NORTHEAST USA

INTEGRATED SCIENCES BUILDING

Final Report

Variable Primary Flow Analysis

Thermal Storage Analysis

Solar Photovoltaic Array Analysis

Thermal Storage System Construction Logistical Analysis

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Thesis Abstract

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Integrated Sciences Building
Northeast USA

Project Team

Owner | Information not for Publication
Architect | Diamond+Schmitt Architects, Inc.
Associate Architect | H2L2 Architects & Planners, LLC
General Contractor | Turner Construction Company
MEP Engineer | Crossey Engineering, Ltd.
MEP Engineer | Spotts, Stevens, & McCoy, Inc.
Structural Engineer | Halcrow Yolles Ltd.
Associate Structural Engineer | Keast & Hood Co.
Civil/Landscape | Stantec Consulting Services, Inc.

Mechanical

- (9) Variable Air Volume Air Handler Units
- Heat Recovery Loops with Estimated 29% Energy Savings
- (2) 620-ton Electric Centrifugal Chillers
- (2) 620-ton Induced Draft Single Cell Cooling Towers
- 200psi Campus Steam for Heating & Humidification
- (2) 250-gallon Domestic Hot Water Tanks
- Penthouse Level Reverse Osmosis, Multiple-Purification purified water tank
- Laboratory piping systems for CO_2, Natural Gas, & Liquid Nitrogen

Project Information

Size | 133,847 Square Feet
5 Stories Above Grade
6th-Level Mechanical Penthouse
Partial Basement
Occupancy | Educational & Research Laboratory
Construction Cost | $52.1 million
Construction Schedule | October 2009-July 2011
Delivery Method | Design-Bid-Build

Architecture

- LEED Gold Certification
- 4-Story Bio Wall Air Filtration feature
- 240-Seat Auditorium
- Cutting edge Laboratories & Science Classrooms
- Ground Floor Café
- Recycled Stone Exterior Cladding

Structure

- 83 Reinforced Concrete Columns
- 83 Underground Reinforced Concrete Caissons
- Concrete Shear Walls providing Lateral Support
- Filigree Slab Floor System with Integrated Beams
- Structural Steel Roof & Penthouse Framing
- Elevated Structural Steel Auditorium Sloped Floor

Lighting & Electrical

- 13.2 kV Supplied from Campus Switchgear
- 2500 kVA Main Transformer stepdown to 480Y/277V
- 600 kW, 480V, 3-Phase Diesel Generator
- 5000A, 480Y/277V Main Switchboard
- T8 Linear Fluorescent Lighting
- Dimmable Lighting Controls & Occupancy Sensors
- 59 Lighting Fixture Types

http://www.engr.psu.edu/ae/thesis/portfolios/2011/csp5028/index.html

Executive Summary

This report is intended to explain the results of several investigations into the proposed alternatives systems for the Integrated Sciences Building as a part of the Penn State Architectural Engineering program. The Integrated Sciences Building is an urban University facility that will be home to several scientific research departments in addition to becoming the first LEED Gold certified building at one of the most prestigious education institutions in the northeastern United States.

The original mechanical system uses primarily Variable Air Volume (VAV) systems with terminal to meet the thermal loads of the office, educational, and research spaces within the building, with Constant Volume air systems serving the 240-seat auditorium and atrium. The laboratory VAV systems supply 100% Outside Air to guarantee acceptable air quality and to purge the building of any contaminants generated during experiments. Two 620-ton centrifugal chillers provide chilled water to meet cooling demands and a district steam system provides heating capacity of up to 20,000 pounds per hour.

The proposed alternatives include a Variable Primary Flow (VPF) chilled water pumping system in place of the existing Primary/Secondary system. After performing parametric energy analysis on a number of different pumping configurations, it was determined that the VPF system has the potential to save over 32,000 kWh annually, which translates to 3.97% of the total chilled water plant energy consumption. Over the 30-year life cycle, this comes to a net benefit of over $40,000.

The next alternative study performed was Thermal Energy Storage (TES) system which was designed to flatten the cooling load profile of the Integrates Sciences Building in attempts to reduce the monthly billable electrical demand. Calculations showed that annual electrical demand charges would be reduced by over $10,500 annually, but the penalty of ice production for thermal storage resulted in an annual increase in electrical consumption bills by over $9,000 annually. In the end, due to the capital cost and reduced annual maintenance estimates, the life cycle costs showed that a TES system would produce a 30-year net benefit of over $450,000.

A solar photovoltaic energy production system was also designed to take advantage of the renewable energy production capabilities. The system, sized at 80kW is projected to produce over 99,000kWh of electricity per year. With state and government financial grants and incentives, the payback period on the system comes to only five years.

Finally, a Construction and Logistics study was performed to estimate the effect the thermal storage system construction would have on the project schedule. Using RSMeans scheduling data, it was determined that the thermal storage system could be installed concurrently with other construction tanks, minimally effecting the overall schedule. An extra 375 man hours would be added to the schedule and the entire thermal storage tank and piping system could optimistically be installed in little over four weeks.

Building Summary

The Integrates Sciences Building is a 133,000 square foot building located in an urban Northeastern United States environment. The building will be the new home of the Department of Biosciences at a major University with 39 laboratories for Biomedical Engineering, Biology and Chemistry, and a Fossil Preparation Laboratory. It will also incorporate classrooms, faculty offices, a ground floor café, 240-seat auditorium, and a Career Development Center. The design is by the renowned firm, Diamond + Schmitt Architects, Inc. and is intended to reinvigorate the lively urban campus with excellent modern and sustainable architecture. Besides being an environmentally responsible building, its most prominent features are the grand 5-story atrium with skylights and natural daylighting, 5-story Bio Wall and a modern spiral staircase. The exterior appearance of the building is truly unique featuring a facade made of recycled stone panel cladding with metal accents as well as varied geometries like the elliptical rotunda lounge and the saw-tooth atrium skylight window pattern. The many unique features of the building combine to create a sophisticated new building with a modern presence.

The Integrated Sciences Building was initially intended to be the University's first building certified by the Leadership in Energy and Environmental Design (LEED) program which is administered by the United States Green Building Council (USGBC). The original building design was slated to be certified as LEED Silver, but the effort has been upgraded and the new target is LEED Gold status on the LEED v2.2 scale. There are 43 LEED points targeted including water use reduction, optimized energy performance, use of regional and recycled materials, construction waste management and enhanced Indoor Environmental Quality, among others.

One of the goals of the building was to set a new standard for architectural and sustainable design that will “spark students and faculty to engage in an interactive learning environment.” Front and center in this effort is the fact that the Integrated Sciences Building is going to be the first United States University building with a living, breathing Bio Wall. The Bio Wall is a 5-story living wall with live plant species that is intended to offer improved energy efficiency and indoor air quality. The wall is expected to cool the large atrium during summer months and act as a humidifier during the winter. It will also remove carbon dioxide from the indoor atmosphere and filter Volatile Organic Compounds as some return air passes through on its way back to an air handler. The Bio Wall is a visual representation of all of the sustainable features of the building, from the enhanced mechanical system, of which it aids by enhancing air quality, to the renewable materials and growth of technology that was used throughout the building.

Mechanical System Description

Design Objectives & Requirements

The Heating, Ventilation, and Air Conditioning (HVAC) system for the Integrated Sciences Building was designed to accomplish many different objectives. Obviously, comfort of the human occupants based on temperature and humidity were of foremost concern. Another main concern was to provide ventilation air of good quality, which is a more difficult task to achieve in buildings with science laboratories such as the Integrated Sciences Building. The system was designed to meet or exceed ASHRAE Standard 62.1 as well as ASHRAE Standard 90.1. Energy efficiency was another focus of utmost concern in order to reduce operating costs and minimize the carbon footprint of the building.

The building owner has an outline of minimum building performance requirements for all of its many construction projects. This building was required to follow those requirements, which apply to all areas of design and construction, including mechanical systems. In addition to those requirements, the owner wanted the Integrated Sciences Building to excel as an icon of energy efficiency. The original goal of the building was to become LEED Silver certified, but through the design phase, it is expected to reach LEED Gold status.

Finally, since the building is ultimately owned by a private educational organization, construction and operating costs are of great concern as well. The energy efficiency and comfort objectives were to be met in ways that made sense in a financial manner. A large emphasis in selecting systems was the ability to recoup the original installation and construction costs through energy and operational savings.

Major Mechanical Equipment

The mechanical systems that serve the HVAC needs of the Integrated Sciences Building are very complex and designed to be efficient in pursuit of points toward LEED Certification. There are nine air handling units that serve the building. Three of the units are constant volume, terminal reheat systems that serve the electrical and data closets, auditorium, and atrium as well as providing adequate air for pressurization of the building. These systems do have reset controls to modify the supply air flow during unoccupied operation hours, but they do not modulate air flow according to population or room temperature during occupied hours. The other air handlers are all Variable Air Volume (VAV) Systems with hydronic terminal reheat coils that serve classrooms, offices, teaching laboratories, and research laboratories. The laboratory air handlers supply 100% outdoor air to the spaces they serve in order to provide adequate ventilation to the occupants and the purge the building of any contaminants that may result from the experiments in those spaces. The laboratories are negatively pressurized relative to the remainder of the building to ensure that no contaminants enter the other occupied areas. Laboratory spaces are also equipped with VAV fume hood controls to limit the exhaust of the hoods and save energy when possible.

The thermal loads of the building are served by a chiller plant and purchased steam system. The chiller plant consists of two 620-ton two-stage centrifugal, water-cooled chillers with a coefficient of performance (COP) of 5.56 each. Heat is rejected via condenser water that is cooled by two 620-ton cooling towers. The purchased steam enters the building at 200 psi and maximum rate of 15,000 lbs/hr.

The steam is used in four heat exchangers to make hot water to serve heating loads in the building's air handlers and VAV hydronic reheat coils. Steam is also used to provide domestic hot water. Natural gas is used in the building, but not as a significant energy source. The natural gas consumption is limited to bench and fume hood use in the laboratories for scientific experiments.

Air Handlers

The air handling units (AHUs) used for the Integrated Sciences Building are each of unique size and capacity, designed to match the spaces they serve. In Table 1, a list of pertinent air handler information is listed.

AHU Tag	Supply Fan Capacity [CFM]	Minimum Outside Air [CFM]	Return Fan Capacity [CFM]	Cooling Coil Capacity [MBH]	Heating Coil Capacity [MBH]	Glycol Heat Recovery Coil Capacity [MBH]
AHU-1	6000	3400	N/A	259	206	N/A
AHU-2	19000	9500	9500	994	631.4	N/A
AHU-3	16000	7900	8100	1190	1244	N/A
AHU-4	20000	10000	9600	924	864	N/A
AHU-5	42500	100%	N/A	3123	2754	1261.4
AHU-6	28,000	100%	N/A	2046	1878	831
AHU-7	27000	100%	N/A	1963.2	1835	801.3
AHU-8	20000	100%	N/A	1448.4	1338	593.6
AHU-9	12000	2000	12000	942	N/A	N/A

Table 1 – Air Handler Information

Note that AHU-9 has no heating coils and is for cooling only. Air AHUs 5-9 are 100% outdoor air units and have glycol heat recovery coils as an energy saving measure. This heat recovery system harvests heat energy from the exhaust streams and redistributes it to the incoming air stream via a pumped water and glycol solution.

Chilled Water Plant

Chiller information is listed in Table 2Table 2 – Original ISB Chiller Schedule. Both chillers are water-cooled centrifugal chillers manufactured by York International, the HVAC equipment subsidiary of Johnson Controls Incorporated. The chillers are located in the mechanical penthouse at the top level of the building. The condenser water is cooled by the Cooling Towers, listed in the Table 3. The cooling towers and chiller equipment are conveniently located very closely. The cooling towers are direct, induced draft cooling towers manufactured by Baltimore Aircoil Company.

Chiller Schedule								
Chiller	Type	Nominal Capacity [Tons]	kW/Ton	Model	Two-Pass Evaporator		Two-Pass Condenser	
					EWT [°F]	LWT [°F]	EWT [°F]	LWT [°F]
CH-1,2	Water Cooled Centrifugal	620	0.632	York YKGQEVP8-CTG	56	42	85	97

Table 2 – Original ISB Chiller Schedule

Cooling Tower Schedule							
Cooling Tower	Nominal Capacity [Tons]	Water Flow [gpm]	Wet Bulb EAT [°F]	EWT [°F]	LWT [°F]	Fan Capacity [CFM]	Model
CT-1,2	620	15550	78	97	85	149,090	Baltimore Aircoil Company 3604C

Table 3 – Original ISB Cooling Tower Schedule

Hot Water Heat Exchangers

Table 4 lists information about the four heat exchangers that use district steam to heat the building. Two of the heat exchangers heat a 30% glycol mixture and the other two use only hot water.

Steam Heat Exchanger Schedule								
	Process - Side 1				Service - Side 2			Model
	Fluid	EWT [°F]	LWT [°F]	GPM	Fluid	Pressure [psi]	Capacity [lbs/hr]	
HE-1	30% P.G.	150	180	340	Steam	12	5105	Taco E12210-S
HE-2	30% P.G.	150	180	340	Steam	12	5105	Taco E12210-S
HE-3	Water	150	180	290	Steam	12	4474	Taco E10208-S
HE-4	Water	150	180	290	Steam	12	4474	Taco E10208-S

Table 4 - Steam Heat Exchanger information.

Major Pumps

All pumps listed below, Table 5, are from the manufacturer Bell & Gossett. Glycol mixing tank pumps are not listed but are shown on the schematics. They were not selected by the design engineer and are a part of the glycol mixing tank subcontract package.

Pump Tag	Service	GPM	Head [ft]	Motor		Notes
				HP	RPM	
P-1	Primary Chilled Water Pump (Duty)	1100	50	20	1750	-
P-2	Primary Chilled Water Pump (Duty)	1100	50	20	1750	-
P-3	Primary Chilled Water Pump (Standby)	1100	50	20	1750	-
P-4	Condenser Water Pump (Duty)	1550	90	50	1770	-
P-5	Condenser Water Pump (Duty)	1550	90	50	1770	-
P-6	Condenser Water Pump (Standby)	1550	90	50	1770	-
P-7	Secondary Chilled Water Pump (Duty)	1100	76	40	1150	VSD
P-8	Secondary Chilled Water Pump (Duty)	1100	76	40	1150	VSD
P-9	Secondary Chilled Water Pump (Standby)	1100	76	40	1150	VSD
P-10	Glycol Heating Pump (Duty)	340	100	20	1750	VSD
P-11	Glycol Heating Pump (Duty)	340	100	20	1750	VSD
P-12	Hot Water Heating Pump (Duty)	290	95	15	1750	VSD
P-13	Hot Water Heating Pump (Standby)	290	95	15	1750	VSD
P-14	AHU-1 Glycol Circulating Pump	14.3	30	0.4	3250	-
P-15	AHU-2 Glycol Circulating Pump	43.8	30	1	1750	-
P-16	AHU-3 Glycol Circulating Pump	86.4	30	2	1150	-
P-17	AHU-4 Glycol Circulating Pump	60	30	1.5	1750	-
P-18	AHU-5 Glycol Circulating Pump	190	30	3	1150	-
P-19	AHU-6 Glycol Circulating Pump	129	30	2	1750	-
P-20	AHU-7 Glycol Circulating Pump	125	30	2	1750	-
P-21	AHU-8 Glycol Circulating Pump	92	30	1.5	1750	-
P-22	Primary Glycol Heat Recovery Pump (Duty)	300	90	15	1750	VSD
P-23	Primary Glycol Heat Recovery Pump (Standby)	300	90	15	1750	VSD
P-24	Secondary Glycol Heat Recovery Pump (Duty)	100	60	5	1750	-
P-25	Secondary Glycol Heat Recovery Pump (Standby)	100	60	5	1750	-
P-26	AHU-5 Heat Recovery Booster Pump	99	30	2	1750	-
P-27	AHU-6 Heat Recovery Booster Pump	65	30	1.5	1750	-
P-28	AHU-7 Heat Recovery Booster Pump	64	30	1.5	1750	-
P-29	AHU-8 Heat Recovery Booster Pump	47	30	1.5	1750	-

Table 5 – ISB Pump Schedule

Mechanical Systems Operation

Air Side Systems

Air side systems can be separated into two main types for the Integrated Sciences Building: those that serve laboratory spaces, and those that do not.

The non-laboratory air handlers are conventional variable air volume (VAV) systems with terminal reheat. All supply and return fans are equipped with variable speed drives. System startup is based on the building automatic system time of day schedule. Upon startup, the outside air intake and exhaust air dampers open. When the variable speed drives are activated and proof of flow is confirmed, the fans speed up and bring the system to modulating control. In cooling mode, zone controllers send requests to the air handler to modulate the supply air temperature by means of adjusting the cooling coil control valve. Airside free cooling is activated when all heating commands are zero and the outside air temperature is less than 65°F.

The laboratory air handling units do not have return fans, but instead exhaust fans which send air directly to the outdoors. These exhaust fans are equipped with glycol heat recovery coils that preheat new supply air that is 100% outdoor air. All fans on these systems are equipped with variable speed drives so that the amount of air supplied to and extracted from the spaces can be modulated based on the minimum air change rates for laboratories for occupied and unoccupied hours. These schedules are assigned by the Building Automation System upon system setup. More information on the glycol heat recovery system, as well as a schematic can be seen in the water-side portion of this section. The temperature control is achieved in the spaces by conventional variable air volume coil controls such as those described for the non-laboratory airside systems. Table 6 gives details on operation and features of the air handlers.

Unit	Mixing Box	30% Filter	90% Filter	Heat Recovery Coil	Glycol Heating Cool	Chilled Water Cooling Coil	Reheat Coil	Steam Injection Humidifier	Supply Fan	Return Fan
1	X	X	X		X	X	X		VAV	VAV
2	X	X	X		X	X			VAV	VAV
3	X	X	X		X	X			VAV	VAV
4	X	X	X		X	X			VAV	VAV
5		X	X	X	X	X		X	VAV	
6		X	X	X	X	X		X	VAV	
7		X	X	X	X	X		X	VAV	
8		X	X	X	X	X		X	VAV	
9	X	X	X		X	X			CAV	

Table 6 – Air Handler Details

Water Side Systems

Chilled Water System

The chilled water system is a Primary-Secondary system. There are two 620-ton water cooled, centrifugal chillers. The condenser water system includes two 620-ton induced draft cooling towers. The condenser water pumps and the primary chilled water pumps are operate at constant volume while the secondary chilled water pumps are equipped with variable speed drives. The chiller water system initiates when the outside air temperature rises above 55°F. There are three condenser water pumps, with one as a standby and two which operate in alternation. The cooling tower fans are controlled by the cooling tower leaving water temperature. They activate and increase in speed via variable frequency drive when the leaving water temperature rises above the set point. The cooling tower leaving water set point is either 65°F or the outside air wet bulb temperature plus the 12°F cooling tower range, whichever is higher. Primary chilled water pumps cycle to meet the load of the chilled water system and to maintain an acceptable pressure drop between supply and return.

Steam & Hot Water

The heating needs of the building are served by a 200 psi distributed steam system. There are two pressure reducing stations from 200 psi to 60 psi and then 60psi to 12 psi. The 12 psi steam goes to four heat exchangers. Two of the heat exchanges are steam to glycol mixture while the other two are steam to hot water. The heated glycol mixture is distributed to the air handler heating coils. The hot water is distributed to the reheat coils. Some steam is also sent to direct injection humidifiers in the laboratory air handlers as well as domestic hot water heating tanks. Condensate waste heat is utilized in the glycol heat recovery system.

The glycol heating system consists of a 30% glycol-water mixture that enters the heat exchangers are 150°F and leaves at 180°F. The mixture is heated in steam heat exchangers 1 and 2 and sent to the air handler heating coils. The flow through the heating coils is controlled by a temperature sensor of the air downstream of the heating coil in the air handler. Pumps 10 and 11 provide flow through the system and each are equipped with a variable frequency drives. Pump flow is controlled to maintain a differential in pressure between the supply and return headers. Flow based on capacity is controlled by increasing pump energy to meet the set point and pressure differential set point. Overall system heat consumption is measured using temperature sensors upstream and downstream of the heat exchangers and the flow measure quantity of water through the heat exchangers.

The hot water heating system that serves the terminal reheat coils operates in much the same way as the glycol heating system except that water is distributed to the VAV box coils rather than air handlers. Pumps 12 and 13 provide pumping for the system and are alternated to equalize run times. One pump is designated the “lead” pump and the other the “standby” pump. These pumps are also controlled in the same way as the glycol heating system.

Glycol Heat Recovery

The glycol heat recovery system captures energy from the exhaust air streams of the laboratory and its fume hood exhaust. Pumps 22 and 23 are the primary pumps for the system, and operate at constant volume. Pimps 24 and 25 integrate a steam condensate heat recovery loop that uses waste heat from the steam system condensate to help with energy recovery. Both sets of pumps alternate such that one is the "lead" and one is "standby" in order to equalize run times. Booster pumps circulate water though the glycol heat recovery heating coils in the laboratory air handlers. A mixing valve modulates flow through the coil and the bypass based on air temperature downstream of the coil and the leaving glycol mixture temperature.

Energy Sources & Rates

The fuel costs used for this analysis were the same used by the design engineer who performed the LEED energy analysis for the building. Since the building is under construction at the time of this report, these energy rates will likely change by the time of completion. For example, upon speaking with the building owner's Facilities Management, it was learned that electricity rates will be increasing between 20-40% in the year after construction is scheduled for completion. Nonetheless, energy prices that were used for the building consisted of electricity, natural gas, and purchased district steam. These rates are listed in Table 7, Table 8, and Table 9 below.

Electricity (PECO - HT)	
Customer Monthly Charge	$291.43
Charge per kWh [For first 150 hours]	$0.0635
Charge per kWh [After 150 hours - Up to 7,500,000 kWh]	$0.0442
Charge per additional kWh	$0.0253
Demand Charge per kW	$8.79

Table 7 – PECO HT Electricity Rate

Natural Gas (Philadelphia Gas Works)	
Customer Monthly Charge	$18.00
Cost per Therm	$1.22

Table 8 – Philadelphia Gas Works Rate

District Steam (Trigen Rate S)	
Winter (October - May)	
Consumption: Charge per first 100 Mlbs	$29.08
Consumption: Charge per Additional Mlbs	$28.17
Demand: Charge per first 300 lb/hr	$1.84
Demand: Charge per next 39,700 lb/hr	$1.24
Demand: Chare per Additional lbs/hr	$1.09
Summer (June-September)	
Consumption: Charge per first 100 Mlbs	$27.78
Consumption: Charge per Additional Mlbs	$26.87
Demand: Charge per lb/hr	$0.00

Table 9 – Trigen State Rate S

ASHRAE Standard 62.1 Compliance Evaluation Summary

The ventilation systems in the Integrated Sciences Building comply very closely with ASHRAE Standard 62.1-2007. The only part of Section 5, Systems and Equipment, which the HVAC systems do not specifically follow, is the bird screen mesh size. Table 10 shows which requirements from ASHRAE Standard 62.1 - Section 5 are met, and which are not. This breach in compliance is relatively minor in the grand scope of the standard and can easily be resolved. More details on all ASHRAE 62.1 Compliance evaluation sections is available in Technical Report I. One area in which the system design exceeds this section is the filtering equipment. The MERV 7 and MERV 13 filters which will be installed ensure that the quality of the supply air is very good, which is appropriate for a building with LEED Gold status.

ASHRAE 62.1 Compliance Evaluation		
Section	Topic	Compliance
5.1	Natural Ventilation	N/A
5.2	Ventilation Air Distribution	Yes
5.3	Exhaust Duct Location	Yes
5.4	Ventilation System Controls	Yes
5.5	Airstream Surfaces	Yes
5.6	Outdoor Air Leaks	No
5.7	Local Capture of Contaminants	Yes
5.8	Combustion Air	Yes
5.9	Particulate Matter Removal	Yes
5.10	Dehumidification Systems	Yes
5.11	Drain Pans	Yes
5.12	Finned-Tube Coils and Heat Exchangers	Yes
5.13	Humidifiers and Water Spray Systems	Yes
5.14	Access for Inspection, Cleaning & Maintenance	Yes
5.15	Building Envelope and Interior Surfaces	Yes
5.16	Buildings with Attached Parking Garages	N/A
5.17	Air Classification and Recirculation	Yes
5.18	Requirements for Buildiings Containing ETS Areas and ETS-Free Areas	Yes

Table 10 – ASHRAE Standard 62.1 Section 5 Compliance Evaluation Checklist

The building also exceeds all minimum outside air requirements for the ventilation rate calculation procedure outlined in Section 6 of ASHRAE Standard 62.1-2007. The air handlers all have minimum outdoor air intake settings that exceed all requirements based on the building occupancy and space types. Since this building will be used for scientific research and therefore has the potential to hold many different contaminants, the large amounts of outdoor air will enhance the ability of the mechanical systems to purge the building of any harmful substances. Also, since the air handlers serving the laboratory spaces use 100% outdoor air, they guarantee that the contaminants released in these spaces will enter other spaces in the building. Air flow rates of the air handlers, as well as the ASHRAE minimum air flow requirements, are shown in Table 11.

Unit	Calculated Outdoor Air Required (62.1)	Maximum Design Supply Air	Minimum Design Outdoor Air	ASHRAE 62.1 Compliance
AHU-1	2257	6000	3400	Yes
AHU-2	2986	19000	9500	Yes
AHU-3	2305	16000	7800	Yes
AHU-4	2517	20000	6900	Yes
AHU-5	5206	42500	100% OA	Yes
AHU-6	4781	28000	100% OA	Yes
AHU-7	2466	27000	100% OA	Yes
AHU-8	5350	20000	100% OA	Yes
AHU-9	1030	12000	1200	Yes

Table 11 – ASHRAE Standard 62.1 Section 6 Compliance Evaluation Checklist

ASHRAE Standard 90.1 Compliance Evaluation Summary

Due to the governing characteristics of the Integrated Sciences Building, the Prescriptive Building Envelop Option was used to determine compliance with ASHRAE Standard 90.1-2007. The building is striving for LEED Gold certifications, and efforts to make the building as energy efficient as possible exceed the expectations of ASHRAE Standard 90.1-2007. The building complies very closely with this standard. A simplified outline of the compliance of different Sections of ASHRAE Standard 90.1-2007 is shown in Table 12 and a more detailed examination of compliance can be found in Technical Report I.

