

ECONOMIC ANALYSIS OF SOLAR ASSISTED
ABSORPTION CHILLER FOR A
COMMERCIAL BUILDING

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DEDICATION

I would like to dedicate this work to my wife Shanthi Antonyraj, my sons Benedict Antonyraj and Francis Antonyraj, for their encouragement, continuous support, and sacrifices they made during my times of away from home for school work, and to my parents Athinarayanan Gnananesan and Paranjothi Samuvel Saroja a.k.a Clara Stella for their sacrifices for my education.

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ABSTRACT

Dwindling fossil fuels coupled with changes in global climate intensified the drive to make use of renewable energy resources that have negligible impact on the environment. In this attempt, the industrial community produced various devices and systems to make use of solar energy for heating and cooling of building space as well as generate electric power. The most common components employed for collection of solar energy are the flat plate and evacuated tube collectors that produce hot water that can be employed for heating the building space.

In order to cool the building, the absorption chiller is commonly employed that requires hot water at high temperatures for its operation. This thesis deals with economic analysis of solar collector and absorption cooling system to meet the building loads of a commercial building located in Chattanooga, Tennessee. Computer simulations are employed to predict the hourly building loads and performance of the flat plate and evacuated tube solar collectors using the hourly weather data. The key variables affecting the economic evaluation of such system are identified and the influence of these parameters is presented. The results of this investigation show that the flat plate solar collectors yield lower payback period compared to the evacuated tube collectors and economic incentives offered by the local and federal agencies play a major role in lowering the payback period.

TABLE OF CONTENTS

DEDICATION	iii
ACKNOWLEDGEMENTS	iv
LIST OF TABLES	viii
LIST OF FIGURES	ix
LIST OF ABBREVIATIONS	xi
NOMENCLATURE	xii
CHAPTER	
I. INTRODUCTION	1
Literature Survey	2
Mechanical Vapor Compression Cycle vs Absorption Cycle	3
Double Effect Absorption Cycle	5
II. MODELING OF ABSORPTION CHILLERS & SOLAR COLLECTORS	8
Modeling of Absorption Chillers	8
Determination of Solar Angles	11
Modeling of Solar Collectors	15
Evacuated Tube Solar Collectors	29
III. ESTIMATION OF BUILDING LOADS	31
Sol-Air Temperature	33
Mathematical Formulation for Heat transfer through walls, roofs, and Windows	36
Transfer Function Method	37
External Heat Gain from Roofs and Walls	40
Mathematical Model for the Thermal Storage Tank	42
IV. RESULTS	44

V. CONCLUSIONS AND RECOMMENDATIONS.....	59
REFERENCES.....	62
APPENDIX	
A. VARIOUS RATED CAPACITIES OF YAZAKI'S CHILLERS.....	63
B. PERFORMANCE DATA OF YAZAKI ABSORPTION CHILLERS.....	65
C. SOLAR COLLECTOR CERTIFICATION AND RATING.....	69
VITA.....	72

LIST OF TABLES

2.1	Radiational Characteristics of Glazing Materials.....	22
2.2	Impact of Glazing and Gas on U_L	24
4.1	Evacuated Tube Solar Collector Performance.....	52
4.2	Flat Plate Solar Collector Performance.....	53
4.3	Comparison of Evacuated Tube Collector and Flat Plate Collector Performance.....	54
A.1	Various Rated Capacities of Yazaki's Chillers.....	64

LIST OF FIGURES

1.1	Mechanical Vapor Cycle vs Absorption Cycle.....	4
1.2	Single Effect Absorption Cycle.....	5
1.3	Double Effect Absorption Cycle.....	6
2.1	Schematic of Single-Effect Absorption Chiller.....	8
2.2	Illustration of Latitude, hour angle and solar declination.....	11
2.3	Solar Angles for a Tilted Surface.....	12
2.4	Flat Plate Collector.....	16
2.5	Flat Plate Collector with Glass Covers.....	16
2.6	Thermal Circuit of a Flat Plate Solar Collector.....	19
2.7	Equivalent Thermal Circuit.....	19
2.8	Absorber Plate and Bonded Tube Dimensions.....	26
2.9	Energy Balance on a Fluid Element Within a Tube.....	27
2.10	Thermal Performance Curve for a Double-Glazed Flat Plate Liq. Collector.....	28
2.11	Evacuated Tube Solar Collector.....	30
3.1	Thermal Storage Effect in Cooling Load from Lights.....	35
3.2	Transient Heat Transfer in a Composite Wall.....	36
3.3	A Typical Solar Heating and Cooling System.....	42
4.1	Flow Chart for Computer Simulations.....	46
4.2	Results of Computer Simulations Produced by TABLET for the Base Case.....	47

4.3	Typical Operation of Solar Heating and Cooling Systems for Different Seasons	48
4.4	Results of Computer Simulations Produced by TABLET for Evacuated Tube Solar System	50
4.5	Variation of Operating Cost, Heat Gained, Heat Wasted with Size of Evacuate Tube Solar Collectors and Storage Tank.....	55
4.6	Variation of Operating Cost, Heat Gained, Heat Wasted with Size of Flat Plate Solar Collectors and Storage Tank	56
4.7	Variation of Payback Period with Size of Evacuated Tube Solar Collectors and Storage Tank	57
4.8	Variation of Payback Period with Size of Flat Plate Solar Collectors and Storage Tank	58
B.1	Performance Data of Yazaki Absorption Chillers.....	66

LIST OF ABBREVIATIONS

ABS, Absorber

COP, Coefficient of Performance

CON, Condenser

EVA, Evaporator

GE, Generator

HE, Heat Exchanger

SC, Series Chiller

SH, Series Heater

SP, Solution Pump

WFC, Water Fired Chiller

NOMENCLATURE

A	Area	[m ²]
A _c	Solar Collector Surface Area	[m ²]
COP	Coefficient of Performance	
c _p	Specific Heat	[kJ/kgK]
I _{dir}	Direct Solar Radiation	[W/m ²]
I _{dif}	Diffuse Solar Radiation	[W/m ²]
I _{dif,hor}	Diffuse Horizontal Solar Radiation	[W/m ²]
I _{glo,hor}	Global Horizontal Radiation	[W/m ²]
I _o	Extra Terrestrial irradiance	[W/m ²]
M	Mass	[kg]
Q	Heat capacity	[kW]
t	Time	[hr]
T	Temperature	[°C]
T _a	Ambient Temperature	[°C]
T _c	Temperature cold input fluid	[°C]
T _g	Temperature of generator fluid	[°C]
T _{s,t}	Temperature of storage tank fluid	[°C]
U	Overall heat transfer coefficient	[W/m ² K]
U _L	Overall heat loss transfer coefficient	[W/m ² K]

CHAPTER I

INTRODUCTION

During the summer months of year 2012, numerous parts in the United States experience drought due to insufficient rain and excess heat. The farmers reported loss of crops due to drought. There was a foreseeable rise in food prices due to reduction in transportation in our major water ways due to lower water levels. These were not a sudden unexpected event, but a gradually increasing effect of global warming, a phenomenon described as a rise in temperature of planet earth, the rise sea water. Scientific community attributed the release of greenhouse gases such as CO₂, NO_x, and CH₄ as primary cause of global warming. These gases are released due to increased use of fossil fuels to generate electric power and heat energy.

The energy demand is an increasing trend due to extreme warm or extreme cold conditions. Greater consumption of fossil fuels results in greater amount of greenhouse gases, exacerbating the global phenomenon. In order to prevent further rise in global warming phenomenon, drastic changes in energy use should be made or shift from use of fossil fuels to renewable energy resources such as solar or wind energy.

It was found that nearly 72 percent of electric power produced in the U.S. is employed in energy needs of buildings. One way to reduce such a large chunk of electric power demand is the use of solar energy for heating and cooling of the buildings. Solar energy is typically captured by use of solar collectors such as evacuated tube or flat plate solar collectors, through which air or water is passed through. The collector fluid then can be directly employed to heat

the building space by passing through the heating coils a device similar to the radiator of a car. In order to cool the building, chilled water is necessary which is typically produced by use of solar absorption refrigeration cycle.

There are two types of solar absorption chillers, namely single-effect and double-effect absorption chillers. Single-effect absorption chillers are less efficient with COP values in the range of 0.4 – 0.72, but requires hot water at a temp range of 70-95 °C as input energy resource, which is easily obtained from use of flat plate or evacuated tube solar collectors. Double-effect absorption chillers have COP range of 1.2 – 1.3, but requires use of hot water at temperatures around 200°C.

Literature Survey

Saman, N. F and Sa'id, W. K [1] developed a simple model of small capacity absorption chiller, based on first law of thermodynamics with lithium bromide -Water solution absorption system and concluded that COP can reach up to 0.8 for a single-effect chiller. Ghaddar, N. K, Shihab, M, and Bdeir, F [2] conducted a simulation with lithium bromide - water solution solar absorption system for a small residential application at all climatic conditions of Beirut, Lebanon. They concluded that the solar cooling system is marginally competitive only when combined with domestic water heating, which corresponded to a solar fraction of 20% to 26% for constant load extraction and 38% to 44% for a variable load extraction.

Li, Z. F and Sumathy, K [3] conducted a simulation of lithium bromide - water solution solar absorption system and concluded that the COP of the system is higher for partitioned hot water storage tank connected to the solar collectors. Assilzadeh et al [4] presented a simulation of a LiBr solar absorption system with evacuated tube collectors, in Malaysia. They concluded that for optimum performance of 3.5kW (1Ton) capacity system, to meet the peak cooling load

of 7150 MJ in the month of February, 0.8m³ hot water storage tank is essential. The solar collector area should be 35m², with the collectors sloped at 20°.

Lazzarin, R. M [5] modeled a solar absorption chiller system to improve the seasonal performance of the system, where the on-off control capacity mode can be very penalizing. In his model, the load for the chiller can be supplemented by adding a cold storage tank. Instead of a steady operating COP of 0.6-0.7, the seasonal COP can be lower than 0.2. A cold storage tank properly sized and operated can be of great help to improve the seasonal performance of the cooling plant.

Boehm, R. F [6] developed a model of single-effect solar absorption system with 0.45 m³ storage tank, 63.7 m² collector area, and a 35 kW chiller. Economic cost benefits analysis of the model showed an annual savings from \$3,448 to \$1,737 over the traditional 28 kW vapor compression system, with simple payback periods of 18 to 36 years depending on collector efficiencies and electrical rate variations. The model also proved that the system can supply over 20 kW of continuous cooling for 8 hours on a typical summer day. Bahman, A [7] presented a model of single-effect solar absorption system, for the weather conditions of Tampa, FL. His model proved that for lower lithium bromide concentration in the lithium bromide - water solution, the COP can reach up to 0.94.

Present investigation considers a 70' x 70' commercial building located in Chattanooga, Tennessee, and is heated and cooled by two types of solar collectors, in conjunction with single-effect absorption chillers.

Mechanical Vapor Compression Cycle vs Absorption Cycle

Absorption chillers use heat, instead of electrical or shaft power, to provide cooling. The mechanical vapor compressor is replaced by a thermal compressor (as seen in the Figure 1.1) that

consists of an absorber, a generator, a pump, and a throttling device. The refrigerant vapor from the evaporator is absorbed by a solution mixture in the absorber. This solution is then pumped at significantly less power to the generator where the refrigerant is re-vaporized using a waste steam or hot water heat source. The refrigerant-depleted solution is then returned to the absorber via a throttling device. The two most common refrigerant/absorbent mixtures used in absorption chillers are water/lithium bromide and ammonia/water.

Compared to mechanical chillers, absorption chillers have a low coefficient of performance ($COP = \text{chiller load}/\text{heat input}$). Nonetheless, they can substantially reduce operating costs because they are energized by low-grade waste heat, while vapor compression chillers must be motor or engine driven. Low-pressure, steam-driven absorption chillers are available in capacities ranging from 10 to 1,500 tons. Absorption chillers come in two commercially available designs: single-effect and double-effect.

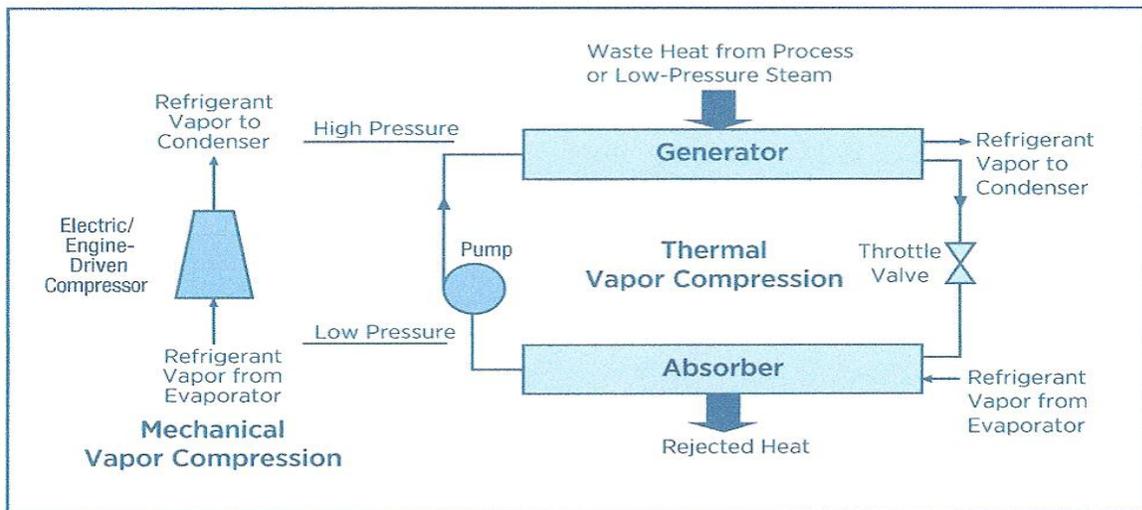


Figure 1.1 Mechanical Vapor Cycle vs Absorption Cycle

Single-effect machines provide a thermal COP of 0.7 and require about 18 pounds of 15-pounds-per-square-inch-gauge (psig) steam per ton-hour of cooling. Double-effect machines are about 40 percent more efficient, but require a higher grade of thermal input, using about 10 pounds of 100- to 150-psig steam per ton-hour.

Double Effect Absorption Cycle

Absorption chillers are generally classified as direct- or indirect-fired, and as single, double – or triple-effect. In direct-fired units, the heat source can be gas or some other fuel that is burned in the unit. Indirect-fired units use steam or some other transfer fluid that brings in heat from a separate source, such as a boiler or heat recovered from an industrial process. Hybrid systems, which are relatively common with absorption chillers, combine gas systems and electric systems for load optimization and flexibility. The single-effect cycle refers to the transfer of fluids through the four major components of the refrigeration machine - evaporator, absorber, generator and condenser, as seen in the Pressure-Temperature diagram in Figure 1.2.

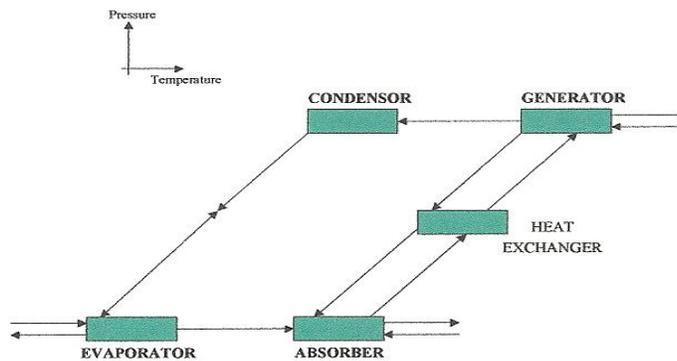


Figure 1.2 Single Effect Absorption Cycle

Single-effect absorption chillers have COPs of approximately 0.6-0.8 out of an ideal 1.0. Since the COPs are less than one, the single-effect chillers are normally used in applications that recover waste heat such as waste steam from power plants or boilers.

The desire for higher efficiencies in absorption chillers led to the development of double-effect LiBr/H₂O systems. The double-effect chiller differs from the single-effect in that there are two condensers and two generators to allow for more refrigerant boil-off from the absorbent solution.

Figure 1.3 shows the double effect absorption cycle on a Pressure-Temperature diagram. The higher temperature generator uses the externally supplied steam to boil the refrigerant from the weak absorbent. The refrigerant vapor from the high temperature generator is condensed and the heat produced is used to provide heat to the low temperature generator.

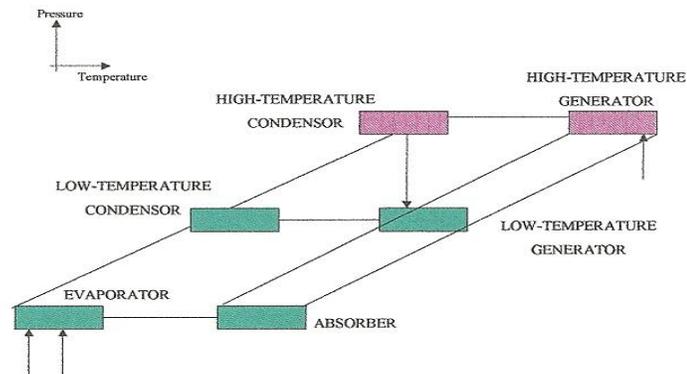


Figure 1.3 Double Effect Absorption Cycle

Double-effect absorption chillers are used for air-conditioning and process cooling in regions where the cost of electricity is high relative to natural gas. Double-effect absorption chillers are also used in applications where high pressure steam, such as district heating, is

readily available. Although the double-effect machines are more efficient than single-effect machines, they have a higher initial manufacturing cost. There are special materials considerations, because of increased corrosion rates (higher operating temperatures than single-effect machines), larger heat exchanger surface areas, and more complicated control systems.

Double-effect absorption chillers have COPs of approximately 1.0 out of an ideal 2.0. While not yet commercially available, prototype triple effect absorption chillers have calculated COPs from 1.4 to 1.6.

CHAPTER II

MODELING OF ABSORPTION CHILLERS AND SOLAR COLLECTORS

Modeling of Absorption Chillers

The absorption chillers produce chilled water by use of hot water or steam to power it. A simple schematic of the single-effect absorption chiller is shown in Figure 2.1. In a solar powered absorption chiller, hot water produced from solar collector is supplied to the generator of the chiller at temperature T_g , while the cooling water from the cooling tower (not shown in the Figure) is supplied to the absorber and condenser of the chiller at temperature T_c .

