The Hearst Tower:
Combining Steam Driven Absorption Cooling with a DOAS/Radiant System

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Senior Thesis-Mechanical Option

Hearst Tower

959 Eighth Avenue
New York, NY

Project Team

Owner:
Hearst Corporation

Development Manager:
Tishman Speyer Properties

Architect:
Foster and Partners

Associate Architect:
Adamson Associates

Structural Engineer:
Cantor Seinuk

MEP:
Flack+Kurtz

Construction Manager:
Turner Construction

Lighting:
George Sexton

Construction

- December 2004-June 2006
- Expected to earn LEED gold certification
- Original 6 story headquarters facade preserved with new tower constructed through center

Mechanical System

- Low temperature air distributed using central VAV system
- 4-4000 ton cooling towers serve the chiller plant
- Chiller plant contains 2-1200 ton and 1-400 ton centrifugal chillers
- Radiant floor heating and cooling serves the lobby

Architecture

- 856,000 Square Feet
- 42 stories, 597 feet tall
- Glass and metal curtain wall façade
- Diagrid design features triangular glass panels four stories high
- Primarily office space, but will also house an auditorium, test kitchens, executive dining hall, and a television studio

Structural System

- Composite steel and concrete floor system with 40 foot column free interior span
- Tower is connected to the landmark façade by a horizontal skylighting system
- Diagrid members are wide flange rolled steel sections connecting at nodes
- The secondary lateral system is a braced frame at the service core

Electrical & Lighting

- 460/265v, 3 phase, 4 wire system
- Utilizes daylighting sensors to control lighting
- General lighting system consists of compact fluorescent downlights and linear fluorescent troffers
- Atrium utilizes incandescent par38 accent lights

Jessica Lucas Mechanical Option
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Acknowledgements

"A successful person is one who can lay a firm foundation with the bricks that others throw at him."

I'd like to take this opportunity to thank those who have thrown those bricks at me.

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To my Greta, my twin…taking a two hour phone break from thesis is fine if there's gossip involved! Thank you for always knowing when I need you most.

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Last and certainly not least, Thomas J, my sanity and strength. I am amazed everyday by the confidence you have in me…if I could only see myself through your eyes. Even in a city of 8 million it will always be you and I.

I love you all. God Bless.

Introduction

The senior thesis project is a culmination of five years of study in the Architectural Engineering Program at the Pennsylvania state University. During the first three years in the program, students take courses in four main subject areas, which include: Mechanical Engineering, Structural Engineering, Electrical Engineering/Lighting Design, and Construction Management. After exploring each option, students choose one of the four areas to study in depth during their final two years.

The senior year is considered a professional year focusing mainly on the infamous “Senior Thesis.” During the fall semester students are asked to research a building of their choice through a series of three technical assignments. After the completion of the technical assignments, students are asked to propose an alternative design approach for their building. The proposal is carried out through a depth study in the student’s option area and two breadth studies in option areas affected by the proposed system design.

The following Thesis report investigates a proposed redesign for the Hearst Tower located in New York City. The main focus of the mechanical depth study is the implementation of Steam Driven Absorption Cooling with a Dedicated Outdoor Air System paralleled by a radiant floor heating and cooling system. The new mechanical system will directly impact the electrical and structural systems of the building and these topics will therefore serve as breadth studies. The main goal of this proposal is to explore the feasibility of such a system with the Hearst Tower and to serve as a learning experience to enhance professional development. The thesis project is solely for educational purposes and in no way reflects error in the original building design.

Enjoy…

Executive Summary

The Hearst Tower is an 856,000 square foot office tower designed to consolidate over 2,000 Hearst Corporation employees. The new tower will be constructed along Eighth Avenue between 56th and 57th Street in New York City, which is the site of the original Hearst headquarters built in 1928. The new tower will rise from the center of the original landmark façade and is scheduled for completion by Summer 2006.

Since the Hearst Corporation is a media mogul, the new office tower is designed to accommodate a variety of functions under one roof. The 42 stories of the tower will include office space, an auditorium, a full service television studio, executive dining areas, and Good Housekeeping test kitchens. Because of the size of the building and complexity of the current mechanical system, only eleven floors of the tower portion will be considered for the proposed design.

The current mechanical system includes a central chilled water plant serving a central low temperature variable air volume (VAV) system. The chiller plant currently contains (2)-12000 ton and (1)-400 ton electric chillers. The office tower is served by (4)-110,000 CFM air handling units that are capable of supplying 100% outdoor air. Heating is provided to the building via a steam to hot water heat exchanger, which subsequently distributes hot water to fan powered VAV boxes located throughout each floor.

In accordance with ASHRAE Std. 62-2004, each office floor requires 2326 CFM of outdoor air. The typical cooling load for one floor is 29 tons, while the heating load is 82.7 MBH.

This report examines the use of steam driven absorption cooling with a dedicated outdoor air system (DOAS) paralleled by radiant floors for heating and cooling, in hopes of reducing annual energy costs and improving indoor air quality. It was ultimately determined that the DOAS/radiant system helped reduce the size of major equipment such as chillers and air handling units. The use of steam driven cooling as opposed to electric chillers also helped to save on annual energy costs because of incentives provided by the NYC steam utility, Consolidated Edison. This system is an attractive idea in an area such as NYC where the electricity grid is already overextended. Considering the increased capital cost of the new system and the annual energy savings, a simple payback period of 4.5 years was determined.

Aside from the cost savings, the DOAS system also helps to secure the LEED point for Indoor Air Quality. Since the Hearst Tower is striving for LEED gold, obtaining this point is very important. Overall, the annual energy savings, short payback period, and sustainability of the system make the proposed system a feasible alternative to the original design.

Project Background

Building an Icon

In 1927, William Randolph Hearst set out to build a headquarter location to house several magazines, which he owned at the time. Located between 56th and 57th street on Eighth Avenue, the 40,000 square foot six story "International Magazine Building" rose to become an important piece of Columbus Circle history. The original design had several key features: the design would reflect the musical and artistic heritage of the neighborhood in which it resides and the building would be structurally reinforced to support the addition of an office tower to this 6-story podium.

The façade of the building is precast limestone with a two story base supporting four recessed stories. Each corner of the building features sculptural representations of music, art, commerce, and industry. The main entrance features "Comedy and Tragedy" on the left and "Music and Art" on the right side. "Printing and Sciences" appear on the 57th street corner, while "Sport and Industry" are located on the 59th street corner.

Although it was Hearst's original intention to build an office tower atop the original building, this dream would not take shape until 2004. When the new Hearst family decided to carry out plans for an office to house over 2,000 employees, the result would be one of both innovation and prestige.

The 42-story Hearst Tower rises from Midtown Manhattan and stakes its claim in the New York City skyline with its eclectic diagrid design. Further inspection of the building as a system and not just an architectural icon reveals a building striving for excellence not only in structural innovation, but in sustainable building design as well.

Combining the 20th and 21st Centuries

Appearing to "float" over the landmark façade, the Hearst Tower makes a bold statement to visitors of Midtown Manhattan. Although the podium building is nearly 77 years old, the latest technologies in energy conservation will be utilized.

With an ultimate goal of achieving a Gold Rating under the Leadership in Energy and Environmental Design (LEED) by the US Green Building Council, the building design teams set out to make "environmentally friendly" design decisions. Utilizing such technologies as daylight and occupancy sensors, storm water reclamation, and the world's largest sustainable water feature, the Hearst Corporation Tower will be the first highrise building in NYC to obtain this status.

In addition to environmentally friendly systems designs, the architecture of the building lends itself to conservation and efficiency. During demolition, about 85% of the original materials were recycled for future use. The diagrid design also allows for maximum structural efficiency. The triangulated system of A-frames used about 2,000 tons less steel than a conventional moment frame design. A detailed description on the existing structural system will be discussed in the "Structural Breadth" section of this report.

With an overall area of 856,000 square feet, with 42 stories above ground and a double height basement, the Hearst Building was able to achieve more LEED points than most buildings half that size. Thus far, the Tower has been designed to secure 37 LEED points, with the possibility of obtaining an additional 6 points. Even without the additional 6 points, the Hearst Tower has earned enough points to qualify as a LEED Silver Building.

Site and Architecture

The Hearst Tower is located in the Columbus Circle neighborhood of New York City. The original Hearst Magazine building occupies a 200 by 200 square block between 56th and 57th street along Eighth Avenue. William Randolph Hearst began purchasing land for the headquarter location around the 57th street neighborhood in 1895. This neighborhood was particularly attractive to Hearst because he believed it would become a home for NYC's growing theater district. With Carnegie Hall around the corner on 57th street and Seventh Avenue, the area did, as Hearst had expected, experience tremendous commercial growth in the 1920's.

With its precast limestone façade and eclectic architectural style, the original headquarter location stood out against the Art Deco style of the time by combining different architectural styles. The building features large fluted columns and carved balustrades on the first two stories. The top four stories are recessed back from the base and are accentuated by massive piers. With almost eighty years of dedication to the city of New York, Hearst's building became a designated landmark site by the Landmark Preservation Commission in 1988.

Existing Mechanical System

Designing for Sustainability and Reliability and Marketability and the List Goes On…

Flack+Kurtz (F+K), the MEP engineer for the project, set forth several design objectives prior to developing the Tower's mechanical system. The team would design a class A office building that would house a variety of spaces including offices, dining areas, test kitchens, assembly spaces, and an auditorium. A class "A" office building is typically characterized by having state of the art mechanical, electrical, and life safety systems as well as high quality exterior and interior finishes. Buildings in this class have a strong market presence and can therefore demand higher rental rates.

In addition to maximizing the buildings marketability, F+K would also strive for a system with low capital and annual energy costs. With these three main objectives at the top of their list, the mechanical design team developed a design matrix as a basis of comparing eleven possible design schemes. The criteria set forth in the matrix included:

- Initial Capital Cost
- Annual Energy Cost
- Speculative Tenant Marketability
- MEP Space Requirements
- Building Height
- Reliability
- Flexibilty
- Maintenance
- Indoor Air Quality
- Building Operation Efficiency
- Interior HVAC Alterations
- Acoustics

As mentioned before, this criterion above was used to evaluate eleven possible system configurations. Ultimately, Flack+Kurtz decided upon using a Central Variable Air Volume System with low temperature supply air and a Central Chilled Water Plant.

Waterside System

The waterside of the air conditioning system is made up of (4)-4000 ton cooling towers, (2)-1200 ton chillers, and (1)-400 ton chiller. Currently, these chillers are electric motor driven, centrifugal chillers. The condenser and evaporator are shell and tube type and circulate R-134a refrigerant for the heat transfer process. The chillers are located in the basement chiller plant and are headered by eight vertical inline pumps, four serving the condenser side and four serving the chilled water loop.

In addition to providing chilled water to the building's AHU's, the chillers also serve the radiant floor in the lobby. The hydronic radiant floor provides both heating and cooling for this space. Using radiant floor heating/cooling is an ideal application for atrium or lobby spaces, as these spaces tend to have very high ceilings and a high window/wall ratio. In the case of the Hearst Tower, the lobby/atrium is 80 feet tall with a skylighting system on the roof. The radiant floor is therefore used to remove the solar radiation incident upon the floor in the summer. During the winter months, radiant floors help to maintain thermal comfort in the occupied zone without having to heat the entire 80 foot height.

Heating for the building is provided by high-pressure steam purchased from the Consolidated Edison steam grid. After being brought into the building, the steam is passed through the building’s pressure reducing station. This low-pressure steam is then passed through heat exchangers to heat water for the buildings hot water heating coils.

Airside System

The current airside system of the Hearst Tower is a central variable air volume (VAV), low temperature system. By providing low temperature supply air, 44F, the mechanical designers were able to downsize the mechanical shafts by 25%. Fan powered terminal units provide the supply air to each space and contain the fan and motor assembly, heating coil, and VAV damper.

There are four large (110,00 CFM) air-handling units located on the 28th floor, four smaller air handling units in the basement, and dedicated units serving the auditorium and kitchen areas. The air-handling units operate such that 100% outdoor air is being delivered during 50% of the occupied hours, 50% outdoor air during 22% of the occupied hours, and minimum outdoor air, as required, during the remaining time period.

Sequence of Operation

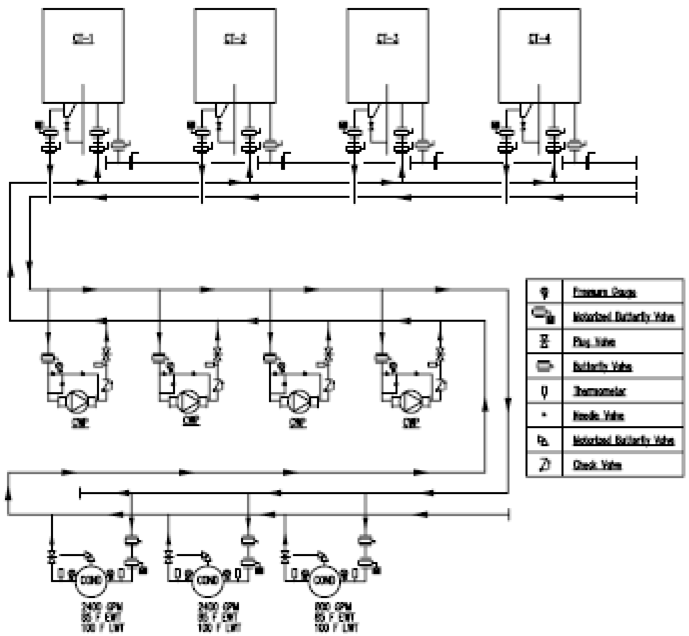
The Building Management and Control System (BMCS) is responsible for both the control and monitoring the building HVAC and electrical equipment. The operating philosophy of the management system is through Direct Digital Control (DDC). Control of the system is performed by the building operator in the central operator’s workstation, but can also be performed at the remote operator’s workstation. The system works to maintain building management and environmental control while facilitating energy conservation.

<u>Cooling Towers</u>

Four induced-draft cooling towers serve nine water pumps in the condenser water system. As seen in the figure titled “Primary Condenser Water Loop,” four of the nine pumps are dedicated to the chillers while the remaining three primary and two secondary water pumps serve the secondary condenser water system. The secondary condenser water system consists of Tenant Condenser Water System and the Dual Temperature Water System.

All cooling towers are started and stopped by the BMCS. The cooling towers only operate when the chillers are operating or when other water-cooled equipment is in operation.

Primary Condenser Water Loop



Primary Condenser Water System

The condenser water supply temperature governs operation of the cooling tower. When the condenser water supply temperature is below the 85F setpoint, the cooling towers will shut off and the condenser water will bypass the fill and return to the chillers. Upon a rise in condenser water temperature, the isolation valves in CT-1 will open and the condenser water bypass valve modulates to bypass less water directly to the chillers and more water to the cooling towers.

When 100% of the condenser water is flowing to the cooling towers, and there is a continued rise in condenser water temperature, the cooling tower fan starts on low speed. The tower fan speed increases as needed to maintain the condenser water setpoint temperature.

If CT-1 is operating at 100% and the condenser water temperature is continuing to rise, the cooling tower with the least amount of run-time hours will be started. The isolation valves of this tower are opened when the tower is in service. When the condenser water temperature drops, the reverse process will occur.

Tenant Condenser Water System

The tenant condenser water loop is a dual heat exchanger operation. Condenser water supplied by the cooling towers passes through plate and frame heat exchangers, PHX-B-1 and PHX-B-2, and heat is transferred from the tenant condenser water return to the cooling tower condenser water. The tenant condenser water is then pumped to water-cooled air conditioning units.

When condenser water is needed for this loop, the secondary control valve for the heat exchanger will open and the lead primary and secondary pumps will start at low speed. A temperature sensor located in the secondary condenser water supply will modulate the speed through the primary pump in order to maintain the 87.5F setpoint temperature in the secondary loop. A pressure sensor within the secondary loop varies the speed of the secondary pump to maintain the setpoint pressure differential.

Dual Temperature Water System

The dual temperature water system (DTW) serves 4-pipe fan coil units (FCU) located on the third floor of the Hearst Tower. The perimeter of the large cafeteria space is lined by twenty-two FCU's. This area is also conditioned by a radiant floor (heating and cooling) and 3-30,000 CFM air handling units located in the basement.

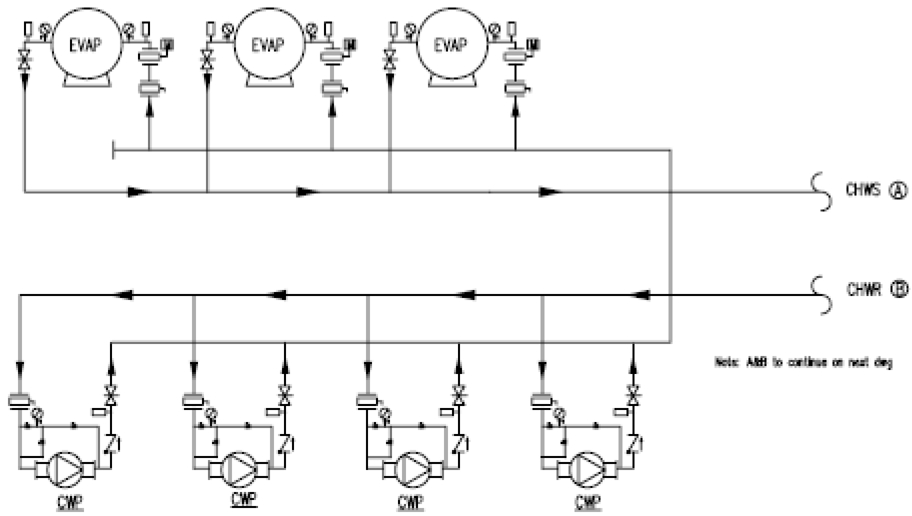
The DTW system is dual heat exchanger operation for both heating and cooling. During cooling operation, the system can operate in three modes: mechanical cooling, pre-cooling, or free-cooling. Corresponding isolation valves will open depending on the operating mode of the DTW system.

Chiller Plant

The chiller plant consists of three chillers, four headered chilled water pumps, and four headered condenser water pumps. The BMCS operator will manually start and stop the chillers.

When a chiller is called to start, the chilled and condenser water isolation valves open and the corresponding lead pumps start. Water is supplied from the cooling towers to the condenser water loop, an R-134a refrigerant loop operates between the condenser and evaporator of the chiller, and chilled water passes through the evaporator to the loads and back through chilled water pumps (See “Chilled Water Loop” Figure). The chilled water temperature is then modulated depending upon the building loads and outdoor air temperature.

Chilled Water Loop



Radiant Loop

The use of the radiant slab for either heating or cooling of the lobby area is determined by the BMCS operator and is initiated by flipping a switch. A dual heat exchanger operates on the heating/cooling side of the primary system, while two common pumps serve the heat exchangers from the secondary system. Temperature sensors in the slab will maintain the slab temperature at a minimum of 68F in the summer and 85F in the winter.

When an operating mode is chosen, the primary control valve for the heat exchanger and the secondary isolation valves open and the lead pump on the secondary side is started. During cooling, temperature sensors in the secondary supply loop control the primary loop control valve in order to maintain the setpoint. This setpoint is determined based upon the dewpoint temperature in the lobby (the area served by the radiant slab) plus one degree. The minimum setpoint temperature is set at 61F for this operating mode.

For heating, the isolation valves for the steam heat exchanger are open and again, temperature sensors in the secondary system control the primary loop control valves. The secondary loop setpoint for this condition is 105F. In the event that one heat exchanger is incapable of maintaining this setpoint, the second heat exchanger will be placed online.

Chilled Water Loop

Air Handling Units

The Hearst Tower contains approximately 32 air-handling/conditioning units. For the purpose of demonstrating the sequence of operations, only AHU-28-1,2,3,4 will be discussed. These units are considered the central air-handling units and serve the heating and air conditioning needs of the 10^{th}-41^{st} floors. These units are made up of airside economizer dampers, heating coil, humidifier, cooling coil, and dual supply fans. These units are supplied chilled water by the chiller plant as discussed previously.

The AHU's serve the supply header and work together to supply occupied floors from 6am-6pm Monday through Friday. When a floor is in occupied mode, the isolation dampers in the supply and return ducts open and the lead supply, return, and exhaust fans will start. The supply and return fans will start in low speed for one minute to allow the system to stabilize. When the exhaust fans are called to start, the respective discharge air damper will also open. Static pressure sensors on the 16^{th} and 32^{nd} floors are compared and the lower static pressure measurement is inputted into the BMCS. If the static pressure drops, the fan speed will be increased as necessary until the fan reaches 100%. If the pressure continues to drop with the fan at the 100% condition, the next AHU (in numerical order) will be called to start. Once a floor is placed into occupied mode, the floors directly above and below the floor are also put into occupied mode.

When a floor is in unoccupied mode, all fans and corresponding dampers will be off/closed.

Fan Powered Boxes

The fan-powered boxes (FPB) are either in occupied or unoccupied mode. In the occupied mode, the series fan runs at constant speed. If the temperature of the zone is below the setpoint temperature of 75F, the primary air damper will supply the minimum CFM. When the zone temperature exceeds the setpoint, the damper will modulate until the setpoint temperature is reached. When the zone is unoccupied and the base building AHU's are off, the series fan will be off and the damper will be at the minimum CFM position (See Figure "Typical Distribution From Air Shaft to VAV or Fan Powered VAV Box").

Fan Powered Boxes with Hot Water Heating Coil

These FPB's also operate in either occupied or unoccupied mode. In occupied mode, the series fan again operates continuously. When the zone temperature is between the heating setpoint (72F) and the cooling setpoint (75F), the primary air damper supplies minimum CFM. However when the zone temperature rises above the cooling setpoint, the damper modulates to bring in more air. When the temperature drops below the heating setpoint, the hot water coil control valve modulates to increase the zone temperature and minimum air is brought in through the primary air damper.

Variable Air Volume Boxes

Variable air volume (VAV) boxes utilize a primary air damper, which modulates to maintain the zone temperature, as sensed by the temperature sensor.

Mechanical Depth Study:
Combining Steam Driven Absorption Cooling with a DOAS/Radiant Floor System

Absorption Cooling

Using Heat to Cool

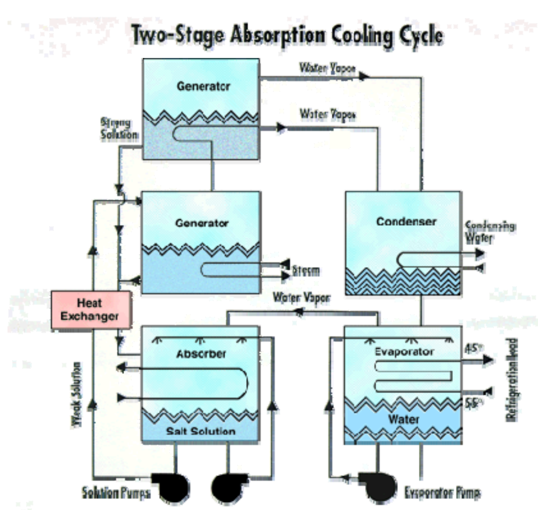
The absorption refrigeration cycle is a conditioning process driven by a heat source, rather than electricity. Typical energy sources for such a process include steam, hot water, or the combustion of natural gas. In the case of the Hearst Tower, a steam driven, double effect absorption process will be analyzed and will therefore be the process described herein.

The basic idea behind both a vapor compression and absorption cooling process is the same: to move heat from a low temperature reservoir to one at a relatively higher temperature. This process is done by evaporating and condensing refrigerant vapors at different pressures within a chiller. The distinct difference between the two refrigeration cycles is that the vapor compression cycle requires mechanical compression to produce the pressure required by the condenser. In an absorption cooling cycle, a thermo-chemical process involving two fluids with a high affinity for one another establishes the required pressure differential.

The two fluids used in the thermo-chemical process are a refrigerant/absorbent pair. To have the most effective absorption cooling cycle, the refrigerant should be able to change between liquid and vapor phase easily and have a high affinity, or ability to dissolve within the absorbent. Because of this, water is typically chosen as the refrigerant and either Lithium Bromide or ammonia may be chosen as absorbents.

Within a two-stage absorption chiller there are several key components: a high and low stage generator accompanied by a high and low stage heat exchanger, a condenser, an evaporator, a solution pump, and an expansion valve. A diagram of a double-effect absorption chiller can be seen below in Figure 1.

Figure 1 Absorption Cooling Cycle



Starting in the evaporator, heat is transferred from chilled water (which serves the building loads) to the refrigerant. The refrigerant boils in the evaporator and the vapors travel to the absorber. Within the absorber the concentrated lithium bromide absorbs water vapor. Heat from this process is rejected to condenser water from the cooling tower. The lithium bromide water solution is then pumped through the low stage heat exchanger where it is pre-heated by concentrated lithium bromide returning from the low-stage generator. Some of the solution is then sent through the high stage heat exchanger while some is sent to the low stage generator.

In the low stage generator, rejected heat (refrigerant vapor) from the high stage generator is used to drive off the water in the lithium bromide-water solution, thus re-concentrating the lithium bromide absorbent. The refrigerant vapor driven off is then returned to the condenser. The returning lithium-bromide concentrate preheats the portion of the solution passing through the high stage heat exchanger. Once preheated, the solution then moves to the high stage heat exchanger where high-pressure steam adds the heat necessary to drive water off of the solution.

As stated previously, the rejected refrigerant vapor is sent to the low stage generator where it transfers heat to the solution and therefore condenses. This condensed vapor is also returned to the condenser. Within the condenser, the refrigerant vapor is cooled further by condenser water from the cooling tower, thus “condensing” the refrigerant. The condenser refrigerant is then passed through and expansion valve that lowers the pressure of the fluid before returning to the evaporator. The condensed refrigerant is now able to accept rejected heat from the chilled water system.

