FINAL REPORT

INTERDISCIPLINARY SCIENCE AND ENGINEERING BUILDING

UNIVERSITY OF DELAWARE CAMPUS
NEWARK, DE 19716

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Interdisciplinary Science & Engineering Building

Project Team

Owner:
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Architect:
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Project Information

Owner: University of Delaware
Locations: Newark Delaware
Size: 194,000 SF
Floors: 5 above grade (including penthouse)
Project budget: $140M
Construction Budget: $105M
Delivery Method: Design-Bid-Build

Lighting/electrical

- 2 primary service entrance feeders (34.5kV)
- Distribution throught building @ 480Y/277V
- Emergency power provided by campus engine generator system
- Lighting primarily flourescent (T5, T8, CF)
- Filtered Lenses or lamp sleeves for UV elimination in clean rooms
- Daylight harvesting for spaces w/ abudent natural light. (With photocell control of space light fixtures)

Structural

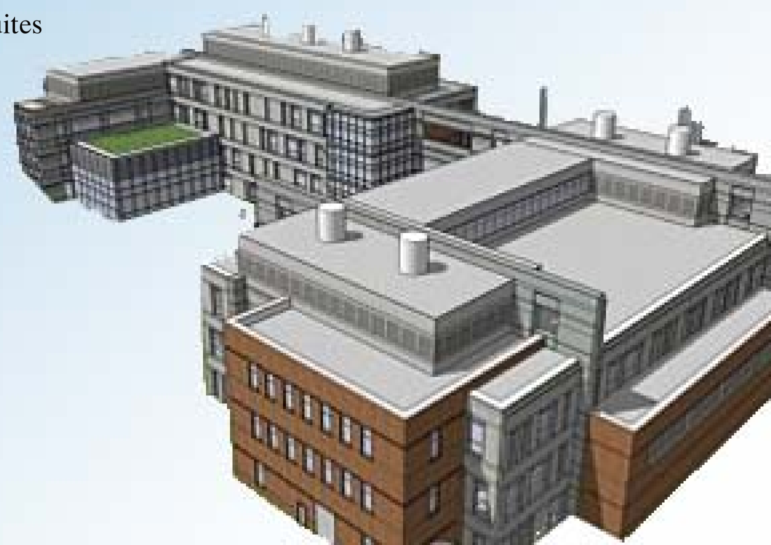
- Ordinary reinforced concrete shear walls
- Gravity system:
  * two-way slabs w/ edge beams
  * Flat plate & one-way slabs
  * Joists
- Strict vibrastion control needed for imaging suites (33-2,000 micro inches per second)
- Steel pedestrian bridge joins research and classroom wings

Mechanical

- Heating provided via steam-to-water HX's in builing fed from campus Central Utilities Plant(CUP)
- Cooling supplied by water-to-water flat plate HX's connected to the campus chilled water loop
- Electric drive standby chiller in basement for labs. 6 modules @ 50 tons each, 2 utilize hot gas bypass.
- 10 total AHU's reside in the buildings penthouses
- 7 AHU's are 100% OA with desicant wheels, and either enthalpy wheels or heat pipes depending on the space served
- High plume dilution type exhaust fans for laboratory exhaust

Architecture

- Building allows for connection between research and learning environments.
- East wing = Research Wing
- West wing = Classrooms/Office Wing
- Pedestrian brige joins the two environments (physicaly and symbolicaly)
- Building features:
  *Brick, stone, metal & glass facade
  *Green roof on west wing commons
  *Photo voltaic solar panels
  * Daylighting incorporated throughout bldg.



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http://www.engr.psu.edu/ae/thesis/portfolios/2011/jpp5060/index.html

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The University of Delaware

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1.1 Executive Summary

The fallowing report is based upon the design of the University of Delaware's new Interdisciplinary Science and Engineering building (ISEB). An in depth analysis of the building's mechanical system reveled that the owners and designers of ISEB left little room for improvement. One area that all laboratory buildings designers look to in order to save energy is the ventilation system. Although the current ventilation system implements some of the most current energy efficient design approaches, such as energy recovery wheels and variable flow fume hoods, this report proposes two alternatives methods of handling the laboratory ventilation air.

Although ISEB contains over 17 laboratory spaces, only the 8 instructional laboratory spaces and their accompanying prep rooms were analyzed for this report. This was done for two reasons; first, these instructional labs have lower hood densities than the research laboratories and therefore offer more advantages for the proposed alternative systems. Second, the air handling units serving these instructional labs also serve non-lab spaces, which also presents energy saving opportunities.

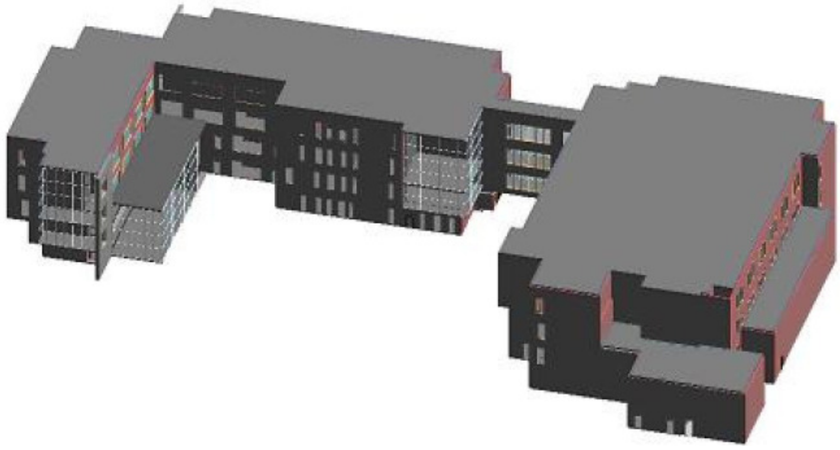
The first alternative method proposed in this report involves separating the lab and non-lab spaces to their own air handling systems, implementing a demand based air change rate system in the lab spaces, and implementing passive chilled beams to meet any sensible load that is not met by the (now lowered) ventilation air change rates. An energy simulation of this proposed alternative showed an annual reduction in the building chilled water consumption by 18,111 ton-hrs (2%), a reduction in steam consumption by 405 MBTU (15%), and a reduction in electrical consumption by 55,479 kW-hrs (1%).

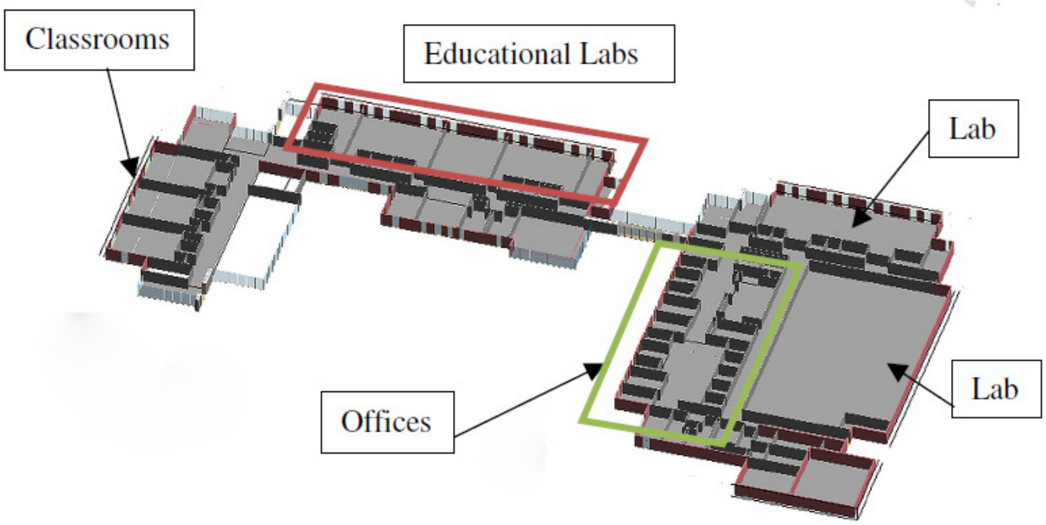
The second alternative method proposed for this report takes advantage of the fact that AHU's 3 and 4, which serve the instructional labs, also serve non-lab spaces. In this approach, room air from the non-lab spaces (offices, classrooms, and corridors) is transferred to the laboratories to reduce the amount of outdoor air brought in to the space to meet the minimum air change rate or fume hood requirements, whichever is greater. This method reduced the annual building chilled water use by 24,898 ton-hrs (2%), and steam use by 256 MBTU (5%).

When comparing the two alternative approaches it was concluded that moving the lab and non-lab spaces to their own air handling units along with implementing a demand based lab air change rate system would provide the most energy savings while maintaining good occupant safety. Analysis revealed that the use of the passive chilled beam system in redesign approach 1 actually increased chilled water and steam consumption while accounting for 52% of this approach's capitol cost. Although energy savings were also seen from alternative method 2, the amount of control needed to maintain proper space pressure relationships and prevent contaminant spread to the rest of the building would require a trained building operating staff and may not be feasible.

1.2 Building Information

The University of Delaware's new Interdisciplinary Science and Engineering building (ISEB) will be built on university property, will be approximately 194,000 square feet, and is scheduled for completion in Fall 2013. When complete, ISEB will facilitate both research and educational needs, each having its own wing of the building. The University of Delaware has seen a need to connect their classroom curriculum with what is going on in their research facilities. This connection will allow for real life problem based learning in order to tackle such issues as renewable energy and sustainability.





1.3 Architectural Information:

To delineate the two uses, the architect divided the building into two wings: An east (research) wing and a west (instructional) wing. These wings are joined together by a bridged walkway to maintain that bond between the classroom and research. The architect has meshed together brick, stone and glass to give an organic feel to a cutting edge building. The buildings day lighting, solar panels, and some rooftop vegetation give a hint to a passer by or an occupant that this building is one in which today's energy issues will be at the forefront.

The architect has incorporated many different materials and wall systems to give the building an organic feel and demonstrate its many uses. At the core of the building are the classrooms and laboratories, which can be identified by the red brick veneer. The interior spaces such as the offices, cafeteria, group study areas, and open offices have a more open feel and can be identified by the many different types of glass, stone, and metal wall assemblies. In total, the wall facades for this building include brick veneer, aluminum curtain wall, stone rain screen, insulated metal panel wall, and a non-operable aluminum window system.

1.3 Existing Mechanical System:

1.3.1 Water/Steam Side:

The building receives steam and chilled water from the East Campus Utilities Plant (ECUP). The steam is converted to hot water in steam-to-water heat exchangers, which provides the buildings heating requirements. The building utilizes a 160°F heating water supply temperature and a 120°F return temperature. Chilled water, from the University of Delaware's campus chilled water plant, enters the building at 43°F and is fed to a water-to-water, flat plate, heat exchanger to meet the buildings chilled water needs. An electric drive stand-by chiller is on site, in the basement mechanical room, and consists of 6 modules each sized at 50 tons (two of which incorporate hot gas bypass). The condenser heat from this chiller is recovered and injected into the buildings heating/reheat loops. Two fluid coolers with a nominal cooling capacity of 240 tons are on site to provide heat rejection from the standby chiller if the heating/reheat loads are low. (Schematics Provided in Appendix C)

1.3.2 Air Side:

There are ten air handling units (AHU's) serving the building, each of which is located in one of three mechanical penthouses. Each of these ten AHU's fall into one of two system types, either recirculating or 100 percent outdoor air.

Air handling units 1, 2, & 10 are of the recirculating air system type. They serve the builds classrooms, offices, common spaces, and corridors. Pressure independent, Variable Air Volume (VAV) terminal units are provided for each temperature control zone of the system. Each will be equipped with a hot water reheat coil to maintain space temperature. Because of the extreme variance in occupancy over a large span of operating hours, in spaces served by this type of unit, these systems are designed to minimize energy consumption through the use of unoccupied modes of operation. Supply fan volume control is accomplished through the use of Variable Frequency Controllers, which modulate fan speed (and air flow) to maintain a constant duct static pressure.

Once in the building, air is supplied to each zone by a variable air volume (VAV) terminal unit. The fan powered terminal units are equipped with hot water reheat coils. The use of reheat is minimized through the use of sequencing and will usually only occur once the terminal unit has reached its minimum setting.

The other seven AHU's (3, 4, 5, 6, 7, 8, & 9) are 100% outdoor air units, which serve the build's cleanroom, microscopy, research, and instructional labs. Six of the seven 100% outdoor air units contain some form of energy recovery, with the exception of AHU-9, which serves the building's clean room and contains no form of energy recovery. Enthalpy wheels are used for spaces in which contamination of the supply air from the exhaust air is not critical and heat pipes for the units in which supply air contamination cannot be risked.

Supply and exhaust air terminal units serving each lab space are pressure independent, single duct, and variable volume. All fume hoods in the building laboratories are variable volume, served by high plume, constant volume exhaust fans on the roof. (Schematics Provided in Appendix C)

1.4 Mechanical System Evaluation

1.4.1 ASHRAE Standard 62.1 Analysis:

After evaluating ISEB based on Standard 62.1 section 5, which covers building issues such as, prevention of mold growth, measures to prevent re-entry of contaminated air, and particulate filtration, it was concluded that it is fully compliant with all of the requirements in this section. Both the drawings and specifications indicate that the designers of ISEB consulted standard 5 when designing its ventilation, exhaust, and further HVAC systems. The mix use of laboratory and education of ISEB makes the need for proper ventilation and exhaust imperative.

The ventilation rate calculation procedure, set by Standard 62.1 section 6, was completed on all of the spaces throughout the building and the results were compared with the design supply airflow rates and outdoor airflow rates. After this analysis was completed it was concluded that every space and zone throughout the building meets, or surpasses, the requirements set by section 6.

1.4.2 ASHRAE Standard 90.1 Analysis:

When the MEP systems and building envelope were compared to ASHRAE standard 90.1 it became evident that energy conservation was at the forefront of ISEB's design. The building complied, or came close to complying, with almost every section of this standard. The fact that this building receives its heating and cooling from a central campus steam/chilled water plant makes compliance with 90.1 much easier because it alleviates a major portion of on site energy usage. Laboratory systems always pose problems for mechanical designers due to their complexity but they also breed some of the most innovative designs. From variable frequency drives, to DDC controls, and energy recovery systems, ISEB has made all of the right moves in order to be a energy conscience building. This building is going to cultivate many great minds and achievements in the field of sustainability, which is why the designers have gone to such great lengths to comply with standard 90.1 and make energy conservation a major identity of ISEB.

1.4.3 Load and Energy Simulation of Designed Mechanical System:

In order to properly evaluate any changes made to the current design of the ISEB's mechanical system it was important to model the current systems as accurately as possible. Trane's TRACE 700 program was used to run building load calculations and energy simulations. The design outdoor and indoor design conditions used for all simulations in this report are shown below in tables 1 and 2. All occupancy, lighting, and equipment schedules can be found in appendix A.

Table 1: Design Outdoor Air Conditions

ASHRAE HOF 2009 CH.14 APPENDIX	
Wilmington, DE	dB Temp
0.4% Cooling	93.1 °F
99.6% Heating	11.7 °F

Table 2: Indoor Space Conditions

Indoor Design Conditions		
	Winter	Summer
Lab Spaces	72 °F	72 °F
Non Lab Spaces	70 °F	75 °F

The TRACE peak-cooling load was 2% lower than the design peak-cooling load and the TRACE peak-heating load was less than 1% higher than the design peak-heating, as seen in table 3 below.

Table 3: Load Calculation Comparison

	Peak Cooling	Peak Heating
Design Docs.	1350 Tons	11,628 MBH
TRACE CALC	1323 Tons	11,633 MBH
Difference	2%	< 1%

The chilled water consumption calculated by TRACE was 19% greater while the steam consumption was 19% less than the design documents indicated. The building's electric, chilled water, and steam consumption calculated by the TRACE energy analysis program resulted in a yearly energy bill of $1,085,495, which is 11% higher than the cost calculated by the design team. The TRACE energy analysis results and the simulation ran by the design engineer are shown in table 4 below.

Table 4: Energy Simulation Comparison

	Electricity (kW-hr/yr)	Chilled Water (ton-hr/yr)	Steam (MBTUH/yr)	Total Cost/Year
Design Docs	6,998,096	946,640	6,958	$1,085,495
TRACE Sim	6,059,252	1,166,349	5,605	$964,507

1.4.4 Final Evaluation:

Through the use of TRACE load/energy simulations, LEED analysis, ASHRAE standard 90.1 analysis, and ASHRAE 62.1 analysis, it is easy to see that ISEB's mechanical systems are were designed with energy conservation as the goal. The owner and design teams have really put in the time and money to make sure this building will operate efficiently. One of the main design concerns with any laboratory type building is handling of the ventilation and/or make-up air. Conditioning outdoor air can prove to be very costly and designers have come up with unique of ways to reduce these costs. The current design of ISEB's airside mechanical systems implements some of the most cutting edge technology such as, enthalpy wheels, run around coils, variable flow fume hoods and occupancy sensors, yet there are other options that may be explored.

2.0 Mechanical System Redesign

Even though the current design of ISEB's ventilation systems employs some of the latest and most energy efficient approaches this report looks at two possible alternatives. The particular areas of interest for this report are the instructional lab air handling systems. These systems were chosen for analysis for two reasons:

1. These systems serve both lab and non-lab spaces. The requirements of the lab spaces (all air must be exhausted) means that all supply air to the non-lab spaces is currently 100% outdoor air. This is an area where excess air is being conditioned and brought into the building.
2. The Instructional Labs have lower hood densities and lower internal loads than the research laboratories, as well as less stringent environmental requirements.

2.1 Redesign Objectives:

- ✓ Reduce ventilation loads on air handling units 3 and 4.
- ✓ Maintain occupant safety in both lab and non-lab spaces
- ✓ Compare alternative approaches and weight pro's/con's

2.2 Alternative Approaches:

2.2.1 Approach 1:

This approach aims at reducing the ventilation load on the building by first separating AHU's 3 and 4 so that all lab spaces are on one 100% outdoor air unit while all of the non-lab spaces are on one recirculating type unit. This will allow all of the non-lab spaces currently on AHU 3 or 4 to receive only enough outdoor air to meet ventilation requirements based on ASHRAE standard 62.1, instead of receiving 100% outdoor air as currently designed. The next step to this approach is to implement a demand based control ventilation system for the instructional labs. This system will vary the air change rates to each lab independently based on room air "cleanliness" and/or hood airflow requirements. With demand based control of the ventilation system, there will now be times when the airflow needed to meet the sensible load in the lab spaces is greater than the air change rate being supplied to the space. Instead of conditioning outdoor air to meet these loads, a chilled beam system will be implemented in each lab space to achieve additional energy savings.

2.2.2 Approach 2:

A common misconception is that labs must be supplied with 100% outdoor but this is not the case. The code simply states that lab air may not be recirculated. This means that room air from “non-lab” spaces, like the ones on the same system as the instructional labs, may be routed to the lab spaces as make-up air. This approach takes advantage of the mixed space types on AHU’s 3 & 4 by transferring the air from the non-lab spaces to the labs. This “transfer air” then reduces the amount of outdoor air brought into the AHU as makeup air for the lab hoods.

3.0 Redesign Method 1: Demand Based ACH Rate Control

The fact that there are non-lab spaces, spaces in which room air may be recirculated (classrooms, offices, corridors), on a 100% outdoor air unit was one area identified as a possible source of energy savings. In building mechanical systems, excess outdoor air equals excess energy use. The first redesign method proposed aims at reducing the ventilation load on air handling units 3 and 4. It involves the fallowing design changes:

1. Move all non-lab spaces onto one air handling unit (AHU-3) and all lab spaces onto another (AHU-4).

This way, AHU-3 can now be a recirculating type unit, with outdoor air being brought in and conditioned only to meet the ventilation needs of the spaces and AHU-4 can remain a 100% outdoor air unit serving all of the lab spaces.

2. Implement a demand air change rate control system for the labs.

This system uses space sensors in the lab to sense any volatile particles that may be in the air and adjusts the exhaust/makeup airflow as needed. This demand- based control over the ventilation system can reduce the air change rates from the current 12/6 ACH (occupied/unoccupied) to as low as 4/2 ACH (occupied/unoccupied) depending on lab fume hood make-up air requirements.

3. With lower air change rates, there will be times when the airflow needed to meet the sensible (cooling) loads is greater than the airflow needed to ventilate the space. To deal with this situation, chilled beam systems will be implemented in the lab spaces to handle the cooling load.

Although fan coil units could have been used to meet the cooling load of these spaces, chilled beams are able to utilize a much warmer supply water temperature (58ºF) compared to fan coil units. This means that 60ºF return water from the AHU's can be mixed with 44ºF chilled water to obtain the 58ºF chilled beam supply water. Resulting in additional energy savings.

3.1 Air handling unit reassignment analysis:

3.1.1 Procedure:

TRANE TRACE was used to calculate the savings from moving the non-lab and lab spaces to separate air handling units. The fallowing procedure was used to model this scenario…

- All non-lab spaces on existing AHU's 3 & 4 were taken off these units.

- All lab spaces on AHU-3 were put onto AHU-4, fan affinity laws were used to resize static pressure and fan power for AHU-4.
- All non-lab spaces were put onto AHU-3. The heat pipe system for AHU-3 was removed.
- A TRACE simulation was conducted so that loads and energy usage due to the non-lab spaces (being supplied with 100% OA) could be recorded.
- All non-lab spaces on AHU-3 were assigned ventilation values based on ASHRAE 62.1 requirements, instead of being 100% OA as in the previous case
- A TRACE simulation was conducted so that new loads and energy usage for the non-lab spaces (now being supplied with only enough outdoor air to meet ventilation requirements) could be recorded and compared to the previous simulation.

