4309

A COMPREHENSIVE METHODOLOGY TO DEFINE APPROPRIATE REGULATING CRITERIA FOR SOLAR DOMESTIC HOT WATER SYSTEMS

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We certify that we have read this thesis and that, in our opinion, it is satisfactory in scope and quality as a thesis for the degree of Master of Science in Mechanical Engineering.

THESIS COMMITTEE

Chairperson

To my mothers across the ocean;

my Mom, Debra,
who never doubted.

Jayn
for her wisdom.

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ABSTRACT

The current state of the environment, as well as socioeconomic factors, demands a rapid decrease in energy use in every area of society. This situation is why different efforts are in place to promote energy conservation. State building codes using standards such as ASHRAE 90.1, require compliance with minimum efficiency levels. Federal limits mandate individual product efficiency. There are also voluntary programs and benchmarks including ENERGY STAR® and LEED®.

Domestic hot water (DHW) production is the second highest energy demand under HVAC for residential buildings. DHW generated by solar thermal energy is a renewable way to reduce this demand. Yet, the incorporation of distributed solar thermal energy systems into standards and regulations is just recently being addressed, with the language at best being loosely stated. This is primarily because there are a wide variety of design options available and the systems' performance is dependent on demand characteristics as well. ENERGY STAR®, a voluntary labeling program to promote energy efficient products established as of April 1st new criteria to include solar domestic hot water (SDHW) systems. Their approach has already been questioned in the field.

This research incorporates a numerical study done using TRNSYS software to simulate 560 different SDHW system scenarios by changing the demand characteristics including location, as well as system characteristics. A regression analysis was done next to quantify the dependencies by fitting the data and by direct comparisons. The results were then compiled to support a proposed new rating procedure utilizing unique dual criteria to be used to define appropriate regulating language for SDHW systems. The underlying methodology of the proposed rating procedure is dependent on the establishment of a new set of climate categories.

Defining appropriate regulating criteria is needed to both promote higher levels of product efficiency as well as incorporate SDHW systems as an option in residential building codes. This is also a step towards defining appropriate use of solar technology in all aspects of building design including active and passive solar heating and air-conditioning technology.

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NOMENCLATURE

AEC	annual energy cost
A_c	collector area
$C_{p,c}$	specific heat of collector-side flow loop
$C_{p,t}$	specific heat of tank-side flow loop
CPC	Compound Parabolic Collector
F_R	Collector heat removal factor
$F^{'}$	collector plate efficiency factor
HVAC	Heating, Ventilation, and Air-Conditioning
I_T	Global radiation incident on collector (tilted surface)
I_{bT}	Direct Radiation on a tilted surface

$K_{\tau\alpha}$	Incidence angle modifier
NP	number of people in living unit
NSR	number of sleeping rooms
NTU	Number of heat Transfer Units
M	time of maximum growth;
m_{opt}	optimal mass flow rate
$m_{t,opt}$	tank-side optimal mass flow rate
m_c	collector-side mass flow rate
m_t	tank-side mass flow rate
R^2	square of the multiple correlation coefficient
R_{opt}	optimal heat capacity ratio
RSR	Relative Solar Rating
SF	Solar Fraction
SRCC	Solar Rating and Certification Corporation
SDHW	Solar Domestic Hot Water
SEF	Solar Energy Factor
TRNSYS	TRaNsient SYstems Simulation program

T_i	inlet water temperature
T_a	ambient temperature
U_L	collector overall heat loss coefficient
$(UA)_x$	overall heat conductance of heat exchanger
UA	Overall heat conductance
$U_{L/T}$	Heat loss coefficient dependency on T
Y_{actual}	Solar Fraction found by TRNSYS
$Y_{estimated}$	Solar Fraction found by multiple variable regression

Greek Symbols

ρ	reflectance
ρ_g	ground reflectance
$(\tau\alpha)$	Product of the cover transmittance and the absorber absorptance
$(\tau\alpha)_b$	$(\tau\alpha)$ for beam radiation
$(\tau\alpha)_n$	$(\tau\alpha)$ for normal radiation
$(\tau\alpha)_g$	$(\tau\alpha)$ for ground reflected radiation
β	Collector slope above the horizontal plane

η	solar thermal collector efficiency

Subscripts and superscripts

a	ambient
b	beam
c	collector
g	ground
i	incident, inlet
l	loss
n	normal
opt	optimal
p	plate
r	radiation
T	tilted, temperature

CHAPTER I

PRINCIPAL BASIS OF RESEARCH

I.1 OVERVIEW

I.1.1 CURRENT FOSSIL FUEL USE

Our way of life and economy in the United States is heavily dependent on fossil fuels. Along with the positive that brings we are seeing the negative impacts as well. There are environmental damages such as global warming, which is happening at an alarming rate because of the increase of CO_2 in the atmosphere. There are other impacts as well such as increased risks as the fossil fuel sources are becoming scarcer. For instance, extraction processes themselves are creating greater environmental damage per fuel unit obtained. There are also increased pressures among societies around the world as each is vying to establish energy for their needs. To complicate the situation, worldwide there is an increase in demand as population grows.

Continuing the rate of fossil fuel consumption that is happening right now will only leads to more and more problems that will be detrimental to our world as we know it. Attention has to be given to remedy these problems. In this context, there are basically two ways to lower fossil fuel use: (i) by energy-efficiency and (ii) by using renewable energy sources, see Figure I.1.

Energy-efficiency is usually regarded as an individual product or process efficiency. Yet, there are other types of energy efficiency such as effective design and operation. An effective design that would also be a form of energy efficiency is an efficient lighting layout design in a building for example. Effective operation of a power plant could also save energy and thereby also be energy efficient. And yet another type of energy efficiency is called load management in the utility industry. Load management covers a vast array of options such as load shifting or time-of-use rates in order to optimize customers' energy use with the most efficient operation of the utilities' power generation.

Whereas energy efficiency reduces the total amount of energy needed, and therefore fossil fuel, required to accomplish a task; renewable energy supplies what energy is needed from a sustainable source. Common commercially available renewable energy technology includes wind power, solar thermal power, photovoltaics, geothermal, and micro-hydro power. Sometimes matters of scale are the difference between what would be called a renewable energy source or not. Micro-hydro for instance, is considered renewable, unlike much larger water-power dams, because they are designed to take a sustainable amount of water from their source. Other energy sources, such as solar thermal can be designed as major power generation plants using parabolic trough technology. Solar thermal power can also be used on a smaller scale such as for solar domestic hot water systems. This particular type of power generation is also known as a distributed generation technology; which signifies generating the power at the site of the demand.

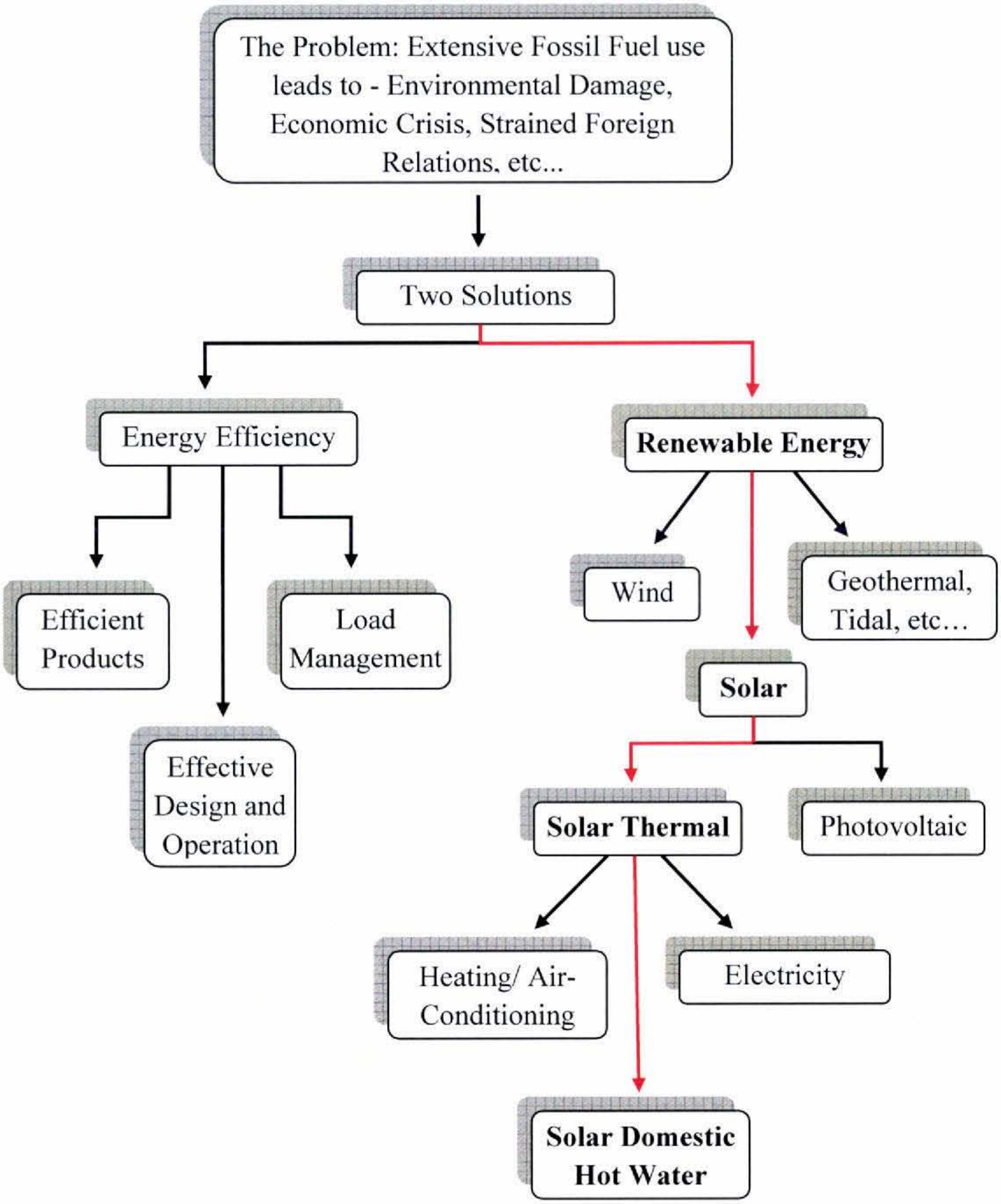


Figure I.1 Energy Efficiency and Renewable Energy are two solutions to lowering fossil fuel use in the US.

I.1.2 Residential Energy Use

Residential energy use in the US is almost ¼ of the total annual energy consumption (22% of 100,368.6 trillion btus including residential, commercial, industrial, transportation categories) (EIA 2005). Currently, there are a few different ways that energy use is regulated with in the residential sector. One way is with building codes. Building codes are used by states to set a minimum requirement for energy efficiency levels for different building energy demands. For the majority of the states this is ASHRAE 90.2 for residential buildings, if there is one even used, and ASHRAE 90.1 for commercial buildings (ASHRAE 2004-1, ASHRAE 2004-2). Another way energy can be regulated is by governmental mandates on individual products. There are also voluntary programs and benchmarking standards that can be followed including ENERGY STAR® and LEED®. Limits on energy use for some types of products such as refrigerators can be set independent of demand characteristics including load size or location. Other products like windows have efficiency criteria that are dependent on local climatic variables.

I.1.3 Domestic Hot Water (DHW)

Domestic hot water (DHW) is second highest single consumption in residential energy use after space conditioning per household in the US regardless of energy source (EIA 2001-1). In Hawaii it is the highest energy use if there is no air conditioning used in the home. Thus, anywhere from 35% to 50% of an average family of four's total household electricity demand is for DHW. This is roughly 3,500 kWh/year or $720year at 20.6 cents per kWh, which is the residential price on Oahu (HEI 2006).

Roughly 39% of all domestic hot water is produced from electricity (EIA 2001-2). This process requires usually some type of fossil fuel to be consumed in a steam-electric generating power plant of 35% efficient, meaning the other 65% of the available energy is turned into waste heat (EIA 2000). The generated electricity, which is a high-exergy form of energy, is then moved far distances with 7% additional energy losses in the grid (Lovins et al. 2002) to then simply be turned back into heat, a lower form of exergy, by the electric resistor element in the hot water tank.

Gas water heaters are more efficient, 65%, as they eliminate the power plant and grid losses by just combusting the fuel to create heat and thus do not need to transform the fuel energy into electricity (US DOE 2006). Yet, even though the production of the heat is more efficient for gas heaters, the source of fuel is still a non-renewable source. Another option is to use solar power for the energy source, which is going to be available onsite anyway.

I.1.4 Solar Thermal Energy for DHW

Solar Domestic Hot Water (SDHW) systems are considered distributed generation, because the energy source and production are located at the same site as the demand or need. They have significant potential to lower residential energy use. In Hawaii, for example, SDHW systems can produce essentially 100% of domestic hot water demands.

Currently some utilities offer monetary rebates for using SDHW systems (NCSC 2008). There are federal and state tax credits available as well (NCSC 2008). Yet, there are no standards or codes that include SDHW. Even voluntary programs simply give credit for the use of these systems, but do not detail any requirements or levels that should be met. There are two basic reasons for this. The first reason being that there are a number of different types of solar thermal technology as well as system configurations, with each one working on separate set of principles. Thus, to define appropriate regulating criteria one would have to be able to take into account all of these variables of the system. This is much more complex than defining efficiency levels for an electrical resistor element water heater. The second reason is that SDHW systems' performance is dependent on location and demand characteristics. Therefore, regulating language would also have to be acceptable across a variety of conditions.

Incorporating SDHW systems in residential buildings is already becoming the next big energy saver on the forefront. As of April 1, 2008, ENERGY STAR® passed new voluntary labeling levels for the following DHW technologies; Gas Storage Tanks, Gas Tankless, Heat Pumps, and SDHW (US EPA & US DOE 2008). The criteria for the SDHW system would use the Solar Rating and Certification Corporation's (SRCC) OG-300 Solar Water Heating Systems Certification.

The SRCC is a non-profit organization that was established to conduct a rating program for solar collectors, the OG-100, as well as solar water or air heating systems, the OG-300 (SRCC 2008). The nationally recognized OG-100 program evaluates collectors for their maintainability as well as gives each collector a thermal performance rating based on a series of tests performed under a specified set of conditions. The OG-

300 program gives solar water or air heating systems a performance rating based on the OG-100 collector ratings as well as other system tests.

The SRCC-OG-300 systems certification is based on a ratio known as the Solar Fraction (SF). The Solar Fraction is equal to the portion of the conventional hot water heating load (including losses) provided by solar energy (SRCC 2008). The ENERGY STAR® criterion is to require the eligible SDHW systems to equal a 0.5 Solar Fraction rating or higher. The SRCC-OG-300 systems certification is based on the calculation of the Solar Energy Factor (SEF) to calculate the given SF rating.

There has been some concern expressed about using the SRCC rating in this manner. The ACEEE has stated that one fault in the offered Energy Star criterion is the inability to take into account the actual performance variance of SDHW systems due to different climate zones (ACEEE 2007). The SRCC expressed disagreement with the use of the OG-300 SEF to determine the Solar Fraction (SEIA & SRCC 2007). This is again because of the SEF calculation being based on one climate profile and not showing the difference in performance among real locations. The SRCC recommends that Energy Star should consider criterion that is climate based but as uncomplicated as possible.

Other possible concerns with the SRCC SEF Ratings arise from the SRCC-OG-300 documentation (SRCC 2008). First, the different systems are run at different flow rates with a default flow rate (kg/hr-m2) being specified for any system where the flow rate is not given by the manufacture. As will be discussed later flow rates highly affect system performance. Second, the SRCC SEF Ratings are done using different collector areas between systems. Thus, a system with a much larger collector area could end up with a higher SEF rating as compared to a more efficient system with that has a much smaller collector area. Third, the SRCC SEF Ratings are for one type of load scenario, which is sized around an average household. Therefore, the resulting SEF rating shows the ability of a given system to meet that load scenario and it could be an over or under rating for another type of load scenario.

I.2 Statement of Research

The focus of this research is to find what factors affect a relative performance comparison among different SDHW systems in order to define appropriate regulating criteria. Instead of either comparing SDHW systems to find the most optimal one or performing a parametric study on the effects of certain variables, this research combines those parts together to define a methodology that can unbiasedly show the optimal SDHW system for a range of conditions. There are two important yet slightly different ways this type of regulating criteria can be used. The first being to promote higher levels of product efficiency, such as with an Energy Star rating or federal minimum product efficiency ratings. Thus, procedures must be able to rate systems where the relative merit can be judged accurately between them. The second way is to promote SDHW system's incorporation into building standards. This means that procedures must use criterion that is sensitive to changing load and climate conditions.

This research incorporates a numerical study using TRNSYS software to quantify SDHW Systems' dependency on demand characteristics, location, and configuration. The following four loads are analysis; a 302 kg/day load at a set point temperature of 48.89°C, a 227 kg/day load at 60°C, a 151 kg/day load at 48.9°C, and a 3,028 kg/day load at 48.89°C. Two different flat plate panels, one compound parabolic concentrator (CPC) panel, and two evacuated tube panels' characteristics are inputted as both direct and indirect system configurations into the analysis. Yearly evaluations for each system configuration are done for fourteen climatically different locations in the United States. This is a total of 560 different scenarios which are then correlated to define appropriate regulating language. Existing literature and documented variable characteristics such as for tank size and flow rate have been taken into account whenever possible. An extended evaluation has been applied to optimize flow rates using the GenOpt program available through the Lawrence Berkley National Laboratory.

I.3 LITERATURE REVIEW

With SDHW systems being a mature technology, many studies have been done to quantify their performance, but not necessarily how to compare performance among systems. With respect to evaluating performance, there are two main groups of studies. There are the ones that try to optimize system configuration or components (Carvalho 1988, AL-Ibrahim 1998, Abdel-Dayem 1999, Rosengarten et al. 1999). Then, there are the parametric studies that evaluate the effects of certain variables to the performance of the system (Christensen et al. 2000, Kenna 1984-1, Kenna 1984-2, Morrison 1986, Frei et al. 2000). Both of these types of studies are very important in developing our knowledge of how SDHW systems work. However, their results do not provide a non-biased way to promote the technology's efficiency.

There are even whole books written on how to design and build SDHW systems for different climates as well as discuss specifics of hardware details (Ramlow & Nusz 2006, Steven Winter Associates 1997, Ramsey 2002, Kemp 2006, Clive 2007). These again fill an important niche in the field by providing consumers with the information needed to decide what SDHW system would be optimal for their specific need. These types of design details should be left up to the consumers and solar industry to choose freely in order to support technology advancement as well as the free market. Thus, a methodology for rating SDHW systems should be able to compare technology fairly yet independent of these details.

Other documentation predicts how SDHW systems will perform, including literature on analyses completed to validate the current SRCC rating procedure (Klein et al. 1983, Minnerly et al. 1991, Davidson et al. 1992, Davidson et al. 1993). A study done by Cuadros et. al. describes a similar approach in the sense that they tried to fit their data to an equation as a numerical fit (Cuadros et al. 2007). Yet, again their results are for predicting SDHW system performance. Their results are also not consistent over collector area size as well as the calculation is climate specific. Brinkworth shows numerical work using dimensionless groups to create correlation curves, thus illustrating the sensitivity of the SDHW system to the many affecting variables (Brinkworth 2001). Again, though this study is useful in predicting system performance, solely it could not be used as a way to regulate collector performance appropriately.

Morrison and Tran proposed two standardized rating methods of hot water heaters in Australia for energy efficiency and CO_2 pollution (Morrison & Tran 1992). These two rating methods would be used to compare electric, gas, solar, and heat pump water heaters with the same method. They state that the current rating scheme for electric and gas water heaters cannot be applied to solar or heat pump technology as the later is affected by more factors that are not taken into account. Their approach, like the present study used TRNSYS calculations to perform the analysis. Even though a variety of solar technology was evaluated in the study (flat plate, concentrating, and evacuated tube solar collectors), their results on performance are just a generalized group of solar technology as compared to the other categories; gas, electric, and heat pump. Both rating methods show, as is expected, that solar technologies uses less grid or gas produced energy as well as produce less CO_2 as compared to the others. Yet, Morrison and Tran's work did not give a rating method to compare different solar technology's performance, or in any way suggest how select optimal SDHW systems for different locations or demand scenarios.

