Mechanical Systems Design

After much deliberation and a long process of brainstorming ideas for potential mechanical system designs for the Hilton Hotel at BWI Airport, it was finally decided to design a central chilled water plant for the building. This report clearly details the ideas for replacement and improvement of the hotel's mechanical systems. The goals for the project and the steps taken in this design process are described next.

Design Objectives

The original design of the mechanical systems for the Hilton Hotel at BWI Airport could be classified as a possible solution or workable design for the hotel. However, it is not by any means the best possible solution, and it is definitely possible to develop improved mechanical systems that will either replace or supplement the existing design.

In order to improve the mechanical systems design, a means of measuring the performance and relative costs must be established. The primary goals of this thesis design project for the BWI Hilton are to increase the energy efficiency of the building and decrease the life cycle costs of the systems and equipment. Other objectives sought for in this project include sustainability, design innovation, and improvement of the overall indoor environmental quality of the hotel.

As stated above, one of the major goals is to improve the energy efficiency of the BWI Hilton, and it is the primary objective of this thesis design project. All design ideas and decisions should be made based on this principle. To accomplish this goal of using energy more effectively, it is necessary to decrease the building's annual energy consumption of electricity and natural gas. In doing so, it will also be possible to make a more environmentally-friendly facility with reduced emissions.

Additionally, improvement to the mechanical systems could also result in a reduction of the life cycle costs of the BWI Hilton. Reduced life cycle costs will also cause the payback period on some of the equipment to become more reasonable. However, despite the lower life cycle costs, the first costs of the equipment could possibly increase. Unfortunately, too many times the primary objective of a building project is to decrease the first costs. The lower first costs look good for the owner, but this could be counter-productive in the long run. This is why the life cycle costs are more of a concern than the first costs for this design project.

Original Mechanical System Design

The original mechanical systems use a simple condenser water loop throughout the entire building. Water source heat pumps are used in all the guest rooms to exchange heat between the air and the water to cool the rooms. The four air handling units all have cooling coils served by the condenser water, which in turn operate similarly to heat pumps. The six rooftop units all use air-cooled DX coils for the cooling. Any heating in the system is provided by a boiler system that adds heat to the condenser water loop via a heat exchanger between the two systems.

The overall goal of energy efficiency for this thesis project could be realized in a variety of ways. The means chosen to accomplish this goal will be based on the existing design of the Hilton Hotel at BWI Airport and the potential to greatly improve the mechanical systems. A new central chilled water plant system to replace this condenser water loop system will be designed, developed, and to a certain degree, optimized. Many design alternatives and a variety of choices of equipment are available, but only a few could be properly researched and used for this thesis design project.

Chilled Water Plant Design

The design of a central chilled water plant system is a very long and complicated process. However, two books, many magazine articles, and the opinions of several manufacturers and mechanical designers served as guidance throughout the design of the chiller plant. According to the “CoolTools Chilled Water Plant Design Guide”, there are seven main steps in the design of a chilled water plant. The steps taken in the process are described next.

Pre-requisite:

To be able to follow these seven steps, it was first assumed that the required building cooling load was previously determined.

In this case, the peak cooling load for the chiller plant was found to be 640 tons by using HAP to simulate the hourly loads on the BWI Hilton. But to slightly oversize the system and provide some cushion in case any of the load calculations are not exact, a 10% safety factor was used. This then caused the total cooling load to be 700 tons.

Step 1:

> According to "CoolTools", the first step is to choose the chilled water flow distribution arrangement. If multiple pumps are used, the possible arrangements either use series pumping or parallel pumping. Possible choices for distribution include constant primary flow, constant primary/constant secondary flow, constant primary/variable secondary flow, and variable primary flow.

The chilled water flow distribution flow arrangement was chosen to be a two-pump parallel configuration with variable primary flow (VPF). This was done for several reasons. First, after running a HAP analysis to model each of the possible pumping arrangements, it was determined that the VPF system was the most energy efficient. This is mostly due to the reductions in pump energy as compared to the other main possibility of a constant primary/variable secondary flow configuration.

Additional savings can be realized when a comparison is made with the number of required pumps. In a constant primary/variable secondary flow system, two constant speed primary pumps and two variable speed secondary pumps are needed, and both sets are piped in parallel. In a VPF system, only two primary pumps with variable speed drives are required. Even though the two pumps will be larger than the primary pumps of the primary/secondary arrangement, the first costs will still be less than having to purchase four smaller pumps.

Step 2:

> The second step in the "CoolTools" design process is to determine the characteristics of the chilled water system. These characteristics include the chilled water supply (CHWS) temperature, maximum flow rate, and main pipe sizes. A variety of choices are available for all the CHWS temperature, flow rate and pipe sizes.

A range of CHWS temperature could be used in the chiller plant, as can be seen in Table 12 – Temperature Limits. After running some HAP simulations and discussing the options with several mechanical design engineers and manufacturer representatives, it was decided that the CHWS temperature would be best at 44 F.

Table 12 – Chiller Plant System Limits

	CHW		CW	
Limit	**LWT (F)**	**ΔT (F)**	**EWT (F)**	**ΔT (F)**
Low	40	10	85	10
High	48	20	85	20
Typical	45	10	85	10

A range of delta-Ts could be used for the design chilled water (CHW) temperature difference (delta-T). A similar method was used to choose this design condition as was described above. The most traditional CHW delta-T used is 10 F, but a higher delta-T of 12 F was chosen to be used for this chiller plant design. This was done because higher delta-Ts typically result in lower water flow rates and less pump energy consumption, as is described next.

Next, a maximum flow rate must be determined. A range of CHW and condenser water flow rates are listed in Table 13 – Relative Flow Rates. These flow rates are all directly related to the corresponding delta-Ts. At a typical delta-T of 10 F, the CHW flow rate would be 2.4 gpm/ton. But with the selected range of 12 F, a CHW flow rate of 2.0 gpm/ton will be used for this design. Therefore, with a peak cooling load of 700 tons, the CHW flow rate will be no more than 1400 gpm.

Table 13 - Flow Characteristics at Various Delta-Ts

ΔT (F)	CHW gpm/ton	CW gpm/ton	CHW Btuh/gpm	CW Btuh/gpm
10	2.4	3.0	5000	5000
12	2.0	2.5	6000	6000
14	1.71	2.14	7018	7009
15	1.6	2.0	7500	7500
16	1.5	1.88	8000	7979
18	1.33	1.67	9023	8982
20	1.2	1.5	10,000	10,000

The main pipe sizes right at the chiller(s) are selected next. In this case, with a maximum flow rate of 1400 gpm on the CHW side, and an assumed friction rate of 3 ft/100 ft, the required piping has an 8 in pipe size.

For a simple representation of the chilled water and condenser water systems, please see Figure 4 – Chilled Water Flow Diagram on the following page. Assumed temperatures and flow rates are shown for both the condenser water and the chilled water, as is discussed here in the design steps.