Fig.1 : Existing Air systems (AHUs 3 and 4)

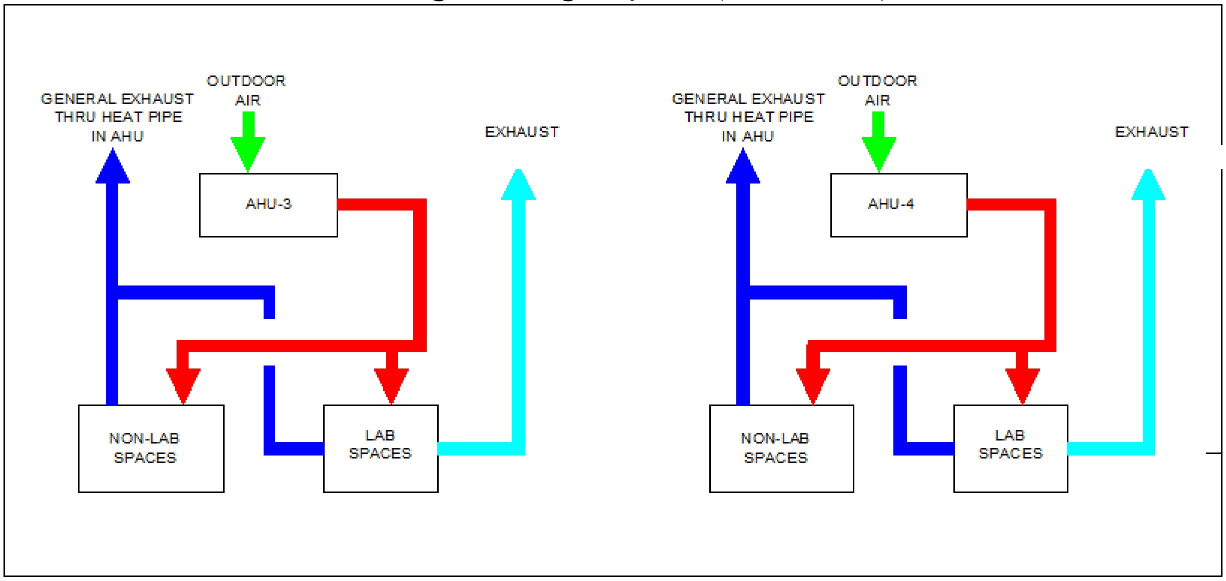


Fig.2 : Proposed Redesign Air systems (AHUs 3 and 4)

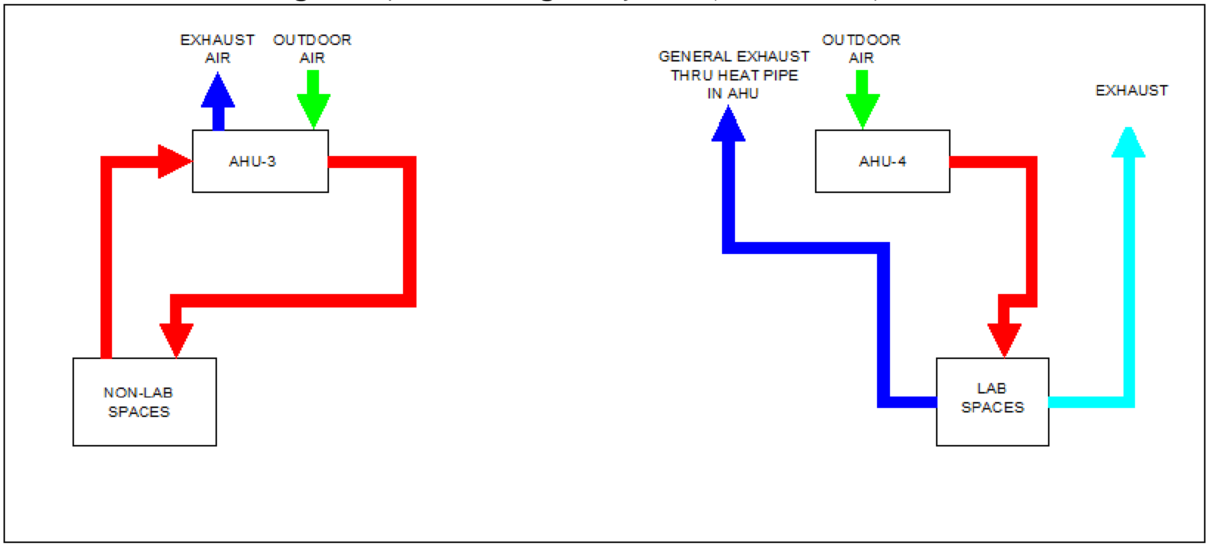
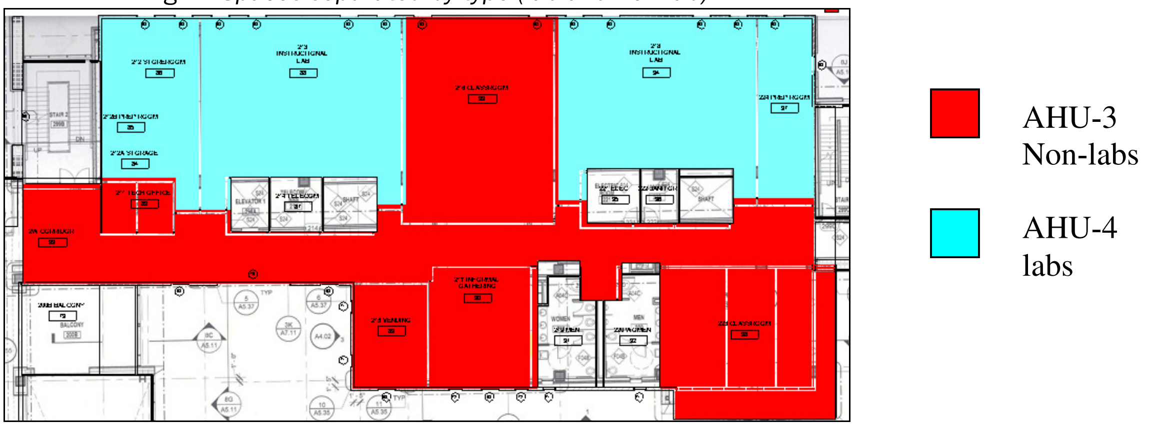


Fig.3: Lab and non-lab spaces on same AHU's

Fig. 4: Spaces separated by type (lab and non-lab)



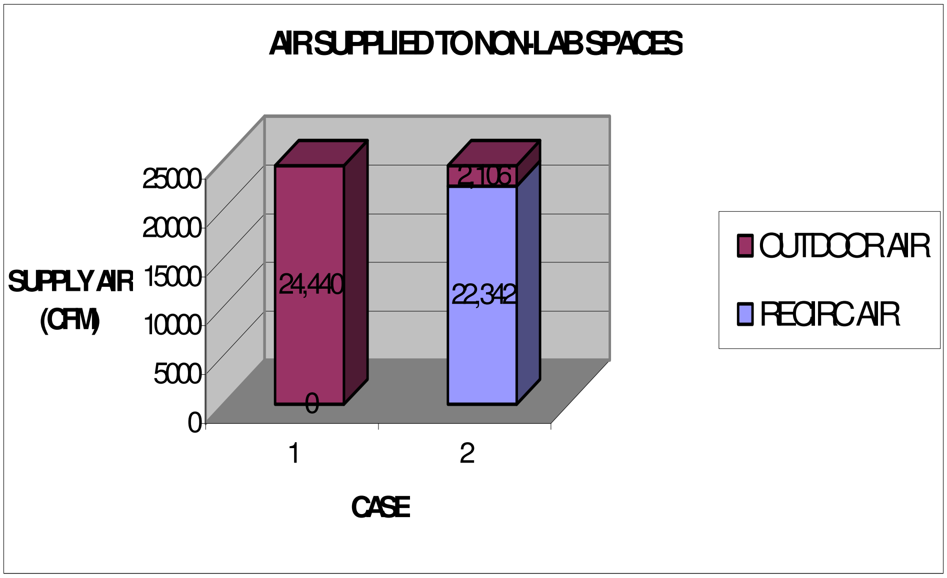
3.1.2 Results

Just as expected, separating the non-lab and lab spaces to different air handling units reduced the peak cooling and heating loads in the non-lab spaces but 38% and 32% respectively. These savings are a direct result of reducing the amount of outdoor air being brought into the air handling unit. These simulations revealed that at times of peak cooling load, these spaces were bringing in and conditioning 22,342 cfm of excess outdoor air (table 5 below).

Table 5: Load Reduction Due to AHU Reassignment

SIMULATION RESULTS		CFM	PEAK COOLING LOAD (TONS)	Peak HEATING LOAD (BTUH)	PERCENT OF COIL LOAD
AS 100% OA UNIT	TOTAL SYSTEM	24,440	134.4	1,713,818	---
	OUTDOOR AIR	24,440	134.4	1,713,818	100%
AS RECIRCULATING TYPE UNIT	TOTAL SYSTEM	22,342	51	553,761	---
	OA TO MEET 62.1	2,106	7.6	125,096	15%
	PEAK LOAD REDUCTION	-	-38%	-32%	

Fig. 5: Outdoor Air Reduction from AHU Reassignment



3.2 Demand Control ACH Rate System Analysis:

3.2.1 Lab Ventilation Background

Reducing the amount of ventilation air can result in large energy savings, but this has to be done while maintaining a safe lab environment. Historically, the way of ensuring "clean" lab air is to set minimum air change rate (sometimes with unoccupied or nighttime setback rates). These minimum rates are recommended by the NFPA and

OSHA, and can range from 6-20 ACH. The purpose of these minimum air change rates is to dilute any fugitive emissions that might be in the lab, but these dilutions rates are not substitution for fume hood performance. Often, the lab air is clean but the ventilation system is still pumping 12 air changes per hour into the space. When there is a spill however, these fixed air change rates are often lower than the amount of ventilation air needed to purge the room. A demand (or dynamic) air change rate allows the system to sense contaminant levels and adjust the air change rate according, offering energy savings when the lab air is clean and safety for times when a spill may require an air change rate higher.

3.2.2 Dynamic Air Change Rate (DACR) Application

A DACR system consists of the fallowing components…

- Air Sensors in each lab space
    - Collect Sample of Lab air ~ every 60 seconds
- Air data routers
    - Transport air sample to be Sensor Suite
- Sensor Suite
    - Collects air samples from all labs
- Vacuum Pump
    - Provides vacuum for transport of air samples from data routers to the sensor suite, through the structured cable
- Information Management Server
    - Communicates the air sample results to the building automated control system, knowledge center and to a web browser. With access to the browser interface, the laboratory air conditions can be tracked by the building owner, the management staff, or even the DCV manufacturer.

Fig. 6: Demand Based Air Change Rate System Component Diagram

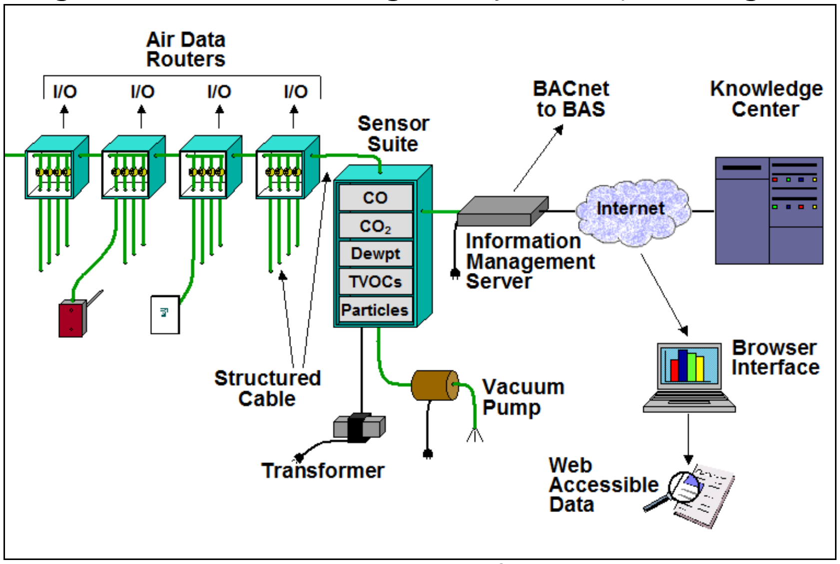


Illustration courtesy of Aircuity

FIG.7: Typical ISEB DACR System Layout Per Floor

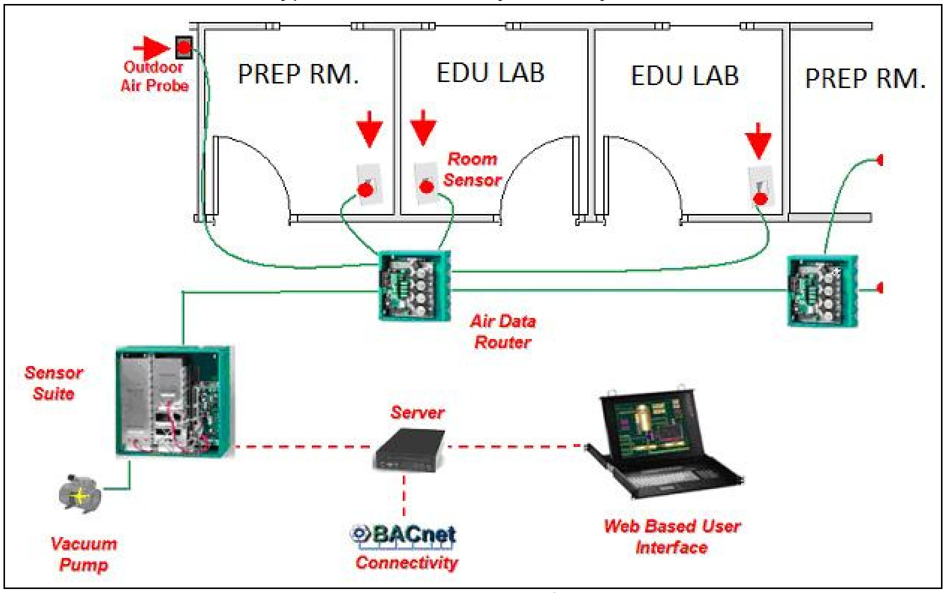


Illustration courtesy of Aircuity

3.2.3 Analysis Procedure

In order to analyze the energy savings from implementing a demand air change rate system the TRACE model created in the air handling unit reassignment procedure needed to be modified. The fallowing changes were made…

- Model labs and lab prep rooms in TRACE as currently designed; 12 ACH occupied and 6 ACH unoccupied. This schedule can be found in appendix A.
- Determine air change rates at maximum and minimum hood airflows for the lab spaces. Any reduction in airflow due to the Aircuity Optinet system will be modeled as a percentage of this maximum air change rate and be limited by the minimum rate.
- Create a new "Demand Based ACH Rate Schedule" and assign to labs and prep rooms in AHU-4 to simulate demand control ventilation. This schedule can be found in table of appendix A.
- Simulate load and energy use of AHU-4 with use of demand control System.
- Determine number of sensors, sensor location, and analyze control scheme.

The ventilation schedules modeled in TRACE prescribe the ventilation rate per hour as a percentage of the maximum daily air change rate. For the prep rooms, this maximum values was input as 4ACH, recommended by OSHA for high-risk spaces. For the lab spaces, the hood airflow dictates how low the air change rate may be throughout the day. Therefore, in order to input a maximum air change rate, which can be reduced by the demand control system when the hoods do not dictate, the maximum hood airflow for each room needed to be converted to an air change rate. Also, the minimum hood air change rates were calculated. These minimum hood air change rates dictate how low the proposed system may reduce the air change rates in the labs. The results of these calculations can be seen below in table 6.

3.2.4 Example Lab Room Air Change Rate Calculation

$$\frac{\text{room cfm}}{\text{(room volume)}} \times \frac{60 \text{ min.}}{1 \text{ hr.}} = \text{ACH}$$

Typical Lab Max Air Change Rate Dictated By Hood Needs:

Number of Hoods = 6

Hood Min Airflow = 300 cfm

Hood Max Airflow = 800 cfm → $\boxed{\begin{array}{l}\text{4.3 ACH min} \\ \text{11.5 ACH max}\end{array}}$

Room Max. Hood Flow = 6 x 800 cfm = 4800 cfm

Room Min Airflow = 6 x 300cfm = 1800 cfm

Room Volume = 25,120 ft^3

Table 6: Maximum air change rates (based on fume hood needs)

ROOM	Floor Area	HEIGHT	No. of Hoods	Max Hood ACH	Min Hood ACH
106 PREP	778	10	0	0.0	0.0
107 INSTRUC LAB	1570	16	6	11.5	4.3
112 INSTRUC LAB	1519	16	6	11.8	4.4
119 PREP	475	10	0	0.0	0.0
213 INTRUCTIONAL LAB	1562	16	6	11.5	4.3
218 INSTRUCTIONAL LAB	1484	16	6	12.1	4.5
224 PREP	419	10	0	0.0	0.0
312 GLASS WASH	835	10	0	0.0	0.0
313 INSTRUC LAB	1603	16	6	11.2	4.2
318 INSTRUC LBA	1641	16	6	11.0	4.1
324 PREP	358	10	0	0.0	0.0
406 PREP	791	10	0	0.0	0.0
407 INSTRUC LAB	1534	16	6	11.7	4.4
412 INSTRUC LAB	1511	16	6	11.9	4.5
418 PREP	340	10	0	0.0	0.0

* Rooms with ACH = 0 were assigned a minimum DCV airflow of 2 ACH as recommended by Aircuity

The maximum lab air change rates where then reduced, based on the occupancy schedule. The possible hourly reductions in the lab air change rates, due to the demand based control system, were compared to the base design of 12/6ACH (occupied unoccupied) and are shown below in figure 8.

Fig 8: Estimated Hourly ACH Rate Reductions From the Proposed System

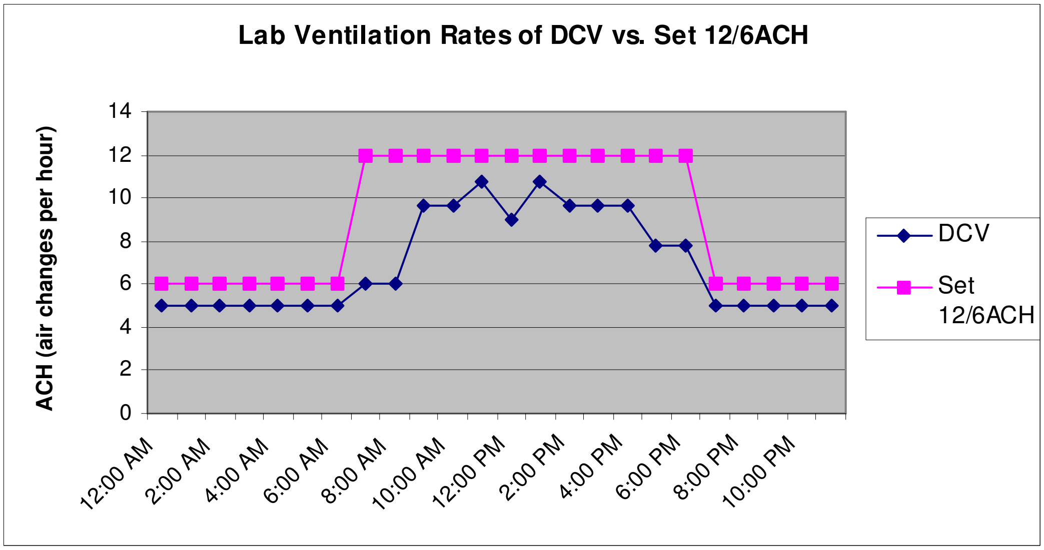


Table7 : Reduction in Lab Ventilation Air due to Demand Based Control

Weekday Time	Ventilation Air Reduction(CFM)
12:00 AM	3163
1:00 AM	3163
2:00 AM	3163
3:00 AM	3163
4:00 AM	3163
5:00 AM	3163
6:00 AM	3163
7:00 AM	19770
8:00 AM	19770
9:00 AM	7908
10:00 AM	7908
11:00 AM	3954
12:00 PM	9885
1:00 PM	3954
2:00 PM	7908
3:00 PM	7908
4:00 PM	7908
5:00 PM	13839
6:00 PM	13839
7:00 PM	3163
8:00 PM	3163
9:00 PM	3163
10:00 PM	3163
11:00 PM	3163

Table 7 above shows the ventilation air savings for each hour in the day, for an average day. The ability of the proposed system really depends on the fume hood use. If the fume hoods are used heavily throughout the day, then the space air change rates may not be able to reduce the amount of ventilation air to the levels shown in this study.

3.3 Passive Chilled Beam System Analysis:

With lower air change rates, there will be times when the airflow needed to meet the sensible (cooling) loads is greater than the airflow needed to ventilate the space. To deal with this situation a passive chilled beam system was proposed, in lieu of DX coil cooling, because the chilled beams utilize a much higher water temperature. This means that 60ºF return water from the AHU's can be mixed with 44ºF chilled water to obtain the 58ºF chilled beam supply water. The fallowing procedure was used to design the passive chilled beam system for each instructional lab.

3.3.1 Procedure:

- Obtain sensible loads for lab spaces from TRACE simulation.
- Determine minimum air change rate for each lab space. Although the Aircuity system allows for rates as low as 2 ACH, the minimum hood airflow of 300 cfm dictates the minimum air change rate in all lab spaces for this analysis. These minimum air change rates can be seen in appendix A.
- Determine Chilled Beam load at minimum ventilation rate.
- Use Halton HIT design software to size and layout Chilled Beam system.

3.3.2 Calculation Example:

Room 107 Instructional Lab:

Step 1) 6 Hoods @ 300cfm/hood (minimum flow) = 1800 cfm

Step 2) Supply Air Cooling Capacity = 1.08 x 1800cfm x (72-55) = 33,048 BTUH

Step 3) Chilled Beam Capacity = Room Sensible Load – 33,048 BTUH = 6,552 BTUH

Step 4) Use Halton HIT Design Software to Size/Layout Chilled Beam System for 6,552 BTUH.

