Mechanical Systems Evaluation

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George W. Hays PK-8

Cincinnati Public School
Cincinnati, OH

Prepared For
Dr. Jelena Srebric

By

Rodrick A. Crousey
The Pennsylvania State University
Architectural Engineering
Mechanical Option

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Executive Summary

This report looks at the initial mechanical design goals of the Architect, Owner, and Mechanical Engineer of the George W. Hays PK-8 School in Cincinnati, OH, and then looks at the individual design components to see how well the building met with its design goals. Initial goals of the design team included goals of energy use, thermal comfort, adequate ventilation, and mechanical space equipment usage. In general these goals were met by the final design of the building system.

The 66,000 ft^2 Public School design team looked towards ASHRAE Standard 90.1 as a guide to reduce energy consumption of the building. Analysis of the structure shows an expected HVAC equipment energy consumption of 53 $kBTU/ft^2/yr$. This consumption is a mixture of electricity used to run fans and pumps, electricity used to created chilled water, and natural gas energy used for the hot water system.

To obtain thermal comfort in all spaces, a careful controls system was mapped out, along with research into the best type of system for educational spaces. Design experience produced a system containing one chiller, two boilers, three air handling units, three total energy wheels, two radiant panels, and the ability for each system to enter into full economizer mode. This system was also carefully controlled to ensure optimum space conditions in all seasons and during all occupied hours.

Ventilation requirements for the space were found by local code requirements. The design ventilation goals were met, even though they fell bellow ASHRAE Standard 62.1. A carefully thought out controls system was implemented to ensure each space was always provided an adequate amount of minimum OA.

Design goals regarding the area used by mechanical equipment were obtained by a good interaction between the engineer and architect to ensure the integrity of the building aesthetics was maintained while allowing adequate space for the building mechanical components.

1. Design Goals

According to the mechanical engineer, the main mechanical focus for all projects for the Cincinnati Public Schools is energy conservation. From previous experience, the design team anticipated a high percentage of Outdoor Air (OA) in each of the three zones. Zone 1, consisting of mainly classrooms and Zone 2, consisting of a mixture of classrooms, offices, and general spaces were expected to have an OA percentage around 50%. The zone for the gymnasium and the gymnasium support area, Zone 3, was expected to have an even higher OA percentage around 70%. Due to the high percentage of required outdoor air, complete enthalpy wheels were implemented as an early design objective. The design team was also focused on implementing high efficiency boilers to supply the decoupled heating and domestic hot water systems.

Thermal comfort is a goal that is incorporated into every design by the mechanical designer. Thermal comfort means creating an atmosphere at which the occupants are expected to be comfortable in terms of both Dry Bulb Temperature (DBT) and Relative Humidity. This goal is achieved by combining design experience and advice from professional journals such as ASHRAE to combine the components of work level and clothing level to determine a desired DBT and Relative Humidity along with providing a reasonable level of occupant control.

Mechanical designers are also restrained by space limitations. Though there were no initial specific floor area limits on the mechanical system, an initial goal by a mechanical designer is to place the equipment within an area that is agreeable by the architect and owner. Excess mechanical space can result in lost rentable space, or even the possibility of affecting the overall aesthetics of the building.

2. Design Conditions

Design conditions include the desired Indoor Air temperature in addition to the various determined design OA temperatures. The specific values for these various conditions can be found on Table 2.1.

Table 2.1 Design Conditions

	SUMMER†			WINTER††
	DBT [F]	WBT [F]	% RH	DBT [F]
OUTDOOR 1*	88	73	-	5
OUTDOOR 2**	-	75	100	-
INDOOR	74	-	50	70
UTILITY SPACES	65	-	-	-

*Design condition based off of DBT
**Design condition based off of WBT
†Summer OA conditions based off of 2% ASHRAE Fundamentals 2001
††Winter OA conditions based off of 99.6% ASHRAE Fundamentals 2001

All of the OA conditions were determined by the mechanical engineer using ASHRAE Fundamentals 2001. Two different possible summer design OA conditions were of interest to the designer. The first is the 2% Dry Bulb Temperature (DBT) condition. This value is the DBT that is surpassed 2% of the hours in a year (175 hours per year). A 2% design condition is acceptable because of the thermal mass of a building allowing the building to maintain indoor air conditions when the OA conditions exceed design for a limited span of time. This span of time is not expected to be exceeded when using a 2% design condition. Associated with the design DBT is a Wet Bulb Temperature (WBT). This value gives the designer a point on the psychrometric chart to base the design of the system on. The other potential design condition is the 2% WBT condition. Similar to the 2% DBT condition, the 2% WBT condition is the WBT that is surpassed 2% of the hours in a year. This design WBT has the potential of accumulating a latent load large enough to require more tons of cooling then would be required if looking at the design DBT alone.

The Indoor Air conditions were decided by analyzing the conditions of the respective spaces keeping in mind energy usage and thermal comfort.

