FINAL THESIS REPORT

MECHANICAL SYSTEM ALTERNATIVES ANALYSIS

New Braunfels Regional Rehabilitation Hospital

New Braunfels, TX

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Mechanical Option

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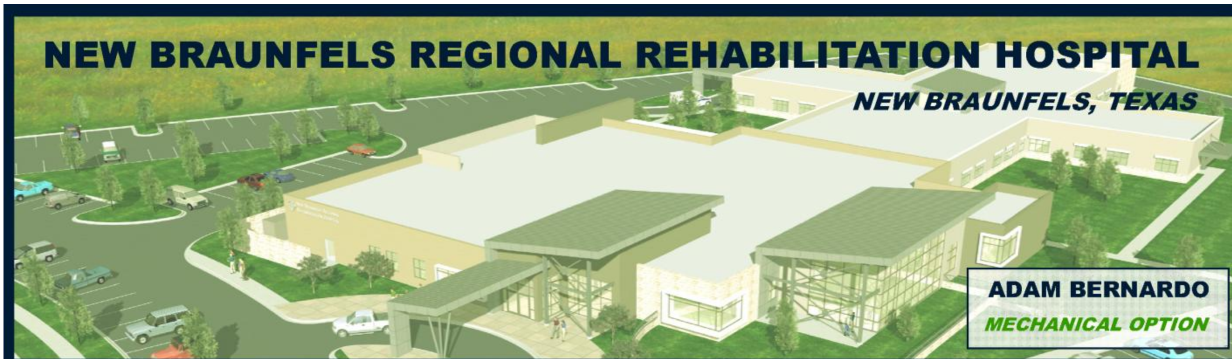
Date Submitted: 04/04/12



»GENERAL PROJECT INFO

Function 40-bed, acute-care hospital **Size** 49,488 square feet **Height** 18-27 feet, 1 story **Construction Dates** July, 2010 - June, 2011 **Delivery Method** Single Prime Contract

»ARCHITECTURE

- The facility contains a hospital wing as well as outpatient day treatment rooms, exam rooms, and a therapy pool.
- 40 patient rooms are laid out in cross design maximizes the number of rooms in the area while providing each with an exterior view.
- Large amounts of southeast glazing create a visually appealing entrance and invite sunlight into public areas of the clinic.
- Exterior shading devices near entrances help shade vehicles used to drop off patients.

»STRUCTURAL SYSTEM

- 1 ½" metal roof deck is supported throughout the building by K-series bar joists and wide-flange girders of various sizes.
- Vertical loads are transferred to HSS columns and down to 6' x 6' spread footings .
- The floor is composed of a 5" thick slab on grade.
- Glass curtain walls are supported by variably-sized wide-flange columns and cross braced with hollow structural sections for resisting lateral loads.

»LIGHTING/ELECTRICAL SYSTEMS

- Patient, exam, and therapy rooms contain recessed T8 lamps in a static fluorescent troffer with electronic ballast control.
- Circuit breakers are served by a central, 1600-amp circuit breaker switchboard with a 1000-amp backup circuit breaker.
- The building's power is distributed through 5 dry-type transformers with a total power capacity of 272.5 kVA.

»PROJECT TEAM

Owner Ernest Health, Inc.

Architect Dekker/Perich/Sabatini

Structural Engineer Dekker/Perich/Sabatini

MEP/FP Engineer JBA Consulting Engineers

General Contractor MJ Harris, Inc.

»MECHANICAL SYSTEMS

- Conditioned air is supplied to the facility from three packaged roof-top units that utilize direct expansion cooling and gas-fired heating.
- Air is distributed to VAV terminal units that serve each zone. Most units contain reheat coils that are served by two gas-fired boilers.
- The entire facility utilizes a fully-ducted return system.
- A specialized pool dehumidification system controls the indoor environmental quality of the pool area while maintaining proper pool temperatures.
- Kitchen and dining functions are served by a 100% outdoor air makeup air unit.

»PLUMBING/FIRE PROTECTION

- Domestic hot water is generated by two gas-fired, 130-gallon water heaters and distributed via two hot water circulating pumps located in the mechanical room.
- Emergency domestic potable water is kept in a 500-gallon insulated vertical storage tank.
- Fire and smoke dampers are controlled by duct detector sensors at zone level.

http://www.engr.psu.edu/ae/thesis/portfolios/2012/ASB5112/index.html

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Acknowledgements

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Bobby George	Dekker/Perich/Sabatini Architects
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Fellow AE Students	

(1.0) Executive Summary

For this senior thesis project, three mechanical depth studies were performed to investigate possible energy-saving or better-controlled systems for the New Braunfels Regional Rehabilitation Hospital (NBRRH). The first depth study performed proved the hypothesis that a central plant re-design, including condensing boilers and the addition of a chilled water system, would not be a cost-effective alternative design. While all central plant alternatives decreased energy consumption, the increased first cost was too great to overcome in a twenty year lifecycle analysis. A basic, water-cooled chilled water plant had a simple payback period of over 48 years.

The second depth study performed involved introducing six multi-split, variable refrigerant flow (VRF) systems to serve the heating and cooling needs of patient room and office zones. These systems would work in heat recovery operation to allow for simultaneous heating and cooling of zones connected to the same outdoor condensing unit. These systems allowed for the removal of the largest rooftop air-handling unit and required the introduction of a dedicated outdoor air unit to serve the ventilating and dehumidifying requirements of these spaces. VRF technology is relatively unused in the United States, but an energy model showed a simple payback period of just under 6 years.

The third depth study investigated the viability of installing a solar thermal system to heat domestic hot water, water in the therapy pool, and supplement space heating energy needs. An in-depth analysis of a forced-circulation solar thermal system with flat-plate collectors showed the optimal collector area to be 390 square feet of rooftop-mounted collectors at an angle of 40° from horizontal, facing 33° degrees from true south. This arrangement, with a stratified hot water storage tank for thermal storage, allowed for the following loads to be met by the solar energy collected:

- 76% of the domestic hot water load, 38% by direct gains, 38% by storage
- 22% of the space heating load, 8% by direct gains, 14% by storage

These energy savings resulted in a simple payback of just over 2 years, and a decrease in the net present value of the system by over $325,000.

Two breadth studies were also performed to analyze the impact of the new rooftop units on the structural roof framing design and the impact of the VRF system on patient room acoustics. The elimination of the largest rooftop unit was able to save $2,900 in material costs and is factored into the economic analysis of the VRF system. The patient room acoustics study showed a slight increase in the noise criteria (NC) sound pressure level in a typical patient room from 36 dBA to 39 dBA. While this slight increase is noticeable, it still fits into the acceptable NC levels for a private hospital room.

This thesis shows the viability of three potential mechanical system redesigns or additions and determines that, while a chilled water system is not economically viable, a VRF system may be depending on the owner's payback threshold and a solar thermal system is a very economically-plausible alternative to create energy savings.

(2.0) Building Overview

Facility Description

The New Braunfels Regional Rehabilitation Hospital is a 40-bed, acute-care hospital and physical rehabilitation clinic located about 30 miles northeast of San Antonio, Texas. Managed by Ernest Health, Inc., the nearly 50,000 square foot facility is located on a several hundred thousand square foot site that was previously a country club. Ernest Health operates 14 similar acute-care hospitals in various regions of the United States.

All of the patient rooms and hospital-specific functions are located in the northern wing of the building, which is arranged in a cross design. The south-facing sections of the building house public functions with a large amount of glazing. These include administrative offices, the entrance lobby and reception area, and the physical therapy and exercise room. Other functions included in the southern wing of the facility are the hospital's kitchen and patient dining areas, exam and therapy rooms, service rooms, and additional office space. A graphical description of building function layout is shown in Figure 1 below.

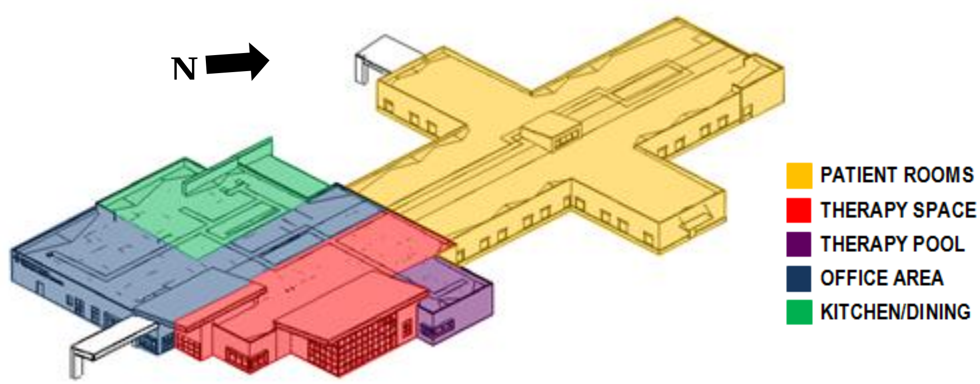

Figure 1: Building Function Layout

Mechanical System Overview

Three packaged rooftop units supply most of the facility with conditioned air. Each of these units is air-cooled and utilizes gas-fired heating. One 26,000 CFM unit serves the entire north patient wing of the building with air for ventilation and space conditioning. The other two units, totaling 29,500 CFM, serve the therapy, administrative, and kitchen/dining functions of the facility. The kitchen and dining functions of the building are supplemented by a 100% outdoor air makeup air unit. All zones are supplied by VAV terminal units and utilize a fully-ducted return system. Two gas-fired boilers provide heating hot water to reheat coils located in the VAV boxes at zone level.

The therapy pool is served by a split-system pool dehumidification unit that is controlled to automatically dehumidify the pool room and maintain proper pool water temperatures.

PART I: EXISTING SYSTEMS EVALUATION

(3.0) Design Load Estimation Procedure

The heating and cooling loads for the New Braunfels Regional Rehabilitation Hospital were estimated using Trane Trace 700 software. The building itself and mechanical systems were modeled using mechanical and architectural design drawings and documents along with a number of assumptions and data, outlined in this report. Because of the manageable size of the facility, a room-by-room method was used to estimate the loads on the building.

(3.1) Load Calculation Assumptions

To perform the load estimation, several general assumptions were made that both accurately simulate design conditions and make the estimation easier to accomplish. It was assumed that the facility is fully operational at all times of the day throughout the entire year. This assumption is valid because of the critical functions occurring in the spaces and makes a difference in load profiles because spaces will need to be heated, cooled, and ventilated around the clock. Additionally, there were simplifications made to some design load data in order to make the modeling process time-efficient.

(3.2) Weather Data

Typical weather data for San Antonio, TX was obtained from the 2009 ASHRAE Handbook of Fundamentals. The measurements for this data were taken at the San Antonio International Airport, approximately 32 miles from the facility, so the data was assumed to be an accurate representation of the weather conditions that the site will see. A summary of the design conditions is shown in Table 1 below, while the entire ASHRAE Weather Data Sheet is provided in Appendix B. The listed design cooling and heating conditions are 0.4% and 99.6% values, respectively.

Table 1: ASHRAE Weather Data

Design Condition	Outdoor DB	Outdoor WB	DB Range	Indoor Design DB
Cooling	98.5 ºF	73.5 ºF	20.1 ºF	75 ºF
Heating	27.4 ºF	-	-	72 ºF

(3.3) Building Envelope

Building U-Factors were obtained from the basis of design performed by JBA Consulting Engineers and confirmed by the architect's model in Autodesk's Revit Architecture program. These values are shown in Table 2 on the next page. All exterior walls in the facility have a structure of 6" metal studs with insulation and have a gypsum wall board interior face. Two exterior facades exist in the facility, so for the purpose of this analysis an average U-Factor was used for all exterior faces. All exterior glazing including components of the southeast curtain wall system was assumed to have the same U-Factor and shading coefficient.

Table 2: Building Envelope U-Factors

Envelope Element	Description	U-Factor (BTU/hr-ft^2-°F)	Shading Coefficient
Floor Slab	4" HW Concrete	0.6587	-
Roof	Insulated Metal Deck	0.03569	-
Exterior Walls	Steel Frame, 6" Insulation	0.05543	-
Glazing	Steel Framed, Double-Pane	0.35	0.95

(3.4) Design Loads

Design loads used in this load estimation are shown below in Table 3 and discussed in the following two sections.

Table 3: Design Load Summary

Template Name	People	Equipment	Lighting	Ventilation	
	SF/Person	W/SF	W/SF	CFM/Person	CFM/SF
Breakroom	33.3	0.5	1.2	5	0.06
Classroom	20	0.5	1.4	10	0.12
Conference	20	0.5	1.3	5	0.06
Corridor	0	0.0	1.0	0	0.06
Custodian	0	0.0	0.9	0	0.12
Dining	10	0.0	0.9	7.5	0.18
Electrical	0	20.0	1.5	0	0.06
Files	0	0.0	1.1	0	0.12
Gym/Exercise	50	2.0	0.9	20	0.06
Kitchen	0	1.0	1.2	0	0
Laundry	0	5.0	0.6	7.5	0.06
Lobby	16.7	0.0	1.3	5	0.06
Locker Room	0	0.0	0.6	0	0
Mechanical	0	10.0	1.5	0	0.06
Nurse Station	143	0.5	1.0	5	0.06
Office	143	0.5	1.1	5	0.06
Pool	50	0.0	0.9	20	0.06
Restroom	0	0.0	0.9	0	0
Storage	0	0.0	0.9	0	0.12
Vestibule	0	0.0	1.3	0	0
Template Name	**# of People**	**Equipment**	**Lighting**	**Ventilation**	
		W/SF	**W/SF**	**Air Changes/Hour**	
Bathing	2	2.0	0.9	10	
Body Holding	0	2.0	0.9	10	
Clean Linen Storage	0	0.0	0.9	2	
Medical Storage	0	0.0	1.4	8	
Patient Room	2	2.0	0.7	6	
Patient Toilet	1	0.0	0.9	10	
Pharmacy	3	2.0	1.2	4	
Soiled Linen Storage	0	0.0	0.9	10	
Therapy	2	1.0	1.5	6	

(3.4.1) Design Occupancy and Ventilation

The design occupancy for spaces in the administrative, dining, and physical therapy areas were determined using the preset occupancy values in the Trace program based on the use of the space. In the hospital-specific spaces of the building, the occupancy density used by the mechanical engineer was used when available. If these values were not available, a reasonable estimate was made based on room function.

The ventilation requirements for the administrative, dining, and physical therapy areas were determined using Table 6-1 of ASHRAE Standard 62.1-2007 because this method was also used by the mechanical designer to calculate ventilation airflows. In the hospital-specific areas of the facility, Table 7-1 of ASHRAE Standard 170 was used to determine the required air changes per hour for this ventilation estimation.

(3.4.2) Lighting and Miscellaneous Loads

Lighting power densities used to generate lighting loads in the Trace model are based on Table 2 in Chapter 18 of ASHRAE Fundamentals 2009. The miscellaneous loads used in the model are based on best judgment of the likely equipment to be in the space.

(4.0) Design Load Estimation Results

As an energy model accuracy check, a system analysis was performed where the modeled loads on each RTU were compared to the existing RTUs, as designed. During the redesign phase of this thesis, a more detailed zone analysis was conducted in order to size systems that would serve smaller areas.

Shown in Figure 2 below are the areas that each of the rooftop units serve. RTU-1 delivers conditioned air to patient rooms and hospital-related functions in the northern wing of the facility. The physical therapy and exercise areas are served by RTU-2, and RTU-3 primarily serves the kitchen and dining area as well as administrative and back-of-house functions.

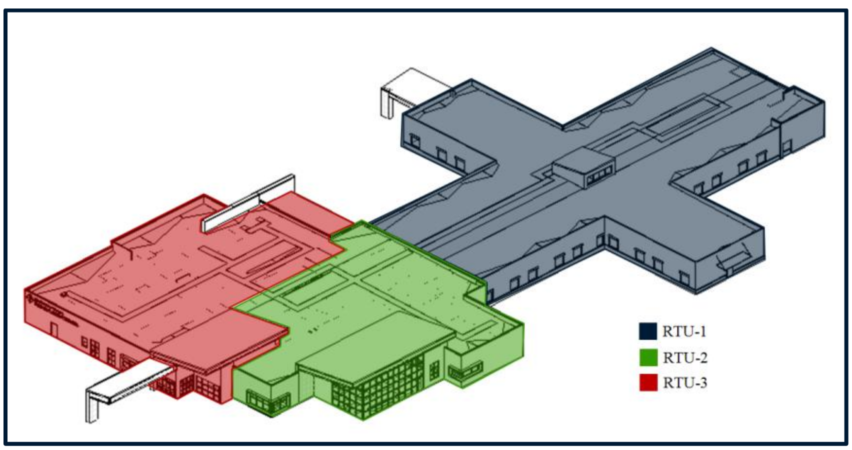

Figure 2: RTU Areas

Results of the load estimation for each system are shown in Table 4 on the following page, which also compares these results to the as-designed systems. A number of discrepancies exist between the modeled and existing systems.

Table 4: System-Level Load Comparison

	System	Area (SF)	Exterior Wall Area (SF)	Glazing Area (SF)	Cooling Load (tons)	Supply Airflow (CFM)	Heating Load (MBh)	Cooling kBTU/SF/yr	CFM/SF
Modeled	RTU-1	22215	13085	1719	63.3	26283	440.2	300	1.183
	RTU-2	11378	5460	1977	35.9	10099	343.5	332	0.888
	RTU-3	10456	6203	593	44.4	11135	267.5	446	1.065
	Totals:	**44049**	**24748**	**4289**	**143.6**	**47517**	**1051.2**	**343**	**1.079**
As Designed	RTU-1	22215	13085	1719	76.1	26000	520.0	360	1.170
	RTU-2	11378	5460	1977	34.8	12000	400.0	322	1.055
	RTU-3	10456	6203	593	57.2	17500	400.0	575	1.674
	Totals:	**44049**	**24748**	**4289**	**168.1**	**55500**	**1320**	**401**	**1.260**

The modeled heating load is less than the load that can be handled by the existing system. An explanation for this could be that, when designed, the heating capacity of the units may have been increased due to concerns of occupant safety and comfort.

Systems RTU-1 and RTU-3 also have significantly higher cooling capacities than what was estimated by the Trace load calculation. A likely cause of this difference is that the mechanical engineer may have used more conservative assumptions for process or miscellaneous power densities in these areas or built in factors of safety into equipment selection.

The modeled system has a reasonable square footage per ton of cooling, approximately 306 SF/Ton, when compared to ASHRAE Fundamentals, which gives a rule of thumb of about 275 SF/Ton for a hospital. The modeled heating and cooling loads, while not exactly what the existing units are sized for, will be considered adequate models of energy use for the facility.

(5.0) Energy Consumption and Operating Costs

Using the results of the Trace load estimation, an analysis of the energy consumption and operating cost of the New Braunfels Regional Rehabilitation Hospital was performed. All systems were modeled as variable air volume systems with zone-level reheat. It is important to note that the accuracy of this yearly energy estimation is impossible to determine at the time of this report because the facility has only been occupied and operational for about eight months.

(5.1) Annual Energy Consumption

Five main elements of the mechanical and electrical systems in the building contributed to the energy consumption of the facility. Direct expansion cooling, lights, supply and return air fans, and receptacle loads all contributed to the electricity consumed by NBRRH, while gas-fired space heating contributed to the natural gas consumed by the facility. A monthly summary of how each element used energy is shown in Table 5 on the next page, which is consistent with the results of the Trane Trace energy model.

Table 5: Monthly Energy Consumption

Month	Cooling (kWh)	Lights (kWh)	Fans (kWh)	Receptacles (kWh)	Space Heating (therms)
Jan	48700	12505	19480	2175	585
Feb	40609	11291	16244	2175	551
Mar	63933	12505	25573	2175	344
Apr	80573	12102	32229	2175	185
May	98530	12505	39412	2175	115
Jun	103836	12102	41534	2175	85
Jul	115763	12505	46305	2175	79
Aug	116918	12505	46767	2175	79
Sep	102696	12102	41078	2175	86
Oct	72794	12505	29118	2175	263
Nov	62769	12102	25108	2175	330
Dec	49537.92	12505	19815	2175	567

Total Electrical Consumption (kWh): 1512650
Total Gas Consumption (therms): 3269

As expected, the cooling load and associated fan energy dominate the electrical consumption of the facility in the summer months because of the hot, humid climate of Texas. Because the facility is occupied year-round at all hours of the day, the lighting system accounts for consistent electricity draw each month as evident in Figure 3 below, which shows the breakdown of the building's monthly energy consumption.

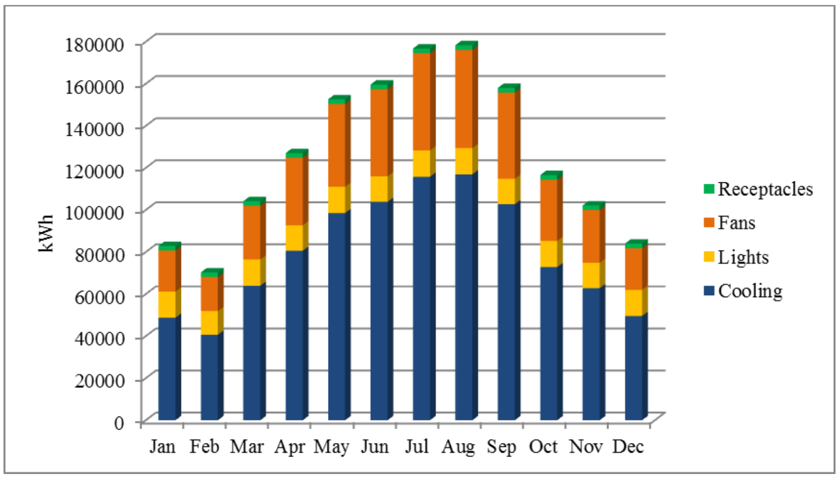

Figure 3: Monthly Electrical Energy Consumption

The yearly natural gas consumption profile for space heating in the New Braunfels Regional Rehabilitation Hospital is shown in Figure 4 on the following page. As seen previously in Table 5 above, the heating demand decreases greatly during summer months and thus there is a large drop-off in the amount of natural gas is consumed. In addition to the natural gas required for space heating, there is a significant amount of natural gas needed to heat domestic hot water usage, which is discussed in Section 20.1 of this report.

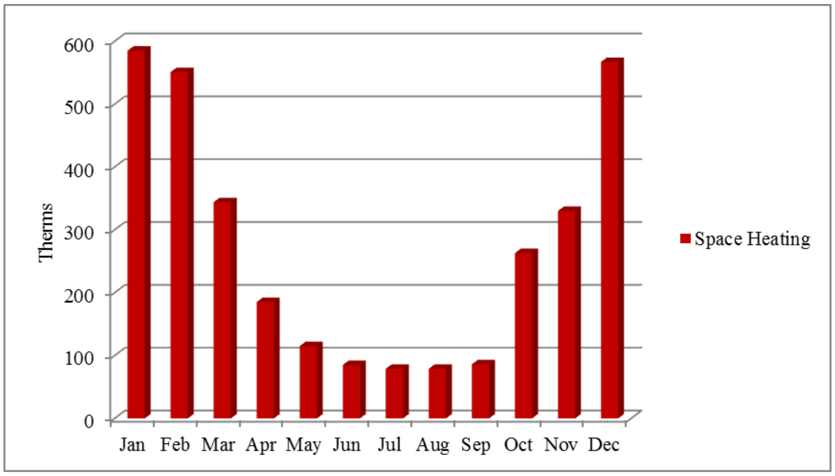

Figure 4: Monthly Natural Gas Consumption

(5.2) Equipment Operating Costs

Using the energy analysis from the previous section, the building's annual operating cost was determined. Electricity and water utility rates for New Braunfels were acquired through the New Braunfels Utility website while an average cost of natural gas in Texas was acquired through Center Point Energy's website. Table 6 below summarizes the utility rate structure that was used for this economic analysis and the total associated electricity and natural gas costs.

Table 6: Monthly Energy Costs

Month	Electricity Cost ($/kWh)	Natural Gas Cost ($/therm)	Total Electricity Cost ($)	Heating Cost ($)
Jan	0.04	0.9573	3314.35	560.02
Feb	0.04	0.9573	2812.74	527.47
Mar	0.04	0.9573	4167.42	329.31
Apr	0.04	0.9573	5083.14	177.10
May	0.04	0.9573	6104.86	110.09
Jun	0.05	0.9573	7982.31	81.37
Jul	0.05	0.9573	8837.40	75.63
Aug	0.05	0.9573	8918.26	75.63
Sep	0.05	0.9573	7902.52	82.33
Oct	0.04	0.9573	4663.66	251.77
Nov	0.04	0.9573	4086.14	315.91
Dec	0.04	0.9573	3361.30	542.79
		Totals:	**$67,234**	**$3,129**

A distribution of monthly operating costs, broken down by component, is shown on the following page in Figure 5. Although heating loads dominate the energy consumption in winter months, the total annual operating cost is dominated by the electricity used to cool the facility. An interesting feature to notice in this profile as opposed to the electrical energy consumption shown in Figure 3 is the sharper increase in cost from May to June and the sharper drop-off from September to October. This can be attributed to the cost of electricity rising in the summer months. This analysis shows that the system having the most effect on energy consumption and operating cost in the building is by far the cooling system.

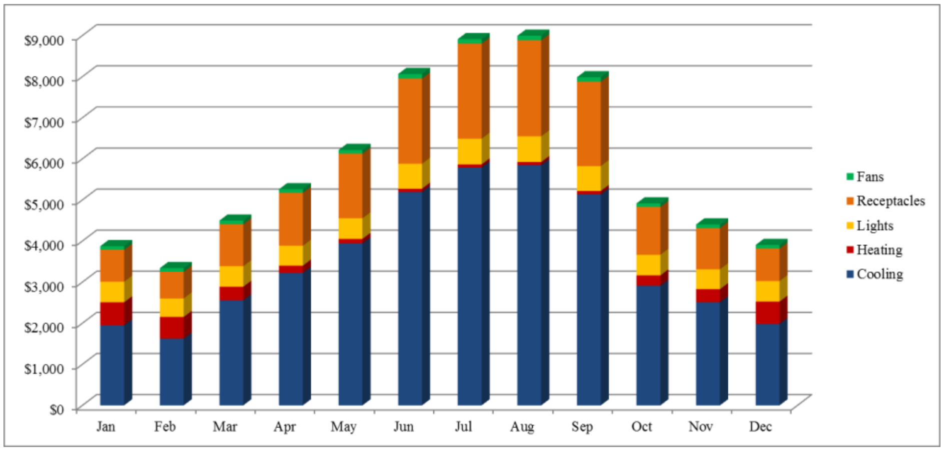

Figure 5: Monthly Operating Costs

(5.3) System Emissions

Important to consider in the system energy use of a building are the potentially harmful emissions associated with the use of this energy. The New Braunfels Regional Rehabilitation Hospital is located in the Electric Reliability Council of Texas (ERCOT) Interconnection, as shown below in Figure 6, taken from the National Renewable Energy Laboratory's (NREL's) Source Energy and Emission Factors for Energy Use in Buildings Report. That document also outlines the amount of energy generated in each region by each source of energy shown in Table 7, displayed below.

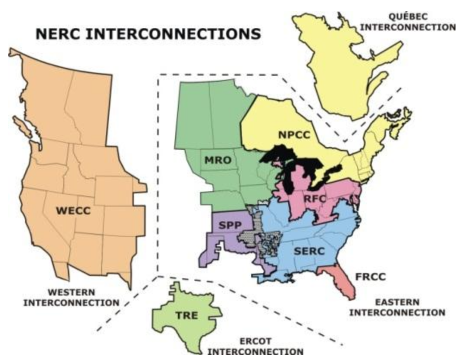

Figure 6: NERC Interconnections Map

Table 7: Percent Electricity Generation by Energy Type

Energy Type	National %	Eastern %	Western %	ERCOT %	Alaska %	Hawaii %
Bituminous Coal	27.8	34.3	13.1	0.0	0.0	1.0
Subbitumious Coal	19.8	19.6	19.8	21.4	9.9	13.1
Lignite Coal	2.3	1.4	0.0	14.8	0.0	0.0
Natural Gas	18.3	12.7	27.4	49.4	55.5	1.5
Petroleum Fuels	2.8	3.6	0.5	0.5	11.5	77.4
Other Fossil Fuel	0.2	0.2	0.3	0.2	0.0	0.2
Nuclear	19.9	23.0	9.9	12.4	0.0	0.0
Hydro	6.8	3.4	24.6	0.3	23.0	0.8
Renewable Fuels	1.5	1.7	1.3	0.2	0.1	4.2
Geothermal	0.4	0.0	2.1	0.0	0.0	1.9
Wind	0.4	0.1	1.0	0.9	0.0	0.1
Solar (PV)	0.0	0.0	0.1	0.0	0.0	0.0
Fossil Fuel Total	71.2	71.8	60.9	86.2	76.9	93.1
Renewable (non hydro)	2.2	1.8	4.6	1.1	0.1	6.1

The NREL's Energy and Emissions Report also specifies the volume of natural gas that needs to be delivered to a site in order to produce a certain capacity of heating. The calculation of delivered natural gas for NBRRH is shown below in Table 8 to be used later in the total emissions calculation.

Table 8: Delivered Natural Gas Calculation

Heating Capacity (BTU)	Natural Gas Heating Value (BTU/ft^3)	Natural Gas Delivered (ft^3)
326900000	1010	323663

Below, Table 9 shows the emission factors associated with the use of electrical energy and on-site combustion of natural gas for twelve prominent pollutants and the calculated annual mass of those pollutants associated with each form of energy. The most abundant pollutants associated with the energy used by the facility are CO_2, CO_{2e} (equivalent carbon dioxide), and solid waste.

Table 9: Emission Factors and Pollutant Mass

Pollutant	Electricity Emission Factor (lb pollutant/kWh electricity)	Mass of Pollutant (lbm/year)	Pre-Combustion Emission Factor (lb pollutant/1000 ft^3 Natural Gas)	Mass of Pollutant (lbm/year)
CO_{2e}	1.84E+00	2783276.00	2.78E+01	8997.75
CO_2	1.71E+00	2586631.50	1.16E+01	3754.49
CH_4	5.30E-03	8017.05	7.04E+01	22785.88
N_2O	4.02E-05	60.81	2.35E-04	7.61E-02
NO_X	2.20E-03	3327.83	1.64E-02	5.31E+00
SO_X	9.70E-03	14672.71	1.22E+00	394.87
CO	9.07E-04	1371.97	1.36E-02	4.40E+00
TNMOC	7.44E-05	112.54	4.56E-05	1.48E-02
Lead	1.42E-07	0.21	2.41E-07	7.80E-05
Mercury	2.79E-08	0.04	5.51E-08	1.78E-05
PM10	1.30E-04	196.64	8.17E-04	2.64E-01
Solid Waste	1.66E-01	251099.90	4.21E+02	136262.12
Delivered Electricity = 1,512,650 kWh				
Delivered Fuel = 323,663 ft^3 Natural Gas				

Though the carbon dioxide values dominate the above figure, the levels of the other pollutants should not be ignored. Sulfur oxides and nitrogen oxides in particular are common results of the combustion process and are significant contributors to the greenhouse effect, acid rain, and local air pollution.

(6.0) Existing Mechanical and Plumbing Systems

(6.1) Equipment

Rooftop Air Handling Units

Three packaged rooftop units supply most of the facility with conditioned air. RTU-1 serves the patient-room wing of the facility, while RTUs 2 and 3 serve the patient therapy and dining/administration portions of the building, respectively. A 100% outdoor air makeup unit serves the kitchen and dining functions in the area served by RTU-3. All of these units are air-cooled and utilize direct, modulating gas-fired heating. A summary of the rooftop units are shown in Table 10 on the next page.

Table 10: Packaged Rooftop Units

Tag	Airflows			Fans		Cooling MBH	Heating MBH
	Supply CFM	Return CFM	OA CFM	Supply RPM	Return RPM		
RTU-1	26,000	26,000	6,850	1,238	652	913	650
RTU-2	12,000	12,000	2,015	1,508	1,023	418	500
RTU-3	17,500	17,500	4,550	1,478	637	686	500
MAU-1	3,500	-	3,500	2,274	-	128	200

RTUs 1-3 are supplied with factory-mounted variable frequency drives on the supply air and exhaust air fans to save fan energy. These VFDs range from 5 to 40 horsepower and operate at 3 phase and 460 volts.

Rooftop units also contain two sets of filter banks, each with a filter differential pressure transducer. These filters are rated MERV 7 and MERV 14, in compliance with ASHRAE Standard 170.

