Development of Renewable Energy

Design & Simulation of Closed-loop Active Solar Thermal System

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A Senior Project Submitted

In Partial Fulfillment of the Requirements for the Degree of Bachelor of Science in

Renewable Energy Engineering

Oregon Institute of Technology
Wilsonville, OR

Abstract

Renewable energy is more popular today than it was yesterday. Solar thermal is one form that is used towards hot water and space heating in residential and commercial applications. When designed appropriately, a solar thermal system can provide most of the annual hot water needs and effectively offset fossil fuel use.

Once a system is designed, it is beneficial to simulate the system. Simulations provide a method of accurately estimating the performance of the system without the need of testing a prototype. Unfortunately most systems are not simulated prior to construction and are assumed to operate as expected.

An active solar thermal system was designed using the f-chart method, without concern of cost. The system consisted of six evacuated tube collectors that used a 30% propylene glycol solution as the working fluid. The fluid ran through a heat exchanger, which passed the solar energy to a water storage tank. The glycol and water fluid loops had a pump to provide circulation. The pumps were controlled by a differential ON/OFF control algorithm.

The system was simulated using Matlab 2014a by calculating the hourly storage temperature. The storage temperature was found by determining the hourly solar energy collected and subtracting the energy used by the load and the storage tank losses over each hour. The storage tank temperature was considered over July 21 and December 21 for a home in Medford Oregon. The days were chosen to simplify the simulation by estimating performance at the climate extremes of the area.

The proposed solar thermal system was successfully designed and simulated using straightforward methods. The process showed that not only design, but also simulations can be done in a simple manner to produce a system that will meet demands and operate as expected.

Solar thermal is a growing technology, and the system design can grow as well. It can be expanded upon to include subsystems like solar cooling or heat dumps. Components like the load heat exchanger can be removed and replaced with radiant floor heating or another form of energy delivery. The simulation can also be altered to show weeks or even a year for study of long term operation.

The proposed system can benefit areas like Medford where solar thermal and renewable energy in general are underutilized. Solar thermal technology can ultimately help offset traditional fossil fuel energy sources and support the environment and a healthy life.

Acknowledgements

I would like to thank Charlie Whitley and Spring Air Inc., for providing permission and time to use their Right-Suite Universal 2012 license, without which this project would not have been possible.

Nomenclature

A	System altitude [km]
A_i	Inside pipe surface area [m^2]
A_o	Outside pipe surface area [m^2]
a_0	Atmospheric constant
a_1	Atmospheric constant
C^*	Maximum to minimum capacitance rate ratio
C_p	Fluid heat capacity [J/kg·°C]
D_i	Inside pipe diameter [m]
D_o	Outside pipe diameter [m]
F'	Collector efficiency factor
F_R	Collector heat removal factor
F_R'	Collector heat exchanger efficiency factor
$F_R(\tau\alpha)$	Collector Optical Efficiency
F_RU_L	Collector Loss Coefficient [W/m^2·°C]
G	Irradiance [W/m^2]
G_b	Beam irradiance [W/m^2]
G_d	Diffuse irradiance [W/m^2]
G_{on}	Extraterrestrial radiation flux [W/m^2]
G_{sc}	Solar constant [W/m^2]
G_T	Solar Irradiance on a tilted surface [W/m^2]
H_T	Daily irradiance [J/m^2·day]
I_T	Hourly irradiance [J/m^2·h]
K	thermal conductivity [W/m·°C]
K	atmospheric constant
$k_{\tau\alpha}$	Incidence angle modifier
L	Daily total heating load [J]
$\dot{m}$	Mass flow rate [kg/s]
$\dot{m}\ C_p$	Fluid capacitance rate [W/m^2·°C]
n	Day number
NTU	Number of transfer units
Q_u	Useable Energy [W]
r	Working fluid/flowrate correction factor
R_b	Ratio of beam radiation on a tilted surface to horizontal
ST	Solar time [h]
T_a	Ambient temperature [°C]
$\bar{T}_a$	Daily average ambient temperature [°C]
T_i	Collector inlet temperature [°C]
T_h	Home setpoint temperature [°C]
T_r	Water storage tank room temperature [°C]
T_{ref}	Empirically derived reference temperature [°C]
T_s	Water storage temperature [°C]
T_s^+	Water storage temperature after hourly energy transfer [°C]
U_d	Pipe loss coefficient [W/m^2·°C]
U_L	Collector heat loss coefficient [W/m^2·°C]
U_L'	Modified collector heat loss coefficient [W/m^2·°C]
$(UA)_{hx}$	Overall heat transfer-area product of heat exchanger (W/°C)
$(UA)_s$	Overall heat transfer-area product of storage (W/°C)
X	Collector losses to heating load ratio
Y	Absorbed solar radiation to heating load ratio
β	Collector tilt angle [°]

δ	Solar declination [°]
ΔT	Hour time step [h]
Δt	Day time step [sec]
ε	Heat exchanger effectiveness
η	Collector efficiency
θ_z	Zenith angle [°]
φ	System latitude [°]
ρ_g	Ground reflectance
τ_b	Atmospheric transmittance for beam radiation
τ_d	Atmospheric transmittance for diffuse radiation
$(\tau\alpha)$	Transmittance-absorptance product
$(\tau\alpha)'$	Modified transmittance-absorptance product
ω	Hour angle [°]
$()_c$	Collector
$()_{max}$	Maximum
$()_{min}$	Minimum
$[]_{test}$	SRCC collector test parameters
$[]_{use}$	Collector parameters in system

Introduction

Renewable energy is more popular today than it ever has been before. People want to better the environment and diminish their need of dwindling fossil fuel sources. In the average American home, heating the home and water for showers and laundry is one of the largest uses of energy. Americans can offset their energy use greatly with solar thermal technology.

Solar thermal systems use energy collected from the sun to heat water for many applications, most popular being pool heating and domestic hot water (DHW) heating. Following these uses of solar hot water, solar thermal can be designed to provide space heating as well. Since solar thermal systems rely on solar radiation, they work better in some areas than others. This requires careful system design and simulation to assess a solar thermal system's performance and feasibility. It is much cheaper and easier to run a simulation than it is to physically build and test the system at the desired site or one with similar conditions.

Southern Oregon, specifically Medford Oregon, is an example of a colder climate that negatively affects system performance. Although summers provide plenty of solar energy, the colder months provide much less solar energy and ambient temperatures frequently drop below freezing during the night. Careful design however can negate some of the poor conditions in winter and lead to an effective system that will offset some of the traditional sources for heating.

Solar thermal technology is still in its infancy, and development lags behind more popular renewable energy technologies like photovoltaics and wind power. Areas like Medford would benefit from solar thermal system development to increase awareness and to help develop solar thermal as a whole. The interest in Medford and the current situation with solar thermal technology lead to the desire to design and simulate a system to determine potential in the area.

Design and simulation were done using Matlab 2014a to assess the performance of a solar thermal system in a residential building in Medford Oregon, which is shown in the appendix. This method can be used as a template for similar systems in any desired location by contractors. The template would prevent companies from having to start from scratch and allow them to get a better idea how their design will function relatively easy.

The system will provide DHW all year round and space heating when necessary. The building was assumed to be maintained at 21 °C (70 °F). Heating and DHW loads were supplied by Right-suite Universal 2012 and MSc respectively. Conducting a simulation of the system was an effective way to test the performance of the system without physically constructing it. Component selection in the design of the system was done using the f-chart method.

Background

Solar thermal technology is the collection of the sun's energy in a fluid medium for use. The collected thermal energy can then be used for building space heating, domestic water heating, industrial process heat, power production, and even refrigeration. This technology has been available since the early 1900s, but as of 2007 less than 0.4% of households in the United States use solar thermal technology (Schelly). These systems collect solar radiation through collectors and the working fluid is used to carry the energy to storage by either a pump or passive means. Solar thermal like so many other renewable technologies is still in the development stages and attractive only to few consumers, but also has begun to grow in popularity due to the general public's increased concern in the environment.

Space heating is one of the common applications of solar thermal energy today. Systems used for this purpose can come in many forms, but are most commonly active systems. Active systems use pumps to circulate a fluid through collectors to absorb thermal energy and to transport it to a storage tank to hold the energy until it is needed. When the energy is needed, the energy can be transferred to air or the fluid can be pumped through the space to radiate heat. Typically active systems are used in areas where freezing is a concern. Antifreeze mixtures typically of propylene glycol and water are used as the working fluid to protect the collector from pipe burst and other problems that could occur when the fluid freezes in cold environments. Propylene glycol also has the benefit of being nontoxic and safe for the environment. The collectors must then be in a closed loop to separate the heat transfer fluid from the water. Heat energy is transferred from the collector loop to the storage tank via a heat exchanger. The system shown in figure 1 is an example of a simple active solar thermal system.