Specific variables taken into account include the amount of clothing occupants are expected to wear, the expected level of activity for the occupants, and the OA conditions. Clothing has an effect on the amount of heat the occupants' bodies are able to reject due to varying thermal resistance. Activity level has been proven to have a direct effect on the occupant's metabolic rate, or the energy that the person is creating. This metabolic variance changes the occupant's perception of what defines comfortable conditions. Finally, the OA temperature has two major influences on the decided space temperature. First, cooling summer air requires more energy then cooling winter air and the converse is also true. Therefore to be energy conscious, summer Indoor Air conditions can be decided to be warmer then the winter Indoor Air conditions. The second effect deals with acclimatization. A human body adjusts over time to different temperatures. Therefore, in the winter months the occupant will define thermal comfort as being cooler then the defined thermal comfort in the summer.

3. System Descriptions

The mechanical system for the building was designed with the goal of maintaining thermal comfort with minimum energy usage. The components of the system include a single centrifugal chiller, two hot water boilers, and three air handling units. The diagrams for these systems may be found in Appendix A and specific information about the major system components may be found in Appendix B.

All of they systems work together to achieve the mechanical goals of the system. To help ensure a proper monitoring and coordination of this system, a direct digital control system was called for that allows the owner to monitor and record all of the major system components from locations away from the site. The system components work together to manage four daily timeframes: Unoccupied, Startup, Occupied, Coast-Down. Since no occupants are expected to be in the space, the Unoccupied timeframe has no requirements for ventilation or thermal comfort. To save energy the system is turned off during the night hours. Towards the end of the first Unoccupied period the Startup is activated, where the system activates prior to occupancy. This Startup period is necessary because of the lag systems naturally have due to thermal mass and unconditioned air in the space overnight. Because of unknown factors to the response of a system prior to construction, the building controls system has a memory that continuously adjusts the Startup time based upon previously recorded data. During the Occupied hours the system is run in a way to achieve thermal comfort and required ventilation to the space. Towards the end of the Occupied timeframe is the Coast-Down. During the Coast-Down period the thermal components of the system begin to turn off with the anticipation of the thermal lag of the building maintaining thermal comfort conditions until the Occupied period is over and the second Unoccupied period of the day begins. Like the Startup, the Coast-Down period changes based on previously recorded data. By implementing this system thermal comfort is ensured in the early hours of the day and energy is saved in the afternoon by taking advantage of the natural lag of the building.

3.1 Cooling Systems

The only active cooling system for the building is a single 170 ton centrifugal chiller, CHLR-1. This system is only responsible for serving the cooling coils in the three AHU's. The chiller is designed to run at a set supply water temperature of 43°F. The controls logic calls for the chiller to be activated any

time the Outdoor Air (OA) temperature is greater then 55°F degrees and at least one AHU is in occupied mode. When the OA temperature falls bellow 50°F, the chiller is disabled and the AHU's are put into full economizer mode, which is further discussed in the description of the AHU's. The chilled water bypass valve is staged according to the Differential Pressure (DP) of the evaporator to ensure the minimum recommended flow rate stated by the chiller manufacturer.

A single 300 gpm pump provides the required pressure drop for the circuit. To help prevent cavitation, a suction diffuser is incorporated at the inlet of the pump. Flow conditions can be verified with a DP gage across the pump and suction diffuser. A single gage is used in this application connected to pipes from three locations: prior to the diffuser, in between the pump and diffuser, and after the pump. This gage reads absolute pressures at the different locations at different points in time. The absolute pressures are then subtracted to find the differential pressure across the desired component. Having a single gage instead of multiple gages ensures an accurate DP even if the gage is not reading the proper absolute pressure. Details such as the single gage are implemented to ensure the future maintenance team having access to adequate knowledge about the operating conditions of the system.

3.2 Heating Systems

The central heating system for the building is served by a hot water system containing two identical 1,500-MBH non-condensing boilers. In accordance with initial design goals the boilers are high efficiency each with two variable frequency drive secondary pump motors. The boilers are designed for a supply water temperature of 180°F. Hot water supply temperatures vary based on OA temperature. In addition to the central heating system, the boilers serve several cabinet unit heaters and local reheat coils at the Variable Air Volume (VAV) boxes.

Previous experiences by the mechanical engineer had shown that school reception areas are more likely to receive complaints about not falling within the bounds of the occupants desired thermal comfort region. For this reason the design called for a 1280 MBH electric radiant panel in the reception area that included a thermostat that could be controlled by the occupant.

3.3 AHU-1

The Air Handling Units (AHU's) in this building are responsible for supplying adequate amounts of OA and conditioning the air to account for the bulk of the space latent and sensible loads.

Air Handling Unit 1 supplies the three-story classroom wing of the building. The unit has a single supply VAV fan that moves 22,000 cfm with a static pressure drop of six in WG. This fan is responsible for the pressure drop from the OA intake to each of the VAV boxes. From there the individual VAV boxes supply an adequate pressure to supply the air to the individual spaces. The return fan has a capacity of 19,000 cfm with a design static pressure drop of 3 in WG. This fan draws the air from a plenum return to a short length of duct where it is then either blown into the mixed air supply or blown out of the building as exhaust air.

