



HEAT TRANSFER ANALYSIS OF A PARABOLIC TROUGH COLLECTOR PROTOTYPE UNDER BOTSWANA'S TERRESTRIAL CONDITIONS

by

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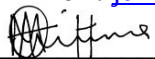
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I, Ndakidzilo Nthoiwa, declare that this dissertation is my own original work and that it has not been presented and will not be presented to any other university for a similar or any other degree award.

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CERTIFICATION

The undersigned certifies that he has read and hereby recommends for acceptance by the Faculty of Science a dissertation titled: **“Heat transfer analysis of a parabolic trough collector prototype under Botswana`s terrestrial conditions”**, in partial fulfillment of the requirements for the degree of Master of Science (Physics) of the Botswana International University of Science and Technology.



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DEDICATION

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LIST OF PUBLICATIONS

Reviewed conference articles (Abstracts and presentations)

- (1) N. Nthoiwa, A. O. Juma, `A survey of the renewable energy sources and their utilization in Botswana`, International conference on clean energy for sustainable growth in developing countries, 16th -18th September 2015, Palapye, Botswana, ISBN: 978-9998-0-475-5**
- (2) N. Nthoiwa, A. O. Juma, `Untapped potential for solar energy technology implementation in Botswana`, 1st Africa Energy Materials conference (AEM 2017), 28th -31st March 2017, Pretoria, South Africa (Abstract).**
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- (4) N. Nthoiwa, C. Ramotoroko, A. O. Juma, ` Analysis of the potential of solar thermal technologies for power generation in Botswana`, Research and innovation symposium, 12th - 14th June 2017, Palapye, Botswana (Abstract submitted)**

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LIST OF ABBREVIATIONS

A_a	Aperture area
A_c	Collector area
A_g	Glass envelope surface area
A_r	Receiver surface area
C	Concentration ratio
C_{max}	Maximum concentration ratio
c_p	Specific heat capacity
D_{ia}	Receiver internal diameter
D_{oa}	Receiver external diameter
ε	Emissivity
ε_g	Emissivity of glass
ε_r	Emissivity of the receiver tube
F_R	Heat removal factor
$\dot{F}$	Collector efficiency factor
h_g	Conduction heat transfer coefficient
h_r	Radiation heat transfer coefficient
$h_{r,g-a}$	Radiation heat transfer coefficient between glass and the ambient
$h_{r,r-g}$	Radiation heat transfer coefficient between receiver and glass envelope
h_w	Convection heat transfer coefficient
I_o	Extraterrestrial solar radiation
I_{sc}	Solar constant
K_a	Prandtl number for air
$K(\theta_i)$	Incidence angle modifier
m	Mass flow rate
Nu_a	Nusselt number for air
Re_a	Reynolds number for air
U_L	Overall heat loss coefficient
U_o	Overall heat loss coefficient
θ_z	Solar zenith angle
ϕ	Solar latitude angle
G	Global horizontal irradiance
G_d	Diffuse horizontal irradiance
G_b	Direct normal irradiance

GHI	Global horizontal irradiance
DHI	Diffuse horizontal irradiance
DNI	Direct normal irradiance
n	Day of the year
ω	Hour angle
ST	Solar time
δ	Solar declination angle
α	Solar altitude angle
α_r	Absorbance of the receiver
σ	Stefan-Boltzmann constant
τ_g	Transmittance of glass
ρ_a	Reflectance of the collector
γ	Intercept factor
γ_z	Solar azimuth angle
θ	Angle of incidence
λ	Wavelength
r	Collector radius
r_r	Maximum collector radius
f	Focal length
D	Receiver diameter
w_a	Aperture width
L	Length of the collector
ϕ_r	Rim angle
θ_m	Half solar acceptance angle
H_p	Latus rectum
T_a	Temperature of the ambient
T_c	Temperature of the cold reservoir
T_{fi}	Inlet temperature of the working fluid
T_{fo}	Outlet temperature of the working fluid
T_g	Glass envelope temperature
T_h	Temperature of the hot reservoir
T_r	Receiver temperature
Q_h	Heat from a high temperature reservoir
Q_u	Useful heat energy
v	Wind velocity
W	Workdone
X_{END}	End losses
ΔT	Change in the working fluid temperature
η_c	Carnot efficiency
η_o	Optical efficiency

η_{th} Thermal efficiency

ABSTRACT

Botswana has about 84 % of its land area as part of the semi-arid Kalahari desert and it experiences over 320 clear sky days per annum, with a solar insolation of 21 MJ/m^2 per day on a horizontal flat surface. This favors energy generation using concentrated solar thermal technologies, especially parabolic trough collector (PTC). PTC is the most mature technology and contributes up to about 90 % of installed concentrated solar thermal power in the world. A PTC prototype was fabricated and tested for the first time at the Botswana International University of Science and Technology (BIUST) campus in Palapye. A stainless steel sheet lined with an adhesive Mylar film of high reflectivity (≥ 0.94) was used as the collector. Two receivers, a coated copper tube of 0.015 m external diameter and an evacuated commercial receiver tube were used. The highest outlet temperature of 76.0°C at a flow rate of 0.0026 kg/s was achieved for the coated copper tube, while 91.9°C was recorded for the commercial receiver at a mass flow rate of 0.0020 kg/s . Thermal and Optical performances of the coated and the commercial receiver tubes were analyzed. Results showed better performance for the commercial receiver with a maximum thermal efficiency of 24.2 % in contrast to 22.5 % for the coated copper tube receiver. The effect of the wind speed, mass flow rate of the water and the irradiation on the Carnot and thermal efficiencies of the system were studied for the two receivers. When the average wind speed varied from 3.0 m/s to 4.0 m/s during experimental days, the Carnot and thermal efficiencies of the coated receiver decreased from 53.6 % to 38.5 % and 22.5 % to 14.1 %, respectively. At constant wind speed, a decrease in the flow rate from 0.0036 kg/s to 0.0012 kg/s resulted in an increase in the Carnot efficiency from 38.5 % to 54.1 % due to an increase in the outlet temperature, but the thermal efficiency dropped by almost half from 14.1 % to 8.2 %. For the commercial receiver, the effect of wind speed was eliminated because of the insulation around the tube. The thermal efficiency was observed to be 11.0 % and 24.2% for mass flow rates of 0.0012 kg/s and 0.0036 kg/s , respectively. These increase in the mass flow rate resulted in a drop of the Carnot efficiency from 56.4 % to 46.2 % as a result of a drop in the outlet temperature. An increase in the flowrate raises the volume of water that needs to be heated by the same amount of irradiation; which results in a lower outlet temperature. The average outlet temperatures were generally between 69°C and 81°C . This temperature range is suitable for industrial process heat applications such as water desalination, water heating, cooling and refrigeration.

1 Introduction

1.1 Motivation

Research has estimated an average population growth rate of 3.86 billion from 7.35 billion in 2015 to 11.21 billion in the year 2100 [1]. This shows an average annual world population growth rate of 45.46 million. This increase in the world population will increase the world energy demand and the annual global energy-related emissions of carbon-dioxide (CO_2) is therefore expected to increase from the 650 million tonnes since 2000 to about 36 gigatonnes by 2040 [2]. The effects of the conventional energy sources on the environment have raised worldwide alarm due to the resulting global warming. A handful of disasters such as the rising of the sea level resulting in displacement of human settlements could be imminent from this. Many countries worldwide have embarked on drafting policies and goals to curb global warming issues, and as such driving the whole world to switch towards alternative renewable energy sources such as solar energy, hydro-power, wind energy, bio-fuels and geothermal power.

In 2014, the renewable energy generation accounted for 28 % of the total gross electricity generation in the EU, equivalent to about 400 GW [3]. This was an increase of 191 % from the year 1990. Currently there is an increase in interest on sustainable energy resources worldwide with more emphasis on USA, China and the Northern part of Africa [4]. The US, which is the second largest emitter of CO_2 in the world at 5.6 gigatonnes is also at a milestone. The US witnessed an increase from 11 % renewables in the total energy mix for the year 2010 to 14 % in 2013 [5]. In 2014, 49.6 % of the new power capacity was accounted for by renewables. An additional 12 400 MW of new power generating capacity was added, which is 64.8 % of the total installed power in 2015. By 2016 19.2 % of the nation's energy mix was generated from renewables [6]. Some states in USA such as Hawaii [7], California [8] and many more, have introduced an energy bill to achieve 100 % renewables by 2045. Of the 24.5 GW of Europe's new capacity installed in 2016, 21.1 GW which represents 86 % came from renewables [9]. Scotland has set a 50 % target of renewable energy by 2030. It set another target to reduce greenhouse gas emissions by 66 % by 2032. This is in line with their draft energy strategy which has a

vision for transition from oil and gas dependency to a low carbon economy by 2050 [10]. These countries are mainly focused on solar photovoltaics (PVs) and concentrated solar power (CSP) considering the abundance of the solar resources and its cleanliness as opposed to fossil fuels. Combustion of fossil fuels empties billions of tons of heat trapping gases such as carbon dioxide into the atmosphere resulting in pollution and global warming.

The Mojave desert in USA occupying over 64 750 km² and located within the south eastern part of California, is a host to more than 1070 MW of concentrated solar thermal power plants including the world's biggest 392 MW Ivanpah solar facility, and over 944 MW of solar PVs. The nine units of the Solar energy generating systems (SEGs) which forms what remains to be the largest parabolic trough collector (PTC) plant with a combined output of 354 MW are also located at the Mojave desert [11].

With the 9.4×10^6 square kilometres Sahara desert [12], the North African countries including Morocco, Algeria, Egypt and Tunisia are at an advantage of tapping into the solar resource. Morocco has a total installed capacity of 185 MWe from the Ain Beni Mathar project which integrates 20 MW from PTC, the 160 MW from the Noor I and the 5MW combined from demonstration projects and a pilot project [13]. After commissioning of the Noor II and III by 2017, Morocco will be host to one of the largest solar thermal facilities in the world with a combined output of 510 MW [13]. Egypt and Algeria have installed 20 MW of solar thermal power each, and the parabolic trough collector projects were commissioned in June and July 2011, respectively.

In the southern region of Africa, South Africa is at an advanced stage of investing in solar thermal power. 205 MW has been commissioned with 400 MW expected to be added on the grid before the end of 2018. Of the seven solar thermal power stations in South Africa, only two employs the solar tower technology while the remainder uses the parabolic trough collectors [14]. Other countries in the Southern African Development Community region (SADC) including Namibia, Zambia and Zimbabwe are yet to develop solar thermal plants. Botswana on the other hand has no known CSP plant though a bankable feasibility study for a 200 MW plant conducted in 2013 identified Jwaneng and Letlhakane as the most favourite sites. An expression of interest (EOI) was floated to independent power producers (IPPs) in 2015 for the construction and commissioning of a

50 MW solar plant in Jwaneng and the North Western part of Botswana. The outcome of the EOI is not known to date.

Parabolic trough collector system is one of the most mature and widely adopted solar thermal technologies in the world [15]. It uses the principle of concentrating solar irradiation from a bigger surface area to a smaller area. It employs the use of parabolic shaped reflectors which focuses solar irradiation onto a central receiver tube carrying a heat transfer fluid (HTF) in the form of synthetic oil, molten salts or water to produce steam directly or indirectly in order to propel turbines to produce electricity. This technology is not limited to power generation since it can be used for industrial applications such as hot water generation [16] and steam production [17] for sterilization of equipment in hospitals, food processing, refrigeration, space heating and in the textile industry.

In this dissertation, a parabolic trough collector prototype was designed, fabricated and mounted within the Botswana International University of Science and Technology campus. Several experiments were carried out to measure the performance of the prototype. This prototype was displayed at shows in Francistown and Gaborone to raise awareness on the technology and its potential in Botswana. It also attracted public interest and as such was used for demonstration of the concentrated solar power technology both to students and the general community. It was also acknowledged in the BIUST 2015/2016 annual report as an innovative project.

This dissertation consists of five chapters and they are briefly summarised as follows;

Chapter one introduces the potential of solar energy and then gives a background of concentrated solar power projects in the world, Africa and in Botswana.

Chapter two discusses solar thermal technologies including the non-concentrating collectors and the concentrating collectors. It further describes research conducted by different groups on the thermal and optical performance of parabolic trough collectors. The theoretical background is presented covering concepts of the solar radiation, spectrum and solar constant as well as details of the geometry of the parabolic trough collector, optical efficiency and loss calculations, thermal efficiency and loss calculations.

In chapter three, the description of the experimental set-up for the two receivers and all the parameters used is presented. It also describes the fabrication of the parabolic trough collector prototype and the experimental procedure for data collection.

Chapter four presents the measured data obtained in this study and the calculated optical and thermal efficiencies of the coated copper tube receiver and the evacuated commercial receiver tube. A discussion of the results and how they relate with literature is also presented.

Chapter five draws out a summary of the thesis and main results and an outlook on how to improve the prototype for higher output and performance efficiencies.

1.2 Statement of the problem

Botswana is still dependent on other neighbouring countries such as South Africa, Mozambique and Namibia to meet the average peak power demand of 610 MW [18]. About 99 % of the locally produced power is from coal at Morupule power station and peak diesel generators at the Orapa and Matsiengabedi emergency generation plants. The remaining 1 % is constituted by renewable energy sources, mainly small scale solar PV systems mainly from the tourism sector, solar water heating systems and biomass plants.

Fossil fuels are the leading emitters of greenhouse gases (GHGs) which accumulate in the atmosphere and promote global warming. Particulate matter, nitrogen oxides, carbon monoxide and sulphur oxides are the main pollutants which are given out during the burning of fossil fuels and they have a huge impact on the environment as well as on human health. Despite the importation of power, Botswana continues to suffer power outages posing an adverse impact on the industrial and business sector as well as the economy.

Botswana has abundant solar irradiation with over 3200 hrs of sunlight annually at an insolation of 21 MJ/m² per day, one of the highest in the world [19] and cattle population of 2.2 million heads [20], yet it ranked poorly as position forty-eight out of the fifty-five climatescope countries in 2014 [21]. Lack of technical knowhow and expertise in renewable energy has remained a hindrance to the progress in investment and power generation from renewable sources such as solar.

1.3 Objectives

General objectives

To construct a PTC and receiver prototype and investigate its heat transfer and performance for process heat applications in Botswana. .

Specific objectives

1. Design and build a PTC and receiver prototype.
2. Test the system under Botswana solar irradiation conditions.
3. Study the performance of the solar PTC and receiver system without a solar tracker (the study period was not enough to build a tracker) .
4. Compare performance of a coated receiver tube and the commercial receiver.

1.4 Significance of the study

Considering that Botswana is amongst the countries with the highest solar insolation in the world with an average direct normal irradiation (DNI) of over 3000 KWh/m² in a year, advanced initiatives should be made to strive for solar power investment. There is need to train experts in solar technology to spearhead the process of harnessing this abundant and clean resource for different industrial applications and power production. The initiative taken by the government to open a new University of Science and Technology to train and increase the number of expertise has opened a pathway to promote research and investment in renewable energy. The power output from solar thermal CSP in the neighbouring republic of South Africa (RSA) is estimated to reach 604.5 MW by 2018 from the three operational CSP plants and other additional four which are still under construction [23]. Botswana can benefit from the development in South Africa by conducting a benchmarking exercise because five out of the seven CSP plants in RSA have adopted the use of PTC technology. The draft energy policy that was presented in Botswana parliament in November 2015 [24] will help in facilitating the implementation of solar thermal CSP projects among others to meet the energy needs and increase the rankings of the country in clean energy investments. This work will contribute to the development of new expertise and knowhow in solar thermal technology and serve as a case study and proof of concept that the technology is feasible and deserves implementation. PTC is a proven solar technology that can be implemented to solve

Botswana's perennial energy challenges. This project will also serve to demonstrate the viability of solar thermal technologies for different industrial purposes and power generation in this country.

2 Literature Review

The idea of harnessing solar energy dates back to around 212 BC when the ancient Greek Mathematician, Physicist, Engineer, Inventor and Astronomer Archimedes came up with a method that employed the use of concave metallic mirrors to burn the Roman fleet [25]. This idea matured over the eighteenth and nineteenth century through the construction of furnaces to melt iron, copper and other metals, to construct solar power steam engines [25]. In 1912, the world largest solar water pumping plant was constructed in Meadi, Egypt using parabolic cylinders [26]. During the past 60 years several technologies have been developed to convert solar radiation into useful thermal energy for various applications.

2.1 Solar thermal technologies

This is a technology that utilises the radiant heat from the sun to heat up a transfer fluid such as water, molten salts or thermal oils for a wide number of applications. Solar thermal technologies can be categorised into two types: non-concentrating and concentrating solar power collectors. Non-concentrating collectors include the evacuated tube and the flat plate collectors.

2.1.1 Non concentrating collectors

2.1.1.1 Evacuated tube collectors

Figure 2.1 shows the schematic diagram of an evacuated tube collector. These collectors contain a copper tube coated with a specialised solar paint to increase their absorptivity. The copper tube is accommodated inside a vacuum sealed tube and they are normally connected parallel to each other. Each sealed copper tube protrudes into the heat collection pipe which carries the water that is being heated. There is a small volume of fluid such as methanol in each heat pipe, and during the absorption of the solar radiation it transforms into hot vapour and rises up to the heat exchanger where the heat gets transferred to the flowing water. It then cools down, condenses and flows back into the heat pipe. The process keeps on repeating provided there is radiation from the sun. These types of

collectors make use of both the diffuse horizontal irradiation (DHI) and direct normal irradiation (DNI) with optimum temperatures of 50 °C up to 100 °C possible [25]. These collectors are used for space heating, powering absorption chillers for solar air conditioning systems, domestic and commercial water heating.

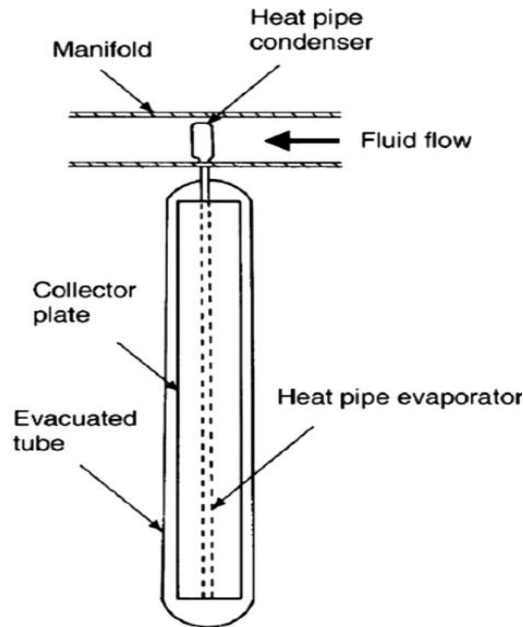


Figure 2.1: Schematic diagram of an evacuated tube collector [27]

2.1.1.2 Flat plate collectors

A Flat plate solar collector is usually made of a darkened absorber plate which houses absorber tubes. The main components of a flat plate collector are shown in Figure 2.2. At the top of the plate there is a glass cover, while below there is an insulation to prevent heat loss by conduction. The transparent glass cover is meant to reduce heat loss by convection and to trap the long wavelength heat from the absorber plate hence preventing heat loss by radiation. These collectors are normally fixed and do not track the sun, and they utilise both the DHI and the DNI. Their temperature range is quiet low, at about 30 °C to 80 °C [25]. Flat plate collectors are used in solar stills, solar ponds, thermal desalination, domestic water heating and space heating.

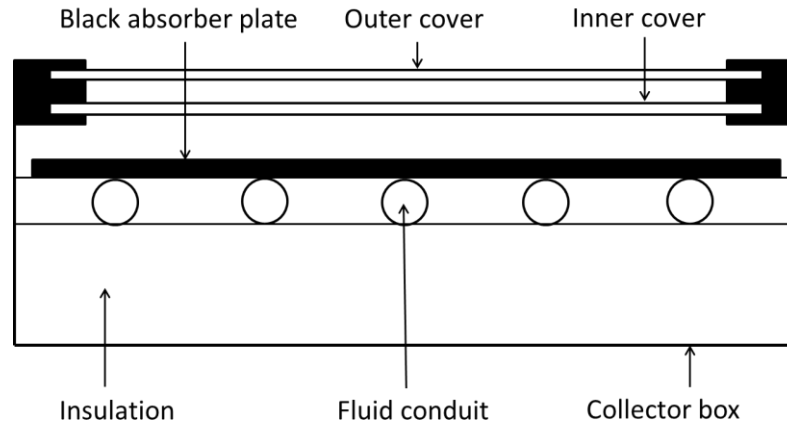


Figure 2.2: Cross-sectional view of a flat plate collector

2.1.2 Concentrating collectors

These collectors are categorised into two namely, point focusing and line focusing collectors. The point focusing collectors are those that focus the irradiation to one central point. They include the solar tower (Heliostat field collector) and the parabolic dish collector. Line focusing collectors concentrate the irradiation on a line that resembles the principal focus of the collector, and they include the Linear Fresnel collector and the parabolic trough collector. A tube is placed along this line and a transfer fluid flowing through it absorbs heat focused by the collector or mirror. The heated HTF is then carried directly to a power generation system or it is channelled to a thermal energy storage tank from which it can be drawn and used when needed for industrial thermal processes or power generation.

2.1.2.1 Solar Tower

Heliostat field collector (HFC) also known as a solar tower is the most recent technology. A solar tower of about 75 - 150 m in height accommodates a receiver at the top. An array of flat or slightly concave mirrors (heliostats) is distributed around the solar tower to focus the direct normal irradiation (DNI) to the receiver as shown in Figure 2.3(a). Each mirror follows the sun's movement using a two-axis tracking system. The heated fluid then drives a turbine to produce electricity at a conversion efficiency of about 17 % [28]. Fluid outlet temperatures can be as high as 2000 °C as shown in Table 2.1. All HFC plants are very large and produce a minimum of 10 MW power output. The first HFC solar plant (Solar

one) was built in 1981 and the second one (Solar two) in 1995 situated in the Mojave Desert, California [29]. Other solar plants include the 11 MW Planta Solar (PS) 10 and the 20 MW PS20 in Spain, and the 5 MW plant in Sierra SunTower, California.