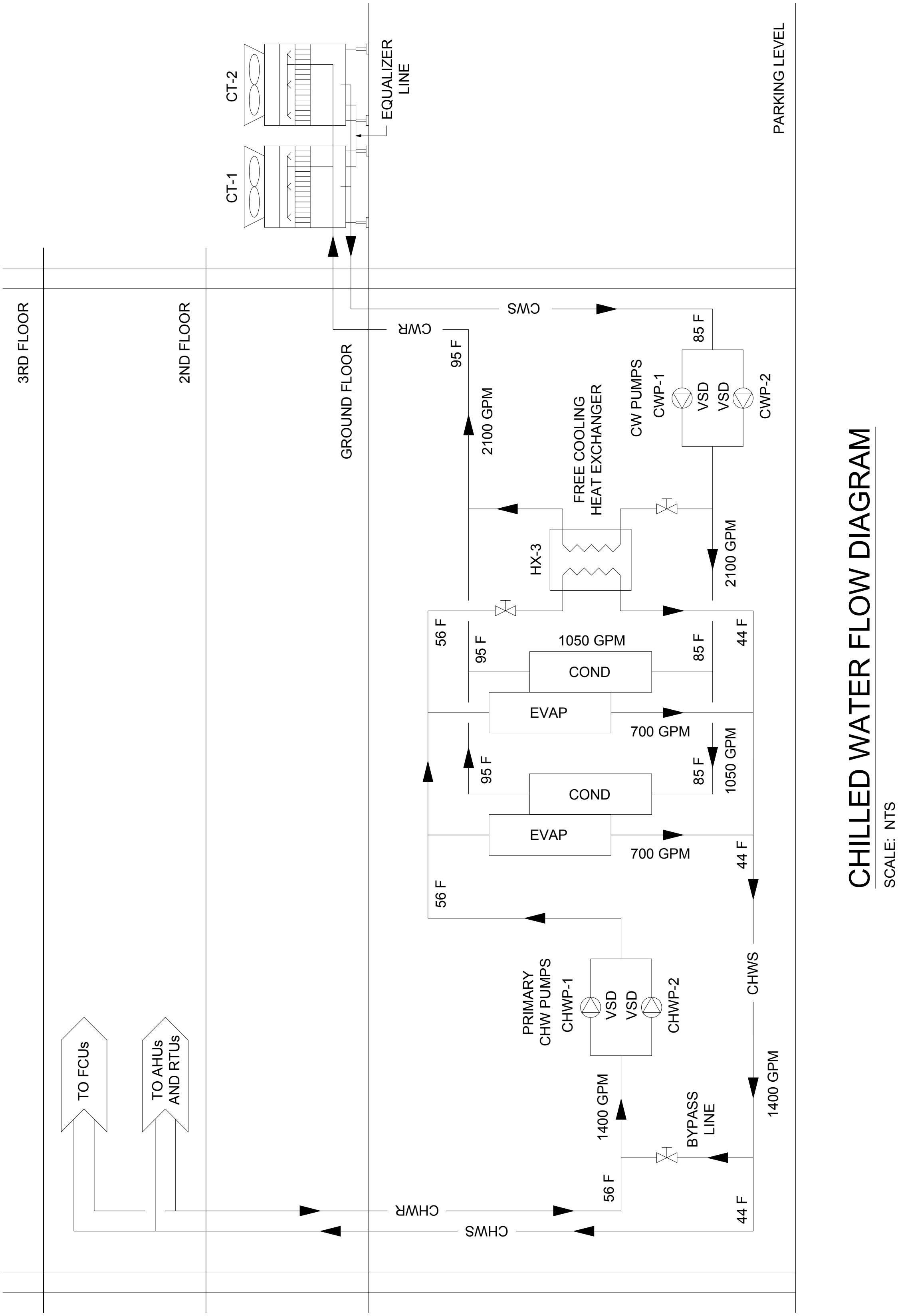

Figure 4 - Chilled Water Flow Diagram

Step 3:

The third step is to choose the characteristics of the condenser water (CW) system. These include the CW temperature, delta-T, CW flow rate, cooling tower fan speed control, and cooling tower efficiency. After making all these decisions, a cooling tower size and selection should be made.

A range of acceptable CW delta-Ts was listed above in Table 12 – Temperature Limits. The most typical case uses a 10 F delta-T, and that is what will be used in this design. The condenser water supply (CWS) temperature (leaving the cooling tower) is the standard 85 F.

The cooling tower range (R) can be found using the following equation: R = CWR – CWS, where CWR is the condenser water return temperature (entering the cooling tower). Since the range is 10 F and the CWS is 85 F, the CWR was found to be 95 F.

With a summer design wet bulb temperature (WBT) of 79 F (not the coincident wet bulb temperature), the cooling tower approach (A) can be found next. Using the appropriate equation: A = CWS – WBT, the approach was calculated to be 6 F.

Next, the flow rate of the CW can be determined. Knowing the CW delta-T of 10F, the CW flow rate is 3.0 gpm/ton, as noted in Table 13 – Relative Flow Rates. This corresponds to a total CW flow rate of 2100 gpm with a 700 ton cooling load.

Several different choices are available for the cooling tower fan control. Possibilities include: one-speed motors, 100%/67% two-speed motor, 100%/50% two-speed motor, pony motors, or variable speed drives. The best option to use with the BWI Hilton is the one with variable speed drives (VSD) because it provides the greatest amount of control for the fan speeds related to the changing cooling tower loads.

The cooling tower efficiency is comprised of two different things – fan type and fill pressure drop. The efficiency is the ratio of CW flow rate to the motor horsepower (gpm/hp) at Cooling Tower Institute (CTI) standard conditions (CWR = 95 F, CWR = 85 F, and WBT = 78 F). For the cooling tower fan type, the choices are either centrifugal fans or propeller fans. The propeller fans use half the energy centrifugal fans, so propeller fans were chosen. For the BWI Hilton project, draw-through type propeller fans will be used. The pressure drop in the fill affects the size and efficiency of the cooling tower. The greater the fill area is, the less pressure drop, the larger the tower size, and the greater the efficiency. It was recommended by “CoolTools” that the tower efficiency be greater than 80 gpm/hp at CTI standard conditions.

Step 4:

The fourth step, as outlined by "CoolTools", is to choose the characteristics of the chillers for the chilled water plant. These include the type of chillers, how many of them, and their sizes. Other chiller selection options are looked at, such as controllers, inlet guide vanes, motor drives, and refrigerant choices.

When selecting the type of chillers to use, choices include air-cooled chillers, water-cooled chillers, and absorption chillers. There are other varieties of each chiller type, as well, like single-effect or double-effect absorption chillers, or screw or reciprocating electric chillers, for example. Another possibility that has benefits in some applications is a hybrid plant. A hybrid plant incorporates different types of chillers in the same cooling plant, which is oftentimes used to offset peak electric loads in an electric chiller by using a natural gas chiller during those times. Based on the total cooling load of the building of 700 tons and comparing capacities of different types of chillers, it was decided that the best option for the BWI Hilton would be centrifugal water-cooled chillers.

When comparing the number and size for the centrifugal chillers, there were several possibilities to choose from. The first option is just using one chiller to meet the full peak load. The second option would be to have two chillers in parallel that each meets half the peak load. A third option would be to have two chillers sized at a 60/40 split of the peak load. However, this option was not as lucrative as the regular 50/50 load split when compared to the chiller load profiles developed in HAP. The 280 load of the 40% chiller (at 280 tons) would only meet the cooling demands of a few hours during the winter months. The 60% chiller (at 420 tons) would have to operate for almost the entire year. In comparison, one of the 50% chillers (at 350 tons) could meet the cooling loads of the building for about four months out of the year. Additional options for the chiller plant arrangement include using three chillers each at one-third of the peak load (at 233 tons each). This was not a feasible solution for the same reasons described for the 60/40 split.

After comparing all these possibilities, it was determined that two centrifugal chillers should operate in parallel and split the load in half at 350 tons each. This way, if one chiller would break down, the second chiller could meet up to half the load of the building. This is much better than if there was just one chiller to meet the entire cooling load: if the single chiller required servicing, there would not be any cooling capacity available for the hotel during those hours. A hotel is not a critical facility like a hospital or data center, but some redundancy is important and logical.

Another benefit to using the 50/50 load split with the two chillers is the typical operating efficiencies. The most efficient operating range for chillers is in the 40-80% capacity range. With a 350 ton chiller, this range is from 140-280 tons. And with both chillers operating together, even at low loads, the minimum cooling load can be met in 5 months out of the year. This provides some flexibility in the way the chillers are cycled on and off, since this method could be about as energy efficient, if not more, than that of just operating one chiller. Further analysis of the chiller loading profiles would be required to determine the optimum loading and unloading sequences for the two chillers. But as was stated previously, it is not imperative that the chillers respond immediately to decreased loads since the two chillers operating together at part load conditions is still very energy efficient.

Another design consideration deals with the speed control of the compressors on the chillers. It is recommended to use VSDs on the chillers to increase the overall energy efficiency. This slightly increases the energy usage at peak loading with the rated kW/ton of the chillers. But in perspective, the chiller plant will only be operating at peak load conditions for a very small percentage (5% or less) of the time during the year. About 95% of the time is spent at part load conditions which operate more efficiently with the use of VSD controllers on both chillers.

There are several other chiller selection options for electric-drive chillers that affect the maintenance costs. A good design choice is to vary the impeller speed by using inlet guide vanes. This is another means to improve the energy efficiency of the chillers. Other methods to reduce maintenance costs include using direct-drive motors in place of gear-drive motors and hermetic centrifugal compressors instead of open-drive centrifugal compressors.

