THE DESIGN OF SOLAR CHIMNEY POWER PLANT FOR SUSTAINABLE POWER GENERATION

by

David Asante
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Declarations

I hereby declare that this thesis is a record of work carried out by myself, that it has not been the subject of any previous application for a degree, and that all sources of information have been acknowledged.

David Asante (PG6307311)	……………………	…………………..
Student Name and ID	Signature	Date

Certified by:

Dr. Emmanuel W. Ramde	…………………...	……………….
Supervisor's Name	Signature	Date

Certified by:

Dr. M.N Sackey	……………..………	………………….
Head of Department	Signature	Date

Abstract

The solar chimney power plant (SCPP) also known as 'solar updraft tower' is a non-concentrating solar thermal technology, which employs both solar and wind energy to operate. The plant essentially consists of three basic parts; a large greenhouse collector, which surrounds a tall chimney and a wind turbine geared to a generator at the base of the chimney. The collector converts solar radiation into thermal energy by means of greenhouse effect to heat the air beneath; subsequently the heated air is converted into kinetic energy in the chimney to drive the wind turbine to produce power.

The mathematical model as a tool was used to analysis the design and performance of the SCPP for electrical power production in Takoradi city. Based on the model, mathematical equations were obtained for the solar radiation collection, useful energy, and the electrical power output by using the wind turbine. The model was then used to estimate for the power output of a 25 m diameter size collector and 50 m tall chimney, and obtained an output of 48 kW. Based on the result obtained, it was observed that, the output power is effectively dependent on the solar irradiance, chimney height and collector diameter and that the power output could be enhanced when the setup is built on a large scale in an area where there is abundance of solar radiation.

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Acronyms

SWERA=Solar and wind energy resource assessment

SNEP= Strategic National Energy plan

SCPP= Solar chimney power plant

IGVs= Inlet guide vanes

CSP= Concentrating solar power

FSC= Floating solar chimney

CFD= Computational fluid dynamics

DNI= Direct solar beam

PCU= Power conversion unit

EC= Energy Commission

Nomenclature

A_{sc}	Area of solar chimney (m^2)
A_c	Area of collector (m^2)
C_p	Specific heat capacity of air (kJ/kg. K)
d_h	Hydraulic diameter (m)
d_{sc}	Chimney diameter (m)
F_R	Heat removal factor
g	Acceleration due to gravity (m/s^2)
Gr	Grasshof's Number
$\overline{H}$	Monthly average daily solar radiation on a horizontal surface (MJ/m^2)
$\overline{H}_d$	Diffuse component of the monthly average daily radiation (MJ/m^2)
$\overline{H}_T$	Monthly average daily total solar radiation (MJ/m^2)
$\overline{H}_o$	Monthly average extraterrestrial radiation (MJ/m^2)
H_c	Height of solar collector (m)
H_{sc}	Chimney height (m)
h_{cf}	Convective heat exchange between cover and the airflow(W/m^2K).
h_{gf}	Convective heat exchange between the airflow and the ground (W/m^2K).
h_{gap}	Collector air gap (m)
$h_{conv\ g-c}$	Convective heat transfer coefficient ground to cover (W/m^2K).
h_{rcg}	Radiation heat transfer coefficient ground to cover (W/m^2K)
h_{rs}	Radiation coefficient from cover to ambient air / sky (W/m^2K)
h_w	Wind convection loss coefficient (W/m^2K)
$\overline{K_T}$	Monthly average daily clearness index (deg)
K_{air}	Thermal conductivity of air (W/mK)
N_u	Nusselt Number
P	Collector perimeter (m)
P_d	Dynamic pressure (N/m^2)
P_e	Electrical power (kW)
P_{sc}	Pressure in the solar chimney (N/m^2)
P_{tot}	Total pressure (N/m^2)

P_t Pressure drop at the turbine (N/m^2)

P_r Prandtl Number

$\dot{Q}_u$ Useful power (kW)

$\dot{Q}_{loss}$ Rate of heat loss (kW) .

Q_{loss} Heat energy loss (kWh)

$\dot{Q}_{opt}$ Optical power (kW)

Q_{opt} Optical energy (kWh)

R_a Rayleigh's Number

R_{col} Radius of collector (m)

$\bar{R}_b$ Monthly average beam factor

R_2 Thermal resistance cover to ambient air/sky (K/W)

R_1 Thermal resistance ground to cover, (K/W)

$\overline{S_1}$ Monthly average daily absorbed solar radiation by collector cover (W/m^2)

S_2 Solar radiation absorbed by the ground soil (W/m^2)

T_a Ambient air temperature (K)

T_f Collector airflow temperature (K)

T_c Cover temperature (K)

T_g Ground soil/absorber temperature (K)

T_s Average sky temperature (K)

U_e Collector edge loss (W/m^2K)

U_g Ground loss (W/m^2K)

U_t Top loss (W/m^2K)

U_L Overall heat transfer coefficient (W/m^2K)

V_{wind} Average Wind speed of location (m/sec)

V_{sc} Velocity of airflow through the chimney (m/s)

V_{max} Maximum airflow rate through the chimney (m/sec)

Greek symbols

γ Azimuth angle (deg)

$(\overline{\tau\alpha})$ Collector average monthly absorbance - Transmittance product

$\mathcal{E}_c$	Collector emissivity
δ	Declination (deg)
ρ_{air}	Density of air (kg/m^3)
η_c	Efficiency of collector
η_{sc}	Efficiency of chimney
η_t	Efficiency of turbine
η_{scpp}	Efficiency of SCPP
$\mathcal{E}_g$	Ground emissivity
α	Ground absorptivity
ρ_g	Ground reflectance/ albedo
υ	Kinematic viscosity of air (m^2/sec)
$\emptyset$	Latitude of location (deg)
σ	Stefan-Boltzmann constant (W/m^2.K^4)
β	Slope or angle of tilt (deg)
w_s	Sunset hour angle (deg)
β'	Volumetric coefficient of expansion of air (1/K).

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CHAPTER 1

1.0 INTRODUCTION

1.1 Background

Sustainable energy also known as renewable or alternative energy is one, which is derived from naturally occurring phenomenon such as wind, biomass, solar, tidal, wave, and hydroelectric power. They are sustainable because, their usage today will have no effect in the next generation and are also environmentally benign.

The increasing concerns over the depletion of fossil fuels, increasing energy prices and the threat to climate change have triggered engineers and scientists to find new methods to reduce the usage and consumption of non-renewable energy sources particularly fossil fuels. Statistical records by the International Energy Agency (IEA 2012) showed that, there is high dependence on the use of non-renewable energy in the share of the global total primary energy supply (TPES) than any other energy resource. (See Figure 1.1)

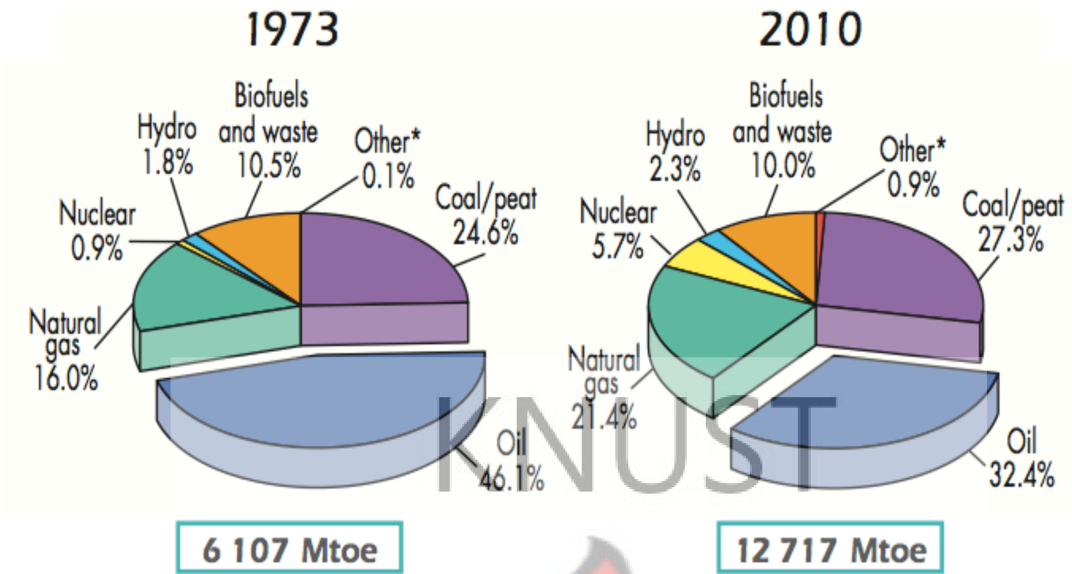

Figure1.1: Fuel Shares of TPES 1973 and 2010. Source: International Energy Agency, 2012

In the power-generating sector, coal dominates the total share of fuel for generating electricity than any other fuel source with a share of about 40 percent in the year 2010.

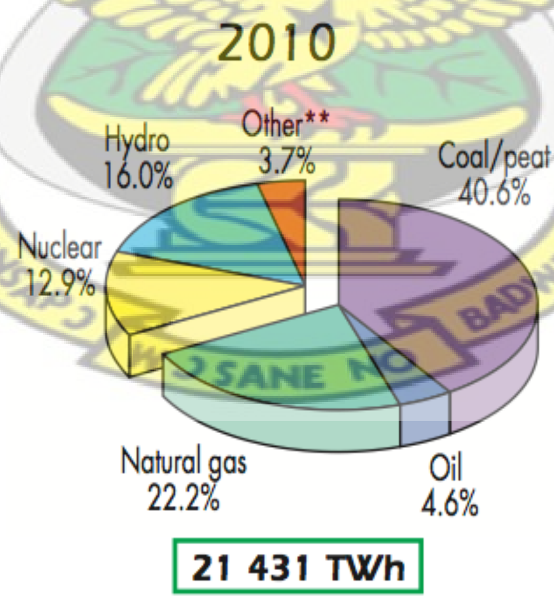

Figure 1.2 Fuel share of electricity generation, 2010. Source (International Energy Agency, 2012)

Apart from hydropower generation, electricity generation from other renewable energy sources has been very insignificant, a total of about 3.7 percent, which includes solar, wind, geothermal and biofuels (Figure 1.2).

The over dependence on non-renewable energy resources especially fossil fuels comes with very hard challenges, a classical example is global warming, which is caused by the emissions of green house gasses (carbon dioxide and methane gas) from the combustion reaction of fossil fuels. Coal, which is largely used for producing power, is the highest emitter of carbon dioxide among the fossil fuels. Figure 1.3 shows fuel share of carbon dioxide emission from 1971 to 2010.

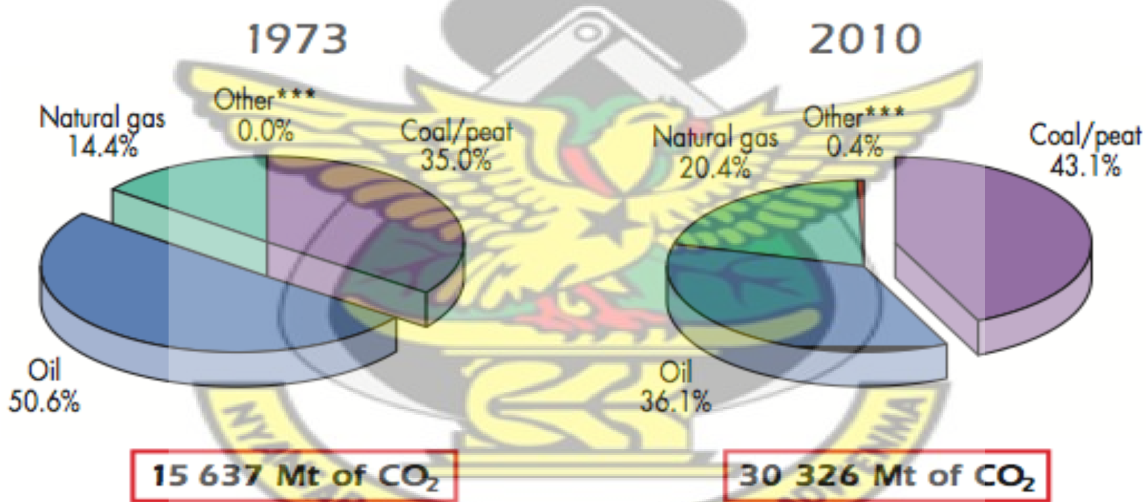

Figure 1.3: Fuel share of carbon dioxide emission from 1971 to 2010. Source (International Energy Agency, 2012)

There is an intensifying natural climate change owing to the emission of carbon dioxides. In 2007, the Intergovernmental Panel on Climate Change (IPCC) predicted the global temperature in figure 1.4.

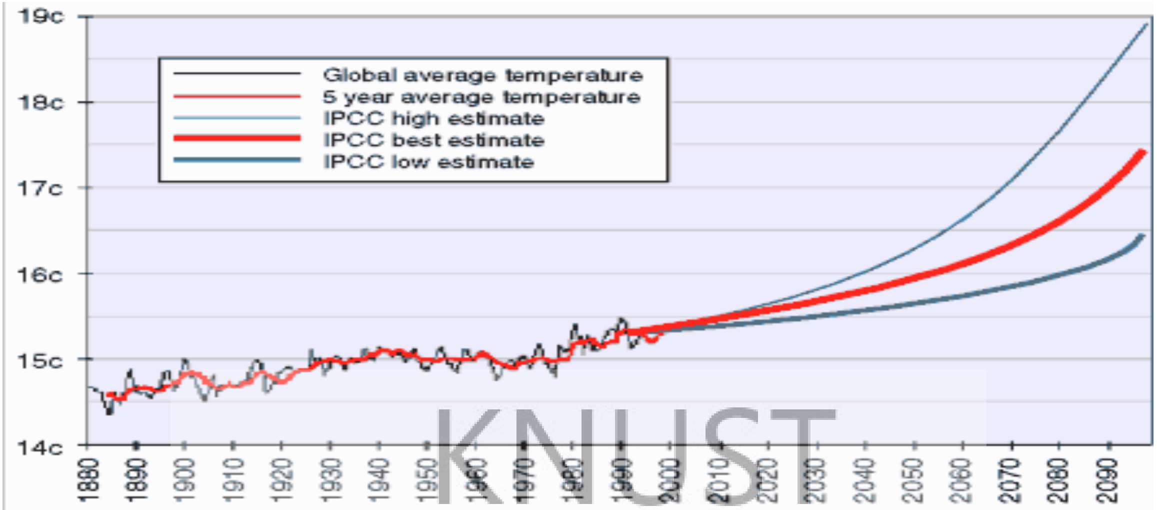

Figure 1.4: Current global temperatures and future predictions, Source: IPCC 2007

The conclusion that can be drawn from the prediction is that, in the few decades to come if the measures to reduce carbon dioxides emissions are not well strengthened a more dangerous climatic change is feared to happen. The call therefore to shift our energy preferences to a more sustainable and environmentally benign sources especially in the power generating sector is cannot be over emphasized.

