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**EXPERIMENTAL AND NUMERICAL STUDY OF
SOLAR WATER HEATER ENHANCEMENT
INTEGRATED WITH PHASE CHANGE MATERIAL
AND HEAT EXCHANGER**

Fahad Saadoon Fahad AL-BEHADILI

Master`s Thesis

Supervisor

Asst. Prof. Dr. İbrahim KOÇ

Istanbul, 2022

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Signature

DEDICATION

I dedicate this humble research to my dear parents who have done everything in their power since my childhood to push me forward for educational attainment, as well as my beloved wife and children who have devoted their efforts to creating the appropriate environment and creating an encouraging atmosphere to continue research and analysis, as well as to every researcher looking forward to collecting information, which I tried as much as possible to collect as much as possible how much can be related to the topic of research.



PREFACE

Before everything, I would like to thank God for endowing me with health, knowledge, and aloe to complete this work.

I would like to extend my sincere thanks to Assistant Professor Dr. İbrahim KOÇ, lecturer at Altınbaş University, Department of Mechanical Engineering, who was my supervisor during the writing of this research and who gave me the right direction by providing much correct advice, instructions, and information. I would also like to thank Associate Professor Dr. Süleyman BAŞTÜRK, Head of the Mechanical Engineering Department at Altınbaş University, who in turn assisted immediately to solve all technical problems from laboratories and others. As well as thanks to all the friends who supported, encouraged me and gave me valuable advice and information.

ABSTRACT

EXPERIMENTAL AND NUMERICAL STUDY OF SOLAR WATER HEATER ENHANCEMENT INTEGRATED WITH PHASE CHANGE MATERIAL AND HEAT EXCHANGER

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In current work a practical and numerical study to improve solar energy storage by using PCM and helical coil tube heat exchangers. These components were incorporated into flat solar plate collectors and storage tanks. Phase change material covered the heating water pipe inside the collector, while suitable capsules were fixed inside the storage tank. The heat exchange using helical coils was added to the storage tank as an external load.

The experimental trials were conducted in four separate months (September 2021, April, May, and June 2022), especially on the first day of every month. The experimental section of the present work was conducted in Basra city in Iraq (30.50 N Lat., 47.780 E Long.). During the course of the study, the influence of factors such as air and water temperature, the amount of solar radiation, and the volume flow rate of inlet water were analyzed.

For numerical study, ANSYS-FLUENT 2020 R1 was used to employ the finite volume method FVM for resolving Navier-Stokes, energy, and standard $k-\epsilon$ model equations to assess system performance and the influence of flow rate and input temperature on the system. The effectiveness, heat gain, and significance of the phase change material in increasing heating efficiency throughout the day were studied using a range of variables, including inlet water

volume flow rate (2, 3, 4, 6, and 8 L/min) and inlet water temperature (25, 30, and 35 °C) respectively.

The results illustrate that a low starting water temperature of 25°C enhances the daily efficiency range from 58% to 65% and that the daily final collector outlet temperature was over 65 °C. The results also indicate a noticeable increase in the time of water heating in the absence of the sun due to the heat emitted by the PCM, which can be observed in the evening hours.

Finally, the comparison between the practical and numerical works shows well compatibility between both works where the error in between both works does pass (2 %) assuring the validity of the used numerical study.

Keywords: Solar energy, Phase Change Material, Experimental Study, Numerical Study, Heat Exchanger.

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ABBREVIATIONS

CFD	: Computational Fluid Dynamics
CPC	: Compound Parabolic Collectors
CSP	: Concentrated Solar Power
DPSC	: Dimple Plate Solar Collector
DHW	: Domestic Hot Water
ETC	: Evacuated Tube Collectors
FPSC	: Flat Plate Solar Collector
FVM	: Finite Volume Method
FPC	: Flat Plate Collectors
LHS	: Latent Heat Storage
HTF	: Heat Transfer Fluid
PCM	: Phase Change Material
PV	: Photo Voltaic
PW	: Paraffin Wax
SIMPLE	: Semi-Implicit Method for Pressure Linked Equations
SHW	: Solar Water Heating
SHS	: Sensible Heat Storage
TES	: Thermal Energy Storage

LIST OF SYMBOLS

μ	:	Dynamic Viscosity
E_g	:	Emissivity of Glass
E_p	:	Emissivity of Plate
ρ	:	Density
α	:	Thermal Diffusivity'
β	:	Thermal Expansion
Θ	:	Solar Collector Slope
σ	:	Boltzmann Constant
η	:	Efficiency
ε	:	Rate of Dissipation

1. INTRODUCTION

1.1 INTRODUCTION

Some consider hot water to be mostly a luxury enjoyed by the developed world and has little practical use in the field of appropriate technology. Water is heated using a variety of heat sources. According to one source on the topic, the "3rd World" can heat the hot water that is required by using fuels like wood or fossil fuels that also heat the house and water. Such dismissals are dangerous on two fronts [1].

While wood or fossil fuels do heat both water and the home, they are also natural resources that are not readily available in many developing nations, which is why appropriate technology attempts to reduce waste and boost efficiency in the use of natural resources. The leading cause of death in third-world countries, ahead of AIDS and famine, is long illness, which is brought on by poor ventilation and too much smoke from cooking fires. In contrast, the sun is always there and continues to produce energy whether or not it is used.

The availability of a cost-effective method to obtain hot water within the constraints imposed by the available local resources is also essential whenever there is a need for it. If this technology began to be widely used, the ecological impact that traditional methods of heating water have on a region would be significantly reduced [1].

1.2 RENEWABLE POWER SOURCES

Sustainable energy is defined as never exhausting or having no bounds, like the sun. When the term "alternative energy" is used, renewable energy sources are commonly included. It refers to energy sources that can be used in place of the most commonly used non-sustainable sources, such as coal. As shown in Figure 1.1, the most widely employed at the now, renewable energy sources are [2]: solar, wind, geothermal power, tidal, and biomass energy.

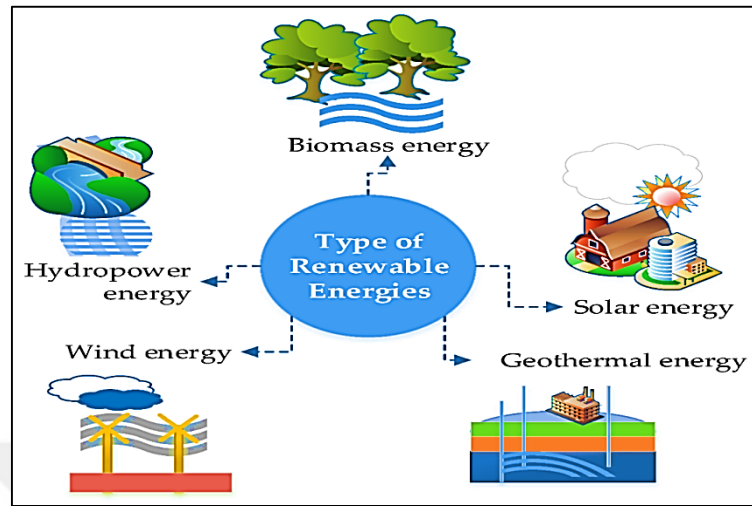


Figure 1.1: Examples of renewable energy resources [3].

The increasing number of people increases need for power to run our homes, businesses, and communities. Innovation and increasing the use of renewable energy sources are necessary to maintain a sustainable energy level and prevent climate change. Currently, 26% of the world's electricity comes from renewable sources, but the International Energy Agency (IEA) predicts by 2024, that percentage rise to 30% [3].

1.3 SOLAR POWER

The most available renewable energy source is solar energy. 60% of the total energy which the sun transmits to the earth is heat energy [2]. The foundation of all recognized energy resources like wind energy, fossil fuels, and others is solar energy. Also, solar energy is converted into electricity (photovoltaic (PV) electricity) and thermal energy [4]. Figure 1.2 shows the solar radiation spectrum with different wavelengths.

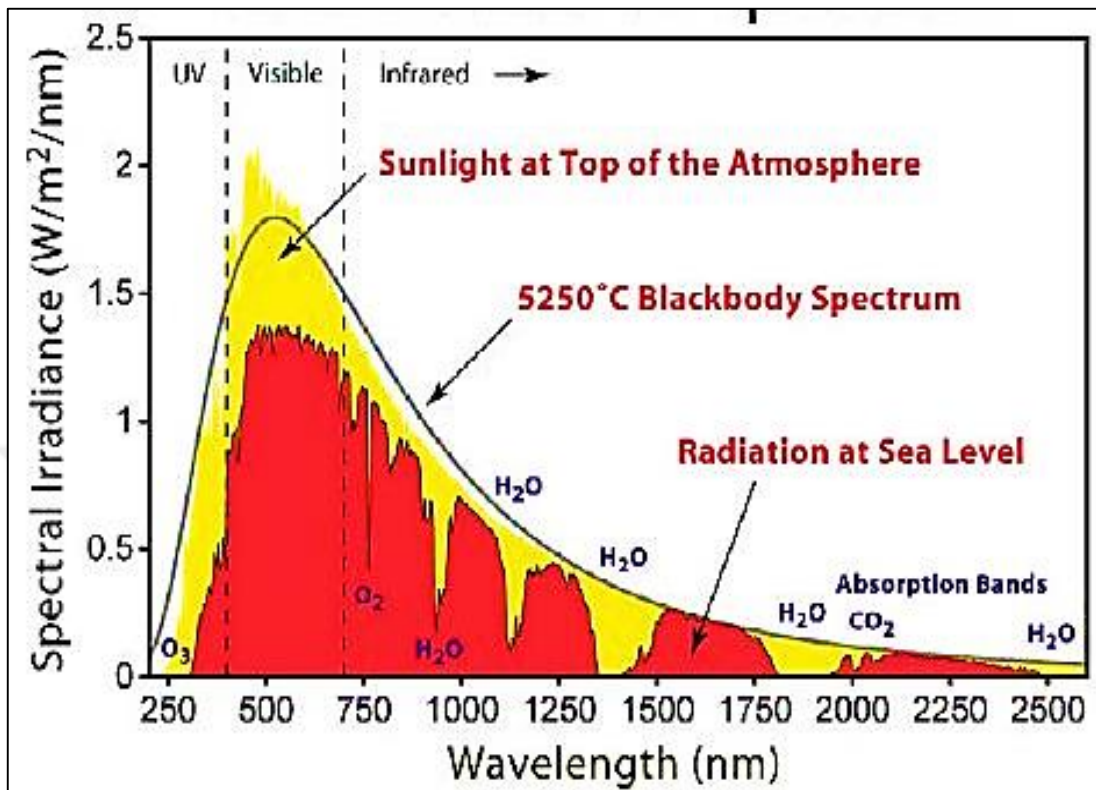


Figure 1.2: Solar radiation spectrum [4]

There are two issues with the use of renewable energy: first the high price of output, and second, the low performance. Increasing the number of instruments that generate this energy may solve the first problem. The second issue can be overcome by using vast amounts of available and free solar energy [5].

Solar thermal technology can either be:

- i. A concentrating technology
- ii. A non-concentrating technology.

Systems that may generate solar energy by reflecting a lot of sunlight onto a receiver are known as concentrated solar power (CSP), also known as concentrating solar power or concentrated solar thermal.

In 2018, the overall installed capacity of CSP increased to 5,500 MW from 354 MW in 2005. Despite the fact that no new capacity has been put into commercial operation in Spain since

2013, the country still accounted for nearly at 2,300 MW, it has half the capacity, USA comes in second with 1,740 MW. There is also a lot of interest in the Middle East, North Africa, India, and China. Parabolic-trough plants, which at one point made up 90% of CSP plants, initially controlled the global market. Due to the higher temperature operation of central power tower CSP, up to 565 °C vs. trough's maximum of 400 °C, which promises greater efficiency, from roughly 2010, it has been preferred in new plants [5].

In a non-concentrating solar collector, they are collecting solar energy and absorbing solar radiation are identical in non-concentrating type collectors. They go by the name "flat plate collectors." With the annual production expected to increase from 40 GW in 2013 to 500 GW by 2020, this market is expected to develop at the quickest rate. China dominates the market for photovoltaic (PV) cell production, which is a billion-dollar industry. With a 37 GW installed capacity, Germany is one of the top nations for installing photovoltaics (PV) nationwide. It takes the lead, followed closely by China, the US of America, Japan, and Italy. Renewable energy sources are the solution to the world's expanding energy needs because fossil fuel costs are always fluctuating and tragedies like Fukushima and Chornobyl are raising major concerns about nuclear power. Water is the other abundant renewable energy source; hydropower raises environmental concerns. Every year, the Earth receives 174 Petawatts of solar energy. It is the planet's biggest source of energy. Solar energy farms may be quickly created that can harness electricity, and the produced electricity is simply given to the grid [5]. To produce energy from other resources such as oil, gas, water, coal, etc., a lot of work and processes are required.

1.4 SOLAR ENERGY APPLICATION

When alien solar radiation moves through the atmosphere, some of it is scattered, some are reflected into space, and some are absorbed. Beam radiation refers to the solar radiation that reaches the surface of the planet while diffuse radiation belongs to the scattered radiation from the sky that reaches the surface. Global or terrestrial solar radiation is the sum both of beam and diffuse radiation. Sunshine is the power produced by absorbing the Sun's heat and light. The photovoltaic (PV) behavior is the part used to describe the team generating electricity from the

sunshine. Utilizing a semiconductor material is accomplished. The other form of obtaining solar energy is through thermal technologies, which are explained below [6].

1.4.1 Solar Water Heating (SWH)

A metal flat plate collector that has been blackened and connected to metal tubing that faces the general direction of the sun makes up an SWH unit. Underneath the thermal insulation is a transparent glass cover that covers the plate collector. A pipe connects the collector's metal tubing to an insulated tank that holds hot water during gloomy days. By absorbing sun rays, the collector then uses gravity or a pump to transmit the heat to the water that is circulated via the tube. Through the connected metal tubing, the storage tank collects this hot water. In addition to domestic and commercial buildings, this method of water heating is frequently employed in hotels, guest homes, tourist bungalows, hospitals, and canteens. Application of solar heating as [6]:

- i. Solar building heating.
- ii. Distillation using the sun.
- iii. Solar pumping.
- iv. Drying food and animal products with the sun.
- v. Solar furnaces.
- vi. Solar cooking.
- vii. Solar ventilation.

1.4.2 Solar Electricity Production

Using PV cells, electric energy may be generated directly from solar energy. A mechanism for converting energy, the PV cell turns solar photons directly into electricity. It is built of semiconductors, which absorb photons from the sun and release highly energetic free electrons as a result. An electric field causes these highly energetic free electrons to flow out of the semiconductor and perform valuable work. In solar cells, this electric field is often created by a p-n junction of materials with various electrical characteristics. To make these cells as efficient

as possible, many fabrication techniques are used. Applications for solar electricity generation include [6]:

- A. Solar transportation
- B. Solar power plant

1.5 SOLAR COLLECTOR

The solar collector roof is mainly constructed using PVC sheets or glass. It is mainly placed at a suitable high above the ground and with an inclined angle according to the longitude and latitude of the city's location. When the sun ray falls on the roof of the solar collector, most of the radiation is absorbed into the downstream layer. The amount of ray intensity absorbed and transmitted by the collector cover depends on the covering material's optical characteristics. Solar collectors are divided into many types [7].

1.5.1 Flat Solar Collector Plates

The most established and extensively researched technology for solar-powered household hot water systems is now flat plate solar collectors Figure 1.3. The heat from the sun warms a dark, flat surface, which absorbs as much of the energy as it can before transferring it to any fluid for later use. Flat plate system typically operates and achieves maximum efficiency between 30 and 80°C, however, modern collectors that use vacuum insulation can operate at high temperatures (up to 100 °C). The stagnant fluid temperatures in FPC illustrated below were improved to 200 °C as a result of the introduction of selective coatings.



Figure 1.3: Flat plate collector [7].

1.5.2 Evacuated Collectors

These collectors are used to get higher fluid temperatures compared to flat plate collectors. They are made of double pipes; the working fluid moves through the inner tube which is made of glass because it keeps a vacuum efficiently compared to other materials while the outer tube is manufactured of transparent material to allow the solar ray to be giant by the fluid in the inner pipe as shown in Figure 1.4.

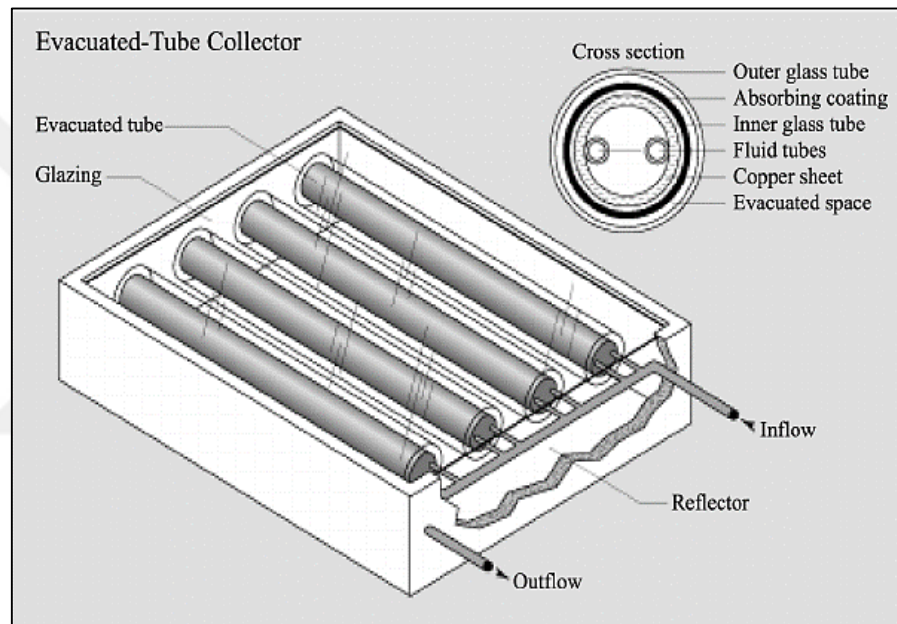


Figure 1.4: Evacuated tube collector [8].

1.5.3 Concentrating Collectors

These devices are rarely used in water desalination systems. They are more used in solar PV to produce electricity. However, when used in water heating it is usually made of parabolic troughs that concentrate the solar energy on an absorber tube that contains water. Figure 1.5 illustrates how the solar rays are concentrated in the receiver tube.

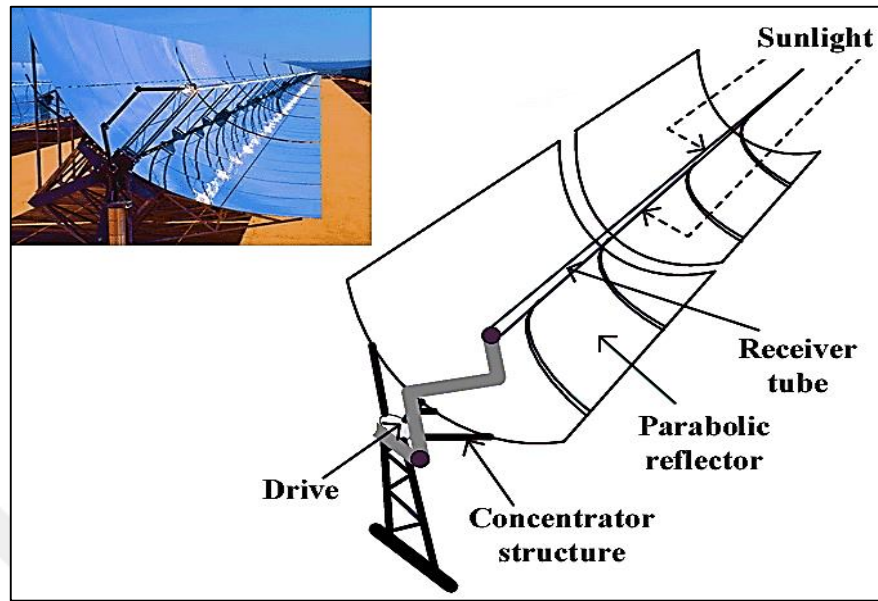


Figure 1.5: Concentrating solar collector [8].

Mirrors are used to concentrate the beam of sun rays on a clear sunny day. This collector consists of a parabolic reflector that directs the radiation onto a receiver tube that absorbs the radiation and turns it into another form of energy.

1.6 SOLAR HEATING WATER

Solar hot water systems are devices that heating water by using solar energy. Selecting the most effective and efficient solar hot water system for a particular environment may be difficult.

1.6.1 Types of Solar Hot Water Systems

The best system configuration should be chosen after taking into account a number of important criteria. These factors involve, among others, the level of construction skill, the amount of solar insolation, the climate, the cost of constructing, implementing, and purchasing materials, the system's location and accessibility, the amount of water that needs to heat, the frequency of using hot water, the availability of electricity, the availability of materials, and the cost of construction, installation, and materials [9].

1.6.1.1 Open loop

Open loop refers to a device in which the fluid that has to be heated circulates through the solar-powered hot water collector itself. There isn't a connecting fluid. For the lone exception to this description, where this solar collector is the sole juncture in a closed loop system where water flows, see “Closed Loop” heading below. Heat exchangers are not used in open-loop systems.

Open loop systems have the advantage of being the easiest to design and build. Due to their small number of moving parts, they often require the least amount of building and could last longer without major maintenance.

This system has some disadvantages. These: In regions where temperatures might fall below freezing, loop systems are not recommended. The pipes may freeze, forcing the liquid water to turn into ice and expand, rupturing the pipes. The system needs to be shut off before it gets cold in order to save pipes. It is suggested to employ a closed loop system to exchange heat to prevent the freezing of the pipes in locations where the temperatures fall below freezing. The system drains that are either manually operated or thermostatically controlled or a pump that circulates traditionally heated water through the system during cold weather are two examples of preventative methods. When the circulation pump is not running, the system may also be set up so that the water drains from the collector by gravity as in Figure 1.6.

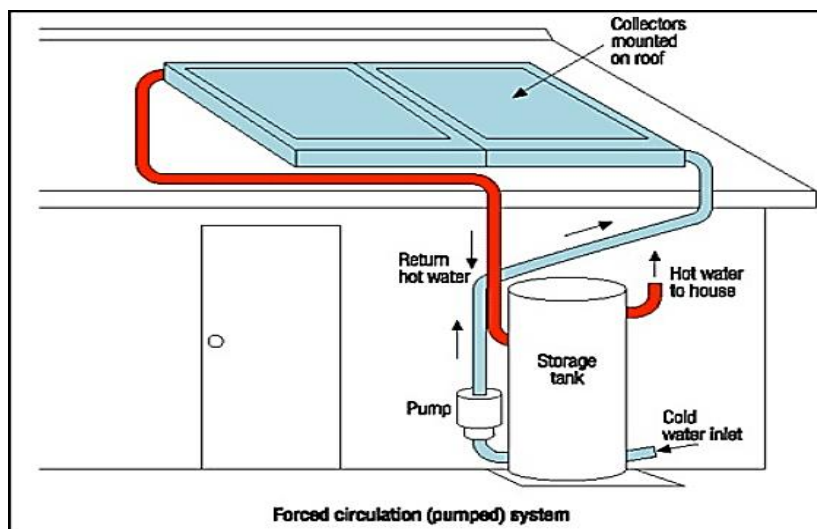


Figure 1.6: Open loop solar water heater [9].

1.6.1.2 Closed loop

Due to its ability to be used in a larger variety of climates, the creation and implementation of closed-loop systems are more complex than open-loop systems.

A liquid that has a much less freezing point than water and is easily heated and cooled is used in closed-loop systems. A notable example is an antifreeze used in automobile radiators. This fluid flows through the solar collector, heats up, and then enters a heat exchanger of some sort. To move heat from the collector to the water, heat exchangers are needed. In addition to using water to heat an area, use in closed-loop systems also is feasible for radiant flooring and wall systems [9].

Closed Loop systems, in contrast to the majority of Open-Loop systems, can be employed in areas with subfreezing temperatures. Water has certain characteristics that might cause pipes to rupture when it freezes. Employing a substance with a freezing point that is substantially lower than water, is avoided.

The need for a heat exchanger is a drawback of closed-loop systems. This causes additional system inefficiencies that need to be taken into account while designing and sizing the system. A system's cost may increase if extra material is required for the heat exchanger.

1.6.1.3 Active solar hot water

Figure 1.7 presents the active system, that depends on outside power to operate. In most systems, this energy is delivered by an electric pump that forces the fluid through the collector. Pumps powered by sunlight energy are frequently found as the external input. Due to the fact that water is only heated and thus pumped while the sun is out, solar pumping is suitable in many cases [10].

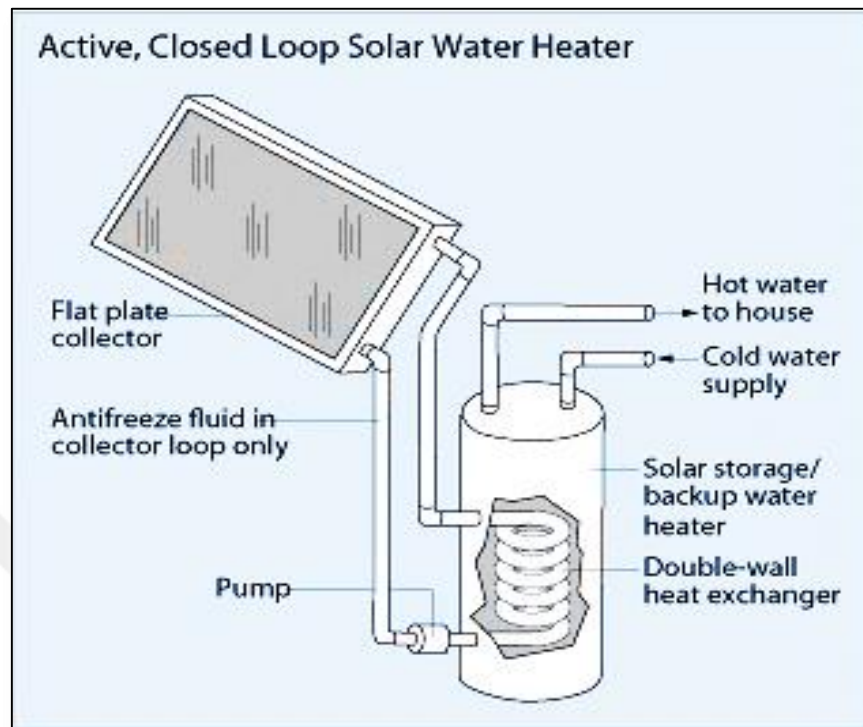


Figure 1.7: Active solar water heater [10].

Pumping water through the collectors eliminates the need to store heated water above the collectors, which can be difficult to design and is constrained by the laws of hydrology and thermodynamics.

The use of a pump requires the procurement of a pump as well as the need for a power source, most often electricity.

1.6.1.4 Passive solar hot water

Passive-type solar systems sometimes referred to as thermosiphon systems, are those that do not rely on an external power source. The phrase "passive solar" refers to much more than just solar hot water. Passive solar techniques include, for instance, painting a swimming pool black to absorb more energy or installing windows on a home's south side (in the northern hemisphere) to let in more of the sun's energy.

The physics of hydrology and thermodynamics must be considered while designing a solar water heater. This technique utilizes a thermosiphon. A thermosiphon utilizes the sun's energy to lift

water or another collecting air vertically. When a liquid has higher viscosity due to varying temperatures, this happens. The hot water tank for storage heat or heat exchanger must be located at the high-level plan, since the hottest fluid always travels higher pushing the colder water lower [10]. While most solar cells are placed on house roofs, this may be significant. There are now multiple design alternatives that make it possible to store hot water above the system.

There is no need for external energy to move liquid through the system. Less moving parts means fewer broken items.

1.6.1.5 The heat exchanging of systems

In regions in which the temperature falls below cold, heat exchangers are used. The heat collected in the collector is the home's user and supplied by a heat exchange unit, which is linked to a closed loop system. By transferring the heat from the sun collection fluid into the hot water tank, a typical design enhances a conventional hot water heater.

Copper tubing is an ideal material for efficiently moving liquid thermal energy. Another better design of a heat exchanger is a barrel with two parallel copper wires. One tube leaves the solar collector, travels in a closed loop back to the solar collector, exits the barrel at the bottom, spirals around the barrel's edge, and enters the barrel at the top. The Domestic Hot Water (DHW) supply is the second tube that runs parallel to the first.

Hot water exits the barrel at the top from where it enters the barrel at the bottom, parallel to the collector tubing.

The closed- loop system is capable of using a wide range of fluids. The liquids that are used more frequently and are more easily accessible are silicones, hydrocarbon oils, antifreeze for automobiles, and glycol/water mixes. These all have pros and disadvantages [10].

Heat exchangers are used primarily to avoid the system from freezing and as a way to enhance another water heating source.

A hot water system is made more complex by heat exchangers. When using a heat exchanger, there are more system components that need to be planned for and designed. Due to the extra

components, there is also a higher likelihood of items breaking or malfunctioning. Additionally, effort must be taken to guarantee that, should the closed loop system fail, the collector fluid won't contaminate the potable water supply or that the likelihood of this happening is minimal.

1.6.2 Advantages of Solar Thermal Energy

Unlike other renewable energy sources, solar heating energy does not require any fuel. The significant advantage over other fossil fuels, whose prices are rising dramatically each year. Most of the world's electricity prices are rising far more quickly than overall inflation. These days, using fossil fuel energy carries a significant risk of price shocks caused by excessive fuel expenses.

Solar thermal power plants can generate power 24 hours a day since they save energy in the molten state. For example, many sources of renewable power, like photovoltaic (PV) solar and wind energy are periodic in space.

One of the greatest benefits of solar heating energy is that does not generate pollution. New technologies such as, solar thermal energy uses equipment that doesn't require significant equipment and material changes, such as thermal mirrors and turbines, which are produced in big amounts at a low cost by the greening industrial sources [7].

1.6.3 Disadvantages of Solar Thermal Energy

Over the last three to four years, solar thermal costs have remained stable at least Euro 3.5/watt. These costs, nevertheless, are excessive because solar PV already costs Euro 2.5/watt and, even on a conservative basis, see a 5% cost savings over the next ten years, bringing its cost down to half that of solar thermal technology by the year 2020.

With daily news of improvements in PV technology materials and manufacturing processes, solar energy has evolved into a hotbed of innovation.

Solar heating systems are used a lot of water, which is a big issue in arid reason. If non-water colling has been used, the cost of CSP projects increases excessively. For the reason use

seawater, however, it is unclear if this idea can be put into effect as it would need construction plants quite close to the coast [7].

It has been observed that the use of large mirror installations has a negative influence on desert animals, jeopardizing endangered species.

Solar thermal energy can only be produced in locations with significant solar radiation. They usually need one large land space and can be built in deserts. This means that it is impossible to construct them in populous places. Table 1.1 shows the operating temperature for several solar heating technologies for the collectors and their efficiency.