Table 14 - Refrigerant Environmental Impacts

Refrigerant	Type	Global Warming Potential	Ozone Depletion Potential	Heat of Vaporization (Btu/lbm)	Safety Group
R-11	CFC	4000	1	81	A1
R-12	CFC	7100	1	65	A1
R-22	HCFC	1700	0.055	86	A1
R-123	HCFC	93	0.016	66	B1
R-134A	HFC	1300	0	83	A1
R-718	Water	0	0	1070	A1

The final major chiller selection issue deals with the refrigerant choice used in the machines. Chillers are designed to be used with one of several different refrigerants. However, certain refrigerants have more detrimental effects on the environment than others. For example, chlorofluorocarbons (CFCs) have the highest global warming potential and the highest ozone depletion potential. Hydrochlorofluorocarbons (HCFCs) have a lower global warming potential and lower ozone depletion potential than CFCs. Use of newer hydrofluorocarbons (HFCs) have lower global warming potential than CFCs and zero ozone depletion potential. Please see the data above in Table 14 –

Refrigerant Environmental Impacts for a full listing of all the refrigerant types being used recently. For definitions of the various safety groups, please refer to Table 15 – Refrigerant Safety Groups.

Table 15 - Refrigerant Safety Groups

Flammability	Lower Toxicity	Higher Toxicity	Comments
Higher Flammability	A3	B3	LFL </= 0.10 kg/m3 or heat of combustion >/= 19 kJ/kg
Lower Flammability	A2	B2	LFL > 0.10 kg/m3 and heat of combustion < 19 kJ/kg
No Flammability	A1	B1	No LFL in air at 21C and 101 kPa
	No toxicity </= 400 ppm	Evidence of toxicity < 400 ppm	

As is listed below in Table 16 – CFC and HCFC Refrigerant Phase-out, CFCs used as refrigerants have not been produced for ten years. Also, HCFCs like R-22 and R-123 are on their way out of production within the next 25 years. However, there are no restrictions on HFCs like R-134A. These restrictions could play a significant role in deciding on what type of chiller to use. R-134A is the most likely choice if phase-out and restrictions are a major concern. But, R-123 will still be available through the probable life-cycle of the chillers chosen.

Table 16 - CFC and HCFC Refrigerant Phase-out

Refrigerant	Type	Year	Restrictions
R-11	CFC	1996	End of production
R-12	CFC	1996	End of production
R-22	HCFC	2010	End of production and no use in new equipment
		2020	End of production
R-123	HCFC	2015	End of production
		2020	No use in new equipment
		2030	End of production
R-134A	HFC	-	None
R-718	Water	-	None

Step 5:

The fifth step in the chiller plant selection process is to adjust the cooling tower sizing and number of cells, if necessary. According to “CoolTools”, this ultimately depends on the chiller configuration and its effects on the condenser water system.

In the case of the BWI Hilton, the chiller selection was done prior to the actual selection of the cooling towers. Step 3 was done to outline the process of looking at the different aspects of cooling towers, but the actual selection of the cooling towers was done in Step 5. Please refer to Step 3 for a detailed description of this selection process

Step 6:

The sixth step in the "CoolTools" process is to optimize the piping design and pumps for the chilled water and condenser water systems.

In this case, the actual piping design and layout for both the CHW and CW systems was not done, as it was beyond the scope of this thesis report. However, the friction loss through the piping systems was estimated, and the calculations were used to estimate and select the size of the pumps used in both systems. End suction pumps were selected for both the CHW and CW pumps since they are typically good for this application, and since they were used in the original design. The pump head calculations are shown below in Table 17 – CHW Pump Head Calculation and Table 18 – CW Pump Head Calculation.

Table 17 - CHW Pump Head Calculation

Pipe Friction Loss		
Height to Penthouse	143	ft
Guestroom Riser Height	127	ft
Horizontal Distances	300	ft
System Length	1140	ft
Friction Rate	3	ft/100 ft
Multiplier	1.5	
Pipe friction loss	51.3	ft wg
Other Head Losses		
Coil Head Loss	15	ft wg
Control Valve Head Loss	10	ft wg
Evaporator Head	15	ft wg
Heat Exchanger	5	ft wg
Total Other Losses	45	ft wg
Total Pump Head		
Pipe Friction Loss	51.3	ft wg
Other Head Losses	45	ft wg
Subtotal	96.3	ft wg
Safety Factor	15	%
Total Pump Head	**110.7**	**ft wg**

Table 18 - CW Pump Head Calculation

Pipe Friction Loss		
Distance - CH to CT	150	ft
System Length	200	ft
Friction Rate	3	ft/100 ft
Multiplier	1.5	
Pipe friction loss	9.0	ft wg
Other Head Losses		
Cooling Tower Lift	15	ft wg
Heat Exchanger Head	5	ft wg
Control Valve Head Loss	10	ft wg
Condenser Head	20	ft wg
Total Other Losses	50	ft wg
Total Pump Head		
Pipe Friction Loss	9.0	ft wg
Other Head Losses	50	ft wg
Subtotal	59.0	ft wg
Safety Factor	20	%
Total Pump Head	**70.8**	**ft wg**

Other piping design issues deal with the pumping arrangement. This was discussed previously in Step 1 with the CHW distribution arrangement. It is recommended that a reverse return piping system be used instead of a direct return system. This is because the reverse return system will self-balance itself to a certain degree because the lengths of the supply piping to the loads and the return piping back from each load is nearly the same. This is not the case in a direct return system where some loads may have the shortest (or the longest) supply and return water piping.

Step 7:

The seventh step in the chiller plant selection process deals with optimizing the control sequences for the entire central chilled water plant.

For the BWI Hilton project, the controls were not studied in great detail since it is beyond the scope of this thesis. However, some recommendations can be made as to what should be done with the staging of the chillers, pump operations, CHW temperature reset, CW temperature reset, and thermal storage.

With the controls of staging chillers, issues of energy efficiency at part load conditions were discussed previously in Step 4. According to “CoolTools”, it is typically more efficient to run two chillers in parallel at part load than one chiller at full load. It is recommended to not run the VSDs on the chillers at less than 20-35% of the load conditions. Also, there are less complex staging issues with using VSDs because it is not crucial to stage the chillers on and off with precision. This was discussed previously in Step 4.

The pumps used for the CHW and CW systems were described previously in Step 6. It is recommended that these pumps all be equipped with VSDs. A minimum pump flow should also be determined. This is important because pumps do not respond to loads, but rather to flow and head requirements.

Another pumping control issue deals with the arrangement of the pumps in a variable primary flow, as there is in the BWI Hilton project. It is recommended that the two pumps in parallel should be headered together and then piped to each of the chillers. This is a better arrangement than having one pump directly serving each chiller for several reasons. First, a pump is not required to start immediately when a chiller starts. This is evident when only one chiller is in operation. If the second chiller starts, the one headered pump may be able to meet the flow and head demands of both chillers until the second pump can start up. The other benefit of the headered pumps is that they also provide some redundancy. In case one of the pumps goes down for repairs, the other pump can meet half the flow requirements and all the head for the CHW system. Both chillers could be in operation at part loads, as well. This would not be possible in a "one pump per chiller" arrangement.

The control strategies necessary to accomplish CHW temperature reset are not defined in this thesis report because they go beyond the scope of the research and study. In spite of this, the chilled water reset may not be that much of an advantage in the case of the BWI Hilton, anyway. This is because it is not as beneficial for the selected chilled water plant design as with other possible configurations. CHW reset is more beneficial in applications with screw chillers and constant speed pumping, and the BWI Hilton uses centrifugal chillers with variable primary flow pumping.

Similarly to the case above, the condenser water temperature reset is also not defined. However, CW reset is used with the application of water-side free cooling during the winter months. More explanation of this can be found later in the section on "Free Cooling".

The last control strategy deals with thermal energy storage. It could have been possible to size and use a chilled water storage tank with the BWI Hilton. However, this was not considered during the research or design stages of this thesis project. Thermal energy storage could be its own topic of future study to see if it also contributes to the energy efficiency improvements of the building's mechanical systems.

Step 8:

The final step in the "CoolTools" guide is to calculate the life cycle costs of the chiller plant with optimized design and selection of both the chillers and cooling towers.

The selection procedures to select the best possible chillers and cooling towers were previously described in Step 4 and Step 5, respectively, so please refer to those steps for more information.

