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Straumann USA
Andover, MA

Mechanical Technical Report 3
Mechanical Systems Existing Conditions Evaluation

Straumann USA
Andover, MA

November 21, 2006

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1.0 Executive Summary

The Straumann USA renovation project featured the replacement of the airside systems of the facility while using the existing heating and cooling central plants of the building. Ten rooftop units serve a variety of spaces including manufacturing areas, offices, an auditorium, and dental operatory suites. The project was designed to comply with the requirements of the Massachusetts State Building Code 780 CMR.

Based on an ASHRAE Standard 62.1-2004 analysis, all rooftop units comply with the required ventilation rates even though the project was designed around ASHRAE Standard 62.1-2001 An energy analysis of the building found that the predicted electric ($671,489) and fuel costs ($42,958) of the building was close to the actual costs for 2005. The total estimated costs were $714,456 and the actual costs were $697,650.

The rooftop air-handling units operate in one of six different modes: occupied, warm-up, cool-down, unoccupied (normal off), night heating, and night cooling. A zero to one hundred percent airside economizing option is also included with each of the rooftop units. A detailed explanation of rooftop unit operation can be found in the full report.

The system selected for Straumann USA was certainly a logical one, and serves the building well. However, given the time and resources, it would be very interesting to estimate how some other systems would compare. One possible system to compare would be a DOAS system with parallel sensible cooling system. During the design of the Straumann USA project renovations to the central plants were considered but the decision was made to use the existing plants. This could also allow for some interesting options to be explored such as thermal storage, or direct-fired absorption chillers.

2.0 Background and Design Considerations

The Straumann USA facility is located in Andover Massachusetts. Straumann USA occupies close to half of the 100 Minuteman building. The entire building is 327,000 square feet and is owned by The Brickstone Companies. It is a two-story building featuring first floor and mezzanine levels. The Straumann facility occupies 153,000 square feet and is separated from the rest of the building by a firewall in order to comply with maximum floor area codes. The areas of the building Straumann USA occupies can be seen below in Figure 2.1.

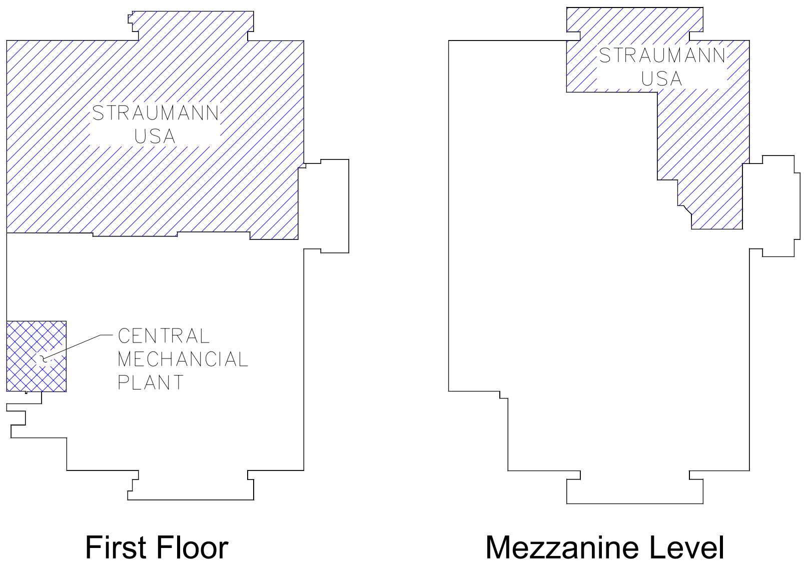

Figure 2.1 – Straumann USA Occupancy Locations

The Straumann USA facility features a variety of spaces. It is largely a combination office and light manufacturing building. However, other unique spaces include a dental operatory suite, a dental training room, and an auditorium seating up to 95 people.

Straumann USA is served by 10 rooftop air handling units. Nine of the units are variable air volume ranging from 21,000 cfm to 33,000 cfm and the tenth unit that serves the auditorium area is a 6,400 cfm constant air volume unit. All 10 of the units condition air with a chilled water cooling coil and a steam heating coil. Table 2.1 breaks down the type of areas each rooftop unit serves and lists the size of each unit. Figure 2.2 displays the location of each zone within the building.

Roofop Unit Summary				
	Max CFM	Number of People per Unit	Square Feet Served	Areas Served
RTU-1	33,000	126	27,139	First floor manufacturing support areas and mezzanine level server room
RTU-2	33,000	248	19,968	First floor office and dental operatory areas
RTU-3	6,400	151	3,303	First floor auditorium
RTU-4	33,000	204	20,602	First floor and mezzanine office areas
RTU-5	21,000	81	11,126	First floor manufacturing support areas
RTU-6	21,000	118	17,326	Mezzanine office areas
RTU-7	33,000	20	5,850	Manufacturing area
RTU-8	33,000	20	5,850	Manufacturing area
RTU-9	33,000	20	5,850	Manufacturing area
RTU-10	33,000	20	5,850	Manufacturing area

