AE 482: Architectural Engineering Senior Project
The Pennsylvania State University
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Design and Analysis of LA Fitness Mechanical System
West Oaks Location
Houston, TX

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Mechanical Option
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Executive Summary:

The West Oaks location of LA Fitness is a 45,000 sf exercise facility located in Houston, Texas. The focus of this report is to critically analyze the mechanical systems of that building and to implement the best system design. The best system design will be selected as such based on these three design criteria: low energy consumption, reduced emissions, and economic feasibility.

Air System Design:

The building's original design documents suggest that the best way to deliver air to the building is through the use of single zone packaged rooftop units. These are constant volume units, and they are fueled by natural gas. It is quite difficult to find a less expensive system to meet the cooling loads than the one initially proposed. The results from the different unit configurations are shown in Table 0.1 below. The results show that the best way to meet the design criteria will be a modification of the original rooftop units instead of implementing a hydronic alternative.

Criteria	Energy	Economics		Emissions			
Configuration	Consumption (kWh)	First Cost	Annual Cost	Particulates	SO_2	Nox	CO_2
RTU with Desiccant Modifications	935,053	$563,662	$46,771	183	1935	2178	883154
Packaged RTU Design	1,255,859	$419,000	$62,818	166	1937	2589	1099593
Water-Cooled Chiller	843,084	$433,345	$65,761	546	6347	3682	1068328
Air-Cooled Chiller	1,018,379	$427,990	$79,434	625	7266	4215	1222938

*RTU = Original Rooftop Unit Configuration
**Emission Data is in lbm/yr

Table 0.1 – Air Side System Comparisons

In the modified rooftop unit design, all of the outdoor air for the building is sent to one unit where it is preconditioned by a desiccant wheel and a sensible wheel working in tandem. The dehumidified air is then ducted to the rooftop unit's outdoor air intakes where it is mixed with return air and sent to the direct expansion coils for cooling and possibly more dehumidification.

Hot Water Design:

Three different types of solar collectors were studied to see if the existing water heaters' natural gas usage could be reduced with the aid of solar energy. A glazed flat plate solar collector proved to be quite effective at reducing gas usage. This option has an attractive payback period for the site, and it is included in the final design.

Technology	Model	Energy Delivered Per Year (MMBtu)	% Demand Per Year	First Cost	Payback Period (Years)
Glazed Flat Plate Collector	Heliodyne Gobi 408	33.94	53.6%	$5,589	5.3

Table 0.2 – Solar Collector Used in Final Design

Acknowledgements

I would like to take this opportunity to thank some of the many people who have offered me their knowledge and support on this journey.

I would like to give special thanks to my parents, Barbara and Vito, and my sister Cheri for always being there for me and supporting my decisions through the years.

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LA Fitness, West Oaks

8906 Highway 6 South
Houston, TX

http://www.arche.psu.edu/thesis/eportfolio/current/portfolios/djm364

David Melfi
Mechanical Option

Architecture
- Two story, 45,000 ft^2 fitness center
- Large double height cardio space
- Mezzanine with exercise equipment overlooking cardio space
- Spaces have distinct different functions such as basketball, strength training, swimming, child care, racquetball, and aerobics

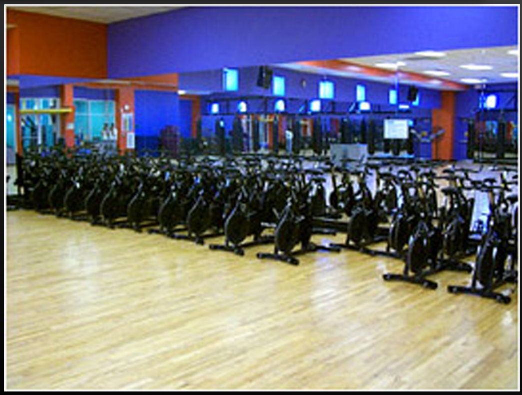
Primary Project Team
- Owner: LA Fitness International, LLC
- Occupant: LA Fitness
- Architect: Heights Venture Architects
- MEP Engineers: Advanced Technologies, Inc.
- Structural Engineers: BGA Engineers, Inc.
- Civil Engineers: Cobb Fendley & Associates
- General Contractor: Ridgemont Construction
- Interior Designers: Senninger Walker Architects
- Construction Management: LA Fitness

Structural
- Composite floor construction
- 4-1/2" normal weight concrete slabs over 20 gauge composite steel deck supported by steel beams
- Tilt-up construction envelope made of 8" thick normal weight concrete
- Wood truss system on roof supported by metal plates

Mechanical
- Mechanical load served by (13) packaged rooftop units
- Energy source for units is natural gas
- Energy recovery wheel used in pool zone
- Pool area is a critical space regarding indoor air quality and condensation control due to Houston's extremely high levels of humidity

Electrical/Lighting
- Service utility transformed to 277/480V outside the building
- (4) 600kCMIL conduits run in from underground to serve building
- One main electrical room with (3) 120/208V panels and (2) 277/480V panels
- 2 additional 120/208V panels serve pool and juice bar areas
- Emergency lighting provided by battery backup wall sconces
- Interior lighting is primarily 2x4 fluorescent fixtures
- Exterior façade has wall-mounted metal halide lighting
- Parking lot served by pole-mounted HID floodlights

Construction
- Project cost: \$4.5 million
- Start Date: 5/9/05
- Turnover Date: 12/9/05
- Delivery: Design-Bid-Build
- Tilt-up construction used on exterior façade for quicker delivery



1.0 Introduction

1.1 Scope

This report represents the culmination of a student designer's five year professional education in the field of Architectural Engineering. The focus of this study lies in the design of an energy efficient mechanical system. However, other building systems have been affected by the design and they have been addressed and analyzed as well.

The building analyzed in this study is a 45,000 sf LA Fitness exercise facility located in Houston, Texas. Design documents for this building were obtained, studied, and critically analyzed over an eight month time period.

It should be noted that the findings of this report are specific to the initial constraints of building size, function, and location among other factors. It is not the intent of the designer to generalize the findings of this report to other building projects without a similar thorough analysis being conducted.

1.2 Design Objectives

The designer set out to achieve four goals over the course of the study. The three design goals to be achieved were energy conservation, cost reduction, and reduced emissions. The final goal that was set was to gain practical experience designing mechanical systems.

Energy conservation is absolutely critical. The United States Energy Information Administration's (EIA) most recent report shows that approximately 72% of the energy in the country goes to buildings as an end-user.

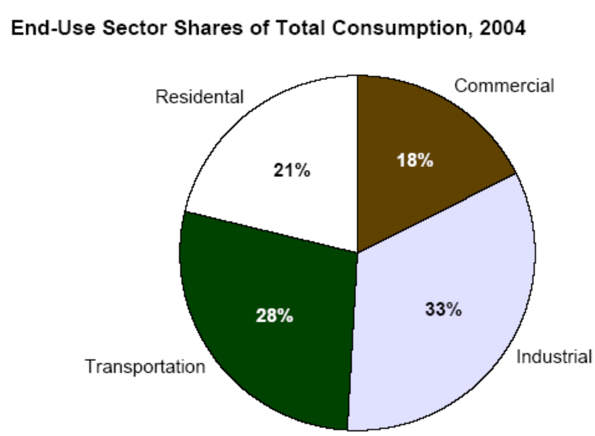


Figure 1.1 – 2004 EIA Reported Energy Consumption by Sector

With this fact in mind, finding and implementing more energy efficient systems should be a major focus of any good mechanical design. If the reduction of fossil fuel sources and the increase of energy consumption are not motivators to drive energy conservation, the resulting price increases will be.

Cost drives decisions. It is this simple unavoidable fact that keeps designers searching for lower cost design alternatives. An economic analysis is included for each of the systems being compared. An effort was made to lower the life cycle cost of the design. For many owners, first cost is the biggest factor. With this in mind, a special consideration was granted for systems with low first cost systems or attractive payback periods.

Another goal of the design is to make the building more environmentally friendly by the means of reducing harmful building emissions. Chemicals like CO_2, NO_x, and SO_x are harmful to the environment. These chemicals are responsible for air pollution and global warming among many other unpleasant topics. The final design will attempt to reduce building emissions.

The designer's final goal is to learn more about the practical application of conceptual design. It is the nature of classes to present a number of isolated concepts. Unfortunately, there is typically not enough time in a given course to focus on what it takes to fully integrate these conceptual designs into practical use. This analysis represents an opportunity to perform that integration.

1.3 Referenced Standards

The American Society of Heating Refrigeration and Air-Conditioning Engineers (ASHRAE) publishes a number of standards for better building design. The standards used in this design are described below.

ASHRAE Standard 62.1 Addendum n

This is the standard that is used to regulate how much ventilation air is required for buildings. A thorough analysis of this standard for the building is included in Section 3.1 of this report.

ASHRAE Standard 90.1

This standard is published with the intent of providing minimum requirements for the energy-efficient design of buildings. The analysis provided contains specific discussions of Section 5 (Building Envelope) and a redesign for Section 9 (Lighting) of the standard.

1.4 Methodology

In any engineering analysis, there must be special attention devoted to the standards of measurement. It is the purpose of this section to discuss the metrics by which the design alternatives that will be presented will be weighed.

Energy Savings

The final design selected will have to be a more energy efficient design than the one proposed in the design documents. The alternatives presented will be analyzed for their performance over a one year time span via computer simulation on Trane's energy modeling program TRACE. The amount of energy necessary for one year of operation will be one of the selection criteria.

Emissions:

Related very closely to energy savings is the issue of emissions. LA Fitness is located in Texas which is not connected to the power grid for the rest of the United States. Texas energy production can be broken down into three categories: coal, natural gas, and nuclear. Shown below is a breakdown of the types of emissions produced from the energy being considered at the site.

Texas Grid lbm/year				
Fuel	**Particulates/kWh**	**SO_2/kWh**	**NO_x/kWh**	**CO_2/kWh**
Coal	4.51E-04	5.24E-03	3.04E-03	8.82E-01
Natural Gas	0	6.21E-06	1.17E-03	6.17E-01
Totals	4.51E-04	5.25E-03	4.21E-03	1.50E+00

Table 1.1 - Texas Emissions Rates/kWh

Reduced yearly emissions will also serve as a critical component of the final selection criteria.

Economics

The final contributing component to the selection criteria will be a cost analysis. The options that were selected will be analyzed on either the basis of first cost, life cycle cost, or payback period depending on which is most appropriate.

2.0 Building Description

2.1 Client Overview

Client: LA Fitness International, LLC

LA Fitness International, LLC is renowned for establishing high quality exercise facilities across the United States. The company was established in 1984 in Southern California. LA Fitness currently owns and operates a total of 135 sports clubs in California, Arizona, Florida, Georgia, Pennsylvania, New Jersey, New York, Connecticut, Texas, and Washington. The company takes great care to ensure that their facilities are on the cutting edge of fitness.

The West Oaks location in Houston Texas is 45,000 square foot, two story facility. Each LA Fitness location offers different activities for their users to enjoy. This location of LA Fitness has diverse spaces including cardio rooms, basketball and racquetball courts, offices, juice bars, a child care center, locker rooms, and an indoor pool.

LA Fitness states on their website (http://www.lafitness.com) that although each of their clubs has its own distinct personality, the company strives to have a common thread or elicit a feeling of familiarity between each location's indoor environment.

2.2 Building Statistics

Building Name:	LA Fitness, West Oaks
Address:	8906 Highway 6 South Houston, TX 77083
Owner/Occupant:	LA Fitness International, LLC
Function:	Sports club and exercise facility
Size:	2 stories; 45,000 ft^2

Primary Project Teams:

General Contractor:	Ridgemont Construction
Construction Manager:	LA Fitness International, LLC
Architects:	Heights Venture Architects, LLP
MEP Engineers:	Advanced Technologies, Inc.
Structural Engineers:	BGA Engineers
Civil Engineers:	Cobb Fendley & Associates
Interior Designers:	Senninger Walker Architects

Dates of Construction:	Start: 5/9/05 Finish: 12/9/05
Cost of Building:	$4.5 Million for building project (cost does not include 5 acres of land)
Project Delivery Method:	Design-bid-build

2.3 Architectural Design

Heights Venture Architects designed this building to serve as a temple to fitness. There is much diversity as to the function of the interior spaces. A goal of the designers was to make the spaces inside feel open and comfortable. This was primarily accomplished by leaving the middle of the first floor open to a 25 foot ceiling and designing a U-shaped mezzanine looking down to the first floor. The exercise equipment on the main level dictates the flow and paths of motion through the double-heighted space.

The front façade is brick veneer. The exterior walls make use of tilt-up construction with minimal 4" stud walls furred out on the inside of the tilt. These walls have an insulation value of R-13. A structural steel system supports the tilt wall. The windows are 1" tinted insulated glass. There is a built up roof with an overall R-22 insulation value.

2.4 Original Mechanical System Description

The LA Fitness mechanical system is unique because of the variety of spaces that it serves. In this design, the ventilation and conditioned air for the building is handled by thirteen direct expansion (DX) packaged rooftop units. These are natural gas fired units. Eleven of these units serve single zones. This design also allows for less ductwork because many of the rooftop units are located right above the zone that the unit serves. The water side load for the building is mainly handled by three natural gas fired water heaters. Water from these heaters is provided at 120^{0}F. There is also one small electric water heater to serve the juice bar area.

2.5 Other Building Systems of Interest

Structural:
The structural system makes use of steel as the primary means of handling the load. This LA Fitness has a composite floor construction. The floor consists of 4-1/2" normal weight concrete slabs over 20 gauge composite steel deck which is all supported by steel beams. The envelope walls use a tilt-up construction system that is made of 8" thick normal weight concrete. There is a flat roof that uses long span steel joists for its support.

Electrical:
The service utility is transformed down to 277/480 V outside of the building. (4) 600 kcmil conduits are run underground and pulled into an area on the exterior façade of the building. There is one main electrical room which serves the majority of the panels for the building. In this room the high voltage panels can be served directly. There is also a transformer in this room that takes the power down to 120/208 V to serve the remaining panels. Outside of the electrical room, there are panelboards located at the Juice Bar and the pool room to better serve these areas.

Lighting:
There are four types of lamps used in this building: fluorescent, metal halide, ceramic metal halide, and LED. Most of the interior lighting is served by 4-lamp fluorescent fixtures that hang in 2' x 4' recessed troffers. The lighting power density of the existing design is 1.2 W/ft^{2}.

3.0 Design Parameters

There are certain parameters that need to be addressed in mechanical design. These parameters include the heating and cooling loads for the building, the required ventilation rates, and a few other factors inherent to the building and location. The following sections include the parameters that the final design must meet, regardless of which system is selected.

3.1 Ventilation

AHSRAE Standard 62.1-2004 governs the minimum ventilation air rate requirements for buildings. The spaces inside of LA Fitness are very diverse in both function and size. For this reason, a thorough analysis of ventilation rates for this building had to be performed.

Space	Area (ft^2)	Design Occupancy
Aerobics	3083	61.7
Racquetball	835	4
Racquetball	835	4
Racquetball	835	4
Racquetball	835	4
Racquetball	835	4
Child Restrooms	148	1.2
Storage	228	0
Kid's Club	1840	36.6
Free Weights	2974	59.5
Basketball	3810	20
Storage	460	8.8
Sp. Exercise	1366	27.3
Equipment Room	147	0.5
Cardiovascular	10520	210.4
Mezzanine	3000	60
Trainer's Office	217	2.7
Spinning	1141	22.8
Pool Equipment	290	1
Pool & Spa	4112	82.3
Locker Rooms	4125	8.8
Reception	1420	14
Membership Sales	687	8
Juice Bar	280	2.6

Table 3.1 Space Function, Area, and Occupancy

The ventilation rate procedure was utilized to find the required minimum outdoor air for each of the thirteen rooftop units. It is assumed that air mixes perfectly in these calculations. The total mixed air supply for the original design is 84,000 cfm that contains 19.4% outdoor air for ventilation purposes. The results from the Standard 62.1 calculation show that the building requires 20.9% outside air if the same total supply cfm is used. While these numbers are approximately equal, they only represent the total percentage requirements for the building. When each rooftop unit is examined, it can be seen that only 3 of the 11 units (RTU-1, 4, and 5) have enough ventilation air for the zones that they each serve.

	V_{oz}	V_{ot}	Actual Design OA	Total Airflow	Design %OA	62.1n %OA
RTU-1	154	154	500	5000	10.0	3.1
RTU-2	950	1188	700	5000	14.0	23.8
RTU-3	3620	3620	3500	10500	33.3	34.5
RTU-4	807	807	3350	8300	40.4	9.7
RTU-5	629	629	750	7500	10.0	8.4
RTU-6	1368	1368	750	6000	12.5	22.8
RTU-7	1419	1419	1000	10000	10.0	14.2
RTU-8	651	651	500	4000	12.5	16.3
RTU-9	2420	2420	1675	6600	25.4	36.7
RTU-10	2420	2420	1675	6600	25.4	36.7
RTU-11	944	1049	750	5500	13.6	19.1
RTU-12	524.5	524.5	500	3500	14.3	15.0
RTU-13	1380	1380	750	5900	12.7	23.4
Entire Building	17286	17628	16400	84400	19.4	20.9

Table 3.2 – Comparison of 62.1 Required OA and Actual Design OA for LA Fitness

These findings show that the new design will need to provide adequate ventilation air to each zone that is being served on a space by space method instead of looking at the entire building totals.

3.2 Heating and Cooling Loads

Houston, TX is a relatively hot location, especially on design days. The 2005 ASHRAE Handbook of Fundamentals was used to find the design day temperature conditions.

	Cooling		Heating
	Dry Bulb	Wet Bulb	Dry Bulb
Fort Bend County, Houston, TX	96.9	80.1	27.7

Table 3.3 – 0.4% and 99.6% Design Conditions from ASHRAE Handbook

The design documents from the original design provided the heating and cooling loads that were found for each of the thirteen rooftop units. Those loads can be seen below in Table 3.3. These loads give an accurate “ballpark figure” as to what type of cooling demand needs to be met at the site. Most of the space loads have a very high latent percentage. This is a result of the activity that takes place inside of the building; besides having an indoor pool and a locker room, the facility houses other high latent activities such as basketball, racquetball, aerobics, and other cardiovascular exercises that cause people to breathe heavily and also to produce sweat.

General Description			Fan Section		Heating Section		Cooling Section		
Unit	Area Served	Energy Used (MBh)	Total Air (cfm)	Outside Air (cfm)	Output (MBh)	Efficiency (%)	Sensible (MBh)	Total (MBh)	Temperature Leaving Unit (F)
RTU-1	Reception	250	5000	500	203	81	119.5	162.0	58.6
RTU-2	Kid's Club	250	5000	700	203	81	102.0	145.8	58.6
RTU-3	Pool	500	10500	3500	400	80	223.3	302.4	58.0
RTU-4	Lockers	350	8300	3350	284	81	136.0	168.1	60.5
RTU-5	Basketball	250	7500	750	203	81	159.8	226.5	58.3
RTU-6	Free Weights	250	6000	750	203	81	133.8	199.2	55.2
RTU-7	Aerobics	500	10000	1000	400	80	220.3	319.5	56.5
RTU-8	Racquetball	150	4000	500	122	81	79.0	118.6	57.6
RTU-9	Cardio	400	6600	1675	324	81	161.5	250.1	56.9
RTU-10	Cardio	400	6600	1675	324	81	161.5	250.1	56.9
RTU-11	Lower Stairs	250	5500	750	203	81	120.0	172.0	57.1
RTU-12	Spinning	150	3500	500	122	81	80.8	117.8	57.7
RTU-13	Mezzanine	250	5900	750	203	81	133.8	199.2	55.2

Table 3.4 – Original Design Rooftop Unit Abridged Schedule (See Appendix A for Full Schedule)

Using Trane’s energy simulation calculator, TRACE, a model was set up to simulate the thermal loads on the building. The model was updated to include the correct ventilation rates that will meet ASHRAE Standard 62.1-2004. Also, other factors were accounted for such as the sensible heat gain from equipment such as treadmills and exercise bikes. The treadmills were simulated to each have a sensible heat output of 590 Btu/hr. This estimation was found in the 2005 ASHRAE Handbook of Fundamentals.

Zone:	Equipment Load	Exercise Loads/Person	
	Sensible	Sensible	Latent
Cardio	8910	350	500
Racquetball	0	350	500
Basketball	0	710	1090
Free Weights	0	350	500
Aerobics	0	710	1090
Spinning	6138	710	1090
Mezzanine	5940	710	1090

Table 3.5 – Modified 2005 ASHRAE Fundamentals Heat Gains for Spaces

The results of this more detailed model showed that the rooftop units may have been oversized beyond standard safety factors. The thought process that preceded the over-sizing of the units is discussed at greater length in Section 8.0 of this report.

3.3 Critical Zones

The indoor pool and the locker room need to be evaluated with special care. The pool area needs to be kept at a lower pressure than the adjacent zones in the final design. The rationale behind this decision is to ensure that there is not humid air being transferred into the reception/lobby area that is located next to the pool. If such a transfer were to occur there would be issues of mold, odor, and an increase of latent load on the zone. The locker room will also need to be designed slightly negative compared to the adjacent spinning and exercise rooms for the same reasons.

