Final Thesis Report

System Alternatives Analysis

Delaware County Community College

STEM Center

Media, PA

Dan Saxton

Mechanical Option

Faculty Consultant: Dr. Stephen Treado

Date Submitted: 4/7/11

DELAWARE COUNTY COMMUNITY COLLEGE

Media, PA

STEM CENTER

science
technology
engineering
math

PROJECT TEAM:
Owner: Delaware County Community College
General Contractor: E.R. Steubner, Inc.
Construction Manager: Reynolds Construction Management, Inc.
Architectects: Burt Hill
MEP and Fire Protection Engineers: Burt Hill
Structural Engineers: O'Donnell Naccarato MacIntosh

PROJECT INFORMATION:
Size: 105,000 SF, 4 stories
Cost: $28.7 Million ($274 per square foot)
Project Delivery Method: Design-Bid-Build
Construction Dates: Jan. 2008 - Sep. 2009

ARCHITECTURE:
The STEM Center is the new forefront building for the college, standing out by its size and features. Glass curtain wall wraps the south side main entrance, and a glass "prism" highlights the south-east end of the S-shaped building.

ELECTRICAL INFORMATION:
- Designed to follow existing campus electrical installation
- Transformers provide 208Y/120V power
- 250kW generator for Normal/Emergency power
- (2) 1000kVA transformers in a double-ended, main-tie-main substation

STRUCTURAL INFORMATION:
Floor System: Steel composite construction with 3" metal decking and 2-1/2" lightweight concrete
Typical Floor Sizes:
Beams: W16x26 Girders: W21x50
Interior Columns: W12x109
Exterior Columns: W12x87
Typical Structural Bay: 30' x 30'
Roof System: Open web steel bar joists spaced 6' on center and 1-1/2" 22 gage metal decking

MECHANICAL INFORMATION:
- Labs/prep rooms with fume hoods exhaust to roof fans with refrigerant coils for heat recovery
- Existing: (3) 400 ton gas-fired absorption chillers
- New: 650 ton electrical centrifugal chiller
(4) 600 ton crossflow cooling towers
(2) Bryan 250 BHP dual fuel boilers
80,000 CFM roof mounted AHUs
- Finned tube radiation used on perimeter of Atrium glazing to prevent condensation

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Delaware County Community College

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Executive Summary

(Photos Provided by Burt Hill)

The Delaware County Community College Science, Technology, Engineering, and Mathematics (STEM) Center is a new addition to their Marple Campus, and is part of the two-building STEM Complex. At 105,000 square feet and four stories it is a focal point for the campus, and stands out with both architectural and sustainable features.

With a sophisticated existing design of the mechanical systems, alternative solutions were analyzed for the core of the building, excluding the classroom and office spaces. This report outlines the mechanical redesign through a radiant floor heating and cooling system with a supplemental dedicated outdoor system, as well as an investigation into the potential use of natural ventilation.

The analysis shows that a radiant floor design has the means to completely handle the sensible heating and cooling capacity for the spaces selected for redesign. Additionally, it is specified that the ventilation and necessary latent load be accounted for by an outdoor air system that means in increase in air handling equipment, but overall reduction in ductwork and terminal units. By using this system, and significantly decreasing the airside energy usage, a yearly cost savings is calculated.

The construction management breadth implications for the use of these alternate systems includes a higher calculated upfront cost and increase in construction time necessary through the special construction required for a radiant floor slab.

Positive implications of the radiant system exist for the acoustical breadth analysis include a lower overall sound pressure level and room criterion for the spaces of concern. Addition analysis proves that the necessary floor finish alteration to correspond with the radiant slab design has minimal effect on the acoustical reverberation characteristics.

Finally, investigation of the use of natural ventilation, including full computational fluid dynamics simulation, determines this as a feasible alternative design to further aid the reduction of operating cost achieved by the radiant floor system. Altogether it is determined that these proposed alternative systems would improve the overall mechanical design of the STEM Center on the Delaware County Community College campus.

SECTION 1 – Mechanical System Description

1.1 – New Construction Background

The STEM Center is a part of new construction located adjacent to the existing buildings that make up the Delaware County Community College campus. Though neighboring other campus buildings, it was necessary to implement all unique systems for both the STEM Center and Technology Building mechanical designs. The considerations going into this all new mechanical design varied in degree, but included low initial and operating costs, excellent air quality, adequate heating and cooling, and high energy efficiency. A goal for the project from the very beginning was to achieve LEED certification to show achievement of high building sustainability.

Special considerations for the STEM Center, specifically, mainly stemmed from the unique occupancy and space types that were included in this type of science-based building. Room types such as laboratories and preparation rooms, as well as other standard educational spaces, meant heightened requirements for air quality, ventilation, and exhaust. Laboratories for the building include physics, biology, anatomy/physiology, earth and space, CAD, and both organic and general chemistry.

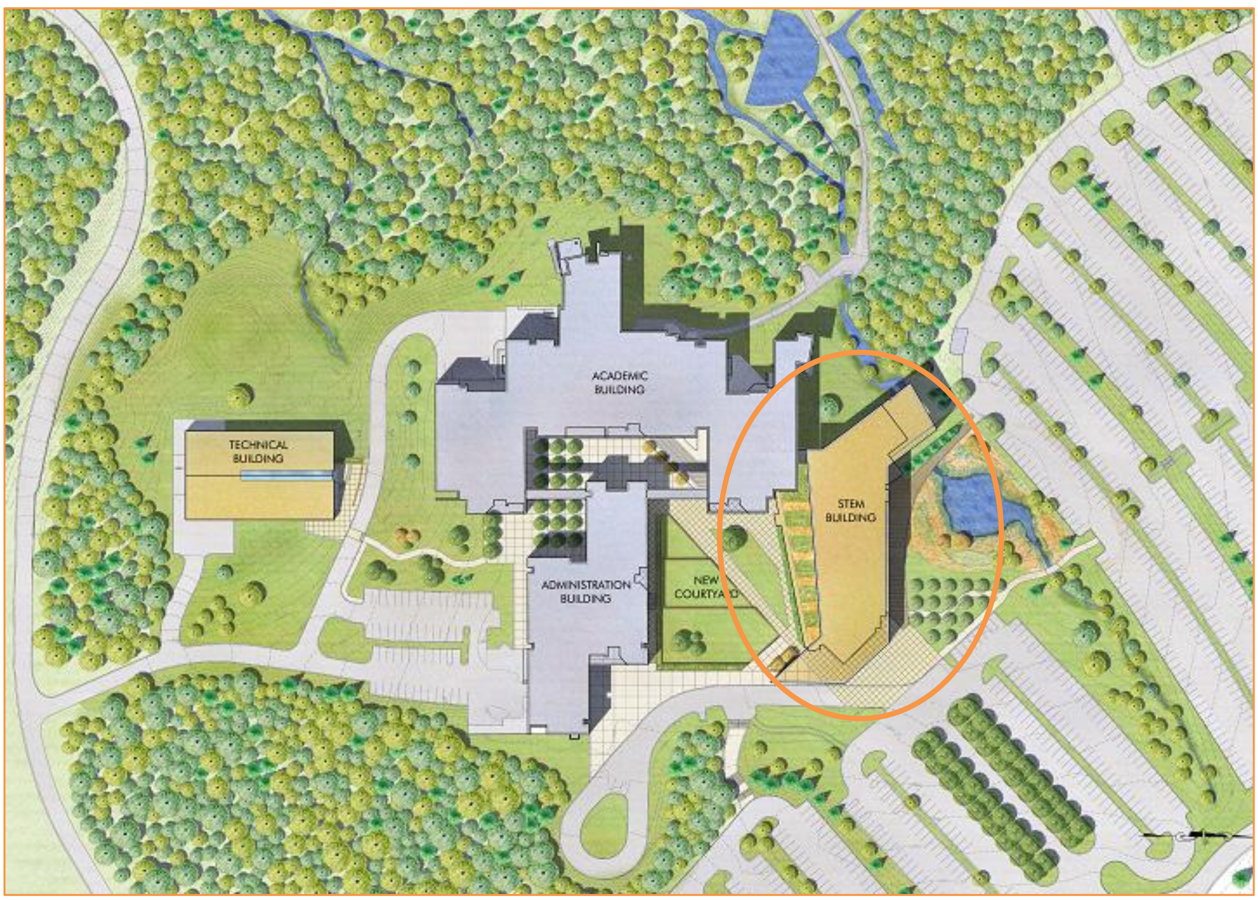

Figure 1: Site Plan – New Construction
(Photo Provided by Burt Hill)

1.2 – Mechanical Design Objectives

A major concern was placed on the operating cost of the mechanical system, as for a college such as DCCC, the hope was to be capable of maintaining the building systems with ease and as little expense as possible. Considerations were taken to provide energy efficient equipment to potentially decrease utility bills.

Overall, limiting factors on the STEM Center design were minimal. With the site being on the DCCC Marple Campus, in the greater Philadelphia area, it was obvious that there would be a slightly higher emphasis on heating than cooling, but a need for good balance and control altogether. The building design and architecture itself, containing a sizeable glass façade on south side at all four levels, suggested a concern for high amount of solar gain as well. The orientation of the building is suitable though, as the majority of the glazing won't receive the significant glare they might if facing east or west.

Figure 2: Site Plan – Aerial View

(Photo Provided by Burt Hill)

1.3 – Mechanical Equipment Breakdown

The mechanical system for the STEM Center is characterized by fewer, yet larger, pieces of equipment. As mentioned in Section 1.1, air handling is taken care of by just two AHUs designed at 89,500 cfm apiece, and there is one chiller and two boilers for the building. The information for these pieces of equipment is shown below in Tables 1 through 4.

Air Handling Units			
Name	**Airflow (CFM)**	**Min. OA (CFM)**	**Outside Air %**
AHU-4	89,500	66,000	74
AHU-5	89,500	66,000	74

Table 1: Air Handling Units

	Cooling Coil			Heating Coil		
Name	**Entering Air Temp. (F)**	**Leaving Air Temp. (F)**	**MBH**	**Entering Air Temp. (F)**	**Leaving Air Temp. (F)**	**MBH**
AHU-4	84.4	50.5	5,351	10	50	3,861
AHU-5	84.4	50.5	5,351	10	50	3,861

Table 2: Cooling and Heating Coils

Water Cooled Chiller		Evaporator		Condenser	
Capacity (tons)	**kW/ton**	**EWT (F)**	**LWT (F)**	**EWT (F)**	**LWT (F)**
700	0.58	56	45	85	97

Table 3: Water Cooled Chiller

Gas-Fired Boilers				
Mark	MBH In	MBH Out	EWT (F)	LWT (F)
B-1	6,000	5,040	160	190
B-2	6,000	5,040	160	190

Table 4: Gas-Fired Boilers

Additionally, there are the four cooling towers for the building, rated at 604 tons each. The information for these as well as a breakdown of the 16 major pumps used in the mechanical system is shown below in Tables 5 and 6. As can be seen in Table 6, the majority of the pumps are equipped with variable frequency drives.

Cooling Towers						
Mark	Capacity (tons)	EWT (F)	LWT (F)	CFM	GPM	Motor HP
CT-1	604	97	85	136,170	1524	30
CT-2	604	97	85	136,170	1524	30
CT-3	604	97	85	136,170	1524	30
CT-4	604	97	85	136,170	1524	30

Table 5: Cooling Towers

Pumps					
Mark	GPM	Min. Efficiency (%)	RPM	VFD? (Y/N)	HP
P/7	1527	80	1160	No	25
P/8	1527	80	1160	No	25
P/9	3327	90	1760	Yes	125
P/10	3327	90	1760	Yes	125
P/11	1623	82	1760	No	40
P/12	392	70	1760	No	7.5
P/13	392	70	1760	No	7.5
P/14	20	40	1760	No	0.5
P/15	20	40	1760	No	0.5
P/16	320	74	1760	Yes	10
P/17	320	74	1760	Yes	10
P/18	200	67	1760	Yes	7.5
P/19	200	67	1760	Yes	7.5
P/20	320	69	1760	Yes	15
P/21	320	69	1760	Yes	15
P/CW	75	69	1160	Yes	1

Table 6: Pumps

1.4 – Lost Usable Space

Although there is no basement in the STEM Center, usable floor area is occupied by mechanical space on all four floors. At the ground floor, a sizeable mechanical room is located in the western portion of the building with a total floor area of 1,852 square feet.

For floors 2 through 4, there are four mechanical shafts that run vertically through the building. Two of the shafts (Shaft 3 and Shaft 4) run through the middle of the building where science labs are located on the 2nd and 3rd floor. These two are not a factor on the fourth floor, which is roof space in the middle portion of the building. The other two (Shaft 1 and Shaft 2) occupy all of the top three floors and are at each end of the building and smaller in size. Additionally found on the outdoor roof portion of the 4th floor is 3,687.45 square feet occupied by mechanical equipment.

A complete breakdown of the lost usable space due to mechanical systems is shown below in Table 7, and totals 6,651.43 square feet of floor area.

Floor	Space	Area (SF)
1	Mech. Room	1,851.79
2	Shaft 1	80.50
2	Shaft 2	91.24
2	Shaft 3	106.29
2	Shaft 4	192.18
3	Shaft 1	80.50
3	Shaft 2	91.24
3	Shaft 3	106.29
3	Shaft 4	192.18
4	Shaft 1	80.50
4	Shaft 2	91.24
4/R	Mech. Space	94.79
4/R	Mech. Space	145.94
4/R	Mech. Space	3,172.08
4/R	Mech. Space	28.00
4/R	Mech. Space	246.67
ALL	TOTAL	6,651.43

Table 7: Lost Usable Space

1.5 – Description of System

Air Side Description

The two aforementioned air handling units that serve the entire building are roof mounted and each of these units contains a heat recovery coil, pre-heat coil, and a chilled water coil. A detailed drawing of AHU-4 is seen in Figure 3. Ductwork traveling down vertically through four different mechanical shafts lead to variable air volume (VAV) terminal units that are equipped with heating coils, with capacities ranging up to 4400 cfm. Fans for supply and return are all provided with variable frequency drives and full economizer capabilities. In terms of control, flow measuring stations are also used for outdoor air control.

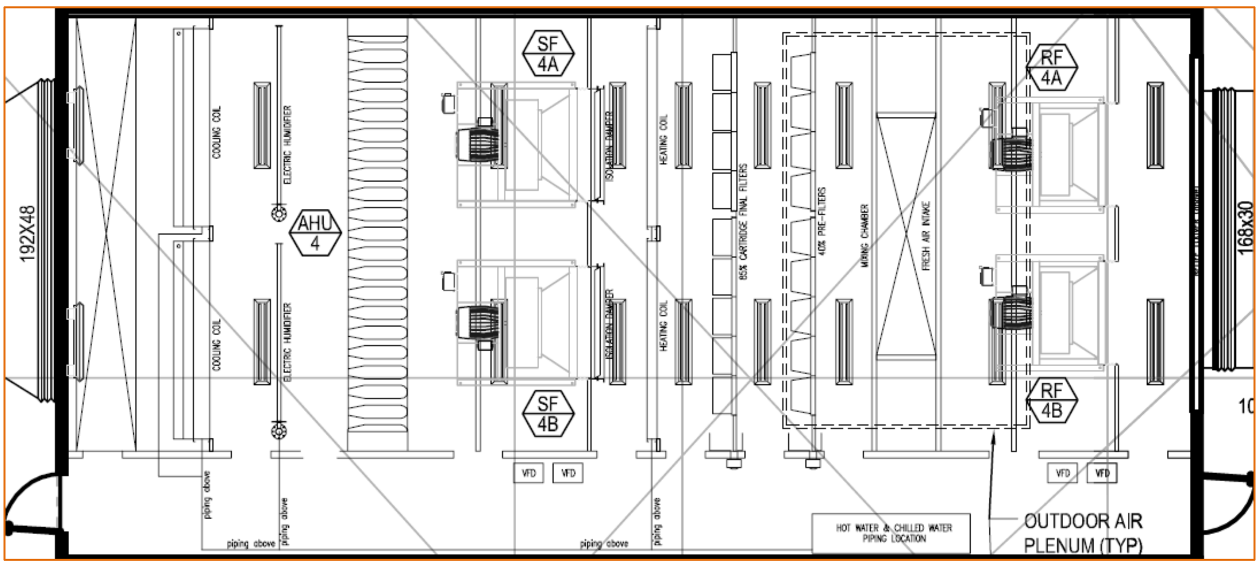

Figure 3: Air Handling Unit (AHU-4) Detail

While the supply and return paths for the air side mechanical system are relatively simple and straightforward, the exhaust system for the building is slightly more unique in order to cater to the laboratory and preparation spaces located on the top three floors. As was previously discussed, there are 13 science labs, most of which have dedicated preparation or storage rooms that require adequate exhaust of air. To comply, fume hoods are located in each lab and preparation room in the building. These are ducted vertically to the roof mounted exhaust fans, which are equipped with refrigerant coils for heat recovery systems. Pressure is controlled using a space pressurization monitoring system, and each exhaust fan is provided with a variable frequency drive.

Water Side Description

On the water side of the mechanical system, a 700 ton water cooled chiller accounts for the chilled water. This is arranged with (2) 25 hp primary chilled water pumps and (2) 125 hp secondary chilled water pumps to make a primary/secondary chilled water system as is shown in the in Figure 5: Chilled Water Schematic. In Figure 5, pumps P-7 and P-8 are primary chilled water pumps and pumps P-9 and P-10 are secondary chilled water pumps, and all primary/secondary pumps are equipped with variable frequency drives.

With the construction of the STEM Center, the intent in the mechanical design was to replace the existing cooling tower, and this was done with the addition of (4) 600 ton induced draft crossflow cooling towers (shown in Figure 4: Condenser Water Schematic). The new chiller and cooling towers are adequately served by the new 40 hp in-line condenser water pump, which is P-11 in Figure 4.

For the hot water system, (2) Bryan 250 BHP dual fuel heating hot water pumps account for the hot water heating using natural gas, as discussed in Section 1.3. For the heating hot water system, a primary/secondary arrangement is utilized as well, consisting of (2) 7.5 hp pumps (P-11 and P-12). Additionally for the hot water system, and also located in the boiler room, are two pumps for the heat exchanger, two for the reheat coils, two for the preheat coils, and also two in-line pumps for the fin-tube radiation heating that is utilized for the exterior glass façade on the south side of the building to prevent condensation.

Shown below in Figures 4 through 6 are the Condenser Water, Chilled Water, and Heating Hot Water Schematics drawn using Microsoft Visio software with reference to the design documents.

System Schematics

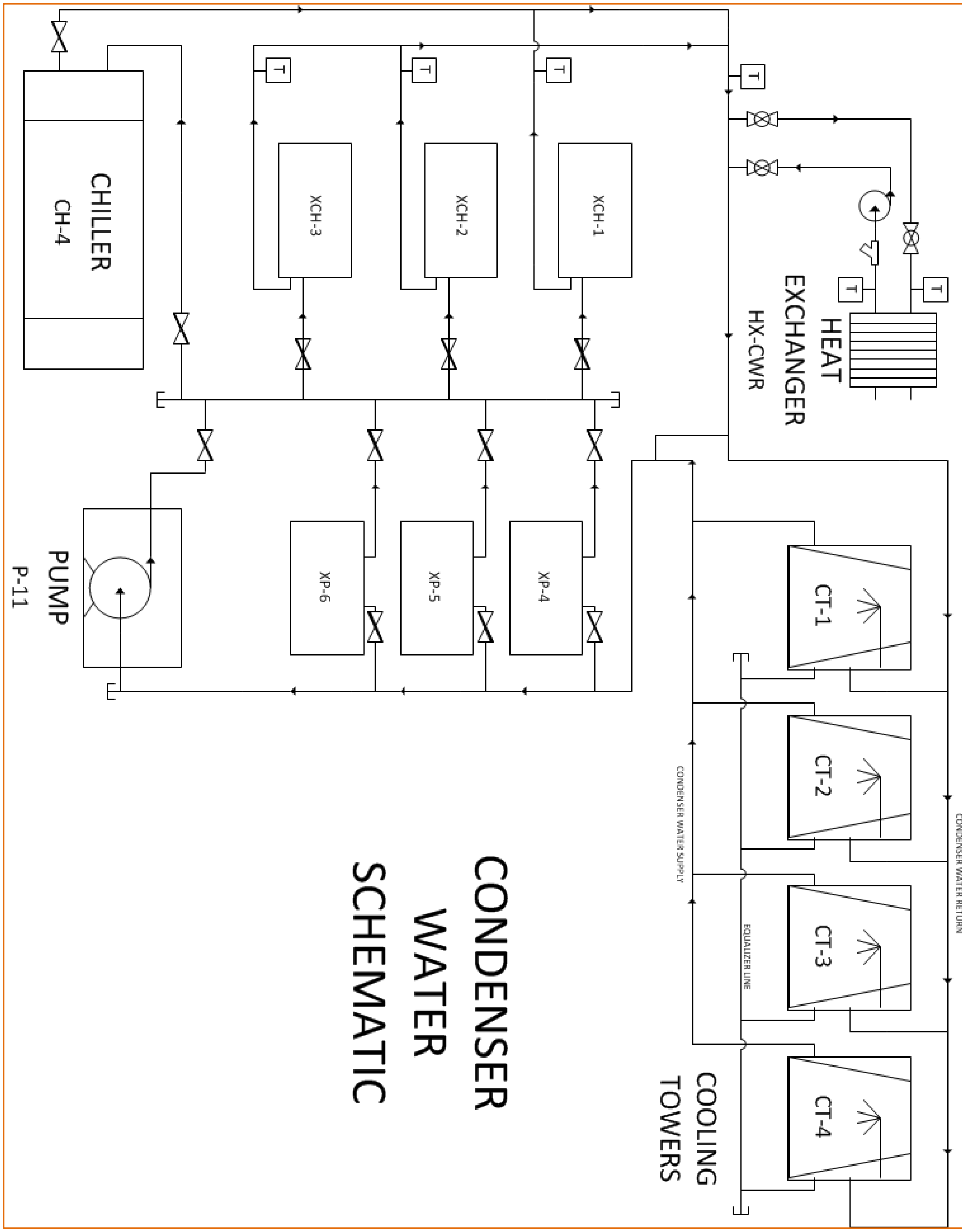

Figure 4: Condenser Water Schematic

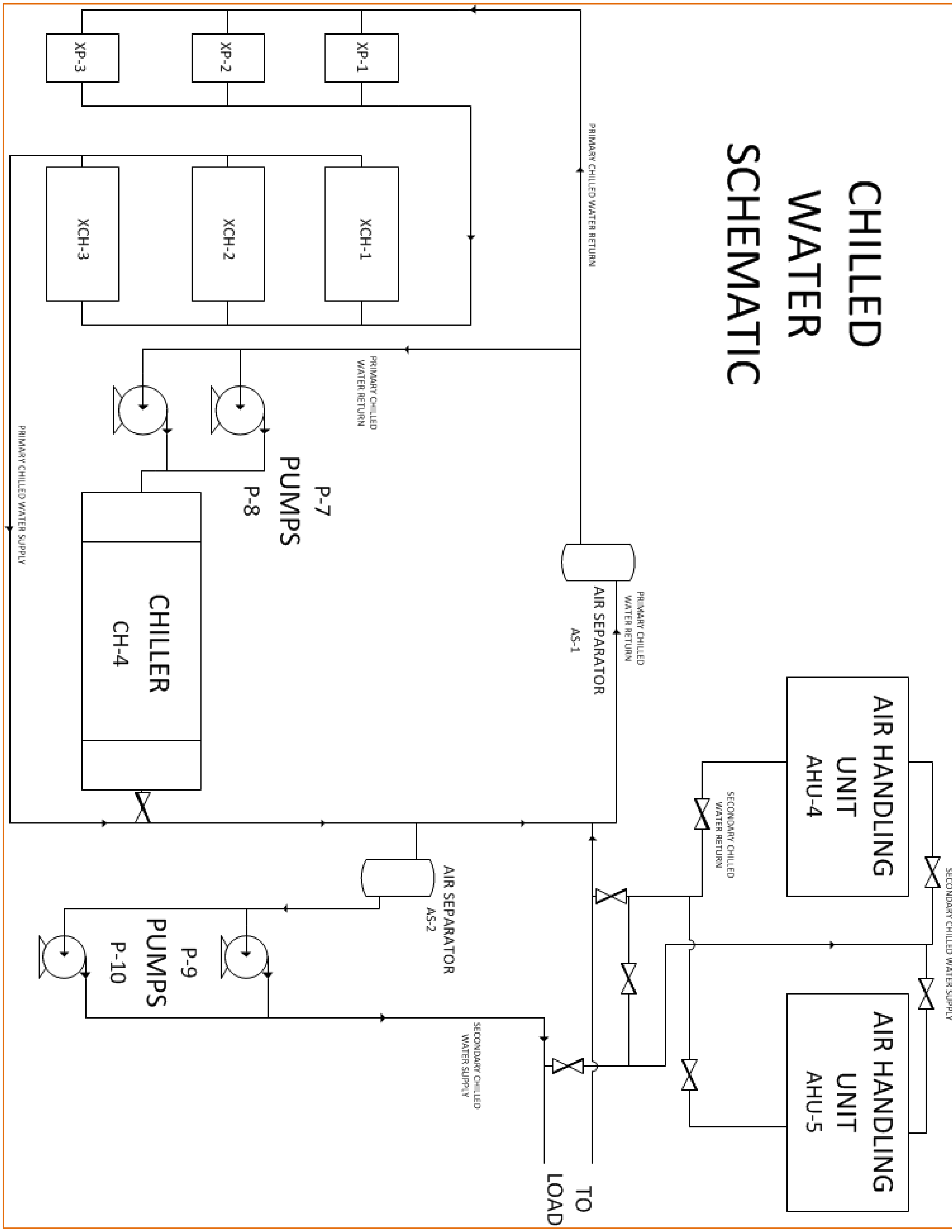

Figure 5: Chilled Water Schematic

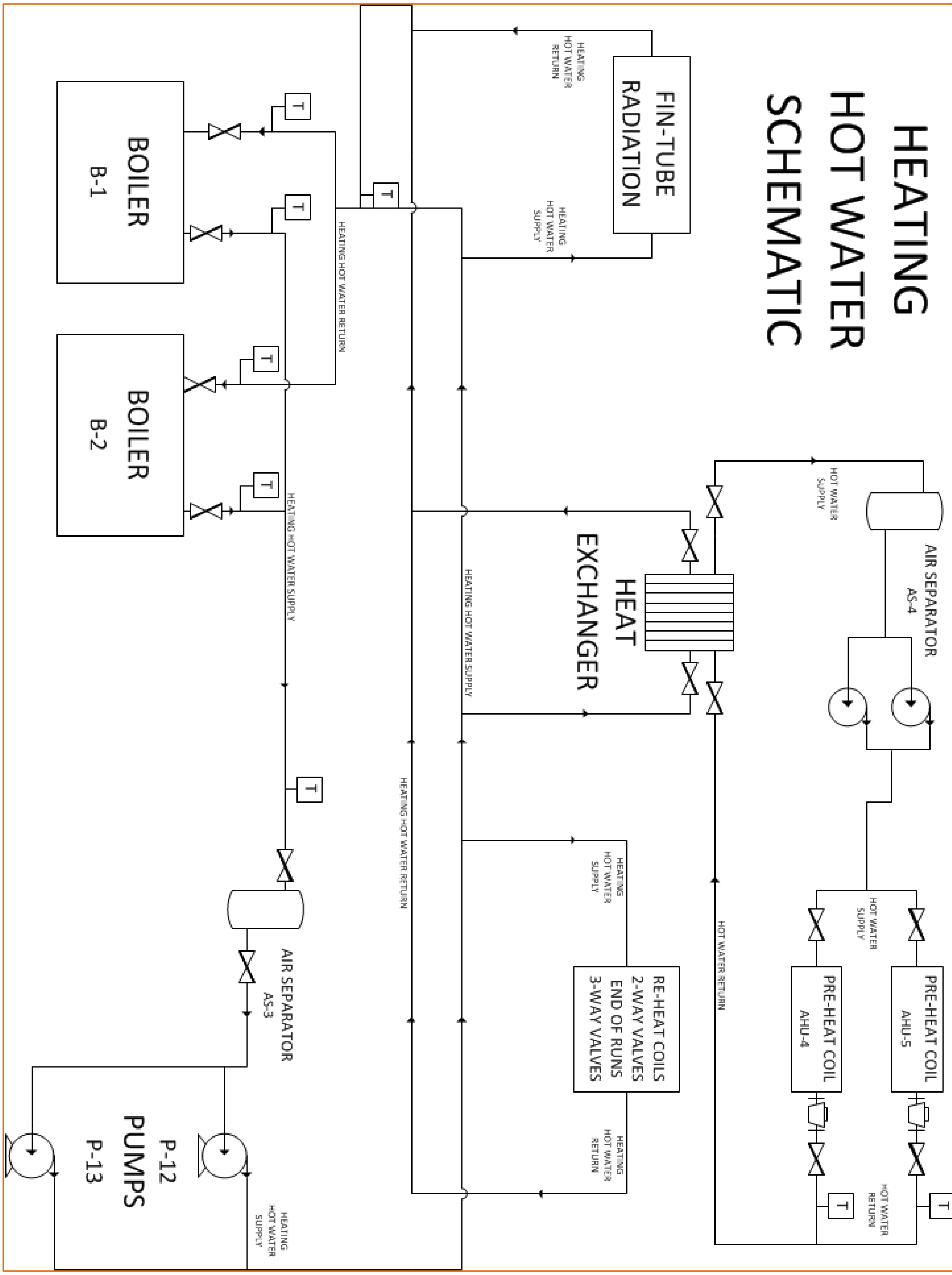

Figure 6: Heating Hot Water Schematic

1.6 – Energy Sources and Rates

For the STEM Center, energy usage consists of electricity and natural gas, which are primarily for cooling and heating, respectively. Although it is an educational facility on a main campus, neither the heating nor cooling are district systems, and are specific to this building. The rates used for simulations for each energy source were a yearly average taken from the design documents provided by Burt Hill for electricity and gas. These values are shown in Table 8.

Energy Source	Energy Rate
Electricity	$0.089 /kWh
Natural Gas	$1.347 /therm

Table 8: Energy Rates

1.7 – Design Considerations

Indoor and Outdoor Air Conditions

From the ASHRAE Handbook of Fundamentals 2009, and using weather data for Philadelphia, PA (within 10 miles of the campus), the indoor and outdoor air conditions were determined to be as shown below in Table 9.