Air Terminal Units

Conditioned air is distributed from each rooftop unit to variable air volume terminal units associated with each zone. RTU-1 supplies 58 VAV boxes in the patient-room wing, while RTU-2 supplies 20 VAV terminal units and RTU-3 delivers air to 30 VAV boxes. Each terminal unit is pressure-independent and is controlled by a supply duct temperature sensor. All VAV boxes also contain zone-level reheat, with the exception of the four terminal units that serve electrical or telecommunication rooms.

Pool Dehumidification Unit

A split-system, air-cooled dehumidification unit maintains occupant comfort at 50%-60% relative humidity in the therapy pool area. This system is automatically controlled to dehumidify the pool room while recycling latent energy back into the pool water and air. By doing so, the pool water heating and space heating requirements are reduced. A summary of the dehumidification unit's characteristics are shown below in Table 11.

Table 11: Pool Dehumidification Unit

Tag	Total Airflow	Hot Water Flow	Moisture Removal	Cooling MBH	Water Reheat MBH
PAC-1A/1B	1900 CFM	10 GPM	20 lb/hr	43	56

Hot Water Boilers

Zone-level reheat in the VAV terminal units are served by hot water from two gas-fired boilers located in the mechanical room. A summary of each boiler's flow and heating capacity are shown in Table 12 below. These non-condensing boilers utilize a Cupro-Nickel heat exchanger and have a glass-lined cast iron lining to limit the common erosion problems associated with Texas's hard water. Each boiler exceeds the less than 10 ppm NOx emission requirement of the Texas Department of Health Services.

Table 12: Heating Hot Water Boilers

Tag	Flow Rate	Input MBh	Output MBH
HWB-1	57 GPM	999	849
HWB-2	57 GPM	999	849

Hydronic Pumps

Two parallel in-line, close-coupled pumps circulate hydronic hot water to the VAV reheat coils throughout the facility. Each pump has a maximum flow of 95 GPM and is furnished with a 5 horsepower variable frequency drive motor for pumping energy savings.

Water Heaters

Two gas-fired commercial water heaters serve the domestic hot water load of the New Braunfels Regional Rehabilitation Hospital. Each unit operates with 96% thermal efficiency that results from a glass-lined tank that prevents lime scale buildup and reduces associated energy losses. Capacity and temperature difference data for these water heaters is shown below in Table 13.

Table 13: Domestic Water Heaters

Tag	Storage Capacity	ΔT	Set Temperature	Input BTUh
WH-1	130 Gallons	100˚F	120˚F	400
WH-2	130 Gallons	100˚F	140˚F	400

Hot Water Circulating Pumps

An in-line, close-coupled pump is headered with each of the water heaters and circulate domestic water to plumbing fixtures throughout the facility. These pumps operate at different flow rates to accommodate the difference in set points of the water heaters.

(6.2) System Operations

Air-Side Operation

As outside air enters each of the packaged outdoor rooftop units, it is mixed with return air via the air-side economizer contained within each unit. The volume of outside and return air that is mixed is based on readings from temperature and humidity sensors located in the outside air duct and return air duct, respectively. This mixed air is drawn through subsequent sections of the air handler by the supply air fan located downstream.

Mixed air then passes through the first bank of filters, which is monitored by a differential pressure transducer (labeled as DPT in Figure 7 on the next page) to ensure the system operator is aware of any flow rate changes through the filter.

Air is then drawn through direct expansion cooling coils arranged in a multi-row, staggered tube configuration. Each unit is equipped with two independent refrigerant loops and interlaced coil circuiting to keep the coils fully active at any load condition.

Cool air then passes through the stainless steel heat exchanger associated with the natural gas fired furnace. The combustion furnace and the combustion air fan are only operational in periods of heating and can modulate between 33% and 100% of the rated capacity.

Conditioned air is then drawn through the supply air fan, which is controlled by a variable frequency drive motor that acts in response to measurements from a pressure sensor in the supply air duct. Supply air passes through a second filter bank before being delivered to each VAV terminal unit. Before return air mixes with outdoor air in the economizer, a portion of air is exhausted via the exhaust air fan, which is controlled through fan tracking.

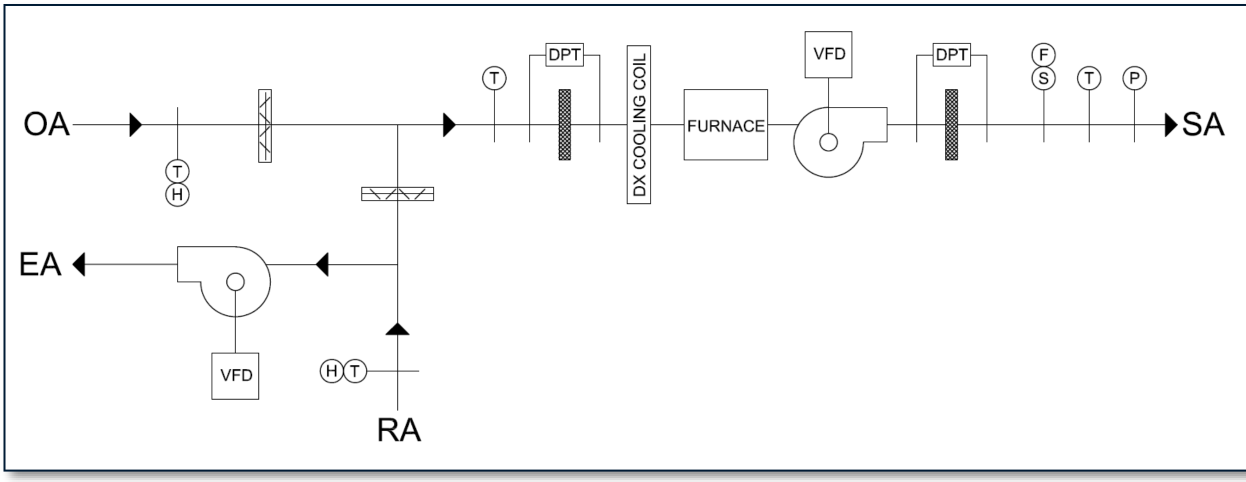

Figure 7: Rooftop Air Handler Schematic

Room humidity level in the therapy pool area is controlled with a specialized pool dehumidification unit that uses vapor compression to both dehumidify air and help heat pool water. Warm, humid air from the pool area passes through the unit's evaporator, causing condensation on the evaporator coil and thus dehumidifying and cooling the air. Refrigerant then gains heat in the compressor before being used to reheat the pool air and then rejecting its remaining heat to pool water to reduce pool heating energy and cost. A schematic of this process is shown below in Figure 8.

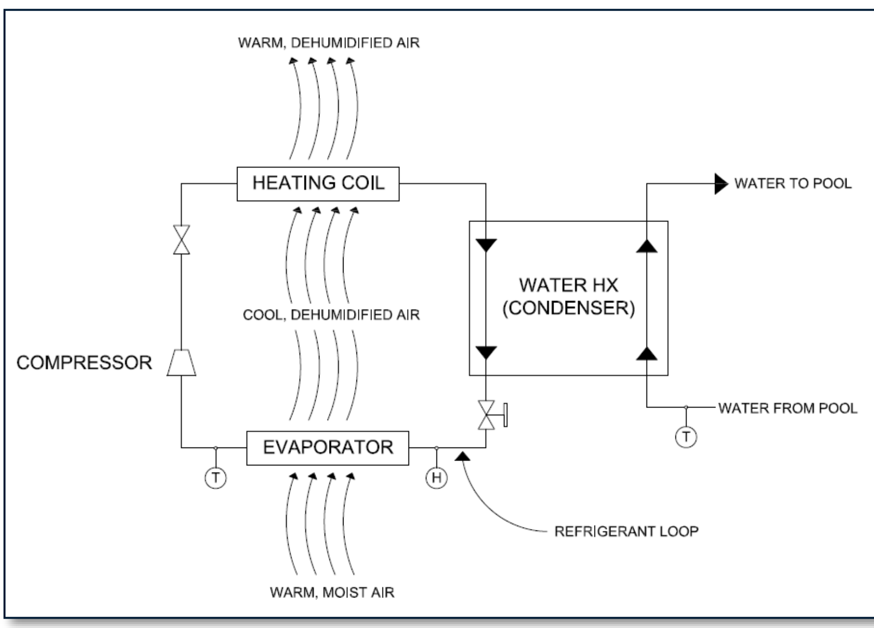

Figure 8: Pool Dehumidification Process

Water-Side Operation

The heating hot water system for the facility utilizes two hot water boilers to heat the primary-secondary loop hydronic hot water system. Heating hot water is primarily circulated through the system by pumps header mounted on each boiler. These pumps are controlled by a boiler control panel that processes differential pressure measurements across space heating loads at various points throughout the facility.

Two auxiliary heating hot water pumps (labeled HP-1 and HP-2 in the hydronic hot water system schematic in Figure 9 below) are arranged in parallel and included on the hydronic supply line. Only one auxiliary pump is operational at a given time; both are controlled by a pressure sensor monitor that allows the stand-by pump to ramp up upon failure of the leading pump.

Note that the facility does not currently utilize any chilled water, so there is no central cooling plant included in the facility or this evaluation of existing systems.

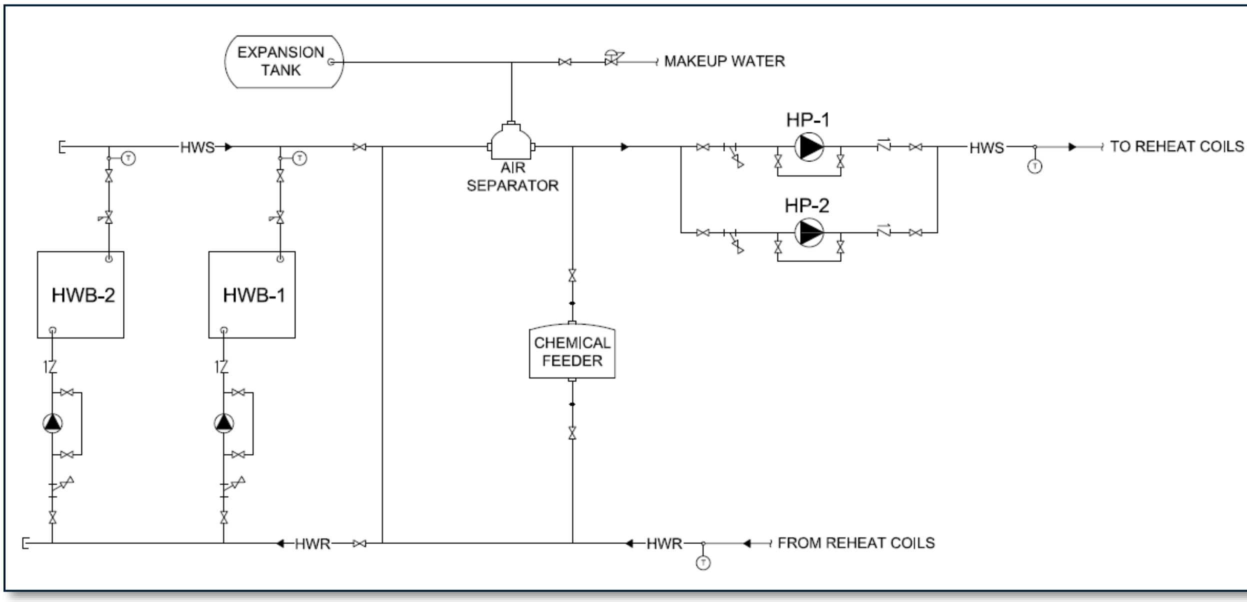

Figure 9: Hydronic Hot Water Piping Schematic

Plumbing Operation

City cold water enters the building and has four possible paths: it could be delivered to the building directly as domestic cold water, be mixed with 140ºF or 110ºF hot water return, or be mixed with 110ºF domestic hot water supply out of Water Heater 1. WH-1 is set to operate at 120ºF and supplies to two three-way mixing valves before being delivered to plumbing fixtures at 110ºF. Water heater 2 supplies high-temperature domestic water directly to plumbing fixtures at 140ºF. A schematic flow diagram of mechanical room plumbing features is shown in Figure 10 on the next page.

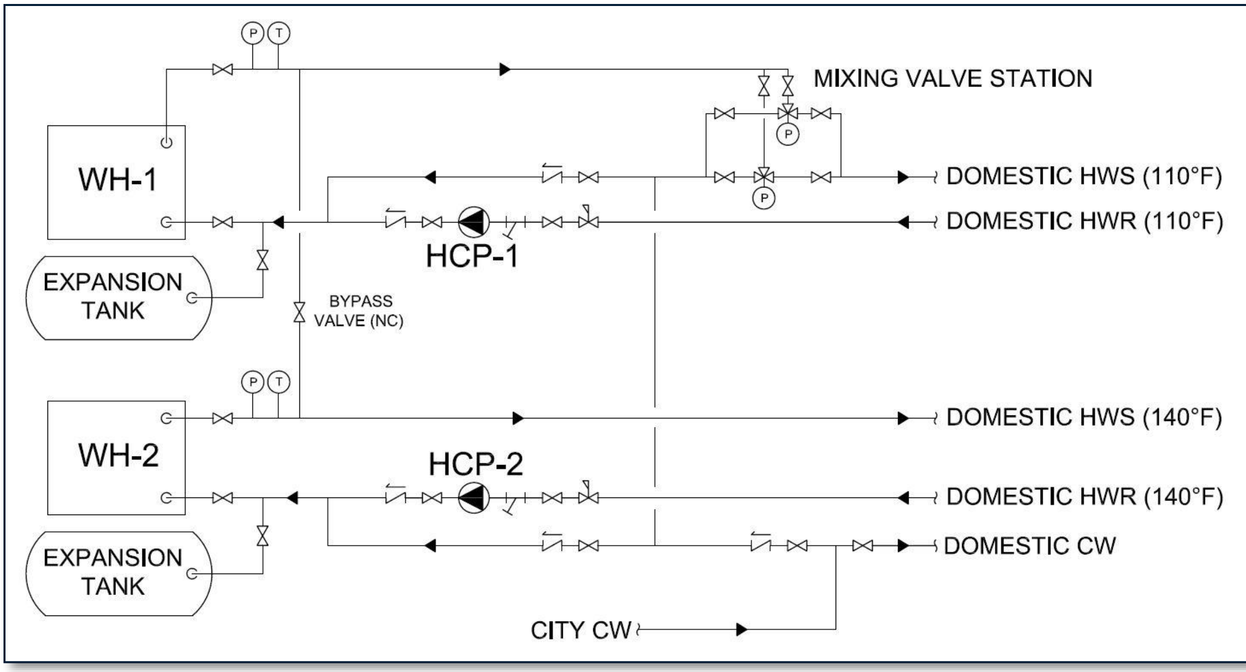

Figure 10: Domestic Water Piping Schematic

(6.3) Space Requirements

The space used by mechanical and plumbing systems is listed below in Table 14. Included in the total lost usable space are the mechanical room, which houses the hot water boilers, domestic water heaters, expansion tanks, and all associated pumps. The pool equipment room that contains the pool dehumidification equipment is included as well as medical gas pump rooms and medical gas access rooms.

Table 14: Lost Usable Space

Room	Area (SF)
Mechanical Room	276
Pool Equipment	323
Medical Gas Pump Room	188
Medical Gas Access	99
Total Lost Usable Space:	886

The total space used by mechanical, plumbing, and medical gas systems is only 1.8% of the total building area, which is significantly lower than industry average and may cause space issues when considering system alterations. Not included in this lost usable space calculation are electrical and telecommunication rooms.

A layout of the southwest mechanical room, which houses boilers, water heaters, expansion tanks, and all in-line pumps is shown on the following page in Figure 11.

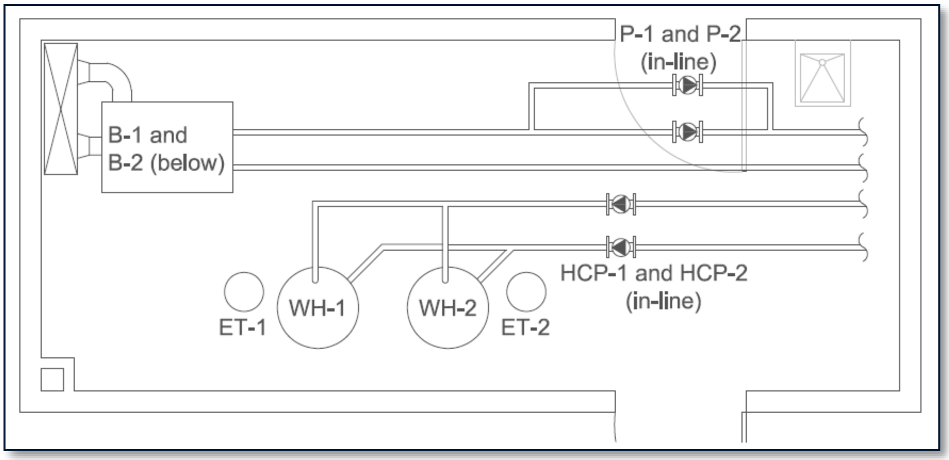

Figure 11: Existing Mechanical Room Layout

(7.0) System Energy

(7.1) Energy Sources and Rates

NBRRH receives electricity and city water from the New Braunfels Utility (NBU) company, which delivers electricity and water to the entire municipality. The utility company designates the rehabilitation hospital as a large general service facility, the rate structure of which is shown for these utilities in Table 15 below. NBU also charges large general service facilities an annual fee of $1,437.53 for potable water delivery.

Also delivered directly to the facility is natural gas for all space and water heating processes. NBU does not deliver natural gas, so an average natural gas price for the state of Texas was taken from Center Point Energy and used for the energy analysis.

Table 15: Energy Rates

Utility	Rate	
	October - May	June - September
Electric Consumption	$0.04 / kWh	$0.05 / kWh
Electric Demand	$4.40 / kW	
Natural Gas	$0.9573 / therm	
City Water	$1.922 / thousand gallons	

(7.2) Energy Use

There currently exists no data on the annual energy consumed by NBRRH because the facility has only been occupied for eight months. Additionally, the mechanical designer did not perform an energy analysis that attempted to model the facility's energy consumption because of the accelerated project schedule. The only available indication of the actual energy used within the facility is the Trane Trace analysis performed.

While this analysis was fairly comprehensive, it is still impossible to assert these results as accurate due to several assumptions made, the variability of weather conditions, and the limitations of the modeling software available. The heating and cooling loads calculated in this model were considerably lower than what the mechanical engineer designed the equipment to handle, so it could be likely that the facility will see higher energy consumption than the load calculation shows.

(8.0) Mechanical System First Cost

The total cost of material and labor associated with the mechanical system of the New Braunfels Regional Rehabilitation Hospital is $1.3 million, according to the mechanical contractor's records. A breakdown of each component's equipment and labor costs is shown below in Table 16. The two largest contributors to this total include ductwork and mechanical piping, which are very labor-intensive. This total mechanical system cost translates to $26.29 per square foot. Not included in this system cost summary are plumbing, fire protection, and medical gas equipment.

Table 16: Mechanical System Cost Breakdown

Item	Material	Labor	Total
RTUs	$ 199,391	$ 12,000	$ 211,391
Air Distribution Equipment	$ 114,480	$ 18,000	$ 132,480
Ductwork and Insulation	$ 190,277	$ 206,566	$ 396,843
MAU System	$ 91,797	$ 20,000	$ 111,797
Pool Dehumidification Unit	$ 20,757	$ 5,000	$ 25,757
Boilers and Control Interface	$ 23,624	$ 7,000	$ 30,624
Hydronic Distribution Equipment	$ 4,450	$ 7,000	$ 11,450
Mechanical Piping	$ 73,889	$ 155,869	$ 229,758
DDC Controls	$ 44,700	$ 105,200	$ 149,900
Totals:	**$ 763,365**	**$ 536,635**	**$ 1,300,000**

(9.0) ASHRAE Standard 62.1 Compliance Evaluation

ASHRAE Standard 62.1 – 2007 addresses ventilation requirements for acceptable indoor air quality. This standard, while comprehensive for many building applications, was considered insufficient for an analysis of a health care facility. For this particular compliance evaluation, ASHRAE Standard 170 – 2008, Ventilation of Health Care Facilities, was used in areas where Standard 62.1 was deemed insufficient. Such areas are explicitly stated in this report.

(9.1) Section 5: Systems and Equipment

Section 5.1: Natural Ventilation

For purposes of occupant safety, all windows in the facility are inoperable. Natural ventilation was not used in this facility due to the complex ventilation requirements of hospitals.

Section 5.2: Ventilation and Air Distribution

VAV terminal units have fixed outdoor air damper positions that comply with the minimum required ventilation airflow for each space as defined by Section 6 of Standard 62.1, which is discussed later in this report. The mechanical system design utilizes fully-ducted supply and return air, so the ventilation system is not affected by issues common to a plenum air distribution system.

Section 5.3: Exhaust Duct Location

All exhaust ducts are specified to be negatively pressurized and operate at a SMACNA -4.0" w.g. static pressure class so that potentially harmful exhaust air cannot leak into the plenum space. The maximum velocity of these exhaust ducts is specified to be 4000 FPM.

Duct and seam and joint construction, sheet metal thicknesses, and hangars and supports for all ductwork, including exhaust, are specified to comply with SMACNA's *HVAC Duct Construction Standards – Metal and Flexible."*

Section 5.4: Ventilation System Controls

The outdoor and return air dampers are controlled by a modulating, spring-return actuator within the air handling units. The outdoor air dampers modulate in response to the unit's temperature control system. An adjustable enthalpy control is also included in the units to monitor the outdoor air's dry-bulb temperature and relative humidity. If the outdoor air is deemed suitable, free-cooling can be achieved via the position of the outdoor air dampers.

Section 7.2.2 of ASHRAE Standard 170 establishes that protective environment (PE) rooms must maintain proper pressurization levels. In this facility, airflow to each patient room is regulated by a differential pressure sensor, ensuring that the patient rooms are always positively pressurized in relation to the attached toilet room and corridor. Although these patient rooms are not classified as PE rooms, this control system is safe practice and is in compliance with Section 7 of Standard 170.

Section 5.5: Airstream Surfaces

Interior duct lining is specified to be coated with an antimicrobial, erosion-resistant coating. This also acts as a moisture repellant, prohibiting mold growth along the airstream. This antimicrobial compound is tested for efficacy in HVAC systems by a nationally recognized testing laboratory registered by the EPA.

The solvent-based adhesive for this coating complies with NFPA 90 and ASTM C 916 and has a VOC content of less than 80 g/L. This adhesive also complies with the requirements of *Standard Practice for the Testing of Volatile Organic Emissions from Various Sources Using Small-Scale Environmental Chambers*, printed by the Texas Department of Health Services.

Section 5.6: Outdoor Air Intakes

All outdoor air intakes are located such that the minimum distance to any potential air-contaminating source complies with Table 5-1 of Standard 62.1.

Specifically, all outdoor air intakes for the rooftop air handling units are located more than 30 feet from any rooftop exhaust fan. Similarly, the air intake for the 100% outdoor air makeup air unit serving the kitchen is located over 30 feet from all kitchen exhaust fans.

All intake and exhaust ductwork are pitched for rain water runoff per SMACNA guidelines and are provided with support rails and galvanized bird screen hoods.

Section 6.3 of ASHRAE Standard 170 requires outdoor air intakes to be a minimum of 25 feet from cooling towers and all discharge air vents for units serving health-care functions. RTU-1 complies with this more stringent requirement, as stated above.

Section 5.7: Local Capture of Contaminants

None of the non-combustion equipment in the facility produces contaminants, so Section 5.7 does not apply to this analysis.

Section 5.8: Combustion Air

All fuel-burning rooftop units are not enclosed, and thus have the necessary available combustion air needed for proper operation. The gas-fired, non-condensing boilers located in the mechanical room are served by a 48" x 16" combustion air intake louver, which provides adequate combustion air, according to the manufacturer's product data. Adequate combustion air is provided to the gas-fired water heaters used to heat domestic hot water through sidewall inlets as specified by the manufacturer.

Section 5.9: Particulate Matter Removal

All outside air is pre-filtered through 2" MERV 7 filters and additionally filtered by 12" MERV 14 final filters before being supplied to the VAV terminal units. The filter media used is micro-glass fiber and the separator material is a thermoplastic resin spaced at 25 mm intervals. This draw-through air filtration sequence surpasses the required rating of MERV 6 prescribed in this section.

Section 6.4 of ASHRAE Standard 170 requires two filter banks: the first placed upstream of heating and cooling coils to filter mixed air, and the second downstream of all cooling coils and the supply fan. Both of these requirements are met in compliance with Standard 170, as shown in Figure 12 below.

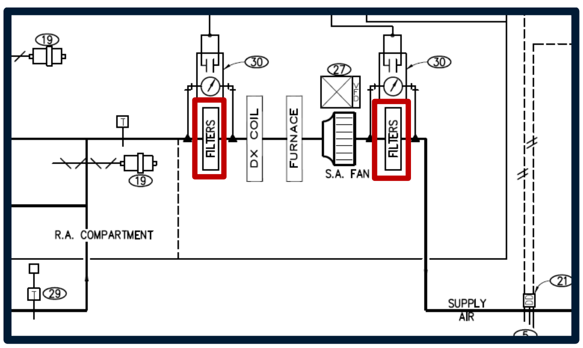

Figure 12: Filter Locations

Section 5.10: Dehumidification Systems

Dehumidification for the majority of the facility is achieved using modulating hot gas reheat, which conditions air to less than 60% relative humidity, which exceeds the requirements of this section.

A packaged dehumidification system is used to maintain occupant comfort and swimming pool environment in the therapy pool area. This unit recovers sensible and latent heat as needed to put back in the air or pool water. The unit is designed to maintain the room at 86ºF and 60% relative humidity while keeping the pool water at 80ºF.

Section 5.11: Drain Pans

Stainless steel drain pans are provided with the cooling coils on all rooftop units. Each pan has a minimum slop of 1/8" per foot to ensure positive draining. The pans are specified to be connected to a threaded drain connection that extends through the base of the unit. Each pan is specified to extend underneath the cooling coil connections and beyond the leaving side of the coil.

Section 5.12: Finned-Tube Coils and Heat Exchangers

Drain pans are specified to be installed beneath all mechanical dehumidification units, as noted in specification section 238416. No condensate-producing heat exchangers are used in the facility, so drain pans are not required for this application.

All equipment cooling coils have more than 18 inches of access space, and thus the required minimum pressure drop stated in Section 5.12.2 is not applicable to this project.

Section 5.13: Humidifiers and Water-Spray Systems

None of the equipment used in the building utilizes humidifiers or water-spray systems, so Section 5.13 is inapplicable in this evaluation.

Section 5.14: Access for Inspection, Cleaning, and Maintenance

Service doors are included on both sides of all sections of the rooftop air-handling units for maintenance purposes. A safety catch is provided in the latch system of each door to protect against injury if the door is opened during fan operation.

Air distribution equipment within the building is located above a lay-in ceiling so maintenance personnel can easily access equipment for cleaning and repairs.

Section 5.15: Building Envelope and Interior Surfaces

All exterior surfaces in the building contain a layer of moisture protection. Exterior walls have a ⅝" thick WPC sheathing and all roof types contain a ¼" thick TPO membrane, in compliance with Section 5.15.1 of Standard 62.1.

Specification section 230700 notes requirements for HVAC insulation to prevent condensation from occurring on exterior duct surfaces, in order to comply with Section 5.15.2 of this standard.

Section 5.16: Buildings with Attached Parking Garages

Parking for the rehabilitation hospital is not attached to the facility, so Section 5.16 is not explored in this evaluation.

Section 5.17: Air Classification and Recirculation

All air in the areas served by RTU-2 and RTU-3, other than in the kitchen and restrooms, are classified as Class 1 air, and can thus be recirculated to any space. Air in the kitchen and restrooms are exhausted straight out of the building and are not recirculated, so the air classification of these spaces is inconsequential.

Air in the area served by RTU-1, including patient and treatment rooms, is classified as Class 2 air, which may be recirculated to other areas with a Class 2 air classification. Exhaust air from the toilet rooms in this area is taken straight out of the building, so the classification of this air was not considered.

Section 5.18: Requirements for Buildings Containing ETS Areas

As a health care facility, the New Braunfels Regional Rehabilitation Hospital does not have any environmental tobacco smoke areas, so Section 5.18 is not applicable to this facility.

Additional Requirements of ASHRAE Standard 170

In accordance with Section 6.1.1 of Standard 170, all patient rooms must maintain the ventilation requirements stated in Section 3.2.1 of this report upon loss of electrical power. This requirement is noted in the electrical specification section 230900.

(9.2) Section 6: Procedures

Section 6 of ASHRAE 62.1 outlines a ventilation rate procedure to determine the minimum outdoor ventilation air required for a system based on the occupancy distribution and type as well as zone size. Other factors taken into account in a ventilation analysis prescribed by this standard are zone air distribution effectiveness and the primary outdoor air fraction.

ASHRAE Standard 170 was used in conjunction with 62.1 to determine the ventilation air required for the rooftop unit serving the patient rooms in the facility. Standard 170 outlines an air changes per hour procedure based on individual room function and volume.

(9.2.1) Ventilation Rate Procedure

The equation used to calculate the breathing zone outdoor airflow (V_{bz}) for each zone is given by Equation 6-1 in Standard 62.1, shown below.

$$V_{bz} = \left(R_p \cdot P_z\right) + \left(R_a \cdot A_z\right) \qquad \textbf{(Equation 1)}$$

where R_p = outdoor airflow rate required per person (from Table 6-1)
R_a = outdoor airflow rate required per unit area (from Table 6-1)
P_z = largest number of people expected to occupy that zone
A_z = net floor area of the zone

The zone air distribution effectiveness (E_z) of the distribution system is determined to be 1.0, in accordance with Table 6-2 of Standard 62.1. Thus the Zone Outdoor Airflow (V_{oz}) is the same as V_{bz}, given by Equation 6-2 of Standard 62.1, shown below.

$$V_{oz} = {}^{V_{bz}}\!/_{E_z} \qquad \textbf{(Equation 2)}$$

The primary outdoor air fraction (Z_p) is the minimum percentage of supply air that is outdoor ventilation air, and is calculated by taking a ratio of zone outdoor airflow to the zone primary airflow (V_{pz}), as shown in Equation 6-5 of Standard 62.1 on the following page.

$$Z_p = {}^{V_{oz}}\!/_{V_{pz}} \qquad \textbf{(Equation 3)}$$

The total outdoor air intake is then determined to be the product of the primary outdoor air fraction and the total supply air used by the system.

Using this prescriptive procedure, the total ventilation rates of RTU-2 and RTU-3 were determined to be compliant with the minimum requirements given by Table 6-1 of Standard 62.1. Detailed calculations are shown in Tables C2 and C3 of Appendix C.

(9.2.2) Air Changes per Hour Procedure

An ASHRAE Standard 170 air changes per hour procedure was performed to determine the ventilation compliance of areas that 62.1 did not address. These spaces were primarily patient rooms, patient toilet rooms, medical gas storage, and other hospital-specific rooms.

To perform this compliance procedure, the volume of each room was calculated and used to convert the supply air CFM to air changes per hour and outdoor air changes per hour. For each room, these values were then compared to those given in Table 7-1 of Standard 170, a sample of which is shown below in Figure 13.

TABLE 7-1 Design Parameters

Function of Space	Pressure Relationship to Adjacent Areas (n)	Minimum Outdoor ach	Minimum Total ach	All Room Air Exhausted Directly to Outdoors (j)
SURGERY AND CRITICAL CARE				
Classes B and C operating rooms, (m), (n), (o)	Positive	4	20	N/R
Operating/surgical cystoscopic rooms, (m), (n) (o)	Positive	4	20	N/R
Delivery room (Caesarean) (m), (n), (o)	Positive	4	20	N/R
Radiology waiting rooms (q)	Negative	2	12	Yes
Class A Operating/Procedure room (o), (d)	Positive	3	15	N/R
INPATIENT NURSING				
Patient room (s)	N/R	2	6	N/R
Toilet room	Negative	N/R	10	Yes
Newborn nursery suite	N/R	2	6	N/R
Protective environment room (f), (n), (t)	Positive	2	12	N/R

Figure 13: Example Air Change Rate Requirements

Each room was evaluated to determine if they met both air change requirements of Standard 170. 92 of the 105 rooms evaluated, including all patient rooms and toilet rooms, complied with these requirements. Areas that did not comply with these requirements included the hospital's public restrooms, medical gas storage, and linen storage rooms.

