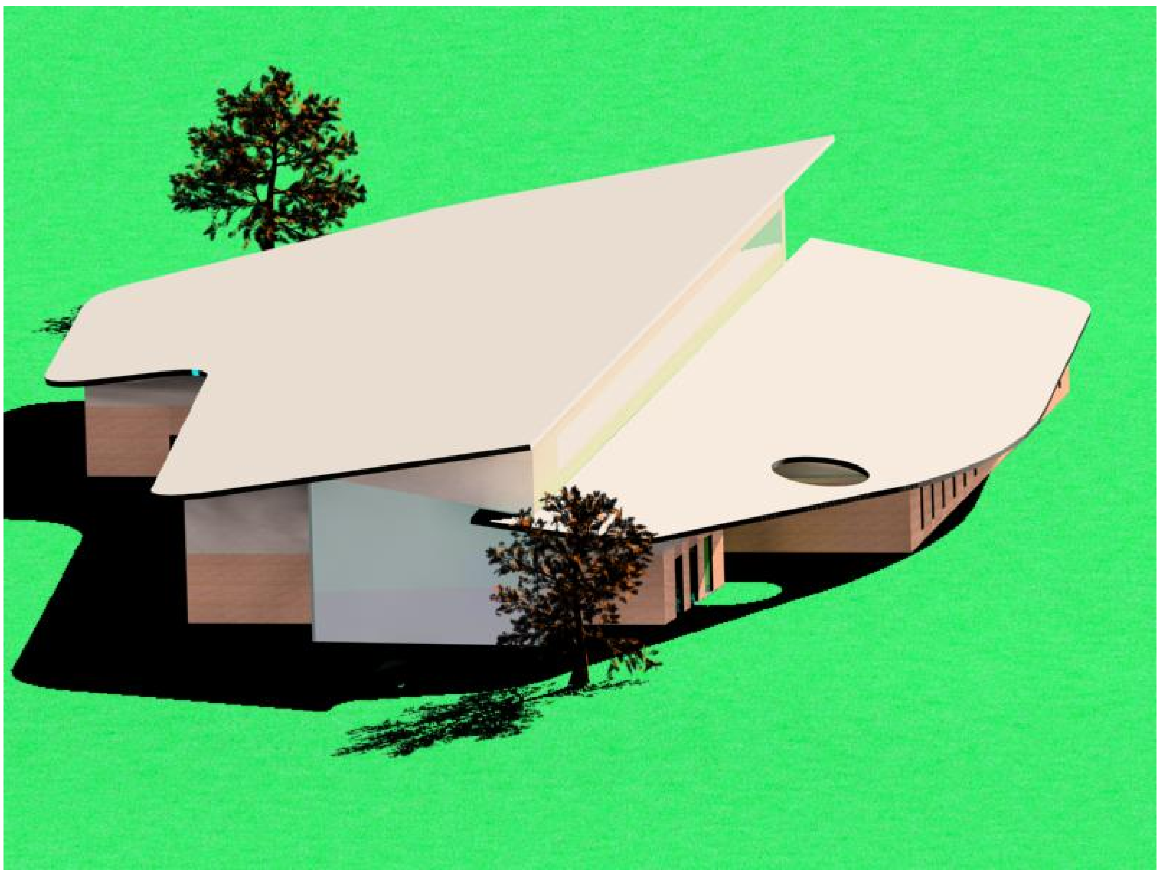
Student:	David M. Potchak	Mechanical Option
Faculty Consultant:	Dr. W. P. Bahnfleth	Spring 2007

DEA Clandestine Laboratory Training Center
Quantico Marine Corps Base, Quantico, VA



Heat Recovery from Laboratory, Classrooms, and Office Spaces

DEA Clandestine Laboratory Training Center

Quantico Marine Corps Base, Quantico, VA

Project Team:

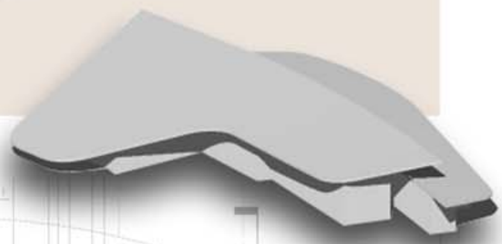
Owner: United States Navy
Occupant: Drug Enforcement Agency Training Academy
A/E Firm: Kling
Contractor: Unknown, Out To Bid
CM: Unknown, Out To Bid

General Data:

Size: 34,152 sq ft
Number of Stories: 1 story plus Mechanical Mezzanine Level
Dates of Construction: October, 2006 – December, 2007
Cost: $10 million (available construction funds)
Project Delivery Method: Invitational Bid for GMP

Electrical:

- (2) parallel 13.2 kV feeders from utility
- 750 kVA main transformer steps down to 480V
- 1,200 A main distribution panel
- Outdoor 230 kW emergency generator powered by #2 fuel oil



Architecture:

- Slab-on-grade masonry building
- Brick veneer accented with courses of CMU
- Large expanses of curtain-wall glazing for natural light
- Curved outline of 2:12 sloped standing seam metal roof
- Low building profile with deep roof over hangs
- Clerestory windows spanning west elevation

Structural:

- Concrete strip footings along exterior walls
- Concrete spread footings under interior columns
- 6" concrete slab on grade reinforced by welded wire fabric
- Steel wide-flange superstructure
- Columns full height from slab to roof
- Typical bay size 34'x28'
- Steel tube cross-bracing and exterior support of roof canopy

Mechanical:

- (5) draw-through AHU's ranging up to 10,880 cfm
  - o (3) VFD and (2) constant volume
  - o (1) 100% OA unit serving analytical lab
  - o (2) economizers with return fan integral to AHU
- (2) 105.5 ton air-cooled chillers serving AHU's
  - o (6) hermetic scroll compressors per chiller, (1) step each
- (3) ACU's serving electrical rooms and LAN equipment room
  - o Up to 2,200 cfm and 34.9 ton
  - o Each with separate air-cooled condensing unit
- (2) 1500 MBH natural gas fired boilers with #2 fuel oil back-up
- Fin tube radiators for sensible heating of exterior zones
  - o Up to 18 MBH with 27' active length
- Cabinet unit heaters in stairs, corridors, vestibules, and showers
  - o Up to 420 cfm and 50.8 MBH

David M. Potchak
Mechanical Option
http://www.arche.psu.edu/thesis/eportfolio/2007/portfolios/DMP287/

Table of Contents

Acknowledgements

Thanks to the following parties for making my thesis possible:

KlingStubbins – Architects, Engineers, Interiors, Planners

In particular, thanks to Ray Doyle, Becky Seitzmeyer, Jeff Brantner, and Mike Schwartz for their assistance and advice.

Penn State AE Mechanical Faculty

Special thanks: Dr. Jae-Weon Jeong – Fall Semester Thesis Advisor
Dr. William P. Bahnfleth – Spring Semester Thesis Advisor

Air Tectonics

In particular, thanks to Rick Galie for his assistance.

Executive Summary

The DEA Clandestine Laboratory Training Center is located on the Quantico Marine Corps Base in Quantico, VA. It is a one-story building with a Mechanical Mezzanine Level that encompasses approximately 34,000 sq ft. The building spaces include multiple function types such as laboratories, classrooms, office space, and physical training areas.

The purpose of this document is to propose a redesign of certain aspects of the Training Center. The main focus of this article is on the redesign of mechanical systems. More specifically, it centers on the analysis of new heat recovery equipment that will salvage waste heat from offices, classrooms, and laboratory spaces. However, two other breadth areas involving the integration of the newly specified equipment into the electrical and structural disciplines will also be discussed. Other alternatives that have been considered for the redesign are covered, as well as the reasons for not selecting those options. The final proposed selections have been analyzed in detail, and the new design is compared and contrasted with the one presently in place. The methods and calculations that were employed in the redesign process are described, and the expected benefits of the proposed changes are discussed.

Project Information

General Building Data:

The DEA Clandestine Laboratory Training Center is located on the DEA Training Academy campus, a sector of the Quantico Marine Corps Base in Quantico, VA. It is a one-story building with a Mechanical Mezzanine Level that encompasses approximately 34,000 sq ft. The building contains spaces of numerous function types including two classrooms, a multi-purpose room, an analytical lab with chemical and lab storage spaces, a mock lab, Firearms Training System (F.A.T.S.) facility, a physical training room, a smoke-filled room, chemical clothing and equipment try-on rooms, a raid training facility, a conference room, a break room, administrative offices, and a glass-wash facility. Supplementary spaces include laundry facilities, showers/restrooms, storage, mechanical and electrical spaces, and an outdoor obstacle area.

Building Use:

The building's main function is to prepare Drug Enforcement Agency trainees for their eventual career. From traditional instruction in classrooms and laboratory spaces to preparation for field work in the raid facility and F.A.T.S. facility, the students receive a thorough education. They are also required to maintain physical fitness via the obstacle areas and the aerobic and weight training rooms.

Architecture:

The Clandestine Laboratory Training Center is a one-story (plus a mechanical mezzanine level), slab-on-grade masonry building designed to be aesthetically compatible with neighboring buildings on the DEA Training Academy campus. By utilizing materials such as the brick, concrete masonry, metal panels, and glazing similar to those already employed on campus, the Laboratory Training Center complements adjacent buildings, but still manages to retain its individuality.

One of the most notable architectural features of the building is the curved outline of its metal roof which forms deep overhanging canopies. Because of the building's relatively low profile, these projections add elements of depth and shadow that are quite evident, while also providing weather protection at entrances. Another prominent feature is a strip of clerestory windows that spans almost the entire length of the west elevation. This stretch of windows is broken in a few places by relief louvers of the same height, flush with the windows to minimize the interruption of the strip effect.

Due to the sloped roof, ceiling heights vary from space to space, with some exterior areas being full-height. Located in an exterior zone, the analytical lab utilizes the generous space between the roof and a drop-ceiling for its extensive exhaust ductwork. Progressing inward, nearing where the roof reaches its peak, a mechanical mezzanine level resides 12’ above the first floor. The raid facility is a full-height space with an observatory catwalk 10’ above the first floor level.

At the northeast corner of the building, the main entrance opens into a lobby which supplies access to the central corridor. Another entrance in the middle of the east elevation also provides access to this corridor, creating a U-shaped circulation system off of which the majority of spaces can be reached.

Building Systems:

Electrical:

The building is served via (2) parallel 13.2 kV utility feeders. The 750 kVA main transformer steps incoming power down to 480V to serve the 1,200 A main distribution panel. Emergency power is provided by a 230 kW outdoor generator that is driven by fuel oil.

Structural:

Foundation: Concrete strip footings provide the base along the exterior walls. Square spread footings ranging from 4 to 12 sq ft support exterior and interior columns.

Ground Floor: The 6” slab on grade is reinforced with 6x6 welded wire fabric. The slab was poured on a vapor retarder atop 6” of compacted porous fill over compacted subgrade. A grid of control joints isolates the slab around each column.

Superstructure: Typical steel columns are W10x33 or 39, with the largest ranging up to W10x60. These extend all the way to the roof at varying heights. Diagonal HSS 8x8x3/8 steel tube columns on the exterior provide support of the roof canopy. The Mechanical Mezzanine is framed by beams ranging from W12x14 to W16x31, supported by girders from W14x22 to W24x76. The typical bay size is 34’x28’. The typical size of roof framing members is W16x26, but others range from W12x14 up to W21x44. Girders supporting the roof range from W18x35 to W27x84. The curved outline of the roof is formed by MC 13x40 steel channels.

Lateral Force Resisting System: The building primarily makes use of a braced-frame system, especially on exterior walls, with tubes from 4”x4” to 7”x7” providing lateral support. Moment-frame connections are used in some places on the interior.

Building Envelope:

Roof System: The roof employs a standing seam metal system with 5” of rigid insulation on a 1½” metal deck. Its two 2:12 slopes are offset to allow for 6’ clerestory windows overlooking a main interior corridor. In a few locations, relief louvers of the same height are aligned flush with the clerestories. The roof slopes terminate at a curved steel channel painted to match the metal roof.

Exterior Wall: A brick masonry veneer backed by concrete masonry and 2” of rigid insulation constitutes the majority of the exterior wall system. This is accented by courses of CMU and large curtain-wall expanses of glazing with aluminum mullions which allow natural light to enter the corridors. Much of the wall under the eaves and in the clerestory section is comprised of metal panels backed either by concrete masonry or 6” cold-formed metal studs with 2’ of rigid insulation.

Existing Mechanical System Summary

<u>Heating System</u>:

Primary Heating System:

Two 1,500 MBH water tube boilers located in the Boiler Room supply 180°F hot water for space heating as well as for heating of domestic and laboratory hot water via a plate and frame heat exchanger. "Water tube" means that the water being heated flows through small tubes which are surrounded by the hot combustion gases in the boiler. Due to the relatively small amount of water in the tubes, this type of system can respond rapidly to changes in load and can decrease the amount of time needed for start-up.

The heating hot water is distributed via two parallel inline centrifugal pumps to the AHU's for primary heating; to unit heaters (UH), cabinet unit heaters (CUH), and finned-tube radiators (FTR) for auxiliary heating; and also to VAV terminal units for reheat. This piping arrangement is called "variable primary flow" because it uses pumps with variable frequency drive to directly adjust the amount of flow to the system. Variable primary flow is thus different than primary-secondary flow, which generally uses constant-flow pumps to circulate the water in a loop around the boilers, and then makes use of valves to control the distribution out to the secondary system. This distribution arrangement is illustrated in the *Heating Hot Water Schematic*.

The majority of the heating load is to be handled by the heating coils located in the air handling units. There is only one heating coil in AHU's 1, 2, and 3. These coils must be capable of heating the air from its design low incoming condition to its intended supply temperature. AHU's 4 and 5 both contain two heating coils, allowing the heating load to be divided between the coils. For entering and leaving air temperatures of the coils, please see the *Air Handling Unit Schedule*. AHU-2 handles the largest load of the heating coils because of the large amount of outdoor air that it must condition at low ambient temperature. Also note that the water side of the system is based on a 20°F drop in temperature across the loads, meaning that a relatively constant temperature of 160°F is returning to the boilers.

With the exception of AHU-5, all air handling units utilize an ultrasonic humidifier to increase the relative humidity that was "lost" across the heating coil. In these humidifiers, a metal diaphragm vibrates at a very high frequency causing condensation to occur.

Auxiliary Heating Systems:

As mentioned above, heating hot water is also piped to unit heaters serving various spaces such as stairs, entry vestibules, corridors, shower areas, and mechanical spaces. These unit heaters range in size up to 50.8 MBH, and all require a supply of around 3 gpm of hot water at design capacity. Like the AHU heating coils, they are also based on a 20°F drop in water temperature. For more information on these components see the *Unit Heater and Cabinet Unit Heater – Hot Water – Schedule*.

Serving exterior zones on the north side of the building are finned-tube radiators. These FTR's provide sensible heating to counteract what would otherwise be a low mean radiant temperature due to the large expanses of glass on that façade. At a total active length of about 73 ft, they offer another 45.5 MBH of auxiliary heating and require roughly 4.5 gpm of hot water.

Many VAV terminal units in the system contain reheat coils which also require hot water. These are typically used when the air at the cooling coil was cooled to a lower temperature than is desired for supply to a space. This is done in order to achieve an acceptable humidity level by condensing some of the moisture out of the air. The reheat coil then adds sensible heat to bring the air back up to its desired supply temperature. Reheat coils are also useful to maintain the desired temperature when certain spaces are unoccupied. The largest reheat coils utilized in this system are capable of over 30 MBH of sensible heat and require about 3.3 gpm of hot water.

Cooling System:

Primary Cooling System:

The Training Center uses a chilled water system to meet the vast majority of its cooling needs. Chilled water containing 30% propylene glycol is cooled via two 105.5 ton chillers. The propylene glycol additive acts as an antifreeze. Propylene glycol was most likely selected because, in the event that it came in contact with potable water, it is generally non-toxic in small concentrations.

The chillers contain air-cooled condensers, so they are located in the utility yard in order to reject heat to the outside. Both chillers contain six hermetic scroll compressors that are electrically driven and handle one step of compression each. The refrigerant specified is R407C, a hydrofluorocarbon mixture that has minimal negative effects on the environment. Chilled water leaves the evaporator at 45°F, and is sent via two centrifugal pumps in parallel to the cooling coils in the air handling units.

Like the heating hot water system, the chilled water system also makes use of a variable primary pumping configuration as illustrated in the *Chilled Water Schematic* in the appendices. This means that the pumps must be capable of VFD operation to control the distribution of chilled water to the coils. One challenge of this type of configuration is to keep the chilled water flow rate through the evaporator within the limits specified in the *Air-Cooled Scroll Chiller Schedule*. This is achieved via a modulating control valve in the bypass line (see *Chilled Water Schematic*) that maintains the correct flow through the evaporator despite the varying load conditions.

All five air handling units contain cooling coils that remove both latent and sensible heat from the supply air. As with the heating coils, the cooling load on AHU-2 coil is the greatest due to the large amount of outdoor air being conditioned from high ambient temperature and humidity levels. The water side of the system is based on a 10°F rise in temperature through the coils, meaning that a near constant temperature of 55°F returns to the chiller. From the *Air Handling Unit Schedule* you'll notice the leaving air temperature of the coils is generally a few degrees cooler than typically supplied in these types of systems. However, this value is the temperature leaving the coil and has not yet accounted for heat gain from the AHU draw-through supply fans. This, in combination with the reheat capability at VAV terminal units, explains for the apparent low leaving air temperature.

Auxiliary Cooling System:

The lone component type that provides cooling but does not does not use chilled water is the Air Conditioning Unit (ACU). This equipment supplies specially conditioned transfer air to electric rooms and LAN closets. Instead of using chilled water, the ACU's use the refrigerant itself, R407C, to extract heat from the space. This is known as the direct expansion (DX) technique. Like the chillers, the ACU's compressors are electrical and the condensers are air-cooled. Having an air-cooled condenser mandates that the refrigerant must be piped to remote condensing units to reject the heat to the outdoors. These remote units are the Air-Cooled Condensing Units (ACCU), and this type of configuration is called a "split system." For more detailed information on ACU's and ACCU's see the *Air-Conditioning Unit Schedule* and the *Air-Cooled Condensing Unit Schedule*.

Air Handling:

Air Handling Units:

Five air handling units located in the Mechanical Mezzanine Level supply the building with conditioned air. AHU's 1 through 5 serve Classrooms, the Analytical Lab, Offices, Raid Facility, and Smokehouse, respectively. AHU-2 is a 100% OA unit serving the labs. Outdoor air is ducted to the AHU's from an intake louver in the mechanical room. AHU's 1 and 3 are capable of economizer mode, in which outdoor air meeting temperature and humidity requirements is supplied directly to the space. Return air is routed back to the AHU's through the plenum using transfer ducts and some longer duct runs where necessary. The Mechanical Room is both ventilated and pressurized by relief air from AHU's 1 and 3 which then is forced out of relief louvers that are ducted from the Mechanical Room.

Fans:

All AHU's utilize draw-through centrifugal supply fans. Supply fans in AHU's 1, 2, and 3 are capable of VFD operation, while AHU's 4 and 5 are constant volume. Integral to

AHU's 1 and 3 are centrifugal return fans, both capable of VFD, recirculating air from offices and classroom areas. A ducted, constant volume, inline centrifugal return fan recirculates air for AHU-5 or exhausts it depending on the operating mode of the Smokehouse. AHU-4 relies on the supply fan to draw sufficient return air back for recirculation. See the *Air Handling Unit Schematics* for a better illustration of flow to and from the spaces.

Mixed flow, induced dilution exhaust fans in the lab (EF-1,2,3) purge the space of contaminant-ridden air through large fumehoods. EF-4 exhausts air from toilet rooms and janitor closets. The Boiler Room is supplied OA for both ventilation and combustion air via a constant volume inline centrifugal fan. See the *Fan Schedule* for more detailed information.

Design Objectives and Requirements

The following sections present the various criteria used in the design of the building's mechanical system. This data includes both assumptions of outdoor design conditions and intended interior space conditions. Estimates of interior thermal loads to be handled by the mechanical system and several other factors are also outlined here. The systems are then developed under these fundamental constraints.

Ambient Weather Criteria:

Outdoor design conditions used were those specified in ASHRAE Fundamentals 2005 as 0.4% occurrence in Richmond, VA.

Ambient Winter Conditions:

T_{DB}	%RH	V_{WIND}*
16°F	50	6 mph

*340° prevailing direction

Ambient Summer Conditions:

T_{DB}	T_{WB}*	V_{WIND}**
94°F	77°F	5 mph

*Mean coincident wet-bulb with design dry-bulb
**230° prevailing direction

Interior Space Temperature and Humidity Criteria:

Office and Support Area Conditions:

max T_{DB}*	min T_{DB}*	max %RH	min %RH	max T_{DB}**	min T_{DB}**
75°F	72°F	50	30	85	60

*Occupied condition **Unoccupied condition

Laboratory Areas:

max T_{DB}*	min T_{DB}*	max %RH	min %RH	max T_{DB}**	min T_{DB}**
72°F	70°F	50	30	75	72

*Occupied condition **Unoccupied condition

Thermostatic Zone Criteria:

This section outlines criteria that were used in the designation of thermostatic zones. Each of the following spaces was designated its own thermostatic zone: Break Room, Lobbies, Laundry Room, Physical Training, Chemical Clothing Try-on, Equipment Try-on, Lab Equipment Storage, Firearms Training System (F.A.T.S.) facility, and the Smokehouse. No more than one of the following types of spaces was allocated to a single thermostatic zone: Laboratories, Mock Lab, Raid Facility, Classrooms, and

Conference Rooms. Open Interior Offices were limited to 3,000 ft^2 per zone, and Open Exterior Offices were limited to 1,500 ft^2 per zone of similar exposure. No more than four Closed Offices were permitted in a single zone.

Ventilation Criteria:

The design of the mechanical system was intended to meet the requirements of ASHRAE Standard 62. Listed below are the objective air change rates used in the design.

Minimum Air Changes Per Hour	
Offices	Labs
2	6

The design and location of air intakes and discharges were required to be compliant with the Anti-Terrorism Force Protection (AT/FP) standards as mandated by Naval Facilities Engineering Command (NAVFAC).

Internal Thermal Load Criteria:

This section outlines estimates of internal loads used in the calculation of peak demand loads and in energy analyses. In offices and support spaces, equipment loads were assumed to be 2 W/ft^2, while the lighting load was calculated from the actual fixtures present. To account for diversity, the population density for office areas was estimated at 140 ft^2 per person. In the laboratory areas, equipment loads were estimated at 6 W/ft^2, and the lighting load was assumed to be roughly 3 W/ft^2. The population of the lab areas was set at 49 people.

Pressurization Criteria:

All spaces were designed to be positively pressured relative to the outside environment with the exception of lab areas. In order to ensure that no harmful substances present in the lab leak into adjacent spaces, the labs were designed for negative relative pressure. This was achieved by way of large exhaust fume hoods, as evident in the *Laboratory Airflow Table* in the appendices.

Building Operating Schedule:

Office spaces are intended to operate 12 hours per day for 5 days each week. Laboratory spaces are intended to operate 24 hours per day, 7 days a week.

Acoustical Criteria:

The mechanical system has been designed to meet the following Noise Criteria levels, excluding occupant-generated noise and noise generated by equipment within the spaces in question:

Space	NC Rating
Enclosed Offices	35
Open Office Area	40
Conference Room	35
Classrooms	35
Laboratories	55
Support Spaces	55

Reliability of Systems:

One of the major intentions of the mechanical system design was to ensure reliability. In the event of a component failure or other type of emergency situation, the remaining components were designed to be capable of maintaining a certain capacity. These capacities are outlined below.

Reliability Requirements:*

Component	min % Capacity
Boiler	50
Heating HTW Pump	50
Chiller	75
CHW Pump	50
Lab Exhaust Fan	66

*In the event of a component failure, the remaining components must be capable of providing the % of design capacity listed here.

