RENEWABLE ENERGY SYSTEMS ANALYSIS

Solar Thermal Conversion for Domestic Hot Water and Space Heating at Hawaii Volcanoes National Park

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EXECUTIVE SUMMARY

Hawaii Volcanoes National Park (HAVO) lies about 30 miles southwest of Hilo on the Big Island. The main visitor center is nearly 4000 feet above sea level, resulting in a climate colder than one generally associates with Hawaii. On average, HAVO tends to be 10-15 degrees Fahrenheit cooler than Hilo, so that while Hilo has no appreciable space heating need, HAVO exhibits one year-round. The National Park Service (NPS) maintains several structures on site, including small residences used for visitors, seasonal employees, permanent employees, and offices. These residences are fairly old buildings, and while tied to the electric grid, lack central heating. The National Park Service wishes to upgrade these structures for year-round habitation in as green as way as possible. Since the primary building energy requirements are for domestic hot water and space heating, a solar thermal system is an appropriate approach.

The year round consistency in the local climate is advantageous for solar system economics. In climates with significant seasonal variation (e.g. the Midwest) a solar space heating system would see almost no use in the summer months, but heavy demand in the winter. Since the economic viability of a solar thermal system depends largely upon degree of utilization, substantial year-round use will prove beneficial. The high cost of electricity (up to 25 ¢/kW hr) and heating fuel for HAVO further improve the economic attractiveness of a solar thermal system, which requires a substantial capital outlay, but has low operating costs.

There are three main functions to a solar thermal system: energy collection and storage, domestic hot water, and space heating. The collector loop functions by passing a fluid (e.g. water) through panels mounted on the roof. The fluid's temperature rises as it absorbs solar energy incident on the panels. This hot fluid then heats water in a large (e.g. 200 gallon) storage tank. One of the principle difficulties in using solar energy is that supply and demand are fundamentally out of synch. For example, space heating needs are highest at night when, by definition, there is no solar resource. The storage tank overcomes this shortcoming by buffering supply and demand. Energy is added whenever the solar resource is sufficient and removed as demand necessitates. The domestic hot water and space heating functions are nearly identical to conventional fossil-fuel systems – with the exception of the boiler. Where a conventional system burns fossil fuel in a boiler to heat water for domestic use and space heating, here the storage tank indirectly heats water for both applications. Since it is generally uneconomic to design a system to meet 100% of the annual load using solar energy, solar thermal systems incorporate auxiliary (fossil fuel) heaters as back-up.

In order to optimize the design of a solar thermal system it is important to understand the loads placed on the system and resource available to meet those loads. Domestic hot water loads are easiest to model – a general assumption being 20 gallons per day per person. The load on the system is the heat required to bring the hot water up to a suitable delivery temperature, between 110^{o}F and 140^{o}F. Space heating loads are more complex. Heat is lost through the walls and windows of the residence and also by the exchange of air between the residence and environment (e.g. through cracks around window frames). Balanced against this is heat generated by the building occupants or appliances (e.g. stove) and solar energy which passes through windows and is absorbed by the structure. The net difference between these competing effects will be the space heating energy requirement. The energy available to the system for space heating and domestic hot water is, broadly, the sum total of solar energy incident on the rooftop collectors.

Clearly, a number of factors influence load and supply. If the house is more heavily insulated, less heat can escape, reducing the space heating load. Likewise, additional panels on the roof allow more solar energy to be captured. Obviously, a system with considerable insulation and a large number of panels will meet a high fraction of the annual energy requirement with solar energy. However, such a system may not be economically preferred. Therefore, it is important to create an integrated engineering and economic model to differentiate potential design options.

We developed a computer simulation which models the loads and solar resource on an hourly basis for the residences. The model pulls together data on climate, building structure, use, and costs to determine lifecycle economics and the fraction of energy requirement met by the solar thermal system.

Four key parameters were varied to understand their impact on the system. The first is number of panels. Increasing the number of panels, as previously mentioned, increases the energy captured by the system, allowing a higher fraction of the load to be met by solar energy. There is also a corresponding increase in system cost for additional panels purchased. Secondly, we considered the thermostat set temperature. The higher the target temperature for the house, the more heat will be lost attempting to maintain it. For the purpose of this analysis, two set-points were considered: 65^{o}F and 70^{o}F. Thirdly, the degree of insulation in the residences strongly influences the total load. Insulating the walls, floors, and ceilings dramatically reduces the space heating load. Further reduction is possible by replacing the existing windows with more efficient, double-paned windows. Finally, the type of heater has a significant impact on the overall system efficiency. Two types of heaters were considered – conventional baseboard heaters (high temperature) and radiant floor heating (low temperature). The lower the operating temperature, the better the chance the energy requirements may be met by the solar resource.

The two following plots show the performance of the various systems considered. The horizontal axis is the fraction of the annual load met by solar energy. The higher the fraction, the "greener" the system. The vertical axis is estimated life cycle cost – that is, the cost of purchasing and operating the system for its assumed twenty year lifetime. The number beneath each data point indicates the number of panels used. The letter beneath each data point indicates whether the system uses a propane (p) or electric (e) auxiliary heater. Data is grouped by type of space heater (baseboard or radiant floor) and degree of insulation. The 'base' insulation case is the residence as it stands today – without insulation. The 'insulation' case involves insulating walls, floors, and ceilings. The 'windows' case would replace the existing windows in addition to insulating the residences.

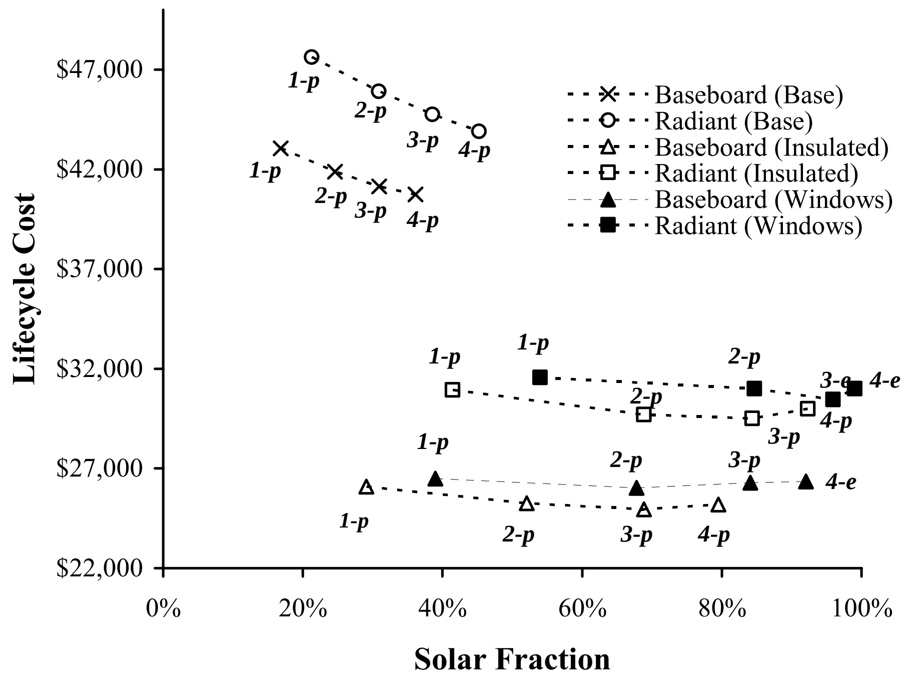


Figure 1 - Economic and Engineering Performance (70°F Set Point)

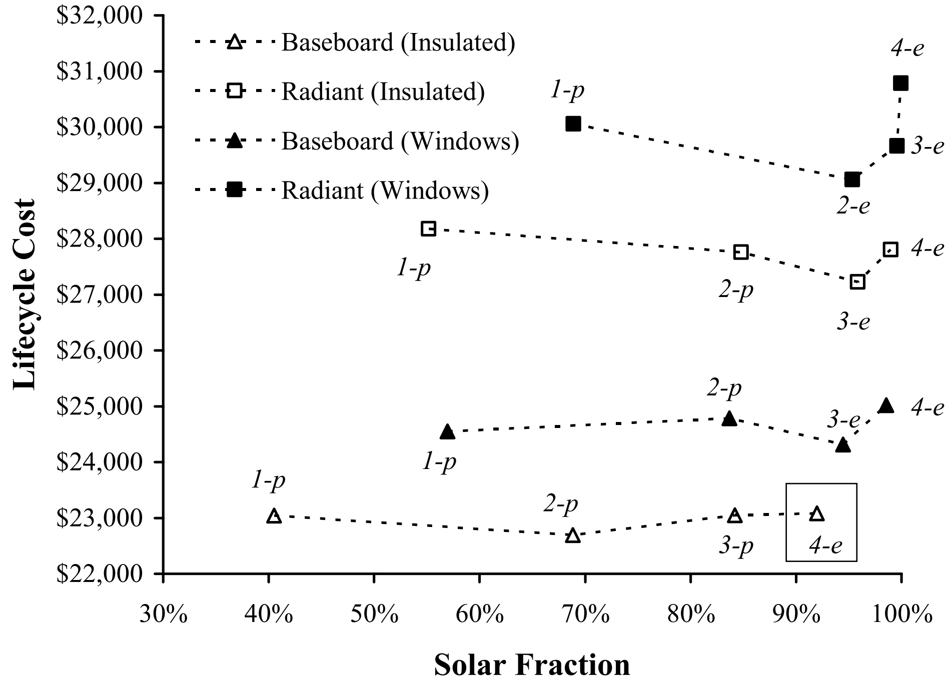


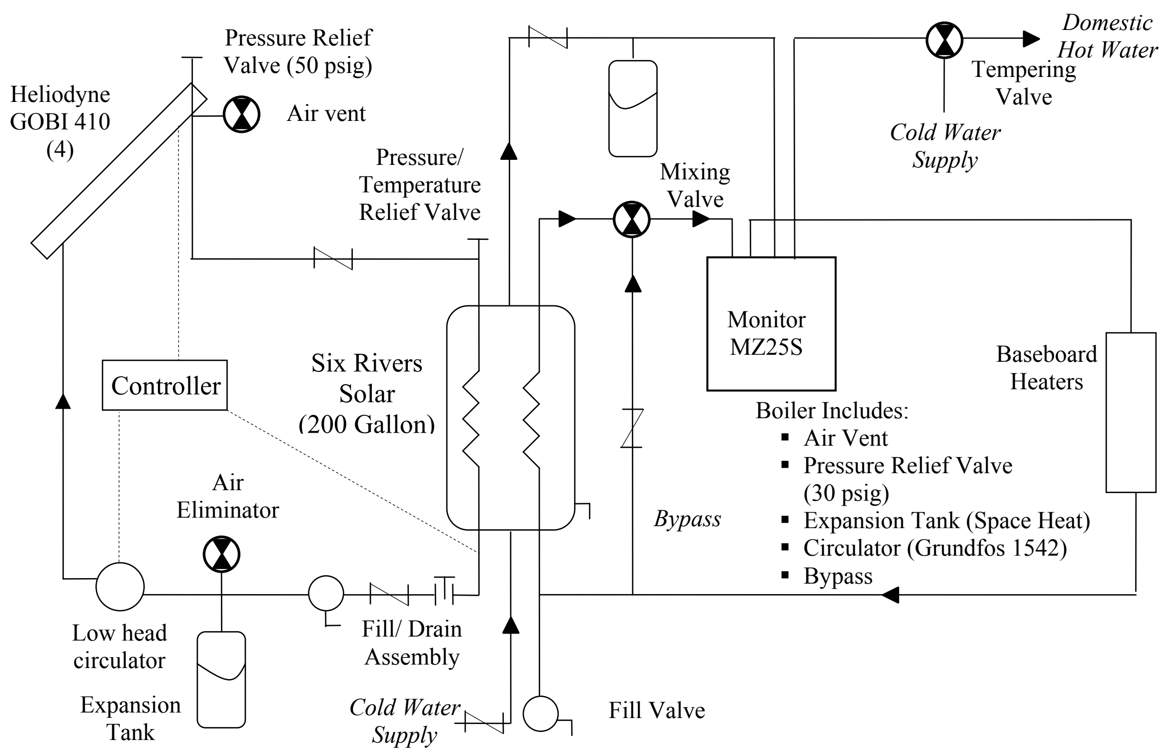
Figure 2 - Economic and Engineering Performance (65°F Set Point)

At a high level, the charts may be broken down into four quadrants. The lower right represents an ideal renewable design – low cost and a high solar fraction. The top right is representative of a "green" design, where economic considerations are subordinate to reducing fossil fuel use. The bottom left is the domain of more conventional systems – low cost, minimal solar contribution. The upper left is a poorly designed system – expensive with little environmental benefit.

These results offer a number of interesting conclusions with relation to "green" design. Reducing load through insulation is absolutely critical to managing the lifecycle cost of the system. Note from figure 1 that even with four panels, the 'base' case system would barely meet 30% of the annual load. Additionally, lowering the thermostat set point substantially reduces energy requirements. At 65°F it is possible to meet more than 90% of the annual load with a solar thermal system at a cost below the cheapest 70°F system. These two findings underline a key principle of renewable system design – conservation should be the first option considered. Furthermore, if a green system is prioritized, replacing windows appears a more economic method of increasing solar performance than installing radiant floor heating.

As a balance between cost and environmental impact we would recommend installing a four panel system with propane auxiliary. This recommendation is contingent upon verification that installing four panels on the south facing roof does not pose a structural concern. The residence should be insulated, but the windows not replaced. Low temperature baseboard heating should be used to meet space heating needs. We recommend a propane auxiliary boiler. While an electric boiler would be somewhat more economic at a 65^{o}F indoor temperature, it is considerably more expensive at 70^{o}F. Specifically, using a propane auxiliary at 65^{o}F adds $800 to the lifecycle cost versus using an electric auxiliary, but using an electric auxiliary at 70^{o}F adds $2500 to the lifecycle cost versus using a propane auxiliary. Since the thermostat set temperature will be largely at the discretion of the building residents, it does not appear prudent to install an electric auxiliary.

At 70^{o}F, the chosen system will meet 80% of the annual load with an estimated life cycle cost of $25,000. At 65^{o}F, the chosen system will meet 92% of the annual load with an estimated lifecycle cost of $23,000. These figures compare favorably with a traditional fossil-fuel heating system which could be installed and operated with a lifecycle cost 10-15% lower than the solar option.



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1. INTRODUCTION

Hawaii Volcanoes National Park lies about 30 miles southwest of Hilo on the Big Island. The main visitor center is nearly 4000 feet above sea level[1] resulting in a climate colder than one generally associates with Hawaii. The National Park Service (NPS) maintains several structures on site, including small residences used for visitors, seasonal employees, permanent employees, and offices. These residences are fairly old buildings, and while tied to the electric grid, lack central heating. The National Park Service wishes to upgrade these structures for year-round habitation in as green as way as possible. Since the primary building energy requirements are for domestic hot water and space heating, solar thermal systems are an appropriate approach.

There are two fundamentally different types of solar thermal systems: active and passive. Passive systems use various techniques to heat a building by capturing and storing solar energy within the structure. All structures do this to a certain extent, but explicitly passive solar designs are more efficient at capturing and using the solar energy. One example of a passive solar system is a Trombe Wall. Normally located beyond a wall of south facing windows, these heavy objects store much of the solar heat absorbed over the course of a day, and release it at night, warming the house. In general, these systems contain no moving parts, and require limited yearly maintenance. Unfortunately, passive solar designs generally are only suitable for new construction, or a building undergoing serious alteration. For this reason, passive solar designs were not considered for this project.

Active solar thermal systems capture the sun's energy, storing for later use. This stored heat may be used for space heating and domestic hot water. Active systems are mechanical, involving one or more loops of water-based heat transfer fluid. A representative schematic diagram is shown in figure 1.

[1] CASTNET

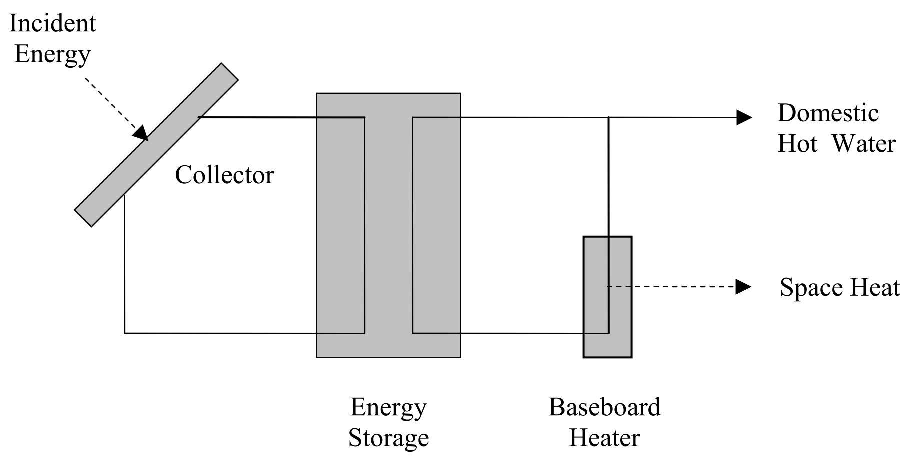

Figure 1 – Active Solar Thermal System

Fluid circulated through the collector is warmed by the incident solar energy and transfers this heat to a water storage tank. The energy in this tank may be then tapped for either hydronic space heating or domestic hot water. Since it is uneconomic to design a collector to meet 100% of the projected building load, an important part of any active system is an auxiliary heater. This may be either an electric or directly fossil fuel fired system. This auxiliary heater sits between the storage tank and end-use application (e.g. hot water tap), increasing water temperature to desired levels if needed.

It is often challenging to design solar thermal systems to economically compete with fossil fuel systems. An optimization involves maximizing the fraction of the building energy requirement met by solar, while minimizing the system cost. Since the "fuel" for a solar system is free, and maintenance costs are relatively low, the biggest expense is for the purchase and installation of equipment. The system then pays for itself by displacing the cost associated with fossil fuels or electricity consumed by a traditional heating system. Two circumstances specific to HAVO allow a well designed solar thermal to compete with traditional fossil-fuel heating: climate and fuel costs.

Outdoor temperature determines space heating requirements. Since the park sees relatively cool conditions throughout the year, with limited seasonal variation, there exists a year-round heating need. Therefore, any system installed will displace fuel costs throughout the year. Additionally, a design intended to meet a significant fraction of the winter load, will also see use throughout the summer. Contrast this situation to a more variable climate, such as the Midwest. A large space heating demand exists in the winter, but none in the summer. As a result, displaceable fuel costs only exist for approximately half the year, reducing the economic competitiveness of a solar thermal system with traditional heating. In general, return on investment will be highest when the purchased equipment sees the most use.

Hawaii has no native fossil fuel resources. As a result, fuel for heating and power generation must be delivered to the island by seagoing tanker. This high transportation cost is reflected in the price of electricity and propane, which are substantially higher than for the continental US. Therefore, the fuel or electric cost displaced by the solar system will be higher for HAVO than at a comparable continental location.

For these reasons, the use of solar energy to provide domestic hot water and space heating is appropriate for HAVO not only for environmental reasons, but is economically competitive as well.

2. CLIMATE

2.1 Overview

Located on the Big Island of Hawaii, Volcanoes National Park enjoys a temperate climate with limited seasonal and yearly variation. The park is located roughly 30 miles southwest of Hilo, but experiences cooler average temperatures due to its elevation (3933 ft).

The creation of an hourly model requires a detailed base of solar and meteorological data to draw upon. Fortunately, sufficient data exists for HAVO and the surrounding region to create such a model without relying on empirical correlations.

2.2 Sources of Meteorological Data

The three primary sources of solar and meteorological data used in this analysis are described in detail as follows.

2.2.1 NCDC – National Climate Data Center

The NCDC is a branch of the NOAA (National Oceanographic and Atmospheric Administration) that maintains a large archive of worldwide weather data. Daily information for HAVO is available starting July, 1993. Recorded data is limited to:

- Precipitation
- Temperature at time of observation
- Maximum daily temperature
- Minimum daily temperature

2.2.2 CASTNET – Clean Air Status and Trends Network

CASTNET is a set of monitoring stations maintained by the EPA. Hourly data for a range of solar and meteorological conditions is available from late 1999 onwards. While only one monitoring station exists for the Hawaiian Islands, it is sited at HAVO. The CASTNET database contains:

- Temperature
- Relative humidity
- Global solar radiation falling on a horizontal surface
- Ozone
- Precipitation
- Wind speed

2.2.3 NREL – National Renewable Energy Laboratory

Hourly, detailed solar insolation information is available from NREL for 1961-1990. While no data series was compiled for HAVO, one does exist for Hilo. It is important to recognize that the NREL data set is not entirely based on physical observation. In the case of Hilo, most of the data has been derived through empirical relations applied to data recorded by a monitoring station elsewhere in the islands.

NREL data is available for:

- Extraterrestrial horizontal radiation
- Extraterrestrial direct normal radiation
- Global radiation on a horizontal surface
- Direct normal radiation
- Diffuse radiation on a horizontal surface

2.3 Temperature

In the case of temperature, daily averages from the broader NCDC data set were compared against the three years of CASTNET temperatures. Since NCDC data is only available for maximum and minimum temperatures for a given day, the average temperature will be defined as $(T_{max} + T_{min})/2$, rather than a more accurate hourly average. Results of this comparison are given in Table 2.3.1.

Table 2.3.1 – Monthly Average Temperature Comparison

Month	CASTNET (^oF) (Sept 1999 – Nov 2002)	NCDC (^oF) (Jul 1993 – Jul 2002)	Variation
January	57.7	58.6	-5%
February	57.4	58.5	-4%
March	58.3	59.2	-3%
April	59.5	59.9	-2%
May	60.8	61.5	-2%
June	61.9	62.8	-3%
July	63.0	63.9	-3%
August	63.3	64.6	-4%
September	62.4	64.2	-6%
October	62.1	63.7	-5%
November	61.0	61.5	-2%
December	58.6	59.2	-2%

This comparison indicates the shorter run CASTNET data set is nearly equivalent to the results obtained by a longer run data. If anything, CASTNET temperatures are slightly cooler, resulting in more conservative simulation inputs.

Temperatures for HAVO show limited seasonal variation, with an average temperature variation of only 6^{o}F between the warmest and coolest months of the year. On average, daily highs and lows vary by approximately 12^{o}F (with a maximum recorded variation of 30^{o}F within the past four years). Figure 2.3.1 shows monthly average temperatures obtained from CASTNET data. These temperatures are lower than those presented in

Table 2.3.1 as they represent a monthly average of hourly temperatures, rather than an average based on daily high and low temperature.

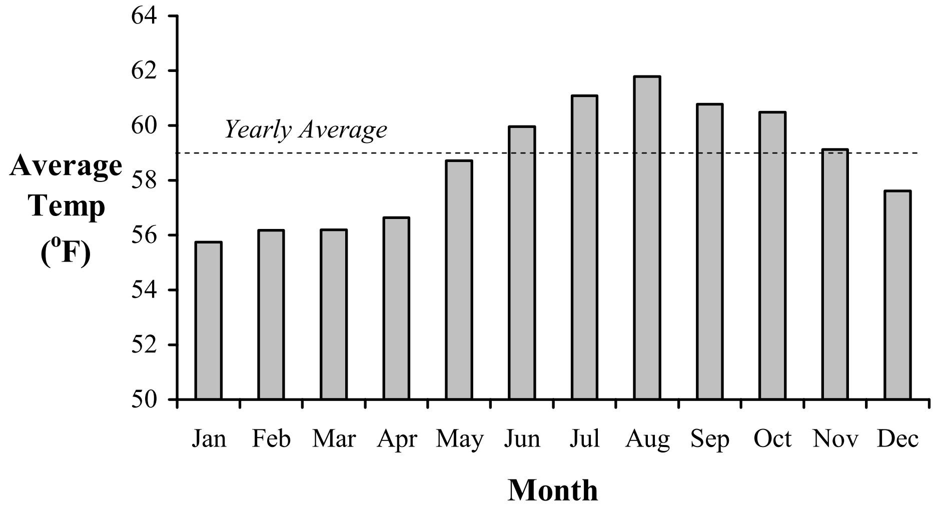

Figure 2.3.1 – Monthly Average Temperature

2.4 Solar Insolation

Accurate solar data proved the most difficult to obtain. Accurate calculation of solar system performance requires knowledge of solar radiation components (e.g., diffuse, direct beam) in addition to total radiation. While component radiation may be calculated from total radiation via empirical correlations, these methods have associated errors. The next best option would be to compute the ratio of hourly global horizontal insolation between HAVO and Hilo for a given hour, then scale diffuse and direct beam radiation by the same ratio. However, this is not possible because the NREL data set for Hilo ends in 1990 and CASTNET recording does not begin until 1999. Rather than make additional assumptions in an attempt to tie together the data, the decision was made to rely exclusively on the Hilo solar data. We believe the error associated with this approximation is likely no worse than the error associated with decomposing total radiation site specific to HAVO using empirical correlations.

The Hilo area receives significant, stable year round solar flux, as shown in figure 2.4.1.

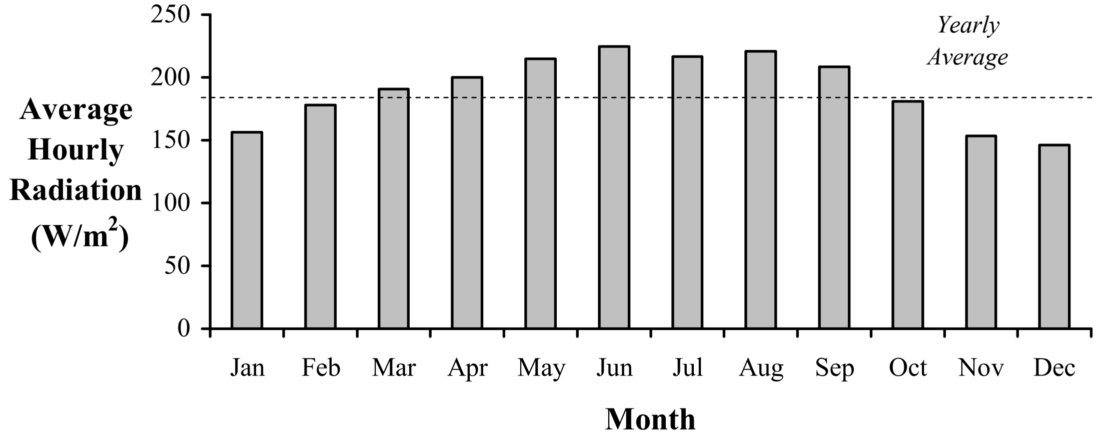

Figure 2.4.1 – Hilo Monthly Average Global Horizontal Radiation (NREL)

However, day-to-day variation is considerable. Figure 2.4.2 shows daily variation in global horizontal radiation for one month at HAVO.

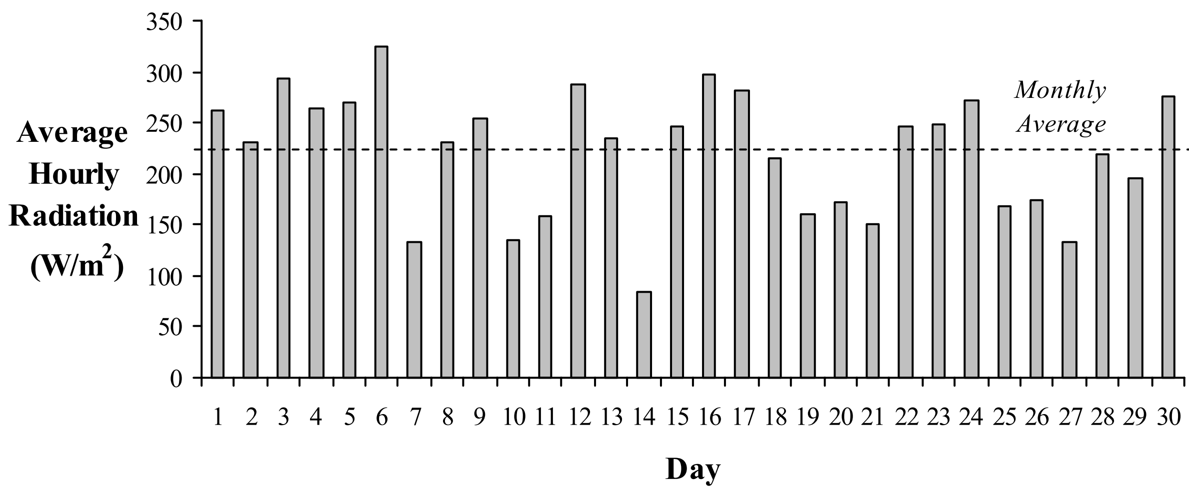

Figure 2.4.2 – HAVO Daily Global Horizontal Radiation – June, 2000 (CASTNET)

2.5 Wind Speed

Site wind speed is of interest for two reasons. In considering renewable options for the site, wind generated electricity would be an obvious complement to a solar thermal system. However, average wind speeds at HAVO are quite low (only a little over 3 mph), with almost no gusting. This indicates the site would be a poor location for a wind turbine. Fortunately, low wind velocity also reduces convective thermal losses from the surface of the building, resulting in a lower space heating energy requirement. This effect will be discussed further in Section 5 of this report.