Thus, by far the majority of documented literature on SDHW systems is geared towards evaluating or predicting performance and not so much towards finding appropriate ways to regulate the technology to promote optimized systems. These studies are crucial though in defining a methodology that can accomplish the task of this research. Thus, any relevant studies are taken into account whenever possible to avoid repetition.

CHAPTER II
METHODOLOGY

In this chapter, a detailed discussion on the numerical programs used to simulate SDHW systems as well as the inputs and outputs parameters is carried out. The chapter is divided in two main sections: (i) thermal energy modeling, and (ii), modeling inputs and outputs. TRNSYS (TESS 2007) a commercial thermal energy modeling program, was used to simulate the performance of SDHW systems for a range of different inputs. Certain inputs were optimized using GenOPT® software, which can be downloaded for free online (LBNL 2004). The data obtained with TRNSYS and GenOpt® was then numerically correlated using non-linear regression analysis as well as analyzed statistically. The correlated data is later used in Chapter 4 to show which parameters need to be accounted for when defining regulating criteria for SDHW systems, as well as show how to account for them. The modeling programs and inputs used in the programs are discussed below.

II.1 Modeling Programs

II.1.1 TRNSYS

TRNSYS, which stands for a TRaNsient SYstems Simulation program was chosen to do the modeling of the SDHW systems (TESS 2007). It was originally created by the University of Wisconsin's Solar Energy Laboratory in the 1970s. TRNSYS has become the reference software for researchers and engineers around the world as it is considered the standard for solar thermal energy analysis. Different from most commercial packages, TRNSYS offers a wide range of built in libraries for the thermal simulation of renewable energy related processes. TRNSYS is also assisted by a graphical user interface (GUI) picturing all the systems' components. Its main applications include; solar systems (solar thermal and photovoltaic systems), low energy building design including HVAC systems, cogeneration, and fuel cells. TRNSYS is also the program used for all SRCC system rating analysis. This fact made it an obvious fit for this research.

There was extensive research conducted to verify the use of TRNSYS for SRCC's system rating. It was found to be within +- 5% accurate when modeling the same system configuration as tested by the ASHRAE 95 standard test procedure (Minnerly et al. 1991). When the system configuration was changed by collector area, flow rate, and storage tank volume or design from the ASHRAE tested configuration, TRNSYS was found to be within +- 10% accurate (Davidson et al. 1993).

Additionally, there has been other experimental validation of the program independent of the SRCC rating program. Some studies focus on particular systems. Mason et al. found TRNSYS to be within 15% accurate when modeling evacuated tube integral collector-storage SDHW systems (Mason et al. 1995). Braun et al. found TRNSYS to be within 3% accurate when modeling thermosiphon SDHW systems (Braun et al. YEAR). Morrison et al. also validated TRNSYS with experimental data specifically looking at thermosiphon SDHW systems and found it to be within +-10% accurate (Morrison et al. 1985). Thermosyphon systems were again modeled by Kalogirou et al. in 2000 to find TRNSYS capable of showing a mean deviation of 4.7% when compared to experimental data (Kalogirou et al. 2000). Other studies have looked at validating particular component models available in the TRNSYS program. Kleinbach et al. validated the appropriate use of three different types of storage tank models (Kleinbach et al. 1993).

II.1.2 GenOpt

GenOpt is an optimization program used to find the minimization of a cost function that is evaluated by external program (LBNL 2004). Its main application field is building energy use or operation cost optimization. It is the result of PhD work done by Michael Wetter in 2004 being funded by Lawrence Berkeley National Laboratory. Generalized Pattern Search algorithms (the Hooke-Jeeves and the Coordinate Search algorithm) as well as Particle Swarm Optimization algorithms (for continuous and/or discrete independent variables) are available optimization techniques in the program. A hybrid global optimization algorithm using Particle Swarm Optimization for the global optimization and Hooke-Jeeves for the local optimization is another optimization technique as well. Other techniques in the program include the Discrete Armijo Gradient algorithm, Nelder and Mead's Simplex algorithm, and Golden Section and Fibonacci algorithms.

TRNSYS has a component available in its library that allows the user to connect to the GenOpt Program from the TRNSYS GUI. At first this method was used, but once extensive numbers of runs needed to be performed, a short code was written to execute GenOpt from the DOS command prompt in batch mode. This made it possible to run the optimization method on numerous scenarios automatically by instructing the program to save the data to specific file locations. The Hooke-Jeeves algorithm was found to be capable of converging at the appropriate global minimization during the validation stage, thus it was the method used for all GenOpt analysis.

The TRNSYS simulations take less than two minutes to run an annual simulation for any particular scenario. GenOpt, on the other hand, takes over 15 minutes to converge for direct system scenarios in which only one flow rate has to be found and takes over an hour to converge for indirect system scenarios in which two different dependent flow rates are to be optimized. The main computer used for computation is a Intel® Core ™ 2 CPU 6400 @ 2.13GHz, with 2.00 GB of RAM running Microsoft Windows XP Professional. Six additional computers were used to run simulations as well as. They all used Microsoft Windows XP Professional, have 1 GB of RAM and Intel® Pentium 4 CPUs @ 3.20 GHz.

II.2 System Inputs

The components involved in the operation of SDHW system can be broken down into demand-side and supply-side inputs, see Figure II.1. Demand-side refers to the inputs having to do with the environmental conditions the SDHW is operating in as well as the characteristics of the actual demand required to fulfill. The demand-side inputs are:

- the location/ climate characteristics
- load size and temperature
- incoming water temperature
- load profile.

Supply-side refers to all inputs dealing with the actual SDHW system itself; that being the total system of energy generation. The supply-side inputs are:

- the collector type and size
- configuration
- tilt angle
- tank size
- auxiliary heating
- heat exchanger size
- heat exchanger location
- flow rates
- pumps
- piping

Each input is described in depth below.

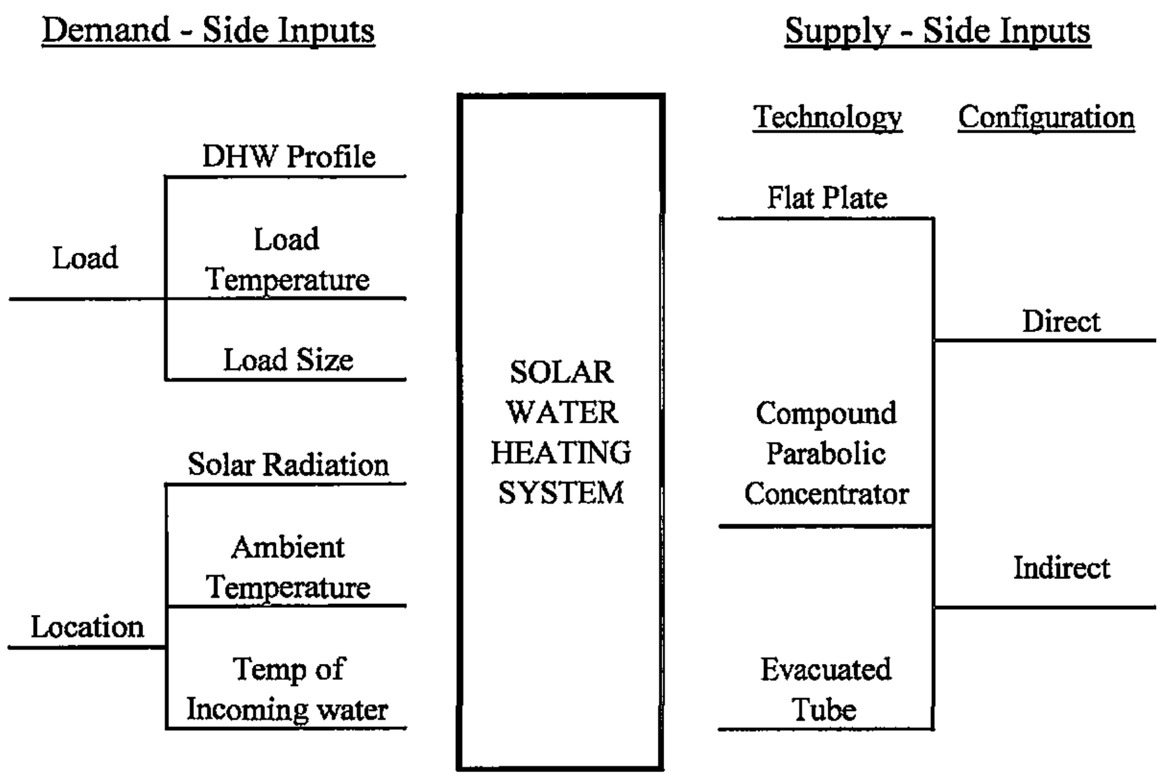


Figure II.1 Inputs of SDHW Systems

II.2.1 DEMAND SIDE CHARACTERISTICS

II.2.1.1 Location Data

As expected, the location of the SDHW system dictates the annual solar radiation available as well as ambient temperatures throughout the year. The system's performance is obviously affected by the amount of solar energy available. But, perhaps not as obvious is the fact that ambient temperature is an important factor as well since a thermal energy system operates on the difference between temperatures. Therefore, the surrounding temperature will affect the rate of heat loss from the system to the ambient and must be taken into account. A third component of climate data that is a factor in the performance of SDHW systems is called the clearness index. This number represents how cloudy a location is as a ratio. Cloudy or clear conditions will affect what type of collector optics is favorable to use to convert sunlight into useful heat energy. For instance a cloudy sky will have more diffuse solar radiation whereas a clear sky will be comprised of more direct solar radiation. The type of radiation affects what angle the solar hits the collector plate, and therefore what optics are desired for the absorption of the solar energy.

In order to collect enough data that would represent different combinations of annual solar irradiation, ambient temperatures, and clearness index, 14 different locations were carefully selected and are used in the models, see Table II.1. The selection criteria for selecting the locations was loosely based on ASHRAE's climate designation for the United State with at least one location to represent each of the first 7 ASHRAE climate categories.

ASHRAE Climate Zone	Location	Average Solar Radiation kWh/m2/day	Average Annual Clearness Index
1a	Miami, FL	5.2	0.53
1a	Honolulu, HI	5.7	0.57
2a	Houston, TX	4.8	0.50
2b	San Antonio, TX	5.4	0.55
3b	Daggett, CA	6.6	0.68
4a	Asheville, NC	4.9	0.51
4a	Wichita, KS	5.2	0.55
4b	Albuquerque, NM	6.4	0.66
4c	Arcata, CA	4.4	0.49
5a	Youngstown, OH	3.9	0.46
6a	Massena, NY	4.3	0.50
6b	Helena, MT	4.7	0.54
7a	Caribou, ME	4.2	0.49
7a	Minot, ND	4.7	0.54

Table II.1 Annual Average Solar and Clearness Index Values for each Location

II.2.1.2 Load Size and Temperature

Four different load scenarios were applied to the model. The first is a residential scenario based off of an average household of four. For this scenario the load temperature is set to 48.9°C (120°F) which is the maximum allowable for showers and tubs as set by the Uniform Plumbing Code^{TM} (IAPMO 2006) and International Plumbing Code® (ICC® 2000). The load size is 302 liters/day (80gal/day) based on a use rate of 75.5 liters (20 gals) per person. This is the upper end of domestic hot water use as documented for the United States (Sikora & Wiehagen 2003) so was taken to be the upper requirement for a residential home. The second load scenario uses a daily load size 151 liters (40 gals) at a set point temperature of 48.9°C (120°F). This scenario is to represent the lower end of domestic hot water use in an average household or to represent a household of 2 persons. The third load scenario uses a set point temperature of 60°C (140°F) and a daily load size of 227 liters (60 gals). This scenario models roughly the same overall energy requirements at the 302 liters/ day scenario but at a higher temperature to separate out the possible effects of varying the set-point temperature. The fourth scenario considers a daily load size of 3,028 liters (800 gals) at a set point temperature of 48.9°C (120°F). This is to simulate a multi-residence building of roughly 40 occupants.

II.2.1.3 Incoming Water Temperature

As stated before, SDHW systems are thermal energy systems and thus overall efficiency is dependent on temperature differences within the system as well as between the system and surrounding environment. Thus, accurate modeling of the incoming water temperature is critical to simulating performance. Incoming water temperature is dependent on the source of the water. The source can be from a city main, private well, or a local spring. ASHRAE 90.2 lists temperatures to use for their calculations that change based on location but not for time of year (ASHRAE 2004-2). Yet realistically, no matter what the source of incoming water, city main, private well, or local spring, the temperature will change throughout the year based on ambient temperature and thus ground temperatures as well.

Fourteen individual monthly temperature profiles were created for the fourteen locations based on monthly average ambient temperature of each location. As also with ASHRAE 90.2, in each of these sets of temperature profiles no incoming temperature is below 5°C or above 24°C (ASHRAE 2004-2). In Appendix VI.1 the incoming water temperature profiles are shown for each location modeled. This method also draws a parallel with current research evaluating the governing equations of incoming water temperatures (Burch & Christensen 2007).

II.2.1.4 Load Profile

Because the effect of the daily water draw profile has been shown to highly influence the storage tank performance (Morrison et al. 1992, Spur et al. 2006, Buckles et al. 1980, Saltiel et al. 1985), the use of a realistic load profile in the modeling was important to ensure unbasised results. Take for instance, a profile consisting of 100% of the daily load drawn at 7pm after the sun has been heating the water in the tank all day will require less energy than a profile consisting of 100% of the daily load drawn at 7am. This is because there will be heat losses as the water sits in the tank overnight. Thus it is important to use an accurate load profile which matches the realistic time requirements of a residential water heater. The hour ratios used to model the load profile in all simulations is taken from the ASHRAE 90.2 Standard. Independent reviewers have compared the load profile used in ASHRAE 90.2 to alternate profiles finding that it fits with real data very well (Fairey et al. 2004). The SRCC system testing currently uses a load profile inconsistent with realistic domestic water draw (SRCC 2008). Both the ASHRAE 90.2 and SRCC load profiles are given in Appendix VI.2.

II.2.2 SUPPLY SIDE CHARACTERISTICS

II.2.2.1 Collector Type and Size

In order to implement a solar collector model, one must account for the optics as well as heat transfer processes involved in each collector type. The first and arguable the most important equation to be outlined, is the general form of the overall efficiency product for a solar collector as defined by Duffie and Beckman for any type of solar thermal collector (Duffie & Beckman 2006).

$$\eta = F_R(\tau\alpha)_n - F_R U_L \frac{(T_i - T_a)}{I_T} - F_R U_{L/T} \frac{(T_i - T_a)^2}{I_T} \tag{II.1}$$

Where: F_R = collector heat removal factor

$\tau\alpha$ = transmittance-absorptance product

U_L = collector overall heat loss coefficient

T_i = inlet temperature

T_a = ambient temperature

I_T = Global radiation incident on collector (tilted surface)

The first term, $F_R(\tau\alpha)_n$, accounts for how solar energy is absorbed by the solar collector. As can be seen, F_R, which is the ratio of the actual useful energy gain by the maximum possible energy gain, is affected by $\tau\alpha$, which is evaluated normal to the collector plane (Duffie & Beckman 2006). The transmittance-absorptance product is a combination of the transmission and absorption properties of a solar collector's cover. In the second and third terms F_R is affected by U_L, the collector overall heat loss coefficient. Thus, the first term evaluates the heat gain of the collector minus the second and third terms which evaluate the heat losses to the surrounds.

Equation 2.1 can be rewritten as Equation 2.2 to symbolize the collector's performance by three coefficients as is used in the ASHRAE 93-2003 test standard (ASHRAE 2003).

$$\eta = a_0 - a_1 \frac{(\Delta T)}{I_T} - a_2 \frac{(\Delta T)^2}{I_T} \tag{II.2}$$

Where: $a_0 = F_R(\tau\alpha)_n$

$a_1 = F_R U_L$

$a_2 = F_R U_{L/T}$

Furthermore, by introducing the "incidence angle modifier" (IAM), also known as $K_{\tau\alpha}$, which represents the ratio of the transmittance-absorptance product at some angle to the transmittance-absorptance product at normal incidence (Duffie & Beckman 2006), one can correct the first term in Equation 2.1 for when the solar incidence angle is not normal to the collector plane. The IAM is found using the following equation based on knowing the transmittance-absorptance product normal to the solar collector (Duffie & Beckman 2006).

$$K_{\tau\alpha} = \frac{(\tau\alpha)}{(\tau\alpha)_n} = \frac{I_{bT}\frac{(\tau\alpha)_b}{(\tau\alpha)_n} + I_d\left(\frac{1-\cos\beta}{2}\right)\frac{(\tau\alpha)_d}{(\tau\alpha)_n} + \rho_g I\left(\frac{1-\cos\beta}{2}\right)\frac{(\tau\alpha)_g}{(\tau\alpha)_n}}{I_T} \quad \text{(II.3)}$$

Equation 2.3 can also be rewritten as Equation 2.4 to follow the ASHRAE 93-2003 test notation (ASHRAE 2003). Flat plate collector covers have optically symmetrical characteristics for both axes thus Equation 2.4 only has to be found once. Yet, optically nonsymmetrical collectors, such as evacuated tube and some parabolic concentrator collectors require the use of biaxial IAMs (McIntire et al. 1983). Thus, a separate calculation is done to find the IAMs' dependence for both the transverse (perpendicular) plane and longitudinal (parallel) plane.

$$\frac{(\tau\alpha)_b}{(\tau\alpha)_n} = 1 - b_o\left(\frac{1}{\cos\theta} - 1\right) - b_1\left(\frac{1}{\cos\theta} - 1\right)^2 \quad \text{(II.4)}$$

Where: $(\tau\alpha)_b$ = $(\tau\alpha)$ for beam radiation

$(\tau\alpha)_n$ = $(\tau\alpha)$ at normal radiation

θ = Incidence angle for beam radiation

The coefficients, a_0, a_1, a_2, b_0, and b_1 are input into the TRNSYS solar collectors models, which in turn uses equations 2.2 and 2.4 as part of calculating the simulation of a particular solar collector's performance for a given scenario. The coefficients for each type are taken from published values by the SRCC and the coefficients for the governing equations are shown in Tables II.2 and II.3 (SRCC 2008).

		Flat Plate A	Flat Plate B
Efficiency	a_0	0.784	0.612
	a_1	4.2805	4.3317
	a_2	0.0048	0.0206
IAM	b_0	0.2947	0.0507
	b_1	0.0119	0.1253

Table II.2 SRCC's Coefficients for 2 Flat Plate Collectors

		Compound Parabolic Concentrator	Evacuated Tube A	Evacuated Tube B
Efficiency	a_0	0.591	0.525	0.416
	a_1	4.5502	0.886	0.9646
	a_2	0.0189	0.0074	0.0023
IAM	Perpendicular			
	b_0	0.6317	0.1441	1.1718
	b_1	1.2396	0.0948	0.847
	Parallel			
	b_0	0.016	0.28	0.13

Table II.3 SRCC's Coefficients for CPC and 2 Evacuated Tube Collectors

Flat Plate panels, which are the most common type for SDHW systems, use both beam and diffuse radiation. Beam radiation, also known as direct has not been scattered by dust or water droplets in the atmosphere. Diffuse radiation has had its direction changed by elements it encountered through the atmosphere. The collectors have an absorbing surface enclosed under an envelope that is transparent to solar radiation but reduces convection and radiation losses to the atmosphere. The other three sides are insulated (Duffie & Beckman 2006). CPC panels, which stands for compound parabolic concentrator, is a particular type of concentrating collector that can utilize both beam and diffuse radiation with no tracking needed. Each side of the trough is a parabola focusing all incoming radiation onto the receiver tube in the middle. CPCs will usually be enclosed under a transparent envelope like flat plate panels, yet they can reach higher temperatures (Duffie & Beckman 2006). They are also insulated on all three sides. Evacuated tube panels have an evacuated space between the receiver tube and the outside envelope, therefore essentially eliminating convective heat losses to the ambient. Because of this design they have no insulated sides and thus can receive light from any direction. Evacuated tubes panels can also reach higher temperatures than a flat plate. In this study two different flat plate collectors, one CPC collector and two evacuated tube collectors are evaluated across all the other parameters.

