2.0 Depth Work - Alternative Mechanical Designs

2.1 Objectives

The goal of designing alternative systems is to compare energy usage, system costs, emissions, lost rentable space, and constructability. By considering the original concerns of the Philadelphia School District, one final design will be recommended based on the four topics listed above. These comparisons can be used for reference in designing other similar office buildings.

2.2 Overview of Systems

The concentration in the depth work of this report focuses on two concepts. The first is the comparison of two different airside systems: VAV and dedicated outdoor air. The second is the comparison of the existing DX-electric system to a central water plant which supplies cold and hot water to the airside system. In all cases, the fourth floor will be simulated as a constant air volume system. Also, cooling for the atrium and lobbies will not be considered. The analysis will concentrate on open office loads and the energy usage by different systems when used in commercial office buildings. The following five systems will be analyzed:

System 1: VAV with DX coil and electric heating coils
System 2: VAV with chilled water and hot water coils
System 3: DOAS/VAV with DX coils and electric heating coils
System 4: DOAS/VAV with chilled water and hot water coils
System 5: DOAS/Radiant with chilled water and hot water coils

2.2.1 Airside

There will be a comparison of 2 different airside systems. The existing VAV system will be analyzed and a dedicated outdoor air system will be analyzed. The dedicated outdoor air system will be modeled separately with VAV as a parallel system and with radiant panels as a parallel system.

VAV — Systems 1 and 2

The VAV system can exist as a stand alone system as it does currently in the Philadelphia School District Administration Headquarters. Because of the nature of the loads within the SDP building, cooling is needed all year. Depending on the cooling load within each space, a certain quantity of air will be delivered to that space. For heating needs, return air in the plenum will be circulated directly back to the room and heated via a fan and heating coil in the parallel fan powered box.

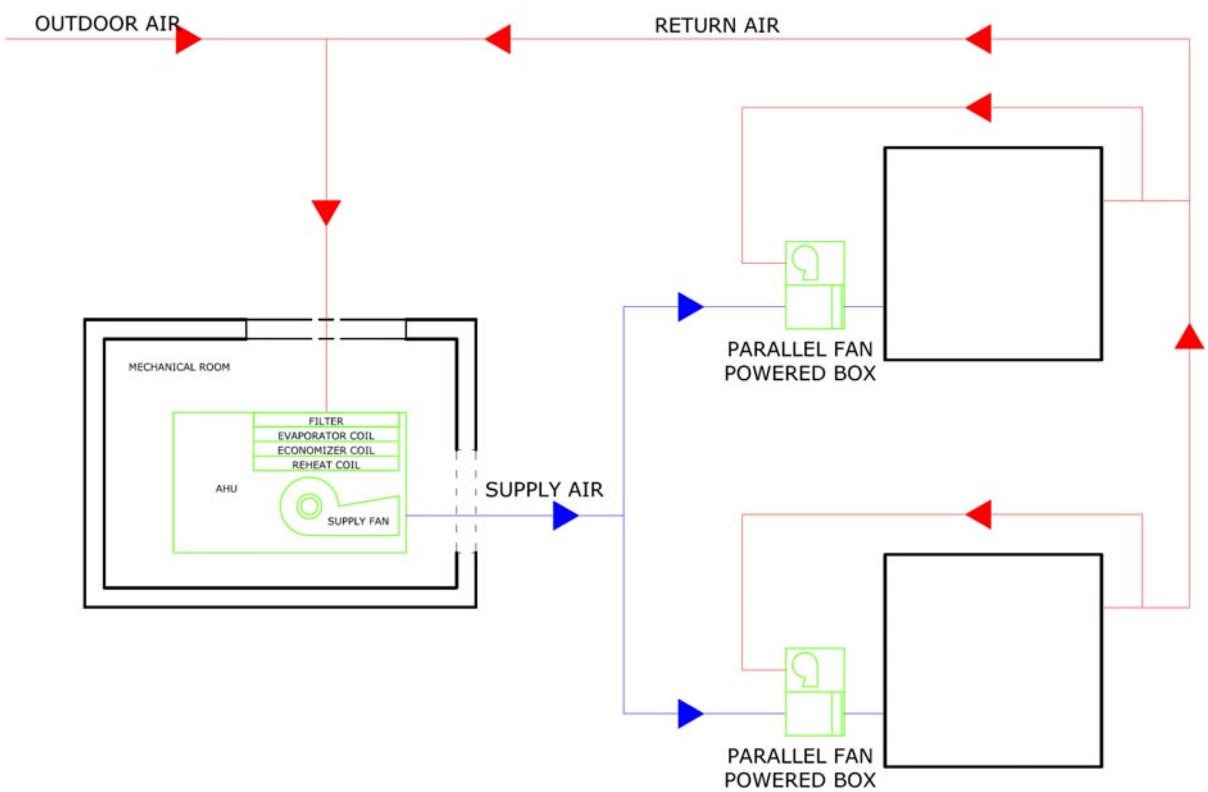

Figure 2.2A. Plan of Airside VAV System.

DOAS — Systems 3, 4, and 5

A dedicated outdoor air system (DOAS) supplies the ventilation requirement for a particular space and is responsible for removing the latent load of that space. It is used with a parallel system which satisfies the rest of the sensible load of the space. In this report two parallel systems will be considered. The first is the existing VAV system with parallel fan-powered boxes. The second is radiant panels used for cooling and radiant baseboard heating. The DOAS in this report uses an enthalpy wheel with an effectiveness of 80 percent. Outdoor air is sent through the enthalpy wheel first. Depending on the season, temperature and humidity are increases or decreased.

During cooling, the enthalpy wheel decreases the outdoor air temperature and humidity by use of heat recovery from the exhaust air stream. The outdoor air is sent through a cooling coil next where its humidity is reduced and its temperature is reduced to 45F. This air can either be reheated to 55F for supply air using a sensible wheel in the outdoor air/exhaust air streams or it can be delivered directly to a space at 45F. The sensible wheel only increases the temperature of the air; it does not alter the humidity of the air. When considering the supply air temperature of the DOAS, it is important to remember that the DOAS is responsible for removing the entire latent load of the space. The supply temperature was determined based on the properties of the parallel systems.

During heating, the DOAS increases the temperature and humidity of the outdoor air by use of heat recovery from the exhaust stream. The enthalpy wheel has the ability to increase the temperature to 58.2F. This can only be done with the VAV system for reasons explained later. With the need for cooling during the winter, the air handling unit does not need a heating coil.

DOAS Analysis

Depending on the parallel system, the conditions of the supply air for a dedicated outdoor air system may be more or less important to control. An analysis was done on the components of the dedicated outdoor air system's air handling units. The supply air conditions required in this report were not considered in this study. The purpose of this study is to compare the cooling capacity of different configurations with the assumption that the entire latent load is satisfied by the DOAS.

Heat recovery is not a “required” component of a dedicated outdoor air system. A cooling coil can be used to remove the entire latent load of a space; however, this idea is not practical. It requires an enormous cooling coil, has a high first cost, and consumes a lot of energy. Therefore, heat recovery in a DOAS is a must. Heat recovery in the form of an enthalpy wheel, as used in the simulations in this report, reduces the load on the cooling coil. The cooling coil will reduce the outdoor air temperature to 45F and saturated. The remaining option in a DOAS is the use of reheat. This air at 45F can be supplied to the room or it can be reheated to the traditional supply air temperature of 55F. Reheat requires another wheel, a sensible wheel. This component does only sensible cooling or heating. It recovers heat from the exhaust stream to increase the temperature of the 45F air to the supply air requirement. The humidity ratio of the conditioned air does not change as it moves through the sensible wheel. Appendix B provides state point conditions and cooling capacity for the three different cooling models: 1. DOAS with enthalpy and sensible wheel, 2. DOAS with enthalpy wheel, and 3. DOAS without heat recovery. Table 2.2A summarizes the results for space FL-1 NE. The enthalpy wheel only configuration requires the least amount of cooling capacity. The supply temperature for configuration 1 with the sensible wheel is 55F which means the supply air cooling capacity is less than that of configuration 2 which supplies low temperature air at 45F.

		Required Cooling Capacity		
		DOAS	Parallel	Total
		tons	tons	tons
1	Enthalpy Wheel and Sensible Wheel	9.33	47.84	57.17
2	Enthalpy Wheel Only	11.8	36.39	48.14
3	Cooling Coil Only	153.4	38.64	192.02

Table 2.2A. DOAS Analysis Comparison for Space FL-1 NE.

The load across the cooling coil for both cases is the same because the supply air should be saturated in order to satisfy the latent load, but since the air in configuration 2 is heated back to 55F, the parallel system is required to cool more air than the parallel system in configuration 1. By adding another wheel to the outdoor air stream, the first cost of the system will increase on for both the DOAS and the parallel system. The results of this analysis makes the evaluation of reheat seem a little more important when designing the dedicated outdoor air handling unit.