Table 2.1 – Spaces Served by Each Rooftop Air Handling Unit

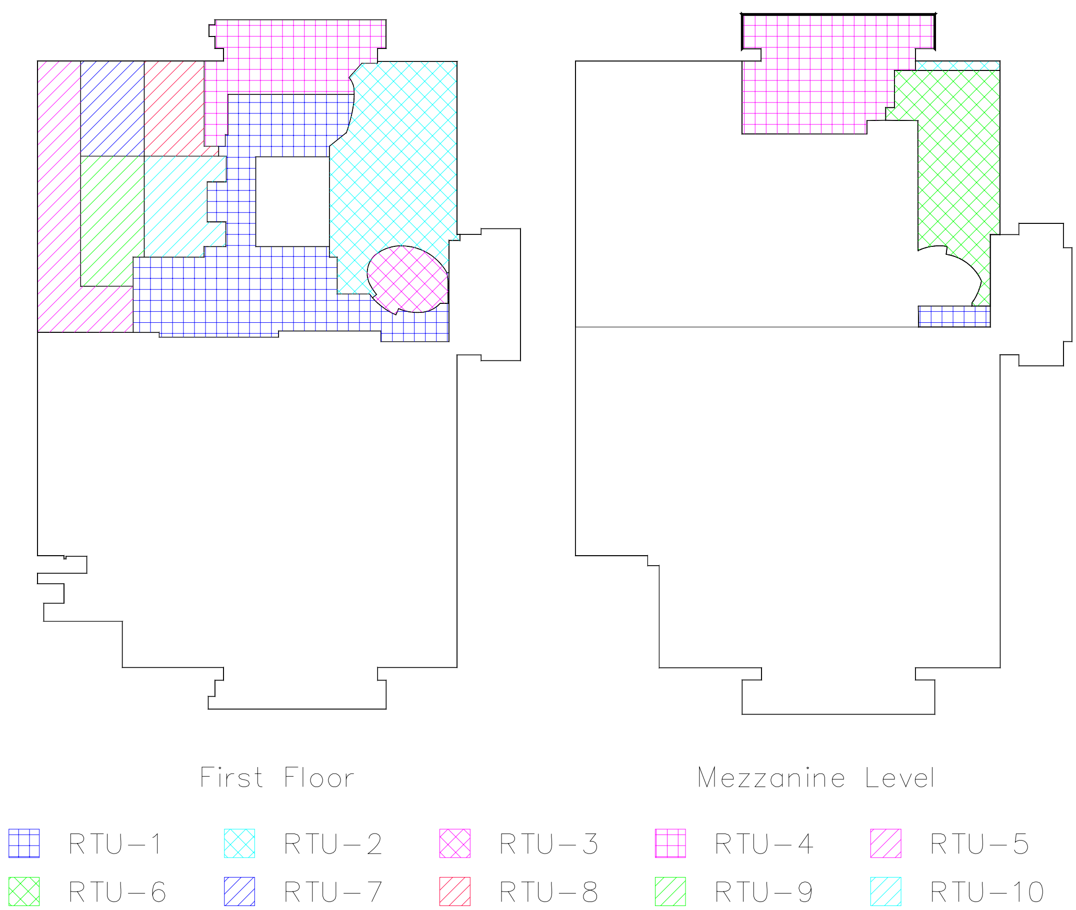

Figure 2.2 – Rooftop Air-Handling Unit Zones

The central plant produces building chilled water and steam for the entire building not just the Straumann USA facility. Figure 2.1 shows the location of the central plant in the building. The central plant includes three water-cooled electric centrifugal chillers of 750, 500, and 350 tons. Heat is rejected from the condenser water system with two cooling towers of 680 and 750 tons. The system is equipped with a waterside free cooling mode that directly rejects heat from the chilled water loop to the condenser water loop by using a plate heat exchanger. Refer to Figure A.2 in Appendix A for the chilled water schematic. High pressure steam is produced for the building by two 11.7MBH fuel oil or natural gas fired boilers. Steam is then reduced to a lower pressure (15psi) and routed to the heating coils in the rooftop units. A shell and tube heat exchanger uses the steam to heat the building hot water used by the fintube radiators at the perimeter of the building. Refer to Figure A.1 in Appendix A for the steam and hot water schematic.

Several design considerations were taken into account during the renovation of the Straumann USA facility. Interior spaces were designed to fall within the comfort region of AHSRAE Standard 55.1-2004. The project was also designed to comply with the requirement of the Massachusetts State Building Code 780 CMR. Since the central plant systems were existing several assumptions were made with regard to the supply conditions. Building chilled water was designed assuming the supply temperature would be 47°F and the return temperature would be 54°F. The building hot water system was designed with supply and return temperatures of 228°F and 180°F respectively. The rooftop units were designed with the assumption that 12psi steam would be available for the heating coils.

Exhaust systems were designed for the toilet room at a rate of 75cfm per fixture. Janitors closets would also be exhausted at rate of 6 air changes per hour. The manufacturing area would be equipped with a general exhaust to prevent excessive temperatures. Refer to the control strategies in section 8 for more details. The final area to be provided with exhaust was the loading dock. It was designed with a dedicated exhaust of 6 air changes per hour with the makeup being drawn through the manufacturing area.

3.0 Energy Sources and Rates

Natural gas and electricity are the two available energy sources at the Straumann USA facility. The rates for are summarized in Table 3.1 for 2006.

Energy Source Rates		
	Monthly Electric Cost per kWh	Cost per Therm
Jan	$0.1672	$1.6803
Feb	$0.1973	$1.5686
Mar	$0.1980	$1.5267
April	$0.1774	$1.5703
May	$0.1510	$1.3863
June	$0.1211	$1.5230
July	$0.1301	$1.5172
August	$0.1341	$1.6792
Sept	$0.1325	$1.6127
Oct	$0.1099	$1.4964
Nov	$0.1134	$1.6266
Dec	$0.1492	$1.7743
Yearly Average	$0.1484	$1.5801

Table 3.1 – Natural Gas and Electric Rates for 2006

4.0 Design Conditions

Straumann USA was designed to meet interior conditions of 70°F/50% relative humidity during the winter months, and 75°F/50% relative humidity during the summer months. The external design conditions that the system was designed for are listed in Table 4.1. Displayed in Table 4.2 are the internal loads considered for the Straumann USA facility.

External Design Conditions		
Summer	Dry Bulb	95°F
	Mean Coincident Wet Bulb	75°F
Winter	Dry Bulb	10°F

Table 4.1 External Design Conditions

Interior Loads		
Occupants	Sensible	250 Btu/h
	Latent	240 Btu/h
Manufacturing Area	Lighting	2.2 W/ft^2
	Equipment	58.4 W/ft^2
Nonmanufacturing Area	Lighting	1.3 W/ft^2
	Equipment	3 W/ft^2

Table 4.2 Interior Loads

Several other factors were taken into consideration during the mechanical system design. An allowance of 0.3 air changes per hour was made for all rooms located on the perimeter of the building. NC levels were designed to fall between 30 and 40 for all HVAC related equipment.

5.0 Design Ventilation Requirements

The ventilation requirements for the Straumann USA facility were calculated based on ASHRAE Standard 62.1-2004 and compared to the amount of ventilation air in the original design. The units were sized for the project based on ASHRAE Standard 62.1-2001, however, based on the results in summarized in Table 5.1 the ventilation rates meet or exceed those specified in ASHRAE Standard 62.1-2004. Each rooftop unit was actually oversized to allow for interior space layouts, occupancies, and sizes to changes without having to alter or replace the rooftop units in order to provide the required ventilation air.

ASHRAE 62.1-2004 Ventilation Requirements					
	ASHRAE Standard 62.1-2004 Ventilation Requirements (Vot) (CFM)	H.F. Lenz Ventilation Requirements	Nominal OA (Σvoz) (CFM)	Critical Zp Value	Compliance with ASHRAE Standard 62.1-2001
RTU-1	4299	5830	2580	0.54	Yes
RTU-2	3953	7949	2372	0.54	Yes
RTU-3	1096	3302	877	0.27	Yes
RTU-4	4009	6150	2406	0.47	Yes
RTU-5	2957	3883	1774	0.47	Yes
RTU-6	1996	4070	1397	0.38	Yes
RTU-7	902	990	902	0.09	Yes
RTU-8	902	990	902	0.09	Yes
RTU-9	902	990	902	0.09	Yes
RTU-10	902	990	902	0.09	Yes

Table 5.1 ASHRAE 62.1-2004 Ventilation Requirements

6.0 Design Heating and Cooling Loads

Several values were assumed in order to produce the load analysis for the Straumann USA facility. Table 6.1 summarizes the values used for the estimated load, and the original design load for the facility.

