Final Report

David H. Koch Institute for Integrative Cancer Research

Massachusetts Institute of Technology

Cambridge, Ma

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1.0 Executive Summary

The David H. Koch Institute for Integrative Cancer research lab presents a multitude of HVAC design challenges. With MIT's expectation of achieving LEED Gold Certification, design engineers were forced to provide innovative design solutions, resulting in an energy efficient HVAC system. In this report, the mechanical systems of the Koch Institute are evaluated, critiqued and redesigned.

The existing design for the Koch Institute utilizes a central VAV ventilation cooling system, with heat pipe heat recovery between the supply and exhaust airstreams. Comprised of (10) 50,000 CFM Factory Built-Up AHU and EAHU's, the central system is responsible for the majority of heating and cooling in the building. An additional 13 packaged modular air handlers provide spot cooling for mechanical and electrical rooms and stair-shafts. The building is heated with hot water reheat coils and a perimeter radiant panel heating system. High intensity load and perimeter spaces are conditioned via fan coil units and chilled beam induction cooling to aid the central VAV system.

Supplying energy to this system is MIT's cogeneration plant which utilizes a 25MW Combustion Turbine Generator. This generator provides 80% of the electricity consumed by the campus by burning Natural Gas purchased from NSTAR based on a large commercial service rate (G-43). A heat recovery steam generator utilizes the exhaust from the generator creating high pressure steam. This steam is distributed to campus as well as to absorption chillers that use the steam to create chilled water for the campus which is fed through 24" mains.

In this report, alternative methods were evaluated to provide spot cooling and stair heating/cooling. A vertical closed loop ground source heat pump was designed to provide chilled water to the 13 packaged modular air handlers for spot cooling of the penthouse, basement, stairs and electrical rooms. The required length of pipe for the GSHP was sized utilizing equations from *Chapter 32 of the 2007 ASHRAE Handbook-HVAC Applications* entered into EES. The resulting ground source heat pump design cost an additional **$191,765** and provided an annual savings of **$87,651**. Therefore, the payback for the system would be **2.21** years.

A glycol run around heat recovery loop was added to the design to recover energy from 12 exhausts to heat the east and west stairwells. The existing design employed (4) 3,600 cfm packaged air handlers to heat and cool the stairs. The heat recovery loop added two preheating coils to the stair pressurization fans that supplied outdoor air to the space, allowing for the removal of 2 AHU's. With the savings from the elimination of 2 AHU's, the heat recovery loop costs **$4,143** with a **4.29** year payback.

The Construction Management Breadth of this report consists of a borehole optimization study that calculates the optimum number and depth of boreholes utilizing pricing estimates from *RS Means Mechanical Cost Data – 2009*. This study compares construction duration with overall pricing and provides alternate drilling schemes if problems arise in the drilling process.

The Electrical Breadth evaluates the increased load on the building electrical system with the addition of mechanical equipment loads. A distribution panel and feeders were sized to incorporate all of the new mechanical equipment into the existing system. A one line schematic shows how the new distribution panel is tied into the building electrical system.

2.0 Project Information

2.1 Design Goals

The Massachusetts Institute of Technology developed the David H. Koch Institute to integrate the work of their prestigious engineers and cancer biologist under one roof. Combining the life scientist's understanding of cancer biology with the analytical skills of the engineers creates and environment for success. Research performed in the building requires very sophisticated equipment and strictly controlled environments. To meet this challenge, MIT desired a LEED Gold Certified building be designed to efficiently meet all the needs of the building occupants.

2.2 Location

The Koch Institute is located on MIT's campus in Cambridge, Ma surrounded by MIT's prestigious science department buildings. The site is highlighted in yellow in the aerial view of MIT's campus shown to the left in ***Figure 1*** to the left. It will become a signature building for MIT with a large presence on Main Street. The addition of a quad adjacent to the Koch Institute provides the campus with usable outdoor space for students and faculty.

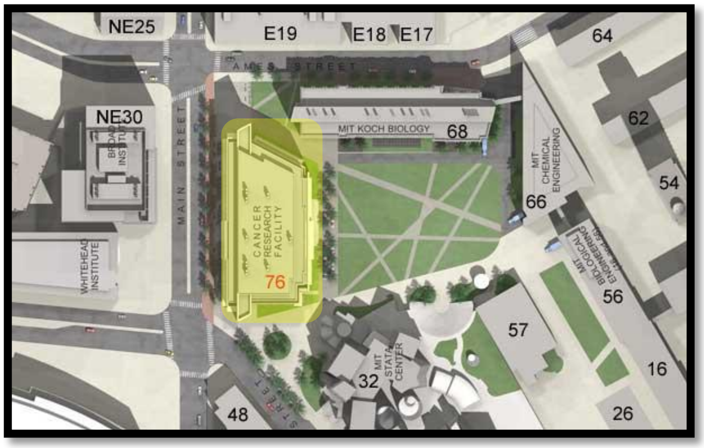


Figure 1 *–Project Site*

2.3 Project Team

Owner	*Massachusetts Institute of Technology*
Architect	*Ellenzweig Architecture*
MEP Engineer	*Bard, Rao + Athanas Engineers, LLC*
Structural Engineer	*LeMessurier Consultants, Inc.*
Lighting Consultant	*Lam Partners, Inc.*
Plumbing/Fire Protection/Codes	*R.W. Sullivan Engineering*
Civil Engineer	*Nitsch Engineering, Inc.*
Leed/Sustainable Design	*The Green Engineer, LLP*
Landscape Architect	*Reed Hilderbrand Associates, Inc.*
Telecommunications	*Communications Design Group, Inc.*

3.0 Building Overview & Existing Conditions

3.1 Architecture

Standing 7 stories tall with a penthouse and basement, the Koch Institute blends well with existing campus buildings utilizing similar massing and materials. It is designed to be welcoming and easily navigable through prominent entries and a transparent appearance. Curtain walls with varying mullion layouts give the main street façade a modern exterior. Varying thicknesses and verticality between facades creates unique views of the building around its entire exterior.

Figure 2 to the left is a rendered view of the Koch Institute from Main Street, depicting some of the fore mentioned architectural characteristics. The program includes research and core laboratories, vivarium, conference facilities, meeting spaces, cafeteria as well as offices and administrative functions. The administrative offices and meeting rooms are all located on the ground floor along with the Core Laboratories.

Figure 2 *–Main Street View*

3.2 Sustainability Features

The Koch Institute for Integrative Cancer Research is designed to achieve LEED Gold certification. The design therefore includes a plethora of sustainable features both architecturally and within the building systems. These sustainable features include:

- Storm and water filtration system
- Reflective roof materials to reduce the heat island effect
- Heat recovery methods incorporated into the HVAC system
- Right sizing of HVAC equipment and utilization of a VAV system to reduce energy use
- Low-emitting materials including adhesives, sealants, paints and carpets
- Low flow fume hoods to reduce ventilation requirements
- Low velocity duct work to reduce fan energy
- Construction waste management plan that recycles and salvages waste
- Exterior solar shading, light shelves for day lighting
- High performance glazing and enhanced building insulation

3.3 Building Enclosure

The building is primarily enclosed by an aluminum and glass curtain wall system. There is a large amount of metal paneling accented by aluminum and stone coping.

3.4 Electrical System

The electrical utility is connected via an existing MIT manhole rated at 15 KV to a new manhole adjacent to the building. From the manhole, the service enters the basement in a concrete encased ductbank, which terminates in a pull box. The service is then fed to 2 double-ended substations (A & B) through G & W 15 KV two position load interrupting switches. Here the power is stepped down to 480Y/277 V through 2000 KVA frame size transformers and distributed throughout the building. Substation A also feeds optional standby receptacle and lab equipment loads through a 1600A 4 pole ATS as well as emergency lighting through a 400A 4 pole ATS. Substation B also feeds emergency and optional standby loads through 6 ATS' of varying sizes. Emergency power is provided by a 2000KW/2500KVA diesel generator powering its own standby switchgear.

3.5 Lighting System

The lighting of the Koch Institute is energy efficient, utilizing mainly linear fluorescent T5 and T8. The few exceptions to this general lighting design are public spaces, labs, MRI room and darkroom. Most public spaces located on the first level employ halogen sources to light the space. The labs, MRI room and darkroom all require special luminaires due to the nature of the work performed and the sensitivity of equipment within the space. All of these lighting systems are controlled by Lutron lighting control panels and dimmers. The control system uses photocells, occupancy sensors and time of day control to optimize the energy consumption of the system.

3.6 Structural System

The superstructure of the Koch Institute employs individual steel columns ranging from W14x43 to W14x233. These columns tie into the orthogonal steel bracing system that provides lateral force resistance throughout the building. The substructure consists of concrete column footings, a foundation wall and slab on grade construction. The floor system is made up of 4.5" normal weight concrete on a 3" deep, 18 gage minimum composite steel deck. This floor system is supported by a beams and girders that vary in size due to the complexity of the layout and column spacing. The interior bays generally utilize W24x55 and W24x68 steel beams to carry the load to the girders. Exterior bays utilize W16x31 beams for the 26'-2" sections and W21x50 for the 30'-2" sections to carry the loads to the girders.

3.7 Fire Protection

The fire alarm system utilizes multiple ADA compliant audio/visual alarms. These alarms send out both a strobe and audio alert. The animal holding spaces use a chime tone indication differing from the rest of the system. A 125 horsepower fire pump supplies water throughout the fire protection system to maintain the prescribed flow rate (gpm) to all sprinkler heads. The fire pump receives its power through a 1600A, 3 pole ATS that is fed by both Substation B and the Emergency Power Switchgear. This ensures that the pump will always have sufficient power in the case of an emergency.

3.8 Transportation

The building can be entered through vestibules leading to the lobby on both the North and South facades. Vestibules on the Northwest and Northeast corners of the building grant access into the gallery space and West and East stair shafts respectively. There are (2) passenger elevators which open to the lobby and rise from the basement to the sixth level. Adjacent to the passenger elevators is a service elevator and vivarium elevator that are not accessible from the lobby. These elevators are reached through vestibules on each floor, branching off of the northern corridor. The vivarium elevator terminates on level seven. The service elevator is the only of the four to span the entire length of the building, restricting access to the penthouse.

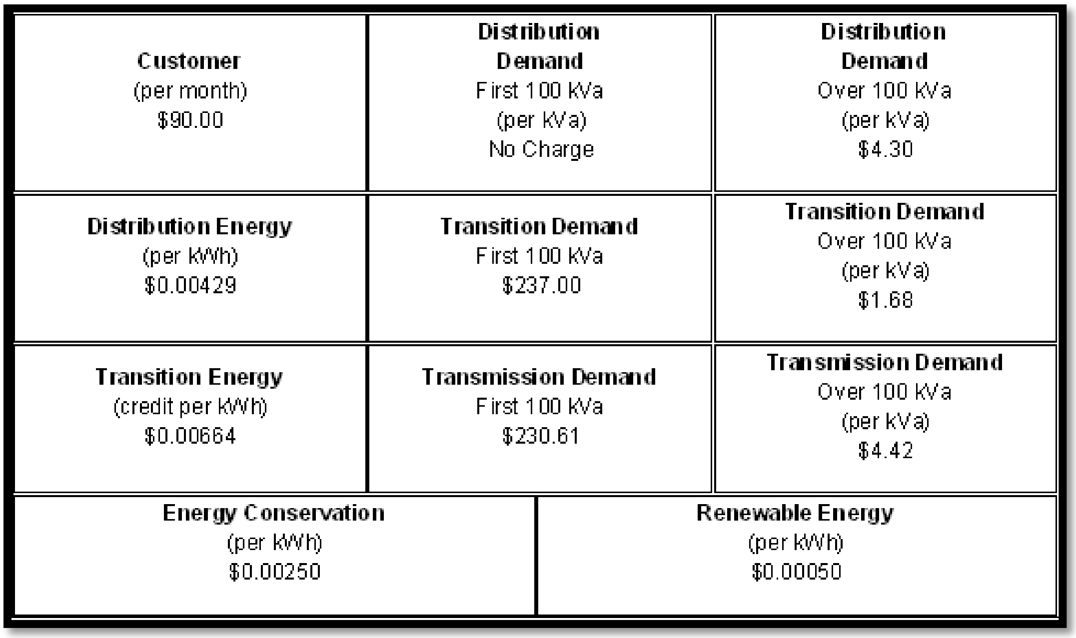
3.9 Telecommunications

The Koch Institute telecommunication service is fed into the basement through an existing manhole. The telecommunications is split into two zones and consists of a main distribution frame (MDF) in the basement and multiple intermediate distribution frames (IDF) located throughout the building. Each floor has and east and west IDF room providing telecommunications to its respective zone. Every IDF room receives data from a 48 strand armored singlemode fiber optic riser cable that is terminated at a rack mounted fiber panel. From the IDF rooms the data is distributed horizontally throughout the zone through (6) 4”conduits typ. providing telecommunications outlets.

3.10 Utility Rates

3.10.1 Electrical Rates

The utility rates for the Koch Institute are based on NSTAR's Large General Time-of-Use – 13.8 kV Service (G3). This service is required for buildings with loads exceeding 100 kW for at least 12 consecutive billing months. ***Figure 3*** below breaks down the specifics of this service.

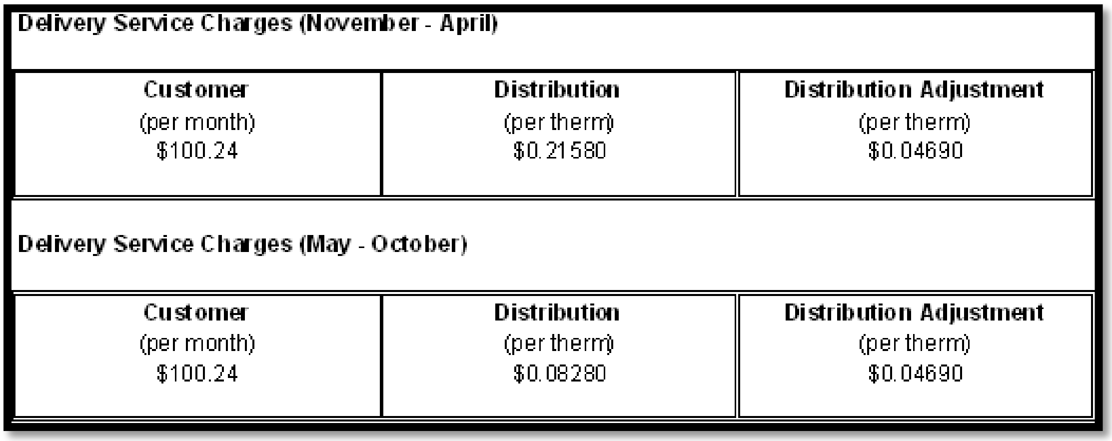


Customer (per month) $90.00	**Distribution Demand** First 100 kVa (per kVa) No Charge	**Distribution Demand** Over 100 kVa (per kVa) $4.30
Distribution Energy (per kWh) $0.00429	**Transition Demand** First 100 kVa $237.00	**Transition Demand** Over 100 kVa (per kVa) $1.68
Transition Energy (credit per kWh) $0.00664	**Transmission Demand** First 100 kVa $230.61	**Transmission Demand** Over 100 kVa (per kVa) $4.42
Energy Conservation (per kWh) $0.00250	**Renewable Energy** (per kWh) $0.00050	

Figure 3 *–NSTAR Electric Rates*

3.10.2 Natural Gas Rates

The natural gas supplied to MIT's gas turbine is supplied by NSTAR based on a Low Load Factor General Service – Large (G43) categorization. This rate is for non-residential customers consuming at least 100,000 therms of gas per year. The cost of gas is also factored into these rates and is set at **$0.7703/therm** as of November 1, 2009. ***Figure 4*** breaks down the specifics of this service.



Delivery Service Charges (November - April)		
Customer (per month) $100.24	**Distribution** (per therm) $0.21580	**Distribution Adjustment** (per therm) $0.04690
Delivery Service Charges (May - October)		
Customer (per month) $100.24	**Distribution** (per therm) $0.08280	**Distribution Adjustment** (per therm) $0.04690

Figure 4 *–NSTAR Natural Gas Rates*

4.0 Existing Mechanical Systems Summary

4.1 Introduction

A central 98% outdoor air VAV ventilation/cooling system conditions the Koch Institute, utilizing heat recovery between the supply and exhaust air streams. The remaining 2% is made up with two small return fans that dump a total of 30,000 cfm into the outdoor air plenum that the large units pull from. The central VAV ventilation/cooling system is made up of (10) 50,000 cfm factory built-up AHU's coupled (10) 50,000 cfm EAHU's, and is responsible for supplying and exhausting the entire building. The building is heated through hot water reheat coils and a perimeter radiant panel heating system. High intensity load and perimeter spaces are conditioned with fan coil units and chilled beam induction cooling to supplement the central VAV system.

4.2 Design Criteria and Objectives

It is essential in the design of any HVAC System to ensure that all spaces are properly ventilated, meeting all requirements of the occupants. A good design can meet these ventilation requirements while also creating comfortable space conditions by controlling temperature and humidity to pre-determined levels. Due to the diversity of building and space types, every project presents new challenges which results in uniquely designed HVAC systems.

In the case of the Koch Institute, a number of critical space types and occupancy requirements drove the design. A large amount of laboratory and classroom spaces demanded that the HVAC system be capable of delivering large amounts of outdoor air to properly ventilate all spaces. Very large equipment loads required the design to adjust quickly to increased loads during equipment operation. Also, the nature and importance of the research being performed in the building called for a sophisticated, reliable emergency power system.

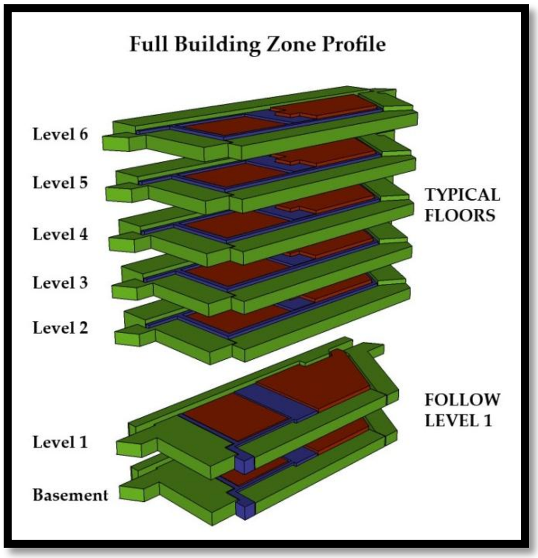


Figure 5 *–Building Zone Profile (level 7 not shown)*

Along with these space and occupancy criteria, the architecture of the Koch Institute presents additional challenges. The glass enclosed building is subjected to large amounts of solar gain, making all perimeter spaces critical. With a mechanical penthouse and two main shafts in the East and West sections of the building, the mechanical design engineers chose to employ a large centralized system that is divided between East and West service. With the exception of the seventh level, the spaces on each level were nearly identical and therefore could be treated similarly. Therefore, the architectural layout of the building dictated that the vivarium space on the seventh floor be conditioned separately than the other levels.

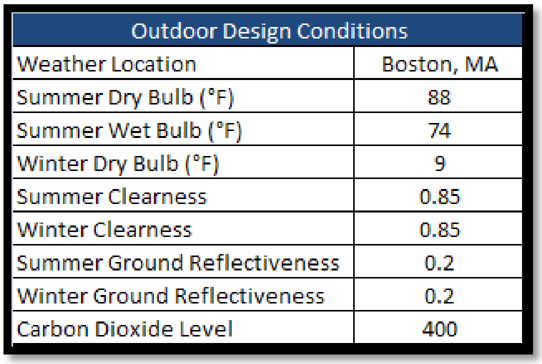
With all of these challenges comes another extremely important factor of the design, and that is the owners design intent. The Koch Institute is designed with the goal of achieving LEED Gold Certification. Energy conscious design makes up a large portion of LEED Credits, and therefore, the mechanical system must be very efficient in all areas.

Having all of these challenges and design requirements in mind, the design engineer has all the tools needed to design the optimum system for the owner. The components of the system must be selected to operate efficiently over the buildings lifetime to reduce large energy costs for the owner.

4.3 Outdoor and Indoor Design Conditions

The desired indoor conditions and the location specific outdoor conditions heavily influence the design of a building. The Koch Institute is located in Cambridge, MA where a New England climate produces harsh winters and hot summers. This area experiences the same outdoor conditions as Boston, MA which has the ASHRAE Weather Data found in ***Figure 6*** below.

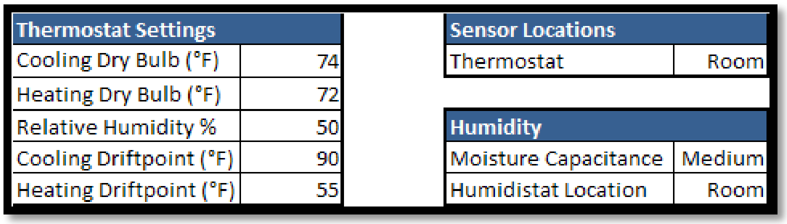
With summer temperatures in the high 80's and winter in the single digits, the building will be exposed to high heating and cooling loads. The system will have to overcome these loads to condition the spaces to desired thermal conditions, while also maintaining proper humidity levels. With laboratories and classroom space making up a large portion of the building, the indoor design conditions follow the requirements associated with these space types.



Outdoor Design Conditions	
Weather Location	Boston, MA
Summer Dry Bulb (°F)	88
Summer Wet Bulb (°F)	74
Winter Dry Bulb (°F)	9
Summer Clearness	0.85
Winter Clearness	0.85
Summer Ground Reflectiveness	0.2
Winter Ground Reflectiveness	0.2
Carbon Dioxide Level	400

Figure 6 *–Outdoor Design Conditions*

These indoor design conditions are shown in ***Figure 7*** to the left. The individual room temperatures may vary based on zone set points or changes in thermostat settings.



Thermostat Settings	
Cooling Dry Bulb (°F)	74
Heating Dry Bulb (°F)	72
Relative Humidity %	50
Cooling Driftpoint (°F)	90
Heating Driftpoint (°F)	55

Sensor Locations	
Thermostat	Room

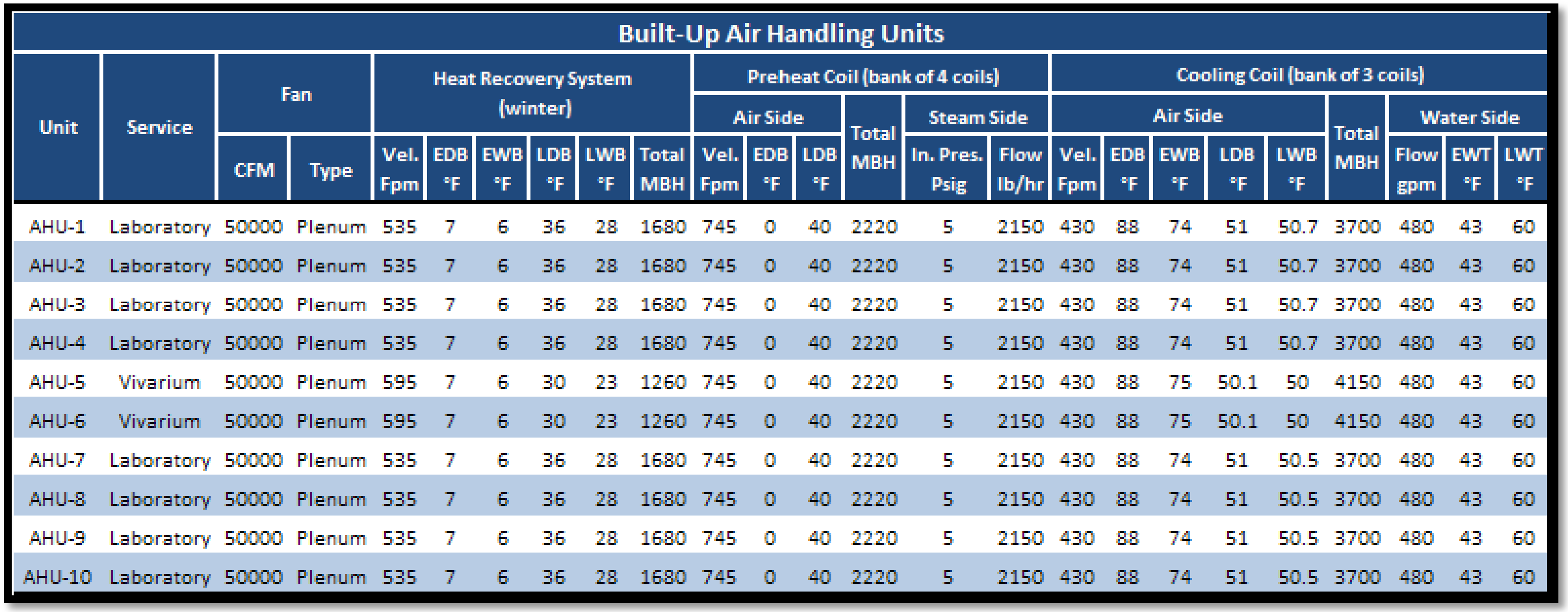
Humidity	
Moisture Capacitance	Medium
Humidistat Location	Room

Figure 7 *–Indoor Design Conditions*

The humidity levels in the spaces are controlled by dehumidification performed in the main air handling system in the penthouse. The only floor to need additional humidification is level seven due to its vivarium spaces and specific space needs. Therefore, level seven has its own dedicated air handlers AHU-5 and AHU-6 that are supplemented by individual ducted humidifiers that provide the appropriate humidity levels for the spaces they serve.

4.4 Air Supply System

The Primary air supply system utilizes (10) 50,000 CFM Factory Built-Up AHU's that utilize 98% outdoor air and 2% return air to the entire building. These units make up the entire central VAV ventilation/cooling system that was described in the introduction of this report. These air handlers are divided up into 3 groups, AHU – 1 to 4; AHU-5 & 6; and AHU-7 to 10. AHU's 1-4 deliver 200,000 cfm of conditioned air down the west shaft to the west zones of levels B-6. AHU-5 & 6 serve the seventh level vivarium spaces and AHU's-7-10 deliver 200,000 cfm of conditioned air down the east shaft to the east zones of levels B-6. These AHU's are summarized in the following table shown in ***Figure 8***. As can be seen, these are cooling units that utilize a heat recovery system from their respective exhaust airstream to pre-condition the incoming outdoor air.



Built-Up Air Handling Units

Unit	Service	Fan		Heat Recovery System (winter)						Preheat Coil (bank of 4 coils)						Cooling Coil (bank of 3 coils)								
										Air Side			Total MBH	Steam Side		Air Side					Total MBH	Water Side		
		CFM	Type	Vel. Fpm	EDB °F	EWB °F	LDB °F	LWB °F	Total MBH	Vel. Fpm	EDB °F	LDB °F		In. Pres. Psig	Flow lb/hr	Vel. Fpm	EDB °F	EWB °F	LDB °F	LWB °F		Flow gpm	EWT °F	LWT °F
AHU-1	Laboratory	50000	Plenum	535	7	6	36	28	1680	745	0	40	2220	5	2150	430	88	74	51	50.7	3700	480	43	60
AHU-2	Laboratory	50000	Plenum	535	7	6	36	28	1680	745	0	40	2220	5	2150	430	88	74	51	50.7	3700	480	43	60
AHU-3	Laboratory	50000	Plenum	535	7	6	36	28	1680	745	0	40	2220	5	2150	430	88	74	51	50.7	3700	480	43	60
AHU-4	Laboratory	50000	Plenum	535	7	6	36	28	1680	745	0	40	2220	5	2150	430	88	74	51	50.7	3700	480	43	60
AHU-5	Vivarium	50000	Plenum	595	7	6	30	23	1260	745	0	40	2220	5	2150	430	88	75	50.1	50	4150	480	43	60
AHU-6	Vivarium	50000	Plenum	595	7	6	30	23	1260	745	0	40	2220	5	2150	430	88	75	50.1	50	4150	480	43	60
AHU-7	Laboratory	50000	Plenum	535	7	6	36	28	1680	745	0	40	2220	5	2150	430	88	74	51	50.5	3700	480	43	60
AHU-8	Laboratory	50000	Plenum	535	7	6	36	28	1680	745	0	40	2220	5	2150	430	88	74	51	50.5	3700	480	43	60
AHU-9	Laboratory	50000	Plenum	535	7	6	36	28	1680	745	0	40	2220	5	2150	430	88	74	51	50.5	3700	480	43	60
AHU-10	Laboratory	50000	Plenum	535	7	6	36	28	1680	745	0	40	2220	5	2150	430	88	74	51	50.5	3700	480	43	60

Figure 8 *–Built-Up Air Handling Unit Summary*

On the following page, ***Figure 9*** shows a schematic of the supply system shows the three groups of built up AHU's without the exhaust system for simplification and reading purposes.