Table 1.1: Types of solar heating water with efficiency [7]

Solar Thermal collectors		
Types of solar heating collectors	Operating temperature(°C)	Efficiency (%)
Flat plate collector	35-65	45-60
Evacuated tube collector	50-100	30-50
Parabolic trough collector	350-450	14-16
Parabolic collector dish	500-1200	20-30
Solar power tower	500-1000	12-18

1.7 STORAGE OF THERMAL ENERGY

The temporary storing of thermal energy in the form of hot or cold substances for use at a later time is known as thermal energy storage (TES) [8]. The use of TES devices that work together and store energy by cooling, heating, melting, solidifying, or vaporizing a substance can satisfy these requirements. TES is a key technology in systems including sustainable energy sources and other energy resources because that can enhance the profitability of their operation. In other words, TES is effective in balancing the supply and demand of energy [5,6]. TES systems may increase the efficiency with which thermal energy equipment is used and make large-scale fuel commuting easier [2]. Numerous parameters, such as storage time, cost, supply and usage

temperature, amount of storage and heat losses, affect the choice of a TES system for a given application [6].

Sensible and latent TES are the two main categories. Sensible TES systems store energy by single phase temperature of the storage space, which might be water, brine, rock, soil, etc.

Latent TES equipment involves phase shifts to store energy, such as cold storage employing water or ice and heat storage using paraffin wax (PW) melting. Latent TES units are often smaller than sensible storage units. TES can be reduced based on storage. The entire TES process consists of three steps charging, storage, and discharging. Some activities (like charging and storing) may occur simultaneously in real-world systems, and each step may occur more than once during the course of a single storage cycle. Figure 1.8 displays a simple storage cycle with three distinct phases.

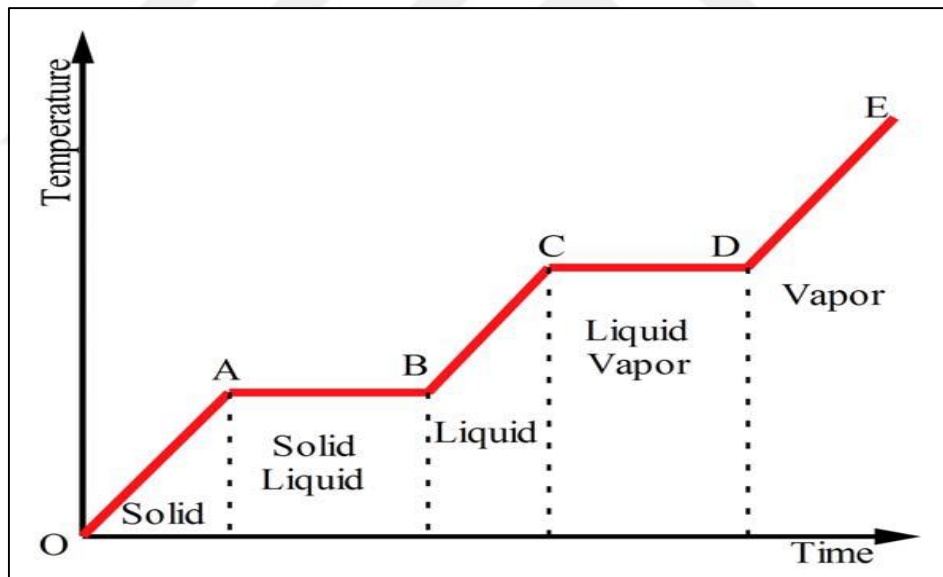


Figure 1.8: Temperature distribution diagram for phase change material [3]

The well-known outcome is a temperature rise (sensible TES) or a phase change (latent TES). When heat is added to a substance that is initially in a solid state at reference O, the solid is first sensibly heated (area O-A), then a solid- to- liquid phase change (line A-B), and finally a liquid- to-gas phase change (region O-B). A liquid- to-vapor phase shift, sensible heating of the liquid (regions B and C), and sensible heating of the vapor (regions C and D) are all possible (regions

D–E). The following formula [4] can be used to express the total amount of heat: One of the most effective methods of storing thermal energy is Latent Heat Storage (LHS) [5].

Energy is stored at the phase change in latent TES systems (e.g. melting, evaporating crystallization). For a given system size, the latent heat change typically exceeds the sensible heat change because of the typical medium's high specific heat and the large enthalpy change during phase shift [5]. The LHS method offers a significantly higher storage density than the Sensible Heat Storage (SHS) approach and a lower temperature differential between giant and releasing energy [5].

Latent TES devices enable the storage of thermal energy in PCMs when become to change phase, often from liquid to solid (energy is used to change water to ice, steam to water and to heating. The transformation of water into ice is the most typical instance of latent TES. Because a significant quantity of energy is stored through the phase shift, ice storage cooling systems have a notable size benefit over chilled-water units of equal capacity [3]. For PCMs (i.e., ice), the transition from liquid -to -solid (freezing) absorbs energy, which is then released during the transition from solid to liquid. In contrast, with the change in positive (high) temperature from solid- to- liquid at constant temperatures, PCMs both absorb and release energy through the liquid- to -solid transition. Latent heat [6] is the total amount of energy that is both absorbed and emitted in each situation. The phase transition of PCM from solid to liquid is essentially shown in Figure 1.9.

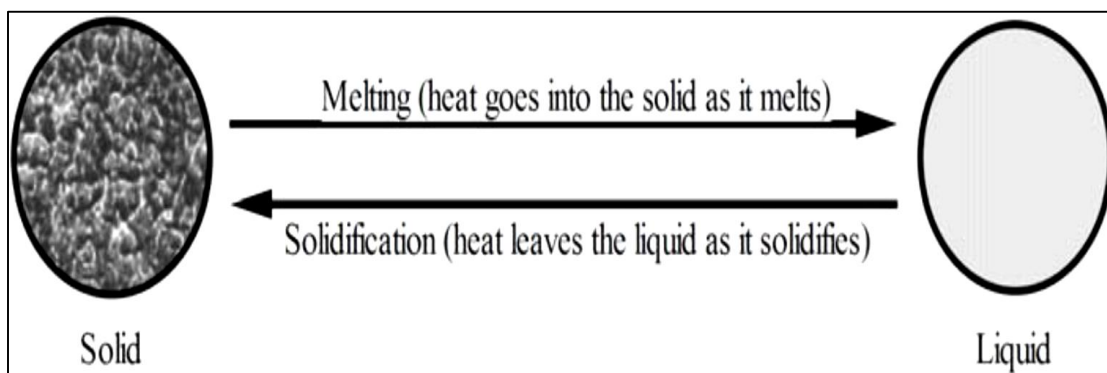


Figure 1.9: Phase change material mechanism [6]

Due to the substantial heat transfer during melting and the lack of significant temperature change during crystallization, in natural settings, PCM is a fascinating source of heat storage material. The heat was absorbed by the PCM microcapsules and stored in the liquid phase transition materials when the temperature increased. The release of this thermal energy by the PCM microcapsules when the temperature drops cause PCM freezing [5].

The energy of vaporization and fusion at boiling and melting degrees, respectively, are the terms used to describe the energy needed to bring about these processes. In the design phase, it's crucial to consider the fusion or vaporization-specific heat as well as the temperature at which the phase shift takes place. PCMs can be enclosed as self-contained elements, contained in specialized containers such as pipes, shallow panels, or rubber bags, or hidden inside normal building materials like wallboard and ceilings. Figure 1.10 present the range of PCM melting temperature and applications. Solar heating paraffine is located within the range of 40-80 °C.

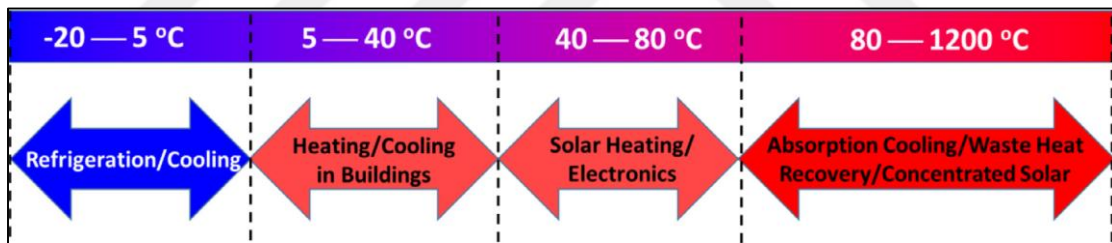


Figure 1.10: Temperature range of PCMs and applications.

1.7.1 Development of the Solar Heating System

Researchers have extensively developed the methods applications and users of solar energy for water heating. Utilizing extremely effective absorbent materials was the first step in enhancing the solar collectors' effectiveness, followed by creating fluid transport corridors, utilizing reflectors, and utilizing composite collectors [8].

It is well recognized that using sufficient TES systems in industrial and building settings offers significant potential for energy savings. The implementation of TES enables the use of waste energy, peak load shifting strategies, and sensible thermal energy consumption in active and passive systems, thereby resolving the mismatch between the energy supply and demand. The

use of TES in an energy system can improve overall efficiency and dependability, but it can also improve economics, reduce environmental pollution, and reduce CO₂ emissions [8].

1.7.2 Storage Material

Over the past three decades, a large number of investigators with an interest in TES have focused a great deal of research attention on PCMs because they are readily available in a wide number of engineering disciplines and have high thermal energy densities per unit volume/mass. For solar thermal energy applications, the use of PCM thermal energy storage is advised in order to increase energy efficiency and fill the current electricity supply gap. Among the numerous TES systems, the PCM thermal storage system ensures performance and dependability due to its high storage density and practically constant thermal energy.

Thermal storage materials are presented into two groups (sensible heat and LHS unit). The most common medium for saving energy in the shape of perceptible heat is water. Common ceramic materials like cement and concrete are also widely used, along with some natural stones like marble and granite, clay and sandstone, and polymers like Synthetic Polymers (PS), and Polyvinyl chloride (PVC).

A PCM that can work within the system's specified temperature range makes up the LHS unit, together with the other two components stated below. A surface for heat exchange that can transmit heat from the environment to the PCM is encased within the PCM.

Three different types of phase shifts may take place in a material [9]: Solid/liquid phase change, Solid/gas, and Liquid /gas phase change.

Because phases change of solid/gas and liquid/gas are needed a significant amount of volume change, the solid/liquid phase is a favored LHS mode.

Most thermal storage TES units use paraffin as the storage medium since it is one of the most reliable PCMs. It's widely utilized due to its properties, like high latent heat and thermal characteristics. Various phase transition temperatures, low vapor pressure in molten form, negligible supercooling, chemical stability, and thermal value are the thermal properties of paraffin[8],[9]. Figure 1.11 shows the classification of three important types of PCM.

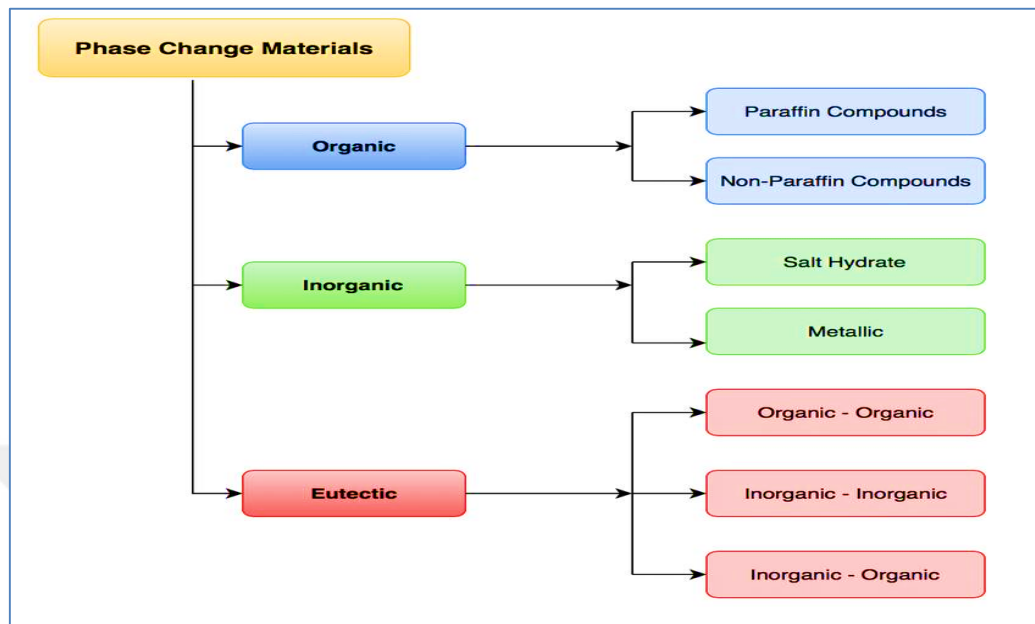


Figure 1.11: Classification of phase change materials [9].

1.7.3. Paraffin

A set of saturated hydrocarbons with remarkably comparable characteristics is known as the C_nH_{2n+2} paraffin.

Between C_5 and C_{15} , paraffin is liquid; the remaining paraffin is waxy solids. The most widely used organic heat storing PCM for industrial purposes is PW. It is made of hydrocarbons with a straight chain and melting points with a range of 23 and 67 degrees Celsius [10].

The advantages and drawbacks of paraffin are as follows [10, [11]:

There is a wide range of temperature ranges in which paraffin is being used, they don't tend to super-cool, they are chemically stable, their fusion temperatures are high, they don't separate, they are safe, they're non-corrosive and have low heat conductivity.

Commercial paraffin had sensitive and specified melting points. Technical grade paraffin's qualities are inferior to those of pure paraffin. It is frequently used as PCMs for TES since it is less expensive than pure paraffin.

1.8 THE AIMS OF THIS WORK

Main objectives of the present study have been summarized in several points as follows.

- a- - To build an experimental station for heating solar water consisting of an FPSC filled with PCM for domestic purposes during the night or cloudy days.
- b- To investigate PCM's influence on the storage energy with some variables.
Numerically, to build a model to simulate PCM performance and external load into the TES tank, solar collector and analyze the influence of several FPSC operating conditions.
- c- To ensure the accuracy of practical results, a comparison should be made between experimental and numerical results.

2. LITERATURE REVIEW

2.1 INTRODUCTION

Solar energy can be stored, and efficient technologies have been developed to preserve sustainable energy, for solar energy storage, it is extremely challenging to maintain a balance between supply and demand.

Sources of solar and wind power are not constant. Both the sun and the wind do not rise and set every day. Additionally, customer demand fluctuates. After 5 PM, when most consumers are going home and utilizing more electricity, residential demand peaks. In order for an electric grid to work, supply and demand must be balanced. Voltage changes are brought on by imbalances. Energy storage is therefore essential to maintain a balance between intermittent supply and changing demand [11].

Numerous experimental and computational investigations looked into ways to boost the effectiveness of FPSCs and storage tanks.

A review of the literature for studies that focused on creating a solar energy system employing PCMs is included in this chapter. During the previous contract, the use of PCMs for cooling and heating solutions was accomplished. PCMs have been widely used in a variety of storage systems, including heat pumps, thermal control systems, and solar engineering, which heavily rely on PCMs. This chapter has been used to execute the history of (TES) with a solid-liquid phase shift. After review, these studies were divided into two groups: History of SWH, developments in solar heating collectors, and applications.

2.2 SOLAR HEATING WATER HISTORY

The most popular use of solar energy is for water heating. The three primary types of solar collector-based solar water heaters on the market are compound parabolic collectors, evacuated tube collectors, and flat plate collectors (FPC) (CPC). Designing, building, and conducting trials on a water heating system, have been assessed the device's performance under various real-world working circumstances [12]. In order to test solar water heaters, Nahar [13] matched

Cu-Al fin with Cu-Cu fin in flat-plate collectors. S.Sadhishkumar T. Balusamy [14] researched and evaluated the behavior of a different solar heating water as TES. The thermal behavior of an integrated solar water heater with a corrugated absorber surface was tested by Rakesh and Rosen [15]. In addition to evaluating the characteristics of evacuated tube-based solar water heaters, Morrison et al. [16] also examined the circulation rate through single-ended tubes. They conducted the simulation underneath the assumption that there was no contact between adjacent tubes in the collector array and created a computer model of the heat transfer and fluid flow within a single-ended evacuated tube. The results were based on single-ended thermosyphon with a constant pressure condition imposed across the open end functioned as the computational domain. The CFD model employed was validated using Particle Image Velocimetry (PIV) data. This chapter is divided into the following sections.

2.3 SOLAR ENERGY COLLECTORS

The literature review on solar heating systems, namely solar water heaters for practical uses, is covered in this part. Water heating is the most common use of solar energy. The three primary types of solar collector-as heating applications system on the market are compound parabolic collectors, evacuated tube collectors, and flat plate collectors.

2.3.1 Flat Plate Solar Collector

Gunerhan and Hepbasli [17] examined the exergy analysis-based system performance analysis for solar water heaters. The experiments were conducted in the province of Izmir (Turkey). Their primary goal was to analyze the system's various parts, namely the FPSC, the heat exchanger, circulating pump, and storage tank. They also wanted to look into how variations in water inlet temperature affected the energy efficiency of the other 29 parts and the system as a whole. On a fuel basis production, the energy efficiency was found to vary between 2.02 and 3.37 % for the collector, between 16 and 51.72 % for the heat exchanger, and between 10 and 16.67 % for the circulating pump, while it was found to be around 3.27 and 4.39% from across board.

Ceylan [18] studied the temperature distribution of solar water heater using an experimental system. At 40, 45, 50, and 55 degrees Celsius, experiments were conducted. The developed system was also contrasted with the thermosiphon system. By calculating the amount of energy

that has been stored, the temperatures of the fluid in the storage with different volume, and the system efficiencies were analyzed. At 40°C, the largest amount of water was discovered to be 108 kg.

Alvarez et al., [19] experimentally analyzed and numerically verified the Thermal and hydrodynamic performance of FPSC with a modern design as shown in Figure 2.1. This model is distinguished by its increased surface area that is directly coupled to HTF and its corrugated serpentine duct. As the HTF, a mixture of 30 percent water and 30 percent ethylene glycol was used with a fluid flow rate range of 16 to 29 l/min. To be exposed to the most solar energy, the collector angle was 35° southerly. The proposed thermal model has been mathematically solved by a FEM utilizing the COMSOL software. By contrasting the findings with the analytical findings of the fin and tube solar collector, the results were validated. The water tank's heating curve demonstrated the results, demonstrating that serpentine geometry outperformed parallel geometry. the HTF temperature enhanced in the collector from 15 to 33 °C.

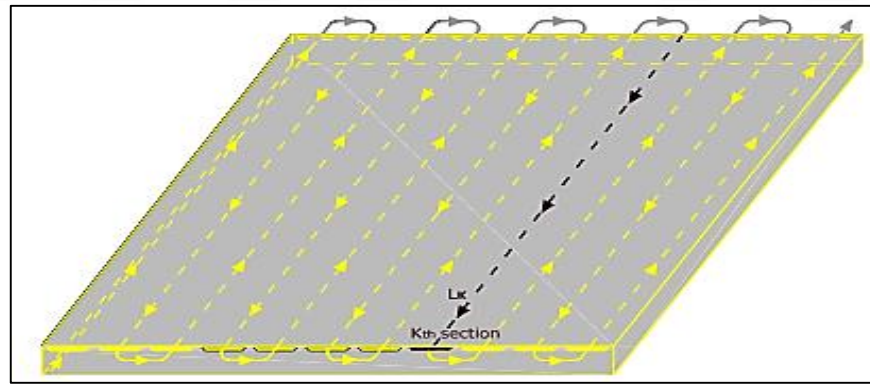


Figure 2.1: Solar collector sketch with serpentine pipe-flow path

Manjunath et al. 2012 [20] presented a numerical model to calculate the impact of the absorber plate surface geometry on the thermal behavior of solar collectors. The 3D-computational program consisted of a flat absorber plate integrated into a circular absorber tube. Water was available as the HTF at the tube's intake. The collector was angled toward the horizon at a 30° angle. Figures 2.2(a) and 2.2(b) illustrate the Dimple-Plate Solar Collector (DPSC) and FPSC, two examples of plates with and without dimples used. These plates were heated uniformly and

continuously at 1000 and 600 W/m². The CFD analysis led the researchers to the conclusion that the DPSC was more thermally efficient than the FPSC. Due mostly to the expansion of the heat transfer's diffusion area, the DPSC boosts heat transfer to the absorption pipe. Additionally, the bulk temperature of the absorber plate for the DPSC increased by almost 50 °C in comparison to an FPSC. Most noteworthy, when a DPSC is used instead of an FPSC, the mean outlet water temperature presents an enhancement to 5.50 °C.

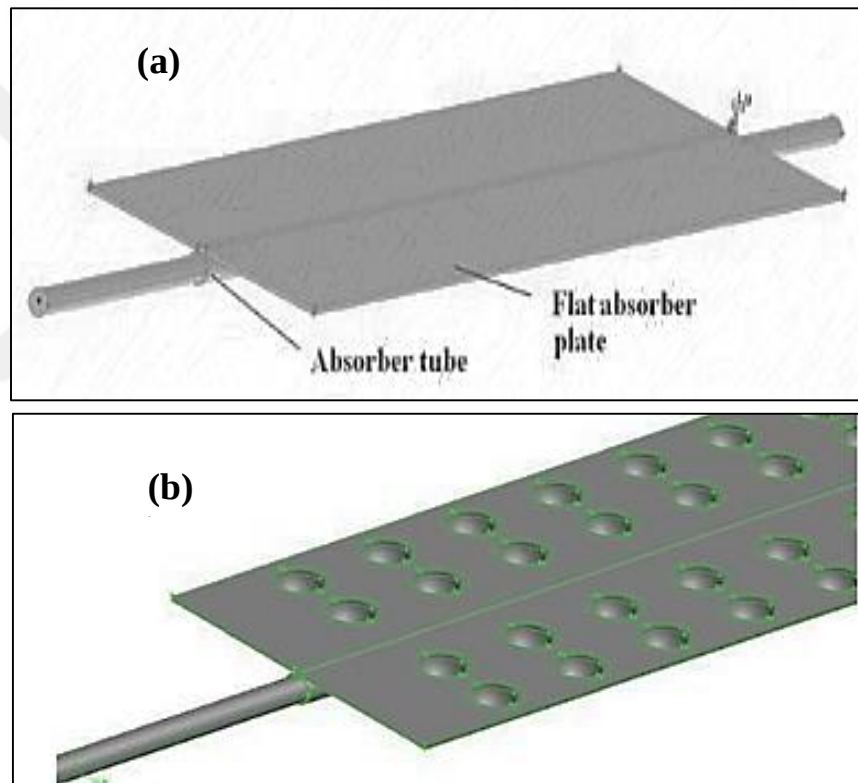


Figure 2.2: 3-D model of a solar collector with (a) flat plate, (b) dimple absorber plate

2.3.2 Parabolic Dish Solar Collector

The dual axis tracking parabolic dish solar collector is an excellent form of focus system to solve the energy sources issue. Due to its high concentration ratio as compared to other solar technologies, it produces a lot of heat [8].

Mayur G. Et al. [21] designed solar parabolic trough, which depends on the following factors: the concentrator's aperture, the inner and outer dimensions of the absorber tube, the glass tube, length of the parabola, ratio of concentration, collector aperture size, Focuser's specular reflectivity, sun radiation transitivity of glass, absorber tube, glass emissivity, the intercept factor, and the emissivity of the absorber tube surface. Performance analysis is based on experimental data collection and calculations of the overall loss coefficient, heat correlations, and thermal performance estimations. The internal heat transfer coefficient between the cover and the absorber tube was analyzed. According to the research, solar concentration collectors enable us to utilize solar thermal energy for a variety of uses without the need for an external power source or separate devices for each application.

Hamdi Kessentini and Chiheb Bouden have reported on the building of double-glazed integrated collector storage (ICS) with compound parabolic concentrators (CPC) reflectors, as well as its design, production, and numerical modeling [22]. The design of the double glazing is optimized using numerical models that are tested in the Tunisian environment. The various combined-glazing models that have been proposed are compared and contrasted with a single glass ICS model illustrated as a baseline. According to the simulation's findings, using combined glazing with an argon layer and a lower glass that is coated with low-E significantly improves the ICS heater's thermal behavior and significantly reduces its thermal losses at night. The construction of a double-glazing ICS prototype collector takes into account these numerical results. The size, geometry, optical properties, and other details of the shape were discussed.

Hamdi Kessentini and Chiheb Bouden [23] developed a numerical study to evaluate the thermal performance of a dual tank ICS solar water heater with asymmetric CPC reflectors to test an experimental setup validation. The water temperature in the storage tanks increased dramatically during the day and was maintained at a suitable level at night, according to the results of the experimental testing. The bottom tank showed larger capture because of its better exposure to the sun during the day, however, the top tank showed a lower water temperature reduction at night because of its superior thermal insulation giving excellent preservation of hot water temperature to the ICS system. The numerical model was created using a heat transfer study of the various solar system components. The model is able to predict either an ICS system with

one or two glass covers in transient circumstances. When calculating the energy balances at a specific time, the absorbed solar energy, as well as the thermal heat losses from the ICS water heater through convection, conduction, and radiation, were taken into account. According to a numerical analysis, the effects of specific design elements, the use of low emissivity absorbers, upgrading the quality of the reflectors, and employing double glass have a significant impact on the ICS system's thermal performance. The ICS system's overall heat losses are reduced by 47 % whenever low emissive absorbers are used, and its daily efficiency rises from 18 to 25 percent as a result. The yearly diurnal usable heat gain increases by 3% when the concentrators' reflectivity is increased from 0.75 to 0.95. Last but not least, using double glazing can reduce the overall loss coefficient by 47%. The design and construction of an upgraded CPC-type ICS system with respect to its thermal performance served as the major goals of this study. The thermal behavior of an ICS system with CPC reflectors and one or more glass coverings was predicted using a numerical code. The models made use of normal Tunisian meteorological data.

Saied Farahat [24] developed a simulation tool for thermal and energetics computations in this study, which focused on flat-plate solar collectors' energetic optimization. To assess the exergetic efficiency, exergy flows, and losses for a typical system, as well as its thermal and optical performance FPSC under certain operating conditions, thorough energy, and exergy analysis was conducted. The exergy efficiency is significantly impacted by the optical efficiency. The optical efficiency is raised by combining FPSCs with optical concentrators. With operational settings, it was discovered that energy efficiency rises without reaching extremes. However, there are local maximum points and a global maximum point for the exergy efficiency. It was discovered that the exergy efficiency was rapidly declining when the ambient temperature and air speed increased.

2.3.3 Evacuated Tubes Solar Collector

Both the fluid flow rate of the naturally moving and evacuated glass tubes into the diffuse reflector were coupled by Budihardjo et al. [25]. This study was recorded by placing collectors on top of a diffuse reflector, which causes the heat input to be focused near the top of the tube. The correlation may be utilized to forecast the flow rate at any time of day had no impact on the

circulation flow rate through the tubes. When a concentrating reflector was placed beneath the collection, different flow configurations in the tube could be viewed.

Zambolin and Del Col [26] examined the heating efficiency of two distinct solar collector types, an evacuated tube collector and a glazed flat plate collector. The two collectors were put next to one another and tested under the same operational circumstances. The evacuated collector was a straight flow-through design with an exterior compound parabolic concentrator (CPC) reflector. The main aim of this study effort was to compare the performance of evacuated tube and FPSCs tested under steady-state and quasi-dynamic test. The evaluation and comparison of the daily energy performance of the aforementioned collectors was the secondary goal of the research. Efficiency and average temperature decrease have been plotted. The results were shown that the daily efficiency could be estimated using the parameters of the quasi-dynamic model. The optical efficiency of the flat plate collector decreases in the morning and afternoon due to additional reflection losses. The design of the vacuum tube collector minimizes efficiency loss, but the majority of the absorber area is more frequently exposed to radiation with a quasi-normal incidence. The advantage of an evacuated collector over a flat plate collector has finally been found.

Ayompe et al. [27] performed a flat plate collector (FPC) and heat pipe evacuated tube collector (ETC) performance review of many types of solar water heaters in Dublin's moderate environment (Ireland. For yearly solar radiation of 1087 kWh/m^2 , the 4 m^2 FPC and 3 m^2 ETC systems. The average annual collector efficiencies for the FPC and ETC were found to be 46.1 % and 60.7 %, respectively, while the system efficiencies were found to be 37.9 % and 50.3%. The economic analysis also revealed that neither system is commercially viable; for the FPC and ETC, the basic payback periods ranged from 13 to 48.5 years, respectively.

Hayek et al. [28] investigated how well two different evacuated tube solar collector design ideas tubes and heat-pipe designs—performed in the context of the regional weather patterns seen along the Mediterranean Sea's eastern shore. The investigations were conducted between 30 November and 1 January, or during the months of winter-like weather on clear days. The experiment included the construction of different evacuated tubes, a tank, a circulation system,

and measurement tools. The experimental analysis demonstrated that the efficiency of the ETC with heat pipe design was found to be nearly 15 to 20 % greater than that of the water-in-glass designs. However, because of their greater beginning costs in the local market, their payback times were found to be substantially longer.

2.4 STORAGE OF THERMAL ENERGY WITH PCM

The practice of temporarily storing thermal energy as hot or cold objects for use later is known as TES. The amount of energy needed varies daily, monthly, and seasonally. These requirements can be satisfied by TES systems that work together and store energy by cooling, heating, melting, solidifying, or vaporizing a substance. The thermal energy becomes accessible when the process is reversed. By bridging the period between energy collection and consumption, TES can enhance the efficiency of systems employing other energy resources as well as renewable energy sources. In other words, TES aids in maintaining a balance between energy supply and demand. Several options for effective energy use and conservation are provided by TES systems. PCMs for TES are compounds that offer thermal control by absorbing and releasing heat from the medium at particular phase change temperatures. TES in general and PCMs, in particular, have been the subject of research for the past 30 years; nevertheless, despite the great amount of material available, it is also widely spread throughout the literature. In the phase change range, PCMs collect energy during the heating process while phase change occurs and release it to the environment during the reverse cooling process. PCMs have the capacity to change their state through latent thermal energy at specific temperatures. Since PCMs for TES are typically solid-liquid PCMs, they must be enclosed. When thermal energy system using PCMs as a storage medium offer benefits including large TES heat capacity, small unit size, and isothermal behavior during charging and discharging.

Prakesh et al. [29] analyzed a storage water heater that was built-in and had a layer of PCM at the bottom. The water warms up in the sunlight, which then radiates heat to the PCM beneath it. Due to the buildup of energy in the form of latent heat, the PCM melts. When there is no sunlight, cold water is utilized instead of hot water, and the cold water receives its energy from the PCM. When the PCM changes from a liquid to a solid state, energy is released. This type of system might not function effectively because PCM and water don't conduct heat well.