All over the globe alternative energy sources are now being considered especially in the power-generating sector. Among the renewable energy sources (solar, wind, geothermal and biofuel) solar and wind energy are found to be attractive and are rapidly growing in terms of power generation. The solar chimney power plant, which is being considered in this thesis combines both solar and wind energy resources to generate electricity.

The solar chimney power plant is such that, it has the ability to produce hot air under the collector roof by green house effect, the heated air by buoyancy rises through a tall structure place at the middle of the collector known as the solar chimney. The hot air as it rises drives a set of air turbine, which is placed inside the chimney to generate electrical power. A typical illustration of solar chimney power configuration is seen in Figure 1.5

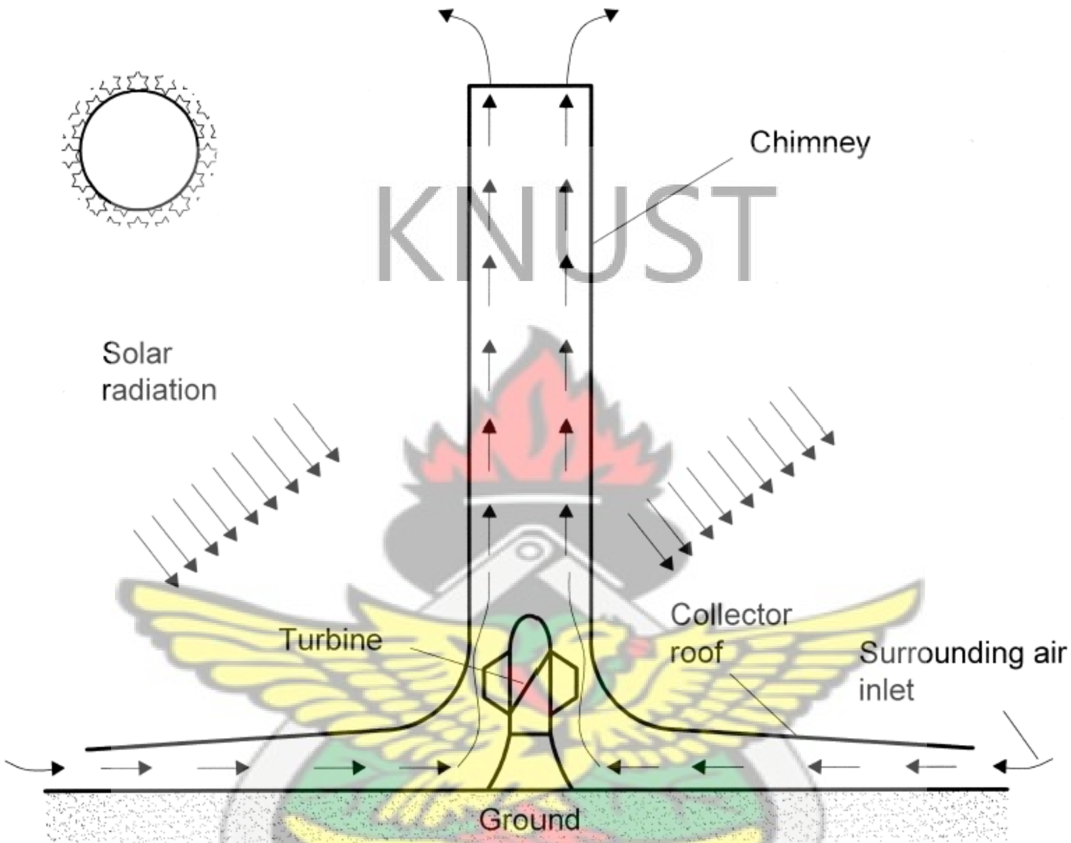

Figure 1. 5: A solar chimney power plant

1.2 Statement of the problem

Ghana as a developing country is faced with lot of challenges. One of the most pressing challenges lies in the power sector. The bulk of Ghana's energy supply is generated from hydro generating resources, Akosombo, Kpong and Bui with a total installed capacity of 1310 MW. The rest are thermal power generation with total installed capacity of 1,168 MW. See Table 1.1.

Table 1.1: Grid Power generation Capacity available for 2013. Source. (Energy commission, April, 2013)

GENERATION PLANT	CAPACITY (MW)		Plant Availability Factor	Expected Energy (GWh)
	Installed	Dependable		
Hydro Power Plants				
Akosombo	1,020	960	0.90	7,568.64
Kpong	160	140	0.90	1,103.76
Bui	130	100	0.90	788.4
Sub-Total	*1,310*	*1,200*		*9,460.80*
Thermal Power Plants[20]				
TAPCO (CC)	330	300	0.70	1,839.6
TICO (SC)	220	200	0.80	1,401.6
Sunon – Asogli (gas)	200	180	0.68	1,072.224
Tema Thermal Plant – TT1PP	110	100	0.85	744.6
Tema Thermal Plant – TT2PP	50	45	0.85	335.07
Takoradi 3 (T3)	132	120	0.50	525.6
Mines Reserve Plant (MRP)	0	35	0.75	229.95
CENIT Energy Ltd	126	120	0.70	735.84
Sub-Total	*1,168*	*1,065*		*6,654.53*
Solar Power Plants				
VRA Solar grid-inter-tied	2	2	0.3	5.26
Sub – Total	*2*	*2*		*5.26*
Total	**2,480**	**2,267**		**≈16,121**

The major setback in hydropower generation since 1982 has been periodic hydrological shocks, which arise as a result of the uncertainty in the rainfall pattern and water inflow. The thermal power generation also suffers the challenge of fuel supply, a situation that lead into the investment in the West Africa Gas Pipeline project with the aim to supply cheap natural gas from Nigeria to power the thermal plants; Takoradi thermal plant, Tema thermal plant and particularly the Sunon-Asogli 200 MW thermal plant which runs solely on gas. However this attempt has also not been smooth.

The demand for electrical power in Ghana keeps rising (See Table 1.2) and is fast outpacing supply, this is so evidenced in the increased frequency in power outages, and deliberate load shedding to ensure that the available power is equitably distributed.

Table 1.2: Grid electricity supply, share and growth to the demand sectors since 2000. Source (Energy commission, April, 2013)

Year	Demand sectors										
	Industry			Non-residential			Residential			Total	
	1000 GWh	% share	%Gr	1000 GWh	% share	%Gr	1000 GWh	% share	%Gr	1000 GWh	%Gr
2000	4.31	68.0	0	0.55	8.7	0	1.49	23.5	0	6.34	0
2001	4.33	66.4	0.5	0.58	8.7	5.5	1.61	24.7	8.1	6.53	3.0
2002	3.90	63.2	-9.9	0.60	9.8	3.4	1.67	27.1	3.7	6.17	-5.5
2003	2.21	48.6	-43.3	0.62	13.6	3.3	1.73	38.0	3.6	4.55	-26.3
2004	2.03	44.8	-8.1	0.66	14.6	6.5	1.78	39.3	2.9	4.53	-0.4
2005	2.54	49.2	25.1	0.70	13.6	6.1	1.92	37.2	7.5	5.16	13.9
2006	3.59	55.1	41.3	0.79	12.1	12.9	2.13	32.7	10.9	6.51	26.2
2007	2.70	48.3	-25.0	0.80	14.3	1.3	2.10	36.6	-1.4	5.59	-14.1
2008	2.97	48.2	10.0	0.93	15.1	16.3	2.27	36.9	8.1	6.16	10.2
2009	2.94	47.2	-1.0	0.88	14.1	-5.4	2.41	38.7	6.2	6.23	1.1
2010	3.16	46.1	7.5	0.97	14.1	10.2	2.74	39.9	13.7	6.86	10.1
2011	3.90	48.9	23.4	1.31	16.4	36.1	2.76	34.6	0.7	7.98	16.3
2012	4.20	51.9	7.7	1.30	16.0	-0.8	2.60	32.1	-5.8	8.10	1.5
Mean growth			2.2			7.3			4.5		2.8
Note: Gr is growth rate											

In 2006, the Energy Commission projected under the Strategic National Energy plan (SNEP) report that, in order for Ghana to ensure secured and uninterrupted electricity supply by the year 2020, the existing installed capacity of 1760 MW must be doubled. Fortunately, Ghana happens to have a very good solar radiation simply because it is located in the continent that receives the highest amount of solar radiation between 300 and 350 W/m^2 annually (Brew-Hammond *et al,* 2008). The monthly average solar radiation across the country ranges between 4.4 kWh/m^2-day to 5.6 kWh/m^2-day with the average sunshine hours also in the range between 5.3 hrs. in Kumasi to 7.3 hrs. in Wa.

The solar radiation if harnessed and well utilized, could minimize the present energy crisis prevailing in the country and also enhance the process of rural electrification.

The solar chimney power plant operates using solar and wind energy therefore it stands as a way to boost the current electricity production. Solar and wind energy being renewable sources will also provide energy without pollutant and green house gas emission. This can go a long way to mitigate the adverse effect of global warming and contribute to sustainable energy development.

1.3 Objectives

The main aim of this thesis is to analyse the design of a solar chimney power plant for sustainable power generation. The specific objectives are to

i. Develop a mathematical model for analysing the design of the solar chimney power plant.

ii. Carry out a solar resource calculation on an inclined collector surface for a worse case scenario and determine the useful energy required or input.

iii. Carry out calculations for a reasonable chimney height and determine the total pressure and air velocities resulting from the chimney effect and finally

iv. Estimate the theoretical electrical power output and the overall efficiency based on the plant design model specifications.

1.4 Research questions

i. What are the technologies available in harnessing solar energy to produce power?

ii. How do solar chimney power plant (SCPP) works and in what areas are they practically feasible?

iii. What factors affect or influence the power output of a solar chimney power plant?

iv. Which kind of turbine is used in solar chimney power plant?

v. What kind of materials are employed for the solar chimney power plant construction

1.4 Methodology

The mathematical model will be used as the main tool to analyse the plant design and its performance. A mathematical model for the solar collector and the solar radiation collection will be presented. Based on the model, the useful energy and the power required by the airflow mass will be determine. Other mathematical models that will be presented will include the solar chimney and the air turbine. Mathematical equations will be derived to theoretically predict the total airflow pressure to be developed and the maximum air velocity through the chimney. The electrical power output of the wind turbine and the overall plant efficiency will then be estimated.

1.5 Limitations

The major limitation to the thesis is that, the analysis is performed based on the weather conditions in Takoradi. Another limitation is that, the analysis of the SCPP does not cover the structural and economic viability of the power produced by the plant as compared to other conventional power generating technologies such as gas or coal fired thermal plant.

1.6 Thesis organisation

The whole thesis is structured into five major chapters, chapter one is the introduction. Chapter two is the review of past literature, chapter three presents the detailed mathematical model upon which mathematical equations are derived to analyse the design and performance of the SCPP. Chapter four is the design calculations and analysis. The last chapter, named chapter five provide the final conclusions and recommendations for future work

CHAPTER 2

2.0 LITERATURE REVIEW

2.1 Introduction

Converting solar energy into electricity is possible by either using a solar photovoltaic device or by converting the solar energy into thermal energy and subsequently converting the thermal energy into electric power. The later technology is known as solar thermal technology.

A solar thermal technology can either be

i. A concentrating technology or

ii. A non-concentrating technology.

The concentrating technologies employ powerful collectors that are able to generate high temperature thermal energy to drive either steam turbines or gas turbines to produce power. The parabolic trough, the dish technology, and the heliostat are examples of concentrating power technologies simply put as CSPs. The non-concentrating technology on the other hand employs collectors that are unable to concentrate the solar radiations i.e. they capture both the direct and diffuse solar irradiation. Invariably they are unable to attain very high temperatures. The solar chimney power technology to be investigated in this literature review falls under the non-concentrating power technologies.

2.2 The concept of solar chimney power plant (SCPP)

The solar chimney power plant generates power based on two basic principles, green-house effect and buoyancy- driven flow. Solar irradiation from the sun passes through the glass of the collector, is absorbed by the ground below. By greenhouse effect the air under the collector gets heated. The high-temperature, lower density air then rises toward the chimney as a result of buoyancy. By this principle (buoyancy), a pressure difference is created in the column of the chimney, which actually drives the air from the base of the chimney to its upper outlet. A wind turbine placed in the chimney and geared to an appropriate generator, captures the kinetic energy of the moving air and produce power.

2.3 History of solar chimney power

1903: As with any other inventions, Isodoro Cabanyes a Spanish engineer, brought up the idea of generating electricity from solar chimney. This concept is illustrated in Figure 2.1 (Omri, 2013)

Figure 2.1 Solar chimney proposed by Isodoro Cabanyes

1926: Prof Engineer Bernard Dubos proposed to the French Academy of Sciences the construction of a Solar Aero-Electric Power Plant in North Africa with its solar chimney on the slope of the high height mountain after observing several sand whirls in the southern Sahara (Figure 2.2). (Hamilton, 2011)

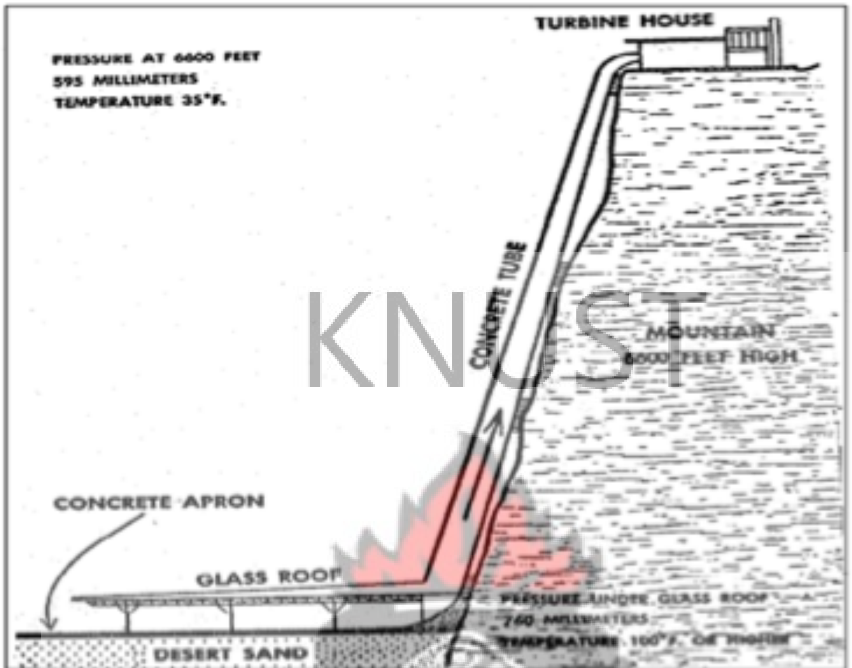

Figure 2.2 Solar chimney proposed by Bernard Dubos (Ley, 1954)

He anticipated an ascending air velocity of 50 m/sec could drive a wind turbine to generate power.