The design mixed air temperature for the chilled water coil is 81.1°F DBT and 66.1°F WBT. 936 MBH of cooling is required to bring the supply air conditions to 52.4°F DBT and 51.8°F WBT. A heating coil of 741 MBH of heating is required to bring the heating design entering coil temperature of 33.8°F to the winter supply temperature of 65°F. The supply air volume is determined by adjusting flow based upon the static pressure in the ductwork with a minimum flow volume above 11,066 cfm to ensure the required minimum value of OA is always supplied. The pressure in the ductwork changes as the local VAV boxes adjust airflow volumes based on space temperature.

To help reduce loads on the building and to achieve design goals, a total enthalpy wheel is used to pre-condition the OA. This design is effective because of a high percentage of OA. The high OA percentage directly correlates to a high Exhaust Air (EA) volume. Though necessary for ventilation, the high EA rate results in a rejection of Return Air (RA). Since the RA is many times closer to the SA conditions then the OA is, the high OA percentage increases the load across the heating and cooling coils. A total energy wheel transfers both latent and sensible energy between the RA and OA. Dumping energy into the OA stream in winter and extracting energy from the OA stream in summer reduces the load on both of the coils.

For transition seasons, AHU-1 enters economizer mode when the OA temperature is closer to the desired SA temperature then the RA temperature is. In economizer mode the system brings in above design OA to save on

energy use. Since the OA is closer to supply conditions then the RA is, the load on the coil is reduced.

3.4 AHU-2

Air Handling Unit 2 is responsible for conditioning air and supplying OA to the auxiliary classrooms and office section of the building. Because of the similarities between AHU-1 and AHU-2 a second detailed description will not be given. Specific details pertaining to AHU-2 can be found in Appendices A & B.

3.5 AHU-3

Air Handling Unit 3 is responsible for maintaining the air supply and air conditions in the gymnasium and the gymnasium support area. Because of the high OA percentages and the availability of implementing a total energy wheel, AHU-3 is a 100% OA system. Because of the uniqueness of this system, the controls are determined directly by space temperature, not duct pressure. Other components of AHU-3 are similar to those of AHU's 1 & 2. Specific information about AHU-3 may be found in Appendices A & B.

4. System Requirements

In order to meet the design goals, the systems must meat minimum requirements for ventilation and have the capacity to maintain thermal comfort throughout the entire year. In previous assignments specific values of these components were analyzed and compared with design values.

4.1 Ventilation

A ventilation analysis of the building was done in accordance with ASHRAE Standard 62.1. Table 4.1 compares the ASHRAE 62.1 ventilation rates with the designed ventilation rates, based off of the Ohio Mechanical Code requirements. As shown, the ASHRAE 62.1 is not equal to the designed minimum OA. This is due to differing assumptions about how the spaces will be used along with different cfm requirements between ASHRAE 62.1 and the Ohio Mechanical Code. In general the design values for OA were lower then the ASHRAE 62.1 values. However, in AHU-3 the designed values were higher then the ASHRAE 62.1 calculated values.

TABLE 4.1 VENTILATION OVERVIEW

Unit	Type	Zone Served	Net Area Served [Sq.ft]	Calculated OA [cfm]	Designed OA [cfm]
AHU-1	VAV	Three story classroom wing	24,700	13,529	11,066
AHU-2	VAV	1st and 2nd floor classrooms and auxiliary areas	19,100	9,078	8,269
AHU-3	VAV	Gymnasium and the gymnasium support areas	6,900	6,535	8,632

Calculated OA values were based off of ASHRAE Standard 62.1. Designed OA values were based off of the Ohio Mechanical Code requirements.

4.2 Heating & Cooling

The design team incorporated ASHRAE Standard 90.1 as a guide for low energy usage. Therefore, all mechanical equipment was selected in accordance with ASHRAE Standard 90.1.

In a previous report, the heating and cooling loads were analyzed using Carriers Hourly Analysis Program, or HAP. These values were then compared

with the design values taken from design documents. Though the HAP analyzed chilled water was close to the design values, both the chilled water and boiler components of the system varied from the design documents. Factors causing these discrepancies include possible different assumptions about the building when the two analyses were done and the potential of different daily schedules for the building.

The chilled water system was designed for a maximum load of 165 tons, which resulted in a requested 170 ton chiller for the building. The distribution of these loads for both the design and the HAP analyzed values are shown in Table 4.2.

