6.0 Mechanical Redesign – Depth

In an attempt to reduce energy consumption costs for the Struamann USA facility, several mechanical system alternatives will be compared. On the air-side of the mechanical system a dedicated outdoor air system (DOAS) with a parallel radiant cooling panel system will be compared to a variable air volume (VAV) system. Two different chiller types will be explored on the waterside, electric centrifugal and direct fire absorption. Two different piping arrangements will also be explored for taking advantage of free cooling, parallel and series.

6.1 Air Systems

The airside analysis of the Straumann facility will compare a common VAV system with a combination DOAS and radiant cooling panel system. VAV systems are probably the most popular type air system installed in the United States. While they have become very popular in buildings, there are other types of systems that can also be explored. When comparing a VAV and DOAS systems, the are advantages to implementing both systems.

A VAV system, as seen in Figure 6.1-1, is capable of providing both ventilation air and thermal cooling all from the same air system. A DOAS system, shown in Figure 6.1-2, typically provides ventilation and latent cooling from a smaller air system and must be coupled with a separate parallel system, in this case radiant panels, in order sensibly cool a building. DOAS air handling units are smaller than those required by a VAV system since DOAS units are usually only supplying air to meet minimum ventilation requirements. Often the DOAS unit will supply slightly more air than required by minimum ventilation standards in order to provide latent cooling for spaces. This prevents condensation from becoming a problem with any parallel systems like radiant panels.

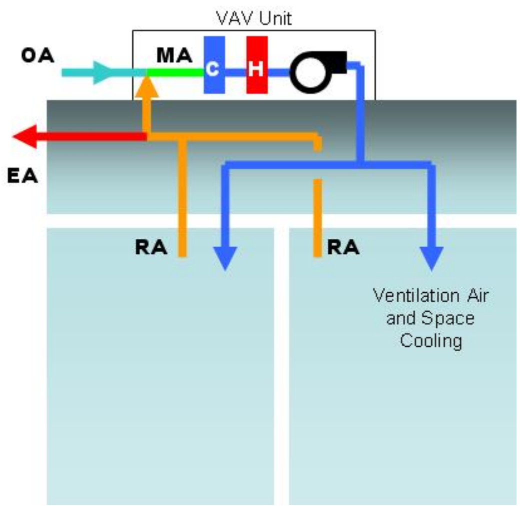

Figure 6.1-1: VAV System Schematic

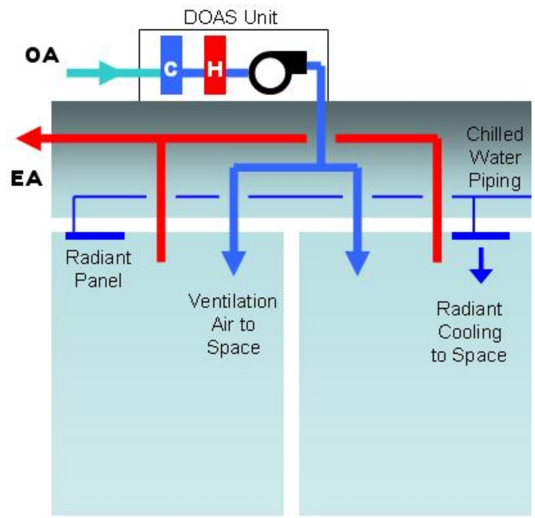

Figure 6.1-2: DOAS System Schematic

Several advantages result in a DOAS system due to the reduced air handling unit size. Since less air is required, smaller fans are used and annual fan energy, which can be a major portion of the annual mechanical operating costs, is reduced. For the Straumann USA facility, fan energy accounts for nearly 30% of the annual electric costs for the mechanical systems. A reduction in fan energy has the potential to result in significant annual operating costs. Another benefit is that a DOAS unit typically supplies less ventilation air than required in a VAV system. This results in a lower energy requirements to condition the ventilation air.

When using a parallel radiant cooling panel system with DOAS, it will result in higher pumping costs than associated with VAV systems. In order to sensibly cool a space, chilled water is pumped to panels above the ceiling where it radiantly cools the occupied spaces. Even though increased pumping energy costs are seen, the reduction in fan energy costs are typically higher, which still results in the DOAS system reducing annual energy costs.

While it may seem that DOAS systems will save yearly energy, it is important to compare not just yearly energy costs, but the first cost of the systems as well. In order to determine the best system for Straumann USA, an energy analysis and first cost comparison for major equipment will both be taken into consideration before making a recommendation.

Carrier's Hourly Analysis Program will be used to calculate the annual energy costs associated with both a VAV and a DOAS system for Straumann USA. Table 6.1-1 lists the design conditions that will be applied both systems.

Load Analysis Assumptions	
OA Ventilation Rates	ASHRAE Standard 62.1-2004
Lighting Loads	
Office	1.3 W/ft2
Manufacturing	2.2 W/ft2
Equipment Loads	
Office	3.0 W/ft2
Manufacturing	38W/ft2
Design Conditions	ASHRAE Fundamentals 2005 (0.4%)
Summer	
Dry Bulb	90.8
Mean Coincident Wet Bulb	73.1
Winter	
Dry Bulb	7.7

Table 6.1-1: Design Assumptions

In order to perform the analysis for the DOAS system new zones must be selected for the DOAS rooftop units. The new DOAS zones are displayed in Figure 6.1-3 and Table 6.1-2 gives a brief description of each. Similar figures and descriptions for the VAV system is found in section 4.7 Existing Mechanical Conditions. The DOAS rooftop units are designed around the Carrier Centurion packaged DX rooftops units, but any equivalent DX rooftop unit could be used. The radiant panels are designed around the Barcol-Air REDEC-CB radiant panel which has a cooling capacity of up to 54 Btu/ft^2. An initial estimate of loads and sensible cooling capabilities of the radiant panels determined a DOAS and radiant panel system would not work in the manufacturing area. Therefore, a VAV system will continue to be utilized in this area being served by RTU-5,6,7,8. The DOAS system considered will be a combination VAV system for the manufacturing area and a DOAS system for the remainder of the facility. For the design, the dew point is at 55°F so the mean radiant temperature based on the sterling design guide will be 56.5°F. The chilled water supply and return temperatures to the radiant panels will be designed using 54°F/59°F.