The minimum outdoor air change per hour calculated was then used, in conjunction with the Standard 62.1 analysis of the other rooms on the system, to determine the ventilation air requirements for RTU-1. It was found that, even with these more stringent requirements, the system as a whole met the necessary ventilation rate outlined in Standard 62.1. A summary of calculations for RTU-1 is shown in Table C1 of Appendix C.

Results of both the prescriptive ventilation rate procedure and the air change per hour procedure are shown on the next page in Table 17.

Table 17: Ventilation Compliance Summary

System	ASHRAE 62.1			ASHRAE 170	
	OA Required	Minimum OA	Compliant?	% Rooms Compliant	Ventilation Compliant?
RTU-1	3730 CFM	6850 CFM	**Y**	88%	**Y**
RTU-2	1327 CFM	2015 CFM	**Y**	N/A	N/A
RTU-3	2314 CFM	4550 CFM	**Y**	N/A	N/A

(9.3) Standard 62.1 Analysis Conclusions

This analysis has determined that the New Braunfels Regional Rehabilitation Hospital is completely compliant with ASHRAE Standard 62.1 – 2007. Several areas of the HVAC system design, such as humidity control and air filtration, well exceed the requirements put in place by this standard. On a system level, the amount of ventilation air supplied to the building exceeds what is required by 62.1.

The facility is also very close to overall compliance with ASHRAE Standard 170, Ventilation of Health Care Facilities. An analysis of the patient room wing shows that this unit complies with Standard 62.1, even by the more stringent air changes per hour ventilation rate calculation. However, several individual rooms do not meet the necessary outdoor air change rate designated by Standard 170.

(10.0) ASHRAE Standard 90.1 Compliance Evaluation

ASHRAE Standard 90.1 is the energy standard for non-low rise, residential buildings. Though compliance with this standard is not required, the sections discussed in the standard should be considered for a responsible mechanical system design.

(10.1) Section 5: Building Envelope

The New Braunfels Regional Rehabilitation Hospital is located in climate zone 2A, defined by Table B-1 of ASHRAE Standard 90.1. This zone implies a hot, humid climate, as shown in Figure 14 below.

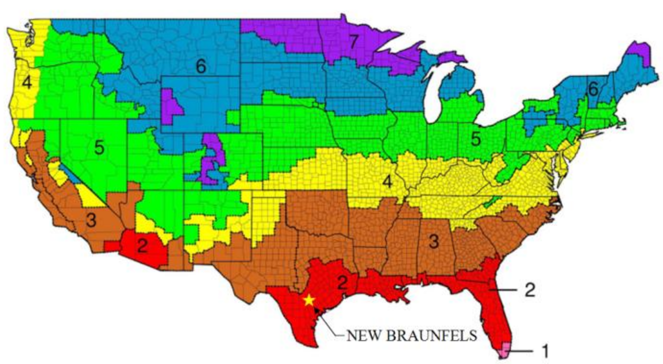

Figure 14: ASHRAE Climate Zones

Because the facility has a 20.5% fenestration area, with no skylights, the prescriptive compliance path specified in Section 5.5 of Standard 90.1 can be followed. The building envelope requirements for climate zone 2 are outlined in Table 5.5-2 of the standard.

A summary of the building envelope characteristics is shown below in Table 18. All of the requirements for assembly maximum U-Values are met, though the solar heat gain coefficient (SHGC) of both the windows and the curtain wall system are higher than the maximum allowed SHGC in this standard.

Table 18: Building Envelope Requirements Compliance

Element	U-Value	Max. U-Value	SHGC	Max. SHGC	Compliance	
					U-Value	SHGC
Metal Deck Roof	0.03569	0.048	N/A	N/A	**Y**	N/A
Above-Grade Walls	0.05543	0.113	N/A	N/A	**Y**	N/A
4" HW Concrete Floor	0.6587	F = 0.730	N/A	N/A	**Y**	N/A
Windows	0.35	0.75	0.32016	0.25	**Y**	**N**
Curtian Wall Glazing	0.35	0.7	0.32016	0.25	**Y**	**N**

(10.2) Section 6: Heating, Ventilating, and Air Conditioning

The gross floor area of the New Braunfels Regional Rehabilitation Hospital is over 25,000 ft^2; therefore, the compliance will be determined by an analysis of Section 6.4, Mandatory Provisions and 6.5, Prescriptive Path.

Section 6.4.2: Load Calculations

In accordance with the requirements of this section, the mechanical designer calculated heating and cooling system design loads with ASHRAE standards and professional engineering practices in mind to adequately size equipment and systems.

Section 6.4.3: Controls

Each zone within the facility is individually controlled by temperature sensors within the zone, in compliance with Standard 90.1. In certain areas where proper zone pressurization is integral to the health of the occupants, an override of these temperature controls is provided in the form of a differential pressure sensor.

Section 6.4.4: HVAC System Construction and Insulation

Specification section 230700 notes that the HVAC insulation thickness and R-Values are to comply with ASHRAE Standard 90.1-2004. This specification also states that joint sealants and metal jacket flashing sealants are to comply with local construction requirements.

Section 6.5.1: Economizers

According to Table 6.5.1 of Standard 90.1, there is no economizer requirement for climate zone 2A, though all of the rooftop units are equipped with a 0-100% outside air economizer to take advantage of free-cooling opportunities.

Section 6.5.2: Simultaneous Heating and Cooling Limitation

Section 6.5.2 requires temperature controls that are capable of preventing reheating and cooling to individual zones. VAV-level reheat is included in the mechanical system of this facility, but many of the zones fall under the exception of having special pressurization relationships and cross-contamination requirements, so this section of Standard 90.1 is not applicable.

Section 6.5.3: Air System Design and Control

The fan power limitation requirement outlined in Section 6.5.3 requires fans to comply with Table 6.5.3.1.1A, shown below in Figure 15.

TABLE 6.5.3.1.1A Fan Power Limitation[a]

	Limit	Constant Volume	Variable Volume
Option 1: Fan System Motor Nameplate hp	Allowable Nameplate Motor hp	$hp \le CFM_S \cdot 0.0011$	$hp \le CFM_S \cdot 0.0015$
Option 2: Fan System bhp	Allowable Fan System bhp	$bhp \le CFM_S \cdot 0.00094 + A$	$bhp \le CFM_S \cdot 0.0013 + A$

[a] where
CFM_S = the maximum design supply airflow rate to conditioned spaces served by the system in cubic feet per minute
hp = the maximum combined motor nameplate horsepower
bhp = the maximum combined fan brake horsepower
A = sum of $(PD \times CFM_D/4131)$
where
PD = each applicable pressure drop adjustment from Table 6.5.3.1.1B in in. w.c.
CFM_D = the design airflow through each applicable device from Table 6.5.3.1.1B in cubic feet per minute

Figure 15: Fan Power Limitation Requirements

A summary of fan compliance calculations is shown in Table 19 below. All exhaust fans in the facility were determined to be compliant. The only non-compliant fan was the supply fan serving the IV Prep room.

Table 19: Fan Power Limitation Summary

Fan	CFM	HP	CFM·0.0011	Compliant?
EF-1	700	0.25	0.77	Y
EF-2	1840	0.50	2.02	Y
EF-3	1020	0.33	1.12	Y
EF-4	630	0.25	0.69	Y
EF-5	1070	0.33	1.18	Y
EF-6	300	0.08	0.33	Y
EF-7	1250	0.50	1.38	Y
EF-8	860	0.25	0.95	Y
EF-9	260	0.03	0.29	Y
EF-10	400	0.25	0.44	Y
EF-11	600	0.03	0.66	Y
KEF-1	3600	1.50	3.96	Y
KEF-2	750	0.50	0.83	Y
GEF-1	2000	0.50	2.20	Y
SF-1	570	1.00	0.63	N

Section 6.5.4: Hydronic System Design and Control

The total pump system power of the hydronic hot water system is a maximum of 10 HP, which equals, but does not exceed the requirements needed to apply Section 6.5.4.

Section 6.5.5: Heat Rejection Equipment

None of the fans used in the HVAC system is powered by a motor of above 7.5 HP, and thus Section 6.5.5 is not considered in this report.

Section 6.5.6: Energy Recovery

The rehabilitation hospital's mechanical systems do not use any exhaust air heat recovery or service water heat recovery systems, thus Section 6.5.5 is not applicable to this compliance analysis.

Section 6.5.7: Exhaust Hoods

The largest exhaust hood in the facility is for the kitchen exhaust fan, which pulls 3600 CFM. This is not a large enough airflow rate to warrant an analysis of Section 6.5.7.

Section 6.5.8: Radiant Heating Systems

There are no radiant heating systems in the New Braunfels Regional Rehabilitation Hospital. That being said, Section 6.5.8 is inapplicable.

Section 6.5.9: Hot Gas Bypass Limitation

A hot gas bypass system is used in the condensing section of each of the rooftop units. Control of hot gas bypass is factory installed on one of the refrigerant coils for constant capacity control of up to 25%, as specified. This is in accordance with Table 6.5.9 and Section 6.5.9 of ASHRAE Standard 90.1.

(10.3) Section 7: Service Water Heating

Compliance with Section 7 of ASHRAE Standard 90.1 will be determined by Section 7.4, Mandatory Provisions.

Table 7.8 specifies that the minimum performance requirement for gas-fired hot water heating boilers with input between 300 and 12,500 MBH is 80% thermal efficiency. Both hot water boilers used to serve the VAV reheat coils have an 849 MBH output given a 999 MBH input, according to the schedules on the mechanical design drawings. This gives these boilers a thermal efficiency of 85%, complying with this standard.

This standard also requires the water heaters used for domestic water supply to have an 80% thermal efficiency. According to manufacturer's data, these water heaters have a 96% thermal efficiency in compliance with Standard 90.1.

The dehumidification unit that uses rejected heat to maintain pool temperature has a time switch on the unit and fuse disconnect next to the unit, which is in compliance with Section 7.4.5.

(10.4) Section 8: Power

The New Braunfels Regional Rehabilitation Hospital is noted to comply with the latest version of the National Electric Code (NEC) in specification section 260100. The latest version of the NEC requires feeder conductors requires feeder conductors to be sized for a maximum voltage drop of 3% at design load, whereas Section 8.4.1.1 of ASHRAE Standard 90.1 requires feeder conductors to be sized at a maximum of 2% voltage drop at design load.

The power distribution system of the facility does not necessarily comply with Standard 90.1.

(10.5) Section 9: Lighting

The space-by-space method for calculating interior lighting power allowances, as described in Section 9.6.1 of ASHRAE Standard 90.1, was used to determine lighting power density compliance for the facility. The lighting power density was calculated for a sample of rooms and compared to the allowable lighting power density from Table 9.6.1 of Standard 90.1. A summary of these comparisons is shown in Table 20 below.

A majority of the spaces were not compliant with the maximum lighting power density. Several of these rooms have safety and health issues associated with proper lighting, which could account for the high lighting power densities in such areas.

Table 20: Sample Lighting Power Densities

Sample Room	Lighting Power (W)	Area (SF)	Lighting Power Density (W/SF)	Required Lighting Power Density (W/SF)	Compliant?
Patient Room	180	214	0.84	0.7	N
Restroom	105	51	2.06	0.9	N
Thearpy Room	90	99	0.91	0.9	N
Exam Room	90	100	0.90	1.5	Y
Admin. Office	180	99	1.82	1.1	N
Hospital Lobby	780	859	0.91	0.8	N
Medical Storage	120	99	1.21	1.4	Y
General Storage	300	419	0.72	0.3	N
Dining Room	1320	1435	0.92	0.9	N
Mechanical Room	180	276	0.65	1.5	Y
Electrical Room	120	106	1.13	1.5	Y
Conference Room	220	119	1.85	1.3	N
Therapy Gym	1770	1829	0.97	0.9	N

(10.6) Section 10: Other Equipment

Section 10.4.1 of Standard 90.1 states that electric motors shall comply with the minimum nominal efficiencies outlined in Table 10.8 of the standard. Motors associated with the two heating hot water pumps operate at 1750 RPM with a maximum 5 horsepower, with a minimum efficiency of 58%. This is not in compliance with the required minimum full-load efficiency of 87.5% set in place by Standard 90.1.

In the hot Texas climate, these heating hot water pumps will rarely be operating at full load, so meeting a high full-load efficiency for these pumps was likely not a cost-effective design.

(10.7) Standard 90.1 Analysis Conclusions

The requirements of ASHRAE Standard 90.1 were not completely met by the mechanical system design of the New Braunfels Regional Rehabilitation Hospital.

Due to the critical functions of the building, the focus of the mechanical system design was on occupant safety and comfort, so sustainability may have taken a back seat in some areas. Although the facility did not completely comply with the standard and did not strive for LEED Certification, several areas of Standard 90.1 were met and even exceeded with a responsible design in mind.

The building envelope U-Values are well below those required by this standard, even if the shading coefficient of the glazing system did not comply. All exhaust fans met the fan power limitation requirements and air-side economizers are used, though they are not required for this climate region.

Despite not complying entirely with ASHRAE Standard 90.1, the New Braunfels Regional Rehabilitation Hospital meets the occupants' safety and comfort needs with an environmentally conscious design.

(11.0) LEED Analysis

Leadership in Energy and Environmental Design (LEED) is a rating system developed by The United States Green Building Council (USGBC) to promote the benefits of sustainable building design and construction. The New Braunfels Regional Rehabilitation Hospital did not strive for LEED certification during its design or construction phase due to the accelerated project schedule and cost concerns, but several measures were taken with the environment and energy efficiency in mind.

Energy and Atmosphere and Indoor Environmental Quality are two categories through which a building can gain multiple LEED version 2.2 credits that are particularly important when discussing environmentally-conscious mechanical system design. Each credit in these two categories will be investigated in this report. A more explicit breakdown of how many points each category and credit can earn is included in Appendix C of this report.

(11.1) Energy and Atmosphere (EA)

✓ EA Prerequisite 1: Fundamental Commissioning of the Building Energy Systems

The purpose of this prerequisite is to verify that the facility's mechanical systems are installed to meet the design and construction documents as well as the owner's project requirements. To meet this prerequisite, a Commissioning Authority must oversee the proper installation and of the commissioning activities described in the LEED version 2.2 checklist.

Specification 230800 – Commissioning of HVAC states the Commissioning Authority's responsibilities, which are compliant with this prerequisite.

✘ EA Prerequisite 2: Minimum Energy Performance

The purpose of this prerequisite is to establish a minimum energy efficiency that the building and its systems must meet. In order to meet this prerequisite, the building design must comply with the mandatory provisions and prescriptive requirements of ASHRAE 90.1-2004.

An analysis of Standard 90.1 is included in Technical Report 1, which finds that NBRRH is not completely compliant with the standard's requirements, and thus this prerequisite is not met.

Although the facility will not be able to gain any Energy and Atmosphere credits because this prerequisite is not met, each credit will still be investigated for the purpose of potential future LEED Certification.

✓ EA Prerequisite 3: Fundamental Refrigerant Management

The purpose of this prerequisite is to ensure ozone depletion reduction through the building's use of non-CFC-based refrigerants. No CFC-based refrigerants were used in the facility's systems, so this prerequisite is met.

✘ EA Credit 1: Optimize Energy Performance
Because the facility does not already comply with the baseline building performance prescribed in ASHRAE Standard 90.1-2004, this credit cannot be obtained until improvements are made to bring the building systems to comply with this standard.

✘ EA Credit 2: On-Site Renewable Energy
No techniques are currently used to garner on-site renewable energy, so this credit cannot be obtained. Several renewable energy sources, including geothermal, solar, and hydro, are available on or near the site, so this credit could be obtained through a system re-design.

✓ EA Credit 3: Enhanced Commissioning
Specification 230800 indicates that the Commissioning Agent shall conduct commissioning design reviews at all phases of construction necessary to gain this credit. Additionally, the Commissioning Agent is required to meet all other requirements of this credit, including having documented experience in two previous projects, and developing a future operating systems manual. This credit would be obtained if all EA prerequisites were met.

✓ EA Credit 4: Enhanced Refrigerant Management
This credit can be achieved with refrigerant performance above EA Prerequisite 3. None of the equipment used in the mechanical system of the facility uses refrigerants, so this credit is achieved with the current system design.

✘ EA Credit 5: Measurement and Verification
There is no measurement and verification plan currently employed in the operation of the facility at the point of five months occupancy, so this credit cannot be obtained for the project.

✘ EA Credit 6: Green Power
Because there is no electricity being generated by renewable resources in the facility, this credit cannot be obtained until a renewable energy source is implemented in system design. This credit could be obtained if future system upgrades are able to provide 35% of the building's electricity through a renewable source.

(11.2) Indoor Environmental Quality (EQ)

✓ EQ Prerequisite 1: Minimum IAQ Performance
The purpose of this prerequisite is to enhance the comfort and well-being of building occupants through proper indoor air quality. In order to meet this prerequisite, the facility must comply with the minimum requirements of Sections 4-7 in ASHRAE Standard 62.1-2004. These sections govern the issues of outdoor air quality, systems and equipment, procedures, and construction and system start-up.

Technical Report 1 includes a detailed analysis of Sections 5 and 6 of Standard 62.1 and determines that NBRRH is completely compliant with those two sections.

Specification section 234100 – Particulate Air Filtration states that compliance with ASHRAE Standard 62.1 Sections 4 and 7 is mandatory. Thus, this prerequisite is completely met.

✓ EQ Prerequisite 2: Environmental Tobacco Smoke (ETS) Control

The intent of this prerequisite is to reduce the exposure of occupants and ventilation air to environmental tobacco smoke. One way to meet this prerequisite is to prohibit smoking in the building and designate smoking areas 25 feet from any air entries, including air intakes.

As a hospital and physical therapy facility, smoking is prohibited on premises and thus this requirement is met.

✘ EQ Credit 1: Outdoor Air Delivery Monitoring

The facility has no system that permanently monitors the ventilation system, so this credit cannot be obtained at this time.

✓ EQ Credit 2: Increased Ventilation

The mechanical system in NBRRH has been designed to provide over 30% additional ventilation air above that require by ASHRAE Standard 62.1, as seen in Table 21 below. Therefore, this credit can be obtained.

Table 21: Increased Ventilation Compliance

Unit	Required OA	130% Required OA	Designed OA
RTU-1	3730 CFM	4850 CFM	6850 CFM
RTU-2	1327 CFM	1725 CFM	2015 CFM
RTU-3	2314 CFM	3010 CFM	4550 CFM

✘ EQ Credit 3.1: Construction IAQ Management Plan: During Construction

An Indoor Air Quality Management plan was not implemented during the construction phase of the NBRRH project, and thus this credit cannot be obtained.

✓ EQ Credit 3.2: Construction IAQ Management Plan: Before Occupancy

An Indoor Air Quality Management plan was implemented in the pre-occupancy phase of the building project. The mechanical contractor performed baseline IAQ testing that demonstrated contaminant levels below the prescribed values in this credit description. These tests were performed to the extent of the requirements outlined in the LEED checklist, so this credit can be acquired.

✓ EQ Credit 4.1: Low-Emitting Materials: Adhesives and Sealants

All water-based sealants and adhesives used in the mechanical system are specified to have a maximum VOC content of 75 g/L (less water) according to Specification section 233113 – Metal Ducts. This is below the maximum VOC content prescribed by this credit, so Credit 4.1 can be achieved.

✘ **EQ Credit 4.2: Low-Emitting Materials: Paints and Coatings**

✘ **EQ Credit 4.3: Low-Emitting Materials: Carpet Systems**

✘ **EQ Credit 4.4: Low-Emitting Materials: Composite Wood and Agrifiber Products**

The emittance properties of the architectural coatings within the facility are unknown, so it is impossible to determine if the facility would achieve Credits 4.2 through 4.4. For the purpose of this hypothetical assessment, it is assumed that these credits would not be earned.

✓ **EQ Credit 5: Indoor Chemical and Pollutant Source Control**

Permanent entrances to the facility each have vestibules that are at least 10 feet long in the direction of travel, which exceeds the minimum requirement of 6 feet prescribed by Credit 5's requirements. Additionally, all areas with potentially hazardous gases and chemicals (including soiled linen rooms, the laundry room, and medical gas storage rooms) are mechanically exhausted per the minimum requirements of this credit. Filtration of all supply air occurs through both a MERV 14 and MERV 7 filter, which exceeds the minimum MERV 13 requirement.

Because the three minimum requirements of this credit are achieved, Credit 5 can be obtained.

✘ **EQ Credit 6.1: Controllability of Systems: Lighting**

Because of the critical function of several spaces within the facility, individual lighting controls are provided for less than 90% of the facility, which is the minimum requirement to earn this credit. With the current lighting control strategy and the critical functions occurring within the spaces, this credit cannot be obtained.

✘ **EQ Credit 6.2: Controllability of Systems: Thermal Comfort**

As an acute-care hospital, it is important that many spaces be maintained at a relatively constant temperature and humidity level. For this reason, individual comfort controls are provided for less than 50% of the spaces in the facility, so this credit cannot be obtained with the current control structure.

✓ **EQ Credit 7.1: Thermal Comfort: Design**

Credit 7.1 requires that the building's mechanical system provide a comfortable thermal environment by following minimum requirements of ASHRAE Standard 55-2004: Thermal Comfort Conditions for Human Occupancy. Complete compliance with this standard has been achieved, as the mechanical engineer designed the system to comply with this standard. Thus, Credit 7.1 can be obtained.

✘ **EQ Credit 7.2: Thermal Comfort: Verification**

No thermal comfort survey is planned for the building occupants, and thus Credit 7.2 will not be obtained.

✘ **EQ Credit 8.1: Daylight and Views: Daylight 75% of Spaces**

✘ **EQ Credit 8.2: Daylight and Views: Views for 90% of Spaces**

While all of the patient rooms are day-lit, many of the exam, therapy, and procedure rooms contain private or critical functions that are not exposed to daylight. Less than 75% of the spaces in the facility are exposed to daylight and less than 90% have a line of sight to the outdoor environment. Therefore, this credit has not been obtained.

(11.3) LEED Conclusions

A possible combined 32 points is available from both the Energy and Atmosphere (EA) and Indoor Environmental Quality (EQ) categories of the LEED Version 2.2 rating system. Through this analysis, it was realized that all 17 points available in the EA category could not be obtained due to not meeting the minimum energy performance prerequisite. If this had been met, NBRRH would have received 2 points from this category. All of the prerequisites were met in the EQ category and 5 points were obtained through credits in this grouping.

The facility did not strive for LEED Certification, but should the owner decide to renovate the mechanical system, 7 of the 32 points for these two categories have already been achieved. It is important to note, however, that certain changes may require revisiting of previously achieved credits, such as those involving refrigerants.

(12.0) Existing Mechanical System Evaluation

The complete mechanical system currently being used by the New Braunfels Regional Rehabilitation Hospital adequately meets all space heating, cooling, and ventilation requirements as well as maintains proper indoor air quality and relative humidity required for a medical facility. Many choices made in the system design were done so with an accelerated schedule and strict budget in mind.

So, while the system design currently performs the required functions for the building occupant, there are several system design changes that can be made to increase energy efficiency and overall system reliability. One such option that should be explored is the installation of a chilled water system. While this would greatly increase the first cost of the mechanical system, it could significantly decrease the electrical consumption used to cool the building and thus save the owner on yearly operating costs. However, with such a small amount of mechanical space, this could have repercussions on other important areas in the building.

Improvements could be made to existing air distribution and hydronic systems, as well. There exist many new technologies in airside operations, including chilled beams or radiant panels in non-critical spaces. More expensive condensing boilers could potentially save energy to reheat supply air, and different pumping arrangements could be explored.

Additionally, several renewable energy sources could be explored for this particular site. Solar water heating or photovoltaic cells could be utilized in the sunny Texas climate. The tract of land on which the facility sits is large compared to the building footprint, so a geothermal system could also be a viable option.

With a very basic system, such as exists currently in NBRRH, there are many possible alternatives. Those that are deemed to be most effective for the building's loads and functions as well as the geographic location will be discussed further in Part II of this report.

PART II: REDESIGN PROPOSAL

(13.0) Alternatives Considered

Several components of the mechanical system could be redesigned to minimize operating cost and energy consumption or to improve the reliability and controllability of the systems. A list of redesign options that were considered are shown in the list below.

- Introducing a chilled water system
- Converting the existing rooftop units to be water-cooled
- Converting to energy-efficient condensing boilers
- Using several multi-split units with variable refrigerant flow (VRF)
- Installing radiant panels or chilled beams in non-critical spaces
- Lowering the ceiling in patient rooms to reduce ventilation requirements
- Using a solar thermal system for domestic hot water heating
- Improving the building envelope that encloses the therapy pool

Many of these alternatives will be investigated further for this thesis. Some system changes will only be made to certain parts of the facility while other changes affect the entire building system. The alternatives that will be explored were chosen with practicality, integration to other systems, and sustainability in mind.

(14.0) Proposed Redesign

The following alternatives were considered the most appropriate for the New Braunfels Regional Rehabilitation Hospital based on building function and geographic location. Several areas of the redesign were also chosen for the educational benefit of learning about these systems in depth.

The following alternative suggestions do not imply in any way that there is anything wrong with the existing systems. These are merely to investigate potential operating cost or energy savings that could be realized with different systems.

(14.1) Central Plant Redesign

The first depth study will focus on a redesign of the existing central plant. Specifically, an investigation into the feasibility of a chilled water system will be performed using Trane Trace. Various equipment types and piping arrangements will be explored if a chilled water plant is deemed feasible. Also, the effect converting from the existing gas-fired boilers to high-efficiency condensing boilers will be explored.

(14.2) Multi-Split Systems with VRF

Multi-split systems utilize one external condensing unit or heat pump connected to several indoor terminal units. When these systems use variable refrigerant flow in a heat recovery operation, the system becomes very versatile by allowing the individual indoor units to heat or cool independently of one another. These systems tend to be good options for buildings with small spaces that have varying requirements.

Converting the patient rooms and office spaces to be served by multi-split VRF systems will increase the controllability of each individual space. The reliability of space pressurization is also improved with a constant air volume system in these areas. There is also a potential for energy savings associated with switching to a multi-split system in these areas of the facility, including eliminating gas-fired heating and eliminating zone-level reheat coils. Downsizing of the existing rooftop units will also be explored with this option.

Downsizing the rooftop air-handling units and introducing several rooftop condensing units will change the loads on the roof and possibly have acoustical implications due to potential vibration of the new units.

(14.3) Solar Thermal System

Harnessing solar energy to primarily offset the domestic water heating load, which can be relatively high for a medical facility, will be investigated through the installation of a rooftop solar thermal system. The possibility of incorporation of vertical solar thermal collectors with the envelope of the therapy pool area will also be considered. The secondary function of the solar thermal system will be heating the therapy pool.

Several different configurations exist for solar thermal systems, but flat-plate, roof-mounted collectors will be investigated for this thesis. Different solar collector arrangements and geometries will be examined based on calculated solar gains. Temperature can be modulated in various ways as well, including having an anti-freeze based system or eliminating the anti-freeze and installing a drainback system. All of these options will be studied for this alternative design.

(15.0) Breadth Studies

(15.1) Acoustical Breadth

The changes made to all of the rooftop air handling equipment could potentially have significant implications to the architectural acoustics of the building. Because the system supplying air to the patient rooms is being decentralized and new diffusers and refrigerant piping will be added, there will likely be altered sound propagation into the patient rooms.

An acoustical analysis will determine whether this change in air-borne noise will be significant, based on room noise criteria (NC) levels and will investigate potential ways to counteract the increase in room noise. These measures could include changing wall construction or diffuser location to either minimize sound transmission or flanking noise.

(15.2) Structural Breadth

The structural system of the roof will be analyzed with the new loads due to the additional mechanical equipment and solar thermal collectors. If necessary, the roof's structural framing members will be redesigned to meet these increased loads. The structural investigation will need to be completed prior to the life-cycle cost analyses of systems, because the change in cost of structural members could have an impact on the payback period of the proposed alternatives.

(16.0) Tools for Analysis

Trane Trace 700

To this point, Trane Trace 700 has been used to perform analyses, including economic and energy studies, of the existing mechanical systems, so it was also used to analyze the proposed alternatives.

Microsoft Excel

This program was used to more easily model the solar thermal system based on the procedures that follow in Section III of this report. This was used in place of other available solar thermal system modeling programs because of the potential to customize and find errors in various calculations.

Excel was also used as a tool to aid in the simple payback and lifecycle cost analyses. The energy use data from the Trace simulations was used in conjunction with energy rates and discount rates in an excel file to determine net present values, a built in function of Excel.

Engineering Equation Solver (EES)

In order to solve complex equations for analysis, Engineering Equation Solver was utilized. EES has built-in functions for many properties or equations that relate to HVAC system design and solar thermal system design.

PART III: PROPOSED DESIGN ANALYSIS

(17.0) Depth Study 1: Central Plant Analysis

The existing system utilizes only air-cooled, direct expansion cooling and gas-fired furnaces for heating, so an investigation into implementing a more efficient central plant was performed. Included in this central plant analysis is an investigation into using high efficiency condensing boilers and implementing a completely new chilled water system. These systems would be connected to the existing air-side equipment including rooftop units, kitchen makeup air unit, and terminal reheat boxes.

Because of the modest size of the facility, the hypothesis was that it would be uneconomical to implement a more expensive chilled water system, but the analysis was performed to confirm this hypothesis.

(17.1) Condensing Boiler Investigation

Condensing boilers are a good alternative to be considered in any application because they recover waste heat from the initial heating process and provide significant increases in overall efficiency. Condensing boilers can be used in operation with low return water temperatures that exist in a reheat system such as that in NBRRH. The zone-level reheat in the existing system makes an increased efficiency on the plant-side of the heating system a particularly desirable improvement.

(17.1.1) System Operation and Design

The major difference between condensing boilers and the regular gas-fired boilers that currently exist in the facility is the recovery of waste heat. While non-condensing boilers exhaust all waste heat produced in combustion, condensing boilers condense water vapor produced in the combustion process into liquid water, recovering its latent heat.

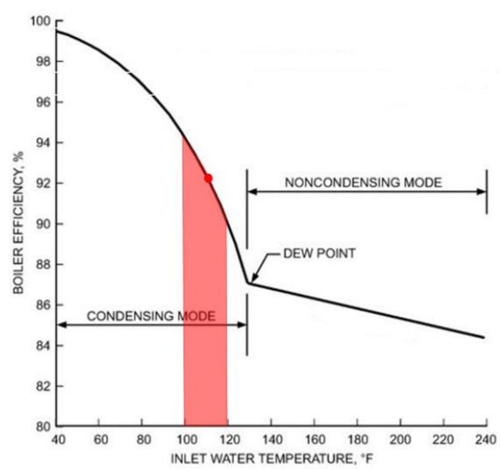

Figure 16: Condensing Boiler Efficiency

Thermal efficiency within condensing boilers increases as the inlet water temperature decreases. Though the inlet water temperature of the current boilers varies with space heating load, it typically falls between 100ºF and 120ºF. This gives an average boiler efficiency of about 92.5%, as shown above on Figure 16, which is taken from Chapter 31: Boilers of the 2008 ASHRAE Systems and

Equipment Handbook. This is about a 7.5% increase in efficiency from the existing non-condensing boilers, which would provide energy savings on the magnitude of about 80 MBh of input energy, as described in the next section of this report.

The feasibility of a 7.5% increase in boiler efficiency as shown is discussed in the energy and economic evaluations that follow.

(17.1.2) Energy Evaluation

The increased efficiency and lower required energy input will result in total energy savings for NBRRH. The Trane Trace model used to calculate the energy use of the existing system was also used to determine the savings in natural gas consumption associated with switching to condensing boilers. The monthly natural gas consumption for each boiler configuration is shown in Figure 17 below.

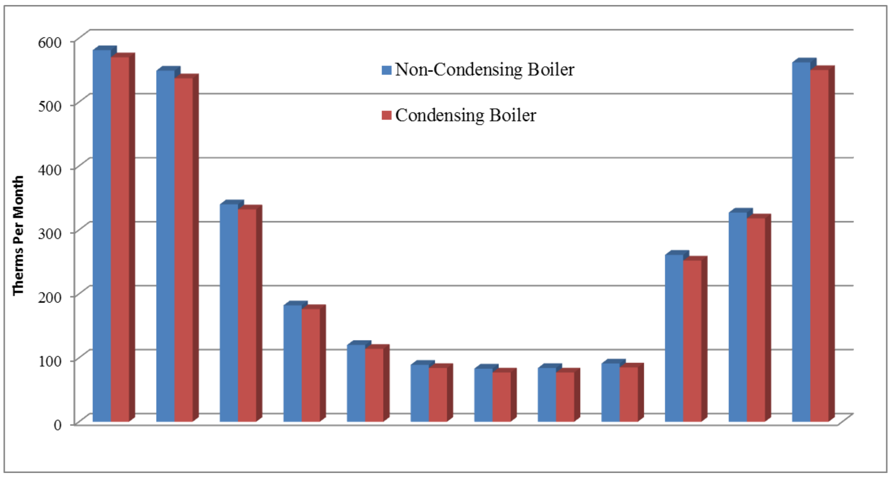

Figure 17: Boiler Gas Consumption Comparison

Though the condensing boiler does decrease total natural gas consumption throughout the year, it is only about 97 therms per year, which is not a significant amount. During the summer, when the mechanical system is in cooling operation, existing pool dehumidification unit and makeup air unit are not returning low enough water temperatures to see the increase in efficiency common to condensing boilers.