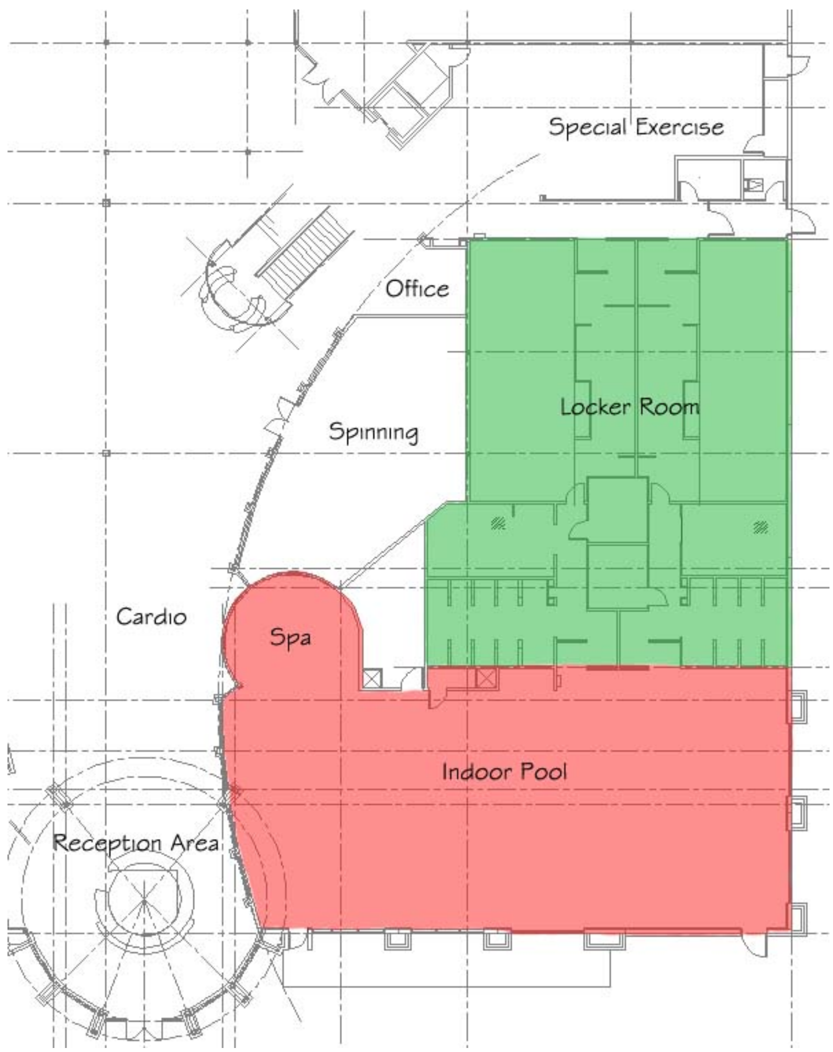


Figure 3.1 – Colored Zones are Critical Spaces for Pressure Balance

3.3.1 Indoor Pool Design

Maintaining adequate humidity levels and dry bulb temperatures to indoor pools presents a unique design challenge. Design loads calculations for natatoriums have to take into consideration all of the normal load parameters such as envelope, lighting, and outdoor air, as well as the loads from occupants and the very high latent load associated with the evaporation of water. ASHRAE has found that swimmers are most comfortable with dry bulb temperatures ranging from 78 to 85 ^{0}F and relative humidity levels ranging from 50% to 60%. With this in mind, the original design supplied air at 80^{0}F and 55% relative humidity. The pool water is maintained at 82^{0}F; a comfortable temperature adequate for recreational or competitive use. At these conditions, the evaporation rate of water was found to be 102.9 lb/hr. This evaporation rate correlates to a latent load of 102,938 Btu/hr. The equation and values used for this calculation can be found in Appendix B. The additional latent load from the pool will considerably lower the sensible heat ratio for this space and will need to be addressed in any good design. The 2003 ASHRAE Handbook of HVAC Applications suggests that there be an air change rate of four to six air changes per hour for this type of space.

3.4 Building Envelope

Section 5 of ASHRAE Standard 90.1-2004 was established to ensure that buildings are not wasting energy through poor building envelope design. The standard provides minimum insulation values for the walls, floors, and roof, and these values or designated by climate zones. The standard also addresses the issue of high solar load through glazing by providing minimum U-values and a solar heat gain coefficient to ensure that there is not too much heat gain from the sun.

The opaque areas of the building were analyzed using the prescriptive building envelope option. In the design documents, the architect calls for a roof assembly that was calculated to have an R-24 insulation value. This assembly exceeds the R-15 minimum that the standard requires. The floor system in place has an R-value of 22 which will satisfy the R-19 requirement between floors.

Item	Description	Insulation Min.
		R-Value
Roofs	Insulation Entirely above Deck	R-15.0
Mass Walls	8” Tilt-wall construction with 2” insulation	Not Required
Floors	Steel Joist	R-19.0
Slab-On-Grade Floors	Unheated	-
Opaque Doors	Swinging	-

Table 3.6 – Opaque Building Envelop Compliance

There are two major factors used when evaluating fenestration: the U-value and the solar heat gain coefficient. The windows used in this design have a U-value of 0.95 Btu/h-ft^2-^{o}F, which complies because it is lower than the standard’s maximum U-value of 1.22. The standard also requires that the total vertical fenestration area is to be less than 50% of the gross wall area. Table 3.7 below shows that this construction far exceeds those criteria as well. The windows had a solar heat gain coefficient of 0.23; this value, while close to the limit, does comply with the standard.

Fenestration	**Operable/Fixed**	**% Glazing**	**Assembly Max. U**	**SHGC**
North	All Fixed	6.96	1.22	0.61
South	All Fixed	6.96	1.22	0.25
East	All Fixed	3.86	1.22	0.25
West	All Fixed	24.9	1.22	0.25

Table 3.7 – ASHRAE Standard 90.1-2004 Fenestration Requirements

4.0 Air Side Alternatives

A number of alternatives had to be compared to find the best system to serve the air loads for the building. The alternatives presented below offer a number of different approaches that were all considered at some point during the design phase of the final system. This section will first discuss each alternative or idea presented on a conceptual basis. Then the idea will be critically weighed by the three metrics set as the selection criteria: energy consumption, environmental impact, and economics.

4.1 Original Rooftop Unit Design

The originally proposed design for this project used natural gas fired packaged rooftop units. The 13 units use direct expansion to meet the cooling load for the building. It has been said that there is no beating this equipment when it comes to first cost. The first cost for the 13 units is $247,000 bringing the initial cost of the airside to $9.31/sf. The performance data and dimensions for these units can be found in Appendix A.

Equipment	$247,000.00
Fans & Grills	$40,000.00
Controls	$14,000.00
Ductwork	$54,000.00
T & B	$13,000.00
Sales Tax	$29,000.00
Misc	$22,000.00
Total	$419,000.00
Total/sf	$9.31

Table 4.1 – Original Mechanical System First Cost

The total energy consumption for this rooftop unit design is 1,624,031 kWh for one year of operation. Of that energy, 368,172 kWh was electric energy that was consumed by the lighting system. The remaining 1,255,859 kWh represents the amount of energy being consumed by the natural gas fired rooftop units. The table below shows the energy and cost breakdown for the current electric and natural gas rates at the site.

End-Use	Energy Consumption	Unit of Energy	Cost/Unit	Enery Cost/Year	First Cost of System
Lighting	368172	kWh	$0.078	$28,717	N/A
HVAC	4285.0	MMBtu	$14.660	$62,818	$419,000

Table 4.2 – Original Rooftop Unit Design Cost Breakdown

The next factor to consider in this design is the environmental impact of using this system. Shown below are the environmental emissions that will be produced by using these fuels at this site.

Building Emissions lbm				
Fuel	**Particulates/yr**	**SO_2/yr**	**NO_x/yr**	**CO_2/yr**
Coal	166	1929	1119	324728
Natural Gas	0	8	1469	774865
Totals	166	1937	2589	1099593

Table 4.3 – Original Rooftop Design Emissions/yr

4.2 Hydronic System

The focus of this section is to consider the option of using a chilled water system to serve the cooling loads for the building. Both water-cooled and air-cooled chillers were initially explored for this analysis as they both presented benefits. It is assumed that the primary air handling units for these scenarios will be variable air volume (VAV) units. In addition, one constant volume unit will serve the four zones on the north side of the building where the pressure balances become a delicate issue because of the indoor pool and the locker room.

4.2.1 Compressor Selection

The first decision to make regarding the chiller alternative was which compressor would best serve the load. From the initial analysis and provided documents, it can be assumed that the cooling demand is approximately 200 tons for the building. This led to the decision to use either a rotary screw or a reciprocating compressor for the chiller. CoolTools Design Guide offered an excellent comparison of these compressors and their typical loads and first costs as seen in Table 4.4 below.

Compressor Type	**Range (tons)**	**First Cost Range ($/ton)**
Reciprocating	50-230	200-250
Screw	70-400	225-275
Centrifugal	200-2000	180-300
Single-Effect Absorption	100-1700	300-450
Double-Effect Absorption	100-1700	300-550
Engine Driven	100-3000	450-600

Table 4.4 – Compressor Performance and Cost from CoolTools Design Guide

A screw compressor was chosen over a reciprocating one for this design based on the size of the load served, ease of maintenance, more efficient part load performance, and comparable price. Reciprocating compressors in chillers have more moving parts and may need rebuilds. Screw compressors have higher part load efficiency and smoother loading than reciprocating engines. In a screw compressor, there is a sliding valve that modulates where refrigerant is introduced to the screw as opposed to a reciprocating compressor where part load performance is achieved by turning pistons on and off.

4.2.2 Water-Cooled Chiller

This alternative includes a rotary screw chiller for refrigerant compression and a cooling tower to reject heat to the outdoors. The chiller used for this energy simulation is a helical rotary chiller made by Trane. The actual chiller selected for analysis is the RTHD model with 200 tons of cooling capacity. The chiller was modeled to perform at 0.66 kW/ton. The cooling tower was then selected for the application. The condenser water flow from the chiller and the outdoor air wet bulb temperatures were used to select a Marley cooling tower for the system. The condenser water flow is 935 gpm for this example. The cut sheets for the chiller and the cooling tower used in this configuration can be found in Appendix C.

This alternative is very conventional and energy efficient, but unfortunately it requires more maintenance and more equipment than the initial design. LA Fitness is a relatively small building; as a retail client, the owners may not be interested in a system that will require more maintenance than it has to have if other alternatives are available. In addition to the extra maintenance of the cooling tower itself, there is also the need to treat the water for the tower, and maintain the associated pumps.

First cost and annual energy data were gathered for this chiller and they are shown below in Table 4.5. The yearly energy cost for this alternative is $2942 more than that of the original design, and the first cost is $14,345 higher. It should be noted that this option does consume 412,777 kWh less on site energy for one year of operation. However, the annual energy savings for this scenario did not translate into operation cost savings because this chiller plant is driven by electricity and the original alternative is driven by natural gas. This difference in fuel rates is an important factor for this location.

End-Use	Energy Consumption	Unit of Energy	Cost/Unit	Enery Cost/Year	First Cost of System
HVAC	843084.0	kWh	$0.078	$65,761	$433,345
Original Design	4285.0	MMBtu	$14.660	$62,818	$419,000
Differential	-412777.7	kWh	N/A	$2,942	$14,345

Table 4.5 – Water-cooled Chiller Associated Energy & Cost Breakdown

The resulting emissions from this alternative (Table 4.6) show that the water-cooled chiller is more harmful to the environment than the original design in every pollutant category discussed excepting CO_2. The reasoning behind this increase of emissions stems from the use of electricity to run the chillers.

Building Emissions lbm				
Fuel	**Particulates/yr**	**SO_2/yr**	**NO_x/yr**	**CO_2/yr**
Coal	546	6347	3682	1068328
Natural Gas	0	0	0	0
Totals	546	6347	3682	1068328
Differential	380	4410	1094	-31267

Table 4.6 – Water-cooled Chiller Associated Emissions/yr

4.2.3 Air-Cooled Chiller

Another alternative for heat rejection is the use of an air-cooled chiller. This option is being evaluated because there is less associated maintenance and first cost for this configuration of equipment. It should first be noted that air-cooled chillers themselves are less efficient than water-cooled chillers. However, air-cooled chillers do have relatively good part load performance. CoolTools Design Guide for chillers reports that as ambient air temperature decreases, the COP of an air-cooled chiller improves considerably; these improvements are only relative to the same chiller's full load performance.

The air-cooled chiller used for this analysis is made by Trane. A 200 ton RTAC model was used for analysis and found to perform at a rate of 1.22 kW/ton from the manufacturer's data which can be found in Appendix D. The first cost of this system is $8,990 more than the original design. The simulation showed that this configuration will save the building 237,482 kWh over a one year time period when compared to the original design. However, there are no resulting savings from the annual purchase of fuel for this equipment. This equipment will cost the owner $16,615 more to run the equipment for the first year. There is no financial benefit to the owner for selecting this equipment. The associated cost figures can be seen in relation to the original design in Table 4.7 on the following page.

End-Use	Energy Consumption	Unit of Energy	Cost/Unit	Enery Cost/Year	First Cost of System
HVAC	1018379.0	kWh	$0.078	$79,434	$427,990
Original Design	4285.0	MMBtu	$14.660	$62,818	$419,000
Differential	-237482.7	kWh	NA	$16,615	$8,990

Table 4.7 – Air-cooled Chiller Associated Energy & Cost Breakdown

The building emissions associated with operating this equipment can be seen below in Table 4.8. There is no environmental benefit to operating this equipment because it is more harmful to the environment than the natural gas fired rooftop units.

Building Emissions lbm				
Fuel	**Particulates/yr**	**SO_2/yr**	**NO_x/yr**	**CO_2/yr**
Coal	625	7266	4215	1222938
Natural Gas	0	0	0	0
Totals	625	7266	4215	1222938
Differential	459	5329	1627	123344

Table 4.8 – Air-cooled Chiller Associated Emissions/yr

4.2.4 Discussion of Chillers

These two hydronic design alternatives are being eliminated from further analysis and from the final selection. The three criteria used for selection are economics, energy reduction, and emissions. The simulation of the water-cooled and the air-cooled chillers for this site each resulted in higher annual operating fuel costs. Both chiller options were also found to have higher first costs when compared to the rooftop unit configuration. This is the reason why the units failed to meet the economic criteria. However, special attention must be paid when analyzing the energy consumption criteria. It appears that these units consume less energy on a yearly basis than the rooftop units did. While it is true that the units do consume less on-site energy over the course of a year, it is important to remember that electricity must be produced at the expense of burning other fossil fuels such as coal before it is delivered to the site. This process is estimated to be approximately 35% efficient. The criteria of energy reduction was established as a means of reducing the fossil fuel energy consumed, regardless of what value is consumed at the site by the end-user. The final design criteria focuses on the reduction of building emissions which is tied back to the overall energy consumption criteria. The simulation proves that these two options are more harmful to the environment in this category as well. It is for these reasons that these chilled water systems are being dismissed at this point in the analysis.

4.3 Building Combined Heat and Power

Combined heat and power (CHP), also referred to as cogeneration, is a technology which uses a prime mover such as a natural gas fired reciprocating engine or turbine to create on-site electricity and uses the waste exhaust heat from that process as a fuel to meet thermal load requirements of the building. Often, these systems will use the hot exhaust stream and a heat exchanger to meet the heating demands. In order to meet the cooling loads for the building an absorption chiller also has to be introduced into the system.

An initial feasibility check for the site showed that there was potential for combined heat and power generation on site. A good measure for feasibility of these systems is the spark spread. The spark spread is defined as the difference in price of alternative fuels per million Btu. This spark spread calculation below compares the price of grid purchased electricity to the price of purchased natural gas per MMBtu of energy.

Spark Spread Analysis:
Assuming 1 ft^3 of natural gas has 1030 Btu of energy:

Rate as of July 2005:

Electricity:	$0.078/kWh	➔ $22.86/MMBtu
Natural Gas:	$8.43/1000 ft^3	➔ <u>$8.18/MMBtu</u>
Spark Spread:		**$14.68/MMBtu**

Table 4.9 – Spark Spread Calculation for July 2005 Rates

The rule of thumb for cogeneration design states that if the spark spread is greater than $12/MMBtu, the site has potential for significant energy cost savings. With this knowledge in mind, an analysis for the site ensued. It was decided that for this analysis an absorption chiller would be necessary to meet the cooling loads.

As time passed, the price of oil increased. The price of natural gas consequently increased as well. The spark spread had to be calculated again to adjust for the change in rates.

Rate as of December 2005:

Electricity:	$0.08/kWh	➔ $23.45/MMBtu
Natural Gas:	$14.92/1000$ft^3$	➔ <u>$14.48/MMBtu</u>
Spark Spread:		**$8.67/MMBtu**

Table 4.10 – Spark Spread Calculation for December 2005 Rates

Cogeneration was eliminated as an option for a number of reasons. The feasibility of a reasonable payback period diminished as oil prices continued to influence the natural gas rates in Texas. This design option would need to include a natural gas fired reciprocating engine to generate electricity and an absorption chiller that would be fueled from the exhaust stream to create cooling as well as air handling units to serve the zones. The maintenance, acoustics, first cost, and long payback period were the final factors that terminated further analysis into this technology for this project; however, the initial analysis, modeling, and calculations provided an excellent opportunity for the designer to explore this technology at great length.

5.0 Hot Water Alternatives

The water system designed for the site includes three natural gas fired water heaters to serve the hot water demand for the building. The following section discusses the possibility of using solar energy to meet some of this demand.

5.1 Solar Water Heating

A solar energy analysis was conducted at the site to see if the water heating loads could be met or reduced within a reasonable payback period by this technology. There are three main types of solar collectors on the market that are capable of providing solar hot water heating: flat plate, glazed flat plate, and evacuated tube. Energy models for these three configurations were set up using RETScreen International's Solar Water Heating analysis program. The tilt on the collectors was decided to be set equal to the latitude line for Houston, TX at 30^0 for maximum exposure to the sun. The collectors are oriented directly south and are located on the building's flat roof.

The next part of this energy model was an estimate of how much load LA Fitness has a demand for. The building was modeled as an 800 person school with showers to decide the hot water demand at the site, because this configuration most closely resembled the building's hot water usage estimation. The building demand for hot water requires 63.28 MMBtu to be delivered for one year of operation.

5.1.1 Flat Plate Collector

An unglazed flat plate solar collector is the most basic type of collector that is widely used to collect the sun's energy. These collectors are made of a black polymer. There is no selective coating on the surface, and typically no frame or insulation on the back. As a result of these imperfections, thermal losses to the environment are significant; this is particularly the case when there are prevailing winds. These collectors are better used for energy delivery at relatively low temperatures.

The flat plate collector used in this energy analysis was an unglazed Heliodyne Mojave 410. Using the simulation software, it was found that using seven collectors at the site would be optimal to balance first cost against the amount of energy delivered. The five collectors have a total gross area of 200 ft^2 and deliver a total of 18.94 MMBtu for one year of operation. This configuration will be able to reduce the energy necessary from the water heaters by 29.9% over the course of a year.

Unglazed flat plate collectors are the most primitive of the three methods studied. However, this type of collector is also the least expensive regarding first cost. The first cost of the installed system is $4,752. This system would provide savings and has a fairly attractive payback period of 8 years. A full breakdown of the energy demand, energy delivered, and economics for this system is included in Appendix E.

5.1.2 Glazed Flat Plate Collector

A glazed collector is very good at collecting energy from the sun due to the selective coating that is applied to the polymer. Also, these collectors are superior to standard flat plate collectors because they are better insulated; they have a glass cover on top of the coating and an insulation panel behind it. Unlike a standard collector, the efficiency of this configuration is almost independent of wind.

The glazed collector used in the energy model is a Heliodyne Gobi 408. This simulation showed that it would be optimal to use five collectors to balance the system's energy collection and capital cost. Using five collectors will result in a total gross collector area of 160 ft^2 and would deliver 33.94 MMBtu over the course of a year. This technology would provide 53.6% of the entire energy needed to heat the water for the building for one year.

The first cost of the glazed collector system is $5,589. The glazed flat plate collector provides the most attractive payback period of the three models studied. The payback period for this model is 5.3 years. The breakdown for this system's energy demand, energy delivered and economics is included in Appendix E.

5.1.3 Evacuated Tube Collector

Evacuated tube collectors also have a selective coating; however, this coating is enclosed in a sealed, evacuated glass tubular envelope. The distinguishing characteristic feature of these systems is their incredibly low thermal losses to the environment. The efficiency of these systems is said to be independent of wind. The typical way in which these systems operate is by the means of a sealed heat-pipe on each tube to extract heat from the absorber. Liquid in contact with the heated absorber is vaporized and then recovered at the top of the tube while the vapor condenses and the condensate returns by gravity to the absorber.

The model used for simulation of the evacuated tube solar technology is the Thermomax Mazdon 20 – TMA 600S. This system is the most efficient of the three models studied. The seven collectors have a total gross area of 228 ft^2 and are capable of delivering 43.26 MMBtu over one year of operation. This correlates to 68.3% of the energy used by the water heaters in a year.

The first cost of this energy efficient system is $16,669 and the payback period is 11.6 years. The breakdown of the energy and economics for the evacuated tube system is included in Appendix E.

6.0 Lighting Breadth

The original LA Fitness lighting selection was designed to meet the 2001 edition of ASHRAE Standard 90.1. This standard specifies that for an exercise facility, the overall lighting power density shall not exceed 1.4 W/ft^2.

The original design provided 53,097 Watts of lighting to the interior spaces. The resulting power density for this configuration is 1.18 W/ft^2. The Table showing this energy calculation can be found in Appendix F. This design meets the 2001 version of the standard. However, the current version of this standard (ASHRAE Standard 90.1-2004) has been updated to include more stringent lighting power density requirements for most facilities in an attempt to further reduce energy consumption and cooling requirements in buildings. The 2004 version of this standard calls for exercise centers to maintain power densities no greater than 1.0 W/ft^2.

Building Area Method:

Building Area Type:	Exercise Center
Gross Lighted Floor Area:	45,000 ft^2
Lighting Watts Used in Spaces:	53,097 W
Original Lighting Power Density:	1.18 W/ft^2

Table 6.1 – Original Lighting Power Density Using Building Area Method

There is one type of lamp that is used very frequently throughout the building's original lighting design. The most frequently used product is a 32 W T8 fluorescent lamp that has a mean output of 2850 lumens. The first step in the redesign of this system included an exchange of all of these lamps for more efficient ones with comparable light output. The lamp selected as a replacement for this bulb is a 30 W T8 lamp that is made to fit in the exact same ballast that was initially used. This lamp has a mean output of 2710 lumens. Each lamp's lumen output is rated at the same condition (performance at 25^{0}C).

Designation	Lamp	Nominal Energy (Watts)	Nominal Length (in.)	Mean Lumens (25 C)	2 Lamps in ballast (Watts)
Original Design	FO32835XPECO	32	47.78	2850	65
Redesign	FO30835XPSSECO	30	47.78	2710	61

Table 6.2 - Comparison of Original Design and Redesign Lamp Choices

With this first change in place, there are savings of 4 Watts for every 2 lamps replaced with only a 5% decrease in lumen output in the redesigned spaces.