Fuel Requirements:

Both boilers and domestic water heaters were chosen with the objective of having dual-fuel capabilities. The boilers selected can be run on natural gas or #2 fuel oil. The design intent was to supply fuel via an existing storage tank located on the DEA Training Academy campus. The Clandestine Laboratory Training Center's generator was also intended to be fired by #2 fuel oil.

Energy Sources and Utility Rates

The major sources of energy utilized by the Training Center are natural gas for heating; #2 fuel oil for back-up heating as well as for the emergency generator; and electricity for cooling, illumination, and to drive many internal activities and pieces of equipment. Actual electricity rates as consumed by the building were not able to be determined because construction has not been completed. Therefore, electric rates were estimated to be similar to those of Virginia Electric and Power Company, a utility company which serves Prince William County, VA, where the DEA campus is located. The assumed rates were selected by comparison with rates charged to buildings having power usage and demand similar to the projected power use of the Training Center. For the entire electricity rate structure, see the *Virginia Electric and Power Company* appendix.

The cost of natural gas was estimated from the rates paid by a neighboring building on the DEA Training Academy campus, and was set at $13.79 per million Btu. When the demand for natural gas exceeds what the supplier is capable of producing, the supplier can "cut off" the supply of natural gas. In this case, the boilers will be fired by fuel oil purchased in bulk through the Marine Corps. When in need of emergency power, the generator will also run off of fuel oil. For simplicity, the energy simulations in the mechanical redesign sections assumed no emergency situations and an unlimited supply of natural gas from the supplier, so #2 fuel oil was not included in the economic analysis. However, by looking at a fuel oil purchase made by the neighboring Justice Training Center building from October 2004 through September 2005 (7,500 gal for $8,626.15) and assuming no inflation from that period, the cost of fuel oil can be calculated to be roughly $ 1.15 per gallon.

Mechanical Redesign – Heat Pipe for Laboratory Energy Recovery

Contaminants Present in Laboratory Exhaust:

In the laboratory area, trainees will concoct methamphetamine (meth) for analytical purposes, as well as to learn safe handling and disposal techniques for their future field work. Entering illegal meth labs is dangerous because of the many highly toxic and explosive gases that are created as byproducts of the meth production process, called "cooking." However, many of these byproducts result unintentionally from incomplete mixing of materials, non-uniform temperature distribution, and other imperfections in the cooking process. In a controlled environment such as the Laboratory Training Center, proper techniques will eliminate the production of many of the harmful substances.

A few hazardous contaminants are still likely to be present in the fumehood exhaust, however. These are phosphine gas, anhydrous hydrogen chloride (hydrochloric gas), vaporized methamphetamine, and possibly iodine particles. Some meth particulate (diameter < 0.1 μm) may be present as well, but gaseous meth is more likely to be found then an aerosol form.

Contact with phosphine and hydrochloric gas can trigger respiratory distress, eye and skin irritation, and even death. Upon contact with air, phosphine gas can also be explosive. Iodine particles can cause respiratory distress as well as eye and skin irritation. Initial effects of inhaling vaporized methamphetamine are generally described as euphoric, with the possibility of elevated heart and breathing rates. Prolonged exposure can produce many serious adverse effects such as fever, stroke, heart failure, malnutrition, skin disorders, ulcers, and eventually psychosis.

For these reasons, it is important to completely eliminate the possibility of cross-contamination when selecting heat recovery equipment for the laboratory exhaust. For the safety of the building occupants, the equipment selected should not provide a path for hazardous materials to enter the supply airstream.

Laboratory Heat Recovery Alternatives:

Several techniques were considered to lessen the load at the coil of the 100% OA unit serving the laboratory areas. The membrane, run-around loop, enthalpy wheel, and heat pipe methods considered all recovery energy from the lab exhaust and transfer it to the supply airstream. The other alternative would utilize chilled water to preheat the incoming outdoor air. This was analyzed in greater detail before ultimately selecting the heat pipe as the method of choice. Please see the table below for a breakdown of selection criteria.

Selection Criteria for Lab Heat Recovery Method

	Effectiveness	Latent Recovery	Availability of Use	Prevention of Cross-Contamination	Duct Reconfiguration Required
Membrane	High	Yes	Continuous	No	Yes
Enthalpy Wheel	High	Yes	Continuous	No	Yes
Chilled Water as Preheat	High	No	Limited	Yes	No
Heat Pipe	Good	No	Continuous	Yes	Yes
Run-Around Loop	Moderate	No	Continuous	Yes	No

The pivotal selection criteria have been highlighted in yellow.

Membrane Heat Exchanger:

This air-to-air heat exchanger places a thin, semi-permeable membrane between airstreams. Across this membrane, both sensible heat and latent energy can be transferred. Along with total energy recovery, high effectiveness is another benefit of the membrane heat exchanger. A major concern when using this equipment in a laboratory exhaust application is the possibility of cross-contamination. Chemicals in the exhaust stream, particularly water soluble ones, could pass through the membrane and enter the supply stream, endangering the occupants of the space. Due to the hazardous nature of the contaminants likely to be present in the laboratory exhaust, the possibility of using a membrane heat exchanger was ultimately discarded.

Run-Around Loop:

A popular choice in laboratory exhaust applications, this heat exchanger completely separates the supply and exhaust airstreams. Sensible heat is exchanged between the exhaust air and a glycol and water solution via a coil. This solution is then routed to a coil in the supply stream for the final energy transfer. Double-walled pipes are common in run-around loops to ensure cross-contamination between airstreams is not an issue. Another benefit of the run-around loop is that, because the airstreams do not need to be routed directly past each other, the intake and exhaust louvers can be placed a safe distance apart, limiting the possibility of contaminant re-entry. Possible drawbacks of this method include added pumping energy and only moderate effectiveness levels. For these reasons, the run-around loop was removed from consideration as a lab energy recovery method.

Enthalpy Wheel:

With high effectiveness and the capability of total energy recovery, the enthalpy wheel is a widely used heat recovery device. Generally, the wheel makes use of a desiccant that rotates between airstreams, transferring both sensible and latent energy. According to a study at the University of Minnesota, certain chemicals, especially those that are water-soluble, were observed to be transferred into the supply stream of the wheel. Many believe that adding a purge section to the wheel will sufficiently cleanse the desiccant, thereby preventing contamination of the supply air. Due to the hazardous nature of the chemicals present in the exhaust air and the varying opinions on the effectiveness of the purge section, the enthalpy wheel was eliminated as an option for lab heat recovery.

Chilled Water as Preheat:

The purpose of this section is to investigate the applicability of using chilled water to preheat incoming outdoor air in the DEA Clandestine Laboratory Training Center. Using the *Engineering Equation Solver* (EES) software program, a 100% OA unit serving laboratory areas is evaluated over a range of ambient temperatures. The cooling coil already in place is utilized to preheat the air, and the leaving air temperatures that can be achieved are determined. This is performed on the existing arrangement of chilled water distribution and also on a proposed reconfiguration. These alternatives will be compared to determine which has the most applicability to the building. Expected benefits include lower loads on the chillers and boilers, the elimination or downsizing of heating or reheat coils, decreased pumping energy, and lower supply fan energy. Also, the risk of re-entry of laboratory exhaust contaminants would be low with this technique.

EES input code and output parametric tables and graphs depicting the results of the model are available. However, the discussion that follows was deemed to be more important in determining why this technique was not used in the redesign.

Introduction:

The analysis is centered on using chilled water to preheat incoming outdoor air. The first step in determining the applicability of this technique is to establish the leaving air temperatures that can be attained when using the cooling coil for preheating. This will be applied to AHU-2, the 100% OA unit serving the laboratory areas. In AHU-2, chilled water enters the cooling coil at 45°F. The dry bulb temperature of the air being supplied to the laboratory spaces ranges from around 55°F in cooling mode to near 75°F when heating is required. This means that, any time the ambient air is between its winter design low of 16°F and the temperature of the chilled water at 45°F, the chilled water is capable of preheating the outdoor air to a state nearer to its intended design supply condition.

The discussion above, using design setpoints currently in place, proves that using chilled water as preheat is possible. Whether or not it has a practical application to the DEA Clandestine Laboratory Training Center is now the question. As evidenced by previous energy analysis, simultaneous operation of the chiller and boiler is sometimes required in the current design. In the summer, the chiller and boiler are run concurrently for humidity control (overcooling the outdoor air and then reheating it to the correct supply temperature). This reheat is also necessary to ensure space temperature control of unoccupied zones (the air handling unit is providing cold air for space cooling of occupied zones, and the reheat maintains proper temperature in unoccupied zones). In the winter months, however, heating of exterior zones near large expanses of glass (such as those served by finned-tube radiators) will be needed even when interior zones require cooling due to equipment and occupant loads. This is where the utilization of chilled water as preheat has potential merit: chilled water is already being produced for cooling of interior zones; and the incoming outdoor air is at a relatively low temperature, in perfect position to benefit from this "free" preheating.

Two different system configurations, each making use of the technique described above, will be considered in the redesign. The first, a "parallel" configuration, is illustrated in the *Parallel System Option* schematic ("parallel" describes the chilled water distribution to the air handling units). In this arrangement, chilled water supply leaves the chiller and splits, one branch serving AHU-2 and the rest serving the other four air handling units in parallel. This is similar to how chilled water is distributed in the existing design. When conditions are right for preheating, the chilled water serving AHU-2 will reject some heat to the colder outdoor air. The branches through the other AHU's, when performing their typical cooling duties, will return at 55°F based on a 10°F temperature drop across the coils. When the returning branches of chilled water mix, the resulting temperature may be very close to that of the 45°F design supply temperature. In this case, little or no cooling energy will be needed to maintain the chilled water supply temperature. At times, it may even be possible to simply circulate the chilled water without running the chiller at all.

The second possibility to be considered is the "series" configuration, illustrated in the *Series System Option* schematic. In this arrangement, the chilled water is first supplied to AHU's 1-4 in parallel. The leaving temperature of the water from these coils will be 55°F. Some of this warmer water will then supplied to AHU-2, providing a greater preheating potential than in the parallel arrangement. This configuration may also allow for preheating to take place at higher ambient temperatures than would be possible in parallel. However, unless a large chilled water temperature drop was achieved across the preheat coil, this arrangement would most likely not provide cool enough return temperatures to allow the chiller to shut off completely, as may be possible in the parallel arrangement.

After being preheated, the outdoor air passing through AHU-2 will experience a heat gain of several more degrees via the draw-through centrifugal supply fan. When cooling is required, the air may now be near its intended supply temperature, especially if the series configuration is in place. If heating of the laboratory spaces is required, the air is most likely still several degrees away from its intended supply temperature. Here, several options can be considered.

The first option is to keep the reheat coils already in place in the lab areas, but perhaps upsize them slightly to provide the rest of the necessary heating. This means that the heating coil in AHU-2 could possibly be eliminated entirely (as illustrated by the dashed-line heating coil in the schematics). However, if lab spaces require heating and no other interior areas require any cooling, no chilled water would be produced and there would be no preheat. This means that the reheat coils would have to be capable of all the necessary outdoor air heating and may become excessively large, requiring substantial amounts of pumping energy for heating hot water distribution.

Another option is to leave the existing heating coil in the air handling unit and remove the reheat coils at the terminal units. In this case, even if no preheat is performed by chilled water, the heating coil will be capable of meeting the space load. This alternative would require less energy from supply fans by removing the reheat coil pressure drop. The pumping of heating hot water to the reheat coils would be completely eliminated, and pipe runs to the reheat coils would no longer be needed. In addition, this could prevent air-side freezing on the coils that may occur without a heating coil in the unit. However, removal of the reheat coils would necessitate another means of dehumidification in summer months. By selecting equipment that is capable of both humidification and dehumidification, the existing ultrasonic humidifiers would not be needed, and some of the cost of the new equipment could be offset.

Analysis:

This section discusses the different aspects that were included in the simulation. The assumptions, methods used, and calculations that have been performed are explained.

AHU-2:

AHU-2 is capable of variable frequency drive operation. Its maximum supply flow rate of 8,040 cfm will only be required when the fumehoods in the laboratory are exhausting the highest amount needed. For this reason, multiple flow rates were considered, ranging down to the terminal unit minimum at 40% of the upper limit. The effectiveness of the cooling coil in transferring energy to the incoming outdoor air was assumed to be 90% via comparison to coils in the existing air handling units. The leaving air temperature from the coil will be calculated.

Outdoor Air:

The dry bulb temperature of the outdoor air is varied from its design low up to the chilled water supply temperature. Its relative humidity is assumed to be at its design winter condition of 50%.

Chilled Water:

The chilled water is actually a propylene glycol solution at 30% concentration. For this analysis, its volumetric flow rate to the cooling coil (used for preheating) was held constant at 165 gpm. The chilled water supply temperature in the parallel alternative is 45°F because it is coming directly from the chiller. In the series case, the chilled water being supplied to AHU-2 is at 55°F because it is actually "returning" from the other AHU's. In both cases, the chilled water exiting the preheat coil is calculated.

Fan Heat Gain:

After the air has left the preheat coil, the break horsepower of the supply fan is assumed to be transferred to the air stream, causing a rise in its temperature. The air's relative humidity at this state was assumed to be 40%.

Remaining Heating Load:

Assuming that the air is to be raised to its heating supply temperature of 74.2°F, the remaining load that would need to be handled by heating hot water (or another means) has been calculated as a percentage of the initial total outdoor air load. The fan heat gain was not included in this calculation, and, if considered, would further decrease the remaining heating load required.

Other Air Handling Units:

The other air handling units are assumed to be performing typical cooling duties in this simulation, and, therefore, the chilled water returning from them is 55°F. The chilled water flow rates through the other cooling coils are held constant for this analysis. The returning chilled water temperature to the chiller is calculated after all returning branches mix.

Discussion:

After passing through the draw-through supply fan, the air temperatures are significantly closer to the intended supply state. As expected, the series arrangement yields the highest air temperatures, saving up to 60% of the total heating energy necessary to bring the outdoor air to its supply condition. Including the fan heat gain in the calculation of the remaining heating load would produce even more savings. The parallel configuration, however, has the lowest return temperature of chilled water, which would save chiller energy. Further simulation would be required to determine which alternative would optimize energy savings.

With lower chilled water return temperatures, there is a possibility that the chiller will operate inefficiently when in preheating mode. Correct staging of the equipment is important here. Consideration could be given to reselecting the chillers, particularly looking at the number and sizes of the machines. Currently there are two identical chillers in place, and this may not prove to be the most efficient selection when utilizing the preheating technique.

The highest leaving air temperatures are achieved when outdoor air approaches the temperature of the chilled water. At these conditions, chilled water is transferring little or no heat to the incoming outdoor air. The energy used creating the chilled water and pumping it through the coil must be compared to the benefits attained from a higher leaving air temperature to determine the setpoints that will control when preheat will or will not occur. This would also require further simulation.

The greatest heating energy savings are produced at the lowest ambient temperatures. Looking at how often the temperature is in this lower range is a way to determine the applicability of chilled water as preheat for the Training Center. On average, the design low of 16°F only occurs 0.4% of the year. According to data from *The Weather Channel*'s website, www.weather.com, the average low temperature in January, Quantico's coldest month, is 26°F. This means that, usually, the lowest ambient temperatures used in the analysis will not be present. Therefore, the greatest energy savings attainable from this system will not often be achieved. More often, moderate heating load reductions will be accomplished.

The amount of air being supplied to the space did not affect the results in proportion to the amount that it was varied in the analysis. Supplying more outdoor air lowered the returning chilled water temperature as would be expected. Increased outdoor air also limited the air temperature that could be achieved after the coil and fan. As was mentioned earlier, the supply of air to the lab will not often be at its maximum, so the more favorable results at lower airflow rates would be present for much of the preheating mode duration.

Conclusions:

Judging by the results obtained from this simulation, using chilled water for preheating does have potential merit in the DEA Clandestine Laboratory Training Center. Lower temperature of returning chilled water was achieved which will save chiller energy. Higher leaving air temperature was achieved which will limit the amount of additional heating required. However, the major issue remaining is the percent of the year that ambient temperatures would be conducive to preheating. On average, the ambient outdoor temperatures in Quantico are below 55°F, the series preheat temperature, only about 42% of the year. For this reason, the use of chilled water for preheat was discarded in favor of an alternative with potential benefits for a larger fraction of the year.

Heat Pipe:

Like the run-around loop, this heat exchanger also completely separates the supply and exhaust airstreams, eliminating the possibility of cross-contamination. Sensible energy is transferred from one airstream to another through a device resembling a pipe that is capable of heat transfer with minimal losses. This device is a container, usually made of copper, aluminum, or another material with a high thermal conductivity, holding a fluid that boils readily under normal conditions. When one end of the heat pipe is heated by the warm airstream, the fluid boils and expands, moving to the cooler side of the pipe. Here, the fluid rejects heat to the cooler airstream and condenses, moving back to the warmer side of the pipe to repeat the process.

Many heat pipes work on buoyancy principles alone, and therefore can only function properly when the hot airstream is aligned vertically below the cold airstream. Horizontally configured heat pipes, however, make use of a wick or capillary structure that allows the working fluid to flow between airstreams by utilizing pressure differences in the evaporated and condensed sections of fluid. A horizontally configured heat pipe can then be left in one position year-round, and will be capable of recovering energy in either cooling or heating mode, depending on which is required by the ambient conditions.

Due to its capability to eliminate cross-contamination, its ability to save energy in both cooling and heating modes throughout the year, and its respectable heat recovery effectiveness, the heat pipe has been chosen as the heat recovery method for the laboratory spaces.

Heat Pipe Selection:

Based on the maximum outdoor air intake and exhaust flows of 8,040 cfm, the *Heat Pipe Technology, Inc.* model HRM-6R has been specified. In a horizontal configuration, energy recovery will be possible year-round. A *Fins-coat* coating, similar to an epoxy but less brittle, will be employed to ensure the heat pipe surfaces do not react adversely with contaminants in the exhaust. By including a bypass damper integrated into the exhaust ductwork, it will be possible to perform maintenance on the heat pipe itself without shutting down the laboratory fumehoods or air handling unit. For more information on the model specification, please see the included *Heat Pipe Technology, Inc.* appendices.

Analysis:

The spreadsheet software program *Microsoft Excel* was used to model the annual energy and cost savings of the specified heat pipe when used to recover energy from the exhaust air of the laboratory spaces served by AHU-2. In the sections that follow, the assumptions, methods, and calculations used in the analysis are described. To fully understand the steps of the analysis, the *Sample Calculations* section of this report is to be used in conjunction with this description.

AHU-2:

AHU-2 is capable of variable frequency drive operation. Its maximum supply flow rate of 8,040 cfm will only be required when the fumehoods in the laboratory are exhausting the highest amount needed. This was estimated to occur 2 hours a day in the winter months and 3 hours a day in the summer, because more supply air is required for the peak cooling load than for the peak heating load. The remainder of the year, the supply flow rate was assumed to be equal to the terminal unit minimum at 40% of the upper limit. As shown in the *Sample Calculations* section of this report, a time-weighted average supply and exhaust flow rate was calculated from the discussion above, resulting in a value of 3,560.18 cfm being used as a constant flow rate in the model.

Outdoor Air:

The hourly ambient air dry bulb temperatures used in the model were average values compiled in Richmond, VA, and obtained from a TMY2 weather data file.

Heat Pipe:

The estimated effectiveness of the heat pipe, derived from bin data obtained from the manufacturer (see attached), was assumed to be constant throughout the year. The value of 0.56 was ultimately assumed because it fell near the middle of the equipment's effectiveness range.

Hourly Energy Savings:

As mentioned earlier in the *Design Objectives and Requirements* section, the laboratory areas are meant to be capable of continuous operation throughout the year. For this reason, the hourly cooling and heating data was simply summed to arrive at the yearly values.

Utility Costs:

The average electric rate used in the model was calculated from the total amount of electric energy used by the building in a year and the total cost of the annual bill containing both consumption and demand charges. These estimated figures were obtained from a previous TRANE TRACE energy model performed on the building. For this reason, no separate demand charge was used in this analysis, because the energy consumption of the rest of the building at any given time is not known. The demand charge was, instead, assumed to be accounted for in the average electric cost per kilowatt-hour.

The cost of natural gas was estimated from the rates paid by a neighboring building on the DEA Training Academy campus, and was set at $13.79 per million Btu. Although the boiler is also capable of being fired by fuel oil, non-standard operation was not accounted for in this analysis, so the cost of fuel oil was not included.

Additional Fan Energy:

Additional fan energy will be required to account for the added pressure drop of the new heat pipe. The maximum pressure drops across the new equipment in both the supply and exhaust airstreams were used to calculate the extra electrical energy that will be necessary.

Life Cycle Cost:

As per the current standard ASHRAE life cycle cost analysis, the annual heat pipe savings were evaluated as a present value from a 20-year period assuming a fixed 5% interest rate. Inflation and maintenance costs were not considered in this analysis. The initial cost is assumed to include installation.

Results:

The annual cooling and heating energy savings are shown below:

Cooling Q (Btu/yr)	Heating Q (Btu/yr)
17,839,836.25	166,624,930.17

After considering the efficiencies of the chiller and boiler, the actual energy savings as they pertain to the equipment electrical and fuel input are shown below:

$W_{CHILLER}$ (kWh/yr)	W_{BOILER} (Btu/yr)
1,982.20	**204,447,767.08**

The additional fan energy results in more electrical energy being required as shown below:

Fan Energy (kWh/yr)
5,905.49

The additional fan energy required, it appears, is actually greater than the amount of electricity saved by lessening the chiller load. In other words, the inclusion of the heat pipe has caused more electricity to be used than was previously. However, a significant amount of heating energy has been saved, offsetting the additional electrical cost. These findings depicted below show that, overall, the heat pipe produces significant energy and cost savings annually.

W_{ELECT} (kWh/yr)	$W_{NAT.\ GAS}$ (Btu/yr)
-3,923.29	**204,447,767.08**

Electrical Savings	Natural Gas Savings
-$319.36	**$2,819.33**

Total Annual Savings
$2,499.98

The present value is a measure of the worth of purchasing and utilizing the heat pipe over the 20-year period analyzed. Calculated in the life cycle cost analysis, the present value was found to be $49,992.05. When compared to the initial cost of the heat pipe at $1,120, the final present value of the heat pipe can be set at $48,872.05. It is obvious that the heat pipe specification was a good economic decision as well as an energy-conscious one.

Conclusions and Discussion:

Several key assumptions and estimations played significant roles in determining the final savings outcomes. The first and most obvious estimation is the average weather data that was used in the analysis. Although average data will provide a fairly good indication of the energy savings that the heat pipe would produce over a number of years, it is not necessarily a good indicator of savings over a one-year period, where high and low temperature conditions are likely to stray from the average.

Assigning the supply and exhaust flow rates a constant value throughout the year proved to be another pivotal assumption. Perhaps a better estimation technique would be to weight the flow rates according to the outdoor temperatures. By increasing the amount of air introduced as temperatures become more extreme in either direction, this would better represent how the air handling unit responds to increased heating and cooling loads. However, this technique would further complicate the model. Another reason for not employing the temperature-weighted estimation is that the occurrences and durations of maximum fumehood exhaust would still not be known. Due to the relative simplicity of using a constant, time-weighted, average flow rate, this method was chosen. The constant value made the calculation of fan energy much simpler as well.

Along with supply and exhaust flow rates, the effectiveness of the recovery device will, in reality, also vary with temperature and other weather conditions. As mentioned earlier, the effectiveness was assumed using bin data obtained from the manufacturer.

Heating savings far outweighed cooling savings, even before the additional fan energy made the numbers more lopsided. As was discussed earlier, if the flow rate had been further increased in the summer months, the results, particularly the cooling savings, would surely be different. However, increasing the flow rate would also increase the fan energy required. Without further analysis, it is not clear whether these changes would result in increased electrical savings.