3. Solar Radiation Incident on a Surface

Creating an accurate engineering model for a solar thermal system requires calculation of solar energy incident on several different surfaces (e.g., roof-top collectors, windows).

3.1 Sun Angles

A number of angles are used to describe the position of the sun relative to a surface. These angles are critical to calculating incident radiation on that surface. Figure 3.1.1 shows a physical representation of a number of these angles.

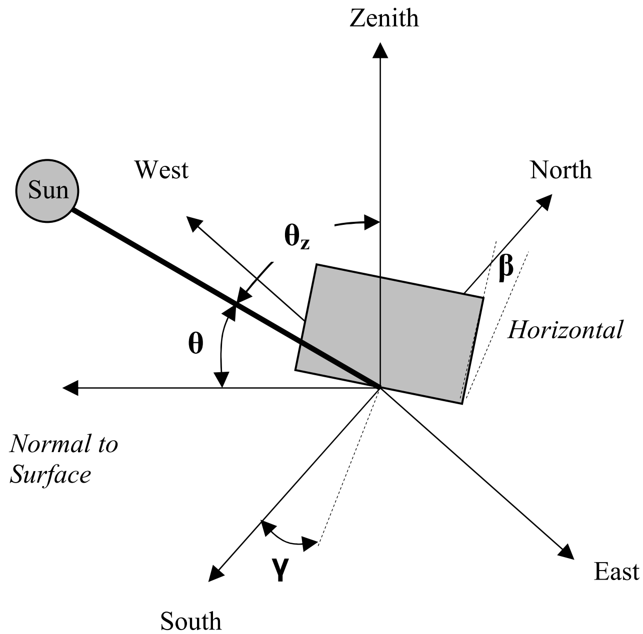

Figure 3.1.1 – Sun Angles[2]

These angles may be readily calculated on an hourly basis for the purpose of simulation.

[2] Beckman, pg. 14

3.1.1 Latitude

Latitude (φ) is the angular location north or south of the equator. Values for latitude range between -90^{o} and 90^{o}. HAVO is located at 19.4197^{o}N [3].

3.1.2 Declination

Declination (δ) is the tilt of the earth's axis relative to the incoming sun at solar noon. Values for declination range between -23.45^{o} and 23.45^{o}.

Declination for a given day of the year may be calculated by:

$$\delta = 23.45 \sin\left(360 \frac{284 + n}{365}\right)$$ Equation 3.1.2.1[4]

where *n* is the day of the year counted from January 1st

3.1.3 Slope

Slope (β) is the angle between the plane of the surface of interest and the horizontal. Values for slope range between 0^{o} (flat ground) and 180^{o} (overhang).

3.1.4 Surface Azimuth Angle

The surface azimuth angle (γ) is the deviation of the normal to the surface from the local meridian; 0^{o} due south, -90^{o} east, 90^{o} west, and 180^{o} north.

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[3] CASTNET

[4] Beckman, pg. 13

3.1.5 Hour Angle

The hour angle (ω) is the angular displacement of the sun east or west of the local meridian due to the rotation of the earth on its axis at 15^o per hour, morning negative, afternoon positive. For example, at 10 am solar time, the hour angle would be -30^o. The hour angle is computed by:

$$\omega = (\mathrm{AST} - 12)15^o \qquad \text{Equation 3.1.5.1}$$

where AST is adjusted solar time

Adjusted solar time varies from local time by the following relation:

$$\text{Solar Time - Standard Time} = 4(L_{\mathrm{st}} - L_{\mathrm{loc}}) + E \qquad \text{Equation 3.1.5.2}$$[5]

where:

L_{st} is the standard meridian for the local time zone (-155.0000^o for HAVO)
L_{loc} is the longitude of the current location (-155.2400^o for HAVO)
E is the equation of time

The equation of time is a correction factor which may be calculated using the following empirical relationship:

$$E = 229.2\begin{pmatrix} 0.000075 + 0.001868\cos B - 0.032077\sin B \\ -0.014615\cos 2B - 0.04089\sin 2B \end{pmatrix} \qquad \text{Equation 3.1.5.3}$$[6]

where $B = (n-1)\dfrac{360}{365}$ and n is the day of the year counted from January 1st

Hour angles are computed at the midpoint of each simulation hour, except for sunrise and sunset. In these cases, angles are computed at the midpoint between sunrise and end of the time period, or sunset and beginning of the time period.

[5] Beckman, pg. 11
[6] Beckman, pg. 11

3.1.6 Angle of Incidence

Angle of incidence (θ) is the angle between the beam radiation on a surface and the normal to that surface. Incidence angles are a function of other solar angles and may be calculated by equation 3.1.6.1.

$$\begin{aligned}\cos\theta = {} & \sin\delta\sin\phi\cos\beta \\ & -\sin\delta\cos\phi\sin\beta\cos\gamma \\ & +\cos\delta\cos\phi\cos\beta\cos\omega \\ & +\cos\delta\sin\phi\sin\beta\cos\gamma\cos\omega \\ & +\cos\delta\sin\beta\sin\gamma\sin\omega\end{aligned}$$

Equation 3.1.6.1[7]

where δ, Φ, β, γ, and ω are as defined above.

Even though this general formulation is complex, the above formula may be simplified for a number of practical cases (e.g. vertical/horizontal surfaces).

3.1.7 Zenith Angle

One particular simplification is for the calculation of zenth angles (θ_z). The zenith angle is defined as the angle between the vertical and the line to the sun (i.e. the angle of incidence of beam radiation on a horizontal surface).

For this case, $\beta = 0$ and the equation for the angle of incidence reduces to

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

Equation 3.1.7.1[8]

where δ, φ, and ω are as defined above.

3.1.8 Sunset Angle

[7] Beckman, pg. 15
[8] Beckman, pg. 16

A special hour angle is the sunset angle (ω_s) – the hour angle for which the zenith angle is 90^o (e.g., sun on the horizon). The sunset angle is given by equation 3.1.8.1.

$$\omega_s = \cos^{-1}\left(\frac{\sin\phi\sin\delta}{\cos\phi\cos\delta}\right)$$ Equation 3.1.8.1[9]

where ω_s, φ, and δ are as defined above

3.2 Incident Solar Radiation

When sunlight strikes the earth's atmosphere, some of it passes straight through, some is scattered, and the remainder is absorbed or reflected back into space. Solar radiation incident on a surface can be considered as three components: beam, diffuse, and reflected. Beam, or direct, radiation is the sunlight which passes through the atmosphere without being scattered or absorbed and strikes a surface at the angle of incidence. Diffuse radiation has been scattered by the atmosphere. Reflected radiation has been reflected off nearby surfaces to the surface in question (e.g. ground to house window). Generally, in terms of strength, beam radiation is greater than diffuse, which is greater than ground-reflected energy. However, on an overcast day, diffuse radiation may dominate due to increased scattering by clouds.

Total radiation on a surface is given as the sum of relevant beam, diffuse, and ground reflected radiation. Assuming diffuse and reflected radiation are isotropic, that is received uniformly from all directions, total radiation is given by equation 3.2.1.

$$I_T = I_b R_b + I_d\left(\frac{1+\cos\beta}{2}\right) + I\rho_g\left(\frac{1-\cos\beta}{2}\right)$$ Equation 3.2.1[10]

where:

I_T is the total radiation incident on the tilted surface
I_b is beam (or direct) radiation on a horizontal surface

[9] Beckman, pg. 19
[10] Beckman, pg. 95

I_d is the diffuse radiation on a horizontal surface
I is the total radiation on a horizontal surface
R_b is the ratio of beam radiation on a tilted surface to beam radiation on a horizontal surface. This is the ratio of incidence angle to zenith angle.
ρ_g is the fraction of radiation incident on the ground which is reflected (0.2 for grass)
β is the slope of the surface

The modifying terms on the second and third terms of equation 3.2.1 are view factors relating the fraction of the sky and ground, respectively that a surface can "see". For example, a surface flat on the ground will have a slope, $\beta = 0$. In this case, the view factor of the surface to the sky (relevant for diffuse radiation) will be unity; or rather the surface will be able to see the entire sky dome. For the same surface, the view factor to the ground will be zero (e.g., a surface on the ground can see none of the ground). For surfaces with $\beta > 0$, the surface will see some fraction of both sky and ground, up to $\beta = 180^o$, where the surface will see only ground. Unshaded vertical surfaces have a view factor of 0.5 for both sky and ground.

As a practical matter, not all diffuse radiation is isotropic. Therefore, computing radiation incident on a surface by equation 3.2.1 will underestimate the actual incident radiation. Diffuse radiation has three components: isotropic diffuse, circumsolar diffuse, and horizon brightening. As mentioned previously, isotropic diffuse radiation is scattered radiation which is received uniformly on a surface from the entire sky. Circumsolar diffuse radiation is directional, following closely on the same axis as beam radiation and results from forward scattering of solar radiation. Horizon brightening is, as the name suggests, diffuse radiation concentrated on the horizon. Figure 3.2.1 shows the components of solar radiation incident on a surface, with isotropic diffuse, circumsolar diffuse, and horizon brightening separately defined.

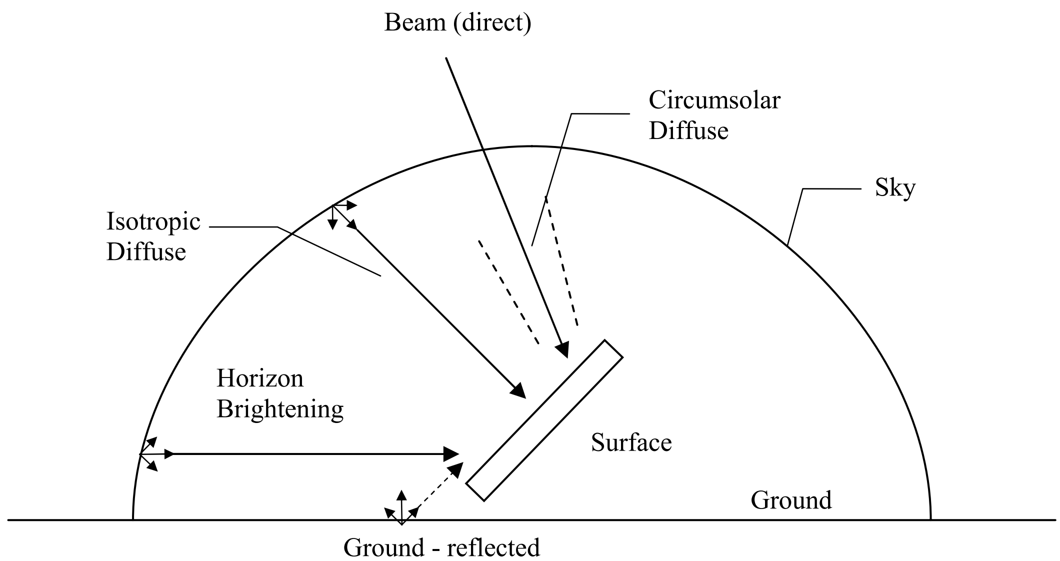


Figure 3.2.1 – Solar Radiation Components[11]

Assuming all circumsolar diffuse is incident at the same angle as beam radiation, circumsolar diffuse radiation on a tilted surface is given by

$$I_{d,Circumsolar} = I_d\left(A_i R_b\right)$$ Equation 3.2.3[12]

where A_i is the anisotropy index, which models the fraction of the diffuse radiation that should be treated as forward scattered. For a clear sky, with limited diffuse scattering, A_i will be highest. The anisotropy index may be computed as the ratio between beam radiation on a horizontal, terrestrial surface and horizontal, extraterrestrial radiation.

$$A_i = \frac{I_b}{I_o}$$ Equation 3.2.4[13]

The remaining balance of diffuse radiation is assumed to be isotropic, expanding the second term of equation 3.2.1 to

$$I_{d,T} = I_d\left(\left(1 - A_i\right)\left(\frac{1+\cos\beta}{2}\right) + A_i R_b\right)$$ Equation 3.2.5[14]

[11] Beckman, pg. 96
[12] Beckman, pg. 97
[13] Beckman, pg. 97
[14] Beckman, pg. 97

Horizon brightening, also non-directional, is accounted for by the addition of a correction factor, F to the isotropic diffuse term. F is given by

$$F = 1 + f \sin^3\left(\frac{\beta}{2}\right)$$ Equation 3.2.6[15]

where f is a correction for cloudiness, given by

$$f = \sqrt{\frac{I_b}{I}}$$ Equation 3.2.7[16]

where I is total radiation incident on a horizontal surface

For clear conditions, with limited scattering, f will be largest, and the contribution by horizon brightening the greatest.

Incorporating horizon brightening, total diffuse radiation on a tilted surface is given by

$$I_{d,T} = I_d\left[(1 - A_i)\left(\frac{1+\cos\beta}{2}\right)F + A_i R_b\right]$$ Equation 3.2.8[17]

Substituting for F, and grouping circumsolar diffuse with beam radiation, the total radiation on a tilted surface is given by

$$\begin{aligned} I_T &= (I_b + I_d A_i)R_b \\ &+ I_d(1 - A_i)\left(\frac{1+\cos\beta}{2}\right)\left(1 + f\sin^3\left(\frac{\beta}{2}\right)\right) \\ &+ I\rho_g\left(\frac{1-\cos\beta}{2}\right) \end{aligned}$$ Equation 3.2.9[18]

This model, referred to as the HDKR model (named such for work by Hay, Davies, Klucher, and Reindl), offers a good balance between agreement with empirical observation and ease of use[19]. The HDKR model has been implemented in our work to calculate incident radiation on a surface on an hourly basis.

[15] Beckman, pg. 97
[16] Beckman, pg. 97
[17] Beckman, pg. 97
[18] Beckman, pg. 98
[19] Beckman, pg. 102

4. BUILDING ENERGY REQUIREMENT

For the purposes of this study, the overall energy requirement for the building is the demand for space heating and hot water.

4.1 Space Heating

Space heating energy requirements are determined by the difference between heat lost from the residence and heat gained (either through internal generation or passive solar).

4.1.1 Heat Loss

Heat loss from the residences occurs through two sets of mechanisms: conduction and infiltration. Conductive losses occur as heat flows through the walls, driven by the temperature gradient between the environment and conditioned spaces. Infiltration losses occur as air moves between the interior and environment via any number leaks (e.g., windows, doors, etc.).

4.1.1.1 Conduction

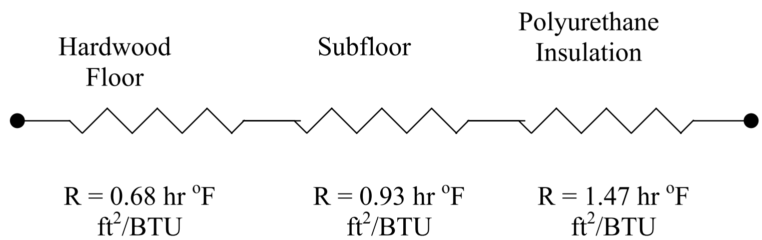
Conductive heat loss is given by the following equation.

$$Q_{\text{Conduction}} = UA\left(\mathrm{T}_{\text{interior}} - \mathrm{T}_{\text{ambient}}\right) \qquad \text{Equation 4.1.1.1}$$

where:

$Q_{\text{Conduction}}$ is the heat loss due to conduction
U is the building loss coefficient (conductance)
A is the structure surface area
$\mathrm{T}_{\text{interor}}$ and $\mathrm{T}_{\text{ambient}}$ are indoor and outdoor temperatures, respectively

Since ambient and interior conditions may be specified with reasonable certainty, and the building surface area is known, the greatest source of uncertainty lies in determining an appropriate loss coefficient. Assuming a series model for resistance to heat flow, loss coefficients may be determined by adding the thermal resistances of individual components. Thermal resistance *R* is the inverse of the conductance *U*. Figure 4.1.1.1 shows the component resistances for a building floor.



$$R_{\text{Total}} = R_{\text{Hardwood}} + R_{\text{Subfloor}} + R_{\text{Insulation}} = 0.68 + 0.93 + 1.47 = 3.08\,\frac{\text{hr }^{\text{o}}\text{F ft}^2}{\text{BTU}}$$

Figure 4.1.1.1 – Series resistance model for building floor

Like solid objects, gasses (in this case air) also have a resistance to heat flow. Therefore, a convective resistance term must be included for both interior and exterior surfaces. To be rigorous, convective resistance is a function of external wind speed, surface roughness, orientation, and film temperature. However, for the purposes of this analysis, constant values are used which are a function only of orientation, in line with ASHRAE guidelines[20]. Table 4.1.1.1 shows a full loss coefficient calculation for a building wall. R values are given in (hr ft^2 $^{\text{o}}$F)/BTU.

Table 4.1.1.1 – Sample Loss Coefficient Calculation

Surface	**R**	**U (=1/R)**
Vertical Interior Surface	0.68	
Drywall	0.45	
Still Air Gap	1.01	
Exterior Sheathing	0.93	
Exterior Siding	1.30	
Vertical Exterior Surface	0.25	
Total	**4.62**	**0.216 BTU/hr ft^2 $^{\text{o}}$F**

[20] ASHRAE Fundamentals 23.12

This calculation is repeated for each building surface, including windows. Complete calculations are presented in Appendix A.

4.1.1.2 Infiltration

Infiltration losses are modeled using the ASHRAE air change method[21]. This method was selected over explicit crack techniques due to ease of use and lack of site specific information. Infiltration losses are given by the following relation.

$$Q_{\text{Infiltration}} = \rho C_p V n \left(T_{\text{interior}} - T_{\text{ambient}}\right) \qquad \text{Equation 4.1.1.2}$$

where:

$Q_{\text{Infiltration}}$ is the heat loss due to infiltration
ρ is the density of air (0.075 lb/ft^3 at standard conditions)
C_p is the specific heat of air (0.24 BTU/lb ^{o}R at standard conditions)
V is the volume of the room
n is the number of air changes per hour for the room
T_{interor} and T_{ambient} are indoor and outdoor temperatures, respectively

ASHRAE recommends the number of air changes per hour based on the number of exterior walls in a given room. Values for each room of the house are shown in Appendix B. Since the number of air changes is an estimate, assumptions have been verified with a professional HVAC engineer[22]. The number of hourly air changes varies from two for rooms with windows or doors on three exterior walls, to one half for rooms without windows or doors.

4.1.2 Offsetting Gains

4.1.2.1 Internal Generation

There are two main sources of heat generation within a building – appliances and occupants. For the purposes of our analysis, we assumed two occupants per residence

[21] ASHRAE Fundamentals, 22.8
[22] Elder

and a minimal number of appliances – stove, refrigeration, and lighting. Heat generation for occupants and appliances is given in Table 4.1.2.1.

Table 4.1.2.1 – Internal Generation Assumptions

Object	Unit Heat Generation (BTU/hr)	Number of Units
Occupant	341	2
Stove	5120	1
Refrigerator	2474	1
Light (60W)	205	15

Combining these generation numbers with usage assumption (e.g. number of lights on in morning and evening) we may construct a simple model for hourly internal generation. A daily generation curve for a HAVO residence is presented in figure 4.1.2.1.

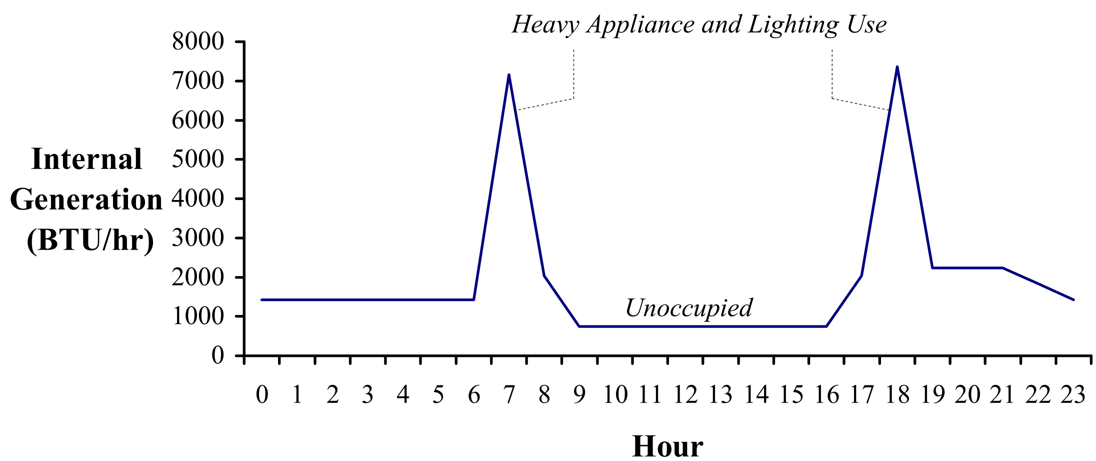


Figure 4.1.2.1 – Daily Internal Generation (BTU/hr)

Most internal generation occurs in the morning and evening when the residence is occupied and appliances see heaviest use. During the day, when the building is empty, the only source of internal generation is assumed to be the refrigerator.

4.1.2.2 Passive Solar Gain

Passive solar gain also serves to offset losses from conduction and infiltration. While all surfaces of the structure (walls, windows, and roof) absorb solar radiation, the only gain considered in most treatments is via windows. In this case, radiation transmitted through the glass is absorbed by interior surfaces. Building walls and roof absorb radiation on the exterior surface, much of which is reradiated to the environment. Passive gains to the living area via the roof are further buffered by the unconditioned space of the attic.

In order to calculate gain through a window one must determine incident radiation on the window (including allowances for shading) and the fraction of that radiation which is transmitted through the glass and absorbed by the room.

While neither residence Q23 nor Q25 is significantly shaded by surrounding trees, the roof does overhang the sides of the building, providing some shading. Overhangs block incident radiation in two ways. For some incidence angles, they block directional radiation – both beam and circumsolar diffuse. They also limit the sky view factor for the overhung surface, reducing isotropic diffuse radiation. Radiation from horizon brightening and radiation reflected by the ground are assumed to be unaffected by the overhang[23].

The fraction of a window shaded from directional radiation is a function of the incidence angle and window geometry, as shown in figure 4.1.2.2.

[23] Bath, Appendices

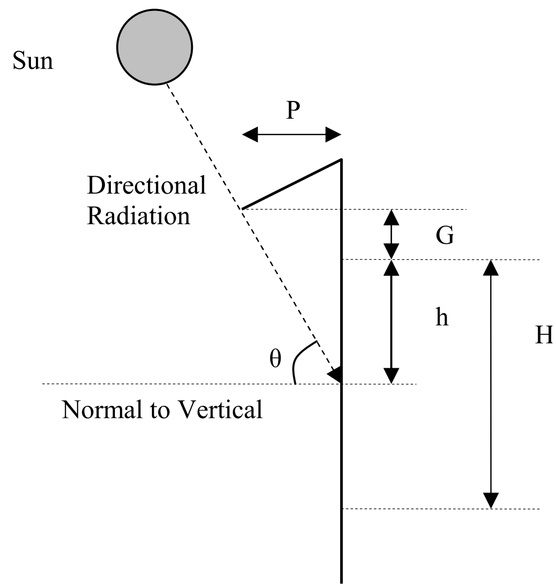


Figure 4.1.2.2 – Directional Shading

where:

P is the horizontal projection of the overhang (est. 18")
G is the gap between the top of the window and overhang projection (est. 5")
H is the height of the window
h is the height of the window which is shaded
θ is the angle of incidence as defined previously

Using basic trigonometry, the fraction of the window shaded for a given angle of incidence is

$$\mathrm{FS} = \frac{P\tan\theta - G}{H} = \frac{h}{H} \qquad \text{Equation 4.1.2.2.1}$$

Directional radiation incident on a shaded surface is therefore, the directional radiation incident on an unshaded surface multiplied by 1-FS.

Utzinger and Klein[24] have developed a method for computing the restricted sky view factor for isotropic diffuse radiation. Normalizing window geometry by the height of the

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[24] Beckman, pg. 558

window (H), yields non-dimensional overhang length (*p*), gap (*g*), and window width (*w*). Tabulated values for sky view factors are given in Duffie and Beckman as a function of *p*, *g*, and *w*. For the small overhang on the HAVO residences the sky view factor is reduced to 0.46, from 0.5 for an unshaded vertical surface.

In addition to radiation blocked by shading, some of the energy transmitted through a window will be re-radiated back through the window. The fraction of incident solar energy absorbed by a room is given by equation 4.1.2.2.2[25].

$$\tau_c \alpha_{eff} = \tau_c \frac{\alpha_i}{\alpha_i + (1-\alpha_i)\tau_d \frac{A_a}{A_i}} \qquad \text{Equation 4.1.2.2.2}$$

where:

τ_c is the transmittance of the window glazing for incident solar radiation (0.87)
τ_d is the transmittance of the window glazing for isotropic diffuse radiation (0.74)
α_{eff} is effective absorptivity of the room
α_i is the absorption of the room surfaces (0.45)
A_a is the area of the windows
A_i is the area of the room (less the windows)

4.1.3 Building Capacitance

Heat stored within the building structure serves to buffer ambient temperature changes. Building materials, such as drywall, store significant amounts of heat during the day and release it at night, as ambient temperatures drop. The relative energy stored in an object is given by equation 4.1.3.1.

$$\mathrm{Q_{Stored}} = \mathrm{mC_p}\left(\mathrm{T_{Object}} - \mathrm{T_{Reference}}\right) \qquad \text{Equation 4.1.3.1}$$

where:

Q_{Stored} is the heat stored by the component

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25 Beckman, pg. 246

m is the object's mass
C_p is the object's specific heat
$T_{Reference}$ and T_{Object} are the reference temperature and object temperature, respectively

In theory, it would be possible to apply equation 4.1.3.1 to every structural component to calculate the energy stored within the structure. In practice, this summation would be overly complex and prone to error given the number of components and spatial temperature gradient across each of them. A simplification would be to assume that the majority of heat is stored within drywall, which is uniformly at the same temperature as the interior. Given the high density and low thermal resistance of drywall relative to other building materials, this assumption would appear to be reasonable. With this approximation equation 4.1.3.1 may be rewritten as

$$Q_{Stored} = \left(mC_p\right)_{Drywall}\left(T_{Interior} - T_{Thermostat\ Set}\right) \qquad \text{Equation 4.1.3.2}$$

where:

Q_{Stored} is the relative heat stored by the structure
mC_p is the product of the mass of interior drywall (walls and ceiling) and specific heat (0.26 BTU / lb ^{o}F)
$T_{Interior}$ is the interior temperature of the structure
$T_{Thermostat\ Set}$ is the thermostat set temperature (65 or 70^{o}F)

4.1.4 Net Energy Requirement

For a given hour, the net energy requirement is given by the difference between total load and total gain. Figure 4.1.4.1 gives a view of competing sources of heat gain and loss.

