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Sizing and Economics of Solar Powered Indoor Swimming Pool

Aman Bhatta
University of Mississippi

SIZING AND ECONOMICS OF SOLAR POWERED INDOOR SWIMMING POOL

by
Aman Bhatta

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Approved by

Advisor: Dr. Tejas Pandya

Reader: Dr. Amrita Mishra

Reader: Dr. Ken Thomas

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Abstract

With increasing global population and aspiration for higher standards for living, there is a rapid increase in demand for energy. With the concerns of climate change becoming ever more pressing, a shift from the conventional energy source to an alternative green version to meet the energy demand could be of significant interest. A large amount of energy is required to maintain water temperature in the swimming pool at a human comfort temperature. Several studies have been performed to calculate and evaluate the feasibility of powering thermal systems for swimming pools using solar energy. Still, extensive analysis using solar thermal energy to power the indoor swimming pool lacks in the United States' southern region. This work is an attempt to study the use of solar thermal energy to meet the thermal energy demand of an indoor swimming pool located in Oxford, MS. The swimming Pool in Turner Center at the University of Mississippi is used as a model for the calculation. One of the major assumptions made for this analysis is that the energy required to maintain the ambient temperature, wind velocity, and relative humidity is not accounted for in the system. ASHRAE prescribes the ambiance temperature to be about 2° F higher than water temperature, wind velocity over the water surface to be between 0.0508-0.1524 mps, and relative humidity to be between 50-60%. We assume these standard conditions, as prescribed by ASHRAE[1], are already maintained. Thus, only calculating the thermal energy load of the swimming pool maintained at this standard condition is performed.

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Nomenclature

Abbreviations

ASHRAE American Society of Heating, Refrigerating and Air-Conditioning Engineers

DOE United States Department of Energy

EIA United States Energy Information Administration

fpm feet per minute

GHGs Greenhouse gases

ISP Internal swimming pool

Mcf one thousand cubic feet

mps meter per second

n Total number of months

O&M Operations and Maintenance

TRNSYS Transient System Simulation Tool

List of Symbols

α Absorptivity of absorber plate

$\dot{Q}_{cond}$ Conductive heat loss rate(kW)

$\dot{Q}_{conv}$ Convective heat loss rate(kW)

$\dot{Q}_{evap}$ Evaporative heat loss rate(kW)

$\dot{Q}_{feed}$ Feed water heat requirement(kW)

$\dot{Q}_{rad}$ Radiative heat loss (kW)

ϵ_w Emissivity constant

ρ Density of water($\frac{kg}{m^3}$)

σ Stefan-Boltzman constant $= 5.67 \times 10^{-11} \frac{kW}{m^2 \cdot K^4}$

τ Transmissivity of glass cover plate

A_p Area of the pool(m^2)

G_w Evaporative mass flow rate$\left(\frac{kg}{s}\right)$

h_e Latent heat of evaporation$\left(\frac{kJ}{kg}\right)$

h_{conv} Convective heat transfer coefficient($\frac{W}{m^2 \cdot {}^\circ C}$)

I_T Solar irradiance$\left(\frac{Wh}{m^2}\right)$

m_{rf} Daily feed water mass(kg)

$P_s(T_a)$ Saturation vapor pressure taken at room air temperature(Pa)

$P_s(T_w)$ Saturation vapor pressure taken at surface water temp(Pa)

q_{abs} Solar irradiation absorbed by collector$\left(\frac{Wh}{m^2}\right)$

q_{loss} Energy loss$\left(\frac{Wh}{m^2}\right)$

q_{useful} Useful energy available$\left(\frac{Wh}{m^2}\right)$

T_a Ambient temperature(°C)

T_c Critical point temperature(647.096 K)

T_d Dew point temperature(°C)

T_f Feed water temperature(°C)

T_{wlr} Wall temperature(K)

T_w Pool water temperature(K/°C)

RH Relative humidity

v wind speed($\frac{m}{s}$)

1 Introduction

Swimming is the fourth most popular recreational activity in United States[6]. In Mississippi, which extends from 30°13' N to 35°N, most optimal months for swimming are between early May to late October. Comfortable swimming conditions are determined by factors like water temperature, ambient air temperature, relative humidity, and wind velocity. Water temperature, however, is the most significant parameter that decides the comfort of a swimmer[7]. In the United States, Water Temperature, as suggested by the Water Fitness Association, averages somewhere between 84°C to 86°C [8]. Tripton and Bradford[9] pointed several risks associated with swimming in cold and warm water. The threats associated with swimming in thermally stressful water ranges from deterioration in performance to life-threatening pathology. The threats to life associated with immersion in cold water include drowning, cardiac problems, hypothermia, and cardiovascular problems on exiting the water. In warm water, the corresponding threats are hyperthermia and cardiovascular problems on exiting the water[9]. Thus, it is critically important to maintain the optimal pool temperature.

The most common heating systems used today are electric pump heaters and gas heaters. Provided that fossil fuel contributed 62.7% of total US energy demand in 2109[10], fossil fuel most likely contributed significant energy demand for any electric pump heater in use. Gas heater solely relies on fossil fuels. In the past decade, several studies have been conducted to investigate the impact of the ever-rising implications of fossil fuel combustion on the environment and human health. Olivier, Peters, and Schure[11] investigated and found that about 70% of the total global GHG emissions are in the form of CO_2 due to the combustion of fossil fuels. Similarly, Lelieveld et al.[12] in their recent 2019 study found that fossil-fuel-related emissions account for about 65% of the excess mortality rate attributable to air pollution. The impact of fossil fuel combustion on the environment and human health is becoming more and more pressing. Atse, Wilfried, André and Rudd[13] mentioned that increasing the installation of solar technology decreases the overall greenhouse gas footprint. Thus, the utilization of solar energy technology could be of considerable interest.

Energy use in buildings comprises around 30% to 40% of the total worldwide energy use[14]. Compared with different categories of buildings, swimming pools have higher energy demand for pool water heating, ventilation, space heating, and operation of circular pumps[15]. Even though our swimming pool is an indoor swimming pool, additional heat is required to ensure indoor thermal comfort levels in addition to comfortable water temperature. Rajagopalan and Jamie[16] studied an increase in indoor humidity due to water evaporation, and that increases energy demand as a ventilation system is required to curb excessive humidity.

Recent studies have made the environmental impact of fossil fuels on climate change[17] more evident. This indicates that the need to shift the economy to green and clean energy is becoming more pressing. A typical household uses a significant amount of energy in heating applications. Solar thermal systems do have great potential to supply energy demands for such applications. Considering such prospects, researchers have developed numerous designs and improvements to increase solar-thermal systems' efficiency. Li et al.[18] designed a solar water heater in a glass evacuated tube with a high heat collection rate by high throughput screening (HTS) method based on machine learning. This study has proved that recent advances in machine learning could aid in increasing the heat collection rate. Alfaro-Ayala et al.[19] used computational fluid dynamics (CFD) to conduct a numerical study and predict water temperature at the outlet of the collector. Li et al.[20] in their 2015 work numerically analyzed the use and design the thermal efficiency solar water heaters using U-type evacuated tube solar collector. The results showed that this solar collector should provide 40.5% of the total energy consumed in the year. Govaer[21] has also analyzed the energy performance of an ISP heating system with solar collectors, using a utilizability method, which was applied to evaluate the utilized thermal energy that solar collectors could provide.

Ahmad et al.[22] used evacuated tubes to fulfill the model swimming pool's thermal demand. They found that due to the solar system, energy reduction was found to be close to 75%. Chow et al.[23] also did a similar study to fulfill energy demand for the public swimming center located in Hong Kong using solar assisted heat pump. They found that the energy-saving factor can reach 79% in November. Their study also found that the present system design's economic payback period is less than five years, which is reasonably attractive. Starke et al.[24] also did a similar study using TRNSYS modeling. They found that their proposed model could reduce the energy load by up to 48%.

The solar water heating system's size depends on the availability of varied factors such as solar radiation availability, delivery water temperature requirement, geographic location, and solar system arrangement. There could be several other factors, but these above mentioned are crucial, with solar radiation being the most important one mentioned by Patel et al. [25]. A large amount of research has been performed on dynamic solar collecter systems as well. Mulaweh et al.[26] designed a system where the solar collector rotates with the change in the sun's position. Their study has shown that such a dynamic system increases the absorbed solar energy and thus, the overall system's efficiency.