Consideration should also be given to whether the existing exhaust fans can handle the added pressure drop of the new heat pipe. The existing Strobic Tri-Stack Exhaust Fans, model BS00518, are capable of meeting a pressure drop of 5.1 in wg. The pressure drop already in the exhaust system, taken from design documents, is 2.65 in wg. An additional 0.72 in wg from the heat pipe brings the total to 3.37 in wg, meaning that the present exhaust fans are capable of providing adequate pressure even with the new equipment.

After a first glance at the energy savings attained from the heat pipe, one may consider downsizing the equipment such as the chiller and boiler, believing that they will no longer have to meet the peak loads as before. However, if the heat pipe requires maintenance and the air is routed around by the bypass damper, the same energy savings will not be present. In these cases, it would beneficial to maintain the peak capabilities that the present equipment has.

One major factor that has been not been included in this analysis is the duct reconfiguration required to route the outdoor air intake past the exhaust to make the heat exchange possible. If the new supply duct path is longer than the existing one, additional first cost for the extra length of duct would need to be included to finalize the economic analysis. Even if the new configuration is approximately the same length as the existing path, the location of the intake and exhaust louvers is now another potential problem. A computational fluid dynamic study in combination with possible wind directions and speeds would be necessary to ensure that re-entry of exhaust contaminants will not pose a threat. This study was outside the scope of this report.

The heat pipe, to be situated atop an existing mechanical equipment support structure, will bear on newly specified dunnage beams. For an analysis of the structural integrity of the new and existing members intended to support the new heat pipe, please see the *Structural Breadth Redesign* section of this report.

Mechanical Redesign – Enthalpy Wheel for Classrooms and Offices

As discussed earlier, the enthalpy wheel is a widely used air-to-air heat recovery device that has the capability of total energy recovery. Its relatively high effectiveness in both sensible and latent energy recovery makes it an attractive choice for many types of applications. While the wheel was eliminated as an option for lab heat recovery due to the hazardous nature of the contaminants in the exhaust, it is able to be used for heat recovery from the relief air of classrooms and offices.

The air handling units that serve the classrooms and offices, AHU-1 and AHU-3, are both capable of variable frequency operation, and both recirculate a large portion of the return air under normal operating conditions. This recirculated air returning to the air handling units has already been conditioned, and large amounts of energy would otherwise be required to condition an equal amount of outdoor air to the supply conditions. The laboratory spaces, on the other hand, were not permitted to recirculate air due to the fact that contaminants would be re-introduced to the space. Under normal operating conditions, both AHU-1 and AHU-3 introduce only the minimum amount of outdoor air to meet ventilation requirements, 2,280 cfm and 1,310 cfm, respectively. With the air handling units sized at a total of 9,420 cfm and 10,880 cfm, this means that 7,140 cfm and 9,570 cfm are usually recirculated.

Both air handling units are, however, capable of economizer mode. This means that, when ambient conditions are right, the amount of outdoor air being introduced is allowed to modulate up. The amount of return air being recirculated also modulates, mixing with the outdoor air until the proper space condition is met. In the current design, economizer mode is enabled when the outdoor air dry bulb temperature is below 70°F; the outdoor air enthalpy is less than the return air enthalpy; and the outdoor air dew point temperature is above 39°F. Under these ambient conditions, the wheel will stop rotating, saving energy and allowing for the "free" heating or cooling that the economizer mode is meant to provide.

Enthalpy Wheel Selection:

The *AIRotor*, model AHR-1600 HRW RVA-Hy-R-0-1A, by *Xetex, Inc.,* has been selected to recovery energy from the relief air of the classrooms and offices. The model numbers and letters signify that the airstreams will be horizontally oriented side by side; the rotor will be hygroscopic ($\epsilon_{SENSIBLE} = \epsilon_{LATENT}$); the unit will be capable of electric speed control to prevent frost accumulation; and the unit will not contain a purge section.

By including a bypass damper integrated into the relief ductwork, it will be possible to perform maintenance on the enthalpy wheel when needed without shutting down the air handling units. The bypass also makes it possible to size the wheel only for the required minimum ventilation air of the two air handling units, totaled at 3,590 cfm. When more outdoor air is introduced in economizer mode, the pressure drop through the wheel may become quite large if this excess air were not routed around the wheel. If the bypass damper were not present, the wheel would have to be sized much larger to handle the additional air, driving up both initial and operating costs.

It is possible to serve both air handling units through the same wheel because their outdoor air intakes are drawn through the same louver, as is the relief air through the relief louver. This means that minimum duct reconfiguration will be necessary to accommodate the new wheel. For more information on the model specification, please see the included *AIRotor* appendices.

Analysis:

As with the heat pipe analysis, the spreadsheet software program *Microsoft Excel* was used to model the annual energy and cost savings of the specified enthalpy wheel when used to recover energy from the relief air of the classrooms served by AHU-1 and the offices served by AHU-3. In the sections that follow, the assumptions, methods, and calculations used in the analysis are described. To fully understand the steps of the analysis, the *Sample Calculations* section of this report is to be used in conjunction with this description.

Operating Schedule:

Unlike the laboratory spaces, the classrooms and offices are designed to be in use 12 hours per day, 5 days per week. Given an hour of warm-up time, the air handling units serving these spaces were assumed to operate between 6:00am and 7:00pm during the weekdays. For the analysis, the enthalpy wheel was assumed to operate during these hours except when the economizer mode is enabled.

AHU's and Economizer Mode:

As discussed previously, the wheel has been sized to handle the total minimum outdoor air requirement of the two air handling units, 3,590 cfm. This value is assumed constant throughout the analysis except when in economizer mode. In this case, the wheel is shut off, and no energy savings is achieved via the wheel. Therefore, it is not vital to know the outdoor and relief air flow rates in economizer mode. This is because the objective of the analysis is to determine savings from the installation of the wheel—not from the existing economizer mode. For the same reason, the economizer mode logic used in this analysis does not differ from that of the current design.

Outdoor Air:

The hourly ambient air dry bulb temperatures used in the model were average values compiled in Richmond, VA, and obtained from a TMY2 weather data file. The humidity ratio and enthalpy of the outdoor air were calculated using the *Engineering Equation Solver* (EES) software program from dry bulb, dew point, and atmospheric pressure data that were obtained from the same TMY2 file.

Enthalpy Wheel:

The estimated sensible and latent effectiveness of the enthalpy wheel were taken from charts provided by the manufacturer. These can be found in the attached *AIRotor* appendices. Although the effectiveness will change slightly with outdoor conditions, it was assumed to be a constant value of 0.77 throughout the year.

Utility Costs:

As with the heat pipe model, an average electric rate was used in the analysis. It was calculated from the total amount of electric energy used by the building in a year and the total cost of the annual bill containing both consumption and demand charges. These estimated figures were obtained from a previous TRANE TRACE energy model performed on the building. For this reason, no separate demand charge was used in this analysis, because the energy consumption of the rest of the building at any given time is not known. The demand charge was, instead, assumed to be accounted for in the average electric cost per kilowatt-hour.

The cost of natural gas was estimated from the rates paid by a neighboring building on the DEA Training Academy campus, and was set at $13.79 per million Btu. Although the boiler is also capable of being fired by fuel oil, non-standard operation was not accounted for in this analysis, so the cost of fuel oil was not included.

Electric Savings:

Unlike the heat pipe, the enthalpy wheel is capable of latent energy recovery as well as sensible. For this reason, savings on sensible cooling *and* dehumidification were assumed to contribute to chiller savings. Humidifier savings, together with chiller savings, make up the total amount of electric savings.

Natural Gas Savings:

As with the heat pipe model, all heating energy savings is attributed to the boiler. This is then transferred to natural gas savings after accounting for the boiler efficiency.

Additional Fan Energy:

Similar to the heat pipe, additional fan energy will be required to account for the added pressure drop of the new wheel. The pressure drops across the equipment in both the supply and exhaust airstreams were used to calculate the extra electrical energy that will be necessary. These were obtained from the charts in the attached *AIRotor* appendices.

Additional Motor Energy:

While the heat pipe makes use of passive heat transfer with no external energy required to drive the heat exchange process, the enthalpy wheel has a motor to rotate the desiccant between airstreams. The annual motor energy required was calculated from the nominal motor horsepower running continuously throughout the normal operating schedule except during economizer mode. This, in conjunction with the additional fan energy, will lessen the electric energy savings achieved by the wheel.

Life Cycle Cost:

As per the current standard ASHRAE life cycle cost analysis, the annual enthalpy wheel savings were evaluated as a present value from a 20-year period assuming a fixed 5% interest rate. Inflation and maintenance costs were not considered in this analysis. The initial cost is assumed to include installation.

Results:

The sensible cooling and heating savings, as well as the latent dehumidification and humidification savings obtained from the analysis are shown below:

Q_{COOL} (Btu/yr)	Q_{HEAT} (Btu/yr)	Q_{DEHUM} (Btu/yr)	Q_{HUM} (Btu/yr)
17,721,741.01	100,026,263.30	90,541,329.27	51,040,222.93

Below, the sensible and latent savings have been totaled, revealing the latent savings to be the greater of the two:

Sensible Savings (Btu/yr)	Latent Savings (Btu/yr)
117,748,004.31	141,581,552.20

After considering the efficiencies of the chiller, boiler, and humidifier, the actual energy and cost savings as they pertain to the equipment electrical and fuel input are shown below:

$W_{CHILLER}$ (kWh/yr)	W_{BOILER} (Btu/yr)	$W_{HUMIDIFIER}$ (kWh/yr)
12,029.23	**122,731,611.41**	**18,698.02**

Cooling Savings	Heating Savings	Humidification Savings
$979.18	**$1,692.47**	**$1,522.02**

The enthalpy wheel was found to operate 33.3% of the year. This is a significant amount considering the operating schedule of the spaces and that the wheel will be shut off during economizer mode. The energy and economic cost of running the wheel are shown below, along with the annual fan work and cost.

Fan (kWh/yr)	Motor (kWh/yr)
3,693.18	16,321.06

Fan Cost	Motor Cost
$300.62	$1,328.53

As with the heat pipe, the added electrical costs prove to be greater than the cooling savings achieved.

Summing the cooling, heating, and humidification savings above, and then subtracting the fan and motor cost yields the total annual savings from the specified enthalpy wheel.

Total Enthalpy Wheel Savings
$2,564.51

The life cycle cost analysis of purchasing and utilizing the enthalpy wheel over the 20-year period resulted in a present value of $51,282.63. When compared to the initial cost of the wheel at $14,360.00, the overall present worth comes to $36,922.63. As with the heat pipe, this energy recovery device has proven to be a sound investment leading to both energy and cost savings.

Conclusions and Discussion:

Like the heat pipe analysis, this model made use of average weather statistics which will likely provide fairly accurate life cycle savings, but are not necessarily representative of the savings attainable in a given year. Also like the heat pipe, this model assumed a constant effectiveness independent of ambient conditions. Aside from these few estimations, this model offers a quite practical look at the benefits achievable from the enthalpy wheel. The constant flow rate assumed for normal operating mode in this analysis is, unlike the heat pipe, likely to be the amount of outdoor air that is actually introduced to the space when not in economizer mode. The economizer mode logic of the current design has been used in this model, ensuring that any savings produced is a result of the enthalpy wheel, and not the existing economizer mode.

As with the heat pipe analysis, sensible heating savings proved to be the greatest benefit of the wheel. Such a small amount of sensible cooling savings were achieved, however, that the total latent savings were greater than the total sensible.

Also similar to the heat pipe, one may consider downsizing the chiller, boiler, and humidifiers, believing that they will no longer have to meet the peak loads as before. However, if the wheel requires maintenance and the air is routed around by the bypass damper, the same energy savings will not be present. In these cases, it would beneficial to maintain the peak capabilities that the present equipment has.

The newly specified wheel is to be situated on the existing slab-on-grade beside the outdoor obstacle area, near the existing outdoor air intake louvers. Duct reconfiguration, although not as great an issue with the enthalpy wheel, has been given little consideration in this model as well. The only major issue is re-routing the relief ductwork past the outdoor air intake. This new configuration appears to result in approximately the same distance that the current relief ductwork passes through. After passing through the existing filtration system in place, re-entry of relief air into the outdoor air intake is likely to be a non-issue, unless large amounts of re-entry are discovered. Once again, a computational fluid dynamic study in combination with possible wind directions and speeds would be necessary to ensure that re-entry levels would not pose a significant problem. This study was outside the scope of this report.

For an analysis of the integration of the enthalpy wheel into an existing panel board serving mechanical equipment, please see the *Electrical Breadth Redesign* section of this report.

Structural Breadth Redesign – Support of Heat Pipe and Existing Equipment

As mentioned in the above section, *Mechanical Redesign – Heat Pipe for Laboratory Energy Recovery*, the newly specified heat pipe is to be situated atop an existing mechanical equipment support structure elevated above the roof level. This structure is built entirely of W10x26 wide-flange steel members, with the exception of (2) W10x33 post columns. The structure itself bears on two existing columns and a W24x76 wide-flange steel girder.

As with the existing exhaust fans, the heat pipe will bear on dunnage beams. Initially, these new dunnage beams will be assumed at the same size as all of the other members in the support frame. Starting with the dunnage support beams and working down through the existing frame, the structural integrity of the members will be evaluated. To better visualize the configuration of the support structure, please see the *Mechanical Equipment Support – Structural Plan* and *Section*.

Analysis:

In the sections that follow, the methods used to assess the members are described. All of the flexural members were evaluated on the bases of their shear, moment, and deflection limitations. The post columns were evaluated based on their effective length. To fully understand the steps of the analysis, the *Sample Calculations* section of this report is to be used in conjunction with this description.

The purpose of this analysis is to determine if the members can handle the additional load—not to re-design the existing members. For this reason, many conservative approaches were utilized.

Snow Load:

Coefficient values used in the calculation of the snow load were obtained from the *DEA Narrative* and from the ASCE7-05 code manual.

Beam A – New Support Dunnage Beams:

The heat pipe is to bear on these newly specified members. The total load on these beams was considered to be a point load centered at the middle of the member for ease of deflection calculation. This is a conservative approach.

Beam B – Existing Support Dunnage Beams:

The existing exhaust fans bear on these members. The total load on these beams was considered to be a point load centered at the middle of the member for ease of deflection calculation. This is a conservative approach.

Beam C – Existing Beams Supporting Dunnage Beams:

The new and existing dunnage beams are to bear on these existing members. The shear and moment were calculated using the actual loading of the member. The total load on these beams was considered to be a point load centered at the middle of the member for ease of deflection calculation. This is a conservative approach.

Beam D – Existing Beam:

The total load on this beam was considered to be a point load at center span for ease of deflection calculation. This is a conservative approach.

Beam E – Existing Beam Supporting Beams C and D:

This beam is to support existing beams C and D as well as snow and metal grating loads. The shear and moment were calculated using the actual loading of the member. The total load on this beam was considered to be a point load centered at the middle of the member for ease of deflection calculation. This is a conservative approach.

Column F – Existing Posts Supporting Beam E:

These posts support beam E at each end and bear on the existing girders below. The effective length of the column was assessed.

Girder G – Existing Beam Supporting Columns F and Roof Joists:

These members support existing roof joists as well as column F. The girders are actually angled at a 2:12 slope and would, therefore, experience some compression. By treating the girder as an entirely flexural, flat member, the shear, moment, and deflections calculations will be conservative.

Results and Discussion:

All of the members evaluated were found to retain their structural integrity even after the additional equipment was included and conservative approaches were employed. The allowable deflection proved be the criterion to pass by the narrowest margin for most members.

The only additional structural costs incurred came from the two new dunnage beams for support of the heat pipe. These members, chosen as W10x26 for simplicity of design, proved to be slightly oversized. If budget problems prove to be an issue, these members could be downsized to avoid superfluous initial cost spending.

Electrical Breadth Redesign – Integrating Enthalpy Wheel into Existing Panel Board

As discussed in the *Mechanical Redesign – Enthalpy Wheel for Classrooms and Offices* section of this report, the newly specified enthalpy wheel is to be situated on the existing slab-on-grade beside the outdoor obstacle area, near the existing outdoor air intake louvers. The wheel is driven by a 7.5 nominal hp motor and is to be served by 3Φ alternating current at 460 V. An existing panel board serving mechanical equipment, MP-2, has been sized to allow for new equipment, leaving a few circuit breaker slots unused. This purpose of this section is to integrate the enthalpy wheel into the existing panel board, MP-2.

Analysis:

In the *Sample Calculations* section of this report, the methods and calculations used to integrate the wheel's motor into the existing panel board are presented. Using the *National Electric Code-2005*, the size of the branch circuit, circuit breaker, and conduit have been calculated.

Results:

It was found that the enthalpy wheel's motor required a 20A / 3 Pole circuit breaker. In panel board MP-2, circuit number 19 had been left as a spare 20A / 3 Pole breaker in the original design. Introducing the enthalpy wheel's motor to this slot brings the total amperage on the panel board to 163.75 A, well below it's 300 A maximum limit. Please note the updated panel board MP-2 below.

Panel Board MP-2

CKT #	Description	VFD/T/HP/kW		Load Amps	Load kVA	Breaker Amps / Poles
1	Humidifier H-1	-	-	9	4.3	15 / 2
2	Humidifier H-2	-	-	11	5.3	15 / 2
3	Humidifier H-3	-	-	4	1.9	15 / 2
4	Humidifier H-4	-	-	3	1.4	15 / 2
5	AHU-1SF	15	VFD	21	17.4	30 / 3
6	AHU-1RF	5	VFD	7.6	6.3	15 / 3
7	AHU-3SF	20	VFD	27	22.4	40 / 3
8	AHU-3RF	7.5	VFD	11	9.1	20 / 3
9	AHU-4SF	3	HP	4.8	4	15 / 3
10	AHU-5SF	1.5	HP	2.6	2.2	15 / 3
11	RF-5	0.5	HP	1	0.8	15 / 3
12	CHWP-1	15	HP	21	17.4	30 / 3
13	CHWP-2	15	HP	21	17.4	30 / 3
14	Washer Machine	1.3	kW	1.6	1.3	15 / 3
15	Washer Machine	1.3	kW	1.6	1.3	15 / 3
16	Hot Water Booster	9	kW	32.5	9	45 / 1
17	Panel PM2 via 30kVA Trans.	30	T	8.1	2.9	50 / 3
18	SPARE	-	-	-	-	15 / 3
19	*Enthalpy Wheel*	*7.5*	*HP*	*13.75*	*6.6*	20 / 3
20	SPACE					
				TOTAL 3Φ Amps	163.75	
				TOTAL kVA	131.6	

Neutral Bus:	100%	Voltage:	480Y/277V, 3Φ, 4W + GRND
Ground Bus:	YES	Mains:	400 A
Isolate Ground Bus:	NO	Main Ckt. Brkr.	300A / 3 Pole Breaker
Mounting:	Surface	AIC Rating:	22,000

References

Kling – Architects, Engineers, Interiors. 2006. DEA Clandestine Laboratory Training Center Design Documents Released Submission.

Kling – Architects, Engineers, Interiors. 2006. DEA Clandestine Laboratory Training Center Design Submission Narrative.

Kling – Architects, Engineers, Interiors. 2004. FDA Life Sciences Consolidation for CDER and CDRH Branches Chapter/Regional Technology Award – Short Form.

Schwedler, M. 2006. Variable Primary Flow Systems Revisited. http://www.trane.com/commercial/library/vol31_4/index.asp#intro

Methamphetamine. 2006. Boulder County, Colorado - http://www.co.boulder.co.us/health/meth/healthImpacts/gasses.htm

Efficient Humidity Control with Heat Pipes. December, 1991. Johannesen and West. http://www.p2pays.org/ref/08/07651.pdf

Sample Calculations

For calculations with multiple values, only the general equation has been provided. All calculations resulting in a single value have been shown in full below.

<u>Heat Pipe</u>:

Air Handling Unit Operating Schedule:

Lab spaces designed to operate 24 hr/day, 365 days/yr

Time-Weighted Average Flow of Exhaust Air:

$$V_{dot} = \frac{[\,(625hr)*(8040cfm) + (8135hr)*(8040cfm*0.40)\,]}{8760hr} = \mathbf{3{,}560.18\ cfm}$$

Mass Flow of Exhaust Air:

$$m_{dot} = \rho * V_{dot} = (0.07518\ lbm/ft^3)*(3560.18\ cfm)*(60\ min/hr) = \mathbf{16{,}059.25 lbm/hr}$$

Temperature of Outdoor Air Exiting the Heat Pipe:

$$T_{OA}OUT = T_{OA}IN + (T_{db}LAB - T_{OA}IN)*\epsilon$$

Hourly Cooling Savings:

IF: $T_{OA}IN > T_{db}LAB$ THEN: Q_{COOL} [Btu/hr] = $\epsilon * m_{dot} * C_p * (T_{OA}IN - T_{OA}OUT)$

Hourly Heating Savings:

IF: $T_{OA}IN < T_{db}LAB$ THEN: Q_{HEAT} [Btu/hr] = $\epsilon * m_{dot} * C_p * (T_{OA}OUT - T_{OA}IN)$

Yearly Cooling Savings:

Q_{COOL} [Btu/yr] = $\sum Q_{COOL}$ [Btu/hr] = $Q_{CHILLER}$ = **17,839,836.25 Btu/yr**

Yearly Heating Savings:

Q_{HEAT} [Btu/yr] = $\sum Q_{HEAT}$ [Btu/hr] = Q_{BOILER} = **166,624,930.17 Btu/yr**

Chiller Energy Savings:

$$W_{CHILLER} = Q_{CHILLER} / EER_{CHILLER}$$

$W_{CHILLER}$ = (17,839,836.25Btu/yr)/[(9.0Btu/W*hr)*(1000W/kW)]

$W_{CHILLER}$ = **1,982.20 kWh/yr**

Boiler Energy Savings:

$W_{BOILER} = Q_{BOILER}/\eta_{BOILER}$=(166,624,930.17 Btu/yr)/0.815 = **204,447,767 Btu/yr**

$W_{BOILER} = W_{NAT.GAS}$

Additional Fan Energy Required:

W_{dot} [hp] = (ΔP)*(V_{dot}) / (η_{FAN}*6350)

W_{FAN} [kWh/yr] = W_{dot}*(2544.5Btu/hr*hp)*(8760hr)*(1kWh/3412.14Btu)

Supply Fan:

W_{dot} = (0.57 in wg)*(3560cfm) / (0.8*6350) = **0.40 hp**

W_{FAN} = (0.40 hp)*(2544.5Btu/hr*hp)*(8760hr)*(1kWh/3412.14Btu)

W_{FAN} = **2,609.40 kWh/yr**

Exhaust Fan:

W_{dot} = (0.72 in wg)*(3560cfm) / (0.8*6350) = **0.50 hp**

W_{FAN} = (0.50 hp)*(2544.5Btu/hr*hp)*(8760hr)*(1kWh/3412.14Btu)

W_{FAN} = **3,296.09 kWh/yr**

Total Additional Fan Energy:

$W_{FAN} = W_{SUPPLY} + W_{EXHAUST}$ = 2,609.40 kWh/yr + 3,296.09 kWh/yr

W_{FAN} = **5,905.49 kWh/yr**

Electrical Energy Savings:

$W_{ELECT} = W_{CHILLER} - W_{FAN}$ = 1,982.20 kWh/yr - 5,905.49 kWh/yr

W_{ELECT} = **-3,923.29 kWh/yr**

Average Electrical Cost (whole building):

$$[¢/kWh] = \frac{[\ \$105{,}510\ (\text{Consumption + Demand included})\ ]}{1{,}295{,}989\ \text{kWh (total use per year)}} *(100¢/\$1)$$

$[¢/kWh]$ = **8.14 ¢/kWh**

Annual Electrical Savings:

$A_{ELECT} = (W_{ELECT})^{*}$ (Avg. Elect. Cost)

A_{ELECT} = (-3,923.29 kWh/yr)*(8.14 ¢/kWh)*($1/100¢) = **-$319.36**

Annual Natural Gas Savings:

$A_{NAT.GAS} = W_{NAT.GAS}$*(Cost/Btu)

$A_{NAT.GAS}$ = (204,447,767.08 Btu/yr)*($13.79/millionBtu) = **$2,819.33**

Total Annual Savings:

$A_{TOTAL} = A_{NAT.GAS} + A_{ELECT}$ = $2,819.33 + (-$319.36) = **$2,499.98**

Present Value from Life Cycle Cost Analysis:

$$PV = \frac{A_{TOTAL}*(1+i)^{n} - 1}{i*(1+i)^{n}}$$, where i = 0.05 and n = 20 years

$$PV = \frac{(\$2{,}499.98)*(1+0.05)^{20} - 1}{0.05*(1+0.05)^{20}} = \mathbf{\$49{,}992.05}$$

Present Value after Initial Cost:

PV = $49,992.05 – $1,120 = $48,872.05

Enthalpy Wheel:

Air Handling Unit Operating Schedule:

IF: Time of day is between 6:00am and 7:00pm (hour of warm-up time)
THEN: Air handling units will operate

Outdoor and Relief Air Volumetric Flow:

V_{dot} = 2280cfm + 1310cfm = **3,590 cfm**

Mass Flow Rate:

$m_{dot} = \rho * V_{dot}$ = (0.07518 lbm/ft^3)*(3590 cfm)*(60 min/hr) = **16,193.77 lbm/hr**

Temperature of Outdoor Air Exiting Enthalpy Wheel:

$T_{OA}OUT = T_{OA}IN + (T_{db}REL - T_{OA}IN)*\epsilon$

Humidity Ratio and Enthalpy of Outdoor Air:

EES Input (dry bulb, dew point, and atmospheric pressure in parametric table):

T_DB = x
T_DP = y
P_atm = z/1013.25

W = HumRat(AirH2O,T=T_DB, D=T_DP, P = P_atm)
h = Enthalpy(AirH2O,T=T_DB, D=T_DP, P=P_atm)

Economizer Mode:

IF: $T_{OA}IN < 70°F$ AND
IF: $h_{OA} < h_{RA}$ AND
IF: $T_{DP} > 39°F$ THEN:

Economizer Mode is activated.

