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Use of a Parabolic Trough Collector in the Kingdom of Bahrain Conditions for Water Desalination

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Abstract

In this study, the use of a parabolic trough collector (PTC) to harness the solar energy has been investigated. An optimum design for the PTC was selected based on the various geometric aspects that affect the performance of the collector. A linear model from the literature was then employed to study the performance of the PTC in the climatic conditions of the Kingdom of Bahrain. As a sample, data for the months of August and January were used to compare the PTC operation in the hot and cold seasons. These two months are generally the hottest and the coldest months of the year. Maximum PTC output temperatures of 55.4°C and 37.5°C were obtained for the months of August and January, respectively, using a collector area of only 1.6 m^2. Such temperatures are enough for low-temperature water desalination technologies like the Natural Vacuum Distillation (NVD). The article further discusses the thermal energy storage options to allow for operation during the non-sunlight hours of the day.

Keywords: Solar Energy, Parabolic Trough Collector, Renewable Energy, Energy Storage

1. Introduction

Solar energy is the most readily available renewable energy source. Due to the increasing demand of energy and the environmental hazards associated with the combustion of fossil fuels, solar energy is gaining more and more attention as an alternative energy source (Delyannis and El-Nashar, 2010). A major advantage of solar energy is that it is environmentally clean (Kalogirou, 2006). It can be utilized through conversion to electricity by means of Photovoltaics (PV) or directly as Solar Thermal Energy (STE). A recent review of the different solar energy technologies by Chu and Meisen (2011), has shown that there is no clear advantage of PV over STE or vice versa. However, they did mention that STE is more predictable and allows cheaper and easier energy storage than PV. Solar thermal energy is harnessed by using solar collectors, which convert the energy from incident solar radiation to useful heat energy.

Solar collectors can be classified into two main categories: non-concentrating and concentrating collectors. Non-concentrating collectors have a concentration ratio (CR) of one or less, while the concentrating types have much higher values. CR is the ratio of the absorber area to the intercepting area of the collector. Another advantage of the concentrating collectors is that they can be designed for solar tracking.

Several designs for the concentrating collectors are available. They include the Cylindrical Trough Collectors (CTC), Parabolic Trough Collectors (PTC), Compound Parabolic Collectors (CPC), Heliostat Field Collectors (HFC), Parabolic Dish Collectors (PDC) and the Linear Fresnel Collectors (LFC). They can be designed for single-axis solar tacking, like the CTC, PTC, CPC and LFC, which have a CR values up to 85 or for two-axis tracking, like the PDC and the HFC, which have CR values of up to 2000 (Kalogirou, 2009). Among the single-axis tracking, concentrating collectors, the PTC has the highest CR values and is simple in design. It is a high-performing solar collector with high heat efficiency for operating temperatures of up to 400°C (Jebasingh and Herbert, 2016).

A PTC has the geometrical benefit of its shape that allows it to achieve high concentrating ratios. By the virtue of its parabolic shape, a PTC focuses direct solar radiation onto its focal line. A receiver tube, made from a high thermal conductivity material and with a working fluid flowing inside, is installed in this focal line. The fluid absorbs the thermal energy from the concentrated radiation and raises its enthalpy. The PTC can easily be set up for single-axis solar tracking to ensure that the solar beams intercepted by the collector are always normal to the collector surface. A PTC makes use only of direct solar radiation, called beam radiation or Direct Normal Irradiance (DNI). This is the fraction of solar radiation hat reaches the Earth's surface as a parallel beam and is not diverted by the presence of clouds or other particles in the air (Fernandez-Garcia et al., 2010).

Natural Vacuum Distillation (NVD) is a method based on using the barometric height of water to create a natural vacuum over the surface of the water with a sealed enclosure above it. A review of this process is presented by Rashid et al. (2016). In such a low-pressure (near-vacuum) condition, water can be evaporated at low temperatures. The use of a PTC with such a desalination technique allows for a low-temperature and low-pressure water desalination using renewable energy. The low pressure within the system allows the process to continue in the non-sunlight hours, but for a more efficient performance, an energy storage option should be considered.