TRNSYS offers quite a few different solar collector models in its library. In this study both the 537-TYPE Flat Plate model and 71-TYPE Evacuated Tube TRNSYS components were used to model flat plate and CPC/ evacuated tube collectors respectively. Yet, both TRNSYS components ,537-TYPE and 71-TYPE, compute only the heat balance and are zero capacitance calculations. This means they do not take into account the mass of the heat transfer fluid in the collectors in the energy balance of the equations. This created an abnormally high fluctuation in the outlet temperatures of the collectors. Thus, a pipe component, type31, was added downstream from the collector to account for the mass of fluid in the collector as recommended by TRNSYS helpdesk therefore, creating more accurate outlet temperatures.

Another important characteristic of the solar collector itself is the available area. As area is increased, usable solar heat gain decreases per square meter. This can be seen with most any in-depth study on solar thermal systems (Duffie & Beckman 2006). There is always an increase of solar energy gains as well as an increase of thermal losses to the surrounding environment with increasing area. This leads to a function of diminishing returns and it will become non beneficial to continue increasing area for a particular load size. Thus, in the present study, the area has been varied from 1-8m2 for both the 302 liters/day and 227liters/day load, from 1-5m2 for the 151liters/day, and from 10-80m2 for the 3,028liters/day load.

II.2.2.2 Configuration

SDHW systems can be set up with different configurations including the following. In general SDHW systems can be direct or indirect configurations. Direct systems (Figure II.2a) are open loop systems in which the working fluid is also the potable water being used. Indirect systems (Figure II.2b) incorporate a heat exchanger between the working fluid which is usually a mix of propylene and potable water. The advantage to using direct systems is that there are no energy losses due to using a heat exchanger such as in indirect systems. Yet, direct systems for the most part are unable or have limited operation in freezing climates.

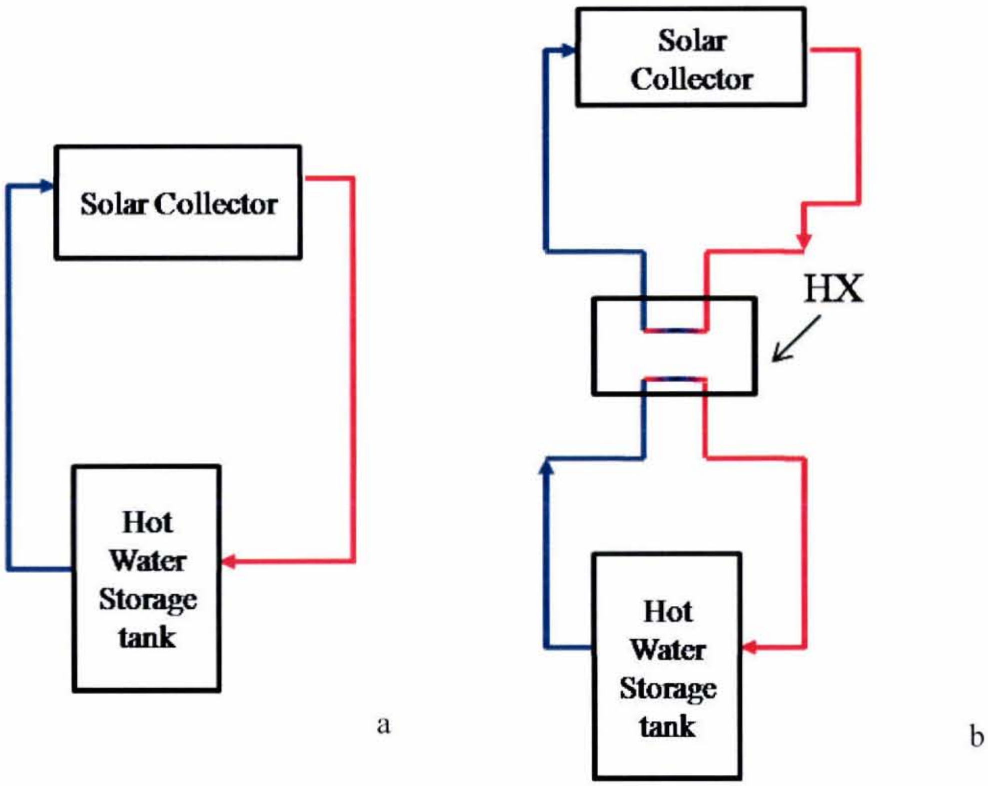


Figure II.2 Direct (a) and Indirect (b) System Configurations

SDHW system configurations also can vary in their overall structure of the fluid flow loop. One type called a thermosyphon system, uses fluid that is circulated by natural convection to the storage tank, which is elevated above the collectors. This can be either a direct or indirect system. Integral Collector Storages systems have the storage tank and collectors as one body (see Christensen et alt. 2000, or Mason & Davidson 1995 for more detail). In this case, there can be large energy losses during the night because the storage tank is located outside and one side of it, i.e. the side that faces the sky, is not insulated. These are usually in a direct system configuration. Drain-back systems refers to an unpressurized system that is separated from the pressurized water used in the house in order to allow the fluid to drain back down when the pump is not in operation. This allows the fluid to be drained during freezing conditions (Dontje 2007). Drain-down also called drain-out systems refer to a completely pressurized direct system using potable water. The water is drained out of the system to waste as a form of freeze protection. Drain-down system configurations are the type of direct system used in this study. The indirect system configuration used in this study is a closed loop system consisting of two separate flow loops connected by one heat exchanger, with both flow loops pressurized with a pump on each side of the heat exchanger.

In this study both a direct system and indirect system were modeled. The direct systems were modeled to operate only during the part of the year when the average daily minimum temperature as taken from NSRDB data (NREL 1994) was above 0°C. This is a seasonal configuration. So, direct systems in ASHRAE climate zones 1, 2, and 3 were modeled to operate year round where as in the other climates operation is seasonal. This also correlates with literature on the probability of solar water heating pipes freezing (Salasovich 2002). The indirect system systems modeled in this study were all design to use 50/50 propylene glycol/ water mix on the collector side of the heat exchanger. Thus every location operates year round. The summary below shows the use of direct SDHW systems for all 14 locations considered:

- 1a Miami FL (25.80°N, 80.27°W)– year round
- 1b Honolulu HI (21.833°N, 157.92°W) – year round
- 2a Houston TX (29.98°N, 95.37°W) – year round
- 2b San Antonio TX (29.53°N, 98.47°W) – year round

- 3b Daggett CA (34.87°N, 116.78°W) – year round
- 4a Ashville NC (35.43°N, 82.53°W) – March to November
- 4a Wichita, KS (37.65°N, 97.42°W) – March to November
- 4b Albuquerque, NM (35.05°N, 106.62°W) – March to November
- 4c Arcata CA (40.98°N, 124.10°W) – year round
- 5a Youngstown, OH (41.27°N, 80.67°W) – March to November
- 6a Massena, NY (44.93°N, 74.85°W) – April to October
- 6b Helena, MT (46.60°N, 112.00°W) – May to September
- 7a Caribou, ME (46.87°N, 68.02°W) – May to October
- 7b Minot, ND (48.27°N, 101.28°W) – May to October

II.2.2.3 Tilt

For this study the tilt of the collector panel is based on each location's latitude. Christianson et al. 2001 showed that for solar thermal collectors (not photovoltaic) in any location; setting the tilt equal to the latitude year round for the collectors will not result in a large loss in energy gain as compared to changing the angle over the course of the year (Christianson et al. 2001). Thus, tilt once set to the latitude, is independent of the location, i.e. to not track in location A vs. location B will equally effect both places. Therefore it is appropriate for this study which is of relative comparisons, to use a set tilt based on the locations latitude.

II.2.2.4 Flow Rate

The flow rate for solar thermal systems has been documented to be one of the most important factors affecting system performance. It is known that there is an optimal flow rate for a given solar thermal system as can be seen in the work completed by Wuestling et al. (Wuestling et al. 1985). Their research suggests the optimal flow rate is based on the average daily load demand of the system. Others have also researched the effectiveness of using low-flow flow rates in SDHW systems (Fanney et al. 1988). This is based on showing that the effectiveness of the storage tank to store hot water increases as stratification of the tank is increased via slower flow rates. Furbo et al. compared using a single flow rate to using a variable flow rate as based on fluid inlet temperature, finding small increases in effectiveness (Furbo et al. 1996). AL-Ibrahim et al. have conducted extensive research on variable speed solar powered pumping schemes (AL-Ibrahim et al. 1998). Hollands and Brunger have shown that there are optimal flow rates for both sides of a heat exchanger in a closed loop SDHW system (Hollands & Brunger 1992).

Based on the above, all simulations in this analysis are modeled running each scenario at its optimized flow rate. The GenOpt program was initially used to find both the direct system and indirect systems optimal flow rates. This data was then used to validate the accuracy of Hollands and Brunger's optimal flow rate equations. The correlations found by Hollands and Brunger (Hollands & Brunger 1992) were for SDHW systems studied by Wuestling et al. (Wuestling et al. 1985) who does not explicitly state the type of systems were analyzed. Thus, it was unknown if these correlations would be valid for the three different types of collectors studied in this research as well as for other study limits such as the maximum collector area the equations could still be used. The constants used in the correlations derived by Hollands and Brunger also had to be found that would fit the current set of SDHW systems studied. Detailed discussion of the verification and validation of these inputs is discussed in Chapter 3. The final set of data used in the regression analysis was run using flow rates calculated by the Hollands and Brunger equations.

II.2.2.5 Tank Size and Characteristics

Tank size has been shown to affect the performance of the system since the requirement to storage heat is a major part of thermal systems in general and is usually shown to correlate with daily load size (Wuestling et al. 1985, Dayan 1997). An optimal tank size is found during the validation phase of this research for every load size and is explained in detail in chapter 3. Studies have also been done to try to find when it is effective to use single or multiple tanks for storing the hot water (Buckles et al. 1980, Mather et al. 2002, Spur et al. 2006). For simplicity all models are run with one storage tank except the 800g daily load scenarios are run using two storage tanks which is more practical for the larger load size. Also, all tank insulation is R-20, which corresponds to 1 kJ/hr-m^2-K heat loss coefficient. There is no standardized requirement for solar tank insulation levels, but this value corresponds a well-insulated tank. The tank geometry is that of a vertical tank. The surrounding temperature is set to 68F to simulate the tank being indoors.

II.2.2.6 Pumps

The type and operation of any pumps in a SDHW system will affect the overall efficiency of the system. Yet the simulation of pump performance is beyond the scope of this project as it would have to be based on experimental data for any type of pump being used. This information is unavailable for this research. The SRCC ratings do take into account for the actual pump operations. Thus, for actual performance comparison the SRCC ratings will accurately model the system performance taking into account the parasitic loads such as pumping power. It is also unnecessary to incorporate pumping power into the model as most companies use the same type of pumps for similar system configurations. Also, pumping efficiency is independent of location, load size, collector type, and load temperature. Therefore, not including pumping power should not affect the relative comparisons within this study.

II.2.2.7 Piping

SDHW systems transfer the solar heat by moving a heat transfer fluid. Thus, there are heat losses as the fluid runs through the piping to the storage tanks and of course from storage to the final load, i.e. the shower. Even though these losses will affect the actual performance of a SDHW system, piping losses were not included in all simulations. This is because pipe losses have been shown to be only 0.97 times, i.e. 0.97% the well-known de Winter penalty factor or less as documented by Marshall at high flow rates (60.12 kg/hr-m2) (Marshall 1999). The de Winter penalty factor accounts for the associated heat exchanger losses within a SDHW system (Duffie & Beckman 2006) and the heat exchanger is accounted for in this research's TRNSYS simulation models. In the same work, Marshall also showed that for low flow rates (12.04 kg/hr-m2) and collector areas (A_c = 1m^2), the same pipe loss value will affect solar collectors with lower U-values, evacuated tube types, more than solar collectors with higher U-values, single glazed non-selective flat plate types. Thus, he found that for a pipe loss of 0.75 W/K the overall performance of a 1 m^2 panel with a U-value of 6 W/m^2-K dropped by 9.2%, as compared to a drop of 11.5% for a panel with a U-value of 2 W/m^2-K. Yet, the gap becomes smaller as the pipe loss value decreases or as the collector area increases. When the pipe loss value was decrease to 0.25 W/K, Marshall found the overall performance decreased by 3.4% and 4.3% for each panel respectively. Then, he also doubled the collector area (A_c =

$2m^2$) to find the overall performance decrease by only 1.7% and 2.2% respectively. Based off of Marshall's findings it is assumed actual systems can be designed that minimize the effects of piping losses on the overall system performance. Therefore, it was concluded for the current work that piping losses could be left out without adversely affecting the comparison between technology types.

II.2.2.8 Heat Exchanger Size and location

As discussed earlier, indirect SDHW systems incorporate a heat exchanger to transfer the collected solar gain from a non-freezable working fluid to the potable water. The heat exchanger in this set of models is set to have a UA of 800W/K. This was seen as a practical number based on size and efficiency limitations found in the literature (Dayan 1997).

This is kept constant with all the models except the 3,028 liters (800 gals) load was done with a much large heat exchange set at 8000W/K. After conducting an analysis of the optimal flow rate for each system, which is dependent on the heat exchanger characteristics, 8000 W/K was found to be the appropriate UA for the larger load size. Since not only the flow rate of system, but also the heat exchanger's UA has been shown to affect the overall performance of a SDHW system (Dayan 1997), it was seen as vital to the current research that all UAs used have the same baseline as compared by calculating the Number of Transfer Units (NTU) per scenario. Thus, starting with the UA of 800W/m2-K as defined for the smaller load sizes, the UA used for the 3,028 liters load was calculated in order that the NTU was kept relatively similar; within 1% between the other scenarios and the 3,028 liters daily load size scenarios. As this is a study to compare relative performance among the model, it is assumed to be most importance that all models have the same advantages or disadvantages with respect to the effects of the heat exchanger component.

Not just the size, but also the location of the heat exchanger in the system can affect the performance of the storage tank as shown by (Thornbloom et al. 1992, Dahm et al. 1998, Marshall 1999). For example, having the heat exchanger located within the hot water storage tank can disrupt the stratification process within the tank and therefore creating less than optimal storage capabilities. Thus, the heat exchanger was modeled outside of the tank to create an optimal situation where the heat exchanger performance will be independent of tank stratification.

II.2.2.9 Auxiliary Heat

SDHW systems use auxiliary heating devices. These are used as backup heat sources to ensure the hot water demand will be met. Auxiliary heating devices, whether gas or electric, are usually triggered by the water temperature to turn on. They can be located either inside the storage tank, in a separate tank, or by using an on-demand heater downstream from the storage tank. The temperature of the stored water in turn can affect the overall performance of the SDHW system, with the location of the auxiliary heating device being a major factor in the degree of the effect on the performance of the system as shown by (Shariah 1997). Thus, the auxiliary heating device is modeled to be downstream from the tank as an on-demand heater to verify that the auxiliary heater does not affect the performance of the storage tank and thus not be affecting the overall system performance.

CHAPTER III
TRNSYS SIMULATION MODEL

Four main TRNSYS projects are able to account for all of the 560 different SDHW system scenarios considered; direct or indirect system; two different flat plate collectors, one CPC collector, or two different evacuated tube collectors; four different load scenarios; and fourteen different locations. This is a total of four degrees of freedom. In Appendix V.3 is pictured the TRNSYS project for an indirect evacuated tube SDHW system. As with any research containing a simulation of the system being studied, the computer model has to be checked for correct operation of calculations.

III.1 VERIFICATION & VALIDATION OF THE MODEL

III.1.1 TIME STEP CONVERGENCE

Simulations are run to show the performance of each SDHW system over the course of a whole year. Since SDHW systems are dependent on climate conditions, annual simulations are assumed to be the appropriate length of time to measure performance over in order to account for seasonal effects. The climate data provided by the TRNSYS program is given in hourly intervals with TRNSYS having the capability to calculate average climate conditions for smaller intervals. Therefore the TRNSYS program performs constant volume, i.e. open system calculations across any time step the user provides. Thus, a convergence study was completed to find an appropriate time step for the models. Figure III.1 shows the results of the test for a range of heat transfer fluid flow rates; 50 kg/hr-m^2, 100 kg/hr-m^2, and 200 kg/hr-m^2. Runs were done for each 1 hour, 30 min, 15min, 6 min, 3 min, 1.5 min, 0.6 min, 0.3 min, and 0.15 min intervals. Smaller flow rates (e.g., 50 kg/hr-m^2) show steady values for the annual solar heat gain by using a large time-step of 30min for the calculations. The higher flow rate, 200 kg/hr-m^2 becomes consistent at less than 5 minutes for the time-step. Thus a time-step of 3 minutes was chosen to run all tests with. This is also consistent with advice from the TRNSYS helpdesk (TRNSYS Helpdesk 2007), who suggest using a time step around 5 minutes.

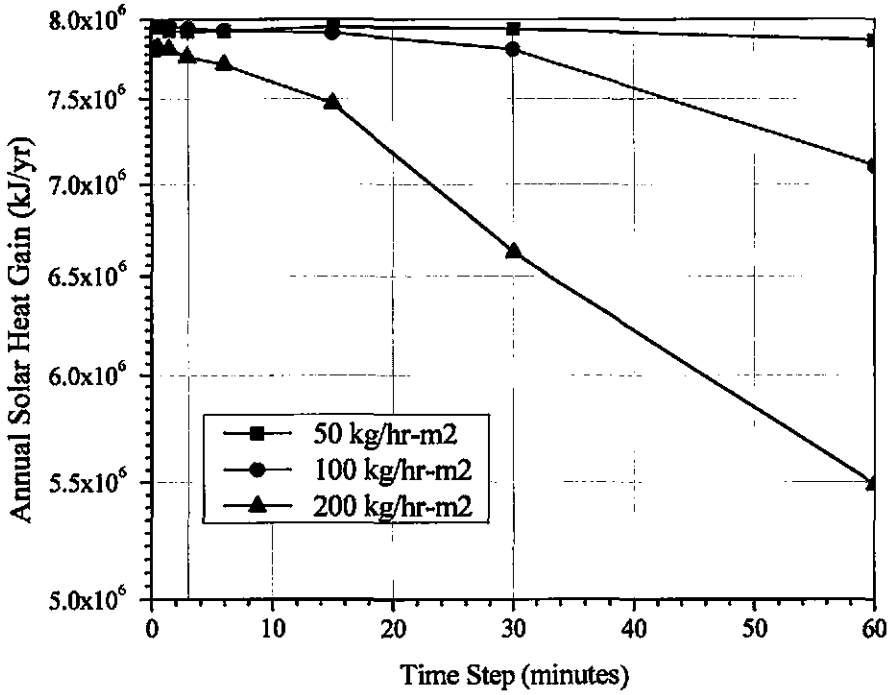


Figure III.1 Time Step Convergence

III.1.2 TANK COMPONENT

A second convergence was performed on the tank component. In reality the storage tank for SDHW systems is stratified. This because water has a lower density as its temperature rises and therefore will circulate above water that is cooler before mixing. SDHW systems are set-up to have in the incoming cold water inlet located at the bottom of the storage tank and the water heated by the solar collectors set to enter near the top of the tank in order to promote this cycle. If the tank is modeled as one complete volume then it would be seen as a fully mixed volume, which is incorrect. Because of that, the storage tank is modeled as having a temperature gradient in the vertical direction "z", while each plane perpendicular to "z" was assumed to have a uniform temperature distribution. The number of nodes used to model the vertical temperature gradient was tested to guarantee the accuracy of the results.