Sensible Energy Savings:

Hourly Cooling:

IF: Air handling units are operating AND
IF: Economizer mode is *not* activated AND
IF: $T_{OA}IN > T_{db}REL$

THEN: Q_{COOL} [Btu/hr] = $\epsilon * m_{dot} * C_p * (T_{OA}IN - T_{OA}OUT)$

Hourly Heating:

IF: Air handling units are operating AND
IF: Economizer mode is *not* activated AND
IF: $T_{OA}IN < T_{db}REL$

THEN: Q_{HEAT} [Btu/hr] = $\epsilon * m_{dot} * C_p * (T_{OA}OUT - T_{OA}IN)$

Yearly Cooling:

Q_{COOL} [Btu/yr] = $(5/7) * \sum Q_{COOL}$ [Btu/hr] = **17,721,741.01 Btu/yr**

(weekday operation only)

Yearly Heating:

Q_{HEAT} [Btu/yr] = $(5/7) * \sum Q_{HEAT}$ [Btu/hr] = **100,026,263.30 Btu/yr**

(weekday operation only)

Latent Energy Savings:

Hourly Dehumidification:

IF: Air handling units are operating AND
IF: Economizer mode is *not* activated AND
IF: $W_{OA}IN > W_{REL}$

THEN: Q_{DEHUM} [Btu/hr] = $\epsilon * m_{dot} * L_V * (W_{OA}IN - W_{REL})$

Hourly Humidification:

IF: Air handling units are operating AND
IF: Economizer mode is *not* activated AND
IF: $W_{OA}IN < W_{REL}$

THEN: Q_{HUM} [Btu/hr] = $\epsilon * m_{dot} * L_V * (W_{REL} - W_{OA}IN)$

Yearly Dehumidification:

Q_{DEHUM} [Btu/yr] = $(5/7) * \sum Q_{DEHUM}$ [Btu/hr] = **90,541,329.27 Btu/yr**

(weekday operation only)

Yearly Humidification:

Q_{HUM} [Btu/yr] = (5/7)* $\sum Q_{HUM}$ [Btu/hr] = **51,040,222.93 Btu/yr**

(weekday operation only)

Chiller Energy Savings:

$Q_{CHILLER} = Q_{COOL}$ [Btu/yr] + Q_{DEHUM} [Btu/yr]

$Q_{CHILLER}$ = 17,721,741.01 Btu/yr + 90,541,329.27 Btu/yr

$Q_{CHILLER}$ = 108,263,070.28 Btu/yr

$W_{CHILLER} = Q_{CHILLER} / EER_{CHILLER}$

$W_{CHILLER}$ = (108,263,070.28 Btu/yr)/[(9.0Btu/W*hr)*(1000W/kW)]

$W_{CHILLER}$ = **12,029.23 kWh/yr**

Boiler Energy Savings:

$Q_{HEAT} = Q_{BOILER}$

$W_{BOILER} = Q_{BOILER}/\eta_{BOILER}$ = (100,026,263.30 Btu/yr)/0.815

W_{BOILER} = **122,731,611.41 Btu/yr**

$W_{BOILER} = W_{NAT.GAS}$

Humidifier Energy Savings:

$Q_{HUM} = Q_{HUMIDIFIER}$

$W_{HUMIDIFIER} = Q_{HUMIDIFIER}/\eta_{HUMIDIFIER}$

$W_{HUMIDIFIER}$ = [(51,040,222.93 Btu/yr) / 0.80] * (1kWh/3,412.14Btu)

$W_{HUMIDIFIER}$ = **18,698.02 kWh/yr**

Annual Chiller Savings:

$A_{CHILLER} = (W_{CHILLER})$* (Avg. Elect. Cost)

$A_{CHILLER}$ = (12,029.23 kWh/yr)*(8.14 ¢/kWh)*($1/100¢) = **$979.18**

Annual Natural Gas Savings:

$A_{NAT.GAS} = (W_{NAT.GAS})*$ (Cost/Btu)

$A_{NAT.GAS}$ = (122,731,611.41 Btu/yr)*($13.79/millionBtu) = **$1,692.47**

Annual Humidifier Savings:

$A_{HUMIDIFIER} = (W_{HUMIDIFIER})*$ (Avg. Elect. Cost)

$A_{HUMIDIFIER}$ = (18,698.02 kWh/yr)*(8.14 ¢/kWh)*($1/100¢) = **$1,522.02**

Additional Fan Energy Required:

W_{dot} [hp] = $(\Delta P)*(V_{dot}) / (\eta_{FAN}*6350)$

W_{FAN} [kWh/yr] = W_{dot}*(2544.5Btu/hr*hp)*(8760hr)*(1kWh/3412.14Btu)

Supply and Exhaust Fan:

W_{dot} = (0.40 in wg)*(3560cfm) / (0.8*6350) = 0.28 hp

W_{FAN} = (0.28 hp)*(2544.5Btu/hr*hp)*(8760hr)*(1kWh/3412.14Btu)

W_{FAN} = 1,846.59 kWh/yr

Total Additional Fan Energy:

$W_{FAN.TOTAL} = \sum W_{FAN}$ (supply and exhaust)

$W_{FAN.TOTAL}$ = 2*(1,846.59 kWh/yr) = **3,693.18 kWh/yr**

Additional Motor Energy Required:

$W_{MOTOR} = W_{HP}$*(0.746 kW/hp)*(Operating Hours)

W_{MOTOR} = (7.5hp)*(0.746 kW/hp)*(2,917 hr) = **16,321.06 kWh/yr**

Additional Fan Cost:

$A_{FAN} = (W_{FAN})*$ (Avg. Elect. Cost)

A_{FAN} = (3,693.18 kWh/yr)*(8.14 ¢/kWh)*($1/100¢) = **$300.62**

Additional Motor Cost:

$A_{MOTOR} = (W_{MOTOR})*$ (Avg. Elect. Cost)

$A_{MOTOR} = (16{,}321.06 \text{ kWh/yr})*(8.14\ ¢\text{/kWh})*(\$1/100¢) =$ **$1,328.53**

Total Annual Enthalpy Wheel Savings:

$A_{WHEEL} = A_{CHILLER} + A_{NAT.GAS} + A_{HUMIDIFIER} - A_{FAN} - A_{MOTOR}$

$A_{WHEEL} = \$979.18 + \$1{,}692.47 + \$1{,}522.02 - \$300.62 - \$1{,}328.53$

$A_{WHEEL} =$ **$2,564.51**

Present Value from Life Cycle Cost Analysis:

$$PV = \frac{A_{TOTAL}*(1+i)^n - 1}{i*(1+i)^n}$$, where i = 0.05 and n = 20 years

$$PV = \frac{(\$2{,}564.51)*(1+0.05)^{20} - 1}{0.05*(1+0.05)^{20}} = \mathbf{\$51{,}282.63}$$

Present Value after Initial Cost:

PV = $51,282.63 – $14,360.00 = **$36,922.63**

Structural Breadth:

Snow Load:

From DEA Narrative:

Exposure C
p_g = 25 lb/ft^2, Snow Ground Load
I = 1.0, Importance Factor

From ASCE7-05:

C_e = 0.9, Exposure Factor (Table 7-2)
C_t = 1.0, Thermal Factor (Table 7-3)

$P_f = 0.7*C_e*C_t*I*(p_g) = 0.7*(0.9)*(1.0)*(1.0)*(25\text{lb/ft}^2) = \mathbf{15.75\ lb/ft^2}$

Beam A – New Support Dunnage Beams:

W10x26 (assumed)
Length = 10'-9"

Snow = (15.75lb/ft^2)*(10.75ft)*(1.0ft) = 169 lb
Equipment = (684lb)/2 = 342 lb
Self Weight = (26lb/ft)*(10.75ft) = 279.5 lb

Total = 791 lb   Assume point loading (conservative):

R_A = 395 lb

Shear:

$V_{MAX} = 0.395\ ^K$

$\varphi V_n = \varphi_v*(0.6)*F_Y*d*t_w$

$\varphi V_n = (0.9)*(0.6)*(50^K/\text{in}^2)*(10.3\text{in})*(0.260\text{in}) = \mathbf{72.31\ ^K > 0.395\ ^K}$ **OK**

Moment:

$M_{MAX} = 2.12\ \text{ft}*^K$

$\varphi M_n = \varphi_b*F_Y*Z_X$

$\varphi M_n = (0.9)*(50^K/\text{in}^2)*(31.3\text{in}^3) = 1{,}409\ \text{in}*^K = \mathbf{117.4\ ft*^K > 2.12\ ft*^K}$ **OK**

Deflection:

$$\Delta_{MAX} = \frac{P*L^3}{48*E*I} = \frac{(0.791^K)*(10.75ft*12in/ft)^3}{48*(29,000\ ^K/in^2)*(144in^4)} = 0.00847\text{ in}$$

$\Delta_{ALLOWED}$ = L/480 = (10.75ft*12in/ft)/480 = **0.269 in > 0.00847 in** **OK**

Beam B – Existing Support Dunnage Beams:

W10x26
Length = 10'-9"

Snow = $(15.75lb/ft^2)*(10.75ft)*(2.5ft)$ = 423.3 lb
Equipment = (4,109lb)/2 = 2,054.5 lb
Self Weight = (26lb/ft)*(10.75ft) = 279.5 lb

Total = 2,757 lb Assume point loading (conservative):

R_B = 1,379 lb

Shear:

$V_{MAX} = 1.38\ ^K$

$\varphi V_n = \varphi_v*(0.6)*F_Y*d*t_w$

$\varphi V_n = (0.9)*(0.6)*(50^K/in^2)*(10.3in)*(0.260in) =$ **$72.31\ ^K > 1.38\ ^K$** **OK**

Moment:

$M_{MAX} = 7.41\ ft*^K$

$\varphi M_n = \varphi_b*F_Y*Z_X$

$\varphi M_n = (0.9)*(50^K/in^2)*(31.3in^3) = 1,409\ in*^K =$ **$117.4\ ft*^K > 7.41\ ft*^K$** **OK**

Deflection:

$$\Delta_{MAX} = \frac{P*L^3}{48*E*I} = \frac{(2.76^K)*(10.75ft*12in/ft)^3}{48*(29,000\ ^K/in^2)*(144in^4)} = 0.0296\text{ in}$$

$\Delta_{ALLOWED}$ = L/480 = (10.75ft*12in/ft)/480 = **0.269 in > 0.0296 in** **OK**

Beam C – Existing Beams Supporting Dunnage Beams:

W10x26
Length = 9'-0"

$\sum M_2 = 0 \;\; => \;\; R_{C1} = 2{,}037.6$ lb
$\sum M_1 = 0 \;\; => \;\; R_{C2} = 1{,}510.4$ lb

Shear:

$V_{MAX} = 2.04\ ^{K}$

$\varphi V_n = \varphi_v*(0.6)*F_Y*d*t_w$

$\varphi V_n = (0.9)*(0.6)*(50^K/in^2)*(10.3in)*(0.260in) = \mathbf{72.31\ ^K > 2.04\ ^K}$ **OK**

Moment:

$M_{MAX} = 4.93\ ft*^K$

$\varphi M_n = \varphi_b*F_Y*Z_X$

$\varphi M_n = (0.9)*(50^K/in^2)*(31.3in^3) = 1{,}409\ in*^K = \mathbf{117.4\ ft*^K > 4.93\ ft*^K}$ **OK**

Deflection:

Assume P = ∑ Point Loads @ Center Span

$$\Delta_{MAX} = \frac{P*L^3}{48*E*I} = \frac{(3.55^K)*(9.0ft*12in/ft)^3}{48*(29{,}000\ ^K/in^2)*(144in^4)} = 0.0223\ in$$

$\Delta_{ALLOWED} = L/480 = (9.0ft*12in/ft)/480 = \mathbf{0.225\ in > 0.0223\ in}$ **OK**

Beam D – Existing Beam:

W10x26
Length = 9'-0"

Self Weight = (26lb/ft)*(9.0ft) = 234 lb

Total = 234 lb Assume point loading (conservative):

$R_D = 117$ lb

Shear:

$V_{MAX} = 0.117\ ^K$

$\varphi V_n = \varphi_v*(0.6)*F_Y*d*t_w$

$\varphi V_n = (0.9)*(0.6)*(50^K/in^2)*(10.3in)*(0.260in) = \mathbf{72.31\ ^K > 0.117\ ^K}$ **OK**

Moment:

$M_{MAX} = 0.527\ ft*^K$

$\varphi M_n = \varphi_b*F_Y*Z_X$

$\varphi M_n = (0.9)*(50^K/in^2)*(31.3in^3) = 1{,}409\ in*^K = \mathbf{117.4\ ft*^K > 0.527\ ft*^K}$ **OK**

Deflection:

$$\Delta_{MAX} = \frac{P*L^3}{48*E*I} = \frac{(0.234^K)*(9.0ft*12in/ft)^3}{48*(29{,}000\ ^K/in^2)*(144in^4)} = 0.00147\ in$$

$\Delta_{ALLOWED} = L/480 = (9.0ft*12in/ft)/480 = \mathbf{0.225\ in > 0.00147\ in}$ **OK**

Beam E – Existing Beam Supporting Beams C and D:

W10x26
Length = 22'-0"

Snow = $(15.75lb/ft^2)*(22.0ft)*(4.5ft) = 1{,}559.3$ lb
Grating = $(15lb/\ ft^2)*(22.0ft)*(4.5ft) = 1{,}485$ lb

Total = 3,044.3 lb Assume point loading (conservative):

$\sum M_1 = 0 \Rightarrow R_{E2} = 2{,}600.2$ lb
$\sum M_2 = 0 \Rightarrow R_{E1} = 4{,}636.3$ lb

Shear:

$V_{MAX} = 2.60\ ^K$

$\varphi V_n = \varphi_v*(0.6)*F_Y*d*t_w$

$\varphi V_n = (0.9)*(0.6)*(50^K/in^2)*(10.3in)*(0.260in) = \mathbf{72.31\ ^K > 2.60\ ^K}$ **OK**

Moment:

$M_{MAX} = 19.89\ ft*^K$

$\varphi M_n = \varphi_b * F_Y * Z_X$

$\varphi M_n = (0.9)*(50^K/in^2)*(31.3in^3) = 1{,}409\ in*^K =$ **$117.4\ ft*^K > 19.89\ ft*^K$** **OK**

Deflection:

$$\Delta_{MAX} = \frac{P*L^3}{48*E*I} = \frac{(2.60^K)*(22.0ft*12in/ft)^3}{48*(29{,}000\ ^K/in^2)*(144in^4)} = 0.239\ in$$

$\Delta_{ALLOWED} = L/480 = (22.0ft*12in/ft)/480 =$ **0.550 in > 0.239 in** **OK**

Column F – Existing Posts Supporting Beam E:

W10x33
Height = 2'-6"

$(R_{E1} = 4{,}636.3\ lb) > (R_{E2} = 2{,}600.2\ lb)$ so check R_{E1}:

$$(KL_{EFF})_Y = \frac{K*L}{(r_x/r_y)} = \frac{(1.0)*(2.5ft)}{(2.16)} = 1.16\ ft, \text{ but } 2.5\ ft > 1.16\ ft$$

So use 2.5 ft: KL = 6.0 can hold $373^K > 4.64^K$ **OK**

Girder G – Existing Beam Supporting Columns F and Roof Joists:

Roof Joists:

Metal Deck = $2lb/ft^2$
Roofing and Insulation = $5\ lb/ft^2$
Total Roof Dead Load = $2lb/ft^2 + 5\ lb/ft^2 = 7\ lb/ft^2$

$P_U = 1.2*(D) + 1.6*(S)$
$P_U = 1.2*(7lb/ft^2) + 1.6*(15.75lb/ft^2) =$ **$33.6\ lb/ft^2$**

W12x19
Length = 22'-0"

Contributing Area to Girder G:

$A_{TRIB} = (6.67ft)*(22.0ft)/2 = 73.33\ ft^2$

Contribution to Point Load on Girder G:

$$P = (73.33ft^2)*(33.6lb/ft^2) = 2,463.9 \text{ lb}$$

W16x26
Length = 34'-0"

Contributing Area to Girder G:

$$A_{TRIB} = (6.67ft)*(34.0ft)/2 = 113.33 \text{ ft}^2$$

Contribution to Point Load on Girder G:

$$P = (113.33ft^2)*(33.6lb/ft^2) = 3,807.6 \text{ lb}$$

Total Point Load from Roof Joists on Girder G:

$$\sum P = 2,463.9 \text{ lb} + 3,807.6 \text{ lb} = 6,271.5 \text{ lb}$$

Girder G:

W24x76
Length = 40'-0"

Conservative to treat as entirely flexural, flat beam:

$\sum M_1 = 0 \quad \Rightarrow \quad R_{G2} = 16.72^{K}$
$\sum M_2 = 0 \quad \Rightarrow \quad R_{G1} = 19.27^{K}$

Shear:

$V_{MAX} = 19.27^{K}$

$\varphi V_n = \varphi_v*(0.6)*F_Y*d*t_w$

$\varphi V_n = (0.9)*(0.6)*(50^K/in^2)*(23.9in)*(0.440in) = \mathbf{283.9^{K} > 19.3^{K}}$ **OK**

Moment:

$M_{MAX} = 208.9 \text{ ft}*^{K}$

$\varphi M_n = \varphi_b*F_Y*Z_X$

$\varphi M_n = (0.9)*(50^K/in^2)*(200in^3) = 9,000 \text{ in}*^{K}$

$\varphi M_n = \mathbf{750 \text{ ft}*^{K} > 208.9 \text{ ft}*^{K}}$ **OK**

Deflection:

Treat 5 equal point loads as distributed load:

$5*(6.27\,^{K})/40ft = 0.784\,^{K}/ft$

$$\Delta_{MAX} = \frac{5\omega*L^4}{384*E*I} = \frac{5*(0.784^{K})*(40ft*12in/ft)^4}{384*(29{,}000\,^{K}/in^2)*(2{,}100in^4)} = 0.74 \text{ in}$$

Point load from Column F:

$$\Delta_{MAX} = \frac{Pab(a+2b)*[3a(a+2b)]^{1/2}}{27*E*I*L}$$

$$\Delta_{MAX} = \frac{(4.64^{K})(9ft)(31ft)(9ft+2*31ft)*[3*9ft*(9ft+2*31ft)]^{1/2}}{27*(29{,}000\,^{K}/in^2)*(2{,}100in^4)*(40ft)}$$

$\Delta_{MAX} = 0.106$ in

Total Deflection:

$\sum\Delta_{MAX} = 0.74in + 0.106in = 0.85$ in

$\Delta_{ALLOWED} = L/480 = (40ft*12in/ft)/480 =$ **1.0 in** > **0.85 in OK**

Electrical Breadth:

Enthalpy Wheel Motor:

(NEC 2005, Table 430.250)

7.5 hp, 3Φ AC, 460 V => **11 Full Load Amps (FLA)**

Sizing Branch Circuit:

(Typically for largest motor on branch, but there is only one motor on branch)

Minimum Circuit Amps = (FLA)*(Demand Factor)

MCA = (11A)*(1.25) = **13.75 A**

(NEC 2005, Table 310.16)

From MCA, using THWN, 75°C, copper wire => #12 AWG

Except from 240.4D:

Overcurrent protection shall not exceed 20A for #12 AWG

So check Maximum Overcurrent Protective Device (MODP):

(NEC 2005, Table 430.52)

Percent of Full-Load Current = 200%

MODP = (FLA)*(% Full-Load Current) = (11A)*(200%) = **22.0 A**

22A > 20A => must use #10 AWG

Circuit Breaker:

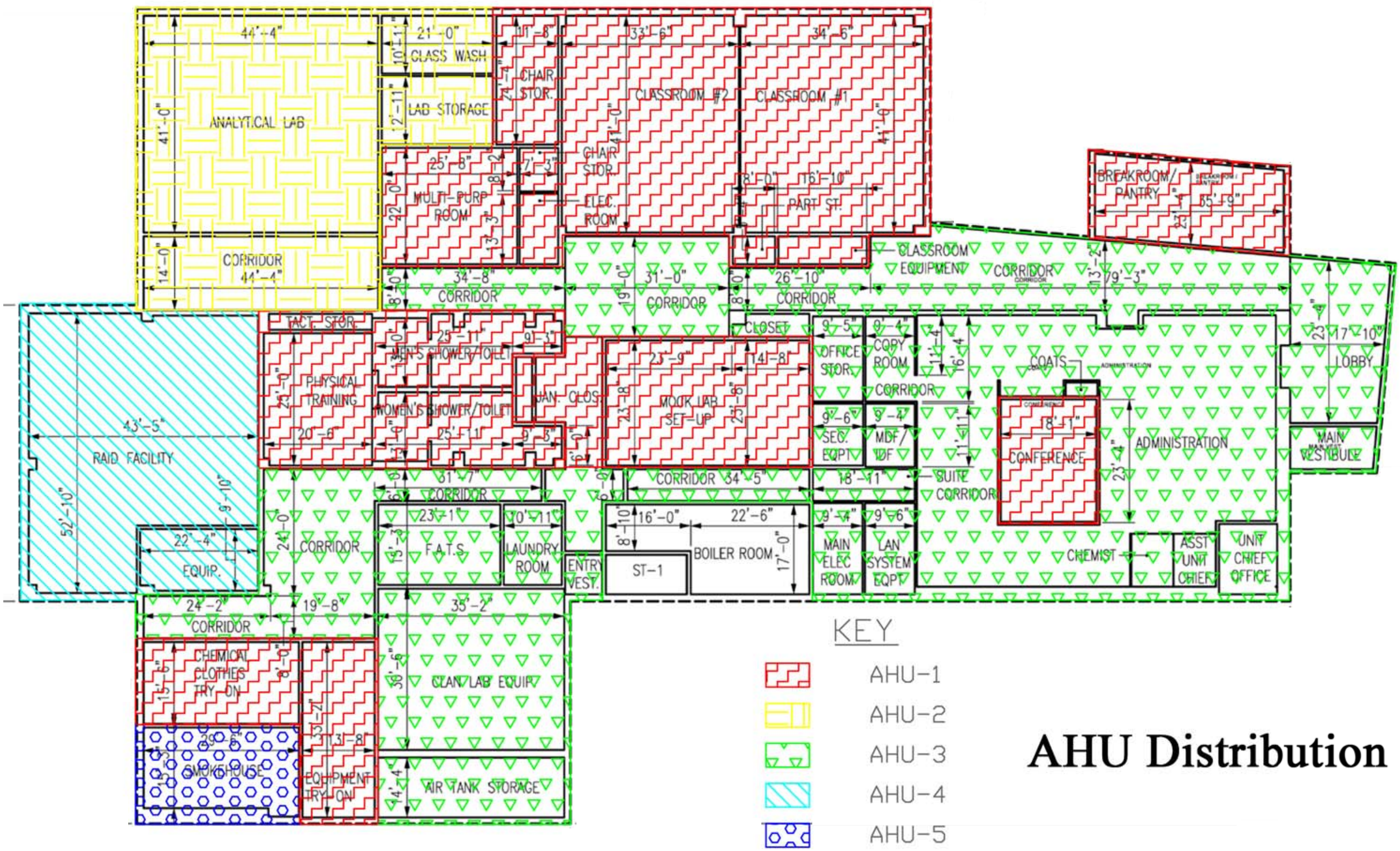
Circuit breaker must be smaller than MODP => next size down = **20 A**