(17.1.3) Economic Evaluation

Though energy can be saved through increased boiler efficiency, there is also an increase in first cost of condensing boilers from non-condensing types. This is because condensing boilers are typically made of more expensive materials like stainless steel or aluminum to prevent against corrosion caused by acidic condensate. Non-condensing boilers, on the other hand, are typically made with cast iron or steel sections. Table 22 on the following page shows the tradeoff between increased efficiency and increased equipment first cost.

Table 22: Boiler Cost and Efficiency Comparison

Equipment	Efficiency	Input MBh	Output MBh	First Cost	Total System First Cost
Existing Boilers	85.0%	999	849	$30,624	$1,400,000
Condensing Boilers	92.5%	918	849	$39,634	$1,409,010

The economic viability of switching to condensing boilers was determined through a simple payback and lifecycle cost analysis, a summary of which is provided in Table I2 of Appendix I of this report.

Because the condensing boilers only save 97 therms per year, this results in less than $100 worth of savings per year. Taking first cost difference, discount rate, maintenance costs, and slightly lowered natural gas use into account, the simple payback period of using condensing boilers rather than non-condensing is just over 97 years, which is not reasonable justification for the installation of a condensing boiler. The condensing boiler system has a higher net present value than the existing system over a life-cycle cost period of 20 years, which means that there is no life-cycle payback.

(17.2) Chilled Water System Investigation

The existing air-cooled, direct expansion units have a considerably low first cost, but the energy required to reject the heat in cooling operation is significantly higher than most traditional chilled water systems. For this reason, a chilled water system investigation was necessary to determine whether it is a viable system alternative. Air-cooled and water-cooled chilled water systems were both investigated.

(17.2.1) System Operation and Design

Air-Cooled System

In an air-cooled chilled water system, chilled water is produced by chillers arranged in parallel and, in this designed system, transported to the loads at the rooftop units through a primary-secondary pumping system. Heat is rejected at the chiller, which means there is no condenser water loop and no mechanical room requirement for the chillers as they will need to be placed outside. Air-cooled systems typically cannot offer as much of a performance increase as water-cooled, but the first cost of an air-cooled system is considerably lower. A schematic of the chilled water loop for both air-cooled and water-cooled is shown below in Figure 18.

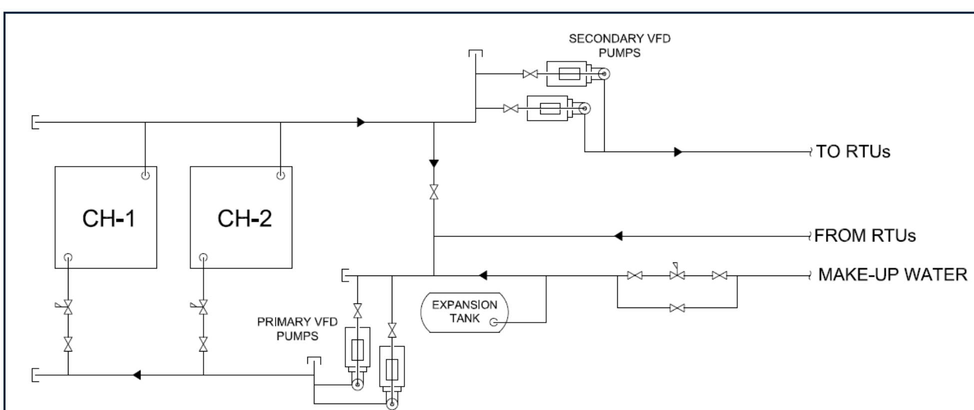

Figure 18: Chilled Water Schematic

Water-Cooled System

A water-cooled chilled water system operates similarly to an air-cooled system, but the difference lies in the method of heat rejection. In a typical water-cooled system, chilled water exchanges heat with condenser water which then travels to a cooling tower and rejects heat to the atmosphere before returning to the chiller. A condenser water loop schematic is shown in Figure 19 below. While there is an increase in performance associated with water-cooled chilled water systems, there is also a considerable increase in first-cost and space requirements for indoor chillers, condenser water distribution equipment, and an outdoor cooling tower.

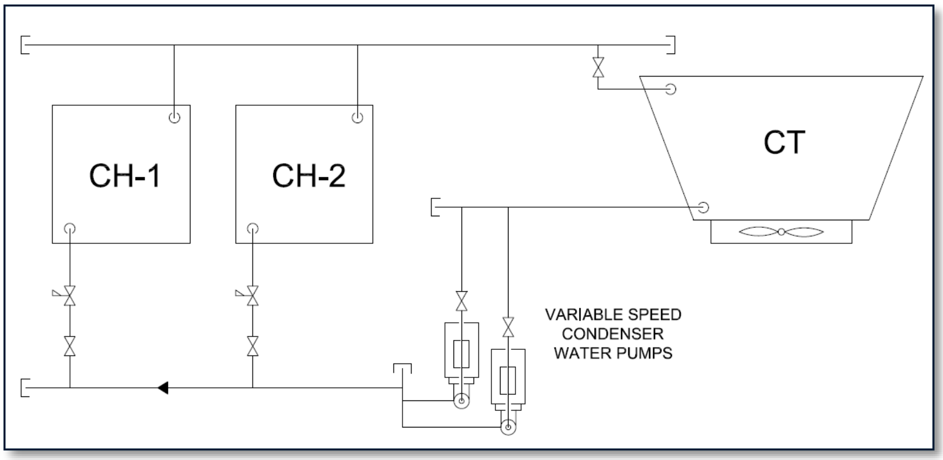

Figure 19: Condenser Water Loop Schematic

(17.2.2) Energy Evaluation

There are considerable energy savings associated with a water-cooled chilled water system, mainly in the compression segment of the refrigerant cooling cycle due to the more efficient heat rejection. Additionally, there are less significant savings in the fan energy required for heat rejection at the cooling tower level. The distribution of equipment energy usage in the three considered systems is shown below in Figure 20.

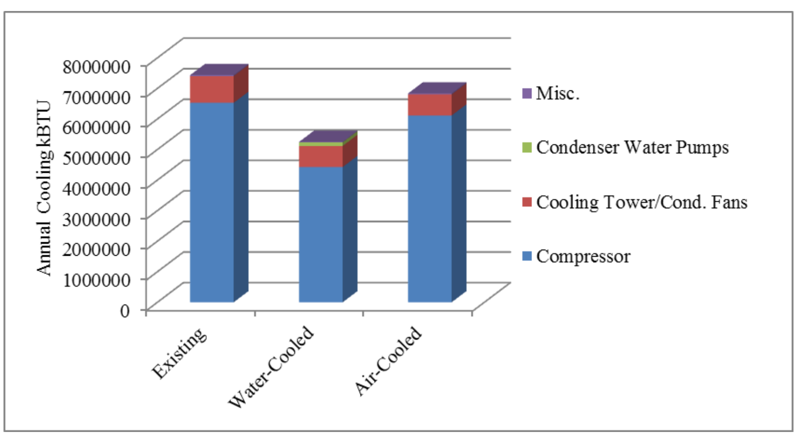

Figure 20: Cooling Equipment Energy Consumption Comparison

The annual savings shown in Figure 20 prove that the water-cooled chilled water system is more efficient than both air-cooled options explored, which includes the existing air-cooled direct expansion units. These savings are most significant during the summer months, when cooling operation is at a maximum, but the savings are barely noticeable in the winter months, when cooling operation is scarce. Figure 21 below shows the monthly distribution of electricity consumption throughout a typical year.

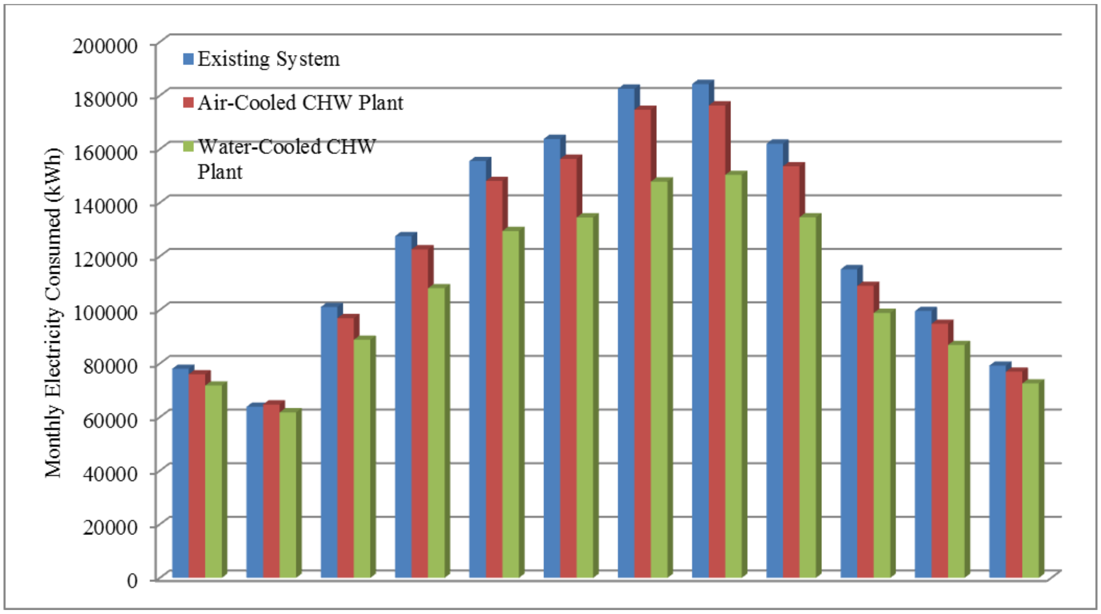

Figure 21: Chilled Water Electricity Consumption Comparison

While there are obvious energy-saving benefits associated with the use of a chilled water plant over the existing direct expansion system, the added first cost of a chilled water system needs to be considered to determine the practicality of such a switch.

(17.2.3) Economic Evaluation

The savings associated with a lower energy use are offset by the increased first cost of chilled water system equipment, including chillers, chilled water distribution equipment, heat rejection equipment, and condenser water distribution equipment. A summary of the total system first cost of the existing system and two alternatives considered are shown in Table 23 below. The existing system cost was acquired through the mechanical contractor and a breakdown is shown in Table 16 of Section 8.0 of this report. The cost of equipment in considered alternatives was acquired through 2011 RS Means Mechanical Cost Data, a breakdown of which is shown in Appendix E.

Table 23: Chilled Water System Cost Comparison

System	Annual kWh Consumed	CHW Equipment First Cost	Total System First Cost
Existing System	1512650	$0	$1,400,000
Air-Cooled CHW Plant	1449986	$414,158	$1,814,158
Water-Cooled CHW Plant	1285496	$470,498	$1,870,498

The simple payback period of an air-cooled chilled water system, as designed, was determined to be 153.7 years, while the simple payback for a water-cooled system was 48.17 years. Additionally, each system had a significantly higher net present value than the existing system, which means the systems would not pay back over a 20-year life cycle. A summary of each system's simple payback period and life-cycle cost analyses are provided in Tables I3-I4 of Appendix I.

(17.3) Overall Evaluation

Despite the annual energy savings, a chilled water plant is not recommended for the New Braunfels Regional Rehabilitation Hospital based on net present values and payback period estimates. A negative net present value means that the investment of a chilled water system will never benefit the owner monetarily, and the energy savings are not large enough to justify the loss of capital for the investment of a chilled water system.

(18.0) Depth Study 2: Variable Refrigerant Flow System

(18.1) System Operation

Multi-split variable refrigerant flow (VRF) systems utilize one outdoor condensing unit connected to multiple indoor evaporating units. Indoor units can be ductless or may be attached to ducts which deliver the required amount of ventilation air to each zone. The ducted options will be used in this alternative to maintain occupant comfort and safety through appropriate ventilation. The amount of R-410A refrigerant sent to each indoor unit is modulated based on user input at the zone level, which allows each zone to be heated or cooled to the occupant's specifications. The flow of refrigerant is modulated through an inverter-driven scroll compressor in the outdoor unit. This increased controllability makes a VRF system ideal for areas such as patient rooms or offices, where it is desirable for the occupant to have temperature control.

VRF systems in heat recovery operation allow for simultaneous heating and cooling of different zones connected to the same outdoor condensing unit. This is achieved through a three-pipe system design. In heating operation, high-pressure hot gas is delivered to the indoor unit and returns low temperature refrigerant to the outdoor unit. In cooling operation, the low temperature line delivers high pressure cold refrigerant to the indoor evaporating unit and a third suction line returns high temperature refrigerant to the compressor or delivers it to other condensing units for heat recovery. This heat recovery is useful when spaces with varying heating and cooling loads are connected to the same outdoor unit.

Ventilation and Humidity Control

Because VRF systems are designed to handle 100% of the heating and cooling load of the zones, the only airflow necessary is that which is required for adequate ventilation and to control the relative humidity of each zone. A dedicated outdoor air handling unit is used to meet ventilation and humidity control requirements by sending the ventilation air at proper humidity to the zone-level air-to-refrigerant heat exchangers.

Heating or Cooling Operation Only

When all zones are in cooling operation, the VRF delivers varying volumes of air-cooled refrigerant to each zone based on occupant input. Modulation of refrigerant is controlled through electronic expansion valves within each indoor unit. This is shown for a typical system with 8 connected indoor units in Figure 22 below.

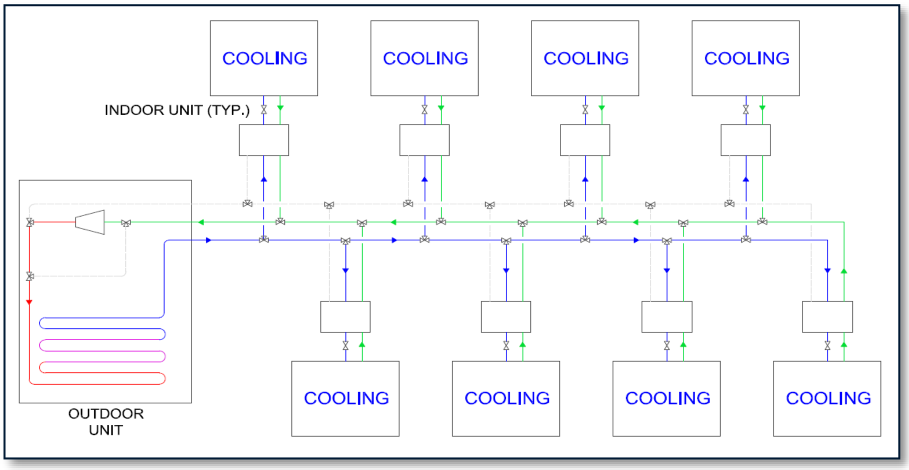

Figure 22: VRF Schematic: Cooling Operation Only

Heating Operation Only

When the system is in heating only operation, the VRF system operates as a heat pump by reversing the flow of refrigerant through the outdoor unit and having the indoor coil now act as the evaporator, as shown in Figure 23 below.

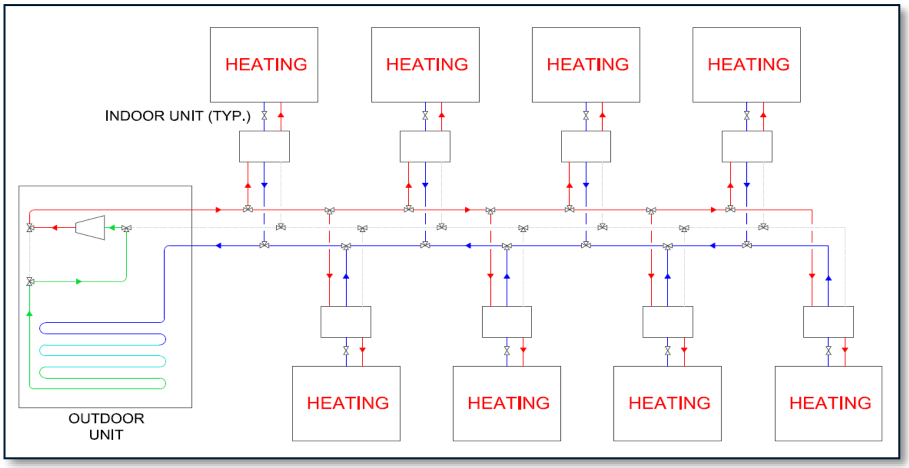

Figure 23: VRF Schematic: Heating Operation Only

Simultaneous Heating and Cooling

If separate zones requiring heating and cooling are connected to the same outdoor condensing unit, heat recovery operation can be employed to lower energy use. In primarily cooling operation, high temperature refrigerant returned from indoor units in cooling operation is sent directly to the compressor and then delivered to the zones requiring heating, as shown on the next page in Figure 24. This eliminates the need for auxiliary heating sources when in cooling mode.

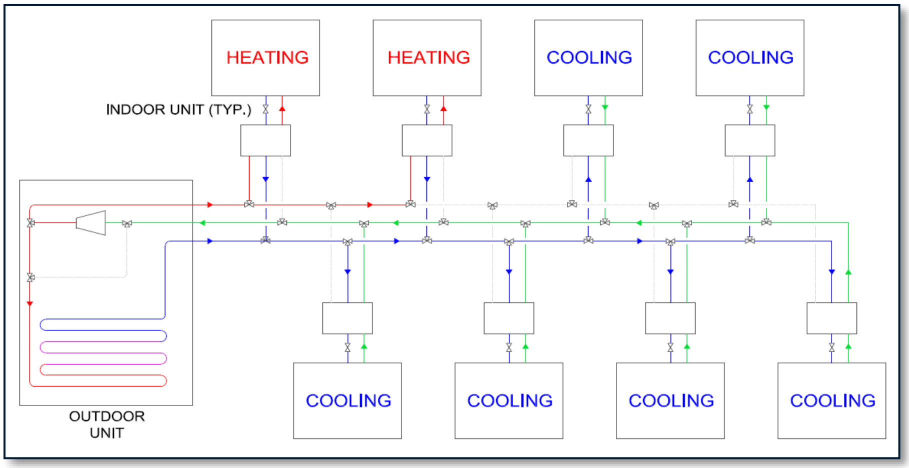

Figure 24: VRF Schematic: Mostly Cooling Operation

In primarily heating operation, low temperature refrigerant returned from indoor units in heating operation is sent directly to indoor units in cooling operation, as shown in Figure 25 below. This eliminates the need for auxiliary cooling energy when in heating mode.

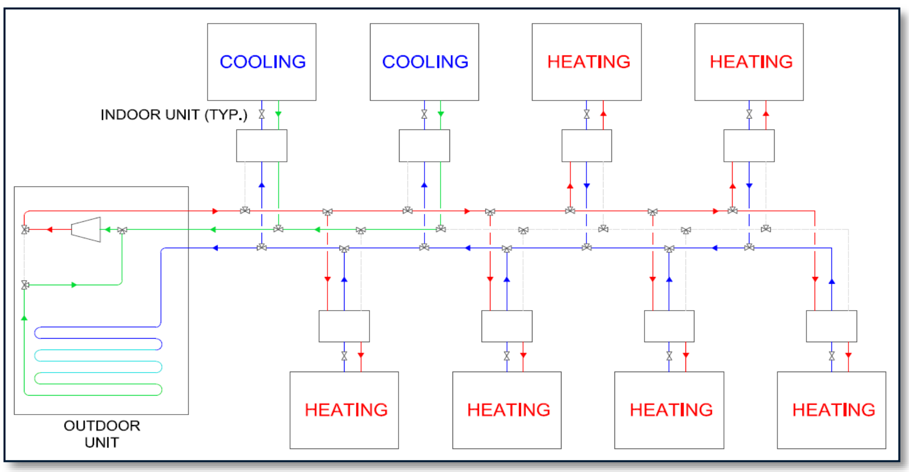

Figure 25: VRF Schematic: Mainly Heating Operation

The ideal situation for a VRF system in heat recovery operation is the exact same heating and cooling load connected to one outdoor unit. While this will likely never happen for a considerable period of time, it is worthwhile to note that periods of operation close to this condition will result in significant energy savings. In this scenario, shown in Figure 26 on the next page, the only energy required by the hydronic system is pumping energy and energy used to operate the inverter-driven scroll compressor in the outdoor unit.

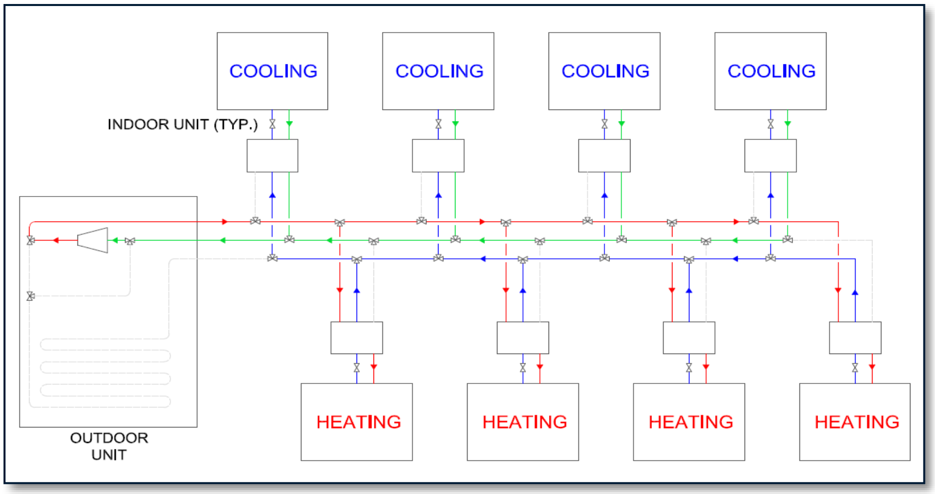

Figure 26: VRF Schematic: Complete Heat Recovery

(18.2) VRF System Zoning

Choosing which zones to interconnect in a heat recovery VRF system is important because the ideal operating condition is complete heat recovery. Because the system is a split system transporting high pressure refrigerant, the total length of piping for each system needs to be minimized per manufacturer specifications. For these reasons, patient rooms were divided into four systems, each corresponding to one branch of the patient room wing. These systems all condition patient rooms facing opposite directions, which will increase the variability of heating and cooling loads due to solar gain for each system. The offices and exam rooms to be conditioned by the VRF systems were split into two systems based on location in the building and the maximum refrigerant piping length. The room assignments to VRF systems are shown in Figure 27 below.

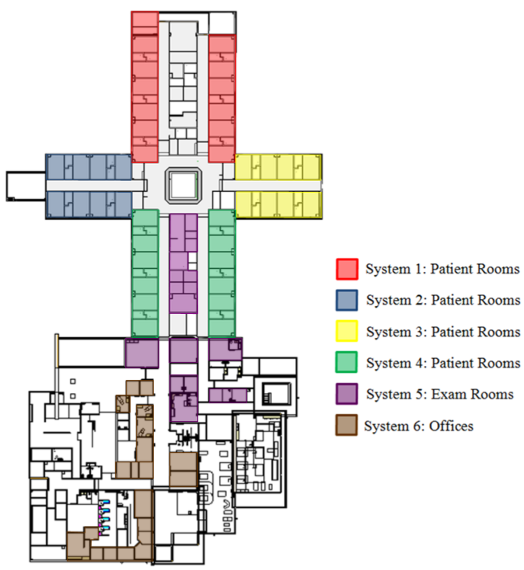

Figure 27: VRF Zoning Diagram

(18.3) System Design

The Trane Trace model used to calculate the original design loads was altered to reflect the new zoning for the VRF system and a new load calculation was performed. The total space still served by the VAV system (shown in white in Figure 27) has a significantly reduced heating and cooling load, which allows for the removal of the largest rooftop air handling unit, RTU-1. The rooms originally served by RTU-1 and not served by the VRF system can now be served by RTUs 2 and 3 without increasing the capacity of those units.

A 5,000 CFM dedicated outdoor air unit would need to be installed in place of RTU-1 to supply the zones served by the VRF system with the appropriate ventilation air. This constant volume unit would also pre-condition the outside air to the necessary relative humidity before delivering it to the packaged indoor evaporators connected to the VRF system.

These indoor units are available in a variety of styles and sizes, so a typical unit was selected from a popular manufacturer for the purpose of cost analysis and a room acoustics analysis discussed in Section 21 of this report. In order to determine the viability of this system, a detailed computational fluid dynamics analysis would need to be performed to prove that room air distribution is appropriate for a medical room. The cut sheets for these indoor units are provided in Appendix F.

The outdoor condensing units were sized based on the maximum heating and cooling required for all zones connected to that unit. Heat recovery between zones was modeled in the Trace model, which reduces the total yearly energy used by each zone. The design cooling and heating capacities of the new outdoor units are shown in Table 24 below and cut sheets for each size of outdoor unit are also shown in Appendix F.

Table 24: VRF Condensing Unit Schedule

Tag	Design Cooling Load (Tons)	Design Heating Load (MBh)	Unit Cooling Capacity (Tons)	Unit Heating Capacity (MBh)	Refrigerant	Input Power (kW)	Weight (lbs)
CU-1	10.8	54.0	12.0	162.0	410-A	10.8	1146
CU-2	6.7	35.1	8.0	108.0	410-A	8.6	573
CU-3	6.8	35.1	8.0	108.0	410-A	8.6	573
CU-4	10.3	51.1	12.0	162.0	410-A	10.8	1146
CU-5	7.5	33.0	10.0	135.0	410-A	10.9	573
CU-6	7.9	30.0	10.0	135.0	410-A	10.9	573

(18.4) VRF System Evaluation

(18.4.1) Energy Evaluation

One of the main advantages to VRF systems, other than increased occupant controllability, is the energy savings associated with lower airflows (less fan energy required) and heat recovery between zones (less heating or cooling energy required). In NBRRH, the cooling operation dominates in the warm Texas climate, so all of the heating and cooling savings occur in cooling operation. When the system is in complete heating operation, ventilation air must be preheated as well, so the system is actually less efficient than the reheat VAV boxes served by the efficient gas-fired boilers; thus, there is a small increase in heating energy from switching to VRF, as shown in Table 25 on the following page.

Table 25: VRF Energy Savings

System	Existing System Zone		VRF System	
	Cooling (Ton)	Heating (MBh)	Cooling (Ton)	Heating (MBh)
VRF-1	11.6	28.6	10.8	54.0
VRF-2	7.1	17.1	6.7	35.1
VRF-3	7.2	17.1	6.8	35.1
VRF-4	11.0	27.3	10.3	51.1
VRF-5	8.1	13.5	7.5	33.0
VRF-6	8.4	20.9	7.9	30.0

Although there is a small increase in heating energy required, the overall yearly energy required decreases with the introduction of the six VRF systems. A comparison of total equipment energy consumption between the existing system and the VRF system, shown in Figure 28 below, gives relative scale to the domination of cooling operation. The decrease in required fan energy accounts for savings in auxiliary energy, while heat recovery and the principles of operation of the VRF system account for the savings in cooling energy shown in the comparison.

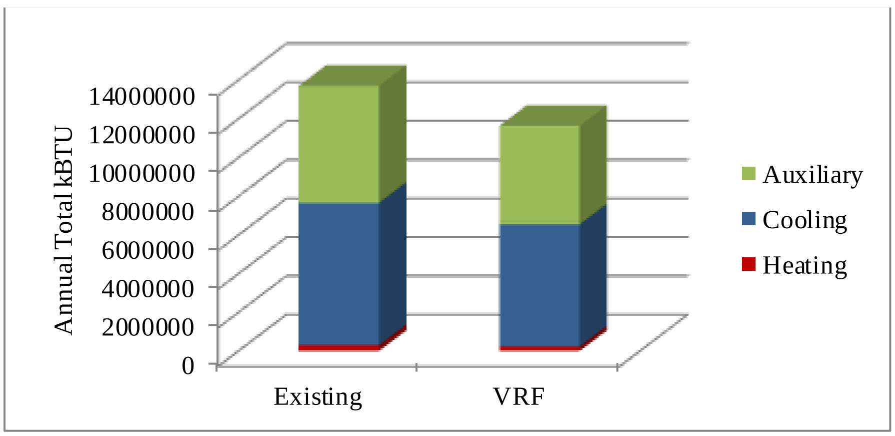

Figure 28: VRF Equipment Energy Consumption Comparison

The benefit of the VRF system is obviously in cooling operation, which is further supported by the annual profiles of electricity consumed, shown in Figure 29 on the next page. The majority of the savings occur in summer months when the existing system would normally be serving the VRF zones by air-cooled, direct expansion rooftop units. In winter months, the electricity savings can be attributed to the decrease in fan energy required to operate a VRF system.

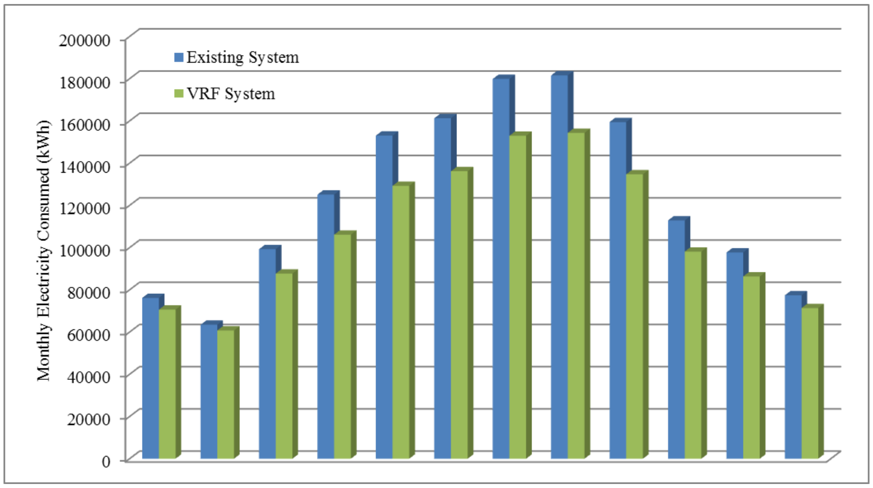

Figure 29: Annual Electricity Consumption Comparison

(18.4.2) Economic Evaluation

Compared to the complete central plant redesign investigated in the first mechanical depth study, the increase in first cost of a VRF system is fairly manageable. Added first costs associated with a VRF system include the actual equipment (outdoor condensing units and indoor evaporating units), refrigerant piping and distribution equipment, additional DOAS unit and ductwork, and the actual R-410A refrigerant itself. The elimination of RTU-1 helps to alleviate the impact of the added first cost of the other equipment. A summary of the first cost breakdown is shown in Table 26 below. The result is approximately a 4% increase in first cost from the original system.

Table 26: VRF System First Cost

Component	Material	Labor	Total
RTUs	$117,800	$7,089	$124,889
Air Distribution Equipment	$114,480	$18,000	$132,480
Ductwork and Insulation	$220,300	$239,400	$459,700
MAU System	$91,797	$20,000	$111,797
Pool Dehumidification Unit	$20,757	$5,000	$25,757
Boilers and Control Interface	$23,624	$7,000	$30,624
Hydronic Distribution Equipment	$4,450	$7,000	$11,450
Mechanical Piping	$49,260	$103,900	$153,160
DDC Controls	$44,700	$105,200	$149,900
Outdoor VRF Condensing Units	$19,340	$4,700	$24,040
Indoor VRF Evaporating Units	$18,696	$10,944	$29,640
VRF Piping and Distribution	$24,630	$51,950	$76,580
R-410A Refrigerant	$6,600	-	$6,600
DOAS AHU	$22,250	$1,475	$23,725
Added First Cost	-	-	$100,000
			$1,460,342
		Structural Savings:	**-$2,900**
			$1,457,442

The energy savings, combined with a conservative assumed increased maintenance cost of $0.05 per square foot per year, yields a yearly savings of $9,906 per year for NBRRH with a VRF system when compared to the existing system. This results in a simple payback period of about 5.8 years.