DOAS Rooftop Unit Summary			
	Max CFM	Square Feet Served	Areas Served
RTU-1	4,273	41,993	First floor manufacturing support areas
RTU-2	3,328	38,549	First floor dental operatory and mezzanine office areas
RTU-3	1,052	4,885	First floor auditorium
RTU-4	3,089	23,361	First floor office and lobby areas
RTU-5	33,000	5,850	Manufacturing area
RTU-6	33,000	5,850	Manufacturing area
RTU-7	33,000	5,850	Manufacturing area
RTU-8	33,000	5,850	Manufacturing area

Table 6.1-2: Spaces Served by Each DOAS Rooftop Air Handling Unit

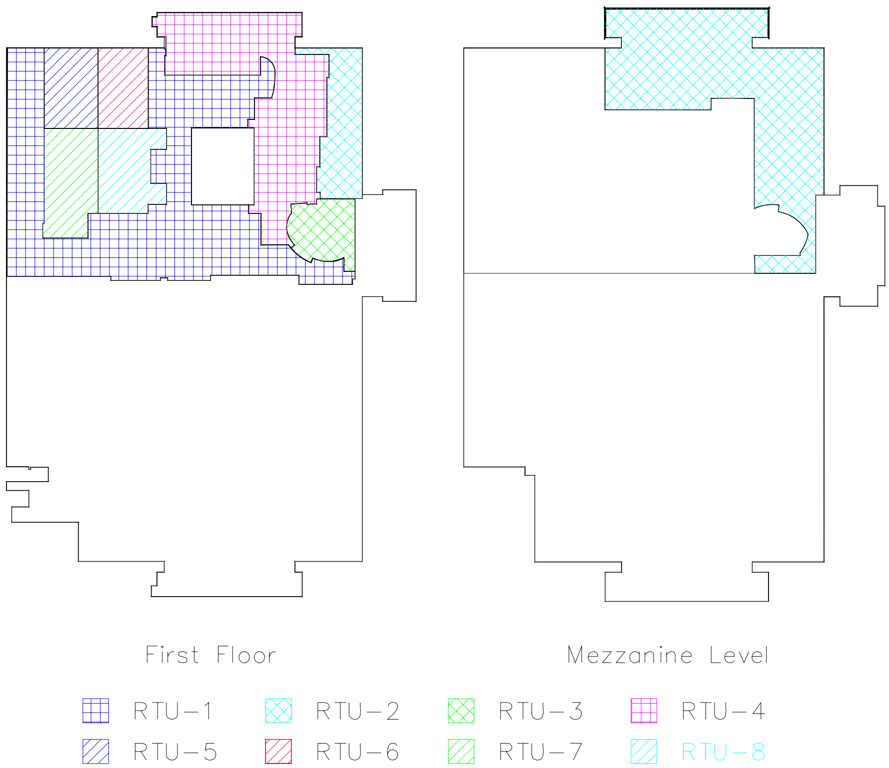

Figure 6.1-3: DOAS Rooftop Air-Handling Unit Zones

The amount of ventilation air introduced into the building decreased by over 50% from the VAV to DOAS systems. ASHRAE Standard 62.1-2004 ventilation rates are still met by the DOAS system. The reason for the increased ventilation requirements for the VAV system is that the critical space ventilation requirement must be met in each zone. This typically results in other spaces being over ventilated. Supplying additional ventilation air is not a problem but does require more energy to condition. By using a DOAS system, each space is supplied with the exact amount of required ventilation, and is not over ventilated. Table 6.1-3 summarizes the amounts of both supply and ventilation air for the two systems.

	VAV (CFM)	DOAS (CFM)	% Reduction by DOAS
Ventilation Air	35,144	15,104	57.0%
Supply Air	510,400	143,742	71.8%

Table 6.1-3: Ventilation and Supply Air Comparison

Annual energy and cost estimates are listed in Table 6.1-4 and Table 6.1-5 respectively. As expected, the DOAS system significantly reduces the amount of fan energy for

Straumann USA. The cooling required for the facility also decreased probably due to the reduction in ventilation air conditioning. The heating energy is also decreased. This is an unexpected benefit but could be a result of supplying a lower minimum air flow to each space which would result in less required reheat at low occupancy conditions. The only increased cost is pump energy and that is to be expected when supplying chilled water to radiant panels rather than just rooftop air-handling units. Overall, this analysis shows that a DOAS system will result in annual energy and cost savings for Straumann USA.

Component	Energy (MMBTU)		
	Electric Centrifugal Chiller		
	Straumann VAV	Straumann DOAS/VAV	DOAS Savings
Air System Fans	1,564	1,093	471
Cooling	1,229	1,202	26
Heating	1,250	616	634
Pumps	356	455	(99)
Cooling Tower Fans	156	155	0
HVAC Sub-Total	**4,554**	**3,521**	**1,032**
Lights	1,509	1,509	0
Electric Equipment	9,326	9,326	0
Non-HVAC Sub-Total	**10,835**	**10,835**	**0**
Grand Total	**15,389**	**14,356**	**1,032**

Table 6.1-4: Annual Energy Comparison

Component	Cost		
	Straumann VAV	Straumann DOAS/VAV	DOAS Savings
Air System Fans	$72,647	$50,727	$21,920
Cooling	$64,415	$62,839	$1,576
Heating	$42,958	$20,298	$22,660
Pumps	$17,916	$24,035	($6,120)
Cooling Tower Fans	$8,961	$8,752	$209
HVAC Sub-Total	**$206,897**	**$166,651**	**$40,245**
Lights	$68,570	$68,570	$0
Electric Equipment	$423,845	$423,845	$0
Non-HVAC Sub-Total	**$492,415**	**$492,415**	**$0**
Grand Total	**$699,312**	**$659,066**	**$40,245**

Table 6.1-5: Annual Cost Comparison

6.2 Central Plant Systems

Although the central plants were not replaced at the 100 Minuteman building at during the renovation work for Straumann USA, a few options will be explored in this report.

The first potential area for energy savings will be explored in comparing electric centrifugal chillers with direct fire absorption chillers. The second area that will be explored for energy savings will be comparing the possibility of changing piping arrangement of the waterside free cooling from a parallel to a series design.