2. PTC Design

The geometrical aspects of a PTC are illustrated in Fig. 1. The aperture area (A_a) is defined as the area through which the incident solar radiation will be intercepted. It is the product of the aperture width ($2W$) and the solar collector length (L). The solar beam incident angle (θ_i) is the angle at which the solar beams hit the collector surface. The focal length (f) is the vertical distance from the vertex to the focus of the parabola.

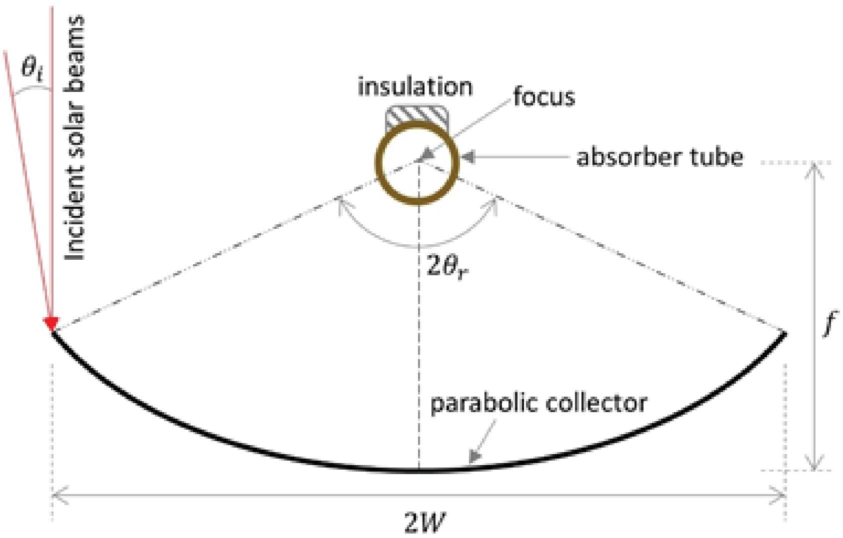

Figure 1: Geometrical Aspects of a Parabolic Trough Collector.

The most important aspect of a PTC is the concentration ratio (CR), which is defined as the ratio between the effective aperture area and the absorber tube area. A relation between the CR, the rim angle (θ_r) and the solar beam incident angle (θ_i) is given by Kandlikar and Vij (1978) as:

$$CR = \frac{1 - \sin\theta_i\ /\sin\theta_r}{\frac{\sin\theta_i}{\sin\theta_r}\left(\pi\frac{\theta_r+90-\theta_i}{180}\right)} \tag{1}$$

According to this equation, it can be seen that the concentration ratio has the highest value at a rim angle of about 70° for all incident angles, see Fig. 2. Hence, this rim angle was selected for the PTC to be used in the process. The width (W) of the PTC was chosen to be 0.8 m. The focal length, f, of the PTC is calculated as:

$$f = \frac{W}{4\tan(\theta_r/2)} \tag{2}$$

For a width of 0.8 m and a rim angle of 70°, f is calculated as 0.286 m. The equation of the parabola is then given by:

$$y = \frac{x^2}{4f} \tag{3}$$

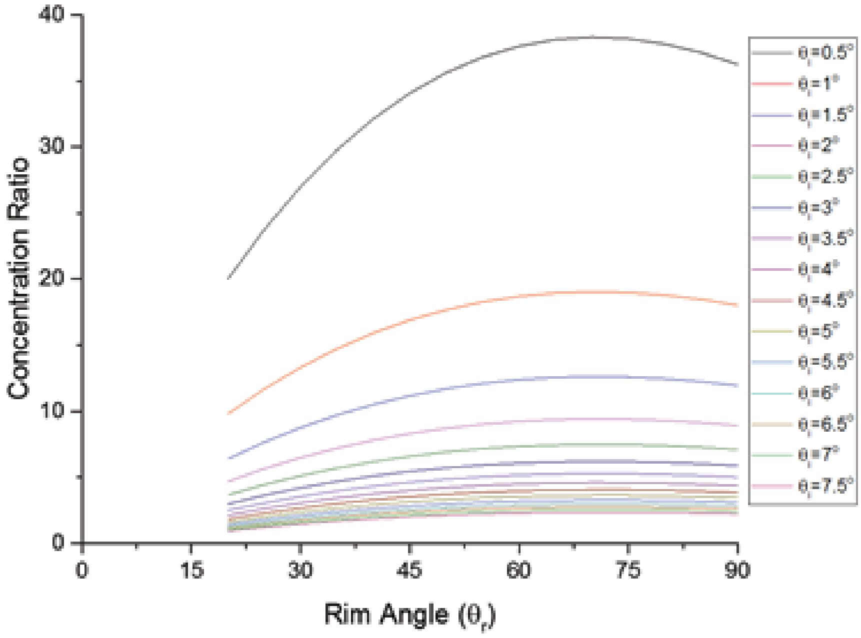

Figure 2: Concentration ratio as a function on the rim angle for different incident angles.

where the x values range from -0.4 m to +0.4 m. The minimum radius (R_{min}) for the absorber tube, which will be able to intercept all the rays focused to the focal point, for a given solar beam incident angle and rim angle is given by the equation:

$$R_{\text{min}} = \frac{2f\sin\theta_i}{1+\cos\theta_r} \tag{4}$$

For an incident angle of 1° and a rim angle of 70°, this is calculated as 0.0037 m (i.e. 3.7 mm) or a diameter of 7.4 mm. The selected absorber tube diameter was, therefore, 12.7 mm (which is the diameter for a ½″ nominal size copper tube). The absorber tube can be shielded by an evacuated glass envelope to minimize convection heat losses from the absorber tube to the environment. The thickness of this glass envelope tube should be enough to withstand the atmospheric pressure, and at the same time, it should not be thick to cause optical losses of the incident solar radiation.

3. Mathematical Modeling

To model the PTC, the following assumptions were made:

- a steady-state model has been assumed
- the model is a one-dimensional model (the temperature rise along the length of the PTC is negligible)
- the sky is a blackbody at an equivalent sky temperature

- the temperature gradients around the absorber tube and the glass envelope are negligible
- the reduction in the incident solar beams on the reflector surface due to the shadow of the absorber tube is negligible
- heat losses due to the joints at the end of the absorber tube and other support structure are negligible

The energy components in the absorber tube are shown in Fig. 3. The red arrows indicate the solar energy absorbed by the glass tube and the absorber tube. The other components of energy flowing in the absorber tube are:

- two conduction heat transfer quantities ($q_{cond,a}$ and $q_{cond,g}$ through the absorber tube and the glass envelope walls respectively),
- three convection heat transfer quantities ($q_{conv,f}$ for the fluid in the absorber tube, $q_{conv,a\rightarrow g}$ in the air trapped between the absorber tube and the glass envelop, and $q_{conv,g\rightarrow air}$ for the convection losses from the glass envelop to the surrounding air),
- and the radiation losses ($q_{rad,a\rightarrow g}$ for the radiation losses from the absorber tube surface to the glass envelope and $q_{rad,g\rightarrow sky}$ for the radiation losses from the surface of the glass envelope to the sky).

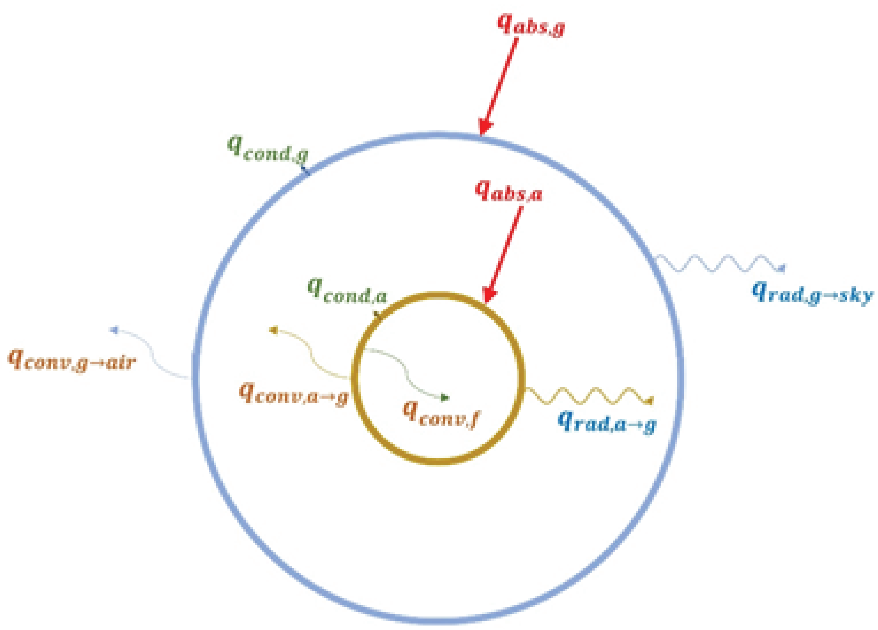

Figure 3: Flow of energy in the PTC absorber tube.