The only portions of ASHRAE Standard 90.1-2007 that are not specifically obeyed are very miniscule details. This includes one insulation thickness on a table that includes requirements for many pipe sizes, and the voltage drop allowance for 480V electrical feeders. However, the building is still under construction, and the enhanced commissioning and construction process, as well as the LEED Accredited professionals on the team are likely to eliminate these issues.

ASHRAE Standard 90.1 Compliance Evaluation		
Section	Topic	Compliance
5	General	Yes
5.4.3.1	Air Leakage	Yes
5.5	Prescriptive Building Envelope Option	Yes
6	HVAC Systems	Yes
6.5.1	Economizers	Yes
6.5.2	Simultaneous Heating and Cooling Limitation	Yes
6.5.3	Air System Design & Control	Yes
6.5.4	Hydronic System Design and Control	Yes
6.5.5	Heat Rejection Equipment	Yes
6.5.6	Energy Recover	Yes
6.5.7	Exhaust Hoods	Yes
6.7	Submittals	Yes
7	Service Water Heating	Yes
8	Power	No
9	Lighting	Yes
10	Electric Motor Efficiency	Yes

Table 12 – ASHRAE Standard 90.1 Compliance Evaluation Checklist

LEED Rating Analysis

The Leadership in Energy and Environmental Design (LEED®) program was created by the United States Green Building Council in order to help building owners and design teams realize the importance of energy efficient and environmentally friendly construction practices. LEED has two primary categories, Energy and Atmosphere (EA) and Indoor Environmental Quality (IEQ), which apply directly to mechanical building systems. Technical Report III discusses the LEED points for the Energy and Atmosphere and Indoor Environmental Quality sections of the rating guidelines in detail. The Integrated Sciences Building used LEED version 2.2 when evaluating the certification rating of the building, and had a primary goal of LEED Silver certification. However, as the building makes progress in construction, it has been realized that the building will likely receive LEED Gold certification.

Many LEED points are awarded for areas of engineering design other than mechanical systems, such as advanced commissioning. Appendix A– Integrated Sciences Building LEED Scorecard shows a breakdown of points that are projected to be awarded according to the LEED analysis engineer-of-record for the Integrated Sciences Building Project. Note that this table does not follow the newest scale, LEED v3. The engineer on the project evaluated the building with the LEED v2.2 scale. However, since this scale will be the one used to base any LEED certification on, it provides a good evaluation of the building.

Mechanical System Cost

The total cost of the plumbing and mechanical system for the Integrated Sciences Building is estimated at $ 11,940,000. Due to the nature of the construction schedule and confidentiality of the pricing, a more detailed breakdown of these costs was not available. This price includes design and construction of the plumbing and mechanical systems. When compared to the overall building construction estimate of $51,100,000, this price is roughly 23% of the total building cost and yields a cost of about $86.52/ft^2. Subtracting the mechanical and electrical space allotted in the basement, shafts, and penthouse, the system cost is about $140.47/ft^2.

Annual Energy Consumption

The total annual energy consumption by source is shown below in Table 13 along with the total energy cost by source along with a comparison of the energy usage and costs as presented by the design engineer's energy analysis. As discussed in Technical Report II, error in the energy analysis could be a result of incorrect fume hood exhaust modeling as well as error in modeling the unusual building pressurization methods.

	TRACE 700 Model		Design Model		ASHRAE 90.1-2004 Baseline Model	
	Annual Consumption	Annual Cost	Annual Consumption	Annual Cost	Annual Consumption	Annual Cost
Electricity	3,249,838 kwh	$257,543.00	4,984,300 kwh	$367,933.00	5020250 kwh	$ 369,182.00
Purchased Steam	3,565,486 kBtu	$ 76,765.68	5,132,000 kBtu	$110,637.00	12939000 kBtu	$ 278,973.00
Natural Gas	23,706 Therms	$ 28,921.00	23,706 Therms	$ 28,921.00	23,706 Therms	$ 28,921.00
Total Annual Cost	-	$363,229.68	-	$507,491.00	-	$ 677,076.00
Annual Cost/ft^2	-	$ 2.63	-	$ 3.68	-	$ 4.91

Table 13 – Annual Energy Consumption Calculation comparison for TRACE Model, Design Engineer, and ASHRAE 90.1 Baseline.

All energy analysis performed for the proposed design changes will be based on the TRACE 700 energy model which was constructed for the purpose of the Thesis Technical Reports. Although the discrepancy in results is very unfortunate and undesirable, it was concluded that the best way to compare the original system to any redesigns would be to use the most complete set of information. Because the design engineer's exact energy model inputs were not available, the TRACE 700 model which was constructed by the student was the most complete energy model available. Thus, it was used as a basis for comparison to justify any modifications.

Overall System Evaluation

One of the goals of the Integrated Sciences Building was to establish the owner, which in this case is an urban University organization, as a leader in sustainable construction and technology. There are many factors that play a role in determining the success of a building's "sustainability" and mechanical systems are major factor in that evaluation. In many cases, the LEED rating systems is used to quantify some of the sustainable features of a building. Whether or not that LEED is a good measure for determining sustainability can be a matter of opinion.

There are some features that immediately provide evidence of a responsible design that was intended to reduce energy consumption. One of the first of these features is the choice of a VAV air system. Although VAV systems are more expensive than constant air volume systems, they have proven to be effective in reducing energy consumption. For the types of occupancy in the Integrated Sciences Building, which includes many classrooms and offices, VAV systems are often used in new buildings for this reason. In the laboratory spaces, the need for high volumes of outside air to ensure air quality also made it easy to see how a VAV system could reduce the energy use of the building. Even a relatively minor change in air volume delivered to a space can change energy consumption a great deal, especially over the course of a year or the life of the building.

The energy sources at the Integrated Sciences Building were another area which could have a huge impact on the sustainability of a project. Since the building has access to a district steam system, the need for a boiler within the building was eliminated. This may be seen as a cost-saving measure, but is often a more efficient use of fossil fuels than on-site combustion. Since this particular steam company uses an efficient cogeneration system, the efficiency of the steam distribution is very efficient.

One of the most impressive features of the Integrated Sciences Building is the Glycol Heat Recovery System which makes large strides in an effort to use energy which would otherwise be wasted. The simple concept of exchanging heat from exhaust air to incoming supply air, as well as heat from steam condensate, is projected to save a large portion of the energy cost of the building.

The cost of initial installation of mechanical systems as well as the operating cost is also a very good way to gauge the success of a building design. The improvements made above the ASHRAE baseline building are projected to save 25% of the annual energy costs of the building, according to the energy modeling engineer. Although this does not qualify the building for the all available LEED points, it is a significant annual savings. The cost of the mechanical system is 23% of the total building cost, but when considering the amount of energy savings, is in line with the typical 15-20% of the total building cost which some engineers use as a rule of thumb for total system cost.

Overall, the building seems to have very simple systems which perform very well. The building is expected to reach LEED Gold certification, which may or may not render it a "success" in the eyes of some engineers. Points and ratings aside, it was very challenging to come up with design proposals to improve the performance of the Integrated Sciences Building. Realizing that improving the building is a tall task, as well as some of the qualitative evaluations in this and previous technical reports, provides reason to render the building successful.

Mechanical System Redesign Proposal

Considered System Modifications

Because the task of modifying the mechanical systems of the Integrated Sciences Building was broad and had very few constraints, there were many good options available as opportunities to expand on education and experience with respect to investigation of HVAC systems. However, the fact that the Integrated Sciences Building mechanical systems were projected to achieve such good energy performance, the task of choosing the right topics to study was difficult. Choosing a topic that would not be able to produce improvements the building was not totally insignificant, but the goal was to improve the building as well as gain knowledge and educational value.

The list of possible alternative studies that was originally compiled for this project were:

- Envelope Load Flattening – Exterior Enclosure material investigation
- Thermal Storage System
- Dedicated Outside Air System (DOAS)
- Chilled Beams
- Radiant Floor Heating
- Demand Control Ventilation
- Variable Primary Flow vs. Primary/Secondary Flow Chiller Configuration
- Bio-Wall Indoor Air Quality Analysis
- Laboratory Indoor Air Quality Simulation

The lack of a large site, which is a result of the urban location of the Integrated Sciences Building, diminishes opportunity for an extensive ground source heat pump system, which could potentially be a rewarding study. Additionally, since the building purchases district steam for heating purposes, very few options are available involving the central heating system of the building. District steam is a good source for energy and eliminates the construction and equipment costs required by that equipment. Life Cycle Cost Analysis (LCCA) is a major factor in determining the success of a redesign, so installing a boiler or heating system is not necessarily a good way to improve the cost performance of a building when a distributed heating sources available.

Two of the above topics were chosen for detailed study and are described below. These topics include a Thermal Storage system, and an investigation into Primary/Secondary Chiller Configuration versus Variable Primary Flow.

Proposed System Alternatives

For the purpose of analyzing the performance of the proposed system alternatives, there must be discussion on the basis for which the results were compared to the original system. Due to the discrepancy between the design engineer’s load calculations and those prepared for this project and Technical Report II, some assumptions were made about the original system. Detailed energy consumption information was not made available for incorporation into the energy model which was prepared for Technical Report II and, as a result, the annual energy consumption figures and design loads did not match that which was provided.

In an effort to ensure a proper comparison between the proposed system alternatives and a similarly sized system, some of the equipment for the original system was resized. For instance, the primary and secondary pumps on the original P/S were sized for the load data which was available from the TRACE energy model instead of the scheduled design documents. Additionally, chiller sizes were modified so that the proper amount of capacity and redundancy was consistent through both the original and proposed alternatives. While this was obviously not an ideal situation, it was deemed the best way to ensure results that compared two similar systems were meaningful. Otherwise, results could have been severely misleading.

Variable Primary Flow Chiller System (MAE Depth)

Variable Primary System vs. Primary-Secondary System Background Information

A Variable Primary Flow (VPF) system is a type of chilled water pumping configuration in which the flow of water through the evaporator of a chiller is varied according to the demand for chilled water throughout the building. This type of system is becoming increasingly favored over the system which has historically been more common, known as the "Primary-Secondary" (P/S) configuration. The possible benefits of a VPF system include the lower capital cost, lower energy consumption, and tolerance to an operating issue commonly referred to as "Low ΔT Syndrome." These benefits are what make this type of pumping configuration so attractive for many building owners and engineers.

The lower capital cost of a VPF system comes from the fact that fewer pumps are required. As shown in the system schematic of a VPF system in Appendix B, there are no secondary pumps in this configuration. Instead, a VPF system relies solely on the primary pumps in a chilled water system to provide the flow for the entire system. The primary pumps in a VPF system are equipped to provide a larger amount of suction head, which would otherwise be split among the primary and secondary pumps in a P/S system. However, despite the need for higher-head pumps, eliminating a full set of pumps that are capable of moving the design flow rate of water drastically reduces the amount of capital cost for the system. Eliminating a full set of pumps also reduces the amount of mechanical space required. This creates another opportunity for capital cost savings. VPF systems do require some up-front expenditures which are not commonly required with P/S systems, including larger VFD motor controllers than the P/S secondary pumps, a modulating control valve in the bypass line, and more expensive instrumentation to control the system operation. However, the overall first-cost benefits, which are primary thanks to the elimination of a set of pumps, normally outweigh these extra costs.

The two sets of pumps in a P/S system, as shown in the schematic in Appendix B, operate separately. The Primary pumps that provide flow through the chillers' evaporators operate a constant speed whenever a chiller is on. The secondary pumps are modulated by VFDs in order to supply chilled water to the system loads as required. Any primary flow that is not required by the system goes through the bypass line back to the primary pumps. This means that, although the secondary pumps only use the energy required to provide chilled water to the cooling coils, the primary pumps each operate at full speed, and therefore full electrical load, at all times. When a chilled water system operates at part load conditions for a large majority of the time, like the Integrated Sciences Building, the constant speed primary pumps often provide capacity that is not necessary to meet loads, and therefore a good deal of that flow gets sent through the bypass and straight back to the primary pumps. This results in energy consumption that exceeds what would be required if the primary flow was modulated according to load, as is done in a VPF system.

Because there is only one set pumps in a VPF system, and those pumps are equipped with Variable Frequency Drives (VFD), they are able to be powered according to the flow rate required for the entire system, and not run continuously at full load like the primary pumps on a traditional P/S system. The pumps operate to maintain a pressure differential across the loads, as shown in the VPF Schematic in Appendix B. The flow meter located before the pumps is used to open and close the low flow bypass during low flow conditions, when the minimum flow exceeds that required flow of chilled water. The ability to reduce the amount of flow provided by the primary pumps allows for decreased energy consumption at part load conditions. Over the course of a year, this can result in significant pump energy savings, and therefore lower operating costs. Lower capital costs, coupled with lower operating costs, are what make VPF systems very attractive for building owners who pay the initial construction costs as well as the monthly and annual electricity bills.

"Low ΔT Syndrome" is a problem in chilled water systems in which the supply and return water flows do not have the temperature difference that is required for good heat transfer and load matching. This problem is caused by improper cooling coil selection, control, and maintenance. While solutions to the problem should initially be focused around reselection or replacement of the cooling coils, increasing the flow rate though an evaporator can be a quick or temporary solution. Dealing with this type of issue in a typical Primary/Secondary system can be difficult, because the primary flow rate through the chillers cannot be raised in order to compensate for the low temperature difference and more chillers must be started to meet loads. Operating more chillers also increases operation costs tremendously because of the condenser water systems that must also be activated. Also, when a temperature difference over an evaporator cannot be maintained, chillers cannot be utilized to their full capacity. In a VPF system, "Low ΔT Syndrome" can be fixed by increasing the evaporator flow rate in order to meet the chilled water demand requirements.

VPF systems are sometimes the subject of some concern for several reasons, which include optimistic energy savings estimates and control instability. Some critics are skeptical that these systems save as much energy as is claimed because the basis for comparison is sometimes not appropriate. For the purpose of the Integrated Sciences Building and this report, comparisons will be made by changing only the type of pumping configuration. This isolates the pumping configuration as the only possible reason for any change in energy consumption. System expansion is not of utmost concern for the Integrated Sciences building because it is an isolated building and the chilled water plant will not be serving other facilities. There is very little space for expansion to the building due its urban environment, and there is additional capacity available due to the redundancy which was included in the design. As for the control stability and reliability concerns, there is little that can be done to guarantee that a system will respond perfectly to controls, especially with lack of experience in that area. However, this type of system has been implemented successfully, and is gaining popularity for a reason. With the help of qualified control systems engineers and modern control instrumentation methods, the chance of a system failing on the basis of controls alone is unlikely.

One other concern with VPF systems is that not all chillers are capable of operating at a wide range of flows or under a high rate of change of flow. However, this worry is more of an issue with older chillers. New chillers are capable of operating at lower flow conditions than before, and many are able to operate when evaporator flow rate is varied at a higher rate than in previous years. The specification of the chiller which is being used was made available by a personal contact who works with York International and Johnson Controls. According to the specifications, the chiller as specified is capable of operating successfully in VPF systems with a minimum allowable evaporator tube velocity of 1.5 feet per second, which is much lower than the traditional tube velocity limit of 3 feet per second. In addition, the York chiller can tolerate a 50% flow rate change per minute. However, for system stability and set point control, this is usually limited to around 30% per minute by the control sequence.

VPF Analysis

Expectations for the VPF analysis versus the original P/S system were that the VPF system would consume considerably less electricity than the P/S system. As described before, this was due to the variation of flow by the primary pumps in a VPF system. Over the course of a year, pumping energy savings are typically between 20 and 40 percent over a comparable P/S system. This translates to a normal savings of anywhere between 0-5% for the entire cooling plant.

Pump Selection and Curves

Before performing calculations, new pumps were selected based on the energy model results from the TRACE simulation from Technical Report II. This included new pumps for both the primary and secondary loops for the P/S system. In addition, pumps were selected for the VPF system based on the design flow rate and the total system head requirements. The pump curves used for this analysis are provided for reference in Appendix B. Pumps were selected using the Bell & Gossett pump curve selection guide, which is also shown in Appendix B. New pumps selected for the P/S configuration are shown in Table 14.

Primary/Secondary System Pump Selection						
Pump Tag	Service	GPM	Head [ft]	Motor		Notes
				HP	RPM	
P-1	Primary Chilled Water Pump (Duty)	625	50	15	1750	-
P-2	Primary Chilled Water Pump (Duty)	625	50	15	1750	-
P-3	Primary Chilled Water Pump (Standby)	625	50	15	1750	-
P-7	Secondary Chilled Water Pump (Duty)	625	76	15	1750	VSD
P-8	Secondary Chilled Water Pump (Duty)	625	76	15	1750	VSD
P-9	Secondary Chilled Water Pump (Standby)	625	76	15	1750	VSD

Table 14 – New Pump Selections for P/S System Energy Analysis

The new pumps for the VPF system were chosen based on the total head of the system, which equaled 126 feet, and the required flow rate for each pump, which was 625 gallons per minute. These pumps are shown in Table 15.

VPF System Pump Selection						
Pump Tag	Service	GPM	Head [ft]	Motor		Notes
				HP	RPM	
P-1	Primary Chilled Water Pump (Duty)	625	126	30	1750	VSD
P-2	Primary Chilled Water Pump (Duty)	625	126	30	1750	VSD
P-3	Primary Chilled Water Pump (Standby)	625	126	30	1750	VSD

Table 15 – Pump Selections for VPF System Energy Analysis

Note that all sets of pumps have two “duty” pumps and a “standby” pump. The standby pump is included for redundancy in case one of the duty pumps should fail. These were included in the original design of the Integrated Sciences Building and were also included here for their effects on the Life Cycle Cost Analysis.

Calculation Method

The calculations for the VPF analysis were performed in Engineering Equation Solver (EES). Initially, attempts were made to perform the calculations in Microsoft Excel. However, after a short time, it was clear that EES was better equipped to handle more complex calculations due to its ability to solve for variables within complex equations. The parametric analysis was based on the hourly flow requirements for both the P/S and VPF systems. It must be noted that the rate of change of flow discussed above, which is a limit of the chiller, was not implemented in this analysis. Because that rate of change of flow rate is controlled on a minute-by-minute basis, it was determined that it would not be of significance on the overall annual scheme of the pumping configuration performance. Also, for the purpose of this analysis, that level of detail was not possible with the amount of time and experience provided.

The pump flow rates were determined using data extracted from the TRACE energy model results which were available from Technical Report II calculations. By knowing the amount of tons of chilled water cooling demand per hour, the amount of chilled water flow was determined using the following equation:

$$\dot{Q} = \dot{m} C_p \Delta T \approx 500(gpm)\Delta T$$

Where:

$\dot{Q} = total\ cooling\ demand\ \left(\frac{Btu}{hr}\right)$

$\dot{m} = flow\ rate\ of\ water\ (\frac{lb_m}{hr})$

$C_p = Specific\ Heat\ Capacity\ of\ Water\ (\frac{BTU}{lb_m{}^\circ F})$

$\Delta T = Chilled\ Water\ Supply\ Temperature - Chilled\ Water\ Return\ Temperature$

From this equation and some conversions based on water density, time, and tons of cooling, the flow of chilled water for each hour of the day was calculated. The equation 500*gpm*ΔT is a simplified version of the equation based on water density, heat capacity, and time conversions, and allows simple calculation from tons of cooling directly to flow rates. This flow was used as the amount of flow required in both the Secondary pumps of the P/S system and the Primary pumps in the VPF system. The flow rates of the Primary pumps of the P/S system were constant at 625gpm at any hour the chiller was on.

Pump Efficiency and Operating Point Calculation

Pump characteristics were modeled in EES by a method of linear regression to a fourth degree polynomial based on flow versus pump head data extracted from pump curves. This allowed an equation to be used for pump head as a function of both flow and motor speed. Using the same method of regression based on pump curve data, an equation was used to calculate pump efficiency as a function of flow rate and pump speed.

Pump Staging

Pump staging was set up in EES by writing calculations to determine the flow rate at which the second pump should be turned on in order to maintain the highest level of efficiency for the overall system. This flow rate was designated the "changeover point" at which the parametric equations would switch the calculations from only one pump to a two-pump operating mode. Knowing the number of pumps in operation helps the program calculate the total energy demand for a set of pumps.

Motor Efficiency Calculation

The efficiency of the pump motors was determined by a regression of the curve in Figure 1. This equation calculates motor efficiency as a function of the percentage of nameplate load the motor is running at during each hour. This equation is a piecewise function due to the extreme slope change that occurs at approximately 18% of nameplate load.

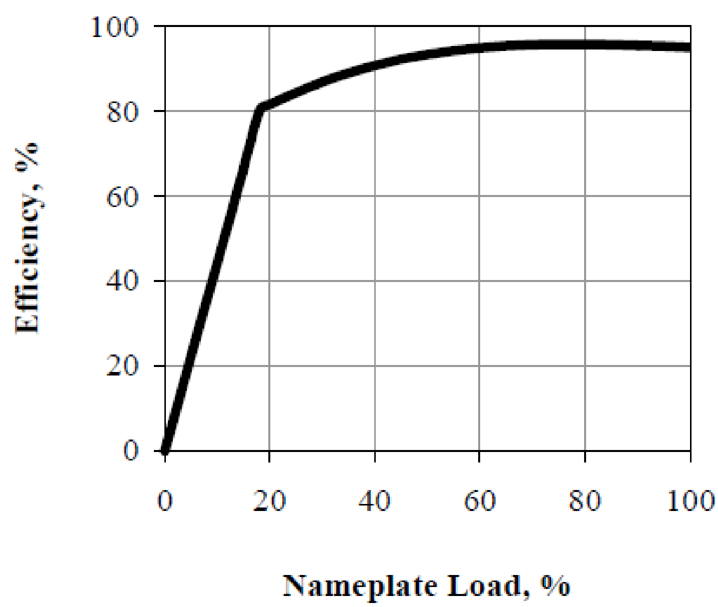

Figure 1 – Pump Efficiency vs. Design Load Percentage Curve.

Variable Frequency Drive Efficiency Calculation

The Efficiency of the Variable Speed Drive was calculated the same way as the motor efficiency. The curve which was used in the regression to get VSD efficiency as a function of the percent of nameplate load is shown in Figure 2.

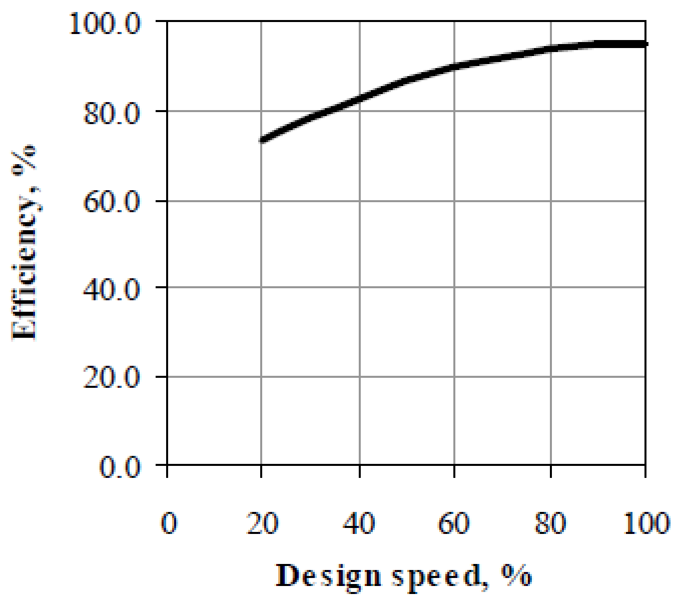

Figure 2 – Variable Frequency Drive vs. Design Speed Percentage curve

Energy Consumption Calculations

To determine the difference of energy between the P/S system and the VPF, total pump energy of the respective systems must be compared. For the P/S system, a calculation must determine the annual consumption of the Primary pumps, and another for the Secondary pumps. For the VPF system, there is only one set of pumps, and thus only one calculation must be made for those pumps.

Pumping energy is calculated using the equation:

$$whp = (Q * H)/(3960 * \eta_{Pump})$$

Where:

Q = Required flow (gpm)

H = Pump Head (feet of water)

η_{Pump} = Pump Efficiency (calculated using Pump curve regression equation)

3960 = Unit conversion constant

Total Pump Electrical energy required is then calculated using the equation:

$$kW = 0.746 * whp/(\eta_{Motor} * \eta_{VFD})$$

Where:

kW = Electrical Energy demand per hour (kW)

0.746 = conversion factor (0.746 BHP = 1 kW)

η_{Motor} = Motor Efficiency (calculated using Motor Efficiency curve regression equation)

η_{VFD} = Variable Frequency Drive Efficiency (calculated using VFD curve regression equation)

Based on the required flow rate, which was known for each hour of the year, the pump head requirements and pump efficiency was calculated by the regression equations described above. Using a parametric table in EES, it was easy to set up a dynamic calculation that would solve for the appropriate efficiencies and pump head based on flow rate. This method of calculating pump energy consumption was used to find the amount of electricity demand by the VPF system pumps as well as the secondary pumps in the P/S system. Because the calculations were performed on an hourly basis for the 8760 hours of the year, these numbers can be considered electricity consumption because 1 kW x 1 hour = 1kWh. Summing this number for each hour of the year provided the amount of annual energy consumption for the different pump sets.

The VPF pumping system energy consumption calculation used an elaborate set of equations, as described above, to optimize the system based on operating with two duty pumps arranged in parallel configuration, both equipped with Variable Frequency Drives. These equations as programmed in EES are provided in Appendix B. Summing the kW of pumping energy for each hour of the year gave a reasonable estimate of the annual energy consumption of the VPF Pumps.

The same method that was used to calculate VPF Pumping energy was also applied to the Secondary pumps of the P/S system. Because both pump sets operate as a set of parallel, VSD driven machines, the equations were very similar. The main difference was pump efficiency, and pump head calculations.

For the Annual P/S configuration pump energy, the amount of energy demand for the primary pumps was calculated in a much simpler way. When either of the primary pumps is on, they operate at full speed and flow in order to achieve constant flow to the evaporators they serve. When no flow is required to the chillers, both primary pumps are off, and power demand is 0 kW. When one of the pumps is on and the other is off, it requires 7.75 kW. When both pumps are on, the primary power demand doubles, to 15.5 kW.

After using this method to calculate the amount of electricity demand in kW for each different flow possibility, the data could be extracted from EES and converted to a spreadsheet. Using Excel lookup functions, the amount of pumping demand can be identified for each hour of the day according to the flow rate required for each hour. In retrospect, this task could have been performed in EES, but was still very simply done in Excel.

