Technical Assignment 3

Mechanical Systems Existing Conditions Evaluation

The Milton Hershey School New Supply Center
Hershey, Pennsylvania

Prepared for:
William P. Bahnfleth, Ph.D., P.E., Professor
Department of Architectural Engineering
The Pennsylvania State University

Prepared by:
Justin Bem
Mechanical Option
November 21st, 2006

TABLE OF CONTENTS

1.0 EXECUTIVE SUMMARY

The Milton Hershey School New Supply Center is a very wide and long single story 110,000 square foot building. Analyzing the existing mechanical systems proves that the design is very practical and energy conscience for this application. However, there are specific features to the HVAC system that have potential for improvements. This document examines the existing mechanical systems and provides in site to areas that need addressed for possible re-design or modifications.

The mechanical design engineer for the supply center is H.F. Lenz Company. The engineers set design objectives based on cost and energy saving criteria. Designing the mechanical systems to reach the overall project goal of LEED Certification is the basis of the majority of the engineering tactics used. Integration of other building systems with the HVAC systems is also another very important project goal. Incorporating a condensate water system for rejection of the heat produced from the supply center's large walk-in freezers with the chilled water plant helps reduce the amount of sensible cooling required in those spaces. Using the same boiler plant for HVAC heating for service hot water heating and steam production for process loads also is a major project goal.

Carefully planed control logic is also a goal of the project and results in optimal energy efficiency for the HVAC system. Using temperature, humidity, enthalpy, and flow measuring devices with direct digital controls (DDC) as the energy management control system, the HVAC system proves to not only save energy, but also meet the most basic requirement of producing comfortable indoor environments.

Even with proper controlling of the HVAC system, alterations or re-designs of the HVAC system may provide cost and energy saving benefits. Areas of the mechanical systems, such as the heat rejection system for the walk-in freezers, need addressed. Incorporating heat recovery systems that will not waste building generated heat has the potential to reduce operating cost. Revaluating the ventilation air systems may mean a significant reduction in the amount of air handling units required for the supply center. First cost savings will result from less material, but energy savings are also a possibility when dedicated outdoor air units are used in replacement of the designed VAV systems. Overall, the design HVAC system is adequate and energy conscience, but there is potential for improvements.

2.0 BUILDING DESIGN BACKGROUND

The Milton Hershey School New Supply Center is a single story 110,000 square foot building with four elevated mechanical mezzanine rooms and contains a variety of spaces. The north and northwest sections of the building consists of general office spaces and conference rooms. Located in the center of the building is the food distribution center for the Milton Hershey School. This area contains large freezers, refrigerators, and temperature controlled storage areas, fifteen in all, totaling to 13,600 square feet to go along with its central food preparation spaces.

Aside from the food production section of the building, the New Supply Center also includes a central mail distribution center for the school and a clothing store with an alterations work area. Complementing the four mechanical mezzanine rooms that house the air handling units, a boiler and chiller plant is located on the north side of the building. The east side is mostly loading docks for deliveries, and the south side accommodates a variety of storage space. There are also two data rooms located in the center of the floor plan. Figure one, shown below, gives a breakdown of the space's location in the building as well as the portion of area each occupancy type consumes.

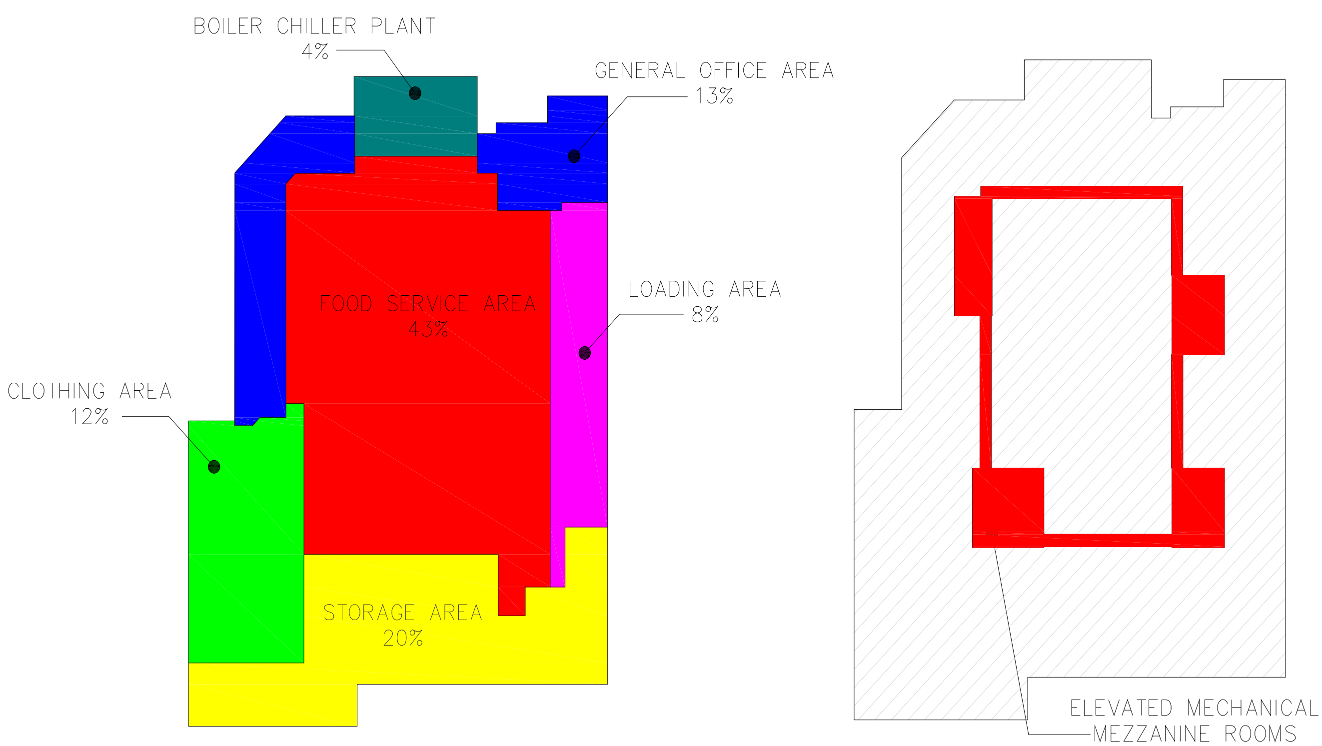


Figure 1 Space relationship and area breakdown

3.0 MECHANICAL SYSTEMS DESIGN OBJECTIVES & REQUIREMENTS

The design of the HVAC systems for the Milton Hershey School New Supply Center is based on cost and energy saving criteria. The MEP engineer for the project is H.F. Lenz Company. H.F. Lenz Company's most general goals of the HVAC system design are:

- The complete HVAC system shall meet the requirements of the 2003 International Building Code, ASHRAE 62.1-2004 Standard for Indoor Air Quality, and all applicable National Fire Protection Association Standards.
- The system will be made as energy efficient as practical in accordance with LEED design principles. Variable volume hydronic pumping and air systems are used where possible.
- The amount of rooftop HVAC equipment is minimized as much as possible to ensure good access for maintenance and to maximize equipment life.

Along with these goals, the system is designed to integrate other building systems. The central kitchen of the supply center, as explained above, consists of large walk-in freezers and coolers. The walk-in coolers are cooled by fan coil units that are supplied 20˚F water from brine chillers. The walk-in freezers are refrigerated by split system DX units using water cooled condensing units. These condensers reject heat to a condenser water loop that is picked up by the HVAC system. A description of the chillers and all other HVAC equipment are located in the following section. Details on the chilled water system are also located in the next section.

Integration of other building systems does not just apply to the chilled water system. The HVAC design objectives also include integrating the process heating loads, dishwashers and steam kettles, with the buildings boiler plant. The buildings boiler plant also provides domestic hot water heating along with HVAC hot water.

Elaborate kitchen exhaust systems are also required in the project's goals. Incorporating energy saving techniques with the exhaust hoods such as variable speed fans help reduce energy cost. Due to an extensive amount of exhaust air in the food service and loading dock sections of the building, the air side mechanical systems include ventilation make-up air units.

The HVAC system, as a whole, generally consists of centrally ducted air handling unit systems. Chilled and hot water systems from the central plant are piped to cooling and heating coils in the AHUs and terminal equipment that both provide the means of conditioning the spaces.

3.1 DESIGN INDOOR AND OUTDOOR CONDITIONS

The design indoor conditions are developed by H.F. Lenz Company and the Milton Hershey School personnel. Collaboration between the two parties results in design heating dry bulb temperatures, cooling dry bulb temperatures, and maximum relative humidity levels for each space. Appendix A includes charts summarizing the design indoor conditions for all occupancy types found in the supply center.

The design outdoor conditions used in building simulation models, calculating thermal loads on spaces, and sizing HVAC equipment are found in the ASHRAE Handbook of Fundamentals 2005 (ASHRAE 2005). Since temperature data for Hershey Pennsylvania is not available in the handbook, the design temperatures are from Harrisburg Pennsylvania. Appendix A includes a summary of the design outdoor conditions used in the design of the supply center's HVAC systems.

4.0 MECHANICAL SYSTEMS SUMMARY

This section breaks down all of the HVAC systems located in the Milton Hershey School New Supply Center. Each system's equipment characteristics are summarized.

4.1 AIR SIDE MECHANICAL SYSTEM

The air side mechanical system for the supply center uses fourteen air handling units. Four of the AHUs are part of multiple zone VAV systems. These air handling units serve offices, dining areas, clothing display and alterations areas, and staff spaces. The air is distributed to these spaces through VAV terminal units with hot water re-heat coils. The perimeter spaces also include fin-tube radiation systems for winter heating. The four VAV air handling units consists of a supply and return fans, 30% and 85% efficient filters, hot water pre-heat and re-heat coils, and a cooling coil.

The remaining ten air handling units are single zone spaces that are either part of VAV or CAV systems. However, since they are single zone units, VAV boxes are not used for air distribution to the spaces. All ten AHUs consists of 30% and 85% efficient filters, hot water pre-heat and re-heat coils, cooling coils, and supply fans.

Six of the ten single zone AHUs are part of constant volume systems. These units provide make up air for spaces requiring excessive amounts of exhaust (kitchens spaces, loading docks, recycling room). Even though the units operate at 100% outdoor air when the spaces are in operation, the AHUs also have the ability to return air during unoccupied times. The final four single zone air handling units not mentioned are part of VAV systems. These units serve dry storage and clothing warehouse areas. The AHUs vary the volume of air supplied at the supply fan via variable frequency drives.

The two data rooms located in the center of the building incorporate two systems to provide cooling year round. The data rooms are served by VAV systems, however, when the central VAV air handling unit is operating at an unoccupied mode, ductless split system air conditioners are used to handle the cooling loads.

Detailed characteristics on each individual AHU and the split system air conditioner are found in Appendix B.

The following figure is a representation of the HVAC air side system zoning plan.

