Longwood University's New Science Building

Farmville, VA

Alicia Carbin

AE Senior Thesis

Mechanical Option

2005

www.arche.psu.edu/thesis/2005/abc164

Longwood University's New Science Building

Farmville, VA

Project Overview:

- Size: 71,800 SF
- Number of Stories: 4 Project Cost: $17.6 million
- University Science Departments: Biology, Chemistry, Earth Science, and Physics
- Building Spaces: 28 faculty offices, 20 classrooms and labs, greenhouse, and additional faculty research spaces

Architectural

- Façade: Brick and Pre-Cast Concrete Veneer
- 15% Glazing
- Roof: Flat with Decorative Metal Enclosure
- "T-Shirt Shaped" Footprint in Plan
- Construction type: 2B 'Non-Combustible Protected'
- Brick U-Shaped walkway directs pedestrian traffic
- Laboratories with the greatest number of fume hoods are located on the upper floors

Mechanical

- Outdoor Design Conditions: 90F DB, 76F WB (Summer), 8F DB (Winter)
- Source of Heating: steam from Central Campus with two-stage pressure reducting station at building
- Source of Cooling: Two 250-ton Screw Compressor Chillers , CV Pumps, CV Cooling Towers
- DDC Energy Management and temperature control system with pneumatic actuators
- Ventilation System: VAV, Air Distributing Ductwork
- Laboratories: 100% Outside Air, Fume Hood Exhaust System, interlocked with VAV Supply system

Structural

- Ground Floor System: Concrete slab-on-Grade
- Floors: Composite System of concrete slab on composite steel deck, supported on composite wide-flange steel beams and girders
- Framing System: Braced Lateral Steel
- Roof: Concrete on Composite Metal Roof Deck

Construction Management

- Construction Cost: $15.1 Million
- Construction Schedule: July '03 – Summer '05
- Delivery method: Design-Bid-Build

Lighting/Electrical

- Main Switchboard: 3000 Amps
- 480Y/277 volts, 3 phase, 4 wire grounded
- Emergency Power: Diesel engine generator
- Fluorescent fixtures with electronic ballast
- T-8 Lamps, low-mercury type with CRI=75
- Occupancy sensors in Classrooms and Labs
- Exterior Lighting for Egress and Security

Penn State Architectural Engineering Senior Thesis

Alicia Carbin, Mechanical Option

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Section 1.0

Project Background and Description

1.0 PROJECT BACKGROUND AND DESCRIPTION

The proposed redesigns of this report in no way imply that the existing mechanical system, or any other system in the building include errors. The analyses done in this report are only for educational purposes only.

1.1 EXECUTIVE SUMMARY

The new science building at Longwood University in Farmville, VA houses the University's Chemistry, Physics, Biology, and Earth Science Departments. It includes 28 faculty and graduate offices, 20 classrooms and labs, as well as additional faculty research space. Many of the laboratory spaces require the use of 100% outdoor air air-handling units. These units create large heating and cooling loads on the system. A run-around heat recovery system utilizing waste exhaust air was analyzed, and it was found that it saves 855,776,768 Btu, which is 15.4% of the total energy used. Hourly energy consumption information was used to produce annual energy cost data. The results of the energy analysis show $3,748 savings per year. The payback period for the heat recovery system was found to be about 4 years.

In addition to the heat recovery system, the use of distributed pumping to replace the existing secondary pumps in the building was analyzed. Overall, the distributed pumps had a 26% increase in first cost over the secondary pumps, however they used 60.8% less horsepower. The distributed pumps save $3,792 per year over the secondary pumps, and the payback period was found to be 10 months.

The impact of using additional pumps on the roof was analyzed for its electrical impact on the existing building service. It was found that the 1- ½ hp pump used in the run-around heat recovery system would fit on panelboard H4, and the 4 small horsepower distributed pumps would fit on panelboard H4A in the 4th floor

mechanical room. An acoustical study was also done for a typical classroom in the science building. It was found that the valves in the VAV boxes have noise levels which are within the RC-ratings recommended for classrooms, however they create a "hissy" sound. The recommended solution to this problem is to decrease the flow velocity through the valves.

1.2 INTRODUCTION

I chose Longwood University's New Science building as my senior thesis subject because I believe the building is a great fit for the requirements that must be completed. The laboratories in the science buildings will work great for a mechanical redesign. The building contains a green house on the roof, which is also a unique feature. The following sketches are of the north and south elevations of the science building done by the architect during the preliminary design.

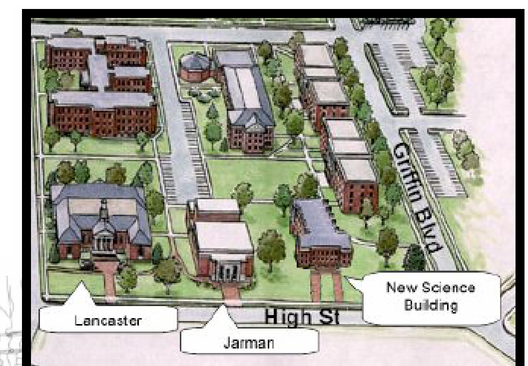
1.3 BACKGROUND

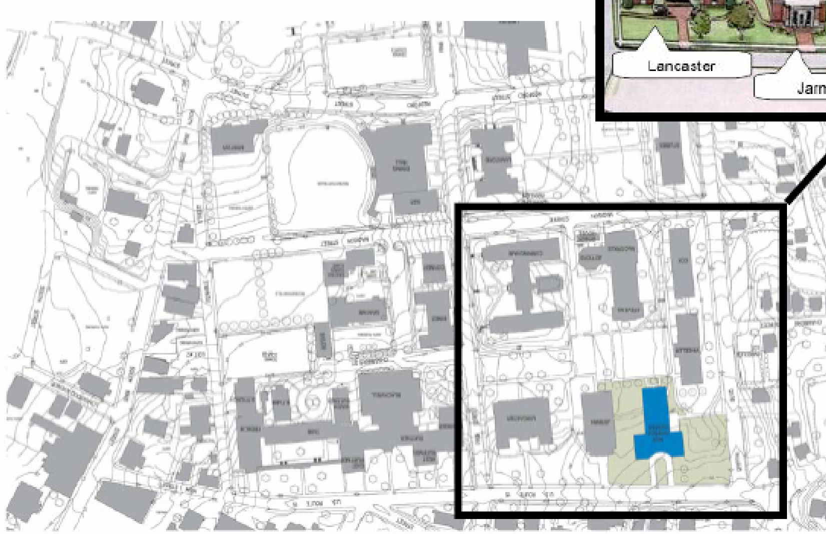
The following sections provide the foundation on which the thesis analyses were built and completed. All significant information relating to the systems and designs analyzed in the building are provided.

1.3.1 LOCATION AND SITE INFORMATION

Longwood University is located in Farmville, VA. The New Science Building is a 71,800 square foot, 4 story building. The total cost for the building, including construction was $17.6 million. The science building houses the University's Chemistry, Physics, Biology, and Earth Science Departments. It includes 28 faculty and graduate offices, 20 classrooms and labs, as well as additional faculty research space. In addition, there is a greenhouse located at the south end of the forth floor.

The site is located to the west of Jarman Hall, situated on the southwest corner of High Street and Griffin Street. There is a 15 ft elevation difference across the site, sloping down to the south.





(Images courtesty of Clark Nexsen and www.longwood.edu)

1.3.2 PRIMARY PROJECT TEAM

BUILDING OWNER:

Longwood University: Farmville, VA

ARCHITECTS & ENGINEERS:

Clark Nexsen: Norfolk, VA

LAB CONSULTANT:

Research Facilities Design: San Diego, CA

GENERAL CONTRACTOR:

Suitt Construction: Greenville, SC

1.3.3 ARCHITECTURE

ARCHITECTURE DESIGN AND FUNCTIONAL COMPONENTS:

The New Science Building has a "T-shirt" shaped footprint (shown in Figure 1.1), with a central loaded corridor. The two main elements of the building are the modular spaced laboratories and the offices. The main entrance to the building will be off of High Street at the second floor level. The secondary entrance is on the first floor on the south side of the building. The laboratories with the greatest number of fume hoods are located on the upper floors to better facilitate the plenum space and vertical chases for mechanical equipment and ductwork.

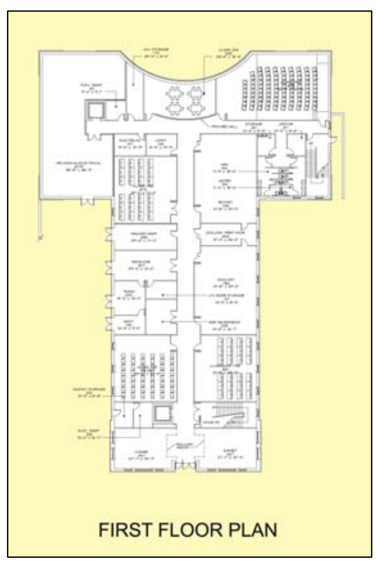


Figure 1.1

MAJOR NATIONAL MODEL CODES:

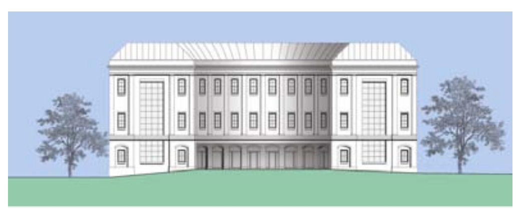
- Virginia Uniform Statewide Building Code
- BOCA 1996 with 2000 amendments

ZONING REQUIREMENTS:

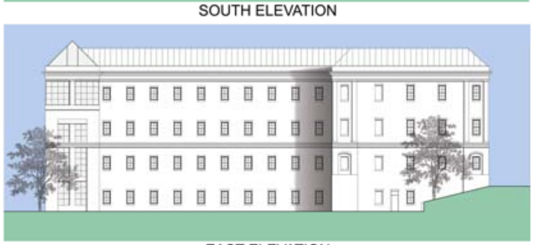
Longwood University wanted the building to have a height to match the height of the surrounding buildings. The following pictures show some of the buildings surrounding the science building site.

SCIENCE BUILDING ELEVATIONS:

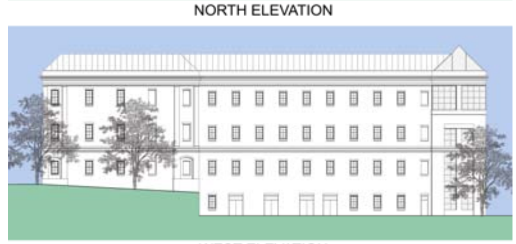
SOUTH ELEVATION



NORTH ELEVATION



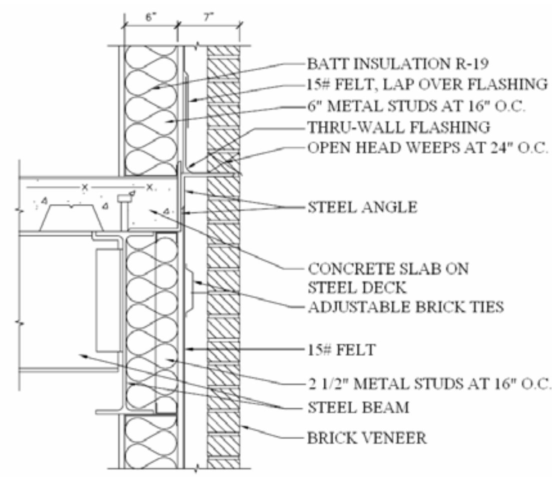
EAST ELEVATION



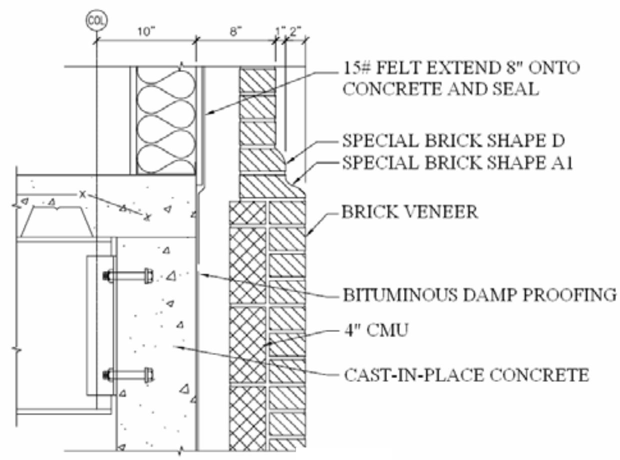
WEST ELEVATION

BUILDING ENVELOPE:

The building has a brick façade with pre-cast veneer. The building is comprised of 15% glazing. The structure of the building is made of a steel framing system. The roof will be flat to accommodate mechanical units and vents which will be screened by a decorative metal enclosure. The figures below show typical exterior wall sections of the science building: (a) Brick wall section, (b) Concrete, CMU, and Brick wall section, (c) Pre-cast Pilaster wall section, (d) Metal closure section.



(a)



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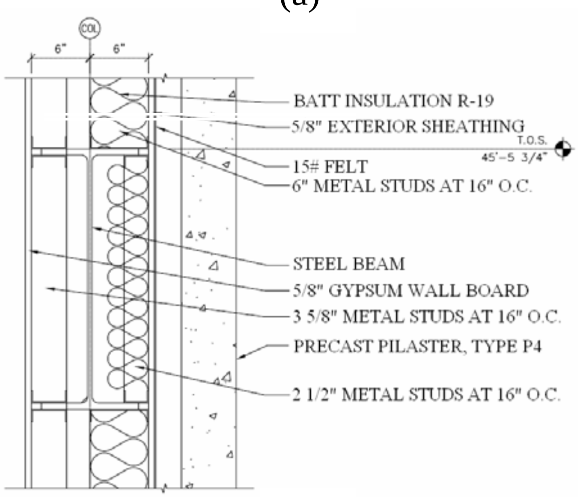

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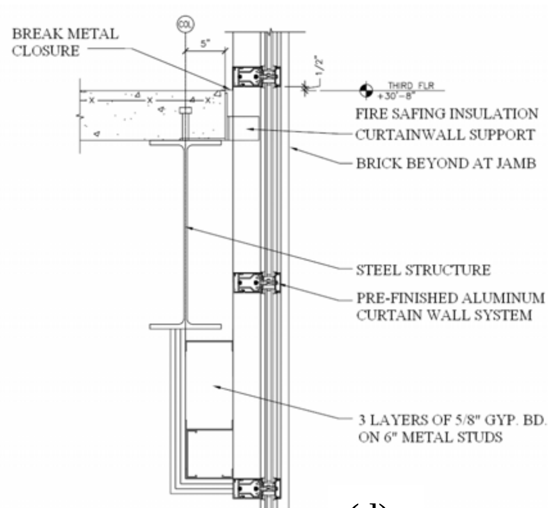


(d)

1.3.4 MECHANICAL SYSTEM

COOLING SYSTEM:

The source of cooling is the two 250 ton screw compressor chillers located in the first floor mechanical room. Each chiller (Table 1.1) is provided with a constant volume, base mounted, centrifugal pump. The condenser water for the chillers are provided by two 250 ton induced draft cooling tower with a constant volume, base mounted, centrifugal condenser water pump. The chilled water distribution is a primary/secondary system with two-way modulating control valves on the terminal cooling equipment in the secondary chilled water loop.

Chiller		CH-1	CH-2
Nominal Capacity (Tons)		250	250
Condenser	Entering Water Temp (F)	95	95
	Leaving Water Temp (F)	85	85
	GPM	750	750
	Max PD (ft wg)	20	20
	Fouling Factor (hr)(sf.)(deg F)/BTU	0.0005	0.0005
Evaporator	Entering Water Temp (F)	55	55
	Leaving Water Temp (F)	45	45
	GPM	600	600
	Max PD (ft wg)	20	20
	Fouling Factor (hr)(sf.)(deg F)/BTU	0.00025	0.00025
Compress. Motor	Max KW Input	153	153
	Volt/Phase/Hertz	480/3/60	480/3/60
	Unit MCA	270	270
	Max KW/ton	0.6	0.6
Starter Type		Star Delta	Star Delta
Refrigerant Type		R-134A	R-134A

Table 1.1

HEATING SYSTEM

The source of heating is steam from the central campus steam distribution system. There is a two-stage steam pressure reducing station at the building in combination with a steam to hot water converter. The building heating system is located in the first floor mechanical equipment room, which includes two hot water heating, base mounted centrifugal circulating pumps with variable frequency drives.

HVAC ENERGY MANAGEMENT AND CONTROL SYSTEM:

A DDC energy management and temperature control system with pneumatic actuators for large dampers and valves is provided for the HVAC system for the Science building.

VENTILATION SYSTEM:

The ventilation system of the science building uses conditioned outside air provided by multiple variable air volume air handling units and distributed to the conditioned spaces throughout the building by air distribution ductwork. All of the air-handling units are located on the roof. Classrooms and office spaces in the science building will be heated and cooled by variable air volume air handling units with fan powered variable air volume terminal units. The air supplied to these spaces will be re-circulated. Laboratory spaces in the science building will be heated and cooled by variable air volume air handling units with non-fan powered variable air volume terminal units working in conjunction with the general and hood exhaust in the labs. The supply air to the laboratory spaces will be 100% outside air. These spaces are not permitted to have any air re-circulated. The green house is served by three evaporative cooling units, which all supply 100% outside air.

LABORATORY EXHAUST SYSTEMS:

The laboratory fume hood exhaust system collects the exhaust air from all fume hoods, downdraft hoods, snorkels and general exhaust registers located in laboratory type spaces. The fume hood exhaust system is a variable volume control system interlocked with the variable air volume supply system. General exhaust registers and variable air volume exhaust valves are provided as required in the laboratories to provide proper laboratory pressurization. Figure 1.2 shows the 60" exhaust duct leading to the four fume hood exhaust fans located on the roof.

Figure 1.2

FIRE PROTECTION SYSTEM:

The building has automatic fire sprinklers throughout the facility. The sprinkler system is a wet pipe system and complies with NFPA 13. Pendant and recessed sprinkler heads are used with suspended ceilings and upright sprinkler heads are used in spaces with exposed ceilings. The water supply for the fire pump is from city water through approved back-flow prevention. A Class I standpipe system is provided in the stairwells. The fire detection and alarm system complies with NFPA 72. Automatic detection is accomplished by smoke detectors, duct smoke detectors and the sprinkler system. Manual pull stations are provided as well, at exits for manual activation of the fire alarm system.