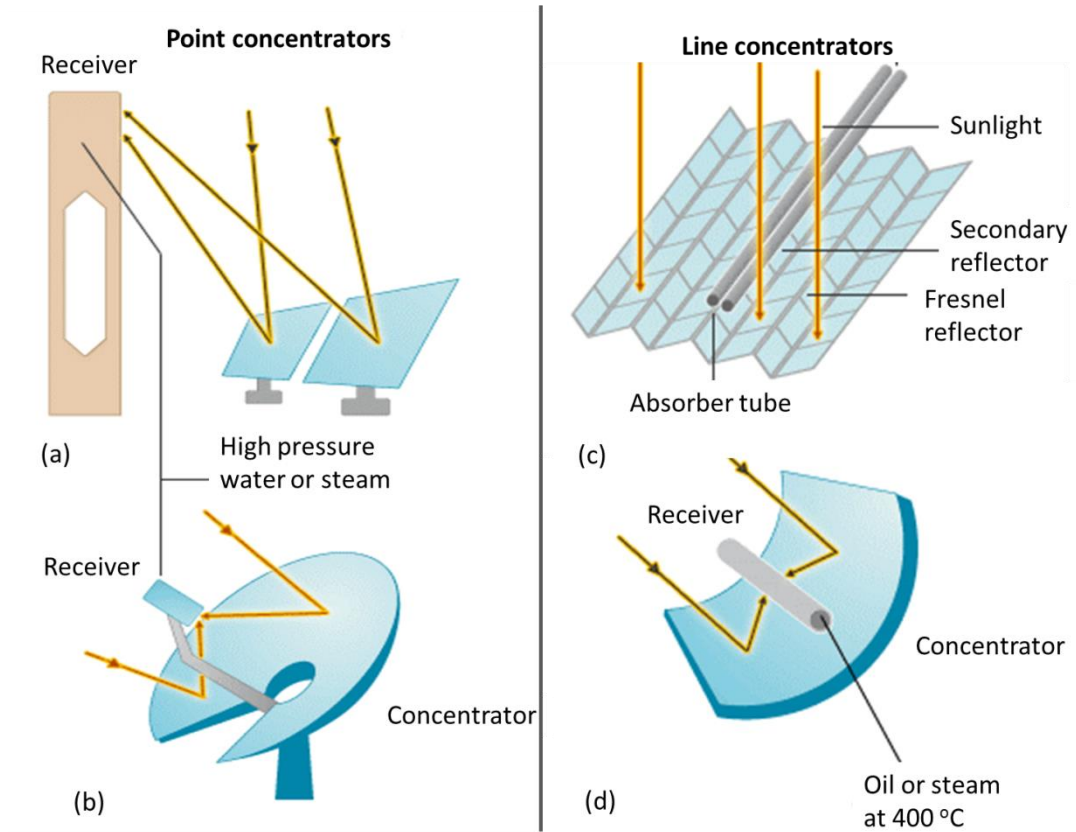


Figure 2.3: (a) Solar tower (Heliostat field collector), (b) Solar dish/Dish Stirling, (c) Linear Fresnel reflector and (d) Parabolic trough collector

2.1.2.2 Solar Dish

A parabolic dish collector (PDC) resembles a household satellite dish receiver with the receiver carrying an HTF mounted at its focal point. The dish focuses a beam of reflected sunlight rays to its focal point where a receiver is mounted as shown on Figure 2.3 (b). The dish is mounted on a two axis tracking system which rotates in the direction of the sun from the East to the West and also in the azimuth direction. Its recorded outlet temperatures can be as high as 1500 °C as shown in Table 2.1. The solar dish is the most efficient of the solar-thermal technologies with conversion efficiency of 24 % [28] and an output of 3.2 kW per dish. Maricopa Solar Project in Arizona (USA) which started operation in January 2010 is the only PDC plant that is operational in the world [30].

2.1.2.3 Linear Fresnel

Linear Fresnel Reflectors (LFR) uses flat or curved mirrors to focus sunlight onto a linear receiver facing downwards at the top of a tall tower of height 10 - 15 m as shown in Figure 2.3 (c). Their highest outlet temperatures can reach a maximum of 300 °C as shown in Table 2.1. Their sunlight to electricity conversion efficiency is normally about 13 % [31]. The first LFR plant was constructed in Germany in March 2009, with a capacity of 1.4 MW, another 30 MW plant in Spain [30] while a 5 MW plant is operational in California, USA.

2.1.2.4 Parabolic trough collector (PTC)

A PTC is made of a sheet of reflective metal such as an aluminium sheet bent into a parabolic shape to form a collector, which focuses the sunlight onto an absorber tube that is mounted at the focal line of the parabola. A parabolic trough collector is shown in Figure 2.3 (d). The receiver, which is a metal tube enclosed within a glass tube features high solar absorption, low thermal emittance, and high transmissivity to increase the system's overall efficiency [29]. Collectors are connected parallel to each other to form a solar field. PTCs normally use a single-axis system to track the sun from east to west even though ideas are in place to track the sun in a dual-axis. Its conversion efficiency (sunlight to electricity) is up to 20 % [28] even though recent studies show efficiencies of over 73% [32]. This is a mature technology accounting to 90 % of installed CSP capacity in the world. The biggest group of nine solar energy generating systems (SEGS) in the Mojave desert gives a total of 354 MWp (Megawatt power) [33]. Other PTC plants include the 50 MW Andasol-1 in Spain, 64 MW Acciona Solar's Nevada Solar One, 64 MW from the Nevada Solar One power plant, 280 MW Solana Power plant (largest PTC plant in the world), United States of America [34], 160 MW Noor I and 472 MW Ain Beni-Mathar Integrated solar plant in Morocco [35], 100 MW Kaxu and 100 MW Bokpoort in South Africa. These power plants are amongst the 20 currently in operation and there are about 27 more still under construction [36].

One of the parameters that determine how efficient a solar collector is the concentration ratio. Concentration ratio (CR) is defined as the ratio of the collecting surface to the receiving surface. Concentrating ratios for point focusing systems can be as high as 1500, and they track the sun along two axis. Line focusing collectors achieve much lower temperatures than the point focusing collectors with CRs of less than 100. It is evident that

an increase in the CR increases the achievable temperature. Table 2.1 gives a summary of the differences between the four solar thermal technologies used for solar concentrators.

Table 2.1: Summary of the differences between the four solar thermal technologies

Collector Type	Operating Temperature Range (°C)	Concentration Ratio (C)	Relative Cost	Thermodynamic Efficiency
PTC	50 – 400	70 – 80	Low	Low
Linear Fresnel	50 – 300	25 – 100	Very low	Low
Solar Tower	300 – 2000	300 – 1000	High	High
Dish Stirling	150 – 1500	1000 – 3000	Very high	High

2.2 Performance of PTC prototypes

Odeh et al., [37] designed and developed an educational solar tracking parabolic trough collector system of aperture 1.8 m, rim angle 74 °, focal line 2.0 m and focal length 0.6 m. The PTC system was used for demonstrative purposes at the University of South Australia. The receiver tube was made of copper with a diameter of 2.0 cm fixed to a long copper sheet. Both the receiver and the copper sheet were coated with a special solar paint and inserted into a rectangular casing insulated at the back and covered with a single glaze at the front. The efficiency obtained for experiments conducted around solar noon was found to be 60 % at a flow rate of 0.0233 kg/s with an open low water flow (water does not flow back into the system). The system tracked the sun while oriented on the North-South axis.

Balghouthi et al., [38] carried out an evaluation of the optical and thermal performance on a medium temperature parabolic trough collector used in a cooling installation located at the centre of Researches and Energy Technology (CRTE) Bordj-Cedria, Tunisia. The maximum outlet temperature achieved was 165 °C when the inlet temperature was 150 °C at about 38 °C ambient. A camera-target method and the method of total errors were used to analyse the optical performance of the system to yield efficiencies of 51.4 % and 48.0 %, respectively. The thermal efficiency was determined under steady state conditions as proposed by ASHRAE standard 93 (1986). The steady state conditions yielded an efficiency value of 58.0 % when the fluid inlet and the ambient temperatures are equal. The thermal efficiency was observed to decrease with an increase in the temperature difference between the fluid inlet and the ambient. Further evaluation revealed significant

optical losses due to collector surface deformations and slope deviations. Thermal losses were attributed to heat losses by conduction, convection and radiation through the collector glass cover, receiver bellows, flexible hose pipe and the support structure.

Chafie et al., [39] also studied the performance of a parabolic trough collector at Borj-cedria in Tunisia. The PTC had a length of 4.0 m, parabola width of 2.7 m and a focal length of 0.835 m utilizing an evacuated glass-steel coated tube receiver. The concentration ratio was 11.77 and the rim angle 76.3° . The outlet temperature of the Transcal N thermal oil used as the heat transfer fluid reached a maximum of 99°C with the system manually rotated to track the sun uniaxial from east to west. When the experimental tests were conducted according to ASHRAE 93 (1986) standards, a maximum thermal efficiency of 55.1 % was achieved with a mass flow rate of 0.2 kg/s at a speed of 2.6 m/s. It was deduced from the analysis that higher thermal efficiencies occurred at around noon due to higher direct solar radiation and higher useful energy gain. Further analysis yielded average thermal efficiencies of 41.09 % and 28.91 % for sunny and cloudy days respectively.

Brooks et al., [40] evaluated the performance of a smaller-scale parabolic solar collector in South Africa for use in solar thermal research programme. The collector was designed with a length of 5.0 m, having an aperture width of 1.5 m and a rim angle of 82.2° . An unshielded and an evacuated glass shielded receiver tubes were used for the purpose of this study. Water was used as the transfer fluid and the performance analysed according to the ASHRAE 93 (1986) standard. Maximum thermal efficiencies of 52.5 % and 53.8 % were obtained for the unshielded and the shielded receiver tubes respectively. The overall heat loss coefficient was observed to be reduced by about 50 % for the shielded evacuated glass tube receiver.

Padilla et al., [41] carried out an investigation to determine the effect of various parameters on the collector efficiency and exergy efficiency. The exergy analysis was done based on work published by other authors. The main parameters analysed were fluid inlet temperature, mass flow rate of the transfer fluid, wind speed, pressure or vacuum of annulus and solar irradiance. Results showed that fluid inlet temperature, solar irradiance and the vacuum in the annulus have a huge impact on the thermal and collector exergy efficiency. The transfer fluid mass flow rate and the wind speed were observed to have little to no impact on the thermal and exergy performance of the parabolic trough collector

system. It was also deduced that when the pressure inside the annulus is above 1 Torr, heat transfer by convection inside the annulus increases, hence an increase in the exergy loss.

Raj et al., [42] analysed numerically the performance of an absorber tube with and without insertion using the commercial CFD code ANSYS CFX 12.0. The working fluid in all these studies was water with different mass flow rates of 0.00917 kg/s, 0.01750 kg/s and 0.02361 kg/s. The parameters measured during the experiment were the inlet, outlet and ambient temperatures and the solar irradiance. It was observed that the absorber tube with insertions yielded fluid outlet temperatures 0.5 °C higher than the case without insertions. It was also noted that the thermal stresses for the tube with insertions were much lower, while the pressure drop was higher than that without insertions.

Natarajan et al., [43] performed a numerical simulation of heat transfer by use of internal flow obstruction in the absorber tube of a parabolic trough collector. A three dimensional numerical analysis was adapted for this study. The analysis was carried out using an inverted triangle and semi-circular inserts which were then compared to the results of a plain absorber tube. The pressure drop and heat transfer were evaluated for a mass flow rate of 0.02361 kg/s using the ANSYS CFX 12.1 software and turbulence was modelled using the SST k- ω model of closure. An observation was made that thermal stresses on absorber tubes with insertions was lower than that of the absorber tube without insertion. The absorber tube with the triangle insertion proved to be the best of the three but with a higher pressure drop of 147 Pa compared to a pressure drop of 48 Pa on the absorber tube without insertion.

Barriga et al., [44] carried out analysis on how solar selective coating to improve the performance of parabolic trough collector systems. Currently, optical values of solar coatings have more than 95 % absorbance and less than 10 % emittance at a temperature of 400 °C. Propositions for a new parabolic trough collector system that would work at a temperature of 600 °C and a lower pressure of 10^{-2} millibars calls for much better solar coatings which can withstand aggressive conditions than what is available. A 4 m long receiver pipe was proposed and experimental analysis on the new solar coating resulted in an absorbance of 95.2 %. This was homogenous throughout the length of the pipe owing to the deposition method employed.

Yaghoubi et al., [45] performed numerical analysis to compare the impact of heat losses on three different receiver tubes; with a vacuum, one with vacuum lost and a broken glass

tube (bare). Numerical results were compared to the experimental data. For each of the three cases, a 4 m long receiver was used with the temperature of the transfer fluid reaching 265 °C. An infrared thermograph (IR) was used to record the temperature around the tubes. The receiver tube with a vacuum reduced the heat losses significantly compared to the broken glass tube and the one without a vacuum. The heat loss of the lost vacuum case was 40 % higher than the one with the vacuum intact. The broken glass tube reduced the thermal performance by 12 - 16 %. It was therefore concluded that glass tube failures should be avoided for any thermal power generation use.

Valan-Arasu & Sornakumar, [46] investigated the performance of a parabolic trough collector system for hot water generation. A copper tube with a glass envelope was used as a receiver at a rim angle of 90 ° with the collector lined SOLARflex foil of reflectance 97.4 %. The simulation was carried out using a MATLAB code and in accordance with the ASHRAE 93 (1986) standard. Smooth variations between the useful heat gain, thermal efficiency and direct normal irradiance was observed and maximum values occurred at noon. This proved that useful heat gain and thermal efficiency are strongly influenced by the irradiance.

Muhlen et al., [47] carried out sensitivity analysis on the effect of key parameters such as mass flow rate, annulus pressure, receiver diameter and the Reynolds number on the performance of parabolic trough collectors. A one dimensional finite element approach was employed in the simulation and the results validated using experimentally acquired data. The results showed linearity in the HTF and absorber temperatures increase and a constant glass envelope temperature. A vacuum maintained at low pressure of 10^{-4} Torr eliminates any possibilities of heat losses from the tube to the envelope. It was also deduced that mirror inefficiencies are the major factors in system's overall efficiencies. When the vacuum was broken the thermal losses from the absorber increased slightly, but the highest losses were observed at the support brackets. As expected, an increase in the HTF temperature has a role in increasing heat losses and decreasing the efficiency of the parabolic trough collector.

Zhang et al., [48] investigated heat losses of a double glazed vacuum U-type solar receiver mounted on a parabolic trough collector for medium-temperature steam generation. Effects of parameters such as wind, vacuum, irradiance and structural characteristics of the receiver on heat losses were evaluated. The thermal efficiency of the receiver was 79.1 %

and 47.2 % in calm and windy days, respectively. That shows that the percentage heat loss was 52.82 % during windy conditions compared with 20.92 % during calm conditions. When the receiver element is considered in the analysis, the thermal efficiencies increased to 79.2 % and 66.3 %, respectively. This proved that heat losses are only increased by forced convection in insulated receiver tubes. Hence it was concluded that for the U-type receiver, characteristics of the structure are very important for the calculations of the thermal efficiency.

2.3 Theoretical background

2.3.1 Earth`s solar energy budget

The sun has a diameter of about 1 391 684 km and is 149 600 000 km away from the earth. It has an effective surface temperature of 5762 K with its central core temperature estimated to be between 8 000 000 K and 40 000 000 K [49]. At this temperatures, it is estimated that the sun produces about 3.8×10^{20} MW of power, which is equal to 63 MW/m² at the sun`s surface. Incoming solar radiation is estimated at about 342 W/m² but only a fraction of this radiation (51%) is absorbed by the surface of the earth. Figure 2.4 shows the distribution of the radiation before and after entering the atmosphere. About 30 % of the radiation is lost due to reflection by clouds and scattering by atmospheric particles while 19 % of the radiation is absorbed by the atmosphere (dust, water vapour and the ozone) [50].

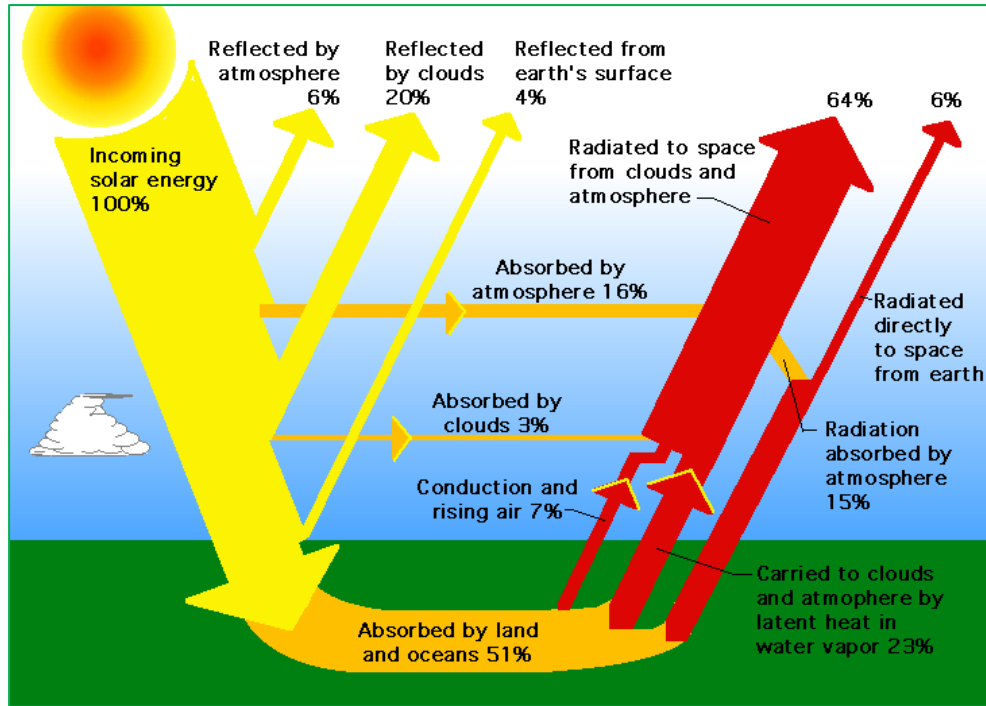


Figure 2.4: Earth's energy budget from the sun [51]

The radiation from the sun can be categorised into extraterrestrial and terrestrial solar radiation. Extraterrestrial radiation refers to the irradiation from the sun outside the earth's atmosphere as shown in Figure 2.5. The value of this radiation at the mean sun-earth distance (1.496×10^{11} m) is equivalent to 1367 W/m^2 and it is known as the solar constant (I_{SC}). Extraterrestrial solar radiation on a plane perpendicular to the sun's rays can be calculated for any given day of the year using the expression [52]

$$I_o = I_{SC} \left[1 + 0.0034 \cos \left(\frac{360n}{365.25} \right) \right] \quad (2.1)$$

where n is the day of the year

Terrestrial solar radiation is the irradiation from the sun within the earth's atmosphere. Only a fraction of the total solar irradiation outside the atmosphere reaches the earth's surface. This is due to reflection by clouds, absorption and scattering by dust and water vapour in the atmosphere. Therefore, the solar radiation that reaches the earth's surface is of two types; direct normal irradiation (DNI) and diffuse horizontal irradiation (DHI).

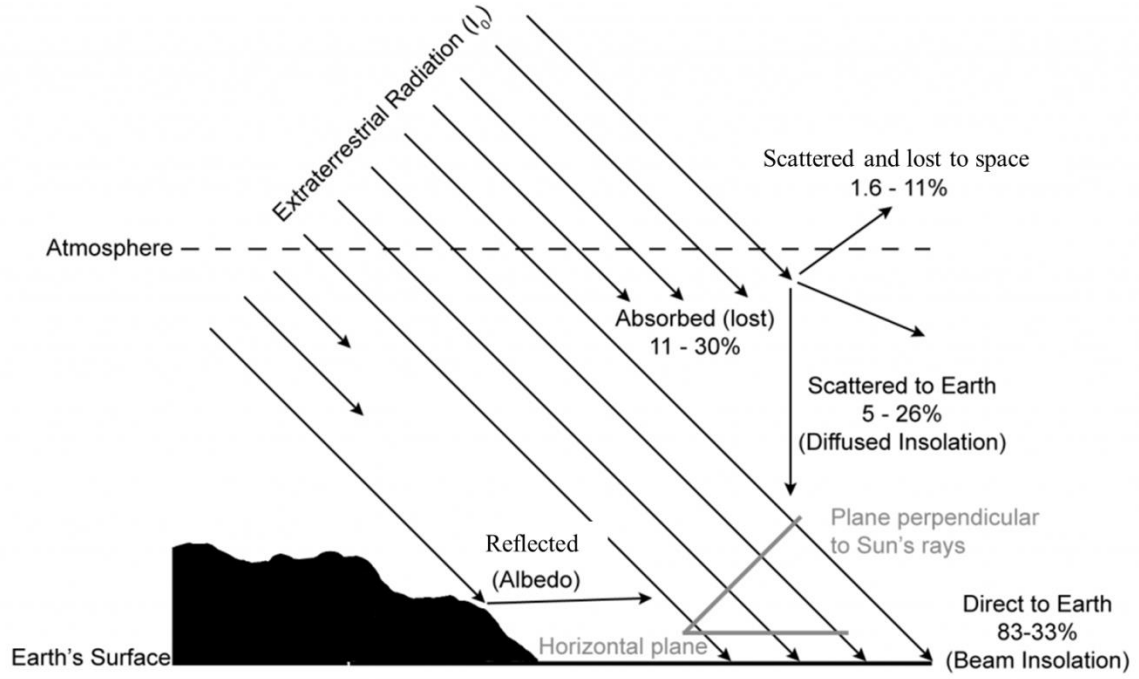


Figure 2.5: Atmospheric attenuation of solar irradiation before reaching the surface of the Earth

DNI refers to the direct irradiance received on a plane normal to the sun over the total solar spectrum [53]. This is the most important component for CSP and can reach a maximum value of 1000 W/m^2 . DHI refers to the solar radiation that is scattered by dust particles, ozone column, clouds and aerosol particles in the atmosphere, and hence it has no direction. The sum of the DNI and the DHI is known as the global horizontal irradiation (GHI) and is given by [54].

$$GHI = DHI + DNI \cos \theta_z \quad (2.2)$$

where θ_z is the solar zenith angle.