To keep hot water warm throughout the night, Chaurasia et al. [30] studied solar energy storage systems based on the latent and sensible heat approach. Two similar storage units were employed for this. One storage unit had 17.5 kg of PW, which has a melting point of roughly 54 °C, packed in a heat exchanger composed of aluminum tubes, while the second unit only contained water in a galvanized drum. The effective area of the collector was taken to be 1 m², and its overall volume was split into five sectors. Experiments in the open air were done to show how a small solar collector could be used to heat water. While the PCM was being charged and discharged, the temperatures over time were measured. While the battery was charging, sun's intensity was noted. The results were performed for rates of water flow ranging from 8.3 to 21.7 kg/h. It was investigated how the water flow rate affected the beneficial heat gain. For the charging process, the heat transfer coefficients were computed. During the charging and discharging procedures, the spread of the melting and freezing fronts was also investigated. The experimental findings demonstrated that as natural convection becomes more powerful during the charging process, the average heat transfer coefficient increases significantly with increasing the thickness of the molten layer. According to research, the beneficial heat gain during discharge increases as the water flow rate does [30]. In Figure 2.3, the cross-sectional schematic view of the experimental apparatus has been given.

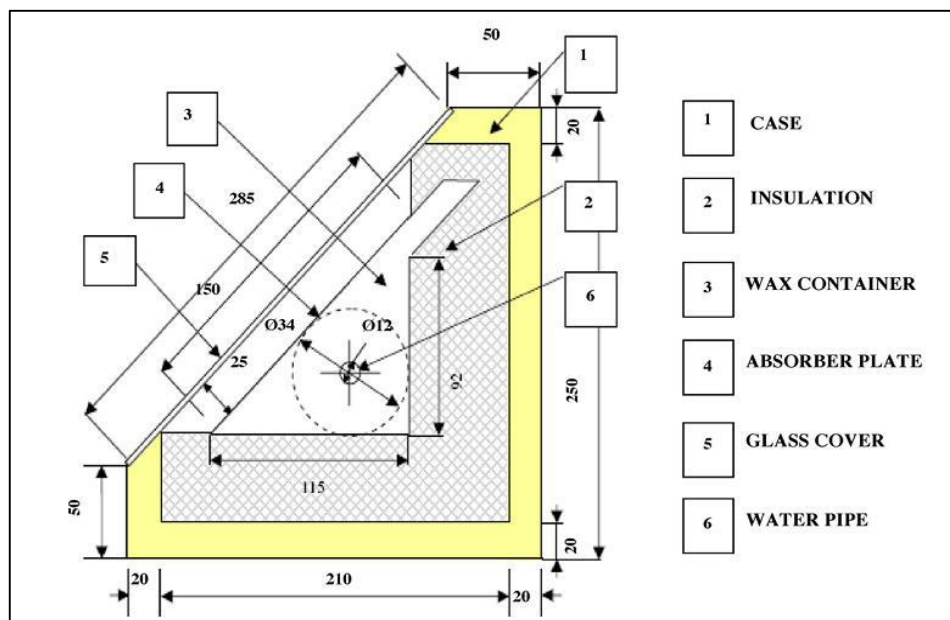


Figure 2.3: The schematic view of the experimental apparatus [30]

Abdul Jabbar N. Khalifa et al. [31] used PCMs as LHS media in solar thermal energy systems, which have a higher storage capacity than SHS. This work combines a back PW container as a PCM thermal storage media with a storage solar collector made of six 80 mm diameter copper pipes connected in series. To demonstrate the storage solar system's capacity to heat water, experiments are carried out outside. Calculations are carried out to analyze the collector's various performance factors in order to determine how well the system works. The temperature of the plate is shown to increase up to 2.5 m from the entry; after that, a very steady temperature is seen. This tendency was in opposition to the earlier growing trend seen for a similar system lacking the PCM back layer. During the hours when it wasn't sunny, the liquid PCM transferred heat to the absorber, some of which was conducted as useful heat in the flowing water. After sunset, the water kept warming because the PCM acted as a heat source. They founded a similar pattern in the fluctuation of the mean plate and wax temperatures over time for the clear days in January and February. However, they found that the wax temperature was higher than the absorber temperature on the partly cloudy day in March, suggesting the PCM's storing capability. The system's usable efficiency varies quite consistently over all months.

Nkwetta et al. [32] experimentally studied the DHW tank performance integrated with various types of PCM in different locations. The PCMs used were PW, industrial-grade granulated PW, sodium acetate trihydrate+10%graphite, and RT58-Rubitherm. This model was numerically simulated using the ANSYS simulation tool, and it was experimentally validated. In comparison to RT58-Rubitherm and industrial-grade granulated PW, the results showed that the PCM composed of 10 kg sodium acetate trihydrate and 10% graphite had the maximum storage energy at around (56812.63 kJ) at a shorter charging time. Additionally, 10 kg of sodium acetate trihydrate plus 10% graphite increases energy storage by 2.98 % in comparison to water. As a result, there was a fairly excellent agreement between the findings.

Beyne et al. [33] experimentally improved the heat transfer by treating the PCM material with foam to increase the thermal conductivity of the PCM. Studying the foam's pore size, porosity, and material type during the solidification process allowed for this (solid phase). Various samples from a metal foam experimentally impregnated with PCM were obtained, and they were compared to another without foam. It was set in a rectangular hollow, with one side cooled by

the coolant and the other sides open to ambient air. Where the temperature of the cavities outside walls, the velocity of coolant flow, and the temperature of the coolant itself could be measured. In comparison to PCM without foaming, the results showed that adding metallic foam shortens cooling time by at least 25%.

Lu et al. [34] conducted a numerical and experimental study on the efficiency of TES tanks containing two distinct PCM types. Sodium acetate and Lauric acid were used as PCM with various melting points and simulation evaluation by the ANSYS software. The researchers came to the following conclusions: the reheat impact of Lauric acid is negligible, but Sodium acetate has a more pronounced reheat effect on the cold water and a substantial heat storage per unit volume. The phase change heat storage water tank may be able to reduce the time needed to store heat to some amount. The two PCM working together might prolong the exothermic process and enhance the efficiency of the heat storage tank. The heat escape from the water tank was also significantly impacted by the increase in PCMs. The PCM calculated for 11.12 % and, 19.17 % of the tank volume could increase the heat release by 24.82 % and, 34.05 %, respectively, under the conditions with the temperature of heat storage of 70°C and flow rate of 5 L/min.

Sathiyaraj et al.[35] improved experimentally the PCM performance for thermal storage (used PW as PCM) by boosting it with graphite powder to increase its thermal conductivity. The improvement was quantified as a time delay during the phase change process, which is shown by a temperature change in the thermocouples. Due to its high thermal conductivity (300 to 550 W/m.K), the paraffin/graphite powder mix outperformed pure PW in terms of heat transmission, according to the data. With pure paraffin, the temperatures of all the thermocouples reached 62 °C in 7228 s. During this time, all of the pure paraffin has melted into liquid. While all thermocouples reached 67 °C in 5879 seconds when paraffin/graphite powder was utilized, this time duration showed that the whole volume of composite PCM was in the liquid phase.

Later, Tayeb [36] examined research on the effectiveness of organic-inorganic compound combinations as energy storage media. In order to find the ideal composition that stores the most energy while releasing it at a steady temperature or within a specific temperature range, two

types of PCMs in various ratios were studied. As a PCM, various ratios of stearic acid and Glauber's salt $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$ were utilized. The effect of inlet air temperature, the flow rate of the cooling fluid, and the addition of a nucleating agent were all evaluated. Based on the experimental findings, a mixture containing 40% stearic acid and 60% Glauber salt might be used to store the most energy.

Qin et al., 2020 [37] numerically investigated the impacts of the different types of PCMs on the heating performance of the hot water tank at various positions, thermal conductivity, and flow rates. As PCM, spherical capsules containing low melting point metals (LMPM), sodium acetate trihydrate (SAT), and PW were employed. Using the ANSYS-FLUENT program, the researchers carried out the numerical simulation. The system with LMPM performs better, per the data, due to its heat conductivity. At the same position and flow rate of the PCM balls, the LMPM filling efficiency and Richardson number were also the highest. With an inlet flow rate of 9 lit/min, the melted portions of the LMPM, SAT, and PW water tanks at the end of the experiment were 0%, 28%, and 47%, respectively, with released heat energies of 17.22 MJ, 16.77 MJ, and 16.37 MJ. Therefore, using LMPM as a PCM in the hot water tank can increase thermal stratification and improve heat storage capacity.

2.5 SUMMARY

It's evident from previous experimental and numerical reviews that the problem of storing solar thermal energy has been extensively studied in order to enhance the system's thermal performance to use in domestic and other applications for heating. Various PCM types were used as a medium for storing thermal energy. All studies were carried out in different climates and environments. Few studies have been conducted to store thermal energy utilizing cylindrical capsules packed with PCM. The present thesis includes an experimental and numerical solution method by ANSYS-FLUENT simulation for a TES tank with PW as a PCM, integrated with external load. The present study was conducted under the condition of Basra city climate. The previously reviewed studies focused on the following points:

- a. The effect of different solar collector types on solar concentric parabolic plate performance.

- b. The effect of the collector constructions on the performance.
- c. Studying the performance of the combined system of solar concentric parabolic plate with different applications.

From the previous reviews there are some points that need more research and explanation which are:

- a- A comprehensive study explains the relation between solar concentric parabolic plate dimensions and ambient conditions with performance and flow field.
- b- According to our review and searches in this field, there are few package programs able to perform the PCM for both solar collectors and storage tanks, otherwise no package programs with external load as heat exchanger.
- c- Also, for Basra city there are few theoretical studies on solar concentric parabolic plates and there is no detailed experimental study including the measurements of the effect of the external load on the storage tank.

3. NUMERICAL ANALYSIS

3.1 INTRODUCTION

The first part of this chapter uses a flat plate collector to combine with a U tube in 10 passes while being filled with a PCM layer. The second component consists of a heat exchanger with a helical 10 coil tube and a storage water tank (shell) contained PCM capsules. ANSYS FLUENT 2020R1 software is used to model each component using finite volumes method (FVM). The numerical solutions that are described in the next sections involve a number of variables and parameters.

3.2 SOLAR RADIATION ANALAYSIS

The sun radiation intensity collected by the solar collectors to heat PCM and water through U tubes is calculated in the current numerical study using a mathematical model. There are several angles that need to be defined and calculated as a way to determine the sun radiation intensity on slanted bodies [38].

Figure 3.1 illustrates the angle at which the earth must rotate to place any location's meridian directly under the sun. The hour angle has a negative value of -180° at any particular place at sunrise. From there, it begins to rise by 15° each hour, increasing until it reaches zero by midday. Hour angle increases from zero to its highest value of $+180^\circ$ over the period from afternoon till sunset. The hour angle value is calculated as in degrees [38]:

$$\omega = 15 \times (ST - 12) \quad (3.1)$$

Where ω is the solar hour angle and ST is the local time of solar.

The angle of latitude, shown in in Figure 3.2, is a geographic coordinate that indicates a location's north-south orientation on the surface of the earth. At the equator, latitude angles range from 0° to 90° (North or South). 30 degrees for Basra, Iraq.

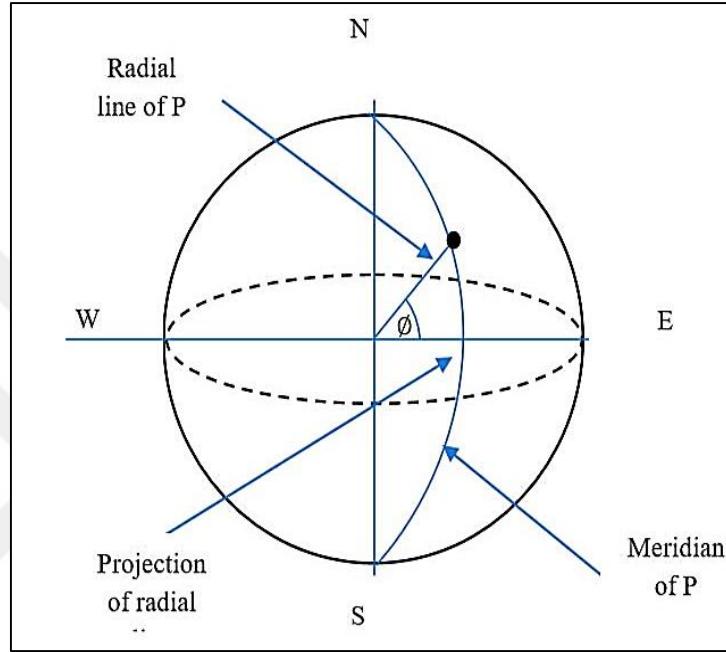


Figure 3.2: Latitude angle of a given point (P) [38].

The angle of solar declination also depicts the line's projection over the earth's equatorial plane. is an angle between a line that connects the centers of the sun and the earth. The earth's rotation around its axis causes changes in the solar declination throughout the course of the year. It changes in the range of +23.45° on June 21st and -23.45° on December 21st. It can be calculated by:

$$\delta = 23.45 \sin \left[\frac{360}{365} (284 + n) \right] \quad (3.5)$$

As shown in Figure 3.3, the zenith angle (z) is the angle formed between the direction of the sun's rays (SP) and the vertical line at a certain place on the surface of the earth (NP). The zenith angle is comparable to the solar altitude angle, but because it is calculated from the vertical

rather than the horizontal, it is equal to 90° minus solar altitude. The zenith angle is close to 90° during dawn and sunset, but it lowers to almost 0° throughout the middle of the day.

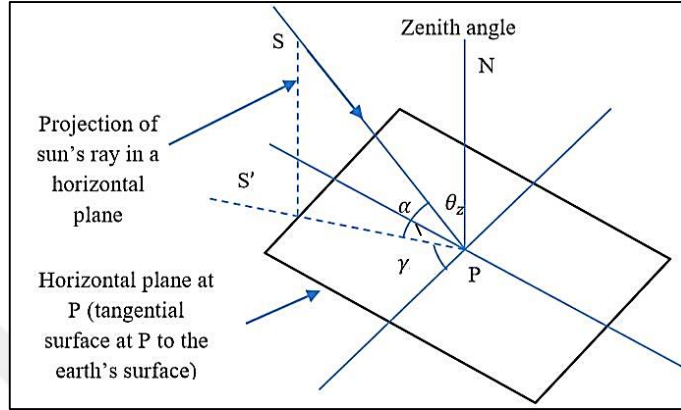


Figure 3.3: Zenith angle, solar altitude angle [38].

As shown in Figure 3.3 solar altitude α' , is the angle in the range of horizontal line (S'P) and the line to the sun rays (SP) ($0^\circ \leq \alpha' \leq 90^\circ$). Zenith angle and solar altitude taken combined equal 90 degrees [38].

$$\theta_z + \alpha' = 90^\circ \quad (3.6)$$

The sun's position affects the solar altitude angle. This angle is close to 0° at sunrise and sunset and gets close to 90° at midday. It can be calculated as follows:

$$\sin(\alpha') = \sin(\phi) \sin(\delta) + \cos(\phi) \cos(\delta) \cos(\omega) \quad (3.7)$$

According to Figure 3.3, the angle of solar azimuth is the range of the sun's ray direction with the horizontal plane (S'P) and the south direction. The angle can be determined by [38]:

$$\sin(\gamma) = \frac{\cos(\delta) \sin(\omega)}{\cos(\alpha')} \quad (3.8)$$

As shown in Figure 3.4, the angle of solar incidence, or is the angle formed by the sun's rays striking a surface and a vertical line on that surface (3.4). This angle varies throughout the day and throughout the year, and it has a significant impact on the solar energy that the receiving

surface receives. As the cosine of the incidence angle increases, the intensity of solar radiation falls. For any surface inclined at an angle of, equation (3.9) yields the angle of incidence:

$$\begin{aligned} \cos \theta = & \sin \delta \sin \varphi \cos \beta - \sin \delta \cos \varphi \sin \beta \cos \gamma \\ & + \cos \delta \sin \varphi \sin \beta \cos \gamma \cos \omega + \cos \delta \cos \varphi \cos \beta \cos \omega \\ & + \cos \delta \sin \beta \sin \gamma \sin \omega \end{aligned} \quad (3.9)$$

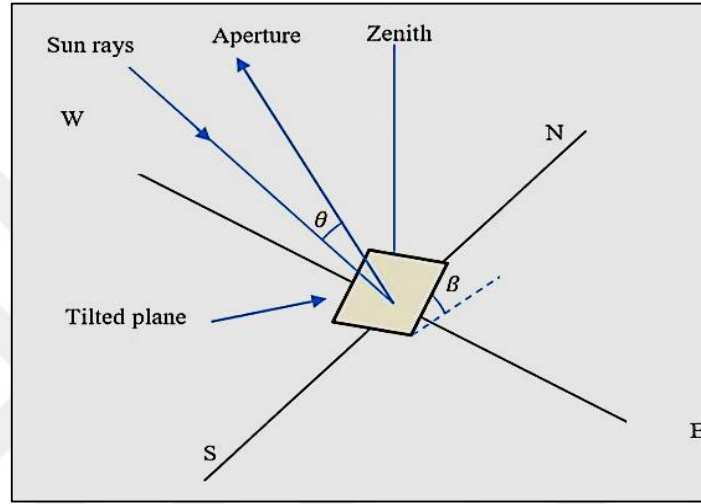


Figure 3.4: The incidence angle on a slanted surface [38].

Direct beam radiation (abbreviated I_b) refers to solar energy that reaches the earth's surface directly without being scattered, whereas diffused radiation refers to solar energy that travels through the atmosphere and is dispersed. Irradiance is the total of diffuse radiation and beam radiation [38].

$$I_s = I_b + I_d \quad (3.10)$$

Beam radiation I_b for tilted surfaces is given by:

$$I_b = I_{bn} \cos \theta \quad (3.11)$$

Where I_{bn} is radiation beam in the direction of the sun rays and θ is angle of incidence.

Thus, the irradiance becomes:

$$I_s = I_{bn} \cos \theta + I_d \quad (3.12)$$

And

$$I_{bn} = A \exp \left[-\frac{B}{\cos \theta} \right] \quad (3.13)$$

$$I_d = C I_{bn} \quad (3.14)$$

Where A, B, and C are constants that vary during the year because of seasonal variation of the atmospheric water-vapor and dust content. These constants are presented by [38]. The revised values are shown in Table 3.1.

Table 3.1: Values of A, B, and C [38]

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
A(W/m ²)	120	1187	1164	1130	1106	1092	1093	1107	1136	1136	1190	1204
B (W/m ²)	0.14	0.14	0.14	0.16	0.17	0.18	0.18	0.18	0.165	0.152	0.144	0.141
C (W/m ²)	0.10	0.10	0.10	0.12	0.13	0.13	0.13	0.13	0.121	0.111	0.106	0.103

3.3 WATER HEATING SOLAR COLLECTOR ANALYSIS

The allocation of actual incident solar radiation into usable energy gain, thermal losses, and optical losses in a solar collector is described using an energy balance. After ten tube passes, the water inside the U tube was heated by the final solar intensity that was exposed to the PCM surface.

The measurements for the flat plate collector consist of a set of information that is used as input in the mathematical model to determine the collector's thermal efficiency. The collection of information also includes PCM parameters, water flow rate, inlet temperature, total collector area, and solar radiation intensity on the collector. The FPSC is seen schematically in Figures 3.5 and 3.6. The 2D governing equations for the unsteady state, and incompressible flows with

Boussinesq approximation are numerically solved using the FVM with the ANSYS FLUENT 2020R1 software suite for both PCM and water flow.

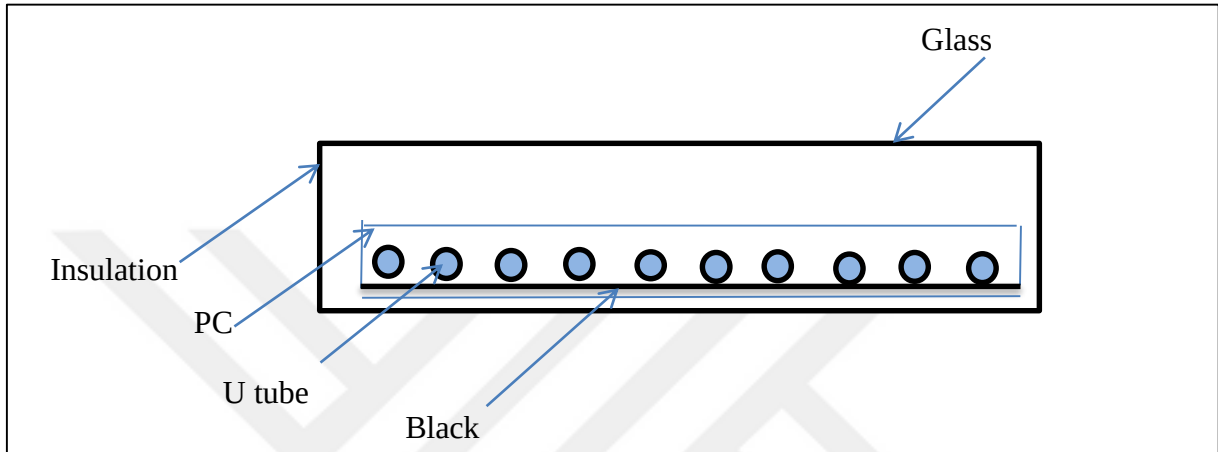


Figure 3.5: Schematic view of the flat heating solar plate collector

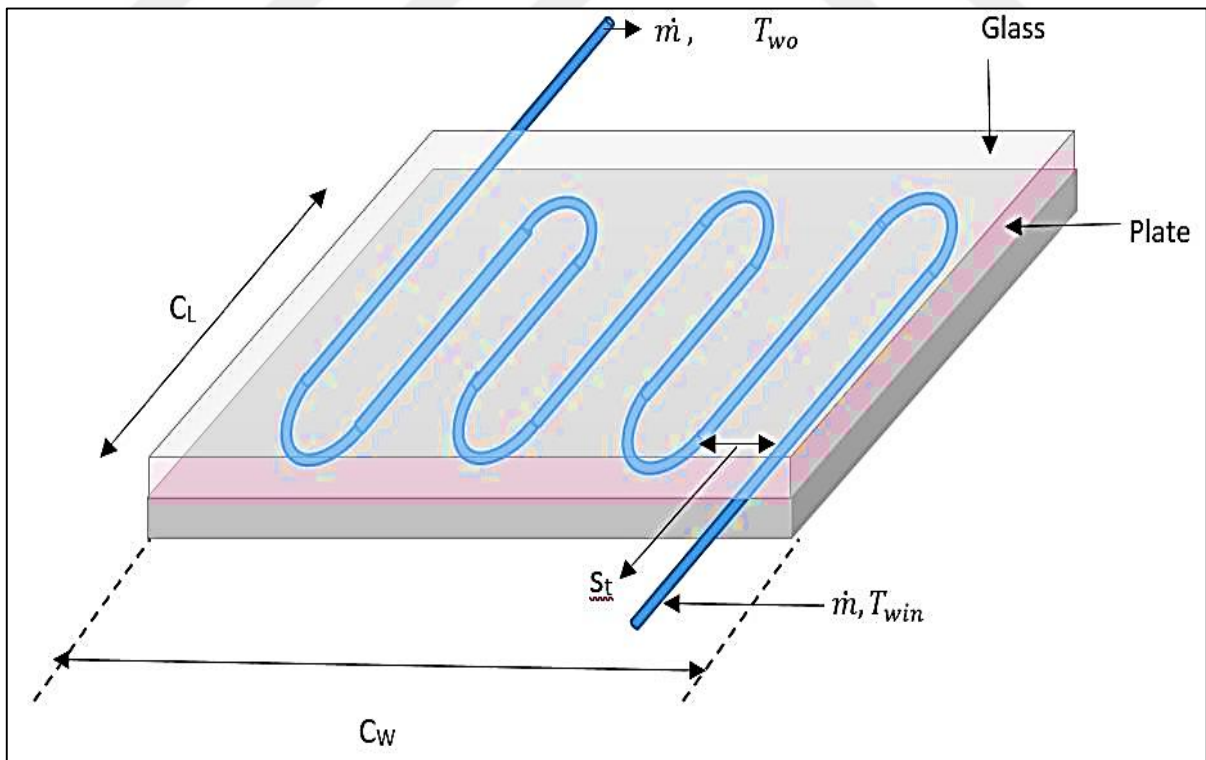


Figure 3.6: PCM-filled flat plate collector diagram.

The ratio of useful energy to pure incident energy on the collector is known as thermal efficiency. The incident energy manifests as incident solar radiation, which manifests as both direct and diffuse radiation as explained in the preceding section, on the collecting area. Figure 3.7 presents the total energy balance and the flat plate collector parts that interested in the present work. The indecent energy is divided into three parts, reflected, absorbed, and transmitted by the PCM layer and water.

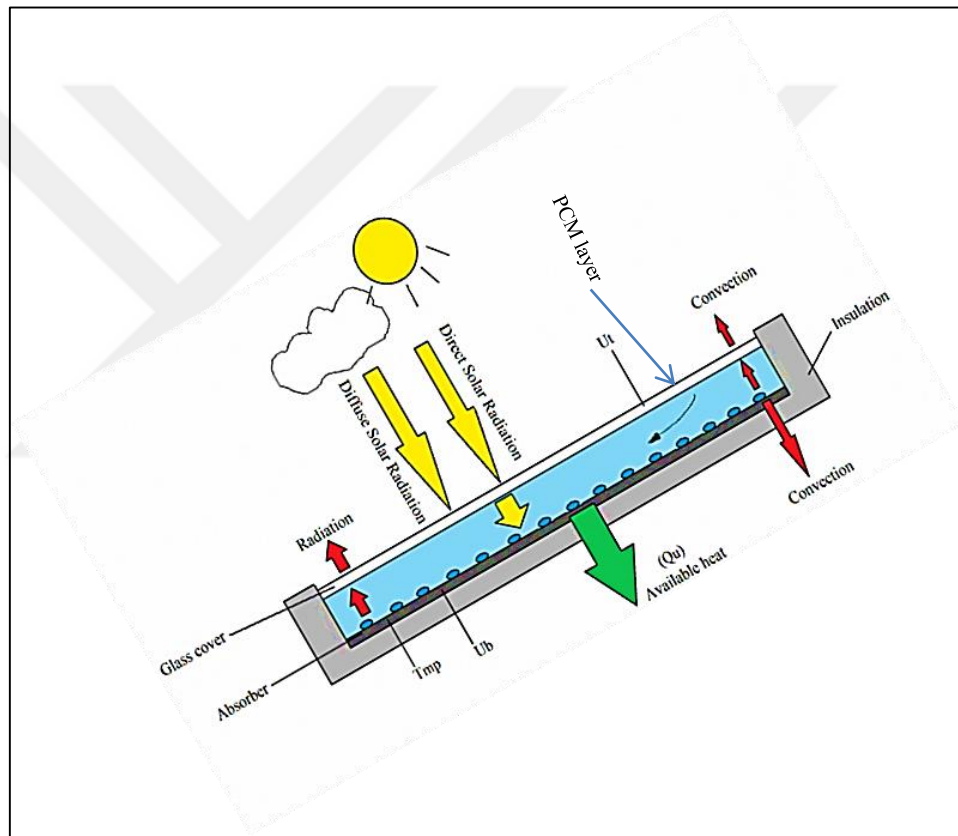


Figure 3.7: Schematic view of the PCM-filled flat plate collector.

The following presumptions were used to write all of the correlations:

- a- There are non-steady state conditions for the collector intensity.
- b- The PCM layer is placed inside the collector.
- c- Only conduction regulates heat transfer because there is no discernible local motion of the liquid due to the difference in densities between liquid and solid PCM.

- d- All the features of the homogeneous, isotropic PCM are distinct in the solid and liquid phases.
- e- The HTF is a turbulent, incompressible flow of water.
- f- The HTF's inflow velocity and temperature are constant, and insulation's thermal capacity is insignificant.
- g- The HTF and the thermal characteristics of the materials employed stay unchanged.
- h- At a temperature less than the melting point, a PCM layer is initially solid.
- i- Fluid has a negligible energy dissipation.

3.3.1 Thermal Analysis on The Solid and Phase Change Material (PCM)

Equation of the energy balance to the flat plate collector is presented as [39]:

$$I_s \times A_c = Q_{loss} + Q_u + Q_{stg} \quad (3.15)$$

Where I_s is solar radiation, A_c is the collector's surface area, Q_{loss} is the heat lost by the collector, and Q_u is the energy that can be used. The energy kept in the collector is known as Q_{stg} . The glass cover reflects some of the solar radiation that strikes the collector back into the surrounding air. Another portion is absorbed by the cover, and the final portion is sent as short-wave radiation to the absorber plate through the cover.

As a result, the solar energy that the collector has absorbed is transformed by multiplying it by the sum of the glass's rate of transmission and the absorber plate's rate of absorption. As a result, [39] is the input heat, Q_i , for the solar collector on the PCM layer:

$$Q_i = I_s(\tau\alpha) \times A_c \quad (3.16)$$

The heat loss can be determined by [39]:

$$Q_{loss} = A_c U_l F' (\overline{T_w} - T_a) \quad (3.17)$$

Where F' is the collector efficiency factor, T_a is the atmospheric temperature and $\overline{T_w}$ is the mean water temperature, which is determined by [40]:

$$\overline{T_w} = T_{win} + \frac{Q_u/A_c}{F_R U_l} (1 - FF) \quad (3.18)$$

Where T_{win} is the water inlet temperature, FF is the flow factor, and is given by [38]:

$$FF = F_R/F' \quad (3.19)$$

From equation 3.18, the outlet water temperature T_{wo} is determined by:

$$T_{wo} = 2 \times \overline{T_w} - T_{win} \quad (3.20)$$

U_l is the collector's overall heat transfer coefficient, which is given by [39]:

$$U_l = U_t + U_b + U_e \quad (3.21)$$

Where, U_b , U_t and U_e are the bottom, top and edge heat losses of the collector, which are calculated by [40]:

$$U_b = \frac{k_b}{l_b} \quad (3.22)$$

$$U_e = \left(\frac{k_e}{l_e}\right) \times \frac{2(C_L + C_w)C_h}{C_L \times C_w} \quad (3.23)$$

Where k_b and k_e are the back and edge insulation thermal conductivities, respectively. l_b and l_e are back and edge insulation thickness, C_L , C_w and C_h are the collector length, width and height, respectively.

$$U_t = \left[\frac{N}{\frac{C}{T_{mp}} \left[\frac{(T_{mp} - T_a)}{(N+f)} \right]^e} + \frac{1}{h_w} \right]^{-1} + \frac{\sigma(T_{mp} + T_a)(T_{mp}^2 + T_a^2)}{(\varepsilon_p + 0.00591N \times h_w)^{-1} + \frac{2N + f - 1 + 0.133 \times \varepsilon_p}{\varepsilon_g} - N} \quad (3.24)$$

Where N is the number of glasses covers, σ is Boltzmann constant, ε_p and ε_g are the plate and glass cover emittance, respectively, and T_{mp} is the collector's mean plate temperature which is calculated by [40]:

$$T_{mp} = T_{win} + \frac{Q_u/A_c}{F_R U_l} (1 - F_R) \quad (3.25)$$

$$f = (1 + 0.089h_w - 0.1166h_w \varepsilon_p)(1 + 0.07866N) \quad (3.26)$$

$$C = 520 \times (1 - 0.000051\beta^2) \text{ for } 0^\circ < \beta < 70^\circ \quad (3.27)$$

$$e = 0.43(1 - \frac{100}{T_{mp}}) \quad (3.28)$$

Where β is the collector tilt angle and h_w is the wind convective coefficient and is determined as [46]:

$$h_w = 5.7 + 3.8 \times V_w \quad (3.29)$$

After calculating U_t , now, the efficiency factor F' can be determined by [40]:

$$F' = \frac{\frac{1}{U_t}}{s_t(\frac{1}{U_t[(s_t-d)F+d]} + \frac{1}{\frac{kb}{C_L} + h_f \pi d_i})} \quad (3.30)$$

Where s_t is the tube spacing in meters, d_i and d_o are the inner and outer tube diameters, C_L is collector length and h_f is the convective coefficient. F is collector fin efficiency and determined by [39]:

$$F = \frac{\tanh[\sqrt{\frac{U_t(s_t-d)}{k\delta}}]}{\sqrt{\frac{U_t(s_t-d)}{k\delta}}} \quad (3.31)$$

Where: k present thermal conductivity of the plate and δ is the plate thickness in meters.