The racquetball courts were the first areas investigated after the lighting power densities for each space were calculated. These rooms each had 1,920 Watts of lighting being provided to 835 ft^2 spaces. This results in power densities of 2.3 W/ft^2 for the courts. These spaces should be kept slightly brighter than other areas because of the nature of the activity in the rooms; however, power densities greater than 1 W/ft^2 will be more than bright enough for the sport (played in a completely white room with a bright blue ball). The courts were redesigned to have a lighting power density of 1.2 W/ft^2.

After these two issues were addressed the building complied with the more stringent lighting power density set forth by the 2004 version of the standard. The redesigned lighting power density result is 0.989 W/ft^2. These calculations can also be found in Appendix F.

The power density in the redesign will lead to reduced annual energy use which correlates to less annual electric cost and a reduction of annual emissions.

Annual Savings Harvested from Lighting Redesign						
	Energy Consumption (kWh)	**Annual Lighting Cost**	**Particulates (lbm)**	**SOx (lbm)**	**NOx (lbm)**	**CO_2 (lbm)**
Original Design	315576	$25,246	142	1654	959	278338
Redesign	260087	$20,807	117	1363	791	229397
Reduction	55489	$4,439	25	291	169	48941

Table 6.3 – Reductions Resulting from Lighting Redesign

Most of the wattage that enters a building for lighting ends up as sensible heat that the building's mechanical system will have to remove. It is estimated that 99% of the total input wattage will result in heat; therefore, the resulting redesign also reduces the necessary cooling demand by 2.7 tons.

The first cost of the 30 watt lamps is rated at $4.29/lamp while the first cost of the 32 watt lamps is rated at $3.79/lamp. The overall first cost of the original lamps is $1224.17. The first cost of the new lamps for the redesign is $1312.17. The redesign is easy to justify economically because of the cost of energy saved. The first cost discussed in this section refers only to the first cost of the lamps that have been changed by the redesign. This is to say that the first cost of the entire lighting design has not been calculated for this study.

7.0 Structural Breadth

The suggested solar water heating system provides energy savings by reducing the natural gas consumed at the building. This equipment will be installed on the building's flat roof and consequently there will have to be a structural analysis to ensure that the existing structure can handle the loads imposed by the additional equipment.

The weight of each collector was conservatively estimated:

Collector Weight:	133 lb
Water in Collector:	230 lb
Heaviest Support Available:	781 lb
Total Weight =	1144 lb/collector

Table 7.1 – Conservative Collector Weight Estimate

Area of each collector: 4 ft x 8ft = 32 ft^2

Pressure and Distributed Load Calculation:

Pressure:
$P = 1144 \text{ lb} / 35.4 \text{ ft}^2$
$P = 36 \text{ lb/ft}^2$

Distributed Load:
$W = 36 \text{ lb/ft}^2 * 4 \text{ ft}$
$W = 144 \text{ lb/ft}$

Table 7.2 – Pressure and Distributed Load Calculations

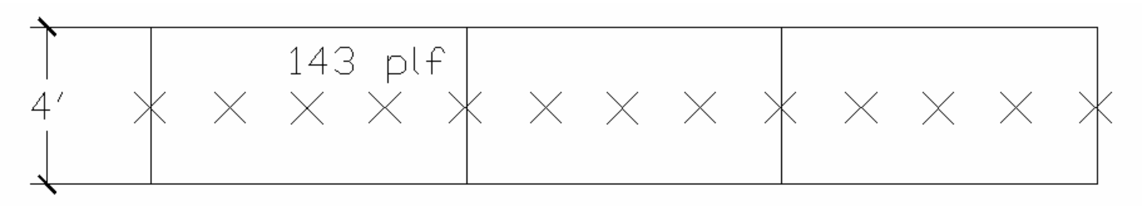


Figure 7.1 – Dead Load Imposed From Solar Collector and Support Structure

For a typical 6 ft tributary area:
Point Load Calculation:
$P = 143 \text{lb/ft} * 6 \text{ ft} \rightarrow P = 858$ pounds

The dead load of the joist used for the roof structure has a dead load of 21 pounds per linear foot (plf).

STANDARD LOAD TABLE/LONG SPAN STEEL JOISTS, LH-SERIES
Based on a Maximum Allowable Tensile Stress of 30 ksi

Joist Designation	Approx. Wt in Lbs. Per Linear Ft. (Joists Only)	Depth in inches	SAFELOAD* in Lbs. Between		CLEAR SPAN IN FEET															
			47-59	60-64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80
40LH08	16	40	16600	16600	254 150	247 144	241 138	234 132	228 127	222 122	217 117	211 112	206 108	201 104	196 100	192 97	187 93	183 90	178 86	174 83
40LH09	21	40	21800	21800	332 196	323 188	315 180	306 173	298 166	291 160	283 153	276 147	269 141	263 136	256 131	250 126	244 122	239 118	233 113	228 109
40LH10	21	40	24000	24000	367 216	357 207	347 198	338 190	329 183	321 176	313 169	305 162	297 156	290 150	283 144	276 139	269 134	262 129	255 124	249 119
40LH11	22	40	26200	26200	399 234	388 224	378 215	368 207	358 198	349 190	340 183	332 176	323 169	315 163	308 157	300 151	293 145	286 140	279 135	273 130

Figure 7.2 – Joist Selection Used In Roof Design

The superimposed dead load is 25 lb/ft^2 which translates to 150 plf because of the 6 ft tributary area. The total dead load for from the joist is 171 plf as calculated in below in Table 7.3.

Joist Dead Load:	21 plf
<u>Superimposed Dead Load:</u>	<u>150 plf</u>
Total Dead Load:	171 plf

Table 7.3 – Dead Load Calculation for Joist

The maximum allowable total moment for the joist selected is 80.9 foot-kips.

$$M_{max} = \frac{w \cdot l^2}{8} + \frac{P \cdot l}{2}$$

$$M_{max} = \frac{171 \cdot 42^2}{8} + \frac{858 \cdot 42}{2}$$

M_{max} = 55.7 foot-kips

Table 7.4 – Maximum Total Moment Applied on Joist from Equipment

55.7 foot-kips < 80.9 foot-kips ➔ The solar equipment can be safely installed on the roof.

8.0 Discussion of Original Design

The original rooftop design is almost impossible to beat when it comes to first cost; however, there are also shortcomings to this configuration. All thirteen rooftop units used at the site are single-zone constant volume mixed-air systems. This type of unit is convenient because it can be located directly above the zone that it serves.

In this systems most basic configuration, the thermostat reads the dry bulb temperature of the space that it is serving and modulates the cooling provided. The means of modulating this cooling capacity is achieved by cycling compressors on and off. At full load, this method of cooling is adequate to meet the temperature and relative humidity setpoints. The unit mixes return air from the space with outdoor air and passes this mixed air through the cooling coil. The full load occupied space adds heat and moisture as the dry bulb temperature and relative humidity levels rise towards their setpoints.

The fan provides a constant volume of air regardless of the space's cooling load. At part load, the system is forced to deliver supply air that is warmer than the full load condition air to avoid overcooling the space. When the compressor starts, the coil's surface will become cold quickly. Beads of water form on the coil until there is enough mass for gravity to overcome the surface tension. At this point, the moisture will fall to the drain pan below. The point is that there are a few minutes of lag time between when the compressor turns on and when the condensation reaches the drain pan for removal. Similarly, when the compressor stops running, the droplets that have not gained enough mass to fall are exposed to the outside air. These droplets evaporate back into the supply air, and actually raise the humidity ratio of the air that is coming across the coil. This will occur until all of the droplets are evaporated and the coil dries off completely.

As the percentage of time that compressor runs decreases, the unit's dehumidification effect also decreases. It should be noted that if the compressor cycles too often, there may be no dehumidification effect at all because the cooling coils would never have an opportunity to dry. Proper humidity control is further diminished if the systems are oversized.

Packaged rooftop units have cfm/ton limits that lead to the delivery of more airflow than the cooling load will require, even at design load. This is the case for the LA Fitness units. Increasing the system airflow will create a lower temperature difference which translates to warmer supply air and less dehumidification at all load conditions.

Below is a psychrometric example of RTU-3's coil performance on an average day in March (this example was taken at a part load condition to get a better idea of how a system may normally operate instead of always comparing to design conditions). The outdoor air is at a condition of 80^0F dry bulb and 70.1^0F wet bulb and the room setpoint is 75^0F and 50% relative humidity. The outdoor air will mix with return air to generate the mixed air condition. This mixed air is sent to the coil. The load is going to be some fraction of the total design load, and the resulting indoor conditions will not allow for humidity control. This is how two of the units (RTU-1 and RTU-2) are configured to operate in the original design. It is important to realize that the line from the supply air to the actual room condition is a constant parameter that represents the sensible and latent load generated within the space. With a constant volume system, the compressor must modulate down or cycle off when it receives a temperature signal that is below the setpoint to increase the temperature of the air coming off of the coils. The result from the analysis below shows that although the temperature requirements for the space have been met, the relative humidity is 11% higher than desired. There is no telling what damage this can lead to when considering the effects of mold on a building and its occupants.

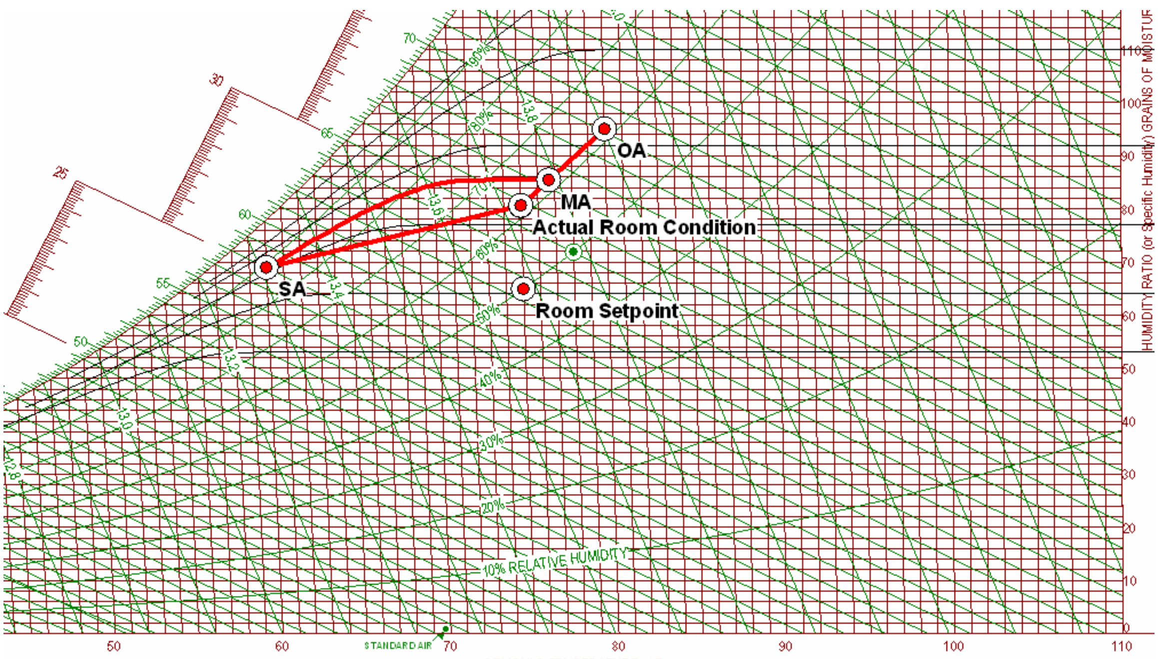


Figure 8.1 – RTU- 3 Loses Humidity Control without Reheat on a Part Load Day in March

The addition of a humidity sensor and reheat is an approach that will work to control some of the humidity problems associated with the systems discussed above. The designers of the original systems realized this and added hot gas reheat to eleven of the thirteen systems in question. It is unclear why the other two units did not receive this option as they receive the same outdoor air at the same conditions. Below is the same scenario as illustrated in Figure 8.1 with a reheat option.

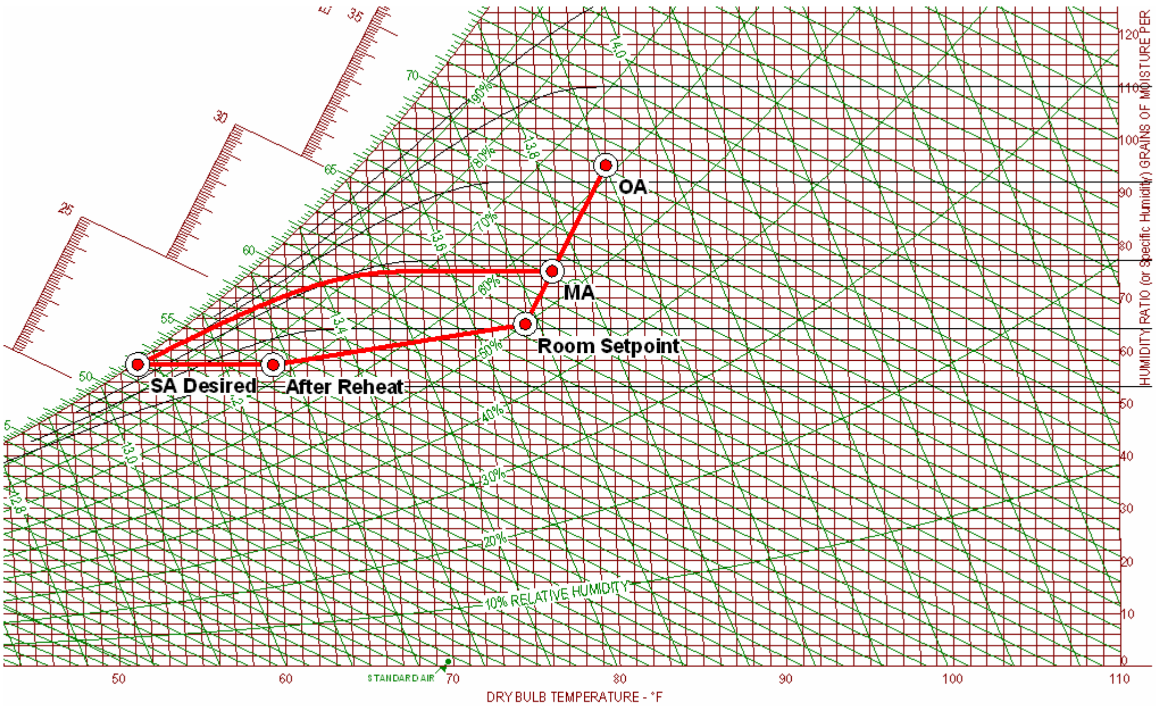


Figure 8.2 – RTU-3 with Hot Gas Reheat: Performance for a Part Load Day in March

A hot gas reheat option works by adding a refrigerant-to-air heat exchanger between the compressor and the condenser. After the coil has removed the required amount of moisture from the air, the heat exchanger adds sensible heat from the hot refrigerant vapor to the supply air downstream of the cooling coil.

It should be noted that the above hot gas configuration will adequately serve the required design and part load conditions. Knowing that this configuration will work, the question then becomes "Is this the best way to meet the loads?" The answer to that question depends on who you ask it to. If you ask a building owner, the answer is probably a simple and cheap "Yes." However, from a mechanical system design standpoint the answer is a hopeful "No." In order to meet the required loads for building, compressor energy is being used to cool the air down to 55^{0}F only to have it reheated back up to 60^{0}F. The reasoning behind this method of meeting space temperatures has been clearly illustrated by the example, but perhaps there is a better approach.

8.1 Desiccant Dehumidification

Desiccant technology relies on changing vapor pressures to perform dehumidification. Desiccants are characterized by having very low surface vapor pressures. When a relatively moist stream of air passes over a desiccant, the lower vapor pressure attracts the moisture out of the air. This removes latent energy from the air stream in a reaction that also generates sensible heat. If this process is accomplished with a solid desiccant, the desiccant is considered an adsorbent. Adsorbents are like sponges that collect and release moisture. There is another type of desiccant called an absorbent; these desiccants undergo some form of chemical or physical change while they dehumidify.

After a desiccant has collected moisture from the air, it becomes useless unless there is a means to release that moisture. This process is referred to as reactivation of the desiccant. A typical way of reactivating solid desiccants is to send a hot, dry stream of air to collect and exhaust the moisture from the surface. Often a hot enough stream is not available from the exhaust alone. In this scenario, a heater or heat exchanger is used to get the exhaust stream to a high enough temperature to collect the necessary moisture and reactivate the wheel. Once heat is added for regeneration, the system is deemed an "active desiccant" system. An example of a passive desiccant system would be an enthalpy wheel.

A rotary honeycomb style desiccant wheel can be sectioned off into two airflow regions as shown below in Figure 8.3. The airflow that is to be dehumidified flows through the larger section of the wheel (75% of the wheel in Figure 8.3) where it deposits moisture and gains sensible heat. The wheel rotates, and the smaller section of the wheel receives a hot and relatively dry air stream that flows in the opposite direction to reactivate the desiccant material for future dehumidification of the process air stream.

Figure 8.3 – Rotor Source's Desiccant Wheel

The processed air leaving the desiccant is now hot and dry. If the process is to be used for a cooling application, it would be wise to run the processed air through a sensible heat recovery wheel that uses the same exhaust stream. This process will lower the dry bulb temperature of the air while leaving the humidity ratio reached by the desiccant process alone. These two processes will bring the processed air to a temperature that is relatively close to the original dry bulb temperature only with a much lower humidity ratio. Figure 8.4 has been created to illustrate how these two wheels work in tandem.

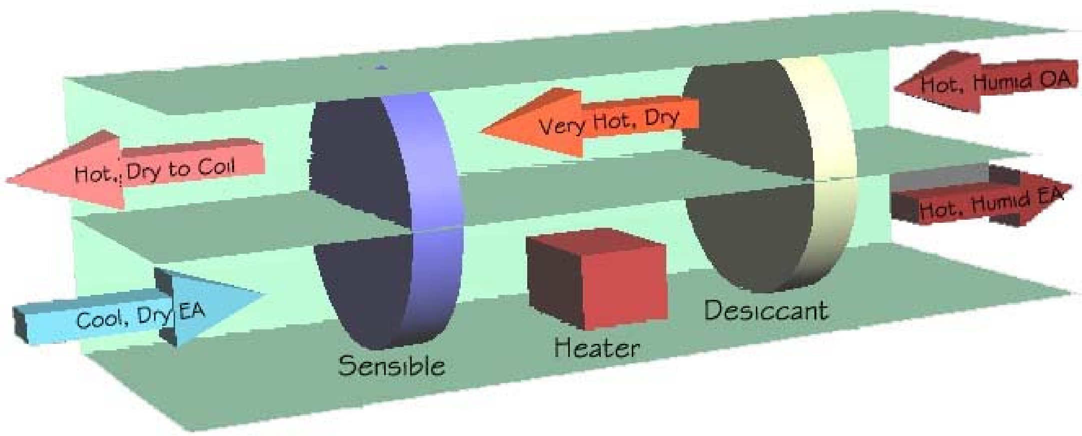


Figure 8.4 – Desiccant Dehumidification with the Addition of a Sensible Wheel Configuration

The outdoor air psychrometric state points will look like some variation of Figure 8.5 for the configuration above on a typical cooling day for Houston.

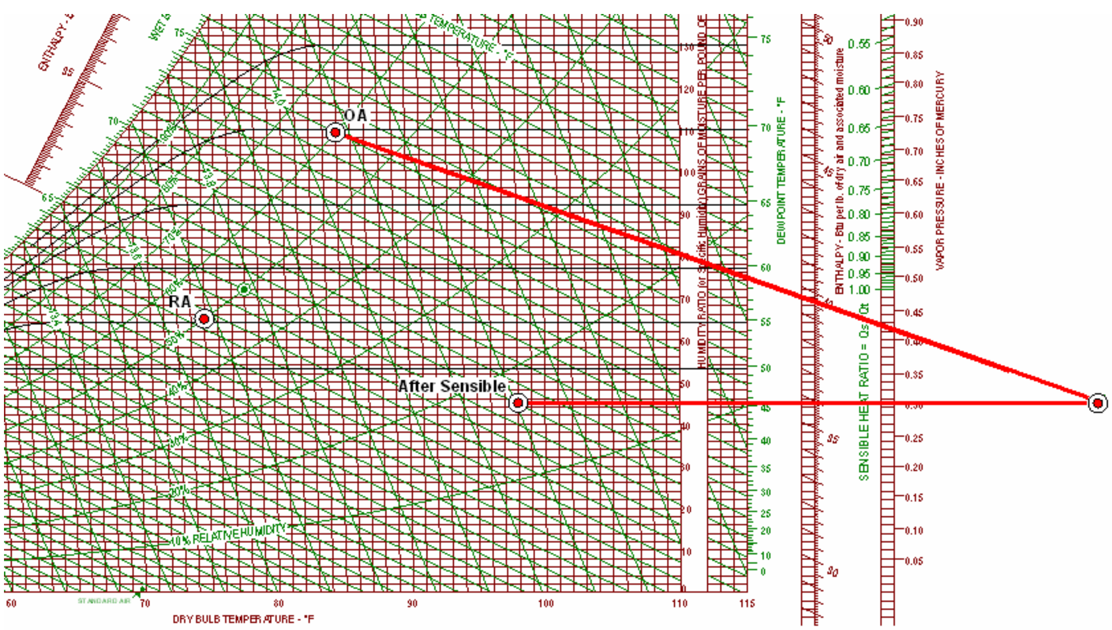


Figure 8.5 – Desiccant Dehumidification of Outdoor Air Followed By Sensible Heat Recovery

If the air is dry enough, the coils on the rooftop units will only need to remove sensible heat from the loads. This system eliminates much of the time that droplets from the coil would be evaporating back into the air stream as discussed before. This system will also undoubtedly save cooling coil energy. The design then lends itself to the question, "Can this design save enough cooling coil energy to have a reasonable payback period?" The recommendation provided in Section 9.0 will provide an analysis and an answer to that question.

9.0 Recommendation

9.1 Air Side System Selection

The original rooftop unit design was compared against several alternatives. After much analysis, an alternative base system could not be found to better meet the criteria set out for the redesign of the LA Fitness, West Oaks location. These systems use less fossil fuel energy, cost less money, and emit less harmful pollutants to the environment.