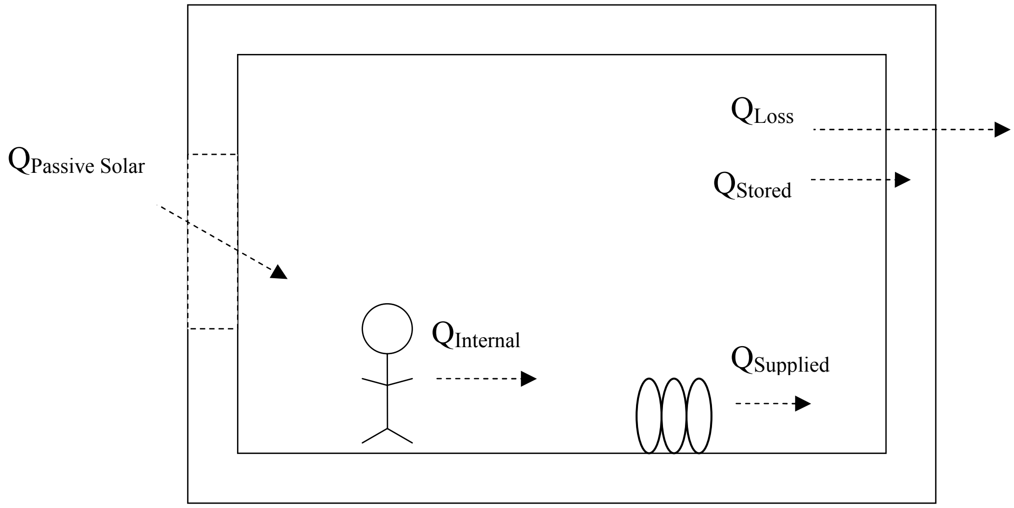

Figure 4.1.4.1 – Sources of Loss and Gain

Building capacitance considerations require calculating the indoor temperature for each hour, to determine how much energy has been stored by the structure.

An energy balance for the residence is given by equation 4.1.4.1.

$$Q_{\text{Passive Solar}} - Q_{\text{Loss}} = mC_p \frac{dT}{dt} - Q_{\text{Internal Generation}} \qquad \text{Equation 4.1.4.1}$$

where:

T is indoor temperature
t is time
mC_p is the product of the mass and specific heat of interior drywall (storage term)
Q's are heat loads and gains as defined previously

Assuming the only variable quantity over the course of the hour is temperature, integration of equation 4.1.4.1 yields

$$T^+ = T + \frac{\Delta t}{mC_p}\left(Q_{\text{Passive Solar}} + Q_{\text{Internal Generation}} - Q_{\text{Loss}}\right) \qquad \text{Equation 4.1.4.2}$$

where:

T^+ is the temperature at the end of the hour

T is the temperature at the beginning of the hour
Δt is the time elapsed from start to end (1 hour)

Incorporating building capacitance into calculations helps close the loop and provides a check to modeled numbers. If loads or gains have been correctly calculated, internal temperatures should fall within a range anticipated by normal use. Ideally, the model could be checked for accuracy by recording indoor and outdoor temperatures at the site over a period of time and comparing the actual to modeled performance. However, such data was not available.

Figure 4.1.4.2 shows the average daily space heating energy requirement for a single residence.

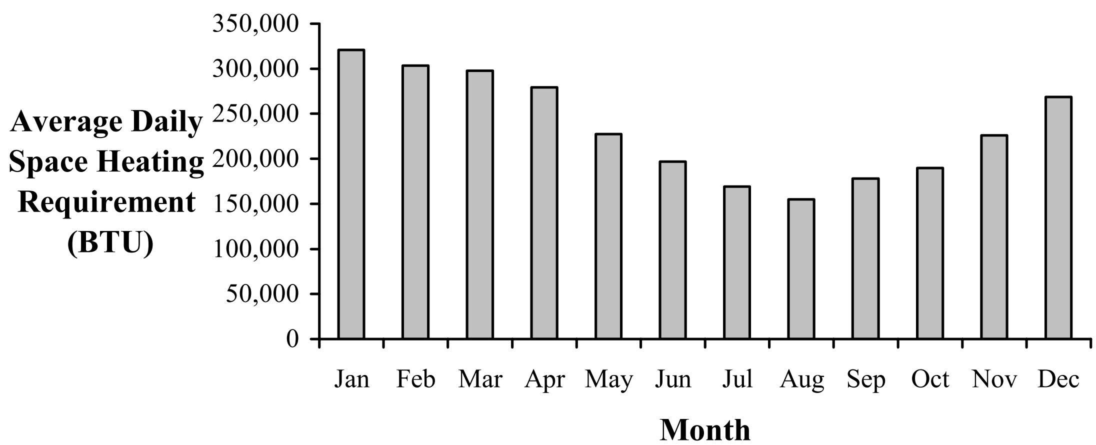

Figure 4.1.4.2 - Daily Average Space Heating Energy Requirement (single residence)

4.2 Domestic Hot Water

Domestic hot water loads are relatively constant throughout the year. While significantly lower than space heating, the load still significantly impacts the overall energy demand. The hot water load is be the energy required to heat supplied water to a temperature desirable for domestic use. Hourly hot water loads are calculated by equation 4.2.1.

$$Q_{DHW} = mC_p\left(T_{Demand} - T_{Supply}\right) \qquad \text{Equation 4.2.1}$$

where:

Q_{DHW} is the domestic hot water load
m is the mass of water demanded for the hour
C_p is the specific heat of water (4.18 kJ/kg at standard conditions)
T_{Supply} and T_{Demand} are, respectively, the temperature of the water supply and the service hot water temperature for the occupants (110 - 140 ^{o}F)

The daily hot water volume is not assumed to vary with season and is considered a function only of the number of occupants. Each occupant is assumed to consume 20 gallons of hot water each day for showering, food preparation, and dish washing[26]. The percentage of the daily 20 gallon per person load consumed each hour is given in figure 4.2.1

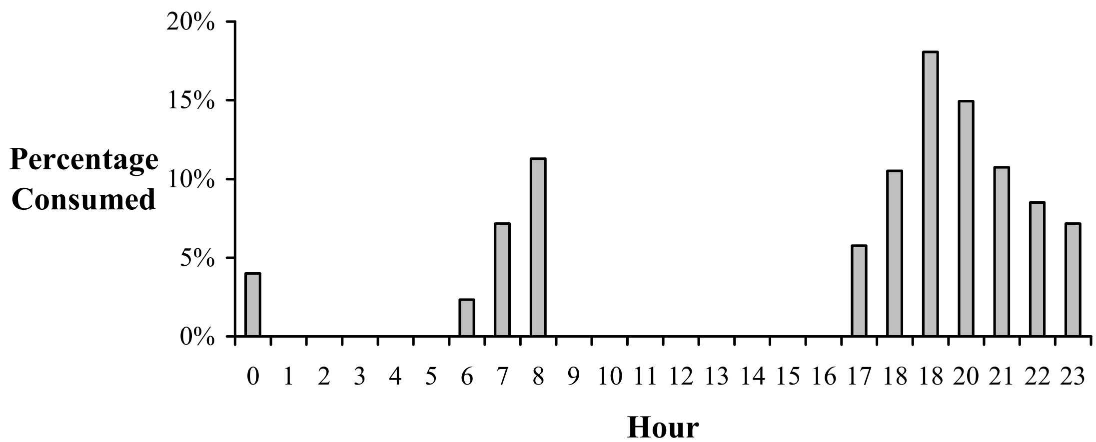

Figure 4.2.1 – Hourly Percentage of Total Domestic Hot Water Consumption

Limited seasonal variation does occur due to fluctuations in supply temperature. HAVO is not connected to a water utility and water needs for the park are satisfied through the collection of rain water. This water is stored in uninsulated, above-ground tanks[27], which are assumed to track closely to ambient temperature. Monthly average ambient temperatures for HAVO were used to model the hot water supply temperature since the

[26] Kreider, pg. 415
[27] Malte

volume of water stored in tanks is likely to buffer daily variations. Calculated average daily hot water loads are shown by month in figure 4.2.

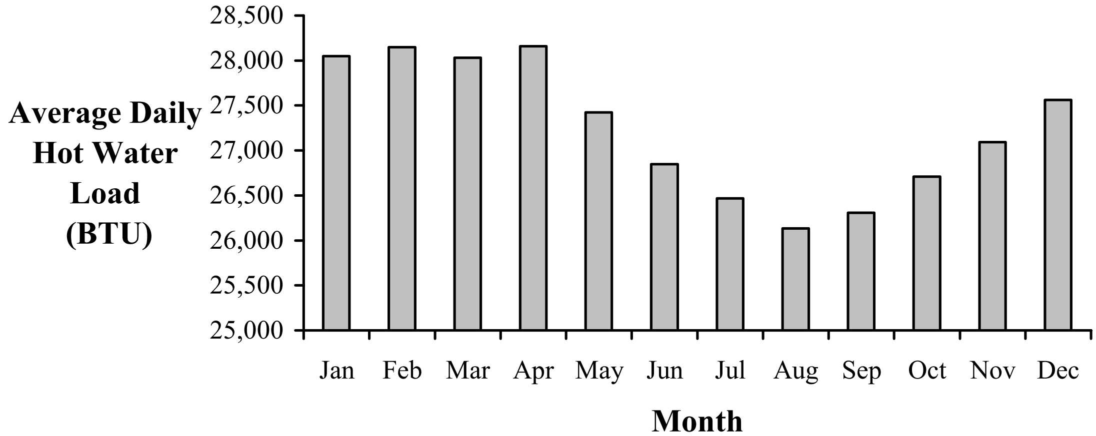

Figure 4.2 – Daily Average Hot Water Load (single residence)

5. STRUCTURE

5.1 Overview

Residences Q23 and Q25 at HAVO are relatively small buildings, designed for comfortable use by two people or a small family. Each building has one floor of conditioned space with a footprint of roughly 800 ft^2. This space is split between two bedrooms, a full bath, kitchen, and living room. Each room has at least one window to provide some daylighting. Unfortunately, there are few south facing windows, limiting the buildings' passive solar gain. The living floor sits roughly five feet off the ground, atop an open space used for storage. The only insulation in the structure is a coating of ¼" spray-on polyurethane applied to the sub-floor and joists beneath the living area. The existing roof is sheet metal, which provides virtually no insulation. A detailed floor plan is given in Appendix C.

5.2 Exterior Walls

The exterior walls of the building are uninsulated tongue and groove construction. Figure 5.2 shows the wall make-up in detail, with R-values for each component[28].

[28] ASHRAE Fundamentals, 23.12-23.32

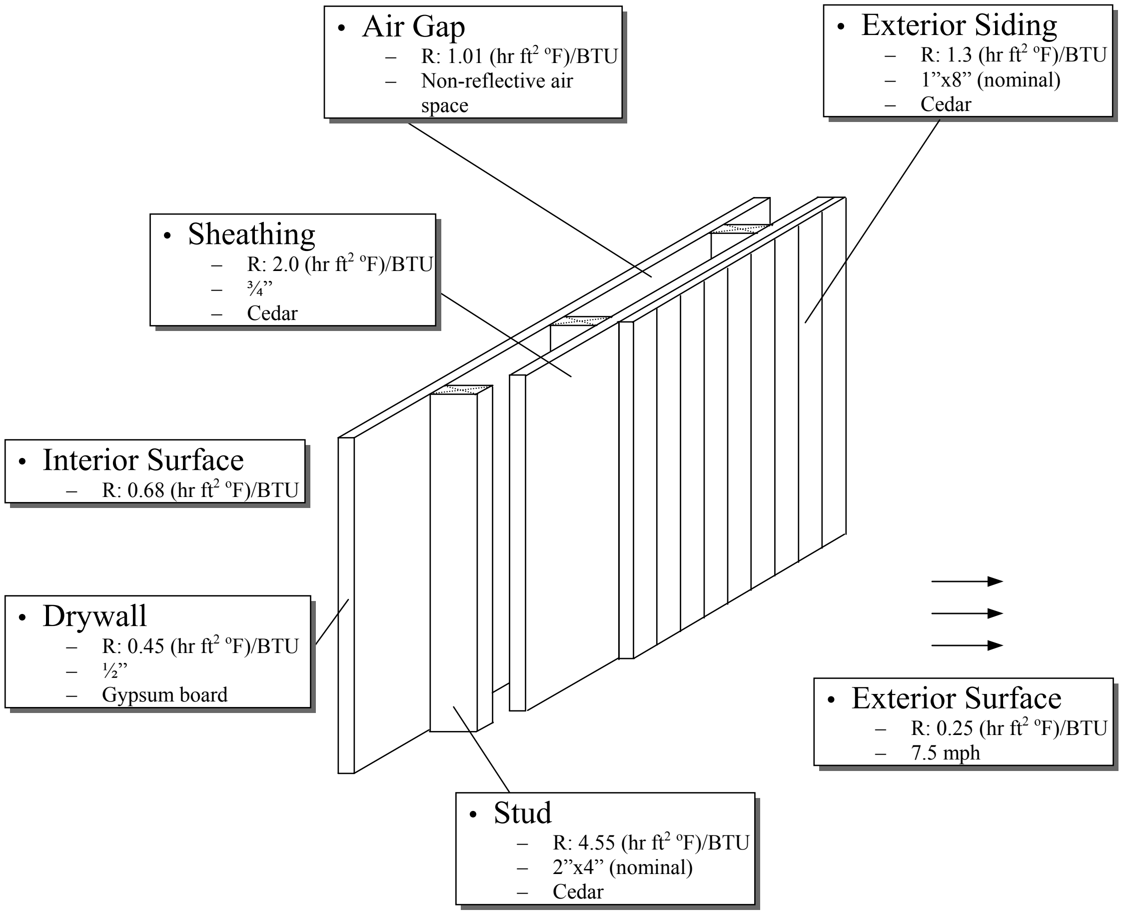

Figure 5.2 – Exterior Wall Construction

Loss coefficients for all components and scenarios considered may be found in Appendix A.

5.3 Windows

Residence windows are single-pane, wood frame. This basic construction results in a relatively low R-value of approximately 1 ft^2 hr ^oF/BTU[29], including both the conductive resistance of the window and convective air films on either side.

[29] Vestergaard-Hansen

5.4 Floor

As in most construction, the sub-floor is overlaid on a frame of 2x8 (nominal) joists. Floor covering varies throughout the residence – hardwood, carpet, and tile. While the joists themselves have a significant resistance to heat flow, the low resistance of the sub-floor means nearly all the heat will bypass the joists and flow directly into the open space beneath the house. If the space between the joists were significantly insulated, then the insulating properties of joists should be accounted for in calculations. This is important when evaluating different scenarios. Figure 5.4 shows R-values for the floor of the residences.

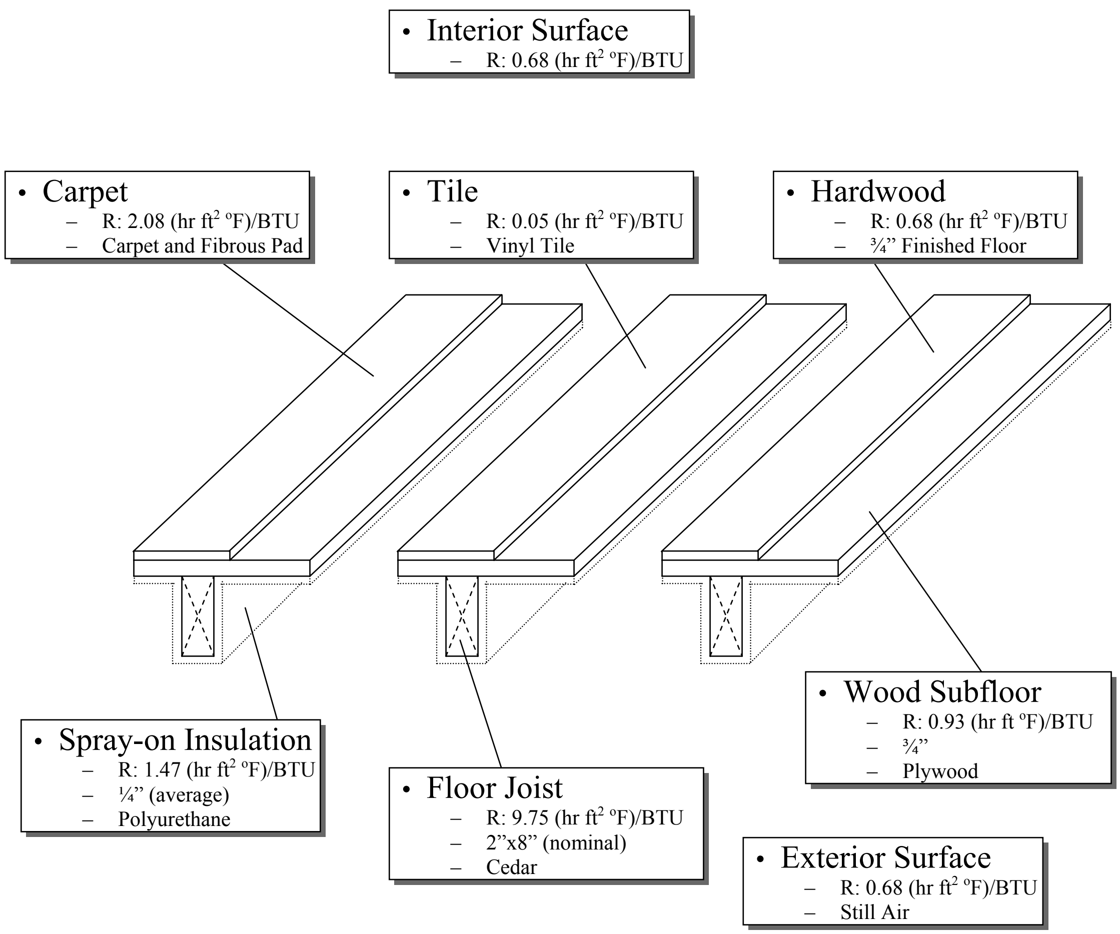

Figure 5.4 – Floor Construction

5.5 Basement

The basement is a half-height unfinished space beneath the living area. One wall is composed entirely of wooden slats, so the basement is assumed to track ambient conditions, but with quiescent air. The floor joists and exposed subfloor have been insulated with spray-on polyurethane to reduce heat loss from the living space above. However, this type of insulation often loses effectiveness over time[30]. Since this insulation was applied several years ago, the R-value used in this study is one half the value suggested by ASHRAE for new insulation.

5.6 Ceiling

The living area ceiling is gypsum board, mounted to a frame of 2x10 (nominal) joists. The ceiling is uninsulated. Standard building practice in more temperate climates is to insulate the space above the ceiling with blown-in fill in order to achieve R values on the order of 33 ft^2 hr ^oF/BTU.

5.7 Roof

The building roof is sheet metal tacked to rafters. While this does serve to keep out the elements, this type of roofing provides negligible insulation owing to the high thermal conductance of metal. Work is underway to replace the existing roof with a more standard shingled surface[31]. In addition to aesthetic improvement, this replacement will further reduce heat loss. A standard roof consists of a layer of thin plywood sheathing over the rafters, a permeable felt membrane, and asphalt shingles. For modeling purposes, a shingled roof is assumed.

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[30] ASHRAE Fundamentals, 23.19
[31] Cortez

5.8 Overall Structure

The product of the U-values and area of each component may be added together and combined with infiltration losses to obtain a loss coefficient for the entire structure. Table 5.8.1 shows the loss coefficients (BTU/hr ^{o}F) for the residences.

Surface	Loss Coefficient (BTU/hr ^{o}F)	Fraction Total
Walls	142	11%
Ceiling	416	32%
Floor	366	29%
Windows	199	16%
Doors	18	1%
Infiltration	143	11%
Total	1283	

Table 5.8.1 – Residence Loss Coefficients

Table 5.8.1 suggests a number of possibilities for reducing the residence's loss coefficient – and which improvements should be prioritized. Since the walls, floors, and ceiling contribute to more than 70% of the total heat loss, additional insulation will significant improve the overall loss coefficient. By comparison, replacing the existing doors with thicker, better insulated doors, will have virtually no impact total heat loss.

5.9 Capital Improvements

To improve the effectiveness of a solar thermal system it is desirable to reduce the building heat loss. Three ways to achieve this would be to add insulation, replace the existing windows, and weatherstripping doors and windows. Table 5.9.1 shows the impact of each of these improvements on the component's loss coefficients and figure 5.9.1 shows the overall impact of each improvement.

Table 5.9.1 – Insulation Upgrade Impact on Loss Coefficients

Surface	Initial Loss Coefficient (BTU/hr ^oF)	Insulation Upgrade	Reduced Loss Coefficient (BTU/hr ^oF)
Walls	142	R-11 Fiberglass Batt Insulation	54
Ceiling	208	R-33 Cellulose Insulation	29
Floor	183	R-33 Fiberglass Batt Insulation	30
Windows	199	Double pane, low emissivity window	68
Infiltration	143	Weatherstripping	95

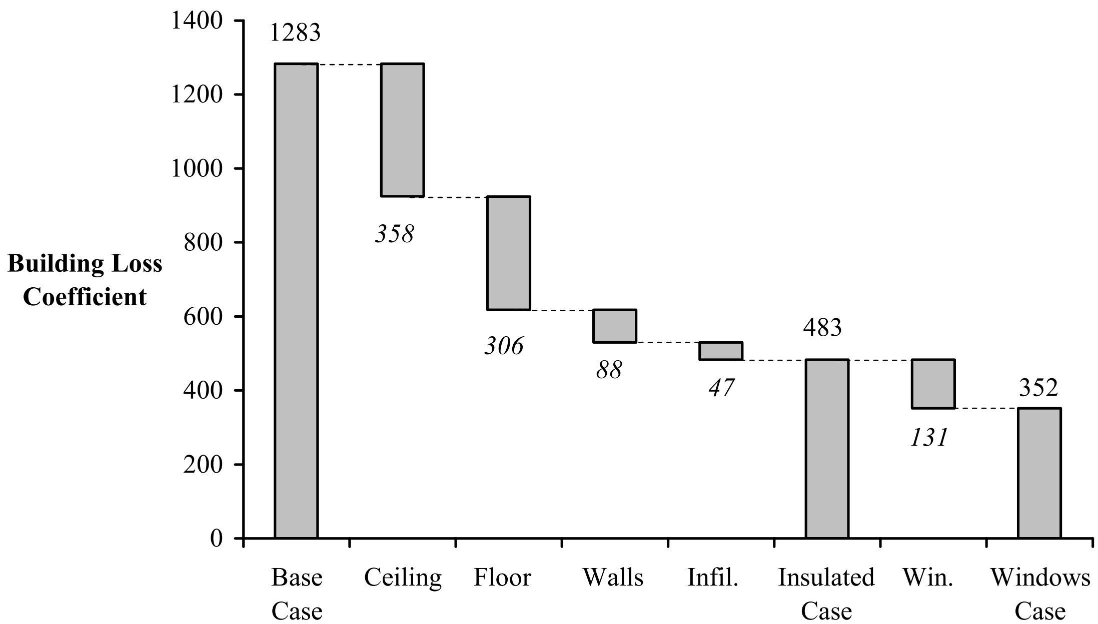

Figure 5.9.1 – Insulation Upgrade Impact on Building Loss Coefficient

Figure 5.9.1 shows two tiers of upgrades. The first set – insulating the ceiling, floor, and walls – is considered to be relatively low cost. The second tier – replacing the windows – is a high cost improvement. Interestingly, the most substantial gains are achieved through lower cost improvements. Each of these upgrades will now be considered in detail.

5.8.1 Insulation

Since neither residence is insulated (beyond the previously mentioned thin layer of polyurethane beneath the living area floor) insulating the walls, ceiling, and floor will significantly decrease heat loss through these surfaces. This would significantly reduce the building loss coefficient at a relatively low cost to the National Park Service. Since interior work is planned to upgrade the wiring[32], it would be logical to insulate the walls in parallel with this work. For wall installations, fiberglass batts are preferred for ease of installation. A 3 ½" thick batt with an R-value of 11 (hr ^{o}F ft^{2})/BTU would be suitable. If wiring upgrades do not occur, it would also be possible to blow loose fill cellulose insulation into the walls to achieve a similar R-value. For the floor, the existing polyurethane could be enhanced through the use of thicker batts installed between the floor joists. 9" thick batts with an R-value of 33 would be appropriate. For the attic, 9" of loose-fill cellulose insulation would also provide an insulating layer with an R-value of 33.

5.9.2 Windows

Double-pane windows would significantly reduce heat loss through the windows. These windows can achieve R-values as high as 2.86 (hr ^{o}F ft^{2})/BTU by filling the space between the two panes with an inert gas, such as Argon. Since replacement windows require more substantial investment than insulation, replacement should be justified by cost-benefit analysis.

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[32] Cortez

5.9.3 Weatherstripping

Weatherstripping could be employed to reduce infiltration losses by roughly 30%[33]. Weatherstripping is installed along the interior seams of windows and doors to reduce leakage. Though infiltration occurs by a fundamentally different mechanism than conductive losses (e.g. loss through walls), the ASHRAE air change method allows us to consider insulation as simply another loss coefficient added to the total. The effective loss coefficient for infiltration losses is 143 BTU/hr ^{o}F, nearly the same as the walls in the base case.

[33] Elder

6. SOLAR THERMAL SYSTEM COMPONENTS

An active solar system has two functions: collection of solar energy and distribution of that energy to end-use applications. This section provides a general overview of solar thermal system components. Specific recommendations for HAVO will be made in Section 9.

6.1 Collection and Storage

6.1.1 Collector

Two types of collectors are commercially available: flat plate and evacuated tube. Flat plate collectors are the most traditional design. A cross section of a flat-plate collector is shown in figure 6.1.1.1.

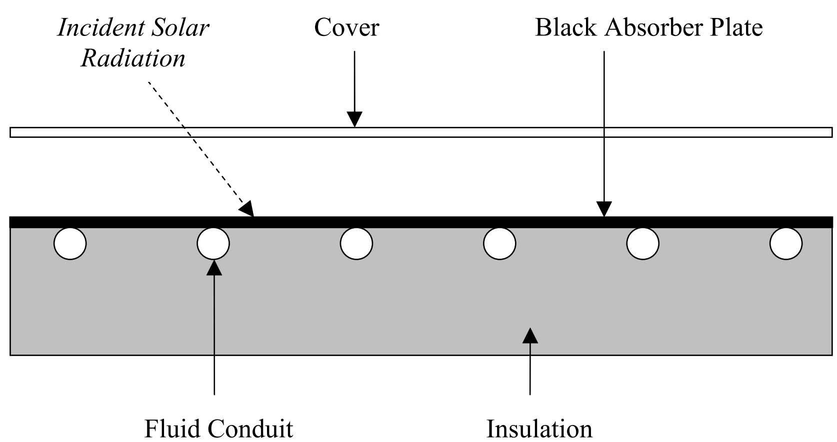

Figure 6.1.1.1. – Flat Plate Collector[34]

Light incident on the collectors in transmitted through the cover plates and absorbed by the black absorber plate, which then transfers the solar energy to the fluid conduits. The gap between absorber plate and cover limits the convective and radiation losses to the environment. Likewise, the insulation behind the fluid conduits reduces conductive

[34] Beckman, pg. 251

losses. Flat plate collectors operate at moderate temperatures, generally lower than 180^{o}F. This makes them a good match with residential space heating and hot water application, which also operate at low temperatures.

Evacuated tube collectors operate on a significantly different principle. A small amount of fluid is located within a tube mounted on an upward slope. Solar radiation vaporizes the liquid, which then rises due to lower density. The vapor then condenses on a heat sink, transferring energy to a working fluid on the opposite side, and falls back to the bottom of the tube to be vaporized again. The working fluid passes through the top of the array, absorbing heat from each of the evacuated tubes as it condenses vapor. The space surrounding the tubes (or heat pipes) is evacuated, reducing conductive losses relative to flat-plate systems. Evacuated tube systems may operate at significantly higher temperatures than flat-plate collectors (up to 302^{o}F). A representative schematic of a single evacuated tube is shown in figure 6.1.1.2.

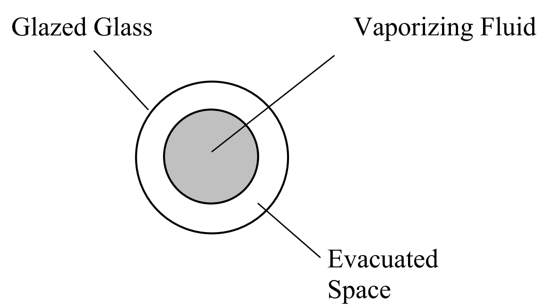

Figure 6.1.1.2 – Evacuated Tube (Top Down View)

While evacuated tube collectors exhibit superior performance, complexity in fabrication and lower sales volume result in higher cost (by about 30%) relative to more traditional flat plate systems. Evacuated tube collectors are more appropriate for higher temperature, commercial applications.

