



Structure of Parabolic Trough Collector Model for Local Heating and Air Conditioning

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Abstract:

Concentrating solar power (CSP) is a type of solar energy that uses mirrors (concentrators) to concentrate sunlight from a large area to a small area where it is absorbed and converted to heat at high temperatures. CSP plants have a big advantage over photovoltaic (PV) power plants because they can use conventional fuels and store thermal energy to make up for the fact that solar energy doesn't always work. In this paper, a parabolic trough collector (PTC) with the following parameters was designed to investigate the efficiency of a small-scale PTC to heat a synthetic heat transfer fluid that may be used for domestic heating or cooling. PTC 2 m in length, 30 cm

rim radius (r_r), 23 cm for the focal length (f), while the receiver tube diameter (D_o) equals 33.4 mm. It was found that the concentration ratio of the PTC was 5 times while the optical efficiency was 25 % considering all imperfections resulted from the manufacturing process. The heat losses from the PTC including convection, conduction, and radiation were calculated, and the overall heat loss coefficient was $2.8 \text{ w/m}^2 \text{ K}$ while the PTC's overall thermal efficiency was 24.1%. The temperature of heat transfer fluid measured by a thermocouple connected to an Arduino board showed that temperatures increased almost linearly with time. The temperature rose by $16 \text{ }^\circ\text{C}$ in five minutes from an ambient temperature of $18 \text{ }^\circ\text{C}$.

Keywords: *Arduino, solar energy, parabolic trough collector, receiver tube, concentrated solar power.*

Introduction

Solar thermal technology has the potential to provide a significant contribution to meeting global energy demand and reducing greenhouse gas emissions. One promising technology for

solar thermal applications is the parabolic trough collector (PTC), which has been widely used for electricity generation and process heat applications. Recently, there has been increasing interest in the use of PTCs for local heating and



air conditioning applications. In this regard, a model of the parabolic trough collector structure for local heating and air conditioning has been proposed by (Shi et al., 2022). The proposed model aimed to optimize the PTC performance by considering the structural design and material selection of the collector. The study conducted a parametric analysis to evaluate the effects of different design parameters on the PTC performance. The results showed that the collector's structural design and material selection significantly influenced the collector's thermal performance.

Predominately, solar thermal technologies innovations depend on concentrating sunlight-based radiation to deliver steam or hot air for power generation (Cabrera et al., 2013). Unlike photovoltaic panels which utilize global heat irradiance to generate electricity, CSP has a high solar power factor, approximately five times higher than PV; however, their implementation is constrained due to technical and financial reasons. Green energy resources, however, have become under increased scrutiny in recent years due to environmental concerns, which increases the likelihood that PTCs will be commercially competitive in the energy market.

Parabolic-shaped mirrors are made of a reflective material that concentrates received sunlight onto the receiver tube at the focal line of the collector. The mirror should be constructed of highly reflective material having high reflectivity for all irradiation wavelengths. silver coated glass is the most common material used in PTC, yet a low-cost silver-coated polymer film called Reflectance which can be placed on any nonporous surface has been developed recently (Mahian et al., 2013; Viebahn, Kronshage, & Lechon, 2008).

Heat transfer fluid (HTF) transfers heat from the receiver tube to a steam-generated heat exchanger by circulation consisting of thermal oil that is stable at high temperatures. Traditionally, PTC uses a synthetic oil called "thermos-oil," a eutectic mixture of biphenyl and diphenyl oxide that is liquid until 120 °C and staples up to 4000 °C. Some PTC plants use mineral oil, for it is less expensive; however, its

instability at relatively low temperatures of 3000 °C is one of its main disadvantages. Recently, a cheaper and more effective HTF was developed, consisting of a mixture of molten salt fertilizers. Molten salt which consists of 60 % NaNO₃ and 40 % KNO₃ is not -toxic or flammable, and it is stable at higher temperatures up to 540 °C. However, its main disadvantage is the higher freezing temperature of 120-220 °C which requires a robust antifreeze system (Duffie & Beckman, 2013).