	Heating Dry Bulb Temperature (F)	Cooling Dry Bulb Temperature (F)	Cooling Wet Bulb Temperature (F)
	99.6%	0.4%	0.4%
Philadelpha, PA	**11**	**93.1**	**75.7**

Table 9: Design Air Conditions

For the interior design, a value of 75°F was used for room temperature and 58°F for supply air temperature. Also, an assumption of 0.11 air changes per hour was made based on information provided in design documents. These air conditions were used for the TRACE™ simulation to adequately model the climate in Media, PA.

1.8 – System First Cost Analysis

First cost for the mechanical system and associated equipment was decreased slightly due to the partial use of existing systems adjacent to the STEM Center site on campus. For the two custom air handling units, the cost was $1,001,335, and for the new cooling towers, it was another $280,580. Those values, as well as the cost of the two new boilers and 600 ton chiller and all the additional necessary costs, totaled the overall mechanical system first cost to **$7,179,242.70**, or roughly $68.37/SF. This number is as anticipated, due to the partial utilization of existing equipment and construction, and the consolidation of air handlers.

SECTION 2 – ASHRAE Standard and LEED Evaluation

2.1 – Design Ventilation Requirements

In preliminary investigation of the original mechanical system design, an analysis of ASHRAE Standards 62.1 and 90.1 was conducted, and the results of this study can be found in Technical Report 1. Among the sections evaluated, the minimum outdoor air ventilation rate was determined for the building based on equations and requirements from ASHRAE Standard 62.1. These values were then compared to the design maximum for the two rooftop air handling units. Nearly all of the building air handling is accomplished by two rooftop AHUs, and a very small percentage (for a few machine rooms) is handled by fan coil units. The two AHUs each are capable of a maximum airflow of 89,500 total cfm, including 66,000 cfm of outdoor air. The total outdoor air intake required according to the ASHRAE Standard 62.1 for all four floors was just 43,748 cfm, as is tallied in Table 10. This comparison is shown below and points out overcompensation on the part of the air handlers for outdoor air requirements.

	Outdoor Air Intake Required (cfm)
First Floor	18,349
Second Floor	10,007
Third Floor	10,462
Fourth Floor	4,930
Total	**43,748**
Design Maximum Primary Air (cfm)	**132,000**
TRACE Simulation Design OA (cfm)	**56,309**

Table 10: Outdoor Air Requirements Comparison

2.2 – Design Heating and Cooling Loads Estimates

For the estimation of the design load for the STEM Center, Trane TRACE™ 700 Version 6.2 was utilized. To simulate the air handling, AHU-4 and AHU-5 were modeled as one unit, treating the whole building as one system, for all 160 spaces. Both air handlers are identical in size and performance, therefore this assumption seemed reasonable for a block load simulation. Shown below in Figure 7 is a 3D view of the main distribution of air throughout the STEM Center, stemming down from the two rooftop units.

All design data for the TRACE load simulation was taken from design documents generously provided by Burt Hill. This included the Autodesk Revit model, which was converted to a gbXML file for import into TRACE. Along with the imported building room dimensions, U-values for floor, roof, wall, and window construction were determined also by provided design documents.

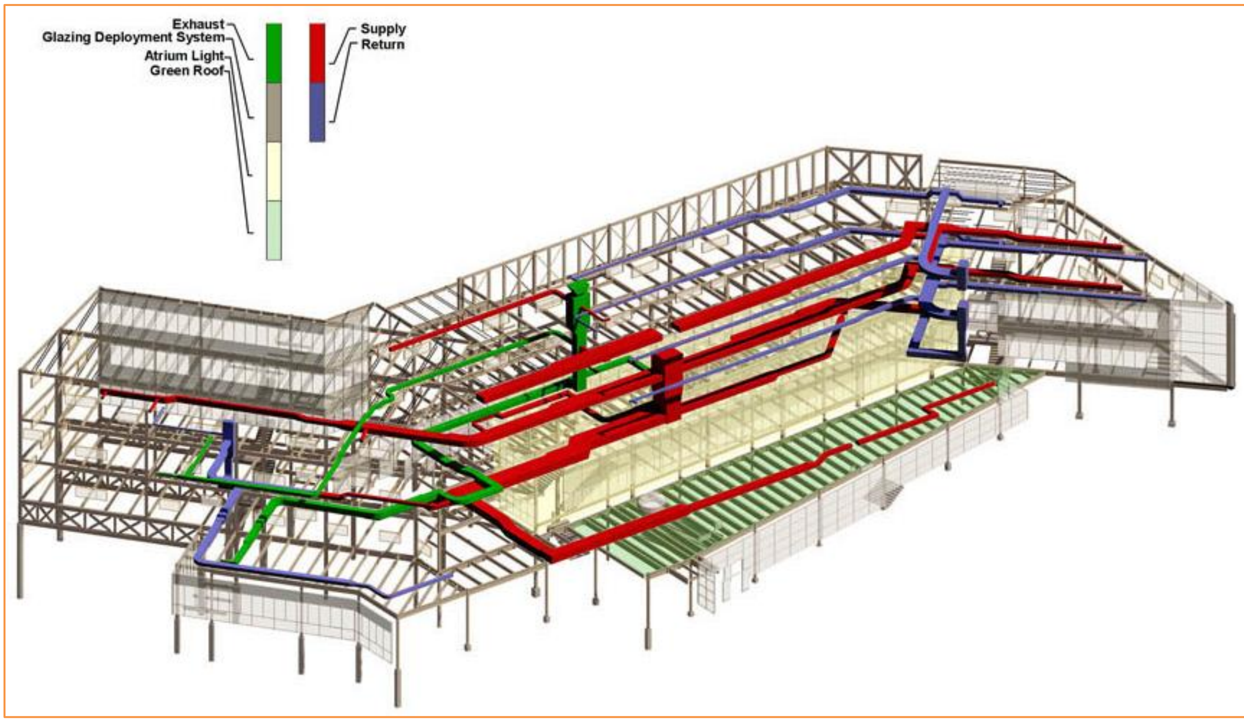

Figure 7: Air Distribution
(Photo provided by Burt Hill)

This TRACE simulation took into account the aforementioned indoor and outdoor air conditions, as well as many other factors, including mechanical equipment and various load sources. With TRACE, a model was done based on several factors from airflow rates to space occupancies to the systems used and even the aforementioned energy rates. For load schedules, the TRACE template for typical College was used for Lighting Loads, Miscellaneous Loads, People Activity, Ventilation, and Infiltration. This was deemed suitable for a building such as the STEM Center that was primarily used for academic purposes on a daily basis with minimal after-hour activity. For this schedule, the highest rates occur between 8 AM and 5 PM. The results from TRACE were compared with the energy modeling results of the designer using IES (Integrated Environmental Solutions), a similar program. The outcomes of the separate models can be seen below in Table 11.

	Cooling (SF/ton)	Heating (BTUh/SF)	Total Supply (cfm/SF)	Ventilation Supply (cfm/SF)
Computed	201.220	40.510	1.328	0.557
Design Documented	197.310	40.696	1.282	0.559
% Difference	1.982	0.457	3.579	0.396

Table 11: Computed and Design Load Comparison

As can be seen in Table 11, the results by the designer using IES software were very near those from the simulation conducted in TRACE. The results for both cooling capacity in square feet per ton were within 2% of each other, and the two computed values for heating capacity were even closer in difference.

2.3 – Energy Usage Estimate

For the STEM Center, electricity is the main energy usage for cooling and primarily natural gas is consumed for heating in the building. By using TRACE for energy modeling, many different loads were taken into account, including lighting, solar, and occupancy loads. The plants selected in TRACE were a water-cooled chiller and a gas-fired boiler, which are described in greater detail in Tables 3 and 4 in Section 1.3 of this report. These two were designed at 700 tons, and 12,000 MBh, respectively.

Energy Results

Shown below in Table 12 are the results of the energy modeling conducted by TRACE. As anticipated, the auxiliary loads from fans and pumps resulted in a significant percentage (48.6%) of the total building energy usage. This particular load category was greatly higher than that of the heating and cooling system and may be the result of an oversimplification somewhere along the way. Still, however, the amount of energy usage by each category is reasonable, and all add up to a total building energy usage of **8,545 mBtu/year**.

	Electric (kWh)	Gas (kBtu)	Water (1000 gal)	% of Total Building Energy	Total Building Energy (kBtu/yr)
Heating					
Primary Heating		650,978			650,978
Heating Accessories	37,888				129,310
Heating Subtotal	**37,888**	**650,978**	**0**	**9.1%**	**780,288**
Cooling					
Cooling Compressor	72,830				248,570
Tower/Cond Fans	109,988		530		375,389
Condenser Pump	113,227				386,445
Cooling Accessories	2,847				9,717
Cooling Subtotal	**298,893**	**0**	**530**	**11.9%**	**1,020,121**
Auxiliary					
Supply Fans	772,751				2,637,400
Pumps	894,410				3,052,620
Aux Subtotal	**1,667,161**	**0**	**0**	**66.7%**	**5,690,020**
Lighting	**282,857**	**0**	**0**	**11.3%**	**965,391**
Receptacles	**26,160**	**0**	**0**	**1.0%**	**89,284**
TOTAL	**2,312,958**	**650,978**	**530**	**100%**	**8,545,104**

Table 12: Energy Usage Breakdown

Energy Consumption Breakdown

A report for monthly energy use was also compiled and showed a general peak of electrical energy usage in the summer months and a peak of fuel energy usage in the winter months. Though these graphs are not as perfectly normally distributed as would be assumed, they still provide evidence of the general pattern of energy usage based on necessary heating and cooling loads throughout the year. The highest therm consumption occurs in January and

February, and likewise the highest amount of kilowatt-hours is in August. The numerical breakdown of monthly energy consumption is shown in Table 13, and Figures 8 and 9 display the pattern of energy usage in kWh and therms, respectively.

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Electric (kWh)	163,326	136,388	200,729	184,997	212,509	215,751	199,753	227,348	194,224	204,208	192,592	181,132	2,312,957
Gas (therms)	1,138	1,117	770	465	282	239	198	250	250	462	580	760	6,511
Water (1000 gal)	7	5	17	25	61	83	105	101	60	31	22	12	529

Table 13: Monthly Energy Consumption

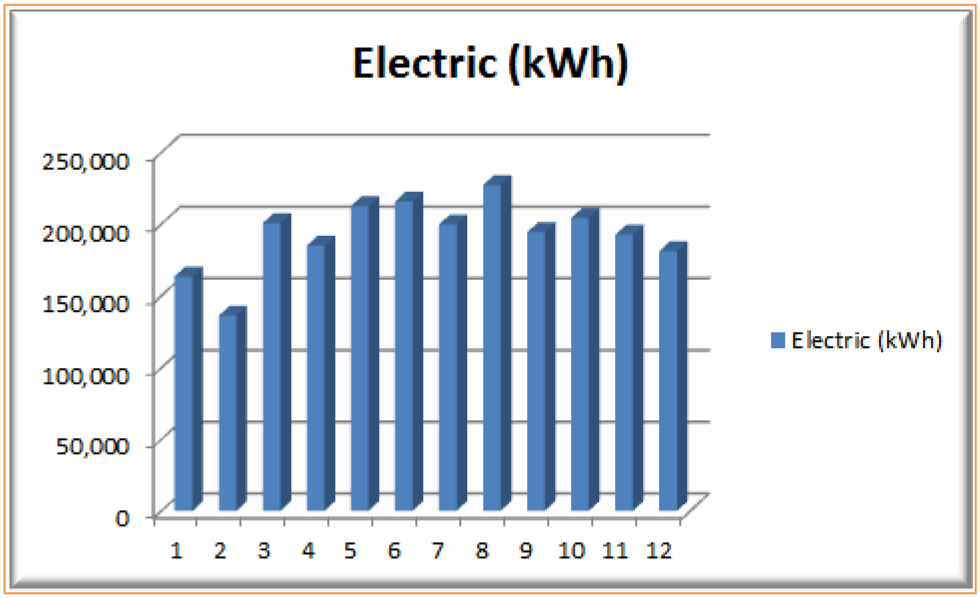

Figure 8: Electrical Energy Consumption

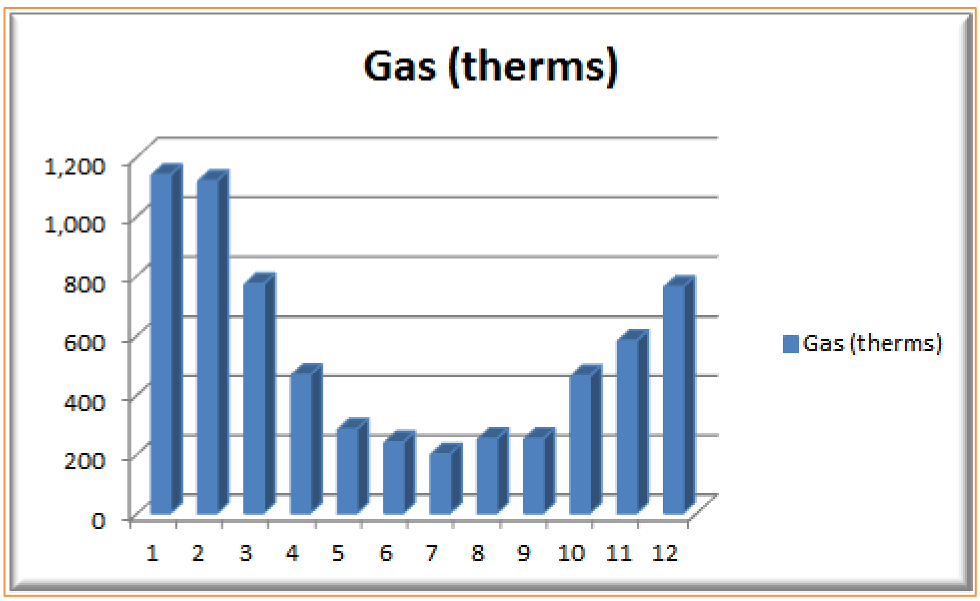

Figure 9: Gas Consumption

The electrical energy total was also broken down by usage, including cooling, heating, fans, lighting, and general equipment, as per the data outputs from TRACE. This breakdown is represented graphically in Figure 10, and once again it can be seen that a substantial amount of energy is used for auxiliary purposes (41.5%), as well as for cooling systems (49.32%).

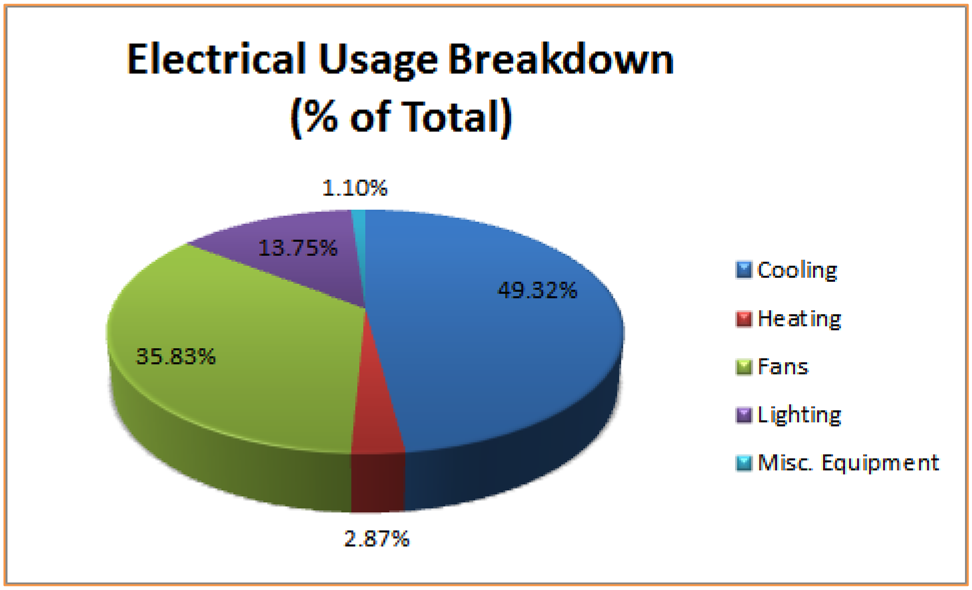

Figure 10: Electrical Usage Breakdown

In total, the utility cost was estimated to be $219,459 per year, which equals $2.10 per square foot. This is a reasonable estimate for a building of this size and specialty, and was just slightly higher than the results produced by the Burt Hill mechanical design team. As was previously mentioned, the designer energy simulation was conducted using IES software. This program was selected largely due to its versatility in providing graphical features and overall ability to model the building loads, specifically those produced by the solar gain from the south side glass curtain wall that is present at all floors. The results for the IES simulation have been generously provided by Burt Hill, and it was seen that the TRACE model produced comparable data for energy usage that erred on the side of an overestimate.

2.4 – LEED©-NC Analysis

A goal for the project from the very beginning was to achieve LEED certification according the requirements and point system outlined by the USGBC (U.S. Green Building Council). The USGBC has existed since 1993 and has been rating building green performance since LEED Version 1.0 debuted in August of 1998. Since then it has developed into a highly respected and pursued standard for excellence in green building, and continues to progress and improve just as the building industry and need for sustainability also advances.

Figure 11: STEM Center Green Roof
(Photo provided by Burt Hill)

The LEED rating for New Construction and Major Renovations takes into account 7 different topics: Sustainable Sites (SS), Water Efficiency (WE), Energy and Atmosphere (EA), Materials and Resources (MR), Environmental Quality (EQ), Innovation in Design (ID), and Regional Priority (RP). It was the aim for the project team to achieve a LEED Silver design, which they decided would be possible without significantly increasing the project budget.

The rating scale used for the building design was version 2.2 and a copy of the LEED Review including the anticipated scoring based on design objectives can be found in Appendix A of Technical Report 3. For version 2.2, there are 69 total points possible, and this report will discuss in detail several of the points from the EA and EQ sections, which are pertinent to the mechanical system design. The building design anticipated scoring 8 points from the Sustainable Sites category, 2 points for Water Efficiency, 4 points from the Materials and Resources category, and 2 points for Innovation in Design. Among those points includes selecting an appropriate site, using high reflectivity non-roof materials, diverting 50% of construction and demolition waste, using 10% recycled materials, and having LEED accredited designers on the project team.

SECTION 3 – Overall System Evaluation

The mechanical system designed for the STEM Center is effective and appropriate for the needs outlined prior to design. The first cost for the mechanical system was able to be decreased with the partial use of existing plants and equipment alongside the addition of larger, consolidated equipment. The total estimated mechanical first cost was $7,179,242.70, or roughly $68.37/SF. There is potential to decrease the initial cost should the equipment become less centralized and ductwork distance decreased.

Operating cost analysis was conducted with the help of TRACE and a yearly cost of $219,459 was calculated. This value cannot be compared with current system reports for the building, as the STEM Center is currently only 11 months beyond the grand opening. The use of heat recovery, variable frequency drives, and economizers aid the operating cost to a degree, and further investigation will be taken to determine where the yearly totals can be decreased even more.

Altogether, the centralization of the equipment to the roof and the adjacent building help to alleviate the loss of usable floor space, and also contributes to the easier maintainability of the building as a whole. Obviously a great deal of mechanical shaft space is needed to ensure proper ventilation and exhaustion of air for various science-related rooms, and this usage is largely unavoidable.

As is summarized in Section 2.4 with discussion of the LEED credits awarded, great steps have been taken by the design team to ensure high quality of air and environmental control. The pursuit of LEED Silver certification accurately summarizes the sustainability and excellence of the building systems as a whole.

SECTION 4 – Proposed Alternative Systems

As mentioned in Section 1.1, the current mechanical design for the STEM Center is very effective and sufficient to fulfill the design requirements and considerations, and the LEED Silver certification is proof of a fully sustainable design. However, there appear to be areas in which energy consumption is higher and efficiency is lower than it could be, particularly in regards to the atrium and lobby space on the ground floor. A handful of alternative solutions will be analyzed for mechanical system improvement, and those are Radiant Heated Flooring, Radiant Cooled Flooring, and Natural Ventilation.

4.1 – Radiant Heated Flooring

At present, the air handling for the ground floor is a cause for concern, particularly in the grand lobby space where ceilings heights are relatively high and even exceed 40 feet in some places. A good alternative to the existing space conditioning arrangement may be to utilize radiant heated flooring. Radiant flooring is particularly useful in instances such as high ceilings, space with increased foot traffic, and site locations with potential for snow and ice entering with occupants. In the event of high ceilings, it is important to focus on conditioning the occupants instead of the space itself that has unoccupied volume. With the current layout of the lobby and atrium, there is a significant amount of foot traffic at nearly all levels, as students, faculty, and other occupants utilize the building. Additionally, being in the greater Philadelphia area and having four vestibules on the ground floor in the lobby general area, there is potential for snow and ice to be tracked in. In this case, radiant heating is desirable to keep the floor surfaces at an ideal temperature to ensure both safety and comfort.

Figure 12: Section View of Lobby and Atrium

(Photo provided by Burt Hill)

This alternative will require significant assessment to see if it is feasible, as there is no basement in the STEM Center and this analysis will be from the ground up. For coordination with the building's existing mechanical system and equipment as a whole, it is noted that radiant heating is provided using hot water carried in flexible cross-linked polyethylene (PEX) tubing. The source for hot water in radiant flooring is typically a gas-fired boiler, just like the existing boilers the STEM Center already employs. Therefore a significant change in equipment should not be necessary, although resizing will potentially be needed to ensure accurate and efficient heating supply for such an alternative system.

4.2 – Radiant Cooled Flooring

Just as radiant floor heating can improve the building loads, so radiant cooling can also provide a significant improvement to the efficiency of the mechanical system design for the STEM Center. Using the same tubing design layout and receiving chilled water from the existing chiller plant, this change will likely only result in hopefully lower up front cost from ductwork, and may also decrease the size of air handlers.

Figure 13: View of Lobby and Glazing

A forefront architectural feature of the STEM Center is the glass curtain wall façade on the south side of the building. There are two sections to this glazing: the portion at the ground floor level and the portion above the green roof at the 2nd and 3rd levels, which is shown in Figure 13. These factors may actually help the use of radiant cooling as the cooling effect will be increased for the floor areas not in the shade.

4.3 – Natural Ventilation

The existing design has 100% mechanical ventilation and does not use any natural ventilation. With the need for adequate air exhaustion for lab spaces, the two rooftop air handlers used for ventilation are substantial and could benefit from the assistance that comes from natural ventilation. Using natural ventilation alongside the current mechanical design could potentially decrease operating and energy usage.

Exceptional ventilation is crucial in educational facilities to keep air fresh and lower CO_2 levels that can have an adverse effect on learning and studying. For educational buildings, natural or hybrid ventilation systems are very common, and full analysis of an alternate ventilation system will be conducted as a potential solution.

The site conditions for the STEM Center provide the prevailing wind required for natural ventilation and computational fluid dynamics (CFD) will be the basis of this analysis to see if this system is a potential source of improvement. This study will not be all inclusive and will require additional future sophisticated design to couple a natural or hybrid ventilation system with the existing design; however, by analyzing the airflow using CFD simulation and comparing with ASHRAE Standards for thermal comfort, it will be determined whether or not it is a feasible source of mechanical systems improvement.

4.4 – Tools and Methods for Analysis

STANDARDS

Throughout the processes described above, attention will be paid to ASHRAE Standards and other necessary requirements to ensure a successful and appropriate redesign. This will be specifically applicable in the implementation of natural ventilation, and the assurance of air quality.

LOAD SIMULATION and ENERGY MODELING

As per the analysis completed for the existing mechanical design, load simulations will be necessary to assess the alternative systems, as well as calculations of operating costs and emissions due to energy usage. Modeling will be conducted using appropriate software, such as Trane TRACE™ 700, which has been used in analysis to this point.

RADIANT DESIGN

To aid the design of a proper radiant flooring system, the program RadiantWorks Pro will be utilized. This program will be used in conjunction with TRACE and the associated building finish and construction data provided by Burt Hill.

CALCULATION METHODS

Altogether, to accomplish necessary calculations for redesign, programs such as Engineering Equation Solver (EES) and Microsoft Excel will be utilized. These programs, and other similar ones, are appropriate for engineering related calculations.

COMPUTATIONAL FLUID DYNAMICS MODELING

The CFD simulations will need to be conducted with a reliable program such as PHOENICS VR 2009, which is the one utilized for educational purposes in the AE559 CFD course. Software such as PHOENICS will allow for advanced analysis of airflow patterns, velocities, temperatures, and more.

DAYLIGHT STUDY SOFTWARE

It was determined throughout the course of the radiant floor design that direct sunlight on a section floor of floor can alter the capabilities of the slab to meet the loads. In order to analyze this fully, a daylight study will be necessary using a program such as AGI32 by Lighting Analysts.

SECTION 5 – Radiant Floor Heating and Cooling

5.1 – Initial Capacity Analysis

The need for a Radiant floor design arose from the high-volume spaces located on the ground floor, in the east half of the building. These spaces, which have high ceilings, receive high occupancy and traffic, and they include lobby, lounge, and classroom settings. It was for these spaces alone that the radiant floor design analysis was conducted, with the hopes to eliminate the need for increased amounts of conditioned air for the high volumes of unoccupied overhead space.

Much inspiration for the radiant analysis was drawn from the work of Bjarne W. Olesen, Ph.D., the president of the European Radiant Floor Association, who has written several key ASHRAE papers on the subject of radiant floor heating and cooling, as well as the research and analysis conducted by Dustin Eplee, founder of Aeris Technologies and Energy Wall. For a radiant floor design, the sensible capacity can be determined with the help of heat exchange coefficients derived by Olesen for both radiant heating and cooling of floors, assuming maximum and minimum floor temperatures of 84°F and 67°F, respectively. These limits are selected based on the ASHRAE Standard 55 section 5.2.4.4 requirement for floor surface temperature according to a maximum of 10% dissatisfaction rating that is common for assurance of thermal comfort. The total heat exchange coefficient for radiant floor heating is 1.94 (BTU/hr*ft²*°F), while the coefficient is lower for radiant floor cooling at 1.24. The capacity for radiant cooling is less due to several factors, including the lack of natural upward convection that is obtained from a heated floor slab. These coefficients are based on a radiant floor slab construction using 1-½" gypsum concrete ("gypcrete") and ½" tile floor finish. This construction adds only 2 inches to the original 6-inch slab-on-grade design, as shown in the detail in Figure 14. The implications of this floor construction are discussed more in Section 9.

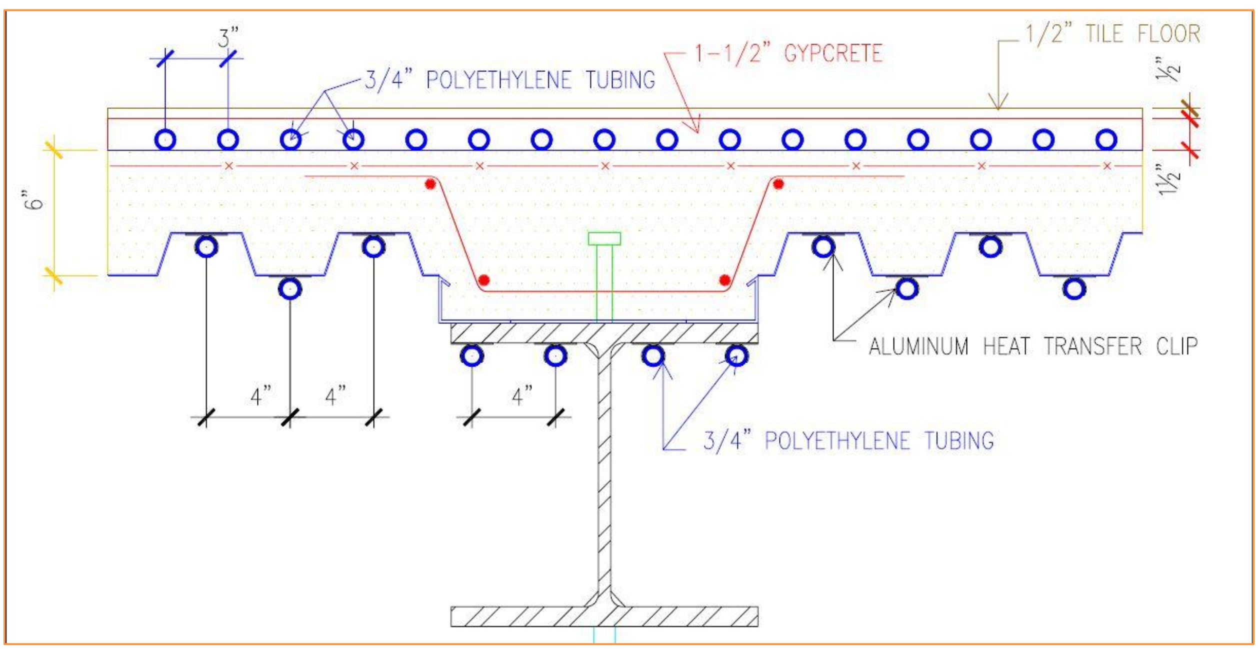

Figure 14: Cross Section of Radiant Floor Slab

Initially studied for adequate radiant capacity were the spaces in the east portion of the ground floor, and they are highlighted in Figure 15. The total number of spaces in this area is fifteen, and they include four sections of lobby area, three sections of lounge spaces, two closets, the welcome desk and office, and the interior egg-shaped portion of the computer lab, meeting room, auditorium, and elevator machine room.