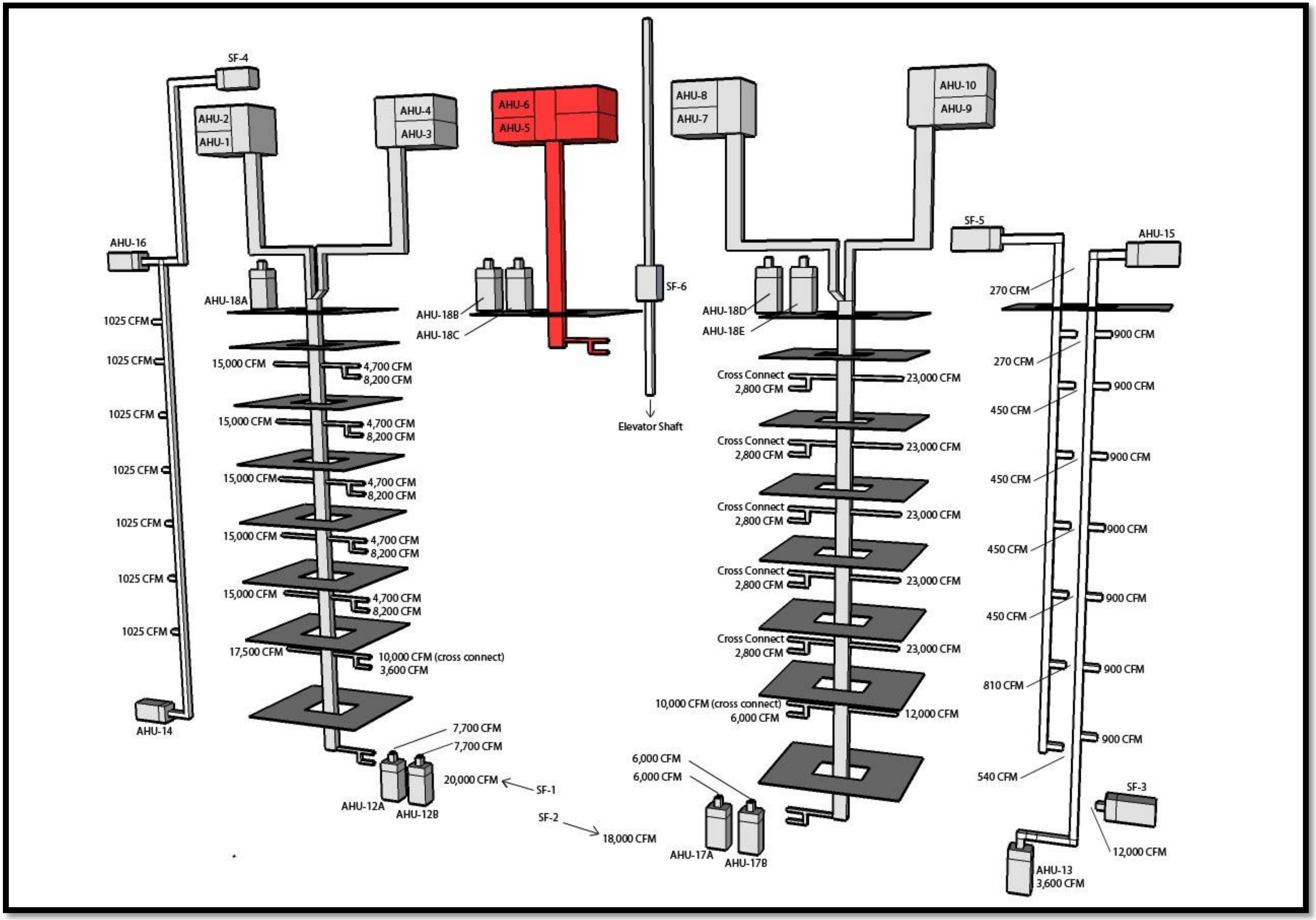


***Figure 9** –Supply Air Riser Diagram*

Figure 9 on the previous page shows the full layout of the central VAV ventilation/cooling system. The 10 large air handling units that make up this system are depicted in what will be the penthouse level of the Koch Institute. The air handling units shown in red (AHU-5 & 6) are not completely depicted in this picture due to their extensive humidification system. To simplify the drawing and maintain readability, a separate drawing for these air handling units was created and is shown below in ***Figure 10***.

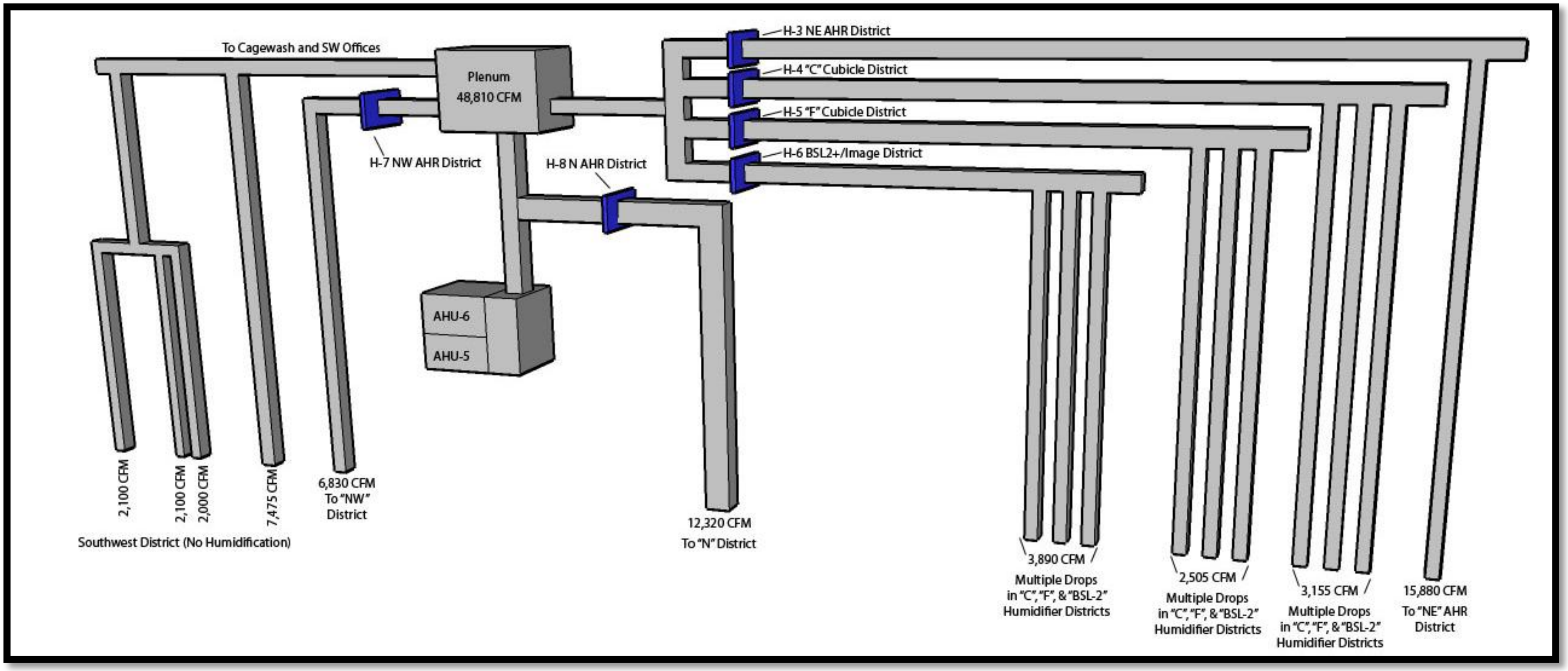
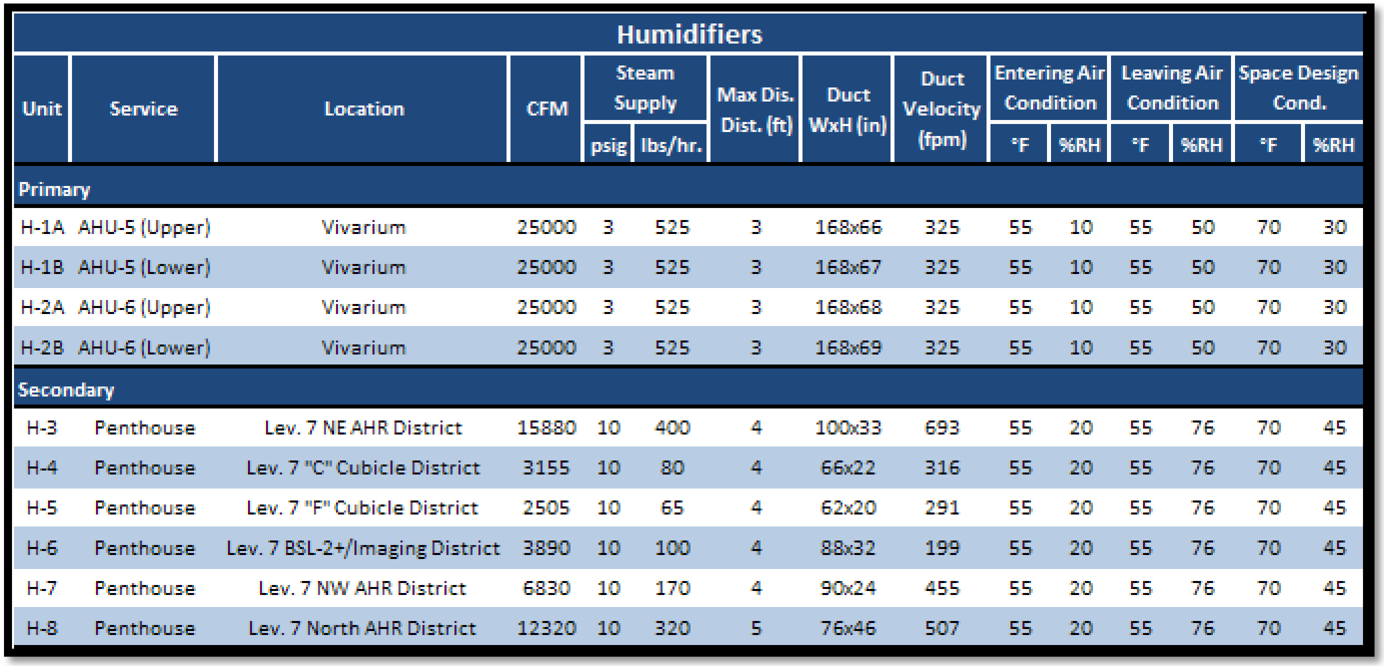


Figure 10 *–AHU 5 & 6 Supply Air Riser Diagram*

To maintain the desired space conditions on level 7, after leaving the (2) 50,000 cfm air handlers, the supply air is humidified by its respective humidifier shown in ***Figure 10***. Each humidifier is controlled by the space it is supplying described in the following table in ***Figure 11*** as "districts". AHU 5 & 6 have internal humidifiers that are also summarized in the table in ***Figure 11***.



Humidifiers

Unit	Service	Location	CFM	Steam Supply		Max Dis. Dist. (ft)	Duct WxH (in)	Duct Velocity (fpm)	Entering Air Condition		Leaving Air Condition		Space Design Cond.	
				psig	lbs/hr.				°F	%RH	°F	%RH	°F	%RH
Primary														
H-1A	AHU-5 (Upper)	Vivarium	25000	3	525	3	168x66	325	55	10	55	50	70	30
H-1B	AHU-5 (Lower)	Vivarium	25000	3	525	3	168x67	325	55	10	55	50	70	30
H-2A	AHU-6 (Upper)	Vivarium	25000	3	525	3	168x68	325	55	10	55	50	70	30
H-2B	AHU-6 (Lower)	Vivarium	25000	3	525	3	168x69	325	55	10	55	50	70	30
Secondary														
H-3	Penthouse	Lev. 7 NE AHR District	15880	10	400	4	100x33	693	55	20	55	76	70	45
H-4	Penthouse	Lev. 7 "C" Cubicle District	3155	10	80	4	66x22	316	55	20	55	76	70	45
H-5	Penthouse	Lev. 7 "F" Cubicle District	2505	10	65	4	62x20	291	55	20	55	76	70	45
H-6	Penthouse	Lev. 7 BSL-2+/Imaging District	3890	10	100	4	88x32	199	55	20	55	76	70	45
H-7	Penthouse	Lev. 7 NW AHR District	6830	10	170	4	90x24	455	55	20	55	76	70	45
H-8	Penthouse	Lev. 7 North AHR District	12320	10	320	5	76x46	507	55	20	55	76	70	45

Figure 11 *–Humidifier Summary*

The smaller packaged air handling units shown in the main supply drawing (***Figure 9***) are responsible for spot cooling the penthouse and basement, as well as heating/cooling the East and West Stair Shafts. These units are packaged AHU's that are summarizes in the following table in ***Figure 12***. The four air handlers that have heating coils (AHU-13-15) are responsible for heating and cooling the stair shafts. The remaining units are utilized to cool the penthouse and electric service room.

Packaged Modular Air Handling Units																			
Unit	Service	Location	Fan		Heating Coil						Cooling Coil								
					Air Side			Total MBH	Water Side		Air Side					Total MBH	Water Side		
			CFM	Type	Vel. Fpm	EDB °F	LDB °F		EWT °F	LWT °F	Vel. Fpm	EDB °F	EWB °F	LDB °F	LWB °F		Flow gpm	EWT °F	LWT °F
AHU-12A	Electrical Service Rm.	Basement	7700	DWDI CENT	-	-	-	-	-	-	400	80	70	62	61	200	45	52	60
AHU-12B	Electrical Service Rm.	Basement	7700	DWDI CENT	-	-	-	-	-	-	400	80	70	62	61	200	45	52	60
AHU-13	East Stair Htg./Clg.	Basement	3600	DWDI CENT	400	70	90	78	180	140	400	75	63	57	56	75	19	52	60
AHU-14	West Stair Htg./Clg.	Basement	3600	DWDI CENT	400	70	90	78	180	140	400	75	63	57	56	75	19	52	60
AHU-15	East Stair Htg./Clg.	Basement	3600	DWDI CENT	400	70	90	78	180	140	400	75	63	57	56	75	19	52	60
AHU-16	West Stair Htg./Clg.	Basement	3600	DWDI CENT	400	70	90	78	180	140	400	75	63	57	56	75	19	52	60
AHU-17A	Basement Spot Cooling	Basement	6000	DWDI CENT	-	-	-	-	-	-	400	80	70	62	61	187	45	52	60
AHU-17B	Basement Spot Cooling	Basement	6000	DWDI CENT	-	-	-	-	-	-	400	80	70	62	61	187	45	52	60
AHU-18A	Penthouse Spot Cooling	Basement	6000	DWDI CENT	-	-	-	-	-	-	400	80	70	62	61	187	45	52	60
AHU-18B	Penthouse Spot Cooling	Basement	6000	DWDI CENT	-	-	-	-	-	-	400	80	70	62	61	187	45	52	60
AHU-18C	Penthouse Spot Cooling	Basement	6000	DWDI CENT	-	-	-	-	-	-	400	80	70	62	61	187	45	52	60
AHU-18D	Penthouse Spot Cooling	Basement	6000	DWDI CENT	-	-	-	-	-	-	400	80	70	62	61	187	45	52	60
AHU-18E	Penthouse Spot Cooling	Basement	6000	DWDI CENT	-	-	-	-	-	-	400	80	70	62	61	187	45	52	60

Figure 12 *–Packaged Modular Air Handling Unit Summary*

Supply Fans												
Unit	Service	CFM	S.P. (in. H$_2$0)	Type	Drive	VFD	Emer. Power	Motor Data at 60 Hz				
								MBHP	MHP	RPM	Volts	Phase
SF-1	Loading Dock	20000	2	Mixed Flow	Belt	Y	N	9.5	15	1750	480	3
SF-2	Pass. Elev. Pressurization	18000	2	Mixed Flow	Belt	N	Y	8.7	15	1750	480	3
SF-3	East Stair Pressurization	12000	1.5	Mixed Flow	Belt	N	Y	5.5	10	1750	480	3
SF-4	West Stair Pressurization	12000	1.5	Mixed Flow	Belt	N	Y	5.5	10	1750	480	3
SF-5	East Stair Vestibule Supply	4100	3	Mixed Flow	Belt	N	Y	3.3	7.5	1750	480	3
SF-6	Pass. Elev. Pressurization	12000	1.5	Mixed Flow	Belt	N	Y	5.5	10	1750	480	3

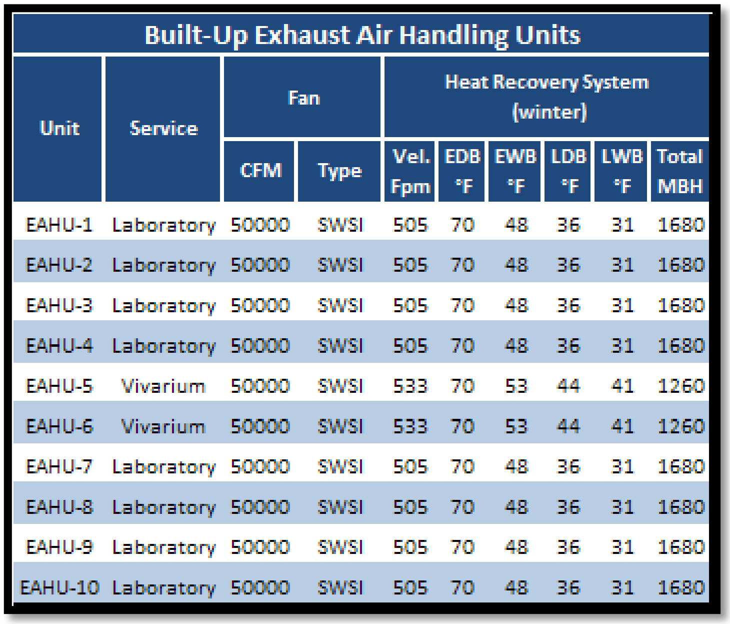
Figure 13 *–Supply Fan Summary*

The remaining supply fans shown in ***Figure 9*** are used to pressurize the loading dock, stairwells and passenger elevator shafts. These fans are summarized in the table to the left in ***Figure 13***.

4.5 Air Exhaust/Return System

The exhaust/return system utilizes (10) 50,000 CFM Factory Built-Up EAHU's to exhaust air from entire building. These exhaust air handlers are paired up with their respective AHU and exhaust air from the same spaces.

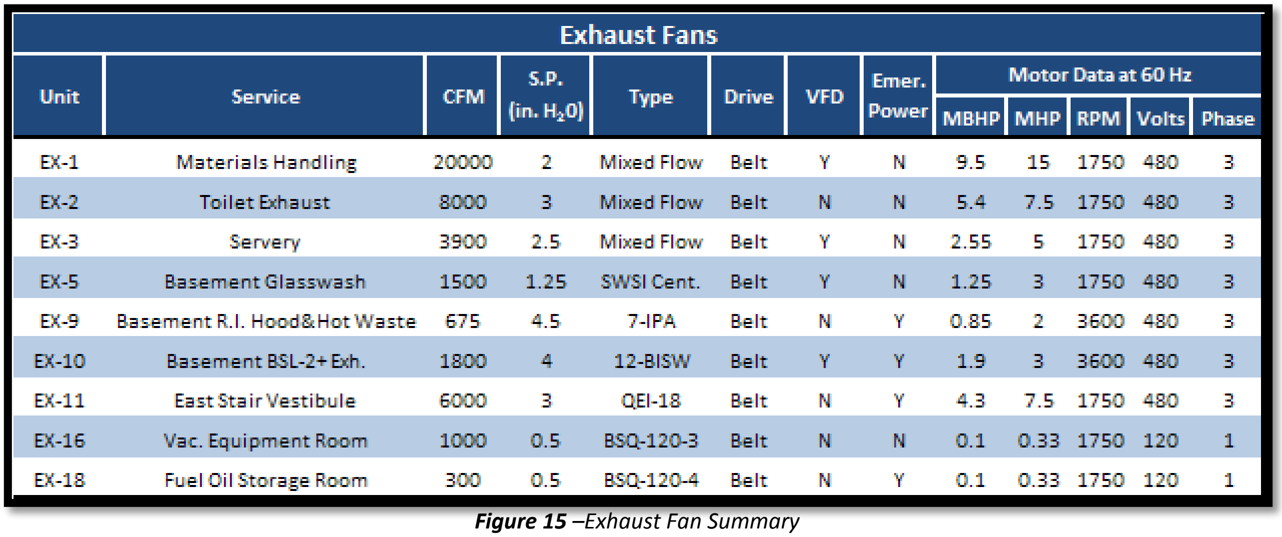
Similar to the supply system, the exhaust system has the 10 main EAHU'S along with a number of smaller Exhaust Fans to deal with smaller spaces. The table to the left in ***Figure 14*** summarizes the main EAHU's, and the table in ***Figure 15*** summarizes these smaller exhaust fans. Lastly, two small return fans that return air directly into the Outdoor Air Plenum are shown in ***Figure 16***. This system is depicted on the following page similar to the supply air system previously outlined. There are a number of future special exhaust fans on the design documents that were not shown in this drawing.



Built-Up Exhaust Air Handling Units

Unit	Service	Fan		Heat Recovery System (winter)					
		CFM	Type	Vel. Fpm	EDB °F	EWB °F	LDB °F	LWB °F	Total MBH
EAHU-1	Laboratory	50000	SWSI	505	70	48	36	31	1680
EAHU-2	Laboratory	50000	SWSI	505	70	48	36	31	1680
EAHU-3	Laboratory	50000	SWSI	505	70	48	36	31	1680
EAHU-4	Laboratory	50000	SWSI	505	70	48	36	31	1680
EAHU-5	Vivarium	50000	SWSI	533	70	53	44	41	1260
EAHU-6	Vivarium	50000	SWSI	533	70	53	44	41	1260
EAHU-7	Laboratory	50000	SWSI	505	70	48	36	31	1680
EAHU-8	Laboratory	50000	SWSI	505	70	48	36	31	1680
EAHU-9	Laboratory	50000	SWSI	505	70	48	36	31	1680
EAHU-10	Laboratory	50000	SWSI	505	70	48	36	31	1680

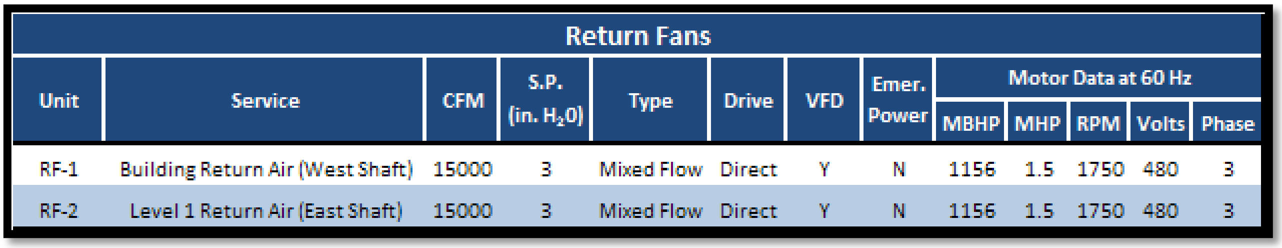
Figure 14 *–Exhaust Air Handling Unit Summary*



Exhaust Fans

Unit	Service	CFM	S.P. (in. $H_2 0$)	Type	Drive	VFD	Emer. Power	Motor Data at 60 Hz				
								MBHP	MHP	RPM	Volts	Phase
EX-1	Materials Handling	20000	2	Mixed Flow	Belt	Y	N	9.5	15	1750	480	3
EX-2	Toilet Exhaust	8000	3	Mixed Flow	Belt	N	N	5.4	7.5	1750	480	3
EX-3	Servery	3900	2.5	Mixed Flow	Belt	Y	N	2.55	5	1750	480	3
EX-5	Basement Glasswash	1500	1.25	SWSI Cent.	Belt	Y	N	1.25	3	1750	480	3
EX-9	Basement R.I. Hood&Hot Waste	675	4.5	7-IPA	Belt	N	Y	0.85	2	3600	480	3
EX-10	Basement BSL-2+ Exh.	1800	4	12-BISW	Belt	Y	Y	1.9	3	3600	480	3
EX-11	East Stair Vestibule	6000	3	QEI-18	Belt	N	Y	4.3	7.5	1750	480	3
EX-16	Vac. Equipment Room	1000	0.5	BSQ-120-3	Belt	N	N	0.1	0.33	1750	120	1
EX-18	Fuel Oil Storage Room	300	0.5	BSQ-120-4	Belt	N	Y	0.1	0.33	1750	120	1

Figure 15 *–Exhaust Fan Summary*



Return Fans

Unit	Service	CFM	S.P. (in. $H_2 0$)	Type	Drive	VFD	Emer. Power	Motor Data at 60 Hz				
								MBHP	MHP	RPM	Volts	Phase
RF-1	Building Return Air (West Shaft)	15000	3	Mixed Flow	Direct	Y	N	1156	1.5	1750	480	3
RF-2	Level 1 Return Air (East Shaft)	15000	3	Mixed Flow	Direct	Y	N	1156	1.5	1750	480	3