Load Analysis Assumptions		
	Estimated	H.F. Lenz Design
OA Ventilation Rates	ASHRAE Standard 62.1-2004	ASHRAE Standard 62.1-2004
Lighting Loads		
Office	1.3 W/ft2	1.3 W/ft2
Manufacturing	2.2 W/ft2	2.2 W/ft2
Equipment Loads		
Office	3.0 W/ft2	3.0 W/ft2
Manufacturing	38W/ft2	38W/ft2
Design Conditions	ASHRAE Fundamentals 2005 (0.4%)	
Summer		
Dry Bulb	90.8	95
Mean Coincident Wet	73.1	75
Winter		
Dry Bulb	7.7	10

Table 6.1 Estimated and Design Load Assumptions

The equipment actually selected and scheduled on the design drawings were oversized in order to prevent a complete renovation of the space if the needs of the tenant changes. The estimated cooling and heating loads are compared to initial load design performed by the mechanical engineering designer. The areas, locations, and occupancies of the spaces may have changed slightly from the initial design, but loads should be a reasonably good source for comparison purposes. The heating and cooling load summaries are located in Table 6.1, along with the airflow rates of each unit.

Overall, the estimated cooling load is slightly higher than the design load. This could be attributed to several factors. First, Trane Trace was used to create the design loads while HAP was used for the load estimates. The design loads were based on the preliminary design, and not the final construction documents. The size of some of the rooms, and occupancies may have changed slightly to create such differences. The total airflow supplied by the estimated units is slightly lower than the design airflow. This could be due to assumptions of load distributions. Since the cooling load is actually larger for the estimate but less air is supplied, it could be possible that a larger amount the roof load was assumed to directly heat the plenum air and not have as great an effect on the occupied space.

The design heating load was slightly lower than the initial design load calculated. Again, this could partially be contributed to a different program performing the analysis.

A larger safety factor may have been used in order to prevent the under sizing of equipment for the actual design.

Estimated and Design Loads and Airflows						
	Estimated Design Cooling Load (MBTU)	H.F. Lenz Design Cooling (MBTU)	Estimated Design Heating Load (MBTU)	H.F. Lenz Design Heating (MBTU)	Estimated Design CFM	H.F. Lenz Design CFM
AC-1,2	120.4	118.3	0	0	5038	5650
AC-3	35.2	34.7	0	0	1275	1250
AC-4,5	120.4	118.3	0	0	5038	5650
AC-6	35.5	34.7	0	0	1275	1250
AC-7	41.3	34.7	0	0	1551	1250
AC-8	35.5	34.7	0	0	1275	1250
RTU-1	1118.6	861.6	408.2	466.4	20151	25704
RTU-2	1174.4	1022.3	601.4	587.8	20941	28329
RTU-3	258.9	260.8	82.6	214	4319	4543
RTU-4	1188.5	961	394.8	534.4	24186	26800
RTU-5	848.2	485.2	251.2	264.2	15360	15876
RTU-6	94.8	562.7	351.2	280.9	15723	16250
RTU-7,8,9,10	4316.6	3559	62.6	438.5	144860	148381
Total	9388.3	8088	2152	2786.2	260992	282183

Table 6.2 Estimated Design Loads and Airflows

Comparison of Estimated and Design Load and Ventilation Indices						
	Estimated Cooling ft^2/ton	Design Cooling ft^2/ton	Estimated Supply Air cfm/ft^2	Design Supply Air cfm/ft^2	Estimated Ventilation cfm/ft^2	Design Ventilation cfm/ft^2
AC-1,2	55	56	9.06	10.16	0.00	0.00
AC-3	57	58	7.63	7.49	0.00	0.00
AC-4,5	53	54	9.54	10.70	0.00	0.00
AC-6	178	183	2.41	2.37	0.00	0.00
AC-7	78	93	5.79	4.66	0.00	0.00
AC-8	66	67	6.57	6.44	0.00	0.00
RTU-1	269	410	0.80	0.87	0.36	0.11
RTU-2	222	281	0.96	1.18	0.64	0.24
RTU-3	182	181	1.10	1.16	0.78	0.77
RTU-4	226	290	1.08	1.15	0.43	0.21
RTU-5	154	282	1.41	1.39	0.56	0.15
RTU-6	1848	301	1.08	1.15	0.66	0.17
RTU-7,8,9,10	67	82	5.98	6.13	0.09	0.07
Total	160	98	2.09	2.13	0.43	0.17

Table 6.3 Comparison of Estimated and Design Load and Ventilation Indices

7.0 Annual Energy Use

There was no energy analysis performed for Straumann USA by H.F. Lenz. There would have been an additional cost for the company to perform such analysis, and the owner decided not to pursue this option.

The energy analysis for performed for this report was compiled using Carrier's Hourly Analysis Program. It was necessary to make several assumptions in regards to schedules, electric and fuel rates which can be found in Appendix D. Extra energy analysis information can be found in Appendix E.

The estimated annual energy costs for Straumann USA was found to be $714,456. The HVAC energy costs account for approximately 31% ($222,041) of the total annual energy cost. Table 7.1 summarizes the costs for each system component. The results of the finding can also be seen in the form of dollars per square foot in Table 7.2. Refer to Appendix E for additional cost breakdowns, and graphs. Along with the annual costs, the annual energy consumption rates were calculated and the results are summarized in Table 7.3.

Based on the results displayed in Table 7.4, the energy model predicts a slightly higher yearly electric cost, however, this could be due to the way the electric rate was calculated (Refer to Appendix D). It does not seem to be a large enough difference to cause any concern. The slight difference could also be caused by the assumed values for lighting and power per square foot differing from the amount of electricity actually consumed.

The predicted fuel cost is much lower than the actual cost. One possible difference could be assuming that half of the heating energy is used by the Straumann facility. At this time, the breakdown of the heating bill by tenant was not provided, just the overall heating cost. Straumann USA occupies the southern portion of the building and may use less than half of the heating for the building since it would have a higher solar heat gain which would decrease the actual costs heating costs. Another factor would be the actual weather the building experienced during 2005. If the weather was warmer than normal during the winter months, it would result in lower heating costs.

Component	Cost
Air System Fans	$72,647
Cooling	$48,432
Heating	$42,958
Pumps	$19,052
Cooling Tower Fans	$38,952
HVAC Sub-Total	**$222,041**
Lights	$68,570
Electric Equipment	$423,845
Misc. Electric	$0
Misc. Fuel Use	$0
Non-HVAC Sub-Total	**$492,415**
Grand Total	**$714,456**

Table 7.1 Annual Component Energy Costs

Component	Cost per Square Foot
Air System Fans	$0.581
Cooling	$0.387
Heating	$0.343
Pumps	$0.152
Cooling Tower Fans	$0.311
HVAC Sub-Total	**$1.775**
Lights	$0.548
Electric Equipment	$3.388
Misc. Electric	$0.000
Misc. Fuel Use	$0.000
Non-HVAC Sub-Total	**$3.936**
Grand Total	**$5.711**

Table 7.2 Annual Component Energy Costs per Square Foot

Component	Site Energy (kBTU)	Site Energy (kBTU/ft^2)	Source Energy (kBTU)	Source Energy (kBTU/ft^2)
Air System Fans	1,564,042	12.501	5,585,866	44.648
Cooling	923,702	7.383	3,298,937	26.369
Heating	2,499,012	19.975	2,809,784	22.459
Pumps	428,824	3.428	1,531,516	12.241
Cooling Tower Fans	765,774	6.121	2,734,908	21.860
HVAC Sub-Total	**6,181,354**	**49.408**	**15,961,011**	**127.577**
Lights	1,508,801	12.060	5,388,574	43.071
Electric Equipment	9,326,197	74.545	33,307,848	266.231
Misc. Electric	0	0.000	0	0.000
Misc. Fuel Use	0	0.000	0	0.000
Non-HVAC Sub-Total	**10,834,998**	**86.604**	**38,696,422**	**309.302**
Grand Total	**17,016,352**	**136.012**	**54,657,433**	**436.879**

Table 7.3 Annual Energy Consumption Rates by System Component

Annual Energy Costs		
	Estimated	Acutal
Fuel Costs	$42,958	$75,000
Electric Costs	$671,498	$622,650

Table 7.4 Annual Fuel and Electric Costs

8.0 Description of System Operation

The control of the new equipment was obtained and will be discussed in this section. However, the control strategies were not available for equipment that was not new to the building, such as chillers, cooling towers, and boilers. Central plant schematic drawings and rooftop air-handling unit figures can be referred to in Appendix A.