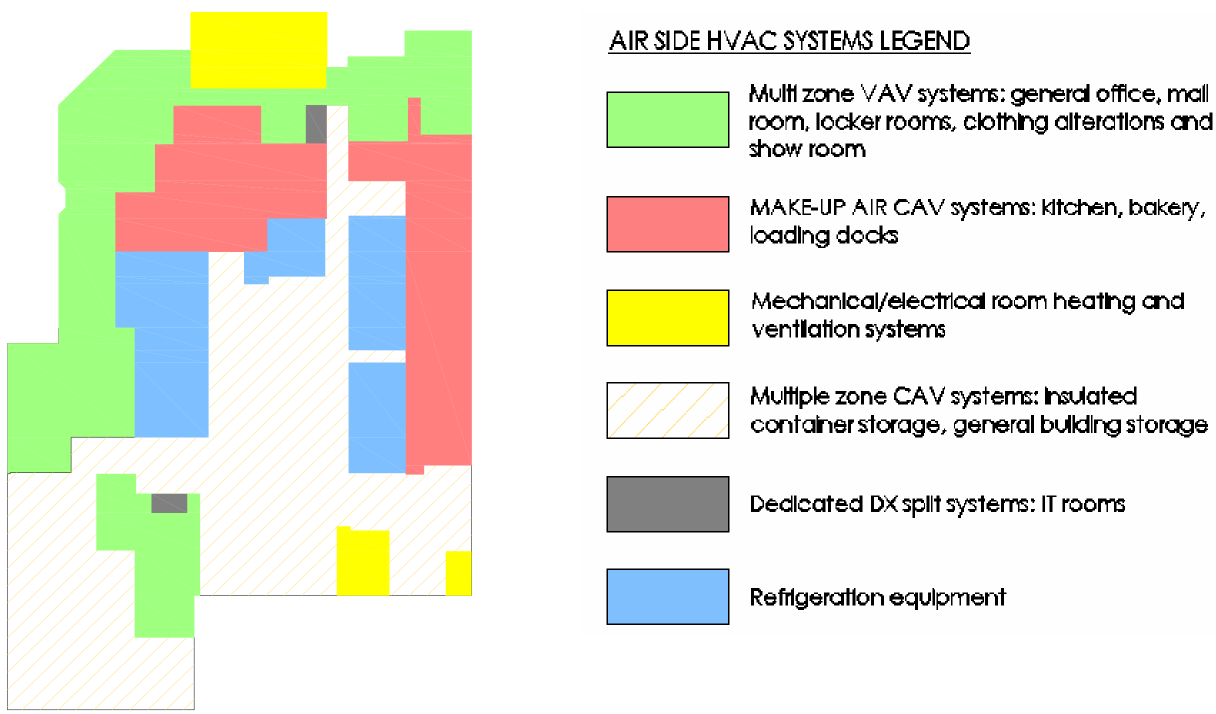


Figure 2 Air side mechanical systems zoning plan

4.2 CHILLED WATER SYSTEM

The chiller plant of the supply center consists of two (one duty one standby) 270 ton electric driven centrifugal water cooled chillers that produce 45°F water. These chillers are used to meet the normal HVAC building loads. Also included in the chilled water system are two (one duty one standby) water cooled brine chillers that produce 20°F water. These chillers service fan coil units located in walk-in-coolers and refrigerated rooms year round. The two sets of chillers in the plant are interconnected in that they all have the capabilities to produce either 45°F or 20°F water for emergency purposes. All four of the chillers operate with R-134a refrigerant and the entire chilled water system (both the HVAC and brine loops) are provided with a 35% propylene glycol solution.

As mentioned above, the 45°F water loop also serves two plate frame heat exchangers to pick up the rejected heat from the walk-in freezers. The rejected heat from the freezers is distributed to a condenser water loop. This water loop is then cooled by the chilled water system before returning to the freezer's condensing units.

The HVAC chilled water loop incorporates a primary-secondary pumping system. Three primary pumps are located in the chilled water plant and are of a duty-duty-standby configuration. Two secondary pumps with VFDs distribute chilled water to the building loop. A similar pumping configuration is used for the brine loop, however, only four pumps are needed (2 primary, 2 secondary).

The chiller room includes a refrigerant leak detection and exhaust system that complies with ASHRAE Standard 15. Detailed characteristics of each chilled water component are found in Appendix B.

4.3 CONDENSER WATER SYSTEM

The condenser water system for the chilled water plant includes two induced draft cooling towers for the heat rejection equipment. These service all four of the chiller's condensers. The walk-in freezer's condenser water loop also utilizes the chiller plant's condenser water system. As stated above, the freezer's rejected heat is handled by the HVAC chilled water loop. This operation occurs in the summer months, or when the ambient outdoor temperature is above 50°F. The freezer's condenser water loop bypasses the plate frame heat exchangers that are served by 45°F chilled water and enters a third plate frame heat exchanger served directly by 60°F condensing water from the cooling tower. This process is used for water side "free" cooling in seasons where the outdoor temperature is below 50°F.

The chilled water flow diagram, located in Appendix C, show the three main water loops (condenser water, chilled water, and freezer condenser water). Design temperatures at which each water loop operates is also located on the flow diagram. The freezer's condenser water loop shows the 3-way valve that is used to bypass the chilled water heat exchangers when the system is operating in "free" cooling mode.

4.4 STEAM BOILER AND HOT WATER SYSTEMS

The boiler plant in the supply center consists of three natural gas-fired fire tube boilers. The two larger boilers, 200 BHP, service the building HVAC heating and domestic water heating loads. The third smaller boiler, 125 BHP, meets the kitchen equipment hot water demands. The boilers also incorporate flue gas recirculation to lower pollution levels. NO_X levels are held to 30 parts per million due to this configuration.

A combination deaerator and condensate storage tank is used to provide feed water to the boilers. Three active feed water pumps operate continuously with feed water valves located on the boilers. The feed water valves are controlled by level sensors so that minimum water levels are met to avoid potential hazards.

As stated above, the steam boilers produce 40 psig steam to service kitchen equipment loads, such as dishwashers. However, hot water for HVAC heating is also produced by these boilers. Hot water is needed to serve fin tube radiators, VAV box reheat coils, and cabinet and horizontal unit heaters. The hot water is produced by conversion of low pressure steam in two (one duty, one standby) shell and tube heat exchangers. Two hot water pumps with VFDs distribute the hot water to the HVAC equipment. Appendix C includes the hot water flow diagram for further illustration of the process.

5.0 MECHANCIAL SYSTEMS CONTROL LOGIC

This section includes summaries of the HVAC control systems of operations. Direct digital controls (DDC) are used as the energy management control system for the HVAC equipment in the supply center.

5.1 TYPICAL SINGLE ZONE AHUs CONTROL LOGIC (AHU-9,12-13, and 14)

AHU-9, 12, 13, and 14 are operated so that in cooling mode, the cooling coil control valve modulates to maintain a fixed supply temperature of 53°F. When the cooling demand decreases, the supply air fans for the AHUs decrease the quantity of air from the maximum scheduled flow rate to the minimum cooling mode airflow rate (50% of the max cfm). Once the minimum airflow rates are met, the cooling coil control valve decreases the amount of chilled water being supplied to maintain desired space temperatures. When there is a demand for heating, the supply fan operates at the maximum airflow, but the outdoor air damper only allows for the minimum required amount of ventilation air to enter the AHU. The chilled water valve is closed in heating mode, and the heating coil control valve modulates to maintain the desired space conditions. The AHU's supply air temperature is monitored so not to fall below 53°F in heating mode.

Controlling the ventilation air for these AHUs relies on the DDC system, the outdoor air dampers, and return air dampers. The supply air quantity is measured using a fan-inlet airflow measuring station. The return air quantity is measured using a duct-mounted airflow measuring station. Minimum outdoor airflow rates are controlled by simultaneously modulation the return damper closed and the outdoor air damper open. All devices used in the controlling process are shown in the overall HVAC schematic in Appendix C.

Economizer modes for the AHUs are activated by enthalpy controlled systems as seen in the overall HVAC schematic. When the outdoor air enthalpy is less than the return air enthalpy and there is a need for cooling in the space, the return/outdoor air dampers modulate to increase the outdoor airflow and maintain the desired space temperature. When the system is operating at 100% outdoor air and there is still a high cooling demand, the coiling coil control valve modulates to maintain the desired space temperatures.

Relative humidity is controlled by the DDC system as well. When space relative humidity rises above the desired upper limit, the AHU operates at the maximum supply air flow rate. The normal temperature control over the cooling coil is overridden, and the control valve is modulated to produce a supply air temperature as low as 51°F to maintain the desired space relative humidity. The reheat coils in the AHUs are then modulated to maintain the desired space dry-bulb temperature.

5.2 VARIABLE VOLUME AHU WITH RETURN FAN CONTROL (AHU-4, 10)

The DDC systems monitors the outdoor air, mixed air, supply air, and return air temperatures and provides a variable supply air temperature as see in the following table.

Table 1 Temperature Reset Schedule

RETURN AIR TEMPERATURE	SUPPLY AIR TEMPERATURE
78°F	51°F
54°F	70°F

A rise in supply air temperature means the hot water supply to the heating coil is reduced first. A further rise in supply air temperature then sees modulation of the return and outdoor air dampers until 100% outdoor air economizer mode is activated. A further rise in supply air temperature requires the DDC system to modulate the cooling coil control valve open. As with the single zone AHUs, economizer mode is activated only when the outdoor air enthalpy is lower than return air enthalpy.

When the outdoor air temperature is above 35°F, the integral face and bypass damper is positioned for full coil flow. The heating coil control valve then modulates to control the discharge air temperature. When the outdoor air temperature is below 35°F, the hot water control valve is positioned for full water flow, and the face and bypass dampers modulate to vary the amount of air passing through the coil.

Since these AHUs are variable volume, the DDC system monitors the supply duct static pressures and varies the supply fan speed to maintain 1.05 inches of water gauge. The DDC systems also measure the supply and return air flow rates. Outdoor airflow rates are measured by duct mounted outdoor air flow measuring stations. As with the single zone AHUs, the return and outdoor air dampers are simultaneously modulated to meet the ventilation requirements. The DDC system then modulates the return fan speed to maintain a return air flow rate equal to the supply air flow rate minus the exhaust air flow rates for proper building pressurization.

The AHUs with relief fans (AHU-3, 11) instead of return fans operate with similar characteristics as AHU-4 and 10 described above, except relief fan speed is modulated by the DDC to maintain building pressurization.

5.3 KITCHEN AND BAKERY AHUs (AHU-1,2, and 8)

The AHUs serving the kitchen and bakery operate with the same temperature control logic as the single zone AHUs. They also include integral face and bypass dampers that operate similarly to the VAV AHUs described above.

The ventilation/make-up air quantities are controlled by having the units operate at 100% outdoor air during occupied modes, and the minimum outdoor air flow rates are 5% of the maximum value during unoccupied modes. Controlling the quantities of return and outdoor air is performed similarly as in the single zone AHUs. However, the pressure differential between the kitchen and bakery with the surrounding corridors is monitored and outdoor air flow rates are increased upon increased kitchen negative deferential pressure beyond the set point of -0.02 inches of water gauge.

5.4 VARIABLE VOLUME BOX CONTROL

Spaces in the supply center served by VAV systems with terminal boxes with reheat include thermostats for controlling the space temperature. The thermostat monitors the space temperature and modulates the VAV box damper. When the VAV box is operating at minimum air flow and there is still a call for heating, the reheat coil control valve modulates the hot water supply.

5.5 45°F CHILLER PLANT CONTROL

The DDC system provides lead/lag control of both chillers, control over the sequencing of all chilled and condenser water pumps, and control over the loading of the chillers. The DDC system enables the chilled water system whenever the outdoor dry bulb temperature is above 51°F for longer than 15 minutes and any single AHU is energized with the chilled water valve positioned to chilled water flow, or whenever the freezer condensate supply and return systems enters a chilled water mode. The chilled water system is disabled whenever the outdoor temperature falls below 48°F for more than 40 minutes and the condensate water supply and return system is operating in economizer mode (water side “free” cooling). The chilled water flow diagram, located in Appendix C, illustrates the 3-way control valve that singles to the chillers that the freezer’s condensate water system is operating in chilled water mode.