Table 8: Chilled Beam Design Loads for Each Lab Space

ROOM	AREA	CEILING	Sensible	Total	Latent	SENSIBLE HEAT	MINIMUM	CHILLED BEAM
	S.F.	HEIGHT	BTUH	BTUH	BTUH	RATIO	AIRFLOW (CFM)	LOAD
107 INSTRUC LAB	1570	16	39600	43226	3626	0.92	1800	6552
112 INSTRUC LAB	1519	16	38468	42054	3586	0.91	1800	5420
213 INTRUCTIONAL LAB	1562	16	38828	41652	2824	0.93	1800	5780
218 INSTRUCTIONAL LAB	1484	16	37657	41236	3579	0.91	1800	4609
313 INSTRUC LAB	1603	16	40440	44040	3600	0.92	1800	7392
318 INSTRUC LBA	1641	16	41330	44938	3608	0.92	1800	8282
407 INSTRUC LAB	1534	16	38172	40991	2819	0.93	1800	5124
412 INSTRUC LAB	1511	16	38227	41861	3634	0.91	1800	5179

Fig 9: Illustration of Passive Chilled Beam

Fig 10: Convection through Passive Chilled Beam

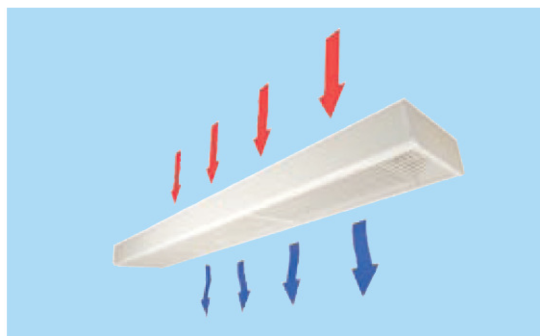


Fig.11: Example Halton Design Software Results for Instructional Lab 107

Cooling	CPA-75-1600-465-1		2009.04
Room:		Supply air flow rate	1800 cfm
Room size:	45.0 x 35.0 x 12.0 ft		1.1 cfm/ft^2
Occupied zone:	h=5.9 ft / dw=1.6 ft	Supply air temperature:	55.0 °F
Room air:	72.0 °F / 55 %	Jet outlet temperature:	67.2 °F
Heat gain:	-	Primary air capacity:	32908 Btu/h
Perforated ceiling:	-	Total pressure drop:	-
Installation height:	11.25 ft	Total sound pressure level:	-
Inlet water temperature:	58.0 °F	Total cooling capacity:	39841 Btu/h
Outlet water temperature:	60.0 °F		25 $Btu/h/ft^2$
Water flow rate:	6.92 gpm (9 x 0.77 gpm)	Dew point temperature:	54.9 °F
Coil capacity:	6933 Btu/h (9 x 770 Btu/h) 168 Btu/h/ft	Velocity control:	-
Water pressure drop:	3.1 "WC		

Velocity point	v3			
v	~45 fpm			
▲T	-3.1 °F			

vlim = 39.4 fpm

3.3.3 Handling of the Latent Load

The TRACE simulation of this proposed method resulted in the increase of both chilled water and steam loads. This is probably due to the fact that the use of chilled beams in any space requires stringent dew point control. If the dew point is allowed to rise above the temperature of the chilled beams (58^{o}F) then condensation will form on the beam. In dedicated outdoor air system (DOAS) applications, this latent load is usually handled by either an enthalpy wheel or a desiccant wheel. The system created for the simulation of the proposed case was modeled with cooling coil dehumidification because the requirements of the lab spaces do not allow the use of an air-to-air enthalpy wheel and natural gas used for reactivation of a desiccant wheel is more expensive than chilled water.

3.3.4 Results:

The theory behind the implementation of the passive chilled beams was that less chilled water would be used to meet part of the load, when the ventilation airflow to the space was low enough due to the ability to mix AHU return water with the chilled water. The simulation conducted for this report showed otherwise. The TRACE simulation of this proposed method resulted in the increase of both chilled water and steam loads. This is probably due in part to three reasons:

1) The inability to model the mixing of the 60^{o}F AHU return water with 44^{o}F building chilled water.
2) While there are significant energy savings that can be see on the cooling design day (T_{OA} = 95^{o}F), the amount chilled water needed to reduce the dew point in the room to an acceptable temperature is greater than the amount of chilled water saved by the chilled beams themselves.
3) In the TRACE model, both the labs and prep rooms were modeled as passive chilled beam systems. In such a system, the room with the lowest humidity ratio dictates the supply air humidity ratio. While the labs have relatively high supply airflow rates and do not require abnormally low humidity ratios (Appendix A), the prep rooms with low air change rates (2-4ACH thanks to the demand based ventilation system proposed) and high latent loads require a significantly low humidity ratio (table Appendix A) and therefore require more chilled water to dehumidify and more hot water to reheat the air.

3.4 Alternative Approach 1: Whole System Analysis

The TRACE simulation of total alternative approach 1 resulted in a reduction of building chilled water usage by 2% and a reduction of building steam usage by 15%, and a reduction in the building electrical consumption by 1%.

Table 9 : Energy Savings from Proposed Redesign Option 1

	Base	Option 1	Change	% Reduced
Chilled Water (ton-hrs)	1,166,349	1,148,238	-18,111	2%
Steam (MBTU)	5,605	4,775	-830	15%
Electricity (Kw-hr)	6,059,252	6,003,773	-55,479	1%

Redesign Approach 1 saves $27,790 a year in operating costs. With the cost of the addition fan coil units, chilled/hot water piping, and controls adding an additional $109,050 the simple payback period is 4.1 years.

Table 10 : Payback Period for Proposed Redesign Option 1

Cost	Base	Option 1
Chilled Water/yr	$115,935	$114,135
Steam/yr	$131,157	$114,175
Electricity/yr	$717,415	$710,846
Totals	$964,507	$936,717
Capitol Cost	$240,000	$349,050
Payback Period	4.1 Years	

3.4.1 Conclusions:

The goal of this proposed alternative approach was to reduce the amount of ventilation air, in turn reducing the total building energy consumption, while maintaining the safety of the occupants. The proposed air handling unit reassignment and demand based air change rate systems both reduced the amount of outdoor air significantly. However, the passive chilled beam system proposed did not reduce the amount of energy consumption but actually increased it. For this reason, it is concluded that the first two changes of this proposal are a viable redesign option, while the passive chilled beam system not.

4.0 Redesign Approach 2: Transfer Air

4.1 Transfer Air Analysis

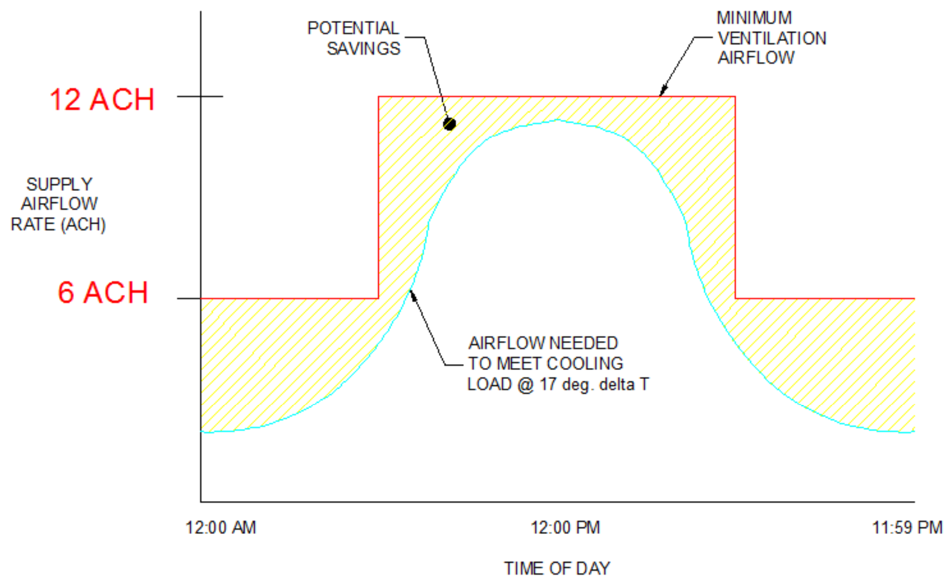
The instructional labs in ISEB have relatively large minimum air change rates (12ACH occupied/6ACH unoccupied). During times of occupancy this ventilation rate requires more supply air than is needed to cool/heat the space. This is illustrated in table 11 below. This means that the majority of the time these air handling units need to reheat the makeup air to avoid sub cooling the lab spaces. This is where the cost of such high ventilation rates really has an effect on operating costs, but the mixed-use characteristic of ISEB may help out with this problem.

Table 11 : Comparison of sensible load and ventilation airflows for time of maximum sensible load

ROOM	Sensible BTUH	CFM to Meet Sensible Load *	Occupied Ventilation CFM (12 ACH)	Excess Outdoor Air CFM	Excess Coil Load (tons)
107 INSTRUC LAB	39600	2157	5024	2867	5.9
112 INSTRUC LAB	38468	2095	4861	2766	5.7
213 INTRUCTIONAL LAB	38828	2115	4998	2884	6.0
218 INSTRUCTIONAL LAB	37657	2051	4749	2698	5.6
313 INSTRUC LAB	40440	2203	5130	2927	6.1
318 INSTRUC LBA	41330	2251	5251	3000	6.2
407 INSTRUC LAB	38172	2079	4909	2830	5.9
412 INSTRUC LAB	38227	2082	4835	2753	5.7
* Sensible load airflow based on (17oΔT)				Total	35.6
** Excess Coil Load based on design cooling outdoor air temp. of 95F and supply temp of 55F					

A common misconception is that air supplied to labs must be 100% outdoor air, but this is not the case. According to AIHA/ANSI Z9.5-2003, *Laboratory Ventilation*, the actual requirement is that "Air from laboratories shall not be recirculated." This means that room air from non-lab spaces (like the ones that are on the same system as the instructional labs) may be transferred to the labs as make-up air. In the case of ISEB, the room air from the non-lab spaces is exhausted anyway so transferring it to the lab spaces in order to reduce the amount of ventilation air seems like a good fit. This approach is particularly effective with labs that have high hood flow rates compared to their sensible load requirements, like the ones found in ISEB (Illustrated in fig. 12 below). This approach looks at transferring room air from the non-lab spaces to the lab spaces in order to reduce the amount of outdoor air brought into the lab spaces to meet ventilation requirements or as make-up air for the hoods.

Fig.12 : Potential Transfer Air Savings



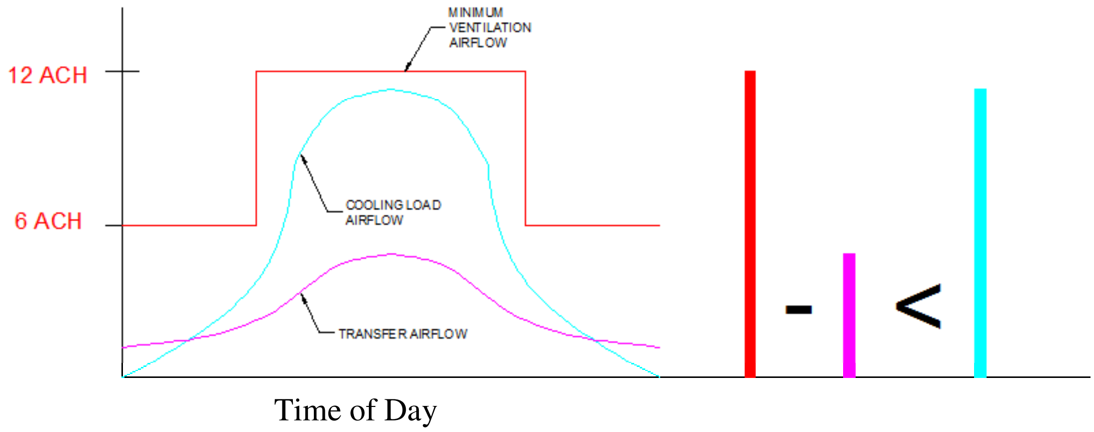
4.1.1 Procedure:

In order to determine the feasibility of the proposed system, the fallowing procedures were fallowed…

- Determine the amount of transfer air available from the non-lab spaces based on…
  - Room Sensible Loads
  - Room Ventilation Airflow Requirements (Based on Stand. 62.1)
  - Positive pressurization of the non-lab rooms compared to the lab spaces
- Determine the amount of Ventilation and/or Make-up Air needed in the labs based on…
  - Hood Airflows
  - Occupancy Schedule
  - Diversity

- Determine the airflow required to meet the sensible loads in the labs based on a 17°ΔT.
- Set up a simulation to determine the energy savings from the transfer air
- Size fan coil units to meet the sensible load for the times when the transferred air to the lab space reduces the outdoor air ventilation airflow below the airflow needed to cool the space, as in fig. below.

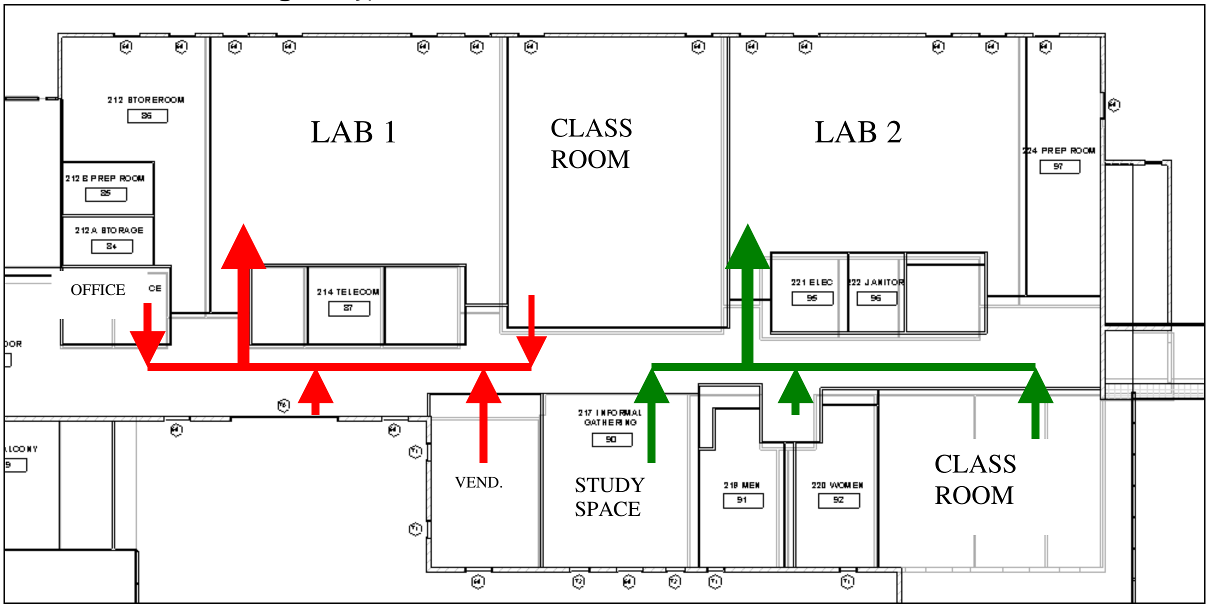
Fig.13 : Time when Lab Ventilation Airflow Minus Transfer Airflow Is Less than the Required Supply Airflow to Meet the Lab Cooling Load; ie: Load is Not Met and FCU is Required



4.1.2 Analysis:

The first step in designing the transfer air systems was to determine which rooms would transfer air to which labs. The rooms were divided so that there would be equal amounts of air available to each lab space, based on peak airflows for each non-lab space. The type of spaces and their location relative to the lab spaces made the amount of transfer air to each lab relatively equal (+/- ~500cfm). Figure 14 below shows the typical airflow pattern for each floor.

Fig14 : Typical Floor Transfer Airflow Pattern to Each Lab



After the transfer airflow systems had been determined, a simulation model was made to estimate the energy savings that this system could provide.

4.1.3 Simulation Set-up:

It was apparent that at the when the non-lab spaces were at peak internal load and the labs were at their maximum ventilation requirement that there was energy to be saved. The problem was, not only do the loads and ventilation rates of the labs vary, but the amount of transfer air available varies in response to the loads and ventilation requirements of the non-lab spaces. For this reason, a multivariable simulation needed to be constructed for each hour of the year. The equations and values of all variables were put into tables in EES, *Engineering Equation Solver*, and solved for each hour in the year. Tables of all variables can be found in Appendix A. The equations used for the simulation are as fallows…

Total Energy Savings from transferred air:

$$\dot{Q}_{saved} = \frac{\dot{V}_{TA} \cdot 1.08 \cdot \left[\left| T_{OA} - 55 \right| \right]}{12000} - \dot{Q}_{UNMET} - \dot{Q}_{TEMPDIFF} \qquad \text{(EQ. 1)}$$

Where:

V_{TA} = Volumetric flow rate of transfer air (cfm)

T_{OA} = Outdoor air temperature

Q_{unmet} = Load created when the transferred air reduces the amount of 55°F make-up air below the amount needed to cool the space. This load will be met by the fan coil units.

$Q_{tempdiff}$ = Load to account for the fact that the transfer air is 75°F which is 3 °F higher than the design room temperature.

EQ.1 is a simplified version of the equation input into EES. In the actual simulation an equation needed to be solved for both chilled water and hot water savings. The full set of equations from EES can be found in Appendix B.

$$\dot{V}_{TA} = \mathbf{Max}\left(V_{load} , V_{vent} \right) \cdot 0.95 \qquad \text{(EQ. 2)}$$

Where:

V_{load} = Volumetric flow rate needed to meet internal loads in non-lab spaces (cfm)

$$V_{load} = \frac{SL_{NONLAB}}{1.08 \cdot (75 - 55)} \qquad \text{(EQ. 3)}$$

Where:

SL_{nonlab} = Sensible Load in the non-lab spaces

75 = Design non-lab room temperature

55 = Supply air temperature to the non-lab spaces

$$V_{vent} = 0.06 \cdot A_{FLOOR} + 8 \cdot P_{non} \cdot Occ \qquad \text{(EQ. 4)}$$

V_{vent} = Volumetric flow rate to meet ASHRAE standard 62.1 requirements (cfm)

A_{floor} = Floor area of non-lab spaces

P_{non} = Number of people in the space (totaled over all of the non-lab spaces)

Occ = Occupancy schedule (Appendix A)

* Due to the fact that there are multiple spaces types comprising the "non-lab floor area" average values for ASHRAE standard 62.1 requirements were used in order to simulate the resulting ventilation rates as accurate as possible while reducing the number of variables in the calculation procedure. The cfm per occupant value used (8 cfm/occupant) was averaged over each occupied space type. The cfm per floor area value used (0.06 cfm/ft^2) was chosen in order to be conservative.

$$\dot{Q}_{UNMET} = \frac{1.08 \cdot (72 - 55) \cdot (\dot{V}_{labcooling} - (\dot{V}_{ACH} - \dot{V}_{TA}))}{12000} \quad \text{(EQ. 5)}$$

Where:

$V_{labcooling}$ = Airflow required to meet the sensible cooling load in the labs (cfm)

V_{ACH} = Ventilation airflow rates to the lab spaces (cfm)

72 = Design Lab air temperature (^oF)

55 = Supply air temperature to lab space (^oF)

* If the term ($V_{labcooling}$ – (V_{ACH} – V_{TA})) in equation 5 is positive, then there is too much air being transferred to the lab spaces and the sensible load is not being met due to the fact that not enough 55^{o}F supply air is being supplied to the space. If this term is negative then equation 5 is neglected from equation 1.

$$\dot{V}_{labcooling} = \frac{SL_{LAB}}{1.08 \cdot (72 - 55)} \quad \text{(EQ. 6)}$$

Where:

SL_{lab} = Sensible cooling load in the lab spaces (Btu/hr)

72= Design Lab air temperature (^oF)

55 = Supply air temperature to lab space (^oF)

$$BHP = \frac{\frac{\dot{V}_{TA}}{8} \cdot SP}{6356 \cdot Fan_{eff}}$$

$$KW = BHP \cdot 0.7457 \cdot 8$$

$$Fan_{eff} = 0.8$$

$$SP = 0.5 \cdot \left[\frac{\dot{V}_{TA}}{8 \cdot 2500}\right]^2$$

(EQ. 7,8,9 & 10)

Where:

BHP = Transfer fan Break horsepower

Kw = Energy consumption of all 8 transfer fans

Fan_{eff} = Fan Efficiency

8 x 2500 = 8 Fans X 2500 CFM ea.

4.1.4 Results:

The resulting transfer air quantities and chilled water/steam savings are shown below in figures 15, 16 and 17. The maximum transfer air quantity from the simulation was 22,500 CFM which is only 5% higher than the sum of the peak design airflows from the design documents which was 21,492 CFM. The peak non-lab internal load was within 1.3% of the TRACE simulation results and the peak lab internal cooling load was within 1.9% of the TRACE simulation results. The negative savings values in figures 15 and 16 represent the hours in the year when the auxiliary fan coil units must be used to condition the lab spaces. When analyzing the results it was apparent that these hours occurred during times of high lab internal load and moderate outdoor air temperature (~53 ^oF to 57^oF). During these times it would actually be more beneficial to bring in the outdoor air and condition it to 55ºF rather than transferring the 75ºF air.

