DOKUZ EYLÜL UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

STEAM GENERATION AND PERFORMANCE ANALYSIS OF EVACUATED TUBE SOLAR COLLECTORS WITH THE INNER PIPE

by
Azat SERVER

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İZMİR

STEAM GENERATION AND PERFORMANCE ANALYSIS OF EVACUATED TUBE SOLAR COLLECTORS WITH THE INNER PIPE

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M.Sc THESIS EXAMINATION RESULT FORM

We have read the thesis entitled **"STEAM GENERATION AND PERFORMANCE ANALYSIS OF EVACUATED TUBE SOLAR COLLECTORS WITH THE INNER PIPE"** completed by **AZAT SERVER** under supervision of **PROF. DR. MUSTAFA SERHAN KÜÇÜKA** and we certify that in our opinion it is fully adequate, in scope and in quality, as a thesis for the degree of Master of Science.

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ABSTRACT

In this study, the thermal performance of evacuated tube solar collector (ETSC) with an inner tube was investigated. For this purpose, a compound parabolic solar collector with tubular absorber has been designed and manufactured. In order to observe system performance, experiments were carried out in different outdoor conditions. During the experiments, the ambient temperature, the outer glass temperature, the inner tube surface temperature, solar radiation and amount of steam generation were measured. The thermal efficiency of the system was determined by using the steam obtained from the system. A mathematical model for one-dimensional steady state, where the energy balance was established by using heat transfer relations, was transferred to the computer via the EES software. Numerical and experimental results are compared and so heat loss was calculated. After that, the thermal performance of the system was determined at different irradiation rates and flow temperatures by using the software.

Keywords: Compound parabolic collector, inner tube, steam generation, thermal performance analysis, heat loss model, solar collector

İÇ BORULU VAKUM TÜPLÜ GÜNEŞ TOPLAYICILARINDA BUHAR ÜRETİMİ VE PERFORMANS DEĞERLENDİRMESİ

ÖZ

Bu çalışmada, yansıtıcı yüzey kullanılan iç borulu vakum tüp güneş toplayıcısının ısıl performansı incelenmiştir. Bu amaçla iç borulu bütünleşik parabolik güneş toplayıcısının tasarım ve imalatı gerçekleştirilmiştir. Sistem performansını gözlemlemek için, farklı dış ortam koşullarında deneyler yapılmıştır. Deneyler sırasında çevre sıcaklığı, vakum tüp dış yüzey sıcaklığı, iç boru yüzey sıcaklığı, güneş ışınım değeri ve üretilen buhar miktarı ölçülmüştür. Elde edilen buhar kullanılarak sistemin ısıl verimi belirlenmiştir. Bir boyutlu kararlı hal durumu için ısı transfer bağıntıları kullanılarak enerji dengesinin kurulduğu bir matematiksel model oluşturulmuş ve EES yazılımı yardımıyla bilgisayar ortamına aktarılmıştır. Sayısal ve deneysel sonuçlar karşılaştırılmış ve ısı kaybı hesaplanmıştır. Ayrıca yazılımın kullanılması ile sistemin farklı ışınım şiddetleri ve akışkan sıcaklıkları için ısıl performansı belirlenmiştir.

Anahtar Kelimeler: Bütünleşik parabolik yansıtıcı, iç boru, buhar üretimi, ısıl performans analizi, ısı kaybı modeli, güneş toplayıcı

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CHAPTER ONE
INTRODUCTION

A brief introduction about energy sources, importance of solar energy and explanation of its features are given in this chapter. Literature review about evacuated tube solar collector systems is also provided to highlight the recent research trends about these systems. Literature survey is specialized in type of evacuated tube solar collector systems, compound parabolic concentrators, and thermal efficiency of the evacuated tube solar collectors. In the following part of the chapter, the motivation and outline of thesis are provided.

1.1 Classification of Energy Sources and Solar Energy

The energy that is used every stage in daily life can be found in different forms such as chemical, nuclear, mechanical, thermal, geothermal, hydraulic, solar, wind, electrical energy and can be converted to each other by appropriate methods. Energy sources are divided into two groups according to their usage patterns, renewable and non-renewable as shown in Figure 1.1 (Koç & Şenel, 2013).

Natural gas and coal are fossil-based, while uranium and thorium are core-based non-renewable energy sources. These sources have led to the search for alternative energy sources due to their greenhouse effect and environmental pollution, high operating costs and non-renewable. The reasons mentioned above have led to the existence of renewable energy sources such as solar, wind, biomass, hydrogen and wave that do not harm the ecological balance.

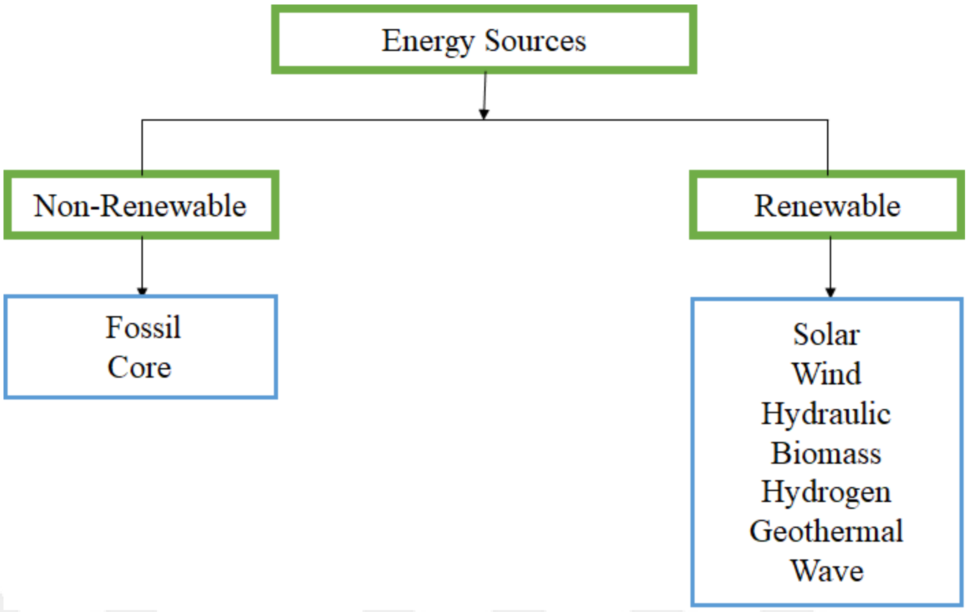

Figure 1.1 Classification of energy sources (Koç & Şenel, 2013)

Solar energy is the most widely used type of energy in these sources. The demand for this energy is increasing day by day since it is very convenient to use in domestic and industrial facilities. Solar collectors are used to store solar energy and convert it into the required form of energy, absorbing the radiation from the sun. In order to increase the amount of solar energy obtained, many solar collector models have been developed and used today.

Evacuated tube solar collectors (ETSC) have become prominent in recent years in hot production with reasonable price and low heat loss. They can also be used at higher temperatures using a reflective surface. Compound parabolic concentrators (CPC) have been widely used in solar energy medium temperature applications, in recent years. The reason of this is the low cost of installation and ability to the required temperatures which these systems. Literature studies about ETSC and CPC are given in.

At direct systems, in which coaxial or U tubes are inserted of all glass vacuum tube, the flow is circulated forced in the tubes. Another way to obtain heat from the evacuated tube is using heat pipe. Heat pipe and direct flow evacuated solar collectors supply higher heat transfer coefficient than all glass solar collectors.

ETSC can be used also absorber tube of compound parabolic collectors (CPC). At these solar collectors, the flow can rise higher temperatures using with reflective surface. CPC's have been widely used in medium temperature applications, in recent years. The reason of this is the low cost of installation and ability to reach to the required temperatures without any solar tracking mechanism.

1.2 Literature Review

There are many experimental and theoretical studies about evacuated tube solar collectors (ETSC) and compound parabolic concentrators in the literature. A large part of this theoretical studies consists of modelling the thermal efficiency of evacuated tube collector systems and the heat losses that occur in these systems. In addition, some studies which about optical efficiency of CPC evacuated tube solar collector and effect of CPC's on these systems efficiency is available. Experimental studies include comparison of different types ETSC efficiencies, optimization of thermal losses and comparison with theoretical results. In this section, some of these studies which mentioned above are included and examined.

Water-in-glass solar water heaters consist of all glass evacuated tubes inserted directly into a storage tank. Heat production from a water-in-glass evacuated tube is driven by natural circulation of the fluid between the collector and the storage tank. These types systems work at low pressure, so they cannot rise to high temperature values. For this reason, they are often used in domestic hot water supply. Some studies about all glass evacuated tubes are examined in detail and given below.

Shah & Furbo (2007) investigated the heat transfer and flow structures for all glass evacuated tubes under different working conditions with computational fluid dynamics (CFD). The collector geometry is determined with a distribution tube which placed vertical and vacuum tubes placed horizontally. The temperature of the fluid entering the collector was fixed at 333 K. Also, CFD model was created for three different tube lengths which 0.59, 1.17 and 1.47 m. Flow rates entering the system were analysed for the 0.05, 0.4, 1.0, 3.0 and 10 kg/min values. The highest efficiency was obtained in the shortest tube and in the flow rate range of 0.4-1 kg/min.

Yadav & Saikhedkar (2017) developed a simulation model with MATLAB software to determine the theoretical efficiency of the all glass evacuated tube collector. They have also investigated the effect of atmospheric conditions on the collector performance. It has been found that the absorber surface and the amount of incoming solar radiation have a significant effect on the collector efficiency. As a result of experimental studies, it was observed that collector efficiency ranged between 71.4% and 75.6% during the day.

Coaxial or U tube carriers put insert of all glass vacuum tube high pressure and high heat transfer can provide. High pressure and good heat transfer can be achieved by placing coaxial and U tube carriers in the all glass vacuum tube. Another way to obtain heat from the evacuated tube is using a heat pipe. Heat pipe and direct flow evacuated solar collectors can rise to higher pressures also can supply higher heat transfer coefficient than all glass solar collectors. Such as mathematical modelling of heat loss, theoretical and experimental thermal efficiency and reduction of heat loss which direct flow solar collectors found in literature was studied.

Ataee & Ameri (2015) have modelled forced convention flow T-type and H-type coaxial evacuated tube collectors. Different working fluids (air, CO_2), delivery tube characteristics and absorber tube; effects of working fluid on temperature distribution were investigated. Both of collectors the effect of mass flow, solar radiation intensity, inlet temperature, ambient temperature, optical efficiency and tube length on outlet temperature were investigated and energy-exergy analyses were performed. The results showed that the fluid outlet temperature and exergy analysis in the H-type model were better for both working fluids that CO_2 and air than the T-type model.

Ma, Lu, Zhang & Liang (2010) investigated the effect of the air layer between the absorber surface and the fin of the U tube which inserted in evacuated tube. Thermal performance of the collector was investigated by analytical method based on conservation of energy principle. The results show that the air resistance that between the absorber surface and copper fin reduces the collector efficiency by 10% and increases the absorber surface temperature by 30°C.

Farjallah, Chaabane, Mhiri, Bournot & Dhaoudi (2016) developed a 3D numerical model for a U tube solar collector. Numerical results showed that collector efficiency increased from 30.9% to 35.6% when the working fluid flow rate was 0.001, 0.002 and 0.003 kg/s. The same numerical method was repeated for copper fin and graphite filled U tube solar collectors and collector efficiencies were calculated as 54% and 64%, respectively.

Ong & Tong (2012) tested the natural and forced convection U tube and heat pipe evacuated tube solar collector under the same operating conditions. As a result of the experimental studies, the efficiency of the U pipe collector was 53% and the efficiency of the heat pipe collector was 65%.

Ghoneim, Shabana, Shabaan & Mohammedien (2016) evaluated the experimental and numerical performance of the U tube solar collector under the conditions of Kuwait. A comprehensive optical and thermal analysis of the collector was performed. A numerical solution was applied to solve the non-linear equations forming the mathematical model. Experimental and numerical simulation results were obtained.