The run time of pumps and chillers is logged by the DDC system and the control system automatically provides lead/lag operation for the chillers and lead-lag-standby operation for both the primary chilled water pumps and secondary condenser water pumps. When an accumulated run time of 300 hours is reached for a lead chiller, the control system shall automatically switch the previous lag unit to the current lead unit status until the run hours limit is reached. If a chiller is in operation when the run time limit is reached, the lag unit starts in addition to the lead unit until the lag unit’s supply water temperature has time to respond to the chiller starting. The lead unit, at this point, stops and the lead/lag notation now shifts.

The flow rate through each chiller’s evaporator is measured. Any time the evaporator water flow drops below 50% of the rated flow, the chiller is disabled and an alarm is issued. The DDC system uses the evaporator differential temperature and evaporator water flow to calculate the actual tonnage of refrigeration that is being produced by each chiller. A secondary chilled water flow station determines the total gpm being delivered throughout the secondary (building loop) chilled water piping system. The chilled water flow diagram in Appendix C illustrates these flow measuring devices.

Similar control logic is used in the 20°F brine chiller plant as described above.

5.6 PRIMARY CONDENSER WAER SYSTEM CONTROL (COOLING TOWERS)

The cooling towers each have a single fan motor controlled through a VFD and they operate in a lead/lag notation. The pumping system is primary secondary with the primary pumps circulating water to the cooling towers. The secondary pumps serve the water cooled chillers and the “free” cooling heat exchanger in the winter.

The DDC system monitors the primary condenser water supply temperature. The condenser water system includes an adjustable primary condenser water temperature set point reset configured to the conditions in the following table.

Table 2 Condenser Water Temperature Set Point Reset

OUTDOOR WET BULB	PRIMARY CONDENSER WATER TEMPERATURE
47°F	55°F
72°F	85°F

As the chilled water flow diagram illustrates, the economizer, or "free" cooling, mode of the walk-in freezer's condenser water system involves the bypassing of the chilled water heat exchangers by means of a 3-way valve. When the outdoor air wet bulb temperature falls below 45°F, the economizer mode is activated and the freezer's condenser water loop is diverted to the plate frame heat exchanger handled directly by cooling tower condensing water. When the outdoor air wet bulb temperature is above 52°F, the economizer mode is off.

5.7 STEAM BOILER AND HOT WATER SYSTEM CONTROL

The hot water system consists of two steam to hot water converters and two hot water pumps. Each converter (shell and tube heat exchanger) is controlled with two control valves having 1/3 and 2/3 capacities.

A temperature sensor is located in the common supply water leaving the heat exchangers and is connected to the DDC system. A proportional plus integral hot water reset control loop is also part of the control system. When the outdoor air temperature is 0°F, the hot water supply temperature is set to 200°F. When the outdoor air temperature is 70°F, the hot water supply temperature is 120°F. The supply water temperature is controlled so that it does not drop below 120°F or exceed 200°F. Hot water flow diagrams, located in Appendix C, further illustrate the hot water control system.

Since the Milton Hershey School New Supply Center is currently under construction, there is no operating history of the HVAC systems.

6.0 DESIGN VENTILATION REQUIREMENTS

ASHRAE Standard 62.1-2004 Table 6-1 (ASHRAE 2004) provides minimum ventilation rates for breathing zones and governs the design outdoor air requirements of the Supply Center. Table 6-1 includes a list of occupancy categories and the required minimum outdoor air rates per person and per square foot for those spaces.

The Ventilation Rate Procedure uses a series of equations in conjunction with tables found in Standard 62.1 (ASHRAE 2004) which calculate the amount of ventilation air required for each space based on the it's use, occupancy, and floor area. This procedure then calculates the amount of outdoor air required for each AHU to intake in order to ensure that each space receives at least the minimum amount of outdoor air. Ventilation rates calculated for a compliance check are summarized in Table 3 shown below. The table illustrates the amount of outdoor air each AHU is to intake in order to comply with the standard and the amount of outdoor air each AHU is scheduled to intake according to the design documents provided by H.F. Lenz Company.

Table 3 Ventilation Compliance Summary

AHU	ΣVOZ	MAX ZP	MIN OA REQ. (VOT CFM)	OA SUPPLIED (CFM)	COMPLIES WITH STD. 62.1
1	1124	0.1	1124	1150	YES
2	795	0.14	795	1150	YES
3	1957	0.53	3215	3640	YES
4	1817	0.52	3008	5585	YES
5	833	0.26	1041	3000	YES
6	1133	0.30	1416	13500	YES
7	549	0.29	686	3000	YES
8	335	0.09	335	7400	YES
9	943	0.38	1348	1000	NO
10	849	0.51	1414	1125	NO
11	354	0.23	394	1045	YES
12	497	0.19	552	1250	YES
13	896	0.25	996	3000	YES
14	582	0.18	647	800	YES

Table 3 shows that two of the fourteen air handlers do not comply with ASHRAE Standard 62.1 – 2004. The reason for the non-compliance is that the occupancy type assumed for certain areas, such as boys and girls fitting rooms, are assumed retail spaces which differ from the original design. Table 6-1 in Standard 62.1 (ASHRAE 2004) requires 7.5 cfm per person and 0.06 cfm per square foot of ventilation air supplied to these spaces. The original design of the supply center did not use any values on a per person basis. This proves that the original design calculations produced smaller amounts of ventilation air than what Standard 62.1 recommends.

There are significant over ventilation results that Table 3 illustrates. Recalling what is mentioned in the control logic section is that these AHUs are 100% outdoor air units for make-up when the kitchen and loading dock areas are in operation. The units must provide enough air to meet the thermal loads and to maintain space pressurization. The resulting actual outdoor air flow rate proves to over ventilate the spaces when compared to the value calculated using Standard 62.1.

7.0 DESIGN HEATING AND COOLING LOADS

The design heating and cooling are found from the construction documents provided by the H.F. Lenz Company. The equipment schedules on the design documents indicate the peak heating and cooling loads on the coils in each air handling unit.

In order to estimate design loads, annual energy consumption, and operating cost of the Milton Hershey School New Supply Center, Carrier's Hourly Analysis Program (HAP) is used as the building energy simulation program. Input data including OA ventilation rates, lights and equipment loads on a W/sq-ft basis, and design occupancy were all taken from design documents that are supplied by H.F. Lenz Company. Table 4 indicates that the computed cooling loads are similar to the design cooling loads. In all but one AHU, the computed load estimates are higher than the design loads.

The design heating loads for the supply center are outlined in Table 5. A new natural gas service is proposed to be extended to the new MHS Supply Center facility as the primary source of fuel/ energy. The building heating, domestic water heating, and kitchen equipment is supplied by natural gas.

Table 4 Cooling Load Comparison

SYSTEM	DESIGN COOLING LOAD (FT^2/TON)	COMPUTED COOLING LOAD (FT^2/TON)	DESIGN CW FLOW (GPM)	COMPUTED CW FLOW (GPM)
AHU-1	67.8	56.1	240.8	253
AHU-2	53.6	40.7	240.8	275.5
AHU-3	378	374.6	57.8	59.91
AHU-4	301	279.0	70.4	74.06
AHU-6	180.7	129	145.3	175.8
AHU-7	389.5	312.4	50.1	47.3
AHU-8	56.2	44.9	79.8	86
AHU-9	516	458.2	38.0	41.2
AHU-10	326	250	40.7	49.3
AHU-11	252	218.9	28.1	38.1
AHU-12	451	381.8	65.8	75.6
AHU-13	325.5	348.2	66.2	51.5
AHU-14	314	313.1	39.1	37.2

Note: AHU-5 does not include a cooling coil - Delivers 100% outdoor air year round

Table 5 Natural Gas Heating Demand Loads

KITCHEN EQUIPMENT	1,820
Building Heating Demand	9,640
Kitchen Domestic Water	3,731
TOTAL STEAM SYSTEM DEMAND	15,191
EQUIPMENT THAT USES GAS DIRECTLY	
Laundry	800
Clothes	360
Direct Gas For All Cooking	6,828
TOTAL DIRECT EQIUPMENT DEMAND	7,988
TOTAL PEAK DEMAND	**23,179**

Note: The loads are expressed in equivalent gas input CFH

8.0 ANNUAL ENERGY USE

The Hourly Analysis Program is also used to estimate the annual energy consumption for the supply center. Using the same data as in the cooling and heating loads simulation, specifying the mechanical equipment (AHUs, chillers, boilers, cooling towers, pumps, and fans) actually used in the design of the supply center allows for the calculation of annual energy consumption. Performance characteristics, such as efficiencies and COPs, of the major equipment are taken from the design documents supplied by H.F. Lenz Company.

The last pieces of information needed to perform a cost estimate are electric utility rates and the cost of natural gas. Meter data or utility bills are not obtainable since the supply center is currently under construction. Therefore, the electricity rates and natural gas cost used in the simulation are from the energy analysis performed by the design engineer (H.F. Lenz Company).

Table 6 Energy Rates

	OFF PEAK	DEMAND CHARGE
Electric Rates	$0.06 / kWh	$8.60 / kW
Natural Gas Rates	$1.35 / therm	

All airflow rates used in the analysis are taken from the design documents. The load analysis already calculated the design flow rates for each space and that the air handling units supply. Table 7 indicates that the calculated values and the actual design values are very similar. Therefore, these flow rates are used in the simulation.

Table 7 Supply Air and Outdoor Air cfm/ft² Comparison

SYSTEM	DESIGN SUPPLY AIR (CFM/FT^2)	COMPUTED SUPPLY AIR (CFM/FT^2)	VENTILATION SUPPLY (CFM/FT^2)
AHU-1	3.72	3.72	3.72
AHU-2	4.71	4.71	4.71
AHU-3	0.97	0.96	0.34
AHU-4	1.26	1.25	0.65
AHU-5	1.83	1.81	1.80
AHU-6	1.43	1.43	1.43
AHU-7	0.81	0.80	0.49
AHU-8	4.60	4.60	4.60
AHU-9	0.76	0.75	0.13
AHU-10	1.46	1.45	0.22
AHU-11	1.77	1.75	0.30
AHU-12	0.94	0.93	0.11
AHU-13	1.00	0.99	0.40
AHU-14	1.34	1.33	0.16

Appendix D includes summaries, charts, and figures from the results of the annual energy consumption simulation. Table 8 gives a breakdown of the annual energy cost for operating the HVAC equipment.

Table 8 Annual Energy Cost Breakdown

COMPONENT	ANNUAL COST ($/YR)	ANNUAL COST/FT² ($/FT² YR)	% OF TOTAL ENERGY COST
HVAC Component			
Electric	74,337	0.843	9.3 %
Natural Gas	32,420	0.368	4.1 %
HVAC Subtotal	106,757	1.210	13.4 %
Non HVAC Component			
Electric	64,049	0.726	8.0 %
Natural Gas	628,763	7.127	78.6
Non HVAC Subtotal	692,812	7.853	86.6 %
TOTAL	799,569	9.063	100 %

9.0 CRITIQUE OF HVAC SYSTEM

The Milton Hershey School New Supply Center's mechanical systems are designed with careful attention towards energy conservation and thermal comfort. Overall, the combination of the HVAC systems ability to incorporate other building systems as well as its sophisticated control methods used to minimize energy use while maintaining thermal comfort classifies it as a very good system for this application. The design engineers at H.F. Lenz Company cut no corners in the design, however, there are still alternatives that need addressed. Adjustments to the current design or redesign of certain areas of the

HVAC system can result in further optimization in first cost, construction cost, and operating cost.