After processing the results of a long analysis, the design goal for the air system shifted from finding a better alternative, to finding the best configuration possible for the system that is in place. It can be seen that much of the energy being consumed by the building's HVAC system was being used for dehumidification across the coils. The humid outdoor air imposes a high latent load on systems residing in Houston, TX.

Active desiccant dehumidification followed by a sensible heat recovery wheel seemed to be the best way to remedy this problem. The initial idea was to change each of the original packaged rooftop units over to built-up custom units with the new dehumidification technology in place. The hot gas reheat from the direct expansion equipment could be modified to reactivate the desiccant in this scenario as well. However, after reviewing the cost of purchasing 13 modified rooftop units, 13 desiccant wheels, 13 sensible wheels, as well as the cost of having these systems built to the designers specifications on site, it was clear that this was not an economically feasible approach.

The outdoor air for the entire building is responsible for the bulk of the latent load during the year. Knowing this, the next design modification consisted of one large custom built unit that would process all of the outdoor air for the building before it was fed to the packaged units for mixing. This design would severely lower the cooling load on the units' compressors by eliminating a sizable portion of the latent load.

This proposed design also takes advantage of economies of scale. Purchasing and installing 26 separate wheels to process the outdoor air will cost more than purchasing and installing two large units (one desiccant, one sensible) that are centralized.

After redesigning the ventilation air to better meet the requirements of ASHRAE Standard 62.1-2004, it was found that the entire building requires 17,630 cfm. This would serve as the process air stream in the desiccant dehumidification system. Rotor

Source's equipment specifications for their PPS model desiccant wheel showed that their wheel would receive 18,000 cfm of process air, but would also require approximately 6000 cfm for the smaller counterflow air stream that is necessary to reactivate the desiccant surface.

This counterflow air stream can be provided by the one of the original design's exhaust fan (EF-5). This exhaust fan is located relatively close to seven of the units in a fairly central location on the roof. The decision to use this air stream is a result of its excellent location and appropriate amount of airflow at relatively dry conditions. CAD files showing this system's integration are provided in Appendix G.

The custom built unit also requires a heat source that is capable of raising the exhaust stream through the unit to a condition that will remove moisture from the desiccant. The heat source necessary will be a natural gas fired air heater. Natural gas is already being provided to the rooftop to serve the packaged units, so this option is logistically sound.

This configuration was simulated to find its energy consumption for the year. For one year of operation, this system will use 935,067 kWh of energy. This figure takes into account the energy usage by the fan motors, the wheel motors, and the reactivation heat used by the dehumidification unit as well as the energy consumed by the packaged units receiving the pre-treated outdoor air. For a point of comparison, the energy reduction compared to the unaltered original rooftop units is 320,792 kWh (1094 MMBtu). The full comparison can be seen below in Table 9.1.

End-Use	Energy Consumption	Unit of Energy	Cost/Unit	Enery Cost/Year	First Cost of System
HVAC	3190.4	MMBtu	$14.66	$46,771	$563,662
Original Design	4285.0	MMBtu	$14.66	$62,818	$419,000
Differential	-1094.6	MMBtu	NA	($16,047)	$144,662

Table 9.1 – Rooftop Modifications Compared to Original Rooftop Design

This is only system in the course of this study that has potential to meet all three of the design criteria. The system undeniably saves energy. As seen in Table 9.2 below, the suggested system produces less harmful emissions. The only consideration to be made lies in the economic analysis.

Building Emissions lbm/year				
Fuel	**Particulates/yr**	**SO_2/yr**	**NO_x/yr**	**CO_2/yr**
Dehumidified RTU	183	1935	2178	883154
Original RTU	166	1937	2589	1099594
Differential	17	-2	-410	-216440

Table 9.2 Emissions for Proposed Modifications

The rooftop unit scenario with preconditioned outdoor air resulted in a much higher first cost of $563,662. However the units do save $16,047 on energy every year. The payback period for these units was calculated to be 11 years. This payback period is conservative because the calculation assumes that the price of natural gas will only increase at a rate of 3% every year.

Many energy analysts are now predicting that the price of natural gas will be increasing at the rate of 7.5-8% per year. This estimation is not at all unjustified. In fact, the impact of rising natural gas rates became the determining factor in eliminating a combined heat and power system as a design alternative from this site. Using a natural gas escalation rate of 8%, the payback period for this technology is reduced to 8.5 years.

It is assumed that this building will have a 20 year life span. Using either escalation rate for gas, it can be seen that it would be a wise decision to install this dehumidification system at the site as a modification to the original design.

Appendix G includes other useful information about this design including: the CAD layouts of the roof, a schematic of the dehumidification system, psychrometric analysis taken at varying load conditions, and a discussion of the application of this system to other sites.

9.2 Water Side System Selection

Solar water heating is a good fit for LA Fitness. The building requires hot water at a fairly low temperature (120^{0}F). After reviewing solar water heating alternatives it can be seen that using a glazed flat plate collector will prove to be the best choice for this site. Using five of these collectors on the roof will reduce the natural gas consumed by the water heaters an average of 33.94 MMBtu per year. The collectors are an attractive choice for a building owner because the payback period is only 5.3 years. Table 9.3 below provides a comparison of the three types of panels that were modeled for use at the site.

Technology	Model	Energy Delivered Per Year (MMBtu)	% Demand Per Year	First Cost	Payback Period (Years)
Unglazed Flat Plate Collector	Heliodyne Mojave 410	18.94	29.9%	$4,752	8
Glazed Flat Plate Collector	Heliodyne Gobi 408	33.94	53.6%	$5,589	5.3
Evacuated Tube Collector	Thermomax Mazdon 20	43.26	68.3%	$16,999	11.6

Table 9.3 – Solar Energy Collector Comparison

The economic analysis provided did not deduct the cost of any of the original water heaters. There are a few reasons why it was decided to keep all of the existing water heaters in place. The system selected for water heating provides 53.6% of the energy used by the original heaters in a typical year. However, this is not to say that any of the existing water heating equipment could be downsized or removed. If there were to be a cloudy day that would require full hot water demand, solar equipment would not be able to meet 53% of the load. Rather, it is safe to say that 53% of the energy used in a year could be saved if these systems supplement the existing water heating system. The existing water heaters also serve as storage tanks for the hot water delivered by the sun.

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Appendix A:

Original Rooftop Unit Complete Schedule:

General Description					Fan Section				Heating Section		Cooling Section		
Unit	Area Served	Fuel	Energy Used (MBh)	Weight (lb)	Total Air (cfm)	Outside Air (cfm)	Static Pressure (" H20)	Motor H.P.	Output (MBh)	Efficiency (%)	Sensible (MBh)	Total (MBh)	Temperature Leaving Unit (F)
RTU-1	Reception	N. Gas	250	2200	5000	500	1.0	3.0	203	81	119.5	162.0	58.6
RTU-2	Kid's Club	N. Gas	250	2200	5000	700	1.0	3.0	203	81	102.0	145.8	58.6
RTU-3	Pool	N. Gas	500	5100	10500	3500	1.2	7.5	400	80	223.3	302.4	58.0
RTU-4	Lockers	N. Gas	350	2400	8300	3350	1.0	7.5	284	81	136.0	168.1	60.5
RTU-5	Basketball	N. Gas	250	2500	7500	750	1.0	5.0	203	81	159.8	226.5	58.3
RTU-6	Free Weights	N. Gas	250	2400	6000	750	1.0	3.0	203	81	133.8	199.2	55.2
RTU-7	Aerobics	N. Gas	500	4000	10000	1000	1.0	7.5	400	80	220.3	319.5	56.5
RTU-8	Racquetball	N. Gas	150	1700	4000	500	1.0	3.0	122	81	79.0	118.6	57.6
RTU-9	Cardio	N. Gas	400	2600	6600	1675	1.0	5.0	324	81	161.5	250.1	56.9
RTU-10	Cardio	N. Gas	400	2600	6600	1675	1.0	5.0	324	81	161.5	250.1	56.9
RTU-11	Lower Stairs	N. Gas	250	2300	5500	750	1.0	3.0	203	81	120.0	172.0	57.1
RTU-12	Spinning	N. Gas	150	1700	3500	500	1.0	3.0	122	81	80.8	117.8	57.7
RTU-13	Mezzanine	N. Gas	250	2400	5900	750	1.0	3.0	203	81	133.8	199.2	55.2

Packaged Cooling & Gas/Electric Rooftops

Voyager™
12½ – 25 Tons — 60 Hz

Packaged Cooling (TC)*

Packaged Gas/Electric (YC)*

August 2005
RT-PRC024-EN

Performance Data (12½, 15 Ton) Standard Efficiency

Table PD-1 — Gross Cooling Capacities (MBH) 12½ Ton Three Phase T/YC*150D3, D4, DW, DK

		Ambient Temperature																							
		85						95						105						115					
	Enter Dry Bulb	Entering Wet Bulb																							
CFM		61		67		73		61		67		73		61		67		73		61		67		73	
Airflow	(F)	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC
4500	75	135.0	108.0	153.0	86.0	162.0	59.3	126.0	103.0	146.0	82.0	158.0	56.7	116.0	98.2	138.0	77.7	152.0	53.8	106.0	93.1	127.0	80.0	145.0	50.5
	80	136.0	128.0	154.0	105.0	163.0	79.5	127.0	123.0	147.0	101.0	159.0	76.5	117.0	117.0	138.0	97.0	153.0	73.2	110.0	110.0	128.0	92.2	145.0	69.6
	85	142.0	142.0	154.0	123.0	164.0	95.9	135.0	135.0	147.0	120.0	159.0	94.2	127.0	127.0	138.0	116.0	153.0	91.7	120.0	120.0	128.0	111.0	146.0	88.5
	90	150.0	150.0	155.0	142.0	166.0	112.0	144.0	144.0	148.0	139.0	161.0	111.0	137.0	137.0	139.0	136.0	154.0	109.0	130.0	130.0	130.0	130.0	146.0	107.0
5000	75	139.0	114.0	155.0	89.8	163.0	60.4	129.0	109.0	149.0	85.4	159.0	57.9	120.0	104.0	141.0	89.5	154.0	55.0	109.0	98.9	131.0	85.1	147.0	51.6
	80	140.0	136.0	156.0	109.0	164.0	81.3	130.0	130.0	149.0	106.0	160.0	79.1	123.0	123.0	141.0	102.0	155.0	75.8	115.0	115.0	131.0	97.5	147.0	72.2
	85	147.0	147.0	157.0	129.0	166.0	98.2	141.0	141.0	150.0	126.0	161.0	96.8	133.0	133.0	141.0	123.0	155.0	94.8	125.0	125.0	131.0	118.0	148.0	91.9
	90	155.0	155.0	158.0	148.0	167.0	115.0	149.0	149.0	151.0	147.0	163.0	115.0	143.0	143.0	143.0	143.0	157.0	114.0	135.0	135.0	135.0	135.0	149.0	111.0
5500	75	142.0	120.0	157.0	91.3	164.0	61.5	132.0	115.0	151.0	88.8	160.0	59.0	122.0	110.0	143.0	84.2	155.0	56.1	112.0	104.0	133.0	89.9	148.0	52.9
	80	143.0	143.0	158.0	112.0	165.0	82.0	135.0	135.0	152.0	110.0	161.0	80.4	127.0	127.0	143.0	107.0	156.0	78.3	119.0	119.0	134.0	102.0	149.0	74.7
	85	152.0	152.0	159.0	133.0	167.0	99.9	145.0	145.0	152.0	132.0	163.0	99.1	138.0	138.0	144.0	129.0	157.0	97.4	130.0	130.0	134.0	125.0	150.0	95.0
	90	159.0	159.0	160.0	154.0	169.0	118.0	153.0	153.0	153.0	153.0	164.0	118.0	147.0	147.0	147.0	147.0	158.0	117.0	140.0	140.0	140.0	140.0	151.0	116.0
6000	75	145.0	125.0	158.0	93.3	165.0	62.5	135.0	120.0	153.0	91.2	161.0	60.1	125.0	115.0	145.0	87.3	156.0	57.2	115.0	110.0	136.0	82.8	149.0	54.0
	80	147.0	147.0	159.0	115.0	166.0	82.9	140.0	140.0	153.0	114.0	162.0	81.7	132.0	132.0	145.0	111.0	157.0	79.3	123.0	123.0	136.0	107.0	150.0	76.5
	85	155.0	155.0	161.0	137.0	168.0	101.1	149.0	149.0	154.0	137.0	164.0	101.0	142.0	142.0	146.0	135.0	158.0	99.8	134.0	134.0	137.0	131.0	151.0	97.8
	90	161.0	161.0	162.0	159.0	170.0	120.0	156.0	156.0	156.0	156.0	166.0	121.0	150.0	150.0	150.0	150.0	160.0	121.0	143.0	143.0	143.0	143.0	152.0	119.0

Notes:
1. All capacities shown are gross and have not considered indoor fan heat. To obtain **NET** cooling capacity subtract indoor fan heat. For indoor fan heat formula, refer to appropriate airflow table notes.
2. MBH = Total Gross Cooling Capacity
3. SHC = Sensible Heat Capacity

Table PD-2 — Gross Cooling Capacities (MBH) 15 Ton Three Phase T/YC*180B3, B4, BW, BK

		Ambient Temperature																							
		85						95						105						115					
	Enter. Dry Bulb	Entering Wet Bulb																							
CFM		61		67		73		61		67		73		61		67		73		61		67		73	
Airflow	(F)	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC	MBH	SHC
5400	75	168.0	133.0	188.0	106.0	198.0	73.0	157.0	128.0	181.0	101.0	194.0	70.4	147.0	122.0	172.0	97.0	187.0	67.3	137.0	117.0	162.0	92.1	179.0	63.2
	80	168.0	156.0	188.0	128.0	199.0	97.3	155.0	155.0	181.0	125.0	195.0	94.3	149.0	143.0	172.0	120.0	188.0	90.8	140.0	137.0	162.0	115.0	179.0	86.7
	85	174.0	174.0	189.0	150.0	200.0	117.0	166.0	166.0	182.0	147.0	196.0	115.0	158.0	158.0	172.0	143.0	189.0	113.0	150.0	150.0	162.0	138.0	180.0	110.0
	90	184.0	184.0	190.0	172.0	202.0	136.0	177.0	177.0	183.0	170.0	197.0	135.0	170.0	170.0	170.0	170.0	190.0	133.0	162.0	162.0	165.0	157.0	181.0	131.0
6000	75	172.0	140.0	190.0	110.0	199.0	74.3	161.0	134.0	184.0	106.0	195.0	71.7	151.0	129.0	175.0	101.0	189.0	68.7	140.0	124.0	165.0	96.1	181.0	64.9
	80	169.0	169.0	191.0	133.0	200.0	99.4	164.0	157.0	184.0	130.0	196.0	97.2	155.0	151.0	176.0	126.0	190.0	93.8	145.0	145.0	165.0	121.0	182.0	89.9
	85	181.0	181.0	192.0	156.0	202.0	119.0	173.0	173.0	185.0	154.0	198.0	118.0	165.0	165.0	176.0	151.0	191.0	116.0	157.0	157.0	166.0	146.0	183.0	113.0
	90	189.0	189.0	193.0	179.0	204.0	140.0	183.0	183.0	183.0	183.0	199.0	139.0	176.0	176.0	178.0	171.0	193.0	138.0	168.0	168.0	170.0	166.0	184.0	136.0
6600	75	176.0	147.0	192.0	112.0	200.0	75.5	165.0	141.0	186.0	110.0	196.0	73.0	154.0	136.0	178.0	105.0	190.0	69.8	144.0	130.0	168.0	99.8	182.0	66.1
	80	178.0	171.0	193.0	136.0	201.0	101.0	169.0	165.0	187.0	135.0	197.0	98.8	160.0	159.0	178.0	131.0	191.0	96.1	150.0	150.0	168.0	127.0	183.0	92.6
	85	185.0	185.0	194.0	161.0	203.0	122.0	179.0	179.0	188.0	160.0	199.0	121.0	171.0	171.0	179.0	158.0	193.0	119.0	163.0	163.0	169.0	153.0	185.0	116.0
	90	193.0	193.0	193.0	193.0	205.0	143.0	188.0	188.0	188.0	188.0	201.0	143.0	181.0	181.0	182.0	178.0	195.0	142.0	173.0	173.0	174.0	174.0	186.0	140.0
7200	75	179.0	152.0	194.0	114.0	201.0	76.7	168.0	147.0	188.0	112.0	197.0	74.2	158.0	142.0	180.0	109.0	191.0	71.0	142.0	142.0	170.0	103.0	183.0	67.2
	80	182.0	177.0	195.0	140.0	202.0	103.0	173.0	172.0	189.0	139.0	198.0	100.0	164.0	164.0	181.0	136.0	192.0	97.6	155.0	155.0	171.0	132.0	184.0	94.4
	85	189.0	189.0	196.0	166.0	204.0	125.0	183.0	183.0	190.0	166.0	200.0	123.0	176.0	176.0	182.0	164.0	194.0	121.0	167.0	167.0	167.0	167.0	186.0	119.0
	90	197.0	197.0	197.0	197.0	206.0	146.0	192.0	192.0	193.0	187.0	202.0	146.0	185.0	185.0	186.0	184.0	196.0	146.0	177.0	177.0	177.0	177.0	188.0	144.0

Notes:
1. All capacities shown are gross and have not considered indoor fan heat. To obtain **NET** cooling capacity subtract indoor fan heat. For indoor fan heat formula, refer to appropriate airflow table notes.
2. MBH = Total Gross Capacity
3. SHC = Sensible Heat Capacity

*Indicates both downflow and horizontal units.

Figure DD-1 — Cooling with Optional Electric Heat & Gas/Electric — 12½-17½ Tons Standard Efficiency, 12½ Ton High Efficiency

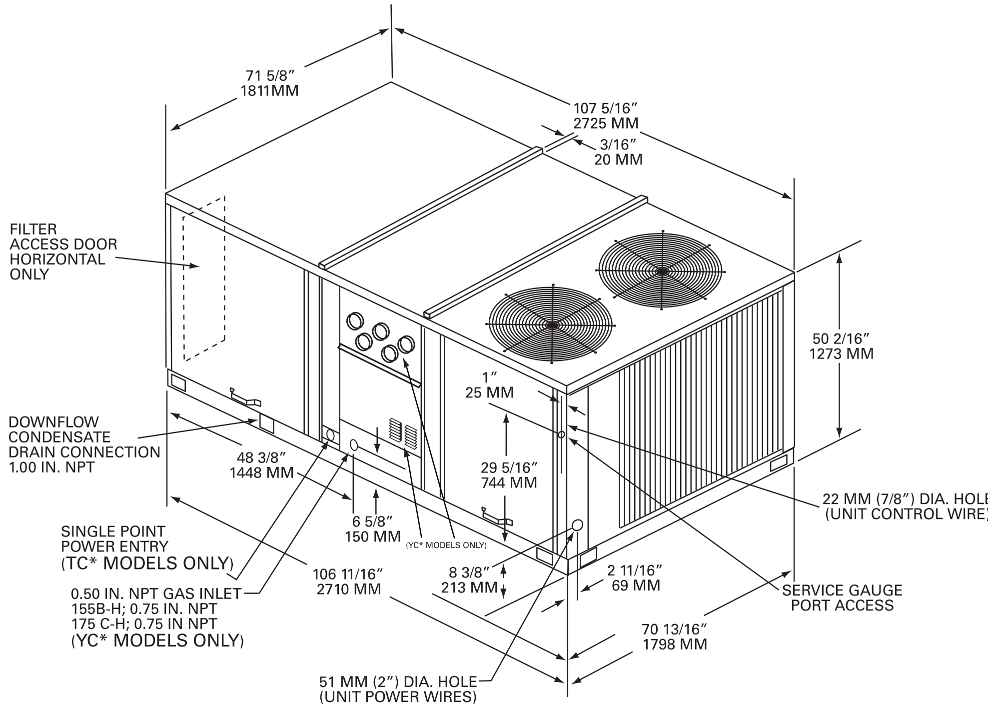
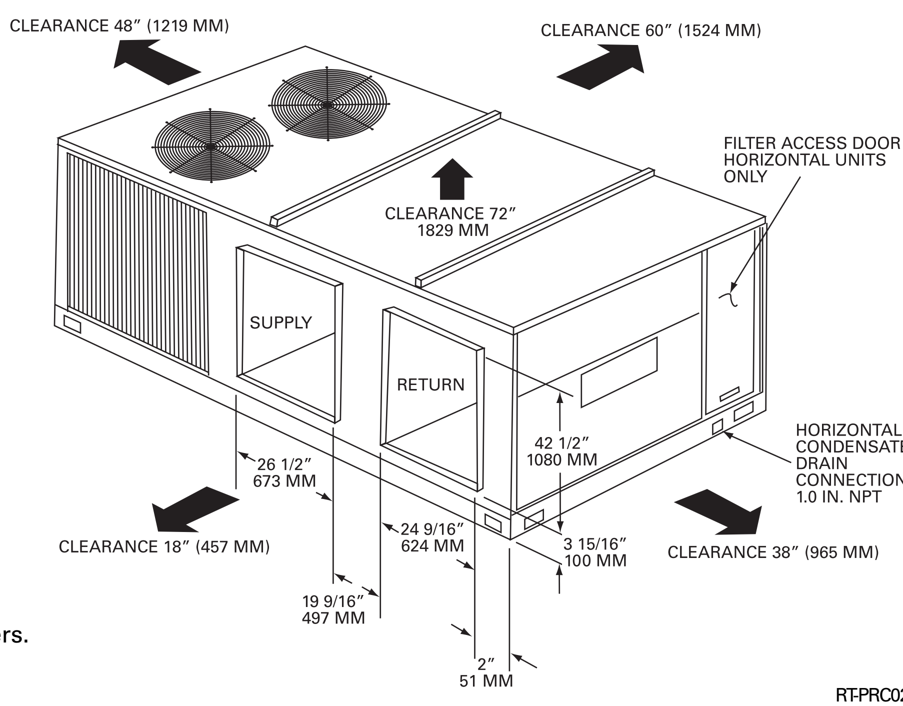


Figure DD-2 — Cooling with Optional Electric Heat & Gas/Electric — 12½-17½ Tons Standard Efficiency 12½ Ton High Efficiency — Downflow Unit Clearance



All dimensions are in inches/millimeters.

