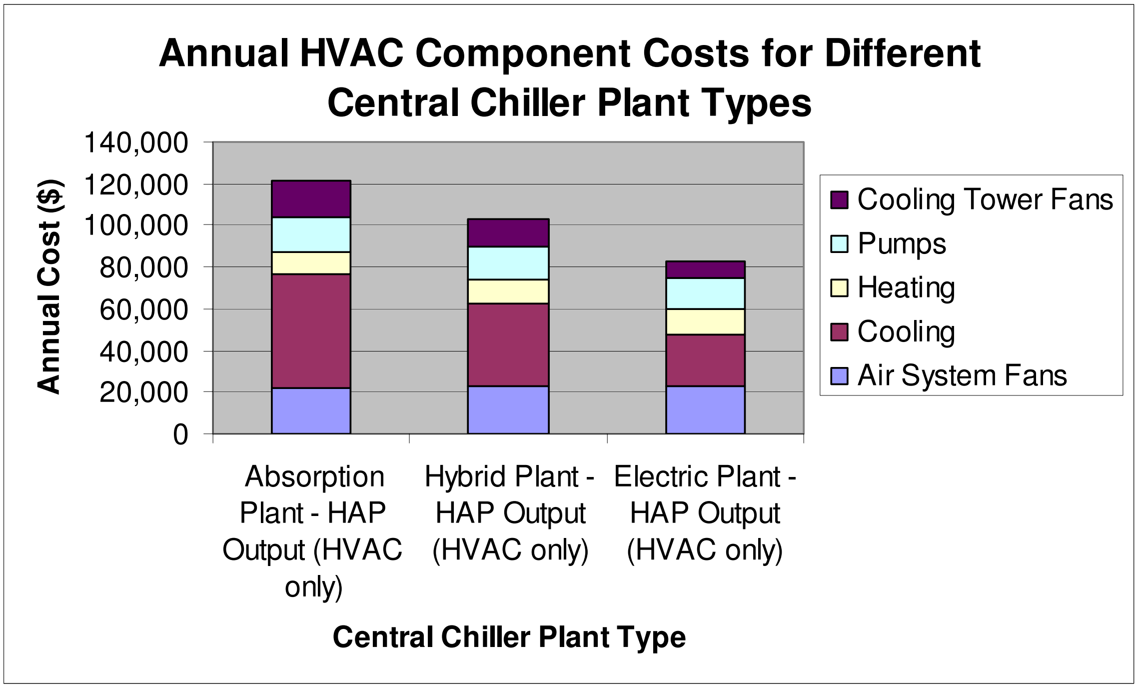
Component	MCCCH_Corrected ($)
Air System Fans	22,692
Cooling	24,654
Heating	12,168
Pumps	15,138
Cooling Tower Fans	8,435
HVAC Sub-Total	**83,087**
Lights	64,991
Electric Equipment	6,097
Misc. Electric	0
Misc. Fuel Use	13,042
Non-HVAC Sub-Total	**84,130**
Grand Total	**167,218**

Table 2. Annual Energy Consumption

Component	MCCCH_Corrected
HVAC Components	
Electric (kWh)	2,615,294
Natural Gas (therms)	46,845
Fuel Oil (na)	0
Propane (na)	0
Remote HW (na)	0
Remote Steam (na)	0
Remote CW (na)	0
Non-HVAC Components	
Electric (kWh)	2,701,812
Natural Gas (therms)	47,263
Fuel Oil (na)	0
Propane (na)	0
Remote HW (na)	0
Remote Steam (na)	0
Totals	
Electric (kWh)	5,317,106
Natural Gas (therms)	94,108
Fuel Oil (na)	0
Propane (na)	0
Remote HW (na)	0
Remote Steam (na)	0
Remote CW (na)	0

Table 4: HAP Output for MCCCH's All Electric Cooling/Heating Plant

A single table summary comparing these three chiller plant's annual operating costs (HVAC only) was compiled and can be found in Appendix A. A graph of this same information is displayed below.



Graph 3: Different Central Chiller Plant Operating Costs for MCCCH (HVAC only)

It was very easy to see that the electric central chilling plant gave MCCCH the lowest annual operating cost. However, in order to determine if that plant was indeed the optimum plant type, a first cost analysis and life-cycle cost analysis on each plant had to be calculated.

Table 5 below shows the first cost breakdowns for each plant type. All cost data was taken from the *2005 R.S. Means Mechanical Cost Data*.

Plant Types - First Cost Breakdowns

Absorption Plant		
	Equipment Cost	*Installation Cost*
Chillers (2)	$270,000.00	$23,300.00
Cooling Towers (2)	$68,400.00	$6,100.00
CW Pumps (2)	$15,200.00	$2,620.00
Piping (200'-10" CW)	$15,600.00	$8,000.00
	Total w/o chiller equipment cost:	**$139,220.00**