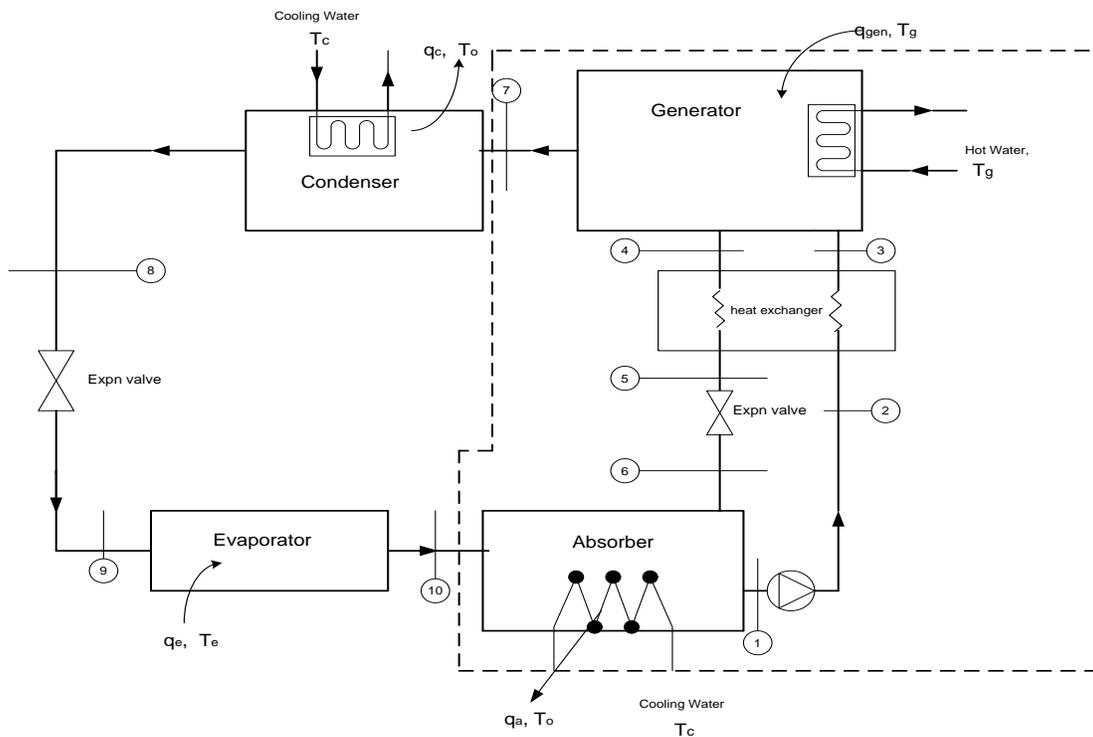


Figure 2.1 Schematic of Single-Effect Absorption Chiller

The cooling capacity of the absorption chiller is primarily dependent upon two key variables the temperature of the hot water (T_g) supplied to power the chiller and temperature of the cooling water (T_c) employed from the cooling tower to the absorber and condenser of the chiller as shown in the Figure 2.1. The rated refrigeration capacity (q_c), which is the rate of heat absorbed in the evaporator of the chiller, the rated heat supplied to the generator (q_g) at temperature T_g at rated conditions of $T_c = 85^{\circ}\text{F}(31^{\circ}\text{C})$, $T_g = 190.4^{\circ}\text{F}(88^{\circ}\text{C})$ with a mass flow rates of $m_c = 5.1 \text{ kg/s}$, $m_h = 2.4 \text{ kg/s}$ are for a ten ton capacity chiller are $q_c = 35.2 \text{ kW}$, $q_g = 50.2 \text{ kW}$.

However, the temperature of the entering cooling water temperature T_c and entering hot water temperature T_g to the generator of the chiller are dependent upon the ambient conditions for a given system. In order to estimate the performance of the chiller for an arbitrary values of the T_c and T_g are obtained by the manufacturers of the absorption chiller and are typically shown in charts as shown in Figure A.1 in the Appendix. Based on this experimental data, the performance of the absorption chiller can be modeled as follows:

The cooling capacity of the absorption chiller q_{cc} is given by

$$q_{cc} = ccf * (35.2 \text{ kW} * 3413 \text{ Btu/hr.kW}) \quad (2.1)$$

where, 35.2 kW is the rated capacity of the 10 ton capacity absorption chiller and ccf is the cooling capacity factor derived from the manufacturers catalog graphical data and is given by

$$ccf = \frac{-0.23834566 + 0.00094052 T_g - 0.0000021632 T_g^2 + 0.017467725 T_c - 0.00034409 T_c^2 + 0.00000187175 T_c^3}{1 - 0.00106584 T_g + 0.00000268177 T_g^2 - 0.0208588 T_c + 0.000122903 T_c^2} \quad (2.2)$$

The energy supplied q_g to the absorption chiller in the form of hot water, typically supplied from the storage tank is given by

$$q_g = hif (50.2 \text{ kW}) \quad (2.3)$$

where, 50.2 kW is the rated heat supplied to the absorption chiller of 10 ton capacity.

and hif is the heating input factor and is given by

$$hif = \frac{-0.25913779 + 0.003078753 T_g - 0.000011292 T_g^2 + 0.001044284 T_c - 0.000016538 T_c^2 + 0.00000012254 T_c^3}{1 - 0.01697269 T_g + 0.0000973525 T_g^2 - 0.00000018863 T_g^3 - 0.00034889 T_c} \quad (2.4)$$

The COP of the absorption refrigeration chiller then would be given as

$$COP = \frac{q_c}{q_g} \quad (2.5)$$

Since, the q_c is related to ccf in turn to the T_c and T_g , from Equation (2.2) and q_g is related to hcf which in turn is related again to the T_c and T_g , from Equation (2.4) it may be concluded that the COP of the absorption chiller is also related to the key variables T_c and T_g .

The hot water supplied to the absorption chiller is provided by the solar collectors. The solar collectors are typically made of copper plates embedded with copper tubes typically placed on the plate in serpentine form through which collector fluid such as glycol solution is circulated to get heated up by the black coated absorber copper plate. The absorber plate is encased in a steel frame backed by the insulation to reduce the thermal loss from the back plate and the front of the absorber plate is covered by one or two glass cover to reduce the thermal loss due to convection and radiation. The performance of the solar collectors is typically expressed in terms of useful energy gained by the collector fluid and is typically dependent upon key variables such

as the magnitude of incident solar flux called solar insolation I_t , and others. In order to determine the I_t , one must determine the following solar angles.

Determination of Solar Angles

The solar energy arriving on the surface of the earth consists of two components, namely the beam or direct solar energy and the diffuse energy. The direct component is the one that exists predominantly on a clear sky conditions coming directly from the Sun in the form of beam. Due to presence of dust particles and gases such as CO_2 , NO_x , H_2O and the clouds the beam radiation is scattered in several directions and results into the diffuse radiation, which exists predominantly on a cloudy day. In the deep space beyond the earth's atmosphere, the solar energy exists only in the direct or beam form.

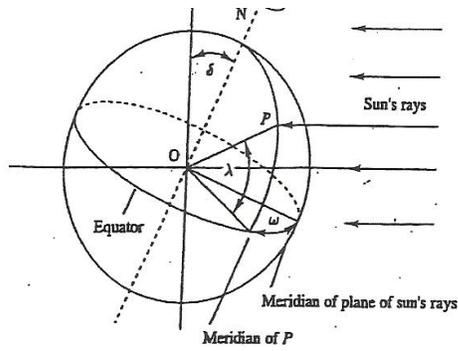
Estimation of Total Incident Solar Irradiance on a Surface on the Earth

Due to rotation of the earth about its own axis as well around the Sun, the estimate of incident solar irradiance, I_t consisting of beam and diffuse components involves determining various solar angles namely:

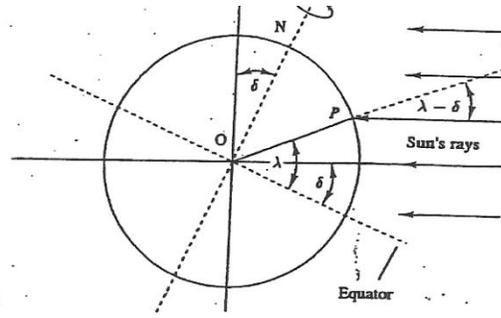
Declination angle (δ) is the angle made by the equator with sun's rays as shown in Figure 2.2

Surface latitude angle (λ) is the angle between the radius vector of the location from the center of the earth and equatorial plane as indicated in the Figure 2.2

Hour angle (ω) is the angle between the meridian of plane of sun's rays make with local meridian at the center of the earth in an equatorial plane as shown in Figure 2.2



(a) Three-Dimensional View



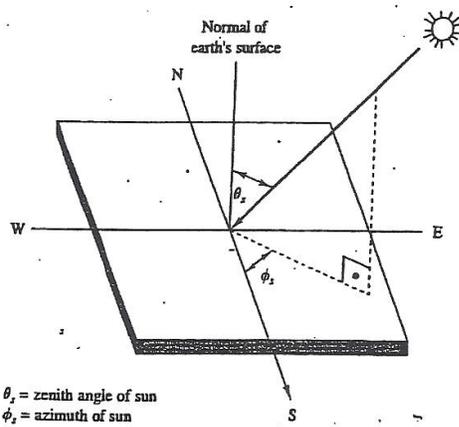
(b) Cross-Sectional View at Solar Noon

Figure 2.2 Illustration of Latitude, hour angle and solar declination

The magnitude this extra-terrestrial solar radiation I_o is given by

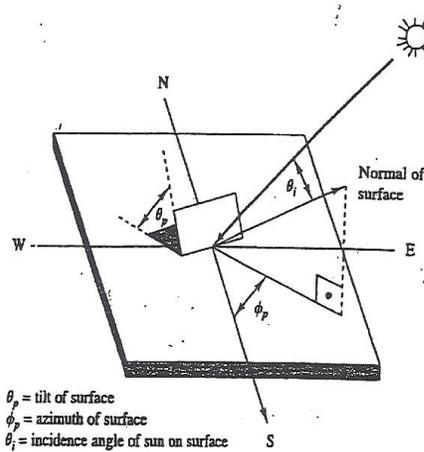
$$I_o \left(\frac{Btu}{hr \cdot ft^2} \right) = 435.2 \left[1 + 0.033 \cos \left(\frac{360^\circ n}{365.25} \right) \right] \quad (2.6)$$

Solar azimuth angle (ϕ_s) is the angle between the projection of the sun's rays on a local horizontal plane and the south direction as shown in Figure 2.3a



θ_z = zenith angle of sun
 ϕ_s = azimuth of sun

(a)



θ_i = tilt of surface
 θ_p = azimuth of surface
 θ_i = incidence angle of sun on surface

(b)

Figure 2.3 Solar Angles for a Tilted Surface

Solar zenith angle (θ_s) is the angle between the sun's rays and the normal on the local horizontal plane as shown in Figure 2.3a

Surface azimuth angle (ϕ_p) is the angle between the normal to the surface with south direction as shown in Figure 2.3b

Surface tilt angle (θ_p) is the angle between the surface and the local horizontal as shown in Figure 2.3b

Solar incident angle (θ_i) is the angle between the normal to the surface with sun's rays as shown in Figure 2.3b

The declination angle (δ) can be given as,

$$\sin \delta = -\sin 23.45^\circ \cos \frac{360^\circ (n+10)}{365.25} \quad (2.7)$$

where, n is the day of the year with January 1 being n = 1.

The hour angle (ω) can be estimated in terms of solar time (t_{sol}) from,

$$\omega = \frac{360^\circ (t_{sol} - 12h)}{24h} \quad (2.8)$$

The solar time, t_{sol} is related to the local standard time, t_{std} as

$$t_{sol} = t_{std} + \frac{L_{std} - L_{loc}}{15^\circ / hr} + \frac{E_t}{60 \text{ min} / hr} \quad (2.9)$$

where t_{std} = local standard time

L_{std} = longitude of the standard time, for United States, Eastern = 75° , Central = 90° ,

Mountain = 105° , Pacific = 120° .

L_{loc} = longitude of the location in degrees.

E_t = equation of time is the difference between the solar noon and noon time based

on local Time and it varies over the year.

It may be noted that solar noon refers to the time when sun reaches the highest point in the sky.

The equation of time E_t is obtained from

$$E_t = 9.87 \sin\left(\frac{360^\circ (n-81)}{364}\right) - 7.53 \cos\left(\frac{360^\circ (n-81)}{364}\right) - 1.5 \sin\left(\frac{360^\circ (n-81)}{364}\right) \quad (2.10)$$

The solar zenith angle (θ_s) as shown in the Figure 2.3 can be estimated from,

$$\cos \theta_s = \cos \lambda \cos \delta \cos \omega + \sin \lambda \sin \delta \quad (2.11)$$

Now, the solar azimuth angle (ϕ_s) in terms of solar zenith angle (θ_s) is obtained as follows

$$\sin (\phi_s) = \frac{\cos \delta \sin \omega}{\sin \theta_s} \quad (2.12)$$

Finally, the solar incident angle (θ_i) is given by

$$\cos \theta_i = \sin \theta_s \sin \theta_p \cos (\phi_s - \phi_p) + \cos \theta_s \cos \theta_p \quad (2.13)$$

where, ϕ_s is the solar azimuth angle given by Equation (4.7) while the surface azimuth angle ϕ_p as shown in the Figure 4.2 is the angle made by the surface normal with south direction. The tilt angle of the surface θ_p is the angle of inclination of the surface with local horizontal surface as shown in Figure 4.3b.

Now the total incident solar load I_t is the sum of

- (i) the solar direct radiation (I_{dir}) incident normal to the surface
- (ii) the solar diffuse radiation (I_{dif}), the diffuse radiation is the radiation scattered from the surroundings and the dust particles present in the atmosphere.
- (iii) the solar radiation reflected from the ground

$$I_t = I_{dir} \cos \theta_i + I_{dif,hor} \frac{1 + \cos \theta_p}{2} + I_{glo,hor} \rho_g \frac{1 - \cos \theta_p}{2} \quad (2.14)$$

where, $I_{\text{glo,hor}}$ is the global horizontal radiation incident on the horizontal surface.

The weather stations in many major cities record hourly data consisting of I_{dir} , $I_{\text{dif,hor}}$, $I_{\text{glo,hor}}$, the ambient air temperature, the dew point temperature, the relative humidity, wind speed and direction, cloud cover factor and many other data. The meteorologists obtained the average of 25 to 30 years of such data and designated this data as the typical meteorological year (TMY) for that city. Use of such data allows a more detailed and accurate estimate of solar loads.

Modeling of Solar Collectors

Flat plate solar collectors are employed in majority of the solar energy installations in the world. They are relatively cheap and easy to maintain compared to other types of solar collectors. A typical flat plate solar collector as shown in Figure 2.4, consists of a flat box containing a black coated copper plate embedded with tubes placed in serpentine fashion or in parallel tubes as shown in Figure 2.5, and the plate is lined with the insulation at the bottom of the plate and with one or two glass or plastic covers on the top. The following derivations are presented based on the references from Duffie and Beckman and from Dhamshala's handouts presented in the class [8 and 9].

Thermal analysis of these collectors starts with recognizing the modes of heat transfer occurring at different components of the collector. The solar radiation striking the top glass cover is transmitted through the glass covers and eventually gets absorbed by the black coated absorber plate. Tiny fractions of energy striking the glass covers and absorber plate gets reflected, but the majority of the energy is absorbed by the absorber plate causing its temperature to rise. The major portion of the absorbed energy by the plate is convected to the collector fluid passing through the tubes bonded to the plate, while the remaining absorbed energy is dissipated to the

ambient air from top and bottom of the collector. The heat loss from the back is essentially controlled and reduced by the insulation placed in the back, while the heat loss from the top of the absorber plate involves natural convection and radiation between the plates. The heat transfer coefficient h_w ($W/m^2.K$) due to convection between the wind with velocity V (m/s) and the top glass cover plate of the solar flat plate collector is given by

$$h_w = 5.7 + 3.8 V \quad (2.15)$$

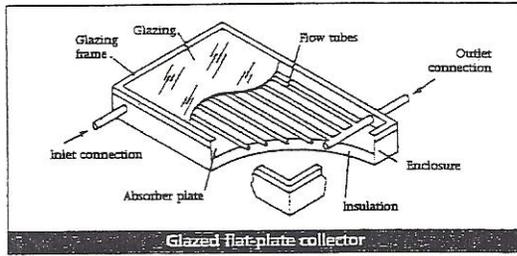


Figure 2.4 Flat Plate Collector

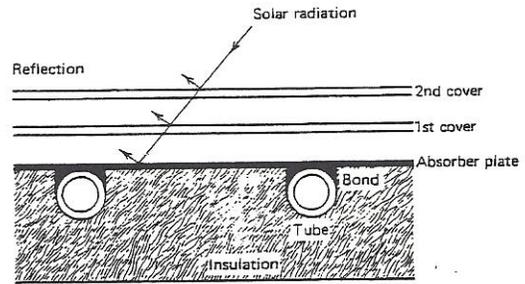


Figure 2.5 A Flat Plate Collector with Glass Covers

The net heat transfer due to radiation between two surfaces of areas A_1 and A_2 is given by

$$q_1 = -q_2 = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1 - \epsilon_1}{\epsilon_1 A_1} + \frac{1}{A_1 F_{12}} + \frac{1 - \epsilon_2}{\epsilon_2 A_2}} \quad (2.16)$$

where

q_1 = net heat transfer between surface 1 and surface 2, in W or Btu/hr

q_2 = net heat transfer between surface 2 and surface 1, in W or Btu/hr

T_1 = absolute temperature of surface 1, in K or R

T_2 = absolute temperature of surface 2, in K or R

ϵ_1 = emissivity of surface 1

ϵ_2 = emissivity of surface 2

F_{12} = view (shape) factor of surface with respect to surface 2

If the surfaces are parallel to each other and the space between these surfaces is relatively small compared to their areas, a typical case with glass covers of a flat solar collector,

then $A_1 = A_2 = A$ and $F_{12} = 1$ for this case the Equation (2.16) reduces to

$$q_1 = -q_2 = \frac{\sigma A (T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \quad (2.17)$$

The equation (2.17) can be expressed in a linear form in terms of radiational heat transfer coefficient h_r as follows.

$$q_1 = -q_2 = h_r A (T_1 - T_2) \quad (2.18)$$

where,

$$h_r = \frac{\sigma (T_1 + T_2)(T_1^2 + T_2^2)}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \quad (2.19)$$

The fraction ($\tau\alpha$) of incident energy on the solar collector is absorbed by the absorber plate and a fraction of $(1 - \alpha)\tau$ is reflected back to the cover system, which is probably more diffuse than specular. The fraction that is reflected back to the absorber plate is $(1 - \alpha)\tau\rho_d$. The multiple

reflection of diffuse radiation continues so that the energy ultimately absorbed is expressed in terms of the transmittance-absorptance product of the plate cover system $(\tau\alpha)_{cs}$ and is given by

$$(\tau\alpha)_{cs} = \frac{\tau\alpha}{1 - (1-\alpha)\rho_d} \quad (2.20)$$

where

τ = transmittance of the glass cover

α = absorptance of the plate

and ρ_d = diffuse reflectance of the glass cover system is found to be approximately equal to 0.16, 0.24, 0.29 and 0.32 for one, two, three and four glass cover systems, respectively.