Conduit:

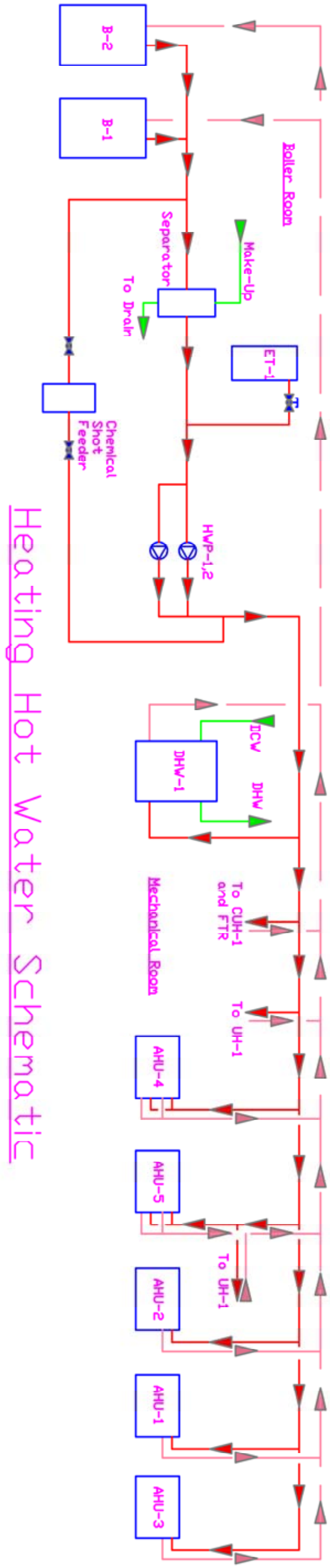
(NEC 2005, Table C-1) **Use 1½" conduit (electric metallic tubing)**

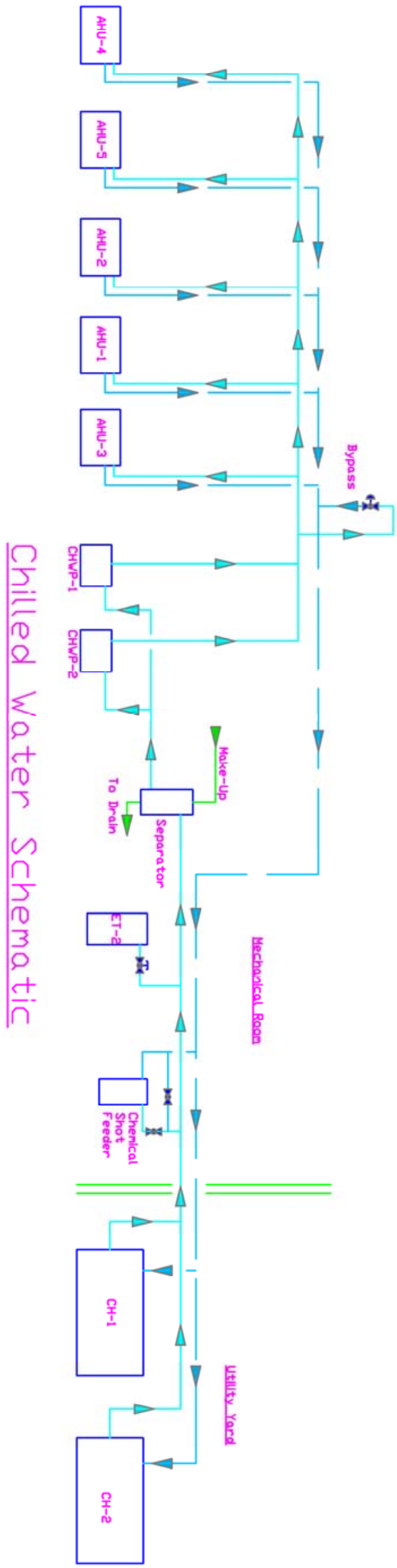
Final Sizing Information:

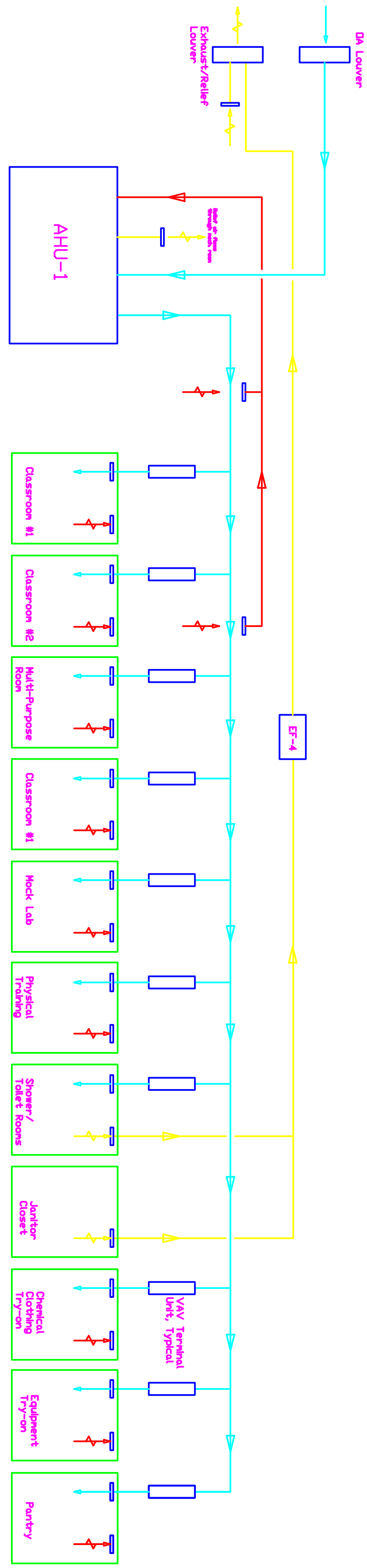
Use (4) #10 AWG wires in 1½" conduit with a 20 A circuit breaker



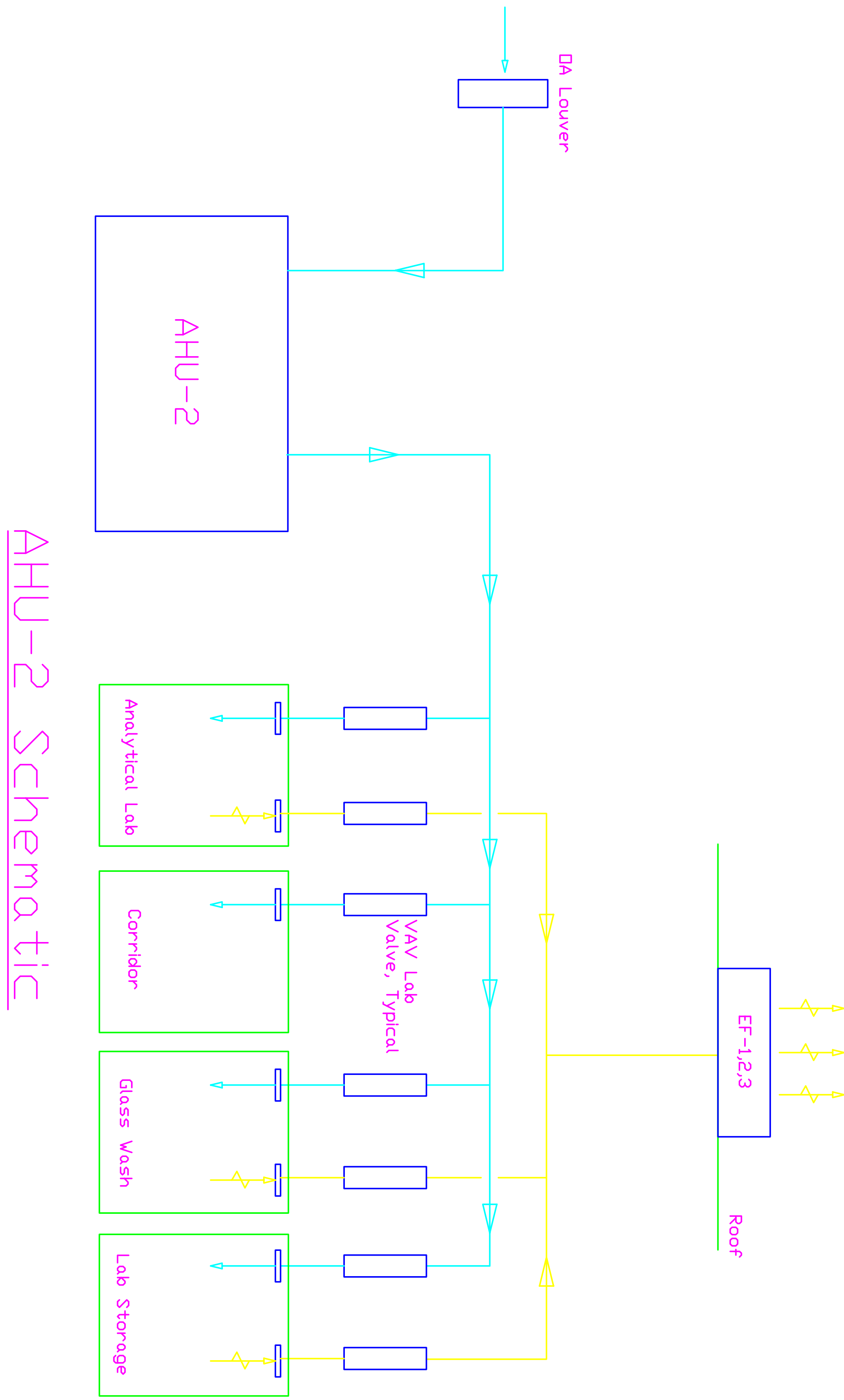
AHU Distribution

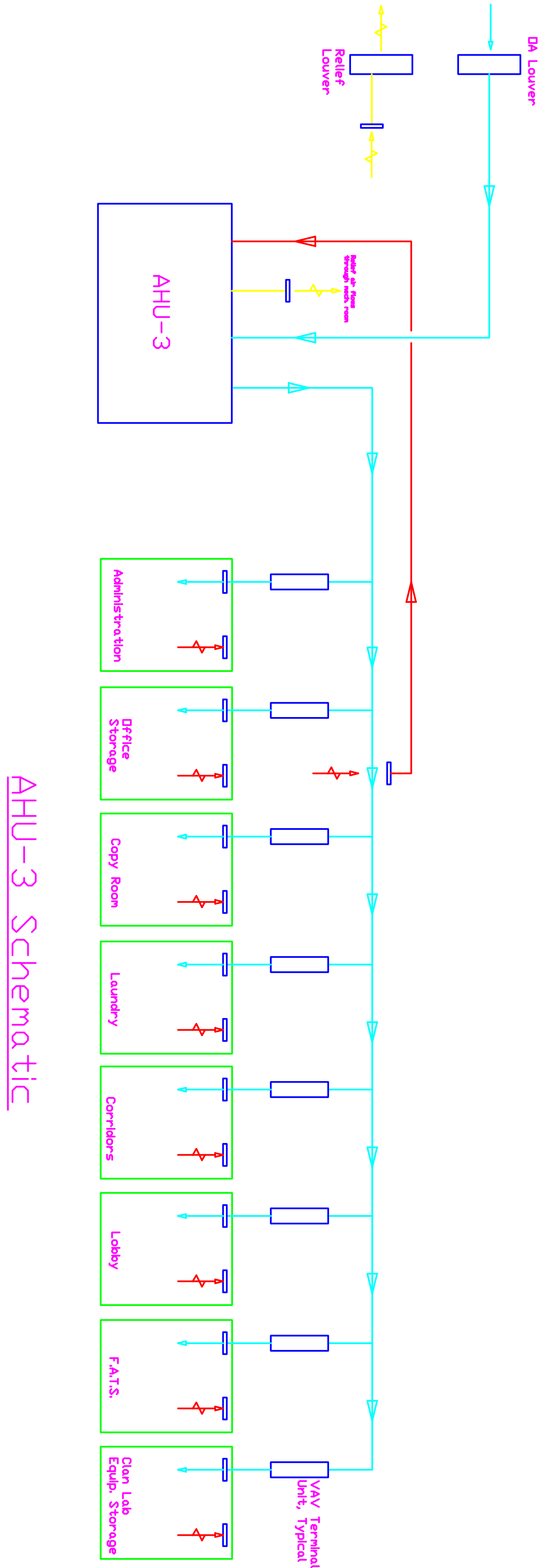

Heating Hot Water Schematic
B-2
B-1
Boiler Room
Make-Up
Separator
To Drain
ET-1
Chemical Shot Feeder
HWP-1,2
ICW
DHW-1
DHW
Mechanical Room
To CUH-1 and FTR
To UH-1
AHU-4
AHU-5
To UH-1
AHU-2
AHU-1
AHU-3

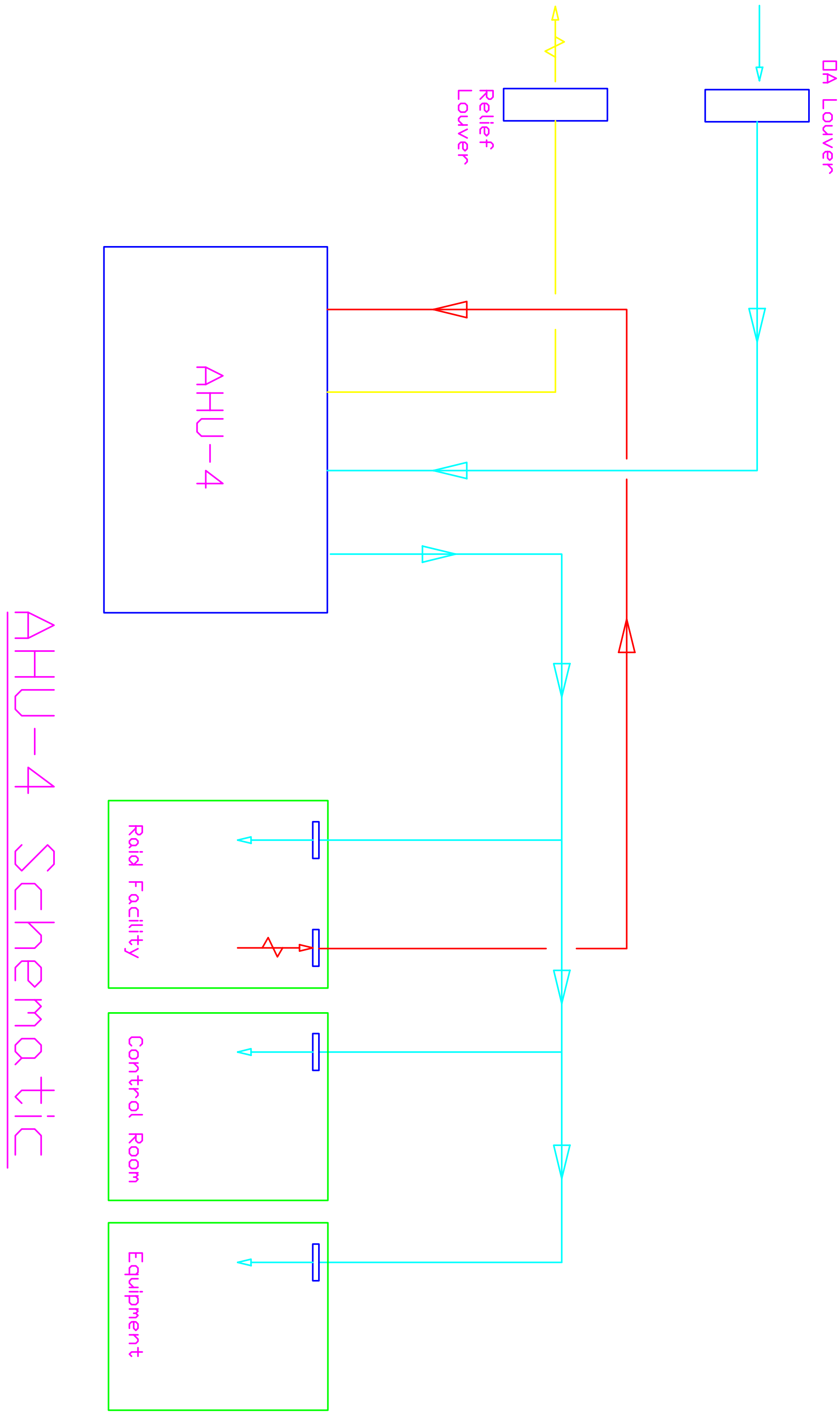

Chilled Water Schematic
AHU-4
AHU-5
AHU-2
AHU-1
AHU-3
Bypass
CHWP-1
CHWP-2
Make-Up
To Drain
Separator
ET-2
Mechanical Room
Chemical Shot Feeder
CH-1
CH-2
Utility Yard



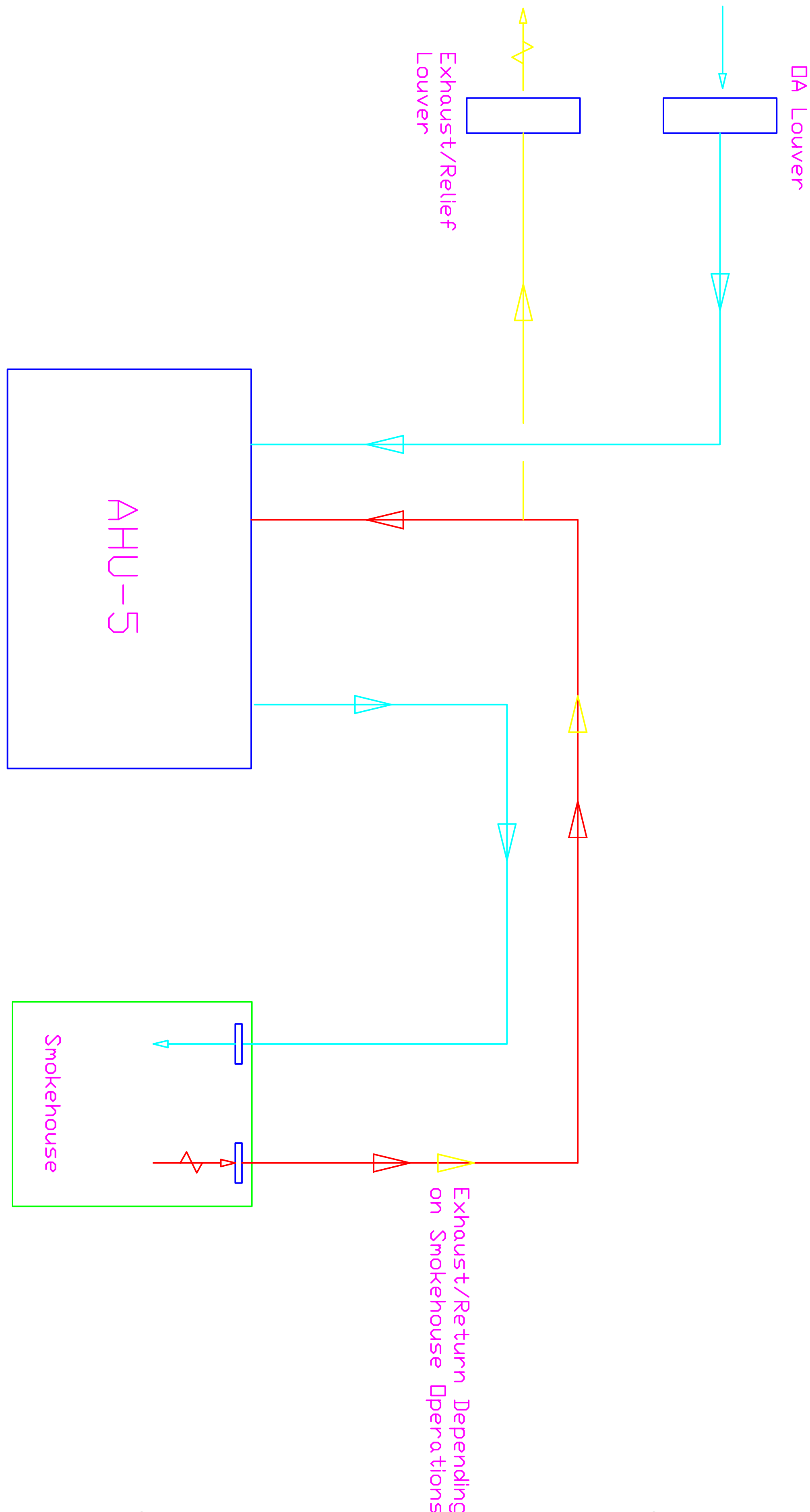
AHU-1 Schematic


AHU-2 Schematic
OA Louver
AHU-2
Analytical Lab
Corridor
Glass Wash
Lab Storage
VAV Lab
Valve, Typical
EF-1,2,3
Roof


AHU-3 Schematic
Relief Louver
OA Louver
AHU-3
Administration
Office Storage
Copy Room
Laundry
Corridors
Lobby
F.A.T.S.
Clan Lab Equip. Storage
VAV Terminal Unit, Typical


AHU-4 Schematic
Relief Louver
OA Louver
AHU-4
Raid Facility
Control Room
Equipment