6.1.2 Energy Storage

Since peak solar incidence (day) and peak thermal energy requirements (night) do not coincide, a storage mechanism is necessary to capture solar energy so it may be used as needed. For liquid collector loops, the most common storage medium is water held in an insulated tank. Over the course of a day, energy absorbed by the collector is transferred to the water tank. This transfer can be accomplished either by drawing the collector working fluid directly from the tank (open loop system) or indirectly via a heat exchanger (closed loop system). Loop types will be discussed in detail in section 6.1.3.

Mathematically, an optimal tank size exists, balancing reduced thermal losses against achievable tank temperature. As tank size increases, the surface to volume ratio decreases. Since the energy stored in a tank scales with the volume and losses scale with surface area, the relative thermal loss decreases with increasing tank size (i.e. losses increase, but the thermal storage capacity of the tank increases even more). However, the higher thermal capacity of a larger tank also means that for the same energy input, a smaller tank will reach a higher temperature than a larger tank. If too large a tank is selected, it will not be possible to achieve operating temperatures required for space heating or domestic hot water from solar energy alone. Recommendations for optimal storage capacity are on the order of 2 gallons storage capacity per ft^2 of collector area[35]. In practice, tanks are available only in incremental sizes (e.g. 100, 200, 300 gallons). Most storage tanks incorporate a single or double walled heat exchanger to accommodate a closed loop collector system.

[35] Beckman, pg. 694

6.1.3 Loop Type

Three types of collector loops are commonly used. These are open loop systems, anti-freeze closed loops, and drainback closed loops. Each system entails differing levels of complexity, as well as mechanisms to protect against freezing or boiling.

6.1.3.1 Open Loop

Open loop systems are the simplest type of collector loop. Water is drawn from the storage tank, pumped through the collectors where it absorbs incident radiation, and returned to the tank. Since cold water is denser than warm, it is possible to draw the coldest water from the bottom of the tank. The warmest water will be at the top of the tank and may be drawn off for domestic hot water or space heating needs. A representative schematic is shown in figure 6.1.3.1.

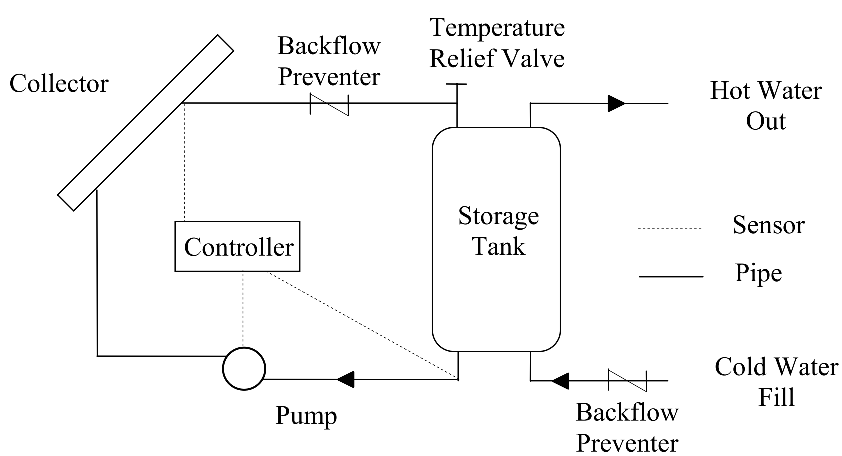

Figure 6.1.2.1 – Open Loop System

Open loop systems contain relatively few components, incorporating a pump, backflow preventer to keep water from being drawn back through the collector when the pump is switched off, and a temperature relief valve to protect the system. A controller, described in detail in section 6.1.5, regulates pump operation.

If water freezes within piping or the collectors, the expansion will cause significant damage to the system. As a result, a critical factor in any design is to assess the potential for freezing and incorporate mechanisms to avoid freezing. Of the three types of systems, open loop designs are the most susceptible to freezing. If temperature sensors detect near-freezing conditions, the pump switches on, drawing warm water from the storage tank to keep the collector and pipes above freezing. This solution results in lowered efficiency since energy used to heat the collector is not available for domestic hot water or space heating. Additionally, if electric power is lost, the pump will be inoperative and freezing may occur.

Open loop systems are named such since as water is drawn from the tank for hot water applications it is replenished by a fresh, cold feed. As such, new oxygen is constantly introduced into the loop, which can lead to corrosion of steel and cast iron. For open loop applications, system components exposed to the working fluid should be either copper, bronze, brass, stainless steel, plastic, or glass[36].

6.1.3.2 Anti-Freeze Closed Loop

One way to provide freeze protection without sacrificing as much efficiency is to use an anti-freeze solution in place of water as the working fluid. Anti-freeze systems are the most common type of closed loop available. Propylene glycol is often used as the working fluid, since it is less toxic than ethylene glycol mixtures commonly used in automobiles. Propylene glycol is not potable and, therefore, may not be used directly for domestic hot water. Heat is transferred between the collector loop and storage tank using a heat exchanger, which keeps the potable and non-potable fluids separate. Most storage tanks sold for residential use incorporate a double-walled heat exchanger to

[36] Olson

accommodate closed loop designs. Figure 6.1.3.2 shows an anti-freeze closed loop system.

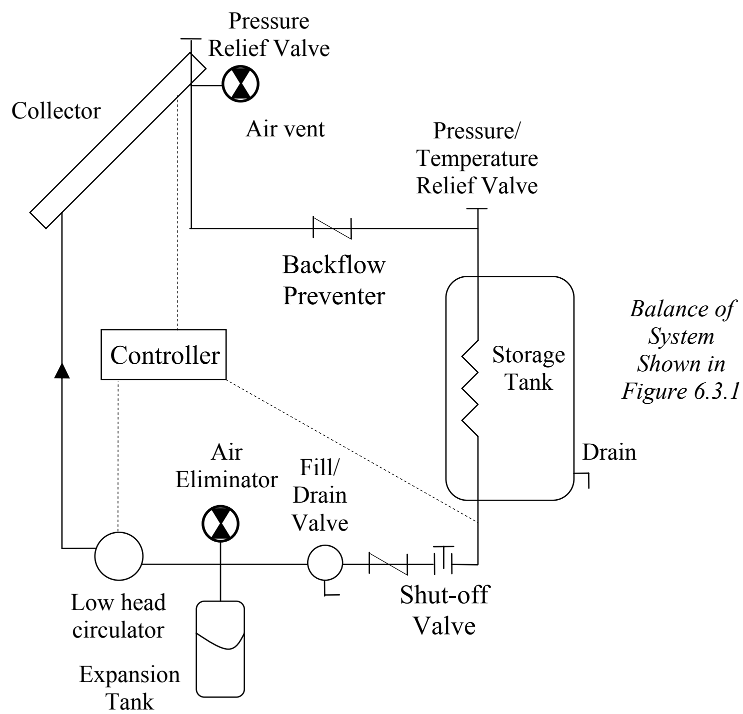

Figure 6.1.3.2 – Closed Loop – Anti-Freeze

Closed loop systems require several additional components beyond those used in open loop systems. In addition to the previously mentioned heat exchanger, an expansion tank, pressure relief valve, and air vent are required for operation. These components will be discussed in detail in section 6.1.4. Because closed loop systems are not constantly exposed to oxygen, cheaper materials, such as cast iron, may be used.

While propylene glycol will not freeze, the fluid breaks down at high temperatures, creating a sludge that can plug the collectors. Extremely high temperature may occur if fluid is allowed to stagnate within the collectors. This may happen if the circulation pump shuts down, either due to power loss or if the storage tank has achieved a target temperature. Much as with open loop freeze protection, the controller can restart the pump if the collector temperature exceeds an upper limit, forcing colder water into the collector. However, just as with open loop freeze prevention, power or mechanical failure can negate this protection. The risk of power failure can be mitigated by operating the pump off a photovoltaic panel. Even normal operation will, over time, cause propylene glycol to break down. Inhibitors can extend the fluid's life, but, in general, it should be purged and recharged every five years. Since closed loop anti-freeze systems are pressurized, this recharge should only be carried out by a qualified contractor.

Additionally, over time, gas dissolved in the working fluid may be forced out of solution and collect within the system. Gas bubbles will, at the least, degrade system performance and at worst burn out circulation pumps if a bubble pocket builds up at the intake. Air vents should be installed at any point where gas might accumulate (high points, 90^o bends) which may be periodically opened to allow trapped gas to escape[37].

6.1.3.3 Drainback Closed Loop

Drainback systems use water as a working fluid, but prevent freezing (or boiling) by allowing the working fluid to drain out the collector and piping (in unconditioned spaces) when the system is not in use. A schematic of a drainback system is given in figure 6.1.3.3.

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[37] Olson

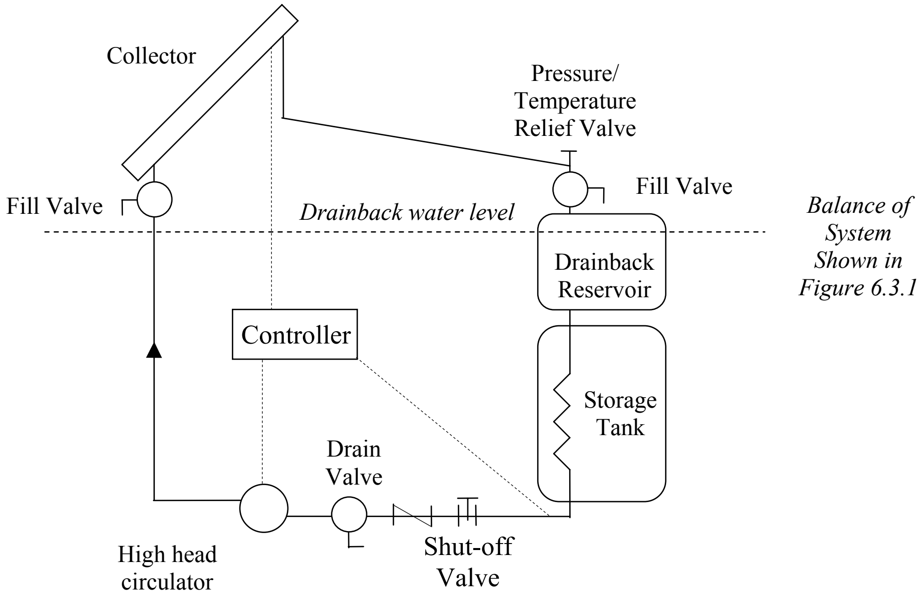

Figure 6.1.3.3 – Drainback Closed Loop

When the circulation pump is turned off, water drains back into piping and a reservoir within a conditioned space. Care must be taken to hang the pipe and mount collectors at a sufficient slope for all the water to drain back. A minimum slope of 15^{o} is required for piping in exterior or unconditioned space[38]. ¾" piping is also recommend (anti-freeze closed loops use ½" piping) to prevent damage from the freezing of residual water within pipes. If installed properly, drainback systems are easier to maintain than anti-freeze systems, and offer inherent protection from both freezing and boiling. However, if the system is improperly installed, it will provide no protection from freezing. In the case of a freeze, significant damage may be done to the system, destroying the entire investment. The drainback reservoir is smaller than the storage tank, with a capacity of 10 – 20 gallons, depending on the length of piping.

[38] Olson

6.1.4 Balance of System

6.1.4.1 Circulation Pumps

Centrifugal pumps are appropriate for solar thermal systems owing to low power consumption and maintenance and high reliability. These circulators are lubricated by the working fluid during regular operation. Pumps are sized based on the required flow rate and head. Static head is a measure of the height of the water, vertically, from the pump to highest point in the system. Dynamic head is a measure of the resistance to flow caused by friction and valves. A pump must overcome both static and dynamic head in a system. For solar thermal systems, static head (if present) will be significantly higher than the dynamic head. In general, the higher the head a given pump must overcome, the lower the flow rate. Typically, flow rates for thermal solar are about 1 – 1.25 gallons per minute per collector[39].

As previously mentioned, cast-iron may be used for closed loop pumps, while open loop pumps must use material which resists oxidation, such as bronze or stainless steel. Additionally, pumps in an open loop system must be high head pumps since a substantial static head exists (the difference between the pressure in the storage tank and pressure required to lift water to the collector). Closed loop anti-freeze pumps need only overcome dynamic losses due to pipe friction (since pressure is constant throughout the system) and may use low head pumps. Drainback systems require high head pumps since a static head must be overcome during the initial filling. However, once all piping has been filled, the static head vanishes, reducing load on the pump and flow rate increases. Table 6.1.4.1 shows a number of pumps suitable for operation in each type of loop.

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[39] Heliodyne

Table 6.1.4.1 – General Pump Selection[40]

Loop Type	Suitable Circulators
Open Loop	TACO 006B, Grundfos 15-18 SU, Hartell MD-31U
Anti-freeze Closed Loop	TACO 008F, Grundfos 15-42 F, Hartell MD-101U
Drainback Closed Loop	TACO 009BF

Low head circulators may be powered by standard grid electricity or a small photovoltaic panel (PV). PV powered systems are often considered more reliable, since the pump will continue to operate even if grid electricity is lost. This strategy is not viable for high head pumps, owing the higher power requirements. High head pumps used in open loop or drainback systems could draw power from a battery bank charged by PV, but such systems are not commercially available.

6.1.4.2 Expansion Tank

As water is heated, its density decreases. Within a closed loop, since the water's mass is held constant, this density drop results in a higher volume, and elevated pressure. Since higher pressures run the risk of damaging valves and fittings, it is important that the pressure within the system remain constant. This is accomplished by incorporating an expansion tank into the loop. An expansion tank contains a gas filled cushion (e.g. air) which may be compressed as the volume of water in the system increases. While loop pressure will rise slightly, it should not reach damaging levels, as might be the case without an expansion tank. A representative schematic of an expansion tank is shown in figure 6.1.4.2.

[40] Olson

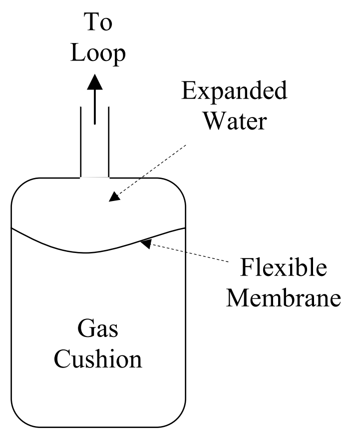

Figure 6.1.4.2 – Expansion Tank

Tanks should be sized so that for a maximum operating temperature, system pressure will not rise to dangerous levels. ASHRAE gives the following relation to calculate the required size of an expansion tank.

$$V_t = V_s \frac{([\nu_2/\nu_1]-1)-3\alpha\Delta t}{(P_a/P_1 - P_a/P_2)}$$ Equation 6.1.4.2[41]

where:

V_t is the tank volume
V_s is the volume of fluid in the closed loop
ν_1 is the specific volume of the fluid at the minimum temperature
ν_2 is the specific volume of the fluid at the maximum temperature
P_a is atmospheric pressure, psia
P_1 is the minimum system pressure, psia
P_2 is the maximum system pressure, psia
α is the coefficient of thermal expansion (9.5e-6 in/in ^{o}F for copper tube)
Δt is the difference between maximum and minimum temperatures

The minimum temperature for closed loop systems is generally the ambient temperature when the system is filled (e.g. 50^{o}F). Maximum temperature is usually assumed to be less than 200^{o}F. Minimum system pressure is set by the pressure to lift the fluid from the tank to the highest point in the system (e.g. 10 psig). Maximum allowable pressure

[41] ASHRAE, HVAC 12.4

should be no higher than 50 psig. Given these constraints, equation 6.1.4.2 suggests the acceptance volume of the expansion tank should be 9% of the total loop volume.

For the systems considered, expansion tanks are required only for anti-freeze closed loop systems. The reservoir in a drainback closed loop incorporates the pressure moderating function of an expansion tank, eliminating the need for a separate piece of equipment.

Expansion tanks are commercially available in any number of sizes. Thermo Technologies manufactures a series of expansion tanks suitable for use in solar thermal systems. Model TT30, with a capacity of 4.4 gallons, will be more than adequate for the collector loop[42].

6.1.4.3 Pressure Relief Valve

A relief valve should be incorporated into the system to protect against pressures higher than the expansion tank's ability to compensate. Pressure relief valves are usually spring operated mechanical devices which open when pressures are sufficient to overcome the force of the spring. In the event that a pressure relief valve opens, it is usually necessary to replace the valve since they often do not reseat properly and result in small leakages. 50 psig relief valves should be used with a closed-loop collector system.

6.1.4.4 Piping

Insulated copper pipe is used in both open and closed loop systems due to its corrosion resistance and relative ease of installation[43]. ½" pipe is suitable for open loop and anti-freeze closed loop systems. However, ¾" pipe should be used in drainback systems to prevent damage from the freezing of any residual water. PEX (cross-linked polyethylene) is not appropriate for the collector loop since it is possible for collector

[42] Thermo Technologies
[43] ASHRAE HVAC 42.1

loop temperatures to exceed PEX thermal limits (200^{o}F) and the piping also degrades from prolonged exposure to UV.

6.1.5 Control

An integral part of the daily operation is the controller which tells the circulation pump when to operate. In general, operation criteria are determined by the temperature delta between the bottom of the tank and the collector outlet. For best performance, the pump should be switched on if the temperature delta is greater than 20^{o}F, and shut down when the delta drops to 5^{o}F [44].

Heliotrope manufactures controllers suitable for open loop, closed loop anti-freeze, or drainback systems (DTT-84 or DTT-94). Temperature measurements are obtained using thermistor type sensors, one at the collector outlet and one at the bottom of the storage tank[45].

6.2 Heating Applications

6.2.1 Domestic Hot Water

Two approaches are accepted for providing domestic hot water. Since hot water delivery temperatures are between 120^{o}F and 140^{o}F, and the solar storage tank will not usually sustain this temperature throughout the day, an auxiliary heater is an integral part of the delivery system. The traditional method is shown in figure 6.2.1.1 and involves a heat exchanger in conjunction with a standard hot water heater.

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[44] Olson
[45] Olson

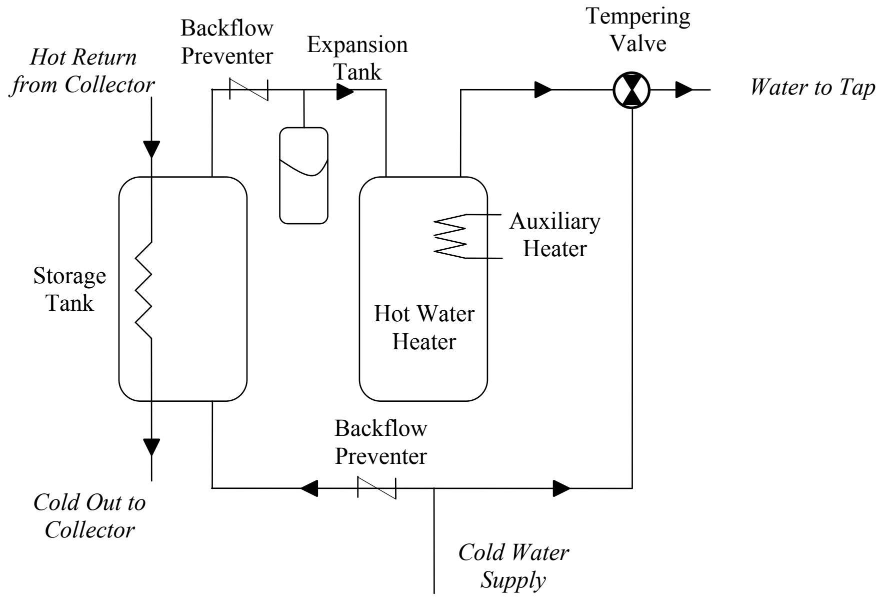


Figure 6.2.1.1 – Traditional Solar Hot Water Heating

These systems are capable of providing more than 6 GPM of hot water for domestic applications. The tempering valve shown in figure 6.2.1.1 limits the maximum hot water temperature to between 110°F and 140°F by mixing in cold supply water. Without a tempering valve, it is possible for much hotter water to be piped to tap, which could scald an occupant. Auxiliary heat may be provided either by a propane burner or electric immersion heater depending on the hot water heater design. The choice of propane versus electricity is discussed further in section 6.2.3. As water is drawn from the hot water heater tank for domestic applications, it is replaced by hot water from the top of the storage tank. This water is then replaced at the bottom of the storage tank by cold water from the external supply. Even though domestic hot water is an open loop, the section of piping between the upper backflow preventer and tempering valve does constitute a closed 'loop' and should incorporate an expansion tank. A tank sized similarly to that used in the collector loop should be adequate.

The second approach is "instant", or tankless hot water heat. Water is drawn directly from the storage tank, passed through an instant hot water heater (auxiliary), and delivered to the hot water application. Unlike a conventional design, instant hot water heaters do not maintain a reserve of hot water, but rather produce it on demand. Instant hot water systems have a lower maximum throughput, commonly under 3 GPM, but this volume of hot water is usually sufficient for a small family residence. Instant hot water units are significantly smaller than a traditional hot water installation, and are appropriate in lower use situations where space is at a premium. Figure 6.2.1.2 shows how an instant hot water heater could be incorporated into a solar thermal system.

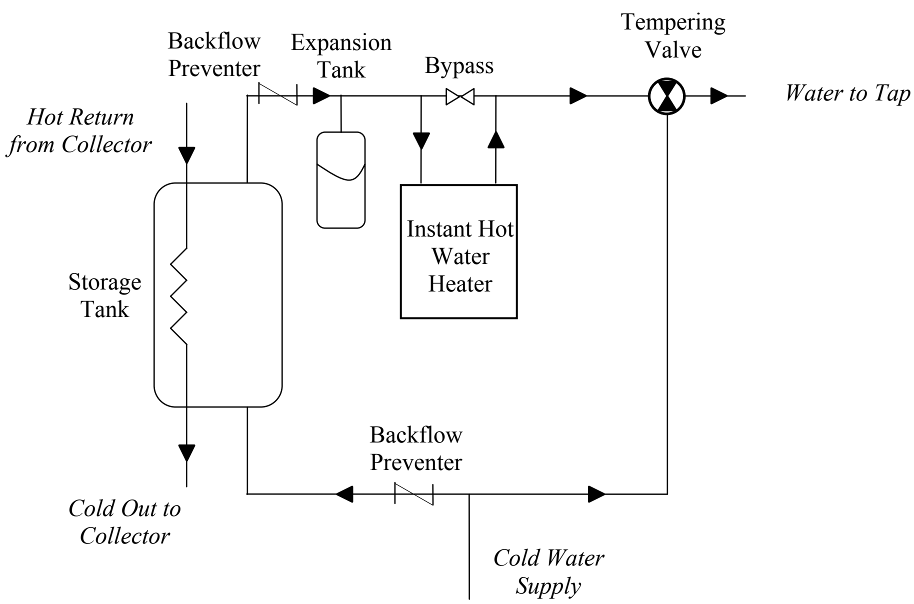


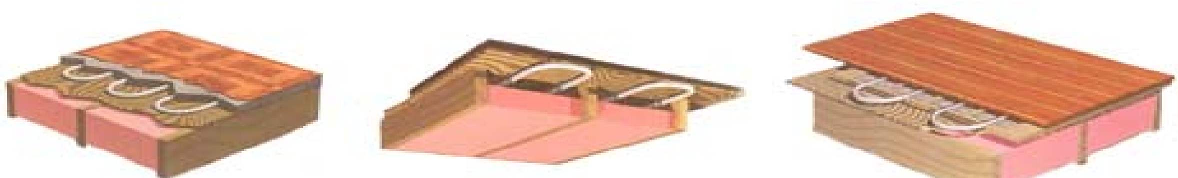
Figure 6.2.1.2 – Instant Solar Hot Water Heating

6.2.2 Space Heating

Given the temperature limits of a solar thermal system, the best mechanism for heat distribution is a low temperature hydronic system. Within this category, one must choose between more traditional baseboard style heaters and radiant floor systems.

Hydronic baseboard heaters generally require water between 130°F and 200°F to efficiently heat a house. The higher the operating temperature, the greater the convective transfer coefficients between the radiator and air and the less baseboard required. At lower operating temperatures an uneconomic amount of baseboard is required to offset lower transfer coefficients with greater surface area.

Radiant floor systems use a series of hot water loops to heat the floor, which in turn serves to heat the room. Because the relative surface area of a radiant floor system is so much higher than a baseboard system, they may operate at significantly lower temperatures (e.g. 95^{o}F) and still deliver adequate heat to the building. Radiant floor systems deliver a more diffuse heat, rather than the point sources of a baseboard system, and are considered superior from a comfort standpoint[46]. However, they are more difficult to install than baseboard systems, which translates to a cost nearly twice as a great as a comparable baseboard installation. Radiant floor systems may be installed in three different ways. The first involves laying out the piping circuit, then pouring a thin layer of cement over the top (figure 6.2.2.1a). This method is generally favored for new construction, but not for renovation. However, radiant floor systems may be added to existing structures by either installing the piping assembly beneath the subfloor (figure 6.2.2.1b) or installing a suspended floor incorporating a radiant system (figure 6.2.2.1c).



a. Thin Cement　　a. Beneath Subfloor　　c. Suspended Floor

Figure 6.2.2.1 – Radiant Floor Installation Options

Note that in all installations, PEX is used rather than copper pipe due to its superior flexibility.

Figure 6.2.2.2 shows how a hydronic heating system could be incorporated into a solar thermal system.

[46] Elder

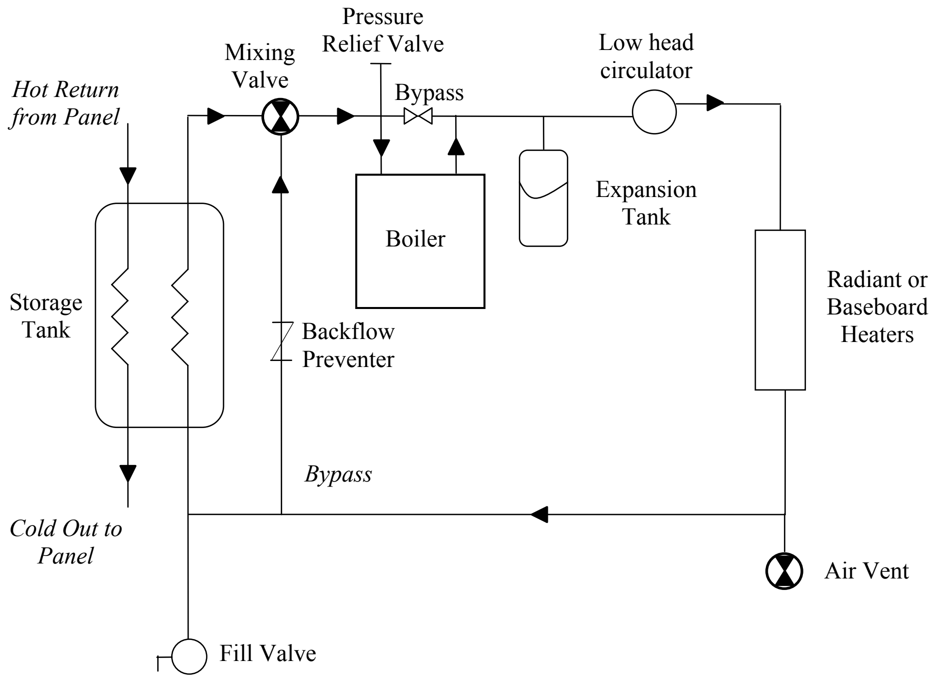


Figure 6.2.2.2 – Hydronic Space Heating

This is a closed loop hydronic heating system. Much like an anti-freeze closed loop collector system, it requires a number of ancillary components: expansion tank, air vent, pressure relief valve, fill valve. A bypass and mixing valve serve to protect the system from water above design temperatures by mixing colder return water with water heated by the storage tank to limit the boiler inlet temperature.