As seen in Figure III.2, testing was done across a range of flow rates for the tank model using a one-node, five node, ten node, and twenty node arrangement. The Solar Fraction was used as the testing parameter as it is a dimensionless parameter that takes into account to total operation of the SDHW system. As defined in Chapter 1, the Solar Fraction is the performance of the SDHW system. Therefore, convergence or non-convergence due to the flow rate, which the SDHW system performance is greatly affected will be seen by plotting the Solar Fraction versus the mass flow rate.

The model shows convergence, using the annual Solar Fraction as the marker, with a tank model using ten nodes or more. Thus, all simulations are run with twenty nodes in the tank component. This also correlates well with the advice given by the TRNSYS helpdesk (TRNSYS Helpdesk 2007).

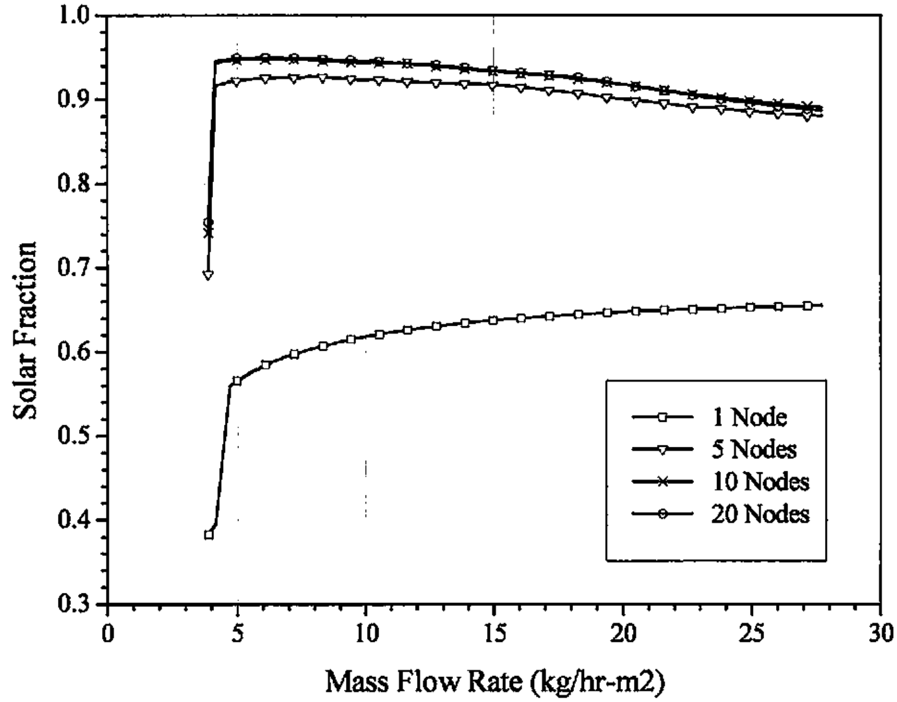


Figure III.2 Tank Node Convergence

III.2 System Parameter Testing & Validation

III.2.1 Storage Tank Size

As discussed previously the tank size will affect the storage heat capacity and thus the overall performance of a SDHW system. Figure III.3 shows the results of a parametric study of the optimal tank size with respect to location and technology type. It shows that the optimal tank size is not strongly dependant on either paramater. This trend is consistant across the other locations studed.

Also, in Figure III.3 it is clear that there is a point at which it becomes inconsequential to continue increasing the tank size for a given load. For the 302 liter/day load size, a tank size of 454 liters, corresponding to a 1.5 ratio of tank size to daily load size, is chosen as the optimal tank size as the increase in solar fraction by increasing the tank size falls below 1% by this point. For the other three load scenarios tank sizes of 379 liters, 227 liters, and (2) 3785 liters were used for 227liters/day load, 151liters/day load, and 3,028liters/day load respectively. This behavior was validated with existing data (Dayan 1997). Dayan's work was for a similar parametric study done for Miami FL and Madison WI; but only for flat plate collector types. Her work also showed that it becomes inconsequential to continue increasing the tank size for a given load at about a 1.5 tank size to load size ratio.

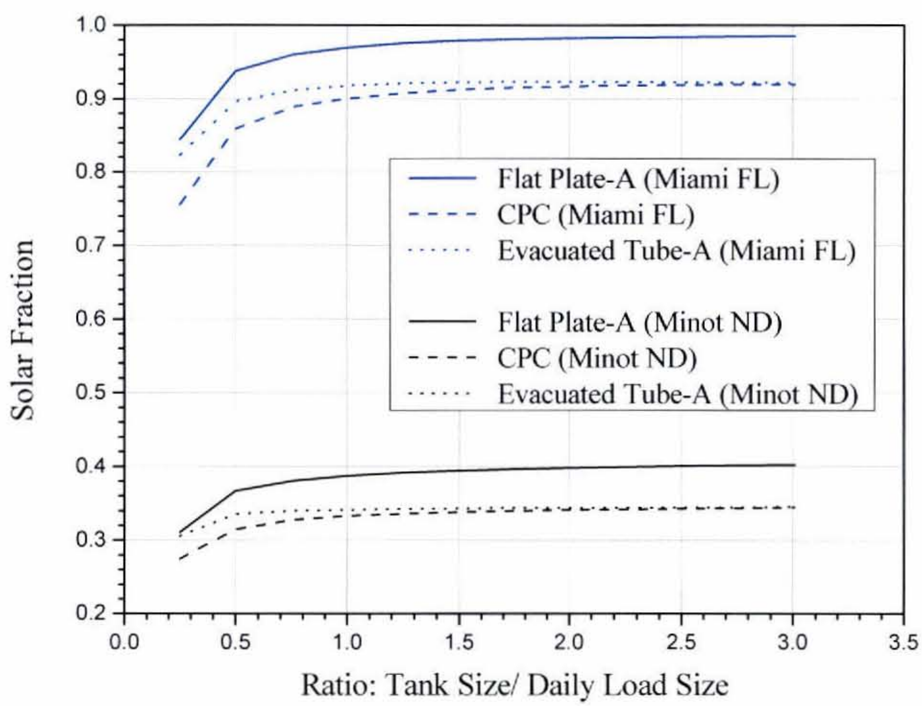


Figure III.3 Optimal Tank Size as a Function of the Ratio of Tank Size to Daily Load Size

III.2.2 Fluid Flow Rate

As stated before, the optimal flow rate must be used for each scenario simulated or the optimal performance will not be found. This would result in comparing optimal scenarios with other scenarios that are not optimal. To overcome this difficulty, GenOpt, the optimization program was initially used to find the optimal flow rate for each simulation run. The Hooke-Jeeves optimization algorithm option available in GenOpt was found to work best for this procedure. Because the function being evaluated is continuous, this derivative-free optimization method can be used. The Hooke-Jeeves algorithm uses a mesh from user-defined inputs for structure and boundaries by which the algorithm calculates the cost function. Based on the answer, the mesh is updated until reaching a user-defined difference in the answers (Wetter 2004).

The optimization process takes roughly fifteen minutes of computing time for the direct system configurations and roughly one hour for the indirect system configurations. The difference in computing time is directly a cause of the cost function being a function of only one flow rate for the direct system configurations and a function of two flow rates which are also dependent on each other for the indirect system configurations. Figure III.4 shows an example of optimal flow rates found using GenOpt for Daggett, CA for all five collector types in direct system configuration.

The overall trend is that the optimal flow rate increases with increasing collector area, yet decreases per unit area (m^2). This is because the collector heat removal factor, F_R, is affected by the inlet collector fluid temperature which will increase with increasing collector area (as more solar energy is absorbed) and thus will lower the overall F_R resulting in lower flow rates per unit area being more optimal with increasing collector area as discussed by (Wuestling et al. 1985, Duffie & Beckman 2006). Though, it is shown that there are other parameters affecting optimal flow rates besides just collector area by the fluctuation of each curve as it increases. It was beyond the scope of this study to define what those parameters are and thus the focus was to verify that the optimal flow rate was found for each scenario simulated in the modeling process.

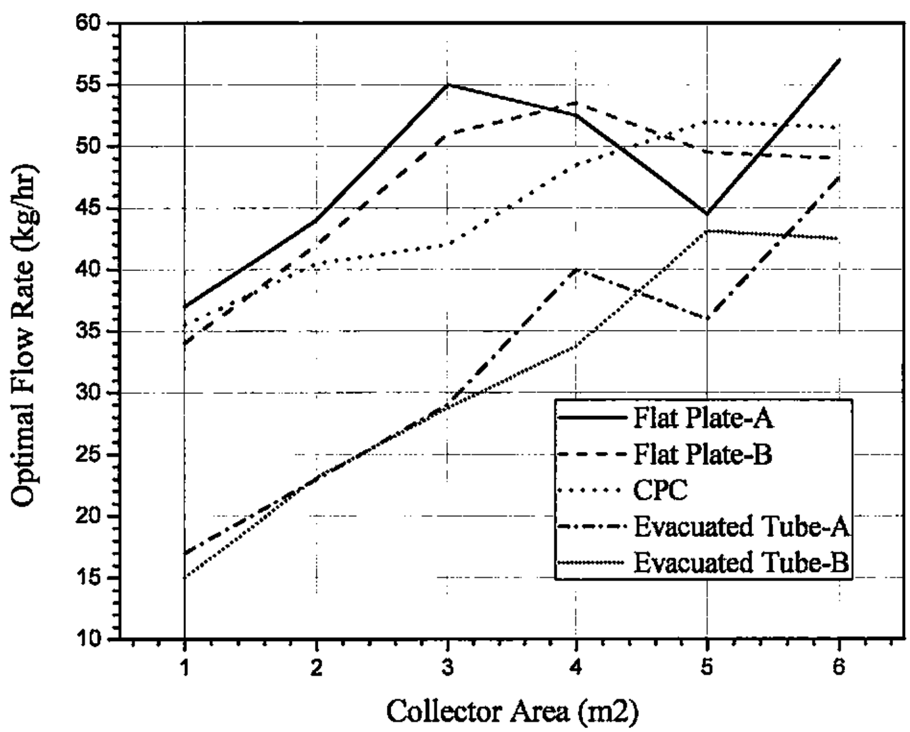


Figure III.4 Optimal Flow Rates found using GenOpt for Direct System Configuration Located in Daggett, CA

Alternatively, a set of equations for finding the optimal flow rate in solar hot water systems are detailed in the available literature (Hollands & Brunger 1992).

Hollands & Brunger use data documented by Wuestling et al. to fit an optimal flow rate equation for a direct system, i.e. one without a heat exchanger (Wuestling 1985). It is unknown from either document if this equation would be valid to use for the three different types of collectors studied in this research as well as for other study limits such as the maximum collector area the equation could still be used. The constants used in the equations also had to be found that would fit the current set of SDHW systems studied. Shown as Equation III.1, Hollands & Brunger do not give a specific number or range of values for the constant 'c' in the equation as well as it will be shown later constant 'q' had to be re-defined for the current research's parameters.

$$\dot{m}_{opt} = c(A_c)^q \qquad \text{(III.1)}$$

Where: c = constant

A_c = collector area

$q \approx 0.4$

By choosing to set 'c' equal to the optimal flow rate found by GenOpt for a collector area of $1m^2$ for Miami, FL the following data set is obtained in Figure III.5. The reason there is poor agreement between the two methods is because the GenOpt program finds the actual minimum of the cost function, which in this case is the Solar Fraction. Whereas the Hollands & Brunger equation is a power-equation fit resulting in a nicely increasing curve.

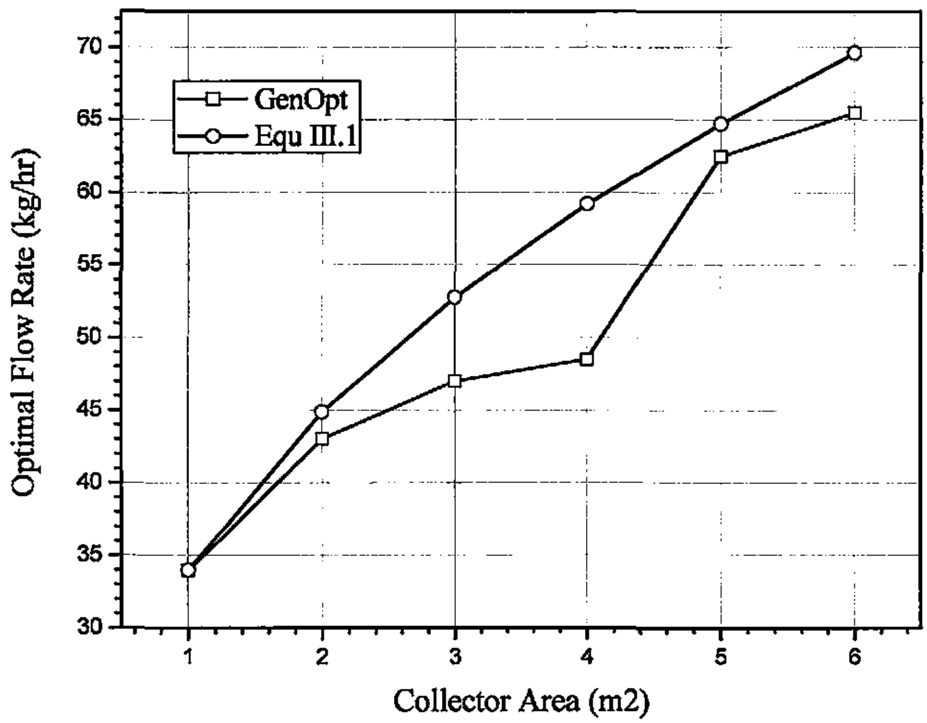


Figure III.5 Optimal Flow Rates found using GenOpt and Equation III.1 for a Direct Flat Plate System

Holland & Brunger's (1992) study also determines a correlation for finding the optimal flow rates for an indirect system, i.e. one with a heat exchanger based off of their fit of Wuestling's data. First, they calculates that R_{opt} (where R_{opt} is the optimal heat capacity ratio) is only a function of the collector plate efficiency and collector heat loss coefficient across the collector area per the overall conductance of the heat exchanger; Equation III.2. This means that the flow rate is independent of solar radiation and ambient temperature. Thus, R_{opt} is a function of the conductance ratio; Equation III.4.

$$R_{opt} = 1 + \frac{(F'U_L A_c)}{(UA)_x} = \frac{(1+\rho)}{\rho} \qquad \text{(III.2)}$$

Where $R_{opt} = \frac{(m_c C_{p,c})}{(m_t C_{p,t})}$ (III.3)

$F^{'}$ = collector plate efficiency factor

U_L = collector heat loss coefficient

$(UA)_x$ = overall heat conductance of heat exchanger

$$\rho = \frac{(UA)_x}{(F'U_L A_c)} \qquad \text{(III.4)}$$

Then, they showed the optimal tank side flow rate for an indirect system can then be found by combining Equations III.1 and III.2 to give Equation III.5.

$$\dot{m}_{t,opt} = \dot{m}_{opt}\left(\frac{\rho}{(1+\rho)}\right)^q \qquad \text{(III.5)}$$

Then, the optimal collector side flow rate is solved for using Equation III.3. Figure III.6 shows the calculated flow rates for an indirect system in Miami FL by (i) using the GenOpt Program (blue), (ii) using the results from running the GenOpt Program for the direct case with Hollands & Brunger's (1992) work (red), and with just (iii) using Hollands & Brunger's (1992) correlations (black). Again the questionable agreement between the three methods because both methods in which the GenOpt program is used to find the direct system flow rate; the program finds the actual minimum of the cost function. Whereas the Hollands & Brunger equation for the direct system flow rate a power-equation fit resulting in a nicely increasing curve.

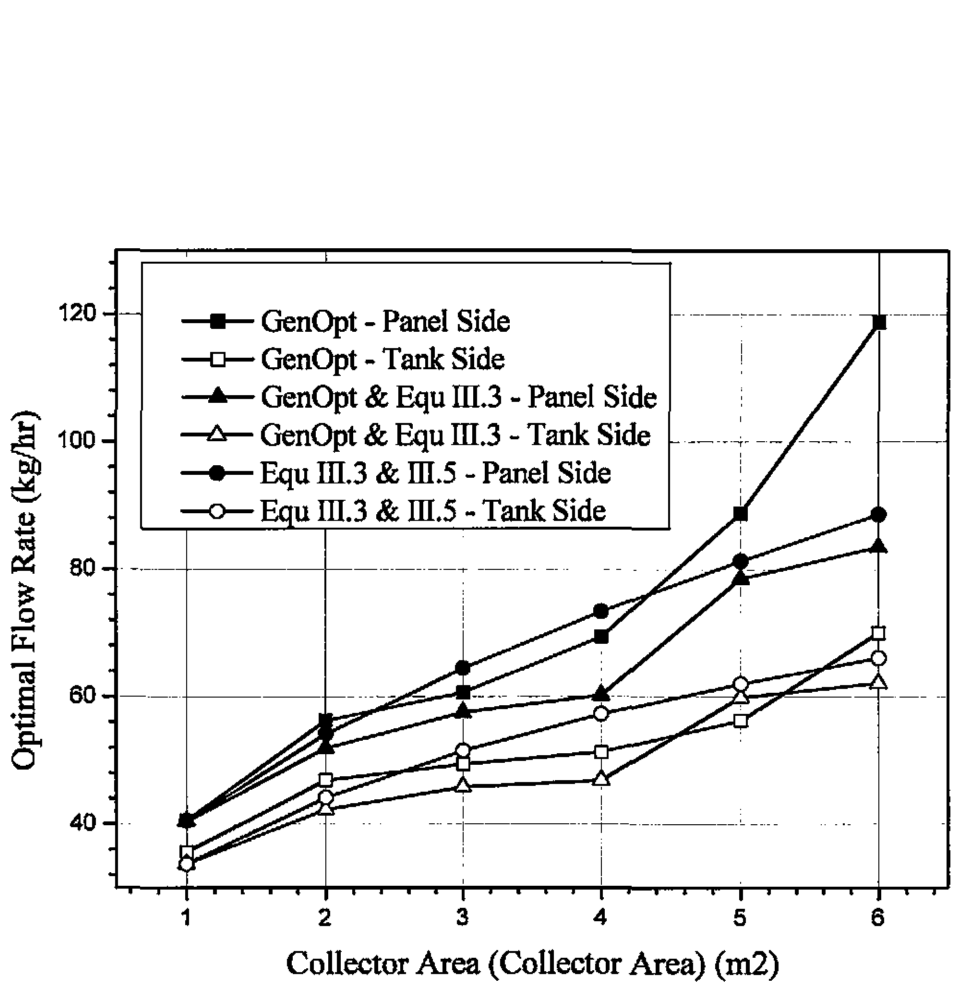


Figure III.6 Optimal Flow Rates found using all 3 Methods for an Indirect Flat Plate System

TRNSYS was then run to find the solar fraction for each flow rate found from each of the methods used. Figure III.7 depicts the results of the comparison for a flat plate collector located in Miami, FL. For the direct system shown, using a flow rate calculated from Equation III.1 compared to flow rates found using the GenOpt program show less than a 0.05% difference across the range of collector areas. For the indirect system shown (Figure III.7), flow rates found using (i) GenOpt, using (ii) GenOpt for the direct case with Hollands & Brunger's (1992) equations to find the optimal indirect flow rates, and using (iii) just Hollands & Brunger's (1992) correlations, show results with less than a

0.12% difference across the range of collector areas, with the largest difference in Solar Fraction occurring at the largest collector area.

Even though the actual flow rates for type of calculation are not similar, the good correlation is most likely due to the fact that the slope of the flow rate versus Solar Fraction function, after a very sharp initial increase (as can be seen in Figure III.2) flattens out in the area of the optimal flow rate. Thus, a range of optimal flow rates can be used without significant losses in performance; as seen by the less than 0.12% difference. This explains why the optimal flow rates found using GenOpt as seen in Figures III.4 – III.6 do not increase nicely, yet are still global minima and not local minima. The curve flattens out at the minima, making it of small difference in Solar Fraction to use a flow rate within that range. Yet, after this long flat optimal range of flow rates, the curve starts to decrease proving what is already known, that there is an optimal flow rate.