1.3.5 STRUCTURAL SYSTEM

FLOOR SYSTEM:

The ground floor consists of a concrete slab-on-grade placed over a vapor retarder over a 6" thick layer of porous fill. The structural floor system of the classroom building will be a composite system with a cast-in-place concrete slab placed on composite steel deck supported on composite wide-flange steel beams and girders. The typical bay size for this structure is 28'-9" x 31'-6". The composite beams are spaced at 10'-6" on center and span 28'-9" between girders. The composite slab is comprised of 3" deep steel deck with a 3.25" thick lightweight concrete topping slab (total slab thickness = 6.25.) The minimum 28-day compressive strength, f'_c, for the slabs on grade and the composite concrete slabs was 3000 psi.

ROOF SYSTEM:

The roof system is similar construction to the elevated floors with a composite, cast-in-place concrete slab supported by composite wide-flange steel beams. The roof beams are sloped at ¼" per foot in the east-west direction to facilitate positive drainage of rainwater. Steel frames, supported by the roof framing, are provided at each mechanical unit. Separate structural steel framing is also be provided along the perimeter of the roof to support the screen wall.

FRAMING SYSTEM:

The gravity framing system is a structural steel framing system consisting of wide-flange beams and columns. Concentric structural steel braced frames will be used to resist the lateral loads imposed on the structure. The rigid concrete floor and roof diaphragms distribute the lateral wind and seismic loads to the braced frames based upon the relative rigidity of these elements.

1.3.6 LIGHTING/ELECTRICAL SYSTEM

LIGHTING:

Lighting fixtures are high quality fixtures selected for efficiency, durability, and quality of lighting produced. Fluorescent fixtures are provided with electronic ballast and linear lamps will be T-8, low-mercury type with CRI of 75. Occupancy sensors are provided in areas that are often unoccupied to help conserve energy use. Local and master switching of light fixtures are located for areas where occupancy sensors are not used. Day lighting is used to the greatest extent where it was economically feasible. In addition, emergency lighting is served from the facility emergency power system. The exterior is lit for means of egress and security as well.

ELECTRICAL SYSTEM:

The electrical service for the facility will be provided at 480Y/277 Volts, 3 phase, 4-wire grounded from an exterior pad mounted transformer. This transformer and the connection to the primary distribution system is provided by Virginia Power. The Main Service Switchboard is rated at 3000 Amps. It will utilize stationary individually mounted feeder circuit breakers, and will feed distribution panels located in electrical rooms located on each floor. The main emergency panel board is fed by an automatic transfer switch. Panel boards will be located in corridors as necessary to support heavy laboratory electrical loads. Emergency power will be provided by a diesel engine driven generator, approximately 400 KW that provides service for emergency lighting, one of the two elevators, select laboratories, 100% of the laboratory hood exhaust fans and any other loads appropriate for emergency power such as the heating system, Fire Detection and Alarm system.

COMMUNICATIONS:

An empty conduit / cable tray / raceway system is provided to facilitate installation of complete telephone and data systems which will be provided by the owner. A central telephone and communication room (used for data acquisition and security) is provided on the first floor. Combination voice/data equipment rooms are provided on each floor.

1.3.7 CONSTRUCTION MANAGEMENT

The project delivery method for the science building was design-bid-build. The total project cost for the science building as $17.6 million, and of that, $15.1 was construction cost. The groundbreaking for the science building was on May 9, 2003. Actual construction of the science building began in July 2003. The pictures below were taken in the early stages of construction.

The building is scheduled to be finished in summer 2005. The pictures below show how far the construction has come. The picture on the left that shows the right side of the building was recently taken from a live web cam at the science building. The picture on the right shows the front of the building as it was on March 24, 2005.

1.4 MECHANICAL REDESIGN CONSIDERATIONS AND DISCUSSION

The following sub-sections of the report include the problems with the original mechanical design, as well as the proposed solutions to the problems. The two mechanical depth designs which are proposed are analyzed in sections 2.0 and 3.0.

1.4.1 PROBLEM STATEMENT

WASTED ENERGY:

Laboratory facilities are four to five times more energy intensive than a typical commercial building. Almost half of Longwood University's 71,800 ft^2 science building is lab space that includes many fume hoods. Fume hoods require a vast amount of energy, since they need to exhaust large quantities of contaminated air. In fact, just a conventional fume hood alone can consume three times more energy than an average American home.

The VAV system of the science building involves reheat terminal boxes that deliver the air to the lab space, which is controlled to match the variable exhaust quantity of the fume hood system. Some of the reasons that a standard VAV lab system wastes energy include:

- The supply air quantity needed to match the exhaust air also exactly meets the quantity required to meet the thermal load, and in reality, this does not occur most of the time.
- A large amount of refrigeration energy is expended to sub-cool outside air in the summer (to properly dehumidify the air), only to have heat returned to it before being supplied to the space.

- High energy outside air is used to generate more supply air to the space to match the exhaust air, which causes this already treated, unpolluted air to be directly exhausted from the space to the outside.
- Heating 100% outside air could take five to 10 times the heat input energy as compared to a re-circulated air system that provides the same heating capacity.

OVERSIZED PUMPS

The chilled water of the science building is forced to the cooling coils in the air handling units by a primary/secondary pumping system. The pump is sized to overcome all frictional losses, including the piping, coils, associated control valves, balancing valves, static losses, etc.

In Wayne Kisner's article in HPAC magazine, titled "The Demise of the Primary/Secondary Pumping Paradigm for Chilled Water Plant Design," he states that there are three main problems associated with Primary/Secondary design. The first problem is that the primary/secondary control scheme is "blinded" by low ΔT central plant syndrome. Furthermore, a primary/secondary control scheme that depends on system flow to gauge system load is virtually blind to load variation. The second problem is that the primary loop is constant flow. According to Kisner, most modern chiller controls no longer require constant flow to keep the chillers out of trouble. A variable-flow chilled water pumping scheme can respond to low system ΔT if the pumps are selected for excess capacity. The final problem presented with secondary pumping is that it is not the most efficient pumping distribution scheme. In a primary/secondary system, only the small fraction of the total chilled water flow going to the most distant load is produced without wasted energy. For this reason, a better way to pump distant loads is via distributed pumping, where the secondary pumps are replaced with distributed load pumps.

FIRST COST/OPERATING COST:

The design of the system was greatly impacted by first cost. It was determined from the life cycle cost analysis done by the engineers at Clark Nexsen, that the two 250-ton screw compressor chillers would be the most economical system. Heat Recovery was ruled out due to the fact that the cost of heat in the winter was so low because the university burns sawdust at the boiler plant. The results of the life cycle analysis showed that incorporating heat recovery devices into the system could nearly double the equipment cost. The energy cost savings were not enough to offset these increased equipment costs of heat recovery, which resulted in a failure to justify the heat recovery. First cost is a problem that a lot of designers face when designing new mechanical systems. Even though other systems may be able operate more efficiently, or use waste heat effectively, their equipment tends to cost more up front.

1.4.2 HEAT RECOVERY PROPOSAL

To solve the problem of waste energy, I propose to explore ways in which the exhaust heat could be recovered, to heat the supply air. There are many choices of heat recovery methods. The type of heat recovery which will be considered in my senior thesis is uses a run-around loop.

Heat recovery systems operate more efficiently with a variable-air-volume system than with a constant-volume system, because VAV systems typically operate at face velocities lower than those of design conditions. Since the science building has a VAV system, it makes sense to use some type of energy recovery. Heat recovery is a practical method to consider when it is not possible to clean the air and return it to the conditioned space. Certain systems of heat recovery are specifically reasonable for use in laboratory facilities, where cross contamination is not

permissible. Heat conservation and recovery can result in significant energy savings whenever there are large volumes of air being exhausted, such as in laboratories.

For energy recovery to be economically feasible, there are number of considerations that should be explored. First of all, the cost of energy should be considered. If the cost of energy is high, heat recovery is favorable, because it will save on energy. The grade of waste energy should also be considered. High-temperature waste energy is more economical to recover than low-temperature energy. The larger the temperature difference between the waste energy source (exhaust air) and the supply air stream, the more economical. The use of heat recovery may reduce the size of major equipment of the primary heating and cooling system, so this should be considered as well. The complete heat recovery analysis is covered in Section 2.0.

1.4.3 DISTRIBUTED PUMPING PROPOSAL

The use of smaller variable-speed distributed pumps throughout the building to replace the secondary chilled water pumps will be explored. Locating these distributed pumps at the air handling units should decrease the amount of power needed to run these pumps, and solve the problem of oversized pumps. In addition, it is predicted that the initial cost of four smaller pumps will be less expensive than the two secondary pumps. The complete distributed pumping analysis is covered in Section 3.0.

1.5 BREADTH REDESIGN DISCUSSION

Along with the two depth analyses, two breadth analyses were also completed. The mechanical changes stemming from the addition of the heat recovery system as well as distributed pumping create the unique opportunity to evaluate how mechanical system changes affect other building systems.

1.5.1 ELECTRICAL BREADTH PROPOSAL

The additions of the heat recovery system as well as distributed pumping introduce more electrical components to the existing system. Both of the mechanical depths introduce pump motors to the roof of the building. Feeder sizing for this new equipment as well as an evaluation of the switchgear serving this new mechanical equipment will be performed. See Section 4.0 for the complete electrical analysis.

1.5.2 ACOUSTICAL BREADTH PROPOSAL

A common complaint of mechanical systems is that they are too noisy. I propose to look into the common problem of VAV terminal unit noise. There are 42 fan powered variable air volume terminal units which condition the non-laboratory spaces. In addition, there are 41 variable air volume supply terminal units condition the laboratory and laboratory support spaces. There are also 77 exhaust terminal units in the science building. The acoustical study of the thesis will consist of exploring the acoustical properties of these VAV boxes. For the thesis, I propose to explore ways in which the noise levels of these units can be decreased. See Section 5.0 for the complete acoustical analysis.

Section 2.0

Heat Recovery
Energy
Depth
Analysis

2.0 HEAT RECOVERY ENERGY DEPTH ANALYSIS

Heat recovery will be utilized in the redesign of the mechanical system. The proposed mechanical redesign will provide an energy efficient solution to the problem of high amounts of wasted energy from the exhaust system for a building which contains laboratory spaces.

The main advantage of heat recovery is the reduction of heating costs in the building due to capturing waste heat and recycling it. According to an article in the February 2001 HPAC Engineering magazine, energy costs for heating and cooling are expected to rise as much as 50%. Since laboratory facilities have the most expensive energy costs for heating and cooling per sq. ft. in the country, heat recovery is a practical solution.

There are a number of things to take into consideration before deciding to design a heat recovery system. These considerations include:

- Energy recovery opportunities
- Location of the supply and exhaust air streams
- Risk associated with cross-contamination of the air streams
- Potential for fouling and corrosion of the devices
- Space requirements for additional equipment
- Operation and maintenance costs, as well as replacement costs

2.1 RUN-AROUND LOOP HEAT RECOVERY SYSTEM DISCUSSION

Run-around heat recovery systems include finned tube-type connections by counter flow piping. The coils are mounted in different airstreams connected by a closed loop hydronic system, either in series or parallel. A typical run-around energy recovery loop has finned tube water coils in the supply and exhaust air streams. A

pump circulates an appropriate heat transfer fluid through the system. This system is seasonally reversible, meaning that the exhaust air coil either preheats or pre-cools the outside air, depending on the season. When the outside air is cooler than the exhaust air, waste heat is recovered to preheat the outside air. When the outside air is warmer than the exhaust air, heat is removed from the outside air and, therefore, it is pre-cooled.

The main advantage of the run-around energy recovery loop is that the two air streams do not need to be adjacent, or even in close proximity, to each other. The distance between the two air steams is limited only by the cost of the piping and pumping systems. Run-around energy recovery systems have no possibility of cross contamination between the air streams. The main disadvantage of these systems is they can only transfer sensible heat between the air streams. Also, run-around loops require additional maintenance of the pumps and glycol system. Figure 2.1 shows a typical run-around heat energy recovery loop.

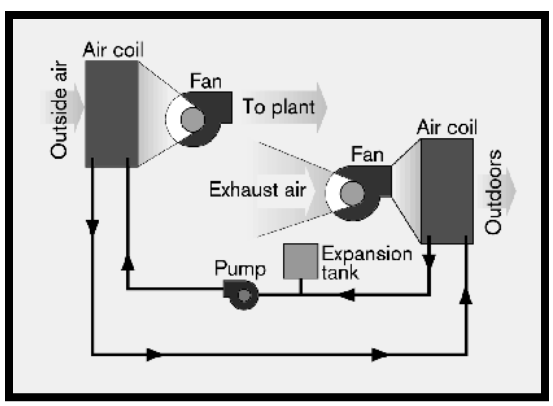


Figure 2.1

The effectiveness of a heat recovery system is defined as the amount of heat that is actually recovered as a percentage of the heat that is theoretically recoverable. According to the article "Laboratories for the 21st Century: Best Practices", put out by

the U.S. Department of Energy, the sensible effectiveness of run-around loops is typically between 55% and 66%. For my thesis, I have designed a run-around heat recovery system for the spaces served by AHU-1, AHU-2, and AHU-3.

2.1.1 COIL LOCATION

The location of the heat recovery coils was the first step in designing the run-around heat recovery system. The air handlers, as well as the exhaust fans are all located on the roof. Therefore, it made sense to look at locations on the roof where the coils could be placed into the air streams.

There are four 23,000 cfm fume hood exhaust fans which exhaust all of the laboratory and laboratory support spaces. Since these account for such a huge amount of exhaust air, compared to that of the nine smaller exhaust fans, only the exhaust ducts connected to the fume hood exhaust fans was considered as locations for the heat recovery coils. There is one 60" round duct which includes all exhaust air from the 1st, 2nd, and 3rd floor laboratory spaces. This duct accounts for 38,874 cfm of exhaust air, which is 52% of the total air exhausted by the fume hood exhaust fans.

There are five other exhaust ducts which connect to the fume hood exhaust fans on the roof, which consist of the remaining 48% (36,368 cfm) of the total air exhausted by the fume hood exhaust fans. There is one 24" round duct that accounts for 6,741 cfm, one 24" round duct that accounts for 6,184 cfm, one 28" round duct that accounts for 8,520 cfm, one 40 X 34 oval duct that accounts for 10,879 cfm, and one 18" round duct that accounts for 4,044 cfm of exhaust air. There is not much space around these five ducts, so it was decided to not include them in the run-around heat recovery design.

The locations for the coils of the supply air streams were determined next. The first thing that needed to be decided was how many supply air streams would be used in the run-around heat-recovery design. There are four air handling units, and it was decided to take advantage AHU-3 which all supplies 100% outside air. AHU-3 has an entering air flow rate of 29,130 cfm which most closely matches the flow rate of the 60" exhaust air duct. This unit accounts for 33% of the total air supplied to the building. Replacing the re-heat coils in the air handling units with the heat recovery coil, as well as adding the heat recovery coil to the existing air handling units were both considered as locations. Replacing the pre-heat coil in the air handler with the heat recovery coil was considered due to the fact that the added pressure drop in the supply system is lower than that of a system that has separate energy recovery and pre-heat coils. However, it was found that the leaving temperature of air due to the heat recovery was not enough to replace the pre-heat coil. It was therefore decided that the heat recovery coil would need to be added to the existing air handling unit (Shown in Figure 2.2). The supply-side heat recovery coil will be located up stream of the preheat coil and down stream of the filters in AHU-3.

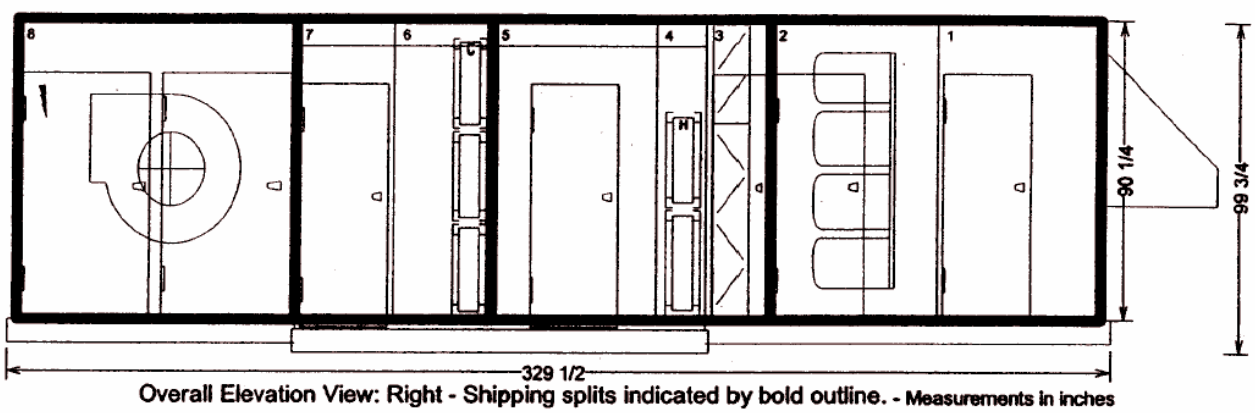
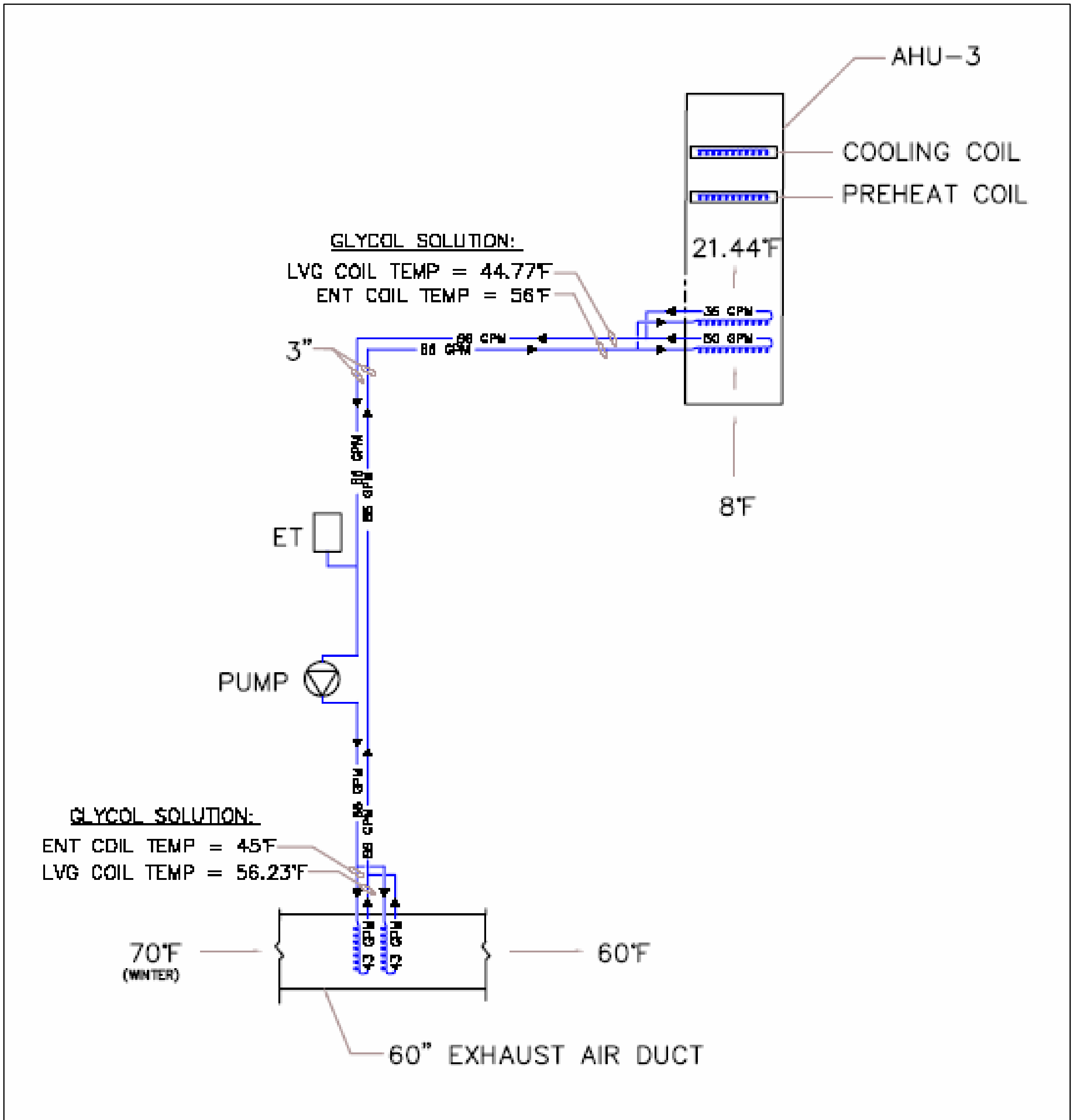


Figure 2.2

(1) Mixing module where the air enters the unit. (2)Bag, cartridge, or HEPA filters. (3) Face or bypass dampers. (4) Pre-heat coil. (5) Access module. (6) Cooling coil. (7) Access panel. (8) Fan.