1982 Schaich, Bergerman and Partners, under the direction of Prof. Dr. Ing. Jorg Schlaigh built an operating model of a solar chimney in 1982 in Manzanares (Spain). The plant specifications were 46000 m^2 area of a green house made of glass, 195 m tall solar chimney with 10 m internal diameter and the rated power was 50 kW. This first prototype demonstrated the technical feasibility of generating power from the solar chimney power plant. (Schlaich, 1995)

TODAY: Based on the Manzanares prototype a lot of research works is being carried out which involves the construction of experimental prototypes to investigate the potential of solar chimneys power all over the world. A few of those prototypes will be outlined.

- **In 2001**, a company called Enviro Mission initiated a plan to build a 200 megawatts solar chimney in southwest Australia that could generate 4000 times more power than the Manzanares system (Figure 2.3).

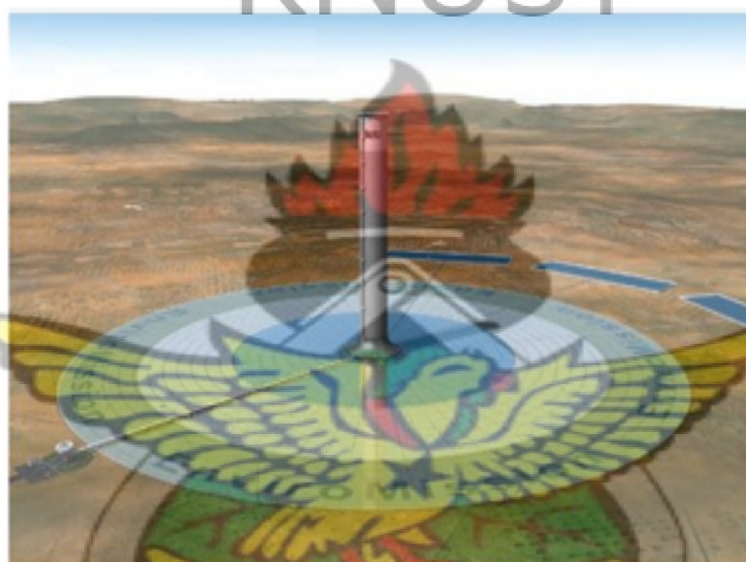

Figure 2.3 Tapered solar collector by Enviro Mission

- **2005** Botswana's Ministry of Science and Technology designed and built a small-scale solar chimney system for research. The chimney height was 22 m tall, with a 2 m internal diameter. The collection base area was approximately 160 m^2

  The rational behind this project was due to the long-term strategic plans for energy in Botswana.

- **2008,** the Namibian government also approved a proposal for the construction of a 400 MW solar chimney called the 'Greentower'. The tower was planned to be 1.5 km tall and 280 m in internal diameter, and the collector base area 37 km^2 .

- **2009-2013** The construction of the Chinese prototype in Jinsha Bay Wuhai in Inner Mongolia. The project was completed in three phases. The first phase generated a total capacity of 200 kW and was completed in December 2010. Second phase started February 2011 completed in December 2011. A total capacity of 2.2 MW of power was generated. The third phase, began in January 2012, and completed in December 2013. The plant rated total power was 27.5 MW. The collector occupied a total area of about 2.51 million meter square. (Omri, 2013)

2.4 Solar chimney plant configuration

The solar chimney plant consist of three key components

i. The collector
ii. The chimney
iii. The wind turbine

The three will be discussed in brief

2.4.1 The collector

The collector is also termed as the greenhouse, it is a special kind of heat exchanger that transforms solar radiation into thermal energy. The collector provides the main natural source of heat to the plant. Unlike CSP collectors, which absorb only the direct normal solar irradiation (DNI), the collector for the solar chimney plant makes use of both the direct and the diffuse solar radiations to generate thermal energy.

The material used for the collector roof is either plastic or glass. The covering or what is termed as the glazing admits the short wave solar radiation component and retains long-wave radiation from the heated ground, in that way the air beneath gets heated. The collector comes in various configuration based on the materials used for its roof (Figure 2.4). It could be circular, or rectangular in shape.

Figure 2.4 Collector design options

Tapered and sloppy collectors have also been proposed. The reason behind this design is that the angle of inclination will aid in providing sufficient and effective area of the collector to receive more solar radiation, which would improve solar collector efficiency and improving the collector efficiency end up in the increase of the amount of useful energy needed to heat the air.

2.4.2 Chimney

The chimney is the main characteristic of the solar chimney station. The chimney is located at the center of the collector and is the thermal engine for the plant. The chimney creates a temperature differential between the cool air at the top and the heated air at the bottom. The change in the air temperature induces a pressure differential, which drives the air from the bottom of the chimney out of the top.

Several factors contribute to the physical design of the chimney. A well designed solar chimney must minimize frictional losses and maximize the pressure difference in the chimney. The pressure difference in the chimney is proportional to the chimney height, given by the relation.

$$\Delta P_{sc} = (\rho_o - \rho_2) g H_{sc} \tag{2.1}$$

therefore maximizing the height of the chimney is very essential in improving the efficiency. Schlaich (1995) stated that, the efficiency of the chimney is given by

$$\eta_{sc} = \frac{P_{tot}}{\dot{Q}} = \frac{g H_{sc}}{C_p T_f} \tag{2.2}$$

The above relation equation (2.2), explains the significance of the chimney's height on the efficiency. The chimney efficiency is again seen to be inversely proportional to the outside air temperature, therefore the lower the air temperature the better the efficiency. This explains why the solar chimney plant is yet able to a produce substantial amount of electrical power during the night and overcast weather conditions. This is made possible as the plant is able to make good use of the low rise in air temperature produced by the heat emitted by the ground.

The material for the chimney construction could be; reinforced concrete tubes, steel sheet tubes supported by guy wires, or cable-net construction with a cladding of sheet metal or membranes (Figure 2.5). Schlaich (1995) suggests that reinforced concrete would be a cost effective way to create a stable tower with a lifespan of up to 100 years.

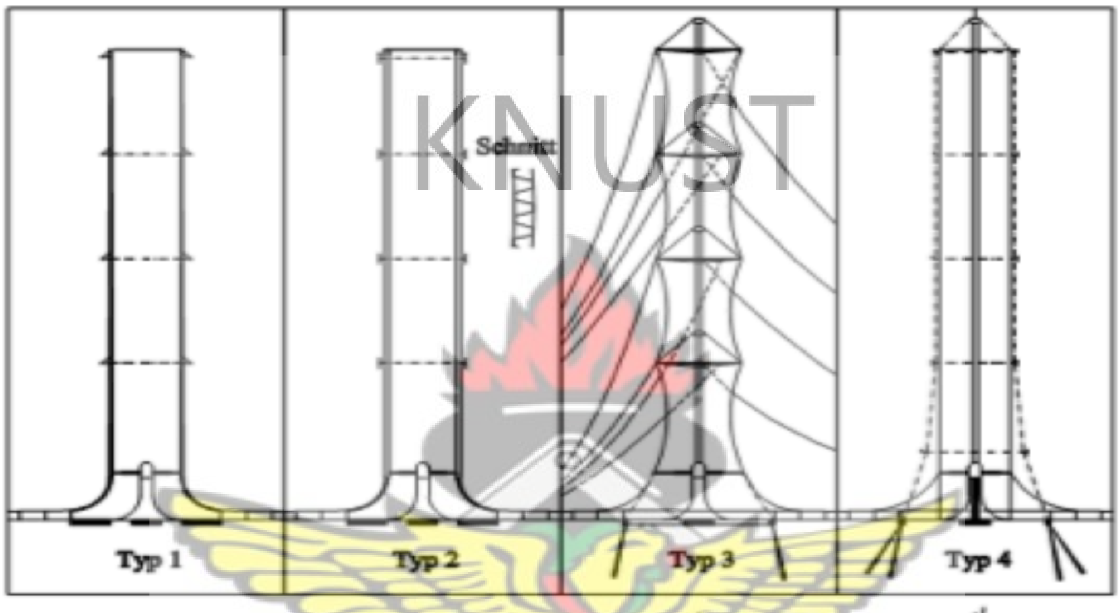

Figure 2.5: chimney construction shapes

2.4.3 Turbine

The SCPP uses a turbine or array of turbines to generate power. The turbine or turbines operate as cased pressure-staged generators, similar to a hydroelectric plant. The turbines employed are basically wind turbines only that they are subjected to relatively steady airflow compared to those of wind generator plants. They are also subjected to less physical stress but they are to cope with high temperatures. The wind turbines used in SCPP are ducted and so their maximum achievable theoretical total efficiency can go beyond 59 % as stipulated by Betz limit in those of wind generators. (Pastohr, 2003).

There are various wind turbine layouts and configurations for solar chimneys power conversion unit (PCU). (See Figure 2.6). A few of those configurations are examined;

i. Single rotor turbine without inlet guide vanes (IGVs): With only one blade row the single rotor turbine without IGVs is the simplest layout. Its biggest disadvantage is that the swirl induced by the rotor cannot be recovered. (Schwarz, (1981).)

ii. Single rotor turbine with IGVs: This layout is a single rotor axial flow turbine stage with inlet guide vanes. The swirl is induced by guide vanes, which are located upstream of the rotor. The rotor turns the flow back to a close to axial direction.

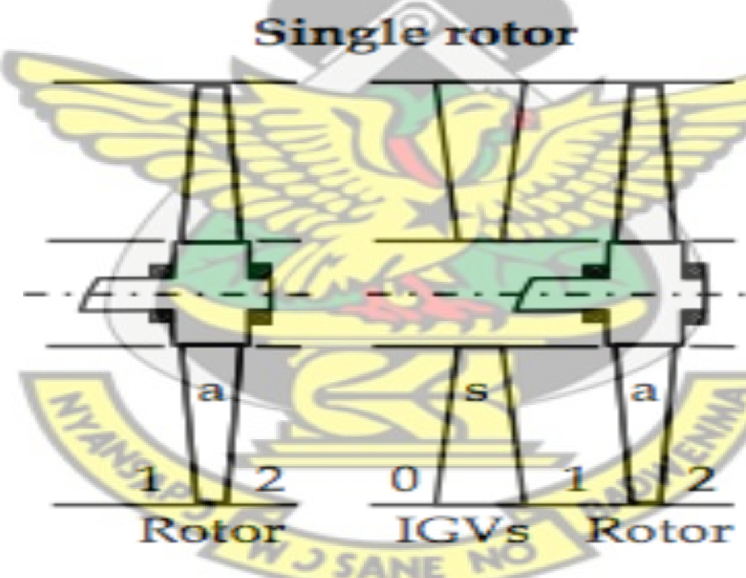

Figure 2.6: Schematic drawings of turbine. Source; Fluri, (2008).

The single rotor layout without IGVs is found to be the simplest and cheapest layout, as it requires comparably few blades and a small drive train. The pilot plant in Manzanares employed this turbine layout design (Schlaich, 1995).

2.5 Energy storage

The solar chimney plant like any other solar thermal power technology has a means of storing heat energy to enable it function or work even after sunset. The ground under the collector roof has been found to serve as or behave as a storage medium. The ground soil can heat up the air for a significant time after sunset depending on its nature. Several research works have been conducted to analyze the effect of the ground soil type on the power output. Pretorius, (2004) investigated six different ground types and they include: sandstone, granite, limestone, and sand, wet soil and water. They concluded that the SCPPs employing the wet soil and the sand have the lowest and highest power outputs respectively, and the different materials lead to varying power outputs during the daytime and at night.

Another concept of thermal storage that has attracted several research work is the use of black tubes filled with water and laid on a black sheet under the solar collector. Here the idea is that since the heat capacity of water (4.2 kJ/kg) is higher than that of the soil (0.75 - 0.85 kJ/kg) the water inside the tubes stores a part of the solar heat and releases it during the night, when the air in the collector cools down.

Kreetz (1997) examined the effect of the ground water storage on the power and his analysis showed the possibility of a continuous day and night operation of the solar chimney. See Figure 2.7

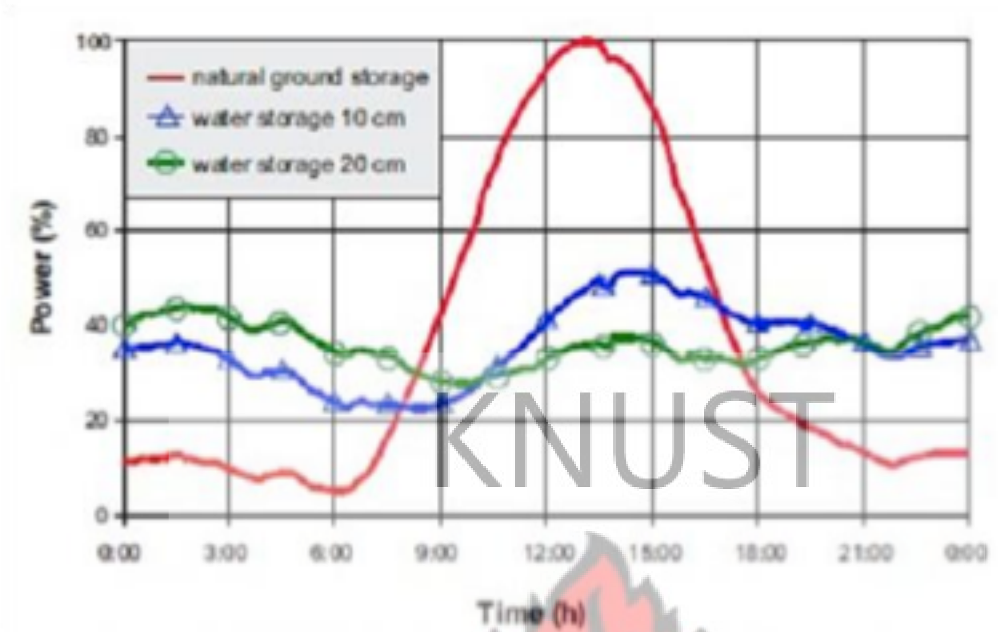

Figure 2.7 Effect of heat storage underneath the collector roof using water-filled black tubes. Simulation results from (Kreetz, 1997)

Hammadi, (2008) also developed a mathematical model for a solar updraft tower with water storage system and stated that the thickness of the water storage layer under the collector also influences peak power output generated by the plant.

2.6. The floating solar chimney power plant (FSC).

Papageorgiou, (2006), presented another concept of the solar chimney known as the floating solar chimney power plant (FSC). The researcher proposed a flexible solar chimney instead of a rigid concrete tower. The floating solar chimney power plant will have it chimney made with a flexible material and will float on air with the help of a lighter gas like helium (see Figure 2.8). The chimney essentially has a heavy base and the walls are filled with a lighter gas. The support rings allow air to enter and pass through them freely, so that the chimney does not yield under wind pressure. (See Figure 2.9).