6.2.1 Chiller Options

Currently water cooled centrifugal electric chillers provide chilled water for the Straumann USA facility. While the renovation of the central chilled water plant is not a part of the original project, it is possible that a change in chillers could provide a reduction in energy savings. This analysis will compare the effects of replacing the current chillers in kind with the possibility of replacing the chillers with direct-fired absorption chillers.

While absorption chillers typically have a lower COP than electric chillers, an absorption chiller can save energy under the right circumstances. A sample of an electric centrifugal and absorption chiller is displayed in Figure 6.2-1 and Figure 6.2-2 respectively. Steam driven absorption chillers can take advantage of large process loads that may need to reject heat to power an absorption chiller. Utilizing district steam to power an absorption chiller is yet another way to reduce electric costs. Unfortunately, there is no district steam or large process loads available on site in order to use a steam driven chiller. This limits the analysis to a direct-fired absorption chiller, which will be powered by natural gas already located on site.

Figure 6.2.-1: Trane Electric Centrifugal Chiller

Figure 6.2-2: Carrier Direct-fired Absorption Chiller

Both electric vapor compression, and absorption chillers provide cooling through condensing and evaporating a refrigerant. Electric chillers mechanically change the pressure of the refrigerant with a compressor while an absorption chiller utilizes a sorption and desportion process instead of a compressor to achieve the same effect.

An electric vapor compression cycle is displayed in Figure 6.2-3. In this type of chiller the refrigerant is heated in the evaporator by the warm chilled water return. An electric compressor then increases the pressure of the refrigerant. Next the refrigerant flows through the condenser where it is cooled by supply water from the cooling tower or other condenser water source. The cycle is completed when the cooled refrigerant passes through an expansion valve reducing the pressure and reentering the evaporator.

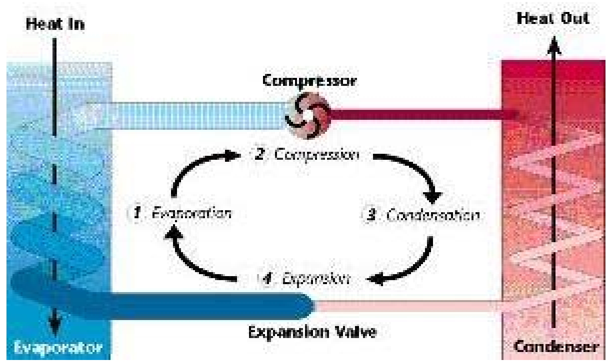

Figure 6.2-3: Vapor Compression Cycle

A double effect absorption chiller is a slightly more complicated process and is displayed in Figure 6.2-4. The process starts once again in the evaporator where it is heated until it becomes a vapor by the warm returning chilled water. The refrigerant than travels into the absorber where it condenses and is mixed with an absorbent. The heat generated in the absorber is removed by the condenser water. The mixture of absorbent and refrigerant is pumped to the low generator. Here some heat is added from the high temperature refrigerant vapors leaving the high generator. This boils some of the refrigerant out of the mixture in the low generator. Some of the mixture of refrigerant an absorbent left in the low generator is mixed with the absorbent returning from the high generator and is sprayed back into the absorber. The rest of the mixture in the low generator is pumped to the high generator. In the high generator heat from burning natural gas boils off the remaining refrigerant which passes into the condenser. The absorbent does not evaporate and travels back to the absorber being cooled along the way by preheating absorbent and refrigerant mixture that is entering both the high and low generators. The refrigerant that entered the condenser in the form of vapor is cooled back to a liquid by condenser water and then passes through an expansion valve before re-entering the evaporator.

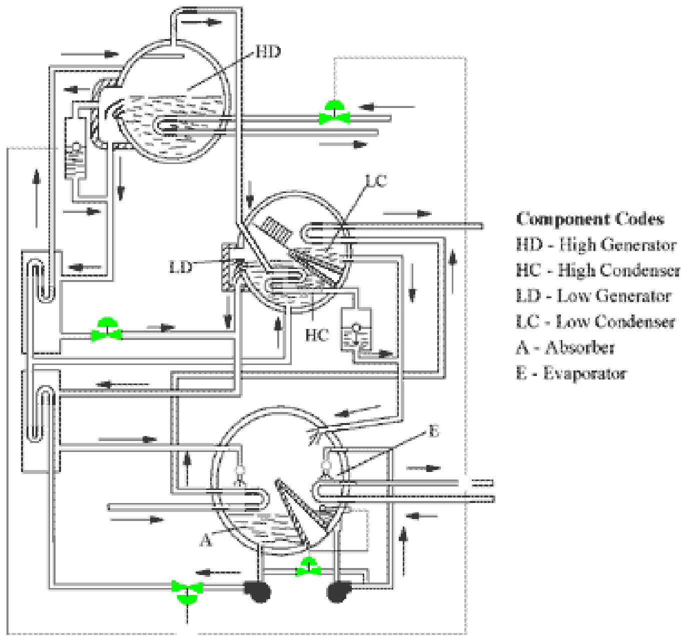

Figure 6.2-4: Double Effect Absorption Cycle

Once again, Carrier's Hourly Analysis Program will be used to perform annual energy analysis. The new absorption chiller will be designed using the same design conditions as the original electric chillers. Condenser water temperatures are 85°F/95°F and chilled water is designed to supply 45°F chilled water. The designs will be based around Carrier's double effect Centurion direct-fire absorption chiller and Trane's EarthWise CenTraVac electric centrifugal chiller. However, any chiller of comparable performance could be used. Both of the previously discussed airside systems will be considered with each type of chiller. Annual energy and cost results are summarized in Tables 6.2-1 and 6.2-2 respectively.