This behavior distinctly becomes more apparent as collector area increases, i.e. the range of 'almost' optimal flow rates becomes larger, thus allowing for larger discrepancies between calculated 'optimal' flow rates. This is most likely because the overall limits of performance (as discussed in detail in Chapter 5) are reached at larger collector areas and therefore flow rate is no longer the deciding factor on system performance.

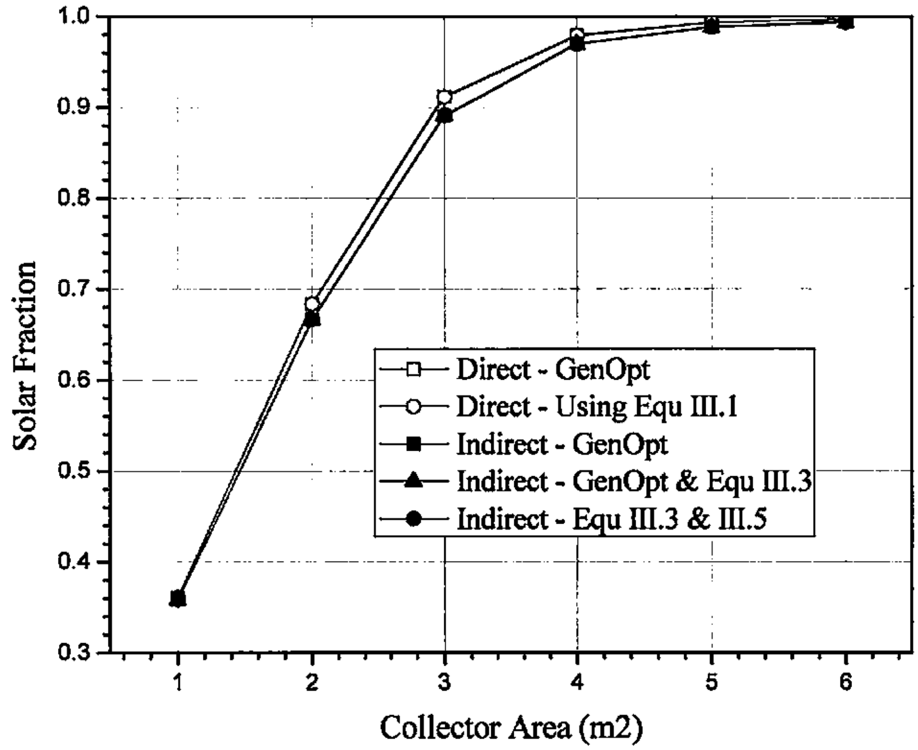


Figure III.7 Solar Fractions for both Direct and Indirect Flat Plate Systems Located in Miami, FL

Hollands & Brunger's (1992) work did not state if Equations III.1-III.5 could be applied to all types of solar collectors, which is a concern as different collectors have different heat transfer capabilities as discussed in Chapter 3. Thus, the same procedure as discussed previously was repeated for the CPC and evacuated tube collectors in an indirect system configuration to compare the results. The three methods to calculate optimal flow rates showed good correlation for use in modeling the CPC collector performance with a difference of 0.14% or less for the range of areas studied as seen in Figure III.8.

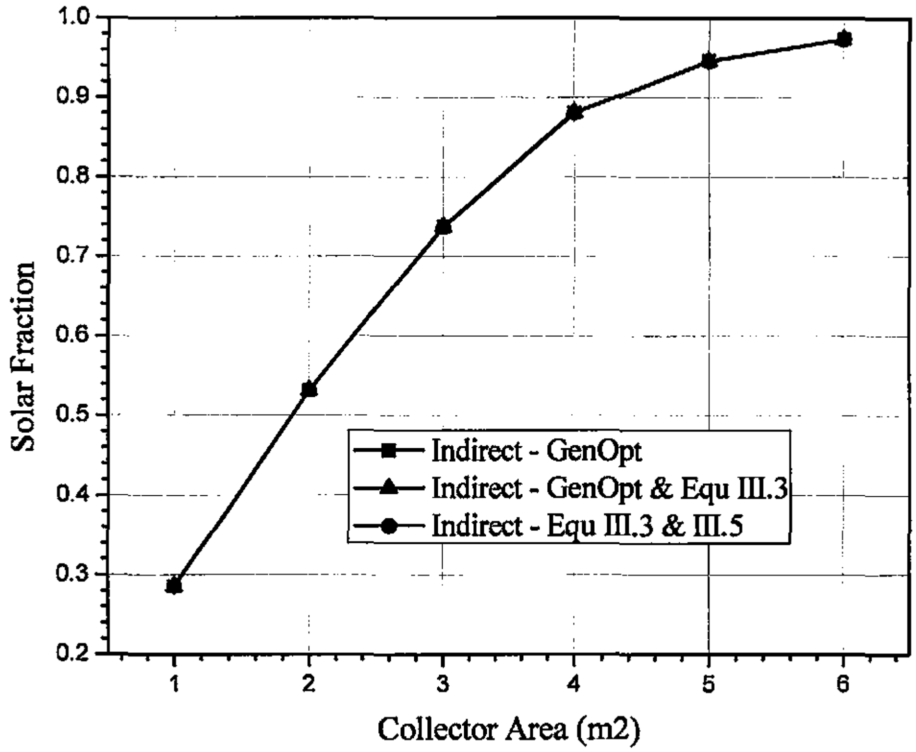


Figure III.8 Solar Fraction for Indirect CPC System Located in Miami, FL

For the evacuated tube collector, the flow rates found with method (i), using just GenOpt, as compared to with method (ii), using GenOpt for the direct flow rate with Hollands & Brunger's (1992) equations to find the optimal indirect flow rates, have a difference of 0.07% or less across the range of areas studied (Figure III.9). Yet, using method (iii) which is just using Hollands & Brunger's (1992) equations, results in much higher differences of 11.12% at the higher collector areas (Figure III.9). Thus, equations III.3 and III.5 were able to effectively define optimal flow rates for the indirect systems' flow rates as shown by good correlations to the GenOpt findings. Yet, equation III.1 did not work to find the direct system's optimal flow rate for evacuated tube collectors.

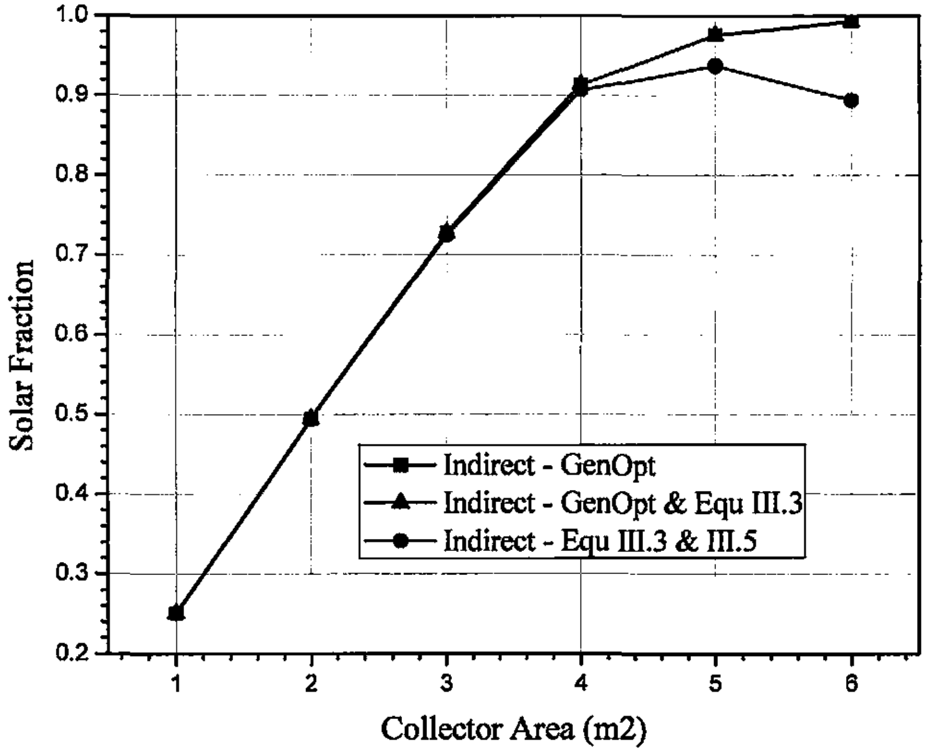


Figure III.9 Solar Fraction for Indirect Evacuated Tube System Located in Miami, FL

Therefore, other values were researched for both the 'c' and 'q' constants for equation III.1. It was found that instead of using the optimal flow rate found by GenOpt for a collector area of 1m^2 for the 'c' constant, it was set to a value of 30. The 'q' constant was redefined to be equal to 0.5 instead of 0.4 as suggested by Hollands & Brunger's (1992) work. After incorporating these changes, a very good fit, less than 1.0% difference was found as compared to using the GenOpt flow rates for the same three collector scenarios (both direct and indirect flat plate, CPC, and evacuated tube) located in Miami FL as just discussed.

Thus, it was determined that the same constants; c = 30 and q = 0.5 could be used to find the optimal flow rates for all three collector types, all locations, and the three smallest load scenarios; 151 kg/day, 227 kg/day, and 302 kg/day with a difference of less than 1.5% as compared to using GenOpt flow rates. Furthermore, as the optimal flow rate is clearly dependent on collector area, the constant 'c' had to be recalculated to be found to equal 95 for the largest load size of 3,028 kg/day. This is because this load size required much more collector area, 10-80m^2 as compared to the smaller load sizes which required 1-8m^2. This also gave a very good fit of less than 1.5% for the largest load size scenarios for all three collector types and all locations as compared to using GenOpt flow rates.

Using Hollands and Brunger's (1992) equations, or the third method, two sets of optimal collector side flow rates were plotted in Figure III.10. As seen, the optimal mass flow rate per collector area decreases as total collector area increase for both the set of optimal flow rates found for the smaller three load sizes as well as for the largest load size. As discussed by Duffie and Beckman (2006) and Wuestling et al. (1985) this is the result of balancing the losses and the gains of the system.

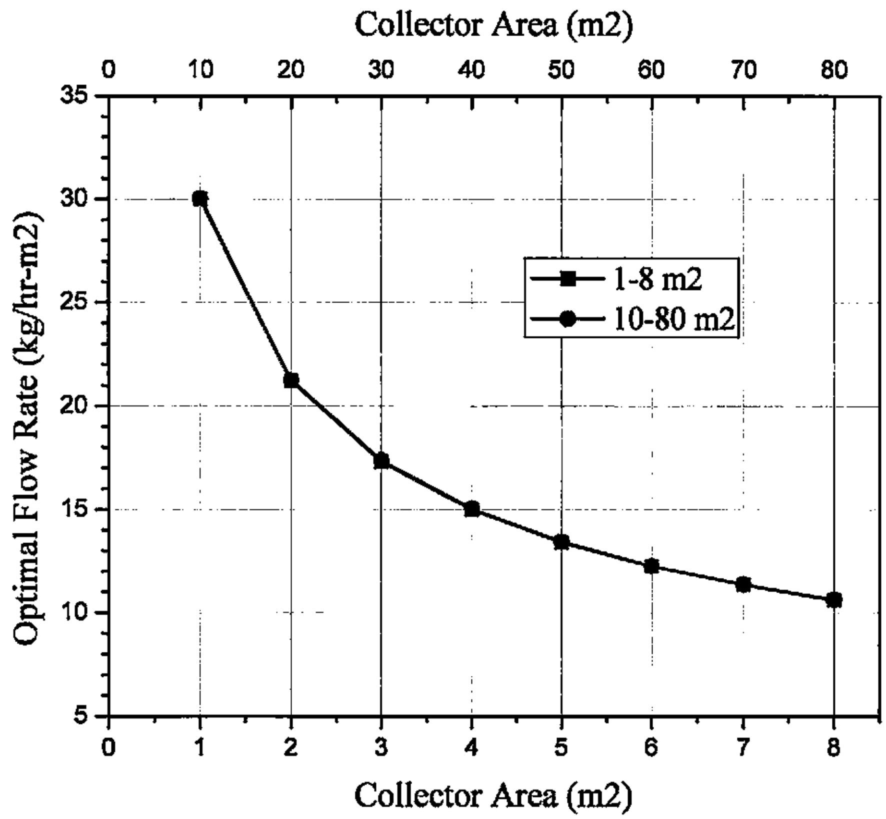


Figure III.10 Optimal Flow Rates per Area as a Function of Collector Area

III.3 RESULTS

The validation testing showed that the final models used in this research were computing values correctly as compared to results found in the available literature (Dayan 1997, Kleinbach et al. 1993, Wuestling et al. 1985). It was also shown that Hollands & Brunger's (1992) equations could be used after finding the correct constants, 'c' and 'q', to find the optimal flow rate in all scenarios studied. This saved having to run every data point through GenOpt, which would have been thousands of hours of computing time.

CHAPTER IV
REGRESSION ANALYSIS

After compiling a complete set of data for the 560 scenarios simulated, an in-depth analysis was conducted to examine the results. This investigation was divided into two sections; (i) the demand-side effects and (ii) the supply-side effects. The importance of different input parameters was quantified numerically by fitting the data or validated by simple comparisons. Also, trends found within the data were evaluated for their significance in affecting overall collector performance. Lastly, these results were compiled to support a proposed rating procedure to be used to define appropriate regulating criteria of SDHW systems.

IV.1 DEMAND-SIDE EFFECTS

Shown in Figure IV.1, is the performance for one type of solar collector, Flat Plate-A as it increases per increasing collector area across the 14 different climates. It can be seen from this figure that there are two physical limits to the performance of the collector as its area is increased. As is only realistic, the upper limit is approached when the output of the solar collector stops increasing as the total demand is met, i.e. as it reaches a solar fraction of 1.0. Locations such as Honolulu, HI or Daggett, CA have the realistic possibility of reaching this type of upper limit. The other, i.e. lower limit can be seen at locations such as Helena, MT or Youngstown, OH. This lower type of limit shows that the incremental increase in performance of the collector becomes significantly small as collector area is increased. This happens as the losses of the system become larger than the incremental increase of gains per unit of time. This trend of having two limits to the performance of the collector as its area is increased, is repeated for all four other collector types evaluated and will not be shown for brevity.

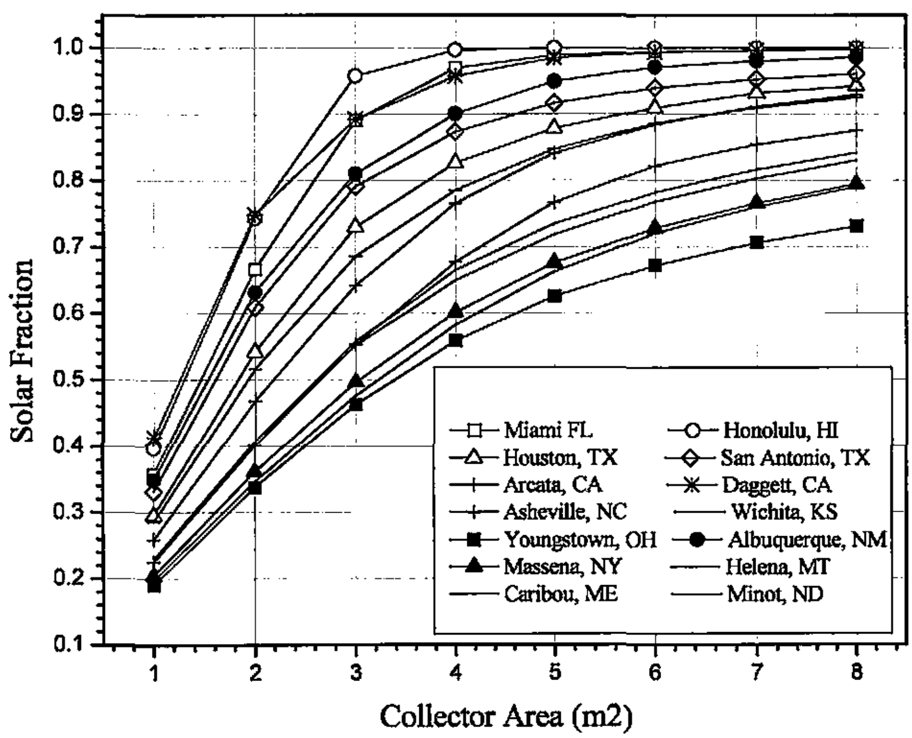


Figure IV.1 Performance of Flat Plate Collector per Area for all 14 Locations

What this means is that appropriate Solar Fractions cannot be designated on a global biases. For example, a Solar Fraction of 0.75 for Honolulu, HI would not be that impressive as it would only take one more m^2 of collector area to be able to reach a Solar Fraction of over 0.95. Yet, a Solar Fraction of 0.75 would be a more intimidating figure to aspire a SDHW system to meet in a location such as Youngtown, OH as this level of performance is closing in on the physical limitations of performance for that location. This figure (Figure IV.1) thus, also justifies location specific concerns brought up in regards to the proposed Energy Star criteria for SDHW systems (ACEEE 2007, SEIA & SRCC 2007)

By looking at each collector type at a set area per load size ($8m^2$ for the 302 l/d and 227 l/d, $4m^2$ for 151 l/d, and $80m^2$ for the 3,020 l/d load), other important trends in the data become clear, as shown in Figures IV.2 - IV.5. In Figure IV.2 it can be seen that the gaps between the performance of individual collectors increase as the limiting factor changes from the upper limit of meeting the total demand to the lower limit of the losses outweighing the gains. For example, Flat Plate-B and Evacuated Tube-A give the same performance for Honolulu, HI. Yet, Evacuated Tube-A outperforms Flat Plate-B by over 10% in Youngstown, OH.

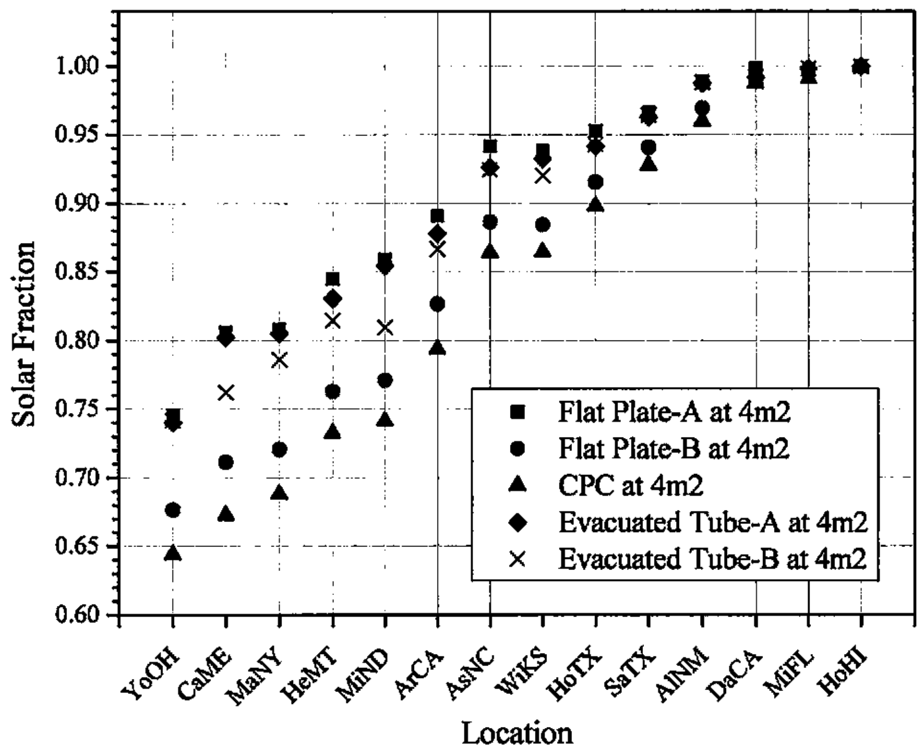


Figure IV.2 Performance of each Solar Collector to meet the 151 l/d Load at a Set-Point Temperature of 48.8°C

What also stands out is the formation of two branches of performance levels for the five different collectors as viewing from left to right in the figure. In Honolulu, HI the five collectors all perform at essentially a 1.0 Solar Fraction for any load size. Yet, as the performance for all the collectors decreases per a location towards the left of the figures, there is an upper level of performance which includes the Flat Plate-A, Evacuated Tube-A, and Evacuated Tube-B collectors. There is also a lower level of performance which includes the Flat Plate-B and CPC collectors. The reason for this branching of the performance of different SDHW systems is based on the climate conditions becoming harsher, including less solar radiation annually, more cloudy days, and colder ambient temperatures. Thus, the efficiency of an individual SDHW system will become more apparent making it clearer which ones actually perform better.