Sizing a solar thermal system for space heating has some drawbacks. To get a high solar fraction in winter, the system will end up with too much energy in the summer. The extra energy can lead to overheating which can damage system components or breakdown the working fluid. The energy therefore needs to be transferred to another location known as a heat dump. The simplest method of heat dump would be to run the system at night. Known as "Nocturnal cooling", the system could be thought of as running in reverse, transferring heat from the tank to the air surrounding the collector. This method can bring the tank down to a temperature where it could utilize the following day's solar energy. Another popular method is to use a swimming pool or hot tub as a heat dump. These hold a large amount of water and are enjoyed when the water temperature is warmer. The other popular method is to have a radiator on the collectors in a secondary loop that the working fluid will pass through when the tank has reached a maximum value (Siegenthaler). An example of this is shown in figure 1 below.

Figure 1. Radiator solar heat dump (Homepower).

All systems when Implemented should have a form of heat dump employed. Unless the system is small, or an extremely large tank, high temperatures will more than likely be reached in the summer. Overheating can become more likely with a summer vacation or another situation where the occupants of the home are gone for an extended amount of time.

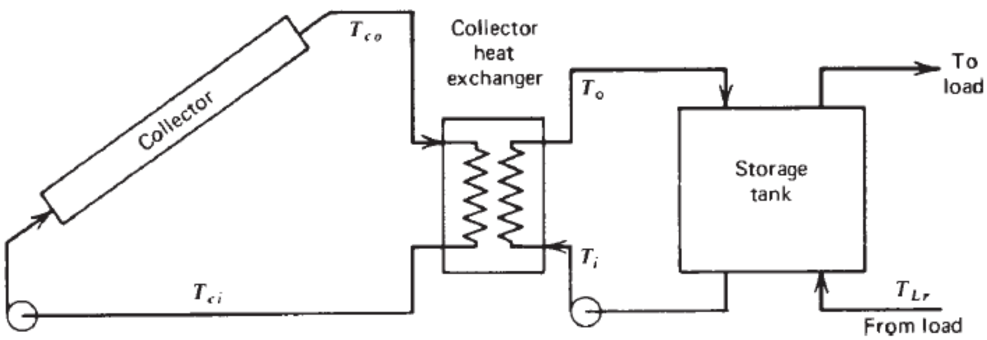

Figure 2. General system (Duffie)

The collector is the most important part of a solar thermal system. It is the component that collects the solar radiation as well as is the biggest impact on system performance. It's important to select the right collector for the climate that the system is in. Flat plate collectors have been the most popular because they have been around much longer than evacuated tube collectors and they are the cheapest with very little maintenance. They look similar to photovoltaic panels, but are essentially an insulated box with a glass cover and metal piping inside that carries the working fluid through the panel to collect the incoming solar radiation. Evacuated tube collectors however have been proven to be more efficient in cooler climates where the ambient temperature is lower, if there are cloudy days, or windy conditions. They use glass tubes around metal pipes that from a vacuum to greatly decrease heat loss due to conduction and convection. The collector is made of multiple tubes (Solar Choices). There is an example of an evacuated tube collector in figure 1 above.

There are three mechanisms of heat transfer that can lead to heat loss from a collector: radiation, conduction, and convection. Radiation is the release of electromagnetic waves. This form of heat loss cannot be addressed since the solar thermal system main working principle is absorbing solar radiation. To reduce heat loss due to radiation would mean to reduce the ability for the system to gain solar radiation. This form of heat loss is also the weakest and therefore can be neglected when compared to conduction and convection (Cengel).

Conduction is one of the major forms of heat loss that should be addressed in a solar thermal system. It is the thermal energy exchange between solids like through the storage tank walls or quiescent fluids like air at the surface of the collector. Conduction therefore cannot occur in vacuum since there is no air, like in the case of evacuated tube collectors (Cengel).

Convection is the principal form of heat loss in a solar thermal system and occurs in moving fluids. In the case of solar thermal systems, this will be the surrounding outside air, especially in windy conditions. As with conduction, the lack of fluid motion in vacuum negates convection (Cengel).

The cylindrical tubes of an evacuated tube collector also provide a level of passive solar tracking, since they allow the sun's beam radiation to be normal to the collector surface for a larger part of the day instead of just solar noon in the case of flat plate collectors. Evacuated tube collectors however lose their advantages when the climate is warmer, because their superior heat loss characteristics are not as prevalent (Solar Choices). Table 1 and Figure 3 & 4 below are visual examples of the collector comparison.

Table 1. Example comparison of flat plate and evacuated tube collectors

COMPARISON PERFORMANCE: FLAT PLATE TO EVACUATED TUBE (SRCC DATA)						
	BTUS PER PANEL PER DAY					
Temp/ condition	Clear day ~2,000 BTU/f2/day		Partly cloudy ~1,500 BTU/f2/day		Cloudy~1,000 BTU/f2/day	
	Flat plate	Evacuated	Flat plate	Evacuated	Flat plate	Evacuated
A. -9°F (Pool 1)	53,000	43,000	42,000	32,000	28,000	22,000
B. 9°F (Pool 2)	48,000	41,000	36,000	31,000	22,000	20,000
C. 36°F (DHW 1)	40,000	38,000	27,000	28,000	13,000	17,000
D. 90°F (DHW 2)	24,000	32,000	12,000	22,000	2,000	12,000
E. 144°F (A/C)	9,000	26,000	1,000	15,000	0	6,000

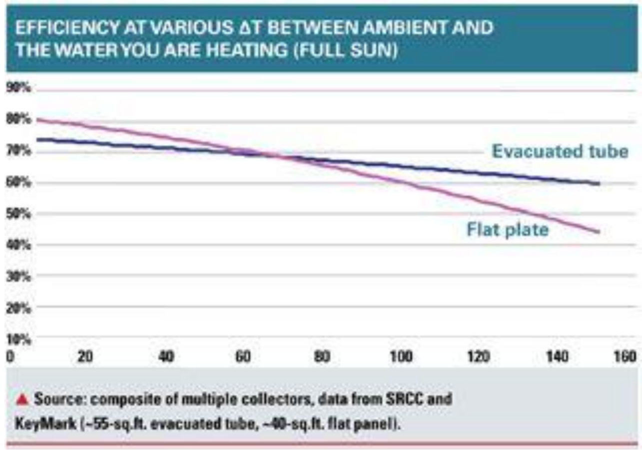

Figure 3. Collector comparison during sunny day.

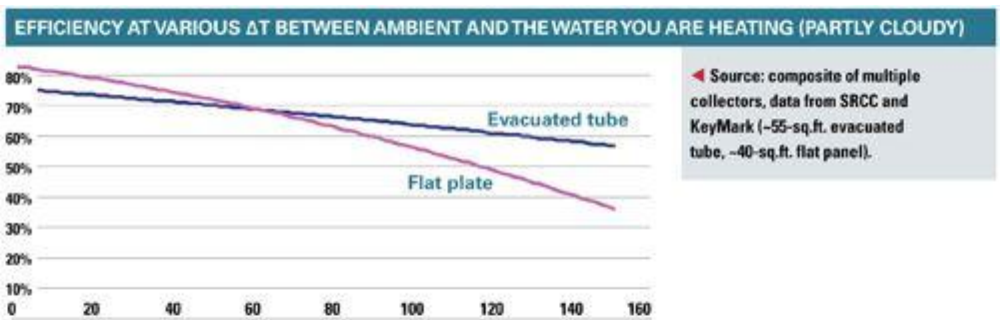

Figure 4. Collector comparison during cloudy day.

The above table and figures compare the same flat plate and evacuated tube collectors. They show that after a certain temperature difference between the collector inlet and ambient temperatures, the evacuated tube becomes more efficient. The point of equal efficiency will be more likely passed in cooler ambient temperatures. Figures 3 & 4 show that as the incident solar radiation gets weaker, the point of equal efficiency decreases, leading to evacuated tubes being more efficient for a wider temperature difference range (Solar Choices).

Active solar thermal systems for space heating applications have been proven to be effective in the past, as explained by Duffie, Salcines, and Schelly. These systems are commonly used for residential space heating, but can also be used for domestic hot water, and pool heating.