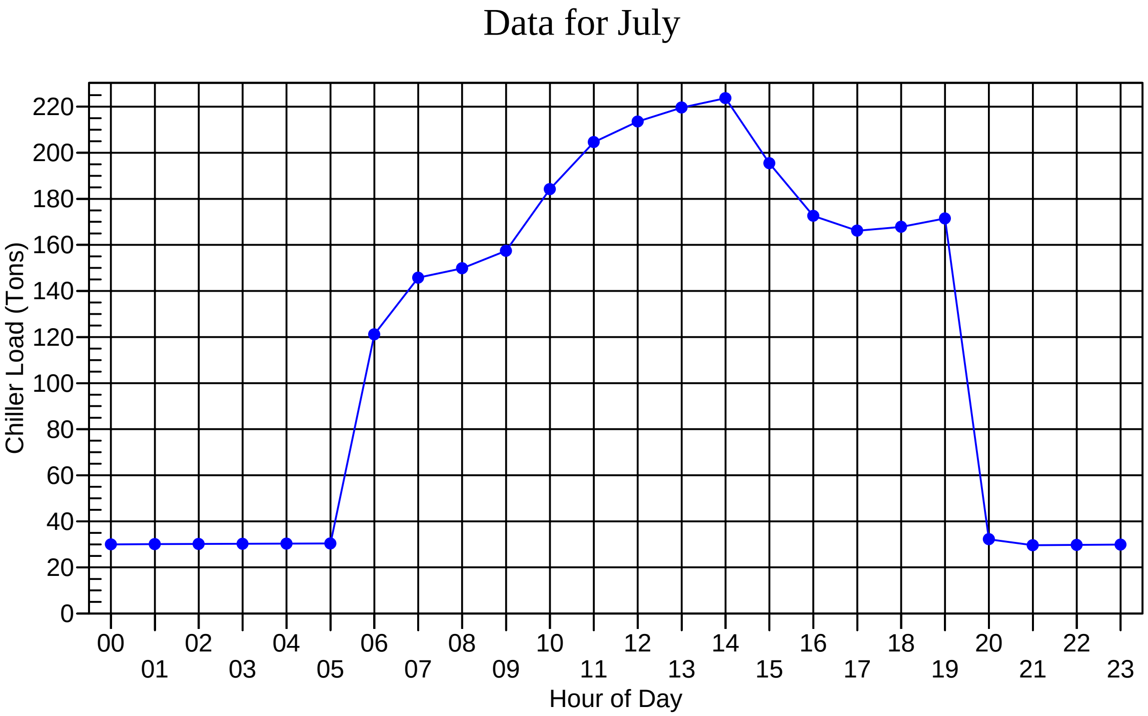
TABLE 4.2 Chilled Water Load Distribution

Air System	Analyzed Cooling Coil Load [ton]	Design Cooling Coil Load [ton]
AHU-1	71.9	64.6
AHU-2	66.5	66.5
AHU-3	31.3	45.6

Compares the design and analyzed cooling coil loads for each of the three major air handling units.

The chiller design is based upon what is called the design day. For this building, the design day occurred in July. The chiller load profile for this day is shown in Table 4.3. Typically load profiles only show a single peak. However, this design day contains the typical peak in the middle of the day, and a second, smaller peak in the afternoon. This second peak is due to a major influx of people into the gymnasium late in the day.

TABLE 4.3 Cooling Design Day Load Profile



Chiller load profile for the chiller design day.

The analyzed hot water load distributions were significantly lower then the design required hot water loads. Only AHU-2 was close to the design documents load capacity. Again, these values most likely deal with differing assumptions about the building and the mechanical system or contain a mistake in the HAP analysis. The distribution of these values for the major Air Handling units are shown on Table 4.4.

TABLE 4.4 Hot Water Load Distribution

Air System	Analyzed Heating Coil Load [MBH]	Design Heating Coil Load [MBH]
AHU-1	253.1	741
AHU-2	567.6	603
AHU-3	229.2	1167

Compares the design and analyzed heating coil loads for each of the three major air handling units.

5. Annual Energy Consumption & Costs

The annual energy consumption and costs of the building were analyzed using the HAP program. As shown on Figure 5.1 and Table 5.1, the major energy cost component of the building was the Air System Fans, followed by Lighting Cooling. The fan component in this building is high because of a high minimum OA flow that must be maintained even while other components of the system such as the chiller and boiler are not in use. Table 5.1 also outlines the cost per area of energy each year in the building. The total HVAC cost per square foot of the building was $1.192/ft^2, or 73.3% of the buildings total energy use.