<u>Radiant Panels — System 5</u>

Cooling

The radiant system uses ceiling panels to distribute cooling through radiation. These panels are 2 by 4 foot panels which are similar to 2 by 4 ceiling tiles. Two by 4 foot ceiling tiles are currently installed in the Administration building and were probably a request of the district or a decision made by the architect. These panels would not be complicated to co-exist with the current ceiling layout. Each radiant panel consists of 5 passes of tubing where chilled water is passed through. The heat transfer mechanism of radiation takes over and cools the building spaces.

The radiant panels pose restrictions on the DOAS. No latent load can be left for the radiant panels and the dew point temperature of the DOAS supply air must be lower than the radiant panel surface temperature in the radiant cooling application. The panel surface temperature must exceed the room dew point temperature also. *Both the room dew point temperature and the supply air dew point temperature must be less than the radiant panel surface temperature.* The design room conditions for cooling are 75F and 50% relative humidity. The room dew point temperature that corresponds to these conditions is 55F. The design heating temperature for winter is 70F and 50% relative humidity which correspond to a 50F dew point temperature. The inlet water conditions for these radiant panels must be at least 1 degree above the room dew point temperature. Because the summer conditions control an inlet water temperature of 56F must be used. Therefore, the dew point temperature of the DOAS supply air must be less than 56F. If the occupancy of the space increased, room dew point would increase. This will cause condensation on the radiant panels if air was supplied at 55F and saturated. Using a supply air temperature of 55F would make the chance of condensation more uncertain. Therefore, a supply temperature of 45F will be used to keep the dew point below panel surface temperature. Because the supply temperature is lower than the normal 55F supply temperature, high induction diffusers will be used with the radiant panels. High induction diffusers supply air through a narrow stream at high velocity. The high velocity causes low pressure relative to the ambient room pressure. Because of this pressure differential, room air is drawn into the supply stream where it is quickly mixed within the room. With the DOAS and radiant panels both the latent and sensible load are satisfied.

Heating

In the winter, the dedicated outdoor air system can supply up to 58.2F air. Because SDPAH requires cooling in the winter, the DOAS must supply at 45 to avoid condensation. The leaving temperature of the enthalpy wheel is controlled by its speed. Therefore, supplying 45F air in the winter is not a problem. For any instances of heating needs, the radiant panel system is supplemented by radiant baseboard heaters along the perimeter of the building. When they sense that the room temperature is below heating thermostat temperature, they will turn on and circulate hot water through finned tubes allowing radiant heating to occur.

VAV — Systems 4 and 5

The cooling supply air temperature in the summer for the DOAS with VAV as the parallel system was also determined to be 45F. For the DOAS/VAV combination the VAV system satisfies the remaining sensible load as does the radiant panels in the previous section. The outdoor air from the DOAS can be mixed with the conditioned air from the VAV unit downstream of the VAV air handling unit. The supply air for the DOAS/VAV system does not have limits on it as it did with radiant panels. This allows the 45F DOAS air to be mixed with the 55F VAV air after both are conditioned. The schematic in Figure 2.2B shows an airside riser diagram for the mixing conditions. The amount of outdoor air being supplied at 45F by the DOAS is small compared to the amount of return air being conditioned to 55F by the VAV system. Because of the ratio of the two supply streams, the supply temperature after they mix is still close to 55F.

During the winter season, the enthalpy wheel increases the temperature and humidity of the outdoor air to 58.2F and 44.2g/lb, respectively. Because all the spaces in the SDPAH require cooling in the winter, this temperature air can mix with the air being supplied by the VAV units. To save energy, the wheel can also be set to heat to 45F where mixing conditions would be very similar to those for the summer. The VAV system has terminal heating at the parallel fan powered boxes. This takes care of any chances of space temperature dropping below heating thermostat temperature.

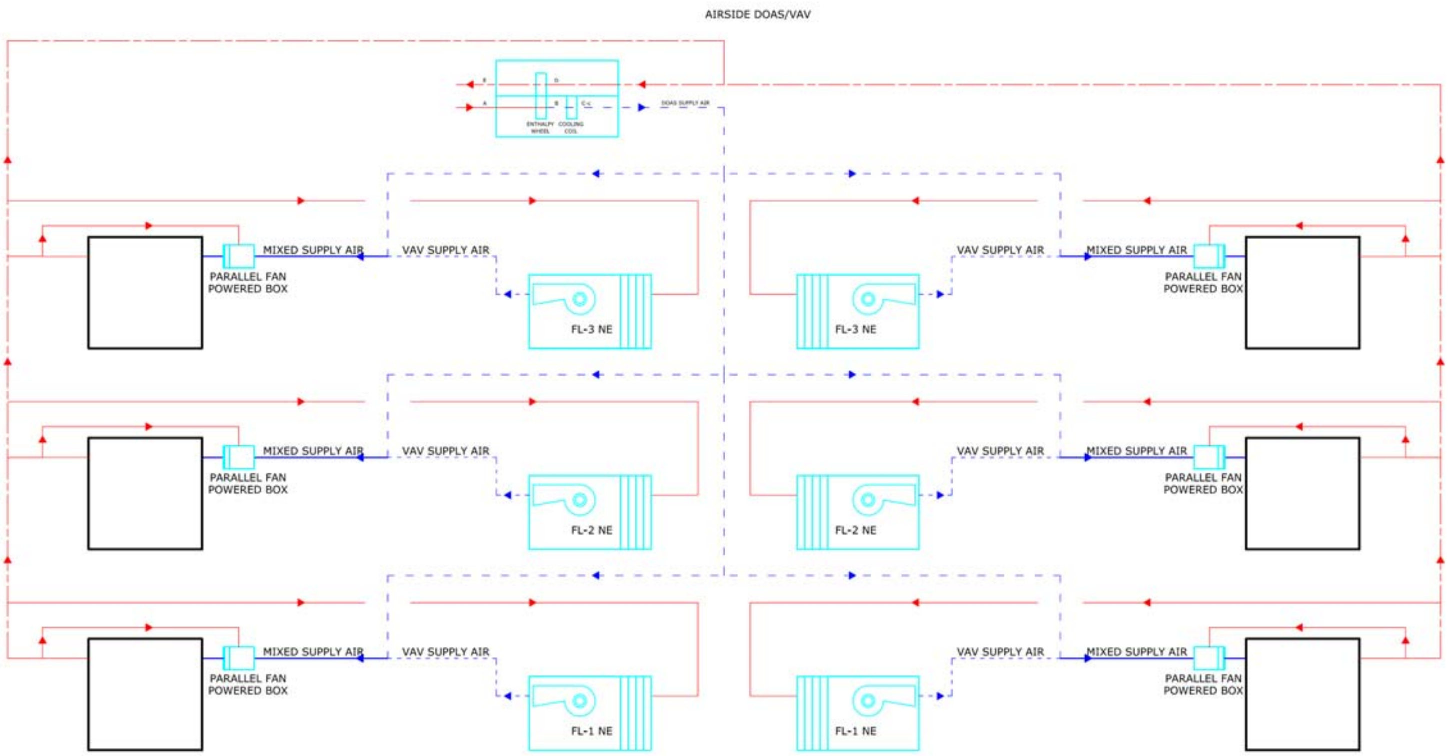

Figure 2.2B. DOAS/VAV Airside Riser Diagram.

CAV—Floor 4

A constant air volume (CAV) system will be used for the fourth floor in all simulations. Floor 4 is known to be utilized as a data center and always has a constant cooling load. Because of this a constant air volume system is the most logical choice.

2.2.2 Water Side

The purpose of comparing the existing DX-electric system to a central water plant is to find yearly energy savings. The current system uses a large quantity of energy (Table 1.3C, Page 19) and a central water plant has the potential to reduce the amount of energy used.

Central Chilled Water Plant

<u>Chiller</u>

In the existing system the refrigeration cycle occurred locally within self-contained packaged air handling units positioned throughout the building using a DX (direct expansion) coil and a waterside economizer coil. An alternative to the existing system is using a central chilled water plant for the main cooling source. For cooling, the refrigeration cycle occurs in a central chiller. A chiller has all of the refrigeration components of the self contained packaged air handling units. Cool refrigerant is passed through an evaporator where it evaporates by taking the heat out of warm water returned from the air handling units or radiant panels. This chilled water is sent to the load and the hot refrigerant vapor passes to a compressor usually at 44F or 45F. A centrifugal compressor is chosen for the models in this report. This compressor has rotating mechanical element that exchanges angular momentum with the refrigerant as it continuously passes through it. The refrigerant at high pressure then passes to the condenser where it will be cooled by water from a cooling tower. The cool refrigerant passes through a pressure reducing element in order for it to enter the evaporator at low pressure. The chiller must be capable of supplying lower temperature water for the DOAS in order to supply 45F air. A heat exchanger must also be used in the application of radiant panels where entering water temperature is 56F. It should be noted that the controls between the chiller and the airside system must be extremely accurate. The focus of this report was on system performance without getting into detail about the system controls.