The actual solar energy absorbed S (W/m^2) by the absorber plate of the solar collector is given by

$$S \text{ (W/m}^2\text{, or Btu/hr.ft}^2\text{)} = [I_t (\tau\alpha)]_{\text{beam}} + [I_t (\tau\alpha)]_{\text{diffuse}} \quad (2.21)$$

Typically the combined value for $(\tau\alpha)$ for beam and diffuse components range from 0.7 to 0.8

The thermal circuit of a three glass cover system is represented in Figure 2.6.

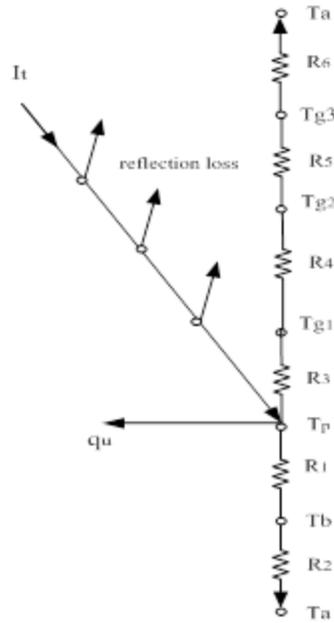


Figure 2.6. Thermal Circuit of a Flat Plate Solar Collector

where

- R_1 = the thermal resistance due to conduction of the back insulation = k_{ins}/L_{ins}
- R_2 = thermal resistance due to convection and radiation to the ambient air from back
- R_3 = thermal resistance due to convection and radiation between the absorber plate and the glass cover 1
- R_4 = thermal resistance due to convection and radiation between the glass covers 1 & 2
- R_5 = thermal resistance due to convection and radiation between the glass covers 2 & 3
- R_6 = thermal resistance due to convection and radiation between the glass covers 3, ambient air and the sky

The magnitude of the resistance R_2 is very small compared to the R_1 and therefore the overall heat transfer coefficient U_b for the back loss can be approximated as

$$U_b = 1 / R_1 = k_{ins} / L_{ins} \quad (2.22)$$

where, k_{ins} = coefficient of thermal conductivity of the insulation, W/m.K or Btu/hr.ft.R
 L_{ins} = thickness of insulation, m or ft

The thermal resistances R_4 is approximately equal to R_5 , and these resistances are not generally equal to R_3 , since R_3 is dependent upon the absorber plate emittance, which is not equal to that of the glass. The absorber plate is coated with selective coating with high value of absorptance for the solar energy and low emittance for the long wave radiation loss from the plate.

Practical limit for number of glass covers is three with most systems use only one or two.

The thermal resistance R_3 can be given as

$$R_3 = \frac{1}{h_c + h_r} \quad (2.23)$$

where

h_c = coefficient of heat transfer due to convection, W/m².K or Btu/hr.ft².R

h_{rp} = coefficient of heat transfer due to radiation, W/m².K or Btu/hr.ft².R

The air between the absorber plate and the glass cover is practically stagnant and therefore it involves natural convection, which is controlled by the magnitude of Grashof number. A convenient relation for h_c for an air space at 45⁰ tilt angle is given by

$$h_c = 1.14 \frac{(\Delta T)^{0.31}}{(L)^{0.07}} \quad (2.24)$$

where

ΔT = temperature difference between the parallel plates in $^{\circ}\text{C}$

L = plate spacing in cm

the h_r can be estimated from Equation (2.19) by substituting the T_p for T_1 and $\varepsilon_p = \varepsilon_1$ and $T_{g1} = T_2$ and $\varepsilon_g = \varepsilon_2$

Similar procedure is employed in determining the thermal resistances, R_4 and R_5 by substituting the temperatures of glass covers 1, 2 and 3.

The thermal resistance R_6 can be given as

$$R_6 = \frac{1}{h_w + h_{r6}} \quad (2.25)$$

The h_w is obtained from Equation 1 and the h_{r6} can be estimated from Equation (5) by substituting the T_{g3} for T_1 and $\varepsilon_g = \varepsilon_1$ and $T_a = T_2$ and $\varepsilon_g = \varepsilon_2$.

The overall heat transfer coefficient U_t for the top can be estimated from

$$U_t = \frac{1}{R_3 + R_4 + R_5 + R_6} \quad (2.26)$$

More than 98 percent of solar energy is within the wavelength band of 0.2 to 3.0 μm (microns, 1 $\mu\text{m} = 10^{-6}$ m), while the heat radiated at ordinary room temperatures is mostly infrared or long wave radiation. Normal window glass is practically opaque to long wave radiation while it allows transmission of more than 80 percent of sunlight. In order to have favorable thermal characteristics, the absorber plates have selective low e- coatings that have low emittance (< 0.1) for long-wave radiation and high absorptance (> 0.90) for solar radiation.

Optical characteristics of the commonly employed glazing (glass cover) materials are shown in Table 2.1.

Table 2.1 Radiational Characteristics of Glazing Materials

Materials	Thickness (mm)	Solar Transmittance	Long-wave Transmittance
Float glass (normal window glass)	3.9	0.83	0.02
Low-iron glass	3.2	0.9	0.02
Perspex	3.1	0.82	0.02
Ploy vinyl fluride (tedlar)	0.1	0.92	0.22
Polyster (mylar)	0.1	0.87	0.18

For a plate temperature of 100°C with ambient and sky temperatures of 10°C , plate spacing of 2.5 cm, tilt of 45° and wind speed of 5 m/s (11 mph) the U_t value varies as follows:

- (i) one cover, $\epsilon_p = 0.95$; $U_t = 8.1 \text{ W/m}^2.\text{K}$
- (ii) one cover, $\epsilon_p = 0.10$; $U_t = 4.0 \text{ W/m}^2.\text{K}$
- (iii) two covers, $\epsilon_p = 0.95$; $U_t = 4.3 \text{ W/m}^2.\text{K}$
- (iv) two covers, $\epsilon_p = 0.10$; $U_t = 2.6 \text{ W/m}^2.\text{K}$

Finally, the overall heat transfer coefficient U_L for the solar collector can be estimated from

$$U_L = U_t + U_b + U_e \quad (2.27)$$

where, U_e = overall heat transfer coefficient for the edges of the collector, which can be insignificant for a large size solar collector.

Performing the energy balance on the absorber plate gives

$$S = q_u + q_t + q_b + q_e \quad (2.28)$$

where

q_u = the rate of energy gained by the collector fluid, W, Btu/hr

q_t = the rate of energy loss by the collector from top, W, Btu/hr

q_b = the rate of energy loss by the collector from bottom, W, Btu/hr

q_e = the rate of energy loss by the collector from the edges, W, Btu/hr

For the case of negligible edge loss from the collector, the total loss from the collector would be from the top and bottom, given by

$$q_{\text{loss}} = U_L (T_p - T_a) \quad (2.29)$$

$$q_u = S - U_L (T_p - T_a) \quad (2.30)$$

The equivalent thermal circuit for the case of negligible edge loss is as shown in Figure 3b.

Typically the U_b is 7 to 10 times lower than that of U_t . On observing the Equation (2.30), one can obtain greater useful energy q_u for low values of U_L . Recently produced windows employ heavy gases in place of air space to minimize the heat convection loss and low E-coatings to reduce the radiation heat loss. Impact of these techniques on U_L can be seen from the data presented in Table 2.2.

It may be noted that the useful energy gained by the collector is related to the mean temperature of the absorber plate, which is very inconvenient to determine. In order to eliminate this parameter, one has to perform the thermal analysis of the absorber plate treating it as a fin between the centerlines of two adjacent tubes, as shown in Figure 2.7.

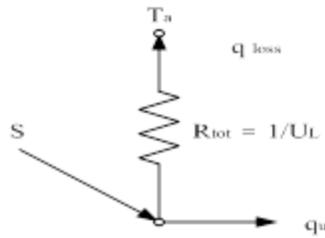


Figure 2.7 Equivalent Thermal Circuit

Table 2.2 Impact of Glazing and Gas on U_L
 Ref: Renewable Energy, Power for a Sustainable Future, by Godfrey Boyle

Window Type	U-Value ($W/m^2.K$)
Single glazed window	6
Double glazed window	3
Double glazed window with low-E-coating	1.8
Double glazed window with low-E-coating plus heavy gas filling	1.5
Three plastic films with low-E-coating plus heavy gas	0.35
Evacuated space with low-E-coating	1.0
Fiberglass insulation (10 cm thick) for comparison	0.4

It may be noted that the useful energy gained by the collector is related to the mean temperature of the absorber plate, which is very inconvenient to determine. In order to eliminate this parameter, one has to perform the thermal analysis of the absorber plate treating it as a fin between the centerlines of two adjacent tubes, as shown in Figure 2.8.

$$q_u = W F' [S - U_L (T_{cm} - T_a)] \quad (2.31)$$

where

W = width between two adjacent tubes in the absorber plate, m or ft

T_a = ambient temperature, $^{\circ}\text{C}$ or $^{\circ}\text{F}$

T_{cm} = mean collector fluid temperature, $^{\circ}\text{C}$ or $^{\circ}\text{F}$

F' = dimensionless parameter representing the ratio of the overall thermal resistance of the collector to that of the collector fluid to the ambient air and is given by

$$F' = \frac{1/U_L}{W \left[\frac{1}{U_L [D + (W - D) F]} + \frac{1}{C_b} + \frac{1}{\pi D_i h_{c,i}} \right]} \quad (2.32)$$

where

D_i, D = inner and outer diameter of the tube attached to the absorber plate, m or ft

C_b = thermal conductance of the bond employed to attach the tube to the absorber plate, for a good fabricated plate with well bonded tube, the bond resistance i.e $1/C_b$ is negligible

$h_{c,i}$ = the heat transfer coefficient between the collector fluid and inner surface of the tube, $\text{W/m}^2 \cdot \text{K}$ or $\text{Btu/hr} \cdot \text{ft}^2 \cdot \text{R}$

F = the fin efficiency and is given by

$$F = \frac{[\tan h m(W - D)/2]}{m(W - D)/2} \quad (2.33)$$

where,

$$m^2 = U_L / k \delta \quad (2.34)$$

k = coefficient of thermal conductivity of the absorber plate, $\text{W/m} \cdot \text{K}$ or $\text{Btu/hr} \cdot \text{ft} \cdot \text{R}$

δ = thickness of the absorber plate, m or ft

The heat transfer coefficient for the collector fluid can be found from

$$N_{ui} = \frac{h_{c,i} D_i}{k_f} = 0.023 R_{ed}^{0.8} P_r^{0.4} \quad (2.35)$$

where

k_f = coefficient of heat transfer due to convection, $W/m^2.K$ or $Btu/hr.ft^2.R$

R_{ed} = Reynold's number based on inner diameter = $\frac{4\dot{m}}{\pi D_i \mu}$

\dot{m} = mass flow rate of the collector fluid, kg/s or lb/hr

μ = dynamic viscosity of the fluid, $kg/m.s$ or $lb/ft.hr$

Pr = Prandtl number of the fluid = $\mu c_p/k_f$

c_p = specific heat at constant pressure of collector fluid, $J/kg.K$ or $Btu/lb.R$

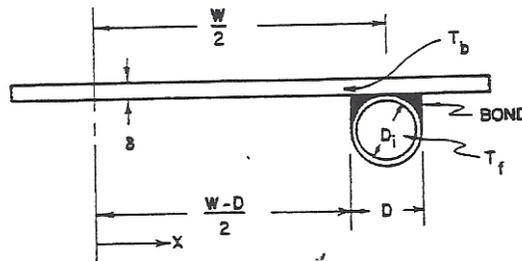


Figure 2.8 Absorber Plate and Bonded Tube Dimensions

However, the useful energy gained by the collector fluid, q_u is related to the mean fluid temperature $T_{c,m}$ which may be inconvenient to obtain. Performing the energy balance on a differential fluid element along the fluid flow direction as shown in Figure 2.9, one can determine q_u as

$$q_u = A_c F_R [S - U_L(T_{c,i} - T_a)] \quad (2.36)$$

where

$T_{c,i}$ = the inlet temperature of the collector fluid, $^{\circ}\text{C}$ or $^{\circ}\text{F}$

A_c = the solar collector surface area, m^2 , ft^2

F_R = the heat removal factor and is given by

$$F_R = \frac{G c_p}{U_L} \left[1 - \exp\left(-\frac{U_L F'}{G c_p}\right) \right] \quad (2.37)$$

where, G = mass flow rate of the collector fluid per unit area = $\left(\frac{\dot{m}}{A_c}\right) \text{kg/s.m}^2$ or lb/hr.ft^2 .

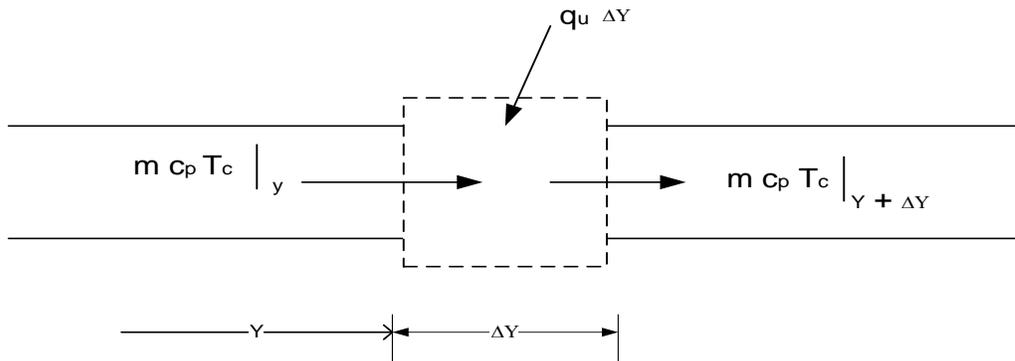


Figure 2.9 Energy Balance on a Fluid Element Within a Tube

The performance of flat plate solar collector is expressed in terms of energy collected (q_c) by the collector fluid as given by

$$q_c = A_c F_R [I_T \eta_o - U_L (T_{c,i} - T_a)]^+ \quad (2.38)$$

where

A_c = collector surface area, m^2 or ft^2

I_T = total radiation intensity on the plane of the collector (W/m^2 or Btu/hr.ft^2)

η_o = optical efficiency or the product of the transmittance and absorptance ($\tau\alpha$) of the cover and absorber plate.

U_L = overall heat transfer coefficient of the collector ($\text{W/m}^2\cdot\text{K}$ or $\text{Btu/hr}\cdot\text{ft}^2\cdot\text{R}$)

F_R = heat removal factor (= actual heat transfer / max heat transfer)

T_{ci} = fluid inlet temperature to the collector ($^{\circ}\text{C}$ or $^{\circ}\text{F}$)

+ sign denotes that negative value in the square parenthesis is set to zero indicating that collector loses more heat energy than it can collect.

Flat plate solar collector efficiency (η_c) is given as

$$\eta_c = \frac{q_c}{A_c I_T} = \frac{(mc_p)_c (T_{c,o} - T_{c,i})}{A_c I_T} \quad (2.39)$$

or alternatively,

$$\eta_c = \left[F_R \eta_o - F_R U_L \left(\frac{T_{c,i} - T_a}{I_T} \right) \right] \quad (2.40)$$

The ASHRAE Standard 93-77 (1978) is the most widely used test procedure to test the solar flat plate collectors and the test data is best curve fitted and displayed as a linear equation form as shown in Figure 2.10.

The equation (2.40) reduces to a simplified linear form based on the SRCC certification document as shown in Appendix Figure A.2

$$\eta_c = 0.638 K_{\theta} - 4.2645 \left(\frac{T_{ci} - T_a}{I_t} \right) \quad (2.41)$$

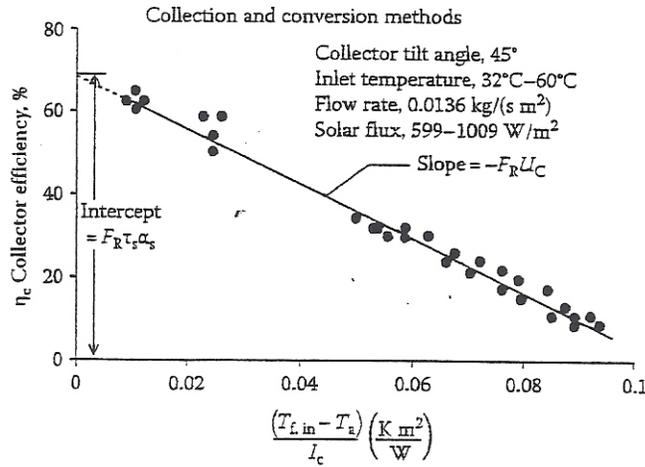


Figure 2.10 Thermal Performance Curve for a Double-Glazed Flat Plate Liquid Collector

where, K_{θ} = Incident angle modifier and is given by $= (1 / \cos \theta) - 1$ (2.42)

θ = angle of incident of the beam radiation with the normal to the solar collector

T_{ci} = temperature of the collector fluid at the inlet to the collector, $^{\circ}\text{C}$

T_a = ambient temperature, $^{\circ}\text{C}$

I_t = total incident solar flux on the tilted solar collector, W/m^2

Efficiency expression as given by Equation (2.41) is used in the computer simulations code.