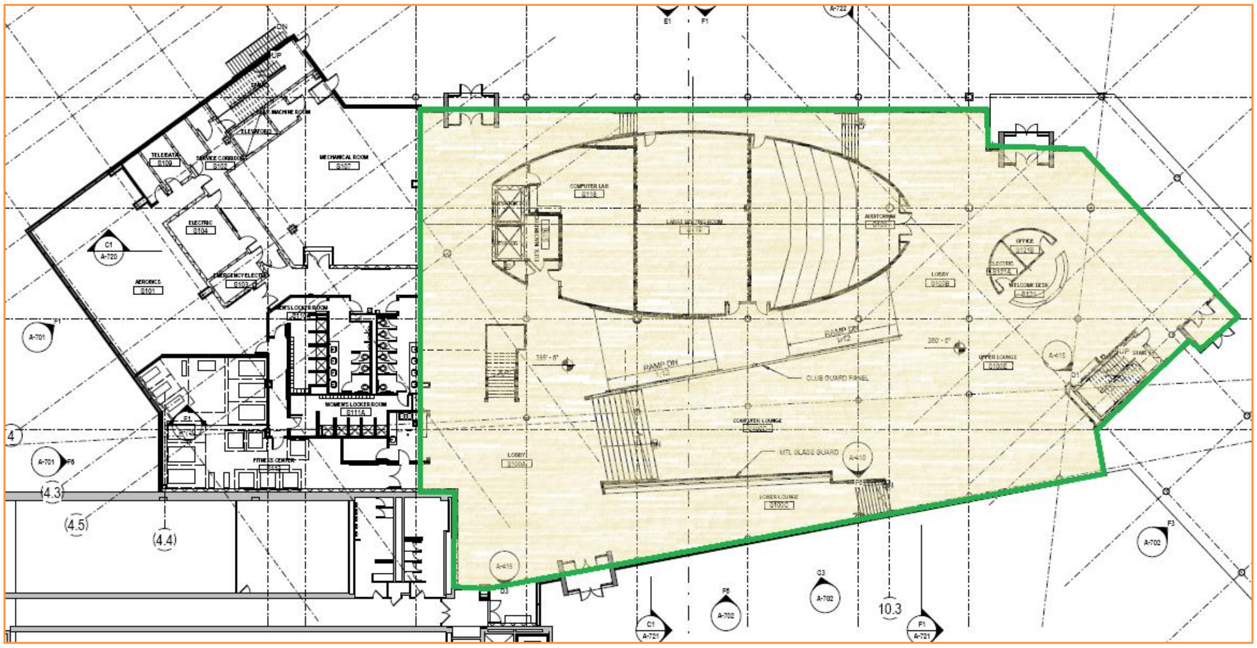

Figure 15: Ground Floor Spaces for Radiant System Design

To begin analysis, the results for the load simulation conducted by TRACE would be used to assess the capabilities of a radiant floor design to handle the loads associated by these spaces. The initial concern was to calculate the capacity of the radiant floor to handle both the heating and cooling peak sensible loads.

Heating and cooling capacity was calculated using the heat exchange coefficients in the following manner:

$$\textit{Capacity} = \textit{Area}\ (ft^2) * \Delta T\ (°F) * \textit{Coefficient}\ (BTU/hr*ft^2*°F)$$

Heating Example (S100C Lower Lounge):

$$1{,}220\ ft^2 * 10°F * 1.94\ BTU/hr*ft^2*°F = \mathbf{18{,}481\ BTU/hr}$$

In this manner, the radiant capacity calculations were conducted for the fifteen spaces mentioned above, and the initial results for these calculations are shown below in Tables 14 and 15. For the heating, only the S116 Elevator Machine Room, S118A Closet, S121A Electrical Closet, and S121B Office were calculated as not having sufficient capabilities to handle the generated sensible load. This further shows the greater capacity of radiant heating to overcome the heating load. These four spaces were deemed insufficient and unnecessary for radiant floor design.

RADIANT FLOOR HEATING SENSIBLE CAPACITY					
Room	Area (SF)	ΔT (°F)	Heating Coefficient	RADIANT CAPACITY (BTUh)	Space Load (BTUh)
C-143 Lobby	5,011	10	1.94	97,213.4	49,628
C-243 Lobby	1,132	10	1.94	21,960.8	11,548
C-343 Lobby	1,154	10	1.94	22,387.6	13,195
C-443 Lobby	1,146	10	1.94	22,232.4	14,205
S100C Lower Lounge	1,220	10	1.94	23,668.0	18,481
S100D Comp Lounge	1,794	10	1.94	34,803.6	14,767
S100E Upper Lounge	3,063	10	1.94	59,422.2	16,479
S116 Elev. Room	109	10	1.94	2,114.6	2,524
S118 Computer Lab	970	10	1.94	18,818.0	14,485
S118A Closet	29	10	1.94	562.6	1,928
S119 Lg. Mtg. Room	1,452	10	1.94	28,168.8	12,757
S120 Auditorium	1,468	10	1.94	28,479.2	13,656
S121 Welcome Desk	183	10	1.94	3,550.2	1,504
S121A Electric	73	10	1.94	1,416.2	2,774
S121B Office	80	10	1.94	1,552.0	2,026

Table 14: Radiant Floor Heating Sensible Capacity

RADIANT FLOOR COOLING SENSIBLE CAPACITY					
Room	Area (SF)	ΔT (°F)	Cooling Coefficient	RADIANT CAPACITY (BTUh)	Space Load (BTUh)
C-143 Lobby	5,011	10	1.23	61,635.3	57,566
C-243 Lobby	1,132	10	1.23	13,923.6	10,924
C-343 Lobby	1,154	10	1.23	14,194.2	12,496
C-443 Lobby	1,146	10	1.23	14,095.8	13,573
S100C Lower Lounge	1,220	10	1.23	15,006.0	28,139
S100D Comp Lounge	1,794	10	1.23	22,066.2	34,993
S100E Upper Lounge	3,063	10	1.23	37,674.9	34,898
S116 Elev. Room	109	10	1.23	1,340.7	2,522
S118 Computer Lab	970	10	1.23	11,931.0	23,851
S118A Closet	29	10	1.23	356.7	1,909
S119 Lg. Mtg. Room	1,452	10	1.23	17,859.6	50,275
S120 Auditorium	1,468	10	1.23	18,056.4	30,501
S121 Welcome Desk	183	10	1.23	2,250.9	2,095
S121A Electric	73	10	1.23	897.9	2,772
S121B Office	80	10	1.23	984.0	2,531

Table 15: Radiant Floor Heating Sensible Capacity

With the cooling capacity, however, multiple lounge spaces and the interior egg spaces were also found to be incapable of meeting the load. While the lounge spaces were close to sufficient, the spaces that make up the interior egg (S118 Computer Lab, S119 Large Meeting Room, and S120 Auditorium) were dismissed from the potential use of radiant floor system.

These omissions were also reasonable due the architectural nature of the spaces. Particularly in the auditorium, which has a stepped down floor landscape, an under-floor hydronic system would have been a much greater difficulty than the basic floors that make up the majority of the other spaces.

5.2 – Increased Capacity Capabilities

Based on this development, there was a need to explore the ability to get an increased capacity, and this could be found through floor exposure to direct sunlight (Olesen). Flooring that is in direct sunlight has a higher cooling capacity due to an increased ΔT that results from the hot rays of the sun hitting the surface of the floor. Just how much this capacity could increase is the issue. The answer to that lies in the research that shows that the standard capacity of 9 BTU/h*ft² increases to 26 BTU/hr*ft², which is nearly three times greater (Eplee).

In order to study the effects of a sunlight factor, it was necessary to conduct a daylight study for the spaces that were potentially in direct sunlight. Further validating the elimination of the interior egg spaces from radiant analysis was the fact that they received no sunlight and had no chance for greater sun-related capacity. The spaces that were in question for a daylight study were the Lower Lounge (S100C) and Computer Lounge (S100D). Both spaces are on the south side of the building, where glass curtain wall forms the façade at all three main levels. The curtain wall is split up in two sections, as shown in Figure 16. Because of the horizontal shading devices used in the upper glass curtain wall, the only ability for direct sunlight to aid the ground floor capacity existed through the 16'-high ground level glass façade. Because of this, a potential increase in cooling capacity existed for both spaces.

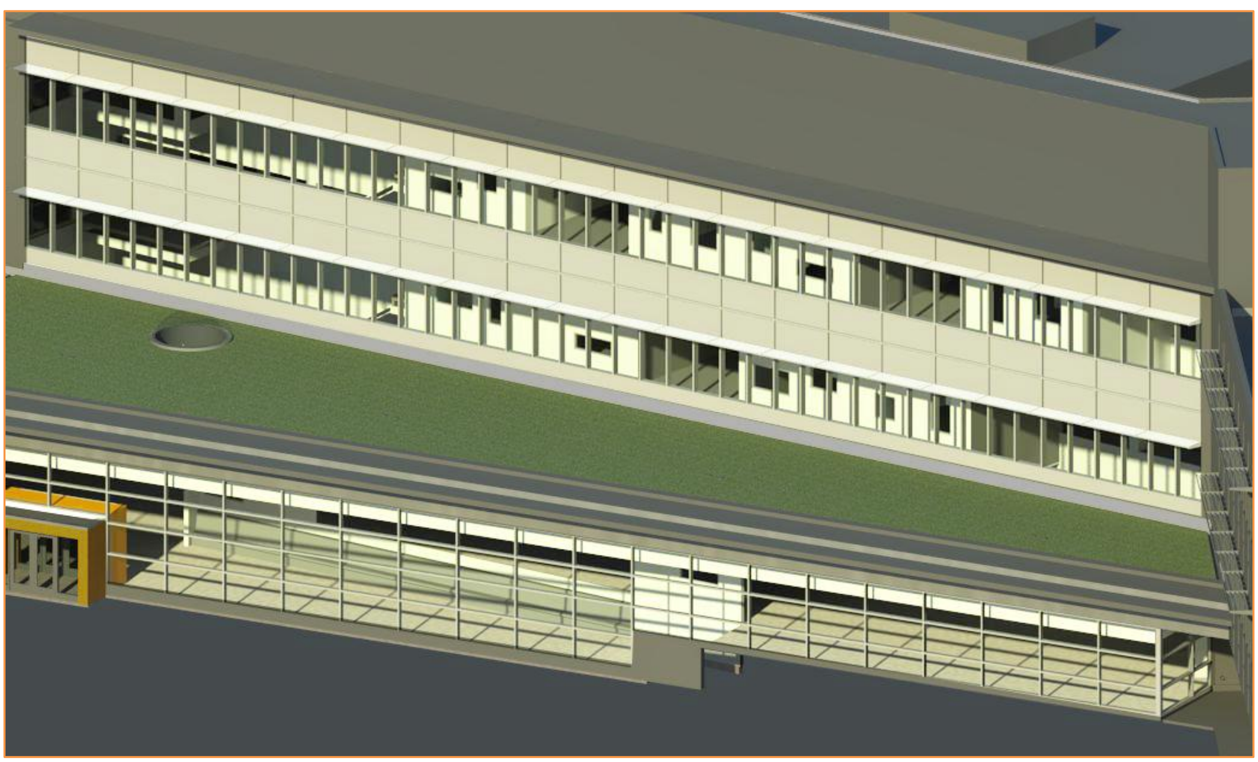

Figure 16: South Side Glass Curtain Wall Façades

To perform the daylight study for these two spaces, the lighting software AGi32 (version 2.1) by Lighting Analysts was utilized. In order to accurately model the solar conditions, this program used solar data for the greater Philadelphia area, the orientation of the STEM Center relative to true north, and a wireframe floor plan imported from the Autodesk Revit Architecture (2011) file by way of Autodesk AutoCAD (2011). The time frame specified for the design was based on the peak days determined by the TRACE load simulation.

5.3 – Daylight Study Results

The highest cooling loads existed for the first few weeks in July, as was anticipated, and so several days in July were selected for daylight simulation. The worst case scenario of those July simulations was kept as a conservative approximation and Figures 17 through 19 shows the results of the AGI32 study, with additional image results in Appendix B. In each image, the red region represents 1000+ fc, which is the standard estimation for full daylight.

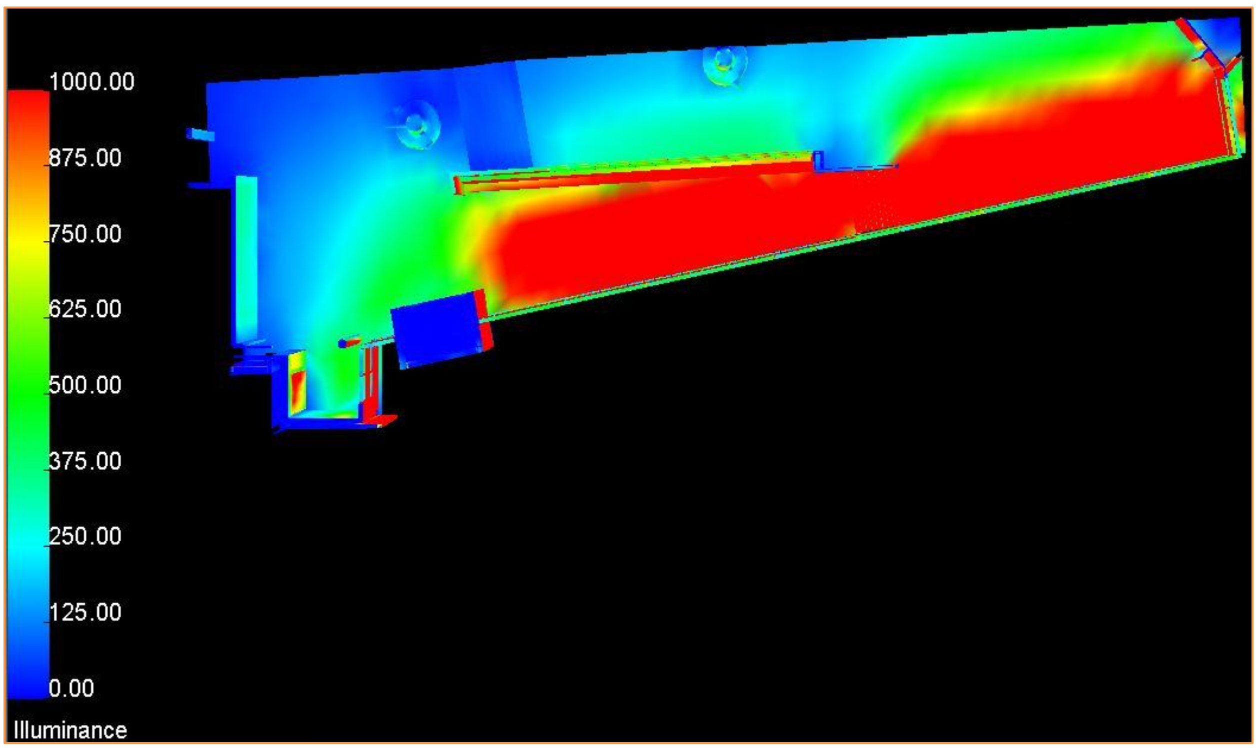

Figure 17: Daylight Study Results for Lower Lounge

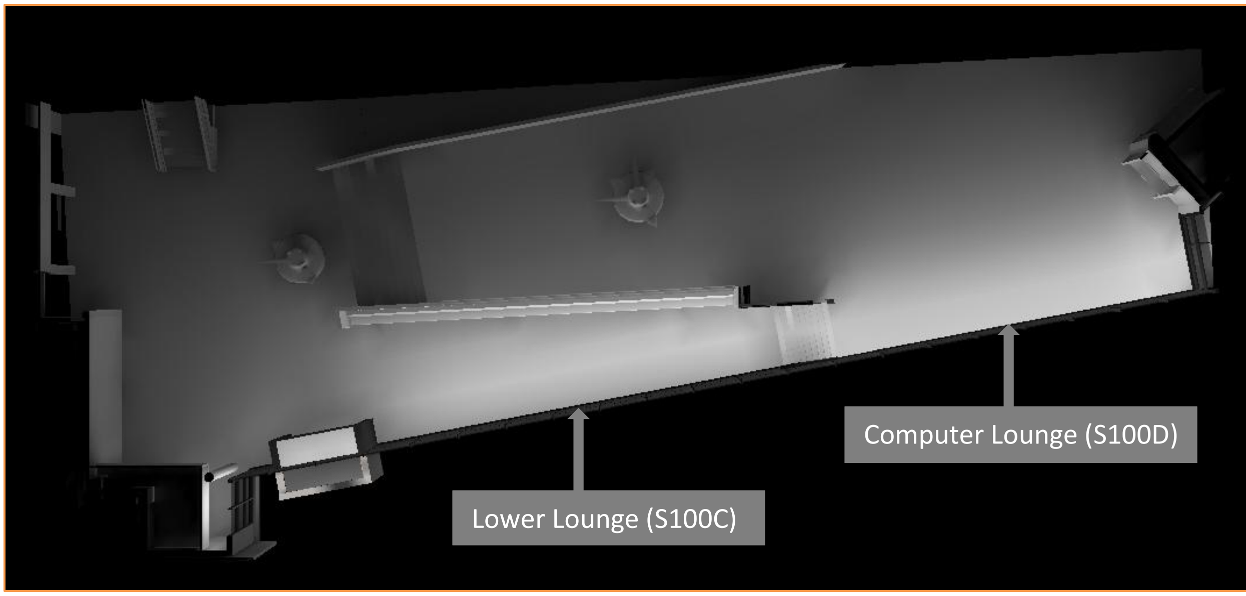

Figure 18: Daylight Study Results for Lower Lounge and Computer Lounge

To calculate an accurate approximation for a sunlight factor for the radiant floor, a grid was displayed over the floor to calculate the percentage of the floor area with over 1000+ fc (in direct sunlight). This percentage was then multiplied by the aforementioned ratio of standard capacity in sunlight to not in sunlight.

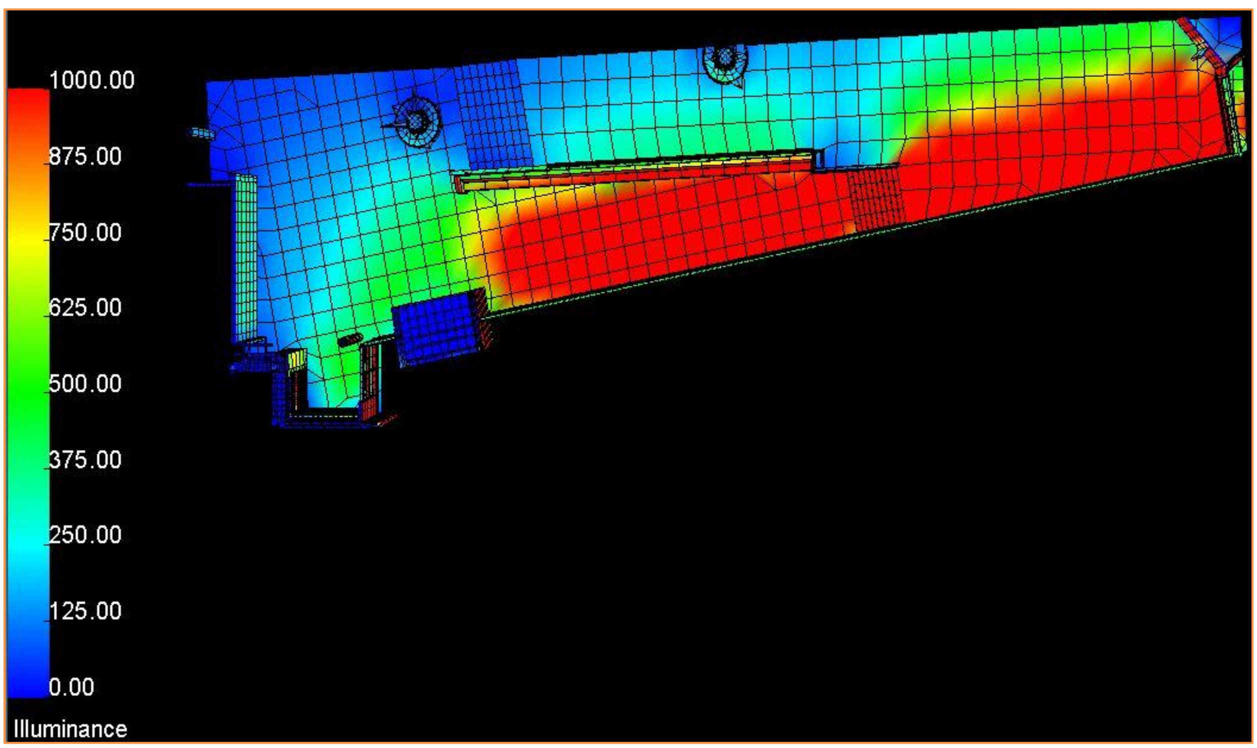

Figure 19: Daylight Study Results with Grid for Calculation

5.4 – Sensible Capacity Results

This daylight study provided sunlight factors for both the Lower Lounge and Computer Lounge equal to 2.51 and 1.60, respectively, and gave the cooling capacity for the radiant flooring in these spaces a significant boost. As shown in the complete calculation breakdown of radiant capacities in Tables 16 and 17, with the greater sunlight factor all lounge spaces were calculated to be capable of meeting the sensible load for cooling.

RADIANT FLOOR HEATING SENSIBLE CAPACITY						
Room	Area (SF)	ΔT (°F)	Heating Coefficient	Sun Factor	RADIANT CAPACITY (BTUh)	Space Load (BTUh)
C-143 Lobby	5,011	10	1.94	1.00	97,213.40	49,628
C-243 Lobby	1,132	10	1.94	1.00	21,960.80	11,548
C-343 Lobby	1,154	10	1.94	1.00	22,387.60	13,195
C-443 Lobby	1,146	10	1.94	1.00	22,232.40	14,205
S100C Lower Lounge	1,220	10	1.94	1.00	23,668.00	18,481
S100D Comp Lounge	1,794	10	1.94	1.00	34,803.60	14,767
S100E Upper Lounge	3,063	10	1.94	1.00	59,422.20	16,479
S121 Welcome Desk	183	10	1.94	1.00	3,550.20	1,504

Table 16: Updated Radiant Floor Heating Sensible Capacity

RADIANT FLOOR COOLING SENSIBLE CAPACITY						
Room	Area (SF)	ΔT (°F)	Cooling Coefficient	Sun Factor	RADIANT CAPACITY (BTUh)	Space Load (BTUh)
C-143 Lobby	5,011	10	1.23	1.00	61,635.30	57,566
C-243 Lobby	1,132	10	1.23	1.00	13,923.60	10,924
C-343 Lobby	1,154	10	1.23	1.00	14,194.20	12,496
C-443 Lobby	1,146	10	1.23	1.00	14,095.80	13,573
S100C Lower Lounge	1,220	10	1.23	***2.51***	37,605.20	28,139
S100D Comp Lounge	1,794	10	1.23	***1.60***	35,305.92	34,993
S100E Upper Lounge	3,063	10	1.23	1.00	37,674.90	34,898
S121 Welcome Desk	183	10	1.23	1.00	2,250.90	2,095

Table 17: Updated Radiant Floor Cooling Sensible Capacity

The results of this analysis are significant in that it was determined that 100% of the sensible load could be sufficiently handled by a radiant cooled and heated floor. This result opened up the door to a significant downsize in the existing airside system design. If it could be shown that the necessary amount of fresh outside air could adequately support the latent load for the eight radiant spaces (a total 14,700 square feet of floor area), then a substantial amount of capital cost would be saved through the reduction of ductwork and variable air volume boxes from the original design.

SECTION 6 – DOAS System

6.1 – Analysis Objective

The results of the radiant analysis based on heat exchange coefficients proved that the sensible load could be handled completely by a radiant floor system for both heating and cooling. What that does not account for, however, is the need for adequate ventilation and the means to handle the latent load for the spaces.

To accommodate the ASHRAE Standard 62.1 requirements for outdoor air, the amount of outdoor air specified in the original design was kept as an appropriate base level. These values had been checked for compliance in Technical Report 1 and were concluded to be sufficient to meet fresh air requirements.

For each of the rooms selected for radiant design, there will be a significant decrease in required air, as was expected. The amount of fresh air required for the radiant spaces was compared with the amount of air being supplied to the same spaces in the original design, and this comparison is shown in Table 18. For just these eight spaces, the decrease in total air flow is nearly 20,000 cfm. This means a decrease in duct size all the way up to the rooftop air handling units that distributed all of the supply air for the STEM Center.

Room	Area (SF)	OA Vent. Needed (CFM)	Old SA Duct Size (CFM)
C-143 Lobby	5,011	4,385	12,529
C-243 Lobby	1,132	991	2,831
C-343 Lobby	1,154	1,010	2,885
C-443 Lobby	1,146	1,003	2,865
S100C Lower Lounge	1,220	1,337	3,820
S100D Comp Lounge	1,794	525	1,500
S100E Upper Lounge	3,063	1,446	4,130
S121 Welcome Desk	183	42	120

Table 18: Comparison of Original and Ventilation Required Airflow

The second big need for an additional system for the radiant-designed spaces was the addressing of the latent load. The humidity level for lounge and lobby spaces such as these can be high due to the high number of inhabitants that produce elevated moisture levels in the air. Because of the need to account for the latent load, the DOAS system needs to include air handling components to remove moisture from the outside air, so that the air that enters the space is dry enough.

6.2 – Design and Equipment for DOAS System

The ASHRAE Handbook of Fundamentals (2009) cooling design conditions for Philadelphia, PA, list the dry bulb temperature at 93.1°F and the wet bulb temperature at 75.7°F for the cooling months (0.4% values). This equates to 45% relative humidity and a humidity ratio of 0.015 lb/lb. In order to meet these conditions, the DOAS system was designed to consist of an enthalpy wheel as well as a heat exchanger and cooling coil system.

The first component of the dedicated outdoor air handling unit is wheel specified to precool and slightly dehumidify the air as it crosses a branch of the exhaust path. This type of heat wheel has total effectiveness of 50%-80%, providing the capacity to precool the 93.1°F DB outside air to roughly 85°F (and 70°F WB). This is a small step in the dehumidification process that continues to the crossflow heat exchanger.

To adequately handle the remainder of the dehumidification there is a high efficient plate heat exchanger design patented by MSP Technology. Figure 20 shows the heat exchange process as displayed on the MSP Technology website. Warm, humid outside air enters the heat exchanger (after going through the enthalpy wheel) at 85°F DB and 70°F WB and undergoes pre-cooling and initial dehumidification by way of the cooler air leaving the heat exchanger.

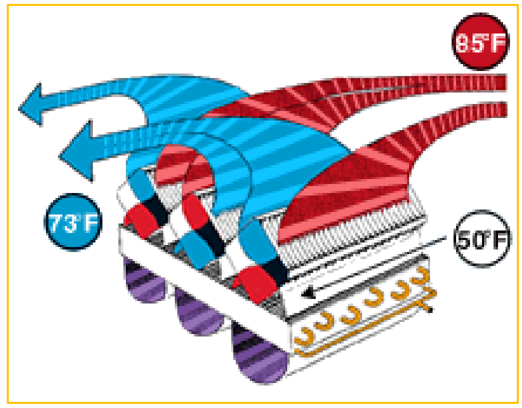

Figure 20: High Efficiency Heat Exchanger
(MSP Technology)

The second process of this high efficiency heat exchanger involves pre-cooled air passing over standard cooling coils twice to undergo the dehumidification and cooling as the dry bulb temperature lowers to roughly 50°F. In the third and final process, this cool, dry air is then drawn back through the heat exchanger where crossing with the initial air stream leads to a rise to a neutral temperature for the ventilation requirement.

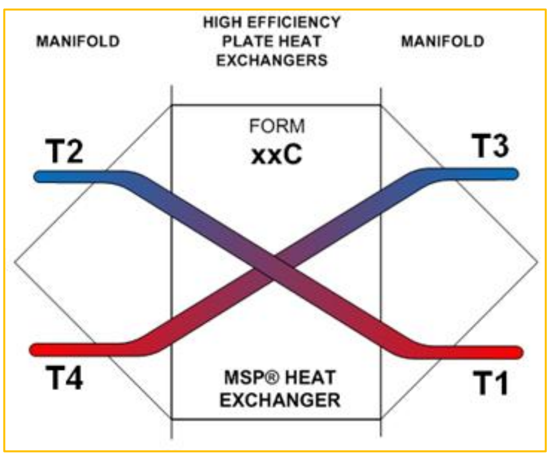

Figure 21: Heat Exchanger Temperatures
(MSP Technology)

This design allows the dedicated outdoor air needed to meet the ASHRAE Standard 62.1 fresh air requirements to be dehumidified sufficiently to account for the latent load in the radiant spaces. This fresh air will have 78.8°F dry bulb temperature, a 54°F wet bulb temperature, a 16.4% relative humidity, and 0.00327 lb/lb humidity ratio, and is shown as “T4” in the Figure 21 illustration of the heat exchange process. All four states and all three processes are outlined in detail in the product sheets found in Appendix C.

According the MSP Technology specifications, each Air-to-Air heat exchange element unit will perform in the following manner:

MSP® HEAT EXCHANGE ELEMENT	
EFFICIENCY	81.4%
HX UNIT SIZE (MM)	190
MASS FLOW (SCFM)	125
PRESSURE DROP (IN WG)*	0.52
PLATE SPACING (MM)	2.0
HX SURFACE -ONE SIDE (SQ FT)	106
EFFICIENCY**	81.4%

* Pressure drop based upon standard air conditions ** Efficiency based on non-condensing performance

Figure 22: Heat Exchanger Specification *(MSP Technology)*

With this technology in place in concordance with a new small air handling unit for adequate outside air ventilation of the radiant spaces, the latent load and fresh air requirements will be obtained to complete the proposed design for the radiant heating and cooling system. Altogether, the equipment need for the new design will include the pumps and piping required for the hydronic system, and the rooftop air handler complete with enthalpy wheel, heat exchanger, and cooling coil.

The air handling unit will be roof-mounted in the same manner as the aforementioned AHU-4 and AHU-5. There exists adequate space adjacent to the existing equipment to install this new unit for the ground floor DOAS system. This is illustrated in Figure 23 showing the mechanical layout of rooftop equipment in the central region.

Figure 23: Roof Plan Showing Air Handling Unit Layout

6.3 – Airside Equipment Reduction

It is understood that the addition of the aforementioned equipment raises the upfront cost for the mechanical system design. However, the overall airside equipment is greatly reduced due to the radiant design that requires only ventilation air. With the radiant floor and dedicated outdoor air system, there is no longer need for much of the ground floor ductwork for the radiant spaces, and the main branches that extend up the core of the building will also decrease in size.

With the use of a radiant system, and the addition of a separate air handling unit for the subsequent DOAS system, there will be 30,680 cfm less supply air needing to be produced by the main air handling units. AHU-4 and AHU-5 (outlined in Figure 23 as “Existing AHUs”) are each sized at 89,500 cfm. The supply air produced by the two units comes together in a main branch sized at 144x48, capable of handling 132,330 cfm. This main branch becomes progressively smaller as it proceeds down the main mechanical shaft in the building until it becomes only 88x70 duct supplying 44,880 cfm for the existing ground floor design.

In the redesign, the main branch is downsized and additional smaller duct will be added to supply the dedicated outdoor air for the radiant spaces. As a whole, the main branch will handle 30,680 cfm less supply air, while the additional duct will be sized for 10,800 cfm.

For the resizing of the existing ductwork and the sizing of the additional ductwork, McQuay Air Conditioning's DesignTools DuctSizer Version 6.4 software was used to provide an appropriate duct size based on flow rate and head loss. A head loss of 0.08" WC/100' was used for the calculation of the supply air duct, and the sizes of the new duct sizes and additional duct sizes are shown in Table 19.