This paper aims to design a small-scale unit of PTC that can be fit on the roof of any house with an integrated heat storage system. The heat collected by the PTC will not be used to produce electricity but it will be well utilized for space heating and air conditioning and providing hot water for domestic purposes. Space heating can be achieved by using a heat exchanger to preheat the boiler water. Space air conditioning can be achieved by using the heat to operate the absorption chiller which can be sold with the system instead of conventional compressor-based air conditioning units. Also, the heat exchanger can be added to provide hot water for domestic use. This article constitutes of six parts. Following this introduction, a literature review will be introduced in part two. Part three describes the framework while part four gives a detailed description of the PTC design and Experimental Setup. Part 5 depicts the results and discussion of the current work while part 6 presents the major conclusions and recommendations.

Literature Review

In this regard, several studies have proposed models for the design and optimization of PTCs for local heating and air conditioning. For instance, Ajbar et al. (2022) presented a detailed model for the design and optimization of PTCs for space heating and cooling applications in buildings. The study evaluated the effect of various design parameters, such as collector length, width, and reflector surface quality, on the thermal performance of the PTC. The results showed that the proposed design could achieve

high thermal efficiency and could effectively provide heating and cooling for buildings.

Mutlak et al. (2011) has studied the key design elements that affect a PTSC's performance, including the mirror-receiver tube intercept factor, the spectral directional reflectivity of the mirror system, receiver tube misalignment, the effect of tracking errors, the end thermal loss, and the incident angle modifier. He carried out the performance analysis of PTSC with synthetic oil and water as working fluids, and he formulate the efficiency of solar parabolic trough collectors in terms of absorber wall temperature, absorber emissivity, wind speed, and radiation level to predict the performance of the PTC with any working fluid.

Similarly, Sathish et al. (2023) proposed a design optimization model for PTCs that incorporates the use of phase change materials (PCMs) for thermal energy storage. The study aimed to optimize the PTC design and PCM selection to achieve high thermal efficiency and reduce the cost of the system. The results showed that the proposed design could increase the PTC's thermal efficiency by up to 60% and reduce the cost of the system by up to 20%.

On the other hand, Clark (1982) has analyzed PTC design parameters that affect PTC performance such as the mirror system's reflectivity, the intercept factor for the mirror-receiver tube, the effect of tracking errors and receiver tube misalignment, thermal losses, and connected them to economic factors of PTC solar fields such investment tax credits, tax credits for energy equipment, the auxiliary system's income tax cost, the collector's installation cost, maintenance and tax charges, and fuel costs, and capital cost to develop a model to analyze energy costs produced by PTC power plants. Furthermore, a study by Goel et al. (Goel & Manik, 2023) proposed a design optimization of PTC for solar heating and cooling applications using genetic algorithm (GA) optimization. The proposed model aimed to maximize the thermal efficiency of the collector while minimizing the cost of the system. The results showed that the optimized design increased the collector's thermal

efficiency by 12% while reducing the cost of the system by 7%.

To overcome system size obstacles, Thomas (1994) has designed a small parabolic collector for a solar thermal research program in South Africa. In this study, the length of the complex is 5 meters, the width of the Aperture is 1.5 m, and the angle of the edges is 82.2 degrees while the surface is made up of Stainless-steel Sheets that are covered with SA-85 film. Two receivers were fabricated for comparison, one of them encapsulated in an evacuated glass cover. It has been found that the peak efficiency of 55.2% and 53.8% were reached with non-shielded and glass cover reception, respectively.

In Jordan, the utilization of parabolic trough collector systems for local heating and cooling is the subject of investigation. Jordan doesn't have any actual CSP extensions, however, there have been several small-scale tests. A parabolic trough collector has been used in the Dead Sea Spa Hotel (Kiwani et al., 2018) to generate boiling water that is used in the summer to power a two-arrange alkali retention chiller. Three lines of illustrated sun-oriented collectors make up the structure. Aluminum sheets with a thickness of 0.8 mm are used to create the reflectors, and an aluminum coating holds them together. Through a 38 mm steel pipe that is lined with 80 mm Rockwool, the HTF is distributed among the collectors.

Al-Salaymeh et al (2012) has been designed a project using parabolic trough collectors including multi-using amid for comparative research. On the roof of Mutah University's engineering department is where this project is situated. A Tri-Generation of cooling and heating, water distillation, and electricity was produced using CSP technology. The project consists of 40 CSP parabolic trough reflector panels that span 240 m² of solar matrix area and can create 150 L/hr. of distilled water in addition to 120 kW of thermal power (kW) peaks with 15 kW of electrical power (kW), 100 kW of heating, and 20 kW of cooling.