The life cycle cost analysis was performed in depth in the section "Overall Cost Analysis". Please refer to that section for details on the comparisons of all the chiller plant equipment selection options with respect to first costs and operating costs.

<u>Chillers</u>

After examining all the chiller selection guidelines, as described in Table 16 – Chiller Selection Criteria, several manufacturer representatives were contacted to get actual chiller selections and pricing. All the necessary information for chiller selection was received from Carrier, McQuay, Trane, and York. All the chillers quoted used R-134A, except for Trane, which used R-123. The McQuay quote also included three different chillers to select from. All these options were examined and comparisons were made to determine which chiller best fits the application at the Hilton Hotel at BWI Airport. Both the first costs and energy costs were studied during this process.

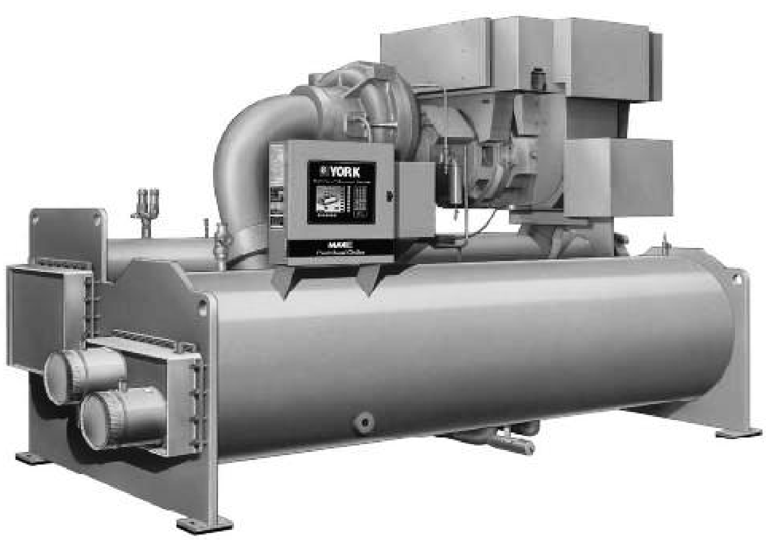

Figure 5 - York Chiller

As was described earlier in Step 2 of the design process, a 12 F delta-T was chosen for this project. However, when price quotes were gathered from manufacturers, one quote was given at a 10 F delta-T. The 10 F delta-T on Chiller Option No 1 used the traditional flow rates associated with the chilled water and condenser water. In the midst of the newer thinking with using lower flows (gpm/ton) with higher delta-Ts, there has been concern over the performance and maintenance issues on the evaporators of chillers. This question was asked to the manufacturer representatives, but they all said the lower flow rates and 12 F delta-Ts will not damage the equipment. A summary of all the chiller selection choices can be seen in Table 19 – Chiller Selection Comparison.

Table 19 - Chiller Selection Comparison

Option No	Manuf	Qty	Refrig	VSD	Capac (tons)	Input Power (kW)	Perf (kW/ton)	Comments
1	Carrier	2	R-134A	Yes	350	231.0	0.661	
2	York	2	R-134A	Yes	350	216.0	0.617	
3	McQuay	2	R-134A	Yes	350	207.6	0.593	
4	McQuay	1	R-134A	IGV	700	403.0	0.576	Inlet Guide Vanes
5	McQuay	1	R-134A	Yes	700	418.2	0.597	
6	Trane	1	R-123	No	350	178.8	0.511	
6	Trane	1	R-123	AFD	350	184.3	0.526	Adaptive Freq Drive
7	Trane	2	R-123	AFD	350	184.3	0.526	Adaptive Freq Drive

Option No	Manuf	CHWR Evap EWT (F)	CHWS Evap LWT (F)	Evap delta-T (F)	Evap Flow Rate (gpm)	CWR Cond LWT (F)	CWS Cond EWT (F)	Cond delta-T (F)	Cond Flow Rate (gpm)
1	Carrier	54.0	44.0	10.0	840	94.5	85.0	9.5	1050
2	York	56.0	54.0	12.0	700	94.3	85.0	9.3	1050
3	McQuay	56.0	44.0	12.0	700	95.0	85.0	10.0	1050
4	McQuay	56.0	44.0	12.0	1400	95.0	85.0	10.0	2100
5	McQuay	56.0	44.0	12.0	1400	95.0	85.0	10.0	2100
6	Trane	56.0	44.0	12.0	696.7	95.0	85.0	10.0	970.7
6	Trane	56.0	44.0	12.0	696.7	95.0	85.0	10.0	976.8
7	Trane	56.0	44.0	12.0	696.7	95.0	85.0	10.0	976.8

After comparing all the first costs and annual operating costs in the life cycle cost analysis, it was decided that Option No 7 with the two Trane Chillers with adaptive frequency drives (same thing as VSD or VFD controls) was the best choice chiller for the BWI Hilton. More information can be found in Appendix A – Chiller Selection with the selected chiller cut sheets from the manufacturers. For all the information regarding the life cycle costs, which incorporate the first costs, energy usage, and annual operating costs, please refer to the section "Overall Cost Analysis".

Cooling Towers

Figure 2 - Marley Cooling Tower

The selection of the cooling towers was based on Marley NC Class cooling towers. The type of unit used is a two-cell, induced-draft, cross-flow, galvanized steel cooling tower. Several choices of cooling towers were looked at for the BWI Hilton project. Two different towers at standard sound ratings (with 1800 rpm fan motors) were examined and compared to two comparable towers that have lower sound ratings (with 1200 rpm fan motors). The energy consumption and acoustics of all four cooling towers were examined prior to the final selection. A comparison of the four cooling towers is shown below in Table 20 – Cooling Tower Selection Comparison.

Table 20 - Cooling Tower Selection Comparison

Cooling Tower Model	No of Cells	Fan				Motor			gpm/hp
		Blades	Length (ft)	Speed (rpm)	Speed (fpm)	Speed (rpm)	Output (BHP)	Air Flow (cfm)	
NC8305FL2	2	8	8	313	7866.5	1200	40	228,800	72.6
NC8306EL2	2	8	10	191	6000.4	1200	30	244,000	99.4
NC8305F2	2	6	8	370	9299.1	1800	40	232100	73.8
NC8307E2	2	6	10	241	7571.2	1800	30	242300	103

Cooling Tower Model	Water Flow (gpm)	HWT (F)	Range (F)	CWT (F)	Approach (F)	WBT (F)	RH (%)	Total Heat Rejection (Btu/hr)	Price
NC8305FL2	2100	95	10	85	6	79	50	10,463,000	$80,600
NC8306EL2	2100	95	10	85	6	79	50	10,463,000	$92,300
NC8305F2	2100	95	10	85	6	79	50	10,463,000	$78,000
NC8307E2	2100	95	10	85	6	79	50	10,463,000	$94,900

After comparing all the first costs and annual operating costs in the life cycle cost analysis, it was decided that Option No 3 with the NC8305F2 was the best choice cooling tower for the BWI Hilton. More information can be found in Appendix B – Cooling Tower Selection with the selected chiller cut sheets from the manufacturer. Please also refer to the section "Overall Cost Analysis" for additional life cycle cost information.

On the BWI Hilton project, the two-cell cooling towers will be located at the same spot as the original cooling towers just north of the building outside of the kitchen. Please see the section "Cooling Tower Acoustical Analysis" for more information on the acoustics study done on the four cooling towers.

Pumps

The pump energy consumed in the Hilton Hotel at BWI Airport is a major contributing factor to the total building energy usage. Therefore, the pump sizing and selection is an important issue with the chilled water plant design. Pumps are needed on the project for both the chilled water and condenser water systems. However, since the exact piping layouts, sizing, lengths, and fittings were not known, an estimate of the total pump head required for each of the pumps was made. These estimates are shown previously in Step 6 of the "Chilled Water Plant Design" section.