Figure 16 *–Return Fan Summary*

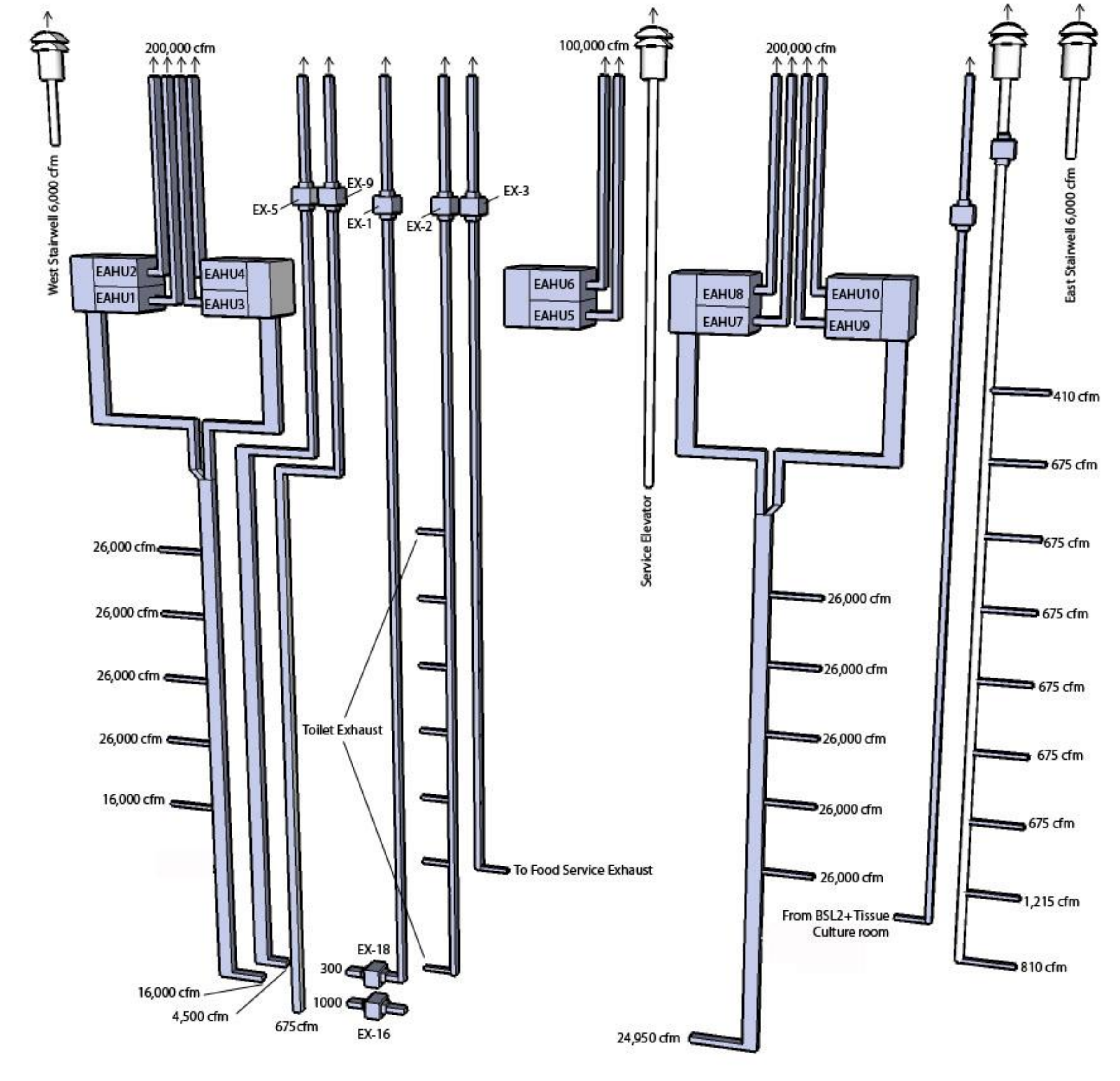


Figure 17 –*Exhaust Air Riser Diagram*

4.6 Chilled Water System

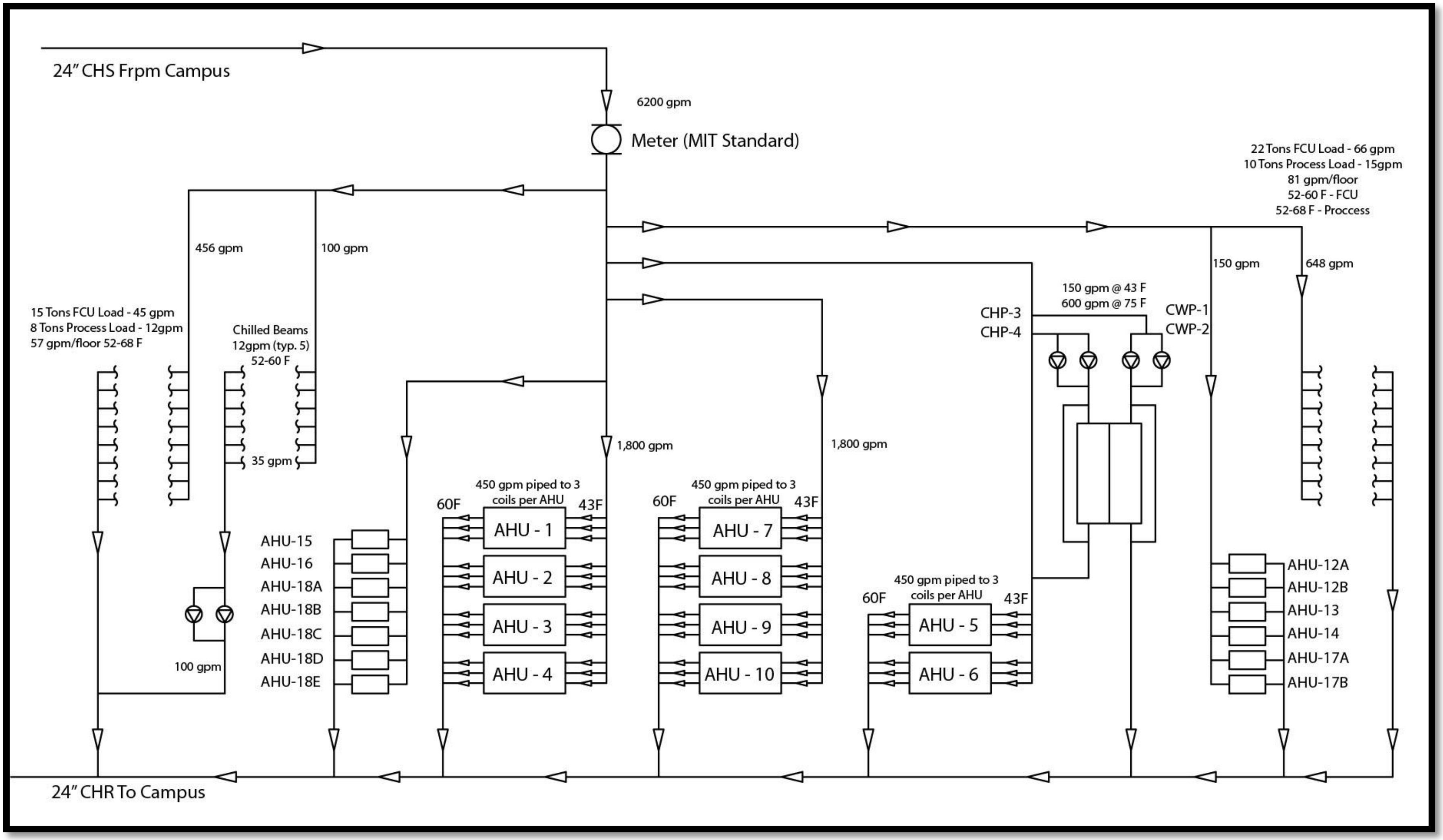
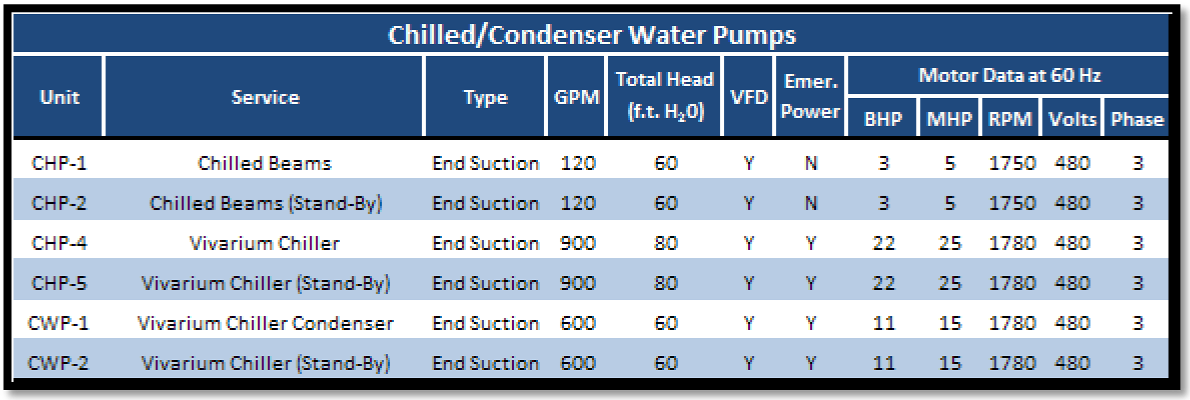


Figure 18 *–Chilled Water Riser Diagram*

The chilled water system for the Koch Institute is fed through an existing MIT Campus chilled water loop. A maximum flow of 6,200 gpm chilled water enters the building through a 24" directly buried supply line and passes through the MIT Standard Meter. The chilled water is then distributed throughout the building. One, 200 ton water cooled rotary screw chiller, was added to the design to provide redundancy for the vivarium spaces. The ten large AHU's require three cooling coils and therefore 450 gpm of chilled water is piped to each through (3) 6" pipes to each unit, which is shown on the previous page in ***Figure 18***. Chilled water also serves fan coil units and process loads on all floors through East and West Risers. The chilled water and condenser water pumps are summarized below in ***Figure 19.***



Chilled/Condenser Water Pumps

Unit	Service	Type	GPM	Total Head (f.t. H_2O)	VFD	Emer. Power	Motor Data at 60 Hz				
							BHP	MHP	RPM	Volts	Phase
CHP-1	Chilled Beams	End Suction	120	60	Y	N	3	5	1750	480	3
CHP-2	Chilled Beams (Stand-By)	End Suction	120	60	Y	N	3	5	1750	480	3
CHP-4	Vivarium Chiller	End Suction	900	80	Y	Y	22	25	1780	480	3
CHP-5	Vivarium Chiller (Stand-By)	End Suction	900	80	Y	Y	22	25	1780	480	3
CWP-1	Vivarium Chiller Condenser	End Suction	600	60	Y	Y	11	15	1780	480	3
CWP-2	Vivarium Chiller (Stand-By)	End Suction	600	60	Y	Y	11	15	1780	480	3

Figure 19 *–Chilled/Condenser Water Pump Summary*

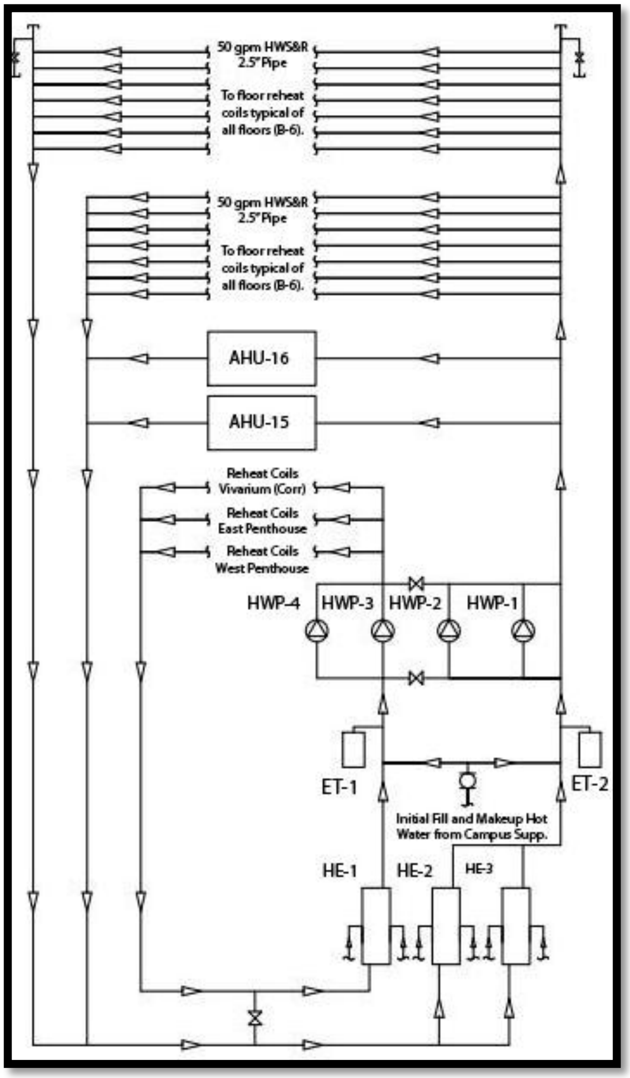


Figure 20 *–Hot Water Riser Diagram*

4.7 Hot Water System

The hot water system for the Koch Institute consists of three shell and tube heat exchangers that produce 180 degree F hot water from low pressure steam (4 psig). As shown in ***Figure 20*** *to the left*, the hot water is then pumped to building reheat, vivarium reheat and AHU's 15 & 16. To maintain separation from the other systems, the vivarium space has its own heat exchanger HE-3.

The three valves shown in the drawing, which are normally closed, are used to either isolate the vivarium from the rest of the hot water system or vice versa. Prior to the expansion tanks (ET-1 & 2) is a connection to the campus hot water system that will act to initially fill the system as well as provide make-up hot water for the system.

The following two tables shown in ***Figure 21 & 22*** summarize the hot water pumps and heat exchangers respectively. Hot water pumps 3 and 4 are responsible for the vivarium space to once again keep it on a separate loop.

Hot Water Pumps											
Unit	Service	Type	GPM	Total Head (f.t. H_20)	VFD	Emer. Power	Motor Data at 60 Hz				
							BHP	MHP	RPM	Volts	Phase
HWP-1	Hot Water Reheat	End Suction	600	75	Y	Y	13.6	20	1750	480	3
HWP-2	Hot Water Reheat (Stand-By)	End Suction	600	75	Y	Y	13.6	20	1750	480	3
HWP-3	Vivarium Reheat	End Suction	120	60	Y	Y	3	5	1780	480	3
HWP-4	Vivarium Reheat (Stand-By)	End Suction	120	60	Y	Y	3	5	1780	480	3

Figure 21 *–Hot Water Pump Summary*

Heat Exchangers									
Unit	Service	Tube Length (ft.)	Tube Length (ft.)	Water Side (Tube)				Steam Side (Shell)	
				EWT °F	LWT °F	GPM	Min MBH	Oper. Pres.	Oper. Temp.
HE-1	Building Reheat	7	20	152	180	600	9600	4	360
HE-2	Building Reheat (Stand-By)	7	20	152	180	600	9600	4	360
HE-3	Vivarium Reheat	6	10	152	180	120	1920	4	360

Figure 22 *–Heat Exchanger Summary*

4.8 Cogeneration Plant

MIT's Central Plant design has become a template for on-site cogeneration across the country. The plant is designed around an ABB GT10A Combustion Generator Set located in the campus Central Plant. With a nominal output of 21MW (electric) and 56MW (thermal), the gas turbine provides MIT with approximately 80% of its electricity. The turbine employs fuel pre-mixing to ensure complete combustion of the fuel source, either natural gas or liquid fuel. Water injection into the combustion zone cools the flame to approximately 2300˚F, reducing thermal NO_x levels. Additionally, a platinum and alumina Carbon Monoxide catalyst removes over 98% of the CO present in the CTG exhaust.

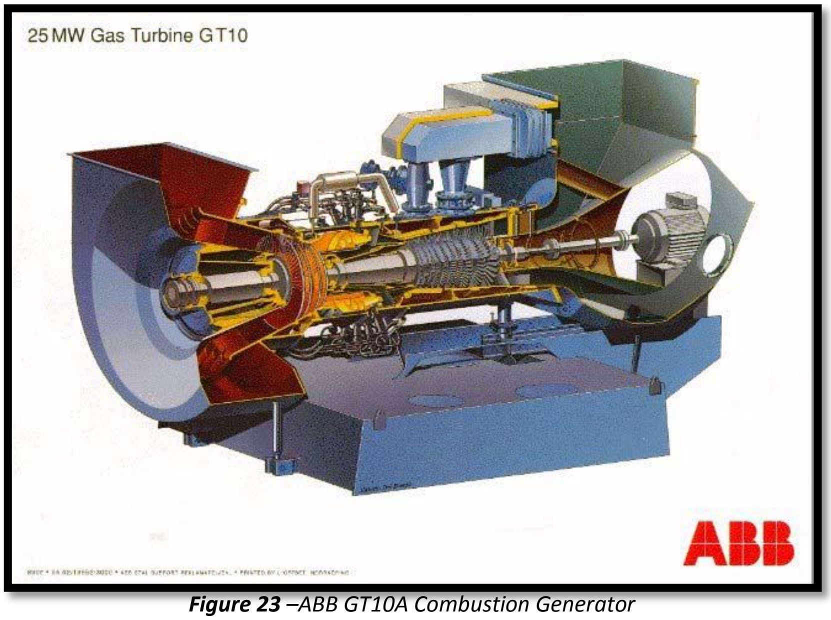


Figure 23 *–ABB GT10A Combustion Generator*

The turbine provides high quality exhaust with a max temperature and flow rate of 1050°F and 628,000 lb_m/ hr respectively. This exhaust is then sent into a Heat Recovery Steam Generator where it is used to create high pressure steam. Steam is then distributed between the MIT Campus and the steam driven absorption chillers. The schematic diagram below in ***Figure 24*** shows the steam generation and distribution to both campus and the chilling plant. The schematic in ***Figure 25*** shows the configuration of the absorption chillers that receive steam from the HRSG and produce chilled water for campus.

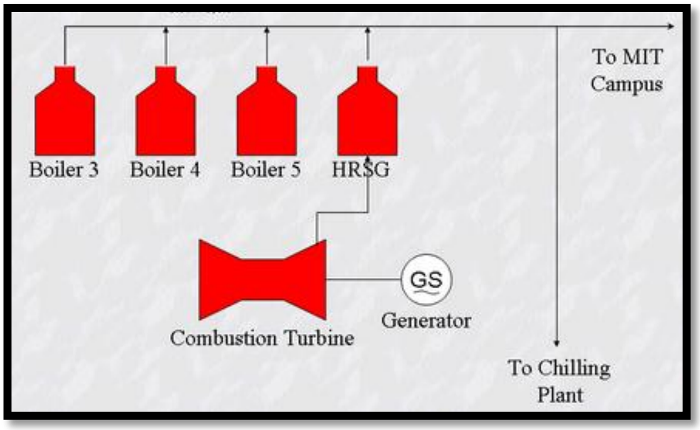


Figure 24 *–Steam Production Schematic*

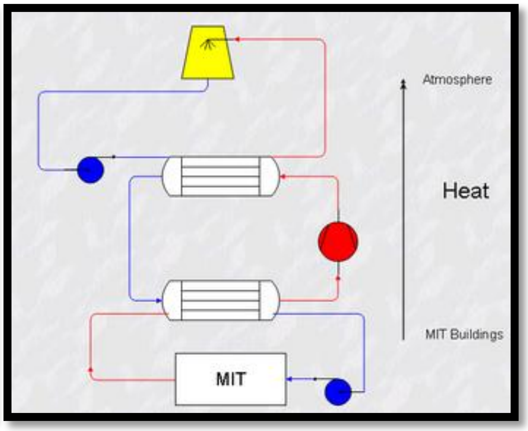


Figure 25 *–Absorption Chiller Schematic*

When the loads on campus exceed 20MW or the cogeneration plant is down, NSTAR power provides transmission and distribution to supplement the cogeneration plant. This service is provided via four umbilicals passing through two current limiting reactors into the 13.8 kV bus. This bus is connected to both the cogeneration plant and the NSTAR service, allowing MIT to switch the electrical source when necessary. Building loads draw from this bus directly. The power is stepped down to 2.4 kV for campus emergency power and for non-building loads across campus. A schematic drawing of MIT's Electrical Distribution is shown below in ***Figure #.***

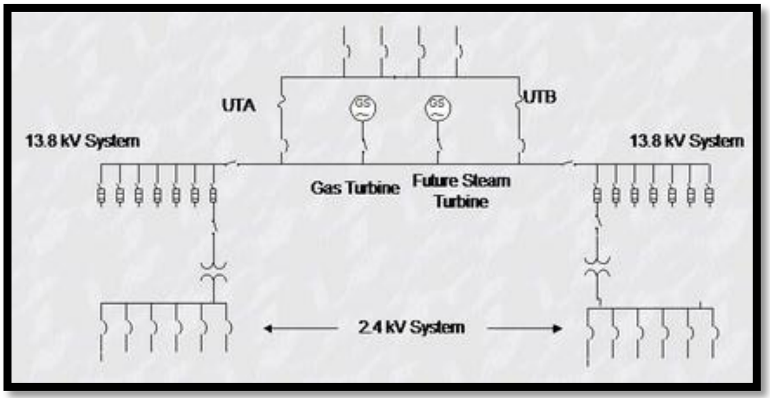
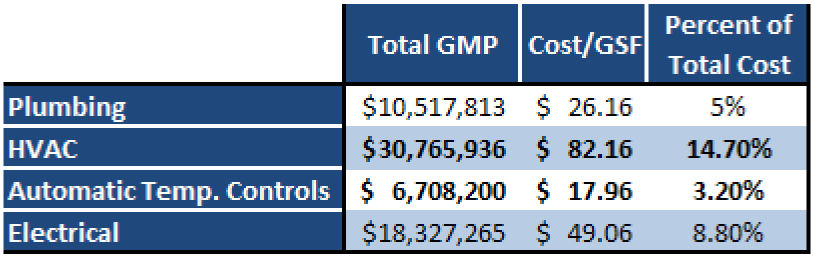


Figure 26 *–Campus Electrical Distribution Schematic*

4.9 Mechanical System Initial Cost

The approximate initial cost for the Mechanical and Plumbing system of the project, as specified in the official estimate are as follows:



	Total GMP	Cost/GSF	Percent of Total Cost
Plumbing	$10,517,813	$ 26.16	5%
HVAC	$30,765,936	$ 82.16	14.70%
Automatic Temp. Controls	$ 6,708,200	$ 17.96	3.20%
Electrical	$18,327,265	$ 49.06	8.80%

Figure 27 *–Initial Cost of Mechanical System*

The HVAC and Automatic Temperature Controls total to **$37,474,136**, which equates to **$100.12/square foot** and accounts for **18%** of the total construction cost. If you include plumbing it raises to $47,991,949, roughly 22% of the total construction cost. Therefore, it is a significant portion of the total cost of the building.

4.10 Evaluation of System

MIT's high expectations for their new research facility presented the design engineers with many challenges. As a result, the systems designed for MIT's new Integrative Cancer Research Lab are extremely sophisticated and have exceeded all expectations. Hybrid air/water system designed for this building was compared to a number of other approaches early in design. The efficiency of the system combined with the Cogeneration Plant resulted leave little room for improvement. Therefore, utilization additional heat recovery and renewable energy sources are the remaining possibilities for system enhancement.

4.10.1 Air System

To meet all internal ventilation requirements and indoor air quality requirements, a nearly 100% outdoor air ventilation/cooling system was necessary for the design. With this type of system in the Boston, MA climate, a large amount of energy is needed to dehumidify and cool the incoming airstream in summer months. The decision to use factory built-up AHU and EAHU's allowed for an efficient heat pipe heat recovery system to be employed, making it possible to precondition the incoming supply airstream, reducing the load on each air handler.

Also, by utilizing some supplemental systems (i.e. chilled beam induction units, fan coil units and radiant panel heating) in high load areas, the main air handlers could all be equally sized. Having ten identical large units, building operators of the building only must familiarize themselves with that particular AHU allowing for successful operation. The system responds well to the needs of the building and utilizes the most efficient techniques to do so.

4.10.2 Chilled/Hot Water System

The campus Cogeneration Plant is responsible for providing chilled and hot water to the Koch Institute. High pressure steam is produced by a heat recovery steam generator and delivered directly to the buildings heat exchangers. These shell and tube heat exchangers then create hot water for the building. Similarly, steam is sent to absorption chillers in the plant that produce chilled water which is supplied to the building through existing mains. Both systems are efficient and utilize heat recovery from the gas turbine's exhaust. Attempting to reduce the chilled/hot water needed in the building with renewable energy sources could shed load at the plant, saving money for the school.

4.10.3 LEED NC Design

This project is projected to be awarded LEED Gold Certification after gaining 42 credits on the LEED NC 2.2 checklist. The design provides the building an energy efficient solution to HVAC and is designed to create a comfortable environment for the occupants. It is possible however that renewable technology can generate enough energy to achieve a reasonable payback period for the owner. In conclusion, the MEP design engineers at BR+A have succeeded in creating an innovative mechanical system that meets the needs of the owner and does so efficiently.

5.0 Existing Design Loads & Consumption

The following charts in ***Figures 28 & 30*** are the calculated design cooling and heating loads for the Koch Institute. Loads were first calculated in Trane TRACE 700, but complications arose with modeling the buildings complex mechanical system and MIT's cogeneration plant. Additional calculations were performed in Microsoft Excel to maintain accuracy.

The Koch Institute has a **2,746 ton** peak cooling load due to large laboratory equipment loads as well as a significant amount of solar gain due to the glass façade. ***Figure 28*** below shows the division of the cooling load throughout the spaces in the building.

	Peak Cooling Load
	Tons
Level 2-7 Laboratory	2,114
Intense Load Areas	440
Penthouse/Stairs/Equip. Rms	160
Level 1 Offices	32
	2,746

Figure 28 *–Peak Cooling Load*

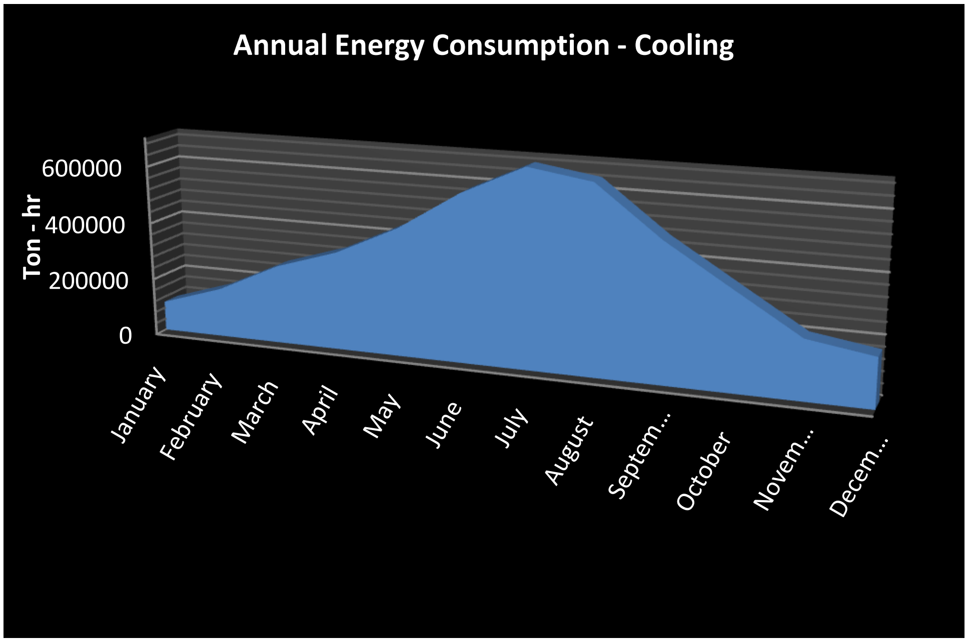
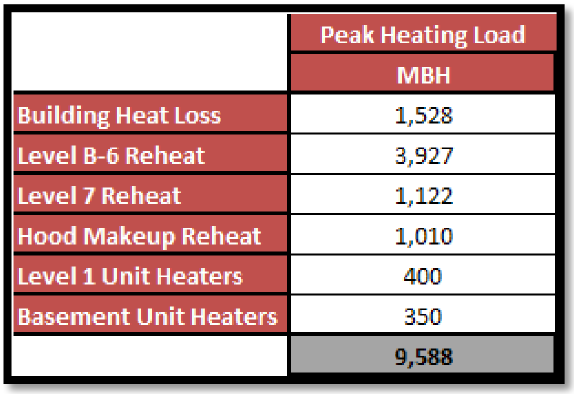


Figure 29 *–Annual Energy Consumption - Cooling*

The peak heating load in the building was calculated to be **9,588 MBH.** This is expected to be significantly lower than the cooling load in a laboratory building due to the heat generated by the equipment and occupants. Calculations for the building total heat loss can be found in the ***Appendix.***



	Peak Heating Load
	MBH
Building Heat Loss	1,528
Level B-6 Reheat	3,927
Level 7 Reheat	1,122
Hood Makeup Reheat	1,010
Level 1 Unit Heaters	400
Basement Unit Heaters	350
	9,588

Figure 30 *–Peak Heating Load*

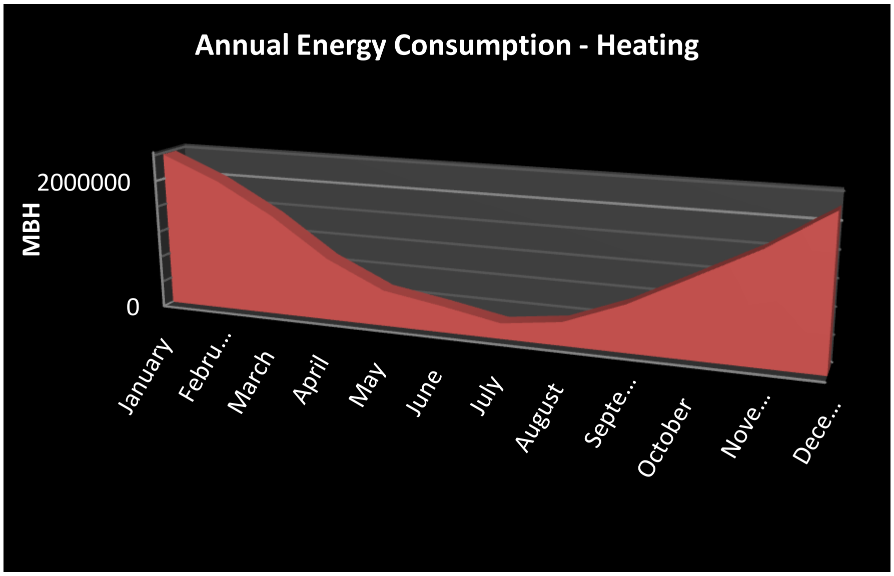


Figure 31 *–Annual Energy Consumption - Heating*

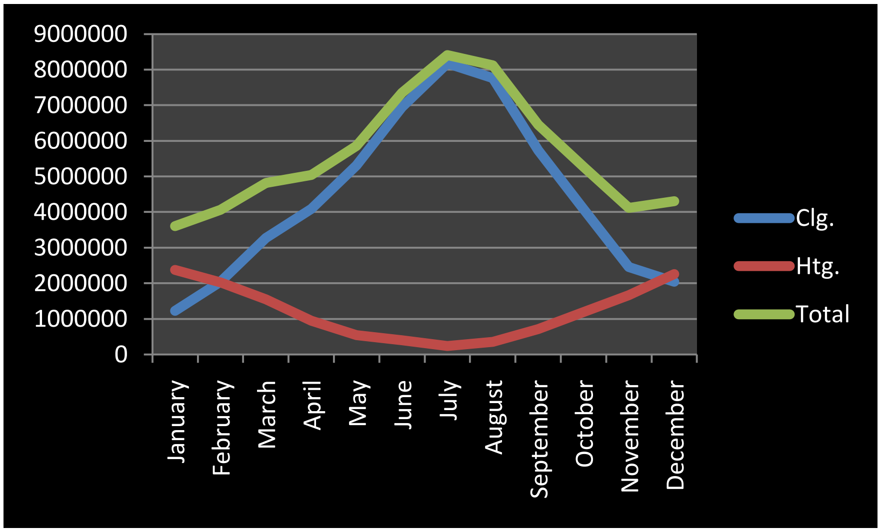


Figure 32 *–Annual Energy Consumption (MBH)*

6.0 Proposed Redesign Overview

6.1 Heat Recovery on Specialty Exhaust

The central ventilation/cooling system utilizes a large heat pipe heat recovery system between the supply and exhaust airstreams. For a nearly 100% outdoor air system this is an energy conscious approach given the strict laboratory ventilation requirements. Due to specific needs, it was also required to provide the Koch Institute with a number of specialty exhaust fans. These fans do not currently employ any heat recovery systems. The exhaust air diagram shown in section **3.5 Air Exhaust/Return Systems** depicts the specialty exhaust fans that are responsible for the stairs and all levels of the building with the exception of level seven.

The building design utilizes **18 specialty exhaust fans** that exhaust **53,810 cfm** of conditioned air without retrieving any energy. To account for increased need for specialty exhaust, the design also includes an additional **14 future exhaust fans (of unknown size)**. Therefore, there is potential for energy savings if energy is recovered from all of these exhaust airstreams and utilized to pre-condition the supply airstream of the AHU-14 and 15 to heat the stairwells.

Due to the varying location of all the specialty exhaust fans, a side by side airstream is not feasible. To incorporate the specialty exhaust heat recovery successfully, a **glycol loop system** was researched. A glycol loop is capable of collecting energy from sources in various locations and placing it all into the supply airstream. Optimizing this system will take time given the varying airstreams, but with 53,810 cfm of conditioned air being exhausted, the energy recovery should be significant.