Component	Energy (MMBTU)			
	Electric Centrifugal Chiller		Direct-fired Absorbtion Chiller	
	Straumann VAV	Straumann DOAS/VAV	Straumann VAV	Straumann DOAS/VAV
Air System Fans	1,564	1,093	1,564	1,093
Cooling	1,229	1,202	5,838	5,072
Heating	1,250	616	1,250	616
Pumps	356	455	439	542
Cooling Tower Fans	156	155	246	146
HVAC Sub-Total	**4,554**	**3,521**	**9,337**	**7,468**
Lights	1,509	1,509	1,509	1,509
Electric Equipment	9,326	9,326	9,326	9,326
Non-HVAC Sub-Total	**10,835**	**10,835**	**10,835**	**10,835**
Grand Total	**15,389**	**14,356**	**20,172**	**18,303**

Table 6.2-1: VAV and Absorption Chiller Annual Energy Comparison

Component	Cost			
	Electric Centrifugal Chiller		Direct-fired Absorbtion Chiller	
	Straumann VAV	Straumann DOAS/VAV	Straumann VAV	Straumann DOAS/VAV
Air System Fans	$72,647	$50,727	$72,647	$50,727
Cooling	$64,415	$62,839	$107,264	$92,452
Heating	$42,958	$20,298	$42,958	$20,298
Pumps	$17,916	$24,035	$21,720	$28,737
Cooling Tower Fans	$8,961	$8,752	$13,779	$9,055
HVAC Sub-Total	**$206,897**	**$166,651**	**$258,368**	**$201,270**
Lights	$68,570	$68,570	$68,570	$68,570
Electric Equipment	$423,845	$423,845	$423,845	$423,845
Non-HVAC Sub-Total	**$492,415**	**$492,415**	**$492,415**	**$492,415**
Grand Total	**$699,312**	**$659,066**	**$750,783**	**$693,685**

Table 6.2-2: VAV and Absorption Chiller Annual Cost Comparison

The energy analysis of centrifugal and absorption chillers provided some interesting results. When comparing similar airside systems an absorption chiller consumes more energy and is more expensive annually. However, on an annual cost basis, using a DOAS airside system and an absorption chiller, it is actually cheaper than the electric chiller with a VAV system. When comparing the amount of energy consumed for these two system the opposite is true, the electric chiller and VAV system actually consumes less energy.

It is possible to use the use the absorption chillers for simultaneous heating and cooling which could also result in energy savings. Rather than using a separate boiler system to provide heating, the chiller might be able to provide both hot water for perimeter fin-tube radiators as well as well as chilled water for radiant panels. The chiller heater option with absorption chillers depends largely on the heating and cooling load profiles. The amount of heating a chiller heater can produce depends on the amount of cooling the chiller is performing. Figure 6.2-5 displays the give and take effect of the heating and cooling capabilities of a chiller heater.

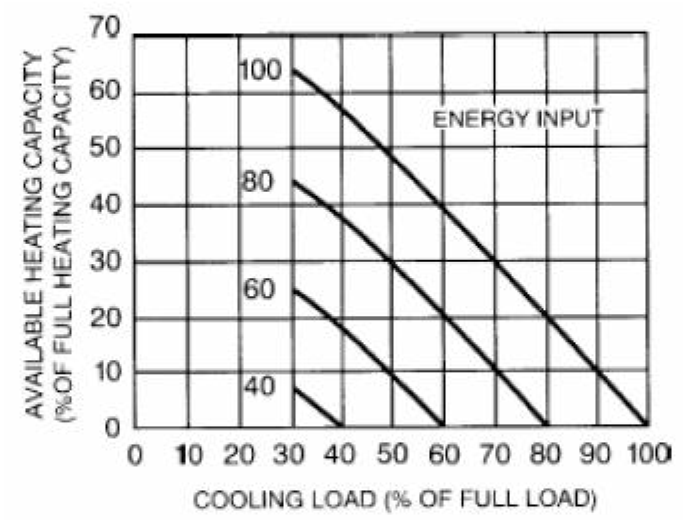

Figure 6.2-5: Heating and Cooling Performance of a Chiller Heater

The heating and cooling load profile displayed in Figure 5.2.1.6 for Straumann USA is used to determine whether a chiller heater would be applicable for the building. The load profile shows that most of the large heating demand occurs when the cooling load is around 400 tons. This poses a bit of a problem because the chiller of 500 tons will be operating at nearly 80% of full capacity. By using the heating and cooling graph in Figure 6.2-5, only 20% of the total heating capacity of the chiller heater can be used. Nearly 2400 MBH or more of heating capacity is needed and at this operation point only 1200 MBH is available. A boiler is still necessary for over half of the heating capacity. While some heating capacity is better than none at all, new boilers are not needed for Straumann USA so there is not additional expense to use the boilers. The boilers are also more efficient at heating than the chiller heater so unless a boiler would need to be replaced and the chiller heater could prevent the purchase of an additional boiler, there does not seem to be any additional benefit from using a chiller heater in this application.

A full analysis of the heating and cooling load profiles resulted in determining that heating is needed 3222 hours during the year at Straumann USA. The chiller heater would be available for combined heating and cooling in only 733 of those hours or 23% of the time. Of the hours a chiller heater could be used nearly one third of the time, 226 hours, a supplement boiler would be necessary to meet the heating load. Overall the chiller heater would only be able to meet the full heating demands of Straumann USA 16% of the time heating is needed.