AHU-5 Schematic



Air Handling Unit Schedule

		AHU-1	AHU-2	AHU-3	AHU-4	AHU-5
General:	Service	Classrooms	Lab	Offices	Raid	Smokehouse
	Total CFM	9,420	8,040	10,880	2,090	560
	min OA CFM	2,280	8,040	1,310	880	340
Supply Fan:	Type	22" Airfoil Cent.	20" Airfoil Cent.	22" Airfoil Cent.	12" Airfoil Cent.	Airfoil Centrifugal
	Ext. S.P. (in w.g.)	2.28	2.54	2.20	1.56	0.75
	Total S.P. (in w.g.)	5.80	6.30	5.60	4.30	3.12
	max BHP	13.1	12.0	14.2	2.5	1.1
	max RPM	1,711	2,006	1,725	2,611	3,499
	Volume Control	VFD	VFD	VFD	Constant	Constant
Motor:	HP	15	15	20	3	1
	max RPM	1,800	1,800	1,800	1,800	1,800
	Volts/Phase/Hz	460/3/60	460/3/61	460/3/62	460/3/63	460/3/64
Sound Attenuator:		SA-1S	-	SA-3S	-	-
Return Fan:	Type	20" Airfoil Cent.	-	22" Airfoil Cent.	-	-
	max CFM	8,270	-	10,880	-	-
	Ext. S.P. (in w.g.)	1.0	-	1.0	-	-
	Total S.P. (in w.g.)	1.1	-	1.1	-	-
	max BHP	3.4	-	4.6	-	-
	max RPM	1,255	-	1,150	-	-
	Volume Control	VFD	-	VFD	-	-
Motor:	HP	5	-	7.5	-	-
	max RPM	1,800	-	1,800	-	-
	Volts/Phase/Hz	460/3/60	-	460/3/62	-	-
Cooling Coil:	CFM	9,420	8,040	10,880	2,090	560
	EAT DB/WB (°F)	82.0 / 67.6	92.0 / 79.0	80.9 / 65.9	82.7 / 69.7	86.7 / 63.1
	LAT DB/WB (°F)	52.1 / 51.1	52.1 / 52.0	52.1 / 52.0	52.1 / 52.0	53.1 / 53.0
	MBH Sensible/Total	310.5 / 446.5	358.3 / 760.4	344.8 / 450.9	70.7 / 115.4	21.3 / 36.7
	max Face Velocity (fpm)	475	500	475	450	450
	max Air ∆P (in w.g.)	0.9	1.3	0.7	0.6	1.0
	% Propylene Glycol	30	30	30	30	30
	EWT / LWT (°F)	45 / 55	45 / 55	45 / 55	45 / 55	45 / 55
	GPM	96.6	165.0	97.5	25.0	7.5
	max Water ∆P (in w.c.)	8.4	16.1	7.2	7.8	20
Pre-Heat Coil:	max CFM	4,940	8,040	4,350	2,090	560
	EAT / LAT Dry Bulb (°F)	44.5 / 69.0	16.0 / 74.2	52.9 / 75.6	48.4 / 63.6	16.0 / 55.0
	MBH	131.7	507.5	107.1	34.5	18.7
	max Face Velocity (fpm)	500	500	500	500	500
	max Air ∆P (in w.g.)	0.1	0.1	0.1	0.1	0.04
	EWT / LWT (°F)	180 / 160	180 / 160	180 / 160	180 / 160	180 / 160
	GPM	13.2	50.7	10.7	3.4	2.0
	max Water ∆P (in w.c.)	1	7	1	1	1
Re-Heat Coil:	max CFM	-	-	-	2,090	560
	EAT / LAT Dry Bulb (°F)	-	-	-	55 / 85	55 / 85
	MBH	-	-	-	68	20.6
	max Face Velocity (fpm)	-	-	-	500	500
	max Air ∆P (in w.g.)	-	-	-	0.1	0.04
	EWT / LWT (°F)	-	-	-	180 / 160	180 / 160
	GPM	-	-	-	6.8	2.2
	max Water ∆P (in w.c.)	-	-	-	4.4	1.0
Humidifier:	5 ft Section	H-1	H-2	H-3	H-4	-
Pre-Filters:	Type	Pleated	Pleated	Pleated	Pleated	Pleated
	Efficiency %	30	30	30	30	30
	∆P Clean/Change (in w.g.)	0.25 / 0.75	0.25 / 0.75	0.25 / 0.75	0.25 / 0.75	0.25 / 0.75
Final Filters:	Type	Cartridge	Cartridge	Cartridge	Cartridge	Cartridge
	Efficiency %	85	85	85	85	85
	∆P Clean/Dirty (in w.g.)	0.75 / 1.25	0.75 / 1.25	0.75 / 1.25	0.75 / 1.25	0.75 / 1.25

Fan Schedule

Fan										Motor		
Equip. No.	Location	Service	Design CFM	S.P. (in w.g.)	Type	max RPM	Volume Control	Drive	max BHP	HP	RPM	V/PH/Hz
SF-1	Boiler Room	Ventilation/ Combustion	1,500	1	Inline Centrifugal	1,214	Constant	Belt	0.44	1/2	1,725	120/1/60
EF-1,2,3	Roof	Lab	2,465	2.65	Mixed Flow Induced Dilution	1,770	VFD	Direct	2.68	3	1,725	460/3/60
EF-4	Mech Room	Toilets/ Janitor Closets	1,185	0.75	Inline Centrifugal	1,125	Constant	Belt	0.28	1/3	1,725	120/1/60
R-5	Mech Room	AHU-5	560	0.75	Inline Centrifugal	1,644	Constant	Belt	0.23	1/2	1,725	460/3/60

Boiler Schedule

	B-1	B-2
Design:		
LWT (°F)	180	180
Boiler HP	44	44
MBH	1467	1500
EWT (°F)	160	160
Operating Pressure (psig)	18	18
min Efficiency at Rated Output	80%	80%
Burners:		
Type	Natural Gas	Natural Gas
Supply Pressure (psig)	7	7
Type	#2 Fuel Oil	#2 Fuel Oil
gal/hr	10.7	10.7
Supply Pressure (psig) max	3	3
Blower:		
HP*	3/4	3/4
Combustion CFM	750	750
V/PH/Hz	460/3/60	460/3/60
Oil Pump:		
gal/hr	35	35

*Includes oil pump horsepower

Air-Cooled Scroll Chiller Schedule

	CH-1	CH-2
Location	Outside	Outside
Service	AHUs	AHUs
Refrigerant	R407C	R407C
Capcity in Tons	105.5	105.5
max kW/Ton	1.3	1.3
Evaporator:		
max GPM	267.6	267.6
min GPM	165	165
EWT (°F)	55	55
LWT (°F)	45	45
ΔP (ft w.c.)	16.9	16.9
No. of Passes	2	2
Fouling Factor	0.0001	0.0001
Condenser:		
Fan Type	Prop	Prop
No. of Fans	8	8
Drive	Direct	Direct
Design EAT (°F)	95	95
Total kW	14.4	14.4
Compressors:		
No. of Compressors	6	6
Steps per Compressor	1	1
Full Load Amps	215.6	215.6
max Inrush	225.0	225.0
V/PH/Hz	460/3/60	460/3/60

30% Propylene Glycol

Expansion Tank Schedule

Equip. No.	Location	Service	Nominal Tank Size (gal)	min Tank Accept. Vol. (gal)	min Operating Temp (°F)	max Operating Temp (°F)	max Operating Pressure (psig)	Remarks
ET-1	Mech Room	Heating Hot Water	45	22	160	180	40	Vertical Tank
ET-2	Mech Room	Chilled Water	10	2.5	40	75	45	Vertical Tank

Supply Air Terminal Unit Schedule (SB)

VAV Box					Reheat Coil***		
Equip. No.	max CFM	min CFM*	ΔP (in w.g.)**	Nominal Inlet Size (in)	min MBH	GPM	max ΔP (ft w.c.)
SB5	200	80	0.5	5	4.3	0.4	1.6
SB6	400	125	0.5	6	7.1	0.7	1.6
SB8	700	210	0.5	8	12.9	1.3	1.6
SB10	1,205	360	0.5	10	22.3	2.2	1.6
SB12	1,800	510	0.5	13x10 oval	33.2	3.3	1.6
SB14	2,300	690	0.5	16x10 oval	30.0	3.0	1.6

*Actual minimum airflows are to be set at 40% of the box maximum
**Discharge static pressure at max CFM
***Only where shown on drawings. Based on 180F EWT and 160F LWT.
Coil capacity and flow on drawings override schedule.

Supply Air Terminal Unit Schedule (SB) (continued)

External Sound Attenuator

Equip. No.	max Face Velocity (fpm)	max ΔP (in w.g.)	WxHxL (in)	125* Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
SB5	700	0.06	10x10x36	6	10	17	23	18	13
SB6	700	0.06	10x10x37	6	10	17	23	18	13
SB8	700	0.06	12x12x36	6	11	17	18	14	11
SB10	700	0.06	21x12x36	6	11	17	21	16	12
SB12	700	0.06	12x31x36	6	11	17	18	14	11
SB14	700	0.06	40x12x36	6	10	17	23	18	13

*Minimum Insertion Loss (dB)

Laboratory Air Valve Schedule

VAV Valve*

Equip. No.	max CFM	min CFM	ΔP (in w.g.)	Nominal Inlet/Outlet Size (in)
SV8	550	35	0.6	8
SV12	3,600	90	0.6	12
SV312	1,200	270	0.6	36x12
EV8	550	35	0.6	8
EV12	1,200	90	0.6	12
EV212	2,400	180	0.6	24x12

*Reheat coils as shown on drawings.
Based on 180F EWT and 160F LWT.
Max water ΔP=2ft, max air ΔP=2in

Laboratory Air Valve (VAV) Schedule (continued)

External Sound Attenuator

Equip. No.	max CFM	max Face Velocity (fpm)	max ΔP (in w.g.)	Inlet/Outlet (in)	125* Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	Service
SV8	550	900	0.1	12x12	5	11	18	17	16	12	Supply
SV12	1,200	900	0.1	24x12	5	11	18	17	16	12	Supply
SV312	3,600	900	0.1	46x12	5	11	18	17	16	12	Supply
EV8	550	-900	0.1	12x12	6	12	19	17	16	11	General Exhaust
EV12	1,200	-1,200	0.07	12x12	5	10	14	9	8	6	Fume Hood Exhaust
EV12	1,200	-900	0.1	24x12	6	12	19	17	16	11	General Exhaust
EV212	2,000	-1,000	0.07	24x12	5	10	14	9	8	6	Fume Hood Exhaust
EV212	2,000	-1,000	0.1	24x12	6	12	19	17	16	11	General Exhaust

*Minimum Insertion Loss (dB)

Sound Attenuator Schedule

Equip. No.	Location	WxHxL (in)	Face Vel (fpm)	max ΔP (in w.g.)	Insertion Loss (dB) 63 Hz	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	8000 Hz
SA-1S	AHU-1	64x46x36	500	0.12	7	14	19	18	25	23	18	16
SA-1R	AHU-1 Return Duct	26x38x36	1250	0.17	4	8	13	18	18	14	11	9
SA-2S	AHU-2 Supply Duct	26x24x60	2000	0.21	3	6	13	22	20	16	12	9
SA-2E	EF-1,2,3 Inlet Exhaust	50x18x60	1250	0.21	11	8	17	18	12	11	9	8
SA-2B	EF-1,2,3 Inlet Bypass	24x18x36	1053	0.60	8	12	20	24	29	23	14	11
SA-3S	AHU-3	64x50x36	500	0.12	7	14	19	18	25	23	18	16
SA-3R	AHU-3 Return Duct	50x26x36	1250	0.17	4	8	13	18	18	14	11	9
SA-4S	AHU-4 Supply Duct	16x20x36	1250	0.17	4	8	12	14	17	15	13	11
SA-4R	AHU-4 Return Duct	16x20x36	1250	0.17	5	10	14	15	18	14	13	11

Laboratory Airflow Table

Condition	Supply		Fume Hood Exhaust					General Exhaust		Transfer	Pressurization
	SV312	SV312	EV212	EV12	EV12	EV12	EV12	EV212	EV12		
min cool / min exh	920	830	-560	-315	-315	-315	-315	-180	-90	280	-60
max cool/ min exh	2460	2060	-560	-315	-315	-315	-315	-1950	-1090	280	-60
max cool/ max exh	3260	2745	-1855	-1055	-1055	-1055	-1055	-180	-90	280	-60
min cool/ max exh	3260	2745	-1855	-1055	-1055	-1055	-1055	-180	-90	280	-60

Notes: All values in CFM. (+) indicates flow into the space. (-) indicates flow out of the space.

Pump Schedule

					Pump					Motor			
Equip. No.	Service	Type	Liquid	GPM	Total Head (ft w.c.)	min. % Efficiency	max BHP	Suction (in)	Discharge (in)	HP	RPM	V/PH/Hz	Remarks
CHWP-1	CHW Primary	Base Mounted Centrifugal	30% Propylene Glycol	201	129	57	11.8	3	2	15	1750	460/3/60	VFD
CHWP-2	CHW Primary	Base Mounted Centrifugal	30% Propylene Glycol	201	129	57	11.8	3	2	15	1750	460/3/61	VFD
HWP-1	Heating HW	Inline Centrifugal	Water	147	65	65	3.7	2.5	2	5	1750	460/3/62	VFD
HWP-2	Heating HW	Inline Centrifugal	Water	147	65	65	3.7	2.5	2.5	5	1750	460/3/63	VFD
P-1	AHU-1 Recirculating	Inline Centrifugal	Water	13.2	2.0	-	-	-	2.5	FRAC	1750	120/1/60	-
P-2	AHU-2 Recirculating	Inline Centrifugal	Water	50.7	10	53	0.24	-	-	0.33	1750	120/1/61	-
P-3	AHU-5 Recirculating	Inline Centrifugal	Water	2.0	2.0	-	-	-	-	FRAC	1750	120/1/62	-
FOP-1	Duplex Fuel Oil Pump	Direct Drive Positive Displacement	#2 Fuel Oil	2.58	115	-	-	0.5	0.5	0.5	1725	460/3/60	-
FOP-2	Fuel Oil Pump	Direct Drive Positive Displacement	#2 Fuel Oil	4.7	115	-	-	-	-	0.75	1725	208/3/60	-

Air Conditioning Unit Schedule

Equip. No.	Fan CFM	Drive	External S.P.	Motor HP	Motor RPM	Cooling Refrig.	Cooling Sens. MBH	Cooling Total MBH	Cooling EAT (°F) DB/WB	Heating Capacity kW	Heating EAT (°F) DB/WB	Electric MCA	Electric V/PH/Hz
ACU-1	2,200	Direct	0.5	1	1750	R407C	34.9	-	80/67	5.1	80/67	18.4	460/3/60
ACU-2	900	Direct	0.4	1/4	1750	R407C	19.4	-	75/63	5.1	75/63	9.8	460/3/60
ACU-3	1,000	Direct	0.4	1/3	1750	R407C	20.3	-	80/67	-	-	2	460/3/60

Air-Cooled Condensing Unit Schedule

Equip. No.	Location	Service	MBH	Design OA (°F)	Refrig.	Electric MCA	Electric V/PH/Hz
ACCU-1	Outside	ACU-1	70	95	R407C	18.4	460/3/60
ACCU-2	Outside	ACU-2	31.5	95	R407C	9.8	460/3/60
ACCU-3	Outside	ACU-3	40.5	95	R407C	2	460/3/60

Ultrasonic Humidifier Schedule

Equip. No.	Service AHU-#	Supply CFM	OA CFM	Capacity (lbs/hr)	max air ΔP (in w.g.)	Electric A	Electric V/PH/Hz
H-1	1	4,940	2,280	61.2	0.1	8	460/1/60
H-2	2	8,040	8,040	171.6	0.1	13	460/1/60
H-3	3	4,350	1,310	30.8	0.1	4	460/1/60
H-4	4	2,090	880	21.1	0.1	4	460/1/60

Notes: All humidifiers have a maximum absorption distance of 30".

Unit Heater and Cabinet Unit Heater - Hot Water - Schedule

Equip. No.	Location	Type	Fan CFM	Throw (ft)	Motor HP	Motor V/PH/Hz	Motor RPM	Coil MBH	Coil EAT (°F)	Coil LAT (°F)	Coil EWT (°F)	Coil GPM	Coil max Water ΔP (ft w.c.)
CUH-1	Corridor	Ceiling Recessed	420	-	FRAC	120/1/60	1050	33.0	60	141	180	3.3	2
CUH-2	Entry Vestibule	Wall Mounted	420	-	FRAC	120/1/60	1050	33.0	60	141	180	3.3	2
CUH-3	Main Vestibule	Wall Recessed	420	-	FRAC	120/1/60	1050	50.8	60	141	180	3.3	2
CUH-4	Stair	Wall Mounted	230	-	FRAC	120/1/60	1050	24.1	60	141	180	2.5	2
CUH-5,6	Shower	Wall Mounted	170	-	FRAC	120/1/60	1050	11.1	60	141	180	3.0	2
UH-1	Various	Horizontal	630	25	FRAC	120/1/60	-	30.0	65	120	180	3.75	5

Schedule GS-2
INTERMEDIATE GENERAL SERVICE

I. APPLICABILITY

Except as modified herein, this schedule is applicable only to a non-residential Customer who elects to receive Electricity Supply Service and Electric Delivery Service from the Company and who has within the current and previous 11 billing months at least three peak measured demands of 30 kW or more and not more than two peak measured demands of 500 kW or more.

For a Customer served under this schedule whose peak measured demand has decreased to less than 30 kW, this schedule shall remain applicable to the Customer and the Customer shall not have the option to purchase electricity under Schedule GS-1 until such time the maximum measured demand has remained at less than 30 kW during all billing months within the current and previous 11 billing months.

At such time the Customer no longer meets the above applicability requirements, the Customer will remain on this schedule for the period (not exceeding two additional billing months) required to achieve an orderly transfer to the applicable schedule.

For new service, this schedule is applicable when the anticipated kW demand meets the above criteria.

II. 30-DAY RATE

A. Non-Demand Billing

1. Distribution Service Charges

a. Basic Customer Charge
Basic Customer Charge $21.17 per billing month.

b. Plus Distribution kWh Charge
All kWh @ 2.433¢ per kWh

2. Electricity Supply Service Charges

a. Electricity Supply kWh Charge
1) For the billing months of June – September
All kWh @ 4.795¢ per kWh

2) For the billing months of October – May
All kWh @ 4.075¢ per kWh

(Continued)

Schedule GS-2
INTERMEDIATE GENERAL SERVICE

(Continued)

II. 30-DAY RATE (Continued)

2. Electricity Supply Service Charges (Continued)

b. Each Electricity Supply kilowatthour used is subject to Fuel Charge Riders A and B.

B. Demand Billing

1. Distribution Service Charges

a. Basic Customer Charge
Basic Customer Charge $21.17 per billing month.

b. Distribution Demand Charge

All kW of Demand	@	$ 3.387 per kW

2. Electricity Supply Service Charges

a. Electricity Supply Demand Charge

1) For the billing months of June – September

All kW of Demand	@	$ 2.844 per kW

2) For the billing months of October – May

All kW of Demand	@	$1.406 per kW

b. Plus Electricity Supply kWh Charge

First 150 kWh per kW	@	4.617¢ per kWh
Next 150 kWh per kW	@	2.588¢ per kWh
Next 150 kWh per kW	@	1.119¢ per kWh
Additional kWh	@	0.272¢ per kWh

c. Each Electricity Supply kilowatthour used is subject to Fuel Charge Riders A and B.

(Continued)

Schedule GS-2
INTERMEDIATE GENERAL SERVICE

(Continued)

II. 30-DAY RATE (Continued)

C. The minimum charge shall be the highest of:

1. The Basic Customer Charge in Paragraph II.A.1.a. or II.B.1.a., whichever is applicable.

2. The amount as may be contracted for.

3. The sum of the charges in Paragraph II.A. or II.B., whichever is applicable, plus $1.480 multiplied by the number of kW by which any minimum demand established exceeds the demand determined under Paragraph IV.

4. If the demand determined under Paragraph IV is 50 kW or greater, the minimum charge for Non-Demand Billing under Paragraph II. A. shall not be less than $3.13 per kW of demand determined.

III. NON-DEMAND BILLING VS. DEMAND BILLING

A. The non-demand billing charges of Paragraph II.A. apply to customers whose kWh usage for the current month does not exceed 200 kWh per kW of the demand as determined under Paragraph IV.

B. The demand billing charges of Paragraph II.B. apply to customers whose kWh usage for the current month exceeds 200 kWh per kW of the demand as determined under Paragraph IV.

IV. DETERMINATION OF DEMAND

The kW of demand will be determined as the highest average kW load measured in any 30-minute interval during the billing month.

V. MINIMUM DEMAND

The minimum demand shall be such as may be contracted for, however:

A. When the kW demand determined has reached or exceeded 500 kW during the current or preceding eleven billing months, the minimum demand shall not be less than the highest demand determined during the current and previous eleven billing months.

V. MINIMUM DEMAND (Continued)

(Continued)

Schedule GS-2
INTERMEDIATE GENERAL SERVICE

(Continued)

B. When the Customer's power factor is less than 85 percent, a minimum demand of not less than 85 percent of the Customer's maximum kVA demand may be established.

VI. METER READING AND BILLING

A. Meters may be read in units of 10 kWh and bills rendered accordingly.

B. When the actual number of days between meter readings is more or less than 30 days, the Basic Customer Charge, the Distribution Demand Charge, the Electricity Supply Demand, the quantity of kWh in the first three blocks of the Demand Billing Electricity Supply kWh Charge and the minimum charge of the 30-day rate will each be multiplied by the actual number of days in the billing period and divided by 30.

VII. STANDBY, MAINTENANCE OR PARALLEL OPERATION SERVICE

A Customer requiring standby, maintenance or parallel operation service may elect service under this schedule provided the Customer contracts for the maximum kW which the Company is to supply. Standby, maintenance or parallel operation service is subject to the following provisions:

A. Suitable relays and protective apparatus shall be furnished, installed, and maintained at the Customer's expense in accordance with specifications furnished by the Company. The relays and protective equipment shall be subject, at all reasonable times, to inspection by the Company's authorized representative.

B. In case the maximum kW demand determined in Paragraph IV. or the minimum demand determined in Paragraph V. exceeds the contract demand, the contract demand shall be increased by such excess demand.

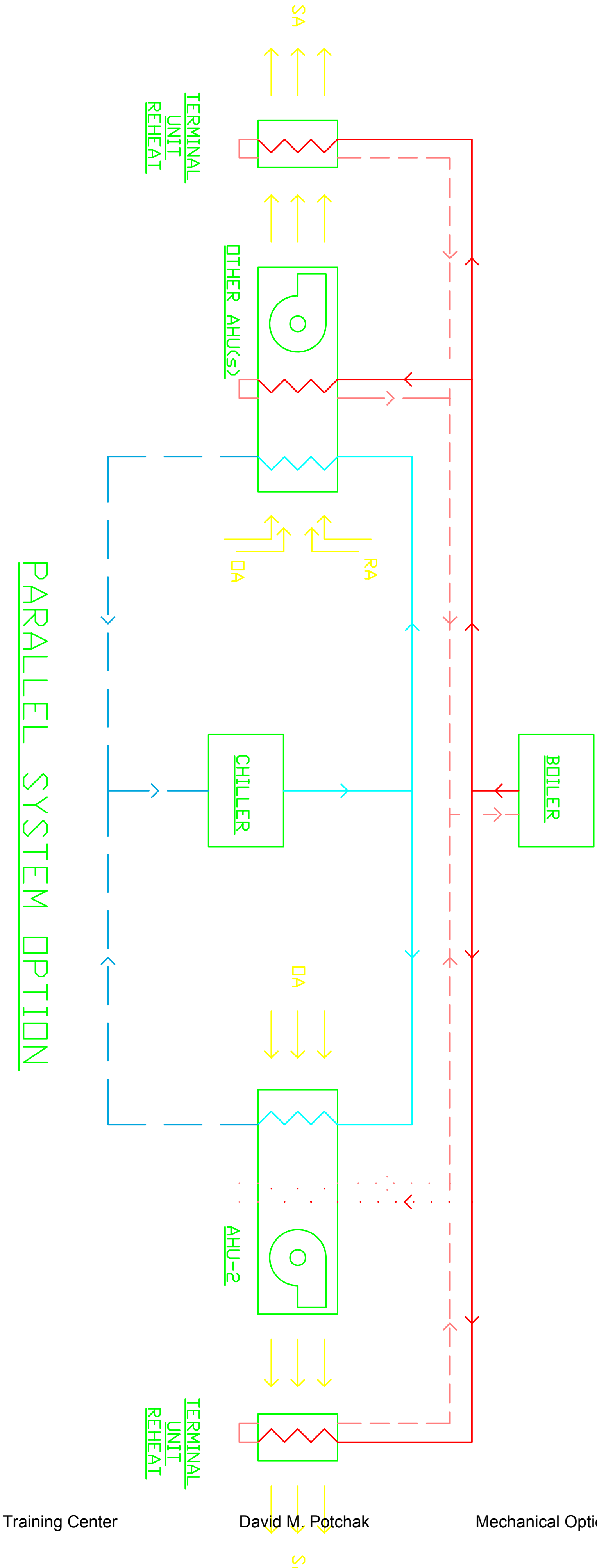
C. The demand billed under Paragraph II.B.2.a.1) or II.B.2.a.2) shall be the contract demand.