PEX (cross-linked polyethylene) piping is an easier to install alternative to the copper pipe used for the collector loop and is appropriate for connecting a series of baseboard heaters or as the piping loops in a radiant floor installation. For closed-loop space heating, the specific PEX should be impermeable to oxygen in order to prevent corrosion of system components. Copper piping should be utilized between the mixing valve and storage tank heat exchanger due to the elevated temperatures that portion of the loop may achieve. ¾” PEX and copper pipe will be adequate for the standard residential heating flow rate of 1 GPM.

The circulator is a low head pump – TACO 006B or equivalent. An expansion tank equivalent to that used in the collector loop should be adequate. Space heating systems commonly operate at 20-22 psig, with a pressure relief valve at 30 psig mounted at the boiler outlet.

6.2.3 Auxiliary Heating

An auxiliary heater is an integral part to any design, since it is uneconomic to design a solar system to meet 100% of the yearly energy requirement. For this project, two types of auxiliary heaters were considered: propane and electric. The decision between the two is governed strictly by economics. Electric units are relatively cheap and convert fuel (electricity) to heat at 100% efficiency, but electric rates are very high at HAVO – as much as 25¢ per kWh. Propane furnaces cost more to purchase, and operate at a lower efficiency (75-90%), but propane fuel is less expensive than electricity. As a result, for situations where the auxiliary heater sees extensive use, a propane furnace will be more appropriate, since the higher investment is mitigated by the lower fuel price over the life of the unit. The opposite is true to low use systems, where electric heating is more competitive.

The auxiliary heating for domestic hot water and space heating may be handled by a single, more complex unit or by two, simpler units. Combined systems are less common and appear to only be available for propane fired systems. However, their compact profile makes them preferable in installations where space is at a premium.

6.3 System Layout

An integrated system to provide hot water and space heating using solar energy is presented in figure 6.3.1.

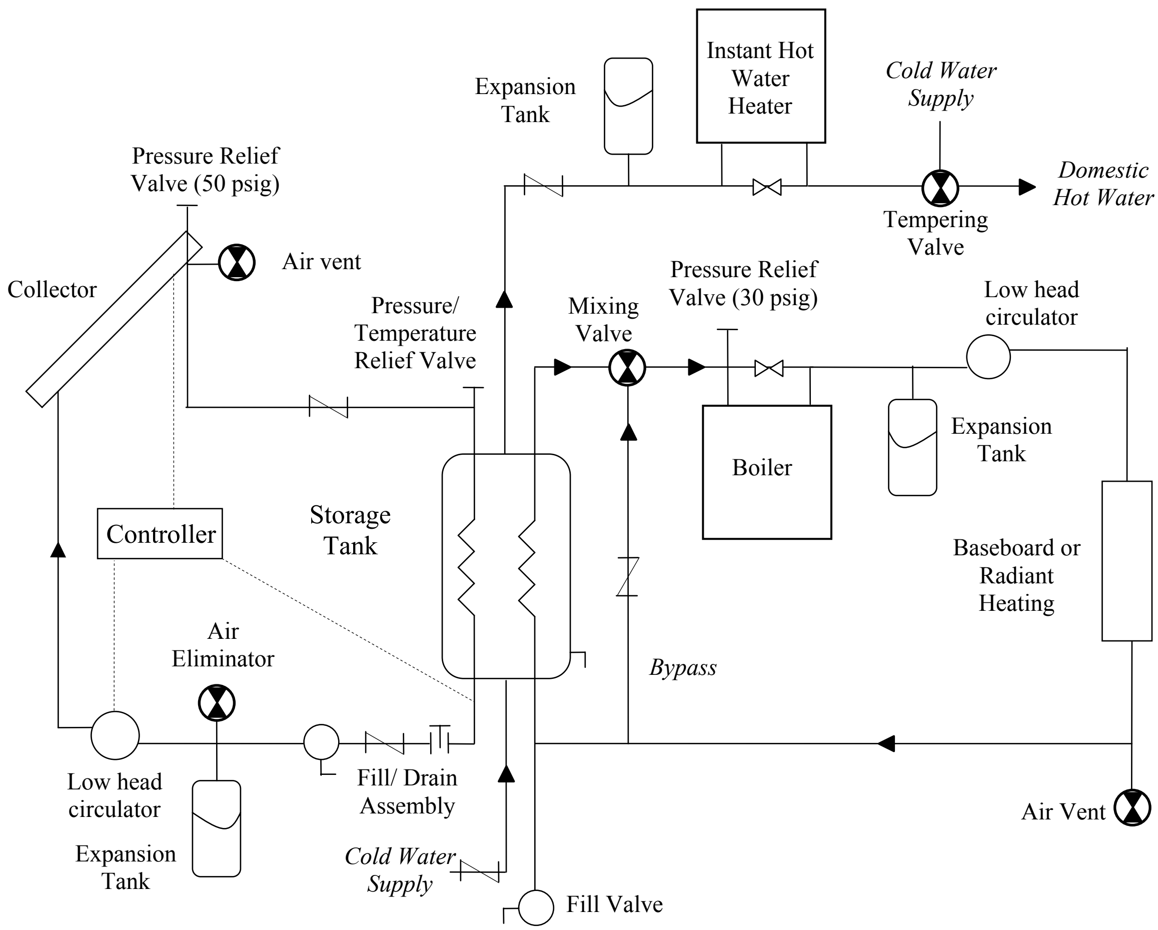


Figure 6.3.1 – General System Layout

7. ENGINEERING ANALYSIS

7.1 Methods for Evaluating Performance

Two methods are commonly used to predict the operating performance for solar thermal systems. *f*-charts are a graphical tool developed to provide quick results with minimal knowledge of a system. Physical models are more complex – reflecting more of the fundamental mechanics of the systems in question. While *f*-chart analysis is quite powerful, it has been developed for forced air heating, and is, therefore, not directly applicable to the hydronic space heating options discussed in Section 6. Here, *f*-chart analysis will be used as a comparison to the results predicted by the physical model.

7.1.1 f-Chart

f-charts are useful for quickly evaluating residential solar applications where detailed system modeling may not be cost-effective. They are correlations developed from hundreds of test simulations that allow evaluation of system effectiveness with relatively few inputs. The two dimensionless variables on an *f*-chart plot are representative of the energy incident on the collector surface and the losses associated with the entire system. The intersection of these two values gives the monthly fraction of the load met by solar energy. The loss variable (X) is given by equation 7.1.1.1.

$$X\,(\text{System Loss}) = \frac{A_c F_R' U_L \left(T_{\text{Ref}} - \overline{T}_a\right)\Delta t}{L}$$ Equation 7.1.1.1[47]

where:

A_c is the collector surface area (m^2)
F_R' is the collector-heat exchanger efficiency factor
U_L is the collector loss coefficient (W/m^{2}C)
$\overline{T}_a$ is the monthly average ambient temperature (C)
T_{ref} is a fixed reference temperature (100^{o}C)

[47] Beckman, pg. 690

L is the monthly load for space heating and domestic hot water (J)

A correction to X to account for different sized storage tanks is given by Equation 7.1.1.2.

$$\frac{X_c}{X} = \left(\frac{Actual\ Storage\ Capacity}{Standard\ Storage\ Capacity}\right)^{-0.25}$$ Equation 7.1.1.2[48]

where:

X_c is the corrected system loss variable
X is the system loss variable as given by equation 7.1.1.1
Standard Storage Capacity is 75 L of water per square meter of collector

The incident solar variable (Y) is given by equation 7.1.1.3.

$$Y\left(\text{Incident Solar}\right) = \frac{A_c F_R^{'}(\tau\alpha)\overline{H}_T N}{L}$$ Equation 7.1.1.3[49]

where:

$\tau\alpha$ is the monthly average transmittance-absorbed product of the collector surface
$\overline{H_T}$ is the daily average incident radiation on the collector surface
N is the number of days in the month

$F_R^{'}$ and $\tau\alpha$ are properties of the particular collector chosen for the application. These values have been determined under test conditions and are tabulated by the Solar Rating and Certification Corporation (SRCC)[50].

Once X and Y, have been calculated, the monthly fraction of the load met by solar energy may be looked up on an *f*-chart. Duffie and Beckman provide a curve fitted correlation for f given a range of X and Y, which is more amenable to numerical simulation. This correlation for the monthly solar fraction is presented in equation 7.1.1.4.

$$f = 1.029Y - 0.065X - 0.245Y^2 + 0.0018X^2 + 0.0215Y^3$$ Equation 7.1.1.4[51]

[48] Beckman, pg. 694
[49] Beckman, pg. 690
[50] Solar Rating and Certification Corporation
[51] Beckman, pg. 692

f-charts agree to a good approximation with yearly observations, but effectiveness is more variable on a month-to-month comparison. As previously mentioned, *f*-charts are based on systems using forced air for heating and the results are not appropriate for predicting the performance of hydronic space heating. However, knowing the expected relative performance of hydronic options relative to space heating is useful for calibrating the physical model discussed in section 7.1.2.

7.1.2 Physical Model

In order to optimize a hydronic space heating system it is necessary to create a physical model which uses explicit energy balances to assess system performance. Such a model is more transparent than *f*-chart analysis, but significantly more complex. Such a model was implemented using MS EXCEL (for calculation) and MS ACCESS (for data storage). The model calculates storage tank temperature via an energy balance which accounts for energy gain by the collector, demand for space heating and domestic hot water, and tank thermal losses. This relation is given by equation 7.1.2.

$$T^{+} = T + \frac{\left(Q_{Collector} - Q_{Tank\ Loss} - Q_{Space\ Heat} - Q_{DHW}\right)}{mc_{p}} \qquad \text{Equation 7.1.2}$$

where:

T^{+} is the temperature in the storage tank at the end of the hour
T is the temperature in the storage tank at the beginning of the hour
$Q_{\mathrm{Collector}}$ is the energy gained by solar radiation incident on the collector
$Q_{\mathrm{Tank\ Loss}}$ is thermal energy lost from the storage tank
$Q_{\mathrm{Space\ Heat}}$ is heat used to provide space heating
Q_{DHW} is hot water used for domestic application
m is the mass of water in the storage tank
c_p is the specific heat of water

The storage tank is assumed to be at a uniform temperature (perfectly mixed). Stratification effects found in actual tanks are beyond the model's capability and are neglected. Stratification generally increases overall system performance[52]. Tank temperature is limited to 200°F in accordance with standard practices. In the case that calculations predict a higher temperature at the end of an hour, temperature is reset to 200°F, simulating the opening of a thermal relief valve.

7.1.2.1 Collector Gain

Collector efficiency is modeled using empirical curve fits developed by the Solar Rating and Certification Corporation (SRCC)[53]. SRCC publishes efficiency relations for certified collectors of the form:

$$\eta = AK_{\alpha\tau} + B\frac{(T_i - T_a)}{G} + C\frac{(T_i - T_a)^2}{G} \qquad \text{Equation 7.1.2.1}^{54}$$

where:

η is the collector efficiency – the fraction of incident energy absorbed
A, B, and C are empirical coefficients determined by collector lab testing
G is the incident solar flux (W/m^2)
T_i and T_a are, respectively, collector inlet temperature and ambient temperature
$K_{\alpha\tau}$ is an incidence angle modifier which accounts for the decrease in efficiency as the sun moves away from the normal to the collector

The incidence angle modifier is also empirically derived and is given by equation 7.1.2.2.

$$K_{\alpha\tau} = 1 - D\left[\frac{1}{\cos\theta} - 1\right] \qquad \text{Equation 7.1.2.2}^{55}$$

where:

D is an empirical coefficient determined by collector lab testing
θ is the incidence angle between collector and sun (as defined previously)

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[52] Beckman, pg. 678
[53] Solar Rating and Certification Corporation
[54] Huggins
[55] Huggins

For the collectors considered in this analysis, A is a positive number, while B and C are negative. As a result, as the storage tank is heated and delta between collector inlet and ambient increases, the collector efficiency decreases. Additionally, for values of θ greater than zero, $K_{\alpha T}$ will be negative, resulting in decreased efficiency.

The presence of a heat exchanger between the tank and collector loop further reduces the effective energy gain. A heat exchanger factor of 0.97 is used to account for this penalty (e.g. 97% of absorbed heat is transferred to the storage tank).

7.1.2.2 Domestic Hot Water

Domestic hot water is drawn off the storage tank as needed and replaced by cold water from the supply. The energy drawn off the tank is given by equation 7.1.2.2.1.

$$Q_{DHW} = mc_p\left(min\left\lfloor T_{Delivery}, T_{Tank}\right\rfloor - T_{Supply}\right) \qquad \text{Equation 7.1.2.2.1}$$

where:

Q_{DHW} is the energy drawn off the tank for domestic hot water
m is the mass of water drawn off and replaced
c_p is the specific heat of water
T_{Tank} is the temperature of water in the tank
$T_{Delivery}$ is the delivery temperature for hot water (120^{o}F)
T_{Supply} is the temperature of the cold water supply

The bracketed term models the temperature valve on the hot water circuit. If tank temperature is above the delivery temperature, a smaller volume of tank water will be mixed directly with supply water at the tempering valve.

If the tank temperature is below delivery, the auxiliary unit will further heat the water. Auxiliary energy input is given by equation 7.1.2.2.2.

$$Q_{Aux,DHW} = mc_p\left(T_{Delivery} - T_{Tank}\right)^{+} \qquad \text{Equation 7.1.2.2.2}$$

where:

$Q_{Aux,DHW}$ is the auxiliary energy input for domestic hot water
m is the mass of water drawn off and replaced
c_p is the specific heat of water
T_{Tank} is the temperature of water in the tank
$T_{Delivery}$ is the delivery temperature for hot water (120^{o}F)

The + notation in equation 7.1.2.2.2 indicates that only positive values are considered.

7.1.2.3 Space Heating

Water for space heat is assumed to circulate through the baseboard radiators at 1 gallon per minute (GPM). For baseboard heating, water is assumed to leave the boiler at 140^{o}F and return at 110^{o}F, for an average operating temperature of 125^{o}F. In order to meet design loads, it is possible to incorporate a controller into the boiler with multiple set points. For example, above 50^{o}F outdoor temperature (norm for HAVO) the boiler temperature would be specified at 140^{o}F. However, for temperatures below 50^{o}F, a higher set point could be used to increase the total heat delivered by the system. A radiant floor design is assumed to also have a 30^{o}F delta, but only heat the supply water to 110^{o}F (average operating temperature of 95^{o}F). The amount of heat delivered to the residence each hour is regulated by varying the time the circulator operates. Return water is raised by to tank temperature by the use of a heat exchanger, with maximum temperatures controlled by the mixing valve. If water temperature is below the design criteria (140^{o}F or 110^{o}F), the auxiliary boiler further increases the temperature. Auxiliary energy input is given by equation 7.1.2.3.1.

$$Q_{Aux,Space\ Heat} = mc_p\left(T_{Delivery} - T_{HX}\right) \qquad \text{Equation 7.1.2.3.1}$$

where:

$Q_{Aux,Space\ Heat}$ is the auxiliary energy input for space heating
m is the mass of water passing through the boiler in an hour
c_p is the specific heat of water
$T_{Delivery}$ is the delivery temperature for space heating

T_{HX} is the temperature of water leaving the storage tank heat exchanger

The mass of water passing through the boiler is given by equation 7.1.2.3.2.

$$m = \dot{m}\,t \qquad \text{Equation 7.1.2.3.2}$$

where:

m is the mass of water passing through the boiler in an hour

$\dot{m}$ is the mass flow rate of water in the system
t is the time (in minutes) the pump operates each hour

The pump operating time is given by equation 7.1.2.3.3.

$$t = \frac{Q_{\text{Space Heat}}}{\dot{m}\,c_p} \qquad \text{Equation 7.1.2.3.3}$$

where:

t is the time (in minutes) the pump operates each hour
$Q_{\text{Space Heat}}$ is the space heating load for the hour

$\dot{m}$ is the mass flow rate of water in the system
c_p is the specific heat of water

The temperature of the water leaving the heat exchanger will be lower than the tank temperature due to thermodynamic constraints. Simulation indicates that for the operating range of the storage tank, water could leave the heat exchanger within 2^{o}F of the tank temperature. Details of this simulation may be found in Appendix D.

The heating load as, as calculated in section 5, is increased by 2% to account for pipe losses in the distribution system.

7.1.2.4 Tank Losses

Storage tank losses are calculated as with conductive losses from the structure. Tank losses are given by equation 7.1.2.4.1.

$$Q_{Tank\ Loss} = UA(T_{Tank} - T_{Ambient}) \qquad \text{Equation 7.1.2.4.1}$$

where:

$Q_{Tank\ Loss}$ is the energy lost by conduction and convection from the tank
UA is the tank loss coefficient
T_{Tank} is the temperature of water in the tank
$T_{Ambient}$ is the ambient temperature

Tank losses are, in practice, 2-4 times higher than would be predicted by calculating losses from insulation alone due to fittings and inlet/outlet pipe losses[56].

7.2 System Model

The system model consists of a number of modules, each performing a different function. Modules have been implemented in Microsoft Excel since it is a commonly understood interface. The bulk of the simulation data is stored in a Microsoft Access database. Data is transferred from the databases to the modules using Structured Query Language (SQL) implemented in Microsoft Visual Basic for Applications (VBA).

To limit the complexity of the model, individual modules do not to link directly to each other. Directly linking modules may lead to a veritable spiderweb of complicated, difficult to maintain links. Rather, individual modules pull data from a central database, perform a calculation, and return the result. This approach also allows the same module to perform several different calculations. For example, rather than having a module

[56] Beckman, pg. 416

which calculates collector incidence for north, south, east, and west facing panels, a single module may be used which calculates incidence with orientation as an input.

All user units are held in a single file to simplify the user interface. A future user of this model need not understand the mechanics behind the calculations, so long as the inputs are fully specified. The simulation database is, by intention, not meant as a medium to view or analyze data. All normal user interaction should be constrained to the input module and the analysis modules.

Figure 7.2.1 shows the flow for the system model. For an example of how modules gather and return information to the database consider the "Collector Incidence" and "Physical Model" modules. The "Collector Incidence" module calculates radiation incident on the rooftop collectors from information on collector geometry supplied by the user (e.g. number of panels). The "Physical Model" module then uses the incident radiation numbers as part of a calculation of overall system performance. A detailed description of module input, output, and calculations may be found in Appendix E. Please contact Dr. Philip Malte[57] or Brian Polagye[58] for an electronic copy of the model spreadsheets and database.

[57] Dr. Philip Malte, University of Washington, Dept. of Mechanical Engineering (malte@u.washington.edu)
[58] Brian Polagye, University of Washington, Dept. of Mechanical Engineering (bpolagye@u.washington.edu)

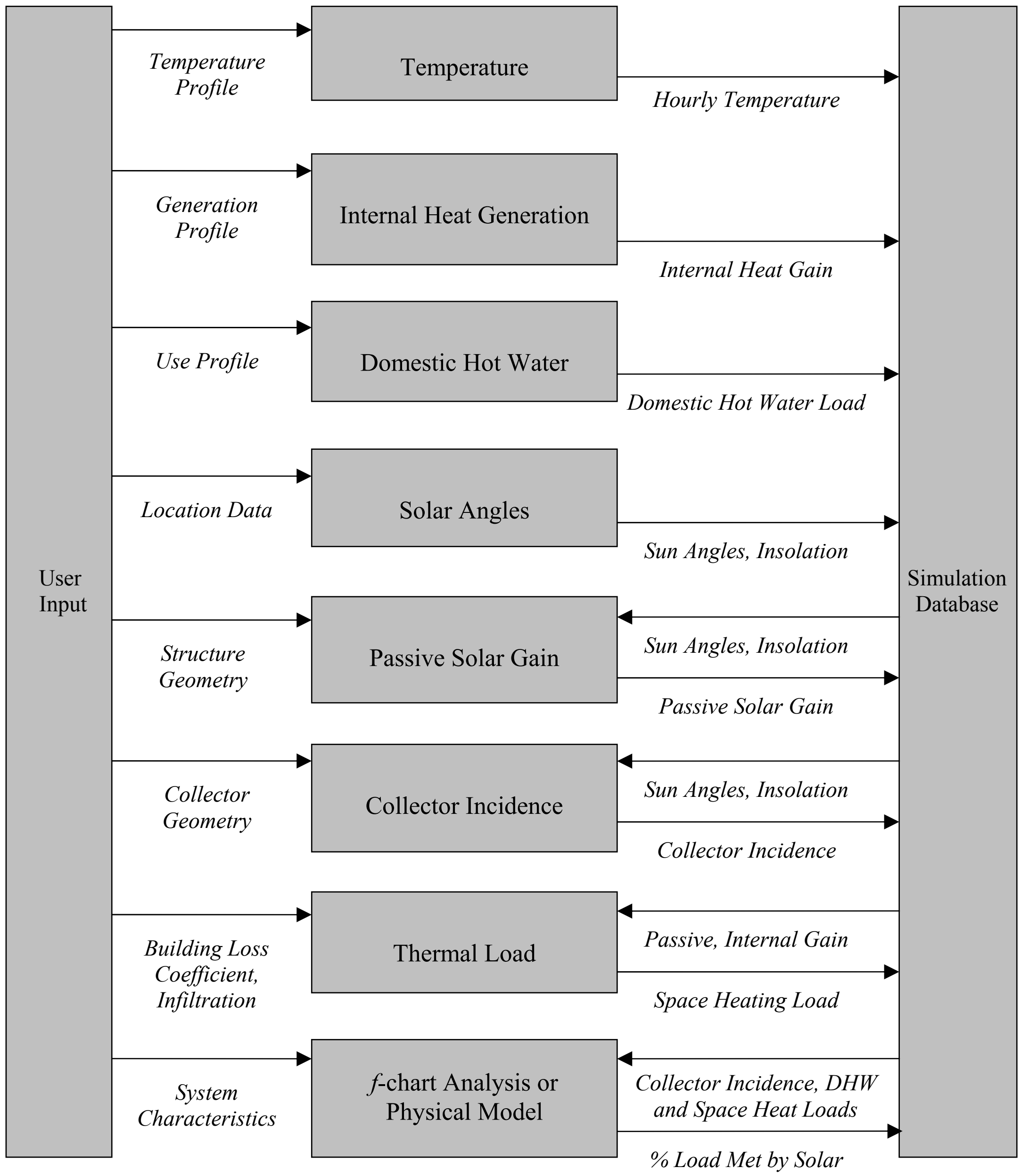

Figure 7.2.1 – System Model Flow

7.3 Key Parameters

Four key parameters were varied to gauge their impact on system performance: degree of insulation, thermostat set temperature, space heating type (radiant vs. baseboard), and number of panels.

7.3.1 Insulation

The 'base' case assumed the house as it stands today, with only a thin layer of insulation beneath the floor. The 'insulated' case assumed walls, ceilings, and floors had been insulated and weatherstripping applied to doors and windows. The 'windows' case assumes all the insulation of the 'insulated' case, along with the replacement of existing windows with more efficient ones.

7.3.2 Thermostat Set Temperature

Two set temperatures were considered for the interior thermostat, 65^{o}F and 70^{o}F. Setting the thermostat lower reduces heat loss to the environment relative to a high thermostat set. Most energy conscious homeowners set their thermostats to 65^{o}F without noticeable discomfort. However, for cultural and geographic reasons, Hawaiians may prefer warmer environments and, thus, set the thermostat to 70^{o}F.

7.3.3 Space Heating

The baseboard system in this analysis is assumed to operate at an average temperature of 125^{o}F, the radiant radiant systems at 95^{o}F. The cooler the operating temperature, the better the chance will be that the solar collectors will meet the daily load without assistance from the auxiliary heater.

7.3.4 Number of Panels

The number of panels was varied between one (40 ft^2) and four (160 ft^2). Aesthetic considerations and weight on the roof limit the number of panels to four. A situation without panels was also considered to serve as a comparison of the solar thermal systems to fossil fuel heating. This case is considered further during economic analysis as the solar engineering result of 0% solar fraction may be trivially obtained.

7.4 System Performance

System performance for the cases considered is presented in Tables 7.4.1 and 7.4.2. Table 7.4.1 is for a 70°F thermostat set temperature, Table 7.4.2 for 65°F. Performance is measured by the annual solar fraction – the percentage of the total energy requirement met by the solar thermal system. Results are grouped by insulation case.

Table 7.4.1 – Modeled System Performance (70°F Set Point)

Insulation	Panels	*f*-Chart Forced Air	Physical Model Baseboard	Physical Model Radiant
Base	1	14%	17%	21%
	2	25%	25%	31%
	3	35%	31%	39%
	4	44%	36%	45%
Insulated	1	42%	29%	41%
	2	68%	52%	69%
	3	83%	69%	84%
	4	91%	80%	92%
Windows	1	59%	39%	54%
	2	86%	68%	85%
	3	95%	84%	96%
	4	98%	92%	99%

Table 7.4.2 – Modeled System Performance (65^{o}F Set Point)

Insulation	Panels	*f*-Chart Forced Air	Physical Model Baseboard	Physical Model Radiant
Base	1	23%	20%	28%
	2	40%	33%	44%
	3	54%	44%	56%
	4	65%	52%	65%
Insulated	1	61%	41%	55%
	2	86%	69%	85%
	3	95%	84%	96%
	4	98%	92%	99%
Windows	1	79%	57%	69%
	2	97%	84%	95%
	3	100%	94%	100%
	4	100%	99%	100%

The validity of the modeled results will be discussed in Section 7.4.1. The implication of these results on design selection will be discussed in Section 7.4.2.

7.4.1 Physical Model Validation

Before the results obtained from the physical model are used to perform economic analysis, it is important to first consider the validity of the model. Figure 7.4.1.1, adapted from Kreith and Kreider, shows the qualitative performance of forced air, baseboard, and radiant floor heating.

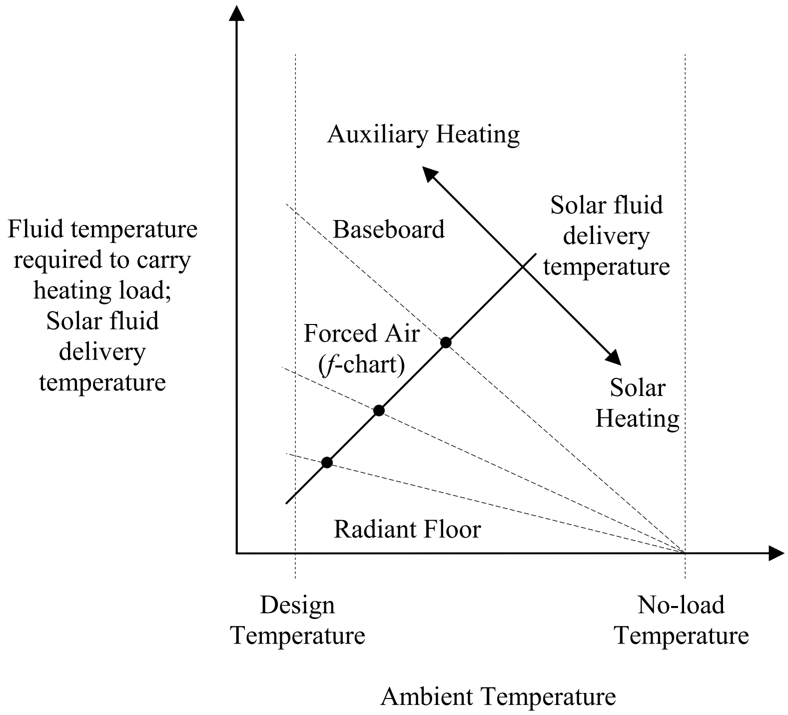

Figure 7.4.1.1 – Heating Load Diagram[59]

As the ambient temperature drops, higher temperature working fluid is required to meet the heating demand. The point at which the heating demand curve (dashed lines) intersects the solar delivery curve (solid line) is the cross-over point at which auxiliary heating becomes necessary. From this figure, it is apparent that for a given ambient temperature, radiant floor heating will require the least auxiliary heating, followed by forced air, followed by baseboard heaters. For example, as previously mentioned, radiant floor systems may operate at a lower temperature than baseboards and meet the same heating load. As a result, there is a higher probability for radiant floor operating temperatures to be maintained by the solar resource alone (and require less auxiliary heat).