9.1 GENERAL SYSTEM CRITIQUE

The supply center is a very wide and long single story 110,000 square foot building. With that said, the design of the air side mechanical system makes perfect sense. The placement of the 14 AHU's on the elevated mezzanine rooms creates neat and organized ductwork runs for each zone. The placement of the boiler and chiller plants also works well for the building lay out. Located at the north side of the building, the building plants are able to distribute hot water, chilled water, and medium pressure steam with little space impact by running piping in the ceiling plenums and up to the mezzanine rooms. As shown in figure 1 and analyzed in Technical Assignment 2, there is a very small percentage of rentable space occupied by the HVAC systems, which creates for an excellent system in this sense.

9.2 AIR SIDE SYSTEMS CRITIQUE

Variable air volume and constant air volume systems traditional create humidity problems if not controlled properly. However, as seen in the mechanical systems control logic section (section 5.0), proper air temperature monitoring and modulation of the AHU's cooling and heating coil's control valves creates acceptable indoor air quality.

The use of the mezzanine rooms, VAV units, and CAV units opens the doors for possible optimization. Potential energy, initial cost, and construction cost savings will result if dedicated outdoor air systems (DOAS) are used. The use of a DOAS to maintain the space humidity levels and ventilation needs with a parallel system can drastically lower the number of air handling units that are required. The use of another air system is possible for the parallel system, but if a water system is used, this will result in fewer AHUs required. Requiring a lesser amount of air handling units will most defiantly save in first cost, and DOAS are proven to save fan energy. There is potential in construction cost savings in terms of the mezzanine rooms as well. Currently, the elevated mezzanine rooms only house the air handling units and their VFDs. Using outdoor units and locating them on the roof of the supply center means that the elevated mezzanine rooms are not required. Construction and material cost saving potential is now available.

Parallel cooling/heating systems are required if DOAS is implemented. This opens the doors for eliminating the use of VAV units, and for finding more energy efficient systems. A water source heat pump system is an option, and the current ductwork layouts will not need major changes. The DOAS creates the need for smaller duct main sizes since only ventilation air is supplied, and the correct duct sizes for each zone will not need changing. Only replacement of VAV boxes with the terminal units is required for the system change.

The use of a DOAS system with parallel cooling does create maintenance issues. VAV systems are easier to maintain and most facility technicians are more familiar with this design because of its long history. Implementing a heat pump system, for example, creates the potential for additional training of the system operators and more extensive up-keep on the terminal units as compared to VAV boxes. Therefore, the VAV systems originally designed are easily maintained and improving or preserving this category is a challenge.

9.3 WATER SIDE SYSTEMS

Recovering the walk-in freezer's rejected heat in the plate frame heat exchangers eliminates the additional cooling load in the kitchen spaces. However, the freezers generate this heat year round and it is basically wasted in the heat exchangers. Heat recovery improvements to this portion of the HVAC systems will ultimately optimize the process. Integrating the water source heat pumps, as stated above, with the freezer condensate water system is an option for heat recovery. Maintaining the loop temperature with the freezer's waste heat instead of a boiler can prove to decrease the size of the boilers scheduled, and ultimately save in fuel consumption. As previously stated, the overall HVAC system is very good, but addressing the issues stated above has the potential to create an even more sustainable system.

10.0 REFERENCES

ASHRAE. 2004, ANSI/ASHRAE, Standard 62.1 – 2004, Ventilation for Acceptable Indoor Air Quality. American Society of Heating Refrigeration and Air Conditioning Engineers, Inc., Atlanta, GA. 2004.

H.F. Lenz Company. 2006, Mechanical Construction Documents. H.F. Lenz Company, Johnstown, PA. 2006.

H.F. Lenz Company. 2006, HVAC Design Objectives Proposal. H.F. Lenz Company, Johnstown, PA. 2006.

APPENDIX A – DESIGN OBJECTIVES DATA

Table A-1 Design Indoor Conditions

	HEATING DBT (˚F)	COOLING DBT (˚F)	MAX % RELATIVE HUMIDITY
Main offices	70	74	None (3)
Lobby and Office Corridors	70	74	None (3)
Public Toilets	70	78	None (3)
Servery	70	74	None (3)
Dining / Conference	70	74	None (3)
Meal Van Loading Dock	60	82	65
Kitchen	70	80	60
Dishwashing	70	80	60
Kitchen Training	70	80	60
Chef's Office	70	74	None (1)
Bakery	70	80	60
Bakery Dry Storage	68	78	50 (4)
Food Service Offices	70	74	None (3)
Kitchen dry goods	68	78	50 (4)
Clothing Storage	68	78	None (3)
Clothing Shipping / Receiving	70	74	None (3)
Used Clothing	70	74	None (3)
Alterations	70	74	None (3)
Clothing Work Room	70	74	None (3)
Clothing Display and Waiting	70	74	None (3)
Dry Storage	68	78	50 (4)
Bulk Storage / Bulk Staging	68	78	None (3)
Staging and staging corridor	68	80	None (1)
Transportation Offices	70	74	None (3)
Washer / dryer	68	80	None (1)
Program Support Inventory	70 (5)	74 (5)	None (3)
Storage for "year round" experience	68	78	None (3)
Garbage / recycling	50	95	None
Receiving and General Building Storage	68	80	None (1)
Mail	70	74	None (3)
IT Rooms (2)	68	70	None (1)
Mechanical / Electrical Rooms	60	100	None
Custodial	70	80	60
Break Room	70	74	None (3)
Lockers	70	74	None (1)
Bread/Bakery Staging	68	80	None (1)
Catering Staging	68	80	None (1)

Footnotes:
(1) No direct, active humidity control is planned but should not rise above 60% RH under normal operating conditions
(2) Wintertime humidification may or may not be necessary.

(3) No direct humidity control is planned, but should not rise above 50% RH under normal operating conditions.
(4) Will require a separate de-humidification unit to control humidity.
(5) Assumes there will be staff working in this room consistently

Note: All of these temperatures are "occupied mode" temperatures (i.e not unoccupied hours). Food storage spaces will not have changes in temperature/humidity requirements according to building occupancy.
No humidification systems are anticipated in the building design.

The design indoor condition data is supplied by H.F. Lenz Company and designed by H.F. Lenz in conjunction with the Milton Hershey School.

Table A-2 Design Conditions for Harrisburg, PA

LATITUDE	40.22°
Longitude	76.85°
Elevation	338 ft
Design Summer DBT	92.8°F
Design Summer WBT	73.7°F
Design Winter DBT	8.3°F

APPENDIX B – HVAC MAJOR EQUIPMENT CHARACTERISTICS

CHILLED WATER SYSTEM EQUIPMENT

Centrifugal Water Cooled Chillers

Quantity	Evaporator				Compressor				Condenser		
	GPM	EWT	LWT	Max Water PD ft	Quantity	Type	NPLV (kW/ton)	kW/ton Input	GPM	EWT	Max Water PD ft
2	648	55	45	19.8	1	Centrifugal	0.395	0.627	810	85	22.9

Water Cooled Brine Chillers

Quantity	Evaporator				Compressor				Condenser		
	GPM	EWT	LWT	Max Water PD ft	Quantity	Type	Min Cap'y Step	kW/ton Input	GPM	EWT	Max Water PD ft
2	155	25.8	20	17.3	2	Screw	10%	1.228	250	85	9.7

Induced Draft Cooling Towers

Quantity	Fan CFM	EAT WB °F	LWT °F	GPM	Fan Motor HP	Sump Heaters			VFD?
						kW	V	Ph	
2	103,700	76	95	85	25	14	480	3	YES

Plate Frame Heat Exchangers

Quantity	Freezer Cond. Closed Loop				Cold Side					Notes
	Flow Rate GPM	EWT °F	LWT °F	Max PD psi	Flow Rate GPM	Fluid	EWT °F	LWT °F	Max PD psi	
1	135	84	65	2.3	260	Open Loop CDW	60	69.8	8.5	Used for "free" cooling
2	135	84	65	3.5	170.5	Chilled Water	45	60	5.5	Used in summer time operation

Air-Cooled Ductless Split System (IT Rooms)

Supply CFM	OA CFM	Nominal Cooling MBH	EAT DB	EAT WB	System EER
559	0	27	80	67	12.1

AIR HANDLING UNITS

Air Handling Unit Fans																			
AHU	Serves	Supply Fan Data										Return Fan Data							
		CFM	OA CFM	Ext SP	Total SP	Drive	Type	RPM	HP	BHP	VFD?	Ext SP	Drive	Type	RPM	HP	Total SP	BHP	VFD?
1	Kitchen	22,000	22,000	4	6.18	Belt	Plenum	1115	40	31.2	Yes	None							
2	Kitchen	22,000	22,000	4	6.18	Belt	Plenum	1115	40	31.2	Yes	None							
3	Offices	9,100	3,640	4	5.09	Belt	DWDI AF	1846	20	11.5	Yes	None							
4	Misc Spaces	10,875	5,585	4	5.51	Belt	DWDI AF	1122	20	15	Yes	1.87	Belt	SWSI FC	700	10	1.87	6.3	Yes
5	Trash Area	3,000	3,000	2	3.28	Belt	Plenum	2362	5	3	No	None							
6	Loading Dock	13,500	13,500	3	5.1	Belt	DWDI AF	1123	25	19.3	Yes	None							
7	Staging	5,000	3,000	3	4.37	Belt	Plenum	1769	10	5.9	No	0.85	Belt	SWSI FC	562	3	0.85	1.2	No
8	Bakery	7,400	7,400	3	4.64	Belt	Plenum	1438	15	8.2	Yes	None							
9	Dry Storage	6,000	1,000	3	4.78	Belt	Plenum	1890	10	7.1	Yes	None							
10	Clothing	7,500	1,125	4	5.21	Belt	Plenum	1510	15	9.4	Yes	0.84	Belt	SWSI FC	517	3	0.86	2.2	Yes
11	Misc Spaces	6,145	1,045	4	5.59	Belt	Plenum	2006	15	8.6	Yes	None							
12	Clothing Storage	10,850	1,250	3	4.39	Belt	Plenum	1127	20	11.1	Yes	None							
13	Receiving	7,500	3,000	3	6.15	Belt	Plenum	2231	20	11.6	Yes	None							
14	Dry Storage	6,500	800	3	4.99	Belt	Plenum	1977	15	8.1	Yes	None							