The heat removal factor F_R is given by [39]:

$$F_R = F' \times \frac{\dot{m}C_p}{A_c U_l F'} \left(1 - \exp\left(-\frac{A_c U_t F'}{\dot{m}C_p}\right) \right) \quad (3.32)$$

Now, the collector's usable energy is determined by [40]:

$$Q_u = F_R A_c [I_s(\tau\alpha) - U_l(T_{win} - T_a)] \quad (3.33)$$

Collector performance, which is calculated as the ratio of usable gain during a certain incident of solar energy for the same time period presents the collection efficiency. [40] estimates the thermal effectiveness of the collector.

$$\eta_c = F_R(\tau\alpha) - F_R U_l \quad (3.34)$$

A suitable and simplified variant of the differential energy equation of heat flow in solid paraffin is shown below.

$$\frac{\partial T_s}{\partial t} = \frac{k_s}{\rho c_p} \left(\frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} \right) + \frac{Q_{gen}}{\rho c_p} \quad (3.35)$$

Thermal energy balance of PW layer is.

$$Q_{in} = Q_{out} + \rho c_p \frac{dT}{dt} dv \quad (3.36)$$

Where ρ is PCM's density in kg/m^3 , c_p is PCM's specific heat in kJ/kg .

$\frac{dT}{dt}$ is temperature gradient with time $^\circ\text{C/s}$.

3.3.2 Thermal Analysis of Fluid

Heat transfer through fluid, HTF's velocity, and temperature fields are governed by the following system of two-dimensional transient differential equations [41].

Mass conservation equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial y}(\rho v) = 0 \quad (3.37)$$

Forced conservation equation:

x-momentum equation

$$\frac{\partial}{\partial t}(\rho u) + \frac{\partial}{\partial x}(\rho u u) + \frac{\partial}{\partial y}(\rho u v) = -\frac{\partial p}{\partial x} + \mu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \quad (3.38)$$

y-momentum equation

$$\frac{\partial}{\partial t}(\rho v) + \frac{\partial}{\partial x}(\rho u v) + \frac{\partial}{\partial y}(\rho v v) = -\frac{\partial p}{\partial y} + \mu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) - \rho g \beta (T_\infty - T) \quad (3.39)$$

Energy conservation equation

$$\frac{\partial}{\partial t}(\rho c_p T) + \frac{\partial}{\partial x}(\rho u c_p T) + \frac{\partial}{\partial y}(\rho v c_p T) = K \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) + q''' \quad (3.40)$$

3.4 TURBULENT MODEL

The turbulence model that simulates turbulence eddies most frequently is the (k-ε) model. The transport equation for kinetic turbulence energy (k) and its rate of dissipation (ε) is required by this model. The equation model (k-ε) is frequently used in engineering applications and can be applied to a wide range of various flows. It is now one of the best-tested turbulence models because it can handle complex recirculation flows in a lot less computer time than other models. The mechanisms that generate turbulent kinetic energy are the main focus of the k-model. To be effective in a general-purpose CFD code, a turbulence model needs to be widely applicable, accurate, straightforward, and affordable to execute. The three different model types are standard, RNG, and Realizable. The (k-ε) model, which may be stated as follows for three dimensions, is used in the current study to solve turbulent flow. Cartesian coordinates as follows [41]:

Turbulent kinetic energy (k):

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_i} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_i} \right] + G_k - \rho \varepsilon + S_k \quad (3.41)$$

and its rate of dissipation (ε).

$$\frac{\partial(\rho \varepsilon)}{\partial t} + \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_i} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial \varepsilon}{\partial x_i} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} (G_k + C_{3\varepsilon}) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} + S_\varepsilon \quad \dots (3.42)$$

Where G_k is the mean velocity gradients that induce the generation of turbulent kinetic energy, G_b is the neutrally buoyant generation of turbulent kinetic energy, and Y_M represents the fluctuating dilation compressible turbulence. S_ε and S_k are source terms defined by user. $C_{1\varepsilon}$, $C_{2\varepsilon}$, $C_{3\varepsilon}$, and C_μ are constants as follow:

$C_{1\varepsilon}=1.44$, $C_{2\varepsilon}=1.92$, $C_{3\varepsilon}=\tanh\frac{v}{u}$, $C_\mu=0.09$, σ_k and σ_ε are values for the turbulent kinetic energy and its rate of dissipation. Furthermore, derived by experiment, these have the following definitions:

$$\sigma_k=1.0, \quad \sigma_\varepsilon=1.3$$

Turbulent (or eddy) viscosity at each site by:

$$\mu_t = \rho c_\mu \frac{k^2}{\varepsilon} \quad (3.43)$$

where c_μ is constant and defined above.

G_k and G_b are representing the kinetic energy of the turbulence caused by the buoyancy and mean velocity gradient, respectively.

The dilation dissipation term, Y_M , which accounts for turbulence from compressibility effects and is used in the ongoing investigation because it uses fairly low velocities, is as follows:

$$Y_M = 2\rho\varepsilon M_t^2 \quad (3.44)$$

Where M_t is the turbulent Mach number, given as:

$$M_t = \sqrt{\frac{k}{a^2}} \quad (3.45)$$

Where a is the speed of sound ($\equiv \sqrt{\gamma RT}$), Y_M (dilation dissipation) is generally not taken into account while modeling incompressible flow.

3.5 THERMAL ENERGY STORAGE TANK WITH PCM INTEGRATED AND HELICAL COIL HEAT EXCHANGER ANALYSIS

By resolving the Navier-Stokes and energy equations for the storage medium and heat exchanger, it is possible to simulate the 2D-transient model with forced convection at turbulent flow (4L/min) numerically. The aim of the solution is to calculate the thermal stored energy,

PCM melting time, and PCM mass fraction under various temperature conditions. In order to assess the impact of PCM performance during the day, thermal load utilizing a helical coil is also explored.

3.5.1 Physical Description

For the purpose of using the heat generated from solar heating water for domestic or factory applications, a heat exchanger was used inside the hot tank to absorb energy. Figure 3.8 shows the helical coil heat exchanger with all dimensions and directions of flow is modeling by ANSYS 2022 R1 Design Modeler. The coil outer diameter of helical coil heat exchanger is 30 cm, and copper tube outside diameter is 12.25 mm. This coiled tube is fixed inside the storage tank.

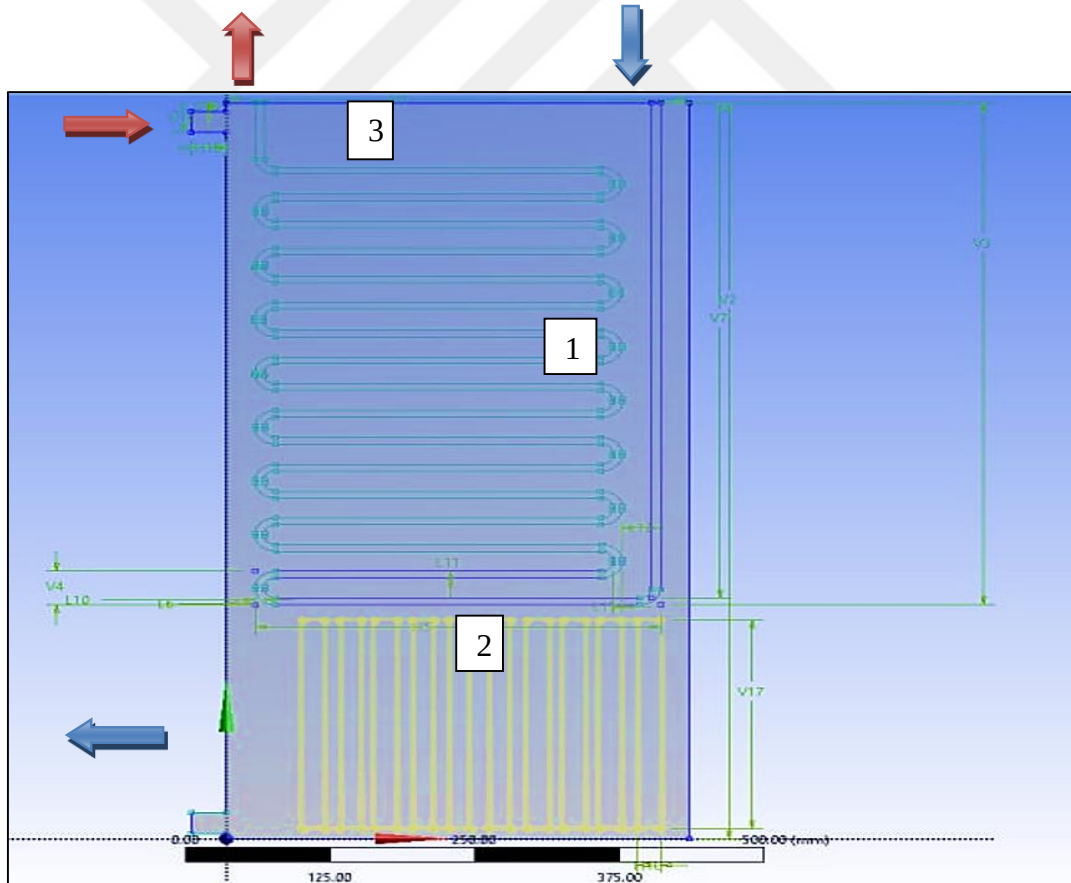


Figure 3.8: Simulation model of helical coil heat exchanger inside storage tank. The numerical model by ANSYS Design Modeler of storage tank component shown in figure 3.8 as follow: a. Helical coil heat exchanger, b. PCM capsules, 3. Water storage tank.

Figure 3.9 depicts a conceptual 2D computational model for the TES. A cylindrical TES tank and cylindrical PCM capsules make up the device. TES tank with a 40 cm diameter and 88 cm height that is filled with water. The HTF exit is located at the bottom of the water tank, while the HTF input from the collector is located at the top. HTF enters the inlet at a constant temperature and speed. PCM capsules with a 2 cm diameter and 20 cm height are packed with PW as two layers, which melts to provide the hot water's heat-absorbing component. Each layer consists of twenty number of capsules (5 kg). The PCM melts and changes from the solid phase to the liquid phase as a result of the heat transfer from the HTF to the PCM. The impact of 40 PCM capsules with PW (10 kg) and the water flow rate for both the inflow water and the heat exchanger water is investigated using numerical simulation.

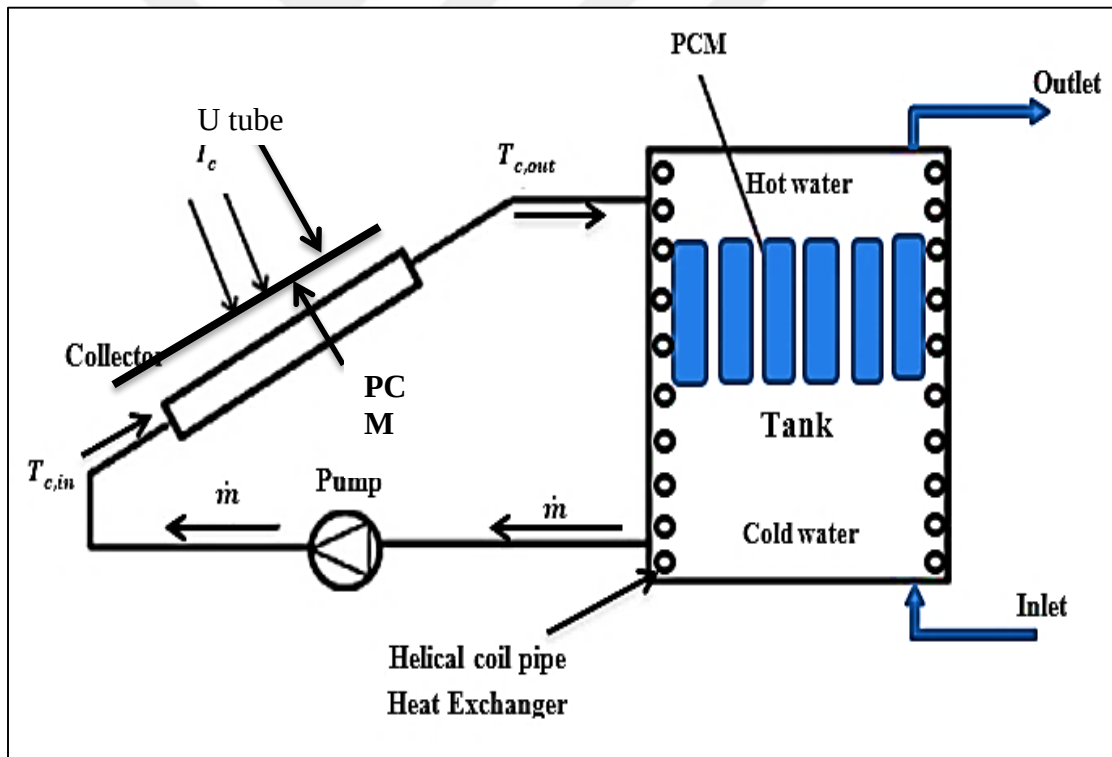


Figure 3.9: Simulation model of solar collector and TES tank with PCM integrated and helical coil heat exchanger.

Important presumptions are included for the transient condition model for both the PCM melting process within the TES tank and the heat exchanger-developed model.

- a. PCM (Paraffine Wax) undergoes transitory processes, and temperature is a function of sun intensity in a 2D model.
- b. The water inside the tube is counter flow.
- c. The fluid flow is Incompressible flow and Newtonian.
- d. The PW has uniform, isotropic characteristics.
- e. The thermo-physical characteristics of PCM and water remain constant.
- f. There is very little volumetric expansion with PCM.

3.5.2 Storage Tank Analysis

TES system modeling was done with ANSYS-FLUENT 2020 R1 and is based on the enthalpy formulation approach. Where liquid fraction denotes the portion of each cell domain's volume that is monitored in a liquid state. In the mushy zone, the liquid fraction increases when heated by hot water from 0 to 1, but under normal conditions, it falls from 1 to 0 due to heat loss and hardens. Energy balance between heat gains and losses through the heat exchanger is performed.

3.5.2.1 The PCM governing equation

Because the sensible heat of the PCM, like that of PW, is 100 times less than its latent heat, as is the case for other solids, less material is required to store the same amount of heat. Because PCM stores heat close to its melting point, the output temperature of the collector is stabilized when PCM is used as the storage medium [8]. Due to the tiny capacity, high heat conductivity in the storage medium is thus no longer necessary [9].

A suitable problem for PCM is the Stefan problem. By converting the power equation into a nonlinear equation system with an enthalpy, the enthalpy is playing a crucial part in the solution of the Stefan problem. The mushy phase spontaneously forms between the two phases after the interface boundary criteria, hot water and heat exchanger are met into equation below [42].

$$m_p c p_p \frac{dT_p}{dt} = \dot{Q}_{cond}^{pcm} + \dot{Q}_w = Q \text{ heat exchanger} \quad (3.46)$$

Where $m_p c p_p \frac{dT_p}{dt}$ is the rate of heat storage through the overall PCM, $\dot{Q}_{cond}^{pcm}$ is the rate of conduction through PCM. $\dot{Q}_w$ is the rate of heat through the water.

The enthalpy transport equation is used to analyze the paraffin phase change phenomenon. This equation presupposes that sensible and latent heat is added to form specific enthalpy ΔH [42]:

$$\rho_p \cdot \frac{\partial H}{\partial \tau} = k_p \cdot \nabla^2 \cdot T \quad (3.47)$$

$$H = h + \Delta H \quad (3.48)$$

$$h = h_{ref} + \int_{T_{ref}}^T C_p dT \quad (3.49)$$

Where T_{ref} and h_{ref} are the reference temperature, reference enthalpy respectively. The latent heat content ΔH of PCM written as:

$$\Delta H = \beta \cdot L \quad (3.50)$$

Where β is liquid fraction through the phase change of PCM between the solid and liquid phase when the temperature is ($T_S < T < T_L$):

$$\beta = \begin{cases} 0 & \text{solid phase} & T < T_S \\ \frac{T - T_S}{T_L - T_S} & \text{mushy phase} & T_S < T < T_L \\ 1 & \text{liquid phase} & T > T_L \end{cases}$$

3.5.2.2 Energy balance for the HTF

The energy balance between energy through the water and all heat gained by water and PCM is obtained from equation below.

$$m_w c_{p_w} \frac{dT_w}{dt} = \dot{Q}_{flow} + \dot{Q}_{conv}^{pcm/w} = \dot{Q}_{exchanger} \quad (3.51)$$

Where $\dot{Q}_{flow} = m_w c_{p_w} \frac{dT_w}{dt}$ is the energy through the water per unit time, is heat transfer through water from collector per unit time, $\dot{Q}_{conv}^{pcm/w}$ is the rate of convective energy through PCM.

The ratio between the heat actually retrieved by the HTF during the PCM solidification process and the heat received from the HTF during the PCM melting phase that can be stored in the PCM is the definition of the effectiveness of PCM thermal energy storage. The efficiency of heat storage can be mathematically represented as [42]:

$$\eta = \frac{\text{Energy recovered from PCM during discharging}}{\text{energy input to PCM during charging}} \quad (3.52)$$

$$\eta = \frac{Q_{\text{discharging}}}{Q_{\text{charging}}} = \frac{m_{\text{dis}} C_p w (T_{\text{out}} - T_{\text{in}})}{m_{\text{ch}} C_p w (T_{\text{i}} - T_{\text{out}})} \quad (3.53)$$

Where $Q_{\text{discharged}}$ is the actual heat retrieved by the HTF during the PCM solidification process. Q_{charging} is the rate of heat removing by HTF during the PCM melting process.

3.5.2.3 Heat exchanger equations

The issue at hand is the forced convection through the heat exchanger's counter-flow heat exchanger's helical coil. The helical coil heat exchanger had used transient energy from hot water received from the collector or PCM at night. As in the thermal analysis of HTF through the collector, the current model is controlled by partial differential equations that contain continuity, momentum, and energy equations. Equations (3.37), (3.38), (3.39), and (3. 40) are the Navier-Stokes and energy equations that regulate the mathematical model of the thermal storage tank.

3.6 INITIAL AND BOUNDARY CONDITIONS

The input and outflow boundary conditions must be given in order to obtain accurate simulation solutions. There are several inlet boundaries that can be employed, including mass flow inlet, velocity inlet, and inlet pressure, with mass flow inlet being most frequently used to define the intake mass flow rates. The outlet boundary condition could be defined using outflow in other circumstances. On each occasion, the TES tank's starting temperature was 298 K. The intake water temperatures for the heat exchanger alone were 298, 303, and 308 K, while the inlet mass-flow rates for the collector and heat exchanger combined were 0.033 kg/s for 2 L/min and 0.033 for 2 L/min, 0.0667 kg/s for 4 L/min.

3.6.1 Initial Conditions

$$(time) = 0, T_{(x,y,t=0)} = 298 \text{ K}$$

3.6.2 Boundary Conditions

There was a no-slip marking on the wall.

$$U=V=0 \quad (3.53)$$

Assuming that the wall, top, and bottom wall boundary conditions are adiabatic.

$$\frac{\partial T}{\partial x} = \frac{\partial T}{\partial y} = 0 \quad (3.54)$$

Collector tube and helical tube with water is coupling boundary layer.

The boundary conditions for the inlet water of the heat exchanger were as follows:

$$t > 0, T = T_{in}, \dot{m} = \dot{m}_{in}$$

The boundary conditions for the inlet water of the collector were as follows:

$$t > 0, T = T_{in}, \dot{m} = \dot{m}_{in}$$

Figure 3.10 shows the important boundary conditions for storage tank which are selected to the solution program.

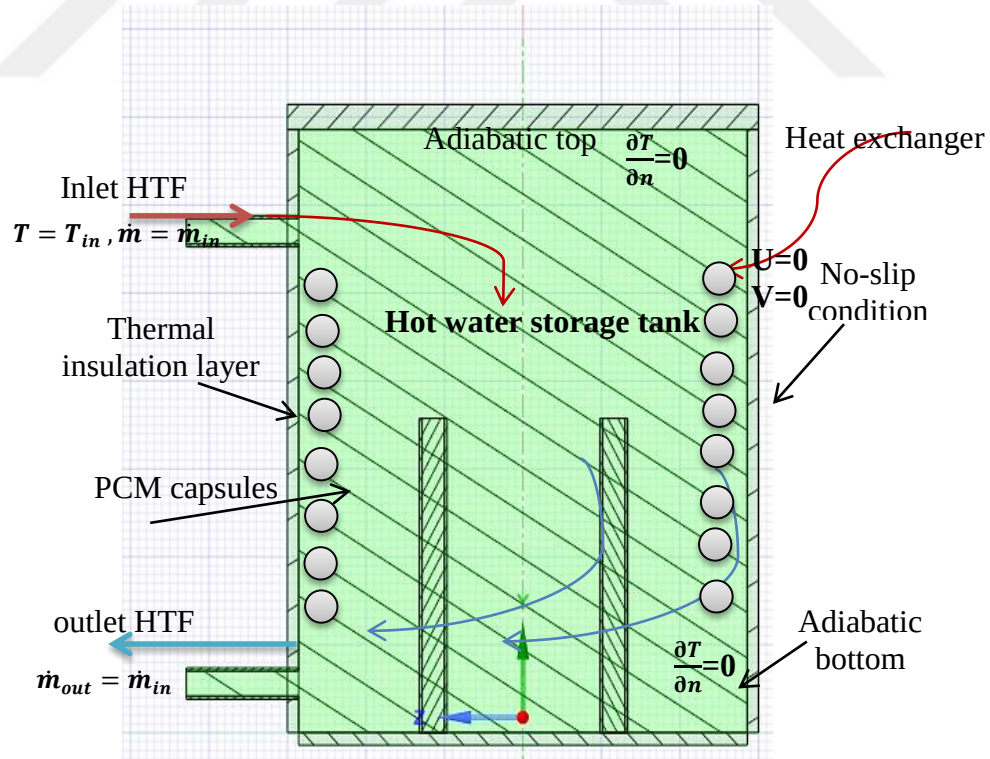


Figure 3.10: Schematic drawing of storage tank to define boundaries condition.

In Table 3.4, the boundary conditions are given.

Table 3.4: Boundary conditions

Location	Boundary condition	Comments
Inlet	$T = T_{in} , \dot{m} = \dot{m}_{in}$	Volume flow inlet and temperature condition (for collector and heat exchanger)
Outlet	$\dot{m}_{out} = \dot{m}_{in} , P = 0$	Outflow condition (for collector and heat exchanger)
Wall	$U=V= 0$	No-slip condition
Top, and bottom-wall of tank	$\frac{\partial T}{\partial x} = \frac{\partial T}{\partial y} = 0$	Adiabatic condition
Points of solid-fluid contact interface	$k_s \frac{\partial T_s}{\partial n} = k_f \frac{\partial T_f}{\partial n}$ and $T_s=T_f$	Where, k_s and k_f are thermal conductivity of solid, liquid of PCM and coil pipe respectively

3.7 MAIN THERMAL PARAMETERS

There is important parameter which are governed the problem of forced convection.[44].

3.7.1 Reynolds Number (Re)

The dimensionless Reynolds Number (Re) is the ratio of inertia force to viscous force.

$$Re = \frac{\rho u D}{\mu} \quad (3.55)$$

3.7.2 Prandtl Number (Pr)

Prandtl number (Pr) depicts the relationship between heat diffusivity and momentum diffusivity.

$$Pr = \frac{\nu}{\alpha} \quad (3.56)$$

3.7.3 Nusselt Number (Nu)

Nusselt number (Nu) is the ratio between convection heat transfer to the conduction heat transfer within a fluid.

$$Nu = \frac{\text{convection heat transfer}}{\text{conduction heat transfer}} = \frac{hD}{k} \quad (3.57)$$

3.7.4 The Mean Temperature (T_b)

Bulk temperature of the worked fluid is calculated in Fluent by volume integral by means of mass-weighted average which is also called (bulk) temperature is expressed by [44]:

$$T_b = \frac{\int T_i \rho dV_i}{\int \rho dV_i} = \frac{\sum_{i=1}^n T_i \rho |V_i|}{\sum_{i=1}^n \rho |V_i|} \quad (3.58)$$

Where T_b is the bulk temperature of working fluid in a closed cavity. dV is the differential volume for each control volume.

The average surface temperature is calculated by surface integral by means of the area-weighted average.

$$T_s = \frac{1}{A} \int T_i dA_i = \frac{1}{A} \sum_{i=1}^n T_i |A_i| \quad (3.59)$$

Where T_s is the mean temperature of the storage plate. Now, the average convection heat transfer coefficient can be calculated by selecting the obtained T_b and T_s in the equation below:

3.7.5 Heat Transfer Coefficient

Convective heat transfer coefficient can be calculated from the energy balance between solar intensity and fluid convection equations as below [44].

$$h = \frac{q''}{\bar{T}_s - T_b} \quad (3.60)$$

The value of the heat flux is calculated from the solar intensity.

3.7.6 Effectiveness

Heat exchanger effectiveness presents the important thermal performance and can be derived from the energy balance between hot and cold fluids.

$$Q \doteq \dot{m}_c C p_c \Delta T_c = -\dot{m}_h C p_h \Delta T_h \quad (3.61)$$

Where the subscripts c and h refer to cold and hot fluid and the i , o to inner and outer

$$\Delta T_c = T_{co} - T_{ci} \quad \text{and} \quad \Delta T_h = T_{ho} - T_{hi} \quad (3.62)$$

All heat exchangers are formed as multi-layer pipe.

$$Q = U A_s F \Delta T_{ln} = U_i A_i F \Delta T_{ln} = U_o A_o F \Delta T_{ln} \quad (3.63)$$

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} \quad (3.64)$$

$$\Delta T_{ln} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} \quad (3.65)$$

Where Q is heat transfer between two fluids and U is the overall heat transfer coefficient.

$A_s = 2\pi r L$ (separated area), $A_i = 2\pi r_i L$ [Inner separated area]

$A_o = 2\pi r_o L$ [outer separated area of the pipe]

F = correction factor $0 \rightarrow 1$

$$\varepsilon = \frac{q_{\text{actual}}}{q_{\text{max}}} = \frac{C_h (T_{hi} - T_{ho})}{C_{\min} (T_{hi} - T_{ci})} = \frac{C_c (T_{co} - T_{ci})}{C_{\min} (T_{hi} - T_{ci})} \quad (3.66)$$

3.7.7 PCM Heat Transfer [42]

The useful heat gained through the collector and storage tank is divided into four sections:

3.7.7.1 Heat giant from solar intensity to water

At solar collector U tube is heated by solar intensity through the PCM layer.

$$Q_{us} = m_w C_{pw} \Delta T_w \quad (3.67)$$

3.7.7.2 Heat transfer through the solid PCM

At starting time, the water tube is covered by a solid PCM layer, which means heat giant:

$$Q_{giant} = \frac{m_{s,pcm} C_{pcm} \Delta T_{s,pcm}}{3600} \quad (3.68)$$

3.7.7.3 Heat transfer through the phase change PCM

When the PCM layer receives solar energy directly, sensible energy is beginning and until the melting part of this layer, latent heat is beginning.

$$Q_{lat} = \frac{m_{pcm} L_{pcm}}{3600} \quad (3.69)$$

Where L_{PCM} is the talent heat for PCM and the m_{cpm} is the flow rate of the liquid PCM

3.7.7.4 Heat transfer through the liquid PCM

When the PCM layer receives solar energy directly, melting all of this layer, sensible heat is begun.

$$Q_{sensible} = \frac{m_{L,pcm} C_{L,pcm} \Delta T_{L,pcm}}{3600} \quad (3.70)$$

The daily effectiveness of the PCM-equipped collector is [42]:

$$\eta_{with\ pcm} = \frac{m_w \cdot C_{pw} \Delta T_w}{\Sigma (I A_{abs} + Q_{phase\ change} + Q_{s,pcm} + Q_{L,pcm})} \quad (3.71)$$

3.8 NUMERICAL METHODOLOGY

In ANSYS Workbench Space-Claim and the mesh of the geometry, all schematic geometry is produced. The simulation is carried out using the two-dimensional double precision code once the mesh has been loaded into ANSYS-FLUENT 2020R1. It is necessary for FLUENT to be able to distinguish between the two different fluid materials used in the PCM and HTF domains. Figure 3.11 shows the complete geometry which drawn by ANSYS workbench.

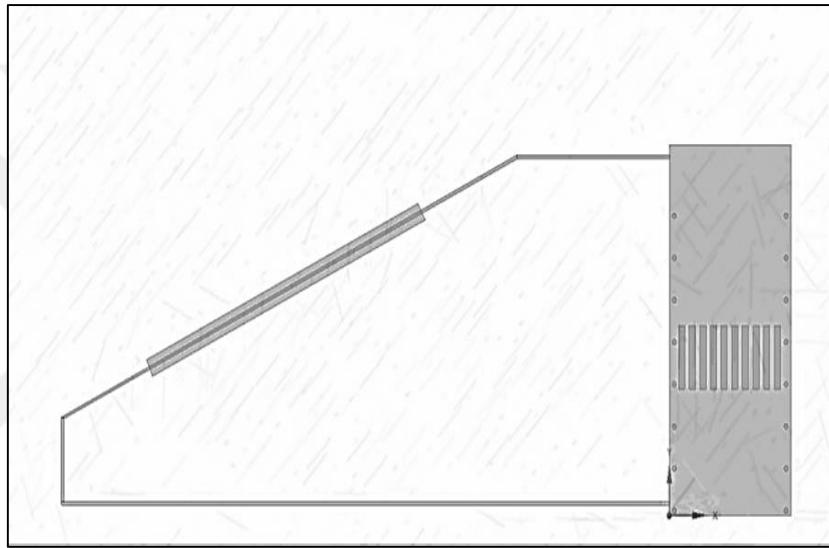


Figure 3.11: Schematic drawing by ANSYS Workbench software for the numerical geometry.

Each part of the geometry is determined at boundary conditions and defined by the program for the purpose of numerical solution and then move a step forward to the adjacent part in a step-by-step solution method. For example, the collector is will be defined as one part, and the boundary conditions, steps of the numerical solution shall be determined on it, so that the hot water will be inlet to the tank as shown in Figure 3.12.