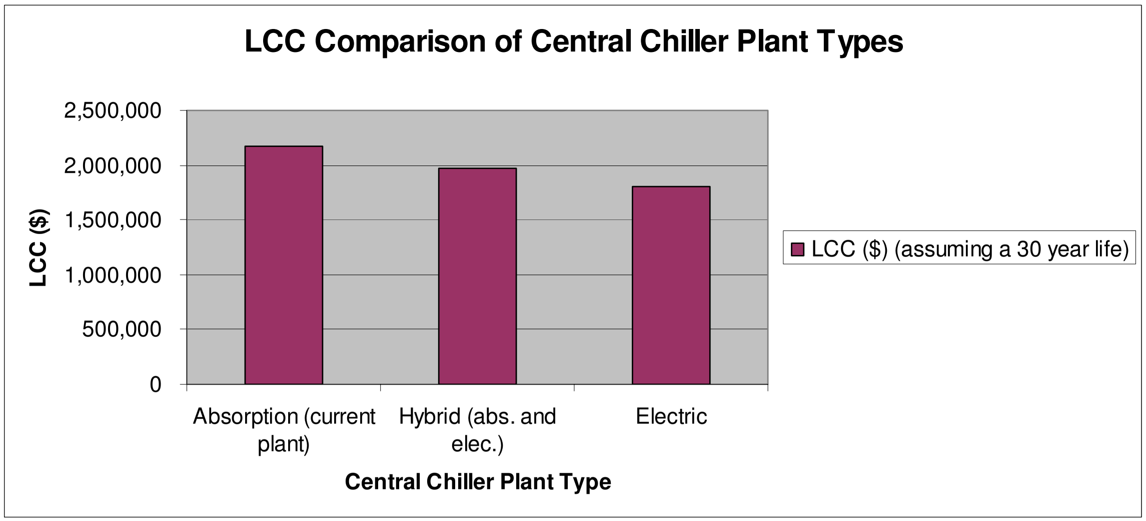
Hybrid Plant		
	Equipment Cost	*Installation Cost*
Chillers (2)	$240,000.00	$22,050.00
Cooling Towers (2)	$56,900.00	$5,375.00
CW Pumps (2)	$7,225.00	$1,880.00
Piping (200'-8" CW)	$6,500.00	$5,900.00
Boiler (1)	$25,000.00	$3,500.00
	Total w/o chiller equipment cost:	**$134,330.00**

Electric Plant		
	Equipment Cost	*Installation Cost*
Chillers (2)	$210,000.00	$20,800.00
Cooling Towers (2)	$45,400.00	$4,650.00
CW Pumps (2)	$6,850.00	$1,140.00
Piping (200'-8" CW)	$6,500.00	$5,900.00
Boiler (2)	$50,000.00	$7,000.00
	Total w/o chiller equipment cost:	**$148,240.00**

Table 5: Chiller Plant Type - First Cost Breakdowns

The following table and graph depict the life-cycle cost analysis of the three central plant types. Detailed life-cycle cost analysis calculations can be found in Appendix A.

Central Plant Option	Description	First Cost ($)	Additional First Cost ($)	Total First Cost ($)	First Cost Rank	Total Building Energy Costs/Year (HAP), ($)	Energy Cost Rank	LCC ($) (assuming a 30 year life)	LCC Rank	Comments
Absorption (current plant)	2-300 ton, direct-fired, natural gas	~270,000	139,220	409,220	3	202,297	3	2,168,988	3	
Hybrid (abs. and elec.)	1-300 ton, abs. and 1-300 ton, elec. (centrifugal)	~240,000	134,330	374,330	2	184,825	2	1,975,994	2	
Electric	2-300 ton, centrifugal, electric	~210,000	148,240	358,240	1	167,218	1	1,805,054	1	(select)



Graph 4: MCCCH Chiller Plant Type – LCC Analysis

From the above calculations, it was very apparent that an electric central chilling plant design of 2-300 ton centrifugal, vapor-compression chillers coupled with 2 natural gas boilers, was the most economical for the Montgomery County Conference Center and Hotel. Its L.C.C. was much lower than the other two plant options; one being the original plant design! However, before calculating the total overall savings, further optimization was performed.

Further Chiller Plant Optimization and Design According to Chiller Manufacturers

In an effort to further optimize MCCCH's central chilling plant design, Pacific Gas and Electric Company's CoolTools-*Chilled Water Plant Design and Specification Guide* was consulted. In there, a process for optimizing among the many different chiller manufacturers was found. The following is CoolTools' recommended optimization approach:

- Calculate or estimate the required plant total tonnage
- Pick a short list of vendors based on past experience and local representation
- Request chiller bids based on a performance specification
- Adjust bids for other first cost impacts
- Estimate the energy usage of options with a detailed computer model of the building/plant
- Calculate life cycle cost differences between options
- Select the chiller option with the lowest LCC

Therefore, the CoolTools approach was very close to what was done for the initial optimization of MCCCH's central cooling plant design. So, for this part of the optimization between different chiller manufacturers:

- The plant tonnage had already been calculated
- The list of vendors included Carrier, McQuay, Trane, and York

- Representatives from each manufacturer were contacted and chiller models and bids were requested (each manufacturer was to size and price 2-300 ton centrifugal chillers and send operating information)
- The bids were received and were corrected for first cost impacts (all models were received as well as one additional option – one McQuay 600 ton chiller (double compression) which was also analyzed)
- MCCCH's energy usage and operating costs for each chiller manufacturer's cooling plant were simulated by using Carrier's HAP.
- The life-cycle cost of MCCCH's central cooling plant was calculated for each manufacturer and compared.
- Finally, the manufacturer/cooling plant with the lowest L.C.C. was selected.

The following table and graph depict the HAP output/operating costs for each chiller manufacturer's cooling plant in MCCCH. Only HVAC operating costs were examined. Full HAP results for each chiller manufacturer's cooling plant in MCCCH can be found in Appendix C.