From the literature review, it can be seen that comprehensive study to size a solar thermal collector lacks in the southern region of the United States of America. Therefore, this study presents the sizing of solar thermal collectors to supply the indoor swimming pool's thermal energy demand in Oxford, MS. First of all, the reference swimming pool is determined. Then, assuming the pool conditions to be the same as prescribed by ASHRAE, the mathematical models were developed to describe heat transfer processes in swimming pools, such as the heat losses due to evaporation, convection, radiation, and feed water. Since this is an indoor pool setting, no heat is directly gained from the sun. Based on the thermal model and localized ambiance of Oxford, MS, thermal calculations were performed. A brief overview of the solar thermal collector is provided. Then, the mathematical model to calculate the efficiency of a thermal collector is presented. Based on this mathematical model for efficiency, the thermal collector was sized to supply the reference swimming pool's energy demand.

2 Pool and Heat Loss Schematic

The swimming pool at the University of Mississippi's Turner Center is used as a reference to model the thermal loss and size the solar thermal heater accordingly. The general representation and dimension of the swimming pool are shown schematically in Figure 1. Similarly, the general schematic of heat loss is shown in Figure 2.

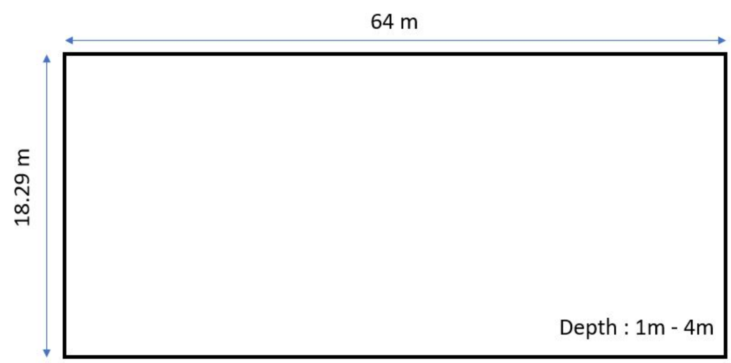

Figure 1: Schematic Diagram of Swimming Pool

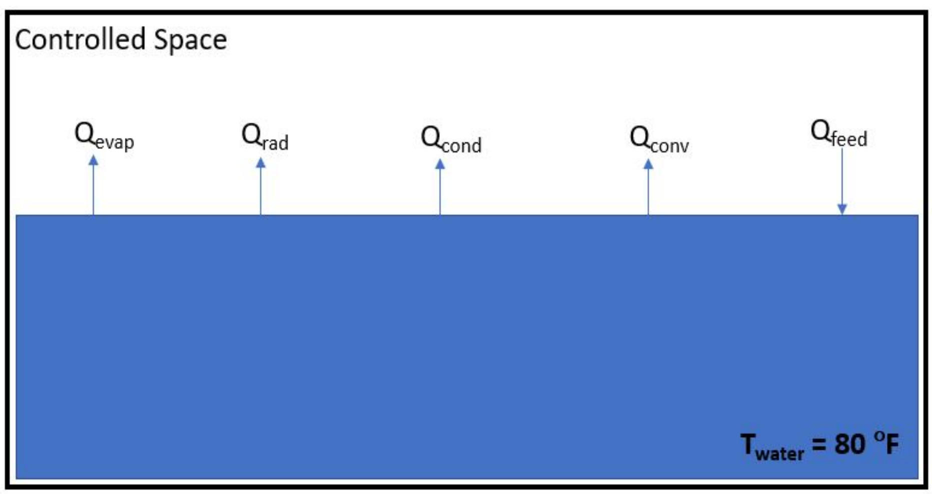

Figure 2: Schematic Diagram of Heat Loss

3 System Description

3.1 Pool Dimesions and Water Temperature

The dimensions of the swimming pool were obtained from The University of Mississippi campus recreation website[27]. Length and Width of the model swimming pool are 64m and 18.29m respectively. Depth of the pool ranges from 1m to 4m. Average depth value of 2.5m was used for calculation. Water temperature at the pool is maintained at around 76-84 °F throughout the year, so average value of 80 °F was used for calculation.

3.2 Major Assumptions

Some of the assumptions used for the sizing of the system are listed below:

1. The ambient of the model indoor swimming pool is maintained under the conditions as specified by ASHRAE. It is to be noted that the average value of these specified conditions is used in the calculation. The conditions specified by ASHRAE is shown as:

   - Relative Humidity = 50 - 60 %
   - Temperature of pool space air = T_w + 2 °F
   - Average velocity of air over water surface = 0.0508 - 0.1524 $\frac{m}{s}$

2. Conductive heat loss through the wall of swimming pool is neglected [28][29].

3. Temperature of the ambient space is assumed to be at steady state at all times. In other words, heat transfer gradient causing the change in the temperature of the ambient space is neglected.

3.3 Current Heating and Circulation System

The gas boiler used in heating the pool water in the turner center is a part of the campus hot water loop. So, there is not a single burner unit just dedicated to heating the pool water. A 25 HP pump, a part of the filtration system that runs 24/7, circulates the water. The pool water runs through the heat exchanger piping system from a gas boiler on the recirculation route. A thermostat controls a solenoid valve on the heat exchanger to attain desirable pool temperature. The hot water produced through the central gas boiler unit heat the pool water through a heat exchanger. Thus, there is no mixing of the pool water and the boiler hot water. A simple representative schematic of this hot water loop is shown as in Figure 3.

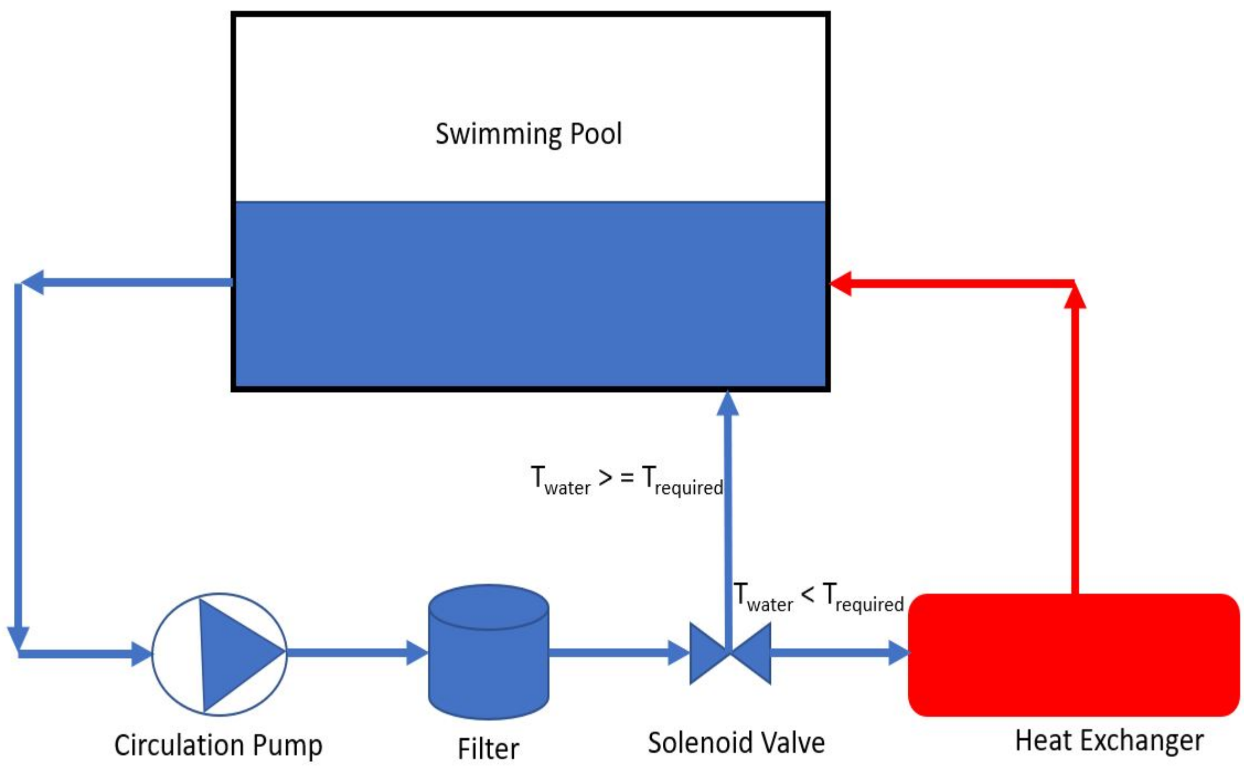

Figure 3: Simple schematic representation of circulation loop

3.4 Need of the Thermal Model

As described in section 3.3, there is no dedicated gas boiler to heat the water in the swimming pool. So, the amount of actual energy consumed or equivalent fuel burnt to sustain the swimming pool's thermal load was impossible to isolate from the energy data of the central system. But, this is a crucial piece of information to size the solar thermal panel. Thus, to study and determine the thermal load required to maintain the swimming pool at the desired temperature, a thermal model is proposed in this work. All the essential calculations are performed based on this thermal model.