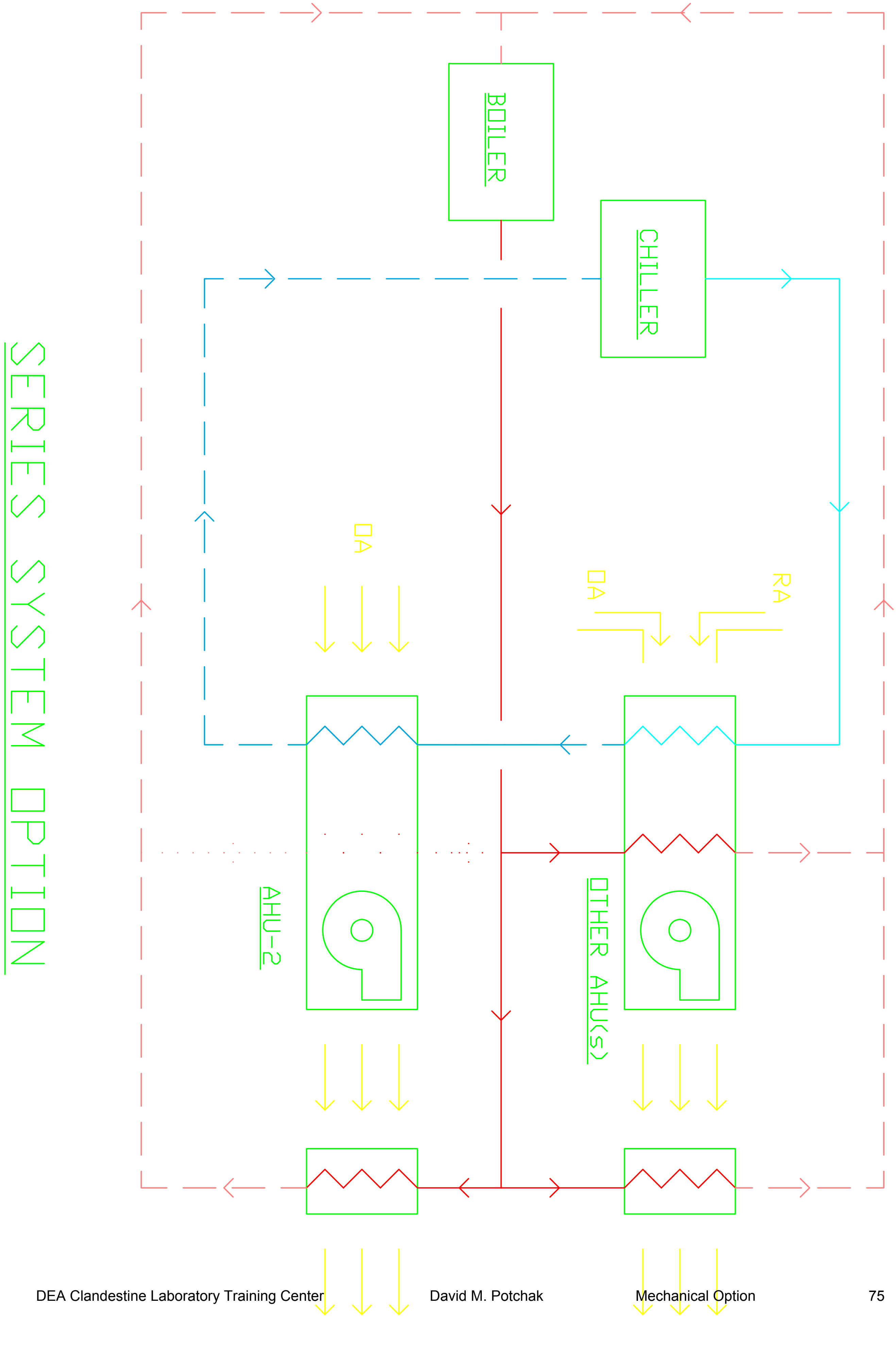
Schedule GS-2
INTERMEDIATE GENERAL SERVICE

(Continued)

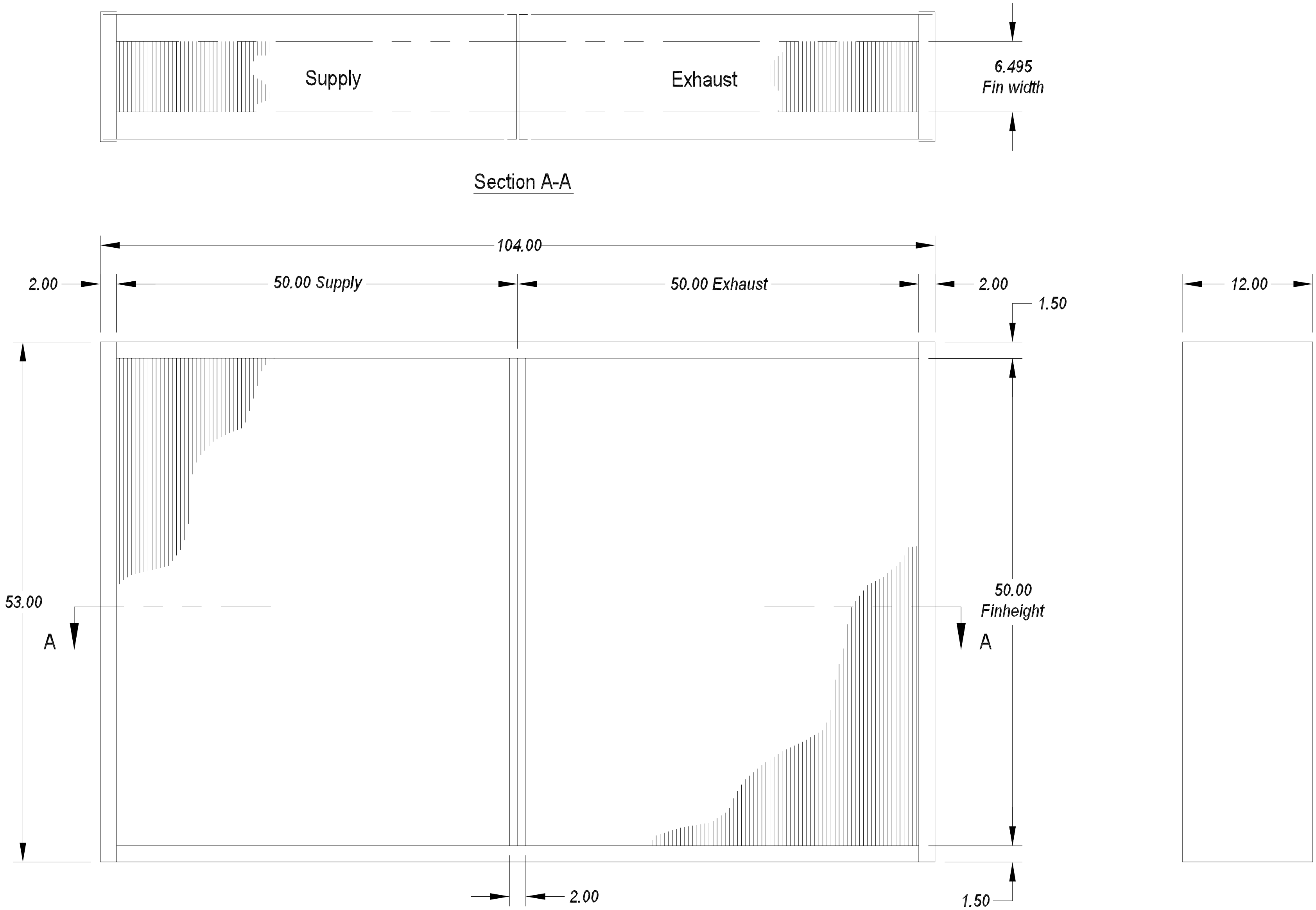
VIII. TERM OF CONTRACT

The contract shall be open order unless (a) standby, maintenance or parallel operation service is provided, or (b) the Customer or the Company requests a written contract. In such cases, the term of contract for the purchase of electricity under this schedule shall be as mutually agreed upon, but for not less than one year. During the minimum term of applicability, the Customer may be billed under the corresponding Unbundled Rate Schedule GS-2U, if applicable.


SA
TERMINAL UNIT REHEAT
OTHER AHU(S)
OA
RA
PARALLEL SYSTEM OPTION
CHILLER
BOILER
OA
AHU-2
TERMINAL UNIT REHEAT
SA


SERIES SYSTEM OPTION
BOILER
CHILLER
OA
OA
RA
AHU-2
OTHER AHU(S)

Coil Nomenclature		
Tubes	Fins	Frame
Diameter: 1/2" 6 Rows	FPI: 12 Coating: None	Box Type: Side Plate
Material: Copper	Material: Aluminium	Material: 16 ga Galvanized Steel
Surface: Rifled	Surface: Sine Wave	



Heat Pipe Technology, Inc.
4340 NE 49th Ave., Gainesville, Florida 32609
Phone: (352)367-0999 Fax: (352)367-1688

HRM-6R 050.00 x 100.00 (Approx. Weight: 684 lbs) Penn St U		
Date: 3/27/2007 10:43:26 AM	Drawn By: HPT 4.1F Software	Scale: N/A
Approved:	Project No:	Dwg No:

ENERGY RECOVERY HEAT PIPE BIN ANALYSIS

Heat Pipe Technology, Inc.

Project Name: Penn St U,
Date: 3/27/2007 By: Engineer:

	Supply	Exhaust
Air Flow (SCFM)	8,040	8,040
Fin Height (in.)	50.00	50.00
Finned Length (in.)	50.00	50.00
Face Area (SF)	17.36	17.36
Face Velocity (SFPM)	463	463
Max Pressure Drop (in. WC)	0.57	0.55 Dry 0.72 Wet
No. Rows of Tubes(1/2"OD)	6	6
Fin Density, FPI	12	12
Fin Material	Aluminum	Aluminum
Fin Type	Standard	Standard
Desired Leaving Temps.	55°F	36°Fmin

Heat pipe installed as retofit

Cooling EER	10.
Electric Rate	8.0 ¢/kwh
Heating	Gas/Fuel
Gas/Fuel Rate	95.0 ¢/them
Burner Efficiency	75.0%
Motor Efficiency	90.0%
Fan Efficiency	70.0%
Evaporative Cooling	No
Refrigerant	R-22

Bin: Williamsport, PA, US @ 24-7, All Days, All Hours, All Months. *File: pennst.hbn*

Outside Air Supply Entering			Supply Air Lvg	Exhaust Air Ent	Exhaust Air Lvg	Heat Pipes Heat Recovery		Run Time	Savings	Fan Cost	Net Savings
HeatPipe CFM	DB Bin °F/°F	MCWB °F	DB/WB °F/°F	DB/WB °F/°F	DB/WB °F/°F	Eff %	Rate Btuh	h/Yr	$/Year	$/Year	$/Year
Cooling Recovery:											
8,040	95/100	76.88	82.7/72.9	71/57	86.1/62.5	55.9	132,345	8	$8	$1	$7
8,040	90/95	76.26	80.5/73.0	71/57	83.2/61.5	55.9	107,418	17	$15	$2	$12
8,040	85/90	71.42	78.2/68.7	71/57	80.4/60.5	56.1	82,272	71	$47	$9	$37
8,040	80/85	69.42	76.0/67.4	71/57	77.5/59.5	56.1	57,325	342	$157	$45	$112
8,040	75/80	67.32	73.9/66.2	71/57	74.7/58.4	56.1	32,392	466	$121	$61	$60
8,040	70/75	64.93	71.7/64.7	71/57	71.8/57.3	56.2	7,472	654	$39	$85	-$46
Economizer Mode:											
0	65/70	62.63	67.5/62.6	71/57	71.0/57.0	0.0	0	670	$0	$43	-$43
0	60/65	58.57	62.5/58.6	71/57	71.0/57.0	0.0	0	1009	$0	$65	-$65
0	55/60	52.93	57.5/52.9	71/57	71.0/57.0	0.0	0	689	$0	$44	-$44
Modulated Economizer Mode:											
1,099	50/55	47.89	55.0/49.0*	71/57	68.5/56.0	13.5	21,952	654	$182	$42	$140
2,913	45/50	43.07	55.0/46.7*	71/57	63.5/54.0	31.9	65,713	600	$499	$41	$458
4,687	40/45	38.56	55.0/44.9*	71/57	58.6/52.0	43.8	109,314	537	$744	$42	$701
6,595	35/40	34.28	55.0/43.6*	71/57	53.6/49.8	52.2	152,888	978	$1,894	$97	$1,797
Full Heating Recovery:											
8,040	30/35	29.54	54.3/41.7	71/57	49.3/47.9	56.7	190,627	710	$1,714	$88	$1,626
8,040	25/30	24.44	52.2/38.8	71/57	46.6/46.6	56.8	215,315	458	$1,249	$57	$1,193
8,040	20/25	19.93	50.3/36.8	71/57	45.1/45.1	57.3	241,912	306	$938	$39	$899
8,040	15/20	15.46	48.4/35.0	71/57	43.6/43.6	57.8	269,057	318	$1,084	$41	$1,042
8,040	10/15	10.75	46.6/33.2	71/57	42.0/42.0	58.3	296,553	164	$616	$22	$594
8,040	5/10	6.07	44.8/31.6	71/57	40.4/40.4	58.7	324,301	81	$333	$11	$322
8,040	0/5	1.75	43.0/30.4	71/57	38.7/38.7	59.1	352,245	27	$120	$4	$117
8,040	-5/0	-2.56	41.2/29.3	71/57	37.0/37.0	59.5	380,297	1	$5	$0	$5
							Totals:	8,760	$9,764	$841	$8,924

* Recombined supply air

AIRotor
by XeteX

Air-to-Air Heat Recovery

XeteX Inc.
3530 East 28th Street
Minneapolis, MN 55406
(612) 724-3101
(612) 724-3372 Fax

General

The AIRotor heat recovery unit is a rotary heat exchanger which operates on the air-to-air principle of heat transfer and has the following features:

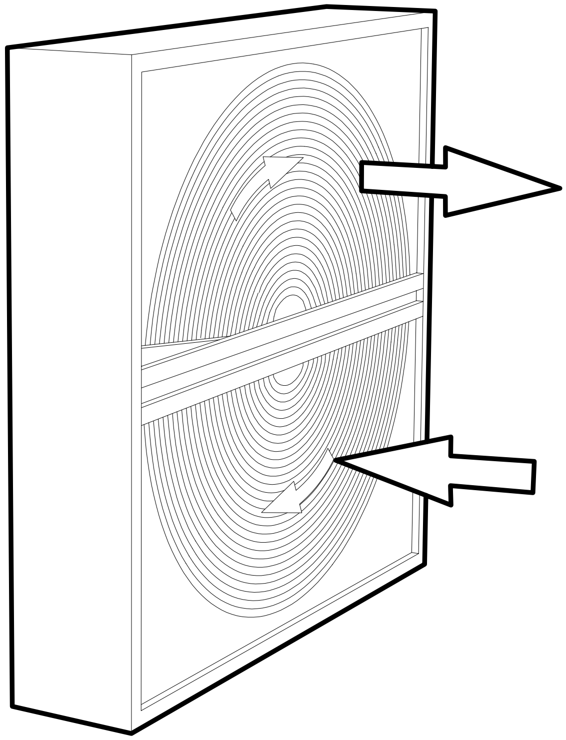
- Available in 16 sizes, with a nominal flow range of 500-28,000 cfm.
- Total energy recovery efficiencies as high as 90%.
- Rotor has smooth air channels to ensure a low pressure drop and reduce the risk of fouling.
- Rotor surface is manufactured absolutely smooth allowing for tight fitting seals between airstreams.
- Available with electronic speed control for variable rotor capacity.
- Hygroscopic rotor provides latent and sensible heat recovery.

Design

The RVA heat recovery unit is constructed from a rigid tubular steel welded frame, with insulated galvanized sheet metal cover plates and hatches. The frame is reinforced to prevent deflect of the rotor from static pressure drops to less than 0.03".

The rotor is assembled from alternate layers of flat and corrugated thin sheet aluminum. The smooth channels formed by this construction ensure that the air flow is laminar, thereby ensuring that the pressure drop is low and minimizing the risk of fouling by dirt or dust. Dry particles up to 900 microns shall pass freely through the rotor without clogging the media. The rotor media can be cleaned with low temperature steam without degrading unit performance.

The hygroscopic rotor equally transfers both sensible and latent heat. Moisture is transferred between airstreams in the vapor stage so media remains dry and no drain pan is required.



The rotor, which may be removed from the frame, is mounted in sealed permanently-lubricated spherical ball bearings. The bearings can be serviced or replaced without removing the rotor from the case.

The exchanger is sealed with brush seals between airstreams and around the perimetor of the rotor. Because of the the smooth rotor surface, the brush seals provides an extremely effective seal with very little contact pressure, resulting in extended service life.

An adjustable purging sector is provided to ensure continuous cleaning of the rotor and to virtually eliminate cross-contamination between the exhaust air the supply air.

The standard AIRotor heat recovery unit is supplied with a perimeter self adjusting drive belt and worm gear drive for on/off operation.

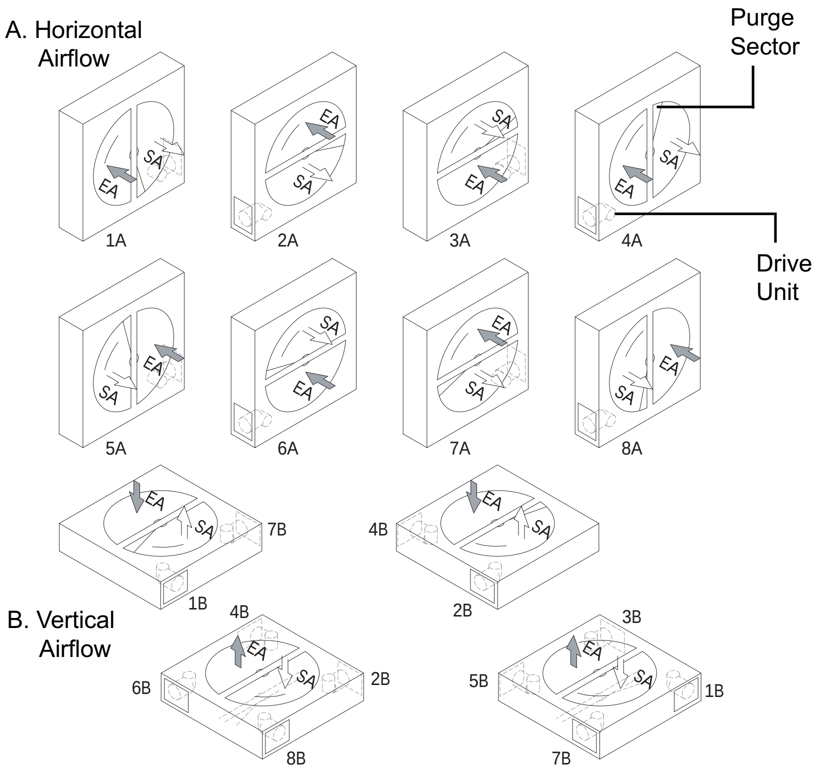
For installations where there is a requirement for controlling heat recovery capacity and/or rotor frost control, the heat recovery unit is equipped with an electronic control unit that varies rotor speed from maximum speed down to an automatic purge cycle of 1/20 rpm.

Specification

HRW RV(X)-a-b-c-d-e-f

RVB	0600, 0700, 0850, 1000, 1160	a
RVA	0600, 0700, 0850, 1000, 1250, 1500, 1750, 2000, 2250, 2500, 2750, 3000	a
Rotor Type	No = Non Hygros. Hy = Hygroscopic	b
Drive Unit	K = Constant Speed R = Electronic Speed Control (ESC)	c
Purge Sector	0 = Without 1 = With	d
Unit Config.	1, 2, 3, 4, 5, 6, 7, 8 (See Below)	e
Air Flow	A = Horizontal B = Vertical	f

Rotor Configuration



Specifications and dimensions are subject to change without notice.

Accessories

RVAT-x-x-x-x

Flanged Duct Connections	01
Epoxy Treated Rotor	02
Filter Sections	
2" Pleated	03
Washable Filters	04

Control Options

Constant Speed Drive

Speed Detector (w/Alarm Contact) 05
(Standard with Elect. Spd Control)

Electronic Speed Control

Frost Control	06
Economizer Control	07
Summer Changeover	08

Description of Controls

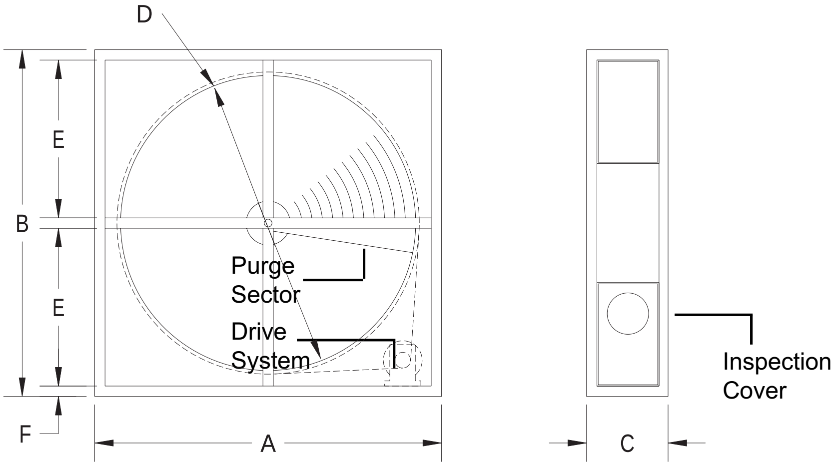
Frost Control monitors the exhaust temperature leaving exchanger and reduces rotor speed to prevent exhaust temperature from dropping below setpoint.

Economizer control monitors supply discharge temperature and reduces rotor speed to prevent discharge from rising above setpoint.

Summer Changeover control monitors outdoor air and return air temperatures and automatically switches rotor to maximum recovery speed when the outside air temperature is higher then the return air temperature.

TECHNICAL SPECIFICATIONS

Dimensions and Weights



MODEL #	DIMENSIONS						WEIGHT (lbs)
	A	B	C	D	E	F	
RVB-0600	24.43	31.00	14.25	18.90	14.10	0.70	90
RVB-0700	31.00	31.00	14.25	23.23	14.10	0.70	155
RVB-0850	36.00	36.00	14.25	29.13	16.60	0.70	190
RVB-1100	43.60	43.60	14.25	37.50	20.40	0.70	265
RVC-1300	53.54	54.72	14.96	46.46	22.25	1.58	350
RVC-1600	62.05	63.23	14.96	57.09	27.00	1.58	440
RVC-1900	76.77	77.95	14.96	70.87	32.10	1.58	640
RVC-2100	85.04	86.22	14.96	78.35	37.00	1.58	735
RVA-2250	88.58	88.58	17.32	81.10	40.75	2.36	880
RVA-2500	98.74	98.74	17.32	90.94	45.67	2.36	1035
RVA-2750	108.58	108.58	17.32	100.79	50.59	2.36	1210
RVA-3000	118.43	118.43	17.32	110.63	55.51	2.36	1365

AIRotor Drive System

AIRotors are driven with a belt around the perimeter of the rotor. An AC gear reduced motor with permanently sealed bearings is easily serviced through an access panel in the corner of the wheel housing. A spare belt can be provided with each wheel to reduce downtime in the event of belt failure.

Constant Speed Drive

The AIRotor constant speed drive is provided with On/Off dry contacts for control by a thermostat or building control system. An optional speed detector is available which closed a normally open contact when wheel stops turning for over 20 minutes.

