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Thesis Introduction

The Johns Hopkins Hospital in Baltimore Maryland is currently nearing construction on a new building. The building is a medical office building (abbreviated as MOB in this report) for use as doctors' offices and hospital faculty offices. Currently the building uses a system picked for its low first cost. The system is a VAV system with packaged rooftop DX units, and all electric terminal reheat. The purpose of this thesis is to evaluate the MOB for utilization of a VAV system with chilled water cooling coils as well as the use of a DOAS system paired with active heated/chilled beam parallel system.

After the building details, as well as the existing system ventilation and energy analysis, the operation and merits of both standard VAV systems and DOAS systems is reviewed. The main calculation process for the new mechanical system for the MOB is shown in the equipment selection section. Analysis of the equipment for indoor air quality performance as well as a first cost and operating cost analysis round out the main mechanical depth work.

The report is closed with the breadth work. The first breadth topic is a cost impact study for the reduction in electrical distribution equipment without the DX units and all electric reheat system. The second is a constructability review for connection of the MOB with the remote source chilled water and steam generated on the far side of the block containing the MOB.

Section 1

Building Introduction

1.1 Purpose

The MOB is a new medical office building. The bottom two floors are tenant space consisting of examination rooms, non-invasive care rooms, such as dialysis rooms, blood infusion rooms and various ancillary spaces. The top two floors are mainly faculty space for Johns Hopkins Hospital staff as well as conference rooms.

1.2 Location

Along with the rest of the main Johns Hopkins Hospital campus, the MOB is located in Baltimore Maryland. It's situated along N. Wolfe St. just south of Orleans St. It's newly developed location makes it the new south eastern corner of the JHH campus, and puts it in proximity to a parking garage, a loading dock, the South of Orleans Energy Plant and an abandoned project.

1.3 Size

88,000 sq ft equally divided between four floors, with the basement being below grade.

1.4 Project Team

Architect – ZGF, Zimmer Gunsul Frasca Partnership
CM firm - Atlantic Builders
MEP firm – Leach Wallace Associates, Inc.
Structural – Columbia Engineers

1.5 Dates of Construction

June 2005 – Late March 2006

1.6 Project Delivery Method

Design-Bid-Build

1.7 Cost Information

Approximate project cost is 15.5 million.

Section 2

Existing Ventilation Analysis

2.1 Overview of Assumptions and Analysis

Each of the six existing rooftop units delivers mixed air at a10%OA ratio. The analysis of std. 62-2004 compliance took place at design air delivery values.

Because the MOB is somewhat of a specialized building, it contains many spaces not listed in ASHRAE std. 62-2004 table 6-1. The following table details their approximations as existing ASHRAE defined spaces.

Space Approximation Table

Category Approximated	MOB space name	occ rate/ 1000 sq ft	Rp (cfm/occ)	Ra (cfm/sq ft)
Office	exam, treatment, radiology, infusion, dialysis, files copy, work	5	5	0.06
main entry lobby	Lobby	7	7.5	0.06
corridor	Vest			0.06
pharmacy	Meds	10	5	0.18
reception	Waiting	30	5	0.06
conference	Consult	50	5	0.06
science laboratories	Lab	10	5	0.18

After the analysis in the Air Ventilation Spreadsheet, found in Technical Assignment 1, it was found that the original MOB design met ASHRAE std. 62-2004 guidelines. Below is listed the final conclusions on space ventilation demands from the MOB.

Zone Ventilation Conclusions

	AHU 1-2	AHU 3-4	AHU 5	AHU 6
SUM Vpz	33635	37765	17600	15040
Vpz*.1	**3364**	**3777**	**1760**	**1504**
SUM Vbz	2785	3043	863	972
Zp (max	0.249	0.247	0.236	0.243
Ev	0.9	0.9	0.9	0.9
Vou	**3095**	**3381**	**959**	**1081**
compliance	YES	YES	YES	YES

2.2 Ventilation Conclusions

At design conditions, the MOB satisfies ASHRAE std. 62-2004 requirements. However, one of the design goals of my thesis will be to purposely over ventilate many of the common spaces of the MOB as well as exam rooms. The logic here is that these rooms will often have contaminant generation of pathogenic viruses and germs from ill patients. An updated ventilation summary and comparison will be given later in the report.

Section 3

ASHRAE std. 90.1-2004 Energy Compliance

3.1 Background to ASHRAE std. 90.1-2004

Standard 90.1 is the general energy usage standard for ASHRAE. It is designed to encourage more efficient buildings. The method used to analyze the MOB was the building area method. This is a method whereby the building area is multiplied by an energy usage per unit area number. For the MOB this number was 1.1 W/sqft which is used for office buildings.

3.2 Energy Usage Conclusions

The MOB did not pass the building general electrical usage guidelines set out in ASHRAE std. 90.1-2004. The MOB electrical usage is based strictly on the number of circuits and average loading. The number may seem high, but even with an assumed load of only 8amps per circuit the building still is above compliance with std 90.1. The electrical usage totals also do not reflect the six packaged AHUs on the roof or the many electric reheat VAV boxes in the building.

Building General Energy Usage Chart

Area	Total Building Area	Exempted Area	Adjusted Area
	88,260	491	87,769

Allowed Power Use

Allowed power density as per Std 90.1-2004 Table 9.5.1 for Building Area Method

Office Use (W/ft^2)	Adjusted Area	Total Power Allowed
1.1	87,769	**96,546**

Actual Power Use	Circuits	Unit Amperage	Voltage	Power (Watts)
	44	15	277	**182,820**

Section 4

DOAS Supply Air Demands
vs.
VAV Supply Air Demands

4.1 VAV Supply Air Demands

The typical office HVAC application is currently a VAV set up. VAV stands for variable air volume. It works on the principal that air is supplied from the central air handling units at medium pressure to localized VAV boxes that regulate the amount of air each zone receives.

Cooling - The air, being at a temperature and humidity level suitable to remove both latent and sensible loads from the spaces based on a cooling application is modulated in accordance with zonal controls that monitor temperature and humidity. In the MOB, the AHUs supply air at 53 F with a wet-bulb temperature of 52 F.

Heating - Any heating needs are typically accomplished via either some sort of heating coils in the individual VAV boxes or in-room sensible heating such as baseboard heaters. The MOB has its heating needs satisfied via the individual VAV boxes which contain electrical resistance heaters.

Ventilation - Ventilation is delivered at design conditions from the supply air which is typically itself a mixture of recycled indoor air and outdoor air. In the MOB, at design conditions, the AHUs pull in 10% outdoor air each for a total of 11,400 cfm of outdoor air.

4.1.1 VAV airside Pros

-Satisfies cooling loads of space with only supply air.
-Relatively high supply cooling supply air temperatures reduce danger of cold drafts.
-With high volumes of supply air and filters installed particle contaminant levels fall quickly within space.

4.1.2 VAV airside Cons

-Supplies entire zone to the needs of the most demanding space within a zone.
-Low outdoor air percentage necessitates large volumes of supply air with high fan costs and increased size of ductwork.

4.2 DOAS Supply Air Demands

DOAS is an acronym for dedicated outdoor air system. Although its initial cost is often higher that standard VAV applications, it can often save both money and energy in the long run while delivering increased amounts of outdoor air to a space. In a DOAS system, the entire volume of supply air is non-recycled outdoor air. Despite this fact is has several critical differences from the 100% outdoor air systems that have been used in hospitals and buildings with sensitive security issues for years. In a 100% outdoor air system the air is supplied at similar parameters as the VAV system where a large volume of air is needed to remove sensible and latent loads from the space. DOAS does not approximate a standard VAV system in its supply air parameters. Supply air in a DOAS system is typically colder and therefore drier than the supply air of a VAV system. This means that the internal latent load demands as well as ventilation requirements are satisfied with a smaller volume of air. Since space sensible loads are still not satisfied with the small amount of air, a parallel cooling system, often chilled radiant panels or chilled beams are used to remove sensible load in excess of the supply airs removal capacity.

Cooling – The primary cooling purpose of DOAS supply air is to remove the entire latent load. It is important that the supply air be able to do this, because if the latent load is not removed it can cause problematic condensation on the parallel cooling equipment. To assure that the entire latent load is removed by the smaller volumes of supply air, the air itself is supplied at lower temperatures. The lower temperature of the supply air means that it is extremely low in moisture content and better able to remove humidity from the space. In the case of the MOB, air will be supplied at 45 degrees. Its moisture content is only 35 grains of moisture per pound of dry air as compared to the standard VAV supply air with 48 grains. This over doubles the supply airs latent removal capacity.

The second part of space cooling in a DOAS application is a parallel cooling system. Because the supply air volume is so low, it typically cannot successfully remove the sensible load. To remove excess sensible load, a parallel system is used within the individual spaces. Often times this parallel system is ceiling mounted chilled radiant panels. These panels approximately split the load removal between radiant heat transfer, which tends to improve thermal comfort, and convective heat transfer.

Parallel cooling in the MOB is achieved through a slightly different system called an active beam. A chilled beam is a device with extended heat transfer surfaces that increase its heat transfer capacity. An active chilled beam is a chilled beam that uses high induction nozzles coupled with the supply air to induce room air to flow past the heat transfer surfaces thereby increasing its convective heat transfer. The particular beams used in the MOB are active chilled and heated beams. They supply air via high velocity, high induction nozzles. The room air induced into the unit by the supply air, enters through a centrally located hydronic cooling or heating coil and then is redistributed to the room from the sides of the unit.

Heating – A secondary heating system is as important to a DOAS system as a parallel cooling system. This is because in a DOAS system the low supply temperatures create a danger of overcooling. Usually the systems used for parallel cooling cannot easily be made to provide heating. Chilled radiant panels and passive beams, if used to provide heat, tend to lead to stratification problems as the units themselves are located on the ceiling. However, in the MOB, the fact that the parallel equipment actively distributes air through induction means that heating can better be accomplished without worry of temperature stratification.

Ventilation – Ventilation in a DOAS system is improved over standard systems for that fact that all of the air supplied is outdoor air. Typically, the amount of supply air needed to remove the entire latent load is greater than the required air for ventilation. Therefore, DOAS systems almost always over ventilate the spaces they serve.

4.2.1 <u>DOAS airside Pros</u>

-Ventilation is greatly improved over standard systems.
-Smaller volume of supply air requires smaller HVAC equipment sizes as well as diminished duct sizes.

4.2.2 <u>VAV airside Cons</u>

-With lower volumes of supply air, some airborne particle contaminants may linger longer than with standard VAV applications.
- Without high induction diffusers, there is a danger of cold drafts in cooling conditions and temperature stratification under heating conditions.
-Parallel cooling system must be employed to meet sensible load requirements.