In another study, Alguacil et al (2014) designed a prototype of a parabolic trough collector located in Sanlucar la Mayor Solar Platform, in Seville

(Spain) to generate about 8 MW of thermal power. This prototype consists of containing this prototype Three parallel loops of 800 meters each making up the evaporator solar park (EVAP). Kolb et al. (2011) has shown that it was possible to evaluate potential next-generation high-temperature Molten-Salt Power Towers. A solar receiver has been innovated with the help of the U.S. Department of Energy's SunShot Program to concentrate solar irradiation using an air-particle mixture to move a gas turbine or a combined cycle at a much higher temperature than the state-of-the-art molten salt receivers.

Arzhakov et al (2016) have developed an expedited process for designing a parabolic trough solar field for use in industrial heating applications. Meteorological data from Ipoh, Malaysia, have been used to develop and simulate a parabolic trough solar heating system. The researchers have anticipated the development of parabolic trough solar collector systems for steam power cycles that generate energy, but there is also room for this technology to be applied to industrial heating needs. Al-Asfar et al (2014) has contributed to the development and evaluation of a parabolic trough collector's performance for PTC's commercial viability. Overall, the development of models for the design and optimization of PTCs for local heating and air conditioning has significant potential for reducing carbon emissions and promoting sustainable energy use. The optimization of the collector design, reflector surface quality, and incorporation of PCMs for thermal energy storage can significantly enhance the collector's performance and increase its economic viability.

Materials and Methods

In this study, a small house in the Al Hussein Bin Talal University Housing Residential Complex in Ma'an city-Jordan with an area of 150 m^2 has been chosen to implement this project (1). According to the ASHRAE handbook, table 1 shows the specifications of the location while figure 1 is a schematic view of the house and the parameters shown in Table 1.

Table 1. Average Climate Properties of the Location According to the ASHRAE Handbook

Parameter	Value
Latitude:	31.956° North
Longitude:	35.945° East
Wind speed:	3.6 m/s
Height above sea level:	1100 m
Outside temperature design(summer):	34 C°
Outside temperature design(winter):	3 C°
Outside relative humidity (Summer):	35 %
Outside relative humidity (winter):	60 %
Type of building:	Residential
The number of occupants:	Five

The following elements make up a space's heating load: 1) The rate of heat loss via all exposed walls, the ceiling, the floor, the windows, the doors, and the walls separating heated and unheated areas (parts of walls); 2) The amount of heat needed to reheat outside cold air that has entered the heated room through the window and door gaps as well as outside cold air that has entered as a result of doors opening and closing; 3) The load for home hot water; 4) all other loads, such as the load for emergency heating and the safety factor. Heat loss rate due to filtration has been calculated by air change method as shown in equations 1-3.

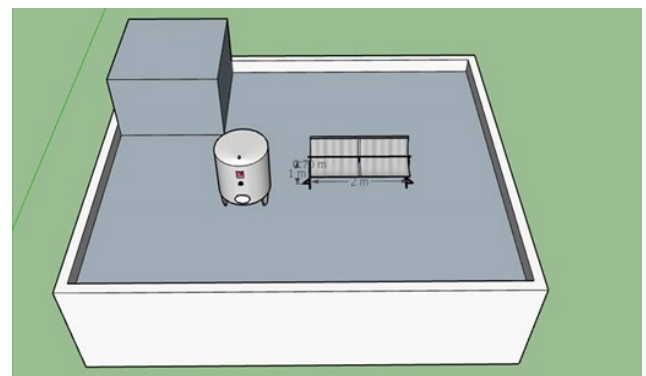


Figure 1. Schematic Diagram of the Proposed PTC Mounted on a Roof of a House

$$Q_{L,f} = \dot{m}_f(h_i - h_o) \quad (1)$$

$$V_f = N \times V_{room} \quad (2)$$

$$\dot{m}_f = \rho_i V_f \quad (3)$$

here, $Q_{L,f}$ is the heat loss due to infiltration (w); h_i, h_o is the heat transfer coefficient (kj/kg) for inside and outside the room; \dot{m}_f : mass flow rate of the infiltrated air (kg/s); V_{room} denoted the room volume (m³); N represented the number of times the air is changed in the room; and ρ_i is the density of the air (1.2 kg/m³). The heating load for domestic hot water is calculated as follows assuming daily consumption of 100 L/day and 10 % as the factor of safety to ensure the availability of hot water.