The losses from the system can be calculated by the equations shown below using the thermal network shown in Fig. 4.

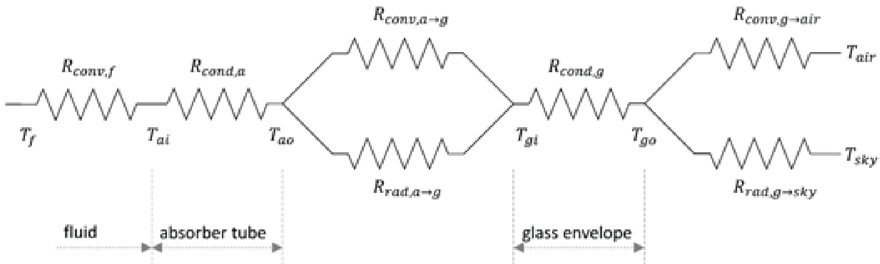

Figure 4: Thermal resistance network showing the heat losses in the PTC.

The energy balances on the absorber tube and the glass envelope gives us the equations 5 to 14. The various heat flux values ($\dot{q}'$) are functions of the temperatures at the respective boundaries and can be solved to find the temperatures at the different boundaries, as shown in Fig. 4.

$$\dot{q}'_{focus} = \frac{I_{dn}.\rho.\gamma.I_{AM}.A_a}{L} \tag{5}$$

$$\dot{q}'_{abs,g} = \dot{q}'_{focus}\alpha_g \tag{6}$$

$$\dot{q}'_{abs,a} = \dot{q}'_{focus}\tau_g\alpha_a \tag{7}$$

$$\dot{q}'_{abs,g} + \dot{q}'_{rad,a\to g} + \dot{q}'_{conv,a\to g} = \dot{q}'_{rad,g\to sky} + \dot{q}'_{conv,g\to air} \tag{8}$$

$$\dot{q}'_{abs,a} = \dot{q}'_{rad,a\to g} + \dot{q}'_{conv,a\to g} + \dot{q}'_{conv\_f} \tag{9}$$

$$\dot{q}'_{Abs} = \dot{q}'_{abs,g} + \dot{q}'_{abs,a} \tag{10}$$

$$\dot{q}'_{Thermal\,Loss} = \dot{q}'_{rad,g\to sky} + \dot{q}'_{conv,g\to air} \tag{11}$$

$$\dot{q}'_u = \dot{q}'_{Abs} - \dot{q}'_{Thermal\,Loss} \tag{12}$$

$$\dot{q}'_{cond\_a} = \dot{q}'_{conv\_f} \tag{13}$$

$$\dot{q}'_{cond\_g} = \dot{q}'_{rad,a\to g} + \dot{q}'_{conv,a\to g} \tag{14}$$

where $\dot{q}'_{focus}$ is the total solar radiation reflected on to the glass envelope, I_{dn} is the direct normal solar irradiation, ρ is the parabolic surface's reflectance (= 0.9), γ is the intercept factor of the absorber tube (= 0.95) as given by Goswami and Kreith (2008), I_{AM} is the incident angle modifier given by I_{AM}= $1{+}0.00031780_i{-}0.0000398\,5\theta_i^2$, $\dot{q}'_{abs,a}$, $\dot{q}'_{abs,g}$ and $\dot{q}'_{Abs}$ represent the heat absorbed by the absorber, the glass tube and the total solar energy absorbed, $\dot{q}'_u$ is the useful energy in the PTC.