Why two-stage, steam driven absorption cooling

Since New York City is home to approximately 8 million people and hundreds of thousands of buildings, the electric utility grid is nearly exhausted and the cost of electricity is extremely high. The Hearst Tower incurs an electricity cost of about \$0.20/kWh and utilizes approximately 7 million kWh per year solely for the mechanical system. Currently, the Hearst chiller plant has 2800 tons of cooling capacity produced by 3-electric motor driven centrifugal chillers. With absorption cooling, electricity is only required for the pumps and controls.

The Tower is also connected to the Consolidated Edison (Con Ed) district steam grid, which provides high pressure (165 psi) steam. Con Ed offers rebates to building owners utilizing steam driven cooling, especially in months when cooling is at a peak and the electricity grid becomes scarce. The rate structure applied to such buildings can be seen in Table 3 in the section of this report titled “System Simulation.”

Two stage absorption coolers are a way of improving the energy efficiency of an absorption chiller. As described in the previous section, rejected heat from the high-stage generator is used to drive off more water in the low-stage generator; therefore water is recovered by two generators but energy only needs purchased for one.

In addition to energy benefits, absorption cooling offers some key features. Since water is used as the refrigerant, no chlorofluorocarbons (CFC's) or hydrochlorofluorocarbons (HCFC's) are used. Although absorption chillers are much larger than vapor compression chillers, there are fewer moving parts which increases the chiller's reliability and decreases maintenance costs.

Limitations & Additional Considerations

Double effect absorption chillers usually have a COP on the order of 1-1.2, which is lower than that of a typical vapor compression cycle. In general, the size of condenser water pumps is a function of the flow rate per unit of cooling capacity. Since the COP of absorption chillers is lower, a higher condenser water flow rate and thus larger pumps are needed. To compensate for the increased condenser water flow rate, additional cooling tower capacity is also required.

In the case of the Hearst Tower, implementing a Dedicated Outdoor Air System with parallel Radiant Floor Cooling allowed the chiller plant to be reduced by approximately 37% (a discussion of this system is available in the "Combining Steam Driven Absorption Cooling with a DOAS/Radiant Floor System" portion of this report.) Since the chiller plant was reduced, the existing cooling towers were enough to compensate for the needed condenser water flow rate. If the new secondary system had not been implemented, the existing cooling towers would need to be resized.

In addition to COP, chiller efficiency can be measured using Integrated Part Load Value (IPLV). IPLV is a way of calculating an annual COP based on the typical load profile and part load characteristics of the chillers. The basic assumption behind this method is that the chiller operates at a specific part load for a specific number of hours throughout the year. The following equation defines IPLV:

$$\text{IPLV} = \frac{1}{(0.17/A) + (0.39/B) + (0.33/C) + (0.11/D)}$$

Decisions, Decisions

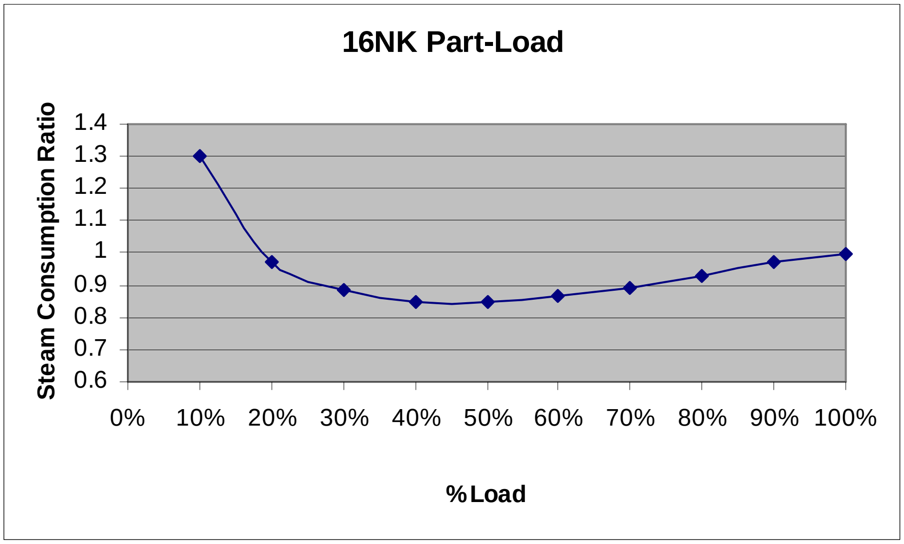
Based on the characteristics described throughout this section and the design load provided by Carrier's Hourly Analysis Program (HAP v4.20), industry consultation was used to determine the best absorption chiller plant configuration. With the help of Scott McDonough of Carrier's Water Cooled Chiller Marketing Division, 3-600 ton double effect absorption chillers arranged in parallel were selected to meet the 1800 ton design load.

Since some allowance is needed between the boiling temperature and the freezing point of the water refrigerant, the lowest attainable chilled water temperature with this configuration is 41F. This is slightly higher than the 39F chilled water temperature achieved by the original centrifugal chillers. The chosen absorption chiller model

(16NK71) also requires steam at 114 psig. A pressure reducing valve will be implemented to reduce the pressure of the 165 psig steam provided by the district grid.

In addition to chiller specifications, part load performance curves and IPLV calculations were also provided by Carrier and are included below in Figure 2.

Figure 2 Part Load Operating Characteristics



The double effect absorption chillers selected are rated in accordance with ARI-560.

Pumping

Because of a decreased chiller plant capacity, resulting from the changes to the airside of the system discussed later in this report, the condenser water and chilled water pumping capacity also decreased. Appling the pump affinity laws, new pumps were selected. The original pumps selected were Armstrong split-coupled vertical in-line centrifugal pumps. Utilizing the Armstrong online catalog, the same type of pumps, with lower RPMs were chosen.

Condenser Pumps: 1731 GPM/2400 GPM=X/1800 rpm, therefore X=1300 rpm
Selected: (3)-1500 rpm

Chilled Water Pumps: 1029 GPM/1800 GPM=Z/1800 rpm, therefore Z=1200 rpm
Selected: (3)-1200 rpm

For detailed specifications on the pumps selected, including corresponding performance curves, see Appendix B at the end of this report.

Dedicated Outdoor Air Systems

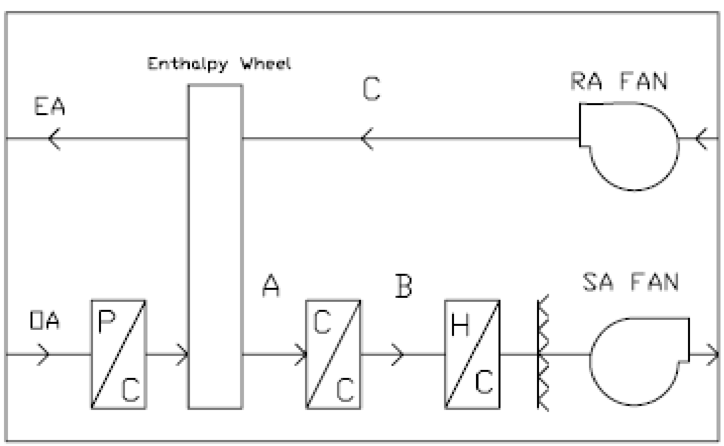
A dedicated outdoor air system (DOAS) is a constant volume system which supplies the minimum amount of outdoor air (OA) required by ASHRAE Std. 62-2004 or the local code requirements, whichever is greater. The basic concept behind a DOAS system is that this ventilation air is used to completely satisfy the space latent load and a portion of the sensible load using heating coils, cooling coils, and energy recovery wheels.

The remaining sensible load is satisfied using a parallel system. Examples of parallel systems include radiant ceiling panels, radiant floors, fan coil units, heat pumps, or a variable air volume system (VAV). In the case of the Hearst Tower, a radiant floor parallel system was analyzed in conjunction with the DOAS system.

How does the DOAS system work?

With all of the pieces of the puzzle mentioned above, a DOAS air handling unit conditions ventilation are by recovering energy from the exhaust air (EA) stream. During the cooling season, the DOAS unit will supply cool, dry air at a low dew point temperature, usually around 45F. In the wintertime, the unit supplies warm and humidified air. Figure 3 shows the DOAS configuration applied to the Hearst Tower.

Figure 3 Dedicated Outdoor Air Unit



An enthalpy wheel, which is capable of absorbing both heat and moisture, rotates between the OA and EA streams. During summer operation, moisture and heat are transferred to the exhaust stream, while in the winter the opposite process occurs. Recovering energy in this manner preconditions the air before it reaches the heating/cooling coil thus reducing the load on these coils (See Figure 4).

The coils are then used to meet the latent load not met by the wheel. In the case of the Hearst Tower, the DOAS air handling unit was able to satisfy about 33% of the total space load, as seen in the sample calculation (Summer Design) included on the following page. A sample calculation for winter design conditions is also included. The third calculation page shows the resulting cooling coil load for a Variable Air Volume (VAV) configuration, which acts as a basis of comparison for the newly proposed system.

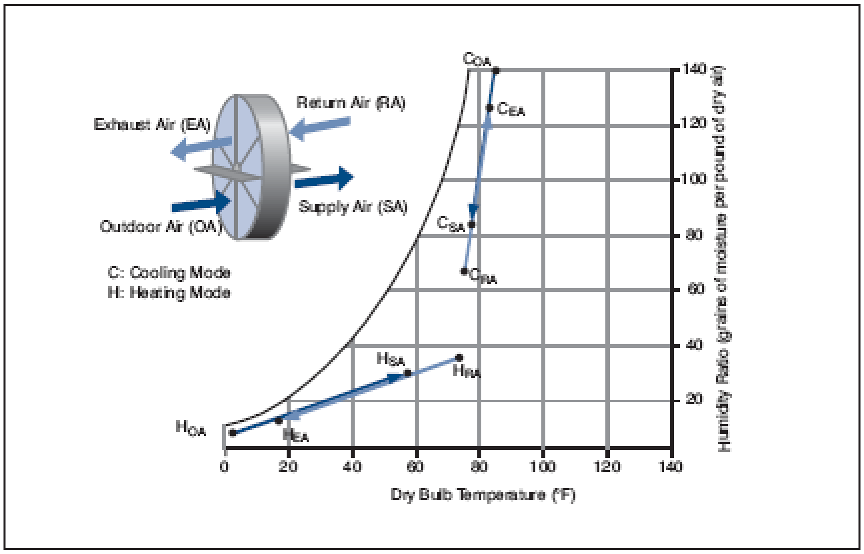
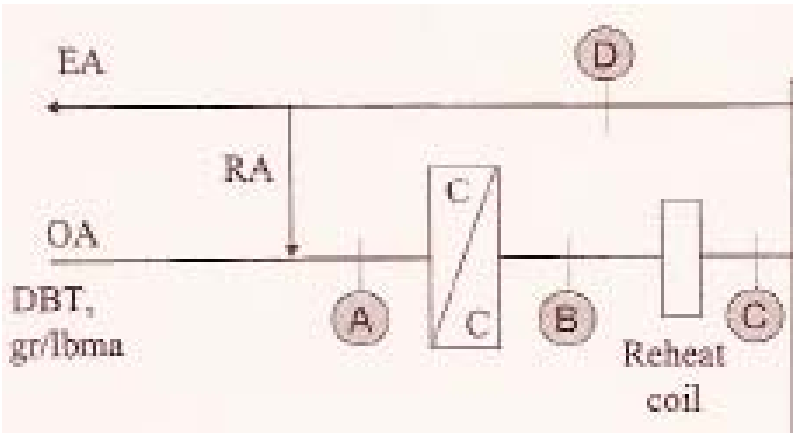


Figure 4 Typical Heating/Cooling Operating Conditions

The following figure, Figure 5**,** shows a diagram of a typical Variable Air Volume System.

Figure 5 Typical VAV AHU



Sample Calculation #1 for one Typical Floor (Summer) Using DOAS

Outdoor Design Conditions: 92F, 98gr/lbma
Room Design Conditions: 75F, 65 gr/lbma

Design Loads (For one typical open office floor):
Sensible: 319067 BTU/hr
Latent: 31629 BTU/hr

Enthalpy Wheel Characteristics (Based on Semco Design Guide)
Latent Effectiveness (ε_l) =0.8
Sensible Effectiveness (ε_s) =0.8

With Supply air conditions set at 45F saturated air, the minimum amount of OA required by ASHRAE Std. 62.1-2004 was not enough to meet the latent load. Also, supplying 30% more OA than required by the standard, a LEED point can be obtained. Therefore 2326 CFM was used to meet the latent load.

<u>Point A</u> *(After enthalpy wheel)*

$W_{EA-EW}=W_{OA}-\varepsilon_l*(W_{OA}-W_{RA})$
$W_{EA-EW}=98-0.8*(98-65)=$**71.6 gr/lbma**

$T_{EA-EW}=T_{OA}-0.8*(T_{OA}-T_{RA})$
$T_{EA-EW}=92-0.8*(92-75)=$**78.4F**

<u>Point B</u> *(Supply Air to the space)*

$W_{SA}=45$ gr/lbma
$T_{SA}=45$F

<u>Cooling Coil Loads</u>

$Q_{LATENT}=0.68*(CFM)*(W_{EA-EW}-W_{SA})$
$Q_{LATENT}=0.68*2326*(71.6-45)=$ **42072.7 BTU/hr**

$Q_{SENSIBLE}=1.08*(CFM)*(T_{EA-EW}-T_{SA})$
$Q_{SENSIBLE}=1.08*2326*(78.4-45)=$ **83903.5 BTU/hr**

$Q_{TOTAL}=(Q_{LATENT}+Q_{SENSIBLE})/12000$ (BTU/hr)/ton=**10.5 tons**

<u>DOAS Sensible Cooling Capacity</u>

$Q_{DOAS\_SENS}=1.08*(CFM)*(T_{RA}-T_{SA})=1.08*2326*(75-45)=$ **75362.2 BTU/hr**

<u>Required Cooling Capacity of Parallel System</u>

$Q_{DESIGN\_SENS}-Q_{DOAS\_SENS}=(319067-75362.2)$ BTU/hr= **243704.6 BTU/hr (20.3 tons)**

Sample Calculation #2 (Winter) using DOAS

Outdoor Design Conditions: 13F, 0gr/lbma
Room Design Conditions: 70F, 55 gr/lbma

Design Loads (For one typical open office floor):
Sensible: 82784 BTU/hr
Latent: 0 BTU/hr

Enthalpy Wheel Characteristics (Based on Semco Design Guide)
Latent Effectiveness (ε_l) =0.8
Sensible Effectiveness (ε_s) =0.8

<u>*Point A*</u> *(After enthalpy wheel)*

$W_{EA\text{-}EW}=W_{OA}+\varepsilon_l*(W_{RA}-W_{OA})$
$W_{EA\text{-}EW}=0+0.8*(55-0)=$**44 gr/lbma**

$T_{EA\text{-}EW}=T_{OA}+0.8*(T_{RA}-T_{OA})$
$T_{EA\text{-}EW}=13+0.8*(70-13)=$**58.6F**

<u>*Point B*</u> *(Supply Air to the space)*

$W_{SA}=55$ gr/lbma
$T_{SA}=90$F

<u>*Heating Coil Load*</u>

$Q_{SENSIBLE}=1.08*(CFM)*(T_{SA}-T_{EA\text{-}Ew})$
$Q_{SENSIBLE}=1.08*2326*(90-58.6)=$ **78879.36 BTU/hr**

<u>*Humidification*</u>

$m_{dotvapor}=m_{dotair}(W_{SA}-W_{EA\text{-}EW})$
$m_{dotair}=(2328cfm*60\ min/hr)/13.83\ cf/lbma=$**10091 lbma/hr**

$m_{dotvapor}=10091$ lbma/hr*(0.0078-0.0063 lbmv/lbma)

Therefore, required steam flow rate is 15 **lbm/hr**

<u>*DOAS Sensible Heating Capacity*</u>

$Q_{DOAS\_SENS}=1.08*(CFM)*(T_{SA}-T_{RA})=1.08*2326*(90-70)=$ **50241.6 BTU/hr**

<u>*Required Heating Capacity of Parallel System*</u>

$Q_{DESIGN\_SENS}-Q_{DOAS\_SENS}=(82785-50241.6)$ BTU/hr= **32543.4 BTU/hr**

Sample Calculation #3 One Typical Floor using VAV (Summer)

Outdoor Design Conditions: 92F, 98gr/lbma
Room Design Conditions: 75F, 65 gr/lbma

Design Loads (For one typical open office floor):
Sensible: 319067 BTU/hr
Latent: 31629 BTU/hr

Finding the supply air quantity necessary to satisfy sensible load (V_{DOT_SA})

$Q_s=1.08*(V_{DOT\_SA})*(T_{RA}-T_{SA})$
->solving for $V_{DOT\_SA} = Q_s/(1.08*(75-45))=$**9847.7 cfm**

Required OA Ventilation Flow Rate (V_{DOT_OA})*:* **2326 cfm** (std 62.1-2004 + 30%)

<u>Point A</u>

$V_{DOT\_OA}+V_{DOT\_RA}= V_{DOT\_MA}$ $V_{DOT\_MA}= V_{DOT\_SA}$
-> solving for $V_{DOT\_RA}= V_{DOT\_MA}- V_{DOT\_OA}$=9847.7-2326=**7521.7 cfm**

Energy Balance: $V_{DOT\_OA}*T_{OA}+ V_{DOT\_RA}*T_{RA}= V_{DOT\_MA}*T_{MA}$

-> solving for $T_{MA}= \frac{(2326cfm*92F)+(7521.7cfm*75F)}{9847.7\ cfm}$ = **78F**

-> solving for $W_{MA}= \frac{(2326cfm*98gr/lbma)+(7521.7cfm*65gr/lbma)}{9847.7\ cfm}$ = **70.9 gr/lbma**

<u>Point B</u>

If supply at 45F and saturated conditions, the air temperature after the cooling coil will be at this temperature as well.

<u>Cooling Coil Load</u>

$Q_{SENSIBLE}=1.08* V_{DOT\_SA}*(T_{MA}- T_{SA})$=1.08*9847.7*(78-45)=**350972 BTU/hr**
$Q_{LATENT}=0.68* V_{DOT\_SA}*(W_{MA}-W_{SA})$=0.68*9847.7*(70.9-45)=**173437.7 BTU/hr**

Q_{TOTAL}=(350972+173437.7 BTU/hr)/12000 (BTU/hr/ton)= **43.7 tons**

Why DOAS/Parallel Radiant Floor over conventional VAV systems?

Using a dedicated outdoor air system has several benefits over using a conventional VAV system. When applying a DOAS system, each space is guaranteed to receive the minimum amount of outdoor air required by the standard, since each space receives only 100% outdoor air. In an all air VAV system, OA is normally mixed with a percentage of air returning from the space. Once mixed, this air is then distributed to the spaces according to VAV box settings. Theoretically, the proper amount of outdoor air should reach the occupants even if the VAV box is set at a minimum, but this is very hard to prove. As seen in the following example, the amount of outdoor air being brought in by a VAV system is determined by the minimum damper setting and the control of the system. The following calculation example was created by Dr. Stanley Mumma for the AE 555 Building Automation and Controls course.

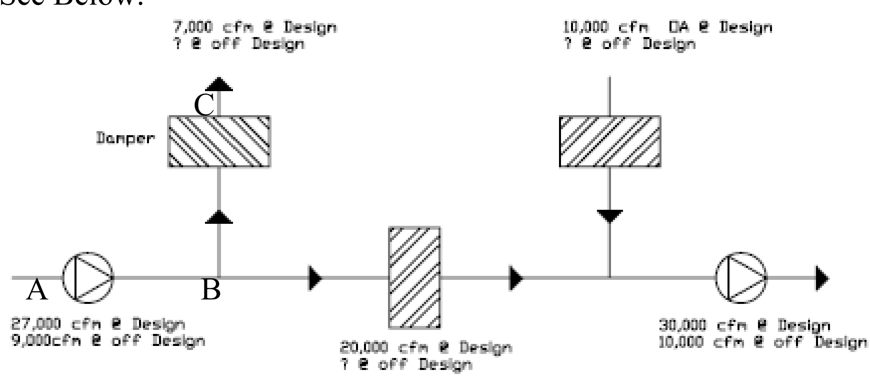
Example Problem

Objective: This problem illustrates some potential problems with maintaining the minimum outdoor air requirements in VAV systems, when a minimum outdoor air damper position is selected.

Assumptions: The return fan is controlled so that return flow is 10% less than supply.
Under all conditions the OA, exhaust, and recirculation control dampers remain fixed at the positions reflected in the figure below.
The pressure at point A is 0.25 in. wg at design.

Problem Statement: To find the air flowrates through the dampers at off design.

Solution: See Below.



Find the system characteristic based on design: (A-B) $P=c*Q^2$, $c=0.25/27000^2=$**3.43E-10**
(B_C) $c=0.25/7000^2=$**5.1E-9**

Now, find the new pressure to be maintained based on off design: $P=3.43E\text{-}10*9000^2$
P=**0.0168 in. wg**

Moving to branch B-C, applying the same equation:

$Q_{exhaust}$= sqrt(P/c)= sqrt(0.0168/5.1E-9)= **2343 cfm**

$Q_{recirculated}$= (9000-2343) cfm= **6657 cfm**

Therefore to have 10,000 cfm supplied to the space, there is only 10,000-6657= **3343 cfm OA** supplied, which is less than the 10,000cfm OA required.

In the existing design of the Hearst Tower, (4)-110,000cfm VAV air handling units serve the office floors. Air is supplied and returned at a height of 9.5 ft, making it difficult to prove that the required amount of outdoor air actually reaches the breathing zone of the occupants. This essentially makes it very hard to obtain the LEED point for Indoor Air Quality.

Another benefit of bringing in the minimum amount of outdoor air is the resulting reduction in total load on the cooling coil. As seen in the sample calculations appearing on Pages 14 & 15, the total cooling coil load for the DOAS unit was about 10.5 tons and the load for the parallel system was about 20.3 tons, resulting in a total capacity of 30.8 tons from the chiller plant. In the VAV calculation, the total coil load was approximately 44 tons. This system not only results in a larger coil, but also requires 14 more tons of capacity from the chiller plant.

Choosing a DOAS Air Handling Unit

Since the building is so large and the spaces from floors 10 through 41 are used for the same activity, office work, floors 10 through 21 will serve as a representative system for modeling purposes. Activities that would be of particular concern, such as printing, copying, and cooking are done on the third floor and are served by dedicated air handling units, and are therefore not investigated in this report.

Using the SEMCO Technical Guide, a packaged energy recovery system was chosen. According to the analysis done in Technical Assignment #1, the office portion of the Hearst Tower requires approximately 75000 CFM total of outdoor air. Since the Hearst Tower is striving for a LEED Gold rating, 30% more than this requirement (97500 CFM) will be supplied to ensure the building obtains a point for Indoor Air Quality. Based on this amount of air, the EPCH-43 model was chosen. Three identical units with full load capabilities of 40,000 CFM OA will be used; with one unit serving floors 10^{th} -21^{st}, one serving the 22^{nd} -31^{st}, and the third serving the 32^{nd}-41^{st} floors. This unit provides year-round supply air conditioning and includes an enthalpy wheel, a cooling coil, heating coil, and steam humidification.

The next step in the selection process was to determine the internal static pressures for the supply and return side. A summary of these results can be seen in the following table, Table 1.

Table 1 Internal Static Pressure Calculations

Internal Static Pressures for SA And RA side			
SA @40000 CFM		RA @ 40000 CFM	
OA Opening	0.04	EA Opening	0.04
SA Opening	0.14	RA Opening	0.26
Damper	0.19	Damper	0.19
OA Filter	0.62	RA Filter	0.98
Wheel	0.98	Wheel	0.98
CHW	0.83	Casing	0.3
HW	0.13	Total (in.wg)	2.75
Casing	0.3		
Total (in.wg)	3.23		

The total fan static pressure is then determined by adding the internal static pressure to the external static pressure required.

SA Side: 3.23+1=4.23 in. wg
RA Side: 2.75+0.5=3.25 in. wg

Using the tables provided in the Technical Guide, which can be seen in the appendix section of this report, it was determined that 2662 CFM of outdoor air are needed for enthalpy wheel purge. Therefore the fan will be required to deliver 42662 CFM and overcome the total static pressures found above. Using the given fan tables, the SA fan will require a 55 hp motor while the RA fan will require a 50 hp, taking into account a 10% factor for drive losses and safety factors.

The variable speed option will also be selected for the wheel. This option not only helps protect the wheel from freezing and condensation problems, but helps the new design to remain consistent with energy conservation steps taken in the existing system. In the original design for the Hearst Tower, the owner was awarded rebates from the New York State Energy Research and Development Authority for using variable speed/frequency drives wherever possible.

Enthalpy Wheel Composition

The SEMCO enthalpy wheel uses a 3 angstrom molecular sieve desiccant coating to help reduce the risk of cross-contamination between the exhaust and supply air streams. The wheel itself is formed into a honeycomb shape and then coated with this desiccant to help protect the aluminum from corrosion. Other desiccant coatings such as silica gel and oxidized aluminum are available; however molecular sieves have a high affinity for water vapor. This high affinity results in a higher rate of absorption and subsequent higher latent transfer rate.

Radiant Floor Heating and Cooling

Radiant floor heating is an idea that has been in employed in buildings for centuries; however the use of radiant floors for cooling is an idea only recently gaining understanding and implementation. In the case of the Hearst Tower, radiant floors have been integrated into the lobby/atrium space to supplement a hydronic cooling system made up of fan coil units and forced cool air.