$$Q_{HW} = F.S \times \frac{1}{3600} \dot{m}_{HW} C_p (T_h - T_c) \quad (4)$$

$$Q_{HW} = 1.1 \times \frac{1}{3600} \times 1000 \times 4.186 \times 50 = 6.4Kw \quad (5)$$

Therefore, the total boiler load is the sum of the room's heating load and the domestic hot water load.; hence, the boiler load is 16.6 KW. Hot water heating systems consist of the piping network, circulating pump, radiators, burner expansion tank, fuel tank, hot water cylinder, fittings, and safety devices. The forced circulation hot water systems for winter heating included many types such as single pipe systems, double pipe systems, or a combination of single and double pipe systems or radial plastic pipe systems. The type used in the house used for this project is of double-pipe system arrangement, this type consists of a separate supply pipe network and a separate return pipe network. The separate supply pipe network is used to deliver hot water to each radiator while the separate return pipe network is used to collect the hot water leaving each radiator. A system of this type requires more pipe length than that of the single pipe system, but it is the main advantage is the even distribution of the temperature of all radiators. The double pipe systems are of two

types: reversed return pipe system and direct return system which are used in this project. An equivalent length for losses in fitting is added to the piping length, and it is nearly 50 % of the system length, in this project the equivalent length of heating pipes length is 45m.

PTC Design and Experimental Setup

PTC has a trough cross-section which has the shape of a parabola. It was envisaged that PTC reflective surface can be made of mirrors, but it was found that using pieces of a mirror to form a parabolic surface will not be effective. after a long search in the local market, it was found that stainless steel with a mirror surface that is used in building interior design may be used to design the reflective surface of PTC. the challenge, however, was to accurately reshape the stainless-steel sheet to form a parabolic shape with the required parameters. therefore, a professional stainless-steel fabricator was asked to do the job. The design parameters of PTC are as follows:

Rim radius (r_r) which is the distance between the focal point of PTC and the vertex (rim) of the reflective surface is 30 cm while the focal length (f) is the vertical shortest distance between r the reflector focal point and the reflector surface. It is worth mentioning that the rim radius and focal length are the longest and shortest radius of the PTC, respectively. The reflectivity (ρ) of the mirror stainless steel is about 0.65 as found in the manufacturer's specifications. This value is low and may be due to the composition of the surface, coated with chromium instead of silver which absorb solar radiation. The receiver tube is made also of stainless of an outside diameter (D_o) equal to 33.4 mm and an inside diameter (D_i) equal to 31.4mm is placed in the focal line of the PTC. plexiglass (transparent plastic) tube of inner diameter (D_{gi}) equals 41.6 mm is coaxially installed to cover the receiver tube (Figure 2), and both ends of the receiver tube are connected with flexible pipes into the heat exchanger tank. (Figure 3) shows a photo of the designed PTC.



Figure 2. Receiver and Glass Cover Tubes Arrangements Used in



Figure 3. General View of the Designed PTC



Figure 4. The 12V DC Hydraulic Pump Used in PTC

The heat transfer fluid is circulated in the receiver tube by a 12v DC hydraulic pump as shown in Figure 4. The inlet and exit of the heat transfer fluid to the heat exchanger tank are equipped with temperature and pressure sensors which are programmed with Arduino to read the values of the sensor values into an excel file. Figure 5 shows the arrangement of temperature

and pressure sensors. Finally, the whole system is positioned toward the sun by single-axis tracking powered by an AC motor as shown in Figure 6.