Evacuated Tube Solar Collectors

Evacuated tube solar collector consists of a black coated copper tube containing a heat pipe fluid enclosed in an evacuated glass tube. Typically either 20 or 30 tubes are placed in row and the top of the copper tubes is connected to a heat exchanger placed in the top header of the collector as shown in the Figure 2.11. The collector fluid is passed through the header where its gets heated when the tubes are exposed to the solar energy.



Figure 2.11 Evacuated Tube Solar Collector (Courtesy of SunMaxx Co.)

The major advantage of evacuated tube collectors the heat loss due to convection from the hot absorber surface to the surroundings is eliminated due to vacuum conditions and the only loss is due to the radiation, which is typically not significant. Higher collector fluid temperatures can be achieved from this collector as compared to the flat plate unit and also high efficiency is possible even at a very low surroundings temperature. A copy of the SRCC document for such collector is shown in the Appendix Figure A.3 and the collector efficiency for this unit can be given as

$$\eta_c = 0.4772 K_\theta - 0.9447 \left(\frac{T_{ci} - T_a}{I_t} \right) \quad (2.42)$$

where, K_θ = Incident angle modifier and is given by $= (1 / \text{Cos } \theta) - 1$ (2.42)

θ = angle of incident of the beam radiation with the normal to the solar collector

T_{ci} = temperature of the collector fluid at the inlet to the collector, $^{\circ}\text{C}$

T_a = ambient temperature, $^{\circ}\text{C}$

CHAPTER III

ESTIMATION OF BUILDING LOADS

The transient ambient conditions of temperature, humidity and intensity of solar energy causes varying rates of heat and moisture transfer across the building envelop throughout the year. The moisture diffusion occurs due to difference in vapor pressure of the moisture present in the outside and indoor air that leaks through the cracks present in the building envelop or along the edges of doors and windows. The space in an air conditioned building is maintained at desirable conditions either for the comfort of the occupants. The desirable conditions are typically characterized by the specification of space humidity, temperature, air movement and air free from impurities, dust particles, viruses and other pollutants. A relative humidity range of around 35-55 percent and a space temperature of about 23-26⁰C (72-78⁰F) are considered to be comfortable for most of the occupants and these conditions are commonly referred to as *the design indoor air conditions*. A certain amount of outside air is ventilated through the space in order to maintain indoor air quality (IAQ) free of impurities and excess amounts of carbon dioxide. In most cases, during summers, the accumulated moisture due to perspiration of the occupants and other moisture and heat releasing sources such as occupants, appliances, and lights is mixed with the outside ventilation air of high humidity and temperature causing the space temperature and humidity to rise above the desired levels. Under these circumstances, the desirable indoor air conditions are often maintained at near constant values by supply of relatively dry chilled air to absorb the excess heat and moisture. During winters, the direction of

moisture and heat transfer would be from inside to the outside air. The difference in humidity of indoor air from the desirable level contributes to the building load known as *latent load* as opposed to the *sensible load* that refers to the heat transfer due to temperature difference across the building envelop or that produced from the indoor heat sources. The space heat gain from solar energy transmitted through windows, and the heat released from the internal sources such as occupants, lights and appliances are absorbed by the building envelope and furnishings causing their temperature to rise. Due to thermal capacitance of the envelope and its contents, there exists a time lag before these heated walls of the enclosure and furnishings dissipate heat by convection to the air in the space. The required rate of heat removal from the space air to maintain it at the summer design indoor air conditions for the given hour is called the *hourly cooling load*. Similarly, the required rate of heat to be supplied to the space air to maintain it at the winter design indoor air conditions for the given hour is called the *hourly heating load*. The maximum rate of heat removal from the indoor air to keep it at the *summer design indoor air conditions* during a year depends upon the characteristics of the building, its location and the extent of internal load sources the building has. The magnitude of this maximum rate of heat removal is commonly referred to as *peak cooling load* while the *peak heating load* refers to the magnitude of the maximum rate of heat supplied to the indoor air to keep it at the *winter design indoor air conditions*. Typically, the sizes of the cooling and heating system of the building are determined from the peak cooling and heating loads.

The weather stations in many major cities record hourly data consisting of I_{dir} , $I_{dif,hor}$, $I_{glo,hor}$, the ambient air temperature, the dew point temperature, the relative humidity, wind speed and direction, cloud cover factor and many other data. The meteorologists obtained the average of 25 to 30 years of such data and designated this data as the typical meteorological year (TMY)

for that city. Use of such data allows a more detailed and accurate estimate of hourly building loads.

Sol-Air Temperature

As presented in the last chapter that the outside air temperature (t_o) and the total solar energy incident on a unit area of a surface (I_t) on the earth are functions of time, location, day of the year, and tilt angle of the surface. These two variables directly affect the amount of heat gain through a wall or glass window of a building. In order to couple the effect of these two variables into a single parameter, *sol-air temperature*, t_e is defined to represent the outdoor air that in the absence of all radiation effects gives the same heat gain into the surface as would be the combination of incident solar radiation exchange with the sky and other outdoor surroundings, and convective heat exchange with outdoor air. Heat flux, q'' into sunlit external surface can be given as

$$q'' = q/A = \alpha I_t + h_o (t_o - t_s) - \varepsilon \Delta R \quad (3.1)$$

Based on the definition of the sol-air temperature, t_e the same heat flux can be given as

$$q'' = q/A = h_o (t_e - t_s) \quad (3.2)$$

Equating the above two equations, one can get

$$t_e = t_o + \frac{\alpha I_t}{h_o} - \frac{\varepsilon \Delta R}{h_o} \quad (3.3)$$

where t_e = sol-air temperature

t_o = current hour dry-bulb temperature

α = absorptance of surface for solar radiation

ε = emittance of the surface

I_t = total incident solar irradiation (W/m^2 or $Btu/hr.ft^2$)

h_o = heat transfer coefficient by long-wave radiation and convection at the outer surface
($W/m^2.K$ or $Btu/hr.ft^2.R$)

ΔR = difference between the long-wave radiation incident on the surface from the sky and the surroundings and the radiation emitted by the black surface at the outdoor air temperature.

$\frac{\varepsilon \Delta R}{h_o}$ = long wave radiation factor = $3.7^{\circ}C$ ($6.7^{\circ}F$) for horizontal surfaces;

= $0^{\circ}C$ ($0^{\circ}F$) for vertical surfaces

$\frac{\alpha}{h_o}$ = surface color factor = 0.026 for light colors and 0.052 for dark colors

It may be noted that for horizontal surfaces that receive long-wave radiation from sky only, an appropriate value for $\Delta R \approx 63.07 W/m^2$ ($20 Btu/hr.ft^2$) and if $\varepsilon \approx 1.0$ and $h_o \approx 17.0 W/m^2.K$ ($3.0 Btu/hr.ft^2.^{\circ}F$) then $\frac{\varepsilon \Delta R}{h_o} = 3.7^{\circ}C$ ($6.7^{\circ}F$).

Vertical surfaces receive radiation energy from the ground, sky and the surrounding walls. Typically during the day the temperature of the ground and surrounding walls may be greater than that of the outdoor air. The sky temperature is relatively low, therefore the net radiation exchange may be considered to be approximately zero giving the value of $\Delta R \approx 0$

In the above relations, the solar irradiance incident on a surface of a roof or wall as well as a relation to obtain the solar heat gain through a fenestration. In addition, the heat transfers by conduction through the wall, roof or glass window due to temperature difference between the outdoor and indoor air. The sol-air temperature has been defined in the previous section to couple the contributions of incident solar energy and outdoor air temperature.

The heat energy is also gained by indoor air from the heat releasing sources such as incoming solar energy, lights, appliances and occupants. The instantaneous *heat gain* to the space is the sum of instantaneous heat gain from all these sources. The variation of cooling load contributed by the lights with time is shown as thick lines in Figure 3.1. It may be noted from this figure that the thermal capacitance of the envelope and the space furnishings will continue to add to the cooling load even after the lights are turned off. Similar trends in thermal lag of the cooling loads occur from the incoming solar heat gain through windows and heat released due to appliances and occupants.

But the heat gain to the space air is contributed through heat and mass transfer i.e sensible and latent loads from air infiltration and ventilation, occupants, appliances, lights, solar heat gain through windows and heat conduction through walls, roofs, floors and windows.

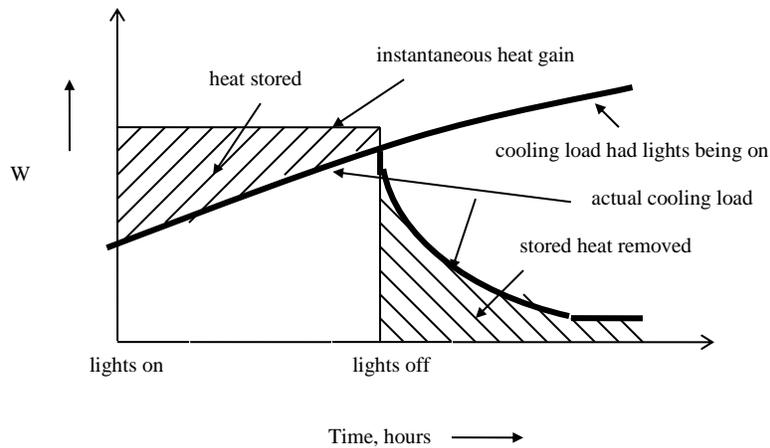


Figure 3.1 Thermal Storage Effect in Cooling Load from Lights

After adjusting for the time delay due to thermal capacitance of envelope and furnishings as seen in Figure 4.4, one may estimate the current *cooling load*, which is the rate of heat

removal from the space air in order to keep it at desired conditions. Typically this heat transfer is provided at the coil located in the duct serving the space. However, the coil capacity or the coil load may not exactly match the cooling load of the space at that moment causing slight oscillations in humidity and temperature of the indoor air. It is also to be noted that the conditioned air is brought to the building space from the air-conditioning equipment located outdoors through the ducts. The heat gain in the duct will occur from the fan motor as well as from the surrounding air which may not be conditioned. Thus the thermal capacity of the equipment or the *equipment load* sometimes it is called the *rate of extraction* will be sum of the *cooling load* and heat gain in ducts from outside air and fans.

Mathematical Formulation for Heat transfer through walls, roofs, and windows

The transient heat transfer in a composite wall is shown in Figure 3.2. The governing differential equations for a transient heat transfer a one-dimensional heat transfer due to conduction through the walls and roofs can be expressed as

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \tag{3.4}$$

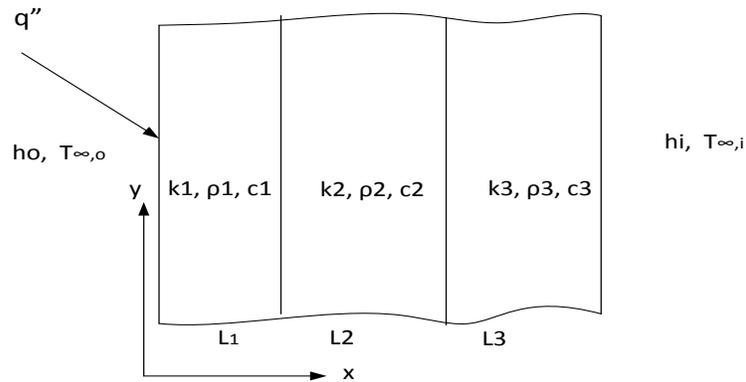


Figure 3.2 Transient Heat Transfer in a Composite Wall

The boundary conditions are of 3rd kind at $x = 0$ and at $x = L_1 + L_2 + L_3$ while at the interfaces.

The boundary conditions have same heat transfer rates on either side of the interfaces. The exact solution to such a system of partial differential equations are difficult to obtain, a reasonably accurate solution can be obtained from transfer function method, which is described below:

Transfer Function Method

The estimation of the annual or monthly energy requirements of a building to maintain it at comfortable conditions require the knowledge of building for each hour of the year. Hourly loads are done typically, since the weather data is available for most of the cities on a an hourly basis. The weather data collected for each city reflects the historical average for a period of 20 to 30 years.

Transfer function method takes into account the thermal storage effect of the solar energy, occupants, lights and equipment. For instance, the heating or cooling load Q can be considered as the response of a building or room to the effects that the temperature of the space (T_i), the temperature of the environment outside (T_o), or adjoining spaces, and the solar heat transfer rate (\dot{Q}_{sol}), etc. have on that building or room. The temperature of the space, the temperature of the environment outside, or adjoining spaces, the solar heat transfer rate, heat energy from occupants, equipment, and lighting ($T_i, T_o, \dot{Q}_{sol}, etc$) are known as the driving terms. The Transfer Function Method calculates the response of a system by making the following three assumptions [10, 11]:

1. **Discrete time steps:** all functions of time are represented as series of values at regular time steps. (Hourly in this case).

2. **Linearity:** the response of a system is a linear function of the driving terms and of the state of the system.

3. **Causality:** the response at time t can depend only on the past, not on the future.

Take into consideration, for example, the following driving term $u(t)$ (or sometimes represented as u_t) and its response $y(t)$ (or sometimes represented as y_t). To indicate the time dependence of the driving term and its response to make it more readable, a linear series relationship between the response and the driving term is assumed to be in the form:

$$y_t = -(a_1 y_{t-1\Delta} + a_2 y_{t-2\Delta} + \dots + a_n y_{t-n\Delta}) + (b_0 u_t + b_1 u_{t-1\Delta} + b_2 u_{t-2\Delta} + \dots + b_m u_{t-m\Delta}) \quad (3.5)$$

where the *time step* $\Delta t = 1$ hour and a_1 to a_n and b_0 to b_m are coefficients that characterize the system.

The coefficients a_1 to a_n and b_0 to b_m are independent of the driving term or response.

Equation (3.5) satisfies the assumption of causality because y_t depends only upon the past values of the response ($y_{t-1\Delta}$ to $y_{t-n\Delta}$) and on present and past values of the driving terms (u_t to $u_{t-n\Delta}$).

The thermal inertia of the system is taken into account with the coefficients a_1 to a_n and b_0 to b_m . If these coefficients are zero, then the response is instantaneous. The greater the number and magnitude of the coefficients, the greater the weight of the past has with the system. And, the accuracy of the model increases as the number of coefficients increases and as the time step is reduced. Hourly time resolution and a handful of coefficients per driving term will be enough for load calculations. The coefficients are called transfer function coefficients.

In the symmetric form, the relationship between u and y, as seen above, in equation (3.5) becomes:

$$a_o y_t + a_1 y_{t-1\Delta} + \dots + a_n y_{t-n\Delta} = b_o u_t + b_1 u_{t-1\Delta} + \dots + b_m u_{t-m\Delta} \quad (3.6)$$

Equation (3.6) can be generalized to the case where there are many driving terms. For example, in the case of heating and cooling load calculations, if the response of the indoor temperature T_i is determined by two driving terms, heat input into the space \dot{Q} , and the temperature outside T_o , then the transfer function model can be written as follows:

$$\begin{aligned} a_{i,0} T_{i,t} + a_{i,1} T_{i,t-1\Delta} + \dots + a_{i,n} T_{i,t-n\Delta} = & a_{o,0} T_{o,t} + a_{o,1} T_{o,t-1\Delta} + \dots + a_{o,m} T_{o,t-m\Delta} \\ + a_{Q,0} \dot{Q}_t + a_{Q,1} \dot{Q}_{t-1\Delta} + a_{Q,2} \dot{Q}_{t-2\Delta} + \dots + a_{Q,r} \dot{Q}_{t-r\Delta} \end{aligned} \quad (3.7)$$

Equation (3.7) can be considered as an algorithm for calculating $T_{i,t}$, hour by hour, given the previous value of T_i and the driving terms T_o and \dot{Q} . Likewise, \dot{Q} could be calculated as the response if T_i and T_o were given as the driving terms.

Any set of response and driving terms can be handled as above. In other-words, for any driving terms such as meteorological data, building occupancy, heat gain schedules, etc: the cooling and heating loads can be calculated hour by hour. Once the necessary numerical values of the transfer function coefficients have been calculated, the calculation of the peak loads is simple enough for a spreadsheet.

The Transfer Function Method applies a series of weighting factors, or conduction transfer function (CTF) coefficients to the various exterior opaque surfaces and to differences between sol-air temperature and inside space temperature to determine the heat gain with the appropriate reflection of thermal inertia of such surfaces.

These CTF coefficients relate an output function at a given time to the value of one or more driving functions at a given time and at a set time immediately preceding. The TFM applies

a second series of weighting factors known as Room Transfer Functions (RTF) to heat gain and cooling load values from all load elements that have radiant components. The purpose is to account for the thermal storage effect in converting heat gain to cooling load. RTF coefficients relate specifically to the special geometry, configuration, mass, and other characteristics of the defined space in order to reflect weighted variations in thermal storage effect on a time basis rather than a straight-line average.

Calculating the conductive heat gain (or loss), $\dot{Q}_{cond,t}$ at time t through the roof and walls can be done with the following relationship:

$$\dot{Q}_{cond,t} = -\sum_{n \geq 1} d_n \dot{Q}_{cond,t-n\Delta t} + A \left(\sum_{n \geq 0} b_n T_{os,t-n\Delta t} - T_i \sum_{n \geq 0} c_n \right) \quad (3.8)$$

where: A = area of the roof or wall, can be in units of m² or ft².

Δt = time step, which is 1 hour.

$T_{os,t-n\Delta t}$ = sol-air temperature of outside surface at time t - Δt

b_n, c_n, d_n are the coefficients of conduction transfer function

The following equations are used in calculating the cooling loads utilizing the Transfer Function Method:

For walls, the layers of wall construction can be identified from a table like the example above and with the R-value of the dominant material, can identify the R-value Range Number from the corresponding Wall Group Number Tables.