Section 5

HVAC Equipment Selection

Section 5.1

Current HVAC Equipment

5.1.1 System Basics

The MOB currently is served by six York packaged rooftop units using a DX cooling system. The units are rated at 21,000 cfm and 61 Tons of cooling each. These units were chosen primarily due of their low first cost. The AHUs are supported by a VAV system using fan powered boxes with all electric reheat coils. This system is easy to install since it has no plumbing associated with it and more importantly has a very low first cost compared to the other systems to be analyzed.

Current Design Supply Air – 114,036 cfm
This value determined from an individual fixture count.
Gross Cooling Capacity – 4,414 MBH (LWA specs)
Gross Sensible Capacity – 3,199 MBH (LWA specs)
Gross Latent Capacity – 1,215 MBH
Latent Removal at Design Airflow – 775 MBH
This value is based off of design air supply.
Determined by the following formula:

$$\boxed{Qs = 0.68 \cdot CFM \cdot (Gra - Gsa)}$$

Gra is 58Gr/lbma from the air parameters of 72F and 50%RH
Gsa is 48Gr/lbma from LWA specs
This represents 63% of system capacity for latent removal.
Gross Heating Capacity – 380 kW or 1,296,613 BTU/hr
This value is spread out amongst the 118 fan powered VAV boxes in the building.

5.1.2 Current DX AHU system electricity needs

Total Power
Two of the units are specified at 460V and 207 MCA
The other four units are specified at 460V and 184 MCA
Total Power = 2 * 460V * 207Amps/1.25 + 4 * 460V * 184Amps/1.25
The total maximum demand for all six rooftop units is 423 kW.
Part of this total it the 145.5 kW of the total fan energy
Coolant Circulation Power
Therefore the non-fan power consumption of the six rooftop units is 423kW-145.5kW
Total, non-fan, power consumption of rooftop units is 277.5 kW

Total AHU Electrical Power Use – 423 kW
Fan Power Use – 145.5 kW
Non-Fan Power Use – 277.5 kW
*These figures do not include the 380 kW maximum heating capacity.

Section 5.2

Alternative #1 Standard VAV System with Water Cooled AHUs

5.2.1 System Basics

The first alternative for the MOB is simply replacing the inefficient packaged DX air handling units on the rooftop with more efficient units that utilize the nearby available chilled water. The powered VAV boxes will also be analyzed for use with hot water heating coils instead of electrical resistance heating coils.

This system will be a standard VAV application just like the current system. For this reason the supply air volume and parameters will remain the same. Just as the original system, Alternative #1 will use 10% outdoor air.

After investigation a York Custom air handling unit was selected that will provide the needed supply air capacity with six units configured in the same way the current system delivers air. The new units have a maximum cfm of 22,500 cfm but equipped with the same fan and total system pressure drop as the original system their capacity is closer to 21,000 cfm.

Supply Air – 114036 cfm
Gross Cooling Capacity – 4,074 MBH

This Value determined from the following equation
Total Cooling = Number of Units * (Sensible Load + Latent Load)
Sensible Load = 1.08 * SCFM * (T mixed air – T coil leaving air)
Latent Load = .68 * SCFM * (G mixed air – G coil leaving air)

$$Qt = 6 \cdot (1.08 \cdot 21000 \cdot (73.9 - 52.9) + 0.68 \cdot 21000 \cdot (62.2 - 48))$$

Gross Sensible Capacity – 2,858 MBH
Gross Latent Capacity – 1,216 MBH

The basic concept behind using a chilled water coil instead of a DX cooling coil is that the cheaper cost of central chilled water versus electricity offsets any losses from additional pumping necessary to deliver the chilled water to the site, through the cooling coils, and back to the central plant.

The electrical needs of the water cooled AHU will consist of fan energy, and pumping cost for circulation of the cooling water. Because the system is essentially the same system as the original DX system, fan power will be assumed to be the exact same. The pumping power for the fan powered box reheat coils will be considered separately.

5.2.2 Chilled Water Pump Selection

Pump selection is a function of the total system head loss as well as the volumetric flow rate of the chilled water.

System Head Loss

Because the same volume of chilled water is supplied and returned to the central plant, all pipe sizes for the AHU cooling water will be the same. The main run from the central plant to the MOB is 432 feet. The vertical rise through the building is 55 feet. The rooftop delivery will be approximated as a 150 foot long pipe which will be long enough to distribute from the southern mechanical chase to the units on the north side of the roof. The following friction factor and flow velocity was found from the flow rate table for Schedule 40 steel piping on page 33.5 of ASHRAE Fundamentals Handbook 1997.
Flowrate = 680 gpm
Pipe Size – 6”
Frictional Loss – 2.8 feet wg per 100 feet of piping
Flow Velocity – 7.2 FPS

The following formula was used to compute overall piping pressure losses.
Supply pressure = total length * frictional loss + vertical rise height
Return pressure = total length * frictional loss – vertical fall height

$$\boxed{SupplyPressure = (432 + 55 + 150) \cdot \frac{2.8}{100} + 55}$$

$$\boxed{ReturnPressure = (432 + 55 + 150) \cdot \frac{2.8}{100} - 55}$$

Total system piping pressure loss = 73 ft – 37 ft = 36 ft wg

The cooling coil head loss is approximated from a similar coils internal pressure drop from Carrier’s AHU builder v.5.42.
Cooling coil pressure loss = 12.3 ft wg.

Total System Head Loss = 36 + 12.3 **= 48.3 ft wg**
Total System Flowrate = 680 gpm
Pump Selected – Bell & Gossett 5x5x9 ¾ 1750RPM with a 9 ¾ “ impeller and a 12 hp motor rated at 72% efficiency

Water Cooled AHU system electricity needs
Pumping Power

Pumping Power is dependant on the total system head loss as well as the volumetric flow rate of the chilled water as shown above.

Pump electrical power = pump horse power *.746/efficiency

Pumping Power = 12.4 kW

Fan Power
The same fan power is used as the original system of 145.5 kW.
Total AHU Electrical Power Use – 158 kW
Fan Power Use – 145.5 kW
Pumping Power Use – 12.4 kW
***** These totals do not include pumping cost and fan cost for fan powered VAV boxes with hot water heating.

5.2.3 <u>Hot Water Pump Selection</u>

Pump selection is a function of the total system head loss as well as the volumetric flow rate of the hot water.

System Head Loss
The pumping distance for hot water will only be from the basement mechanical room where a steam-water heat exchanger will be located, to the various floors of the building, through the distribution to the fan coil units, and then back to the basement.

The following friction factor and flow velocity was found from the flow rate table for Schedule 40 steel piping on page 33.5 of ASHRAE Fundamentals Handbook 1997.
With a flowrate of 216 gpm the following pipe sizes were found.

Riser Pipe Size – 4”
Frictional Loss – 2.7 feet wg per 100 feet of piping
Flow Velocity – 5.5 FPS

Floor Main Branch Pipe Size – 2”
Frictional Loss – 1.3 feet wg per 100 feet of piping
Flow Velocity – <3 FPS

Floor Individual Distribution Pipe Size – 1 ¼ ”
Frictional Loss – 1.7 feet wg per 100 feet of piping
Flow Velocity – <3 FPS

The following formula was used to compute overall piping pressure losses.
Supply pressure = total length * frictional loss + vertical rise height
Return pressure = total length * frictional loss – vertical fall height

$$\text{SupplyPressure} = \frac{80 \cdot 2.7 + 25 \cdot 1.3 + 129 \cdot 1.7}{100} + 50$$

$$\text{ReturnPressure} = \frac{80 \cdot 2.7 + 25 \cdot 1.3 + 129 \cdot 1.7}{100} - 50$$

Water - Air coil Pressure Drop – 8.1 ft wg approximated from Lytron water coil selector.
Steam – Water coil Pressure Drop – 12 ft wg
Total System Head Loss = 54.7ft – 45.3ft + 8.1ft + 12ft = 29.5 **=> 32 ft wg**
Total System Flowrate = 216 gpm
Pump Selected – Bell & Gossett 4x4x9 ¼ L 1150RPM with a 9 3/8 “ impeller with a 2.5 hp motor rated at 73% efficiency

Section 5.3

Alternative #2 DOAS System Paired with Enthalpy Wheel and Parallel Active Heated/Chilled Beams

5.3.1 System Basics

The second alternative for the MOB is again replacement the inefficient packaged DX air handling units on the rooftop with more efficient units that utilize the nearby available chilled water. However this time the units will be serving a DOAS system that requires less supply air. Because the units supply less air in a DOAS application, fewer units will be used. Overall the energy demands as a whole should be diminished because of the lessened amount of supply air needing to be treated. In addition to the air handling units, a parallel system for removal of sensible load and heating will be used.

After investigation, a York Custom air handling unit paired with a Semco enthalpy wheel was selected that will provide the needed supply air capacity with only three units supplying all spaces within the building. The new units have a maximum cfm of 19,000 cfm but are designed to operate at just below 17,000 cfm per unit.

The design method for the DOAS application is shown in the appendix in the Parallel Equipment Sizing Spreadsheet. System set points are based initially off of the larger of the latent load or ventilation requirements of individual spaces. In most cases, the space had excess sensible load not removed by the minimum supply air. To remove the sensible load, either the supply air volume was increased or a chilled beam was introduced to the space. Because the Trox beams can function as either chilled or heated beams, they were only added for heating purposes to spaces that had a heating demand and were not already equipped with a beam for cooling.

Because the chilled beams have recommended air volumes per set cooling capacity, different amounts of supply air are sent through the unit depending on how much cooling capacity is needed from the beam. Beam specifications are given in the parallel system sizing section.

Current Design Supply Air – 49,581 cfm

This value determined from air needed to remove latent load from fixture count method.

Maximum Supply Air – 57,000 cfm
Gross Cooling Capacity – 3,333 MBH
AHU Cooling Capacity – 3,113 MBH
Parallel Design Cooling – 220 MBH
Gross Sensible Capacity – 1,896 MBH
Gross Latent Capacity – 1,217 MBH

Gross Heating Capacity – 884,381 BTU/hr (259 kW)

This value is based off the all of the parallel units being used for maximum heating.

The basic goal of DOAS is that the lowered electrical and thermal loads of supplying lower volumes of air and using parallel cooling and heating systems will save money over the long run versus a relatively inefficient VAV system.

The electrical needs of the DOAS system will consist of fan energy for the AHU, pumping cost for circulation of the cooling water to the AHU, cooling water to the parallel units and heating water to the parallel units.