The layout in Figure 2.3 shows the run-around system schematic. There are two heat recovery coils in the 60" exhaust stream, as well as two heat recovery coils in AHU-3. The temperatures shown are for winter conditions.



RUN-AROUND HEAT RECOVERY LOOP

Figure 2.3

2.1.2 COIL SELECTION

The coils for the run-around heat recovery design were chosen based on the cfm, and temperatures of the exhaust and supply air streams. The size of the air streams where the coils are being placed are also considered when selecting coils. Using Trane's TOPPS Selection program, it was found that the coil for the exhaust stream (in the 60" round duct) would have a flow rate of 86 GPM. Two heat recovery coils were chosen to be used in the exhaust air stream. Using the given air flow rate of 38,874 cfm, and assuming a design face velocity of 500 fpm, the coils are required to have a face area of 78 ft^2. The design face velocity is typically 500 fpm or less in laboratory applications, because lower face velocities result in lower pressure drops, higher effectiveness, and lower operating costs. The coils selected for the exhaust air duct each have a height of 45", and a finned length of 126" for a total face area of approximately 79 ft^2, which meets the required area. This rectangular shaped coil needs to be retrofitted into the round duct. Each exhaust-side coil has a pressure drop of 5 ft. H20.

The supply-side heat recovery coils were also chosen using the Trane's TOPPS Selection program. Again, a face velocity of 500 fpm was used to determine the face area needed for the heat recovery coils. Two heat recovery coils were chosen to be used in AHU-3. For the 29,130 cfm entering AHU-3, it was found that a face area of 58 ft^2 would be needed. The first coil selected for AHU-3 has a height of 24", and a finned length of 126", and the second coil has a height of 33", and a finned length of 126". The face area of these coils is approximately 50 ft^2, which is close enough to meet the required area.

Another important consideration taken into account when selecting the heat recovery coils was the fact that the coils themselves would increase the pressure drop

across the supply and exhaust fans. The horsepower of the fans were checked to be sure that they could handle the additional load.

2.1.3 PIPING AND PUMP SELECTIONS

Once the coils were chosen, both the flow rate and the pressure drop of the coils were known. The design of the piping and pumps was based on a typical hydronic system. The flow rate of the system is 86 GPM, and Bell & Gossett's Pipe Syzer was used to select the pipe size of 3" (Copper tubing). The heat recovery pipe requires 2" insulation made of closed-cell phenolic foam, and cellular glass with gasket in a PVC jacket and vapor retarder.

The pressure drop of the system was found to be 31.45 ft H20, assuming 4 feet of head per 100 feet of pipe length plus the pressure drop of the coils. The flow rate and pressure drop were used in Bell & Gossett's online pump selecting program to select the pump. The pump chosen is a series 1510, which is base mounted with an end suction, which will be located on the roof. The pump curve and information for this pump can be found in Appendix B. Table 2.1 shows the pump details for the run-around heat recovery loop:

Pump	Cooling Coil (GPM)	Head (ft)	Pump Speed (RPM)	Pump Motor (HP)	Pump Efficiency
HRP-1	86	31.45	1750	1 1/2	65.63%

Table 2.1

Ethylene glycol was used in the loop to keep the system from freezing in the winter. In its pure form, ethylene glycol is an odorless, colorless, syrupy liquid with a sweet taste. It is toxic, and should only be used in closed systems where ingestion

would not be possible. It was determined that 30% glycol would be used in the heat recovery system.

2.1.4 EXPANSION TANK SELECTION

Since the run-around system is a closed loop, it requires an expansion tank. The expansion tank is used to accommodate the expansion from cold to hot, so the entire fluid volume is computed. The total system water content volume is 72 gallons, calculated by using 3" diameter piping and the volume of water given in the coil selection data. Since the system is located outside, the maximum water temperature was assumed to be 100 F (in hot summer conditions), and the minimum water temperature is 45 F. The minimum operating pressure of the system is 12 psig, and the maximum operating pressure of the system is 30 psig. The system pressure is primarily being maintained so that if there would be a leak in the system, it would be known. The Wessels Company Tank Sizing Program was used to size the expansion tank, and it was found that a 2.7 gallon expansion tank would be needed in the system. The expansion tank will be located on the pump suction side of the loop, and will also be placed on the roof. The tank selection calculations can be found in Appendix C.

2.2 ENERGY ANALYSIS

Bin data taken for Lynchburg, VA was taken to model the performance of the run-around heat recovery addition to the existing mechanical system. The bin data supplied the outside temperatures which are reached in Lynchburg, as well as how many hours each of the temperatures are reached. The total number of hours in one year is 8,760, however only the hours which reach 55F and below were used to determine the heat recovery savings. Table 2.2 shows this bin data, as well as the calculated energy savings at each temperature. The savings were calculated using the

equation: **Savings = 1.08 * cfm * ΔT * hours**. The flow rate of air entering AHU-3 is 29,130 cfm. The "best case" heat recovery was calculated at an outdoor temperature of 8F. At this temperature, the heat recovery is ΔT=13.44F. The "worst case" heat recovery was calculated at an outdoor temperature of 55F. At this temperature, the heat recovery is ΔT=3.22F. The ΔT's in between were calculated by using a rate of decay= 0.2174 for every 1F of outdoor air.

Outside Air Temp (Bin) (°F)	Hours	ΔT (°F)	Temp after HR (°F)	Energy Savings (Btu)
55	404	3.22	58.22	40,926,205
53	288	3.65	56.65	33,115,479
51	272	4.09	55.09	34,997,183
49	265	4.52	53.52	37,722,201
47	275	4.96	51.96	42,908,179
45	321	5.39	50.39	54,477,409
43	286	5.83	48.83	52,450,503
41	224	6.26	47.26	44,144,841
39	269	6.70	45.70	56,693,632
37	320	7.13	44.13	71,820,419
35	270	7.57	42.57	64,292,568
33	253	8.00	41.00	63,706,016
31	218	8.44	39.44	57,875,566
29	125	8.87	37.87	34,895,758
27	121	9.31	36.31	35,434,593
25	87	9.74	34.74	26,668,083
23	62	10.18	33.18	19,853,113
21	45	10.61	31.61	15,025,199
19	55	11.05	30.05	19,116,632
17	36	11.48	28.48	13,005,250
15	32	11.92	26.92	11,998,040
13	24	12.35	25.35	9,326,894
11	18	12.79	23.79	7,241,443
9	9	13.22	22.22	3,743,858
7	8	13.66	20.66	3,437,328
5	1	14.09	19.09	443,348
3	1	14.53	17.53	457,030

Table 2.2

The total energy saved by the heat recovery system was found to be 855,776,768 Btu (855.777 K lb.st.) per year. The total amount of steam consumed by the science building for heating is 55,645 therms (5,564,500,000 Btu or 5,564.5 K lb.st.). Therefore, the energy saved by the heat recovery system is 15.4% per year.

2.3 LIFE CYCLE COST ANALYSIS

The total first cost of the run-around heat recovery system was found to be $14,147. The first costs of the run-around heat recovery equipment are broken down in Table 2.3.

Equipment	Cost	Source
150' of 3" Copper Pipe	$3,225	RS Means
150' of Phenolic Foam Pipe Insulation Material	$1,592	RS Means
150' of 5" PVC Protective Jacketing	$633	RS Means
1- ½ hp Centrifugal, Base-mounted Pump	$2,380	Bell & Gossett
Sheet metal Enclosure for Pump	$150	RS Means
(4) Heat Recovery Coils	$5,600	Trane
Expansion Tank (model NTA15)	$567	Wessels Company

Table 2.3

With the cost of steam being $4.38 per thousand pounds of steam, it was found that the heat recovery system saves $3,748 per year. Therefore, the payback period for the heat recovery system is just under 4 years.

Section 3.0

Distributed Pumping Depth Analysis

3.0 DISTRIBUTED PUMPING DEPTH ANALYSIS

Distribution systems can be categorized as centralized or distributed systems. Centralized pumping systems, or "source-distributed" systems have secondary pumps located only in the central plant. Distributed pumping systems have pumps located at each zone, with no secondary pumps. The existing system in the science building is centralized pumping system, however the following sections will discuss distributed pumping systems, and how one may be introduced into the science building.

3.1 MULTIPLE BUILDINGS VS. SINGLE BUILDING

Distributed pumping is suited very well where there are multiple buildings connected to a single chiller plant. Campus applications present the opportunity for optimizing energy by distributed pumping. This section will discuss the theory of distributed pumping systems for multi-building use, and then discuss and show how this same principle could be applied to a single building, such as the science building.

3.1.1 DISTRIBUTED PUMPING FOR MULTI-BUILDINGS USE

Distributed pumping was first proposed by Wilber Shuster of Cincinnati (Kisner). In a distributed pumping system, the pumps are remotely located around the loop, and are therefore closer to the load. This reduces the pump head and total horsepower compared to a layout where the pumps are directly adjacent to the chillers and boilers. There are no secondary pumps needed in a distributed pumping system. According to Tekworx.us, distributed pumping typically saves 25-40% of the pumping energy when a single thermal energy plant must serve multiple buildings.

For a multi-building system (shown in Figure 3.1), each building has its own variable volume pump. Each building may also have a DP transmitter at the hydronic

end of the building loop. The DPs are wired to the central controller which modulates the speed of each building¹s pump to satisfy its respective thermal load.

The most important characteristics of distributed pumping are:

- The pumps are located in each zone
- No secondary pumps in the plant
- Flexible to system diversity and expansion

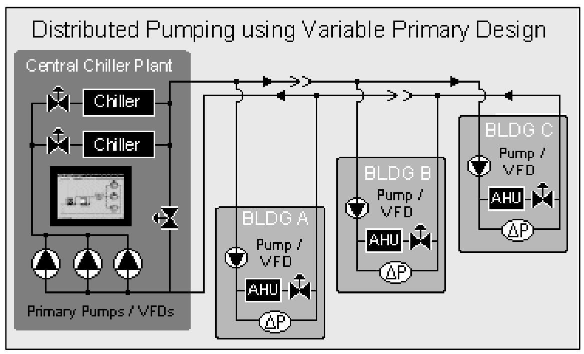
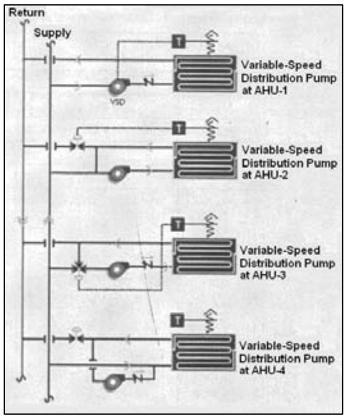


Figure 3.1

3.1.2 DISTRIBUTED PUMPING FOR A SINGLE BUILDING

In a variable-speed distributed pumping arrangement, constant flow primary pumps re-circulate the chilled water source in a primary source loop, and the variable-speed zone pumps draws flow from main distribution loop and distributes to the branches of each distributed pump. A controller measuring zone differential pressure across supply-return mains or across selected critical zones determines the speed of the zone distributed pumps. Two-way control valves in the load terminal return branch vary the flow required in the load.

In this type of system, the primary source pump would only need to supply water to these distributed pumps, and the distributed pumps will only need to be sized to overcome the frictional losses of the branch it serves. This system will reduce the



total head pressure, and total energy requirements. On the chilled water side, locating the distributed pumps directly at the cooling coils decentralizes the pumping and eliminates the need for two-way control valves. Figure 3.2 shows distributed pumps located directly at the air handling units.

Figure 3.2

3.2 EXISTING PRIMARY/SECONDARY CHILLED WATER SYSTEM

Primary/secondary pumping has been around since 1954. Until the past few decades, they were used mostly for larger commercial heating and chilled water cooling systems. Primary/secondary pumping is based on a simple fact: when two circuits are interconnected, flow in one will not cause flow in the other if the pressure drop in the piping common to both is eliminated. The key to all primary-secondary applications is the use of a common pipe which interconnects the primary and secondary circuits. The existing chilled water distribution system of the science building is a primary/secondary system with two-way modulating control valves on the terminal cooling equipment in the secondary chilled water loop.

There are two primary chilled water pumps and two secondary chilled water pumps in the science building. All of these pumps are located in the first floor mechanical room. P-1 and P-2, which are provided with variable frequency drive, service the primary chilled water, and P-3 and P-4 service the secondary chilled water. All of the chilled water pumps are mounted on a 6” concrete pad. Table 3.1 shows the pump schedule for the existing primary/secondary pumps, and Figure 3.3 shows the existing chilled water diagram.

Mark	GPM	Head (ft)	Type	Max RPM	Motor	
					HP	Volt/Phase/Hertz
P-1	1175	117	Base Mounted	1750	40	460/3/60
P-2	1175	117	Base Mounted	1750	40	460/3/60
P-3	600	40	Base Mounted	1750	15	460/3/60
P-4	600	40	Base Mounted	1750	15	460/3/60

Table 3.1

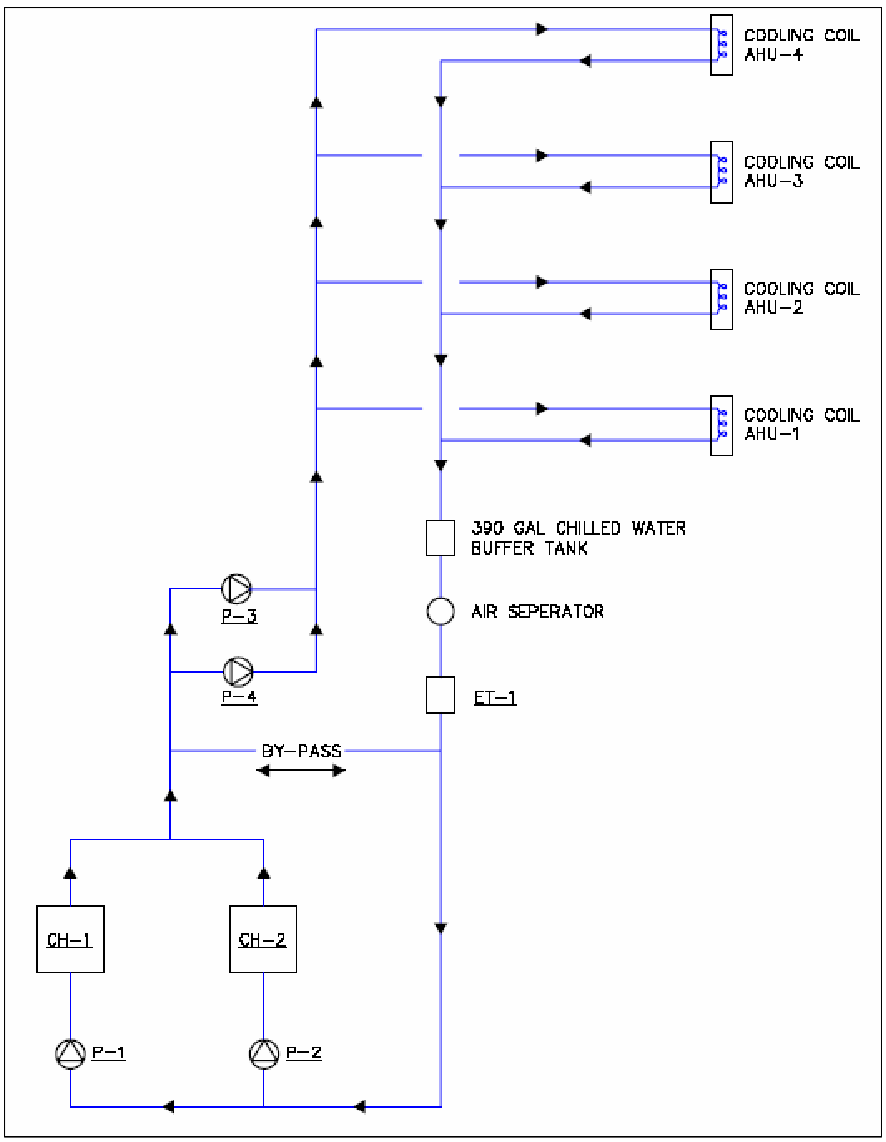


Figure 3.3

3.3 Addition of Distributed Pumps to Science Building

The proposal of the distributed pumping section of this thesis suggests replacing the secondary pumps with smaller variable-speed distributed pumps at the cooling coils. It is hoped that replacing these pumps will both increase the efficiency and decrease the first costs of the system. Figure 3.4 shows the chilled water system with the distributed pumping.