Cooling Tower

Condenser water from the cooling tower helps to condense the refrigerant after the cooling process. After heat rejection from the refrigerant occurs, heat rejection by the condenser water must occur. After the condenser water leaves the chiller, it is usually 95F. This hot water needs to be cooled again to flow back to the chiller. This occurs by evaporation. The goal is to expose as much water surface area to the air to induce the evaporation process. For the simulation in this report, a “direct induced draft” cooling tower is used. It is direct because the water is in direct contact with the air. It is induced draft because the air is being pulled through the tower by a high pressure water spray instead of blown through, which requires mechanical energy. The induced draft tower is widely used because of its energy efficiency. It uses an axial fan instead of a centrifugal fan. A centrifugal fan requires almost twice as much energy as an axial fan.

Economizer

An economizer cycle is used to model the different systems in this report. A plate and frame heat exchanger can be connected between the condenser water loop and the chilled water loop but the two loops are kept separate. The chilled water loop is connected to the cooling tower loop to transfer heat while the cooling tower loop can bypass the chiller and can provide free cooling to the space when outdoor air conditions are favorable. Favorable outdoor air conditions are typically below 55F.

Central Hot Water Plant

Perimeter heating is needed when the internal load is not large enough to satisfy the perimeter heat losses. Comparisons will be made between the existing electric heating coils and hot water coils served by the hot water plant. A gas fired hot water boiler will be used in the analysis of hot water verses electric heating coils. The boiler will be located on the roof and serve the preheat coils in the AHUs and the coils in the parallel fan powered boxes for the VAV system and the baseboard heaters for the DOAS/Radiant heating system.

2.3 Equipment

The simulations in this report were based on properties of equipment used for each system.

2.3.1 Self Contained Packaged VAV Air Handling Units

The air handling units in the existing system are McQuay self contained water cooled plenum discharge (SWP). The same units were selected for the VAV parallel system in the DOAS application. These units were selected in the existing system for ease of installation and ease of maintenance. The selection of the units depended on the supply air flow for each space and the face velocity of the air across the coil. A small area and high velocity will give a small unit, small cost, and small mechanical room. However, condensate carryover must be considered. As hot air flows over the evaporator or cooling coil, condensate may occur. If the face velocity is too high the condensate will enter the duct system. It may cause damage to the ducts and may threaten indoor air quality. The face velocity rule of thumb of 500 feet per minute (FPM) was used to size the AHUs.

Figure 2.3A. McQuay Packaged SWP Air Handling Unit in Third Floor South Mechanical Room.

		DX-Electric Units			
Model		SWP095D	SWP095D	SWP095D	SWP065F
Floor		3	3	3	5
Quantity		2	2	1	2
Nominal cfm		33540	33540	33540	24900
SA cfm		30000	29450	27200	20700
Face Area of Unit	SF	55.9	55.9	55.9	41.5
Maximum Face Velocity	FPM	537	527	487	499
Condenser Flow Rate	GPM	257	257	256	170
DX					
Total Heat Capacity	MBH	638	632	595	429
Sensible Heat Capacity	MBH	656	654	636	454
Economizer					
Total Capacity	MBH	595	584	533	388
Sensible Capacity	MBH	576	563	505	370
Unit Size					
W	IN	84	84	84	81
L	IN	156	156	156	120
H	IN	88	88	88	88

Table 2.3A. Sample Schedule of SWP Units on Floor 3 and 5 of DOAS/VAV System.

2.3.2 DOAS Units

The dedicated outdoor air system requires the use of an enthalpy wheel. In this case the SEMCO TE3-was selected as the manufacturer for the air handling units. The selection of the units depended on the supply air flow for each space and the face velocity of the air across the coil. Because the DOAS units only supply outdoor air, the total quantity of the air is much smaller than what is being supplied by the VAV units. Thus the location and areas served by these units are different than the VAV units. The building was divided into a North section and a South section, each being served by a DOAS unit sized for the required outdoor air for the spaces within those sections. The face velocity parameter of 500 feet per minute (FPM) was used to size the AHUs.

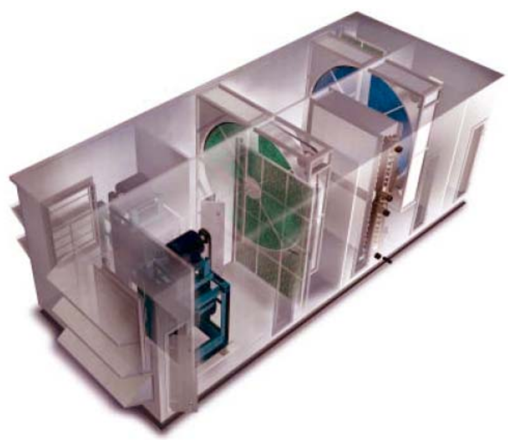

Figure 2.3B. SEMCO Packaged Energy Recovery System, EPC-24/EP-43

	Airflow	Chilled Water	Motor	
			Supply	Return
	CFM	GPM	HP	HP
EPC-24	16150	119	15	15
EPC-43	29100	256	20	20

Table 2.3B. DOAS AHU Schedule.

2.3.3 Enthalpy Wheel

The enthalpy wheel in the SEMCO unit can be individually selected. In this case the EXCLU-SIEVE enthalpy exchanger is chosen. The enthalpy wheel slowly takes the sensible and latent energy from the exhaust stream and shifts it to the transfer core which is made of an aluminum coating with a 3Å molecular sieve desiccant. The energy from the exhaust is transferred to the supply stream saving energy used by traditional systems. This occurs because the air entering the cooling coil after leaving the enthalpy wheel is cooler than it would have been if it skipped the heat recovery process by the enthalpy wheel. The cooling coil is then required to do less work.

In the winter, frost is a possibility on the enthalpy wheel in cold climates. The winter room design conditions are 70F, 50% RH and the winter outdoor air design conditions are 11F, 30% RH. If these two points are connected on a psychrometric chart and the line crosses the saturation curve, then frost may occur. By doing this, it is found that frost does not occur.

When the outdoor air temperature starts to get near the supply temperature an economizer cycle begins. The wheel's speed will slow in response to the set supply temperature. The wheel effectiveness is decreased, however, free cooling is being provided.

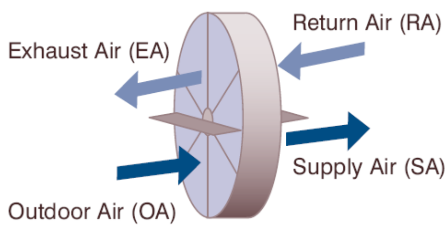

Figure 2.3C. SEMCO Model TE3-24-3Å/TE3-43-3Å

2.3.4 High Induction Diffusers

High induction diffusers must be used to supply low temperature air. Trane was chosen for these diffusers. These diffusers supply air tightly against the ceiling so that it induces the room air into its stream and the two can fully mix.

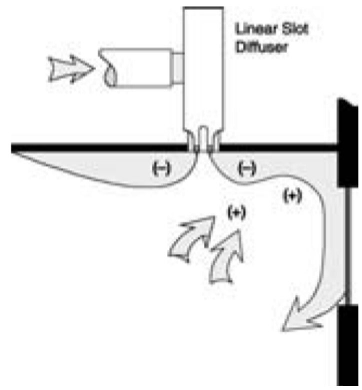

Figure 2.3D. Trane's High Induction Diffuser Performance.

2.3.5 Radiant Panels and Baseboard Heating

Sterling's smooth face linear extrusion radiant panel Type D at 35Btu/SF was used to complete the DOAS/Radiant calculations. The linear extrusion radiant panels will architecturally work with the current ceiling of 2 by 4 acoustic tiles. Sterling's Versa-Line finned tube baseboard radiators were selected for the baseboard heating in the DOAS/Radiant application.

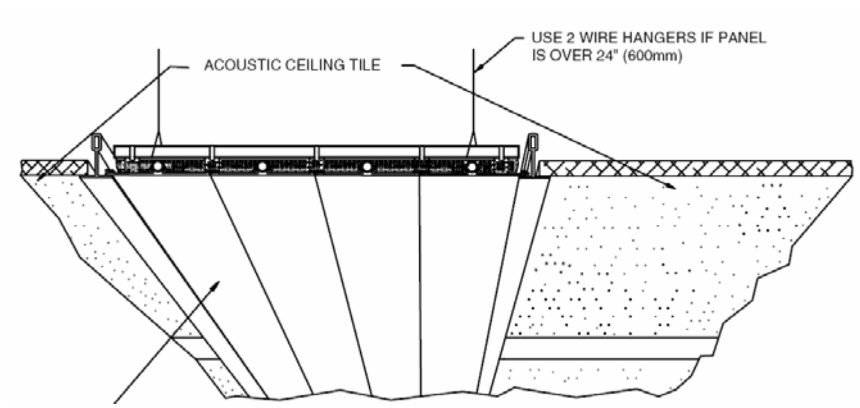

Figure 2.3E. Smooth Face Linear Extrusion Panel Type D by Sterling.