External Heat Gain from Roofs and Walls

$$q_{e,t} = A \left[\sum_{n=m} b_n (T_{o,t-nt}) - \sum_{n=m} d_n \left[\frac{(q_{e,t-nt})}{A} \right] - T_{rc} \sum_{n=0} c_n \right] \quad (3.9)$$

where: $q_{e,t}$ = the rate of heat gain through the wall or roof at time t

b_n, c_n and d_n = conduction transfer coefficients with units (Btu/hr.ft².°F; Btu/hr. °F; ft²),

respectively and are generally given in separate tables one for roofs, and the other for walls.

t = hour for which calculation was made

δ = time interval (1 hr)

m = number of hours for which the values are significant

e = element under analysis, roof or wall assembly

A = area of element under analysis

$T_{ot,t}$ = sol-air temperature at time t

T_{rc} = room or space air temperature at time t

With the specification of materials that make up the roof, one can substitute the conduction transfer coefficients appropriate for each roof or wall group. The Equation (3.9) estimates the heat gain through a wall or roof for any hour during the year for which the ambient air temperature and solar flux are available for that hour.

In the detailed approach, the hourly weather data is employed to determine the sol-air temperature for the hour and this sol-air temperature is applied in the above equation to estimate the heat gain and later this hourly heat gain is transformed to obtain the hourly cooling load. A detailed method using transform function method is presented in the references [10,11] The reference [11] also presents the detailed procedure to evaluate the cooling load contributed by lights, solar gain through the windows, heat release from equipment and occupants. Total contributed all of these components are added for each hour and the HVAC equipment would

match the supply air conditions to the building space to the hourly load in order to keep the space at temperature at the set value of the thermostat.

As shown in the Figure 3.3, a typical solar heating and cooling system consists of solar collectors, thermal storage tank, absorption refrigeration systems and a heating/cooling coil. The thermal storage is employed to store the heat energy during the day to meet the loads during the hours of night when no solar energy is available.

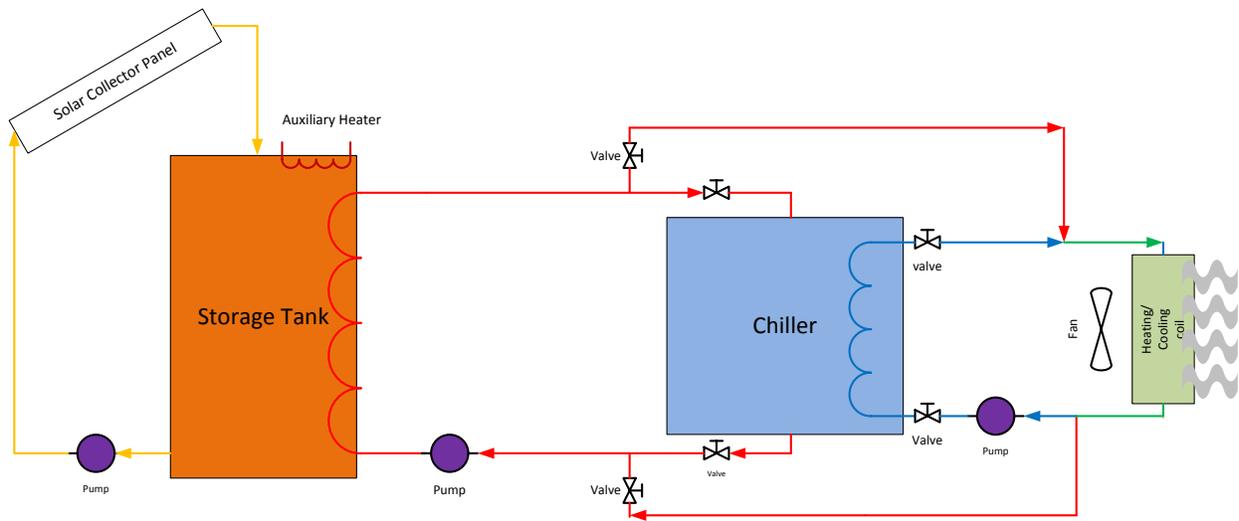


Figure 3.3 A Typical Solar Heating and Cooling System

Mathematical Model for the Thermal Storage Tank

The following equation can be employed to estimate the temperature of the **mixed storage fluid** of the storage tank at time $(t + \Delta t)$.

$$T_s^{t+\Delta t} = T_s^t + \Delta t [Q_u - Q_l - (UA)_s (T_s^t - T_a)] / (M C_p) \quad (3.10)$$

where,

Q_l = the building load for the hour (kJ) obtained from software TABLET

Q_u = the useful energy gain by the solar collector for the hour (kJ) obtained from Equation (2.39)

$(UA)_s$ = product of overall heat transfer coefficient and surface area of the storage tank, assume it to be $0.86 \text{ W/}^\circ\text{C}$ or $(0.86 \times 3600/1000) \text{ kJ/}^\circ\text{C}$

T_s = the temperature of the storage tank at time t, in $^\circ\text{C}$

M = mass of the storage tank fluid, assume it to be 795 kg

C_p = the specific heat of the storage fluid, 4.18 kJ/kg.K,

Based on the Equation (3.10), the hourly thermal storage temperature is obtained.

CHAPTER IV

RESULTS

The energy demand for heating and cooling of the building space is obtained from the transient analysis of the building envelope using the hourly weather data by use of the software TABLET developed by Professor Dhamshala. A 70'x70' commercial building with 10 occupants, an equipment load of 5 kW, a lighting load of 2 W/ft², an infiltration of 0.3 ACH and a ventilation load of 15 cfm/occupant is assumed. The roof is made of 4" low weight concrete deck, 6" of insulation and suspended ceiling with an overall heat transfer coefficient of 0.036 Btu/hr.ft².⁰F, while the walls are made up of face brick, 6" low weight concrete block, and 6" of insulation with an overall heat transfer coefficient of 0.038 Btu/hr.ft².⁰F. The south and north walls, each have a window area of 200 ft². The office has electrical equipment such as computers, copiers and others with an equivalent load of 5 kW and the lighting load during the office hours are about 2 W/ft² of floor area of 4,900 ft². The infiltration of outside air is assumed to be 0.3 ACH, a ventilation load of 15 cfm/occupant, electrical power cost of \$0.12/kWh, an electrical demand cost of \$10/kW in excess of 10 kW limit, the gas cost of \$0.90/ therm, and the state/federal subsidies of 40% of the total cost of the solar equipment are assumed for this investigation.

A building with the above characteristics with no solar equipment is considered as a base case. The base case system uses an air-conditioner to cool the building space, while a natural gas powered furnace to heat the space with an electric water heater for producing the domestic hot water. The operating cost to heat and cool the building and to meet the electrical power

requirements of operating the HVAC equipment, electrical equipment and the lighting for the space is evaluated for each hour of the year by taking into account the variation of building loads, the efficiency of the HVAC equipment for the given weather conditions of the hour. This estimation is repeated for each hour of the year as shown in the computational flow chart shown in Figure 4.1. A summary of the yearly computer simulation is as shown in Figure 4.2.

In this chart, the name of the city, where the building is located along with the latitude, longitude and the angle for the standard time is shown on the top of this Figure followed by the building envelope data containing the area of roof, walls, windows and overall heat transfer coefficient ('U') factors. The other input data such the number of occupants, electrical load, night thermostat set-backs and energy cost profiles are presented. Based on this input data, the TABLET software provides the output data as shown in the middle of the Figure 4.1. The summary of the output data falls into two categories, the first one is the building load profiles for heating and cooling of the building space on the respective Peak Heating Day and Peak Cooling Day of the year for the 24 hours on these days. The second part of the output consists of Utility Cost Analysis that presents the monthly expenses for air-conditioning (ac), the equipment operation, auxiliary heating (aux), lighting (lit), electrical demand (edmd) and for water heating (wat) followed by the total electrical costs (telc) for each month and total for the year for each

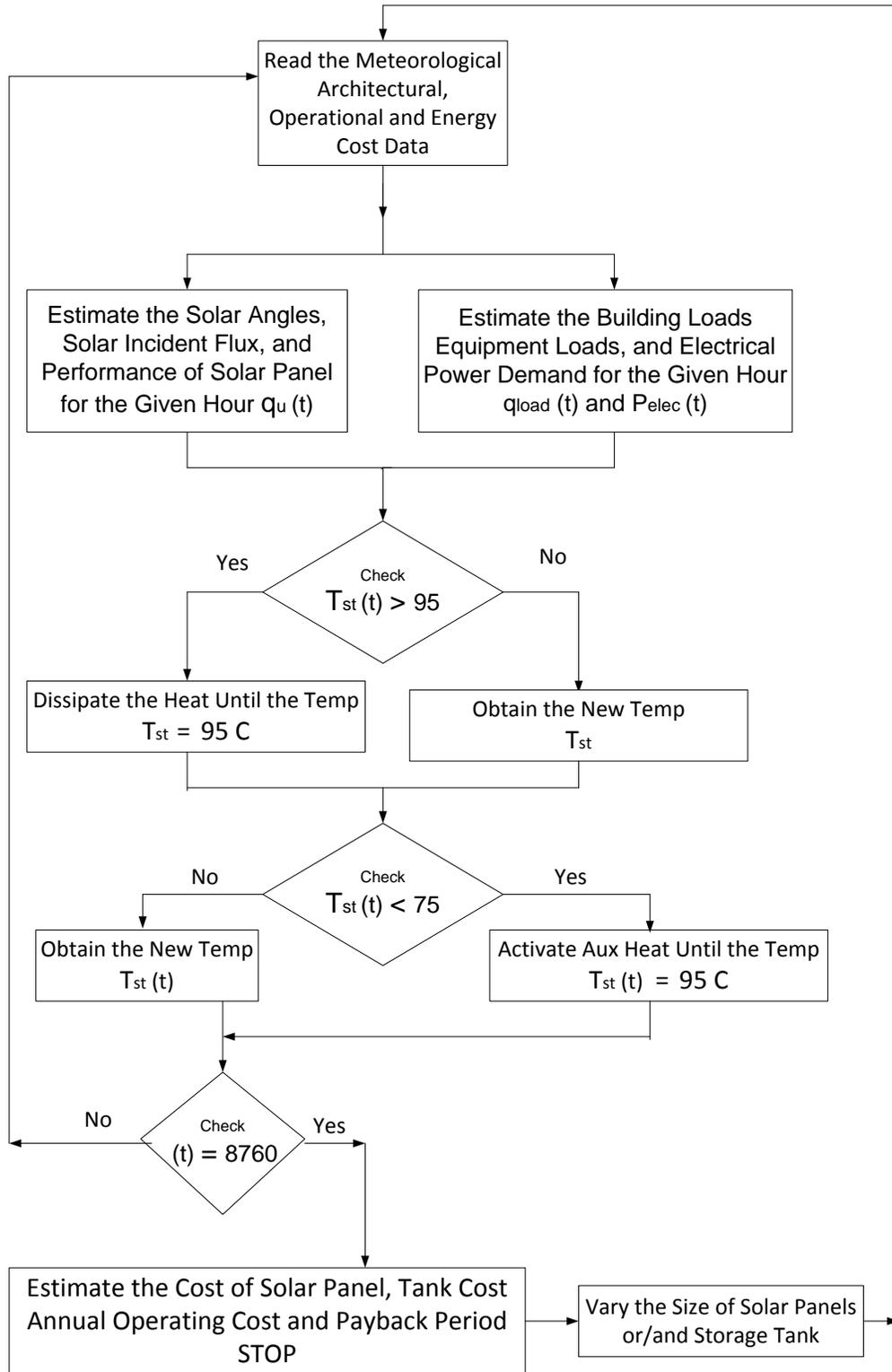


Figure 4.1 Flow Chart for Computer Simulations

category. Just below this line represents the expenses for space heating (heat) for each month and the total for the year. The term (tuty) represents the total utility cost for each month.

CHATTANOOGA TN 35 2 85 12 75
 Time 4:34:53 PM Date: 12/6/2012
 (TABLET) Transient Analysis of Building Loads and Energy Requirements Zone:1

Envelop:	Roof	East Wall	South Wall	West Wall	North Wall	Floor	NE Wall	SE Wall	SW Wall	NW Wall
Area(ft ²):	4900	700	500	700	500	4900	0	0	0	0
Glass Area (ft ²):	0	0	200	0	200	0	0	0	0	0
U-Factor(Btu/hr.ft ² .F):	0.036	0.038	0.038	0.038	0.038	0.04				
Type:	17	31	31	31	31	6				

No of Occupants (day) = 10 (night) = 0 (holidays) = 0 Renew System:1 Mode:1 Life(yrs): Util P.Fac:2 Cap Cost:5 Payback =
 Equip Elec Load, kW (day) = 5 (night) = 0 (holidays) = 0 No hrs/yr. Heat Eqmpt on: 0 Cool Eqmpt on: 0 Eqmpt Off: 0
 Elec. Lights, W/ft² = 2 (night) = 0.01 (holidays) = 0.01 DCV System: Off Zc = Energy Recovery System: Off Sen.Eff: 0.85 Lat.Eff: 0.85
 Infiltration, fraction of ACH = 0.3 Ventilation, cfm/person = 15 Hot Water Cons, gal/person/day = 2 U- of Glass(Btu/hr.ft².F) = 0.8
 S.C of Glass = 0.85 Elec Power Cost, cents/kWh = 12 Gas Fuel Cost, cents/therm = 90 Economiser: Off Elec kW limit: 10 kW cost:10
 Wint:Therm set = 72 F N. Setback = 72 F Throtl Range = F :Sum: Thrm set = 76 F N.Setback = 76 F Throtl Range = F

Pk H/lat.Load, tons = -4.4711.1 in Month = 12 on Day = 20 at hr = 7 : Peak C/maxil, tons = 10.22/1.7 in month = 6 on Day = 28 at hr = 16

Approx Recommended Cap of Heating Eqmpt, tons = 7 On the Peak Heating Day :						Approx Recommended Cap of Cooling Eqmpt, tons = 12 On the Peak Cooling Day					
Time	A.Temp	S.Load	H.Added	S.Flux	S.Temp	Time	A.Temp	S.Load	Heat Extr	S.Temp	S.Flux
hr	(deg F)	(Btu/hr)	(Btu/hr)	(W/m ²)	(deg F)	hr	(deg F)	(Btu/hr)	(Btu/hr)	(deg F)	(W/m ²)
1	23	-50767	-49579.6	0	72.2	1	75.9	33741	27948	75.1	0
2	19.94	-52215	-51006.9	0	72.2	2	75	32595	27399	75.1	0
3	19.04	-52733	-51584.3	0	72.2	3	73	28592	23336	75.2	0
4	19.94	-52665	-51564.4	0	72.2	4	73	27791	23490	75.2	0
5	19.04	-53109	-52125	0	72.2	5	73	26994	23484	75.2	0
6	19.04	-53344	-52513	0	72.2	6	72	24487	21387	75.2	0
7	19.04	-53601	-52963.4	0	72.2	7	75.9	27784	25601	75.1	0
8	19.94	-51244	-50438.7	0	72.2	8	82	59095	61493	74.6	0
9	21.92	-49233	-47951.2	0	72.2	9	84.9	93595	100231	74.1	0
10	24.98	-45772	-43481.7	0	72.1	10	90	104603	111714	73.9	0
11	28.04	-40132	-37103.6	0	72	11	91.9	110586	117038	73.9	0
12	30.02	-35577	-32202.5	0	71.9	12	95	114923	121471	73.8	0
13	33.08	-31176	-27872.9	0	71.9	13	95	118967	125092	73.8	0
14	33.98	-29313	-25751.3	0	71.9	14	97	121328	127436	73.7	0
15	35.96	-28324	-24351.2	0	71.8	15	97	121774	126794	73.7	0
16	37.04	-29574	-26518.4	0	71.9	16	98.1	122673	126785	73.7	0
17	35.06	-33778	-31103.3	0	71.9	17	97	115210	117267	73.9	0
18	33.98	-37902	-36287	0	72	18	95	110894	112504	73.9	0
19	32	-43830	-43049.3	0	72.1	19	93	79485	75741	74.4	0
20	32	-44980	-44119.3	0	72.1	20	89.1	72553	68066	74.6	0
21	30.92	-47616	-47124.2	0	72.2	21	84.9	42026	34165	75	0
22	30.92	-47687	-46438.1	0	72.1	22	84	39753	32622	75	0
23	30.02	-48048	-47277.6	0	72.2	23	82.9	38192	31372	75.1	0
24	28.94	-48535	-47136.5	0	72.2	24	80.1	37804	32377	75.1	0

	Jan	Feb	March	April	May	June	July	Aug	Sept	Oct	Nov	Dec	t/year
ac:	0	0	0	231	453	634	715	680	498	284	0	0	3495
eqp:	126	120	132	126	132	126	126	138	114	138	126	114	1518
aux:	69	58	59	53	53	58	63	61	45	53	54	68	694
lit:	250	238	262	250	262	250	250	274	227	274	250	227	3014
edmd:	75	75	75	165	166	178	177	173	167	162	75	75	1563
wat:	6	6	5	5	5	4	4	4	4	5	5	5	58
telc:	526	497	533	831	1070	1249	1335	1330	1054	915	510	488	10338
heat:	169	133	96	54	15	0	0	0	4	54	97	175	797
RevRn:	0	0	0	0	0	0	0	0	0	0	0	0	0
tuty:	695	630	629	885	1085	1249	1335	1330	1058	969	607	663	11135

Pk kW Dem: 27.82 kWh. Comp/yr: 77799 kWh Cost/yr, \$: 8776 therms/yr: 0 GasCost, \$: 0 E.D.Cost, \$/yr: 1563

Figure 4.2 Results of Computer Simulations Produced by TABLET for the Base Case

As an alternative to the base case, a solar system typically consists of either flat plate or evacuated tube solar collector with a storage tank equipped with an auxiliary back up heater, an absorption chiller and a heating/cooling coil as shown in Figure 4.3. The solar collectors produce the hot water during the day and store it in the thermal storage tank and power the absorption chiller that produces the chilled water typically supplied to the cooling coil across which a mixture of outside air plus recirculated air from the building is blown to obtain the chilled air. This chilled air is then circulated throughout the building to cool the building space. During the winter, the hot water from the thermal storage tank by-passes the absorption chiller and is directly sent to the heating coil, across which a mixture of outside air plus recirculated air from the building is blown to obtain the warm air. This warm air is then circulated throughout the building to heat the building space.