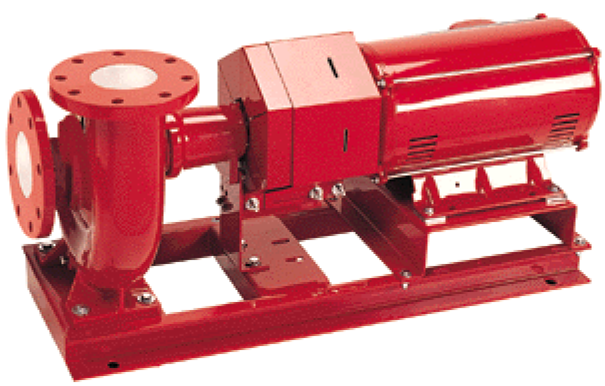

Figure 7 - Bell & Gossett Pump

For both systems, two Bell & Gossett Series 1510 end suction pumps piped in parallel will meet the calculated flow and head requirements. All four pumps were selected with variable frequency drives (VFDs) with manual bypasses for improved partial flow performance. A summary of the pumps selected for the condenser water and chilled water systems can be seen below in Table 21 – Pump Selections. More information on the pump curves and cut sheets can be found in Appendix C – Pump Selection.

Table 21 - Pump Selections

Chilled Water Pumps			Condenser Water Pumps		
No of Pumps	2	pumps	No of Pumps	2	pumps
Pump Flow	700	gpm ea	Pump Flow	1050	gpm ea
Pump Head	110.7	ft wg ea	Pump Head	70.8	ft wg ea
Pump Speed	1750	rpm	Pump Speed	1750	rpm
Manufacturer	Bell&Gossett		Manufacturer	Bell&Gossett	
Model	1510 Base mounted centrifugal		Model	1510 Base mounted centrifugal	
Size	5G		Size	5E	
Impeller Dia	10.875	in	Impeller Dia	9.875	in
Motor Size	40	HP	Motor Size	25	HP
BHP	24.91	BHP	BHP	23.16	BHP
Efficiency	79.19	%	Efficiency	82.14	%

<u>Water-Side Free Cooling</u>

One of the energy saving techniques used in the Hilton Hotel at BWI Airport project involved water-side free cooling. Free cooling can be used when the outdoor wet bulb temperature is below a certain temperature, and the cooling tower can produce lower temperature condenser water. The chilled water bypasses around the chillers and goes through a plate-and-frame heat exchanger that is piped in parallel to the chillers. Here the CHWR water is cooled down by giving up its heat to the CWS water instead of passing through the evaporator of the chiller. An example of a plate-and-frame heat exchanger is shown below in Figure 8 – Bell & Gossett Heat Exchanger.

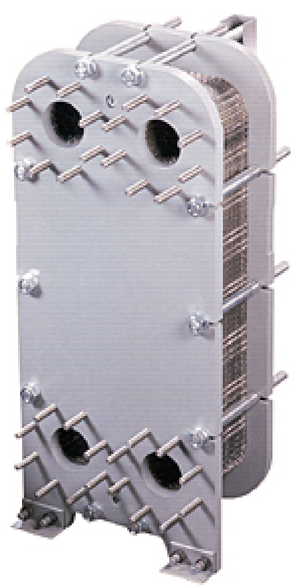

Figure 8 - Bell & Gossett Heat Exchanger

There can be significant energy savings realized by installing the heat exchanger and piping to utilize water-side free cooling. A simple comparison was done in HAP to determine how much energy was saved by using free cooling with (2) York MaxE Chillers (Option 2). The results are summarized below in Table 22 – Free Cooling Energy and Cost Savings.

Table 22 - Free Cooling Energy and Cost Savings

Component	With Free Cooling Site Energy (kBtu)	No Free Cooling Site Energy (kBtu)	Savings with Free Cooling (kBtu)	% Savings
Air System Fans	3,423,614	3,423,614	0	0.00%
Cooling	3,452,357	4,255,716	803,359	18.88%
Heating	17,442,574	17,442,574	0	0.00%
Pumps	1,605,084	1,604,931	-153	-0.01%
Cooling Towers	759,293	686,163	-73,130	-10.66%
HVAC Sub-Total	**26,682,921**	**27,412,998**	**730,077**	**2.66%**

Component	With Free Cooling Annual Cost	No Free Cooling Annual Cost	Savings with Free Cooling	% Savings
Air System Fans	$70,297	$70,209	-$88	-0.13%
Cooling	$79,914	$93,363	$13,449	14.41%
Heating	$36,124	$36,121	-$3	-0.01%
Pumps	$33,274	$33,232	-$42	-0.13%
Cooling Towers	$17,486	$16,247	-$1,239	-7.63%
HVAC Sub-Total	**$237,094**	**$249,172**	**$12,078**	**4.85%**

For the BWI Hilton project, it was determined that when the WBT is 30 F or lower, water-side free cooling can be used to cool the chilled water to its specified setpoint of 44 F. After talking with a Marley sales representative, it was determined that the minimum CW temperature can be 40 F. This is because air passing through the cooling tower fill will be stratified and it could potentially produce CW that is near freezing when the average CW temperature gets below 40 F. In order to get 42 F CWS entering the heat exchanger, a 6 F range (or delta-T) on the cold side of the heat exchanger is used to get a wet bulb temperature (WBT) of about 30 F. Please see the graph below in Figure 9 – CWS vs WBT (the lowest line was used since it represents the 6 F range).

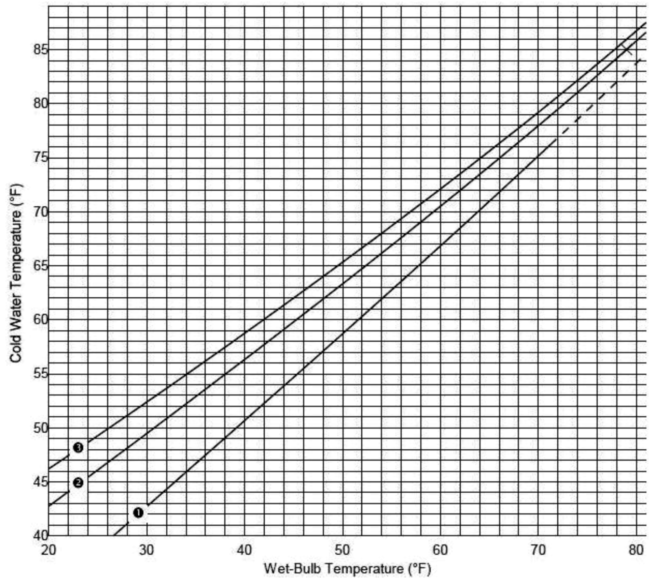

Figure 9 - CWS vs WBT

If a plate-and-frame heat exchanger with a 2 F approach is selected and the CWS temperature entering the cold side of the heat exchanger is 42 F, then the CHWS temperature leaving the heat exchanger on the hot side is 44 F. This works because 44 F is the desired CHWS temperature setpoint. The number of hours when the outdoor WBT is less than or equal to 30 F can be seen in Table 23 – Free Cooling at WBT < 30 F. This occurs for a total of 1357 hours each year, which is about 15.5% of the year.

Table 23 - Free Cooling at WBT < 30 F

No of hr WB </= 30F	
January	432
February	312
March	184
April	45
May	3
June	0
July	0
August	0
September	0
October	0
November	75
December	306
Total	**1357**
% of yr	15.49%

To size the plate-and-frame heat exchanger, the total load on it needs to be calculated. Using hourly simulation results of the chiller load profiles from HAP, the number of hours when free cooling occurs was determined. This can be seen above in Table 23 – Free Cooling at WBT < 30 F above. Also, the maximum free cooling load that occurs during those hours is shown below in Table 24 – Max Free Cooling Load. Using these values, the maximum load is 3523 MBH, which is about 294 tons. Using a safety factor of 10% and rounding to the next even size, the total load on the heat exchanger was found to be 325 tons.

Table 24 - Max Free Cooling Load

January	3406.1
February	3459.7
March	3523.0
April	3462.0
May	3476.2
June	0.0
July	0.0
August	0.0
September	0.0
October	3463.2
November	3449.8
December	3435.8
Max (MBH)	3523.0
Max (tons)	293.58
Safety Factor	10%
Total Load (tons)	322.94
HTX Size (tons)	**325**

Using this total cooling of 325 tons (Q=3,900,000 Btu/hr), the water temperature conditions and flow rates could be calculated. On the cold side of the heat exchanger, the CW delta-T is known to be 6 F, since that is the selected range for the cooling tower. Using the equation Q = 500 x gpm x delta-T, the CW flow rate was calculated. The maximum CW flow rate is 1300 gpm during free cooling operations. This should not be a problem for the cooling towers or the CW pumps since they all have VSDs to modulate their speeds.