Radiant floors work in areas such as lobbies and atriums because the ceilings are usually very high and the occupant level is closer to the floor. The area of thermal comfort concern becomes a height of 6-8 feet above the floor and the remaining height of the space is not of concern. For heating this allows occupants to feel comfortable, even if the surrounding air temperature is lower than 70F. Since radiant flooring is normally the most massive part of the room, heat is transferred to the surrounding objects in the room from the floor. The occupants therefore radiate little to no heat to the surroundings and thus feel warmer. Radiant floor applications also provide an even distribution of heating and cooling when designed correctly, adding to the energy conservation ability of the system.

The radiant flooring system analyzed for the Hearst Tower is a hydronic radiant floor embedded in the composite floor slab. The floor covering will be carpet and padding with a total thickness of 0.79 in. Based on a radiant floor study conducted in AE 554, System Simulation and Optimization, and a Radiant Floor Model program written by Dr. Xia of Flack and Kurtz, the thermal resistance of the coverings is assumed to be 0.18 Fhrft^2/BTU. The Radiant Model program will then be used to estimate the cooling capacity of each radiant zone based on the solar gain intensity received by that zone.

Using the Wirsbo Installation Handbook, 5/8" Wirsbo hePEX tubing was chosen. Wirsbo hePEX is cross-linked polyethylene heat transfer tubing with an oxygen diffusion barrier. This is also the type of tubing currently in place for the existing radiant floor in the lobby.

Estimating the Solar Gain Intensity

In order to optimize the zoning of the radiant floor, a daylighting study was conducted using the AGI32 Daylighting Study Program. The floor plan and surrounding buildings were entered into the study to get an accurate, or as close to accurate as possible, portrayal of the shading on the tenth floor. Again, the tenth floor was used as a representative floor because of the size of the building.

The daylighting study was conducted on four days of the year: December 21st, March 21st, June 21st, and September 21st. After viewing the solar patterns on the floor for each day from 6 am to 6 pm, the following zoning pattern was developed (see Figure 6).

Figure 6 Radiant Floor Zoning

Once this zoning was established, the incidence solar beam radiation on that particular zone was found using the Hottel Clear Sky Model (1976). A spreadsheet detailing all of the necessary angles and intensities is included in Appendix A.

The basic technique described herein follows the procedure set forth in "Solar Engineering of Thermal Processes" (Duffie & Beckman 1991).

The atmospheric transmittance for beam radiation is given by: $\tau_b = a_0 + a_1 \exp(-k/\cos\theta_z)$

The Zenith Angle, θ_z, is the angle between the vertical and the line to the sun.

The coefficients are found by multiplying the following values by a correction factor:

$$a_0^* = 0.4237 - 0.00821*(6-A)^2$$
$$a_1^* = 0.5055 - 0.00595*(6.5-A)^2$$
$$k^* = 0.2711 - 0.01858*(2.5-A)^2$$

(Where A is the altitude of the observer, or in the case of the Hearst analysis the altitude of the 10^{th} floor.)

New York is considered Midlatitude Summer, and therefore the following correction factors are: $r_0=0.97$, $r_1=0.99$, $r_k=1.02$.

From these values, the clear sky instantaneous beam radiation (G_b) can be determined.

G_b=1367*(1+0.033cos(360*n/365)* τ_b*cosθ_z [J/s*m^2], where n is nth day of the year.

For hourly beam radiation, I_b= G_b*3600 s/hr.

These values are for a horizontal surface being hit directly by sunlight. The actual scenario is a vertical fenestration with solar beam radiation incident upon the surface. A portion of this incidence radiation is then transmitted through the window and reaches the floor. In order to correct the horizontal beam radiation for the vertical fenestration case, a geometric factor, R_b, was multiplied by I_b .

$$R_b=\cos\theta/\cos\theta_z$$

Once the corrected hourly beam radiation is determined, the corrected instantaneous beam radiation can be found.

$$G_{bcorrected}=I_{bcorrected}/(3600\ s/hr)$$

This value was further corrected based on the transmissivity (25%) of the glass curtain wall.

These values were then entered into the Radiant Floor Model discussed previously to determine the cooling capacity of each zone. Using a consist spacing of 6”, it was determined that allowing zones control over the radiant system is the best choice. During times of the day when a particular zone has large solar gain intensity, the flow rate through that zone can be modulated as necessary to maintain a delta T of 5F.

Figure 7 Radiant Floor Schedule

RADIANT HEATING/COOLING SCHEDULE (WIRSBO)

MANIFOLD NO.	AREA	NUMBER OF LOOPS PER MANIFOLD	EWT-LWT HEATING	EWT-LWT COOLING	TUBE TYPE (hePEX)	APPROX. LOOP LENGTH (FEET)	GPM/LOOP HEATING	GPM/LOOP COOLING
1	1418	8	105-75	61-66	5/8"	300	1	2
2	1425	8	105-75	61-66	5/8"	300	1	2
3	2025	10	105-75	61-66	5/8"	300	1	2
4	2455	10	105-75	61-66	5/8"	300	1	2
5	2025	10	105-75	61-66	5/8"	300	1	2
6	1425	10	105-75	61-66	5/8"	300	1	2
7	1418	10	105-75	61-66	5/8"	300	1	2

Combining Steam Driven Absorption Cooling with a DOAS/Radiant Floor System

As described above, the proposed mechanical system for the Hearst Tower will utilize Steam Driven Absorption Cooling in conjunction with a Dedicated Outdoor Air/Radiant Floor system. The system will also deliver low temperature (45F) supply air. Employing a low temperature DOAS air side system resulted in a 36% reduction in the size of the chiller plant, in addition to a 70% decrease in the size of air handling equipment. Therefore the new system configuration consists of:

(3)-600 ton double effect absorption chillers
(3)-40,000 cfm packaged energy recovery AHU's
Radiant slabs installed from floors 10-42

The new absorption units will be located in the basement and arranged in parallel. The existing pressure reducing station will also house the pressure reducing valve necessary to achieve the 114 psig steam required by the new chillers.

Since the new chillers are of a smaller capacity than the existing chillers, new condenser water and chilled water pumps will be installed. The Armstrong pumps that are currently used can be downsized to 1500rpm from 1800rpm, while the chilled water pumps are reduced from 1800 rpm to 1200 rpm. Again the configuration of the pumps will remain the same as the existing design: four headered chilled water pumps and four headered condenser water pumps.

The new chiller plant will also be tied into the current Building Management and Control System which operates on DDC control logic. Chilled water temperature will thus be modulated depending upon the building loads and outdoor air temperature.

The packaged energy recovery air handling units will be placed on the 28th floor, which is an existing mechanical floor. Currently this floor contains the four large (110,000cfm) AHU's and room for a future AHU of the same size, therefore additional space will not be needed for the proposed air system design.

The new 40,000cfm units are sized to handle the minimum amount of required outdoor air plus 30%, as well as an additional 2700 cfm necessary for purging the enthalpy wheel. Freezing and condensation will be controlled by the variable speed drive applied to the wheel. A dewpoint temperature sensor will modulate the speed of the wheel, which in turn limits the amount of energy recovery from the exhaust stream.

As stated before, a DOAS unit is responsible for satisfying the entire latent load and a portion of the sensible, with the remaining sensible load being met by the parallel system. In a VAV system, the air handling unit must meet the entire latent and sensible loads, therefore more air must be circulated and fans need to be sized to meet this flowrate. The existing AHU's in the Tower require 100 hp supply and return fans while the new DOAS-Radiant design will only require 55 hp fans on the supply side and 50 hp fans for return. Smaller fan power requirements and the implementation of variable frequency drives will help this system conserve more energy, as compared to the original. The annual energy savings will be discussed in "System Simulation" section of this report. As necessary, all of the equipment and instrumentation changes discussed above were included in the system simulation.

System Simulation

As there is no program that exists to model DOAS/Radiant systems, manipulations to Carrier's Hourly Analysis Program (HAP v4.20) were necessary to get a model for comparison to the original VAV system. With the advisement of Dr. Stanley Mumma, one air handling unit serving floors 10-21 served as a representative system for this analysis. These floors are open office spaces and are typical for the majority of floors in the tower.

Initially, a load analysis was conducted in HAP to find the cooling and heating loads for the representative floors. For one floor, the resulting design conditions were as follows:

Cooling Design Conditions	
Outdoor Air DB/WB	92F/74F
Sensible Load (BTU/hr)	319067
Latent Load (BTU/hr)	31629
Room Air DB/RH	75F/50%

Heating Design Conditions	
Outdoor Air DB/WB	13F/10.4F
Sensible Load (BTU/hr)	82785
Latent Load (BTU/hr)	0
Room Air DB/RH	70F/50%

With the space sensible and latent loads determined, all of the existing equipment in the current VAV system was entered into HAP to produce a model that would serve as a basis of comparison for the newly proposed system. Annual energy cost results for this system are shown in detail later in this section in Table 2- Table 7.

To begin the calculations for the new DOAS system, the procedure described in the sample calculation in the "Dedicated Outdoor Air Systems" section of this report was conducted. This calculation helped determine the amount of cooling load satisfied by the DOAS system as well as the capacity needed for the parallel radiant flooring system. Approximately 75,363 BTU/hr of sensible load and the entire latent load were satisfied with the DOAS unit, leaving 243,705 BTU/hr of sensible load to be met by the radiant floors.

Upon inspection of the zone loads initially calculated using HAP, the load due to people, electrical equipment, and outdoor air added up to approximately the same load that the DOAS unit should meet. This load was therefore assigned to a constant volume DOAS unit, and the unit was modeled as an interior space, i.e. no wall/window exposure. The following figure, Figure 8, shows the load output assigned to the representative system (approximately 10 tons). The results from the DOAS hand calculation (shown previously) determined a total DOAS cooling capacity of 10.5 tons, for the given design conditions. Therefore modeling in this manner gave a close approximation (within 5%)

to the total load the DOAS system is responsible for and will ultimately be used to calculate the energy use by a DOAS system handling such a load.

Figure 8 Loads Assigned to DOAS Unit

	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Jul 1500 COOLING OA DB / WB 92.0 °F / 74.0 °F			HEATING DATA AT DES HTG HEATING OA DB / WB 13.0 °F / 10.4 °F		
ZONE LOADS	Details	Sensible (BTU/hr)	Latent (BTU/hr)	Details	Sensible (BTU/hr)	Latent (BTU/hr)
Window & Skylight Solar Loads	0 ft²	0	-	0 ft²	-	-
Wall Transmission	0 ft²	0	-	0 ft²	0	-
Roof Transmission	0 ft²	0	-	0 ft²	0	-
Window Transmission	0 ft²	0	-	0 ft²	0	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Loads	0 ft²	0	-	0 ft²	0	-
Floor Transmission	0 ft²	0	-	0 ft²	0	-
Partitions	0 ft²	0	-	0 ft²	0	-
Ceiling	0 ft²	0	-	0 ft²	0	-
Overhead Lighting	0 W	0	-	0	0	-
Task Lighting	0 W	0	-	0	0	-
Electric Equipment	264000 W	900764	-	0	0	-
People	1782	436584	365310	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	0% / 0%	0	0	0%	0	0
>> Total Zone Loads	-	1337348	365310	-	0	0
Zone Conditioning	-	0	365310	-	0	0
Plenum Wall Load	0%	0	-	0	0	-
Plenum Roof Load	0%	0	-	0	0	-
Plenum Lighting Load	0%	0	-	0	0	-
Return Fan Load	33264 CFM	0	-	33264 CFM	0	-
Ventilation Load	33264 CFM	0	0	33264 CFM	0	0
Supply Fan Load	33264 CFM	0	-	33264 CFM	0	-
Space Fan Coil Fans	-	0	-	-	0	-
Duct Heat Gain / Loss	0%	0	-	0%	0	-
>> Total System Loads	-	0	365310	-	0	0
Precool Coil	-	1439247	365310	-	0	0
Preheat Coil	-	-487329	-	-	0	-
>> Total Conditioning	-	951918	365310	-	0	0
Key:	Positive values are clg loads Negative values are htg loads			Positive values are htg loads Negative values are clg loads		

(951918+365310 BTU/hr)/11 Floors= 119748 BTU/hr total load for one floor

(119748 BTU/hr)/12000 BTU/hr/ton= 10 tons

Setting up the radiant floor system in HAP became the next challenge. The same basic idea was used for assigning the loads. According to Sample Calculation #1, a total parallel capacity of about 20 tons is needed to condition the space. Assigning loads due to solar gain, window and wall transmission and lighting to the radiant system resulted in a total capacity of 18 tons of cooling provided by the radiant floor. This is within 10% of the required load calculated in the sample problem and therefore will serve as an adequate model for energy consumption calculations.

Another issue with modeling radiant floors is that this system is not available as a “system” within the HAP analysis program. At the advisement of Dr. Jeong, the parallel system was modeled as a 2-pipe fan coil unit, with the supply temperature set at 61F, which is well above the space dewpoint temperature (i.e. the unit will not handle any latent load) and will be the supply temperature of the radiant floor. The fan coil unit static pressure was also set to zero, since the radiant flooring system does not have fans. With the loads assigned, the created fan coil/radiant floor system will be used to calculate energy consumption. Figure 9 below shows the loads assigned to the model.

Figure 9 Loads Assigned to Radiant Floor

	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Jul 1500 COOLING OA DB / WB 92.0 °F / 74.0 °F			HEATING DATA AT DES HTG HEATING OA DB / WB 13.0 °F / 10.4 °F		
ZONE LOADS	Details	Sensible (BTU/hr)	Latent (BTU/hr)	Details	Sensible (BTU/hr)	Latent (BTU/hr)
Window & Skylight Solar Loads	73805 ft²	967013	-	73805 ft²	-	-
Wall Transmission	9356 ft²	12360	-	9356 ft²	30651	-
Roof Transmission	0 ft²	0	-	0 ft²	0	-
Window Transmission	73805 ft²	233561	-	73805 ft²	925508	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Loads	0 ft²	0	-	0 ft²	0	-
Floor Transmission	0 ft²	0	-	0 ft²	0	-
Partitions	0 ft²	0	-	0 ft²	0	-
Ceiling	0 ft²	0	-	0 ft²	0	-
Overhead Lighting	332640 W	1134946	-	0	0	-
Task Lighting	0 W	0	-	0	0	-
Electric Equipment	0 W	0	-	0	0	-
People	0	0	0	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	0% / 0%	0	0	0%	0	0
>> Total Conditioning	-	2347880	0	-	956160	0

2347880 BTU/hr/11 floors= 213443.6 BTU/hr for one representative floor
213443.6 BTU/hr/12000 BTU/hr/ton= 18 tons

The chiller plant and corresponding cooling towers selected and described in the "Absorption Cooling" section of this report where then added to the simulation. Absorption cooling was analyzed because of the incentives offered by Con Ed for buildings utilizing steam as a power source for cooling and because of the high cost of electricity in New York City ($0.18/kWh). Such buildings fall into Service Classification No. 2 and are offered special incentives set forth in "Special Provisions part D," (a copy of Service Classification No. 2 is included in the Appendix B of this report). The basic incentive offered by Con Ed is a reduction of $2.00 per 1000 lbs. of steam used in excess of the first 250,000 lbs.

With the energy analysis completed in HAP, monthly energy consumption reports for each energy source were calculated. These results were then tabulated in Excel and multiplied by the corresponding rate structure for steam and flat rate for electricity. From the following figures, Table 2 though Table 4, the total cost for the proposed system was approximately $505,000 per year. Tables 5 through 7 show the results for the original electric chiller/VAV system. These results reflect a total cost of $862,000 per year. Therefore the new system provides an annual savings of about 41%.

Table 2 Yearly Steam Cost for Absorption Cooling

Steam Cost for Absorption Cooling/DOAS System					
Month	therm	lb	Cost $	Monthly Charge $	Total Monthly Cost $
January	4455	1400206.5	25114.51	2455.35	27569.86
February	4572	1436979.6	25856.19	2455.35	28311.54
March	5275	1657932.5	30083.27	2455.35	32538.62
April	4088	1284858.4	22788.06	2455.35	25243.41
May	3981	1251228.3	8079.18	2455.35	10534.53
June	7136	2242844.8	14482.05	2455.35	16937.40
July	10879	3419269.7	22078.22	2455.35	24533.57
August	11412	3586791.6	23159.91	2455.35	25615.26
September	6176	1941116.8	12533.79	2455.35	14989.14
October	3345	1051333.5	6788.46	2455.35	9243.81
November	4295	1349918.5	24100.26	2455.35	26555.61
December	4755	1494496.5	27024.24	2455.35	29479.59
TOTAL		22116976.7			271552.35

Rate Structure applying Special Provision "D":

Table 3 Steam Rate Structure

May thru Oct		
250,000	0.006777	$/lb
750,000	0.007047	$/lb
over 1 mil lbs	0.006457	$/lb

Nov thru April		
250,000	0.007664	$/lb
1,250,000	0.020169	$/lb
3,500,000	0.018717	$/lb

Applying a flat rate as discussed above:

Table 4 Yearly Electricity Cost (HVAC only)

Electric Cost Absorption/DOAS System		
Month	kWh	Cost $
January	39,941	7189.38
February	36,076	6493.68
March	40,498	7289.64
April	50,886	9159.48
May	80,986	14577.48
June	179,476	32305.68
July	268,452	48321.36
August	273,655	49257.90
September	154,734	27852.12
October	75,606	13609.08
November	55,918	10065.24
December	39,975	7195.50
TOTAL		233316.54

Table 5 Yearly Steam Cost for Electric Chillers/Steam Heat

Steam Cost for Electric Cooling/VAV System					
Month	therm	lb	Cost $	Monthly Charge $	Total Monthly Cost $
January	4354	1368462.2	22850.26	2455.35	25305.61
February	4479	1407749.7	31249.60	2455.35	33704.95
March	5275	1657932.5	35147.02	2455.35	37602.37
April	4018	1262857.4	25451.90	2455.35	27907.25
May	802	252068.6	1726.98	2455.35	4182.33
June	869	273126.7	1903.48	2455.35	4358.83
July	1000	314300	2275.97	2455.35	4731.32
August	961	302042.3	2085.08	2455.35	4540.43
September	789	247982.7	1680.58	2455.35	4135.93
October	542	170350.6	1099.95	2455.35	3555.30
November	4213	1324145.9	26599.04	2455.35	29054.39
December	4680	1470924	31785.92	2455.35	34241.27
TOTAL					213319.98

Rate structure when Special Provision "D" is not applied:

Table 6 Steam Rate Structure

May thru Oct		
250,000	0.006777	$/lb
750,000	0.009047	$/lb
over 1 mil	0.008457	$/lb

Nov thru April		
250,000	0.007664	$/lb
1,250,000	0.020169	$/lb
3,500,000	0.018717	$/lb
20,000,000	0.017973	$/lb

Applying a flat rate for electricity:

Table 7 Yearly Electric Cost (HVAC only)

Electric Cost For Electric/VAV System		
Month	kWh	Cost $
January	155,155	27927.90
February	145,610	26209.80
March	184,792	33262.56
April	237,712	42788.16
May	321,134	57804.12
June	436,479	78566.22
July	519,952	93591.36
August	535,144	96325.92
September	397,500	71550.00
October	294,995	53099.10
November	210,075	37813.50
December	165,068	29712.24
		648650.88

First Cost Analysis

With the aid of documents provided by Flack+Kurtz, the first cost for the existing chillers, ahu's, and all related equipment were determined. As seen in Table 8, the first cost for the chiller plant is $3.0 million while the cost of a representative air system (11 floors) is $5.28 million. Therefore the total first cost of the representative existing system is $8.28 million.

Table 8 First Cost of Existing Equipment

ITEM	COST
Chiller Plant	
3-Electric Centrifugal Chillers	1,043,750
Primary CW pumps	180,000
Primary CHW Pumps	135,000
Secondary CHW Pumps	225,000
Primary CHW Piping	268,750
Secondary CHW Piping	562,500
Primary CW Piping	581,250
	$2,996,250.00
Air Systems	
1-Central AHU's ($2/cfm)	220,000
Perimeter FPTU w/HW coils	254,100
Perimeter FPTU w/HW coils piping distribution	385,000
Interior FPTU Boxes	13,200
Horizontal Supply Ductwork (Typ Tower Floor)	2,117,500
Horizontal Return Ductwork (Typ Tower Floor)	1,980,000
Supply duct risers	309,375
	$5,279,175.00
System Total	*$8,275,425.00*

The DOAS/Radiant system on the other hand has a much higher first cost. The steam driven absorption chillers and corresponding water pumps total approximately $3.1 million. The packaged energy recovery units used for the DOAS System are $3.45 million and the parallel radiant floor system $3.16 million. The new proposed system has a capital cost of approximately $9.71 million. Table 9 on the following page shows the first cost of the system based on the cost of each component.

First cost data for the original system was calculated using information provided by Flack+Kurtz. This information was then scaled down to a representative system of 11 floors. Based on the information from this spreadsheet, the building HVAC system should cost approximately $38.00/SF. Using this value, it was determined that the actual system probably cost is closer to $6.69 million (1600SF/Floor*11 Floors*$38.00/SF). Since using the smaller of the two is being more conservative when calculating the payback period, this value will be used instead of the $8.28 million previously calculated.

As for the proposed system, cost estimates were found using *CostWorks 2005,* industry consultation, and pieces of the cost spreadsheet describing the original system cost.

Using economic information from the Penn State DOAS-Radiant website, $1/CFM for VAV and $4/CFM for DOAS AHU's were used to estimate the cost of airside equipment. Upon consulting with an engineer from O'dea, Lynch, Abbitista CE, an estimate of $16/SF for installed radiant flooring was used to determine the first cost of that system (An estimate of $12.00/SF plus 30% because the building is located in NYC was given).

Table 9 First Cost of Proposed Equipment

ITEM	COST
Chiller Plant	
3-Double Effect Absorption Chillers	1,387,500
Primary CW pumps	129,825
Primary CHW Pumps	77,175
Secondary CHW Pumps	225,000
Primary CHW Piping	241,875
Secondary CHW Piping	506,250
Primary CW Piping	523,125
	$3,090,750.00
Air Systems	
Central AHU ($4/cfm)	160,000
Horizontal Supply Ductwork (Typ Tower Floor)	1,588,125
Horizontal Return Ductwork (Typ Tower Floor)	1,485,000
Supply duct risers	232,031
	$3,465,156.25
Radiant System	
Radiant Floor ($16/SF)	2152656
Flooring (Cost of additional slab)	704,000
Pumps	80,056
Insulation	176,000
Heat Exchangers	50,000
	$3,162,712.00
System Total	*$9,718,618*

Based on the first costs calculated above and the annual savings generated by the system simulation reports, a simple payback period can be determined.

$$\text{Payback Period} = \frac{\text{(Difference in Initial Cost)}}{\text{Annual Savings}}$$

$$\text{Payback Period} = \frac{(\$8.28M - \$6.69M)}{862{,}000 - 505{,}000} = \mathbf{4.5\ years}$$

With the life of a typical building being somewhere in the area of 20-25 years, a payback period of 4.5 years is definitely attractive.

Proposed System Conclusions

Overall, the proposed system of using Steam Driven Absorption Cooling with a Dedicated Outdoor Air/Radiant Flooring System is a definite possibility for the Hearst Tower. The main criteria set forth at the beginning of this study were as follows:

- To secure the LEED point for Indoor Air Quality (IAQ)
- To lessen the Tower's dependence on an already overextended electric grid
- To lower annual energy cost associated with the HVAC System

The results from the system simulation and subsequent annual operating cost calculations illustrate the decreased energy costs associated with the proposed system. This cost savings is largely associated with the incentive programs offered by the utility, Consolidated Edison, for using steam driven cooling instead of electric cooling, especially in months when cooling is at a peak.

The use of a DOAS/Radiant System also contributed greatly to the annual energy cost savings and initial capital cost savings. Decoupling the sensible and latent loads and using an enthalpy wheel for energy recovery from the exhaust air stream allowed for reductions in coiling coil size, chiller size, pumping size, and duct sizes throughout the building. In addition to cost savings, greater indoor air quality is achieved by introducing a constant volume of outdoor air into the building, an amount equivalent to ASHRAE Std. 62-2004 plus 30%. Using the minimum plus 30% helps secure the LEED point for IAQ and is actually the amount of air required to meet the latent load of the space given the supply air conditions, 45F and saturation.

Additional Considerations

Although the results came out favorable for this design analysis, there are some additional considerations that should be made. Upon reading the special provision section of Service Classification No. 2, a rate structure change will probably take place, for buildings using steam power, in September of 2006. If a building starts service under this agreement before October 2006, then Con Ed will provide service under these provisions for two years. After two years, a new rate structure may be applied to the steam service. Since the payback period is longer than the amount of time the utility can guarantee the service rate, using the proposed design may not be best case scenario. To be absolutely sure, the building owner should definitely try to negotiate a longer service contract with the utility.

Another consideration to take into account is that there is no accurate way to model a DOAS/Radiant System using HAP. The simulation performed did have some discrepancies from the DAOS hand calculations, as mentioned in the System Simulation Section of this report. Therefore, the DOAS/Radiant system may use slightly more energy than estimated by the model. Having an accurate rate structure for electricity use would also produce a more accurate cost analysis, although using $0.18/kWh is a conservative estimate.

Another key issue with the proposed system is the affect of installing radiant floors throughout the office tower on the construction schedule. The current delivery method for the project is a guaranteed maximum price contract of $252 million. Installing radiant floors may have increased the construction time of the floors, and since time is money, this may have translated into a higher construction cost.

Ultimately the goals set forth for the proposed design were met without sacrificing the goals set forth by the MEP design team, Flack+Kurtz. Although there are additional considerations that would be investigated if the system were actually to be installed, the proposed design did serve the purpose of providing an educational model and meeting the design criteria set forth. The proposed system also has little impact on the architectural program of the Hearst Tower. All of the new equipment is smaller than the original, thus allowing for more space in MER rooms for maintenance and future expansion.