There are many methods for design of active solar thermal systems. Many were developed as performance predictions with what Duffie referred to as “short-cut” methods. These are meant for residential-scale applications and provide adequate estimations of long-term performance. According to Duffie, the f-chart method is used for estimating the solar fraction, the percent of load met by the solar energy. These systems can be active liquid or air heating systems for residential applications. The method also made the assumption that the minimum temperature of water delivery was near 20 °C (68 °F).

The f-chart method was developed from hundreds of thermal performance simulations of solar heating systems. The simulations led to correlations that were created from simulation conditions that were varied over ranges of parameters of practical system designs. This led to the solar fraction becoming a function of two dimensionless parameters. One of the dimensionless parameters, X, was related to the ratio of collector losses to heating loads, and the other, Y, is related to the ratio of absorbed solar radiation to heating loads (Duffie).

Although the f-chart method is based around the collector parameters, $F_R(\tau\alpha)$ and F_RU_L, it is important to account for the other components as well. The collector parameters can be modified to include piping and heat exchanger characteristics. Without these modifications, the parameters would make the system appear as just a collector array connected directly to a storage tank. The unmodified parameters therefore are initially provided by the SRCC for the collector itself (Directory).

The SRCC is the Solar Rating & Certification Corporation. As the name implies, the SRCC is a nationally recognized non-profit organization that tests solar thermal collectors. They provide test methods or evaluate independent test results for certification. Collectors that are certified give confidence to the user of the collector's credibility and durability. The certification process is based on minimum standards (Solar-rating).

The modified values are also important for a more accurate simulation. They add to the accuracy of storage temperature calculations, which was the basis of the simulation.

Matlab is one of many programs that can be used to simulate solar thermal systems. Developed by Mathworks, Matlab is a language used by engineers and scientists. The program allows users to complete tasks from solving systems of equations and other numeric computation to system modeling and algorithm development. It allows work to be done across many disciplines like signal processing and control systems (Mathworks).

Design

All equations associated with the design and simulation of the purposed system provided by Duffie.

Determining Solar radiation

The system components used are based on the available solar radiation, which is calculated in the equation below. The ground reflectance was assumed to be 0.2 since there was rarely any snowfall in Medford.

$$G_T = G_b R_b + G_d \left(\frac{1 + cos\beta}{2}\right) + G\rho_g \left(\frac{1 - cos\beta}{2}\right)$$

Before this equation can be used however, the variables in the above equation must first be found. First, the solar declination or angular position at solar noon was calculated using the following equation.

$$B = (n-1)\frac{360}{365}$$

$$\delta = \frac{180}{\pi}(0.006918 - 0.399912\cos B + 0.070257\sin B - 0.006758\cos 2B + 0.000907\sin 2B - 0.002697\cos 3B + 0.00148\sin 3B)$$

Where B is a dimensionless parameter. The extraterrestrial radiation flux was then found using the equation below.

$$G_{on} = G_{sc}(1.00010 + 0.034221\cos B + 0.001280\sin 2B + 0.000719\cos 2B + 0.000077\sin 2B)$$

Where the solar constant was assumed to be 1367 W/m^2, as adopted by the World Radiation Center (WRC) (Duffie).

Next the constants for the atmosphere with visibility of 23 km were calculated using the equation below.

$$a_0 = r_0(0.4237 - 0.00821(6-A)^2$$

$$a_1 = r_1(0.5055 - 0.00595(6.5-A)^2$$

$$k = r_k(0.2711 - 0.01858(2.5-A)^2$$

Where r_0, r_1, and r_k are correction factors for changes climate types from standard atmosphere are shown in table 2 below. The altitude for Medford, Oregon is 0.4 km.

Table 2. Climate type correction factors

Climate Type	r_0	r_1	r_k
Midaltitude Summer	0.97	0.99	1.02
Midaltitude Winter	1.03	1.01	1.00

The coefficients for the atmosphere were used to find the atmospheric transmittance for beam and diffuse radiation respectively.

$$\tau_b = a_0 + a_1\, exp\left(\frac{-k}{\cos\theta_z}\right)$$

$$\tau_d = 0.271 - 0.294\tau_b$$

Where the zenith angle and hour angle were found using the following equations respectively.

$$\cos\theta_z = \cos\varphi\cos\delta\cos\omega + \sin\varphi\sin\delta$$

$$\omega = 15\,(ST - 12)$$

Now the beam and diffuse irradiance were determined using the following equations.

$$G_b = G_{on}\cos\theta_z \tau_b$$

$$G_d = G_{on}\cos\theta_z \tau_d$$

The last variables to determine were the ratio of beam radiation on a tilted surface to horizontal and the total irradiance on a horizontal surface, which were found with the equations below. With all parameters known, solar irradiance on a tilted surface could be calculated.

$$R_b = \frac{\cos(\varphi - \beta)\cos\delta\cos\omega + \sin(\varphi - \beta)\sin\delta}{\cos\varphi\cos\delta\cos\omega + \sin\varphi\sin\delta}$$

$$G = G_b + G_d$$

Selecting the collector

Collectors from the SRCC 2003 directory were compared to each other in Matlab as shown in the appendix to find the most efficient panel using the equation below.

$$\eta = F_R(\tau\alpha) - \frac{F_R U_L}{G_T}(T_i - T_a)$$

Where the collector loss coefficient and optical efficiency were supplied by the SRCC. The collector inlet to ambient temperature difference (T_i – T_a) was assumed to be 50 °C for water heating in cool climates, based on the chart supplied by the SRCC. The collectors were compared at solar noon on December 21 at an arbitrary collector tilt angle equal to the latitude (42°). The Mazdon 30 by Thermo Technologies was ultimately selected for having one of the top efficiencies at 42% and the best energy gain for an inlet-ambient temperature difference of 50 °C, as stated by the SRCC. This selection made sense since the Mazdon 30 was an evacuated tube collector, Medford Oregon had a cool climate during the winter, and the system was designed to focus on space heating during the winter. This Mazdon 30 is pictured in the appendix.

Determining heating load

The thermal load was determined for both summer and winter. In winter the load included both space heating and DHW, while summer was DHW only. The design and simulation were based on the extremes, middle of winter and middle of summer. The DHW load was selected based on a project in Glasgow, Scotland that determined DHW loads. The DHW load on this system was selected to be an average between 3 people week days and 3 people weekends for a peak load of 510 W (MSc). The peak space heating load was determined to be 20104 Btu/h, or 5887 W, by using Right-Suite Universal 2013. The program's output is shown in the appendix. For design purposes the DHW load was considered constant year round and system sizing was done based on the peak DHW and space heating loads.

Right-Suite Universal 2013 by Wrightsoft is a HVAC software solution for designing residential or commercial systems including ducts, hydronic, radiant, or geothermal loop systems. The software calculates loads as well as can produce parts lists and project proposals. This product is nationally recognized as an accurate source for determining thermal loads and designing heating systems (Wrightsoft).

Modification of collector parameters

The collector optical efficiency and loss coefficient that were developed by the SRCC through tests to describe the collector characteristics only and for when water was used as the thermal fluid at a certain flowrate. The two parameters did not take into account the heat exchanger, piping losses, or the use of the propylene glycol solution. To modify the parameters for the correct working fluid, the following equations where used.

$$F'U_L = -\frac{\dot{m}C_p}{A_C} ln\left[1 - \frac{A_C F_R U_L}{\dot{m}C_p}\right]$$

$$r = \frac{\frac{\dot{m}C_p}{A_C F' U_L}\left[1 - exp\left(-\frac{A_C F' U_L}{\dot{m}C_p}\right)\right]_{use}}{\frac{\dot{m}C_p}{A_C F' U_L}\left[1 - exp\left(-\frac{A_C F' U_L}{\dot{m}C_p}\right)\right]_{test}}$$

The collector parameters were corrected to account for the 30% propylene glycol mix that would be used in the collector loop. It should be noted that accounting for a different thermal fluid involves accounting a different flowrate along with different heat capacities. Since mass flowrates are used, they changed from fluid to fluid even when the same volumetric flowrates were used as tested by the SRCC. Propylene glycol was necessary to protect the system against freezing, and 30% glycol will be enough for protection while maximizing the dilution of the glycol. Water has better heat absorption characteristics than propylene glycol, as propylene glycol has a heat capacity of 70% that of water. This led to a 30% propylene glycol solution with a heat capacity of 3915 J/kg·°C (Engineeringtoolbox).