Mishra & Saikhedkar (2014) described the system elements and properties that get the U tube solar collector in this study. According to the reference sources used, the transmittance of the outer glass tube as 92% and absorptivity of the surface coating as 92%. For the same operating conditions, U tube collectors have 25-40% higher efficiency than flat plate solar collectors.

Some studies about the type of fluid in the heat pipe, the diameter of the heat pipe, the length of the condenser and evaporator zone are examined and stated below.

Daghigh & Shafieian (2016) evaluated the theoretical and experimental performance of the heat pipe solar water heating system. A mathematical model was created for thermal and exergy analysis of the collector. Then this system was experimentally tested under outdoor conditions. The results showed that the collector output temperature reached a maximum temperature (64°C) between 15:00-16:00 hours and the optimal heat pipe number was 15.

Topal (2018) investigated the effects of working fluids used in heat pipe evacuated tube solar collectors on system performance by using alternative nanofluids. New nanofluids were obtained by adding copper oxide and aluminium oxide nanoparticles to working fluids such as antifreeze, methanol and acetone used in heat pipes. As a result of the experiments, the highest system efficiency (58.4%) was obtained from acetone-aluminium oxide mixture.

Ersöz (2016) studied the effects of different working fluids on performance in heat pipe solar collectors. Energy and exergy analyses of the system were performed using six different working fluids: ethanol, methanol, acetone, hexane, petroleum ether, chloroform. An air heating device design and the system performance was evaluated for air speeds of 2 m/s, 3 m/s and 4 m/s. As a result of the experiments, the best exergy efficiency was achieved with acetone at 2 m/s and chloroform at 3m/s and 4 m/s air speed.

Kumar et al. (2017) modelled the heat generation of the heat pipe evacuated tube solar collector. The heat transfer analysis and design of the evacuated tube solar collector which length of 1.8 m and internal and external diameters of 0.049 m and 0.058 m and made of borosilicate glass was made. As a result of the experiments, it was determined that the ratio of the length of the evaporator to the length of the condenser area was an important variable for the collector design.

CPC are used to effectively reach higher temperatures. The studies about the solar collectors used in CPC are examined in detail.

Tzivanidis, Bellos, Korres, Antonopoulos & Mitsopoulos (2015) examined the effectiveness of parabolic trough solar collector at high temperatures. The parabolic trough collector is designed and simulated with Solidworks software for different operating conditions. The main purpose of the study is efficiency analysis of this model and heat transfer events. Heat flux was given perpendicular to the absorber surface and the temperature distribution on this surface and inner pipe was obtained. The convection coefficient that in tube has been calculated and compared with the studies in the literature. At high temperatures, collector efficiency was achieved over of 75%. This high efficiency was explained by the high input temperature and the heat loss coefficient being between 0.6–1.3 W/m^2K.

Brunold, Frey & Frei (2010) examined the effect of compound parabolic concentrator on system efficiency for U tube solar collectors. A comparison was made between for the flat plate, concentrator and non-concentrator collector at high temperatures. The system which include concentrator efficiency was calculated as 35%, the non-concentrator system's efficiency as 25% and flat plate collector system's efficiency as 13%.

Pei, Li, Zhou, Ji & Su (2012) examined the effect of CPC use on a U tube solar collector. Two systems with and without CPC were created by combining the glass tubes with three series and four parallel ones with 47 mm outer diameter. Experiments were conducted for input temperatures of fluid at 55, 65, 75, 85 and 95°C. While the water temperature in the tank was 70°C, the efficiency of the system without CPC was found to be 29% and the efficiency with CPC was 35% and efficiency increase was obtained that 20% by adding CPC.

Aguilar-Jiménez, Velázquez, Acuña, López-Zavala, & González-Uribe (2018) compared the thermal efficiency of two different collectors with parabolic concentrator surfaces positioned in the North-South and East-West direction numerically and experimentally. The results showed that the optical efficiency of the compound parabolic collector positioned in the East-West direction was 57.5% and the optical efficiency of the CPC collector positioned in the North-South direction was 51.3%.

Aboulmagd, Padovan, Oliveski & Col (2014) developed a new model by performing performance evaluation for U tube evacuated tube solar collectors with and without CPC in their study. The mathematical model was solved in MATLAB for heat losses. Experimental and analytical data was compared and it was found that the U tube collector with concentrator gave better results in terms of optical and thermal efficiency than non-concentrator collector.

Widyolar, Jiang, Ferry & Winston (2018) designed a CPC which can be used commercially at temperatures between 100– 250°C temperatures. The CPC's optical and thermal efficiencies were simulated as 71% and 50% respectively, at temperature of 200°C with an aperture area of 4.56 m^2. However, on experimental studies, the optical efficiency was 62% and the thermal efficiency was 50%. The installation cost of the system is 0.58 $/W and its lifetime is approximately 20 years.

Yadav & Saraswat (2016) studied the collector efficiency by performing steam production from evacuated tube solar collector experimentally. Natural circulation occurs in the system due to the slopes of the distribution tube and the evacuated tubes connected to it. Experimental results showed that the highest efficiency was between 14:00-15:00 hours and 46.26%.

Vendan, Shunmuganathan, Manojkumar & Thanu (2012) calculated to the steam production under atmospheric pressure by comparing flat plate and evacuated tube solar collectors. The aperture area of the collector containing 4 tubes was calculated as 0.31 m^2 and the theoretical efficiency as 25.1%. Since the fluid temperature in the evacuated tube collector can rise above 200°C, it is stated that it is more applicable in steam production than flat plate solar collectors.

Liu, Tao, Lu & Wang (2014) developed a low-cost compound parabolic concentrator evacuated tube steam generator. The aperture area is 32 m^2 and each unit of the steam generator, which consists of 60 collector units, has a metal pipe placed inside the evacuated tube. Produced steam temperature 0.10-0.55 MPa pressure range up to 200°C. The air gap between the inner glass tube and the concentric metal tube was filled heat transfer oil and graphite powder. Experimental results showed that the maximum steam outlet temperature was 200% and 0.55 MPa and the collector thermal efficiency was above 30%.

Mehta & Rane (2011) conducted performance tests for a heat pipe solar collector with a compound parabolic concentrator. They obtained saturated steam with a pressure of 1 bar and a temperature of 100°C with the collector having a aperture area of 7 m^2 with using tap water. At the end of the 6 hours experiment, the average solar radiation ranged between 685 W/m^2 and 719 W/m^2, while the average efficiency of the system was between 48% and 54%.

1.3 Motivation and Content of Thesis

When studies in the literature are examined, it is observed that evacuated tube solar collectors are also heat pipes or U-pipes. In this study, in contrast to the literature, a steel inner tube was placed in an all glass evacuated tube collector. RayViz® and TracePro® software were used to analyse the beam for the CPC surface. As a result of the analysis, optical efficiency was calculated and the effect of the concentrator surface on the collector efficiency was evaluated. A mathematical model was created by a one-dimensional thermal resistance network based on the solar radiation coming to the absorber surface. The heat losses and thermal efficiency of the system were calculated with EES® by reference to this surface. Then an experimental setup was established using one- evacuated tube which includes an inner tube and compounds with parabolic concentrator. Optical and thermal efficiency and availability for steam generation of the system was evaluated experimentally.

The chapters of the thesis are as in the following: In chapter 2, the general structure of the sun, solar energy applications and solar collector types are investigated and detailed information is given. In chapter 3, a CPC with tubular absorber was modelled. The ray analysis was made at different incident angles and the optical efficiency of the system was obtained using Trace Pro software. One-dimensional mathematical model of thermal resistance network was set and thermal performance at various temperatures and irradiation rates was calculated. In chapter 4, an experimental study has been described in detail and applicability for steam production was evaluated. In chapter 5, the numerical and experimental results were summarized and evaluated.

CHAPTER TWO
SOLAR ENERGY APPLICATIONS AND SOLAR COLLECTORS

A brief about structure of the Sun and Solar energy applications is given in this chapter. After that, collector systems were investigated which common use in solar energy applications. Finally, evacuated glass tube solar collectors were classified.

2.1 Structure of the Sun and Solar Constant

The sun is the largest star in the solar system that a density of approximately 1.5×10^{11} diameter and 1.39×10^{9} m spherical shape composed of dense and hot gases. It is estimated that the temperature in the central interior areas is 8×10^{6} to 40×10^{6} K, and the density is approximately 100 times the water density. The surface temperature is approximately 5777 K. During the nuclear reactions that occur in the sun, hydrogen is transformed into helium, during which a quantity of energy is produced (Duffie & Beckman, 2013). A schematic structure of the sun is shown in Figure 2.1.

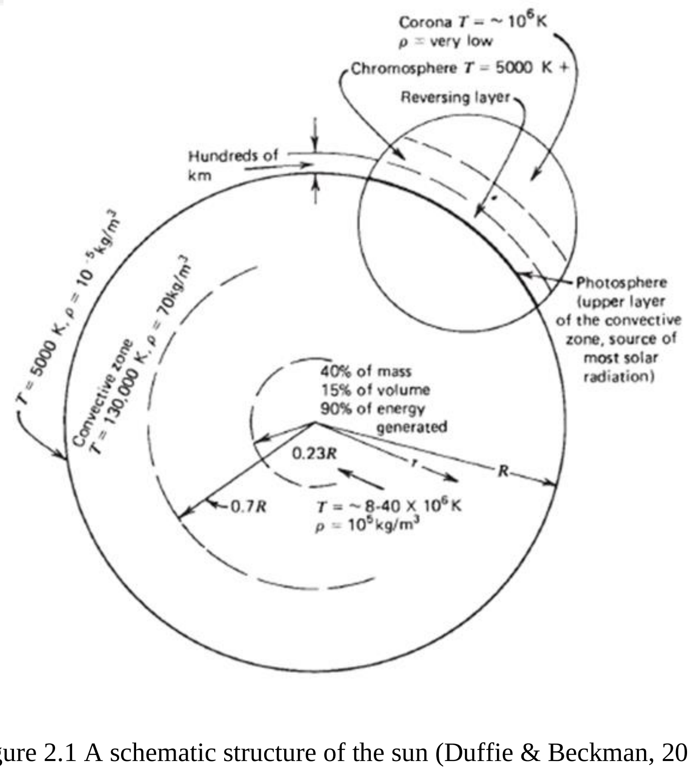

Figure 2.1 A schematic structure of the sun (Duffie & Beckman, 2013)

2.2 Solar Energy Applications

Solar energy systems vary in terms of methods, materials and technologies. The main purpose of these systems is to convert solar energy into thermal energy. This heat energy; can be used for different purposes as heating and cooling applications, steam production, electricity generation, industrial drying, cooking. The most common uses of solar energy are photovoltaic systems and solar thermal systems.

2.2.1 Photovoltaic Systems

Photovoltaic (PV) is a technology that converts solar energy into electric current. Materials such as silicon have the property of directly converting solar energy into electrical energy, which is called a photovoltaic effect. Photovoltaic solar cells are systems that consist of semiconductor materials that convert solar energy directly into DC electrical energy when sunlight falls on it. Semiconductor materials such as Silicon, Gallium, Arsenite, Cadmium, Telluride or Copper indium are used during this process.

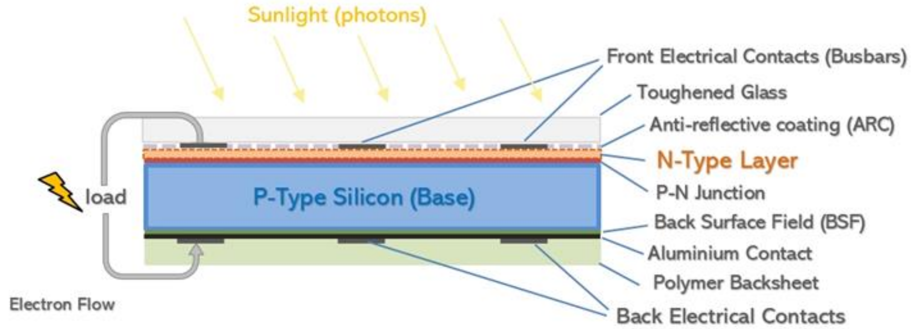

Figure 2.2 Schematic diagram of a solar cell (Clean energy reviews, 2019)

Solar cells operate on the base of photovoltaic principle and when the light falls on them, electric voltage is generated at the ends. Solar cells absorb to photons (components of solar rays) and convert their energy into electrical molecules (electrons). These molecules are collected in the front and back parts of the cells. A Solar Cell schematic diagram was given in Figure 2.2. The voltage created here creates an electric current. The cells mounted on the modules and panels are also connected with together in series in order to obtain a sufficiently high voltage.