[59] Kreider, pg. 434

Figure 7.4.1.1 may be adapted to a format more amenable for comparison to modeling results. Figure 7.4.1.2 shows the annual solar fraction (fraction of total energy requirement met by solar) versus panel area.

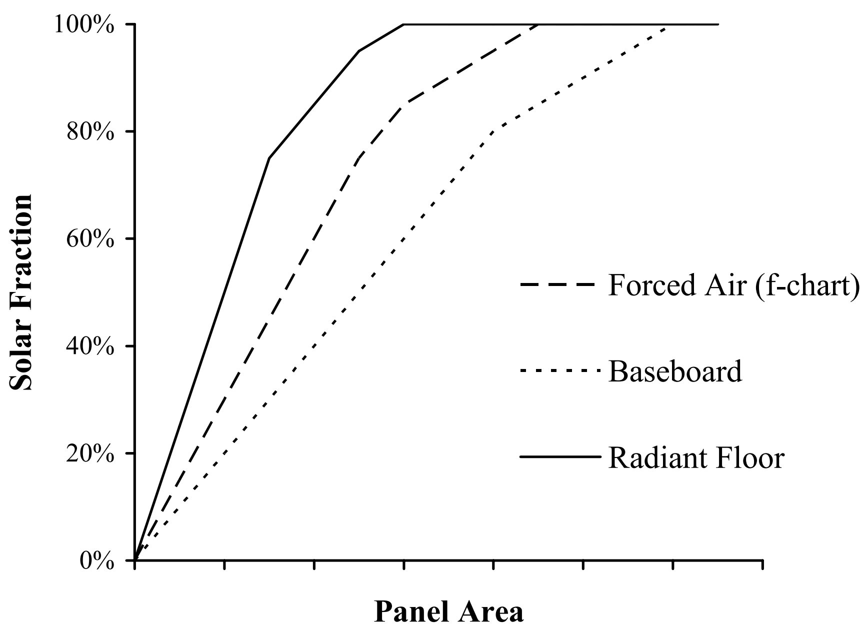

Figure 7.4.1.2 – Theoretical System Performance

Solar fraction met grows with panel area, approaching 100%. The curvature effect occurs when a system is capable of meeting more than 100% of the summer load, but still only a fraction of the winter (averaging to a value below 100%).

Figure 7.4.1.3 shows modeling results for a single case (70°F indoor temperature, insulated case) in the same format.

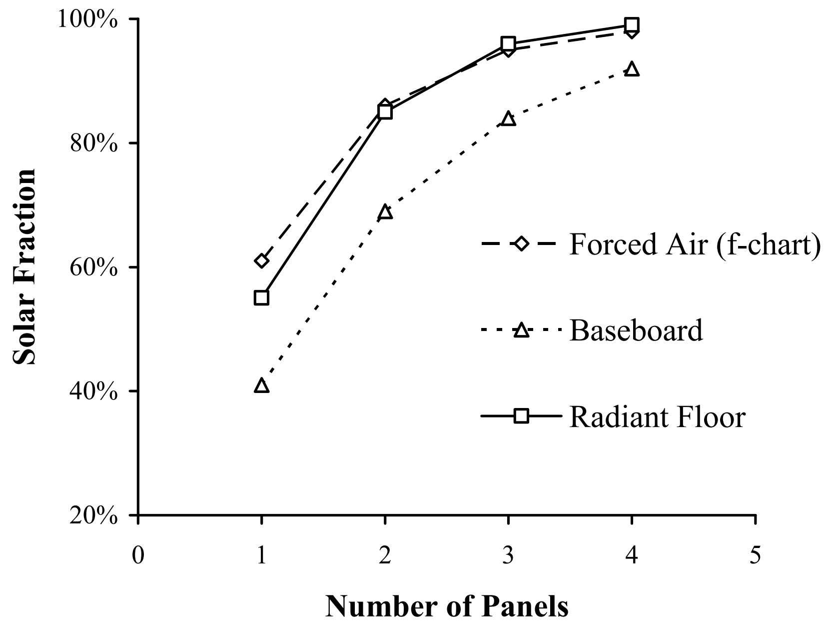

Figure 7.4.1.3 – Modeled System Performance (70^{o}F Set Point, Insulated Case)

Comparing the modeled system performance to the qualitative, expected system performance, one may readily verify that the performance of the baseboard system relative to forced air or radiant floor is consistent. However, while one would have expected the radiant floor system to outperform the forced air, modeling predicts comparable performance. This inconsistency may be reconciled by considering a key difference between *f*-chart analysis and the physical model constructed for this study. As discussed in section 7.1.2, uniform temperature is assumed within the storage tank for the physical model, while *f*-chart analysis assumes some temperature stratification.. Using a TRNSYS model, Duffie and Beckman describe a 6% improvement relative to a uniform tank model with a 3-node stratification model[60]. Therefore, it is likely the physical model systematically underestimates the performance benefit resulting from stratification. Making a qualitative correction for this understatement would result in the expected superior performance for the radiant physical model versus the *f*-chart and helps to justify

[60] Beckman, pg. 678

model results. Since we lack sufficient information to apply a quantitative correction to the physical model output, the results will be used for economic analysis “as-is”.

7.4.2 Discussion of Results

The tabulated modeled performance from tables 7.4.1 and 7.4.2 are shown graphically in figures 7.4.2.1 and 7.4.2.2.

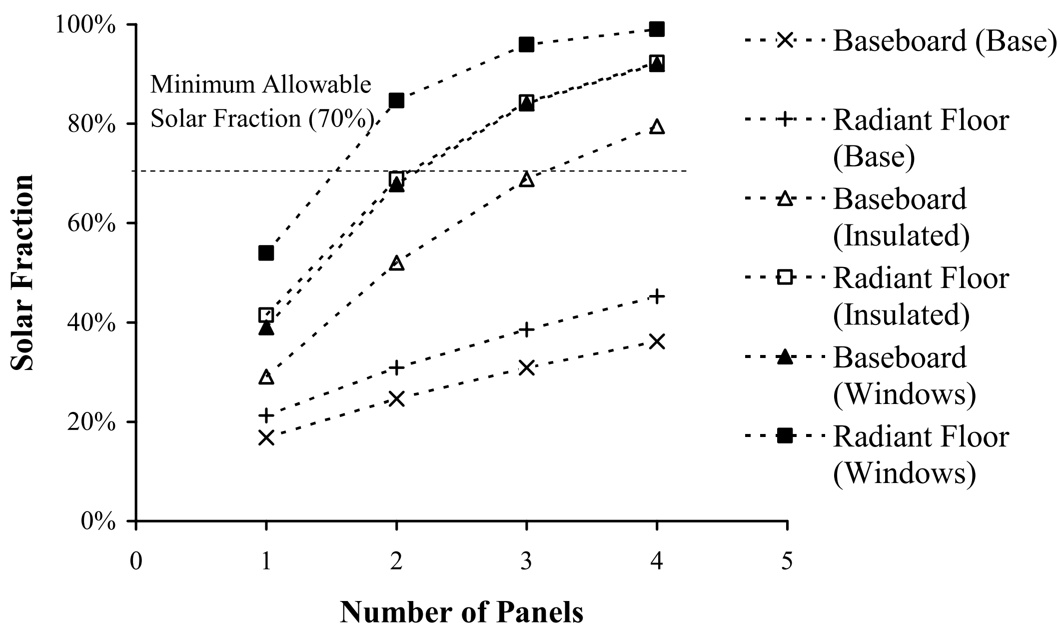

Figure 7.4.2.1 – Modeled System Performance (70oF Indoor Temperature)

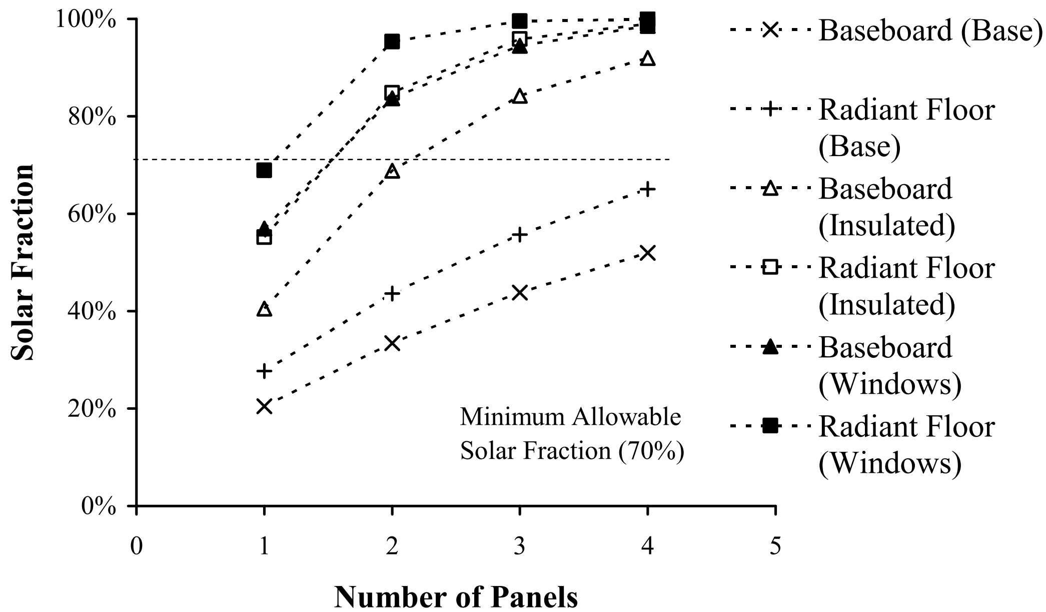

Figure 7.4.2.2 – Modeled System Performance (65°F Indoor Temperature)

Results for both cases are in-line with expectations. Additional insulation (and lower heat loss) improves the solar fraction – as evidenced by comparing base to insulated to windows cases. Radiant floor systems, which are able to make better use of the solar resource due to lower operating temperatures, outperform baseboard heaters by a substantial margin. Increasing the number of panels for a given load (e.g. radiant floor, insulated case) improves the solar fraction, since more energy is collected, while load remains constant. Finally, lower thermostat set temperatures significantly improve the solar fraction by decreasing the load met by the system.

From a standpoint of design selection, an imposed minimum solar fraction of 70% (dashed line) will disqualify many of the scenarios. For example, in the base case (no insulation), even a radiant system with four panels and a 65°F set temperature will only provide 60% of the annual load. By comparison, a well insulated system appears capable of meeting nearly 100% of the annual load. Economics, discussed in the following section, will be ultimate differentiator between competing designs.

8. ECONOMIC ANALYSIS

8.1 Components of Economic Analysis

8.1.1 Operating Expenses

Operating expenses are costs which occur over the life of a system as part of operation. The cost of fuel for the auxiliary heating unit is an example of an operating expense. Table 8.1.1 gives a listing of all operating costs and the way in which they have been modeled. Costs are in 2003 dollars.

Table 8.1.1 – Operating Expenses

Category	Item	Driver	Model	Cost
Fuel	Propane	Gallons consumed	Cost per gallon	\$1.30/gallon
	Electricity	kW hr consumed	Cost per kW hr	\$0.22/kW hr
Maintenance	Boiler and Distribution System	Installed cost	2% of initial investment	Function of equipment selection
	Collector	Installed cost	1% of initial investment	
	Collector Recharge	Gallons anti-freeze	Cost per gallon	

Additional detail for each cost item may be found in following sections.

8.1.2 Capital Investment

Capital investments, or capital costs, are one-time charges generally relating to the purchase or installation of some tangible asset (e.g. solar collector). In this design, all capital charges are incurred at installation. Table 8.1.2 gives a listing of capital costs for each portion of the system. Additionally, labor costs are assumed at $30/hr and estimates have been made for the installation time for individual components. General accounting practices allow for the capitalization of installation labor.

Table 8.1.2 – Capital Investment

Category	Equipment Cost	Install Cost	Key Drivers
Collector Loop	$3,521 - $6,221	$525	Number of panels, storage tank
Domestic Hot Water	$902 (Electric)	$255	Instant Water heater
	$204 (Propane)	$135	Combined hot water/space heat boiler
Space Heating	$6,425 (Baseboard – Electric)	$1,875	Boiler, baseboard
	$8,442 (Baseboard – Propane)	$1,620	Boiler, baseboard
	$10,625 (Radiant – Electric)	$1,875	Boiler, radiant panels
	$12,622 (Radiant – Propane)	$1,620	Boiler, radiant panels
Structure	$1,365 (Insulation)	$660	Cavity volume
	$2,625 (Windows)	$630	Window purchase

Additional detail for each investment may be found in following sections.

8.1.3 Discounted cash flow analysis

Discounted cash flow (DCF) analysis was used to calculate the cost of different engineering scenarios. The cost of a system over its lifetime will be the sum of capital and operating costs. However, a straight sum neglects the timing of costs. As an example, consider a hypothetical situation. Two people are given $100. One immediately purchases a $100 light bulb with a twenty year life. The other spends $5 a year for twenty years on cheaper light bulbs. Over twenty years, both people will have spent $100. However, the person spending $5 a year on light bulbs can invest the unspent money and expect to receive a reasonable return. This is an opportunity the first person forgoes by buying the $100 long-lasting bulb. In the case of solar thermal systems, this argument is analogous to making a large up-front investment in collectors and storage, versus paying smaller, but substantial, fuel costs each year for a traditional heating system.

To accurately compare two designs, one must take the present value of costs over the life of the system. This involves discounting the costs which occur further on in the life of the system by a pre-determined discount rate to account for the time-value of money. The sum of these discounted costs may be compared on a like-to-like basis. The present value of a stream of cash flows is given by equation 8.1.3

$$NPV = \sum_{i=1}^{n} \frac{\text{cost}_i}{(1+\text{rate})^i} \qquad \text{Equation 8.1.3}$$

where:

NPV is the net present value of the costs
i is the number of years since the initial investment
n is the number of years of costs to be taken into account (in this case 20 year)
cost_i is the system cost in year i
rate is the discount rate (in this case 2%)

The determination of the discount rate is often subjective and open to interpretation. A low discount rate has been chosen here in line with work[61] regarding comparison of renewable and fossil fuel technologies. Awerbuch argues that the high uncertainty regarding the future price of fossil fuels warrants a low effective discount rate when comparing to renewable technologies. A discount rate of 2% is used for this analysis.

8.2 Component Analysis

As with the engineering analysis, economic analysis is broken out between the collector loop, energy storage, auxiliary heating, and distribution loop.

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[61] Awerbuch

8.2.1 Collector Loop

The collector loop is comprised of collectors, pump, expansion tank, piping, and valves.

The bulk of the capital investment for the collector loop is in the collector panels themselves. The other equipment costs within the collector loop are nearly incidental by comparison. Note that the pump controller is accounted for with the distribution system. Hardware costs are given in Table 8.2.1.1. The pump must be replaced every ten years. Other components are assumed to last the entire assumed 20 year life of the system.

Table 8.2.1.1 –Collector Loop Capital Investment

Item	Cost	Comment
4x10' GOBI Collector	$852	
PV Pump	$395	12V DC
Expansion Tank	$45	4.4 gallon capacity
Auto-fill Valve	$75	Includes check valve

The only operating costs incurred by the collector loop are maintenance and a recharge of the working fluid every five years. Operations and maintenance has been fixed at 1% of the original installed capital cost. Due to the high cost of electricity on HAVO, the PV pump will be economically advantaged if the system operates for more than 3 hours per day. See Appendix F for detail.

8.2.2 Storage

A 200 gallon storage tank has been selected at a cost of $1,995. The tank is good quality and assumed to last 20 years without replacement. Operations and maintenance has been fixed at 1% of the original installed capital cost.

8.2.3 Auxiliary Heating

Fuel for auxiliary heating is responsible for the majority of system operating costs. Electric and propane units have different comparative economics. The propane fired unit requires a costly capital investment, but has lower fuel cost, while electric heating has a relatively minimal investment with a high operating cost. Clearly, propane heating will be advantaged in a situation where the auxiliary unit sees heavy use, while electric heating will be preferable in situations where auxiliary heating demands are relatively low. As previously mentioned, for geographic reasons, fuel costs (both propane and electricity) are significantly higher for HAVO than for the continental US.

8.2.3.1 Propane Auxiliary

The furnace selected for HAVO is a high quality, combination condensing boiler, with an efficiency of 95%. The specific model (Monitor MZ25S) costs $3,519 and has a maximum output of 94,500 BTU/hr. Design loads are on the order of 15,000 BTU/hr. The boiler provides auxiliary heating for space heat and incorporates a tankless water heater.

Propane is liquefied petroleum gas commonly used in residential heating applications. The fuel has a higher heating value (HHV) of 92,000 BTU/gal. The cost of propane for HAVO was assumed to be $1.30/gal. Yearly propane is cost is given below.

$$C_{Propane} = Q_{aux} \frac{E_{Propane}}{\eta} c_{Propane} \qquad \text{Equation 8.2.3.1}$$

where:

C_{Propane} is the yearly cost of propane in dollars
Q_{aux} is the total auxiliary heating load in BTU/yr
E_{Propane} is the energy density of propane in BTU/gallon
η is the efficiency with which the furnace converts propane to heat (70-95%)
c_{Propane} is the unit cost of propane for a given year in USD/gallon

Yearly maintenance for the boiler was assumed to be 2% of installed capital to account for electrical usage and system complexity.

8.2.3.2 Electric Auxiliary

Combination electric boilers do not appear to be widely available. As a result, two separate units will be required for auxiliary heating. A tankless electric hot water heating with an output comparable to the MZ25S will cost roughly $700. An electric boiler with the capacity to meet the design load will cost roughly $1,500. These boilers convert electricity to heat at 100% efficiency.

Electricity at HAVO is assumed to cost $0.22/kW-hr, roughly 3 times the cost on the continental US. Yearly cost for electricity is given by Equation 8.2.3.2.

$$C_{Electricity} = Q_{aux} * c_{Electricity} \qquad \text{Equation 8.2.3.2}$$

where:

$C_{Electricity}$ is the yearly cost of electricity in USD
Q_{aux} is the total auxiliary heating load in kW-hr/yr
$C_{Electricity}$ is the unit cost of electricity for a given year in USD/kW-hr

Yearly maintenance for the units was assumed to be 2% of installed capital.

8.2.3.3 Economic Comparison

In total, a propane system will cost $1,200 more than a comparable electric system to install. However, propane costs $0.05/kW hr, while electricity costs $0.22/kW hr. Accounting for the effects of discount rate and maintenance over a 20 year life, a design which consumes more than 15 gallons of propane per year is a suitable candidate for propane heating. Systems designed to consume less than 15 gallons of propane per year would operate more economically with an electric auxiliary.

8.2.4 Heat Distribution

Heat distribution costs are primarily capital in nature, with some yearly maintenance.

8.2.4.1 Baseboard Heating

Baseboard heater costs are driven by the length of baseboard required per room. Assuming design conditions of 40^{o}F ambient temperature, with no internal or passive solar gains, the total load on the house for the insulated case will be 14,477 BTU/hr. Dividing this load between the major rooms (e.g. excluding halls, closets) on the basis of floor area, it is possible to develop a design load per room. Length of baseboard required may be determined by dividing this load by the rated output of a baseboard radiator (given in BTU/hr/ft). Rated outputs are a function of flow rate and operating temperature. Increased flow rates fractionally increase output. Increased operating temperatures also increase output since this creates a greater temperature delta between radiator and room, improving convection coefficients. 28 feet of baseboard radiator should be adequate to meet design loads with an operating temperature of 150^{o}F. Operating at lower temperature via a set point switch as discussed in section 7.1.2.3 will be sufficient to meet the auxiliary heating load in cases where the house has been insulated.

The length of PEX required may be obtained by calculating the circuit length, then adding 3 ft of additional piping for each baseboard heater (10) and 10 ft for boiler piping[62].

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[62] Kestenholz

Major installation costs for a baseboard heating system are summarized in table 8.2.4.1.1. Yearly operations and maintenance are assumed to be 2% of installed capital.

Table 8.2.4.1 – Key Baseboard Installation Costs

Item	Equipment Cost
Baseboard Radiator	28 ft @ $150/ft = $4,200
PEX	152 ft @ $0.69/ft = $105
Controller and Thermostat	$228

8.2.4.2 Radiant Floor Heating

The cost of a radiant floor design requires specification beyond the scope of this project – usually done using proprietary supplier software (e.g. Wirsbo). For the purposes of analysis, a radiant floor system has been assumed to cost twice as much as a comparable baseboard installation.

8.2.4.3 Domestic Hot Water

Aside from an electric auxiliary, there are few costs associated with the domestic hot water system. Costs are summarized in table 8.2.4.2.1. Yearly operations and maintenance are assumed to be 2% of installed capital.

Table 8.2.4.3 – Key Domestic Hot Water Equipment Costs

Item	Equipment Cost
Tempering Valve	$95
Instant Heater (Electric)	$698
Expansion Tank	$45

8.2.5 Building Upgrades

Two types of building upgrades were considered – insulation of walls, floors, and ceilings and replacement of existing windows with low loss double-paned windows.

Since the National Park Service has plans to rewire the house – which will necessitate replacing existing drywall – the only material costs considered here are for insulation and windows. No allowance has been made for drywall and painting, as would be the case if the walls were being opened exclusively for the purpose of insulating.

8.2.5.1 Insulation

As previously mentioned, building heat loss could be substantially reduced through the introduction of insulation into the walls, floors, and ceiling. Table 8.2.5.1 shows the cost of insulating the residences. Total cost shown is for materials and labor. Keeping with building loss coefficient calculations, coverage figures assume studs or joists occupy 15% of the space within the walls, floor, or ceiling.

Table 8.2.5.1 – Insulation Costs

Cavity	Insulation	Unit Cost	Coverage	Total Cost
Walls	R-11 Fiberglass	$0.26/ft^2$	802 ft^2	$501
Floor	R-30 Fiberglass	$0.61/ft^2$	673 ft^2	$971
Ceiling	R-30 Cellulose	$0.12/ft^2$	673 ft^2	$250

8.2.5.2 Replacement Windows

Assuming a replacement window cost of $250 per window, the 11 windows in the house would cost $2,750. Labor is estimated at $60/window.

8.3 Economic Modeling

Life cycle costs have been calculated for each of the engineering scenarios considered in Section 7. In addition to the capital and operating cost assumptions stated above, the following general economic assumptions were employed.

Table 8.3.2.1 – General Economic Assumptions

Assumption	Value
Discount Rate	2%
Inflation Rate	3% yearly
Starting Year	2003
Salvage Value	$0

Discounted life cycle costs are given in table 8.3.2.2 (70°F indoor temperature) and 8.3.2.3 (65°F indoor temperature). Propane and electric auxiliaries are used where relatively advantaged by economics. The lowest cost solar option is indicated in bold.

Table 8.3.2.2 – Lifecycle Costs (70°F indoor temperature)

Insulation	Panels	Physical Model Baseboard	Auxiliary Type		Physical Model Radiant	Auxiliary Type	
			Prop	Elec		Prop	Elec
Base	0	$ 41,180	x				
	1	$ 43,056	x		$ 47,629	x	
	2	$ 41,873	x		$ 45,902	x	
	3	$ 41,149	x		$ 44,753	x	
	4	$ 40,748	x		$ 43,910	x	
Insulated	0	$ 23,213	x				
	1	$ 26,099	x		$ 30,933	x	
	2	$ 25,252	x		$ 29,695	x	
	3	**$ 24,950**	x		$ 29,504	x	
	4	$ 25,187	x		$ 29,983	x	
Windows	0	$ 23,469	x				
	1	$ 26,487	x		$ 31,564	x	
	2	$ 26,032	x		$ 31,001	x	
	3	$ 26,290	x		$ 30,451		x
	4	$ 26,343		x	$ 30,994		x

Table 8.3.2.3 – Lifecycle Costs (65°F indoor temperature)

Insulation	Panels	Physical Model Baseboard	Auxiliary Type		Physical Model Radiant	Auxiliary Type	
			Prop	Elec		Prop	Elec
Base	0	$ 29,330	x				
	1	$ 31,813	x		$ 36,468	x	
	2	$ 30,715	x		$ 34,853	x	
	3	$ 30,072	x		$ 33,913	x	
	4	$ 29,826	x		$ 33,458	x	
Insulated	0	$ 20,016	x				
	1	$ 23,041	x		$ 28,178	x	
	2	**$ 22,695**	x		$ 27,761	x	
	3	$ 23,049	x		$ 27,227		x
	4	$ 23,085		x	$ 27,803		x
Windows	0	$ 21,493	x				
	1	$ 24,551	x		$ 30,055	x	
	2	$ 24,783	x		$ 29,062		x
	3	$ 24,316		x	$ 29,661		x
	4	$ 25,021		x	$ 30,787		x

Consideration of the above tables offers a number of insights into the economics of solar thermal design.

Insulating the residences is shown to be highly cost-effective, substantially reducing the lifecycle cost. However, replacing the windows does not decrease the life-cycle cost for any of the scenarios considered.

For the 70°F set point case the lowest lifecycle cost (for a solar thermal system) is for an insulated building with three panels. Note that this system is only 7% more expensive than a conventional fossil-fuel system (no panels). For the 65°F set point case, a two panel system is cheapest and is 13% more expensive than a conventional fossil-fuel system.

Baseboard heating appears economically preferable to radiant floors in all cases. However, given the crudity of estimates for radiant panel cost (doubling of baseboard cost), it would be prudent to obtain an estimate for HAVO from a local contractor before purchasing equipment. If the cost is lower than estimated, a radiant system would make significantly better use of the available solar resource and provide higher quality space heating.

An electric auxiliary is only appropriate for scenarios with very low auxiliary heating needs. However, it should be noted that since the cross-over point between electric and propane auxiliaries is so low (15 gallons propane/year), the cost savings of switching from propane to electric is only on the order of a few hundred dollars.

Having considered the economics and engineering performance of these systems separately, an integrated comparison may now be performed to choose an optimal system.

9. Recommended Design

Final determination of a system design is based on a consideration of the both the engineering performance and the economics. Scenarios are rated by plotting engineering performance (% load met by solar) against economic performance (lifecycle cost).

Considered in quadrants, the lower right represents an ideal renewable design – low cost and a high solar fraction. The top right is representative of a "green" design, where economic considerations are subordinate to reduced fossil fuel consumption. The bottom left is the domain of more conventional systems – low cost, minimal renewable contribution. The upper left is a poorly designed system – expensive to operate and costly to install with little environmental benefit.

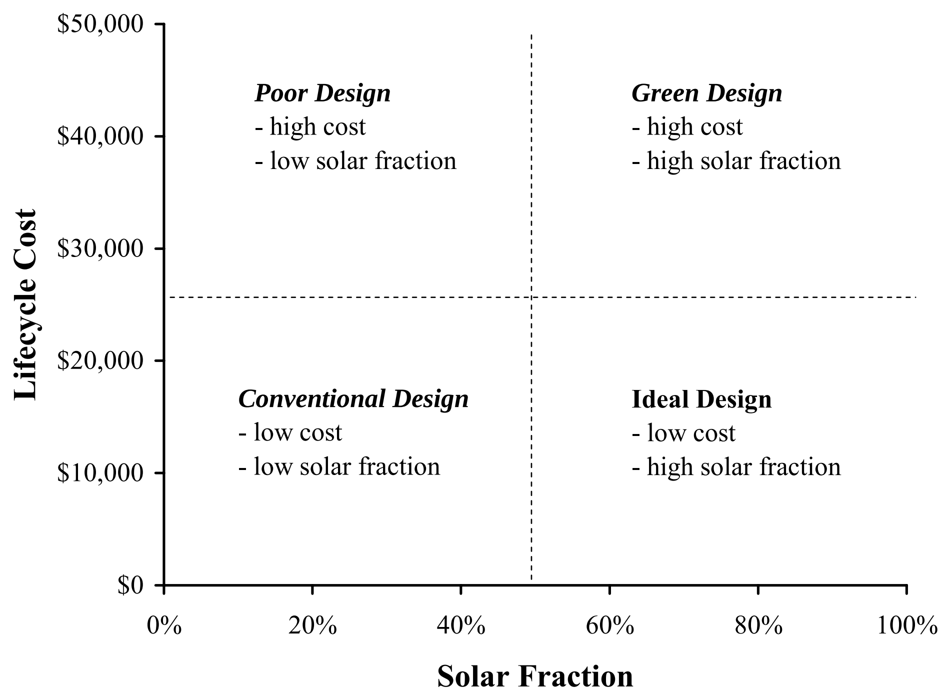


Figure 9.1 – General Design Types

Results are shown in figures 9.2 (70°F set point) and 9.3 (65°F set point). Each point on the plot represents one scenario. The data series are grouped by insulation case and type of space heating. Beneath each data point is a number indicating the number of panels and a "p" for propane auxiliary or "e" for electric auxiliary. Note that the base case is not shown on these plots because the predicted solar fraction is unacceptably low (<40%) and cost is very high (>$30,000).