5.3.2 Chilled Water Pump Selection

Pump selection is a function of the total system head loss as well as the volumetric flow rate of the chilled water.

System Head Loss

Because a lower volume of chilled water is supplied and returned to the central plant versus the Alternative #1, the pipes will be resized using the flow rate table for Schedule 40 steel piping on page 33.5 of ASHRAE Fundamentals Handbook 1997. The main run from the central plant to the MOB is 432 feet. The vertical rise through the building is 55 feet. The rooftop delivery will be approximated as a 110 foot long pipe which will be long enough to distribute from the southern mechanical chase to the unit on the north side of the roof.
The following friction factor and flow velocity was found from the flow rate table for Schedule 40 steel piping on page 33.5 of ASHRAE Fundamentals Handbook 1997.

Flow Rate – 556 gpm
Pipe Size – 6”
Frictional Loss – 1.9 feet wg per 100 feet of piping
Flow Velocity – 6 FPS

The following formula was used to compute overall piping pressure losses.
Supply pressure = total length * frictional loss + vertical rise height
Return pressure = total length * frictional loss – vertical fall height

$$\text{SupplyPressure} = (432 + 55 + 110) \cdot \frac{1.9}{100} + 55$$

$$\text{ReturnPressure} = (432 + 55 + 110) \cdot \frac{1.9}{100} - 55$$

Total system piping pressure loss = 66.3 ft – 43.7 ft = 22.6 ft wg

The cooling coil head loss is approximated from a similar coils internal pressure drop from Carrier’s AHU builder v.5.42.
Cooling coil pressure loss = 20.4 ft wg.

> The increased water pressure drop is due to the large volume of water supplied to the three units.

Total System Head Loss = 22.6 + 20.4 **= 43 ft wg**
Total System Flowrate = 556 gpm
Pump Selected – Bell & Gossett 6x6x9 ¾ 1750RPM with a 7 ¾ " impeller and a 7.5 hp motor rated at 77% efficiency

Water Cooled AHU system electricity needs
Pumping Power
Pumping Power is dependant on the total system head loss as well as the volumetric flow rate of the chilled water as shown above.

Pump electrical power = pump horse power *.746(kW/hp)/efficiency

Pumping Power = 7.3 kW

Fan Power
The fan power for the DOAS application is found by using the same system static pressure drop of 3.8 inches wg as LWA spec'd for the existing units but lowering the supply volume from the old 21,000 cfm capacity to the new 19,000 cfm capacity.

Using the Greenheck product selection guide a fan speed of 1250RPM and 17hp per unit was found. Assuming an electrical efficiency of .8 the fan electrical power can be calculated in the equation below.

$$\boxed{\mathrm{Fan}_{\mathrm{Electrical,Power}} = 3 \cdot \mathrm{Fan}_{\mathrm{Horse,Power}} \cdot \frac{0.746}{\mathrm{Efficiency}}}$$

Total AHU Electrical Power Use – 54.9 kW
Fan Power Use – 47.6 kW
Pumping Power Use – 7.3 kW
***** These totals do not include pumping costs for the parallel system hot and cold water loops.

Section 5.4

Enthalpy Wheel Selection

5.4.1 Enthalpy Wheel Basics

A crucial part of any DOAS system is some sort of energy recovery system. In any system the outdoor air used may be at an undesirably high or low temperature, as well as being too humid or too dry. In comparison, return air is very close to optimal temperature and humidity parameters. In a standard system, the outdoor air is brought closer to the necessary parameters by simply mixing it with the return air before mechanically treating the air. However, in a DOAS system the outdoor is completely unadulterated by return air. Therefore a different method of removing sensible and latent energy from the air during cooling conditions and adding sensible and latent energy during the heating season is needed.

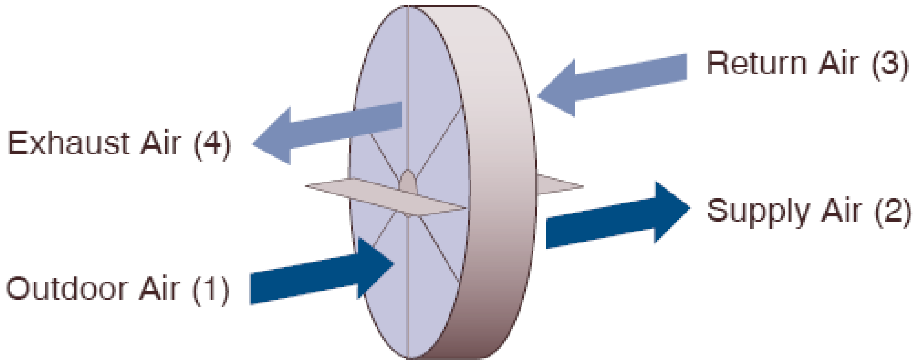
This is where an enthalpy wheel finds its use. An enthalpy wheel uses a single wheel to transfer both sensible and latent energy between airstreams. The sensible load is transferred via the aluminum wheel spokes themselves. The latent energy is transferred by a more sophisticated system. The aluminum spokes are coated in desiccant that is structured with three angstrom holes within itself. For a reference value, this is six times the diameter of an atom of Hydrogen. The holes give the desiccant the quality of being a selective absorption medium, transferring water vapor but not other contaminants, in addition to greatly increasing its surface area and therefore water affinity.

5.4.2 Enthalpy Wheel Sizing

The first step in sizing an enthalpy wheel is determining the amount of supply air needing to be treated. In the case of the MOB, the maximum supply air is 19,000 cfm. Using the Semco sizing chart, TE3-43 wheel was selected. This unit has a maximum flow rate of 21,450 cfm and a resultant face velocity of 500 fpm. The fact that the MOB design value is lower than this means the face velocity will be lower and the efficiency higher than what it's rated for. The efficiency rating for this wheel is 82.5 for transfer for both latent and sensible energy. This efficiency rating represents the percent of difference in either dry bulb temperature or grains of moisture between the return air and the outdoor air able to be transferred. It is represented with the following equation.

$$\text{Efficiency} = \frac{X1 - X2}{X1 - X3}$$

$$X2 = X1 - \text{Efficiency} \cdot (X1 - X3)$$



This Efficiency equation is applied to the outside air to generate the wheeled outside air in the System Sizing Spreadsheet in the appendix.

5.4.3 Enthalpy Wheel Freezing Precaution

Because the wheel will operate at very low outdoor air temperatures during the heating season, it is important to check that it not be in danger of becoming frosted. The procedure from Semco to determine whether this is a danger is as follows.

Step 1

Locate the RA point on the psychometric chart.

Step 2

Locate winter outdoor design condition (ASHRAE Fundamentals 1997 99.6% heating DB) Connect the two points with a straight line.

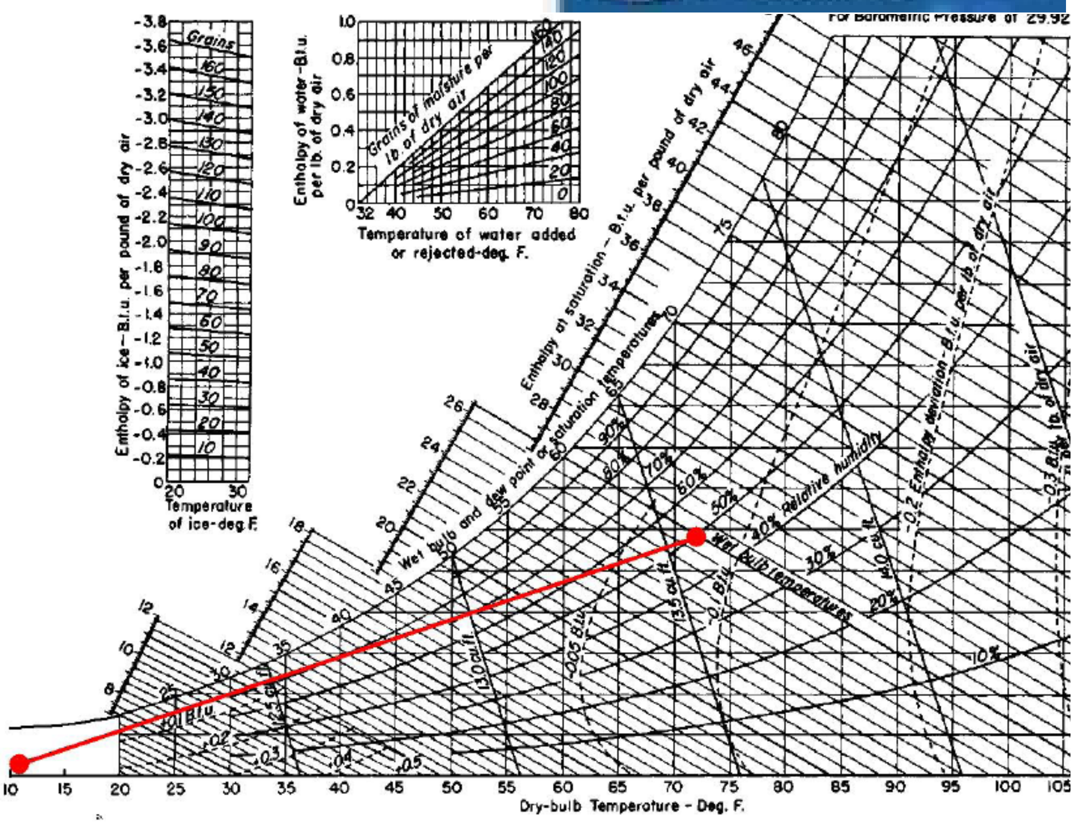
Step 3

Determine the higher dry bulb temperature at which this line intercepts the saturation curve.

Step 4

Add 2 degrees F to this point and make that the system set point for preheating.

In the MOB application the line never reaches the system saturation curve because of Baltimore's relatively mild winter temperatures. This negates the need for preheating. The psychometric chart diagram used for steps one and two is shown below.


For Barometric Pressure of 29.92
Enthalpy of ice - B.t.u. per pound of dry air
Grains
Temperature of ice-deg. F.
Enthalpy of water - B.t.u. per lb. of dry air
Grains of moisture per lb. of dry air
Temperature of water added or rejected-deg. F.
Enthalpy at saturation - B.t.u. per pound of dry air
Wet bulb and dew point or saturation temperatures
Relative humidity
Wet bulb temperatures
Enthalpy deviation - B.t.u. per lb. of dry air
Dry-bulb Temperature - Deg. F.

5.4.5 Wheel Cross Contamination Concerns

One of the main concerns with inclusion of an enthalpy wheel in an all outdoor air system is the contamination of the incoming outdoor air from the wheel that is also in contact with the exhaust air stream.