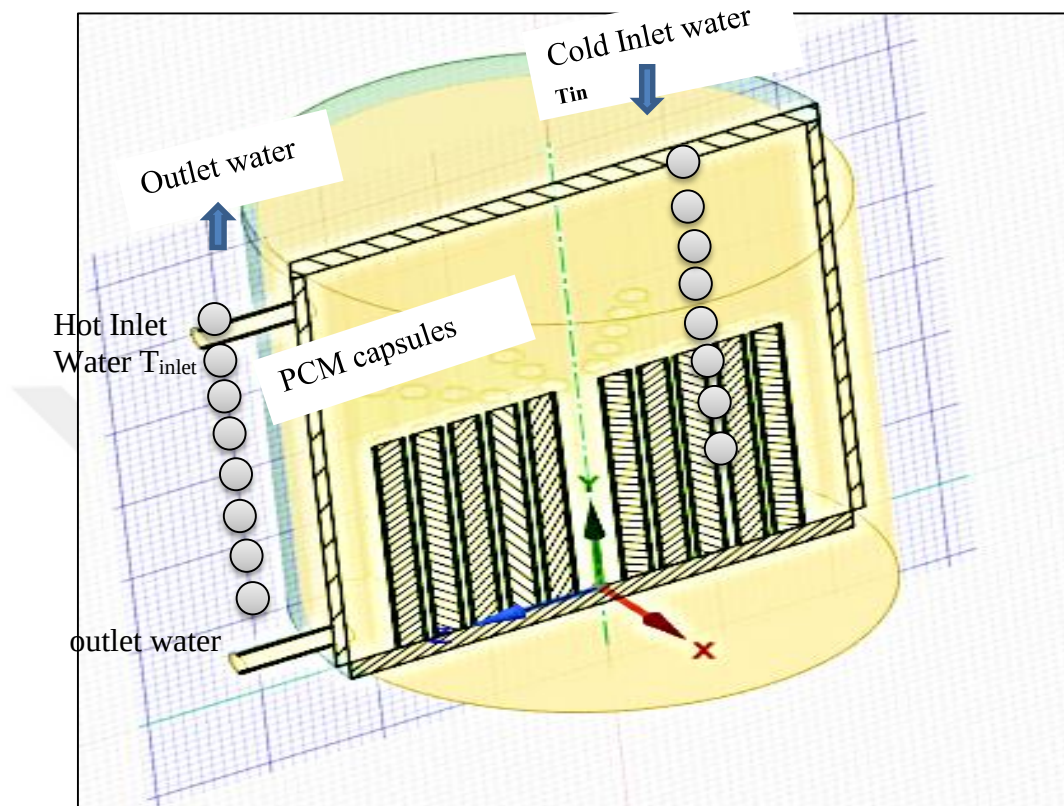


Figure 3.12: Cross section of storage tank with three components, helical coil heat exchanger, PCM capsules and collector water.

At the beginning of ANSYS FLUENT simulation setup is arranged as shown in table 3.1. For coupling velocity pressure equation there are number of schemes used like, SIMPLE, SIMPLER or SIMPLEC. Also, for turbulent flow in viscous layer is active and solidification and melting processes for PCM. For thermal analysis energy equation is doing.

Table 3.1: Setup for ANSYS FLUENT simulation.

Description	Type
Solver	Pressure-Based-SIMPLE
Time	Transient
Energy	On
In viscous	Turbulent flow
Solidification & Melting	On

The phase-changeable PW is employed as the PCM and water are used as the HTF. Table 3.2 lists the thermophysical characteristics of water, PCM, and galvanized iron assigned in FLUENT.

Table 3.2: Thermophysical parameters of copper pipe, water, PCM, and galvanized iron.

Parameters	Water	PCM	Copper	Galvanized iron
Density [kg/m^3]	998.2	834.36	8900	7869
Specific heat [kJ/kg K]	4.182	2.9 (solid) /2.2 (liquid)	0.39	0.4484
Thermal conductivity [W/m K]	0.6	0.22	400	79.86
Viscosity [kg/m.s]	0.0010	0.008	–	–
Melted degree [K]	273	324	1358	–
Melting heat [kJ/kg]	333.55	70.006	206	–
Solidus temperature [K]	–	292	–	–
Liquidus temperature [K]	–	300	–	–

3.8.1 Grid Independence

The grid independence test is one of the most important tests that should be performed on a numerical problem. The number of grids should have no bearing on the grid independence test's goal of validating the results. The number of grids formed always affects the outcomes of numerical problems. As a result, if the number of grids changes as well, the results vary also. The smallest number of grids at which the results remain unchanged when the number of grids is altered is known as the optimal grid size. A two-dimension geometry of TES tank with PCM capsules similar to the experimental setup was created by utilizing ANSYS-FLUENT software. Several simulation runs were carried out with changing total numbers of tetrahedral cells between (112,633 and 115,452) to study the convergence behavior. Grid influence was checked at mass flow rate and initial temperature of water HTF (298K and 0.033 kg/s), respectively, with 80 liters of water in the storage tank. In table 3.3, the numerical results (PCM liquid-fraction y_{pcm} and inlet water temperature in storage tank from the collector obtained were listed. During

the simulation, the results of a grid were chosen with element size 39 mm and elements numbers is 113,490 because the results with small size element were stabled.

Table 3.3: Grid effect check at coil pipe, $T_{in}=303$ K and $\dot{m}=0.05$ k/s.

S.no.	Size of elements (mm)	Numbers. of volume elements	Numerical results	
			y_{pcm}	T_{inlet} (K)
1	35.0	115,452	0.0033	395.60
2	39.0	113,490	0.0041	395.53
3	40.0	113,209	0.00357	395.69
4	41.7	112,699	0.0034	395.60
5	42.0	112,633	0.0041	395.53

3.8.2 Parameters for Converged Solution

Numerous tests conducted throughout the simulation revealed that specific parameters were crucial in achieving the converged solution. The following is a list of a few of these crucial parameters:

The PCM density varies with increase/decrease in temperature through its melting and solidification process. The FLUENT takes care of PCM density in the mushy zone. It is important to set the default reference values for the geometry and the values of PCM, HTF, and the properties of the model. The pressure-velocity coupling is solved using the SIMPLE (Semi-Implicit-Method for Pressure-Linked Equations) algorithm, and second-order upwind is chosen. In addition, the k- ϵ descriptions of the turbulence models are solved using the pressure-based solver.

Relaxation factors Can be used to stop the solution's divergence. Under relaxation factors, FLUENT had a default value of (1.0). Default Fluent values that provide a decent convergence are utilized, depending on the solution method [42].

Time step size is one of the most crucial factors that needed to be closely monitored to meet the convergence solution's requirements. In calculations, the time step is set to 10 s, with a

maximum of 20 iterations per time step. The solution process should be iterated until convergence is reached; for the energy equation, the convergence condition is set at 10^{-6} while all other variables are set at 10^{-3} .

3.8.3 Numerical Analysis

The numerical representation of temperature and velocity distributions in partial differential equations is explained in this section. PCMs provide a numerical solution that is more precise than the analytical solution and applicable to a variety of flow and thermal issues. Equations relating to fluid and heat movement can be solved using CFD as illustrated at Appendix A. In a variety of technical applications, including thermal storage in buildings and electronic cooling technology, the CFD software mimics the use of PCM. CFD may be used to create PCM thermal storages, which led to extremely accurate experimental results. The application of finite volume helps to clarify the computational behavior of the governing differential equations. The discretization equations are solved using the SIMPLE method according to the upwind scheme formula. Using the ANSYS FLUENT 2020R1 code, the flow field in a heating water system with PCM was numerically analyzed [45].

4. EXPERIMENTAL STUDY

4.1 GENERAL

This chapter includes information and specifics on the constructed experimental rig as well as descriptions of the experiments. Field tests on the solar heating system with thermal load model that had been tested at Basra city climate in September 2021, April, May, and July 2022, respectively, were carried out in the experimental part. PCM should be positioned at the collector and storage tank so that heat can easily and directly flow from the absorber to the storage medium and then, in the absence of sunlight, to the water. As a result, with the suggested configuration, the PCM only touched the absorber plate directly from below. By using an external load as a heat exchanger, the useful heat can be obtained.

4.2 THE DEVICE DESCRIPTION

The test rig is made up of two main parts: storage tank with PCM and a heat exchanger, and flat plate collectors loaded with PCM and heating pipe.



Figure 4.1: The experimental test rig

The experimental setup components shown with numbers in Figure 4.1 are as follows: 1. Storage tank, 2. Collector, 3. Thermocouples and digital, 4. Rotameters, 5. Pvc pipes.

The following sections provide explanations of each component's specifics:
Solar collector as shown in Figure 4.1. Frame with suitable angle to Basra position (30°) as shown in Figure 4.2.

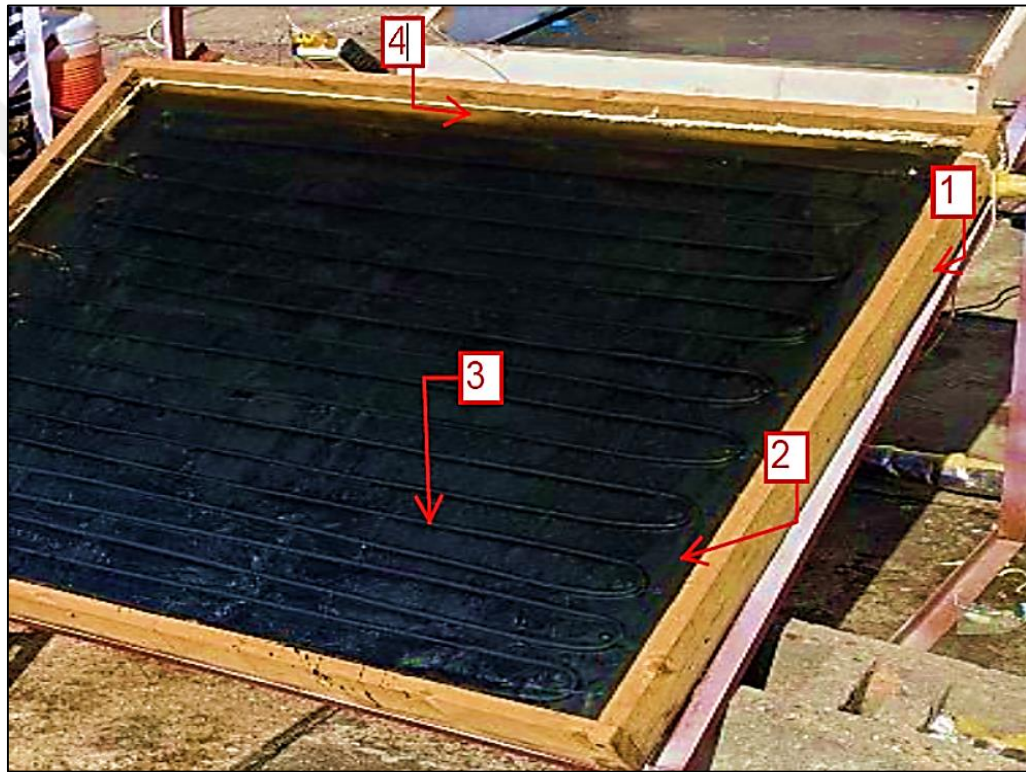


Figure 4.2: Flat plate heating water collector

The flat plate heating water collector components shown with numbers in Figure 4.2 are as follows: 1. Wood frame. 2. Black absorber (galvanized steel). 3. U tube copper pipe covered with PCM and 4. Glass cover.

The outer connection of PVC water pipes is 1.25 cm in diameter, check valves and rotameter for inlet water to collector. The details of each component are explained in the table 4.1 below:

Table 4.1: Components of flat plate collectors and dimensions

No.	Component	Dimensions	Properties
1	Absorber plate (galvanized steel) (2)	1m x 1m x 0.8 m	Thermal conductivity = 60 W/m K
2	Copper pipe (3)	do = 10 mm di = 8 mm	Thermal conductivity = 400 W/m K
3	Insulation (cork)	1 x 1 x 0.03 m	Thermal conductivity = 0.04 W/m K
4	Glass cover (4)	1m x 1m x 0.005 m	Absorptivity = 0.88
5	PCM layer (3)	5 cm thickness	Paraffine wax
6	Black observer plate (2)	0.1 cm thickness	-----
7	Thermocouples	8 thermocouples Ni-Cr and Ni-Al	-40 °C to 250 °C

Figure 4.3 shows the storage tank and all components with temperature measuring instruments of the PCM, inlet and outlet water flow. There are two rotameters, two electric water pumps, and two digital temperature readings. Also, the 16 K- type thermocouples were connected to the digital measuring device.

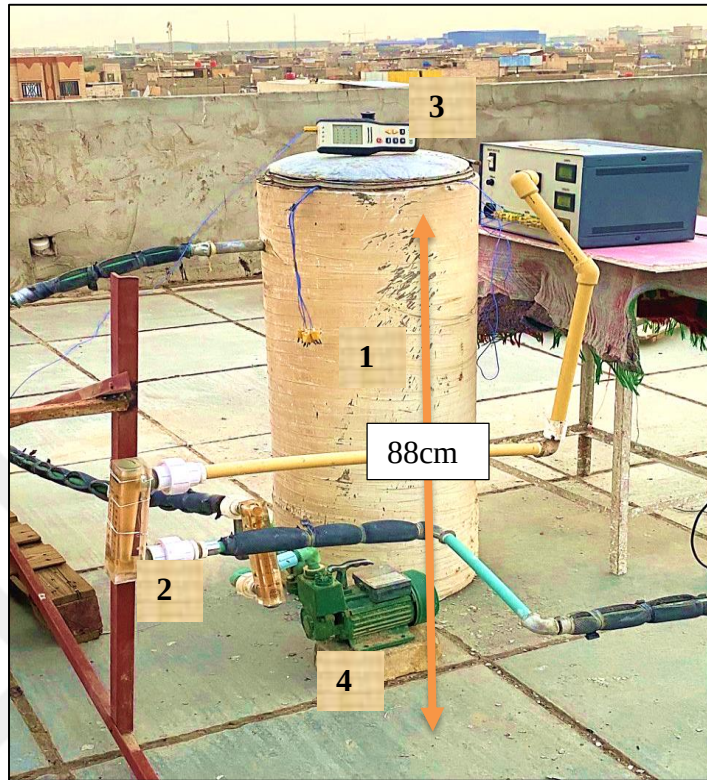


Figure 4.3: Photograph view of solar water heater (storage tank)

The storage tank components shown with numbers in Figure 4.3 are as follows: 1. storage tank, 2. rotameters, 3. digital thermocouples instruments and 4. water pump.

Storage tank size (80 Liter) consists of PCM 40 capsules size (10 kg) and Thermocouple's type K (No 16) as shown in Figure 4.4. Also, the storage tank with PCM capsules, thermocouples, a helical coil tube inside tank.

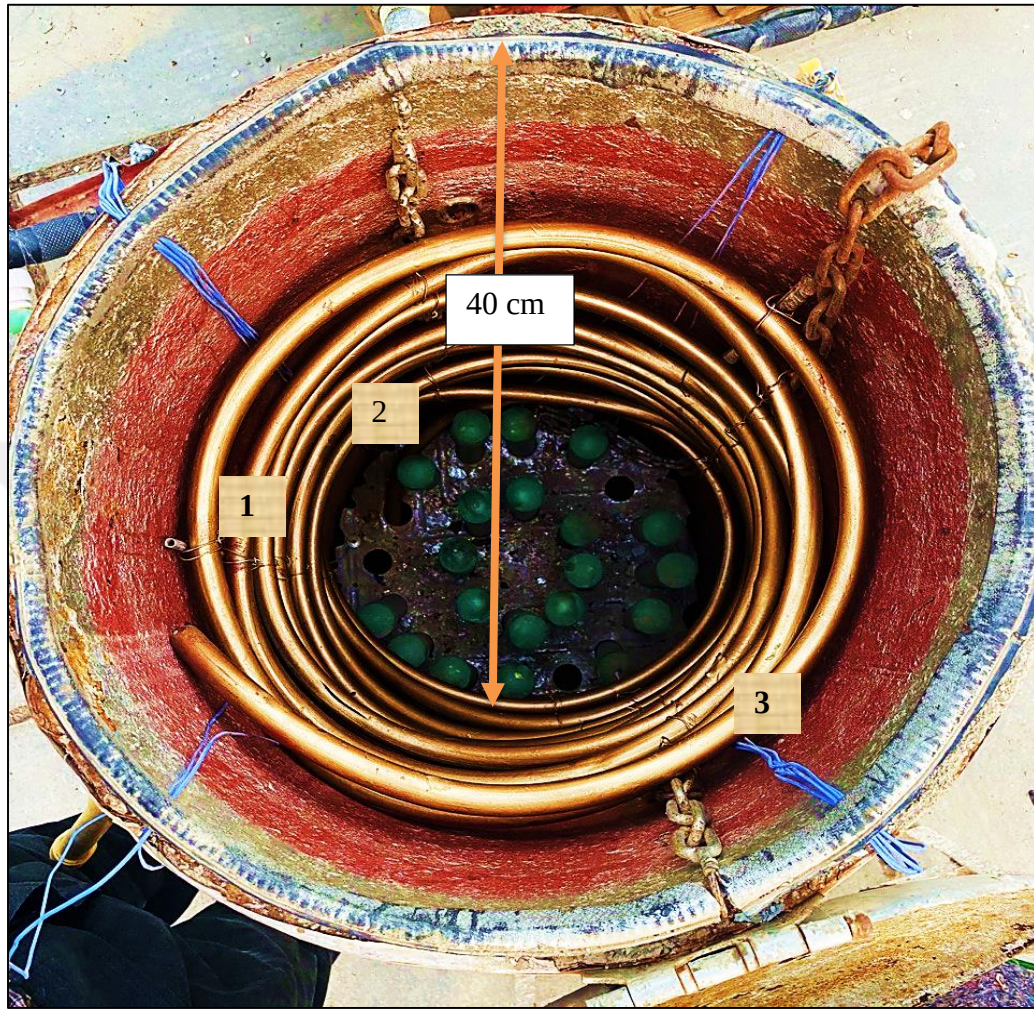


Figure 4.4: a.Helical coil inside the storage tank, b.PCM capsules and 3.Thermocouples.

Storage tank components with PCM capsules and dimensions are given in Table 4.2.

Table 4.2: Storage tank components with PCM capsules and dimensions

No.	Component	Dimensions	Properties
	Cylinder tank	88 cm length 40 cm diameter	Size (80) liter
2	Capsule of PCM	20 cm high ,22 mm diameter,40 samples	Weight (0.25kg for each capsule) Two layers (40 samples)
3	Copper pipe (helical coil pipe)	do= 10 mm di= 8 mm Step=10 mm (turn No=8)	Thermal conductivity = 400 W/m K
4	Insulation (cork)	1m x 1m x 0.03m	Thermal conductivity = 0.04 W/m K
5	Thermocouples type K	8 thermocouples Ni-Cr and Ni-Al	-40 °C to 250 °C

All components that support the completion of this work are given in the attached table 4.3.

Table 4.3: Accuracies components.

No.	Component	Dimensions	Properties
1	Water pump	Section 19 mm, head =40 m	50Hz, 220V 0.5Hp/0.37Kw
2	Rotameter	Acrylic _{max} =90 degree. Reading (1-10 LPM)	accuracy of $\pm 4\%$.
3	Solar Power Meter	Portable	2000 W/m ²
4	PVC pipes	Diameter 12.5 cm	High insulation PVC

Paraffin wax consists of a mixture of straight-chain n-alkanes, as both the latent heat and melting point increase with the chain length. In this present work, paraffin is employed as a phase change

material PCM with a melting point of 51°C is packed in each cylindrical capsule and collector. The original form of PCM is shown in Figure 4.5.



Figure 4.5: Blocks of Paraffine Wax

Paraffin wax's thermophysical properties [45] are given Table 4.4.

Table 4.4: Paraffin wax's thermophysical properties [45].

Parameters	Values
Density	834.36 [kg/m ³]
Specific heat	2.9 (solid) /2.2 (liquid) [kJ/kg K]
Thermal conductivity	0.22 [W/m K]
Melting temperature	324 [K]
Melting heat	70.006 [kJ/kg]
Solidus temperature	292 [K]
Liquidus temperature	300 [K]
Viscosity	0.008 [kg/m.s]

4.3 HOT WATER TANK INTEGRATED WITH PCM AND HEAT EXCHANGER

The 80-liter water tank is constructed from two layers of 2 mm thick galvanized steel. This tank is used to store water and is loaded externally using a helical coil heat exchanger and PCM. Table (4.2) list all PCM capsules and tube coil configurations. The water tank storage in its final configuration, with eight type K thermocouples, is shown in Figures. 4.1,4.3and 4.4. Two layers of glass wool are used to insulate every surface in the tank's layers.

4.4 EXPERIMENTAL INSTRUMENTS

4.4.1 Thermocouple Circuit and Test

At the solar collector eight thermocouples are divided as below: Three collections of - K-type Chrome-Aluminum were used to measure the local temperature of the bottom wall of the collector. Three thermocouples are attached to the PCM layer, while the two- K-type are fixed to the inlet and outlet collector water.

Eight thermocouples were connected to a selector switch, which was coupled to a digital temperature gauge type (heat transfer service unit). These thermocouples were calibrated against a common digital multimeter before the tests, and it was discovered that they were accurate to within 0.3 °C across the temperature measurement range.

At the storage tank eight thermocouples are divided as below: Two -K-type thermocouples fixed through the flow input and output heat each other up for the storage hot water tank. Four thermocouples were evenly spaced apart and joined to a PCM capsule. In order to measure the hot water inside the tank, Two-K type thermocouples fixed through the flow outside the helical coil heat exchanger were used.

In this thesis, the following instruments have been used to collect data:

4.4.2 Solar Power Meter

PV modules were tested at a 30° longitude-latitude angle, and the total solar energy they received was measured using a solar meter of type TES-1333. Table 4.5 lists the attributes and capabilities of the TES-1333 solar power meter gadget. Every half-hour during the testing

process, the amount of solar radiation was also measured and recorded by solar meter Figure 4.6.

Table 4.5: TES-1333 specifications.

Maximum Value	2000 W/m ²
Accuracy	±0.5%
Angular accuracy	Less than 60 °
Sampling Time	Approx. 0.4 second

Total solar energy they received was measured using a solar meter of type TES-1333 is shown in Figure 4.6



Figure 4.6: TES-1333 Solar Power Meter.

4.4.3 Rotameter

To measure the flow rate of collector water, a flow meter is mounted near the storage tank's outlet. The flow meter's operational range is between 1 and 18 LPM. This thesis flow meter is depicted in Figure 4.7. To measure the flow rate of cold water from the main supply flow meter

is mounted to inlet helical coil pipe. Both with a reading range of 1-18 LPM or 0.2 – 2 GPM) with an accuracy of $\pm 4\%$.



Figure 4.7: Flow meter.

4.4.5 Thermocouples

As seen in Figure 4.8 K-type thermocouples are used to gauge the temperatures in this system since they are suited for an extensive range of operational temperatures (-40°C - 250°C). Due to its nickel base and strong corrosion resistance, it can typically be used in all environmental conditions, including natural and chemical ones, and for the majority of applications, manufactured of K-type (Ni-Cr and Ni-Al). Additionally, it is compatible with K-type temperature measurement instruments and has a quick response time. The distribution of the 16 thermocouples is as stated above.

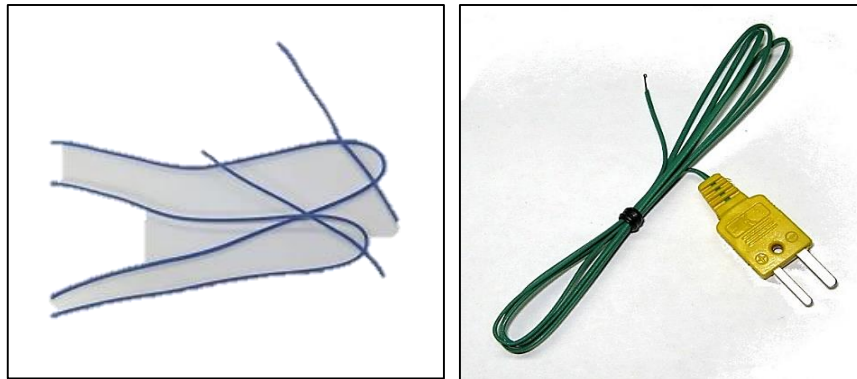


Figure 4.8: K-type thermocouple.

4.4.5.1 Temperature measurement

The backlit HT-9815 Digital-LCD display is simple to see. Figure 4.9 illustrates a 4-channel thermocouple thermometer (K-type thermocouple) that simultaneously displays and measures temperature. It offers conversion between Kelvin (K), Celsius ($^{\circ}\text{C}$), and Fahrenheit ($^{\circ}\text{F}$). Display of (Min, Max, Average Data, and Data Hold. Function), with 0.1°C precision.



Figure 4.9: Temperature Measuring

4.4.5.2 Heat transfer service unit, H112

A bench-top heat transfer service unit that is fully instrumented and offers regulated, controllable AC power for all optional modules in addition to a number of auxiliary power outlets. The equipment in the unit comprises digital current and voltage meters as well as a 12-point temperature indicator, which is common to all modules. To facilitate the reading of 12 thermocouple, point temperatures at the same time as a data logger is used, shown in Figure 4.10. This device is convenient to work with a number of thermocouples, so that gives a stable temperature reading.



Figure 4.10: Heat transfer service unit (H112).

4.4.5.3 Calibration of temperature sensors

The temperature sensors that are being used are fixed in different locations. The true value of the needed temperature is difficult to determine because these sites are exposed to solar radiation of various intensities. For the temperature sensors that are exposed to full radiation, partial radiation (glass and ground), and shade locations, a site calibration is therefore required (the measuring points along the centerline of the storage tank). Installing a regular thermocouple very close to the temperature sensor enables this on-site calibration to be done. Additionally, calibrations of measurements (PCM, Solar water, and helical coil pipes) not subject to radiation are carried out.

According to the calibration results, it is evident that temperature sensors that are completely exposed to solar radiation (i.e., those measuring air temperatures between the solar collector's floor and roof) record temperatures that are higher than the actual temperatures because they are receiving extra heat from the sun. All results of calibration are presented in Appendix B.

4.5 EXPERIMENTAL PROCEDURE

The following actions are taken into consideration when conducting an experiment.

- Add measured amounts of salt water to the tank.
- Turn on the primary power source so the electric water pump may function.
- Use the valve to regulate the system's water flow rate.
- After the system has run for about 10 minutes to reach a steady state, record the readings from each thermocouple, as well as the amount of distilled water used and the solar radiation strength.
- Repeat step 5 for the appropriate number of readings, repeating it every hour.

4.6 EXPERIMENTAL CALCULATION

The experimental results can be obtained from the flowing equations which are dependent on the measuring temperatures, flow meter and solar intensity.

4.6.1 Flat Plate Collector [47]

The data obtained during the experiments as values are recorded.

Heat Transfer by PCM

All range of storage energy through the PCM in the collector can explained by sensible and latent heat

$$Q = \underbrace{m_p c_{p,s} (T_m - T_i)}_{\text{Sensible heat stored in PCM}} + \underbrace{m_p \beta \Delta h_m}_{\text{Heat transfer during phase transitions}} + \underbrace{m_p c_{p,l} \Delta T}_{\text{Sensible heat stored in PCM}} \quad (4.1)$$

$C_{p,s}$ is PCM specific heat of solid, $C_{p,L}$ is PCM specific heat of liquid, T_m is melting temperature of PCM, Δh_m is Melting heat, β is Thermal expansion of PCM, its value is calculated as:

$$\beta = \begin{cases} 0 & \text{if } T < T_s \\ \frac{T - T_s}{T_L - T_s} & \text{if } T_s < T < T_L \\ 1 & \text{if } T > T_L \end{cases} \quad (4.2)$$

Where T is PCM temperature, T_s is Solidus temperature, T_L is Liquidus temperature, $\dot{m}$ is the mass flowrate of water inside the tubes in kg/s, and C_p is the heat capacity of water which equals to 4.2 kJ/kg K, T_{out} and T_{in} are outlet and inlet water temperatures in Celsius, respectively. And I is the solar radiation intensity in W/m^2 , A is total collector area which equals $1 m^2$.

4.6.2 Estimation of PCM Amount into the Storage Tank

The mass of one capsule with PCM = 0.25kg

The mass of PCM in the layer:

$$m_p = N_{caps} \cdot M_{PCM} \quad (4.3)$$

N_{caps} is the number of capsules in the layer.

If one layer of PCM is 20 capsules, then two layer is 40 capsules.

$$m_p = 40 \cdot 0.25 = 10 \text{ kg}$$

4.6.3 Calculation of Thermal Storage Energy for the Water Tank

To examine how the PCM affects the TES Q (kJ) in the current investigation. Equations are used to determine the thermal energy in the water tank both with and without a PCM:

Solar hot water heat loss equals heat gain from a helical coil heat exchanger plus heat storage from a tank PCM.

Heat gains by the heat exchanger are the sum of heat losses from PCM and the collector hot water.

$$Q_{in} = \int_{T_i}^{T_o} m_w c_{pw} dT = m_w c_{pw} \Delta T \quad (4.4)$$

$$m_w = \rho_w \cdot V_w \quad (4.5)$$

Where m_w is water mass, ρ_w is water density, ρ_w is 1000 kg/m^3 , V_w is the volume of water in storage tank per liter (1000lit= $1m^3$), C_{pw} is the water specific heat, C_{pw} = 4.180 KJ/kg °C, T_i is

initial temperature, T_o is outlet temperature, ΔT is a difference in water temperature is calculated for each time interval of 1 hour.

Heat giant by the helical coil heat exchanger:

$$Q_c = \dot{m} C_{p, c} (T_{c, out} - T_{c, in}) \quad (4.6)$$

$$Q_h = \dot{m} C_{p, h} (T_{h, out} - T_{h, in}) \quad (4.7)$$

Heat exchanger overall coefficient (U)[47]:

$$U = \frac{\bar{Q}}{A_s \Delta T_{ln}} \quad (4.8)$$

where: $\bar{Q}$ is mean value of heat between (Q_{cold} , Q_{hot}).

4.6.4 Efficiency Calculation [48]

The amount of heat transmitted to the water Q_u (W) divided by the total amount of solar energy that reached the solar collector during the same time period is known as the collector's efficiency (η_c) in this study. This is how it calculates:

The following equation yields the thermal efficiency for each solar collector:

$$\eta = \frac{P_{out}}{P_{in}} \quad (4.9)$$

Where P_{out} is the output power that the PCM and water carry from the collector, and P_{in} is the power absorbed by the collector from the sun.

$$P_{out} = \dot{m} \times C_p \times (T_{out} - T_{in}) \quad (4.10)$$

$$P_{in} = I \times A \quad (4.11)$$

Where I is solar intensity and A is the collector cross section area.