The direct normal irradiation component of the radiation is very useful in concentrated solar power systems. This is because concentrators are positioned to face the sun, hence only the direct rays from the sun are utilised in contrast to the diffuse horizontal irradiation which strikes the collector from all directions.

2.3.2 Basic Sun-Earth angles and seasonal changes

Figure 2.6 shows various important angles between the sun and the earth as explained below;

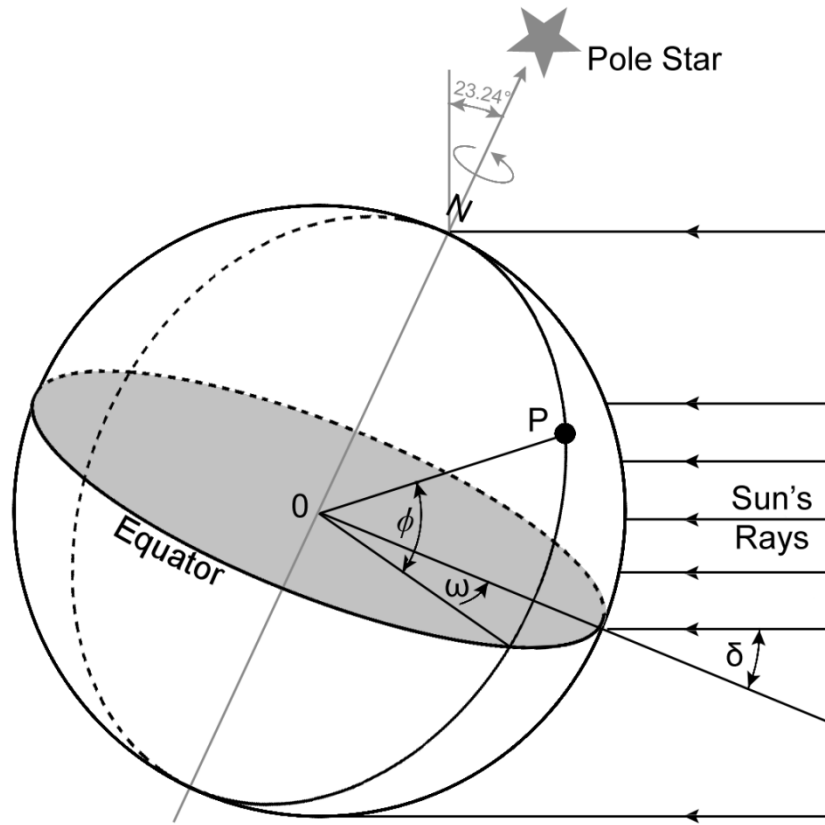


Figure 2.6: Solar angles

The angle between a radial line drawn from a point P to the centre of the earth and its projection on the equator is called the Latitude angle (ϕ).

The hour angle (ω) of point P on the surface of the earth is defined as the angle of its meridian and the meridian that is parallel to the sun's rays. The hour angle is negative at sunrise, decreases to zero at solar noon then increases positively until sunset. The magnitude of the hour angles at sunrise and sunset are the same, only the signs differ [55].

$$\omega = [(ST - 12) * 15] \quad (2.3)$$

where ST is the solar time in hours.

The angle between the earth's equatorial plane and a straight line drawn from the sun into the centre of the earth is called the declination angle (δ). This angle can either be drawn from the south or the north of the equator. The maximum declination angle achieved is $+23.45^\circ$ when the sun is along the tropic of cancer. This occurs on the 21st June and it is called the summer solstice, because it would be summer in the northern hemisphere. A minimum declination angle of -23.45° occurs on the 21st December, when the sun is above the tropic of Capricorn. This is called the winter solstice since it would be winter in

the northern hemisphere. When the sun is directly above the equator, the declination angle is equal to zero. This occurs twice within a year, on the 22nd March and the 23rd September, resulting in a phenomenon called vernal and autumnal equinox, respectively. An illustration of the declination angles together with the solstice and equinox are shown in Figure 2.7.

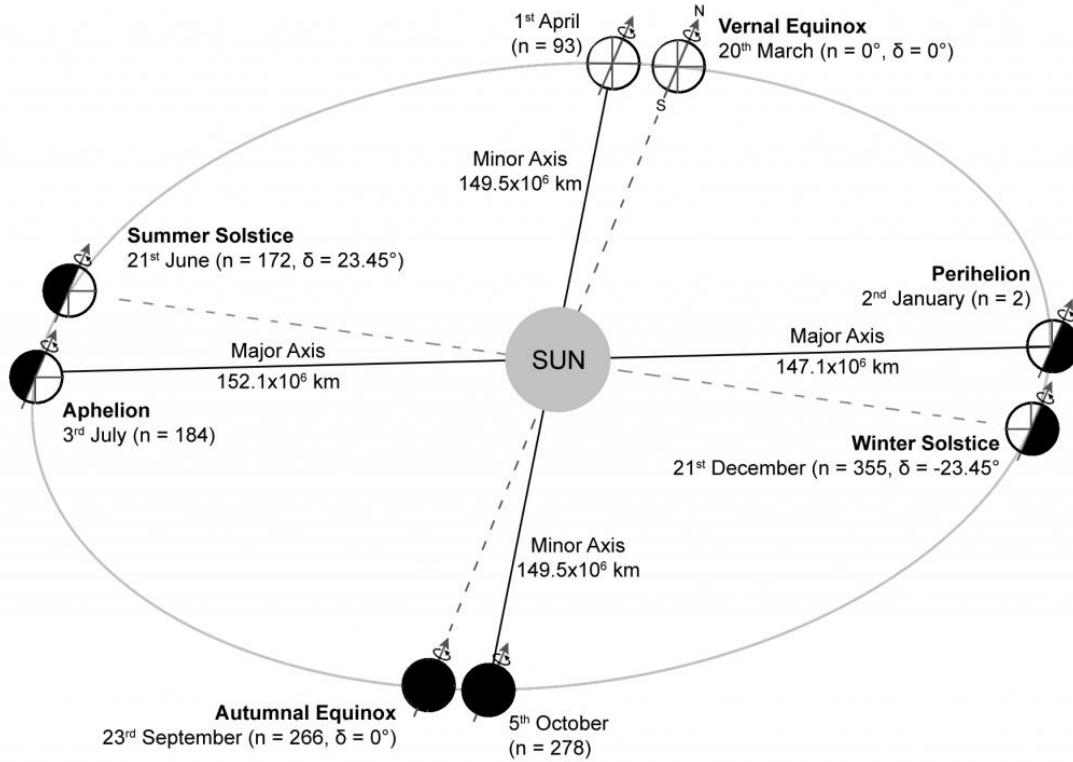


Figure 2.7: Earth`s movement around the sun and seasonal changes

The declination angle is estimated by [56]

$$\delta = 23.45^\circ \sin \left[360 \frac{284+n}{365} \right] \quad (2.4)$$

Where n is the day of the year and always ranges between 1 and 365.

2.3.3 The Solar spectrum

The spectral wavelength range of the extraterrestrial radiation is about 0.1 to 50 μm [60], but this range is narrowed to 0.3 up to 3 μm due to absorption and scattering by water vapour and dust. The ozone layer (O_3) also plays a role in the absorption of the spectra, affecting the ultra violet (UV) radiation of wavelengths below 0.29 μm as well as the visible light. Water vapour absorbs a small portion of the UV radiation within the range of 0.38 to 0.78 μm . A stronger effect of the water vapour is noticeable in the infra-red (IR) region which ranges from 0.78 to 3.00 μm . The solar spectral distribution of about 8.3 % is contributed by UV, 42.3 % by visible light and the majority 49.4 % by IR [61] as shown in Figure 2.9.

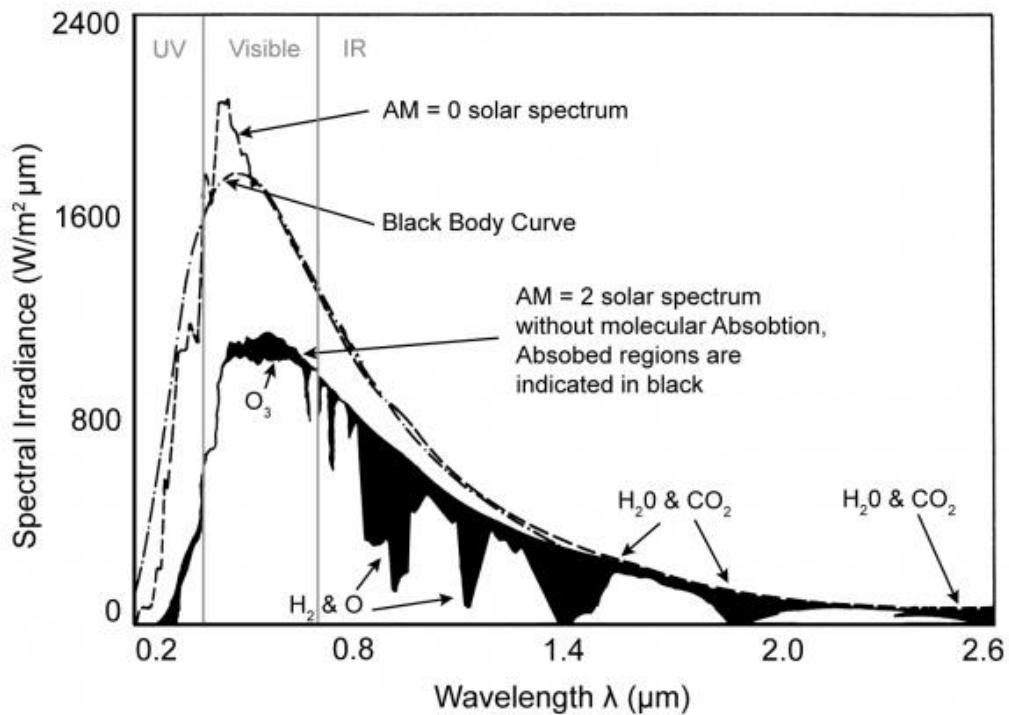


Figure 2.9: Solar spectral distribution [62]

2.4 Performance of the PTC

2.4.1 PTC geometry

A parabolic trough collector consists of three most important components; the reflecting system, a receiver tube and the rotational mechanism. Figure 2.10 shows a schematic diagram for the cross sectional view and a dimensional analysis of the parabolic trough collector, respectively. The reflecting system is a parabolic shaped sheet of metal which is highly reflective, and it is called the collector. This collector is sometimes lined with a self-adhesive Mylar sheet, glass mirrors, silvered-glass or anodized sheets of aluminium which are resistant to degradation under harsh weather conditions, to maintain high reflectivity of the irradiation falling on it. A receiver tube is placed along the focal line of the collector and this tube is sometimes shielded by a glass envelope and evacuated to prevent heat loss from the transfer fluid flowing inside the metal tube. To increase the absorptivity of the tube, it is coated using a specialised solar coat with low emissivity. A rotational mechanism is also designed, either to manually or automatically track the sun.

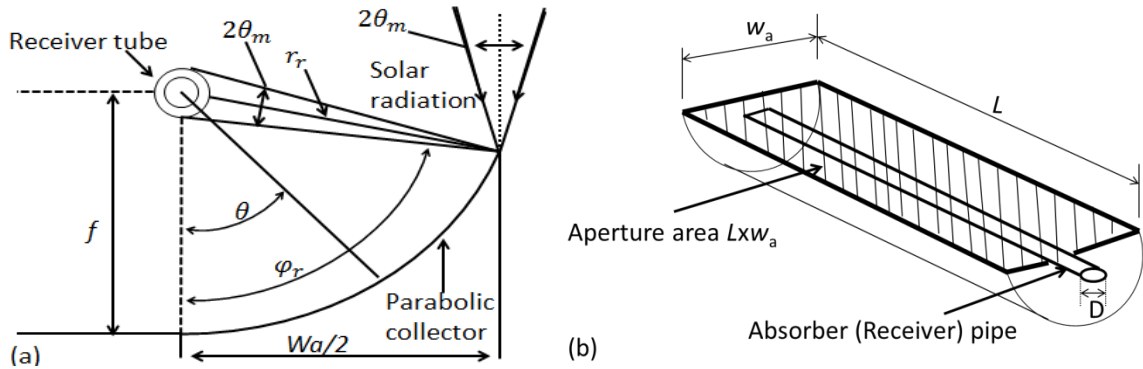


Figure 2.10: (a) Schematic diagram for the cross section of a PTC and (b) a dimensional analysis of a PTC

The parabolic collector is defined in terms of its length (L), aperture diameter (w_a), rim angle (φ_r) diameter of the receiver (D) and its focal length (f). Theoretically, the receiver diameter of the parabolic trough solar system should intercept all the solar irradiation that falls on the collector. That is only possible if all the optical losses are ignored and the system is assumed to be ideal, something that is not practically viable. The receiver diameter can be determined using the equation [63]

$$D = 2r_r \sin \theta_m \quad (2.8)$$

where θ_m is half the solar acceptance angle, and r_r is the collector radius.

The radius of the collector at any point of the collector as shown in Figure 2.10 (a) is given by the equation [63]

$$r = \frac{2f}{1+\cos \theta} \quad (2.9)$$

where θ is the angle between a line drawn from the vertex of the parabola and the reflected ray. The angle θ has a minimum value of zero and a maximum value equal to the rim angle φ_r . The radius r also varies from a minimum equal to the focal length to a maximum value which equal to the collector radius, r_r .

Equation 2.9 can be re-written as

$$r_r = \frac{2f}{1+\cos \varphi_r} \quad (2.10)$$

where φ_r is the rim angle.

The aperture width of the collector is deduced from the relationship between the collector radius and the focal length [64]

$$w_a = 4f \tan\left(\frac{\varphi_r}{2}\right) \quad (2.11)$$

The latus rectum of the parabola H_p , is a line segment that is drawn through the focal point and whose end points lie on the parabola. This line segment is parallel to the directrix and perpendicular to the axis of the parabola. The latus rectum is calculated from [65]

$$H_p = 4f \quad (2.12)$$

and for a parabola with a rim angle of 90° , the latus rectum is equivalent to the aperture width, .

$$H_p = w_a = 4f \tan(45^\circ) \quad (2.13)$$

The curved length (S) of the parabola is given by [66]

$$S = \frac{H_p}{2} \left\{ \sec\left(\frac{\varphi_r}{2}\right) \tan\left(\frac{\varphi_r}{2}\right) + \ln \left[\sec\left(\frac{\varphi_r}{2}\right) + \tan\left(\frac{\varphi_r}{2}\right) \right] \right\} \quad (2.14)$$

The geometric concentration ratio, C is defined as the ratio of the collector aperture area to the receiver area [25]. From Figure 2.10 (b), C can be given by

$$C = \frac{\text{Effective aperture area}}{\text{Receiver tube area}} = \frac{A_a}{A_r} = \frac{w_a L}{\pi D L} = \frac{w_a}{\pi D} \quad (2.15)$$

Substituting Equation 2.8 and 2.11 into Equation 2.15 gives the concentration ratio as

$$C = \frac{\sin\phi_r}{\pi\sin\theta_m} \quad (2.16)$$

The value of C is always greater than one for concentrating solar collectors, and for a PTC it ranges between 70 and 80 [63]. The maximum concentration is obtained when the rim angle is 90° and Equation 2.16 becomes

$$C_{max} = \frac{1}{\pi\sin(\theta_m)} \quad (2.17)$$

where θ_m is half the acceptance angle as shown in Figure 2.10. It is important to optimize the acceptance angle ($2\theta_m$) in order to increase the amount of solar radiation that falls on the collector. This angle is basically the angular field within which the solar radiation is collected and focused on the receiver without tracking [25].

For a perfect dual tracking system, the maximum concentration ratio is much higher as it depends only on the sun's disk which has a width of 0.53° ($32'$) $= 2\theta_m$ [59]

$$C_{max} = \frac{1}{\sin^2(\theta_m)} = \frac{1}{\sin^2(16')} \quad (2.18)$$

The accuracy in solar tracking and perfection in the construction of collectors is always essential in increasing the value of the concentration ratio.

2.4.2 Optical Performance for a PTC

The optical model of the PTC takes into account factors including optical properties of materials used in construction of the system, size of the receiver tube relative to the collector, tracking errors, collector surface imperfections and geometrical errors. These factors rule out the possibility of a perfect system since they result in losses, hence reducing its performance.

2.4.2.1 Optical efficiency (η_o)

Optical efficiency is defined as the ratio of energy absorbed by the receiver to the energy incident on the collector's aperture [67]. It depends on four terms. The optical efficiency is expressed as [25];

$$\eta_o = (\rho_a \tau_g \alpha_r \gamma) K(\theta) X_{END} \quad (2.19)$$

where ρ_a is the collector reflectivity, γ is the intercept factor which is ratio of the energy intercepted by the receiver to the energy reflected by the collector [59], τ_g is the transmittance of the glass tube and α_r is the absorptivity of the receiver.

The incidence angle modifier $K(\theta_i)$ describes an additional impact of the incidence angle to the collector output results. It is defined as the ratio of thermal efficiency at a given angle of incidence to the thermal efficiency at a normal incidence [68]. The incidence angle modifier is presented as an empirical fit to experimental data and is described by a polynomial function of the value of the angle of incidence [69].

$$K(\theta_i) = \cos \theta_i + 0.000884(\theta_i) - 0.00005369(\theta_i)^2 \quad (2.20)$$

The intercept factor (γ) is defined as the fraction of incident direct normal irradiation that is intercepted by the receiver tube at normal incidence. For an ideal parabolic trough collector system, the intercept factor should be 1. Practically, this is never the case because of inaccuracy in tracking the sun, imperfections of collector surface and misalignments of the collector. Other factors include accumulation of dirt on the collector and receiver tube, as well as shadowing from bellows at the end of the receiver tubes.

When the angle of incidence is equal to zero, all the radiation that fall on the collector is reflected such that it illuminates the entire length of the receiver tube which is equal to the collector length. Any deviation in the angle of incidence to any angle greater than zero results in one of the receiver length not being illuminated, hence loss of the reflected irradiation. This is called the end loss error. Figure 2.11 shows a schematic diagram on the occurrence of the end losses. For long collectors, the effects of end losses are not significant and can be ignored.

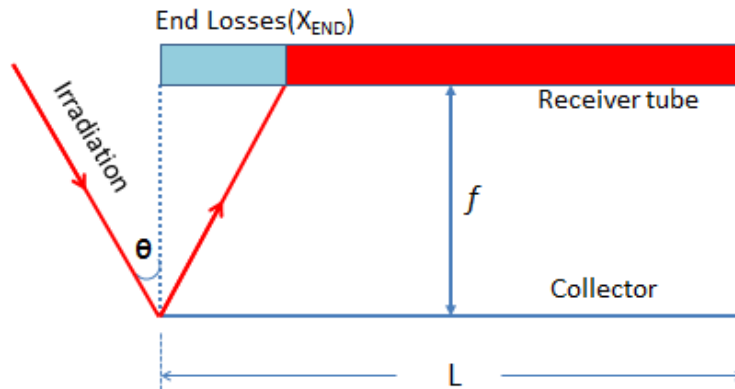


Figure 2.11: The end loss effect on a parabolic trough collector

The end loss is determined by the focal length (f) of the collector, angle of incidence (θ) and the collector length (L) as [70]

$$X_{END} = 1 - \frac{f}{L} \tan(\theta) \quad (2.21)$$

To minimize the effects of end losses, accurate two axis sun tracking is the solution. Most parabolic trough collector systems track the sun along a single axis from sunrise to sunset. For this study, the end losses are significant because manual tracking of the sun was adopted and the collector was also short.

2.4.3 Carnot Efficiency

Carnot efficiency is defined as the ratio of the work done by a heat engine to the heat drawn out of the high temperature reservoir of the engine. The thermodynamic efficiency limit for a PTC system depends on the temperature of the ambient (cold reservoir) and the HTF outlet temperature (hot reservoir). Therefore the thermal efficiency of a PTC is always equal to or less than the Carnot efficiency as shown below

$$\eta_c = \frac{W}{Q_h} = 1 - \frac{T_c}{T_h} \quad (2.22)$$

$$\eta_{th} \leq \eta_c \quad (2.23)$$

where W is workdone, Q_h is the heat energy of the hot reservoir, T_c is the temperature of the cold reservoir and T_h is the temperature of the hot reservoir.

2.4.4 Thermal model of a PTC

The thermal model estimates the amount of heat energy transferred to the working fluid and all the energy losses experienced through conduction, convection and radiation during the process. The thermal efficiency of a PTC is the ratio of the useful energy from the collector to the solar radiation incident on the collector [71]. Thermal efficiency is dependent on the useful heat gain by the transfer fluid, direct normal irradiation and the collector aperture area and it is given by [72]

$$\eta_{th} = \frac{Q_u}{A_a G_b} \quad (2.24)$$

where Q_u is the useful heat gain, A_a is the aperture area and G_b is the direct normal irradiation.

The useful heat gain Q_u is given by

$$Q_u = \dot{m}c_p\Delta T \quad (2.25)$$

Where $\dot{m}$ is the mass flow rate, c_p is the specific heat capacity of the transfer fluid and ΔT is the difference in the outlet and inlet temperature

The useful heat energy can be expressed in terms of the optical efficiency η_o of the system, heat removal factor F_R , heat loss coefficient U_L and the fluid inlet and ambient temperatures [25]

$$Q_u = F_R[G_b\eta_o A_a - A_r U_L(T_{fi} - T_a)] \quad (2.26)$$

Combining Equation 2.24 and Equation 2.26 yields the thermal efficiency

$$\eta_{th} = F_R \left[\eta_o - U_L \left(\frac{T_{fi} - T_a}{G_b C} \right) \right] \quad (2.27)$$

Equation 2.27 can be written as

$$\eta_{th} = F_R \eta_o - \frac{F_R U_L}{C} \left[\frac{T_{fi} - T_a}{G_b} \right] \quad (2.28)$$

where C is the concentration ratio.