Air Handling Units Cooling Coils																	
AHU	Chilled Water Coil									Hot Water Pre-Heat Coil							
	EAT °F		LAT °F		SP (in)	GPM	EWT °F	#Rows/FPI	WPD	FB Damper Type	EAT °F DB	LAT °F DB	SP (in)	GPM	EWT °F	#Rows/FPI	WPD
	DB	WB	DB	WB													
1	95	72	51	51	1.02	240.8	45	8/14	11.9	Integ. FB	0	75	0.32	119.1	180	3/12	13.1
2	95	72	51	51	1.02	240.8	45	8/14	11.9	Integ. FB	0	75	0.32	119.1	180	3/12	13.1
3	81.2	64.5	51	51	0.37	57.8	45	6/11	6.5	Integ. FB	39.2	81	0.22	27.5	180	2/12	3.4
4	81.2	64.5	52	51	0.64	70.4	45	8/11	2.2	Integ. FB	39.9	80	0.24	32.1	180	3/6	5.8
5	NONE									Integ. FB	48	92	0.32	14.4	180	3/9	0.3
6	95	72	52	52	1.06	145.3	45	8/14	8.7	NONE	33	92	0.17	87.3	180	2/8	7.2
7	87	68.3	51.9	51.8	0.66	50.1	45	8/11	1.7	NONE	60.4	96.3	0.11	19.7	180	1/8	2.5
8	95	72	52	52	0.71	79.8	45	8/14	2.4	Integ. FB	0	79	0.31	42.3	180	3/12	19
9	79.2	64.6	51	51	0.84	38	45	8/11	4.6	NONE	56.7	90	0.14	22.2	180	1/8	3.2
10	74.5	60.6	51.2	50.8	0.43	40.7	45	8/11	2.7	NONE	62.9	98.1	0.11	28.8	180	1/8	2.4
11	75.9	61.4	51	51	0.61	28.1	45	8/14	2.1	NONE	58.1	91	0.15	22.1	180	1/8	3.2
12	78.2	64.1	52	52	0.65	65.8	45	8/11	2	NONE	60.8	97	0.11	43.3	180	1/8	5.9
13	86	69	52	52	1.36	66.2	45	8/14	12.6	Integ. FB	35	74	0.69	21.3	180	2/12	7.1
14	78.4	64.2	52	52	0.95	39.1	45	8/11	4.8	NONE	60.2	91	0.17	22.3	180	1/8	3.2

Air Handling Units Re-heat Coil and Filters																	
AHU	Hot Water Re-heat Coil							Pre-Filter Data					Final Filter Data				
	EAT °F DB	LAT °F DB	SP (in)	GPM	EWT °F	#Rows/FPI	WPD	Eff %	Depth	Type	Final SP	Total Area Sq ft	Eff %	Depth	Type	Final SP	Total Area Sq ft
1	52	88	0.15	86.6	180	1/8	6.8	30	2"	Flat	0.6	52.08	85	4"	Flat	1.2	52.08
2	52	88	0.15	86.6	180	1/8	6.8	30	2"	Flat	0.6	52.08	85	4"	Flat	1.2	52.08
3	NONE							30	2"	Flat	0.6	33.33	85	4"	Flat	1.2	33.33
4	NONE							30	2"	Flat	0.6	33.33	85	4"	Flat	1.2	33.33
5	NONE							30	2"	Flat	0.6	11.11	NONE				
6	52	87	0.16	52.2	180	1/8	8.4	30	2"	Flat	0.6	33.33	85	4"	Flat	1.2	33.33
7	52	90.8	0.1	21.3	180	1/8	2.9	30	2"	Flat	0.6	15	85	4"	Flat	1.2	15
8	52	91	0.1	31.9	180	1/8	2.9	30	2"	Flat	0.6	20.83	85	4"	Flat	1.2	20.83
9	50	86	0.14	23.6	180	1/8	3.6	30	2"	Flat	0.6	15	85	4"	Flat	1.2	15
10	NONE							30	2"	Flat	0.6	20.83	85	4"	Flat	1.2	33.33
11	NONE							30	2"	Flat	0.6	15	85	4"	Flat	1.2	15
12	52	91	0.11	47	180	1/8	6.9	30	2"	Flat	0.6	33.33	85	4"	Flat	1.2	33.33
13	52	83	0.21	25.8	180	1/8	3.1	30	2"	Flat	0.6	15	85	4"	Flat	1.2	15
14	52	86	0.16	24.1	180	1/8	3.7	30	2"	Flat	0.6	15	85	4"	Flat	1.2	15

HOT WATER/STEAM SYSTEM EQUIPMENT

Fire Tube Boilers						
Quantity	Fire Rating Gas CFH	Gross Output MBH	Boiler HP	Comb. Air Fan Motor HP	Heating Surface Sq Ft/hp	Water Volume Gallons
2	8,165	6,695	200	15	5	8,625
1	5103	4,184	125	10	5	5,750

Air-Cooled Ductless Split System (IT Rooms)					
Supply CFM	OA CFM	Nominal Cooling MBH	EAT DB	EAT WB	System EER
559	0	27	80	67	12.1

Horizontal Unit Heater (hot water)					
CFM	RPM	HP	MBH	GPM	Max PD (ft)
1100	1050	1/30	44.8	5.3	0.23
1050	1050	1/30	37.3	4.4	0.17
450	1550	9 WATT	14.9	1.8	0.014

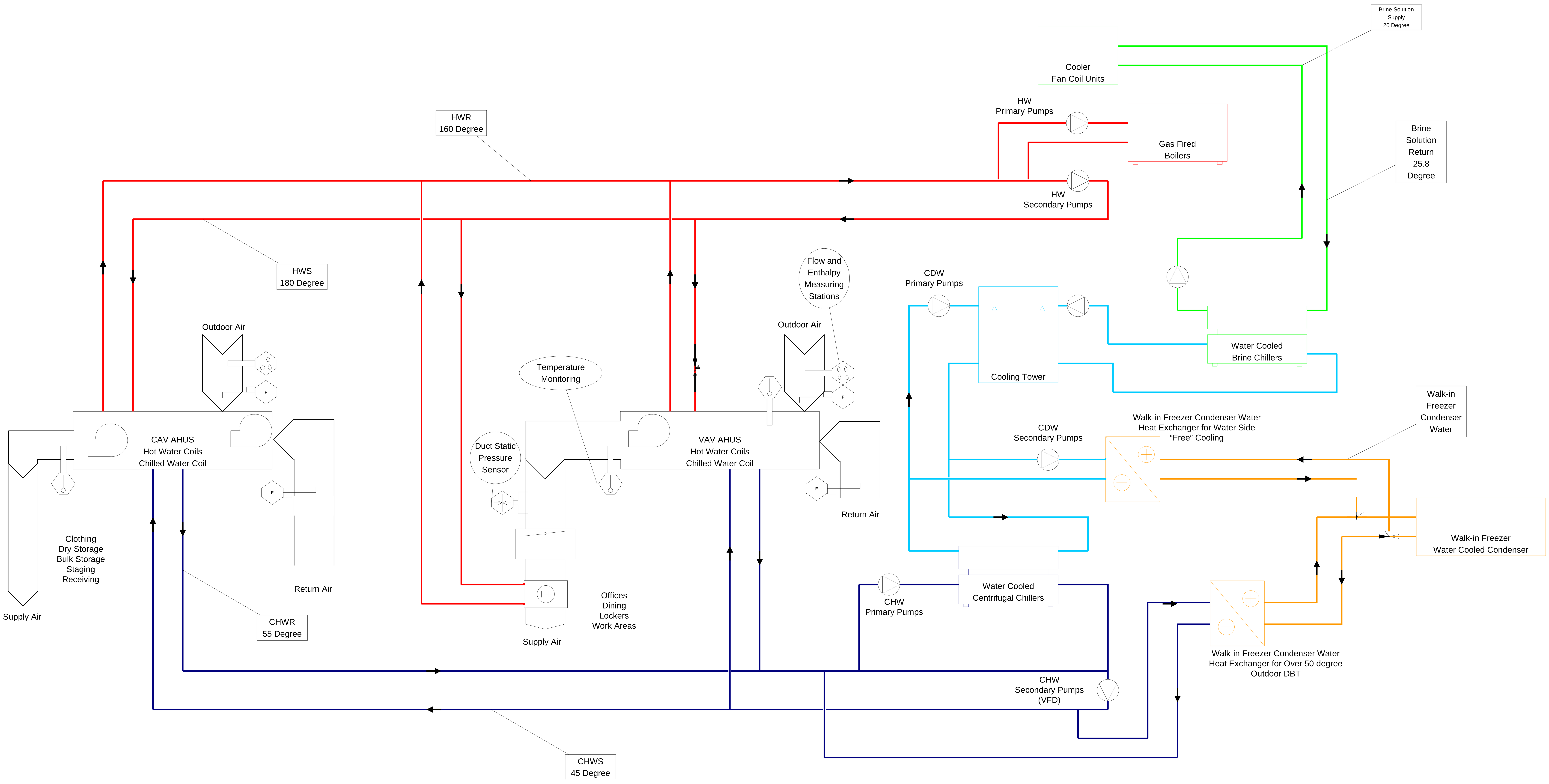
Cabinet Unit Heater (hot water)						
CFM	MBH	GPM	PD (ft)	# Rows	HP	RPM
630	41.1	4.9	1.6	1	1/10	1050
430	25.5	3.1	0.61	1	1/10	1050
230	14.1	1.7	0.19	1	1/15	1050

Shell and Tube Heat Exchangers							
Quantity	Circulating Fluid (Tubes)				Heating Fluid (Shell)		
	GPM	In °F	Out °F	PD ft	Fluid	lbs/hr	Press.
2	750	155	180	0.9	Steam	9.544	5 psi