Fig. 15 : Chilled Water Savings from Transfer Air Approach

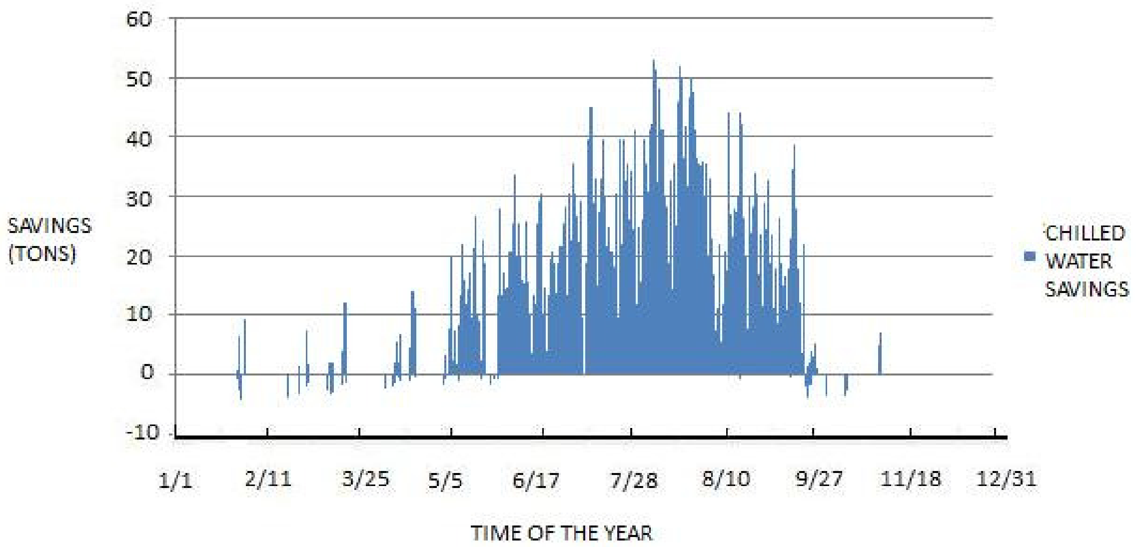


Fig. 16 : Steam Savings from Transfer Air Approach

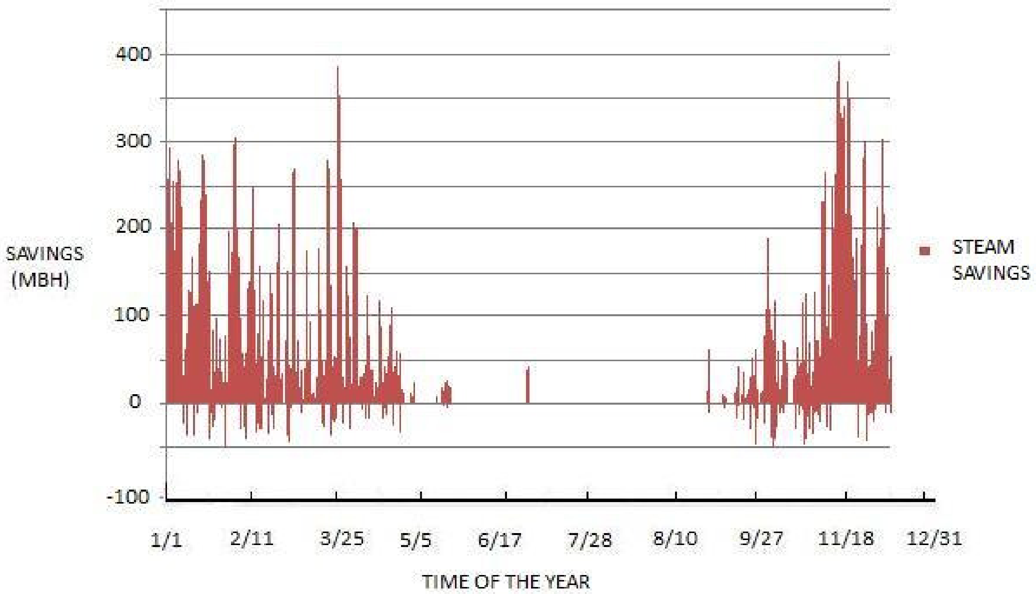


Fig. 17 : Amount of Transferred Air by Hour

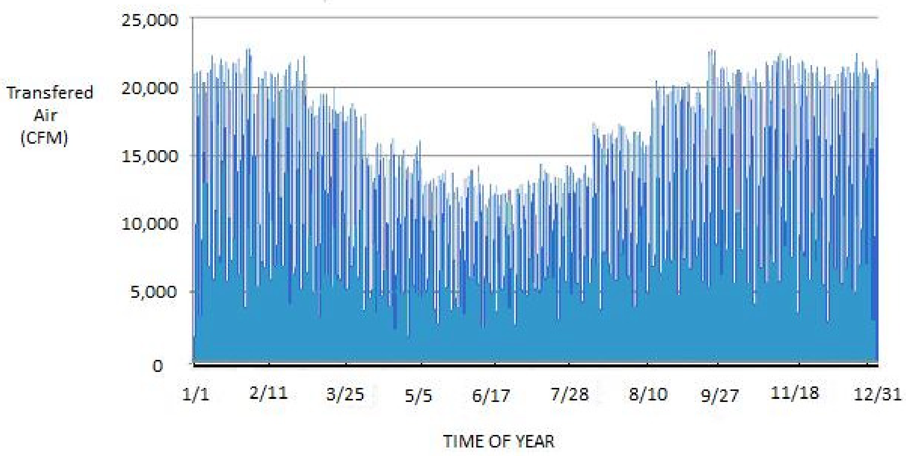


Table 12 : Simulation Model Accuracy

	Peak Non-Lab Airflow	Peak Non-lab Sensible Load	Peak Lab Sensible Load
TRACE SIM.	21,492 CFM	43.3 tons	26.1 tons
Simulation	22,500 CFM	43.9 tons	26.5 tons
ERROR	4.5%	1.3%	1.9%

4.2 Alternative Approach 2: System Analysis

Simulation of the transfer air redesign method resulted in a reduction of building chilled water usage by 2% and a reduction of building steam usage by 5%.

Table 13 : Energy Savings From Redesign Approach 2

	Base	Option 2	Change	Reduction %
Chilled Water (ton-hrs)	1,166,349	1,141,451	-24,898	2%
Steam (MBTU)	5,605	5,349	-256	5%
Electricity (Kw-hr)	6,059,252	6,050,256	-8,996	0%

Option 2 saves $14,007 a year in operating costs. With the cost of addition fans, fan coil units, chilled/hot water piping, and controls adding an additional $46,877 the simple payback period is 3.3 years.

Table 14 : Payback Period for Redesign Approach 2

Cost	Base	Option 2
Chilled Water/yr	$115,935	$112,470
Steam/yr	$131,157	$121,680
Electricity/yr	$717,415	$716,350
Totals	$964,507	$950,501
Capitol Cost	$240,000	$286,887
Payback Period	3.3 Years	

4.2.1 Conclusions:

The goal of this proposed system was to take advantage of the mixed use of this section of the building to reduce the chilled water and steam loads due to ventilation air. Although simulation in TRACE was not possible, basic energy equations and affinity laws were used to model this system to within 5% of the predicted values. While this proposed system does save energy, and offers a relatively short payback period, further investigation into the control strategy is needed. Pressure relationships between the lab and non-labs spaces must be carefully monitored in order to prevent any possibility of airflow from the labs to the non-lab spaces.

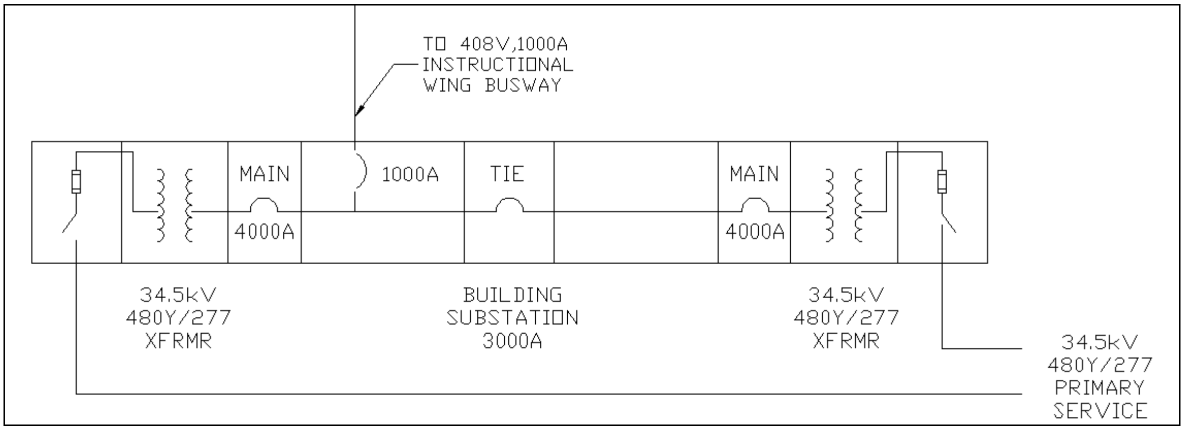
5.0 Electrical Analysis

Although energy use is a big factor is mechanical system design, the effects on the electrical system have to be considered. In order to approach the mechanical redesigns presented in this report in a holistic way, the effects of adding equipment and resizing equipment on the building electrical system was analyzed.

5.1 Existing Conditions:

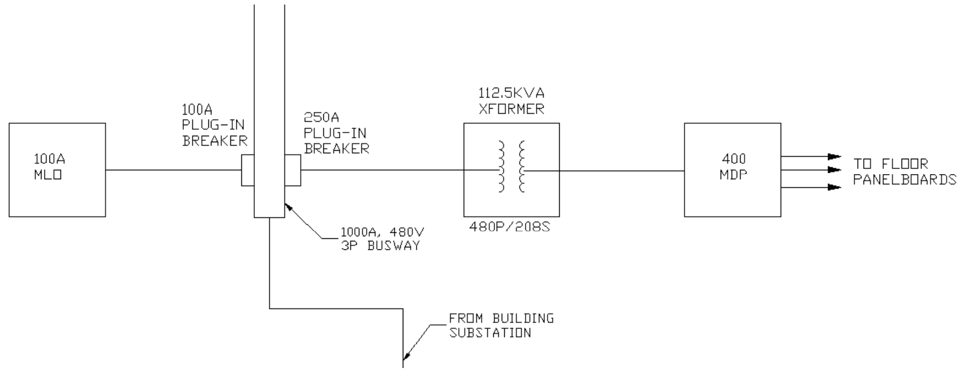
Electricity enters ISEB by two 34.5 kV, medium voltage, service entrance feeders. These feeders serve a dual primary-secondary doubled ended unit substation. Two medium voltage fused switches, on each side of the substation, allow for selection of either service feeder. The indoor substation transformers are rated at 1500/1995 kVA and are 34.5kV delta primary, 480Y/277 volt secondary. These transformers feed the 3000A main-tie-main distribution section.

Fig18. :Building Service Entrance Diagram



Electrical distribution throughout the building is at 480Y/277 volt. Busway systems feed vertically through stacked electrical rooms on each floor of each section of the building. Lighting and receptacle panel boards are fed from 480volt – 208Y/120 step down transformers located in each electrical room.

Fig.19 :Typical Instruction Wing Floor Electrical System Diagram



5.2 Option 1 Electrical:

Mechanical redesign option 1 incorporates the addition of fain coil units and dynamic ventilation controls to the existing design. This equipment adds additional electrical load to the building electrical system and must be taken into consideration. Each of the eight fan coil units implemented in the prep rooms contains two blower fans, each with its own motor. Also, electric heating was selected for these fan coil units, due to the small and infrequent heating loads in these spaces, and therefore must be met by the building electrical system. Not only does the new equipment need to be incorporated into the existing electrical system, but any existing equipment that was resized must be re-evaluated. The equipment re-evaluated in this electrical study were the supply fans for air handling units 3 & 4, due to the fact that these units were downsized. In order to determine the additional electrical load created by each piece of equipment the fallowing information needed to be obtained and calculated…

- Horsepower and/or Amp draw
  - ⇒ Manufacturer Literature
- Motor Full Load Amps (FLA)
  - ⇒ NEC TABLE 430-148

- Minimum Circuit Ampacity
  - ⇒ MCA = 1.25 * FLA
- Conductor Size
  - ⇒ Based on MCA
  - ⇒ NEC TABLE 310-16 (75oC, Copper, THWN)
- Overcurrent Device
  - ⇒ NEC TABLE 250-95
- Conduit Size
  - ⇒ NEC CH.9 TABLES 3A & 3B

Once all of the above information was obtained for each blower motor and electric heater, the information was organized into a table (table 15 below) for analysis. The next step in the process was to calculate the load created by each piece of equipment so that a panel board could be selected/sized. The fallowing equations were used to determine the equipment loads.

1 Phase:

Load (VA) = FLA x Voltage

3 Phase:

Load (VA) = FLA x Voltage x 1.73

The load for each piece of equipment was calculated and summed. When this was done the total load from the addition of the blower fans and heating coils was determined to be 26,704 VA.

Then, in order to determine size of the panel board needed to house these loads, this total load (VA) was divided by the three-phase voltage serving the system (208V).

Panel Ampacity = Total VA / (208*1.73)

This resulted in a panel load of 74.2 Amps (table 16 below). In order to handle this load (plus a 10% growth factor) a 100 Amp, 30-pole panel board was selected. A Main Circuit Breaker (MCB) type panel was selected due to the fact that this panel will be fed from a circuit breaker in a main distribution panel. The panel board layout can be seen below in figure 20.

Table 15 :Redesign Approach 1 Equipment Electrical Information

	Motor				Controller	Disconnect		Wiring			
EQUIPMENT	HP	Volts	Phase	FLA	Type	Size	Type	CCT	MCA	CB	Panel
SUPPLY FAN-3	10	480	3	14	NEMA 1	30	CB	3-#10 + 1#10G, IN 1/2"C	17.5	30	SM5C
SUPPLY FAN-4	10	480	3	14	NEMA 1	30	CB	3-#10 + 1#10G, IN 1/2"C	17.5	30	SM5C
FCU-1 BM1	1/4	115	1	5.8	NEMA 00	30	NFD	2#12 + 1#12G, IN 1/2" C	13.05	20	PJP
FCU-1 BM2	1/4	115	1	5.8							
FCU-2 BM1	1/4	115	1	5.8	NEMA 00	30	NFD	2#12 + 1#12G, IN 1/2" C	13.05	20	PJP
FCU-2 BM2	1/4	115	1	5.8							
FCU-3 BM1	1/4	115	1	5.8	NEMA 00	30	NFD	2#12 + 1#12G, IN 1/2" C	13.05	20	PJP
FCU-3 BM2	1/4	115	1	5.8							
FCU-4 BM1	1/4	115	1	5.8	NEMA 00	30	NFD	2#12 + 1#12G, IN 1/2" C	13.05	20	PJP
FCU-4 BM2	1/4	115	1	5.8							
FCU-4 BM1	1/4	115	1	5.8	NEMA 00	30	NFD	2#12 + 1#12G, IN 1/2" C	13.05	20	PJP
FCU-5 BM2	1/4	115	1	5.8							
FCU-6 BM1	1/4	115	1	5.8	NEMA 00	30	NFD	2#12 + 1#12G, IN 1/2" C	13.05	20	PJP
FCU-6 BM2	1/4	115	1	5.8							
FCU-7 BM1	1/4	115	1	5.8	NEMA 00	30	NFD	2#12 + 1#12G, IN 1/2" C	13.05	20	PJP
FCU-7 BM2	1/4	115	1	5.8							
FCU-8 BM1	1/4	115	1	5.8	NEMA 00	30	NFD	2#12 + 1#12G, IN 1/2" C	13.05	20	PJP
FCU-8 BM2	1/4	115	1	5.8							

EQUIPMENT	Volts	Phase	Amps	Control	CCT	MCA	CB	Panel
FCU ELEC. HEATER-1	120	1	16.7	T ' STAT	2#10 + 1#10G, IN 1/2"C	16.7	30	PJP
FCU ELEC. HEATER-2	120	1	16.7	T ' STAT	2#10 + 1#10G, IN 1/2"C	16.7	30	PJP
FCUELEC. HEATER-3	120	1	16.7	T ' STAT	2#10 + 1#10G, IN 1/2"C	16.7	30	PJP
FCU ELEC. HEATER-4	120	1	16.7	T ' STAT	2#10 + 1#10G, IN 1/2"C	16.7	30	PJP
FCU ELEC. HEATER-5	120	1	16.7	T ' STAT	2#10 + 1#10G, IN 1/2"C	16.7	30	PJP
FCU ELEC. HEATER-6	120	1	16.7	T ' STAT	2#10 + 1#10G, IN 1/2"C	16.7	30	PJP
FCU ELEC. HEATER-7	120	1	16.7	T ' STAT	2#10 + 1#10G, IN 1/2"C	16.7	30	PJP
FCU ELEC. HEATER-8	120	1	16.7	T ' STAT	2#10 + 1#10G, IN 1/2"C	16.7	30	PJP

Once the panel board was selected the feed wires to the panel needed to be size. First, table 310-16 of the NEC handbook was used to size the conductors. Next the conduit was sized using NEC table C1 in Appendix C. Table 17 below shows the results of this process.

Table 16 :Redesign Approach 1 Equipment Loads

	Motor				Load
EQUIPMENT	HP	Volts	Phase	FLA	(VA)
FCU-1 BM1	1/4	115	1	5.8	667
FCU-1 BM2	1/4	115	1	5.8	667
FCU-2 BM1	1/4	115	1	5.8	667
FCU-2 BM2	1/4	115	1	5.8	667
FCU-3 BM1	1/4	115	1	5.8	667
FCU-3 BM2	1/4	115	1	5.8	667
FCU-4 BM1	1/4	115	1	5.8	667
FCU-4 BM2	1/4	115	1	5.8	667
FCU-4 BM1	1/4	115	1	5.8	667
FCU-5 BM2	1/4	115	1	5.8	667
FCU-6 BM1	1/4	115	1	5.8	667
FCU-6 BM2	1/4	115	1	5.8	667
FCU-7 BM1	1/4	115	1	5.8	667
FCU-7 BM2	1/4	115	1	5.8	667
FCU-8 BM1	1/4	115	1	5.8	667
FCU-8 BM2	1/4	115	1	5.8	667
EQUIPMENT		Volts	Phase	Amps	Load (VA)
FCU ELEC. HEATER-1		120	1	16.7	2004
FCU ELEC. HEATER-2		120	1	16.7	2004
FCUELEC. HEATER-3		120	1	16.7	2004
FCU ELEC. HEATER-4		120	1	16.7	2004
FCU ELEC. HEATER-5		120	1	16.7	2004
FCU ELEC. HEATER-6		120	1	16.7	2004
FCU ELEC. HEATER-7		120	1	16.7	2004
FCU ELEC. HEATER-8		120	1	16.7	2004
				Total	26704
				Panel Size (Amps) @ 208, 3P	**74.2**

Table 17 :Panel board Feeder Information

Conductor Type	Temp Rating	Ampacity	Conductor Size
THWN	75	74 amps	# 4

Conductor sizing:

Conductors = 3 phase wires, 1 neutral

4 - #4 conductors = 1-1/4" conduit

Fig. 20:New Panel PJP Layout

PANELBOARD: PJP				BUS RATING	100A								MAIN: 3P 100A MCB
MIN AIC: 35,000				VOLTAGE:	208Y/120V								PHASE(S): 3
ENCLOSURE: NEMA 1				MOUNTING:	SURFACE								WIRES: 4
LOCATION:				NOTES:									
						LOAD, VA							
BRANCH CIRCUIT DESIGNATION	P	TRIP	CKT#	ϕA		ϕB		ϕC		CKT#	TRIP	P	IRCUIT DESIGNATION
BM-1	1	20	1	1354	1354					2	20	1	BM-5
BM-2	1	20	3			1354	1354			4	20	1	BM-6
BM-3	1	20	5					1354	1354	6	20	1	BM-7
BM-4	1	20	7	1354	1354					8	20	1	BM-8
Heater 1	1	30	9			2004	2004			10	30	1	Heater 5
Heater 2	1	30	11					2004	2004	12	30	1	Heater 6
Heater 3	1	30	13	2004	2004					14	30	1	Heater 7
Heater 4	1	30	15			2004	2004			16	30	1	Heater 8
SPACE	–	–	17					0	0	18	–	–	SPACE
SPACE	–	–	19	0	0					20	–	–	SPACE
SPACE	–	–	21			0	0			22	–	–	SPACE
SPACE	–	–	23					0	0	24	–	–	SPACE
SPACE	–	–	25	0	0					26	–	–	SPACE
SPACE	–	–	27			0	0			28	–	–	SPACE
SPACE	–	–	29					0	0	30	–	–	SPACE
PHASE CONNECTED LOAD, VA				9424		10724		6716					
PHASE BALANCE				35.08%		39.92%		25.00%					
TOTAL CONNECTED LOAD, VA		26864											
FUTURE GROWTH - 10%		2686											
TOTAL + FUTURE LOAD, VA		29550											
TOTAL CURRENT, A		82											

Once the new equipment loads had been analyzed and the panel board had been sized, I needed make sure the main distribution panel had enough spare capacity for the new panel. The loads from each phase on panel PJP were added to the main distribution panel DRGC to determine if there was enough capacity. Once the loads were added and a new amperage was calculated it was determined that the new equipment loads did fit onto DRGC, while leaving 5% future growth.

Table 18 :New Calculated Load on Main Distribution Panel DRGC

	MDP DRGC
Location	Elec Rm 115
Size	400 Amps
Previous Load	110,082 VA
Added Load	26,704
New Amp Draw	379 Amps
Okay?	Yes
Future Growth	5%

5.2 Option 2 Electrical:

Mechanical redesign option 2 incorporates the addition of transfer fans on each floor, as well as fan coil units with electric heaters. The same process that was conducted for redesign option 1 was conducted for option 2 and the results were as fallows.

Table 19 :Redesign Approach 2 Equipment Electrical Information

	Motor				Controller	Disconnect		Wiring			
EQUIPMENT	HP	Volts	Phase	FLA	Type	Size	Type	CCT	MCA	CB	Panel
Transfer Fan 1	1/3	115	1	7.2	VFD	30	NFD	3#14 + 1#14, IN 1/2"C	9.0	15	PJP2
Transfer Fan 2	3/4	208	3	3.1	VFD	30	NFD	3#14 + 1#14, IN 1/2"C	3.9	15	PJP3
Transfer Fan 3	1/3	115	1	7.2	VFD	30	NFD	3#14 + 1#14, IN 1/2"C	9.0	15	PJP4
Transfer Fan 4	1/2	208	3	2.2	VFD	30	NFD	3#14 + 1#14, IN 1/2"C	2.8	15	PJP5
Transfer Fan 5	1/2	208	3	2.2	VFD	30	NFD	3#14 + 1#14, IN 1/2"C	2.8	15	PJP6
Transfer Fan 6	1/3	115	1	7.2	VFD	30	NFD	3#14 + 1#14, IN 1/2"C	9.0	15	PJP7
Transfer Fan 7	1/2	208	3	2.2	VFD	30	NFD	3#14 + 1#14, IN 1/2"C	2.8	15	PJP8
Transfer Fan 8	1/3	115	1	7.2	VFD	30	NFD	3#14 + 1#14, IN 1/2"C	9.0	15	PJP9
FCU-1	1/2	208	3	2.2	NEMA 00	30	NFD	3#14 + 1#14, IN 1/2"C	2.8	15	PJP10
FCU-2	1/2	208	3	2.2	NEMA 00	30	NFD	3#14 + 1#14, IN 1/2"C	2.8	15	PJP11
FCU-3	1/2	208	3	2.2	NEMA 00	30	NFD	3#14 + 1#14, IN 1/2"C	2.8	15	PJP12
FCU-4	1/2	208	3	2.2	NEMA 00	30	NFD	3#14 + 1#14, IN 1/2"C	2.8	15	PJP13
FCU-5	1/2	208	3	2.2	NEMA 00	30	NFD	3#14 + 1#14, IN 1/2"C	2.8	15	PJP14
FCU-6	1/2	208	3	2.2	NEMA 00	30	NFD	3#14 + 1#14, IN 1/2"C	2.8	15	PJP15
FCU-7	1/2	208	3	2.2	NEMA 00	30	NFD	3#14 + 1#14, IN 1/2"C	2.8	15	PJP16
FCU-8	1/2	208	3	2.2	NEMA 00	30	NFD	3#14 + 1#14, IN 1/2"C	2.8	15	PJP17
EQUIPMENT		Volts	Phase	Amps	Control		CCT		MCA	CB	Panel
FCU ELEC. HEATER-1		120	1	16.7	T' STAT		2#10 + 1#10G, IN 1/2"C		16.7	30	PJP17
FCU ELEC. HEATER-2		120	1	16.7	T' STAT		2#10 + 1#10G, IN 1/2"C		16.7	30	PJP18
FCUELEC. HEATER-3		120	1	16.7	T' STAT		2#10 + 1#10G, IN 1/2"C		16.7	30	PJP19
FCU ELEC. HEATER-4		120	1	16.7	T' STAT		2#10 + 1#10G, IN 1/2"C		16.7	30	PJP20
FCU ELEC. HEATER-5		120	1	16.7	T' STAT		2#10 + 1#10G, IN 1/2"C		16.7	30	PJP21
FCU ELEC. HEATER-6		120	1	16.7	T' STAT		2#10 + 1#10G, IN 1/2"C		16.7	30	PJP22
FCU ELEC. HEATER-7		120	1	16.7	T' STAT		2#10 + 1#10G, IN 1/2"C		16.7	30	PJP23
FCU ELEC. HEATER-8		120	1	16.7	T' STAT		2#10 + 1#10G, IN 1/2"C		16.7	30	PJP24

Table 20 :Redesign Approach 2 Equipment Load

	Motor				Load
EQUIPMENT	**HP**	**Volts**	**Phase**	**FLA**	**(VA)**
Transfer Fan 1	1/3	115	1	7.2	828
Transfer Fan 2	3/4	208	3	3.1	1116
Transfer Fan 3	1/3	115	1	7.2	828
Transfer Fan 4	1/2	208	3	2.2	792
Transfer Fan 5	1/2	208	3	2.2	792
Transfer Fan 6	1/3	115	1	7.2	1432
Transfer Fan 7	1/2	208	3	2.2	792
Transfer Fan 8	1/3	115	1	7.2	828
FCU-1	1/2	208	3	2.2	792
FCU-2	1/2	208	3	2.2	792
FCU-3	1/2	208	3	2.2	792
FCU-4	1/2	208	3	2.2	792
FCU-5	1/2	208	3	2.2	792
FCU-6	1/2	208	3	2.2	792
FCU-7	1/2	208	3	2.2	792
FCU-8	1/2	208	3	2.2	792
EQUIPMENT		**Volts**	**Phase**	**Amps**	**Load (VA)**
FCU ELEC. HEATER-1		120	1	16.7	2004
FCU ELEC. HEATER-2		120	1	16.7	2004
FCUELEC. HEATER-3		120	1	16.7	2004
FCU ELEC. HEATER-4		120	1	16.7	2004
FCU ELEC. HEATER-5		120	1	16.7	2004
FCU ELEC. HEATER-6		120	1	16.7	2004
FCU ELEC. HEATER-7		120	1	16.7	2004
FCU ELEC. HEATER-8		120	1	16.7	2004
				Total Load	29772
			Panel Size @ 208V, 3P (amps)		82.7

Just as in case 2, a 100 amp, 30 pole, MCB panel may be used to serve all of the new equipment loads.