Variable Volume Rooftop Units 1, 2, 4-10

Each of the rooftop units supplied with variable frequency drives operates in either the occupied or unoccupied mode. Within the occupied schedule, the unit can enter a warm up mode when the space temperature is below the set point of 70°F/75°F (winter/summer) or the cool-down mode when the space temperature is above the set point. Night heating and night cooling are available settings during the unoccupied mode. However, night heating and cooling only take effect when a space falls below 65°F or above 85°F.

The rooftop unit can only enter the warm-up mode one time per day. During this mode, the supply fan starts and the variable frequency drive for the supply air fan is at its minimum setting. The relief and outside air dampers are set to allow 100% return air and no outside air, with the cooling coil valve closed. Over a period of ten minutes, the variable speed drive increases to its maximum speed, while the heating coil valve modulates to maintain the supply air temperature set point from the discharge air sensor, which is located downstream of the heating coil but before the cooling coil. If the outside air temperature is less than 35°F the discharge air temperature set point is 55°F. If during the warm-up mode, the schedule change from unoccupied to occupied, the outdoor air damper will modulate open to its minimum position over a period of at least five minutes.

Like the warm-up mode, the cool-down mode can only happen one time per day. When the cool-down mode is entered, the supply fan starts. The heating coil valve, mixing dampers, and cooling coil valve modulate without overlap in order to satisfy the space set point. To further explain, first the heating coil valve is modulated to its off position. If the set point is not met, and the outdoor dry bulb temperature is below the return air temperature then the outside air damper opens. The outdoor air damper is modulated to try to maintain the set point. If the outdoor air damper is opened to 100% and the space set point is still not met, or if the outdoor air temperature is greater than the return air temperature, the cooling coil valve is modulated to maintain the space set point. If at any time the schedule changes from unoccupied to occupied, the outside air damper is opened to its minimum position unless it is in the economizer operation.

In the occupied mode, the supply fan starts, or continues to run if the unit was previously in the warm-up or cool-down mode. If the outside air dry bulb temperature is above the return air temperature, the mixing dampers are placed in the minimum outdoor air setting. The heating and cooling coil valves then modulate without overlapping to maintain the space set point. Modulating without overlap means that only one valve is open or modulating at a time. The heating coil valve must fully closed before the cooling coil valve will open, as the cooling coil valve must be closed before the heating coil valve can open. If the outdoor air dry bulb temperature is below the return air temperature, the rooftop unit will use the economizing mode and will allow the mixing dampers, heating coil valve, and cooling coil valve to modulate without overlap to satisfy the space set point, without allowing the mixed air temperature to fall below 48°F. The cooling coil valve will not open until the mixing dampers are in 100% outdoor air operation and the space set point is not met. The heating coil valve will not open until the mixing dampers are at minimum settings and either the space set point is not met, or the mixed air temperature is below 48°F

During the unoccupied (normal off) mode, the supply fan is off, the heating and cooling coil valves are closed and the outdoor air dampers close to prevent any outdoor air from entering.

When the space temperature falls below 65°F during an unoccupied schedule, the night heating mode is entered. In this situation, the cooling coil valve is closed along with the damper for outside air. The supply fan starts with the variable speed drive at its minimum setting with the heating coil valve open to maintain the space set point temperature of 65°F.

Night cooling is possible during the unoccupied schedule when the space rises above the set point of 85°F. The heating coil valve remains closed and the supply fan starts with its variable speed drive at its minimum setting. If the outside air dry bulb temperature is above the return air temperature, the mixing dampers remain closed to outside air and the cooling coil valve modulates to maintain the space set point. If the outside air dry bulb temperature is below the return air temperature, the outside air damper modulates to maintain the space set point. If the outside air damper is modulated to fully open economizing and the space set point is still not satisfied, the cooling coil valve will modulate to satisfy the space set point.

The supply fan variable frequency drive will modulate to maintain a static pressure of 1.5 in. wg. two thirds of the way downstream of the longest duct run. The return fan variable frequency drive will modulate to maintain a constant space static pressure.

Constant Volume Rooftop Unit 3

Rooftop unit 3 operates in the same way as the variable volume rooftop units however, there is no variable frequency drive on the supply or return fan. Occupied, warm-up, cool-down, unoccupied, night heating, and night cooling modes are all utilized as well as full air side economizing.

Freeze Protection Pumps 1-10

The freeze protection pump for a rooftop unit will start when the outdoor air temperature is less than 35°F. When the outdoor air temperature rises above 35°F the pump will stop.

Exhaust Fans 3, 5, 9, 10, 12, 13, 15-18, 20, 21

The exhaust fans located in the toilet and loading dock areas start and run constantly when the areas served are in the occupied mode. Then fans stop when the mode changes to unoccupied.

Exhaust Fans 1, 4, 7, 8, 19

The fans will start when the space temperature rises above 80°F. When the fan is started, the exhaust and transfer dampers associated with the fan also open. When the space temperature is satisfied the dampers close and the fan is turned off.

Exhaust Fans 2, 6, 11, 14

The fans will run continuously in the electric rooms.

Data and Electric Room Air Condition Units 1-8

When the space set point of 78°F is not satisfied, the AC unit on call for cooling will turn on. The corresponding air-cooled condenser (ACCU) is hard wired to the AC unit and turns on when the AC unit is started. If the AC unit is unable to maintain the space set point, the variable air volume box will open from it normally closed position to supply air from the rooftop unit to satisfy the space set point. When the space set point is satisfied the VAV box closes allows the AC unit to handle the cooling. If the split system (AC-ACCU) fails, the rooftop unit serving the space is turned on if it is not already running.

9.0 Critique of System

Overall, I think the mechanical system selected for Straumann USA is a very reasonable one. Since building chilled water, and steam were available, it makes sense that both would be utilized for the coils of the air handling units. Also, using fintube radiators along the perimeter of the building to help account for skin losses was certainly a reasonable decision since it was already present in some locations of the building and would only need to be added to areas where it was not present. I think including airside economizing with the rooftop units was a very good decision. It would be my assumption, that it could be utilized for a significant portion of the year in Andover, MA.