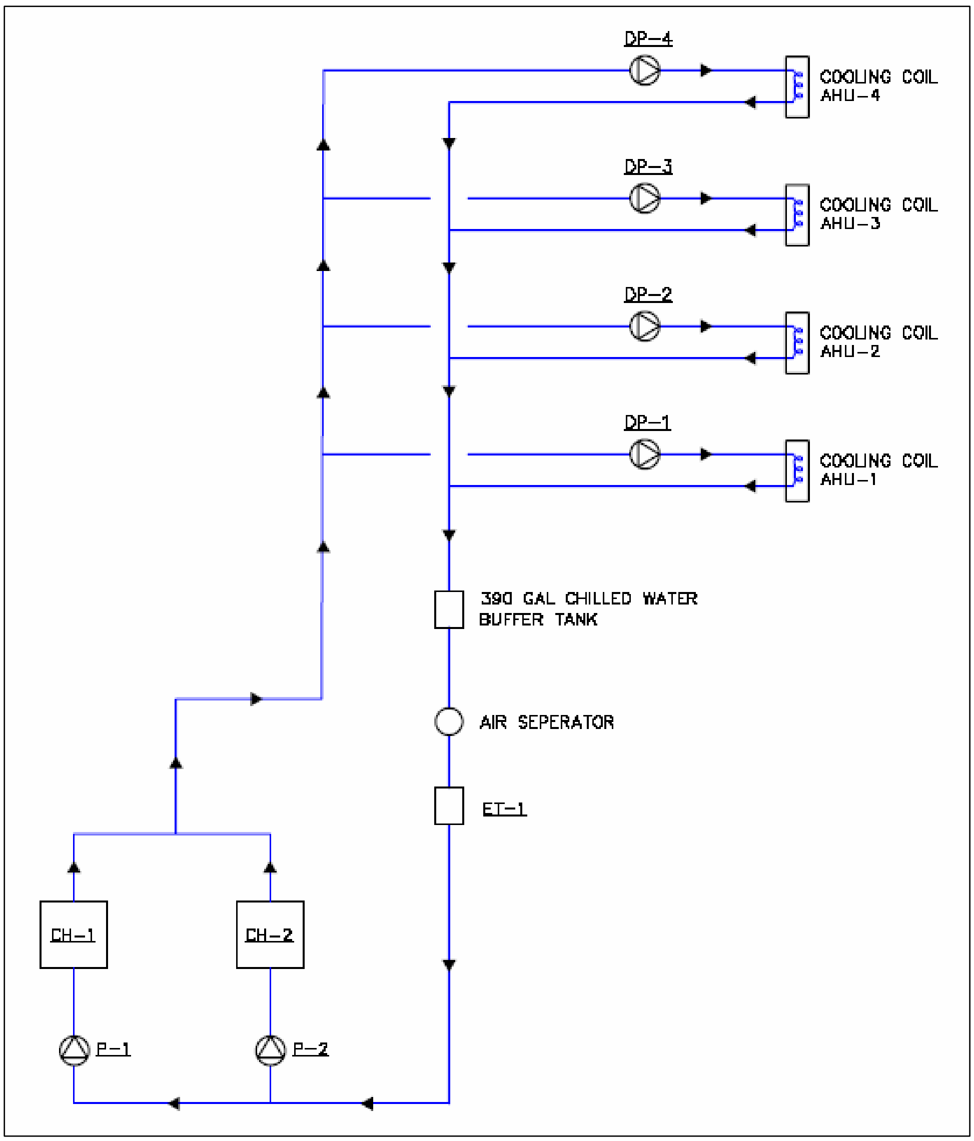


Figure 3.4

3.3.1 LOCATION OF DISTRIBUTED PUMPS

The primary chilled water pumps supply chilled water through piping from the first floor mechanical room up to the 4th floor. The piping is networked through the 4th floor ceiling to locations under each of the air handling units on the roof.

The placement of the chilled water piping does not leave many options for the location of the distributed pumps. Since the distributed pumps need to each be located in their separate zone, they must be placed between the main branch, and the cooling coil itself. In order to place these pumps between the main branch and the air handling units, the pump would need to be located in the 4th floor ceiling. For acoustical, structural, and space reasons, this location would not be sensible for the distributed pumps.

Another location that was considered for the distributed pumps was the mechanical room on the first floor, where the existing secondary pumps are located (shown in Figure 3.5). This location would be available after the secondary pumps were removed, however this arrangement would require about four times the amount of supply and return chilled water piping.

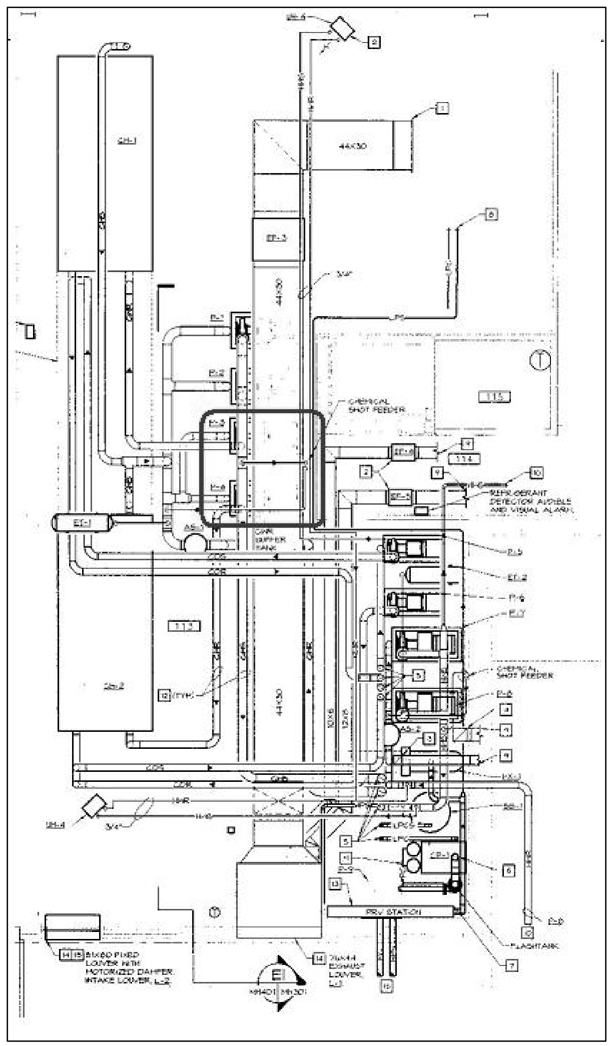


Figure 3.5

The roof was ultimately chosen for the location of the distributed pumps was the roof. With just a little additional piping, it would be possible to locate the distributed pumps directly next to each air handling unit, as Figure 3.6 shows. Pumps are not typically located outside due to the fact that they can freeze in low temperatures, however these pumps will only be operational in the summer months when cooling is needed. Protective enclosures, will, however be needed to shield the pumps from the weather year-round.

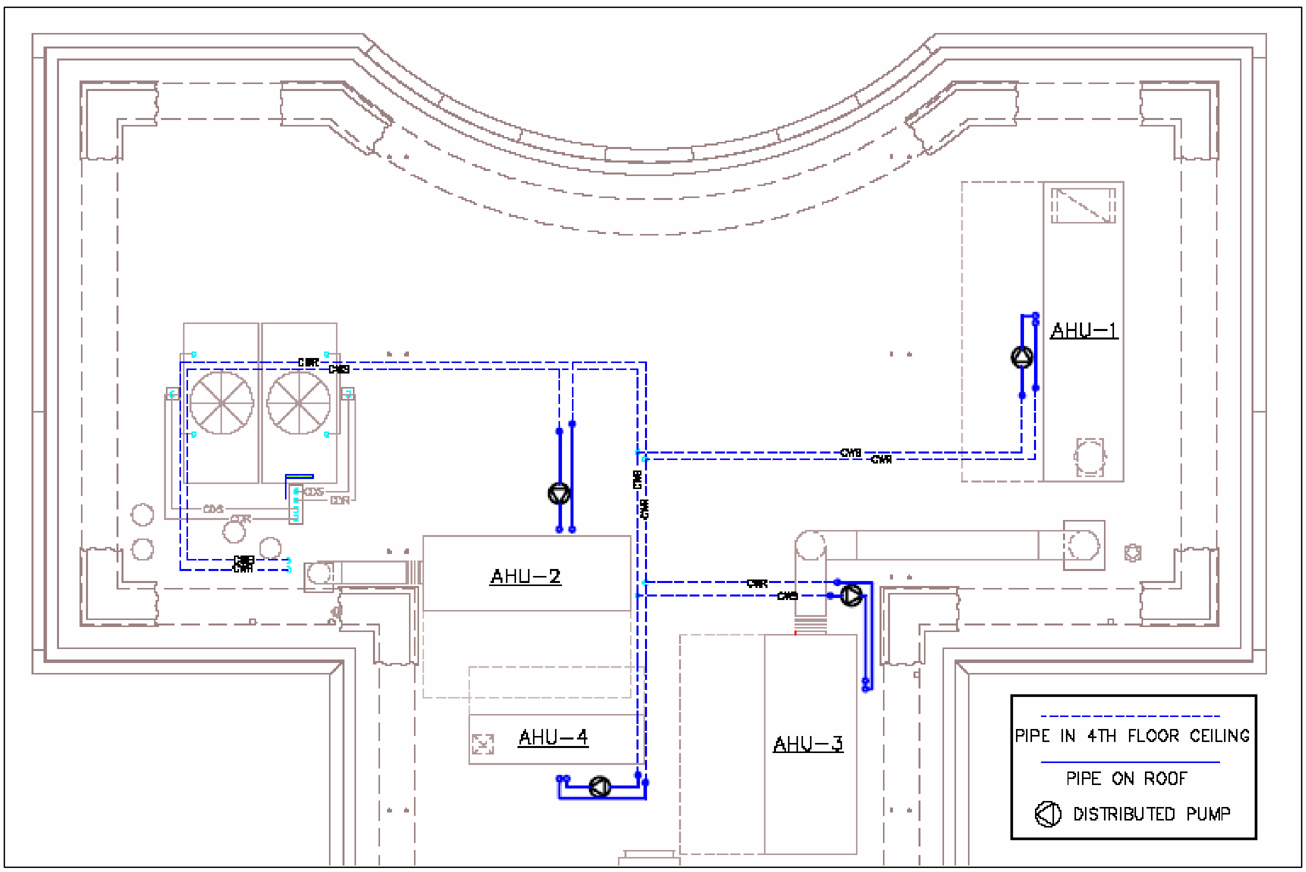


Figure 3.6

3.3.2 PUMP SELECTION AND PROTECTIVE ENCLOSURES

The distributed pumps to be located on the roof were selected as follows:

Pump	AHU Served	Cooling Coil (GPM)	Head (ft)	Pump Speed (RPM)	Pump Motor (HP)	Pump Efficiency
DP-1	AHU-1	299.34	29.8	1750	5	76.96%
DP-2	AHU-2	288.27	29.16	1750	3	76.56%
DP-3	AHU-3	455.25	16.56	1750	3	69.48%
DP-4	AHU-4	139.46	11.4	1150	3/4	65.54%

All pumps were chosen using Bell & Gossett's online pump selecting program. The pumps are all series 1510 (shown in Figure 3.7), which are base mounted with end suction. The pump curves and information for these pumps can be found in Appendix D.

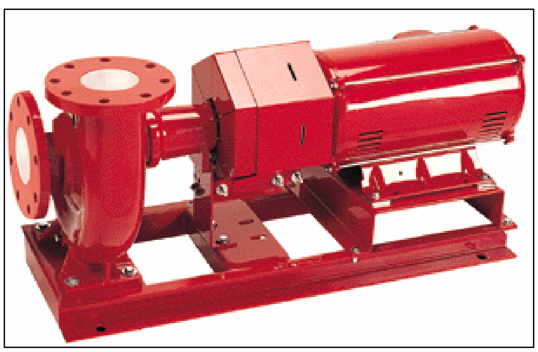


Figure 3.7

All pumps are located on the roof where they are exposed to the outside weather conditions. To protect the pumps from the elements, protective enclosures fabricated from sheet metal are needed to cover the pumps.

3.4 POWER AND COST ANALYSIS OF DISTRIBUTED PUMPING

In the distributed pumping system, each pump is sized to deliver its load's flow at just the head needed to pump the building hydronic loads and draw the chilled water through the mains from the central plant. No balancing valves, or by-pass piping are needed to eat up excess head because there is none. The pump speeds are controlled by Variable-Speed-Drives that receive signals from differential pressure switches at the end of the loop in each loop. There are many advantages to distributed pumping systems. The first is that it minimizes the pumping power. Furthermore, it minimizes the potential for low system ΔT by eliminating crossover bridges at the branches, and if it does become a problem, the pumps and chillers can effectively deal with it. Lastly, distributed pumping systems reduce head pressure imposed on the equipment.

The distributed pumps which were selected for the science building minimize pumping power. The existing secondary pumps in the building require two 15 hp motors. The distributed pumps selected to replace the secondary pumps have motor sizes of 5 hp, 3 hp, 3 hp, and ¾ hp. It is obvious that that distributed pumping system requires less power overall, in fact it requires only 39.2% of the horsepower required by the secondary pumps. There are 4,471 total annual cooling hours for the science building. When total difference in horsepower for the distributed pumps are converted to KW, and multiplied by the total annual cooling hours, the annual savings can be found. It was found that the distributed pumps save 60,870 kWh per year. Since the price of electricity is 6.23 cents per kWh, this is a savings of $3,792 per year.

The first cost of the two existing 600-GPM secondary pumps in the system was $8,800. The additions of four distributed pumps have a total first cost of $11,945. This 26% increase in cost is more than offset by the amount of power saved, as mentioned above. The payback period for the distributed pumps is approximately 10 months.

Section 4.0

Electrical Breadth Analysis

4.0 ELECTRICAL BREADTH ANALYSIS

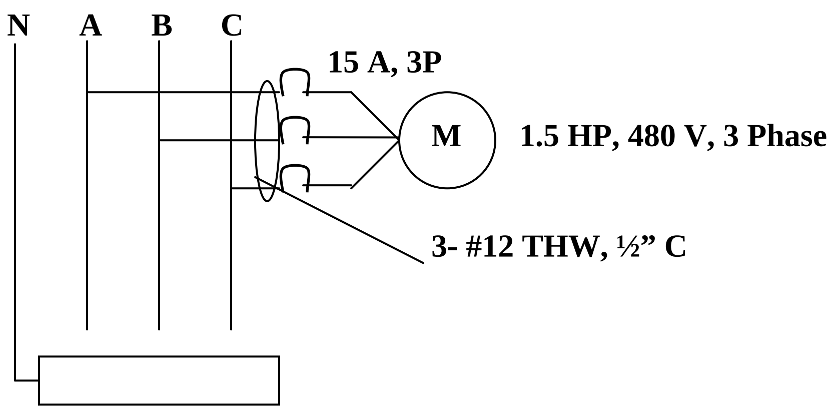
There were 5 pumps which were added to the roof of the science building in the mechanical redesign depths. These pumps have motors which create additional electrical load to the existing system. The existing panelboards were checked to see whether the additions will work with the existing electrical system, or whether a new panelboard would have to be added.

4.1 HEAT RECOVERY PUMP ADDED ELECTRICAL LOAD

The run-around loop heat recovery system includes one pump that is located on the roof. This pump has a motor which requires power from the building's electrical system. A power factor of 0.90 was assumed. The following calculations were completed to size the circuit breaker, conduit, and disconnect for the new motor.

PUMP HRP-1:
(1 1/2 HP, 480 V, 3 Phase, P.F. = 0.90)

NEC Table: 430-150 → FLC=**3 A**
KVA= 1.73 * 3 A * 480V * 0.90 = **2.242 KVA**
Total Demand: 2.242 + j (2.242/0.90)*(sin(cos^{-1}(0.90))) =**2.242 + j 1.086**
Circuit Breaker Size: 3 A * 175% = 5.25 A → **15 A, 3 P Breaker**
Conductor Size: 3 A * 125% = 3.75A
NEC Table 310.16→ **#12 THW (Cu)**
Conduit Size:
NEC Table C1 → **½ "**



It was found that the heat recovery pump requires a 15 A breaker. The electrical room on the 4th floor, room 411, has the closest panelboards to supply the power to the new load on the roof. Panelboard H4 had a spare circuit that could fit the new motor load, so a new panelboard would not be required. Figure 4.1 shows the panelboard with the 1-1/2 hp motor addition in circuit 32. In addition, the added motor requires a disconnect switch. The disconnect switch chosen is a 3P, 30/NF, 3R. 3R is the outdoor NEMA rating.

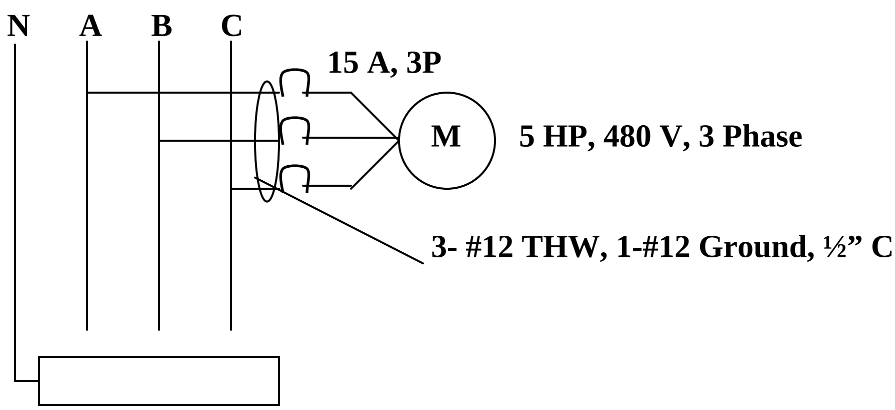
PANELBOARD H4 SCHEDULE														
400	AMP BUS		400	AMP MLO		480Y/277 Volts		3 PH, 4 W, SN,			MIN. 65 KAIC			Surface Mounted
LOAD SERVED	**LOAD AMPS**			**BKR TRIP**	**WIRE SIZE**	**CKT NO.**		**CKT NO.**	**WIRE SIZE**	**BKR TRIP**	**LOAD AMPS**			**LOAD SERVED**
	A	**B**	**C**								**A**	**B**	**C**	
EC-1	7.6			15	12	**1**		**2**	12	15	7.6			EC-2
		7.6										7.6		
			7.6										7.6	
EC-3	3.0			15	12	**7**		**8**	10	40	21.0			CT-1
		3.0										21.0		
			3.0										21.0	
BASIN HEATER	14.4			20	12	**13**		**14**	10	40	21.0			CT-2
		14.4										21.0		
			14.4										21.0	
BASIN HEATER	14.4			20	12	**19**		**20**	8	40	21.0			RETURN FAN
		14.4										21.0		AHU-1 15 HP
			14.4										21.0	
PANEL H4A	35.8			70	4	**25**		**26**	6	90	52.0			SUPPLY FAN
		26.6										52.0		AHU-1 40 HP
			19.4										52.0	
SPACE						**31**		**32**	**12**	**15**	**3.0**			**HRP-1**
												3.0		
													3.0	
LIGHTS	8.1			20	12	**37**		**38**	4/0	225	228.3			PANEL L4 VIA
LIGHTS		5.4		20	12	**39**						217.0		XFMR
SPARE				20		**41**							207.9	
SUB-TOTAL AMPS	83.3	71.4	58.8								353.9	342.6	333.5	SUB-TOTAL AMPS

Figure 4.1

4.2 DISTRIBUTED PUMPS ADDED ELECTRICAL LOAD

The addition of the distributed pumping system introduces four small pumps to the roof of the building. Each pump has a motor which requires power from the building's electrical system. A power factor of 0.90 was assumed. The following calculations were completed to size the circuit breaker, conduit, and disconnect for the new motor.