4 Thermal Analysis

The thermal transfer model for the selected swimming pool should first be set up to size the solar thermal heaters. Ruiz and Martinez[30] modeled and studied the thermal transfer model of the swimming pool, as similar to the reference swimming pool used in this work, using TRNSYS. Their study found that heat loss in swimming pools can occur through conduction, convection, evaporation, long-wave radiation exchange, and heating of daily renovated water. The thermal loss rate can be expressed as:

$$\dot{Q}_{total} = \dot{Q}_{cond} + \dot{Q}_{conv} + \dot{Q}_{evap} + \dot{Q}_{rad} + \dot{Q}_{feed} \tag{1}$$

where $\dot{Q}_{total}$ is the total heat loss rate, $\dot{Q}_{cond}$ is the heat loss rate by conduction through the side and bottom surfaces of the swimming pools, $\dot{Q}_{conv}$ is the heat loss rate by convection at pool surface, $\dot{Q}_{evap}$ is the heat loss rate by water evaporation, $\dot{Q}_{rad}$ is the heat loss by long-wave radiation exchange and $\dot{Q}_{feed}$ is the heat loss rate to daily renovated feed water. Each of the above-mentioned thermal losses is studied and described in detail.

4.1 Conductive Heat Loss

Rakopoulos and Vazeos[28] in their study mentioned that ground conduction losses could be calculated using classical heat transfer methods, but the magnitude is relatively small in well-built pools, and it can be ignored. Similarly, in a different study Hahne and Kübler[29] mentioned that conduction heat losses to the ground usually amount to less than 1 %, and thus, it can be safely neglected in the calculation.Thus, Equation 1 can be updated as:

$$\dot{Q}_{cond} \approx 0 \tag{2}$$

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{evap} + \dot{Q}_{rad} + \dot{Q}_{feed} \tag{3}$$

4.2 Evaporative Heat Loss

When the liquid water of the pool changes its state to gaseous form, evaporative heat loss occurs. This endothermic process cools down the remaining water in the pool. Evaporative heat loss is directly proportional to rate of evaporation and can be expressed below as:

$$\dot{Q}_{evap} = h_e \cdot \ G_w \tag{4}$$

where h_e is the latent heat of evaporation, and G_w is the evaporative mass flow rate. Many methods for evaluating evaporation from water basins have been proposed over the years, although only a few are related explicitly to indoor swimming pools[31]. Some researchers have used the data from experimental measurements in real pools, and some have used evaporation as depicted by the pool or basin's energy balances model. Some researchers have even used the amount of condensate on the air-conditioning unit's cooling coil, assuming that this is equal to the amount of water evaporated from the pool surface. Out of several such studies, the evaporative model proposed by Asdrubali[32] is adopted in this study. The equation to calculate evaporative mass flow rate is shown as follows:

$$G_w = K \cdot A \cdot [P_s(T_w) - \phi \cdot P_s(T_a)] \tag{5}$$

where G_w is the evaporative mass flow rate, A is the area of pool, $P_s(T_w)$ is the saturation vapor pressure taken at surface water temp, $P_s(T_a)$ is the saturation vapor pressure taken at room air temperature and K is the mass transfer coefficient.

The value of the mass transfer coefficient was determined experimentally by Asdrubali[32]. The experimental setup conditions for the experiment performed by Asdrubali were the same as the conditions prescribed by the ASHRAE. Since the reference pool used is also maintained in the requirements specified by ASHRAE, the mass transfer coefficient is valid in our case as well.

Saul and Wagner[33] devised an equation in their study to accurately calculate the saturation vapor pressure at the given temperature. The equation is shown below:

$$log_e \frac{P_s}{22064000} = \frac{T_c}{T}\Bigg(- 7.859517838 \cdot \left(1 - \frac{T}{T_c}\right) - 1.84408259 \cdot \left(1 - \frac{T}{T_c}\right)^{1.5} - 11.7866497 \cdot \left(1 - \frac{T}{T_c}\right)^{3} + 22.6807411 \cdot \left(1 - \frac{T}{T_c}\right)^{3.5} - 15.9618719 \cdot \left(1 - \frac{T}{T_c}\right)^{4} + 1.80122502 \cdot \left(1 - \frac{T}{T_c}\right)^{7.5}\Bigg) \quad (6)$$

where P_s is the saturation water vapor pressure at the given temperature, T_c is the critical point temperature and T is the temperature of the fluid.

Few notable simplifications were made while calculating evaporative heat loss. First, the evaporation rate is different in an occupied versus an unoccupied pool. Occupied pools should have a higher rate of evaporation due to an increase in the contact area. Also, higher occupancy means higher movement of air particles around the water surface. This increase in air velocity aids evaporation as well. For calculation, a baseline activity factor was assumed. This means that the pool was considered to be unoccupied. Second, we can see from Equation 5 that the evaporation rate is directly proportional to the air velocity over the water surface. But as stated earlier, this model swimming pool is maintained under ASHRAE specifications. ASHRAE[1] suggests the air velocity to be held between 0.0508-0.1524 mps to restrain evaporative heat loss. This process requires auxiliary ventilation, but operational energy demand for the auxiliary ventilation system is not accounted for in the calculation.

4.3 Radiative Heat Loss

Radiation heat transfer occurs in the form of heat emission from the body to the ambient environment in the form of long-wave radiation[34]. The radiative heat loss can be calculated by using the classical Stefan-Boltzman Equation as shown below:

$$\dot{Q}_{rad} = A_p \cdot \epsilon_w \cdot \sigma \cdot \left(T_{wlr}^4 - T_w^4\right) \tag{7}$$

where $\dot{Q}_{rad}$ is the radiative heat loss rate, A_p is the area of the pool, σ is the Stefan-Boltzman Constant, T_{wr} is the pool water temperature, T_{wlr} is the wall temperature, and ϵ_w is the Emissivity Constant. Chow et al.[23] have used the value of 0.9 in their recent study. The reference system used in this study is similar to the one used by Chow et al.[23], so a value of 0.9 can be used in this study.

4.4 Convective Heat Loss

Convective heat transfer occurs due to the movement of the pool water and ambient air. Convective heat transfer can be quantified by using Newton's Law of Cooling Equation as shown below:

$$\dot{Q}_{conv} = h_{conv} \cdot A_p \cdot (T_a - T_w) \tag{8}$$

where $\dot{Q}_{conv}$ is the convective heat loss rate, h_{conv} is convective heat transfer coefficient, A_p is the area of the pool, T_w is the pool water temperature, and T_a is the ambient air temperature.

Similar to evaporative heat loss, convective heat loss is also directly proportional to the wind speed. Australian Standard[35] proposed a model to evaluate convective heat transfer coefficient as shown below:

$$h_{conv} = 3.1 + 4.1 \cdot v \tag{9}$$

where h_{con} is the convective heat transfer coefficient, and v is the air velocity over the water surface.

4.5 Feed Water Heat Requirement

Water is lost to dry air molecules in the ambiance constantly due to evaporation. This loss in water volume has to be replenished frequently. In addition to that, the freshwater is circulated in the pool for sanitation as well. With supplementary feed water, it is safe to assume that this added water needs additional heat to reach the comfortable swimming temperature. Chow et al.[23] in their study mentioned that the daily feedwater flow rate, in general, is 5-10% of the total pool volume. For calculation, 5% refill rate is used. The mathematical model proposed by Buanamo et.al[36] is used to calculate refill water heat, as shown below :

$$\dot{Q}_{feed} = c_w \cdot m_{rf} \cdot (T_p - T_{rf}) \tag{10}$$

where, $\dot{Q}_{feed}$ is the feedwater heat lost rate, m_{rf} is the mass flow rate of refilling water, T_w is the pool water temperature, C_w is the specific heat of the water, and T_{rf} is the feedwater temperature.