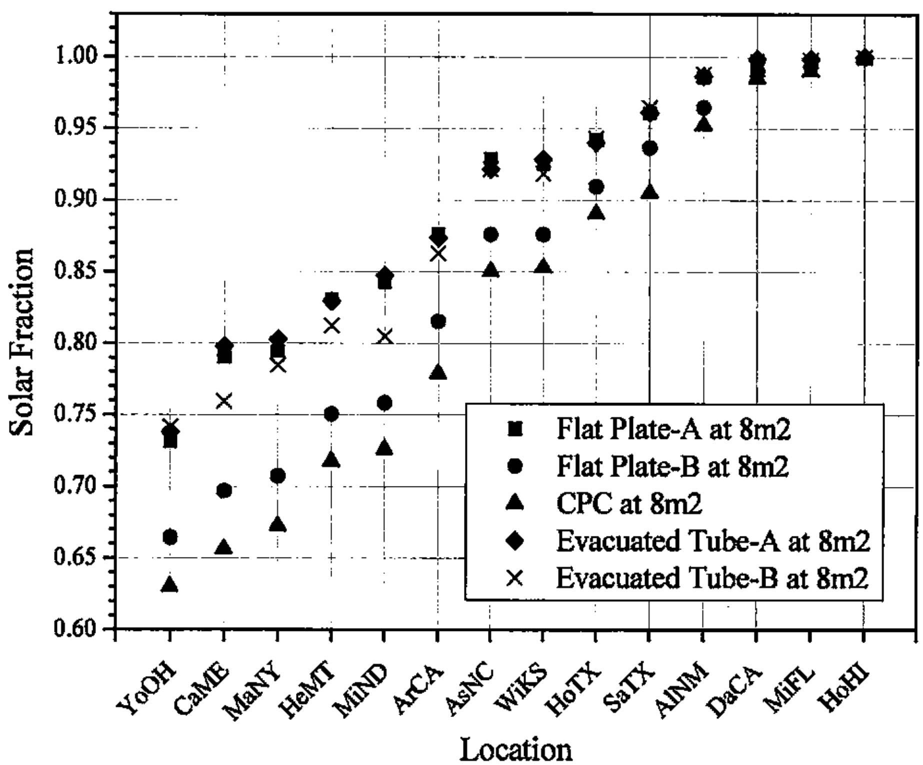


Figure IV.3 Performance of Each Solar Collector to Meet the 301 l/d Load at a Set-point Temperature of 48.8°C

Figure IV.3 and IV.4 shows the same trends as seen in Figure IV.2 for the daily load sizes of 301 l/d and 3,020 l/d respectively. The gaps between the collectors' performance increases from right to left in the figures. Also, there is the clear branching off of the more efficient set of collectors and the less efficient set of collectors as the climate conditions become worse. In each of these figures (Figures IV.2 - IV.5), the ordering of the locations was simply based off of worst performance to best performance (left to right) with the order of the locations being the same even as the load size and hot water temperature set point was varied.

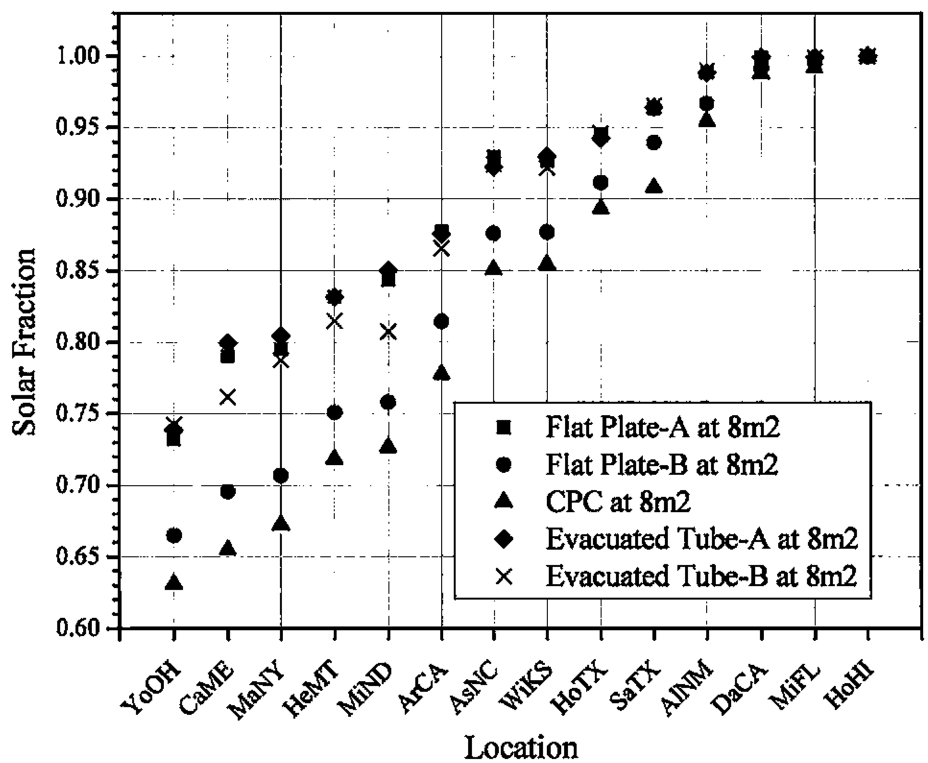


Figure IV.4 Performance of Each Solar Collector to Meet the 3,020 l/d Load at a Set-point Temperature of 48.8°C

For the three loads (Figures IV.2 - IV.4) all with set point temperatures of 48.8°C, the performance of each collector with respect to location is almost identical. The 227 load (Figure IV.5) which has a set point of 60°C, although being very similar in the general trends, shows another slightly different trend in performance. The evacuated tube collectors perform, as compared to the other types of collectors, better than for the loads with set point temperatures of 48.8°C. The overall annual energy requirements for the two load sizes, 227 l/d and 302 l/d are within 3% of each other per location with the 302 l/d total load having on average the larger requirements. Yet, it should also be pointed out that overall performance for all types of collectors is lower by ~3% than for the 302 l/d load size even though the collector areas are the same. Thus, the importance of testing collectors at the same set point temperature is clear. Also, it is advantageous to not run SDHW systems at higher set point temperatures than actually needed. Overall the performance will be lower.

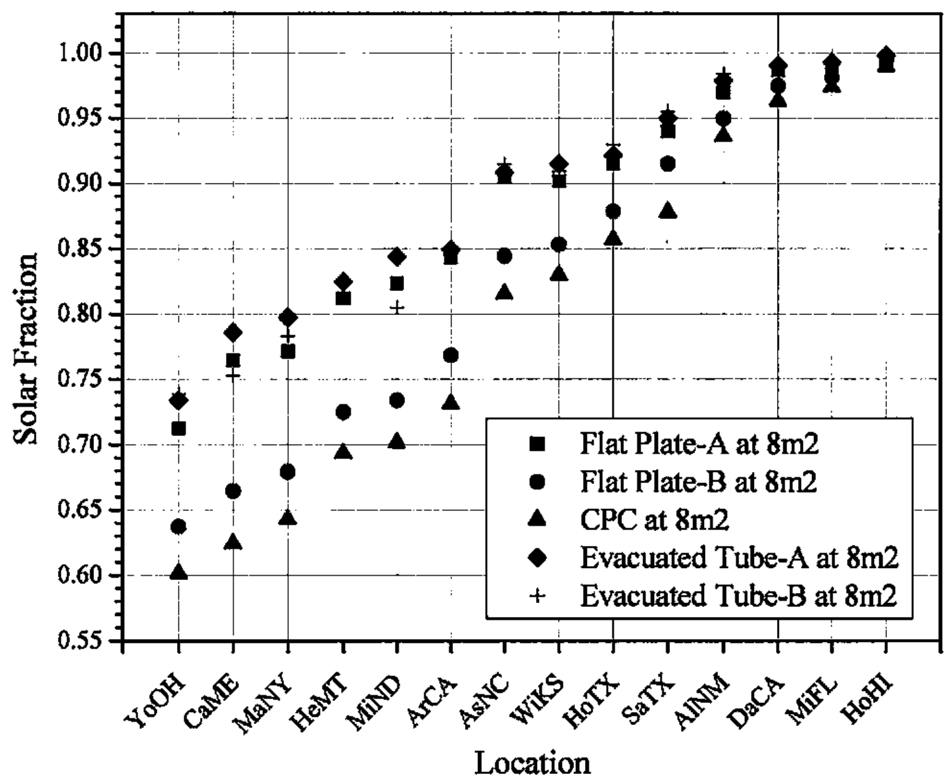


Figure IV. 5 Performance of Each Solar Collector to Meet the 227 l/d Load at a Set-point Temperature of 60°C

It is clear from Figures IV.2 - IV.5 that per collector type there is a repeatable trend in performance across location. After attempting to find a function that specifically addressed this location-based trend; multiple linear regression was used to fit the simulated data modeled in TRNSYS. The equation is of the form $y = a + b_1x_1 + b_2x_2 \ldots b_nx_n$. Microsoft® Office Excel offers a macro called LINEST which will fit a straight line for a set of x's and y's using the least squares method. This program was used to analyze different combinations of independent x's to find what group of variables would result in the best fit. Besides the obvious independent variable, available solar radiation; any correlation to ASHRAE's (ASHRAE 2004-1) already established climate zones was reviewed. Yet, since those climate zones are based on number of heating and cooling days required for a given location, an appropriate correlation simply using those climate designations was not found for predicting the performance of SDHW systems across different locations in this study. In other words, the ASHRAE climate zones didn't linearly fit the data set.

Thus, it was found that SDHW systems can be shown to be linearly dependent on the group of the following five factors: the location's average annual clearness index, average annual solar radiation, total annual load size, as well as average ambient and incoming water temperatures. These five independent variables were capable of accounting for the majority of the difference in a SDHW system's performance for different locations.

Then each of the five indirect SDHW systems were plotted with their Solar Fraction calculated from the TRNSYS models (Y_{actual}) against their Solar Fraction found from the multiple linear regression ($Y_{estimated}$) for all fourteen locations with the upper limit set to a SF of 1.0 as it is realistically impossible to provide more than 100% of the load; i.e. over a 1.0 SF. Figure IV.6 shows (with Table IV.1 listing the constants used) these five systems fit for the 151 l/day load size with each collector having an area of $4m^2$. As shown, each fits within +- 5% accuracy to the simulated data. The R^2 value also known as the square of the multiple correlation coefficient (Dowdy et al. 2004), is given for each equation next to the SDHW system type it represents. Note that all R^2 values are close to 1 showing the equations fit the data well.

		a	b_1	b_2	b_3	b_4	b_5
Weighting factor for:		N/A	Average Annual Clearness Index	Average Annual Solar Radiation	Average Incoming Water Temperature	Average Ambient Temperature	Total Annual Load Size
Indirect SDHW System	Flat Plate-A	-41.029	-2.776	2.093	41.076	-0.153	34.028
	Flat Plate-B	-44.657	-2.690	2.135	44.392	0.074	36.838
	CPC	-43.190	-2.661	2.167	42.956	0.072	35.574
	Evacuated Tube-A	-42.072	-2.721	2.072	42.208	-0.242	34.841
	Evacuated Tube-B	-21.570	-2.420	1.855	21.748	0.141	18.179

Table IV.1 Constants for Multiple Linear Function Fit for the 151 l/d Load with Collector Areas of $4m^2$

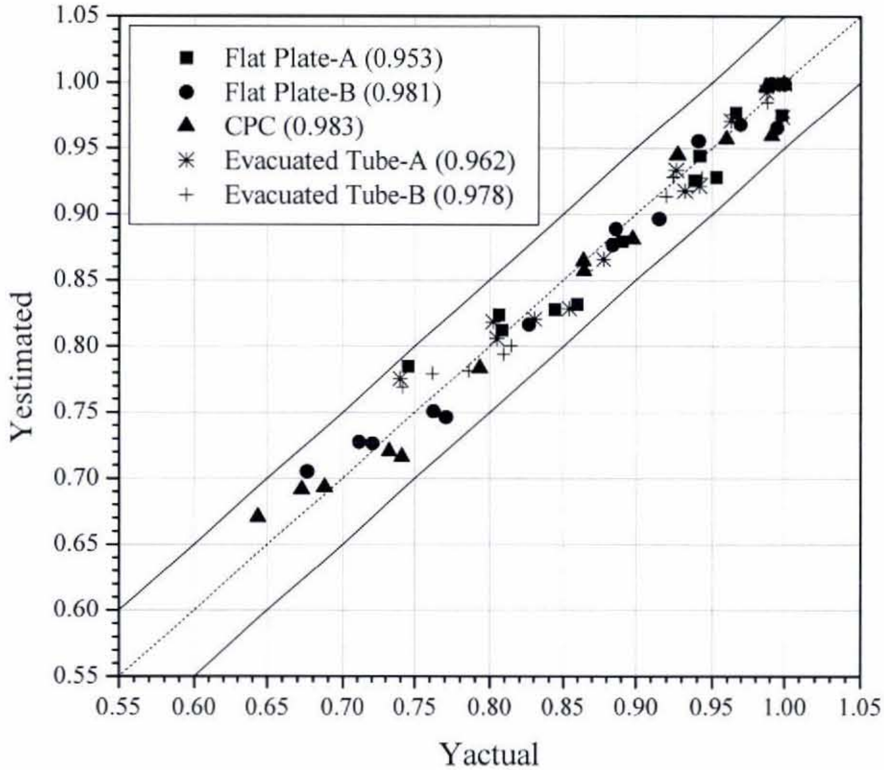


Figure IV.6 Multiple Linear Regression for 151 l/d Load with Each Collector Area Equal to 4m^2

Figure IV.7 shows (with Table IV.2 listing the constants used) these five systems fit for the 302 l/day load size with each collector having an area of 8m^2. Again, each fits within +- 5% accuracy to the simulated data. Also, all R^2 values are close to 1 showing the equations fit the data well. Only one collector area size was chosen to fit all the functions to instead of a range of areas because manufactures only offer SDHW systems in set collector area sizes.

Weighting factor for:		a N/A	b_1 Average Annual Clearness Index	b_2 Average Annual Solar Radiation	b_3 Average Incoming Water Temperature	b_4 Average Ambient Temperature	b_5 Total Annual Load Size
Indirect SDHW System	Flat Plate-A	-41.060	-2.690	2.062	41.111	-0.140	34.011
	Flat Plate-B	-42.182	-2.608	2.095	41.967	0.100	34.794
	CPC	-43.724	-2.485	2.075	43.494	0.083	35.959
	Evacuated Tube-A	-38.750	-2.486	1.935	38.909	-0.183	32.111
	Evacuated Tube-B	-15.710	-2.320	1.788	16.001	0.165	13.407

Table IV.2 Constants for Multiple Linear Function Fit for the 301 l/d Load with Collector Areas of 8m²

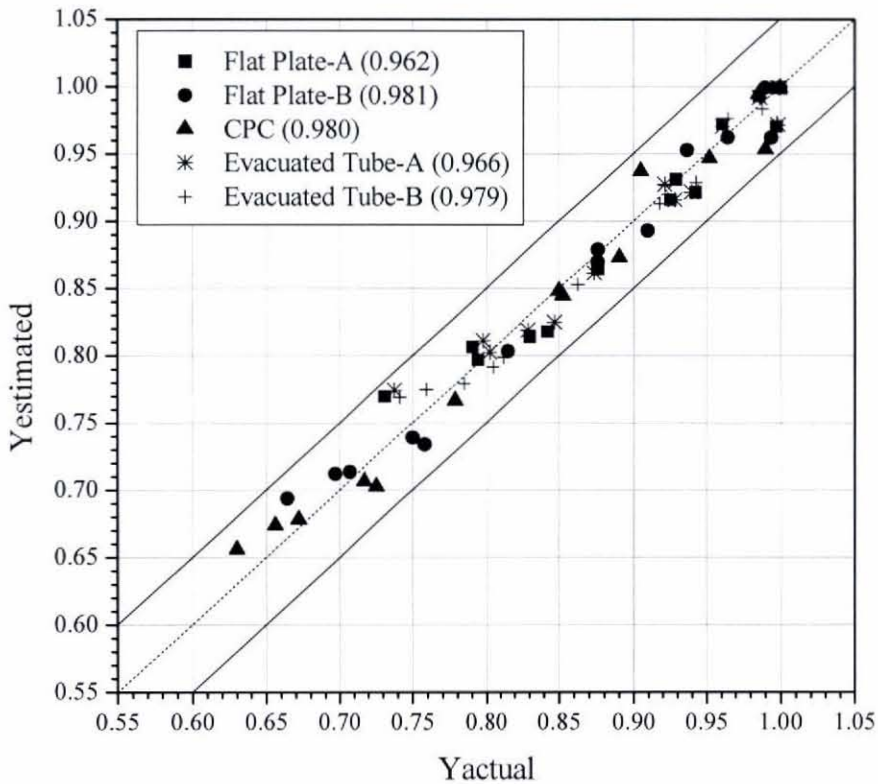


Figure IV.7 Multiple Linear Regression for 301 l/d Load with Each Collector Area Equal to 8m²

Figure IV.8 shows (with Table IV.3 listing the constants used) these five systems fit for the 227 l/day load size with each collector having an area of $8m^2$. Again, each fits within +- 5% accuracy to the simulated data. Also, all R^2 values are close to 1 showing the equations fit the data well.

		a	b_1	b_2	b_3	b_4	b_5
Weighting factor for:		N/A	Average Annual Clearness Index	Average Annual Solar Radiation	Average Incoming Water Temperature	Average Ambient Temperature	Total Annual Load Size
Indirect SDHW System	Flat Plate-A	-52.788	-2.490	1.974	53.699	-0.277	45.328
	Flat Plate-B	-46.170	-2.330	1.993	46.856	-0.016	39.548
	CPC	-48.914	-2.183	1.963	49.608	-0.033	41.781
	Evacuated Tube-A	-42.244	-2.179	1.773	43.280	-0.343	36.378
	Evacuated Tube-B	-17.113	-2.132	1.694	17.789	0.086	15.121

Table IV.3 Constants for Multiple Linear Function Fit for the 227 l/d Load with Collector Areas of $8m^2$

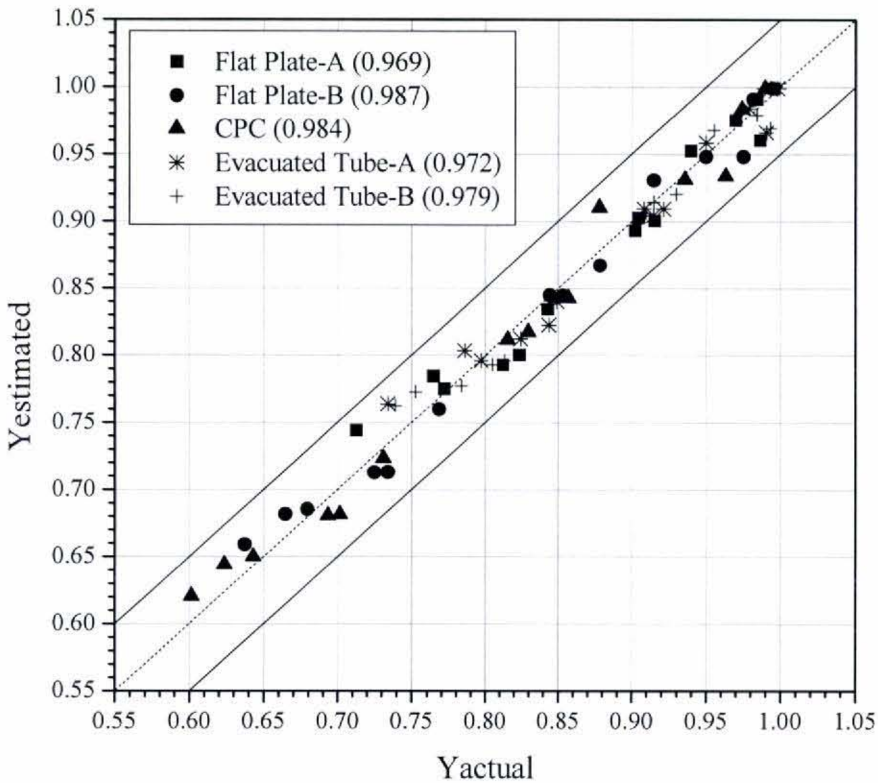


Figure IV.8 Multiple Linear Regression for 227 l/d Load with Each Collector Area Equal to 8m^2

Figure IV.9 shows (with Table IV.4 listing the constants used) these five systems fit for the 3,020 l/day load size with each collector having an area of 80m^2. Again, each fits within +- 5% accuracy to the simulated data. Also, all R^2 values are close to 1 showing the equations fit the data well.