The following pollutants were independently tested by the Georgia Tech Research Institute. Notice the independently verified water transfer efficiency.

Pollutant Tested	Pollutant Concentration*	Measured Cross-Contamination
Isopropanol	20 ppm	None
Methyl-Isobutyl-Ketone	1840 ppb	None
Xylenes	7100 ppb	None
Carbon Dioxide	500 ppm	None
Propane	82 ppm	None
Sulfur Hexafloride	212 ppm	None
Water Vapor	4000 ppm	80%
*Concentrations selected by GTRI to reflect worst case for typical application		

5.4.6 Wheel Control

Much like an airside economizer, an enthalpy wheel need not operate all of the time at full capacity. There are also times when the outdoor air parameters are closer to the supply air parameters than the return air is. In these cases, the wheel is actually impairing system performance.

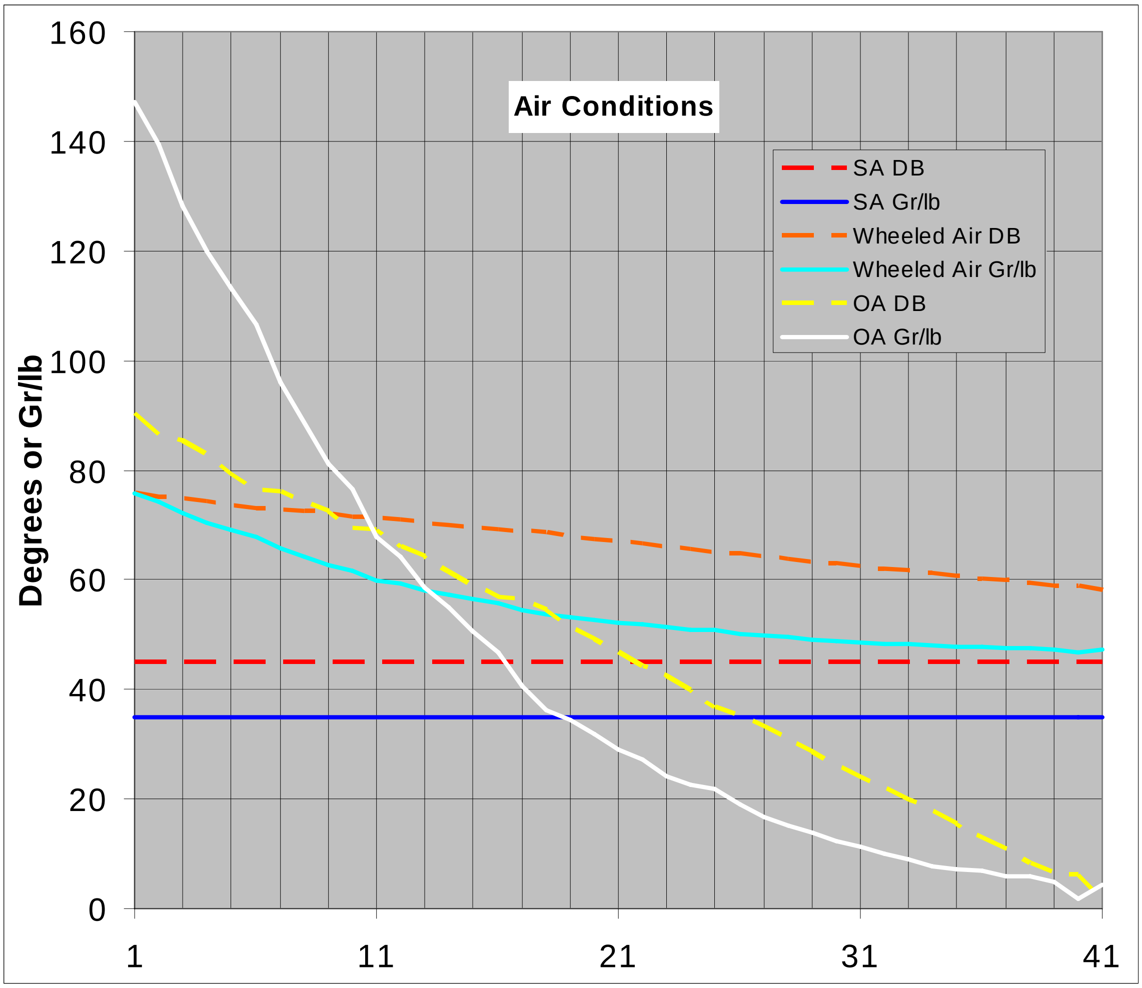
The following graphs were generated by using BIN weather data for Baltimore, MD. The enthalpy wheel is .8, the return air parameters for are 72 degrees F and 58 Gr/lb.

<u>Trial One</u>

Total Cooling Energy with E-Wheel at Full Capacity

The following graph shows the outdoor parameters, return air parameters, and air parameters after being pre-treated by the enthalpy wheel. The three following graphs are based on the values found in the E-Wheel Chart in the appendix. This graph represents the enthalpy wheel operating at full capacity all of the time.

Air Parameters with E-Wheel at Full Capacity



The point in this graph where it would be more economical for sensible performance to use only outside air is where the outside air dry bulb temperature line (dotted yellow) dips below the wheeled air dry bulb temperature (dotted orange). Likewise for latent performance the point is where the white line representing outdoor air water content dips below the light blue line which represents air water content after the enthalpy wheel.

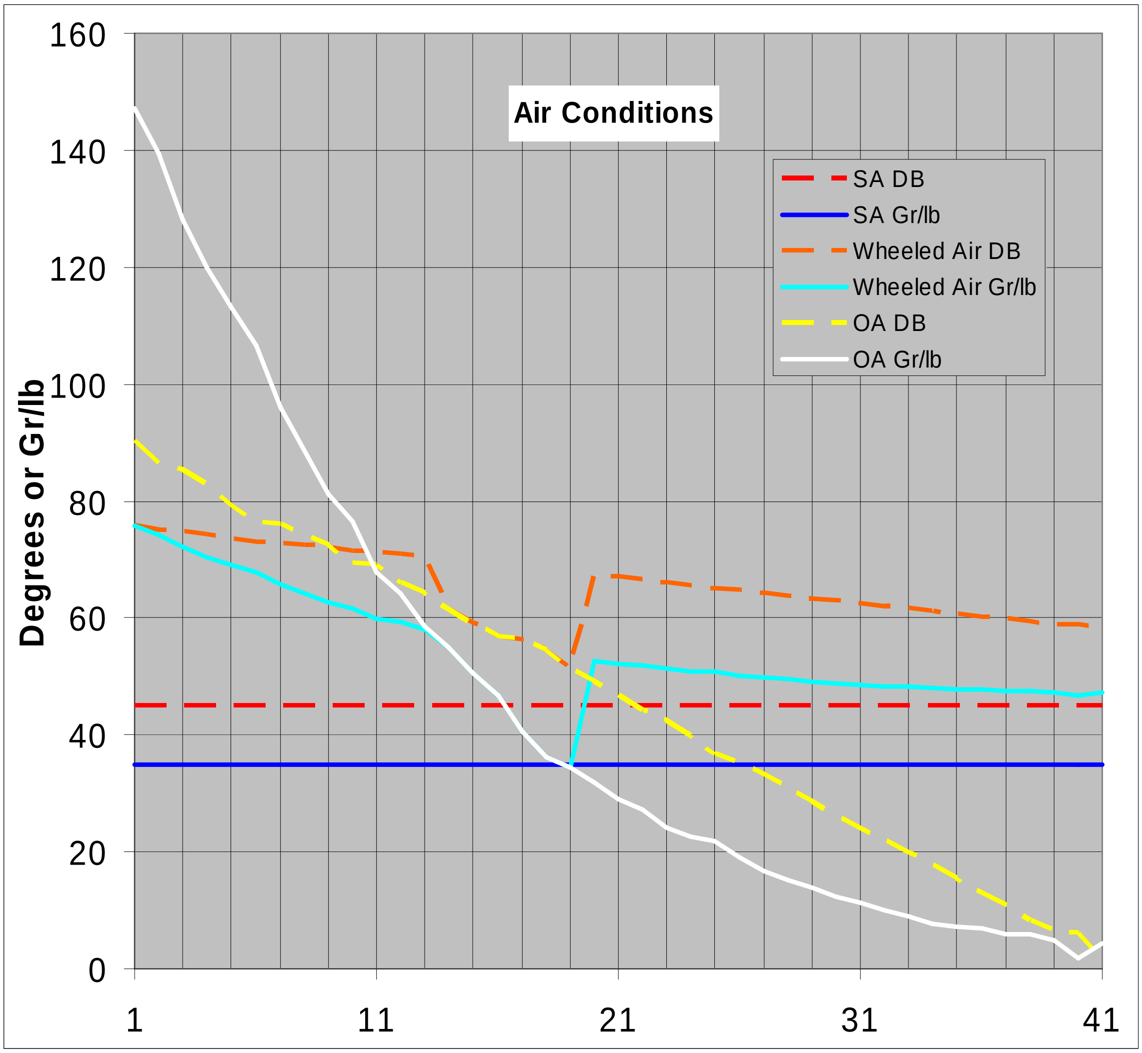
Resultant Total Cooling Energy – 19,532 MMBTU

<u>Trial Two</u>

Total Cooling Energy with Free Cooling when Available

In this trial, the E-Wheel is disused when the latent outdoor latent parameters are between the wheeled air and the minimum supply air latent level. In other, when free cooling is available it is used until overcooling occurs.

Air Parameters with Free Cooling



In this trial, outdoor air is used as soon as the outdoor latent parameters drop below the wheeled air latent parameters as seen at the x-value of thirteen. Once there is an occasion of overcooing as at the x-value of nineteen, the wheel is turned on again to full capacity.

Resultant Total Cooling Energy – 17,861 MMBTU

This value is an 8.5% savings over the E-Wheel being used at full capacity all of the time.

Trial Three

Total Cooling Energy with Free Cooling and Modified E-Wheel use

In this trial, the E-Wheel is disused when the latent outdoor latent parameters are between the wheeled air and the minimum supply air latent level just as before. However, once the outdoor air parameters drop below the desired supply air parameters, the E-Wheel is used at part capacity to warm up the air. The energy transfer capacity of the E-Wheel is assumed to be in a direct linear relationship with the rotation speed of the wheel. This relationship is expressed by the following equation.