The VRF system will also have a positive net present value of approximately $66,800 more than the existing system when analyzed in a 20-year life-cycle cost analysis. A summary of the economic analysis is provided in Table I5 of Appendix I. The economic analysis of the system is reasonable justification to consider a VRF system for the New Braunfels Regional Rehabilitation Hospital following a more in-depth ventilation and air quality analysis of patient rooms.

(19.0) Depth Study 3: Solar Thermal System

The introduction of a solar thermal system was investigated for the primary purpose of heating domestic hot water, which has a relatively constant demand profile throughout a typical day and during all times of the year. Secondary functions of the solar thermal system are therapy pool water heating and space heating when there is demand.

(19.1) Design Load Estimation

Primary Load: Domestic Hot Water Heating

The hourly domestic hot water load for the facility was estimated using values in Table 10 of the 2011 ASHRAE HVAC Applications Handbook, which shows hot water demand per fixture and demand factor adjustments for various types of buildings. The building was broken into three wings based on probable hot-water use schedule. The patient room wing is assumed to have use of hot water at all hours of the day, while the therapy/exam wing is assumed to be in use from 6:00 AM – 10:00 PM and the kitchen wing from 6:00 AM – 6:00 PM. The calculations of hot water demand for each wing are shown in Tables 27-29 below.

Tables 27-29: Domestic Hot Water Demand Loads

Patient Room Wing			
Fixture	#	Gallons/hour (per fixture)	Total gph
Sink, Private Lav	83	2	166
Sink, Public Lav	1	6	6
Dishwasher	0	150	0
Sink, Kitchen	2	20	40
Service Sink	11	20	220
Bathtub	1	20	20
		Total:	452
		x Demand Factor (0.25):	**113**

Therapy Wing			
Fixture	#	Gallons/hour (per fixture)	Total gph
Sink, Private Lav	6	2	12
Sink, Public Lav	5	6	30
Dishwasher	0	150	0
Sink, Kitchen	3	20	60
Service Sink	11	20	220
Bathtub	0	20	0
		Total:	322
		x Demand Factor (0.25):	**80.5**

Kitchen			
Fixture	#	Gallons/hour (per fixture)	Total gph
Sink, Private Lav	0	2	0
Sink, Public Lav	0	6	0
Dishwasher	1	150	150
Sink, Kitchen	8	20	160
Service Sink	0	20	0
Bathtub	0	20	0
		Total:	310
		x Demand Factor (0.25):	**77.5**

The facility is estimated to use about 5,200 gallons of domestic hot water per day. The gallon per hour load was converted to an energy requirement, in MBh, to be combined with the pool water heating load and secondary space heating load. The daily domestic hot water heating load is assumed to be approximately 1,022 MBh, which equates to a DHW heating load of about 373,000 MBh yearly. This energy use was not factored into energy or economic analyses of the other mechanical systems discussed in this report.

Primary Load: Pool Water Heating

The required energy to heat the therapy pool water was estimated by methods described in Chapter 4 of the 2011 ASHRAE HVAC Applications Handbook which has a section on natatoriums. The rate of evaporation of pool water, in pounds of water per hour, is given by Equation 5-2 of the handbook, shown below in Equation 4 of this report:

$$W_p = 0.1A(p_w - p_a)F_a \qquad \textbf{(Equation 4)}$$

where W_p = Evaporation rate of pool water (lb/hr)
A = Area of pool surface (400 ft^2)
p_w = Saturation vapor pressure at surface water temperature (in. Hg)
p_a = Saturation pressure at room air dew point (in. Hg)
F_a = Typical activity factor (0.65 for a therapy pool)

For the 400 ft^2 therapy pool in use in NBRRH, the estimated water evaporation rate is approximately 11.75 pounds of water per hour, or 11.75 MBh daily. This is considered a constant load, as it is required to keep pool water at an occupiable temperature at all hours. This load is primary for the solar thermal system, and is thus added to the domestic hot water heating load for the analysis.

Secondary Load: Space Heating

Because of the varying space heating load during the day and, most importantly, throughout the year, space heating applications were designed to be a secondary load for the solar thermal system. Using the load calculations performed in the Trane Trace energy model, heating demand schedules were acquired for a typical day in each month of the year.

(19.2) System Operation

The operations of the various components of a forced-circulation solar thermal system are described in the sections that follow. A schematic diagram of the whole system, from solar gain to delivery to load, is shown in Figure 30 on the next page. The collectors and storage tank used for this design were based on components found in the Lochinvar product catalog.

Flat-Plate Collectors

High efficiency commercial flat plate collectors were used for this solar thermal system design. The tilted collectors include copper absorber coils welded to an absorber plate inside the collector casing. The casing itself has a 0.15" thick prismatic glass cover for optimal light transmission with heat-resistant, rigid foam insulation backing. A circulating pump moves antifreeze solution through the absorber coils within the collectors. This fluid absorbs the sun's energy and travels to a water-to-water heat exchanger. The antifreeze solution is controlled to only flow when the collector outlet temperature is greater than the inlet temperature.

Water-to-Water Heat Exchanger

The plate and frame water-to-water heat exchanger is then used to transfer the harnessed solar energy from the antifreeze solution to the potable hot water taken to the stratified hot water storage tank. Plate heat exchangers are very efficient and can have an approach of as little as 1°-2°F, so a 90% efficiency heat transfer was assumed and incorporated when calculating the total amount of energy transferred to the storage tank.

Stratified Hot Water Storage Tank

A stratified water storage tank was used for thermal storage to maximize the solar energy available at all times of demand. Stratified tanks utilize the buoyancy differential of water at different temperatures to separate hot and cold water and deliver the maximum temperature water to loads and the minimum temperature water back to the heat exchanger with the flat plate collector loop.

The storage tank available from the manufacturer used also has an indirect water heating system, which allows the water in the storage tank to remain at the necessary temperature to meet the hot water demands at all hours of the day. This makes the elimination of the existing hot water boilers possible, which significantly decreases the first cost of the system.

Delivery to Loads

The solar-heated hot water is first sent to the domestic hot water loads in the building. If there is an excess of stored hot water available, that water is sent to another plate and frame heat exchanger to help with the heating of the indoor therapy pool water. If the solar gain available exceeds both of these loads, the water is sent to the hot water boilers to reduce the space heating load on the building. The control of the heated hot water is determined by supply and return temperature sensors into and out of the mechanical room, where the storage tank and boilers are located.

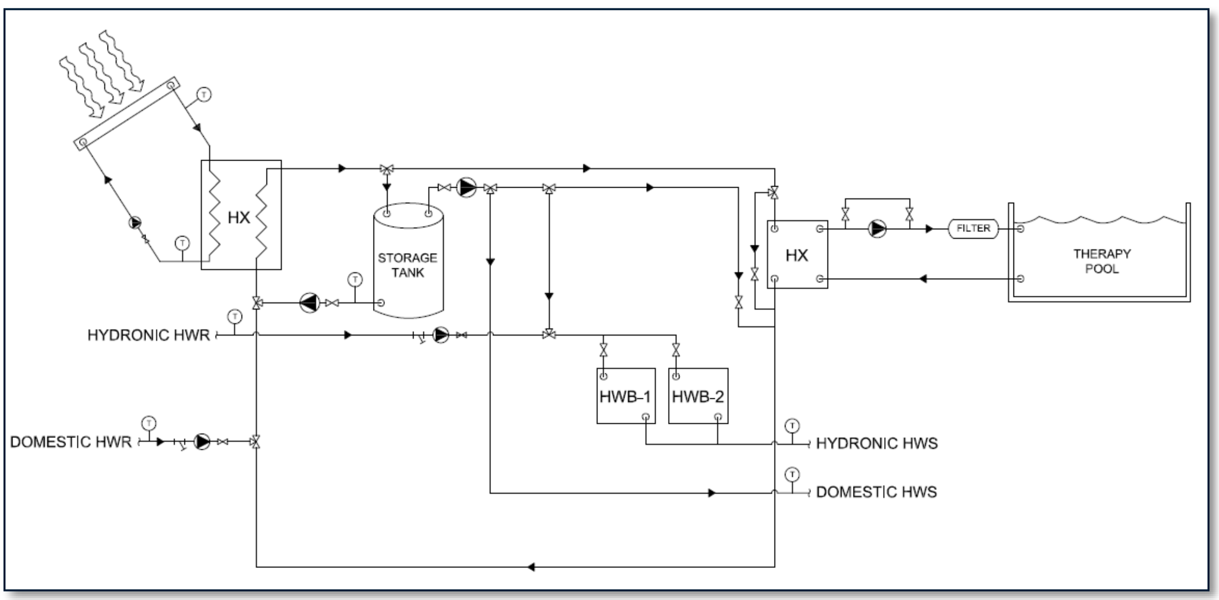

Figure 30: Solar Thermal System Schematic

(19.3) System Design

The design of the solar thermal system was based on methods presented in *Solar Engineering of Thermal Processes* by Duffie et al. All equations used in the following sections have been taken from this text, though the analyses and interpretations presented here are original.

(19.3.1) Solar Radiation Calculations

To accurately estimate how much energy can be absorbed by a solar thermal collector, the geometry of the system, location of the system, time of day, and climate data need to be considered. To begin the analysis, the zenith angle (Θ_z) and angle of incidence (Θ) were calculated in a Microsoft Excel file for every hour over a yearlong period. By creating an Excel program to calculate absorbed solar energy, the effect of different variations in collector tilt and orientation on total absorbed energy was explored easily. The equations used to calculate these angles are shown below in Equations 5 and 6:

$$\cos(\Theta_z) = \cos(\Phi)\cos(\delta)\cos(\omega) + \sin(\Phi)\sin(\delta) \qquad \textbf{(Equation 5)}$$

$$\cos(\Theta) = \sin(\delta)\sin(\Phi)\cos(\beta) - \sin(\delta)\cos(\Phi)\sin(\beta)\cos(\gamma) + \cos(\delta)\cos(\Phi)\cos(\beta)\cos(\omega) + \cos(\delta)\sin(\Phi)\sin(\beta)\cos(\gamma)\cos(\omega) + \cos(\delta)\sin(\beta)\sin(\gamma)\sin(\omega)$$

(Equation 6)

where Θ_z = Zenith angle, or incident angle on a horizontal surface
Θ = Angle of incidence between the sun's ray and the surface normal
Φ = Latitude of New Braunfels, TX (29.69ºN)
δ= Solar declination, or the angular position of the sun at noon (varies with date)
β = Collector tilt angle from horizontal
γ = Surface azimuth angle, or the angle at which the collector is aimed, measured counter-clockwise from true south.
ω = Hour angle, or east-west displacement of the sun from the local meridian

To determine the total extraterrestrial radiation that reaches the collector during a given hour (I_o) in MJ/m^2, the following equation was used:

$$I_o = \frac{12 \cdot 3600 \cdot G_{sc}}{\pi} \cdot \left(1 + 0.033\cos\left(\frac{360n}{365}\right)\right) \cdot \left(\cos(\Phi)\cos(\delta)(sin\omega_2 - sin\omega_1) + \frac{\pi}{180}(\omega_2 - \omega_1)\sin(\Phi)\sin(\delta)\right)$$

(Equation 7)

where I_o = Total extraterrestrial radiation in MJ/m^2
G_{sc} = Solar constant (1367 W/m^2)
n = Julian date (1-365)
ω_1= Hour angle at the beginning of the hour
ω_2 = Hour angle at the end of the hour

Equation 7 above calculates the estimated extraterrestrial radiation for a given hour, assuming a perfectly clear day. In order to more accurately estimate the amount of solar gain available, calculated values and measured radiation values were compared to determine an hourly clearness index, K_T. This clearness index is then used to determine the fraction of diffuse and beam radiation that makes up the total radiation value. Equation 8 below shows the calculation of K_T. For this analysis, TMY3 data for New Braunfels, TX was used to obtain average measured values of I throughout the year.

$$K_T = \frac{I}{I_o} \quad \textbf{(Equation 8)}$$

The Erbs et al. correlation has determined a peace-wise correlation between the hourly clearness index and the ratio of diffuse radiation, I_d, to averaged measured extraterrestrial radiation. Beam radiation, I_b, can then be assumed to be responsible for the rest of the measured radiation striking a horizontal surface. The third component of the total radiation that a collector sees is the ground reflected radiation, which is found simply by multiplying the total measured radiation on a horizontal surface by the ground reflectance value. Because the area around NBRRH is asphalt or cement walkway and the roof is a white roof, the ground reflectance value is assumed to be 50%.

These three components of the total incident radiation on a solar collector will be used in conjunction with collector characteristics to determine the total absorbed solar radiation and the total useful energy harnessed.

(19.3.2) Collector Characteristics

When determining the total amount of incident energy that can be absorbed, several collector properties need to be considered, including loss coefficients, fluid type and flow rate, area of individual panels, transmittance and absorptance values, and tube diameter and spacing.

The single-cover flat-plate collectors used for this analysis were Lochinvar Commercial Solar Flat Plate Collectors, which are available in 65-130 square-foot options. The 130 square-foot collectors were used due to the ample roof space available. The profiles of calculated useful harnessed energy compared to the heating demand profiles to determine the optimal number of collectors, as described in Section 19.3.4 of this report. The physical dimensions of this collector are shown in Figure 31 below.

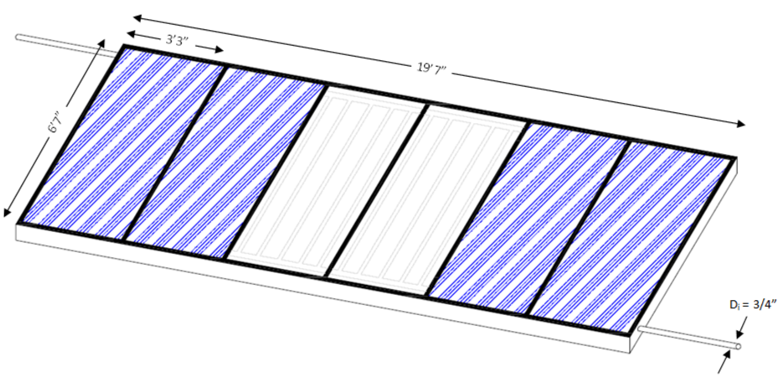

Figure 31: Solar Collector Dimensions

The other necessary collector characteristics were either taken from the manufacturer's catalog or an assumed typical value. Table 30 below summarizes these characteristics and their source.

Table 30: Flat Plate Collector Characteristics

Property	Symbol	Value	Units	Source
Cover Reflectance	ρ_d	0.50%	-	Assumed Typical Value
Cover Transmittance	τ	75%	-	Assumed Typical Value
Plate Absorptance	α	90%	-	Assumed Typical Value
Distance between Tubes	w	0.15	m	Calculated based on geometry
Tube Outer Diameter	D_o	0.02	m	Calculated based on flow rate
Tube Inner Diameter	D_i	0.01905	m	Calculated based on flow rate
Bond Conductance	C_b	25	$W/m^2 °C$	Assumed Typical Value
Heat Transfer Coeffient in Tubes	h_{fi}	300	$W/m^2 °C$	Assumed Typical Value
Plane Thermal Conductivity	k	400	W/m°C	Assumed Typical Value
Plate Thickness	δ	5	mm	Assumed Typical Value
Fluid Inlet Temperature	T_i	35	°C	Assumed Typical Value
Fluid Mass Flow	$\dot{m}$	0.082	kg/s	Manufacturer Information
Top Loss Coefficient	U_T	5.0		Assumed Typical Value
Bottom Loss Coefficient	U_B	3.0	$W/m^2 °C$	Assumed Typical Value
Edge Loss Coefficient	U_E	1.0		Assumed Typical Value

In order to perform a full analysis of the collector, a transmittance-absorptance product ($\tau\alpha$) was calculated. This value is not the product of the two properties, but a cover-absorber property calculated by the following equation:

$$(\tau\alpha) = \frac{\tau\alpha}{1-(1-\alpha)\rho_d} = 0.67534$$ **(Equation 9)**

Though this property varies with the angle of incidence of the sun, it is assumed for this analysis that the ($\tau\alpha$) value is constant for all hours of the day. This value will be used in determining the fraction of beam, diffuse, and reflected radiation that is absorbed by the collector.

Through the series of equations that follow, the heat removal factor, F_R, was calculated. The first calculated value was m, a unit less variable to be used in later equations:

$$m = \sqrt{\frac{U_L}{k\delta}} = 2.1213$$ **(Equation 10)**

where U_L = Overall collector loss coefficient ($U_T + U_B + U_E$)

The collector fin efficiency, F, was then determined:

$$F = \frac{\tanh\left(\frac{m(W-D_O)}{2}\right)}{\frac{m(W-D_O)}{2}} = 0.9937$$ **(Equation 11)**

F' is defined as the collector efficiency factor and was calculated by Equation 11:

$$F' = \frac{{}^{1}/_{U_L}}{W \cdot \left[\frac{1}{U_L(D_O+(W-D_O)F)} + \frac{1}{C_b} + \frac{1}{\pi D_i h_{fi}}\right]} = 0.8813$$ **(Equation 12)**

F_R was then able to be calculated:

$$F_R = \frac{\dot{m}c_p}{A_c U_L}\left[1 - exp\left(-\frac{A_c U_L F'}{\dot{m}c_p}\right)\right] = 0.8812$$ **(Equation 13)**

This collector heat removal factor was then used to yield a more accurate estimation of useful energy harnessed by the calculated incident solar radiation, as described in the next section of this report.

(19.3.3) Useful Harnessed Energy

The total incident solar radiation and the collector characteristics were then used to determine the total useful energy harnessed, per unit area of collector, by the given collector. To determine the total incident radiation that would actually be absorbed by the collector, the incident radiation values are combined with the (τα) values calculated above, and the ratio of the angle of incidence to the zenith angle, R_b, shown below:

$$R_b = \frac{\cos(\Theta)}{\cos(\Theta_z)}$$ **(Equation 14)**

The total absorbed radiation, S, is then calculated through Equation 10 below:

$$S = I_b R_b (\tau\alpha)_b + I_d (\tau\alpha)_d \frac{1+\cos(\beta)}{2} + I\rho_g (\tau\alpha)_g \frac{1-\cos(\beta)}{2}$$ **(Equation 15)**

where S = Total absorbed solar radiation in MJ/m^2
I_b = Beam incident radiation
I_d = Diffuse incident radiation
I = Total incident radiation ($I_d + I_b$)
ρ_g = Ground reflectance (assumed to be 0.5)

With all of the previous information, the total useful energy harnessed by the collector per unit area, Q_u/A_c, was determined by Equation 11 below:

$$\frac{Q_u}{A_c} = F_R(S - U_L(T_i - T_a))$$ **(Equation 16)**

where T_i = Inlet fluid temperature
T_a = Ambient outdoor temperature, taken from TMY3 data

With these equations input into the custom-made Excel program, the values for the collector tilt (β) and surface azimuth (γ) were manipulated to determine the combination that gave the greatest useful harnessed energy per unit area.

Common tilt angles were explored and compared, as shown in Figure 32 on the following page. The range of tilt angles between 30˚-45˚ from horizontal yielded the greatest averages, and it was determined that the optimal collector tilt in terms of total yearly energy harnessed is 37.2º from horizontal. For the purposes of practicality, a 40˚ collector tilt will be used for the design of the system.

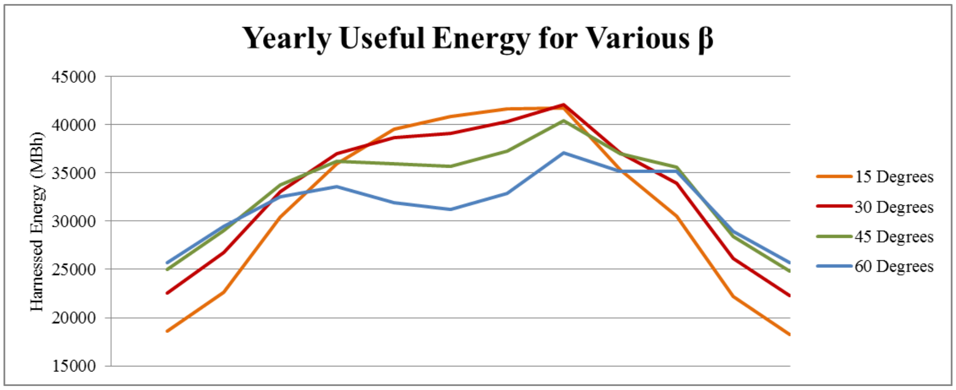

Figure 32: Collector Tilt Angle Comparison

A similar strategy was used to determine the optimal surface azimuth angle, or the direction in which the collector faces, measured from true south. The optimal surface azimuth angle was determined to be 37.5˚ west of south. Again for the purposes of practicality, the surface azimuth used in analysis was 33˚W so it would line up with the geometry of the building.

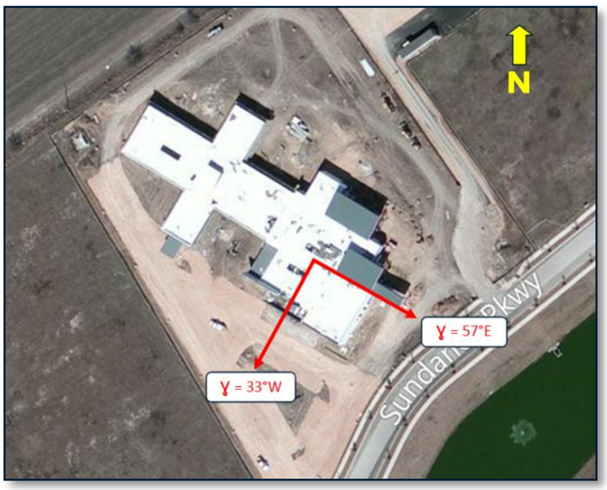

Figure 33: Surface Azimuth Angles

(19.3.4) Solar Thermal Load Profiles

Thermal storage makes it possible for excess absorbed energy collected during peak collection times to be used when the peak heating load exceeds the amount of energy being collected. The amount of excess collected energy is dependent on the number of collectors, and it is desirable to find the best tradeoff between the number of collectors that need to be purchased and the amount of hours of load that can be met.

For a typical day in October, when there is a moderate space heating requirement, the total peak heating demand (including domestic hot water heating, pool water heating, and secondary space heating) is plotted with the total solar gain for one, two, three, and four flat-plate collectors. The total demand in October is representative of the average for the year because the secondary space heating is close to the average for the year and the hot water heating demand is constant. These sample profiles are shown in Figures 34-37 on the next page.

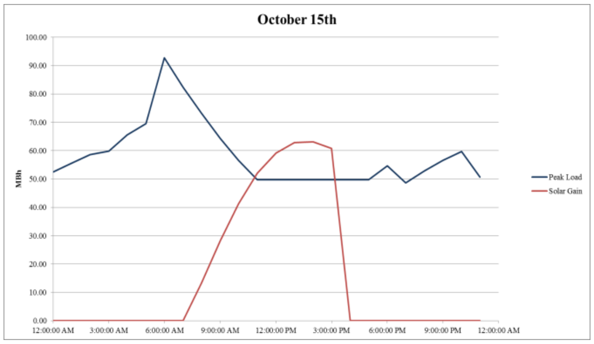

Figure 34: 1 Collector Load Profile

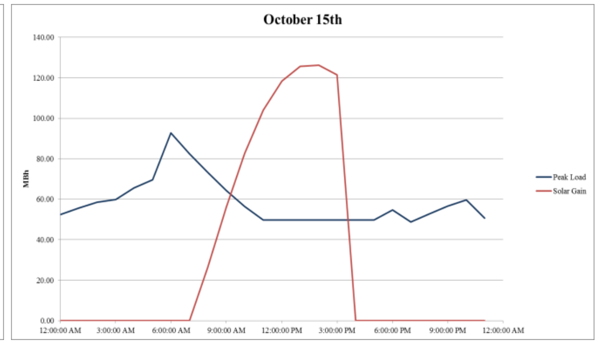

Figure 35: 2 Collectors Load Profile

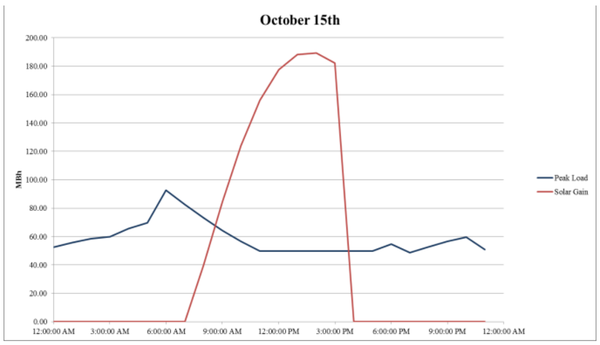

Figure 36: 3 Collectors Load Profile

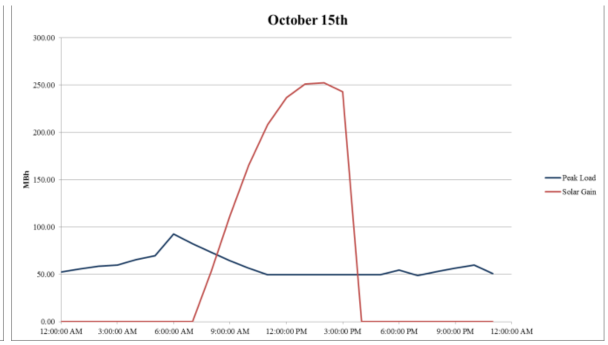

Figure 37: 4 Collectors Load Profile

A similar analysis was performed for a typical day of each month, and it was determined that three 130 square-foot collectors (390 square feet of collector total) yielded the most even trade-off between excess collected energy and heating demand not met at the time of collection. A load profile for each month of the year with three collectors is shown in Appendix G.

A stratified solar thermal storage tank was then chosen from the collector manufacturer. This manufacturer recommends approximately one gallon of storage capacity per square foot of collector space. Since there is a significant amount of excess collected energy during collection times, a 400-gallon tank was chosen to handle the solar thermal storage.

(19.3.5) Collector Arrangement

Using the calculated optimal area of collector and collector tilt angle, the array of collectors were located on the roof just about the mechanical room in the southwest corner of the facility. This will reduce heat loss from piping to the storage tank and heating loads. The collectors were located in such a way where they avoid shading from each other and from other rooftop equipment, as shown in Figure 38 on the next page.

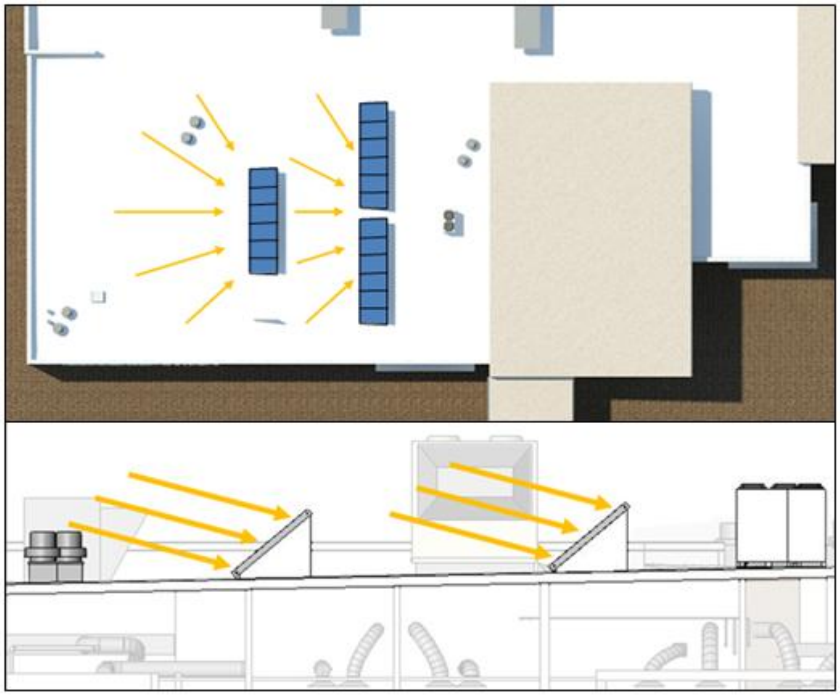

Figure 38: Solar Thermal Collector Arrangement

(19.3.6) Mechanical Room Redesign

The elimination of the hot water heaters for the combination storage tank and water heater frees up room in the southwest mechanical room for the water-to-water heat exchanger on the collector side. Additionally, the room becomes less cluttered and is able to house the additional circulating pumps and increased piping, as shown below in Figure 39.

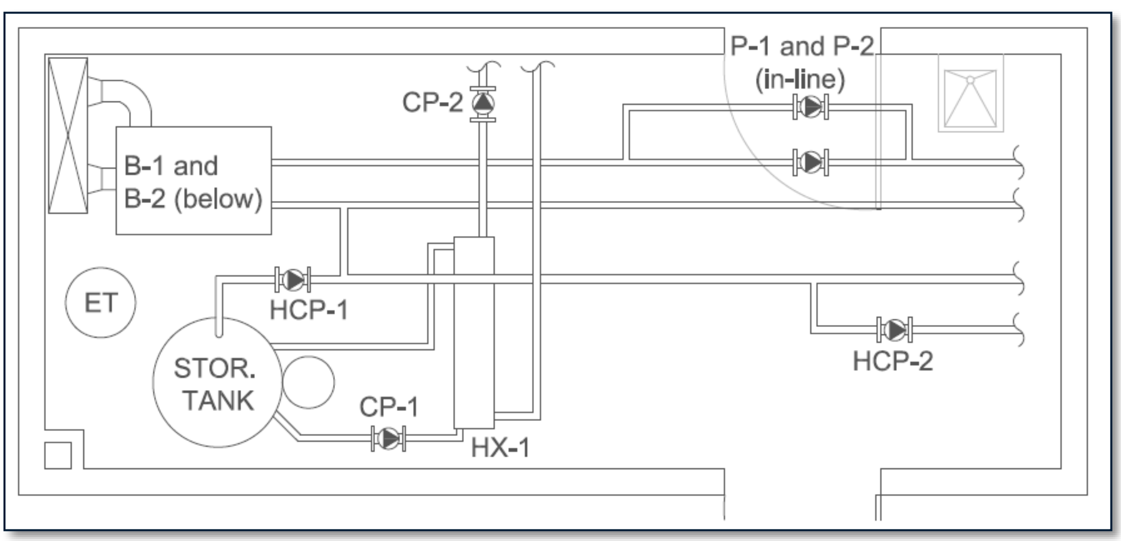

Figure 39: Mechanical Room Layout for Solar Thermal System

(19.4) Solar Thermal System Evaluation

(19.4.1) Energy Evaluation

The primary purpose of the solar thermal system and its thermal storage is to heat domestic hot water and therapy pool water, while space heating is supplemented when solar energy is in excess. As designed, the solar thermal system is able to offset an average total of 76% of the annual domestic hot water heating costs for the facility, with about 38% coming from direct solar energy gained and an additional 38% coming from thermal storage. The annual domestic water heating load profile is shown below in Figure 40 with percentages of load met by the solar thermal system shown for each month.

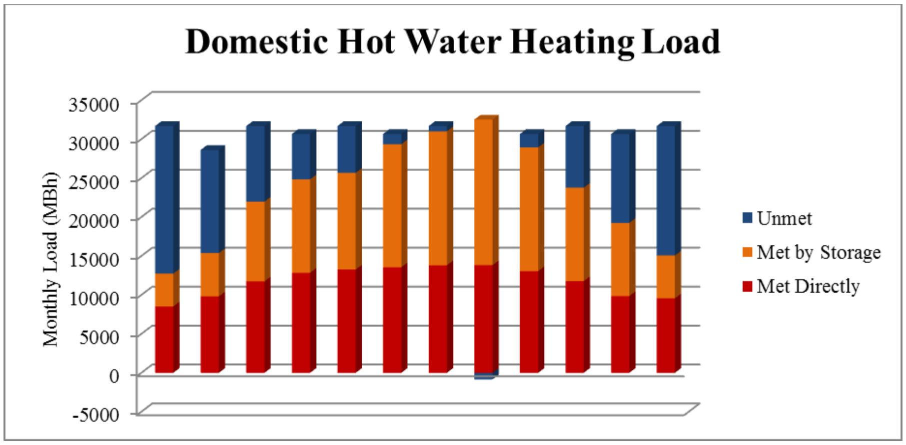

Figure 40: Domestic Hot Water Heating Load Met

The secondary application is able to reduce energy consumption by about 22% due to gains from the solar thermal system. Around 8% of these savings come from direct solar energy gained while 14% is supplemented by thermal storage. The annual space heating load profile is shown below in Figure 41 with percentages of load met by the solar thermal system shown for each month.