Weighting factor for:		a N/A	b_1 Average Annual Clearness Index	b_2 Average Annual Solar Radiation	b_3 Average Incoming Water Temperature	b_4 Average Ambient Temperature	b_5 Total Annual Load Size
Indirect SDHW System	Flat Plate-A	-38.055	-2.673	2.055	38.094	-0.087	31.575
	Flat Plate-B	-39.331	-2.570	2.079	39.103	0.149	32.477
	CPC	-41.379	-2.449	2.061	41.129	0.131	34.053
	Evacuated Tube-A	-36.691	-2.484	1.938	36.838	-0.148	30.445
	Evacuated Tube-B	-15.651	-2.324	1.794	15.910	0.185	13.364

Table IV.4 Constants for Multiple Linear Function Fit for the 3,020 l/d Load with Collector Areas of 80m²

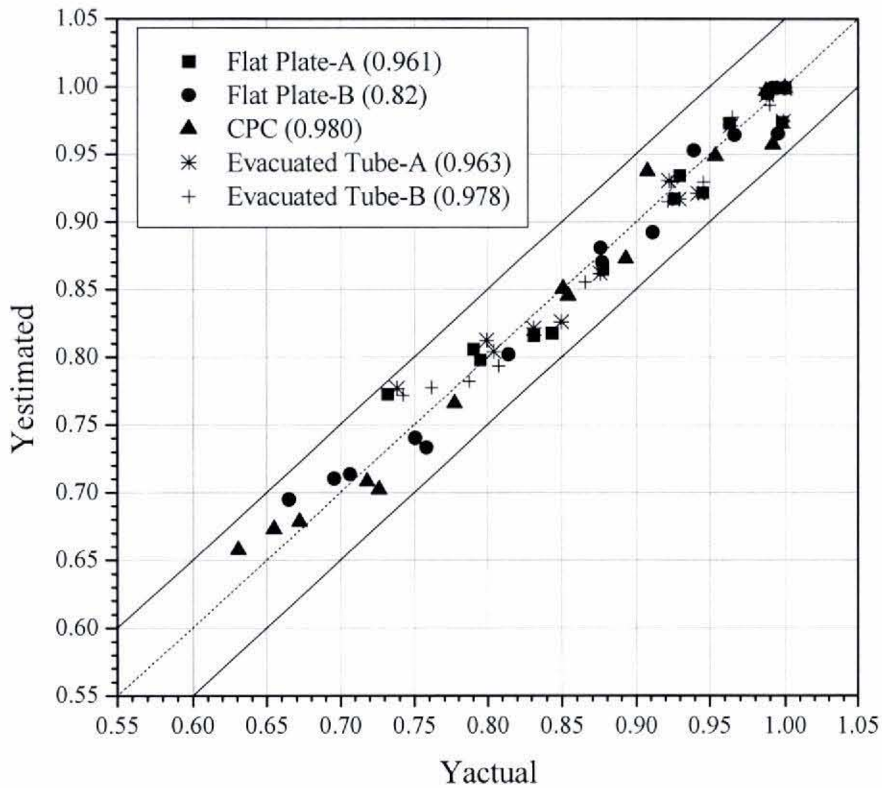


Figure IV.9 Multiple Linear Regression for 227 l/d Load with Each Collector Area Equal to 8m²

IV.2 Supply-Side Effects

It is known that performance per additional collector area is a function of diminishing returns (Duffie & Beckman 2006) and can also be seen in Figure IV.1 at the beginning of this chapter. Yet, the non-linear dependencies of solar collectors have not yet been fit with one equation. Thus, six different growth-rate functions were analyzed for their correlation to the data. These functions were picked because of their ability to appropriately model crop yield rates. The crop yield per area will increase as the number of crops planted is increased. Yet similarly as to what is seen with SDHW system performance, the crop yield is a non-linear function of diminishing returns with increasing number of crops. Therefore, the losses will start to outweigh the gains and the crop yield will go towards a limit; showing the same behavior as the performance increase per increasing collector area in SDHW systems.

As part of this research, this non-linear behavior was analyzed further to see if this would lead to a way to simplify or group the characteristics of SDHW systems further. The growth-rate functions analyzed for their agreement to the non-linear characteristics of SDHW systems include the following.

- The Von Bertalanffy's Model (Rawlings 1998)
- The Saturation Growth Rate Model (Chapra & Canale 2006)
- The Mitscherlich Growth Model (Rawlings 1998)
- The Grompertz Growth Model (Rawlings 1998)

- The Logistic or Autocatalytic Growth Model (Rawlings 1998)

- The Generalized Logistic or Richard's Growth Model (Wikipedia® 2007)

For each function the following procedure was done to find its ability to fit the data. First, the function was input into the Genetic Algorithm toolbox available in MATLAB© R2006b to find possible coefficients for the function. Then these coefficients were used as starting points for analyzing the function with the Optimization Toolbox available in MATLAB© R2006b. In the Optimization toolbox, the fsolve non-linear equation solver with the medium scale Gauss-Newton algorithm was used. After running MATLAB© R2006b for each of the previously mentioned functions it was found that the Generalized Logistic or Richard's Growth Model is the one found to fit the data most accurately (Wikipedia® 2007).

$$Y = A + \frac{C}{\left[1 + Te^{-B(X-M)}\right]^{1/T}} \qquad \text{(IV.1)}$$

Where: A - lower asymptote;

C - upper asymptote minus A;

M - time of maximum growth;

B - growth rate;

T - affects near which asymptote maximum growth occurs.

After the generalized logistic growth function showed to fit the data simulated by TRNSYS better than the other functions, two programs were created to fit all of the TRNSYS simulated data for the 560 different scenarios by running them in batch files to speed up the process. Figure IV.10 shows the function fit, with Table IV.5 listing the coefficients found, for a load size of 227 l/d in both Miami FL and Youngstown OH. The generalized logistic curve was fit with an error of 0.0056(Flat Plate-A), 0.009(Flat Plate-B) and 0.0029 (Flat Plate-A), 0.0011 (Flat Plate-B), for each location respectively. This figure is representative of the accuracy the generalized logistic curve was capable of in fitting any of the scenarios simulated in this study.

	Coefficients:	A	C	T	B	M	error
Miami, FL	Flat Plate-A	0.1642	0.8224	0.2517	1.0474	1.6690	0.0056
	Flat Plate-B	-0.0899	1.0734	0.7535	0.7776	1.6415	0.0090
Youngstown, OH	Flat Plate-A	0.1334	0.6217	-0.6928	0.3893	1.4052	0.0029
	Flat Plate-B	-0.7201	1.4342	0.6044	0.3425	-0.6804	0.0011

Table IV.5 Coefficients for Generalized Logistic Curve Fits

After fitting all of the sceneries simulated in this research, the five coefficients were examined statistically to look for any trends among the data. Although, this study did not find any relationship useful towards the research's final goal, this fit is significant towards defining a way of accounting for the performance of an individual SDHW system per area used. Grouping SDHW systems performance characteristics with the type of processes that are defined by growth-rate functions does bring insight to the overall pattern of the SDHW system.

The collectors in SDHW systems are not manufactured for any area. Thus, SDHW systems are rated by SRCC for actual panel dimensions. Systems that utilize more than one panel, such as two $2m^2$ panels would be rated a separate system than the same system utilizing just one $2m^2$ panel. It is appropriate to rate these systems individually as performance is highly dependent on area. Yet, to compare the two ratings against each other without accounting for the difference in area would not show the true different in performance. For example, in Figure IV.10 for both locations, if a Flat Plate-B system at $4m^2$ of collector area and a Flat Plate-A system with $2m^2$ were directly compared, it would show the Flat Plate-B system to be more efficient. Yet, when looking at the whole curve of performance as it is affected by collector area it is clear that the Flat Plate-A system would actually be more efficient per area. The performance of different systems needs to be compared with accounting for collector area. Neither the SRCC nor Energy Star discusses the effect of area in any of their rating calculations.

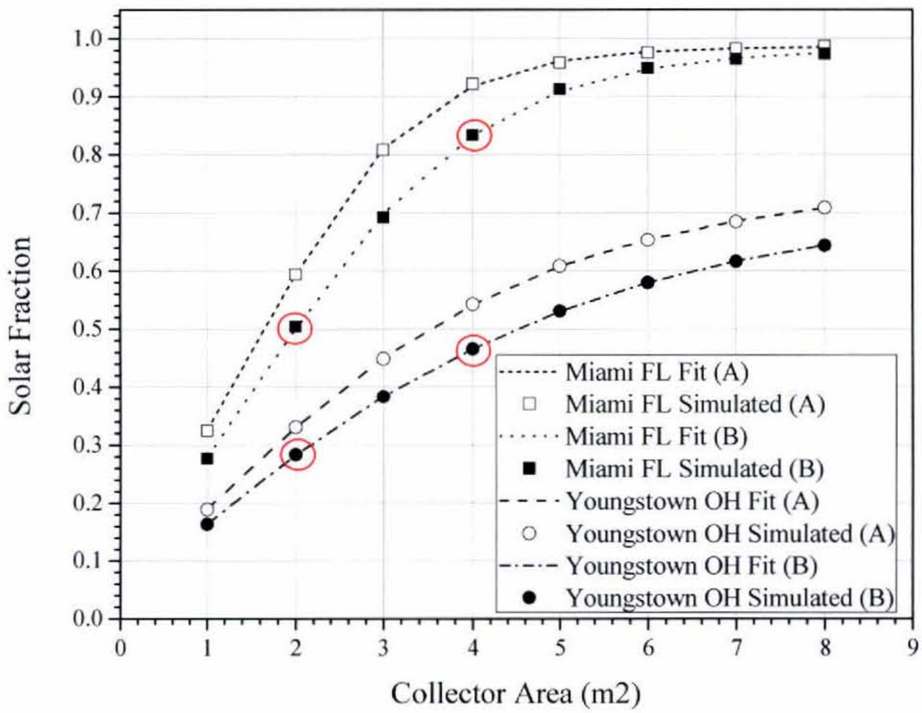


Figure IV.10 Non-linear Fit of Data with Generalized Logistic Growth Model

Not only is there a non-linear dependency of the SDHW system's performance on collector area, but the slope of this dependency also changes per collector type (Figures IV.11 & IV.12). In Figure IV.11, the performance of all five collector types is shown per increasing collector area for the 302 l/d load size in Minot, ND. The two evacuated tube collectors, shown in red, have the worst performance at $1m^2$ of area. Yet, as area is increased the two evacuated tube collectors become more efficient than both the CPC and Flat Plate-B collectors. With the 151 l/d load size, Figure IV.12, the Evacuated Tube-A collector actually becomes the most efficient by $8m^2$.

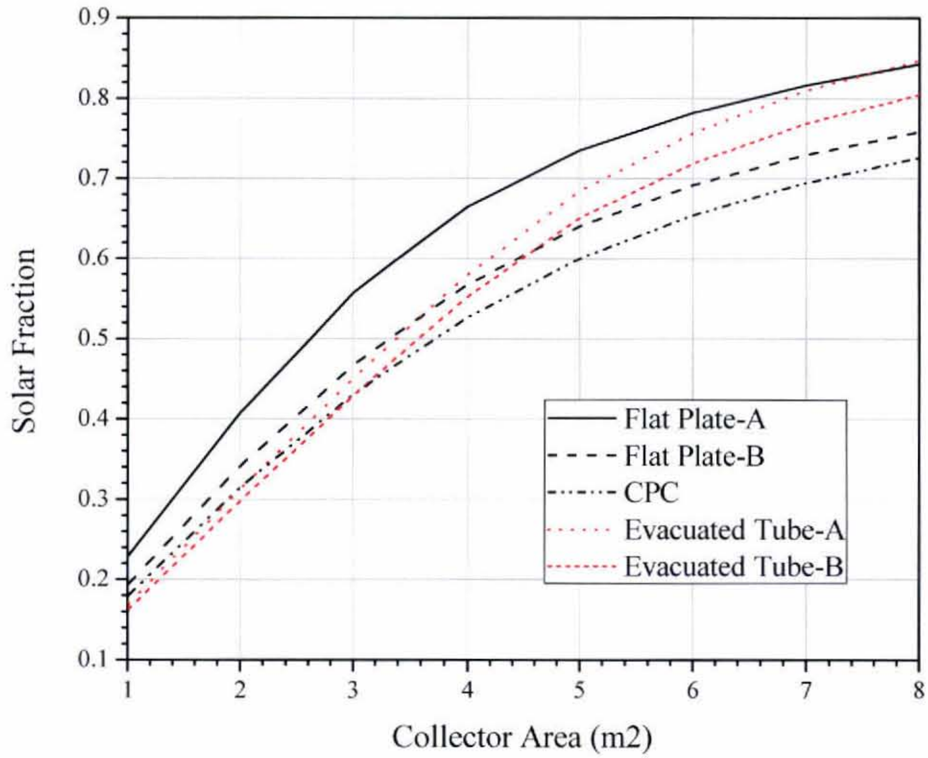


Figure IV.11 Different Performance Slopes for Each Collector Type Across Area for 301 l/d Load

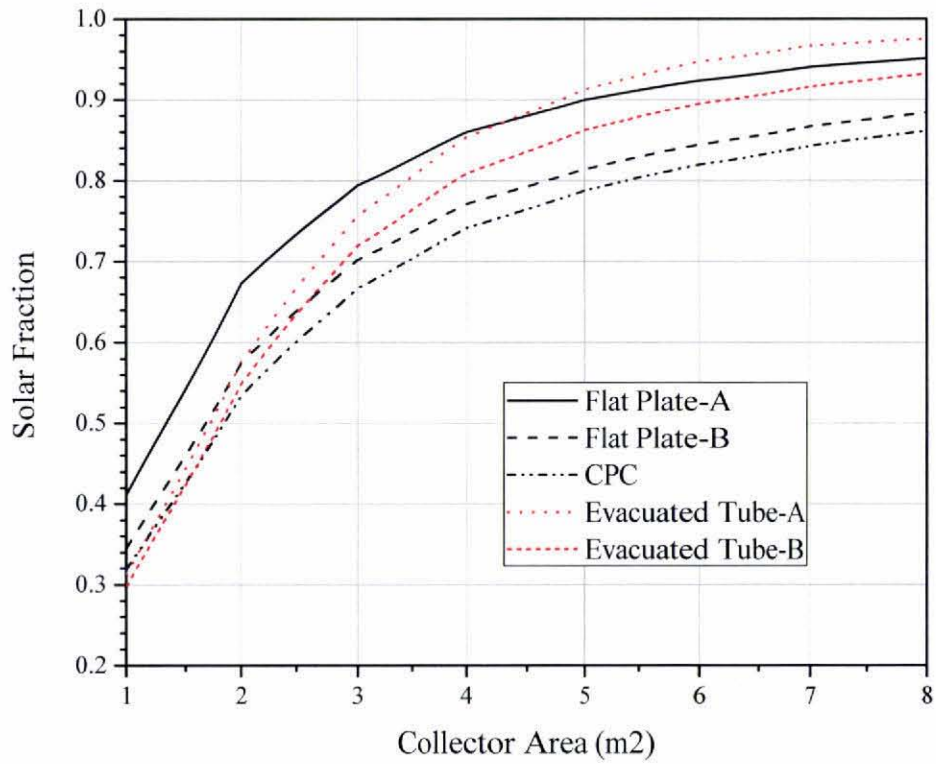


Figure IV.12 Different Performance Slopes for Each Collector Type Across Area for 151 l/d Load

Also, Figures IV.11 and IV.12 show that the performance of the SDHW system is affected by how much of the total load is being met, or in other words how close the system is to meeting the limit of performance for that location. For instance, at $4m^2$ the Evacuated Tube-A collector will meet 58% of the total load for the 302 l/d demand and the Flat Plate-A collector will meet 66% of the demand. Yet, for the 151 l/d demand scenario, at $4m^2$ the Evacuated Tube-A collector will meet 83% of the total demand whereas the Flat Plate-A collector will meet 85% of the demand. Thus, the relative comparison between SDHW systems is dependent on the load size that it is being evaluated for.

IV.3 Results

The objective of this research is to define a methodology or procedure that can unbiasedly show the optimal SDHW system for a range of conditions by taking into account the dynamics of SDHW systems that were found in the earlier part of this work. For example, lighting energy use can be regulated effectively without limiting the engineer's ability to design for specific situations. A common procedure is to allow for an allotted number of watts per meter squared of floor space in a building. Thus, the total energy load for lighting has an upper limit, but the engineer is still left with the ability to create customized lighting schemes. In other words, this type of regulating of energy use does not restrict the technology and therefore possibly smother further advances in energy efficient lighting technology and lighting design. SDHW systems would require the same characteristics for any energy efficiency requirements. Yet, their system dynamics are quite more complex as just examined.

Based on the present study, there are four parameters, i.e. four degrees of freedom, which any rating calculation of SDHW system should be able to account for. They are climate, load size/ temp, technology type, and collector area. As stated before, a rating procedure that could be used by building codes must be able to appropriately lower energy use fairly across different climate zones as well as different demand sizes as based on number of occupancies per dwelling. A rating procedure to be used to set levels of minimum efficiency for the product such as ENERGY STAR® or federal minimum product efficiency ratings, is concerned with all the same criteria as for building codes yet doing it with efficient products. As follows, a new rating procedure is proposed.

IV.3.1 Proposed Rating Procedure

The proposed rating procedure is discussed in detail below. The format of this procedure is modeled after the regulatory language that is used in energy efficiency building code as well as for defining energy efficient HVAC installation procedures. The overall rating procedure and then implementation structure would follow the following criteria.

1. All testing and/ or calculations should be done using a proven average demand profile for domestic hot water use such as the one used in this research, which is the ASHRAE demand profile and is shown in Appendix VI.2 (ASHRAE 2004-2).

2. The water demand set point temperature used in the calculations procedures should be no higher than actual use. The set point temperature of 48.88°C is used for this research and following examples.

3. Since SDHW systems performance is shown to be dependent on climate, representative climate categories should be used to rate system performance and thus system criteria per location. Using a location's average annual clearness index, average annual solar radiation, average ambient and incoming water

temperatures, the appropriate representative climate category can be calculated to figure out which performance criteria is applicable to that location. Representative climate categories are already used to define criteria for HVAC systems and windows. Thus, a new set of climate categories needs to be used to define SDHW system criteria.

4. The concept of Standard Capacity Ratings, such as evaluated for air-conditioning systems should be used to evaluate SDHW systems. For example, air conditioning systems are tested based on manufacture's stated level of heat removal rate. They are not all tested at just one arbitrary load size. Therefore, an air-conditioning system that is designed to remove heat from a smaller room is not incorrectly compared to a system that is designed to remove the heat from an area twice the size.