Ductwork Section (floor to floor)	CFM	Old Duct Size	**If 30,680 fewer CFM...**	**NEW DUCT SIZE**	**Additional Duct CFM**	**Additional Duct Size**
AHU-4	132,330	192x48	**101,650**	**132x48**	**10,800**	**48x24**
AHU-5	132,330	192x48	**101,650**	**132x48**		
AHU to R/4th	132,330	144x48	**101,650**	**132x48**	**10,800**	**48x24**
R/4th to 3rd	117,890	132x88	**87,210**	**88x60**	**10,800**	**48x24**
3rd to 2nd	75,970	88x88	**45,290**	**72x45**	**10,800**	**48x24**
2nd to 1st	44,880	88x70	**14,200**	**48x28**	**10,800**	**48x24**

Table 19: Duct Resizing Calculations

Additionally, the supply VAV boxes will no longer be needed as there will be only a constant volume of air providing the outdoor air at neutral temperature. These boxes are used all throughout the existing design for the ground floor mechanical system, including a total of 15 that serve the radiant spaces in the original design. These boxes are sized at anywhere from 570 to 3850 cfm, and are tallied up in Table 20.

VAV Boxes removed:		
Type	#	CFM Range
VVR-4	1	450-600
VVR-5	1	600-850
VVR-6	1	850-1000
VVR-7	2	1000-1600
VVR-8	4	1600-2400
VVR-9	3	2400-3200
VVR-10	3	3200-4400

Table 20: Terminal Units Removed

The calculations for the savings associated with the equipment changes are outlined in Section 9, along with the complete construction management cost and schedule implications of the radiant floor and DOAS systems.

6.4 – Energy Calculations

Along with the equipment and general initial cost implications of the mechanical redesign, there is a change in anticipated operating costs. As discussed in Section 1.3, the mechanical system for the STEM Center includes a gas-fired boiler and a water-cooled electric chiller, and the energy usage associated with these and other equipment were calculated for the original design using Trane's TRACE™ 700.

For the estimation of the energy usage and utility cost implications of the mechanical system redesign, TRACE was once again used. It is important to note that while TRACE is a very credible energy modeling software, there are restrictions with being able to predict the actual performance of the system with complete accuracy. For this analysis, a new system was created in the software using an arrangement involving only piping and no fans, as well as a simple DOAS system. The air handler and pump specifications were based on the previously discussed analysis and sizing.

The results of the energy simulation were close to as expected, with a slight reduction in overall energy consumption and utility costs. The implications of a radiant floor design are the reduced need for air treatment and increased need for water for pumping through the hydronic system. Shown in Table 21 is the comparison of the energy modeling results of the old and new mechanical systems designs. As expected, the pump energy is increased and the fan energy is decreased.

	Energy Consumption Summary					
	OLD SYSTEM			NEW SYSTEM		
	Elec. (kWh)	Gas (kBTU)	% of Total	Elec. (kWh)	Gas (kBTU)	% of Total
Heating	37,788	664,750	9.1%	29,288	731,125	9.6%
Cooling	351,382	-	13.7%	341,124		13.4%
Auxiliary						
Fans	772,751	-	30.2%	744,123		29.2%
Pumps	894,290	-	34.9%	909,443		35.7%
Lighting	282,857	-	11.1%	282,857		11.1%
Receptacle	26,160	-	1.0%	27,152		1.0%
	2,365,228	664,750		2,333,987	731,125	

Table 21: Energy Consumption Summary

As can be seen, the total kWh consumption decreased slightly, while the gas consumption increased. This is most likely due to the greater reliance on the existing boiler arrangement to handle the hot water through the radiant hydronic system. While the chiller usage is increased for the chilled water needed for the hydronic system as well, the kWh consumption is less due to the decrease in auxiliary energy usages such as fans. The total yearly energy consumption breakdown in kWh and therms is seen in Table 22.

Yearly Energy Consumption		
	OLD SYSTEM	NEW SYSTEM
Electric	2,365,228 kWh	2,333,985 kWh
Gas	6,648 therms	7,311 therms

Table 22: Yearly Energy Consumption

As was previously discussed in Section 1.6, the rates used for the calculation of the total utility cost were provided by Burt Hill in the design documentation as a standard average for natural gas (therms) and electrical consumption (kWh). These rates were applied to the energy results and the total yearly utility costs are shown in Table 23.

Yearly Utility Costs		
	OLD SYSTEM	NEW SYSTEM
Electric	$210,505	$207,725
Gas	$8,954	$9,848
Total	**$219,459**	**$217,573**

Table 23: Yearly Utility Costs

This analysis concludes that the redesign of the mechanical system for the ground floor lobby, lounge, and welcome desk spaces produced a yearly energy savings of close to $2,000. Therefore, this proposed alternative mechanical system design for the portion of the building under consideration is capable of decreasing the operating cost for Delaware County Community College.

SECTION 7 – Natural Ventilation Study

7.1 – Analysis Procedure

For the STEM Center, the original mechanical design for ventilation consists of 100% mechanically ventilated air. Based on this and the knowledge that the prevailing northwest wind hits the south side of the building at a suitable angle, it was decided that a study would be taken to assess the feasibility of using natural ventilation. The spaces chosen for the natural ventilation study are the core of the building, from the ground floor to the roof, excluding the wings. This analysis would potentially lead to a sophisticated design for natural or hybrid ventilation, but the purpose of it was to simply assess the potential of harnessing the prevailing wind to further relieve the current airside mechanical design, and to ensure that the natural airflow in the specified spaces would still achieve adequate thermal comfort as laid out in ASHRAE Standard 55.

In order to analyze the use of natural ventilation, a computational fluid dynamics study was conducted with the purpose of looking at flow patterns, airspeed in the occupied spaces, and temperature stratification. The program used for the CFD analysis was PHOENICS 2009, and while PHOENICS is not the most sophisticated CFD software on the market, it delivers clear, basic results in a graphical format that is easy to understand.

The CFD model was set up based on the approach of using operable windows at the previously discussed ground level glass façade along with a the exhaust fans that eject air at the roof level at different points throughout the building. Operable windows were specified as the size of the existing ones, while the exhaust fans, which are located throughout the top floor, were represented by four main outlets with a calculated velocity based on the known fan area and flow rate (in cfm).

Instead of modeling the entire building, the domain for the CFD study consisted of just the middle core of the STEM Center, where the ground floor lobby and lounge spaces extend up through the glassed-in atrium to the second and third floor main corridors. At the ceiling of the third floor corridor the exhaust outlets (in red) were placed with specified velocity and direction. Figure 24 shows the domain and model elements set up for the simulations (with windows in green).

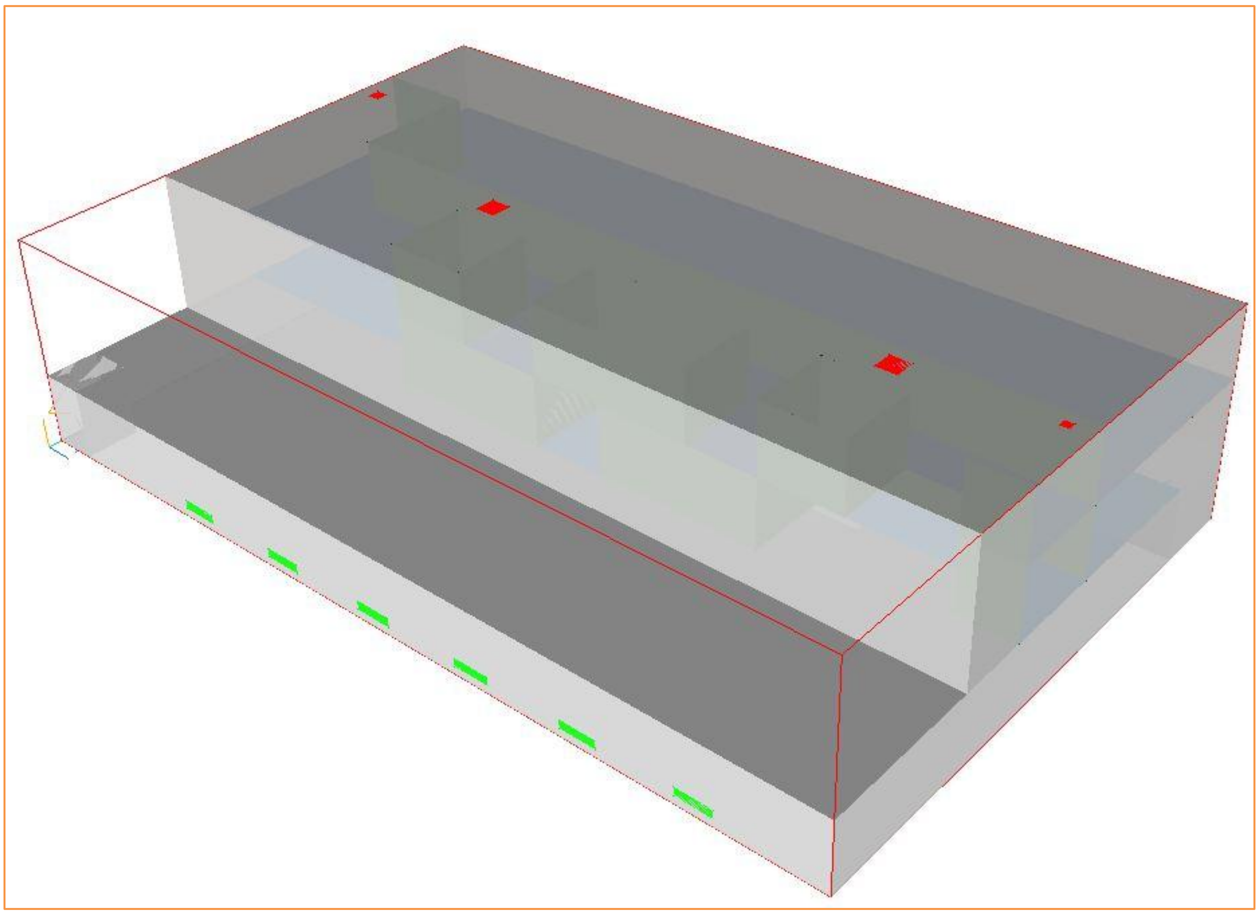

Figure 24: Domain and Diffusers for CFD Simulation

7.2 – CFD Simulation Conditions

The boundary conditions for the study were an air exchange rate of 35.6 m^3/s specified for the exhausts fans, while the conditions at the windows were kept at neutral so as to avoid over-prescription of airflow. In order to model this, the exhaust fans were treated as “inlets” with corresponding velocities in the positive-z direction (out the roof), and the windows were treated as “outlets”. With no velocity applied to the windows, it meant that the pressure difference created by the exhaust units would cause the airflow through the windows to create the means for ventilation. In terms of the overall domain conditions and size and position of the individual airflow components, they can be found in Appendix D.

For CFD modeling, many elements need to be taken into account, from the turbulence model to the numerical differencing schemes, all of which are crucial to the processes and laws of computational fluid dynamics. For this particular study, a standard k-ε model and hybrid (central and upwind) was utilized, which both are typical for the majority of CFD modeling applications.

In regards to convergence of the simulation, the convergence criterion was set at 0.1%, while a goal for mass and temperature residual was less than 3%. For large simulations such as this, it is a difficult process to achieve appropriate convergence of results. The overall mesh of the domain was refined to 113x84x32 for a total of 303,744 cells, and great care was taken to ensure a finer mesh at the edges of the domain and at each inlet and outlet, which can be seen in Figure 25.

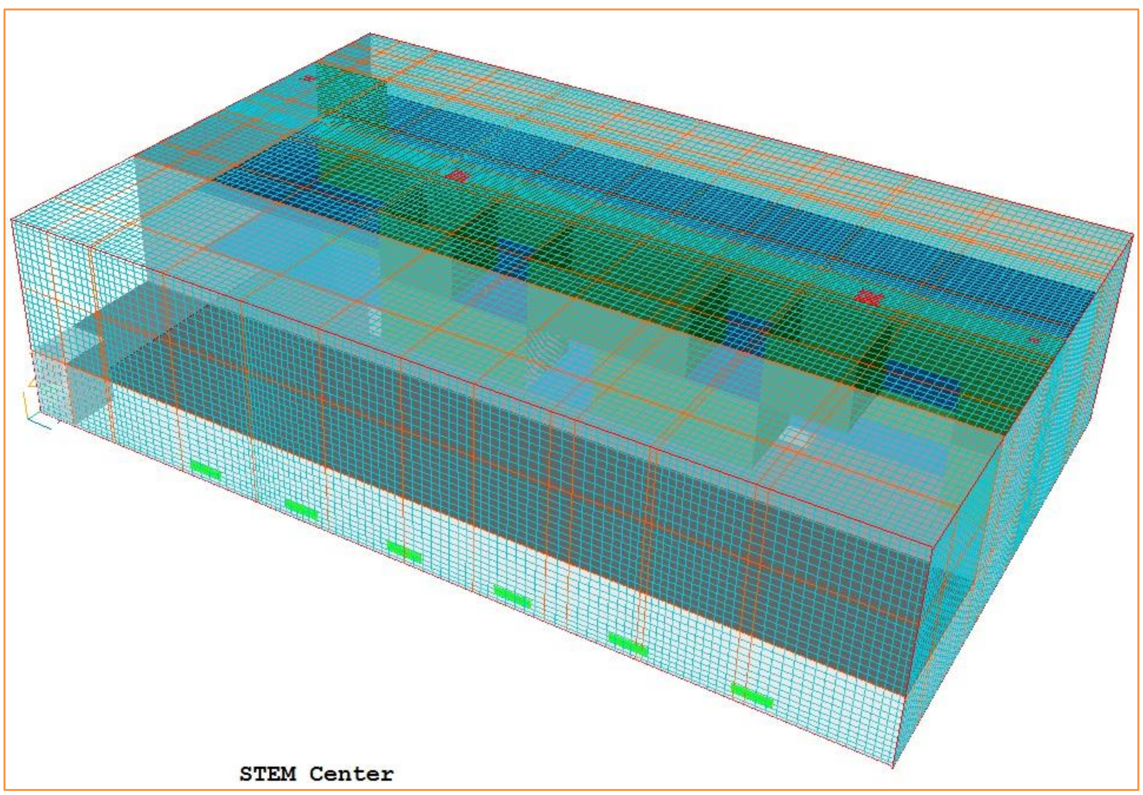

Figure 25: CFD Domain Showing Mesh Refinement

For the inlets and outlets, the ambient temperature was set at 20°C (68°F) while the outside temperature entering through the windows was a cooler 17°C (62.6°F), an arbitrary value for a typical day when open windows would create more pleasant interior conditions. As mentioned above, it is the prescribed velocity for the exhaust boundaries that creates the pressure difference at the windows. This air exchange rate approximately matched the prevailing wind of 4.5062 m/s (10.1 mph) in the northwest direction as specified by weather data for Philadelphia, PA. To simulate the internal load for the spaces of interest, a surface heat flux was applied to the floor at ground level based on the load calculation previously performed by TRACE. This load was calculated to 22 W/m².

7.3 – CFD Results

Upon running numerous simulations over the course of several weeks, good results were obtained based on the aforementioned conditions and approximately 1130 iterations. Shown in Figures 26-35 are images from the output results taken from the PHOENICS software, both for temperature and velocity. Figures 26-30 illustrate the temperature stratification resulting from the cooler ventilation air entering at the ground floor level and eventually being exhausted at the top of the atrium. (Note the temperature scale for reference)

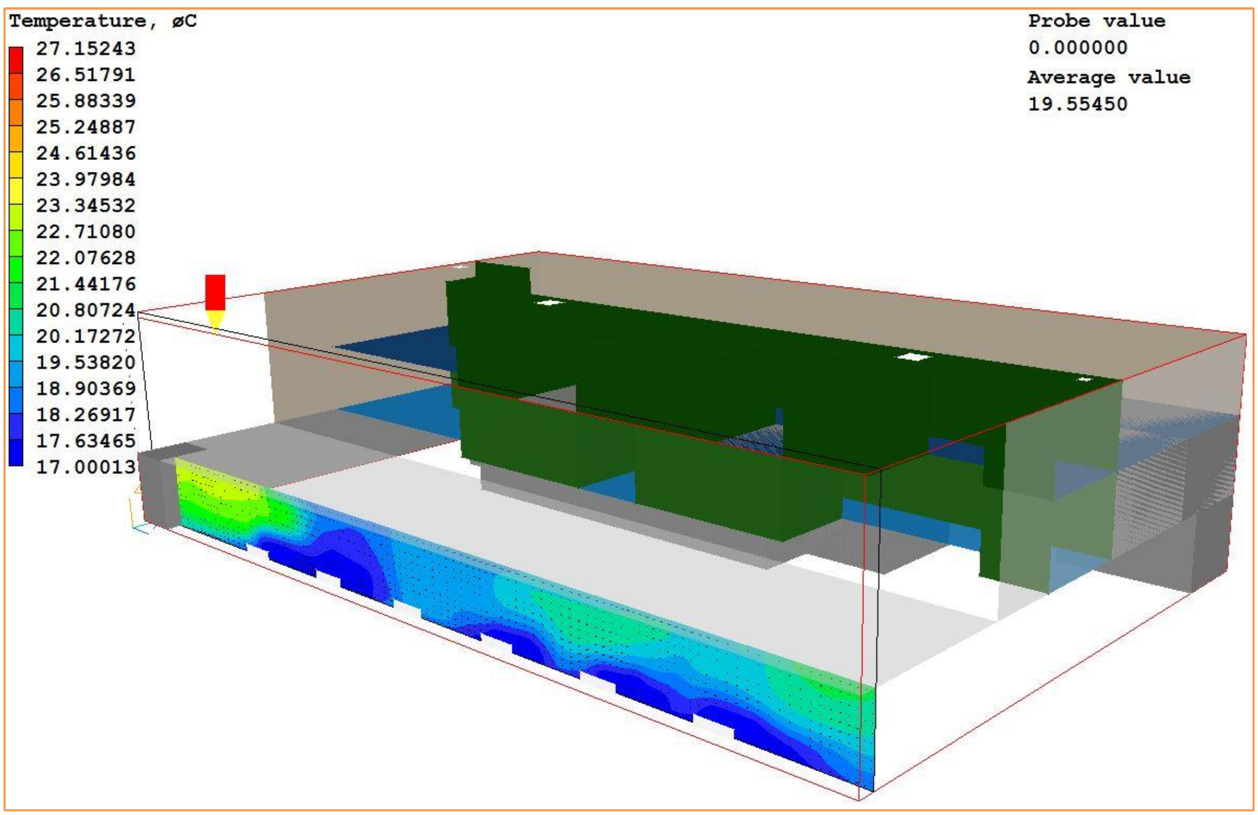

Figure 26: Temperature Profile in Y-Plane

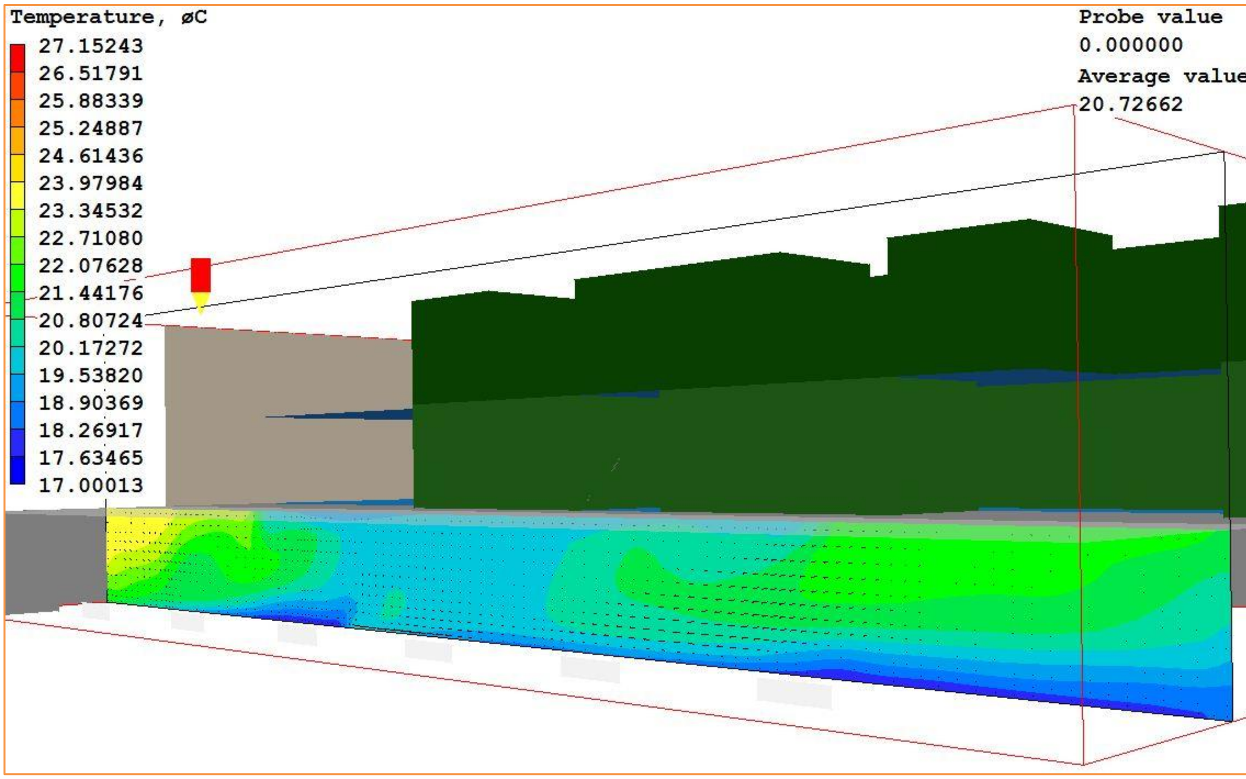

Figure 27: Temperature Profile in Y-Plane

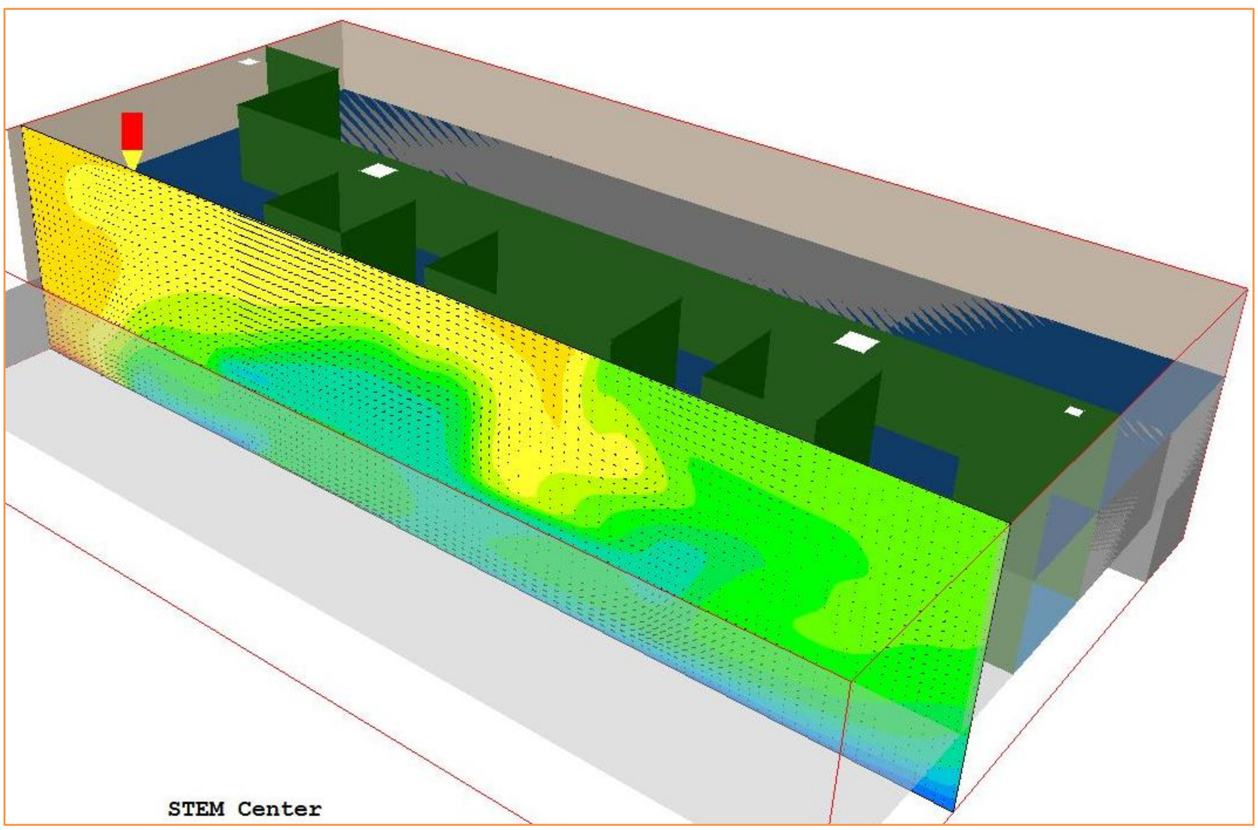

Figure 28: Temperature Profile in Y-Plane

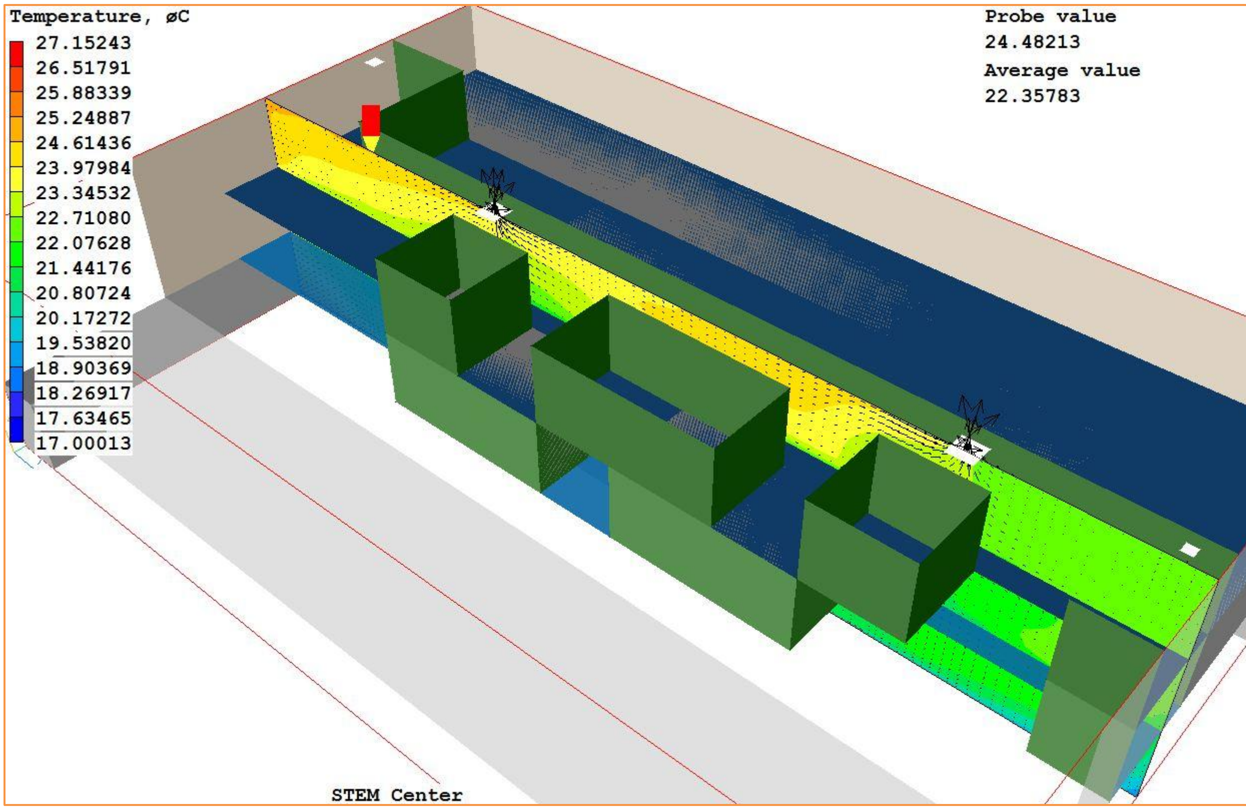

Figure 29: Temperature Profile in Y-Plane

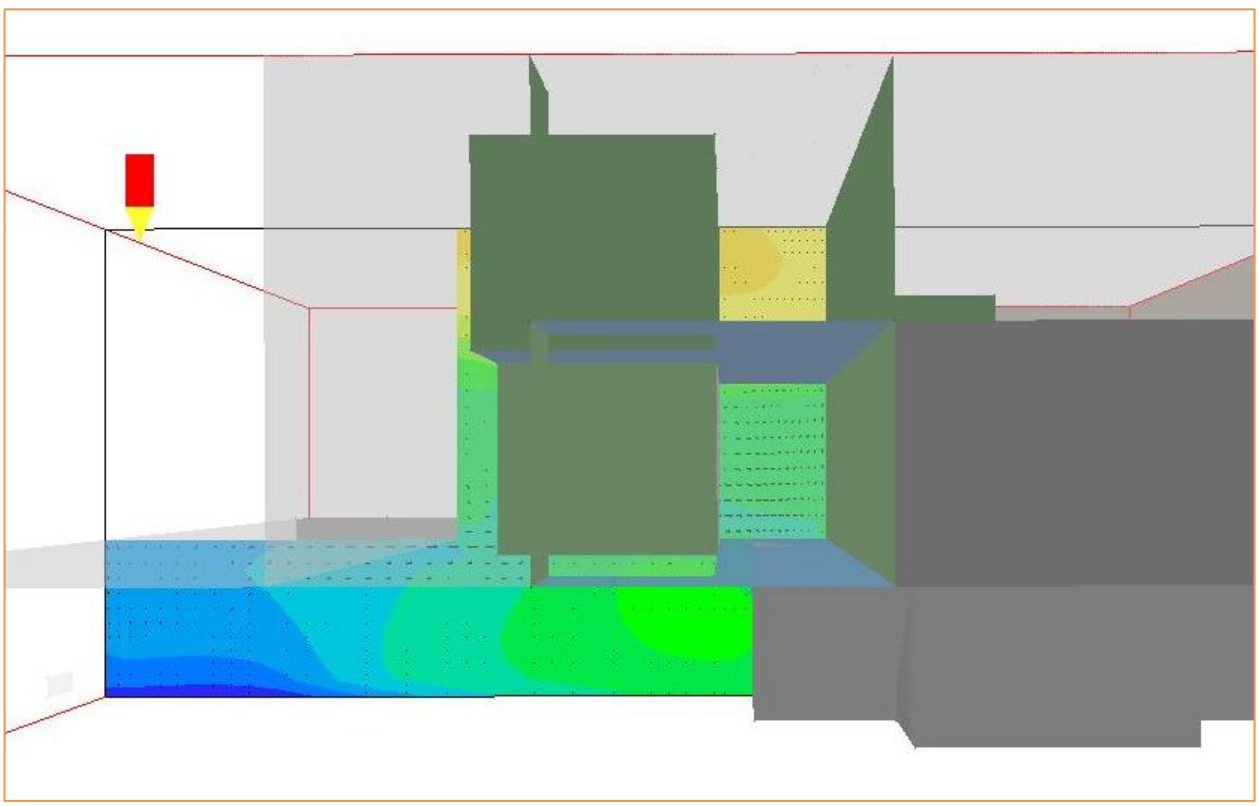

Figure 30: Temperature Profile (27.15°C scale) in X-Plane

These results showed good temperature stratification and provided a good representation of the means for the ventilation air to assist the room loads in providing thermal comfort for the occupants. Just as important is the velocity of the airflow in the spaces, and Figures 31-35 graphically illustrate these results. Figures 31-32 in particular show the airflow results achieved near the boundary conditions, while Figures 33-35 show the overall velocity field and vector results.