Figure 5. Temperature and Pressure Sensors Connection to Arduino Uno



Figure 6. AC Motor and Track Used in the Single-Axis Tracking System

Results and Discussions

Firstly, the overall heat losses (U_L) from the PTC are calculated. These losses include heat radiation, conduction, and convection. However, since a glass cover tube is inserted concentrically with the receiver tube to minimize air circulation causing a significant reduction in convection heat transfer, it may possible to eliminate heat loss by convection term. Therefore, overall PTC heat losses including only conduction and radiation are given by the following equation:

$$U_L = \left[\frac{A_r}{(h_w + h_{r,c-a}) \times A_g} + \frac{1}{h_{r,r-c}} \right]^{-1} \quad (6)$$

where, U_L : is the overall heat losses per receiver unit area ($w/m^2.k$). A_r : is the receiver area (m^2). $A_r = \pi D_o L = \pi \times 0.0334 \times 2 = 0.2099m^2$. h_w are the heat losses by convection due to the wind from the glass cover ($w/m^2.k$). $h_{r,c-a}$ are the heat losses by radiation from glass cover to ambient ($w/m^2.k$). A_g is the area of the glass cover (m^2). $A_g = \pi D_g L = \pi \times 0.0416 \times 2 = 0.2613 m^2$. $h_{r,r-c}$ are the heat losses by radiation from the receiver to glass cover ($w/m^2.k$).

Heat losses by the wind from the glass cover to ambient ($h_w = h_{c,c-a}$) depends on Nusselt Number which in turn depends on Reynold number and is calculated from the following equation.

$$h_w = h_{c,c-a} = \frac{(Nu) \times k}{D_g} \quad (7)$$

where, Nu : is Nusselt number. D_g : is the diameter of the glass cover (m). For laminar flow Nusselt number is calculated from equation 8 while the Reynold number for wind around the glass cover is calculated by equation 9 at ambient air temperature.

$$Nu = 0.3 \times Re^{0.6} \quad (8)$$

$$Re = \rho \times V \times D_g / \mu \quad (9)$$

At ambient temperature ($T_a = 20 C$), air density (ρ), air viscosity (μ), and air thermal conductivity (κ) are $1.1774 kg/m^3$, $1.983 \times 10^{-5} kg/m.s$, $0.02624 w/m.k$ respectively. Assuming wind speed (V) is 5 m/s then the diameter of the glass cover (D_g) equals 41.6 mm while wind speed (V) is assumed to be 5 m/s.

$$Re = 1.1774 \times 5 \times 0.0416 / 1.983 \times 10^{-5} \quad (10)$$

$$Nu = 0.3 \times (12350)^{0.6} \quad (11)$$

$$h_w = h_{c,c-a} = \frac{85.5 \times 0.0264}{0.416} = 5.43 w/m^2.k \quad (12)$$

Heat loss by radiation from the receiver to the glass cover ($h_{r,c-a}$) is calculated by equation 13 but with the glass cover temperature (T_g) slightly higher than the receiver tube temperature. Therefore at $T_g = 25c$, heat loss by radiation will be:

$$h_{r,c-a} = \varepsilon_g \sigma (T_g + T_a) (T_g^2 + T_a^2) \quad (13)$$

where, ε_g : is the emissivity of the glass cover which equals 0.94 for plexiglass. σ : Stefan-Boltzmann constant. Therefore:

$$h_{r,c-a} = 0.94 \times 5 \times 10^{-8} (298 + 293) (298^2 + 293^2) = 4.85 w/m^2.k \quad (14)$$

But the heat loss by radiation from the receiver to the cover ($h_{r,r-c}$) is calculated by equation 15 as shown below where the emissivity of the receiver (ε_r) is 0.6.

$$h_{r,r-c} = \frac{\sigma (T_r + T_g) (T_r^2 + T_g^2)}{\frac{1}{\varepsilon_r} + \frac{A_r}{A_g} \left(\frac{1}{\varepsilon_g} - 1 \right)} \quad (15)$$

where, A_r, A_g : are the areas of the receiver and glass cover, and their values are $0.2099m^2$ and $0.2613m^2$, respectively. T_r : receiver temperature. its value is assumed to be 30 C i.e 303 k.

$$h_{r,r-c} = \frac{5.67 \times 10^{-8} (303 + 298) (303^2 + 298^2)}{\frac{1}{0.6} + \frac{0.2099}{0.2613} \left(\frac{1}{0.94} - 1 \right)} = 3.58 w/m^2.k \quad (16)$$

$$U_L = \left[\frac{0.2099}{(5.43+4.85) \times 0.2613} + \frac{1}{3.58} \right]^{-1} = 2.80 \text{ w/m}^2.k \quad (17)$$