2.2.2 Solar Collectors and Hot Water Systems

Solar thermal applications are divided into three according to working temperature;

1- Low temperature applications (20-110°C)
2- Medium temperature applications (110-400°C)
3- High temperature applications (>400°C)

Thermal applications of solar energy consist of three basic system elements. These are solar collectors, heat exchangers and storage units. The solar collector is the equipment that converts the solar energy into heat energy. This heat energy is transferred to the working fluid which in the solar collector. There are basically two types of solar collectors: stationary and concentrating (Kalogirou, 2013).

Stationary solar collectors are classified in three main groups: flat plate, evacuated tube and compound parabolic collectors. Parabolic dish, parabolic trough and solar tower fall into the concentrating collector types and is also shown in Figure 2.3.

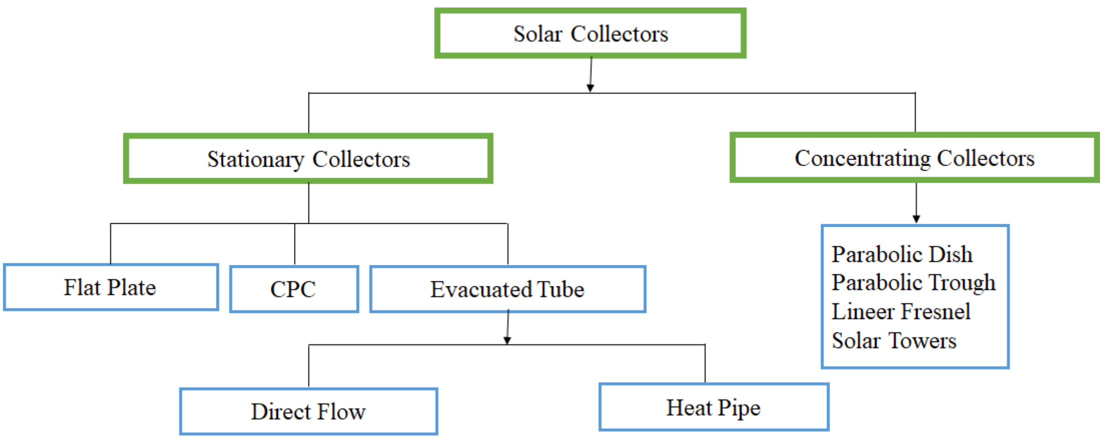

Figure 2.3 Classification of solar collectors

In this section, detailed information about the structure of evacuated tube collectors are given. Structures, operating temperatures, and application areas for these types of collectors are indicated.

2.3 Evacuated Tube Solar Collectors

Evacuated tubes consist of two glass tubes, where heat loss is reduced by emptying the air between them with a vacuum. The inner glass tube outer surface is covered with a selective coating (Al/N/Al, AlN/AlN-SS/Cu) that can absorb 93% of the sun's rays. Antifreeze is not needed in this system as the vacuum process eliminates the risk of freezing the water inside the tubes. Evacuated tubes, thanks to their shape, can receive the sun's rays upright at all hours of the day. Evacuated glass tube sectional view is shown Figure 2.4.

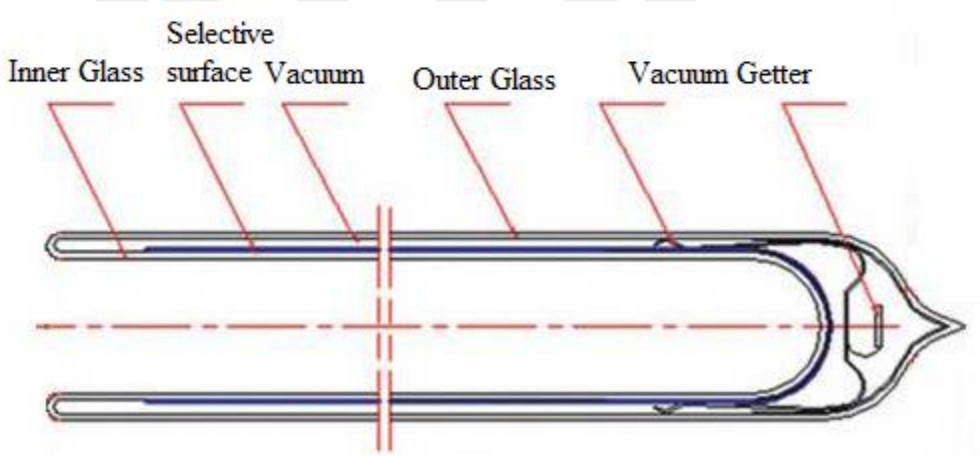

Figure 2.4 Evacuated glass tube sectional view

There are three types of evacuated tube solar collectors, all glass, direct flow and heat pipe. The heated fluid rises in the evacuated tube by natural circulation for all glass collector types and also shown in Figure 2.5.

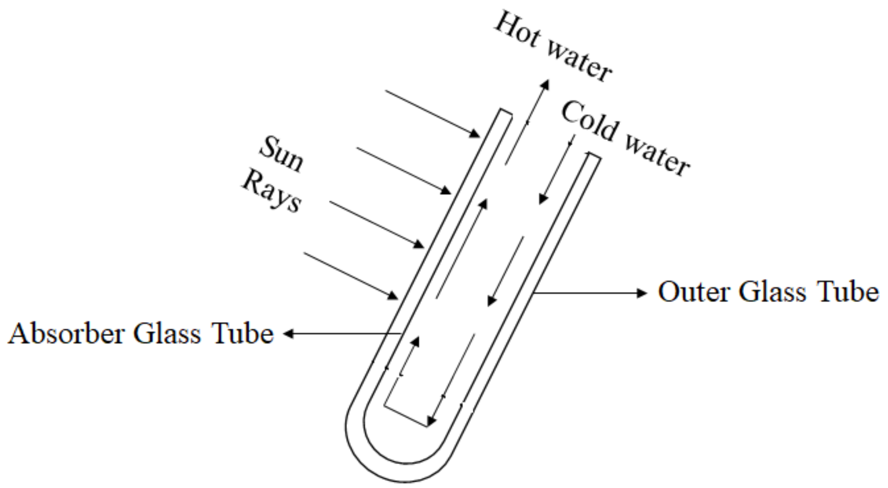

Figure 2.5 A schematic structure of all glass evacuated tube

Therefore, it has been developed other collectors which using direct flow (coaxial, U-tube) or heat pipe to supply the circulation and an increasement of heat transfer to the fluid.

2.3.1 Direct Flow Evacuated Tube Collectors

Direct flow evacuated tube collector systems consist of evacuated glass tube, inner tube and manifold. An inner tube is placed in the evacuated tube concentrically and the working fluid moves within the tube. The most commonly use types of carrier systems are those with coaxial and U tube.

There are two co-axis carrier pipes in coaxial tubes. The inner tube bottom surface is left open and the heated water rises between the inner and outer pipe and goes to the manifold. The gap between the outer tube and the vacuum tube is filled with Aluminium fin or gap material (graphite) to supply conduction of heat from the absorber surface to the carrier pipe. The working method of the coaxial evacuated tube solar collector is shown in Figure 2.6.

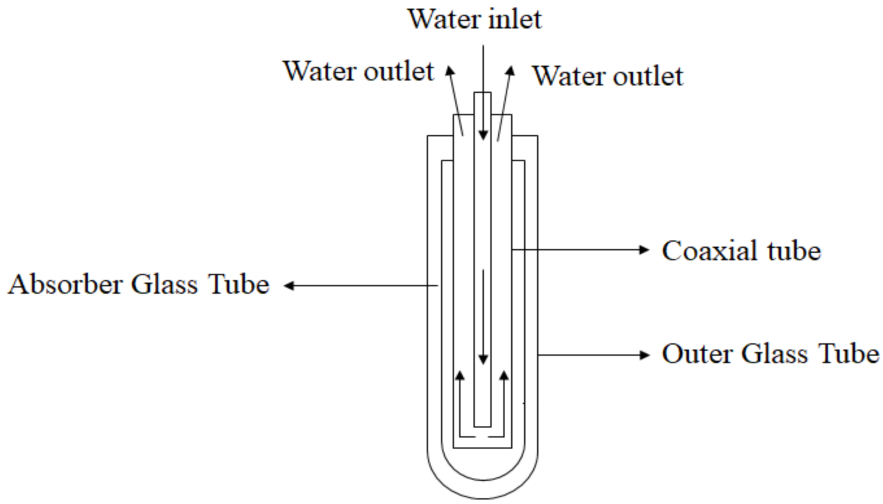

Figure 2.6 A schematic structure of coaxial evacuated solar collector

In the U tube evacuated tube solar collector, the U tube is placed in the evacuated tube. As the heat conduction coefficient is high, copper is often used as tube material. A U tube solar collector system structure is given in Figure 2.7.

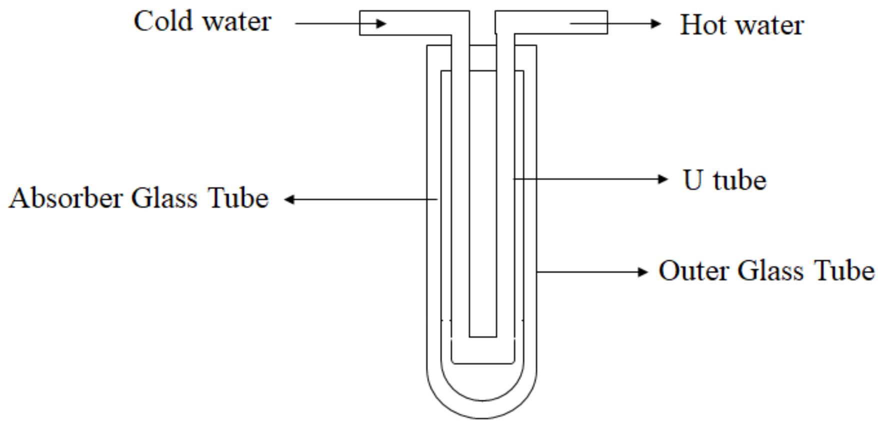

Figure 2.7 A schematic structure of U tube evacuated solar collector

2.3.2 Heat Pipe Evacuated Tube Solar Collectors

Heat pipes are two phase devices which carry a high amount of heat due to the effect of temperature difference between hot and cold sources. Heat pipes of very different types are widely used in many industrial applications in order to improve heat transfer. One of these applications is heat pipe evacuated tube solar collectors. A heat pipe evacuated solar collector illustrated in Figure 2.8.

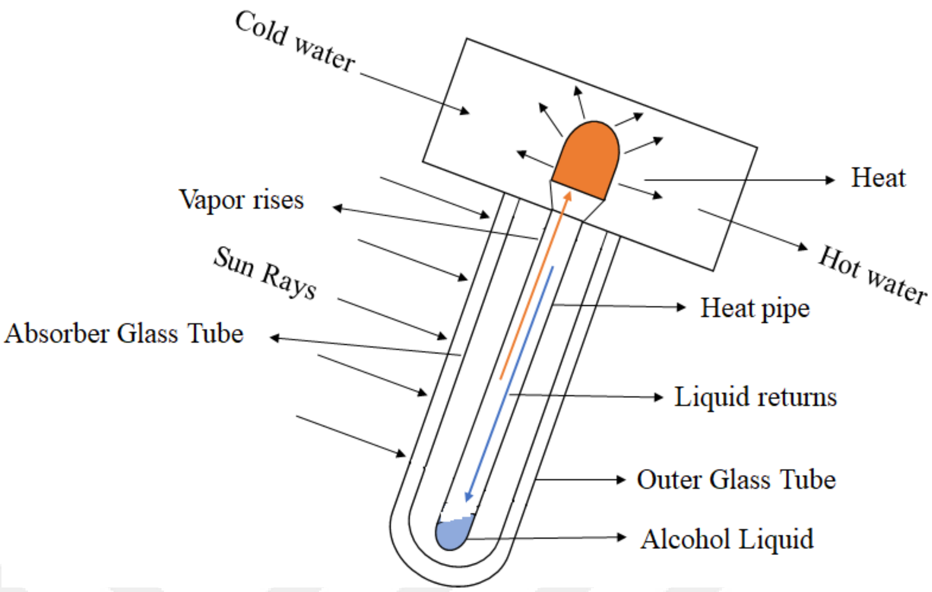

Figure 2.8 A schematic structure of heat pipe evacuated solar collector

The heat pipe is placed concentrically in the heat pipe evacuated tube solar collector. The heat pipe heats up under the influence of sunlight. Therefore, the working fluid in the pipe reaches high temperatures and begins to evaporate. The working fluid, which starts to boil, rises in the heat pipe and reaches the condenser part and is put into use. This cycle continues as long as the collector surface is exposed to radiation. When selecting working fluids in heat pipes, the freezing point and boiling point temperatures of the fluid are taken into consideration. Therefore, working fluids such as methanol, acetone, ammonia is generally used. Since the thin walled, high level corrosion resistance and easy to shape copper is mostly used as heat pipe material.