The system's energy and mass balances can be used to anticipate the efficiency of the PCM-integrated solar water heater. The useful energy that solar radiation can be used to convert into is:

$$\eta_c = \frac{Q_u}{I_i A_c} = \frac{\dot{v}_w \rho_w c_{pw} \Delta T_c}{I A_c} \quad (4.12)$$

Where $\dot{v}_w$ is volumetric flow rate of water, ΔT_c is the difference between the outlet T_{out} , inlet T_{in} temperatures of the solar collector.

When using PCM, the above useful general equation for heat gain can be further broken down into three formulae. PCM contains both sensible and latent heat because it is a PCM.

Heat gain when PCM is in a solid state:

$$\text{Heat Gain (PCM)} = \frac{m_{PCM} C_{p,PCM} (T_{s2} - T_{s1})}{60} \quad (4.13)$$

Where $C_{p,PCM}$ = Specific heat of PCM during solid phase (J/kg°C), T_s = Temperature at solid state, m_{PCM} = mass of PCM (kg).

Heat gain during phase change,

$$\text{Latent heat} = \frac{m_{pcm} C_{p,latent,pcm}}{60} \quad (4.14)$$

Where $C_{p,latent,PCM}$ is Specific latent heat PCM (J/kg).

$$\text{Heat gain when PCM is in liquid state} = \frac{m_{pcm} C_{p,l,pcm} (T_{L2} - T_{L1})}{60} \quad (4.15)$$

where $C_{p,L,PCM}$ is Specific latent heat PCM (J/kg), T_L is Temperature at liquid state.

The experiment's solar water heater operates both during the day and at night. Solar radiation provides heat during the day, whereas PCM provides warmth at night. The following equation is used to translate the prediction of the solar collector efficiency:

Efficiency during the day and night (without PCM), Collector Efficiency.

$$\eta_{col} = \frac{Q_u}{I_i \cdot A_c} = \frac{\dot{m}_{water} C_{p, water} \Delta T_{water}}{I_i \cdot A_c} \quad (4.16)$$

where η_{col} is instantaneous efficiency of collector (%), I_i is solar radiation (W/m^2), A_c is solar collector area (m^2) and ΔT_{water} is water temperature different ($^{\circ}C$).

The calculation of efficiency at night is shown below, with water receiving heat only via PCM heat discharge.

Day and Night-time efficiency with PCM,

$$\eta_{col} = \frac{Q_u}{I_i \cdot A_c + \frac{m_{pcm} C_{p, pcm} \Delta T_{pcm}}{60} + \frac{m_{pcm} C_{p, Latent pcm}}{60}} \quad (4.17)$$

Where ΔT_{PCM} is PCM temperature difference ($^{\circ}C$) and $C_{p_{PCM}}$ is PCM specific heat ($J/kg^{\circ}C$).

The results without PCM and the results with PCM both use equations (4.16&4.17) to estimate daytime efficiency.

5. RESULTS AND DISCUSSION

5.1 INTRODUCTION

Solar energy can only be used when it's sunny outside. Therefore, solar heating is only efficient during the day and decreases at night or on overcast days. Consumer energy needs have a distinct seasonal structure, and solar energy cannot completely meet those needs. In order to satisfy customer demand, energy storage is essential. In order to maximize the use of solar energy and to enhance the energy and efficiency of the solar absorption system, superior thermal properties of sophisticated materials, such as PCMs, are performed [47].

In the current study, PCM is integrated with SWH and fed into a storage tank and flat plate collector. Helical coil heat exchangers were added to the storage tank as an external load. The trials were conducted in four separate months (September (2021), April, May, and June 2022) that were chosen on the first day of BASRA climate (Latitude 30°.54, longitude 47°.64). The effectiveness, heat gain, and significance of the PCM in increasing heating efficiency throughout the day were studied using a range of variables, including water volume flow rate (2, 3, 4, 6, and 8 L/min) and inlet water temperature (25, 30, and 35 °C). This chapter includes experimental results and numerical results with above variables. The results showed that, given an initial temperature of 25 °C, the daily efficiency range was 0.58 to 0.65 and that the daily final outlet temperature was over 65 °C. Additionally, on all test days, the heat released by the PCM was audible in the evening.

5.2 EXPERIMENTAL RESULTS

In order to get the overall accuracy, experimental results show the average value of each test at the start for all 4 months. The FPSC's performance at a 30-degree tilt. With different inlet HTF temperatures of 25, 30, and 35 °C, and 80 liters of water inside the tank, the performance of the storage tank unit for hot water storage is examined. Every case is handled both with and without the PCM. Controlling the melting time is another way to confirm PCM's performance impact. Ten hours are dedicated to recording the readings throughout the day. The thermal energy is initially held as sensible heat inside the capsules until the PCM gets close to melting. The PCM

subsequently undergoes superheating, at which point the energy is transformed to latent heat and stored in the liquid PCM. The PCM and HTF temperatures are recorded every hourly interval. As a heat exchanger, all stored heat was lost to the external load.

5.2.1 Weather Conditions of BASRA Climate

Figures 5.1&5.2 present the solar intensity and daily temperature value per daytime of BASRA CITY weather conditions for four months from morning 9.00 AM to evening 19.00 PM on the first day of each month respectively. Figure 5.1 shows the same behavior of the four months at the time period from 12 o'clock to 2 PM in the afternoon The temperature data was recorded in conjunction with the solar radiation readings for each 1st day of testing with the constant volume of water into the tank with and without PCM, and with the stability of the flow rate. Figure 5.2 can note the relative convergence of the ambient temperature curves for all months. Maximum solar intensity reached 1200 W/m² in June 2022 compared to other months. Also, weather temperature reached 46 °C in the same month and 12.30 PM.

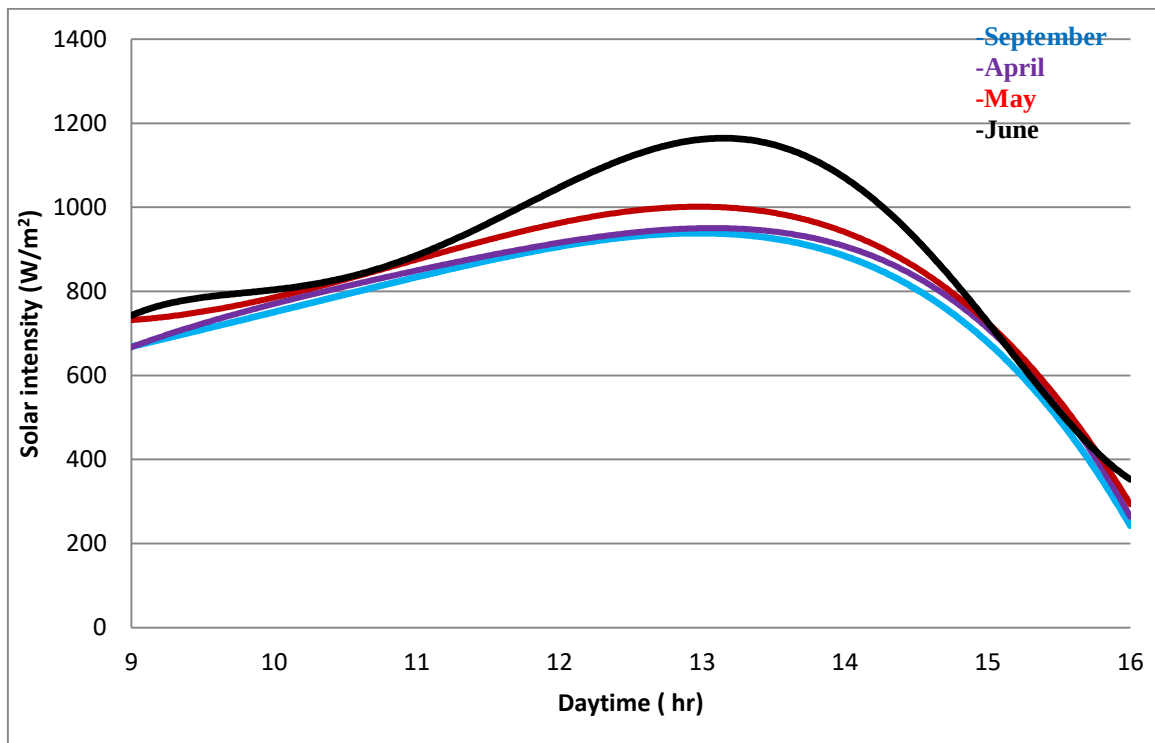


Figure 5.1: Solar intensity on the FPSC with different months and daytime.

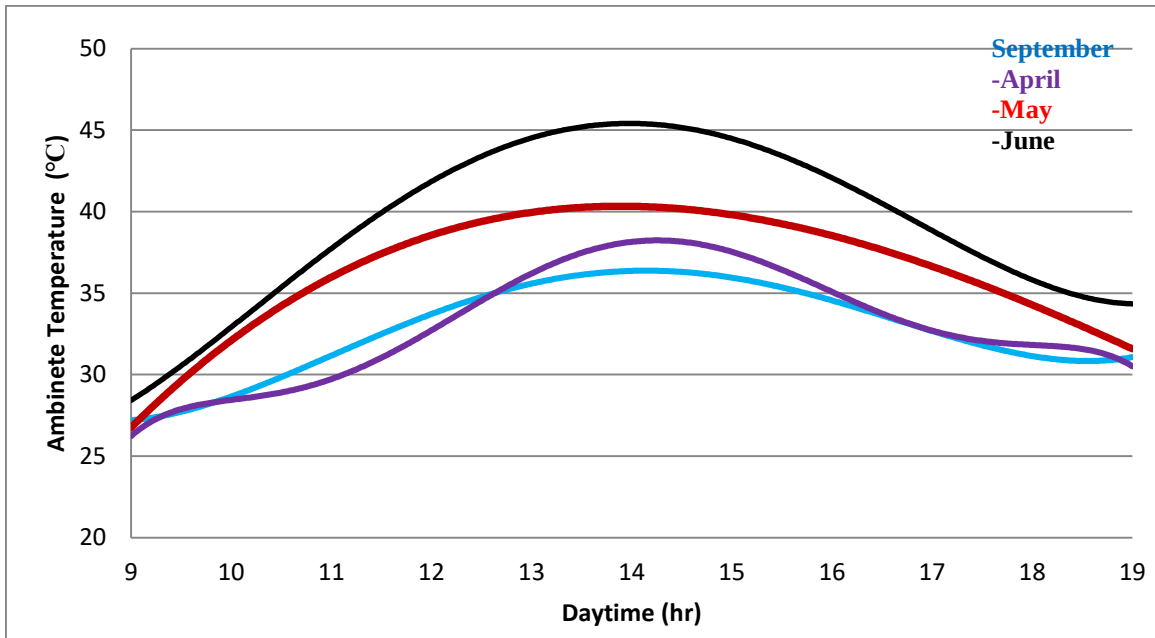


Figure 5.2: Weather temperature with different months and daytime

5.2.2 Collector Temperatures Distribution

Temperature distribution along the solar collector was recorded experimentally by eight K-type thermocouples as explained in chapter four.

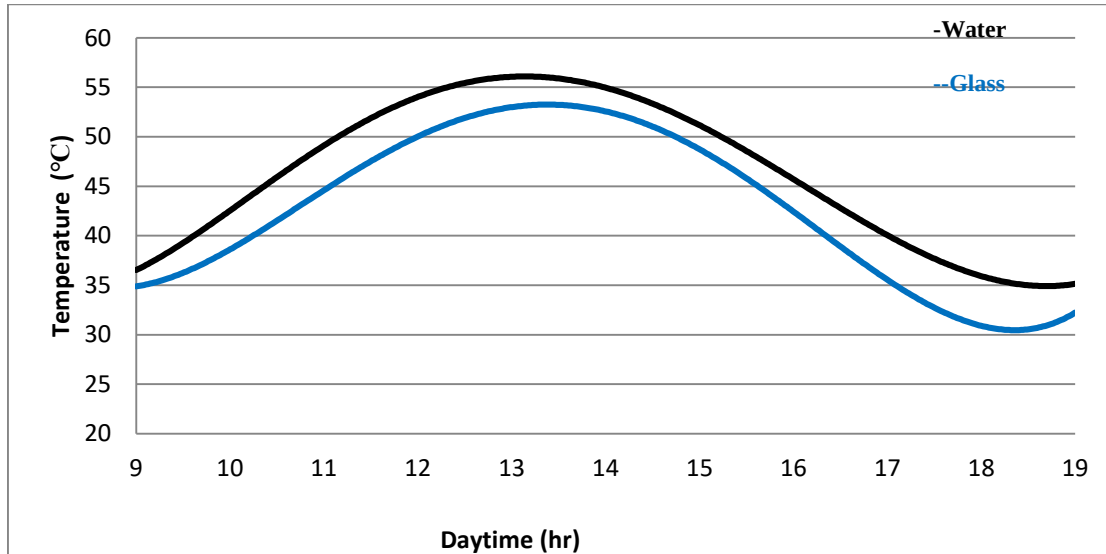


Figure 5.3: The average temperature through FPC glass and water at daytime ,1st May 2022, $T_{in}=25$ °C, $V_{in}= 4$ L/min in to coil pipe

Figure 5.3 presents the experimental of the average temperature through FPC glass and water during daytime ,1st May 2022, $T_{in} = 25\text{ }^{\circ}\text{C}$ and $V_{in} = 4\text{ L/min}$. Due to the water heat capacity, the average temperature increased with the daytime hour and reached $56\text{ }^{\circ}\text{C}$ at 13.00 PM. The average temperature decreases with the decrease in the intensity of solar radiation until six o'clock in the afternoon and then stabilizes with the heat gained from the wax paraffine.

Figures 5.4, 5.5, 5.6 and 5.7 present the FPC with PCM performance at four months and daytime (along ten hours each 1st day) at fixed collector flow rate water 4L/min and storage tank water flow rate 4L/min.

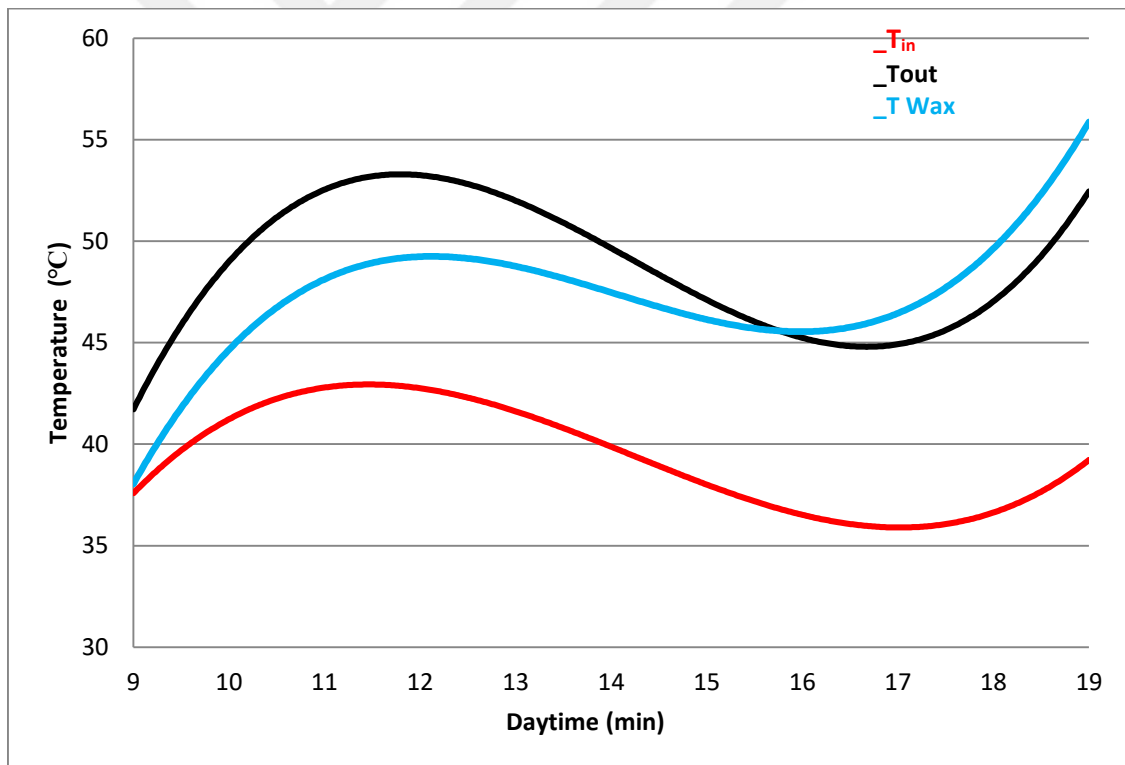


Figure 5.4: Collector performance at daytime with coil pipe $V_{in} = 4\text{ L/min}$, collector $V_{in} = 4\text{ L/min}$ (1st Sep 2021).

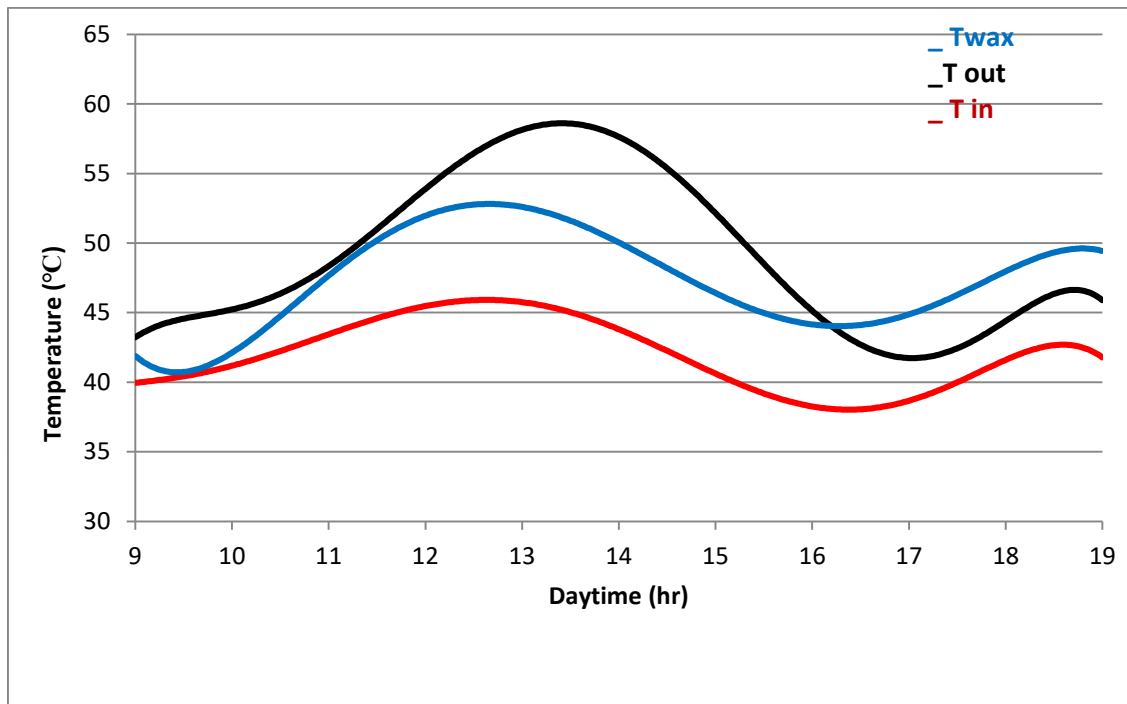


Figure 5.5: Collector performance at daytime with coil pipe $V_{in} = 4$ L/min and collector $V_{in} = 4$ L/min (1st April 2022).

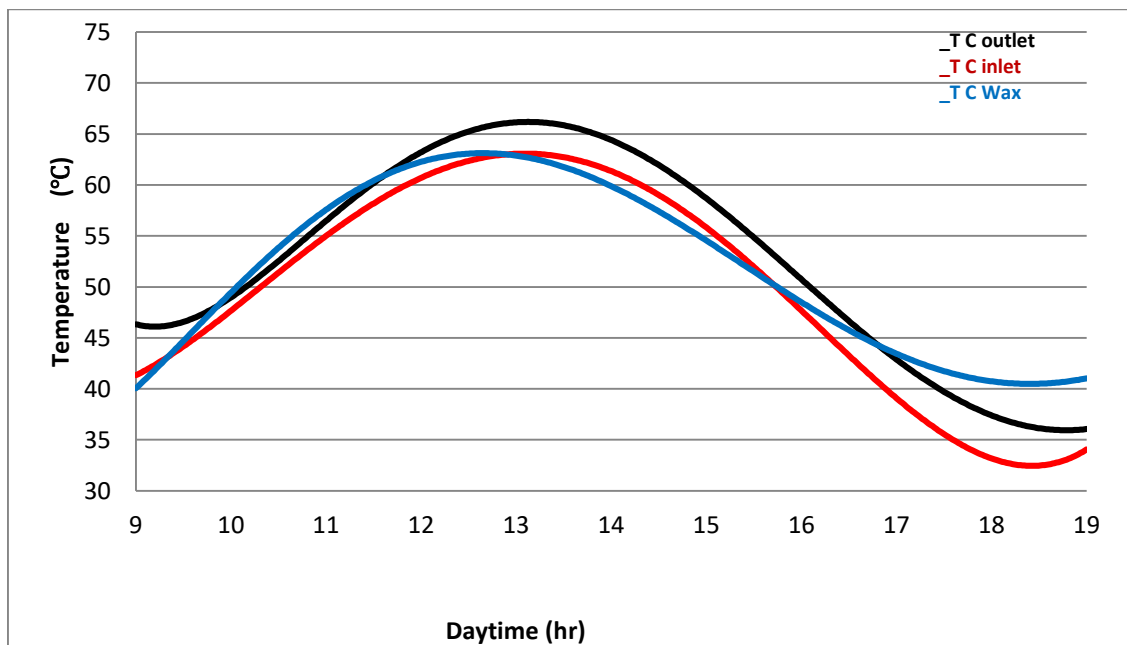


Figure 5.6: Collector performance at daytime with coil pipe $V_{in} = 4$ L/min and collector $V_{in} = 4$ L/min (1st May 2022).

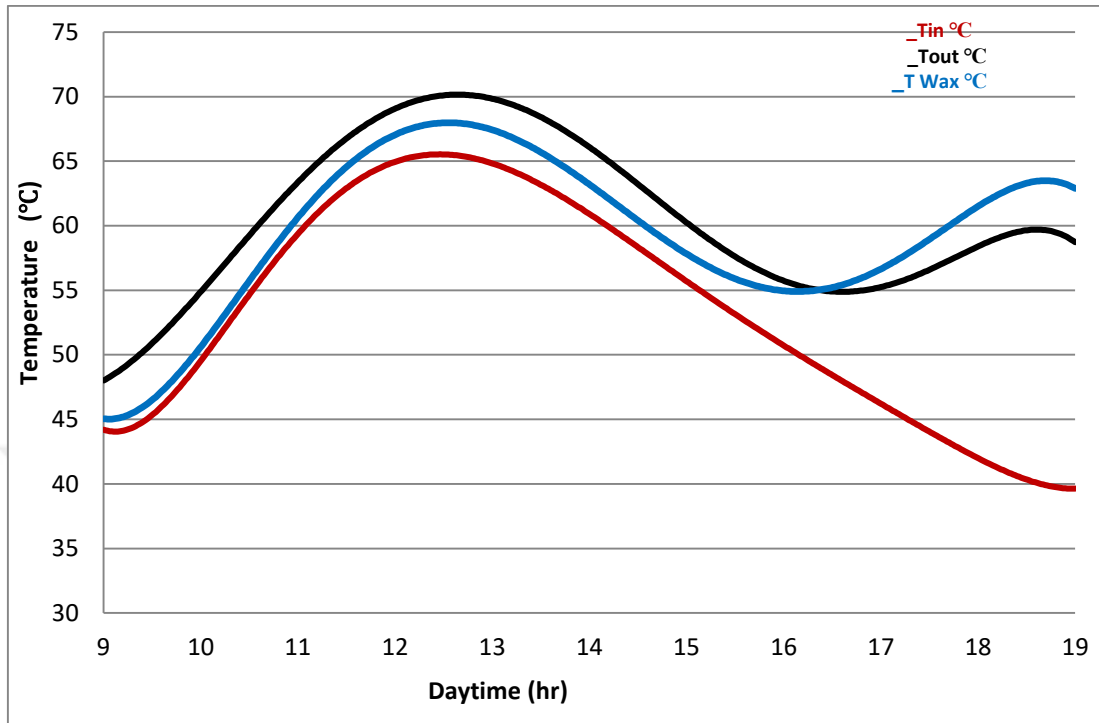


Figure 5.7: Collector performance at daytime with coil pipe $V_{in} = 4$ L/min and collector $V_{in} = 4$ L/min (1st June 2022).

All results indicate the inlet and outlet water temperatures through the collector and the mean temperature of PW from 9.00 Am to 19.00 Pm. All results indicate an increase in temperature with the increase in the intensity of solar radiation for all months, and it gradually decreases in the afternoon. At sunset, the effect of phase-changing materials begins to lose heat to the incoming water, which leads to an increase in its temperature with the demise of the sun. All results show that the temperature value of PCM is greater than the exit temperature due to better release of energy from the PCM at sunset time,

To demonstrate how the incoming water flow rate affects the efficiency of the collector, 2L/min was selected on 1st May 2022. Figure 5.8 was noticed the maximum outlet temperature of collector water reached 67 °C at midday of 1st May 2022 due to reducing in the income flow rate of the heat exchanger to 2L/min. This is due to Newton's convective law.

Figure 5.9 presents the comparison of collector performance with and without collector Wax on 1st May 2022.

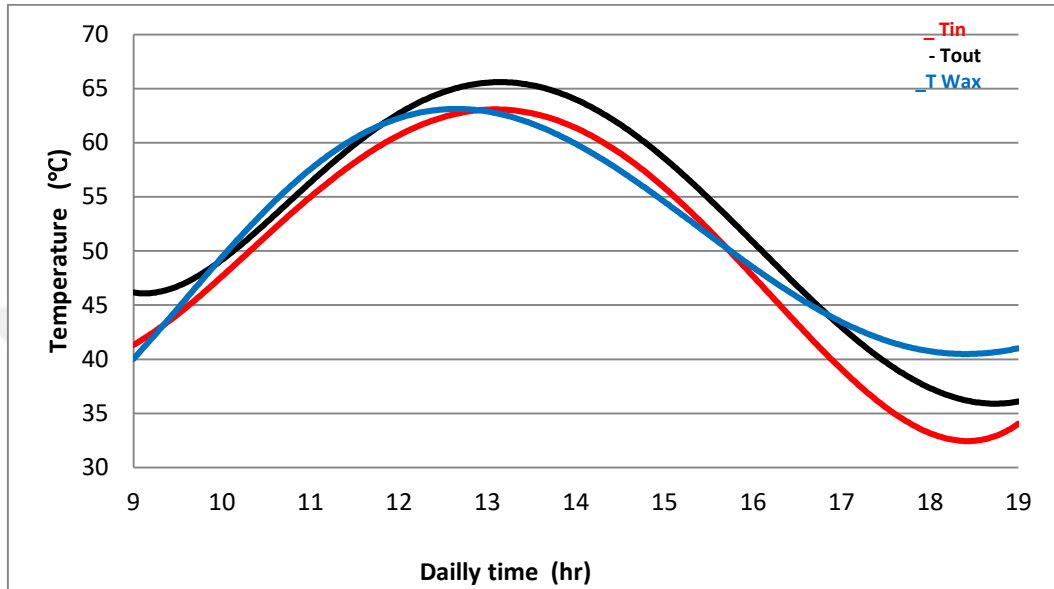


Figure 5.8: Collector performance at daytime with tank flow rate 2L/min, collector flow rate 4 L/min. (1st May 2022).

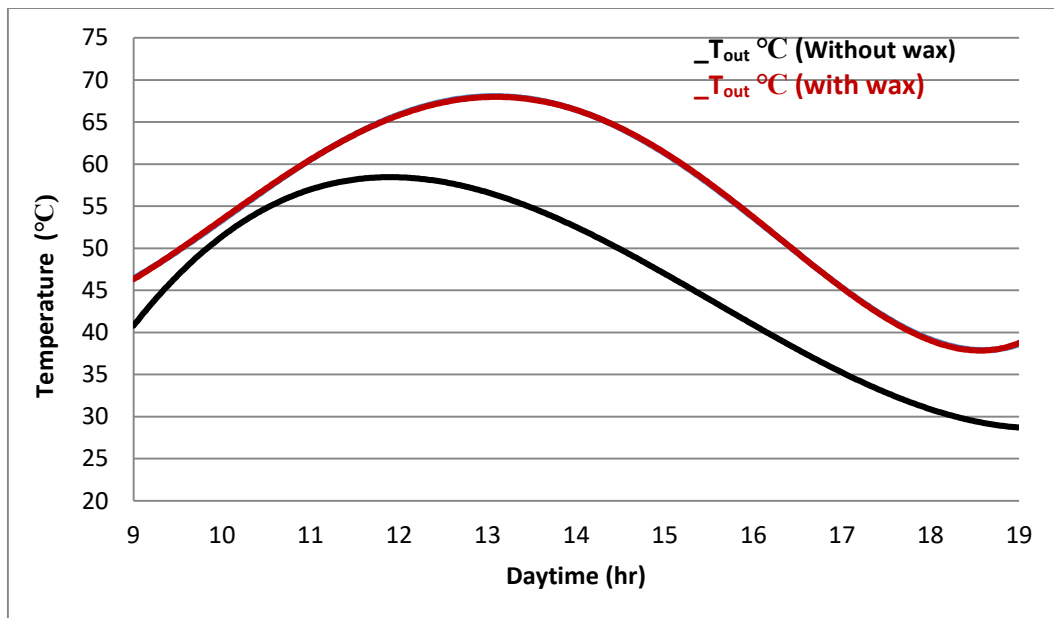


Figure 5.9: The effect of PCM on the outlet collector temperature at daytime, $V_{in} = 4\text{L/min}$, 1st May 2022.

The effect of heat-retaining materials is very little in the daytime compared to sunset due to the PCM behavior during the phase change period. At the same conditions water outlet of the collector increases about 10°C.

5.2.3. The Effect of Inlet Temperature on the Performance of the Collector

To explain the effect of inlet temperature of the heat exchanger, three values are selected 25,30 and 35 °C respectively.

Figure 10 illustrates the results from an experiment on the collector outlet temperature with different helical heat exchanger inlet water temperatures (25,30 and 35°C) during all daytime of 1st May 2022, and the volumetric flow rate is 4L/min respectively.