Electronic Speed Control

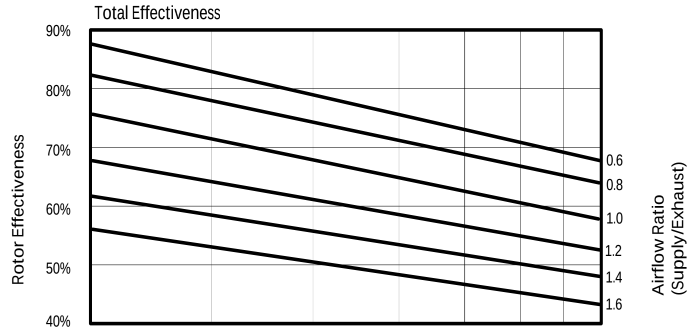
The AIRotor Electronic speed control consists of a motor control center and drive motor. The control center incorporates functions for purging, speed detection, motor protection and alarm. For speed control the control center is built to receive 0-10 VDC or 4-20 mA input from temperature controller.

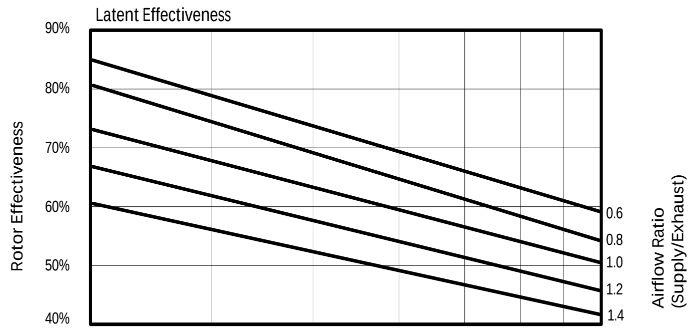
Frost, Economizer, & Summer/ Winter Changeover Control

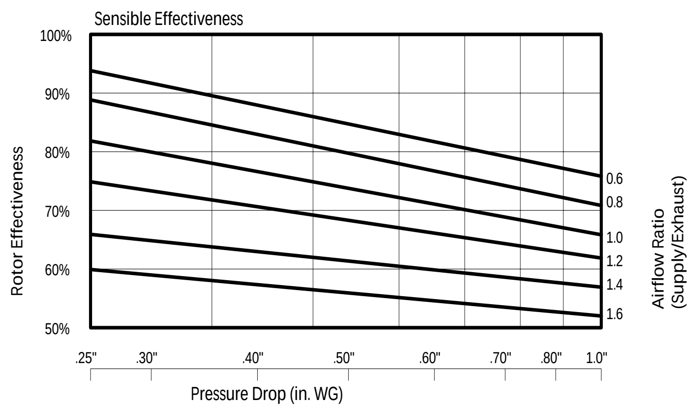
The AIRotor can be supplied with built-in temperature controller that automatically modulates rotor speed to prevent frost build-up, reduce heat recovery to prevent overheating space (economizer), and switch to maximum recovery during the summer (W/S Changeover). AIRotor is supply with integral control panel, digital temperature readout, and four remote mounted temperature sensors.

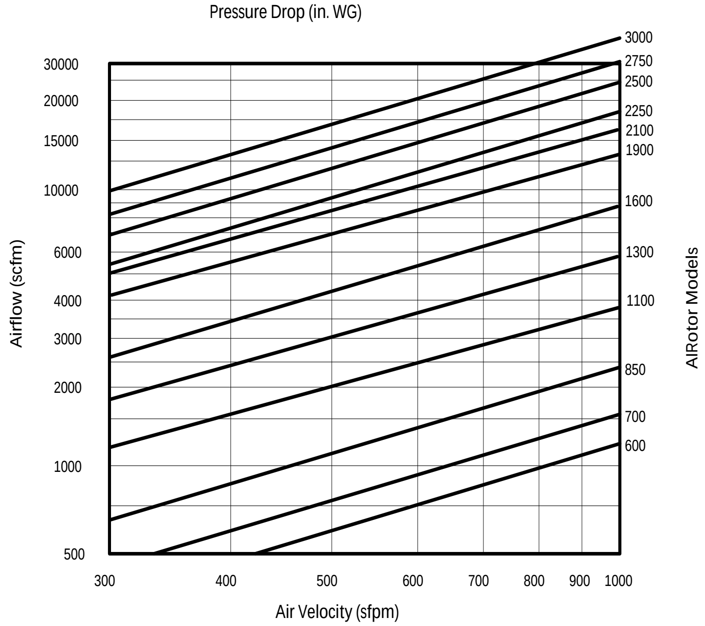
Specifications and dimensions are subject to change without notice.

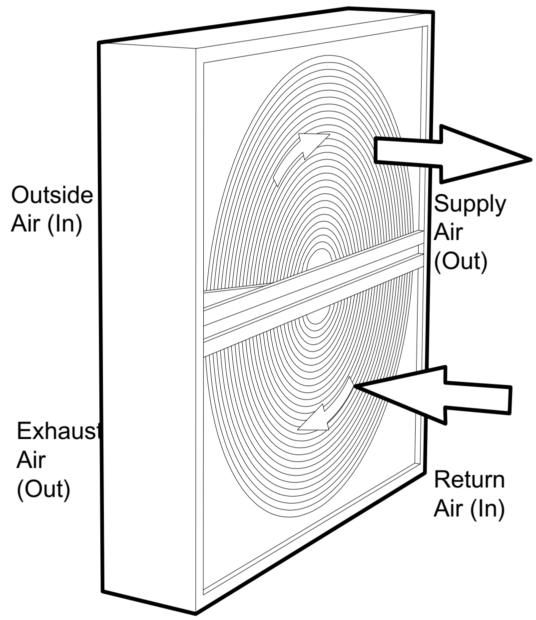
PERFORMANCE CHARTS









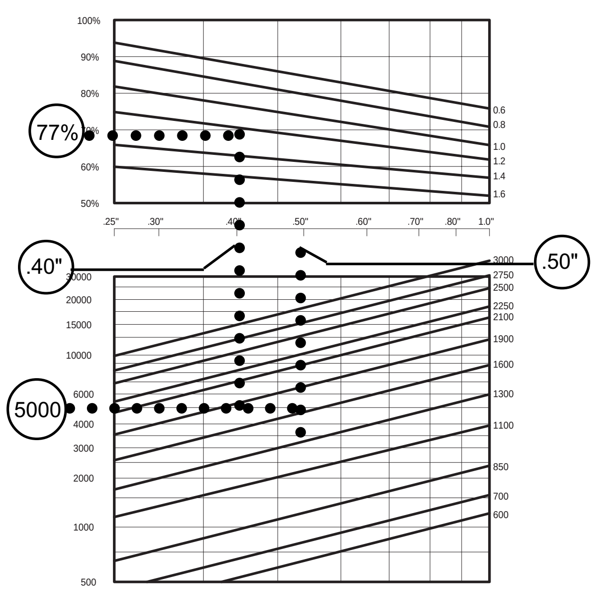


Example

Given the following conditions:
Supply Airflow = 5000 cfm
Return Airflow = 5000 cfm
Maximum Pressure Drop = 0.5" WG

Select AIRotor 1750

From the charts:
Pressure Drop = 0.40"
Effectiveness = 77%

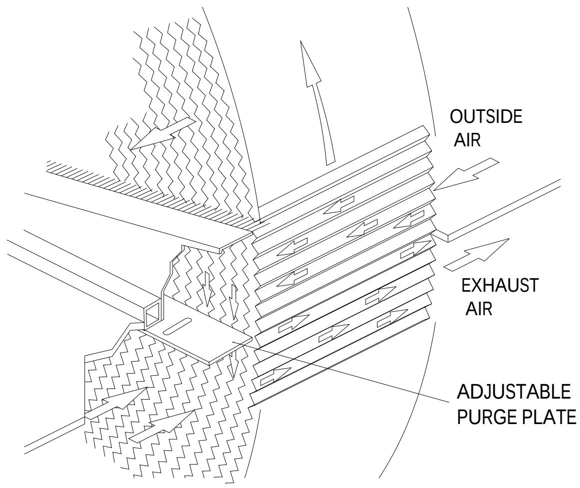


Specifications and dimensions are subject to change without notice.

PURGING AND LEAKAGE AIRFLOW

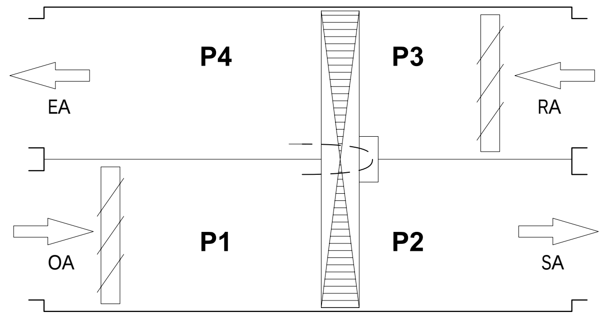
In rotary heat exchangers a certain amount of leakage inevitably takes place, in both directions, between the supply air and exhaust air sides, the leakage air being transferred by the rotor.

The purging sector is used to clean the rotor to eliminated leakage from the exhaust air to the supply air side. A detail of the purge sector is shown below.



Purge Sector Detail

When installing a unit provided with a purging sector, the fans should be located so that P1 > P4 and
P2 > P3, as shown in the figure below. If required, an adjusting damper may be used to obtain the required pressure balance.



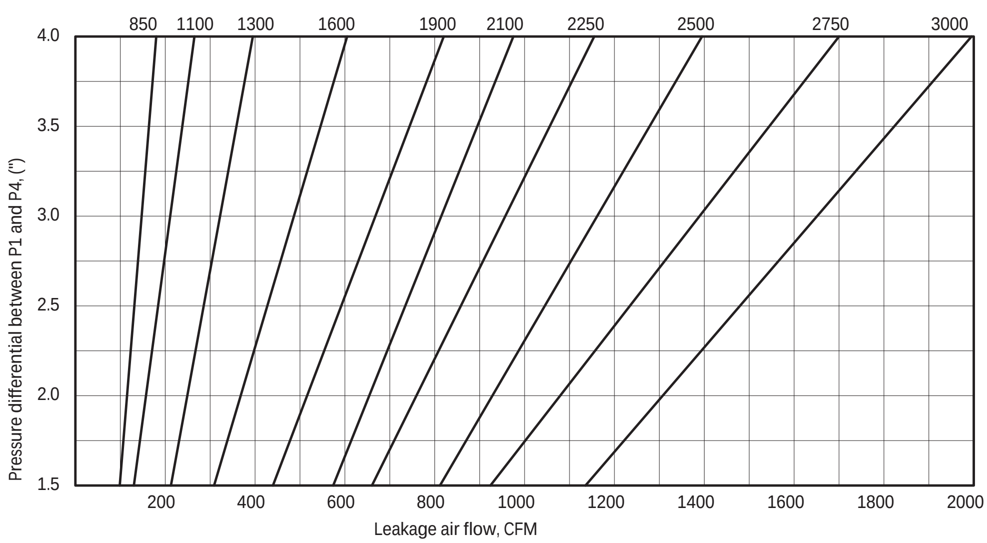
Purge Schematic

The chart below shows the leakage flow throught the purging sector. Allowance for high differential pressures should be made when selecting the fan.

Purge Airflow Chart

AIRotor Model

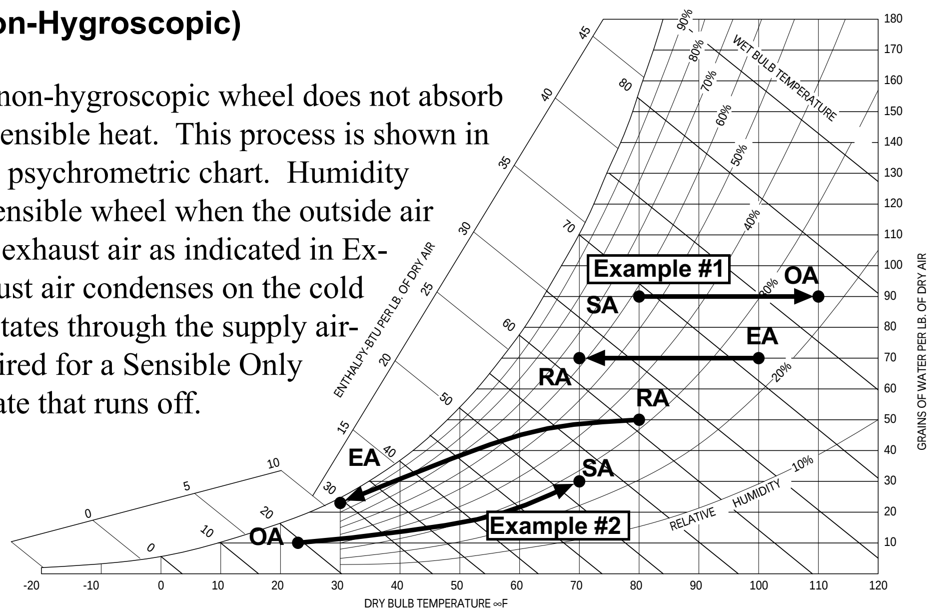
Approximate leakage through the purging sector and seals
HYGROSCOPIC ROTOR @ 20 RPM



PSYCHROMETRIC ANALYSIS

Sensible Only Wheel (Non-Hygroscopic)

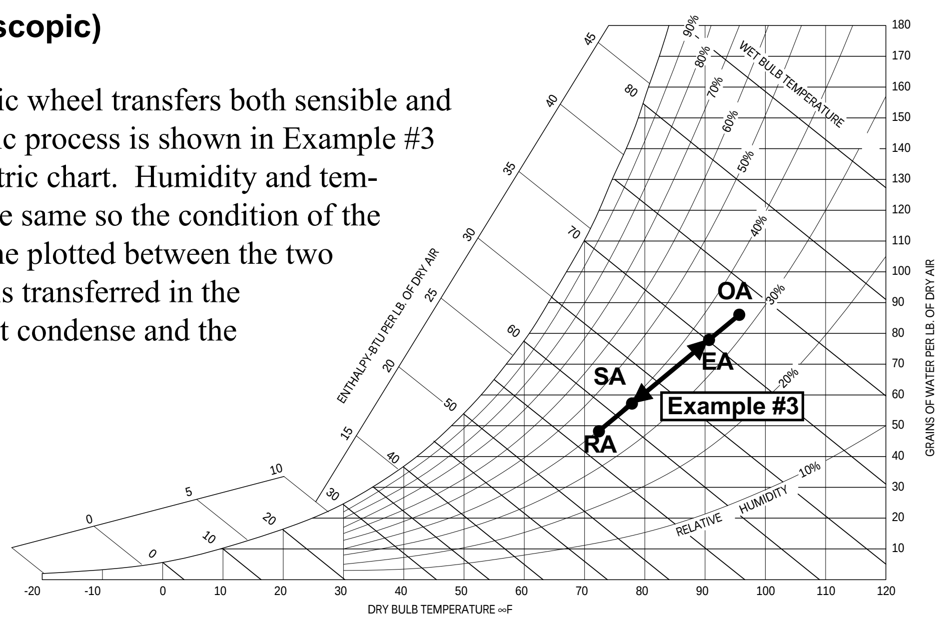
The Sensible Only wheel or non-hygroscopic wheel does not absorb moisture and transfers only sensible heat. This process is shown in Example #1 on the adjoining psychrometric chart. Humidity transfer only occurs with a sensible wheel when the outside air is below the dewpoint of the exhaust air as indicated in Example #2. The warmer exhaust air condenses on the cold wheel and evaporates as it rotates through the supply airstream. A drain pan is required for a Sensible Only wheel to collect the condensate that runs off.



SA = Supply Air
RA = Return Air
OA = Outside Air
EA = Exhaust Air

Enthalpy Wheel (Hygroscopic)

The Enthalpy or Hygroscopic wheel transfers both sensible and latent heat. This hygroscopic process is shown in Example #3 on the adjoining psychrometric chart. Humidity and temperature effectiveness are the same so the condition of the air varies along a straight line plotted between the two inlet conditions. Humidity is transferred in the vapor stage so the air doesn't condense and the wheel stays dry.



GENERAL SPECIFICATION

ROTARY AIR-TO-AIR HEAT EXCHANGER

Furnish an "AIRotor" rotary air-to-air heat exchanger manufactured by XeteX, Inc. Exchanger shall include hygroscopic rotor, constant or variable speed drive, rotation detector with alarm connection, and speed controller with temperature sensors.

ENTHALPY RECOVERY WHEEL

Exchanger shall be constructed of alternate Layers of corrugated and flat aluminum sheet material. Both sides of the exchanger shall be completely smooth with less then 0.005" variation between alternate layers to allow for optimum sealing surface for brush seals. The rotor shall have smooth air channels to ensure laminar airflow for low pressure drops. Dry particles up to 900 microns shall pass freely through the rotor without clogging the media. The rotor media shall be capable of being cleaned with low temperature steam without degrading unit performance. The rotor media must be made of aluminum which is coated to prohibit corrosion. All surfaces shall be coated with a nonmigrating adsorbent specifically developed for the selective transfer of water vapor.

* Verification in writing must be presented from independent laboratory evaluations confirming that the desiccant adsorbent surface does freely transmit water vapor without detectable gaseous cross-contamination. Specially formulated aluminum compound of "Micro-Sieve" shall permanently bond the selective adsorbent desiccant to the hygroscopic (enthalpy) recovery AIRotor by XeteX.

* Sensible and latent recovery efficiencies must be clearly documented through a certification program conducted in accordance with ASHRAE 84-1991 and ARI 1060 standards that verify actual performance to be *independent phenomena and there is no reason to expect that ...(*efficiencies*)... will be equal.* Performance is derived by assuming equal sensible and latent recovery effectiveness.

UNIT HOUSING

The rotor housing shall be constructed using a heavy duty welded tubular steel frame (rotors under 42" shall have a heavy duty galvanized frame) with galvanized sheet metal cover plates and inspection hatches. Adjustable brush seals must be provided along the periphery of the rotor and between the inlet and outlet air passages to effectively prevent air leakage and cross-contamination between airflows. Total airflow between airstreams from leakage and purge shall be less than 10% @ 2.5"w.g. differential pressure between airflows. Rotor and casing shall be reinforced to prevent deflection from differential pressures to less than .03 inches. All rotors shall be mounted on sealed permanently-lubricated spherical bearings. All rotors over 42" in diameter must have flanged or pillow block bearings that can be serviced or replaced without removal of the rotor from the rotor housing.

PURGE SECTOR

* The unit must be provided with a factory set, field adjustable purge sector designed to limit cross contamination at qualified appropriate design conditions to operate at less than .04 percent of that of the exhaust air stream concentration. Independent laboratory evaluations must indicate purge sector configurations, rotor construction, gasses, air pressure differentials, rotor speeds and other phenomena that constitute "appropriate design conditions" required to limit cross-contamination and air leakage.

DRIVE SYSTEM/SPEED CONTROL

The rotor drive system shall consist of a self adjusting belt around the rotor perimeter driven by an AC motor with gear reduction. The variable speed drive shall be specifically designed for heat wheel applications to include: an AC inverter, soft start/stop, rotation detection w/alarm contacts, automatic self cleaning jog cycle, and self testing capability. The speed controller shall be capable of accepting any control signal (potentiometer, VDC, and mA).

AUTOMATIC TEMPERATURE CONTROL

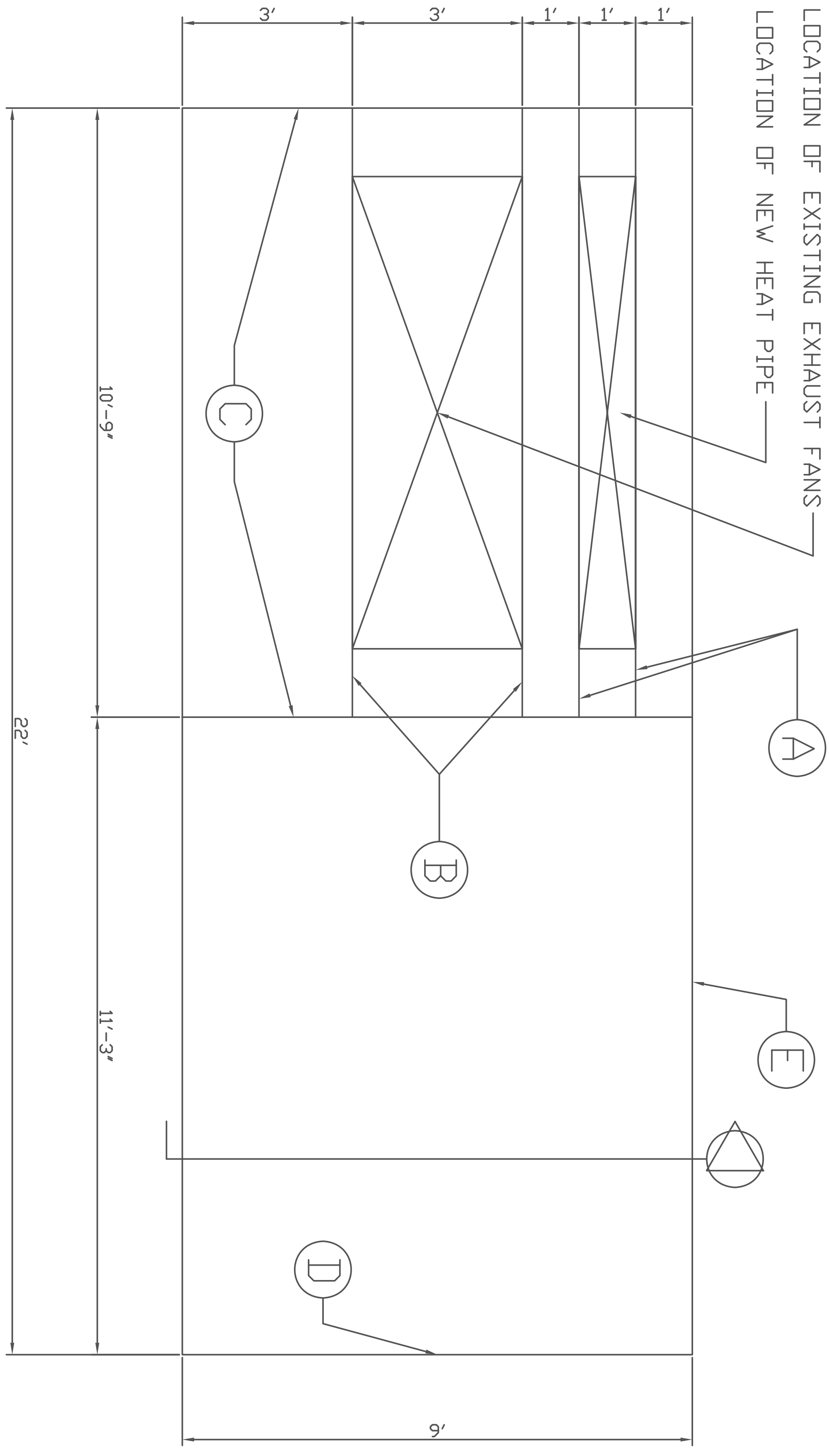
The temperature control system shall consist of an integral control panel with remote temperature sensors mounted in each of the four airstreams to monitor exchanger performance. The control shall modulate rotor speed to (1) prevent frost build-up, (2) reduce heat recovery for economizer mode, (3) switch to maximum heat recovery when outdoor temperature is higher than indoor temperature. A rotation detector/alarm shall be built into control panel with contactor provided for connection building control system.

*** Refer to independent performance tests of XeteX AIRotor Total Energy Recovery Wheels conducted, evaluated and verified for the specified characteristics by research assistants from the Department of Mechanical Engineering, University of Minnesota, Minneapolis. Detailed Technical Reports that certify Thermal Effectiveness and Cross-Contamination performance are available on request.**

XeteX Inc.
3530 East 28th Street
Minneapolis, MN 55406
(612) 724-3101
(612) 724-3372 Fax

Form 2.2E04 01 Printed in USA

MECHANICAL EQUIPMENT SUPPORT - STRUCTURAL PLAN



MECHANICAL EQUIPMENT SUPPORT - STRUCTURAL SECTION