2.3.6 Chillers and Boilers

For purposes of modeling the alternative mechanical systems, the standard chillers and boilers were used from Trace. Two electric centrifugal chillers piped in parallel with variable primary flow characteristics were chosen each with a 0.52 kW/ton rating. The boiler chosen in Trace was a gas-fired 83% efficient boiler.

2.3.7 Pumps

Pumps were sized using Bell and Gossett's VSCS model catalog. Based on the flow and head required for each system the motor horsepower was taken from the pump curves at an average efficiency of about 80%.

2.4 Procedure and Calculations for Alternative Designs

The following sections give the details in designing the alternative systems. The procedure and calculations of the five base systems were described in this section. The fourth floor CAV system design will be described first. Based on the energy consumption of the fourth floor with and without an economizer the main building systems for the other floors will be analyzed with a waterside economizer.

System 1: VAV with DX coil and electric heating coils
System 2: VAV with chilled water and hot water coils
System 3: DOAS/VAV with DX coils and electric heating coils
System 4: DOAS/VAV with chilled water and hot water coils
System 5: DOAS/Radiant with chilled water and hot water coils

2.4.1 New Space Designations

For simplicity of analysis, new spaces designations were created. The general idea of serving five different zones was kept; however, they were squared off from the center of the building so that there are four basic quadrants along with the south wing section. The cardinal directions were used in the designations, which will make it easier to know what unit is serving what space once familiarity sets in with the orientation of the building. “FL-1 NE” is referring to the equipment serving the northeast corner of the first floor. Use Figure 2.4F and this key to correspond with the analysis ahead in this report.

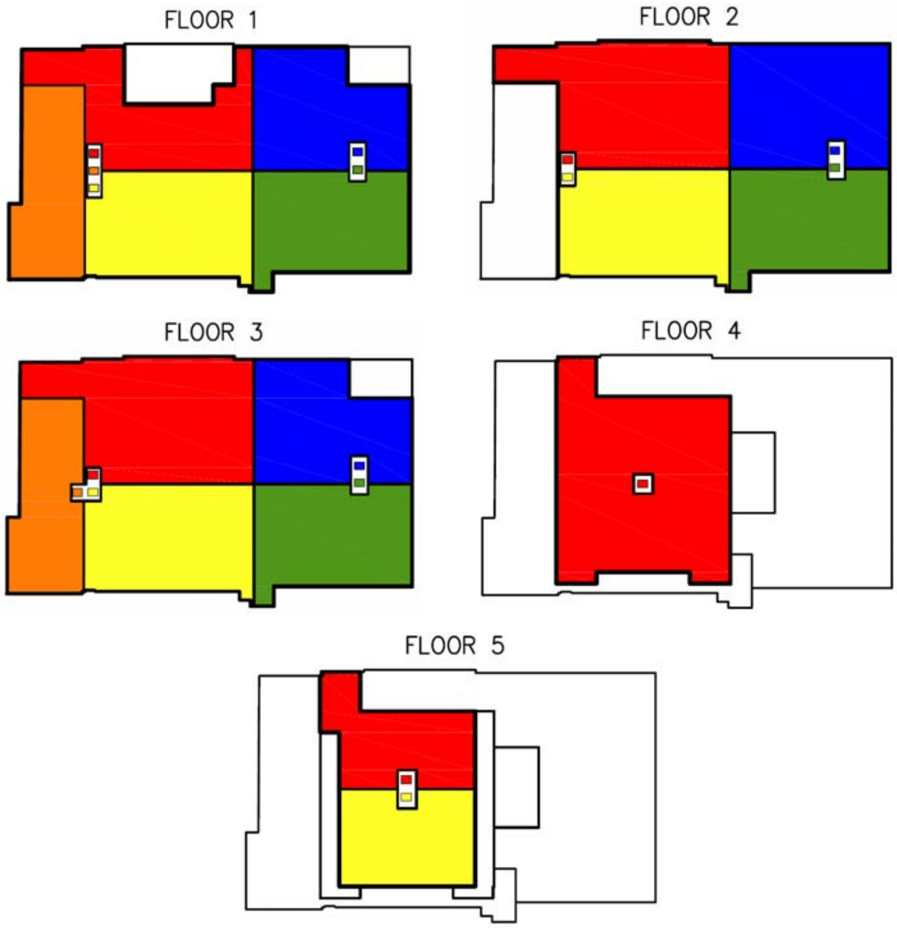

Figure 2.4A. New Service Designation for AHUs.

There will be 16 VAV air handling units in the alternative designs. There will be two DOAS air handling units: One serving the north section and one serving the south section.

2.4.2 Space Loads

Each system was modeled in a separate file using the software Trace by HVAC manufacturer Trane. Spaces were set up in the "Rooms" section of this program according to the floor area served by each air handling unit, the exterior wall, window, and roof areas, the internal load of the building, and other space properties. Once the simulation of the spaces was run, space loads were given. The simulations with VAV as an airside system all had the same space loads because the air handling units where included in the lighting/equipment internal loads. The simulation for the DOAS-Radiant model had slightly smaller space loads because the internal lighting/equipment loads were reduced without the air handling units on each floor. The following tables give each space heating and cooling peak load used in the simulations and calculations in the next sections. It should be noted that the sensible load in winter is a positive value, meaning that cooling is necessary in the winter.

	Cooling		Heating
	Sensible	Latent	Sensible
FLOOR 1			
FL-1 NE	517725	20000	346867
FL-1 NW	480276	20000	360694
FL-1 SE	510438	20000	357927
FL-1 SW	428994	20000	386150
FL-1 T	540264	20000	348706
FLOOR 2			
FL-2 NE	577357	20000	423699
FL-2 NW	555183	20000	435325
FL-2 SE	581823	20000	450197
FL-2 SW	575225	20000	456172
FLOOR 3			
FL-3 NE	582810	20000	384254
FL-3 NW	609033	20000	385486
FL-3 SE	579808	20000	440467
FL-3 SW	590399	20000	442701
FL-3 T	515911	20000	413353
FLOOR 4			
FL-4	17202746	0	16812040
FLOOR 5			
FL-5 E	401390	20000	224655
FL-5 W	418200	20000	226799

Table 2.4A. Space Heating and Cooling Loads with VAV as an Airside System [Btu/h].

	Cooling		Heating
	Sensible	Latent	Sensible
FLOOR 1			
FL-1 NE	517725	20000	346867
FL-1 NW	480276	20000	360694
FL-1 SE	510438	20000	357927
FL-1 SW	428994	20000	386150
FL-1 T	540264	20000	348706
FLOOR 2			
FL-2 NE	577357	20000	423699
FL-2 NW	555183	20000	435325
FL-2 SE	581823	20000	450197
FL-2 SW	575225	20000	456172
FLOOR 3			
FL-3 NE	582810	20000	384254
FL-3 NW	609033	20000	385486
FL-3 SE	579808	20000	440467
FL-3 SW	590399	20000	442701
FL-3 T	515911	20000	413353
FLOOR 5			
FL-5 E	401390	20000	224655
FL-5 W	418200	20000	226799

Table 2.4B. Space Heating and Cooling Loads with Radiant Panels as the Parallel System [Btu/h].

2.4.3 Air Properties for Designs

All simulations and calculations were done using the same design air temperatures. The design outdoor air temperatures and properties for heating and cooling based on weather for Philadelphia, PA were taken from ASHRAE Fundamentals. The room air and supply air temperatures and properties were taken as typical design procedure for commercial office buildings. For the DOAS, the supply air temperature was decided to be 45F previously in this report. The following designations are used in Table 2.4C: OA—Outdoor Air, RA—Room Air, SA—Supply Air, DB—Dry Bulb Temperature, RH—Relative Humidity, WB—Wet Bulb Temperature, h—Enthalpy, rho—Density, HR—Humidity Ratio.

Properties	Units	Heating		Cooling		
		OA	RA	OA	RA	SA
DB	F	11.0	70.0	92.0	75.0	55.0
RH	%	30.0	50.0	47.0	50.0	80.0
WB	F	7.2	58.6	75.0	62.4	51.6
h	Btu/lb	3.11	25.3	38.4	28.2	21.1
rho	lb/cf	0.070		0.084		
HR	gr/lb	2.8	54.5	113.5	64.0	52.0

Table 2.4C. Air Properties for Design VAV Conditions.