The systems directly affected by the mechanical system changes are the electrical system and the structural system of the building. These two areas therefore served as the breadth studies for this report. The following two sections will discuss the changes made to both systems due to the addition of major mechanical equipment. Further conclusions about the feasibility of the DOAS/radiant system will be discussed within each breadth section.

Electrical Breadth Study:

Determining the Effects of the Proposed Mechanical Equipment on the Electrical System

Electrical Breadth

The Hearst Tower is currently served by (4)-4000 amp service takeoffs. Each takeoff is then served by one 6000A service switch. Secondary service to the building is 480/277 V, three phase from the primary transformers.

Since the building is set to achieve LEED Gold certification, steps were taken to make the electrical/lighting systems energy efficient. Occupancy sensors are installed throughout the tower and work to control the lighting system, shutting down the lights and unused electrical equipment when a space is unoccupied. Daylight sensors are also used in conjunction with the lighting controls to reduce energy use and maximize the use of natural light. Aside from the energy conservation benefits, studies have shown natural light improves productivity and occupant health.

Electrical Equipment: Existing and Proposed

The current chiller plant serving the Tower contains three electric centrifugal chillers. There are (2)-1200 ton chillers and (1)-400 ton chiller served by four headered chilled water pumps and four headered condenser water pumps. All eight pumps are 100 hp split-coupled, vertical inline centrifugal pumps.

Since chiller one and two (both 1200 ton) are so large and require a large electric power input, both chillers are connected directly to a service switch, service switch #2. Each chiller is fed from the switch by a 2000A fuse. The chillers are wired to the service switch using six sets of 3-#400 MCM with 1-#3/0 ground in six 3" conduits.

Chiller three, the smallest of the three, is fed from switchboard #1 using an 800A fuse. This chiller is wired with two sets of 3-#500 MCM with a #2/0 ground wire in two 3.5" conduits.

The proposed chiller plant contains three 600 ton steam driven absorption chillers. Since these chillers use steam as a power source, very little electric power is required. In addition to the annual energy savings calculated in the "System Simulation" section of this report, changing the chillers results in a large first cost savings. According to the chiller specifications provided by Carrier, the total electric current needed for the chillers is 30A. Therefore, the new chillers can be fed by a 40A fuse and 3-#6 wires with one #10 ground wire in 1" conduit.

The airside of the system analyzed in this report includes four large central air handling units located in the 28th floor mechanical equipment room. These units are designed for a maximum airflow of 110,000 cfm each, and therefore have very large supply fans. Each unit has 100 hp dual supply fans with 124A full load ampacity (FLA) each. The supply fans are wired using 3-#2/0 with one #4 ground wire in 2" conduit.

The return fans are separate from the air handling units and are also located in the 28th floor mechanical room. The return fans are 50hp 65A full load ampacity fans. These fans are wired using 3-#2/0 with a #8 ground in a 1.5" conduit. A complete motor control schedule describing the current equipment is provided in Figure 10.

Figure 10 Existing AHU Motor Control Schedule

EXISTING MOTOR CONTROL CENTER SCHEDULE

FLOOR MOUNTED | VOLTS: 460Y/265V | BUS RATING: 2000A | PHASE: 3 | WIRES: 4

DESIGNATION	HP or (kW)	FULL LOAD AMP	STARTER NEMA SIZE	SWITCH SIZE	FUSE SIZE	FEEDER SIZE
PANEL HV-28-M	(42.94kW)	54	–	200AS	200AF	200 NG
AHU-28-1-1	100	124	–	200AS	175AF	175 G
AHU-28-1-2	100	124	–	200AS	175AF	175 G
AHU-28-2-1	100	124	–	200AS	175AF	175 G
AHU-28-2-2	100	124	–	200AS	175AF	175 G
AHU-28-3-1	100	124	–	200AS	175AF	175 G
AHU-28-3-2	100	124	–	200AS	175AF	175 G
AHU-28-4-1	100	124	–	200AS	175AF	175 G
SPARE	100	124	–	200AS	175AF	–
SPARE	100	124	–	200AS	175AF	–
AHU-28-4-2	100	124	–	200AS	175AF	175 G
RETURN FAN RF-28-2	50	65	–	100AS	100AF	115 G
RETURN FAN RF-28-3	50	65	–	100AS	100AF	115 G
RETURN FAN RF-28-5	50	65	–	100AS	100AF	115 G
RETURN FAN RF-28-6	50	65	–	100AS	100AF	115 G
EXHAUST FAN EF-28-7	15	21	2	60AS	60AF	60 G
EXHAUST FAN EF-28-8	7.5	11	1	30AS	20AF	20 G
EXHAUST FAN EF-28-9	15	21	2	60AS	60AF	60 G
EXHAUST FAN EF-28-10	0.5	1.1	1	30AS	15AF	20 G
HOT WATER HEATER	15	19	–	30AS	30AF	30 G
SPARE		–	1	60AS	–	–
SPARE	50	65	–	100AS	100AF	–
SPARE	–	–	1	100AS	–	–
PUMP MUP-28-1	3	4.8	1	30AS	15AF	20 G
SPARE	–	–	1	100AS	–	–
SPACES	–	–	–	–	–	–
TOTAL CONNECTED LOAD (AMPS)		1,697				

The motor control center (MCC) described above will support a total connected load of 1697 amps. Using the 1.25 multiplier for wire sizing, the MCC requires six sets of 3-#500 MCM wires with one #250 MCM ground in six runs of 3.5" conduit.

The proposed air side changes will require much smaller AHU's than those currently installed in the Hearst Tower. The new system will contain (3)-40,000 cfm dedicated outdoor air units containing both a supply and return fan. The supply fans will be 50 hp, 77A full load ampacity and will be wired using 3-#2 with 1-#8 ground in 1.5" conduit. The return fans are slightly smaller, requiring only 50 hp and 65A (FLA). These fans require 3-#4 wires with 1-#8 ground wire in 1.5" conduit. A schedule showing the proposed equipment motor control is shown below in Figure 11.

Figure 11 Proposed Equipment Motor Control Schedule

NEW MOTOR CONTROL CENTER SCHEDULE						
FLOOR MOUNTED	VOLTS: 480Y/265V		BUS RATING: 2000A		PHASE: 3	WIRES: 4
DESIGNATION	HP or (kW)	FULL LOAD AMP	STARTER NEMA SIZE	SWITCH SIZE	FUSE SIZE	FEEDER SIZE
PANEL HV-28-M	(42.94kW)	54	–	200AS	200AF	200 NG
AHU-28-1-SF	55	77	–	100AS	100AF	100 G
AHU-28-1-RF	50	65	–	100AS	90AF	85 G
AHU-28-2-SF	55	77	–	100AS	100AF	100 G
AHU-28-2-RF	50	65	–	100AS	90AF	85 G
AHU-28-3-SF	55	77	–	100AS	100AF	100 G
AHU-28-3-RF	50	65	–	100AS	90AF	85 G
SPARE	–	–	–	100AS	100AF	–
SPARE	–	–	–	100AS	90AF	–
EXHAUST FAN EF-28-7	15	21	2	60AS	60AF	60 G
EXHAUST FAN EF-28-8	7.5	11	1	30AS	20AF	20 G
EXHAUST FAN EF-28-9	15	21	2	60AS	60AF	60 G
EXHAUST FAN EF-28-10	0.5	1.1	1	30AS	15AF	20 G
HOT WATER HEATER	15	19	–	30AS	30AF	30 G
SPARE		–	1	60AS	–	–
SPARE	50	65	–	100AS	100AF	–
SPARE	–	–	1	100AS	–	–
PUMP MUP-28-1	3	4.8	1	30AS	15AF	20 G
SPARE	–	–	1	100AS	–	–
SPACES	–	–	–	–	–	–
TOTAL CONNECTED LOAD (AMPS)		623				

The proposed motor control center (MCC) will support a much lower connected load than the original, serving on 623 amps. Again, using the 1.25 multiplier for sizing wire, the MCC requires three sets of 3-#500 MCM with one #2/0 ground wire. The three sets will be placed in three runs of 3.5" conduit. Figure 12 shows a feeder schedule, which explains the feeder sizing used in Figure 10 & 11.

Figure 12 Feeder Schedule

FEEDER SCHEDULE							
		COPPER CONDUCTORS (75° C) PER SET					
FEEDER DESIGNATION	No. OF SETS	PHASE CONDUCTORS	P PARTIAL NEUTRAL CONDUCTOR	N NEUTRAL CONDUCTOR	NN OVERSIZED NEUTRAL CONDUCTOR	G GROUND CONDUCTOR	I ISOLATED & EQUIPMENT GROUND CONDUCTOR
15	1	3 # 12		1 # 12	1 # 10	1 # 12	2 # 12
20	1	3 # 12		1 # 12	1 # 8	1 # 12	2 # 12
30	1	3 # 10		1 # 10	1 # 6	1 # 12	2 # 12
50	1	3 # 8		1 # 8	1 # 2	1 # 12	2 # 12
60	1	3 # 6		1 # 6	1 # 1	1 # 10	2 # 10
85	1	3 # 4		1 # 4	1 # 2/0	1 # 8	2 # 8
100	1	3 # 2	1 # 8	1 # 2	1 # 3/0	1 # 8	2 # 8
115	1	3 # 2	1 # 6	1 # 2	1 # 4/0	1 # 8	2 # 8
130	1	3 # 1	1 # 6	1 # 1	1 # 300	1 # 6	2 # 6
150	1	3 # 1/0	1 # 4	1 # 1/0	2 # 1/0	1 # 6	2 # 6
175	1	3 # 2/0	1 # 2	1 # 2/0	2 # 2/0	1 # 4	2 # 4
200	1	3 # 3/0	1 # 2	1 # 3/0	2 # 3/0	1 # 4	2 # 4

Cost Savings

Thus far, only the nominal size differences between the wiring for the existing and proposed mechanical systems have been discussed. The differences in sizes do however equate to substantial first cost savings in some instances. The wiring currently installed throughout Hearst is of type THHN cooper, since this wire type is pretty standard, it will also be used with the redesigned system. The table below, Table 10, shows the pricing differences between both systems. All cost data was found from R.S. Means 2005 sections 16120 "Conductors and Cable," and section 16132 'Conduit."

Table 10 Wire and Conduit Cost Table

Existing System

Size	Item	Service	Unit Cost (per 100 L.F.)
400 MCM	THHN Wire	Chiller 1&2	660
#3/0	THHN Wire	Chiller 1&2 GRND	386
500 MCM	THHN Wire	Chiller 3	765
#2/0	THHN Wire	Chiller 3 GRND	296
#2/0	THHN Wire	AHU-28-1,2,3,4-SF	296
#4	THHN Wire	AHU-28-1,2,3,4-SF GRND	136
#2	THHN Wire	AHU-28-1,2,3,4-RF	178
#8	THHN Wire	AHU-28-1,2,3,4-RF GRND	78

Size	Item	Service	Unit Cost (per L.F.)
3"	Conduit	Chiller 1&2	41.50
3.5"	Conduit	Chiller 3	49.50
2"	Conduit	AHU-28-1,2,3,4-SF	20.50
1.5"	Conduit	AHU-28-1,2,3,4-RF	16.40

Proposed System

Size	Item	Service	Unit Cost (per 100 L.F.)
#6	THHN Wire	Chiller 1,2,3	100
#8	THHN Wire	AHU-28-1,2,3 SF&RF GRND	78
#2	THHN Wire	AHU-28-1,2,3 SF	178
#4	THHN Wire	AHU-28-1,2,3 RF	136

Size	Item	Service	Unit Cost (per L.F.)
1"	Conduit	Chiller 1,2,3	12.13
1.5"	Conduit	AHU-28-1,2,3 SF&RF	16.40

From the cost tables above, it was determined that the wiring for Chiller 1&2 would cost about $391.00/linear foot each. Wiring for Chiller 3 would cost $150.00/ linear foot. With the proposed system requiring only a 40 amp feed each, wiring would only cost around $15.60/linear foot. Since the chillers would be located in the same space as the existing chillers, the distance to the main electrical room from each chiller would be the same and these numbers can therefore be used to estimate cost savings.

Original design for chiller plant (2@$391.00+$150.00): $932.00/linear foot
Proposed design for chiller plant (3@$15.60): $46.80/linear foot

Therefore, the cost of wiring the new chiller plant is about 5% of the cost of the original plant.

The wiring cost for the original air handling units is approximately $213.00/ linear foot, while the new air system for all three units is $131.34/linear foot. The new air system results in a 40% savings in wiring costs.

All wire sizes were determined using the method described in Chapter 11 and Table 11.1 of “Electrical Systems in Buildings” (Hughes 1988). The tables provided within this chapter are replicated from the National Electric Code. Power requirements for the proposed equipment were found in the equipment specifications included in the Appendix section of this report.

Downsizing of the major mechanical equipment in the Hearst Tower resulted in large first cost savings of wiring and conduit. Installing equipment will less required power input allows circuit breaker frame size to be decreased which saves on panel board space and allows for easier installation; ultimately this results in lower cost circuit breakers with respect to both first cost and installation. Essentially the main goal of the proposed design was achieved, lowering the electrical consumption of the building and lessening the dependence on the district electric grid of NYC.

Structural Breadth Study:

Determining the Effects of Radiant Floors on the Building Structure

Structural Breadth

Landmark Façade

As mentioned before, the building has two distinct pieces: the historic landmark façade and the new office tower. The two portions are brought together by a skylighting system and clearstory that wraps the base of the new tower. The 200 square foot base building will house lobby space and 80 foot atrium space.

Preserving the 1928 landmark façade posed several challenges for the structural design team and construction manager. Demolition of the existing Hearts Headquarters called for the six story building to be completely gutted leaving three remaining facades on 56th and 57th street and Eighth Avenue. Aside from simply constructing the new tower, the construction team also introduced a new framing plan to support the exterior walls and upgraded the existing façade to meet the seismic requirements of the New York City Building Code.

Foundation

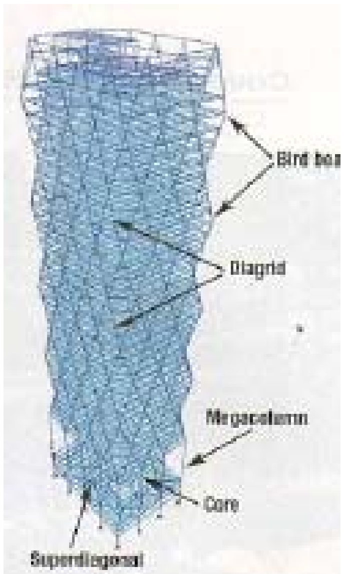
Tests done at the site revealed that there is a sharp drop in the elevation of the rock below grade. The rock elevation varied from 30 feet down to only a few feet in some spots. Because of this variation in elevation, half of the tower is supported by spread footings on rock, while the other half is supported by caissons embedded into the rock below.

Lateral System

Three sides of the building are open to the street as mentioned above, however the fourth face is against an existing high-rise to the west. From an architectural program standpoint, the most logical placement for the service corridor was toward the west side of the tower, since having a window view in these areas is unnecessary. Locating the corridor in this offset manner however eliminated the benefit of using the core as the "main spine of the tower" (Rahimian 2006). To address the issue of instability, the structural design team developed a plan to use the perimeter structure to create the needed stability. The system design resulted in a triangulated system of horizontal A-frames.

Secondary Lateral System

Although the primary lateral system will produces the needed amount of strength and stiffness, the diagonal elements need additional bracing between the 40 foot nodal modules. The goal of the secondary lateral system is to provide stability for gravitational loads and construction tolerances at each level. The secondary lateral system in this case is a braced frame at the service core.



Diagrid

The most striking feature of the Hearst Tower is the diagrid design that forms a triangular curtain wall façade. Starting at the tenth floor, each triangular glass pane is 4 stories high and the intersections of these panes form diagrid nodes. The nodes, set on 40 foot modules, then act to redirect member forces. Essentially, all four faces come together at chamfered "Bird's Mouth" corners and create a very efficient tube structure.

The office tower begins at the tenth floor and utilizes a composite steel and concrete flooring system. Each floor has 40 foot column free spans to allow for maximum open office planning. The office floors are typically 16,000 sf in area with 9.5' ceilings.

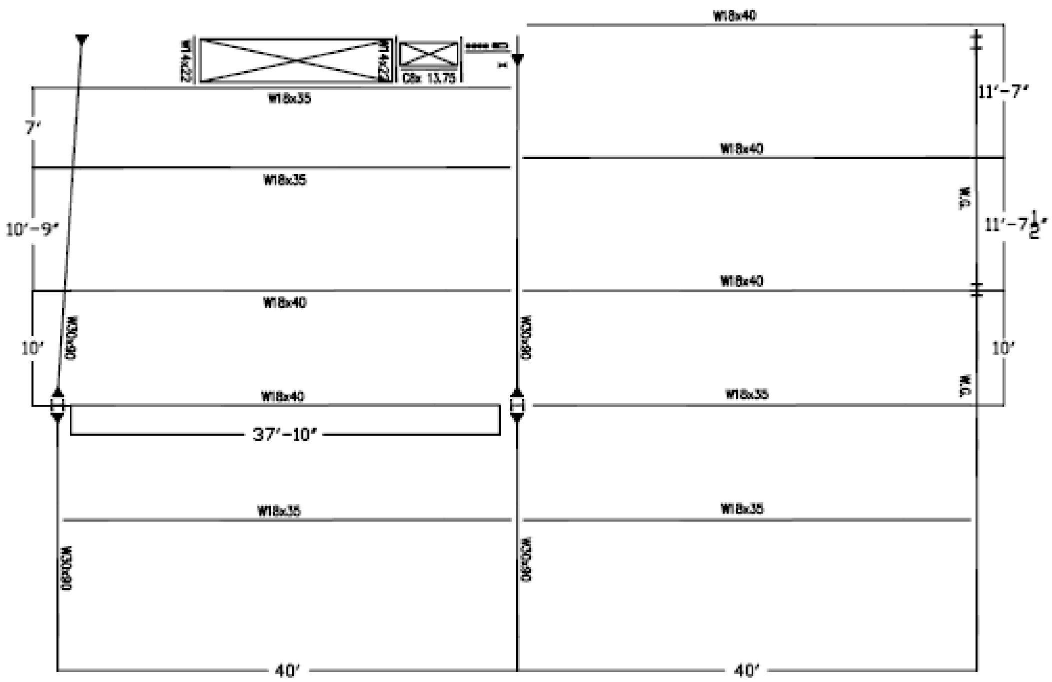
Having no vertical members on the façade allowed for a savings of 2,000 tons of steel. In all, the diagrid uses approximately 10,000 tons of wide flange rolled sections.

The left hand side of the **Figure 1s** below shows an interior bay of the Hearst Tower. Since the proposed mechanical design calls for radiant flooring, checking the fire rating for the new slab and subsequent loading on the members is necessary.

Columns, Mega Columns, and Super Diagonals

The diagrid façade is used to transfer loads at the tenth floor to 12 perimeter megacolumns. These megacolumns have an unbraced length of 85 feet and continue all the way to the foundation. The megacolumn system is designed to respond to the large unbraced height. Eight-ninety foot super diagonals slope inward from the megacolumn nodes on the third floor to the columns lines at floor ten. Additional superdiagonals are provided below floor ten to assist the composite core system with stability. The megacolumns are made out of built-up steel tube sections filled with concrete.

Figure 1s Typical Interior Bay



Procedure for Checking

Using the Vulcraft Steel Deck and Floor Deck Catalog, under the Section titled "Floor-Ceiling Assemblies with Composite Deck," 3VLI deck was chosen to satisfy the 2 hour fire rating required by the 2003 International Fire Code (IFC 2003). Since the current slab is 6" deep, an additional 2" of concrete is needed to protect the tubes.

Once this deck was selected, a check was performed to make sure the 19 gauge deck used currently in the Hearst Tower was sufficient to hold the additional 2" of slab. With the additional slab weight, the total load is 95 PSF. According to the deck catalog, and looking at the 3VLI19 deck with 6" slab (to be conservative, since top 2" is not structural), the total superimposed live load that can be supported for 12' spans is 110 PSF.

After checking the deck, four interior beams were checked. Using the Load Resistance Factor Design Method (LRFD Method), in conjunction with ASCE7-2005, it was determined that all of the beam sizes need to increase to withstand the additional loading. Since this is a structural breadth study, the beams were checked as if they are noncomposite with the slab and deck. A table of the existing beams and the required beams for the new design are shown below in **Table 1s**. Appendix A contains hand calculations for each beam check performed.

Table 1s Beam Sizes

Existing Structure	New Structure
W18X35	W21X48
W18X40	W21X55
W18X40	W21X55
W18X35	W18X40

After the beams were checked and resized as necessary, the girder on the right hand side of Figure 1s was also checked. Hand calculations and STAAD Pro results for this procedure are also included in the Structural Breadth Appendix of this report. The increased load resulted in an increased girder size from W30X90 to W30X132. This was done to keep the same floor to ceiling heights, since the W30X90 was the deepest member in the flooring system.

Cost Considerations

The purpose of this analysis was not necessarily to determine cost savings or increases, but rather to show the impact the changes in one building system (Mechanical in this case) may have on other systems of the building. For the case of the Hearst Tower, using an estimate of $1/lbm, the increased beam sizes only result in an increase of $48/ typical bay. Since this number is small in comparison to the $38 million dollar steel contract, structural cost will not serve as an argument for or against the proposed mechanical system design.

References

American Society of Civil Engineers. ASCE Standard 7-02. 2003.

American Institute of Steel Construction. Manual of Steel Construction, Load and Resistance Factor Design, Third Edition. 2001.

ASHRAE Handbook of Fundamentals. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 2001

Beckman, William & Duffie, John. Solar Engineering of Thermal Processes. 2nd Edition. John Wiley&Sons. 1991.

CostWorks. R.S. Means Inc. 2005

Hughes, Davis S. Electrical Systems in Buildings. Delmar Publishers Inc. 1988.

International Fire Code. International Code Council. 2003

McQuiston, Faye, Parker, Jerald, Spitler, Jeffrey. Heating, Ventilation, and Air Conditioning Analysis and Design. 5th Edition. John Wiley&Sons. 2000.

Vulcraft Steel Roof and Floor Deck Catalog.

www.doas-radiant.psu.edu

www.hearstcorp.com

Appendices

SERVICE CLASSIFICATION NO. 2
ANNUAL POWER SERVICE

Applicable to Use of Service for

Power, Heat, or Power and Heat

Character of Service

Continuous; supplied at an average gauge pressure in excess of 125 pounds.

Rate

Base Rate (in each monthly period)

During monthly billing periods terminating within the months of May to October, inclusive:

For the first	250,000 pounds	$ 6.777	per 1000 pounds
For the next	750,000 pounds	9.047	per 1000 pounds
For excess over	1,000,000 pounds	8.457	per 1000 pounds

During monthly billing periods terminating within the months of November to April, inclusive:

For the first	250,000 pounds	$ 7.664	per 1000 pounds
For the next	1,250,000 pounds	20.169	per 1000 pounds
For the next	3,500,000 pounds	18.717	per 1000 pounds
For the next	20,000,000 pounds	17.973	per 1000 pounds
For excess over	25,000,000 pounds	16.882	per 1000 pounds

Customer Charge

$2,455.35 per month, exclusive of increase in rates and charges. A Customer who requests a temporary discontinuance of service will be required to pay the monthly customer charge for each month that service is discontinued. A Customer whose service is discontinued for non-payment and who applies for service within twelve months at the same location will be regarded as having taken a temporary discontinuance of service and will be required to pay the monthly customer charge for each month that service was discontinued.

Fuel Adjustment

The charges to the Customer set forth herein shall be subject to a fuel adjustment per 1000 pounds of steam supplied hereunder when changes from the base cost of fuel occur (as explained in General Information Section VII).

Increase in Rates and Charges

The rates and charges under this Service Classification including fuel adjustment and customer charge are increased by the applicable Percentage as explained in General Information Section VIII and shown on the related Statement.

(Service Classification No. 2 - Continued on Leaf No. 21-A)

Date of Issue: August 25, 2005
Date Effective: October 1, 2005

Issued by Joan S. Freilich, Executive Vice President and
Chief Financial Officer
4 Irving Place, New York, N.Y. 10003

Issued under authority of order of the Public Service Commission dated September 27, 2004 in Case 03-S-1672.

SERVICE CLASSIFICATION NO. 2 - Continued
ANNUAL POWER SERVICE

Terms of Payment

Net cash on presentation of bill, subject to late payment charge in accordance with provisions of General Rule III- 4(j).

Term

The term of the agreement for service shall be for a period ending no less than one year from the beginning of service under this Service Classification, and shall continue after the end of such period until terminated by the Customer upon thirty days' prior written notice, or by the Company in accordance with law or the provisions of this Rate Schedule.

Special Provisions

A. Service will not be available under this Service Classification to Customers eligible for service under Service Classification No. 4.

B. A Customer may request a temporary discontinuance of service by notifying the Company in writing at least five days in advance of the date on which each temporary discontinuance is to begin specifying the date on which service is to be discontinued and the date on which service is to be resumed. The Company may elect not to physically disconnect service at the service valve. In the event that a Customer takes service during a temporary discontinuance period the Customer will be billed in accordance with this Service Classification.

C. A Customer having transferred from this Service Classification to another Service Classification may not transfer back to this Service Classification except on a date in any year which corresponds with the date when service was first supplied under this Service Classification.