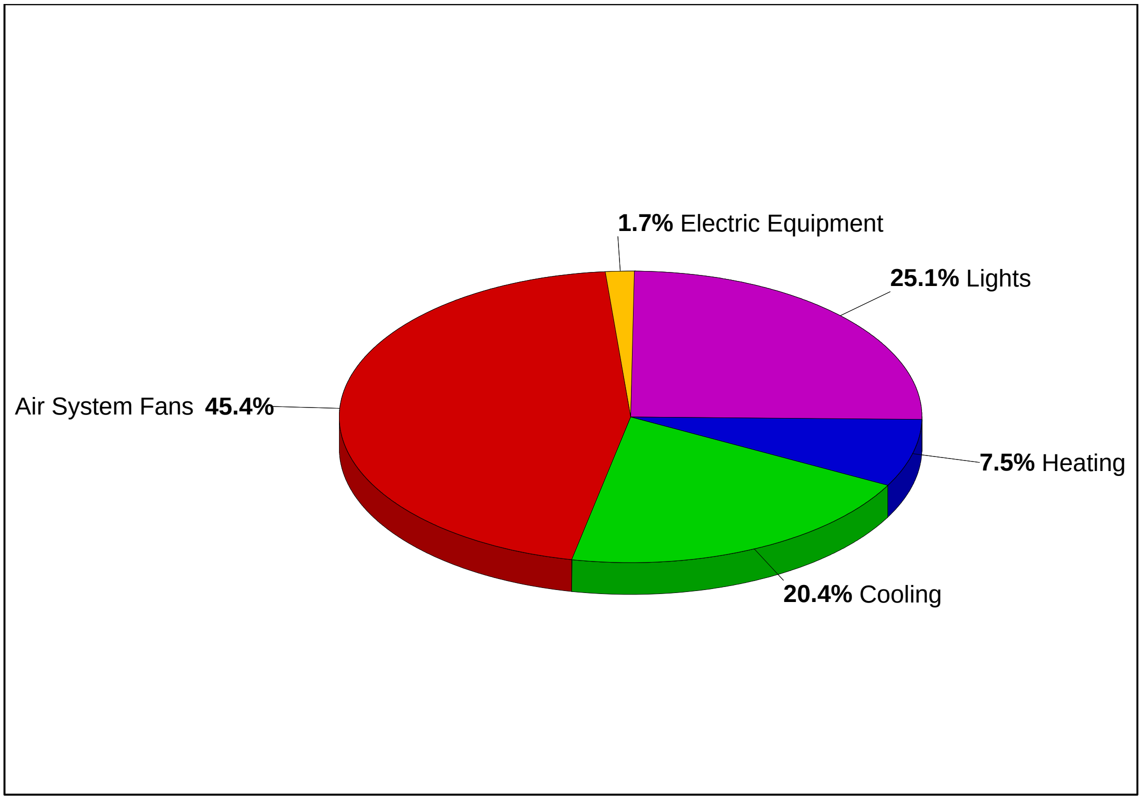


FIGURE 5.1 Percentage Annual Energy Costs by Components

TABLE 5.1 Annual Energy Cost, Annual Cost Per Area, and Percent Total Annual Energy Cost by Component.

Component	($)	($/ft²)	(%)
Air System Fans	49,077	0.738	45.4
Cooling	22,105	0.332	20.4
Heating	8,107	0.122	7.5
HVAC Sub-Total	**79,289**	**1.192**	**73.3**
Lights	27,101	0.407	25.1
Electric Equipment	1,790	0.027	1.7
Non-HVAC Sub-Total	**28,892**	**0.434**	**26.7**
Grand Total	**108,180**	**1.626**	**100**

Figure 5.2 looks at the expected energy costs of the building on a monthly basis. As expected, the cooling costs peak in July and August, while the much smaller heating costs peak during the winter months. As predicted by the analysis of Figure 5.1, the fan energy costs is relatively constant throughout the year.

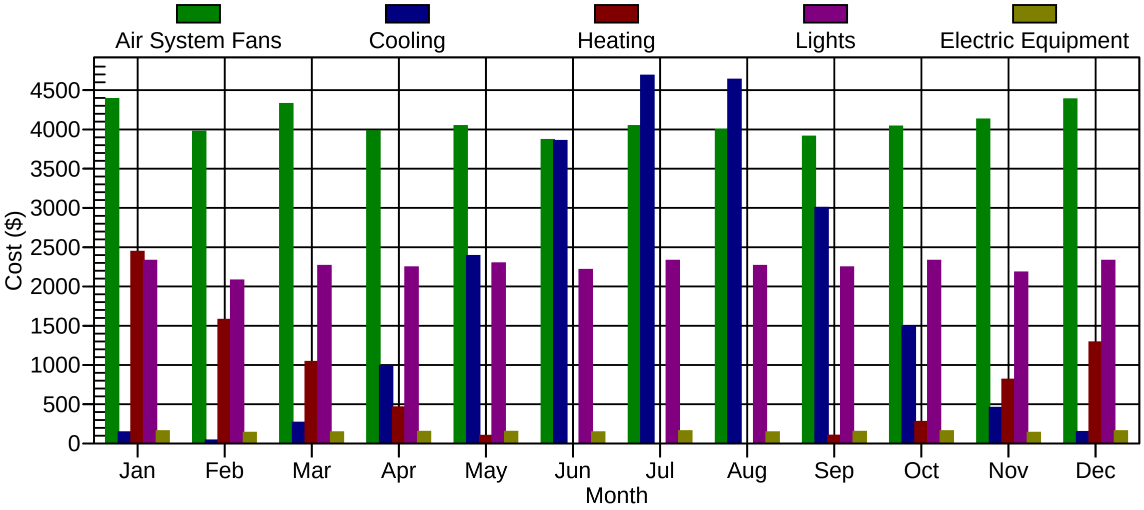


FIGURE 5.2 Monthly Energy Costs by Component

Finally, Table 5.2 displays the energy usage in terms of energy. The HVAC system is shown to consume 3,541,551 kBTU/yr. This correlates to 53.235 kBTU/ft^2/yr. Because energy rates per kBTU are different for the multiple energy sources, Table 5.2 shows the annual HVAC energy consumption to be 75% of the total building usage, slightly higher then the annual HVAC energy costs percentage. These values also only include equipment that is directly related to the building systems. Components such as domestic hot water and gas-fired stoves are not included in this analysis.