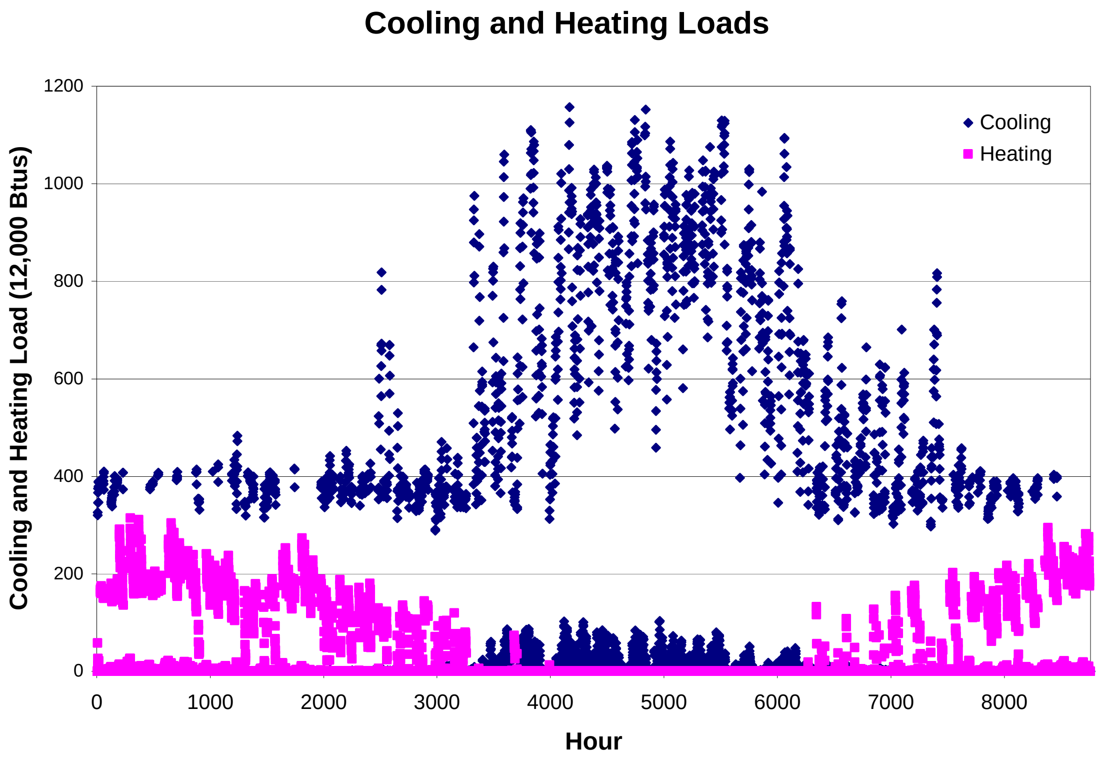

Figure 6.2-6: Simultaneous Heating and Cooling Load

6.2.2 Free Cooling Options

There are several opportunities with any mechanical system to reduce energy consumption, and save annual operating costs. One way to do so is to include waterside free cooling. When a building is experiencing reduced load conditions, and low wet bulb temperatures exist, it is possible to reject heat from the chilled water loop without the use of a chiller. Any hour that the chiller is turned off, significant amounts of energy can be saved, since a chiller is typically one of the largest energy consuming pieces of equipment. Depending on the number of hours waterside free cooling can be utilized, a building owner can receive a significant reduction in the yearly energy costs.

Two main types of water side free cooling exist: direct and indirect. Direct free-cooling simply allows the chilled water return to bypass the chiller and directly enter the cooling tower where it is cooled and supplied to the loads at the chilled water supply temperature. The main disadvantage of this type of free cooling is that debris can enter the chilled water system through the cooling tower. The second major type of waterside free cooling, which is present in at Straumann USA, is the indirect method. In this configuration, the chilled water and condenser water loops remain separated. Heat is transferred from the chilled water line to the condenser water line typically through a plate and frame heat exchanger. While this type of waterside free cooling requires the additional first cost of a plate heat exchanger, it makes certain no debris from the cooling tower enters the chilled water loop where cooling coils could possibly get clogged.

While there are two types of waterside free cooling, indirect free cooling can also be piped in one of two ways. The first is a parallel arrangement in which the heat exchanger utilized for free cooling is placed in parallel with the chiller, refer to Figure 6.2-7 for a schematic of the system.

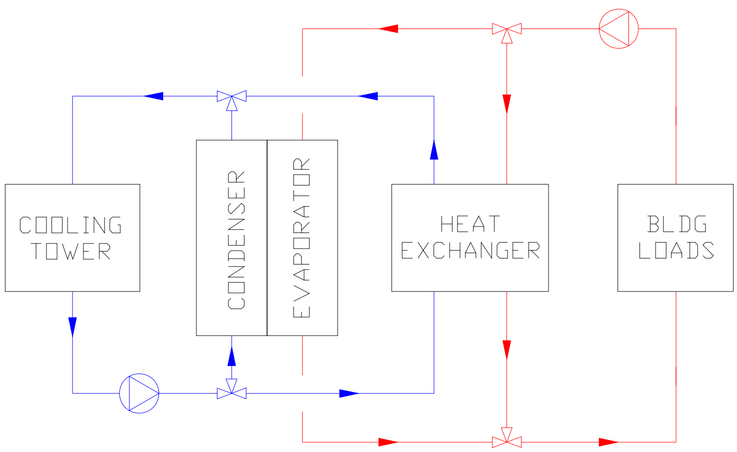

Figure 6.2-7: Parallel Waterside Free Cooling Schematic

This seems to be the most common way an indirect system is piped. In this arrangement, waterside free cooling can only be utilized when it can reject enough heat to produce the chilled water supply temperature. Any conditions that do not reject the entire load of the chilled water loop require the operation of the chiller

The second piping configuration for an indirect free cooling system is a series arrangement. In this layout, the heat exchanger is placed in series with the chiller, refer to Figure 6.2-8 for a schematic of the system.

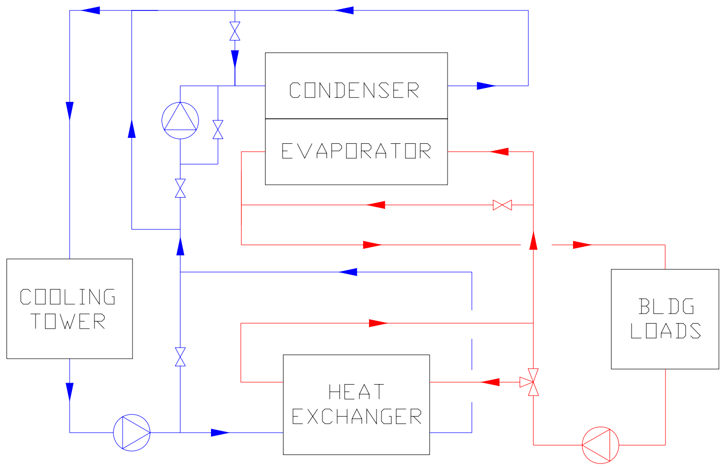

Figure 6.2-8: Series Waterside Free Cooing Schematic

In such an arrangement, free cooling can be used as soon as the cooling tower is able to produce condenser water that is below the chilled water return temperature. This allows waterside free cooling to be used more hours each year. It is not necessary for the condenser water loop to reject the all the heat from the chilled water loop. In this configuration, free cooling can be utilized to pre-cool the chilled water return before it enters the evaporator, resulting in a lower load seen by the chiller. When the condenser loop is able to reject all the heat from the chilled water loop, the chiller can be turned off and the system will operate just like a parallel piping arrangement. Some disadvantages of the series system include more advanced controls, and an extra pump on the condenser water loop.