Solar thermal systems contain more than just a collector, and the losses associated with the other components must be accounted for in design and simulation. The following equations were used to correct the collector parameters for the pipe run of the system.

$$U_d = \frac{2K}{D_o \ln\left(\frac{D_o}{D_i}\right)}$$

$$\frac{(\tau\alpha)'}{(\tau\alpha)} = \frac{1}{1 + \frac{U_d A_O}{(\dot{m}C_p)_C}}$$

$$\frac{U_L'}{U_L} = \frac{1 - \frac{U_d A_i}{(\dot{m}C_p)_C} + \frac{U_d (A_i + A_O)}{A_C F_R U_L}}{1 + \frac{U_d A_O}{(\dot{m}C_p)_C}}$$

Where the pipe run was 12 m of Urecon Sunpipe with K = 0.026 W/m·°C and is pictured in the appendix (Urecon). The pipe loss coefficient was found by assuming the pipe insulation was the significant resistance to heat loss through the pipe.

The steps to determining the heat exchanger removal factor are shown below. The effectiveness was found using the collector as minimum capacitance rate and the storage side as the maximum. This was because both sides of the heat exchanger have the same flowrate but water has the better heat capacity, leading to the larger capacitance rate.

$$C^* = \frac{(\dot{m}C_p)_{min}}{(\dot{m}C_p)_{max}}$$

$$NTU = \frac{(UA)_{hx}}{(\dot{m}C_p)_{min}}$$

$$\varepsilon = \frac{1 - e^{-NTU(1-C^*)}}{1 - C^* e^{-NTU(1-C^*)}}$$

$$\frac{F_R'}{F_R} = \left[1 + \left(\frac{A_C F_R U_L'}{(\dot{m}C_p)_C}\right)\left(\frac{(\dot{m}C_p)_C}{\varepsilon(\dot{m}C_p)_C} - 1\right)\right]^{-1}$$

The heat exchanger was an Alfa Laval CB14-14H with a UA = 850 W/°C (Alfa Laval). The selected heat exchanger is a single-walled, because they are more efficient. Also, propylene glycol is a food-grade fluid that would not be dangerous if a leak occurred in the heat exchanger. Corrosion through the heat exchanger wall was also not a concern. The Alfa Laval CB14-14H is pictured below in the appendix.

Selecting the pump

After calculating the flowrate in the collector loop to be 0.08 kg/s, or 1.2 GPM, and determining that the max head would be 15 feet, all parameters were known for selecting the circulation pump. The pump selected was the Taco model 009. This pump was selected because it met the specifications and its size and efficiency. The Taco model 009 is shown in the appendix (Taco-hvac). The same pump will be used on the collector and storage sides of the heat exchanger.

Sizing collector array and storage

With the collector parameters modified to account for a system comprised of a collector, piping, and a heat exchanger, the collector array and water storage tank could be sized. The f-chart method was used for its simplicity and that the proposed system's parameters were within the range of the f-chart correlations. The collector losses and absorbed solar radiation to the heating load ratio equations and the solar fraction equation are presented below. The time step is one day since the simulation was done for a summer day and a winter day, so all data used were for a separate day to estimate how the system performs on the middle of summer and winter. The reference temperature was 100 °C as stated by Duffie, while the average ambient temperature was found based on hourly data for July 21 and December 21 from the National Renewable Energy Laboratory (WeatherSpark).

$$\Delta t = \left(3600\frac{sec}{h}\right)\left(24\frac{h}{day}\right)$$

$$X = \frac{A_c F_R' U_L' \left(T_{ref} - \bar{T}_a\right)\Delta t}{L}$$

$$Y = \frac{A_c F_R' (\tau\alpha)' H_T}{L}$$

$$f = 1.029Y - 0.065X - 0.245Y^2 + 0.0018X^2 + 0.0215Y^3$$

The solar fraction was calculated for a range of array sizes, as shown in the appendix. The parameter equations above were modified to be used for estimating daily solar fraction instead of a monthly value. This led to the time step (Δt) of seconds in a day instead of a month. Considering performance only, a six-panel array was selected. This size array was selected because the summer solar fraction was 164% while only providing a third of the winter load. For the site used in this design, up to twenty panels could be installed on the roof facing south. Twenty panels would provide over 80% of the winter load, but would provide over 62 times the summer load. Depending on the heat dump methods employed, the entire south facing roof can be utilized. It should be noted that these two solar fractions are the minimum and maximum provided, with the majority of the year having a solar fraction between the two calculated values.

With six Mazdon 30 collectors and using storage that was 75 liters per square meter of collector, the storage tank would need to be at least 2061 liters, or 545 gallons and have an assumed $(UA)_s$ = 2 W/°C. This heat loss-are product was selected based on values provided by Duffie. Sizing the storage tank at 75 l/m^2 was selected as it matched the storage tanks used in the development of the f-chart method and thereby simplified the equations. The selection also provided a good balance between of thermal storage capability and space, when compared to the typical 30-100 l/m^2 range as described by Duffie. Salcines demonstrated that as storage increased, the solar fraction increased logarithmically, with the large storage volumes providing diminished returns.

The collector array selected along with the other system components would not cover the entire load during the winter. The system will need an auxiliary heat source to provide a maximum of at least 4.3 kW to make up for the load that was not supplied by the solar thermal system. For solar system downtime, it may be beneficial to have an auxiliary system like a boiler that can provide all the load, which would be at least 6.4 kW.

Now that the entire system was sized and components were selected, the system could now be simulated to estimate its performance.

Simulation

The Matlab simulation *SolarThermalSim* was developed to first calculate the solar radiation incident on the collector array, then the solar energy collected by the system, and then the change in the storage temperature. To get useful information from the collector while keeping computation at a minimum, the simulation performed the calculations for every hour of the day. For this purpose a simulation with a resolution of seconds would not have been necessary.

The solar radiation incident on the collector array was calculated using the same method as shown above for collector selection. This was done with the modified collector optical efficiency and loss coefficient values.

The next step was to find the energy collected by the system. The energy that is successfully collected by the solar thermal system is known as the useable energy, and was calculated with the following equations.

$$K_{\tau\alpha} = 1 - 0.1441\left(\frac{1}{\cos\theta_z} - 1\right) - 0.0948\left(\frac{1}{\cos\theta_z} - 1\right)^2$$

$$Q_u = A_C F_R'[(\tau\alpha)' G_T K_{\tau\alpha} - U_L'(T_s - T_a)]$$

The storage temperature replaced the typical collector inlet temperature (T_i) in the useable energy equation. The inlet temperature was assumed to be equal to the storage temperature since the pipe run was insulated and short enough that the losses would be minimal, making the calculations simpler. The incident angle modifier is added to account for the change in the incidence angle of the incoming beam radiation on the collector array.

Knowing the energy collected by the system, it was possible to calculate the storage temperature. The storage equation, shown below, essentially takes the solar energy collected over the hour and subtracts that hour's load and losses, ultimately adding to the storage temperature at the beginning of the hour. The equation was developed by using Euler integration on the storage temperature differential equation, dT_s/dt.

The load was a heat exchanger, often a baseboard heater, and the losses are from the storage tank. The load heat exchanger was assumed to have the same properties as the Alfa Laval CB14-14H. The time step in this case was in hours, so $\Delta T = 1$. The efficiency of the system could also be calculated using the useable energy collected for the day and divided by the total incident radiation of the day and collector array area. This is shown as the second equation below.

$$T_s^+ = T_s + \frac{\Delta T}{(\dot{m}C_p)_s}\Big\{3600 A_c F_R'[G_T(\tau\alpha)' - U_L'(T_s - T_a)]^+ - 3600(UA)_s(T_s - T_r) - 3600\varepsilon(\dot{m}C_p)_{min}(T_s - T_h)\Big\}$$

$$\eta = \frac{\sum_{i=1}^{24} 3600 Q_{u,i}}{H_T A_C}$$

Solar thermal systems also need controls to function efficiently. Without control, the pump would pull heat from the storage tank and release through the collector to the atmosphere at times with low solar radiation, likely near sunrise and sunset. The proposed system was designed to use differential control to turn the pump on or off based on the temperature difference between the collector and storage tank. When the collector was at least T_{on} = 10 °C above the storage, the pump will turn on. The pump will turn off when the temperature difference T_{off} falls beneath a ratio of turn-on criterion to turn-off criterion that was calculated using the equation below.

$$\frac{T_{on}}{T_{off}} \geq \frac{(\dot{m}C_p)_c}{A_c F_R' U_L'}$$

Following the above ratio, the controller will shut the system off if the temperature difference falls beneath 1 °C, while keeping the pumps from cycling.