Primary/Secondary Demand Profiles

The total P/S energy requirements are a combination of the Primary pumping energy and the Secondary energy. Figure 3 shows the energy demands of the two sets of pumps, as well as the total energy of the P/S system. It is easy to see the point where the Primary portion of the system switches from one pump operation to two pumps, at the flow level of 625 gallons per minute. Because of the Variable Frequency Drives, however, the secondary pump energy gradually increases as the required flow increases. This indicates that the pump staging methodology is operating very well. In real situations, staging controls are not this simple, but for an hourly energy simulation, it is a fairly accurate method. The "Total P/S Power" curve versus flow still shows that there is a considerable energy increase when the second primary pump is activated, which is due to the discontinuity of the primary pump energy demands.

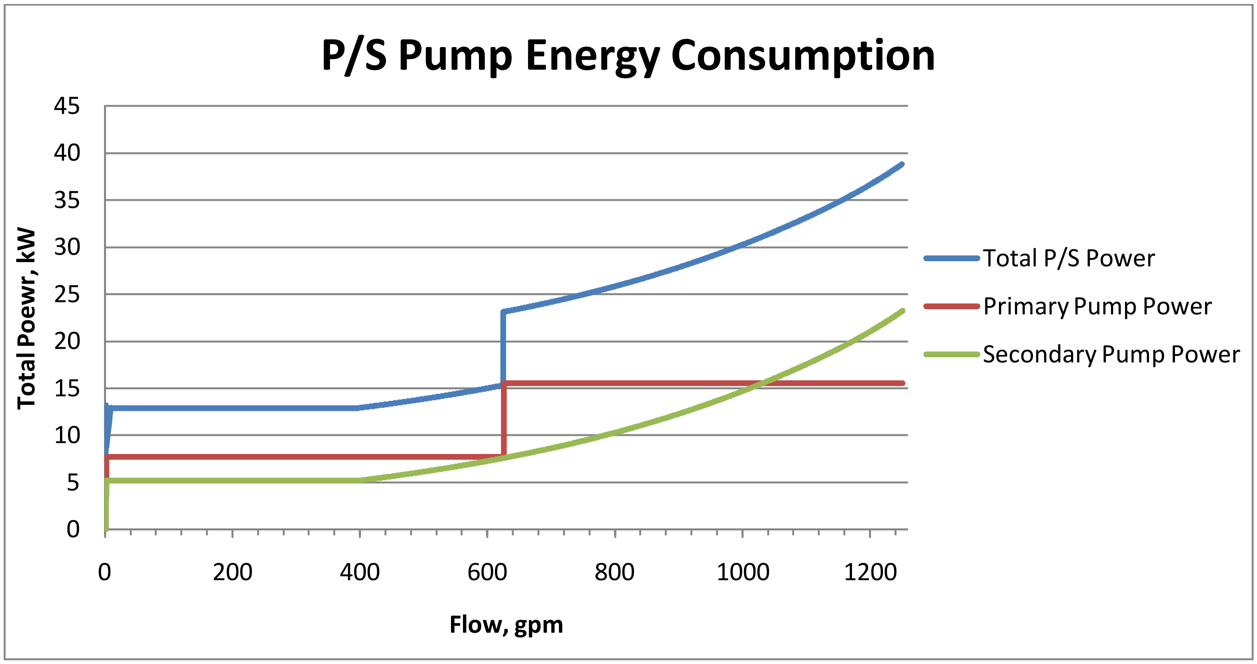

Figure 3 – Primary Pumping Power Demand versus Flow

To find the annual energy consumption of the P/S System, the load profile of the Integrated Sciences Building, as shown in Figure 4 is a very useful visual tool. The chart shows that during only 30% of the total year, there is a significant cooling water flow. It also shows that the ISB only sees a large percentage of the design flow for fewer than 5% of the year. This means that for a large portion of the year, flow requirements are very low. Therefore, primary pumps are likely providing excess flow to the evaporator of the chiller that is in operation. This offers a prime opportunity for savings with a VPF system.

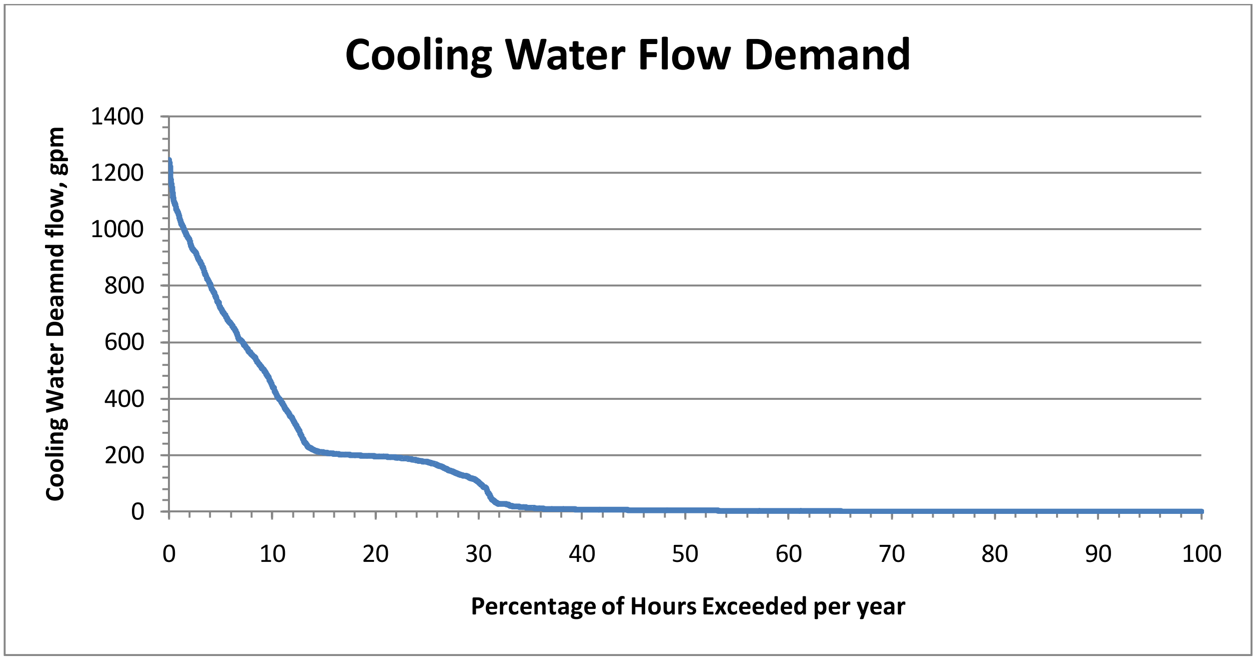

Figure 4 – Annual Cooling Water Flow Demand curve

Annual consumption was calculated by identifying the energy demand for the total system on an hourly basis according to the required flow. Over the course of a typical meteorological year, it was calculated that the P/S pumping energy required is 77,154 kWh.

VPF Demand Profiles

Calculation of the pumping energy according to the amount of flow for the VPF system was performed in the same way as for the Secondary pumps in the P/S system. This time, there were no primary pump energy demand numbers to include. The profile of energy consumption based on flow requirements is shown in Figure 5. The profile discussed above for the P/S system is also included for comparison. Note that the curve for the VPF system indicates that less energy is required compared to the P/S system for a large portion of the range of flows that are required. However, the portion of this graph to near the maximum flow requirement of 1250 gallons per minute seems unusual. The pump was initially selected using a Bell & Gossett Selection guide. The model of pump which was used in this calculation, Bell & Gossett's 1510-3G model, had very low pumping efficiency of 65% at the design flow. Because of this low efficiency, more power was required to provide the proper amount of suction to the system.

Under this configuration, labeled "Original VPF," annual electricity consumption was calculated as 54,289 kWh, which is a 22,625kWh savings or 29% over the original P/S System. For the total plant energy, the VPF pumping configuration would provide a 2.78% savings in electrical consumption. These figures are presented in Table 16 at the end of this section.

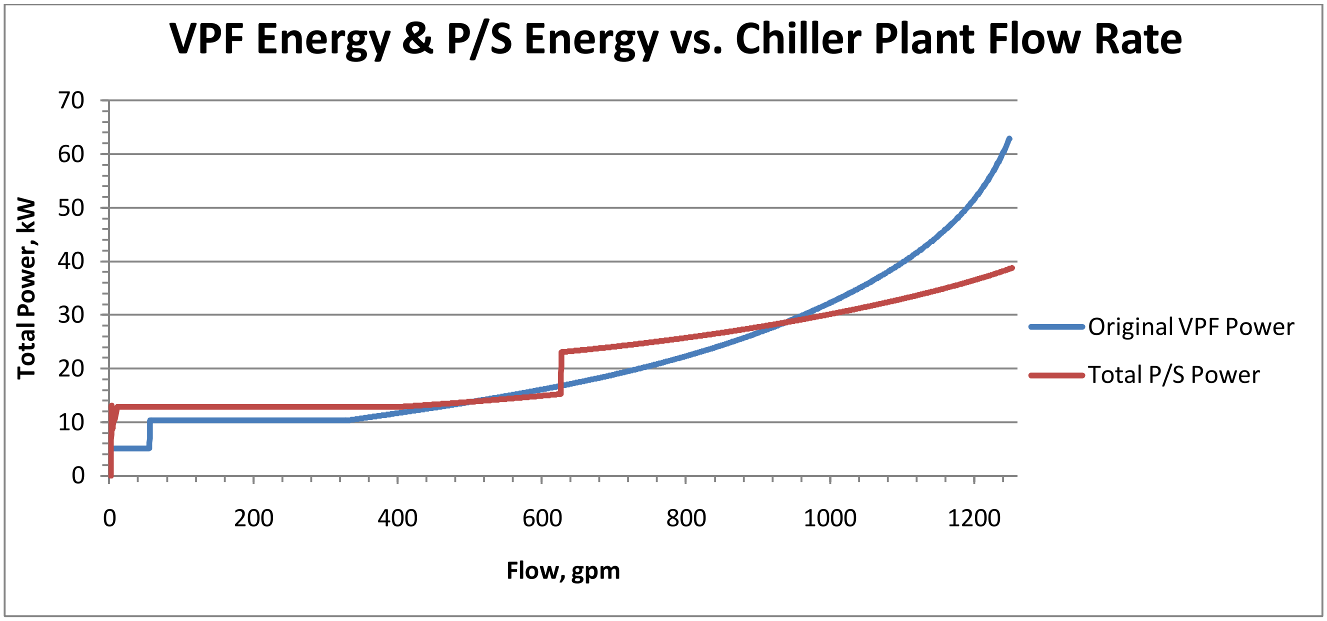

Figure 5 – VPF Pumping Power and P/S Total Pumping Power versus Flow Curve

After examining the possibilities with the initial pump selection, it was discovered that a different VPF pump selection could improve the energy savings of the VPF system over the P/S System even further. By selecting a pump that operates more efficiently at the design load, more savings should be possible. Because of the good efficiency of modern Variable Frequency Drives, savings with a larger pump should not need to be sacrificed at lower operating conditions. To investigate, another pump was chosen, specifically, the Bell & Gossett 1510-4GB.

Using the Bell & Gossett 1510-4GB, a new energy profile was calculated to show the energy required at given flow rates. This curve is shown in Figure 6 with the same P/S curve overlaid as before. Compared to the original VPF energy curve shown above, this new simulation shows a VPF demand curve that uses less electrical power at nearly all flow conditions. Because the Bell & Gossett 1510-4GB pumps have a higher pumping efficiency at their full flow of 625 gallons per minute each, the total efficiency when both pumps are running at full capacity is substantially better. As a result, less electricity consumption is required.

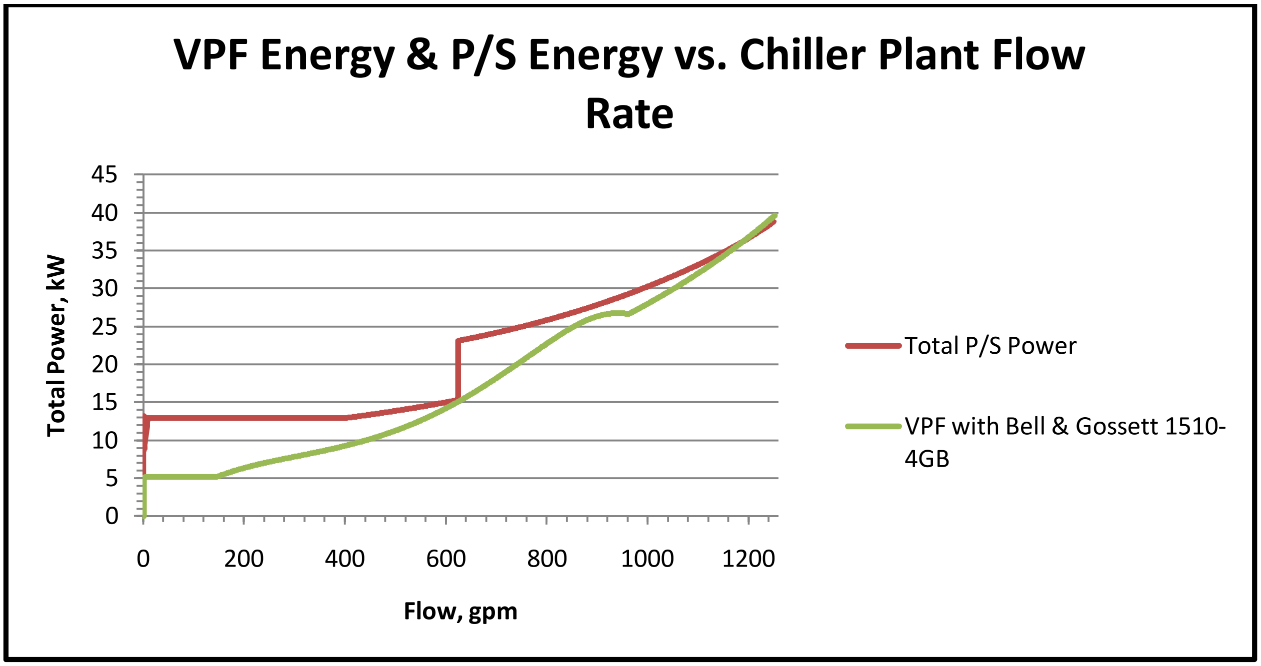

Figure 6 – VPF Pumping Power (with B&G 1510-4GB Pumps) and P/S Pumping Power versus flow

Under this configuration, labeled "VPF with 1510-4GB," annual electricity consumption was calculated as 44,910 kWh, which is a 32,245kWh savings or 42% over the original P/S System. For the total plant energy, the VPF pumping configuration would provide a 3.97% savings in electrical consumption. These figures are also presented in **Error! Reference source not found.** below.

P/S vs. VPF Parametric Study Results			
	P/S System	VPF - 3G	VPF - 4GB
Annual Consumption (kWh)	77154.67	54,528	44,909
Savings (kWh)	-	22,625	32,244
Savings % over P/S	-	29%	42%
Total Plant Savings	-	2.78%	3.97%
Annual Consumption Cost	$3,718.86	$2,628.30	$2,164.66

Table 16 – VPF versus P/S Pumping Configuration Parametric Study Results

Total VPF Energy Savings

The pump energy savings associated with Variable Primary Flow have been discussed for both of the different VPF pump types in comparison to the original Primary/Secondary pumping configuration. Figure 7 shows a graphical representation of the pumping energy for each of the three configurations discussed above. Clearly the potential for savings with a VPF system is beneficial with regard to energy consumption improvements.

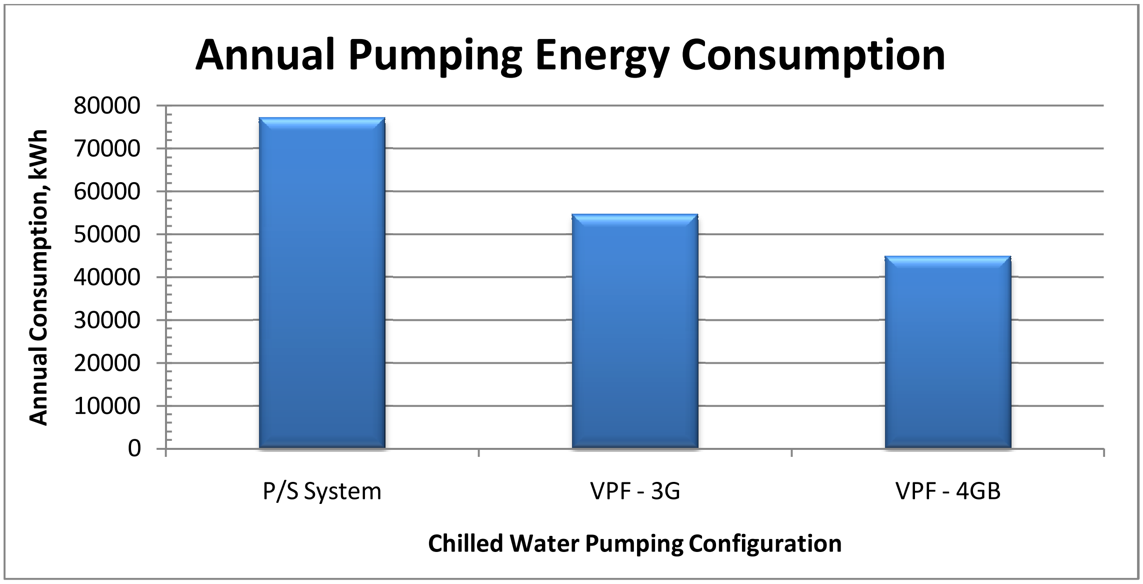

Figure 7 – Annual Energy Consumption comparison for Original P/S, VPF with 1510-3G Pumps, and VPF with 1510-4GB Pumps.

Additional Energy Savings with VPF

There is some potential for energy savings in VPF systems that was not calculated for the purpose of this report, but that deserve discussion nonetheless. The ability to change the amount of flow through an evaporator allows more water flow through fewer chillers at some design points. If fewer chillers are able to meet the cooling demand at a higher flow rate, this means that fewer chiller auxiliaries are required, which includes condenser water pumps and cooling tower fans. Fewer auxiliaries operating translate into increased savings in condenser pump and cooling tower fan electrical consumption. Also, depending on the chiller characteristics, operating at peak load can also increase the overall chiller efficiency. Special care must be taken in the control systems to ensure that flow is within the maximum limits of the chiller's evaporator flow rating and that the pump motor power is not exceeded. In Primary/Secondary systems, this control over evaporator flow is not possible and getting better chiller efficiency based on evaporator flow is not an option.

VPF Financial Analysis & Life Cycle Cost

In examining the financial analysis for the VPF system, comparisons were made about the capital cost, operating cost, and overhaul costs for the original P/S pumps and the VPF system with the Model 1510-4GB that saved 42% of the annual pumping energy. Capital costs for the pumps and VFD motor controllers were taken RS Means Mechanical Cost Data guide. The prices include materials and installation cost. Because of the standby pumps that ensure redundancy, the cost of three pumps was used for each set of pumps. First Cost data for the P/S system and the VPF System can be found in Appendix B.

Energy cost was based on $0.0482/kWh for consumption. Demand charges were not included in this calculation due to the fact that all energy consumption numbers were based solely on pump energy and the interactions with the overall peak demand is difficult to ascertain manually without a software package like Trane TRACE. Life Cycle Cost based on 30 years of operation was the method used to determine financial performance. Included in the Life Cycle Cost Analysis calculations were the annual energy consumption costs, as well as the 30-year OMB discount rate of 2.7% and electricity escalation factors from the 2010 NIST Life Cycle Cost Supplement.

The overhauls included are intended to account for the replacement of VPF controllers. VFD controllers have a life span of approximately 100,000 hours. Assuming they operate for most of the year, this suggests that they need to be replaced every 12 years. The VFD overhauls cost more for the VPF system because the same number of VFDs is required; however they are rated for 30 HP pumps instead of the 15HP pumps on the Secondary of the P/S system.

Maintenance costs, which are difficult to determine for most HVAC equipment, was estimated at $1000.00 per year for both pumping configurations.

The complete life cycle cost spreadsheet is provided in **Error! Reference source not found.**. The esults of that analysis are shown in Table 17. The total benefit after 30 years comes to $46,069.

30-Year Life Cycle Cost Breakdown		
LCC 30-year Net Present Value	Primary/Secondary	VPF [1510-4GB]
Capital Costs	$70,725	$51,050
Overhauls	$8,966	$14,671
Maintenance	$20,383	$20,383
Electricity Consumption	$76,806	$44,707
Total 30-year Life Cycle Cost	$176,880	$130,811
	30-year Savings	$46,069

Table 17 – 30-year Life Cycle Cost Comparison of P/S System and VPF System with 1510-4GB Pumps.

Emissions

Saving electricity in a region of the United States that still produces a large percentage of electricity from fossil fuel combustion means that fewer emissions are exhausted into the atmosphere.

Annual Emissions for Electrical Consumption		
Pollutant	Eastern Interconnection	VPF Emissions Savings per Year [lb]
Electric Use	1 kWh	32244.818 kWh
CO_{2e}	1.74E+00	5.61E+04
CO_2	1.64E+00	5.29E+04
CH_4	3.59E-03	1.16E+02
N_2O	3.87E-05	1.25E+00
NO_x	3.00E-03	9.67E+01
SO_x	8.57E-03	2.76E+02
CO	8.54E-04	2.75E+01
TNMOC	7.26E-05	2.34E+00
Lead	1.39E-07	4.48E-03
Mercury	3.36E-08	1.08E-03
PM10	9.26E-05	2.99E+00
Solid Waste	2.05E-01	6.61E+03

Table 18 – Annual Emissions savings as a result of VPF system.

Recommendation

Because the Capital cost of installation for the VPF is lower than the cost of P/S system, no payback period is required. Instead, financial benefits can be collected from the very first year of installation. This is one of the benefits of VPF that is making it increasingly attractive to building owners. In addition, the annual energy consumption reduction produces savings in operating costs when compared to a similar P/S system. Because the system analyzed for this report was based on two duty pumps and two duty chillers, the savings were also very positive. For larger chilled water plants with multiple chillers or primary pumps, staging is used to optimize the system operation and VPF is less beneficial than in this case. However, according to the calculations performed for this report, the Integrated Sciences Building is a prime building candidate that could benefit immediately from a Variable Primary Flow chilled water pumping system.

Thermal Storage System (MAE Depth)

Thermal Storage is a method of optimizing a chilled water system so that it reduces operating cost, even if it does not necessarily mean that energy consumption goes down. For this portion of the project, an investigation was pursued to determine the feasibility, based on financial success as well as constructability, of applying a thermal ice storage system to the Integrated Sciences Building chilled water system.

Thermal Storage Type – Latent vs. Sensible

Sensible thermal storage could be applied to this system, but because of the relatively small capacity of the thermal storage system in relation to past applications, as advised by Dr. William Bahnfleth and several ASHRAE publications, ice storage was chosen for the Integrated Sciences building. Until chilled water storage tanks get very large, they are rarely cost effective. In practice, ice storage systems average about 3,100 ton-hours, which turned out to be near the size of this system. As a result, thermal ice storage has proven to be the most cost effective method for thermal storage systems below 10,000 ton-hours of capacity.

Ice storage causes an inherent efficiency loss and increased energy consumption because of the greater temperature difference the chiller is asked to provide during icemaking. The Carnot efficiency can be used to illustrate the icemaking energy penalty:

$$COP_{Carnot} = T_L/(T_H - T_L)$$

Where:

$T_L = Low\ Temperature\ (Water\ out\ of\ Chiller)$

T_H = High Temperature (Water into Chiller)

It is easy to see that an increased temperature difference, which is present during icemaking mode where the supply temperature is below the freezing point of water, decreases chiller efficiency.

Thermal Storage Method – Full Storage vs. Partial Storage

To determine the type of thermal storage system that would work best at the Integrated Sciences Building, a basic understanding of the two types of systems and their control methods was a very important learning experience. Thermal storage can be done by storing either sensible (cold water) or latent (ice) storage. Additionally, selecting the capacity of the system depends on whether the goal is load flattening using partial storage, or complete load shifting using full storage.

The full storage option is used when chillers are run at night to store all of the cooling capacity required during the daytime hours. During the day, they do not run at all and therefore do not consume energy at "peak" pricing hours. Because the Integrated Sciences Building has a flat energy rate that increases drastically during the daytime hours, full storage was not considered because it does not reduce chiller size as much or have as large of an impact in reducing electricity demand compared to partial storage.

Partial thermal storage configuration is a method of flattening the chilled water load over the course of a daily period. This is appropriate for buildings like the Integrated Sciences Building, which have very uneven daily load profiles. Figure 8 gives a visual representation and explanation of how partial storage works. The chillers are run continuously to flatten the load profile. During the nighttime hours, when load is low or negligible, the thermal storage tanks are in a charging mode, which operates the chiller at temperatures below the freezing point of water. During the daytime hours, when the load profile peaks, the chillers run at higher temperature and the ice capacity is used to meet loads when chilled water demand exceeds chiller capacity.

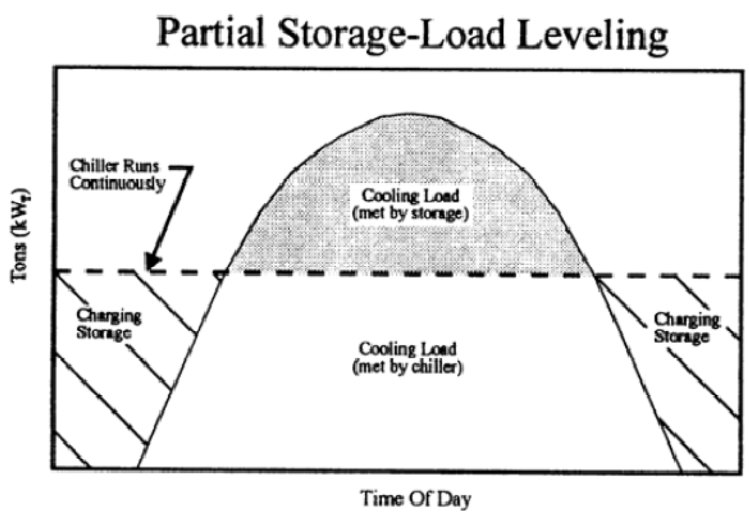

Figure 8 – Partial Thermal Storage Load Leveling Load Profile Illustration

Figure 9 shows the peak summer design cooling day for the Integrated Sciences Building, which happens to be August 14th. In comparison to Figure 8, the original load curves are very similar. The Red line in Figure 9 represents the average cooling load for the 24 hour period. This is a very crude estimate for the capacity of a chiller that would run continuously. Clearly, chiller capacity can be much lower, near 300 or 400 tons, than a non-thermal storage system which would need a 730 ton capacity to meet the full peak load. However, chiller capacity cannot be selected based on average load alone, as will be discussed below.