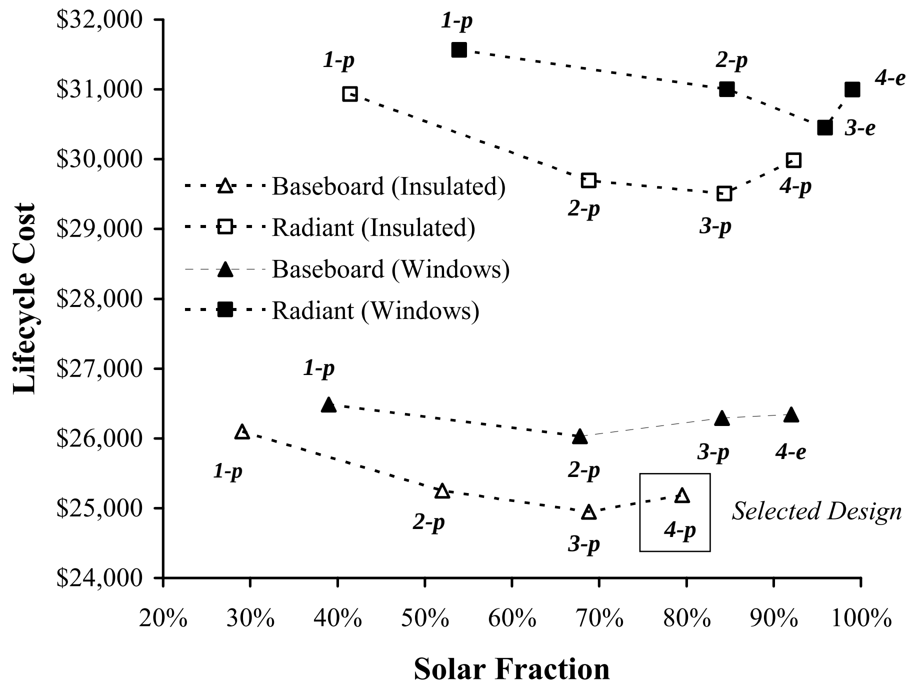


Figure 9.2 – Economic and Engineering Performance (70°F Set Point)

For a 70°F set point, a four panel system using baseboard heat in a house with insulated walls arguably offers the best trade-off between cost and performance. While only marginally more expensive than a three panel system, the additional panel increases the solar fraction by more than 10%. However, it is important to note that actually mounting four panels on the limited roof area available on the HAVO residences may be problematic for structural and aesthetic reasons. A professional contractor should be asked to make an inspection before a four panel system may be fully endorsed.

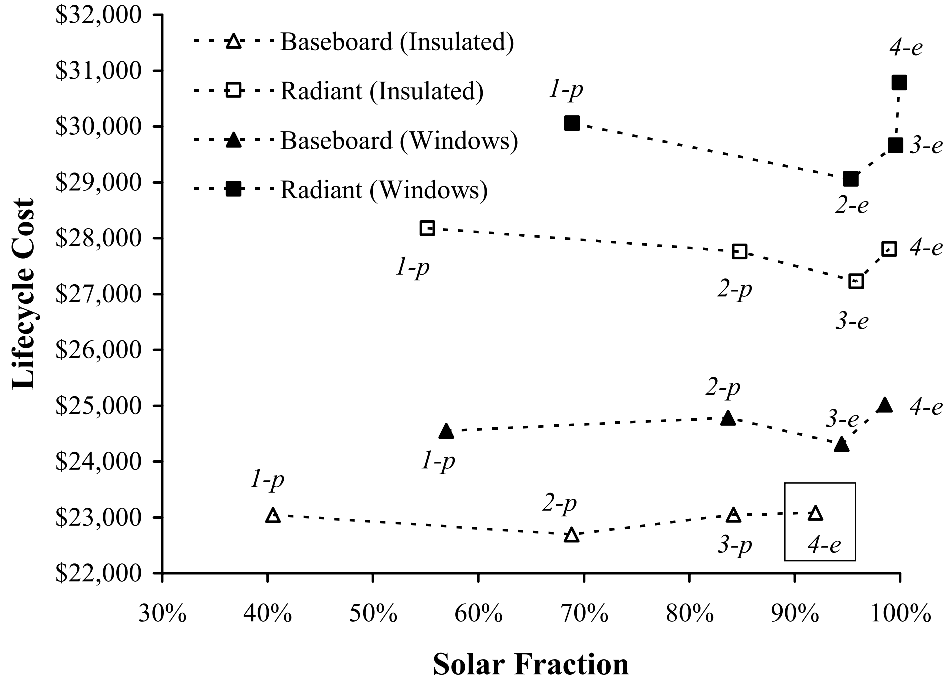


Figure 9.3 – Economic and Engineering Performance (65^{o}F Set Point)

For the 65^{o}F set point, the choice is again between a three or four panel installation. Provided structural and sighting issues can be resolved satisfactorily, a four panel system appears optimal.

Consideration of these results offers a number of interesting conclusions with relation to "green" design. The cheapest way to improve the solar fraction is to lower the thermostat set temperature. At 65^{o}F it is possible to meet more than 90% of the annual load with a solar thermal system which is cheaper over its life than the lowest cost 70^{o}F system. This underlines a key principle of renewable system design – conservation should be the first option considered. Furthermore, replacing windows appears a more economic method of increasing solar performance than installing radiant floor heating.

If a four panel system is practical to install, we would recommend doing so. Economic analysis indicates a strong need to insulate the residences, but we would not advocate replacing the windows. Baseboard heating appears preferred over radiant, though we would strongly recommend obtaining a price quote from a local contractor before making a final decision. A propane auxiliary is recommended. Note that this will somewhat increase the cost of operating the four panel system at 65^{o}F, where electric heating is advantaged. However, this increased cost is preferable to operating a four panel electric system at 70^{o}F, where economics more strongly favor propane. Specifically, using a propane auxiliary at 65^{o}F adds $800 to the lifecycle cost versus using an electric auxiliary, but using an electric auxiliary at 70^{o}F adds $2500 to the lifecycle cost versus using a propane auxiliary. If at all possible, the thermostat set point should be lowered to 65^{o}F to conserve energy – though we understand this will be dependent on the residence occupants.

9.1 Collector Loop

9.1.1 Collectors

Flat plate-collector systems are recommended over evacuated tubes, since it is unlikely the additional cost could be justified for a site with HAVO's climate. Additionally, no vendors on the Big Island appear to have experience installing evacuated tube systems – which raises practicality issues. Panels should be located on the south facing roof and mounted at 15^{o} (slightly shallower than the roof) for optimal efficiency (1% improvement). Heliodyne manufactures a high quality 4'x10'collector (GOBI 410) which would be appropriate for this project[63].

[63] Heliodyne

9.1.2 Energy Storage

Six Rivers Solar produces a line of storage tanks with capacities ranged from 80 to 500 gallons[64]. A 200 gallon tank with dual heat exchangers – for the collector loop and space heating – should be adequate to meet storage needs.

9.1.3 Loop Type

Even though temperatures have not dropped below the freezing point of water at HAVO in the past 10 years, temperatures in the upper 30's have been recorded. Taking into account clear sky radiation, it is possible that water in a collector could freeze on a clear, cold night even if the ambient temperature remains above 32^{o}F since the sky tends to be several degrees colder than ambient. For this reason, a closed-loop system will be most appropriate. Given the potential difficulties in finding a contractor on the Big Island to correctly plumb a drainback system, a closed-loop anti-freeze system has been selected, using propylene glycol as the working fluid. More detail on a potential drainback system is presented in Appendix G. To avoid stagnation boiling and reduce the electrical consumption of the system, the circulator should be powered by a small PV system. Heliodyne offers a PV system sized for use with low-head circulators.

[64] Six Rivers Solar

9.1.4 Balance of System

The major components for the balance of the system are specified in table 9.1.4.

Table 9.1.4 – Balance of System Components

Component	Specification
Pump	Helix PV -12V DC
Expansion Tank	TT 4.4 gallon, or equivalent
Pressure Relief Valve	50 psig, generic
Pipe	½” copper

9.2 Heating Applications

9.2.1 Auxiliary Heating

The MZ25S Monitor boiler is compact, highly efficient over a range of loads. This is a combination heater, providing heat for both hydronic space heating and domestic hot water. The boiler directly incorporates a number of components in the space heating loop: expansion tank, circulator (Grundfos 1542), air eliminator, and pressure relief valve. With a maximum output of 94,500 BTU/hr, this boiler will be more than sufficient to meet the design heat load.

9.2.2 Domestic Hot Water

Instant hot water is supplied by the tankless propane heater incorporated into the MZ25S. The tempering valve will provide a measure of safety to the building occupants in the case the storage tank reaches temperatures capable of scalding. Copper pipe used should be of the same diameter as the existing piping to kitchen and bathroom.

9.2.3 Space Heating

Sterling manufactures baseboard heaters suitable for installation in the residence[65]. Catalog model R05 should provide sufficient heat at 1 GPM flow rate and operating temperature of 125^{o}F and be able to meet design loads with a high set point triggered by low ambient temperatures.

The expansion tank, pressure relief valve, air elimination vent, and circulator are incorporated into the MZ25S.

Recommended lengths of baseboard heater by room are given in Table 9.2.3. Baseboard heaters should be installed under windows to optimize comfort for occupants. A complete schematic of baseboard location and pipe routing may be found in Appendix H.

Table 9.2.3 – Baseboard Heater Length by Room

Room	Suggested Length*
Bedroom 1	6 ft
Bedroom 2	6 ft
Kitchen	6 ft
Living Room	8 ft
Bathroom	2 ft

*At design load of 40oF ambient temperature, insulated residence, 1 GPM flow rate

[65] Sterling

9.2.4 System Layout

Figure 9.2.4 shows the full layout for the recommended solar thermal system.

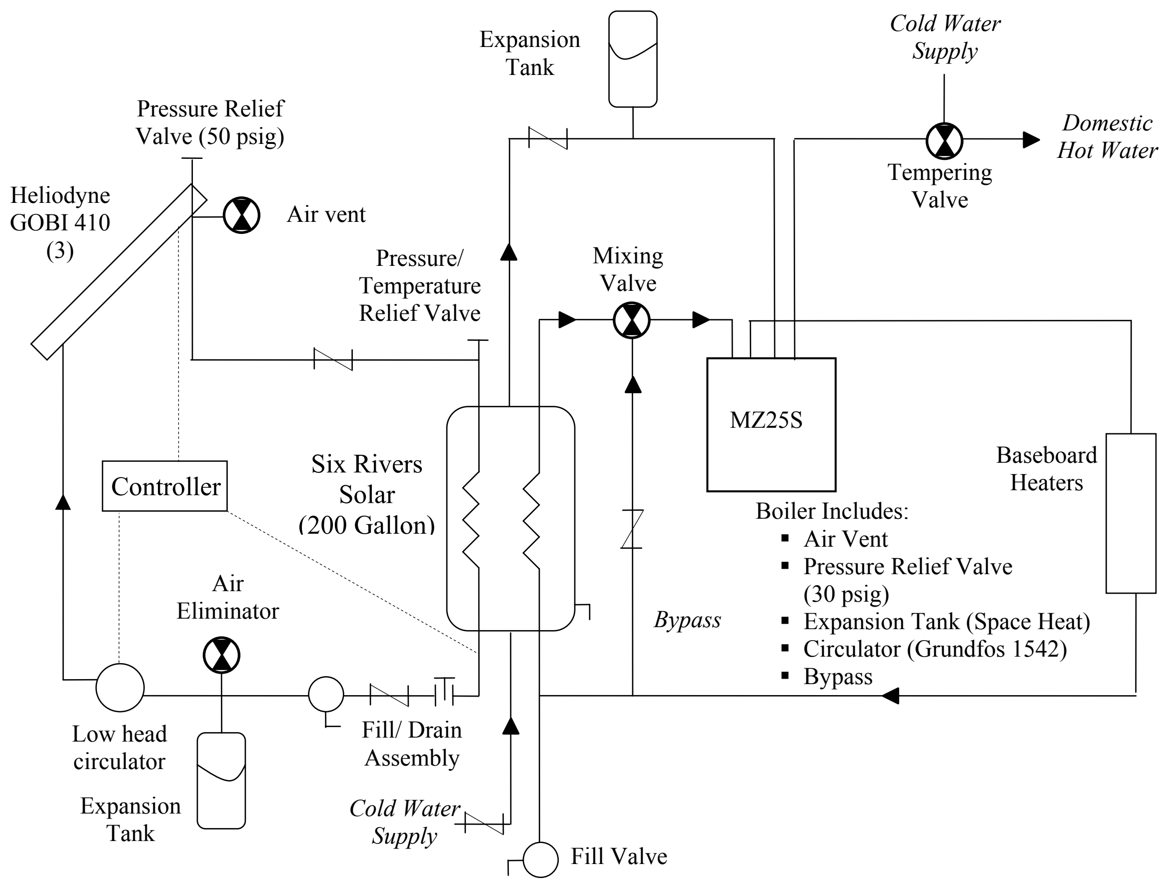


Figure 9.2.4 – Recommended System Layout

The collector panels should be mounted on the south facing roof above the external utility closet. Storage tank and boiler should be located in the closet. The MZ25S is light enough to hang from a wall without additional reinforcement. Once the residence roofs have been upgraded from sheet metal to standard construction, the weight of the collectors should not present a structural issue.

10. REFERENCES

ASHRAE Fundamentals, American Society of Heating and Refrigeration, Atlanta, GA (1981)

ASHRAE HVAC Applications, American Society of Heating and Refrigeration, Atlanta, GA (1991)

Awerbuch, Shimon, *Renewable Energy World,* "Determining the Cost: Why Renewable Power is More Cost-Competitive than Previously Believed", (2003)

Bath, Thomas, *NREL Solar Radiation Manual for Buildings,* http://rredc.nrel.gov/solar/pubs/bluebook/appendix.html#calcincrad

CASTNET, Location Data, *CASTNET Sampling Stations,* http://www.epa.gov/castnet/sites.html

CASTNET, Meteorological Data, *CASTNET Hourly Meteorological Data*, http://www.epa.gov/castnet/metdata.html

Cortez, Rose, National Park Service, Personal Communication

BRACO, *SAVEnergy Audit, Volcano National Park*, Report to the Department of Energy (2000)

Duffie, John and William Beckman, *Solar Engineering of Thermal Processes*, John Wiley & Sons, Inc., New York, NY (1991)

Elder, Keith, Coffman Engineering, Personal Communication

Heliodyne Corporation, Personal Communication

Heliodyne Website, *GOBI Solar Collectors*, http://www.heliodyne.com/gobi_solar_collectors/gobi_solar_collectors.htm

Huggins, Jim, Technical Director, Solar Rating and Certification Corporation, Personal Communication

Kestenholz, Harold, *Designing Home Heating Systems Using Warm Water*, (1996) http://64.226.150.132/hydronic/des/hotwater.pdf

Kreider, Jan and Frank Kreith, *Principles of Solar Engineering*, Hemisphere Publishing Corporation (1978)

Incropera, Frank and David DeWitt, *Introduction to Heat Transfer,* John Wiley & Sons, Inc., New York, NY (2002)

Malte, Philip C., University of Washington, Personal Communication

McDonald, William, Coffman Engineering, Personal Communication

NCDC, *Daily Surface Data,* National Climatic Data Center, Ashenville, NC (2001)

NREL, Solar Insolation Data, *NSRDB Hourly Data Files,* http://rredc.nrel.gov/solar/old_data/nsrdb/hourly/

O'Connor, Brett, Mechanical Agents, Personal Communication

Olson, Ken., "Solar Hot Water: A Primer", *Home Power*, Vol. 84 (2001)

Olson, Ken., "Solar Hot Water for Cold Climates: Closed Loop Anti-Freeze Components", *Home Power*, Vol. 85 (2001)

Olson, Ken and Tom Lane, "Solar Hot Water for Cold Climates: Part 2 Drainback Systems", *Home Power*, Vol. 86 (2001)

Six Rivers Solar, *Thermal Tanks*, http://www.sixriverssolar.com/TANK_SUBWEB/MASTER_Tanks.htm

Solar Rating and Certification Corporation, *Directory of SRCC Certified Collector Ratings*, http://www.solar-rating.org/ratings/og100directories/OG100DIRFULL_20030611.pdf, (2003)

Sterling Heating Equipment, *LB-2, Light Commercial Enclosure for to the Floor Installations,* Westfield, MA

Thermo Technologies, *Product Descriptions*, http://www.thermomax.com/

Vestergaard-Hansen and Philip C. Malte, *Moving Toward Energy Sustainability at the Marblemount (Skagit) Ranger Station,* University of Washington Report to the National Park Service, Seattle, WA (2001)

APPENDIX A – BUILDING LOSS COEFFICIENTS

Base Case	UA
Walls	142
Windows	199
Doors	18
Ceiling	416
Floor	366
Total	1141

Exterior Walls	R	U	A	UA
Windows				
Window	0.98			
Windows Total	0.98	1.020	195.4	**199.4**
Doors				
Door	2.63			
Doors Total	2.6	0.380	46.2	**17.6**
Walls				
Stud Area				
Vert. Interior Surface	0.68			
Drywall	0.45			
Stud	4.55			
Exterior Sheathing	0.93			
Exterior Siding	1.3			
Exterior Surface	0.25			
	8.16	0.123	105.4	12.9
Cavity Area				
Vert. Interior Surface	0.68			
Drywall	0.45			
Air Gap	1.01			
Exterior Sheathing	0.93			
Exterior Siding	1.30			
Exterior Surface	0.25			
	4.62	0.216	597.0	129.2
Walls Total				**142.1**

Ceiling	R	U	A	UA
Stud Area				
Slop. Interior Surface	0.62			
Roof Sheathing	0.77			
Felt Membrane	0.06			
Shingles	0.44			
Exterior Surface	0.25			
Horz. Interior Surface	0.61			
Drywall	0.45			
Horz. Interior Surface	0.61			
	3.81	0.262	237.6	62.36
Cavity Area				
Slop. Interior Surface	0.62			
Roof Sheathing	0.77			
Felt Membrane	0.06			
Shingles	0.44			
Exterior Surface	0.25			
Horz. Interior Surface	0.61			
Drywall	0.45			
Horz. Interior Surface	0.61			
	3.81	0.262	1346.4	353.39
Ceiling Total				**415.75**

Floor	R	U	A	UA
Hardwood				
Stud Area				
Horz. Interior Surface	0.61			
Hardwood	0.68			
Subfloor	0.93			
Spray Insulation	1.47			
Horz. Interior Surface	0.61			
	4.3	0.233	91.8	21.3
Cavity Area				
Horz. Interior Surface	0.61			
Hardwood	0.68			
Subfloor	0.93			
Spray Insulation	1.47			
Horz. Interior Surface	0.61			
	4.3	0.233	520.2	121.0
Tile				
Stud Area				
Horz. Interior Surface	0.61			
Tile	0.05			
Subfloor	0.93			
Spray Insulation	1.47			
Horz. Interior Surface	0.61			

Floor	R	U	A	UA
Carpet				
Stud Area				
Horz. Interior Surface	0.61			
Carpet	2.08			
Subfloor	0.93			
Spray Insulation	1.47			
Horz. Interior Surface	0.61			
	5.7	0.175	86.4	15.2
Cavity Area				
Horz. Interior Surface	0.61			
Hardwood	0.68			
Subfloor	0.93			
Spray Insulation	1.47			
Horz. Interior Surface	0.61			
	4.3	0.233	489.6	113.9
Floor Total				**365.8**

Insulated Case	UA
Walls	54
Windows	199
Doors	18
Ceiling	57
Floor	59
Total	388

Exterior Walls	R	U	A	UA
Windows				
Window	0.98			
Windows Total	0.98	1.020	195.4	**199.4**
Doors				
Door	2.63			
Doors Total	2.6	0.380	46.2	**17.6**
Walls				
Stud Area				
Vert. Interior Surface	0.68			
Drywall	0.45			
Stud	4.55			
Exterior Sheathing	0.93			
Exterior Siding	1.3			
Exterior Surface	0.25			
	8.16	0.123	105.4	12.9
Cavity Area				
Vert. Interior Surface	0.68			
Drywall	0.45			
R-11	11.00			
Exterior Sheathing	0.93			
Exterior Siding	1.30			
Exterior Surface	0.25			
	14.61	0.068	597.0	40.9
Walls Total				**53.8**

Ceiling	R	U	A	UA
Stud Area				
Slop. Interior Surface	0.62			
Roof Sheathing	0.77			
Felt Membrane	0.06			
Shingles	0.44			
Exterior Surface	0.25			
Horz. Interior Surface	0.61			
Drywall	0.45			
Floor Joist	9.75			
Horz. Interior Surface	0.61			
	13.56	0.074	237.6	17.52
Cavity Area				
Slop. Interior Surface	0.62			
Roof Sheathing	0.77			
Felt Membrane	0.06			
Shingles	0.44			
Exterior Surface	0.25			
Horz. Interior Surface	0.61			
Drywall	0.45			
R-30 Cell	30.00			
Horz. Interior Surface	0.61			
	33.81	0.030	1346.4	39.82
Ceiling Total				**57.34**

Floor	R	U	A	UA
Hardwood				
Stud Area				
Horz. Interior Surface	0.61			
Hardwood	0.68			
Subfloor	0.93			
Floor Joist	9.75			
Horz. Interior Surface	0.61			
	12.58	0.079	91.8	7.3
Cavity Area				
Horz. Interior Surface	0.61			
Hardwood	0.68			
Subfloor	0.93			
R-30	30.00			
Horz. Interior Surface	0.61			
	32.83	0.030	520.2	15.8
Tile				
Stud Area				
Horz. Interior Surface	0.61			
Tile	0.05			
Subfloor	0.93			
Floor Joist	9.75			
Horz. Interior Surface	0.61			

Floor	R	U	A	UA
Carpet				
Stud Area				
Horz. Interior Surface	0.61			
Carpet	2.08			
Subfloor	0.93			
Floor Joist	9.75			
Horz. Interior Surface	0.61			
	13.98	0.072	86.4	6.2
Cavity Area				
Horz. Interior Surface	0.61			
Hardwood	0.68			
Subfloor	0.93			
R-30	30.00			
Horz. Interior Surface	0.61			
	32.83	0.030	489.6	14.9
Floor Total				**59.5**

Insulated Case	UA
Walls	54
Windows	68
Doors	18
Ceiling	57
Floor	59
Total	257

Exterior Walls	R	U	A	UA
Windows				
Window	2.86			
Windows Total	2.8571429	0.350	195.4	**68.4**
Doors				
Door	2.63			
Doors Total	2.6	0.380	46.2	**17.6**
Walls				
Stud Area				
Vert. Interior Surface	0.68			
Drywall	0.45			
Stud	4.55			
Exterior Sheathing	0.93			
Exterior Siding	1.3			
Exterior Surface	0.25			
	8.16	0.123	105.4	12.9
Cavity Area				
Vert. Interior Surface	0.68			
Drywall	0.45			
R-11	11.00			
Exterior Sheathing	0.93			
Exterior Siding	1.30			
Exterior Surface	0.25			
	14.61	0.068	597.0	40.9
Walls Total				**53.8**

Ceiling	R	U	A	UA
Stud Area				
Slop. Interior Surface	0.62			
Roof Sheathing	0.77			
Felt Membrane	0.06			
Shingles	0.44			
Exterior Surface	0.25			
Horz. Interior Surface	0.61			
Drywall	0.45			
Floor Joist	9.75			
Horz. Interior Surface	0.61			
	13.56	0.074	237.6	17.52
Cavity Area				
Slop. Interior Surface	0.62			
Roof Sheathing	0.77			
Felt Membrane	0.06			
Shingles	0.44			
Exterior Surface	0.25			
Horz. Interior Surface	0.61			
Drywall	0.45			
R-30 Cell	30.00			
Horz. Interior Surface	0.61			
	33.81	0.030	1346.4	39.82
Ceiling Total				**57.34**

Floor	R	U	A	UA
Hardwood				
Stud Area				
Horz. Interior Surface	0.61			
Hardwood	0.68			
Subfloor	0.93			
Floor Joist	9.75			
Horz. Interior Surface	0.61			
	12.58	0.079	91.8	7.3
Cavity Area				
Horz. Interior Surface	0.61			
Hardwood	0.68			
Subfloor	0.93			
R-30	30.00			
Horz. Interior Surface	0.61			
	32.83	0.030	520.2	15.8
Tile				
Stud Area				
Horz. Interior Surface	0.61			
Tile	0.05			
Subfloor	0.93			
Floor Joist	9.75			
Horz. Interior Surface	0.61			

Floor	R	U	A	UA
Carpet				
Stud Area				
Horz. Interior Surface	0.61			
Carpet	2.08			
Subfloor	0.93			
Floor Joist	9.75			
Horz. Interior Surface	0.61			
	13.98	0.072	86.4	6.2
Cavity Area				
Horz. Interior Surface	0.61			
Hardwood	0.68			
Subfloor	0.93			
R-30	30.00			
Horz. Interior Surface	0.61			
	32.83	0.030	489.6	14.9
Floor Total				**59.5**

APPENDIX B – INFILTRATION LOSSES

Volume (Conditioned Space)

Length	24.0
Width	35.0
Height	8.0
Volume	6336.0

Air Constants

specific heat	0.24	BTU/lb R
density	0.075	lb / ft3

Air Changes

	Room Type	No Weatherstrip
A	*No windows or doors*	0.5
B	*Windows or exterior doors on one side*	1
C	*Windows or exterior doors on two sides*	1.5
D	*Windows or exteror doors on three sides*	2
E	*Entrance halls*	2

Weatherstrip No

	Type	Air Changes	Length	Width	Area	Volume
Living Room	C	1.5	17.0	12.0	204.0	1632
Kitchen	C	1.5	12.0	12.0	144.0	1152
Bedroom 1	C	1.5	12.0	12.0	144.0	1152
Bedroom 2	B	1.0	12.0	12.0	144.0	1152
Closet	A	0.5	7.0	7.0	49.0	392
Hall		0.0	6.0	3.0	18.0	144
Porch	B	1.0	5.0	7.0	35.0	280
Bath	B	1.0	9.0	6.0	54.0	432
Average (per hour)		1.26				

Air Change Mass

cpρnV 142.5

APPENDIX C – BUILDING LAYOUT

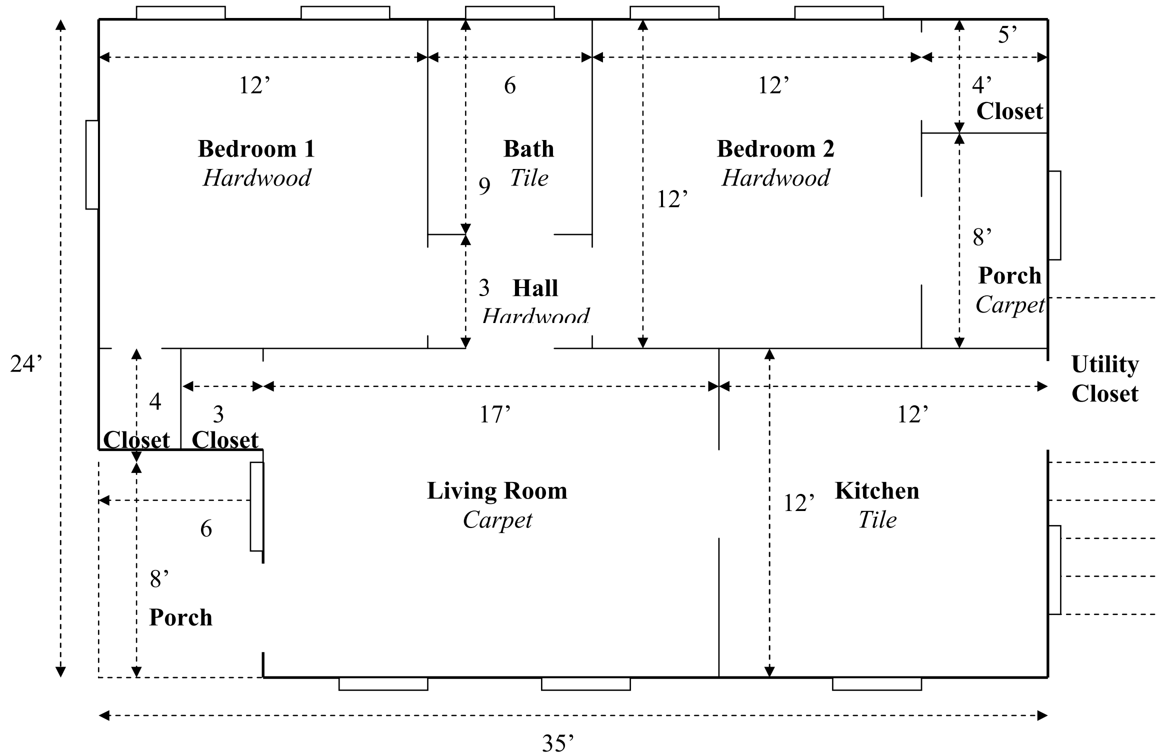


Figure C.1 – Floor Plan Detail

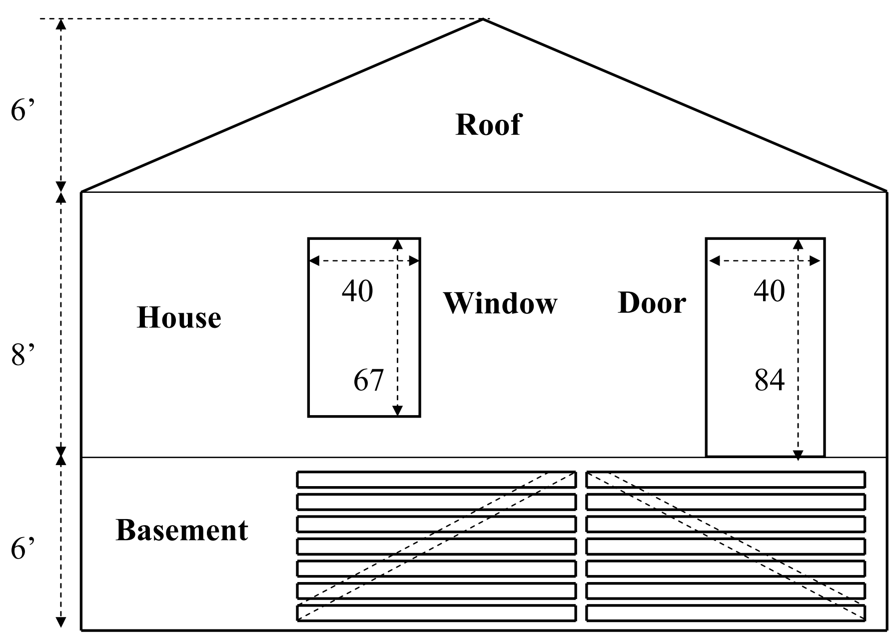


Figure C.2 – Side View

APPENDIX D – HEAT EXCHANGER EXIT TEMPERATURE

The space heating heat exchanger may be approximately modeled by a set of equations and a number of simplifying assumptions.