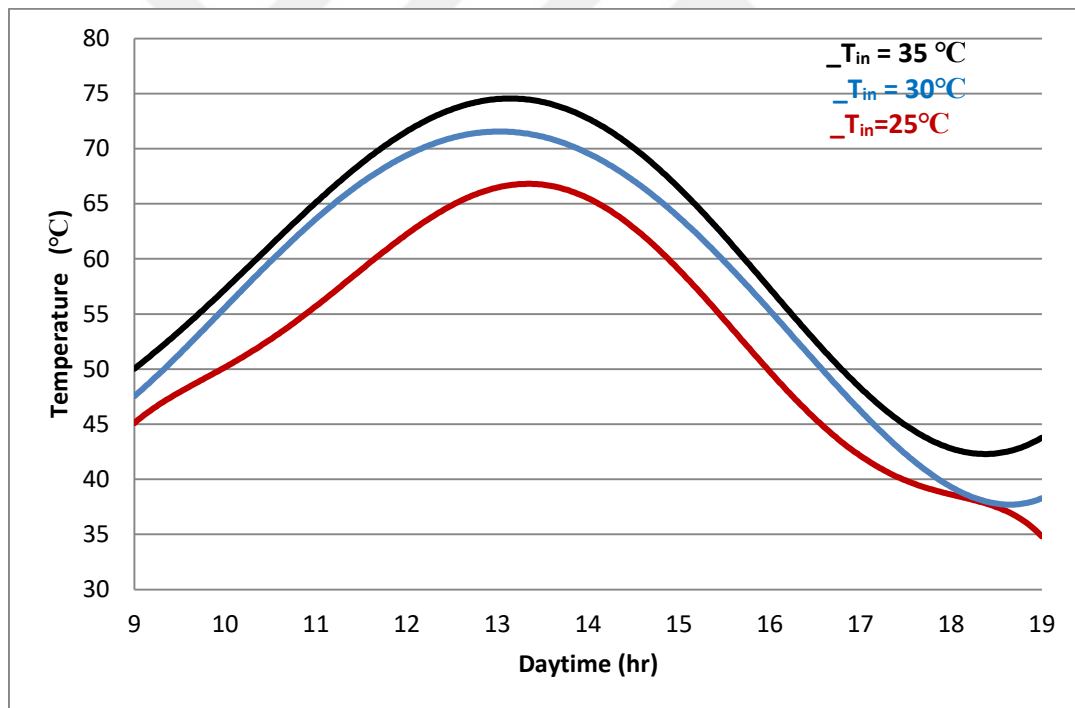


Figure 5.10: The effect of inlet temperature on the collector performance, coil pipe $V_{in} = 4$ L/min, May 2022.

The effect of entry temperature in a helical coil heat exchanger almost has little impact on the solar collector's performance, except in the peak hours of the day, while the greatest effect is PCM release energy at sunset of the day due to the stored energy.

Figure 5.11 shows the effect of inlet HTF & water inlet temperature on the collector wax melting time, in May 2022.

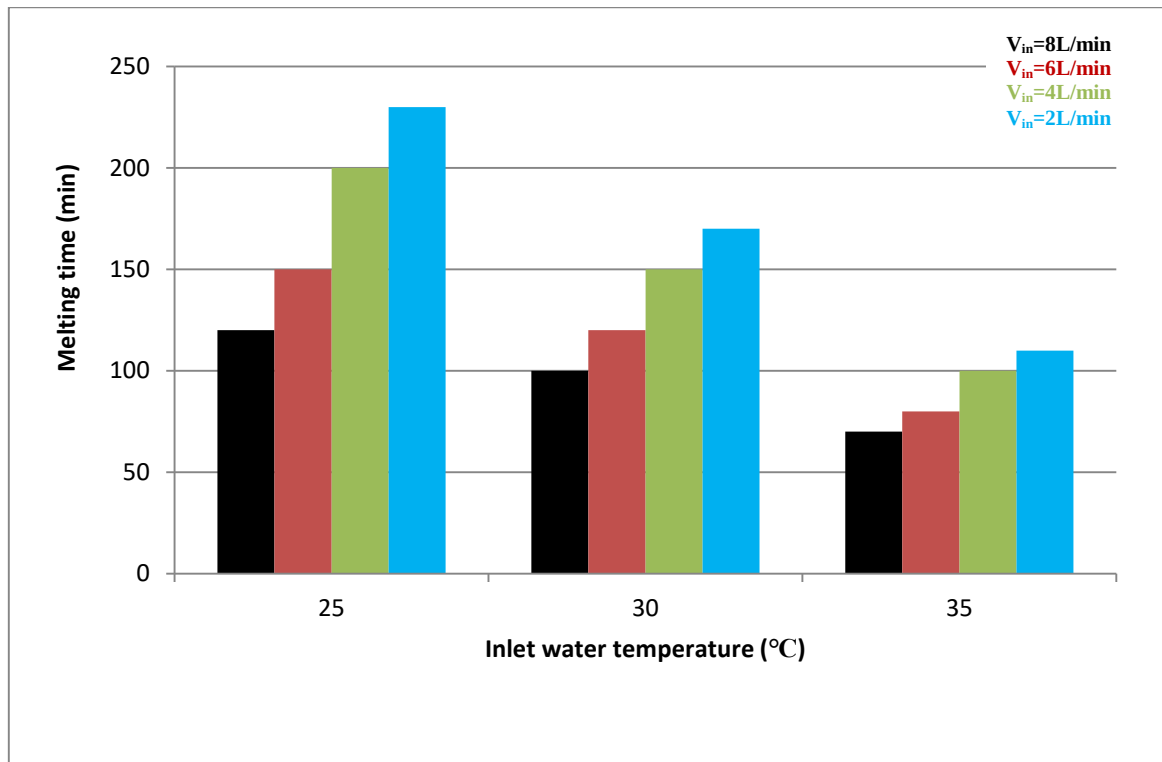


Figure 5.11: The effect of inlet HTF & water inlet temperature on the Collector Wax melting time, May 2022.

Collector wax is exposed directly to solar radiation and heat losses to water. Melting time is a good indication of the two-phase flow region and then latent heat as heat giant to water. The results have shown low inlet HTF and inlet temperature give high Wax melting time compare to others. At low flow rate $V_{in}=2\text{ L/min}$ melting time is reached to 225 min at 25 °C.

5.2.4 Collector Efficiency

This section presents the analysis of collector in the month of May 2022 is selected only. The heating performance of the solar collector, represented by the efficiency and the effect of radiation intensity, water inlet temperature and flow rate of the water entering the exchanger, was studied.

Figure 5.12 displays the impact of inlet water temperature on the collector storage energy at a fixed heat exchanger inlet volume flow rate(4L/min) during the daytime on 1st May 2022.

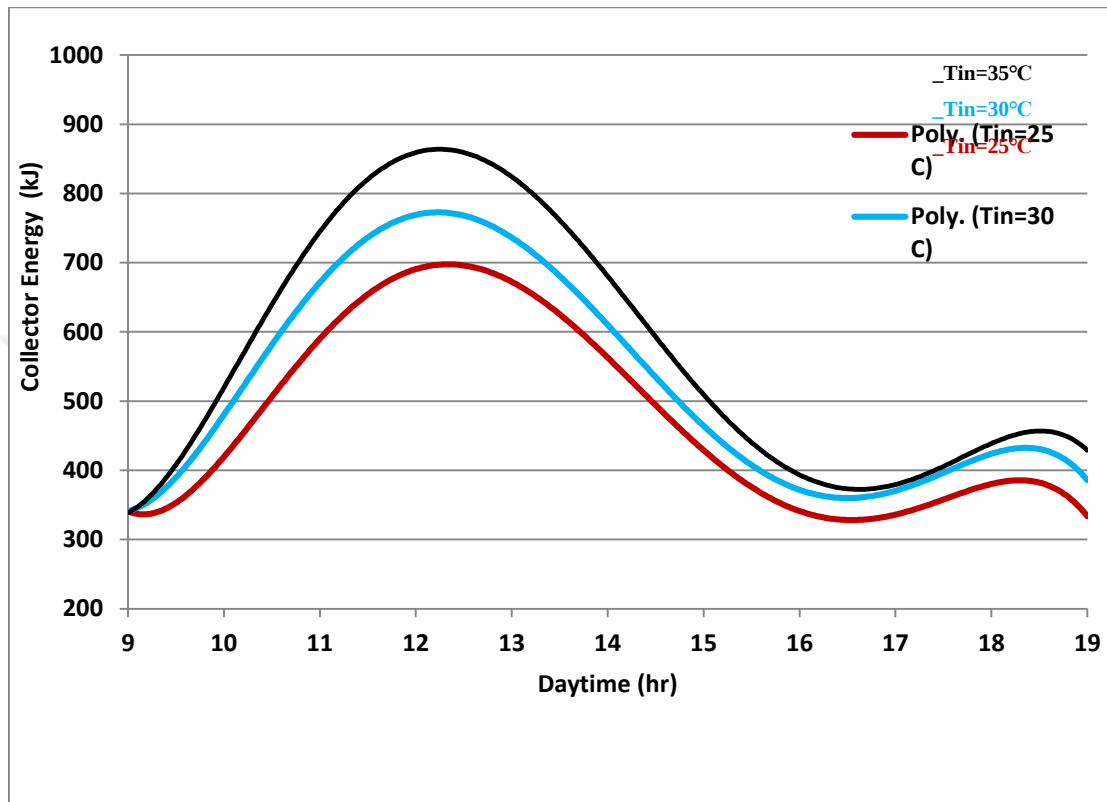


Figure 5.12: Collector storage energy at daytime and inlet water temperature, $V_{in} = 4\text{L/min}$, 1st May 2022.

Heat storage the heat is stored inside the paraffine wax during the day to lose it in the evening. High energy was registered at high inlet temperature during the daytime. The results recorded a 180 kJ increase in storage energy at an entry temperature of 35 °C compared to an entry temperature of 25 °C. The effect of the PW on the storage energy appeared after time 5.00 Pm, by increasing the energy stored around 100 kJ at 7.00 Pm.

The relation between the solar intensity to the collector efficiency was shown in Figures 5.13 & 5.14 with different inlet water flow rates during the daytime of 1st May 2022 respectively.

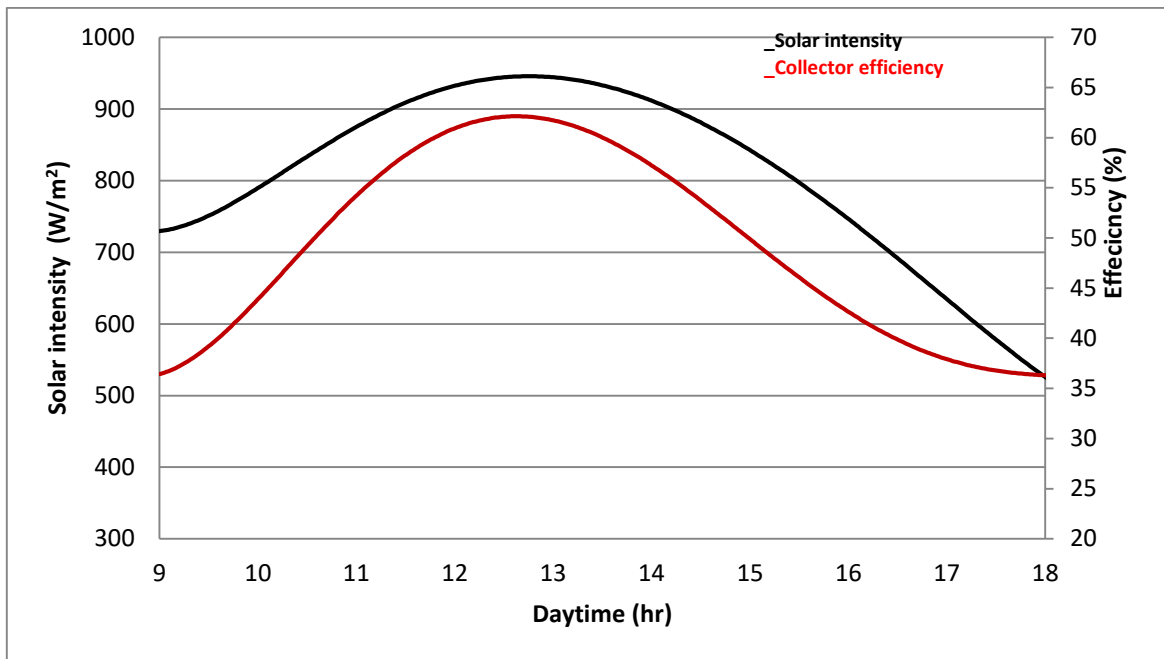


Figure 5.13: Effect of solar intensity on the collector efficiency with daytime (May 2022) $T_{in} = 25\text{ }^{\circ}\text{C}$, $V_{in} = 2\text{ L/min}$.

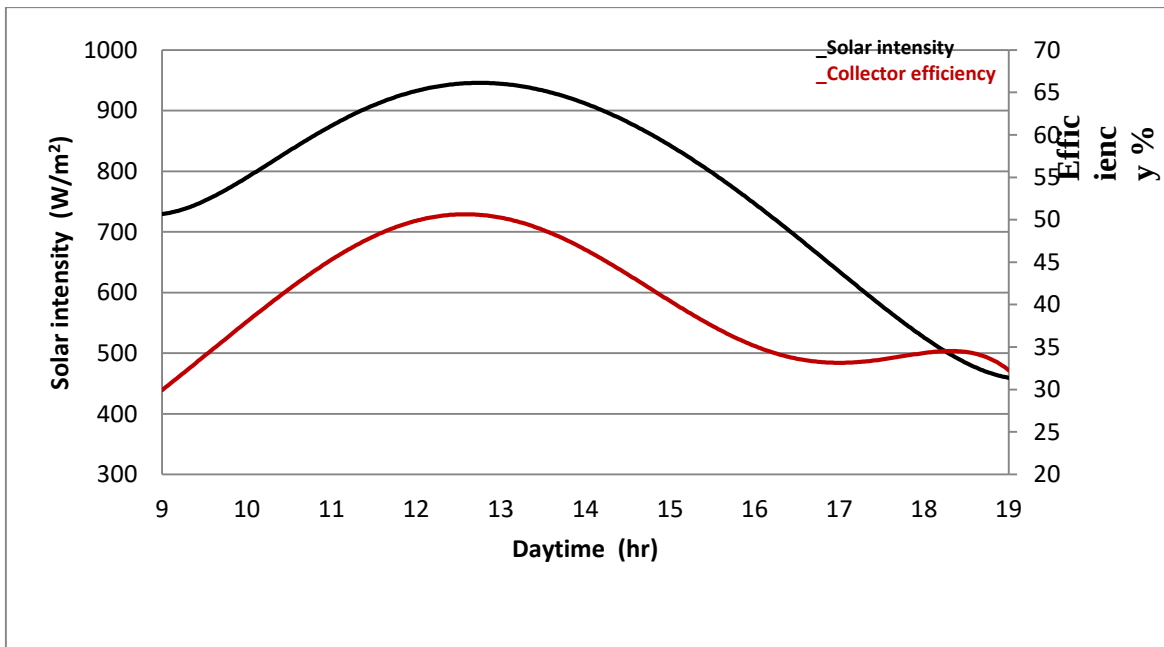


Figure 5.14: The effect of solar intensity on the collector efficiency with daytime (September 2021) $T_{in} = 25\text{ }^{\circ}\text{C}$, $V_{in} = 4\text{ L/min}$.

The results show high solar intensity gives the highest efficiency to the FPC. In the midday solar radiation reaches 980 watts, giving an efficiency of about 62% at $V_{in} = 2\text{L/min}$ compared to 53% at $V_{in} = 4\text{L/min}$.

The effect of solar intensity value on the collector efficiency has been shown in Figures 5.15&5.16 respectively.

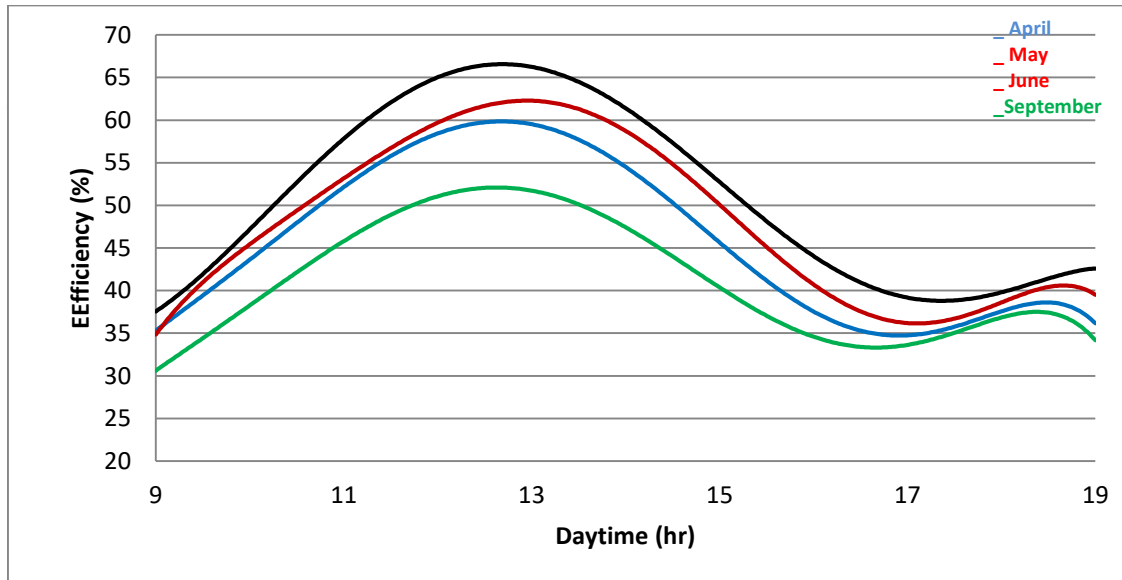


Figure 5.15: Collector efficiency at Month and daytime at constant $T_{in} = 25^\circ\text{C}$, tank $V_{in} = 2\text{L/min}$

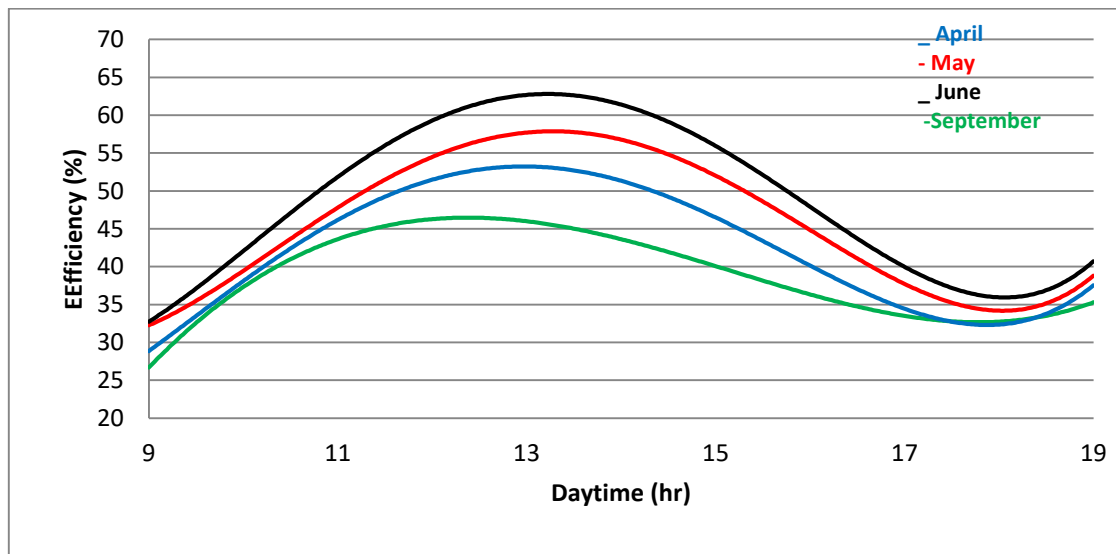


Figure 5.16: Collector efficiency at Month and daytime at constant. $T_{in} = 25^\circ\text{C}$. tank $V_{in} = 4\text{ L/min}$.

The comparison between the collector efficiency at Month and daytime $T_{in} = 25^{\circ}\text{C}$, $V_{in} = 2 \text{ L/min}$ and $V_{in} = 4 \text{ L/min}$ has been examined. High solar efficiency shows in June month compared to the other months. Collector efficiency value reached 67% at midday of June 2022 compared to 53 % on September 2021. High solar intensity. The results ensure that its storage heat is higher in June month, and that appears at the evening the efficiency of the FPC is higher than others by 12% at $V_{in} = 2 \text{ L/min}$.

The effect of the heat exchanger water flow rate on the collector efficiency on 1st May 2022 is shown in Figure 5.17 at constant inlet water temperature (25°C).

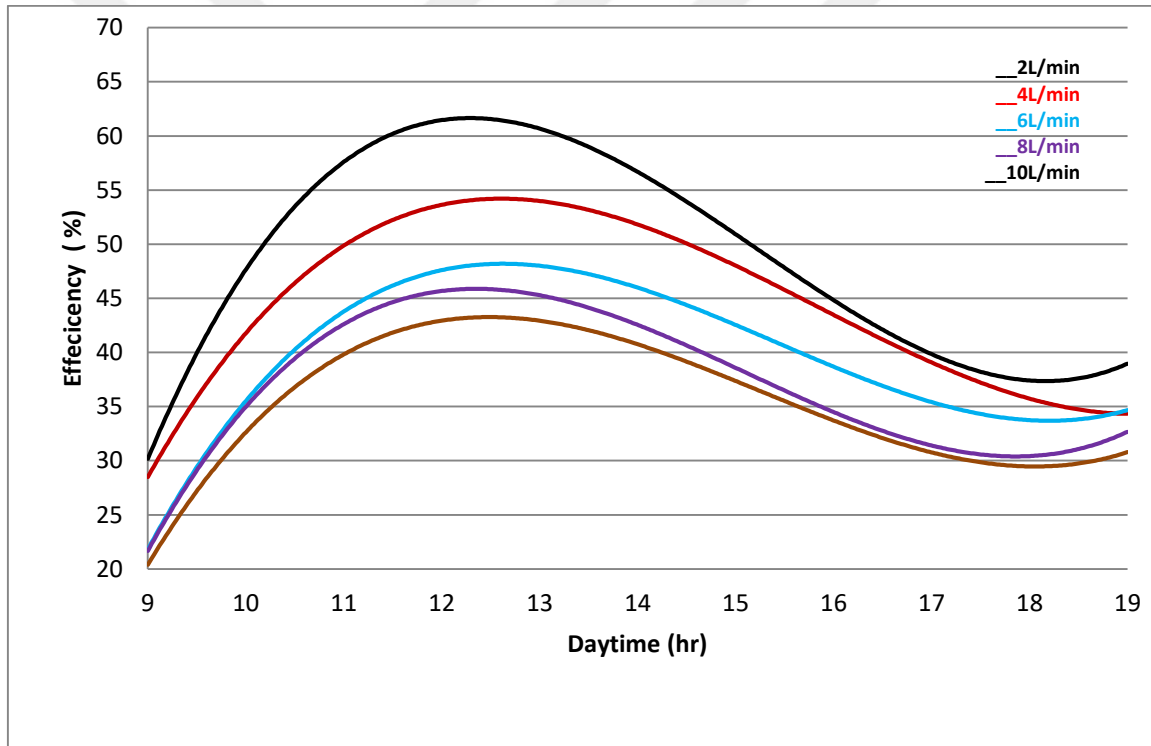


Figure 5.17: The effect of heat exchanger flow rate on the collector efficiency on 1st May 2022.

The high value of collector efficiency is recorded at low water inlet flow due to high outlet collector temperature. At $V_{in} = 2 \text{ L/min}$ the results show an average collector efficiency was increased of 20% at $V_{in} = 2 \text{ L/min}$ than the other flow rate at the mid daytime due to high inlet solar energy to the collector.

The effect of inlet water flow rate and inlet temperature on 1st May 2022 is presented in Figure 5.18 with different water inlet temperatures 25,30 &35°C respectively.

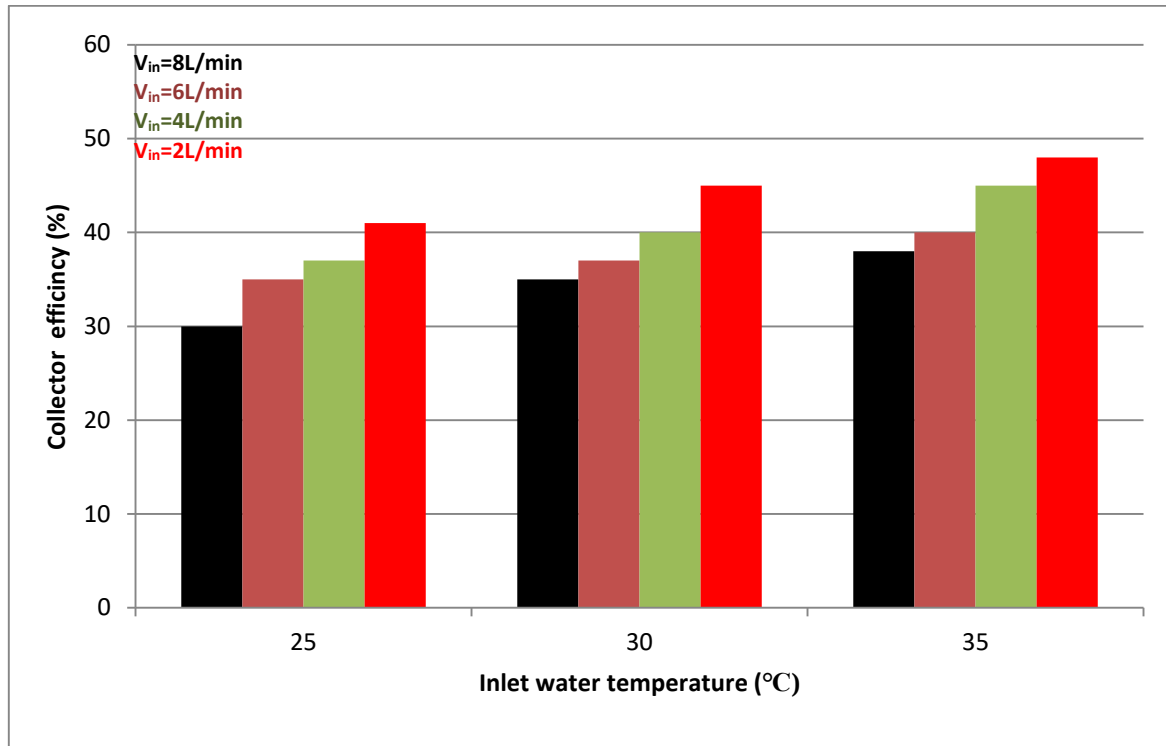


Figure 5.18: Collector efficiency with different water inlet temperatures and inlet HTF on 1st May 2022.

The results show the average collector efficiency is enhanced by about 10% with high inlet temperature and low inlet water flow rate ($V_{in} = 2 \text{ L/min}$). These results became clear as the effect of the storage materials and low water flow rate on the collector outlet temperature.

5.2.5 Storage Tank Performance

Storage tank performance is improved with capsules of phase changer material and integrated with the external load as a helical coil heat exchanger. So, the results indicate the effect of solar intensity, water flow rate, and inlet temperature on both PCM and heat exchanger storage enhancement.

Figures 5.19 & 5.20 show the experimental storage tank performance as, inlet water temperature (from the collector), Wax temperature and outlet temperature (to collector). All temperatures are increased with increasing solar intensity and decreased with decreased solar intensity. The effect of storage material PCM appears in the evening time.

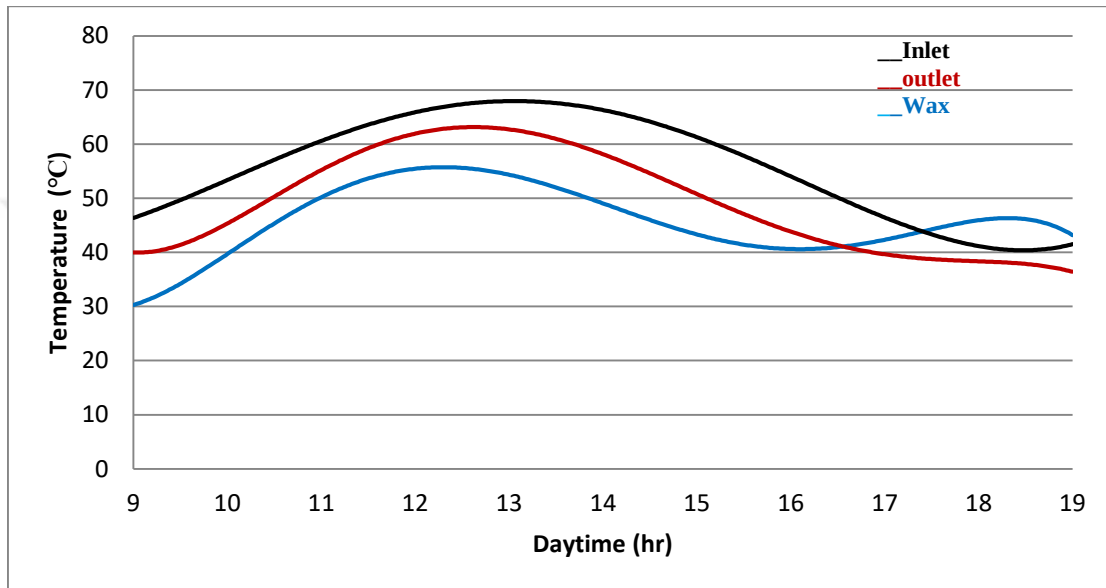


Figure 5.19: Storage tank performance with daytime at inlet volume flow 2 L/min, 1st May 2022.

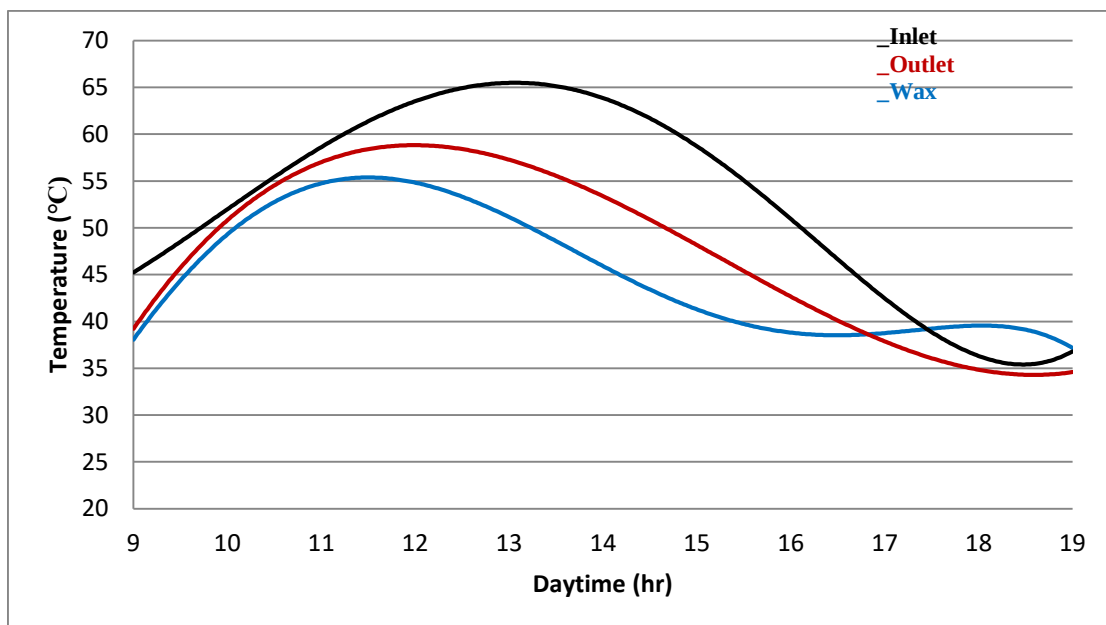


Figure 5.20: Storage tank performance with daytime at inlet volume flow 4 L/min on 1st May 2022. Thermal behavior is represented by, a temperature distribution diagram, overall heat transfer coefficient and effectiveness through Figures 5.21 to 5.25.