Carrier Plant - HAP Output (HVAC only)	
Component	**Annual Cost from Energy Usage ($)**
Air System Fans	22,659
Cooling	20,744
Heating	12,168
Pumps	15,116
Cooling Tower Fans	7,502
HVAC Total:	**78,189**

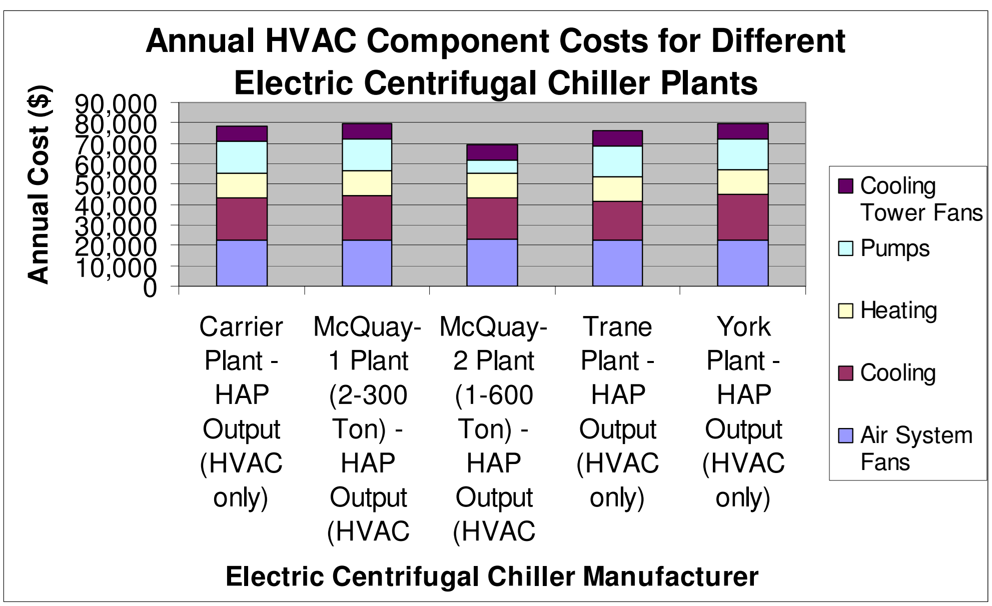
McQuay-1 Plant (2-300 Ton) - HAP Output (HVAC only)	
Component	**Annual Cost from Energy Usage ($)**
Air System Fans	22,674
Cooling	21,874
Heating	12,168
Pumps	15,125
Cooling Tower Fans	7,576
HVAC Total:	**79,417**

McQuay-2 Plant (1-600 Ton) - HAP Output (HVAC only)	
Component	**Annual Cost from Energy Usage ($)**
Air System Fans	22,868
Cooling	20,316
Heating	12,168
Pumps	6,402
Cooling Tower Fans	7,323
HVAC Total:	**69,077**

Trane Plant - HAP Output (HVAC only)	
Component	**Annual Cost from Energy Usage ($)**
Air System Fans	22,634
Cooling	18,828
Heating	12,168
Pumps	15,099
Cooling Tower Fans	7,377
HVAC Total:	**76,106**

York Plant - HAP Output (HVAC only)	
Component	**Annual Cost from Energy Usage ($)**
Air System Fans	22,678
Cooling	22,252
Heating	12,168
Pumps	15,129
Cooling Tower Fans	7,600
HVAC Total:	**79,827**

Table 7: **HAP Results between Different Chiller Manufacturers**



Graph 5: HAP Results between Different Chiller Manufacturers

Finally, the next two tables and graph depict the first cost breakdowns and life-cycle cost analyses for each manufacturer's central cooling plant in MCCCH. All cost data was taken from the *2005 R.S. Means Mechanical Cost Data* and more detailed life-cycle cost analysis calculations can be found in Appendix B.

Electric Plants - Additional First Cost Breakdowns

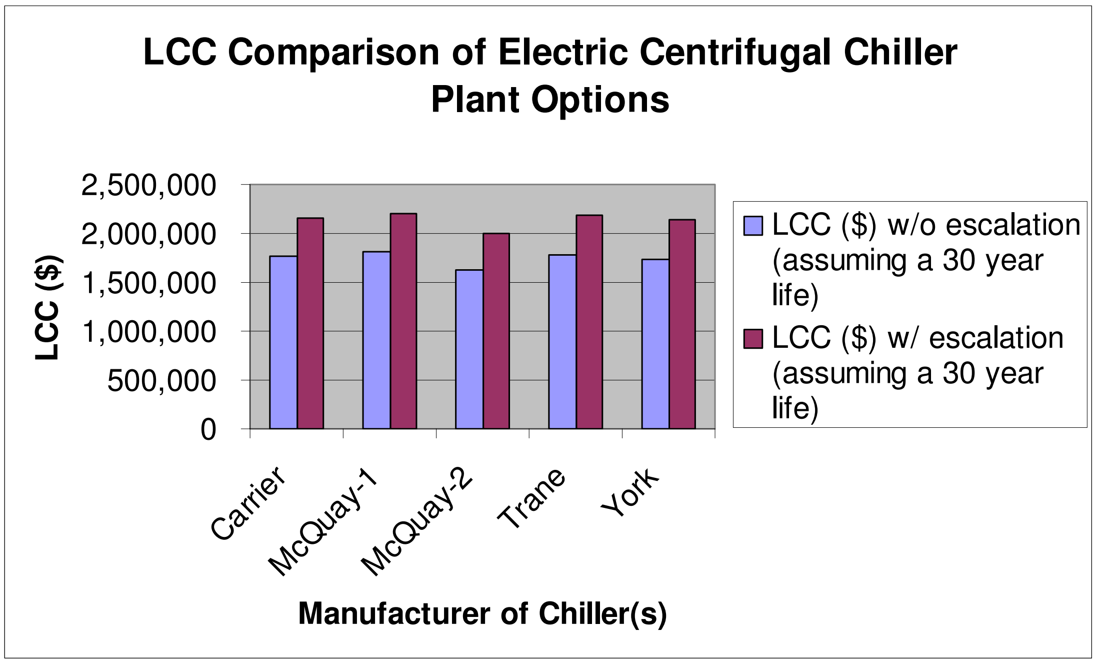
Electric Plant		
	Equipment Cost	*Installation Cost*
Chillers (2)	(manufacturer)	$20,800.00
Cooling Towers (2)	$45,400.00	$4,650.00
Pumps (2)	$6,850.00	$1,140.00
Piping (200'-8" CW)	$6,500.00	$5,900.00
Boiler (2)	$50,000.00	$7,000.00
	Total w/o chiller equipment cost:	**$148,240.00**