2.3.3 Compound Parabolic Collectors

Compound parabolic collectors consist of two parts, the reflective and the absorber surface. The reflective surface is called the compound parabolic concentrator and consist of two parabola arms which reflecting sunlight to absorber surface. Because of their parabolic structure, they focus the sun's rays coming from different angles easily on the absorber surface. A compound parabolic collector geometry is illustrated in Figure 2.9.

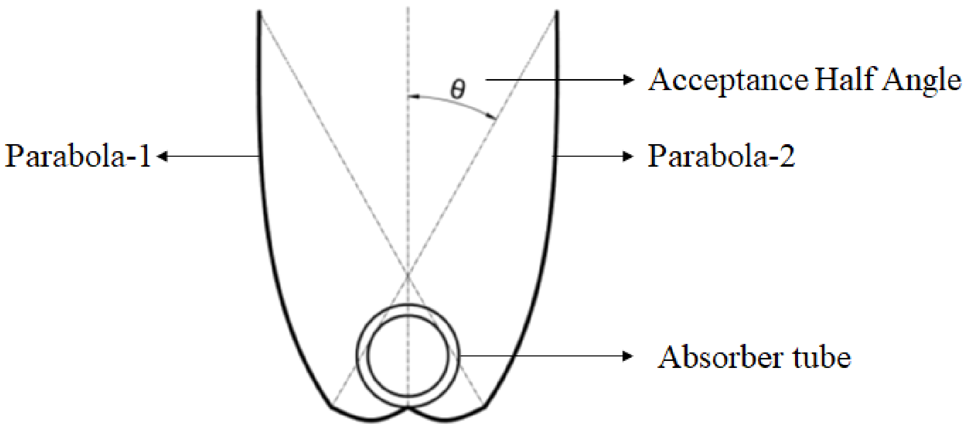

Figure 2.9 Compound parabolic collector geometry

The concentration ratio is defined as the ratio of the reflective surface aperture area to the absorber surface area. This ratio is between 1-5 for compound parabolic collectors. Therefore, they are included in the class of low concentrator solar collectors (Kalogirou, 2013).

Compound parabolic collectors do not require a tracking system like linear Fresnel, parabolic trough collectors which used in high temperature applications. Therefore, the installation costs are lower than those of such collectors. Compound parabolic collectors usually use with tubular or flat absorbers. However, tubular absorbers are generally preferred in high temperature fluid applications. The All glass, U tube or heat pipe evacuated tubes can be used as tubular absorber.

CHAPTER THREE
MODELLING AND HEAT TRANSFER ANALYSIS OF A COMPOUND PARABOLIC COLLECTOR WITH INNER TUBE

In this chapter, modelling and analysis of a CPC with inner tube has been presented in detail. The model is mainly consisting of two parts which optical and thermal analysis of system. In the first part, a CPC model was created and optical analysis of this structure is performed. The purpose of this part is to determine the optical efficiency of the system. In the second part, thermal analysis of the inner tube solar collector was performed. Finally, the energy balance is written of the system and the thermal efficiency of inner tube has been achieved.

3.1 Modelling of CPC Inner Tube and Optical Analysis

Solar angles are important parameters in the calculation of net heat energy transferred to the absorber surface. Some of these parameters were used in this study for calculation of the optical efficiency. For this reason, brief information about these aspects is given below. θ is the angle between the direct ray and normal of the surface, γ is the divergence angle between the local meridian and the normal to the surface, γ_s is between projection of direct ray from the sun on the horizontal surface and the south, β is the angle between a plane and horizontal surface, θ_z is the angle between the vertical and the line to the sun. Also, these angles are shown in Figure 3.1, (Duffie & Beckman, 2013).

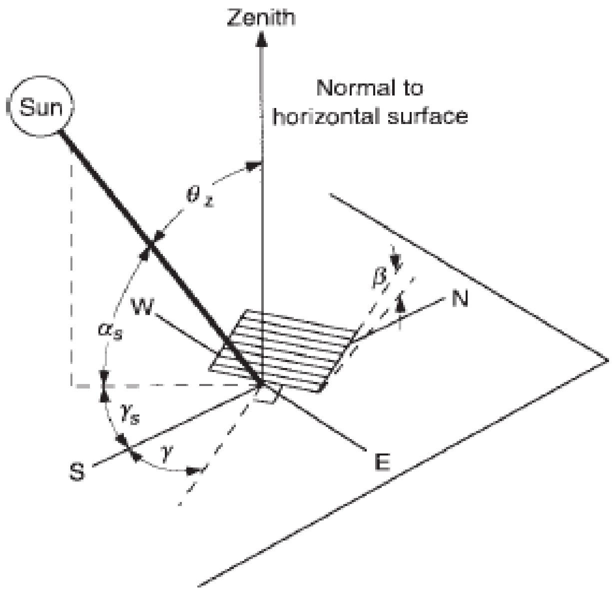

Figure 3.1 Solar angles for a sloped surface (Duffie & Beckman, 2013)

Some parameters such as the irradiation rate to the collector surface depending on the incident angle. Incident angle was calculated with Eq. (3.1).

$$Q_{abs} = \ cos\theta_z cos\beta + sin\theta_z sin\beta cos(\gamma_s - \gamma) \tag{3.1}$$

The acceptance half angle is a main factor of the concentrator surface. The sun rays that are equal or less than the acceptance half angle of is collected on the absorber surface.

Involute reflects the rays which come nearly vertical to the aperture. This part is used only in concentrator's with circular absorber. Tubular absorber, parabolic surfaces, the focal point of left and right parabolas and involute surface of a CPC is given as schematically in Figure 3.2.

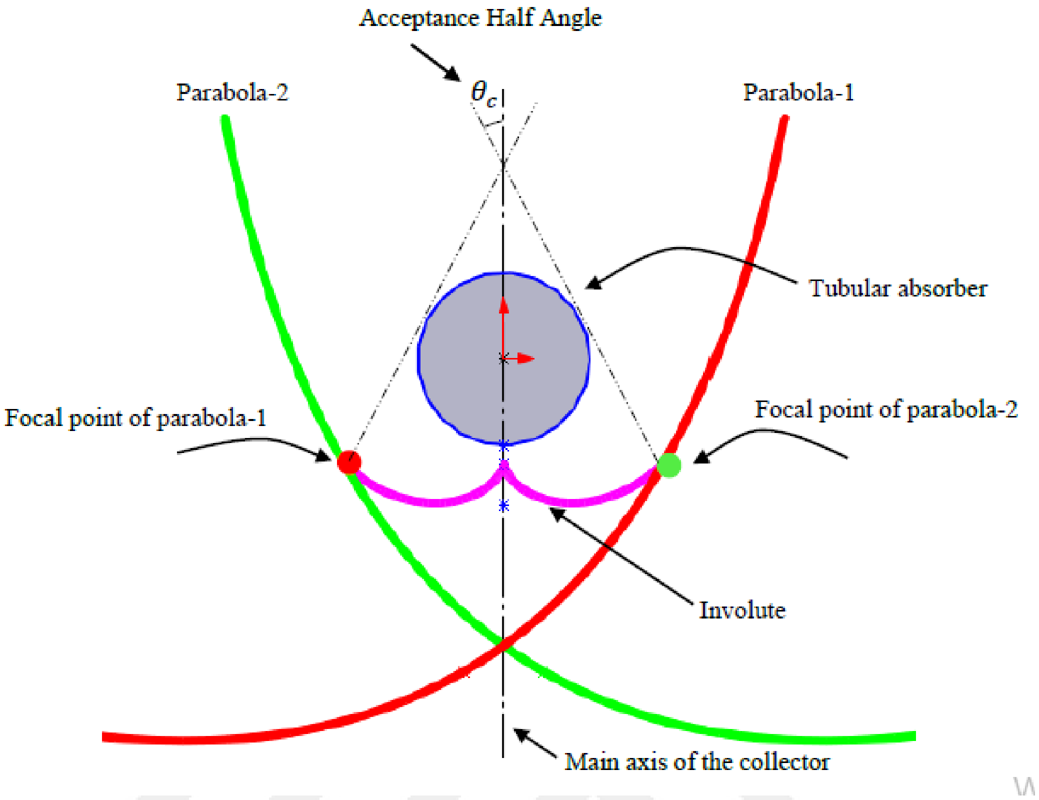

Figure 3.2 A tubular absorber, parabola, focal point and involute in a CPC drawing (Yaman, 2020)

The CPC was designed with a 30° of acceptance half angle and 47 mm absorber diameter. The height of the reflective wings was 237 mm originally. The height of the reflectors is reduced to prevent shadowing to each other. One of third of the reflector height was truncated in this study. Untruncated and truncated reflectors are shown in Figure 3.3 and Figure 3.4, respectively.