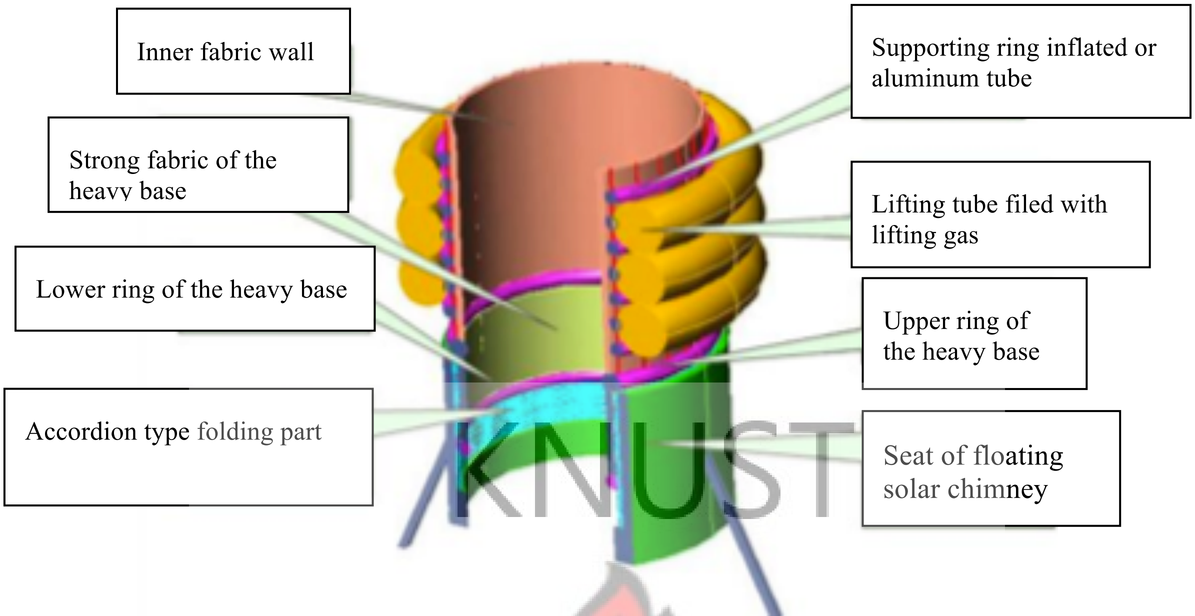

Figure 2.8: FSC proposed by Papageorgiou (2006)

Figure 2.9: FSC in operation

The major advantage behind this concept is that it seeks to replace conventional concrete solar chimneys due to it lower construction cost as compared to that of concrete. But basically the FSC comprises of a collector, the floating chimney and a set of air turbines geared to an appropriate electric generator place inside or around the chimney just as in other SCPP plants.

2.7 Characteristics of the solar chimney power plant

The solar chimney power plant has certain features or characteristics that make it attractive for power generation. They include the following:

i. Clean technology using renewable solar energy as a heat source which produces neither greenhouse effect gasses nor hazardous wastes.
ii. Efficient use of solar radiation, the solar collector utilizes both the direct and diffuse solar radiation. The plant therefore is able to generate power under cloudy conditions although reduced.
iii. The soil under the collector acts as heat storage, avoiding sharp fluctuations and allowing power supply after sunset.
iv. The Plant has a low maintenance cost as compared to other conventional power technologies (Schlaich, *et al.,* 2005). This is considered a key advantage especially in areas with enough sunshine.
v. The SCPP power plant does not require any cooling water to operate. This makes them advantageous in areas where water supply is a challenge.

Nevertheless the SCPP also have certain features or disadvantages that make them less suitable for some sites.

i. They require large areas of flat land. This land should be available at low cost, which means that there should be no competing usage, like e.g. intensive agriculture, for the land.

ii. Zones with frequent sand storms should also be avoided, as either collector performance losses or collector operation and maintenance costs would be substantial there.

CHAPTER 3

3.0 MATHEMATICAL MODEL

3.1 Introduction

The analysis of a solar chimney power plant depends mainly on the following parameters, which influence the power output

- The ambient conditions, which are represented by the solar insolation, ambient temperature and the wind velocity.
- The solar chimney plant configuration, which is represented by the dimension of the chimney and the collector: the chimney height, the chimney diameter, the collector radius and the collector's periphery height. (See Figure 3.1)

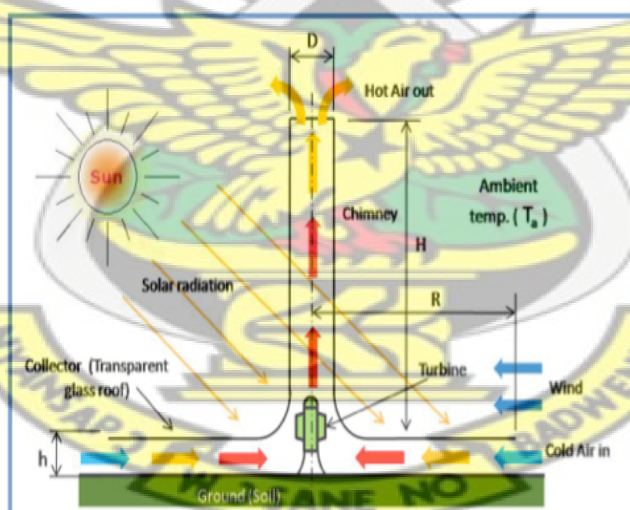

Figure 3.1 Solar Tower Schematic drawing showing the ambient condition and tower configuration Bilgen, E. and Rhealt, J. (2005)

Other factors that influence the power output are the materials used for the collector and chimney construction, the soil or rock content under the collector.

A more detailed analysis of the design and performance of solar chimney plants are often done by using Computational Fluid Dynamics (CFD). CFD software has now been developed, and by using the software, it is possible to simulate a prototype for studies without necessarily building real and expensive prototypes or models.

Another method which is valid and used to analysis the design and performance of the solar chimney plant is the mathematical model. In this thesis, a mathematical model will be developed based on energy balance to analyse the plant design and performance. The elements to be analysed will include; solar radiation collection, useful energy developed, the chimney airflow speed, and total pressure developed as a result of buoyancy and then the electrical power output of the turbine and the overall plant efficiency. The following are the assumptions under which the model is developed

1. Performance of the power plant is analyzed at steady state.
2. Air is an ideal gas
3. No friction or leakage in the system
4. Airflow in the system is due to buoyancy force
5. Heat loss from the collector is purely by convection and radiation.
6. The flow in the collector is considered as a flow between two parallel plates (Bernardes and Weinrebe, 2003)

3.2 The solar collector

According to Duffie and Beckman (1991), the energy balance equation for the solar collector is given as,

$$\dot{Q}_u = \dot{Q}_{opt} - \dot{Q}_{loss} \ldots\ldots\ldots\ldots \text{General equation} \tag{3.1}$$

$$\dot{Q}_u = A_c F_R\left[\overline{S_1} - U_L\left(T_g - T_a\right)\right] \tag{3.2}$$

$$\dot{Q}_u = A_c F_R\left[(\overline{\tau\alpha})\overline{H}_T - U_L\left(T_g - T_a\right)\right] = \eta_c A_c \overline{H}_T \tag{3.3}$$

The efficiency of the collector/optical efficiency is therefore given as

$$\eta_c = \frac{\dot{Q}_u}{A_c \overline{H}_T} = (\overline{\tau\alpha}) - \frac{U_L\left(T_g - T_a\right)}{\overline{H}_T} \tag{3.4}$$

3.3 Thermal losses

From Duffie and Beckman, (1991) equation (3.02), it follows that not all the solar radiation that incidents on the collector eventually turn out to be useful energy. As the collector absorbs heat its temperature gets higher than that of the surrounding and so heat is lost to the atmosphere by convection and radiation.

The rate of heat loss $\dot{Q}_{loss}$ depends on the collector overall heat transfer coefficient U_L and the collector/ absorber temperature.

$$\dot{Q}_{loss} = U_L A_c (T_g - T_a) \tag{3.5}$$

3.3.1 Collector overall heat loss coefficient U_L

The overall heat transfer coefficient U_L is the effective combination of all losses out of the collector. It accounts for all heat transfer modes from all surfaces of the collector assemblies. (See Figure 3.2a). A typical way to break this down is to consider the losses from the top glass plate, bottom surface, and edges separately. This gives the following.

$$U_L = U_t + U_g + U_e \tag{3.6}$$

The solar collector has the ground soil which serves as an absorber and the edges are freely opened to allow or admit fresh air into the collector. It is assumed that heat loss through the bottom and edges are negligible and ignored, therefore the top glass cover will only contribute to the heat loss. The top loss is therefore contributed by convection and radiation between the ground soil and the glass cover creating a thermal resistance (R_1), conduction through the cover and convection and radiation heat from the cover to the ambient creating another thermal resistance (R_2).

In a more detail analysis, there is a convective heat exchange between the cover and the airflow inside the collector h_{cf}, and between the airflow and the ground h_{gf} but these two convective heat exchanges are considered equal and put together as convection between ground soil and the cover $h_{conv\ g-c}$. Figure 3.2b (Chergui, *et.al.,* 2011).

The resistance from conduction through the cover is also considered as negligible and ignored. This then leaves the model with only two resistances in series. The overall heat loss coefficient U_L is therefore equal to

$$U_L = U_t = \frac{1}{R_1 + R_2} \tag{3.7}$$

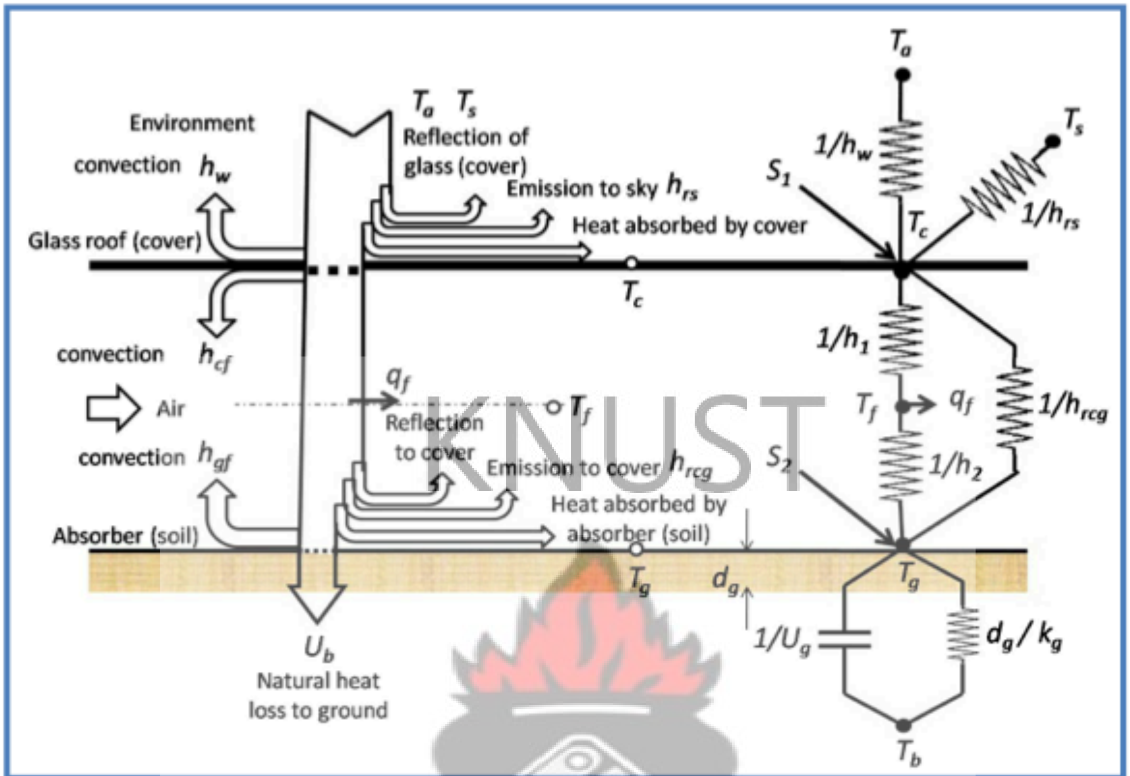

Figure 3.2a: Detailed heat transfer mechanisms through a collector with a single cover. Source Ali, (2013).

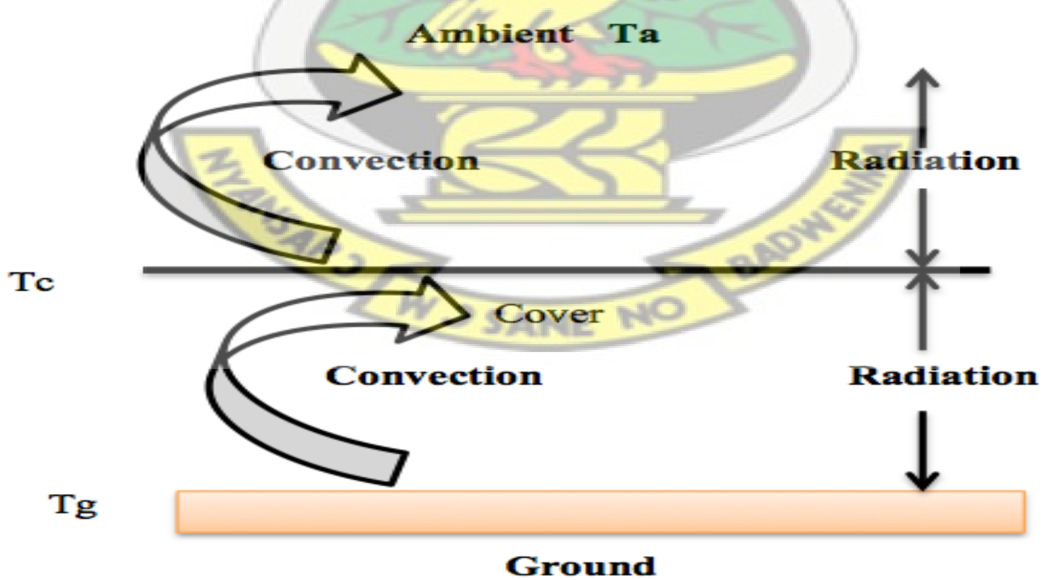

Figure 3.2b: Simplified heat transfer mechanisms through a collector with a single cover