There was no annual energy estimate requested by the owner during the design, so one was not performed. I would be curious as whether or not another system might be able to save some energy over the course of a year. Two large fans were provided with each rooftop unit and account for nearly one third of the yearly HVAC costs. I think it would be interesting to explore a DOAS system with either chilled beams or radiant panels to see if any energy savings could be obtained. The fan energy would certainly decrease in fan energy while the pumping energy increased, however the fan energy savings may be enough to validate the use of such a system.

According to the mechanical designer for the Straumann renovation, there was a consideration at one point of also renovating the central heating and cooling plants of the building. The new load on the building was going to certainly max out the existing chilled water plant, and the possibility of renovating was explored but ultimately the existing plant was used. If this had been followed through with, I think some interesting options could have been explored such as direct-fired absorption chillers since natural gas is available on site. Another interesting idea that could have been explored even for the current system with the existing chilled water plant is thermal storage. The cooling load of the building occurs only during the standard working hours of 8-5. If thermal storage was explored, the maximum load on the chiller plant could have been decreased allowing the chillers to run at more constant rate and perhaps save energy.

The system selected for Straumann USA was certainly a logical one, and serves the building well. However, given the time and resources, it would be very interesting to estimate how some other systems would compare.

Appendix A – Mechanical System Schematics

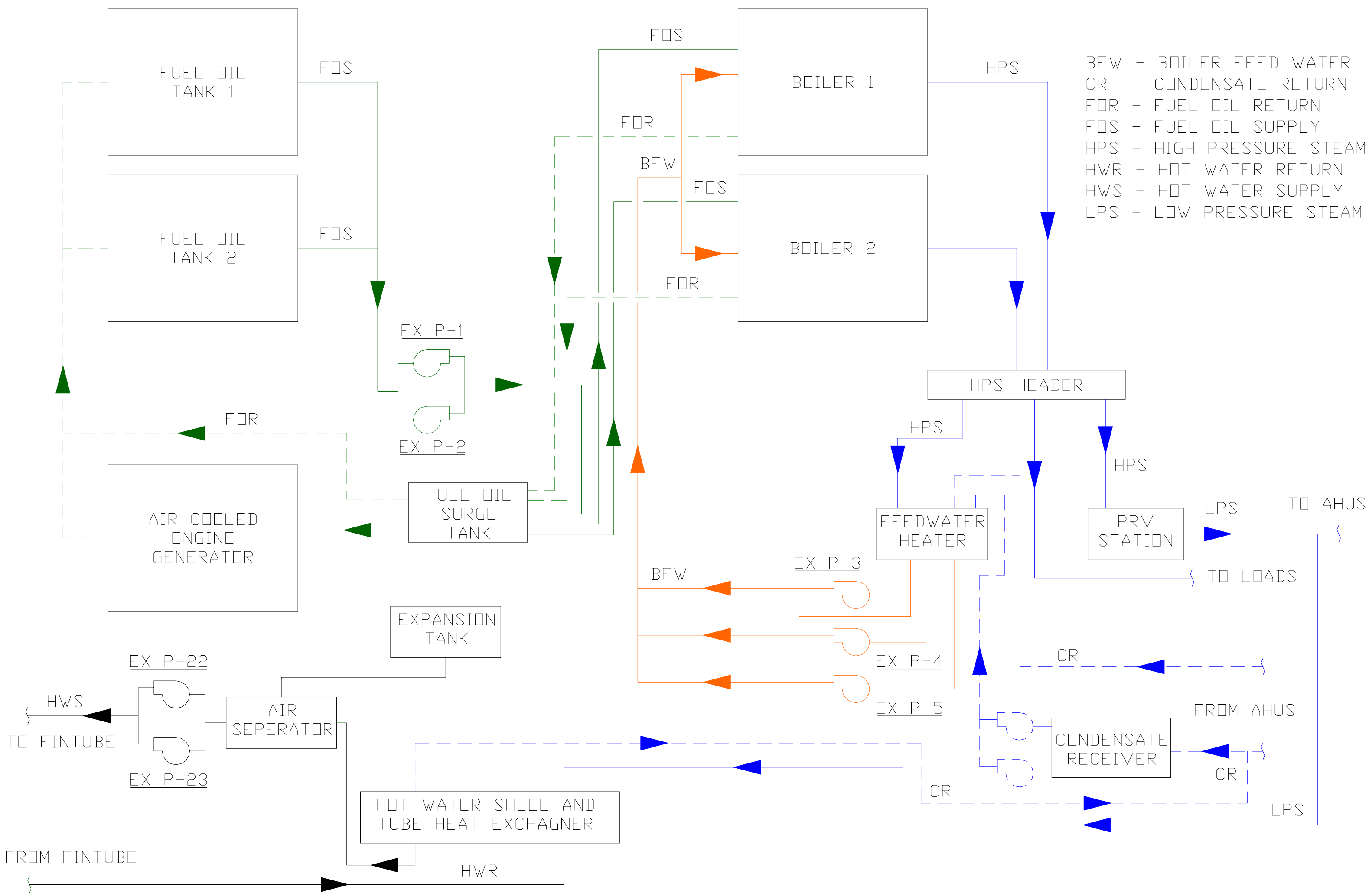

Figure A.1 – Central Heating Plant Schematic

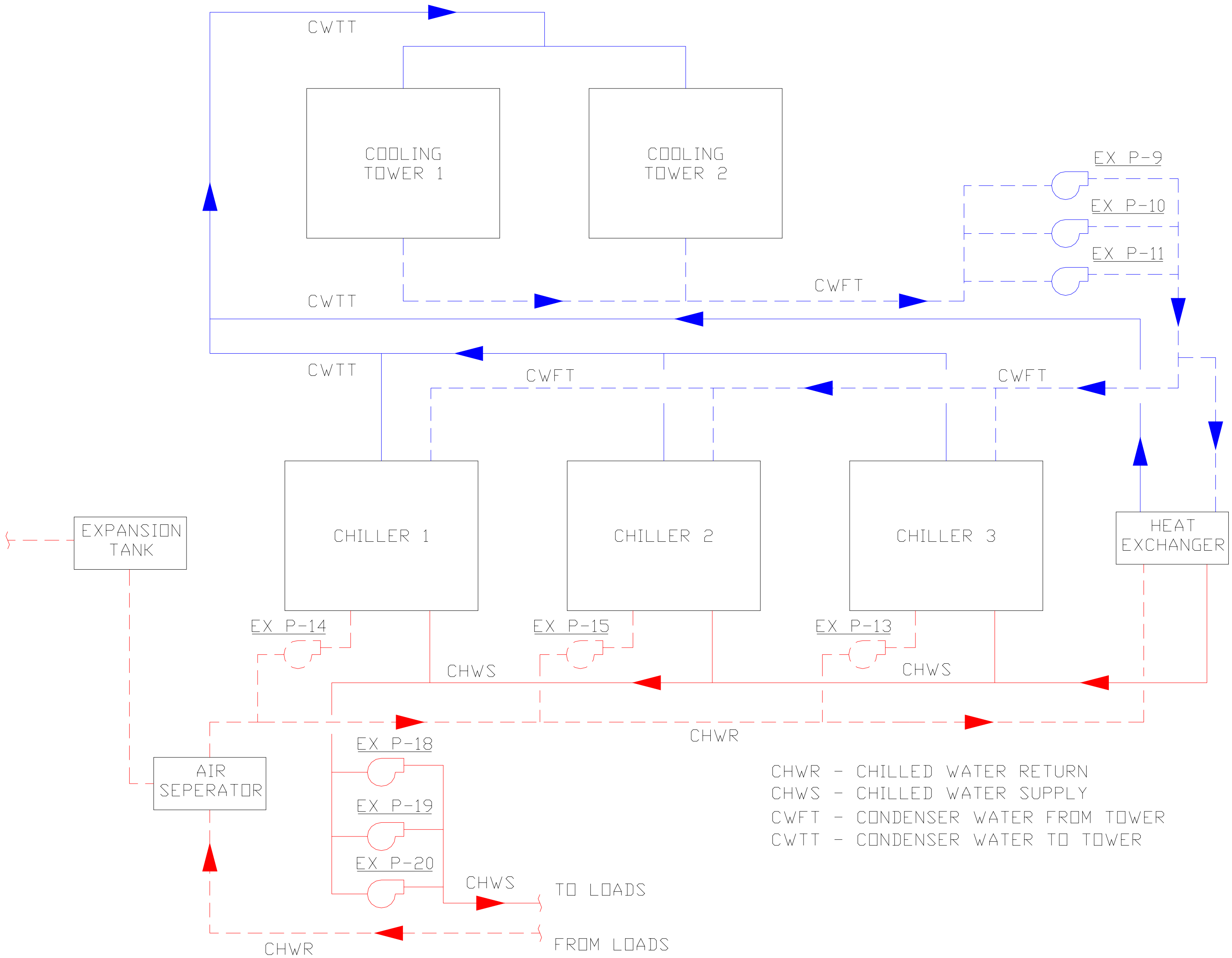

Figure A.2 – Central Cooling Plant Schematic

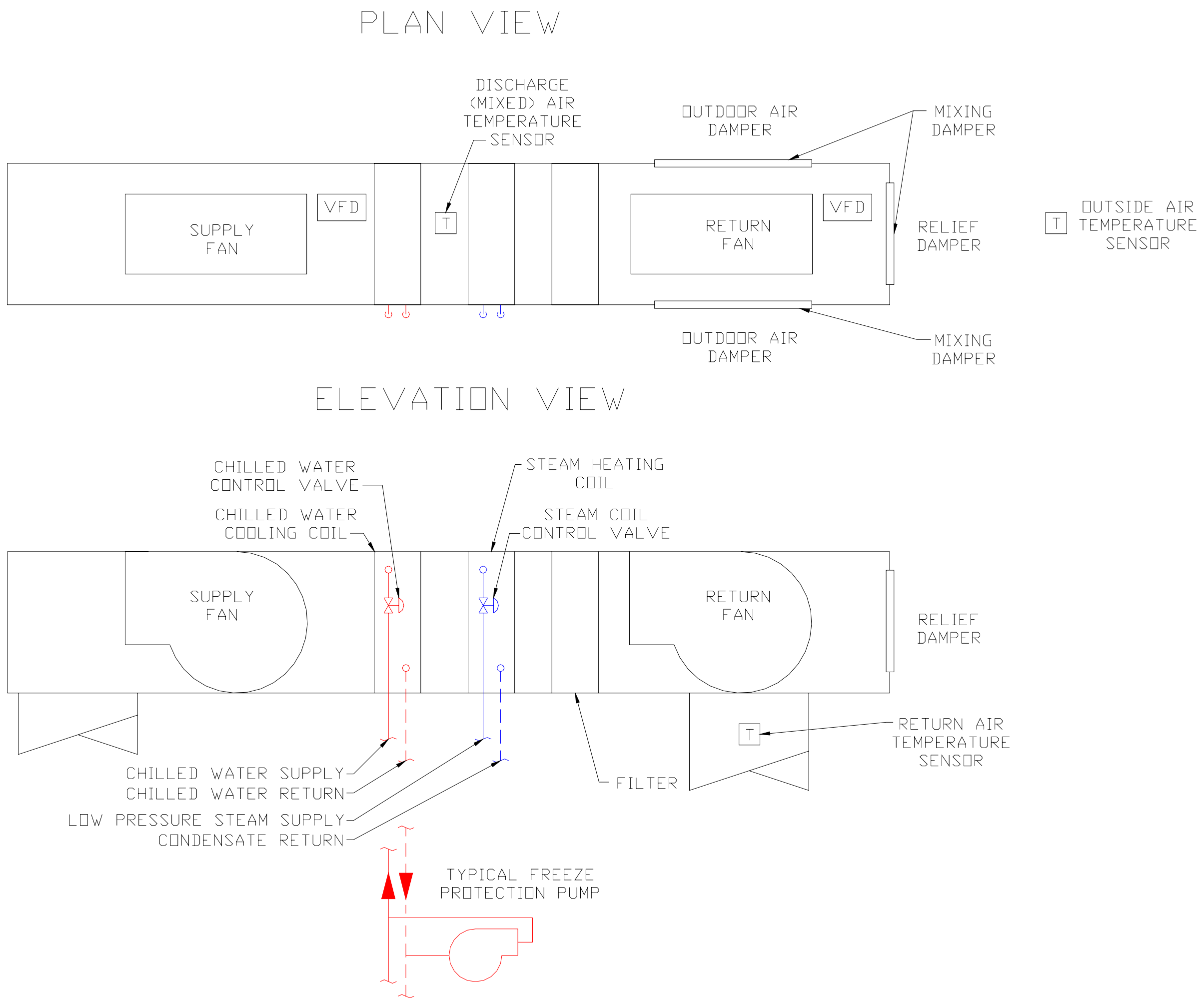

Figure A.3 – Typical Rooftop Unit

Appendix B – New Mechanical Equipment

The following equipment was included in the Straumann USA renovation project.

Rooftop Units

Unit	Supply Fan					
	CFM	Drive	Type	HP	TOT SP	Var Freq Drive
RTU-1	33,000	Belt	Air Foil	50	5.52	Yes
RTU-2	33,000	Belt	Air Foil	50	5.52	Yes
RTU-3	6,400	Belt	Air Foil	5	2.47	No
RTU-4	33,000	Belt	Air Foil	50	5.52	Yes
RTU-5	21,000	Belt	Air Foil	30	5.89	Yes
RTU-6	21,000	Belt	Air Foil	30	5.89	Yes
RTU-7	33,000	Belt	Air Foil	50	5.52	Yes
RTU-8	33,000	Belt	Air Foil	50	5.52	Yes
RTU-9	33,000	Belt	Air Foil	50	5.52	Yes
RTU-10	33,000	Belt	Air Foil	50	5.52	Yes