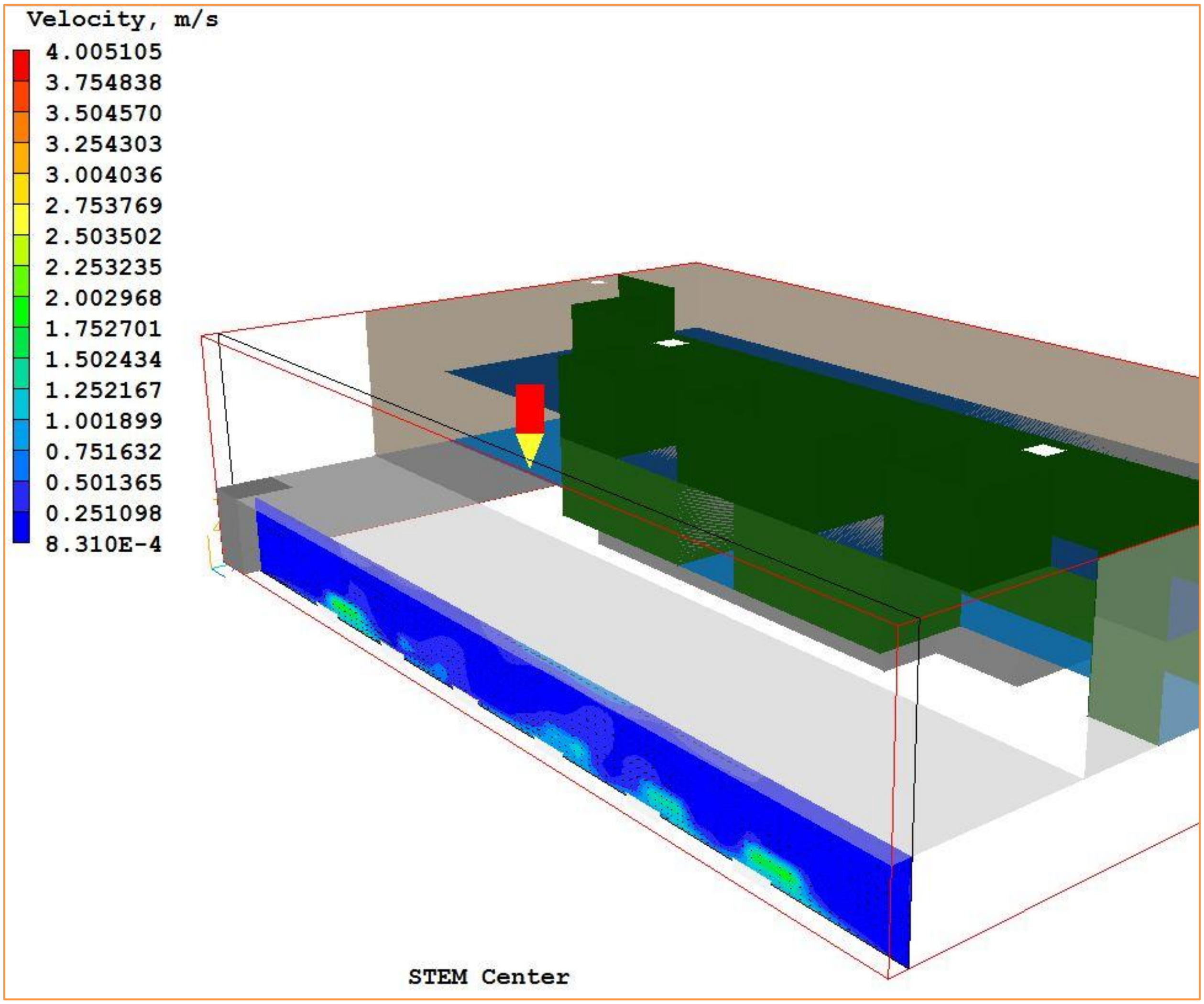

Figure 31: Velocity Profile (4 m/s scale) in Y-Plane at Windows

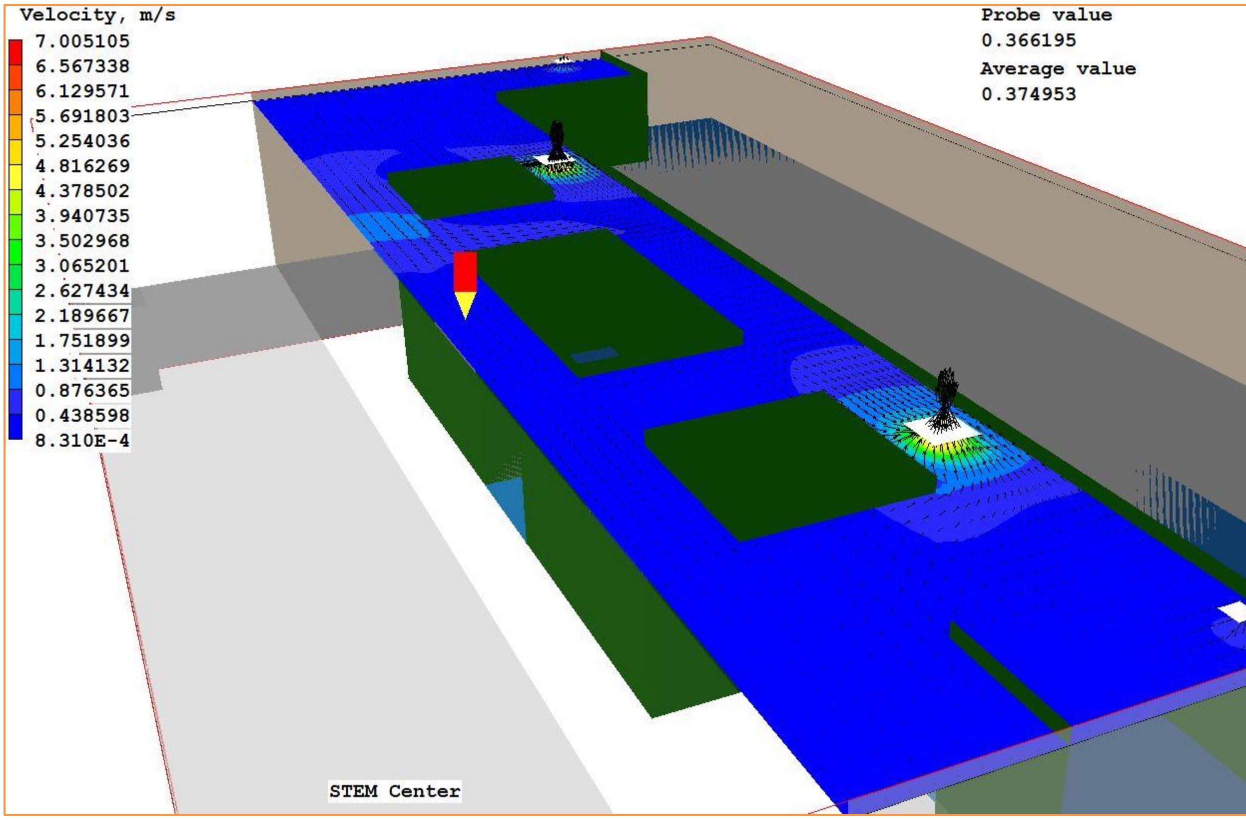

Figure 32: Velocity Profile (7 m/s scale) in Z-Plane at Exhaust Diffusers

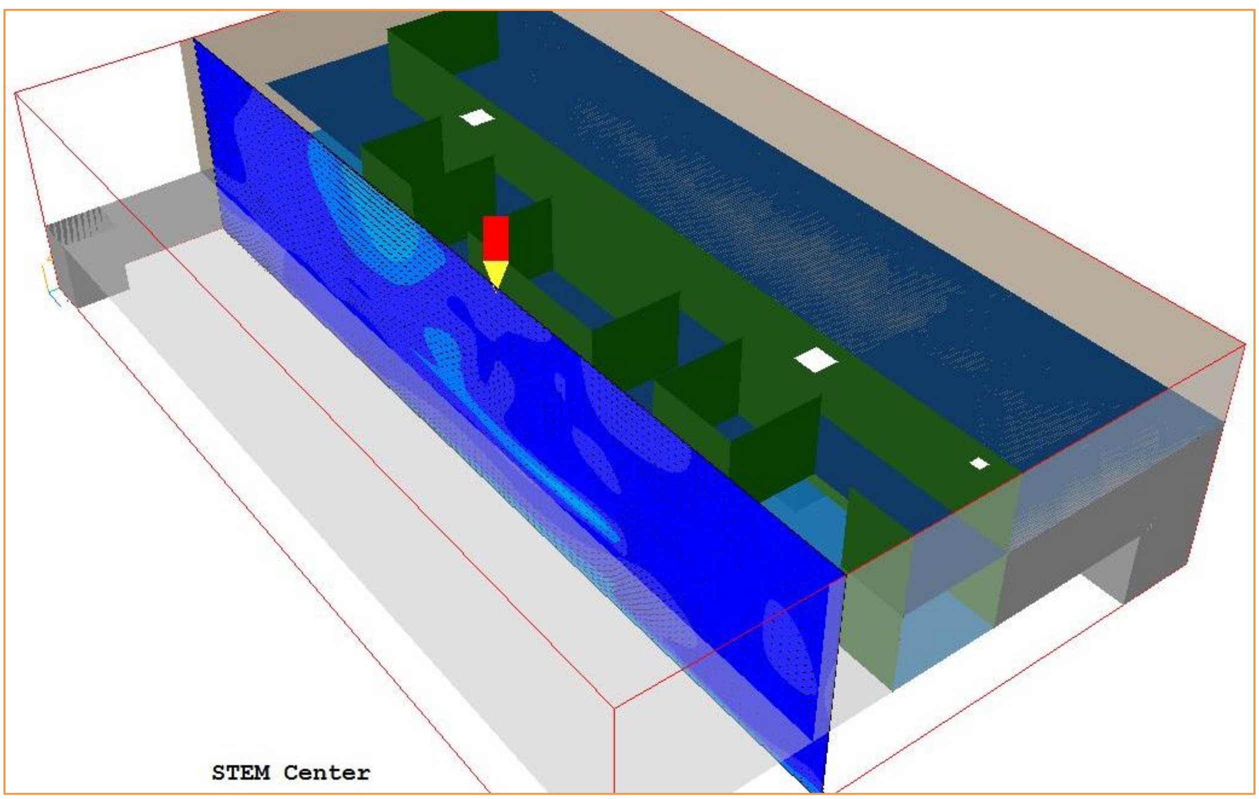

Figure 33: Velocity Profile (4 m/s scale) in Y-Plane at Atrium

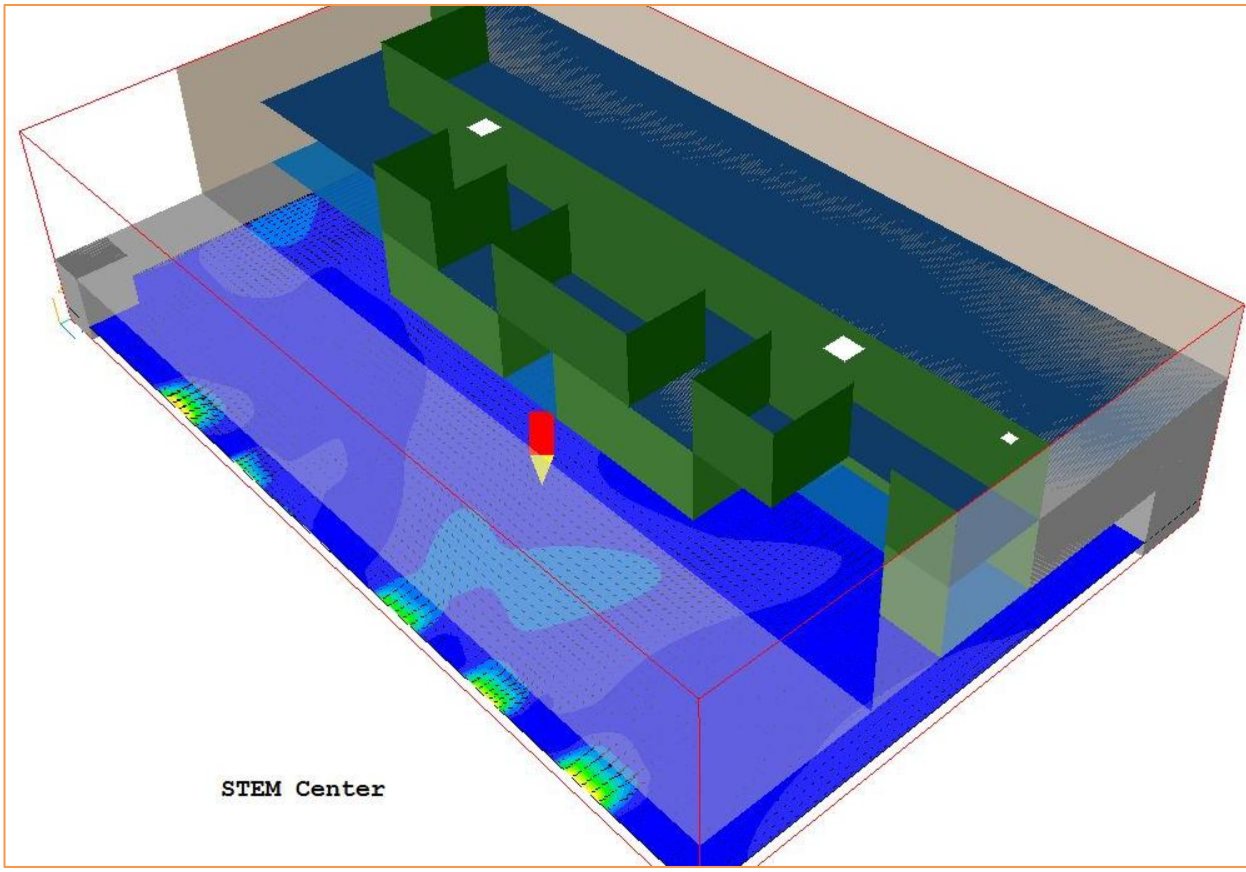

Figure 34: Velocity Profile (4 m/s scale) in Z-Plane

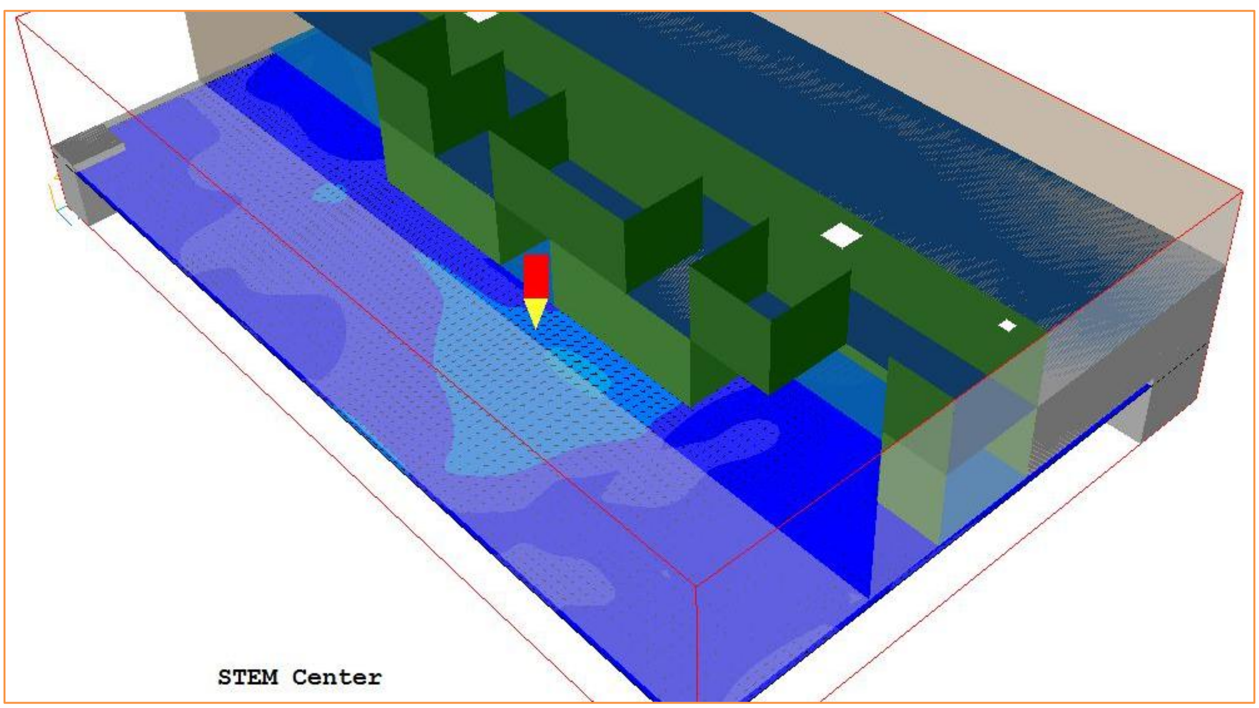

Figure 35: Velocity Profile (4 m/s scale) in Z-Plane

As can be seen by the image results of the CFD simulation, the airflow through the ground level windows provided good circulation of air without reaching very high velocities. This is important in regards to maintaining thermal comfort in the ventilated spaces.

In testing the quality of the results, the mass and temperature residuals were taken, and a 0.1% convergence criterion was set for the simulations, with a goal of less than 3% for the mass and temperature residuals. While the overall results were of high quality and the temperature residual was an appropriate 2.3567%, the mass residual was calculated to be 84.2084%. Because of this, a final simulation was run at 10,000 iterations, which produced slightly worse results, but produced much improved residuals. The whole-field mass residual was 2.261 and the nett source was 43.911419, which calculated to a mass residual of 5.149%. Improvement was seen for the temperature as well, with a whole-field value of 6,897, a nett source value of 1.281764 x 10^7, and a temperature residual of 0.0538%.

7.4 – ASHRAE Standard 55 Assessment

ASHRAE Standard 55 (2004) Thermal Environmental Conditions for Human Occupancy provides appropriate standards for the flow and temperature of air in a space and were highly applicable in the analysis of the use of natural ventilation for the STEM Center. The two main areas of concern in this assessment were in achieving a low enough temperature difference between ankle-level and head-level for a typical occupant to ensure good stratification, as well as low enough velocity relative to the draftiness of the room.

To test the requirements in section 5.2.4.3 Vertical Air Temperature Difference, the PHOENICS ability to measure temperature values at individual points in the domain was utilized. This section requires that the temperature difference between the head and ankle of typical occupant is less that 3°C (or 5.4°F) to have less than 5% PPD (Percentage of People Dissatisfied), which is the respective standard. Approximate heights of ankle and head, respectively, were specified as 0.1 and 1.6 meters (4" and 5'-3"), and at these distances in the z-direction from the ground floor, measurements were taken. All six points of measurement were within the ground level lounge spaces where the majority of the occupants would be and where the temperature stratification would be of greatest concern due to the incoming wind airflow. The results in vertical air temperature difference are shown in Table 24, illustrating that section 5.2.4.3 is very adequately met in ensuring good temperature stratification.

Vertical Level	x (m)	y (m)	z (m)	Probe T (°C)	Difference (°C)
Ankles	30	1	0.1	18.01	1.11
Head	30	1	1.6	19.12	
Ankles	30	5	0.1	18.10	1.22
Head	30	5	1.6	19.32	
Ankles	30	10	0.1	19.97	0.61
Head	30	10	1.6	20.58	
Ankles	37	1	0.1	17.95	1.42
Head	37	1	1.6	19.37	
Ankles	37	5	0.1	18.53	2.13
Head	37	5	1.6	20.66	
Ankles	37	10	0.1	19.43	1.48
Head	37	10	1.6	20.92	

***Table 24:** Section 5.2.4.3 Analysis for Vertical Air Temperature Difference*

For testing the requirements of section 5.2.4.2 Draft, only a 20% dissatisfaction percentage (defined as DR) is required to meet the standard. The standard provides an equation for calculating the DR-value as the following:

$$DR = ([34\text{-}t_a] * [v\text{-}0.05]^{0.62}) * (0.37 * v * Tu + 3.14)\ ,$$

where

DR = predicted percentage of people dissatisfied due to draft;

t_a = local air temperature, °C;

v = local mean air speed, m/s, based on v_a, the mean velocity; and

Tu = local turbulence intensity, %.

The local air temperature and mean air speed were determined for several different points by using the probe point value feature in PHOENICS, just as was done for the temperature difference analysis. In order to find the local turbulence intensity (Tu), a chart shown in Figure 36 was needed, and these calculated values were based on the known t_a and v variables.

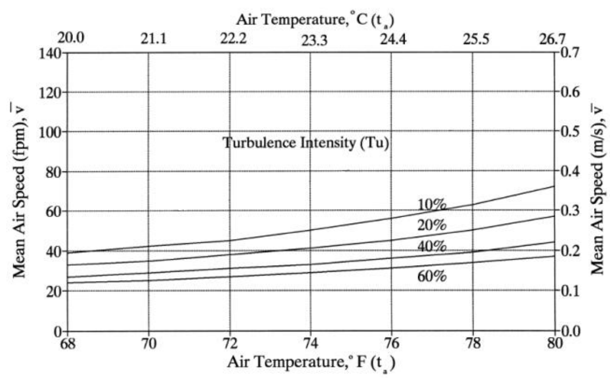

Figure 36: Section 5.2.4.2 Turbulence Intensity Chart

The positions tested for dissatisfaction due to draft were at several different heights, including the ground floor as well as the 2nd and 3rd floor corridors. Additionally, two x-planes were assessed for sufficiently low airflow draft levels. As mentioned previously, the requirements for percentage dissatisfied due to draft were only 20%, and the results for the calculated DR-values are shown in Table 25.

Percentage Dissatisfied Due to Draft				Probe Position		
DR (%)	ta (°C)	v (m/s)	Tu	x (m)	y (m)	z (m)
19.75%	21.24	0.27	0.10	-	-	1
18.46%	21.70	0.19	0.25	-	-	2
18.76%	21.84	0.18	0.30	-	-	3
18.74%	21.72	0.27	0.10	-	-	4
16.85%	21.88	0.24	0.10	-	-	6
17.56%	21.95	0.25	0.10	-	-	7
17.89%	22.08	0.26	0.10	-	-	8
17.37%	23.05	0.28	0.10	-	-	10
18.09%	23.23	0.30	0.10	-	-	11
18.81%	23.44	0.32	0.10	-	-	12
16.85%	22.62	0.25	0.10	10	-	-
16.25%	21.45	0.19	0.15	50	-	-

Table 25: Section 5.2.4.2 Analysis for Percentage Dissatisfied Due to Draft

7.5 – Natural Ventilation Conclusions

Overall, the two main concerns for natural ventilation, temperature stratification and draft, were shown to be adequate to meet the requirements outlined in ASHRAE Standard 55, Section 5.2.4. These results of the PHOENICS simulation of computational fluid dynamics show that natural ventilation is a system that is able to be utilized as an aid to the existing means to handle the building loads. The orientation of the site in relation to the prevailing northwest wind provides a reasonable wind flow that can enter the building at the ground level through operable windows. Exhaust fans at the top of the 3rd floor make this arrangement a sort of hybrid ventilation scheme, and one that, while it would need to be explored further in more sophisticated manners, leads to a conclusion that this portion of the STEM Center could benefit from the utilization of natural ventilation.

SECTION 8 – Acoustical Breadth

8.1 – Acoustical Analysis Procedure

Based on the mechanical system redesign for the ground floor, there are a few minor changes in the acoustical characteristics that merited analysis to determine the implications, both positive and negative, of the proposed design. The two main causes of alterations in the acoustics of the radiant spaces are the change from carpet to tile floor finish in two rooms as well as the removal of VAV boxes as discussed in Section 6.3. In order to calculate the acoustical changes associated with these design alterations, methods and equations from the textbook *Architectural Acoustics*, by Marshall Long (2006) were utilized.

8.2 – Reverberation Time Analysis

With the change to a radiant floor slab, there was a need for tile flooring in each of the radiant spaces. While nearly all of the radiant spaces already used an unglazed porcelain tile by Stone Source, two of the spaces were finished with carpet flooring. The necessary floor finish change covered only a total of 4,829 SF between the two spaces, and was a minor aesthetic alteration. The acoustical change is in the sound absorption coefficients of the two different materials. Table 26 shows the comparison between those two absorption coefficients from 125 Hz to 4000 Hz octave bands.

	Absorption Coefficients					
Material	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
CARPET	0.02	0.06	0.14	0.57	0.6	0.65
TILE	0.01	0.01	0.015	0.02	0.02	0.02

Table 26: Absorption Coefficients of Flooring Materials

For the acoustical analysis associated with the floor finish changes, the reverberation time was calculated for each room for both the old and new design. The Sabine equation was used to determine the reverberation time (T60), which is defined as the time it takes (in seconds) for the sound to decay by 60 dB. The Sabine equation takes into account the volume of the space and the total absorption of each material at all the aforementioned frequencies. Tables 27 and 28 show the results of the reverberation time calculations for each space, including the absorption coefficients used for each material.

Computer Lounge		Absorption Coefficients					
Material	Area (SF)	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
CARPET	3,615.08	0.02	0.06	0.14	0.57	0.6	0.65
TILE	3,615.08	0.01	0.01	0.015	0.02	0.02	0.02
ACT	3,615.08	0.72	0.84	0.7	0.79	0.76	0.81
GWB	779.09	0.29	0.1	0.05	0.04	0.07	0.09
GLASS	886.54	0.18	0.06	0.04	0.03	0.02	0.02
Volume	**18,651.73**						
Computer Lounge	Reverberation Times						
Old T60	0.298606	**0.27**	**0.2938**	**0.1837**	**0.1832**	0.170324	
New T60	0.302175	**0.2853**	**0.3437**	**0.3061**	**0.316**	0.295928	

Table 27: Reverberation Calculations for Computer Lounge

Lower Lounge		Absorption Coefficients					
Material	Area (SF)	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
CARPET	1,213.38	0.02	0.06	0.14	0.57	0.6	0.65
TILE	1,213.38	0.01	0.01	0.015	0.02	0.02	0.02
ACT	1,213.38	0.72	0.84	0.7	0.79	0.76	0.81
GWB	612.96	0.29	0.1	0.05	0.04	0.07	0.09
GLASS	1,268.12	0.18	0.06	0.04	0.03	0.02	0.02
Volume	**16,961.19**						
Lower Lounge	Reverberation Times						
Old T60	0.637384	**0.676**	**0.7551**	**0.4852**	**0.4836**	0.448742	
New T60	0.643371	**0.7111**	**0.8758**	**0.795**	**0.8191**	0.764134	

Table 28: Reverberation Calculations for Lower Lounge

While the changes are minimal at the lower frequencies, the reverberation time of the tile floor design at the higher frequencies is quite substantial. Because the middle frequencies (the main levels for human speech) are the ones that are of concern for these two lounge spaces, the T60 values of those four (250, 500, 1000, and 2000 Hz) were averaged for an overall reverberation time comparison. The results of this comparison are shown in Tables 29 and 30, and they highlight the increase in reverberation time for both spaces.

Computer Lounge	T60 (sec)
Old (Carpet)	**0.23**
New (Tile)	**0.31**

Table 29: Reverb Time for Computer Lounge

Lower Lounge	T60 (sec)
Old (Carpet)	**0.60**
New (Tile)	**0.80**

Table 30: Reverb Time for Lower Lounge

As can be seen, the result is a slightly higher T60, meaning a greater amount of echo and time needed for sound decay in these spaces. These numbers, when compared to the chart in Figure 37, lie roughly in the common band of reverberation times. Sound decays in these spaces very quickly due to the incomplete enclosure and relative openness.