Secondly, the heat transfer coefficient to heat transfer fluid inside the receiver need to be calculated, and this can be done by equation 18.

$$U_0 = \left[\frac{1}{U_L} + \frac{D_o}{h_{fi} \times D_i} + \frac{D_o \ln[D_o/D_i]}{2\kappa} \right]^{-1} \quad (18)$$

where, h_{fi} : convective heat transfer coefficient for the heat transfer fluid inside the receiver which equals $300 \text{ w/m}^2.k$. κ is the thermal conductivity of the receiver which equals 47.6 w/m.k . Therefore, heat transfer coefficient from the receiver to the heat transfer fluid is given by equation 19.

$$U_0 = \left[\frac{1}{2.8} + \frac{0.0334}{300 \times 0.0314} + \frac{0.0334 \times \ln[0.0334/0.0314]}{2 \times 47.6} \right]^{-1} = 2.77 \text{ w/m}^2.k \quad (19)$$

The useful heat (Q_u) delivered to the receiver is calculated by equation 21 where (G_B) is the beam radiation. the average beam radiation in Ma'an area is 700 w/m^2 . The receiver aperture area (A_a) is:

$$[w_a - D_g]L = [0.508 - 0.0416] \times 2 = 0.9328 \text{ m}^2 \quad (20)$$

$$Q_u = G_B \times \eta_o \times A_a - A_r U_L (T_r - T_a) \quad (21)$$

$$Q_u = 700 \times 0.25 \times 0.9328 - 0.2099 \times 2.80(303 - 293) = 157.4 \text{ w} \quad (22)$$

The PTC thermal efficiency (η_{th}) is calculated by equation 23 as follows:

$$\eta_{th} = \frac{Q_u}{G_B \times A_a} \quad (23)$$

$$\eta_{th} = \frac{157.4}{700 \times 0.9328} \times 100\% = 24.1 \quad (24)$$

If the heat transfer fluid temperature (T_i) and receiver temperature (T_r) are equal, the exit temperature (T_o) of the heat transfer fluid is given by equation 25.

$$Q_u = \dot{m} c_p [T_o - T_i] \quad (25)$$

where, \dot{m} : is the mass flow rate of the heat transfer fluid. it is value is 0.01 kg/s . c_p : is the specific heat transfer fluid. it is valued is 1.67 J/kg.k . Therefore, the exit temperature of the heat transfer fluid is

$$T_o = T_i + \frac{Q_u}{\dot{m} c_p} = 303 + \frac{157.4}{0.01 \times 1670} = 312.4 \text{ K} = 39.4 \text{ C} \quad (26)$$

Table 2. Temperature and Pressure of PTC heat transfer fluid on Jan 11

Time (hh:mm:ss)	Temperature (C)	Pressure (kpa)
1:09:20 pm	18	80.2
1:10:08 pm	19	82.6
1:11:10 pm	21	87.7
1:12:15 pm	33	92.7
1:13:10 pm	34	86.1

Table 3. Temperature and pressure of PTC heat transfer fluid on Jan 12

Time (hh:mm:ss)	Temperature (C)	Pressure (kpa)
10:33:26 AM	31	80.2
10:34:25 AM	31	80.3
10:35:24 AM	31	80.4
10:36:23 AM	31	80.6
10:38:29 AM	32	81.3
10:39:26 AM	34	82.1
10:40:00 AM	40	99.7
10:41:02 AM	41	82.6
10:42:01 AM	40	82.7
10:43:00 AM	39	82.8
10:44:02 AM	38	82.9
10:45:09 AM	37	83.1

10:46:00 AM	37	83.1
10:47:03 AM	38	83.1
10:48:05 AM	37	83.2
10:49:00 AM	37	83.2
10:50:03 AM	36	83.2
10:51:02 AM	36	82.8
10:52:04 AM	36	82.7
10:53:03 AM	35	82.7
10:54:02 AM	35	82.8
10:55:04 AM	35	82.8
10:56:00 AM	35	82.9
10:57:02 AM	34	82.9
10:58:01 AM	34	83.2
10:59:00 AM	34	83.2
11:00:02 AM	34	83.2
11:01:01 AM	34	83.2
11:02:04 AM	34	83.2
11:03:03 AM	34	83.2
11:04:02 AM	34	83.2
11:05:04 AM	34	83.2
11:06:03 AM	33	83.1
11:07:02 AM	33	83.1
11:08:01 AM	33	83.1
11:08:56 AM	33	83.1
11:09:33 AM	32	82.2
11:10:06 AM	32	82.2
11:11:01 AM	32	82.2
11:12:03 AM	32	82.2
11:27:05 AM	33	83.1
11:28:04 AM	33	83.2
11:29:03 AM	32	83.2
11:32:29 AM	32	83.6
11:33:45 AM	38	104.9
11:34:31 AM	41	84.6
11:35:00 AM	46	86.6
11:36:03 AM	43	85.6
11:37:05 AM	40	92.8