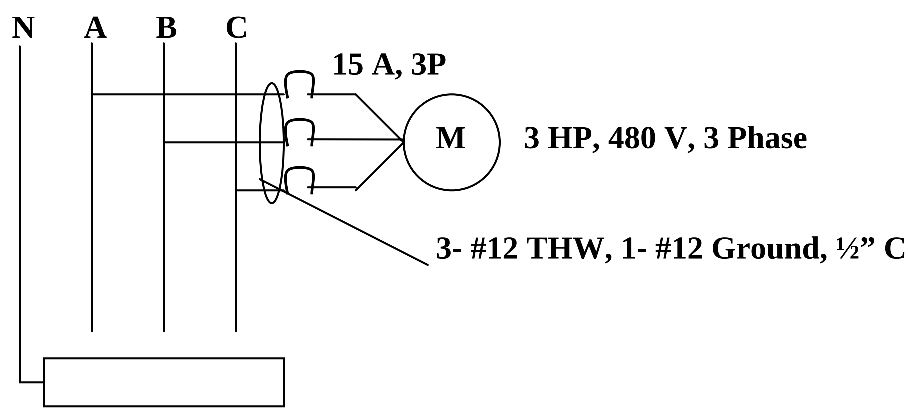
PUMP DP-1:
(5 HP, 480 V, 3 Phase, P.F. = 0.90)
NEC Table: 430-150 → FLC= 7.6 **A**
KVA= 1.73 * 7.6 A * 480V * 0.90 = 5.680 **KVA**
Total Demand: 5.680 + j (5.680/0.90)*(sin($\cos^{-1}$(0.90))) =**5.680 + j 2.751**
Circuit Breaker Size: 7.6 A * 175% = 13.3 A → **15 A, 3 P Breaker**
Conductor Size: 7.6 A * 125% = 9.5A
NEC Table 310.16 → **#12 THW (Cu)**
Conduit Size:
NEC Table C1 → **½ "**



It was found that DP-1 requires a 15 A breaker. Figure 4.2 shows panelboard H4A with the 5 hp motor addition in circuit 19.

PUMP DP-2:
(3 HP, 480 V, 3 Phase, P.F. = 0.90)

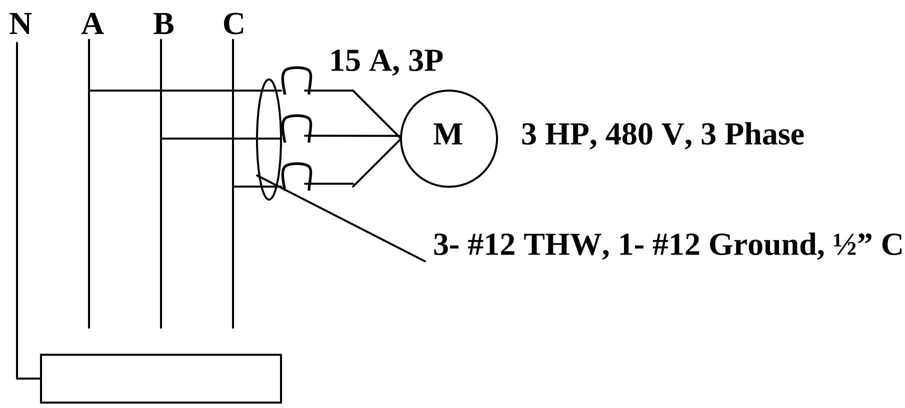
NEC Table: 430-150 → FLC=**4.8 A**
KVA= 1.73 * 4.8 A * 480V * 0.90 = **3.587 KVA**
Total Demand: 3.587+ j (3.587/0.90)*(sin(cos^{-1}(0.90))) =**3.587 + j 1.737**
Circuit Breaker Size: 4.8A * 175% = 8.4 A → **15 A, 3 P Breaker**
Conductor Size: 4.8 A * 125% = 6 A
NEC Table 310.16 → **#12 THW (Cu)**
Conduit Size:
NEC Table C1 → **½ "**



It was found that DP-2 requires a 15 A breaker. Figure 4.2 shows panelboard H4A with the 3 hp motor addition in circuit 2.

PUMP DP-3:
(3 HP, 480 V, 3 Phase, P.F. = 0.90)

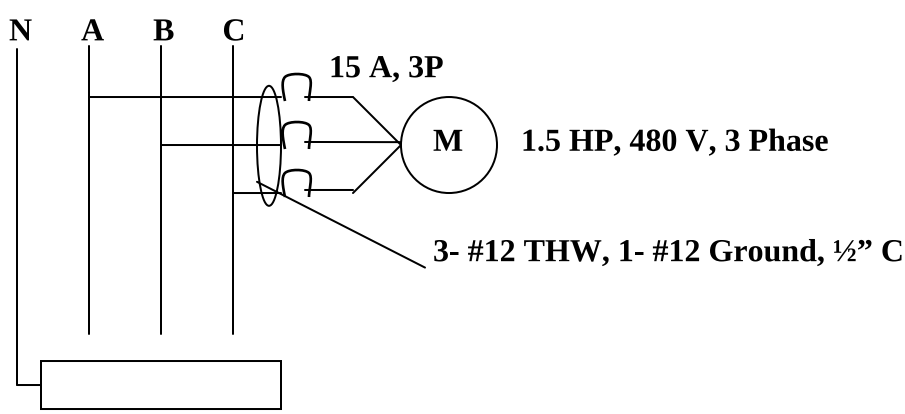
NEC Table: 430-150 → FLC=**4.8 A**
KVA= 1.73 * 4.8 A * 480V * 0.90 = **3.587 KVA**
Total Demand: 3.587+ j (3.587/0.90)*(sin(cos^{-1}(0.90))) =**3.587 + j 1.737**
Circuit Breaker Size: 4.8A * 175% = 8.4 A → **15 A, 3 P Breaker**
Conductor Size: 4.8 A * 125% = 6 A
NEC Table 310.16 → **#12 THW (Cu)**
Conduit Size:
NEC Table C1 → **½ "**



It was found that DP-3 requires a 15 A breaker. Figure 4.2 shows panelboard H4A with the 3 hp motor addition in circuit 14.

PUMP DP-4:
(3/4 HP, 480 V, 3 Phase, P.F. = 0.90)

NEC Table: 430-150 → FLC=**1.6 A**
KVA= 1.73 * 1.6 A * 480V * 0.90 = **1.196 KVA**
Total Demand: 1.196+ j (1.196/0.90)*(sin($\cos^{-1}$(0.90))) =**1.196 + j 0.579**
Circuit Breaker Size: 1.6 A * 175% = 2.8 A → **15 A, 3 P Breaker**
Conductor Size: 1.6 A * 125% = 2 A
NEC Table 310.16 → **#12 THW (Cu)**
Conduit Size:
NEC Table C1 → **½ "**



It was found that DP-4 requires a 15 A breaker. Figure 4.2 shows panelboard H4A with the 3/4 hp motor addition in circuit 20.

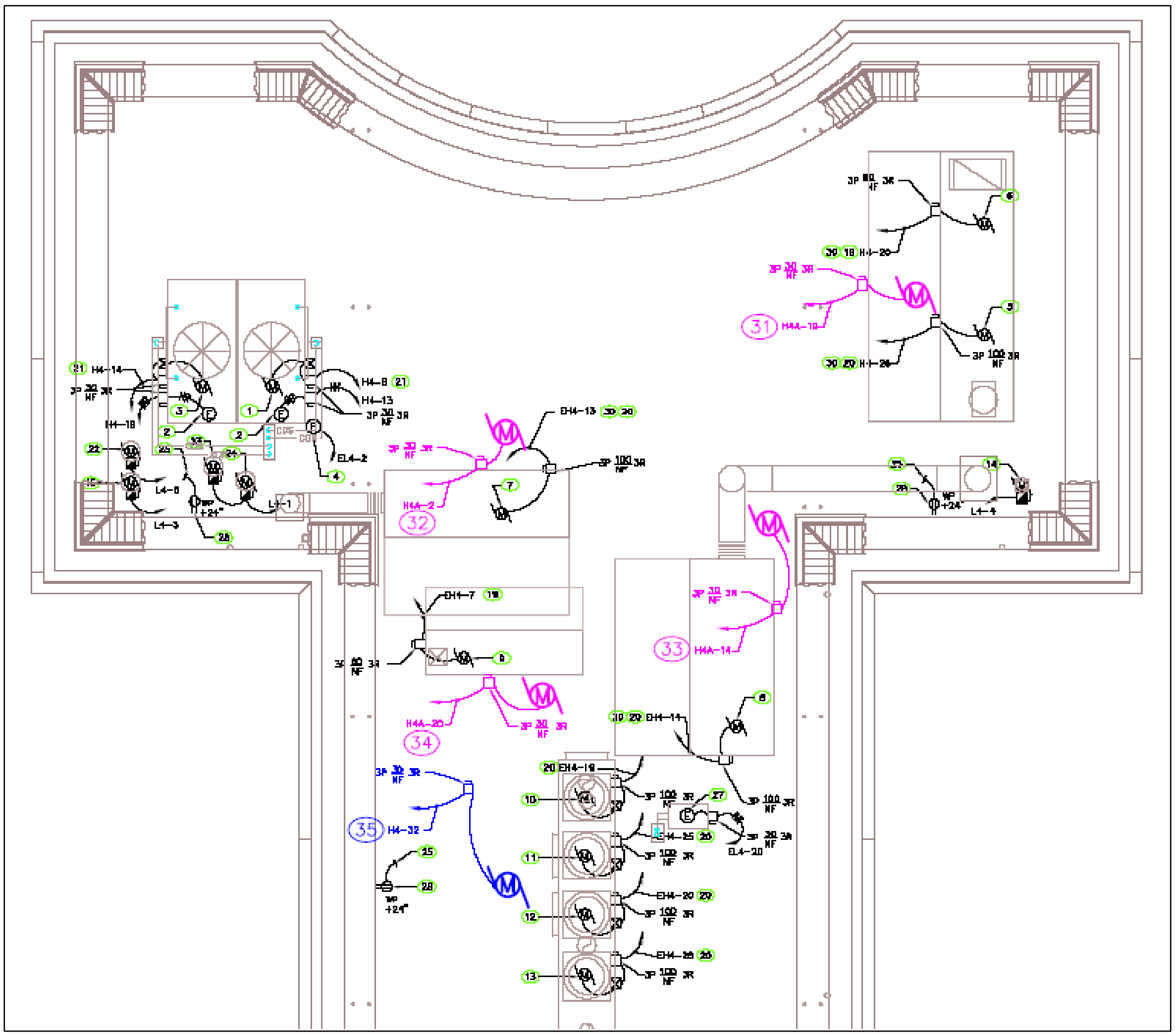
PANELBOARD H4A SCHEDULE

100 AMP BUS 100 AMP MLO 480Y/277 Volts 3 PH, 4 W, SN, MIN. 65 KAIC Surface Mounted

LOAD SERVED	LOAD AMPS A	LOAD AMPS B	LOAD AMPS C	BKR TRIP	WIRE SIZE	CKT NO.		CKT NO.	WIRE SIZE	BKR TRIP	LOAD AMPS A	LOAD AMPS B	LOAD AMPS C	LOAD SERVED
LIGHTING	10.4			20	12	1		2	12	15	4.8			DP-2
LIGHTING		11.4		20	12	3						4.8		
LIGHTING			10.0	20	12	5							4.8	
LIGHTING	7.8			20	12	7		8	12	20	8.8			MECH-VAV Units
LIGHTING		7.7		20	12	9		10		20				SPARE
LIGHTING			9.4	20	12	11		12		20				SPARE
LIGHTING	8.8			20	12	13		14	12	15	4.8			DP-3
CRAC		7.5		20	12	15						4.8		
SPARE				20		17							4.8	
DP-1	7.6			15	12	19		20	12	15	1.6			DP-4
		7.6										1.6		
			7.6										1.6	
SUB-TOTAL AMPS	34.6	34.2	27.0								20.0	11.2	11.2	SUB-TOTAL AMPS

Figure 4.2

There were no additional panelboards needed, because panelboard H4A in room 411 had enough space to house all of the additional loads. In addition, all added motors required a disconnect switch. The disconnect switches chosen were 3P, 30/NF, 3R. "3R" is the NEMA outdoor enclosure which is rain-tight and sleet resistant. Figure 4.3 shows the locations of the new electrical loads on the roof.



31 DP–1, 5 HP, 480V, 3 PHASE
32 DP–2, 3 HP, 480V, 3 PHASE
33 DP–3, 3 HP, 480V, 3 PHASE
34 DP–4, 3/4 HP, 480V, 3 PHASE
EXISTING ELECTRICAL LOADS
35 HRP–1, 1–1/2 HP, 480V, 3 PHASE
M DISTRIBUTED PUMPING MOTORS
M MOTOR FOR HEAT RECOVERY PUMP

Figure 4.3

Section 5.0

Acoustical Breadth Analysis

5.0 ACOUSTICAL BREADTH ANALYSIS

Controlling the HVAC system noise is essential to the quality of the listening environment in classrooms. Often there are noise levels created by the mechanical equipment which exceed the allowable levels set for classrooms. The recommended RC rating for HVAC noise in classrooms is 25-30 dB. The purpose of this analysis is to ensure that a typical classroom in the science building meets this specified RC rating. The conclusion of this section proposes recommendations that could be added to the existing system.

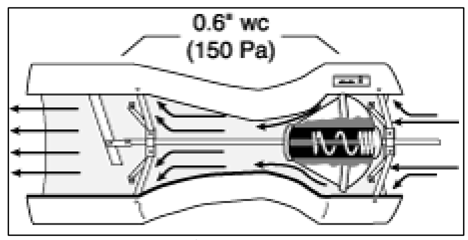


Figure 5.1

Classroom 130 on the first floor is a 9 ft high x 30 ft long x 29 ft wide room, and is typical of the classrooms in the science building. There is one supply VAV terminal unit, as well as one exhaust VAV terminal unit in the room. The airflow rates of the supply VAV box vary from 1320 to 1522 cfm, and the airflow rates of the exhaust VAV box vary from 1256 to 1320 cfm. Each VAV unit includes an airflow control valve, as shown in Figure 5.1. The sound performance data for the valves in these VAV units were obtained from Phoenix Control's Laboratory Sourcebook. The data is broken into sound power levels for discharge, exhaust, and radiated operating conditions. Discharge sound is the sound generated by the valve that is transmitted through the discharge duct into the occupied space. Exhaust sound is the sound generated by the valve that is transmitted through the inlet duct into the occupied space. The radiated sound is the sound transmitted through the valve body. Attenuations for the elbow in the duct, as well as the terminating point were subtracted from the given levels for the discharge and exhaust sound levels. Attenuations for the VAV housing was subtracted from the given levels for the radiated sound levels. These three sound power levels (Lw's) were combined to obtain the sound power levels of each VAV

unit at each frequency (see Tables 5.1 and 5.2). These combined levels for each VAV unit were then combined to obtain a total combined level at each frequency (see Table 5.4). The equation used to combine the levels is:

$Lw\ (Combined) = 10 * \log(10^\wedge(L_D/10)+10^\wedge(L_E/10)+10^\wedge(L_R/10))$

Frequency	Exhaust VAV Unit Lw's			
	Discharge	Exhaust	Radiated	**Combined**
125	53	52	51	**57**
250	47	44	43	**50**
500	43	39	28	**45**
1000	51	42	26	**52**
2000	45	39	18	**46**
4000	39	33	9	**40**

Table 5.1

Frequency	Supply VAV Unit Lw's			
	Discharge	Exhaust	Radiated	**Combined**
125	58	57	57	**62**
250	53	49	48	**55**
500	49	46	32	**51**
1000	56	51	30	**57**
2000	51	49	22	**53**
4000	51	49	19	**53**

Table 5.2

These combined sound power levels (Lw's) as well as the room constants (R_T's) were needed to find the sound pressure levels (Lp's) at each frequency. The

room constant for each frequency was a function of the room materials, as well as the room dimensions. The ceiling in the room is made of acoustical tile, the floor is made of vinyl composition tile (VCT), and the walls are painted gypsum wallboard. The absorption coefficients for these materials are listed in Table 5.3. The ceiling and floor area of the room is 870 ft^2, and the wall area is 1062 ft^2.