2.4.4 CAV System

The fourth floor was modeled as a constant volume system. The space loads, room air conditions, and supply air conditions are needed to find the required air flow for the space.

$$CFM_L = \frac{Q_L}{0.68 \cdot (W_{RA} - W_{SA})} \qquad CFM_S = \frac{Q_S}{1.08 \cdot (T_{RA} - T_{SA})}$$

Using the latent load and sensible load equations above, the flow rate can be found. There is no latent load in the space because it is used as a data center and people will not occupy the space for more time than maintenance requires. If there is a case where it is occupied for more than a short time, outdoor air is still supplied based on the floor air requirement. The internal load of the space was 100W/SF which was recommended by a faculty consultant. This is a huge number which can be seen by the sensible load in Table 2.4D.

Floor 4 Design	
Sensible Load	17202.75 Mbh
Supply Air Quantity	799423 CFM
Outdoor Air Quantity	3000 CFM

Table 2.4D. Peak Cooling Design Floor 4 CAV.

Floor 4 was simulated 5 different ways, all with CAV as the airside system. The original DX packaged units were modeled first. The second set of simulations involved the use of an electric centrifugal chiller with an electric boiler and with a gas-fired boiler. The last set of simulations included the use of the electric centrifugal boiler with an economizer and electric boiler and with an economizer and a gas-fired boiler.

The total system capacity is about 1550 tons. By the results of this analysis, it was decided that the systems being analyzed for the open office space will all be modeled with a waterside economizer.

2.4.5 VAV with DX coils and electric heating coils — System 1

A model of the existing system was built in order to have a comparison for the models of the alternative systems. Loads from the model were used to find the supply air quantity. The supply air requirement for each space was used to size the air handler. The total load served by the air handlers were used to size the cooling tower. Heating is provided by parallel fan powered boxes.

Required for Calculations

Space Sensible Loads	Occupancy	Room Dry Bulb Temperature
Space Latent Loads	Floor Area	Supply Dry Bulb Temperature

Calculation Procedure

Cooling Supply Air Quantity

Cooling is needed in both summer and winter for the SDPAH building. In both cases, the required outdoor air quantity should be calculated based on ASHRAE Standard 62.1. For open office space 5CFM/person and 0.05CFM/SF is required. The constant 0.8 is used as a

$$CFM_{vent} = \frac{0.06 \cdot Area + 5 \cdot People}{0.8}$$

This value will be used if it is more than the outdoor air required to satisfy the latent load. The outdoor air required to satisfy the latent load is determined by the following equation:

$$CFM_L = \frac{Q_L}{0.68 \cdot (W_{RA} - W_{SA})}$$

CFM_L is the outdoor air quantity, Q_L is the latent load in Btu/hr, W_RA is the humidity ratio of the room air, and W_SA is the humidity ratio of the supply air. If the quantity required to satisfy the latent load is larger then it is supplied to the space. The rest of the space load, the sensible load, is satisfied by the air quantity calculated by the following equation:

$$CFM_S = \frac{Q_S}{1.08 \cdot (T_{RA} - T_{SA})}$$

The outdoor air and the air that satisfies the sensible load combined is the total supply air to the space. The full spreadsheet of space loads and required supply air quantities is found in Appendix B. This supply air quantity is used to size the air handling unit. A sample selection is provided in Appendix B. It should be noted that during the summer the room air temperature is 75F and humidity ratio is 64 and during the winter the room air temperature is 70 and humidity ratio is 54.5. During the winter a lower sensible load must be met and there is no latent load to be met. The lack of a latent load means that the minimum required outdoor air quantity based on Standard 62.1 must be supplied to the space.

The final cooling capacity for System 1 is 1150 tons.

Sample Calculation for Summer Cooling for FL-1 NE.

CFM_vent = (0.06*Area + 5*People)/0.8

CFM_vent = 0.06*25000 + 5*90 = 1950CFM

CFM_L = Q_L/(0.68*(W_RA-W_SA))

CFM_L = 18000/(0.68*(64-52)) = 2206CFM

*2206CFM controls

CFM_S = Q_S/(1.08*(T_RA-T_SA))

CFM_S = 625550/(1.08*(75-55)) = 28961 CFM

CFM_total = CFM_L + CFM_S = 2206 + 28961 = **31167CFM**

2.4.6 VAV with chilled water and hot water coils — System 2

The source of cooling and heating for the VAV system with a central water plant is chilled and hot water. The procedure for finding the required supply airflow for the DX-electric units should be used to find the supply airflow for air handling units being provided chilled and hot water. Because the airflow is the same and the standard face velocity is the same (500FPM) for both cases, the same minimum face area is required. Although the size and flow rate for the coils are not necessary inputs for Trace to do an energy analysis, they are needed for cost analysis. Carrier's Air Handling Unit Builder was used to find the sizes of the coils based on the load requirements, airflow, and the entering air dry bulb and wet bulb temperatures. For simplicity in all selections the entering dry bulb and wet bulb temperatures in the summer were assumed based on mixing conditions to be 76.2F and 63.3F, respectively. In the winter entering dry bulb and wet bulb temperatures were assumed to be 65.7F and 55.3F, respectively. The mixing equation used to find these temperatures was based on the ventilation outdoor air mixing perfectly with the return air from the space.

$$T_{Mixed} = \frac{T_{OA} \cdot CFM_{OA} + T_{RA} \cdot CFM_{RA}}{CFM_{SA}}$$

See Appendix B for full results of the mixing calculations.

The final equipment size for System 2 is 1150 tons, the same as System 1.

2.4.7 DOAS / VAV — System 3 and System 4

SEMCO's packaged energy recovery systems give the choice of cooling with DX or chilled water coils and heating with electric or hot water coils. The load satisfied by the DOAS must be known to do calculations and equipment selection for the parallel system.

Required for DOAS Calculations

Sensible Load	Supply Air Conditions	Room Air Conditions
Latent Load	Outdoor Air Conditions	Ventilation Airflow

Enthalpy Wheel Sensible & Latent Effectiveness

Calculation Procedure

<u>DOAS Cooling Conditions</u>

The first calculation that should be done is the required outdoor airflow. This should be based on ASHRAE Standard 62.1 and on the latent load (Refer to System 1 of the Procedure and Calculations section for more details on these calculations). The larger of the two values should be used in the DOAS calculations. The supply airflow is also what is used to select a SEMCO air handling unit. A sample selection is provided in Appendix B for the North DOAS air handling unit.

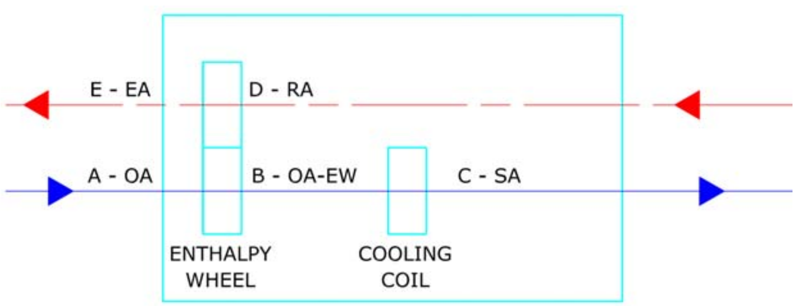

Figure 2.4B. DOAS Air Handling Unit State Points.

Figure 2.4B shows the different state points within the DOAS air handling unit used for the calculation procedure. Outdoor air enters the enthalpy wheel at summer conditions, 92F DBT and 75WBT, an ASHRAE design guideline. From this we can get the different properties

of the outdoor air. Supply air conditions were set at 45F and saturated in the previous discussion and the room air conditions are 75F dry bulb and 50% relative humidity. Table 2.4E gives the conditions of the outdoor, supply, and room air.

Property	A - OA - Outdoor Air	C - SA - Supply Air	D - RA - Room Air
DBT [F]	92	45	75
WBT [F]	75	45	62.4
% RH	47	100	50
W [g/lb]	113.5	44	64
h [Btu/lb]	38.4	17.6	28.2

Table 2.4E. Summer DOAS Outdoor, Supply, and Room Air Conditions.

The temperature and humidity ratio at state B governs the effectiveness of the enthalpy wheel. The temperature and humidity ratio at state B can be calculated using the enthalpy wheel effectiveness. The sensible effectiveness can be calculated using the following equation:

$$\varepsilon_{SEN} = \frac{DBT_A - DBT_B}{DBT_A - DBT_D}$$

The effectiveness is assumed to be 0.80, therefore, this equation can be solved for the dry bulb temperature at state B (DBT_B).

$$DBT_B = DBT_A - \varepsilon_{SEN} \cdot (DBT_A - DBT_D)$$

The humidity ratio at state B can be calculated using a similar equation.

$$W_B = W_A - \varepsilon_{LAT} \cdot (W_A - W_D)$$

Building space FL-1 NE will be used to illustrate these calculations.