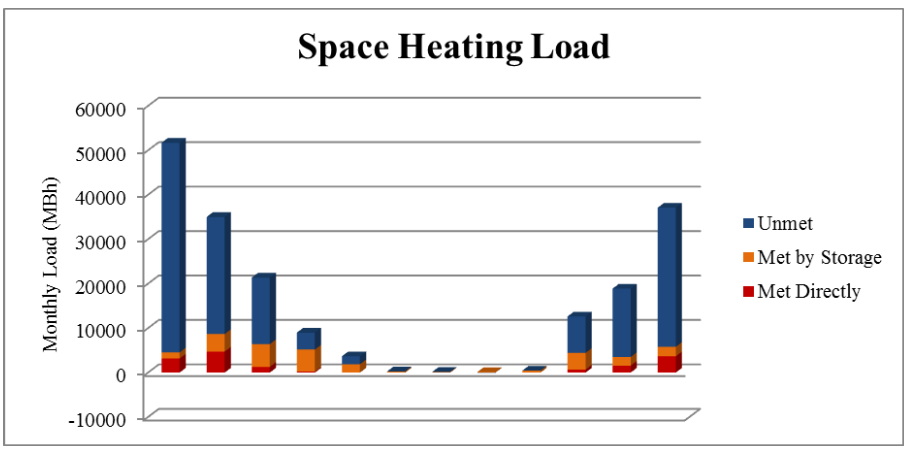

Figure 41: Space Heating Load Met

A combined total of 51.37% of the domestic hot water and space heating energy is able to be supplemented by the solar thermal system. This accounts for an energy reduction of nearly 290,000 MBh per year, summarized in Table 31 below.

Table 31: Solar Thermal System Energy Savings

Month	DHW Load (MBh)			Space Heating Load (MBh)		
	Directly	Storage	Unmet	Directly	Storage	Unmet
January	8521.28	4218.17	18939.14	3162.93	1365.55	47173.94
February	9815.68	5565.28	13231.96	4711	3968.72	26336.24
March	11741.87	10238.68	9698.04	1287.12	5121.2	14964.63
April	12839.7	12004.2	5812.8	225.3	4965.6	3795
May	13254.98	12421.08	6002.53	0	1884.18	1799.86
June	13557.9	15790.2	1308.6	0	252.9	2.7
July	13806.78	17197.25	674.56	0	146.63	0.31
August	13819.18	18689.9	-830.49	0	117.49	-0.93
September	13039.2	15896.7	1720.8	0	459	0
October	11763.88	12001.34	7913.37	662.78	3767.12	8194.85
November	9855.9	9397.8	11403	1572.6	1927.2	15386.1
December	9552.34	5506.53	16619.72	3665.75	2122.26	31307.21
Savings:	**141568.7**	**138927.1**		**15287.48**	**26097.85**	
				Total Energy Savings:	***321881.2***	***MBh***

(19.4.2) Economic Evaluation

An economic analysis was performed separately for the solar thermal system because, unlike the previous systems considered, the solar thermal operation involves the heating of domestic hot water. Therefore, the amount of natural gas used in this analysis is considerably higher than the other economic analyses.

There is a significant increase in first cost for a solar thermal system because all of the equipment mentioned in this section is auxiliary equipment and the existing system must remain in place as well. A summary of the first cost increase of $61,906 is shown in Table 32 below.

Table 32: Solar Thermal System First Cost

Component	Material	Labor	Total
RTUs	$199,391	$12,000	$211,391
Air Distribution Equipment	$114,480	$18,000	$132,480
Ductwork and Insulation	$190,277	$206,566	$396,843
MAU System	$91,797	$20,000	$111,797
Pool Dehumidification Unit	$20,757	$5,000	$25,757
Boilers and Control Interface	$23,624	$7,000	$30,624
Hydronic Distribution Equipment	$4,450	$7,000	$11,450
Mechanical Piping	$73,889	$155,869	$229,758
DDC Controls	$44,700	$105,200	$149,900
Flat-Plate Collectors	$19,198	$7,560	$26,758
Stratified Storage Tank	$3,685	$985	$4,670
Collector to Water HX	$4,028	$895	$4,923
Water to Pool Water HX	$2,225	$1,255	$3,480
Piping and Distribution Equipment	$3,665	$7,890	$11,555
Solar Thermal Controls	$4,200	$6,320	$10,520
Added First Cost	-	-	$100,000
			$1,461,906

The energy savings yield a decrease of operating cost by $30,814 per year, which gives a simple payback of just over 2 years. This is an appropriate payback period for the size of the system installed and the climate of San Antonio. Any incentives that may be available were not factored into the payback period.

The net present value of the solar thermal system is over $327,000 greater than the existing system over a 20-year life-cycle period, as summarized in Tables I6-I7 of Appendix I. The short simple payback period combined with the high net present value of the solar system makes it a very viable redesign alternative.

(20.0) Breadth Study 1: Structural Analysis of Roof

A structural analysis of the roof framing system was performed to determine whether the existing roof would be able to meet the new requirements of a VRF and/or solar thermal system. New framing members were designed as necessary for both systems and the new cost of steel was factored into the life cycle analysis of each system as discussed previously.

(20.1) Existing Roof Framing System

The existing metal deck roof is supported by variably-sized K-Series bar joists and wide-flange joist girders. There are also several wide flange beams supporting the roof under the existing rooftop units. The takeoffs for these members were included in the wide flange girder category. A summary of the quantity of each member and its total associated cost is shown below in Table 33. Framing plans of each area as described in the table are shown in Appendix H with mechanical equipment loads shown.

Table 33: Roof Framing Cost with Existing System

Member	Material $/ft.	Linear ft.		Total Cost
		Area 1	Area 2	
Wide Flange Girders				$ 124,798.04
W10x12	$ 16.50	0	0	$ -
W12x14	$ 17.40	0.00	268.25	$ 4,667.55
W12x16	$ 19.80	0.00	68.58	$ 1,357.95
W12x19	$ 23.40	0.00	214.00	$ 5,007.60
W12x24	$ 29.50	486.33	83.17	$ 16,800.25
W12x26	$ 32.00	53.50	0.00	$ 1,712.00
W14x22	$ 27.00	102.83	282.25	$ 10,397.25
W16x26	$ 32.00	876.92	445.75	$ 42,325.33
W16x31	$ 38.50	0.00	28.00	$ 1,078.00
W16x36	$ 44.61	0.00	28.00	$ 1,249.11
W18x35	$ 43.50	280.83	263.17	$ 23,664.00
W18x40	$ 49.50	124.00	90.00	$ 10,593.00
W21x44	$ 54.50	0.00	28.00	$ 1,526.00
W21x50	$ 62.00	0.00	34.00	$ 2,108.00
W24x55	$ 68.00	0.00	34.00	$ 2,312.00
K-Series Joists				$ 30,898.65
10K1	$ 3.19	0.00	57.00	$ 181.83
12K1	$ 3.19	627.75	112.00	$ 2,359.80
16K2	$ 3.50	0.00	155.00	$ 543.16
16K3	$ 4.01	1979.50	0.00	$ 7,937.80
18K3	$ 4.20	267.50	1519.42	$ 7,505.05
20K4	$ 5.15	878.50	710.83	$ 8,185.07
24K4	$ 5.27	0.00	793.83	$ 4,185.94
Total Structural Framing Cost:				$ 155,696.70

The roof live and dead loads were assessed via methods described in ASCE 7-10. Framing self-weight was determined to be 5 psf, which is a common assumption for K-Series open bar joists and wide flange joist girders. A standard superimposed dead load for a roof of 8 psf was also applied. The maximum snow load of 20 psf for Texas was taken from the USDA Forest Service – Missoula Technology & Development's snow load data. The 1½" metal deck was determined to have a 2.14 psf self-weight per information in the Vulcraft Deck Catalog. The uniformly distributed live load for a roof was determined to be 20 psf for an ordinary flat roof according to ASCE 7-10 Table 4-1.

These live and dead loads were then adjusted per the load combination that usually controls for roof framing members not part of the lateral load restraining system, shown in Equation 17:

$$Load = 1.2D + 1.6(L_r \; or \; S) \qquad \textbf{(Equation 17)}$$

where D = Total dead load = member self-weight + deck weight + superimposed
L_r = Roof live load
S = Snow load

The total of live and dead loads acting on the roof was calculated to be 50.17 psf. When comparing the joist sizes to this calculated load, it is apparent that additional factors of safety were used to size the joists, so these were evaluated on a bay-by-bay basis when redesigning structural members for the new mechanical systems.

(20.2) Roof Framing with VRF System

The elimination of RTU-1 allows for the beams that support the roof underneath that unit to be replaced by the same K-Series joists that support the rest of that patient room wing roof. However, the additional weight of the exterior condensing units, a layout of which is shown on the partial framing plan in Figure 42 below, requires additional roof support.

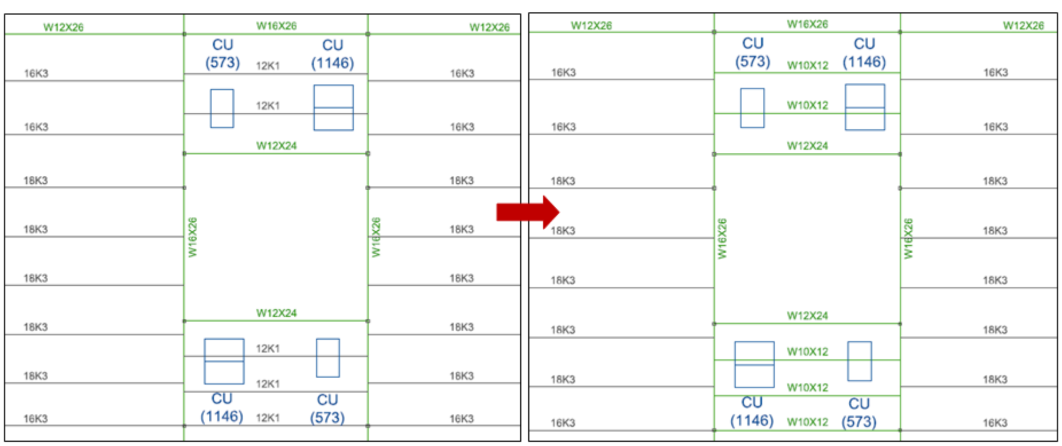

Figure 42: New Roof Framing Members

The area of each of the bays that support the units is approximately 235 square feet. It was determined that the 12K1 Series joists that exist in these bays were designed to handle a maximum load of 630 pounds per linear foot; thus a live and dead load of 160.85 psf was assumed based on the previous sizing and the 5 foot tributary width.

The condensing units add a total of 1,719 pounds over the bay's total area, which causes the total load on the roof to increase to 168.18 psf. This is converted into a load of 0.659 kips per linear foot, which is then analyzed in terms of shear and moment on the beam, as shown in Figure 43 below.

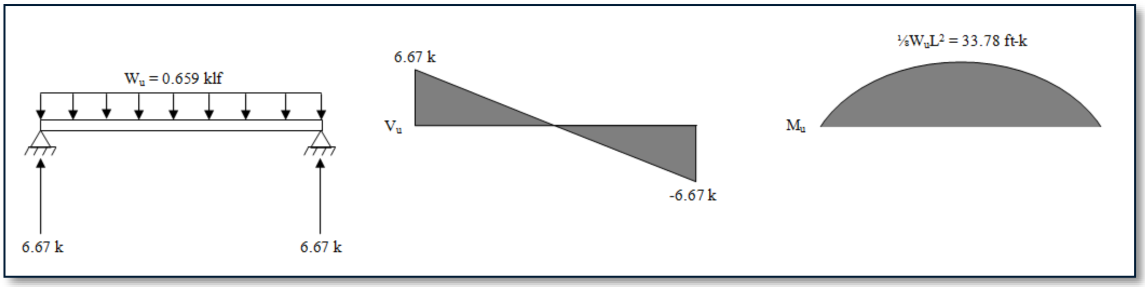

Figure 43: Shear and Moment Diagrams

A W10x12 beam was determined to be the most economical based on the moment applied to the beam. The flexural and shear strength of the beam were confirmed via Equations 18-19 before it was determined that the W10x12 beam was acceptable for the new loads.

$$M_u \leq \phi M_n \quad \textbf{(Equation 18)} \qquad\qquad V_u \leq \phi V_n \quad \textbf{(Equation 19)}$$

where $\phi = 0.9$

The new structural roof framing system cost was calculated with the elimination of the wide flange beams supporting RTU-1 and the addition of beams supporting the condensing units. It was determined that the new structural framing system can save over $2,900 in material cost. The summary of members and total cost is shown in Table 34 below.

Table 34: Roof Framing Cost with VRF System

Member	Material $/ft.	Linear ft. Area 1	Linear ft. Area 2	Total Cost
Wide Flange Girders				**$ 121,689.67**
W10x12	$ 16.50	101.25	0	$ 1,670.63
W12x14	$ 17.40	0.00	268.25	$ 4,667.55
W12x16	$ 19.80	0.00	68.58	$ 1,357.95
W12x19	$ 23.40	0.00	214.00	$ 5,007.60
W12x24	$ 29.50	324.33	83.17	$ 12,021.25
W12x26	$ 32.00	53.50	0.00	$ 1,712.00
W14x22	$ 27.00	102.83	282.25	$ 10,397.25
W16x26	$ 32.00	876.92	445.75	$ 42,325.33
W16x31	$ 38.50	0.00	28.00	$ 1,078.00
W16x36	$ 44.61	0.00	28.00	$ 1,249.11
W18x35	$ 43.50	280.83	263.17	$ 23,664.00
W18x40	$ 49.50	124.00	90.00	$ 10,593.00
W21x44	$ 54.50	0.00	28.00	$ 1,526.00
W21x50	$ 62.00	0.00	34.00	$ 2,108.00
W24x55	$ 68.00	0.00	34.00	$ 2,312.00
K-Series Joists				**$ 31,092.45**
10K1	$ 3.19	0.00	57.00	$ 181.83
12K1	$ 3.19	688.50	112.00	$ 2,553.60
16K2	$ 3.50	0.00	155.00	$ 543.16
16K3	$ 4.01	1979.50	0.00	$ 7,937.80
18K3	$ 4.20	267.50	1519.42	$ 7,505.05
20K4	$ 5.15	878.50	710.83	$ 8,185.07
24K4	$ 5.27	0.00	793.83	$ 4,185.94
			Total Structural Framing Cost:	**$ 152,782.11**

(20.3) Roof Framing with Solar Thermal System

A similar analysis was performed with the new loads due to the rooftop solar thermal collectors. The two bays that support these new loads have a total area of 2,658 ft^2. Additionally, the outdoor condensing unit serving VRF system 6 is located on the roof over these bays, so it was factored into this analysis. It was determined that the joists spanning these bays were sized for a maximum of 378 pounds per linear foot, so it was assumed that the designer used a roof load of approximately 75.6 psf.

The additional equipment load is only 2,415 pounds spread over 2,658 square feet, which accounts for an increase in load of less than a pound per square foot, so it was determined that the roof joists supporting the roof under the solar thermal collectors were sufficient to support the new loads. There is no redesign of roof framing members necessary for the installation of this solar thermal system.

(21.0) Breadth Study 2: Room Acoustics Investigation

A room acoustics evaluation was performed to determine the noise criteria (NC) levels in a typical patient room for the existing system and for the VRF system. For this breadth analysis, only mechanical system sound sources within the room or in the ceiling plenum were considered, although there would be a small amount of sound propagation through walls from adjacent rooms. Additionally, exhaust fans from adjacent patient restrooms were not taken into account because the patient rooms would not be occupied when the patient was using the restroom.

This analysis is based on methods presented in *Architectural Acoustics* by Long et al. All equations used in the following sections are based on sections of this text, though the analyses and interpretations presented here are original.

(21.1) Existing System Room Acoustics

In the existing system there are three sound sources from the mechanical system, shown below in Figure 44: the supply air diffuser, return air grille, and VAV terminal unit in the ceiling plenum. Unweighted sound pressure levels were taken from manufacturer information or information included in the mechanical designer's specifications.

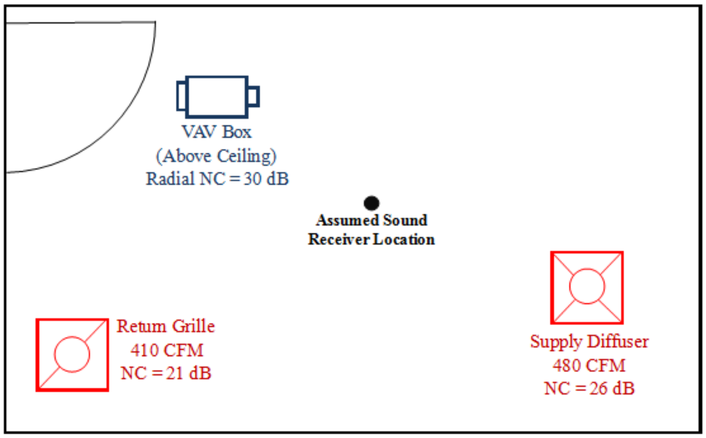

Figure 44: Patient Room Sound Sources

Common material sound absorption and transmission values were assumed for the patient room surfaces and are summarized in Table 35 below.

Table 35: Patient Room Material Properties

Surface	Total Area (ft^2)	Absorption Coefficient (α)	Transmission Coefficient (τ)
Plaster Walls	552	0.05	-
Vinyl Tile Floor	221.67	0.03	-
Lay-In Ceiling	217.67	0.77	0.39

From these values a total area absorption, A, was calculated by taking the sum of the products of surface area and absorption coefficients and determined to be 201.9 sabins. The amount of sound pressure lost though the air in 60 seconds, ΔL_{air}, was then calculated through Equation 20:

$$\Delta L_{air} = 0.049\frac{Volume}{A}$$ **(Equation 20)**

This air loss was determined to be 0.48 dB in 60 seconds of traveling through the air. The room constant, R_r, was then calculated through Equation 21 and determined to be 3737 sabins.

$$R_r = A + 4\cdot\left(4\frac{L_{air}}{4.34}\right)V$$ **(Equation 21)**

Using these values, the sound pressure level at the receiver was calculated through Equation 22 below and determined to be 17 dB.

$$L_r = L_s - 10\log(\tau_{ceiling}) - 10log\left[\frac{S_wQ}{16\pi\left[z+\sqrt{\frac{S_wQ}{4\pi}}\right]^2} + \frac{S_w}{R_r}\right]$$ **(Equation 22)**

where L_r = sound pressure level at the receiver, in dB
L_s = sound pressure level from source, in dB
S_w = area of transmitting surface, in ft^2
z = distance from the source to receiver
Q = directivity of the wall = 2.0

The NC values for each source were then A-weighted per Equation 23 below and added together through decibel addition to determine the NC level of the existing system in a typical patient room.

$$NC \cong 1.25(L_A - 13)$$ **(Equation 23)**

where NC= Noise criteria level, from manufacturer information, in dB
L_A = sound pressure level in dBA

A summary of each noise source and its associated sound pressure levels are shown on the following page in Table 36, which gives a total NC level of 36 dBA for a typical patient room. Recommended noise criterion levels for a private hospital room range from 35-45 dBA, so the existing mechanical system is able to achieve a proper noise level.

Table 36: Patient Room Sound Pressure Levels

Source	dB Level	dBA Level
Supply Diffuser	26	34
Return Grille	21	30
VAV Box	17	26
	Total:	**36**

(21.2) VRF System Room Acoustics

With the installation of the proposed VRF system, the supply diffuser and VAV terminal unit sound sources are eliminated and the indoor evaporation unit introduces a new sound source. According to manufacturer data, the NC level of these units ranges from 39-45 dBA. When this sound pressure level is added to the level of the return grille, it is determined that the grille's level does not have a noticeable effect on the overall NC level of the room.

The total NC level of a patient room served by the VRF system will be dictated by the indoor evaporating unit and will vary from 39-45 dBA. This is a noticeable increase from the existing system's level of 36 dBA, but it is still within the range of acceptable noise criteria levels for a private hospital patient room.

(22.0) Conclusion

Although air-cooled, direct expansion units seem to be a very inefficient option for a hot Texas climate, the relatively small size of the New Braunfels Regional Rehabilitation Hospital creates obstacles in the feasibility of a chilled water system. While this system would save cooling energy, the increased upfront cost of the system does not make it economically practical.

When all of the design alternatives presented in this report are considered, it is apparent that a solar thermal system used to primarily heat domestic hot water is a very cost-effective way to save energy in the New Braunfels Regional Rehabilitation Hospital. Additionally, a variable refrigerant flow system may be a feasible alternative to the existing system, depending on the owner's payback threshold. In order to confidently install a VRF system, a more detailed analysis into occupant health and comfort in patient rooms where new indoor units were installed would need to be performed.

While great efforts have been taken to provide accurate and complete information in the pages of this report, the modifications and changes presented here are solely the interpretation of this writer. Many of the assumptions made in calculations or estimations may not be entirely accurate and should be confirmed if these systems are investigated further. The changes and discrepancies in no way imply that the original design contained errors or was flawed.

Appendix A: References

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Appendix B: ASHRAE Weather Data Sheet

2009 ASHRAE Handbook - Fundamentals (IP)

SAN ANTONIO INTL AP, TX, USA

WMO#: 722530

Lat: 29.53N Long: 98.46W Elev: 810 StdP: 14.27 Time Zone: -6.00 (NAC) Period: 82-06 WBAN: 12921

Annual Heating and Humidification Design Conditions

Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB	
			99.6%			99%			0.4%		1%			
	99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD
1	27.4	31.6	10.3	9.6	38.9	15.6	12.5	44.6	24.4	46.8	20.9	50.2	8.3	10

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB	
		0.4%		1%		2%		0.4%		1%		2%			
		DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD
8	20.1	98.5	73.5	96.9	73.6	95.2	73.7	78.0	88.0	77.3	87.1	76.7	86.2	9.6	160

Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Hours 8 to 4 & 55/69
0.4%			1%			2%			0.4%		1%		2%		
DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB	
75.9	139.4	80.1	75.2	136.0	79.8	74.4	132.6	79.8	42.0	88.0	41.2	86.9	40.6	86.0	690

Extreme Annual Design Conditions

Extreme Annual WS			Extreme Max WB	Extreme Annual DB				n-Year Return Period Values of Extreme DB							
				Mean		Standard deviation		n=5 years		n=10 years		n=20 years		n=50 years	
1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max
20.2	18.2	16.6	82.9	21.5	102.2	5.8	2.6	17.4	104.1	14.0	105.6	10.8	107.1	6.6	109.0

Monthly Climatic Design Conditions

			Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Temperatures, Degree-Days and Degree-Hours	Tavg		69.5	52.2	55.6	62.1	69.5	76.7	82.0	84.6	85.1	79.7	71.2	61.1	53.1
	Sd			9.13	9.55	8.47	7.10	5.30	3.77	2.95	2.97	5.64	7.44	9.10	9.95
	HDD50		239	83	47	13	0	0	0	0	0	0	1	15	80
	HDD65		1480	406	283	154	39	1	0	0	0	2	30	179	386
	CDD50		7350	150	204	389	586	828	961	1071	1089	891	658	348	175
	CDD65		3115	8	20	65	175	364	511	606	624	443	221	62	16
	CDH74		32766	44	172	442	1390	3224	5615	7251	7674	4688	1870	341	55
	CDH80		15154	4	44	98	440	1283	2666	3733	4062	2162	607	52	3
Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		78.8	85.4	87.1	93.2	96.5	99.2	100.4	100.2	99.6	92.2	85.0	79.1
		MCWB		60.5	60.9	64.8	68.1	72.5	74.7	72.9	73.7	72.3	71.6	68.6	63.9
	2%	DB		74.3	79.0	82.4	88.1	93.0	96.3	98.3	98.4	95.6	89.2	81.0	75.2
		MCWB		60.6	61.1	65.0	67.9	72.8	74.5	73.4	73.8	73.5	71.1	67.5	62.6
	5%	DB		71.0	74.5	79.0	84.7	89.9	93.8	96.4	96.8	93.1	86.5	78.0	72.1
		MCWB		60.1	60.3	63.8	67.9	72.8	74.3	73.8	73.8	73.2	70.3	66.5	62.3
	10%	DB		67.7	70.6	75.3	81.9	87.0	91.5	94.2	95.1	90.6	83.3	74.9	69.0
		MCWB		59.8	60.4	62.9	67.1	72.4	74.4	74.0	74.0	72.7	69.6	65.5	62.1
Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		68.4	68.8	71.3	75.2	78.3	78.7	78.4	78.3	78.2	76.8	73.2	69.4
		MCDB		71.3	73.8	78.6	83.5	88.3	89.5	88.7	88.9	86.3	83.5	78.6	72.7
	2%	WB		66.5	67.3	69.7	73.4	76.7	77.8	77.6	77.4	77.1	75.5	71.5	67.8
		MCDB		69.8	71.9	75.3	80.9	86.0	88.2	87.9	87.3	85.2	81.9	76.3	71.1
	5%	WB		64.3	65.6	68.5	72.1	75.4	77.1	77.0	76.8	76.3	74.5	70.1	66.1
		MCDB		68.1	69.7	73.9	79.1	83.6	87.0	86.9	86.7	84.2	80.8	74.8	69.7
	10%	WB		61.1	63.1	66.9	70.8	74.4	76.3	76.3	76.2	75.5	73.0	68.5	63.7
		MCDB		66.4	69.0	72.4	77.3	82.5	85.7	85.8	86.0	83.3	79.3	73.3	68.1
Mean Daily Temperature Range		MDBR		20.7	20.8	21.0	21.1	18.9	18.7	19.1	20.1	20.1	20.4	20.3	20.5
	5% DB	MCDBR		25.5	26.7	24.6	24.6	21.3	21.0	21.9	21.9	21.8	21.7	20.9	23.0
		MCWBR		15.4	13.9	12.0	11.3	7.1	5.5	4.6	4.6	5.8	8.2	11.1	13.7
	5% WB	MCDBR		17.6	17.9	17.1	18.0	17.7	18.0	18.4	19.3	17.7	16.1	16.4	17.1
		MCWBR		13.7	12.5	10.0	9.7	7.3	6.1	4.7	5.0	5.9	7.6	11.2	13.2
Clear Sky Solar Irradiance	taub			0.336	0.351	0.366	0.395	0.421	0.435	0.441	0.449	0.423	0.368	0.345	0.331
	taud			2.422	2.371	2.308	2.223	2.189	2.174	2.167	2.144	2.228	2.398	2.443	2.501
	Ebn,noon			285	290	292	286	277	271	269	267	270	281	280	282
	Edh,noon			33	36	41	45	47	48	48	48	43	35	32	29

CDDn Cooling degree-days base n°F, °F-day
CDHn Cooling degree-hours base n°F, °F-hour
DB Dry bulb temperature, °F
DP Dew point temperature, °F
Ebn,noon } Clear sky beam normal and diffuse hori-
Edh,noon } zontal irradiances at solar noon, Btu/h/ft2
Elev Elevation, ft
Enth Enthalpy, Btu/lb
HDDn Heating degree-days base n°F, °F-day
Hours 8/4 & 55/69 Number of hours between 8 a.m. and 4 p.m with DB between 55 and 69 °F
HR Humidity ratio, grains of moisture per lb of dry air
Lat Latitude, °
Long Longitude, °
MCDB Mean coincident dry bulb temperature, °F
MCDBR Mean coincident dry bulb temp. range, °F
MCDP Mean coincident dew point temperature, °F
MCWB Mean coincident wet bulb temperature, °F
MCWBR Mean coincident wet bulb temp. range, °F
MCWS Mean coincident wind speed, mph
MDBR Mean dry bulb temp. range, °F
PCWD Prevailing coincident wind direction, °, 0 = North, 90 = East
Period Years used to calculate the design conditions
Sd Standard deviation of daily average temperature, °F
StdP Standard pressure at station elevation, psi
taub Clear sky optical depth for beam irradiance
taud Clear sky optical depth for diffuse irradiance
Tavg Average temperature, °F
Time Zone Hours ahead or behind UTC, and time zone code
WB Wet bulb temperature, °F
WBAN Weather Bureau Army Navy number
WMO# World Meteorological Organization number
WS Wind speed, mph

Appendix C: Ventilation Rate Calculations

Table C1: RTU-1 Ventilation Rate Calculation

RTU-1 VENTILATION CALCULATIONS

Min OA (CFM): 6850
Total Airflow (CFM): 26000
Outdoor Air Fraction (Z_p): 0.2635

Standard 170 Analysis

Room Name	Occupancy Category	Total Airflow (Supply/Exhaust) (CFM)	Area (SF)	Height (ft)	Total ACH	Total Outdoor ACH	Minimum Total ACH	Minimum Outdoor ACH	Room Complies?	Required Ventilation Air (CFM)
Corridor	Corridor	550	576	9	6.4	1.7	2	0	Y	0.0
Patient Room	Patient Room	440	215	9	13.6	3.6	6	2	Y	64.5
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	410	214	9	12.8	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	440	215	9	13.6	3.6	6	2	Y	64.5
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	410	214	9	12.8	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	440	215	9	13.6	3.6	6	2	Y	64.5
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	410	214	9	12.8	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	500	215	9	15.5	4.1	6	2	Y	64.5
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	430	214	9	13.4	3.5	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Corridor	Corridor	660	718	9	6.1	1.6	2	0	Y	0.0
Patient Room	Patient Room	440	214	9	13.7	3.6	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	440	214	9	13.7	3.6	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	440	214	9	13.7	3.6	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	440	214	9	13.7	3.6	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	440	214	9	13.7	3.6	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	380	214	9	11.8	3.1	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	380	214	9	11.8	3.1	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	440	214	9	13.7	3.6	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	440	214	9	13.7	3.6	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	440	214	9	13.7	3.6	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	440	214	9	13.7	3.6	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	440	214	9	13.7	3.6	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Corridor	Corridor	680	699	9	6.5	1.7	2	0	Y	0.0
Corridor	Corridor	660	718	9	6.1	1.6	2	0	Y	0.0
Patient Room	Patient Room	420	214	9	13.1	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	420	214	9	13.1	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	420	214	9	13.1	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	420	214	9	13.1	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Isolation Patient Room	Patient Room	420	214	9	13.1	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Isolation Patient Room	Patient Room	500	214	9	15.6	4.1	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	380	214	9	11.8	3.1	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	420	214	9	13.1	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	420	214	9	13.1	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	420	214	9	13.1	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	420	214	9	13.1	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	420	214	9	13.1	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Corridor	Corridor	340	632	9	3.6	0.9	2	0	Y	0.0
Corridor	Corridor	550	575	9	6.4	1.7	2	0	Y	0.0
Patient Room	Patient Room	410	214	9	12.8	3.4	6	2	Y	64.2

(Table continued on following page)

Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	480	214	9	15.0	3.9	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	410	214	9	12.8	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	480	214	9	15.0	3.9	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	410	214	9	12.8	3.4	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	480	214	9	15.0	3.9	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	430	214	9	13.4	3.5	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Patient Room	Patient Room	500	214	9	15.6	4.1	6	2	Y	64.2
Toilet	Toilet Room	70	51	8	10.3	2.7	10	0	Y	0.0
Nourishment	Pharmacy	100	103	9	6.5	1.7	10	0	N	0.0
Alcove	Corridor	100	102	9	6.5	1.7	2	0	Y	0.0
Alcove	Corridor	100	103	9	6.5	1.7	2	0	Y	0.0
Assisted Bathing	Bathing Room	150	172	9	5.8	1.5	10	0	N	0.0
IV Prep	Sterilizing	570	78	9	48.7	12.8	10	0	Y	0.0
Pharmacy	Pharmacy	190	183	9	6.9	1.8	4	2	N	54.9
Med Room	Pharmacy	100	102	9	6.5	1.7	4	2	N	30.6
Equipment Storage	Sterile Storage	70	65	9	7.2	1.9	4	2	N	19.5
Toilet	Toilet Room	40	58	8	5.2	1.4	10	0	N	0.0
Toilet	Toilet Room	40	58	8	5.2	1.4	10	0	N	0.0
Clean Linen Supply	Clean Linen	170	190	9	6.0	1.6	2	0	Y	0.0
Soiled Linen Utility	Soiled Linen	200	174	9	7.7	2.0	10	0	N	0.0
Respiratory Storage	Sterile Storage	100	99	9	6.7	1.8	4	2	N	29.7
Body Holding	Body Holding	50	79	9	4.2	1.1	10	0	N	0.0
Restroom	Toilet Room	50	66	8	5.7	1.5	10	0	N	0.0
Corridor	Corridor	340	166	9	13.7	3.6	2	0	Y	0.0
Med Gas Pump Room	Medical Gas Storage	500	188	9	17.7	4.7	0	8	N	225.6
Med Gas Access	Medical Gas Storage	70	99	9	4.7	1.2	0	8	N	118.8
Equipment Storage	Sterile Storage	400	290	9	9.2	2.4	4	2	Y	87.0