Pumps

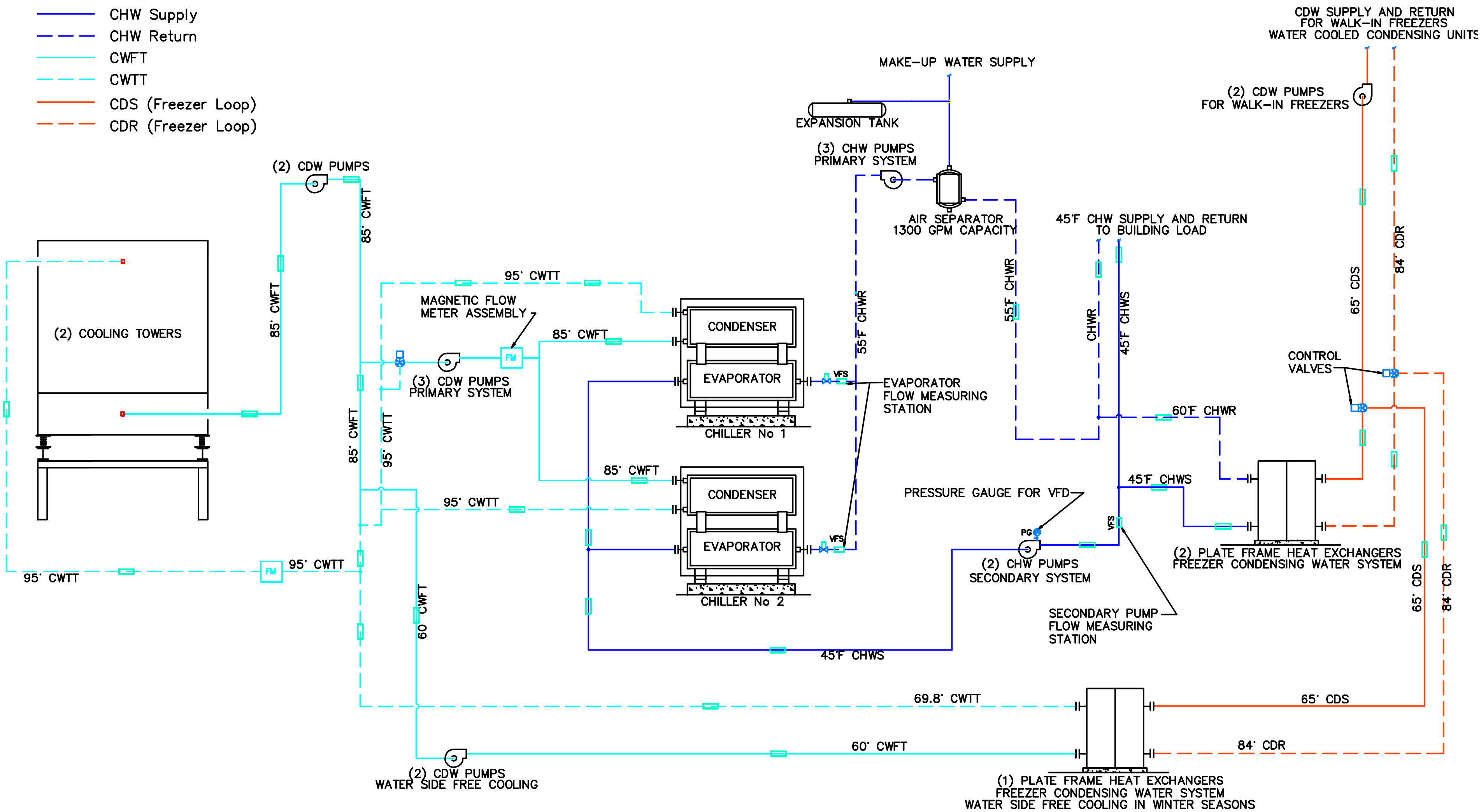
Pump No	Type	System	Split/Close Coupled?	Operation: Duty/Standby?	Fluid Type	Max GPM	Shut Off HD (ft)	Operation Point bhp	Max bhp	Motor hp	RPM	VFD?	Impeller Diam.
CHWP-1	Flr Mtd	Secondary 45° CHW	Split	Duty	Water	1460	115	32	32.8	40	1750	Yes	11"
CHWP-2	Flr Mtd	Secondary 45° CHW	Split	Standby	Water	1460	115	32	32.8	40	1750	Yes	11"
CHWP-3	Flr Mtd	Primary 45° CHW	Split	Duty	Water	885	44	7	7.6	10	1150	No	10.125"
CHWP-3A	Flr Mtd	Primary 45° CHW	Split	Duty	Water	885	44	7	7.6	10	1150	No	10.125"
CHWP-4	Flr Mtd	Primary 45° CHW	Split	Standby	Water	885	44	7	7.6	10	1150	No	10.125"
CHWP-5	In-Line	Primary 20° CHW	Close	Duty	35% Brine	272	40	2.3	2.8	3	1750	No	6.5"
CHWP-6	In-Line	Primary 20° CHW	Close	Standby	35% Brine	272	40	2.3	2.8	3	1750	No	6.5"
CHWP-7	Flr Mtd	Secondary 20° CHW	Split	Duty	35% Brine	262	78	4.1	5.1	5	1750	Yes	8.5"
CHWP-8	Flr Mtd	Secondary 20° CHW	Split	Standby	35% Brine	262	78	4.1	5.1	5	1750	Yes	8.5"
CDWP-1	Flr Mtd	Primary Condenser water	Split	Duty	Water	3100	54	24.5	25.8	30	1150	Yes	11.625"
CDWP-2	Flr Mtd	Primary Condenser water	Split	Standby	Water	3100	54	24.5	25.8	30	1150	Yes	11.625"
CDWP-3	In-Line	Chiller-1,2 Condenser Water	Split	Duty	Water	1205	51	9.7	10.2	15	1150	No	7.5"
CDWP-3A	In-Line	Chiller-1,2 Condenser Water	Split	Duty	Water	1205	51	9.7	10.2	15	1150	No	7.5"
CDWP-4	In-Line	Chiller-1,2 Condenser Water	Split	Standby	Water	1205	51	9.7	10.2	15	1150	No	7.5"
CDWP-5	In-Line	Chiller-3,4 Condenser Water	Split	Duty	Water	390	40	2.7	2.8	3	1750	No	6.5"
CDWP-6	In-Line	Chiller-3,4 Condenser Water	Split	Standby	Water	390	40	2.7	2.8	3	1750	No	6.5"
CDWP-7	In-Line	Closed Loop Freezer Condensers	Split	Duty	Water	155	91	4.15	4.39	5	1750	No	9.125"
CDWP-8	In-Line	Closed Loop Freezer Condensers	Split	Standby	Water	155	91	4.15	4.39	5	1750	No	9.125"
CDWP-9	In-Line	Open Loop Freezer Condensers	Split	Duty	Water	425	48	3.67	4.75	5	1150	No	10.375"
CDWP-10	In-Line	Open Loop Freezer Condensers	Split	Standby	Water	425	48	3.67	4.75	5	1150	No	10.375"
HWP-1	Flr Mtd	Heating Hot Water	Split	Duty	Water	1410	110	28.3	30.2	40	1750	Yes	10.5"
HWP-2	Flr Mtd	Heating Hot Water	Split	Standby	Water	1410	110	28.3	30.2	40	1750	Yes	10.5"
HWP-3	In-Line	HWC-1	Close	Duty	Water	41	19	0.2	0.22	0.25	1725	No	4.6875"
HWP-4	In-Line	AHU-6 Pre-heat coil pump	Close	Duty	Water	165	14.9	0.48	0.6	0.75	1150	No	6"
HWP-5	In-Line	AHU-7 Pre-heat coil pump	Close	Duty	Water	32	20	0.2	0.22	0.25	1725	No	4.6875"
ERP-1	In-Line	AHU-6 Energy Recovery	Close	Duty	30% Brine	185	55	2.2	3	3	1750	No	7.5"

APPENDIX C – SYSTEM FLOW DIAGRAMS



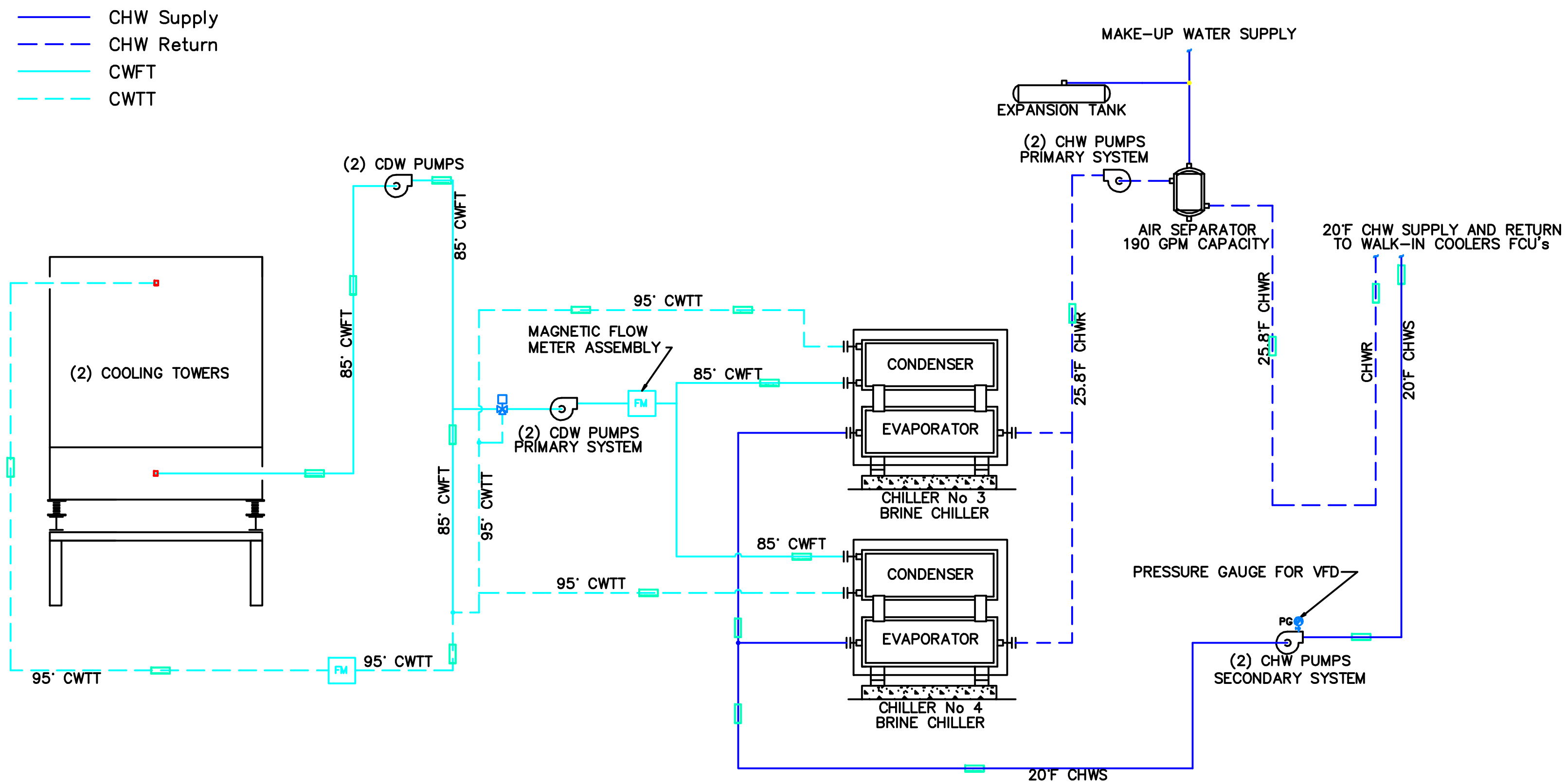
Overall HVAC Schematic

Key:
CHWS/R – Chilled Water Supply/Return
CDWS/R – Condenser Water Supply/Return (Freezers)
CWFT – Condenser Water From Tower
CWTT – Condenser Water To Tower

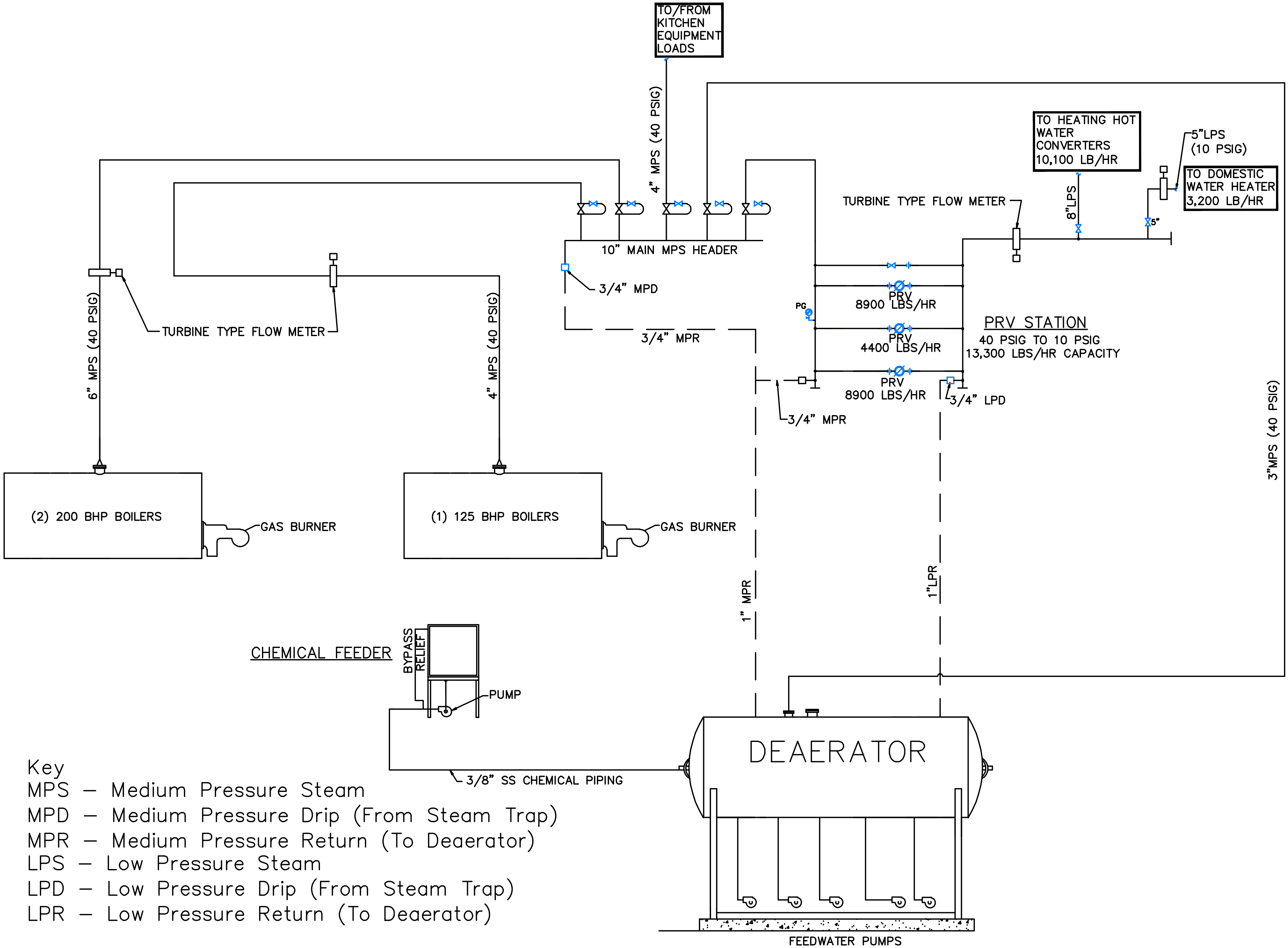


CHILLED WATER FLOW DIAGRAM (45°F WATER)