6.2 Ground Source Heat Pump

MIT's campus plant utilizes a 25 MW Micro-turbine to produce 80% of the campuses electrical energy, while also utilizing the waste heat in the turbine's exhaust for a number of applications within the plant. Therefore, the existing energy sources are very efficient. That being the case, any attainable renewable sources that help to reduce the buildings energy consumption can largely benefit the already efficient campus system. For that reason, over the application of geothermal heat pumps to reduce the load on the central plant was analyzed.

In commercial applications where cooling loads exceed heating loads, as is the case in this project, the long term efficiency can drop due to an increase in the ground temperature. To avoid this dilemma, a hybrid system was looked at for a number of possible applications.

Incorporating a ground source heat pump into the following three options was evaluated:

- **Providing chilled water to the fan coil units**
- **Providing chilled water to the chilled beams**
- **Providing chilled water to the packaged AHU's (Spot Cooling Units & Stair Units)**

These options were evaluated and a geothermal system will be employed to reduce loads on the existing systems, resulting in a reduction in work for the central plant as well as the designed ventilation/cooling system. It will provide redundancy as well as shed some load from the existing system during proper conditions. The following graph in ***Figure 33*** shows that at shallow depths, variation in the average ground temperature is higher. With the reliability of the system in mind, it is more feasible to utilize the vertical loop system that reaches deeper into the ground.

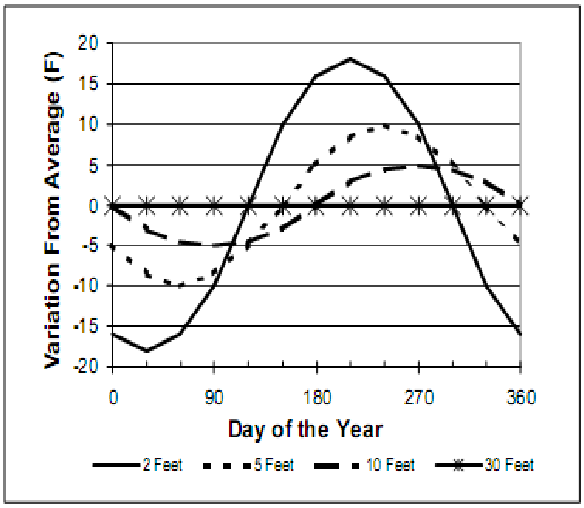


Figure 33 *–Average Ground Temp. Variation with Depth*

Also, the land area necessary for proper amounts of heat transfer for a system of this size deems a horizontal loop system impractical. With a city environment, there is not enough land to utilize a horizontal system, making vertical loop the most obvious choice. There are drilling challenges in the Boston area due to high levels of rock that was evaluated to ensure the system's feasibility. If properly designed and installed, this system could greatly decrease the load on MIT's campus plant, while also adding renewable energy to the Koch Institute's plethora of energy conscious initiatives.

The theoretical vertical loop system shown to the right in ***Figure 34*** was taken from McQuay's *Geothermal Heat Pump Design Manual*. After comparing this picture to the site of the Koch Institute, the quad to the south of the Koch Institute presents itself as a possible location for this system. This area is depicted on the following page in ***Figure 11***.

Figure 34 *–Conceptual Vertical Loop System*

The quad located in the center of ***Figure 35*** is the proposed area for the vertical loop geothermal heat pump. This closed loop design will reduce pump work and will require approximately 250 to 300 ft^2/ton to be a succesfully sized system.

The next steps following this proposal will be to research the feasibility of this system in regards to the forementioned applications. Data on the specific ground content of the site will be gathered and evaluated to ensure efficient heat transfer between the fluid and the ground.

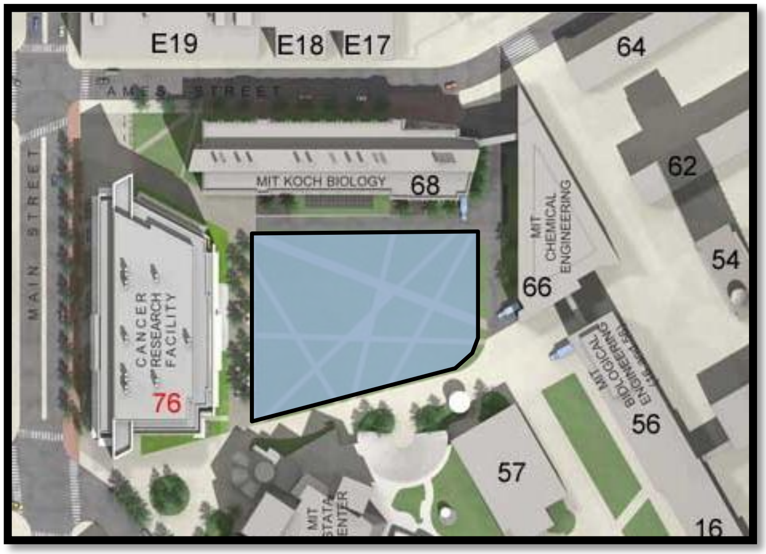


Figure 35 *–Proposed Location for the GSHP*

7.0 Heat Recovery – Mechanical Depth

7.1 Redesign Objective

The objectives of the system redesign were to reduce the overall energy consumption, and incorporate renewable energy into the project.

7.2 Glycol Run Around Heat Recovery

The existing design contained 18 exhaust fans that exhausted a total of 53,650 cfm without any energy recovery. Of these 18 exhaust fans, 12 are ducted directly through the penthouse to the roof. They vary in size and location, making them prime canditates for a run around heat recovery system. In ***Figure 36*** below, the locations of the twelve exhaust airstreams are shown in red, and the 10 EAHU airstreams are depicted in black.

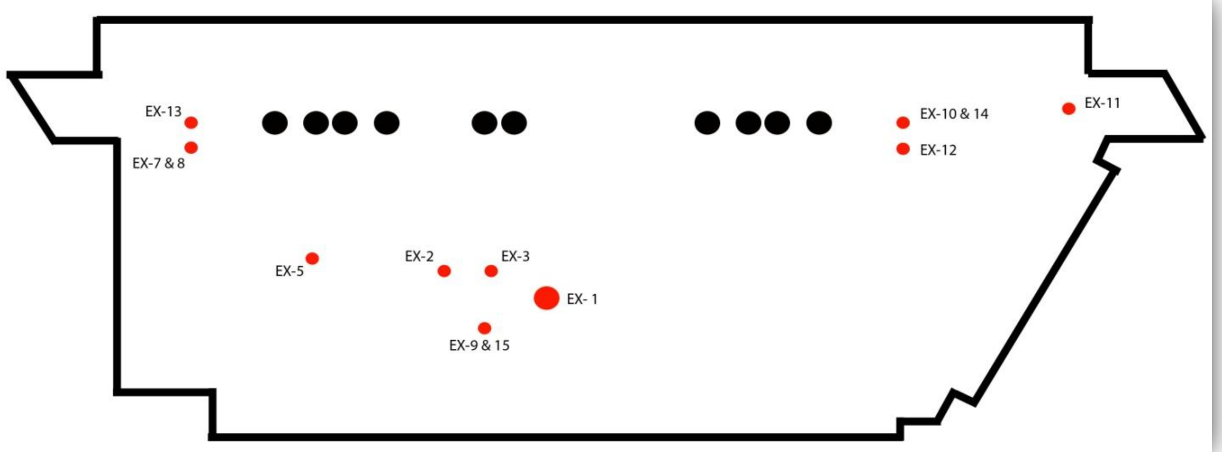


Figure 36 *–Exhaust airstream locations for Glycol Loop*

Advantages

- does not require that the two air streams be adjacent to each other
- several air streams can be used
- has relatively few moving parts - a small pump and control valve
- relatively space efficient
- the cooling or heating equipment size can be reduced in some cases
- the moisture removal capacity of existing cooling equipment can be improved
- no cross-contamination between air streams

Disadvantages

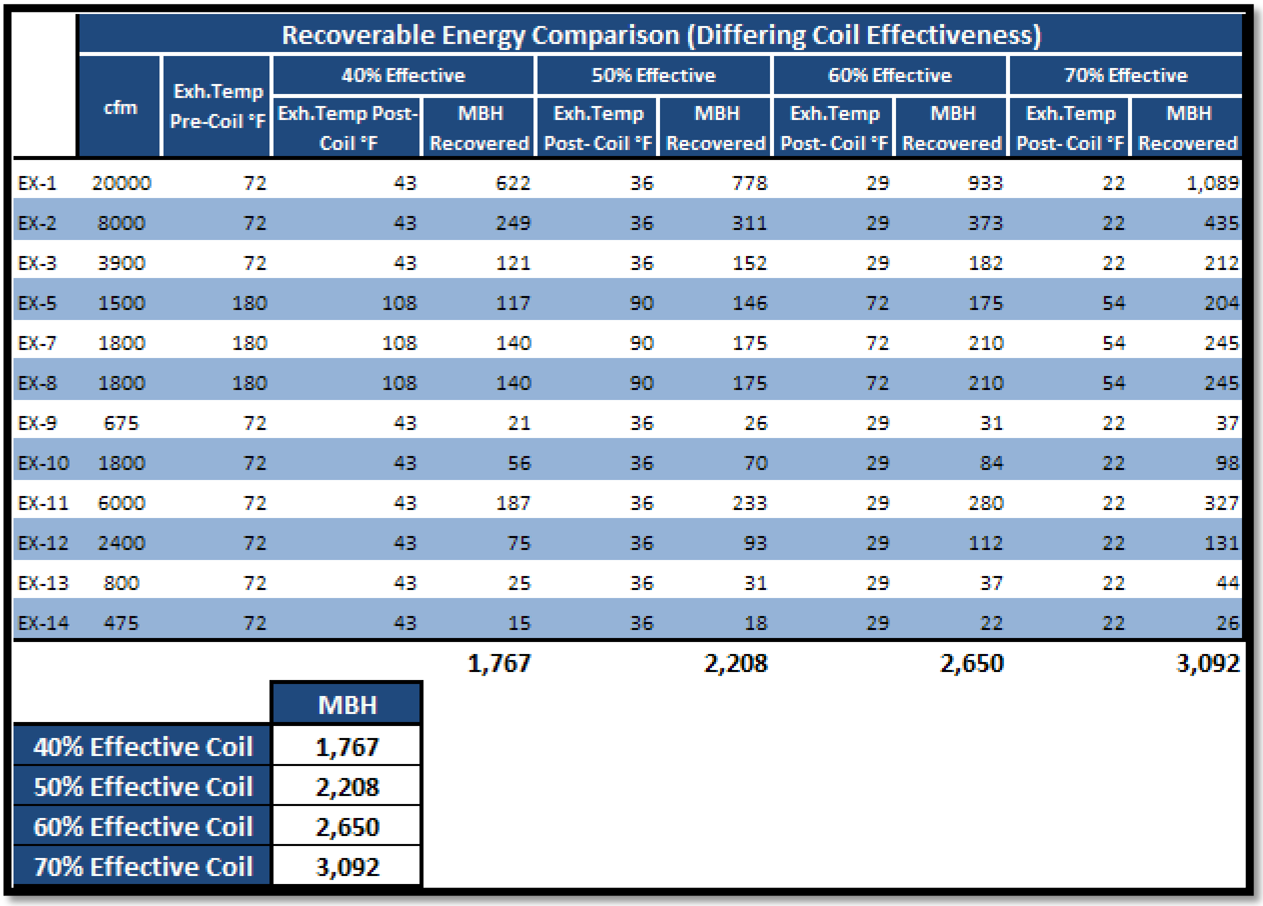
- adds to the first cost, to the fan power to overcome add coil pressure drop, and for the glycol circulating pump
- requires added glycol pump and piping, expansion tank, and a three-way freeze protection control valve,
- requires that the air streams must be relatively clean and may require filtration

7.2.1 Assumptions

All exhaust airstreams with unknown temperatures were conservatively assumed to be at 72°F to calculate the energy available for recover. Exhausts from rack, tunnel and cage washers were assumed to be at 140°F due to their use of medium pressure steam during the washing process.

7.2.2 Recoverable Energy Calculation

The sensible heat transfer between the exhaust airstream and the heat recovery coil depends on the surface area of the coil bank. Coils with larger surface area available for heat transfer are more effective yet they introduce pressure drop to the system. Four coil types with differing heat recovery capabilities were evaluated ranging from 40%-70%. This comparison is shown below in ***Figure 37***.



Recoverable Energy Comparison (Differing Coil Effectiveness)

	cfm	Exh.Temp Pre-Coil °F	40% Effective		50% Effective		60% Effective		70% Effective	
			Exh.Temp Post-Coil °F	MBH Recovered	Exh.Temp Post- Coil °F	MBH Recovered	Exh.Temp Post- Coil °F	MBH Recovered	Exh.Temp Post- Coil °F	MBH Recovered
EX-1	20000	72	43	622	36	778	29	933	22	1,089
EX-2	8000	72	43	249	36	311	29	373	22	435
EX-3	3900	72	43	121	36	152	29	182	22	212
EX-5	1500	180	108	117	90	146	72	175	54	204
EX-7	1800	180	108	140	90	175	72	210	54	245
EX-8	1800	180	108	140	90	175	72	210	54	245
EX-9	675	72	43	21	36	26	29	31	22	37
EX-10	1800	72	43	56	36	70	29	84	22	98
EX-11	6000	72	43	187	36	233	29	280	22	327
EX-12	2400	72	43	75	36	93	29	112	22	131
EX-13	800	72	43	25	36	31	29	37	22	44
EX-14	475	72	43	15	36	18	29	22	22	26
				1,767		**2,208**		**2,650**		**3,092**

	MBH
40% Effective Coil	1,767
50% Effective Coil	2,208
60% Effective Coil	2,650
70% Effective Coil	3,092

Figure 37 *–Recoverable Energy Analysis Table*

Though it recovers energy less efficiently than the others, the 40% Effective coil was to avoid increasing fan size. An increase in fan energy would outweigh the energy saved by the heat recovery system. Therefore, at peak operation the glycol heat recovery loop is capable of recovering 1,767 MBH from the ten exhaust airstreams.

7.2.3 Airside Redesign

The existing design utilizes recirculating AHU's 13, 14, 15 and 16 to provide heating and cooling to the east and west stair shafts. The stair pressurization is done independently by SF-3, 4 and 5 which supply 100% unconditioned outdoor air to the space. The glycol heat recovery loop is designed to precondition the incoming outdoor air to SF- 4 & 5 directly in the penthouse, thus reducing the heating load on the recirculating AHU's. ***Figures 38 & 39*** show the air riser diagrams for the west and east stairshafts with all forementioned AHU's and SF's.

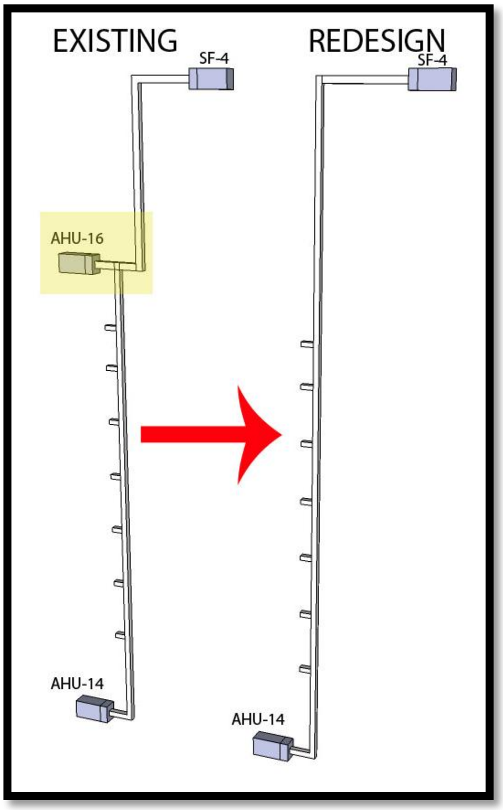


Figure 38 *–East Stair Shaft*

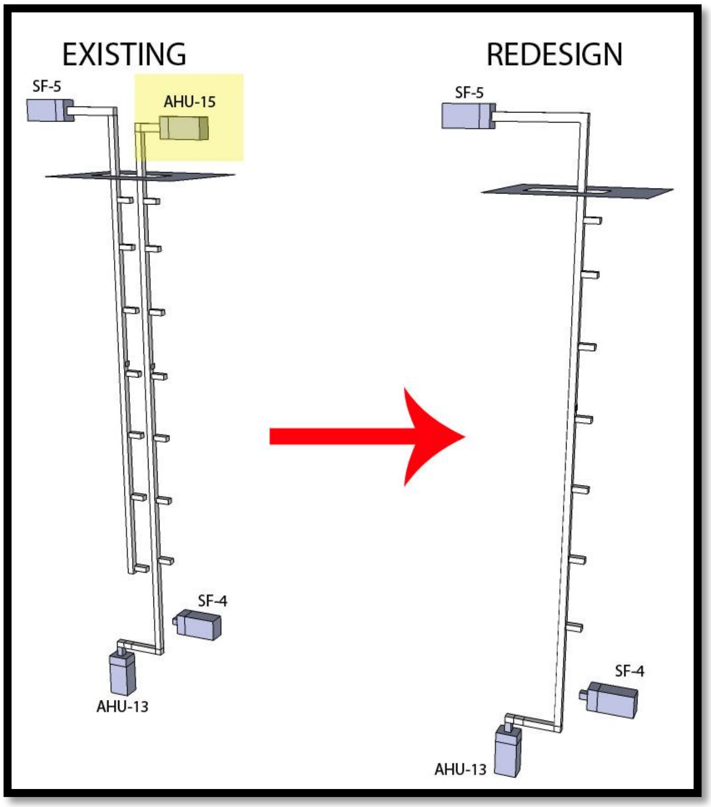


Figure 39 *–West Stair Shaft*

The two figures above show the resdesigned air riser diagrams for the East and west stair shafts. With the preconditioning of incoming outdoor air at SF- 4 and 5, the heating load decreases enough to **remove AHU-15 and 16** and some resulting ductwork.

7.2.4 Pumping – Configuration and Selection

The design of the runaround heat recovery system is shown below ***Figure 40.*** The entire run around loop is contained within the penthouse where the tweleve exhausts are accessable. The preheat coils for supply fans 4 and 5 are also located in the penthouse at there respective outdoor intakes.

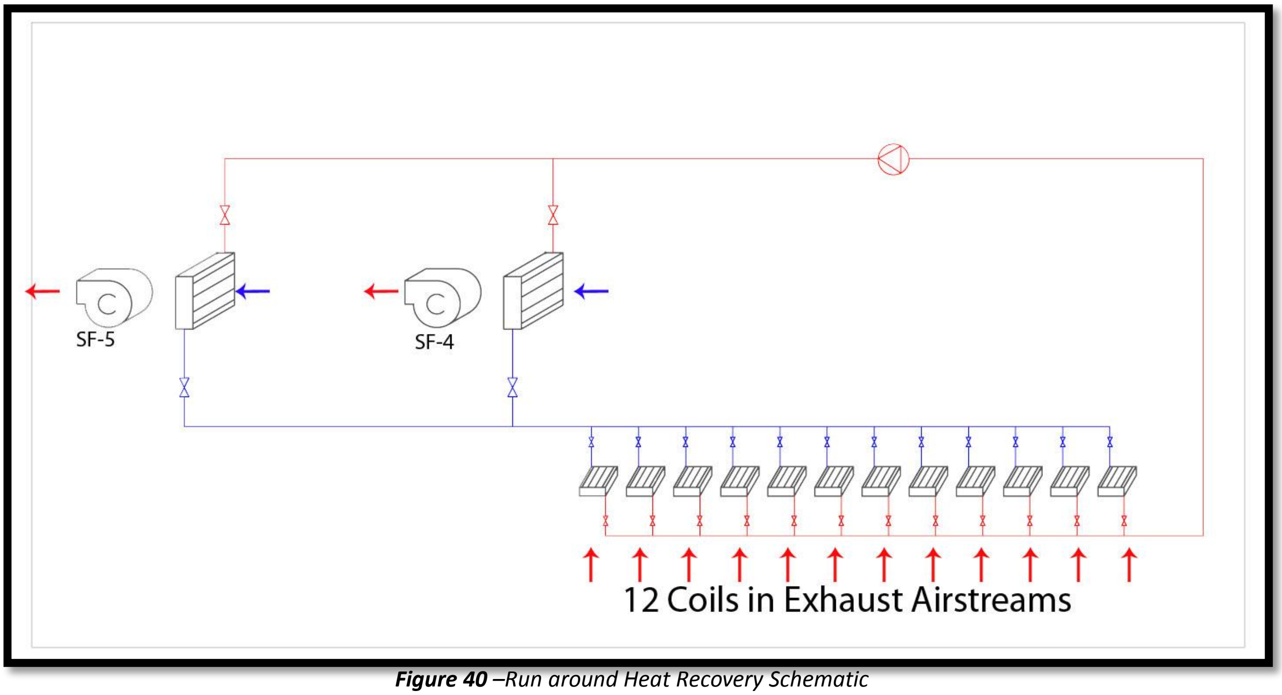
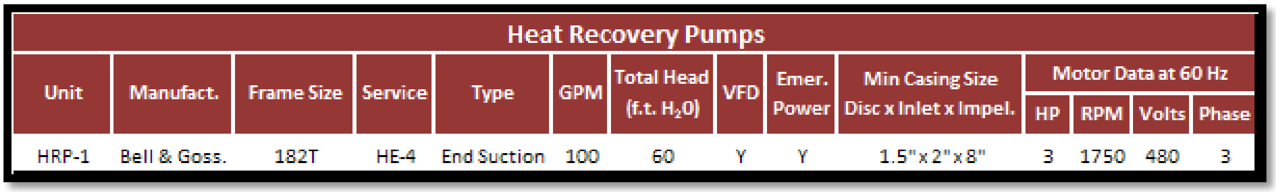


Figure 40 *–Run around Heat Recovery Schematic*

The system was calculated to have a head pressure of approximately 60 ft H_2O and requires a flow of 100 gpm. With these system characteristics, a 3HP Bell & Gossett pump was selected from the *Bell and Gossett Curve Booklet B-260G Series 1510* (pump curves found in ***Appendix***). ***Figure 41*** below shows the new pumping schedule for the heat recovery system.



Heat Recovery Pumps

Unit	Manufact.	Frame Size	Service	Type	GPM	Total Head (f.t. H_2O)	VFD	Emer. Power	Min Casing Size Disc x Inlet x Impel.	Motor Data at 60 Hz			
										HP	RPM	Volts	Phase
HRP-1	Bell & Goss.	182T	HE-4	End Suction	100	60	Y	Y	1.5" x 2" x 8"	3	1750	480	3

Figure 41 *–Heat Recovery Pump Schedule*

8.0 Ground Source Heat Pump – Mechanical Depth

8.1 Redesign Objective

The majority of Koch Institutes heating, ventilation and cooling loads are handled by the 10 Factory Built-Up units located in the penthouse. This central ventilation/cooling system coupled with MIT's on-site cogeneration plant is extremely efficiently. Therefore, the target areas for this depth study is improving the smaller systems in the building that supplement the central system. This section discusses the proposed design of a ground source heat pump to supply chilled water to the (13) Packaged Modular Air Handling Units, shown earlier in ***Figure 8***.

8.2 Site Geology Study

The site geology has an enormous impact on successful design of a ground source heat pump. The soil and rocks located underground have varying thermal qualities that are essential to the effectiveness of heat transfer to and from the ground. A proper analysis of the site's geology for a project of this size involves expensive borehole testing to specifically analyze the soil and rock properties. The nature of this project does not allow for this level of detail, so a detailed investigation into the known rock and soil types was performed. The map below in ***Figure 42*** is taken from the USGS (United States Geological Survey). It depicts the geology of massachusetts.

Figure 42 *–MA Geology Map*

The section shaded green in ***Figure 42*** depicts the area shown in ***Figure 43*** that provides closer to view the Koch Institute's site specific geology. An arrow has been drawn on ***Figure 43*** to the location of the Koch Institute on the geoligical map. The map has been color coated to differentiate the soil and rock types that are specific to each location. The Koch Institute is located in a section whose primary rock type is comprised of Cambridge Argillite. The secondary and tertiary rock types are quartile and sandstone. These rock and soil types were then compared to the those in Table 5 in Chapter 32 of the ASHRAE Handbook of Fundamentals. The closest matches for soil and rock types were Heavy Sand 5% water and Sandstone, which were utilized in the thermal resistance calculations shown in the following pages.

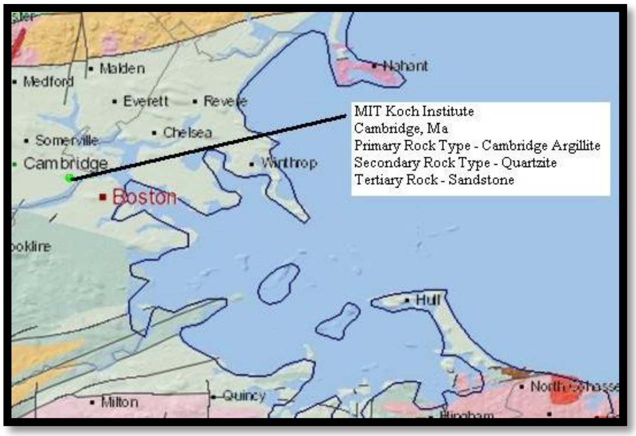


Figure 43 *–Cambridge Geology Map*

8.3 Sizing Method

8.3.1 Bore Length Equation

To assure that the ground source heat pump is properly sized, a method from *Chapter 32 of the 2007 ASHRAE Handbook-HVAC Applications* was followed. This method accounts for the change in thermal resistance of the ground per unit length over three heat pulses. The equation shown below in ***Figure 44*** calculates the required bore length for the heat pump. The three heat pulses are represented in the various thermal resistance values which were generated using equations shown on the following page. To accurately calculate the value of Lc, these equations were entered into Engineering Equation Solver. The results generated from EES can be found in the ***Appendix***.

$$L_c = \frac{q_a \cdot R_{ga} + [q_{lc} - 3.142 \cdot W_c] \cdot [R_p + PLF_m \cdot R_{gm} + R_{gd} \cdot F_{sc}]}{t_g - \left[\frac{t_{wi} - t_{wo}}{2}\right] - t_p}$$

Figure 44 *–Equation for Required Borehole Length*

F_{sc}= *short circuit heat loss factor*
L_c= *required bore length for cooling, ft*
q_a= *net annual average heat transfer to ground, Btu/h*
q_{lc}= *building design cooling block load, Btu/h*
R_{ga}= *effective thermal resistance of ground (annual pulse), h-ft-°F/Btu*
R_{gd}= *effective thermal resistance of ground (daily pulse) , h-ft-°F/Btu*
R_{gm}= *effective thermal resistance of ground (monthly pulse) , h-ft-°F/Btu*
R_p= *thermal resistance of pipe and borehole, h-ft-°F/Btu*
t_g= *undistributed ground temperature, °F*
t_p= *temperature penalty for interference of adjacent bores, °F*
t_{wi}= *liquid temperature at heat pump inlet, °F*
t_{wo}= *liquid temperature at heat pump at outlet, °F*
W_c= *power input at design cooling load, Btu/h*
PLF_m= *part load factor during design month*

8.3.2 Heat Pump Temperatures (t_g, t_{wi}, t_{wo}, t_p)

The spot cooling air handling units require a chilled water supply at 52°F and return 60°F. From the geological study the temperature of the ground in Cambridge, MA was found to be 50°F. A temperature penalty of 2.4°F was found using Table 7 in Chapter 32 of the 2007 ASHRAE Handbook-HVAC Applications. With these parameters known, the temperatures entered into the equation for length are shown in the following table in ***Figure 45.***

t_g	t_{wi}	t_{wo}	t_p
50°F	60°F	52°F	2.4°F

Figure 45 *–Heat Pump Temperature*

8.3.3 Calculating Thermal Resistances (R_p, R_{ga}, R_{gm}, R_{gd})

Ground source heat pumps rely heavily on their ability to transfer and extract heat to and from the ground. For this to be effective, minimizing the amount of thermal resistance between the ground and the fluid is imperitive. To optimize this process, a number of formulas shown below were utilized. To accurately calculate the effective thermal resistances for three heat pulses, three values of τ are defined. These values of τ, measured in days, were set to one year, one month and 4 hours as suggested in Chapter 32 of the 2007 ASHRAE Handbook of Fundamentals. Once fourier's number has been calculated, the G-Factor's (G_f, G_1, and G_2) for ground thermal resistance are acquired via Figure 15 in Chapter 32 of the 2007 ASHRAE Handbook of Fundamentals. Lastly, these G-Factors along with the thermal conductivity of the ground are used to define the effective thermal resistances for each heat pulse R_{ga}, R_{gm}, and R_{gd}.

$$F_{of} = \frac{4 \cdot \alpha \cdot \tau_f}{d_p^{\,2}} \qquad R_{ga} = \frac{G_f - G_1}{k_g}$$

$$F_{o1} = \frac{4 \cdot \alpha \cdot \left[\tau_f - \tau_1\right]}{d_p^{\,2}} \qquad R_{gm} = \frac{G_1 - G_2}{k_g}$$

$$F_{o2} = \frac{4 \cdot \alpha \cdot \left[\tau_f - \tau_2\right]}{d_p^{\,2}} \qquad R_{gd} = \frac{G_2}{k_g}$$

Figure 46 *–Thermal Resistance Equations*

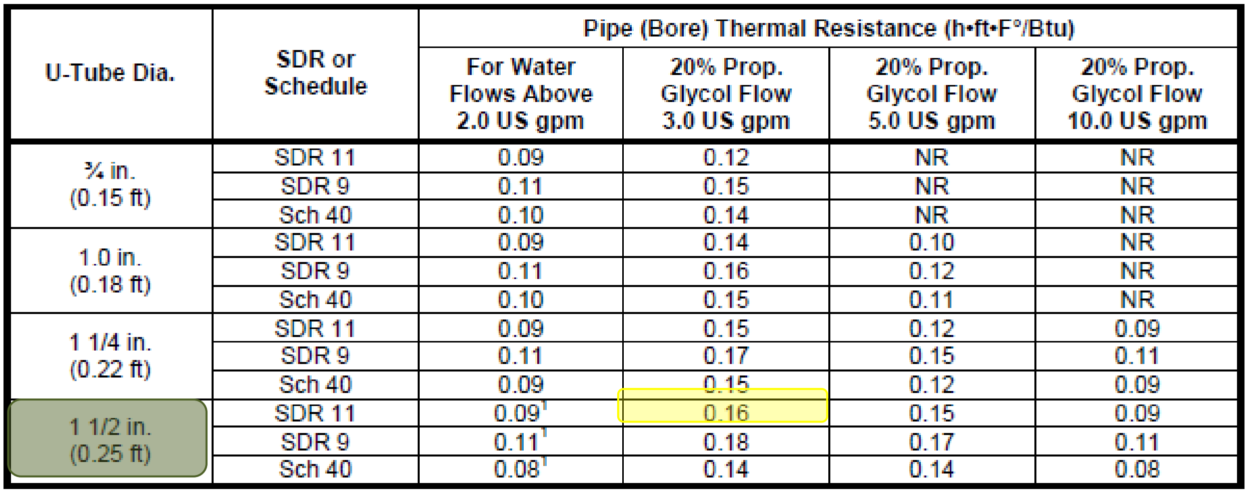
F_{of} = *Fouriers number for* τ_f
F_{o1} = *Fouriers number for* τ_1
F_{o2} = *Fouriers number for* τ_2
α = *Thermal diffusivity of the ground,* m^2*/day*
d_p= *Outside diameter of pipe, ft*
k_g= *Thermal conductivity of the ground, Btu /h-ft-°F*

The effective thermal resistances shown above in ***Figure 46*** (R_{ga}, R_{gm}, R_{gd}) are used to account for the long term heating of the ground source. The thermal resistance of the pipe, R_p, and borehole is found based on the conductivities of the natural soil, grout and the thermal resistance of the High Density Polyethylene U-Tube. The tables shown on the following page, taken from McQuay's Geothermal Heat Pump Design Manual, demonstrate the process utilized to obtain a final value for R_p, the total thermal resistance of the pipe and borehole.