Standard 62.1 Analysis

Room Name	Occupancy Category	Total Supply Air (CFM)	Area (SF)	Max Occupants	CFM/SF	CFM/Person	Required Ventilation Air (CFM)
Day Room	Break Room	1120	423	8	0.06	5	65.4
Tele/Data	Equipment Room	800	131	0	0.06	0	7.9
Storage	Storage	200	198	0	0.12	0	23.8
Electrical	Equipment Room	800	123	0	0.06	0	7.4
Office	Office Space	300	252	4	0.06	5	35.1
Office	Office Space	80	79	4	0.06	5	24.7
Office	Office Space	80	78	4	0.06	5	24.7
Nursing Admin	Office Space	80	80	4	0.06	5	24.8
Nursing Admin	Office Space	80	80	4	0.06	5	24.8
Storage	Storage	30	33	0	0.12	0	4.0
Housekeeping	Storage	70	47	0	0.12	0	5.6
Pharmacy Office	Office Space	90	97	4	0.06	5	25.8
Staff Breakroom	Break Room	180	157	2	0.06	5	19.4
Respiratory Office	Office Space	100	101	4	0.06	5	26.1
Housekeeping	Storage	0	56	0	0.12	0	6.7
Storage	Storage	100	97	0	0.12	0	11.6
Nurse Station	Office Space	3840	2355	23	0.06	5	256.3
							3729.4

Table C2: RTU-2 Ventilation Rate Calculation

RTU-2 VENTILATION CALCULATIONS

Min OA (CFM): 2015
Total Airflow (CFM): 12000
Outdoor Air Fraction (Z_p): 0.1679

Room Name	Occupancy Category	Total Supply Airflow (CFM)	Area (SF)	Max Occupants	CFM/SF	CFM/Person	Required Ventilation Air (CFM)
Outpatient Day Treatment	Health/Aerobics	1900	1710	18	0.06	10	282.6
Storage	Storage	50	36	0	0.12	0	4.3
Storage	Storage	50	67	0	0.12	0	8.0
Therapy Gym	Health/Aerobics	4620	1829	18	0.06	10	289.7
Charting	Office Space	500	402	4	0.06	5	44.1
Therapy Reception	Lobby	230	350	0	0.06	5	21.0
Clinical Director	Office Space	100	90	1	0.06	5	10.4
Toilet	Restroom	50	63	0	0	0	0.0
Private Therapy	Office Space	100	101	2	0.06	5	16.1
Housekeeping	Storage	50	49	0	0.12	0	5.9
ADL Suite	Office Space	380	363	1	0.06	5	26.8
Corridor	Corridor	130	188	0	0.06	0	11.3
Locker	Restroom	120	89	0	0	0	0.0
Locker	Restroom	150	105	0	0	0	0.0
Speech Therapy	Office Space	100	112	4	0.06	5	26.7
Speech Therapy	Office Space	100	112	4	0.06	5	26.7
Classroom	Classroom	440	350	18	0.12	10	222.0
Transition Suite	Office Space	500	291	1	0.06	5	22.5
Hallway	Corridor	500	189	0	0.06	0	11.3
Hallway	Corridor	320	389	0	0.06	0	23.3
Reception Coordination	Office Space	150	201	2	0.06	5	22.1
Files	Storage	50	69	0	0.12	0	8.3
Toilet	Restroom	50	52	0	0	0	0.0
Storage	Storage	50	44	0	0.12	0	5.3
Exam	Office Space	120	104	6	0.06	5	36.2
Medical Director	Office Space	80	99	2	0.06	5	15.9
Clinical Director	Office Space	80	99	2	0.06	5	15.9
Hallway	Corridor	150	218	0	0.06	0	13.1
Exam	Office Space	90	100	6	0.06	5	36.0
Emergency Treatment	Office Space	160	813	6	0.06	5	78.8
Waiting	Lobby	160	206	6	0.06	5	42.4
							1326.8

Table C3: RTU-3 Ventilation Rate Calculation

RTU-3 VENTILATION CALCULATIONS

Min OA (CFM): 4550

Total Airflow (CFM): 17500

Outdoor Air Fraction (Z_p): 0.2600

Room Name	Occupancy Category	Total Supply Air (CFM)	Area (SF)	Max Occupants	CFM/SF	CFM/Person	Required Ventilation Air (CFM)
Maint Office	Office Space	190	120	1	0.06	5	12.2
Dry Storage	Storage	190	133	0	0.12	0	16.0
Dietician	Office Space	120	126	1	0.06	5	12.6
Housekeeping	Storage	50	48	0	0.12	0	5.8
Dietary Breakdown Receiving	Storage	100	87	0	0.12	0	10.4
M Staff Toilet	Restroom	150	70	0	0	0	0.0
Dish Wash	Kitchen	300	171	0	0	0	0.0
Kitchen	Kitchen	1800	867	10	0	0	0.0
Serving Line	Cafeteria	790	1021	18	0.18	7.5	318.8
Dining	Cafeteria	3700	1435	88	0.18	7.5	918.3
Day Room	Break Room	700	465	8	0.06	5	67.9
W Staff Toilet	Restroom	150	123	0	0	0	0.0
Vestibule	Lobby	600	145	4	0.06	5	28.7
Lobby	Lobby	750	859	15	0.06	5	126.5
Waiting	Lobby	750	137	2	0.06	5	18.2
Admission	Conference	510	274	10	0.06	5	66.4
Conference Room	Conference	370	265	12	0.06	5	75.9
Restroom	Restroom	200	173	0	0	0	0.0
Restroom	Restroom	200	173	0	0	0	0.0
Hallway	Corridor	400	1328	0	0.06	0	79.7
Conference Room	Conference	450	278	14	0.06	5	86.7
Quality	Office Space	110	109	1	0.06	5	11.5
Marktg	Office Space	100	102	1	0.06	5	11.1
CEO	Office Space	450	147	1	0.06	5	13.8
CFO	Office Space	430	119	1	0.06	5	12.1
Marketing	Office Space	100	94	1	0.06	5	10.6
Workroom	Office Space	330	117	1	0.06	5	12.0
Admin	Office Space	100	338	2	0.06	5	30.3
Business Office	Office Space	540	553	4	0.06	5	53.2
Payroll	Office Space	100	88	1	0.06	5	10.3
BOM Office	Office Space	400	87	1	0.06	5	10.2
Medical Records	Storage	380	88	0	0.12	0	10.6
Medical Records	Storage	550	300	0	0.12	0	36.0
General Storage	Storage	350	419	0	0.12	0	50.3
Mechanical	Equipment Room	240	276	0	0.06	0	16.6
Electrical	Equipment Room	500	106	0	0.06	0	6.4
ATS/Emergency Electric	Equipment Room	370	202	0	0.06	0	12.1
Maintenance Shop	Equipment Room	240	309	0	0.06	0	18.5
Telecom	Equipment Room	500	95	0	0.06	0	5.7
Soiled Linen	Storage	130	59	0	0.12	0	7.1
Storage	Storage	60	71	0	0.12	0	8.5
Corridor	Corridor	400	491	0	0.06	0	29.5
Hallway	Corridor	270	349	0	0.06	0	20.9
Conference Room	Conference	200	119	6	0.06	5	37.1
Files	Storage	50	67	0	0.12	0	8.0
Human Resources	Office Space	150	118	2	0.06	5	17.1
Tray Return	Storage	100	80	0	0.12	0	9.6
							2313.3

Appendix D: Possible LEED Credits

Energy & Atmosphere — 17 Possible Points

Prereq 1	**Fundamental Commissioning of the Building Energy Systems**	Required
Prereq 2	**Minimum Energy Performance**	Required
Prereq 3	**Fundamental Refrigerant Management**	Required
Credit 1	**Optimize Energy Performance**	1–10
Credit 2	**On-Site Renewable Energy**	1–3
Credit 3	**Enhanced Commissioning**	1
Credit 4	**Enhanced Refrigerant Management**	1
Credit 5	**Measurement & Verification**	1
Credit 6	**Green Power**	1

Indoor Environmental Quality — 15 Possible Points

Prereq 1	**Minimum IAQ Performance**	Required
Prereq 2	**Environmental Tobacco Smoke (ETS) Control**	Required
Credit 1	**Outdoor Air Delivery Monitoring**	1
Credit 2	**Increased Ventilation**	1
Credit 3.1	**Construction IAQ Management Plan**, During Construction	1
Credit 3.2	**Construction IAQ Management Plan**, Before Occupancy	1
Credit 4.1	**Low-Emitting Materials**, Adhesives & Sealants	1
Credit 4.2	**Low-Emitting Materials**, Paints & Coatings	1
Credit 4.3	**Low-Emitting Materials**, Carpet Systems	1
Credit 4.4	**Low-Emitting Materials**, Composite Wood & Agrifiber Products	1
Credit 5	**Indoor Chemical & Pollutant Source Control**	1
Credit 6.1	**Controllability of Systems**, Lighting	1
Credit 6.2	**Controllability of Systems**, Thermal Comfort	1
Credit 7.1	**Thermal Comfort**, Design	1
Credit 7.2	**Thermal Comfort**, Verification	1
Credit 8.1	**Daylight & Views**, Daylight 75% of Spaces	1
Credit 8.2	**Daylight & Views**, Views for 90% of Spaces	1

Appendix E: Chilled Water Plant First Cost Breakdown

Existing System

Component	Material	Labor	Total
RTUs	$199,391	$12,000	$211,391
Air Distribution Equipment	$114,480	$18,000	$132,480
Ductwork and Insulation	$190,277	$206,566	$396,843
MAU System	$91,797	$20,000	$111,797
Pool Dehumidification Unit	$20,757	$5,000	$25,757
Boilers and Control Interface	$23,624	$7,000	$30,624
Hydronic Distribution Equipment	$4,450	$7,000	$11,450
Mechanical Piping	$73,889	$155,869	$229,758
DDC Controls	$44,700	$105,200	$149,900
Added First Cost	-	-	$100,000
			$1,400,000

Water-Cooled CHW Plant w/ Existing Boilers

Component	Material	Labor	Total
Reciprocating Chillers and Controls	$122,500	$12,300	$134,800
Cooling Tower	$23,400	$1,950	$25,350
CHW Pumps	$7,400	$940	$8,340
CW Pumps	$4,300	$500	$4,800
RTUs	$199,391	$12,000	$211,391
Air Distribution Equipment	$114,480	$18,000	$132,480
Ductwork and Insulation	$190,277	$206,566	$396,843
MAU System	$91,797	$20,000	$111,797
Pool Dehumidification Unit	$20,757	$5,000	$25,757
Boilers and Control Interface	$23,624	$7,000	$30,624
Hydronic Distribution Equipment	$8,900	$70,000	$78,900
Mechanical Piping	$147,778	$311,738	$459,516
DDC Controls	$44,700	$105,200	$149,900
Added First Cost	-	-	$100,000
			$1,870,498

Air-Cooled CHW Plant w/ Existing Boilers

Component	Material	Labor	Total
Air-Cooled Condenser	$82,850	$20,960	$103,810
CHW Pumps	$7,400	$940	$8,340
CW Pumps	$4,300	$500	$4,800
RTUs	$199,391	$12,000	$211,391
Air Distribution Equipment	$114,480	$18,000	$132,480
Ductwork and Insulation	$190,277	$206,566	$396,843
MAU System	$91,797	$20,000	$111,797
Pool Dehumidification Unit	$20,757	$5,000	$25,757
Boilers and Control Interface	$23,624	$7,000	$30,624
Hydronic Distribution Equipment	$8,900	$70,000	$78,900
Mechanical Piping	$147,778	$311,738	$459,516
DDC Controls	$44,700	$105,200	$149,900
Added First Cost	-	-	$100,000
			$1,814,158

Appendix F: VRF Indoor and Outdoor Unit Details

DAIKIN AC
absolute comfort

Submittal Data: FXSQ30MVJU
Indoor Unit – Concealed Ceiling Unit

Job Name:	Location:
Purchaser:	
Engineer:	
Submitted To:	For: ☐ Reference ☐ Approval ☐ Construction
Submitted By:	Date:
Unit Designation: Schedule #:	Model No.:

VRV SYSTEM
THE INTELLIGENT AIR CONDITIONING SYSTEM
R-410A

Capacities:

Cooling	30,000 Btu/h
Heating	34,000 Btu/h

Cooling Mode Nominal Conditions:
Indoor: 80°F DB / 67°F WB
Outdoor: 95°F DB
Pipe Length: 25ft
Level Difference: 0ft

Heating Mode Nominal Conditions:
Indoor: 70°F DB
Outdoor: 47°F DB / 43°F WB
Pipe Length: 25ft
Level Difference: 0ft

Fan:

Fan Type	Sirocco Fan
Fan Motor Output	225 Watts
Airflow Rate (H/L)	950/720 cfm
External Static Pressure (High/Standard)	0.57"/0.39" WG
Drive	Direct Drive

Power Supply:

V/PH/Hz	208-230/1/60
Minimum Circuit Amps (MCA)	1.8 Amps
Nominal Power Input – Cooling	315 Watts
Nominal Power Input - Heating	295 Watts
Maximum Fuse Amps (MFA)	15 Amps
Full Load Amps (FLA)	1.4 Amps

Refrigerant & Piping:

Refrigerant	R-410A
Control	Electronic Expansion Valve
Liquid Refrigerant Piping	ϕ3/8" Flare
Gas Refrigerant Piping	ϕ5/8" Flare
Condensate Drain Piping (OD)	ϕ1-1/4"

Unit Data:

Sound Pressure Level (H/L) - at 5 ft below bottom suction grill of unit	45/39 dBA
Weight	119 lbs

Ductwork Opening:

Supply Opening Height (Fixing Hole Center)	9-5/8 inches
Supply Opening Width (Fixing Hole Center)	46-5/8 inches
Return Opening Height (Fixing Hole Center)	9-7/8 inches
Return Opening Width (Fixing Hole Center)	53-1/8 inches

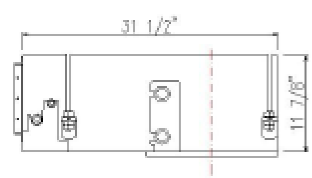
Standard Features:

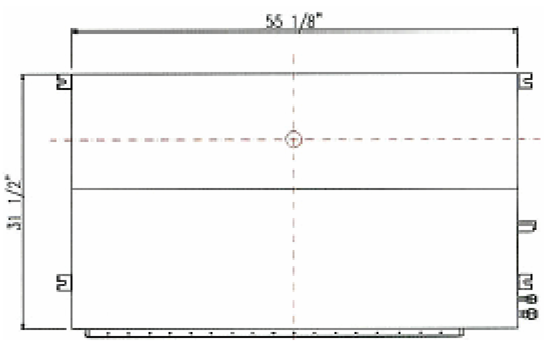
- One year limited labor warranty
- One year parts warranty
- Built-in drain pump
- Washable long life filter with mildew proof resin

Options:

- Wired remote controller
- Wireless remote controller
- Simplified controller
- Remote sensor kit
- Suction panel

Unit shown with optional decoration panel (P/N BYBS_ _DJW1)





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SDS FXSQ30MVJU 09-07 www.daikinac.com

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Submittal Data: RXYQ96PYDN

8 Ton VRV®III Outdoor Unit – Heat Pump

Job Name:	Location:
Purchaser:	
Engineer:	
Submitted To:	For: ☐ Reference ☐ Approval ☐ Construction
Submitted By:	Date:
Unit Designation: Schedule #:	Model No.:

Capacities:

Cooling Capacity	96,000 Btu/h
Cooling Input Power	8.6 kW
Heating Capacity	108,000 Btu/h
Heating Input Power	8.7 kW (29,684 Btu/h)

Cooling Mode Nominal Conditions:
Indoor: 80°F DB / 67°F WB
Outdoor: 95°F DB
Pipe Length: 25ft
Level Difference: 0ft

Heating Mode Nominal Conditions:
Indoor: 70°F DB
Outdoor: 47°F DB / 43°F WB
Pipe Length: 25ft
Level Difference: 0ft

Operating Range:

Cooling	23°F – 110°F DB
Heating	0°F – 77°F DB -4°F – 60°F WB

Compressor:

Type	Daikin High Pressure Dome Scroll
Quantity	2 (1 PAM Inverter and 1 on/off)
Capacity Control Range	Module: 14% – 100%
Motor Rating Output	(4.5 + 2.2) kW

Fan:

Fan Type x Quantity	Propeller Fan x 1
Fan Motor Output	750 Watts
Airflow Rate	6,530 cfm
Drive	Direct Drive

Power Supply:

V/PH/Hz	460V/3/60
Minimum Circuit Amps (MCA)	20.3 Amps
Maximum Fuse Amps (MFA)	25 Amps
Maximum Starting Current (MSC)	65 Amps
Total Overcurrent Amps (TOCA)	31.5 Amps

Refrigerant & Piping:

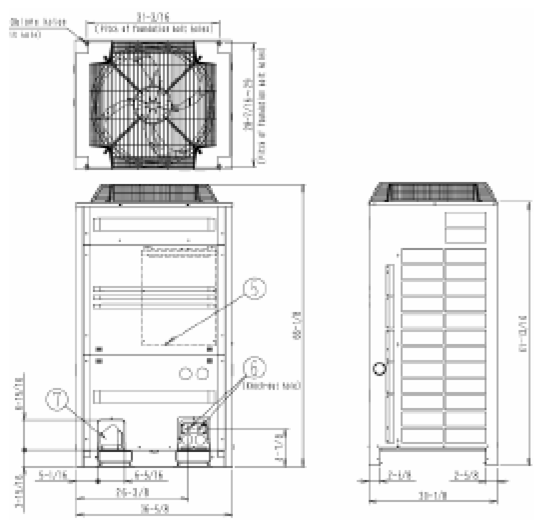
Refrigerant Type/Charge	R-410A/19.8 lbs
Control	Electronic Expansion Valve
Liquid Piping (Main Line)	ϕ3/8" Brazing
Suction Gas Piping (Main Line)	ϕ7/8" Brazing

Unit Data:

Max. Number Indoor Units	16
Sound Pressure Level at 3 ft	58dB (A)
Weight	573 lbs

Standard Features:

- Six year compressor warranty
- One year parts warranty
- PE coated condenser coil to prevent corrosion
- VFD controlled outdoor motor fan



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SDS RXYQ96PYDN 5-09 www.daikinac.com

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DAIKIN AC
absolute comfort

Submittal Data: RXYQ120PYDN

10 Ton VRV®III Outdoor Unit – Heat Pump

Job Name:	Location:
Purchaser:	
Engineer:	
Submitted To:	For: ☐ Reference ☐ Approval ☐ Construction
Submitted By:	Date:
Unit Designation: Schedule #:	Model No.:

Capacities:

Cooling Capacity	120,000 Btu/h
Cooling Input Power	10.9 kW
Heating Capacity	135,000 Btu/h
Heating Input Power	11.4 kW (38,897 Btu/h)

Cooling Mode Nominal Conditions:
Indoor: 80°F DB / 67°F WB
Outdoor: 95°F DB
Pipe Length: 25 ft
Level Difference: 0 ft

Heating Mode Nominal Conditions:
Indoor: 70°F DB
Outdoor: 47°F DB / 43°F WB
Pipe Length: 25 ft
Level Difference: 0 ft

Operating Range:

Cooling	23°F – 110°F DB
Heating	0°F – 77°F DB -4°F – 60°F WB

Compressor:

Type	Daikin High Pressure Dome Scroll
Quantity	2 (1 PAM Inverter and 1 on/off)
Capacity Control Range	Module: 14% - 100%
Motor Rating Output	(4.5 + 3.5) kW

Fan:

Fan Type x Quantity	Propeller Fan x 1
Fan Motor Output	750 Watts
Airflow Rate	7,060 cfm
Drive	Direct Drive

Power Supply:

V/PH/Hz	460V/3/60
Minimum Circuit Amps (MCA)	20.5 Amps
Maximum Fuse Amps (MFA)	30 Amps
Maximum Starting Current (MSC)	65 Amps
Total Overcurrent Amps (TOCA)	31.5 Amps

Refrigerant & Piping:

Refrigerant Type/Charge	R-410A/20.1 lbs
Control	Electronic Expansion Valve
Liquid Piping (Main Line)	ϕ1/2" Brazing
Suction Gas Piping (Main Line)	ϕ1-1/8" Brazing

Unit Data:

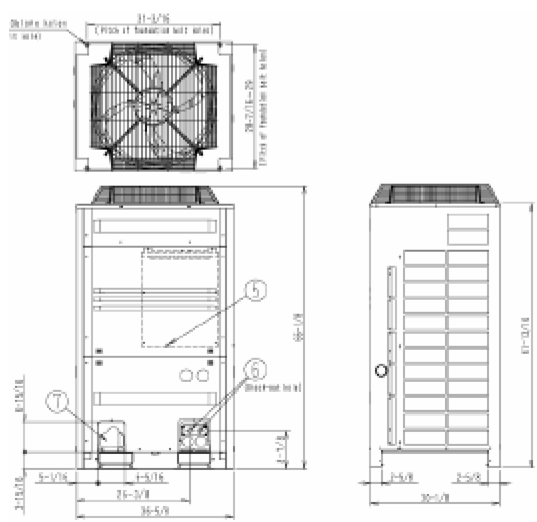
Max. Number Indoor Units	20
Sound Pressure Level at 3 ft	60dB(A)
Weight	573 lbs

Standard Features:

- Six year compressor warranty
- One year parts warranty
- PE coated condenser coil to prevent corrosion
- VFD controlled outdoor motor fan

VRV III

R-410A INVERTER



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SDS RXYQ120PYDN 5-09 www.daikinac.com

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Submittal Data: RXYQ144PYDN

12 Ton VRV®III Outdoor Unit – Heat Pump

(RXYQ72PYDN + RXYQ72PYDN + Manifold Kit)

Job Name: Location:

Purchaser:

Engineer:

Submitted To: For: ☐ Reference ☐ Approval ☐ Construction

Submitted By: Date:

Unit Designation: Schedule #: Model No.:

Capacities:

Cooling Capacity	144,000 Btu/h
Cooling Input Power	10.8 kW
Heating Capacity	162,000 Btu/h
Heating Input Power	12.0 kW (40,944 Btu/h)

Cooling Mode Nominal Conditions:
Indoor: 80°F DB / 67°F WB
Outdoor: 95°F DB
Pipe Length: 25ft
Level Difference: 0ft

Heating Mode Nominal Conditions:
Indoor: 70°F DB
Outdoor: 47°F DB / 43°F WB
Pipe Length: 25ft
Level Difference: 0ft

Operating Range:

Cooling	23°F – 110°F DB
Heating	0°F – 77°F DB -4°F – 60°F WB

Compressor:

Type	Daikin High Pressure Dome Scroll
Quantity	4 (2 PAM Inverter and 2 on/off)
Capacity Control Range	System: 13% - 100%
Motor Rating Output	(4.7) x 2 kW

Fan:

Fan Type x Quantity	Propeller Fan x 2
Fan Motor Output	750 + 750 Watts
Airflow Rate	6,530 + 6,530 cfm
Drive	Direct Drive

Power Supply:

V/PH/Hz	460V/3/60
Minimum Circuit Amps (MCA)	20.2 + 20.2 Amps
Maximum Fuse Amps (MFA)	25 + 25 Amps
Maximum Starting Current (MSC)	69 Amps
Total Overcurrent Amps (TOCA)	63.0 Amps

Refrigerant & Piping:

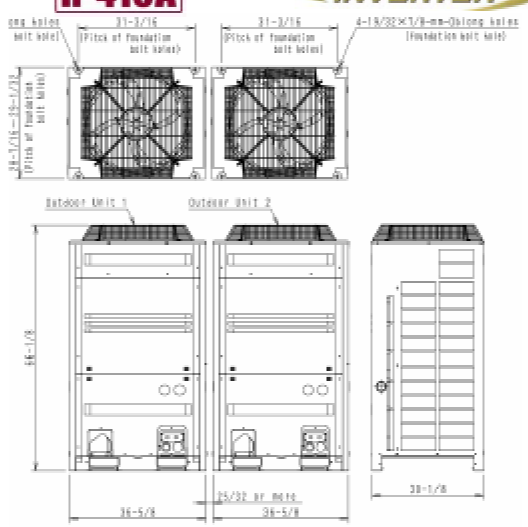
Refrigerant Type/Charge	R-410A/18.1 + 18.1 lbs
Control	Electronic Expansion Valve
Liquid Piping (Main Line)	ϕ1/2" Brazing
Suction Gas Piping (Main Line)	ϕ1-1/8" Brazing
High Pressure Equalizer Pipe	ϕ3/4" Brazing
Low Pressure Equalizer Pipe	ϕ3/4" Brazing

Unit Data:

Max. Number Indoor Units	25
Sound Pressure Level at 3 ft	61dB (A)
Weight	573 + 573 lbs

Standard Features:

- Six year compressor warranty
- One year parts warranty
- Continuous heating during defrost operation
- PE coated condenser coil to prevent corrosion
- VFD controlled outdoor motor fan
- Manifold Kit Part Number = BHFP22P100U



For further information about the installation allowances on manifolded systems, please refer to the Installation and Engineering Guides

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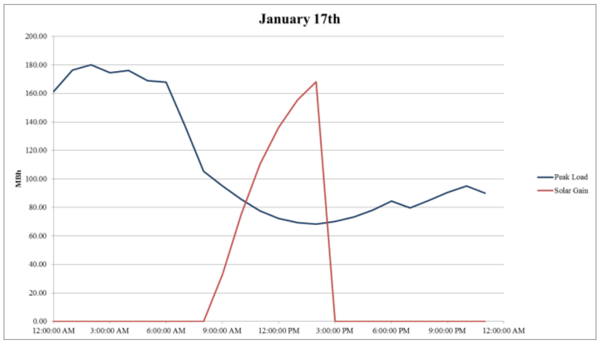
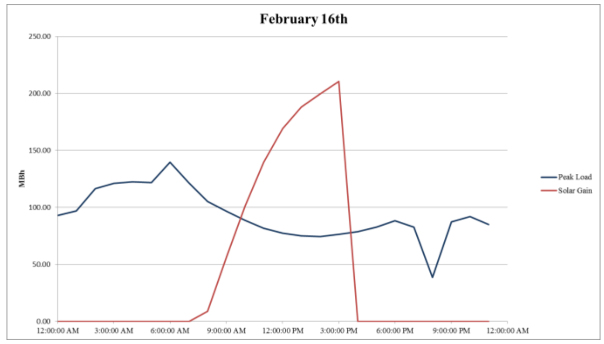
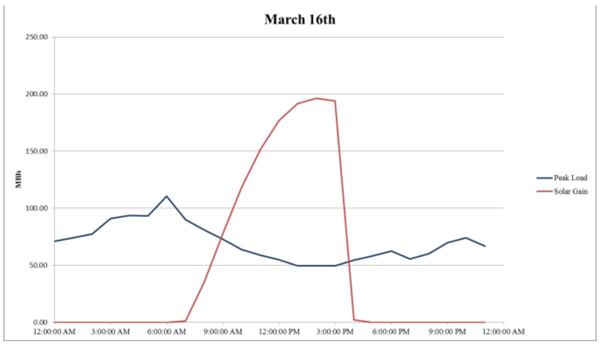
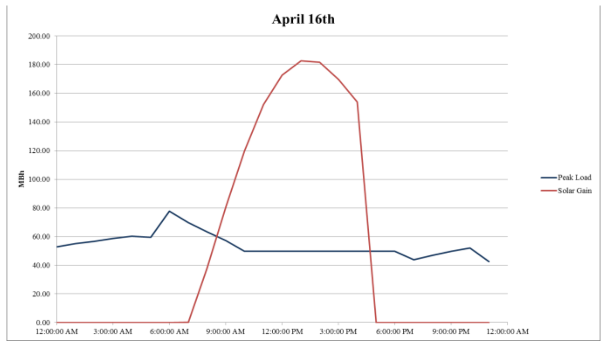
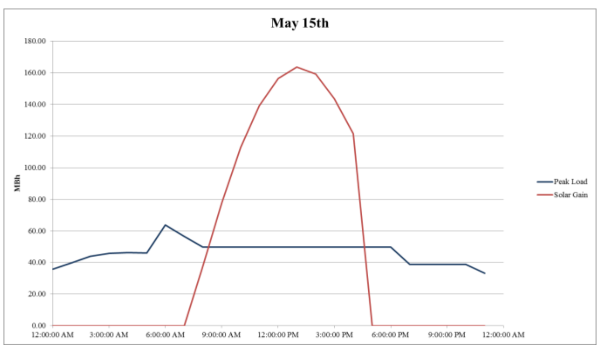
Appendix G: Solar Collector Load Profiles