5 Solar Thermal Collector

5.1 Introduction

Solar energy is free and abundantly available in the atmosphere; it's just a matter of capturing it and converting it to the required form of energy. A Solar Thermal Collector, as the name suggests, collects heat by absorbing sunlight. Solar Collector Technology is regarded as one of the main applications of Solar Energy Engineering. There are several configurations of solar thermal collector; some major ones are mentioned below[37]:

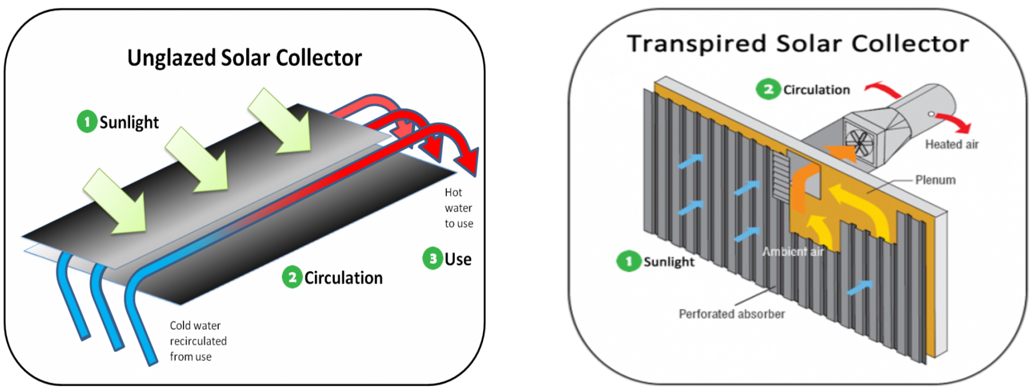

Figure 4: Unglazed and Transpired Solar Collectors[3]

Figure 4 shows an unglazed and transpired solar collector, respectively. An unglazed solar collector is the simplest form of a solar thermal collector. The unglazed solar collector consists of a receiver plate that receives heat from sunlight and a tubing pipe through which heat transfer fluid is circulated. The receiver plate is made of a material with high solar absorptivity, thus absorbing a significant amount of thermal energy from the sun and transferring it to a tubing pipe where the water is heated. The transpired solar collector is reasonably straightforward as well. It is comprised of two major components: a receiver plate and a fan. The receiver plate absorbs solar energy, whereas a fan circulates air through a plenum in the back of a receiver. Transpired solar collector is primarily used in applications where heating of space is required.

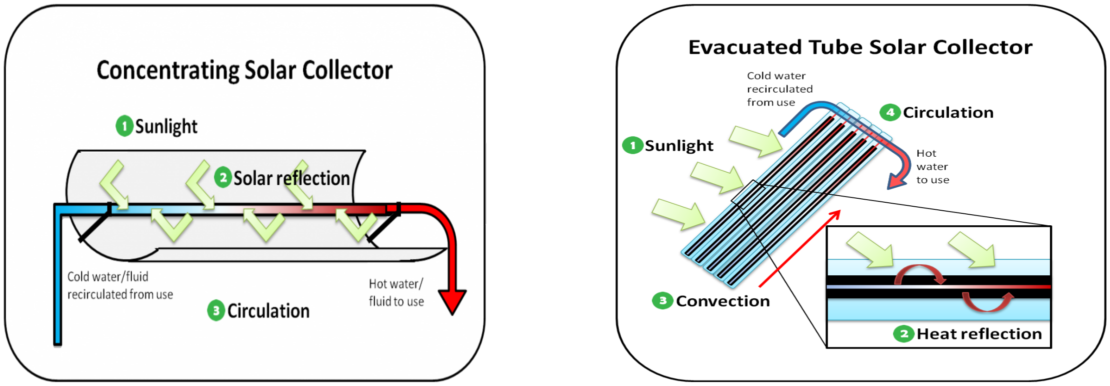

Figure 5: Concentrating and Evacuated Tube Solar Collectors[3]

Figure 5 shows concentrating and evacuated tube collector, respectively. Concentrating solar collectors consist of three major components: trough or parabolic-shaped reflector, receiver, and tubing pipe. The reflector is usually made up of material with low solar absorptivity and high solar reflectivity. Solar irradiation is reflected through the reflector surface to the absorber plate made up of material with high solar absorptivity. This absorbed heat is transferred to the tubing pipe, where the heat transfer fluid is heated. This type of thermal collector is typically used where higher temperatures are required. An evacuated tube solar collector is mainly comprised of evacuated cylinder and tubing. Evacuated cylinder, made up of material with high solar absorptivity, absorbs thermal energy and transfers it to tubing usually made out of copper.

A flat-plate Collector configuration is the one that is widely used in residences, commercial, and industrial buildings for hot water and space conditioning (heating). The flat plate collector system is similar to the unglazed solar collector, except that it has a glazing layer to retain long-wavelength thermal radiation inside the collector. Sunlight strikes the receiver plate, which absorbs solar thermal energy and heats up. A cool fluid, usually water, is circulated through the tubing of the collector, where the heat exchange occurs. This heated fluid can be used for various applications as desired.

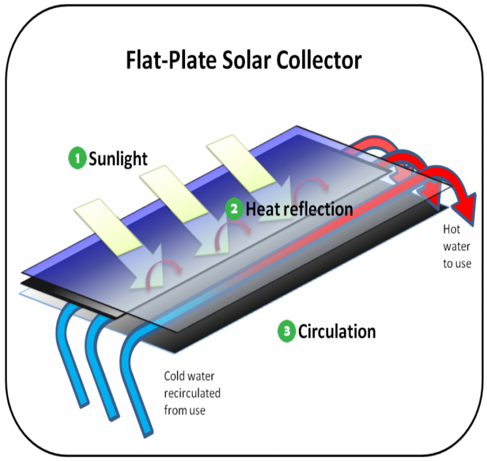

Figure 6: Flat Plate Solar Collector[3]

5.2 Fundamentals

Figure 7 below shows the cut view of a flat plate solar collector. All the components of the collector are housed in an enclosure that provides structural support and protection. Glazings are transparent cover sheets that pass most of the solar radiation to the absorber and provide protection. Glazing is typically made of glass with high solar transmissivities. The absorber plate is the primary component that absorbs heat. The absorber plate is generally made of a material that has high solar absorptivity and low emissivity. Water flowing through the tubes is heated by energy from the absorber plate. To minimize the heat loss, the collector bottom that is in contact with the absorber plate is highly insulated.

Figure 8 shows the simple schematic of the flat plate solar collector system[3]. This system consists of two primary flow loops: one from the solar collector to the storage and one from the storage system to application load. The hot fluid is drawn from the repository for whatever application it is intended, and cool fluid is returned at reduced temperature to the storage tank. Solar energy supply is not consistent, so an auxiliary heating system is essential in energy deficiencies. Adding few control system components can make this system sophisticated but effective. For instance, using the real-time difference between inlet and outlet water temperature and available solar irradiance, the circulation system can be

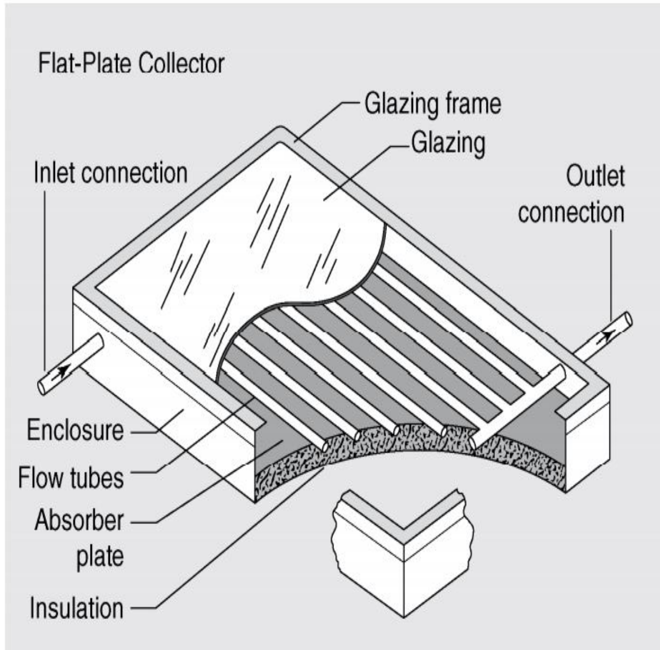

Figure 7: Flat Plate Collector Cut-away [3]

controlled to regulate optimal temperature. This can improve the efficiency of the system by quite a bit.