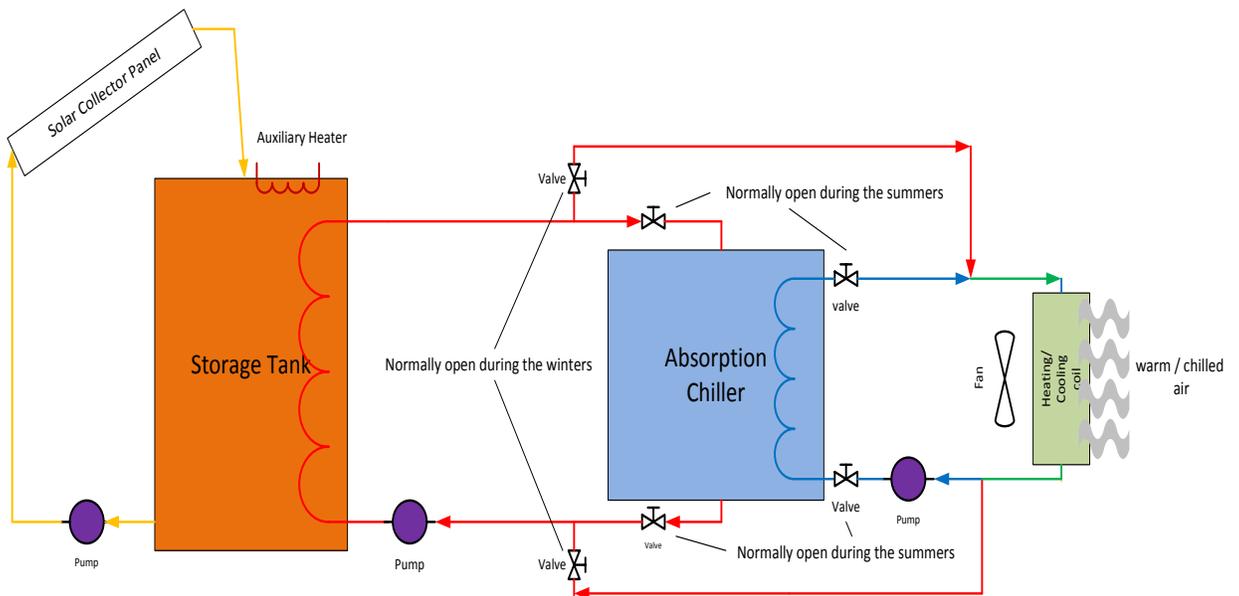


Figure 4.3 Typical Operation of Solar Heating and Cooling Systems for Different Seasons

Typically, the single-effect absorption refrigeration systems require hot water supplied to it at a temperature within the range of 70 to 95°C. Anytime the temperature of the storage tank goes beyond the 95°C, the heat energy need to be dissipated from the storage tank for a satisfactory operation of absorption chiller. The heat energy thus dissipated is wasted or termed as dumped. Although the minimum temperature of operation of the chiller is 70°C, auxiliary back up heater is activated whenever the temperature of the tank falls below 75°C for the computer simulations. In this investigation, three different sizes of thermal storage tanks are considered with diameters of 0.5, 1.0 and 1.5 m with the height of the tank in each case to be twice that of the diameter. The number of solar collector panels are varied from 10, 30 to 50 for the evacuated tube and flat plate solar collector systems. It is assumed that the panels are installed at a tilt angle of 36° from the horizontal facing the south direction for best results of solar gains during the summer and winters.

A typical summary of computer simulations obtained from the TABLET program for an evacuated tube solar panel system is as shown in Figure 4.4. The description of the output data is already discussed for Figure 4.2. However, for a system consisting of solar system additional data is presented such as number of solar collector panels, the diameter of the storage tank, capital cost of the storage tank, surface area of the storage tank for the selected diameter, the volume of the storage tank, the total thermal energy gained by the solar system over the year in terms of therms along with the total heat dissipated over the year in order to keep the fluid temperature within the tank below the 95°C limit. The payback period of the system as presented in the Figure 4.4 is obtained by dividing the total cost of the solar system by the difference in total annual operating cost compared to the base case with no solar installation. The total cost of the solar system includes the capital cost of the solar panels, storage tank, additional cost of the

absorption chiller over that of the conventional air-conditioner (assumed to be \$ 100/ton of cooling capacity) and the installation cost ((assumed to be 10% of total capital cost of solar system).

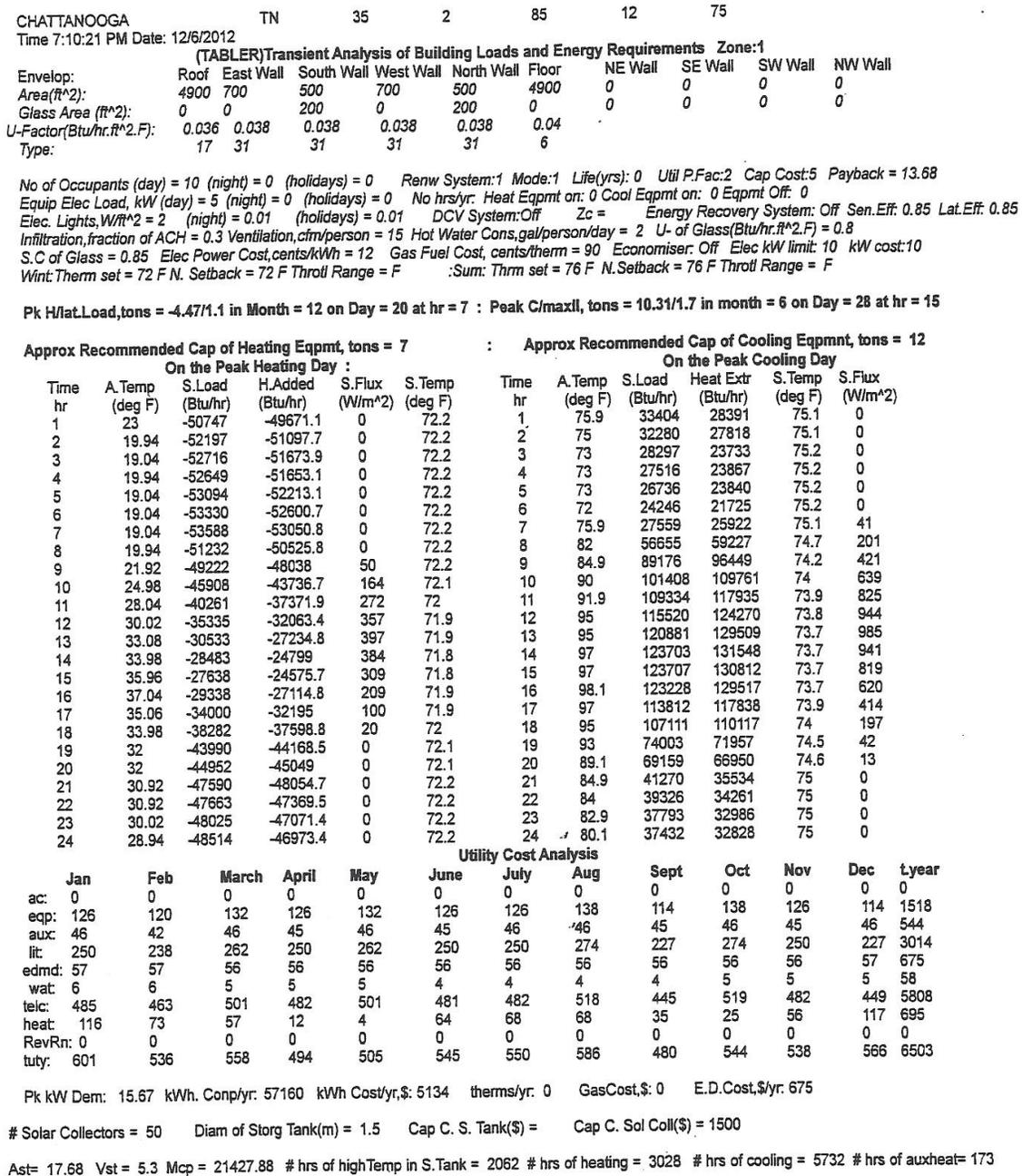


Figure 4.4 Results of Computer Simulations Produced by TABLET for Evacuated Tube Solar System

After repeating the computer simulations using TABLET software for different sizes of thermal storage tank and for the various solar collector panel sizes, a summary table for the evacuated tube collector is developed as shown in Table 4.1. A similar table for the flat plate collector is developed as shown in Table 4.2. A comparison table for both types of solar collectors is prepared and is as shown in Table 4.3.

Figures 4.5 and Figure 4.6 are obtained based on the results presented in Tables 4.1 and Table 4.2. The capital cost of the thermal storage tank is primarily dependent upon the volume of the storage tank typically expressed in gallons. The installation cost of the solar system varies from case to case, however in this research it is expressed as the percentage of the total capital cost of the solar system including that of the storage tank. The formulation to determine the payback period is shown at the bottom portion of the Tables 4.1 - 4.3. Figures 4.7 shows the variation of payback period with solar system size for evacuated tube solar collector system, while the Figure 4.8 shows the variation of payback period with solar system size for flat plate solar collector system.

Table 4.1 Evacuated Tube Solar Collector Performance

70'x70' Building; $U_r = 0.036$, $U_w = 0.038$ Btu/hr.ft ² .°F; Occupants: 10; ACH: 0.3; Ventilation: 15 cfm/occupant; Lighting: 2 W/ft ² ; Summer Peak Load: 10.3 tons, Windows: 200 ft ² on South and North Walls Energy Cost = \$0.12/kWh; Demand Cost = \$10/kW; Elect Demand limit = 10 kW; Gas Cost = \$0.90/therm; Solar Panel Cost = \$1500 /panel															
	Diameter of Storage Tank, D =0.5 m					Diameter of Storage Tank, D =1.0 m					Diameter of Storage Tank, D =1.5 m				
# Panels	Opr Cost (\$/yr)	Gas Cost (\$/yr)	Heat S.Coll (thr/yr)	Heat dump (thr/yr)	Pay back (yrs)	Opr Cost (\$/yr)	Gas Cost (\$/yr)	Heat S.Coll (thr/yr)	Heat dump (thr/yr)	Pay back (yrs)	Opr Cost (\$/yr)	Gas Cost (\$/yr)	Heat S.Coll (thr/yr)	Heat dump (thr/yr)	Pay back (yrs)
0	<i>11,135</i>	<i>797</i>	-	-	-	<i>11,135</i>	<i>797</i>	-	-	-	<i>11,135</i>	<i>797</i>	-	-	-
10	8,540	2,732	1,263	111	4.46	8,558	2,750	1,263	77.5	5.42	8,597	2,789	1,263	30.7	8.06
30	7,365	1,557	3,787	1,461	5.64	7,250	1,442	3,787	1,293	8.69	6,914	1,106	3,787	866	9.52
50	7,053	1,245	6,312	3,672	8.87	6,912	1,104	6,312	3,479	12.61	6,503	695	6,312	2,979	12.89

The numbers in italics refers to the base case with no solar installation, the other numbers are for the evacuated solar tube collector system.

Formulation for Economic Analysis

Total Capital Cost of Solar System , C_s = [({ # panels x Cost/panel}) + {(Additional cost of Absorption Chiller/ton) x (cooling capacity in tons)} + {Cost of Storage Tank}]

Additional cost of Absorption Chiller = \$ 100/ton of chiller capacity.

Cost of Storage Tank, (\$) = 6000 + { 10 [V (gal) - 500] } , where, V = volume of storage tank in gallons

Cost of Installation (\$) = typically, it is expressed as certain percentage of the total capital cost of the system without installation = $(1+C_s) \{1+(i_{\text{instal}}/100)\}$

Total Installed cost of Solar system = [({ # panels x Cost/panel}) + {(Additional cost of Absorption Chiller/ton) x (cooling capacity in tons)} + {Cost of Storage Tank}] x {1+(i_{instal} /100)}

where, i_{instal} represents the installation costs as percentage of the total capital cost of the solar panels systems C_s as given in the above equation.

$$\text{Payback Period, PB (yrs)} = \frac{\text{Total Installed Capital Cost of SolarSystem over Conventional Sytem}}{(\text{Annual Cost of Operation of Conventional System} - \text{Annual Cost of Operation of Solar System})}$$

Table 4.2 Flat Plate Solar Collector Performance Evaluation Data Sheet

70'x70' Building; $U_r = 0.036$, $U_w = 0.038$ Btu/hr.ft ² .°F; Occupants: 10; ACH: 0.3; Ventilation: 15 cfm/occupant; Lighting: 2 W/ft ² ; Summer Peak Load: 10.3 tons, Windows: 200 ft ² on South and North Walls															
Energy Cost = \$0.12/kWh; Demand Cost = \$10/kW; Elect Demand limit = 10 kW; Gas Cost = \$0.90/therm; Solar Panel Cost = \$ 500 /panel; Cost of Installation = 10%															
	Diameter of Storage Tank, D =0.5 m					Diameter of Storage Tank, D =1.0 m					Diameter of Storage Tank, D =1.5 m				
# Panels	Opr Cost (\$/yr)	Gas Cost (\$/yr)	Heat S.Coll (thr/yr)	Heat dump (thr/yr)	Pay back (yrs)	Opr Cost (\$/yr)	Gas Cost (\$/yr)	Heat S.Coll (thr/yr)	Heat dump (thr/yr)	Pay back (yrs)	Opr Cost (\$/yr)	Gas Cost (\$/yr)	Heat S.Coll (thr/yr)	Heat dump (thr/yr)	Pay back (yrs)
0	<i>11,135</i>	<i>797</i>	-	-	-	<i>11,135</i>	<i>797</i>	-	-	-	<i>11,135</i>	<i>797</i>	-	-	-
10	7,667	2,888	879	47	1.43	7,692	2,913	877	20	2.14	7,762	2,983	877	7.1	4.11
30	6,482	1,703	2,631	651	2.48	6,409	1,630	2,631	489	2.95	6,302	1,523	2,631	295	4.23
50	6,035	1,256	4,384	1,921	3.56	5,885	1,106	4,384	1,718	3.92	5,551	772	4,384	1,293	4.85

The numbers in the italics refers to the base case with no solar installation, the other numbers are for the flat pale solar collector system.

Formulation for Economic Analysis

Total Capital Cost of Solar System , $C_s = [(\# \text{ panels} \times \text{Cost/panel}) + \{(\text{Additional cost of Absorption Chiller/ton}) \times (\text{cooling capacity in tons})\} + \{\text{Cost of Storage Tank}\}]$

Additional cost of Absorption Chiller = \$ 100/ton of chiller capacity.

Cost of Storage Tank, (\$) = 6000 + { 10 [V (gal) - 500] }, where, V = volume of storage tank in gallons

Cost of Installation (\$) = typically, it is expressed as certain percentage of the total capital cost of the system without installation = $(1+C_s) \{1+(i_{\text{instal}}/100)\}$

Total Installed cost of Solar system = $[(\# \text{ panels} \times \text{Cost/panel}) + \{(\text{Additional cost of Absorption Chiller/ton}) \times (\text{cooling capacity in tons})\} + \{\text{Cost of Storage Tank}\}] \times \{1+(i_{\text{instal}}/100)\}$

where, i_{instal} represents the installation costs as percentage of the total capital cost of the solar panels systems C_s as given in the above equation.

$$\text{Payback Period, PB (yrs)} = \frac{\text{Total Installed Capital Cost of Solar System over Conventional Sytem}}{(\text{Annual Cost of Operation of Conventional System} - \text{Annual Cost of Operation of Solar System})}$$

Table 4.3 Comparison of Evacuated Tube Collector and Flat Plate Solar Collector Performance Evaluation Data Sheet

70'x70' Building; $U_r = 0.036$, $U_w = 0.038$ Btu/hr.ft².⁰F; Occupants: 10; ACH: 0.3; Ventilation: 15 cfm/occupant; Lighting: 2 W/ft²; Summer Peak Load: 10.3 tons, Windows: 200 ft² on South and North Walls
Energy Cost = \$0.12/kWh; Demand Cost = \$10/kW; Elect Demand limit = 10 kW; Gas Cost = \$0.90/therm; Solar Panel Cost = \$ 1500 (ETC) and \$500 (FP) /panel

	Diameter of Storage Tank, D =0.5 m					Diameter of Storage Tank, D =1.0 m					Diameter of Storage Tank, D =1.5 m				
# Panels	Opr Cost (\$/yr)	Gas Cost (\$/yr)	Heat S.Coll (thr/yr)	Heat dump (thr/yr)	Pay back (yrs)	Opr Cost (\$/yr)	Gas Cost (\$/yr)	Heat S.Coll (thr/yr)	Heat dump (thr/yr)	Pay back (yrs)	Opr Cost (\$/yr)	Gas Cost (\$/yr)	Heat S.Coll (thr/yr)	Heat dump (thr/yr)	Pay back (yrs)
0	<i>11,135</i>	<i>797</i>	-	-	-	<i>11,135</i>	<i>797</i>	-	-	-	<i>11,135</i>	<i>797</i>	-	-	-
10	7,667 <i>8,540</i>	2,888 <i>2,732</i>	879 <i>1,263</i>	47 <i>111</i>	1.43 <i>4.46</i>	7,692 <i>8,558</i>	2,913 <i>2,750</i>	877 <i>1,263</i>	20 <i>77.5</i>	2.14 <i>5.42</i>	7,762 <i>8,597</i>	2,983 <i>2,789</i>	877 <i>1,263</i>	7.1 <i>30.7</i>	4.11 <i>8.06</i>
30	6,482 <i>7,365</i>	1,703 <i>1,557</i>	2,631 <i>3,787</i>	651 <i>1,461</i>	2.48 <i>5.64</i>	6,409 <i>7,250</i>	1,630 <i>1,442</i>	2,631 <i>3,787</i>	489 <i>1,293</i>	2.95 <i>8.69</i>	6,302 <i>6,914</i>	1,523 <i>1,106</i>	2,631 <i>3,787</i>	295 <i>866</i>	4.23 <i>9.52</i>
50	6,035 <i>7,053</i>	1,256 <i>1,245</i>	4,384 <i>6,312</i>	1,921 <i>3,672</i>	3.56 <i>8.87</i>	5,885 <i>6,912</i>	1,106 <i>1,104</i>	4,384 <i>6,312</i>	1,718 <i>3,479</i>	3.92 <i>12.6</i>	5,551 <i>6,503</i>	772 <i>695</i>	4,384 <i>6,312</i>	1,293 <i>2,979</i>	4.85 <i>12.8</i>
										1					9

The numbers in the boldface shown in the above Table refers to the flat plate collector system, while those in italics refers to the base case with no solar installation, the other numbers are for the evacuated solar tube collector system.