4. Results and Discussion

A linear model for the simulation of a PTC has been presented by Qu et al. (2006) which was validated using experimental data. A similar model was used with the solar irradiation and temperature data from the Kingdom of Bahrain to investigate the performance of the PTC in Bahrain conditions. The parameters used are listed in Table 1. Climatic data for the months of August and January were studied. These are generally the two hottest and coldest months of the year respectively. The hourly temperatures and solar irradiation values were obtained from the Bahrain Meteorological Directorate for the above-mentioned months. These are the average values for the years 2006-2010.

TABLE 1: Parameters used in the PTC model.

Parameter	Value	
PTC length (L)	2.0	M
PTC width (W)	0.8	M
Focal Length (f)	0.286	M
Glass tube transmissivity (τ_g)	0.82	
Glass tube absorptivity (α_g)	0.11	
Glass tube diameter (d_g)	6.35	cm
Glass tube thickness (t_g)	2.8	mm
Glass tube thermal conductivity (k_g)	1.05	W/m.K
Absorber tube transmissivity (τ_a)	0.01	
Absorber tube absorptivity (α_a)	0.96	
Absorber tube diameter (d_a)	12.7	mm
Absorber tube thickness (t_a)	1.245	mm
Absorber tube thermal conductivity (k_a)	377	W/m.K

The results of the simulation of the PTC along with the solar irradiation values and the ambient temperatures are shown in Fig. 5 and 6 for the months of August and January respectively. The solar irradiation value for August peaked at around 800-850 W/m^2 while it was only around 600-700 W/m^2 during January. The PTC output can be observed to have a maximum temperature of around 55.4°C during August and around 37.5°C during January. These temperatures are high enough for domestic heating or for processes that operate at low to moderate temperatures. During the non-sunlight hours, the PTC output temperature is seen to be same as the ambient temperature.

Figures 7 and 8 show the comparison of the total energy absorbed by the solar collector and the useful energy gained by the water flowing in the absorber tube for the selected days in the months of August and January respectively. The total solar

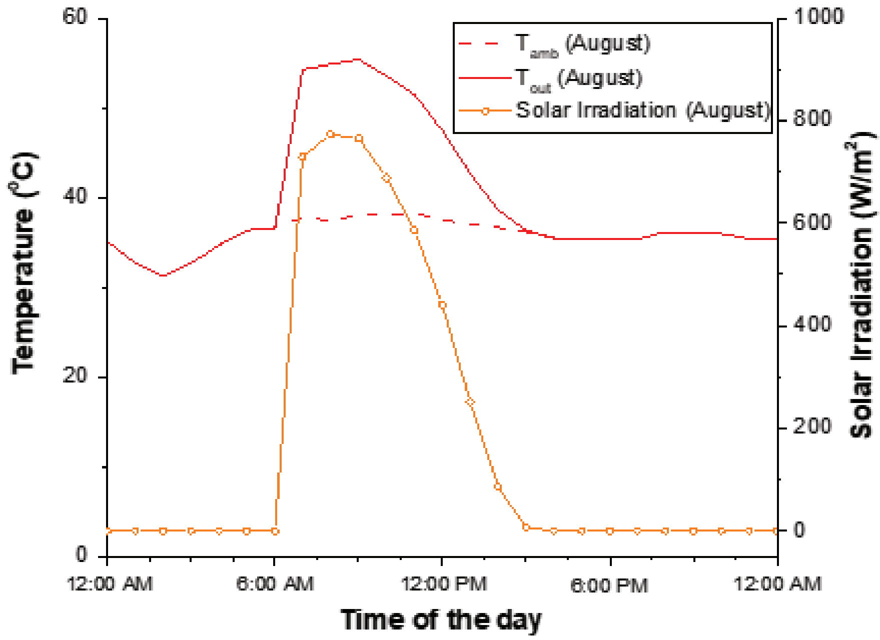

Figure 5: Solar Irradiation and Input and Output temperatures of the PTC during the month of August.