Unit	Return Fan					
	CFM	Drive	Type	HP	TOT SP	Var Freq Drive
RTU-1	26,500	Belt	Air Foil	15	1.5	Yes
RTU-2	26,500	Belt	Air Foil	15	1.5	Yes
RTU-3	6,400	Belt	Air Foil	3	1	No
RTU-4	26,500	Belt	Air Foil	15	1.5	Yes
RTU-5	21,000	Belt	Air Foil	10	1.5	Yes
RTU-6	21,000	Belt	Air Foil	10	1.5	Yes
RTU-7	26,500	Belt	Air Foil	15	1.5	Yes
RTU-8	26,500	Belt	Air Foil	15	1.5	Yes
RTU-9	26,500	Belt	Air Foil	15	1.5	Yes
RTU-10	26,500	Belt	Air Foil	15	1.5	Yes

Steam Heating Coil				
Unit	EAT DB	LAT DB	MBH TOT	Lbs/Hr
RTU-1	57.59	102.8	1333	1400
RTU-2	57.59	102.8	1333	1400
RTU-3	10	97.5	615	637
RTU-4	57.59	102.8	1333	1400
RTU-5	-	-	-	-
RTU-6	-	-	-	-
RTU-7	57.59	102.8	1333	1400
RTU-8	57.59	102.8	1333	1400
RTU-9	57.59	102.8	1333	1400
RTU-10	57.59	102.8	1333	1400

Air Conditioning Units															
Unit	CFM	Fan hp	Cooling Coil Data				Compressor			Electric Coil		Humidifier		Elect Char	
			EAT		Total	Sensible	Type	Quanity	hp each	kW	steps of control	Capacity	kW	V-PH-HZ	FLA
			DB	WB	MBH	MBH									
AC-1	5650	2.0	72	60	118.3	108.7	scroll	2	-	15	3	17.4	7.9	3-60-480	36.5
AC-2	5650	2.0	72	60	118.3	108.7	scroll	2	-	15	3	17.4	7.9	3-60-480	36.5
AC-3	1250	0.5	75	61	34.7	27.5	scroll	1	3	7.3	3	4.3	1.5	3-60-480	13.2
AC-4	5650	2.0	72	60	118.3	108.7	scroll	2	-	15	3	17.4	7.9	3-60-480	36.5
AC-5	5650	2.0	72	60	118.3	108.7	scroll	2	-	15	3	17.4	7.9	3-60-480	36.5
AC-6	1250	0.5	75	61	34.7	27.5	scroll	1	3	7.3	3	4.3	1.5	3-60-480	13.2
AC-7	1250	0.5	75	61	34.7	27.5	scroll	1	3	7.3	3	4.3	1.5	3-60-480	13.2
AC-8	1250	0.5	75	61	34.7	27.5	scroll	1	3	7.3	3	4.3	1.5	3-60-480	13.2

Data and Electric Room AC Units

Chilled Water Cooling Coil										
Unit	EAT		LAT		SP IN	EWT	LWT	WPD	MBH TOT	MBH SENS
	DB	WB	DB	WB						
RTU-1	78.4	64.7	51.1	50.9	1.16	45	55.2	11.8	1325	984.9
RTU-2	78.4	64.7	51.1	50.9	1.16	45	55.2	11.8	1325	984.9
RTU-3	99.3	75	55.1	54.3	0.25	45	54.9	9.8	447	310
RTU-4	78.4	64.7	51.1	50.9	1.16	45	55.2	11.8	1325	984.9
RTU-5	80	67	50.5	50.3	1.38	45	55	11.15	1044	677
RTU-6	80	67	50.5	50.3	1.38	45	63.2	11.15	1044	677
RTU-7	78.4	64.7	51.1	50.9	1.16	45	55.2	11.8	1325	984.9
RTU-8	78.4	64.7	51.1	50.9	1.16	45	55.2	11.8	1325	984.9
RTU-9	78.4	64.7	51.1	50.9	1.16	45	55.2	11.8	1325	984.9
RTU-10	78.4	64.7	51.1	50.9	1.16	45	55.2	11.8	1325	984.9

Air Cooled Condensing Units

Air Cooled Condensing Units					
Unit	Condenser Fans			Elect Char	
	Quantity	hp each	CFM	V-PH-HZ	FLA
ACCU-1	2	-	12000	3-60-480	4.2
ACCU-2	2	-	12000	3-60-480	4.2
ACCU-3	1	0.5	3000	3-60-480	7.4
ACCU-4	2	-	12000	3-60-480	4.2
ACCU-5	2	-	12000	3-60-480	4.2
ACCU-6	1	0.5	3000	3-60-480	7.4
ACCU-7	1	0.5	3000	3-60-480	7.4
ACCU-8	1	0.5	3000	3-60-480	7.4

Freeze Protection Pumps

Chilled Water Freeze Protection Pumps								
Unit	Type	Motor hp	RPM	VFD	GPM	Eff	Feet Head	Impeller Size
P-1	In-Line	3	1750	No	60	52.02	60	8.125
P-2	In-Line	3	1750	No	60	52.02	60	8.125
P-3	In-Line	3	1750	No	60	52.02	60	8.125
P-4	In-Line	3	1750	No	60	52.02	60	8.125
P-5	In-Line	3	1750	No	60	52.02	60	8.125
P-6	In-Line	3	1750	No	60	52.02	60	8.125
P-7	In-Line	3	1750	No	60	52.02	60	8.125
P-8	In-Line	3	1750	No	60	52.02	60	8.125
P-9	In-Line	3	1750	No	60	52.02	60	8.125
P-10	In-Line	3	1750	No	60	52.02	60	8.125

Appendix C – Existing Mechanical Equipment

The following equipment was sized and selected based on the load prior to the renovation of Straumann USA portion of the building and may no longer be running at the selected conditions.

Cooling Towers

Cooling Towers					
Symbol	Tons	EWT	LWT	Fan Speed Control	WB
EX CT-1	680	94.5	85	2 speed	78
EX CT-2	750	94.5	85	2 speed	78

Pumps

Existing Pumps					
Unit	System Served	RPM	VFD	GPM	Feet Head
EX P-1	Fuel Oil	1800	No	5	50 PSIG
EX P-2	Fuel Oil	1800	No	5	50 PSIG
EX P-3	Boiler Feedwater	3550	No	30	N/A
EX P-4	Boiler Feedwater	3550	No	30	N/A
EX P-5	Boiler Feedwater	3550	No	30	N/A
EX P-9	Condeser Water	1800	No	1550	85
EX P-10	Condeser Water	1800	No	1550	85
EX P-11	Condeser Water	1800	No	1550	85
EX P-14	Chilled Water Chiller	1800	No	900	30
EX P-15	Chilled Water Chiller	1800	No	900	30
EX P-16	Chilled Water Chiller	1800	No	900	30
EX P-18	Chilled Water Building Loop	1800	No	900	175
EX P-19	Chilled Water Building Loop	1800	No	900	175
EX P-20	Chilled Water Building Loop	1800	No	900	175
EX P-22	Heating Hot Water	1800	No	190	125
EX P-23	Heating Hot Water	1800	No	190	125

Hot Water Heat Exchangers

Not Available

Waterside Free Cooling Plate Heat Exchanger

Not Available

Chillers

Electric Centrifugal Chillers												
Symbol	nominal tons	evaporator				condensor						
		EWT (F)	LWT (F)	GPM	ΔP (ft)	EWT (F)	LWT (F)	GPM	ΔP (ft)	kW	compressor	refrigerant
EX Chiller 2	750	56	42	1284	13.8	85	94.5	2250	26	272	hermetic centrifgual	R-134a
EX Chiller 3	500	56	42	856	N/A	85	94.5	1500	N/A	192	hermetic centrifgual	R-134a
EX Chiller 4	350	56	42	600	N/A	85	94.5	1050	N/A	157	hermetic centrifgual	R-134a