Table 21:Approach 2 Calculated Load on Main Distribution Panel DRGC

	MDP DRGC
Location	Elec Rm 115
Size	400 Amps
Previous Load	110,082 VA
Added Load	26,704
New Amp Draw	379 Amps
Okay?	Yes
Future Growth	5%

Fig. 21: New Panel PJP2 layout

PANELBOARD: PJP2				BUS RATING	100A								MAIN: 3P 100A MCB
MIN AIC: 35,000				VOLTAGE:	208Y/120V								PHASE(S): 3
ENCLOSURE: NEMA 1				MOUNTING:	SURFACE								WIRES: 4
LOCATION:				NOTES:									
						LOAD, VA							
BRANCH CIRCUIT DESIGNATION	P	TRIP	CKT#	ϕA		ϕB		ϕC		CKT#	TRIP	P	IRCUIT DESIGNATION
TRANSFER FAN 1	1	20	1	828	791.6					2	20	1	TRANSFER FAN 5
TRANSFER FAN 2	1	20	3			1116	1432			4	20	1	TRANSFER FAN 6
TRANSFER FAN 3	1	20	5					828	791.6	6	20	1	TRANSFER FAN 7
TRANSFER FAN 4	1	20	7	791.6	828					8	20	1	TRANSFER FAN 8
Heater 1	1	30	9			2004	2004			10	30	1	Heater 5
Heater 2	1	30	11					2004	2004	12	30	1	Heater 6
Heater 3	1	30	13	2004	2004					14	30	1	Heater 7
Heater 4	1	30	15			2004	2004			16	30	1	Heater 8
BLOWER MOTOR 1	–	–	17					791	791	18	–	–	BLOWER MOTOR 5
BLOWER MOTOR 2	–	–	19	791	791					20	–	–	BLOWER MOTOR 6
BLOWER MOTOR 3	–	–	21			791	791			22	–	–	BLOWER MOTOR 7
BLOWER MOTOR 4	–	–	23					791	791	24	–	–	BLOWER MOTOR 8
SPACE	–	–	25	0	0					26	–	–	SPACE
SPACE	–	–	27			0	0			28	–	–	SPACE
SPACE	–	–	29					0	0	30	–	–	SPACE
PHASE CONNECTED LOAD, VA				8829		12146		8792					
PHASE BALANCE				29.66%		40.80%		29.53%					
TOTAL CONNECTED LOAD, VA		29767											
FUTURE GROWTH - 10%		2977											
TOTAL + FUTURE LOAD, VA		32744											
TOTAL CURRENT, A		91											

5.3 Conclusions:

In both mechanical redesign options, some form of equipment load was added. These loads needed to be considered and checked to make sure they worked with the current electrical system. In both cases, these loads were able to fit onto one 100 amp panel board fed by the 400 amp main distribution panel on the first floor of the building. While these loads took up space in the panel, option 1 and 2 left 5% and 3% spare capacity for future growth respectively.

6.0 Acoustical Analysis

The transfer fans incorporated in mechanical redesign option 2 have save energy but like all fans, they create noise. These fans are in such close proximity to the rooms in which they serve that an acoustical analysis needed to be conducted. The purpose of the acoustical analysis was to determine the amount of attenuation needed in order to meet the required NC levels in these spaces. It was determined for this report that analyzing the fan sound transmission to the classrooms would suffice. This is because the classrooms have the most stringent NC requirements and are closest to the transfer fans. If attenuation to an acceptable level can be achieved for the classrooms, then the offices and corridors will be covered as well.

6.1 Fan Noise

The first step in conducting such an analysis was to determine the amount of sound that each fan generates. Although there are three different sizes of transfer fans, they all produce sound levels (Lw) that are fairly similar. This data was obtained from the manufacturer's literature and is shown in table 22 below.

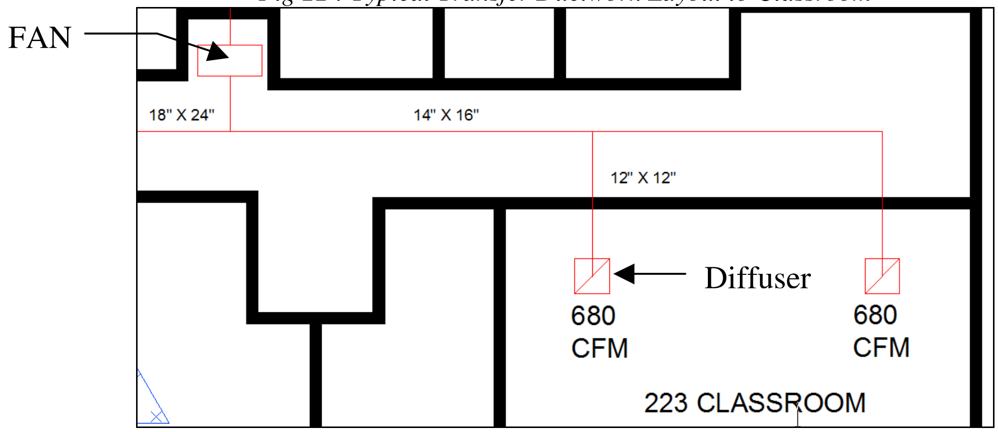
Table22 : Manufacture's Data Transfer Fan Self Noise

			Hz							
	RPM	SP (in.wg.)	62	125	250	500	1000	2000	4000	8000
Fan 1	950	0.25	74	80	70	69	68	65	65	62
Fan 2	750	0.375	69	76	73	64	63	60	58	54
Fan 3	750	0.375	69	76	73	64	63	60	58	54
Fan 4	700	0.375	73	78	72	67	64	65	63	55
Fan 5	750	0.375	69	76	73	64	63	60	58	54
Fan 6	850	0.5	71	77	74	66	64	62	59	55
Fan 7	700	0.375	73	78	72	67	64	65	63	55
Fan 8	750	0.375	69	76	73	64	63	60	58	54

6.2 Duct System Attenuation and Self Noise

The next step in determining the sound transmitted to the space was to determine the self noise and attenuation for each part of the air system. The typical ductwork system is illustrated in figure 22 below. The fallowing equations were used to determine the self noise of each piece…

Fig 22 : Typical Transfer Ductwork Layout to Classroom



Junction Self Noise:

$$L_w = K_j + 10\log(f_o/63) + 50\log(U_B) + 10\log(S_B) + 10\log(D_B) + C_B$$

Where:

K_j = Characteristic of Junction type

f_o = center frequency of octave band

U_B = Velocity in branch duct (ft/s)

S_B = cross sectional area of branch duct (ft^2)

D_B = equivalent diameter of branch duct (ft)

C_B = Constant based on junction type

Table 23 : Dcut System Attenuation

	Hz					
Attentuation	125.0	250.0	500.0	1000.0	2000.0	4000.0
T-junction	2.7	2.7	2.7	2.7	2.7	2.7
Straight	4.8	2.7	1.5	0.8	0.5	0.3
Branch takeoff (P/S = 3.2)	0.2	0.2	0.2	0.2	0.2	0.2
90 degree	0.0	1.0	5.0	8.0	4.0	3.0
Diffuser	0.0	0.0	0.0	0.0	0.0	0.0

Table 24 : Dcut System Self Noise

	Hz					
Self Noise	125.0	250.0	500.0	1000.0	2000.0	4000.0
Fan	80.0	70.0	69.0	68.0	65.0	65.0
T-junction	60.7	63.7	66.7	69.7	72.7	75.7
Branch takeoff	53.2	56.2	59.2	62.2	65.2	68.3
90 degree	62.0	65.1	68.1	71.1	74.1	77.1
Diffuser	29.0	26.0	20.0	12.0	1.0	0.0

6.3 Room Effect

The next step was to determine the room effect. The room effect is dependent on the materials in the room (their absorbance values) and the location of the diffuser in the room. For this analysis it was assumed that the diffuser was near a two wall junction.

Room effect:

$Lw_{room} = 10\log_{10}((Q/(4*\pi*r) + (4/R)$

Where:

Q = 4 (for a two wall junction)

r = distance from the diffuser

R = room constant
= α x S
α = average room absorption
S = total room surface area (ft^2)

Table 25 : Room Effect Calculation

	alpha (a) per frequency (HZ)						
Material	**125**	**250**	**500**	**1000**	**2000**	**4000**	**SA (ft^2)**
Glass 1/4" Thick	0.18	0.06	0.04	0.05	0.02	0.02	100
1/2" Gyp. Board on 2x4 Stud	0.29	0.1	0.05	0.04	0.07	0.09	1300
Terrazzo Floor	0.01	0.01	0.0115	0.02	0.02	0.02	1184
ACT Mineral Fiber 5/8"	0.68	0.76	0.6	0.65	0.82	0.76	1184
Door	0.42	0.21	0.1	0.08	0.06	0.06	21
Aeverage	0.322	0.030	0.021	0.022	0.026	0.025	3789
Room Effect	-14	-11	-10	-11	-11	-11	-14

6.4 Sound Pressure Level Transmitted to Room

Once the attenuation, self noise, and room effect had been obtained the next step was to determine the fan noise transmitted to the room. When calculating the transmitted sound pressure level it is important to remember that you directly subtract the attenuation values and use decibel addition for the self noise values. These results were obtained (table 26 below) and compared against the NC curve (fig 23 and table 27 below). As previously stated, the noise criteria for a classroom is 25-30. For this analysis the transmitted sound levels were compared against the NC 25 curve in order to be conservative.

Table 26 : Fan Noise Transmission to Classroom against NC level

	Hz					
	125	250	500	1000	2000	4000
Fan (self noise)	80.0	70.0	69.0	68.0	65.0	65.0
T-Junction (Attenuation	-2.7	-2.7	-2.7	-2.7	-2.7	-2.7
sum	77.3	67.3	66.3	65.3	62.3	62.3
Self Noise	**57.7**	**60.7**	**63.7**	**66.7**	**69.7**	**72.7**
sum	77.3	68.3	68.3	68.3	63.3	62.3
Stright Duct (14x16)	**-4.8**	**-2.7**	**-1.5**	**-0.8**	**-0.5**	**-0.3**
sum	72.5	65.6	66.8	67.5	62.8	62.0
Self Noise	**0.0**	**0.0**	**0.0**	**0.0**	**0.0**	**0.0**
sum	72.5	65.6	66.8	67.5	62.8	62.0
Branch	**-0.2**	**-0.2**	**-0.2**	**-0.2**	**-0.2**	**-0.2**
sum	72.3	65.4	66.6	67.3	62.6	61.8
self noise	**50.2**	**53.2**	**56.2**	**59.2**	**62.2**	**65.2**
sum	72.3	65.4	66.6	68.3	65.6	63.8
90 Elbow	**0.0**	**-1.0**	**-5.0**	**-8.0**	**-4.0**	**-3.0**
sum	72.3	64.4	61.6	60.3	61.6	60.8
self noise	**59.1**	**62.0**	**65.1**	**68.1**	**71.1**	**74.1**
sum	72.3	66.4	63.6	61.3	62.6	60.8
Diffuser	**0.0**	**0.0**	**0.0**	**0.0**	**0.0**	**0.0**
sum	72.3	66.4	63.6	61.3	62.6	60.8
self noise	**30.0**	**29.0**	**26.0**	**20.0**	**12.0**	**1.0**
sum	72.3	66.4	63.6	61.3	62.6	60.8
Room Effect	**-14**	**-11**	**-10**	**-11**	**-11**	**-11**
Lp Transmitted	58.6	55.2	53.1	50.7	51.7	50.0
NC 30	44	37	31	27	24	22
Acceptable?	NO	NO	NO	NO	NO	NO

Fig. 23: Noise Criteria Curves

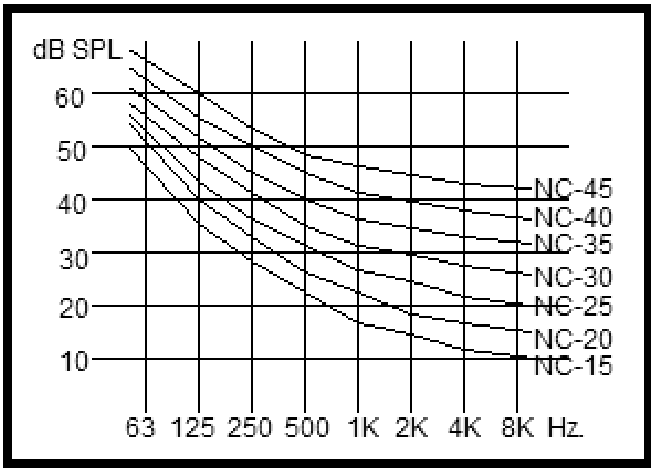


Table 27 : Noise Criteria Sound Pressure Levels

Noise Criterion	Octave Band Center Frequency *(Hz)*							
	63	125	250	500	1000	2000	4000	8000
	Sound Pressure Levels *(dB)*							
NC-15	47	36	29	22	17	14	12	11
NC-20	51	40	33	26	22	19	17	16
NC-25	54	44	37	31	27	24	22	21
NC-30	57	48	41	35	31	29	28	27
NC-35	60	52	45	40	36	34	33	32
NC-40	64	56	50	45	41	39	38	37
NC-45	67	60	54	49	46	44	43	42
NC-50	71	64	58	54	51	49	48	47
NC-55	74	67	62	58	56	54	53	52
NC-60	77	71	67	63	61	59	58	57
NC-65	80	75	71	68	66	64	63	62

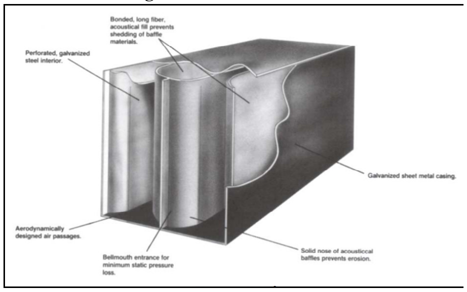
6.5 Additional Attenuation

The results displayed in table 26 above show that although the fan noise is reduced when it gets to the classroom, it is above the NC 25 curve and is not acceptable. In order to reduce this noise further, some additional attenuation must be incorporated into the ductwork system. The choice used for this analysis was to insert a sound attenuator in the straight section of duct, between the classroom and the T-junction. A low frequency duct silencer (fig 24 below) was selected and inserted into the system for recalculation of the fan noise transmission. The results of this calculation are shown below in table 28.

Table 28 : New Transmitted Fan Sound Pressure Level With Duct Silencer

	Hz					
	125	250	500	1000	2000	4000
Fan (self noise)	80.0	70.0	69.0	68.0	65.0	65.0
T-Junction (Attenuation	-2.7	-2.7	-2.7	-2.7	-2.7	-2.7
sum	77.3	67.3	66.3	65.3	62.3	62.3
Self Noise	**0.0**	**60.7**	**63.7**	**66.7**	**69.7**	**72.7**
sum	77.3	68.3	68.3	68.3	63.3	62.3
Silencer	**-21.0**	**-35.0**	**-41.0**	**-41.0**	**-28.0**	**-21.0**
sum	**56.3**	**33.3**	**27.3**	**27.3**	**35.3**	**41.3**
Self Noise	**0.0**	**0.0**	**0.0**	**0.0**	**0.0**	**0.0**
sum	**56.3**	**33.3**	**27.3**	**27.3**	**35.3**	**41.3**
Stright Duct (14x16)	**-4.8**	**-2.7**	**-1.5**	**-0.8**	**-0.5**	**-0.3**
sum	51.5	30.6	25.8	26.5	34.8	41.0
Self Noise	**0.0**	**0.0**	**0.0**	**0.0**	**0.0**	**0.0**
sum	51.5	30.6	25.8	26.5	34.8	41.0
Branch	**-0.2**	**-0.2**	**-0.2**	**-0.2**	**-0.2**	**-0.2**
sum	51.3	30.4	25.6	26.3	34.6	40.8
self noise	**0.0**	**53.2**	**56.2**	**59.2**	**62.2**	**65.2**
sum	51.3	30.4	25.6	26.3	34.6	40.8
90 Elbow	**-2.0**	**-3.0**	**-4.0**	**-5.0**	**-6.0**	**-8.0**
sum	49.3	27.4	21.6	21.3	28.6	32.8
self noise	**38.0**	**34.0**	**28.0**	**18.0**	**4.0**	**0.0**
sum	49.3	28.4	22.6	23.3	28.6	32.8
Diffuser	**0.0**	**0.0**	**0.0**	**0.0**	**0.0**	**0.0**
sum	49.3	28.4	22.6	23.3	28.6	32.8
self noise	**0.0**	**29.0**	**26.0**	**20.0**	**12.0**	**1.0**
sum	49.3	31.4	24.6	25.3	28.6	32.8
Room Effect	**-14**	**-11**	**-10**	**-11**	**-11**	**-11**
Lp Transmitted	35.6	20.2	14.1	14.7	17.7	22.0
NC 25	44	37	31	27	24	22
Acceptable	**YES**	**YES**	**YES**	**YES**	**YES**	**YES**

Fig. 24 : Duct Silencer



6.6 Conclusion:

Although the transfer fans incorporated in the mechanical redesign option 2 do create significant noise and are in close proximity to the spaces it is serving it is possible to attenuate the noise to an acceptable level. It should also be noted that other attenuation techniques, like using duct liner, were not examined in this report but may be acceptable alternatives to the duct silencer.

7.0 CFD Passive Chilled Beam Study

7.1 Introduction

The goal of this study was to compare two options of passive chilled beam layouts, as well as to determine the feasibility of using these passive chilled beams with the laminar diffusers specified in the existing lab design. Beam layout option 1 (figure 25 below) places the passive chilled beams between each supply diffuser. This is not the conventional way of using passive chilled beams but the use of laminar diffusers may make this a practical option. Layout option 2 (figure 26 below) puts the chilled beams at the perimeter of the room, allowing warm air to flow through the passive beam through convection, while the supply diffusers take care of the load in the center of the room. This option is the standard way of using a passive chilled beam and is prescribed in the passive chilled beam manufacturer's literature.

The laminar diffusers in the current mechanical design of this lab are laminar diffusers with adjustable guide vanes for directional control over the airflow. This study compares a vertical and horizontal air discharge pattern in order to determine which one is best suited to parallel chilled beam systems. The effectiveness of these systems will be based upon ability to meet the space cooling load (meet the specified lab air temperature of 72°F (22.2°C)) and ability to achieve a vertical temperature difference < 3°C in the occupied zone, as prescribed by section 5.2.4.3 of ASHRAE standard 55.

Table 5.2.4.3 out of ASHRAE Standard 55

TABLE 5.2.4.3
Allowable Vertical Air Temperature Difference Between Head and Ankles

Vertical Air Temperature Difference °C (°F)
< 3 (< 5.4)

Page 10, ASHRAE Standard 55-2004

Although table 5.2.4.3 was used in the analysis of each simulation in this report, it should be noted that standard 55 states, "Section 5.2.4.3 applies to temperature difference where the head level is warmer than the ankle level. Thermal stratification in the opposite direction is rare, is perceived more favorably by occupants, and is not addressed in this standard". The temperature stratification in this lab room is in this "opposite direction" due to the fact that the internal loads for each case in this report were modeled as a floor heat flux and that the cooling system being proposed is a chilled beam system. This results in a conservative analysis approach.