3.4 The thermal resistances

3.4.1 Ground to Cover/collector, resistance R_1

The thermal resistance between the ground and glass cover is predominantly by natural convection inside the collector and radiation between the two surfaces. (Figure 3.3). The total resistance is therefore represented by the following expression

$$R_1 = \frac{1}{h_{conv\,g-c} + h_{rcg}} \tag{3.8}$$

The radiation coefficient from the ground soil to the cover (h_{rcg}) is given **by** Bilgen and Rheault, (2005) as

$$h_{rcg} = \frac{\sigma\left({T_g}^2 + {T_c}^2\right)\left(T_g + T_c\right)}{\frac{1}{\varepsilon_g} + \frac{1}{\varepsilon_c} - 1} \tag{3.9}$$

The convective heat exchange between the cover and the air flow and between the airflow and the ground are considered equal, and is determined from the following equation Chergui, *et al* (2011)

$$h_{conv,g-c} = \frac{N_u \times K_{air}}{d_h} \tag{3.10}$$

Where K_{air} is the thermal conductivity of air and d_h is the hydraulic diameter or characteristic length given as

$$d_h = \frac{4\times\, Collector\ Area}{Collector, Perimeter} = \frac{4A_c}{P} \tag{3.11}$$

The collector is circular with radius R_c and height H_c therefore the hydraulic diameter by (Chergui, *et al*. 2011) is:

$$d_h = 2H_c \tag{3.12}$$

The Nusselt Number for natural convection over surfaces from empirical correlations is given as (Chergui, *et al* 2011)

$$N_u = 0.54Ra^{0.25} \text{ For } 2\times10^4 < Ra < 8\times10^7 \tag{3.13}$$

$$N_u = 0.15Ra^{0.33} \text{ for } 8\times10^7 < Ra < 8\times10^{11} \tag{3.14}$$

Where the Prandtl number is given as

$$Pr = \frac{\upsilon}{\alpha} \tag{3.15}$$

The Raleigh's number given by Bahrami, (2011) is

$$Ra = \frac{g\beta'(T_g - T_{f,})h^3{}_{gap}}{\upsilon\alpha} = \frac{g\beta'(T_g - T_f)h^3{}_{gap}Pr}{\upsilon^2}$$

$$Gr = \frac{g\beta'(T_g - T_f)h^3{}_{gap}}{\upsilon^2} \tag{3.16}$$

$$Ra = GrPr \tag{3.17}$$

β' Is the coefficient of volumetric thermal expansion, it is given according to Bilgen and Rheault, (2005) as:

$$\beta' = \frac{1}{0.25T_a + 0.75T_f} \tag{3.18}$$

3.4.2 Cover to ambient air/sky, total thermal resistance R_2

The thermal resistance between the glass cover and the ambient air/sky is predominantly by forced convection from the wind and radiation exchange with the sky.

$$R_2 = \frac{1}{h_w + h_{rs}} \tag{3.19}$$

The radiation coefficient from the cover to the ambient air / sky is given by Bernardes and Weinrebe, (2003)

$$h_{rs} = \frac{\mathcal{E}_c \sigma (T_c + T_s)\left(T_c{}^2 + T_s{}^2\right)(T_c - T_s)}{(T_c - T_a)} \tag{3.20}$$

The wind convection loss coefficient is given by Annaratone, (2010):

$$h_w = 5.7 + 3.8V_{wind} \tag{3.21}$$

Where V_{wind} is the average wind speed at the location.

The average sky temperature is given as, Xinping *et al.,* (2006):

$$T_s = 0.0552T_a{}^{3/2} \tag{3.22}$$

3.5 The solar chimney

The chimney is the plant's actual thermal engine. It is a pressure tube with low friction loss because of its optimal surface volume ratio. The main purpose of the chimney is to convert the thermal energy in the collector into kinetic energy.

The pressure developed in the chimney from the inlet to the exit from equation (2.1) is given as

$$\Delta P_{sc} = (\rho_o - \rho_2) g H_{sc}$$

It is assumed that the air density variation is linear between the entrance and exit.

The total pressure developed due to buoyancy (equivalent to the power contained in the flow) is given by (Bilgen, 2005) as

$$\Delta P_{tOt} = \rho_{air} g \left(H_{sc} + {}^{H_c}/_2 \right) \tag{3.23}$$

The total pressure difference created is spent, in part, for friction, (ΔP_f) losses in the collector and the chimney, kinetic energy losses at the chimney exit, ΔP_{KE} and the rest is used by the turbine, ΔP_t. Thus,

$$\Delta P_{tOt} = \Delta P_t + \Delta P_f + \Delta P_{KE} \tag{3.24}$$

The kinetic energy losses at the chimney exit is given as,

$$\Delta P_{KE} = \frac{1}{2} \rho_{air} V_{sc}{}^2 \tag{3.25}$$

The friction losses in the collector and the chimney, ΔP_f is given by the Darcy weisbach equation:

$$\Delta P_f = \frac{4fL}{D} \left(\frac{V^2}{2g} \right) \tag{3.26}$$

Assuming the pressure loss due to friction is negligible then:

$$\Delta P_{tOt} = \Delta P_t + \Delta P_{KE}$$

$$\Delta P_{tOt} = \Delta P_t + \frac{1}{2}\rho_{air}{V_{sc}}^2 \qquad (3.27)$$

Equation (3.27) shows that the total pressure is a combination of the static pressure drop at the turbine and a dynamic pressure, which is converted to kinetic energy.

In the absence of a wind turbine, a maximum airflow speed is achieved and the whole pressure difference is used to accelerate the air and converted into kinetic energy (Schlaich, *et al* 2005). The maximum chimney airflow rate and the maximum air velocity is therefore expressed as (Zhou, *et al* 2008):

$$\Delta P_{tOt} = \frac{1}{2}\rho_{air}{V_{sc}}^2$$

$$V_{max} = \sqrt{\frac{2\Delta P_{tOt}}{\rho_{air}}} \qquad (3.28)$$

Zhou, *et al* (2008) stated again that, the maximum power is drawn when the chimney airflow rate velocity V_{sc} is one third of the maximum speed V_{max} in the case of turbine being on load. Combining equations (25) and (30), the velocity in the chimney V_{sc} can therefore be expressed as:

$$V_{sc} = \frac{1}{3}\sqrt{\frac{2\Delta P_{tOt}}{\rho_{air}}} = \frac{1}{3}\sqrt{\frac{2\rho_{air}g\left(H_{sc} + {}^{H_c}/_2\right)}{\rho_{air}}} = \frac{1}{3}\sqrt{2g\left(H_{sc} + {}^{H_c}/_2\right)} \qquad (3.29)$$

The chimney efficiency from equation (2.02) is then expressed as, (Schlaich, 1995)

$$\eta_{sc} = \frac{P_{tot}}{\dot{Q}} = \frac{gH_{sc}}{C_pT_f}$$

3.6 Turbine

Schlaich (1995), stated that the maximum electrical power output that can be generated by using an air turbine is:

$$P_{wt,max} = \eta_{wt}V_{sc}A_{sc}\Delta P_t \text{ Where } \eta_{wt} = \frac{2}{3} = 67\%$$

Therefore the electrical power output is

$$P_e = P_{wt,max} = \frac{2}{3}V_{sc}A_{sc}\Delta P_t \qquad (3.30)$$

3.7 Solar radiation model

The solar energy received by the collector is determined using the monthly average daily total radiation on a horizontal surface, calculation of the monthly average of daily total radiation on a sloped surface (collector) at any given location, for a particular day, follows the well established methods in literature given by Duffie and Beckman, (1991).

The monthly average extraterrestrial radiation on a sloped surface is given as

$$\overline{H}_0 = \frac{243600G_{sc}}{\pi}\left(1 + 0.033cos\frac{360n}{365} \times \left(cos\emptyset cos\delta sin\, w_s + \frac{\pi w_s}{180}\sin\phi\, sin\delta\right)\right) \qquad (3.31)$$

Where the declination

$$\delta = 23.45sin\left(360\frac{284+n}{365}\right) \qquad (3.32)$$

The sunset hour angle

$$w_s = \cos^{-1}(-tan\phi \tan\delta) \qquad (3.33)$$

The monthly average daily solar clearness index is determined as follows,

$$\bar{K}_T = \frac{\bar{H}}{\bar{H}_0} \qquad (3.34)$$

The diffuse component of the monthly average daily radiation is calculated as:

For $\quad 0.3 < \bar{K}_T < 0.8 \; and \; w_s > 81.4^0$

$$\bar{H}_d = \bar{H}\left(1.311 - 3.022\bar{K}_T + 3.427\bar{K}_T^{\;2} - 1.821\bar{K}_T^{\;3}\right) \qquad (3.35)$$

For $\quad 0.3 < \bar{K}_T < 0.8 \; and \; w_s \le 81.4^0$

$$\bar{H}_d = \bar{H}\left(1.391 - 3.560\bar{K}_T + 4.189\bar{K}_T^{\;2} - 2.137\bar{K}_T^{\;3}\right) \qquad (3.36)$$

The monthly average of $\bar{R}_b$ for a surface sloped towards the equator in the northern hemisphere is given as. In the case of Ghana, the surface azimuth angle $\gamma = 0$.

$$\bar{R}_b = \frac{\cos(\phi - \beta)\cos\delta\sin w_s' + \frac{\pi}{180} w_s' \sin(\phi - \beta)\sin\delta}{\cos\phi\cos\delta\sin w_s + \frac{\pi}{180} w_s \sin\phi\sin\delta} \qquad (3.37)$$

With $\quad w_s^{\;1} = min[\cos^{-1}(-tan\phi\; tan\delta), \cos^{-1}(-tan(\phi - \beta)tan\delta)] \qquad (3.38)$

Where ‘min’ means the smaller of the two items in the bracket.

The **monthly average daily total solar radiation** on a south-facing sloped surface or collector **(isotropic sky model)** is calculated as

$$\bar{H}_T = \bar{H}\left(1 - \frac{\bar{H}_d}{\bar{H}}\right)\bar{R}_b + \bar{H}_d\left(\frac{1 + \cos\beta}{2}\right) + \bar{H}\rho_g\left(\frac{1 - \cos\beta}{2}\right) \qquad (3.39)$$

Therefore the **monthly average absorbed solar radiation** by the south facing collector is given as

$$\bar{S}_1 = (\overline{\tau\alpha}) \times \bar{H}_T \qquad (3.40)$$

CHAPTER 4

4.0 DESIGN CALCULATIONS AND RESULT ANALYSIS

4.1. Design input for the model

In other to conduct a numerical analysis and estimate the theoretical power output for the solar chimney plant, the following parameters and specifications are proposed as the input for the model Table 4.1. The geographical location to be considered for the model is Takoradi city, Ghana which is on latitude 4.9° N and longitude 1.8° E. The Monthly average daily solar radiation, $\overline{H}$ experienced on a horizontal is 5.011 kWh/m^2 given by the Energy Commission in its solar and wind energy resource assessment (SWERA) report.

Table 4.1: Preliminary design inputs

Parameter	**Value**
Physical properties of plant	
Chimney height	50 m
Chimney diameter	4 m
Collector height	2 m
Collector diameter	25 m
Angle of inclination of collector	45 °

Cover material	**Glass (single glazing)**
Cover emissivity	0.9
Average transmittance -absorptance product	0.8
Ground type	**sandstone**
Ground emissivity	0.9
Ground absorptivity	0.9
Ground reflectance/ albedo	0.36
Geographical location	**Takoradi**
Takoradi city latitude	4.9° N
Takoradi city longitude	1.8° E
Monthly average daily solar radiation, $\overline{H}$	5.011 kWh/m^2 or 208.79 W/m^2

4.2 Solar resource calculation

Performing the solar resource calculation on an inclined solar collector for the worse case scenario.

- **The declination,** from equation (3.32) is given as

$$\delta = 23.45 sin\left(360\frac{284+n}{365}\right)$$

Where n the average day of the year, Taking July 17th as the average day of the month day, $n = 198$,

$$\delta = 23.45 sin\left(360\frac{284+198}{365}\right) = 21.2\,°$$

- From equation (3.33), **the sunset hour angle:**

$$w_s = \cos^{-1}(-tan\phi \tan\delta)$$

$$w_s = \cos^{-1}[-\tan(4.9) \times \tan(21.2)] = 91.91°$$

- The **monthly average extraterrestrial radiation,** from equation (3.31)

$$\overline{H}_0 = \frac{243600 G_{sc}}{\pi}\left(1 + 0.033cos\frac{360n}{365} \times \left(cos\emptyset cos\delta sin\, w_s + \frac{\pi w_s}{180}\sin\phi\, sin\delta\right)\right)$$

$$\overline{H}_o = \frac{24\times3600\times1367}{\pi}\left(1 + 0.033cos\frac{360\times198}{365} \times \left(cos4.9\times cos21.2\times sin\, 91.91 + \frac{\pi\times91.91}{180}\sin 4.9\times sin21.2\right)\right)$$

$$= 37595198.69(0.968167732)(0.92840034 + 0.049549809)$$

$$\overline{H}_o = 37595198.69(0.968167732)(0.977950149)$$

$$\overline{H}_o = 35.60 MJ/m^2$$

- The **monthly average daily solar clearness index** from equation (3.34)

$$\overline{K_T} = \frac{\overline{H}}{\overline{H}_0} = \frac{18.03}{35.60} = 0.506 \approx 0.51$$

For $$\mathbf{0.3 < \overline{K}_T < 0.8\ and\ w_s > 81.4^0}$$

- Using equation (3.35), **the diffuse component of the monthly average daily radiation** is determined as

$$\bar{H}_d = \bar{H}\left(1.311 - 3.022\bar{K}_T + 3.427{\bar{K}_T}^2 - 1.821{\bar{K}_T}^3\right)$$

$$\bar{H}_d = 18.03(1.311 - 3.022(0.51) + 3.427(0.51)^2 - 1.821(0.51)^3)$$

$$= 18.03(1.311 - 1.541 + 0.891 - 0.242)$$

$$\bar{H}_d = 7.554 \approx 7.56\ MJ/m^2$$

- The **beam component** of the monthly average daily radiation

$$\bar{H}_b = \bar{H} - \bar{H}_d \tag{4.01}$$

$$\bar{H}_b = 18.03 - 7.56$$

$$\bar{H}_b = 10.47\ MJ/m^2$$

- From Equation (3.38), **the sunset hour angle** is the minimum value between the two terms.

$${w_s}^1 = min[\cos^{-1}(-tan\phi\ tan\delta), \cos^{-1}(-tan(\phi - \beta)tan\delta)]$$

Assuming a tilt angle of 45 ° of the collector,

$$\cos^{-1}(-tan\phi\ tan\delta) = \cos^{-1}(-tan\ 4.9\times tan\ 21.2) = \mathbf{91.91}°$$

$$\cos^{-1}(-tan(\phi - \beta)tan\delta) = \cos^{-1}(-\tan(4.9 - 45)\times \tan 21.2) = \mathbf{70.94}°$$