The same calculation (with the same Q) can be used on the hot side of the heat exchanger. After talking with a sales representative from Bell & Gossett, it was determined the the CHW delta-T = 10 F during winter operation. With the CHWS = 44 F, the CHWR must be 56 F. Therefore, the maximum flow rate of the CHW is 780 gpm, when the same Q equation is used as above.

For a schematic of the heat exchanger with the free cooling design temperatures and flow rates, please see Figure 10 – Free Cooling Heat Exchanger below.

For more information on the heat exchanger used and the cut sheets, please see Appendix D – Heat Exchanger Selection.

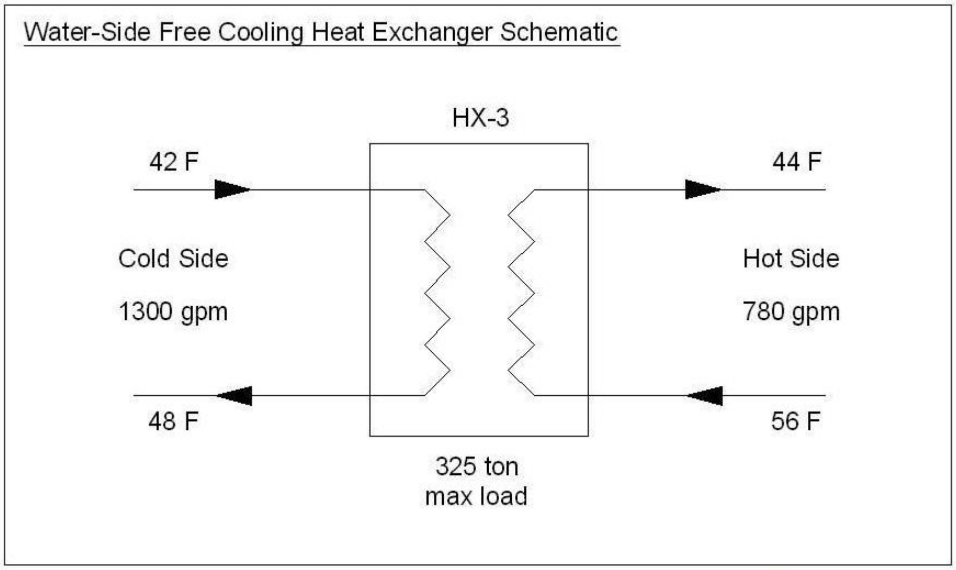

Figure 10 - Free Cooling Heat Exchanger

Please note that if CHWS temperature reset was used, which it was not for this project, then higher WBTs could be used to achieve higher CWS temperatures. However, additional calculations would be necessary to determine if all the chilled water coils in the building could handle a higher temperature CHWS. The total building cooling loads would also have to be compared to the maximum cooling capacity of the heat exchanger (325 tons) to see if this is feasible. Not enough time was allotted to all these supplementary equations, and the CHWS temperature reset option was not studiesd. Despite the more complicated controls, the main benefit to the CHWS temperature reset is that free cooling could be used for a greater percentage of hours during the year. This can be seen from the data shown in Table 25 – Free Cooling WBTs.

Table 25 - Free Cooling WBTs

No of hr WB </= 35F		No of hr WB </= 40F	
January	582	January	679
February	414	February	524
March	351	March	501
April	99	April	208
May	26	May	62
June	0	June	0
July	0	July	0
August	0	August	0
September	0	September	0
October	20	October	85
November	192	November	406
December	514	December	648
Total	**2198**	**Total**	**3113**
% of yr	25.09%	% of yr	35.54%

Impact on Air-Side Equipment

The addition of the chilled water central plant in the Hilton Hotel at BWI Airport did not have many major effects on the air-side equipment of the existing mechanical systems design for the building. The design conditions for all four air handling units (AHUs) and the six rooftop units (RTUs) remained the same as in the original design. The loads from all the spaces in the hotel also did not change. However, the new calculated HAP outputs were used to determine the size and capacities of all the AHUs and RTUs. The other significant impact on the equipment design and selection equipment was the changeover to operating with the new chilled water system instead of the original condenser water system, so the new AHUs and RTUs were selected to have chilled water coils.

Air Handling Units

All the data for the AHUs is based on selection of Carrier 39MN air handling units. All four units were selected to have a mixing box, hot water heating coil, chilled water cooling coil, and draw-through supply fan. A sample cross-section view of a Carrier AHU can be seen below in Figure 11 – Carrier AHU.

Figure 11 - Carrier AHU

The original unit price for each individual air handling unit was not known, so a cost analysis between the existing design and the new AHUs was not calculated. Please refer to the original equipment cost estimate in Table 53 – Original Mechanical Equipment Costs in the “Overall Cost Analysis” section. Please see Appendix E – Air Handling Unit and Rooftop Unit Selection for information and cut sheets for the new AHUs with chilled water cooling coils.

Rooftop Units

All the data for the RTUs is based on the selection of Carrier 39MW rooftop units. Five of the six units were selected to have a mixing box, hot water preheat coil, chilled water cooling coil, hot water reheat coil, and draw-through supply fan. The only unit that did not change is RTU-11, which serves the elevator machine room in the penthouse. This unit remained the same as in the original design since it had such a small load and a minimal impact on the operating costs of the rest of the building. Therefore, RTU-11 was not included in any of the cost calculations or energy usage comparisons between the original design and the new design.

The original unit price for each individual rooftop unit was not known, so a cost analysis between the existing design and the new RTUs was not calculated. Please refer to the original equipment cost estimate in Table 53 – Original Mechanical Equipment Costs in the “Overall Cost Analysis” section. Please see Appendix E – Air Handling Unit and Rooftop Unit Selection for more information and the cut sheets for the new RTUs with chilled water cooling coils and hot water preheat and reheat coils.

Guest Room Indoor Air Quality

In the original design of the Hilton Hotel at BWI Airport, two air conditioning units located in the penthouse provided 60 cfm of ventilation air to each of the guest rooms. However, during the value engineering process for the project, these two air conditioning units in the penthouse were eliminated. After eliminating the primary source of outdoor air, a variance was granted for the BWI Hilton project to use an alternative method of ventilation. This method takes the supply air that the RTUs provide for pressurization and ventilation to the guest room tower corridors and has it drawn into each of the guest rooms through the undercut in the doors. The corridor air is drawn into the guest rooms either by the continuously operating mechanical exhaust fans through all the bathrooms or by the use of operable windows in each guest room.

However, this method does not seem to be a very effective method to adequately ventilate the guest rooms. Unfortunately, there is no way to accurately measure, predict, or record the levels of carbon dioxide or other airborne contaminants in each guest room. But, the indoor air quality (IAQ) of the guest rooms is one of the biggest design issues of the building. There is major concern with the ventilation of the 279 guest rooms of the BWI Hilton. An improved system is needed to provide fresh air to the rooms, as well as reducing the concentration of odors and other contaminants that result from poor ventilation techniques.

Therefore, to increase the IAQ of the spaces, it was decided to install new outdoor air units to provide ventilation for all the guest rooms. Two new units will be located in the penthouse on top of the guest room tower, and the air will be ducted down through vertical risers, which was the same way it was done in the original design before it was valued engineered out of the project. This will be accomplished through the use of two Dedicated Outdoor Air System (DOAS) units with energy recovery wheels. In this case for the BWI Hilton, the term "DOAS" is used to simply refer to a unit that continuously provides conditioned 100% outside air.

The DOAS units will provide ventilation air into the guest rooms through the fan coil units. The DOAS units supply 60 cfm of pre-conditioned ventilation air at continuous operation. The fan coil units will be used to meet the space cooling and heating loads in all 279 guest rooms of the BWI Hilton. The FCU recirculates a certain amount of the room air and mixes the two air streams together. After the mixing, the air stream passes over the coil section where it is either cooled or heated, depending on what the room calls for. The draw-through fan then supplies the guest room with the conditioned air.

During times when no additional cooling loads are called for by the FCU, the 60 cfm of ventilation air will still be supplied to the space. This has several benefits which are described below.