D. Where a Customer installs a new or replacement steam air conditioning system on or after October 1, 2004, base rates in each monthly billing period for service furnished under this Service Classification during monthly billing periods terminating within the months of May to October, inclusive, will be reduced by an amount equal to $2.00 per 1000 pounds of steam used in excess of the first 250,000 pounds of steam used. A Customer applying for the foregoing reduction must notify the Company at least two months before installation of the air conditioning system. Such Customer will be eligible for the rate reduction effective with the first summer monthly billing period that commences subsequent to the date on which the Company receives written notice from the Customer that the installation is complete. Service will be provided under this Special Provision under the following conditions:

(1) in total, no more than 75,000 tons of new or replacement steam air conditioning capacity will be accepted under this Special Provision and Special Provision D of Service Classification No. 3;
(2) the Company will offer service to new applicants under this Special Provision until at least September 30, 2006, after which time the level of the rates and charges and the terms and conditions of service may vary from those originally offered;
(3) the Company will provide service under the terms of this Special Provision for a two-year period to Customers who commence service under this Special Provision on or after October 1, 2004, but before October 1, 2006; and
(4) to facilitate Customers' investment decisions, the Company may, at its sole discretion, negotiate the term of service under this Special Provision with a Customer who agrees to use steam for air conditioning and other purposes during the service term. In lieu of monthly bill reductions set forth under this Special Provision, the Company may, at its discretion, make a one-time payment of the net present value of the bill reduction, provided that the Customer commits to a contract term that is equivalent to the term value of the bill reduction.

(Service Classification No. 2 - Continued on Leaf No. 21-B)

SERVICE CLASSIFICATION NO. 2 - Continued

ANNUAL POWER SERVICE

Special Provisions – Continued

E. Where a Customer does not own a steam air conditioning system and installs a new steam air conditioning system to replace or supplement existing electric air conditioning equipment, base rates in each monthly billing period for service furnished under this Service Classification during monthly billing periods terminating within the months of May to October, inclusive, will be reduced by $2.00 per 1000 pounds of steam used in excess of the first 250,000 pounds of steam used. A Customer applying for the foregoing reduction must notify the Company at least two months before installation of the air conditioning system. Such Customer will be eligible for the rate reduction effective with the first summer monthly billing period that commences subsequent to the date on which the Company receives written notice from the Customer that the installation is complete. Service will be provided under this Special Provision under the following conditions:

(1) service under this Rider will not be available until after a total of 25,000 tons of new or replacement steam air conditioning capacity is accepted under Special Provision D of this Service Classification and Special Provision D of Service Classification No. 3;

(2) in total, no more than 25,000 tons of new steam air conditioning capacity will be accepted under this Special Provision and Special Provision E of Service Classification No. 3;

(3) steam purchased from the Company must be used solely for operating air conditioning equipment and only during the months of May to October, inclusive;

(4) the Company will offer service to new applicants under this Special Provision until at least September 30, 2006, after which time the level of the rates and charges and the terms and conditions of service may vary from those originally offered;

(5) the Company will provide service under the terms of this Special Provision for a two-year period to Customers who commence service under this Special Provision on or after October 1, 2004, but before October 1, 2006; and

(6) to facilitate Customers' investment decisions, the Company may, at its sole discretion, negotiate the length of the term of service under this Special Provision with a Customer who does not own a steam air conditioning system and installs a new steam air conditioning system to replace or supplement existing electric air conditioning equipment. In lieu of monthly bill reductions set forth under this Special Provision, the Company may, at its discretion, make a one-time payment of the net present value of the bill reduction, provided that the Customer commits to a contract term that is equivalent to the term value of the bill reduction.

F. The foregoing rates and charges shall apply to all steam service supplied hereunder on and after the effective date hereof. Where a bill includes periods before the effective date and after the effective date, the rates and charges applicable will be prorated based on the number of days of service rendered before the effective date and on and after the effective date related to the total number of days in the billing period.

G. The following Riders may be applied to this Service Classification: A, B, C, D, and E on Leaf Nos. 11 to 13-B, inclusive.

H. For form of application under this Service Classification, see Leaf Nos. 16-18.

Date of Issue: October 1, 2004 Date Effective: October 1, 2004
Issued by Joan S. Freilich, Executive Vice President and
Chief Financial Officer
4 Irving Place, New York, N.Y. 10003

Issued under authority of order of the Public Service Commission dated September 27, 2004 in Case 03-S-1672.

Zone 1 Cooling Calculations for One Typical Floor
10-28, 34,35

Sensible Load (BTU/hr)	Latent Load (BTU/hr)	Std. 62.2 OA Req'd (CFM)	Req'd OA for Load (CFM)	Std. 62 OA +30% (CFM)
319067	31629	1770	2326	2301

	OA	Point A	Point B	RA
Temp (F)	92	78.4	45	75
HR(gr/lbma)	98	71.6	45	65

	ε_L	ε_S
Enthalpy Wheel Effectiveness	0.8	0.8

Cooling Coil Loads	Sensible (BTU/hr)	Latent (BTU/hr)	Total (Tons)
	83903.5	42072.7	10.5

DOAS Sensible Cooling Capacity (BTU/hr)	75362.4	Total doas cap (tons)	9.8

Parellel Capacity Needed	(BTU/hr)	Tons
	243704.6	20.3

Zone 2	
29th Floor	*same as 30-33 floors

Sensible Load (BTU/hr)	Latent Load (BTU/hr)	Std. 62.2 OA Req'd (CFM)	Req'd OA for Load (CFM)	Std. 62 OA +30% (CFM)
276592	28080	1880	1376.47059	2444

	OA	Point A	Point B	RA
Temp (F)	92	77.89	45	75
HR(gr/lbma)	98	71.6	45	65

	ε_L	ε_S
Enthalpy Wheel Effectiveness	0.8	0.83

Cooling Coil Loads	Sensible (BTU/hr)	Latent (BTU/hr)	Total (Tons)
	86813.8	44207.1	10.9

DOAS Sensible Cooling Capacity (BTU/hr)	79185.6

Parellel Capacity Needed	(BTU/hr)	Tons
	197406.4	16.5

Zone 3
36th Floor

Sensible Load (BTU/hr)	Latent Load (BTU/hr)	Std. 62.2 OA Req'd (CFM)	Req'd OA for Load (CFM)	Std. 62 OA +30% (CFM)
174926	13330	920	980.1470588	1196

	OA	Point A	Point B	RA
Temp (F)	92	77.89	45	75
HR(gr/lbma)	98	71.6	45	65

	ε_L	ε_S
Enthalpy Wheel Effectiveness	0.8	0.83

Cooling Coil Loads	Sensible (BTU/hr)	Latent (BTU/hr)	Total (Tons)
	42483.4	21633.2	5.3

DOAS Sensible Cooling Capacity (BTU/hr)	38750.4

Parellel Capacity Needed	(BTU/hr)	Tons
	136175.6	11.3

Zone 4
37th, 9th, 38th Floor

Sensible Load (BTU/hr)	Latent Load (BTU/hr)	Std. 62.2 OA Req'd (CFM)	Req'd OA for Load (CFM)	Std. 62 OA +30% (CFM)
233257	16655	1333	1225	1733

	OA	Point A	Point B	RA
Temp (F)	92	77.89	45	75
HR(gr/lbma)	98	71.6	45	65

	ε_L	ε_S
Enthalpy Wheel Effectiveness	0.8	0.83

Cooling Coil Loads	Sensible (BTU/hr)	Latent (BTU/hr)	Total (Tons)
	61554.7	31344.7	7.7

DOAS Sensible Cooling Capacity (BTU/hr)	56145.96

Parellel Capacity Needed	(BTU/hr)	Tons
	177111.0	14.8

Zone 5
39th,11th,40th floor

Sensible Load (BTU/hr)	Latent Load (BTU/hr)	Std. 62.2 OA Req'd (CFM)	Req'd OA for Load (CFM)	Std. 62 OA +30% (CFM)
261014	12215	1423	898	1850

	OA	Point A	Point B	RA
Temp (F)	92	77.89	45	75
HR(gr/lbma)	98	71.6	45	65

	ε_L	ε_S
Enthalpy Wheel Effectiveness	0.8	0.83

Cooling Coil Loads	Sensible (BTU/hr)	Latent (BTU/hr)	Total (Tons)
	65710.7	33461.0	8.3

DOAS Sensible Cooling Capacity (BTU/hr)	59936.76

Parellel Capacity Needed	(BTU/hr)	Tons
	201077.2	16.8

Zone 6
41st floor

Sensible Load (BTU/hr)	Latent Load (BTU/hr)	Std. 62.2 OA Req'd (CFM)	Req'd OA for Load (CFM)	Std. 62 OA +30% (CFM)
388124	52540	4270	3863	5551

	OA	Point A	Point B	RA
Temp (F)	92	77.89	45	75
HR(gr/lbma)	98	71.6	45	65

	ε_L	ε_S
Enthalpy Wheel Effectiveness	0.8	0.83

Cooling Coil Loads	Sensible (BTU/hr)	Latent (BTU/hr)	Total (Tons)
	197178.2	100406.5	24.8

DOAS Sensible Cooling Capacity (BTU/hr)	179852.4

Parellel Capacity Needed	(BTU/hr)	Tons
	208271.6	17.4

Sample Solar Gain Intensity Calcualtion

June 21st

Solar Time	w	cos(thetaZ)	theta z	α_s	γ_s	γ_{s-w}	tau b	tau d	theta	profile angle	R_b	I_{b1} (J/m^2)	$I_{b1}R_b$	G_{b1} (J/s*m^2)	G_{b2}	Gon
5-6 AM	-97.5	0.16	80.56	9.44	112.77	67.77	0.20	0.21	68.09	23.73	2.27	154082.71	350470.67	42.80	97.35	1322.62
6-7 AM	-82.5	0.35	69.66	20.34	104.06	59.06	0.37	0.16	61.18	35.79	1.39	613269.13	850620.63	170.35	236.28	1322.62
7-8 AM	-67.5	0.52	58.35	31.65	95.34	50.34	0.48	0.13	57.09	44.00	1.04	1205669.93	1248420.12	334.91	346.78	1322.62
8-9 AM	-52.5	0.68	46.87	43.13	85.75	40.75	0.55	0.11	56.43	51.03	0.81	1788662.63	1446744.03	496.85	401.87	1322.62
9-10 AM	-37.5	0.81	35.58	54.42	73.73	28.73	0.59	0.10	59.33	57.90	0.63	2283834.95	1432515.85	634.40	397.92	1322.62
10-11 AM	-22.5	0.91	25.17	64.83	55.65	10.65	0.61	0.09	65.30	65.21	0.46	2640657.93	1219363.03	733.52	338.71	1322.62
11-12 AM	-7.5	0.95	17.72	72.28	23.17	-21.83	0.62	0.09	73.59	73.48	0.30	2827009.61	838492.32	785.28	232.91	1322.62
12-1 PM	7.5	0.95	17.72	72.28	23.17	-21.83	0.62	0.09	73.59	73.48	0.30	2827009.61	838492.32	785.28	232.91	1322.62
1-2 PM	22.5	0.91	25.17	64.83	55.65	10.65	0.61	0.09	65.30	65.21	0.46	2640657.93	1219363.03	733.52	338.71	1322.62
2-3 PM	37.5	0.81	35.58	54.42	73.73	28.73	0.59	0.10	59.33	57.90	0.63	2283834.95	1432515.85	634.40	397.92	1322.62
3-4 PM	52.5	0.68	46.87	43.13	85.75	40.75	0.55	0.11	56.43	51.03	0.81	1788662.63	1446744.03	496.85	401.87	1322.62
4-5 PM	67.5	0.52	58.35	31.65	95.34	50.34	0.48	0.13	57.09	44.00	1.04	1205669.93	1248420.12	334.91	346.78	1322.62
5-6 PM	82.5	0.35	69.66	20.34	104.06	59.06	0.37	0.16	61.18	35.79	1.39	613269.13	850620.63	170.35	236.28	1322.62
6-7 PM	97.5	0.16	80.56	9.44	112.77	67.77	0.20	0.21	68.09	23.73	2.27	154082.71	350470.67	42.80	97.35	1322.62

Declination	A (km)	a0*	a1*	k*	a0	a1	Beta	k	Latitude	n	γ
23.45	0.043	0.13	0.75	0.38	0.13	0.75	90.00	0.39	40	172	45

* Transmittance of 29%

90% actually absorbed by floor

Solar Time	G_{actual}	0.9*Gactual
5-6 AM	7.72	6.9
6-7 AM	18.73	16.9
7-8 AM	27.48	24.7
8-9 AM	31.85	28.7
9-10 AM	31.54	28.4
10-11 AM	26.84	24.2
11-12 AM	18.46	16.6
12-1 PM	18.46	16.6
1-2 PM	26.84	24.2
2-3 PM	31.54	28.4
3-4 PM	31.85	28.7
4-5 PM	27.48	24.7
5-6 PM	18.73	16.9
6-7 PM	7.72	6.9

RADIANT FLOOR MODEL

Input:

Airside parameters:

Indoor air temperature = 75 [F]

Overall heat transfer coefficient above floor = 2.008 [Btu/(h ft^2 F)]

Solar gain intensity = 6 [Btu/(h ft^2)]

Covers parameters:

Total thickness of covers ΣX_i = 0.79 [in]

Sum of products of thickness and conductivity of covers $\Sigma(X_i*K_i)$ = 0.024 [Btu/ (h F)]

Thermal resistance of coverings = 0.18 [F h ft^2 / Btu]

Slab & tube parameters:

Thickness of slab L = 1.5 [in]

Thermal conductivity of slab K = 0.87 [Btu/(h ft F)]

Tube center-to-center spacing W = 6 [in]

Tube interior diameter = 0.625 [in]

Tube exterior diameter = 0.75 [in]

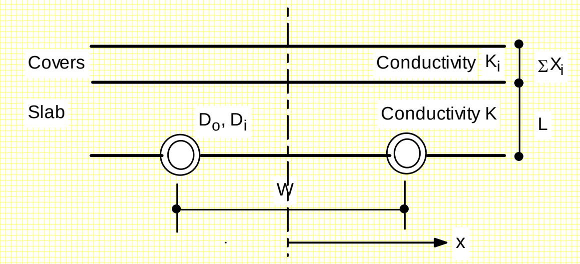
Tube thermal conductivity = 0.22 [Btu/(h ft F)]

Water loop parameters:

Water loop length = 300 [ft]

Inlet water temperature = 61 [F]

Water flowrate = 1 [GPM]



x/(W/2) = 0

File menu:

Calculate Save Print

Output:

Radiant floor capacity = 17.05 [Btu/(h ft^2)]

Outlet water temperature = 66.11 [F]

Mean floor temperature T_{pm}= 69.5 [F]

Floor surface temperature at x = 70.1 [F]

```
"First Cost of Pumps"

"Knowns"

    "Temperatures"

        T_fluid_in=61 [F]
        T_fluid_out=66 [F]
        T_mean=(T_fluid_in+T_fluid_out)/2

    "Pipe Dimensions"

        d_inner=0.625/12 [ft]
        d_outer=0.75/12  [ft]
        r_inner=d_inner/2 [ft]
        r_outer=d_outer/2 [ft]

        Length_circuit=300 [ft]
        Number_circuits=10
        Number_zones=7

    "Flow Characteristics"

        v_dot=2 [gpm]
        rho=density(water,t=T_mean,p=14.7)
        mu=viscosity(water,t=T_mean,p=14.7)*convert(lbm/ft-hr,lbm/ft-s)

"Velocity of Water Through Pipe"

    Flow_velocity=(convert(gpm,ft^3/s)*v_dot)/(pi*(r_inner^2))

"Reynold's Number"

    Re_D=(rho*Flow_velocity*d_inner)/mu

"Pressure Drop"

        DELTAP_total=(0.0160185)*(f*Length_circuit*(Flow_velocity^2)*rho)/(2*d_inner*g_c)*Number_circuits*Number_zones

        g_c=32.2

    f=1/((1.82*(LOG10(Re_D)))-1.64)^2

"Pump Characteristics"

    eta_pump=0.8
    eta_motor=0.75
    HP_pump=v_dot*Number_circuits*Number_zones*DELTAP_total/(3960*eta_pump*eta_motor)

"First Cost"

    C_pump= 325+(1550*HP_pump)
```

Appendix B

Equipment Specifications
Equipment Cut Sheets

v3.5F

Reference Specifications

Project : Hearst Tower — Ref. US -06

Model name:16NK71 — 3 units

Cooling capacity					600	USRT
Chilled water	Inlet temperature				55.0	^oF
	Outlet temperature				41.0	^oF
	Flow rate				1,029	USgpm
	Pressure drop				53.3	$ft.H_2O$
	Pass number (EVAP)				5	
	Connection diameter(ANSI)				8	inch
	Max. working pressure				150	psig
	Fouling factor				0.0001	ft^2h^oF/Btu
	Brine				0.0	%
Cooling water	Inlet temperature				85.0	^oF
	Outlet temperature				100.0	^oF
	Flow rate				1,731	USgpm
	Press. drop				21.2	$ft.H_2O$
	Pass number (ABS/COND)				3 + 2	
	Connection diameter(ANSI)				16	inch
	Max. working pressure				150	psig
	Fouling factor				0.00025	ft^2h^oF/Btu
	Brine				0.0	%
Steam line	Condition				Saturated	
	Supply pressure				114	psig
	Consumption				5,420	lb/h
	Steam inlet connection(ANSI)				6	inch
	Drain outlet connection(ANSI)				2-1/2	inch
	Max. working pressure				114	psig
Electric	Power source		3 Phase	460 V	60	Hz
	Electric consumption			20.0813 kW	23.6	kVA
	Total electric current				30.0	A
	Motor output	No.1 ABS pump		9.0 kW	20.0	A
		No.2 ABS pump		1.5 kW	4.6	A
		Refrigerant pump		1.3 kW	3.1	A
		Purge pump		0.75 kW	1.45	A
Overall dimensio	Length (L)				254	inch
	Width (W)				130	inch
	Height (H)				136	inch
	Space of tube removal				237	inch
Weight	Operation weight				83,800	lbs
	Max. shipping weight				61,300	lbs
	Total shipping weight				72,800	lbs
Shipping method					1	section

Note) 1) Electric consumption kW = 0.85 x KVA

2)Rated in accordance with ARI-560.

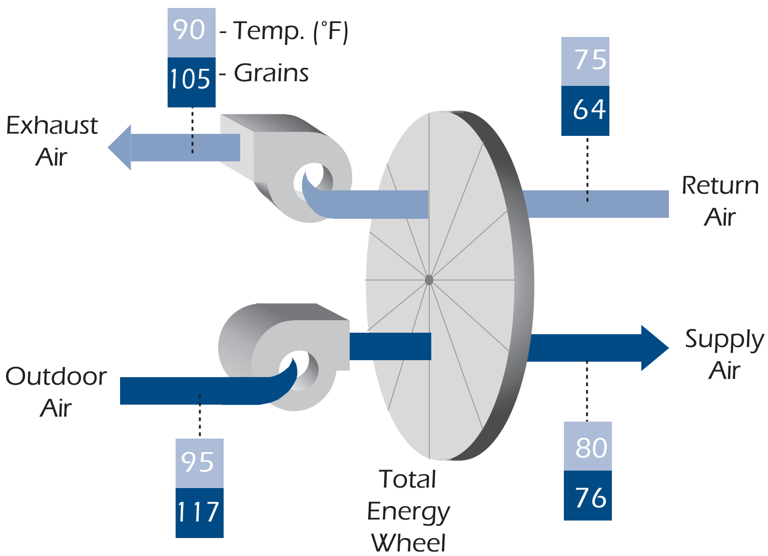
Available Equipment Configurations

EP

In addition to the SEMCO EXCLU-SIEVE energy recovery wheel, this dual-wall system contains backward curved supply and exhaust fans, outdoor air and return air filtration and an optional, full-electrical package with a single-point electrical connection. All EP family products are designed for either indoor or outdoor mounting.

EP Product

• Suitable for new construction and can be retrofitted to most existing facilities.
• Precools and dehumidifies outdoor air during the cooling season.
• Preheats and pre-humidifies the outdoor air during the heating season.
• Supplies preconditioned outdoor air to conventional HVAC systems, allowing them to effectively increase outdoor air percentages.
• Preconditioned outdoor air can be introduced to the return air plenum serving a central HVAC system.
• It can also be supplied directly to the conditioned space since the system's recovery efficiency ranges between 74 and 85% (in balanced flow operation).

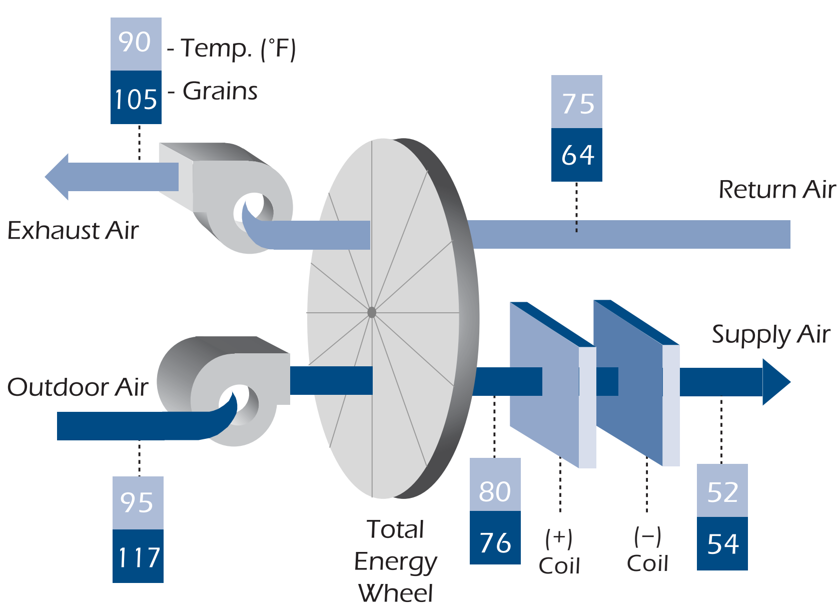


EPH, EPC, EPHC

These products build on the EP product mentioned above. However, unlike the EP, they integrate full heating and cooling options. The cooling options include either chilled water or DX cooling coils, with options regarding the number of fins per inch and the number of row options. The heating options include either hot water, steam or electric coils.

EPH, EPC, EPHC Products

These products are applied to installations where there is a need for 100 percent outdoor air. The SEMCO system is the primary source for temperature and/or humidity control. This includes hospitals, manufacturing areas, laboratories and casinos. These products are also used to precondition buildings where the outdoor air goes directly to the space, but requires additional post heating or cooling to supplement what is being provided by the energy recovery wheel.



EP Equipment Summary

Model Size	3	5	9	13	18	24	28	35	43
Width	86.25	86.25	98.25	98.25	122.25	122.25	146.25	146.25	182.25
Height	50.25	62.25	74.25	86.25	98.25	110.25	122.25	134.25	146.25
Supply Air CFM Range [1] Return Air CFM Range [1]	2,000-3,000	3,000-4,500	4,500-8,000	6,000-10,000	8,000-15,000	11,000-18,000	15,000-23,000	18,000-27,000	26,000-40,000
Fan size (standard)	APF 141	APF 181	APF 221	APF 251	APF 281	APF 321	APF 351	APF 391	APF 441
Fan size (option X)	-	APF 201	APF 251	APF 281	APF 321	APF 351	APF 391	APF 441	APF 491
Fan size (option XX)	-	APF 221	APF 281	APF 321	APF 351	APF 391	APF 441	APF 491	-
Purge volume (single wheel) [3]	513	695	906	1168	1440	1735	1961	2297	2662
Heat/cool coil total fin height	33 in	45 in	54 in	66 in	78 in	90 in	99 in	111 in	123 in
Heat/cool coil total fin length	30 in	30 in	42 in	42 in	54 in	54 in	66 in	66 in	90 in
Number of stacked coils (height)	(1) 33 in	(1) 45 in	(2) 27 in	(2) 33 in	(2) 39 n	(2) 45 in	(3) 33 in	(2) 36 in	(2) 42 in
	-	-	-	-	-	-	-	(1) 39 in	(1) 39 in
Supply filter	(1) 24x24	(2) 24x24	(6) 20x24	(6) 24x24	(3) 20x24	(12) 20x24	(12) 24x24	(15) 24x24	(20) 24x24
	(2) 12x24	(2) 12x24	-	-	(9) 20x20	-	(3) 12x24	-	(4) 12x24
Return filter	(1) 24x24	(2) 24x24	(2) 24x24	(3) 24x24	(6) 24x24	(8) 24x24	(6) 20x24	(15) 20x24	(15) 24x24
	(2) 12x24	(2) 12x24	(3) 12x24	(3) 12x24	(2) 12x24	-	(9) 20x20	-	(3) 12x24
Door size (inches)	13x31	13x43	13x56	19x66	19x66	19x66	19x66	19x66	19x66

Notes:
1. Maximum airflow limitations vary. Consult SEMCO before laying out unit with velocities greater than 525 fpm on 2” filters, 525 fpm on cooling coils, and 1100 fpm on wheels.
2. For optional wide RA side, RA side components will be the same as the SA side components.
3. Single wheel purge volume based on 4” P_{OA}-P_{RA}.