Therefore the desired sunset hour angle is ${w_s}^1 = \mathbf{70.94}°$

- From equation (3.37), **the monthly average geometric factor $\bar{R}_b$** for surface sloped towards the equator in the northern hemisphere

$$\bar{R}_b = \frac{\cos(\phi - \beta)\cos\delta\sin w_s' + \frac{\pi}{180} w_s' \sin(\phi - \beta)\sin\ \delta}{\cos\phi\cos\delta\sin w_s + \frac{\pi}{180} w_s \sin\phi\sin\delta}$$

$$\bar{R}_b = \frac{\cos(4.9 - 45)\times\cos(21.2)\times\sin 70.94 + \frac{\pi}{180}\times 70.94\times\sin(4.9 - 45)\times\sin(21.2)}{\cos 4.9\times\cos(21.2)\times\sin 91.91 + \frac{\pi}{180}\times 91.91\times\sin 4.9\times\sin(21.2)}$$

$$\bar{R}_b = \frac{0.6741 - 0.28840}{0.92840 - 0.04955} = 0.4$$

- The **Monthly average daily total solar radiation** incident on a south-facing sloped surface **(isotropic sky model)** from equation (3.39) is given as:

$$\bar{H}_T = \bar{H}\left(1 - \frac{\bar{H}_d}{\bar{H}}\right)\bar{R}_b + \bar{H}_d\left(\frac{1+\cos\beta}{2}\right) + \bar{H}\rho_g\left(\frac{1-\cos\beta}{2}\right)$$

$$\bar{H}_T = 18.03\left(1 - \frac{7.56}{18.03}\right)\times 0.4 + 7.56\left(\frac{1+\cos 45}{2}\right) + 18.03\times 0.36\left(\frac{1-\cos 45}{2}\right)$$

$$\bar{H}_T = 4.188 + 6.453 + 0.951$$

$$\bar{H}_T = 11.59\ MJ/m^2$$

$$\bar{H}_T = 3.22\ kWhr/m^2 \text{ or } 134.2\ W/m^2$$

- Therefore the estimated **monthly average absorbed solar radiation** on a south facing collector inclined at an angle of 45° from equation (3.40) assuming the monthly average transmittance –absorptance product of the glass as 0.8 is

$$\bar{S_1} = 0.8\times 11.59$$

$$\overline{S_1} = 9.30\frac{MJ}{m^2}$$

$$\overline{S_1} = 2.58\ kWh/\ m^2\ or\ 107.5\ W/m^2$$

4.3 Collector design

In other to capture more solar irradiation to produce enough thermal energy, large solar collector areas are often proposed. The collector proposed is a 25 m diameter wide, which is circular in shape. The collector is to be raised at a height 2 m above the ground level to allow for easy access during maintenance and also to allow more air inflow. A single glazing transparent glass material is considered as the cover.

4.3.1 Optical energy produced by the solar collector

From equation (3.2) it follows that the optical power produced by the solar collector is

$$\dot{Q}_{opt} = F_R A_c(\overline{\tau\alpha})\overline{H}_T = F_R A_c \bar{S}$$

Assuming that the heat removal factor, F_R =1

$$\dot{Q}_{opt} = 1\times(\pi\times\frac{25^2}{4})\times107.5$$

$$\dot{Q}_{opt} = 52.768 \text{ kW}.$$

Therefore the required optical energy produced by the collector within 24-hour period is

$$Q_{opt} = 52.768\times24 = 1{,}266.4 \text{ kWh}$$

4.3.2 Energy loss in solar collector

Considering the solar collector ground soil temperature heated to 75 ℃, and taking the ambient temperature as 25 ℃, the rate of energy loss in the collector from equation

(3.02) given the overall heat transfer loss coefficient as 0.71 W/m^{2}K (see appendix 2 for details on overall heat loss coefficient calculation) is therefore calculated as

$$\dot{Q}_{loss} = F_R A_c U_L (T_g - T_a)$$

$$\dot{Q}_{loss} = 1 \times 0.71 \times 490.87 \times (65 - 25)$$

$$\dot{Q}_{loss} = 17.425\ kW$$

Considering the energy loss for a 24-hour period, the total heat energy loss to the surrounding is estimated as,

$$Q_{loss} = 17.425 \times 24$$

$$Q_{loss} = 418.2\ kWh$$

4.3.3 Useful energy in the collector

The estimated useful energy needed to produce hot air in the collector from equation (3.01) is

$$\dot{Q}_u = \dot{Q}_{opt} - \dot{Q}_{loss}$$

$$\dot{Q}_u = 52.768\ \text{kW} - 17.425\ kW$$

$$\dot{Q}_u = 35.343\ \text{kW}$$

Therefore the required useful energy for 24-hour period is

$$Q_u = 848.232\ kW$$

4.3.4 Collector efficiency

The efficiency of the collector is expressed from equation (3.04) as

$$\eta_c = \frac{\dot{Q}_u}{A_c \bar{H}_T}$$

Therefore the collector efficiency

$$\eta_c = \frac{848.232\ kWhr}{490.87\ m^2 \times 3.22\ kWh/m^2} = 0.54 = 54\%$$

4.4 Solar chimney design

In other to achieve high wind velocities and efficiency, chimney heights are often made very tall. A chimney of height 50 m and diameter 4 m is considered and the airflow velocity through the chimney will be estimated.

4.4.1 The total pressure developed in the chimney

The total pressure developed due to buoyancy from equation (3.23) is determined as

$$\Delta P_{tot} = \rho_{air} g \left(H_{sc} + {}^{H_c}/_{2} \right)$$

$$\Delta P_{tot} = 1.23 \times 9.81 \times (50 + {}^{2}/_{2})$$

$$\Delta P_{tot} = 615.38\ pa$$

4.4.2 The wind speed through the chimney

The maximum chimney airflow rate and the maximum air velocity in the chimney in the absence of a wind turbine from equation (3.28) is

$$V_{max} = \sqrt{\frac{2\Delta P_{tot}}{\rho_{air}}}$$

$$V_{max} = \sqrt{\frac{2\times615.38}{1.23}} = 31.2\ m/sec$$

The actual wind speed through the 50 m tall chimney when a turbine is installed from equation (3.29) is found to be

$$V_{sc} = \frac{1}{3}\sqrt{\frac{2\Delta P_{tot}}{\rho_{air}}}$$

$$V_{sc} = \frac{1}{3}\ (31.2)$$

$$V_{sc} = 10.4\ m/s$$

4.4.3 The dynamic pressure

From equation (3.27) the dynamic pressure needed to accelerate the air through the chimney is determined as

$$P_d = \frac{1}{2}\rho_{air}{V_{sc}}^2$$

$$P_d = \frac{1}{2}\times1.23\times(10.4)^2$$

$$P_d = 66.52\ pa$$

4.4.4 Static pressure drops at the turbine

The static pressure drop at the turbine from equation (3.27) is determined as

$$\Delta P_t = \Delta P_{tOt} - \frac{1}{2}\rho {V_{sc}}^2$$

$$\Delta P_t = 615.38 - 66.52$$

$$\Delta P_t = 548.86\ Pa$$

4.5 Chimney efficiency

The efficiency of the chimney from equation (2.01) is found as follows

$$\eta_{sc} = \frac{P_{tot}}{\dot{Q}} = \frac{gH_{sc}}{C_p T_f}$$

$$\eta_{sc} = \frac{9.81 \times 50}{1005 \times 338}$$

$$\eta_{sc} = 1.44 \times 10^{-3}$$

$$\eta_{sc} = 0.14\ \%$$

4.6 Electrical power output.

The maximum electrical power output of the plant from equation (3.30) is therefore estimated as

$$P_e = P_{wt,max} = \frac{2}{3} V_{sc} A_{sc} \Delta P_t$$

$$P_e = \frac{2}{3} \times 10.4 \times 12.57 \times 548.86$$

$$P_e = 47834.246\ Wp$$

$$P_e = 48.0\ kWp$$

4.7 The plant overall efficiency

The overall efficiency of the SCPP can be considered as the product of the partial efficiency contributed by each of the plant component (Figure 4.1), i.e. the turbine, the collector and the chimney.

$$\eta_{scpp} = \eta_c \eta_{sc} \eta_{wt} \qquad (4.02)$$

$$\eta_{scpp} = 0.53 \times 0.0014 \times 0.67$$

$$\eta_{scpp} = 0.00049$$

$$\eta_{scpp} = 0.05\ \%$$

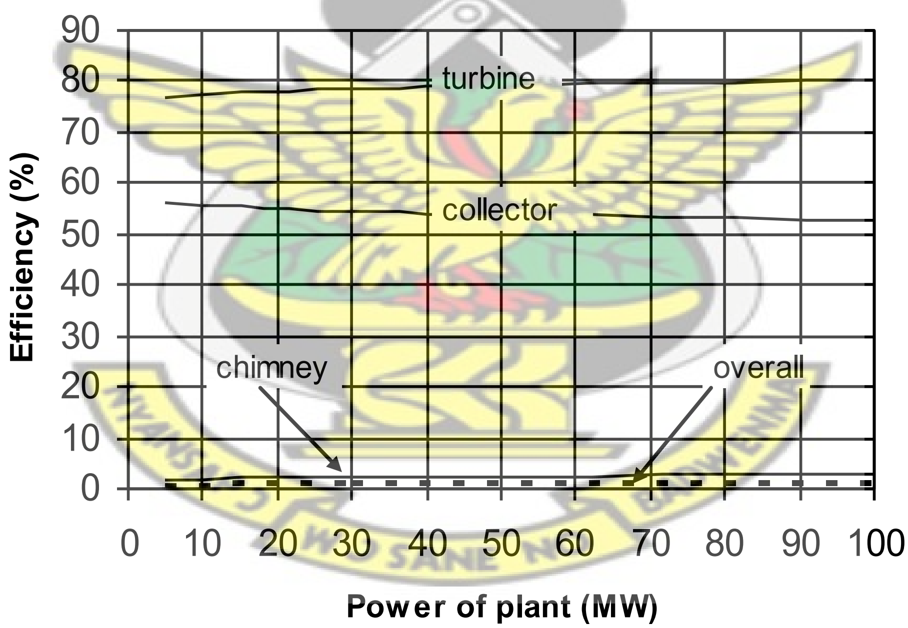

Figure 4.1: Influence of plant components on overall efficiency

4.8 Discussion of results

The SCPP with the following physical dimensions; 25 m diameter collector size, 50 m tall chimney with internal diameter 4 m employing a wind turbine of efficiency 67 % output a theoretical electrical current of 48 Kw based on the climatic conditions of Takoradi city which exhibit an average monthly daily solar radiation of 208.79 W/m^2.

From the results obtained, the electrical power output of the plant is seen to be influenced by the following factors discussed bellow:

1. The collector diameter, increasing the size of the solar collector i.e. the collector area, improves the amount of solar energy absorbed. From equation (3.2), the optical power is seen to be directly influenced by the collector area therefore increasing the collector size causes a rise in the energy input, which contributes, greatly to the power output produced by the plant. Hammadi, (2008) investigated the effect of various collector sizes on the power output and the results depict how the power output is influenced by size of the collector in Figure 4.2.

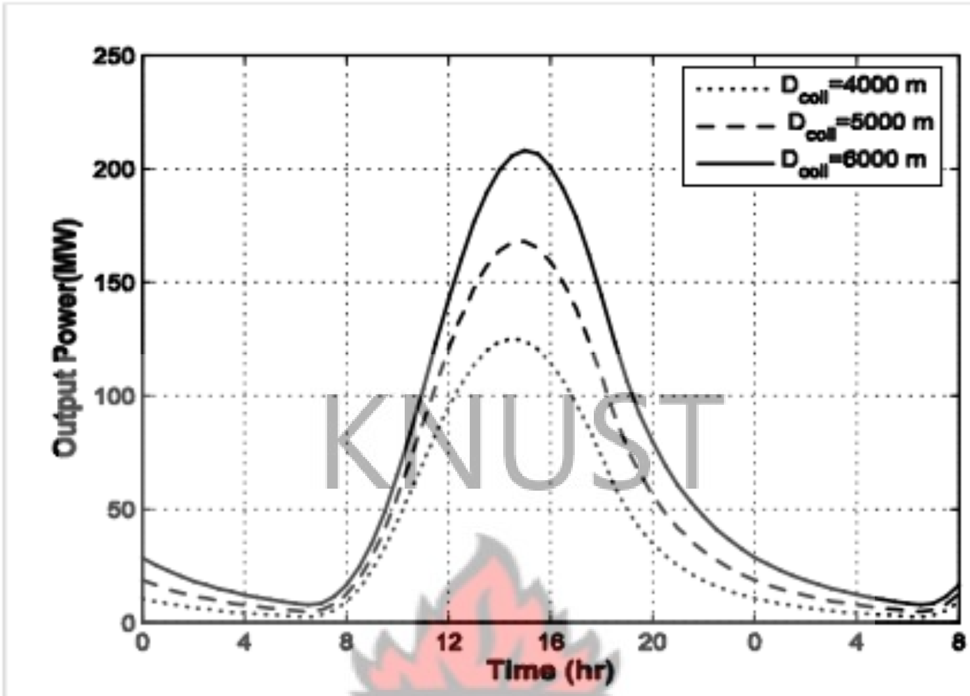

Figure 4.2: The effect of various collector sizes on the power output. Source Hammadi, (2008)

2. The height of the chimney, raising the height of the chimney to an appreciable limit increases the pressure difference which improves the airflow and the draft. In Hammadi, (2008) report on Solar Updraft Tower Power Plant with thermal storage (Figure 4.3,) he studied the effect of the various chimney heights on the power productivity and confirmed the fact that a rise in the chimney height influence the power output. The major challenge to this effect has been the difficulty in putting up tall structures and the huge capital investment that is required. Generally the SCPP portrays very poor overall efficiency and this is attributed to the low efficiency of the chimney. The chimney efficiency is directly proportional to its height and as result increasing the height improves the its efficiency which interns affects the overall plant efficiency. Currently a huge solar

chimney height of 1000 m is being proposed and the effect is that, the efficiency of the chimney will be increased to about 35 % .See Figure 4.4. Owing to this reason Mullet, (1987) in his report on SCPP concluded that the SCPP is essentially a power generator of large scale.