Formulation for Economic Analysis

Total Capital Cost of Solar System , $C_s = [(\# \text{ panels} \times \text{Cost/panel}) + \{(\text{Additional cost of Absorption Chiller/ton}) \times (\text{cooling capacity in tons})\} + \{\text{Cost of Storage Tank}\}]$

Additional cost of Absorption Chiller = \$ 100/ton of chiller capacity.

Cost of Storage Tank, (\$) = 6000 + { 10 [V (gal) - 500] }, where, V = volume of storage tank in gallons

Cost of Installation (\$) = typically, it is expressed as certain percentage of the total capital cost of the system without installation = $(1+C_s)$

{ $1+(i_{\text{instal}}/100)$ }

Total Installed cost of Solar system = $[(\# \text{ panels} \times \text{Cost/panel}) + \{(\text{Additional cost of Absorption Chiller/ton}) \times (\text{cooling capacity in tons})\} + \{\text{Cost of Storage Tank}\}] \times \{1+(i_{\text{instal}}/100)\}$

where, i_{instal} represents the installation costs as percentage of the total capital cost of the solar panels systems C_s as given in the above equation.

$$\text{Payback Period, PB (yrs)} = \frac{\text{Total Installed Capital Cost of SolarSystem over Conventional Sytem}}{(\text{Annual Cost of Operation of Conventional System} - \text{Annual Cost of Operation of Solar System})}$$

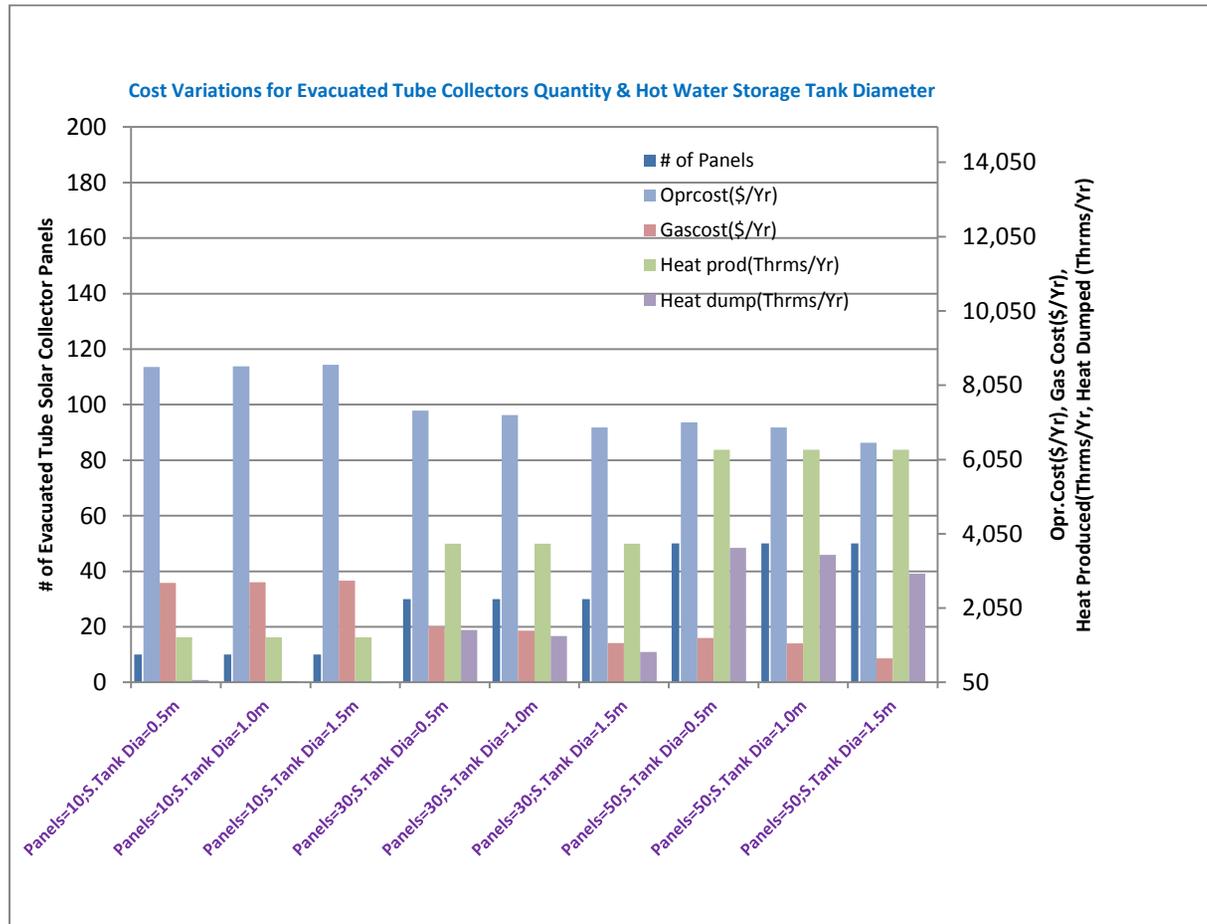


Figure 4.5 Variation of Operating Cost, Heat Gained, Heat Wasted with Size of Evacuated Tube Solar Collectors and Storage Tank

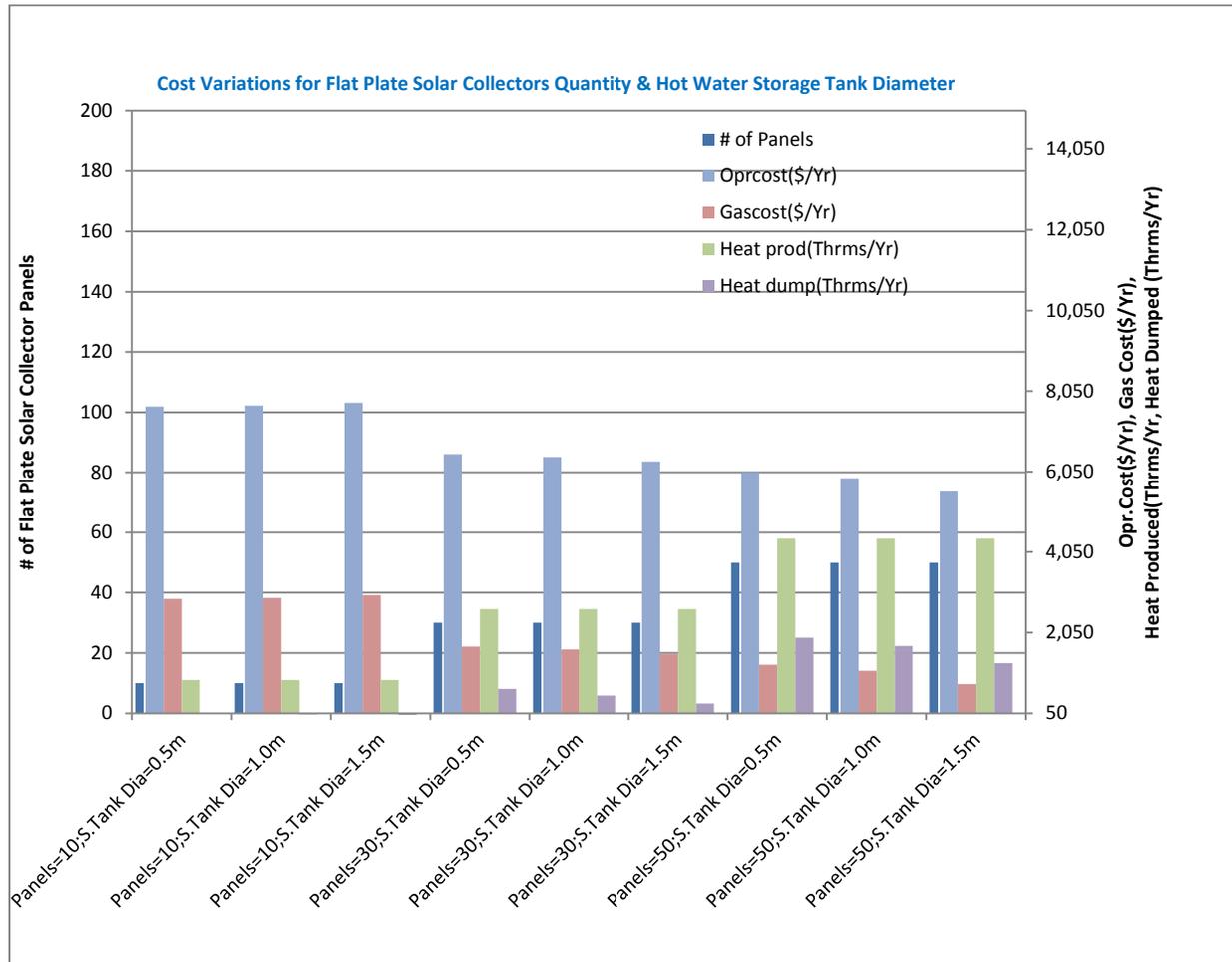


Figure 4.6 Variation of Operating Cost, Heat Gained, Heat Wasted with Size of Flat Plate Solar Collectors and Storage Tank

Evacuated Tube Solar Collector System: Pay-back Years Variation

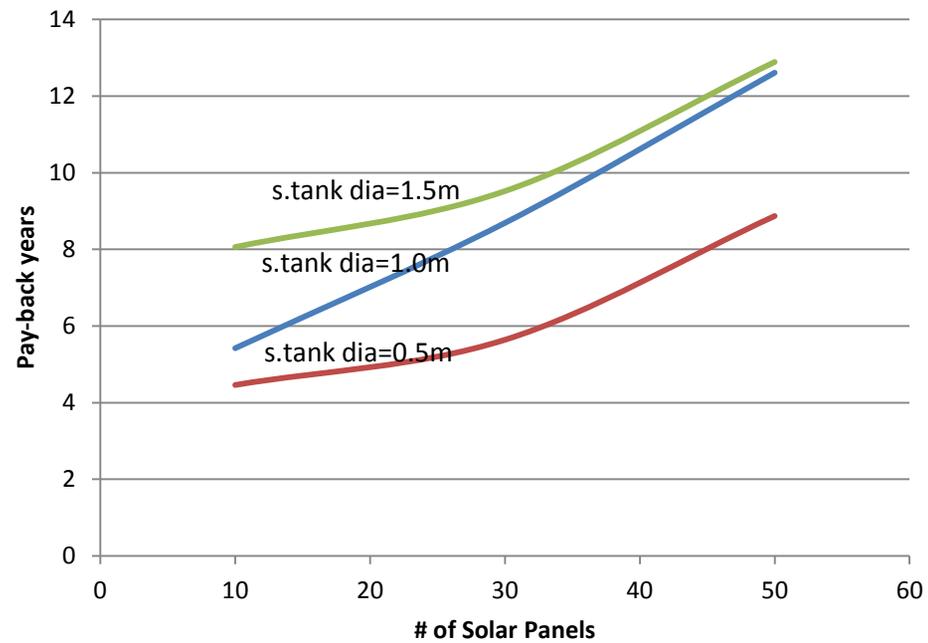


Figure 4.7 Variation of Payback Period with Size of Evacuated Tube Solar Collectors and Storage Tank

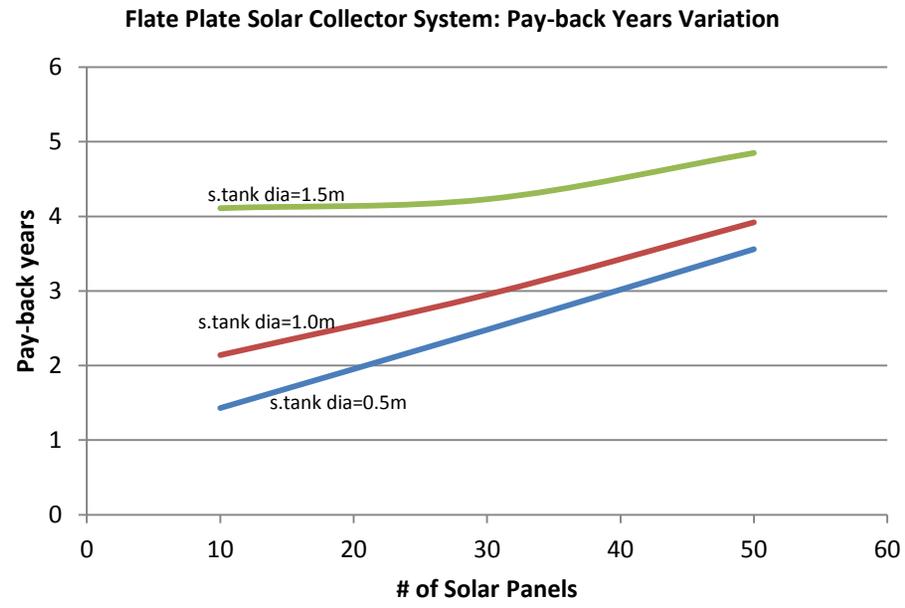


Figure 4.8 Variation of Payback Period with Size of Flat Plate Solar Collectors and Storage Tank

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

The computer simulations of solar assisted absorption chiller performance is evaluated based on the hourly weather data. The evaluation is carried out for a 4900 square foot floor area of a commercial building located in Chattanooga, Tennessee. The building is chosen to reflect the typical electrical, and occupancy load profile. The transient nature of weather as typified by the variation of ambient temperature, humidity, solar flux and that of electrical load profile pose difficult task in evaluating the hourly building loads. In order to overcome this problem Transfer Function Method (TFM) is employed to determine the hourly variation of the buildings. The computer code (TABLET) as developed by Dr. Dhamshala [8] is employed to determine the hourly loads for each hour of the year. The performance parameters of the solar collectors and that of the absorption chiller are obtained based on the manufacturers specifications and performance data. In order to meet the building loads at evening and night hours, a single node thermal storage is employed to store the energy during the day time. During the hours, when sufficient energy is not available in the storage tank, an auxiliary heater is activated to meet the load.

A computer simulation is carried out for base case of a conventional system consisting of a gas furnace and an air-conditioner. It was found that the operating energy cost of the building comes to \$ 11, 135 per year that includes \$ 797 per year of gas cost used for space

heating. Based on the results obtained from computer simulations for an evacuated tube solar collector and a flat plate solar collectors, the following conclusions can be made:

1. The smaller number of panels yield lower payback period for both flat plate and evacuated tube solar collectors. However, the operating cost and gas cost decrease as the number of panels are increased for both the cases.
2. The payback period tends to rise as the size of the thermal storage is increased.
3. The payback period also tends to rise as the size of the thermal storage is increased for a given panel size.
4. The cost of auxiliary gas used reduces as expected for rise in panel size and size of the thermal storage that has a profound positive impact on the environment due to reduction in greenhouse gas emissions.
5. The supply hot water temperature for absorption chillers are in the range of 70 to 95⁰C for a single effect absorption chiller for satisfactory operation. In order to limit the temperature rise in thermal storage to below 95⁰C during hot summer hours, there is a need to dissipate (dump) the heat energy form the storage tank. The magnitude of heat wasted (dumped) increases as the size of the panels is increased or as the size of the thermal storage is decreased for a given size of the panels.
6. It is suggested that a flat plate collectors are more favorable compared to the evacuated tube collectors.

It is recommended that a multi-node thermal storage model may be employed to obtain more accurate predictions for a future research project. It is also recommended that a more elaborate economic analysis may be carried out that takes into account the drop in

collector costs, variable energy costs and the latest in panels such as PV/T panels that are capable of providing electric power and thermal energy at a premium costs.

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APPENDIX A

VARIOUS RATED CAPACITIES OF YAZAKI'S CHILLERS

Table A.1 Various Rated Capacities of Yazaki's Chillers

Specifications

Model		WFC	SC10	SH10	SC20	SH20	SC30	SH30
Cooling	Capacity (Btu/hr x 1000)		120.0		240.0		360.0	
	Chilled Water Temp. (°F)		44.6 Outlet, 54.5 Inlet					
Heating	Capacity (Btu/hr x 1000)	-	166.3	-	332.6	-	498.9	
	Hot Water Temp. (°F)		131.0 Outlet, 117.3 Inlet					
Chilled/Hot Water	Rated Water Flow (gpm)		24.2		48.4		72.6	
	Evap. Press Drop (psi)		6.1		9.6		10.1	
	Water Retention Volume (gal)		4.5		12.4		19.0	
Cooling Water	Heat Rejection (Btu/hr x 1000)		291.4		582.8		874.2	
	Inlet Temperature (°F)		87.8 (Standard)					
	*Rated Water Flow (gpm)		80.8		161.7		242.5	
	Cond./Abs. Press. Drop (psi)		12.3		6.6		6.7	
	Water Retention Volume (gal)		17.4		33.0		51.3	
Heat Medium	Input (Btu/hr x 1000)		171.4		342.8		514.2	
	Inlet Temperature (°F)		190.4 (Standard)					
				Temperature Range 158 (min.) - 203 (max.)				
	Rated Water Flow (gpm)		38.0		76.1		114.1	
	Generator Press. Drop (psi)		13.1		6.7		8.8	
	Water Retention Volume (gal)		5.5		14.3		22.2	
Electrical	Power Supply		208V, 60Hz, 3 ph					
	Consumption (W)		210		260		310	
Capacity Control			On - Off					
Noise Level	Sound Pressure dB(A)		49		49		46	
Piping	Chilled/Hot Water (in)		1-1/2 NPT		2 NPT		2 NPT	
	Cooling Water (in)		2 NPT		2 NPT		2-1/2 NPT	
	Heat Medium (in)		1-1/2 NPT		2 NPT		2-1/2 NPT	
Weight	Dry (lb)		1,100		2,050		3,200	
	Operating (lb)		1,329		2,548		3,975	