Results

Below are tables showing the results of the simulation. The two tables display the values calculated and for the full day of July 21 and December 21.

Table 3. Simulation results for July 21.

Hour	I_T (MJ/m^2)	T_a (°C)	T_S^+ (°C)	Q_U (MJ)	Load (MJ)	η (%)
Start			40.0			
0 - 1	0	19.4	39.9	0	0.83	0
1 - 2	0	18.3	39.8	0	0.77	0
2 - 3	0	17.2	39.7	0	0.72	0
3 - 4	0	16.7	39.6	0	0.74	0
4 - 5	0	15.6	39.5	0	0.76	0
5 - 6	0	15	39.4	0	0.72	0
6 - 7	0.45	16.1	39.2	0	1.69	0
7 - 8	1.09	17.8	38.9	0	1.76	0
8 - 9	1.72	20	40.4	14.46	1.37	30.6
9 - 10	2.23	22.2	43.1	24.26	1.39	39.5
10 - 11	2.56	24.4	46.4	29.97	1.51	42.5
11 - 12	2.68	26.7	49.9	31.77	1.44	43.2
12 - 13	2.56	28.3	53.1	29.57	1.37	42.0
13 - 14	2.23	30.6	55.7	23.66	1.15	38.6
14 - 15	1.72	31.7	57.1	13.74	1.10	29.1
15 - 16	1.09	32.8	57.0	0	1.44	0
16 - 17	0.45	32.8	56.8	0	1.51	0
17 - 18	0	32.2	56.6	0	1.37	0
18 - 19	0	31.1	56.4	0	1.37	0
19 - 20	0	28.9	56.1	0	1.84	0
20 - 21	0	25.6	55.9	0	1.75	0
21 - 22	0	23.9	55.7	0	1.37	0
22 - 23	0	22.2	55.5	0	1.33	0
23 - 24	0	21.1	55.4	0	1.15	0
Total	18.78			167.43	30.44	37.9

Table 4. Simulation Results for December 21.

Hour	I_T (MJ/m^2)	T_a (°C)	T_S^+ (°C)	Q_U (MJ)	Load (MJ)	η (%)
Start			40.0			
0 - 1	0	1.7	38.1	0	16.55	0
1 - 2	0	1.7	36.3	0	14.89	0
2 - 3	0	1.7	34.7	0	13.38	0
3 - 4	0	1.1	33.3	0	12.10	0
4 - 5	0	1.1	32.0	0	10.94	0
5 - 6	0	1.1	30.9	0	9.84	0
6 - 7	0	1.1	29.7	0	9.85	0
7 - 8	0.58	1.1	29.0	2.79	8.96	17.4
8 - 9	1.35	1.7	29.6	13.81	7.96	37.2
9 - 10	2.09	2.2	31.5	24.55	8.53	42.7
10 - 11	2.56	3.3	33.9	31.30	10.18	44.5
11 - 12	2.71	4.4	36.4	33.45	12.12	44.8
12 - 13	2.56	5	38.3	30.86	14.08	43.9
13 - 14	2.09	5.6	39.2	23.81	15.46	41.4
14 - 15	1.35	6.1	38.8	12.98	16.19	34.9
15 - 16	0.58	6.1	37.2	2.20	16.21	13.8
16 - 17	0	5	35.5	0	14.92	0
17 - 18	0	3.9	33.9	0	13.33	0
18 - 19	0	3.3	32.5	0	12.03	0
19 - 20	0	2.8	31.1	0	11.33	0
20 - 21	0	2.8	29.9	0	10.14	0
21 - 22	0	2.2	28.9	0	8.77	0
22 - 23	0	2.2	28.0	0	7.88	0
23 - 24	0	1.7	27.2	0	6.93	0
Total	15.88			175.74	282.57	35.6

The total efficiency in both tables 3 & 4 were calculated as the average efficiency over the period where the useable energy was over zero. Otherwise the daily system efficiency would have been greatly underestimated by the hours of no sun. It should be noted that in table 3, there are times during the summer when there is solar radiation incident on the panels but the useable energy is zero. This was because the pump is off. The storage doesn't lose heat as quickly during the summer, so the pump will come on later and turn off sooner, as shown in figure 5.

The benefit of selecting the collector tilt to be optimized for winter is clearly demonstrated in figure 6. Although the day length is shorter in the winter time, the two days are very comparable on energy collected. The peak irradiation is higher on December 21, but July 21 still has a larger daily radiation. The data is for a clear day however, leading to most winter days in Medford to supply less useable energy.

Another issue to consider is that the storage has trouble containing heat in the winter, as the storage is roughly half of what it was at from beginning to end of the simulation. Added or improved insulation could be a simple way to improve performance of the system in winter. Improving the insulation on the storage would ultimately decrease $(UA)_S$.

Both table 4 and figure 7 are a good representation of the need for a thermal dump during the summer. The storage temperature ended up much higher than where it started. The temperature will continue to increase each day when considering consecutive days in summer. With enough thermal dump, the system size could be increased to cover more of the winter load.

The following graphs present a visual representation of the pertinent information in tables 3 & 4 as well as other information.

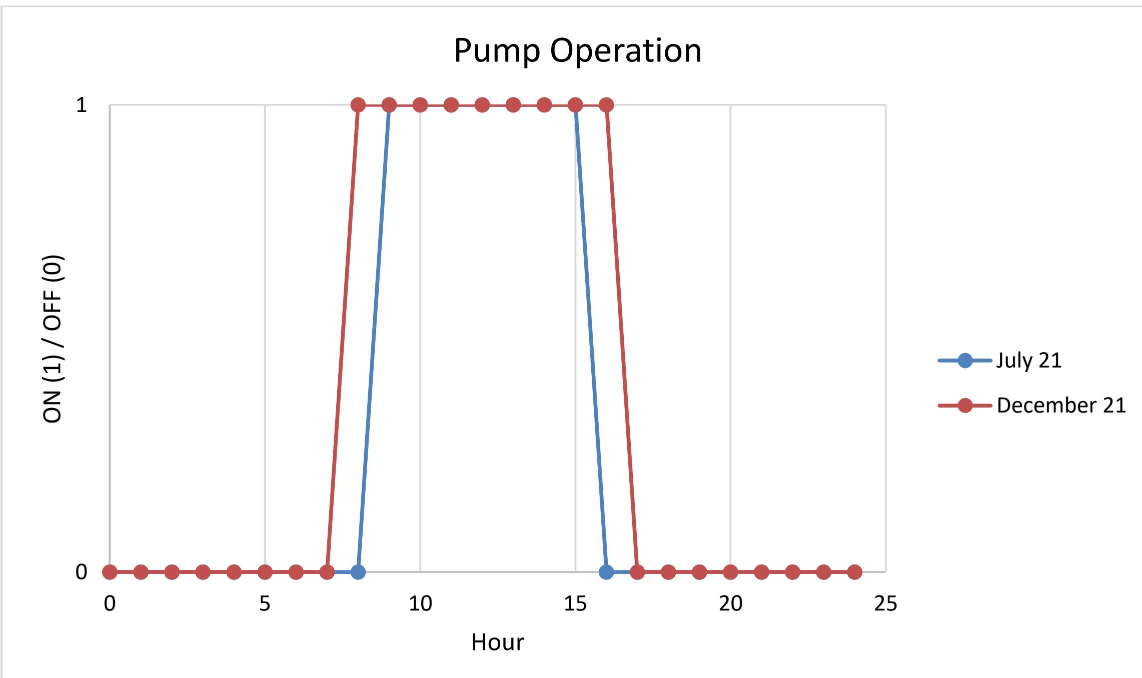

Figure 5. Pump operation over day.