DBT_B = DBT_A – Eff*(DBT_A – DBT_D)

DBT_B = 92 – 0.8*(92 – 75) = 78.4F

W_B = W_A – Eff*(W_A – W_D)

W_B = 113.5 – 0.8*(113.5 – 64) = 73.9 g/lb

The enthalpy wheel reduces the humidity of the outdoor air by 34.9%.

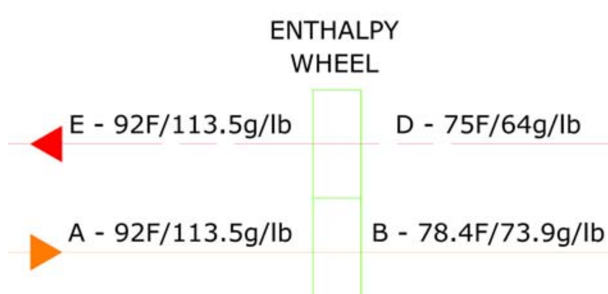

Figure 2.4C. Enthalpy Wheel Effectiveness – Summer.

The dry bulb temperature and humidity ratio reduction at point B will now determine the total cooling coil load. The required outdoor airflow should be used in the following series of equations to calculate the load across the cooling coil.

$$Q_{CC,S} = 1.08 \cdot CFM_{OA} \cdot (DBT_B - DBT_C)$$

$$Q_{CC,L} = 0.68 \cdot CFM_{OA} \cdot (W_B - W_C)$$

$$Q_{total} = Q_{CC,S} + Q_{CC,L}$$

Q_cc,s = 1.08*1950*(78.4 – 45) = 70.340MBH

Q_cc,l = 0.68*1950*(73.9 – 44) = 39.647MBH

Q_total = 70.340 + 39.647 = 109.988MBH = 9.17tons

The total and sensible load across the cooling coil is used to select a coil with Carrier's AHU Builder.

In order to find the parallel system cooling capacity the DOAS cooling capacity must be calculated.

$$Q_{SA} = 1.08 \cdot CFM_{OA} \cdot (T_D - T_{,C})$$

Q_SA = 1.08*1950*(75 – 45) = 63.180MBH

This is where the room sensible load becomes important. The DOAS has satisfied a certain portion of the sensible load (Q_SA) and this can be subtracted from the room sensible load to find the required capacity of the parallel system.

$$Q_{parallel} = Q_{SEN} - Q_{SA}$$

Q_SEN = 625.550MBH

Q_parallel = 625.550 – 63.180 = 562.370MBH

<u>DOAS Heating Conditions</u>

Table 2.4F gives the outdoor, supply, and room air conditions for heating design. Remember, cooling is needed so there is no need for a heating coil within the DOAS unit.

Property	A - OA - Outdoor	C - SA - Supply Air	D - RA - Room Air
DBT [F]	11	45	70
WBT [F]	7.2	45	58.6
% RH	30	100	50
W [g/lb]	2.8	44	54.5
h [Btu/lb]	3.11	17.6	25.3

Table 2.4F. Winter DOAS Outdoor, Supply, and Room Air Conditions.

The same basic equations are used for heating conditions.

$$DBT_B = DBT_A - \varepsilon_{SEN} \cdot (DBT_A - DBT_D)$$

$$W_B = W_A - \varepsilon_{LAT} \cdot (W_A - W_D)$$

Building space FL-1 NE will be used to as an example for the heating calculations.

DBT_B = DBT_A – Eff*(DBT_A – DBT_D)

DBT_B = 11 – 0.8*(11 – 70) = 58.2F

W_B = W_A – Eff*(W_A – W_D)

W_B = 2.8 – 0.8*(2.8 – 54.5) = 44.2 g/lb

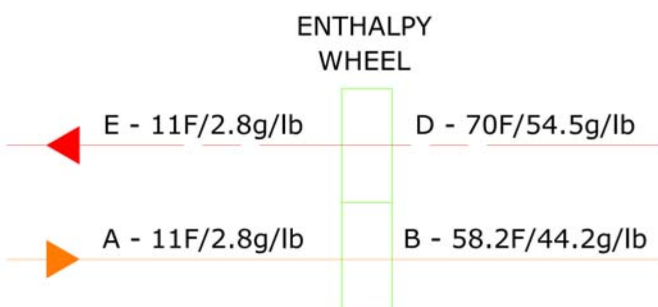

Figure 2.4D. Enthalpy Wheel Effectiveness – Winter.

Appendix B gives a sample selection of the North DOAS air handling unit.

Parallel VAV

The design procedure for System 1 and 2 in the Procedure and Calculations section is used to calculate the supply air required for the parallel VAV system. The supply air for System 3 and 4 is less than that supplied by the existing case. The percent reduction can be found in Table 2.4G. Using the new supply air quantity, McQuay units were selected for the parallel VAV application. A sample selection can be found in Appendix B.

Space	Supply Air Existing	Supply Air System 3 / 4	Percent Reduction
FLOOR 1			
FL-1 NE	28000	25200	**10.00%**
FL-1 NW	28000	25200	**10.00%**
FL-1 SE	28000	23750	**15.18%**
FL-1 SW	28000	23750	**15.18%**
FL-1 T	28000	27300	**2.50%**
FLOOR 2			
FL-2 NE	31500	28500	**9.52%**
FL-2 NW	31500	28500	**9.52%**
FL-2 SE	31500	29000	**7.94%**
FL-2 SW	31500	29000	**7.94%**
FLOOR 3			
FL-3 NE	35000	30000	**14.29%**
FL-3 NW	35000	30000	**14.29%**
FL-3 SE	35000	29450	**15.86%**
FL-3 SW	35000	29450	**15.86%**
FL-3 T	35000	27200	**22.29%**
FLOOR 5			
FL-5 E	28000	20700	**26.07%**
FL-5 W	28000	20700	**26.07%**

Table 2.4G. Supply Air Reduction for Parallel VAV System.

The total supply air for the DOAS/VAV system is that provided by both the DOAS and VAV units. These two air streams are mixed after both have been conditioned. The following equation was used to find the supply air temperature for the mixed air:

$$SAT_{Mixed} = SAT_{VAV} \cdot \frac{CFM_{VAV}}{CFM_{TOTAL}} + SAT_{DOAS} \cdot \frac{CFM_{DOAS}}{CFM_{TOTAL}}$$

SAT—Supply Air Temperature, CFM—Cubic Feet per Minute

See Appendix B for the mixed air calculations.

The final equipment size required by SDPAH for System 3 and 4 is 1236 tons.

2.4.8 DOAS/Radiant with chilled water and hot water coils — System 5

Because the latent load of the space remains the same as the systems change, the dedicated outdoor air design procedure remains the same for all three dedicated outdoor air systems.

Required for Radiant Panel Calculations

Calculated Parallel System Capacity | Panel Entering Water Temperature

Panel Sensible Load Absorption Capacity

Calculation Procedure

Panel inlet water temperature must be at least one degree above room dew point temperature. Room design dew point is 55F, therefore the water temperature must be at least 56F. Using Sterling's design procedure, the estimated temperature rise is 10F—leaving water temperature is 66F. Half of the temperature rise is added to the entering water temperature to find the mean water temperature, which is 61F. The difference between the room dry bulb temperature and the mean water temperature, 14F, is used to find a set of panels that will work with out design. If we restrict the ceiling to 50 percent coverage by radiant panels then the capacity of the panel per square foot can be calculated.

$$Area_{panels} = \frac{Area_{ceiling}}{2}$$

$$Q_{perSF} = \frac{Q_{parallel}}{Area_{panels}}$$

Using FL-1NE as an example:

Q_perSF = 436725Btu/hr ÷ 12500SF = 34.938Btu/hr per SF

Panel selection C at 35Btu/SF will work. Using 2 by 4 panels each panel will remove 280Btu/hr. Dividing the total parallel load by the capacity of the panel will give the number of panels for the space.

Panels = 436725Btu/hr ÷ 280Btu/hr / Panel = 1560 Panels

Optimizing the pressure drop, capacity, and flow rate for a circuit will give the absorption, pressure drop, and flow per circuit. See Appendix B for the results of this optimization. In space FL-1 NE, 1GPM was chosen as the flow rate because the pressure drop for higher flow rates were more than tripled that of the 1GPM flow. This corresponds to a pressure drop of 6.8 FT WG per circuit and absorption of 5000Btu/hr per circuit. The number of circuits can be calculated by dividing the total parallel load by 5000Btu/hr. Eighty eight (88) circuits will be needed for this space. Now the radiant ceiling panels can be laid out.

A circulation pump will be assigned to each space, increasing the required pumping energy. To counterbalance this increase in energy, the fan energy will be decreased tremendously since there are only two DOAS air handling units.