* Minimum cooling water flow

APPENDIX B

PERFORMANCE DATA OF YAZAKI ABSORPTION CHILLERS

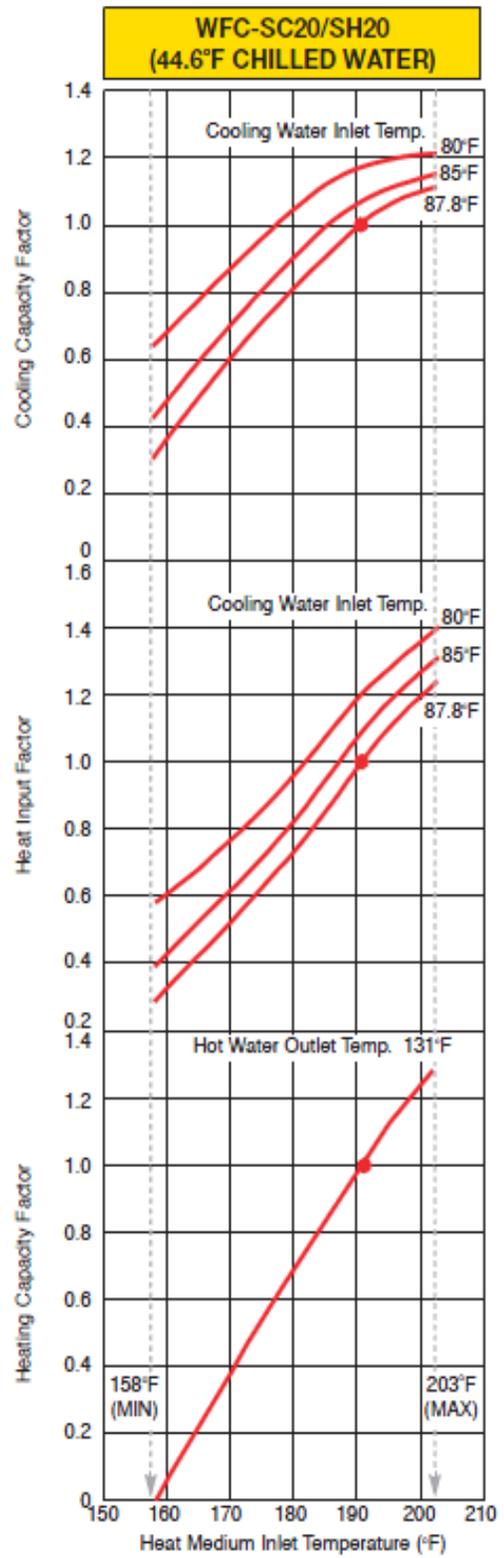
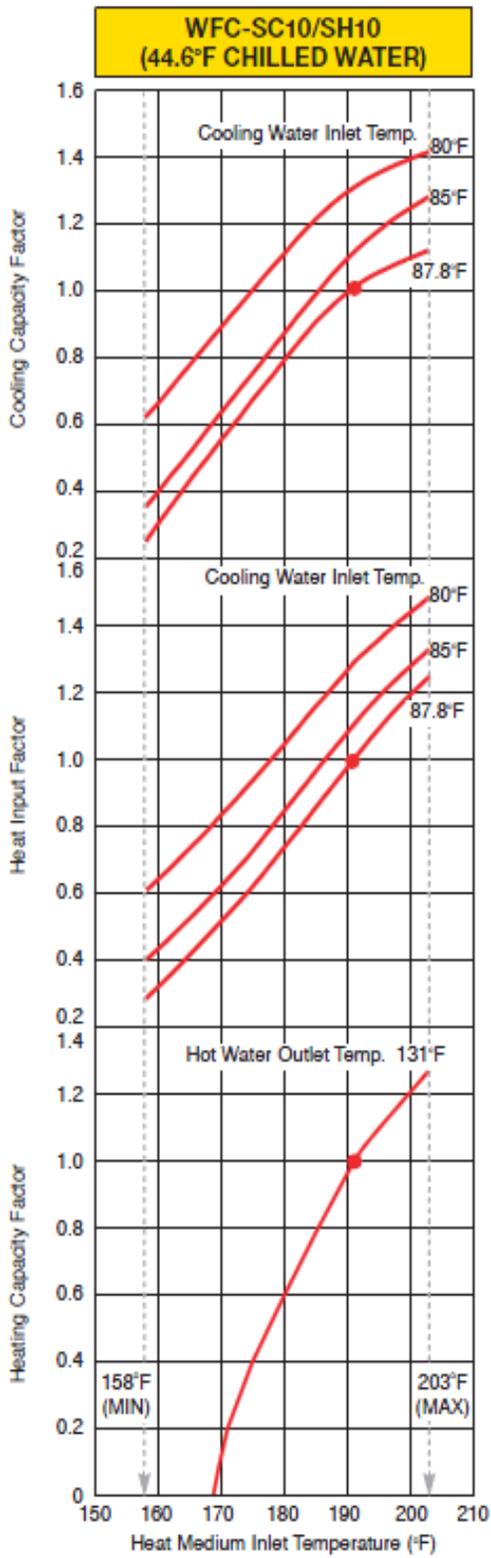


Figure B.1 Performance Data of Yazaki Absorption Chillers

ABSORPTION CHILLER HEAT BALANCE
HEAT IN = HEAT OUT

$$Q_g + Q_e = Q_c$$

Where, Q_g = Heat input to generator
 Q_e = Cooling capacity
 Q_c = Heat rejected to cooling tower

COOLING CAPACITY

$$Q_e = \text{CLG. CAP. FACTOR} \times \text{HM FLOW CORRECTION} \times \text{STD. CLG. CAPACITY}$$

HEAT INPUT (COOLING)

$$Q_g = \text{HEAT INPUT FACTOR} \times \text{HM FLOW CORRECTION} \times \text{STD. HEAT INPUT}$$

HEATING CAPACITY

$$Q_h = \text{HTG. CAP. FACTOR} \times \text{HM FLOW CORRECTION} \times \text{STD. HTG. CAPACITY}$$

Where, Q_h = Heating Capacity

HEAT INPUT (HEATING)

$$Q_g = \frac{\text{HEATING CAPACITY}}{\text{EFFICIENCY}} = \frac{Q_h}{0.97}$$

TEMPERATURE DIFFERENCE (°F)

$$\Delta T = \frac{\text{ADJUSTED CAPACITY OR HEAT INPUT (MBH)}}{0.5 \times \text{FLOW (gpm)}}$$

PRESS. DROP FOR NONSTANDARD FLOW (psi)

$$\Delta P = \text{STANDARD PRESS. DROP} \times \left(\frac{\text{NONSTANDARD FLOW}}{\text{STANDARD FLOW}} \right)^2$$

EXAMPLE 1.

Given design conditions:

Heat medium inlet temperature	195°F
Heat medium flow	114.1 gpm
Cooling water inlet temperature	85°F
Cooling water flow	242.5 gpm
Chilled water outlet temperature	44.6°F
Hot water outlet temperature	131°F
Chilled/hot water flow	72.6 gpm
Absorption chiller-heater model	WFC-SH30

Refer to Capacity Factor curves and Specifications for model WFC-SC30/SH30. Since 114.1 gpm is standard, the Heat Medium (HM) Flow Correction is 1.0.

1. AVAILABLE COOLING CAPACITY:

Cooling Capacity Factor = 1.12

2. HEAT INPUT (COOLING):

Heat Input Factor = 1.17
Heat Medium Flow Correction = 1.0
Standard Heat Input = 514.2 MBH
 $Q_g = 1.17 \times 1.0 \times 514.2 = 601.6$ MBH
Heat Medium $\Delta T = \frac{601.6}{0.5 \times 114.1} = 10.5^\circ\text{F}$
Heat Medium $\Delta P = 8.8$ psi (Standard)

3. HEAT REJECTED TO COOLING TOWER:

$Q_c = Q_g + Q_e = 601.6 + 403.2 = 1004.8$ MBH
Cooling Water $\Delta T = \frac{1004.8}{0.5 \times 242.5} = 8.3^\circ\text{F}$
Cooling Water $\Delta P = 6.7$ psi (Standard)

4. AVAILABLE HEATING CAPACITY:

Heating Capacity Factor = 1.12
Heat Medium Flow Correction = 1.0
Standard Heating Capacity = 498.9 MBH
 $Q_h = 1.12 \times 1.0 \times 498.9 = 558.8$ MBH
Hot Water $\Delta T = \frac{558.8}{0.5 \times 72.6} = 15.4^\circ\text{F}$
Hot Water $\Delta P = 10.1$ psi (Standard)

5. HEAT INPUT (HEATING):

$Q_g = \frac{Q_h}{0.97} = \frac{558.8}{0.97} = 576.1$ MBH
Heat Medium $\Delta T = \frac{576.1}{0.5 \times 114.1} = 10.1^\circ\text{F}$
Heat Medium $\Delta P = 8.8$ psi (Standard)

EXAMPLE 2.

Given design conditions:

Heat medium inlet temperature	203°F
Heat medium flow	57.0 gpm
Cooling water inlet temperature	85°F
Cooling water flow	242.5 gpm
Chilled water outlet temperature	44.6°F
Hot water outlet temperature	131°F
Chilled/hot water flow	72.6 gpm
Absorption chiller-heater model	WFC-SH30

Refer to Capacity Factor curves and Specifications for model WFC-SC30/SH30. Since 57.0 gpm is 50% of standard, the Heat Medium (HM) Flow Correction is 0.86.

1. AVAILABLE COOLING CAPACITY:

Cooling Capacity Factor = 1.22

2. HEAT INPUT (COOLING):

Heat Input Factor = 1.35
Heat Medium Flow Correction = 0.86
Standard Heat Input = 514.2 MBH
 $Q_g = 1.35 \times 0.86 \times 514.2 = 597.0$ MBH
Heat Medium $\Delta T = \frac{597.0}{0.5 \times 57.0} = 20.9^\circ\text{F}$

Heat Medium $\Delta P = 8.8 \times \left(\frac{57.0}{114.1}\right)^2 = 2.2$ psi

3. HEAT REJECTED TO COOLING TOWER:

$Q_c = Q_g + Q_e = 597.0 + 377.7 = 974.7$ MBH
Cooling Water $\Delta T = \frac{974.7}{0.5 \times 242.5} = 8.0^\circ\text{F}$
Cooling Water $\Delta P = 6.7$ psi (Standard)

4. AVAILABLE HEATING CAPACITY:

Heat Capacity Factor = 1.33
Heat Medium Flow Correction = 0.86
Standard Heating Capacity = 498.9 MBH
 $Q_h = 1.33 \times 0.86 \times 498.9 = 570.6$ MBH
Hot Water $\Delta T = \frac{570.6}{0.5 \times 72.6} = 15.7^\circ\text{F}$
Hot Water $\Delta P = 10.1$ psi (Standard)

5. HEAT INPUT (HEATING):

$Q_g = \frac{Q_h}{0.97} = \frac{570.6}{0.97} = 588.2$ MBH
Heat Medium $\Delta T = \frac{588.2}{0.5 \times 57.0} = 20.6^\circ\text{F}$
Heat Medium $\Delta P = 8.8 \times \left(\frac{57.0}{114.1}\right)^2 = 2.2$ psi

APPENDIX C

SOLAR COLLECTOR CERTIFICATION AND RATING

SOLAR COLLECTOR CERTIFICATION AND RATING  SRCC OG-100	CERTIFIED SOLAR COLLECTOR SUPPLIER: Solar Panels Plus 533 Byron Street Suite E Chesapeake, VA 23320 USA MODEL: SPP -30 COLLECTOR TYPE: Tubular CERTIFICATION #: 100-2008-050B
---	---

COLLECTOR THERMAL PERFORMANCE RATING							
Megajoules Per Panel Per Day				Thousands of Btu Per Panel Per Day			
CATEGORY (T _i -T _a)	CLEAR DAY 23 MJ/m ² -d	MILDLY CLOUDY 17 MJ/m ² -d	CLOUDY DAY 11 MJ/m ² -d	CATEGORY (T _i -T _a)	CLEAR DAY 2000 Btu/ft ² -d	MILDLY CLOUDY 1500 Btu/ft ² -d	CLOUDY DAY 1000 Btu/ft ² -d
A (-5°C)	50	38	25	A (-9°F)	47	36	24
B (5°C)	48	36	24	B (9°F)	46	34	22
C (20°C)	45	33	20	C (36°F)	43	31	19
D (50°C)	38	26	14	D (90°F)	36	25	13
E (80°C)	30	19	7	E (144°F)	29	18	7

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

Original Certification Date: January 5, 2009

COLLECTOR SPECIFICATIONS

Gross Area: 4.810 m² 51.78 ft²
 Dry Weight: 86 kg 190 lb
 Test Pressure: 600 kPa 87 psig

Net Aperture Area: 4.157 m² 44.75 ft²
 Fluid Capacity: 1.6 l 0.4 gal

COLLECTOR MATERIALS

Frame: Stainless Steel
 Cover (Outer): Glass Vacuum Tube
 Cover (Inner): None
 Absorber Material: Tube - Copper / Plate - Aluminum
 Absorber Coating: Aluminum Nitride
 Insulation (Side): Vacuum
 Insulation (Back): Vacuum

PRESSURE DROP

Flow		ΔP	
ml/s	gpm	Pa	in H ₂ O
0	0.00	0	0.00
0	0.00	0	0.00
0	0.00	0	0.00

TECHNICAL INFORMATION

Efficiency Equation [NOTE: Based on gross area and (P) = T_i-T_a]

				Y Intercept	Slope	
SI Units:	$\eta = 0.4772$	$-0.9374 (P)/I$	$-0.0066 (P)^2/I$	0.4806	-1.3337	W/m ² -°C
IP Units:	$\eta = 0.4772$	$-0.1652 (P)/I$	$-0.0006 (P)^2/I$	0.4806	-0.235	Btu/hr-ft ² -°F

Incident Angle Modifier [(S) = 1/cos θ - 1, 0° ≤ θ ≤ 60°]
 $K_{inc} = 1.0 + 0.9638 (S) - 1.1856 (S)^2$
 $K_{inc} = 1.0 - 0.27 (S)$ (Linear Fit)

Model Tested: 100-2007-028B
 Test Fluid: Water
 Test Flow Rate: 71 ml/s 1.13 gpm

REMARKS: Tested with long axis of tubes oriented north-south. IAM perpendicular to the tubes is listed above. IAM parallel to the tubes = 1.0 - 0.33(S)

May, 2009

Certification must be renewed annually. For current status contact:
 SOLAR RATING & CERTIFICATION CORPORATION

c/o FSEC • 1679 Clearlake Road • Cocoa, FL 32922 • (321) 638-1537 • Fax (321) 638-1010

SOLAR COLLECTOR CERTIFICATION AND RATING  SRCC OG-100	CERTIFIED SOLAR COLLECTOR SUPPLIER: Alternate Energy Technologies 1057 N. Ellis Road Jacksonville, FL 32254 USA MODEL: American Energy AE-21E COLLECTOR TYPE: Glazed Flat-Plate CERTIFICATION #: 100-1999-001A
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COLLECTOR THERMAL PERFORMANCE RATING							
Megajoules Per Panel Per Day				Thousands of Btu Per Panel Per Day			
CATEGORY (Ti-Ta)	CLEAR DAY 23 MJ/m ² -d	MILDLY CLOUDY 17 MJ/m ² -d	CLOUDY DAY 11 MJ/m ² -d	CATEGORY (Ti-Ta)	CLEAR DAY 2000 Btu/ft ² -d	MILDLY CLOUDY 1500 Btu/ft ² -d	CLOUDY DAY 1000 Btu/ft ² -d
A (-5°C)	28	21	15	A (-9°F)	27	20	14
B (5°C)	25	18	11	B (9°F)	24	17	11
C (20°C)	20	13	6	C (36°F)	19	12	6
D (50°C)	9	3		D (90°F)	8	3	
E (80°C)				E (144°F)			

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

Original Certification Date: June 15, 1999

COLLECTOR SPECIFICATIONS

Gross Area:	1.926 m ²	20.73 ft ²	Net Aperture Area:	1.776 m ²	19.12 ft ²
Dry Weight:	40.8 kg	90 lb	Fluid Capacity:	3.0 l	0.8 gal
Test Pressure:	1103 kPa	160 psig			

COLLECTOR MATERIALS

Frame:	Anodized Aluminum
Cover (Outer):	Low Iron Tempered Glass
Cover (Inner):	None
Absorber Material:	Tube - Copper / Plate - Copper
Absorber Coating:	Moderately Selective Black Paint
Insulation (Side):	Polyisocyanurate
Insulation (Back):	Polyisocyanurate

PRESSURE DROP

Flow	Δ P		
	ml/s	gpm	Pa
20	0.32	55	0.22
50	0.79	306	1.23
80	1.27	745	2.99

TECHNICAL INFORMATION

Efficiency Equation [NOTE: Based on gross area and (P) = Ti-Ta]				Y Intercept	Slope
SI Units:	$\eta = 0.638$	$-4.2645 (P)/I$	$-0.0297 (P)^2/I$	0.66	-6.37 W/m ² ·°C
IP Units:	$\eta = 0.638$	$-0.7515 (P)/I$	$-0.0029 (P)^2/I$	0.66	-1.123 Btu/hr-ft ² ·°F

Incident Angle Modifier [(S) = 1/cos θ - 1, 0° ≤ θ ≤ 60°]			Model Tested:	AE-21E
K _{cor} = 1.0	+0.0248 (S)	-0.0861 (S) ²	Test Fluid:	Water
K _{cor} = 1.0	-0.05 (S)	(Linear Fit)	Test Flow Rate:	39 ml/s
				0.61 gpm

REMARKS:

SRCC sample certification information (SRCC).

VITA

Gnananesan Antonyraj was born in the State of Tamilnadu in India, to the parents of Athinarayanan Gnananesan and P.S. Saroja. Joseph is his sibling brother. Gnananesan earned his Master's degree in Mathematics from St. Xavier's College, Palayamkottai, India, which was affiliated to Madurai Kamaraj University in the year 1988. After his marriage in 1990, Gnananesan immigrated to Canada with his wife Shanthi at the end of 1991. He studied at Seneca College of Applied Arts and Sciences in Toronto, Canada, and earned his diploma in Computer Programming And Analysis. He was employed with Canadian Imperial Bank of Commerce in Toronto and learned Mainframe computing skills. With the Mainframe computing skills, Gnananesan migrated to the United States of America in the year 1999. He started working at BlueCross and BlueShield of Tennessee in Chattanooga, and has held the titles of Technical Systems Analyst and Database Administrator. Gnananesan became the father of his two sons Benedict and Francis in the years 2000 and 2001 respectively. Gnananesan started attending the University of Tennessee as a part-time student and completed his second Master's degree in Mechanical Engineering in December 2012. Gnananesan enjoys travelling. In his spare time, he volunteers at the churches.