Frequency	Absorption Coefficients		
	α ceiling	α floor	α walls
125	0.76	0.02	0.29
250	0.93	0.03	0.10
500	0.83	0.03	0.05
1000	0.99	0.03	0.04
2000	0.99	0.03	0.07
4000	0.94	0.02	0.09

Table 5.3

The equation used to determine the R_T's is:

$$R_T = \Sigma(S_i\alpha_i) / (1 - \Sigma(S_i\alpha_i) / \Sigma(S_i))$$

The R_T's are summarized in Table 5.4, along with the sound power levels which were computed using the equation:

$$Lp = Lw - 10 \log(R_T) + 6$$

Frequency	Total Combined Lw	R_T	Lp
125	63	1523	37
250	56	1418	31
500	52	1122	**27**
1000	58	1392	**33**
2000	54	1464	**28**
4000	53	1394	28

Table 5.4

The total RC rating for the room was determined by taking the average of the 500, 1000, and 2000 octave band sound pressure levels. It was found that the RC rating of the room is 29 dB. Although this rating is at the high end, it does fall within the recommended RC rating of 25-30 dB. This RC rating number corresponds to the speech communication or masking properties of the noise. The quality or character of the background noise can be determined by finding the RC-II rating. The RC-II rating is found by plotting the Lp's on the RC curve at each frequency and taking the difference between the Lp's and the recommended RC rating for the space. These ΔL's are used in the following equations:

$$\mathbf{LF = 10 \log [(10^\wedge(\Delta L_{16}/10) + 10^\wedge(\Delta L_{31.5}/10) + 10^\wedge(\Delta L_{63}/10)) / 3]}$$

$$\mathbf{MF = 10 \log [(10^\wedge(\Delta L_{125}/10) + 10^\wedge(\Delta L_{250}/10) + 10^\wedge(\Delta L_{500}/10)) / 3]}$$

$$\mathbf{HF = 10 \log [(10^\wedge(\Delta L_{1000}/10) + 10^\wedge(\Delta L_{2000}/10) + 10^\wedge(\Delta L_{4000}/10)) / 3]}$$

A base case of 25 dB was used because it is the lowest recommended level for a classroom. Appendix E shows the RC curve that was used. The low-frequency sound pressure levels were assumed, and they are shown on the RC

curve. It was found that the range of frequencies (the Quality Assessment Index) is 20 dB which is clearly unacceptable.

The subjective quality of the background noise was classified as well. The sound pressure levels of the valves at low frequencies were found to be acceptable; however the valves have high sound pressure levels at high frequency which produce a "hiss" sound. Therefore, an (H) would be placed after the RC rating to denote this "hissy" quality.

To conclude, the classroom in the science building was found to have a rating of RC- 29 H. Overall this is a good rating. The best way to reduce the "hiss" would be to reduce the flow velocity through the valves, which will reduce the high sound pressure levels in the classroom at high frequencies.

Section 6.0

Conclusions and Recommendations

6.0 CONCLUSIONS AND RECOMMENDATIONS

The run-around heat recovery applied to the mechanical system of Longwood University's new science building is located on the roof. The system includes 4 heat recovery coils, 150 feet of copper tubing with insulation, an expansion tank, and a small pump. The heat recovery coils were applied to the 60" exhaust duct and AHU-3, which is a 100% outside air unit. It was found that the run-around heat recovery system successfully reduces the heating loads in the science building by 855,776,768 Btu per year, which is a 15.4% savings. The lower than expected first cost of the heat recovery design allows for a payback of only 4 years. The addition of the heat recovery system proved to have little impact on the electrical system. The system takes up minimal space and is operated by a small 1- ½ hp motor. No additional panelboards were needed. Overall, the run-around heat recovery system provides an efficient response to the problem of large energy consumption by the existing 100% outdoor air-handling units.

The distributed pumping system applied to Longwood University's chilled water system includes four small variable-speed pumps which are located on the roof next to each air handler. These distributed pumps replace the use of the two existing 15 hp secondary pumps which are located in the first floor mechanical room. In this system, the primary pumps supply water to these distributed pumps, and the distributed pumps are sized only to overcome the frictional losses of the branch it serves. No balancing valves, or by-pass piping are needed to eat up excess head because there is none. The pump speeds are controlled by Variable-Speed-Drives (VSD's) that receive signals from differential pressure switches at the end of the loop in each loop. This system reduces the total head pressure, as well as energy consumption.

The four distributed pumps run on a total of 11.75 hp, a 60.8% decrease of the two existing secondary pumps. Annually, this saves 60,870 kWh, a total of $3,792 per

year. This savings was enough to account for the increased first cost of the distributed pumps over the secondary pumps. The payback period was found to be 10 months. The addition of these small pumps also proved to have little impact on the electrical system, because no additional panelboards were required.

The acoustical study was performed with the noise levels of the air valves located in the VAV boxes of a typical classroom in the science building. It was found that the classroom has an RC rating of 29 dB, which falls within the recommended level of 25-30 dB. The RC-II rating showed that sound pressure levels of the valves at low frequencies were too high, which result in a "hissy" sound. To reduce this sound, it was determined that the flow velocity through the valves should be reduced.

Section 7.0

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Trane Representatives:

- Kohler, Brian. Pittsburgh, Pa.
- Orr, Mike. Richmond, Va.

Section 8.0

Appendicies

Cooling Coil

Model number

D--B45

Coil Information

Tag	Exhaust Wint	Actual airflow	19437 cfm
Quantity	1	Entering dry bulb	70.00 F
System type	Chilled Water	Entering wet bulb	54.00 F
Unit size	80	Total capacity	212.28 MBh
Coil type	UW	Sensible Capacity	212.27 MBh
Coil tube drainability		Leaving dry bulb	60.00 F
Rows	2	Leaving wet bulb	49.72 F
Nominal coil height	45" (1143 mm)	Actual coil face area	39.38 sq ft
Finned length	126" (3200 mm)	APD	0.18 in H2O
Fin material	Aluminum	Volume	9.35 gal
Fin spacing	143 fins per foot	Elevation	0.00 ft
Fin type	Delta flo E	Face velocity	493.64 ft/min
Tube matl/wall thickness	.016 (0.406 mm) copper	Max Air PD	
Coil coating	No	Max face velocity	
Rigging weight	252.6 lb	Max fin spacing	
Installed weight	344.4 lb	Turbulators	Yes

Note: Ratings outside the scope of the ARI Air-Cooling and Air-Heating Coils Certification Program.

Chilled Water Information

Fluid type	Ethylene Glycol	Fluid PD	5.00 ft H2O
Fluid concentration	30.00 %	Maximum water pressure drop	
Standard fluid flow rate	43.00 gpm	Fluid velocity	2.11 ft/sec
Entering fluid temp	45.00 F	Fouling factor	0.00000 hr-sq ft-deg
Fluid temp rise	11.23 F	Reynolds number	2690.73 Each
Leaving water temperature	56.23 F		

Cooling coil

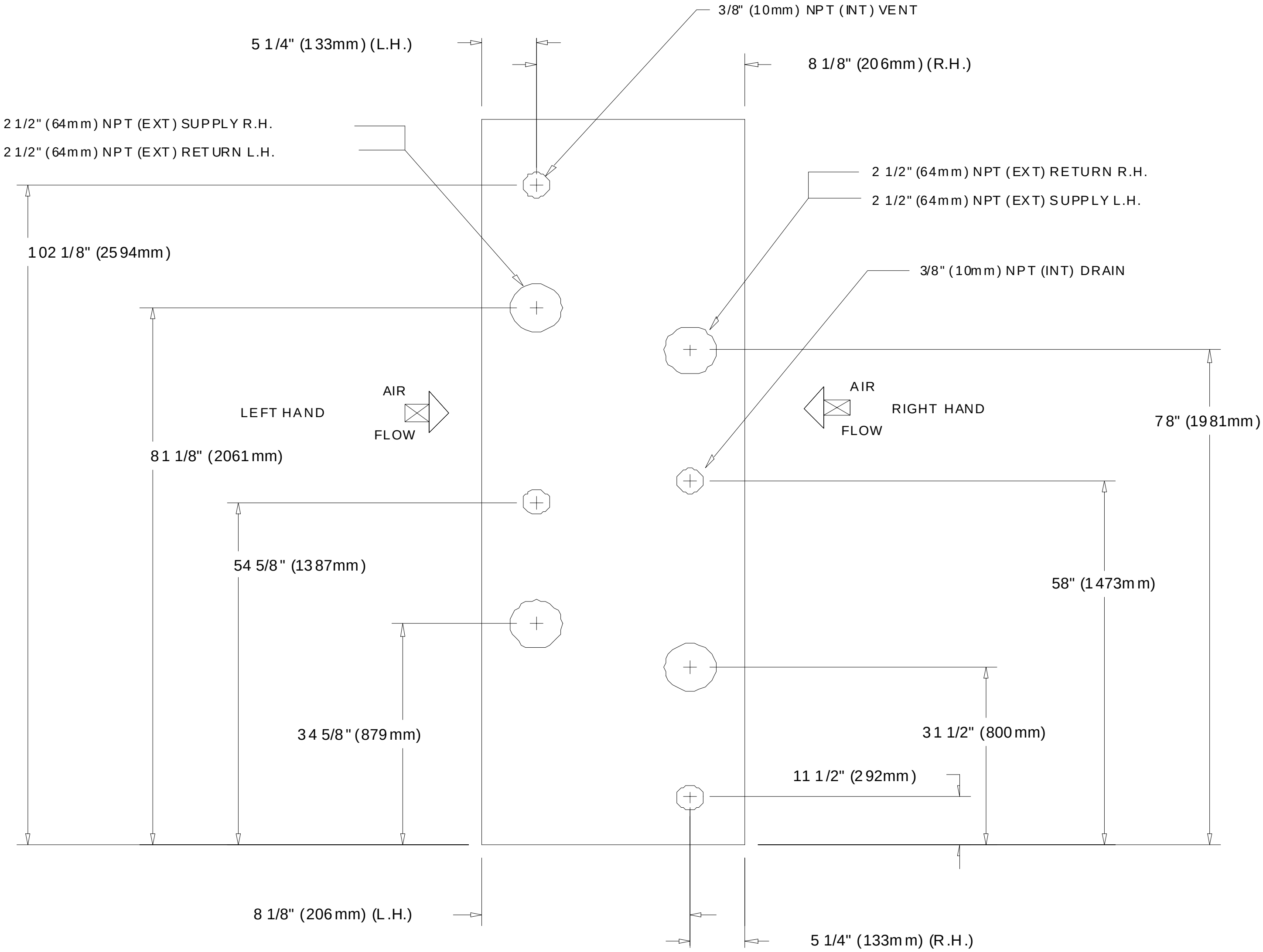
Refrigerant Information

Refrigerant type	Number of distr - ent coil type #1
Liquid temp	Number of distr - lvg coil type #1
Suction temp	Number of distr - ent coil type #2
Circuiting type	Number of distr - lvg coil type #2
Distributor type - ent air side	Number of distr - ent coil type #3
Distributor type - lvg air side	Number of distr - lvg coil type #3
DX circuits - ent air side	Total cap ent coil type #1
DX circuits - lvg air side	Total cap lvg coil type #1
LDB - ent air vert split	Total cap ent coil type #2
Packed elbows	Total cap lvg coil type #2
LWB - ent air vert split	Total cap ent coil type #3
	Total cap lvg coil type #3

Stacked Coil Information

Qty of stacked coil #1		Coil bank total cap	424.56 MBh
Qty of stacked coil #2		Coil bank sensible cap	424.54 MBh
Qty of stacked coil #3		Coil bank standard flow rate	86.00 gpm
Nominal ht stacked coil #1		Coil bank volume	18.71 gal
Nominal ht stacked coil #2		Coil bank rigging weight	505.1 lb
Nominal ht stacked coil #3		Coil bank installed weight	688.8 lb
Coil bank airflow	38874 cfm	Coil bank fluid PD	5.00 ft H2O

Cooling coil (CLCL)
Items: Exhaust Wint Exhaust Wint
Tags: Exhaust Wint Exhaust Wint
03/01/2005 07:15:50 PM



Cooling coil

Job Information

Job01
Richmond

Comments

Model number

D--B45

Coil Information

Tag	Exhaust Wint	Actual airflow	19437 cfm
Quantity	1	Entering dry bulb	70.00 F
System type	Chilled Water	Entering wet bulb	54.00 F
Unit size	80	Total capacity	212.28 MBh
Coil type	UW	Sensible Capacity	212.27 MBh
Coil tube drainability		Leaving dry bulb	60.00 F
Rows	2	Leaving wet bulb	49.72 F
Nominal coil height	45" (1143 mm)	Actual coil face area	39.38 sq ft
Finned length	126" (3200 mm)	APD	0.18 in H2O
Fin material	Aluminum	Volume	9.35 gal
Fin spacing	143 fins per foot	Elevation	0.00 ft
Fin type	Delta flo E	Face velocity	493.64 ft/min
Tube matl/wall thickness	.016 (0.406 mm) copper	Max Air PD	
Coil coating	No	Max face velocity	
Rigging weight	252.6 lb	Max fin spacing	
Installed weight	344.4 lb	Turbulators	Yes

Note: Ratings outside the scope of the ARI Air-Cooling and Air-Heating Coils Certification Program.

Chilled Water Information

Fluid type	Ethylene Glycol	Fluid PD	5.00 ft H2O
Fluid concentration	30.00 %	Maximum water pressure drop	
Standard fluid flow rate	43.00 gpm	Fluid velocity	2.11 ft/sec
Entering fluid temp	45.00 F	Fouling factor	0.00000 hr-sq ft-deg
Fluid temp rise	11.23 F	Reynolds number	2690.73 Each
Leaving water temperature	56.23 F		

Cooling coil

Refrigerant Information

Refrigerant type	Number of distr - ent coil type #1
Liquid temp	Number of distr - lvg coil type #1
Suction temp	Number of distr - ent coil type #2
Circuiting type	Number of distr - lvg coil type #2
Distributor type - ent air side	Number of distr - ent coil type #3
Distributor type - lvg air side	Number of distr - lvg coil type #3
DX circuits - ent air side	Total cap ent coil type #1
DX circuits - lvg air side	Total cap lvg coil type #1
LDB - ent air vert split	Total cap ent coil type #2
Packed elbows	Total cap lvg coil type #2
LWB - ent air vert split	Total cap ent coil type #3
	Total cap lvg coil type #3

Stacked Coil Information

Qty of stacked coil #1		Coil bank total cap	424.56 MBh
Qty of stacked coil #2		Coil bank sensible cap	424.54 MBh
Qty of stacked coil #3		Coil bank standard flow rate	86.00 gpm
Nominal ht stacked coil #1		Coil bank volume	18.71 gal
Nominal ht stacked coil #2		Coil bank rigging weight	505.1 lb
Nominal ht stacked coil #3		Coil bank installed weight	688.8 lb
Coil bank airflow	38874 cfm	Coil bank fluid PD	5.00 ft H2O

Cooling coil (CLCL)
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Tags: Exhaust Wint Exhaust Wint
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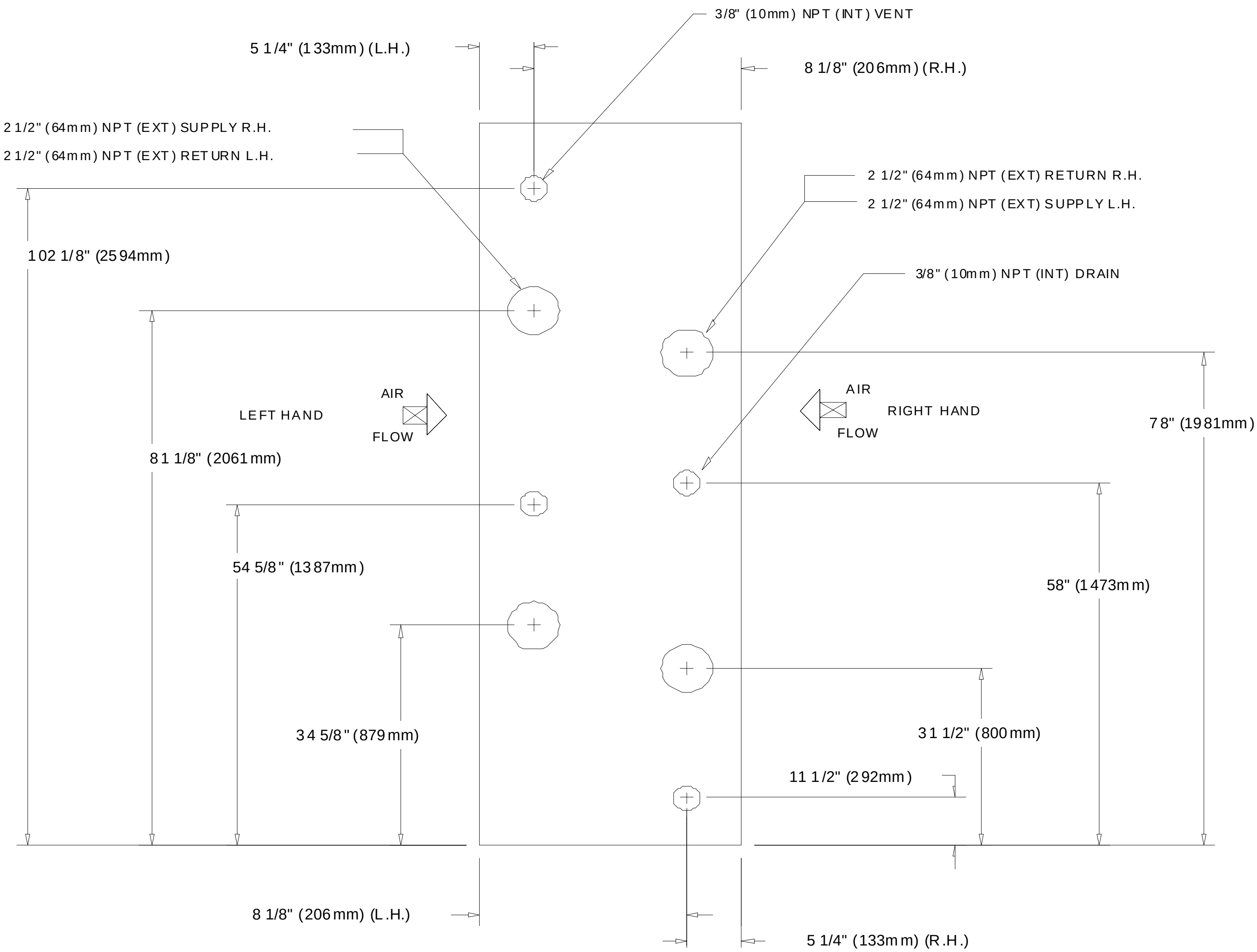


TOPSS Dimension Drawing

ALL WEIGHTS AND DIMENSIONS ARE APPROXIMATE. CERTIFIED PRINTS ARE AVAILABLE UPON REQUEST.