Maximum 50 hp Motor

SIZE 43X, 35XX

	STATIC PRESSURE IN INCHES OF WATER																			
	1" SP		2" SP		3" SP		4" SP		5" SP		6" SP		7" SP		8" SP		10" SP		12" SP	
CFM	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
12000																				
14300	371	3.17																		
16600	389	3.65																		
18900	411	4.22	516	8.46																
21200	435	4.85	532	9.38	624	14.52														
23500	458	5.50	551	10.38	634	15.74														
25800	483	6.25	572	11.45	650	17.11	725	23.26												
28100	508	7.04	595	12.64	669	18.62	738	24.95	807	31.80										
30400	535	7.96	619	13.93	689	20.15	756	26.92	819	33.94	883	41.43								
32700	562	8.93	643	15.29	712	21.92	774	28.82	835	36.25	893	43.85	954	52.15						
35000	591	10.07	667	16.70	735	23.72	795	30.98	853	38.66	908	46.54	963	54.88	1020	63.77				
37300	619	11.24	692	18.23	759	25.67	817	33.21	872	41.14	926	49.48	978	58.08	1029	66.91				
39600	648	12.56	717	19.82	782	27.59	841	35.71	893	43.81	945	52.49	995	61.38	1043	70.40	1141	89.75		
41900	678	14.04	744	21.63	806	29.68	864	38.15	916	46.76	965	55.59	1013	64.72	1060	74.19	1152	93.99	1247	115.09
44200	707	15.57	771	23.52	831	31.95	888	40.81	939	49.73	987	58.95	1033	68.39	1078	78.03	1166	98.34	1253	119.68
46500	737	17.29	799	25.59	856	34.28	911	43.41	963	52.93	1009	62.31	1054	72.14	1097	82.02	1182	102.84	1265	124.79
48800	767	19.14	827	27.76	882	36.80	936	46.36	986	56.07	1033	66.08	1076	76.05	1118	86.34	1201	107.93	1280	130.21
51100	797	21.11	855	30.04	908	39.39	960	49.23	1010	59.46	1056	69.74	1099	80.19	1140	90.82	1219	112.67	1296	135.57
53400	828	23.31	884	32.55	935	42.21	985	52.33	1034	62.94	1080	73.70	1122	84.39	1163	95.53	1239	117.86	1315	141.68
55700	858	25.58	913	35.20	963	45.25	1011	55.67	1058	66.49	1103	77.56	1146	88.91	1186	100.30	1261	123.48	1334	147.70
58000	889	28.08	942	37.99	991	48.42	1037	59.11	1083	70.30	1127	81.71	1169	93.32	1209	105.14	1283	129.07	1353	153.62
60300	920	30.75	972	41.06	1019	51.72	1064	62.80	1108	74.21	1151	85.95	1193	98.07	1233	110.36	1306	134.97		

Class I = Max. 823 RPM Class II = Max. 1070 RPM Class III = Max. 1358 RPM *APF491*

Legend:

Class I = First white section
Class II = Blue shaded section
Class III = White section after blue section
Underlined figures indicate Maximum Static Efficiency

Component Pressure Drop Tables

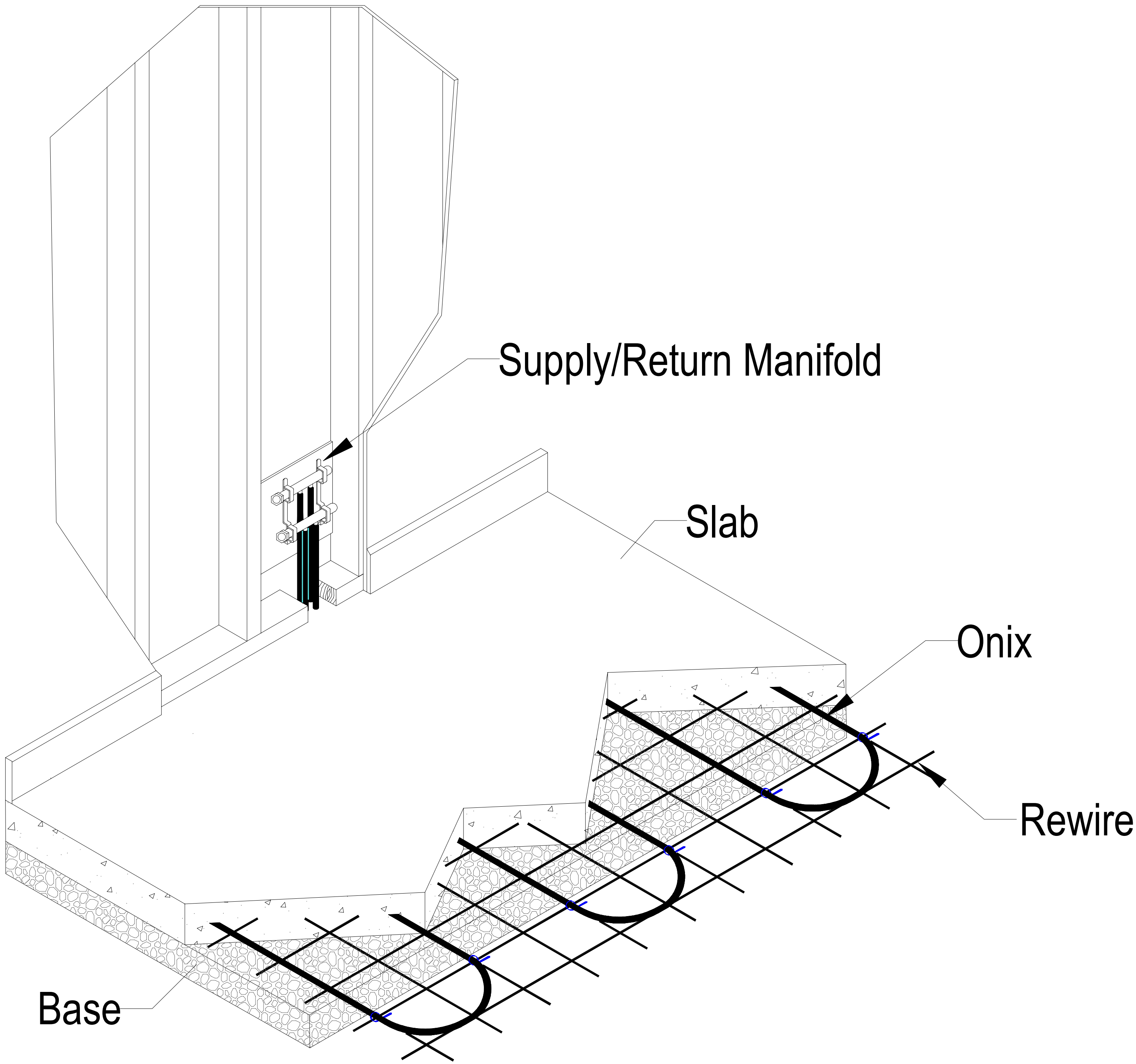
Table 4: Single Wheel Unit Pressure Drops

Size	EP-3			EP-5			EP-9			EP-13			EP-18	
CFM	2000	2500	3000	3000	4000	4500	4500	6000	8000	6000	8000	10000	8000	10000
Enth. wheel purge	513	543	513	695	695	695	906	906	906	1168	1168	1168	1440	1440
Fan cfm	2513	3013	3513	3695	4695	5195	5406	6906	8906	7168	9168	11168	9440	11440
OA opening (w/hood)	0.02	0.04	0.05	0.02	0.03	0.03	0.01	0.02	0.03	0.01	0.02	0.30	0.01	0.01
EA opening (w/hood)	0.04	0.05	0.07	0.08	0.12	0.15	0.07	0.12	0.20	0.08	0.12	0.18	0.07	0.11
RA or EA opening	0.10	0.14	0.19	0.09	0.15	0.19	0.09	0.15	0.24	0.09	0.15	0.22	0.09	0.13
SA or OA opening	0.06	0.10	0.14	0.06	0.11	0.14	0.05	0.08	0.14	0.05	0.08	0.13	0.04	0.06
Damper	0.07	0.10	0.14	0.05	0.08	0.09	0.05	0.08	0.13	0.06	0.09	0.14	0.04	0.07
OA filter	0.26	0.38	0.51	0.25	0.41	0.50	0.19	0.32	0.53	0.24	0.39	0.58	0.19	0.28
RA filter	0.17	0.26	0.37	0.17	0.30	0.37	0.27	0.49	0.87	0.30	0.52	0.82	0.22	0.34
Enth. wheel	0.69	0.92	1.18	0.52	0.74	0.86	0.48	0.67	0.97	0.42	0.57	0.75	0.41	0.51
Cooling coil	0.26	0.40	0.58	0.31	0.56	0.70	0.25	0.44	0.79	0.30	0.53	0.82	0.23	0.36
Heating coil	0.04	0.07	0.09	0.05	0.09	0.11	0.04	0.07	0.13	0.05	0.09	0.13	0.04	0.06
Casing losses	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30
Int. static pressure														
Ext. static pressure														
Total static pressure														

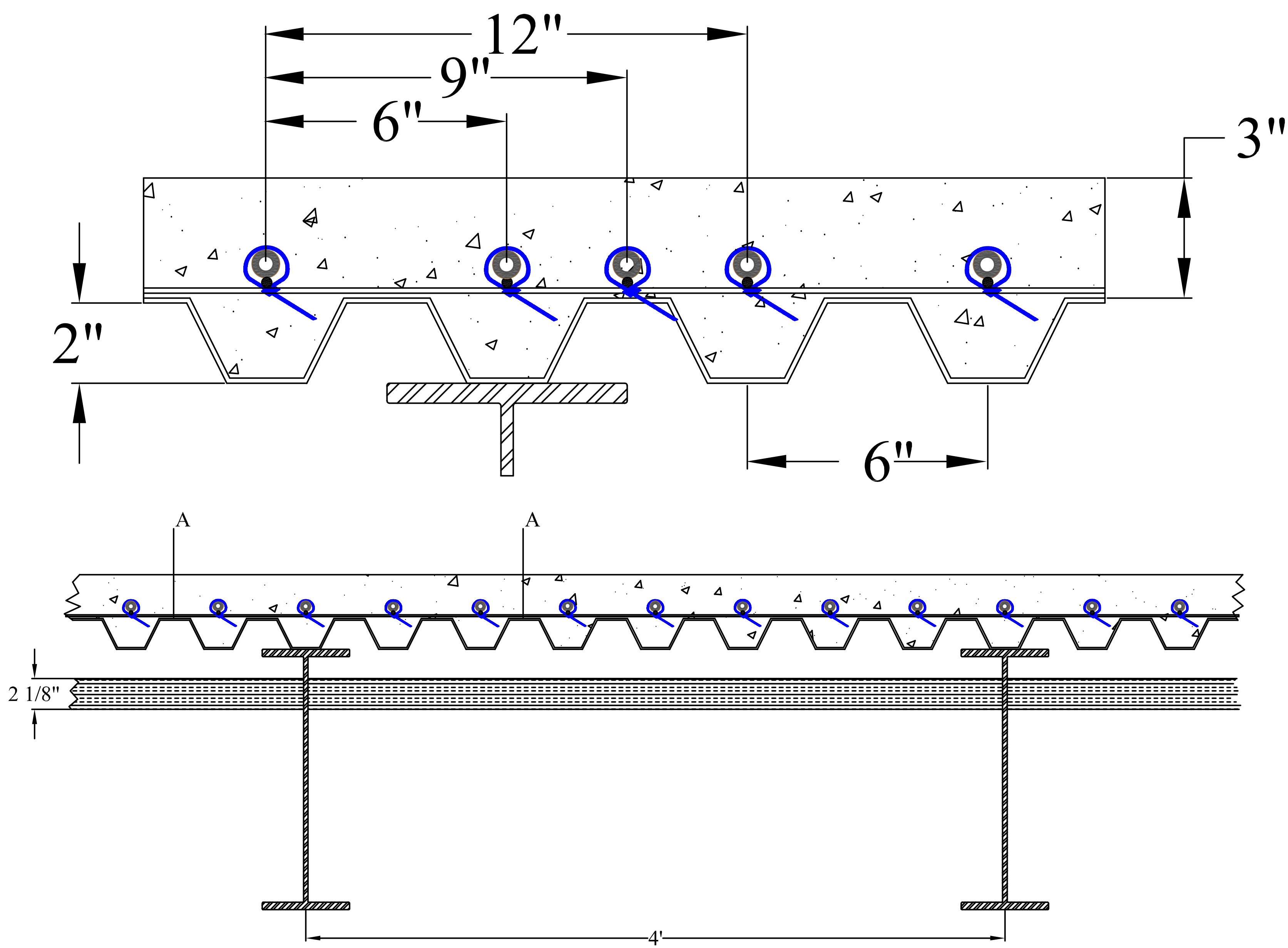
Size	EP-18	EP-24			EP-28			EP-35			EP-43		
CFM	15000	11000	14000	18000	15000	18500	23000	18000	22500	27000	26000	30000	40000
Enth. wheel purge	1440	1735	1735	1735	1961	1961	1961	2297	2297	2297	2662	2662	2662
Fan cfm	16440	12735	15735	19735	16961	20461	24961	20297	24797	29297	28662	32662	42662
OA opening (w/hood)	0.03	0.20	0.02	0.04	0.01	0.02	0.03	0.02	0.02	0.03	0.02	0.02	0.04
EA opening (w/hood)	0.22	0.09	0.14	0.21	0.10	0.15	0.22	0.11	0.16	0.23	0.11	0.14	0.23
RA or EA opening	0.26	0.16	0.24	0.38	0.11	0.17	0.25	0.11	0.17	0.24	0.12	0.15	0.26
SA or OA opening	0.14	0.05	0.08	0.13	0.06	0.09	0.15	0.06	0.10	0.14	0.06	0.08	0.14
Damper	0.14	0.06	0.09	0.13	0.07	0.10	0.14	0.06	0.10	0.13	0.09	0.11	0.19
OA filter	0.59	0.27	0.41	0.65	0.26	0.38	0.57	0.30	0.45	0.63	0.28	0.37	0.62
RA filter	0.76	0.31	0.5	0.84	0.30	0.45	0.69	0.34	0.54	0.77	0.41	0.55	0.98
Enth. wheel	0.83	0.42	0.55	0.74	0.49	0.62	0.81	0.47	0.60	0.75	0.56	0.67	0.98
Cooling coil	0.80	0.32	0.53	0.87	0.33	0.51	0.78	0.38	0.60	0.86	0.35	0.47	0.83
Heating coil	0.13	0.05	0.09	0.14	0.05	0.08	0.13	0.06	0.10	0.14	0.06	0.08	0.13
Casing losses	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30
Int. static pressure													
Ext. static pressure													
Total static pressure													

Notes:

1. Filter pressure drops based on 2 inches thick, 30% efficient Class II filters.
2. Cooling coil pressure drops based on 6 row, 10 fins per inch single-circuited coil.
3. Heating coil pressure drops based on 1 row, 6 fins per inch.
4. Purge volumes based on 4 inches P_{OA}-P_{RA} for wheel.
5. Casing losses include fan inlet losses.


Supply/Return Manifold
Slab
Onix
Rewire
Base

SLAB OVER STEEL DECK



Watts Radiant
Radiant Floors & Snowmelting
Ph: 417/864.6108
Fax: 417/864.8161

GENERAL NOTES:

Cross sections are intended to illustrate typical radiant installations. Floor construction details shown are to be used for illustrative purposes only, not for actual construction. Structural details will need to be determined by the project engineer, architect or other.

SPECIFIER INFORMATION:

ONIX CROSS SECTIONAL DETAILS:

General Legend

- Concrete (Thin Slab)/ Mud
- Joist/Sleepers
- Plywood
- Sand Fill
- Board/ Batt Insulation
- Gravel Bed/ Earth Fill

FILE NO:	43.547
DATE:	July 15, 2003
SUPERSEDES:	5043.9149
DATE:	Apr. 3, 1990
SUPERSEDES:	5043.9069
DATE:	Jan. 9, 1990

ARMSTRONG®

Series 4300 Vertical In-Line 8 x 8 x 13 SUBMITTAL

JOB: ____________________ REPRESENTATIVE: ____________________

ORDER NO: __________ DATE: __________

ENGINEER: __________ SUBMITTED BY: __________ DATE: __________

CONTRACTOR: __________ APPROVED BY: __________ DATE: __________

PUMP DESIGN DATA			
NO. OF PUMPS:			
TAG:			
CAPACITY:	USgpm (L/s)	HEAD:	ft. (m)
LIQUID:		VISCOSITY:	
TEMPERATURE:		SPECIFIC GRAVITY:	

MOTOR DESIGN DATA			
HP:			RPM:
FRAME SIZE:		ENCLOSURE:	
VOLTS:		HERTZ: 60 Hz	PHASE:
EFFICIENCY:	☐ EPAct / NRcan	☐ PREMIUM	LEVEL: %
FLANGE SIZE:	SUCTION: 8" (200 mm)		DISCHARGE: 8" (200 mm)

MATERIALS OF CONSTRUCTION					
ANSI FLANGE RATING	☐ 125 lb. CAST IRON			☐ 250 lb. DUCTILE IRON	
CONSTRUCTION	☐ BF	☐ AI	☐ AB	☐ DBF	☐ DI
CASING	CAST IRON	CAST IRON	BRONZE	DUCTILE IRON	DUCTILE IRON
IMPELLER	BRONZE	CAST IRON	BRONZE	BRONZE	CAST IRON
FLUSH LINE	COPPER	STEEL	COPPER	COPPER	STEEL
PUMP SHAFT	416 STAINLESS STEEL				
COUPLING	HIGH TENSILE ALUMINUM				
GASKET	CONFINED NON-ASBESTOS				

MECHANICAL SEAL DESIGN DATA

See File No. 43.50 for standard mechanical seal details as indicated below

Armstrong Seal Reference Number

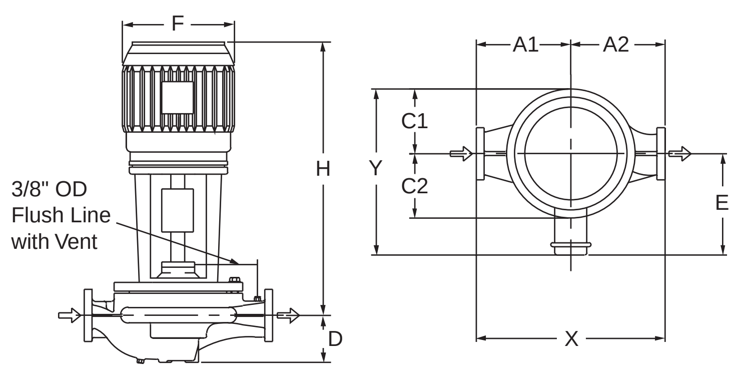
☐ C1 (a) Other____________

SEAL DATA OPTIONS

OPTIONAL EQUIPMENT

SEE FILE NO. 43.50 FOR ENVIRONMENTAL ACCESSORIES

MAXIMUM PUMP OPERATING CONDITIONS	
125 lb. CAST IRON	250 lb. DUCTILE IRON
175 psig at 140°F (12 bars at 60°C) 100 psig at 300°F (7 bars at 149°C)	375 psig at 100°F (26 bars at 38°C) 275 psig at 500°F (19 bars at 260°C)
SEE FILE NO. 43.50 FOR MAXIMUM SEAL OPERATING CONDITIONS	



- COUPLING GUARD (NOT SHOWN) IS SUPPLIED.
- TOLERANCE OF ±0.125" (±3 mm) SHOULD BE USED.
- FOR EXACT INSTALLATION DATA PLEASE WRITE FACTORY FOR CERTIFIED DIMENSIONS.
- SEE PERFORMANCE CURVES ON PAGE 2.
- PUMP EQUIPPED WITH CASING DRAIN PLUG AND 1/4" NPT SUCTION AND DISCHARGE GAUGE PORTS

PUMP DIMENSIONS inches (mm)

A1	A2	C1	C2	D	X
23.00 (584)	19.00 (483)	12.00 (305)	9.75 (248)	9.94 (252)	42.00 (1067)

MOTOR HORSEPOWER							MAX. DIMENSIONS inches (mm)				ASSEMBLY WEIGHT*	
ODP			TEFC			Frame	ODP / TEFC				lbs. (kg)	
3600	1800	1200	3600	1800	1200		E	F	H	Y	ODP	TEFC
---	---	7.5	---	---	7.5	254TC	9.94 (252)	13.38 (340)	37.00 (940)	21.94 (557)	945 (429.0)	990 (449.5)
---	---	10	---	---	10	256TC	9.94 (252)	13.38 (340)	37.00 (940)	21.94 (557)	950 (431.3)	1020 (463.1)
---	---	15	---	---	15	284TC	12.00 (305)	15.31 (389)	43.81 (1113)	24.00 (610)	1025 (465.4)	1030 (467.6)
---	30	20	---	30	20	286TC	12.00 (305)	15.31 (389)	43.81 (1113)	24.00 (610)	1025 (465.4)	1080 (490.3)
---	40	---	---	40	---	324TC	13.91 (353)	17.00 (432)	45.38 (1153)	25.91 (658)	1175 (533.5)	1210 (549.3)
---	50	---	---	50	---	326TC	13.91 (353)	17.00 (432)	45.38 (1153)	25.91 (658)	1175 (533.5)	1350 (612.9)
---	60	---	---	60	---	364TC	16.44 (418)	19.03 (483)	43.56 (1106)	28.44 (722)	1340 (608.4)	1350 (612.9)
---	75	---	---	75	---	365TC	16.44 (418)	19.03 (483)	43.56 (1106)	28.44 (722)	1340 (608.4)	1720 (780.9)

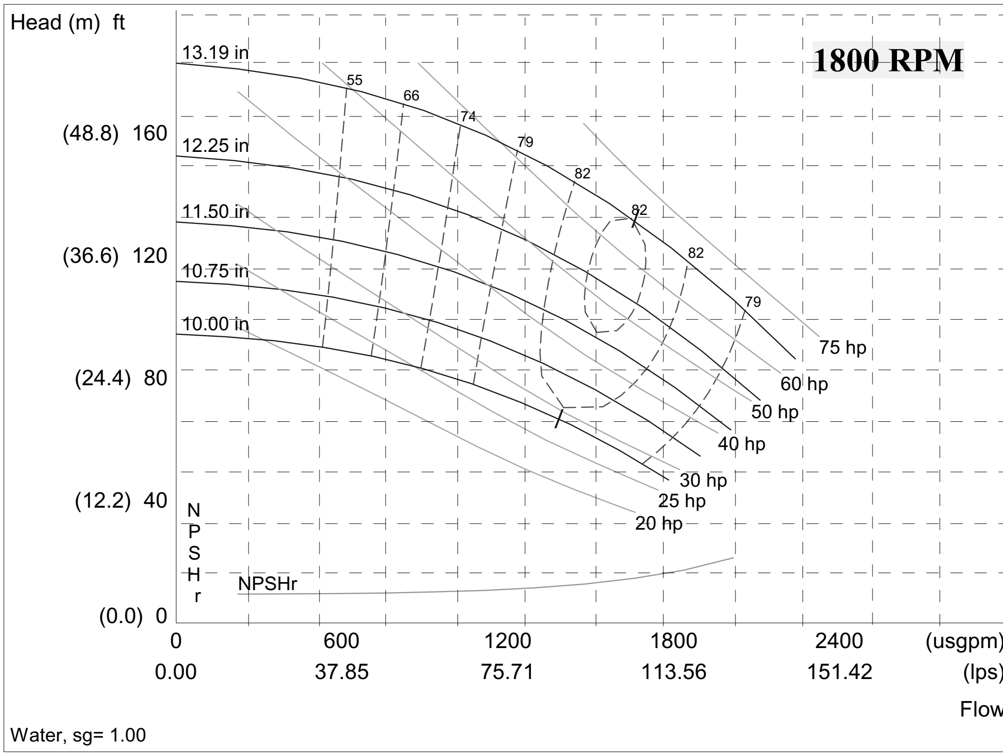
*Assembly weight combines pump and motor.

ARMSTRONG®

PERFORMANCE CURVES

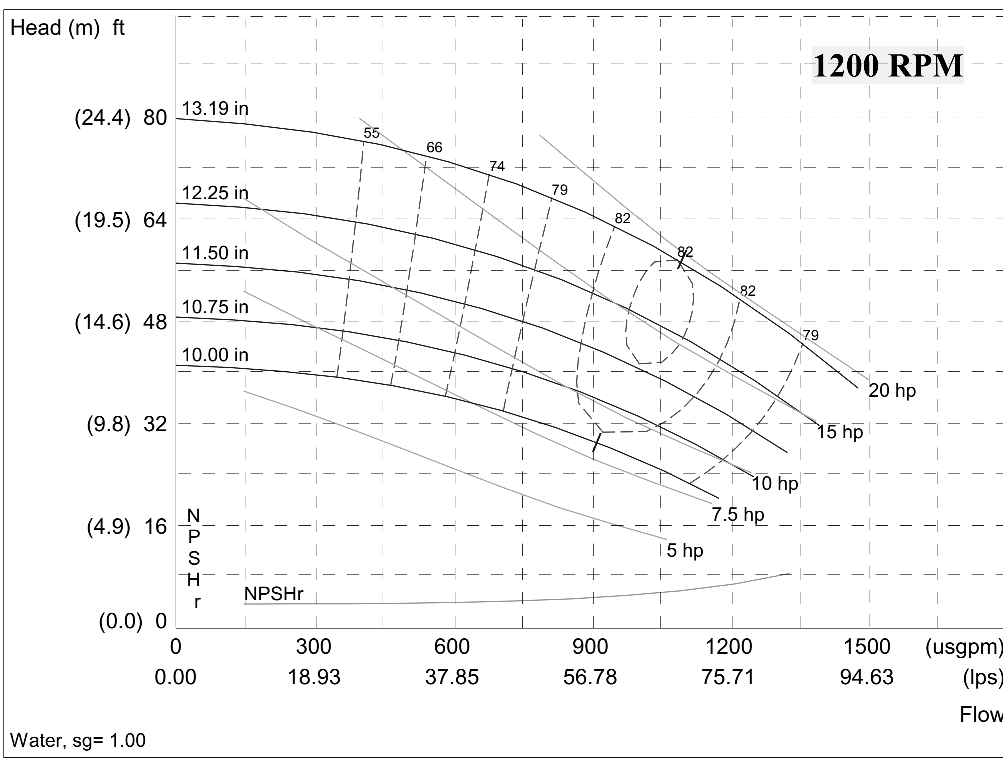
Performance Guaranteed Only At Operating Point Indicated

CURVE SHOWN FOR CLEAR, COLD WATER - SP. GR. 1.0



CURVE NO.
PT114-0-0-1800
SERIES
4300 4380
SIZE
8x8x13
RPM
1800
BHP based on shown Fluid's sp. gr.