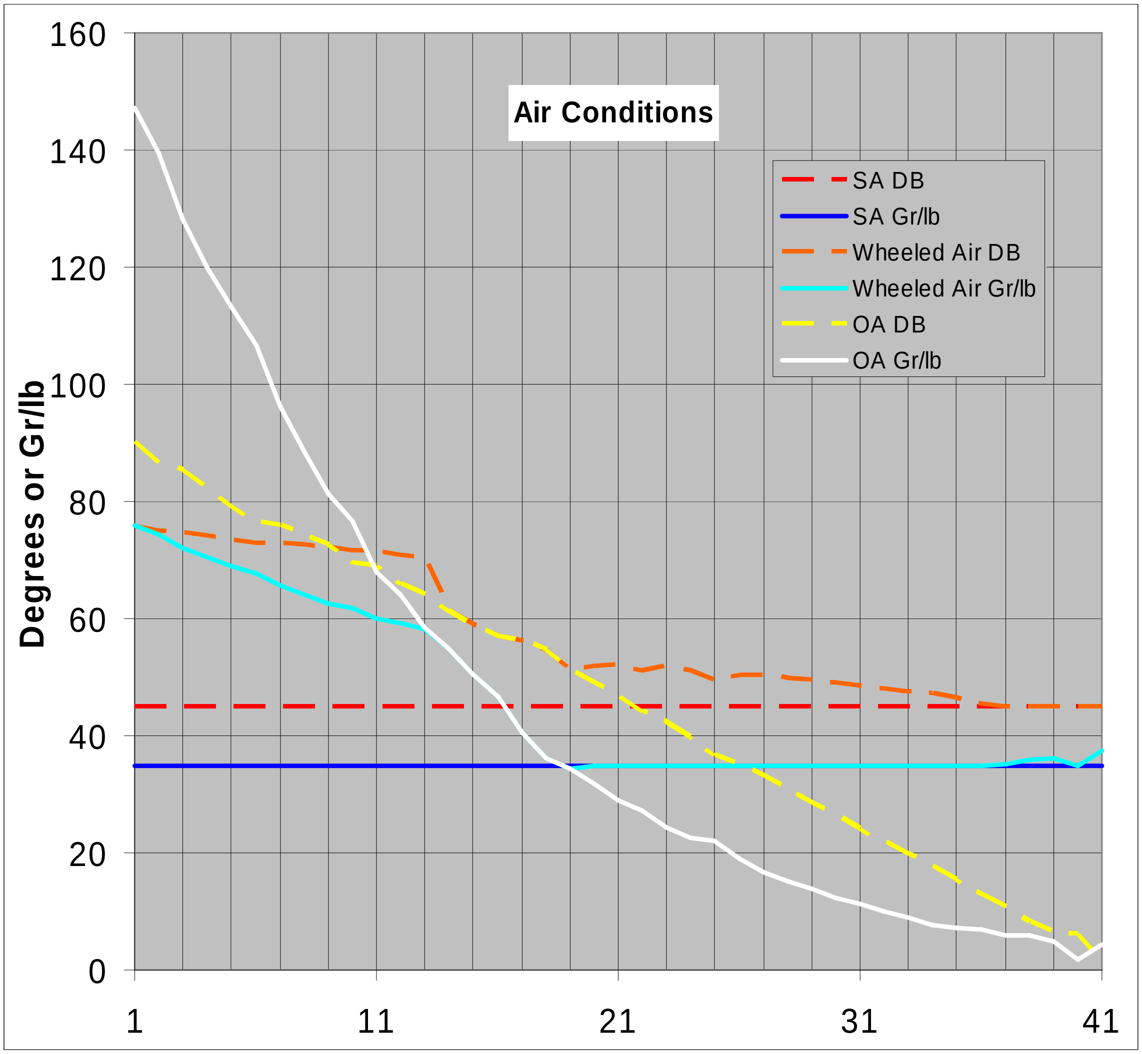
$$\boxed{Gr = Gr_{oa} - X \cdot 0.8 \cdot (Gr_{oa} - Gr_{ra})}$$

This formula simplifies to the formula below, where X equals wheel spin rate as a percentage of full capacity spin rate.

$$\boxed{X = \frac{Gr_{oa} - 35}{0.8 \cdot (Gr_{oa} - Gr_{ra})}}$$

Using this equation the wheel spin rates found in the E-Wheel Chart in the appendix were found. Below is the graph corresponding to this trial.

Air Parameters with Free Cooling and Modified E-Wheel use



In this chart it is evident that latent load was the controlling factor since it exactly matches the desired air parameters from the point when the outside air begins overcooling to the point at the x-coordinate of thirty-seven, where this scheme caused sensible overcooling.

Resultant Total Cooling Energy – 11,134 MMBTU
This value is a 43% savings over the E-Wheel being used at full capacity all of the time.
This value is a 38% savings over the E-Wheel with free cooling.
*Note – This savings will not correlate directly to the overall system performance since it does not include parallel system loads. This comparison was conducted simply to evince the savings possible with intelligent E-Wheel control.

Section 5.5

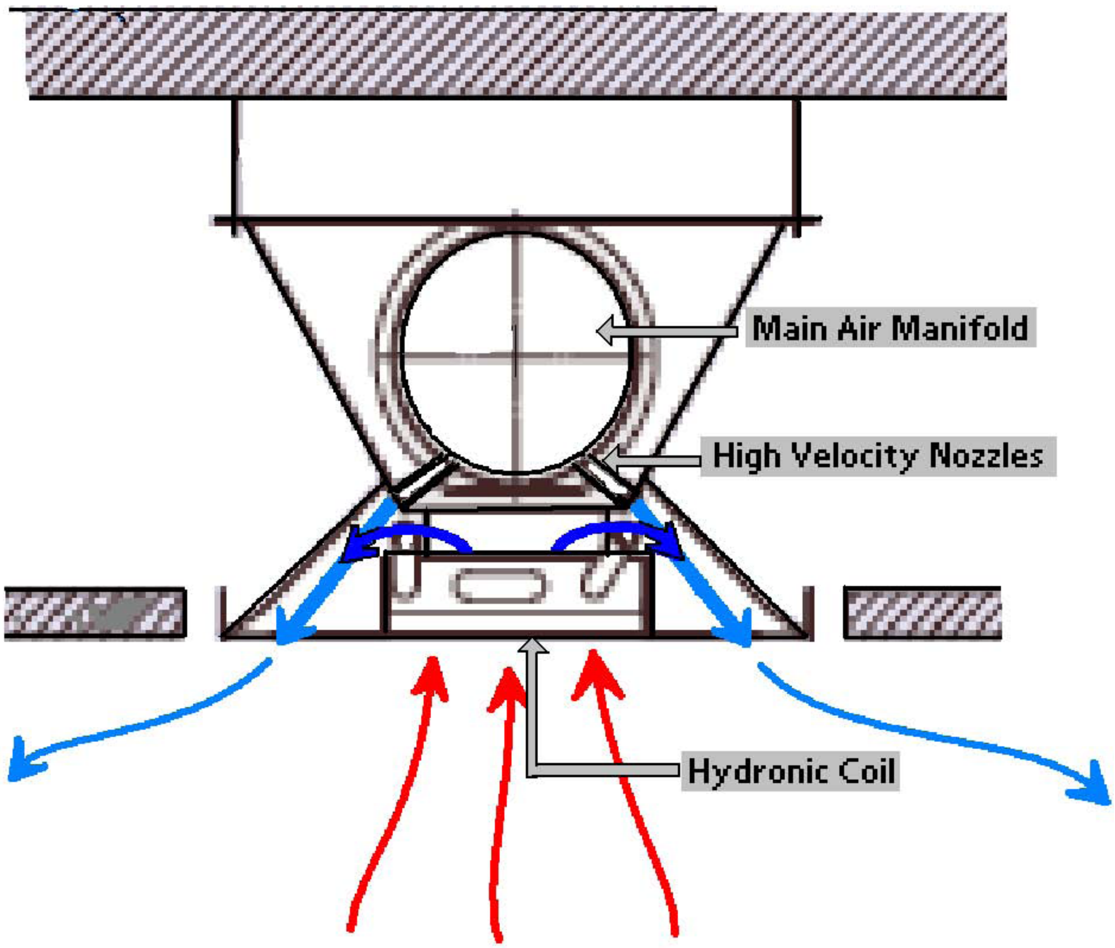
Parallel Equipment Design

5.5.1 Parallel Equipment Basics

Because the supply volumes in a DOAS system are so low, they often cannot remove the entire sensible load in a space. Likewise, if the space has heating needs, they cannot be accommodated by the centrally supplied cold air. For this reason, a parallel system is needed to enable the mechanical system to properly treat the individual spaces.

In the MOB, auxiliary heating and cooling needs will be met by active heated and chilled beams. Active beams are supply terminals that use high velocity, and therefore low pressure, supply air to induce room air through the unit. While passing through the unit, the room air can be either heated or chilled via a centrally located hydronic coil before being mixed with the supply air and delivered to the room. A diagram of the active beam is shown below.

Trox Active Heated/Chilled Beam Section



5.5.2 Parallel System Sizing

The cooling or heating capacities of the beams are determined by the primary airflow and the resulting change in temperature of the water in the hydronic coil. The chilled beams are supplied with water from the main AHU cold water return. This water is mixed with domestic cold water to a temperature of 60 degrees F. The heated beams are supplied hot water from a steam to water heat exchanger in the basement mechanical room. The hot water is at 150 degrees F. The charts below show the unit specific cooling, primary air, water flow rate, temperature change and water pressure drop. These are the values that were used in both the System Sizing Spreadsheet in the appendix.

900mm (2.95ft) long Trox Active beam
Cooling Application

Secondary Air Cooling (BTU/hr)	Needed Primary Air (cfm)	Chilled Water Flowrate (gpm)	Water Temp Rise (degree F)	Water Pressure Drop (ft wg)
427	25	0.59	1.8	0.49
491	32	0.62	1.98	0.49
543	38	0.63	2.16	0.49
587	45	0.63	2.34	0.49
624	51	0.62	2.52	0.49
659	57	0.61	2.7	0.49

1200mm (3.94ft) long Trox Active beam
Cooling Application

Secondary Air Cooling (BTU/hr)	Needed Primary Air (cfm)	Chilled Water Flowrate (gpm)	Water Temp Rise (degree F)	Water Pressure Drop (ft wg)
1174	34	0.61	2.34	0.59
1262	42	0.61	2.7	0.59

900mm (2.95ft) long Trox Active beam
Heating Application

Secondary Air Heating (BTU/hr)	Needed Primary Air (cfm)	Hot Water Flowrate (gpm)	Water Temp Drop (degree F)	Water Pressure Drop (ft wg)
256	21	0.24	2.7	0.1
341	21	0.24	3.6	0.1
444	21	0.25	4.5	0.1
682	25	0.27	6.3	0.1
819	25	0.57	3.6	0.1
918	25	0.28	8.3	0.1
1051	32	0.27	9.7	0.1
1157	38	0.28	10.4	0.1
1320	47	0.28	12.1	0.1

1800mm (5.91ft) long Trox Active beam
Heating Applicaton

Secondary Air Heating (BTU/hr)	Needed Primary Air (cfm)	Hot Water Flowrate (gpm)	Water Temp Drop (degree F)	Water Pressure Drop (ft wg)
2474	106	0.28	22.5	0.1

5.5.3 <u>Parallel Equipment Cold Water Pump Selection</u>

Pump selection is a function of the total system head loss as well as the volumetric flow rate of the hot water.