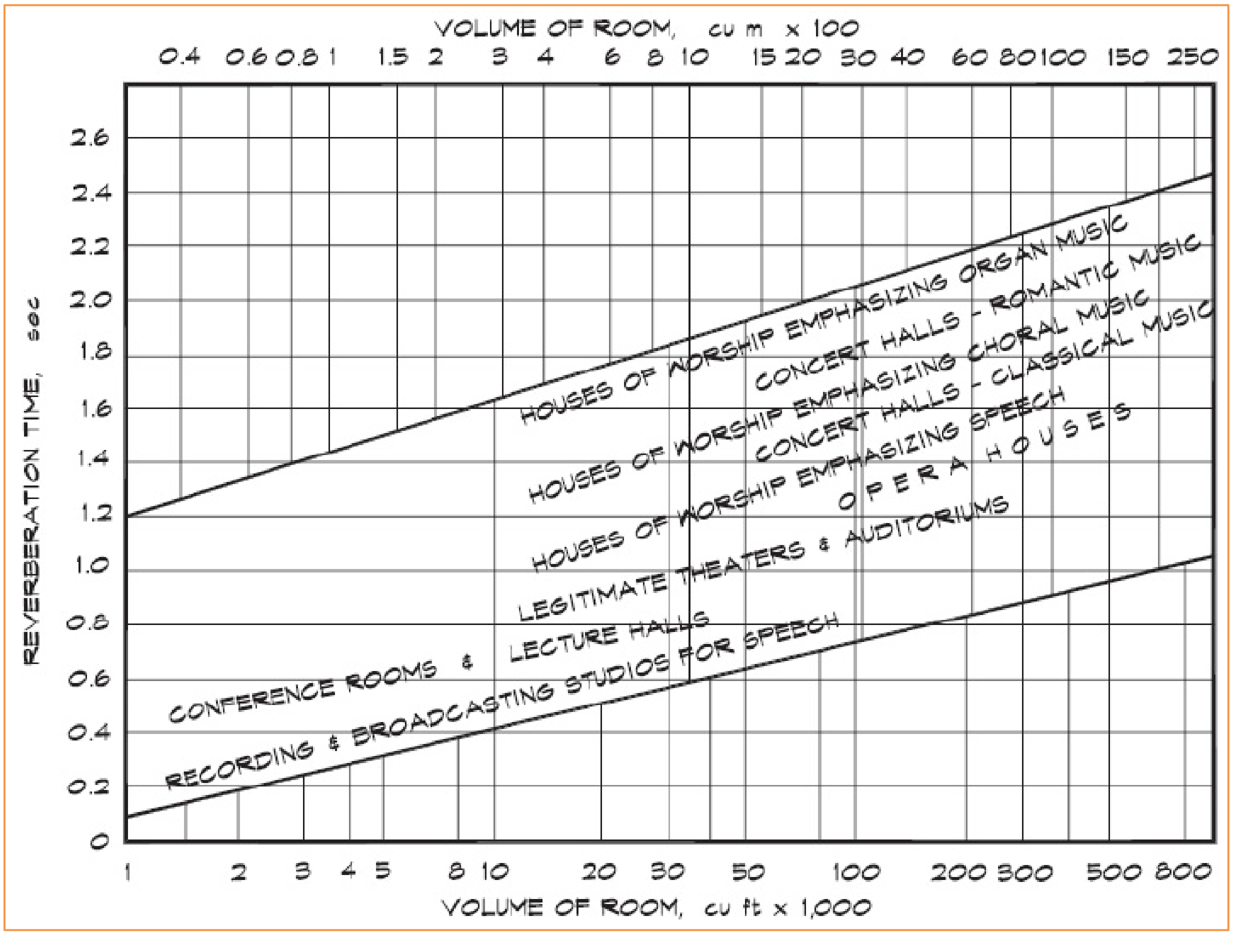

Figure 37: Reverb Time Chart

8.3 – Mechanical Noise Analysis

While reverberation is a substantial concern in architectural and acoustical design, mechanical noise associated with the ductwork leading to a room is at times also a large source of acoustical problems. An acoustical improvement from the mechanical redesign emerged from the elimination of the numerous VAV terminal units used for the eight radiant spaces. Each of these boxes is a source of noise that travels through the ductwork to the supply diffuser, and the elimination of them is a certain advantage for the acoustical setting of each of the spaces.

In particular, the lounges are of greater concern than the lobby areas, and an example calculation was performed for the duct-borne noise from the old and new mechanical designs in room S100C Lower Lounge (shown in Figure 38). Manufacturer data was available for the diffusers in the form of a Noise Criterion (NC) rating, from which sound power levels (L_W values) for the self-noise could be determined.

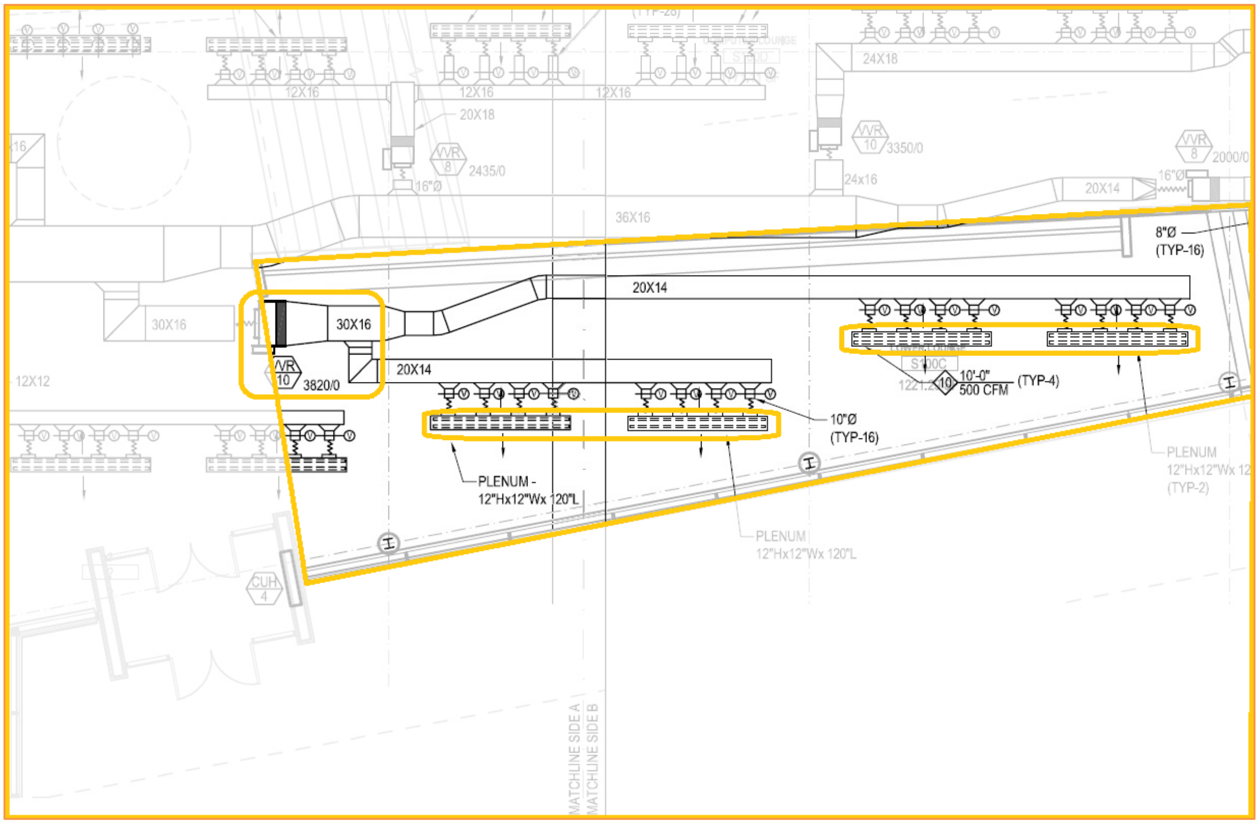

Figure 38: Mechanical Noise Example Space

For the example room, the existing design has a VAV unit (highlighted in Figure 38) sized at 3820 cfm that serves two duct branches and a total of four diffusers. Sound is lost at several different points after the VAV, including the branch split, the elbows, and the runs of duct themselves. The branch splits, elbows, and diffusers are the sources of self-noise that are added with the cumulating sound power level using dB addition as the calculation goes through each step. In Tables 31 and 32, the step-by-step increase and decrease of sound power levels through the duct from the VAV to each of the four diffusers are shown.

Frequency (Hz):		125	250	500	1000	2000	4000
VVR-10	Self Noise	64	62	62	56	51	48
Split	Sound Loss	-3.03721	-3.03721	-3.03721	-3.03721	-3.03721	-3.03721
		60.96279	58.96279	58.96279	52.96279	47.96279	44.96279
Split	Self Noise	41.24766	36.04068	30.8337	25.62672	20.41974	15.21276
		61	59	59	53	48	45
Duct	(B. 1, 26')	-5.65874	-3.78549	-2.53236	-1.69405	-1.13326	-0.75811
20x14	(B. 2, 34')	-7.39988	-4.95025	-3.31154	-2.2153	-1.48196	-0.99138
	Branch (1)	55	55	56	51	47	44
	Branch (2)	54	54	56	51	47	44
Diffuser	Self Noise	48	43	37	34	31	30
Lw (1)		56	55	56	51	47	44
Lw (2)		55	54	56	51	47	44
Lw(1+2)		**59**	**58**	**59**	**54**	**50**	**47**

Table 31: Mechanical Noise Calculations, Diffusers 1 and 2

Frequency (Hz):		125	250	500	1000	2000	4000
VVR-10	Self Noise	64	62	62	56	51	48
Split	Sound Loss	-3.03721	-3.03721	-3.03721	-3.03721	-3.03721	-3.03721
		60.96279	58.96279	58.96279	52.96279	47.96279	44.96279
Split	Self Noise	40.58944	35.38246	30.17548	24.9685	19.76152	14.55454
		61	59	59	53	48	45
Elbow	Sound Loss	-1	-5	-8	-4	-3	-3
		60	54	51	49	45	42
Elbow	Self Noise	10	5	0	0	0	0
		60	54	51	49	45	42
Duct	(B. 3, 4')	-0.87057	-0.58238	-0.38959	-0.26062	-0.17435	-0.11663
20x14	(B. 4, 14')	-3.04701	-2.03834	-1.36358	-0.91218	-0.61022	-0.40821
	Branch (3)	59	53	51	49	45	42
	Branch (4)	57	52	50	48	44	42
Diffuser	Self Noise	48	43	37	34	31	30
Lw (3)		59	53	51	49	45	42
Lw (4)		58	53	50	48	44	42
Lw(3+4)		**62**	**56**	**54**	**52**	**48**	**45**
Lw(1+2)		59	58	59	54	50	47
Lw (total)		**64**	**60**	**60**	**56**	**52**	**49**

Table 32: Mechanical Noise Calculations, Diffusers 3 and 4

Using the total sound power level calculated, the diffuse field level (the sound pressure level at a significant distance from the diffuser, L_P) was determined using the room constant (R) known from the reverberation calculation for the space discussed in Section 8.2. From the diffuse field level for the room, shown for the original design in Table 33, the Room Criterion (RC-value) was determined based on the RC chart taken from *Architectural Acoustics* and shown in Figure 39.

Lw (total)	64	60	60	56	52	49
R	1291.787	1168.756	948.9395	1045.4	1014.706	1087.634
Lp	39	35	36	32	28	25

Table 33: Diffuse Field Calculation for Sound Pressure Level

The total L_W for the new design was solely based on the sound power level emitted from the diffusers, as specified in the mechanical schedule. Highlighted in green are the old sound pressure levels for the diffuse field in the space, and in orange are the new levels.

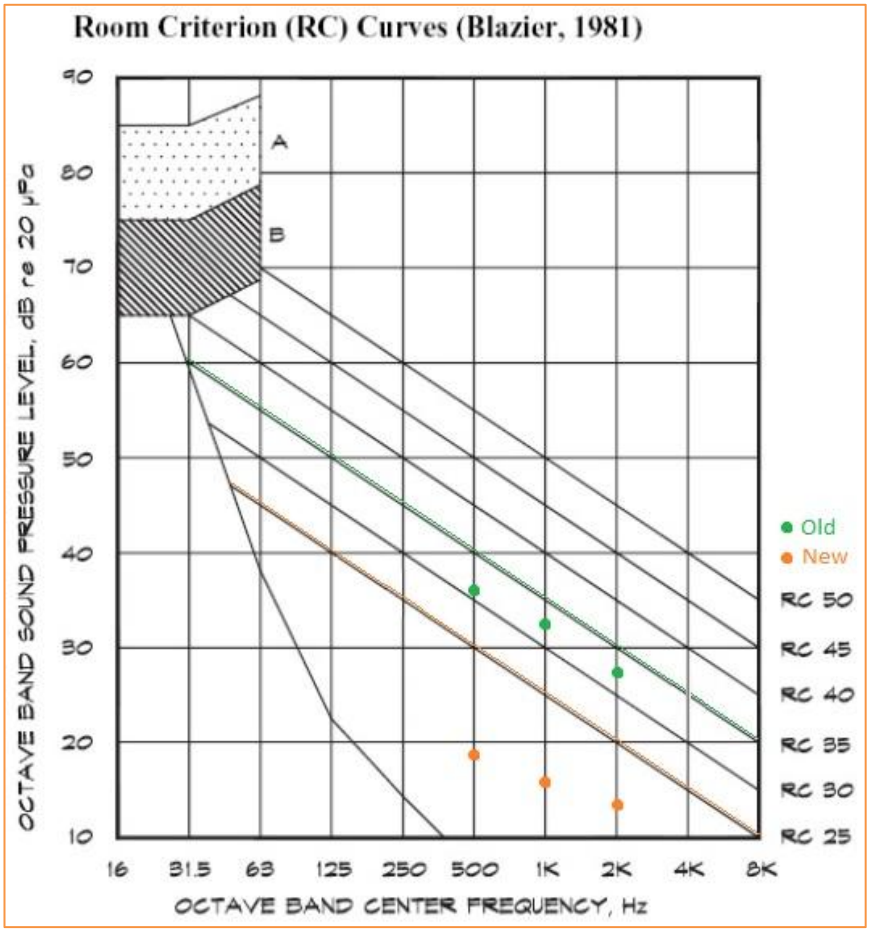

Figure 39: Room Criterion Curves

The Room Criterion rating method was chosen over the Noise Criterion (NC) rating because it is based on the arithmetic average of the three middle octave bands of 500 Hz, 1000 Hz, and 2000 Hz. This means that is zeroes in on the preferred speech interference level, which is of greatest concern in a lounge-type setting where conversations take place and could be interrupted by background mechanical-related noise.

8.4 – Acoustical Conclusions
From the acoustical breadth analysis of the implications of the mechanical redesign, it can be concluded that minimal detractions and a couple improvements will result from the changes. In the way of reverberation time, the T60 calculations showed very little difference for the two spaces that required an alteration in floor finish from carpet to porcelain tile to accommodate the radiant floor design. These spaces are not critical in the way of reverberation and so the minimal change that occurred will be hardly detectable in that regard.

For the mechanical noise entering the space through the ductwork, an improvement in acoustical quality was seen through the removal of VAV terminal units in the ductwork. These fans are a source of great sound power (L_W) and resulted in a RC-value of 35 for the Lower Lounge. As removed, the diffusers alone were tested for noise level and calculated a RC-value of 25. With this in mind, it is concluded that the acoustical implications of the mechanical were not significant, and provided a slight improvement for most spaces in eliminating excess mechanical noise through the supply diffusers.

SECTION 9 – Construction Management Breadth

9.1 – Analysis Procedure
With the implementation of a radiant floor heating and cooling design, there exists a concern for the construction management implications of the design from both a cost and schedule standpoint. A full life cycle cost analysis was needed to determine the feasibility of the proposed mechanical system design, based on values obtained from the RS Means online cost books, the cost data for the existing design, and the previously mentioned utility costs as calculated by TRACE.

The TRACE energy simulation determined that the new radiant floor and DOAS systems would result in a yearly savings of nearly $2,000. This is not a great amount when compared to the yearly utility costs of the STEM Center as a whole; however the design was only for most of the ground floor, and roughly 1/6th of the building.

Several items required an estimated value for their capital cost, including the equipment needed for the radiant floor system and the additional AHU for the ventilation air. For all of these cost estimates, the online version of RS Means (2010) was used for up-to-date and accurate values.

9.2 – Life-Cycle Cost Analysis
Just as there were additional capital costs for added equipment, there were savings calculated due to the reduction of air terminal units and ductwork from the system redesign. These, too, were priced according to RS Means. Table 34 shows the overall costs for each element of the mechanical alternative design, and the broken down costs are shown in Appendix E.

DOAS System Costs		Initial Cost ($)
	AHU	$18,075.00
	HX/Wheel	$13,850.00
Radiant System Costs		
	Piping	$20,615.00
	Pumps	$6,570.80
Overall Redesign Savings		
	Terminal Units	-$27,058.00
	Ductwork	-$1,181.25
TOTAL		$30,871.55

Table 34: Capital Cost Calculations

This increase in cost was expected due to the complexity of installing and equipping a radiant floor design, and it was necessary to calculate the payback period for such an investment according the National Institute of Standards and Technology (NIST) Handbook 135 Life-Cycle Costing Manual. The 1995 Edition was used for the calculations, including the tables for projected electricity and natural escalation factors, organized by end-use sector and fuel type for the Census Region 1, which includes Pennsylvania. The results of the Life-Cycle Cost Analysis, which are seen completely in Appendix F, reveal a payback period of 18 years. After an assumed life cycle of 30 years, the net profit from the mechanical alternative design will be $19,073.15, as shown in Figure 40, and Figure 41 illustrates the yearly savings and gradually decreasing initial investment.

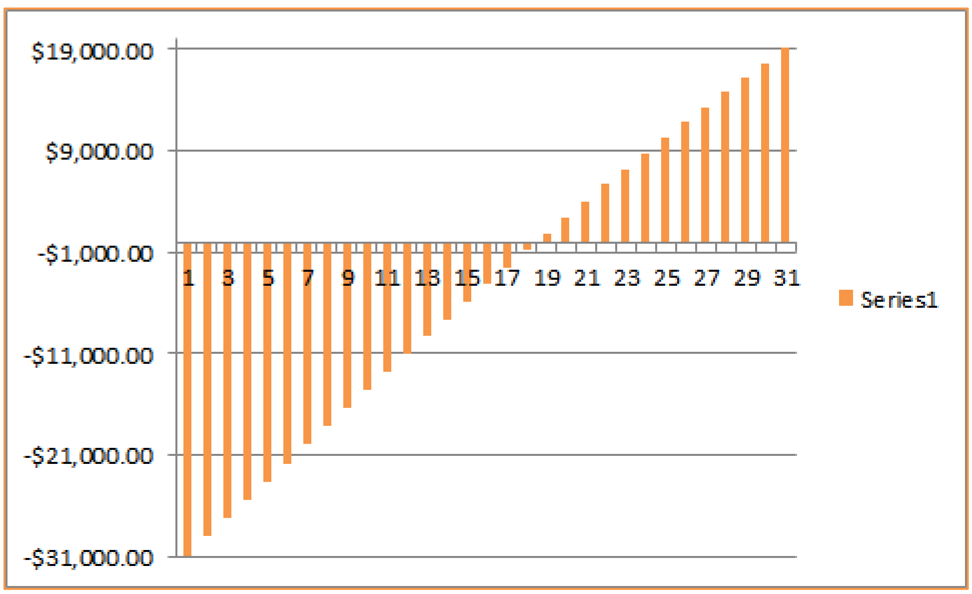

Figure 40: Payback and Profit after 30 years

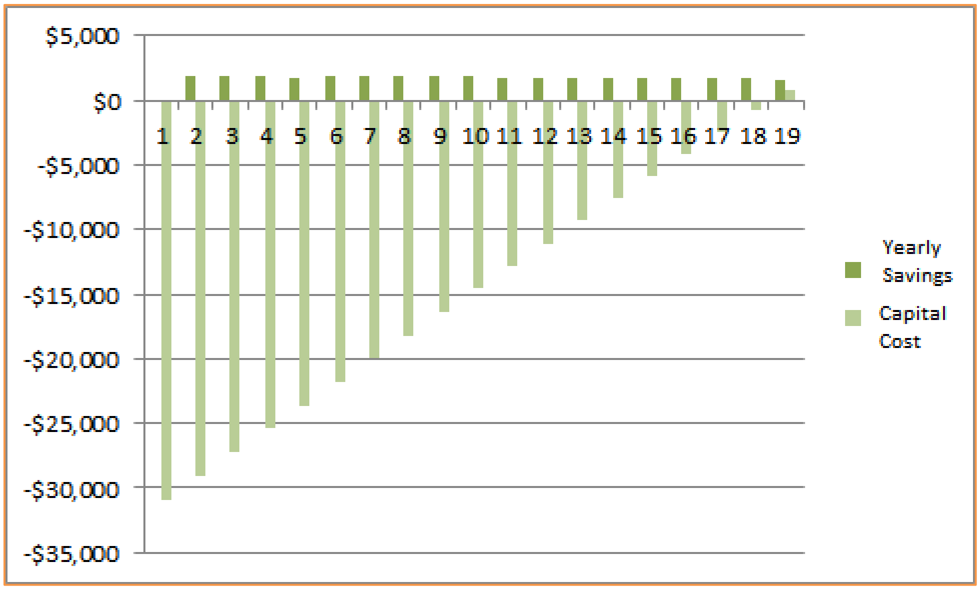

Figure 41: Payback Period

Not included in these calculations is the change in floor finish material discussed in Sections 5.1 and 8.2, from carpeting to porcelain tile for the Lower and Computer Lounges. The acoustical implications have been analyzed, and similarly, the cost comparison results in minimal change. Shown in Table 35 is the comparison that reveals a slight savings of just over $700, based on cost information provided for the porcelain tile and RS Means literature for the carpeting.

	Line Number	Description	Unit	Total O&P	Area (SY and SF)	Price
Carpet	96813101100	Tufted 24"x24"	S.Y.	38.95	536.4956	$20,896.50
Tile	93013103300	Porcelain Tile	S.F.	4.18	4828.46	$20,182.96
					Cost Decrease	**-$713.54**

Table 35: Flooring Cost Implications

9.3 – Schedule Implications

The glaring concern for the added equipment was in regard to the scheduling impact, particularly in the way of the radiant flooring. As discussed in Section 5.1 and shown in Figure 14, a radiant floor system involves a 2” increase in floor thickness with the added PEX (cross-linked polyethylene) tubing and gypsum concrete fill. It is the laying of the piping and gypcrete that causes the greatest impact on the schedule by adding 29 days. As shown in Table 36, after the adding and subtracting of necessary construction days due to the changes in mechanical and architectural feature, there are a total of 24 additional days needed to complete the construction of the STEM Center. This extra time equates to a little over a month of additional work and delay in opening, but the flexibility of time between the completion and occupancy is sufficient to deem this time delay acceptable.

Task	Old Days	New Days	Change Days
Rooftop HVAC Equipment	12	3	3
HVAC Pipe Rough-Ins	40	69	29
HVAC Equipment	20	18	-2
HVAC Duct Rough-Ins	10	9	-1
Porcelain Tile	17	22	5
Carpeting	10	0	-10
TOTAL		**+24 Days**	

Table 36: Schedule Impact of Alternative Design

9.4 – Construction Management Conclusion

The effects of the mechanical redesign are significant and of concern for the implementation of the proposed alternative systems. Based on Life-Cycle Cost Analysis, the payback period for the upfront cost of mechanical equipment is 18 years, which is substantial but will ultimately mean an improvement in energy efficiency for the building to provide the necessary comfort to the occupants. While the schedule impact for the installation of radiant floor slab for the eight designated spaces is sizeable as expected, it is partially offset by the reduction in ductwork and terminal units. Overall, these construction management consequences are a concern for the design, but not an insurmountable roadblock in the way of improving building performance and efficiency. It is concluded that the proposed alternative systems for mechanical design have an impact on the construction management for the STEM Center, but are not an overwhelming deterrent for the redesign and its implications.

SECTION 10 – Conclusion and Recommendations

Based on the analysis of the proposed systems, a suitable alternative exists for the mechanical design of the ground floor of the STEM Center should the owner be willing to supplement the increase in upfront cost. The main source of the change lies in a switch to a radiant heated and cooled flooring system for about 15,000 square feet, which will save energy and operating costs as it has the means to fully meet the sensible loads for the spaces. This design will be coupled with a dedicated outdoor air system to take care of the ventilation requirements for the highly used spaces, and this will be complete with an enthalpy wheel and high efficiency heat exchanger capable of meeting the latent loads. Together, this mechanical system is an upgrade for the existing VAV design system that uses a substantial amount of supply air to condition the high volume spaces instead of the occupants.

From the computational fluid dynamics modeling conducted for the analysis of natural ventilation, there exists potential to decrease the utility costs even further. While further depth of analysis is required to develop a means to couple the two systems, external studies have shown that natural ventilation is a reasonable complement for radiant design, and may be used in this case to improve energy efficiency for the STEM Center even more.

Additionally, the effects of the unique radiant design for the building would have minimal impacts, but overall improvements, for the acoustical conditions in the lounge and lobby spaces. The main source of drawbacks for the implementation of this design comes from a modest initial cost and overall project schedule increase.

To achieve this enhancement of building performance, a capital cost increase for the alternate mechanical equipment would have to be approved of by the owner. If the Delaware County Community College were to undertake this investment for the alternative design of the STEM Center mechanical system, the energy efficiency for this already highly sustainable project would improve and profits from yearly cost savings would take place after eighteen years.

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Appendix A: MAE Course Work

In order to fulfill requirements for the integrated MAE program, the AE Senior Capstone Thesis Project for MAE students must include use of knowledge and technique from Masters course work. Methods and information learned from the following courses were exercised in the contents of this thesis report:

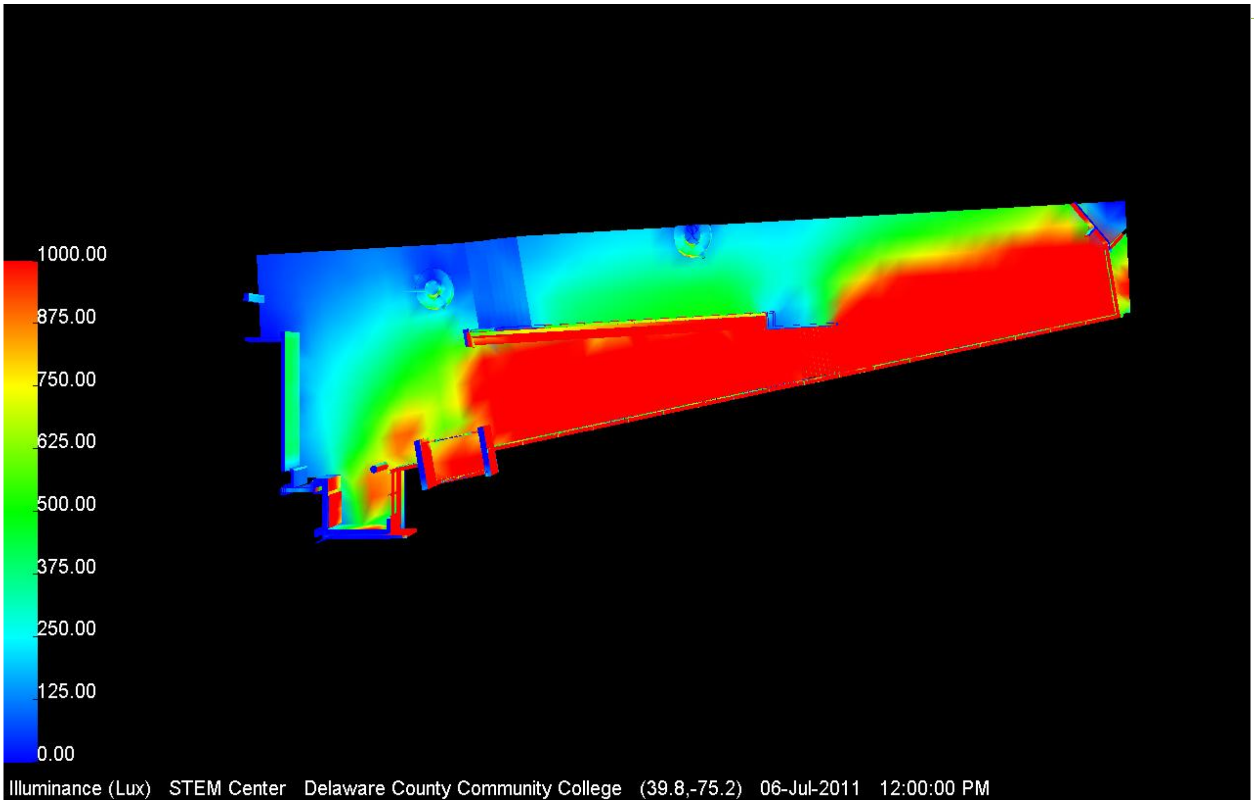
AE 558 – Centralized Heating Production and Distribution Systems

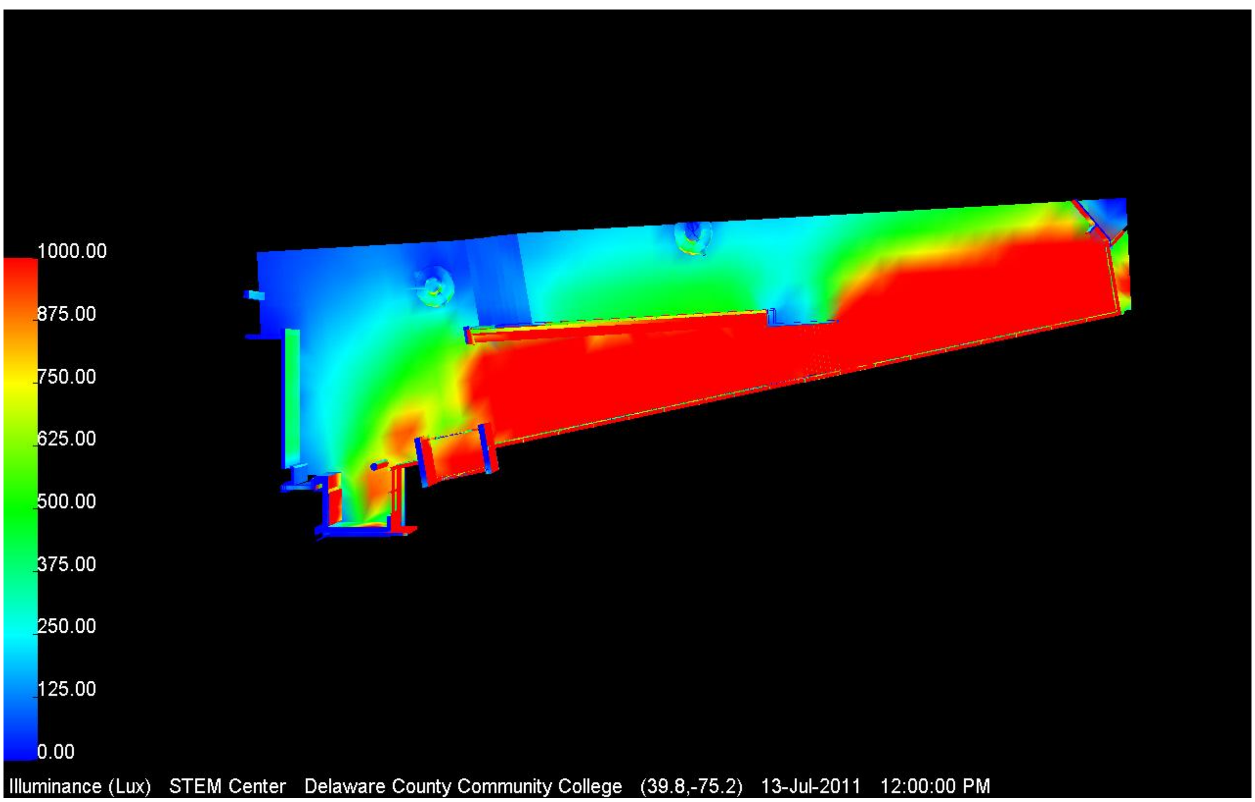
An element of this heating course included the procedures associated with the type of Life Cycle Cost analysis used in this report in Section 9.2. Methods for utilizing escalation factors and inflation rates were helpful in the calculation of the necessary cost information associated with the alternative mechanical systems.