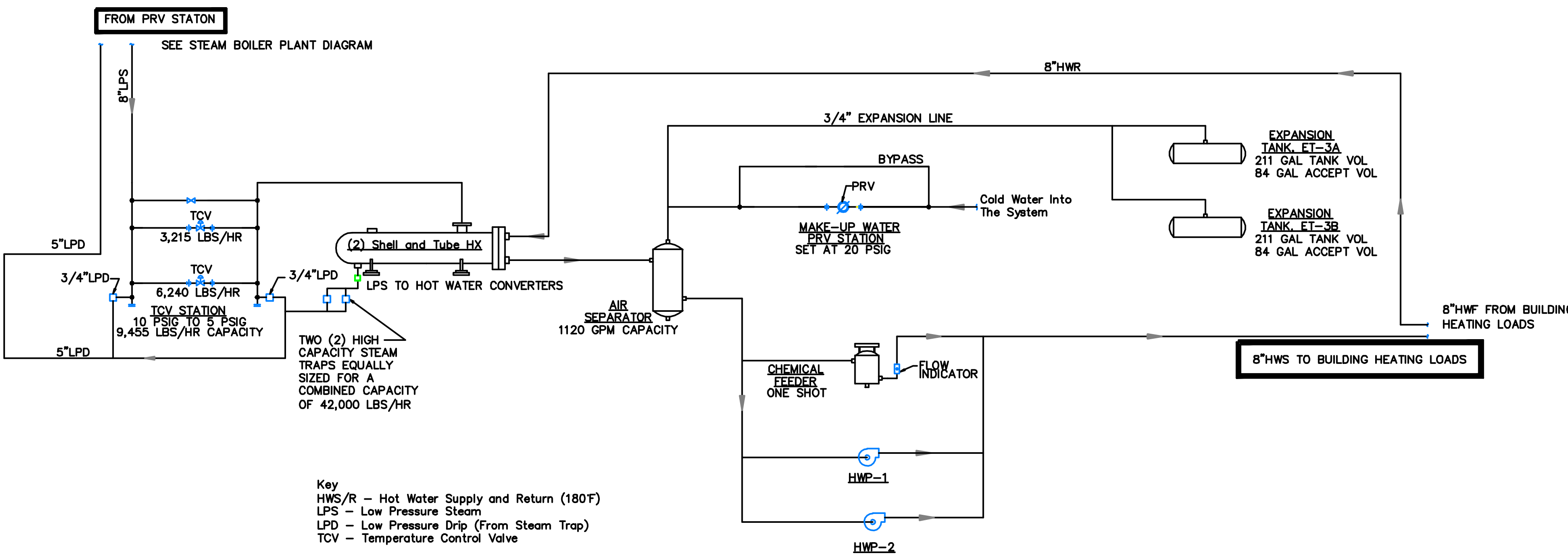
Key:
CHWS/R – Chilled Water Supply/Return
CDWS/R – Condenser Water Supply/Return (Freezers)
CWFT – Condenser Water From Tower
CWTT – Condenser Water To Tower



CHILLED WATER FLOW DIAGRAM (20°F BRINE SOLUTION)



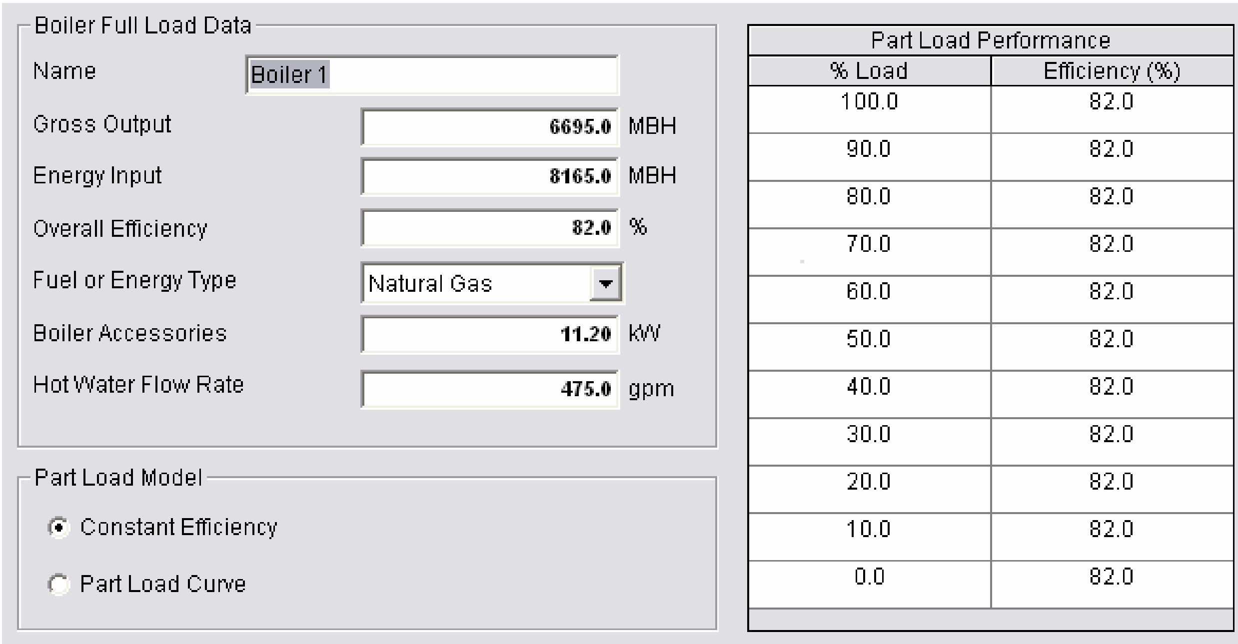
STEAM BOILER PLANT FLOW DIAGRAM



HOT WATER CONVERTER DIAGRAM

APPENDIX D – ANNUAL ENERGY CONSUMPTION AND COST RESULTS

INPUTS



Boiler Full Load Data

Name: Boiler 1

Gross Output: 6695.0 MBH

Energy Input: 8165.0 MBH

Overall Efficiency: 82.0 %

Fuel or Energy Type: Natural Gas

Boiler Accessories: 11.20 kW

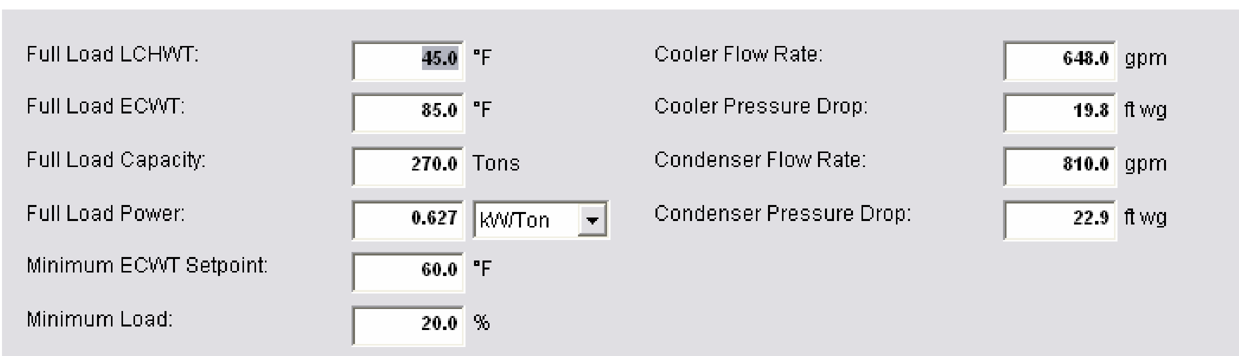
Hot Water Flow Rate: 475.0 gpm

Part Load Model

- Constant Efficiency (selected)
- Part Load Curve

Part Load Performance	
% Load	Efficiency (%)
100.0	82.0
90.0	82.0
80.0	82.0
70.0	82.0
60.0	82.0
50.0	82.0
40.0	82.0
30.0	82.0
20.0	82.0
10.0	82.0
0.0	82.0

Figure D-1 Boiler performance characteristics



Full Load LCHWT: 45.0 °F

Full Load ECWT: 85.0 °F

Full Load Capacity: 270.0 Tons

Full Load Power: 0.627 kW/Ton

Minimum ECWT Setpoint: 60.0 °F

Minimum Load: 20.0 %

Cooler Flow Rate: 648.0 gpm

Cooler Pressure Drop: 19.8 ft wg

Condenser Flow Rate: 810.0 gpm

Condenser Pressure Drop: 22.9 ft wg

Figure D-2 Chiller performance characteristics

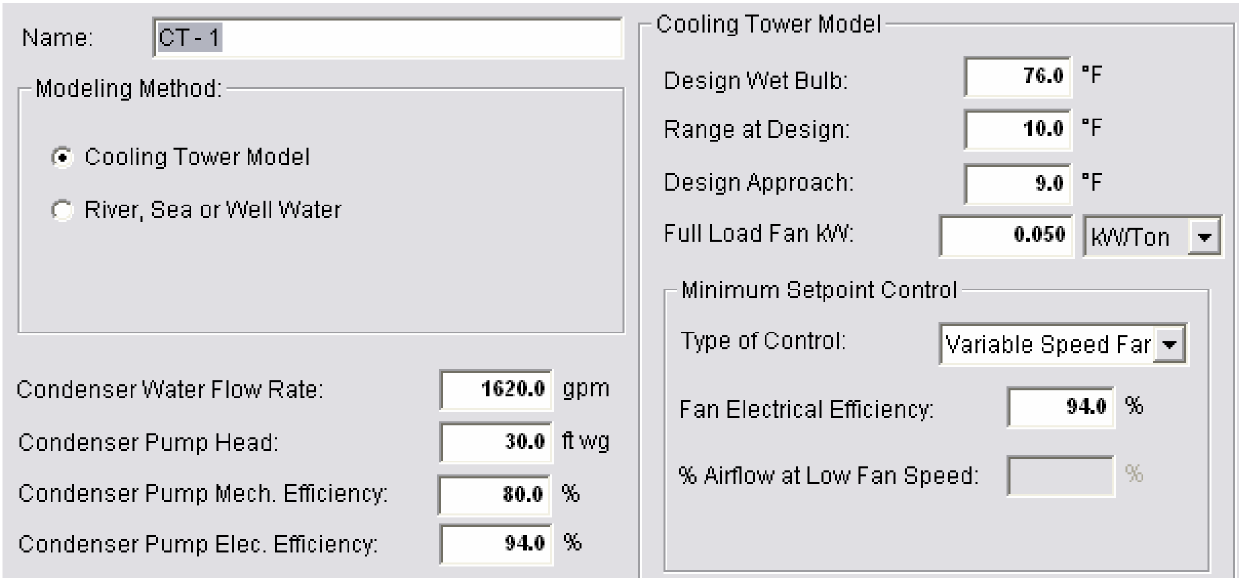


Figure D-3 Cooling tower performance characteristics

Figures D-1 to D-3 indicated the performance characteristics of the major mechanical equipment used in the HAP energy analysis. Also, all major fan and pump efficiencies are used and taken from the design documents provided by H.F. Lenz Company.