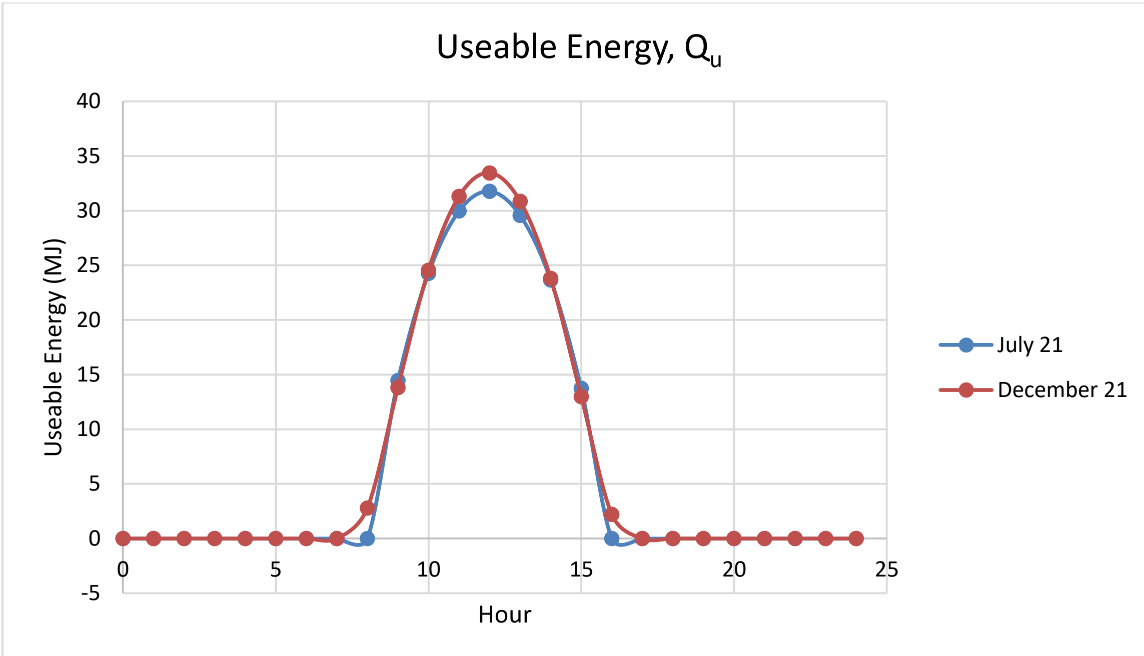

Figure 6. Useable energy collected by system over day.

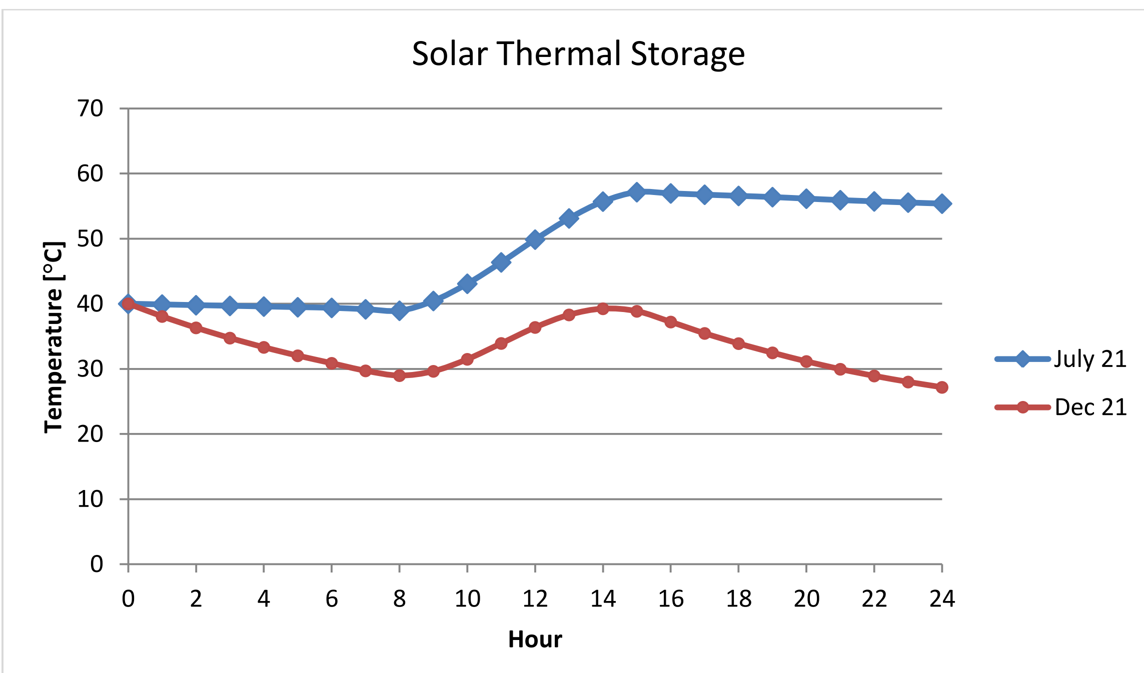

Figure 7. Storage temperature over day.

Conclusion

An active solar thermal was successfully designed and simulated using Matlab2014a for a home maintained at 21 °C in Medford, Oregon. The system was designed using the f-chart method and found that six Mazdon 30 evacuated tube collectors were ideal to cover the winter load as much as possible without supplying too much over the summer load that couldn't be maintained at safe levels with minimal heat dump requirements. With six collectors, 160% and 33% of the summer and winter loads were met respectively. With proper heat dumping, the entire site's south-facing roof could be used for collectors, which is about 92 m^2 and met 80% of the load. This design ignored financial aspects, which could greatly change the final design, depending on financial backing.

After the collector array, the system used SunPipe by Urecon. SunPipe was ¾ in copper piping with 0.026 W/m·°C insulation. The heat exchanger between the collector and storage tank loops was an Alfa Laval CB14-14H brazed plate heat exchanger. The pump used to push the fluids in both collector and storage loops was the model 009 by Taco. The water storage tank was assumed to have a $(UA)_s$ = 2 W/°C and was in a room with a temperature that was assumed to be the average of the house and ambient temperature.

The system was studied through simulation for July 21 and December 21 to estimate how the solar thermal system would operate in summer and winter respectively. There was a 30% propylene glycol solution used in the collector loop for freezing protection at a flowrate of 0.08 kg/s. Geographical data was collected from NREL and the space heating and DHW loads used in the design were found using Right-Suite Universal 2013 and a Scottish study respectively.

The proposed system would help in space heating and DHW for the year by meeting between 33-100% of the load, depending on the time of year. Although the system doesn't cover the entire annual load, and financial constraints could make it cover less than originally designed. The solar thermal system was effective at offsetting some of the energy that would have otherwise came from fossil fuels. Solar thermal if used with other renewable energy sources could provide a home with 100% clean energy.

Recommendations

There are many things that can be done to this system. The system can easily be expanded, as well as use more involved methods for design. Because of the excessive amount of extra energy in the summer diminishes what can be done in the winter, solar cooling could be added to the solar thermal system to make use of the extra solar radiation. The system would than cover even more of the home climate control needs. With research completed on solar cooling technologies, and with the excess summer solar energy known for a system with up to 20 panels, a subsystem can be sized and added to the existing design. The load heat exchanger can be removed to use radiant floor heating instead, which could be done with modification done to the storage temperature equation in the simulation.

Since the f-chart was employed to provide simplicity, other methods can be employed to increase the accuracy of the design and simulation. For instance the utilizability f-chart method can be used for any temperature delivered to the load, whereas the f-chart method assumes 20 °C. The majority of the load on the designed system was space heating, so it was not a stretch to assume 20 °C. The utilizability f-chart method takes more parameters into account to better estimate system performance, at the cost of more involved calculations. The method is an extension of the f-chart method, and many parameters are therefore shared.

The simulation can also be expanded to show a bigger picture of the storage temperature fluctuations. Weeks or even a full year can be simulated to show the long term effects of the climate on the solar thermal system.