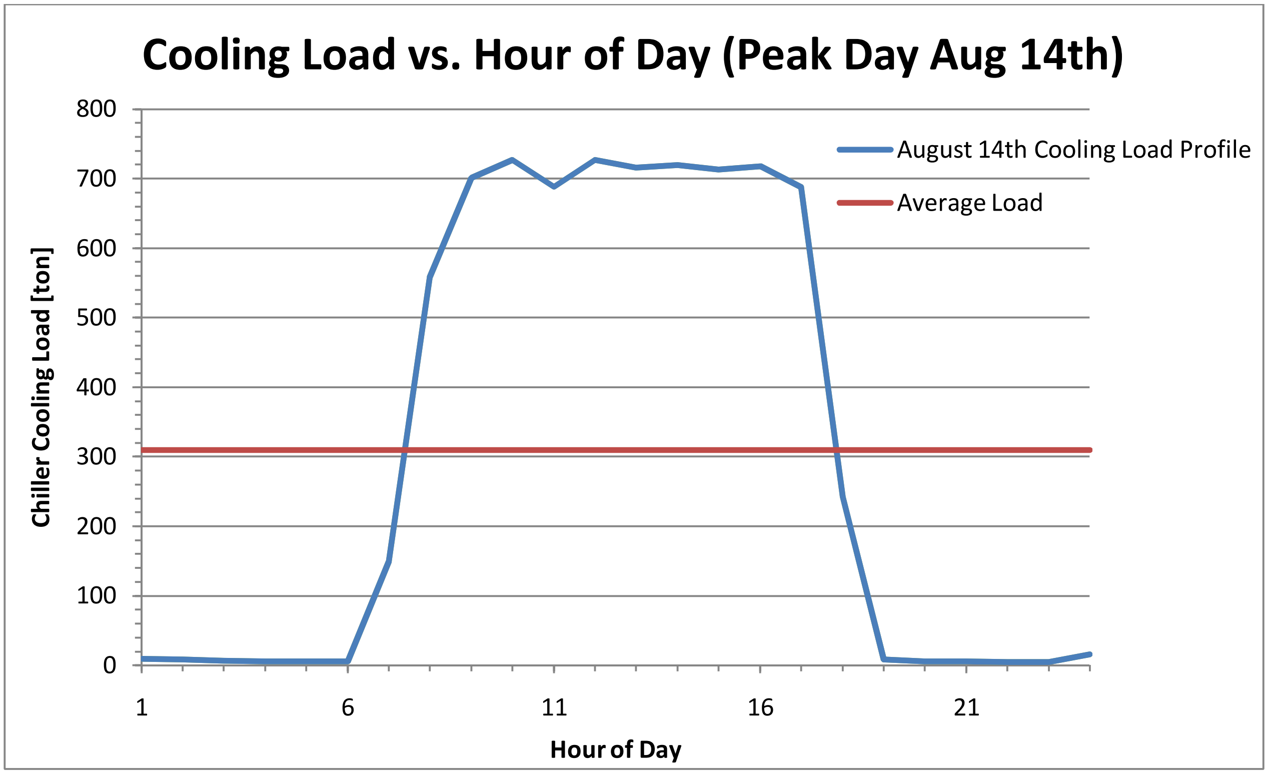

Figure 9 – August 14th Design Day Cooling Load Profile

Savings Opportunity

The savings that can be generated for a partial thermal storage system is not the same as for a full storage system. While full storage systems depend on reduced nighttime electricity rates, commonly called "off peak" rates, partial storage saves money on monthly electrical peak demand costs. The Integrated Sciences Building has a demand charge of $8.97 per kilowatt. This means that over the course of a monthly billing cycle, the maximum electrical demand for that month is recorded and multiplied by $8.97. Because a chiller consumes a large portion of the building's electricity during peak cooling load, it is a major contributor to this demand charge. For example, the August 14th load profile suggests the peak chilled water load for August is 727 tons. At 0.642 kW per ton, the chiller's energy rating, the chiller would consume approximately 467kW. This of course, is an estimate because the chiller runs at different energy consumption rates according load. However, if the load profile is flattened by using thermal storage during the daytime peak load hours, the chiller capacity can be reduced to around 400 tons, which is an estimated 257kW demand. This simple calculation results in a 210kW chiller electrical demand reduction, which at $8.97/kW, comes to over $1800 in savings for that single month's electric bill. Over the course of a year, as will be discovered below, the savings can be very significant.

Tank Selection

The Thermal Storage tanks that were chosen for this system are made by a company called CALMAC who specializes in ice storage tanks with the trade name ICEBANK®. Figure 10 shows an exterior view of the ICEBANK® tanks and a model of the internal coils of the tank.

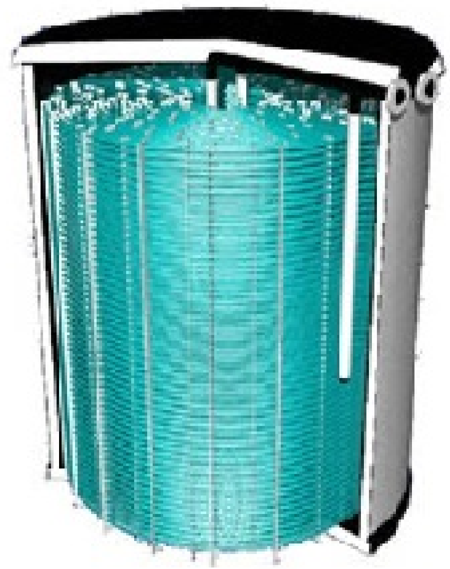

Figure 10 – Exterior of ICEBANK ® tank, right, and internal representation of ice coils, left.

The tank is an internal-melt, ice-on-coil type which means that during both charging and discharging modes, the water flows inside the heat exchanger tubes, as shown in Figure 11. Internal melt ice tanks are advantageous because they operate very simply and require minimal external piping connections. However, a downside is that heat transfer changes over time. During charging, the chiller must operate at colder temperatures as ice continues to form. During discharge, heat transfer continually decreases due to the water formation on the outside of the coils. However, the CALMAC tank spaces the coils very closely in order to minimize this effect.

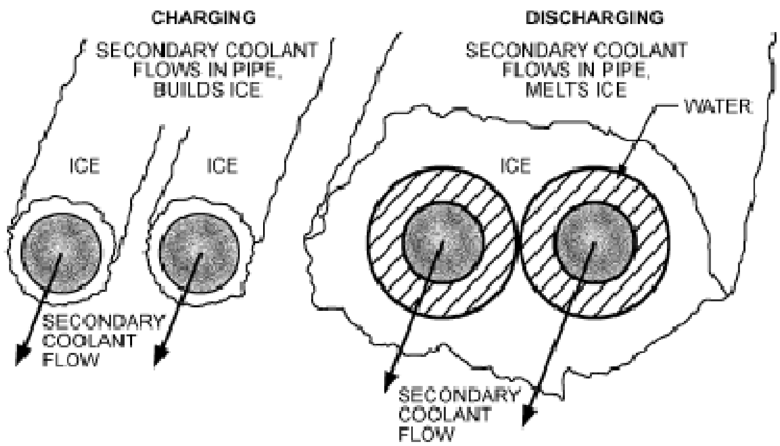

Figure 11 – Internal Melt Ice-on-Coil Charging and Discharging diagram.

After consulting with a CALMAC sales representative, his recommendation, according to the design day load profile, was that 6 model 1500CSF tanks were the best design choice for this ISB system. Model 1500C is essentially a set of 3 of the tanks shown in Figure 10, which are connected by common piping. This reduces installation costs by limiting the amount piping required in the field. Each tank has a net usable capacity of 486 ton-hours. Therefore, the total system capacity comes to 2916 ton-hours. The usable capacity takes into account the losses associated with insulation and piping.

The Tanks are connected in a parallel configuration so that they all receive equal flow. This ensures that they charge and discharge in the most efficient manner. A diagram of the tank piping is shown in Appendix C.

Maintenance of the ICEBANK® ice storage tanks is very minimal, which is another benefit over larger chilled water plants. Maintenance tasks include:

- Annual tank water level check
- Annual heat transfer fluid check (25% Glycol-75% Water Mixture)
- Biennial Biocide Treatment in Tanks

While the annual water level check and the annual glycol solution check can be very easily included in normal chiller maintenance, the Biocide does cost $30 per gallon. This is a very minimal cost and the lack of maintenance is very beneficial during the life cycle cost calculation.

Chiller Selection

Selecting a chiller is a very important consideration when designing an ice storage system. When in ice-making mode, chillers typically have about 40-60% of the capacity of HVAC mode. Therefore, it important to work closely with a knowledgeable sales engineer or manufacturer of chiller equipment to be sure that the chiller that is selected can meet both the ice and HVAC demands.

Because the redundancy methodology of the ISB design documents was unclear and attempts to discover the methodology by communicating with the design engineer failed, the best way to get an accurate comparison of the TES energy consumption compared to the original non-TES system was to perform the analysis as if both systems had the same redundancy. For the purpose of this report, therefore, just as in the VPF Depth, some redesign was performed in order to make a realistic comparison. The decision was made to go with three half-sized chillers for each system. For the original system, with 727 tons of peak load, three identical 370-ton chillers were used in simulation. After speaking with a TRANE representative, a rating of 0.642 kW/ton was an accurate energy rating for this type of centrifugal chiller.

The communication with the TRANE representative resulted in the selection of three centrifugal chillers that have a capacity of 200 tons in HVAC mode with an energy rating of 0.641kW/ton. During ice mode, the capacity decreased to 135 tons with an energy rating of 0.770kW/ton. The increase in kW/ton ratings for the ICE mode relative to the HVAC mode was less than expected. However, according to the HVAC representative, the energy ratings at different modes of operation are unique to different chiller types and sizes. This is something that should be considered in all ice storage system designs.

Knowing the capacity of the chillers during both HVAC and ICE modes was very crucial in being able to perform an energy analysis of both systems that would be useful in comparing performance. The change in capacity and efficiency between HVAC and ICE modes is one of the dynamic factors that can have significant impacts on the success or failure of an ice storage system.

Ice Storage System Controls

The operation of a thermal ice storage chilled water plant system can be very complicated to control, but the theory is very simple. The system has two modes: Charging Mode (ICE mode) and Discharge Mode (HVAC mode). Charging mode occurs at night when the load profile is at very small or negligible load and discharging mode occurs during the daytime when the HVAC capacity of the chiller is exceeded by the chilled water demand.

During “Charging mode”, a three way valve sends a 25% glycol/75% water solution to the thermal storage tanks. The glycol prevents freezing in the supply lines. The chiller must produce water temperatures of about 25°F so that the solution going to the tanks is cold enough to freeze the water in the tanks. The return temperature to the chiller is about 30°F depending on how efficiently the heat transfer in the tank occurs. Because the tanks chosen for this application are internal ice-on-coil style, heat transfer efficiency changes over time. As the glycol solution flows through the ICEBANK heat exchanger coils, absorbs heat from the water in the tank, which begins freezing. This continues for 6-12 hours, depending on the control optimization sequence, until 95% of the water in the tanks is frozen solid. Because the water moves freely throughout the tank while it freezes, and because the coils are spaced very close together, freezing is nearly uniform within the tank. The glycol flow path during “Charging Mode” is highlighted using blue lines in Figure 12.

During “Discharge Mode,” the chiller will be running at full HVAC capacity and the chilled water demand will eventually exceed the normal cooling capacity of the equipment. When this happens, the chiller discharge temperature will raise above the chilled water supply temperature set point of 42°F. At that point, the Temperature Modulating Valve will open and close as necessary so that the supply water flows through the ice storage tanks. The ice tanks will cool the glycol solution, with heat transfer in the opposite direction as during charging mode. As the ice in the tanks melt, the glycol solution cools and returns to the system to be distributed to the cooling coils. The combination of the chiller capacity and the ice capacity combine to the meet the full daily cooling loads. The glycol flow path during “Discharging Mode” is highlighted using blue lines in Figure 13.

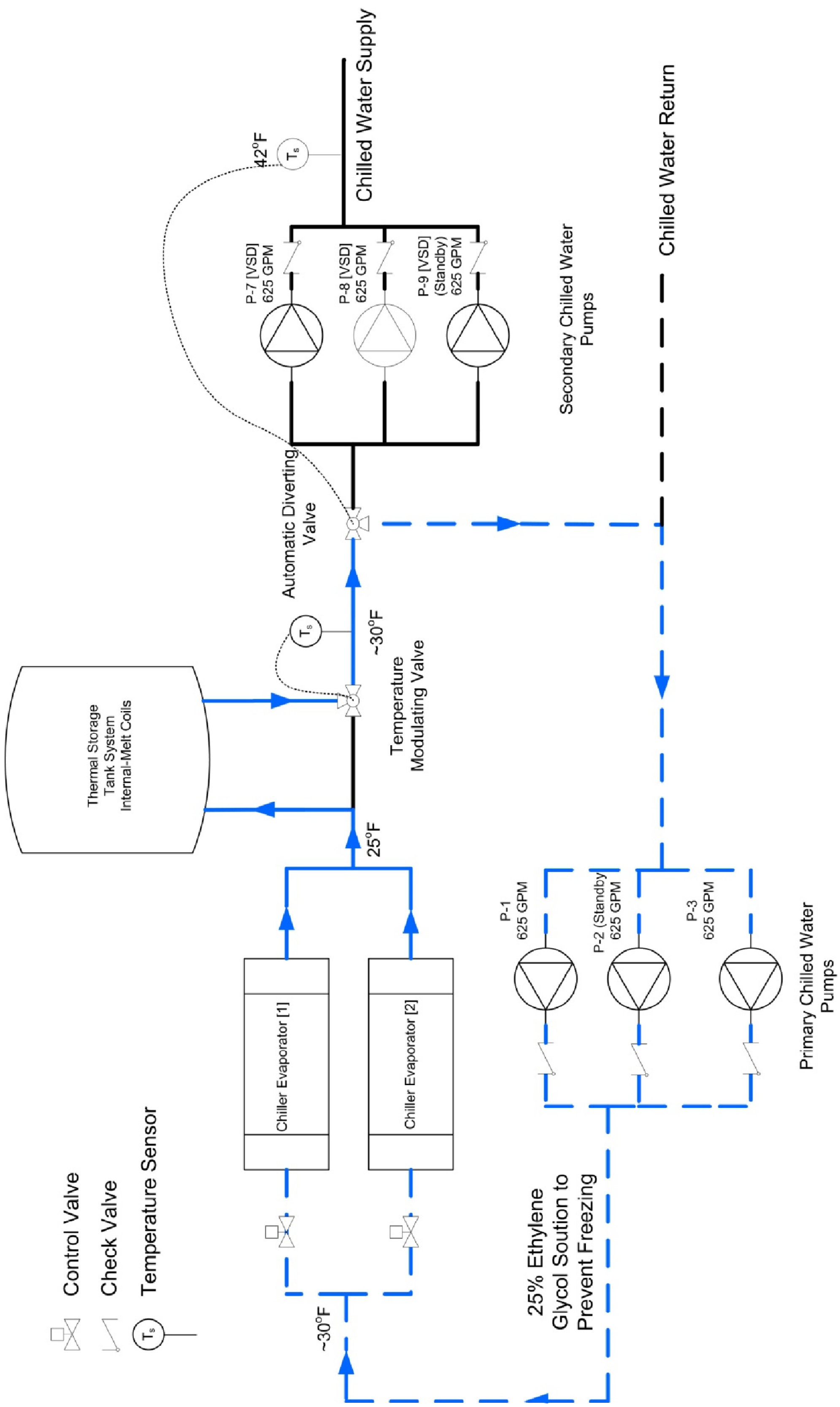

Figure 12 – TES Charging "ICE" Mode Schematic

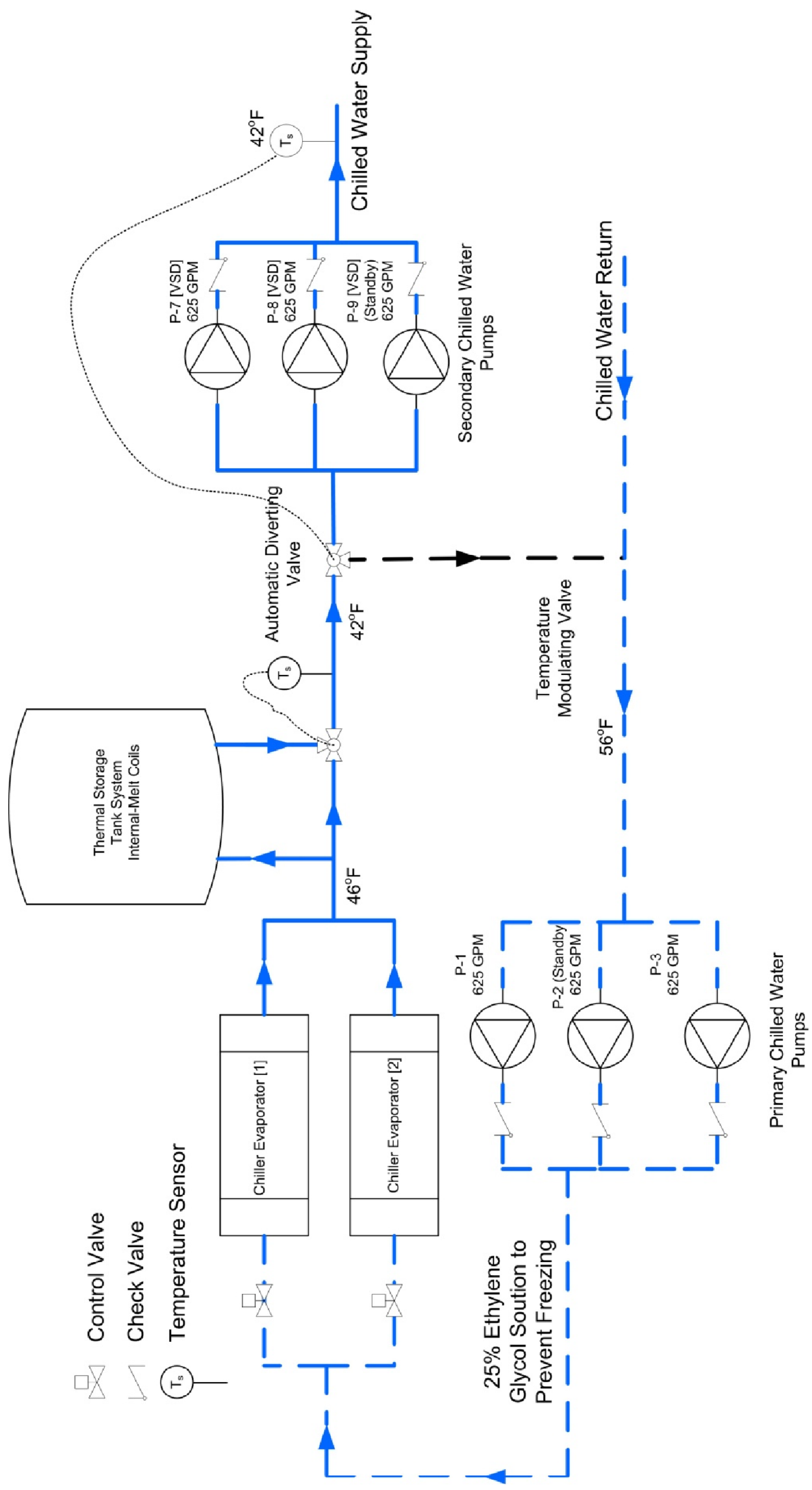

Figure 13 – TES Discharging "HVAC" Mode

Thermal Storage Energy Analysis

In order to learn the true benefit of the Thermal Ice Storage System, an energy analysis must be performed for both the original non-storage system and the TES system that was designed. The energy model must be very sophisticated and dynamic. Because the load is shifted so that chillers are running at night, cooling tower analysis must be included in the calculations to include the benefit of lower wet bulb temperatures that occur during nighttime hours. According to the Typical Meteorological Year 2 (TMY2) data for the location of the Integrated Sciences Building, there can be as much as an 8°F swing in wet bulb temperatures between the day and night during the summer cooling season.

Another variable that must be included in the analysis is that the chiller energy demand changes as the load fluctuates. DOE2 Chiller models should be used to determine the actual kW/ton rating of the chiller at each loading condition. Because chiller efficiency also depends on the condenser water temperature, which in turn depends on the wet bulb temperature, the model can be very complex.

Perhaps one of the most important variables that can be difficult to simulate manually is the complexity of the control systems which are possible with a Thermal Storage system. In order to optimize the system so that the HVAC mode operates to the best of its ability, the low wet bulb temperature is utilized as much as possible, and the thermal storage capacity is used to flatten the load in the most ideal way, a very complex algorithm must be developed based on weather data and wet bulb temperature reset. Unfortunately, this was outside the realm of expertise for this report.

Attempts were made to formulate a program in EES that would account for all of these variables. However, the complexity of the many variables as well as the relative inexperience in programming resulted in a program which produced numbers that could not be consistently correct. As a result, the thermal storage system was modeled in Trane TRACE 700 software. TRACE is a “black box” software program that regrettably does not offer the chance to see exactly how the calculations are formulated. Therefore, the results generated by the program must be treated with reasonable skepticism. If more time and experience was available for this analysis, a complex EES program would be ideal.

After modeling the thermal storage system in EES, and several modifications to the configuration, several sets of results emerged that were valuable. TRACE has different types of control settings that can be applied to Thermal Storage systems. There is a method named “storage priority”, where the thermal storage tanks try to meet the entire cooling load, and mechanical cooling is brought in to satisfy any excess chilled water demand. There is another mode which can be used to specify a maximum amount of mechanical cooling, and when that level is reached, the thermal storage capacity meets the remaining load. The control method which was ultimately the most successful, and did the best job of integrating the many variables was a method named “Optimized.” The Optimization control method is a strategy where software is used to forecast the amount of storage capacity which is necessary to meet the cooling load for each day. If the thermal storage capacity is not enough to satisfy the chilled water demands, it determines the amount of mechanical cooling is required and adds that to the control methodology. The optimized control was determined to be the most ideal method after several trials and examination of energy analysis reports.

Thermal Storage vs. Non-Thermal Storage Energy Results

After discovering several errors in the original TRACE energy model simulation and experimenting with the Thermal Storage control configurations, energy results from the simulations produced results that seemed intuitively accurate.

Electricity Demand

As expected, the peak electrical demand for each month decreased so that the overall electricity bill substantially decreased over the course of a year. Fortunately, TRACE generates many reports of energy consumption data. Some of the reports that were most useful in determining TES energy performance were the breakdowns of consumption and peak demand energy for the various mechanical equipment in the Integrated Sciences Building. By identifying the pieces of equipment that were affected by the modifications made by thermal storage and summing the energy data for each component, it was very simple to compare the two systems. This information is shown for the Non-Thermal Storage system in Table 19 and for the Thermal Storage system in Table 20.

Non-Thermal Storage Chiller Plant Energy Demand (kW)												
Component\Month	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec
Chiller #1	64.2	64.7	68.6	185	198.6	212.7	233.9	240.3	208.6	204.4	209.4	68.5
Cooling Tower #1	11.2	24.1	24.1	24.1	24.1	24.1	24.1	24.1	24.1	24.1	24.1	24.1
CV Chilled Water Pump #1	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4
CV Condenser Pump #1	22.6	22.6	22.6	22.6	22.6	22.6	22.6	22.6	22.6	22.6	22.6	22.6
Chiller #2	0	0	0	0	163.2	212.7	233.9	240.3	195.6	160.9	0	0
Cooling Tower #2	0	0	0	0	24.1	24.1	24.1	24.1	24.1	24.1	0	0
CV Chilled Water Pump #2	0	0	0	0	10.4	10.4	10.4	10.4	10.4	10.4	0	0
CV Condenser Pump #2	0	0	0	0	22.6	22.6	22.6	22.6	22.6	22.6	0	0
Billable Demand (kW)	108.4	121.8	125.7	242.1	476	539.6	582	594.8	518.4	479.5	266.5	125.6

Table 19 – Non-Thermal Storage Chiller Plant Electrical Demand breakdown by component. Total Billable Demand per month at bottom (red).

Thermal Storage Chiller Plant Energy Demand (kW)												
Component\Month	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec
Chiller #1	73.8	77	77	83.6	120.2	127.2	128.7	132.4	127.2	120.8	84.4	79.2
Cooling Tower #1	12.8	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2
CV Chilled Water Pump #1	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6
CV Condenser Pump #1	12.2	12.2	12.2	12.2	12.2	12.2	12.2	12.2	12.2	12.2	12.2	12.2
Chiller #2	73.8	77	77	83.6	120.2	127.2	128.7	132.4	127.2	117.1	84.4	79.2
Cooling Tower #2	12.8	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2
CV Chilled Water Pump #2	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6
CV Condenser Pump #2	12.2	12.2	12.2	12.2	12.2	12.2	12.2	12.2	12.2	12.2	12.2	12.2
Billable Demand (kW)	208.8	216	216	229.2	302.4	316.4	319.4	326.8	316.4	299.9	230.8	220.4

Table 20 - Thermal Storage Chiller Plant Electrical Demand breakdown by component. Total Billable Demand per month at bottom (red).

The Billable demand figures calculated in Table 13 and Table 14 arranged so they can be compared more visually in Figure 14.