The overall heat loss coefficient U_L is a function of the collector inlet and ambient temperatures [25]. It summarises all the heat losses. For a receiver tube without a glass tube the total heat loss is given by a product of the heat loss coefficient with tube area and the temperature difference between the tube and the ambient [73].

$$U_L = h_w + h_r \quad (2.29)$$

where the convective heat transfer coefficient h_w is given by

$$h_w = Nu_a * \frac{K_a}{D_o} \quad (2.30)$$

where the Nusselt number for the air Nu_a is deduced from the following expressions

$$\text{For } 0.1 < Re_a < 1000, \quad Nu_a = 0.4 + 0.54 * Re_a^{0.6}$$

$$\text{For } 1000 < Re_a < 50000, \quad Nu_a = 0.3 * Re_a^{0.6}$$

and the radiative heat transfer coefficient is given by

$$h_r = \varepsilon * \sigma * (T_r + T_a) * (T_r^2 + T_a^2) \quad (2.31)$$

For evacuated receiver tube losses due to conduction, convection and radiation are merged into one coefficient.

$$U_L = h_w + h_r + h_g \quad (2.32)$$

where the linear radiation constant, h_r is estimated from

$$h_r = 4\sigma\epsilon T_r^3 \quad (2.33)$$

where σ is Stefan's constant, ϵ is the emissivity and T_r is the receiver temperature.

In cases of large temperature variations within the receiver, the collector is divided into small segments and the overall heat loss coefficient is given as [74]

$$U_L = \left[\frac{A_r}{(h_w + h_{r,g-a})A_c} + \frac{1}{h_{r,r-g}} \right]^{-1} \quad (2.34)$$

The radiation heat transfer from the receiver to the glass envelope is given by

$$h_{r,r-g} = \frac{\sigma(T_r^2 + T_g^2)(T_r + T_g)}{\left[\frac{1}{\epsilon_r} + \frac{A_r}{A_g} \left(\frac{1}{\epsilon_g} - 1 \right) \right]} \quad (2.35)$$

Where ϵ_r and ϵ_g are emissivity of the receiver and glass respectively.

While the radiation heat transfer from glass to air is given by [74]

$$h_{r,g-a} = \epsilon_g * \sigma * (T_g + T_a) * (T_g^2 + T_a^2) \quad (2.36)$$

The collector heat removal factor (F_R) is defined as the actual useful energy gain by a collector to the useful energy gain if the whole collector surface were at a uniform temperature equivalent to the fluid inlet temperature [75]. F_R gives an insight into the performance of a collector since it relates the actual heat transfer to the possible maximum heat transfer. The value for the heat removal factor is guided by the mass flow rate of the fluid and its specific heat capacity plus the thermal properties of the system's receiver tube. It is expressed as:

$$F_R = \frac{\text{Actual useful energy gain}}{\text{Maximum possible useful energy gain if collector was at fluid inlet temperature}} \quad (2.37)$$

It is calculated mathematically as

$$F_R = \frac{\dot{m}c_p}{A_r U_L} \left[1 - \exp \left(- \frac{A_r U_L F}{\dot{m}c_p} \right) \right] \quad (2.38)$$

where F' is the collector efficiency factor and is defined as the useful energy gained to the energy collected if the entire receiver tube was at local transfer fluid temperature [75]. The collector efficiency factor can be expressed by [74]

$$F' = \frac{U_o}{U_L} = \frac{\frac{1}{U_L}}{\frac{1}{U_L} + \frac{D_o}{h_{fi}D_i} + \frac{D_o}{2K} \ln \frac{D_o}{D_i}} \quad (2.39)$$

The analysis of the PTC prototype in this thesis was analyzed based on the optical and thermal models presented in this chapter. Two systems whose difference is the receiver type were analyzed experimentally to study their performance in Botswana. Two different receivers were used on the collector – one locally made from a copper tube and a commercial receiver.

3 Methodology

This chapter describes the design, materials, fabrication of a parabolic trough collector (PTC) prototype and the data collection process. The parabolic trough collector was installed at a chosen flat site without shading from buildings and trees within the Botswana International University of Science and Technology (BIUST) campus. The geographical coordinates of the site of the project are 22.59404 °S and 027.12455 °E at an altitude of 973 m above sea level.

3.1 Design and fabrication of the PTC prototype

The prototype system comprised of two main assemblies, being the reflecting sheet assembly and the supporting base.

3.1.1 Determining the curvature of the collector

The curvature of the collector was determined using parabolic trough calculator software (a JavaScript programme) for a rim angle of 90 ° [76] to obtain the aperture width (w_a) and a focal length (f). Figure 3.1 shows the parabolic trace obtained from the software. The design software produced a parabola of width 1.072 m with a height of 0.268 m. This was obtained from the actual metal sheet width of 1.230m. The height of the parabola is equivalent to its focal length since the rim angle (φ_r) is 90°.

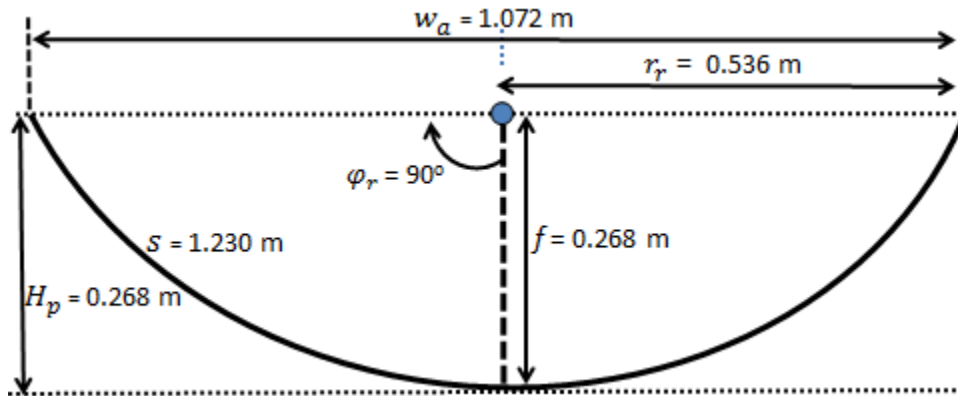


Figure 3.1: Parabolic profile obtained from the parabolic calculator software [76]

The values for the parameters shown in Figure 3.1, being the rim angle (φ_r), parabola radius (r_r), aperture width (w_a), height of the parabola (H_p), curve length of the parabola (S) and the focal length (f) were verified using Equations (2.9) to (2.13).

3.1.2 Designing the base support structure

The base support structure was fabricated from square steel tubes to hold the entire system. The thickness of the square tubes used for the base was 30 mm x 30 mm, and the base area is shown in Figure 3.2. Figure 3.2 (a) shows the design of the base, while Figure 3.2 (b) shows the side support structure to hold the collector's frame.

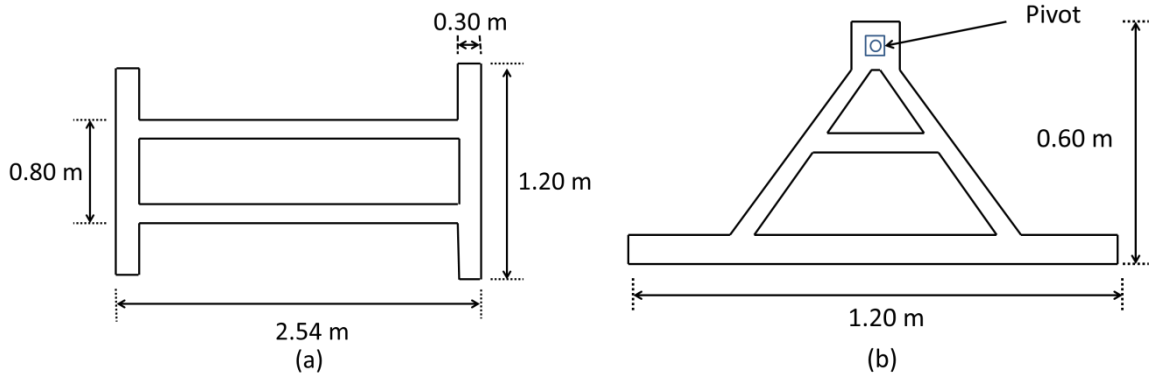


Figure 3.2: (a) Base support assembly, and (b) Side support assembly

3.1.3 Materials, Equipment and specifications

The majority of the materials used were sourced locally to cut on the general construction costs. Metal bars, copper tubes, hose pipes and square tubes used in constructing the mechanical support unit were sourced from local hardware stores in Palapye. A list of these materials is shown in Table 3.1.

3.1.4 Fabrication of complete prototype system

Figure 3.3 shows a complete collector frame joined to the base support structure. Four flat bars of length 1.230 m were bent into a parabola shape to obtain a width (w_a) of 1.072 m and a depth (f) of 0.268 m. Angle iron bars were then joined together to form a rectangle of dimensions 2.440 m by 1.072 m. An angle iron bar of length 2.440 m was then welded along the vertices of the parabolic shaped flat bars to hold the system firmly. Two flat bars were also welded from the vertex of the parabolic bar to the angle iron to maintain the parabolic focal distance.

Table 3.1: A summary of all the materials used to fabricate and install the prototype system

Description of Material	Specifications
Square tubes	30 mm x 30 mm
Flat bars	25 mm x 5 mm
Row bolts	M12 x 10
Copper tube	15 mm diameter
Bolts and nuts	3/8-16 mm
Metal rods	10 mm
Welding rods	5kg
Stainless steel mirror finish sheet	1230 mm x 2440 mm
Cutting discs	T41A Model
Wire mesh	12 m
Gate	Standard
Cement	50 kg
Standpipe	Standard
Hosepipe	10m
Spray paints	500 ml (Matt black, Silver)
Evacuated commercial Sunda solar receiver	2 m long
Stainless steel sheet	1.23 mx 2.44 m
Mylar reflective film	1m x 6m roll
Selective solar coat (Thurmalox)	5 L
Xplorer GLX data logger	PASCO Scientific
Mass flow meter	Inline



Figure 3.3: The PTC prototype system structure comprising of the collector holder and the base support

The reflecting support assembly was joined to the mechanical base unit through a pivot so that it could be tilted in two directions. The reflective stainless steel sheet was fitted onto the support assembly to form a parabolic shaped collector. Self-adhesive Mylar film was then lined on the inner surface of the collector. The reflectance of the film and a summary of properties of the materials used for the prototype are given in Table 3.2. A receiver tube was mounted at the focal line of this collector. A laser was used to ensure that the receiver tube was placed at the correct position by shining it perpendicular to the aperture opening of the collector.

Geared levers were installed to rotate the collector support and secure it rigidly at a desired angle with the collector perpendicular to the sun's position. The completed PTC prototype system before the before setting up for experiments is shown in Figure 3.4. The fabrication was done at a local metal workshop called "Supu Metal Clinic" with the help of a local craftsman. Modifications were done on both ends of the receiver tubes by drilling small holes then inserting Xplorer GLX temperature probes for the measurement of water temperature.

Table 3.2: Key features of the parabolic system using two types of receivers.

Description	PTC system with copper receiver	PTC system with commercial receiver
Collector length L	2.440 m	2.000 m
Collector area	3.001 m ²	2.460 m ²
Aperture area	2.616 m ²	2.144 m ²
Rim angle φ_r	90 °	90 °
Focal distance f	0.268 m	0.268 m
Receiver diameter (External) D_{oa}	15.0 mm	38.0mm
Receiver diameter (Internal) D_{ia}	13.4 mm	-
Collector aperture width w_a	1.072 m	1.072 m
Concentration ratio C	22.7	9.0
Absorber absorptivity α_r	0.96	0.94
Absorber emissivity ε_r	0.52	0.06
Mylar reflectance, ρ_a	0.97	0.97
Glass envelope transmittance τ_g	-	0.95
Mode of tracking	Manual	Manual

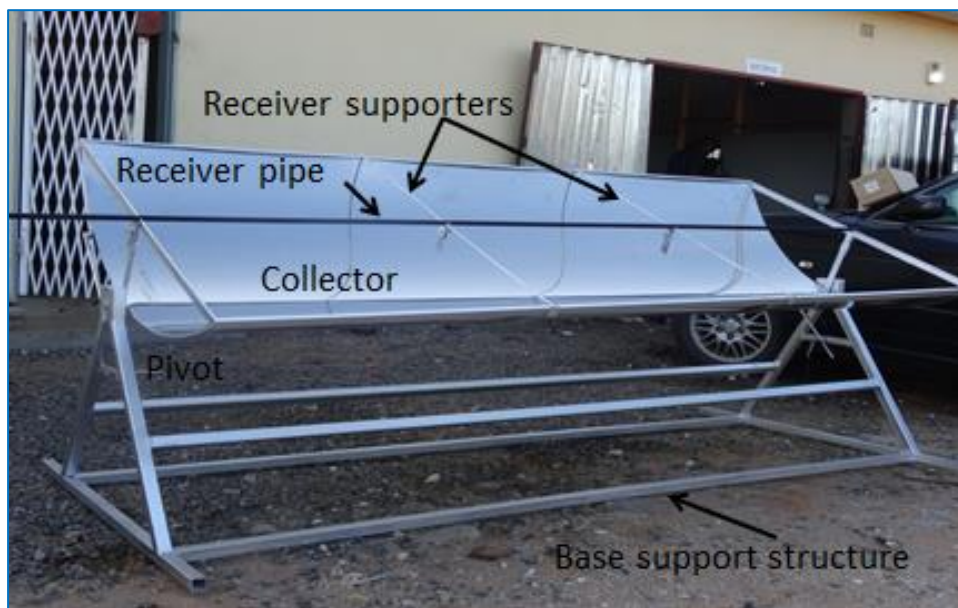


Figure 3.4: The PTC prototype system ready for testing within BIUST campus

3.2 Data Collection

3.2.1 Measurement conditions

The ambient temperature, wind speed and global horizontal irradiation values were recorded every fifteen minutes each day using an electronic temperature sensor, wind vane and a second class Kipp and Zonen CMP 3 pyranometer of sensitivity $16.09 \mu\text{V/W/m}^2$, respectively as shown in Figure 3.5. The equipment is installed at the Mahalapye meteorological weather station, 70 km away from the experimental site. These data was accessed through a sasscalweather [77], with permission from the Department of meteorological services office in Gaborone.

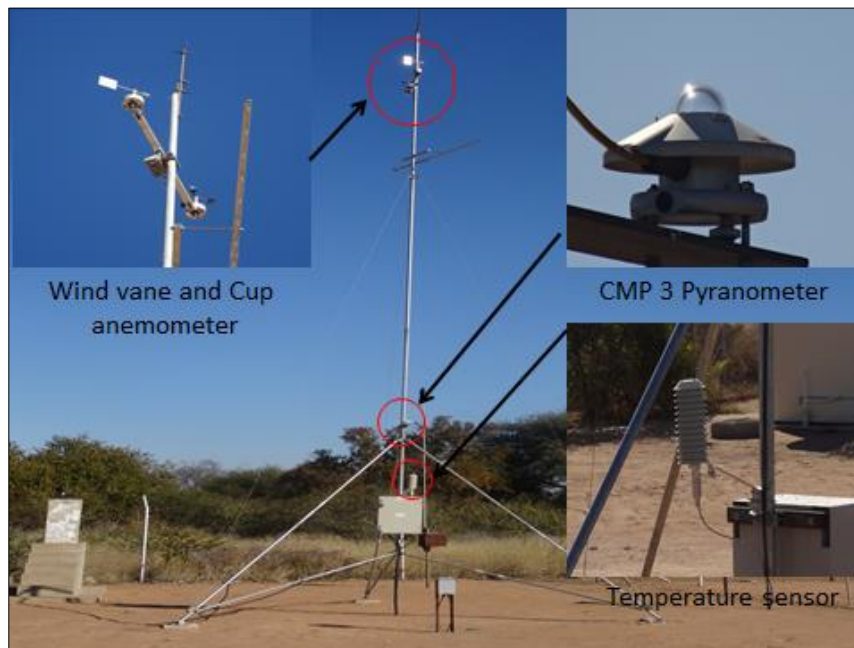


Figure 3.5: Tower comprising of a wind vane, cup anemometer, electronic temperature sensor and a Kipp Zonen CMP 3 pyranometer at Mahalapye meteorological centre

3.2.2 Experiments with coated copper receiver tube

The prototype system was firmly secured on a concrete slab using raw bolts with the collector facing geographic north and the trough tilted at an angle as shown in Figure 3.6. For the receiver to be on focus, the collector aperture was adjusted to be perpendicular to

the position of the sun at solar noon. The angle varied throughout the day, and was determined by the position of the sun which changed seasonally and throughout the day.

In the first set of experiments a copper tube of diameter 0.015 m, coated with a selective solar coat (Thurmalox) was used as a receiver. Water was chosen as a heat transfer fluid as it is readily available and cheap compared to molten salts and specialised oils. Therefore the system was connected to a standpipe using a hosepipe. Water inlet and outlet temperatures were recorded every minute between 09:00 hours and 16:00 hours during clear sky days, while the mass flow rate was kept constant on each day of the experiment. Two Xplorer GLX data loggers were used to record the temperature of the water, one at the inlet and the other at the outlet of the system. The inlet temperature was measured along the hose pipe. This was to avoid heating effects from end pipe absorptions which could lead to false water inlet temperatures. The outlet temperature was measured in the receiver pipe but at about 0.15 m before the exit point of the water. A Teflon float flowmeter connected between the standpipe and the inlet to the receiver tube was used as a regulator to keep the flow rate of the water constant throughout the experiment. The collector was manually rotated to keep the receiver in focus with the irradiation reflected by the collector, and this was done every thirty minutes throughout the experiment.

The recorded data was then used together with the meteorological data to calculate optical and thermal properties of the system. The influence of the mass flowrate on the water outlet temperature, useful heat absorbed and the Carnot efficiency were also deduced.

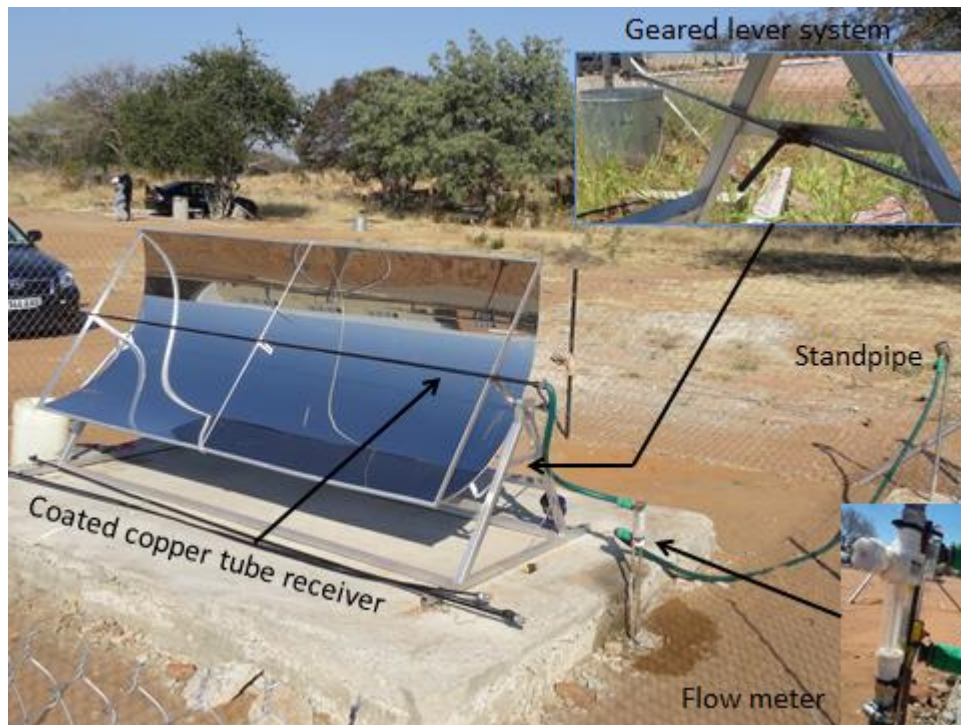


Figure 3.6: Experimental set-up using a solar coated copper tube as a receiver

3.2.3 Experiments with evacuated commercial receiver tube

The copper tube receiver was replaced by a commercial receiver and water inlet and outlet temperatures were recorded for four days while varying the mass flow rate. This was done in a similar way as when the coated copper receiver was used. The experimental set-up using the commercial receiver tube is shown in Figure 3.7.

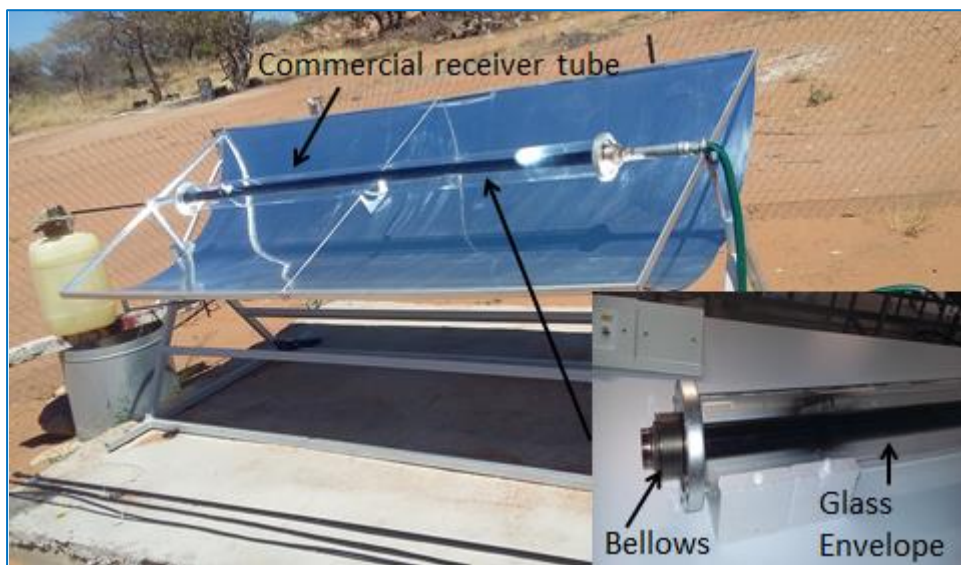


Figure 3.7: PTC prototype system using the commercial receiver supplied by Beijing sunda solar energy technology co. ltd

The receiver is made of a conducting metal tube coated with a specialized solar coat with very low emissivity and high absorptivity. The tube is covered with a glass envelope and has a vacuum between the glass envelope and the metal tube to lower heat losses from the transfer fluid within the tube. Figure 3.8 shows a schematic diagram of the receiver tube used in the experimental work.