The thermal properties of the soil, rock and grout types decided upon in the geological study are displayed in ***Figure 47.*** As can be seen in ***Figure 48,*** a 1 ½ ” U-Tube Diameter SDR-11 pipe with water flows above 2.0 US gpm has a thermal resistance of ***0.16 h-ft-°F/Btu*** . An additional correction factor of ***0.02 h-ft-°F/Btu*** to account for the 6” bore and soil/grout conductivity was then found via ***Figure 49,*** and added to the thermal resistance of the pipe. This process resulted in a thermal resistance **R_p= 0.18 *h-ft-°F/Btu.***

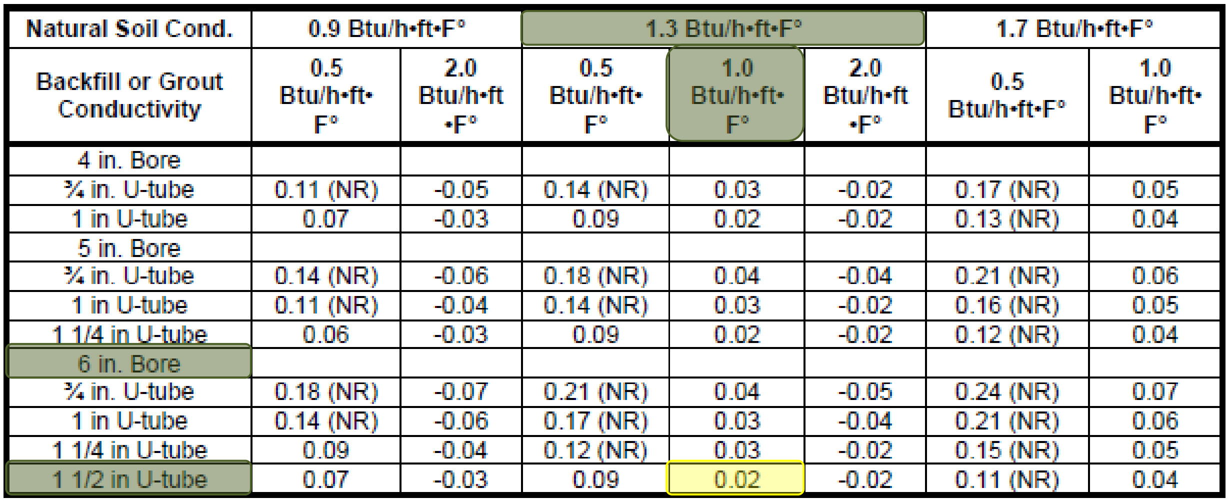
Category	Type	Dry Density lb/ft^3	Conductivity Btu/h-ft-°F	Diffusivity ft^2/day
Soil	Heavy Sand 5% Water	120	1.2 to 1.9	1.0 to 1.5
Rock	Sandstone	-	1.2 to 2.0	0.7 to 1.2
Grouts/Backfill	15% bennonite/85% SiO_2 sand	-	1.00 to 1.10	-

Figure 47 *–Soil, Rock and Grout Characteristics*



U-Tube Dia.	SDR or Schedule	Pipe (Bore) Thermal Resistance (h•ft•F°/Btu)			
		For Water Flows Above 2.0 US gpm	20% Prop. Glycol Flow 3.0 US gpm	20% Prop. Glycol Flow 5.0 US gpm	20% Prop. Glycol Flow 10.0 US gpm
¾ in. (0.15 ft)	SDR 11	0.09	0.12	NR	NR
	SDR 9	0.11	0.15	NR	NR
	Sch 40	0.10	0.14	NR	NR
1.0 in. (0.18 ft)	SDR 11	0.09	0.14	0.10	NR
	SDR 9	0.11	0.16	0.12	NR
	Sch 40	0.10	0.15	0.11	NR
1 1/4 in. (0.22 ft)	SDR 11	0.09	0.15	0.12	0.09
	SDR 9	0.11	0.17	0.15	0.11
	Sch 40	0.09	0.15	0.12	0.09
1 1/2 in. (0.25 ft)	SDR 11	0.09[1]	0.16	0.15	0.09
	SDR 9	0.11[1]	0.18	0.17	0.11
	Sch 40	0.08[1]	0.14	0.14	0.08

Figure 48 *–Pipe Thermal Resistance Table*



Natural Soil Cond.	0.9 Btu/h•ft•F°		1.3 Btu/h•ft•F°			1.7 Btu/h•ft•F°	
Backfill or Grout Conductivity	0.5 Btu/h•ft•F°	2.0 Btu/h•ft•F°	0.5 Btu/h•ft•F°	1.0 Btu/h•ft•F°	2.0 Btu/h•ft•F°	0.5 Btu/h•ft•F°	1.0 Btu/h•ft•F°
4 in. Bore							
¾ in. U-tube	0.11 (NR)	-0.05	0.14 (NR)	0.03	-0.02	0.17 (NR)	0.05
1 in U-tube	0.07	-0.03	0.09	0.02	-0.02	0.13 (NR)	0.04
5 in. Bore							
¾ in. U-tube	0.14 (NR)	-0.06	0.18 (NR)	0.04	-0.04	0.21 (NR)	0.06
1 in U-tube	0.11 (NR)	-0.04	0.14 (NR)	0.03	-0.02	0.16 (NR)	0.05
1 1/4 in U-tube	0.06	-0.03	0.09	0.02	-0.02	0.12 (NR)	0.04
6 in. Bore							
¾ in. U-tube	0.18 (NR)	-0.07	0.21 (NR)	0.04	-0.05	0.24 (NR)	0.07
1 in U-tube	0.14 (NR)	-0.06	0.17 (NR)	0.03	-0.04	0.21 (NR)	0.06
1 1/4 in U-tube	0.09	-0.04	0.12 (NR)	0.03	-0.02	0.15 (NR)	0.05
1 1/2 in U-tube	0.07	-0.03	0.09	0.02	-0.02	0.11 (NR)	0.04

Figure 49 *–Bore and Grout Thermal Resistance Correction Table*

8.3.4 Power Input at Design Cooling Load (W_c)

To account for the power input at the design cooling load, 50,000 BTU/hr was assumed. This includes the heat added to system by the pumps and a safety factor to account for

8.3.5 Part Load Factor (PLF_M)

Without specific building performance data available for the Koch Institute, the part load factor is unknown. To ensure that the ground source heat pump was not undersized, the worst case scenario was assumed and a **PLF_M=1** was entered into EES.

8.4 Results

After entering all of the forementioned parameters into the EES program, the required bore length to meet the cooling loads of the system was calculated to be **40,586 ft**. The detailed results generated with the EES program can be found in the ***Appendix*** of this report.

8.4.1 System Layout

To design an effective layout for the 40,586 ft of underground piping, many variables were evaluated. The available space for the geothermal field, drilling costs, piping costs, impacts on the construction schedule, and the integration into the designed system must be considered. In the Construction Management Breadth Section of this report, these variables were considered and the optimum design resulted in 185 bores drilled to 219 ft each.

The system is designed using Reverse-Return headers that provides self balancing which eliminates the need for additional balancing valves. This set up also reduces head loss, allowing for smaller pumps. A schematic example of the Reverse-Return headers is shown below in ***Figure 50***. The reductions in pipe size to and from the loops helps to prevent air trapping as well as maintain proper pressure within the system.

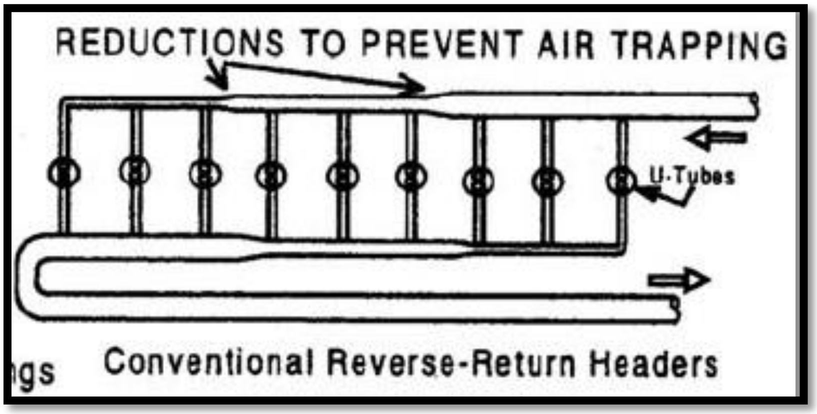


Figure 50 *–Typical Reverse-Return Header Schematic*

To reduce the need for large diameter piping, these 185 bores were layed out in 37 sets with 5 boreholes each. The layout of the boreholes is shown below in ***Figure 51*** . To fit in the available area economically, the 37 borehole sets were divided into three subgroups which makes it possible to use multiple headers, making the system easier to manage and control efficiently. The 2 longer subgroups service 15 sets of boreholes each, and the third services the remaining 7 sets.

The field available for the borehole layout allowed for each borehole to be spaced 15 ft apart horizontally and 20 ft apart vertically. This spacing avoids a rise in ground temperature overtime and allows each borehole to dissapate heat to the ground effectively. This spacing provides an economical solution to an efficient design that fits within the required area.

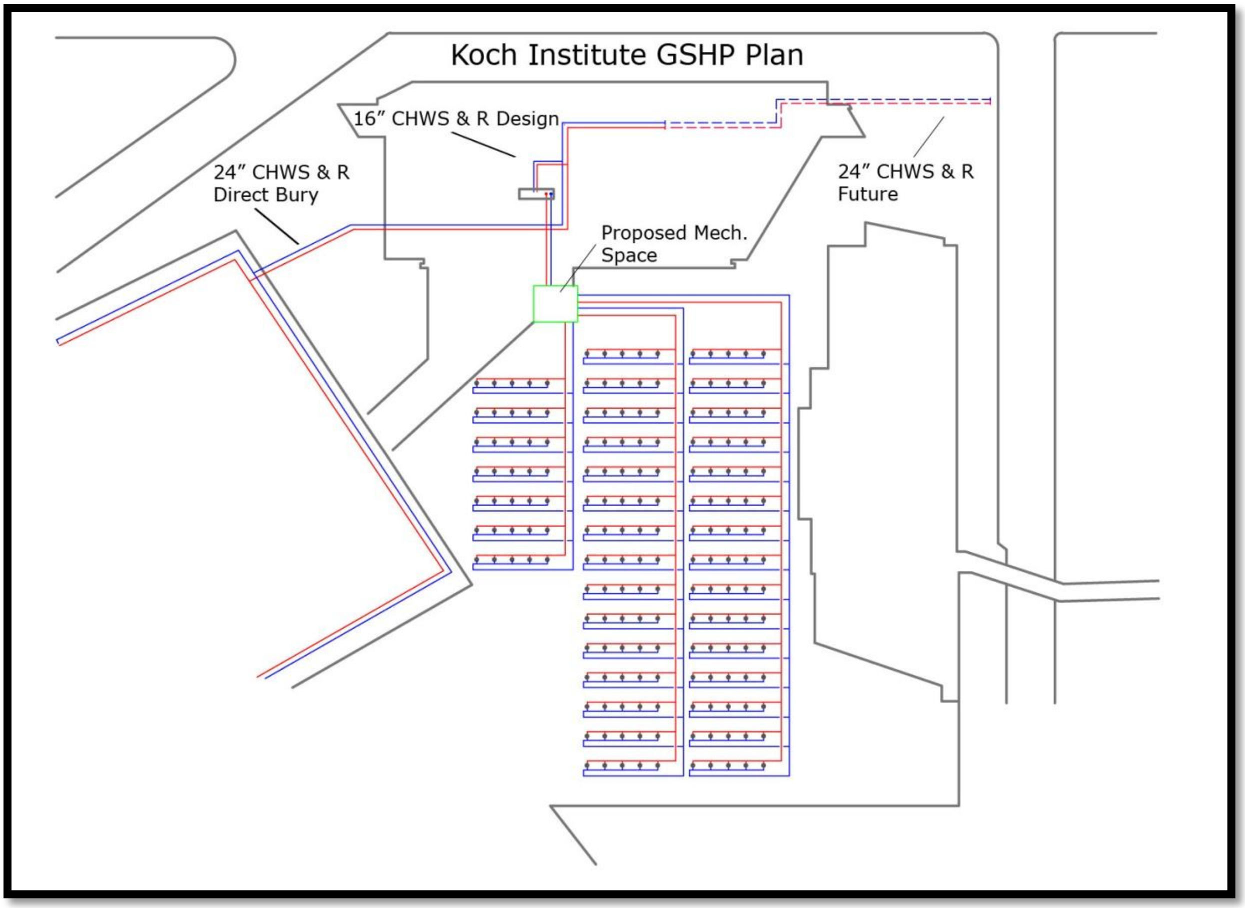


Figure 51 *–Schematic of Ground Source Heat Pump (Boreholes and Piping)*

To reduce the amount of penetrations and piping running into the building, a mechanical space located in the tunnel system has been planned. This area, labeled "Proposed Mech. Space" in the figure above, houses the pumping system for the geothermal wells and a heat exchanger that connects the building load to the heat sink. This mechanical space and pumping system is described in furthur detail on the following page.

8.4.2 Pumping –Configuration and Selection

Dividing the system into three goethermal loops makes it possible to meet part load conditions at higher efficiency. A single geothermal loop sized for the design load will run at part load most of the year, decreasing the efficiency of the pump as well as the capacity of the ground loop. The advantages of creating the three separate loops are as follows:

- **Heat transfer in operating ground loops is maximized at part load (non laminar flow).**
- **Multiple loops builds redundancy and smaller pumps operating at full capacity increases efficiency**
- **Cycling between loops:**
    - **Minimizes the rise ground temperature (t_g) over time**
    - **The fluid in loops that are not in operation fully dissapate heat to the ground.**
    - **The system is able to respond quickly to spikes in chilled water demand utilizing the water in the unoperating loop.**
- **Separate headers reduces pipe size and aids in system flushing.**

Ground Source Chilled Water Pumps 1, 2 & 3 operate their respective ground loop independently. As the demand for Chilled Water increases beyond a single loop capacity, the next pump begins to ramp up, activating another loop. At design load, all pumps operate at full capacity to provide 52°F Chilled Water to the Air Handlers.

At part load, the system cycles between loops to prevent an increase in ground temperature and maximize the heat transfer.

This pumping station is located in the tunnel South of the building, as shown in ***Figure 51*** on the previous page. This location minimizes changes to the original design, only introducing the CHWS & R headers. These headers run 60 ft to the West Shaft, where they tie into the 4” pipe that directly serves the spot cooling Air handlers.

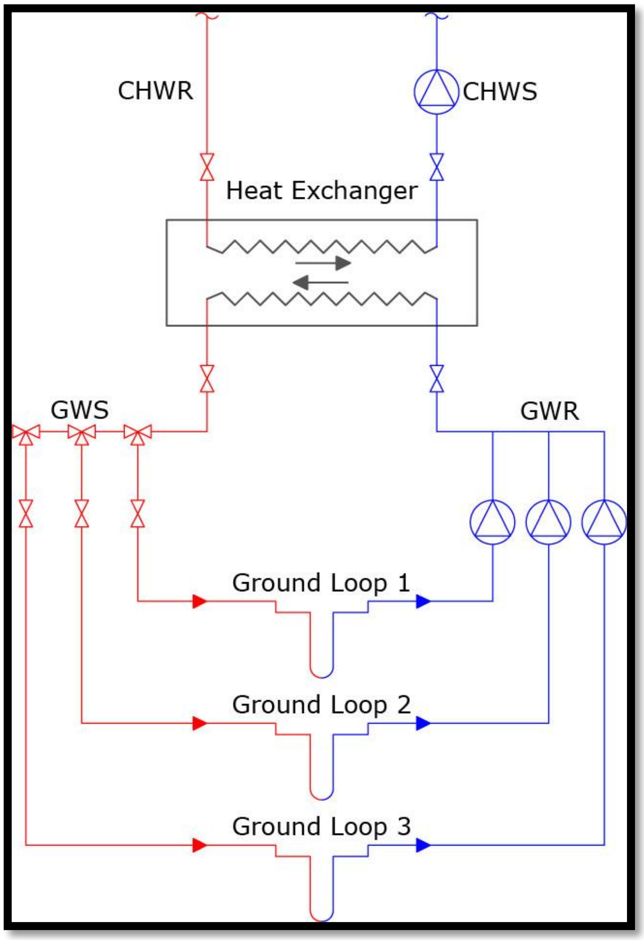


Figure 52 *–Pumping Schematic*

The longest run of loops 1, 2 and three shown in ***Figure 52*** on the previous page are shown below in ***Figures 53 & 54.*** Loops 1 and 2 have nearly identical loop dimensions and therefore have been drawn with one schematic. From these diagrams, system head pressure was calculated based on losses due to friction and fittings to size the 3 loop pumps. (*Isolation valves are not depicted in the figures but were taken into account in the calculations.*)

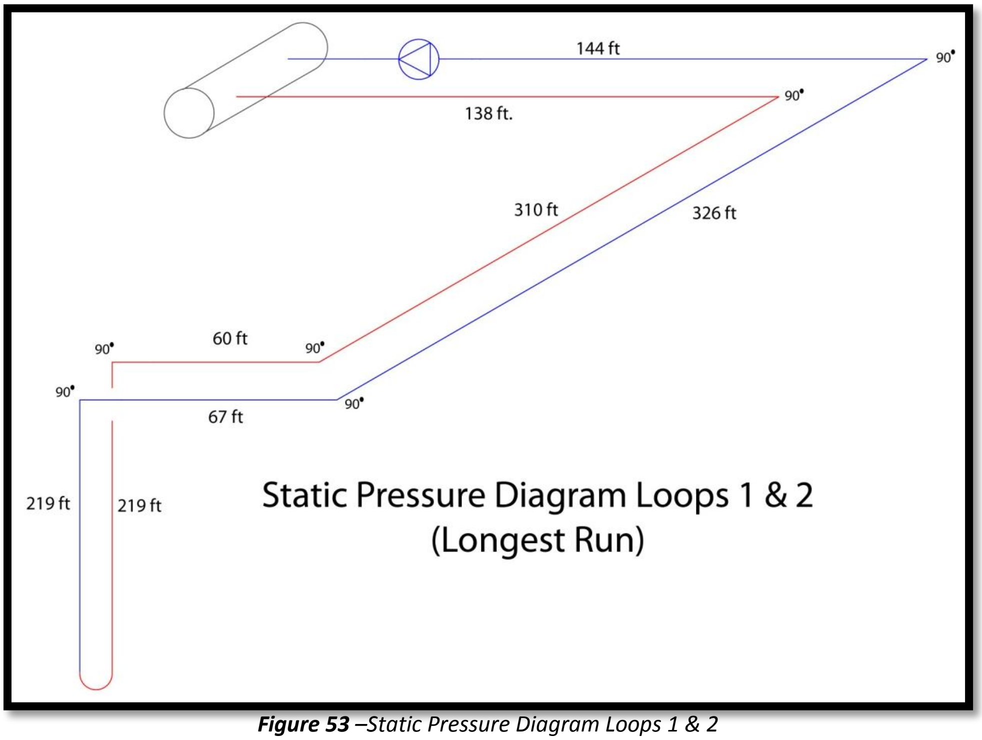


Figure 53 *–Static Pressure Diagram Loops 1 & 2*

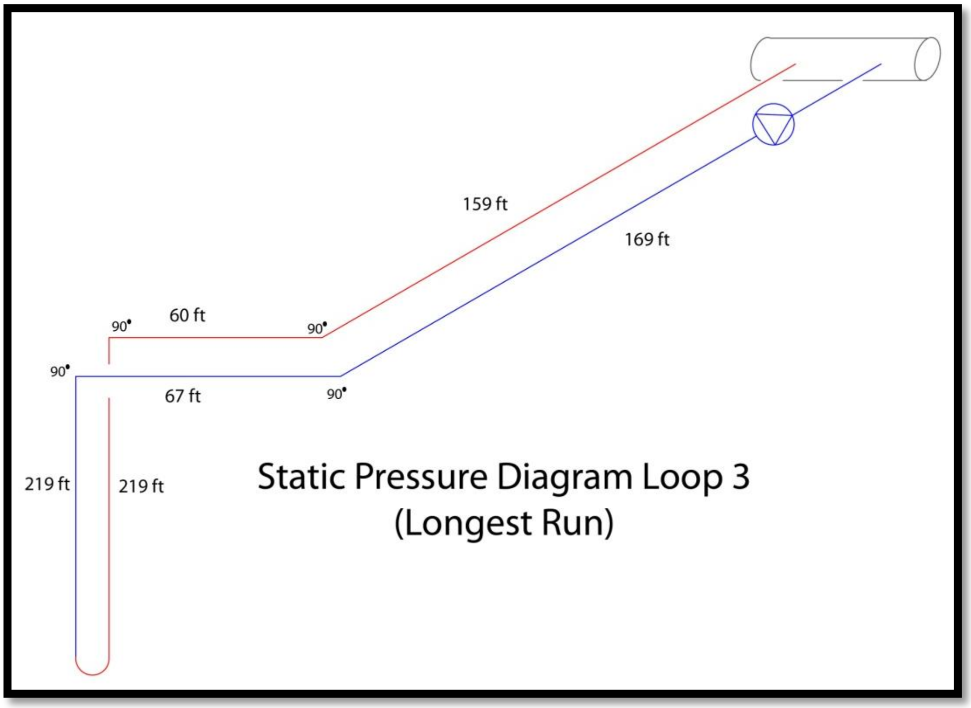


Figure 54 *–Static Pressure Diagram Loop 3*

During the ground loop sizing the gpm for the whole system was determined to be 481 gpm (based on a 3 gpm/ton recommendation by McQuay). The system characteristics utilized for pump sizing are shown in ***Figure 55*** below.

	Ground Loop 1	Ground Loop 2	Ground Loop 3	CHW Loop
Capacity (GPM)	200	200	100	500
Total Head (ft h20)	80	80	60	80

Figure 55 *–System Characteristics for Pump Sizing*

The 3 ground loop pumps and chilled water pump were then selected utilizing pump curves from *Bell and Gossett Curve Booklet B-260G Series 1510* (pump curves found in ***Appendix***). The resulting pumps are 4 Base-mounted, end suction Series 1510 pumps.

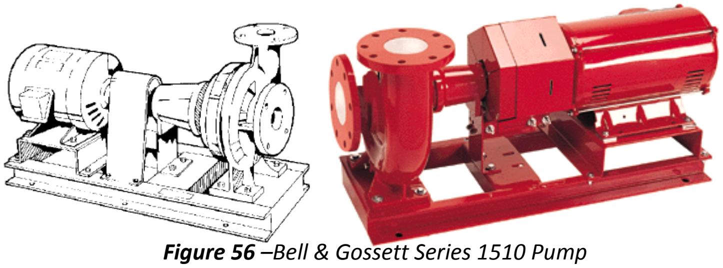


Figure 56 *–Bell & Gossett Series 1510 Pump*

Ground Chilled Water Pumps

Unit	Manufact.	Frame Size	Service	Type	GPM	Total Head (f.t. H_20)	VFD	Emer. Power	Min Casing Size Disc x Inlet x Impel.	Motor Data at 60 Hz			
										HP	RPM	Volts	Phase
GCHWP-1	Bell & Goss.	213T	HE-4	End Suction	200	80	Y	Y	2"x 2.5"x 9.5"	7.5	1750	480	3
GCHWP-2	Bell & Goss.	213T	HE-4	End Suction	200	80	Y	Y	2"x 2.5"x 9.5"	7.5	1750	480	3
GCHWP-3	Bell & Goss.	182T	HE-4	End Suction	100	60	Y	Y	1.5"x 2"x 8"	3	1750	480	3

Assumptions

20% Ethylene Glycol based Water Solution with Specific Gravity @ 50 °F = 1.07 adjusting horsepowers accordingly

Chilled Water Pumps

Unit	Manufact.	Frame Size	Service	Type	GPM	Total Head (f.t. H_20)	VFD	Emer. Power	Min Casing Size Disc x Inlet x Impel.	Motor Data at 60 Hz			
										HP	RPM	Volts	Phase
CHWP-3	Bell & Goss.	245T	HE-4	End Suction	500	80	Y	Y	4"x 5"x 9.5"	15	1750	480	3

Figure 57 *–GCHW and CHW Pump Schedule*

8.4.3 System Piping

High-Density Polyehtelene Piping that is thermally fused is the ideal choice for ground source heat pumps. Each ground loop utilizes 1 ½ " SDR 11 piping that is rated at 100 psi.

8.4.4 System Flow

Each pump is operated by a Variable Frequency Drive with a bypass that maintains the minimum flow rate at 33% of design. To properly balance this with required pumping power, the piping is sized to achieve non laminar flow at design conditions. During part load conditions the oversized loop will offset the loss in heat transfer due to laminar flow conditions.

9.0 Construction Management Breadth Study

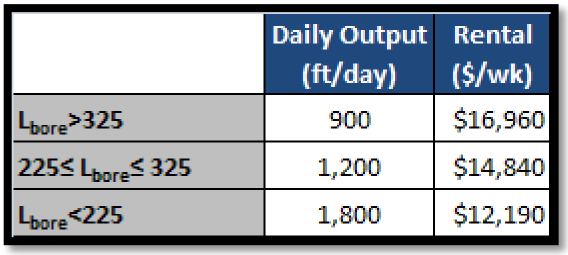
9.1 Objectives

The construction of the large geothermal system outlined in the previous section involves heavy construction in an already time sensitive schedule. The construction of a vertical loop ground source heat pump can be expensive and time consuming. The study in this section of the report was performed to minimize the capital cost and impact on the construction schedule. With the mechanical performance of the ground source heat pump as the driving factor, this study evaluates the cost of drilling, piping, grouting and other miscellaneous site costs to optimize the number and depth of required boreholes. All estimated values of cost and daily outputs of equipment/crew were taken from *RS Means Mechanical Cost Data – 2009* based on the projects construction duration of March 2008 - Winter 2011.

9.2 Estimation Assumtions

9.2.1 Drilling Cost

Drilling costs rely on the equipment utilized and the capabilities of the crew. This study compared the use of three different augers capable of drilling to different depths. ***Figure 58*** shows the daily output and weekly rental cost of each auger, categorized by the borehole depth (L_{bore}) it is capable of drilling.