TABLE 5.2 Annual Energy Consumption and Annual Consumption Per Area by component

Component	Site Energy (kBTU)	Site Energy (kBTU/ft²)	Source Energy (kBTU)	Source Energy (kBTU/ft²)
Air System Fans	2,000,598	30.073	7,144,992	107.402
Cooling	901,103	13.545	3,218,223	48.376
Heating	639,850	9.618	639,850	9.618
HVAC Sub-Total	**3,541,551**	**53.236**	**11,003,065**	**165.396**
Lights	1,104,782	16.607	3,945,649	59.31
Electric Equipment	72,981	1.097	260,647	3.918
Non-HVAC Sub Total	**1,177,763**	**17.704**	**4,206,295**	**63.228**
Grand Total	**4,719,313**	**70.94**	**15,209,361**	**228.624**

6. Critique

Generally, the mechanical system design obtains the initial goals of the project team. Though no building is designed to an optimal standard, this building does deliver the necessary components to fulfill the owners expectations.

To ensure a minimum energy use for the building, the mechanical designers implemented a sophisticated control system to reduce wasted energy. They also implemented three complete energy wheels to recover both latent and sensible energy from the exhaust air. However, the energy consumption using only one chiller may be significantly higher then using two staged chillers. This is because, generally speaking, the maximum efficiency point of a chiller is at full capacity. The load profiles of a chiller express that typical chiller usage is weighted towards the bottom half of its capacity and only achieves full load capacity a few hours each year. By strategically staging multiple chillers the designer may be able to run each chiller closer to design capacity, resulting in an increased overall efficiency. This design will also increase initial costs and increase the amount of space required for the mechanical equipment. It is likely that the designer chose to sacrifice reduced energy consumption because of a mixture of these two reasons.

Thermal comfort was achieved in the building by properly controlling indoor air temperature and humidity. A sophisticated control system, research on the building type, and experience with similar buildings were the key elements to achieving the thermal comfort goal. The controls system helps to ensure comfortable living conditions at all times in the space, including the early hours of occupancy by a Startup period that adjusts as the response characteristics of the building are learned. By research and experience, the designer was able to make predictions about the loads in the building. They also implemented special controls in areas that previous experience had shown a difficulty in maintaining thermal comfort. The primary example of this is the radiant panel in the reception area with local thermostat control. As a VAV system, the building does risk uncomfortable humidity levels under low load conditions. However, because of the presence of total enthalpy wheels, this effect should be kept to a minimum.

The building controls are also effective in maintaining adequate amounts of OA to each of the spaces. Though the design minimum OA does seem to be

bellow that required by ASHRAE 62.1, it does meet with code and was determined to meet the design goals by the design team.

The mechanical team was also successful in confining the mechanical equipment area to a level that was acceptable to the owner and architect. Though initial mechanical room areas were exceeded, the mechanical designer worked with the architect and owner effectively to ensure the expectations for the building were met. These changes required the addition of two mechanical mezzanines, which drastically affected the exterior design of the building. However, the architect was able to work with these adjustments and was pleased with the final outcome of the building.

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Appendix A- System Diagrams

DIAGRAM DEFINITIONS

AS-1	350 gpm air separator
AS-2:	260 gpm air separator
BLR-1,2	Boilers
CHLR-1	Air cooled chiller
CHWP-1	Chilled water pump
CPF	Actual definition of this component not found in design documents, however it is assumed that this device controls the flow of water in the system to ensure that the chiller has a constant supply of 300 gpm
CV	Control valve
P-BLR-1,2	Boiler recirculation pumps
ET-1	Bladder type expansion tank, 33.6 gallons
ET-2	Bladder type expansion tank, 150 gallons
FCV	Flow control valve
FX	Flow meter
HWP-1,2	Heating hot water pumps
M	Motorized component
PI	Pressure indicator
SD-1,2	Suction diffuser
TE	Temperature element
TI	Temperature indicator
VFD	Variable frequency drive