- Fan efficiencies range from 45% - 74%
- Primary loop chilled water pump efficiency is 85%
- Secondary loop chilled water pump efficiency is 77%

RESULTS

Table D-1 Component Annual Cost Breakdown

COMPONENT	ANNUAL COST ($/YR)	ANNUAL COST/FT² ($/FT² YR)	% OF TOTAL ENERGY COST
HVAC Component			
Air System Fans	21,303	0.242	2.7 %
Cooling	23,239	0.263	2.9 %
Heating	32,618	0.370	4.1 %
Pumps	24,405	0.277	3.1 %
Cooling Tower Fans	5,194	0.059	0.6 %
HVAC Subtotal	**106,759**	**1.210**	**13.4 %**
Non HVAC Component			
Lights	18,501	0.210	2.3 %
Electrical Equipment	45,546	0.516	5.7 %
Misc. Fuel Use	628,763	7.127	78.6
Non HVAC Subtotal	**692,810**	**7.853**	**86.6 %**
TOTAL	**799,569**	**9.063**	**100 %**

Table D-1 shows the annual energy cost to operate each component in the supply center. The table also breaks down the cost per square foot.

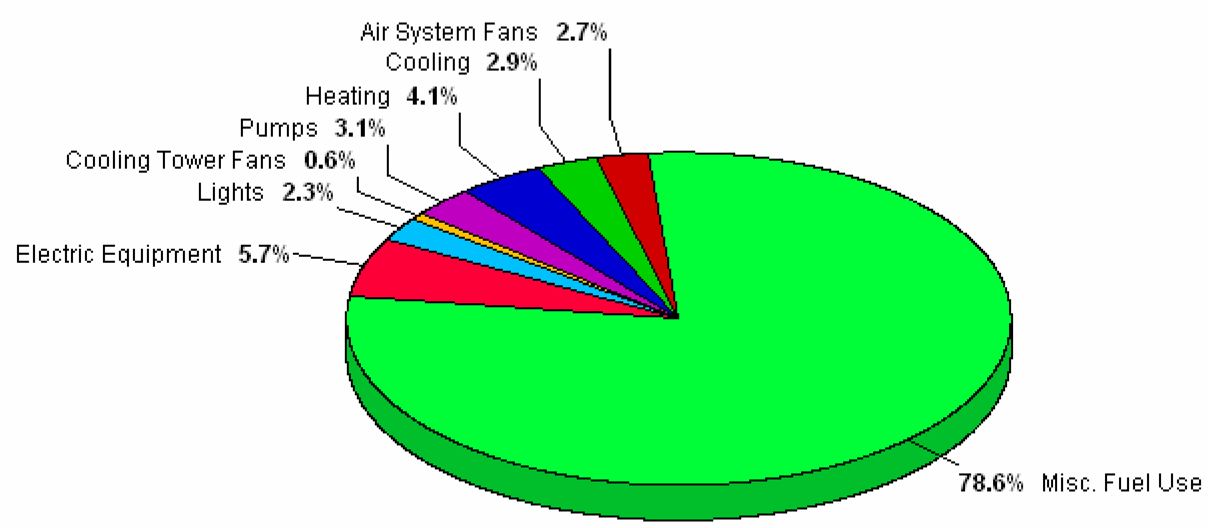


Figure D-4 Annual component cost summary

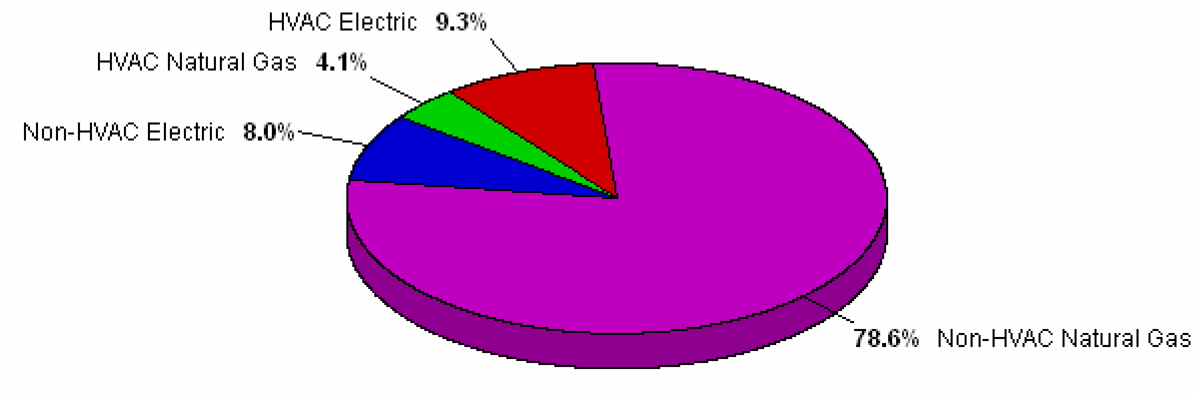


Figure D-5 Annual energy cost by fuel type

Table D-2 Annual Energy Consumption Breakdown

COMPONENT	ANNUAL ENERGY CONSUMPTION
HVAC Component	
Electric (kWh)	1,238,947
Natural Gas (therms)	23,969
Non HVAC Component	
Electric (kWh)	1,067,486
Natural Gas (therms)	464,855
TOTAL	
Electric (kWh)	2,306,433
Natural Gas (therms)	488,824

Aside from energy consumption, it is also important to recognize the amount emissions generated from operating the building. HAP also performs an annual emissions calculation for the production of CO_2, SO_x, and NO_x. Table 15 illustrates the estimated annual emissions produced for operating the supply center.

Table D-3 Annual Emissions Report

EMISSION	AMOUNT PRODUCED
CO_2	280,437 lb
SO_x	652 kg
NO_x	2,253 kg

Figure D-5 and D-6 illustrate monthly cost to operate the supply center. D-5 is a breakdown according to each component where as D-6 breaks down the monthly cost by fuel type.

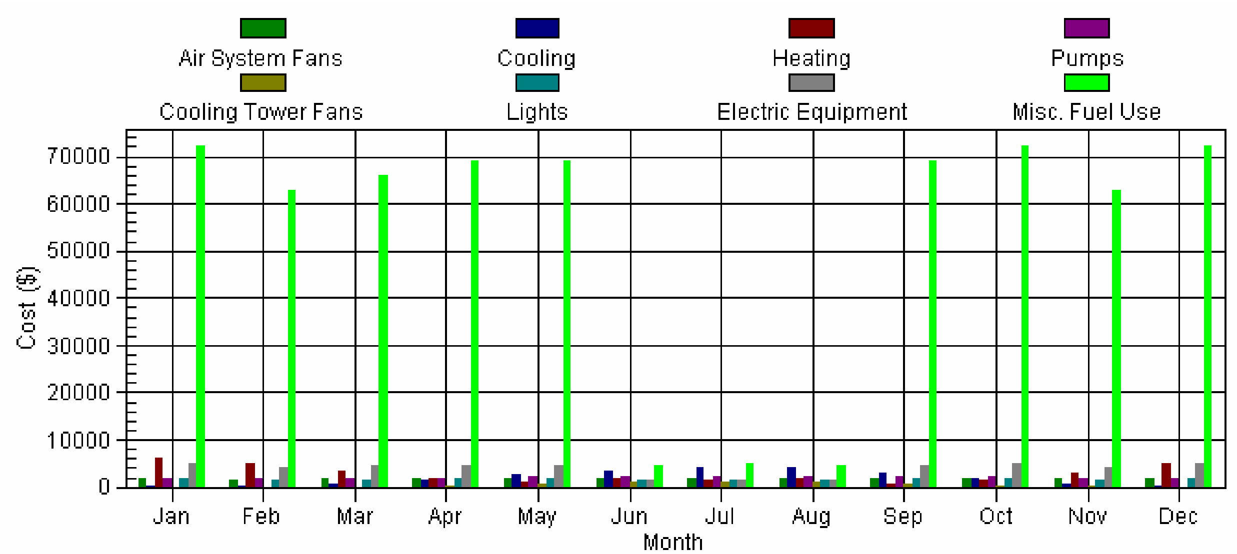


Figure D-6 Monthly cost breakdown by component

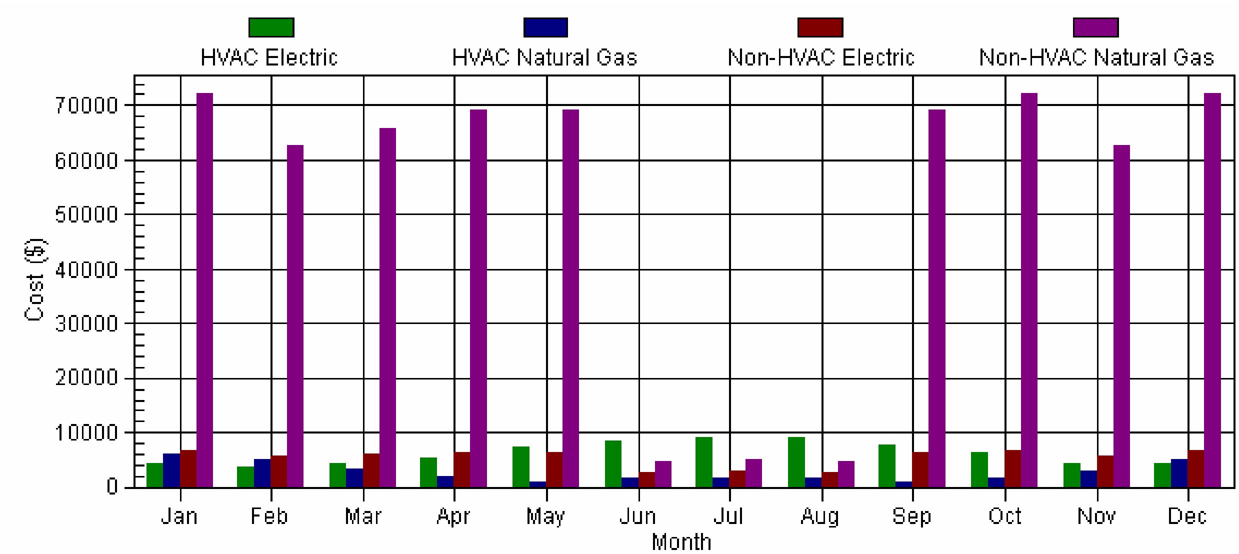


Figure D-7 Monthly cost breakdown by fuel type