System Head Loss

The pumping distance for parallel cooling water will be from the basement mechanical room where a tap off the AHU return water will be located, to the various floors of the building, through the distribution to the parallel cooling beams and then back to the basement.

The following friction factor and flow velocity was found from the flow rate table for Schedule 40 steel piping on page 33.5 of ASHRAE Fundamentals Handbook 1997. With a flowrate starting at 277 gpm and diminishing to 20 gpm the following pipe sizes were found.

Flow Rate – 277 gpm
Basement to First Floor Plenum Riser Pipe Size – 4”
Frictional Loss – 4.5 feet wg per 100 feet of piping
Flow Velocity – 7 FPS

Flow Rate – 191 gpm
First Floor Plenum to Third Floor Plenum Riser – 4”
Frictional Loss – 2.2 feet wg per 100 feet of piping
Flow Velocity – 5 FPS

Flow Rate – 50 gpm
Southern Feeder Pipe Size – 2 ½ ”
Frictional Loss – 2 feet wg per 100 feet of piping
Flow Velocity – 3.4 FPS

Flow Rate – 20 gpm
Southern Branch Pipe Size – 1 ½ “
Frictional Loss – 3 feet wg per 100 feet of piping
Flow Velocity – 3.1 FPS

Because this piping service is smaller and more complicated than those previously considered for the rooftop AHUs it will be calculated in the segments above. It will also have elbows and splitters factored into its head pressure loss.
The Equation below accounts for straight pipe losses and fitting losses. The fitting losses are calculated on an equivalent straight pipe length method and modified for the ratio of fluid flow rate in stream of interest to the fluid flow rate in the alternate flow path.

Pressure Drop =2*(length*frictional loss + number of equivalent elbows♦*equivalent length of fitting♦♦)*frictional loss
♦From Figure 4 on page 33.6 of ASHRAE Fundamentals Handbook 1997
♦♦From Table 6 on page 33.6 of ASHRAE Fundamentals Handbook 1997

$$\mathrm{First}_{\mathrm{Riser,Head}} = 2 \cdot (24 + 0.1 \cdot 13.1 + 0.13 \cdot 10.5) \cdot \frac{4.5}{100}$$

$$\mathrm{Second}_{\mathrm{Riser,Head}} = 2 \cdot (24 + 0.6 \cdot 10.6 + 4 \cdot 9.5) \cdot \frac{2.2}{100}$$

$$\mathrm{Southern}_{\mathrm{Feeder,Head}} = 2 \cdot (25 + 6.2 + 5.6) \cdot \frac{2}{100}$$

$$\mathrm{Southern}_{\mathrm{Branch,Head}} = 2 \cdot (119 + 4.2 + 4.2) \cdot \frac{3}{100}$$

Piping Head Drop = 14.51 ft wg
Water - Air coil Pressure Drop – 0.49 ft wg
This value from equipment manufacturer cut sheet
Total System Head Loss = 14.51 + 0.49 **= 15.0 ft wg**
Total System Flowrate = 277 gpm
Pump Selected – Bell & Gossett 5x5x9 ¾ 1150RPM with a 7 ¾ " impeller and a 1.5 hp motor rated at 74% efficiency

Active Beam Parallel Cooling system electricity needs
Pumping Power
Pumping Power is dependant on the total system head loss as well as the volumetric flow rate of the chilled water as shown above.

$$\text{Fan}_{\text{Electrical,Power}} = 3 \cdot \text{Fan}_{\text{Horse,Power}} \cdot \frac{0.746}{\text{Efficiency}}$$

Pumping Power = 1.5 kW

5.5.4 Parallel Equipment Hot Water Pump Selection

Pump selection is a function of the total system head loss as well as the volumetric flow rate of the hot water.

System Head Loss

The pumping distance for hot water will only be from the basement mechanical room where a steam-water heat exhchanger will be located to the various floors of the building, through the distribution to the parallel heating beams and then back to the basement.

The following friction factor and flow velocity was found from the flow rate table for Schedule 40 steel piping on page 33.5 of ASHRAE Fundamentals Handbook 1997. With a flowrate starting at 126 gpm and diminishing to 15 gpm the following pipe sizes were found.

Flow Rate – 126 gpm
Basement to First Floor Plenum Riser Pipe Size – 3"
Frictional Loss – 3.5 feet wg per 100 feet of piping
Flow Velocity – 5.3 FPS

Flow Rate – 87 gpm
First Floor Plenum to Third Floor Plenum Riser – 3"
Frictional Loss – 1.8 feet wg per 100 feet of piping
Flow Velocity – 3.8 FPS

Flow Rate – 25 gpm
Southern Feeder Pipe Size – 1 ½ "
Frictional Loss – 4.8 feet wg per 100 feet of piping
Flow Velocity – 3.9 FPS

Flow Rate – 15 gpm
Southern Branch Pipe Size – 1 ¼ "
Frictional Loss – 4 feet wg per 100 feet of piping
Flow Velocity – 4 FPS

Because this piping service is smaller and more complicated than those previously considered for the rooftop AHUs it will be calculated in the segments above. It will also have elbows and splitters factored into its head pressure loss except in the main feeder where the high flow through ratio makes the losses to the main flow path negligible. The Equation below accounts for straight pipe losses and fitting losses. The fitting losses are calculated on an equivalent straight pipe length method.

Pressure Drop =2*(length*frictional loss + equivalent length of fitting♦♦)*frictional loss
♦♦From Table 6 on page 33.6 of ASHRAE Fundamentals Handbook 1997

$$\boxed{First_{Riser,Head} = 2 \cdot 24 \cdot \frac{3.5}{100}}$$

$$\boxed{Second_{Riser,Head} = 2 \cdot 24 \cdot \frac{1.8}{100}}$$

$$\boxed{Southern_{Feeder,Head} = 2 \cdot (25 + 4 + 4.4) \cdot \frac{4.8}{100}}$$

$$\boxed{Southern_{Branch,Head} = 2 \cdot (119 + 3.7 + 3.7) \cdot \frac{4}{100}}$$

Piping Head Drop = 15.9 ft wg
Water - Air coil Pressure Drop – 0.1 ft wg
This value from equipment manufacturer cut sheet
Steam – Water coil Pressure Drop – 7 ft wg

Total System Head Loss = 14.51 + 0.49 **= 23 ft wg**
Total System Flowrate = 126 gpm
Pump Selected – Bell & Gossett 4x4x9 ¼ 1150RPM with a 7 3/8 " impeller and a 1.25 hp motor rated at 62% efficiency

Active Beam Parallel Heating system electricity needs
Pumping Power
Pumping Power is dependant on the total system head loss as well as the volumetric flow rate of the chilled water as shown above.

$$Fan_{Electrical,Power} = 3 \cdot Fan_{Horse,Power} \cdot \frac{0.746}{Efficiency}$$

Pumping Power = 1.5 kW

Section 6

Indoor Air Quality Comparison with VAV and DOAS

6.1 Basics on Increased IAQ with DOAS

Aside from energy savings, DOAS has another major advantage over traditional systems. That is the increased level of indoor air quality provided by the 100% outdoor air supply. By not recycling air contaminants too small to be picked up by the filter, DOAS dilutes and removes contaminants much more efficiently than standard systems using mixed air.

Anyone who has been to the doctors' office with a cold of some type, replete with a depressed immune system, can identify with the alarming feeling of being surrounded by people with communicable illnesses. Indoor air quality in a medical office building, although not as critical as facilities with invasive medical procedures or high security risks, is of elevated concern as compared to a normal office building.

6.2 Calculation of Indoor Air Contaminant Levels

In the MOB, contaminant dilution is modeled by using CO2 analysis on a per occupant basis and comparing VAV systems to DOAS systems.

The following formulas are used for determining the current concentration of CO2 in the particular space. The first equation is using a the current VAV system where 90% of the air is re-circulated through the space. The two supply air terms are because of the two sources of supply air; outside air, and return air.

$$C = \frac{Vsa \cdot Camb \cdot 0.1 + Vsa \cdot Cprev \cdot 0.9 - Vra \cdot Cprev}{Volume} + Gocc \cdot Occ \cdot \frac{1000000}{Volume} + Cprev$$

The next equation is for a DOAS application where none of the air is re-circulated. Because none of the air is re-circulated, all of the supply air is outside air, with a low concentration of CO2.

$$C = \frac{Vsa \cdot Camb - Vra \cdot Cprev}{Volume} + Gocc \cdot Occ \cdot \frac{1000000}{Volume} + Cprev$$

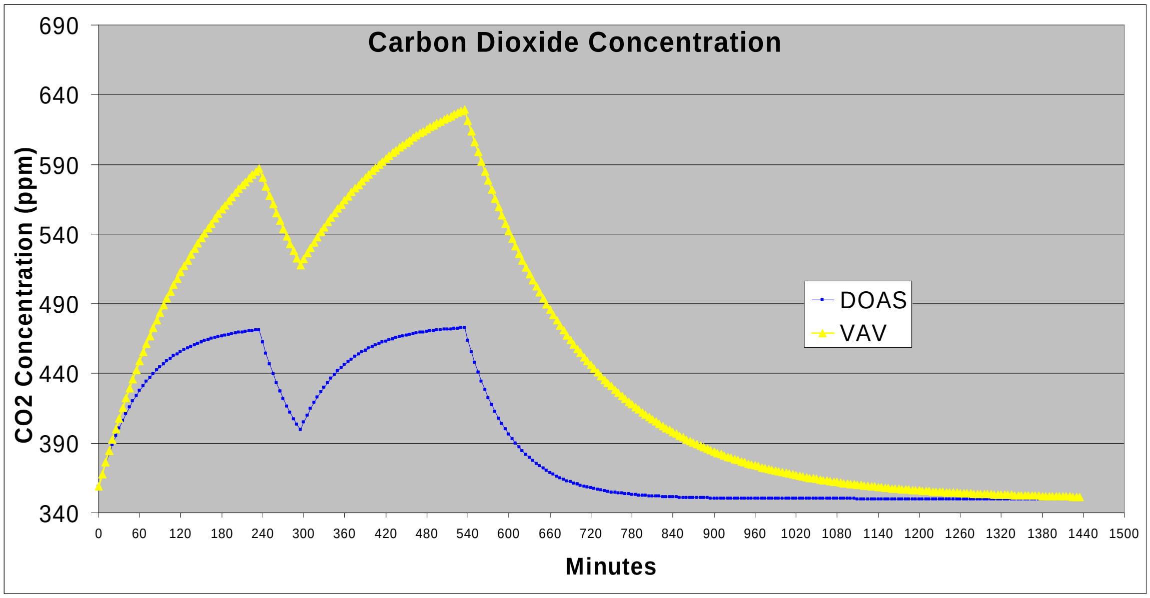
Where C = current concentration (ppm)
Vsa = supply air volume (cfm)
Camb = outside air ambient concentration (ppm)
Cprev = inside air concentration from previous time sample (ppm)
Vra = return air volume (cfm)
Volume = volume of space (ft^3)
Gocc = Generation rate per occupant (cfm)
    * 0.31 l/s was used as an occupant generation rate as per ASHRAE std. 62-2004
Occ = number of occupants in space

The Spreadsheet for generation of the following charts is CO2 Diffusion in the appendix.