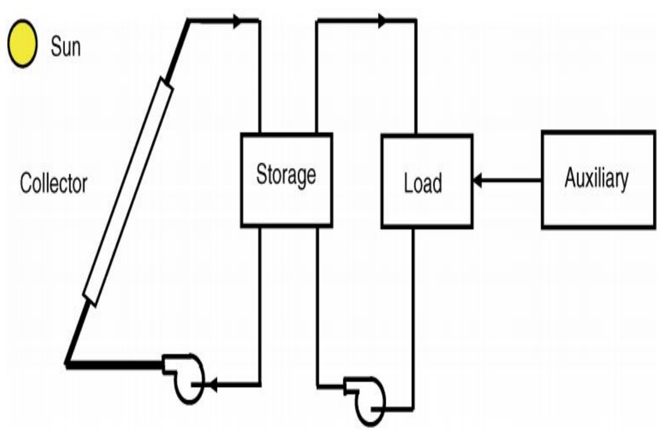

Figure 8: Flat Plate Collector System Schematic[3]

5.3 Efficiency of Flat Plate Collector System

The energy flow for a flat-plate collector system is shown as follows:

$$q_{useful} = q_{abs} - q_{loss} \tag{11}$$

where q_{abs} is the solar irradiation absorbed by the collector, q_{loss} are the conduction and radiation losses from the collector to the environment, and q_{useful} is the useful energy available.

The solar irradiation absorbed by the absorber plate must pass through the glass cover plates. The absorber plate can only absorb a portion of energy, while the glass cover plate can only transmit a part of radiation incident on it. The energy absorbed by the absorber can be mathematically represented as:

$$q_{abs} = I_T \cdot A_c \cdot \alpha \cdot \tau \tag{12}$$

where I_T is the irradiation intensity, α is the absorptivity of the absorber plate, τ is the transmissivity of the glass cover plate, and A_c is the collector surface area.

Heat lost through conduction and radiation from the collector to the environment can be quantified as shown below:

$$q_{loss} = U_L \cdot A_c \left(T_{ave} - T_a\right) \tag{13}$$

where U_L is the overall conductance, T_{ave} is the average temperature of the collector, and T_a is the average ambiance temperature.

The useful energy available from Equation 11, can be updated as follows:

$$q_{useful} = I_T \cdot A_c \cdot \alpha \cdot \tau - U_L \cdot A_c \left(T_{ave} - T_a\right) \tag{14}$$

The efficiency of any given collector can be defined as the proportion of useful energy extracted out of total incident irradiation. It is shown as follows:

$$\eta_C = \frac{q_{useful}}{I_T \cdot A_c} \tag{15}$$

where η_c is the efficiency of the collector, q_{useful} is the useful energy available, and $(I_T \cdot A_c)$ is the total incident energy.

Substituting the expression for q_{useful} from Equation 14 in Equation 15, we get:

$$\eta_C = \frac{I_T \cdot A_c \cdot \alpha \cdot \tau - U_L \cdot A_c \left(T_{ave} - T_a\right)}{I_T \cdot A_c} = \tau \cdot \alpha - U_L \frac{T_{ave} - T_a}{I_T} \tag{16}$$

Therefore, Equation 16 can be used to determine the solar collector's efficiency. It can be seen from the Equation that collector efficiency, η_c is the linear function of temperature difference divided by the irradiation. So, specification for a given thermal collector alone cannot define the system's efficiency, and it depends on several factors, especially irradiation at a particular location. Equation 16, however analyses the collector efficiency in terms of average collector temperature, T_{ave}. A more useful equation involving the relation between inlet temperature and output temperature can be found by introducing the collector heat removal factor,F_R, to Equation 16. This equation is sometimes called the Hottel-Whillier-Bliss equation and is considered the most important equation related to the flat plate collector. Given that we know the irradiation intensity, inlet fluid temperature, and ambiance temperature at a location, SRCC certification for any flat plat collector should have all information to calculate the solar collector's efficiency accurately. Hottel-Whillier-Bliss equation is shown below:

$$\eta_C = F_R \cdot \tau \cdot \alpha - F_R \cdot U_L \frac{T_{ave} - T_a}{I_T} \tag{17}$$

where F_R is the heat removal factor.

6 Thermal Calculation

Using Equation 6, $P_s(T_w)$, saturation vapor pressure at pool water temperature can be calculated as follows:

$$log_e \frac{P_s(T_w)}{22064000} = \frac{647.096}{299.82} \times (-4.02708)$$
$$P_s(T_w) = e^{-8.74929} * 22064000$$
$$\approx 3498.78\ Pa$$

Similarly, $P_s(T_a)$, saturation vapor pressure at surface water temperature can also be calculated using Equation 6 as follows:

$$log_e \frac{P_s(T_a)}{22064000} = \frac{647.096}{300.93} (-3.99971)$$
$$P_s(T_a) = e^{-8.68415} * 22064000$$
$$\approx 3734.26\ Pa$$

Using values of $P_s(T_w)$ and $P_s(T_a)$ as calculated above and from Equation 5, G_w, evaporative mass flow rate can be calculated as follows:

$$G_w = (4.2 \cdot 10^{-8}) \cdot (64 \cdot 18.29) \cdot [3498.78 - 0.5 \cdot 3734.26]$$
$$= 0.0802\ \frac{kg}{s}$$

Finally, using the value of G_w as calculated above and from equation 4, the evaporative heat loss can be estimated as follows:

$$\dot{Q}_{evap} = 2260 \cdot 0.08458 = 181.28\ kW$$

From Equation 7, $\dot{Q}_{rad}$, radiative heat loss per hour can be calculated as follows:

$$\dot{Q}_{rad} = (64 \cdot 1.287) \cdot 0.9 \cdot (5.67 \cdot 10^{-11}) \cdot (300.92^4 - 299.81^4) = 7.19\ kW$$

From Equation 9, h_{conv}, convective heat transfer coefficient can be calculated as:

$$h_{conv} = 3.1 + 4.1 \cdot 0.1016 = 3.516\ \frac{W}{m^2 \cdot^\circ C}$$

Using value of h_{conv} as calculated above and from Equation 8, $\dot{Q}_{conv}$, convective heat loss per hour can be calculated as follows:

$$\dot{Q}_{conv} = 3.516 \cdot (64 \cdot 18.287)\,(27.77 - 26.66) = 4.57\ kW$$

Using Equation 10, $\dot{Q}_{feed}$ can be calculated as follows:

$$\dot{Q}_{feed} = \frac{4.184 \cdot (0.05 \cdot (64 \cdot 18.29 \cdot 2.5)) \cdot 1000 \cdot (26.66 - 15)}{24 \cdot 60 \cdot 60} \approx 83.09\ kW$$

Using values of thermal losses obtained from equations as calculated above, the total thermal $\dot{Q}_{total}$ can be updated as follows:

$$\begin{aligned}\dot{Q}_{total} &= 191.17 + 7.19 + 4.57 + 83.09 \\ &= 276.14\ kW\end{aligned}$$

The proportion of the heat loss per category is shown in the pie-chart below:

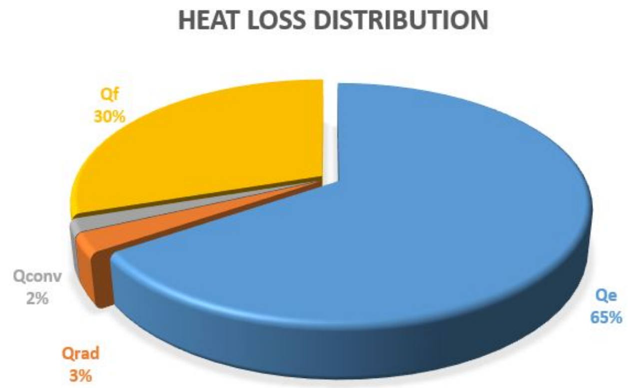

Figure 9: Pie Chart Heat Loss Distribution

7 Sizing Thermal Collector

7.1 Efficiency Calculation of Selected Collector

The solar collector used to supply energy demand for the reference swimming pool is Model AE-40, manufactured by Florida based company named Alternate Energy Technologies. This option was selected because this solar thermal collector model is cost-effective and efficient compared to several others taken into consideration. The OG-100 ICC-SRCC^{TM} certification for the solar collector is shown in the figure 10 below.