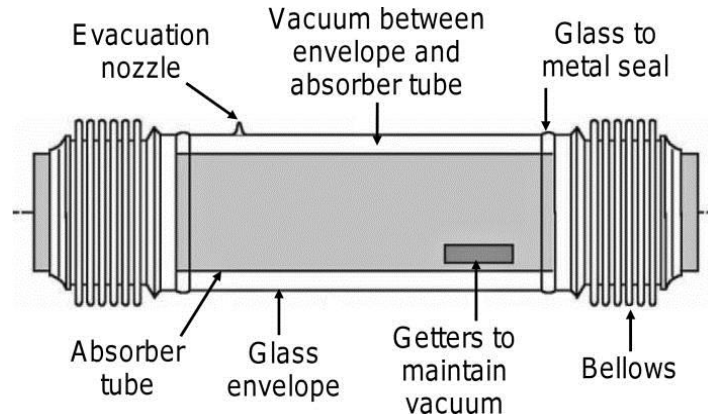


Figure 3.8: Schematic diagram of an evacuated commercial receiver tube

4 Results and Discussions

This chapter discusses the results obtained from two experimental set-ups using a selective solar coated copper tube and a commercial receiver. Optical, thermal and Carnot efficiencies of the two systems were calculated from experimental data collected on different days. The selected days for analysis of the coated receiver and the commercial receiver tube were chosen for the sake of comparison. The other data for different days is presented in the appendix. Parameters such as the air temperature, global horizontal irradiation and wind speed were obtained from Mahalapye meteorological station, for the period starting from February 2014 to December 31st 2016. Annual global horizontal irradiation (GHI), wind speed patterns and sunshine hours over this period of time are discussed.

4.1 Meteorological conditions of the region

Meteorological data from Mahalapye weather station was analysed to understand the annual and monthly terrestrial conditions in Botswana, especially Palapye where the experiment was carried out. Variations in the average ambient temperatures, wind speed, average GHI, sum of GHI and the total sunshine hours are discussed for the years 2014, 2015 and 2016. Figure 4.1 shows the annual variation of the ambient temperatures, wind speed, GHI and the sunshine hours.

Observations from Figure 4.1 (a) shows that the average monthly ambient temperatures are generally higher during the first three months of the year and the last four months of the year. The months observed fall within summer and spring seasons which normally associated with high temperatures. The lowest ambient temperatures were recorded for the months of June and July, and these are winter months where the temperatures are expected to reach their lowest values. For the year 2014, the average ambient temperatures are the lowest for all the months. This shows that there was generally an increase in the ambient temperatures for the year 2015 and 2016.

Figure 4.1 (b) shows that the average monthly wind speeds were erratic throughout the year. Nevertheless, the general trend shows a decrease in the wind speed from the month of January to June and July. In August the wind speed increased with the peak wind speeds recorded in the month of October before decreasing towards November and December.

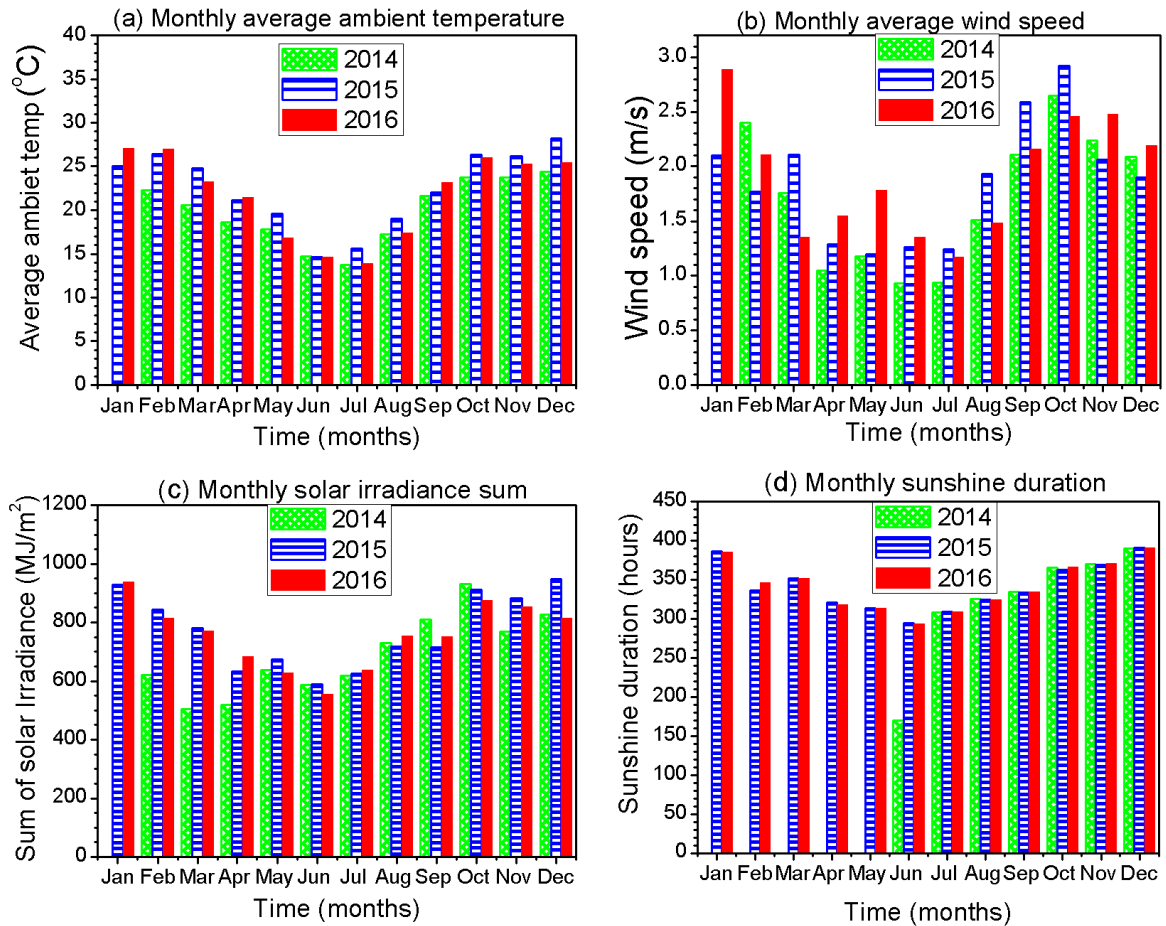


Figure 4.1: (a) Monthly average ambient temperatures, (b) monthly variation of wind speed, (c) monthly solar irradiance sum and (d) monthly sunshine duration for the period from 2014 up to 2016

The highest GHI sum as seen in Figure 4.1 (c) is received during the months of October to January, while the lowest GHI sum is observed between May and July. This trend is in agreement with the elliptical orbit of the earth around the sun, with summer occurring between November and March while winter occurs between May and August. Summers are characterized by much higher ambient temperatures due to increased irradiance and longer sunshine hours. During this period the sun is in the Southern hemisphere, either moving towards the tropic of Capricorn, or the Equator. In May to July, the sun would be in the northern hemisphere, either moving towards the tropic of Cancer or the Equator.

On the other hand, Figure 4.1 (d) shows that the number of sunshine hours experienced in Botswana depends on the seasonal changes. The sunshine hours increase up to a maximum average of 13.50 hours on the 21st December (winter solstice) and decreases to a minimum average of 10.75 hours on 21st June (summer solstice). The sunshine trend clearly shows that the highest clear sunshine hours are experienced in summer, especially during the months of October until January as expected. Lowest sunshine hours are observed during the winter season during the months of May up to July. Calculations further show a total of 4094.6 sunshine hours for the year 2015, while a total of 4104.3 hours were recorded for the year 2016. Conclusive calculations could not be performed for year 2014 because records show data from June to December only. This sunshine exposure in Botswana is one of the highest in the world at an estimated annual average of 3200 hours [20], though actual calculations shows it is much higher.

Ambient, water inlet and outlet temperatures for the coated copper receiver tube were collected on three different days. Figure 4.2 shows variations of the daily average ambient temperature, wind speed, GHI and sunshine hours as a function of time. It is observed from Figure 4.2 (a), that the average ambient temperatures increased from the first days towards the last days of the month. Daily lowest recorded temperatures also increased towards the end of the month. This trend shows a gradual transition from winter into summer season since the temperatures are becoming warmer. The average wind speed as observed from

Figure 4.2 (b) indicates that wind speed also fluctuated throughout the month but an increase is noticeable towards the end of the month. The highest wind speed was recorded on 16th of the month. One can notice that the wind speed was generally increasing towards the end of the month. From Figure 4.2 (c), the average daily GHI fluctuated slightly from a minimum of 250 W/m² to the highest value of 300 W/m². The overall trend shows an increase from the beginning of the month up to the end. Figure 4.2 (d) shows a steady increase in sunshine hours from 10.2 to 10.8 hours throughout the month. Only two days fall out of the linear trend in increasing sunshine hours. These deviations are possibly a result of cloud cover because no rain was recorded during the month. The total recorded sunshine hours for this month was 324.42 hours out of a possible 325.16 hours. The increase in daily GHI sum and sunshine duration can be attributed to the seasonal change from winter into summer.

The experiments with the commercial receiver were performed in the month of October 2016. Figure 4.3 shows variations of meteorological data during this period. The daily average ambient temperature from Figure 4.3 (a) fluctuated but an increase is noticed towards the end of the month. The lowest daily recorded temperatures also increased from 12.8 °C early in the month to about 19.1 °C on the last day of the month. From Figure 4.3 (b), wind speeds are erratic with the daily maximums above 4 m/s. The daily average wind speed varied between 2 m/s and 7 m/s. Overall, the wind speed is observed to increase towards the end of the month. In comparison with the month of August, the wind speeds for October are averagely higher.

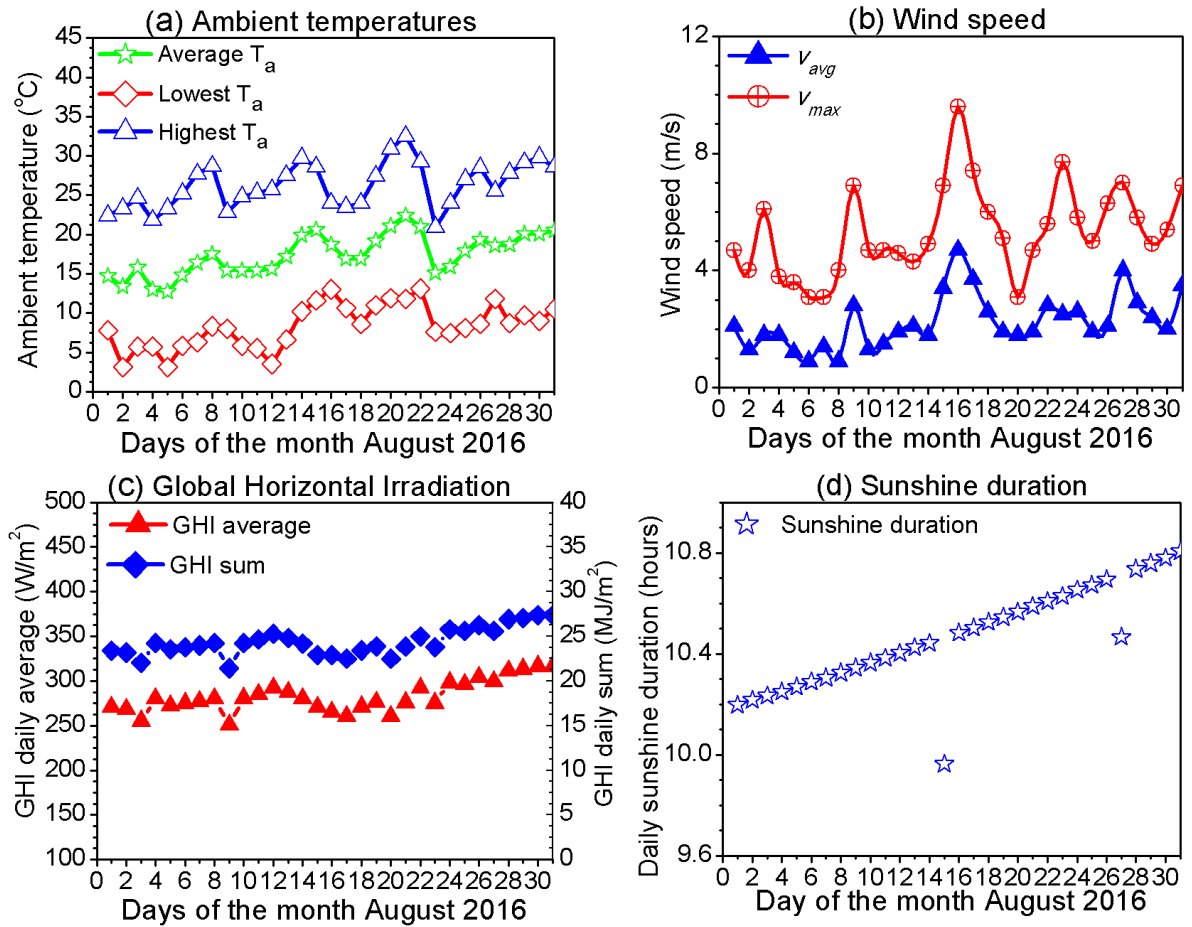


Figure 4.2: (a) Ambient temperatures, (b) wind speed, (c) global horizontal irradiation, and (d) sunshine duration as a function of days for the month of August 2016

Average daily GHI and the GHI sum calculated over 24 hours as seen in Figure 4.3 (c) fluctuated throughout the month with very steep differences observed on the 9th, and between 18th and 21st of the month. Because there was no rainfall recorded throughout the month, the drop in the daily GHI is attributed to cloud cover, humidity or dust

accumulation within the atmosphere. An increase in the average daily GHI and the GHI sum is noticed towards the last days of the month compared to the early days of the month.

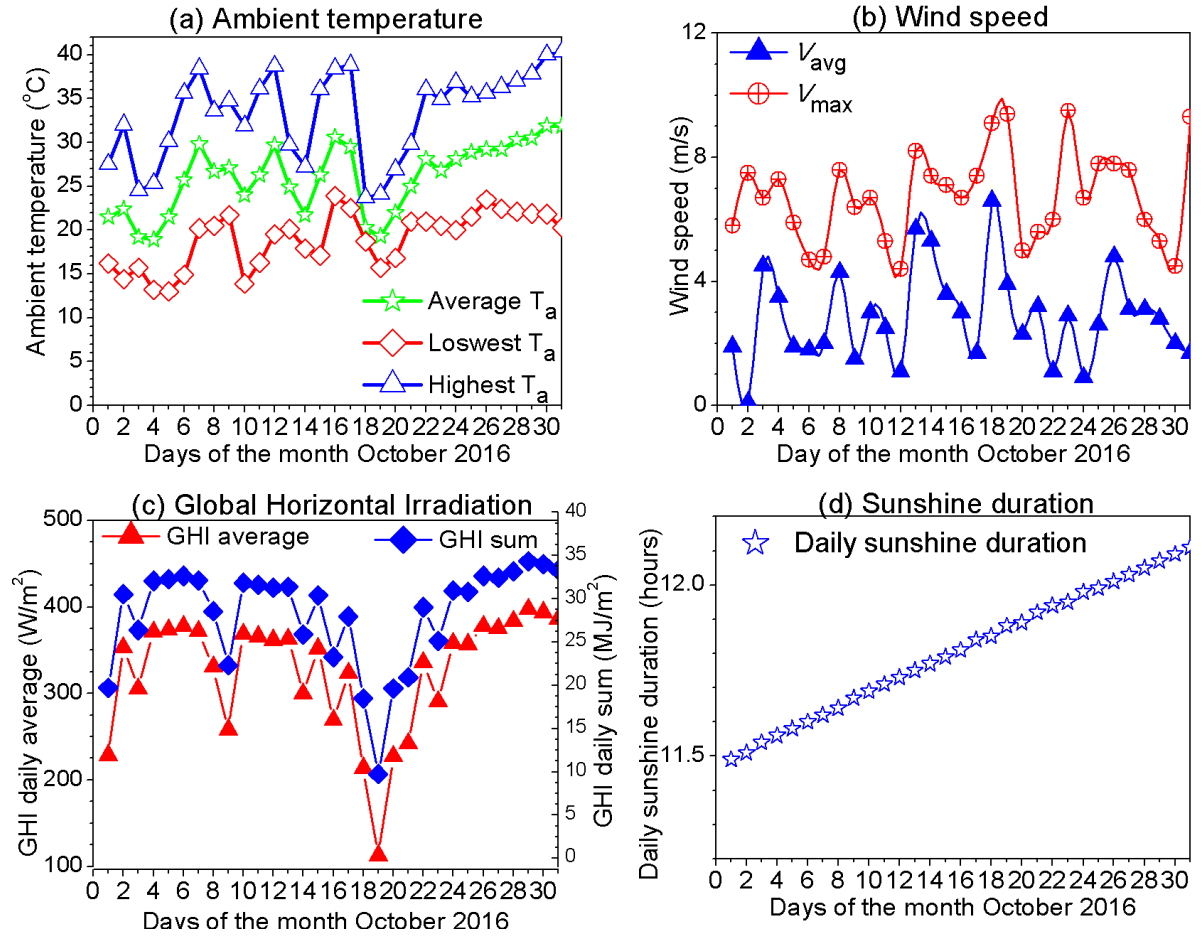


Figure 4.3: (a) Ambient temperatures, (b) wind speed, (c) global horizontal irradiation, and (d) sunshine duration as a function of days for the month of October 2016

The sunshine hours (Figure 4.3 (d)) increased steadily from 11.49 hours to 12.11 hours throughout the month, an average increase of 71.61 seconds on a daily basis. A straight line trend is observed with the progression of the days and corroborates the increase in daytime hours as expected in the gradual transition from winter to summer season. The total recorded sunshine hours for this month was 366.07 hours, an increase of 41.65 sunshine hours compared to 324.42 hours recorded for August 2016.

4.2 Performance of the PTC prototype

4.2.1 Using the coated copper tube

The optical, thermal and Carnot performance of the system are evaluated using the measured inlet and outlet temperatures of water (heat transfer fluid), mass flow rate, collector aperture area, ambient temperature, wind speed and global horizontal irradiation obtained from Mahalapye weather station. The experiments were performed for a period of seven hours with inlet and outlet temperature recordings taken at one minute interval between 09:00 hours 16:00 hours. The effect of factors such as wind speed, wind direction, irradiation and mass flow rate on the thermal performance of the system is also discussed. The heat loss coefficient (U_L) and the heat removal factor (F_R) are deduced for each day of the experiment and discussed. Considering that GHI was used in this experiment instead of direct normal irradiation (DNI), the thermal performance of the system is underestimated.

4.2.1.1 Effect of GHI on the outlet temperature

Variations of GHI, ambient, water inlet and outlet temperature as a function of time for the three different experimental days during the month of August 2016 are shown in Figure 4.4. The mass flow rate was 0.0036 kg/s on day one, 0.0026 kg/s on day two and 0.0012 kg/s on day three.

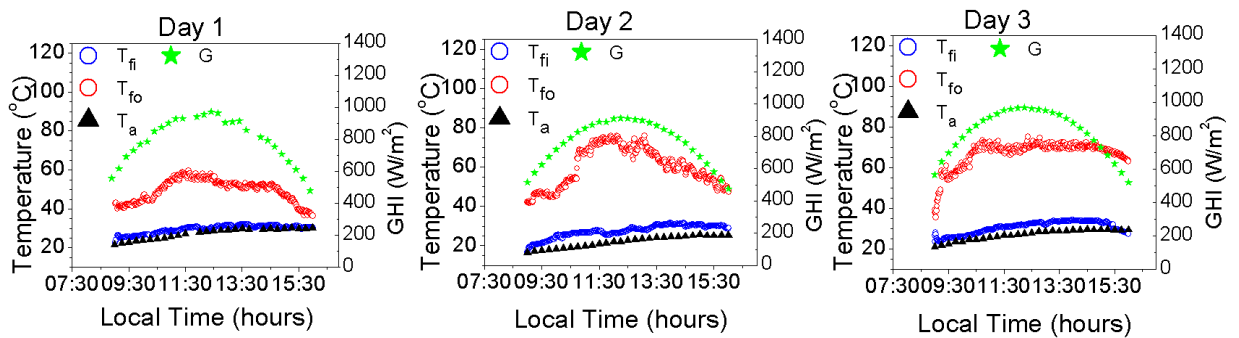


Figure 4.4: Inlet, outlet, ambient temperatures and GHI as a function of time for the PTC prototype using coated copper pipe receiver on three different days.