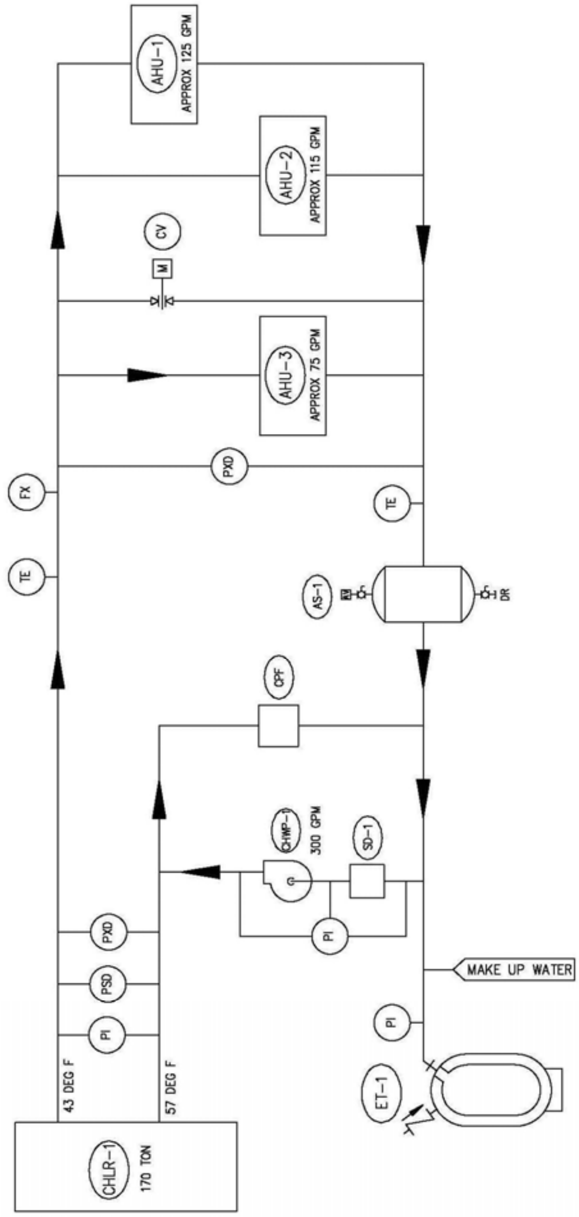

FIGURE A.1 CHILLED WATER SYSTEM SCHEMATIC

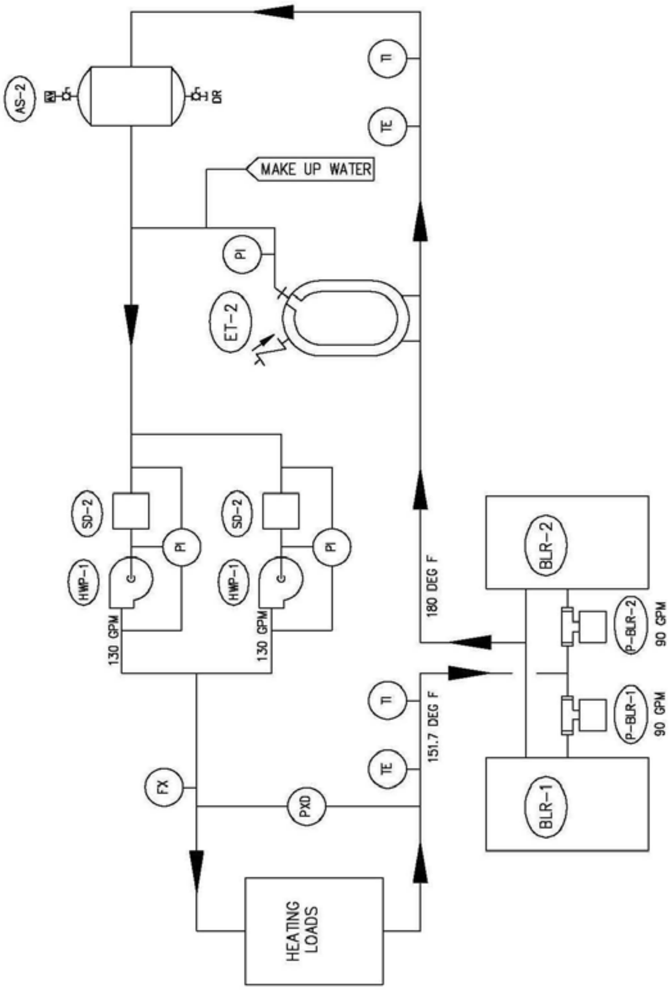

FIGURE A.2 HOT WATER SYSTEM SCHEMATIC

Appendix B- Equipment Tables

TABLE B.1 AIR HANDLING UNIT EQUIPMENT COMPONENTS

DESIGNATION	UNIT LOCATION	FAN DATA			MINIMUM OA FLOW						
		FAN TYPE	TOTAL AIR FLOW CFM	NO. FAN SPEEDS	CFM	COIL TYPE	COIL EAT DB, DEG F	COIL EAT WB, DEG F	COIL LAT DB, DEG F	COIL LAT WB, DEG F	COIL LOAD MBH
AHU-1	3RD FLOOR	SUPPLY	22,000	VFD	11,066	DTW C	81.1	66.1	52.4	51.8	936
	MER 313	RETURN	19,800	VFD	11,066	DTW H	33.8		65		-741
AHU-2	2ND FLOOR	SUPPLY	18,000	VFD	9,000	DTW C	81.1	66.1	52.0	51.6	775
	MER 225	RETURN	16,200	VFD	9,000	DTW H	34.0		65		-603
AHU-3	GYM	SUPPLY	12,000	VFD	12,000	DTW C	81.6	66.6	51.7	51.3	547
	MEZZANINE	RETURN	12,000	VFD	12,000	DTW H	0.0		55		-713
	MER 228					REHEAT	50.0		85		-454