Fig. 25 : Passive Chilled Beam Layout 1 Domain

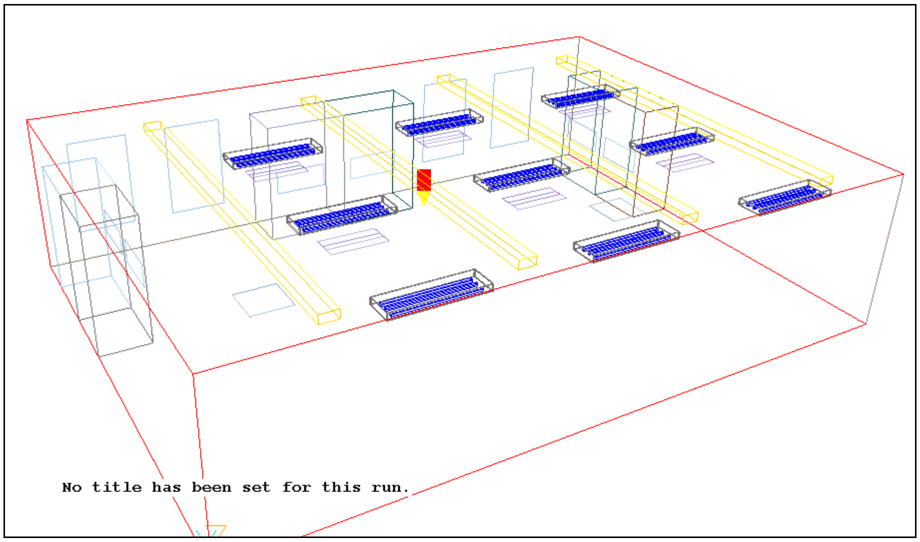
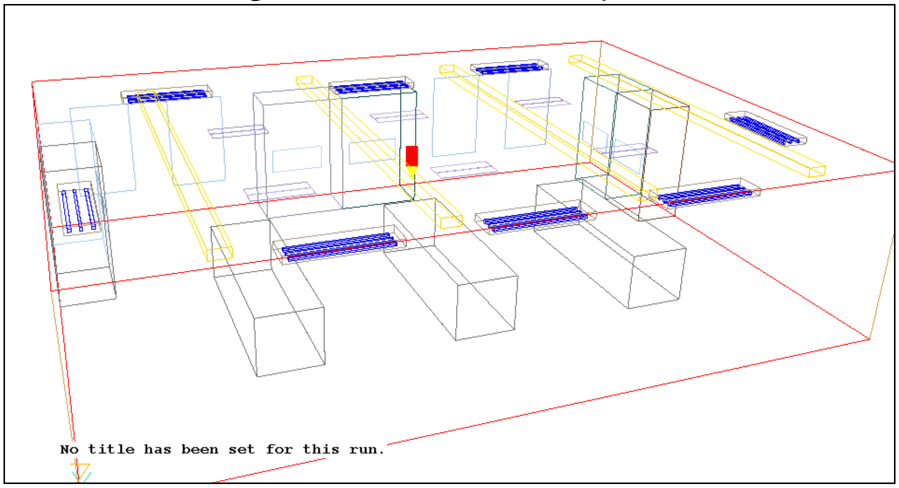


Fig. 26 : Passive Chilled Beam Layout 2 Domain



7.2 Diffuser Airflow Simulation:

Turbulence Model: K-e

Differencing Scheme: Hybrid

Vertical Airflow Pattern

Computation Time (Vertical Pattern): 2hr 48m

Number of Iterations: 4600

Mass Residual: 1.6%

Horizontal Pattern

Computation Tim: 2hr 10m

Number of Iterations: 4600

Mass Residual: 1.3%

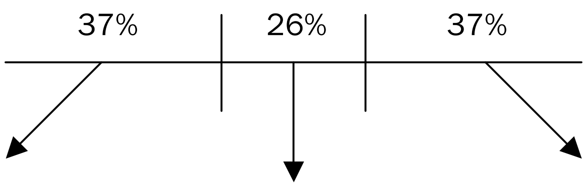
Table29 : Room and Grid (mesh) Size

	Size		
	X (m)	Y (m)	Z (m)
Room Dims.	12.19	8.87	3
Grid Size	109	59	13

Table 30: Diffuser Boundary Conditions

Diffuser	Air Exchange Rate CFM(kg/s)	Supply Velocity fpm (m/s)	Gross Area (Agross) m^2	Area Factor (Ao) m^2	e = Ao/Agross %	Geom. Area Factor m^2
Laminar	285 (0.161)	35.6 (0.244)	0.535	0.397	0.74	0.74

Fig. 27 : Titus Versatec Laminar Diffuser

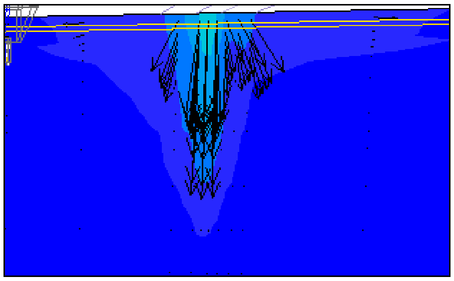


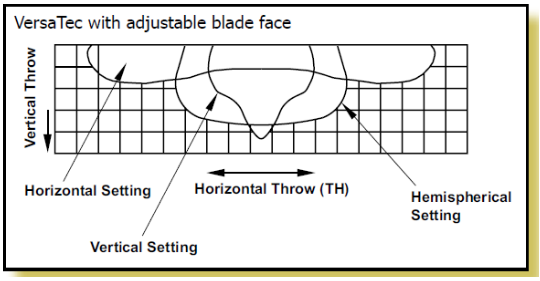
(95 cfm) = 0.294 m/s = 0.207j , -0.207k

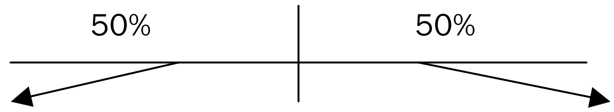
(95 cfm) = 0.418 m/s = -0.418k

(95cfm) = 0.294 m/s = -0.207j , -0.207k

Fig. 28 & 29 : Vertical Pattern Modeled Diffuser vs. Manufacturer Data





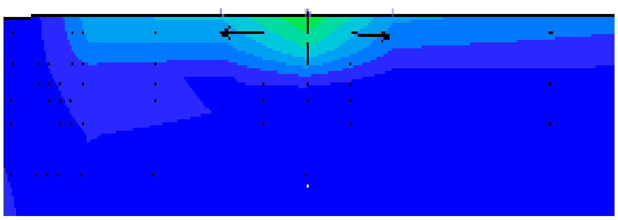


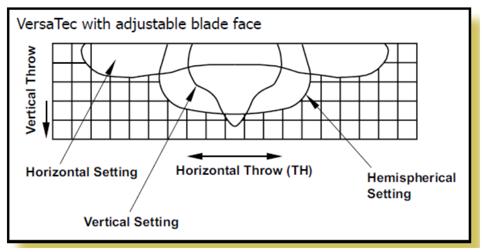
(145 cfm) (145cfm)

= 0.338 m/s = 0.338 m/s

= 0.336i , - 0.029k = -0.336j , -0.029k

Fig. 30 & 31: Horizontal Pattern Modeled Diffuser vs. Manufacturer Data





7.2.1 Results

The diffuser models used for this simulation appear to accurately represent the actual equipment. The airflow patterns obtained in the simulations seem to match the manufacturer's data shown above in figures 30 and 31 . The more accurately the diffusers are modeled the more accurate our room temperature and airflows values will be in the non-isothermal simulations.

7.3 Non-isothermal Simulation:

Table 31: Simulation Room and Mesh Dimensions

	Size		
	X (m)	Y (m)	Z (m)
Room Dims.	12.19	8.87	3
Grid Size	80	60	21

Table 32 : Lab Internal Loads

Internal Load	btu/h	Watts
People, Equip, LTG	24,860	7,286
Solar	1,375	398.75
Glass Conduction	1,010	292.9
Wall Conduction	1,239	359.31
Total	36,434	8,348 (**77.3 W/m^2**)

Table 33 : Lab Surface Temperatures

Boundary	Temp ºF (ºC)
North Wall	80 (26.6)
South Wall	76 (24.4)
East Wall	74 (23.3)
West Wall	74 (23.3)
Ceiling	75 (23.8)
Supply Air	55 (12.7)
Chilled Beam	56 (13.3)

7.3.1 Case 1: Beam Layout 1 with Vertical Discharge Pattern

Turbulence Model: K-e
Numerical Scheme: Hybrid
Computation Time: 2hr 36m
Number of Iterations: 4600
Mass Residual: 1.4%
Temperature Residual: 0.37%

Results:

Fig 32 : Case 1 Y-axis Temperature Distribution through Occupied Zone

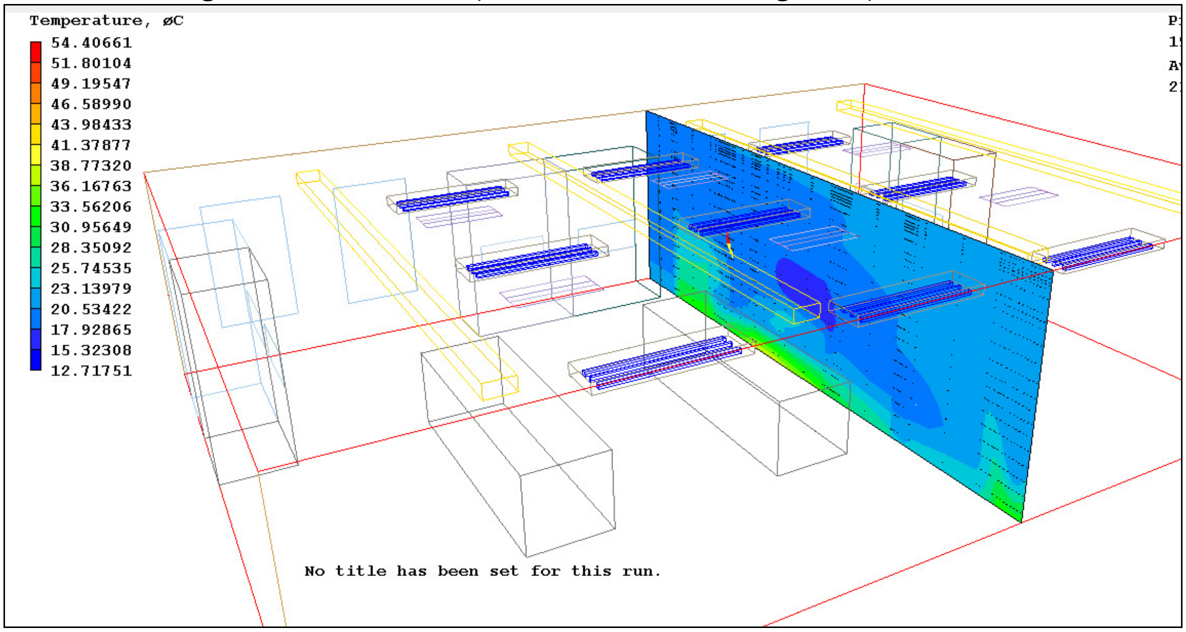


Fig 33 : Case 1 Y-axis Temperature Distribution through Diffusers and CB's

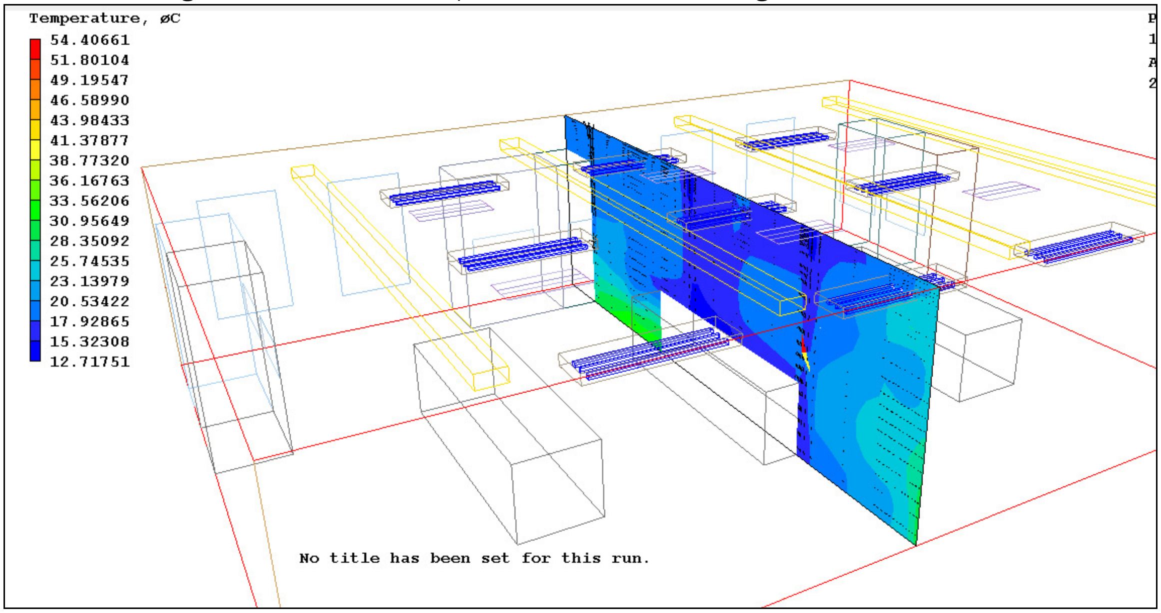
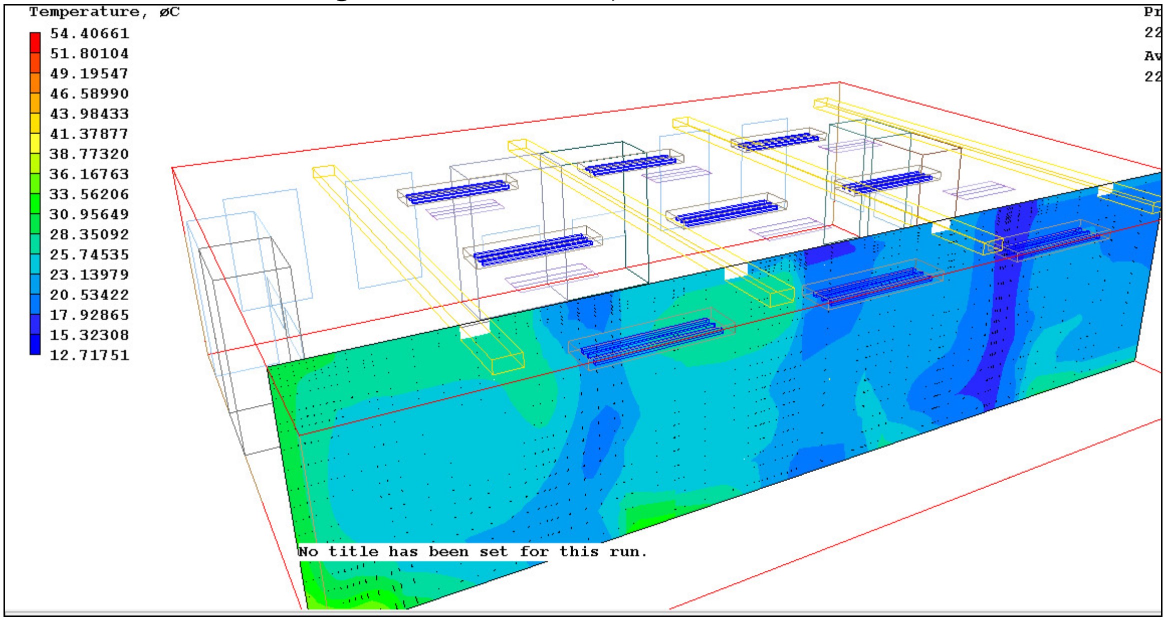


Fig 34 : Case 1 X-axis Temperature Distribution



Case 1 Synopsis:

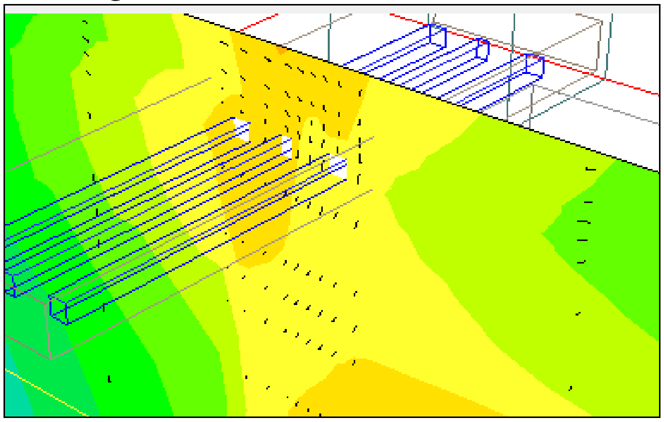
Reviewing the results from the Case 1, non-isothermal simulation, it is apparent that case 1 does not condition the room well. In much of the lab the occupied area air temperatures are greater than or less than the specified 72°F (22.2°C). Vertical temperature differences throughout the room are unacceptable, >10 °C in some areas, which is larger than the < 3 °C specified in section 5.2.4.3 of standard 55. This may lead to a large number of occupants dissatisfied. The larger problem however, seems to be the horizontal temperature difference. While standard 55 does not sepecify guidelines for this it seems as though the cool pockets of air scattered throughout the room would be noticable to an occupant walking through them. The main problem seems to be the diffusers dumping cold air straight down, overcooling the occupied area (fig. 7 above). It seems as though the vertical diffuser pattern is cooling the lower level air and providing enough of a horizontal draft to prevent convection through the chilled beams rendering them ineffective. Also, the flow through the chilled beams seems to be as expected; warm air entering on top and exiting the bottom as it is cooled.

7.3.2 Case 2: Beam Layout 1 with Horizontal Discharge Pattern

Turbulence Model: K-e
Numerical Scheme: Hybrid
Computation Time: 2hr 31m
Number of Iterations: 4600
Mass Residual: 1.3%
Temperature Residual: 0.4%

Results:

Fig 35 : Zoomed In View of Chilled Beam Airflow



*Temperature Scale Enhanced for Better Clarity; Not the Same Scale as in figures 36-38

Fig 36 : Case 2 y-axis Temperature Distribution

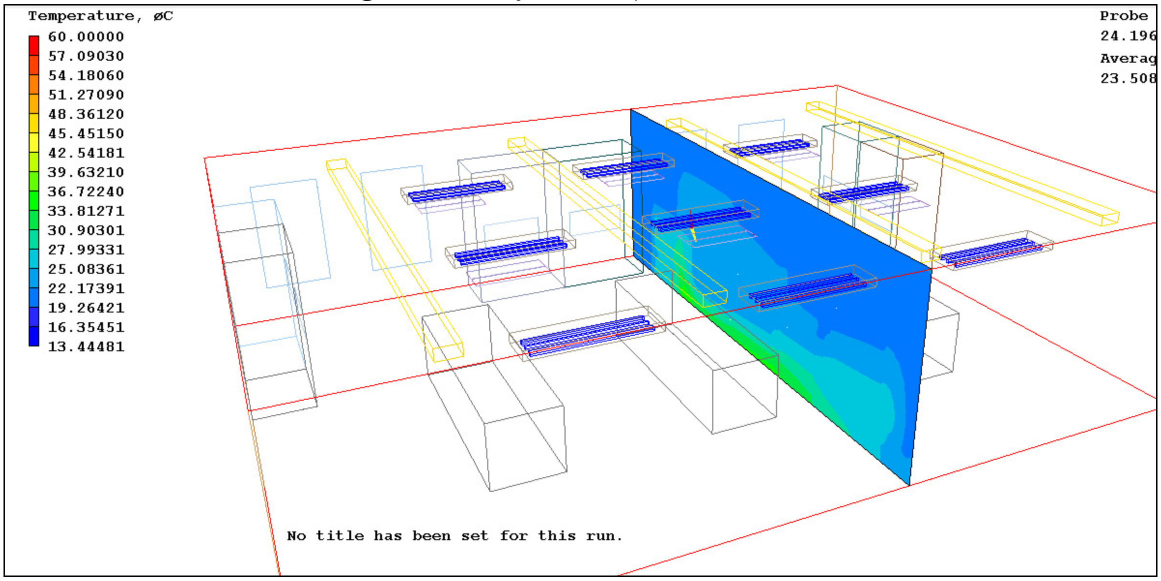


Fig 37 : Case 2 Y-axis Temperature Distribution through Diffusers and CB's

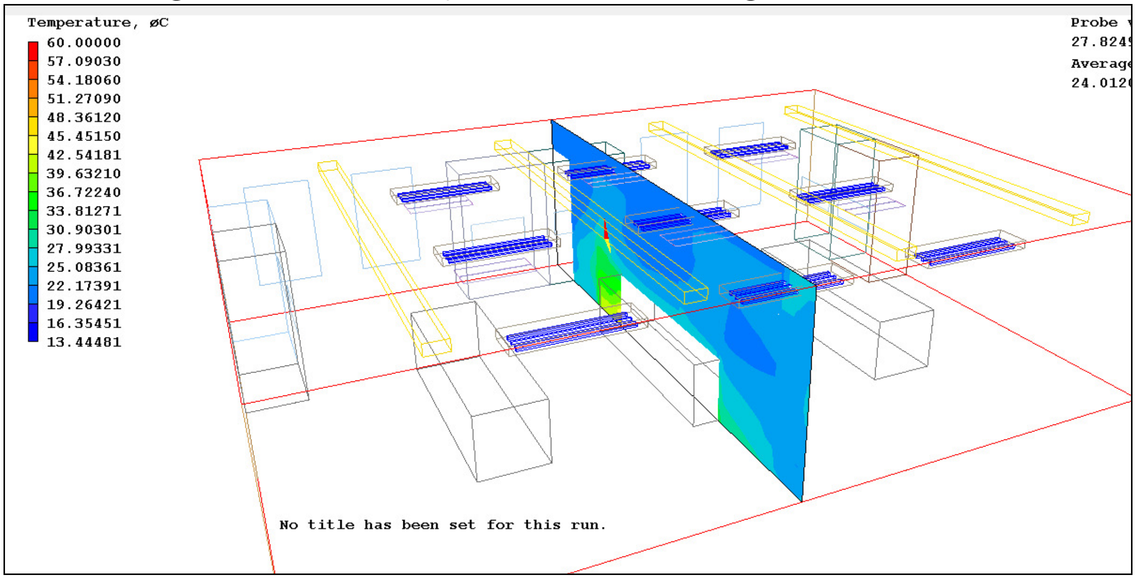


Fig 38 : Case 2 Eddy Formation

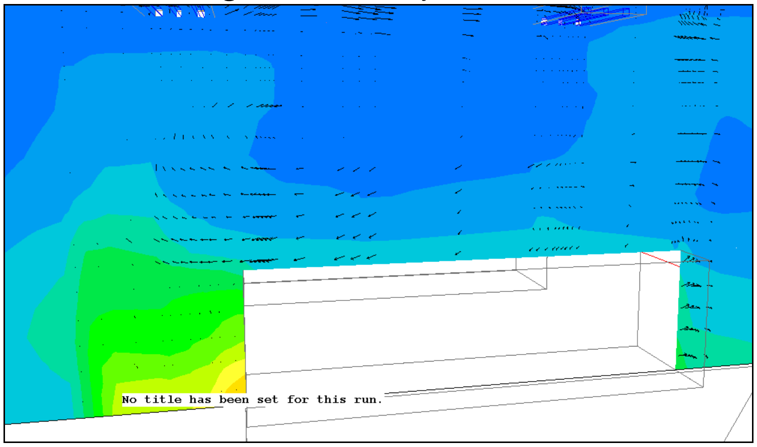
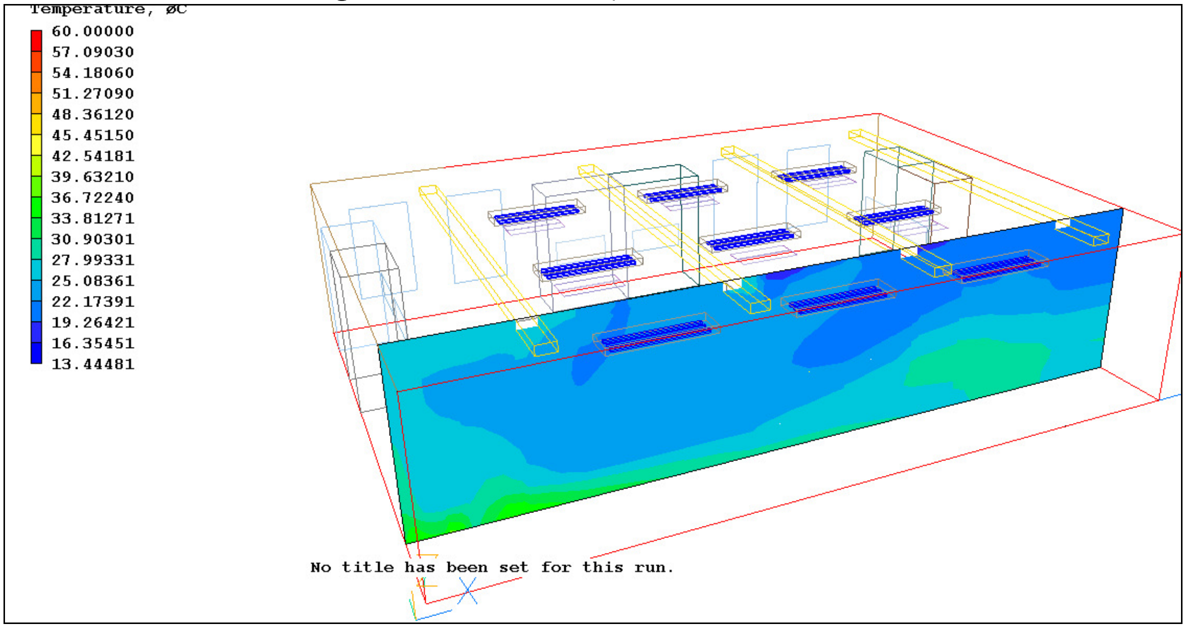