The average GHI for day one was 792.3 W/m² with the highest value of 970.2 W/m² observed at 12:30 hours and a minimum of 478 W/m² recorded at 16:00 hours, while the average water outlet temperature is 49.4 °C. Day two experienced a much lower average GHI as 760 W/m² with the highest value of 911.7 W/m² also at 12:30 hours and a

minimum of 473.2 W/m^2 at 16:00 hours but the daily average water outlet temperature was 60.2°C . Day three received the highest GHI value of 821 W/m^2 with the highest of 968.1 W/m^2 at 12:15 hours and a minimum of 521.2 W/m^2 at 16:00 hours at a daily average water outlet temperature of 66.4°C .

The water outlet temperature would increase from day one to day three if all other parameters such as wind speed, GHI and receiver size are kept constant. It is noticeable that the increase in the average GHI from day two, to day three resulted in an increase in the average outlet temperature. One can deduce that an increase in the GHI while decreasing the mass flow rate increases the output temperature provided the receiver is in focus and all other factors such as the wind speed and receiver tube diameter are kept constant.

4.2.1.2 Effect of wind speed on the outlet temperature

Variations in the wind speed, ambient, water inlet and outlet temperatures as a function of time for the three days within the month of August are shown in Figure 4.5. The average wind speed for day one was 4.0 m/s with the highest and lowest recorded wind speeds being 4.9 m/s at 09:30 hours and 2.8 m/s at 15:00 hours respectively. A drop in the wind speed between 09:30 hours and 11:00 hours resulted in a steady increase in the output temperature to reach a maximum of 59.7°C at 11:30 hours.

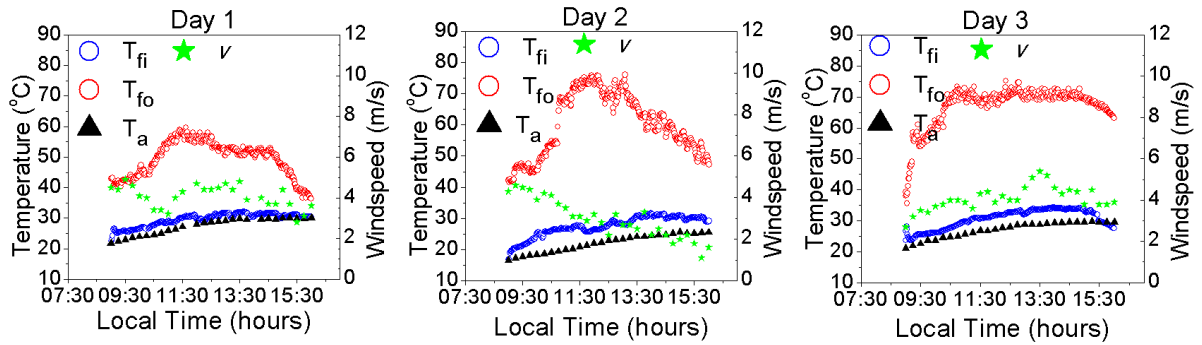


Figure 4.5: Ambient, water inlet, outlet temperatures and wind speed as a function of time for three different days

On day two the average wind speed was 3.0 m/s while the highest and lowest recorded wind speeds are 4.6 m/s at 09:15 hours and 1.1 m/s at 15:45 hours, respectively. The wind speed decreased gradually at 09:00 hours from 4.6 m/s to below 3.0 m/s at noon and 1.1 m/s towards 16:00 hours. It is evident that the wind speed for day two is lower than that for day one. The average output temperature for day two is much higher than that for day

one with the maximum of 75.8 °C observed at 12:07 hours when the wind speed was 2.9 m/s, and the GHI almost at its day maximum. This resulted in a much higher temperature difference for the water. Day three showed an average wind speed of 4.1 m/s with the highest and lowest wind speed recorded at 13:30 hours and 09:00 hours as 5.4 m/s and 2.9 m/s, respectively. The average wind speed on day three is relatively higher than that for day two but lower than day one.

4.2.1.3 Carnot efficiency (η_c)

The Carnot efficiency gives an insight into the theoretical effectiveness of a system that is operating between two temperatures. The instantaneous Carnot efficiencies (η_c) as shown in Figure 4.6 were calculated using Equation (2.22) for the values of the water inlet temperature (T_{fi}) and the water outlet temperature (T_{fo}) recorded by the GLX data logger for the three experimental days.

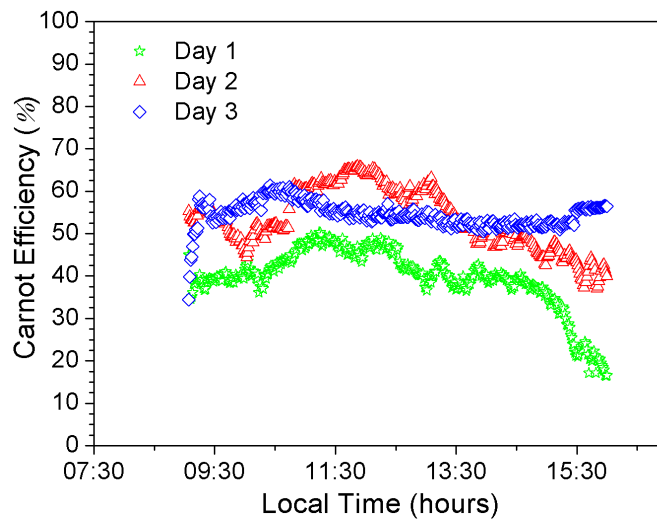


Figure 4.6: The Carnot efficiency of the PTC prototype when using a coated copper pipe receiver on three different days

The average Carnot efficiency (η_c) for day one was 38.5 % with the maximum value of 50.3 % obtained at 11:15 hours. The efficiency dropped gradually to 16.5 % at 16:00 hours as the GHI and the ambient temperature dropped. On day two, the efficiency reached a maximum of 64.8 % at 11:45 hours, with the day's average of 53.6 %. The efficiency then dropped to a minimum of 39.9 % at 16:00 hours. Day three showed a much higher average Carnot efficiency of 54.1 % as compared to the other two days, with the highest output value of 60.4 % achieved at 10:45 hours and then decreased to remain constant at 54.0 %. Carnot efficiency of more than in the range of 45% to 85 % is

considered ideal to achieve very good system performance results for concentration ratios of 10 up to 5000, respectively [78].

4.2.1.4 Useful heat energy output

The useful heat energy output refers to the rate of heat energy being added to a heat transfer fluid passing through the receiver tube [79], and is dependent on the specific heat capacity of the fluid (c_p), mass flow rate ($\dot{m}$) and the fluid's temperature difference (ΔT). From Equation (2.25) it is clear that an increase in the mass flow rate at a constant temperature difference will increase the useful heat energy output since the specific heat capacity is a constant. An increase in the temperature difference at a constant mass flow rate will increase the amount of useful heat energy output as well.

Calculations of the useful heat energy using Equation (2.25) resulted in values of 299.5 W, 356.0 W and 183.2 W for day one, two and three, respectively. Day two gave the highest energy of 356.0 W at a mass flow rate of 0.0026 kg/s. It is worth pointing out that the heat energy of 299.5 W on day one was achieved at the highest mass flowrate of 0.0036 kg/s, 183.2 W in day three was obtained at the lowest flowrate of 0.0012 kg/s. Day one with an average GHI of 792.3 W/m² and a water mass flowrate of 0.0036 kg/s was expected to yield the lowest value of useful heat energy output, but this is not the case even though the average wind speed is higher compared to day two. Day two had a lower GHI of 760.9 W/m² and a much lower mass flowrate of 0.0026 kg/s. The lowest wind speed in day two resulted in a lower heat loss, hence a much higher useful heat energy output from the system.

The useful energy gain on day three is the lowest, and that could be linked to high average wind speed since the average GHI is 821 W/m². One can deduce that when wind speed is increased and the GHI is maintained, the useful heat energy is lowered. A small increase in the water mass flow rate has minimum impact on the useful heat energy output compared to the effect of wind speed at constant GHI.

4.2.1.5 Optical efficiency (η_o)

The optical efficiency (η_o) of the PTC system is the ratio of the energy intercepted by the receiver to that incident on the collector aperture. It is calculated using Equation (2.19). A Gaussian intercept factor (γ) for a concentrator of concentration ratio (C) equal to $80/\pi$ is 0.876 [80]. For this study, this value was used in calculations of the optical efficiency.

The incidence angle modifier ($K(\theta_i)$) and the end loss effect (X_{END}) were calculated using Equation (2.20) and Equation (2.21) to yield 0.97 at an incident angle of 0° . Using values of $K(\theta_i)$, X_{END} and parameters from Table 3.2 and employing Equation (2.19) yielded the value of η_o as 76.75 %. This means that three-quarters of the energy that is focused by the collector is intercepted by the receiver.

4.2.1.6 Thermal efficiency (η_{th})

The thermal efficiency gives an insight into the performance of the parabolic trough collector by considering the ratio of the energy output to that input to the system. The water inlet and outlet temperatures, GHI, the specific heat capacity of water (c_p) and the collector aperture area were substituted in Equation (2.25) and (2.24) to calculate the instantaneous thermal efficiency. The thermal efficiencies for the three experimental days are shown in Figure 4.7. The thermal efficiency for the solar selective coated copper tube receiver was calculated using data recorded between 09:00 hours and 16:00 hours of the experiment. For this work, the GHI was used because of the inability to measure DNI since a pyrheliometer needed for such measurements was not available at BIUST and the Mahalapye meteorology station.

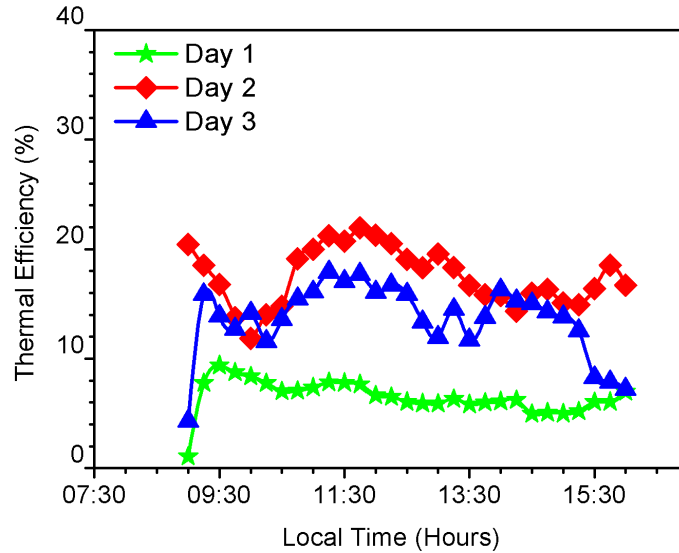


Figure 4.7: Thermal efficiency of the PTC prototype when using a coated copper pipe receiver for three different days

It is worth noting that the efficiencies calculated from the GHI values give the lower limit of the thermal efficiency because the DNI component of the irradiation is always lower than the GHI. The lowest instantaneous thermal efficiency of less than 11.0 % is observed

for day one and it is almost constant throughout the day. The highest thermal efficiency is obtained on day two, and reaches a maximum of 22.5 % around the solar noon. A nose dive in the efficiency is noticed between 09:00 hours and 10:00 hours, probably due to the higher wind speeds or loss of focus by the receiver. From Equation (2.24) combined with Equation (2.25), the specific heat capacity of water, the aperture area and the mass flow rates are constants for each experiment. It is worth noting that the efficiencies stated above are very low compared to values of 36.5 % [81], 48.8% [82], and 60.0 % [83] for similar systems. The lower thermal efficiency could be because we used GHI and others used DNI in calculations. The absence of a glass cover in this study decreased the efficiency of the system [83] [84].

When the instantaneous thermal efficiency η_{th} calculated from Equation (2.24) is plotted against the heat loss parameter $(T_{fi} - T_a)/G_b$, a straight line is obtained if the overall heat loss coefficient U_L is kept constant. This is called the performance curve of the PTC. Figure 4.8 shows the performance curves for the three days when using a coated copper pipe. The vertical intercept represents $F_R\eta_o$ while the gradient of the graph is equivalent to $U_L F_R/C$ as shown by Equation (2.28). The overall heat loss coefficient is a factor that gives an idea of the heat loss per unit area per unit temperature difference between the receiver tube and the ambient. On the other hand, F_R is the heat removal factor and represents the ratio of useful energy gain to the energy gained if the entire receiver tube is at the fluid inlet temperature.

The line of best fit through the plotted data for day one in Figure 4.8 could be described by;

$$\eta_{th} = 0.1490 - 2.009 \left(\frac{T_{fi} - T_a}{G} \right) \quad (4.1)$$

From Equation (4.1), $F_R\eta_o = 0.1490$ and $F_R U_L/C = 2.009 \text{ W/}^\circ\text{Cm}^2$. Because the geometric concentration ratio, C is equivalent to 22.7, $F_R U_L$ becomes $45.60 \text{ W/}^\circ\text{Cm}^2$. From Equation (2.19), the optical efficiency of the system $\eta_o = 76.8 \%$, thus resulting in a heat removal factor $F_R = 0.194$. This in turn gives an overall heat loss coefficient U_L of $235 \text{ W/}^\circ\text{Cm}^2$. The average instantaneous thermal efficiency calculated using Equation (2.24) amount to 10.45 % and is lower than the value of 14.14 % obtained from Equation (4.1) when using the day's average values of T_{fi} , T_a and G . This implies that the ratio of the heat energy collected by the water to the irradiation incident on the collector is only 10.45 to 14.14 %.

For day two the thermal efficiency shown in Figure 4.8 is given by the expression

$$\eta_{th} = 0.2251 - 7.362 \left(\frac{T_{fi} - T_a}{G} \right) \quad (4.2)$$

where $F_R \eta_o = 0.2251$ and $F_R U_L / C = 7.362 \text{ W/}^\circ\text{Cm}^2$. When the value of C is substituted, $F_R U_L$ becomes $167.12 \text{ W/}^\circ\text{Cm}^2$. From Equation (2.19), the optical efficiency of the system $\eta_o = 76.8 \%$, thus resulting in a heat removal factor of $F_R = 0.293$. This in turn gives an overall heat loss coefficient U_L of $570.38 \text{ W/}^\circ\text{Cm}^2$. The average thermal efficiency calculated using Equation (2.24) was 17.48% , slightly lower than the value of 22.51% obtained using Equation (4.2).

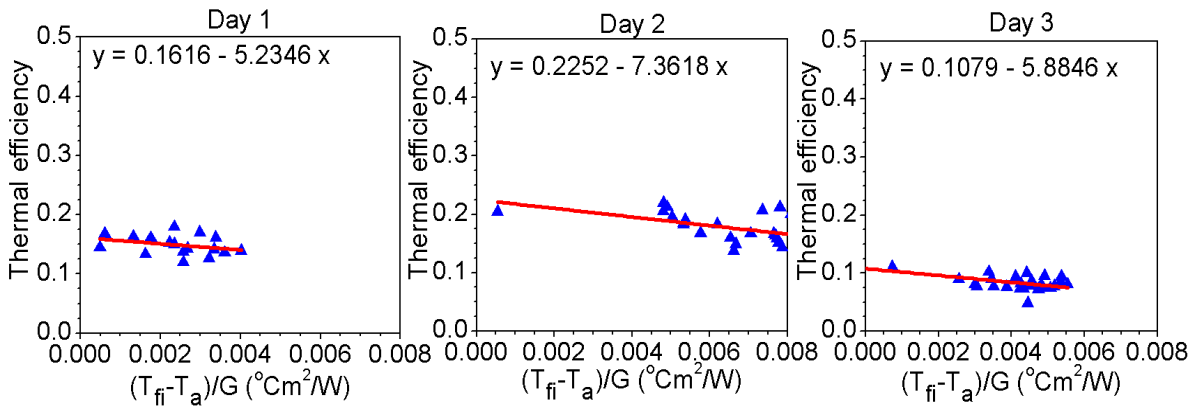


Figure 4.8: Performance curves of the PTC using a coated copper tube as a receiver on three different days

Thermal efficiency for day three is represented by the expression

$$\eta_{th} = 0.1129 - 6.919 \left(\frac{T_{fi} - T_a}{G} \right) \quad (4.3)$$

where $F_R \eta_o = 0.1129$ and $F_R U_L / C = 6.919 \text{ W/}^\circ\text{Cm}^2$. Substituting $F_R U_L = 157.07 \text{ W/}^\circ\text{Cm}^2$ gives the heat removal factor $F_R = 0.147$. The overall heat loss coefficient U_L becomes $1068.62 \text{ W/}^\circ\text{Cm}^2$. The average thermal efficiency calculated using Equation (2.24) was 7.65% and compares well with the value of 8.24% obtained using Equation (4.3).

According to the findings of Yassem [85] an increase in the fluid mass flow rate should increase the heat removal factor. This is because an increase in the mass flow rate decreases the absorber tube temperature, hence decreasing heat losses. For this work, the heat removal factor increased initially with an increase in the mass flow rate and then decreased. A summary of some of the parameters discussed are presented in Table 4.1. The values for the heat removal factor agree with the useful heat gain for the three days.

With an F_R of 0.194 for day one, the useful heat gain was averaged at 299.5 W, while the highest F_R of 0.293 resulted in the best useful heat gain of 356.0 W as expected. Day three with the lowest F_R experienced the lowest heat gain of 183.2 W. it was evident that higher F_R results in an increase in heat gain. The overall heat loss coefficient U_L is observed to be increasing with the mass flow rate of the water. The other factor was the decrease in the mass flow rate, which tends to be associated with the increase in the heat loss [85].

Table 4.1: A summary of all performance parameters for the coated copper receiver system for the three days of experimental measurements.

Measured parameters	Day one	Day two	Day three
Average global horizontal irradiation, GHI (W/m ²)	821.0	760.0	792.0
Average wind speed, v (m/s)	4.0	3.0	4.1
Average ambient temperature, T_a (°C)	28.6	22.7	28.0
Average water inlet temperature, T_{fi} (°C)	30.6	27.6	32.3
Average water outlet temperature, T_{fo} (°C)	54.6	71.2	70.3
Average water temperature difference, ΔT (°C)	24.0	43.6	38.0
Highest water outlet temperature (°C)	74.7	76.0	59.0
Average useful heat energy (W)	299.5	356.5	183.2
Mass flow rate (kg/s)	0.0036	0.0026	0.0012
Optical efficiency, η_o (%)	76.8	76.8	76.8
Carnot efficiency, η_c (%)	38.5	53.6	54.1
Heat loss coefficient, U_L (W/°Cm ²)	235.00	570.38	1068.62
Heat removal factor, F_R	0.194	0.293	0.147
Thermal efficiency, η_{th} (%)	14.1	22.5	8.2

On day one the average water inlet and outlet temperature was 24.0 °C when the wind speed was 4.0 m/s and the mass flow rate was 0.0036 kg/s. The average temperature difference between the water outlet and inlet on day one was 43.6 °C, and the average wind speed (v) was 3.0 m/s while the mass flow rate was 0.0026 kg/s. The thermal efficiency increased from 14.1 % on day one to 22.5 % on day two. Carnot efficiency on day one was 38.5 % while on day two it increased to 53.6 %.

One has to note that the average GHI on day one was 821.0 W/m^2 while on day two it was 760.0 W/m^2 . When the wind speed is almost constant on day one and three, it is noticed that on day one the water inlet and outlet temperature difference was 24.0°C while on day three it was 38.0°C . The Carnot efficiency increases from 38.5 % on day one to 54.1 % on day three. Thermal efficiency decreased from 14.1 % on day one to 8.2 % on day three. It can be deduced that increasing the water mass flow rate decreases the time spent by the fluid within the receiver tube hence reducing heat loss and increasing thermal efficiency.

It is evident that the performance of the PTC system is highly affected by the wind speed. It is also evident that an increase in the mass flow rate from 0.0026 kg/s to 0.0036 kg/s with an increasing GHI did not necessarily increase the thermal efficiency of the system, but instead lowered it from 22.5 % to 14.1 % as a result of the increase in the wind speed. The Carnot efficiency also decreased from 53.6 % to 38.5 %. As expected, the values of the thermal efficiencies as obtained using Equation (2.24) for all the three days were less than the corresponding Carnot efficiencies.

4.2.2 Commercial evacuated receiver tube

The analysis was carried out in the same way as the previous case of the coated receiver tube, though experiments were performed on different days within the month of October 2016.

4.2.2.1 Effect of GHI on the outlet temperature

Variations in GHI, ambient, water inlet and outlet temperatures as a function of time are shown in Figure 4.9 for four experimental days during the month of October 2016. The average GHI is 940.9 W/m^2 on day one, 917.7 W/m^2 on day two, 898.6 W/m^2 on day three and 910.1 W/m^2 on the fourth day of the experiment. The highest GHI of 1102.6 W/m^2 was recorded on day one, 1104.2 W/m^2 on day two, 1081.7 W/m^2 on day three and 1082.5 W/m^2 on day four. The lowest GHI values recorded at 16:00 hours for all the days were above 500 W/m^2 except for day two which experienced some cloudy conditions within the last two hours of the experiment, reducing the GHI to 281.2 W/m^2 . The average GHI resulted in average outlet temperatures of 79.0°C , 80.7°C , 74.0°C and 69.4°C for day one, day two, day three and day four, respectively. The maximum water outlet temperatures reached exceeded 87.0°C in all the experiments, and that occurred around solar noon on each of the days.