TABLE B.2 HOT WATER BOILERS

DESIGNATION	BOILER TYPE	BURNER TYPE	FUEL TYPE	HEATING AREA, SF	MIN INPUT MBTU/HR	MIN OUTPUT MBTU/HR	MAX OPERATING TEMP, F	WATER FLOW DATA DESIGN GPM	WATER FLOW DATA ENTERING TEMP, F	WATER FLOW DATA LEAVING TEMP, F
BLR-1	COPPER TUBE	MODULATING	NAT GAS	206	1,500	1,275	240	130	151.7	180
BLR-2	COPPER TUBE	MODULATING	NAT GAS	206	1,500	1,275	240	130	151.7	180

TABLE B.3 CHILLED WATER CHILLER, CHLR-1

EVAPORATOR				MISCELLANEOUS	
NOMINAL CAPACITY, TONS	FLUID	FLOW, GPM	EWT / LWT DEG F	MAX A WEIGHTED SOUND PRESSURE @ 30 FEET (dBA)	REFRIGERANT TYPE
170	WATER	300	57.0 / 43.0	70	HFC-134a

CONDENSER					
EAT, DEG F	NO. OF FANS	NOMINAL SIZE COMP #1 TONS	NOMINAL SIZE COMP #2 TONS	RLA COMP #1	RLA COMP #2
95	12	85	85	133	133

TABLE B.4a TOTAL AIRSIDE ENERGY WHEELS (OA Flow Stream)

DESIGNATION	FLOW RATE CFM	SUMMER EFF %	WINTER EFF %	SUMMER		WINTER	
				EAT, DB/WB, DEG F	LAT, DB/WB, DEG F	EAT, DEG F	LAT, DEG F
ERU-1	11,066	85.8%	90.8%	88.0 73.0	82.1 67.2	0	51.2
ERU-2	9,000	83.4%	86.0%	88.0 73.0	82.2 67.2	0	48.6
ERU-3	12,000	79.3%	81.9%	88.0 73.0	81.6 66.6	0	53.8

TABLE B.4b TOTAL AIRSIDE ENERGY WHEELS (EA Flow Stream)

DESIGNATION	MIN FAN MOTOR HP	WHEL S. P. IN. WG	FLOW RATE CFM	SUMMER	WINTER	WHEEL MOTOR HP	NO. WHEEL SPEEDS
				EAT, DB/WB, DEG F	EAT, DEG F		
ERU-1	15.0	0.55	8,800	79.8 64.4	69.8	1/2	VFD
ERU-2	15.0	0.62	7,200	79.7 64.4	69.7	1/2	VFD
ERU-3	15.0	0.77	12,000	76.5 63.2	71.1	1/2	VFD

TABLE B.5 PUMPS AND SUCTION DIFFUSERS

DESIGNATION	SERVES	PUMP DATA						ELECTRICAL DATA		SUCTION DIFFUSERS	
		FLOW, GPM	FLUID	MIN HEAD, FT. FLUID	RPM	MIN EFFIC %	TYPE	MOTOR HP	VOLT/FREQ/PH	DESIGNATION	SUCTION SIZE, IN.
CHWP-1	CHLR-1 / CHW LOOP	300	WATER	90	1,750	68	VERTICAL INLINE	15	460 / 60 / 3	SD-1	6
HWP-1	HHW LOOP	130	WATER	100	1,750	60	VERTICAL INLINE	10	460 / 60 / 3	SD-2	2
HWP-2	HHW LOOP	130	WATER	100	1,750	60	VERTICAL INLINE	10	460 / 60 / 3	SD-2	2
BRP-1	BLR-1 (MIN FLOW)	90	WATER	25	1,750	60	INLINE CIRCULATOR	1-1/2	460 / 60 / 3	--	--
BRP-2	BLR-2 (MIN FLOW)	90	WATER	25	1,750	60	INLINE CIRCULATOR	1-1/2	460 / 60 / 3	--	--
HWP-AHU-1-1	AHU-1 DTW COIL	75	WATER	20	1,150	45	VERTICAL INLINE	1	460 / 60 /3	--	--
HWP-AHU-2-1	AHU-2 DTW COIL	65	WATER	20	1,150	45	VERTICAL INLINE	1	460 / 60 /3	--	--
HWP-AHU-3-1	AHU-3 DTW COIL	50	WATER	20	1,150	45	VERTICAL INLINE	1	460 / 60 /3	--	--