5. The number of occupancies per dwelling calculation as currently used by ASHRAE 90.2 (ASHRAE 2004-2) should be used in SDHW system requirements. The calculation taken from page 21 of ASHRAE 90.2 (2004-2) states as follows:

    i. "NP = number of people in living unit; if exact information is unknown, estimate as follows: (1.0)(NSR) for single-family detached and manufactured (mobile) homes with one to four sleeping rooms, plus (0.5)(NSR) for each sleeping room beyond four, or (1.25)(NSR) for multifamily buildings with one to four sleeping rooms per dwelling unit, plus (0.5)(NSR) for each sleeping room beyond four.

ii. "NSR = number of sleeping rooms"

6. All ratings given to SDHW systems for the purpose of evaluating systems relative to one another for minimum efficiency standards or to prove compliance with standard building codes should use ratings that label the Solar Fraction (SF) ratio with a different name. It is important to keep this distinction as the *rated* Solar Fraction for a SDHW system is not the actual Solar Fraction for any given situation. This number would be valid only for standard test (rated) conditions. Therefore it would be valid for comparison only to other systems *rated* for the same demand size (capacity rating) and representative climate category. The example label created for this research is the acronym for Relative Solar Rating or "RSR". The definition of RSR is shown in equation IV.2:

$$RSR = (SF)_{at\ standard\ test\ conditions} = \frac{(Q_s)_{stc}}{(Q_{del} + Q_l + Q_{par})_{stc}} \qquad \text{(IV.2)}$$

Where: $(Q_s)_{stc}$ = Solar Energy provided at standard test conditions

$(Q_{del})_{stc}$ = Energy delivered at standard test conditions (demand)

$(Q_l)_{stc}$ = Energy losses at standard test conditions (heat losses)

$(Q_{par})_{stc}$ = Parasitic energy required at standard test conditions (i.e. pumping power)

7. The requirement should be written in a way that accounts for the fact that the SDHW system's performance is dependent on the collector's area. Thus, a double requirement should be used in order to account for collector area. For instance, the requirement for the incorporating SDHW systems could read:

    i. "0.5 RSR or $2m^2$ gross collector area, whichever incorporates the smaller amount of area"

For this case if just a 0.5 RSR value was required, a very inefficient SDHW system could be installed by using a large collector area. If just the $2m^2$ gross collector area was required, again a very inefficient system could be installed as long as it had at least a $2m^2$ area. Thus the double requirement establishes not only the requirement to install a SDHW system but puts requirements on the efficiency of the system installed. Furthermore, the double requirement can take into account the difference in a system's performance across different climate zones. It would be economically unreasonable to require every location to install a system meeting the same RSR but to have the upper limit be a specified collector area would ensure the upper limit economically as well. In this example the most a contractor would have to spend would be for a $2m^2$ system.

IV.3.2 CASE STUDY

An example is used to explain how to calculate and use RSR ratings for SDHW systems. This example will go through the procedure for incorporating RSRs into building codes first and then move on towards incorporation in minimum energy efficiency regulation criteria.

IV.3.2.1 Building Code Criteria

1. Representative climate profiles will have to be established by the rating body. Then any location could calculate what 'climate profile" they match by using the multiple linear regression analysis. For simplicity, two of the locations' climate profiles used in this research will be used as representative climate categories in this example to depict the high and low ends of the range of locations studied. These two locations will be Miami, FL for the high end and Caribou, ME for the low end.

2. Standard Capacity Rating Categories will have to be established by the rating body. Table IV.6 shows a set of possible Standard Capacity Rating Categories that could be used. For this example the load size of 301 l/d will equate to a NP equal to 4.

Standard Capacity Rating	NP	Daily Load Size (l/d)	Description
A	2	151	Small household - half of average
B	**4**	**302**	**Average household - single family home**
C	8	604	Duplex housing
D	20	1,510	Small Apartment - multi-family
E	40	3,020	Medium Apartment - multi-family

Table IV. 6 Proposed Standard Capacity Ratings for SDHW Systems

3. Individual SDHW systems are tested to find RSRs for the appropriate Standard Capacity Rating Category for both representative climate profiles. For this example SDHW systems for use in the B Standard Capacity Rating Category will be discussed. Table IV.7 shows the RSR values for 15 example SDHW systems for both the high and the low climate profiles.

Standard Capacity Rating	SDHW System			Relative Solar Rating (RSR)	
	Name	Type	Area (m^2)	Climate Zone - High	Climate Zone - Low
B	S1	Flat Plate-A	2	0.666	0.346
B	S2	Flat Plate-B	2	0.567	0.292
B	S3	CPC	2	0.531	0.268
B	S4	Evacuated Tube-A	2	0.492	0.271
B	S5	Evacuated Tube-B	2	0.488	0.258
B	S6	Flat Plate-A	4	0.970	0.582
B	S7	Flat Plate-B	4	0.913	0.490
B	S8	CPC	4	0.878	0.449
B	S9	Evacuated Tube-A	4	0.908	0.502
B	S10	Evacuated Tube-B	4	0.905	0.476
B	S11	Flat Plate-A	7	0.996	0.760
B	S12	Flat Plate-B	7	0.990	0.663
B	S13	CPC	7	0.984	0.621
B	S14	Evacuated Tube-A	7	0.997	0.753
B	S15	Evacuated Tube-B	7	0.998	0.719

Table IV. 7 RSR Values for 15 Example SDHW Systems

4. There is now enough information to show incorporation of SDHW Systems in building codes. Example regulatory language could read as follows:

"shall incorporate a SDHW system with a RSR value of 0.5 or higher OR a collector area equal to the Number of Persons (NP) times $0.5m^2$; whichever results in the smaller total area. The SDHW system must be rated in the Standard Capacity Rating category that corresponds to the NP calculated or higher. (i.e. NP=4 is category B, NP=5-8 is category C, etc.)"

Thus, for Climate Zone – High using a NP=4, the SDHW systems S1, S2, S3, S6, S7, S8, S9, S10, S11, S12, S13, S14, and S15 would meet the requirement (Table IV.7). Then the RSR required value of 0.5 could be divided by each systems' area to result in the smallest system areas to meet the criteria; S1, S2, and S3; i.e. 0.5/2 = 0.25, 0.5/4 = 0.125, and 0.5/7 = 0.071, with 0.25 being the largest number. Then, one more calculation is done by dividing each of those systems' RSR value by their area to result in the most efficient system to meet the requirement; S1; i.e. 0.666/0.2 = 0.333, 0.567/2 = 0.283, and 0.531/2 = 0.266, with 0.333 being the largest number.

For Climate Zone – Low using a NP=4, none of the $2m^2$ systems result in a RSR value of 0.5 (Table IV.7), yet a system having only a $2m^2$ area is needed to comply with the criteria. Therefore, only one additional calculation would be done by dividing each of those systems' RSR value by their area to result in the most efficient system to meet the requirement; which is again S1.

Therefore, it is shown that for this type of regulator language a separate RSR value per climate zone is not needed when a dual path approach is taken to meet the criteria, i.e. RSR value OR collector area. In fact a separate RSR value per climate zone may be undesired as this approach will assure the consumer will not have to spend more money on a system simple because of their location. In this example both locations could result in the same system purchase (not taking into account actual economics of each system).

5. As it could be seen as unfair to require the inclusion of one type of technology, i.e. SDHW systems, an additional statement could be included addressing this weakness. In the same manner as currently allowed by ASHRAE code (2004-1, 2004-2), the calculated annual energy cost (AEC) can be used to show energy efficiency compliance. So to apply this same concept to the incorporation of SDHW systems, the resulting AEC of the hot water demand is calculated for both the SDHW system being installed and not being installed. Thus, the difference in the demand being met without solar and with the required amount of solar is found. It is this difference that can be allowed to be met by any other on-site non-fossil fuel based means, i.e. heat pump, geothermal, wind, microhydro, to count as complying as well.

IV.3.2.2 Minimum Energy Efficiency Criteria

1. The next level of energy efficiency as promoted by ENERGY STAR® or minimum efficiency levels would incorporate regulatory language that was stricter. Example regulatory language could read as follows:

"shall incorporate a SDHW system with a RSR value as stated for appropriate climate zone in Table IV.8 or higher and has a collector area equal to or smaller than the Number of Persons (NP) times constant α as stated for appropriate climate zone in Table IV.8. The SDHW system must be rated in the Standard Capacity Rating category that corresponds to the NP calculated or higher. (i.e. NP=4 is category B, NP=5-8 is category C, etc.)"

Standard Capacity Rating	Climate Zone - High		Climate Zone - Low	
	RSR	α (m^2)	RSR	α (m^2)
A	0.95	1	0.75	1.75
B	**0.95**	**1**	**0.75**	**1.75**
C	0.95	1	0.75	1.75
D	0.95	1	0.75	1.75

Table IV. 8 Example RSR and α Values for 2 Climate Zones

Thus, for Climate Zone – High using a NP=4, the only system that meets this qualification would be S6 (Table IV.7). For Climate Zone – Low using a NP=4, the two systems that meet the qualifications would be S11 and S14 (Table IV.7). Thus for this type of regulation criteria a dual path approach is also advantage as it promotes a large percentage of the hot water load to be meet with solar as well as promotes using the most efficient SDHW systems available to do that.

CHAPTER V
CONCLUSIONS

After an extensive literature review, a verification and validation routine, and individual component optimization were conducted, a comprehensive set of SDHW system simulations were created. This data was evaluated and correlated using regression analysis to find the underlying dependent variables. This understanding gave a basis for the proposed SDHW rating procedure using a ratio termed Relative Solar Rating or "RSR". It was shown, using an example how to apply the new RSR methodology regarding energy efficiency requirements as well as within possible building code language.

The most important parameter with any SDHW system rating methodology has been shown to be the solar collector's area. The proposed method shows how to account for this parameter. It was also discussed how the current approach has negative limitations in it accuracy by not accounting for the area.

Another significant parameter, location and thus local climate conditions, was addressed to account for its effects within a nationally valid methodology by using multiple variable regression. By using average annual clearness index, average annual solar radiation, average ambient and incoming water temperatures for a range of locations, representative climate categories can be created to sort SDHW systems' performance per different climate conditions. This method significantly develops the opportunity to apply locally significant and appropriate SDHW system regulations on a nationally-based scale. As discussed earlier the ASHRAE climate zones (ASHRAE 90.2) were found unable to be used for applying requirements for SDHW systems. Thus, this new set of climate categories is an important supporting feature of the proposed rating procedure.

One of the defining characteristic of the proposed rating procedure is that it is a dual requirement incorporating both an RSR value and a collector area. This enables a way to require the implementation of efficient SDHW systems. In turn this would create the incentive for the industry to develop more and more efficient systems, promoting advancements in the field.

Through the regression analysis is was also found that efficiency of a SDHW system becomes less important as the climate becomes more favorable for the operation of the system. In other words it is more important to have efficient SDHW systems in unfavorable climates as the climate is the limiting factor on performance as favorable climates limiting factor is simply the demand size.

In summary five contributions to the field have been made as listed below.

- A new set of climate categories are defined for SDHW system performance

- The implication of comparing SDHW systems of different collector areas

- The varying importance of the efficiency of SDHW systems with respect to location

- A new term for the SF of a system found at standard test conditions titled the "Relative Solar Rating" or RSR

- A proposed rating procedure using a unique dual criteria of both RSR and area

This research tries to bring SDHW system standards up to par with long-established heating and air conditioning equipment standards. The solar thermal industry has proven in the last thirty years to offer technically as well as economically viable products to the marketplace. The regulating standards now need to mature to the current level of available technology.

The technology discussed, SDHW systems, is currently one of the forerunning ways that the United States can use to change our nations' current faulty energy-use excursion. The importance of having scientifically valid procedures for measuring and regulating SDHW systems has never been more immense than the current time. This is because a solution to a problem will never work unless it is appropriately designed, applied, and built from the beginning. This beginning in many aspects is when regulating standards are written.

Solar power has the possibility to save immense amounts of fossil fuel from needing to be consumed in the United States as well as around the world. In turn real change can be made for the better in remedying our detrimental environmental, economical, and social problems as they relate to societies' energy use.

CHAPTER VI
APPENDIX

VI.1 Monthly Water Temperature Data

Miami, FL (°C)

January 19.6	February 20.3	March 22.1	April 24	May 24	June 24
July 24	August 24	September 24	October 24	November 23.1	December 20.6

Honolulu, HI (°C)

January 22.7	February 22.8	March 23.6	April 24	May 24	June 24
July 24	August 24	September 24	October 24	November 24	December 23.4

Houston, TX (°C)

January 10.2	February 12.2	March 15.9	April 20.2	May 23.6	June 24
July 24	August 24	September 24	October 20.9	November 16.1	December 11.9

San Antonio, TX (°C)

January 9.6	February 11.9	March 16.5	April 20.7	May 24	June 24
July 24	August 24	September 24	October 21.2	November 15.8	December 11.2

Arcata, CA (°C)

January 7.8	February 8.7	March 8.9	April 9.5	May 11.2	June 12.8
July 13.6	August 13.8	September 13.2	October 11.7	November 9.9	December 8

Daggett, CA (°C)

January 9.3	February 12.1	March 14.6	April 18.2	May 23.1	June 24
July 24	August 24	September 24	October 20.7	November 13.8	December 9.2

Asheville, NC (°C)

January 5	February 5	March 8.6	April 12.9	May 17.2	June 20.8
July 22.7	August 22.2	September 19	October 13.3	November 8.6	December 5

Wichita, KS (°C)

January 5	February 5	March 7.4	April 13.6	May 18.7	June 24
July 24	August 24	September 21.3	October 14.8	November 7.1	December 5

Youngstown, OH (°C)

January 5	February 5	March 5	April 8.5	May 14.2	June 19
July 21.3	August 20.4	September 16.8	October 10.7	November 5	December 5

Albuquerque, NM (°C)

January 5	February 5	March 8.3	April 12.9	May 17.9	June 23.4
July 24	August 24	September 20.3	October 13.9	November 6.8	December 5

Massena, NY (°C)

January	February	March	April	May	June
5	5	5	5.9	12.9	17.7
July	August	September	October	November	December
20.7	19.2	14.4	8.3	5	5

Helena, MT (°C)

January	February	March	April	May	June
5	5	5	6.3	11.4	16.7
July	August	September	October	November	December
20.7	19.7	13	7.3	5	5

Caribou, ME (°C)

January	February	March	April	May	June
5	5	5	5	10.5	15.8
July	August	September	October	November	December
18.6	17.1	12.1	6.2	5	5

Minot, ND (°C)

January	February	March	April	May	June
5	5	5	5.7	12.4	17.9
July	August	September	October	November	December
21.1	19.9	13.5	7.3	5	5

VI.2 Load Profile

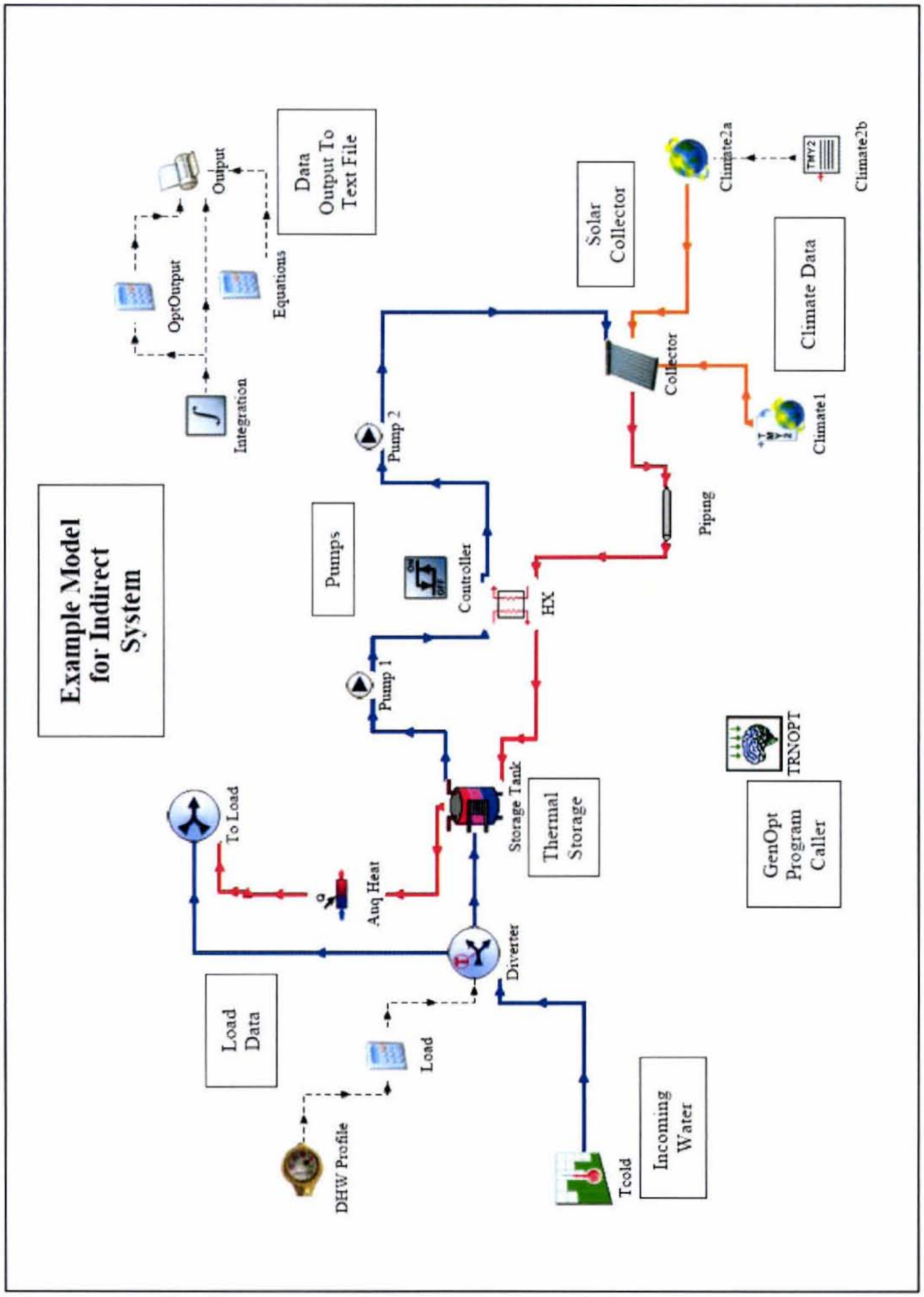
VI.2.1 Daily Domestic Hot Water Load Profile from ASHRAE 90.2 (20004-2)

Time of Day	Factor
MID - 1 a.m.	0.0085
1 - 2 a.m.	0.0085
2 - 3 a.m.	0.0085
3 - 4 a.m.	0.0085
4 - 5 a.m.	0.0085
5 - 6 a.m.	0.0100
6 - 7 a.m.	0.0750
7 - 8 a.m.	0.0750
8 - 9 a.m.	0.0650
9 - 10 a.m.	0.0650
10 - 11 a.m.	0.0650
11 - NOON	0.0460
12 - 13 p.m.	0.0460
13 - 14 p.m.	0.0370
14 - 15 p.m.	0.0370
15 - 16 p.m.	0.0370
16 - 17 p.m.	0.0370
17 - 18 p.m.	0.0630
18 - 19 p.m.	0.0630
19 - 20 p.m.	0.0630
20 - 21 p.m.	0.0630
21 - 22 p.m.	0.0510
22 - 23 p.m.	0.0510
23 - MID	0.0085

VI.2.2 Daily Load Profile Used in SRCC OG-300

The SRCC daily load profile is based on a specific amount of heat being drawn throughout a day rather than a volume of hot water. Their load profile starts at 9:30 a.m. solar time using a mass flow rate of 0.189 l/s until up to 243 l is drawn. This draw is repeated every hour for six separate draw cycles until the total amount of heat is drawn.

VI.3 Example Layout of TRNSYS Code



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