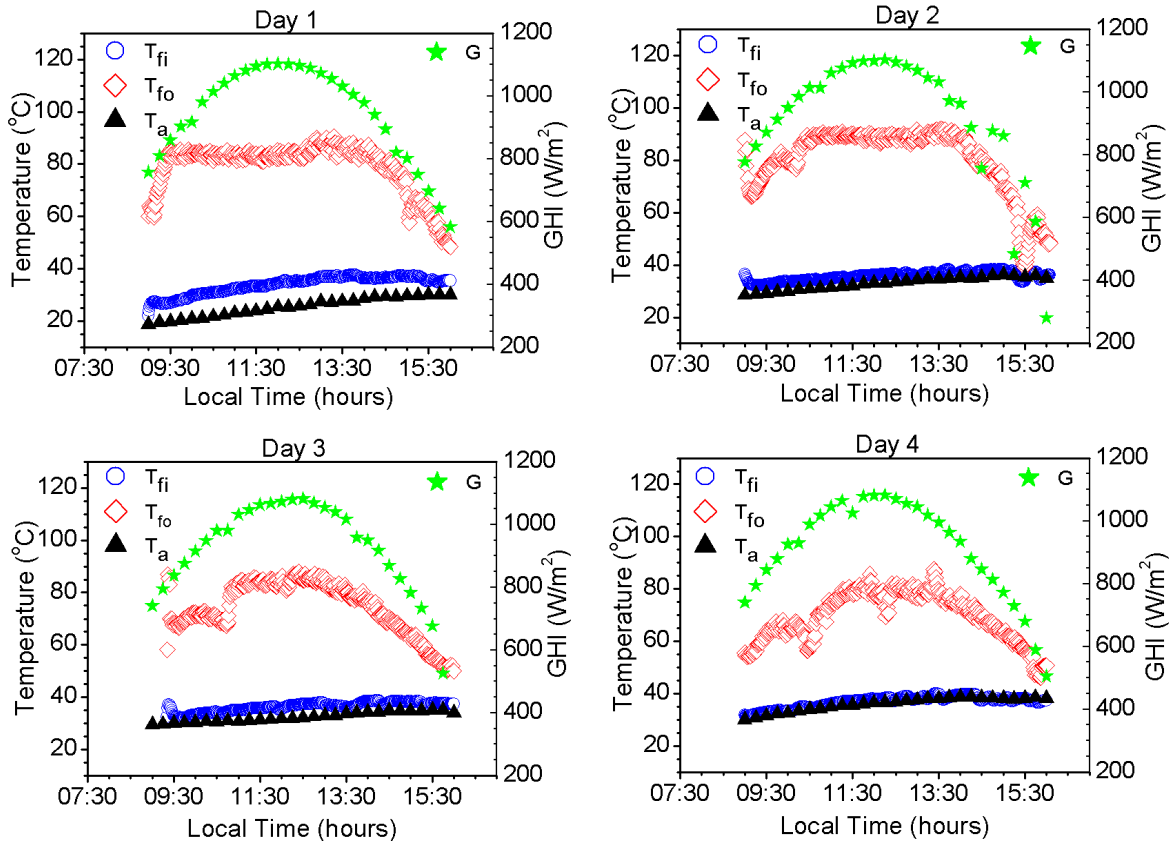


Figure 4.9: The inlet, outlet, ambient temperatures and GHI as a function of time for four different days

4.2.2.2 Effect of wind speed on the outlet temperature

Figure 4.10 shows the variations in the wind speed, ambient, water inlet and out temperature variations as a function of time for the four days in which the experimental data was collected. The calculated average wind speeds were 3.2 m/s, 3.6 m/s, 5.4 m/s and 2.6 m/s for day one, day two, day three and day four, respectively. The highest recorded wind speed for each day were very high, with day one reaching 5.9 m/s, 5.7 m/s on day two and 7.8 m/s on day three. Day four recorded a much lower wind speed of 3.7 m/s. The recorded highest wind speeds for day one, day two and day three were observed between 09:00 hours and 09:45hrs. The highest recorded wind speed for day four occurred late in the afternoon at 15:45 hours. The outlet temperatures on all the four days seemed to be less affected by the air motion. The wind speed did not have a significant effect on the outlet temperature. This is attributed to the insulation by the vacuum between the glass envelope and the metal absorber tube, preventing heat loss by air motion around the receiver tube.

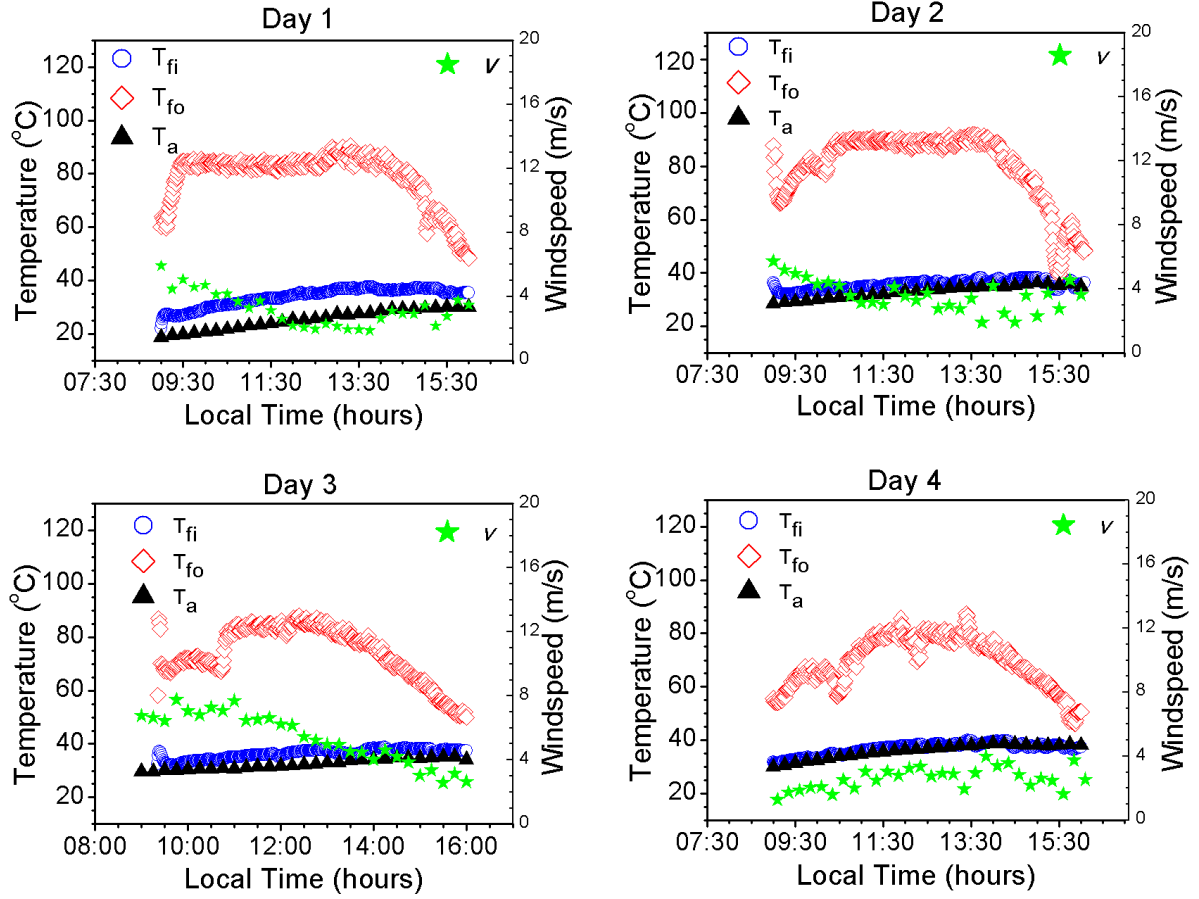


Figure 4.10: Inlet, outlet and ambient temperatures and wind speed as a function of time for four different days

4.2.2.3 Optical efficiency (η_o)

The optical efficiency (η_o) for the system using the commercial receiver is calculated in a similar way to the previous experiment with the coated copper tube. The parameters used in Equation (2.19) were substituted from Table 3.2. The concentration ratio was 9.0, therefore a Gaussian intercept factor (γ) was assumed [80]. The incidence angle modifier ($K(\theta_i)$) and the end loss effect (X_{END}) were calculated using Equation (2.20) and Equation (2.21) to yield 0.97 and 0.96, respectively. Equation (2.19) was then used to compute η_o to obtain a value of 80.34 %. This value tells us that over 80 % of the energy that is focused by the collector is intercepted by the receiver.

4.2.2.4 Carnot efficiency

Figure 4.11 shows the Carnot efficiency as a function of time for the four days during which the experiments were performed.

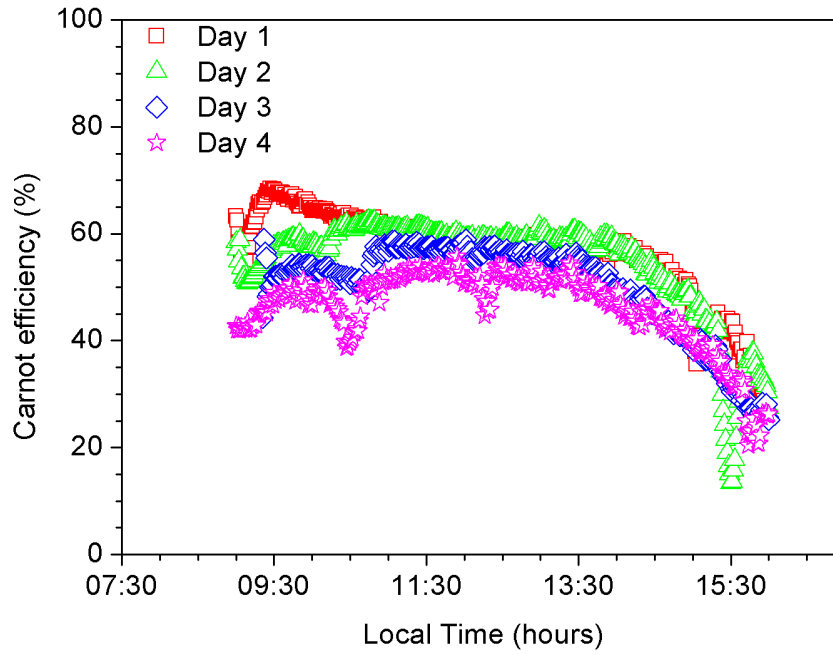


Figure 4.11: The Carnot efficiency of the PTC prototype when using commercial receiver on four different days

The Carnot efficiency depends on the inlet and outlet temperatures of the system. Day one showed high Carnot efficiency of about 70.0 % at 09:30 hours, which then dropped to an average of 56.2 % for about 5 hours before further dropping to 35.0 % at the end of the experiment. Day two showed a relatively lower average Carnot efficiency of 54.9 % with the highest value of 62.0 % recorded at 10:45 hours and the lowest value of 27.4 % in the afternoon. On day three the average Carnot efficiency was 48.0 % with the highest value of 57.5 % observed at 11:30 hours and the lowest efficiency of 23.3 % at 09:00 hours. Day four experienced a much lower average Carnot efficiency compared to all the three days at 45.7 % and with a peak of 53.3 % at 12:30 hours and the lowest of 26.7 % at 16:00 hours. The average Carnot efficiency for the four days ranged between 45.0 % and 56.7 %. For all the four days, it is worth noting that beyond 14:30 hours the Carnot efficiency dropped sharply, probably as a result of end loss errors due to the position of the sun and a drop in the GHI.

4.2.2.5 Useful heat output energy

Calculations using Equation (2.25) proves that the useful output heat increased from day one to day four from a minimum of 224.3 W on day one, 379.4 W on day two, 386.8 W on day three to 441.3 W on day four. The mass flow rate was 0.0012 kg/s on day one, 0.0020 kg/s on day two, 0.0026 kg/s on day three and 0.0036 kg/s on day four. From the change

in the wind speed from 3.2 m/s on day one, to 3.6 m/s on day two, and to 5.6 m/s on day three before dropping to 2.6 m/s on day four, the useful heat energy output on day three should have decreased. But this is not the case because of the glass envelope which eliminated the effects of winds. From literature, it has been shown that an increase in the fluid mass flow rate decreases the heat loss of a receiver tube [85], hence an increase in the useful heat energy output. Since the effect of wind speed was eliminated by the insulation around the receiver, and the GHI remains almost constant, we conclude that the increase in the water mass flow rate increased the useful heat energy output.

4.2.2.6 Thermal efficiency

The mass flow rate, specific heat capacity, absorber area, inlet & outlet temperatures and the corresponding GHI were substituted into Equation (2.25) and then into Equation (2.24) to calculate the instantaneous thermal efficiency. Figure 4.12 shows the instantaneous thermal efficiency as a function of time for the four days.

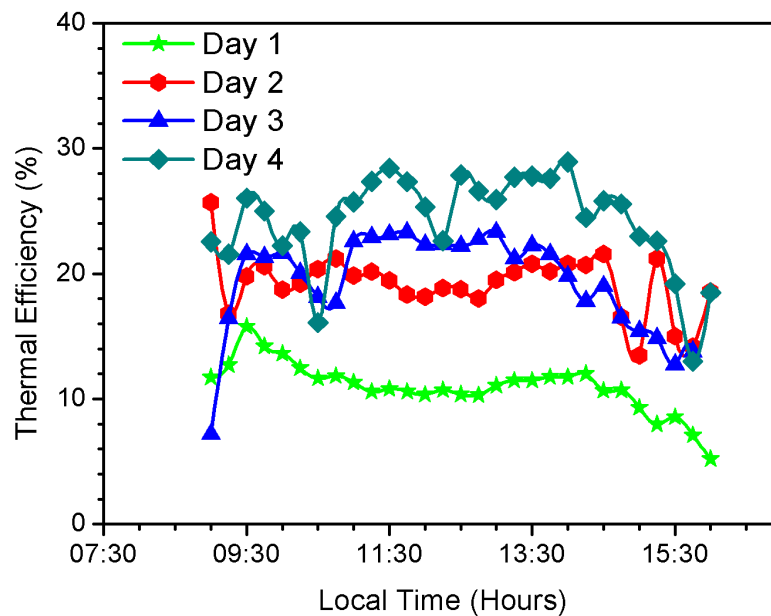


Figure 4.12: Instantaneous thermal efficiency as a function of time for the PTC with the commercial receiver tube on four different days

The general trend from Figure 4.12 shows that even though the average instantaneous efficiency fluctuated throughout each day, it relatively increased from day one up to day four. Average thermal efficiencies for the four days are 10.97 % for day one, 19.20 % for day two, 20.68 % for day three, and 24.23 % for the fourth day. The temperature difference between the inlet and the outlet is almost the same for the first two experimental

days, being 45.2 °C and 45.1 °C for day one and two, respectively. The temperature difference dropped slightly to 37.7 °C and 32.7 °C on day three and four, respectively. This is in contrast to the increasing thermal efficiency from day one up to day four. This increase in the thermal efficiency occurred with the increase in the water mass flow rate from the first day to the fourth day. Since the average GHI is within the same range of 898.6 W/m² to 940 W/m², it appears that the increase in the water mass flow rate played a role in reducing heat loss from the system, hence giving better thermal efficiencies. The fluctuations observed could be due to loss of focus of the receiver because of manual tracking of the sun.

4.2.2.7 Performance curves for the PTC with the commercial receiver

The instantaneous thermal efficiency, η_{th} as calculated from Equation (2.24) was plotted against the loss in temperature difference $(T_{fi}-T_a)/G_b$. Figure 4.13 shows the performance graphs during the four days of the experiment.

For day one, the line of best fit is described by Equation (4.4).

$$\eta_{th} = 0.1074 - 0.2085 \left(\frac{T_{fi}-T_a}{G} \right) \quad (4.4)$$

Equation (4.4) gives $F_R\eta_o = 0.1074$ and $F_RU_L/C = 0.2085 \text{ W/}^\circ\text{Cm}^2$. Substituting for the geometric concentration ratio, $C = 9.0$ gives $F_RU_L = 1.8765 \text{ W/}^\circ\text{Cm}^2$. Taking the optical efficiency of the system, $\eta_o = 80.3 \%$ gives the heat removal factor, $F_R = 0.134$. This in turn gives an overall heat loss coefficient, U_L of $14.00 \text{ W/}^\circ\text{Cm}^2$. The average thermal efficiency as calculated using Equation (2.24) was 11.0 % and compares well with 10.6 % obtained by substituting average values of T_{fi} , T_a and G into Equation (4.4).

For day two, the thermal efficiency is given by expression

$$\eta_{th} = 0.2083 - 4.5564 \left(\frac{T_{fi}-T_a}{G} \right) \quad (4.5)$$

Equation (4.5) gives $F_R\eta_o = 0.2083$ and $F_RU_L/C = 4.5564 \text{ W/}^\circ\text{Cm}^2$. Substituting the geometric concentration, C gives $F_RU_L = 45.60 \text{ W/}^\circ\text{Cm}^2$. The heat removal factor then becomes $F_R = 0.259$ and the overall heat loss coefficient U_L of $158.34 \text{ W/}^\circ\text{Cm}^2$. The average thermal efficiency calculated using Equation (2.24) amounted to 19.2 % and compares well with the value obtained using Equation (4.5) of 19.6 %.

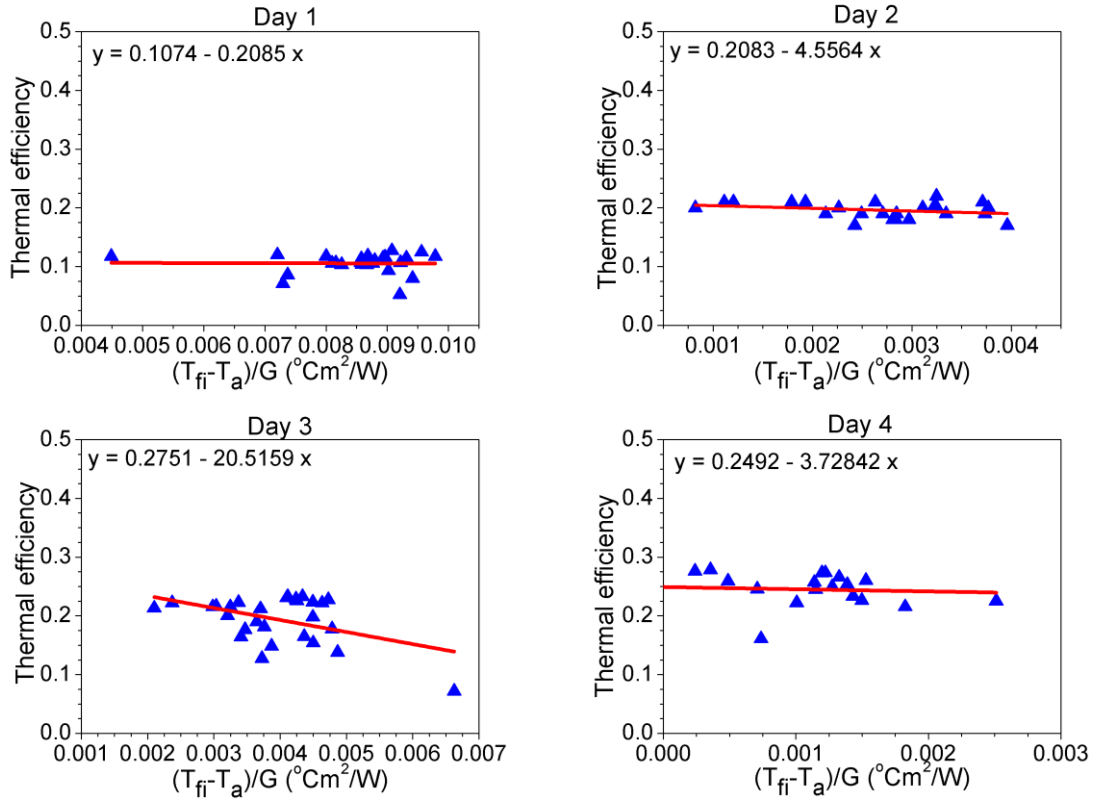


Figure 4.13: Thermal efficiency of the PTC using the commercial receiver for the four days in the month of October 2016

Thermal efficiency for day three is represented by the expression

$$\eta_{th} = 0.2751 - 20.5159 \left(\frac{T_{fi} - T_a}{G} \right) \quad (4.6)$$

Equation (4.6) gives $F_R \eta_o = 0.2751$ and $F_R U_L / C = 20.5159 \text{ W}^{\circ}\text{Cm}^2$. When the value of C is substituted, $F_R U_L$ becomes $-184.64 \text{ W}^{\circ}\text{Cm}^2$. The heat removal factor amounts to $F_R = 0.343$ while the overall heat loss coefficient U_L was $538.95 \text{ W}^{\circ}\text{Cm}^2$. The average thermal efficiency calculated using Equation (2.24) is 20.7 % and is slightly higher than the value of 18.9 % obtained using Equation (4.6).

The line of best fit through the plotted data for day four is described using the expression below.

$$\eta_{th} = 0.2492 - 3.7284 \left(\frac{T_{fi} - T_a}{G} \right) \quad (4.7)$$

Equation (4.7) gives the heat removal factor F_R as 0.310 and the overall heat loss coefficient U_L becomes $108.13 \text{ W}^{\circ}\text{Cm}^2$. The average thermal efficiency calculated using

Equation (2.24) amounted to 24.2 % and compares well with the value of 24.6 % obtained using Equation (4.7). This is the highest achieved efficiency of the four experimental days.

The heat removal factor is observed to increase in the first three days of the experiment from 13.4 % on day one, to 25.9 % on day two, and then 34.3 % on day three before dropping to 31.0 % on day four as shown in Table 4.2. The trend for the first three days is expected, because an increase in the mass flow rate of the working fluid (water) reduces heat loss, hence increasing the heat removal factor. An increase in the heat removal factor reduces the heat loss within the system, leading to an increase in the useful heat energy output as observed above. There seem to be a direct proportionality between the wind speed and the heat removal factor, since an increase in the heat removal factor is noticed to increase with an increase in the wind speed.

Table 4.2: A summary of all performance parameters for the commercial receiver system for the four days of experimental measurements.