To size the thermal collector, the selected model's efficiency needs to be calculated at the specified location. Since the thermal collector's efficiency depends on Solar Irradiance, we need to find and input the average solar irradiance value for Oxford, MS. Oxford's latitude is close to 34° N. Several researchers have collected the irradiance data at latitude 32° N, which passes through Meridian, MS. The irradiance value at latitude 32° N is highly similar to that in Oxford, MS, and it can be used to size the thermal collector at Oxford. The Table 1 shown below is reproduced from Principles of Solar Engineering (2^{nd} edition), showing average daily solar irradiance values at each month of the year.

From SRCC certification for Model AE-40 as shown in Figure 10, the intercept and slope value can be obtained for efficiency calculation:

$$\text{Intercept} = 0.760$$

$$\text{Slope} = \text{-}6.215\ \frac{W}{m^2 \cdot {}^\circ C}$$

Now the efficiency of the collector can be calculated using Equation 16 can be calculated as:

$$\eta_C = F_R \cdot \tau \cdot \alpha - F_R \cdot U_L \frac{T_{ave} - T_a}{I_T}$$

$$\eta_C = 0.760 - 6.215 \times \frac{28.61 - 16.38}{6964} = 0.749 \approx 75\%$$

OG-100 ICC-SRCC™ CERTIFIED SOLAR COLLECTOR # 10001820

SOLAR RATING & CERTIFICATION CORPORATION

Supplier:
Alternate Energy Technologies
1345 Energy Cove Court
Green Cove Springs, FL 32043 USA
www.aetsolar.com

Reference Standard:
SRCC Standard 100-2010-08

Brand:	AlternateEnergy
Model:	AE-40
Collector Type:	Glazed Flat Plate
Certification #:	10001820
Original Certification:	April 16, 2013
Renewal Expiration Date:	November 30, 2020

Certifications must be renewed annually.

This solar collector has been evaluated and certified by the Solar Rating & Certification Corporation™ (ICC-SRCC™), an ISO/IEC 17065 accredited and EPA recognized Certification Body, in accordance with the SRCC OG-100 Certification Program, as defined in SRCC OG-100, Operating Guidelines and Minimum Standards for Certifying Solar Collectors. This award of certification is subject to all terms and conditions of the current OG-100 Program Agreement and the documents incorporated therein by reference. This document must be reproduced in its entirety.

COLLECTOR THERMAL PERFORMANCE RATINGS

Kilowatt-hours (thermal) Per Panel Per Day				Thousands of Btu Per Panel Per Day			
Climate -> / Category (Ti-Ta)	High Radiation (6.3 kWh/m².day)	Medium Radiation (4.7 kWh/m².day)	Low Radiation (3.1 kWh/m².day)	Climate -> / Category (Ti-Ta)	High Radiation (2000 Btu/ft².day)	Medium Radiation (1500 Btu/ft².day)	Low Radiation (1000 Btu/ft².day)
A (-5 °C)	17.5	13.2	9.0	A (-9 °F)	59.6	45.1	30.8
B (5 °C)	15.3	11.1	6.9	B (9 °F)	52.4	37.9	23.6
C (20 °C)	12.2	8.1	4.0	C (36 °F)	41.7	27.5	13.6
D (50 °C)	6.3	2.7	0.1	D (90 °F)	21.6	9.1	0.2
E (80 °C)	1.3	0.0	0.0	E (144 °F)	4.6	0.0	0.0

A- Pool Heating (Warm Climate) **B**- Pool Heating (Cool Climate) **C**- Water Heating (Warm Climate) **D**- Space & Water Heating (Cool Climate) **E**- Commercial Hot Water & Cooling

TECHNICAL RESULTS				**Tested in accordance with:** ISO 9806:1994			
ISO Efficiency Equation [NOTE: Based on gross area and (P)=Ti-Ta]							
SI UNITS:	η= 0.755 - 5.22470(P/G) - 0.01950(P²/G)			**Y Intercept:**	0.760	**Slope:**	-6.215 W/m².°C
IP UNITS:	η= 0.755 - 0.92081(P/G) - 0.00191(P²/G)			**Y Intercept:**	0.760	**Slope:**	-1.095 Btu/hr.ft².°F
Incident Angle Modifier							
θ	10^0	20^0	30^0	40^0	50^0	60^0	70^0
Kτα	1.00	1.00	0.99	0.97	0.93	0.79	
Impact Safety Rating: 11							

COLLECTOR SPECIFICATIONS					
Gross Area:	3.690 m²	39.72 ft²	**Dry Weight:**	69.4 kg	153.0 lb
Net Aperture Area:	3.475 m²	37.40 ft²	**Fluid Capacity:**	4.6 liter	1.2 gal
REMARKS:					

Shawn Martin

Technical Director

Print Date: October 30, 2020 Page 1 of 4
Please verify certification is active on the SRCC website www.solar-rating.org.

Figure 10: OG-100 Certification for Model AE-40 by Alternate Energy Technology[4]

Month	Solar Insolation at 32°with Hoz.$\left(\frac{\mathbf{W}}{\mathbf{m^2}}\right)$
January	6329
February	7250
March	7574
April	7426
May	7199
June	7042
July	7092
August	7237
September	7275
October	6960
November	6241
December	5951
Average	**6964**

Table 1: Average Solar Insolation (Oxford,MS)[2]

7.2 Sizing of Selected Thermal Collector

The total hourly energy requirement from the calculations as done in the above section is :

$$\begin{aligned} \text{Hourly Energy Requirement} &= 276.14 \text{ kW} \times 1h \\ &= 276.14\ kWh \end{aligned}$$

The Turner Center pool has an average run-time of 10 hours. So, assuming we only need to meet the thermal demand for 10 hours, the total energy requirement for the day is :

$$\begin{aligned} \text{Total Daily Energy Requirement} &= 276.14 \times 10 \\ &\approx 2761.4\ kWh \end{aligned}$$

From Table 1, the approximated average solar insolation value for Oxford, MS is :

$$\text{Average Solar Insolation} = 6964\ \frac{\text{Wh}}{\text{m}^2 \cdot \text{day}}$$
$$= 6.964\ \frac{\text{kWh}}{m^2 \cdot \text{day}}$$

From Equation 7.1, the efficiency of the selected thermal collector, i.e. Model AE-40 from Alternate Energy Technologies at Oxford, can be approximated to be 75%. This means that only 75% of the incident solar energy is converted to useful energy. Thus, useful average solar insolation is :

$$\text{Useful Average Solar Insolation} = 6.964 \times 0.75$$
$$\approx 5.223\ \frac{\text{kWh}}{m^2 \cdot \text{day}}$$

Given the total energy demand and applicable incident solar radiation, we can calculate the total collector area required to meet the energy demand. The calculation is :

$$\text{Collector Area required} = \frac{2761.4}{5.223}$$
$$\approx 530\ m^2$$

But, we know that sunlight is not available all 365 days a year. To design a self-sustaining system, we need to consider this fact and size the thermal collector accordingly. The number of clear days per year in Oxford is :

$$\text{Number of Clear Days in Oxford, MS} = 217\ \text{days[38]}$$
$$\text{Fraction of Clear Days per Year} = \frac{217}{365} \approx 0.6$$

This fraction obtained can be used to scale up the system to meet the annual demand. One thing to note is that since solar energy is not constantly available for this system to function as expected, a way to store excess energy needs to be devised.

$$\text{Actual Collector Area} = \frac{530}{0.6} \approx 890m^2$$

From Figure 10 i.e OG-100 ICC-SRCC^{TM} certification for the selected thermal collector model AE-40, the area of the single collector panel can be found as:

$$\text{Area of single collector panel} = 3.475\ m^2$$

Given the collector area required to meet the thermal energy demand and area of individual area of Model AE-40, the total number of units of Model AE-40 required can be calculated. The calculation is shown as follows:

$$\text{Required Units of AE-40 Panels} = \frac{890}{3.475} \approx 250\, units$$

The area of the collector panel alone might not provide the scale of the project. To fathom the scale of the project, a ratio between the area of the pool and the area of the thermal collector is calculated as shown below:

$$\text{Ratio of Collector Area to Pool Area} = \frac{890}{1152} \approx .75$$

This means that the total solar thermal collector area is 75% of the entire pool surface area. One important thing to note here is that this number gives a straightforward comparison of the thermal collector area to the surface area of the pool area. This ratio doesn't give us an idea of the area required to install these solar thermal collectors. To find the actual land area to install required units, several factors like inclination angle of thermal collector, spacing between two thermal collectors, etc., must be considered.