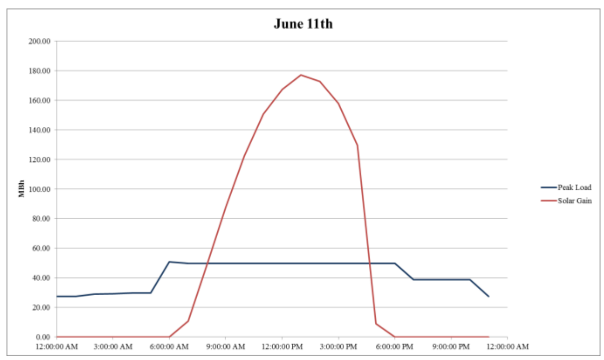


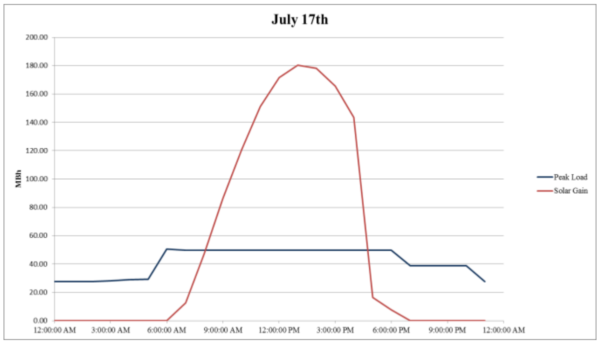


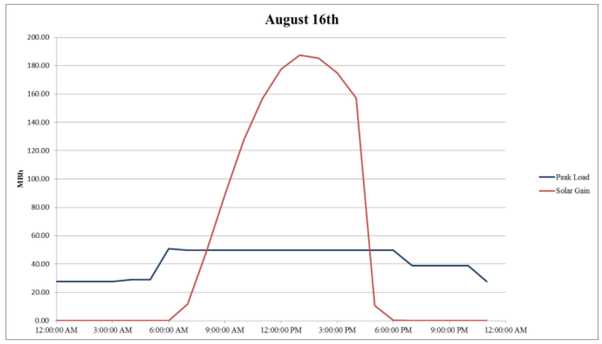


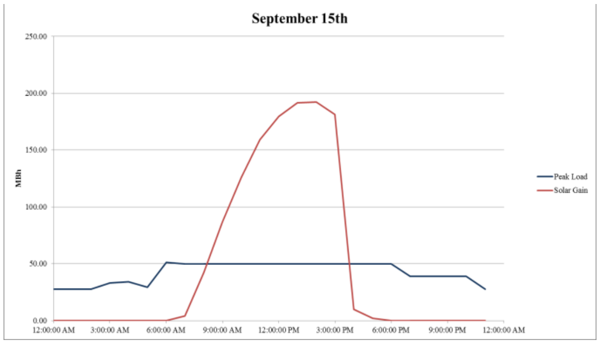


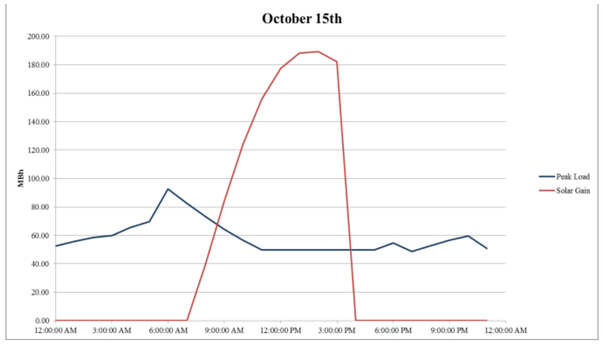


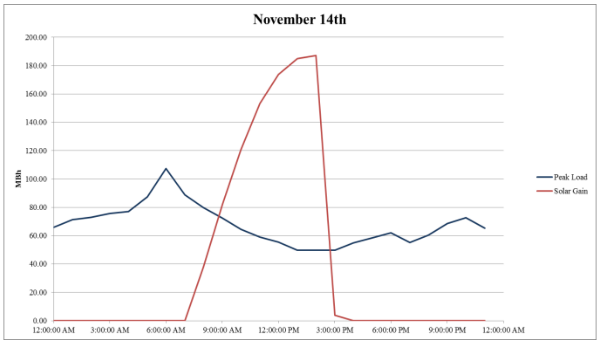


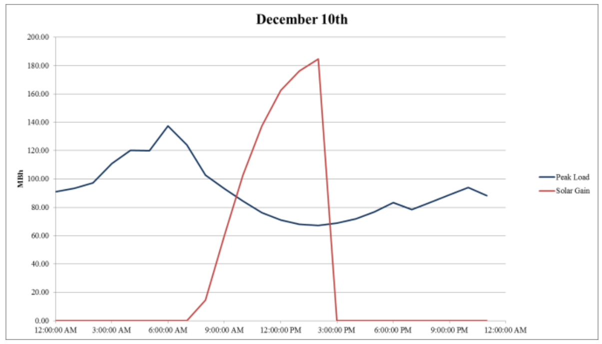

July 17th
MBh
Peak Load
Solar Gain


August 16th
MBh
Peak Load
Solar Gain


September 15th
MBh
Peak Load
Solar Gain

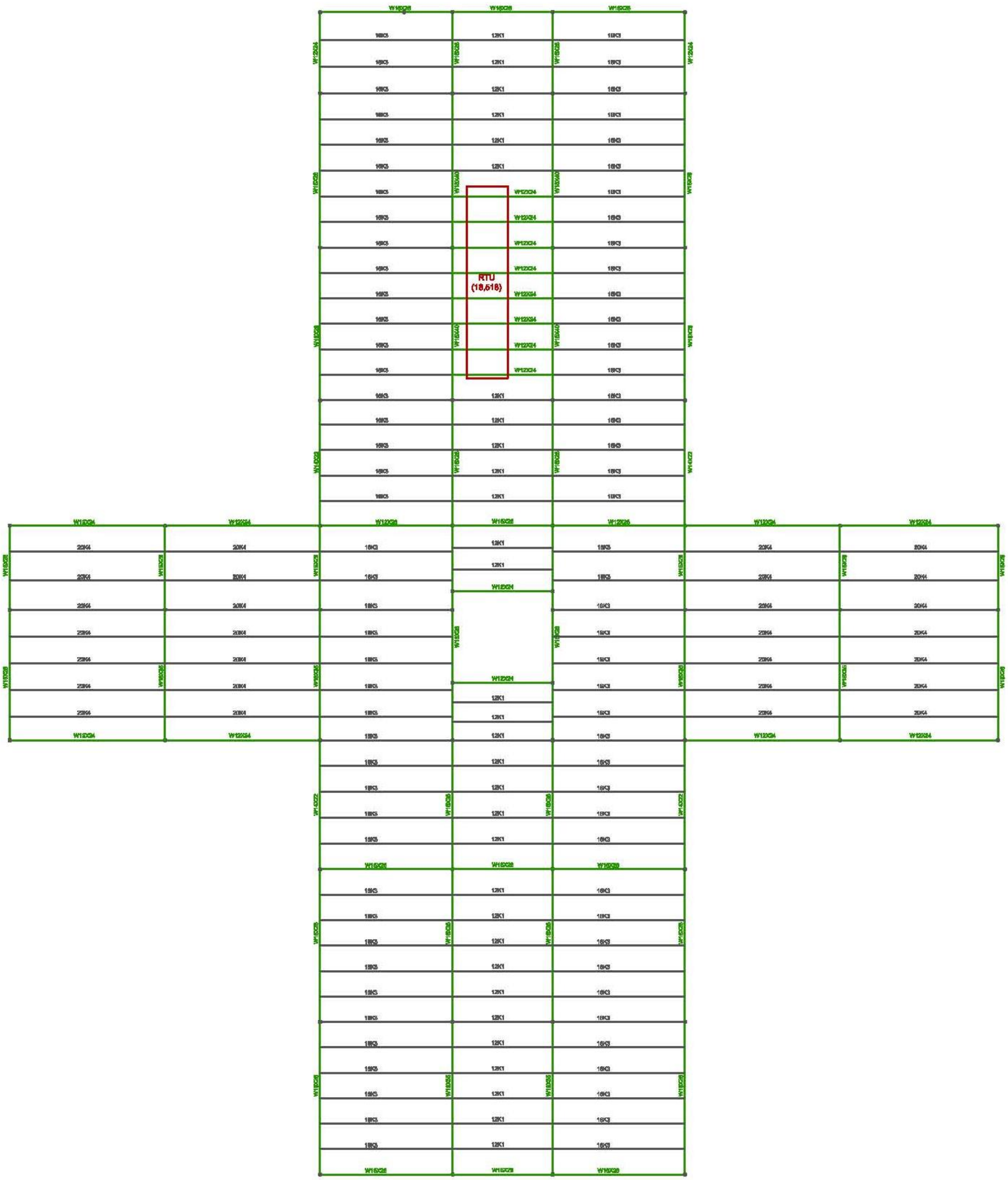

October 15th
MBh
Peak Load
Solar Gain


November 14th
MBh
Peak Load
Solar Gain

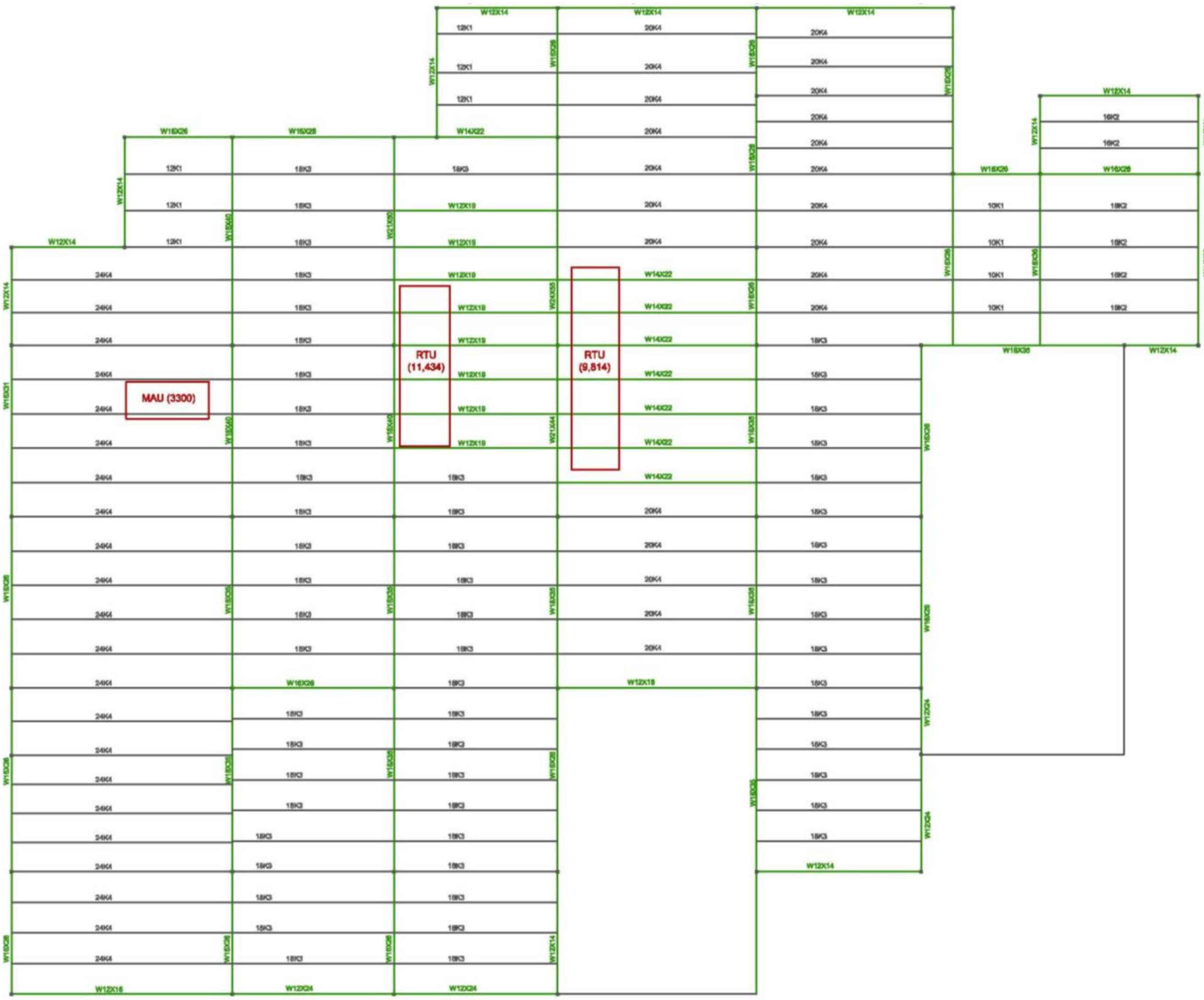

December 10th
MBh
Peak Load
Solar Gain

Appendix H: Structural Roof Framing Plans

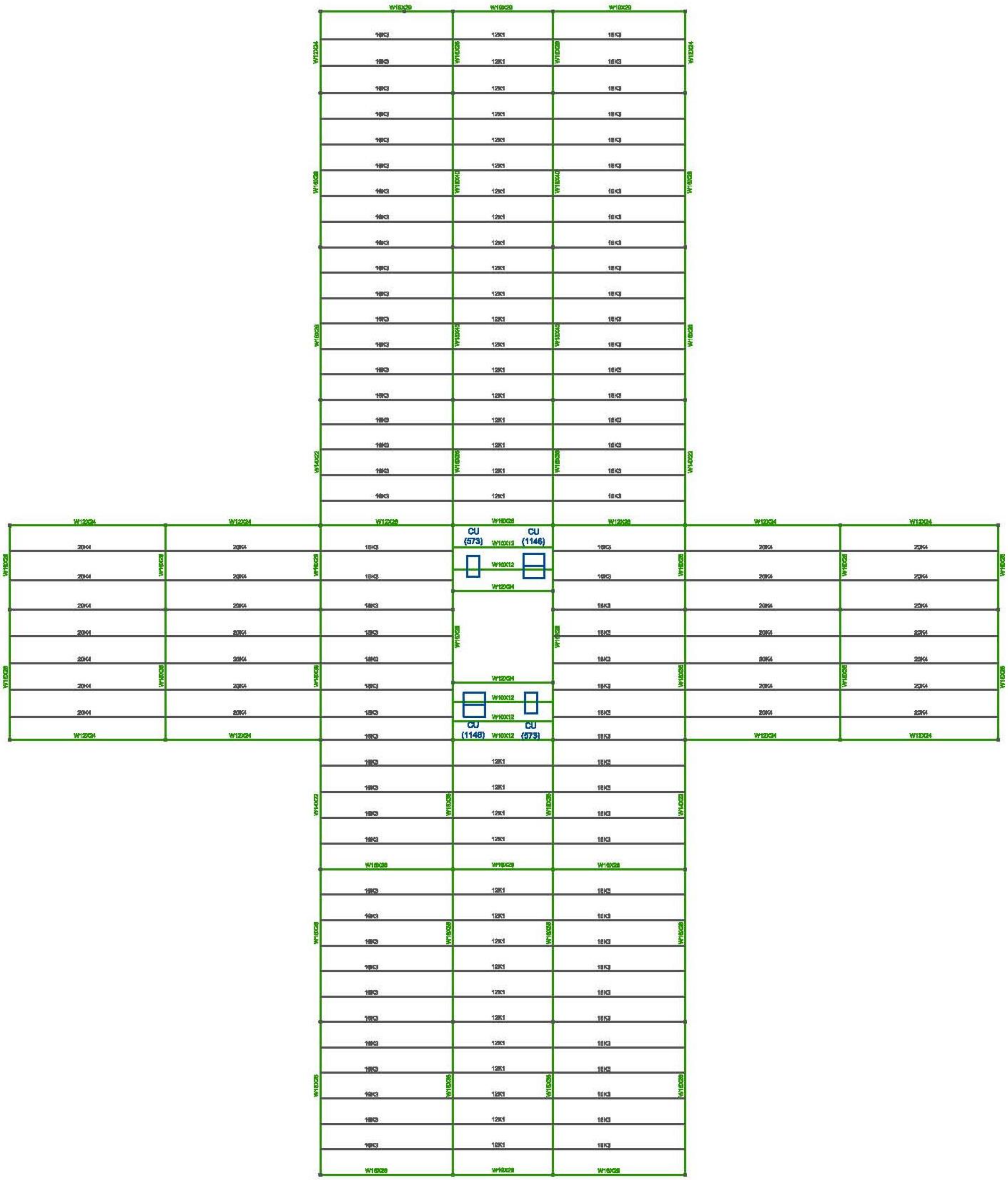
Area 1: Existing Mechanical System



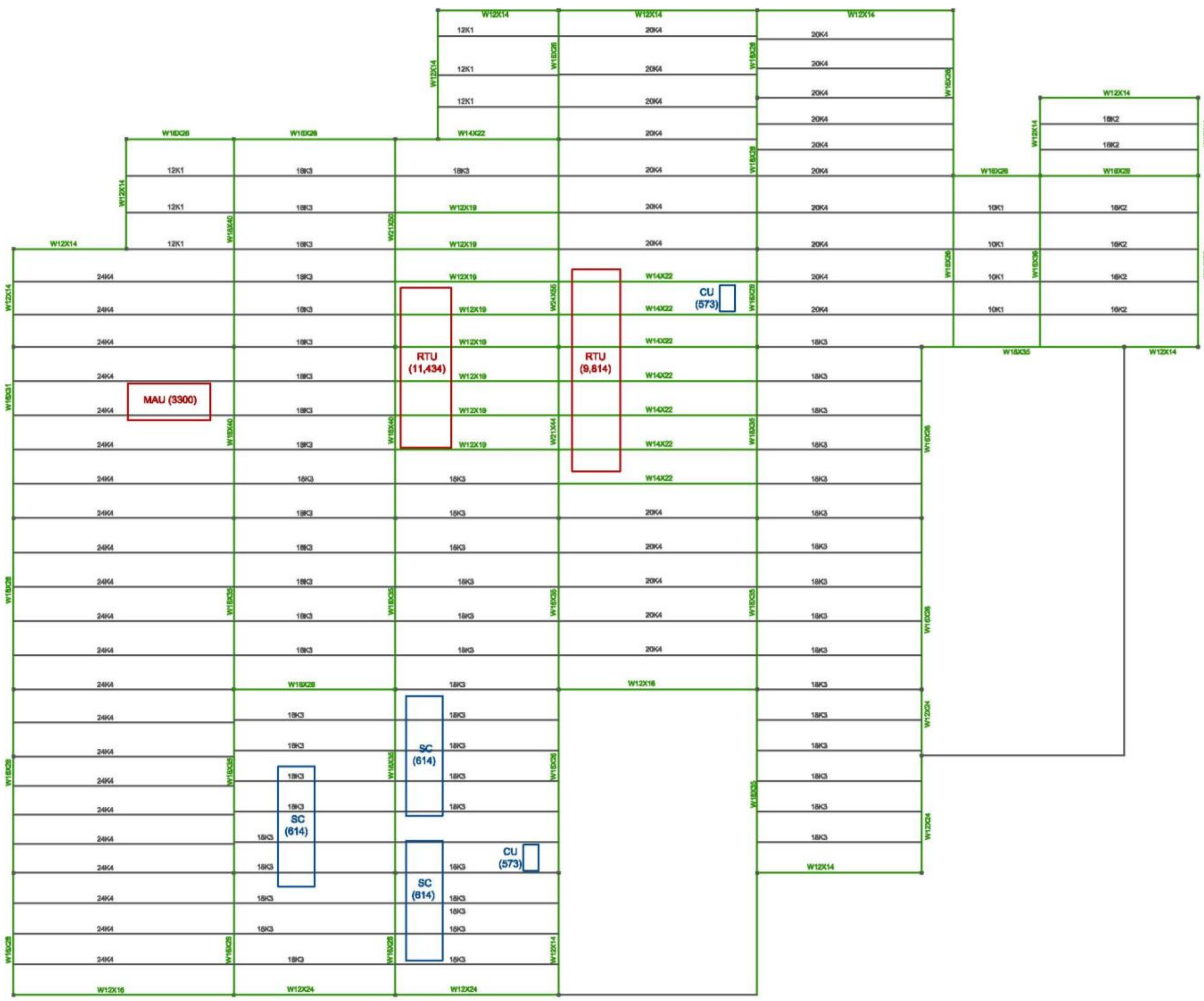
Area 2: Existing Mechanical System



Area 1: VRF Mechanical System



Area 2: VRF Mechanical System



Appendix I: Economic Summary: Comparison of Alternatives

Table I1: Existing System

System:	Existing System		Electricity Cost ($/kWh):	$ 0.04	
Added First Cost:	$ -		Natural Gas Cost ($/therm):	$ 0.96	
	Discount Rate (%):	Electric Use (kWh):	Natural Gas Use (therms):		
	2.1	1512650	3269		
Year	**Annual Maintenance:**	**Electrical Escalation:**	**Natural Gas Escalation:**	**Electricity Cost:**	**Natural Gas Cost:**
1	$ 4,405.00	1.00	1.00	$ 65,044	$ 3,129
2	$ 4,405.00	0.99	0.99	$ 64,394	$ 3,098
3	$ 4,405.00	0.98	0.97	$ 63,743	$ 3,036
4	$ 4,405.00	0.97	0.94	$ 63,093	$ 2,942
5	$ 4,405.00	0.97	0.95	$ 63,093	$ 2,973
6	$ 4,405.00	0.97	0.95	$ 63,093	$ 2,973
7	$ 4,405.00	0.98	0.96	$ 63,743	$ 3,004
8	$ 4,405.00	0.99	0.96	$ 64,394	$ 3,004
9	$ 4,405.00	0.99	0.97	$ 64,394	$ 3,036
10	$ 4,405.00	1.00	0.99	$ 65,044	$ 3,098
11	$ 4,405.00	1.00	1.01	$ 65,044	$ 3,161
12	$ 4,405.00	1.00	1.03	$ 65,044	$ 3,223
13	$ 4,405.00	1.00	1.05	$ 65,044	$ 3,286
14	$ 4,405.00	1.00	1.07	$ 65,044	$ 3,348
15	$ 4,405.00	1.01	1.10	$ 65,694	$ 3,442
16	$ 4,405.00	1.01	1.11	$ 65,694	$ 3,474
17	$ 4,405.00	1.02	1.13	$ 66,345	$ 3,536
18	$ 4,405.00	1.02	1.14	$ 66,345	$ 3,568
19	$ 4,405.00	1.02	1.15	$ 66,345	$ 3,599
20	$ 4,405.00	1.02	1.15	$ 66,345	$ 3,599
Net Present Value:	**$71,337.78**	-	-	**$ 1,048,517**	**$ 51,863**
Total System NPV: $1,171,718					

Table I2: Existing System with Condensing Boilers

System:	Existing with Condensing Boilers		Electricity Cost ($/kWh):	$ 0.04	
Added First Cost:	$ 9,010.00		Natural Gas Cost ($/therm):	$ 0.96	
	Discount Rate (%):	Electric Use (kWh):	Natural Gas Use (therms):		
	2.1	1512650	3172		
Year	**Annual Maintenance:**	**Electrical Escalation:**	**Natural Gas Escalation:**	**Electricity Cost:**	**Natural Gas Cost:**
1	$ 4,405.00	1.00	1.00	$ 65,044	$ 3,037
2	$ 4,405.00	0.99	0.99	$ 64,394	$ 3,006
3	$ 4,405.00	0.98	0.97	$ 63,743	$ 2,945
4	$ 4,405.00	0.97	0.94	$ 63,093	$ 2,854
5	$ 4,405.00	0.97	0.95	$ 63,093	$ 2,885
6	$ 4,405.00	0.97	0.95	$ 63,093	$ 2,885
7	$ 4,405.00	0.98	0.96	$ 63,743	$ 2,915
8	$ 4,405.00	0.99	0.96	$ 64,394	$ 2,915
9	$ 4,405.00	0.99	0.97	$ 64,394	$ 2,945
10	$ 4,405.00	1.00	0.99	$ 65,044	$ 3,006
11	$ 4,405.00	1.00	1.01	$ 65,044	$ 3,067
12	$ 4,405.00	1.00	1.03	$ 65,044	$ 3,128
13	$ 4,405.00	1.00	1.05	$ 65,044	$ 3,188
14	$ 4,405.00	1.00	1.07	$ 65,044	$ 3,249
15	$ 4,405.00	1.01	1.10	$ 65,694	$ 3,340
16	$ 4,405.00	1.01	1.11	$ 65,694	$ 3,371
17	$ 4,405.00	1.02	1.13	$ 66,345	$ 3,431
18	$ 4,405.00	1.02	1.14	$ 66,345	$ 3,462
19	$ 4,405.00	1.02	1.15	$ 66,345	$ 3,492
20	$ 4,405.00	1.02	1.15	$ 66,345	$ 3,492
Net Present Value:	**$71,337.78**	-	-	**$ 1,048,517**	**$ 50,324**
Total System NPV: $1,179,189					
Simple Payback (Years): 97.03					

Table I3: Existing System with Air-Cooled CHW

System:	Existing with Air-Cooled CHW		Electricity Cost ($/kWh):	$ 0.04	
Added First Cost:	$ 414,158.00		Natural Gas Cost ($/therm):	$ 0.96	
	Discount Rate (%):	Electric Use (kWh):	Natural Gas Use (therms):		
	2.1	1449986	3269		
Year	**Annual Maintenance:**	**Electrical Escalation:**	**Natural Gas Escalation:**	**Electricity Cost:**	**Natural Gas Cost:**
1	$ 4,405.00	1.00	1.00	$ 62,349	$ 3,129
2	$ 4,405.00	0.99	0.99	$ 61,726	$ 3,098
3	$ 4,405.00	0.98	0.97	$ 61,102	$ 3,036
4	$ 4,405.00	0.97	0.94	$ 60,479	$ 2,942
5	$ 4,405.00	0.97	0.95	$ 60,479	$ 2,973
6	$ 4,405.00	0.97	0.95	$ 60,479	$ 2,973
7	$ 4,405.00	0.98	0.96	$ 61,102	$ 3,004
8	$ 4,405.00	0.99	0.96	$ 61,726	$ 3,004
9	$ 4,405.00	0.99	0.97	$ 61,726	$ 3,036
10	$ 4,405.00	1.00	0.99	$ 62,349	$ 3,098
11	$ 4,405.00	1.00	1.01	$ 62,349	$ 3,161
12	$ 4,405.00	1.00	1.03	$ 62,349	$ 3,223
13	$ 4,405.00	1.00	1.05	$ 62,349	$ 3,286
14	$ 4,405.00	1.00	1.07	$ 62,349	$ 3,348
15	$ 4,405.00	1.01	1.10	$ 62,973	$ 3,442
16	$ 4,405.00	1.01	1.11	$ 62,973	$ 3,474
17	$ 4,405.00	1.02	1.13	$ 63,596	$ 3,536
18	$ 4,405.00	1.02	1.14	$ 63,596	$ 3,568
19	$ 4,405.00	1.02	1.15	$ 63,596	$ 3,599
20	$ 4,405.00	1.02	1.15	$ 63,596	$ 3,599
Net Present Value:	**$71,337.78**	-	-	**$ 1,005,081**	**$ 51,863**
Total System NPV: $1,542,439					
Simple Payback (Years): 153.70					

Table I4: Existing System with Water-Cooled CHW

System:	Existing with Water-Cooled CHW		Electricity Cost ($/kWh):	$ 0.04	
Added First Cost:	$ 470,498.00		Natural Gas Cost ($/therm):	$ 0.96	
	Discount Rate (%):	Electric Use (kWh):	Natural Gas Use (therms):		
	2.1	1285496	3269		
Year	**Annual Maintenance:**	**Electrical Escalation:**	**Natural Gas Escalation:**	**Electricity Cost:**	**Natural Gas Cost:**
1	$ 4,405.00	1.00	1.00	$ 55,276	$ 3,129
2	$ 4,405.00	0.99	0.99	$ 54,724	$ 3,098
3	$ 4,405.00	0.98	0.97	$ 54,171	$ 3,036
4	$ 4,405.00	0.97	0.94	$ 53,618	$ 2,942
5	$ 4,405.00	0.97	0.95	$ 53,618	$ 2,973
6	$ 4,405.00	0.97	0.95	$ 53,618	$ 2,973
7	$ 4,405.00	0.98	0.96	$ 54,171	$ 3,004
8	$ 4,405.00	0.99	0.96	$ 54,724	$ 3,004
9	$ 4,405.00	0.99	0.97	$ 54,724	$ 3,036
10	$ 4,405.00	1.00	0.99	$ 55,276	$ 3,098
11	$ 4,405.00	1.00	1.01	$ 55,276	$ 3,161
12	$ 4,405.00	1.00	1.03	$ 55,276	$ 3,223
13	$ 4,405.00	1.00	1.05	$ 55,276	$ 3,286
14	$ 4,405.00	1.00	1.07	$ 55,276	$ 3,348
15	$ 4,405.00	1.01	1.10	$ 55,829	$ 3,442
16	$ 4,405.00	1.01	1.11	$ 55,829	$ 3,474
17	$ 4,405.00	1.02	1.13	$ 56,382	$ 3,536
18	$ 4,405.00	1.02	1.14	$ 56,382	$ 3,568
19	$ 4,405.00	1.02	1.15	$ 56,382	$ 3,599
20	$ 4,405.00	1.02	1.15	$ 56,382	$ 3,599
Net Present Value:	**$71,337.78**	-	-	**$ 891,062**	**$ 51,863**
Total System NPV: $1,484,760					
Simple Payback (Years): 48.17					

Table I5: VRF System

System:	VRF System			Electricity Cost ($/kWh):	$ 0.04
Added First Cost:	$ 57,442.00			Natural Gas Cost ($/therm):	$ 0.96
	Discount Rate (%):	Electric Use (kWh):	Natural Gas Use (therms):		
	2.1	1288299	2999		
Year	**Annual Maintenance:**	**Electrical Escalation:**	**Natural Gas Escalation:**	**Electricity Cost:**	**Natural Gas Cost:**
1	$ 6,600.00	1.00	1.00	$ 55,397	$ 2,871
2	$ 6,600.00	0.99	0.99	$ 54,843	$ 2,842
3	$ 6,600.00	0.98	0.97	$ 54,289	$ 2,785
4	$ 6,600.00	0.97	0.94	$ 53,735	$ 2,699
5	$ 6,600.00	0.97	0.95	$ 53,735	$ 2,727
6	$ 6,600.00	0.97	0.95	$ 53,735	$ 2,727
7	$ 6,600.00	0.98	0.96	$ 54,289	$ 2,756
8	$ 6,600.00	0.99	0.96	$ 54,843	$ 2,756
9	$ 6,600.00	0.99	0.97	$ 54,843	$ 2,785
10	$ 6,600.00	1.00	0.99	$ 55,397	$ 2,842
11	$ 6,600.00	1.00	1.01	$ 55,397	$ 2,900
12	$ 6,600.00	1.00	1.03	$ 55,397	$ 2,957
13	$ 6,600.00	1.00	1.05	$ 55,397	$ 3,014
14	$ 6,600.00	1.00	1.07	$ 55,397	$ 3,072
15	$ 6,600.00	1.01	1.10	$ 55,951	$ 3,158
16	$ 6,600.00	1.01	1.11	$ 55,951	$ 3,187
17	$ 6,600.00	1.02	1.13	$ 56,505	$ 3,244
18	$ 6,600.00	1.02	1.14	$ 56,505	$ 3,273
19	$ 6,600.00	1.02	1.15	$ 56,505	$ 3,302
20	$ 6,600.00	1.02	1.15	$ 56,505	$ 3,302
Net Present Value:	**$106,885.21**	-	-	**$ 893,005**	**$ 47,579**
Total System NPV: $1,104,911					
Simple Payback (Years): 5.80					

Table I6: Existing System with DHW Heating Energy Included

System:	Existing with DHW Energy			Electricity Cost ($/kWh):	$ 0.04
Added First Cost:	$ 21,063.00			Natural Gas Cost ($/therm):	$ 0.96
	Discount Rate (%):	Electric Use (kWh):	Natural Gas Use (therms):		
	2.1	1512650	56334		
Year	**Annual Maintenance:**	**Electrical Escalation:**	**Natural Gas Escalation:**	**Electricity Cost:**	**Natural Gas Cost:**
1	$ 6,600.00	1.00	1.00	$ 65,044	$ 53,928
2	$ 6,600.00	0.99	0.99	$ 64,394	$ 53,389
3	$ 6,600.00	0.98	0.97	$ 63,743	$ 52,310
4	$ 6,600.00	0.97	0.94	$ 63,093	$ 50,692
5	$ 6,600.00	0.97	0.95	$ 63,093	$ 51,232
6	$ 6,600.00	0.97	0.95	$ 63,093	$ 51,232
7	$ 6,600.00	0.98	0.96	$ 63,743	$ 51,771
8	$ 6,600.00	0.99	0.96	$ 64,394	$ 51,771
9	$ 6,600.00	0.99	0.97	$ 64,394	$ 52,310
10	$ 6,600.00	1.00	0.99	$ 65,044	$ 53,389
11	$ 6,600.00	1.00	1.01	$ 65,044	$ 54,467
12	$ 6,600.00	1.00	1.03	$ 65,044	$ 55,546
13	$ 6,600.00	1.00	1.05	$ 65,044	$ 56,624
14	$ 6,600.00	1.00	1.07	$ 65,044	$ 57,703
15	$ 6,600.00	1.01	1.10	$ 65,694	$ 59,321
16	$ 6,600.00	1.01	1.11	$ 65,694	$ 59,860
17	$ 6,600.00	1.02	1.13	$ 66,345	$ 60,939
18	$ 6,600.00	1.02	1.14	$ 66,345	$ 61,478
19	$ 6,600.00	1.02	1.15	$ 66,345	$ 62,017
20	$ 6,600.00	1.02	1.15	$ 66,345	$ 62,017
Net Present Value:	**$106,885.21**	-	-	**$ 1,048,517**	**$ 893,732**
Total System NPV: $2,070,197					

Table I7: Solar Thermal System

System:	Solar Thermal System			Electricity Cost ($/kWh):	$ 0.04
Added First Cost:	$ 61,906.00			Natural Gas Cost ($/therm):	$ 0.96
	Discount Rate (%):	Electric Use (kWh):	Natural Gas Use (therms):		
	2.1	1512650	24145		
Year	**Annual Maintenance:**	**Electrical Escalation:**	**Natural Gas Escalation:**	**Electricity Cost:**	**Natural Gas Cost:**
1	$ 15,400.00	1.00	1.00	$ 65,044	$ 23,114
2	$ 15,400.00	0.99	0.99	$ 64,394	$ 22,883
3	$ 15,400.00	0.98	0.97	$ 63,743	$ 22,421
4	$ 15,400.00	0.97	0.94	$ 63,093	$ 21,728
5	$ 15,400.00	0.97	0.95	$ 63,093	$ 21,959
6	$ 15,400.00	0.97	0.95	$ 63,093	$ 21,959
7	$ 15,400.00	0.98	0.96	$ 63,743	$ 22,190
8	$ 15,400.00	0.99	0.96	$ 64,394	$ 22,190
9	$ 15,400.00	0.99	0.97	$ 64,394	$ 22,421
10	$ 15,400.00	1.00	0.99	$ 65,044	$ 22,883
11	$ 15,400.00	1.00	1.01	$ 65,044	$ 23,346
12	$ 15,400.00	1.00	1.03	$ 65,044	$ 23,808
13	$ 15,400.00	1.00	1.05	$ 65,044	$ 24,270
14	$ 15,400.00	1.00	1.07	$ 65,044	$ 24,732
15	$ 15,400.00	1.01	1.10	$ 65,694	$ 25,426
16	$ 15,400.00	1.01	1.11	$ 65,694	$ 25,657
17	$ 15,400.00	1.02	1.13	$ 66,345	$ 26,119
18	$ 15,400.00	1.02	1.14	$ 66,345	$ 26,350
19	$ 15,400.00	1.02	1.15	$ 66,345	$ 26,582
20	$ 15,400.00	1.02	1.15	$ 66,345	$ 26,582
Net Present Value:	**$249,398.83**	-	-	**$ 1,048,517**	**$ 383,067**
Total System NPV: $1,742,889					
Simple Payback (Years): 2.01					