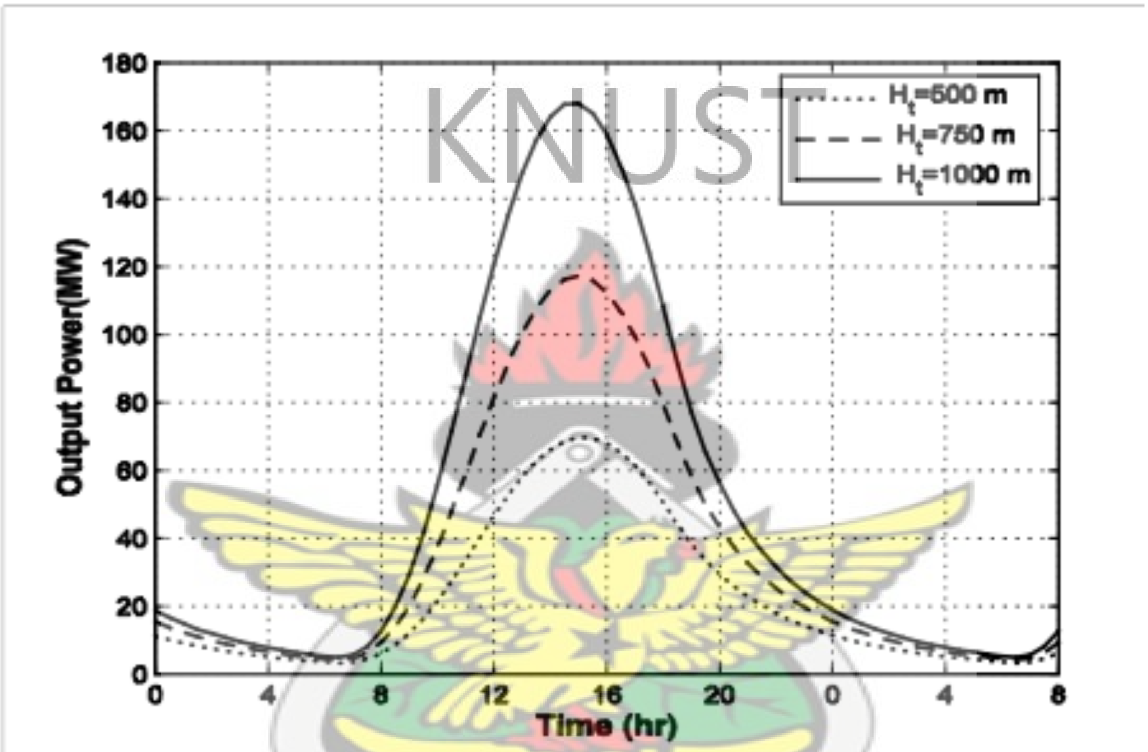

Figure 4.3: **The effect of chimney height on power output. Source, Hammadi, (2008)**

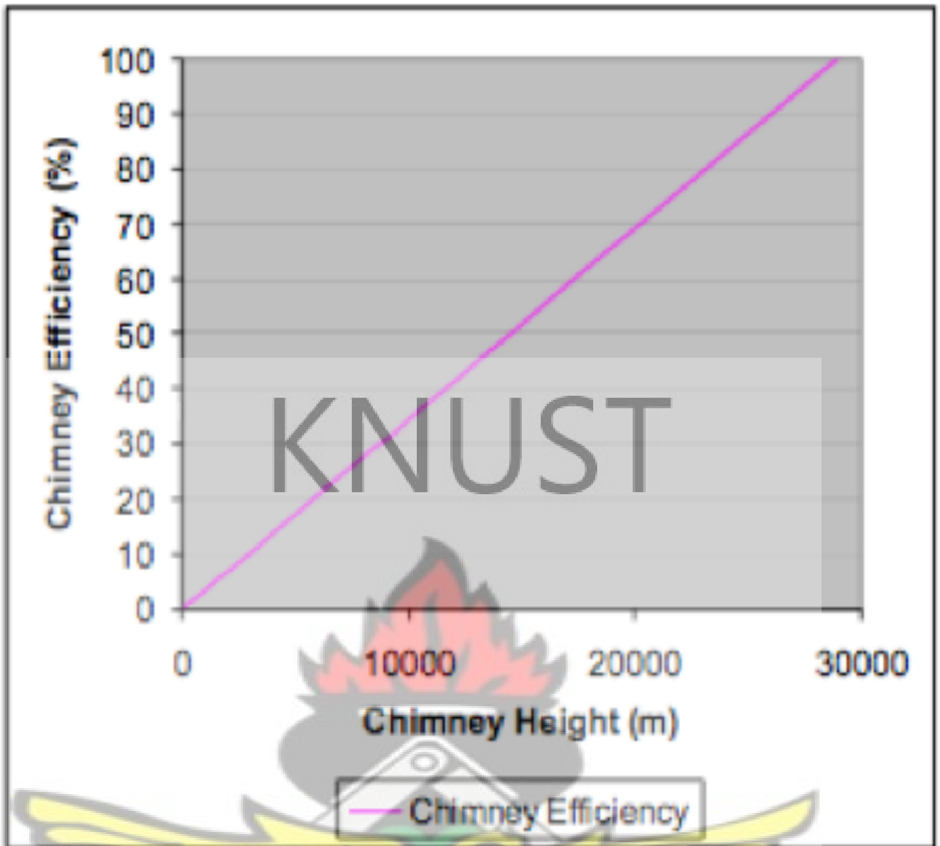

Figure 4.4: Variation of chimney efficiency with height (Schlaich, 1995).

3. The solar irradiance and ambient temperature. These two external factors are key to the power productivity of the solar chimney plant. Its been verified by Dai, *et al.,* (2003) that, the solar radiation is in a dominant position to affect the power generation in the solar chimney plant, in comparison to the ambient temperature in any weather or climatic conditions. See Figure 4.5. In general, the abundance of solar radiation is the active drive to this power technology.

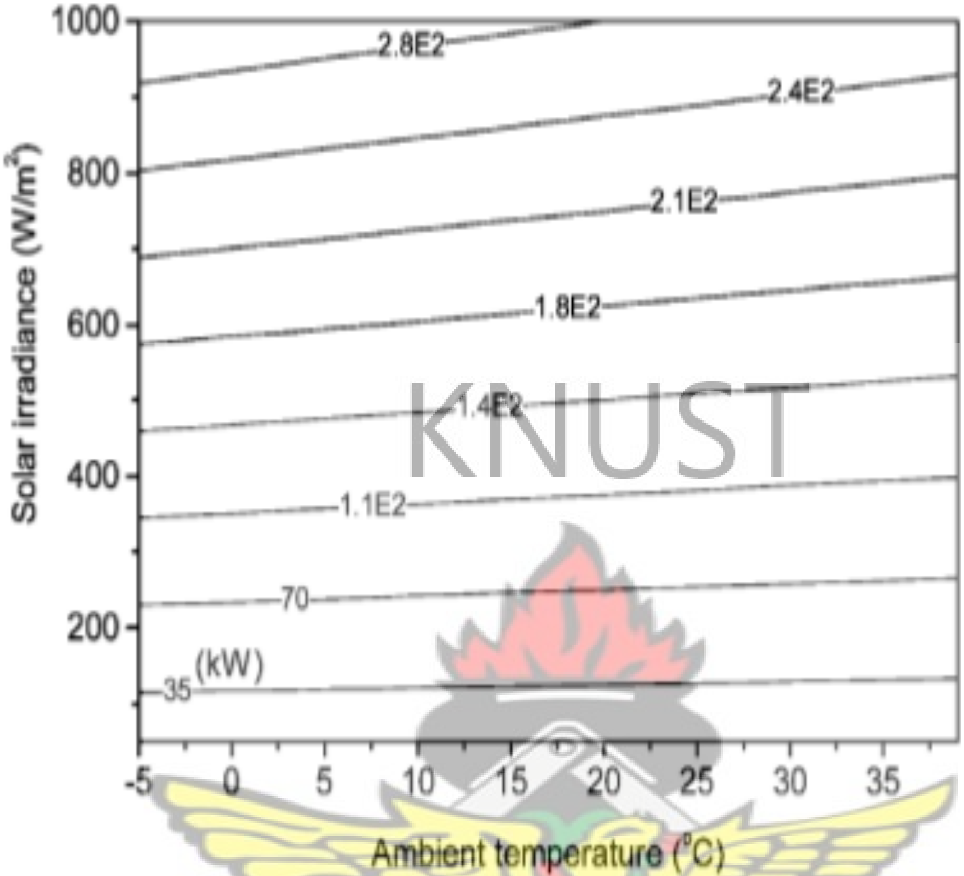

Figure 4.5: The effect of solar irradiation and ambient temperature on the output power. Source *Dai etal.,(***2003)**

CHAPTER 5

5.0 CONCLUSION AND RECOMMENDATION

5.1 Conclusion

The mathematical model as a tool has been used to analysis the design and performance of the SCPP. Mathematical derivations where provided to predict the theoretical power output of the SCPP based on the climatic weather conditions in Takoradi city. As an input to the model, a small scaled plant with the following physical parameters were considered: 25 m diameter collector, 50 m tall and 4 m internal diameter solar chimney. A collector with a single glazing was considered. With a monthly average solar irradiance of 208.79 W/m^2 the plant output a theoretical electrical power of 48 kW. Based on the analysis and result obtained, it was identified that the power output of the plant is greatly influenced by:

1. The size of the solar collector, since the solar energy absorbed increases by increasing the diameter of the collector.
2. The chimney height and
3. The solar irradiation and the surrounding air temperature

It is therefore concluded that the SCPP stands to achieve good performance in areas such as Wa and Navrongo which exhibits the highest monthly average daily solar irradiance of 5.520 kWh/m^2 (230 W/m^2) and 5.505 kWh/m^2 (229 W/m^2) respectively (see Table 4 appendix 1) but only on a large scale.

5.2 Recommendations

Based on the conclusions the following suggestions and recommendations might be found useful in future work.

i. Increasing the height and the diameter of the collector to improve on the power output and efficiency.

ii. As part of government strategy in promoting renewable power development, the Energy Commission of Ghana could conduct a comprehensive economic analysis on the commercial viability of the power produced by the SCPP.

iii. The Mechanical Engineering Department and the Energy Center could as well provide technical support and financial assistance to undergraduate students to mount a pilot plant to promote further studies and research into this area.

iv. Using the fluent software to perform the complex heat transfer analysis to increase the level of accuracy of the model analysis.

v. Finally, another area that calls for further investigation is the usage of thermal storage tanks to improve the efficiency. It has been suggested that by placing extra thermal mass under the collector in the form of black containers with water, the plant could generate power at night. In order to confirm the validity it is therefore recommended that this be experimented with any model to be built to help improve the efficiency and to guarantee continuous power output.

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APPENDIX 1

List of Tables

Table 1. A table showing the recommended average days for months and Values of n by months

	For the average day of month			
Month	**n for *i*th day of month**	**Date**	**n, day of year**	**Declination**
January	i	17	17	-20.9
February	31 + i	16	47	-13.0
March	59 + i	16	75	-2.4
April	90 + i	15	105	9.4
May	120 + i	15	132	18.8
June	151 + i	11	162	23.1
July	181 + i	17	198	21.2
August	212 + i	16	228	13.5

September	243 + i	15	258	2.2
October	273 + i	15	288	-9.6
November	304 + i	14	315	-18.9
December	334 + i	10	344	-23.0

Table 2: Albedo for some natural surfaces

Natural surface types	**Approximated albedo**
Blackbody	0
Grassland and cropland	0.1- 0.25
Dark-Coloured soil surfaces	0.1-0.2
Dry sandy soil	0.15-0.35
Sand	0.2-0.4
Mean albedo of the earth	0.36
Granite	0.3-0.35

Water	0.1-1

Table 3. Properties of ground soil type (sandstone)

Ground type	**Sandstone**
Emissivity	$\mathcal{E}_g = 0.9$
Absorptivity	$\alpha_g = 0.9$
Density	$\rho_g = 2160\ kg/m^3$
Specific Heat capacity	$C_g = 710\ J/kgK$
Thermal conductivity	$K_g = 1.83\ W/mK$

Table 4: Solar irradiation for some locations in Ghana

Synoptic Station	Ground (kWh/m^2-day)	Satellite (kWh/m^2-day)	% Error
Kumasi	4.633	5.155	-11.3
Accra	5.060	5.180	-2.3
Navrongo	5.505	5.765	-4.7
Abetifi	5.150	5.192	-0.8
Akuse	4.814	5.58	-15.9
Wa	5.520	5.729	-3.7
Akim Oda	4.567	5.177	-13.3
Wenchi	5.020	5.093	-1.5
Ho	5.122	5.223	-2.0
Kete Krachi	5.280	5.345	-1.3
Takoradi	5.011	5.200	-3.8
Yendi	5.370	5.632	-4.8
Bole	5.323	5.570	-4.6

Source: SWERA Report , *Energy Commission*

APPENDIX 2

Calculation of heat transfer coefficient and overall heat loss coefficient

1.0 Heat transfer co efficient

- **The convective heat transfer coefficient, ground to cover. $h_{conv\ g-c}$**

$$h_{conv\ g-c} = \frac{N_u \times K_{air}}{d_h} \qquad (3.10)$$

Where K_{air} is thermal conductivity of air taking as K_{air}= 0.0261 W/mK (Duffie and Beckman, 1991)

d_h is the hydraulic diameter or characteristic length of the collector given as

$$d_h = 2H_c \qquad (3.12)$$

$$d_h = 2\times 2$$

$$d_h = 4\ m$$

The Nusselt Number Nu is found by, equation (3.13)

$$N_u = 0.54Ra^{0.25} \text{ For } 2\times 10^4 < Ra < 8\times 10^7$$

The Rayleigh's Number from equation (3.17) is given as

$$R_a = GrP_r$$

depends on the both the Grasshof's number Gr and the Prandtl Number Pr

The Prandtl Number:

$$P_r = \frac{\nu}{\alpha_g} \qquad (3.15)$$

Taking the kinematic viscosity of air as $\upsilon = 1.96\times10^{-5} m^2/sec$ and the ground soil type absorptivity (sandstone) $\alpha_g = 0.9$ (see appendix 1)

$$P_r = \frac{\nu}{\alpha_g} = \frac{1.96\times10^{-5}}{0.9} = 2.17\times10^{-5}$$

Grasshof's Number:

$$Gr = \frac{g\beta'(T_g - T_f)h^3{}_{gap}}{\upsilon^2} \qquad (3.16)$$

Volumetric coefficient of expansion of air in the collector is

$$\beta' = \frac{1}{0.25T_a + 0.75T_f} \qquad (3.18)$$

Assuming the ground under the collector is heated to a temperature T_g of 75 °C and intend transfers the heat to the adjacent moving wind to a temperature of about T_f 65 °C due to natural convection. Taking ambient temperature T_a as 25 °C.

$$\beta' = \frac{1}{(0.25\times25) + (0.75\times65)}$$

$$\beta' = 0.018$$

Taking the collector air gap is h_{gap} as equal to the collector height, H_{col}, 2 m

$$Gr = \frac{9.81\times0.018\times(75-65)\times2^3}{(1.96\times10^{-5})^2}$$

$$Gr = 3.67\times10^{10}$$

Therefore from equation (3.17), Rayleigh's Number:

$$R_a = GrP_r$$

$$R_a = 2.17\times10^{-5}\times3.67\times10^{10}$$

$$R_a = 8.0\times10^{5}$$

Using the equation (3.13), $N_u = 0.54Ra^{0.25}$ due to the condition being satisfied:

$2\times10^{4} < Ra < 8\times10^{7}$

$$N_u = 0.54Ra^{0.25}$$

$$N_u = 0.54\times(8\times10^{5})^{0.25}$$

$$N_u = 16.15$$

The convective heat transfer coefficient, from equation (3.10) therefore is:

$$h_{conv\ g-c} = \frac{N_u\times K_{air}}{d_h} = \frac{16.15\times0.0261}{4}$$

$$h_{conv\ g-c} = 0.1054\ W/m^2K$$

- **The radiation heat transfer coefficient, ground to cover.** h_{rcg}**:**

Assuming collector cover (glass) surface temperature T_c to be 55 °C **ie** between the ground temperature and the ambient temperature

The Stefan Boltzmann constant σ, taken as $5.67\times10^{-8}\ W/m^2\ K^4$

From equation (3.09)

$$h_{rcg} = \frac{\sigma(T_g{}^2 + T_c{}^2)(T_g + T_c)}{\frac{1}{\varepsilon_g} + \frac{1}{\varepsilon_c} - 1} = \frac{5.67\times10^{-8}[(348\ K)^2 + (328\ K)^2][(348 + 328)K]}{\frac{1}{0.9} + \frac{1}{0.9} - 1}$$

$$= \frac{8.765}{1.2222}$$

$$h_{rad\ g-c} = 7.173\ W/m^2K$$

- **Radiation coefficient from the cover to the ambient air / sky h_{rs}**

The radiation coefficient from the cover to the ambient air / sky h_{rs} is given:

$$h_{rs} = \frac{\varepsilon_c \sigma (T_c + T_s)(T_c{}^2 + T_s{}^2)(T_c - T_s)}{(T_c - T_a)} \qquad (3.20)$$

The average sky temperature from equation (3.22) is given as:

$$T_s = 0.0552\times T_a{}^{3/2}$$

$$T_s = 0.0552\times 25^{3/2}$$

$$T_s = 7\ ℃$$

The glass cover emissivity ε_c is taken as 0.9, therefore

$$h_{rs} = \frac{(0.9\times5.67\times10^{-8})[(328) + (280)][(328K)^2 + (280K)^2)][(328 - 280)K]}{328 - 298}$$

$$h_{rs} = \frac{5.103\times10^{-8}\times(608)(185{,}984)(48)}{30}$$

$$h_{rs} = 923{,}261{,}475.2\times10^{-8}$$

$$h_{rs} = 9.23\ W/m^2K$$

- **The wind convection loss coefficient h_w is given as:**

$$h_w = 5.7 + 3.8V_{wind} \qquad (3.21)$$

The average wind speed at the location Takoradi at ground level is taken as 3.2 m/sec. (satellite value, from Retscreen)

$$h_w = 5.7 + (3.8 \times 3.2)$$

$$h_w = 17.86\ W/m^2 K$$

2.0 The overall heat loss coefficient

- The total thermal resistance between the ground and cover/collector, R_1 from equation (3.08) is found to be

$$R_1 = \frac{1}{h_{conv\ g-c} + h_{rcg}}$$

$$R_1 = \frac{1}{0.1054 + 7.173}$$

$$R_1 = 1.37\ K/W$$

- Total thermal resistance from cover to ambient air/sky, R_2, from equation (3.19)

$$R_2 = \frac{1}{h_w + h_{rs}}$$

$$R_2 = \frac{1}{9.23 + 17.86}$$

$$R_2 = 0.0369\ K/W$$

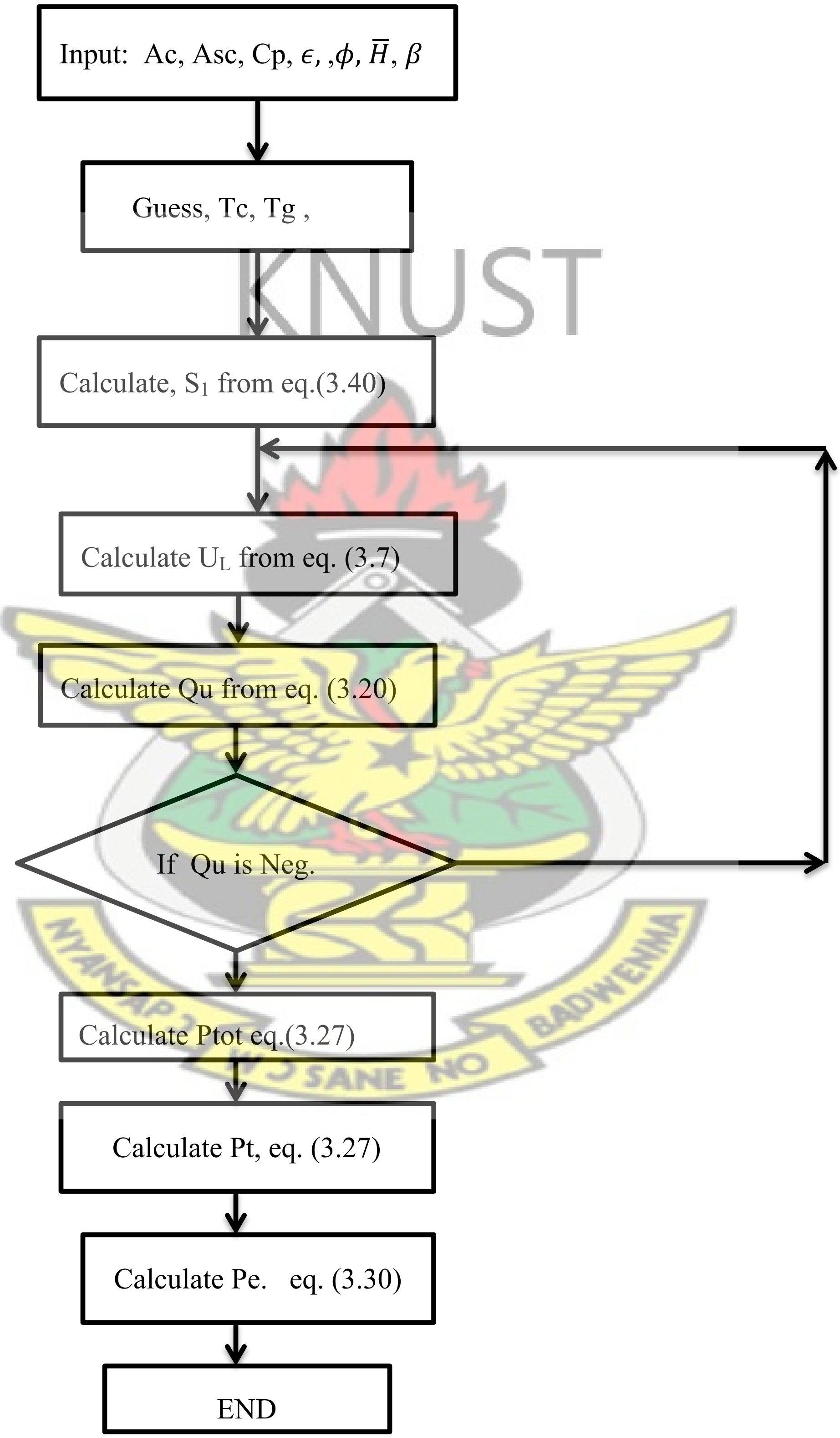
- The overall heat transfer coefficient from equation (9) therefore is:

$$U_L = U_t = \frac{1}{R_1 + R_2} = \frac{1}{1.37 + 0.0369}$$

$$U_L = 0.71\, W/m^2K$$

APPENDIX 3

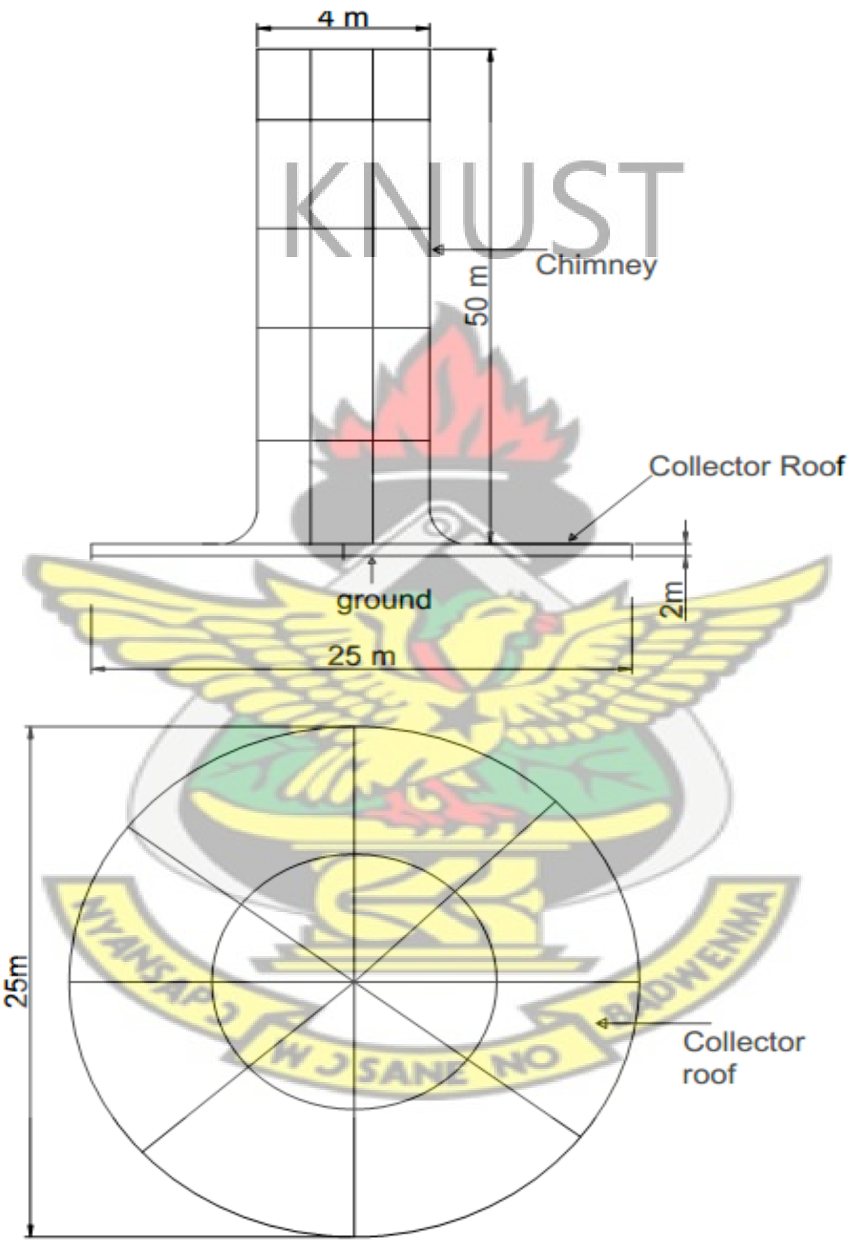
Summary of flow chart showing solution procedure of the mathematical model.



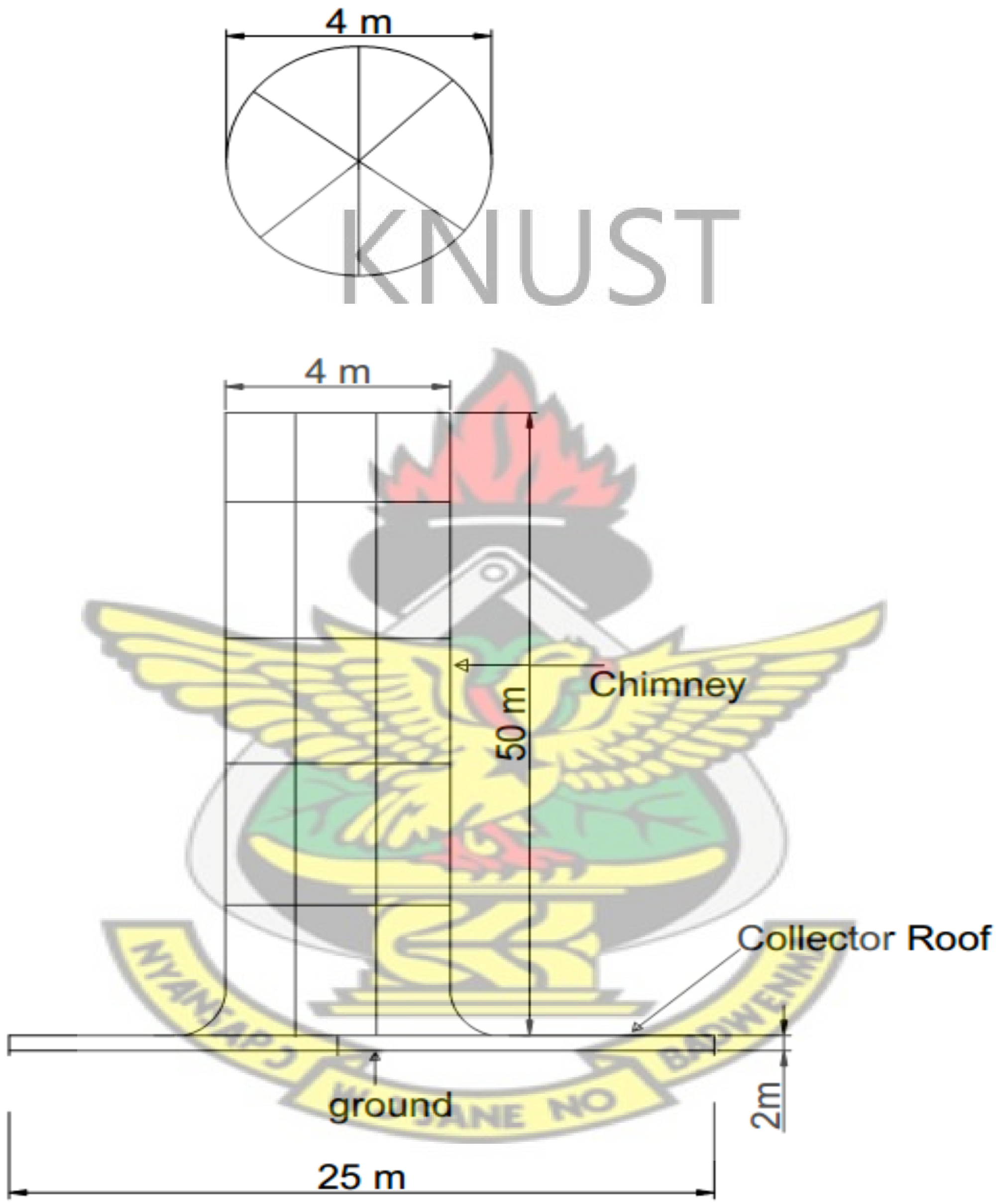
APPENDIX 4

Design drawings

Drawing 1: Solar chimney with collector



Drawing 2: Solar chimney



Drawing 3: Solar collector