Even though ASHRAE Standard 62.1-2004 only requires a minimum ventilation rate of 5 cfm/person and 0.06 cfm/sf in hotel guest rooms, the DOAS units were sized and selected to provide 60 cfm of conditioned outside air to each guest room. The 60 cfm of outdoor air was chosen for several reasons. First, 60 cfm was the quantity used in the original design prior to the value engineering, and this way a comparison can be made between the two scenarios. Next, it was known that all guest rooms have 50 cfm of continuous exhaust from the bathrooms. Pressurization of the guest rooms is important to prevent air flow in the wrong direction, so the guest room should always have positive pressure compared to the spaces around it.

There are also some benefits to having positive pressure in the guest rooms. This will prevent unwanted infiltration from the moist summer outside air, which has the potential to cause mold in the walls. Compared to the bathroom, the guest room will always be provided with 10 cfm more than is exhausted. This will help to prevent air flow and unwanted odors from quickly moving from the bathroom into the guest room. Also, the guest rooms will have slightly higher pressure as compared to the corridors to prevent any unwanted infiltration of odors from the more public areas.

Continuous supply of outdoor air to all the guest rooms and proper pressurization are the main reasons to use DOAS units to maintain improved indoor air quality of the guest rooms. In comparison, this method is a significant improvement over the original method of ventilation.

Dedicated Outdoor Air System

All the data for the DOAS units is based on selection of Semco Pinnacle units. Both units were selected to have a total energy enthalpy wheel, chilled water cooling coil, passive dehumidification wheel, supply fan, and exhaust fan. Please see Appendix F – Dedicated Outdoor Air System Unit Selection for cut sheets and more information on the selection of the two new DOAS units.

The two DOAS units both use two energy recovery wheels. The first wheel does total energy recovery with an exchange between the outside and exhaust air streams. This enthalpy wheel has a 3A (Angstrom) molecular sieve with a desiccant coating that prohibits the adsorption of any particles larger than a water molecule, which is 2.8A. The second wheel is a passive dehumidification energy recovery wheel that is used to dehumidify the outdoor air after it passes through the cooling coil. This wheel has adsorbent desiccant material that has optimum dehumidification performance. Also, both wheels operate with variable speed drives to modulate to the appropriate speed at part load based on the indoor and outdoor air conditions.

Figure 12 - Semco DOAS Unit

A schematic of one of the DOAS units is shown below in Figure 13 – DOAS Unit Schematic. The one shown actually shows the typical operation of the unit during peak sensible cooling load.

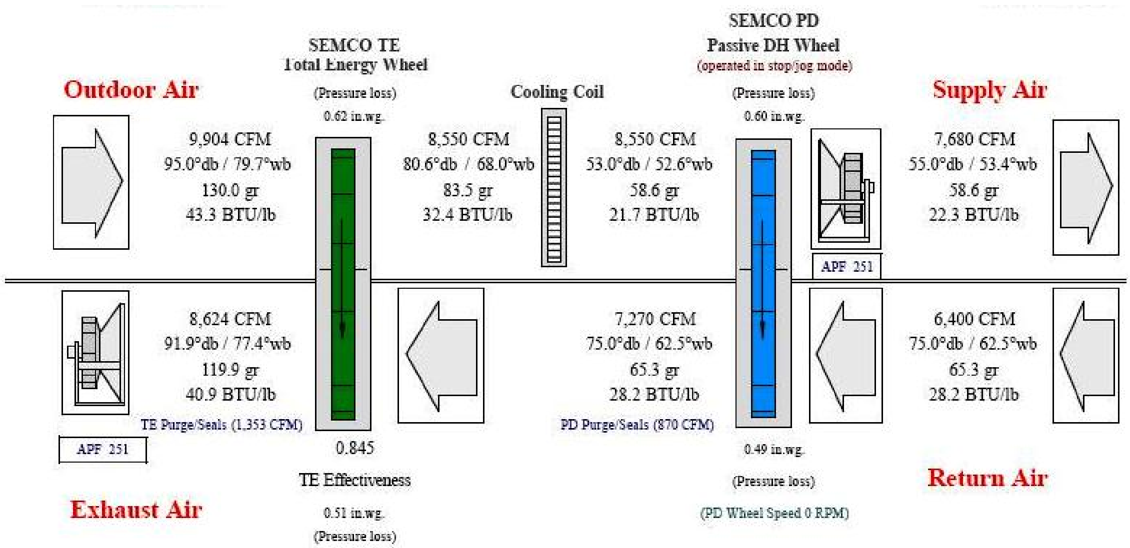

Figure 13 - DOAS Unit Schematic

The airflow sizing of both of the units was done with the following procedure for the calculation of the ventilation air and the exhaust air. The ventilation sizing is shown in Table 26 – DOAS Ventilation Air Quantity Sizing. The exhaust sizing is shown in Table 27 – DOAS Exhaust Air Quantity Sizing.

Table 26 - DOAS Ventilation Air Quantity Sizing

Unit	Floor(s)	No of Floors	No of Rooms	Units/Room	Airflow/Room (cfm)	Total Airflow (cfm)
DOAS-1	3	1	14	1	60	840
	4 - 10	7	14	1	60	5880
	11	1	10	1	60	600
	11	1	3	2	60	360
					Total	**7680**
DOAS-2	3	1	18	1	60	1080
	4 - 10	7	18	1	60	7560
	11	1	10	1	60	600
	11	1	2	3	60	360
					Total	**9600**

Table 27 - DOAS Exhaust Air Quantity Sizing

Unit	Floor(s)	No of Floors	No of Rooms	Units/Room	Airflow/Room (cfm)	Total Airflow (cfm)
DOAS-1	3	1	14	1	50	700
	4 - 10	7	14	1	50	4900
	11	1	10	1	50	500
	11	1	3	2	50	300
					Total	**6400**
DOAS-2	3	1	18	1	50	900
	4 - 10	7	18	1	50	6300
	11	1	10	1	50	500
	11	1	2	3	50	300
					Total	**8000**

Fan Coil Units

Four-pipe fan coil units (FCUs) were selected to be used in each guest room. These FCUs are vertical stack units that are made specifically for applications where multiple units will be lined up and stacked above each other, like in hotels. This way, minimal piping is required between units on adjacent floors. The connections for all four pipes are made at each unit the entire way up the riser.

A four-pipe FCU system was chosen to replace the existing water source heat pump (WSHP) system in the original design. This was done for two reasons. The primary reason was because the overall system could operate more efficiently using a central cooling plant as compared to individual units that require refrigerant loops and compressors to provide space cooling. Another benefit of using the FCU system was the ability to have units that do not contain a compressor. This reduces both the noise levels of the units as well as the maintenance needs.

The four-pipe was also chosen over a two-pipe FCU system because of the increased comfort levels and flexibility that it provides, especially in the mid-seasons. Some rooms call for heating while other rooms call for cooling oftentimes during the spring and fall. With a two-pipe system, there is limited flexibility and all the rooms must either use all heating or all cooling. With the four-pipe system, both the boiler and the chiller can be running at the same time during these swing periods to provide room-by-room options for either heating or cooling. The four-pipe FCU system has much simpler controls than a two-pipe FCU system because of the changeover between hot and chilled water.

A two-pipe system with electric reheat was also considered as an alternative. However, it was presumed that the four-pipe system could operate more efficiently than a half-electric system, provided the natural gas fuel rates do not escalate greatly in the future, it will continue to be more economical to use natural gas boilers for heating in place of electric reheat coils.

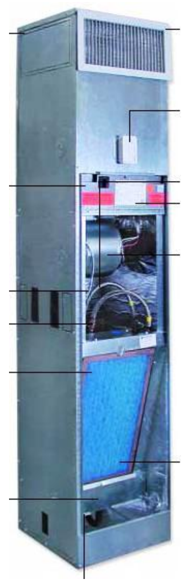

Figure 14 – Enviro-Tec Fan Coil Unit

Using HAP, the block loads for a typical guest room were calculated, and they are shown in Table 28 – Guest Room Block Loads. The main loads on the spaces were from the solar loads, lighting, electric equipment, and the people. These broken loads are shown for two sample rooms, one from each FCU. They are shown below in Table 29 – FCU-1 Sample Guest Room Loads and Table 30 – FCU-2 Sample Guest Room Loads. There were some slight variations between some of the rooms, but all the loads of the rooms will be met easily by either of the possible units shown above in Table 31 – FCU Comparison.