Measured parameters	Day one	Day two	Day three	Day four
Average global horizontal irradiation, GHI (W/m^2)	940.9	917.7	898.6	910.1
Average wind speed, v (m/s)	3.2	3.6	5.4	2.6
Average ambient temperature, T_a ($^{\circ}\text{C}$)	25.4	33.1	32.5	36.0
Average water inlet temperature, T_{fi} ($^{\circ}\text{C}$)	33.8	35.6	36.3	36.7
Average water outlet temperature, T_{fo} ($^{\circ}\text{C}$)	79.0	80.7	74.0	69.4
Average water temperature difference, ΔT ($^{\circ}\text{C}$)	45.2	45.1	37.7	32.7
Highest water outlet temperature ($^{\circ}\text{C}$)	90.1	91.9	88.0	87.5
Average useful heat energy (W)	224.3	379.4	386.8	481.3
Mass flow rate (kg/s)	0.0012	0.0020	0.0026	0.0036
Optical efficiency, η_o (%)	80.3	80.3	80.3	80.3
Carnot efficiency, η_c (%)	56.4	54.5	49.8	46.2
Heat loss coefficient, U_L ($\text{W}/^{\circ}\text{Cm}^2$)	14.00	158.34	538.95	108.13
Heat removal factor, F_R (%)	13.4	25.9	34.3	31.0
Thermal efficiency, η_{th} (%)	10.97	19.20	20.68	24.23

The overall heat loss coefficient, U_L was $14.00 \text{ W}/^{\circ}\text{Cm}^2$ on day one, $158.34 \text{ W}/^{\circ}\text{Cm}^2$ on day two, $538.98 \text{ W}/^{\circ}\text{Cm}^2$ on day three and $108.13 \text{ W}/^{\circ}\text{Cm}^2$ on day four of the experiment.

Because of insulation by the glass envelope and the vacuum on the commercial receiver, the heat loss coefficient should be low. This is because heat loss through convection and radiation has been minimised. This is not the case in our experiment and the high values observed could be attributed to heat loss through conduction at the ends of the receiver tube. An increase in the wind speed cools the exposed uninsulated pipe extensions at both ends of the receiver tube enhancing heat loss by convection. Hence we deduce that the effect of wind in this case is notable through the non-insulated ends of the pipe of length 0.220 m on either side of the receiver. The heat removal factor is observed to be proportional to the overall heat loss coefficient, its increase results in an increase in the heat loss coefficient.

4.3 Discussion

During the two experiments, only the receivers were changed without any modifications to the collector system. The concentration ratio with the commercial receiver (9.0) was lower than that of the system with the coated receiver tube, C (22.7). In order to achieve the same concentration ratios, one would have to dismantle the whole set-up and build a new one with new dimensions. If the concentration ratios were equal, the commercial receiver would probably give much higher outlet temperature and hence higher Carnot efficiencies than for the coated receiver, under the same environmental conditions. The borosilicate glass envelope and the vacuum in between lowers thermal losses for the commercial receiver compared to the bare coated copper tube which was not shielded against heat losses. The coated receiver tube is more susceptible to heat loss through convection and radiation leading to poor thermal performance. These heat losses are exacerbated by the increase in the wind speed even at high GHI. The GHI was averagely high throughout all the experiments but there is a stronger influence by the wind speed on the outlet temperature than by GHI.

GHI instead of DNI was used in the calculations for this work because of unavailability of a Pyrheliometer to measure the DNI. It is worth noting that the efficiency of a concentrating solar power system depends on the useful heat gain and the interception of the DNI component of the irradiation. Useful heat gain is determined by the mass flow rate of the water and the temperature difference of the water that enters and exits the parabolic trough collector and receiver system. Considering that GHI is the sum of DNI

and diffuse horizontal irradiation (DHI), the actual efficiency of the prototype is much higher than what is reported in this work.

Table 4.3: Comparison of the performance of parabolic trough collector with the coated copper tube receiver to other similar systems from literature

Dimensions ($w_a \times L$) m	Flow rate (kg/s)	Irradia tion (W/m²)	Inlet tempera ture (°C)	Outlet tempera ture (°C)	Temperature difference ΔT (°C)	Thermal efficiency (%)	Ref
1.072* 2.44	0.002600	760.0	27.6	71.2	43.6	22.7	This work
1.063*2.44	0.001870	885.3	25.0	107.5	82.5	28.0	[65]
1.040*1.43	0.0000694	431.7	30.0	106.0	76.0	28.3	[86]
0.500*0.95	0.0003400	783.6	34.0	47.3	13.3	50.6	[87]

As the first PTC prototype to be tested in Botswana, the results obtained are comparable to those obtained by other researchers elsewhere. The value for thermal efficiency for this work is low but the output temperature falls within the range of industrial process heat applications and it is higher than that given by Macedo-Valencia et al [87]. Findings by Jaramillo et al [65] and Rizwan et al [86] were higher with outlet temperatures of 107.5 °C and 106.0 °C, respectively, but at much lower mass flow rates than in this work. The thermal efficiency obtained by Jaramillo et al (35.0 %) is slightly larger than what was obtained for this work, and this could be as a result of higher irradiation values and the lower mass flow rate. As for the findings from Rizwan et al [86], thermal efficiency (28.3 %) is slightly greater than for this work but this could because they used automatic sun tracking system in contrast to our manual sun tracking. In a nutshell, the low thermal performance for our system could be attributed to factors such as higher wind speeds, high emissivity factor for the ThurmaloX special solar coat, high end loss errors due to the orientation of the collector system as well as the use of the GHI instead of the DNI in calculations. The different terrestrial conditions of the geographical locations of the experiment also play a role in the different thermal performance of the systems, even though solar irradiation and wind maps show a similarity between the two places.

A comparison is also made on the performance of the parabolic trough collector system using the commercial receiver tube with similar but large scale commercialised systems as presented in Table 4.4. In comparison to the commercial systems listed in Table 4.4, the irradiation values are similar to our experiments. The inlet temperatures for these systems are above 100 °C yet the temperature differences (ΔT) are lower than in this dissertation, except for [88]. This shows that our system, despite the limitations in tracking and lower concentration ratio, performed quite well. The higher thermal efficiencies obtained for the commercial system are due to their advanced development and optimization of design and construction. The concentration ratios are relatively high and they use expensive and highly polished collectors with automated solar tracking mechanisms.

Table 4.4: Comparison of the performance of parabolic trough collector with the commercial receiver tube to other similar systems from literature

Flow rate (kg/s)	Irradiation (W/m²)	Inlet temperature (°C)	Outlet temperature (°C)	Temperature Difference ΔT (°C)	Thermal efficiency (%)	Ref
0.0036	910.1	36.7	69.4	32.7	24.2	This work
0.8462	968.2	151.0	170.3	19.3	62.2	[89]
0.3333	900.0	165.0	181.0	16.0	55.3	[90]
4.2028	815.0	104.8	143.1	38.3	76.0	[88]
0.8109	933.7	102.2	124.0	21.8	72.5	[91]

On the other hand, average outlet temperatures obtained for this work are in the range of 69 °C to 81 °C and are suitable for industrial process heat applications such as water heating, water desalination, cooling and refrigeration. Temperatures above 90 °C were recorded around solar noon for each experimental day. According to Kalogirou [92] temperatures for solar industrial process heat applications ranges from 60 °C to 260 °C. Further work can be done on the prototype to improve its performance.

5 Summary and Recommendations

A parabolic trough collector prototype was designed, fabricated and tested on BIUST campus. Experimental tests were carried out using an evacuated commercial receiver and a coated copper tube receiver. The water inlet and outlet temperatures were measured using a data logger while parameters such as the irradiation, wind speed and the ambient temperature were obtained from Mahalapye meteorological station. The highest water outlet temperature recorded using the commercial and coated receiver tubes were 91.9 °C and 76.0 °C, respectively. The maximum thermal efficiencies obtained for the two receivers were 24.2 % and 22.5 %, respectively. The outlet temperatures and thermal efficiency for the coated receiver are lower than those for the commercial receiver system because they are affected by the wind speed. Those for the commercial receiver are higher due to the influence from GHI. The maximum outlet temperatures from other authors using similar systems ranged between 47.3 °C and 107.5 °C. Their thermal efficiencies are relatively higher (28.0 % to 50.6 %) due to the use of highly polished reflectors, high absorbing, less emitting solar paints and the use of DNI in calculations compared to GHI for this work. The prototype used for this study is suitable for domestic hot water applications and industrial process heat applications such as blanching, evaporation, pasteurization, distillation, dyeing, etc.

The highest outlet temperatures for the coated receiver tube were low (76.0 °C) due to heat losses as a result of the wind. To counter the effect of wind, an evacuated commercial receiver tube can was used. Since the prototype was designed for the coated receiver tube, the system can be re-designed for the commercial receiver to achieve the highest possible concentration ratio. This will lead to outlet temperatures higher than 100 °C, resulting in steam. An automatic tracking system can be installed on the prototype to improve tracking of the sun. A complete loop system with an insulated storage tank will help improve on heat retention of the system. This can help to optimise the system to produce high temperature saturated steam that can be used for direct applications such as sterilization of equipment in hospitals, industrial process heat applications and small scale power generation if scaled up.

6 References

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7 APPENDIX A

Coated receiver tube: Effect of GHI

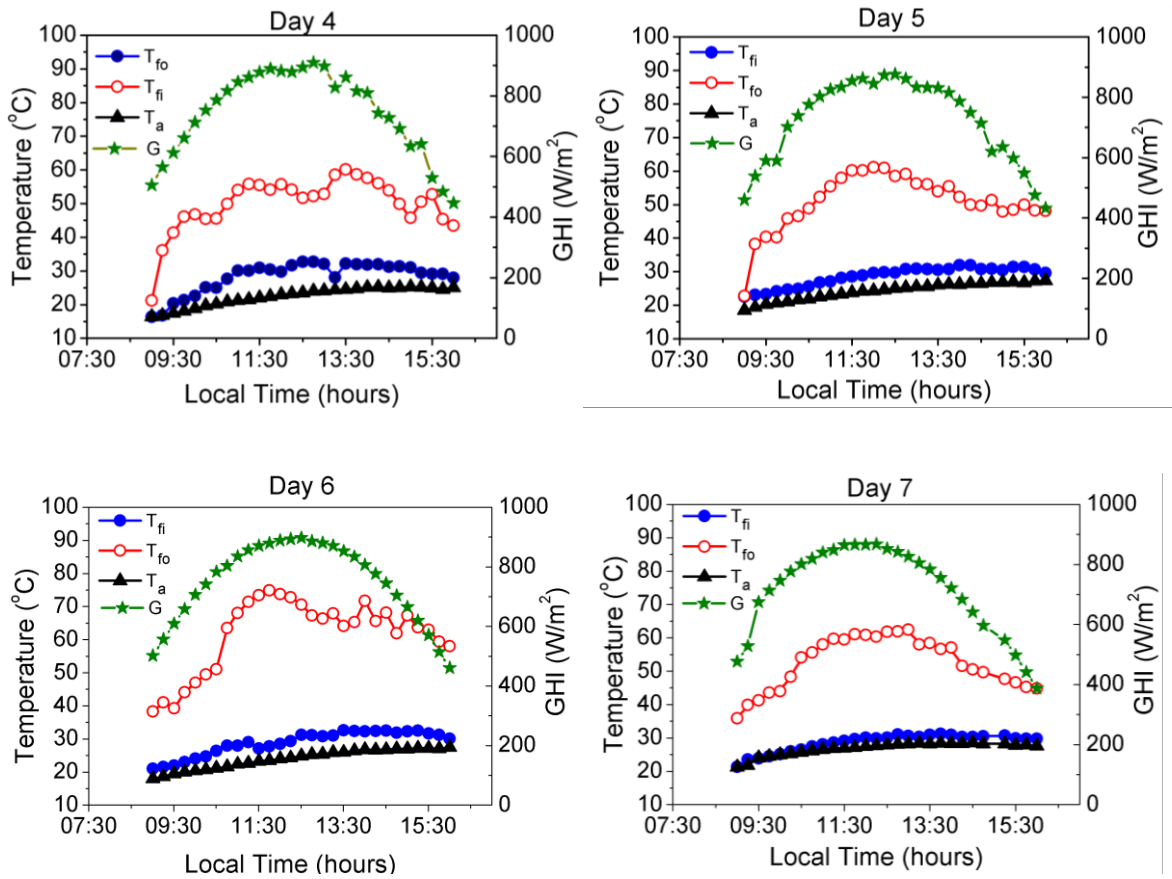


Figure A1: Inlet, outlet and ambient temperatures and GHI as a function of time for day four, five, six and seven shows that the outlet temperatures are dependent on the GHI

Coated receiver tube: Effect of wind speed

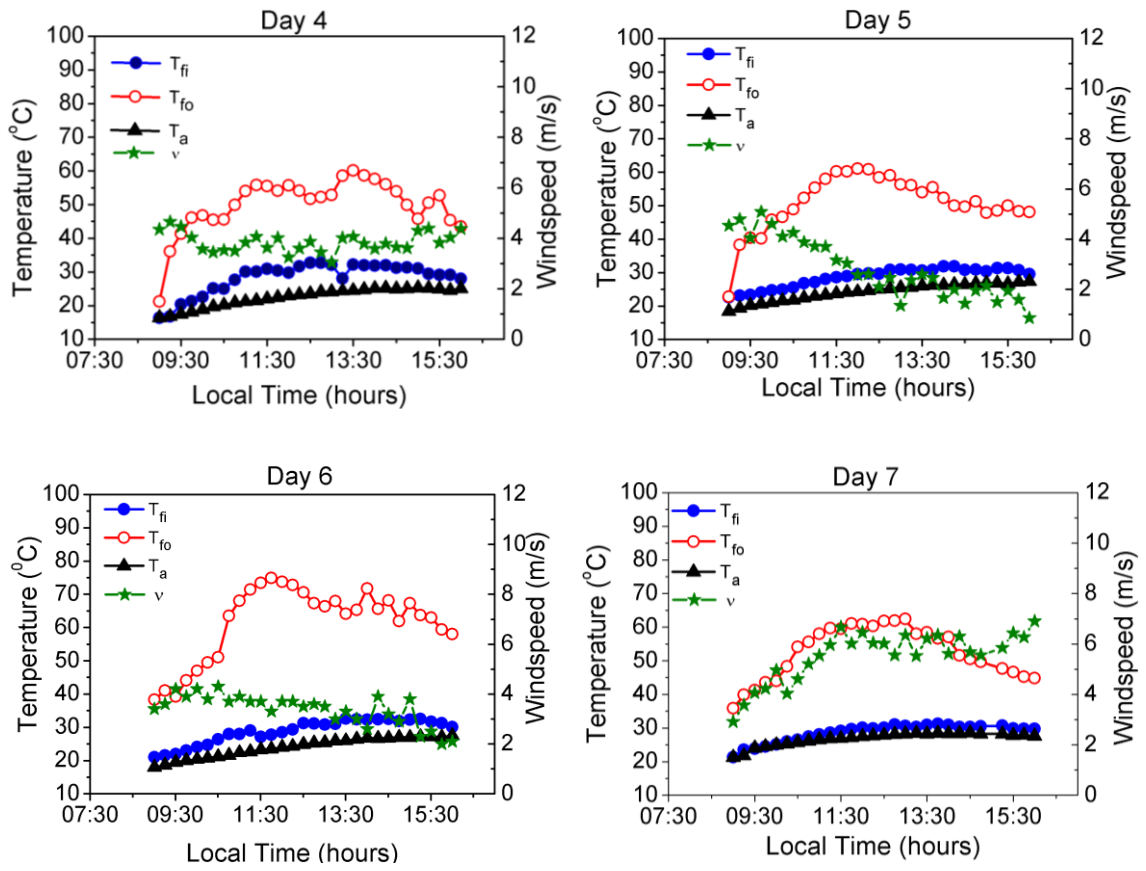


Figure A2: Inlet, outlet and ambient temperatures and wind speed as a function of time for day four, five, six and seven shows that wind speed increases heat losses hence lowering the outlet temperatures

8 APPENDIX B

Commercial receiver tube: Effect of GHI

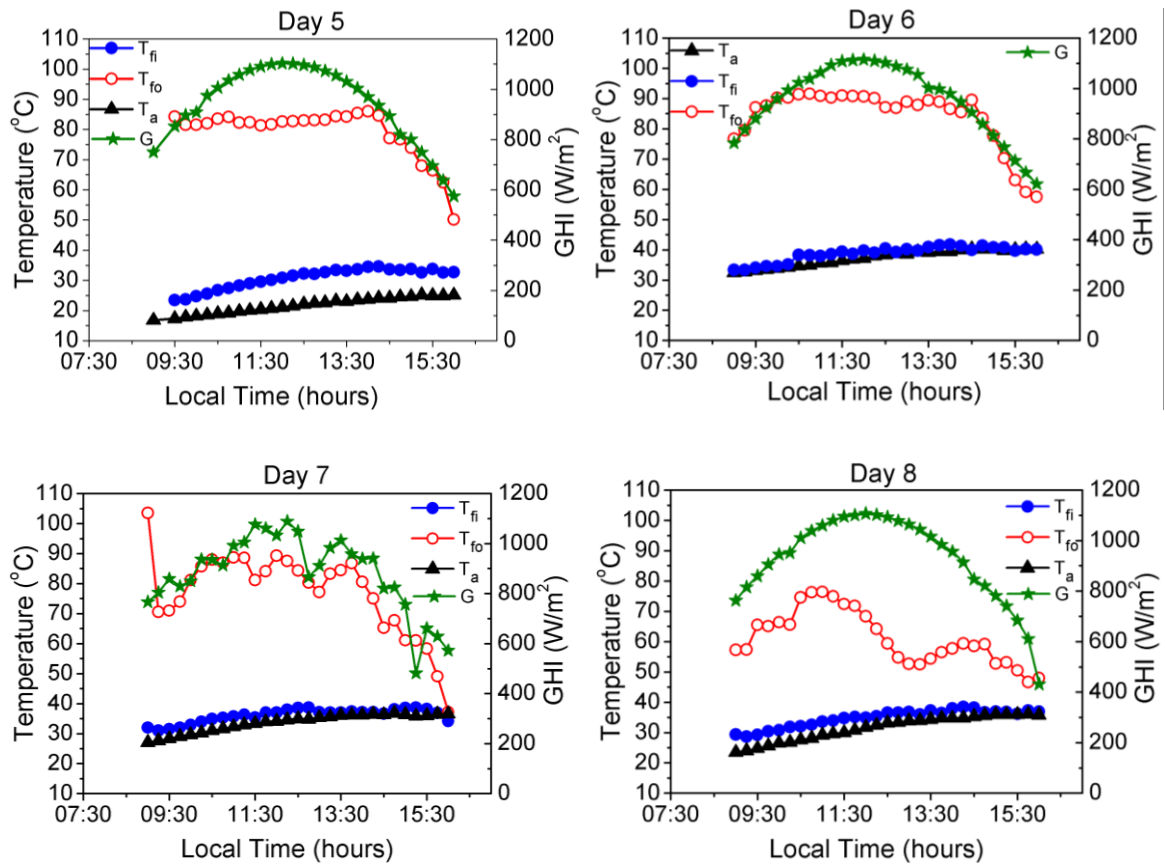


Figure B1: Inlet, outlet and ambient temperatures and GHI as a function of time for day five, six, seven and eight shows that high outlet temperatures are influenced by high GHI

Commercial receiver tube: Effect of wind speed

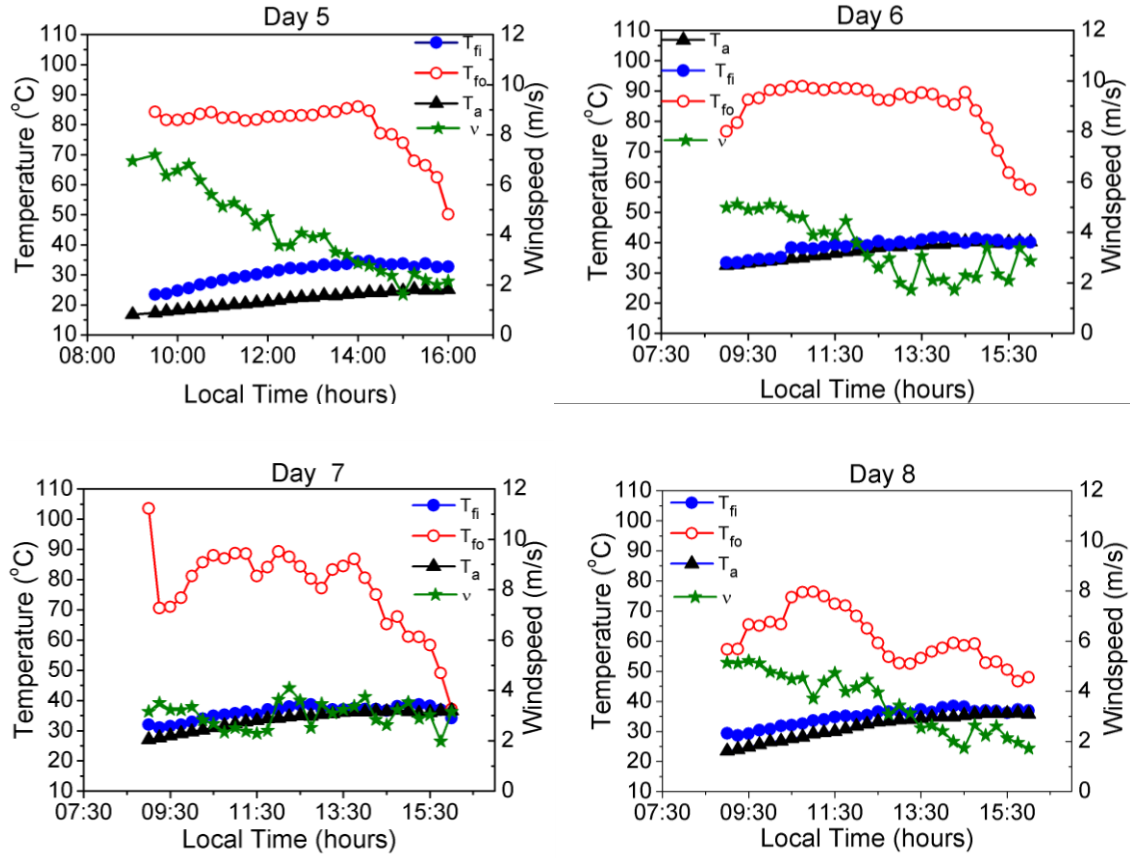


Figure B2: Inlet, outlet and ambient temperatures and wind speed as a function of time for day five, six, seven and eight shows that the wind speed has less or no effect on the outlet temperatures