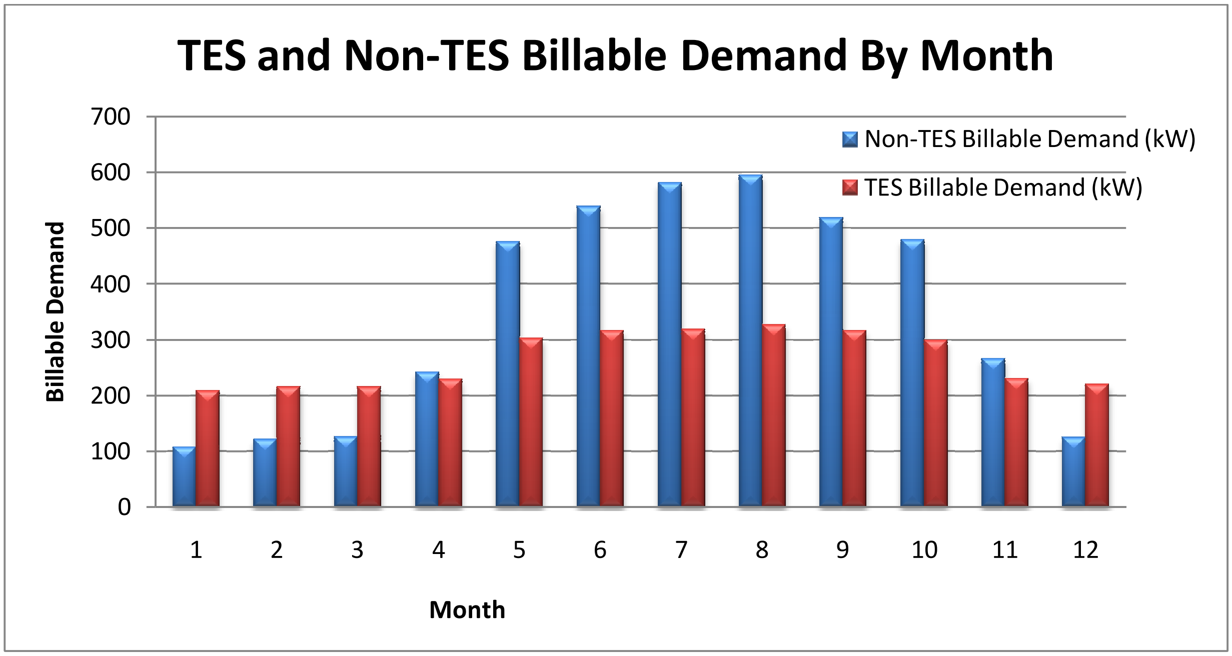

Figure 14 – Visual Comparison if Total Billable Demand per month comparing TES and non-TES systems.

Annual electrical demand difference between the original non-system and the thermal storage system produced a net benefit of 1348.7 billable kW. At $8.97 per kW, that savings comes to $10,555.00 annually, not including price fluctuation. This calculation is shown in Table 21.

Monthly Demand Charges	Non Storage		Storage	
	kW	Demand Fee	kW	Demand Fee
January	108.4	$972.35	208	$1,865.76
February	121.8	$1,092.55	216	$1,937.52
March	125.7	$1,127.53	216	$1,937.52
April	242.1	$2,171.64	229.2	$2,055.92
May	476	$4,269.72	302.4	$2,712.53
June	539.6	$4,840.21	316.4	$2,838.11
July	582	$5,220.54	319.4	$2,865.02
August	594.8	$5,335.36	326.8	$2,931.40
September	518.4	$4,650.05	316.4	$2,838.11
October	479.5	$4,301.12	299.9	$2,690.10
November	266.5	$2,390.51	230.8	$2,070.28
December	125.6	$1,126.63	22.4	$200.93
Annual Billing Demand kW	4180.4	$37,498.19	2795.7	$26,943.19
Net Benefit			1384.7	$10,555.00

Table 21 – Net billable demand savings and financial savings.

Electricity Consumption

Due to the decreased efficiency during ice making, the thermal storage system does consume more electricity per year than the original system. This information is provided TRACE outputs in the same format as the Demand output for each piece of chiller plant equipment. The data in Table 22 shows that the consumption is nearly an extra 200,000 kWh per year with the Thermal Storage system. As mentioned above, this is largely due to the ice making efficiency penalty.

Energy Costs		
	Original Energy Consumption	TES Electricity Consumption
Annual Energy Consumption	685,734 kWh	875,578 kWh
Consumption Costs	$33,052.36	$42,202.85
Net Loss		$9150.49

Table 22 – Annual Energy Consumption for Non-Storage and Thermal Storage Systems

Life Cycle Cost Analysis

As with any major system overhaul or redesign, the Life Cycle Cost should be a major factor in the decision to proceed with a certain alternative. Thermal storage systems are no exception. To gauge the net financial benefit of the Thermal Storage System, a 30-year Life Cycle Cost Analysis was performed based on the knowledge available. It included the initial cost of material and construction. This included the cost of the Chiller plant, thermal storage tanks, three-way valves, above-ground and underground piping to and from the thermal storage tanks, concrete foundation, burial, and fencing around the tanks. A full spreadsheet of the included cost data is provided in Appendix C. Some assumptions were made about the costs due to the lack of detail in RS Means for this type of intense design. Chiller plants were estimated at $1500/ton installed, which included all chiller auxiliaries, pumps, cooling towers, valves, and piping. The cost of the thermal storage tanks was estimated at $150/ton-hour as recommended by the CALMAC sales representative, which included the tanks, concrete slab, glycol solution, controls, and local pumping additions. Trenching was required for the underground piping as will be discussed in the Construction and Logistics Breadth Study below. The costs of excavation, pipe installation, and trench fill labor and materials were included. Also, because the tanks measure approximately 8 feet tall and will be placed next to the Integrated Sciences Building and a neighboring facility, it was decided that a slab on grade would have caused an unpleasant visual constriction in the area. Therefore, the tanks were buried 4 feet and placed inside a privacy fence to hide them to some degree. Slab excavation and fence costs were added to account for this.

First Cost

Unexpectedly, the final cost of the original system was determined to be more than the cost of the thermal storage system. This was due to the drastic reduction in chiller equipment and auxiliaries thanks to the thermal storage system. When chillers decrease in size, their cooling towers, fans, and condenser water pumps also decrease in size. The final cost figures are shown in Table 23 and the more complete version with details on the RSMeans data is provided in Appendix C. The thermal storage system first cost came to be over $215,000 less than the non-thermal storage system.

First Cost		
	Original System	TES
Chiller Plant	$1,642,500.00	$900,000.00
Tanks (Includes Slab, Glycol, Controls, Local Piping	$0.00	$437,400.00
3-Way Valve	$0.00	$3,000.00
A/G Piping & Insulation	$0.00	$13,090.00
U/G Piping & Insulation	$0.00	$62,400.00
U/G Piping Excavation	$0.00	$936.00
U/G Piping Fill	$0.00	$982.80
Concrete Pad Excavation (4-foot tank burial)	$0.00	$2,755.50
Privacy Fence	$0.00	$6,620.00
Total First Cost	$ 1,642,500.00	$ 1,427,184.30

Table 23 – Cost Breakdown of Non-Storage and Thermal Storage systems. (Details in Appendix C).

Maintenance Costs

As discussed above, one major benefit of the Thermal Storage system is that very little maintenance is required. Attempts were made to find a good source of information for estimating annual chiller maintenance. Although there were several credible sources, it was clear that chiller maintenance is very difficult to estimate, no matter the size, type, or configuration. After considering several sources, including one ASHRAE publication, reasonable estimates were decided at $15,000 per year for the TES system and $25,000 per year on the non-storage system. The reason for the difference was simply the size of the equipment and its auxiliaries.

Electrical Costs

Escalation factors for electricity in the region where the Integrated Sciences Building were from the NIST 2010 LCC Supplement. The same factors were used for the Demand costs as they were expected to ruse in accordance with consumption charges. It should be noted that the University organization who owns the building is expecting a 20-40% increase in electrical rates in the near future as a result of de-regulation of electricity rates in the region.

Total Life Cycle Cost Comparison

The total 30-year life cycle cost comparison is shown in Table 24. It shows a 30-year estimated benefit of nearly $450,000. A full spreadsheet showing the Life Cycle Cost Analysis is provided in Appendix C.

30-Year Life Cycle Cost Breakdown		
LCC 30-year Net Present Value	Non-Storage	Thermal Storage
Capital Costs	$1,642,500	$1,427,184
Maintenance	$509,572	$305,743
Electricity Costs	$1,457,096	$1,428,088
Total 30-year Life Cycle Cost	$3,609,168	$3,161,016
	Total 30-year Savings	$448,152

Table 24 – Thermal Storage and non-Thermal Storage System 30-year Life Cycle Cost Comparison

Emissions

Just as saving electrical energy reduces the amount of emissions created at power plants, increasing consumption adds to the emissions generated. Because the thermal storage system increases electrical consumption of the Integrated Sciences Building by nearly 200,000kWh, emissions are greatly increased, according to the calculation in Table 25.

Annual Emissions for Increase from TES		
Pollutant	Eastern Interconnection	TES Emission Increase per Year [lb]
Electric Use	1 kwh	189,844 kwh
CO_{2e}	1.74E+00	3.30E+05
CO_2	1.64E+00	3.11E+05
CH_4	3.59E-03	6.82E+02
N_2O	3.87E-05	7.35E+00
NO_x	3.00E-03	5.70E+02
SO_x	8.57E-03	1.63E+03
CO	8.54E-04	1.62E+02
TNMOC	7.26E-05	1.38E+01
Lead	1.39E-07	2.64E-02
Mercury	3.36E-08	6.38E-03
PM10	9.26E-05	1.76E+01
Solid Waste	2.05E-01	3.89E+04

Table 25 – Annual Emissions Increase as a Result of TES system.

Recommendation

Because of the significant benefit of the 30 year life cycle cost, as well as the energy demand savings of the thermal storage system, installing such an alternative is shown to have clear benefits. Adding the reduced maintenance requirements and ease of operation, thermal storage is a very good option for the Integrated Sciences building. What could be added, beyond convenience or cost savings, is the fact that reducing peak electrical demand is vital to the energy grid in the United States. As power plants run at peak load and the national grid begins degrading even further, reducing electrical demand is a very valuable move for utility companies. They often offer financial benefits for demand reduction, and although individual rebates are based up on specific case and were not available for this study, the possibility of further financial assistance remains.

MAE Coursework

Both of the mechanical depths discussed in this report use information taken directly from MAE-level graduate courses. The Variable Primary Flow pumping flow system information and configuration methodology was presented in AE 557, Centralized Cooling Production and Distribution Systems, by Dr. William Bahnfleth.

Along with schematics and control discussion, a parametric energy study was provided that compared VPF systems of various sizes, locations and building types to similar Primary/Secondary systems. Although the concept of VPF systems is very simple, the parametric study uncovered some of the complexities of the system, limitations of the chiller equipment with respect to flow rate modulation, and the effect of VPF operation on auxiliaries. There was also a great deal of information presented about pump selection and operation, as well as parallel pump staging.

Information was also presented in AE 557 about the basics of thermal storage systems. Discussion focused on the different types of storage systems, including full and partial storage, as well as sensible and latent thermal storage. Different techniques for control systems were discussed, as well as rules of thumb for sizing the capacity of the system. Information was presented about how savings are achieved based on different operation types and electrical rates. Although the thermal storage information was near the end of the semester, and no homework problems were assigned, the basics of system operation were thoroughly covered with plenty of references.

Additionally, the entire AE 557 course provided information on chillers, cooling towers, condenser water systems, and distribution systems that were used in determining the performance of both the VPF and thermal storage systems. Without the knowledge provided by Dr. Bahnfleth in AE 557 and during consultations during the project, this report may not have been possible.

Solar Photovoltaic Panel System (Electrical Breadth)

Because the Mechanical Depth studies for this project have so far involved the reduction of electricity consumption or a modification to how electricity is used at the Integrated Sciences Building, an analysis of Solar Photovoltaic panels and their potential benefit fits directly into the goals of the project. The VPF study seeks to reduce electrical consumption, largely by reducing chiller plant pumping energy. The thermal storage study does not necessarily reduce electrical consumption, but it significantly reduces the electrical demand of the Integrated Sciences Building and, along with the potential for utility company rebates, reduces the annual electrical bills. By producing electricity the Photovoltaic system will contribute to the goals of the other systems as well.

The size of the photovoltaic array for the Integrated Sciences Building was determined to be a maximum of about 60-90kW. This is a very large photovoltaic system in comparison to other buildings located in an urban environment. The system could be larger if more efforts were made to maximize the total available roof area by the mechanical engineers and architect. There are several protrusions located in the wide open planes of the roof that may be able to be relocated in pursuit of more area for photovoltaic panels. Likewise, the penthouse of the Integrated Sciences Building rises a full level above the other half of the roof. It also happens to be on the west side of the roof, which would severely inhibit that effectiveness of any solar cells on the eastern portion of the roof during the afternoon hours. Both of these factors contributed to the 80kW array size. However, modifications to the building profile, if the architects and engineers desired to pursue them, could increase the potential area for solar panels.

The solar panel which was chosen for this study was based on the fact that there was still a very large amount of area that must be covered in order to reach the 89kW array. The BP 3230T is one of the larger modules produced by BP Solar. It produces 230-Watts with a fully loaded efficiency of 13.8%. This efficiency is one of the best available for comparable modules at the time of the study and it also has good part load efficiency with only a 5% reduction at 20% solar intensity. The main undesirable characteristic of this module is the weight. It averages about 5 pounds per square foot (psf). However, many structural engineers include factors of safety and extra allowances for added loads that would likely be enough to allow for this weight. If this system were examined more closely for an actual installation, instead of a feasibility study, structural burden would undoubtedly be an area that would require consultation of a structural engineer and closer analysis. A BP 3230T product information sheet is shown in Appendix D. To see a simple schematic of how the system and its components would be arranged, see Appendix D.

This analysis also does not include a battery storage system for electrical energy storage. Because the electrical production hours of this system are during the main hours of operation, and because electrical demand is consistently above 1000kW during every month of the year, it is unlikely that the system will be producing electricity during hours when the energy is not used. In addition, the Integrated Sciences Building is in a locality that legally requires net metering of any electrical generating photovoltaic system. Therefore, even if there is an unlikely surplus of energy generated, the energy could be sold back to the grid and not wasted.

To perform the solar cell annual generation performance analysis, data from the National Renewable Energy Laboratory (NREL) was used. NREL provides data for many locations around the United States about how many peak sun hours per day there are on average per month. A "peak sun hour" is the number of hours required for a day's total irradiation to accumulate at a peak sun condition. It also makes adjustments according to location to account for the amount of solar energy available for conversion to direct current electricity.

The energy rate for electrical energy which was used to determine the payback period was the $0.0482/kWh that was used for all other analyses. This rate is well below the average rate for the region, which was given by NREL as $0.1051/kWh. The higher energy rate would help to improve the payback period. Ultimately, the payback period was determined by other incentives and the sale of Solar Alternative Energy Credits.

The tilt angle for the solar system was calculated for three different options. Because the roof is flat, a flat tilt angle was first used to measure the amount of benefit for the most solar panels and simplest arrangement on the roof. Next, a 10° tilt angle was used to ensure that close spacing of the modules would still allow for the closest possible spacing of panels. Finally, a 33° tilt angle was used because NREL data indicated this was the ideal tilt angle for collection at the latitude of the Integrated Sciences Building. However, the 33° tilt angle system was reduced to 69kW because of the required spacing of the panels. Panels must be spaced further apart when the tilt angle is larger because of shading issues associated with panels that are fixed at more vertical positions. Of course, more space between modules reduces that amount of modules that are able to fit onto the roof.

After performing the calculations as to which configuration of solar array size and tilt angle, the 10° array tilt system allowed for the best balance of tilt angle and array size to produce the maximum amount of electricity in the space provided. Therefore, the calculations shown Table 26 in report reflect those for the 10° fixed tilt and 80kW array size.

10° Fixed Tilt			
Month	Peak Sun Hours (kWh/m^2/day)	Days/month	kWh/month
1	2.41	31	4604.437068
2	3.18	28	5487.606432
3	4.65	31	8884.07982
4	5.26	30	9725.34024
5	5.98	31	11425.1177
6	6.36	30	11759.15664
7	6.02	31	11501.5399
8	5.67	31	10832.84572
9	4.91	30	9078.21684
10	3.8	31	7260.10824
11	2.6	30	4807.2024
12	2.18	31	4165.009464
Year	4.42	365	99428.96964

Table 26 – Solar PV Panel Energy Generation potential based on NREL PVWATTS calculation method.

Life Cycle Cost

Financial incentives played a large role in determining the cost effectiveness of a solar panel system. There are many state and federal incentives and tax deductions available for commercial solar panel electricity production. Because solar panel systems traditionally are not able to recoup the full investment required to install such a system, financial incentives from state and federal governments are normally necessary in order to justify their installation. Five different incentives and tax credits were investigated for the Integrated Sciences Building in order to determine the payback period and a 15-year life cycle cost. Certain assumptions about inflation rates and the potential continuation of some programs were necessary to estimate the 15-year financial performance of the system. The five incentives and tax deductions are described below.

Federal MACRS (Modified Accelerated Cost Recovery System Depreciation Tax Deductions

The federal MACRS program allows business and commercial building owners to recover losses due to property deprecation in certain property upgrade investments. The policy establishes different classes of investments for which tax deductions can be made. Solar Photovoltaic systems are classified as a system that depreciates over the first 5 years of the system. Therefore, the tax policy states that in the installation year, and the five following years, the system will depreciate at 10%, 32%, 19.2%, 11.52%, 11.52%, and 5.76% respectively. The owner is entitled to a reimbursement of this percentage of the original net cost of the system after state and federal grants (Pennsylvania Sunshine Rebate and Federal Investment Tax Credit) multiplied by their income tax rate. Table 27 shows the MACRS Tax deduction potential and assumes a net system cost of $393,288.00 and a tax rate of 35%.

MACRS (Modified Accelerated Cost Recovery System Depreciation Tax Deductions)		
Depreciation Year	Net System Cost	$393,288.00
2011	10.00%	$13,765.08
2012	32.00%	$44,048.26
2013	19.20%	$26,428.95
2014	11.52%	$15,857.37
2015	11.52%	$15,857.37
2016	5.76%	$7,928.69

Table 27 – MACRS Tax Deduction Calculations

Pennsylvania Sunshine Solar Rebate Program

This state-run incentive program is budgeted to pay $0.75/kW for the first 10kW of solar PV systems and $0.50/kW for the rest up to 90kW. This program has a budget of $100 million. This analysis assumes these funds will be available. A maximum rebate of $52,500.00 limits the peak incentive amount. For an 80kW system, Table 28 shows the total rebate amount is $42,500.

Pennsylvania Sunshine Solar Rebate Program		
	Rebate $/kW	Rebate Amount
First 10kW	$0.75	$ 7,500.00
Next 70kW	$0.50	$ 35,000.00
	Total	$ 42,500.00

Table 28 – Pennsylvania Sunshine Solar Rebate Program Refund

Pennsylvania Public Utilities Commission - Solar Alternative Energy Credits

According to Pennsylvania's Alternative Energy Portfolio Standard (AEPS), each electric distribution company and electric generation supplier to supply 18% of its energy using alternative energy resources by 2021. It also requires a certain percentage of this to be supplied by photovoltaic systems. This percentage is to be ramped up from 0.0120% in 2009 to an ultimate target of 0.5% in 2020. According to Pennsylvania law, a Solar Alternative Energy Credit (SEAC) is represents proof that 1 MWh of electricity was generated by a qualified PV facility. Certification for this ability requires that the facility is within Pennsylvania and is traditionally fairly easy to obtain. The amount of compensation for SEACs varies based on the market. During April of 2010, it was reported that certificates were approximately $400/MWh or $0.40/kWh. Prices peaked at $600/MWh or $0.60/kWh. Assuming a rate of $0.40/kWh for this credit and annual production calculated at 99,429 kWh, this incentive could annually generate $39,772.00 in benefit towards financing the photovoltaic system at the Integrated Sciences Building.

Federal Renewable Energy Production Incentive (REPI)

This was a policy enacted by the Energy Policy Act of 1992. The Federal Renewable Energy Production Incentive provides payment of $0.015 per kilowatt-hour, adjusted to 1993 dollars for inflation, to qualifying systems for the first 10 years of their operation. Table 29 shows the adjusted rates per kilowatt-hour and the yearly incentive amount based on annual generation of 99,429 kWh.

Year	Rate/kWh	Incentive Amount
2011	$ 0.0230	$ 2,286.87
2012	$ 0.0234	$ 2,328.03
2013	$ 0.0238	$ 2,369.93
2014	$ 0.0243	$ 2,412.59
2015	$ 0.0247	$ 2,456.02
2016	$ 0.0251	$ 2,500.23
2017	$ 0.0256	$ 2,545.23
2017	$ 0.0261	$ 2,591.05
2018	$ 0.0265	$ 2,637.69
2019	$ 0.0270	$ 2,685.16
Total		$ 24,812.80

Table 29 – Federal REPI Incentive Calculations

Federal Business Energy Investment Tax Credit (ITC)

The Federal Business Energy Investment Tax Credit was expanded by the Energy Improvement and Investment Act of 2008 and The American Recovery and Reinvestment Act of 2009. This policy states that until 2016, solar systems that generate electricity are eligible for a credit equal to 30% of expenditures, with no maximum credit. By the calculations of this report, the 80kW system photovoltaic panels cost a total of $236,640.00 and the installation cost is $400,200.00. The total system cost comes to $636,840.00. Therefore, the total tax credit available is $191,052.00.

The total cost analysis of this system is slated to have a net balance which turns positive in year 5 after installation, assuming all incentives previously discussed are available for this system. Based on this information, it is recommended that the Integrated Sciences Building owner seriously consider the installation of a solar photovoltaic system. Calculations performed for the solar photovoltaic system can be found in Appendix ##. After 15 years, the estimated net benefit of the system comes to $401,248.71. These savings are based largely on the fact that the Solar Alternative Energy Credits total $39,772 per year. Without the SEAC credit, the system would likely not ever pay for itself. Figure 15shows the net cash flow over the first 15 years of system operation.

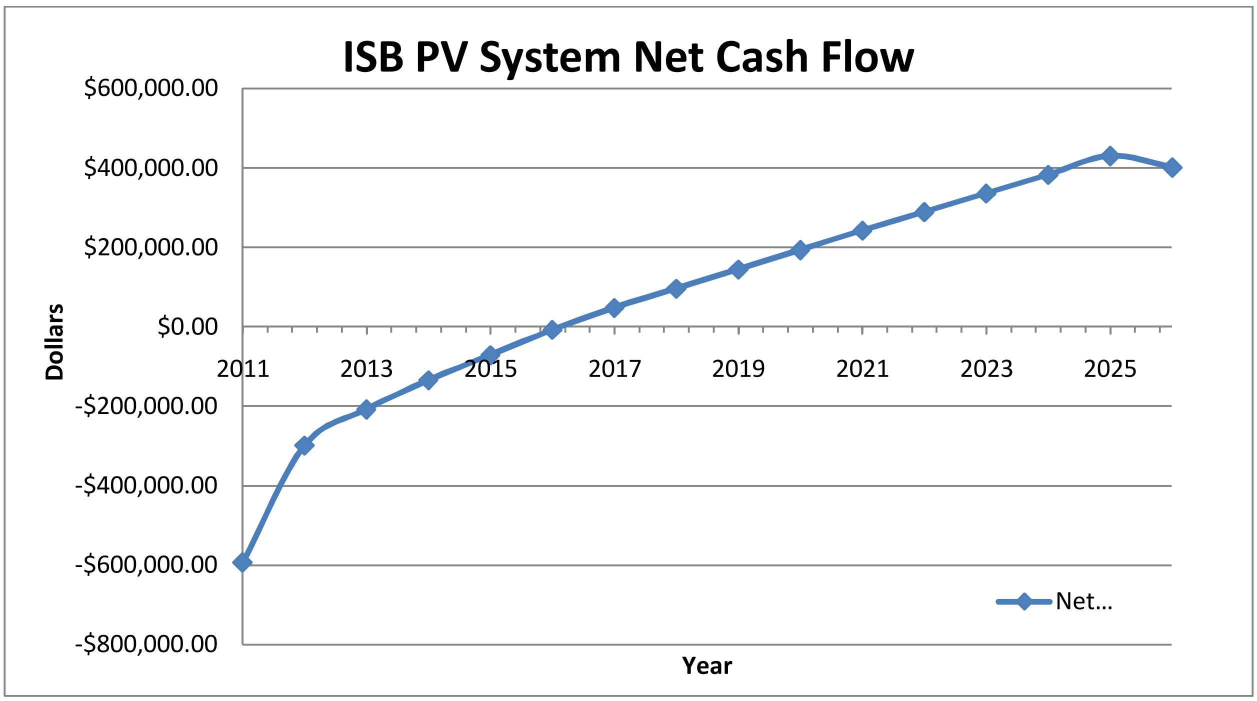

Figure 15 – ISB Photovoltaic System Cash Flow Diagram

Thermal Storage Logistics & Construction Study

Although the Thermal Storage system proposed for the purpose of this report was done largely for educational purposes to determine feasibility of the system size, operation, and cost justification, analysis of the construction logistics is a valuable study and one that deserves attention in order to determine whether the system is truly able to be a success.

Analysis of the Thermal Storage Logistics and Construction included the following parts:

- Thermal Storage Tank Location
- Verification & Resolution of Underground Commodity Clearance
- Determine if tanks will be buried or above ground
- Construction Timeline for Tanks & Components

Tank Location & Clearance

The first major task was to determine a location for the thermal storage tanks and decide what type of above-ground or under-ground arrangement would be suitable. Finding a location that had very no interferences with underground commodities while presenting the required open area for the tanks was difficult. Because the tanks weight approximately 50,600 pounds each, and the system design calls for six of them, it was important to find a location without piping or electrical systems directly below the slab that the tanks would sit upon. Luckily, there was just the right amount of space directly adjacent to the northeast side of the building. The following images show the proposed location of the storage tank foundation. Figure 16 shows the outline of the Integrated Sciences Building in blue and an outline where the second image's outline is in relation to the building and the site.

Figure 17 shows an outline of the space required for the thermal storage tanks in orange. It should be noted that the construction documents indicate a telecommunication conduit duct underground, shown in a crosshatch diagonal line in Figure 17, which could interfere with the thermal storage tank foundation. However, this underground conduit duct was slated for removal at the beginning of construction as a result of the other excavation and underground commodity rearrangement. As a result, it will not interfere with any foundation for these tanks. To the north of the proposed tank location, there is another conduit duct, and to the east there is a steam line. However, there is several feet of clearance at both places. The southwestern corner of the proposed tank foundation location comes very close to the northeast façade of the Integrated Sciences Building. Unfortunately, due to the urban environment where the building is located, clearances for buildings and commodities are less than ideal. In an actual installation, there may have been more interdisciplinary coordination between the architects, engineers, and other designers to help alleviate these close clearances. However, the purpose of this report is to determine feasibility.

For the purpose of this study, it was determined that the location is suitable. Any changes that would be required for excess clearance or more optimal spacing would likely have been resolved in the earlier stages of design if a thermal storage system was originally planned. Also worthy of mentioning, the concrete walk that is shown in Figure 17 was also slated for removal.