Table 28 - Guest Room Block Loads

No	Cooling			Heating	
	Sensible (Btu/hr)	Latent (Btu/hr)	Total (Btu/hr)	Sensible (Btu/hr)	Latent (Btu/hr)
FCU-1	7094	738	7832	938	0
FCU-2	7873	738	8611	938	0

Table 29 - FCU-1 Sample Guest Room Loads

Zone 3	DESIGN COOLING			DESIGN HEATING		
	OCCUPIED T-STAT 74.0 °F			OCCUPIED T-STAT 70.0 °F		
		Sensible	Latent		Sensible	Latent
ZONE LOADS	Details	(BTU/hr)	(BTU/hr)	Details	(BTU/hr)	(BTU/hr)
Window & Skylight Solar Loads	49 ft²	544	-	49 ft²	-	-
Wall Transmission	35 ft²	13	-	35 ft²	97	-
Window Transmission	49 ft²	-7	-	49 ft²	841	-
Overhead Lighting	314 W	1005	-	0	0	-
Electric Equipment	1476 W	4781	-	0	0	-
People	4	757	738	0	0	0
>> Total Zone Loads	-	7094	738	-	938	0

Table 30 - FCU-2 Sample Guest Room Loads

Zone 15	DESIGN COOLING			DESIGN HEATING		
	OCCUPIED T-STAT 74.0 °F			OCCUPIED T-STAT 70.0 °F		
		Sensible	Latent		Sensible	Latent
ZONE LOADS	Details	(BTU/hr)	(BTU/hr)	Details	(BTU/hr)	(BTU/hr)
Window & Skylight Solar Loads	49 ft²	1669	-	49 ft²	-	-
Wall Transmission	35 ft²	19	-	35 ft²	97	-
Window Transmission	49 ft²	-86	-	49 ft²	841	-
Overhead Lighting	314 W	964	-	0	0	-
Electric Equipment	1476 W	4626	-	0	0	-
People	4	681	738	0	0	0
>> Total Zone Loads	-	7873	738	-	938	0

As a point of reference, FCU-1 units are located in all the even-numbered guest rooms, which are the ones that face the north. The FCU-2 units are located in all the odd-numbered guest rooms, which are the ones that face the south.

Table 31 - FCU Comparison

Option No	Tag	Manuf	Model	Dimensions			Airflow (cfm)
				Length (in)	Width (in)	Height (in)	
1	FCU-1	Carrier	42SGA03	17	17	88	330
1	FCU-2	Carrier	42SGA04	17	17	88	400
2	FCU-1	Enviro-Tec	VHC04	18	23.38	88	358
2	FCU-2	Enviro-Tec	VHC04	19	23.38	89	454

Option No	Tag	Total Clg Capac (Btu/hr)	Sens Clg Capac (Btu/hr)	Clg LAT (F)	CHW Flow (gpm)	Sens Htg Capac (Btu/hr)	Htg LAT (F)	HW Flow (gpm)
1	FCU-1	12,633	8825	55.5	2.1	19,597	114.3	2.0
1	FCU-2	13,877	9924	57.3	2.3	21,020	108.1	2.1
2	FCU-1	9,992	7688	55.2	1.66	20,630	123.2	2.12
2	FCU-2	11,854	9263	56.2	1.97	2,443	119.7	2.5

After the basic sizing of the fan coils was complete, a comparison between two manufacturers was done to determine which FCU to use in all the guest rooms, and this can be seen above in Table 31 – FCU Comparison.

After looking at the manufacturer's data for the two fan coil units, the necessary information was put into HAP. Comparing the results of simulations for both options showed very little difference in their operating costs. Option 2 had an annual operating cost that was only $5 more than Option 1. Therefore, the process used to select which units to use was simply done by only comparing their first costs, not their life cycle costs.

By simple observation, it can be seen that the Carrier FCUs cost about $500 less per unit than the Enviro-Tec FCUs. So the Carrier 42SGA units will be used in all the guest rooms in the BWI Hilton. Please see the detail drawings included in Appendix G – Fan Coil Unit Selection.

After the selection of the Carrier FCUs was made, a calculation was done to one of the guest rooms on the north-side of the hotel to ensure that the selected coil size was sufficient to meet the space ventilation and cooling loads. With the DOAS unit providing DBT = 68.1 F and WBT = 56.7 F, the cooling loads required by the fan coil units will be decreased. This is because they were selected based on entering air conditions of DBT = 80.0 F and WBT = 67.0 F. Since 330 cfm is the actual airflow provided by the fan, and there is 60 cfm of ventilation air supplied to the FCU, only 270 cfm of recirculation air is needed. After mixing, the air has a DBT = 75.5 F, and the unit leaving air temperature is DBT = 55.5 F, which gives a delta-T = 20 F. Using the equation Q = 1.08 x cfm x delta-T, the required load of the cooling coil can be calculated. With the above conditions, the required cooling load of the FCU is Q = 7128 Btu/hr. As was expected, this is less than the 12,633 Btu/hr capacity of the cooling coils in the fan coil unit. Therefore, the FCU meets the required load, and it is oversized enough to meet any atypical loads in all the other guest rooms.

Another step taken after the Carrier FCUs were selected was to verify that the units will fit into the area used previously for the water-source heat pumps in the original design. Several items were looked at, and they are described next.

The overall dimensions of each FCU were listed above in Table 28 – FCU Comparison. For the Carrier units, the length and width are both 17 inches. When the available area was measured on the guest room floor plans, it was found to be 23 in x 25 in. The selected units have enough space in that area and the shafts will not have to be resized. This can be seen below in Figure 15 – Typical Maximum Dimensions.

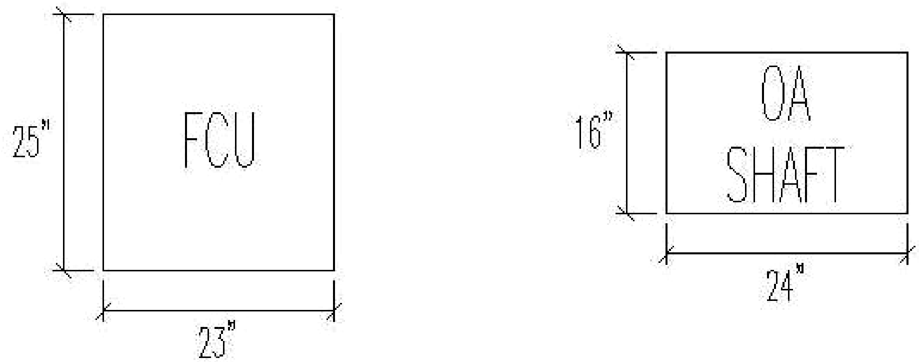

Figure 15 - Typical Maximum Dimensions

The guest rooms all have FCUs adjacent to the neighboring FCU, and they are also beside the outdoor air shaft coming from the DOAS units in the penthouse. This is important because the OA shaft must be tapped at each floor for the connections to the FCUs. The five pipes at each FCU are also shown in the detail. A typical detail of this arrangement is shown in Figure 16 – Typical FCU and OA Detail.

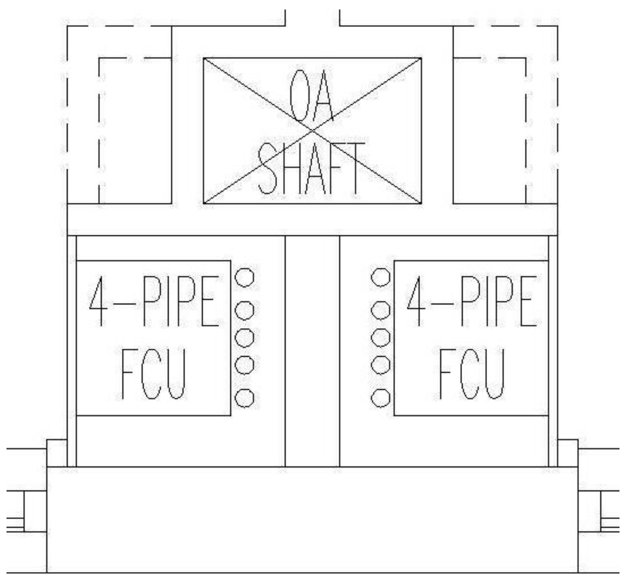

Figure 16 - Typical FCU and OA Detail