***McQuay plant with one 600 ton chiller has less first cost due to having only one chiller and not two. All other first costs for it are relatively the same.*

Table 8: Individual Chiller Manufacturer Electric Cooling Plant-First Cost Breakdowns

Manufacturer	Description	First Cost ($)	Other First Costs ($)	Total First Cost ($)	First Cost Rank	Total Building Energy Costs (HAP) ($)	Energy Cost Rank	LCC ($) w/o escalation (assuming a 30 year life)	LCC ($) w/ escalation (assuming a 30 year life)	LCC Rank	Comments
Carrier	2-300 ton, R-134a, 0.563 kW/ton	205,016	148,240 (in place)	353,256	3	162,224	3	1,759,842	2,160,247	3	
McQuay-1	2-300 ton, R-134a, 0.593 kW/ton	240,000	148,240 (in place)	388,240	4	163,495	4	1,805,064	2,208,598	5	
McQuay-2	1-600 ton, R-134a, 0.587 kW/ton	205,000	108,500 (in place)	313,500	1	153,761	1	1,621,529	2,000,793	1	(not enough redundancy)
Trane	2-300 ton, R-123, 0.512 kW/ton	250,820	148,240 (in place)	399,060	5	160,067	2	1,788,271	2,183,365	4	
York	2-300 ton, R-134a, 0.603 kW/ton	170,000	148,240 (in place)	318,240	2	163,918	5	1,738,472	2,143,047	2	(select)

Table 9: MCCCH Chiller Plant Manufacturers - LCC Analysis



Graph 6: MCCCH Chiller Plant Manufacturers - LCC Analysis

(The escalation factors mentioned throughout the above calculations refer to the U.S. Department of Energy's (DOE) predicted increase in electricity costs over the next 30 years (U.S. Average Fuel Price Indices). See Appendix B calculations for more information.)

Again, from the above calculations, tables, and graphs, it was very apparent that the York central chilling plant with 2-300 ton centrifugal, vapor-compression chillers coupled with 2 natural gas boilers was the most economical for the Montgomery County Conference Center and Hotel. Its L.C.C. was much lower than three of the other plant (manufacturer) options. One plant did have a lower L.C.C. than the York plant but it consisted of McQuay's 1-600 ton, double compression machine. It was decided that this plant did not yield enough redundancy and therefore, could not be selected as the optimum chiller plant for the building.

So, the optimum design seemed to be the York manufactured central chilling plant with 2-300 ton centrifugal, vapor-compression chillers coupled with 2 natural gas boilers. But, were there any ways to realize even more savings?

Cool Thermal Storage System Design

In order to examine the possibility of further operating cost savings, the addition of cool thermal storage to MCCCH's now electric central cooling plant was studied. This method of 'off-peak' air-conditioning seemed like it might be very beneficial to the building type and operating schedule.

ASHRAE's *Design Guide for Cool Thermal Storage* was used as a tool to design and evaluate a thermal energy storage system for MCCCH. The steps outlined in this design guide are listed below:

- Calculating Load Profiles
- Screening Initial Economics
- Selecting Storage Type
- Selecting Operating Strategy

- Determining Storage Interface Parameters
- Sizing the Cooling Plant and Storage
- Evaluating Economics

Everything on this list was covered in the cool thermal storage design for MCCCH. However, some things may fall out of the specific sequence.

Step 1: Calculating Load Profiles

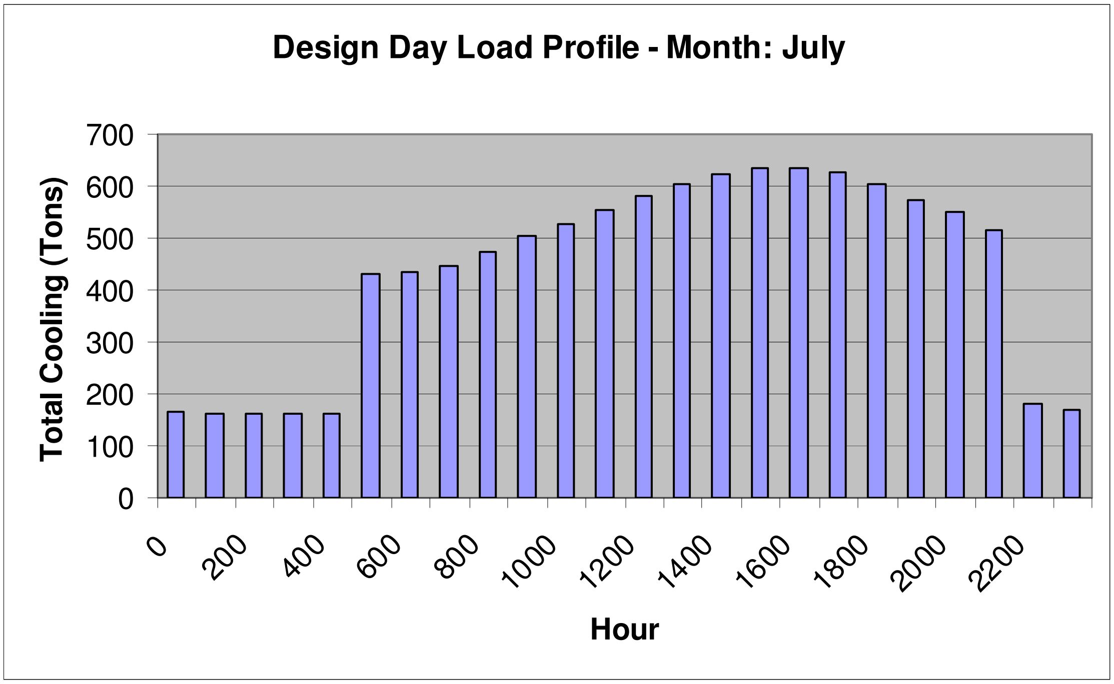
The building cooling load profile for MCCCH was produced by again using Carrier's Hourly Analysis Program. Schedules of the building's occupancy, equipment, lighting, and thermostats were estimated in order to produce MCCCH's overall building load profile as the building's real operating schedules were not known for this project.