Figure DD-3 — Cooling with Optional Electric Heat & Gas/Electric — 12½-17½ Tons Standard Efficiency, 12½ Ton High Efficiency Roof Curb

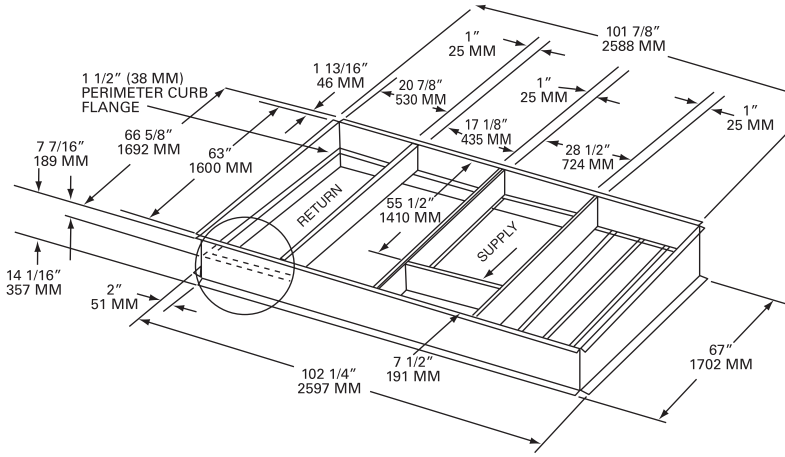


Figure DD-4 — Cooling with Optional Electric Heat & Gas/Electric — 12½-17½ Tons Standard Efficiency, 12½ Ton High Efficiency Downflow Duct Connections — Field Fabricated

Note: Duct flanges mount 7-7/16" down inside the curb on the 1-1/2" curb flanges. Roofcurb is intended for downflow use only.

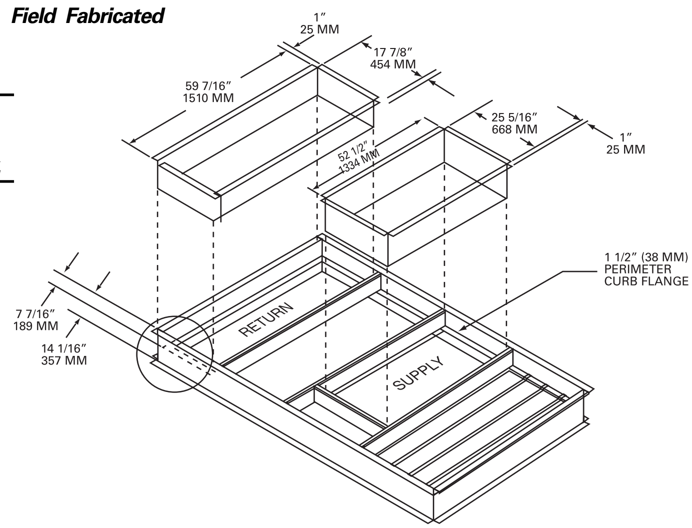
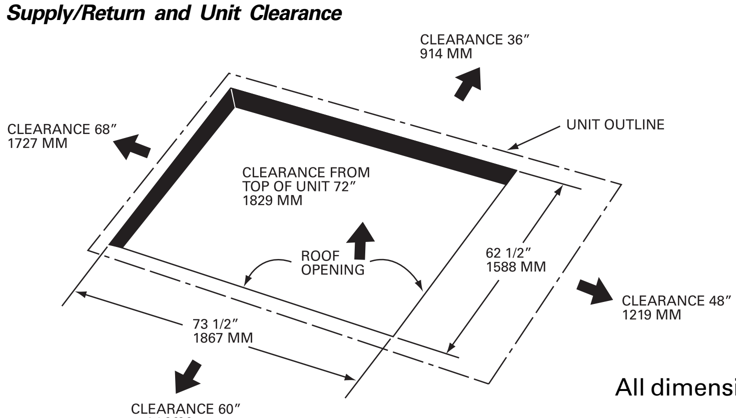


Figure DD-5 — Cooling with Optional Electric Heat & Gas/Electric — 12½-17½ Tons Standard Efficiency, 12½ Ton High Efficiency Horizontal Unit Supply/Return and Unit Clearance



All dimensions are in inches/millimeters.

Appendix B:

Pool Evaporation Equations Used for Section 3.3.1

2003 ASHRAE Handbook of HVAC Application Complete Evaporatoin Equation

$$w_p = \frac{A}{Y} \cdot (p_w - p_a) \cdot (95 + 0.425 \cdot V)$$

w_p = evaporation of water, lb/h

A = area of pool surface, ft^2

V = air velocity over water surface, fpm

Y = latent heat required to change water to vapor at surface water temperature, Btu/lb

p_a = saturatoin pressure at room air dew point, in. Hg

p_w = saturation vapor pressure taken at surface water temperature, in. Hg

2003 ASHRAE Handbook of HVAC Application Shortened Equation

Assuming Y = 1000 Btu/lb and V ranging from 10-30 fpm

$$w_p = 0.1 \cdot A \cdot (p_w - p_a) \cdot F_a$$

F_a = 1.0 for public use

Values used in equation:

p_w = 1.1025 in. Hg
p_a = 0.645 in. Hg
A = 2250 ft^2

Series R™ Helical Rotary Liquid Chillers

Model RTHD
175-450 Tons (60 Hz)
125-450 Tons (50 Hz)

Built for Industrial and Commercial Applications

August 2004

RLC-PRC020-EN

General Data

Nominal Data

Nominal Compressor	B1	B2	C1	C2	D1	D2	D3	E3
Tonnage (60 Hz)	175-200	200-225	225-275	275-325	325-400	375-450	N/A	N/A
Tonnage (50 Hz)	125-150	150-175	175-225	225-275	275-325	300-350	325-375	375-450

Notes:
1. Chiller selections can be optimized through the use of the ARI-Certified Series R selection program and by contacting your local Trane sales office.

General Data

Compressor Code	Evaporator Code	Condenser Code	Evaporator Water Storage Gallons	Evaporator Water Storage Liters	Condenser Water Storage Gallons	Condenser Water Storage Liters	Refrigerant Type	Refrigerant Charge lb	Refrigerant Charge kg
B1	B1	B1	41	155	28	106	HFC-134a	410	186
B1	C1	D1	55	208	31	117	HFC-134a	490	222
B2	B2	B2	45	170	29	110	HFC-134a	410	186
B2	C2	D2	58	220	34	129	HFC-134a	490	222
C1	D6	E5	45	170	29	110	HFC-134a	490	222
C1	D5	E4	52	197	32	121	HFC-134a	490	222
C1	E1	F1	82	310	60	226	HFC-134a	525	238
C2	D4	E4	52	197	32	121	HFC-134a	490	222
C2	D3	E3	78	295	47	178	HFC-134a	490	222
C2	F2	F3	107	405	61	231	HFC-134a	625	284
D1	D1	E1	69	261	44	166	HFC-134a	475	216
D1	F1	F2	102	386	57	216	HFC-134a	625	284
D1	G2	G2	144	545	91	344	HFC-134a	700	318
D2/D3	D2	E2	74	280	47	178	HFC-134a	475	216
D2/D3	F2	F3	107	405	61	231	HFC-134a	625	284
D2/D3	G3	G3	159	602	97	367	HFC-134a	700	318
E3	D2	E2	74	280	47	178	HFC-134a	475	216
E3	F2	F3	107	405	61	231	HFC-134a	625	284
E3	G3	G3	159	602	97	367	HFC-134a	700	318

General Data Water Flow Rates

Minimum/Maximum Evaporator Flow Rates (Gallons/Minute)

Evaporator Code	Two Pass Min	Two Pass Max	Two Pass Nominal Conn Size (In.)	Three Pass Min	Three Pass Max	Three Pass Nominal Conn Size (In.)	Four Pass Min	Four Pass Max	Four Pass Nominal Conn Size (In.)
B1	253	1104	8	168	736	6	—	—	—
B2	288	1266	8	192	844	6	—	—	—
C1	320	1412	8	213	941	6	—	—	—
C2	347	1531	8	232	1022	6	—	—	—
D1	415	1812	8	275	1206	8	—	—	—
D2	450	1980	8	300	1320	8	—	—	—
D3	486	2131	8	324	1417	8	—	—	—
D4	351	1542	8	234	1028	8	—	—	—
D5	351	1542	8	234	1028	8	—	—	—
D6	293	1287	8	196	860	8	—	—	—
E1	450	1980	8	300	1320	8	—	—	—
F1	563	2478	10	376	1655	8	—	—	—
F2	604	2667	10	404	1780	8	—	—	—
G1	—	—	—	505	2218	10	379	1666	8
G2	—	—	—	550	2413	10	411	1807	8
G3	—	—	—	622	2732	10	466	2050	8

Notes:
1. Minimum flow rates are based on **water only**.
2. All water connections are grooved pipe.

Minimum/Maximum Evaporator Flow Rates (Liters/Second)

Evaporator Code	Two Pass Min	Two Pass Max	Two Pass Nominal Conn Size (mm)	Three Pass Min	Three Pass Max	Three Pass Nominal Conn Size (mm)	Four Pass Min	Four Pass Max	Four Pass Nominal Conn Size (mm)
B1	16	70	200	11	46	150	—	—	—
B2	18	80	200	12	53	150	—	—	—
C1	20	89	200	13	59	150	—	—	—
C2	22	97	200	15	65	150	—	—	—
D1	26	114	200	17	76	200	—	—	—
D2	28	125	200	19	83	200	—	—	—
D3	31	134	200	20	89	200	—	—	—
D4	22	97	200	15	65	200	—	—	—
D5	22	97	200	15	65	200	—	—	—
D6	18	81	200	12	54	200	—	—	—
E1	28	125	200	19	83	200	—	—	—
F1	36	156	250	24	104	200	—	—	—
F2	38	168	250	25	112	200	—	—	—
G1	—	—	—	32	140	250	24	105	200
G2	—	—	—	35	152	250	26	114	200
G3	—	—	—	39	172	250	29	129	200

Notes:
1. Minimum flow rates are based on **water only**.
2. All water connections are grooved pipe.

Minimum/Maximum Condenser Flow Rates (Gallons/Minute)

Condenser Code	Two Pass Min	Two Pass Max	Two Pass Nominal Conn Size (In.)
B1	193	850	6
B2	212	935	6
D1	193	850	6
D2	212	935	6
E1	291	1280	8
E2	316	1390	8
E3	325	1420	8
E4	245	1080	8
E5	206	910	8
F1	375	1650	8
F2	355	1560	8
F3	385	1700	8
G1	444	1960	8
G2	535	2360	8
G3	589	2600	8

Notes:
1. Minimum flow rates are based on **water only**.
2. All water connections are grooved pipe.

Minimum/Maximum Condenser Flow Rates (Liters/Second)

Condenser Code	Two Pass Min	Two Pass Max	Two Pass Nominal Conn Size (mm)
B1	12	54	150
B2	13	59	150
D1	12	54	150
D2	13	59	150
E1	18	81	200
E2	20	88	200
E3	21	90	200
E4	15	68	200
E5	13	57	200
F1	24	104	200
F2	22	98	200
F3	24	107	200
G1	28	124	200
G2	34	149	200
G3	37	164	200

Notes:
1. Minimum flow rates are based on **water only**.
2. All water connections are grooved pipe.

Job Information

Selected By

PSU
131 Sowers St Apt. F-10
State College, PA
djm364@psu.edu

David Melfi
Tel 203 912 9346

SPX Cooling Technologies Contact

H & H Associates, Inc.
4510 Westport Drive
Mechanicsburg, PA 17055
frank@hhassociates.com

Tel 717-796-2401
Fax 717-796-9717

Cooling Tower Definition

Manufacturer	Marley	Fan Motor Speed	1200 rpm
Product	NC Class	Fan Motor Capacity per cell	20.00 BHp
Model	NC8305FL1	Fan Motor Output per cell	20.00 BHp
Cells	1	Fan Motor Output total	20.00 BHp
CTI Certified	No	Air Flow per cell	114400 cfm
Fan	8.000 ft, 8 Blades	Air Flow total	114400 cfm
Fan Speed	313 rpm, 7866.5 fpm	ASHRAE 90.1 Performance	72.6 gpm/Hp
Fans per cell	1		

Sound Pressure Level 75 dBA/Cell, 5.000 ft from Air Inlet Face. See sound report for details.

Conditions

Tower Water Flow	935.0 gpm	Air Density In	0.07056 lb/ft³
Hot Water Temperature	95.00 °F	Air Density Out	0.07109 lb/ft³
Range	10.00 °F	Humidity Ratio In	0.01855
Cold Water Temperature	85.00 °F	Humidity Ratio Out	0.02979
Approach	4.90 °F	Wet-Bulb Temp. Out	88.61 °F
Wet-Bulb Temperature	80.10 °F	Estimated Evaporation	10.7 gpm
Relative Humidity	50 %	Total Heat Rejection	4658600 Btu/h

- This selection meets your design conditions.
- The performance for this selection is not guaranteed because the approach is less than 5 °F.
- This selection is not CTI Certified because: the approach is less than 5 °F.

Weights & Dimensions

	Per Cell	Total
Shipping Weight	8870 lb	8870 lb
Max Operating Weight	19170 lb	19170 lb
Width	18.750 ft	18.750 ft
Length	10.896 ft	10.896 ft
Height	12.979 ft	
Static Lift	12.234 ft	

Minimum Enclosure Clearance

Clearance required on air inlet sides of tower without altering performance. Assumes no air from below tower.

Solid Wall	6.051 ft
50 % Open Wall	4.300 ft

Weights and dimensions do not include options; refer to sales drawings. For CAD layouts refer to file NC8305.dxf

Cold Weather Operation

Heater Sizing (to prevent freezing in the collection basin during periods of shutdown)

Heater kW/Cell	18.0	15.0	12.0	9.0	7.5	6.0	4.5
Ambient Temperature °F	-16.14	-6.05	4.04	14.13	19.17	24.22	29.26

Air-Cooled Series R™ Rotary Liquid Chiller

Model RTAC
140 to 500 Tons (60 Hz)
140 to 400 Tons (50 Hz)
Built For the Industrial and Commercial Markets

May 2005

RLC-PRC006-EN

Performance Data

Full Load Performance

Table P-1. 60 Hz standard efficiency machines in English units

Evaporator Leaving Water Temperature (F)	Unit Size Model RTAC	Condenser Entering Air Temperature (F) 85 Tons	85 kW input	85 EER	95 Tons	95 kW input	95 EER	105 Ton	105 kW input	105 EER	115 Tons	115 kW input	115 EER
	140 STD	138.0	139.9	10.9	128.4	152.4	9.4	118.5	166.4	8.0	108.4	182.1	6.7
	155 STD	151.4	152.3	10.9	141.1	165.9	9.4	130.4	181.2	8.0	119.5	198.3	6.8
	170 STD	165.6	165.0	11.0	154.5	179.8	9.5	143.1	196.5	8.1	131.5	215.0	6.9
	185 STD	180.5	183.4	10.8	168.6	199.4	9.4	156.2	217.5	8.0	143.5	237.8	6.8
	200 STD	196.6	202.7	10.7	183.6	219.8	9.3	170.1	239.3	7.9	156.2	261.2	6.7
40	225 STD	215.5	221.8	10.7	201.6	240.7	9.3	187.1	262.1	8.0	172.0	286.2	6.8
	250 STD	236.1	242.2	10.8	220.9	262.7	9.4	205.1	285.9	8.0	188.8	312.0	6.8
	275 STD	267.1	268.2	11.0	249.4	291.5	9.5	231.2	317.8	8.1	212.5	347.2	6.9
	300 STD	298.4	307.1	10.7	278.8	332.7	9.3	258.5	361.8	8.0	237.5	394.5	6.8
	350 STD	338.2	348.1	10.7	316.4	376.8	9.3	293.7	409.5	8.0	270.2	446.3	6.8
	400 STD	400.8	412.7	10.7	374.7	447.0	9.3	347.6	485.9	8.0	319.5	529.6	6.8
	450 STD	440.2	453.6	10.7	412.0	491.1	9.3	382.6	533.7	8.0	352.2	581.6	6.8
	500 STD	481.1	495.5	10.7	450.4	536.1	9.4	418.5	582.3	8.1	385.4	634.2	6.9
	140 STD	143.2	142.9	11.1	133.3	155.5	9.5	123.1	169.6	8.1	112.6	185.4	6.9
	155 STD	157.1	155.5	11.1	146.4	169.2	9.6	135.4	184.7	8.2	124.2	201.8	6.9
	170 STD	171.7	168.5	11.2	160.3	183.4	9.7	148.6	200.2	8.3	136.6	218.8	7.0
	185 STD	187.2	187.4	11.0	174.8	203.5	9.5	162.1	221.7	8.2	149.0	242.1	6.9
	200 STD	203.8	207.2	10.8	190.3	224.4	9.4	176.4	244.1	8.1	162.1	266.1	6.9
42	225 STD	223.4	226.9	10.9	208.9	245.9	9.5	193.9	267.5	8.1	178.4	291.7	6.9
	250 STD	244.8	247.9	10.9	229.0	268.5	9.5	212.7	292.0	8.2	195.7	318.2	6.9
	275 STD	276.9	274.0	11.1	258.6	297.4	9.7	239.9	323.9	8.3	220.6	353.4	7.0
	300 STD	309.2	314.0	10.9	288.9	339.7	9.5	268.0	369.0	8.1	246.3	401.9	6.9
	350 STD	350.6	356.2	10.9	327.9	385.2	9.5	304.4	418.1	8.2	280.1	455.1	6.9
	400 STD	415.1	421.9	10.9	388.1	456.4	9.5	360.1	495.5	8.1	331.2	539.5	6.9
	450 STD	455.9	464.0	10.9	426.7	501.8	9.5	396.4	544.7	8.2	364.9	592.8	6.9
	500 STD	498.3	507.3	10.9	466.6	548.2	9.5	433.6	594.7	8.2	399.3	646.9	7.0
	140 STD	148.4	146.0	11.3	138.2	158.6	9.7	127.7	172.9	8.3	116.9	188.7	7.0
	155 STD	162.9	158.8	11.3	151.9	172.6	9.8	140.5	188.2	8.4	128.9	205.4	7.1
	170 STD	177.9	172.0	11.4	166.2	187.0	9.9	154.1	203.9	8.5	141.8	222.6	7.2
	185 STD	193.9	191.4	11.2	181.2	207.6	9.7	168.0	226.0	8.3	154.5	246.4	7.1
	200 STD	211.0	211.8	11.0	197.2	229.2	9.6	182.8	248.9	8.2	168.0	271.1	7.0
44	225 STD	231.3	232.1	11.0	216.4	251.2	9.6	200.9	272.9	8.3	184.8	297.3	7.0
	250 STD	253.5	253.8	11.1	237.2	274.6	9.6	220.3	298.2	8.3	202.7	324.5	7.1
	275 STD	286.8	279.9	11.3	268.0	303.4	9.8	248.7	330.1	8.4	228.8	359.8	7.2
	300 STD	320.2	321.0	11.0	299.2	346.9	9.6	277.6	376.3	8.3	255.3	409.4	7.0
	350 STD	363.1	364.6	11.0	339.6	393.8	9.6	315.3	426.9	8.3	290.1	464.0	7.1
	400 STD	429.5	431.3	11.0	401.7	465.9	9.6	372.9	505.3	8.3	343.1	549.6	7.0
	450 STD	471.8	474.7	11.0	441.6	512.7	9.6	410.3	555.9	8.3	377.8	604.3	7.1
	500 STD	515.8	519.3	11.0	483.0	560.6	9.6	448.8	607.4	8.3	413.3	659.7	7.1

Notes:
1. Ratings based on sea level altitude and evaporator fouling factor of 0.00010.
2. Consult Trane representative for performance at temperatures outside of the ranges shown.
3. kW input is for compressors only.
4. EER = Energy Efficiency Ratio (Btu/watt-hour). Power inputs include compressors, condenser fans and control power.
5. Ratings are based on an evaporator temperature drop of 10°F.
6. Ambient temperatures 115°F and greater reflect the high ambient condenser option.
7. Interpolation between points is permissible. Extrapolation is not permitted.
8. Rated in accordance with ARI Standard 550/590-98.