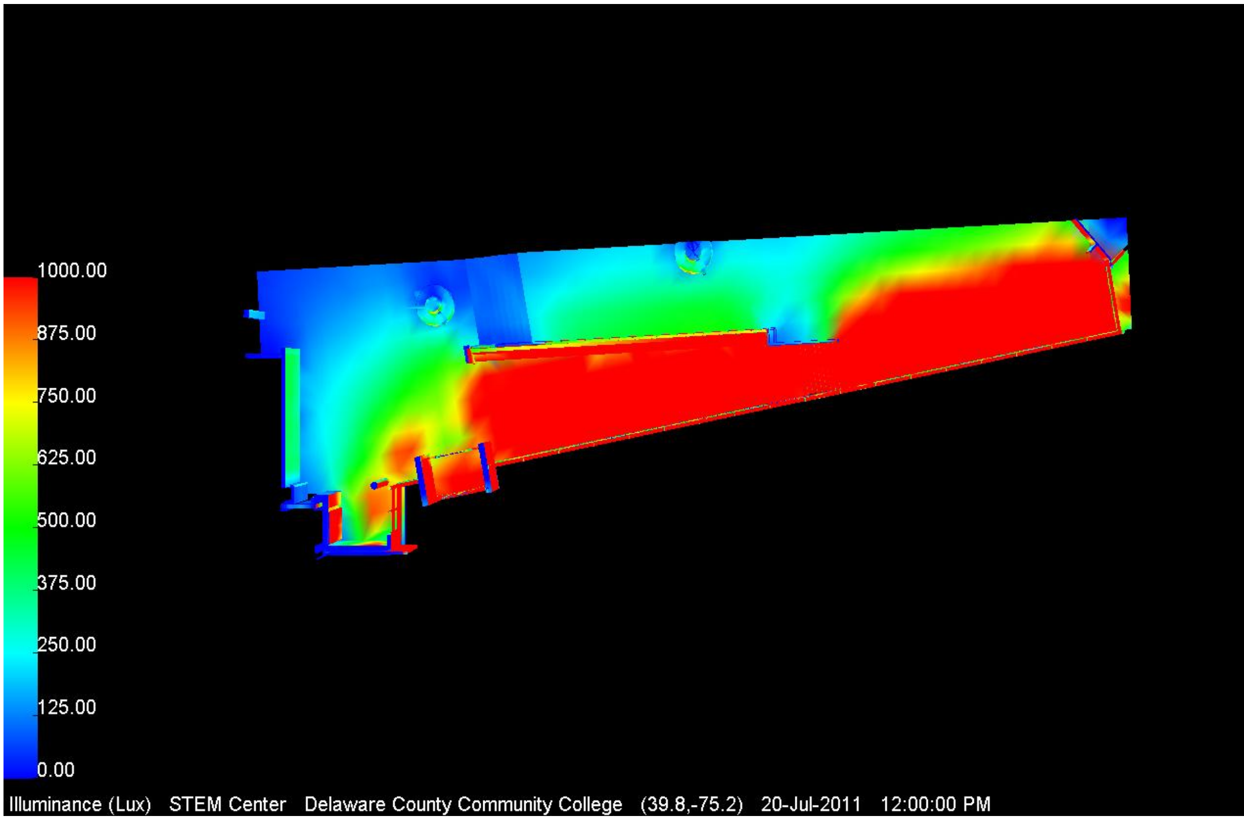
AE 559 – Computational Fluid Dynamics

A great deal of the knowledge of general CFD concepts and the practice with PHOENICS software from this course was critical for the study in natural ventilation. Through the CFD study conducted, airflow characteristics associated with a natural ventilation system were understood and applied to the mechanical depth analysis of this report.

Appendix B: AGI32 Daylight Study Images



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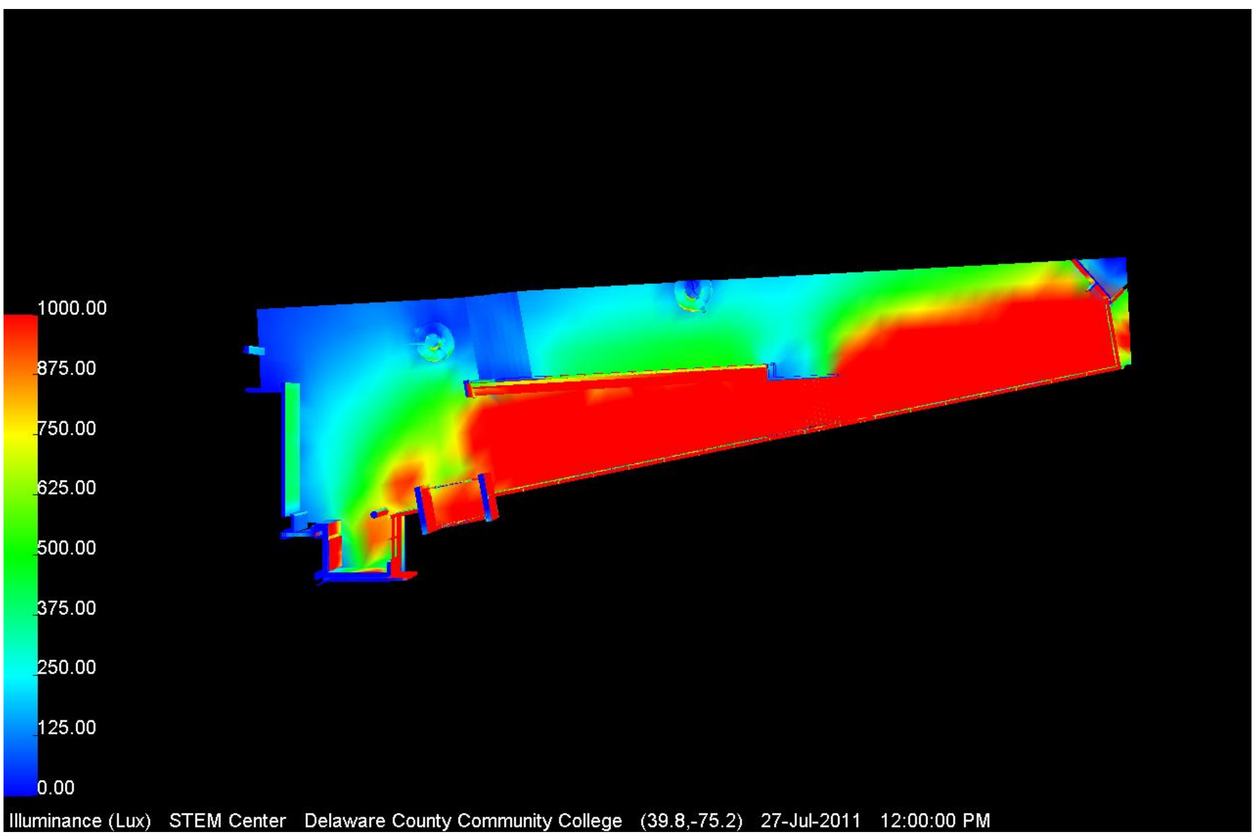
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Appendix C: MSP Technology Heat Exchanger Data

MSP TECHNOLOGY

MSP® HEAT EXCHANGER SELECTION

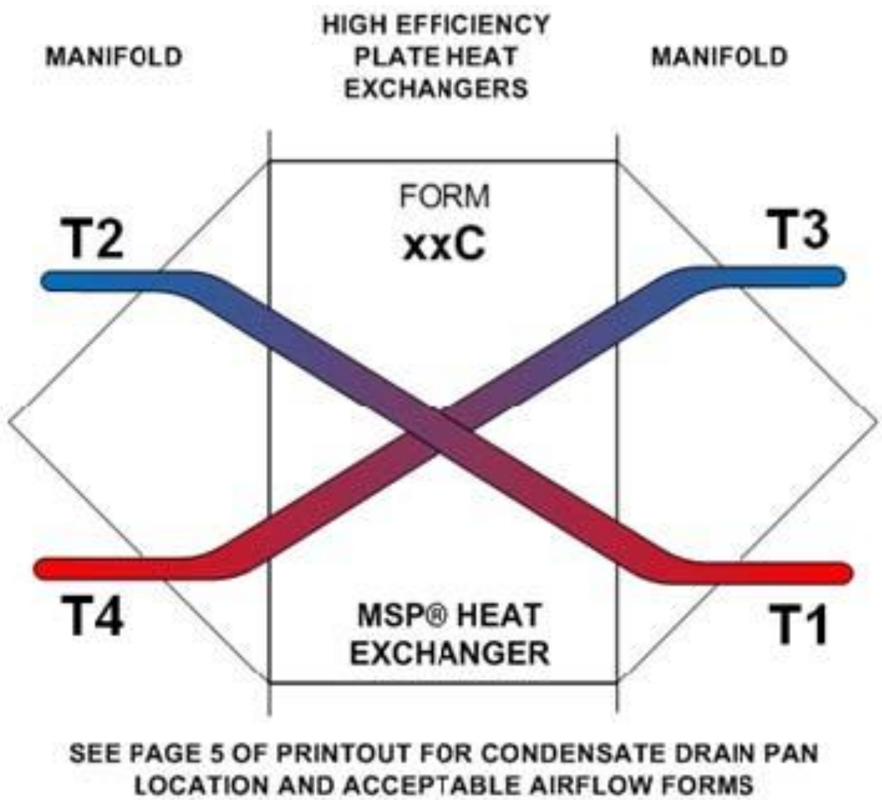
Note: All selections should be checked with factory

Project: STEM Center Model: 0811 Altitude: 118 ft

Unit Description: High Efficiency Plate Heat Exchanger

State 4				State 1			
SCFM:	10,740	W (gr/lb):	24	SCFM:	10,740	W (gr/lb):	87
ACFM:	10,955	DP (F):	29.7	ACFM:	11,170	DP (F):	63.2
DB (F):	78.6	WB (F):	54.0	DB (F):	85.0	WB (F):	70.0
RH (%):	16.4	h (Btu/lb):	22.7	RH (%):	48.2	h (Btu/lb):	34.1

SCFM: b): 87
ACFM: : 63.2
DB (F):): 70.0
RH (%): lb): 34.1



State 2				State 3			
SCFM:	10,715	W (gr/lb):	70	SCFM:	10,740	W (gr/lb):	24
ACFM:	10,822	DP (F):	57.3	ACFM:	10,418	DP (F):	29.7
DB (F):	67.2	WB (F):	60.9	DB (F):	50.0	WB (F):	41.0
RH (%):	70.8	h (Btu/lb):	27.2	RH (%):	44.8	h (Btu/lb):	15.7

MSP TECHNOLOGY

MSP® HEAT EXCHANGER SELECTION

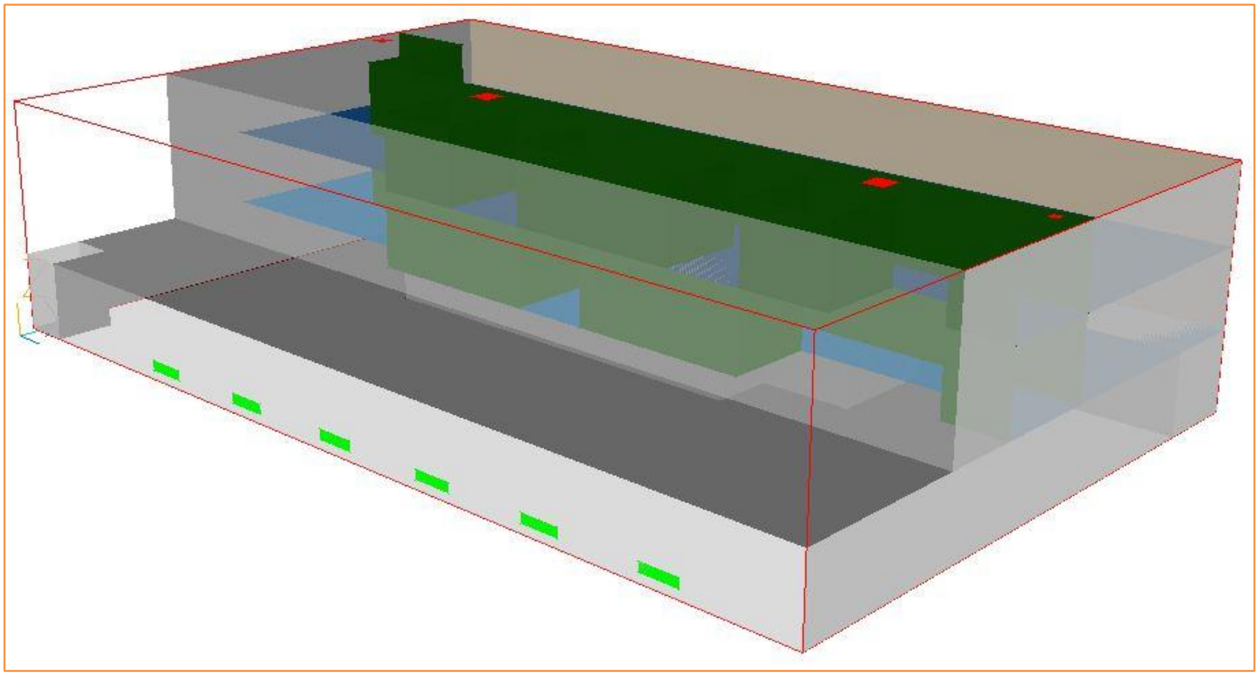
Note: All selections should be checked with factory

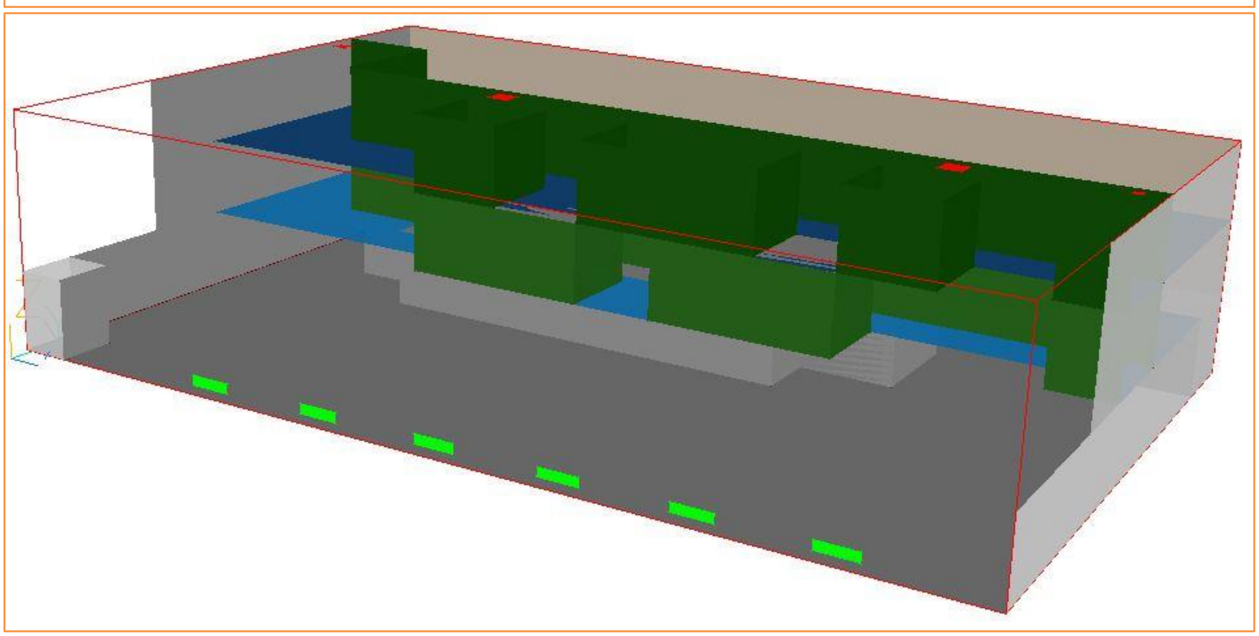
High Efficiency Plate Heat Exchanger **0811 Altitude: 118 ft**

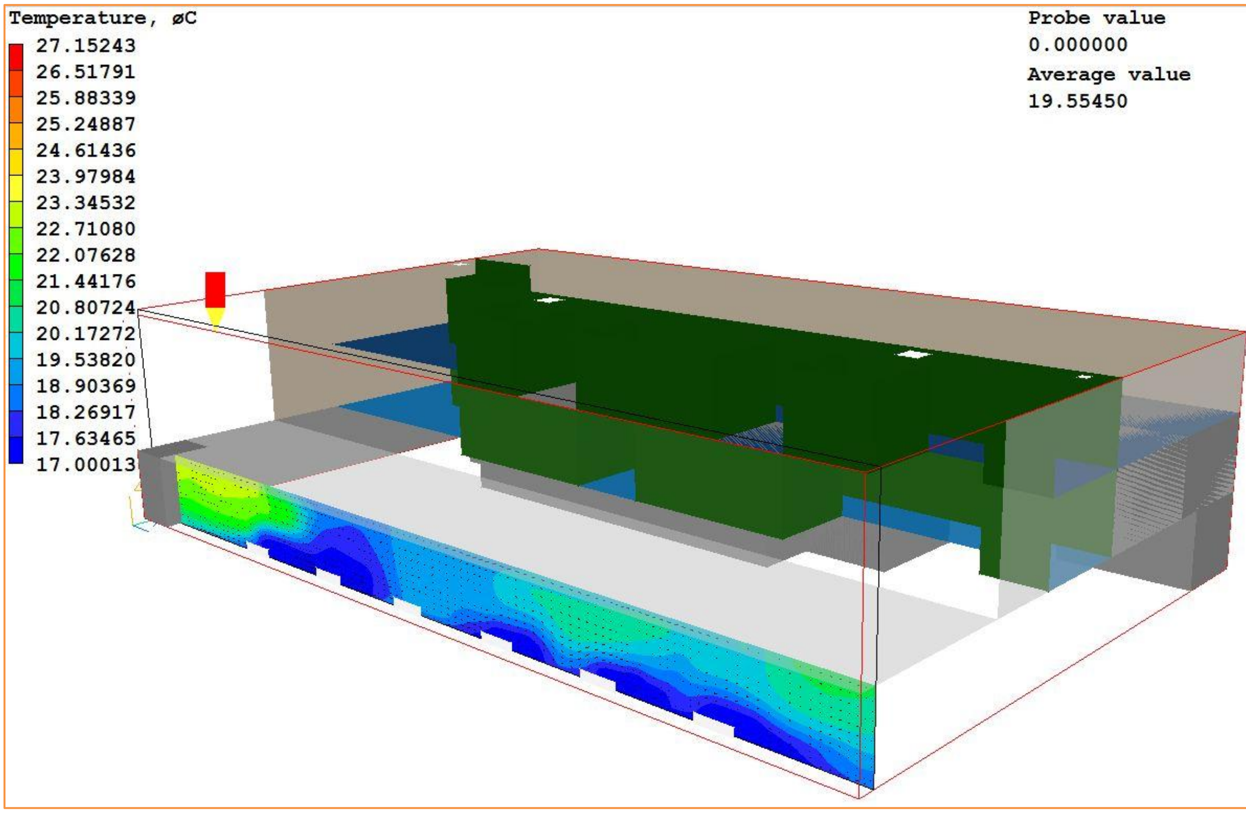
MSP® Heat Exchanger Performance

	Side 1->2	Side 3->4
Pressure drop (in):	0.51	0.42
Effectiveness (%)	51.0	81.8
Sensible Energy Recovered (MBH):	-333.0	333.0
Condensate (lbm/hr):	113.5	0.0