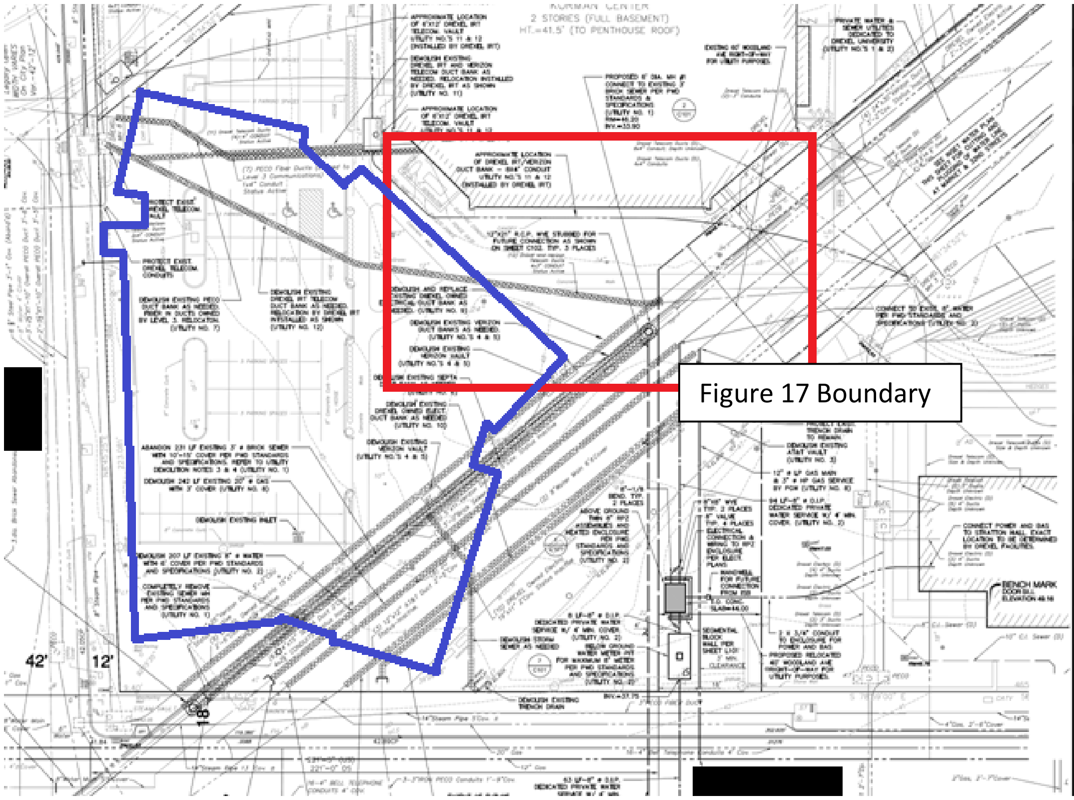

Figure 16 – Proposed Location of Thermal Storage Tank array (Red) in relation to the Integrated Sciences Building. See Figure 17 for a larger view outlined by red region.

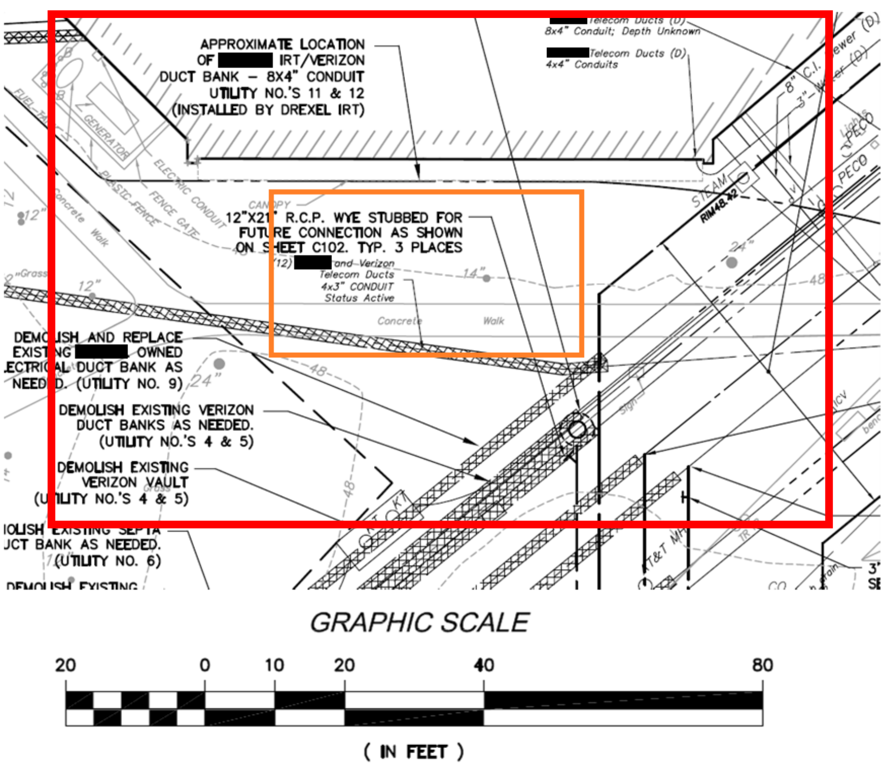

Figure 17 – Orange Box indicates outline of partially buried slab and fence line for thermal storage tank array.

Tank Burial

To determine whether the tanks will be buried, several factors must be considered. First, it should be noted that even when the tanks are buried, they are not completely hidden and out of the way for all purposes. The tanks, although built to withstand the outdoors, are not built to withstand the weight of concrete sidewalks, pavement, or normal foot traffic of any kind. In all applications where they have been buried, they are covered with sand and wood chips in order to hide them. Another downfall is that, even when buried, there is the likelihood that piping will protrude above ground as the supply and return chilled water. The only option for totally removing the tanks from view would be an underground vault. The next consideration for determining whether to bury the tanks is the cost issue. Although the cost of burial would not be exceedingly high, there is still an excavation cost associate with the digging and the slab that is required underground.

Ultimately, for the purpose of this project, it was decided that cost of a vault would be forgone in favor of leaving the tanks partially above ground. The fact that complete burial does not remove the piping from view, as well as the fact that when the tanks are buried, the space is still not usable, outweighed the costs for the purpose of this report. As explained in the Thermal Storage analysis section above, the tanks shall be buried to a depth of 4 feet, which leaves another 4 feet of tanks and piping above ground. This will be shielded from view by a vinyl privacy fence.

Different universities and building owners may prefer the burial of the tanks, or even the vault solution. The decision to bury tank is usually a matter of preference. The only foreseeable legal issue with the tanks as they have been designed for this report is that if the local governing entity deems them a violation of the construction permit or unsuitable, at which point a new solution would need to be investigated. However, it will be assumed for the purpose of this project that the tanks will be located on a grade-level slab and hidden from view by a tall fence.

Installation

The thermal storage tanks will be delivered by truck and have a delivery(empty) weight of 6000 pounds each. There will be six separate tanks. A single tank is actually made up of three smaller tanks that are attached with common piping, so each of the 6 tanks are actually "modules" of a set of three smaller ones. Upon arrival by the delivery truck, they will be moved into place on the slab foundation by the tower crane on site.

The crane that is located on site during construction has the proper length needed to move the tanks from the delivery truck to the final location. Special crane instructions were given by CALMAC, the tank manufacturer, for hoisting them into place. A diagram from the CALMAC Icebank® Ice Storage Installation and Operation Manual is shown in Figure 18. A special rigging bar, which is also shown in Figure 18 is available from CALMAC upon purchasing the tank to ensure that the tanks are rigged properly from hoisting.

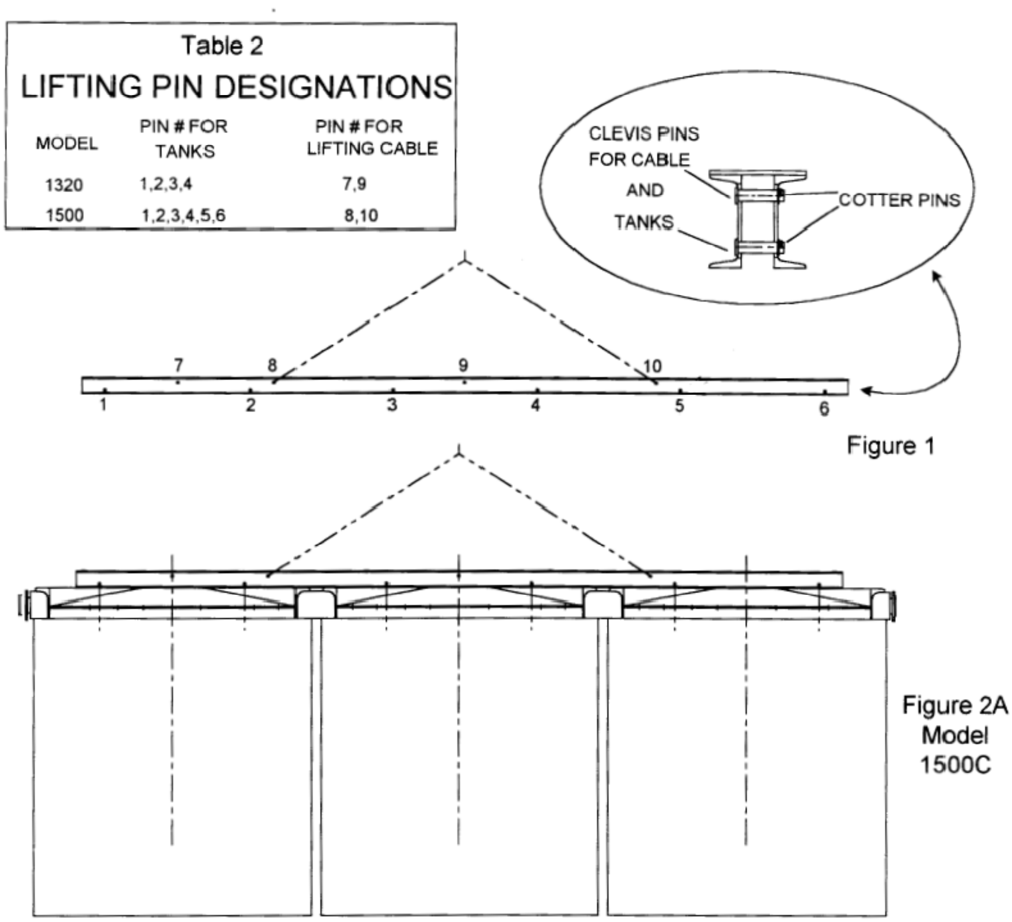

Figure 18 – CALMAC ICEBANK® Rigging diagram

Schedule & Labor

The last consideration for this study was the scheduling and analysis of the increased labor hours. Because the system is very simple and has relatively few components, the schedule for construction is not lengthy. Also, because the location of the tanks is left largely untouched, there is no reason that the excavation, slab construction, tank placement, and underground piping cannot be done during any time throughout the construction schedule.

Using RSMeans construction and cost data, data was collected for the following tasks related to the thermal storage system construction:

- Pad Excavation
- Concrete Preparation & Pouring
- Concrete Curing
- Tank Placement
- Tank Filling
- Fence Installation
- Fence Gate Installation

- Underground Piping Trench Excavation
- Underground Pipe Installation
- Pipe Bedding
- Trench Backfill
- Above ground Pipe Installation

Table 30 provides information from RSMeans as well as the calculation for the duration of each task listed above. While this list may be elementary, it provides a good understanding of what is necessary for installation of a system of this type and magnitude. This calculation indicates that an additional 375 man hours will be necessary to complete the construction of the main parts of the system. Skilled control system installation will be included as a part of the setup of the original system. Because the chiller plant on the Thermal Storage system is smaller, the extra hours that would have been spent setting up a large chiller plant can be spent configuring the controls and instrumentation of the thermal storage tanks and valves.

Task	Measure	Crew Daily Output	Man Hours/Unit	Total Units	Total Man Hours	Days
Pad Excavation	CY	108	0.148	167	24.716	2.000
Concrete Pour	SF	2505	0.029	1125	32.625	1.000
Concrete Cure	SF	85	0	1125	0	14.000
Tank Placement	each	10	10	6	60	1.000
Tank Filling	each	10	10	6	60	1.000
Fence Installation	Lin. Ft	150	0.16	140	22.4	1.000
Fence Gates	Each	9	2.667	2	5.334	1.000
Trench Excavation	L.F	700	0.011	312	3.432	1.000
U/G Pipe Installation	LF	200	0.25	356	89	2.000
Pipe Bedding	Lin. Ft	150	0.16	312	49.92	3.000
Trench Backfill	CY	1000	0.012	9	0.108	1.000
A/G Pipe Installation	LF	296	0.162	170	27.54	1.000

Table 30 – Scheduling Duration Calculation for each construction task.

A condensed schedule of the tasks listed above are is presented in Figure 19. This schedule is optimized, meaning that it is the best case scenerio for the list of tasks given. Because management styles are different and some unknown scheduling conflicts may always arise on a schedule, it cannot be guaranteed. The total length of the optimized schedule is 21 working days, which is just over 4 weeks.

As shown in Figure 20, the chiller plant equipment is scheduled for installation between August 2010 and Febraury 2011. This provides a window of opportunity of about 6 months during which the thermal storage components could be installed. This is plenty of time, given the simplicity of the items.

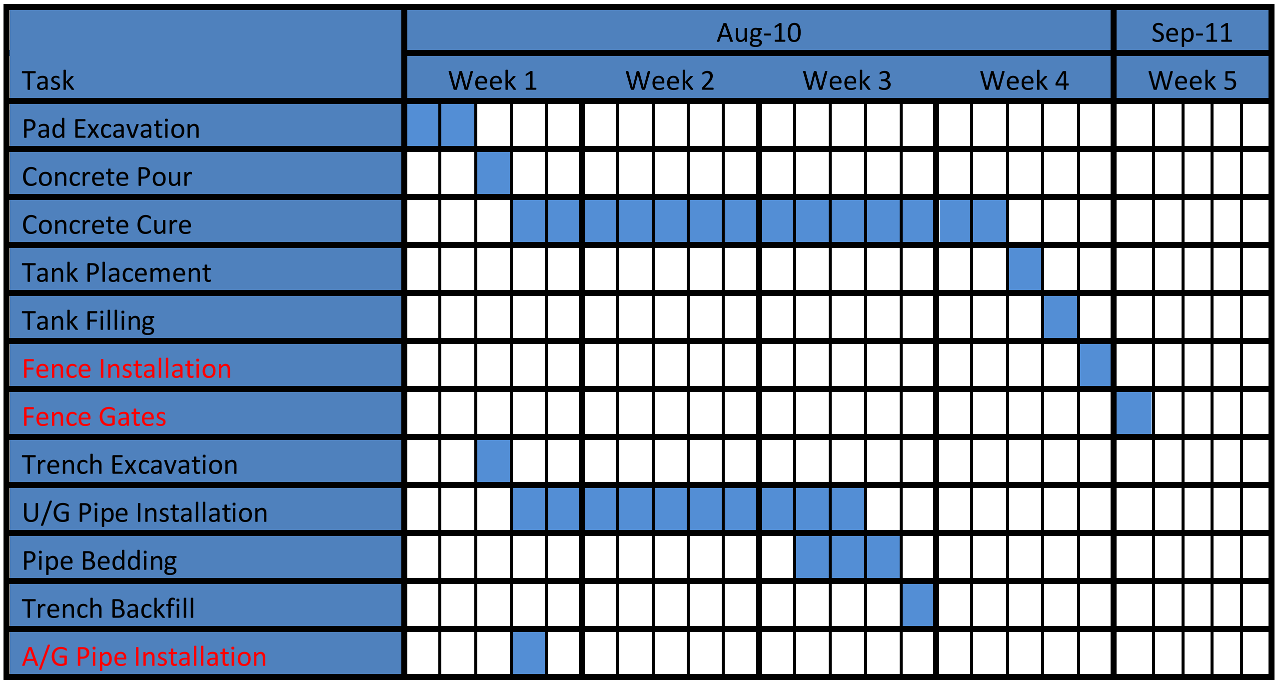

Figure 19 – Optimized Thermal Storage System Installation Schedule.

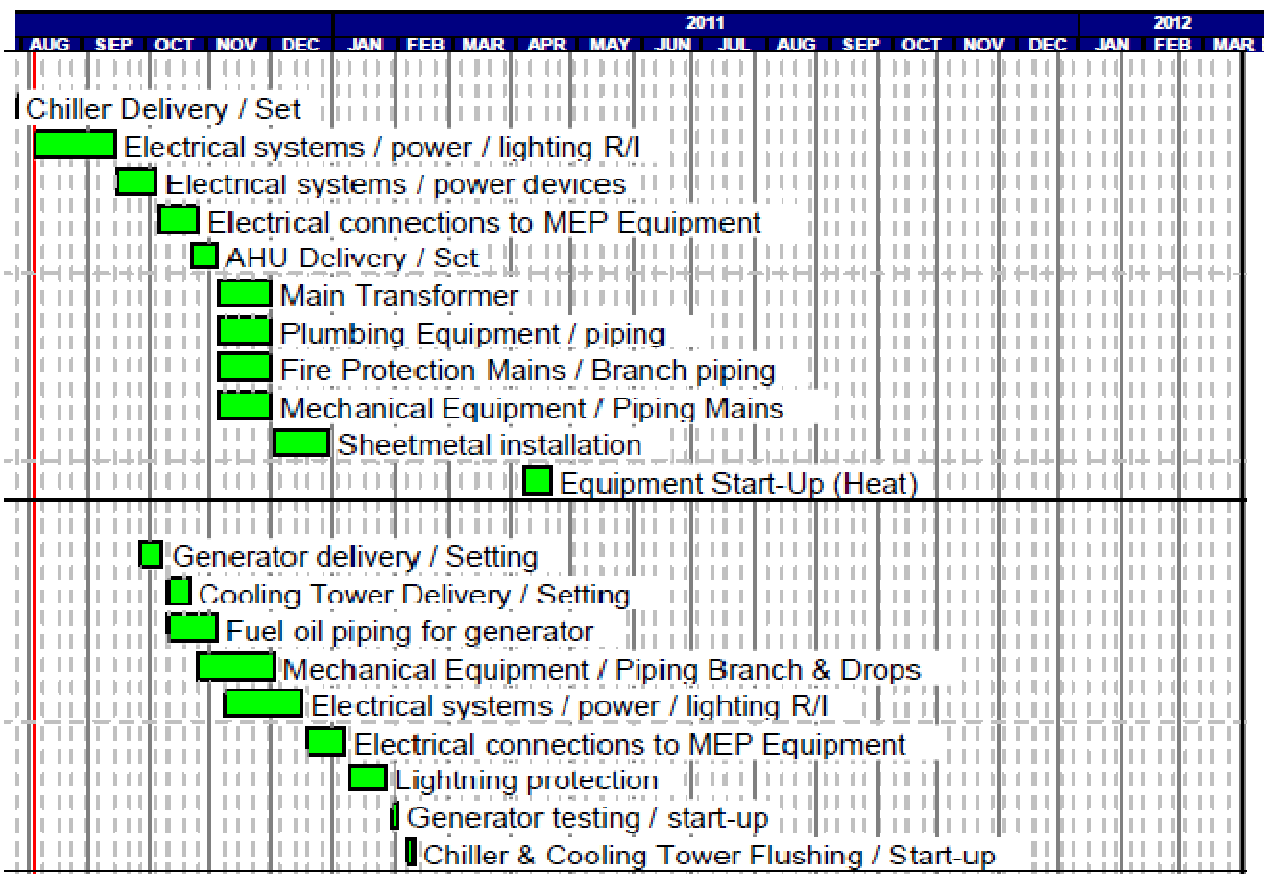

Figure 20 – ISB Construction Schedule for Chiller Plant Equipment August 2010 to March 2011.

Contruction & Logistics Conclusion

After Analyzing the construction tasks, tank location concerns, and scheduling issues associated with installation of the thermal storage system, there are no foreseen issues that would catastrophically affect the success of the thermal storage system. Of course, all construction projects have issues that arise and create trouble. However, for the purpose of this report, the construction and logistics analysis for the thermal storage tank system installation suggests the goals would be met and the project could be successful.

Conclusion

After performing the Variable Primary Flow, Thermal Storage, Construction and Logistics, and Solar Photovoltaic system analyses, it is clear than each investigation is worthwhile and all of the systems can be beneficial. Not only are the systems generally better for the electrical grid and the atmosphere, they are also better with respect to emissions release.

The only question still remaining after performing these studies and realizing the benefits is why more building owners and engineers are not pushing to include these types of design on more new buildings. The misconception that thermal storage and photovoltaic systems are too complex or not worthwhile is one that needs to be remedied in the future if energy efficiency and sustainable performance is truly a goal that engineers and owners intend on addressing.

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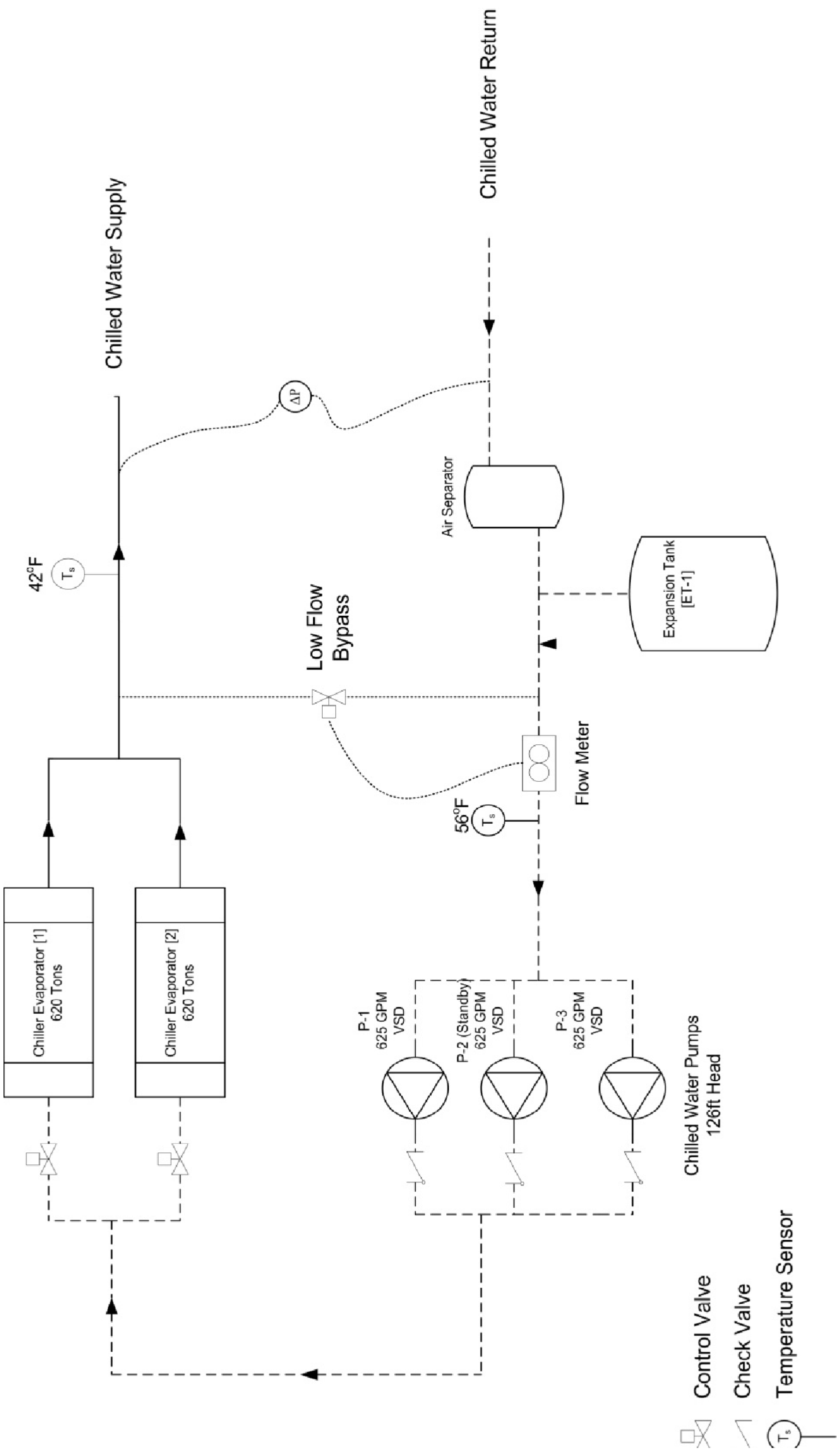
Appendix A– Integrated Sciences Building LEED Scorecard

Targeted	Not Pursued	LEED® Scorecard for Integrated Sciences Building	
10	4	Sustainable Sites	
•		SSP1	Construction Activity Pollution Prevention
1		SSC1	Site Selection
1		SSC2	Development Density & Community Connectivity
	1	SSC3	Brownfield Redevelopment
1		SSC4.1	Public Transportation Access
1		SSC4.2	Bicycle Storage & Changing Rooms
	1	SSC4.3	Low Emitting & Fuel Efficient Vehicles
1		SSC4.4	Parking Capacity
1		SSC5.1	Protect or Restore Habitat
1		SSC5.2	Maximize Open Space
	1	SSC6.1	SW Design - Quantity Control
1		SSC6.2	SW Design - Quality Control
1		SSC7.1	Heat Island Effect, Non-Roof
1		SSC7.2	Heat Island Effect, Roof
	1	SSC8	Light Pollution Reduction
4	1	Water Efficiency	
2		WEC1	Water Efficient Landscaping
	1	WEC2	Wastewater
2		WEC3	Water Use Reduction
9	8	Energy & Atmosphere	
•		EAP1	Fundamental Commissioning of Energy Systems
•		EAP2	Minimum Energy Performance
•		EAP3	Fundamental Refrigerant Management
6	4	EAC1	Optimize Energy Performance
	3	EAC2	On-Site Renewable Energy
1		EAC3	Enhanced Commissioning
	1	EAC4	Enhanced Refrigerant Management
1		EAC5	Measurement & Verification
1		EAC6	Green Power
7	6	Materials & Resources	
•		MRP1	Storage & Collection of Recyclables

	3	MRC1	Building Reuse
2		MRC2	Construction Waste Management
	2	MRC3	Materials Reuse
2		MRC4	Recycled Content Materials
2		MRC5	Regional Materials
	1	MRC6	Rapidly Renewable Materials
1		MRC7	Certified Wood
8	7		Indoor Environmental Quality
•		EQP1	Minimum IAQ Performance
•		EQP2	Environmental Tobacco Smoke Control
	1	EQC1	Outdoor Air Delivery Monitoring
	1	EQC2	Increase Ventilation Effectiveness
1		EQC3.1	Construction IAQ - During Construction
1		EQC3.2	Construction IAQ - Before Occupancy
1		EQC4.1	Low-Emitting Adhesives & Sealants
1		EQC4.2	Low-Emitting Paints & Coatings
1		EQC4.3	Low-Emitting Carpets
1		EQC4.4	Low-Emitting Composite Wood and Agrifibre
	1	EQC5	Indoor Chemical & Pollutant Source Control
	1	EQC6.1	Controllability of Systems - Lighting
	1	EQC6.2	Controllability of Systems - Thermal Comfort
1		EQC7.1	Thermal Comfort - Design
1		EQC7.2	Thermal Comfort - Verification
	1	EQC8.1	Daylight 75% of Spaces
	1	EQC8.2	Daylight 90% of Spaces
5	0		Innovation & Design Process
1		IDC1.1	Green Housekeeping
1		IDC1.2	Building Exterior & Hardscape Management
1		IDC1.3	Exemplary Performance - Water Efficiency
1		IDC1.4	Innovation & Design: Hydration Station
1		IDC2	LEED® Accredited Professional
43	26		Total Points