6.3 Spaces Modeled for Contaminant Levels

When applied to two different rooms in the MOB, the advantage of using DOAS becomes apparent in both lowering of steady state contaminant concentrations and increasing the rate of contaminant dilution. The following graph shows CO2 levels within an office, 3073, on the third floor of the MOB. The yellow line represents the current VAV system and the blue line representing the proposed DOAS system.

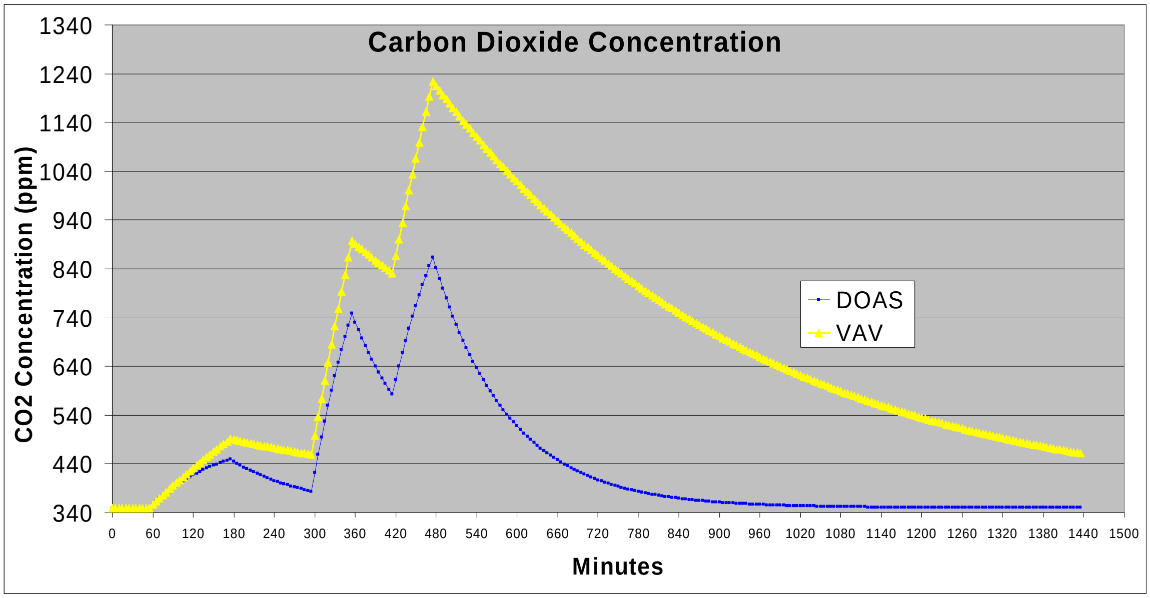
Office 3073, CO2 Dilution



In this graph, the curves represent an eight hour work day with one occupant. The dip beginning at 240 (12:00), and ending at 300 (1:00), represents the lunch hour when the office is unoccupied.

This graph represents a conference room, Room 3080, with intermittent use. Because the room is not constantly occupied and may actually only be occupied a very small percent of the time, a diversity factor is employed in VAV design that reduces the amount of people the space is assumed to be occupied by.

Conference Room 3080, CO2 Dilution



In the morning there are two people using the space for two hours. The shallow dip in levels is the hour before a lunch and the lunch hour, when the conference room is unoccupied. After 300 there is a one hour meeting with twelve people, a one hour break and another one hour meeting, again with twelve people to round out the day. As you can see, because of the diversity factor used in VAV sizing, the ventilation allows a large increase in contaminant concentration when the room is at full occupancy. The DOAS system keeps the contaminant levels significantly lower than that of the currently utilized system.

6.4 <u>Conclusions on DOAS Ventilation</u>

Overall, the DOAS system outperforms the existing VAV arrangement in terms of IAQ. The DOAS system keeps the rooms at a lower contaminant level. It does this because its steady state concentration is lower do to the increased volume of outdoor air. This is evident on the graph of room 3073, where the DOAS system almost reaches steady state concentration while the VAV is clearly not near steady state yet despite its maximum level being over twice as high above ambient concentration as that of the DOAS system.

The DOAS system also reduces the concentration of contaminant faster than the VAV system. This is shown in the graph of room 3080, where the DOAS concentration drops sharply after the first and second meetings as compared to the VAV systems slower decay of contaminant levels.

Section 7

Economic Analysis

7.1 Differing Economic Factors

In any mechanical system there are two main economic considerations the designer takes into account when deciding how to approach the problem. These two factors are the first cost, which covers the initial investment and the operating cost, which includes energy costs and maintenance.

What will the initial price of the buildings systems be? How much will the owner pay up front for the system? Often these questions drive design. This is the case in the MOB. The packaged DX cooling units are pretty much the cheapest option for treating a space. The same can be said about the electric reheat VAV boxes. The individual components of both option #1 and option #2 are higher in first cost.

The operating cost is more of a measure of system efficiency and often comes at a higher premium. The operating cost includes the rate of the driving energy, be it electricity, gas, oil, or other. The operating cost typically also includes maintenance costs. In the MOB the operating cost was of secondary importance to the first cost.

7.2 First Cost Analysis

The first cost of the three systems, Existing, Alternative #1, and Alternative #2 was analyzed in terms of its individual components. Where applicable, Costworks 2005 was used to generate unit costs. Unit quantities for piping were generated from drawing take-offs.

Below is the totaled cost information for the first cost analysis of the MOB. For detailed cost breakdown, reference the First Cost Spreadsheet in the appendix.

First Cost Totals

Assembly Totals	Existing VAV system with DX cooling	242,930.00
	Alt #1 VAV system with chilled water cooling	379,035.00
	Alt #2 DOAS system with chilled water cooling	269,683.20
	Parallel System of Active Heated/Chilled Beams	290,830.30
	Existing VAV box system with electric reheat	109,530.00
	Existing VAV box system with hot water reheat	140,846.40

Combined Totals	Existing system	352,460.00
	Alt #1	519,881.40
	Alt #2	560,513.50

7.2.1 Conclusions on First Cost

The first cost analysis is not surprising. The existing system has a much lower price tag than both Alternative #1, and Alternative #2. Unlike both of those systems, there is no cost for extra piping. The air handling units themselves are also cheaper than the water cooled AHUs used for the two redesigned systems. Much of the increased price for the DOAS system is the cost of the parallel system. The beams themselves, are of English origin and do not have a large market in the US. Also, perhaps they are more economically used in large, open plan office buildings where they will never be used at part capacity as many of the heating beams in the MOB are. However, in the MOB, the danger of overcooling within the many smaller spaces necessitated more units. There were also instances where the inclusion of multiple units where fewer units may have been with a higher volume of supply air per unit. Both of these factors pushed the overall number of beams up. All of this combined to cause the parallel system to negate any of the cost savings of the fewer DOAS AHUs as compared to the other two alternatives.

7.3 Operating Cost Analysis

The operating cost for the existing system and Alternative #1 as applied to the MOB were generated by using Carriers Hourly Analysis Program. HAP did not return a reasonable cost for the DX cooling, other than fan energy cost. For this reason the fan cost from HAP was combined with the annual non-fan energy as per LWA circuit sizing specifications to generate the annual operating cost of the existing DX/VAV system. The electrical system capacities in the System Sizing Spreadsheet in the appendix reflect the LWA supplied sizing. The Values are slightly higher than those determined by Leach Wallace Associates, Inc. However, they are in the same ratio as the costs from LWA and only nine percent higher for the existing system cost and six percent higher for the Alternative #1 cost.

The annual operating cost for Alternative #2, the DOAS system, was not generated using HAP because of its unreliability in evaluating DOAS systems. Instead the DOAS annual operating cost was scaled from the chilled water VAV operating cost by comparing the amount of fan, pumping and chilled water energy used annually. The values for fan energy, chilled water consumption and pumping energy are found in the System Sizing Spreadsheet in the appendix.

Annual Operating Cost Comparison Chart

	Annual Operating Cost (Dollars)	Reference Operating Cost (as per LWA) (Dollars)	Annual Cost Savings vs DX (Dollars)
Existing	165,509	151,399	0
Alternative #1	130,004	122,263	35,505
Alternative #2	106,603		58,906

7.4 Simple Payback Period Analysis

The simple payback period is determined by dividing the difference in cost of two systems by the annual savings.

Alternative #1 vs. Existing System

For alternative #1 as compared to the existing system, the simple payback period is determined as below.
(FC_alt#1 – FC_exist) / (OC_exist – OC_alt#1) = Payback Period
($519,881 - $352,460) / $35,505 per year = **4.7 Years**

Alternative #2 vs. Existing System

For alternative #2 as compared to the existing system, the simple payback period is determined as below.
(FC_alt#2 – FC_exist) / (OC_exist – OC_alt#2) = Payback Period
($560,514 - $352,460) / $58,906 per year = **3.5 Years**

Alternative #2 vs. Alternative #1

For alternative #2 as compared to the existing system, the simple payback period is determined as below.
(FC_alt#2 – FC_alt#1) / (OC_alt#1 – OC_alt#2) = Payback Period
($560,514 - $519,881) / $23,401 per year = **1.7 Years**

7.4.1 Payback Analysis Conclusions

Despite the increased first costs of both the chilled water and steam reheat VAV system, and most notably the DOAS system with parallel active beams, they both showed favorable payback periods. However, in a building that has substantial tenant space, such as the MOB, it is unlikely that either would be selected over the initially cheaper DX system with electric reheat.