The final equipment size required by SDPAH for System 5 is 840 tons.

*All system calculation spreadsheets can be found in Appendix B.

2.5 Results and Recommendations

An analysis on the fourth floor CAV system was completed first to find the savings potential of an electric centrifugal chiller with and without an economizer and to find the savings potential of a gas-fired boiler compared to an electric boiler. The result of the analysis on the fourth floor data center helped to do analysis on the rest of the building spaces. A final recommendation will be made based on energy usage, system costs, lost rentable space, and constructability.

2.5.1 Floor 4 Results

The fourth floor was simulated as a traditional CAV airside system with different waterside systems. The purpose of this analysis was two-fold. The first purpose was to find an energy efficient system that can be useful for the fourth floor's application. The second purpose was to choose what systems the office spaces would be modeled with based on the results from Floor 4's simulations. The full results are detailed in Appendix B. The total energy savings of using a central chilled water and hot water plant can be seen in Chart 2.5A. The base case is the packaged McQuay SWP units. Not only does this system use a lot of energy, 6074637.7 kBtu/yr, but it would require a large amount of floor space to put the units. For data centers, usually air is pushed through underfloor air distribution. The floor panels are perforated and air can be circulated within the electrical and data equipment. Special structural designs should be considered for this system.

From Chart 2.5A, it can be seen that a chiller saves about 350 kBtu per year. Implementing a gas-fired boiler increases this amount by about 6 kBtu per year and using a plate and frame heat exchanger adds another 6 kBtu per year of savings. For the analysis of the office space, it was decided that an electric water cooled chiller with an economizer and a gas-fired boiler would be the basis of comparison to the existing system on the water side of the system.

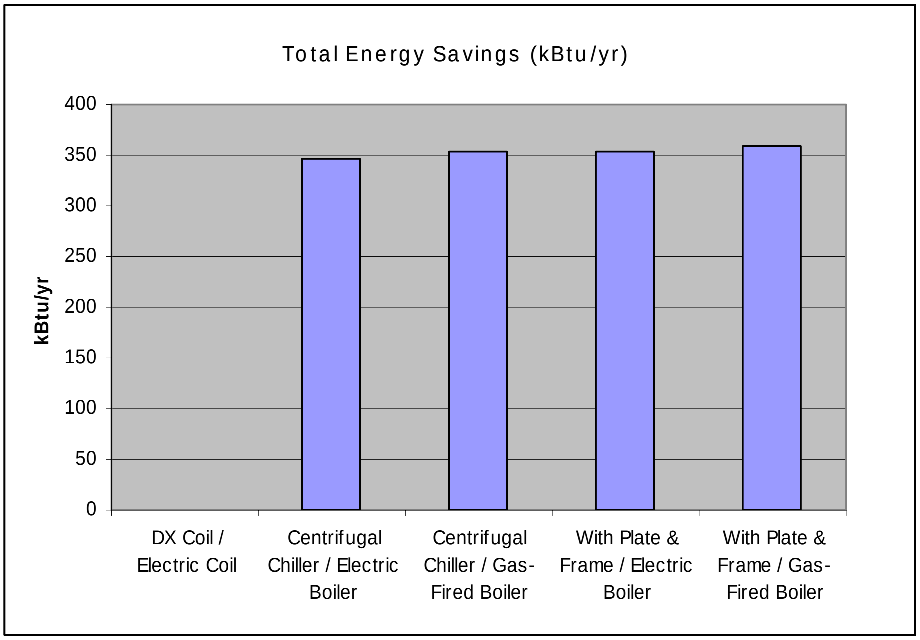

Chart 2.5A. Total Energy Savings Using an Electric Centrifugal Chiller—Floor 4.

2.5.2 Energy Usage and Cost

A good argument for choosing a particular system is yearly energy cost. The total source energy consumed per year by each system is given in Chart 1. "Source" means the part of the mechanical system using energy. The energy represented in this chart includes all forms of energy used in the systems (electric, gas, and water). The values are in MBtu/year—million Btu/hr per year. Compressor energy, the tower fan energy, and the condenser pump energy are all included in the "Primary Cooling" category. It was expected that the DX/Electric systems would use more energy, thus cost more to operate. It was unknown exactly how much more energy the DX/Electric systems would require. System 1—VAV with DX and electric coils and system 3—DOAS/VAV with chilled and hot water coils utilize the largest kW per ton which is why they use the most energy in the end. The systems using VAV as the primary system or as a parallel system require the most fan energy and the radiant system uses the most pumping energy.

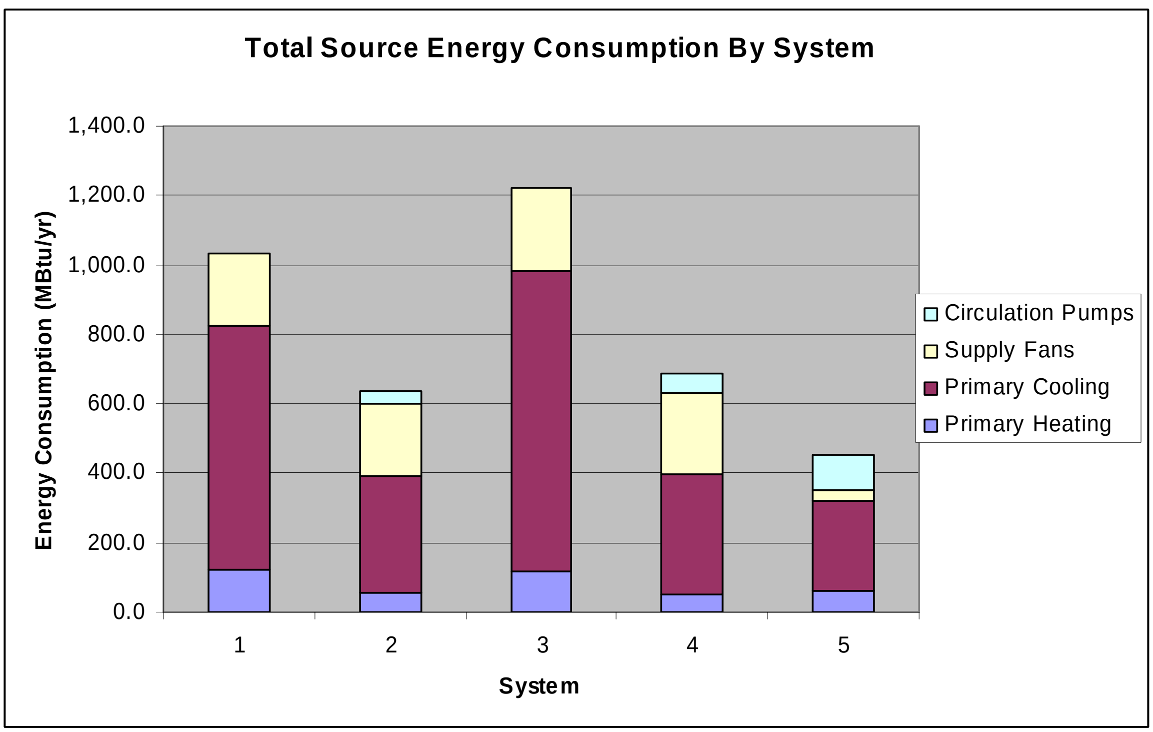

Chart 2.5B. Total Source Energy Consumption by System.

The amount of piping due to the radiant panels is the main reason for this increased piping energy. In general, the high energy consumption is due to the large amount of cooling needed in the summer. During the winter, the chiller can use the economizer cycle to provide free cooling. In general, using a central chilled water plant saves on energy consumption. If the decision of what system should be used was solely based on energy consumption it would be easy—the DOAS/Radiant system would be chosen, however, most of the time first cost is the deciding factor for owners. For full energy consumption results, see Appendix B.

Energy usage is directly related to energy cost. Table 2.5A shows the energy cost per year in US dollars for each system. These costs are based on energy rates from PECO—Philadelphia Electric Company and Philadelphia Water Department (See Appendix B). Rates as well as more detailed yearly cost information can be found in Appendix B. It is not surprising that electricity is the most costly of the energy sources. By choosing a central chilled and hot water plant, savings compared to the all electric systems can be from $287,603 to $634,504 per year. For heating, the DOAS/Radiant system was the highest in cost which should be expected since low temperature air is supplied directly to the space unlike the DOAS/VAV system where it was mixed beforehand. Still, the radiant system holds for the lowest yearly operating cost while all electric systems are highest in operating cost.

	1	2	3	4	5
Electric	$1,697,433	$1,463,243	$1,793,130	$1,495,847	$1,180,379
Gas	$0	$37,455	$0	$34,661	$41,100
Water	$152,379	$140,887	$170,704	$145,723	$107,851
Total	**$1,849,812**	**$1,641,585**	**$1,963,834**	**$1,676,231**	**$1,329,330**

Table 2.5A. Yearly Operating Cost Based on Energy Consumption.