Boilers

Boilers								
Symbol	Input				Output		Max Pressure (PSI)	Blower Motor hp
	BTU	Gas (Therm/hr)	Light Oil (gph)	Heavy Oil (gph)	Steam (lb/hr)	BTU		
EX B-1	14,645,000	146.5	104.5	97.5	12075	11716	200	15
EX B-2	14,645,000	146.5	104.5	97.5	12075	11716	200	15

Appendix D – Schedules and Utility Rates

Auditorium Schedule	- Full load during regular business hours 8am -5pm, three days a week. - Zero load two days a week during business hours. - Zero load during non-business hours, holidays, and weekends.
Office/Manufacturing Schedule	- Full load during regular business hours 8am - 5pm every week day, - Zero load during non-business hours, weekends, and holidays.
Data Cooling Schedules (AC's)	- Full load 24 hours a day, 365 days a year
Data Cooling Fans	- Occupied 24 hours a day, 365 days a year
RTU Fan Schedules	- Full load during regular business hours 8am - 5pm every week day, - Zero load during non-business hours, weekends, and holidays.
Electric Rates	- High: July - October - Medium: January, May, December - Low: February, March, April
Fuel Rates	- High: January August December - Medium: February - April, June, July, September - November - Low: May

Electric and fuel rates were based on information from electric and fuel rates from the past year. The rate were grouped into three categories (high, medium, and low) and then were averaged for each category in order to make a rate structure that could be implemented into HAP. Table D.1 and D.3 provide the monthly rates used to formulate high medium and low rates to apply to electricity and fuel consumption respectively. Tables D.2 and D.3 provide the scheduled rate applied to each category for electricity and fuel.

The monthly consumption and cost data is for the entire 100 Minuteman facility. Struaman USA occupies approximately half of the building and would therefore be responsible for half of the energy costs.

Electric Rates				
	Monthly Electric Cost per kWh	Monthly kWh Consumption	Monthly Cost	Assigned Schedule
Jan	0.1672	658000	$110,018	mid
Feb	0.1973	630000	$124,299	high
Mar	0.198	658300	$130,343	high
April	0.1774	669800	$118,823	high
May	0.151	584600	$88,275	mid
June	0.1211	729800	$88,379	low
July	0.1301	1229100	$159,906	low
August	0.1341	544400	$73,004	low
Sept	0.1325	831600	$110,187	low
Oct	0.1099	654200	$71,897	low
Nov	0.1134	617000	$69,968	low
Dec	0.1492	671600	$100,203	mid
Yearly Totals		8478400	$1,245,300	

Table D.1 Electric Rates

Electric Schedule	
0.124	low
0.156	mid
0.191	high

Table D.2 Electric Schedules

Gas Rates				
	Cost per Therm	Monthly Therm Consumption	Monthly Cost	Assigned Schedule
Jan	1.6803	19671	$33,053	high
Feb	1.5686	16125	$25,294	mid
Mar	1.5267	16574	$25,304	mid
April	1.5703	6154	$9,664	mid
May	1.3863	4181	$5,796	low
June	1.523	1436	$2,187	mid
July	1.5172	681	$1,033	mid
August	1.6792	1302	$2,186	high
Sept	1.6127	1107	$1,785	mid
Oct	1.4964	3731	$5,583	mid
Nov	1.6266	10630	$17,291	mid
Dec	1.7743	22766	$40,394	high
Yearly Totals		104358	$169,569	

Table D.3 Fuel Rates

Fuel Schedule	
low	1.386
mid	1.555
high	1.711

Table D.4 Fuel Schedules

Appendix E – Energy and Cost Analysis

Figure E.1 Percentage of Energy Cost per System Component

Component	Cost	Cost per Square Foot	Percent of Total
Air System Fans	$72,647	$0.581	10.2%
Cooling	$48,432	$0.387	6.8%
Heating	$42,958	$0.343	6.0%
Pumps	$19,052	$0.152	2.7%
Cooling Tower Fans	$38,952	$0.311	5.5%
HVAC Sub-Total	**$222,041**	**$1.775**	**31.1%**
Lights	$68,570	$0.548	9.6%
Electric Equipment	$423,845	$3.388	59.3%
Misc. Electric	$0	$0.000	0.0%
Misc. Fuel Use	$0	$0.000	0.0%
Non-HVAC Sub-Total	**$492,415**	**$3.936**	**68.9%**
Grand Total	**$714,456**	**$5.711**	**100.0%**

Table E.1 Energy Cost per System Component

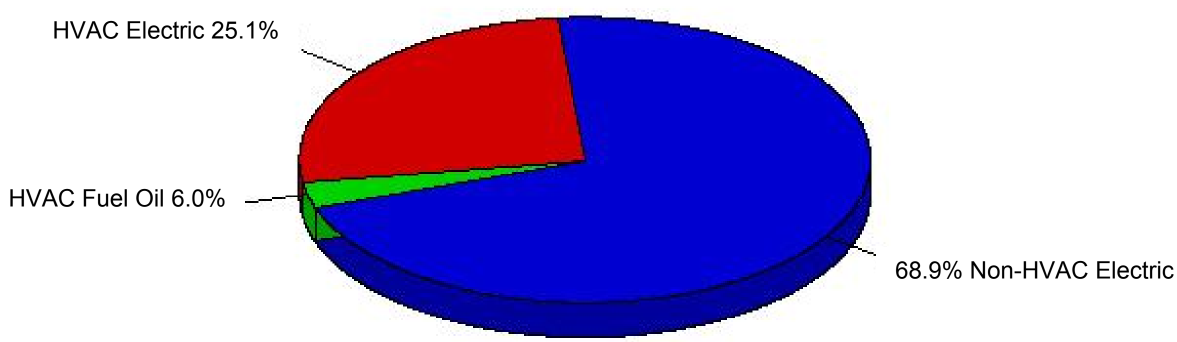

Figure E.2 Percentage of Annual Energy Cost

Component	Cost	Cost per Square Foot	Percent of Total
HVAC	**$222,041**	**$1.775**	**31.1%**
Non-HVAC	**$492,415**	**$3.936**	**68.9%**
Grand Total	**$714,456**	**$5.711**	**100.0%**

Table E.2 Annual HVAC/Non-HVAC Energy Costs

Component	Cost	Cost per Square Foot	Percent of Total
HVAC Components			
Electric	$179,083	$1.431	25.1%
Fuel Oil	$42,958	$0.343	6.0%
HVAC Subtotal	**$222,041**	**$1.775**	**31.1%**
Non-HVAC Components			
Electric	$492,415	$3.936	68.9%
Non-HVAC Subtotal	**$492,415**	**$3.936**	**68.9%**
Total	**$714,456**	**$5.711**	**100.0%**

Table E.3 Annual Costs per Fuel Type