Fig 39 : Case 2 X-axis Temperature Distribution



Case 2 Synopsis:

It is apparent from the reults shown in figures 36-39 above that the horizontal discharge pattern works much more effectively with the passive chilled beam system. Temperature differences throughout the room are are much less severe than in case 1, although some areas are still above the < 3^{o}C allowance, required by ASHRAE standard 55. One problem area identified in the study of case 2 occurs between the lab tables and the north wall (depicted in figures 38 and 39 above). It seems that an eddy has formed in this area, resulting in an inability to properly cool the floor. Also, it seems that with the horizontal diffusers, the lower level air has time to warm up and some convection occurs through the chilled beams.

7.3.1 Case 3: Beam Layout 2 with Horizontal Discharge Pattern

Turbulence Model: K-e
Numerical Scheme: Hybrid
Computation Time: 2hr 40m
Number of Iterations: 4600
Mass Residual: 0.88%
Temperature Residual: 0.1%

Fig 40 : Case 3 Y-axis Temperature Distribution

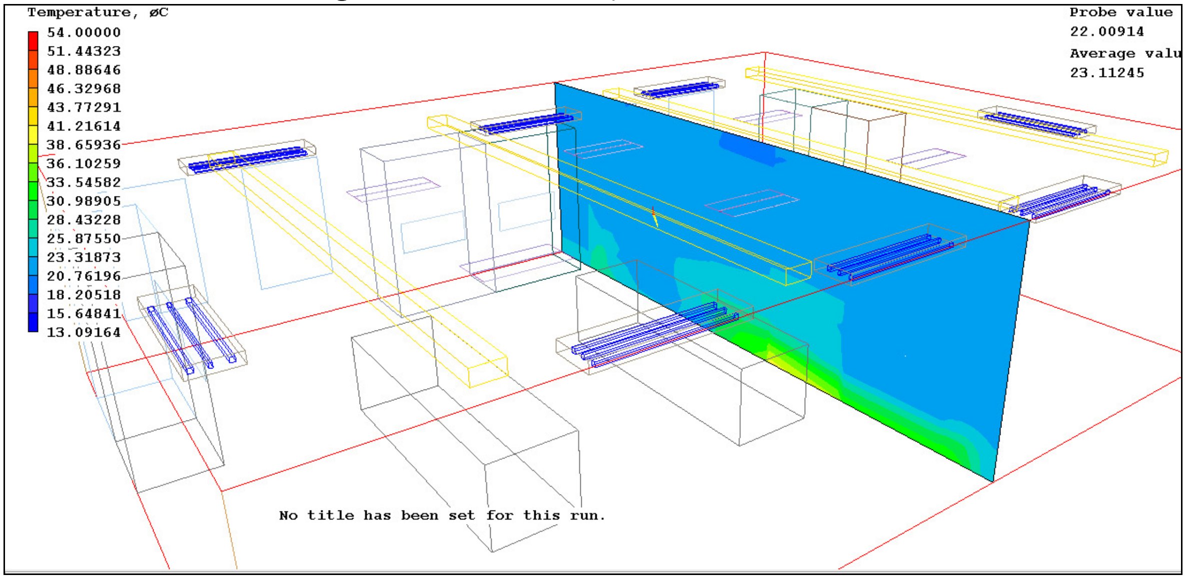


Fig 41 : Case 3 Y-axis Temperature Distribution through Diffusers and CB's

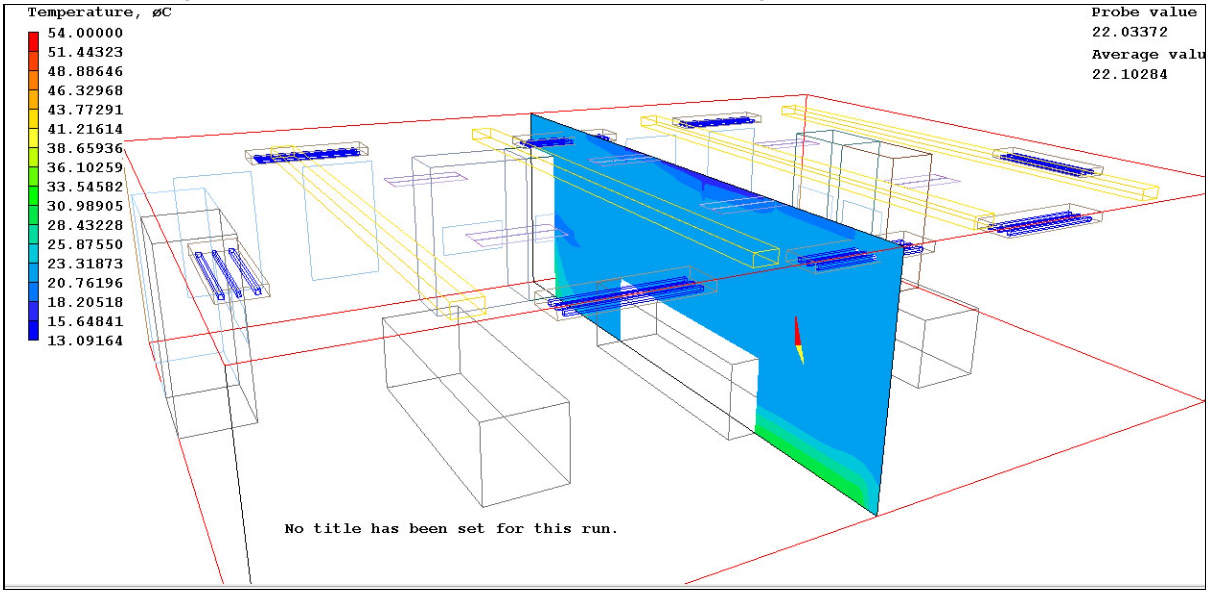
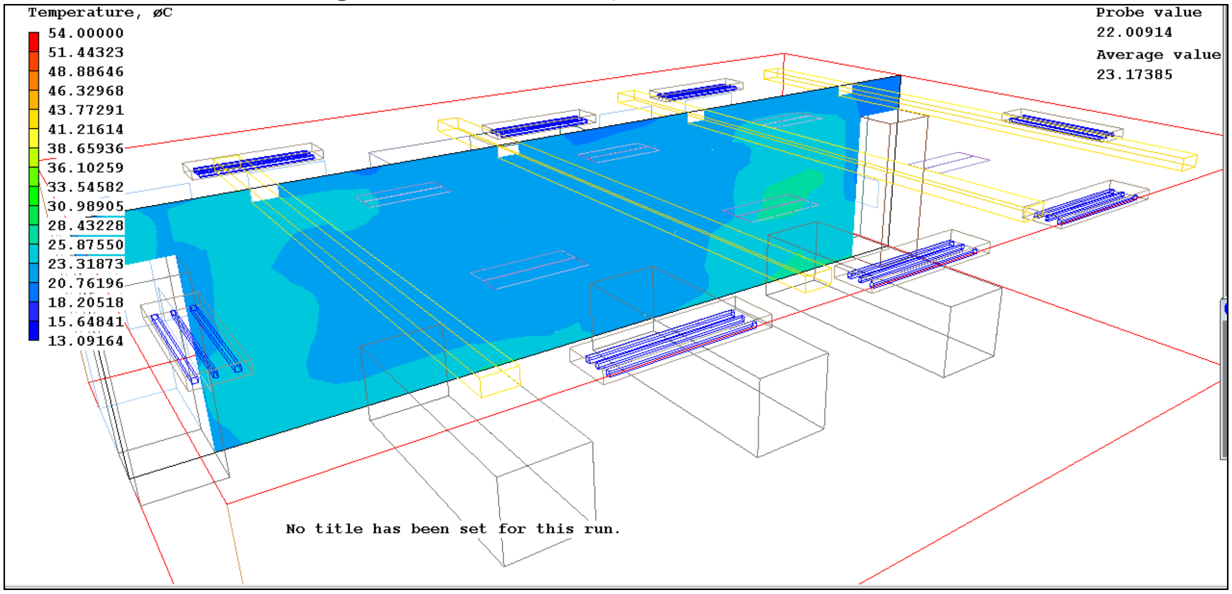


Fig 42 : Case 3 X-axis Temperature Distribution



Case 3 Synopsis:

After reviewing the resutls from the case 3 simtulation in the figures above, it is apparent that case 3 provides the most satisfactory temperature differences and most effectively cools the room to the desired air temperature. These results confirm the passive chilled beam manufacturer's suggestions for the perimeter beam layout.

7.4 Conclusions:

After simulating the three cases presented in this report, it has been concluded that the case that most effectively cools the room to the desired temperature, while meeting the occupant comfort requirements in section 5.2.4.3 of ASHRAE standard 55 is case 3.

The vertical discharge pattern in case 1 put too much cool air straight down into the occupied zone, overcooling some areas while creating warm spots in other areas. This vertical diffuser pattern did not seem to work well with the passive chilled beams. It was hypothesized that the overcooling of the lower level air and air circulating patterns caused by the diffusers made the passive chilled beams Ineffective.

In case 2, the horizontal discharge pattern seemed to solve several of the problems from case 1. The horizontal discharge pattern reduced the amount of cool air streams in the occupied zone and cool/hot pockets throughout the room but there were still areas where the load was not being met.

Case 3 most effectively cooled the room to the desired 72^{o}F (22.2^{o}C) while reducing the horizontal and vertical temperature difference throughout the room. The temperature profiles in case 3 were very steady. The perimeter layout of the passive chilled beams seems to be the best fit in this situation. The supply diffusers properly cool the interior of the space and allow for convection of the perimeter air through the chilled beams.

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Arthur A. Bell, JR. *HVAC Equations, Data and Rules of Thumb*. 2008

APPENDIX A: Simulation Schedules and Variables

Schedules and Variables Used to Model Base Design

Educational Occupancy Weekday		
From	To	% Peak
Midnight	6:00am	0.2
6:00am	7:00am	0.25
7:00am	8:00am	0.55
8:00am	11:00am	0.8
11:00 AM	Noon	0.7
Noon	2:00Pm	0.75
2:00 PM	4:00 PM	0.8
4:00 PM	5:00PM	0.6
5:00 PM	7:00 PM	0.3
7:00 PM	Midnight	0.2

Educational Occupancy Weekend		
From	To	% Peak
Midnight	9:00am	0
9:00am	9:00pm	0.25
9:00pm	Midnight	0

Research Occupancy		
From	To	% Peak
Midnight	6:00am	0
6:00am	7:00am	0.15
7:00am	8:00am	0.55
8:00am	11:00am	0.8
11:00am	Noon	0.7
Noon	2:00pm	0.7
2:00pm	4:00pm	0.8
4:00pm	5:00pm	0.55
5:00pm	6:00pm	0.3
6:00pm	7:00pm	0.25
7:00pm	Midnight	0.2

Educational Lighting		
From	To	% Peak
Midnight	6:00am	0.05
6:00am	7:00am	0.15
7:00am	8:00am	45
8:00am	Noon	100
Noon	1:00pm	70
1:00pm	2:00pm	90
2:00pm	5:00pm	100
5:00pm	6:00pm	90
6:00pm	7:00pm	70
7:00pm	8:00pm	55
8:00pm	9:00pm	45
9:00pm	Midnight	15

Research Lighting		
From	To	% Peak
Midnight	6:00am	40
6:00am	7:00am	45
7:00am	8:00am	80
8:00am	11:00am	100
11:00am	Noon	90
Noon	1:00pm	80
1:00pm	2:00pm	90
2:00pm	4:00pm	100
4:00pm	5:00pm	80
5:00pm	6:00pm	50
6:00pm	Midnight	40

APPENDIX A: Simulation Schedules and Variables

Schedules and Variables Used to Model Base Design

Brick Wall	U Value	% Peak
Roof	9:00am	0
Floor	9:00pm	45
Windows	Midnight	0
Typical Door	9:00pm	45

Educational Misc.		
From	To	% Peak
Midnight	6:00am	70
6:00am	7:00am	85
7:00am	8:00am	95
8:00am	11:00am	100
11:00am	Noon	95
Noon	1:00pm	90
1:00pm	2:00pm	95
2:00pm	4:00pm	100
4:00pm	5:00pm	90
5:00pm	6:00pm	75
6:00pm	Midnight	70

Research Misc.		
From	To	% Peak
Midnight	6:00am	25
6:00am	7:00am	35
7:00am	8:00am	70
8:00am	11:00am	100
11:00am	Noon	90
Noon	1:00pm	75
1:00pm	2:00pm	90
2:00pm	5:00pm	100
5:00pm	6:00pm	90
6:00pm	7:00pm	70
7:00pm	8:00pm	55
8:00pm	9:00pm	50
9:00pm	10:00pm	35
10:00pm	Midnight	25

Assembly	U Value	Shading Coeff.
Brick Wall	0.104	-
Roof	0.048	-
Floor	0.19	-
Windows	0.29	0.44
Typical Door	0.02	-

Unit	Cost
Electricity	**$ 0.1184 / kW-hr**
Chilled Water	**$ 0.828 / Therm**
Steam	**$ 2.34 / Therm**

APPENDIX A: Simulation Schedules and Variables

Table :Redesign Approach 1, TRACE Model Lab ACH Rates
(as % of maximum hood ventilation rate)

Weekday Time	Demand Based	Minimum SET ACH
12:00 AM	42	50
1:00 AM	42	50
2:00 AM	42	50
3:00 AM	42	50
4:00 AM	42	50
5:00 AM	42	50
6:00 AM	42	50
7:00 AM	50	100
8:00 AM	50	100
9:00 AM	80	100
10:00 AM	80	100
11:00 AM	90	100
12:00 PM	75	100
1:00 PM	90	100
2:00 PM	80	100
3:00 PM	80	100
4:00 PM	80	100
5:00 PM	65	100
6:00 PM	65	100
7:00 PM	42	50
8:00 PM	42	50
9:00 PM	42	50
10:00 PM	42	50
11:00 PM	42	50

* Lab max ACH rate = 12
* Prep Room max ACH = 4

APPENDIX A: Simulation Schedules and Variables

Table : Minimum Humidity Ratios for Chilled Beam Analysis (Approach 1)

DOAS	SPACE	Sensible (BTUH)	Total (BTUH)	Latent (BTUH)	SENSIBLE HEAT RATIO	Wmin (gr/lb)	MAX OA AIRFLOW (CFM)	MINIMUM OA AIRFLOW (CFM)
3	106 PREP	55,600	56,865	2,116	0.98	59.00	519	259
3	107 INSTRUC LAB	39,600	43,226	3,626	0.92	63.89	4,800	1,800
4	112 INSTRUC LAB	38,468	42,054	3,586	0.91	63.90	4,800	1,800
4	119 PREP	33,864	35,980	2,116	0.94	55.17	317	158
3	213 INTRUCTIONAL LAB	38,828	41,652	2,824	0.93	64.13	4,800	1,800
4	218 INSTRUCTIONAL LAB	37,657	41,236	3,579	0.91	63.90	4,800	1,800
4	224 PREP	31,112	33,998	2,886	0.92	49.81	279	140
3	312 GLASS WASH	59,352	60,440	1,088	0.98	62.13	557	278
3	313 INSTRUC LAB	40,440	44,040	3,600	0.92	63.90	4,800	1,800
4	318 INSTRUC LBA	41,330	44,938	3,608	0.92	63.89	4,800	1,800
4	324 PREP	27,136	29,969	2,833	0.91	47.54	239	119
3	406 PREP	55,979	58,454	2,475	0.96	58.10	527	264
3	407 INSTRUC LAB	38,172	40,991	2,819	0.93	64.14	4,800	1,800
4	412 INSTRUC LAB	38,227	41,861	3,634	0.91	63.89	4,800	1,800
4	418 PREP	25,842	28,657	2,815	0.90	46.74	227	113

NOTE: SENSIBLE LOAD AIRFLOWS IN RED REPRESENT SPACES IN WHICH HOOD AIRFLOW REQUIREMENTS DICTTATE THE VENTILATION AIRFLOW AND WILL REQUIRE REHEAT DURING TIMES OF PEAK HOOD USE.

Table: Minimum Lab Air Change Rates Based On Minimum Hood Airflow (for Approach 1)

ROOM	AREA S.F.	CEILING HEIGHT	MINIMUM AIRFLOW (CFM)	MINIMUM AIRFLOW AS ACH
107 INSTRUC LAB	1570	16	1800	4
112 INSTRUC LAB	1519	16	1800	4
213 INTRUCTIONAL LAB	1570	16	1800	4
218 INSTRUCTIONAL LAB	1519	16	1800	4
313 INSTRUC LAB	1570	16	1800	4
318 INSTRUC LBA	1519	16	1800	4
407 INSTRUC LAB	1570	16	1800	4
412 INSTRUC LAB	1519	16	1800	4

APPENDIX A: Simulation Schedules and Variables

Cooling Load Factors and Sensible Heat Gain Coefficients for Redesign Approach 2

	WALL CLF	
HOUR	NORTH	SOUTH
12:00 AM	0.23	0.12
1:00 AM	0.2	0.11
2:00 AM	0.18	0.09
3:00 AM	0.16	0.08
4:00 AM	0.14	0.07
5:00 AM	0.34	0.08
6:00 AM	0.41	0.11
7:00 AM	0.46	0.14
8:00 AM	0.52	0.21
9:00 AM	0.59	0.31
10:00 AM	0.65	0.42
11:00 AM	0.7	0.52
12:00 PM	0.73	0.57
1:00 PM	0.75	0.58
2:00 PM	0.76	0.53
3:00 PM	0.74	0.47
4:00 PM	0.75	0.41
5:00 PM	0.79	0.36
6:00 PM	0.61	0.29
7:00 PM	0.5	0.25
8:00 PM	0.42	0.21
9:00 PM	0.36	0.18
10:00 PM	0.31	0.16
11:00 PM	0.27	0.14

	SHGC	
Month	N	S
Jan	20	254
Feb	24	241
Mar	29	206
Apr	34	154
May	37	113
Jun	48	95
Jul	38	109
Aug	35	149
Sep	30	200
Oct	25	234
Nov	18	250
Dec	18	253
@ 40 DEG NORTH LAT.		