Performance Data

Full Load Performance

Table P-1 (Continued). 60 Hz standard efficiency machines in English units

Evaporator Leaving Water Temperature (F)	Unit Size Model RTAC	Condenser Entering Air Temperature (F) 85			95			105			115		
		Tons	kW input	EER	Tons	kW input	EER	Ton	kW input	EER	Tons	kW input	EER
46	140 STD	153.8	149.1	11.4	143.3	161.8	9.9	132.4	176.2	8.4	121.2	192.1	7.1
	155 STD	168.7	162.2	11.5	157.4	176.1	10.0	145.7	191.7	8.5	133.7	209.1	7.2
	170 STD	184.2	175.6	11.6	172.2	190.7	10.0	159.8	207.7	8.6	147.1	226.5	7.3
	185 STD	200.7	195.6	11.3	187.6	211.9	9.9	174.1	230.3	8.5	160.2	250.9	7.2
	200 STD	218.4	216.5	11.2	204.1	234.0	9.7	189.3	253.9	8.4	174.0	276.2	7.1
	225 STD	239.3	237.4	11.2	223.9	256.7	9.7	207.9	278.5	8.4	191.3	303.0	7.1
	250 STD	262.4	259.8	11.2	245.6	280.8	9.8	228.0	304.5	8.4	209.8	331.0	7.2
	275 STD	296.9	286.0	11.5	277.6	309.6	10.0	257.6	336.4	8.6	237.2	366.2	7.3
	300 STD	331.3	328.2	11.2	309.7	354.2	9.8	287.4	383.8	8.4	264.3	417.1	7.2
	350 STD	375.7	373.2	11.2	351.5	402.6	9.8	326.3	435.8	8.4	300.3	473.1	7.2
	400 STD	444.2	440.9	11.2	415.5	475.7	9.8	385.8	515.3	8.4	355.1	559.8	7.2
	450 STD	488.0	485.6	11.2	456.8	523.9	9.7	424.4	567.3	8.4	390.8	615.9	7.2
	500 STD	533.6	531.8	11.2	499.6	573.3	9.8	464.2	620.3	8.4	424.0	665.5	7.2
48	140 STD	159.2	152.4	11.6	148.4	165.2	10.0	137.1	179.6	8.6	125.6	195.6	7.3
	155 STD	174.7	165.7	11.7	163.0	179.7	10.1	151.0	195.4	8.7	138.6	212.8	7.4
	170 STD	190.6	179.3	11.8	178.2	194.5	10.2	165.5	211.6	8.8	152.4	230.5	7.5
	185 STD	207.6	199.8	11.5	194.1	216.2	10.0	180.2	234.8	8.6	165.9	255.4	7.3
	200 STD	225.8	221.3	11.3	211.1	238.9	9.9	195.9	258.9	8.5	180.1	281.3	7.2
	225 STD	247.5	242.8	11.3	231.6	262.2	9.9	215.1	284.2	8.5	197.9	308.8	7.3
	250 STD	271.4	266.0	11.4	254.0	287.1	9.9	235.8	311.0	8.5	216.9	337.6	7.3
	275 STD	307.2	292.2	11.6	287.2	316.0	10.1	266.7	342.8	8.7	245.6	372.8	7.4
	300 STD	342.6	335.6	11.3	320.3	361.7	9.9	297.3	391.5	8.5	273.5	424.9	7.3
	350 STD	388.6	382.1	11.3	363.5	411.6	9.9	337.5	445.0	8.5	304.5	469.5	7.3
	400 STD	459.1	450.7	11.3	429.5	485.7	9.9	398.9	525.6	8.5	367.2	570.2	7.3
	450 STD	504.3	496.8	11.3	472.1	535.3	9.9	438.7	578.9	8.5	394.8	608.1	7.3
	500 STD	551.6	544.5	11.3	516.4	586.3	9.9	479.8	633.5	8.5	427.8	655.7	7.4
50	140 STD	164.7	155.7	11.8	153.5	168.5	10.2	141.9	183.0	8.7	130.1	199.1	7.4
	155 STD	180.7	169.3	11.9	168.7	183.3	10.3	156.3	199.1	8.8	143.6	216.5	7.5
	170 STD	197.1	183.1	11.9	184.4	198.4	10.4	171.2	215.5	8.9	157.8	234.5	7.6
	185 STD	214.6	204.1	11.7	200.7	220.6	10.2	186.4	239.3	8.8	170.9	258.6	7.5
	200 STD	233.3	226.2	11.5	218.2	243.9	10.0	202.5	264.0	8.6	186.3	286.5	7.4
	225 STD	255.8	248.4	11.5	239.4	267.9	10.0	222.3	290.0	8.6	203.1	311.4	7.4
	250 STD	280.6	272.3	11.5	262.5	293.6	10.0	243.7	317.5	8.7	218.2	330.7	7.5
	275 STD	317.6	298.5	11.8	297.0	322.4	10.3	275.9	349.4	8.9	250.8	373.1	7.6
	300 STD	354.0	343.1	11.5	331.0	369.4	10.0	307.3	399.3	8.7	278.7	424.3	7.4
	350 STD	401.7	391.1	11.4	375.7	420.8	10.0	348.8	454.3	8.7	307.4	462.5	7.5
	400 STD	474.2	460.7	11.5	443.7	496.0	10.0	412.1	536.0	8.7	369.9	560.7	7.5
	450 STD	520.9	508.3	11.4	487.7	547.0	10.0	453.1	590.8	8.7	396.9	595.5	7.5
	500 STD	569.9	557.5	11.4	533.5	599.6	10.0	495.5	647.0	8.7	431.5	644.4	7.6

Notes:
1. Ratings based on sea level altitude and evaporator fouling factor of 0.00010.
2. Consult Trane representative for performance at temperatures outside of the ranges shown.
3. kW input is for compressors only.
4. EER = Energy Efficiency Ratio (Btu/watt-hour). Power inputs include compressors, condenser fans and control power.
5. Ratings are based on an evaporator temperature drop of 10°F.
6. Ambient temperatures 115°F and greater reflect the high ambient condenser option.
7. Interpolation between points is permissible. Extrapolation is not permitted.
8. Rated in accordance with ARI Standard 550/590-98.

Performance Data

Part Load Performance

Table P-9. ARI part-load performance for 60 Hz standard efficiency machines in English units

Unit Size	Full Load Tons	Full Load EER	IPLV
140	138.2	9.7	**13.5**
155	151.9	9.8	**13.6**
170	166.2	9.9	**13.9**
185	181.2	9.7	**13.7**
200	197.2	9.6	**13.3**
225	216.4	9.6	**13.4**
250	237.2	9.6	**13.6**
275	268.0	9.8	**13.3**
300	299.2	9.6	**13.3**
350	339.6	9.6	**13.1**
400	401.7	9.6	**14.6**
450	441.6	9.6	**14.7**
500	483.0	9.6	**14.9**

Notes:
1. IPLV values are rated in accordance with ARI Standard 550/590-98.
2. EER and IPLV values include compressors, condenser fans and control kW.

Table P-10. ARI part-load performance for 60 Hz high efficiency machines in English units

Unit Size	Full Load Tons	Full Load EER	IPLV
140	143.9	10.3	**14.0**
155	157.1	10.4	**14.1**
170	171.2	10.4	**14.4**
185	187.1	10.3	**14.2**
200	204.1	10.1	**13.9**
225	223.9	10.2	14.0
250	243.2	10.1	**13.8**
275	277.1	13.7	**13.7**
300	308.8	10.2	**13.6**
350	349.7	10.5	**15.3**
400	415.5	10.1	**14.5**

Table P-11. ARI part-load performance for 50 Hz standard efficiency machines in English units

Unit Size	Full Load Tons	Full Load EER	IPLV
140	133.7	9.3	**14.2**
155	146.0	9.2	**14.1**
170	159.0	9.2	**13.9**
185	175.9	9.3	**13.8**
200	193.9	9.5	**14.2**
250	232.6	9.5	**14.3**
275	259.0	9.4	**14.4**
300	294.4	9.5	**14.0**
350	324.6	9.3	**15.9**
375	360.1	9.4	**16.0**
400	395.1	9.5	**16.1**

Notes:
1. IPLV values are rated in accordance with ARI Standard 550/590-98.
2. EER and IPLV values include compressors, condenser fans and control kW.

Table P-12. ARI part-load performance for 50 Hz high efficiency machines in English units

Unit Size	Full Load Tons	Full Load EER	IPLV
140	140.4	10.2	**15.0**
155	152.4	10.1	**14.9**
170	165.2	14.7	**14.7**
185	183.1	10.1	**14.6**
200	202.2	10.2	**14.9**
250	241.1	10.0	**14.3**
275	269.9	10.2	**14.9**
300	306.1	10.3	**14.5**
350	337.2	10.0	**16.1**
375	374.1	10.1	**16.1**
400	411.8	10.2	**16.2**

RETScreen® Energy Model - Solar Water Heating Project

Training & Support

Site Conditions		Estimate	Notes/Range
Project name		**LA Fitness**	See Online Manual
Project location		**Houston, TX**	
Nearest location for weather data		Houston, TX	Complete SR&HL sheet
Annual solar radiation (tilted surface)	MWh/m²	1.71	
Annual average temperature	°C	19.9	-20.0 to 30.0
Annual average wind speed	m/s	3.8	
Desired load temperature	°C	49	
Hot water use	L/d	1,500	
Number of months analysed	month	12.00	
Energy demand for months analysed	MWh	18.55	

System Characteristics		Estimate	Notes/Range
Application type		Service hot water (with storage)	
Base Case Water Heating System			
Heating fuel type	-	**Natural gas - mmBtu**	
Water heating system seasonal efficiency	%	50%	50% to 190%
Solar Collector			
Collector type	-	Unglazed	See Technical Note 1
Solar water heating collector manufacturer		Heliodyne	See Product Database
Solar water heating collector model		Heliodyne Mojave 410	
Gross area of one collector	m²	3.73	1.00 to 5.00
Aperture area of one collector	m²	3.56	1.00 to 5.00
Fr (tau alpha) coefficient	-	0.73	0.50 to 0.90
Wind correction for Fr (tau alpha)	s/m	0.040	0.030 to 0.050
Fr UL coefficient	(W/m²)/°C	6.08	10.00 to 15.00
Wind correction for Fr UL	(J/m³)/°C	4.37	3.00 to 15.00
Suggested number of collectors		5	
Number of collectors		5	
Total gross collector area	m²	18.7	
Storage			
Ratio of storage capacity to coll. area	L/m²	63.8	37.5 to 100.0
Storage capacity	L	1,135	
Balance of System			
Heat exchanger/antifreeze protection	yes/no	Yes	
Heat exchanger effectiveness	%	80%	50% to 85%
Suggested pipe diameter	mm	13	8 to 25 or PVC 35 to 50
Pipe diameter	mm	12	8 to 25 or PVC 35 to 50
Pumping power per collector area	W/m²	7	3 to 22, or 0
Piping and solar tank losses	%	6%	1% to 10%
Losses due to snow and/or dirt	%	3%	2% to 10%
Horz. dist. from mech. room to collector	m	5	5 to 20
# of floors from mech. room to collector	-	1	0 to 20

Annual Energy Production (12.00 months analysed)		Estimate	Notes/Range
SWH system capacity	kW_{th}	12	
	million Btu/h	0.043	
Pumping energy (electricity)	MWh	**0.15**	
Specific yield	kWh/m²	298	
System efficiency	%	17%	
Solar fraction	%	30%	
Renewable energy delivered	MWh	**5.55**	
	million Btu	18.94	
			Complete Cost Analysis sheet

Version 3.1 NRCan/CETC - Varennes

RETScreen® Solar Resource and Heating Load Calculation - Solar Water Heating Project

Site Latitude and Collector Orientation		Estimate	Notes/Range
Nearest location for weather data		Houston, TX	See Weather Database
Latitude of project location	°N	30.0	-90.0 to 90.0
Slope of solar collector	°	30.0	0.0 to 90.0
Azimuth of solar collector	°	0.0	0.0 to 180.0

Monthly Inputs

(Note: 1. Cells in grey are not used for energy calculations; 2. Revisit this table to check that all required inputs are filled if you change system type or solar collector type or pool type, or method for calculating cold water temperature).

Month	Fraction of month used (0 - 1)	Monthly average daily radiation on horizontal surface (kWh/m²/d)	Monthly average temperature (°C)	Monthly average relative humidity (%)	Monthly average wind speed (m/s)	Monthly average daily radiation in plane of solar collector (kWh/m²/d)
January	1.00	2.66	10.4	74.6	4.1	3.60
February	1.00	3.42	12.2	73.4	4.3	4.16
March	1.00	4.25	16.3	72.7	4.5	4.65
April	1.00	5.01	20.4	74.1	4.4	4.93
May	1.00	5.62	23.8	75.5	4.0	5.11
June	1.00	6.02	26.7	75.0	3.6	5.27
July	1.00	5.95	27.8	74.7	3.2	5.29
August	1.00	5.61	27.6	75.1	3.0	5.34
September	1.00	4.87	25.3	76.3	3.3	5.14
October	1.00	4.19	20.7	74.1	3.4	5.04
November	1.00	3.07	16.1	75.3	3.8	4.13
December	1.00	2.51	12.0	74.9	3.9	3.51

		Annual	Season of Use
Solar radiation (horizontal)	MWh/m²	1.62	1.62
Solar radiation (tilted surface)	MWh/m²	1.71	1.71
Average temperature	°C	19.9	19.9
Average wind speed	m/s	3.8	3.8

Water Heating Load Calculation		Estimate	Notes/Range
Application type	-	Service hot water	
System configuration	-	With storage	
Building or load type	-	School	
Number of units	Student	800	
Rate of occupancy	%	80%	50% to 100%
Estimated hot water use (at ~60 °C)	L/d	1,472	
Hot water use	L/d	1,500	
Desired water temperature	°C	49	
Days per week system is used	d	7	1 to 7
Cold water temperature	-	Auto	
Minimum	°C	16.6	1.0 to 10.0
Maximum	°C	22.7	5.0 to 15.0
Months SWH system in use	month	12.00	
Energy demand for months analysed	MWh	18.55	
	million Btu	63.28	

Return to Energy Model sheet

RETScreen® Cost Analysis - Solar Water Heating Project

Type of project: Pre-feasibility | Currency: $ | Cost references: None

Initial Costs (Credits)	Unit	Quantity	Unit Cost	Amount	Relative Costs	Quantity Range	Unit Cost Range
Feasibility Study							
Other - Feasibility study	Cost	0	$ -	$ -		-	-
Sub-total :				**$ -**	0.0%		
Development							
Other - Development	Cost	0	$ -	$ -		-	-
Sub-total :				**$ -**	0.0%		
Engineering							
Other - Engineering	Cost	0	$ -	$ -		-	-
Sub-total :				**$ -**	0.0%		
Energy Equipment							
Solar collector	m²	18.7	$ 90	$ 1,679		-	-
Solar storage tank	L	1,135	$ -	$ -		-	-
Solar loop piping materials	m	19	$ 7.00	$ 135		-	-
Circulating pump(s)	W	125	$ 1.10	$ 137		-	-
Heat exchanger	kW	10.7	$ 15	$ 160		-	-
Transportation	project	1	$ 100	$ 100		-	-
Other - Energy equipment	Cost	0	$ -	$ -		-	-
Sub-total :				**$ 2,211**	46.5%		
Balance of System							
Collector support structure	m²	18.7	$ 70	$ 1,306		-	-
Plumbing and control	project	1	$ 200	$ 200		-	-
Collector installation	m²	18.7	$ 10	$ 187		-	-
Solar loop installation	m	19	$ 4.00	$ 77		-	-
Auxiliary equipment installation	project	1	$ 50	$ 50		-	-
Transportation	project	1	$ 50	$ 50		-	-
Other - Balance of system	Cost	0	$ -	$ -		-	-
Sub-total :				**$ 1,869**	39.3%		
Miscellaneous							
Training	p-h	4	$ 60	$ 240		-	-
Contingencies	%	10%	$ 4,320	$ 432		-	-
Sub-total :				**$ 672**	14.1%		
Initial Costs - Total				**$ 4,752**	100.0%		

Annual Costs (Credits)	Unit	Quantity	Unit Cost	Amount	Relative Costs	Quantity Range	Unit Cost Range
O&M							
Property taxes/Insurance	project	0	$ -	$ -		-	-
O&M labour	project	1	$ 15	$ 15		-	-
Other - O&M	Cost	0	$ -	$ -		-	-
Contingencies	%	10%	$ 15	$ 2		-	-
Sub-total :				**$ 17**	57.7%		
Electricity	kWh	155	**$ 0.0780**	**$ 12**	42.3%	-	-
Annual Costs - Total				$ 29	100.0%		

Periodic Costs (Credits)		Period	Unit Cost	Amount		Interval Range	Unit Cost Range
Valves and fittings	Cost	10 yr	$ 250	**$ 250**		-	-
	Credit	10 yr		**$ -**		-	-
				$ -		-	-
End of project life		-		**$ -**			Go to GHG Analysis sheet

Version 3.1 NRCan/CETC - Varennes

RETScreen® Financial Summary - Solar Water Heating Project

Annual Energy Balance

Project name		LA Fitness	Electricity required	MWh	0.15
Project location		Houston, TX			
Renewable energy delivered	MWh	5.55			
Heating fuel displaced	-	ıtural gas - mmBtu			

Financial Parameters

Avoided cost of heating energy	$/mmBtu	14.480	Debt ratio	%	0.0%
			Income tax analysis?	yes/no	No
Retail price of electricity	$/kWh	0.078			
Energy cost escalation rate	%	3.0%			
Inflation	%	2.0%			
Discount rate	%	6.0%			
Project life	yr	20			

Project Costs and Savings

Initial Costs				**Annual Costs and Debt**		
Feasibility study	0.0%	$	-	O&M	$	17
Development	0.0%	$	-	Electricity	$	12
Engineering	0.0%	$	-			
Energy equipment	46.5%	$	2,211	**Annual Costs and Debt -Total**	**$**	**29**
Balance of system	39.3%	$	1,869			
Miscellaneous	14.1%	$	672	**Annual Savings or Income**		
Initial Costs - Total	100.0%	**$**	**4,752**	Heating energy savings/income	$	548
Incentives/Grants		$	-			
				Annual Savings - Total	**$**	**548**
Periodic Costs (Credits)						
Valves and fittings		$	250	Schedule yr # 10,20		
		$	-			
		$	-			
End of project life -		$	-			

Financial Feasibility

Pre-tax IRR and ROI	%	11.9%			
After-tax IRR and ROI	%	11.9%			
Simple Payback	yr	9.1	Project equity	$	4,752
Year-to-positive cash flow	yr	8.0			
Net Present Value - NPV	$	2,781			
Annual Life Cycle Savings	$	242			
Benefit-Cost (B-C) ratio	-	1.59			

Yearly Cash Flows

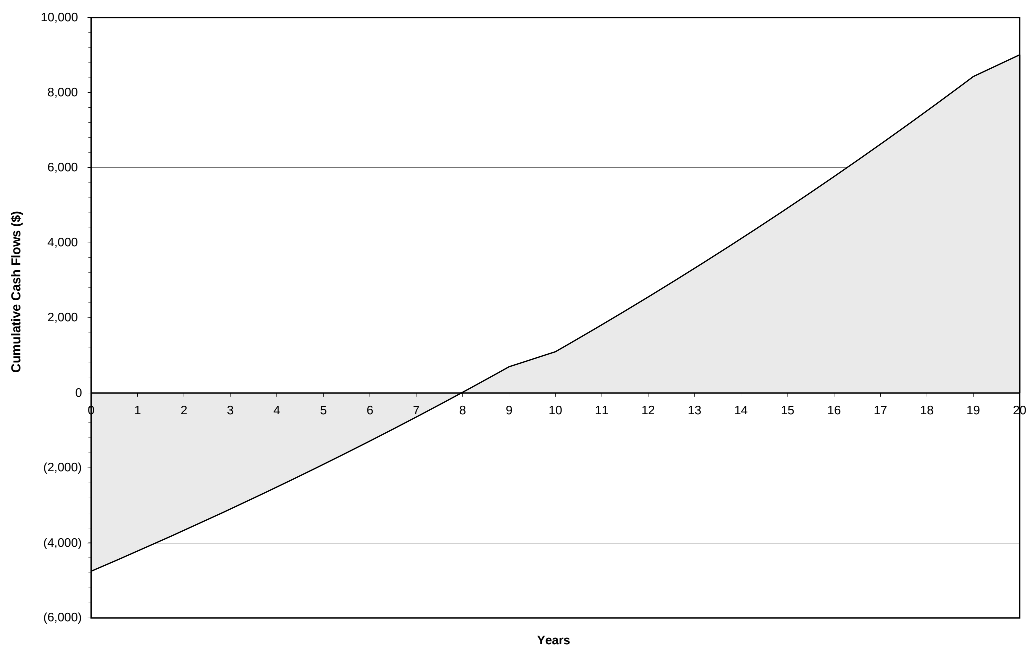
Year #	Pre-tax $	After-tax $	Cumulative $
0	(4,752)	(4,752)	(4,752)
1	536	536	(4,217)
2	552	552	(3,665)
3	569	569	(3,096)
4	586	586	(2,510)
5	604	604	(1,907)
6	622	622	(1,285)
7	641	641	(644)
8	660	660	16
9	680	680	696
10	396	396	1,092
11	722	722	1,814
12	744	744	2,558
13	766	766	3,324
14	790	790	4,114
15	813	813	4,927
16	838	838	5,765
17	863	863	6,629
18	890	890	7,518
19	917	917	8,435
20	573	573	9,008

Version 3.1 NRCan/CETC - Varennes

Cumulative Cash Flows Graph

SWH Project Cumulative Cash Flows
LA Fitness, Houston, TX

Renewable energy delivered (MWh/yr): 5.55 **Total Initial Costs: $ 4,752**



IRR and ROI: 11.9% **Year-to-positive cash flow: 8 yı** **Net Present Value: $ 2,781**

Version 3.1 NRCan/CETC - Varennes

RETScreen® Energy Model - Solar Water Heating Project

Training & Support

Site Conditions		Estimate	Notes/Range
Project name		**LA Fitness**	See Online Manual
Project location		**Houston, TX**	
Nearest location for weather data		Houston, TX	Complete SR&HL sheet
Annual solar radiation (tilted surface)	MWh/m²	1.71	
Annual average temperature	°C	19.9	-20.0 to 30.0
Annual average wind speed	m/s	3.8	
Desired load temperature	°C	49	
Hot water use	L/d	1,500	
Number of months analysed	month	12.00	
Energy demand for months analysed	MWh	18.55	

System Characteristics		Estimate	Notes/Range
Application type		Service hot water (with storage)	
Base Case Water Heating System			
Heating fuel type	-	**Natural gas - mmBtu**	
Water heating system seasonal efficiency	%	50%	50% to 190%
Solar Collector			
Collector type	-	Glazed	See Technical Note 1
Solar water heating collector manufacturer		Heliodyne	See Product Database
Solar water heating collector model		Heliodyne Gobi 408	
Gross area of one collector	m²	3.00	1.00 to 5.00
Aperture area of one collector	m²	2.77	1.00 to 5.00
Fr (tau alpha) coefficient	-	0.74	0.50 to 0.90
Fr UL coefficient	(W/m²)/°C	4.57	1.50 to 8.00
Temperature coefficient for Fr UL	(W/(m·°C)²)	0.00	0.000 to 0.010
Suggested number of collectors		5	
Number of collectors		5	
Total gross collector area	m²	15.0	
Storage			
Ratio of storage capacity to coll. area	L/m²	82.0	37.5 to 100.0
Storage capacity	L	1,135	
Balance of System			
Heat exchanger/antifreeze protection	yes/no	Yes	
Heat exchanger effectiveness	%	80%	50% to 85%
Suggested pipe diameter	mm	13	8 to 25 or PVC 35 to 50
Pipe diameter	mm	12	8 to 25 or PVC 35 to 50
Pumping power per collector area	W/m²	7	3 to 22, or 0
Piping and solar tank losses	%	6%	1% to 10%
Losses due to snow and/or dirt	%	3%	2% to 10%
Horz. dist. from mech. room to collector	m	5	5 to 20
# of floors from mech. room to collector	-	1	0 to 20

Annual Energy Production (12.00 months analysed)		Estimate	Notes/Range
SWH system capacity	kW_{th}	10	
	MWth	0.010	
Pumping energy (electricity)	MWh	**0.22**	
Specific yield	kWh/m²	663	
System efficiency	%	39%	
Solar fraction	%	54%	
Renewable energy delivered	MWh	**9.95**	
	million Btu	33.94	

Complete Cost Analysis sheet

RETScreen® Solar Resource and Heating Load Calculation - Solar Water Heating Project

Site Latitude and Collector Orientation		Estimate	Notes/Range
Nearest location for weather data		Houston, TX	See Weather Database
Latitude of project location	°N	30.0	-90.0 to 90.0
Slope of solar collector	°	30.0	0.0 to 90.0
Azimuth of solar collector	°	0.0	0.0 to 180.0

Monthly Inputs

(Note: 1. Cells in grey are not used for energy calculations; 2. Revisit this table to check that all required inputs are filled if you change system type or solar collector type or pool type, or method for calculating cold water temperature).