	Daily Output (ft/day)	Rental ($/wk)
$L_{bore}>325$	900	$16,960
$225\leq L_{bore}\leq 325$	1,200	$14,840
$L_{bore}<225$	1,800	$12,190

Figure 58 *–Auger Comparison Table*

9.2.2 Piping Cost

As discussed in the previous section, the ground loop piping is comprised of 1 ½ " Thermally Fused High Density Polyethelyne Piping. Price estimates for piping are given in dollar per linear foot values, 1 ½ " HDPE piping is estimated at $0.66/LF. Additional 3" HDPE piping was utilized for headers and is estimated at $1.32/LF. According to RS-Means Mechanical Cost data 2009, every 40 ft. of pipe must be welded together, costing an extra $25/weld and $55/day to rent the proper equipment.

9.2.3 Grouting Cost

The cost of grouting for this system is a fixed cost based on the length of borehole and was estimated to be $8,900. Therefore, this cost remains the same for every combination of borehole depth and number.

9.2.4 Miscellaneous Costs

Throughout the construction miscellaneous costs are inevitable. Upon completion the system will have to be flushed, tested and commissioned to ensure proper operation. This cost also builds in a safety factor to account for poor weather conditions that will not allow for drilling. Due to the equipment rentals, such conditions can incurr extra costs that are included in this segment of the estimation.

9.3 Borehole Optimization Results

With the total required borehole length known to be 40,586 ft, the number and depth of bores was decided upon based on this cost analysis. Figure # on the following page was utilized to find the combination of number and depth of bores with minimum assosciated cost. This analysis resulted in a system that utilizes 185 bores, each at a depth of 219 ft. ***Figure 59*** below shows a graph of the number of bores vs. total cost.

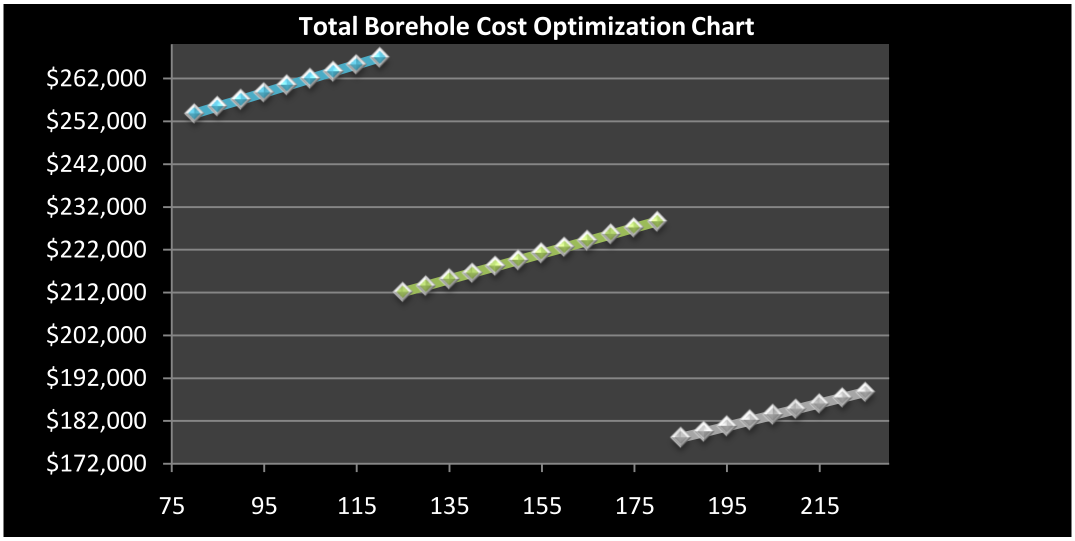


Figure 59 *–Borehole Cost Optimization Chart*

The graph above shows that the auger selection is the driving factor in the overall cost. The three different trends on the graph (blue, green and grey) represent the three augers evaluated. The auger that is only capable of drilling to depths less than 225 ft. has the highest daily output along with the cheapest weekly rental. At these depths, the ground is softer and easier to drill, so the auger is fast and effective, making it the cheapest of the three. The ground temperature below 5 feet is consistent, therefore the depth of 219 ft per borehole will maintain the integrity of the overall system performance.

The tables on the following page were utilized to generate these results and optimize design. As can be seen in ***Figures 60, 61 & 62***, the borehole layout based on the drilling cost, piping cost, grouting cost and miscellaneous costs for each drilling auger by varying the number of boreholes in increments of 5.

	L_{total}	Number of Bores	Bore Depth	day/ bore	days	weeks	Drilling Cost	Piping Cost	Grouting Cost	Misc Cost	Total Cost
$L_{bore}>325$	40586	80	507	0.564	50.056	10.01	$169,788	$50,639	$8,900	$24,500	**$253,827**
	40586	85	477	0.531	50.366	10.07	$170,840	$50,720	$8,900	$25,000	**$255,460**
	40586	90	451	0.501	50.676	10.14	$171,891	$50,802	$8,900	$25,500	**$257,093**
	40586	95	427	0.475	50.986	10.20	$172,943	$50,883	$8,900	$26,000	**$258,726**
	40586	100	406	0.451	51.296	10.26	$173,995	$50,965	$8,900	$26,500	**$260,359**
	40586	105	387	0.429	51.606	10.32	$175,046	$51,046	$8,900	$27,000	**$261,992**
	40586	110	369	0.410	51.916	10.38	$176,098	$51,128	$8,900	$27,500	**$263,625**
	40586	115	353	0.392	52.226	10.45	$177,149	$51,209	$8,900	$28,000	**$265,258**
	40586	120	338	0.376	52.536	10.51	$178,201	$51,291	$8,900	$28,500	**$266,891**

Figure 60 *–Auger Capable of L_{bore} >325 ft*

	L_{total}	Number of Bores	Bore Depth	day/ bore	days	weeks	Drilling Cost	Piping Cost	Grouting Cost	Misc Cost	Total Cost
$225 \leq L_{bore} \leq 325$	40586	125	325	0.271	41.572	8.31	$123,385	$50,808	$8,900	$29,000	**$212,093**
	40586	130	312	0.260	41.882	8.38	$124,305	$50,890	$8,900	$29,500	**$213,595**
	40586	135	301	0.251	42.192	8.44	$125,225	$50,971	$8,900	$30,000	**$215,096**
	40586	140	290	0.242	42.502	8.50	$126,145	$51,053	$8,900	$30,500	**$216,598**
	40586	145	280	0.233	42.812	8.56	$127,065	$51,134	$8,900	$31,000	**$218,099**
	40586	150	271	0.225	43.122	8.62	$127,985	$51,216	$8,900	$31,500	**$219,601**
	40586	155	262	0.218	43.432	8.69	$128,905	$51,297	$8,900	$32,000	**$221,103**
	40586	160	254	0.211	43.742	8.75	$129,825	$51,379	$8,900	$32,500	**$222,604**
	40586	165	246	0.205	44.052	8.81	$130,745	$51,460	$8,900	$33,000	**$224,106**
	40586	170	239	0.199	44.362	8.87	$131,665	$51,542	$8,900	$33,500	**$225,607**
	40586	175	232	0.193	44.672	8.93	$132,586	$51,623	$8,900	$34,000	**$227,109**
	40586	180	225	0.188	44.982	9.00	$133,506	$51,705	$8,900	$34,500	**$228,610**

Figure 61 *–Auger Capable of 225 ≤ L_{bore} ≤ 325 ft*

	L_{total}	Number of Bores	Bore Depth	day/ bore	days	weeks	Drilling Cost	Piping Cost	Grouting Cost	Misc Cost	Total Cost
$L_{bore}<225$	40586	185	219	0.122	34.018	6.80	$82,935	$51,223	$8,900	$35,000	**$178,058**
	40586	190	214	0.119	34.328	6.87	$83,691	$51,304	$8,900	$35,500	**$179,395**
	40586	195	208	0.116	34.638	6.93	$84,447	$51,386	$8,900	$36,000	**$180,733**
	40586	200	203	0.113	34.948	6.99	$85,203	$51,467	$8,900	$36,500	**$182,070**
	40586	205	198	0.110	35.258	7.05	$85,958	$51,549	$8,900	$37,000	**$183,407**
	40586	210	193	0.107	35.568	7.11	$86,714	$51,630	$8,900	$37,500	**$184,744**
	40586	215	189	0.105	35.878	7.18	$87,470	$51,712	$8,900	$38,000	**$186,082**
	40586	220	184	0.102	36.188	7.24	$88,226	$51,793	$8,900	$38,500	**$187,419**
	40586	225	180	0.100	36.498	7.30	$88,982	$51,875	$8,900	$39,000	**$188,756**

Figure 62 *–Auger Capable of L_{bore} < 225 ft*

10.0 Electrical Breadth Study

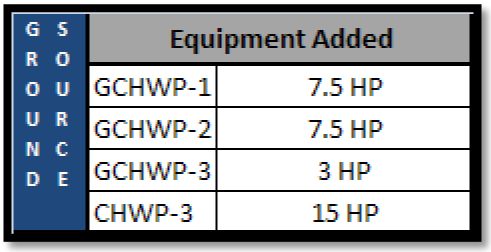
10.1 Objectives

The proposed ground source heat pump and glycol run around heat recovery loops require additional pumps that impose new electrical loads on the buildings electrical system. This study evaluates the impact on the existing system and proposes an additional distribution panel as well as multiple over current protection devices.

10.2 Electrical Load Calculations

10.2.1 Equipment Electrical Loads

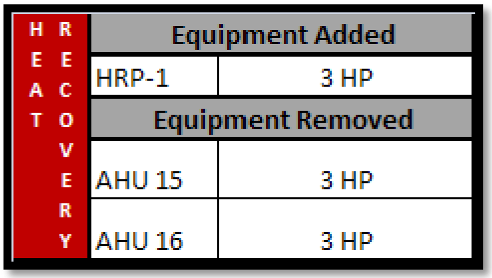
First, the horsepower of the equipment added and removed from each system was determined. ***Figure 63*** below shows the three ground loop pumps and the chilled water supply pump that were added to the design.



GROUND SOURCE	Equipment Added	
	GCHWP-1	7.5 HP
	GCHWP-2	7.5 HP
	GCHWP-3	3 HP
	CHWP-3	15 HP

Figure 63 *–Equipment Added GSHP*

Figure 64 below shows the heat recovery pump added to the design as well as the two air handlers that were removed.



HEAT RECOVERY	Equipment Added	
	HRP-1	3 HP
	Equipment Removed	
	AHU 15	3 HP
	AHU 16	3 HP

Figure 64 *–Equipment Added & Removed Heat Recovery*

10.2.2 Full Load Current

Utilizing ***NEC 2008 Table 430.250*** *Full Load Current, Three-Phase Alternating Current Motors (found in **Appendix**)*, each motor's full load current was specified to be:

- 3 HP Motors @ 460V – 4.8 A
- 7 ½ HP Motors @ 460V – 11 A
- 15 HP Motors @ 460V – 21 A

10.2.3 Over Current Protection Device

With the full load current the over-current protection device can then be sized. Common breaker sizes were taken from ***NEC 2008 240.6 Standard Ampere Ratings*** – (A) Fuses and Fixed-Trip Circuit Breakers.

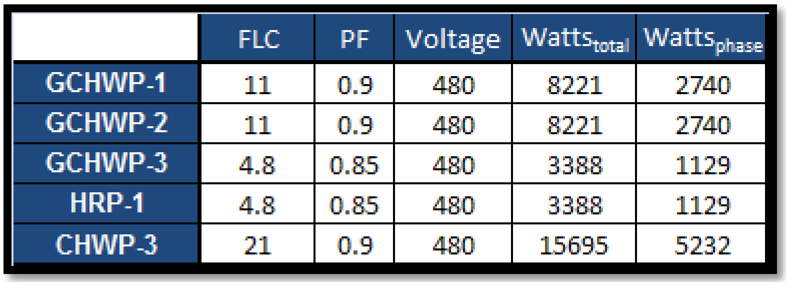
10.2.4 Connected Load

The equation below was used to calculate the connected load. The Watts_{total} are then divided by three to yield the Watts_{phase}.

$$W = FLC \times 1.73 \times Voltage \times PF$$

Power Factors –

*Motors < 5HP – **PF = 0.85***
*Motors > 5HP – **PF = 0.9***



	FLC	PF	Voltage	Watts_{total}	Watts_{phase}
GCHWP-1	11	0.9	480	8221	2740
GCHWP-2	11	0.9	480	8221	2740
GCHWP-3	4.8	0.85	480	3388	1129
HRP-1	4.8	0.85	480	3388	1129
CHWP-3	21	0.9	480	15695	5232

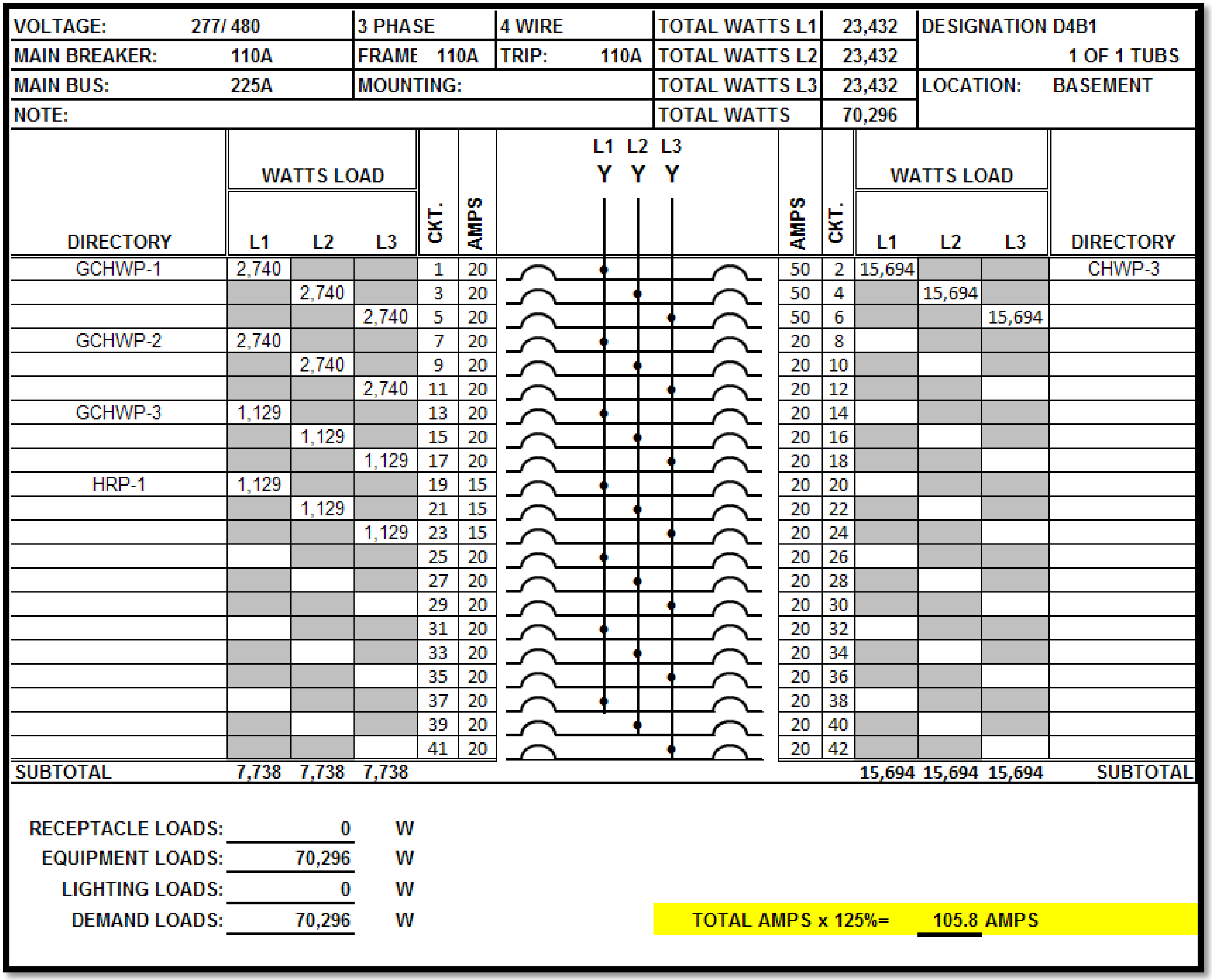
Figure 65 *–Connected Load Calculation Table*

10.2.5 Feeder Sizing

To size the feeders to each motor ***NEC 2008 Table 310.16*** was consulted. The resulting feeders and conduit sizes for the motors are shown in the distribution panel board schedule on the following page in ***Figure 66.***

10.2.6 Panelboard Schedule

The panel board schedule is located on the following page in ***Figure 66.*** To size the feeders to each motor ***NEC 2008 Table 310.16*** was consulted. The 110 A main breaker to the panel was sized from the Total Amps x 1.25 shown at the bottom of ***Figure 66*** in yellow. The 225 A main bus is the next available bus size past 100A.



VOLTAGE:	277/ 480	3 PHASE	4 WIRE	TOTAL WATTS L1	23,432	DESIGNATION D4B1
MAIN BREAKER:	110A	FRAME 110A	TRIP: 110A	TOTAL WATTS L2	23,432	1 OF 1 TUBS
MAIN BUS:	225A	MOUNTING:		TOTAL WATTS L3	23,432	LOCATION: BASEMENT
NOTE:				TOTAL WATTS	70,296	

DIRECTORY	WATTS LOAD L1	WATTS LOAD L2	WATTS LOAD L3	CKT.	AMPS	AMPS	CKT.	WATTS LOAD L1	WATTS LOAD L2	WATTS LOAD L3	DIRECTORY
GCHWP-1	2,740			1	20	50	2	15,694			CHWP-3
		2,740		3	20	50	4		15,694		
			2,740	5	20	50	6			15,694	
GCHWP-2	2,740			7	20	20	8				
		2,740		9	20	20	10				
			2,740	11	20	20	12				
GCHWP-3	1,129			13	20	20	14				
		1,129		15	20	20	16				
			1,129	17	20	20	18				
HRP-1	1,129			19	15	20	20				
		1,129		21	15	20	22				
			1,129	23	15	20	24				
				25	20	20	26				
				27	20	20	28				
				29	20	20	30				
				31	20	20	32				
				33	20	20	34				
				35	20	20	36				
				37	20	20	38				
				39	20	20	40				
				41	20	20	42				
SUBTOTAL	7,738	7,738	7,738					15,694	15,694	15,694	SUBTOTAL

RECEPTACLE LOADS:	0	W
EQUIPMENT LOADS:	70,296	W
LIGHTING LOADS:	0	W
DEMAND LOADS:	70,296	W

TOTAL AMPS x 125%= 105.8 AMPS

***Figure 66** –Distribution Panel-board Schedule D4B1*

10.2.7 One Line Schematic

Figure # and # below show the Panelboard D4B-1's connection to the existing system. D4B-1 ties into an existing 225 Amp spare on D4B located in the basement near the pumping station being served. D4B ties directly into Unit Sub Station A which is fed by the campus power as well as the incoming 13.8 kV service.

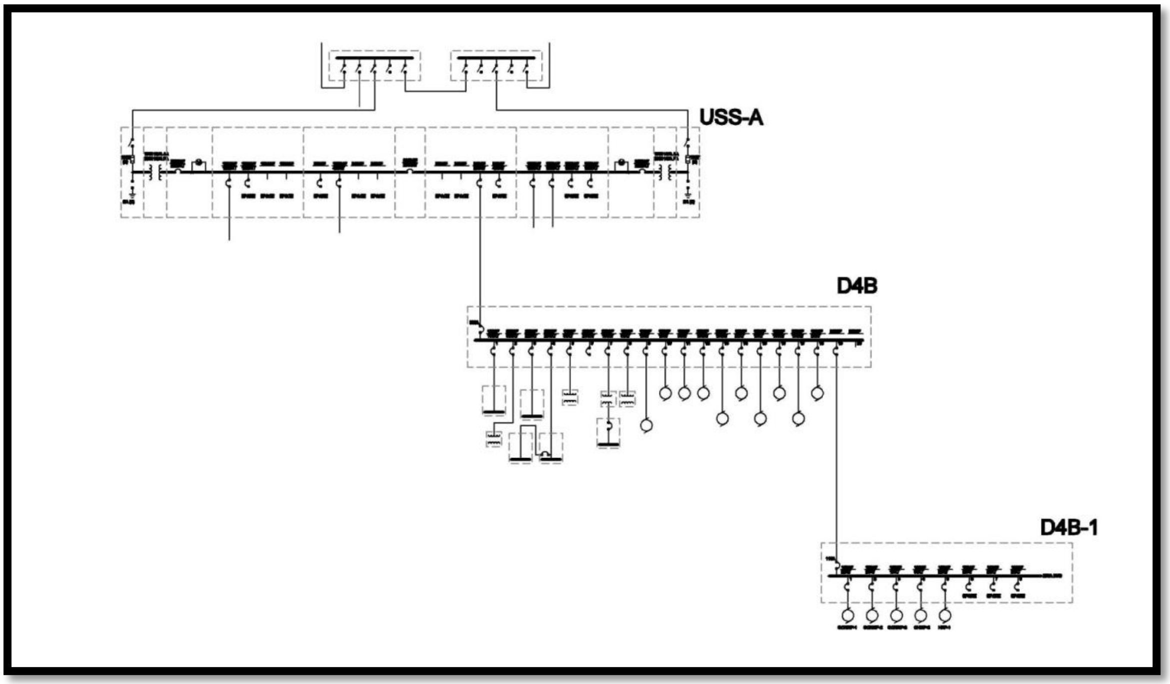


Figure 67 *–One Line Schematic of Redesign Incorporation with Existing Design*

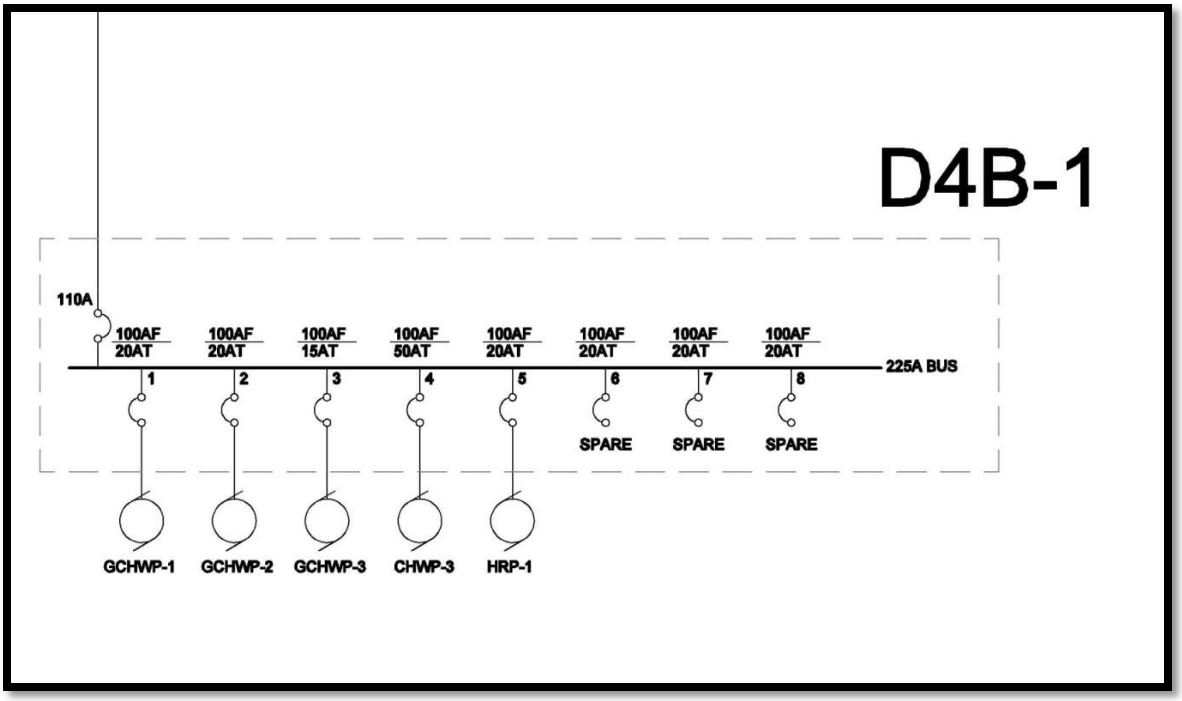


Figure 67 *–One Line Schematic of Added Panel-board D4B1*

11.0 Energy and Cost Evaluation of Redesign

11.1 Energy Savings

The heat recovery system reduced the peak heating load by **400 MBH, therefore from 9,588 to 9,188 MBH**. This is a 4.2 % reduction in the peak heating load that reduces the annual consumption by 784 therms, saving a total of $965. A limited amount of feasible applications for the recovered heat left stair-shaft heating and cooling units the only option. The designed heat recovery loop is capable of recovering up to 1529 MBH. The graph below represents the energy reduction provided by the heat recovery system.

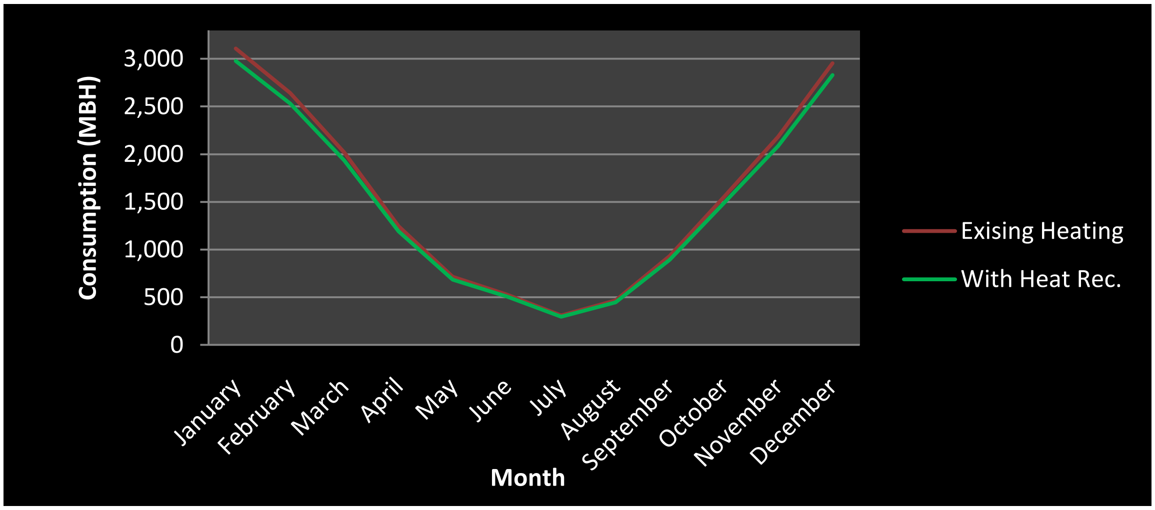


Figure 68 *–Annual Energy Consumption Heating (Existing vs. Redesign)*

The ground source heat pump provides significantly more energy savings than the heat recovery loop. This system reduced the Koch Institute's annual load on the cogeneration plant from 817,137 therms to 729,264 therms, a total of **87,873 therms**. This reduction saves MIT's Cogeneration plant **$86,651/year** in chilled water production.

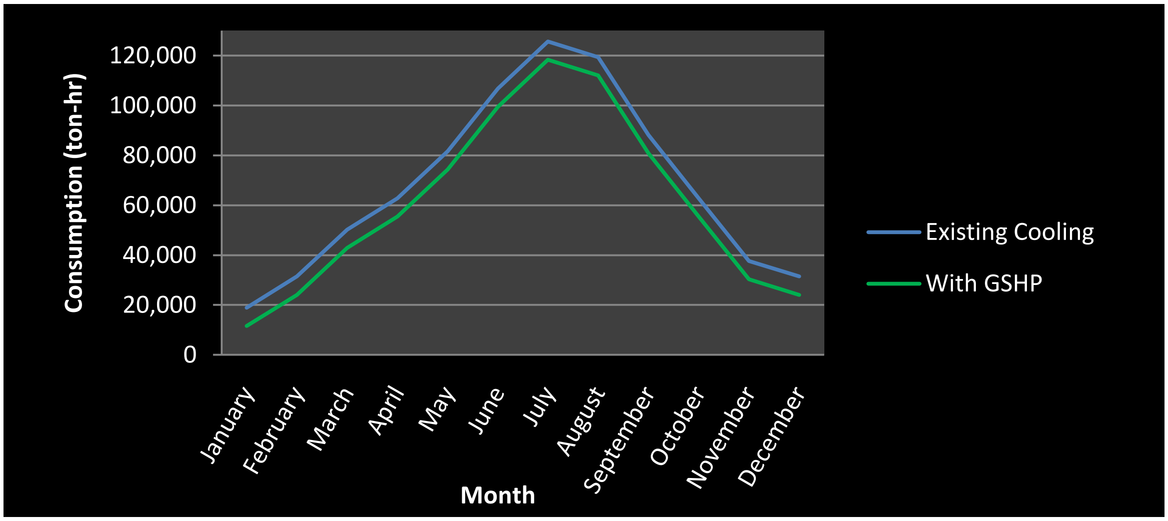
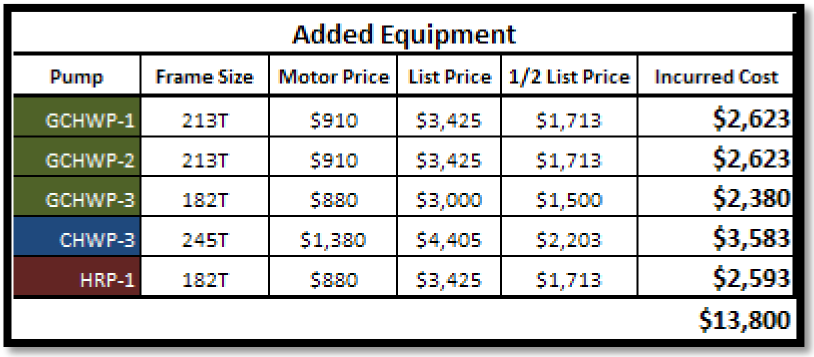


Figure 69 *–Annual Energy Consumption Cooling (Existing vs. Redesign)*

11.2 Equipment (Added & Removed)

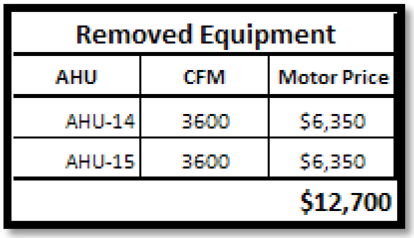
The heat recovery and ground source heat pump systems require additional pumps. Estimates for the pumps were gotten from Thermoflo Equipment Co. and are shown below. The pumps were assumed to be at ½ list price per the recommendation of the sales rep at Thermoflo. Similarly, estimates for the 3600 cfm units (AHU-14 & 15) that were removed from the design were provided by Commercial AIRE Products and are shown below.