The conference center was assumed to be at full occupancy from 8am to 6pm on Monday-Thursday. On the weekends (Friday-Sunday), this half of the building was assumed to be fully occupied from about 8:30am to 9:00pm.

During the week (Monday-Thursday), the hotel portion of MCCCH was expected to be occupied at 100% from 5am to 6pm. For all other times, it was assumed to be at 85%. On the weekends (Friday-Sunday), the hotel was expected to be occupied at 100% at all times.

The building's thermostat schedules throughout the building also followed these hourly trends and the individual graphs of all of MCCCH's operation schedules can be viewed in Appendix D.

Finally, all of this data was input into HAP and the resulting design day cooling load profile for July (the calculated design month for this geographic area) was created.



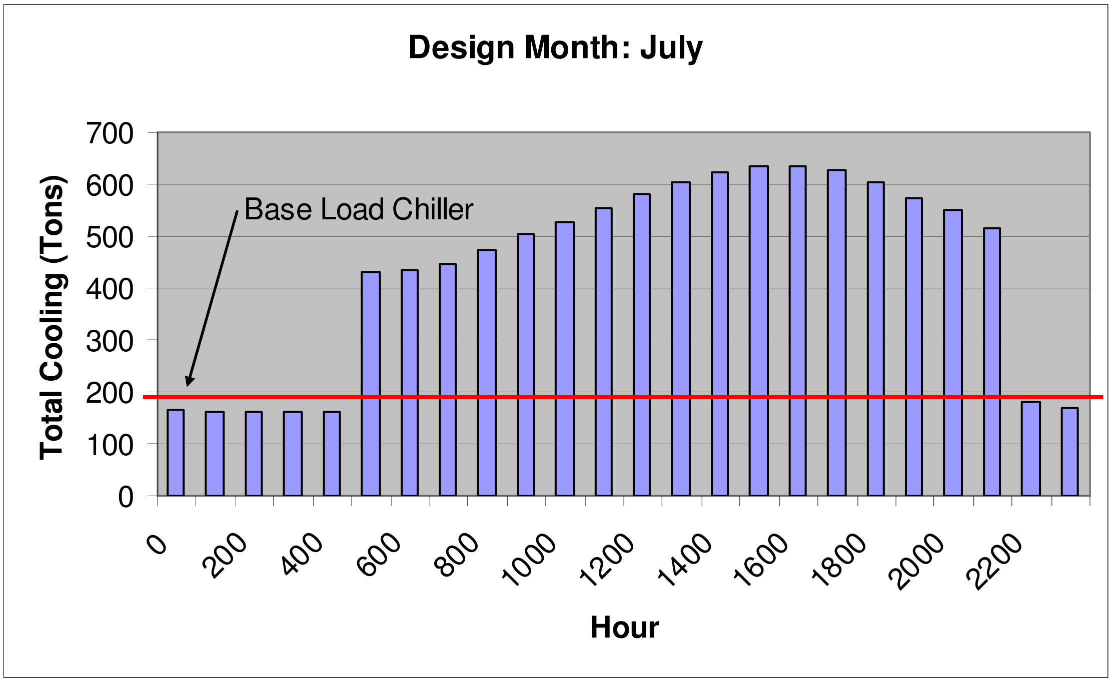
Graph 7: MCCCH's Design Day Load Profile (July)