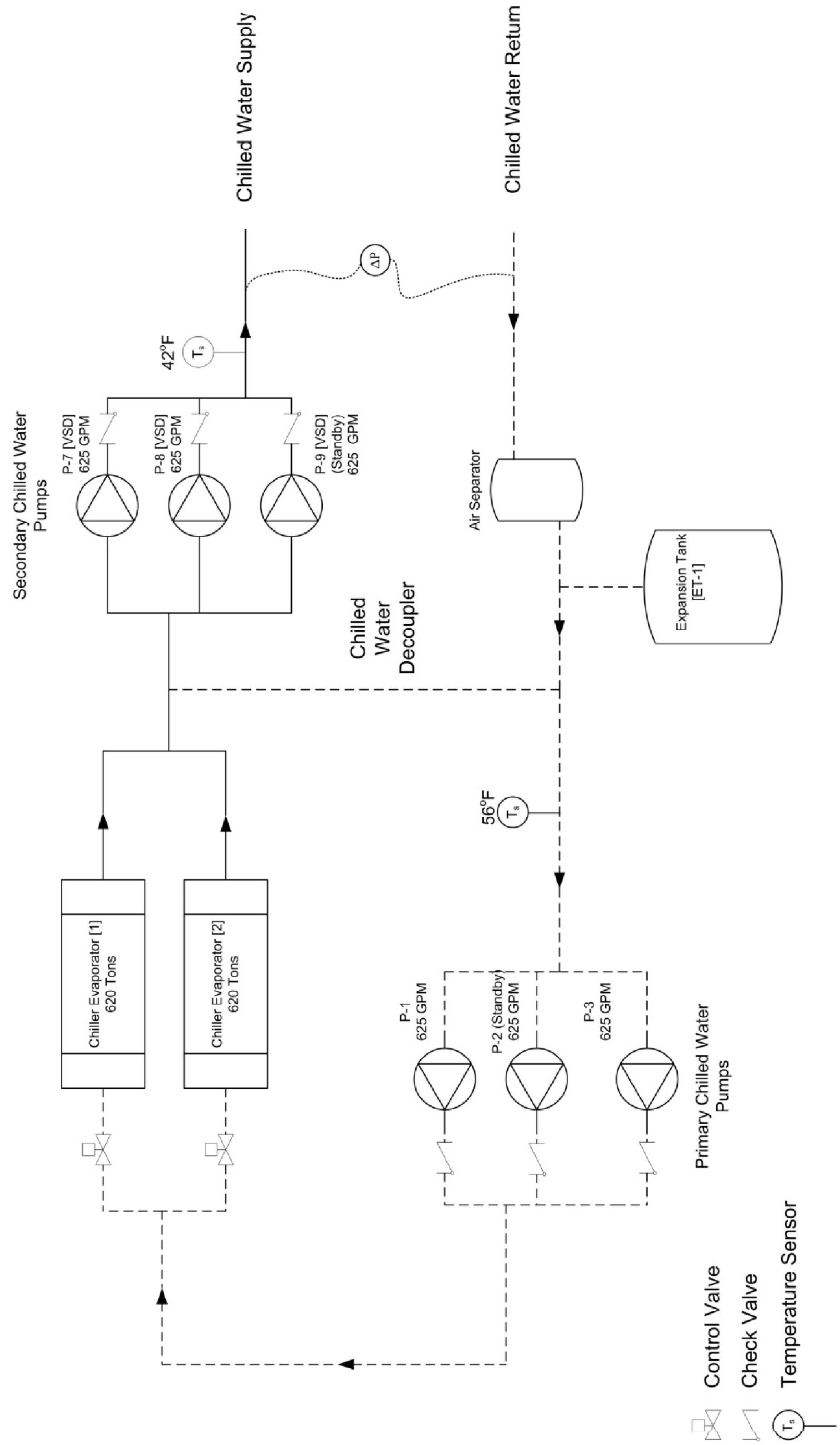
Appendix B - Primary/Secondary vs. Variable Primary Flow Data and Charts

VPF System Schematic



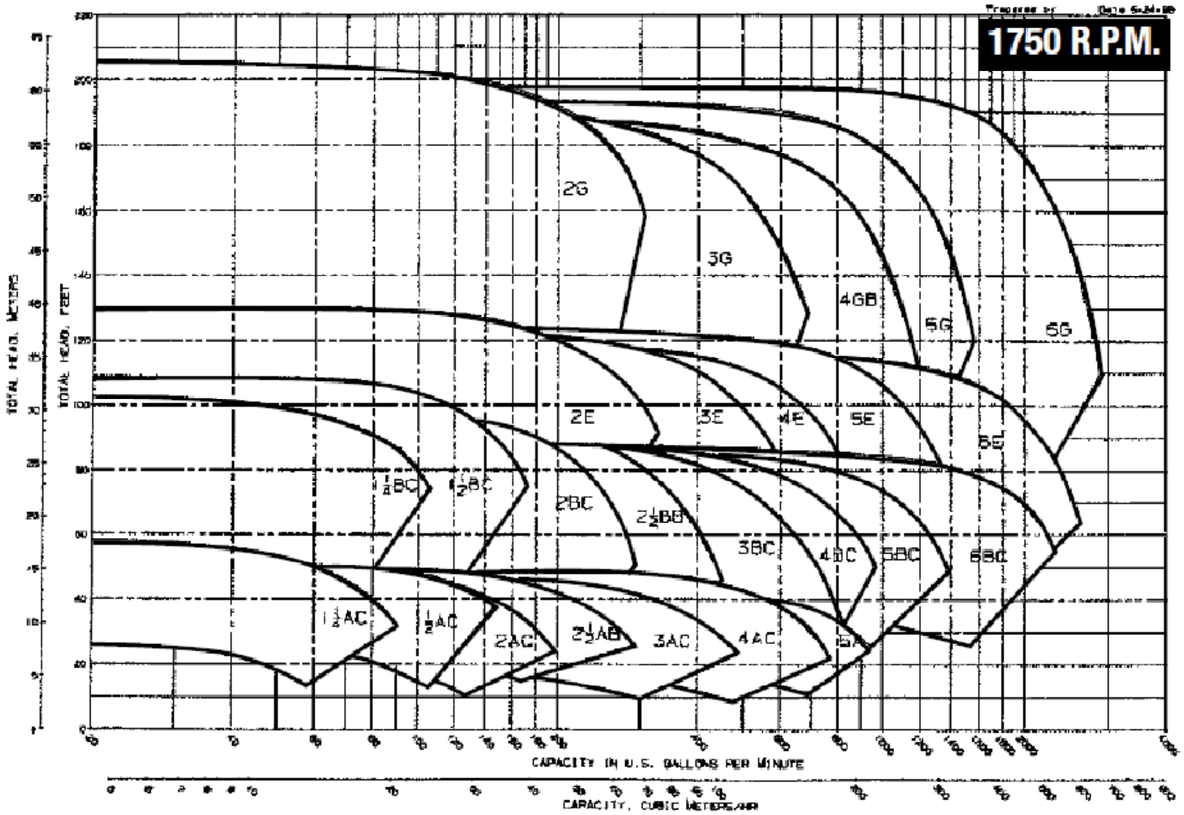
Appendix B - Primary/Secondary vs. Variable Primary Flow Data and Charts

Primary/Secondary System Schematic



Appendix B- Primary/Secondary vs. Variable Primary Flow Data and Charts

Figure 3 – Bell & Gossett Pump Model 1510, 1750 RPM Pump Selection Guide



Appendix B - Primary/Secondary vs. Variable Primary Flow Data and Charts

Figure 4 – P/S Primary Pump Curve – Bell & Gossett Model 1510-3BC

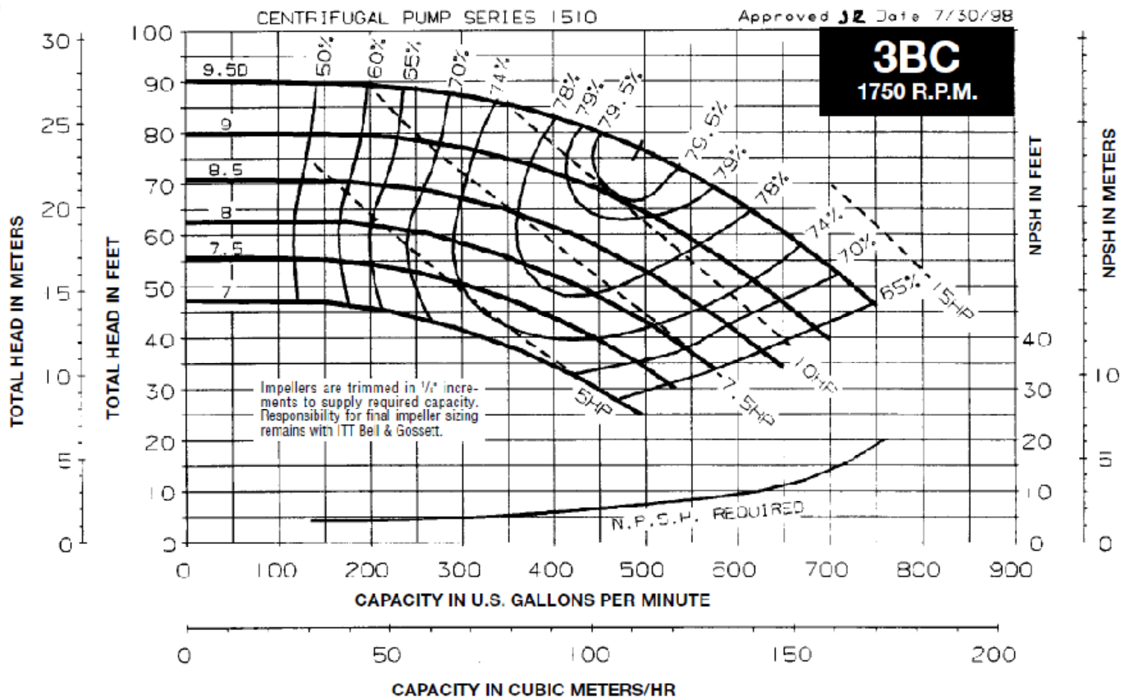
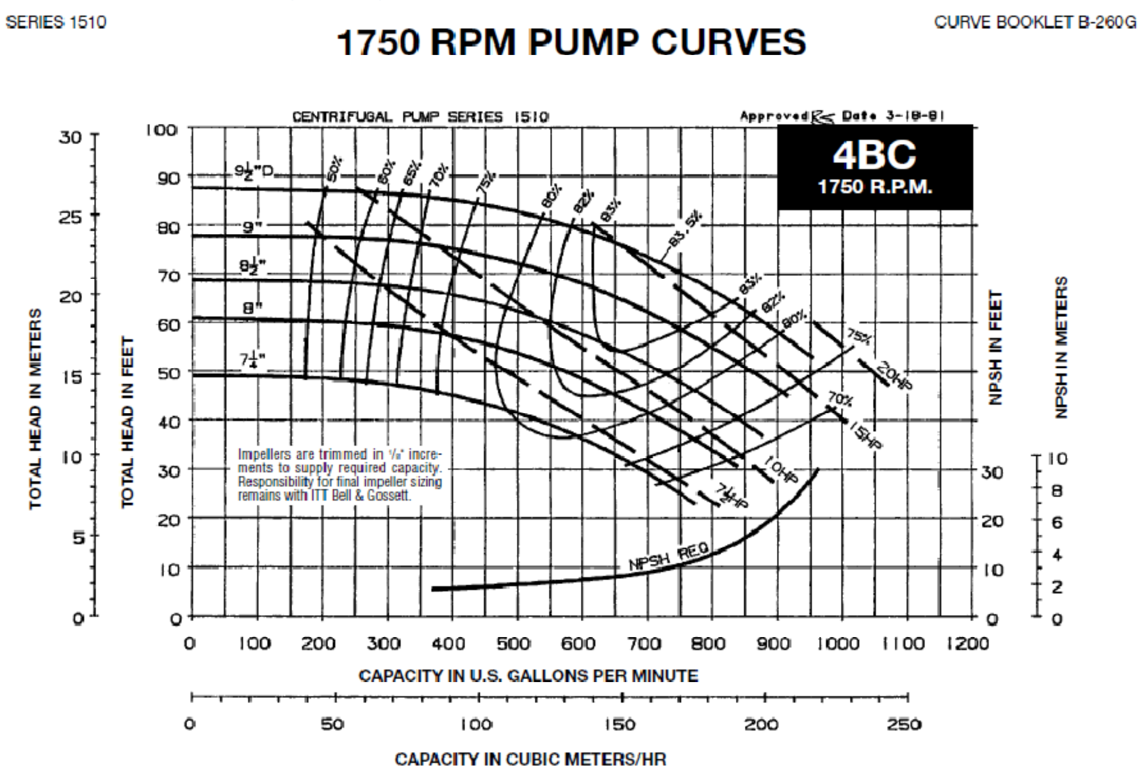


Figure 5 – P/S Secondary Pump Curve – Bell & Gossett Model 1510-4BC



Appendix B - Primary/Secondary vs. Variable Primary Flow Data and Charts

Figure 6 – VPF Pump Curve Original Selection – Bell & Gossett Model 1510-3G

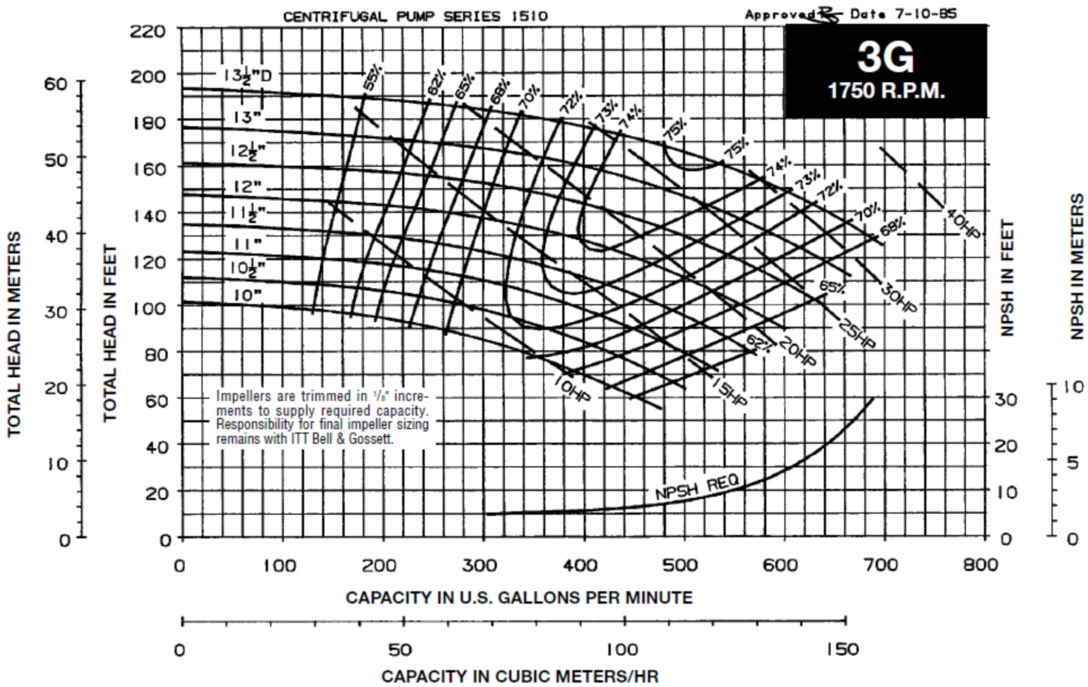
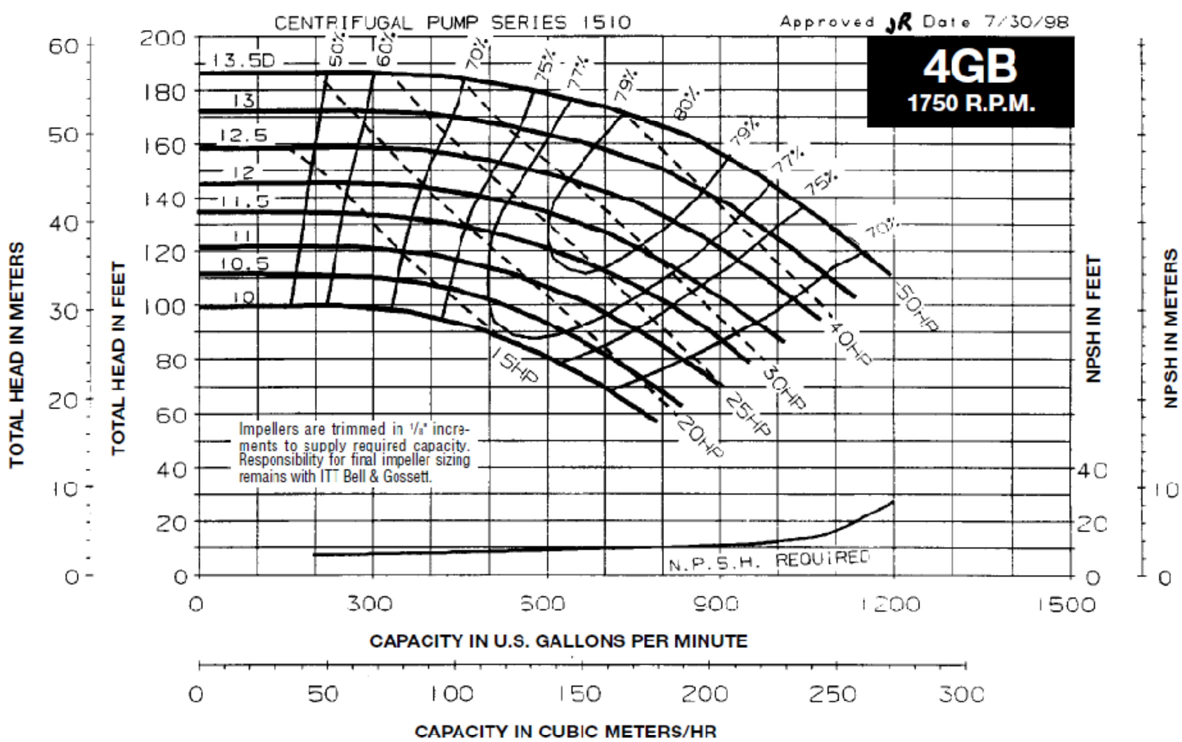


Figure 6 – VPF Pump Curve Re- Selection – Bell & Gossett Model 1510-4GB



Appendix B - Primary/Secondary vs. Variable Primary Flow Data and Charts

Sample Calculations for Parallel Pumps with VSD – Page 1 of 5

File:2Pumps_VPF_kW_per_Q.EES 4/4/2011 10:26:15 PM Page 1
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VPF Energy Consumption

Maximum Load=727 Tons - August 14th

Variable Volume Secondary Pump

Function **$H_{SecondaryPump}$** (Q, N)

$$H_{SecondaryPump} := \left[184.8 - 0.009354 \cdot 1.75 \cdot \frac{Q}{N} - 0.00008027 \cdot \left(1.75 \cdot \frac{Q}{N}\right)^2 + 6.964 \times 10^{-8} \cdot \left(1.75 \cdot \frac{Q}{N}\right)^3 - 2.172 \times 10^{-10} \cdot \left(1.75 \cdot \frac{Q}{N}\right)^4\right] \cdot \frac{N^2}{1750^2}$$

End **$H_{SecondaryPump}$**

Number of Pumps Online

Function **NumPump** (Q, Q_C)

If [$Q < Q_C$] Then

NumPump := 1

Else

NumPump := 2

EndIf

End **NumPump**

Pump Efficiency For Given Speed and Flow for Secondary Pump

Function **Eta_{Pump}** (Q, N)

$$\eta_{Pump} := 0.2925 + 0.002131 \cdot 1750 \cdot \frac{Q}{N} - 0.00000368 \cdot \left[1750 \cdot \frac{Q}{N}\right]^2 + 3.835 \times 10^{-9} \cdot \left[1750 \cdot \frac{Q}{N}\right]^3 - 2.725 \times 10^{-12} \cdot \left[1750 \cdot \frac{Q}{N}\right]^4$$

End **Eta_{Pump}**

Motor Efficiency as a Function of Nameplate Horsepower fraction

Function **Eta_{Motor}** (P_{Shaft}, $P_{Nameplate}$)

From Load Percentage 18% and Up

If $\left[\frac{P_{Shaft}}{P_{Nameplate}} > 0.18\right]$ Then

Appendix B - Primary/Secondary vs. Variable Primary Flow Data and Charts

Sample Calculations for Parallel Pumps with VSD – Page 2 of 5

File:2Pumps_VPF_kW_per_Q.EES 4/4/2011 10:26:15 PM Page 2
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$$\eta_{Motor} := 0.5635 + 1.881 \cdot \frac{P_{Shaft}}{P_{Nameplate}} - 3.751 \cdot \left[\frac{P_{Shaft}}{P_{Nameplate}}\right]^2 + 3.586 \cdot \left[\frac{P_{Shaft}}{P_{Nameplate}}\right]^3 - 1.349 \cdot \left[\frac{P_{Shaft}}{P_{Nameplate}}\right]^4$$

Load Percentage 0 to 18%

Else

$$\eta_{Motor} := 4.4444 \cdot \frac{P_{Shaft}}{P_{Nameplate}}$$

EndIf

End **Eta**$_{Motor}$

VFD as a Function of Variable Speed Drive Efficiency

Function **eta**$_{VFD}$ (N, N_{Design})

$$\eta_{VFD} := 0.88 + 0.35 \cdot \frac{N}{N_{Design}} - 0.6146 \cdot \left[\frac{N}{N_{Design}}\right]^2 + 0.625 \cdot \left[\frac{N}{N_{Design}}\right]^3 - 0.2604 \cdot \left[\frac{N}{N_{Design}}\right]^4$$

End **eta**$_{VFD}$

System Head as a fuction of flow rate, design flow rate, design head, and fixed head

Function $H_{system}(Q, Q_{Design}, H_{Design}, H_{Fixed})$

$$H_{system} := H_{Fixed} + [H_{Design} - H_{Fixed}] \cdot \left[\frac{Q}{Q_{Design}}\right]^2$$

End H_{system}

Secondary Variable Speed Pumps

$Q_{Design} = 1250$

$H_{Design} = 126$

$H_{Fixed} = 80$

$H_{System,V}$=H_{system}(Q,Q,Design,H,Design,H,FV)
$H_{System,V}$=$H_{SecondaryPump}$(Q,N,C2)

Variable Speed Changeover Point

$$H_{SecondaryPump}[Q_C, N_{C1}] = H_{SecondaryPump}[0.5 \cdot Q_C, N_{C2}]$$

Appendix B - Primary/Secondary vs. Variable Primary Flow Data and Charts

Sample Calculations for Parallel Pumps with VSD – Page 3 of 5

File:2Pumps_VPF_kW_per_Q.EES 4/4/2011 10:26:15 PM Page 3
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$$H_{SecondaryPump}[Q_C, N_{C1}] = H_{system}[Q_C, Q_{Design}, H_{Design}, H_{Fixed}]$$

Secondary BHP

$$\eta_{PCV1} = Eta_{Pump}[Q_C, N_{C1}]$$

$$P_{ShaftCV1} = Q_C \cdot \frac{H_{system}[Q_C, Q_{Design}, H_{Design}, H_{Fixed}]}{3960 \cdot \eta_{PCV1}}$$

$$\eta_{PCV2} = Eta_{Pump}[0.5 \cdot Q_C, N_{C2}]$$

$$P_{ShaftCV2} = Q_C \cdot \frac{H_{system}[Q_C, Q_{Design}, H_{Design}, H_{Fixed}]}{3960 \cdot \eta_{PCV2}}$$

Drive Efficiency

$$\eta_{DCV1} = eta_{VFD}[N_{C1}, 1750]$$

$$\eta_{DCV2} = eta_{VFD}[N_{C2}, 1750]$$

Motor Power, kW

$$\eta_{MCV1} = Eta_{Motor}[P_{ShaftCV1}, 30]$$

$$P_{TotalCV1} = \frac{P_{ShaftCV1}}{\eta_{MCV1} \cdot \eta_{DCV1}}$$

$$\eta_{MCV2} = Eta_{Motor}[0.5 \cdot P_{ShaftCV2}, 30]$$

$$P_{TotalCV2} = \frac{P_{ShaftCV2}}{\eta_{MCV2} \cdot \eta_{DCV2}}$$

$$P_{TotalCV1} = P_{TotalCV2}$$

Two Variable Speed Pumps

$$Z_V = NumPump[Q, Q_C]$$

$$H_{SecondaryPump}\left[\frac{Q}{Z_V}, N\right] = H_{system}[Q, Q_{Design}, H_{Design}, H_{Fixed}]$$

$$H_V = H_{SecondaryPump}\left[\frac{Q}{Z_V}, N\right]$$

Brake Horsepower

$$\eta_{PV} = Eta_{Pump}\left[\frac{Q}{Z_V}, N\right]$$

$$P_{ShaftV} = Q \cdot \frac{H_V}{3960 \cdot \eta_{PV}}$$

Drive Efficiency

Appendix B - Primary/Secondary vs. Variable Primary Flow Data and Charts

Sample Calculations for Parallel Pumps with VSD – Page 4 of 5

File:2Pumps_VPF_kW_per_Q.EES 4/4/2011 10:26:15 PM Page 4

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$$\eta_{DV} = \mathbf{eta}_{VFD}\left[N, 1750\right]$$

Motor Power, kW

$$\eta_{MV} = \mathbf{Eta}_{Motor}\left[\frac{P_{ShaftV}}{Z_V}, 30\right]$$

$$P_{TotalSecondary} = \left|0.7457 \cdot \frac{kw}{hp}\right| \cdot \frac{P_{ShaftV}}{\eta_{MV} \cdot \eta_{DV}}$$

$$\eta_{TV} = \left|0.7457 \cdot \frac{kw}{hp}\right| \cdot Q \cdot \frac{H_{system}\left[Q, Q_{Design}, H_{Design}, H_{Fixed}\right]}{3960 \cdot P_{TotalSecondary}}$$