Added Equipment

Pump	Frame Size	Motor Price	List Price	1/2 List Price	Incurred Cost
GCHWP-1	213T	$910	$3,425	$1,713	$2,623
GCHWP-2	213T	$910	$3,425	$1,713	$2,623
GCHWP-3	182T	$880	$3,000	$1,500	$2,380
CHWP-3	245T	$1,380	$4,405	$2,203	$3,583
HRP-1	182T	$880	$3,425	$1,713	$2,593
					$13,800

Figure 70 –Added Pumps



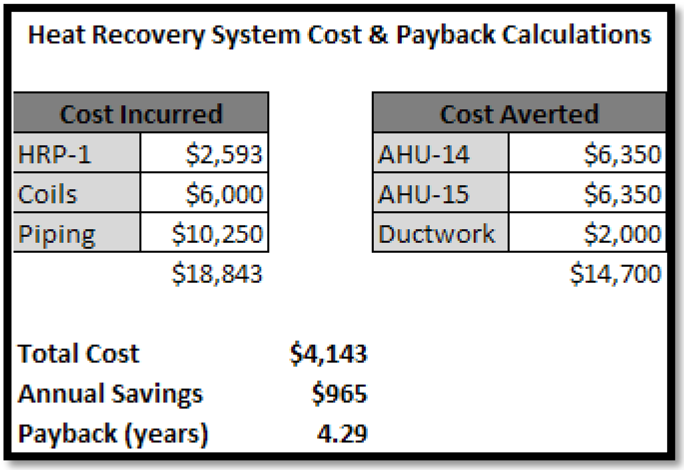
Removed Equipment

AHU	CFM	Motor Price
AHU-14	3600	$6,350
AHU-15	3600	$6,350
		$12,700

Figure 71 –Removed AHU's

11.3 System Cost and Payback

The payback of the heat recovery system was calculated with the specific costs of added equipment and potential savings from energy reduction. The results of this calculation are shown below in ***Figure 72.***



Heat Recovery System Cost & Payback Calculations

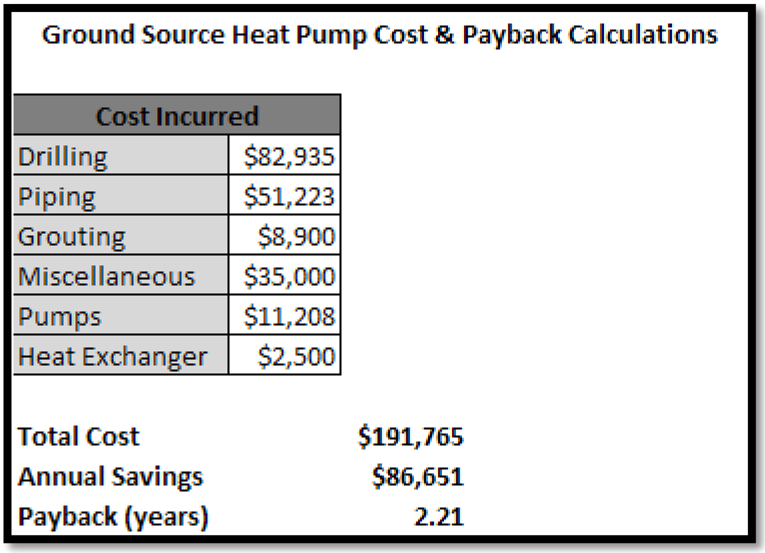
Cost Incurred	
HRP-1	$2,593
Coils	$6,000
Piping	$10,250
	$18,843

Cost Averted	
AHU-14	$6,350
AHU-15	$6,350
Ductwork	$2,000
	$14,700

Total Cost	$4,143
Annual Savings	$965
Payback (years)	4.29

Figure 72 –Heat Recovery Payback

This process was repeated for the ground source heat pump system. The results of this calculation are shown on the following page in ***Figure 73.***



Ground Source Heat Pump Cost & Payback Calculations

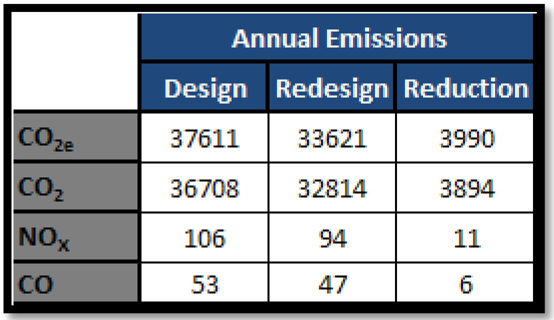
Cost Incurred	
Drilling	$82,935
Piping	$51,223
Grouting	$8,900
Miscellaneous	$35,000
Pumps	$11,208
Heat Exchanger	$2,500

Total Cost	**$191,765**
Annual Savings	**$86,651**
Payback (years)	**2.21**

Figure 73 *–Ground Source Heat Pump Payback*

11.4 Annual Emissions

The redesigned system that incorporates the heat recovery and ground source heat pump significantly reduces the annual emission of pollutants. The table below in ***Figure 74*** shows the reduction of CO_{2e}, CO_2, Nox and CO.



	Annual Emissions		
	Design	Redesign	Reduction
CO_{2e}	37611	33621	3990
CO_2	36708	32814	3894
NO_X	106	94	11
CO	53	47	6

Figure 74 *–Annual Emissions Reduction Table*

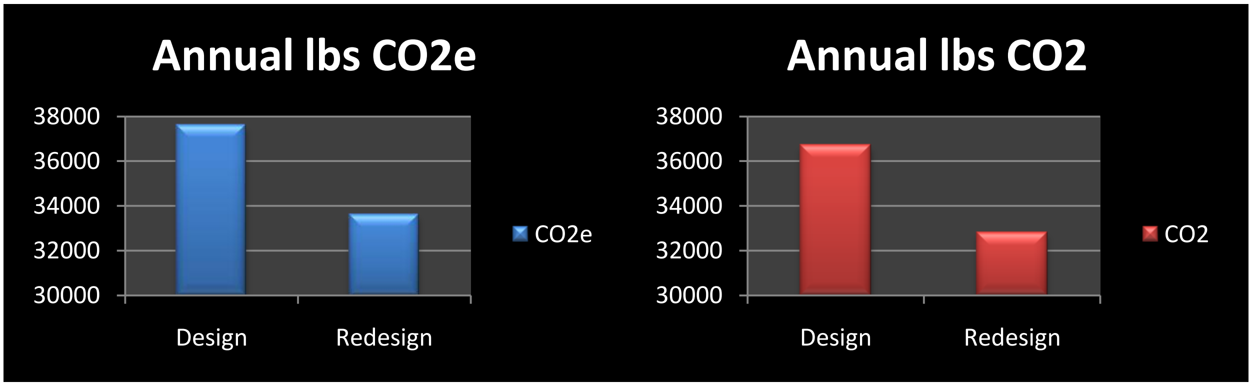


Figure 75 *–Annual lbs CO2e and CO2 (Design vs. Redesign)*

12.0 Credits and Acknowledgements

I would like to express my gratitude to all that have aided me throughout my yearlong study of the David H. Koch Institute for Integrative Cancer Research. A special thanks to:

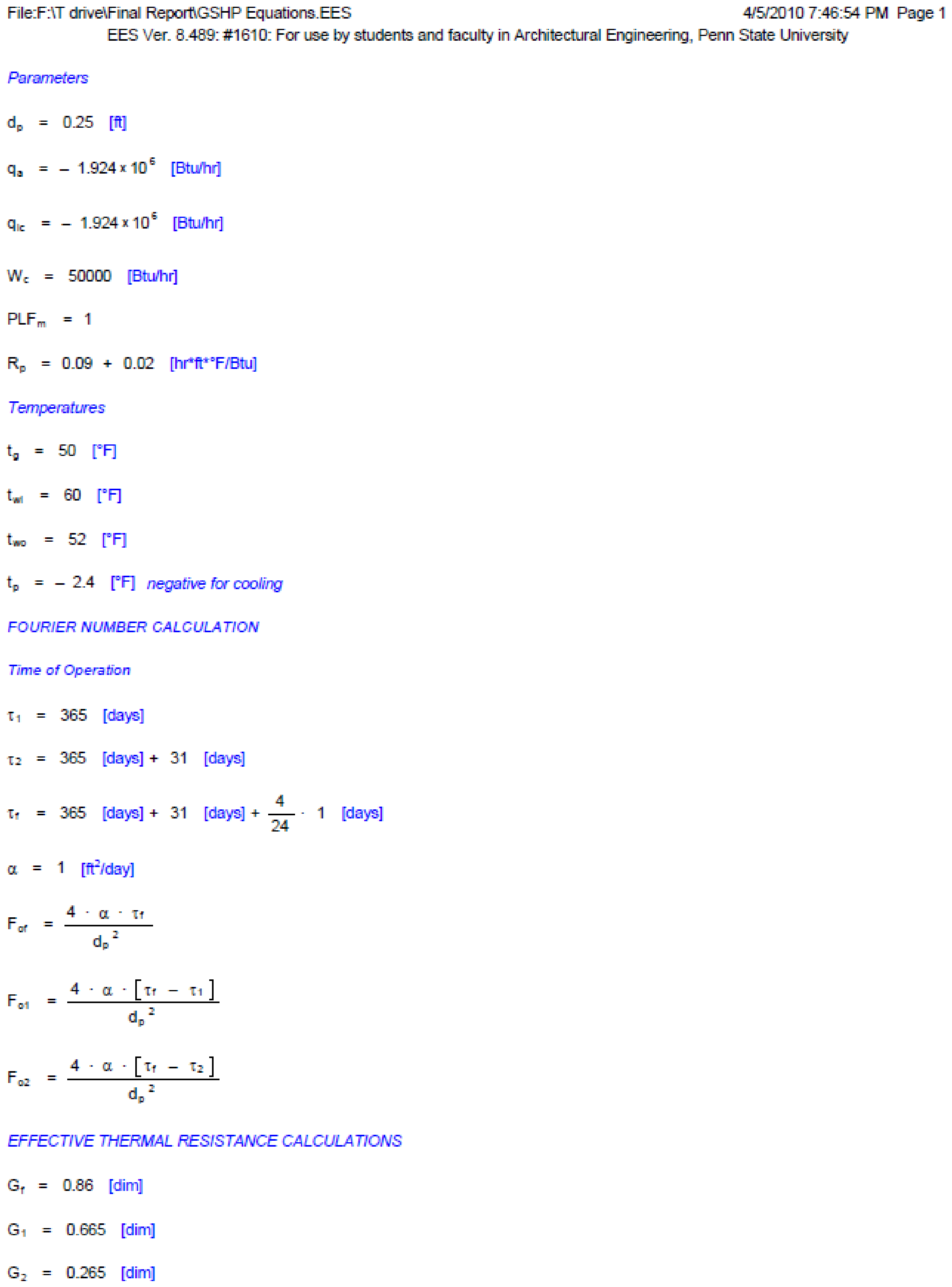
- **Bryan Rydingsward – BR+A Design Engineer**
- **Arne Abramson – MIT Program Manager**
- **James D. Freihaut – Thesis Adviser**
- **Thermoflo Equipment Co.**
- **My Fellow 5th Year Classmates**

13.0 List of Figures

APPENDIX A

Supplemental Tables

EES Ground Source Heat Pump Sizing Calculation



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EES Ver. 8.489: #1610: For use by students and faculty in Architectural Engineering, Penn State University

Parameters

$d_p = 0.25$ [ft]

$q_a = -1.924 \times 10^6$ [Btu/hr]

$q_{lc} = -1.924 \times 10^6$ [Btu/hr]

$W_c = 50000$ [Btu/hr]

$PLF_m = 1$

$R_p = 0.09 + 0.02$ [hr*ft*°F/Btu]

Temperatures

$t_g = 50$ [°F]

$t_{wi} = 60$ [°F]

$t_{wo} = 52$ [°F]

$t_p = -2.4$ [°F] *negative for cooling*

FOURIER NUMBER CALCULATION

Time of Operation

$\tau_1 = 365$ [days]

$\tau_2 = 365$ [days] $+ 31$ [days]

$\tau_f = 365$ [days] $+ 31$ [days] $+ \frac{4}{24} \cdot 1$ [days]

$\alpha = 1$ [ft²/day]

$$F_{of} = \frac{4 \cdot \alpha \cdot \tau_f}{d_p^2}$$

$$F_{o1} = \frac{4 \cdot \alpha \cdot [\tau_f - \tau_1]}{d_p^2}$$

$$F_{o2} = \frac{4 \cdot \alpha \cdot [\tau_f - \tau_2]}{d_p^2}$$

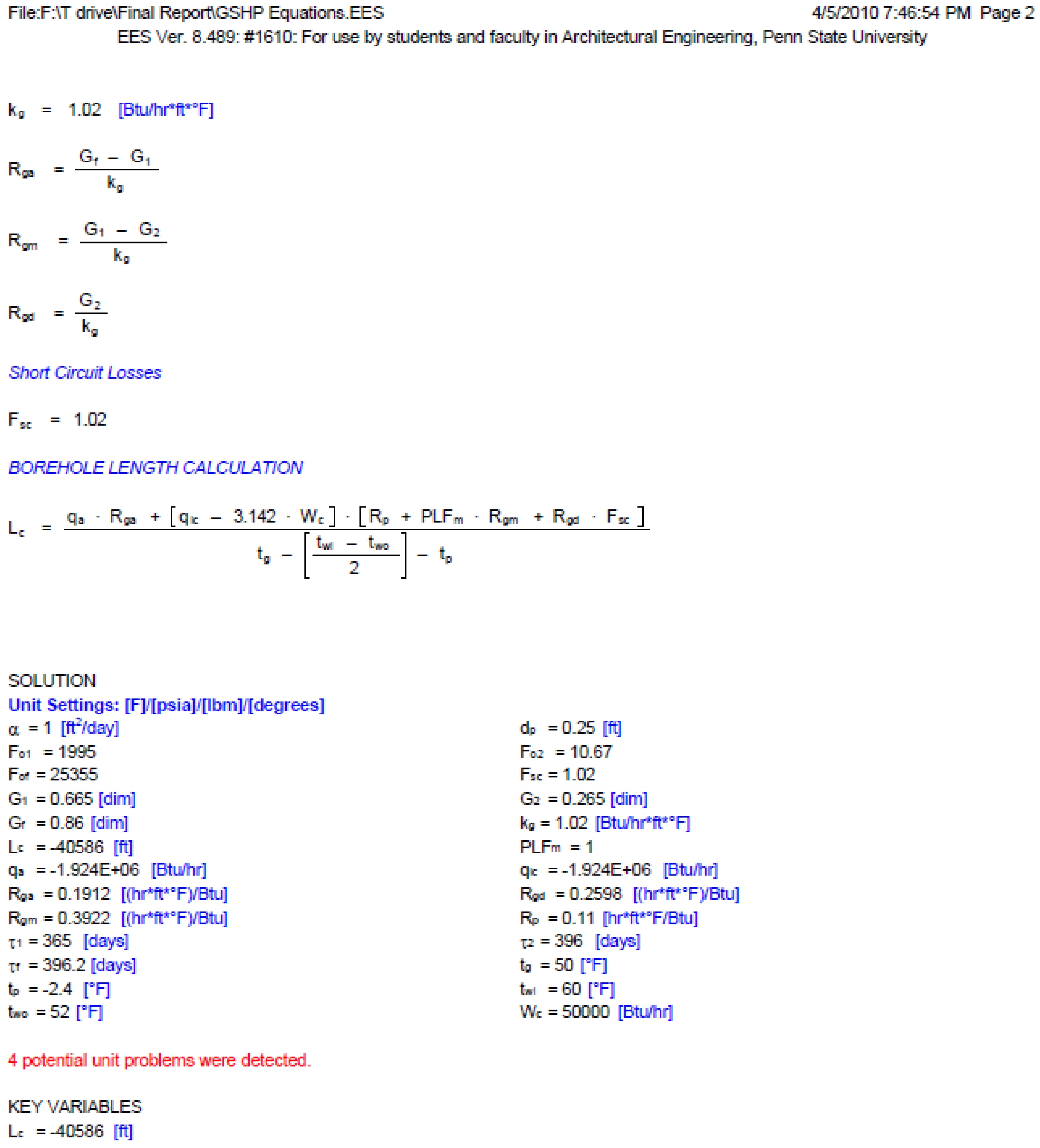
EFFECTIVE THERMAL RESISTANCE CALCULATIONS

$G_f = 0.86$ [dim]

$G_1 = 0.665$ [dim]

$G_2 = 0.265$ [dim]

EES Ground Source Heat Pump Sizing Calculation



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EES Ver. 8.489: #1610: For use by students and faculty in Architectural Engineering, Penn State University

$k_g = 1.02$ [Btu/hr*ft*°F]

$$R_{ga} = \frac{G_f - G_1}{k_g}$$

$$R_{gm} = \frac{G_1 - G_2}{k_g}$$

$$R_{gd} = \frac{G_2}{k_g}$$

Short Circuit Losses

$F_{sc} = 1.02$

BOREHOLE LENGTH CALCULATION

$$L_c = \frac{q_a \cdot R_{ga} + [q_{lc} - 3.142 \cdot W_c] \cdot [R_p + PLF_m \cdot R_{gm} + R_{gd} \cdot F_{sc}]}{t_g - \left[\frac{t_{wi} - t_{wo}}{2}\right] - t_p}$$

SOLUTION

Unit Settings: [F]/[psia]/[lbm]/[degrees]

$\alpha = 1$ [ft^2/day]

$F_{o1} = 1995$

$F_{of} = 25355$

$G_1 = 0.665$ [dim]

$G_f = 0.86$ [dim]

$L_c = -40586$ [ft]

$q_a = -1.924E+06$ [Btu/hr]

$R_{ga} = 0.1912$ [(hr*ft*°F)/Btu]

$R_{gm} = 0.3922$ [(hr*ft*°F)/Btu]

$\tau_1 = 365$ [days]

$\tau_f = 396.2$ [days]

$t_p = -2.4$ [°F]

$t_{wo} = 52$ [°F]

$d_p = 0.25$ [ft]

$F_{o2} = 10.67$

$F_{sc} = 1.02$

$G_2 = 0.265$ [dim]

$k_g = 1.02$ [Btu/hr*ft*°F]

$PLF_m = 1$

$q_{lc} = -1.924E+06$ [Btu/hr]

$R_{gd} = 0.2598$ [(hr*ft*°F)/Btu]

$R_p = 0.11$ [hr*ft*°F/Btu]

$\tau_2 = 396$ [days]

$t_g = 50$ [°F]

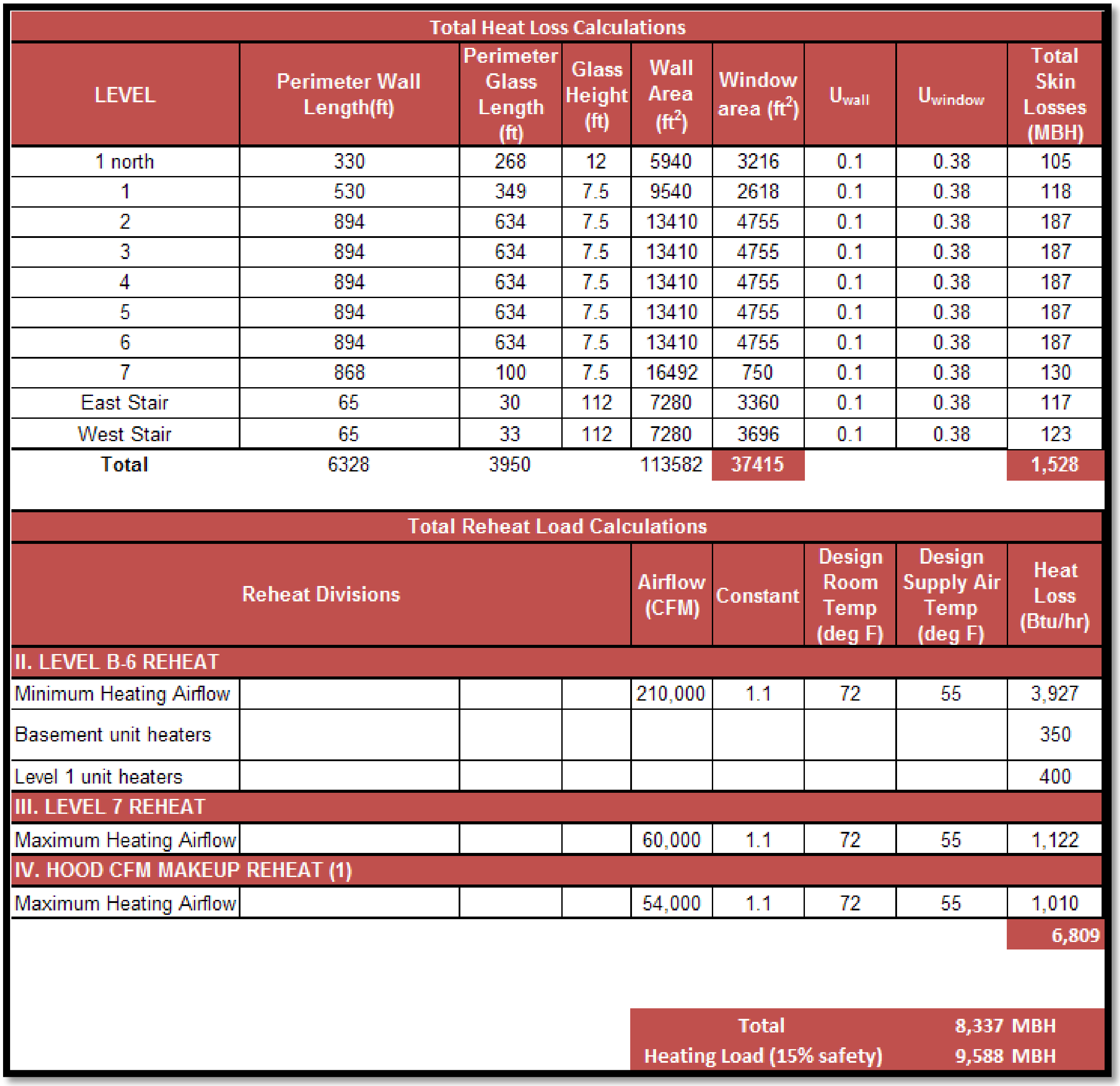
$t_{wi} = 60$ [°F]

$W_c = 50000$ [Btu/hr]

4 potential unit problems were detected.

KEY VARIABLES

$L_c = -40586$ [ft]

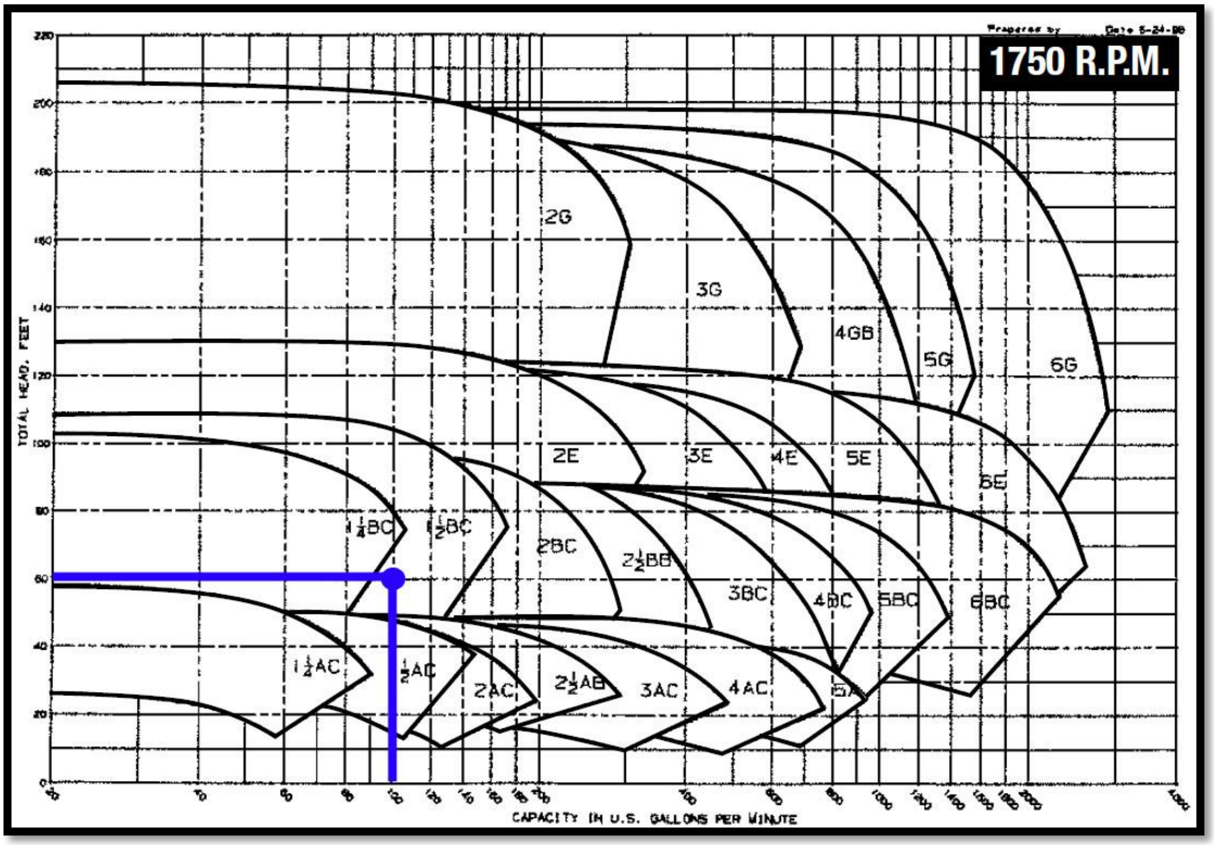


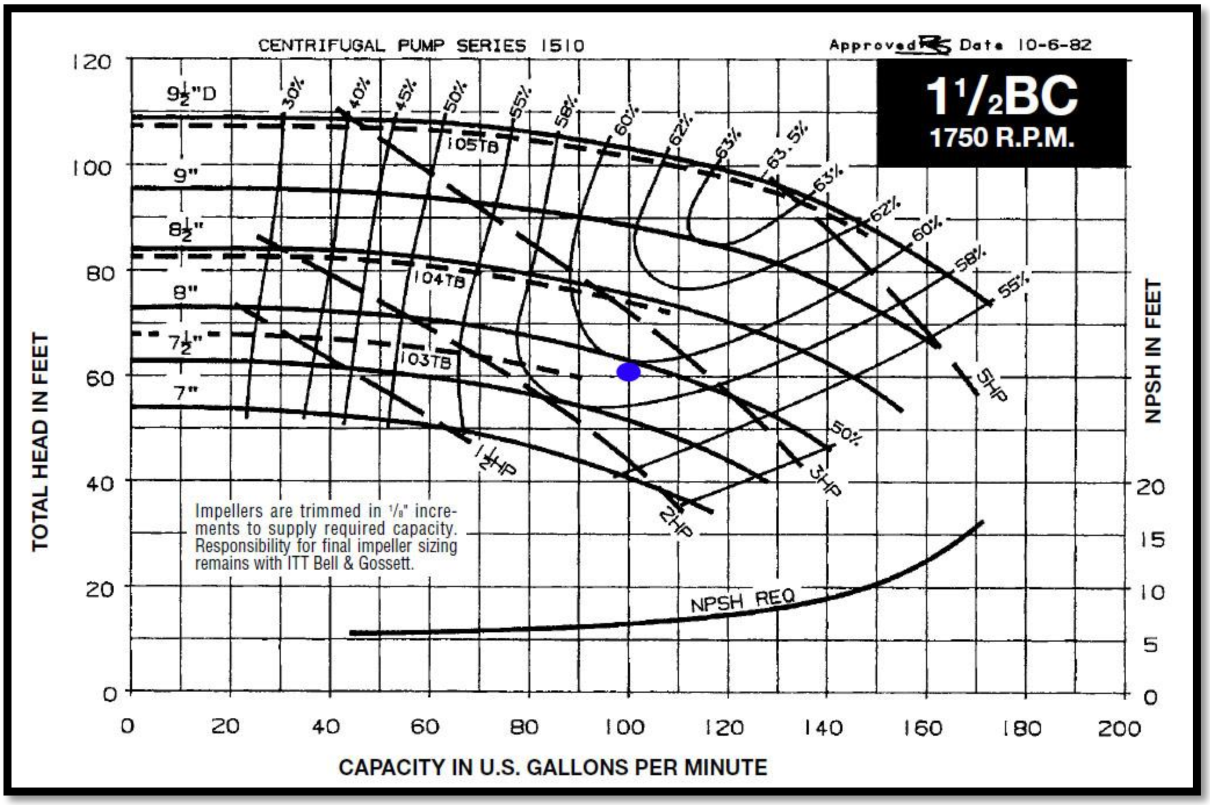
Total Heat Loss Calculations								
LEVEL	Perimeter Wall Length(ft)	Perimeter Glass Length (ft)	Glass Height (ft)	Wall Area (ft^2)	Window area (ft^2)	U_{wall}	U_{window}	Total Skin Losses (MBH)
1 north	330	268	12	5940	3216	0.1	0.38	105
1	530	349	7.5	9540	2618	0.1	0.38	118
2	894	634	7.5	13410	4755	0.1	0.38	187
3	894	634	7.5	13410	4755	0.1	0.38	187
4	894	634	7.5	13410	4755	0.1	0.38	187
5	894	634	7.5	13410	4755	0.1	0.38	187
6	894	634	7.5	13410	4755	0.1	0.38	187
7	868	100	7.5	16492	750	0.1	0.38	130
East Stair	65	30	112	7280	3360	0.1	0.38	117
West Stair	65	33	112	7280	3696	0.1	0.38	123
Total	6328	3950		113582	37415			1,528