Step 2: Initial Economics

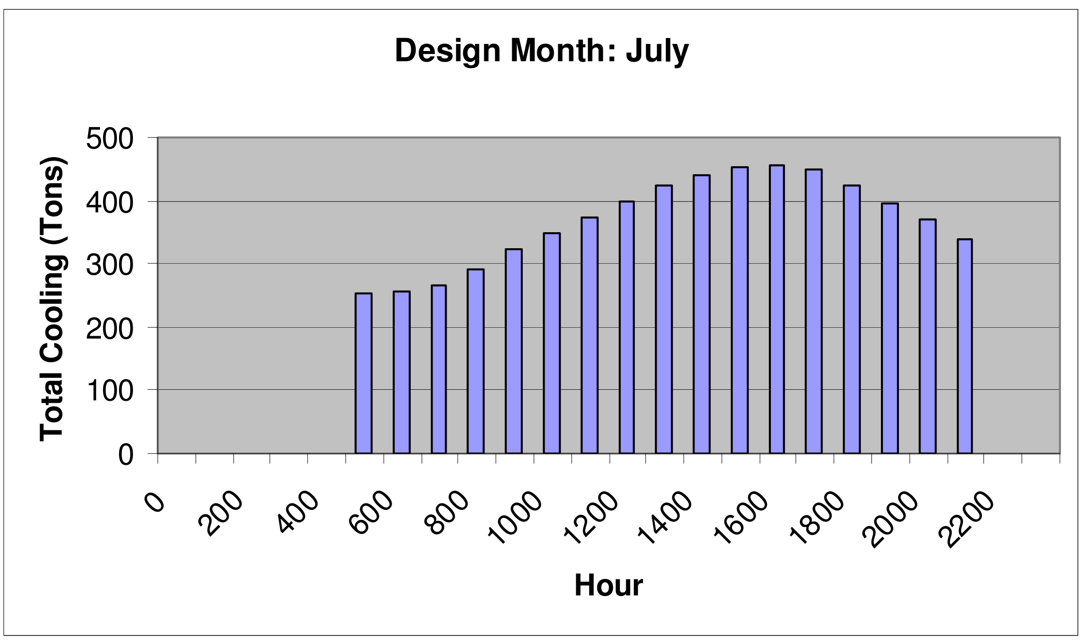
By looking at the building's design day cooling load profile from step 1, one could see that there existed a pretty significant base cooling load at all times. This issue was due to the fact that half of the building was a hotel that operated almost 24 hours every single day (or at least that was how it was assumed to operate). Either way, this situation was not very friendly to the idea of cool thermal storage.

With all types of cool thermal storage, there has to be a certain amount of 'charging' time or, an amount of time when there is no building cooling load. MCCCH's cooling load profile did not allow for any of this 'charging' time.

So, it was decided that the building must be analyzed for the application cool thermal storage in conjunction with a base load chiller. The following two figures depict the process of adding this base load chiller. (The chiller itself was approximately 180 tons.)



Graph 8: MCCCH's Design Day Load Profile (July) – Adding a Base Load Chiller



Graph 9: MCCCH's Modified Design Day Load Profile (July) (After the Addition of a Base Load Chiller)

Given this new cooling load profile, initial economic analyses of cool thermal storage system designs could be conducted for MCCCH. The following storage mediums and control strategies where evaluated as part of this initial study:

Daily, Full Storage
Daily, Full Storage
Daily Partial Storage, Load Leveling
Daily Partial Storage, Load Leveling
Daily Partial Storage, 50% Demand Limiting

(Ice Harvesting was not considered in this project due to the fact that the building would never have 3 or more consecutive days per week of being unoccupied.)

By using the ASHRAE *Design Guide for Cool Thermal Storage's* 'rules of thumb', nominal chiller and storage capacities were calculated for each of the storage mediums and operating strategies listed above. Electrical 'on-peak' and 'off-peak' times and charges were considered as well as MCCCH's modified cooling load profile. The resulting chiller and storage sizes are listed in the table below. Full calculations can be viewed in Appendix D.

Summary of Ice Storage Calculations:			
Operating Strategy	**Storage Type**	**Chiller Size (tons)**	**Storage Size (ton-h)**
Non-Storage	---	420	---
Daily, Full Storage	Chilled Water	894	6258
Daily, Full Storage	Glycol Ice	1277	6258
Daily Partial Storage, Load Leveling	Chilled Water	261	1825
Daily Partial Storage, Load Leveling	Glycol Ice	286	1400
Daily Partial Storage, 50% Demand Limiting	Glycol Ice	467	2288

**Ice Harveting not considered for several reasons.

Table 10: Ice Storage Calculation Results

From here, the design guide outlined a method for cost comparing the different storage and operating types. General numbers and more 'rules of thumb' were given in order to calculate the comparisons. Estimates and cost assumptions that were used for this design project included:

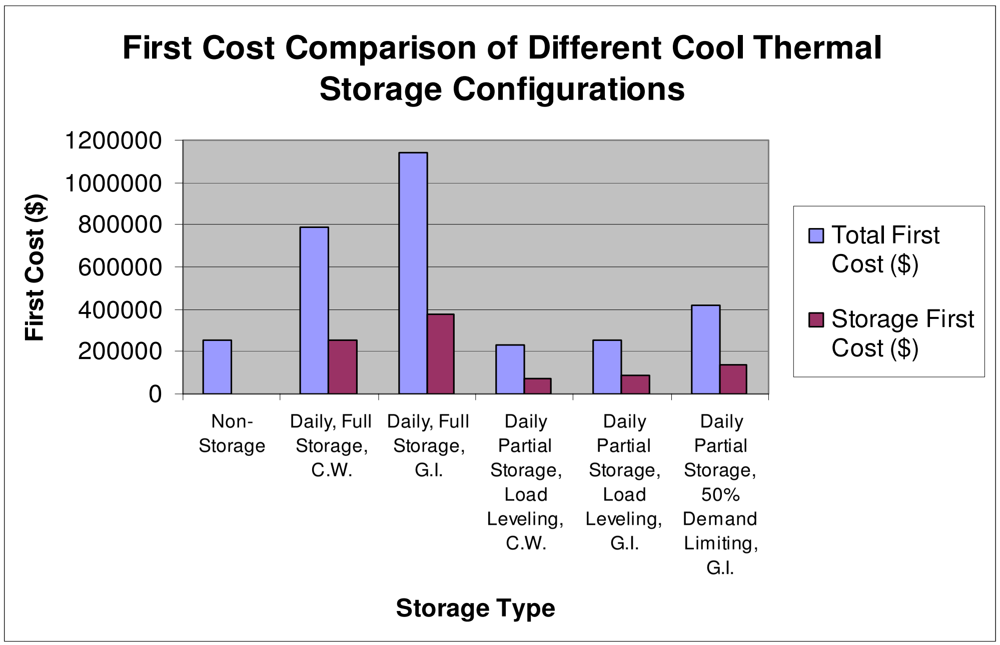
- a chiller efficiency of 70%
- an 'on-peak' demand charge of $8/kW
- a chiller cost of $600/ton