AVAILABILITY	
	4300 All Ratings
	4380 All Ratings



CURVE NO.
PT114-0-0-1200
SERIES
4300 4380
SIZE
8x8x13
RPM
1200
BHP based on shown Fluid's sp. gr.

AVAILABILITY	
	4300 All Ratings
	4380 All Ratings

Armstrong Pumps Inc.
93 East Avenue
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Tel: (716) 693-8813
Fax: (716) 693-8970

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Canada, M1L 2P3
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Armstrong Pumps Limited
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Colchester, Essex
United Kingdom, CO3 0LP
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Fax: +44 (0) 1206 760532

Armstrong Darling
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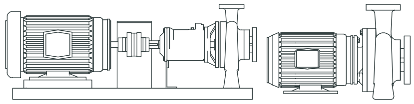
Montreal, Quebec
Canada, H1J 2X9
Tel: (514) 352-2424
Fax: (514) 352-2425

www.armstrongpumps.com

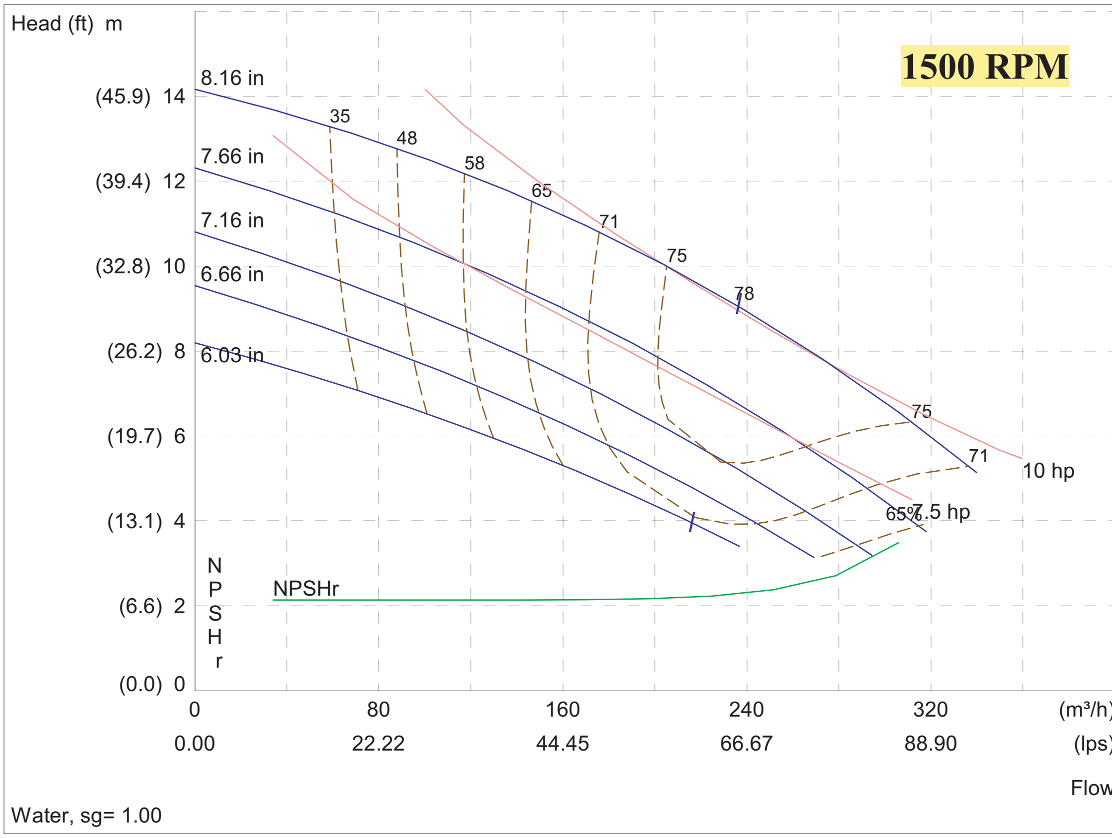
ARMSTRONG®



PERFORMANCE CURVES

Performance Guaranteed Only At Operating Point Indicated

File No:	
Date:	March 15,2002
Supersedes:	NEW
Date:	NEW



CURVE NO.	
PT469-0-0-1500	
SERIES	
4300 **4380**	
SIZE	
8x8x8	
RPM	
1500	
BHP based on shown Fluid's sp. gr.	
AVAILABILITY	4300 All Ratings
	4380 All Ratings

S.A. Armstrong Limited
23 Bertrand Ave.
Toronto, Ontario
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ARMSTRONG®

FILE NO:	43.545
DATE:	July 15, 2003
SUPERSEDES:	5043.9145
DATE:	Apr. 3, 1990
SUPERSEDES:	5043.9065
DATE:	Jan. 9, 1990

Series 4300 Vertical In-Line 6 x 6 x 13 SUBMITTAL

JOB: ______________________ REPRESENTATIVE: ______________________

______________________ ______________________

______________________ ORDER NO: __________ DATE: __________

ENGINEER: ______________________ SUBMITTED BY: __________ DATE: __________

CONTRACTOR: ______________________ APPROVED BY: __________ DATE: __________

PUMP DESIGN DATA			
NO. OF PUMPS:			
TAG:			
CAPACITY:	USgpm (L/s)	HEAD:	ft. (m)
LIQUID:		VISCOSITY:	
TEMPERATURE:		SPECIFIC GRAVITY:	

MOTOR DESIGN DATA			
HP:			RPM:
FRAME SIZE:		ENCLOSURE:	
VOLTS:		HERTZ: 60 Hz	PHASE:
EFFICIENCY:	☐ EPAct / NRcan	☐ PREMIUM	LEVEL: %
FLANGE SIZE:	SUCTION: 6" (150 mm)	DISCHARGE: 6" (150 mm)	

MATERIALS OF CONSTRUCTION					
ANSI FLANGE RATING	☐ 125 lb. CAST IRON			☐ 250 lb. DUCTILE IRON	
CONSTRUCTION	☐ BF	☐ AI	☐ AB	☐ DBF	☐ DI
CASING	CAST IRON	CAST IRON	BRONZE	DUCTILE IRON	DUCTILE IRON
IMPELLER	BRONZE	CAST IRON	BRONZE	BRONZE	CAST IRON
FLUSH LINE	COPPER	STEEL	COPPER	COPPER	STEEL
PUMP SHAFT	416 STAINLESS STEEL				
COUPLING	HIGH TENSILE ALUMINUM				
GASKET	CONFINED NON-ASBESTOS				

MECHANICAL SEAL DESIGN DATA

See File No. 43.50 for standard mechanical seal details as indicated below

Armstrong Seal Reference Number

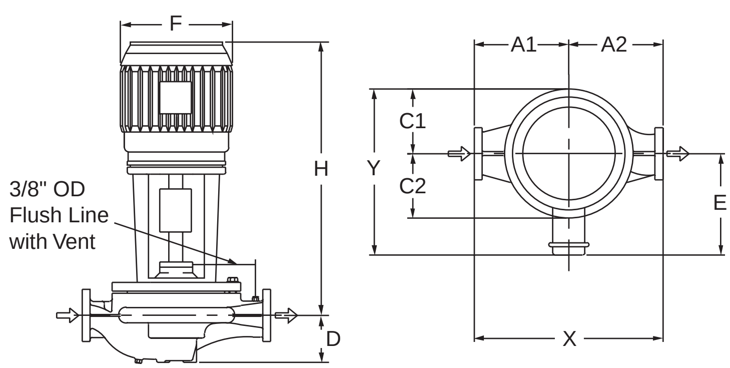
☐ C1 (a) Other____________

SEAL DATA OPTIONS

OPTIONAL EQUIPMENT

SEE FILE NO. 43.50 FOR ENVIRONMENTAL ACCESSORIES

MAXIMUM PUMP OPERATING CONDITIONS	
125 lb. CAST IRON	250 lb. DUCTILE IRON
175 psig at 140°F (12 bars at 60°C) 100 psig at 300°F (7 bars at 149°C)	375 psig at 100°F (26 bars at 38°C) 275 psig at 500°F (19 bars at 260°C)
SEE FILE NO. 43.50 FOR MAXIMUM SEAL OPERATING CONDITIONS	



- COUPLING GUARD (NOT SHOWN) IS SUPPLIED.
- TOLERANCE OF ±0.125" (±3 mm) SHOULD BE USED.
- FOR EXACT INSTALLATION DATA PLEASE WRITE FACTORY FOR CERTIFIED DIMENSIONS.
- SEE PERFORMANCE CURVES ON PAGE 2.
- PUMP EQUIPPED WITH CASING DRAIN PLUG AND 1/4" NPT SUCTION AND DISCHARGE GAUGE PORTS

PUMP DIMENSIONS inches (mm)

A1	A2	C1	C2	D	X
19.00 (483)	17.00 (432)	10.94 (278)	8.95 (227)	10.25 (260)	36.00 (914)

MOTOR HORSEPOWER							MAX. DIMENSIONS inches (mm)				ASSEMBLY WEIGHT*	
ODP			TEFC			Frame	ODP / TEFC				lbs. (kg)	
3600	1800	1200	3600	1800	1200		E	F	H	Y	ODP	TEFC
---	---	5	---	---	5	215TC	8.25 (210)	12.13 (308)	31.99 (813)	19.19 (487)	700 (317.8)	770 (349.6)
---	15	7.5	---	15	7.5	254TC	9.94 (252)	13.38 (340)	36.93 (938)	20.88 (530)	825 (374.6)	870 (395.0)
---	20	10	---	20	10	256TC	9.94 (252)	13.38 (340)	36.93 (938)	20.88 (530)	830 (376.8)	900 (408.6)
---	25	15	---	25	15	284TC	12.00 (305)	15.31 (389)	43.75 (1111)	22.94 (583)	905 (410.9)	910 (413.1)
---	30	---	---	30	---	286TC	12.00 (305)	15.31 (389)	43.75 (1111)	22.94 (583)	905 (410.9)	960 (435.8)
---	40	---	---	40	---	324TC	13.91 (353)	17.00 (432)	45.31 (1151)	24.85 (631)	1055 (479.0)	1090 (494.9)
---	50	---	---	50	---	326TC	13.91 (353)	17.00 (432)	45.31 (1151)	24.85 (631)	1055 (479.0)	1230 (558.4)

*Assembly weight combines pump and motor.

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PERFORMANCE CURVES

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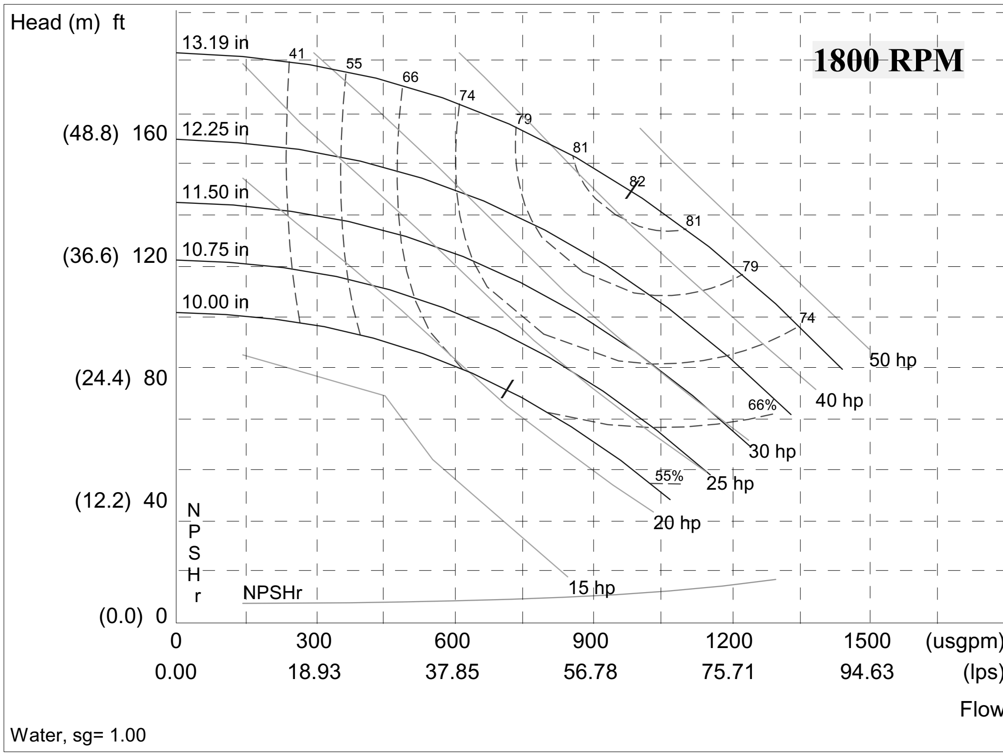
CURVE SHOWN FOR CLEAR, COLD WATER - SP. GR. 1.0



CURVE NO.
PT106-0-0-1800

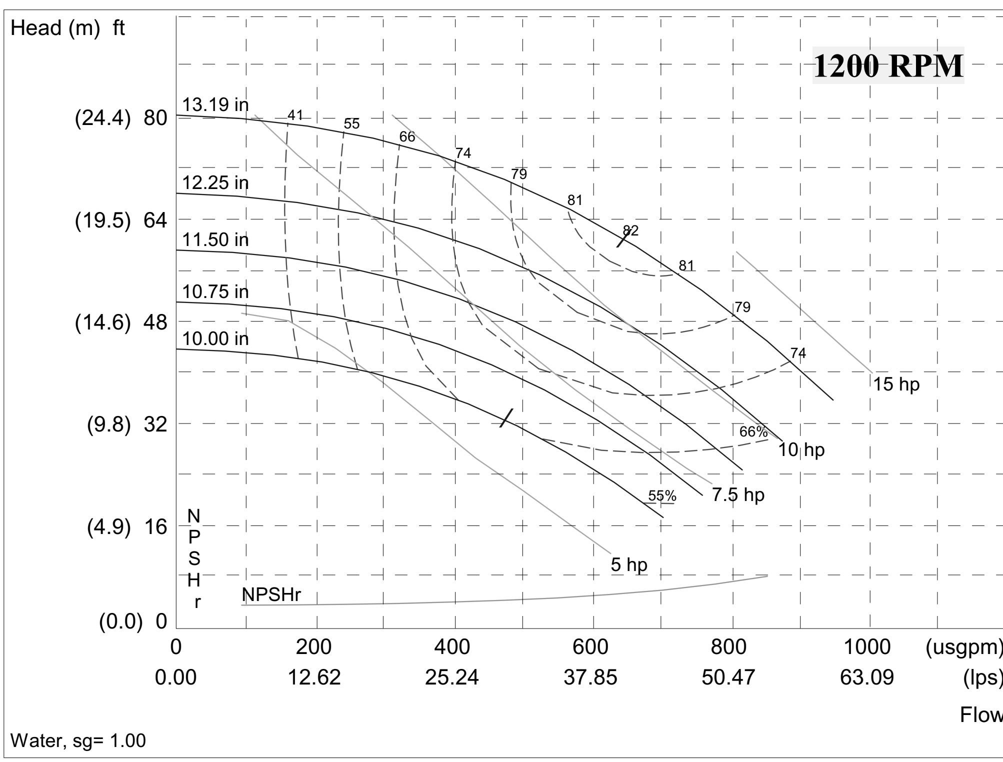
SERIES
4300
4380

SIZE
6x6x13

RPM
1800

BHP based on shown Fluid's sp. gr.

AVAILABILITY	
4300	All Ratings
4380	All Ratings



CURVE NO.
PT106-0-0-1200

SERIES
4300
4380

SIZE
6x6x13

RPM
1200

BHP based on shown Fluid's sp. gr.

AVAILABILITY	
4300	All Ratings
4380	All Ratings

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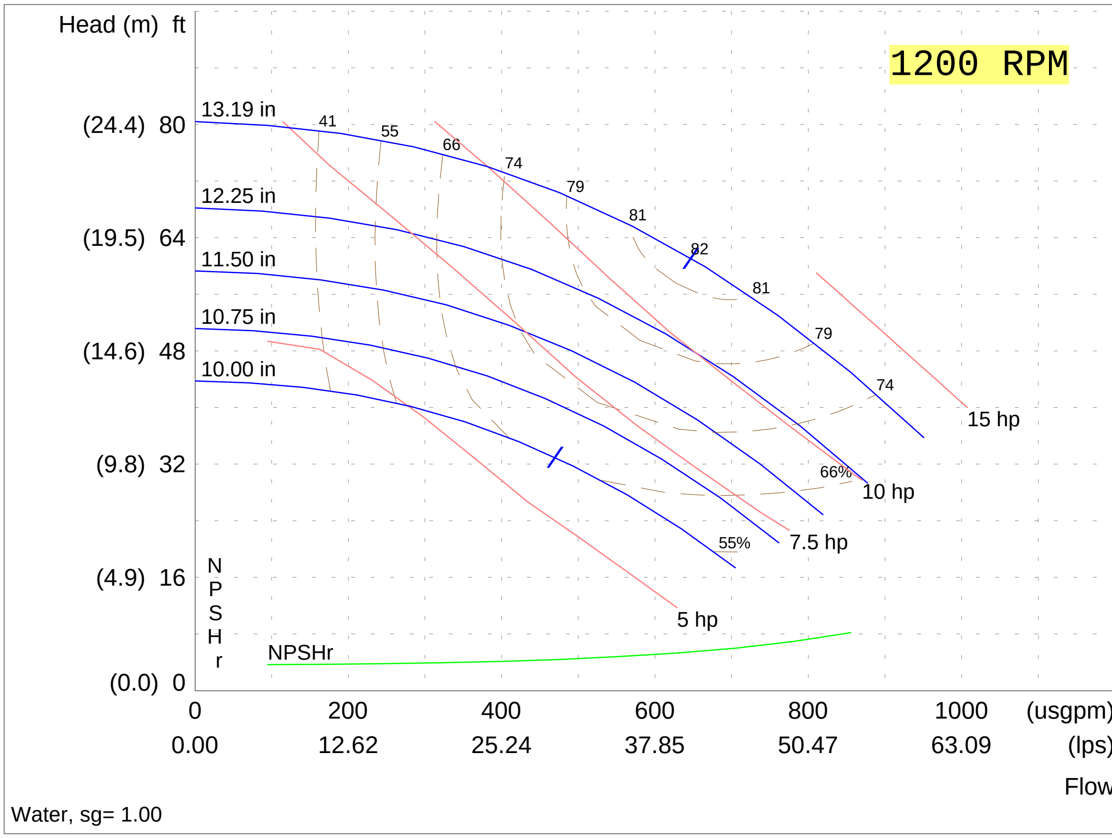
Performance Guaranteed Only At Operating Point Indicated

File No:	
Date:	June 1,2000
Supersedes:	NEW
Date:	NEW



CURVE NO.
PT106-0-0-1200
SERIES
4300 4380
SIZE
6x6x13
RPM
1200
BHP based on shown Fluid's sp. gr.

AVAILABI...	
	4300 All Ratings
	4380 All Ratings

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ARMSTRONG

Series 4030

FILE NO:	40.12
DATE:	July 31, 2004
SUPERSEDES:	40.12
DATE:	July 15, 2001

Horizontal Base Mounted Pumps

Series 4030
Armstrong Pumps - Hallmark of Quality

- Armstrong - Manufacturer of pumps since 1934
- Base mounted pump designs continuously updated
- Traditional features combined with cutting edge concepts

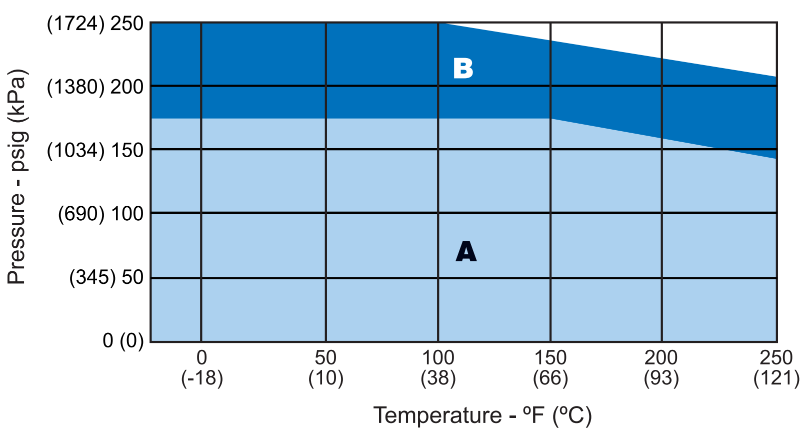
Traditional Features:

- Back Pull Out Design
- One Piece Baseplate
- Base Supported Radially-Split Casing
- Flexible Coupling, with Guard
- Drain and Gauge Connections
- Cast Iron Housing, Bronze-Fitted Construction
- All Iron and Ductile Iron Construction available
- Designed, Manufactured and Inspected to Exacting Standards

Current Design Concepts:

- ANSI style Centerline Discharge Casing
- Large Flow Range
- ANSI Flanged Casing
- Pre-Lubricated and Sealed Ball Bearings
- Confined Casing Gasket
- Mechanical Seal with O-Ring Mounted Seat
- Stainless Steel Shaft Sleeve
- Dynamically Balanced Impellers
- OSHA Coupling Guard

Pressure① / Temperature Chart② Series 4030



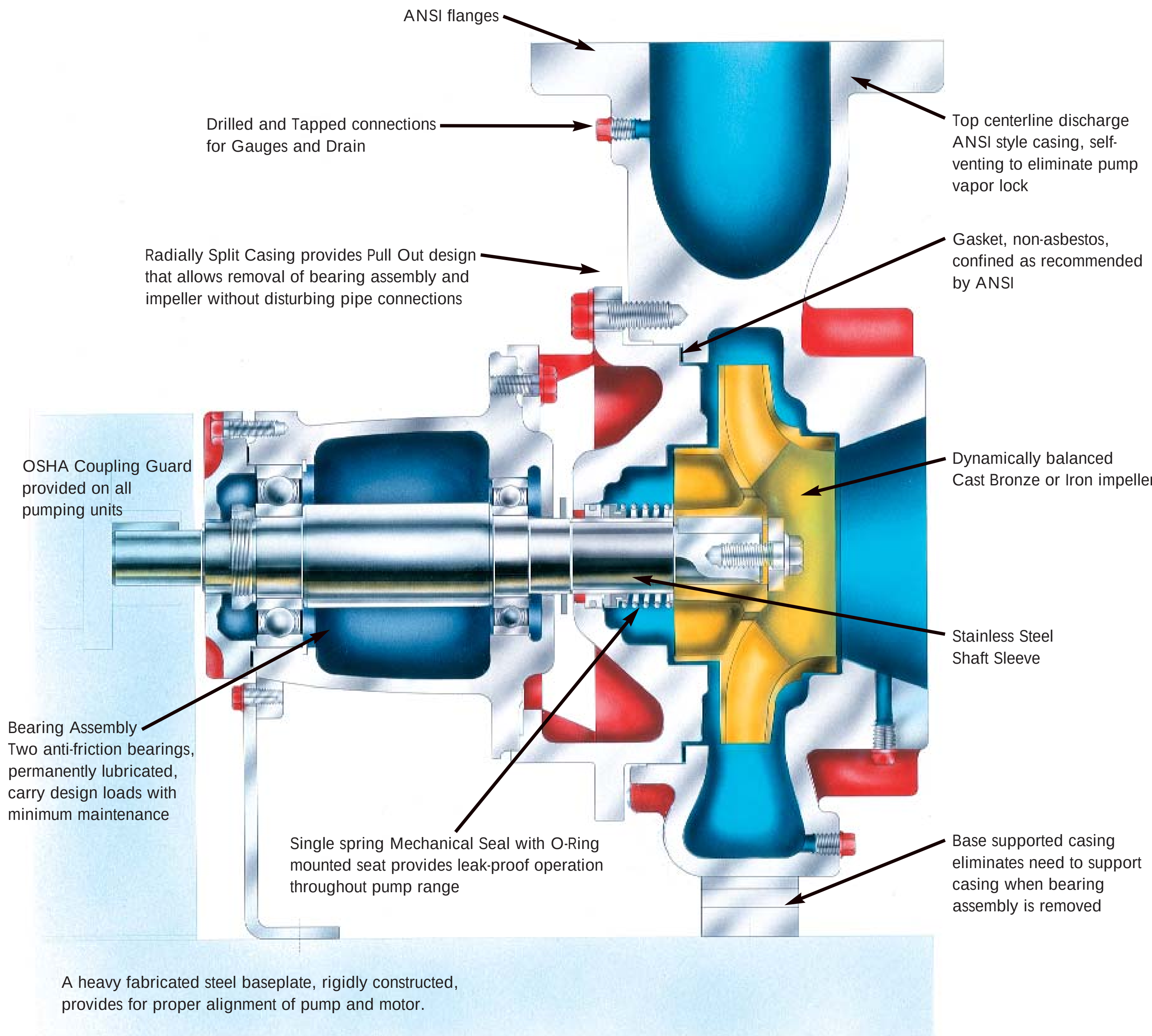
A. CAST IRON - 125 psig flanges
Standard seal

B. DUCTILE IRON - 250 psig flanges
Tungsten Carbide seal

① Hydrostatic test pressure at ambient temperature is 150% maximum working pressure.

② All values are based on clear, clean water. Values may change with other liquids.

Base Mounted Centrifugal Pumps



Armstrong Series 4030 –
simply the best base mounted pump design in today’s HVAC industry!

Typical Specification

Furnish and install, as indicated on the plans and specifications, Armstrong Series 4030 base mounted centrifugal pumps.

The pump shall be single, end suction type with radially split, top center-line discharge, self-venting casing. The casing-to-cover gasket shall be confined on the atmospheric side to prevent blow-out possibility.