Total Reheat Load Calculations								
Reheat Divisions				Airflow (CFM)	Constant	Design Room Temp (deg F)	Design Supply Air Temp (deg F)	Heat Loss (Btu/hr)
II. LEVEL B-6 REHEAT								
Minimum Heating Airflow				210,000	1.1	72	55	3,927
Basement unit heaters								350
Level 1 unit heaters								400
III. LEVEL 7 REHEAT								
Maximum Heating Airflow				60,000	1.1	72	55	1,122
IV. HOOD CFM MAKEUP REHEAT (1)								
Maximum Heating Airflow				54,000	1.1	72	55	1,010
								6,809

Total	8,337 MBH
Heating Load (15% safety)	9,588 MBH

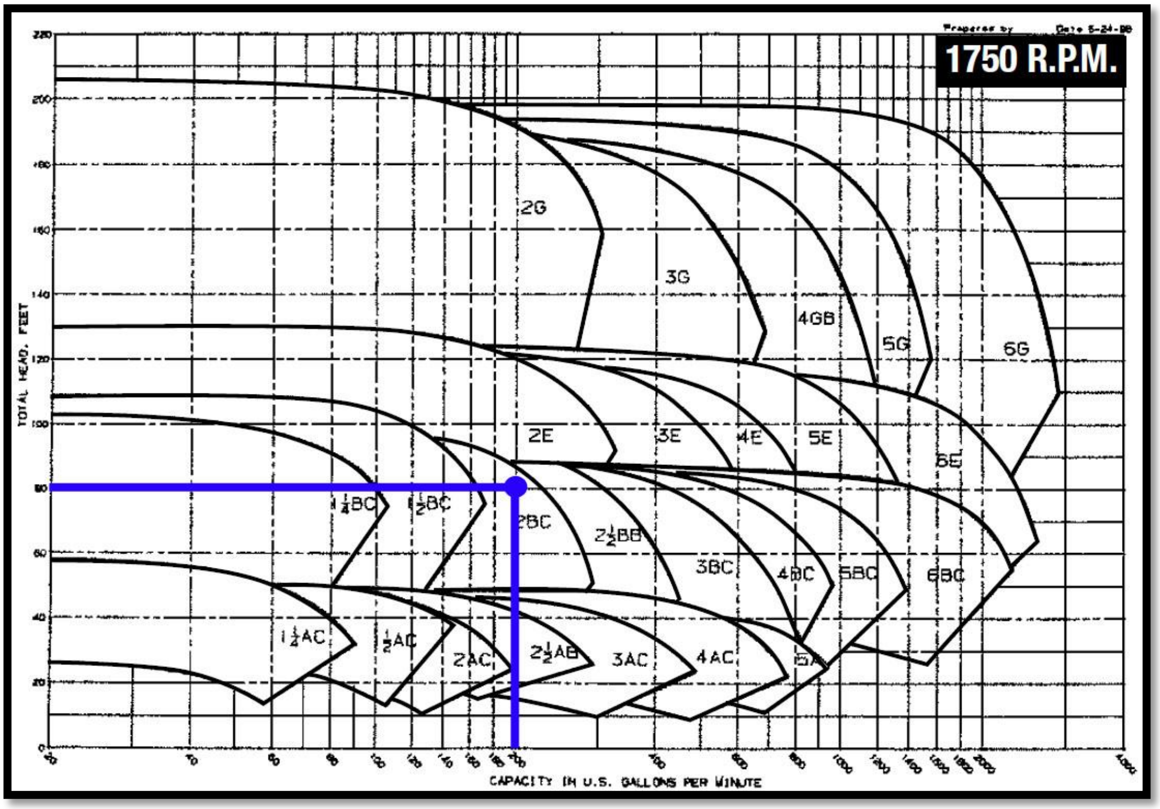
Glycol Run Around Heat Recovery Pump HRP-1

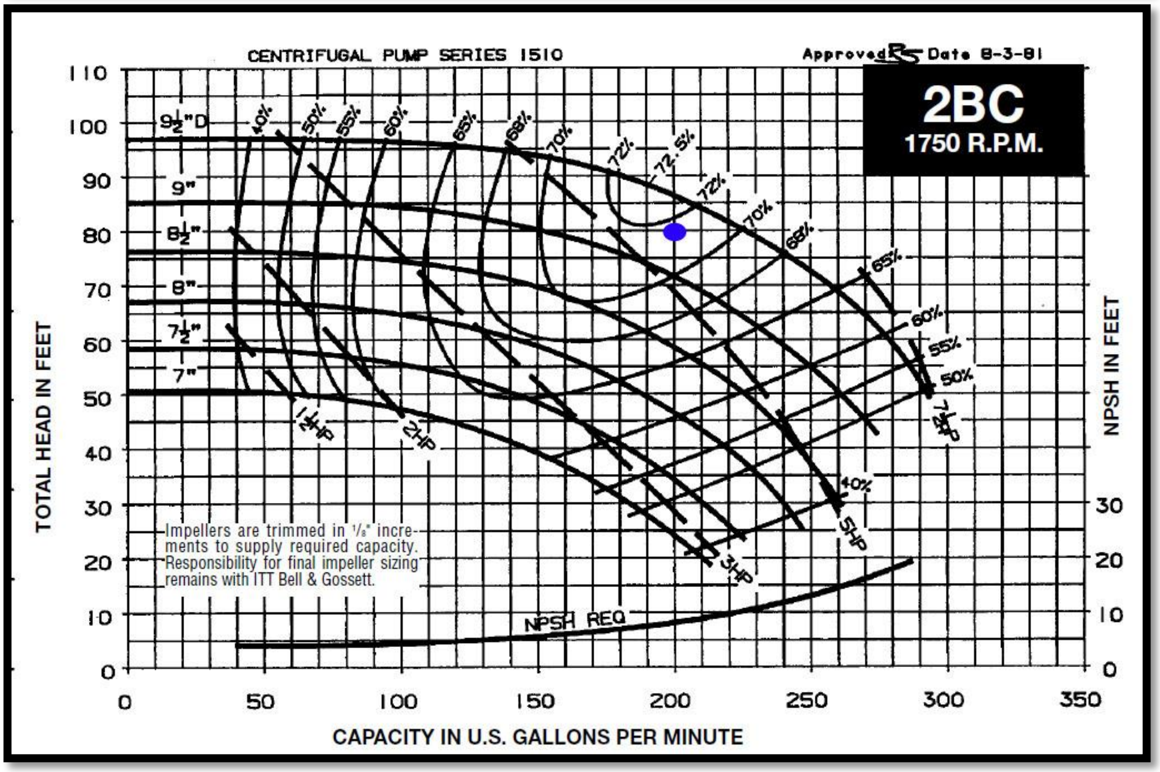




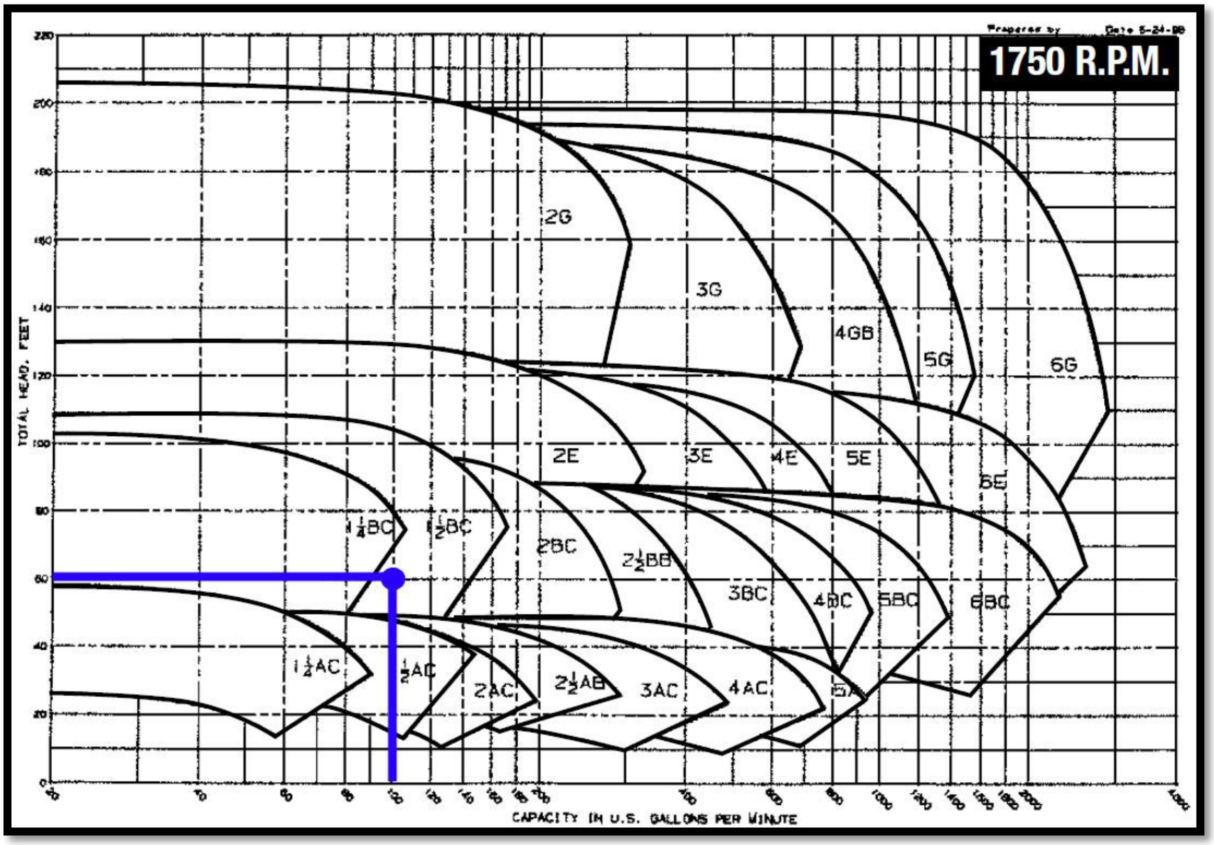
Impellers are trimmed in ⅛" increments to supply required capacity. Responsibility for final impeller sizing remains with ITT Bell & Gossett.

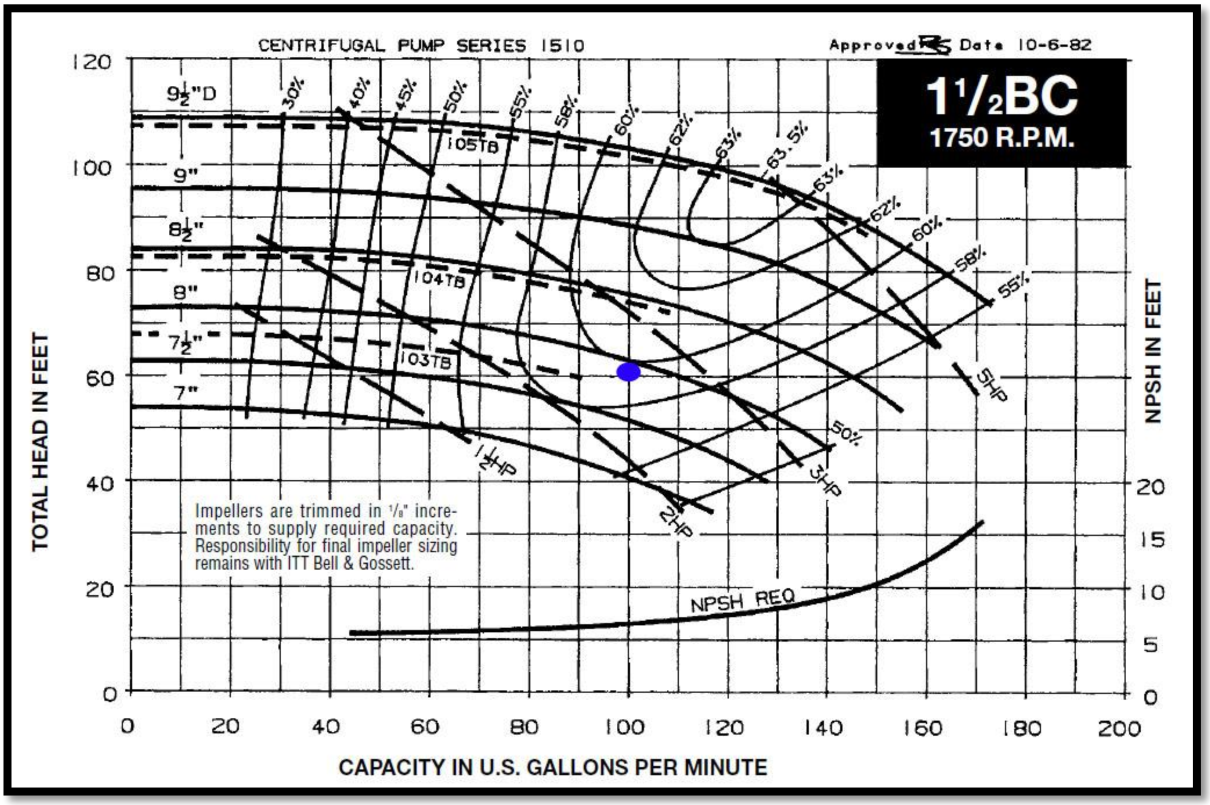
Geothermal Pump Selection GCHWP-1 & 2





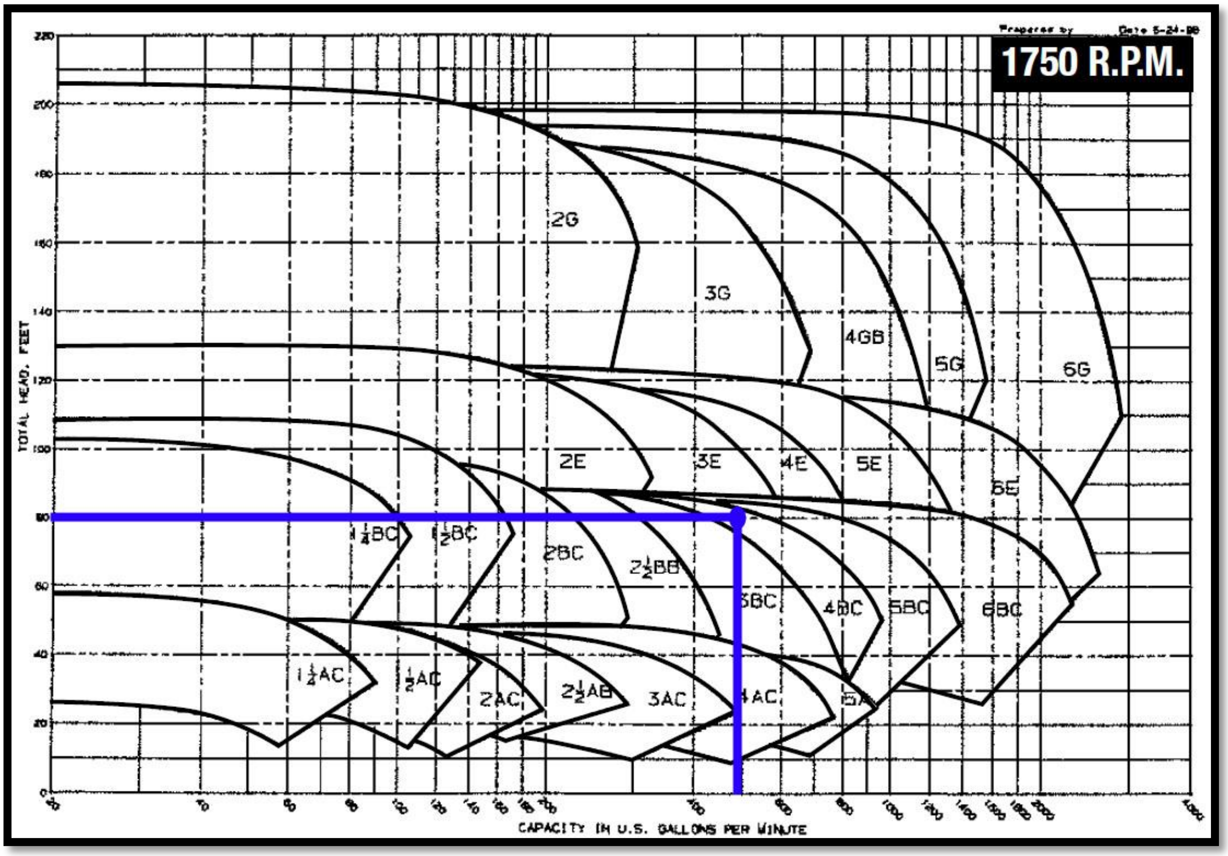
Geothermal Pump Selection GCHWP-3

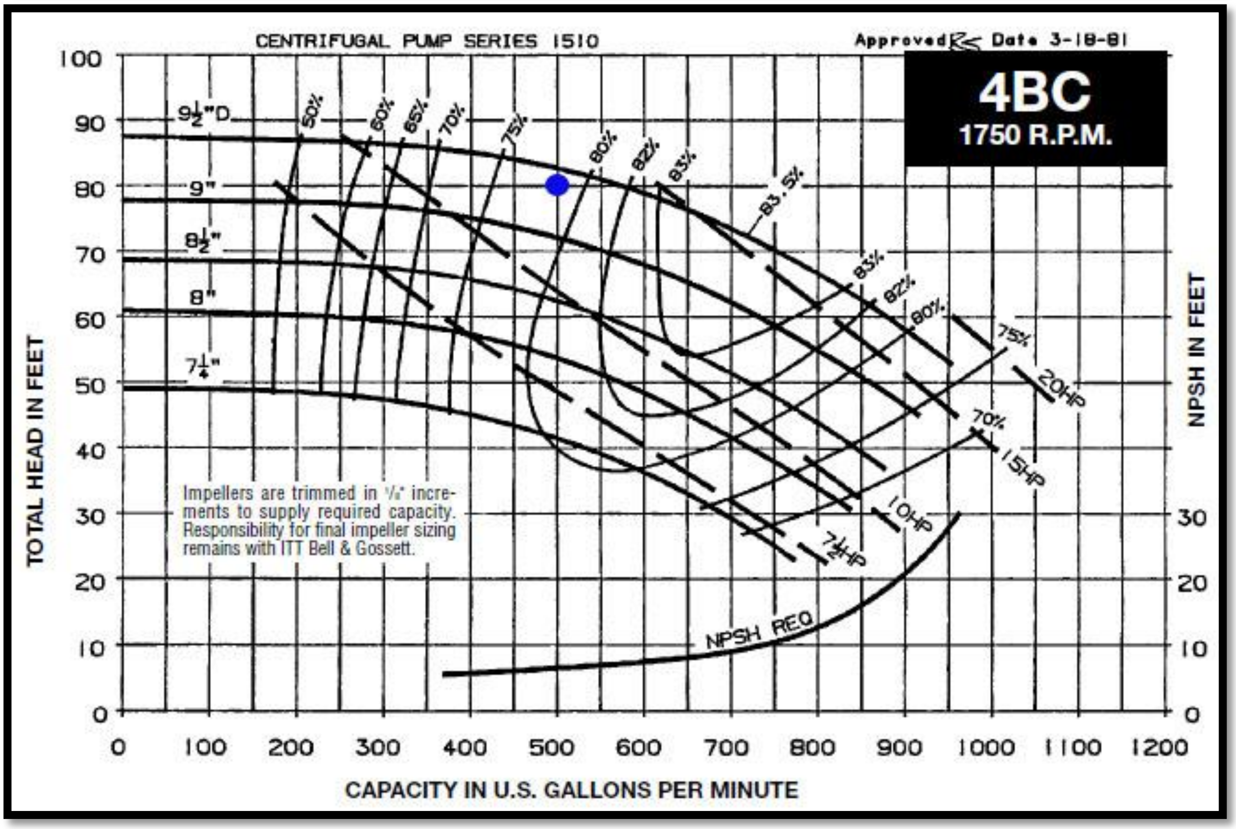


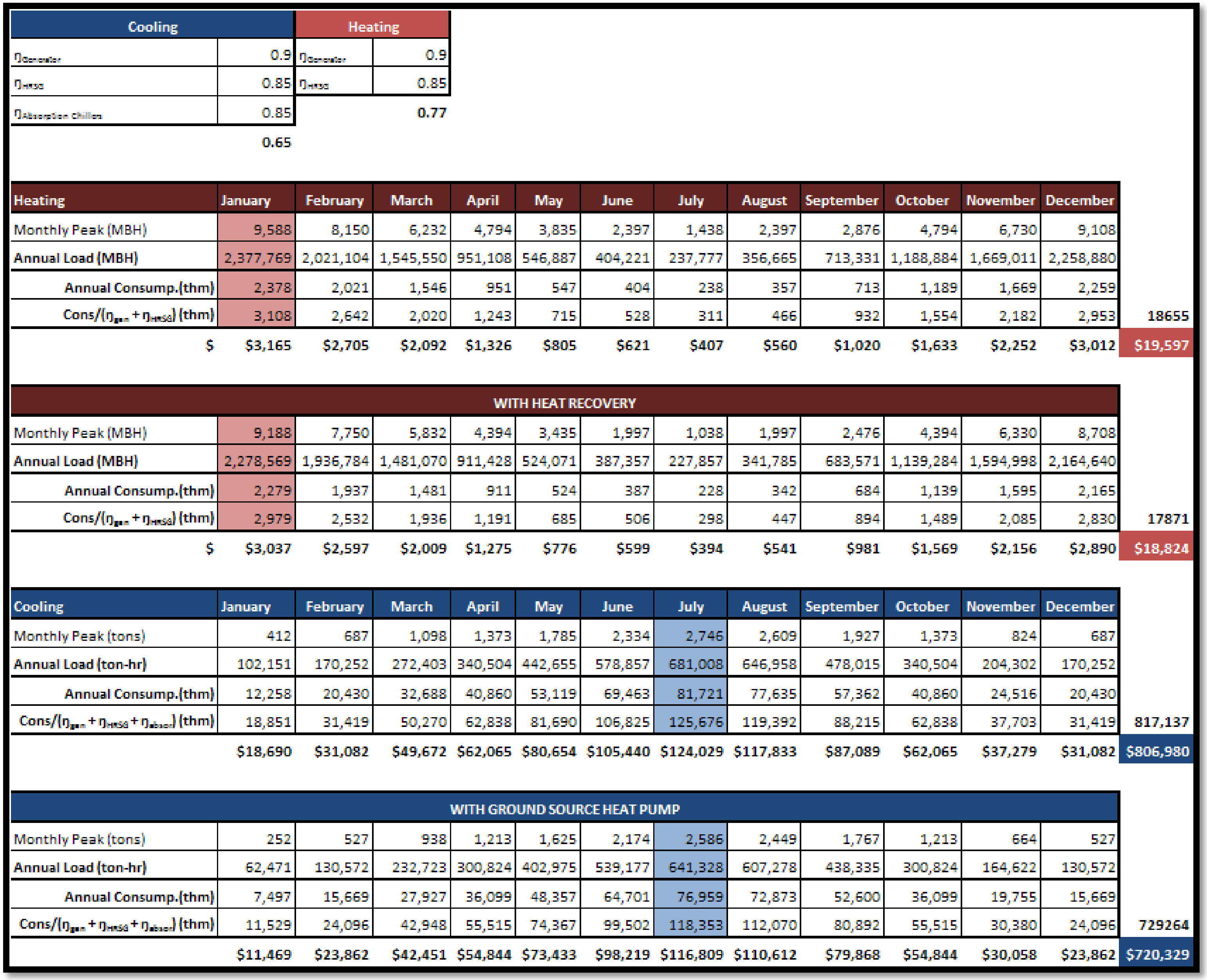


Impellers are trimmed in ⅛" increments to supply required capacity. Responsibility for final impeller sizing remains with ITT Bell & Gossett.

Chilled Water Pump Selection CHW-3







Cooling		Heating	
$\eta_{Generator}$	0.9	$\eta_{Generator}$	0.9
η_{HRSG}	0.85	η_{HRSG}	0.85
$\eta_{Absorption\ Chiller}$	0.85		0.77
	0.65		

Heating	January	February	March	April	May	June	July	August	September	October	November	December	
Monthly Peak (MBH)	9,588	8,150	6,232	4,794	3,835	2,397	1,438	2,397	2,876	4,794	6,730	9,108	
Annual Load (MBH)	2,377,769	2,021,104	1,545,550	951,108	546,887	404,221	237,777	356,665	713,331	1,188,884	1,669,011	2,258,880	
Annual Consump.(thm)	2,378	2,021	1,546	951	547	404	238	357	713	1,189	1,669	2,259	
Cons/(η_{gen} + η_{HRSG}) (thm)	3,108	2,642	2,020	1,243	715	528	311	466	932	1,554	2,182	2,953	18655
$	$3,165	$2,705	$2,092	$1,326	$805	$621	$407	$560	$1,020	$1,633	$2,252	$3,012	$19,597

WITH HEAT RECOVERY													
Monthly Peak (MBH)	9,188	7,750	5,832	4,394	3,435	1,997	1,038	1,997	2,476	4,394	6,330	8,708	
Annual Load (MBH)	2,278,569	1,936,784	1,481,070	911,428	524,071	387,357	227,857	341,785	683,571	1,139,284	1,594,998	2,164,640	
Annual Consump.(thm)	2,279	1,937	1,481	911	524	387	228	342	684	1,139	1,595	2,165	
Cons/(η_{gen} + η_{HRSG}) (thm)	2,979	2,532	1,936	1,191	685	506	298	447	894	1,489	2,085	2,830	17871
$	$3,037	$2,597	$2,009	$1,275	$776	$599	$394	$541	$981	$1,569	$2,156	$2,890	$18,824

Cooling	January	February	March	April	May	June	July	August	September	October	November	December	
Monthly Peak (tons)	412	687	1,098	1,373	1,785	2,334	2,746	2,609	1,927	1,373	824	687	
Annual Load (ton-hr)	102,151	170,252	272,403	340,504	442,655	578,857	681,008	646,958	478,015	340,504	204,302	170,252	
Annual Consump.(thm)	12,258	20,430	32,688	40,860	53,119	69,463	81,721	77,635	57,362	40,860	24,516	20,430	
Cons/(η_{gen} + η_{HRSG} + η_{absorb}) (thm)	18,851	31,419	50,270	62,838	81,690	106,825	125,676	119,392	88,215	62,838	37,703	31,419	817,137
	$18,690	$31,082	$49,672	$62,065	$80,654	$105,440	$124,029	$117,833	$87,089	$62,065	$37,279	$31,082	$806,980

WITH GROUND SOURCE HEAT PUMP													
Monthly Peak (tons)	252	527	938	1,213	1,625	2,174	2,586	2,449	1,767	1,213	664	527	
Annual Load (ton-hr)	62,471	130,572	232,723	300,824	402,975	539,177	641,328	607,278	438,335	300,824	164,622	130,572	
Annual Consump.(thm)	7,497	15,669	27,927	36,099	48,357	64,701	76,959	72,873	52,600	36,099	19,755	15,669	
Cons/(η_{gen} + η_{HRSG} + η_{absorb}) (thm)	11,529	24,096	42,948	55,515	74,367	99,502	118,353	112,070	80,892	55,515	30,380	24,096	729264
	$11,469	$23,862	$42,451	$54,844	$73,433	$98,219	$116,809	$110,612	$79,868	$54,844	$30,058	$23,862	$720,329