- a chilled water storage cost of \$40/ton-hr
- a glycol ice storage cost of \$60/ton-hr
- annual demand savings for full storage = 3 months of peak savings and 5 months of 70% of peak savings
- annual demand savings for partial storage = 100% for 8 months

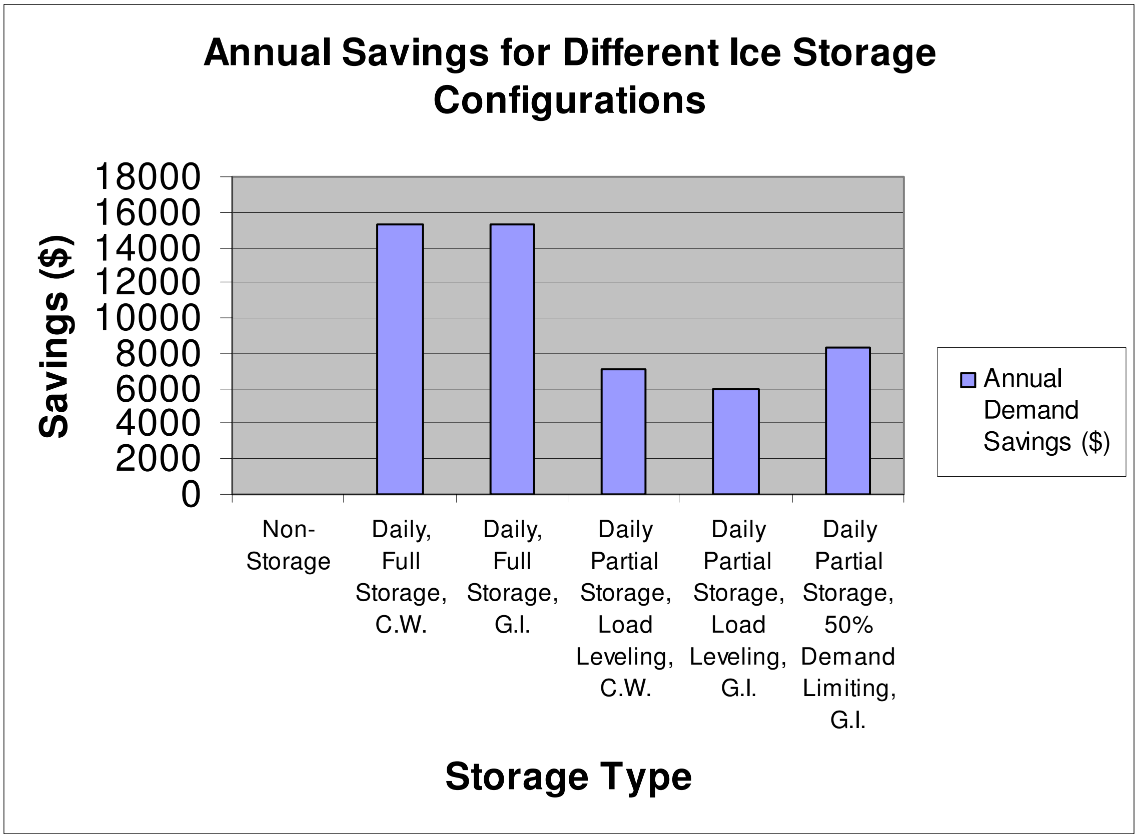
The results of all of these calculations can be seen in the table below. The graphs following the table also depict the calculation results and serve as a visual comparison of all the storage types/strategies.

Ice Storage Economic Comparison:														
Operating Strategy	Storage Type	Chiller Size (tons)	Storage Size (ton-h)	On Peak (kW)	Peak Month Demand Savings (kW)	Annual Demand Savings (kW)	Annual Demand Savings ($)	Chiller First Cost ($)	Storage First Cost ($)	Total First Cost ($)	Difference from Non-Storage ($)			
Non-Storage	---	420	---	294	---	---	---	252000	---	252000	---			
Daily, Full Storage, C.W.	Chilled Water	894	6258	0	294	1911	15288	536400	250320	786720	534720	(more)	Simple Payback (Option 1):	34.98
Daily, Full Storage, G.I.	Glycol Ice	1277	6258	0	294	1911	15288	766200	375480	1141680	889680	(more)	Simple Payback (Option 2):	58.19
Daily Partial Storage, Load Leveling, C.W.	Chilled Water	261	1825	182.7	111.3	890.4	7123.2	156600	73000	229600	-22400	(less)	Simple Payback (Option 3):	0.00
Daily Partial Storage, Load Leveling, G.I.	Glycol Ice	286	1400	200.2	93.8	750.4	6003.2	171600	84000	255600	3600	(more)	Simple Payback (Option 4):	0.60 (select)
Daily Partial Storage, 50% Demand Limiting, G.I.	Glycol Ice	467	2288	163.45	130.55	1044.4	8355.2	280200	137280	417480	165480	(more)	Simple Payback (Option 5):	19.81

Table 11: Results of Initial Ice Storage Economic Analysis



Graph 10: Results of Initial Ice Storage Economic Analysis – First Costs



Graph 11: Results of Initial Ice Storage Economic Analysis – Annual Savings

Step 3-4: Storage and Operating Strategy Selection

By looking at the results, it was easily seen that daily full storage was not the way to go for MCCCH. Even though the yearly savings would be very high, the first costs would also be very high. The chillers would have to be huge along with the thermal storage.