7.3 Inter-Row Spacing for No Shadow Casting

The actual land area required to lay the solar panels is much higher than the total collector area for several reasons. First, for the land area to equal collector area, the panels have to be laid flat. The optimal inclination for maximum energy extraction is 32°; the panels cannot be laid flat. Since the panels are inclined, shadows from the solar panels are generated. Therefore, to ensure no shadow interference among consecutive rows of solar panels, optimal spacing needs to be delegated. The row spacing arrangement is shown in Figure 11 :

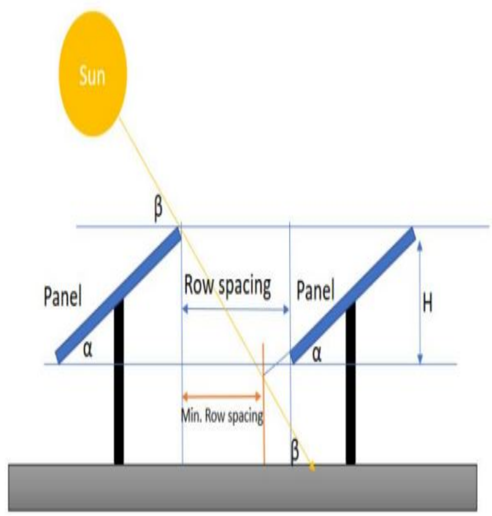

Figure 11: Row Spacing Schematic

Shadow formation and optimal spacing can be calculated using the sun path. Shadows are the longest during the winter solstice, so we use sun path data for that day. The graph obtained from Oregon Lab is shown in Figure 12 below.

Considering the working window to be 8AM-3PM, from Figure 12 we obtain following information:

$$\text{Solar Elevation } (\beta) = 19°$$

$$\text{Azimuth Correction} = 42.5°$$

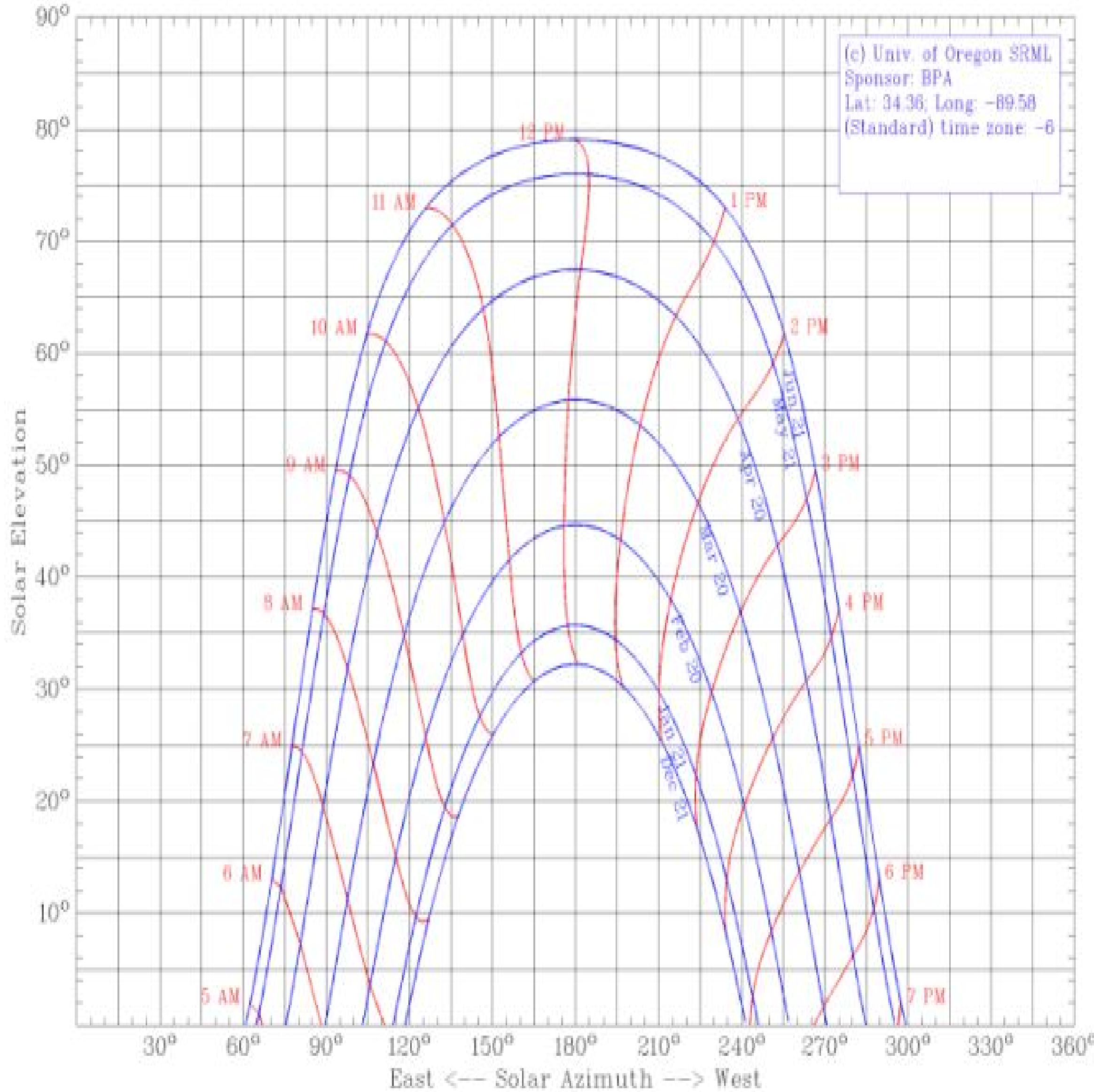

Figure 12: Sun Path Plot[5]

The gross area of the solar collector panel is then used to calculate the width of the panel as shown below:

$$\text{Width (W)} = \sqrt{\text{Area}} = \sqrt{3.690\ \text{m}^2} \approx 1.92\ \text{m}$$

In our setup, we use an angle of tilt (α) for the solar panel to be equal to 32°. Using the angle of tilt, the width of the panel, and Pythagoras Theorem, we calculate the height of the tip of the solar panel from the ground as follows:

$$Height(H) = \text{Width} \times \sin(\alpha) = 1.92 \times \sin(32^\circ) \approx 1.02 \text{ m}$$

Using Solar Elevation(β) as obtained from Figure12 and Pythagoras Theorem,

$$\tan(19^\circ) = \frac{\text{Height}}{\text{Spacing}}$$
$$\text{Spacing} = \frac{1.02}{\tan(19^\circ)}\ m \approx 2.96 \text{ m}$$

This spacing doesn't include azimuth correction, which means that the sun is assumed to be located straight from the solar panel. This assumption is not valid and can skew the result vastly if not corrected. From Figure 12, it can be seen that at 8 AM, the angle between the sun path and trailing edge of the solar panel should be at 42.5°. Using this Azimuth correction angle and Pythagoras Theorem, corrected spacing can be calculated as follows:

$$\sin(42.5^\circ) = \frac{\text{Corrected Spacing}}{2.96}$$
$$\text{Corrected Spacing} = \sin(42.5^\circ) \times 2.96\ m \approx 2.1 \text{ m}$$

Thus, to ensure no shadow casting among the consecutive rows, at least 2.1 m of space is required between rows.

8 Economics of Selected Thermal Collector

The total units of Model AE-40 solar thermal collectors to meet the energy demand has been calculated in the section above. The cost of an individual unit of Model AE-40 solar thermal collectors was obtained from the Web Solar Supplies catalog. The price is shown below:

$$\text{Unit Price of Model AE-40} = \$\ 1024.00[39]$$

The total cost of the solar thermal collector panel is calculated to be:

$$\begin{aligned}\text{Total Cost of Thermal Collector} &= \text{Price per Unit} \times \text{Total Units}\\ &= \$\ 1024 \times 250\\ &\approx \$\ 256{,}000\end{aligned} \tag{18}$$

The life cycle of a solar thermal collector is estimated to be somewhere between 25-30 years[40]. Thus, all the analysis and calculations are done for 25 years mark. To analyze the economic feasibility and environmental impact, the cost of installing and maintaining solar thermal collectors was compared with the cost of grid electricity.