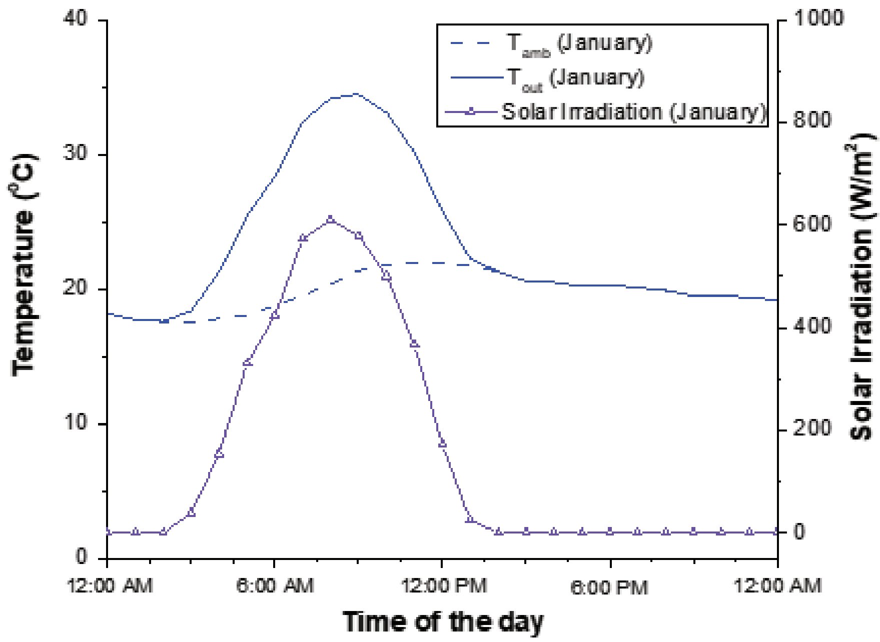

Figure 6: Solar Irradiation and Input and Output temperatures of the PTC during the month of January.

energy absorbed by the collector reaches a maximum value of around 1250 W on the summer day and around 1000 W on the winter day. The useful energy follows a trend similar to the absorbed energy. The difference between the two curves can be attributed to the thermal losses in the collector. It also gives us an idea about the efficiency of the collector. The overall thermal efficiency for the PTC is defined as the

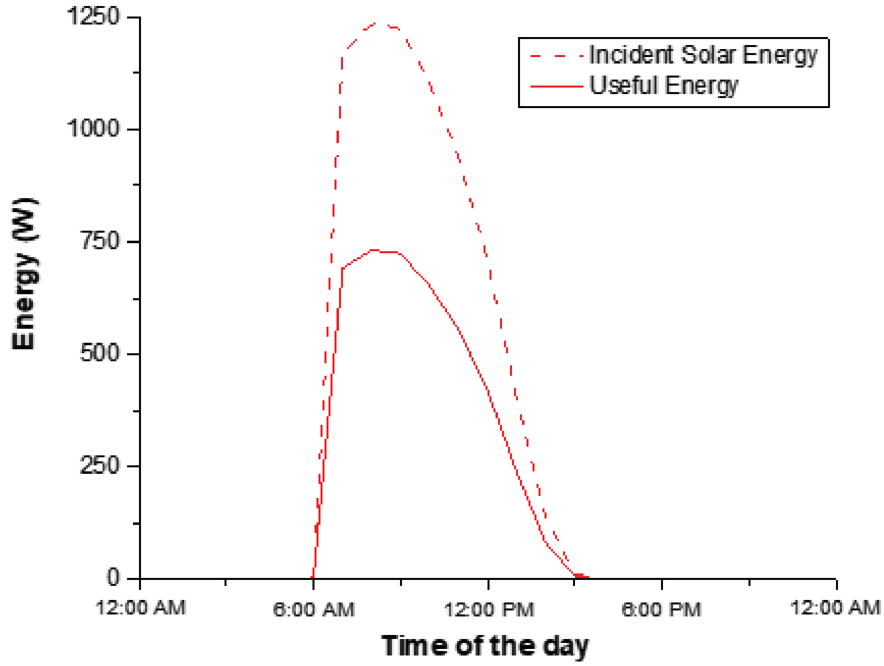

Figure 7: Comparison of the solar energy incident on the collector and the useful energy gained in the month of August.