As previously discussed, waterside free cooling is most effective in climates with a low wet bulb temperature. Figure 6.2-9 shows the predicted wet bulb distribution used by Carrier's Hourly Analysis Program for a year in Andover Massachusetts.

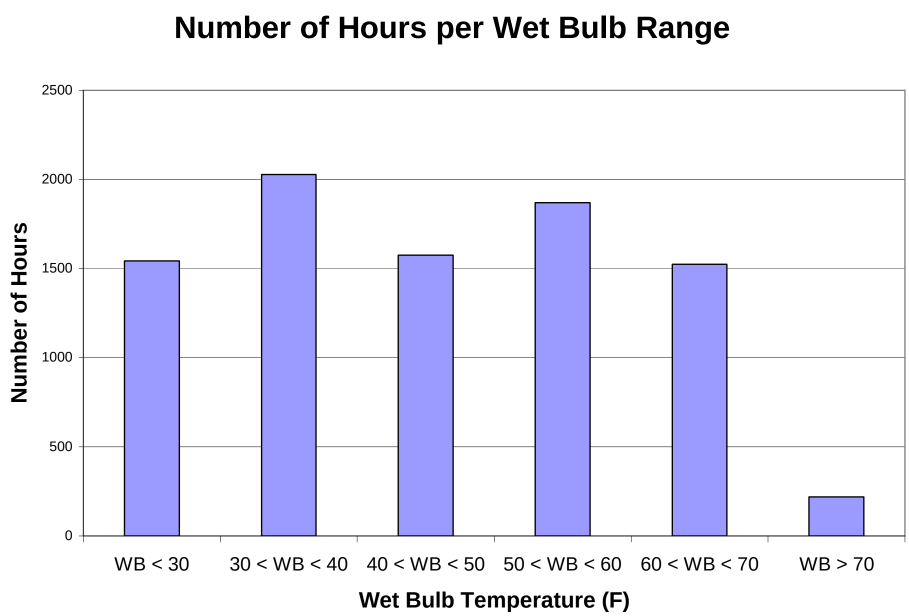

Figure 6.2-9: Yearly Hours per Wet Bulb Range

While it can be seen that over half of the hours in Andover have a wet bulb temperature of less than 50°F, the actual hours where cooling is necessary may not occur during the times of the low wet bulb temperature. Such low temperatures may or may not be capable of providing free cooling depending on the size of the load. Figure 6.2-10 displays the hourly load with the corresponding wet bulb condition. As can be seen by the load distribution, it appears that a load of 250 – 350 tons (slightly less than 50% of the design load) is most common at wet bulb temperatures below 40°F. Based on this comparison of loads and wet bulb temperature, it can be assumed that such a building might be able to effectively utilize waterside free cooling since there are quite a few hours with low wet bulb temperatures and reduced loads.

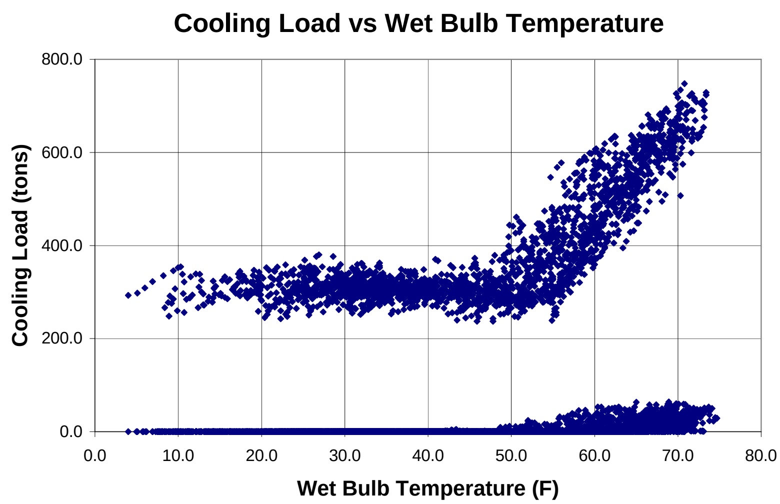

Figure 6.2-10: Straumann Load vs Wet Bulb Temperature

In order to analyze the two types of waterside free cooling, the Engineering Equation Solver (EES) program will be used to calculate the power required at each load coupling a cooling tower, chiller, and heat exchanger. The basis for the cooling tower and chiller models are taken from AE 557, a Penn State course in central cooling systems.

Parallel Waterside Free Cooling Operation

A heat exchanger is modeled between the chiller and cooling tower for the parallel configuration. The cooling tower runs at 100% fan operation until the condenser water temperature reaches 60°F. If the tower is capable of producing condenser water at less than 60°F the fan is modulates between off and full speed to maintain 60°F condenser water. Refer to Figure 6.2-11 for a schematic of the system operation under such conditions. Portions of the system that are "off" or have no flow are shown in grey.

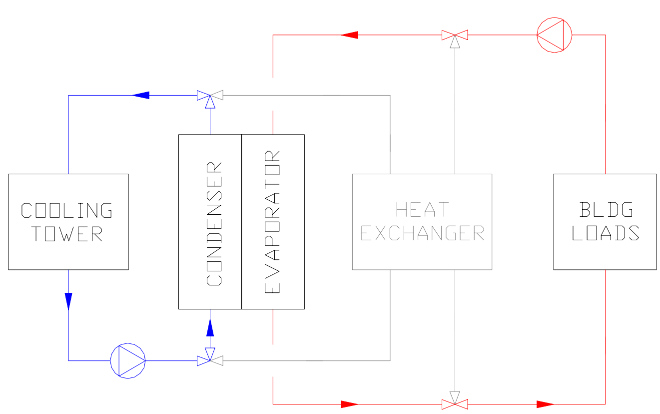

Figure 6.2-11: Parallel System Chiller Mode

If the cooling tower is capable of producing condenser water at temperature low enough to completely reject heat from the chilled water system, the chiller is then turned off. During this operation period, the cooling tower modulates between full speed and off to supply condenser water that maintains 45°F chilled water leaving the heat exchanger. Refer to Figure 6.2-12 for a schematic of the heat exchanger operation in the parallel system. If 45°F chilled water can not be produced by full speed fan operation, the chiller will turn back on until such conditions can again be met.