The best option seemed to be that of daily partial storage, load leveling. Both chilled water and glycol ice had lower total first costs than 'non-storage' and they had significant amounts of annual savings. Additionally, both of their payback periods were less than one year. Therefore, it was decided that MCCCH needed to go with load leveling, partial storage. The only decision involved chilled water storage or glycol ice storage…

The decision was not hard. MCCCH did not have a lot of room surrounding it for large thermal storage tanks. Its land lot was very small and its need for all the parking it could get was very great. Only one small area existed on the exterior of the building where thermal storage tanks might be able to be placed. This location was behind the building, between the conference center and hotel, and beside the generator fuel tank.

Therefore, because of this great lack of space, the smallest type of thermal storage tanks had to be selected. So, ultimately, this meant glycol ice storage (as chilled water storage/stratified tanks are extremely large in size (sensible storage=no phase change=larger tanks)). There was room at the building's site for glycol ice storage tanks.

Step 5: Storage Interface Parameters

Before a detailed design could take place, certain building distribution interface parameters had to be determined and/or verified. The parameters that were of greatest concern are listed below:

- The building was to have a 180 ton, base load, York, electric chiller (screw chiller to be exact).
- The building's chilled water supply temperature was 44F.
- Calmac ICEBANK ice storage tanks had the ability to supply chilled water temperatures between 34F and 44F (internal melt, ice-on-coil).
- It was decided that this project would be based on the Calmac ICEBANK ice storage system.
- The total chilled water flow rate to the building would be 1440 gpm (like the original design).
- The flow rate through the ice storage system would be determined by the Calmac ICEPICK program and the ice-making/cooling, York, electric, centrifugal chiller (whatever size that chiller might turn out to be).
- A heat exchanger would have to be used between the ice storage loop and the building's chilled water loop due to the large size of building/high cooling load > 500 tons

(A.D.G.C.T.S.); this heat exchanger would have a reasonable approach and would have a lot to do with selecting the discharge temperature of the ice storage.

- The system configuration would include the ice-making/cooling chiller upstream of the ice storage tanks for greater chiller efficiencies, the base load chiller would operate on the building chilled water side, and the heat exchanger would be staged with the base load chiller. (A flow diagram of the final design can be found in Appendix G.)

Step 6 and 7: Final Sizing and Economic Evaluations

Both hand calculations and Calmac's ICEPICK program were used in order to determine the final size of MCCCH's ice storage system. All the calculations and program outputs for this sizing process can be found in Appendix D. The final design included:

- (4) Calmac IceBank Model 1500 Tanks
- (1) York MAXE 300 Ton Ice-Maker/Chiller
- (1) Mueller Plate and Frame HTX (w/ 254 Plates/Frame, 1 Frame)
- (1) Bell and Gossett 50 hp pump
- (1) York YCWS Screw Chiller 180 Tons (Base Load)
- (1) 540 gpm Cooling Tower
- (1) 900 gpm Cooling Tower
- (2) Bell and Gossett 15 hp pumps
- (1) Bell and Gossett 10 hp pump
- (1) Bell and Gossett 5 hp pump

A schematic of this design can be found in Appendices F and G along with all of the equipment selection 'cut sheets'. The schematic shows all the design flow rates and the 'cut sheets' show all the design chiller efficiencies and/or kW/ton at different operating capacities.

The 4 Calmac model 1500 tanks were located outside of the building in the location that was mentioned before in this report. Their exact location can be seen on a drawing included in Appendix D. Additionally, the tanks were partially buried as to not affect the aesthetics of the building's exterior and/or the views from hotel guestrooms. Appendix D also shows a figure of a buried Calmac ice tank.

Finally, this combined ice storage and central chilling plant design for MCCCH was modeled and simulated using the DOS version of Carrier's Hourly Analysis Program (Version 3.2). Everything that was designed was modeled and simulated in the program in exactly the same way as the system was described earlier in this report. From the program's output, it was easy to see the reduction in the building's 'on-peak' kW demand. The table below displays this information in comparison to the 'optimized' (York) electric central cooling plant designed earlier in this study.

Energy Consumption Totals			
Billing Period	**York Electric Plant Peak kW Demand**	**Ice Storage Plant Peak kW Demand**	**Demand Savings (kW)**
January	700.7	557.2	143.5
February	699.8	589.6	110.2
March	838.7	793.6	45.1
April	892.1	827	65.1
May	1009.7	922.6	87.1
June	1089.8	931.6	158.2
July	1095.7	927.3	168.4
August	1092.1	934.3	157.8
September	1095.3	929.1	166.2
October	905.2	805.3	99.9
November	839.8	785.7	54.1
December	705.2	559.8	145.4
Totals:	10964.1	9563.1	1401

Table 12: Peak kW Savings with Ice Storage Plant (HAP output)

It was surprising to see that only 1401 peak kilowatts per year were saved by adding the ice storage system to MCCCH's central chilling plant. In effect, this only totaled up to a savings about $3,082.20 in operating costs per year (as the peak demand kW charge is $2.20/kW). Additionally, with a first cost value of

$425,922, the economics yielded a life-cycle cost of $2,249,006 (compare that to the original York electric plant with a L.C.C. of $2,143,047!) and a payback period of over 28 years. (These full economic calculations can be seen in Appendix D.)

So, in concluding this section of the report, it was very clear that the addition of this thermal ice storage design to MCCCH's central cooling plant was not economical and should not be done. Even though the tonnage of the central cooling plant was reduced by 20% and the annual amount of peak demand kilowatts was reduced, it wasn't enough to justify the initial added system first costs. There just weren't enough savings. (One reason for this could have been that the peak electric demand charge was just too small.)