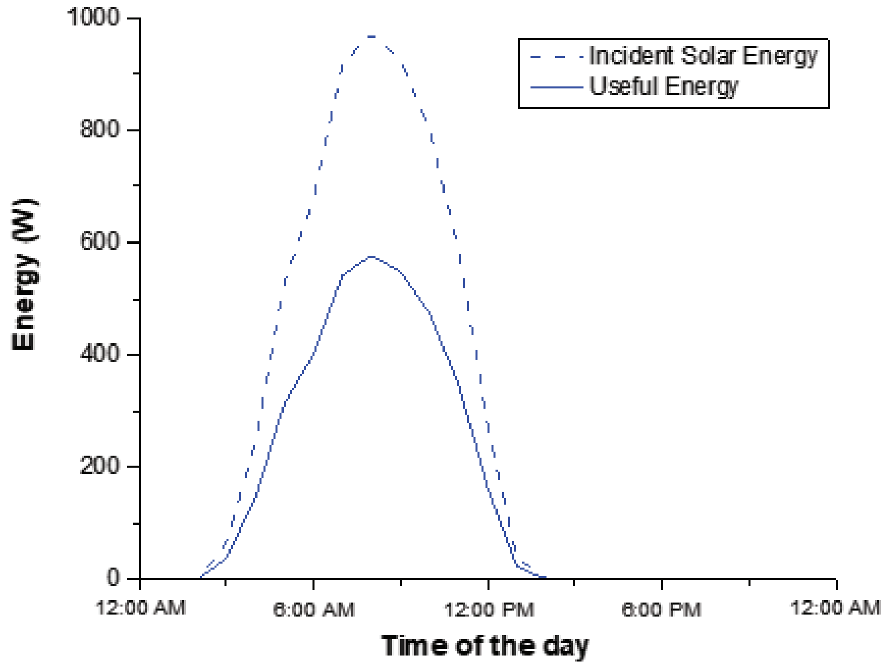

Figure 8: Comparison of the solar energy incident on the collector and the useful energy gained in the month of January.

ratio between the useful energy (Q_u) obtained from the collector to the overall solar energy intercepted. It is given as:

$$\eta = \frac{\int Q_u dt}{A_a \cdot \int I_{dn} dt} \tag{15}$$

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The overall thermal efficiency was calculated as 58.09%.

5. Thermal Energy Storage

Energy storage is becoming a vital part of renewable energy storage systems (Dinker et al., 2017). For a solar thermal collector, thermal energy storage system should be considered. Thermal energy storage (TES) is defined as the technology that stores thermal energy by heating or cooling a storage medium that can be used at a later time for heating or cooling applications (Sarbu and Sebarchievici, 2016). The major TES technologies are shown in Fig. 9.

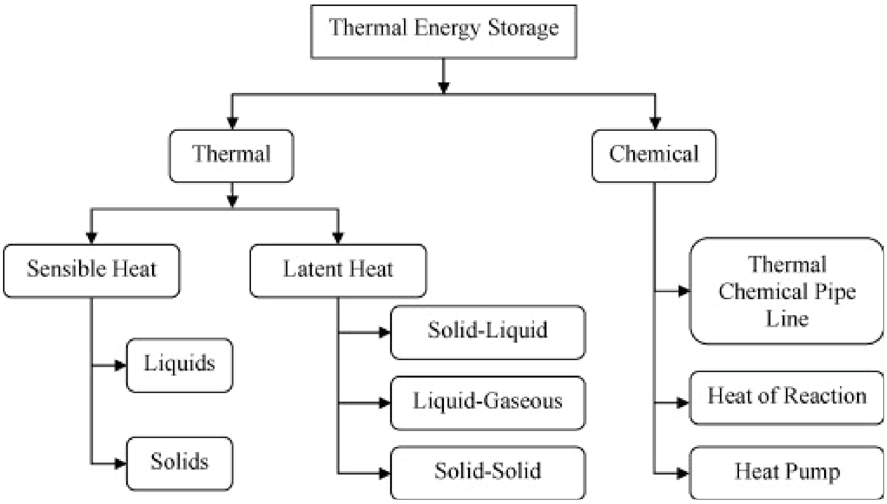

Figure 9: Types of Thermal Energy Storage (as shown by Bal et al., 2010).