Month	Fraction of month used (0 - 1)	Monthly average daily radiation on horizontal surface (kWh/m²/d)	Monthly average temperature (°C)	Monthly average relative humidity (%)	Monthly average wind speed (m/s)	Monthly average daily radiation in plane of solar collector (kWh/m²/d)
January	1.00	2.66	10.4	74.6	4.1	3.60
February	1.00	3.42	12.2	73.4	4.3	4.16
March	1.00	4.25	16.3	72.7	4.5	4.65
April	1.00	5.01	20.4	74.1	4.4	4.93
May	1.00	5.62	23.8	75.5	4.0	5.11
June	1.00	6.02	26.7	75.0	3.6	5.27
July	1.00	5.95	27.8	74.7	3.2	5.29
August	1.00	5.61	27.6	75.1	3.0	5.34
September	1.00	4.87	25.3	76.3	3.3	5.14
October	1.00	4.19	20.7	74.1	3.4	5.04
November	1.00	3.07	16.1	75.3	3.8	4.13
December	1.00	2.51	12.0	74.9	3.9	3.51

		Annual	Season of Use
Solar radiation (horizontal)	MWh/m²	1.62	1.62
Solar radiation (tilted surface)	MWh/m²	1.71	1.71
Average temperature	°C	19.9	19.9
Average wind speed	m/s	3.8	3.8

Water Heating Load Calculation		Estimate	Notes/Range
Application type	-	Service hot water	
System configuration	-	With storage	
Building or load type	-	School	
Number of units	Student	800	
Rate of occupancy	%	80%	50% to 100%
Estimated hot water use (at ~60 °C)	L/d	1,472	
Hot water use	L/d	1,500	
Desired water temperature	°C	49	
Days per week system is used	d	7	1 to 7
Cold water temperature	-	Auto	
Minimum	°C	16.6	1.0 to 10.0
Maximum	°C	22.7	5.0 to 15.0
Months SWH system in use	month	12.00	
Energy demand for months analysed	MWh	18.55	
	million Btu	63.28	

Return to Energy Model sheet

RETScreen® Cost Analysis - Solar Water Heating Project

Type of project: Pre-feasibility | Currency: $ | Cost references: None

Initial Costs (Credits)	Unit	Quantity	Unit Cost	Amount	Relative Costs	Quantity Range	Unit Cost Range
Feasibility Study							
Other - Feasibility study	Cost	0	$ -	$ -		-	-
Sub-total :				**$ -**	0.0%		
Development							
Other - Development	Cost	0	$ -	$ -		-	-
Sub-total :				**$ -**	0.0%		
Engineering							
Other - Engineering	Cost	0	$ -	$ -		-	-
Sub-total :				**$ -**	0.0%		
Energy Equipment							
Solar collector	m²	15.0	$ 200	$ 3,000		-	-
Solar storage tank	L	1,135		$ -		-	-
Solar loop piping materials	m	19	$ 8.00	$ 155		-	-
Circulating pump(s)	W	97	$ 1.10	$ 107		-	-
Heat exchanger	kW	8.3	$ 15	$ 125		-	-
Transportation	project	1	$ 100	$ 100		-	-
Other - Energy equipment	Cost	0	$ -	$ -		-	-
Sub-total :				**$ 3,486**	62.4%		
Balance of System							
Collector support structure	m²	15.0	$ 70	$ 1,050		-	-
Plumbing and control	project	1	$ 220	$ 220		-	-
Collector installation	m²	15.0	$ 10	$ 150		-	-
Solar loop installation	m	19	$ 4.00	$ 77		-	-
Auxiliary equipment installation	project	1	$ 50	$ 50		-	-
Transportation	project	1	$ 50	$ 50		-	-
Other - Balance of system	Cost	0	$ -	$ -		-	-
Sub-total :				**$ 1,597**	28.6%		
Miscellaneous							
Training	p-h	4	$ 60	$ 240		-	-
Contingencies	%	5%	$ 5,323	$ 266		-	-
Sub-total :				**$ 506**	9.1%		
Initial Costs - Total				**$ 5,589**	100.0%		

Annual Costs (Credits)	Unit	Quantity	Unit Cost	Amount	Relative Costs	Quantity Range	Unit Cost Range
O&M							
Property taxes/Insurance	project	0	$ -	$ -		-	-
O&M labour	project	1	$ 15	$ 15		-	-
Other - O&M	Cost	0	$ -	$ -		-	-
Contingencies	%	5%	$ 15	$ 1		-	-
Sub-total :				**$ 16**	48.2%		
Electricity	kWh	217	**$ 0.0780**	**$ 17**	51.8%	-	-
Annual Costs - Total				$ 33	100.0%		

Periodic Costs (Credits)		Period	Unit Cost	Amount		Interval Range	Unit Cost Range
Valves and fittings	Cost	10 yr	$ 250	**$ 250**		-	-
				$ -		-	-
				$ -		-	-
End of project life		-		**$ -**			*Go to GHG Analysis sheet*

RETScreen® Financial Summary - Solar Water Heating Project

Annual Energy Balance

Project name		LA Fitness	Electricity required	MWh	0.22
Project location		Houston, TX			
Renewable energy delivered	MWh	9.95	Net GHG reduction	t_{CO2}/yr	3.94
Heating fuel displaced	-	ıtural gas - mmBtu	Net GHG emission reduction - 20 yrs	t_{CO2}	78.76

Financial Parameters

Avoided cost of heating energy	$/mmBtu	14.480	Debt ratio	%	0.0%
GHG emission reduction credit	$/$t_{CO2}$	-	Income tax analysis?	yes/no	No
Retail price of electricity	$/kWh	0.078			
Energy cost escalation rate	%	3.0%			
Inflation	%	2.0%			
Discount rate	%	6.0%			
Project life	yr	20			

Project Costs and Savings

Initial Costs				**Annual Costs and Debt**		
Feasibility study	0.0%	$	-	O&M	$	16
Development	0.0%	$	-	Electricity	$	17
Engineering	0.0%	$	-			
Energy equipment	62.4%	$	3,486	**Annual Costs and Debt -Total**	**$**	**33**
Balance of system	28.6%	$	1,597			
Miscellaneous	9.1%	$	506	**Annual Savings or Income**		
Initial Costs - Total	100.0%	**$**	**5,589**	Heating energy savings/income	$	983
Incentives/Grants		$				
				Annual Savings - Total	**$**	**983**
Periodic Costs (Credits)						
Valves and fittings		$	250	Schedule yr # 10,20		
		$	-			
		$	-			
End of project life -		$	-			

Financial Feasibility

Pre-tax IRR and ROI	%	19.4%	Calculate GHG reduction cost?	yes/no	No
After-tax IRR and ROI	%	19.4%			
Simple Payback	yr	5.9	Project equity	$	5,589
Year-to-positive cash flow	yr	5.3			
Net Present Value - NPV	$	8,398			
Annual Life Cycle Savings	$	732			
Benefit-Cost (B-C) ratio	-	2.50			

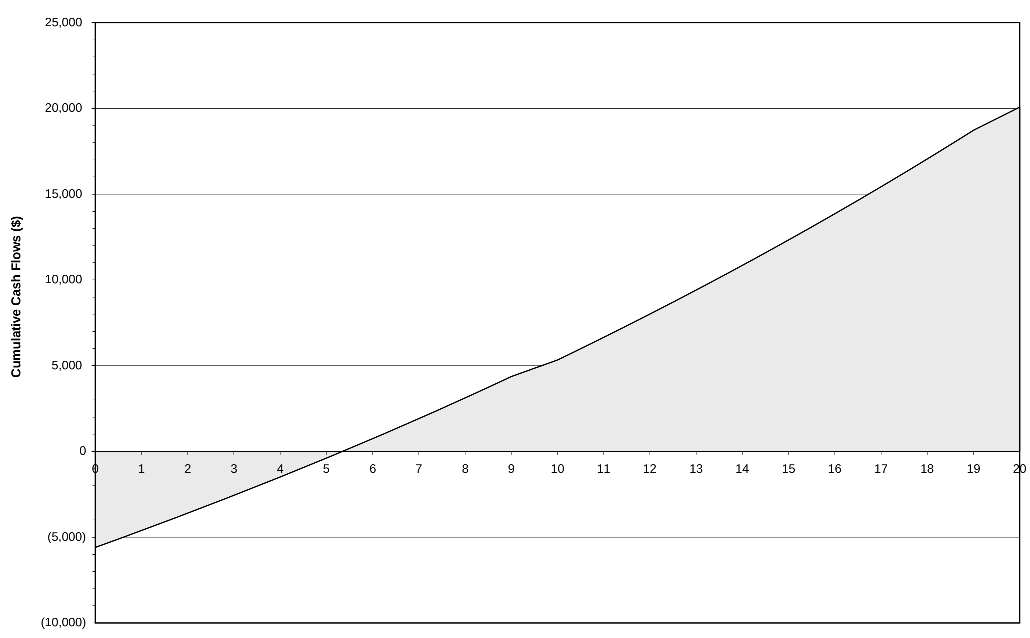
Yearly Cash Flows

Year #	Pre-tax $	After-tax $	Cumulative $
0	(5,589)	(5,589)	(5,589)
1	979	979	(4,610)
2	1,008	1,008	(3,602)
3	1,039	1,039	(2,563)
4	1,070	1,070	(1,493)
5	1,103	1,103	(390)
6	1,136	1,136	746
7	1,170	1,170	1,916
8	1,205	1,205	3,121
9	1,242	1,242	4,363
10	974	974	5,337
11	1,318	1,318	6,655
12	1,357	1,357	8,012
13	1,398	1,398	9,410
14	1,440	1,440	10,851
15	1,484	1,484	12,335
16	1,529	1,529	13,863
17	1,575	1,575	15,438
18	1,622	1,622	17,060
19	1,671	1,671	18,731
20	1,350	1,350	20,081

Cumulative Cash Flows Graph

SWH Project Cumulative Cash Flows
LA Fitness, Houston, TX

Renewable energy delivered (MWh/yr): 9.95 **Total Initial Costs: $ 5,589** **Net average GHG reduction (t_{CO2}/yr): 3.94**



IRR and ROI: 19.4% **Year-to-positive cash flow: 5.3 yı** **Net Present Value: $ 8,398**

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RETScreen® Energy Model - Solar Water Heating Project

Training & Support

Site Conditions		Estimate	Notes/Range
Project name		LA Fitness	See Online Manual
Project location		Houston, TX	
Nearest location for weather data		Houston, TX	Complete SR&HL sheet
Annual solar radiation (tilted surface)	MWh/m²	1.71	
Annual average temperature	°C	19.9	-20.0 to 30.0
Annual average wind speed	m/s	3.8	
Desired load temperature	°C	49	
Hot water use	L/d	1,500	
Number of months analysed	month	12.00	
Energy demand for months analysed	MWh	18.55	

System Characteristics		Estimate	Notes/Range
Application type		Service hot water (with storage)	
Base Case Water Heating System			
Heating fuel type	-	**Natural gas - mmBtu**	
Water heating system seasonal efficiency	%	50%	50% to 190%
Solar Collector			
Collector type	-	Evacuated	See Technical Note 1
Solar water heating collector manufacturer		Thermomax	See Product Database
Solar water heating collector model		Mazdon 20 - TMA 600S	
Gross area of one collector	m²	3.03	1.00 to 5.00
Aperture area of one collector	m²	2.14	1.00 to 5.00
Fr (tau alpha) coefficient	-	0.54	0.40 to 0.80
Fr UL coefficient	(W/m²)/°C	1.07	0.30 to 3.00
Temperature coefficient for Fr UL	(W/(m·°C)²)	0.00	0.000 to 0.010
Suggested number of collectors		7	
Number of collectors		7	
Total gross collector area	m²	21.2	
Storage			
Ratio of storage capacity to coll. area	L/m²	66.3	37.5 to 100.0
Storage capacity	L	993	
Balance of System			
Heat exchanger/antifreeze protection	yes/no	Yes	
Heat exchanger effectiveness	%	80%	50% to 85%
Suggested pipe diameter	mm	13	8 to 25 or PVC 35 to 50
Pipe diameter	mm	12	8 to 25 or PVC 35 to 50
Pumping power per collector area	W/m²	7	3 to 22, or 0
Piping and solar tank losses	%	6%	1% to 10%
Losses due to snow and/or dirt	%	2%	2% to 10%
Horz. dist. from mech. room to collector	m	5	5 to 20
# of floors from mech. room to collector	-	1	0 to 20

Annual Energy Production (12.00 months analysed)		Estimate	Notes/Range
SWH system capacity	kW_{th}	10	
	MWth	0.010	
Pumping energy (electricity)	MWh	**0.36**	
Specific yield	kWh/m²	598	
System efficiency	%	35%	
Solar fraction	%	68%	
Renewable energy delivered	MWh	**12.68**	
	million Btu	43.26	
			Complete Cost Analysis sheet

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RETScreen® Solar Resource and Heating Load Calculation - Solar Water Heating Project

Site Latitude and Collector Orientation		Estimate	Notes/Range
Nearest location for weather data		Houston, TX	See Weather Database
Latitude of project location	°N	30.0	-90.0 to 90.0
Slope of solar collector	°	30.0	0.0 to 90.0
Azimuth of solar collector	°	0.0	0.0 to 180.0

Monthly Inputs

(Note: 1. Cells in grey are not used for energy calculations; 2. Revisit this table to check that all required inputs are filled if you change system type or solar collector type or pool type, or method for calculating cold water temperature).

Month	Fraction of month used (0 - 1)	Monthly average daily radiation on horizontal surface (kWh/m²/d)	Monthly average temperature (°C)	Monthly average relative humidity (%)	Monthly average wind speed (m/s)	Monthly average daily radiation in plane of solar collector (kWh/m²/d)
January	1.00	2.66	10.4	74.6	4.1	3.60
February	1.00	3.42	12.2	73.4	4.3	4.16
March	1.00	4.25	16.3	72.7	4.5	4.65
April	1.00	5.01	20.4	74.1	4.4	4.93
May	1.00	5.62	23.8	75.5	4.0	5.11
June	1.00	6.02	26.7	75.0	3.6	5.27
July	1.00	5.95	27.8	74.7	3.2	5.29
August	1.00	5.61	27.6	75.1	3.0	5.34
September	1.00	4.87	25.3	76.3	3.3	5.14
October	1.00	4.19	20.7	74.1	3.4	5.04
November	1.00	3.07	16.1	75.3	3.8	4.13
December	1.00	2.51	12.0	74.9	3.9	3.51

		Annual	Season of Use
Solar radiation (horizontal)	MWh/m²	1.62	1.62
Solar radiation (tilted surface)	MWh/m²	1.71	1.71
Average temperature	°C	19.9	19.9
Average wind speed	m/s	3.8	3.8

Water Heating Load Calculation		Estimate	Notes/Range
Application type	-	Service hot water	
System configuration	-	With storage	
Building or load type	-	School	
Number of units	Student	800	
Rate of occupancy	%	80%	50% to 100%
Estimated hot water use (at ~60 °C)	L/d	1,472	
Hot water use	L/d	1,500	
Desired water temperature	°C	49	
Days per week system is used	d	7	1 to 7
Cold water temperature	-	Auto	
Minimum	°C	16.6	1.0 to 10.0
Maximum	°C	22.7	5.0 to 15.0
Months SWH system in use	month	12.00	
Energy demand for months analysed	MWh	18.55	
	million Btu	63.28	

Return to Energy Model sheet

RETScreen® Cost Analysis - Solar Water Heating Project

Type of project: Pre-feasibility | Currency: $ | Cost references: None

Initial Costs (Credits)	Unit	Quantity	Unit Cost	Amount	Relative Costs	Quantity Range	Unit Cost Range
Feasibility Study							
Other - Feasibility study	Cost	0	$ -	$ -		-	-
Sub-total :				**$ -**	0.0%		
Development							
Other - Development	Cost	0	$ -	$ -		-	-
Sub-total :				**$ -**	0.0%		
Engineering							
Other - Engineering	Cost	0	$ -	$ -		-	-
Sub-total :				**$ -**	0.0%		
Energy Equipment							
Solar collector	m²	21.2	$ 575	$ 12,196		-	-
Solar storage tank	L	993		$ -		-	-
Solar loop piping materials	m	19	$ 10.00	$ 193		-	-
Circulating pump(s)	W	105	$ 1.10	$ 115		-	-
Heat exchanger	kW	9.0	$ 15	$ 135		-	-
Transportation	project	1	$ 100	$ 100		-	-
Other - Energy equipment	Cost	0	$ -	$ -		-	-
Sub-total :				**$ 12,739**	76.4%		
Balance of System							
Collector support structure	m²	21.2	$ 70	$ 1,485		-	-
Plumbing and control	project	1	$ 300	$ 300		-	-
Collector installation	m²	21.2	$ 10	$ 212		-	-
Solar loop installation	m	19	$ 4.00	$ 77		-	-
Auxiliary equipment installation	project	1	$ 50	$ 50		-	-
Transportation	project	1	$ 50	$ 50		-	-
Other - Balance of system	Cost	0	$ -	$ -		-	-
Sub-total :				**$ 2,174**	13.0%		
Miscellaneous							
Training	p-h	4	$ 60	$ 240		-	-
Contingencies	%	10%	$ 15,153	$ 1,515		-	-
Sub-total :				**$ 1,755**	10.5%		
Initial Costs - Total				**$ 16,669**	100.0%		

Annual Costs (Credits)	Unit	Quantity	Unit Cost	Amount	Relative Costs	Quantity Range	Unit Cost Range
O&M							
Property taxes/Insurance	project	0	$ -	$ -		-	-
O&M labour	project	1	$ 15	$ 15		-	-
Other - O&M	Cost	0	$ -	$ -		-	-
Contingencies	%	10%	$ 15	$ 2		-	-
Sub-total :				**$ 17**	36.8%		
Electricity	kWh	364	**$ 0.0780**	**$ 28**	63.2%	-	-
Annual Costs - Total				$ 45	100.0%		

Periodic Costs (Credits)		Period	Unit Cost	Amount		Interval Range	Unit Cost Range
Valves and fittings	Cost	10 yr	$ 250	**$ 250**		-	-
				$ -		-	-
				$ -		-	-
End of project life		-		**$ -**			Go to GHG Analysis sheet

RETScreen® Financial Summary - Solar Water Heating Project

Annual Energy Balance

Project name		LA Fitness	Electricity required	MWh	0.36
Project location		Houston, TX			
Renewable energy delivered	MWh	12.68			
Heating fuel displaced	-	ıtural gas - mmBtu			

Financial Parameters

Avoided cost of heating energy	$/mmBtu	14.480	Debt ratio	%	0.0%
			Income tax analysis?	yes/no	No
Retail price of electricity	$/kWh	0.078			
Energy cost escalation rate	%	3.0%			
Inflation	%	2.0%			
Discount rate	%	6.0%			
Project life	yr	20			

Project Costs and Savings

Initial Costs				**Annual Costs and Debt**		
Feasibility study	0.0%	$	-	O&M	$	17
Development	0.0%	$	-	Electricity	$	28
Engineering	0.0%	$	-			
Energy equipment	76.4%	$	12,739	**Annual Costs and Debt -Total**	**$**	**45**
Balance of system	13.0%	$	2,174			
Miscellaneous	10.5%	$	1,755	**Annual Savings or Income**		
Initial Costs - Total	100.0%	**$**	**16,669**	Heating energy savings/income	$	1,253
Incentives/Grants		$	-			
				Annual Savings - Total	**$**	**1,253**
Periodic Costs (Credits)						
Valves and fittings		$	250	Schedule yr # 10,20		
		$	-			
		$	-			
End of project life -		$	-			

Financial Feasibility

Pre-tax IRR and ROI	%	6.8%			
After-tax IRR and ROI	%	6.8%			
Simple Payback	yr	13.8	Project equity	$	16,669
Year-to-positive cash flow	yr	11.6			
Net Present Value - NPV	$	1,185			
Annual Life Cycle Savings	$	103			
Benefit-Cost (B-C) ratio	-	1.07			