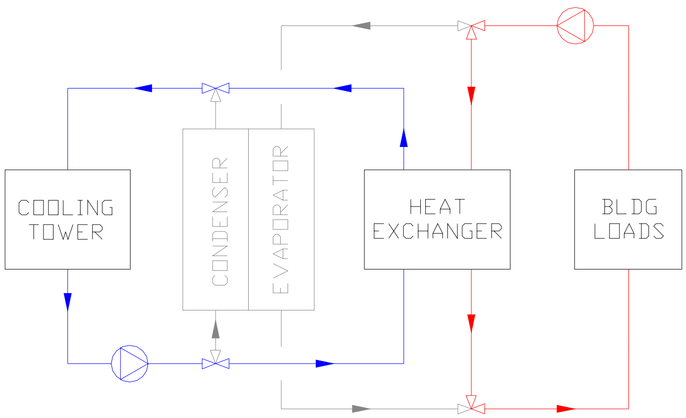

Figure 6.2-12: Parallel System Free Cooling Mode

Series Waterside Free Cooling Operation

The series heat exchanger model is slightly more complicated. Once again, the cooling tower runs with fans at 100% until it produces a condenser water temperature of 60°F for the chiller while bypassing the heat exchanger. When a full speed fan is able produce condenser water temperatures between 55°F and 60°F the fan is modulated between 100% and off to maintain 60°F condenser water for the chiller. Refer to Figure 6.2-13 for a schematic of this operation mode.

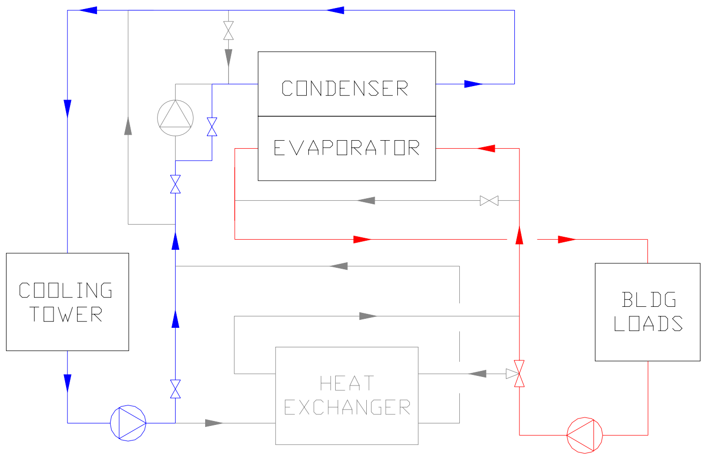

Figure 6.2-13: Series System Chiller and Free Cooling Mode

Once the tower is able to produce condenser water that is lower than 55°F, the heat exchanger is no longer bypassed. The condenser water first passes through the heat exchanger to pre-cool the chilled water before entering the evaporator. The condenser water that leaves the heat exchanger will then mix with some of the water recirculated from the condenser to maintain 60°F entering the chiller. Refer to Figure 6.2-14 for a schematic of this type of operation.

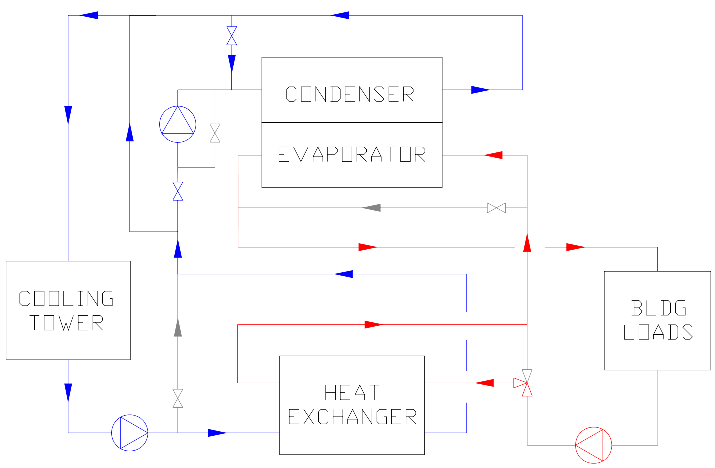

Figure 6.2-14: Series System Chiller and Free Cooling Mode

The condenser water system continues to operate in the combination chiller and heat exchanger mode until the chilled water leaving the heat exchanger reaches 45°F. At this point the chiller turns off, and the condenser water system operates in a full waterside free cooling mode just like the parallel arrangement. Figure 6.2-15 displays a schematic of this type of operation.

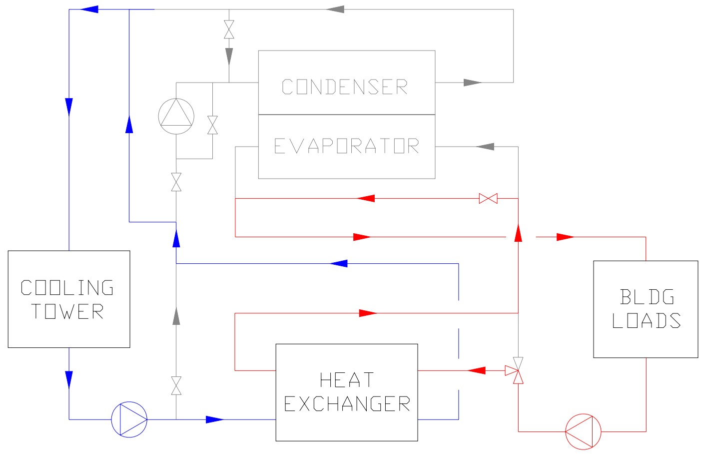

Figure 6.2-15: Series Free Cooling Mode

The results for the two types of waterside free cooling are summarized in Tables 6.2-2 and 5.2-3. One of the surprising results is that free cooling without a chiller can only be used 2 hours of the entire year for the Straumann facility. Another interesting result is that if a series free cooling system is turned on as soon as condenser water can be produced below 55°F, it will use more annual energy than a free cooling system in a parallel arrangement.