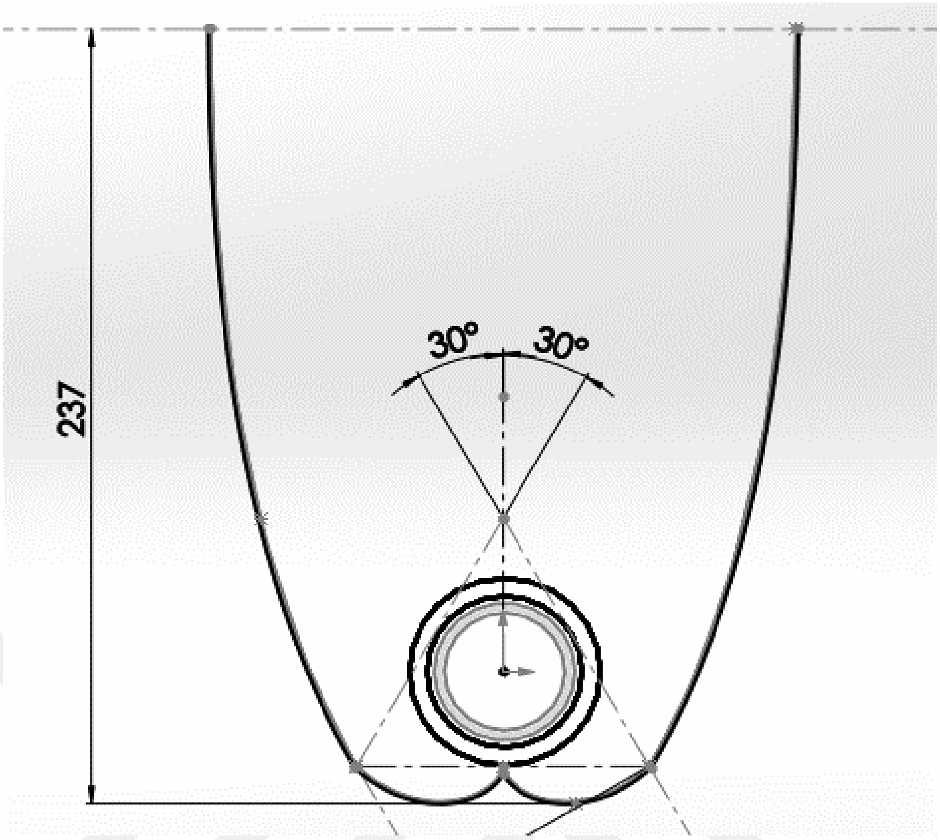

Figure 3.3 Front view of untruncated CPC design

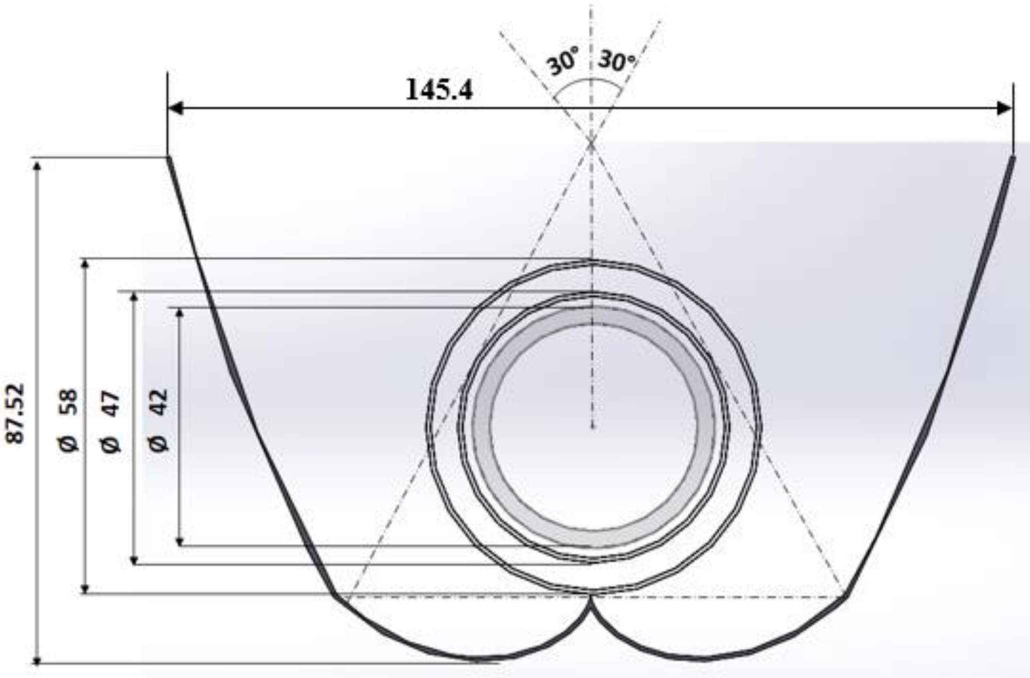

Figure 3.4 Front view of truncated CPC design

The absorbed heat Q_{abs}, is equal to the incident solar power reduced by optical losses, given by

$$Q_{abs} = \eta_{opt} \cdot I_r \cdot A_{CPC} \tag{3.2}$$

where η_{opt} is the optical efficiency, I_r is the global solar irradiance on the reflector plane and A_{CPC} is the aperture area of concentrator in Eq. (3.3). Aperture area is calculated by multiplying length (l) and width (w) of CPC.

$$A_{CPC} = w \cdot l \tag{3.3}$$

The optical efficiency, is defined as the ratio between the absorbed solar power and the total solar power received by the aperture area. The optical efficiency of collector is calculated by (Rachel & Jayaraman 2002):

$$\eta_{opt} = \rho^n \cdot \tau_e \cdot \tau_d \cdot \alpha_i \cdot f_{ref} \cdot p \tag{3.4}$$

where ρ is the reflectance of the CPC material, n is the average number of reflections, τ_e is transmittance of the absorber glass tube, τ_d is the dust stratification factor of evacuated tube, α_i is the absorptance of selective coated material on the absorber, p is the gap loss coefficient due to gap width between outer glass tube and CPC, f_{ref} is multiple reflection coefficients between outer and inner glass tube. Theoretically calculating variables such as the average number of reflections, dust stratification factor and gap loss coefficient is very difficult. Therefore, in order to determine the optical efficiency, ray tracing software is generally used. The parameters which used in software are solar incidence angle, acceptance angle, gap which between concentrator and absorber, and optical properties of reflector surface and glass tubes.

In this study, in order to determine the optical efficiency of the system, firstly the concentrator, virtual ray source and evacuated glass tubes were designed with Solidworks. Concentrator surfaces are defined as reflector, outer surface of inner glass tube is defined as absorber and the outer glass tube is also defined as transmitter and shown in Figure (3.5). The rays which normal to source surface were sent to the aperture area of CPC by observing the sun's incident angle between 0°-30°. Figure 3.6 shows an inner tube solar collector with CPC which was included virtual rays for 0°.

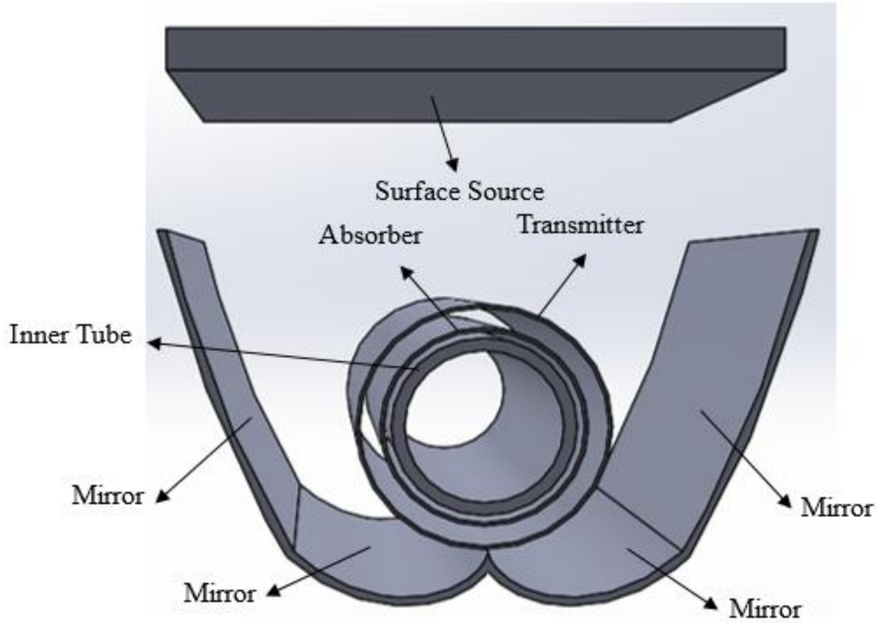

Figure 3.5 Perspective view of CPC with identified surfaces

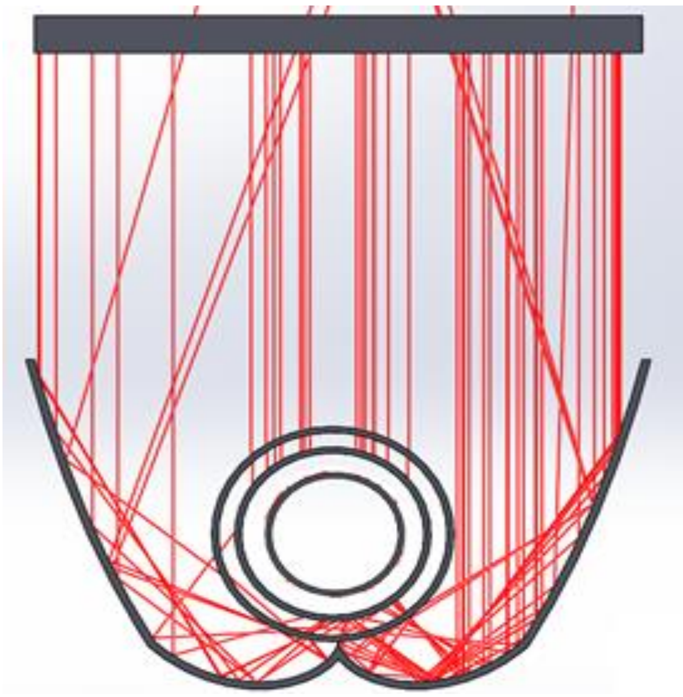

Figure 3.6 Ray paths for incident angle of 0°

After the model was created, optical properties of the surfaces were defined. The coefficients of reflectivity(ρ), transmissivity(τ) and absorptivity(α) were given at Table 3.1.

Table 3.1 The thermophysical properties of CPC

Parameters	Values
Transmissivity of the outer glass tube	0.9
Reflectivity of the concentrator surface	0.8
Absorptivity of the inner glass tube	0.92

The optical properties of the surfaces and the ray paths created by Rayviz software were transferred to TracePro software for determine of the optical efficiency. Optical efficiency values were obtained for 0°-30°. They are given in Table 3.2 and graphed as in Figure 3.7.

Table 3.2 The optical efficiency values

Solar Incident Angle	Optical efficiency
0	0.5857
5	0.5576
10	0.6094
11	0.6355
12	0.6177
13	0.5764
14	0.5573
15	0.5263
16	0.5292
17	0.5536
18	0.5630
19	0.5465
20	0.5546
25	0.6264
30	0.6976

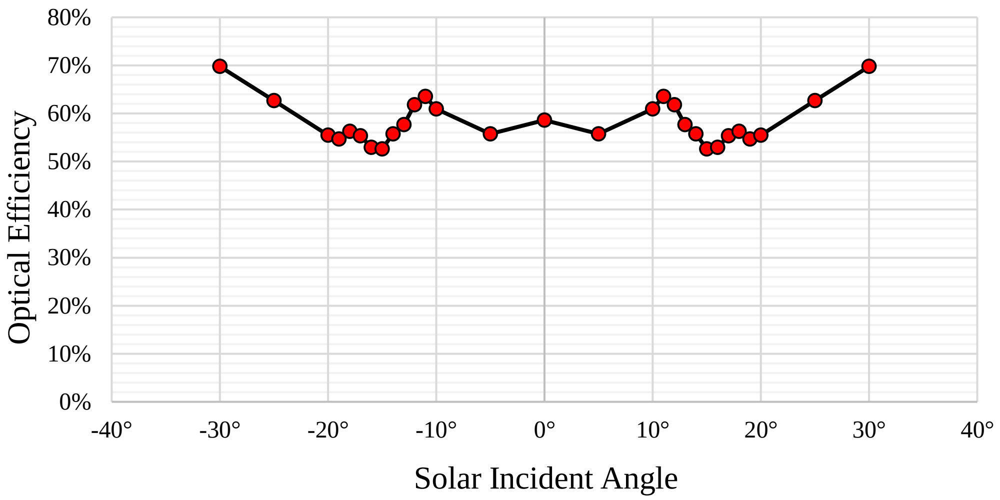

Figure 3.7 Optical efficiency of designed CPC depending on incident angle

3.2 Heat Loss Model of Inner Tube Evacuated Solar Collector

The inner tube evacuated solar collector used in this study which consists of two coaxial glass tubes. A metal inner tube was placed inside of these glass tubes. The gap between the glass tubes is vacuumed with Barium gas. The outer surface of inner glass tubes is coated with selective material which is highly absorbing sunlight. Furthermore, in order to reduce thermal resistance, the space between the absorber glass and the metal inner tube is filled with heat transfer oil. Schematic view of inner tube solar collector without concentrator is given Figure 3.8.

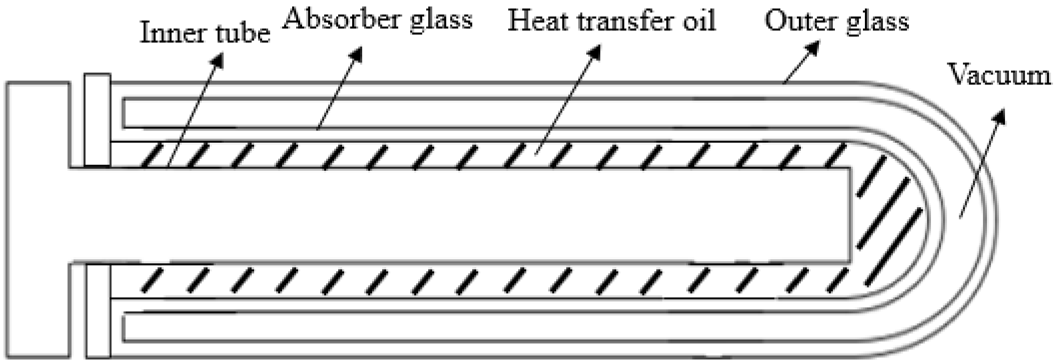

Figure 3.8 Schematic view of inner tube evacuated solar collector without CPC

The following assumptions are used: (a) thermal resistance of the inner and outer glass tubes thickness is negligible; (b) perfect vacuum is assumed between the two glass tubes; (c) the heat flux along the circumferential direction is considered constant; (d) the heat transfer mechanism of the inner tube evacuated solar collector are assumed to be steady.