Pump construction shall be cast iron, bronze fitted (all iron, all bronze, ductile iron) and shall be fitted with a long-life, product lubricated, drip-tight mechanical seal, with O-ring seat retainer, designed for the specified maximum temperature and pressure.

The shaft shall be fitted with a Stainless Steel shaft sleeve and be supported by two heavy duty ball bearings. The design shall allow Back Pull Out servicing, enabling the complete rotating assembly to be removed without disturbing the casing piping connections.

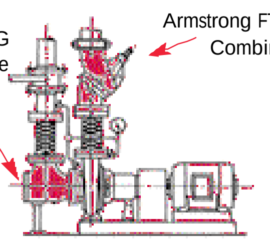
The pump shall be mounted on a rigid, single piece baseplate, with grouting hole, and connected by flexible coupling, with OSHA guard, to a ____ hp, ____ Hz, ___ phase, ____ Volts, _____ rpm, _____ enclosure squirrel cage, induction type motor of Federal approved (premium, ____%) efficiency level and suitable for across-the-line (wye-delta, part wind) starting.

The housing shall be hydrostatically tested to 150% maximum working pressure.

The unit shall be suitable for the conditions shown on the pump schedule.

Need to reduce space requirements and installation costs?

Base mounted

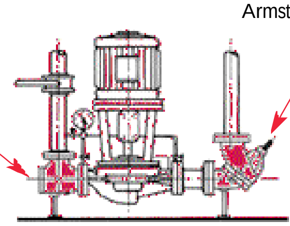


Base mounted pump with Suction Guide and Flo-Trex valve eliminates cost and space of:

Suction: Y-Strainer
Long Radius Elbow
Min. Straight Pipe Run

Discharge: Check Valve
Isolating Valve
Throttling Valve

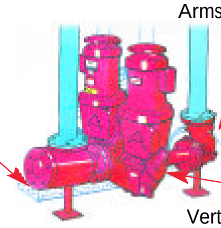
Vertical In-Line



Vertical In-Line with Suction Guide and Flo-Trex valve eliminates cost and space of all the items listed under base mounted pump, plus the following:

- Inertia Base with spring mounts
- Long Radius Elbow
- Flexible Pipe Connectors
- Grouting
- Field Alignment
- Split Couplings available for ease of mechanical seal replacement

dualArm



dualArm Vertical In-Line incorporates two pumps in a casing with single inlet and outlet connections. Enables standby or parallel operation with only one set of piping. Casing design and port valves allow one pump to be removed for service with the second pump still operating. When installed with a Suction Guide and Flo-Trex valve the ***dualArm*** represents the greatest Life Cycle Value in today's commercial HVAC market.

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SERIES 4030
3 x 1½ x 10
CENTERLINE DISCHARGE END SUCTION
BASEMOUNTED

FILE NO.:	40.61
DATE:	April 14, 1992
SUPERSEDES:	5040.9041 (U.S.A.)
DATE:	June 23, 1989
SUPERSEDES:	40.972 (Canada)
DATE:	July 15, 1984

ARMSTRONG **SUBMITTAL**

JOB

ENGINEER ____________________

CONTRACTOR ____________________

REPRESENTATIVE ____________________

ORDER NO. ____________________ DATE ________

SUBMITTED BY ____________________ DATE ________

APPROVED BY ____________________ DATE ________

PUMP DESIGN DATA	
NO. OF PUMPS	
TAG:	
CAPACITY: USGPM	
HEAD: FT.	
LIQUID:	VISCOSITY:
TEMPERATURE: °F	SPECIFIC GRAVITY:

MOTOR DESIGN DATA		
HP:	RPM:	
FRAME SIZE:	ENCLOSURE:	
VOLTS:	HERTZ:	PHASE:

FLANGE: SIZE SUCTION - 3", DISCHARGE - 1½"

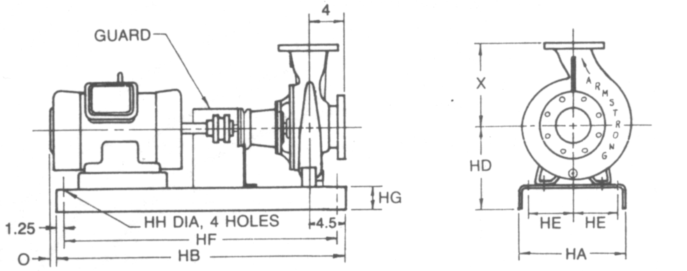
MATERIALS OF CONSTRUCTION				
ANSI FLANGE RATING	☐ 125 lb. CAST IRON		☐ 250 lb. DUCTILE IRON	
CONSTRUCTION	☐ BRONZE FITTED	☐ ALL IRON	☐ BRONZE FITTED	☐ ALL IRON
IMPELLER	BRONZE	C.I.	BRONZE	C.I.
CASING	CAST IRON		DUCTILE IRON	
GASKET	CONFINED NON-ASBESTOS FIBER			
BEARING FRAME	CAST IRON			
BEARINGS	ANTI-FRICTION GREASE LUBRICATED			
SHAFT	CARBON STEEL			
SHAFT SLEEVE	STAINLESS STEEL			
COUPLER	WOODS SURE FLEX OR EQUAL			
COUPLER GUARD	STEEL "U" FORM			
BASEPLATE	FABRICATED STEEL			

MECHANICAL SEAL DESIGN DATA		
PUMP FLANGE RATING	125 lb. CAST IRON	250 lb. DUCTILE IRON
SEAL TYPE	21	2
STATIONARY SEAT	NI-RESIST	TUNGSTEN CARBIDE
ROTATING FACE	CARBON	
SECONDARY SEAL	EPDM (not suitable for use with oil)	
SPRING	STAINLESS STEEL	
ROTATING HARDWARE	STAINLESS STEEL	

MAXIMUM OPERATING CONDITIONS	
125 lb. CAST IRON	250 lb. DUCTILE IRON
175 PSIG AT 150°F 140 PSIG AT 250°F	250 PSIG AT 100°F 210 PSIG AT 250°F

Pump equipped with casing drain plug and ¼" NPT suction and discharge gauge ports.

OPTIONAL EQUIPMENT
PERFORMANCE CURVES..SEE PG. 2



DIMENSIONAL DATA

PUMP SIZE	MOTOR FRAME SIZE i	MOTOR FRAME SIZE ii	HA	HB	HD	HE	HF	HG	HH	X	DIMENSION O DRIP PROOF i	DIMENSION O DRIP PROOF ii	DIMENSION O TEFC i	DIMENSION O TEFC ii	Wt. (lbs.)
3 x 1½ x 10	143T	145T	12	30	10¼	4½	27½	3	¾	8½	—	—	—	—	240
	182T	184T	12	30	10¼	4½	27½	3	¾	8½	—	—	—	—	250
	213T	215T	12	33	10¼	4½	30½	3	¾	8½	—	—	1	2½	282
	254T	256T	15	43	11¾	6	40½	3½	¾	8½	—	—	—	—	364
	284TS	286TS	15	43	11¾	6	40½	3½	¾	8½	—	—	—	1	374
	324TS	326TS	18	48	12¼	7½	45½	4	1	8½	—	—	—	—	412

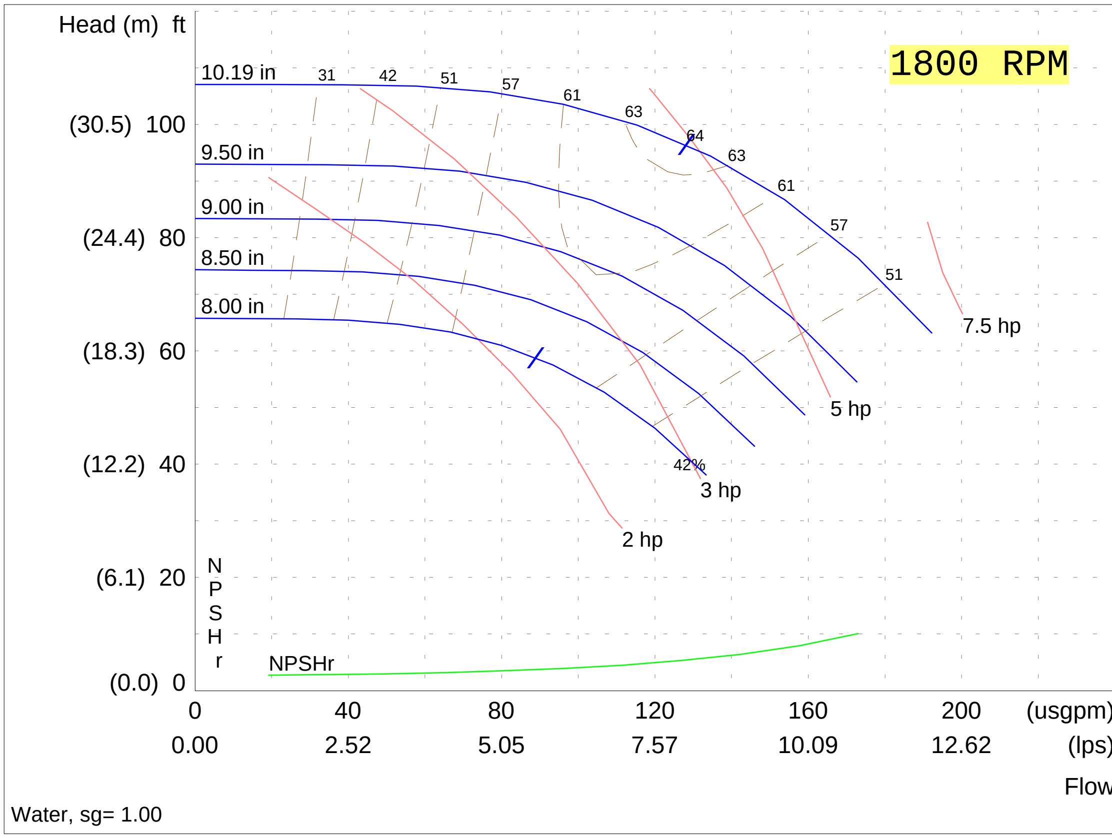
Dimensions and weights shown are for reference only. For exact installation data, contact Factory.

ARMSTRONG®

PERFORMANCE CURVES

Performance Guaranteed Only At Operating Point Indicated

File No:	
Date:	June 1,2000
Supersedes:	NEW
Date:	NEW



CURVE NO.
PT44-1-0-1800
SERIES
4030 4280
SIZE
3x1.5x10
RPM
1800
BHP based on shown Fluid's sp. gr.

AVAILABILI	
	4030 All Ratings
	4280 All Ratings

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Montreal, Quebec
Canada, H9P 2X5
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Fax: (514) 421-2436

Electrical Breadth

Appendix C

Electrical Data for Semco Packaged Energy Recovery Unit
Electrical Wiring Schematic for Enthalpy Wheel

Electrical Data

HP	3 Phase Full Load Amps 208V	240V	480V	Minimum Efficiency Std. Motors	Minimum Efficiency High Eff. Motors
1/6	0.6	0.6	0.3	-	-
1/4	1.0	1.0	0.5	-	-
1/2	2.4	2.2	1.1	-	-
3/4	3.5	3.2	1.6	73	-
1	4.6	4.2	2.1	76.6	82.5
1-1/2	6.6	6.0	3.0	80	84
2	7.5	6.8	3.4	79.9	84
3	10.6	9.6	4.8	83.1	86.5
5	16.7	15.2	7.6	83.4	87.5
7-1/2	24.2	22	11	86.6	88.5
10	30.8	28	14	88.2	89.5
15	46.2	42	21	89.3	90.2
20	59.4	54	27	90.4	91
25	74.8	68	34	90.5	92.4
30	88.0	80	40	89.3	93
40	114	104	52	90	93
50	-	130	65	91.2	94.1
60	-	-	77	92	93.6
75	-	-	96	92.4	94.1
100	-	-	124	92.5	94.1
HP	**3Ø Variable Frequency Drive (VFD)**			**Yaskawa Model #**	
1/2	3.9	3.9	-	CIMR-V7AM20P4	
1/2	-	-	1.6	CIMR-V7AM40P2	
1	6.4	6.4	-	CIMR-V7AM20P7	
1	-	-	4.7	CIMR-V7AM40P7	
	Control Power Transformer (CPT)				
150 VA	0.7	0.6	0.4		
500 VA	2.4	2.0	1.0		
3 KVA	14.4	13.0	6.25		

Note 1: To determine Minimum Circuit Ampacity, add the FLA's for each fan motor, the FLA of the constant speed wheel motor or the Variable Frequency Drive. Then add the CPT amps and 25 percent of the largest motor FLA.

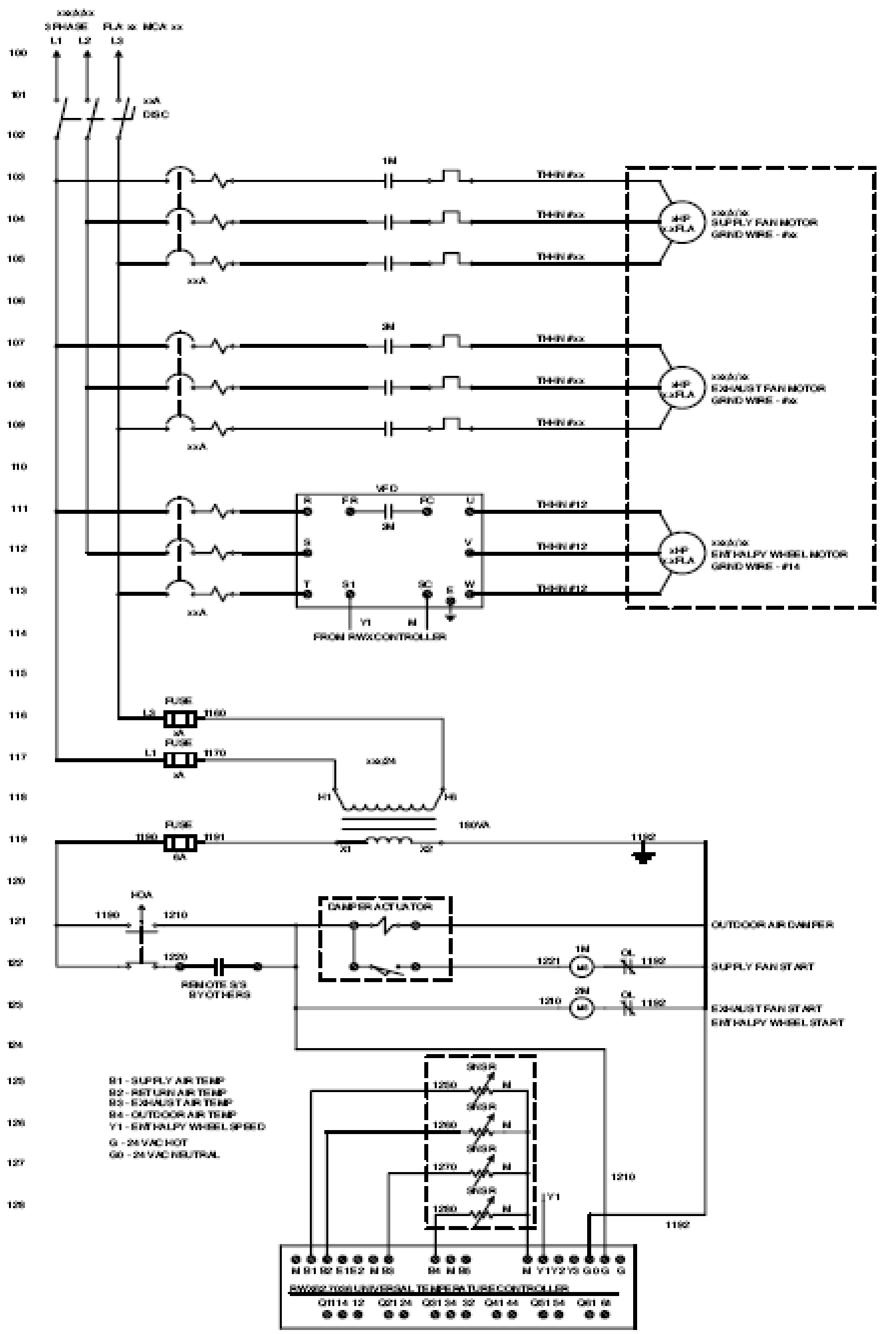
Note 2: Fuse Recommendations: Size fuses at the unit FLA and 75% of the largest motor FLA, then select the next larger size Dual-Element, Time-Delay Fuses (LOW-PEAK®, FUSETRON® or equivalent). If the fuses don't hold, consult N.E.C. for suitability of larger sized fuses.

Note 3: Use a 3KVA transformer for units with 120 volt lights. Otherwise use the 180 VA transformer.

Electrical Data For Semco Packaged Energy Recovery Unit

Single Wheel Electrical Schematic

Typical EP Series Unit with Variable Speed Wheel



Electric Wiring Schematic for Single Enthalpy Wheel Configuration

Structural Breadth

Appendix D

Hand Calculations

Building ______ Project No. ______
Subject 18×40 TO.4 Date ______
Sheet No. ____ of ____ Computed by ______ Checked by ______

LOADS

DEAD
57 PSF (3VLI19 6" SLAB)
2 PSF (CARPET)
2 PSF (FINISH CEILING)
10 PSF (MEP)
24 PSF (XTRA 2" CONCRETE)

LIVE
80 PSF (OPEN OFFICE W/ CORRIDOR THROUGHOUT)

$A_T = (40')\left(\frac{10'+10'}{2}\right) = 400\ FT^2$

$T_W = \frac{10'+10'}{2} = 10'$

$K_{LL} = 2$ (T.4-2 ASCE7-2005 INTERIOR BEAM)

LL REDUCTION (B/C $K_{LL} \cdot A_T > 400\ SF$)

$L = L_o\left(.25 + \frac{15}{\sqrt{2(400)}}\right) = 0.78 L_o > 0.4 \therefore$ USE $\underline{\underline{0.78 L_o}}$.

$L = 0.78(80\ PSF) = 62.4\ PSF$

$(W_u)_{L+D} = 1.2(95) + 1.6(62.4) = 213.84\ PSF$

$W_u = T_W W_u = (10')(213.84\ P/SF) = 2.14\ KLF$

$R = \frac{(2.14\ KLF)(40\ FT)}{2} = 42.8^K$

$M_{MAX} = \frac{WL^2}{8} = \frac{(2.14\ KLF)(40\ FT)^2}{8} = 428\ FT\text{-}K$

CURRENT = 18×40 $\phi M_N = 294\ FT\text{-}K$

$\boxed{\text{TRY W21×55}} \Rightarrow \phi M_N = 473\ FTK > 428\ FTK \Rightarrow$ OK

Flack + Kurtz
475 Fifth Ave.
NY, NY 10017
212-532-9600
WSP
A WSP Group company

Building ______ Project No. ______

Subject W18×40 (2ND one) Date ______

Sheet No. ____ of ____ Computed by ______ Checked by ______

LOADS

DEAD	LIVE
57 PSF (3VLI19 6" SLAB)	80 PSF (OPEN OFFICE W/ CORRIDOR) THROUGHOUT
2 PSF (CARPET)	
2 PSF (FINISH CEILING)	
10 PSF (NEP)	
24 PSF (XTRA 2" CONCRETE)	

$$A_T = (39.33')\left(\frac{10' + 10.75'}{2}\right) = 408.05\ FT^2$$

$$T_W = \frac{10' + 10.75'}{2} = 10.375\ FT$$

$K_{LL} = 2$ (T4-2 ASCE-7-2005 INTERIOR BEAM)

LL REDUCTION (B/C $K_{LL} \cdot A_T > 400$ SF)

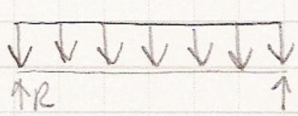
$$L = L_0\left(0.25 + \frac{15}{\sqrt{K_{LL}A_T}}\right) = 0.775 L_0 > 0.4 \therefore \text{USE } 0.775 L_0$$

$$L = 0.775(80\ PSF) = 62\ PSF$$

$$(W_u)_{L+D} = 1.2(95\ PSF) + 1.6(62\ PSF)$$

$$W_u = 213.2\ PSF$$

$$W_u = T_W W_u = (10.375\ FT)(213.2\ PSF) = 2.21\ KLF$$



$$R = \frac{2.21\ KLF(39.33')}{2} = 43.46\ K$$

$$M_{MAX} = \frac{WL^2}{8} = \frac{(2.21\ KLF)(39.33)^2}{8} = 427.3\ FT\text{-}K$$

CURRENT W18×40 $\phi M_N = 294$ FT-K $\qquad \phi M_N = 0.9 \cdot F_y \cdot Z$

$\boxed{\text{NEED W21×55}}$ $\phi M_N = 473$ FT-K

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Building ______ Project No. ______
Subject 18x35 (1ST) Date ______
Sheet No. ____ of ____ Computed by ______ Checked by ______

LOADS

DEAD
51 PSF (3VLI19 6" SLAB)
2 PSF (CARPET)
2 PSF (FINISH CLG.)
10 PSF (MEP)
24 PSF (XTRA 2" SLAB)

LIVE
80 PSF

$A_T = (38.64')\left(\frac{10.75' + 7.0'}{2}\right) = 343\ FT^2$

$T_W = \left(\frac{10.75' + 7.0'}{2}\right) = 8.875\ FT$

$K_{LL} = 2$ (T.4-2 ASCE7-2005 INTERIOR BEAM)

LL Reduction (BC $K_{LL} \cdot A_T > 400\ FT^2$)

$L = L_o\left(0.25 + \frac{15}{\sqrt{2 \cdot (343)}}\right) = 0.823 L_o > 0.4 \therefore$ USE $0.823 L_o$

$L = 0.823(80\ PSF) = 65.8\ PSF$

$\omega_{U_{L+D}} = 1.2(95) + 1.6(65.8) = 219.3\ PSF.$

$W_u = T_W \omega_u = (8.875')(219.3\ PSF) = 1.95\ KLF$

$R = \frac{(1.95\ KLF)(38.64\ FT)}{2} = 37.67\ K$

$M_{MAX} = \frac{W_u L^2}{8} = \frac{(1.95\ KLF)(38.64\ FT)^2}{8} = 364\ FT\text{-}K$

CURRENT: 18x35 $\phi M_N = 249\ FT\text{-}K$

$\boxed{\text{TRY: W21x48}}$ $\phi M_N = 398\ FT\text{-}K > 364\ FT\text{-}K \Rightarrow$ OK!!

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Building ______ Project No. ______
Subject 18x35 (2nd) Date ______
Sheet No. ___ of ___ Computed by ______ Checked by ______

LOADS
57 PSF (3VLI19 6" SLAB)
2 PSF (CARPET)
2 PSF (FINISH CEILING)
10 PSF (MEP)
24 PSF (xtra 2" CONCRETE)

LIVE
80 PSF (OPEN OFFICE/ corridor)

$$A_T = (38.2')\left(\frac{7.0' + 6.0'}{2}\right) = 248.3 \text{ FT}^2$$

$$T_W = \left(\frac{7.0' + 6.0'}{2}\right) = 6.5'$$

$K_{LL} = 2$ (T.4-2 ASCE7-2005 INTERIOR BEAM)

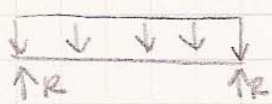
LL REDUCTION (B/C $K_{LL} \cdot A_T > 400 \text{ FT}^2$)

$$L = L_o\left(0.25 + \frac{15}{\sqrt{2(248.3)}}\right) = 0.92 L_o > 0.4 \therefore \text{USE } 0.92 L_o.$$

$$L = 0.92(80 \text{ PSF}) = 73.8 \text{ PSF}$$

$$(\omega_u)_{L+D} = 1.2(95) + 1.6(73.8) = 232.1 \text{ PSF}$$

$$W_u = T_W \omega_u = (6.5')(232.1 \text{ PSF}) = 1.51 \text{ KLF}$$



$$R = \frac{(1.51 \text{ KLF})(38.2')}{2} = 28.84 \text{ K}$$

$$M_{max} = \frac{WL^2}{8} = \frac{1.51 \text{ KLF}(38.2 \text{ ft})^2}{8} = 275.4 \text{ K-FT}$$

CURRENT: W18x35 $\phi M_N = 249$ K-FT

$\boxed{\text{TRY: W18x40}} \Rightarrow \phi M_N = 294 \text{ FT-K} > 275 \text{ FT-K} \Rightarrow$ OK!!

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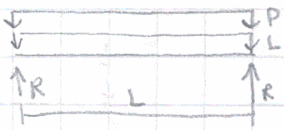
Building ____________ Project No. ______

Subject Girder (T.C) ____________ Date ______

Sheet No. ______ of ______ Computed by ____________ Checked by ____________

W18x40:



$D = (DL)(L)(T_W) = (95PSF)(39.33')(10.375') = 38.8^K$

$L = (80)(39.33)(10.375) = 32.6^K$

$R = \frac{1}{2}(D+L) = 35.7^K$

W18x35: $D = (95PSF)(38.64')(8.875') = 32.6^K$ $\quad R = 30^K$

$L = (80PSF)(38.64')(8.875') = 27.4^K$

W18x35: $D = (95PSF)(38.2')(6.5') = 23.6^K$ $\quad R = 21.75^K$

$L = (80PSF)(38.2')(6.5') = 19.9^K$

W18x40: $D = (95PSF)\left(\frac{10 + 11.625}{2}\right)(40') = 41^K$ $\quad R = 37.8^K$

$L = (80PSF)\left(\frac{10 + 11.625}{2}\right)(40') = 34.6^K$

W18x40: $D = (95PSF)\left(\frac{11.583 + 11.625}{2}\right)(40') = 44.1^K$ $\quad R = 40.6^K$

$L = (80PSF)\left(\frac{11.583 + 11.625}{2}\right)(40') = 37.1^K$

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Printout from STAAD Pro used to analyze the W30X90 Girder.