Heat Exchanger in Parallel						
	Hours	Ton Hours	Fan Energy (kW)	Chiller Energy (kW)	Additional Pump (kW)	Total Energy (kW)
No Cooling	5051	0	0	0	0	0
Free Cooling	2	591	50	0	0	50
Chiller Cooling	3707	904743	103773	665889	0	769662
Total	8760	905334	103822	665889	0	769711

Table 6.2-2: Summary of Parallel Free Cooling Results

Heat Exchanger in Series (55F)						
	Hours	Ton Hours	Fan Energy (kW)	Chiller Energy (kW)	Additional Pump (kW)	Total Energy (kW)
No Cooling	5051	0	0	0	0	0
Free Cooling	2	591	50	0	0	50
Series Cooling	108	33036	3222	11480	934	15635
Chiller Cooling	3599	871707	103773	651008	0	754780
Total	8760	905334	107044	662487	934	770465

Table 6.2-3: Summary of Series Free Cooling Results

After finding that a series free cooling system could actually increase the amount of energy a chilled water plant consumes annually, the series free cooling is optimized to reduce the annual energy to a minimum. In order to operate the system in a way the consumes the least energy, the series free cooling system should operate in a series cooling mode (operating both the heat exchanger and chiller) until the condenser water temperature can be produced at 51°F. Prior to this temperature the system should maintain a condenser water temperature of 60°F and operate only the chiller. The results of the 51°F series free cooing are summarized in Table 6.2-4.

Heat Exchanger in Series (51F)						
	Hours	Ton Hours	Fan Energy (kW)	Chiller Energy (kW)	Additional Pump (kW)	Total Energy (kW)
No Cooling	5051	0	0	0	0	0
Free Cooling	2	591	50	0	0	50
Series Cooling	38	11470	1134	12726	324	14184
Chiller Cooling	3669	893274	103773	651008	0	754780
Total	8760	905334	104956	663734	324	769014

Table 6.2-4: Summary of Optimized Series Results

The three different waterside free cooling systems are compared in Table 6.2-5. The results show that even when optimizing the series free cooling system for Straumann USA, only a minimal savings of 698kW can be expected over the course of a year, while a series free cooling system starting to operate at a condenser water temperature of 55°F will actually consume more energy.

	Ton Hours Free Cooling	Ton Hours Series Cooling	Total Energy (kW)	Savings Compared to Parallel (kW)
Parallel	591	0	769711	-
Series (55)	591	33036	770465	-754
Series (51)	591	11470	769014	698

Table 6.2-5: Summary of Free Cooling Results

The results are particularly interesting. Even though the at first it was assumed that the weather and loading conditions for Straumann USA would result in good application for waterside free cooling, the results tell a different story. It is very possible that a under some conditions, a series free cooling system can actually consume more energy than just running the chiller. When the condenser water is only slightly below the chilled water return temperature, the pre-cooling of the chilled water is minimal. Under such conditions, this means the chiller would require almost as much energy with the slight pre-cooling as without it. The overall increased energy is caused by increases in fan and pumping energy. When both the chiller and heat exchanger are in operation two pumps are running rather than just one. The fan energy of the cooling tower would also increase in order to produce condenser water temperature below 60°F. By minimizing the total energy consumed during potential free cooling hours, it is found that waiting to use free cooling until 51°F condenser water can be produced, the savings in chiller energy outweighs any additional pumping and fan costs.

Based on the results, a plot of cooling load versus wet bulb temperature for each operation mode is displayed in Figure 6.2-16. This shows that for Straumann USA to use free cooling alone, the wet bulb temperature must be less than 6°F. The optimal series cooling can be use when the wet bulb temperature ranges from 6°F to 15°F. Any temperatures above this will solely require a chiller to reject heat from the building chilled water system.

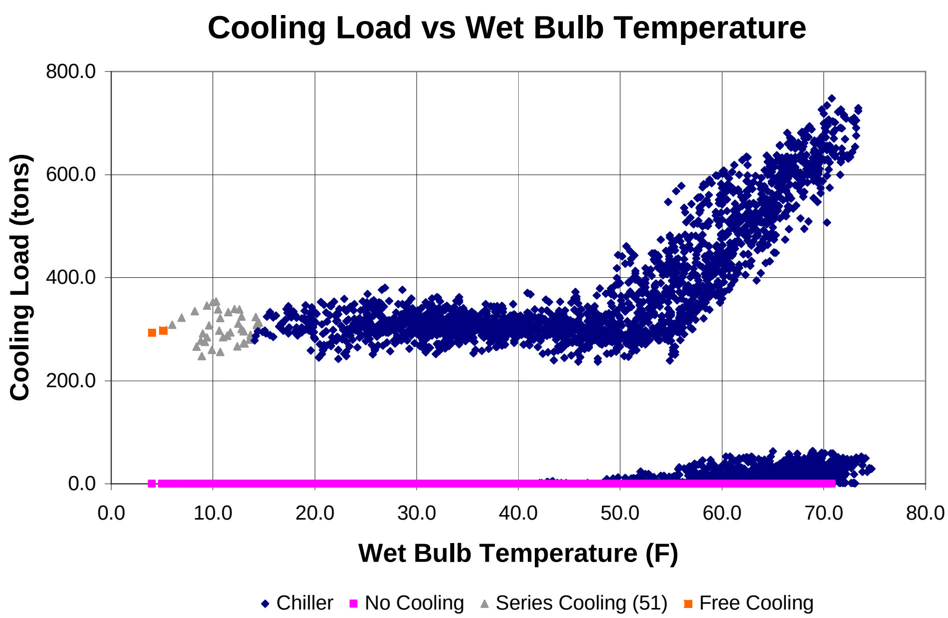

Figure 6.2-16: Cooling Load vs Wet Bulb Temperature at Each Cooling Mode

6.3 Mechanical Conclusions

The analysis of the Straumann USA facility provided some very interesting results. When comparing the airside systems the DOAS system saves on annual energy costs. When comparing the direct-fire absorption and electric centrifugal chillers with the same airside system, the absorption chiller resulted in a higher annual energy cost. However, an absorption/DOAS system did result in a lower annual energy cost than an electric/VAV system. When considering using the absorption chiller to both simultaneously produce hot and chilled water it is found that the heating load for Straumann USA would only be met 16% of the time. Since boilers are already present, there would be no reduction boiler size for the facility so no initial cost savings would be a factor. An analysis of the waterside free cooling capabilities of Straumann USA also provided some interesting results. While a few additional hours of free cooling can be obtained by using a series free cooling arrangement, it must be carefully controlled to prevent the cooling costs from actually increasing if condenser water is supplied between 51°F and 55°F.