3.3 Thermal Resistance Network of Inner Tube Evacuated Solar Collector

An energy balance of one-dimensional at steady state was appeared by taking solar energy into the absorber area, thermal losses from this surface and the useful heat transferred to the working fluid. Before start to the thermal calculations; directions of absorbed solar power and the thermal losses from the absorber surface and the temperatures of surfaces are shown in Figure 3.9.

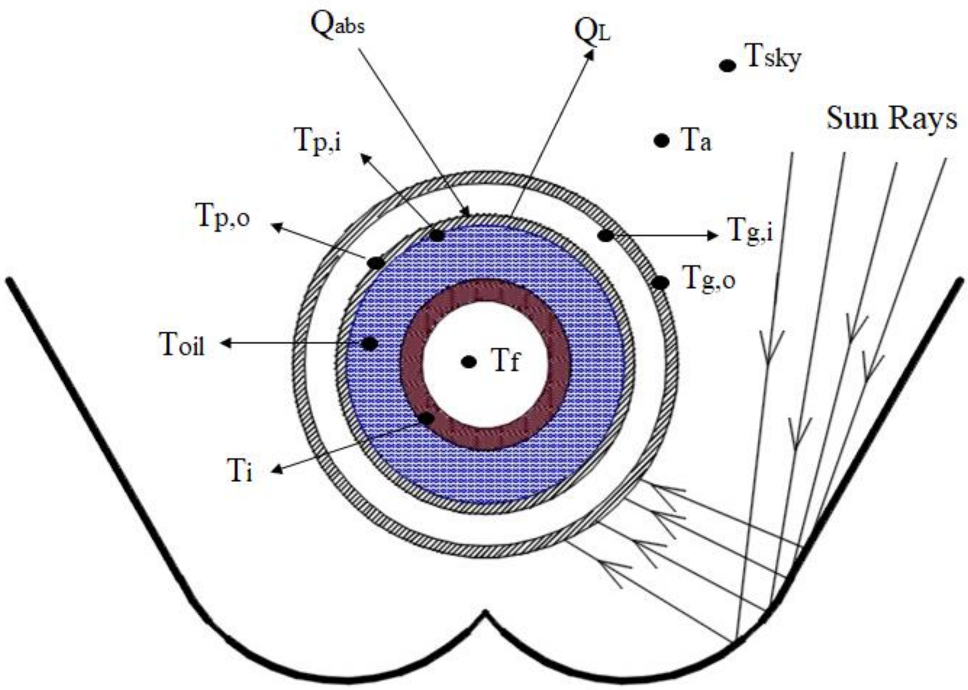

Figure 3.9 Schematic of the inner tube solar collector with reflector

The absorbed solar power Q_{abs}, is the sum of the net heat gained Q_F, by the working fluid and the thermal losses from the absorber surface Q_L, to environment as shown in Eq. (3.5).

$$Q_{abs} = Q_F + Q_L \tag{3.5}$$

The system's thermal resistance network was created while the absorber surface was referenced. Some part of the solar energy coming to this surface is transferred to the working fluid. However, some part of this energy is lost from all systems and pipes by the effect of hot working fluid. Thermal resistance network is given in Figure 3.10.

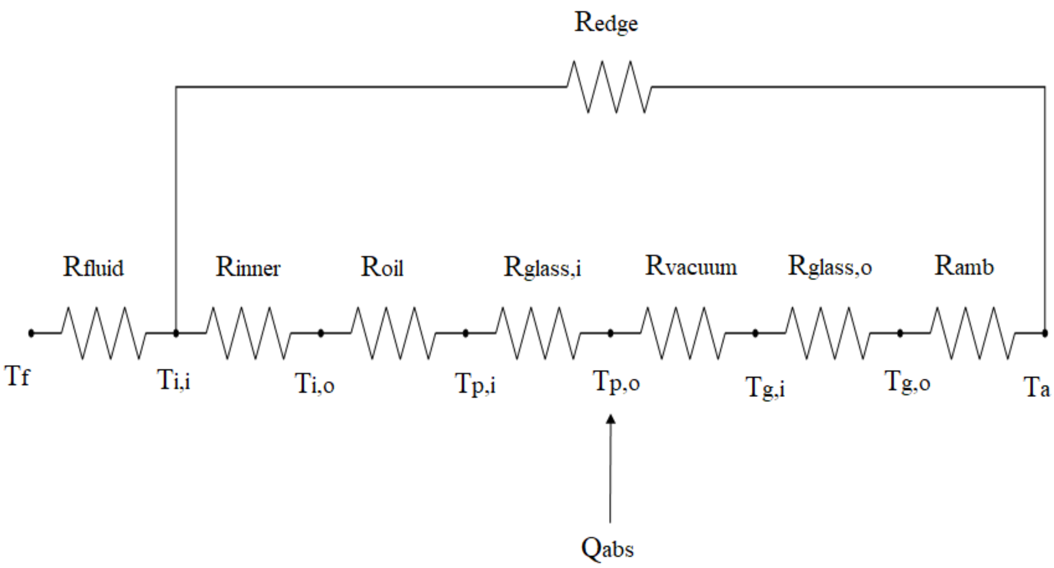

Figure 3.10 Thermal resistance network of inner tube solar collector

Thermal resistances caused by glass tube thicknesses can be neglected. In addition, resistances due to inner pipe thickness and heat transfer oil have usually smaller values than the others. Heat transfer between the inner and the outer glass tubes occurs with radiation and conduction. Also, between the outer glass tube and the ambient occurs with radiation and convection. The new thermal resistance circuit formed by reference to the absorber glass outer surface, A_P, is given at Figure 3.11.

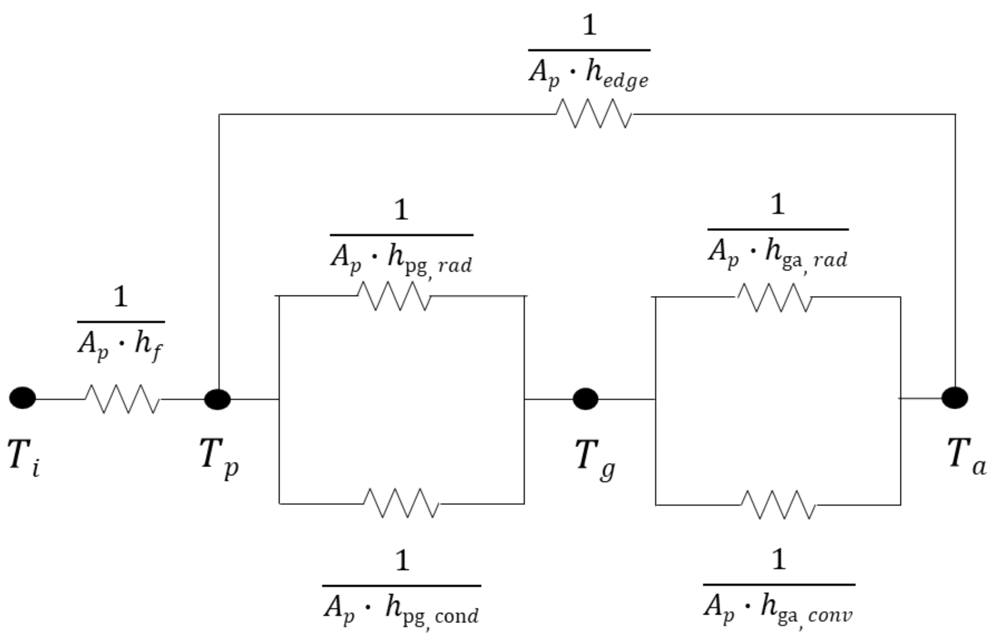

Figure 3.11 Thermal resistance network of inner tube solar collector

The terms used in thermal resistance network: h_f, convection heat transfer coefficient that inside of the inner tube $h_{pg,rad}$, radiation heat transfer coefficient between outer glass tube and environment $h_{ga,conv}$, convection heat transfer coefficient between outer glass tube and environment $h_{ga,rad}$, radiation heat transfer coefficient between outer surface of absorber glass tube and inner surface of outer glass and $h_{Pg,cond}$, is refers to the conduction heat transfer coefficient of metal vacuum getter that located between the outer glass tube and the inner glass tube. Also, h_{edge}, is shown edge loss coefficient between outer surface of inner tube and environment.

3.4 Total Heat Loss Coefficient

Total heat loss to the environment is obtained as follows Eq. (3.6).

$$Q_L = Q_L \cdot (T_p - T_a) \tag{3.6}$$

T_P and T_a values given in this equation are respectively the temperature of the absorber surface and the environment. Also, U_L is the total heat loss coefficient which obtained between outer surface of the absorber glass tube and the environment.

Total heat loss Q_L, is equal to the sum of edge loss from the manifold and connection pipes, Q_{edge}, and heat loss from the absorber surface to the environment, Q_{tube} is given Eq. (3.7).

$$Q_L = Q_{edge} + Q_{tube} \tag{3.7}$$

3.4.1 Edge Loss from Inner Tube to the Environment

Although the insulation, some part of the heat loss throughout manifold and connection pipes in a collector system. This heat loss was calculated with Eq. (3.8).

$$Q_{edge} = U_{edge} \cdot (T_p - T_a) \tag{3.8}$$

Heat loss coefficient from the edges to the environment;

$$U_{edge} = \frac{2 \cdot \pi \cdot k_{ins} \cdot L_e}{ln\left(\frac{D_{ins,o}}{D_{ins,i}}\right)} \tag{3.9}$$

is calculated by the relation in Eq. (3.9). In this equation, L_e is the length of manifold, k_{ins} is the heat conduction coefficient of insulation material, $D_{ins,o}$ and $D_{ins,i}$ respectively, the outer and inner diameters of the insulation pipe, denotes.

If the edge loss coefficient is referenced to the surface area of the absorber glass tube;

$$h_{edge} = \frac{U_{edge}}{A_p} \tag{3.10}$$

Eq. (3.10) is obtained. Also, A_p is calculated as in Eq. (3.11).

$$A_p = \pi \cdot D_p \cdot L_p \tag{3.11}$$

3.4.2 Heat Loss from Absorber Glass Tube to Ambient

The temperature of the absorber glass tube is higher than the environment temperature. Therefore, there is continually heat loss from this surface to the environment. A heat loss coefficient, U_{tube} was determined by taking the absorber surface reference. This coefficient which the heat losses realized by radiation, convection and conduction from the absorber surface to the environment. It can be calculated using the thermal resistance network in Figure 3.12 as follows:

$$U_{tube} = A_p \cdot \left(\frac{1}{h_{pg,cond}+h_{pg,rad}} + \frac{D_p}{h_{ga,conv}+h_{ga,rad}}\right)^{-1} \tag{3.12}$$

The radiation heat transfer coefficient between the absorber and outer glass tubes can be given by Eq. (3.13),

$$h_{pg,rad} = \frac{\sigma \cdot \varepsilon_p}{1+\frac{\varepsilon_p D_p}{\varepsilon_g D_g}(1-\varepsilon_g)}({T_p}^2 + {T_g}^2)(T_p + T_g) \tag{3.13}$$

where ε_p is the emissivity of the selective absorbing coating, ε_g is the emissivity of the inner surface of outer glass tube and is σ the Stefan Boltzmann constant, D_p is the outer diameter of absorber glass tube and is D_g the inner diameter of outer glass tube.