O&M cost is equally crucial as initial seed cost in determining any project's economic feasibility. O&M cost in this report includes module cleaning and vegetation management cost, components part replacement cost, operations administration cost, and pest control cost. Provided that all solar thermal collector panels usually come along with the average of 25 years of the warranty period, which is the average lifetime of collector panels, product repair costs are excluded. Calculation of O&M cost could be pretty complicated as it depends on various factors such as location, availability of resources, and so on, but the United States Department of Energy (DOE) has proposed a mathematical model to approximate it. The equation as proposed by DOE is shown as follows:

$$\text{Estimated Annual Operating Cost per Unit} = \frac{365 \times 12.03}{\text{SEF}} \times \text{EC}[41] \tag{19}$$

where, SEF is Solar Energy Factor and EC is the electricity cost per kWh.

Solar Energy Factor (SEF), as referred to in Equation 19, can be defined as the energy delivered by the system divided by the electrical or gas energy put into the system. The higher the number, the more energy-efficient. Solar energy factors range from 1.0 to 11. Systems with solar energy factors of 2 or 3 are the most common.

For our calculation, we use the average value of 2.5 for SEF, and the electricity cost in Oxford, MS is 10.245¢ per kWh[42]. Using these values, Equation 19 can be updated as follows:

$$\begin{aligned}\text{Estimated Annual Operating Cost per Unit} &= \frac{365 \times 12.03}{2.5} \times 0.10245 \\ &\approx \$180\end{aligned}$$

So, the total estimated cost of the solar thermal collection over one lifetime can be calculated as follows:

$$\begin{aligned}\text{Total Cost Over One Life-Time} &= \text{Annual O\&M} \times \text{Life-Time} \times \text{no. of Units} + \text{Initial Cost} \\ &= \$\ 180 \times 25 \times 250 + \$256,000 \\ &\approx \$1{,}381{,}000 \end{aligned} \tag{20}$$

Thus, from Equation 20, it can be seen that the total cost of installation and maintenance of Model AE-40 from Alternate Technologies to meet the thermal energy demand for swimming pool situated at Turner Center at the University of Mississippi over 25 years is $ 1,381,000.

8.1 Cost Comparison: Solar vs Conventional Energy Source

In this section, a cost comparison is performed between conventional energy sources and solar thermal collector panels. A comparison between electric grid energy cost and solar thermal collector cost is made in the first section. In the second section, a comparison between natural gas cost and solar thermal collector cost is made.

8.1.1 Solar vs Grid Electricity Cost

From Equation 20, the total cost over one lifetime (25 years) for the collection of solar thermal collector panel can be given as:

$$\text{Total Cost Over One Life-Time} = \$1{,}381{,}000$$

The grid electricity in Oxford, MS is 10.245¢ per kWh.The total cost to meet the energy demand for the reference swimming pool using grid electricity can be calculated as shown below:

$$\begin{aligned}
\text{Grid Electricity Cost Over 25 Years} &= \text{Daily Energy Demand} \times \text{Cost} \times \text{Total days in 25 Years} \\
&= 9{,}941{,}145 \text{ kJ} \times \frac{1 \text{ kWh}}{3600 \text{ kJ}} \times \frac{0.102405\,\$}{\text{kWh}} \times 25 \times 365 \\
&\approx \$2,580,200
\end{aligned}$$

So, the potential savings over 25 years using solar thermal collectors instead of grid electricity can is calculated below:

$$\begin{aligned}
\text{Potential Savings Over 25 Years} &= \text{Total Cost Grid Electricity} - \text{Total Cost Thermal Collector} \\
&= \$2,580,200 - \$1,381,000 \\
&\approx \$\,1.2 \text{ M}
\end{aligned}$$

8.1.2 Solar vs Natural Gas Cost

The real-time supply and demand dictate the price of natural gas. It could very well change dramatically over a short period. The average cost provided by United States Energy Information Administration (EIA) could be considered reliable. The average value of natural gas, as given by EIA, for the year 2019 is 10.51 \$/Mcf[43].

Energy density is the amount of energy stored in a given system or region of space per unit volume. Different kind of fuels differ in terms of energy density. The energy density of natural gas is 1050 Btu/ft^3[44].

Using the energy density and cost for natural gas and also, provided that we know the thermal energy demand, the operational fuel cost over 25 years period to meet the thermal load for the reference swimming pool with natural gas can be calculated as:

$$\begin{aligned}\text{Natural Gas Cost} &= \text{Daily Energy Demand} \times \text{Cost} \times \text{Total days in 25 Years} \\ &= \left(9,941,145 \text{ kJ} \times \frac{1 \text{ Btu}}{1.05506 \text{ kJ}}\right) \times \left(\frac{1\, ft^3}{1050 \text{ Btu}} \times \frac{10.50\,\$}{1000 ft^3}\right) \times 25 \times 365 \\ &\approx \$\, 859,789\end{aligned}$$

The cost calculated above only depicts the energy cost and doesn't include any machinery or maintenance cost. Only comparing the energy cost, natural gas is cheaper than grid electricity. So, natural gas would be the obvious choice over the grid electricity, if only the fuel cost is compared. However, this section is intended to indicate the negative environmental impact of using natural gas over perpetual solar energy. The amount of CO_2 emitted per million Btu of natural gas is 117 lbs, as mentioned by EIA. Also, most of the gas pool heaters used nowadays have an efficiency rating of 89-95%[45]. The average efficiency value of 92% is used in the calculation. Using the statistics as mentioned above, the total amount of prevented CO_2 emission over 25 years can be calculated as :

$$\begin{aligned}
\text{Prevented Emmission} &= \text{Emission Rate} \times \text{Energy Demand per day} \times \text{Total days in 25 Years} \\
&= \left(\frac{117\,lbs}{10^6\,Btu \times 0.92}\right) \times \left(9,941,145 \text{ kJ} \times \frac{1 \text{ Btu}}{1.05506 \text{ kJ}}\right) \times 25 \times 365 \\
&\approx \text{10,930,000 lbs of } CO_2 \approx 4960 \text{ metric tons of } CO_2
\end{aligned}$$

Even though natural gas would seem like a better option economically, the environmental impact of burning natural gas is countless. From the calculation above that installation of the solar thermal collector, to meet the thermal energy demand for the swimming pool at Turner Center at the University of Mississippi, could potentially prevent approximately 4960 metric tons of CO_2 emission over an average solar thermal collector panel life cycle which is 25 years.

9 Conclusion and Further Work

From the calculations presented in this report, there is a possibility of huge saving and prevention of significant CO_2 emission by switching from conventional grid electricity to solar thermal energy. The average thermal energy demand for the reference swimming pool was found to be 276.14kW. Assuming an average daily operational time of 10 hours, the total thermal load for a day was found to be 2761.4kWh. Several competing solar thermal collector models were considered, but Model AE-40 from Alternate Energy Technology was most suitable for this region. A solar thermal collector's efficiency depends on two significant factors, i.e., solar irradiance level and operational temperature range. Using the approximated solar irradiance value at Oxford, MS and desired working temperature range for our reference pool, the collector's efficiency was calculated to be 75%. Using the efficiency and average solar irradiation value, the total collector area required to fulfill the energy demand was 530m^2 and equaled 250 individual units of Model AE-40 solar thermal collectors. The total thermal collector area needed to meet the pool's energy demand is 75% of the pool's total surface area.

Regarding the economics, the solar thermal collector's initial seed cost was determined to be $256,000. The solar thermal collector has an estimated average life of 25 years. So, the total cost for the solar thermal collector, including O&M cost over 25 years, was found to be $1.3 million. The grid electricity cost to meet the equivalent thermal demand over 25 years was calculated to be approximately $2.58 million, and thus, the potential savings over 25 years was found to be $1.2 million.The environmental impact of using solar collector panels is immense. Switching to the solar thermal collector from natural gas to supply the thermal load to maintain the swimming pool could prevent 4960 metric tons of CO_2 emission.

This study indicated numerous benefits of switching to solar thermal energy technologies from conventional grid electricity. But a more detailed analysis is yet to be done to draw more conclusive results. The thermal calculations performed were solely for the pool maintained with conditions as prescribed by ASHRAE. A complete analysis of thermal load, including natural ambiance conditions, occupancy of the pool, and all auxiliary operations such as ventilation and space, is required to design a self-sustaining system in a true sense. In addition to that, a more sophisticated system is needed in real life that includes a storage tank, pipings, and other control systems. Heat loss or efficiency of those systems are completely ignored in this study. More efficient results could be obtained, including the variances as mentioned above. All the deficiencies discussed above are some of the potential areas for further study in the future.

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