Sample of the data collected by Arduino from temperature and pressure sensors for the designed PTC for January 11 and January 12 are shown in Tables II and III. Temperature and pressure were recorded every second, but the data reported is for each minute. The average beam solar radiation in Jan 11 and Jan is 700 w/m². the temperatures versus time are plotted and depicted in Figures 7 and 8.

As shown in Figures 7 and 8 the temperatures increased almost linearly with time. For five minutes operation the temperature increased by 16 degrees while for one-hour operation the temperature of the heat transfer fluid temperature was increased by 15 degrees. since the temperature of heat transfer fluid in Table 2

was much lower than in Table 3, it is indicated that lower heat transfer fluid temperature the more apparent the effect of solar heating.

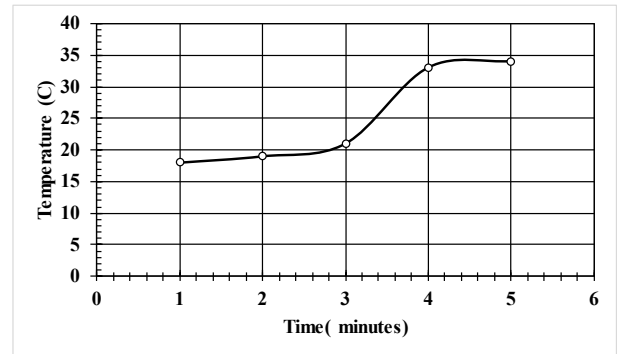


Figure 7. PTC Temperature on Jan 11 During 5 Minutes of Operation

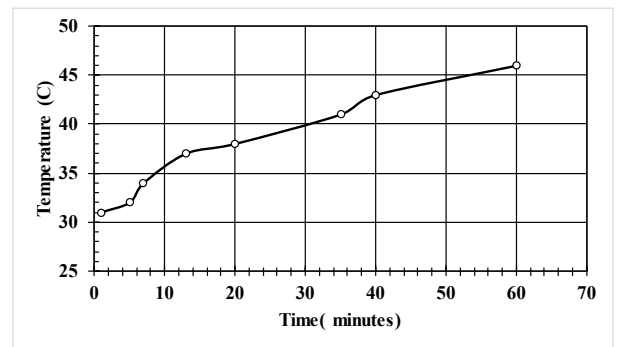


Figure 8. PTC Temperature on Jan 12 During 60 Minutes of Operation

Conclusion

PTC with a 2m length was designed and manufactured for the local market. The collector reflector surface was manufactured from polished stainless steel. The concentration ratio of the PTC was five times higher than the optical efficiency, which was 25 % considering all the imperfections that resulted from the manufacturing process. The heat losses from the PTC, including convection, conduction, and radiation, were calculated, and it was found that the overall heat loss coefficient was w/m^2K while the PTC's overall thermal efficiency (η_{th}) was 24.1%. Measuring the temperature of heat transfer fluid with a thermocouple connected to an Arduino board showed that temperatures

increased almost linearly with time. For five minutes of operation, the temperature increased by 16 °C, while for an hour of operation, the temperature of the heat transfer fluid increased by 15 °C. Since the start temperature of the heat transfer fluid in the first case was higher than that in the second table, it is indicated that the lower the heat transfer fluid temperature, the more apparent the effect of solar heating. Results showed that the reflective surface used in designing the PTC may affect the concentration ratio since it absorbs part of the radiation. It was also difficult to control the parabolic shape because the manufacturing process was not completely accurate, which may result in a deviation between the focal line and the position of the receiver tube.

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Conflict of interests

No conflict of interest.

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