APPENDIX A: Simulation Schedules and Variables

AHU-3 Peak Ventilation and Supply Airflows to Non-Lab Spaces						
	Az	Ra	Pz	Rp	Vbz (= Voz)	SUPPLY AIR
Space	(ft^2)	CFM / ft^2	# People	cfm / person	CFM OA	CFM
111 Informal Gathering	580	0.06	12	5	94.8	658
105 Storage	79	0.12	0	0	9.48	53
100C Corridor	775	0.06	0	0	46.5	896
215 AV	322	0.12	0	0	38.64	193
214 Vending	30	0.06	0	0	1.8	791
200C Corridor	1,175	0.06	0	0	70.5	662
315 PBL Classroom	1,407	0.12	47	10	638.84	1,369
316 Informal Gathering	580	0.06	12	5	94.8	689
310 Tech Offices	198	0.06	2	5	21.88	107
300B Corridor	1,420	0.06	0	0	85.2	1,070
410 PBL Classroom	1,407	0.12	49	10	658.84	1,864
411 Informal Gathering	580	0.06	12	5	94.8	690
400B Corridor	1,175	0.06	0	0	70.5	1,115
405 Tech Office	175	0.06	2	5	20.5	131
	8,553	1	134	40	1,856	9,042

AHU-4 Peak Ventilation and Supply Airflows to Non-Lab Spaces						
	Az	Ra	Pz	Rp	Vbz (= Voz)	SUPPLY AIR
	(ft^2)	CFM / ft^2	# People	cfm / person	CFM OA	CFM
110 PBL Classroom	1,407	0.12	10	10	268.84	1,999
118 Media Services Office	258	0.06	3	5	30.48	247
100C Corridor	775	0.06	0	0	46.5	300
223 Prep Room	424	0.18	10	10	176.32	955
217 Instructional Lab	755	0.18	13	10	265.9	1,995
215 PBL Classroom	1,407	0.12	10	10	268.84	1,364
222 PBL Corner Classroom	511	0.12	20	10	261.32	1,540
200C Corridor	1,018	0.06	0	0	61.08	300
322 PBL Corner Classroom	509	0.12	20	10	261.08	1,620
300B Corridor	1,018	0.06	0	0	61.08	300
417 Seminar	495	0.06	13	5	94.7	1,530
400B Corridor	1,018	0.06	0	0	61.08	300
	9,595				**1,857**	**12,450**

APPENDIX A: Simulation Schedules and Variables

Table : Hourly Outdoor Air Temperature for Newark, DE in 2002 for Approach 2

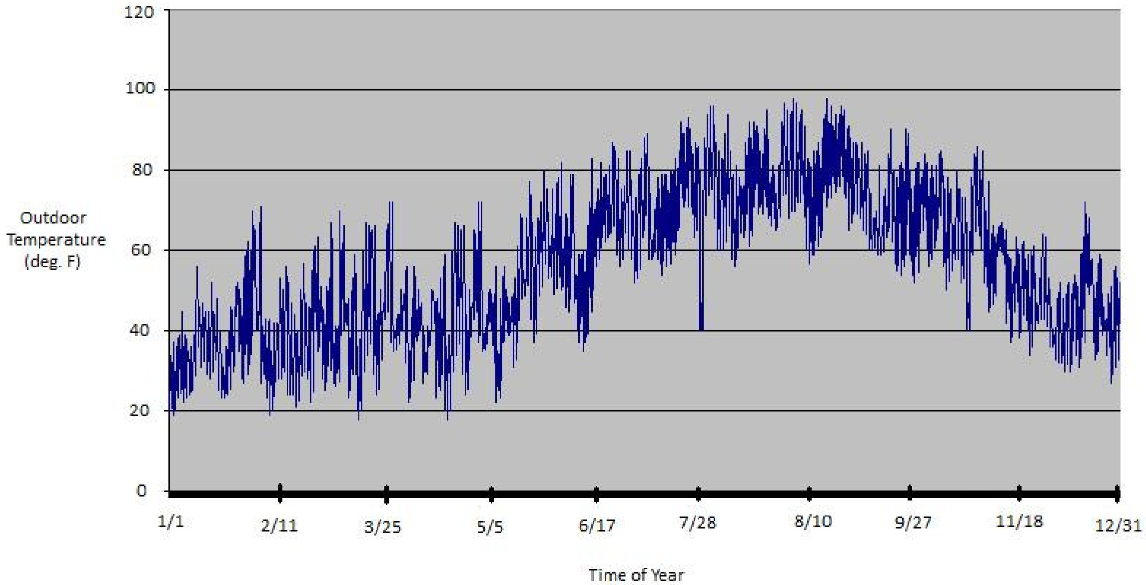


Table : Redesign Approach 2, Hourly Schedules
(As % of Peak)

HOUR	OCC	LTG	VENT	EQUIP
12:00 AM	0.2	0.05	0.5	0.25
1:00 AM	0.2	0.05	0.5	0.25
2:00 AM	0.2	0.05	0.5	0.25
3:00 AM	0.2	0.05	0.5	0.25
4:00 AM	0.2	0.05	0.5	0.25
5:00 AM	0.2	0.05	0.5	0.25
6:00 AM	0.2	0.05	0.5	0.25
7:00 AM	0.25	0.15	1	0.35
8:00 AM	0.5	0.45	1	0.7
9:00 AM	0.8	0.9	1	1
10:00 AM	0.8	0.9	1	1
11:00 AM	0.7	0.9	1	1
12:00 PM	0.75	0.9	1	0.9
1:00 PM	0.75	0.7	1	0.75
2:00 PM	0.75	0.9	1	0.9
3:00 PM	0.8	0.9	1	1
4:00 PM	0.8	0.9	1	1
5:00 PM	0.6	0.9	1	0.9
6:00 PM	0.3	0.7	1	0.7
7:00 PM	0.3	0.7	0.5	0.7
8:00 PM	0.2	0.55	0.5	0.55
9:00 PM	0.2	0.45	0.5	0.5
10:00 PM	0.2	0.15	0.5	0.35
11:00 PM	0.2	0.15	0.5	0.25

APPENDIX B: EES Simulation for Redesign Approach 2

total energy savings from transfer air

$$\dot{Q}_{chilledwater,saved} = \frac{\dot{V}_{TA} \cdot 1.08 \cdot (T_{OA} - 55)}{12000} - \textbf{If}\,(\dot{V}_{labcooling} - (\dot{V}_{ACH} - \dot{V}_{TA}), 0, 0, 0, \dot{Q}_{UNMET}) - \dot{Q}_{TEMPDIFF}$$

$$\dot{Q}_{hotwater,saved} = \frac{\dot{V}_{TA} \cdot 1.08 \cdot (55 - T_{OA})}{12000} - \textbf{If}\,(\dot{V}_{labcooling} - (\dot{V}_{ACH} - \dot{V}_{TA}), 0, 0, 0, \dot{Q}_{UNMET}) - \dot{Q}_{TEMPDIFF}$$

Transfer Air

$$\dot{V}_{TA} = \textbf{Max}\,(V_{load}, V_{vent}) \cdot 0.95$$

Air to meet Sensible Load

$$V_{load} = \frac{SL_{NONLAB}}{1.08 \cdot (75 - 55)}$$

$$SL_{NONLAB} = 230 \cdot P_{non} \cdot Occ + 3.75 \cdot A_{FLOOR} \cdot LTG + A_{WALL} \cdot 0.109 \cdot (T_{OA} - 75) + A_{WND} \cdot SC \cdot SHGF_S \cdot CLF_S$$

$$A_{FLOOR} = 27088$$

$$A_{WALL} = 12209$$

$$A_{WND} = 5087$$

$$SC = 0.55$$

$$P_{non} = 215 \cdot \frac{T}{T}$$

Air to meet Stand. 62.1

$$V_{vent} = 0.06 \cdot A_{FLOOR} + 8 \cdot P_{non} \cdot Occ$$

UNMET LOAD IN LAB TO BE MET BY FCU

$$\dot{Q}_{UNMET} = \frac{1.08 \cdot (72 - 55) \cdot (\dot{V}_{labcooling} - (\dot{V}_{ACH} - \dot{V}_{TA}))}{12000}$$

$$\dot{V}_{labcooling} = \frac{SL_{LAB}}{1.08 \cdot (72 - 55)}$$

$$SL_{LAB} = 230 \cdot P_{LAB} \cdot Occ + 1.4 \cdot A_{FLOOR,LAB} \cdot LTG + A_{WALL,LAB} \cdot 0.109 \cdot (T_{OA} - 75) + A_{WND,LAB} \cdot SC \cdot SHGF_N \cdot CLF_N + 20.48 \cdot A_{FLOOR,LAB} \cdot EQUIP$$

$$A_{FLOOR,LAB} = 12424$$

$$A_{WALL,LAB} = 3921$$

$$A_{WND,LAB} = 1280$$

$$P_{LAB} = 98$$

$$\dot{V}_{ACH} = \frac{12 \cdot A_{FLOOR,LAB} \cdot 12}{60} \cdot VentSched$$

ADDED LOAD FROM PUTTING 75F INTO LAB

$$\dot{Q}_{TEMPDIFF} = \frac{1.08 \cdot \dot{V}_{TA} \cdot (75 - 72)}{12000}$$

Fan Energy

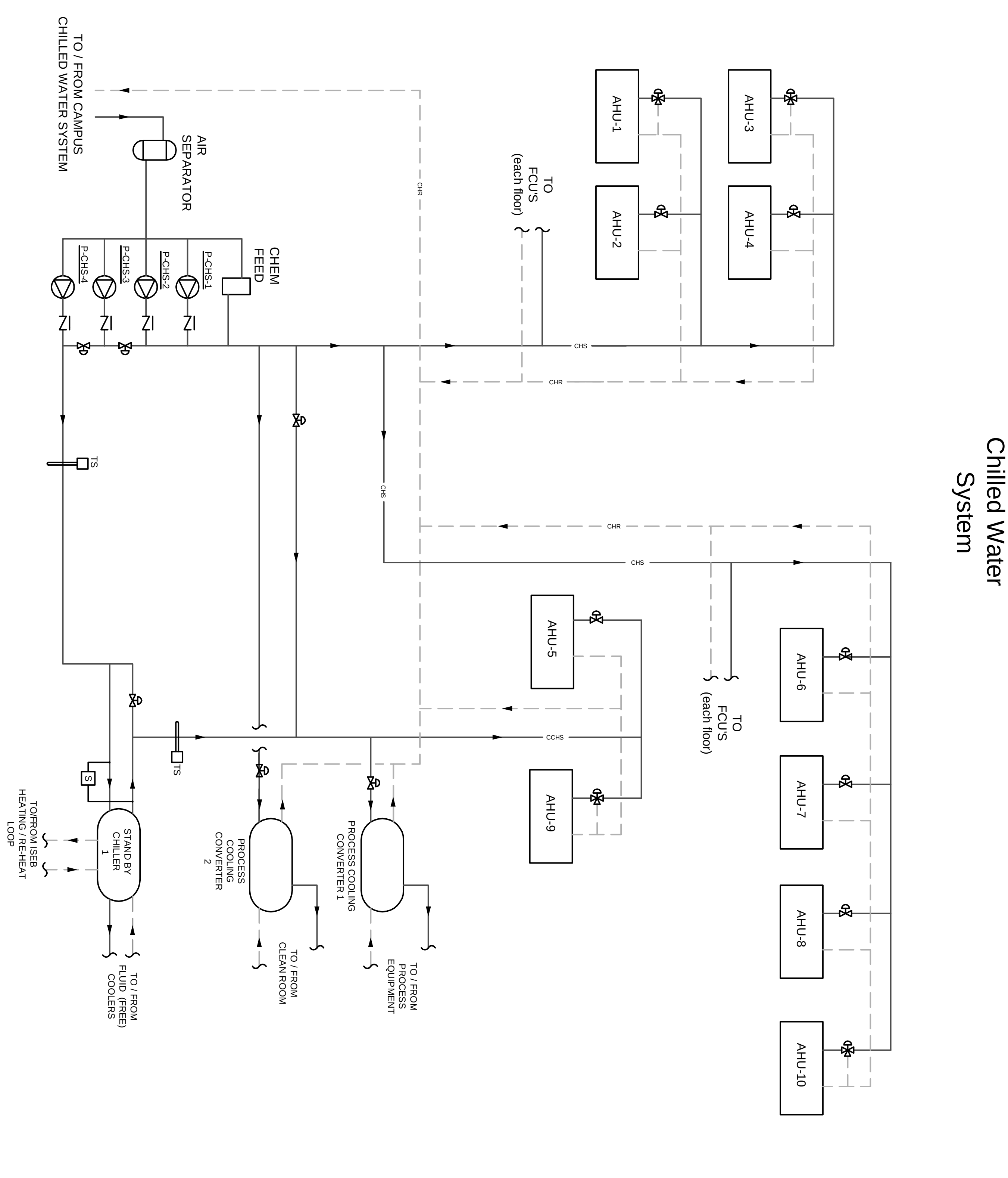
$$BHP = \frac{\frac{\dot{V}_{TA}}{8} \cdot SP}{6356 \cdot Fan_{eff}}$$

$$KW = BHP \cdot 0.7457 \cdot 8$$

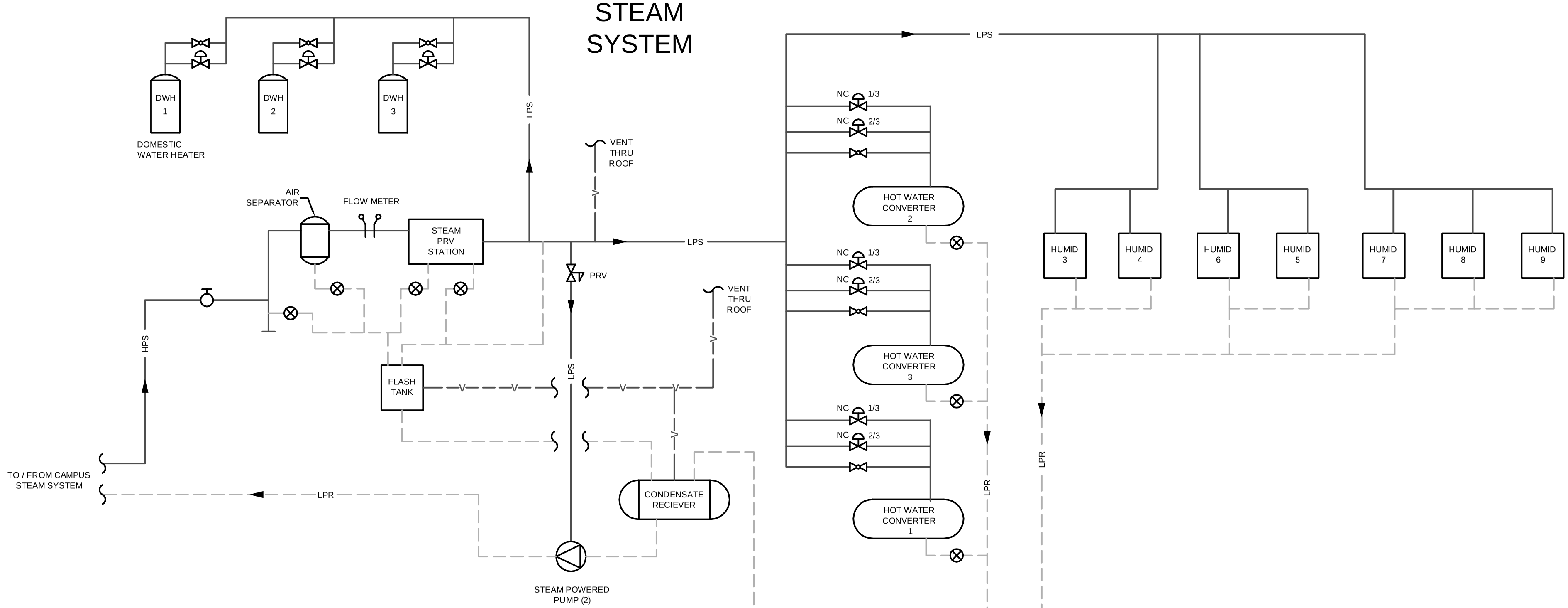
$$Fan_{eff} = 0.8$$

$$SP = 0.5 \cdot \left[\frac{\dot{V}_{TA}}{8 \cdot 2500}\right]^2$$

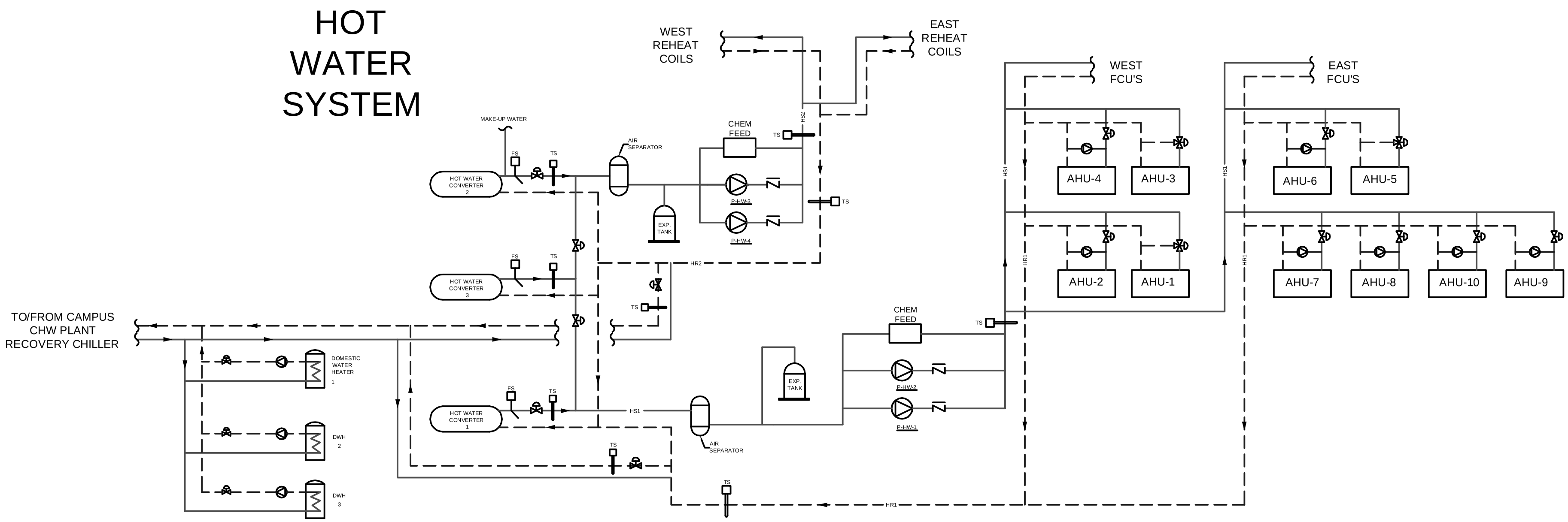
Appendix C: Base Mechanical System Schematics



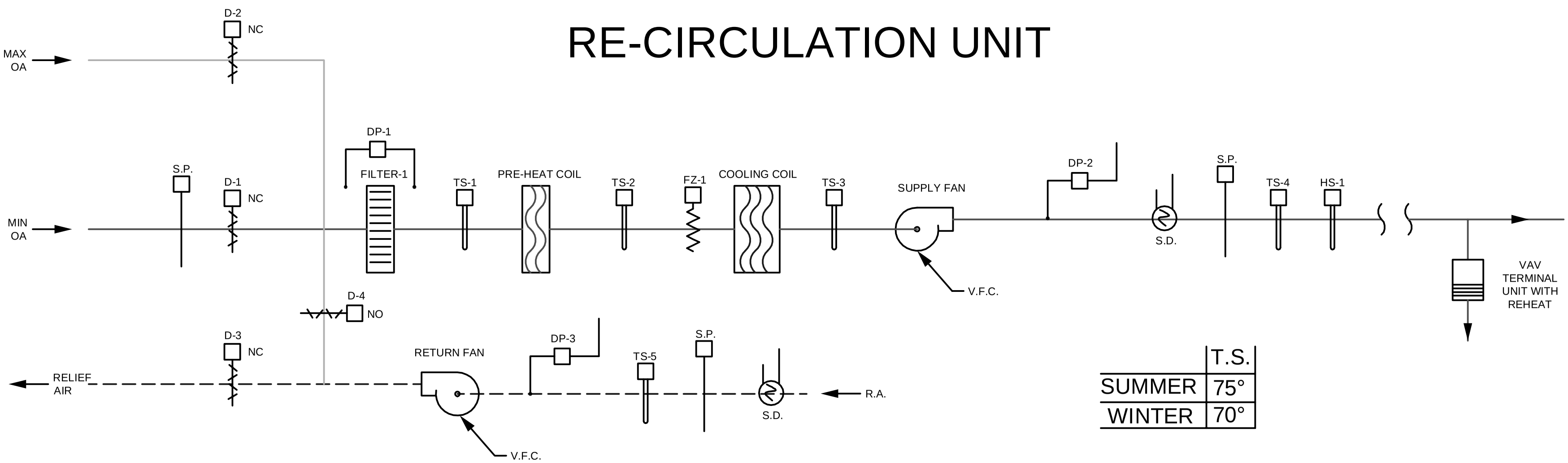
STEAM SYSTEM



HOT WATER SYSTEM

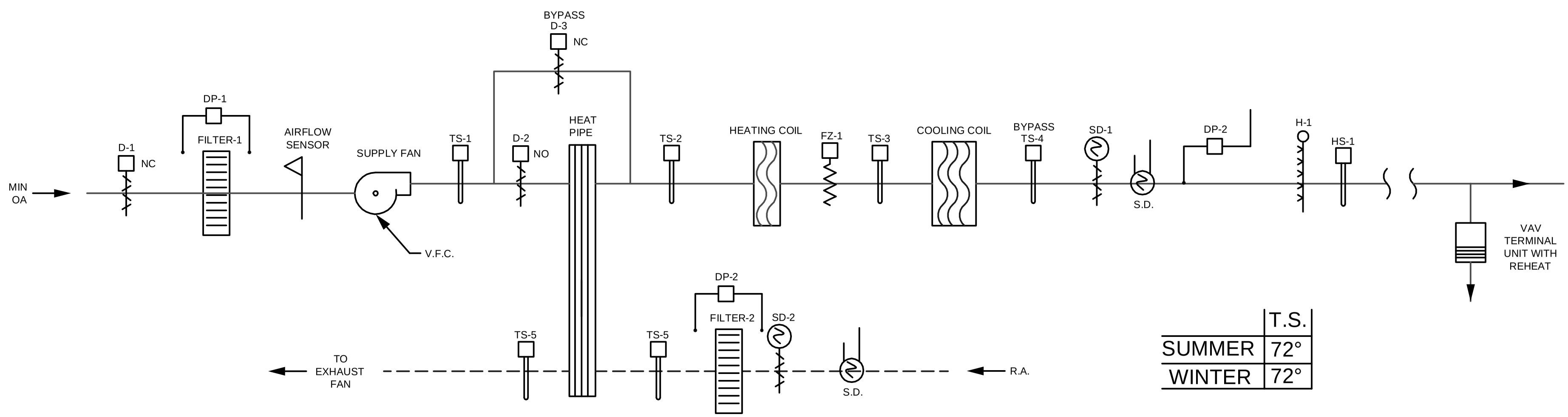


RE-CIRCULATION UNIT



	T.S.
SUMMER	75°
WINTER	70°

100% O.A. UNIT



	T.S.
SUMMER	72°
WINTER	72°

APPENDIX D: Masters Level Course Work

AE 557 and 558: Central Cooling and Heating System

Material learned in these courses was used to size fans, pumps, and piping. As well as provide a solid understanding of the existing mechanical conditions of the building. Also, both of these courses taught the use of the Engineering Equation Solver (EES) program which was used extensively in the analysis for this report.

AE 559: Computational Fluid Dynamics

Material learned in this course was applied for the computational fluid dynamics model of the passive chilled beam system. This course provided the knowledge to use the Phoenics VR software as well as a strong base for the analysis of indoor airflow.