The three major categories of TES, as seen from Fig. 9, are Sensible heat storage, Latent heat storage and Chemical Energy storage. For a process liked NVD which operates at low temperature, sensible heat storage (SHS) is the most suitable because of its low cost and simplicity. This method is based on the storage of thermal energy by heating a liquid or solid storage medium. SHS systems use the heat capacity of the medium and the change in the temperature during the charge and discharge phases, and the amount of energy stored also depends on the amount of storage medium available (Kumar and Shukla, 2015).

Table 2 shows the most used materials for SHS and their properties as presented by Ayappan et al. (2016). As can be seen clearly from the above table that the best option is clearly water. And for a water desalination process, this choice becomes natural. Such an option has been studied by Gude et al. (2012) in their study for a water desalination system using a thermal energy storage tank where water was used for the heat storage. They concluded that this SHS device allows for an uninterrupted

Table 2: Most common used SHS materials and their properties (taken from Ayappan et al., 2016).

Medium	Fluid type	Temp. Range (°C)	Density (kg/m^3)	Sp. Heat (J/kg.K)
Sand	-	20	1555	800
Rock	-	20	2560	879
Brick	-	20	1600	840
Concrete	-	20	2240	880
Granite	-	20	2640	820
Aluminium	-	20	2707	896
Cast iron	-	20	7900	837
Water	-	0-100	1000	4190
Calorie HT43	Oil	12-260	867	2200
Engine oil	Oil	≤160	888	1880
Ethanol	Organic liquid	≤78	790	2400
Propane	Organic liquid	≤97	800	2500
Butane	Organic liquid	≤118	809	2400
Isotunaol	Organic liquid	≤100	808	3000
Isopentanol	Organic liquid	≤148	831	2200
Octane	Organic liquid	≤126	704	2400

operation of the desalination process and also that the larger tanks are more suitable than the smaller tanks. A typical configuration for a water tank used as a thermal energy storage for desalination system using a solar collector is shown in Fig. 10. High specific heat capacity, ease of availability, chemical stability and low cost make water a good storage media suitable for low temperature solar applications.

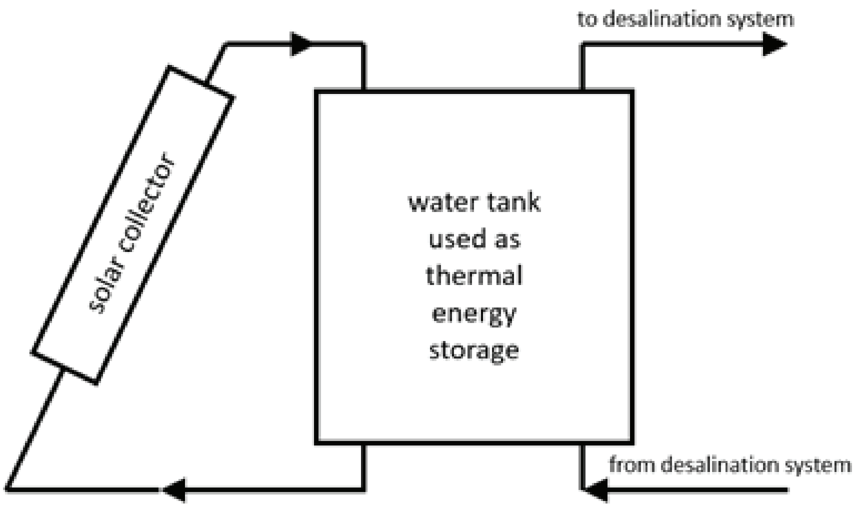

Figure 10: A typical configuration for a water tank TES for solar desalination.

6. Conclusion

A linear one-dimensional model has been developed for a PTC. A 1.6 m^2 PTC was simulated using this model to study its performance in the climatic conditions of the Kingdom of Bahrain. Maximum temperatures of 55.4°C and 37.5°C were obtained for the months of August and January respectively. The overall thermal efficiency for the PTC was calculated as 58.09%. These results show that even with relatively small PTC designs, the output temperatures are high enough for domestic heating or for processes that operate at low temperatures. Uses of PTC include domestic heating, desalination, refrigeration systems, industrial heat, power plants, pumping irrigation water and solar chemistry (Lamba, 2012). When used for water desalination, a thermal energy storage system should be used for continuous operation. A water tank can be used for the easiest and cheapest SHS medium. The size of the tank will depend on the operating temperature and the capacity of production.

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