Appendix D: PHOENICS CFD Simulation Images



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Average value
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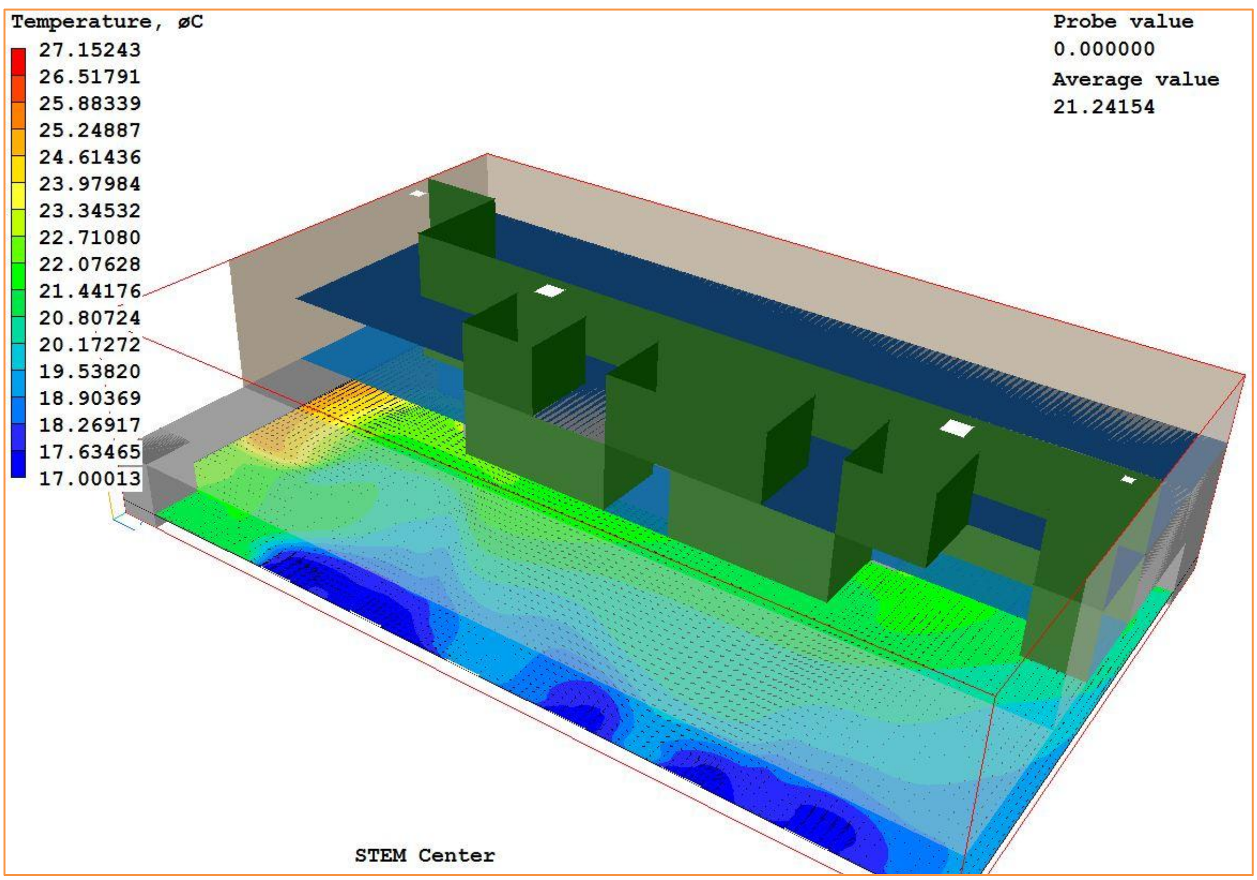
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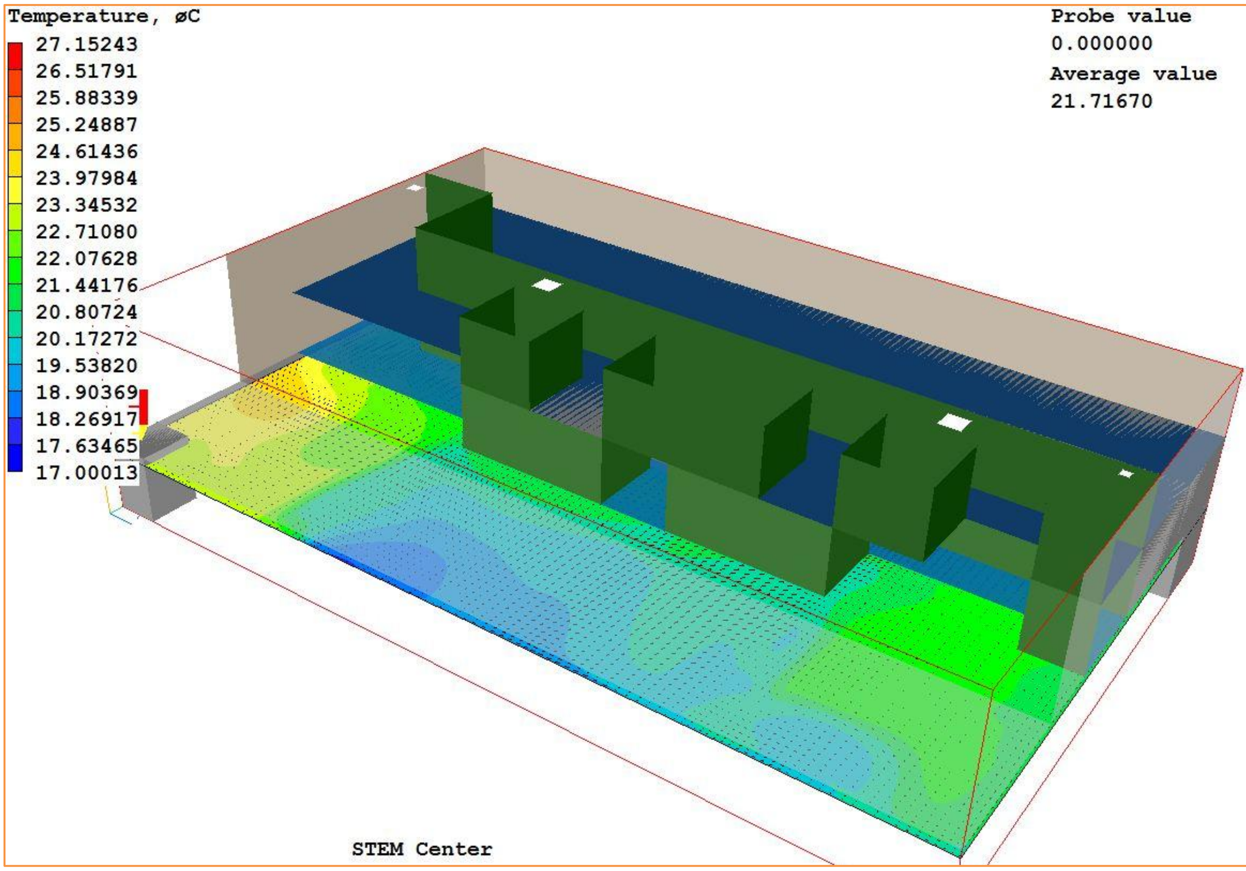
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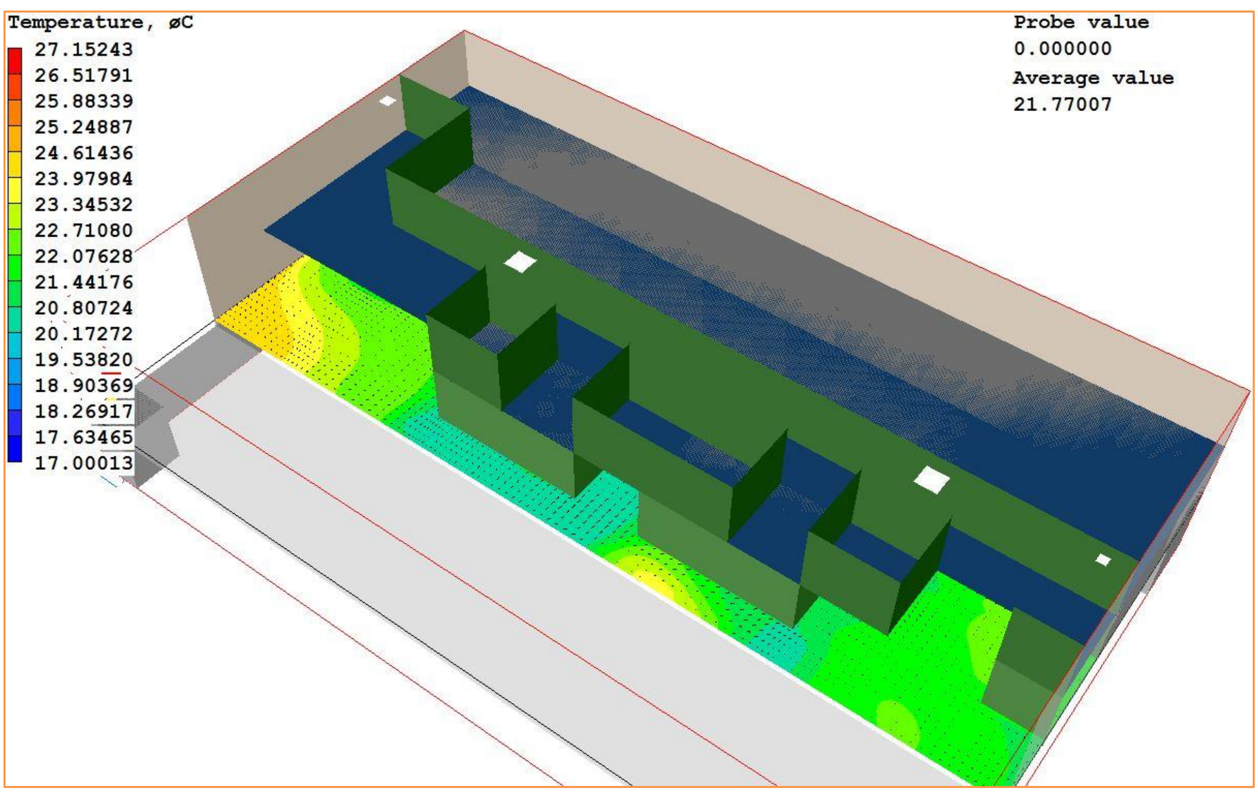
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19.53820
18.90369
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17.00013
Probe value
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Average value
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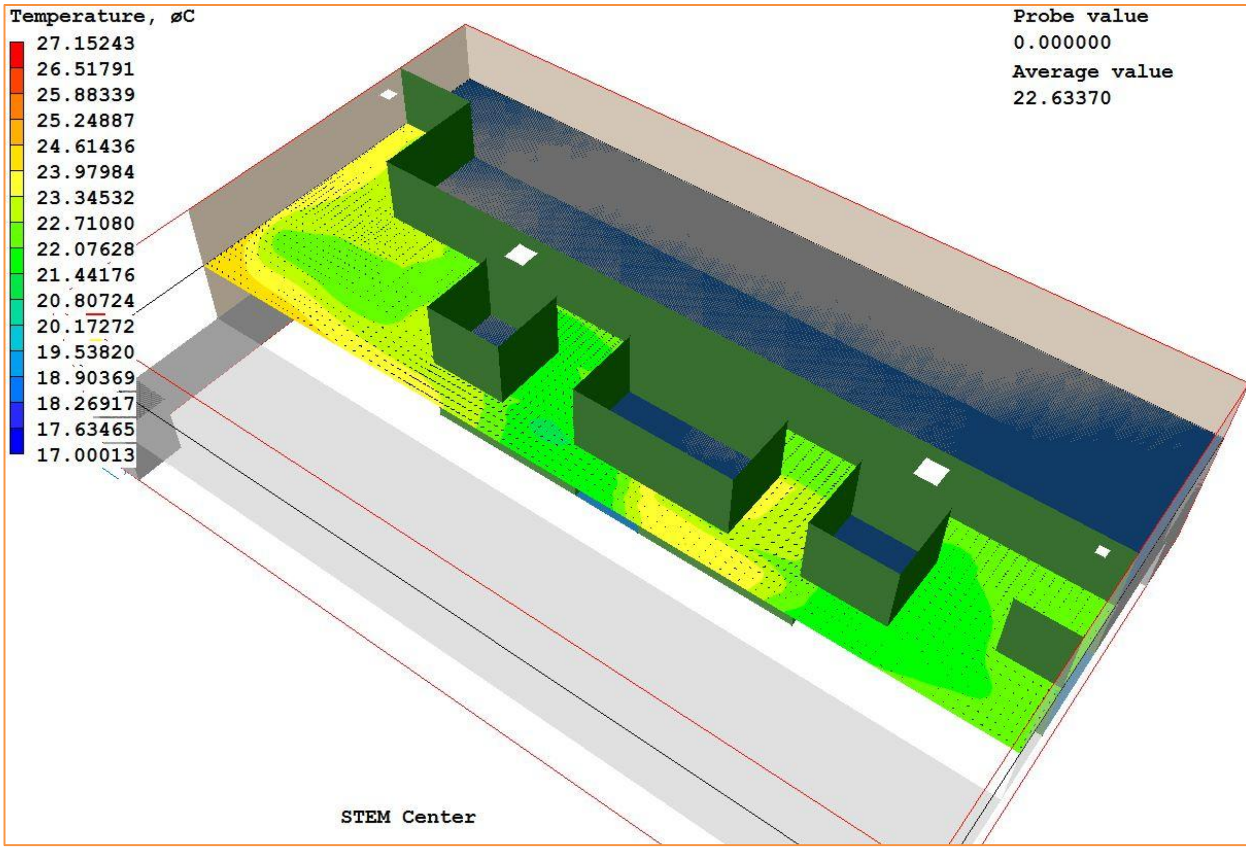
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25.88339
25.24887
24.61436
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23.34532
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21.44176
20.80724
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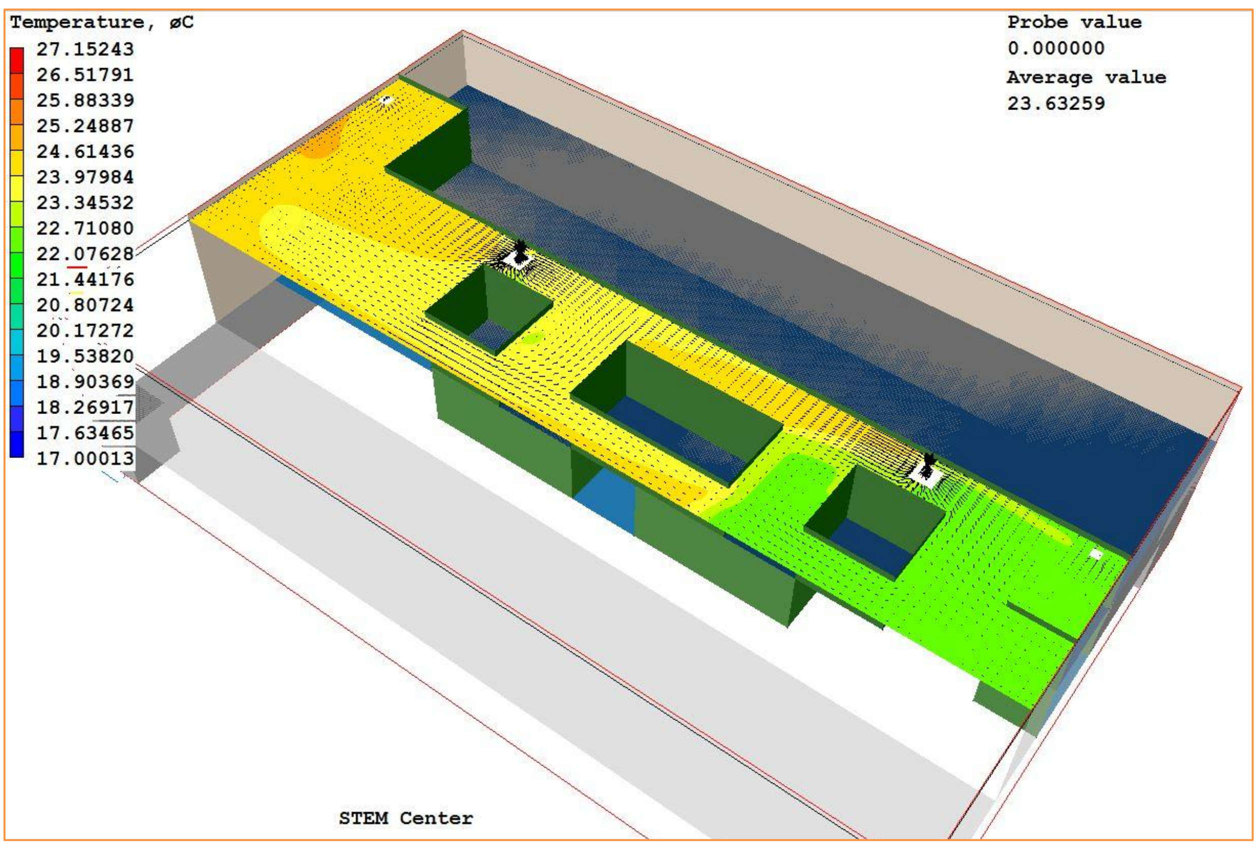
STEM Center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Probe value
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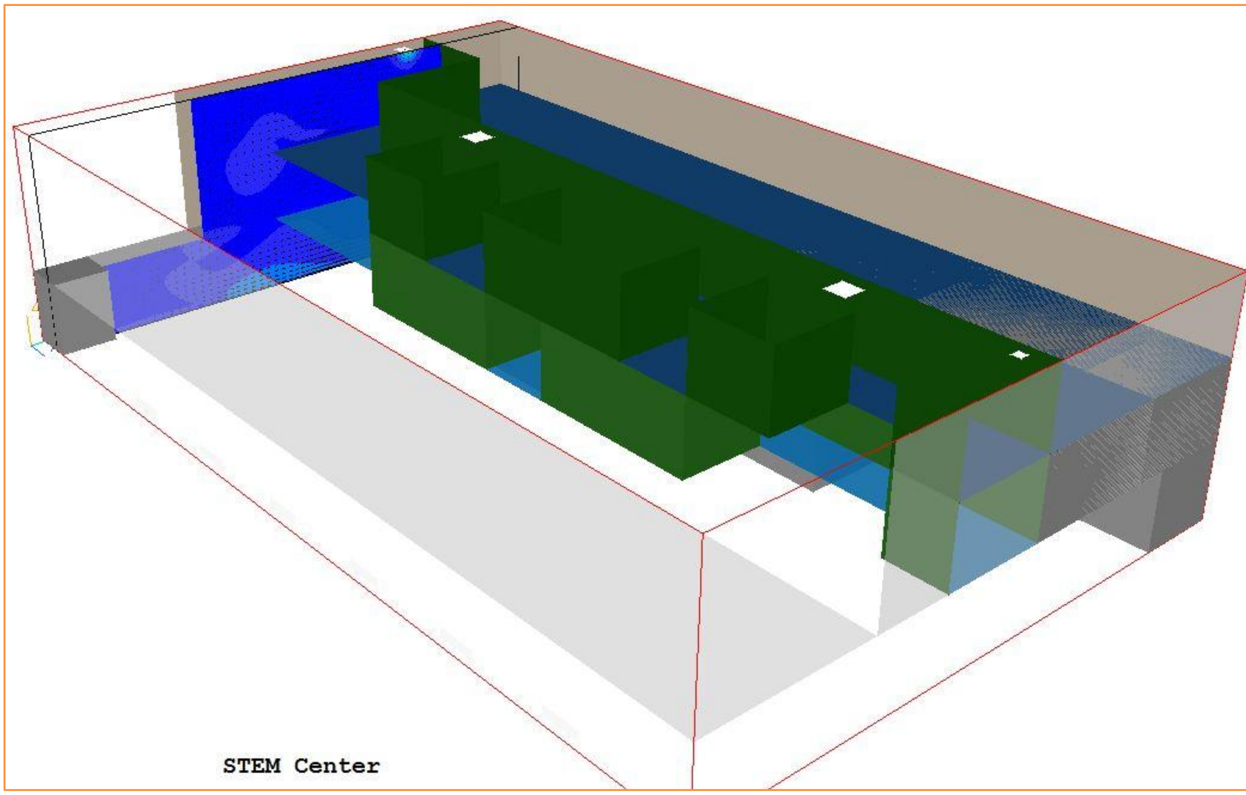
STEM Center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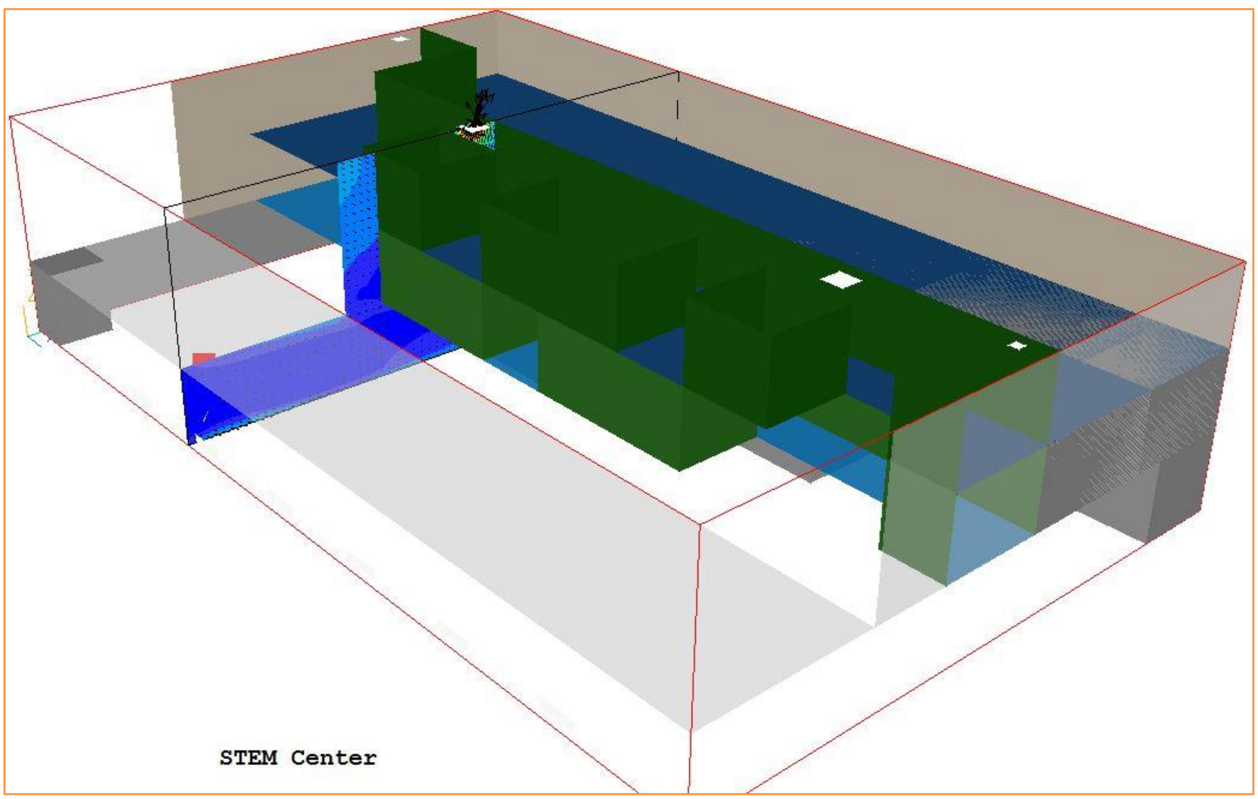
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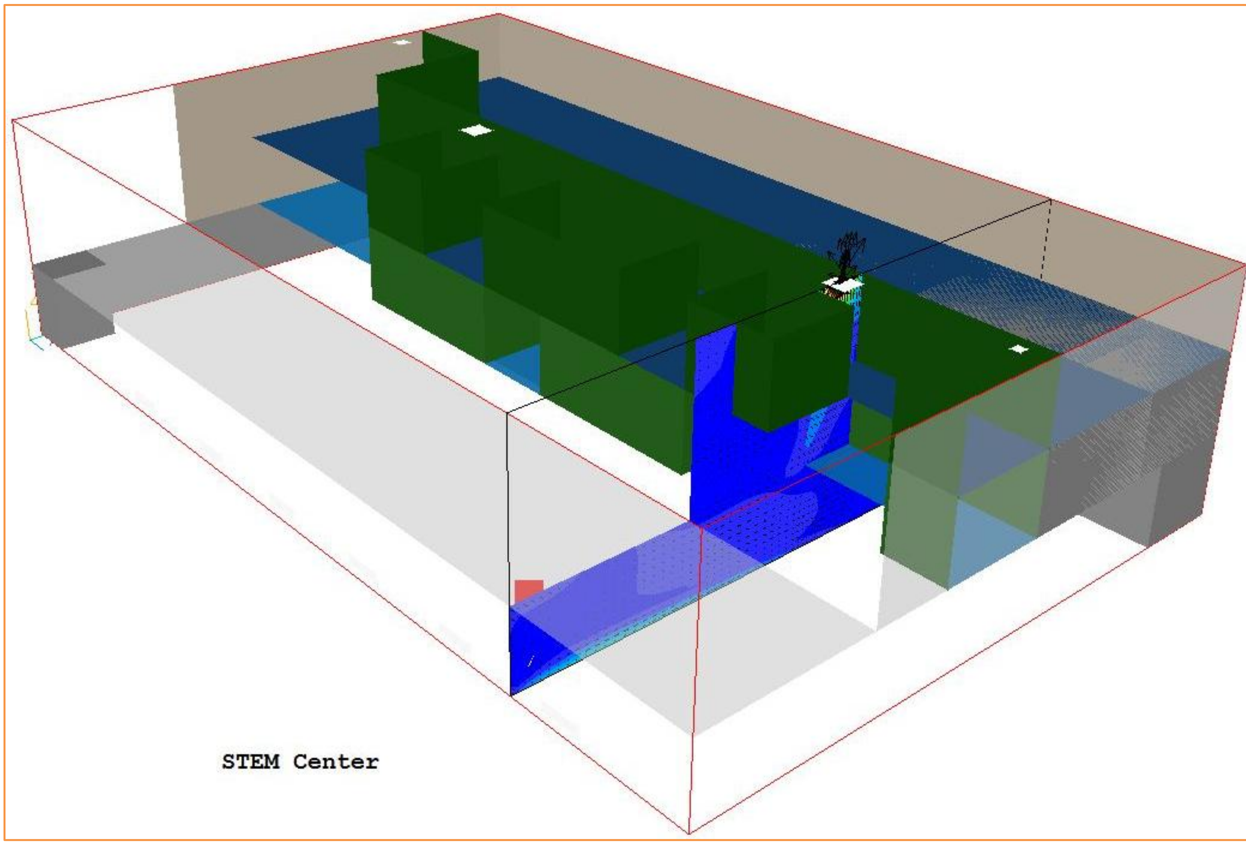
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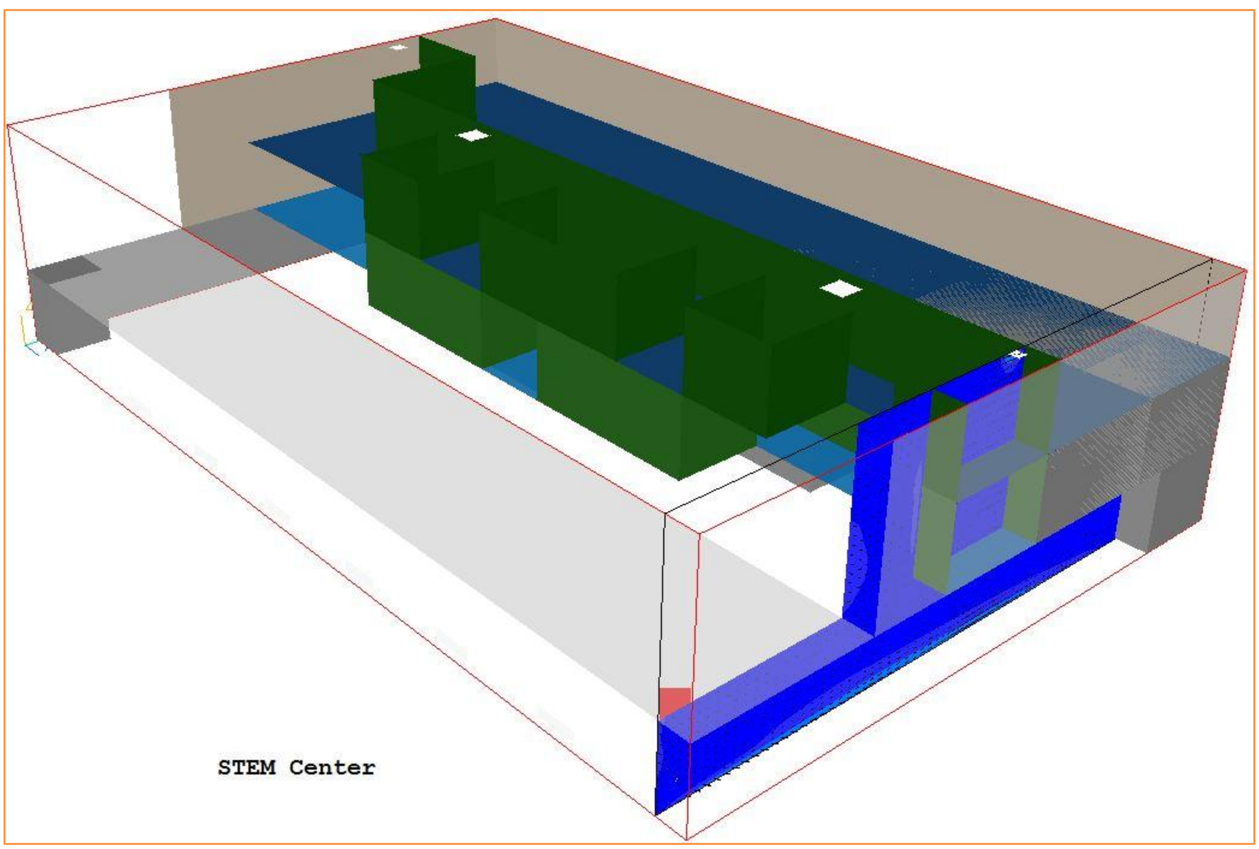
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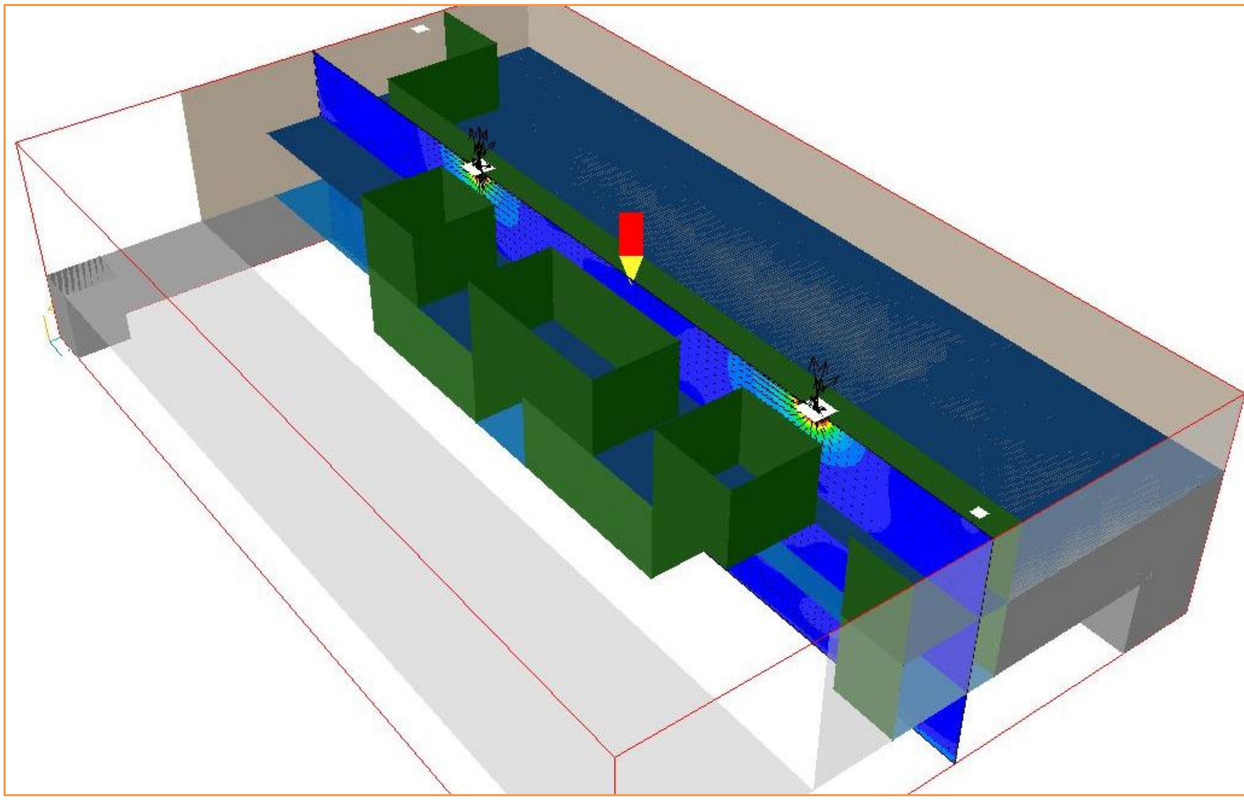

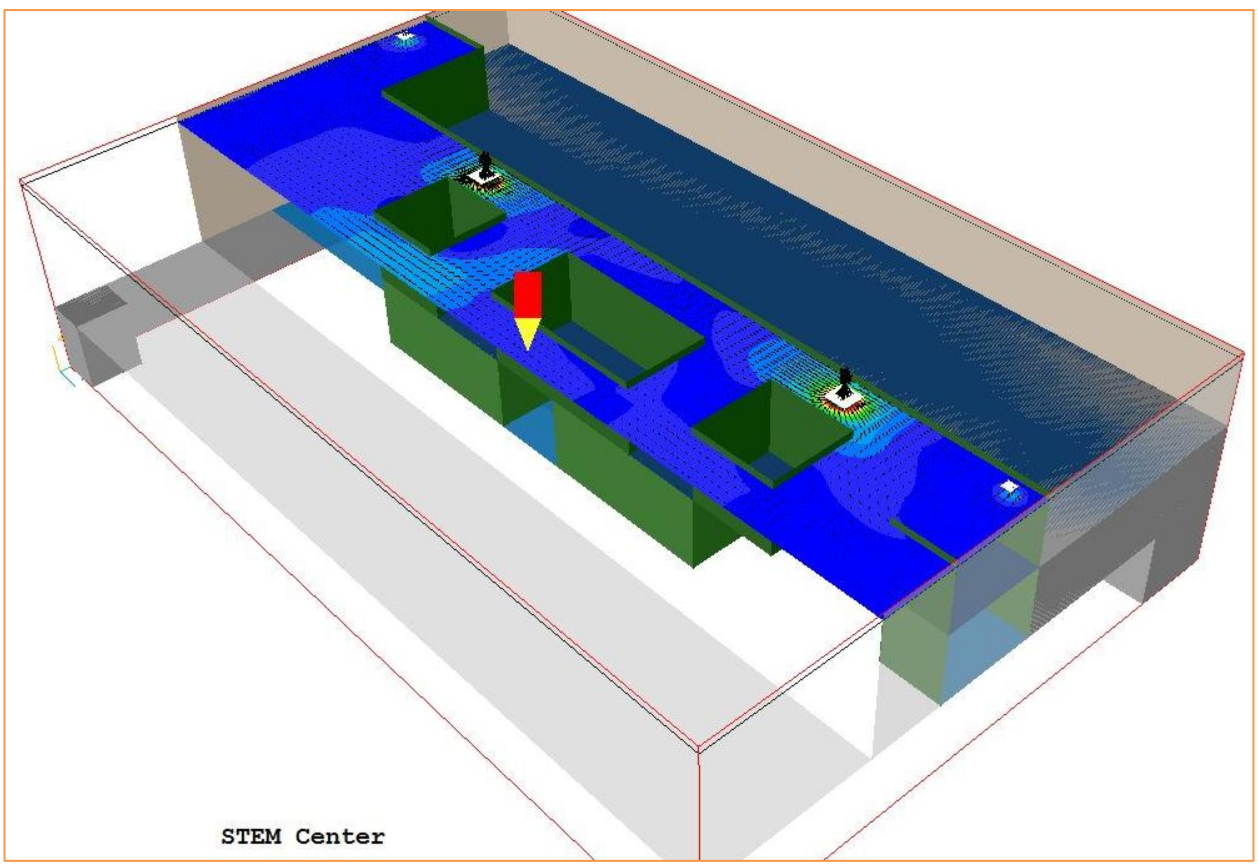
STEM Center

Appendix E: RS Means Pricing Calculations

	Line Number	Description	Unit	Crew	Daily Output	Labor Hours	Bare Material	Bare Labor	Bare Equipment	Bare Total	Total O&P
New AHU	237413103250	10,000 CFM	Each	Q6	0.5	48	13600	2100		15700	**18075**
New Wheel/HX	237213104060	10,000 CFM	Each	Q10	0.9	26.667	11000	1125		12125	**13850**
New Piping	COST VALUES CALCULATED USING RADIANT WORKS PRO SOFTWARE										**20615**
New Manifolds/Fitting											**2880.8**
New Pumps	232123135040	Up to 40 GPM	Each	Q1	2.3	6.957	2950	292		3242	**3690**
										TOTAL	**59110.8**
VAV Units	233616105830	600 CFM	Each	Q9	9	1.778	1150	72.5		1222.5	1362
	233616105840	800 CFM	Each	Q9	7	2.286	1200	93		1293	1469
	233616105850	1000 CFM	Each	Q9	7	2.286	1200	93		1293	1469
	233616105860	1250 CFM	Each	Q9	5	3.2	1375	131		1506	1726
	233616105880	2000 CFM (4)	Each	Q9	3.5	4.571	1475	186		1661	7648
	233616105890	3000 CFM (7)	Each	Q9	3.5	4.571	1475	186		1661	13384
										TOTAL	**27058**
Ductwork	233113130520	200 to 500 lb.	Lb.	Q10	245	0.098	0.6	4.14		4.74	7.06
										TOTAL	**1181.25**

Appendix F: Life Cycle Cost Calculations

Year	2011	2012	2013	2014	2015	2016	2017	2018	2019	2020
Electricity	0.99	0.99	0.99	0.99	1	1	1.01	1.01	1.01	1.01
	$2,752.20	$2,752.20	$2,752.20	$2,752.20	$2,780.00	$2,780.00	$2,807.80	$2,807.80	$2,807.80	$2,807.80
Gas	1.03	1.04	1.06	1.07	1.08	1.09	1.11	1.13	1.13	1.17
	-$920.82	-$929.76	-$947.64	-$956.58	-$965.52	-$974.46	-$992.34	-$1,010.22	-$1,010.22	-$1,045.98
TOTAL	$1,831.38	$1,822.44	$1,804.56	$1,795.62	$1,814.48	$1,805.54	$1,815.46	$1,797.58	$1,797.58	$1,761.82
-$30,871.55	**-$29,040.17**	**-$27,217.73**	**-$25,413.17**	**-$23,617.55**	**-$21,803.07**	**-$19,997.53**	**-$18,182.07**	**-$16,384.49**	**-$14,586.91**	**-$12,825.09**

Year	2021	2022	2023	2024	2025	2026	2027	2028	2029	2030
Electricity	1.01	1.02	1.03	1.04	1.03	1.03	1.03	1.03	1.03	1.03
	$2,807.80	$2,835.60	$2,863.40	$2,891.20	$2,863.40	$2,863.40	$2,863.40	$2,863.40	$2,863.40	$2,863.40
Gas	1.19	1.2	1.25	1.28	1.3	1.33	1.35	1.37	1.39	1.41
	-$1,063.86	-$1,072.80	-$1,117.50	-$1,144.32	-$1,162.20	-$1,189.02	-$1,206.90	-$1,224.78	-$1,242.66	-$1,260.54
Total	$1,743.94	$1,762.80	$1,745.90	$1,746.88	$1,701.20	$1,674.38	$1,656.50	$1,638.62	$1,620.74	$1,602.86
Balance	**-$11,081.15**	**-$9,318.35**	**-$7,572.45**	**-$5,825.57**	**-$4,124.37**	**-$2,449.99**	**-$793.49**	**$845.13**	**$2,465.87**	**$4,068.73**

Year	2031	2032	2033	2034	2035	2036	2037	2038	2039	2040
Electricity	1.03	1.03	1.03	1.03	1.03	1.03	1.04	1.04	1.04	1.04
	$2,863.40	$2,863.40	$2,863.40	$2,863.40	$2,863.40	$2,863.40	$2,891.20	$2,891.20	$2,891.20	$2,891.20
Gas	1.43	1.45	1.48	1.5	1.52	1.55	1.57	1.6	1.62	1.65
	-$1,278.42	-$1,296.30	-$1,323.12	-$1,341.00	-$1,358.88	-$1,385.70	-$1,403.58	-$1,430.40	-$1,448.28	-$1,475.10
Total	$1,584.98	$1,567.10	$1,540.28	$1,522.40	$1,504.52	$1,477.70	$1,487.62	$1,460.80	$1,442.92	$1,416.10
Balance	**$5,653.71**	**$7,220.81**	**$8,761.09**	**$10,283.49**	**$11,788.01**	**$13,265.71**	**$14,753.33**	**$16,214.13**	**$17,657.05**	**$19,073.15**