The convection heat transfer between the inner and outer glass tubes has been neglected. Therefore, some of the heat loss is assumed by conduction from the metal vacuum getter. The conduction heat loss by metal vacuum getter is taken as approximately $1.8 \cdot D_g$ W/mK depends on glass tube diameter (Aboulmagd,2014). Then this coefficient is referenced to the absorber surface and adjusted conduction heat transfer coefficient can be given:

$$h_{pg,cond} = 1.8 \cdot \frac{D_p}{A_p} \tag{3.14}$$

with Eq. (3.14).

Heat transfer occurs by radiation and convection between the outer surface of the glass tube and the ambient. The effect of wind speed should also be taken into account when calculating the heat transfer coefficient on this surface. For this reason, convection heat transfer for around of the outer glass tube is;

$$h_{ga,conv} = 0.6 \cdot \frac{D_g}{D_p}\ (5.7 + 3.8 \cdot V_w) \tag{3.15}$$

obtained as in Eq. (3.15). In this equation V_w, is the velocity of the air(wind) in m/s. Radiation heat transfer coefficient between the outer glass tube and the environment;

$$h_{ga,rad} = \sigma \cdot \varepsilon_g \cdot \frac{({T_g}^2+{T_{sky}}^2)(T_g+T_{sky})(T_g-T_{sky})}{T_g-T_a} \cdot \frac{D_g}{D_p} \tag{3.16}$$

calculated using Eq. (3.16). In this equation T_g and T_{sky} represent the outer glass tube temperature and sky temperature, respectively.

The radiation heat loss coefficient depends on the value of T_g. In order to calculate this value, the total heat loss from the outer glass surface of the evacuated tube to the environment is equalized to the heat loss between the inner and outer glass:

$$\left(h_{pg,rad} + h_{pg,conv}\right) \cdot \left(T_p - T_g\right) = \left(h_{ga,rad} + h_{ga,conv}\right) \cdot \left(T_g - T_a\right) \quad (3.17)$$

by iteration with Eq. (3.17), the outer glass temperature and its associated convection coefficients are calculated.

3.5 Heat Flow to the Working Fluid

The heat transfer is carried out by convection from the absorber surface to the working fluid:

$$Q_F = h_f \cdot A_p \cdot (T_p - T_i) \quad (3.18)$$

h_f coefficient is determined by natural convection between the inner pipe surface and the fluid with Eq. (3.18). The Nusselt Number for the natural convection can be determined from Eq. (3.19) (Kriplani & Nimkar, 2008).

$$Nu = C \cdot Ra^m \quad (3.19)$$

In Eq. (3.19), the constants of C and m are accepted as 10^{-9} and 1.442, respectively; and the Rayleigh number changes between 7×10^5 and 5×10^6. In the range of working temperatures, the convection coefficient varies 60 W/m^2K to 180 W/m^2K. This coefficient is converted by referring to the absorber glass tube surface as in Eq. (3.20).

$$h_f = h_{conv} \cdot \left(\frac{D_i}{D_p}\right) \tag{3.20}$$

3.6 Thermal Performance Evaluation of Modelled CPC

Thermal performance is investigated at different working conditions for a CPC whose dimensions were given in Figure (3.4). The aperture width of modelled CPC is 145.4 mm and the length are 1500 mm; in which the aperture area is calculated as 0.218 m^2. At working temperatures in infrared irradiation, emissivity coefficient of outer glass tube can be taken as 0.92. However, absorber tube is coated as a selective material whose emissivity coefficient is 0.08 and absorptivity for solar radiation is 0.92.

Optical efficiency and environment temperature are taken as 60% and 25°C, respectively. The sky temperature is accepted equal to the environment temperature. The convection coefficient between outer glass and environment is 11.25 $W/m^2°C$, and for working fluid is accepted 100 $W/m^2°C$. Edge loss through the connection pipes is neglected.

The heat transfer rate to the working fluid at changing temperatures for 400-600-800-1000 W/m^2 of radiation intensities as shown in Figure 3.12.

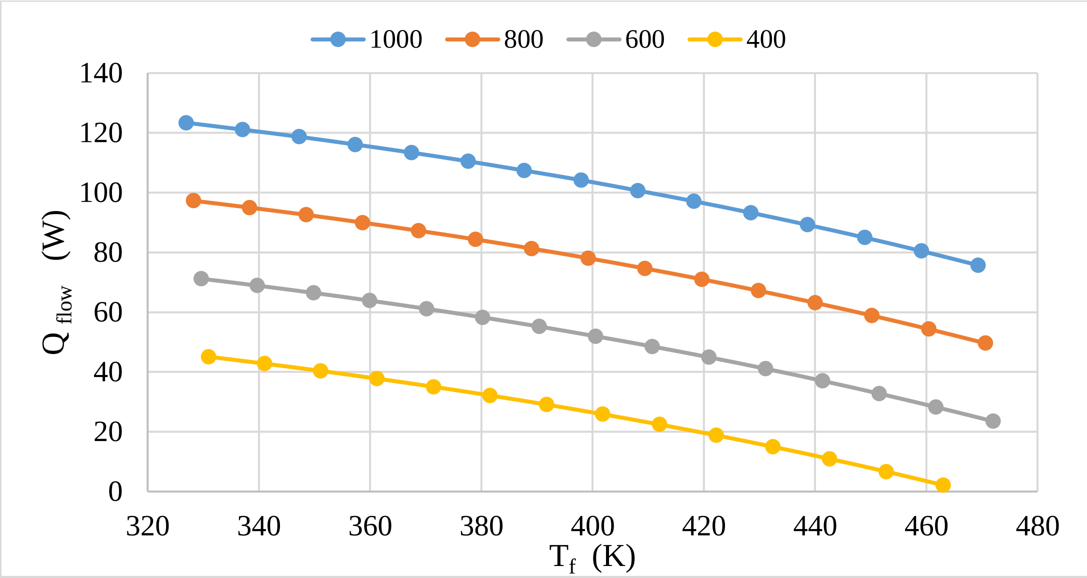

Figure 3.12 Heat flow curves of the working fluid

The thermal efficiency is defined as the ratio of the heat transferred to the working fluid and the radiation intensity that come to aperture area of the CPC. The relation between thermal efficiency and working fluid was given in the Figure 3.13.

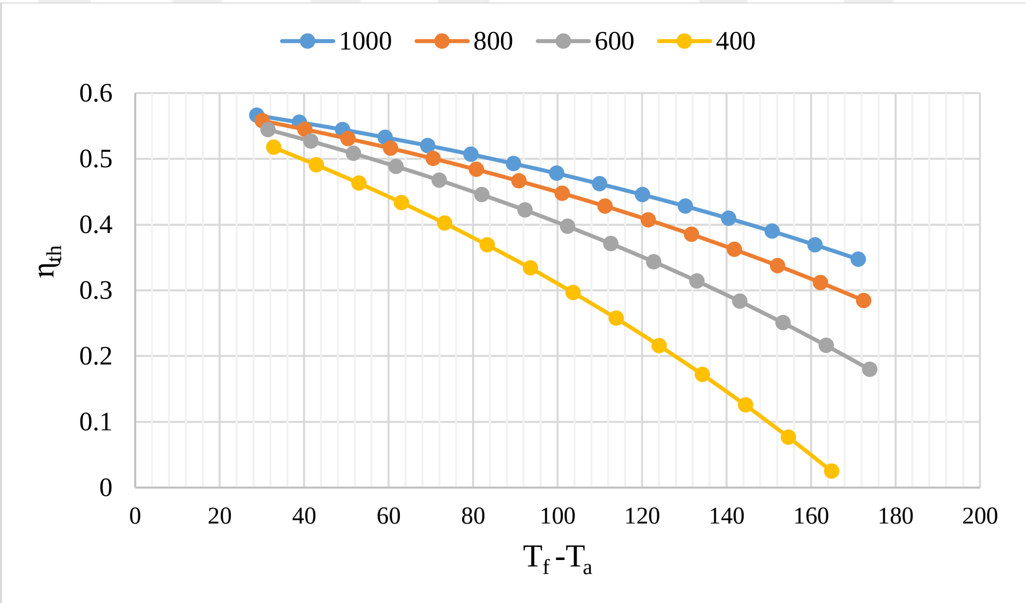

Figure 3.13 Thermal efficiency curves of CPC

As expected, thermal efficiency decreases rapidly depending on amount of radiation intensity and increasing with fluid temperature. The stagnation temperature was found as 466.3 K (193.15°C) at 400 W/m^2. This value gives an idea about the highest temperatures that can be achieved via CPC systems.

CHAPTER FOUR
EXPERIMENTAL STUDY FOR STEAM GENERATION

A detailed information about a prototype of the CPC with inner tube is given in this section. In addition, the components that set up the system, the material properties and design processes are mentioned. Experimental setup was realized. Also, some information about the environment conditions of the experiments and the equipment used in the experimental studies are given. Finally, by conducting tests for the generation of steam, thermal losses and thermal efficiency of the system were obtained.

4.1 Design of the CPC Prototype

A CPC prototype consists of a metal inner tube, vacuum glass tubes and parabolic concentrator surfaces. Evacuated glass tubes are usually produced with a length 1800 mm or 2100 mm. However, a special evacuated glass tube was designed which the length of 1500 mm in this study. The absorber glass tube and outer glass tube which concentrically placed. Outer surface of the absorber glass tube is coated with selective material of (AL-N\AL) which has a high absorption coefficient and low emitting coefficient. The outer glass tube and absorber glass tube diameters which used in this prototype are 58 mm and 47 mm respectively. Furthermore, the space between these two glass tubes was vacuumed. The specifications and dimensions of the evacuated glass tube are given in Table 4.1.

Table 4.1 The glass tubes specifications

Specifications	Values
Length	1500 mm
Outer diameter of the outer glass tube	58 mm
Outer diameter of the absorber glass tube	47 mm
The glasses thickness	1.6 mm
Selective coating material	AL-N\AL
Material	Borosilicate 3.3.
Pressure	P<0.005 Pa

A compound parabolic concentrator has designed to increase the amount of solar radiation coming to the absorber surface. Considering the glass tubes length, a concentrator surface which 1500 mm length and 145.4 mm width was designed as shown in Figure 4.1. Anodized Aluminium was chosen as the concentrator surface material. The concentrator surface properties are given in Table 4.2.

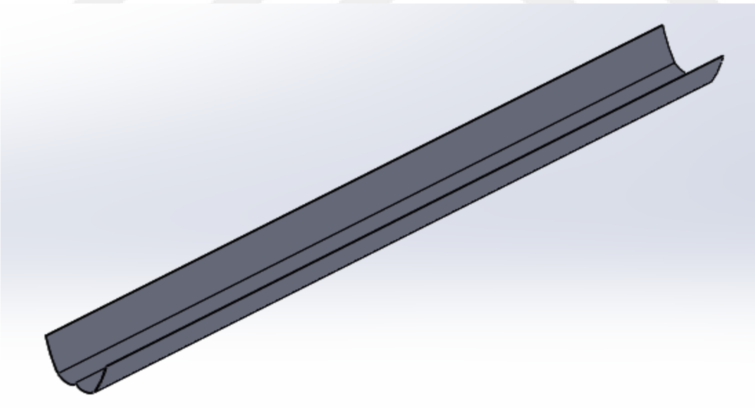

Figure 4.1 Isometric view of the compound parabolic concentrator surface

Table 4.2 The concentrator surface specifications

Specifications	Values
Length	1500 mm
Width	145.4 mm
Thickness	0.7 mm
Material	Anodized Aluminium

The metal inner tube is designed with a length of 1420 mm according to the absorber glass tube length. The tube material was selected as St-37 steel because of its low cost. While determining diameter of this tube, the gap between metal inner tube and absorber glass tube was observed. Therefore, the outer and inner diameter of the metal inner tube were determined by 36 mm and 42 mm respectively. Down side of the tube is closed and upper side was connected to a horizontal manifold. This part was used to fill the system with water and to collect the steam generated in tube. Specifications and dimensions of metal inner tube are given in detailed in Table 4.3. Also, front view of the metal inner tube design is shown in Figure 4.2.