Heating Coil (HTCL)

Job Information

Job01
Richmond

Comments

Model number

D--B24

Coil Information

Tag	AHU-3 Wint	Actual airflow	12265 cfm
Quantity	1	Entering dry bulb	8.00 F
System type	Hot Water	Leaving dry bulb	21.44 F
Unit size	66	Total capacity	178.74 MBh
Coil type	5W	Actual coil face area	21.00 sq ft
Coil tube drainability	Drainable	APD	0.15 in H2O
Rows	1	Volume	3.14 gal
Finned length	126" (3200 mm)	Elevation	0.00 ft
Nominal coil height	24" (610 mm)	Face velocity	584.06 ft/min
Fin type	Prima-Flo H	Max air PD	
Fin material	Aluminum	Max face velocity	
Fin spacing	130 fins per foot	Max fin spacing	
Tube matl/wall thickness	.020 (0.508 mm) copper	Rigging weight	104.9 lb
Coil coating	No	Installed weight	131.0 lb
Turbulators	Yes		

Hot Water Information

Fluid type	Ethylene Glycol
Fluid concentration	30.00 %
Standard fluid flow rate	36.21 gpm
Entering fluid temp	56.00 F
Fluid temp drop	11.23 F
Leaving fluid temp	44.77 F
Fluid PD	20.45 ft H2O
Max fluid PD	
Fluid velocity	5.00 ft/sec
Fouling factor	0.00050 hr-sq ft-deg
Reynolds number	7921.06

Steam Information

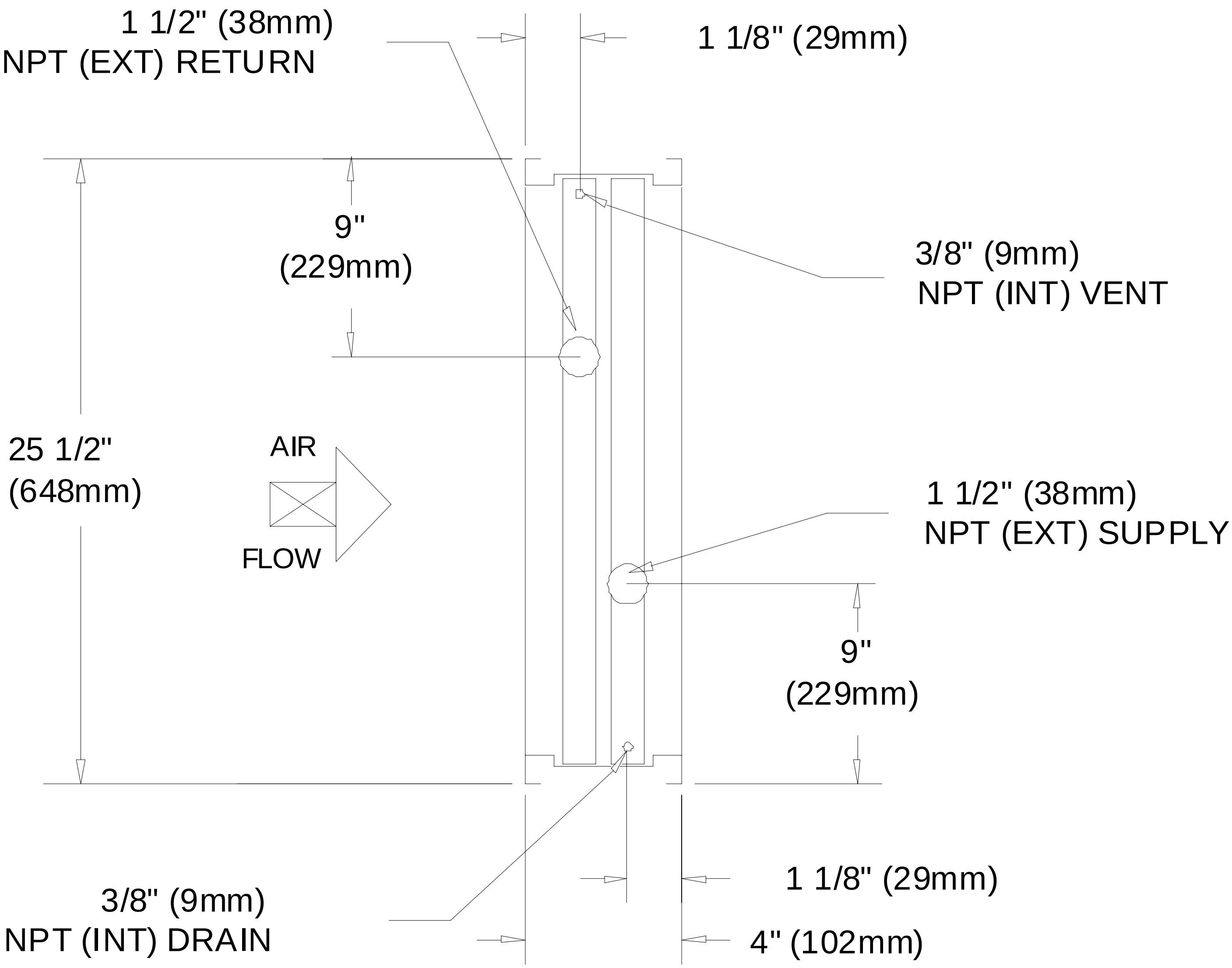
Steam pressure	
Steam PD	
Modulated	
Degrees superheat	
Condensate flow rate	

Stacked Information

Qty of stacked coil #1	
Qty of stacked coil #2	
Qty of stacked coil #3	
Nominal ht stacked coil #1	
Nominal ht stacked coil #2	
Nominal ht stacked coil #3	
Coil bank airflow	29130 cfm
Coil bank total cap	424.50 MBh
Coil bank standard flow rate	86.00 gpm
Coil bank fluid PD	20.45 ft H2O
Coil bank volume	7.45 gal
Coil bank rigging weight	249.2 lb
Coil bank installed weight	311.2 lb

Note: Rated and Certified in accordance with the current edition of ARI Standard 410.

24" 5W 1 ROW


1 1/2" (38mm)
NPT (EXT) RETURN
1 1/8" (29mm)
9"
(229mm)
3/8" (9mm)
NPT (INT) VENT
25 1/2"
(648mm)
AIR
FLOW
1 1/2" (38mm)
NPT (EXT) SUPPLY
9"
(229mm)
1 1/8" (29mm)
3/8" (9mm)
NPT (INT) DRAIN
4" (102mm)

Heating Coil (HTCL)

Job Information

Job01
Richmond

Comments

Model number

D--B33

Coil Information

Tag	AHU-3 Wint	Actual airflow	16865 cfm
Quantity	1	Entering dry bulb	8.00 F
System type	Hot Water	Leaving dry bulb	21.44 F
Unit size	66	Total capacity	245.76 MBh
Coil type	5W	Actual coil face area	28.87 sq ft
Coil tube drainability	Drainable	APD	0.15 in H2O
Rows	1	Volume	4.31 gal
Finned length	126" (3200 mm)	Elevation	0.00 ft
Nominal coil height	33" (838 mm)	Face velocity	584.06 ft/min
Fin type	Prima-Flo H	Max air PD	
Fin material	Aluminum	Max face velocity	
Fin spacing	130 fins per foot	Max fin spacing	
Tube matl/wall thickness	.020 (0.508 mm) copper	Rigging weight	144.3 lb
Coil coating	No	Installed weight	180.2 lb
Turbulators	Yes		

Hot Water Information

Fluid type	Ethylene Glycol
Fluid concentration	30.00 %
Standard fluid flow rate	49.79 gpm
Entering fluid temp	56.00 F
Fluid temp drop	11.23 F
Leaving fluid temp	44.77 F
Fluid PD	20.45 ft H2O
Max fluid PD	
Fluid velocity	5.00 ft/sec
Fouling factor	0.00050 hr-sq ft-deg
Reynolds number	7921.06

Stacked Information

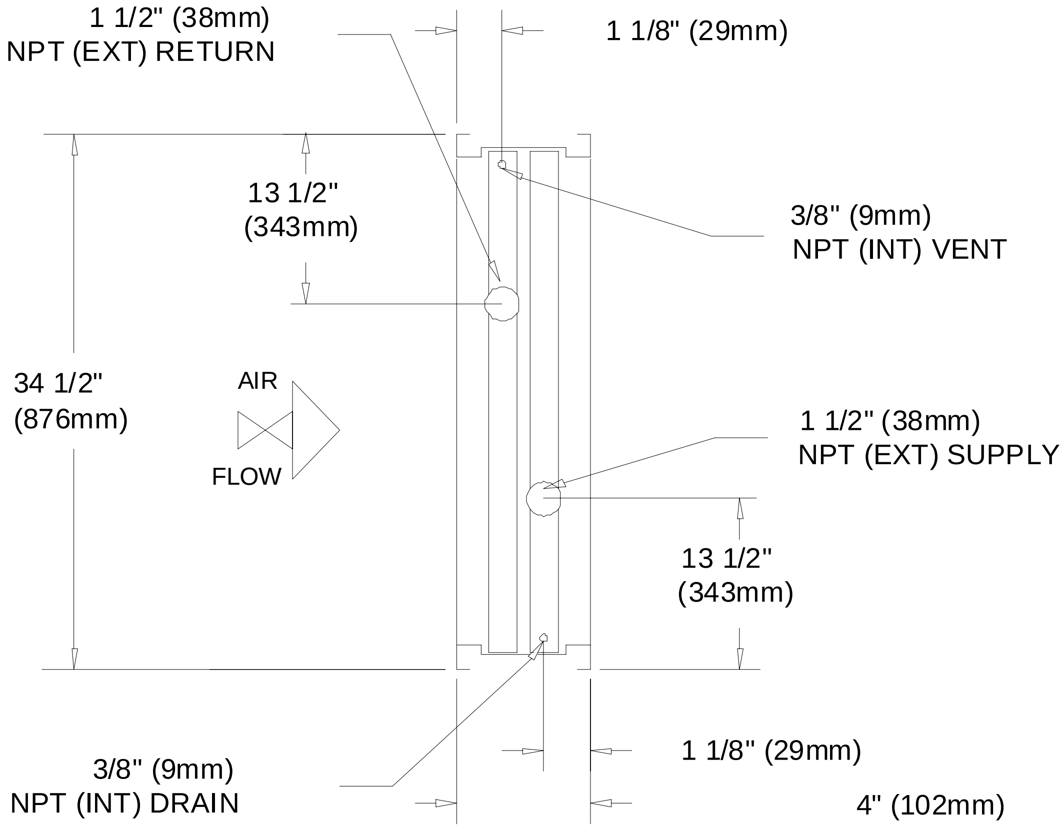
Qty of stacked coil #1	
Qty of stacked coil #2	
Qty of stacked coil #3	
Nominal ht stacked coil #1	
Nominal ht stacked coil #2	
Nominal ht stacked coil #3	
Coil bank airflow	29130 cfm
Coil bank total cap	424.50 MBh
Coil bank standard flow rate	86.00 gpm
Coil bank fluid PD	20.45 ft H2O
Coil bank volume	7.45 gal
Coil bank rigging weight	249.2 lb
Coil bank installed weight	311.2 lb

Steam Information

Steam pressure	
Steam PD	
Modulated	
Degrees superheat	
Condensate flow rate	

Note: Rated and Certified in accordance with the current edition of ARI Standard 410.

33" 5W 1 ROW


1 1/2" (38mm)
NPT (EXT) RETURN
1 1/8" (29mm)
13 1/2"
(343mm)
3/8" (9mm)
NPT (INT) VENT
34 1/2"
(876mm)
AIR
FLOW
1 1/2" (38mm)
NPT (EXT) SUPPLY
13 1/2"
(343mm)
1 1/8" (29mm)
3/8" (9mm)
NPT (INT) DRAIN
4" (102mm)

Appendix B

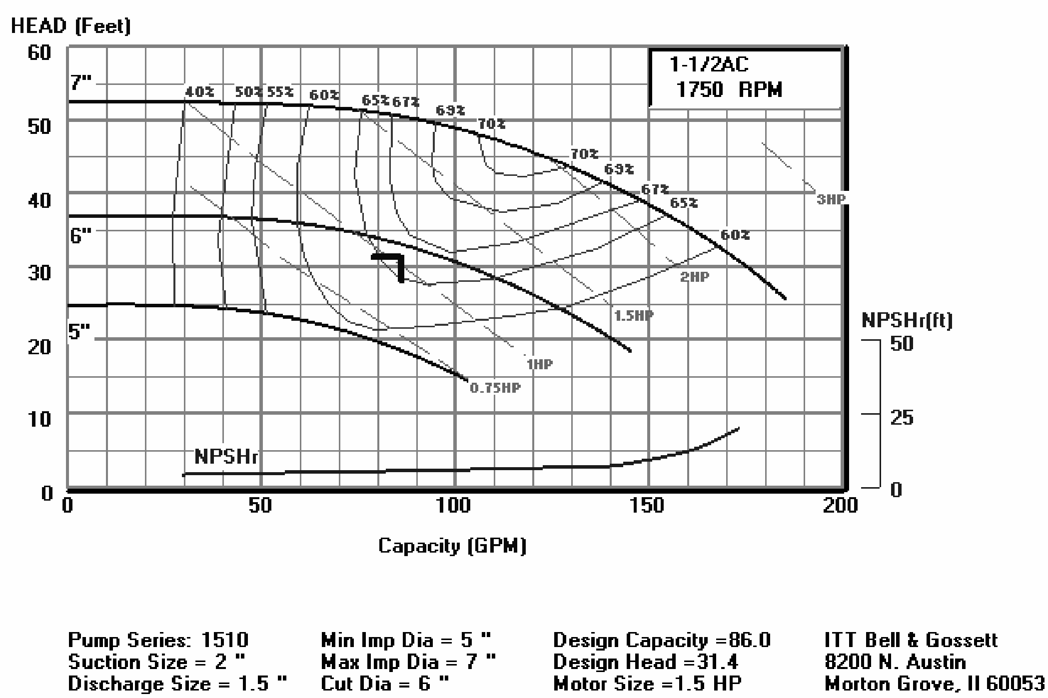
PUMP FOR HEAT RECOVERY SYSTEM

1510 1-1/2AC			
Flow Rate (GPM)	86	Pump Head (Feet)	31.45
Speed (RPM)	1750	NPSHr (Feet)	5.3
Weight (lbs)	170	Cost Index	100%
Suction Size (in.)	2	Suction Velocity (fps)	8.2
Discharge Size (in.)	1-1/2	Discharge Velocity (fps)	13.6
Impeller Size (in.)	6.0	Pump Efficiency (%)	65.63
Max. Flow (GPM)	145	Duty Flow/Max Flow (%)	59.3
Flow @ BEP (GPM)	94	Min. Rec. Flow (GPM)	23.4
Selected Motor Size (HP)	1-1/2	Selected Motor Size (kw)	1.12
Duty-Point Power (BHP)	1.06	Duty-Point Power (kw)	0.79
Maximum Power (BHP)	1.37	Maximum Power (kw)	1.02
Motor Manufacturer	US Prem Eff	Full Load Amps	2.20
Manufacturer Catalog Number	C077	Full Load Efficiency (%)	85.5
Frame Size	145T	Full Load Power Factor (%)	64.0

View Published Pump Curve   Download CAD Drawing

- Generate Pump Curve
- Print Friendly Format
- Calculate Operating Costs
- eMail This Pump Selection
- Generate Submittal

Select Feature



EXPANSION TANK FOR HEAT RECOVERY SYSTEM

SINCE 1908 wessels company

DIAPHRAGM EXPANSION TANKS

Sizing for Hydronic Heating/Cooling Systems

Job Name: LONGWOOD SCIENCE BLDG. Date: 3/16/05

Job Location FARMVILLE, VA Salesman: —

Contact Name: — Model #: ________

Information Required:

1. Total system water content. 72 gallons
2. Temperature of water when system is filled. (MIN. TEMP.) 45 °F
3. Average maximum operating temperature (MAX. TEMP.) 100 °F
4. Minimum operating pressure 12 psig
5. Maximum operating pressure (10% below relief valve) 30 psig

Model Selection:

6. Enter total system water content. (from line 1. above) 72 gallons
7. Using the expansion factor table, find and enter the expansion factor (30% Ethylene Glycol) 0.0147
8. Multiply line 6 by line 7. Enter expanded water volume. 1.0584 gallons
9. Using acceptance factor table, find and enter the acceptance factor. 0.403
10. Divide line 8 by line 9, enter total tank volume required. 2.7 gallons

Line 8. 1.0584 gallons Expanded Water (acceptance volume)

Line 10. 2.7 gallons total tank volume

Select diaphragm expansion tank

NTA Models must satisfy both lines 8 and 10 above.

NLA Models are selected by gallons only from line 10.

NVA Models are selected by gallons only from line 10.

For large systems, multiple tanks can be manifolded together.

CAUTION: This chart is for water only. For expansion factors for glycol solutions contact the Wessels factory or your local Wessels dealer. ✓

NTA Series Expansion Tank — Wesselect

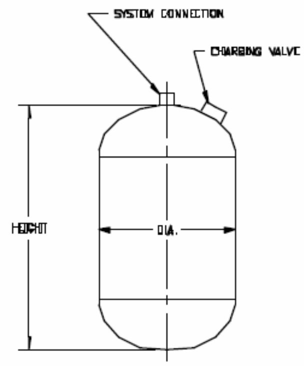
wessels company

SUBMITTAL

Precharged, (125# ASME) Fixed Bladder

JOB	REPRESENTATIVE	
REFERENCE NO.	ORDER NO.	DATE
ENGINEER	SUBMITTED BY	DATE
CONTRACTOR	APPROVED BY	DATE

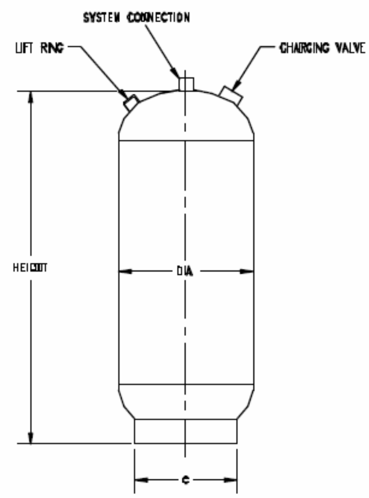
Model No.	Tank Vol. gal.	Accept Vol. gal.	Tank Dimensions inches		Sys. Conn. NPT	Ship Wt. lbs.
			Dia.	Ht.		
NTA 15	7.8	5.0	12	19	1/2	42
NTA 20	10.9	5.0	12	26	1/2	52
NTA 40	25	12	14	42	1/2	84
NTA 60	35	12	14	57	1/2	97
NTA 80	45	24	20	38	1/2	148
NTA 100	60	24	20	49	1/2	175
NTA 120	70	52.5	24	46	1	259
NTA 144	80	52.5	24	49	1	268
NTA 180	90	52.5	24	52	1	283
NTA 200	115	52.5	24	66	1	325
NTA 240	140	52.5	24	78	1	362



NTA 15 and NTA 20

MATERIALS OF CONSTRUCTION	
Shell	Steel
Bladder	Heavy Duty Butyl

MAXIMUM OPERATING CONDITIONS	
Working Temperature	240 ° F
Working Pressure	125 PSIG



NTA 40 thru NTA 240

Typical Specification

Furnish and install, as shown on the plans, Wessels Model NTA-________ ASME Bladder Expansion Tank, stamped 125 PSI working pressure. Each tank will be supplied with a heavy duty butyl diaphragm. Tank shall be supplied with NPT system connection. An air charging valve connection (standard tire valve) shall be provided to facilitate adjusting percharge pressure actual system conditions.