2.5.3 First Cost

The first cost and yearly energy consumption cost usually go hand in hand. Saving money on energy in the future may cause an owner to spend a bit more up front on first cost. A number of sources were used to find the first cost of the different systems analyzed in this report. Quotes from the actual first cost of the existing system were modified to provide first cost for the alternative systems. RS Means was used to find cost of equipment that was not used in the existing system and some online resources were used to compliment RS Means. Results are summarized in Table 2.5B. With the use of DOAS/Radiant, the chiller size is reduced by more than 300 tons from the DX original system. This causes great reductions in chiller cost and when only considering the alternatives with a water cooled chiller, the option of the dedicated outdoor air system and radiant panels is lest costly than the alternatives with VAV as an airside system. The systems using packaged DX units are the least expensive, but their energy use is very high. In this case, for a savings in energy use and a slightly higher first cost, the DOAS/Radiant system should be chosen.

	VAV		DOAS/VAV		DOAS/Radiant
	DX-Electric	CHW-HW	DX-Electric	CHW-HW	CHW-HW
Required Chiller Size [tons]	1150	1150	1236	1236	840
Condenser Water Flow	4000	3500	4300	3750	2500
Chilled Water Flow		2500		2800	2000
Hot Water Flow		2250		2250	2250
AHU	$700,000	$1,050,000	$660,000	$911,000	$81,000
Chiller		$460,000		$495,000	$340,000
Pump		$38,000		$42,000	$30,000
CT/Fan	$120,000	$105,000	$130,000	$115,000	$75,000
Pump	$60,000	$52,000	$65,000	$55,000	$38,000
Boiler		$38,000		$38,000	$38,000
Pump		$34,000		$34,000	$34,000
Resistance Heaters	$50,000		$50,000		
VAV boxes	$310,000	$390,000	$340,000	$430,000	
Radiant Panels					$2,750,000
Pumps					$30,000
Piping					$40,000
Baseboard Heaters					$120,000
CDW/CHW/HW Piping (Primary)	$290,000	$195,000	$320,000	$250,000	$144,000
Ductwork (Mains)	$2,000,000	$2,000,000	$2,000,000	$2,000,000	$500,000
Totals	$3,530,000	$4,362,000	$3,565,000	$4,370,000	$4,220,000

Table 2.5B. First Cost of Base and Alternative Mechanical Systems.

2.5.4 Emissions

Exelon Corporation, the parent company of PECO, generates electricity primarily with the use of nuclear power. Some coal, oil, natural gas, and hydro power also help to generate the electricity provided by Exelon Corporation. The emissions based on the electricity consumed for each system can be calculated by knowing the percentages of the different resources used to make electricity. Using nuclear power as a primary source for producing electricity helps save on emissions greatly as it does not produce any pollutants. Keeping the use of coal to produce electricity to a minimum helps too since it produces the most NOx. Exelon Corporation's 2004 quantities were used to find the amount of particulates, SO2, NOx, and CO2 produced by electricity consumed by each system summarized in Table 2.5D. (The detailed result of these calculations can be found in Appendix B.) It is not surprising that the amount of emissions is proportional to the amount of electricity consumed and therefore System 3 produces the most emissions.

Coal	6.0%
Oil	4.0%
Nat. Gas	1.0%
Nuclear	88.0%
Hydro/Wind	1.0%
All	100.0%

Table 2.5C. Generation Fuel Mix.

System	lbm Pollutant			
	Particulates	SO2	NOx	CO2
1	20808.1	244101.2	143723.3	44685834.0
2	17974.1	210855.6	124148.8	38599804.1
3	**21969.9**	**257730.3**	**151747.9**	**47180815.2**
4	18322.5	214942.7	126555.2	39348004.3
5	14400.1	168929.1	99463.1	30924628.5

Table 2.5D. System Emissions due to Electricity Consumption.

NOx is also produced from the burning of natural gas in a boiler. Since a boiler was used as the heating source in the central hot water plant, NOx calculations can be done based on the NOx emissions for the boiler. The NOx emission from the boiler is 0.144lbm/year per 1 MMBtu, which is quite small compared to the amount given off by the use of electricity.

2.5.5 Lost Rentable Space

The School District of Philadelphia occupies the entire building and therefore any space saved from an alternative system is theirs to use. The alternative systems using VAV as an airside system (Systems 2-4) will require the same mechanical room space that the existing system requires because the amount of air handling units were not reduced. In fact, Systems 3 and 4 with the DOAS, will require more mechanical room space for the new air handling units. The DOAS/Radiant system, System 5, will require the least amount of mechanical room space. For the two DOAS air handling units, one can be placed on the fifth floor and one can be placed on the third floor, North wing. Small pump rooms will be required on each floor for circulation of the cool water in the radiant panels but these rooms will not need nearly as much space as the air handling units do. There will be two 100 GPM pumps in the same location as the air handling units. These require no more than 100 SF of space which is about 1/12 of what the AHUs require.

2.5.6 Integration

The SDPAH is located in an existing building. This calls for special attention to the ability to construct the system. The structural integrity of the building must be checked. Constructing a chiller and boiler on the roof transfers more load to the structural members, the columns, beams, girders. The members must be checked to make sure they are strong enough to transfer the load to the ground. The existing building at 440 North Broad poses an advantage to a typical speculative office building in that it was originally built as a printing facility with floor live loads of 125 PSF. Integration of the structural system will be further investigated in the Breadth section of this paper (3.0).

Constructing the system is another issue that must be addressed. Placing chillers on the roof would require a crane. The location of this equipment is important to the surrounding community. Will it cause disturbances in everyday life of downtown Philadelphia? Also, moving material around within the building is an important concern. The freight elevators will be a great assistance in moving material vertically from floor to floor. These topics too will be discussed in the Breadth section of this paper.

2.5.7 Recommendation

Although the existing system is not expensive and can be easily installed, the DOAS/Radiant with water cooled chiller and waterside economizer provides a great savings in yearly operating cost and would be recommended to an owner if he or she desired a central water plant. At around 1.3 million dollars to operate yearly, the DOAS/Radiant system offers a payback period of about 1 year, 4 months relative to the existing system. Comparatively to the VAV with central chilled water and hot water, the pay back would be 6 months. In either case, the energy savings is worth the first cost.

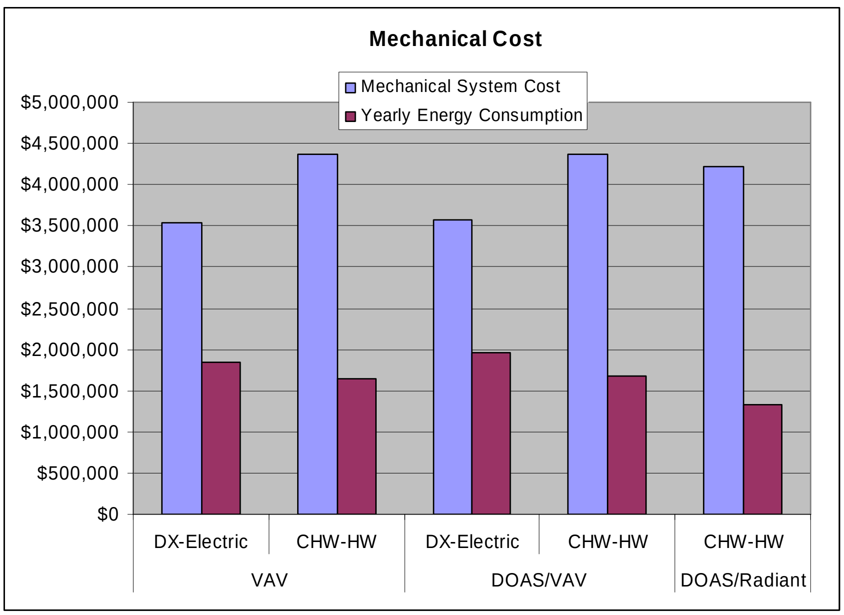

Chart 2.5C. Mechanical System First Cost/Yearly Operating Cost.

A possible consideration for further investigation is using air cooled chillers with and without an airside economizer. Air cooled chillers are smaller than water cooled, usually available up to about 400 tons so they would have to be used in parallel. The air cooled chillers are typically more expensive than water cooled, but save first cost without having to buy a cooling tower.

	VAV		DOAS/VAV		DOAS/Radiant
	DX-Electric	CHW-HW	DX-Electric	CHW-HW	CHW-HW
Mechanical System Cost	$3,530,000	$4,362,000	$3,565,000	$4,370,000	$4,220,000
Yearly Energy Consumption	$1,849,812	$1,641,585	$1,963,834	$1,676,231	$1,329,330

Table 2.5E. Cost Summary.