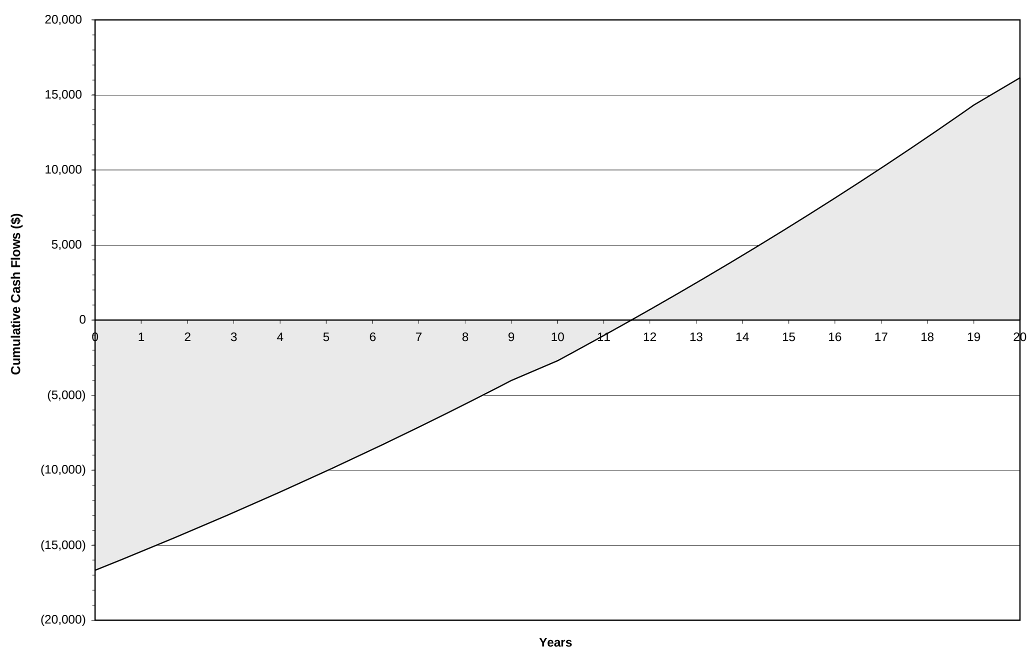
Yearly Cash Flows

Year #	Pre-tax $	After-tax $	Cumulative $
0	(16,669)	(16,669)	(16,669)
1	1,244	1,244	(15,424)
2	1,282	1,282	(14,142)
3	1,320	1,320	(12,822)
4	1,360	1,360	(11,462)
5	1,401	1,401	(10,060)
6	1,443	1,443	(8,617)
7	1,487	1,487	(7,130)
8	1,532	1,532	(5,598)
9	1,578	1,578	(4,020)
10	1,321	1,321	(2,699)
11	1,674	1,674	(1,025)
12	1,725	1,725	700
13	1,777	1,777	2,477
14	1,830	1,830	4,307
15	1,885	1,885	6,193
16	1,942	1,942	8,135
17	2,001	2,001	10,136
18	2,061	2,061	12,197
19	2,123	2,123	14,320
20	1,816	1,816	16,135

Cumulative Cash Flows Graph

SWH Project Cumulative Cash Flows
LA Fitness, Houston, TX

Renewable energy delivered (MWh/yr): 12.68 **Total Initial Costs: $ 16,669**



IRR and ROI: 6.8% **Year-to-positive cash flow: 11.6 yı** **Net Present Value: $ 1,185**

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Appendix F: Lighting Power Density Spreadsheets

Existing Design:

Lighting Power Density Calculation				
Space	**Lamp ID**	**Number**	**Wattage/Type**	**Watts**
Aerobics	A	9	128	1152
	AX	9	128	1152
	BX	1	64	64
	H	5	32	160
Raquetball	D	10	192	9600
Storage	B	3	64	192
Restrooms	B	2	64	128
Kid's Club	A	8	128	1024
	AX	4	128	512
Free Weights	A	15	128	1920
	AX	5	128	640
Basketball	R	8	400	3200
	RX	4	400	1600
Storage	C	2	64	128
Special Exercise	A	13	128	1664
	AX	4	128	512
Equipment Room	C	2	64	128
Cardiovascular	A	39	128	4992
	AX	14	128	1792
	AA2	15	45	675
	B	19	64	1216
	BX	8	64	512
Mezzanine	B	19	64	1216
	BX	5	64	320
Spinning	B	17	64	1088
	BX	4	64	256
Pool Equipment	C	3	64	192
Pool and Spa	K	6	130	780
	L	15	298	4470
	L1	6	220	1320
	Y	10	190	1900
Locker Rooms	B	4	64	256
	G	36	64	2304
	H	12	32	384
	HX	12	32	384
	J	6	26	156
	P	8	28	224
	Z	18	36	648
Reception	K	8	130	1040
	V	2	130	260
	Y	5	190	950
Membership Sales	A	2	128	256
	AX	2	128	256
	AA2	10	45	450
Juice Bar	B	3	64	192
Stairs	E	26	32	832
Total			53097	W
Lighting Power Density			1.18	W/ft^2

Lighting Redesign:

Lighting Power Density Calculation				
Spaces	**Lamp ID**	**Number**	**Wattage/Type**	**Watts**
Aerobics	A	9	112	1008
	AX	9	112	1008
	BX	1	56	56
	H	5	32	160
Raquetball	D	6	168	5040
Storage	B	3	56	168
Restrooms	B	2	56	112
Kid's Club	A	8	112	896
	AX	4	112	448
Free Weights	A	15	112	1680
	AX	5	112	560
Basketball	R	8	350	2800
	RX	4	359	1436
Storage	C	2	56	112
Special Exercise	A	13	112	1456
	AX	4	112	448
Equipment Room	C	2	56	112
Cardiovascular	A	39	112	4368
	AX	14	112	1568
	AA2	15	45	675
	B	19	56	1064
	BX	8	56	448
Mezzanine	B	19	56	1064
	BX	5	56	280
Spinning	B	17	56	952
	BX	4	56	224
Pool Equipment	C	3	56	168
Pool and Spa	K	6	130	780
	L	15	298	4470
	L1	6	220	1320
	Y	10	190	1900
Locker Rooms	B	4	56	224
	G	36	56	2016
	H	12	32	384
	HX	12	32	384
	J	6	26	156
	P	8	28	224
	Z	18	36	648
Reception	K	8	130	1040
	V	2	130	260
	Y	5	190	950
Membership Sales	A	2	112	224
	AX	2	112	224
	AA2	10	45	450
Juice Bar	B	3	56	168
Stairs	E	13	28	364
Total			44497	W
Lighting Power Density			0.989	W/ft^2

Return to: Octron 800 XP

Print Page

Product Number: 21763

Order Abbreviation: FO32/835/XP/ECO

General Description: 32W, 48" MOL, T8 OCTRON XP Extended Performance fluorescent lamp, 3500K color temperature, rare earth phosphor, 85 CRI, suitable for IS or RS operation, ECOLOGIC

Product Information	
Abbrev. With Packaging Info.	FO32835XPECO 30/CS 1/SKU
Actual Length (in)	47.78
Actual Length (mm)	1213.6
Average Rated Life (hr)	24000
Base	Medium Bipin
Bulb	T8
Color Rendering Index (CRI)	85
Color Temperature/CCT (K)	3500
Diameter (in)	1.10
Diameter (mm)	27.9
Family Brand Name	OCTRON® 800 XP®, ECOLOGIC®
Industry Standards	ANSI C78.81 - 2001
Initial Lumens at 25C	3000
Mean Lumens at 25C	2850
Nominal Length (in)	48
Nominal Wattage (W)	32.00

Additional Product Information

Product Documents, Graphs, and Images

Compatible Ballast

Packaging Information

Footnotes

- Approximate initial lumens after 100 hours operation.

Return to: Octron 800 XP Supersaver | Print Page

Product Number:	22060
Order Abbreviation:	FO30/835/XP/SS/ECO
General Description:	30W, 48" MOL, T8 OCTRON SuperSaver fluorescent lamp, 3500K color temperature, rare earth phosphor, 82 CRI, suitable for use on instant ballasts or other T8 ballasts with minimum starting voltage of 550V, ECOLOGIC

Product Information	
Abbrev. With Packaging Info.	FO30835XPSSECO 30/CS 1/SKU
Actual Length (in)	47.78
Actual Length (mm)	1213.6
Average Rated Life (hr)	18000
Base	Medium Bipin
Bulb	T8
Color Rendering Index (CRI)	82
Color Temperature/CCT (K)	3500
Diameter (in)	1.1
Diameter (mm)	27.9
Family Brand Name	OCTRON® 800 XP® SS, ECOLOGIC®
Industry Standards	ANSI C78.81 - 2001
Initial Lumens at 25C	2850
Mean Lumens at 25C	2710
Nominal Length (in)	48
Nominal Wattage (W)	30.00

Additional Product Information

Product Documents, Graphs, and Images

Packaging Information

ECOLOGIC TM

Footnotes

- This lamp may also be operated by the OSRAM SYLVANIA QUICKTRONIC(R) PSN ballast (.88 BF), or the QUICTRONIC PSX ballast (.71 BF).
- Approximate initial lumens after 100 hours operation.
- The life ratings of fluorescent lamps are based on 3 hr. burning cycles under specified conditions and with ballast meeting ANSI specifications. If burning cycle is increased, there will be a corresponding increase in the average hours life.

Appendix G: Desiccant Integration, Performance, & Further Discussion

Design Integration

The addition of the desiccant dehumidification unit to the roof meant that duct work had to be run to each of the rooftop units. The duct runs were kept as short as possible to minimize cost; ductwork for exterior spaces has a higher first cost because it needs to be weather resistant. Figure G.1 shows a rough layout of the duct runs. These runs were used to estimate the first cost of a weatherproof duct system for the economic analysis.

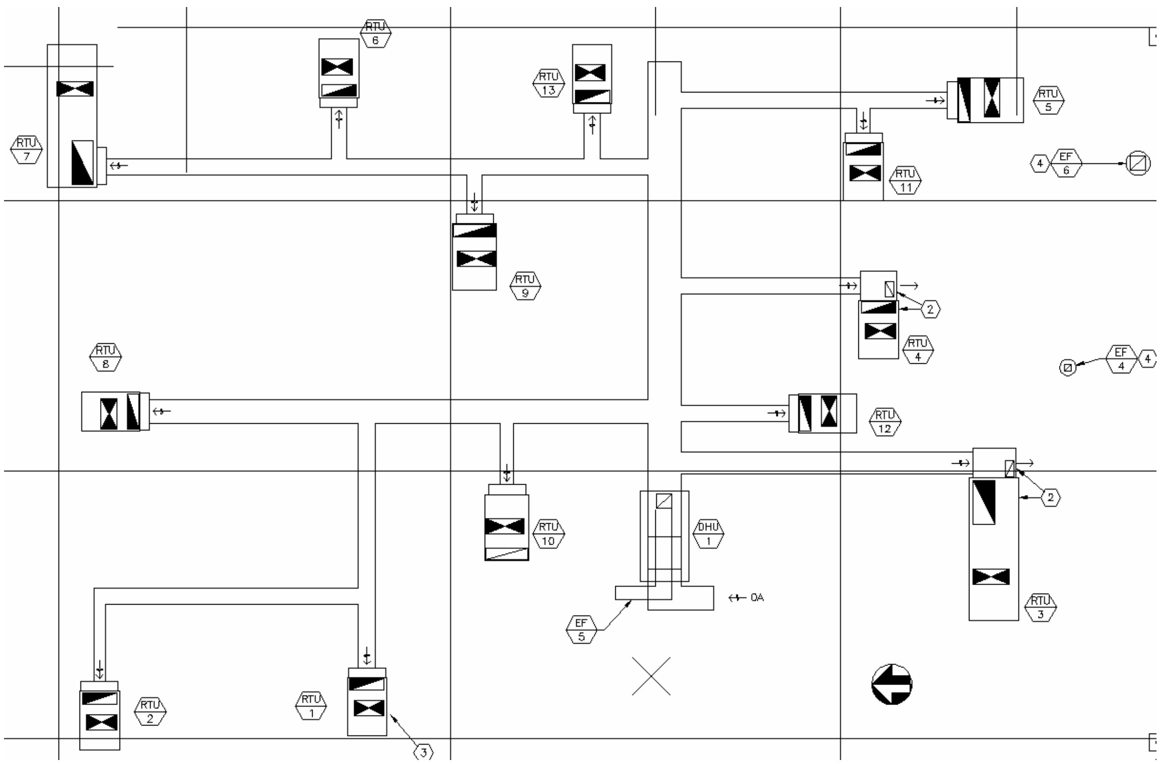

Figure G.1 – Roof Plan Including Desiccant Dehumidification Unit

ASHRAE Standard 62.1 requires that the exhaust outdoor air intakes should be kept 15 feet away from the exhaust outlet. The design provided allows for a 17 ft clearance between the exhaust air outlet and the outdoor air intake. Also, the prevailing winds in Houston are primarily from the south. The outdoor air intake opening is oriented to the south to take advantage of these winds; similarly, the exhaust outlet faces due north for the same advantage.

As each rooftop unit comes installed with fans, the largest fan that needed to be sized for this system only needs to provide enough pressure to overcome the longest roof duct run. This longest duct run for this building is 150 feet.

Performance:

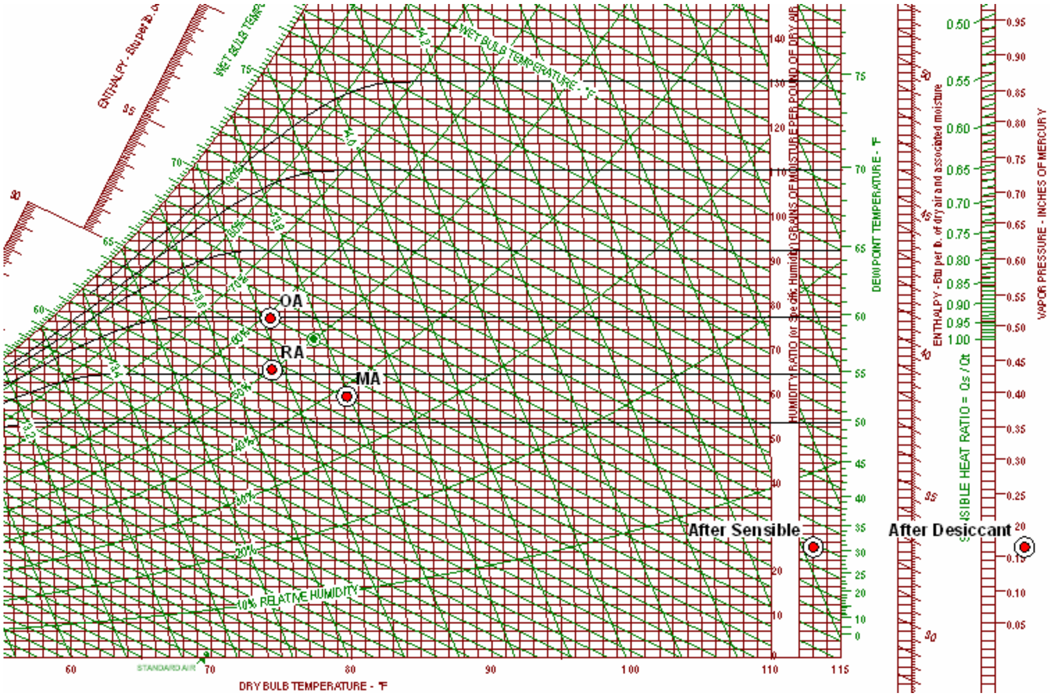

Figure G.2 – Desiccant System Typical January State Points

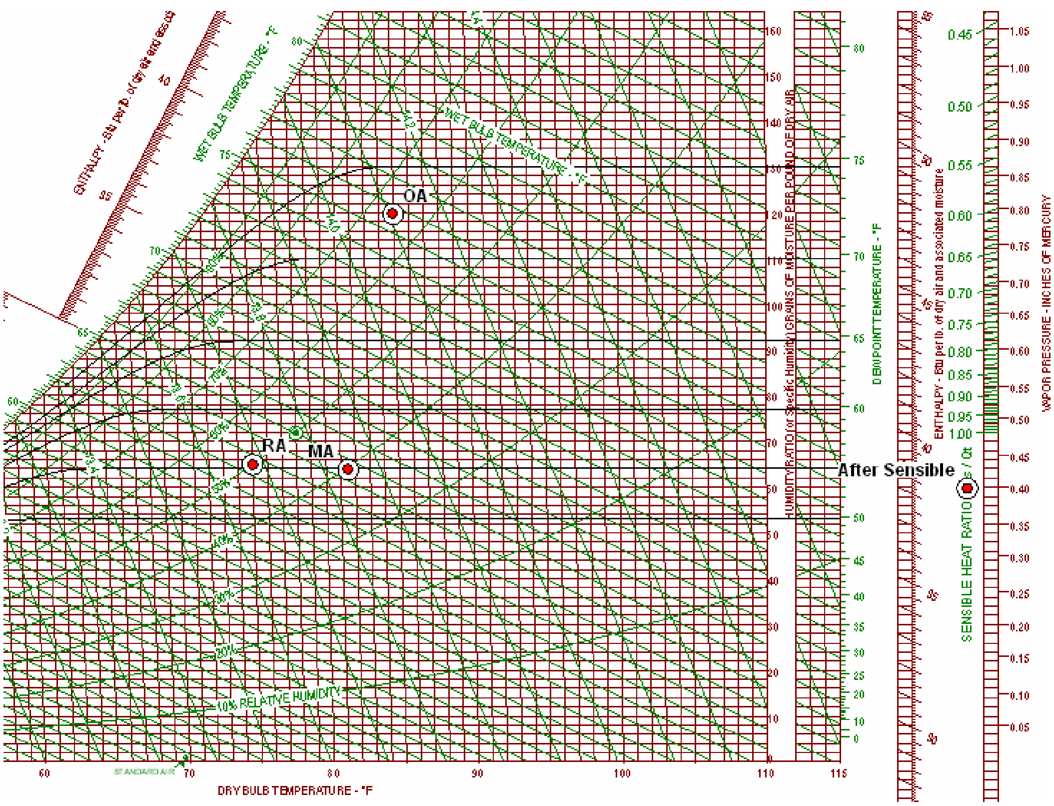

Figure G.3 – Desiccant System Typical April State Points

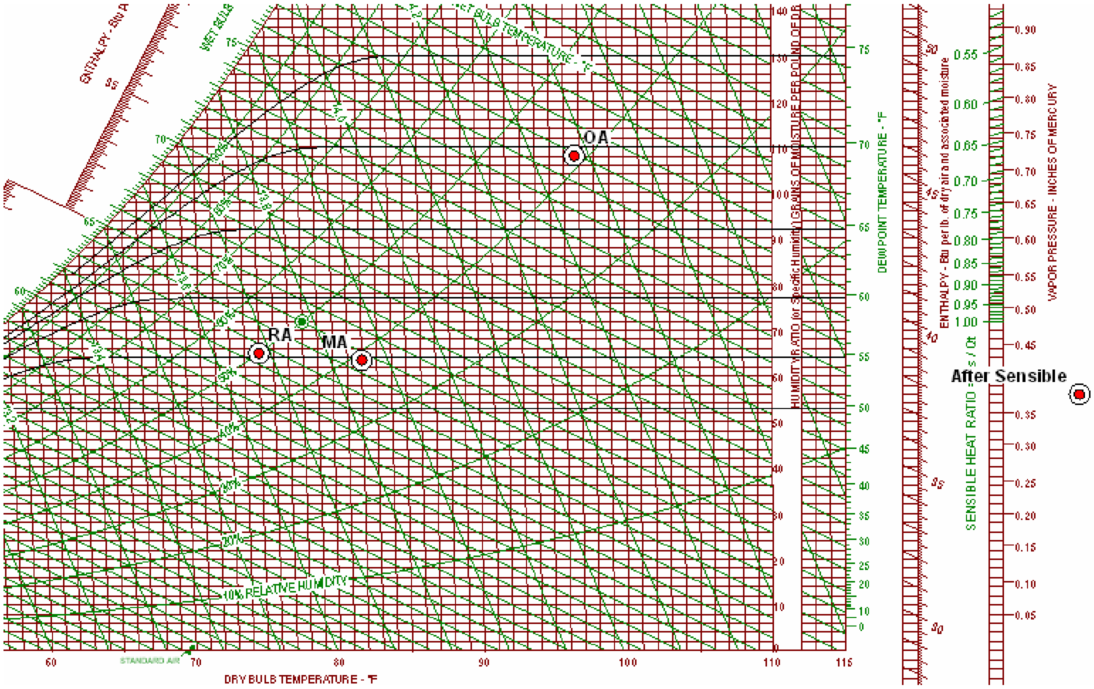

Figure G.4 – Desiccant System Typical July State Points

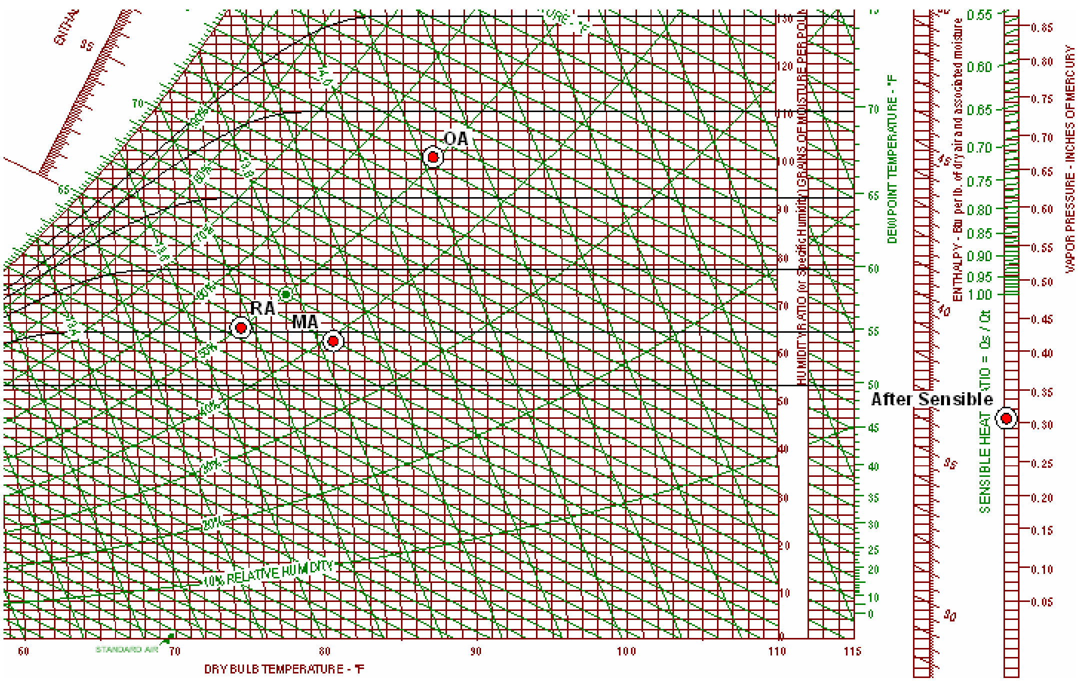

Figure G.5 – Desiccant System Typical October State Points