Lookup Table: Lookup 2

	Q [gpm]	η_{pump}
Row 1	158	0.55
Row 2	218	0.62
Row 3	263	0.65
Row 4	298	0.68
Row 5	325	0.7
Row 6	367	0.72
Row 7	400	0.73
Row 8	423	0.74
Row 9	447	0.745
Row 10	520	0.745
Row 11	545	0.74
Row 12	578	0.73
Row 13	605	0.72
Row 14	625	0.71
Row 15	638	0.7
Row 16	660	0.68

Appendix B - Primary/Secondary vs. Variable Primary Flow Data and Charts

Sample Calculations for Parallel Pumps with VSD – Page 5 of 5

File:2Pumps_VPF_kW_per_Q.EES 4/4/2011 10:28:15 PM Page 5
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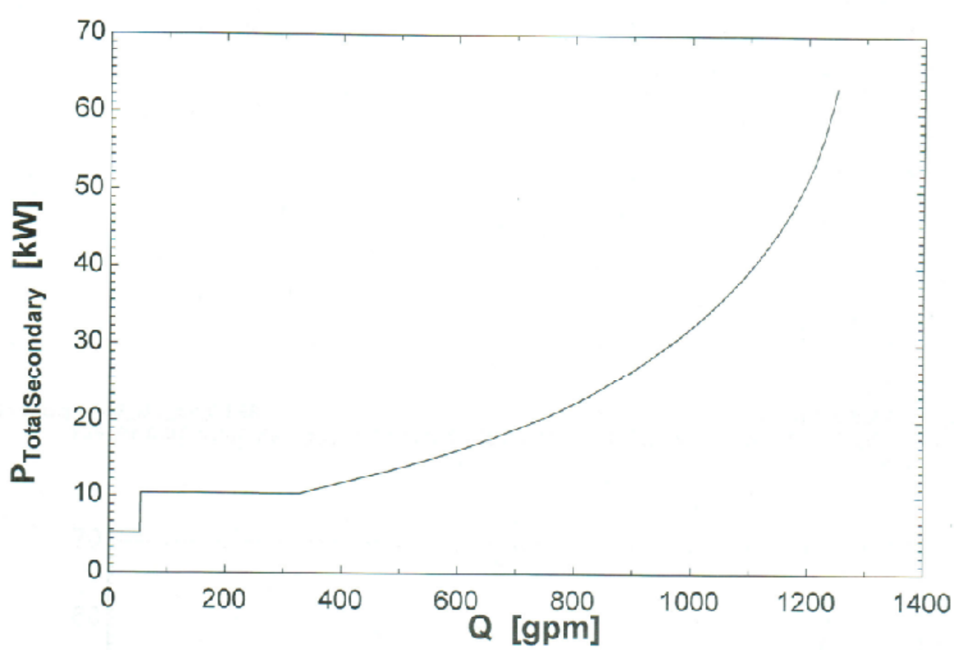


Appendix B - Primary/Secondary vs. Variable Primary Flow Data and Charts

P/S & VPF First Cost Comparison

First Cost				
	Primary/Secondary		VPF	
Primary Pumps	$ 30,000.00	3x $10,200	$ 30,000.00	3x $15,500
Secondary Pumps	$ 30,000.00	3x $12,700	$ -	-
VFDs	$ 10,725.00	3x $3575 (15 HP)	$ 17,550.00	3x $5850 (30 HP)
Modulating Bypass Control Valve	$ -	-	$ 3,050.00	-
Flow Meter	$ -	-	$ 500.00	-
Total First Cost	$ 70,725.00		$ 51,100.00	

Appendix B - Primary/Secondary vs. Variable Primary Flow Data and Charts

P/S System 30-Year Life Cycle Cost

Original P/S System						
	Base Electricity Rate		$0.0482	/kWh		
	Base discount rate (real)		2.7	% (30-year OMB estimate)		
	Annual Pump Energy		77154.674	kWh		
	Annual P/S Energy Cost		$3,718.86			
Date	Year	Capital	Overhauls	Maintenance	Electricity Escalator	Base Electricity Cost
2011	1	$70,725	$0	$1,000	0.90	$3,347
2012	2		$0	$1,000	0.92	$3,421
2013	3		$0	$1,000	0.94	$3,496
2014	4		$0	$1,000	0.93	$3,459
2015	5		$0	$1,000	0.92	$3,421
2016	6		$0	$1,000	0.99	$3,693
2017	7		$0	$1,000	0.95	$3,533
2018	8		$0	$1,000	0.95	$3,533
2019	9		$0	$1,000	0.96	$3,570
2020	10		$0	$1,000	0.97	$3,607
2021	11		$0	$1,000	0.98	$3,644
2022	12		$7,150	$1,000	0.99	$3,682
2023	13		$0	$1,000	1.00	$3,719
2024	14		$0	$1,000	1.01	$3,756
2025	15		$0	$1,000	1.01	$3,756
2026	16		$0	$1,000	1.01	$3,756
2027	17		$0	$1,000	1.02	$3,793
2028	18		$0	$1,000	1.02	$3,793
2029	19		$0	$1,000	1.05	$3,905
2030	20		$0	$1,000	1.06	$3,942
2031	21		$0	$1,000	1.08	$4,016
2032	22		$0	$1,000	1.10	$4,091
2033	23		$0	$1,000	1.12	$4,165
2034	24		$7,150	$1,000	1.13	$4,202
2035	25		$0	$1,000	1.14	$4,239
2036	26		$0	$1,000	1.15	$4,277
2037	27		$0	$1,000	1.15	$4,277
2038	28		$0	$1,000	1.16	$4,314
2039	29		$0	$1,000	1.17	$4,351
2040	30		$0	$1,000	1.17	$4,351
Column NPV		$70,725	$8,966	$20,383	-	$76,806
Total NPV						$176,880

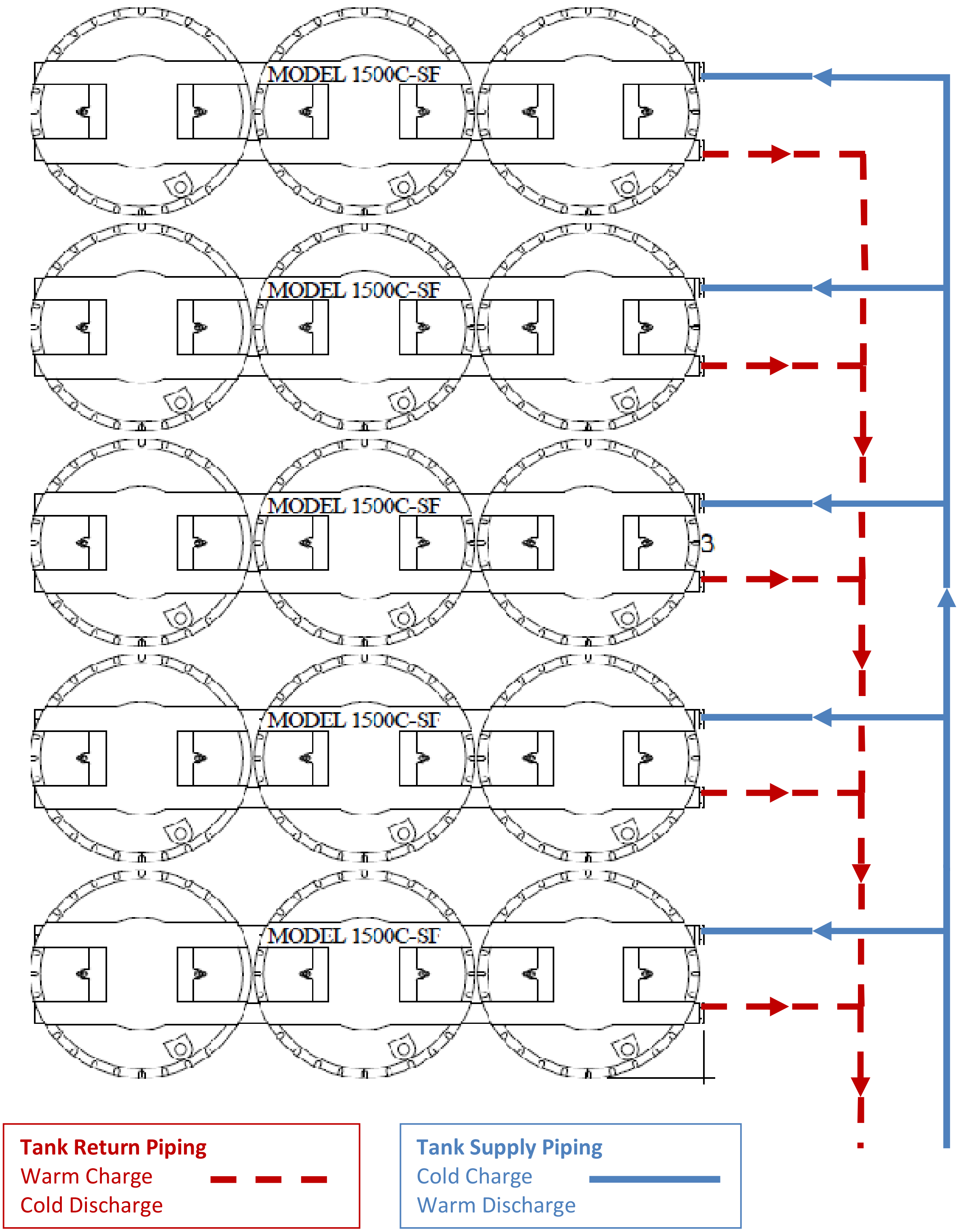
Appendix B- Primary/Secondary vs. Variable Primary Flow Data and Charts

VPF System 30-Year Life Cycle Cost

VPF System						
	Base Electricity rate		$0.0482	/kWh		
	Base discount rate (real)		2.7	% (30-year OMB estimate)		
	Annual Pump Energy		44909.856	kWh		
	Annual P/S Energy Cost		$2,164.66			
Date	Year	Capital	Overhauls	Maintenance	Electricity Escalators	Base Electricity Cost
2011	1	$51,050	$0	$1,000	0.90	$1,948
2012	2		$0	$1,000	0.92	$1,991
2013	3		$0	$1,000	0.94	$2,035
2014	4		$0	$1,000	0.93	$2,013
2015	5		$0	$1,000	0.92	$1,991
2016	6		$0	$1,000	0.99	$2,150
2017	7		$0	$1,000	0.95	$2,056
2018	8		$0	$1,000	0.95	$2,056
2019	9		$0	$1,000	0.96	$2,078
2020	10		$0	$1,000	0.97	$2,100
2021	11		$0	$1,000	0.98	$2,121
2022	12		$11,700	$1,000	0.99	$2,143
2023	13		$0	$1,000	1.00	$2,165
2024	14		$0	$1,000	1.01	$2,186
2025	15		$0	$1,000	1.01	$2,186
2026	16		$0	$1,000	1.01	$2,186
2027	17		$0	$1,000	1.02	$2,208
2028	18		$0	$1,000	1.02	$2,208
2029	19		$0	$1,000	1.05	$2,273
2030	20		$0	$1,000	1.06	$2,295
2031	21		$0	$1,000	1.08	$2,338
2032	22		$0	$1,000	1.10	$2,381
2033	23		$0	$1,000	1.12	$2,424
2034	24		$11,700	$1,000	1.13	$2,446
2035	25		$0	$1,000	1.14	$2,468
2036	26		$0	$1,000	1.15	$2,489
2037	27		$0	$1,000	1.15	$2,489
2038	28		$0	$1,000	1.16	$2,511
2039	29		$0	$1,000	1.17	$2,533
2040	30		$0	$1,000	1.17	$2,533
Column NPV		$51,050	$14,671	$20,383		$44,707
					Total NPV	$130,811

Appendix C– Thermal Storage Information

Tank Piping Diagram



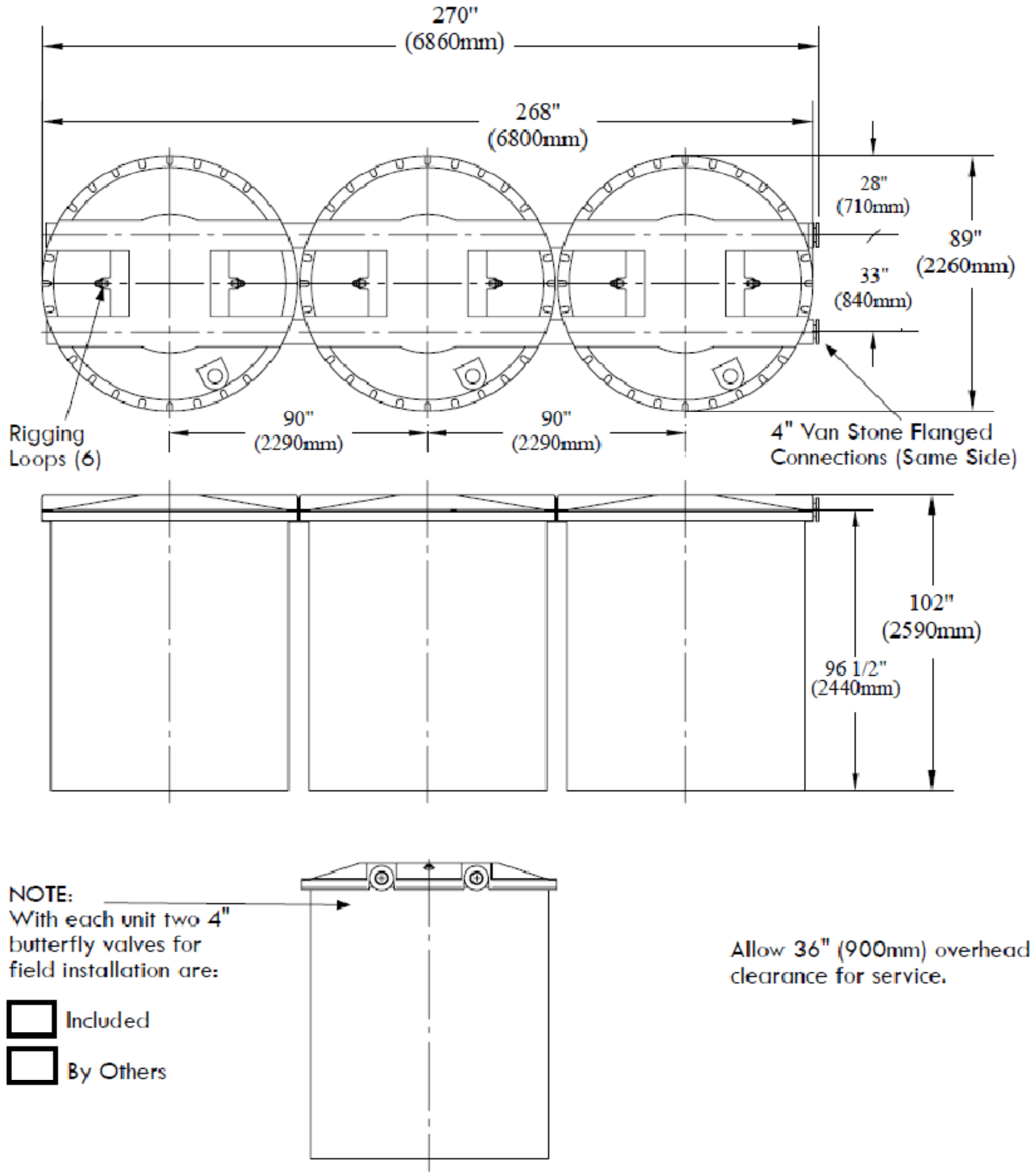
Appendix C – Thermal Storage Information

Tank Dimensions & Diagram

CALMAC IceBank® THERMAL ENERGY STORAGE

DRAWING
Model 1500CSF
August 2007
CS-52

CALMAC Manufacturing Corp. 3-00 Banta Place Fair Lawn, NJ 07410 Tel (201) 797-1511 www.Calmac.com



NOTE: Overall length ±1", Tolerance for all other dimensions is±1/2" (12.5 mm)

Appendix C – Thermal Storage Information

Non- Thermal Storage and Thermal Storage System Detailed First Cost Breakdown

First Cost				
	Original System		TES	
Chiller Plant	$1,542,500.00	3x365 ton @ $1500/ton	$900,000.00	3x 200 ton @ $1500/ton
Tanks (Includes Slab, Glycol, Controls, Local Piping	$		$437,400.00	2916ton hr x $150/ton hr
3 Way Valve	$		$3,000	
A/G Piping & Insulation	$		$13,090.00	2x85 ft @$77/ft
U/G Piping & Insulation	$		$62,400.00	2x156 ft @$200/ft
U/G Piping Excavation	$		$936.00	$3.00/LF
U/G Pipng Fill	$		$982.80	$1.00/LF
Concrete Pad Excavation (4 foot tank burial)	$		$2,756	167CY @ $16.50/CY
Privacy Fence	$		$6,620.00	130 ft @ $60/ft
Total First Cost	$1,542,500.00		$1,427,184.30	

Appendix C – Thermal Storage Information

30-Year Life Cycle Cost of Non-Thermal Storage System

Original P/S System					
	Base fuel rate	Electricity	$0.0482	/kWh	
		Base discount rate (real)	2.7	% (30-year OMB estimate)	
		Annual P/S Energy Cost	$70,550.55		
Date	Year	Capital	Other Maint	Electricity Escalators	Base Electricity Cost
2011	1	$1,642,500	$25,000	0.90	$63,495
2012	2		$25,000	0.92	$64,907
2013	3		$25,000	0.94	$66,318
2014	4		$25,000	0.93	$65,612
2015	5		$25,000	0.92	$64,907
2016	6		$25,000	0.99	$70,057
2017	7		$25,000	0.95	$67,023
2018	8		$25,000	0.95	$67,023
2019	9		$25,000	0.96	$67,729
2020	10		$25,000	0.97	$68,434
2021	11		$25,000	0.98	$69,140
2022	12		$25,000	0.99	$69,845
2023	13		$25,000	1.00	$70,551
2024	14		$25,000	1.01	$71,256
2025	15		$25,000	1.01	$71,256
2026	16		$25,000	1.01	$71,256
2027	17		$25,000	1.02	$71,962
2028	18		$25,000	1.02	$71,962
2029	19		$25,000	1.05	$74,078
2030	20		$25,000	1.06	$74,784
2031	21		$25,000	1.08	$76,195
2032	22		$25,000	1.10	$77,606
2033	23		$25,000	1.12	$79,017
2034	24		$25,000	1.13	$79,722
2035	25		$25,000	1.14	$80,428
2036	26		$25,000	1.15	$81,133
2037	27		$25,000	1.15	$81,133
2038	28		$25,000	1.16	$81,839
2039	29		$25,000	1.17	$82,544
2040	30		$25,000	1.17	$82,544
	Column NPV	$1,642,500	$509,572		$1,457,096
				Total NPV	$3,609,168

Appendix C – Thermal Storage Information

30-Year Life Cycle Cost of Thermal Storage System

TES System					
	Base fuel rate Electricity		$0.0482	/kWh	
	Base discount rate (real)		2.7	% (30-year OMB estimate)	
	Annual TES Energy Cost		$ 69,146.04		
Date	Year	Capital	Other Maint	Electricity Escalators	Base Electricity Cost
2011	1	$1,427,184	$15,000	0.90	$62,231
2012	2		$15,000	0.92	$63,614
2013	3		$15,000	0.94	$64,997
2014	4		$15,000	0.93	$64,306
2015	5		$15,000	0.92	$63,614
2016	6		$15,000	0.99	$68,662
2017	7		$15,000	0.95	$65,689
2018	8		$15,000	0.95	$65,689
2019	9		$15,000	0.96	$66,380
2020	10		$15,000	0.97	$67,072
2021	11		$15,000	0.98	$67,763
2022	12		$15,000	0.99	$68,455
2023	13		$15,000	1.00	$69,146
2024	14		$15,000	1.01	$69,838
2025	15		$15,000	1.01	$69,838
2026	16		$15,000	1.01	$69,838
2027	17		$15,000	1.02	$70,529
2028	18		$15,000	1.02	$70,529
2029	19		$15,000	1.05	$72,603
2030	20		$15,000	1.06	$73,295
2031	21		$15,000	1.08	$74,678
2032	22		$15,000	1.10	$76,061
2033	23		$15,000	1.12	$77,444
2034	24		$15,000	1.13	$78,135
2035	25		$15,000	1.14	$78,826
2036	26		$15,000	1.15	$79,518
2037	27		$15,000	1.15	$79,518
2038	28		$15,000	1.16	$80,209
2039	29		$15,000	1.17	$80,901
2040	30		$15,000	1.17	$80,901
	Column NPV	$1,427,184	$305,743		$1,428,088
Total NPV					$3,161,016

Appendix D – Solar Photovoltaic System Documentation

BP 3230T Solar Photovoltaic Panel Product Sheet

230W Photovoltaic module

BP 3230T

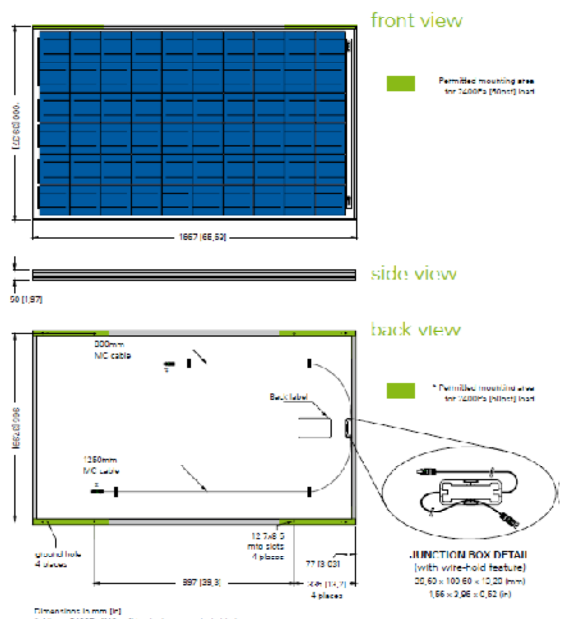
bp solar

Electrical characteristics

	[1] STC 1000W/m²	[2] NOCT 800W/m²
Maximum power (P_{max})	230W	165.6W
Voltage at P_{max} (V_{mpp})	29.1V	25.9V
Current at P_{max} (I_{mpp})	7.90A	6.32A
Short circuit current (I_{sc})	8.40A	6.80A
Open circuit voltage (V_{oc})	36.7V	33.4V
Module efficiency	13.8%	
Tolerance	0/+5%	
Nominal voltage	20V	
Efficiency reduction at 200W/m²	<5% reduction (efficiency 13.1%)	
Limiting reverse current	8.40A	
Temperature coefficient of I_{sc}	(0.065±0.015)%/C	
Temperature coefficient of V_{oc}	-(0.36±0.05)%/C	
Temperature coefficient of P_{max}	-(0.5±0.05)%/C	
[3] NOCT	47±2°C	
Maximum series fuse rating	20A	
Application class (according to IEC 61730:2007)	Class A	
Maximum system voltage (U.S. NEC rating)	600V (U.S. NEC rating) 1000V (IEC 61730:2007)	

1: Values at Standard Test Conditions (STC): 1000W/m² irradiance, AM1.5 solar spectrum and 25°C module temperature
2: Values at 800W/m² irradiance, Nominal Operation Cell Temperature (NOCT) and AM1.5 solar spectrum
3: Nominal Operation Cell Temperature: Module operation temperature at 800W/m² irradiance, 20°C air temperature, 1m/s wind speed

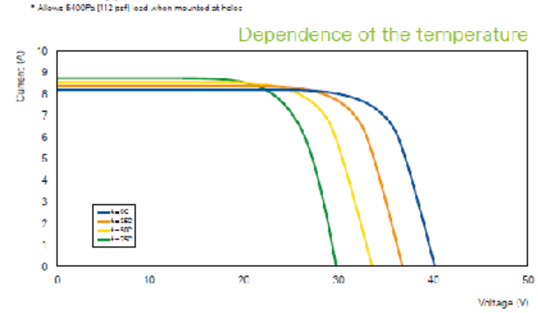
All solar modules are individually tested prior to shipment; an allowance is made within our factory measurement to account for the typical power degradation (LID effect) which occurs during the first few days of deployment.

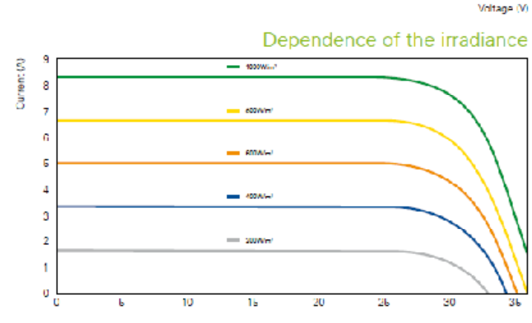


Mechanical characteristics

Solar cells	60 polycrystalline 6" silicon cells (156x156mm) in series
Front cover	High transmission 3.2mm (1/8th in) glass
Encapsulant	EVA
Back cover	White polyester
Frame	Silver anodized aluminum (Universal II)
Diodes	IntegraBus™ with 6 Schottky diodes
Junction box	Potted (IP 67); certified to meet UL 1703 flammability test
Output cables	4mm² cable with latching MC4 connectors Asymmetrical cable lengths: (-)1250mm (49.21in) / (+)800mm (31.50in)
Dimensions	1667x1000x50mm / 65.6x39.4x2.0in
Weight	19.4kg / 42.8lbs

All dimensional tolerances within ±0.1% unless otherwise stated.





Warranty

- Free from defects in materials and workmanship for 5 years
- 93% power output over 12 years
- 85% power output over 25 years

Certification

Certified according to the extended version of the IEC 61215 2005 (Crystalline silicon terrestrial photovoltaic modules - Design qualification and type approval)

Certified according to IEC 61730-1 and IEC 61730-2. (Photovoltaic module safety qualification, requirements for construction and testing)

Listed to UL 1703 Standard for Safety by Intertek ETL (Class C fire rating)

Manufactured in ISO 9001 and ISO 14001 certified factories

Module electrical measurements are calibrated to World radiometric reference via third party international laboratories

This data sheet complies with the EN 50380 requirements.
This publication summarizes product warranty and specifications which are subject to change without notice.

Contact:

Your BP Solar partner

Find more information in: www.bpsolar.com

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Appendix D - Solar Photovoltaic System Schematic