California code-sight glass available on request

DESIGNED AND CONSTRUCTED PER ASME SECTION VIII & STAMPED

Not for domestic Potable Water Systems

MODEL NO. ______________________ **QTY.** ____________

ASME CERTIFICATION REQUIRED __________ **YES** ___________ **NO**

Standard Factory charge is 12 PSIG unless otherwise specified.

Appendix D

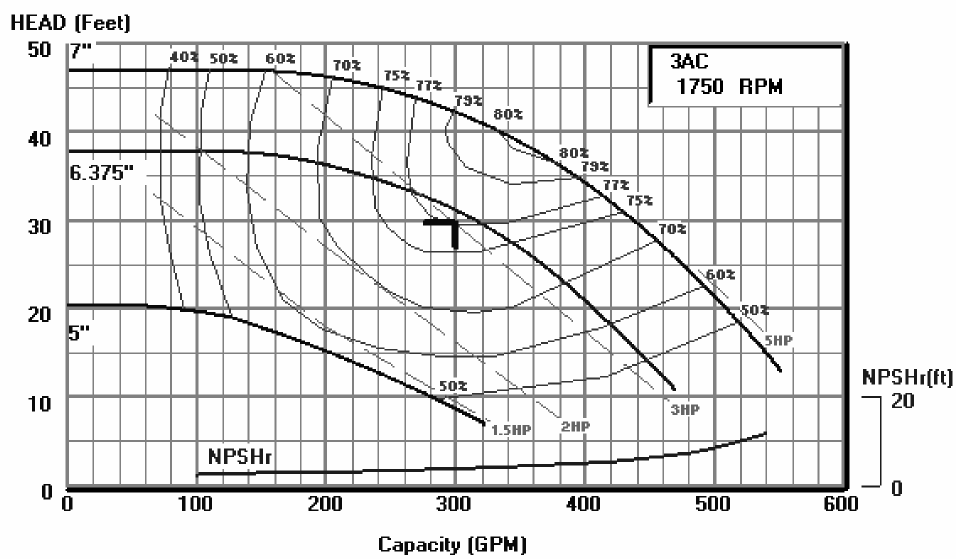
DISTRIBUTED PUMPING PUMP SELECTIONS

AHU-1:

1510 3AC			
Flow Rate (GPM)	299.3	Pump Head (Feet)	29.8
Speed (RPM)	1750	NPSHr (Feet)	3.7
Weight (lbs)	220	Cost Index	100%
Suction Size (in.)	4	Suction Velocity (fps)	7.5
Discharge Size (in.)	3	Discharge Velocity (fps)	13.0
Impeller Size (in.)	6.375	Pump Efficiency (%)	76.96
Max. Flow (GPM)	465	Duty Flow/Max Flow (%)	64.4
Flow @ BEP (GPM)	289	Min. Rec. Flow (GPM)	72.3
Selected Motor Size (HP)	5	Selected Motor Size (kw)	3.73
Duty-Point Power (BHP)	2.95	Duty-Point Power (kw)	2.20
Maximum Power (BHP)	3.19	Maximum Power (kw)	2.38
Motor Manufacturer	US Prem Eff	Full Load Amps	6.50
Manufacturer Catalog Number	F277	Full Load Efficiency (%)	90.4
Frame Size	184T	Full Load Power Factor (%)	74.1

Generate Pump Curve
Print Friendly Format
Calculate Operating Costs
eMail This Pump Selection
Generate Submittal

Select Feature



Pump Series: 1510
Suction Size = 4 "
Discharge Size = 3 "

Min Imp Dia = 5 "
Max Imp Dia = 7 "
Cut Dia = 6.375 "

Design Capacity =299.3
Design Head =29.8
Motor Size =5 HP

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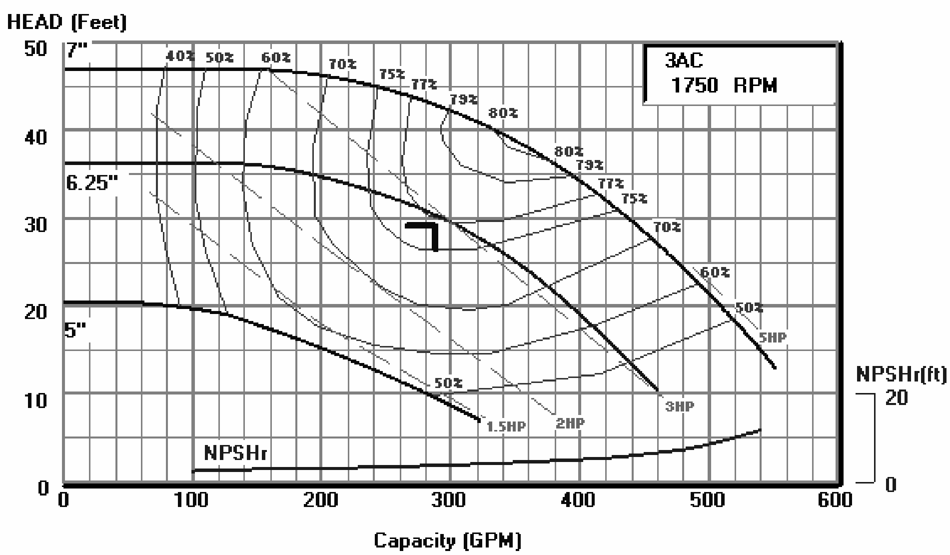
Morton Grove, Il 60053

AHU-2:

1510 3AC			
Flow Rate (GPM)	288.2	Pump Head (Feet)	29.16
Speed (RPM)	1750	NPSHr (Feet)	3.6
Weight (lbs)	210	Cost Index	100%
Suction Size (in.)	4	Suction Velocity (fps)	7.3
Discharge Size (in.)	3	Discharge Velocity (fps)	12.5
Impeller Size (in.)	6.25	Pump Efficiency (%)	76.56
Max. Flow (GPM)	453	Duty Flow/Max Flow (%)	63.6
Flow @ BEP (GPM)	278	Min. Rec. Flow (GPM)	69.5
Selected Motor Size (HP)	3	Selected Motor Size (kw)	2.24
Duty-Point Power (BHP)	2.77	Duty-Point Power (kw)	2.07
Maximum Power (BHP)	2.97	Maximum Power (kw)	2.21
Motor Manufacturer	US Prem Eff	Full Load Amps	4.00
Manufacturer Catalog Number	R333	Full Load Efficiency (%)	90.2
Frame Size	182T	Full Load Power Factor (%)	72.3

◉ Generate Pump Curve ○ Print Friendly Format
○ Calculate Operating Costs ○ eMail This Pump Selection
○ Generate Submittal

Select Feature



Pump Series: 1510
Suction Size = 4 "
Discharge Size = 3 "

Min Imp Dia = 5 "
Max Imp Dia = 7 "
Cut Dia = 6.25 "

Design Capacity =288.2
Design Head =29.2
Motor Size =3 HP

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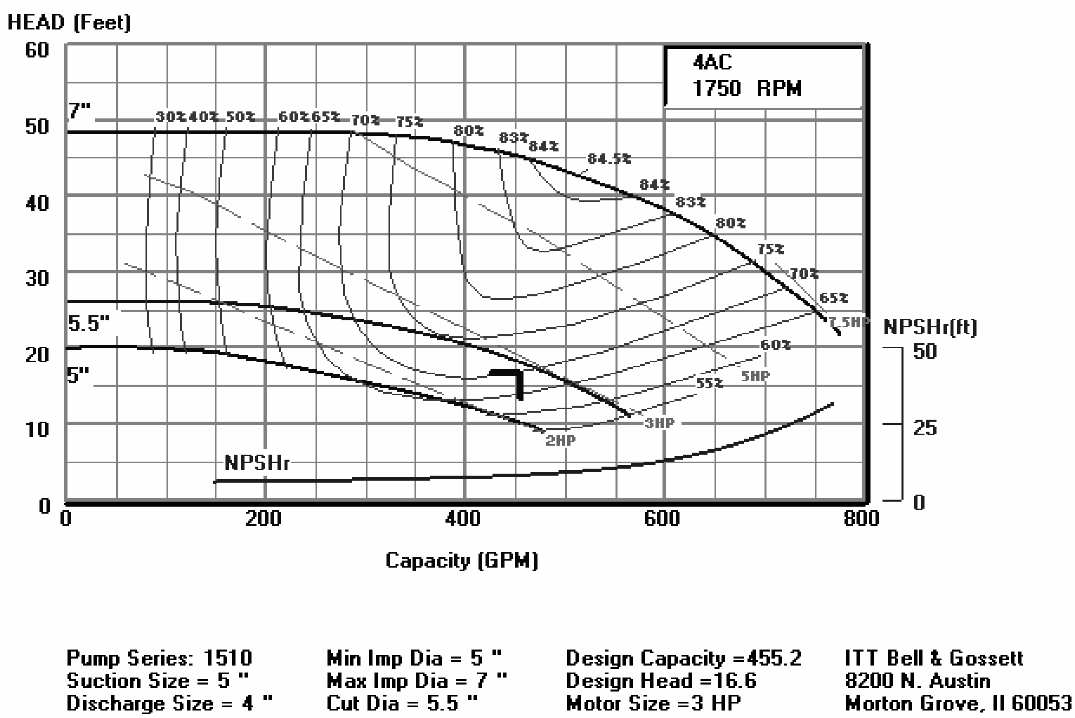
AHU-3:

1510 4AC			
Flow Rate (GPM)	455.2	Pump Head (Feet)	16.56
Speed (RPM)	1750	NPSHr (Feet)	7.9
Weight (lbs)	285	Cost Index	110%
Suction Size (in.)	5	Suction Velocity (fps)	7.3
Discharge Size (in.)	4	Discharge Velocity (fps)	11.5
Impeller Size (in.)	5.5	Pump Efficiency (%)	69.48
Max. Flow (GPM)	566	Duty Flow/Max Flow (%)	80.5
Flow @ BEP (GPM)	374	Min. Rec. Flow (GPM)	93.5
Selected Motor Size (HP)	3	Selected Motor Size (kw)	2.24
Duty-Point Power (BHP)	2.79	Duty-Point Power (kw)	2.08
Maximum Power (BHP)	2.99	Maximum Power (kw)	2.23
Motor Manufacturer	US Prem Eff	Full Load Amps	4.00
Manufacturer Catalog Number	R333	Full Load Efficiency (%)	90.2
Frame Size	182T	Full Load Power Factor (%)	72.3
View Published Pump Curve		Download CAD Drawing	

Generate Pump Curve
Calculate Operating Costs
Generate Submittal
Print Friendly Format
eMail This Pump Selection

Select Feature

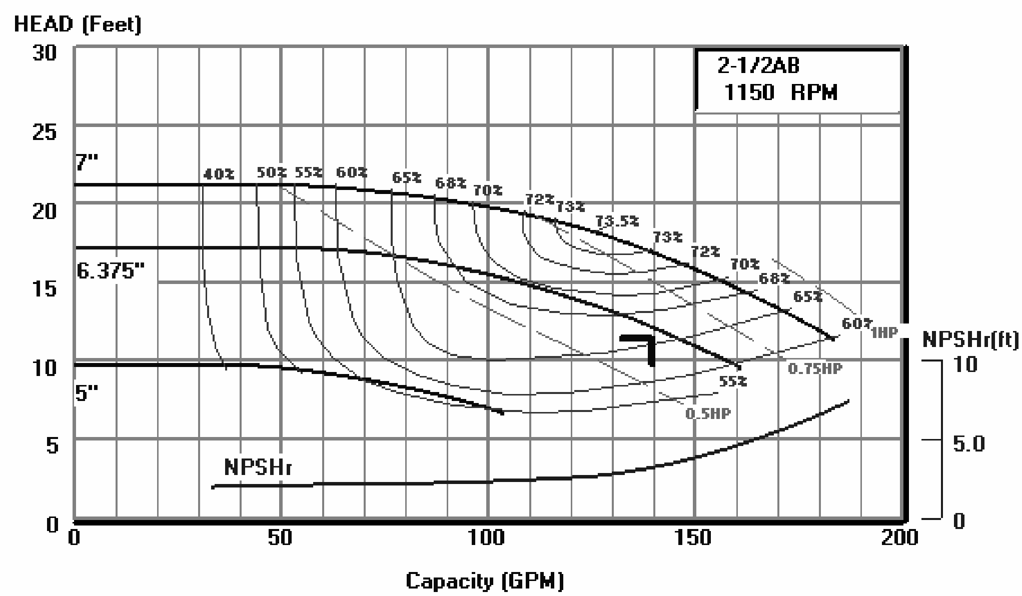


AHU-4:

1510 2-1/2AB			
Flow Rate (GPM)	139.4	Pump Head (Feet)	11.4
Speed (RPM)	1150	NPSHr (Feet)	3.2
Weight (lbs)	183	Cost Index	100%
Suction Size (in.)	3	Suction Velocity (fps)	6.0
Discharge Size (in.)	2-1/2	Discharge Velocity (fps)	9.3
Impeller Size (in.)	6.375	Pump Efficiency (%)	65.54
Max. Flow (GPM)	161	Duty Flow/Max Flow (%)	86.8
Flow @ BEP (GPM)	110	Min. Rec. Flow (GPM)	27.4
Selected Motor Size (HP)	3/4	Selected Motor Size (kw)	0.56
Duty-Point Power (BHP)	0.62	Duty-Point Power (kw)	0.46
Maximum Power (BHP)	0.65	Maximum Power (kw)	0.48
View Published Pump Curve		Download CAD Drawing	

◉ Generate Pump Curve
○ Calculate Operating Costs
Generate Submittal
○ Print Friendly Format
○ eMail This Pump Selection

Select Feature



Pump Series: 1510
Suction Size = 3 "
Discharge Size = 2.5 "

Min Imp Dia = 5 "
Max Imp Dia = 7 "
Cut Dia = 6.375 "

Design Capacity =139.4
Design Head =11.4
Motor Size =.75 HP

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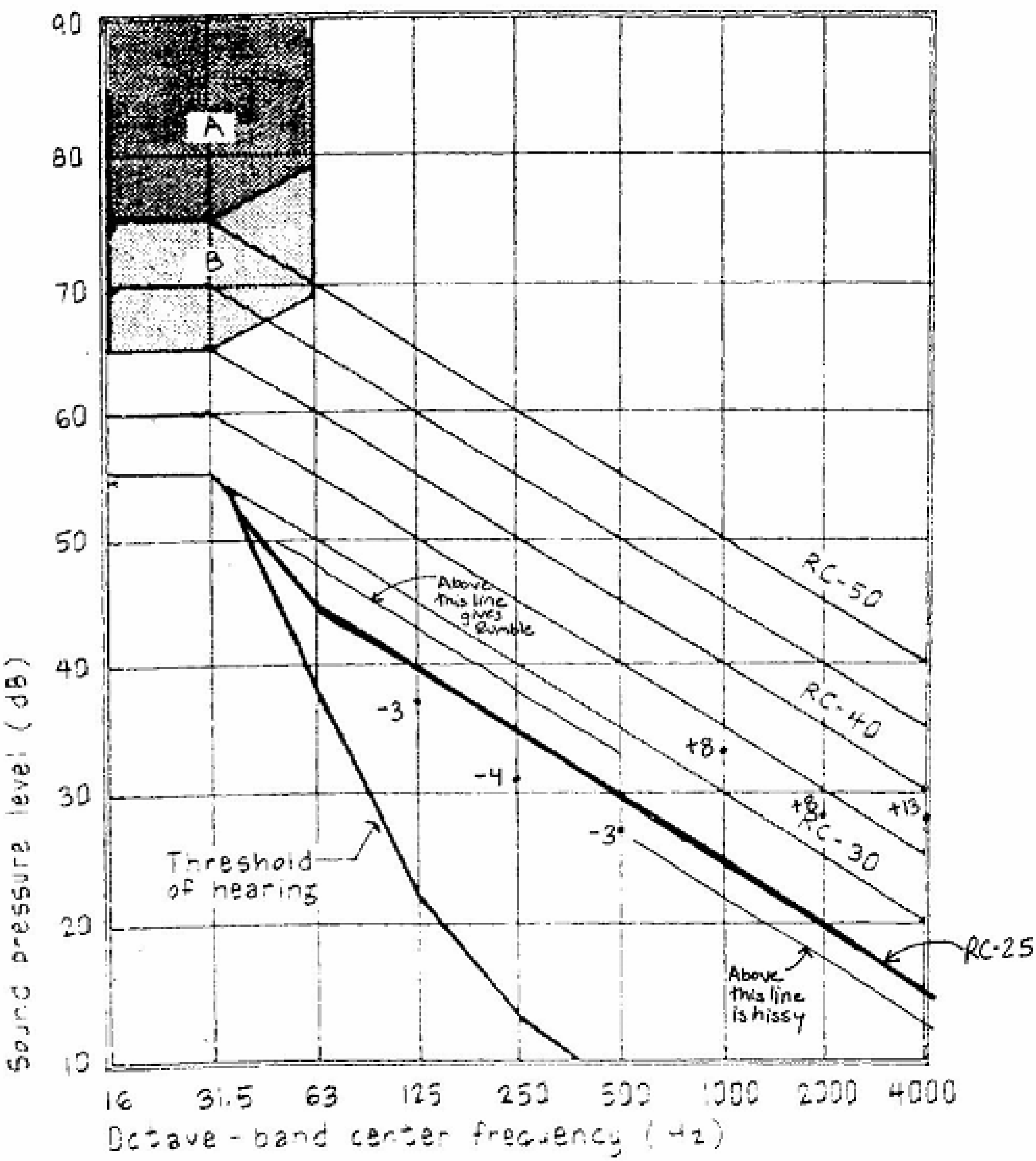
Appendix E

RC-CURVES



42

→ Pts do not fall above "Rumble" line, ∴ There will be no rumble.
→ Pts fall above "hissy" line, ∴ There will be a "hiss."