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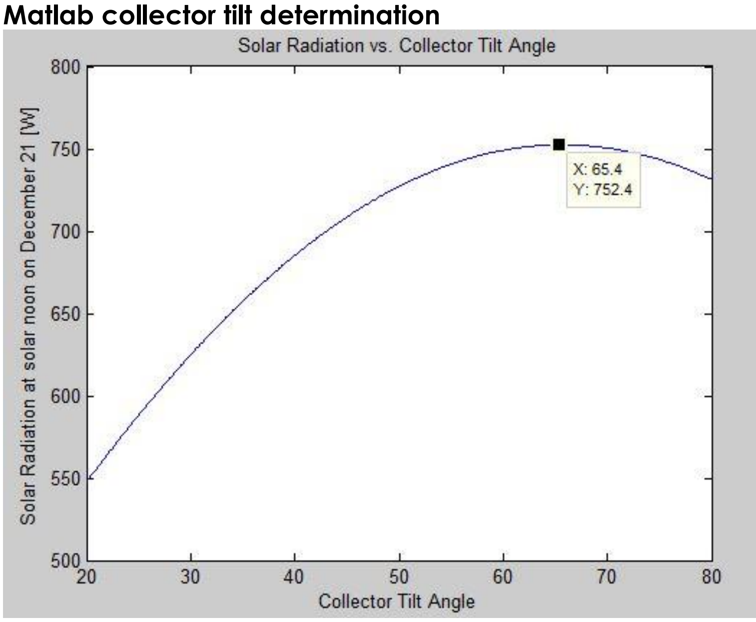
Appendix

Solar thermal system site

Matlab collector tilt determination



Matlab Code for panel selection

```
%Colector selection done by first finding Ht
%Next t=Ti-Ta from SRCC sheets
%Then use t in this code
%Finally collector with highest output is selected
t=50;
Gt=754; %Dec 21 at solar noon

HC30=(-21.31/Gt)*t+.871
FP48=(-15.94/Gt)*t+.794
FS48=(-15.22/Gt)*t+.781
fireball=(-3.73/Gt)*t+.604
AE21=(-4.9099/Gt)*t+.706
AE21E=(-6.37/Gt)*t+.66
AE24E=(-6.37/Gt)*t+.655
ST21E=(-6.02/Gt)*t+.674
Gobi308=(-4.74/Gt)*t+.733
Gobi3366=(-4.68/Gt)*t+.734
Gobi408=(-4.57/Gt)*t+.737
Mojave408=(-6.08/Gt)*t+.726
DP62800=(-1.554/Gt)*t+.502
Sunlast21EPI308SS=(-6.11/Gt)*t+.708
Radco308CHP = (-4.964/Gt)*t+.778
Radco308PHP = (-7.51/Gt)*t+.764
Radco408CHP = (-4.77/Gt)*t+.779
Radco408PHP = (-7.24/Gt)*t+.768
FW48 = (-8.21/Gt)*t+.739
J = (-8.36/Gt)*t+.7724
K = (-5.93/Gt)*t+.759
L = (-7.47/Gt)*t+.625
M = (-4.53/Gt)*t+.625
STG24 = (-7.44/Gt)*t+.7149
EC20 = (-4.1279/Gt)*t+.714
EC24 = (-4.1279/Gt)*t+.7135
EP20 = (-4.5392/Gt)*t+.682
NC32 = (-4.84/Gt)*t+.508
Mazdon30 = (-1.421/Gt)*t+.53

end
```

2003 Certification of Mazdon 30 (Directory)

SOLAR COLLECTOR CERTIFICATION AND RATING	CERTIFIED SOLAR COLLECTOR
SRCC OG-100	SUPPLIER: Thermo Technologies 6193 Wooded Run Drive Suite B Columbia, MD 21044 MODEL: Mazdon 30 COLLECTOR TYPE: Tubular CERTIFICATION #: 100-1998-001A

COLLECTOR THERMAL PERFORMANCE RATING

Megajoules Per Panel Per Day				Thousands of Btu Per Panel Per Day			
CATEGORY (Ti-Ta)	CLEAR DAY 23 $MJ/m^2 \cdot d$	MILDLY CLOUDY 17 $MJ/m^2 \cdot d$	CLOUDY DAY 11 $MJ/m^2 \cdot d$	CATEGORY (Ti-Ta)	CLEAR DAY 2000 $Btu/ft^2 \cdot d$	MILDLY CLOUDY 1500 $Btu/ft^2 \cdot d$	CLOUDY DAY 1000 $Btu/ft^2 \cdot d$
A (-5 °C)	46	35	23	A (-9 °F)	44	33	22
B (5 °C)	45	33	22	B (9 °F)	42	31	21
C (20 °C)	42	30	19	C (36 °F)	40	29	18
D (50 °C)	35	24	13	D (90 °F)	33	23	12
E (80 °C)	27	17	6	E (144 °F)	26	16	6

A-Pool Heating (Warm Climate) **B**-Pool Heating (Cool Climate) **C**-Water Heating (Warm Climate) **D**-Water Heating (Cool Climate) **E**-Air Conditioning

Original Certification Date: April 20, 1998

COLLECTOR SPECIFICATIONS

Gross Area:	4.581 m^2	49.31 ft^2		**Net Aperture Area:**	3.381 m^2	36.39 ft^2
Dry Weight:	89.4 kg	197 lb		**Fluid Capacity:**	0.7 l	0.2 gal
Test Pressure:	1034 kPa	150 psig		**Max. Oper. Temp.:**	130 °C	266 °F

COLLECTOR MATERIALS

Frame:	Stainless Steel
Cover (Outer):	Iron Free Glass Vacuum Tube
Cover (Inner):	None
Absorber Material:	Tube - Copper / Plate - Copper Fin
Absorber Coating:	Black Chrome
Insulation (Side):	Vacuum
Insulation (Back):	Vacuum

PRESSURE DROP

Flow		ΔP	
ml/s	gpm	Pa	in H_2O
40	0.63	935	3.75
80	1.27	3128	12.56
120	1.90	6492	26.06

TECHNICAL INFORMATION

Efficiency Equation [NOTE: (P) = Ti-Ta]

				Y Intercept	Slope	
S I Units:	$\eta = 0.525$	-0.8858 (P)/I	-0.0074 $(P)^2/I$	0.53	-1.421	$W/m^2 \cdot °C$
I P Units:	$\eta = 0.525$	-0.1561 (P)/I	-0.0007 $(P)^2/I$	0.53	-0.250	$Btu/hr \cdot ft^2 \cdot °F$

Incident Angle Modifier [(S) = 1/cos θ - 1, 0° ≤ θ ≤ 60°]

$K_{\alpha\tau}$ = 1.0 -0.1441 (S) -0.0948 $(S)^2$

$K_{\alpha\tau}$ = 1.0 -0.24 (S) (Linear Fit)

Model Tested: 30
Test Fluid: Water
Test Flow Rate: 76 ml/s 1.20 gpm

REMARKS: Collector tested with long axis of tubes oriented north-south. IAM perpendicular to the tubes is listed above. IAM parallel to the tubes = 1.0 - 0.28(S)

June, 2003
Certification must be renewed annually. For current status contact:
SOLAR RATING & CERTIFICATION CORPORATION
c/o FSEC • 1679 Clearlake Road • Cocoa, FL 32922 • (321) 638-1537 • Fax (321) 638-1010

Mazdon 30 (Thermomax)

Urecon Sunpipe (Urecon)

Alfa Laval CB14-14H (Alfa Laval)

Taco Model 009 (Taco-hvac)

Right-Suite Universal 2013 output (in Btuh)

American Standard
HEATING & AIR CONDITIONING

Right-J® Worksheet
Entire House

Job:
Date: May 30, 2013
By:

1	Room name						Entire House				Room1			
2	Exposed wall						176.0 ft				15.0 ft			
3	Room height						8.0 ft			d	8.0 ft			heat/cool
4	Room dimensions										12.0 x 15.0 ft			
5	Room area						1536.0 ft²				180.0 ft²			
	Ty	Construction number	U-value (Btuh/ft²-°F)	Or	HTM (Btuh/ft²)		Area (ft²) or perimeter (ft)		Load (Btuh)		Area (ft²) or perimeter (ft)		Load (Btuh)	
					Heat	Cool	Gross	N/P/S	Heat	Cool	Gross	N/P/S	Heat	Cool
6	W	12C-0sw	0.091	ne	4.03	1.88	512	443	1786	834	120	108	435	203
.	G	1D-c2ov	0.570	ne	25.25	45.82	48	0	1212	2199	12	0	303	550
.	D	11D0	0.390	ne	17.28	10.02	21	21	363	210	0	0	0	0
.	W	12C-0sw	0.091	se	4.03	1.88	192	160	645	301	0	0	0	0
11	G	1D-c2ov	0.570	se	25.25	22.26	32	32	808	712	0	0	0	0
	W	12C-0sw	0.091	sw	4.03	1.88	512	427	1721	804	0	0	0	0
	G	1D-c2ov	0.570	sw	25.25	56.53	64	0	1616	3618	0	0	0	0
	D	11D0	0.390	sw	17.28	10.02	21	21	363	210	0	0	0	0
	W	12C-0sw	0.091	nw	4.03	1.88	192	180	726	339	0	0	0	0
	G	1D-c2ov	0.570	nw	25.25	45.82	12	0	303	550	0	0	0	0
	C	16B-30ad	0.032	-	1.42	1.57	1536	1528	2166	2406	180	180	255	283
	G	8Ci-6w	1.110	-	49.17	153.72	8	0	393	1230	0	0	0	0
	F	19B-19cvcp	0.049	-	1.22	0.56	1536	1536	1879	861	180	180	220	101
6	c) AED excursion									710				-68
	Envelope loss/gain								13981	14987			1214	1070
12	a) Infiltration								3614	872			308	74
	b) Room ventilation								0	0			0	0
13	Internal gains:		Occupants @		230		0			0	0			0
			Appliances/other							0				0
	Subtotal (lines 6 to 13)								17595	15858			1522	1144
	Less external load								0	0			0	0
	Less transfer								0	0			0	0
	Redistribution								0	0			0	0
14	Subtotal								17595	15858			1522	1144
15	Duct loads						14%	15%	2510	2349	14%	15%	217	170
	Total room load								20104	18207			1739	1314
	Air required (cfm)								839	839			73	61

Calculations approved by ACCA to meet all requirements of Manual J 8th Ed.