Section 8
Breadth Topic #1, Electrical System Resizing for DOAS Application

8.1 Existing Electrical Equipment Supporting Mechanical Systems
The current system is served by eleven separate panel boards. These panel boards were found in the electrical power section of the MOB plans. The breakdown of panel boards is as follows:

Existing Panel Description Chart

Quantity	Amperes	Voltage	MCB/MLO	Service to
1	100	208Y/120	MCB	FCU-1, ACCU-1
1	800	480	MLO	AHU-1,2,5
1	800	480	MLO	AHU-3,4,6
1	300	480Y/277	MLO	FCUs, Hot Water Heater
7	225	480Y/277	MLO	FCUs

With the exception of the 100Amp panel board, all of these panel boards are directly related to the building AHUs and terminal reheat units. The 100Amp panel board is out of the scope of this report because the split DX system it powers is used only for the elevator mechanical room and will not be considered for change.

With the introduction of the DOAS system with parallel beams, the central driver for the system will change from electricity to remotely supplied chilled water and steam. The electrical portion of the overall system will go from a primary role, as in high fan energy, and electrical resistance heating to a supporting role, as pumping energy and reduced fan energy.

- Overall the amount of electrical equipment needed to support the AHUs will be cut in half with the number of units.
- The panel boards serving the heavy loads of the many FCUs with their electric reheat will instead be replaced by a panel board(s) serving pumps to circulate the water to the parallel equipment.

8.2 Alternative #2 Electrical Equipment
The system electrical equipment to be used in Alternative #2 is as follows is listed below. The corresponding amperage of the equipment is determined with the following equation.

$$\boxed{kW = \sqrt{3} \cdot Voltage \cdot Amperage}$$

Three AHU Fan - 15.9kW, 480V, 19Amps
One Main Chilled Water Pump – 7.3kW, 480V, 9Amps
One Parallel System Chilled Water Pump – 1.5kW, 480V, 2Amps
One Parallel System Hot Water Pump – 1.5kW, 480V, 2Amps

Even though the entire system Amperage is only 70Amps, conveniently small enough to be placed on one 100Amp panel board, two 100Amp, 480V panel boards will be used.

8.3 Comparison of Electrical Equipment for Existing Equipment and Alternative #2

Currently the electrical equipment cost for the MOB is valued by Costworks 2005 at $24,380. The exact component designations and RS Means Codes can be found in the First Cost Spreadsheet in the appendix.

The proposed two panel boards are valued at $1,260 each. This value is also found in the First Cost Spreadsheet in the appendix.

This makes the total cost savings in supporting electrical equipment from switching from the existing system to DOAS the following.

Cost Savings - $21,860 = $24380 – 2 * $1,260

Section 9

Constructability Review of Connection of Chilled Water And Steam to the MOB from the S. of Orleans Energy Plant

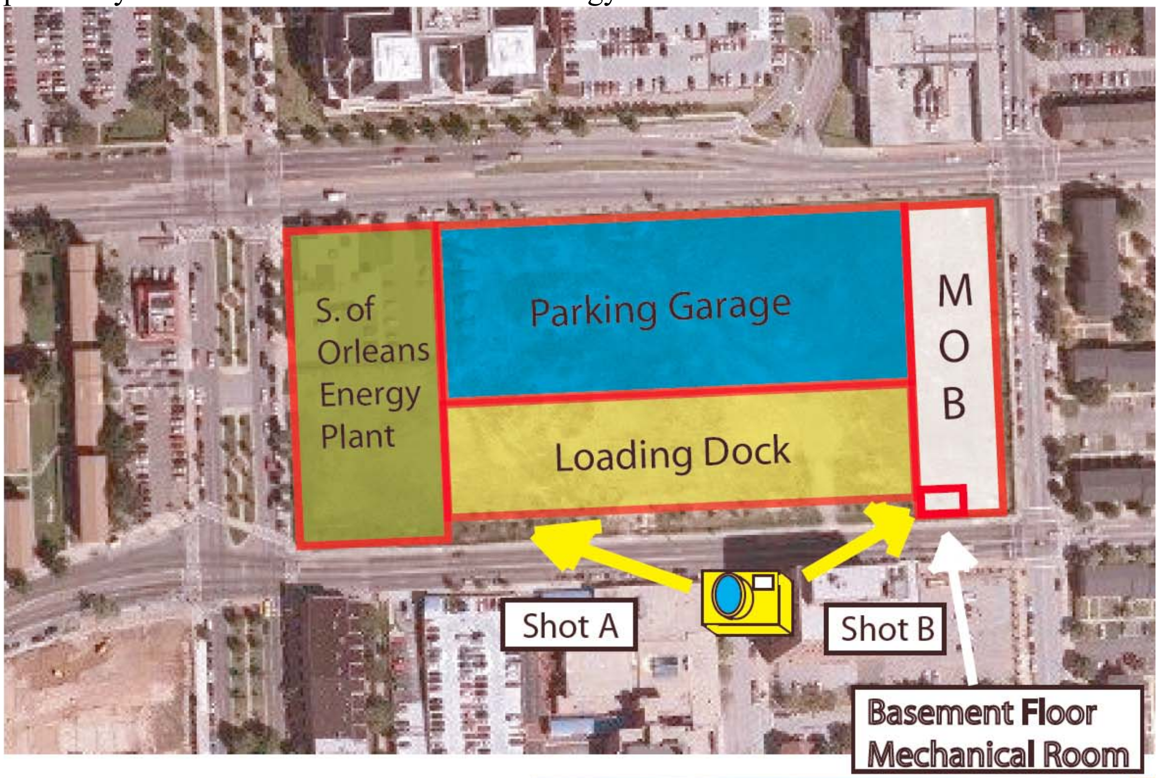
9.1 Overview of Supplying the MOB with Steam and Chilled Water

The central plant from which the MOB will draw its chilled water and steam from in the alternative #1 and #2 schemes is located across the block from the MOB. Between are two new buildings. These buildings are a parking garage on the Orleans St. side and a Loading Dock on the Fayette St. side.

The South of Orleans Energy Plant is approximately 450feet away from the MOB. The easiest way to access the MOB basement mechanical room is through the adjoining loading dock. The photo below shows the area where the MOBs southwestern corner will intersect with the loading dock. In this picture is a decrepit tree, to the left of that and in the background, is a condemned project, and in front of that is a bright yellow excavator. This excavator is approximately where the MOB's Basement main mechanical room is located. This is where the supply and return for the chilled water and steam need to enter and exit the building.

A better overview of the entire block is shown on the next page.

This is the entire block layout with two perspective pictures showing the relative proximity of the South of Orleans Street Energy Plant and the MOB.



9.2 <u>Site Limitations to Installation of Chilled Water and Steam Piping</u>

Installation of the piping raises very few site issues as compared a similar endeavor at a typical job site. This is because the entire block is being developed at the same time. The MOB was planned early enough to facilitate whatever solution was decided upon for the piping route. There are two options for locating the piping for steam and chilled water.

9.2.1 Option One: Buried Pipes

The entire block that the MOB is situated on was developed at the same time and from naught. This made it possible for Johns Hopkins Hospital to bury the steam and chilled water lines in what was at that time, an empty field.

Pros -No impact on planned loading dock.
-The Main Mechanical Room that the pipes need to meet is below grade, so the pipes are closer to their intended destination.
-Pipes are in no danger of damage from equipment in the loading dock.

Cons -Buried utilities may cause schedule delays.
-Access to the pipes will be limited by the overlying concrete slab.

9.2.2 Option Two: Pipes Running Along the Ceiling of the Loading Dock

Because the Loading Dock literally connects points A and B running the pipes, exposed through the loading dock is also an option. Because it an aesthetically unimportant building, to the point of having a barrier wall in front of it, the addition of the pipes would have minimal impact on the building.

Pros -Installation cost may be less expensive without having to deal with excavation.
-Avoids any possible problems with existing underground utilities.
-Pipes will be easily accessible.

Cons -Will necessitate additional fasteners and possibly structural changes to the loading dock.
-Pipes will need to be brought below ground through an area in construction to reach the main mechanical room in the MOB.
- There may be increased risk of damage to the pipes from the equipment activity in the loading dock.

9.3 Schedule Impact of Installation of Chilled Water and Steam Piping

The construction in question is installing piping from the South of Orleans Energy Plant to the MOB. The only length of the piping actually inside the MOB is the few feet entering the mechanical room and supplying the main air handler chilled water pump and the steam-water heat exchanger. For this reason schedule impacts will be limited to those incurred independently of the MOB itself. Such interruptions could include the following:
-With buried pipes, conflicting space demands with existing buried utilizes
-Poor weather impeding excavation of the buried pipe trench.
-Modification of the wall or ceiling structure to the loading dock to accommodate suspended piping.
-Space conflicts with the crew installing the suspended pipe and other tradesmen.

9.4 Conclusion on Constructability of Steam and Chilled Water Piping for the MOB

Overall, the construction of steam and chilled water piping from the South of Orleans Energy Plant to the MOB should be very simple. This is because the entire block has been designed and constructed at the same time, allowing for inclusion of such piping. The area between the MOB and the Energy Plant was originally an empty field able to accommodate buried piping. Suspended piping is also an option because the building between the Energy Plant and the MOB directly connects the two buildings. It is also a purely utilitarian structure and should be well able to accept or be modified to accept the piping.

Thesis Conclusion

In this report the MOB, a medical office building at Johns Hopkins Hospital, was redesigned to use a chilled water VAV system and a DOAS system with parallel active heated/chilled beams.

After the method of design was shown for the MOB both of the systems were evaluated for indoor air quality performance. It was found that the MOB would benefit in terms of indoor air quality from the introduction of a DOAS system, with its high rate of outdoor air supply. The evaluation of the MOB also showed both of the systems to be economically viable options with relatively short payback periods.

The buildings breadth topics analyzed the proposed systems for non-mechanical impact on the MOB. The constructability breadth topics showed that the supply and return piping for the chilled water and steam would not have been a problem to install. Additionally, the electrical supply analysis breadth topic showed that the reduction in electrical support equipment for Alternatives number one and two is significant.

Overall, both the chilled water VAV system and the DOAS application were found to be attractive options for redesign of the Johns Hopkins Hospital MOB.