Table 4.3 Metal inner tube properties and dimensions

Specifications	Values
Length	1410 mm
Manifold Length	300 mm
Outer diameter	42 mm
Inner diameter	36 mm
Thickness	3 mm
Material	St-37

Figure 4.2 Isometric view of the metal inner tube design

The prototype was completed by placing vacuum tubes and metal inner tube concentrically and parallel to the concentrator surface as shown in Figure 4.3.

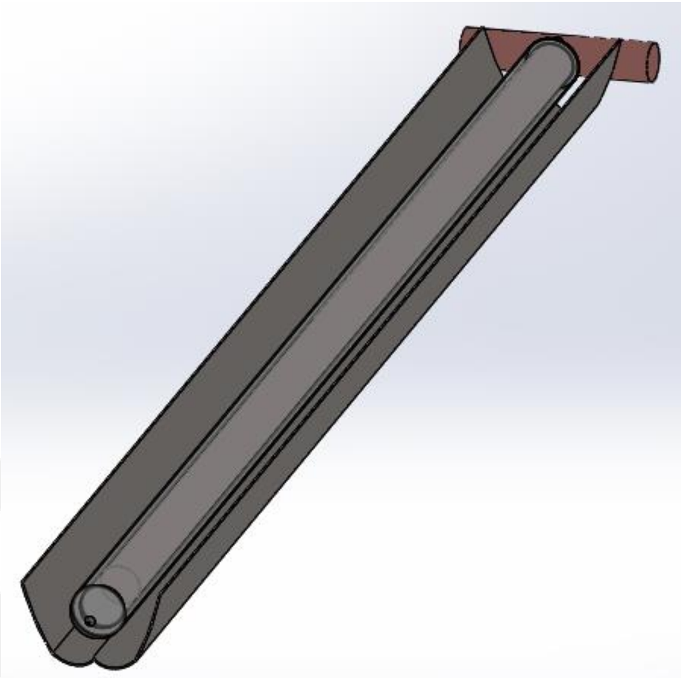

Figure 4.3 Isometric view of the compound parabolic collector design

4.2 Installation of Experimental Study

The inner tube evacuated solar collector with concentrator surface was manufactured. This collector is mounted on a platform with a slope angle of 45°. The gap between the metal inner tube and the absorber glass tube is filled with olive oil. A double pipe heat exchanger is mounted on the steam outlet of the metal inner tube. Right-side view of CPC is shown in Figure 4.4.

Figure 4.4 Right-side view of CPC (Personal archive, 2018)

T-type thermocouples are used for the temperature measurement of the metal inner tube’s outer surface, outer surface of the outer glass tube and the ambient. One of them was placed on the outer surface of outer glass tube, other one was placed into olive oil to measure the temperature on inner tube surface, and the last one was used the temperature of ambient. Solar radiation intensity that come to the collector surface was measured by LP Silicon PYRA-04 which located on the same slope angle as this surface. The measured radiation intensity, surface and ambient temperature values were transferred to the computer with data logger. The components that make up the experimental setup are shown schematically in Figure 4.5.

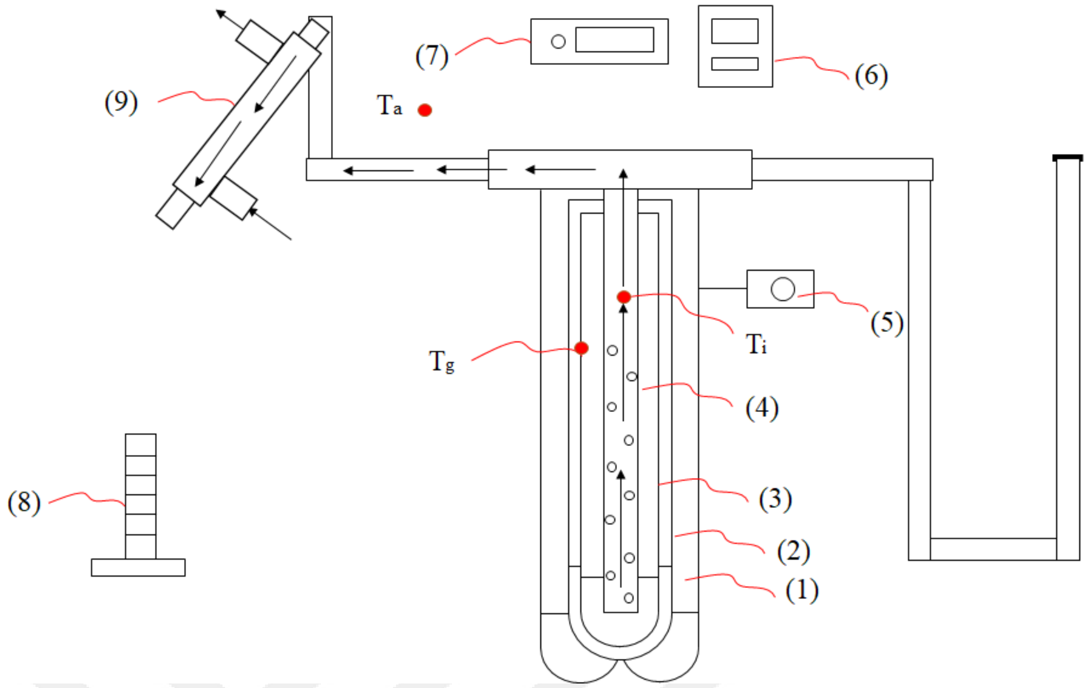

Figure 4.5 Schematic view of the inner tube solar collector with CPC

In Figure 4.5; (5) is pyranometer, (7) is data logger, (6) is computer, (8) is cylindrical measuring vessel, (1) is compound parabolic concentrator surface, (2) is the outer glass tube, (3) is the absorber glass tube, (4) is the metal inner tube, (9) dual pipe heat exchanger. T_i is temperature of metal inner tube surface, T_g is temperature of outer glass tube surface and T_a is the ambient temperature. The steam in the inner tube is condensed in the double pipe heat exchanger and taken out from the left.

4.3 Application of the Experimental Tests

The CPC was placed where the coordinates are 38°22’03.1’’N and 27°12’13.3’’E in Buca, İzmir. Manifold part of the metal inner tube insulated with 5 mm thickness polyethylene material.

At the beginning of the experiment, water was filled to the system at a temperature of about 65 °C. Due to the effect of solar radiation, the water temperature which in the metal inner tube has slowly increased. After that, steam generated in the tube directed to the outlet and condensed in the double pipe heat exchanger. The condense was collected in a cylindrical measuring vessel. During the experiments, temperature and radiation measurements were recorded by data logger. The amount of steam produced was measured and thermal efficiency of the system was calculated. Front view of the experimental setup was shown in Figure 4.6.

Figure 4.6 Front view of experimental setup (Personal archive, 2018)

4.4 Experimental Results

First two experiments conducted on November 8th, 2018. Temperature and irradiation measurements were saved in 5-minute periods and recorded on the data logger.

Steam evaporation started around at 13:00 in the first experiment. Measurements were saved between 13:09-13:39 and time-dependent variations of the temperature and solar radiation data are given in Table 4.4. The average temperature of the oil which between metal inner tube and absorber glass tube was about 102.43°C. The average radiation intensity was calculated as 948.63 W/m^2 at experiment hours. Also, the average ambient temperature and the outer glass tube surface temperature were measured as 24.83°C and 30.16°C, respectively. Amount of condensed steam was measured as 40.2 g in 30 minutes and evaporation heat is calculated as 90.73 kJ and mean value of this heat calculated as 50.41 W.

Table 4.4 Measurement results per 5-minute intervals for Experiments-1

Time	Metal Inner tube surface, °C	Environment Temperature, °C	Outer Glass Surface Temperature, °C	Radiation Intensity, W/m^2
13:09	103.1	24.6	26.4	956.1
13:14	103.2	24.5	29.3	947.2
13:19	102.2	25.6	31.6	951.6
13:24	102.1	25.0	28.9	947.3
13:29	102.1	24.5	31.0	950.4
13:34	102.2	25.3	31.4	944.2
13:39	102.1	24.3	32.5	943.6

According to data, solar energy which coming to the aperture area of CPC during experiment-1 was 372.41 kJ and 206.90 W, as a mean value. So, thermal efficiency of the system was obtained as 24.36%.

The total heat loss from absorber glass tube to the environment was calculated as 17.82 W for the temperatures of experiment-1. The optical efficiency of the CPC system has an average value of 58.7% during experiment hours. The rates of solar radiation, absorbed heat, total heat loss from glass tube and heating rate for steam generation is also given in Table 4.5. The different between absorbed heat and the others can be evaluated as other heat loss.

Table 4.5 Heat losses in the system for Experiment-1

Total solar irradiation (W)	206.90
Absorbed solar power (W)	119.59
Heat loss from absorber surface to ambient (W)	17.82
Generated steam energy (W)	50.41
Other heat loss (W)	51.36
Thermal efficiency	24.36%

Measurements were saved for second experiment between 13:42-14:12 and time-dependent variations of the temperature and solar radiation data are given in Table 4.6. The average temperature of the oil which between metal inner tube and absorber glass tube was about 102.04°C. The average radiation intensity was calculated as 929.83 W/m^2 at experiment hours. Also, the average ambient temperature and the outer glass tube surface temperature were measured as 25.29°C and 31.03°C, respectively. Amount of condensed steam was measured as 41.3. g, in 30 minutes and evaporation heat is calculated as 93.44 kJ and mean value of this heat calculated as 51.91 W.

Table 4.6 Measurement results per 5-minute intervals for Experiments-2

Time	Metal Inner tube surface, °C	Environment Temperature, °C	Outer Glass Surface Temperature, °C	Radiation Intensity, W/m^2
13:42	102.0	24.3	32.5	935.4
13:47	102.0	25.5	32.6	937.3
13:52	102.1	25.3	30.3	935.8
13:57	102.0	25.3	29.3	932.6
14:02	102.0	25.3	30.8	925.2
14:07	102.1	25.7	30.1	925.1
14:12	102.1	25.6	31.6	917.4

According to data, solar energy which coming to the aperture area of CPC during experiment-2 was 365.03 kJ and 202.80 W, as a mean value. So, thermal efficiency of the system was obtained as 25.6%.

The total heat loss from absorber glass tube to the environment was calculated as 17.63 W for the temperatures of experiment-2. The optical efficiency of the CPC system has an average value of 54.9% during experiment hours. The rates of solar radiation, absorbed heat, total heat loss from glass tube and heating rate for steam generation is also given in Table 4.7. The different between absorbed heat and the others can be evaluated as other heat loss.

Table 4.7 Heat losses in the system for Experiment-2

Total solar irradiation (W)	202.80
Absorbed solar power (W)	111.34
Heat loss from absorber surface to ambient (W)	17.63
Generated steam energy (W)	51.49
Other heat loss (W)	42.22
Thermal efficiency	25.6%

In the general evaluation of the test results, it is seen that the optical efficiency is 55% and the steam production thermal efficiency is 25%. The thermal losses from the evacuated glass tube are 9%, and the edge losses from the manifold to the environment are 21%. It will be possible to increase the steam production efficiency slightly by reducing the edge losses.

On the other hand, the total mass of water where in the inner pipe is 1500gr and the rate of evaporated water is 3%. This indicates that the total water volume must be reduced to ensure effective steam production.

CHAPTER FIVE
CONCLUSION

A compound parabolic collector with inner tube was designed and tested in this study. A heat loss model was developed via EES, and heat performance of the CPC was derived at different beam radiation and fluid temperature. Thermal efficiency was reached to 30% above 800 W/m^2 of beam radiation for 200°C of fluid temperature.

A prototype of the CPC with the inner tube was manufactured and the availability of the inner tube carrier system for steam production has been evaluated. Thermal efficiency at the first two experiments was calculated approximately 25%.

The results show that, the CPC systems are able to produce fluid at 200°C or above. These systems can be also generated steam as equipped with inner tube.

Some improvements should be made to achieve an acceptable performance of the system:

1- Since heat loss from insulated distribution pipe to the environment is proportional to the length of the pipe, a shorter connection pipes should be used.
2- By improving the reflective material properties, the amount of beam falling on the collector surface can be increased.

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