ACCA wrightsoft Right-Suite® Universal 2013 13.0.08 RSU05413
...ts\American Standard\2012 homes\cindy house.rup Calc = MJ8 Front Door faces: N

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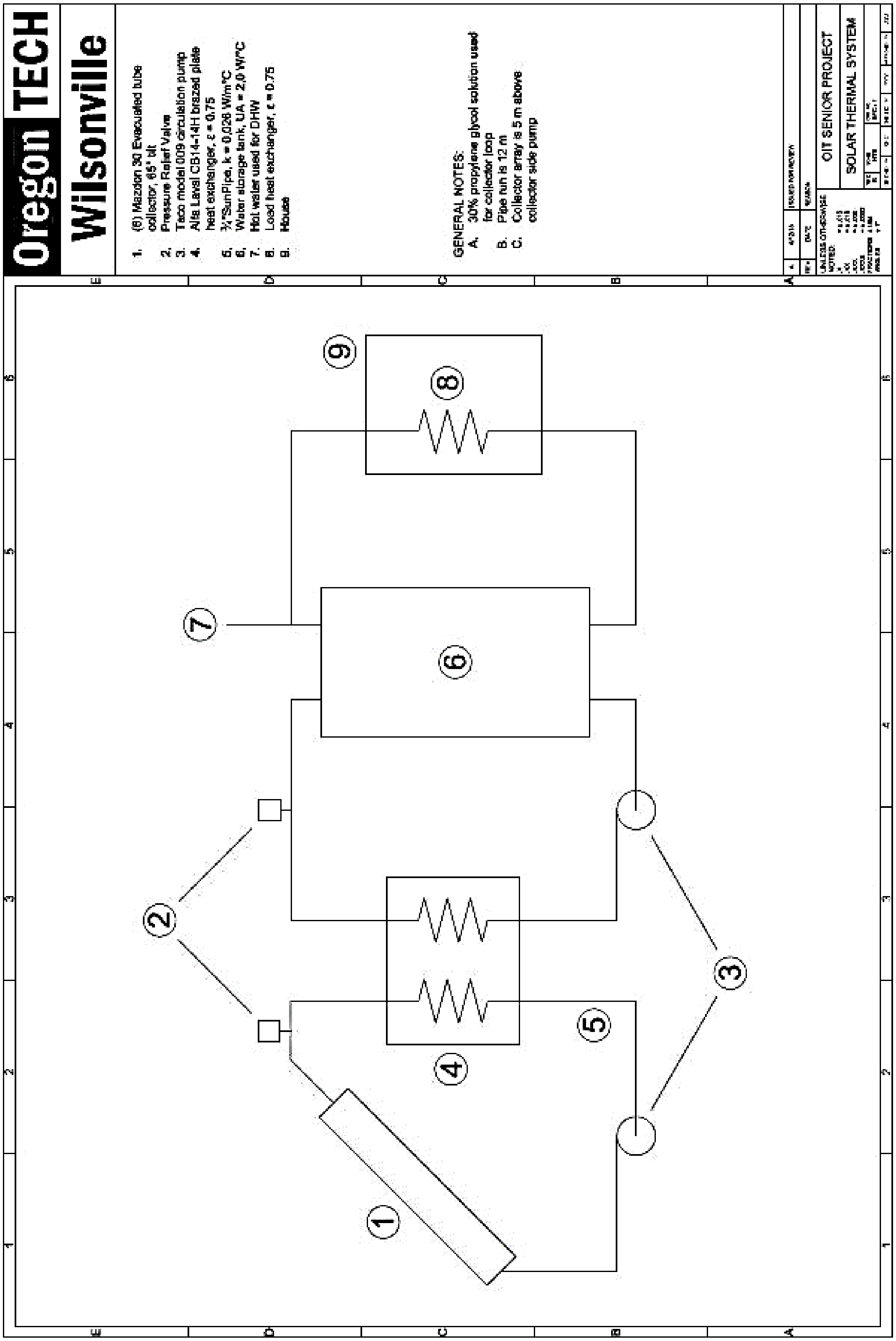
Page 1

System diagram



f-chart design results

Panel	Ac		Fr'(ta)'	Fr'Ul' (J/m2 C h)	Storage Volume		July 21			December 21			Auxiliary (kW)	
	m2	ft2			liters	gal	x	Y	f	X	Y	f	With Solar	No Solar
1	4.58	49.31	0.5224	6553.6	344	91	1.246421635	1.01976096	**0.739**	0.1266	0.068771	**0.061**	6.0	6.4
2	9.16	98.62	0.5186	5758.9	687	182	2.190557114	2.02468619	**1.124**	0.2225	0.136542	**0.122**	5.6	
3	13.74	147.93	0.5148	5470.0	1031	272	3.120998996	3.01477569	**1.279**	0.3171	0.203312	**0.179**	5.3	
4	18.32	197.24	0.5111	5307.9	1374	363	4.038013546	3.9908103	**1.338**	0.4102	0.269135	**0.233**	4.9	
5	22.91	246.55	0.5074	5196.7	1718	454	4.941771933	4.9523996	**1.421**	0.5020	0.333983	**0.285**	4.6	
6	27.49	295.86	0.5038	5111.3	2061	545	5.832673554	5.90071482	**1.641**	0.5925	0.397936	**0.334**	4.3	
7	32.07	345.17	0.5002	5040.8	2405	635	6.710927616	6.83497515	**2.098**	0.6818	0.460941	**0.381**	4.0	
8	36.65	394.48	0.4967	4979.8	2749	726	7.576819404	7.75674223	**2.886**	0.7697	0.523104	**0.425**	3.7	
9	41.23	443.79	0.4932	4925.2	3092	817	8.430463029	8.66484483	**4.088**	0.8564	0.584345	**0.468**	3.4	
10	45.81	493.09	0.4898	4875.3	3436	908	9.272276908	9.56123501	**5.786**	0.9420	0.644796	**0.508**	3.1	
11	50.39	542.40	0.4864	4828.8	3779	998	10.102223	10.4443511	**8.044**	1.0263	0.704353	**0.546**	2.9	
12	54.97	591.71	0.4831	4785.1	4123	1089	10.92087188	11.3165356	**10.933**	1.1094	0.763171	**0.582**	2.7	
13	59.55	641.02	0.4798	4743.6	4466	1180	11.72833765	12.1758365	**14.502**	1.1915	0.821122	**0.617**	2.5	
14	64.13	690.33	0.4765	4703.8	4810	1271	12.52454425	13.0222537	**18.800**	1.2724	0.878203	**0.649**	2.2	
15	68.72	739.64	0.4733	4665.5	5154	1361	13.30989106	13.8587154	**23.887**	1.3521	0.934613	**0.681**	2.0	
16	73.30	788.95	0.4701	4628.5	5497	1452	14.08462533	14.6826838	**29.787**	1.4308	0.990180	**0.710**	1.9	
17	77.88	838.26	0.4670	4592.7	5841	1543	14.84916548	15.4974775	**36.561**	1.5085	1.045128	**0.738**	1.7	
18	82.46	887.57	0.4639	4557.8	6184	1634	15.60316917	16.3001684	**44.215**	1.5851	1.099261	**0.765**	1.5	
19	87.04	936.88	0.4609	4523.8	6528	1724	16.3471499	17.0944653	**52.815**	1.6607	1.152827	**0.791**	1.3	
20	91.62	986.19	0.4578	4490.5	6872	1815	17.08086044	17.8731457	**62.297**	1.7352	1.205340	**0.815**	1.2	