On the space heating side, the flow will pass through a copper tube at 1 GPM, entering at 80°F (radiant case) and exiting at some unknown temperature. The tank temperature is known, and assumed to be constant (e.g. constant temperature external flow).

The exiting temperature will be given by equation D.1.

$$\frac{T_\infty - T_{m,o}}{T_\infty - T_{m,i}} = exp\left(-\frac{\overline{U}A_s}{\dot{m}c_p}\right) \qquad \text{Equation D.1}$$

where:

T_∞ is the tank temperature
$T_{m,o}$ is the temperature of fluid leaving the heat exchanger
$T_{m,i}$ is the temperature of the fluid entering the heat exchanger
U is the average overall heat transfer coefficient for the exchanger
m is the flow rate of fluid through the exchanger
c_p is the specific heat of the fluid in the exchanger
A_s is the surface area of the heat exchanger

The heat exchanger coil was assumed to be ¾" OD copper pipe (thin wall) wrapped in a 10" diameter helix with a 15° rise. This configuration allows for 18 turns in the tank chosen for installation.

Fluid properties on the exchanger side are evaluated at T_m, the average of inlet and outlet temperature. Fluid properties on the tank side are evaluated at T_f, the average of the tube surface temperature and tank temperature.

The mass flow rate of fluid through the heat exchanger will be given by multiplying the volumetric flow rate (1 GPM by the density of fluid.

The overall heat transfer coefficient may be obtained by considering the exchanger and tank as a series network, assuming the tube is sufficiently thin to be neglected. U is given by equation D.2.

$$\overline{U} = \frac{\overline{h}_{Tank}\,\overline{h}_{Tube}}{\overline{h}_{Tank} + \overline{h}_{Tube}} \qquad \text{Equation D.2}$$

where h_{Tank} and h_{Tube} are the respective convective coefficients

Both convective coefficients may be obtained by first calculating the dimensionless Nusselt number (Nu) and applying equation D.3.

$$\overline{h} = \overline{Nu_D}\,\frac{k}{D} \qquad \text{Equation D.3}$$

where:

D is the tube diameter
k is the thermal conductivity of the fluid
Nu_D is the Nusselt number based on tube diameter

For internal flow, as is the case within the tube, the Nusselt number is given by equation D.4.

$$\overline{Nu_D} = 0.023\,Re^{4/5}\,Pr^{0.4} \qquad \text{Equation D.4}$$

where:

Re is the Reynolds number
Pr is the Prandtl number, a property of the flow

The Reynolds number is given by equation D.5.

$$Re = \frac{4\dot{m}}{\pi D \mu}$$ Equation D.5

where:

m is the mass flow rate in the tube
D is the tube diameter
μ is the kinematic viscosity of the fluid

Note that relation D.4 is only valid for fully developed, turbulent flows being heated by an external source. The Re, for relevant operating conditions, indicates turbulence.

Heat transfer from tank to the tube will be via free convection, using a separate set of equations. The Nusselt number in this case will be given by equation D.6.

$$\overline{Nu_D} = \left[0.60 + \frac{0.387 Ra_D^{1/6}}{\left(1 + \left(\frac{0.559}{Pr}\right)^{9/16}\right)^{8/27}} \right]^2$$ Equation D.6

where Ra_D is the Rayleigh number

The Rayleigh number for free convection problems is a dimensionless quantity of the form given in equation D.7.

$$Ra_D = \frac{g\beta(T_\infty - T_s)D^2}{\nu\alpha}$$ Equation D.7

where:

g is the acceleration due to gravity
β is $1/T_f$
T_∞ is the tank temperature
T_s is the temperature on the tube surface
D is the diameter
ν is the dynamic viscosity of the tank fluid

α is the thermal diffusivity of the tank fluid

The system of equations is closed by noting that the heat flux from the tank fluid to tube surface must be equation to the heat flux from the tube surface to the tube fluid. This relation is explicitly defined in equation D.8.

$$\bar{h}_{Tank}\left(T_\infty - T_s\right) = \bar{h}_{Tube}\left(T_s - T_m\right) \qquad \text{Equation D.8}$$

This system of equations requires an iterative solving technique owing the temperature dependence of the fluid properties. Using a software package designed for this purpose (IHT) $T_{m,out}$ may be obtained for a number of operating conditions. Over the range of operation, the simulation indicates the temperature difference between heat exchanger outlet and tank to be no more than 2°F for most cases.

Results are presented in Table D.1.

Table D.1 – Heat Exchanger Simulation Results

Tank Temperature (°F)	Heat Exchanger Exit Temperature (°F)	Delta (°F)
116.6	113.72	2.88
120.2	118.76	1.44
123.8	122.18	1.62
127.4	125.78	1.62
131	129.38	1.62
134.6	132.98	1.62
138.2	136.94	1.26
141.8	140.18	1.62
145.4	143.6	1.8
149	147.2	1.8
152.6	150.8	1.8
156.2	154.4	1.8
159.8	158	1.8
163.4	161.6	1.8
167	165.02	1.98
170.6	168.62	1.98
174.2	165.02	9.18
177.8	176	1.8
181.4	179.6	1.8
185	183.2	1.8
188.6	186.8	1.8
192.2	190.4	1.8
195.8	194	1.8

The simulation converged to a solution with some difficulty, but results appear largely consistent (with the exception of the 9^{o}F outlier).

APPENDIX E – PROGRAM MODULES

Module	File Name	Input	Output
Temperature	Temperature.xls	Ambient temperature profile (average, warmest, coldest)	Ambient temperatures for each hour of the year
Internal Heat Generation	Load_Internal Genration.xls	Daily occupancy and appliance use profile Heat generated by occupants and appliances	Internal heat generation
Domestic Hot Water	Load_Hot Water.xls	Daily hot water consumption profile Cold water supply temperature	Domestic hot water load
Solar Angles	Solar_Location.xls	Latitude Longitude	Declination (δ) Hour Angle (ω) Zenith Angle (θ_z)
Passive Solar	Solar_Passive.xls	Solar Angles Insolation Structure Geometry	Passive gain
Collector Incidence	Solar_Collector.xls	Solar Angles Insolation Panel Geometry	Collector Irradiation (J) Collector Insolation (W/m^2)
Space Heating	Load_Thermal.xls	Internal heat generation Passive solar gain Ambient temperature Building Loss Coefficient Infiltration Building Capacitance	Space heating load
***f*-Chart Analysis**	Analysis_FCHART.xls	Collector irradiation Collector characteristics Domestic hot water load Space heating load	Fraction of load met by solar
System Model	System_Model.xls	Collector irradiation Ambient temperature System specifications Domestic hot water load Space heating load	Fraction of load met by solar
Economic Model	Economics.xls	Cost assumptions Auxiliary heating load Structure and system detail	Lifecycle cost Gallons propane consumed per year

APPENDIX F – SOLAR PV PUMPING ECONOMICS

Inputs

Inflation	3%	
Discount	2%	
Power	33.80	W
Uptime	2.51	hrs
	31.01	kW hr/year

Cost Model

	Cost	CAGR	1	2	3	4	5	6	7	8	9	10
Electric	$ 0.22	3%	$ 0.22	$ 0.23	$ 0.23	$ 0.24	$ 0.25	$ 0.26	$ 0.26	$ 0.27	$ 0.28	$ 0.29
			$ 6.82	$ 7.03	$ 7.24	$ 7.46	$ 7.68	$ 7.91	$ 8.15	$ 8.39	$ 8.64	$ 8.90

NPV $147.00

Cost Comparison

	PV	Electric	Delta
Capital	$ 395	$ 248	$ 147
Operating		$ 147	$ (147)
Total	$ 395	$ 395	$ -

APPENDIX G – DRAINBACK SYSTEM

As discussed previously, while potentially more difficult to install, drainback systems offer a number of advantages over anti-freeze closed loops.

First, drainback systems offer inherent freeze and boil protection – never relying on circulation to prevent either extreme. These systems also contain fewer working parts, thus requiring less maintenance. Thirdly, since the working fluid is never exposed to freezing conditions water may be used without reservation. Distilled water is recommended to avoid build-ups of mineral deposits. The potential to use water in lieu of non-potable fluids is attractive for environmental reasons.

The only substantial drawback to this system – barring improper installation – is the loop circulator must be grid-tied. Power consumption during start-up pressurization makes solar PV uneconomic. Literature suggests a PV-battery assisted system could meet the demand, but is not commercially available[66]. Improper installation of drainback systems, as previously mentioned, may result in catastrophic freeze damage.

A schematic for a drainback system with a propane auxiliary is shown in figure G.1.

[66] Ken Olson, Home Power #86

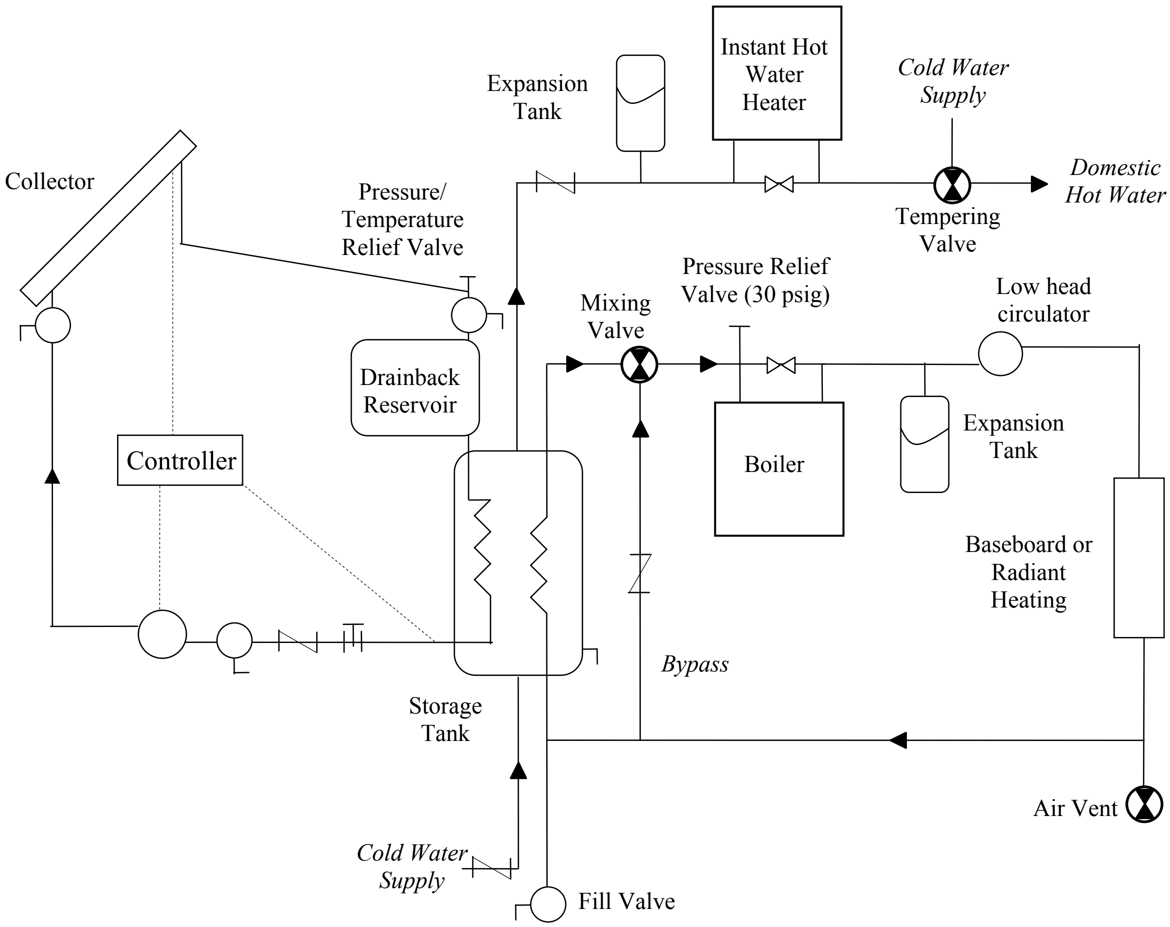


Figure G.1 – Drainback System Layout

APPENDIX H – BASEBOARD HEATER LAYOUT

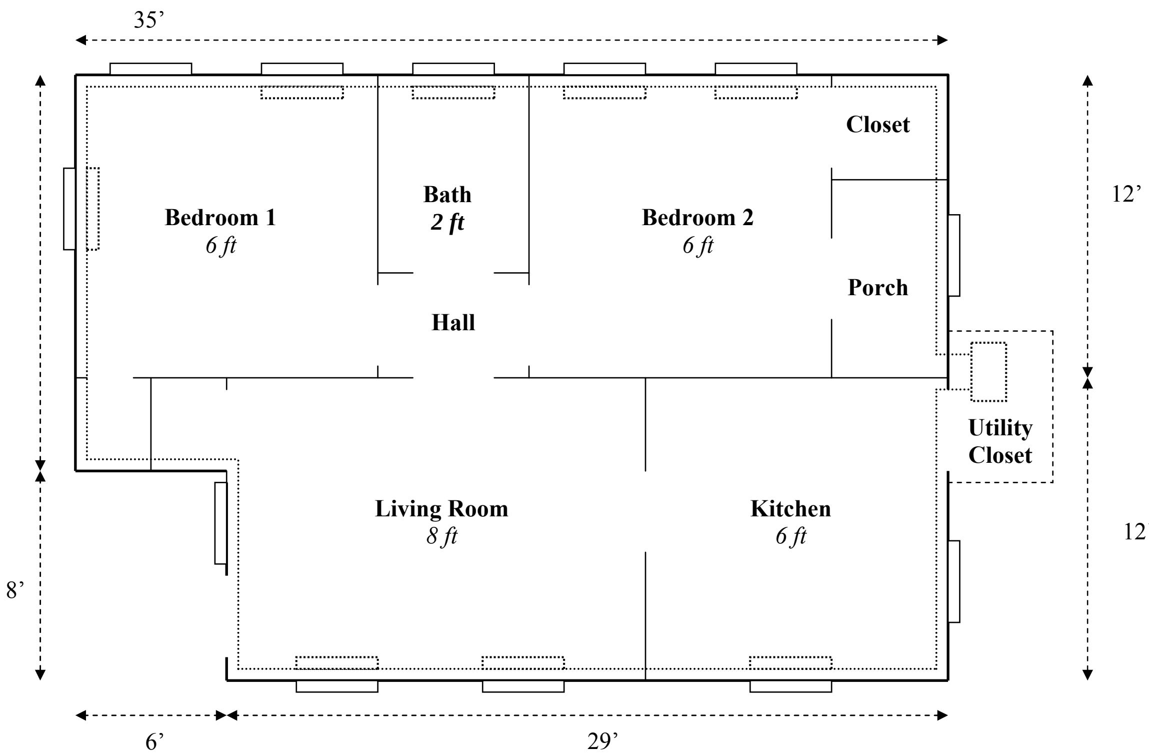


Figure H.1. – Baseboard Heater Layout

APPENDIX I – INTERNAL GENERATION ASSUMPTIONS

Unit Generation

Component	W	BTU/hr
Occupant	100	341
Refrigerator	725	2474
Stove	1500	5120
Lights	60	205

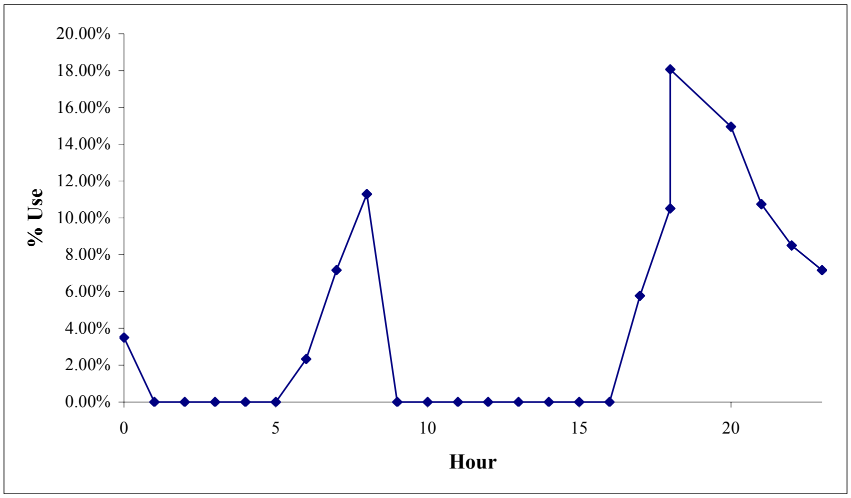
Daily Assumptions

Hour	Occupants	Occupant Heat	Refrigerator Use	Refrigerator Heat	Stove Use	Stove Heat	Light Use	Light Heat	Heat Generated
0	2	683	0.3	742	-	-	-	-	1,425
1	2	683	0.3	742	-	-	-	-	1,425
2	2	683	0.3	742	-	-	-	-	1,425
3	2	683	0.3	742	-	-	-	-	1,425
4	2	683	0.3	742	-	-	-	-	1,425
5	2	683	0.3	742	-	-	-	-	1,425
6	2	683	0.3	742	-	-	-	-	1,425
7	2	683	0.3	742	1.0	5,120	3.0	614	7,159
8	2	683	0.3	742	-	-	3.0	614	2,039
9	-	-	0.3	742	-	-	-	-	742
10	-	-	0.3	742	-	-	-	-	742
11	-	-	0.3	742	-	-	-	-	742
12	-	-	0.3	742	-	-	-	-	742
13	-	-	0.3	742	-	-	-	-	742
14	-	-	0.3	742	-	-	-	-	742
15	-	-	0.3	742	-	-	-	-	742
16	-	-	0.3	742	-	-	-	-	742
17	2	683	0.3	742	-	-	3.0	614	2,039
18	2	683	0.3	742	1.0	5,120	4.0	819	7,364
19	2	683	0.3	742	-	-	4.0	819	2,244
20	2	683	0.3	742	-	-	4.0	819	2,244
21	2	683	0.3	742	-	-	4.0	819	2,244
22	2	683	0.3	742	-	-	2.0	410	1,834
23	2	683	0.3	742	-	-	-	-	1,425

APPENDIX J – DOMESTIC HOT WATER ASSUMPTIONS

Hourly Profile

0	3.50%
1	0.00%
2	0.00%
3	0.00%
4	0.00%
5	0.00%
6	2.34%
7	7.16%
8	11.29%
9	0.00%
10	0.00%
11	0.00%
12	0.00%
13	0.00%
14	0.00%
15	0.00%
16	0.00%
17	5.76%
18	10.51%
18	18.07%
20	14.95%
21	10.75%
22	8.50%
23	7.16%



Yearly Source Profile

Month		C	F
Jan	1	13.4	56.2
Feb	2	13.3	55.9
Mar	3	13.5	56.2
Apr	4	13.2	55.8
May	5	14.5	58.0
Jun	6	15.4	59.8
Jul	7	16.0	60.9
Aug	8	16.6	61.9
Sep	9	16.3	61.4
Oct	10	15.6	60.2
Nov	11	15.0	59.0
Dec	12	14.2	57.6

ρ	1000	kg / m^3
c_p	4180	J / kg K

Delivery T 48.888889 C

Unit Use	76	L/day
Occupants	2	
	0.152	m^3 / day

APPENDIX K – FINANCIAL MODEL

Collector and Storage

System Characteristics

Maint. %	1%	Installed CapEx
Inflation	3%	
Labor Rate	$ 30.00	$/hr
ft3 -> gal	7.4805	

Collector and Storage

		Unit Cost	Units	Mat. Cost	Install Time	Labor	Total Cost	Model
Collector								
	Panel	$ 851.96	1	$ 852	4.0	$ 120	$ 972	Heliodyne GOBI 410
	Mounting Hardware	$ 44.57	1	$ 45	-	$ -	$ 45	Heliodyne
	Blind Unions	$ 9.25	2	$ 18	-	$ -	$ 18	Heliodyne
Pump								
	PV Pump Kit	$ 395.00	1	$ 395	1.0	$ 30	$ 425	Helix PV Pump Statio
Storage Tank								
	Storage Tank	$ 1,995	1	$ 1,995	3.0	$ 90	$ 2,085	Six Rivers Solar 200
Ancillary Components								
	Expansion Tank	$ 45	1	$ 45	1.0	$ 30	$ 75	Generic
	Auto-Fill Valve	$ 75	1	$ 75	0.5	$ 15	$ 90	Generic
	Check Valve	$ 10	2	$ 20	0.5	$ 15	$ 35	Generic
	Pressure Relief Valve	$ 20	2	$ 40	1.0	$ 30	$ 70	Generic
	Air Vent	$ 15	1	$ 15	0.5	$ 15	$ 30	Generic
	Piping	$ 0.42	50	$ 21	6.0	$ 180	$ 201	Copper pipe
Collector and Storage				$ 3,521			$ 4,046	

Loop Fluid

Working Fluid	$ 25.00	per gallon	DYN-O Flow Heat Transfer Fluid
Gallons	1.51		
	1.00	per collector	
	0.51	pipe volume	
Recharge	5	years	

Domestic Hot Water and Space Heating

System Characteristics

Maint. %	2%	Installed CapEx - Electric demand, complexity
Inflation	3%	
Labor Rate	$ 30.00	$/hr

Fuel Characteristics

Propane

Cost	$ 1.30	$/gallon
Efficiency	95%	
Energy Density	92,000	BTU/gallon

Electricity

Cost	$ 0.22	$/kW hr
Efficiency	100%	

Domestic Hot Water

		Unit Cost	Units	Mat. Cost	Install Time	Labor	*Active*	Total Cost	Model
Electric Auxiliary									
	Instant Heater	$ 698	1	$ 698	4.0	$ 120	1	$ 818	SETS 220
Propane Auxiliary									
	Boiler	$ -	1	$ -	-	$ -	-	$ -	MZ25S
Anciliary Components									
	Tempering Valve	$ 95	1	$ 95	1.0	$ 30	1	$ 125	Generic
	Bypass Assy	$ 50	1	$ 50	1.0	$ 30	1	$ 80	Generic
	Expansion Tank	$ 45	1	$ 45	1.0	$ 30	1	$ 75	Generic
	Check Valve	$ 10	1	$ 10	0.5	$ 15	1	$ 25	Generic
	Piping	$ 0.425	10	$ 4	1.0	$ 30	1	$ 34	1/2" copper
Domestic Hot Water				$ 902				$ 1,157	

Space Heating

		Unit Cost	Units	Mat. Cost	Install Time	Labor	*Active*	Total Cost	Model
Electric Auxiliary									
	Boiler	$ 1,394	1	$ 1,394	4.0	$ 120	1.00	$ 1,514	Elect-T-Therm
Propane Auxiliary									
	Boiler	$ 3,519	1	$ -	4.0	$ 120	-	$ -	MZ25S
	Ductwork	$ 255	1	$ -	4.0	$ 120	-	$ -	
Anciliary Components									
	Circulator	$ 242.00	1	$ 242	1.0	$ 30	1	$ 272	TACO 006B
	Expansion Tank	$ 45	1	$ 45	1.0	$ 30	1	$ 75	Generic
	Air Vent	$ 15	1	$ 15	0.5	$ 15	1	$ 30	Generic
	Bypass Assy	$ 50	1	$ 50	1.0	$ 30	1	$ 80	Generic
	Pressure Relief	$ 20	1	$ 20	0.5	$ 15	1	$ 35	Generic
	Check Valve	$ 10	1	$ 10	0.5	$ 15	1	$ 25	Generic
	Mixing Valve	$ 116	1	$ 116	1.0	$ 30	1	$ 146	Generic
	Piping	$ 0.69	152	$ 105	12.0	$ 360	1	$ 465	RAUPEX 1/2"
Radiators									
	Standard	$ 150.00	28	$ 8,400	1.0	$ 840	2	$ 9,240	Sterling
Controls									
	Controller	$ 203.80	1	$ 204	4.0	$ 120	1	$ 324	MODEL
	Thermastat	$ 23.91	1	$ 24	1.0	$ 30	1	$ 54	MODEL
Space Heating				$ 10,625				$ 12,260	

Structure

System Characteristics

Labor Rate $ 30.00 $/hr

Capital Expense

			Unit Cost	Units		Mat. Cost	Install Time	Labor	Total Cost
Insulation	Walls	R-11	$ 0.26	802.4	ft2	$ 209	10	$ 300	$ 509
	Floors	R-30	$ 0.61	1346.4	ft2	$ 821	5	$ 150	$ 971
	Ceiling	R-30 Cell	$ 0.12	1346.4	ft2	$ 160	3	$ 90	$ 250
Windows		Standard	$ -	10.5	number	$ -	2	$ -	$ -
Weatherstrip		Yes	$ 0.05	3495.2	ft2 surface	$ 175	4	$ 120	$ 295
Structure									$ 2,025