Figures 5.21 & 5.22 illustrate the average temperatures through the inlet, the outlet of the coil tube and the shell (storage tank) with the counter flow.

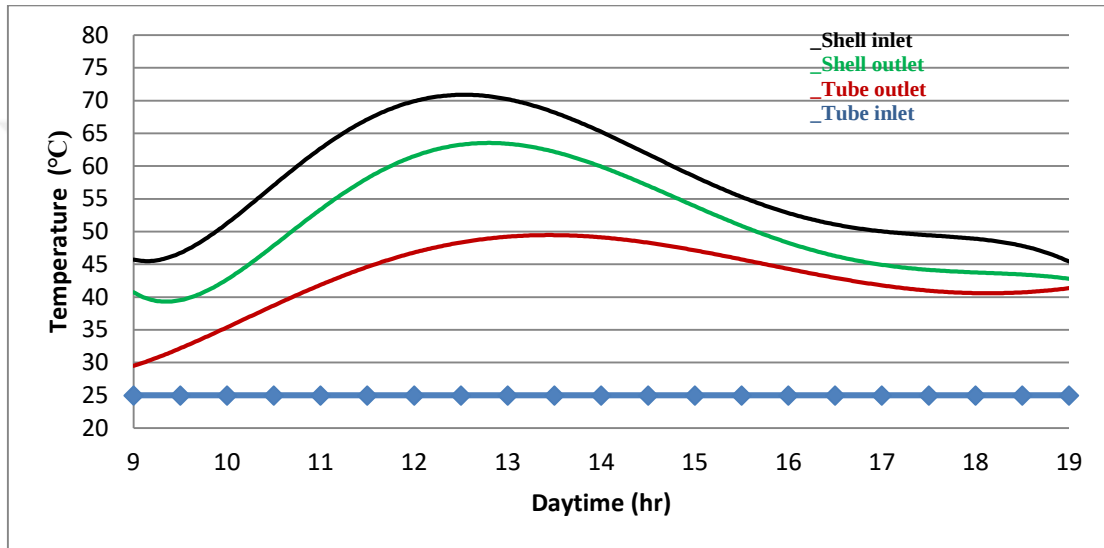


Figure 5.21: Inlet & outlet heat exchanger temperature, ($V_{in} = 2$ L/min), $T_{in} = 25$ °C, (1st May 2022)

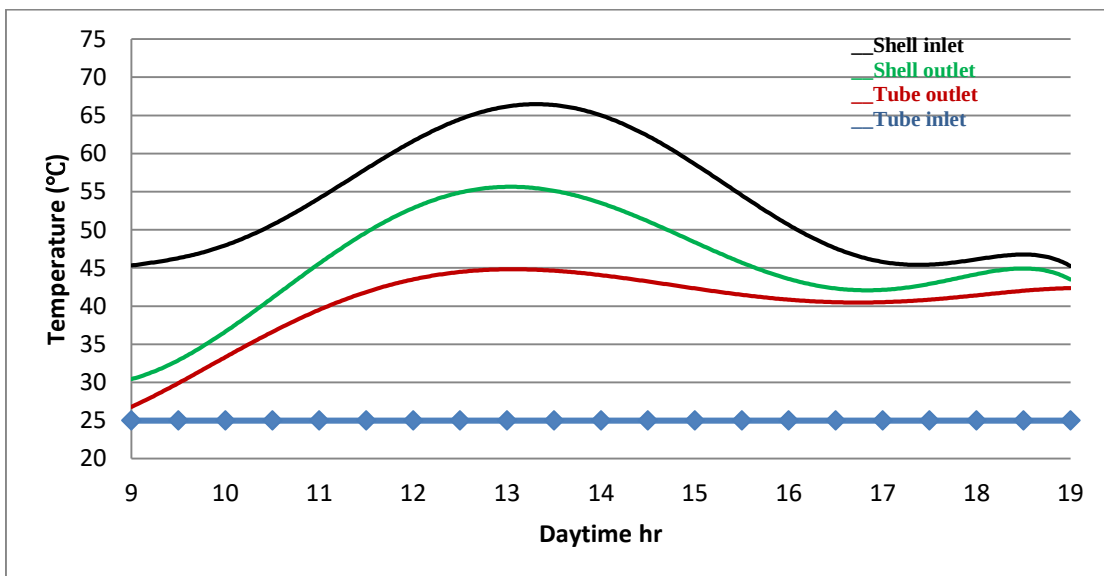


Figure 5.22: Inlet & outlet heat exchanger temperature, ($V_{in} = 4$ L/min), $T_{in} = 25$ °C (1st May 2022)

Inlet water temperature is fixed at 25 °C at all daytime through the tube. Shell inlet water temperature presents the outlet hot water collector while shell outlet the presents inlet collector water temperature. The effect of solar intensity effectively appears at midday. Inlet water temperature at shell reaches 72°C at 12.30 PM.

The heating behavior of the heat exchanger is presented by the overall heat transfer coefficient. Figure 5.23 shows the heat exchanger performance at different months in the daytime.

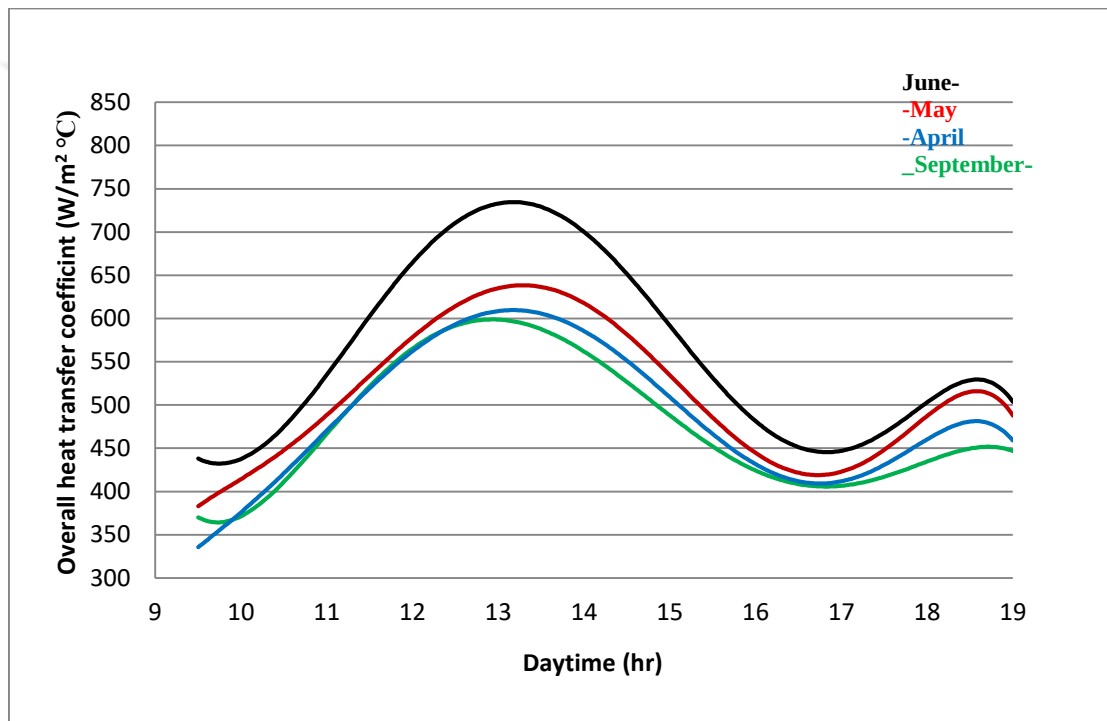


Figure 5.23: Heat exchanger overall heat transfer coefficient with different months and daytime, ($V_{in} = 2 \text{ L/min}$), ($T_{in} = 25^\circ\text{C}$)

The high intensity of solar radiation increases the rate of heat transfer to the water inside the tank. The high intensity of solar radiation increases the rate of heat transfer to the water inside the tank, as indicated by the results in June, higher than the rest of the months. The increase of heat transfer reaches 200 W/m²°C at midday of June 2022. Also, the effect of PCM increases the (U) by about 100 W/m²°C in the afternoon.

The effectiveness of the exchanger can be represented by the following Figures.

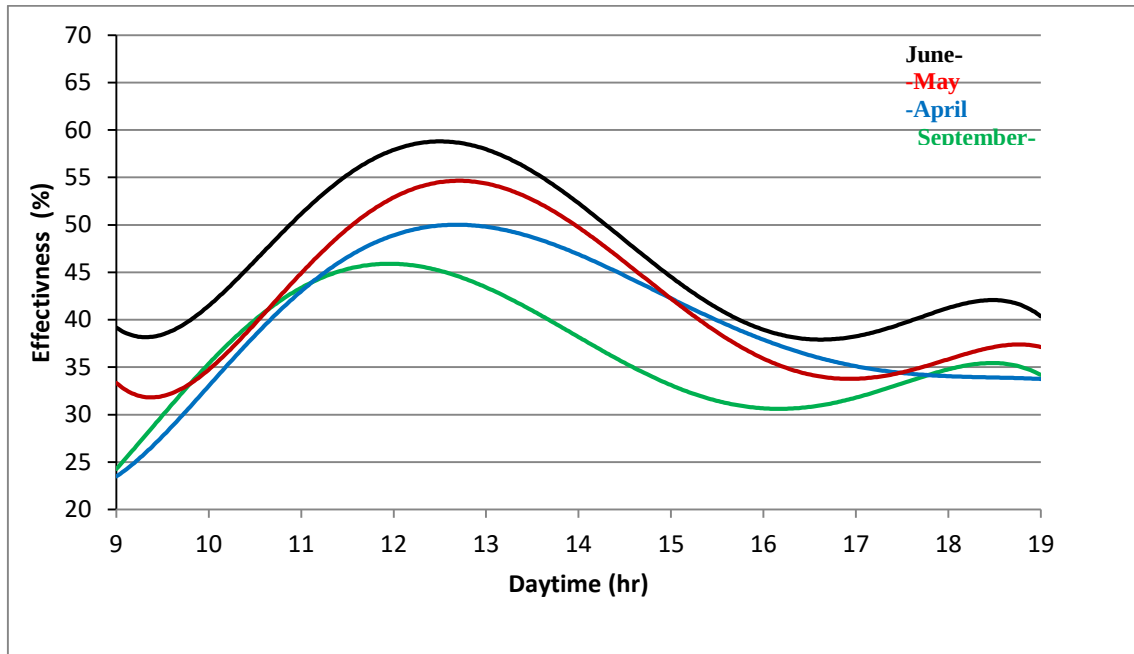


Figure 5.24: Heat exchanger effectiveness with different month and daytime ($V_{in} = 2 \text{ L/min}$, $T_{in} = 25^\circ\text{C}$).

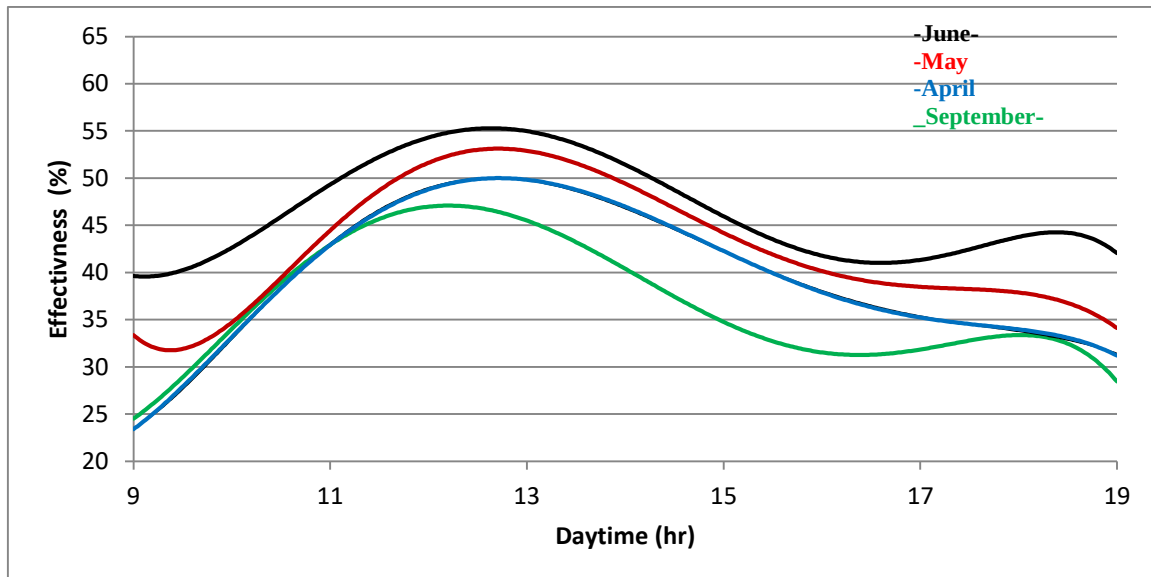


Figure 5.25: Heat exchanger effectiveness with different month and daytime ($V_{in} = 4 \text{ L/min}$, $T_{in} = 25^\circ\text{C}$).

Figures 4.24&5.25 show the heat exchanger effectiveness in different months. Also, the comparison of the effectiveness is presented with different water flow rates, 2 L/min & 4 L/min respectively. High effectiveness is presented at low water flow rate, 2 L/min compared with 4 L/min at same inlet water temperature due to increase of the collector outlet temperature. Maximum effectiveness value reaches 60% at mid-daytime of June 2022 and $V_{in} = 2$ L/min, while high effect of PCM storage energy is observed at flow rate 4 L/min at the surest time.

Overall energy storage through the tanks has been presented in Figures 5.26&5.27 with different inlet water flow rates and temperatures.

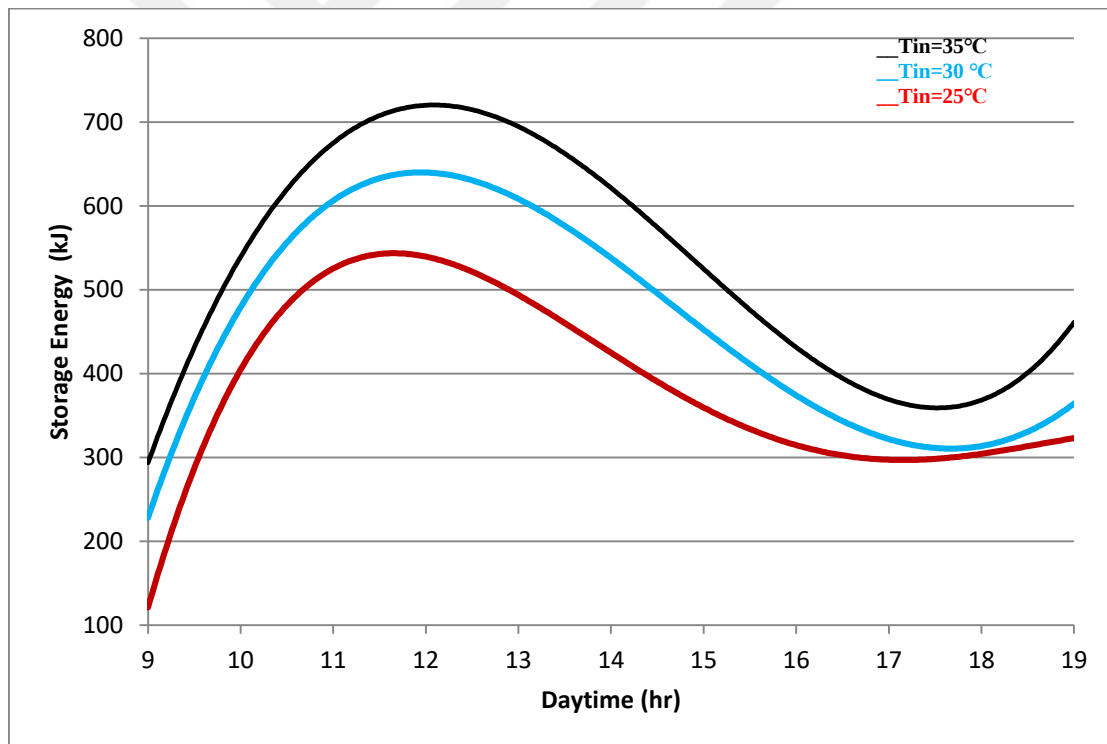


Figure 5.26: Thermal storage for the storage tank with different inlet water temperature, daytime ten hours) at inlet volume flow 2 L/min, May2022.

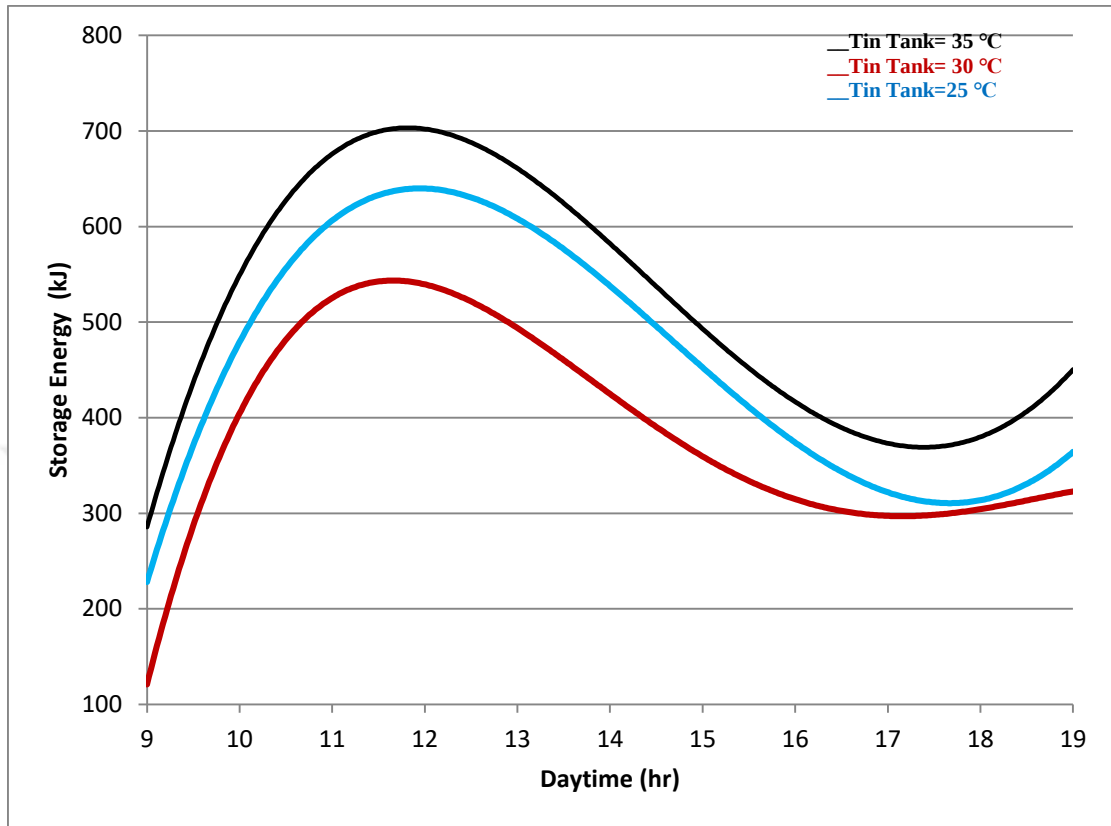


Figure 5.27: Storage tank performance at inlet water temperature and daytime (ten hours) at inlet volume flow 4 L/min, (May2022).

High inlet water temperature ($T_{in} = 35^{\circ}\text{C}$) increases the storage energy to maximum value in June, reached to 750 kJ compared to 700 kJ at flow rate of 4 L/min at the same inlet water temperature. Improving the percentage of stored heat due to a decrease in the flow rate inside the tank. There is no noticeable effect at the rate of flow 4 L/min, especially in the evening compared to the flow rate of 2 L/min.

Figure 5.28 shows the effect of inlet water flow rate and temperature on the overall storage tank efficiency.

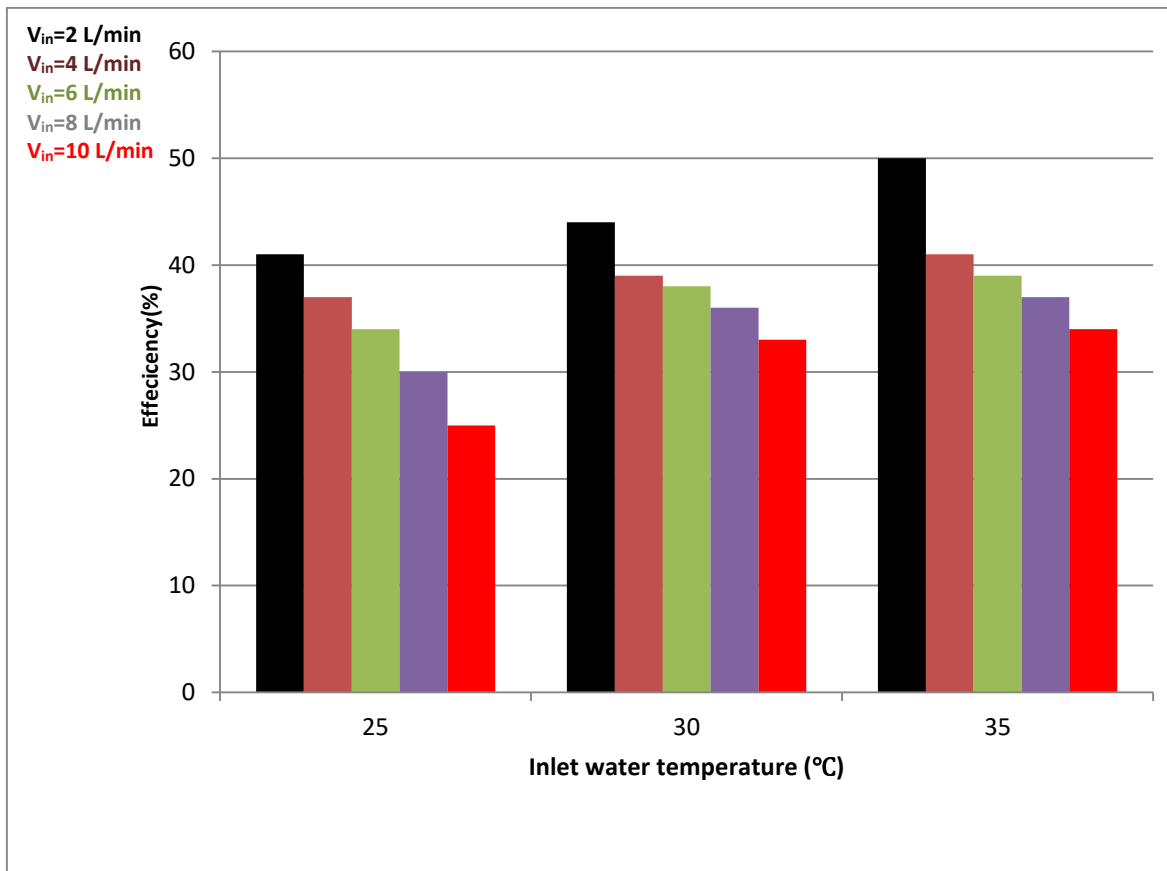


Figure 5.28: Storage tank efficiency with different inlet temp and HTF, (May 2022).

High inlet temperature and low flow rate increase the storage tank efficiency is increased to 50 %. This enhancement due to a low water flow rate which gives enough time to store the largest amount of energy during the day.

Wax performance can be represented by the rate of melting time due to the influence of sunlight. Figure 5.29 shows the storage tank Wax melting time with inlet HTF and inlet temperature on 1st May 2022.

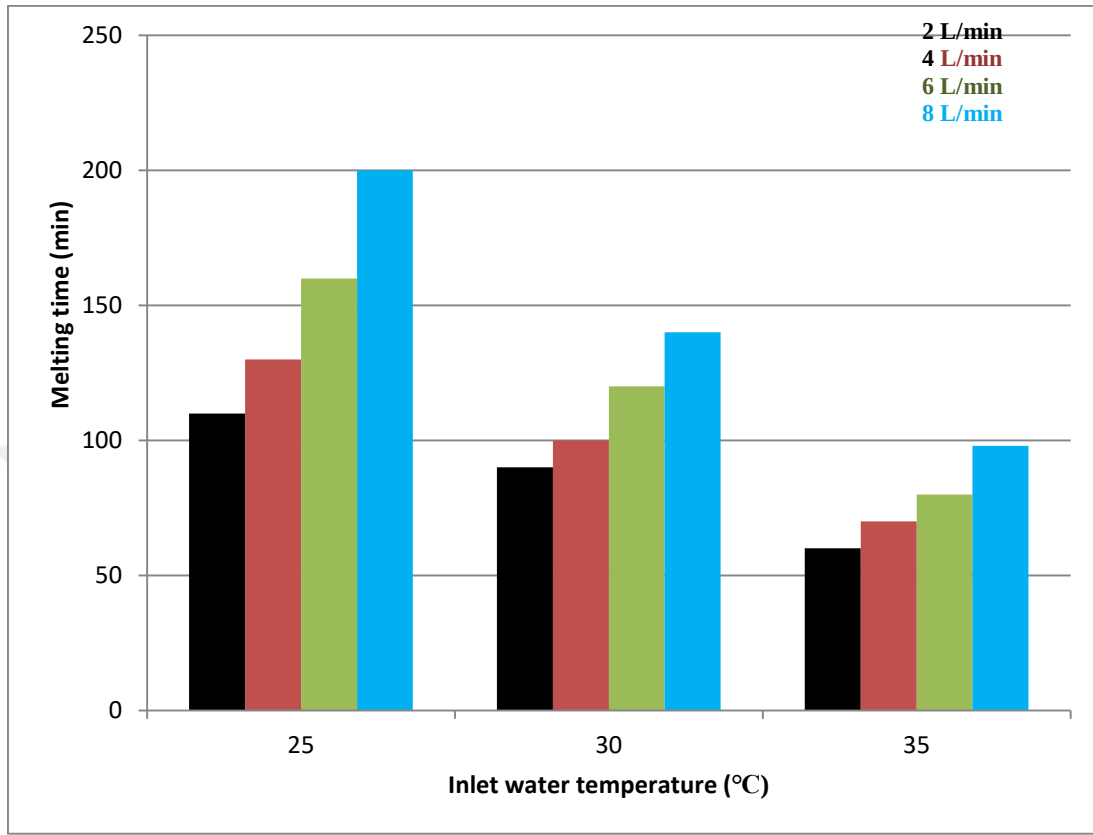


Figure 5.29: Storage tank Wax melting time with inlet HTF and inlet temperature, (May 2022).

Figure 5.29 shows that the HTF flow rate also has a direct impact on how well the TES unit performs in terms of charging time and PCM melting time. Charging and PCM melting take a long time when the flow rate is raised. Furthermore, the performance of the TES unit in terms of charging time and PCM melting time is clearly impacted by the HTF flow rate. By raising the flow rate, charging and PCM melting take less time.

The highest temperatures are recorded at the lowest coil pipe flow rate, and therefore the lowest melting time to be observed. At A high inlet water temperature of 35 °C and water V_{in} of 2 L/min, PW takes a time of 60 min to melt compared to 100 min at $V_{in} = 8$ L/min.

Wax performance can be represented by the rate of melting time due to the influence of sunlight with different months during the daytime. Figure 5.30 shows the storage tank Wax melting time with different inlet temperature at a fixed inlet of flow rate 2 L/min.

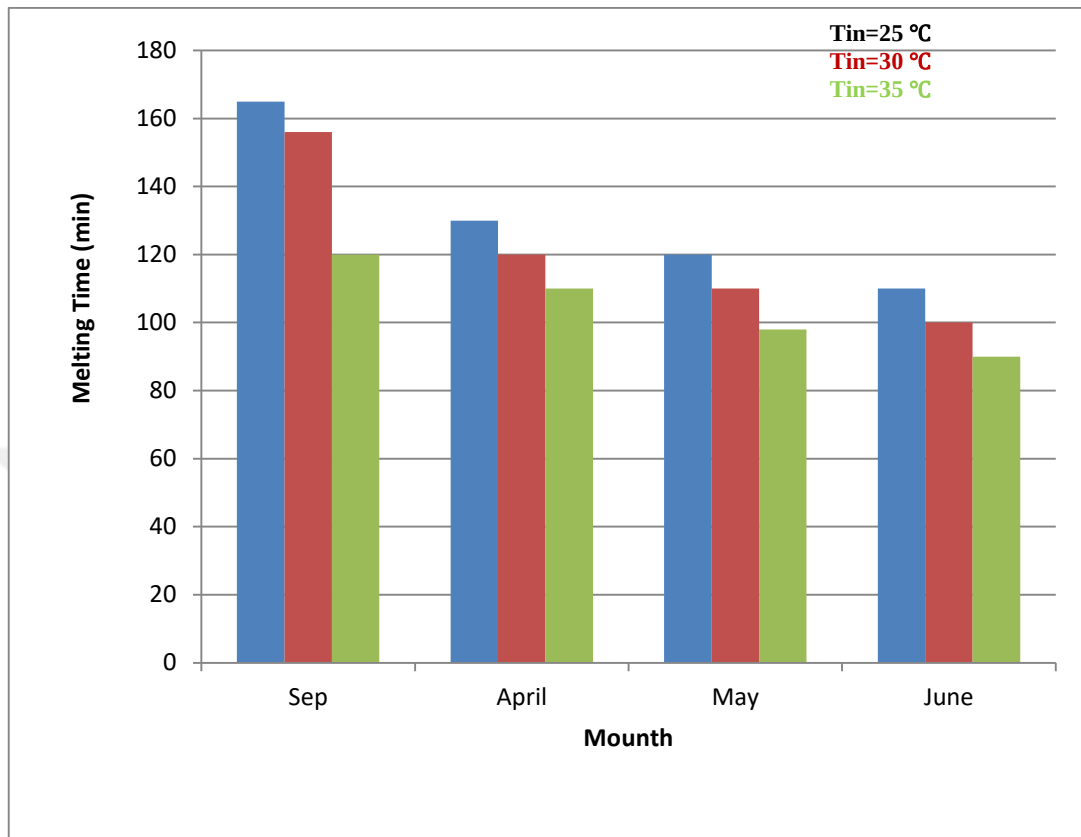


Figure 5.30: Storage tank Wax melting time with the first day of the different months ($V_{in} = 2\text{ L/min}$, $T_{in} = 25\text{ °C}$).

The highest temperatures are recorded in June and the high inlet temperature and therefore the lowest melting time to be observed. At a high inlet temperature of 35 °C and water flow rate of 2 L/min presents time of 90 min to melt.

Figure 5.31 shows the relation between average wax temperature and wax melting time with the different inlet water temperatures on 1st May 2022.

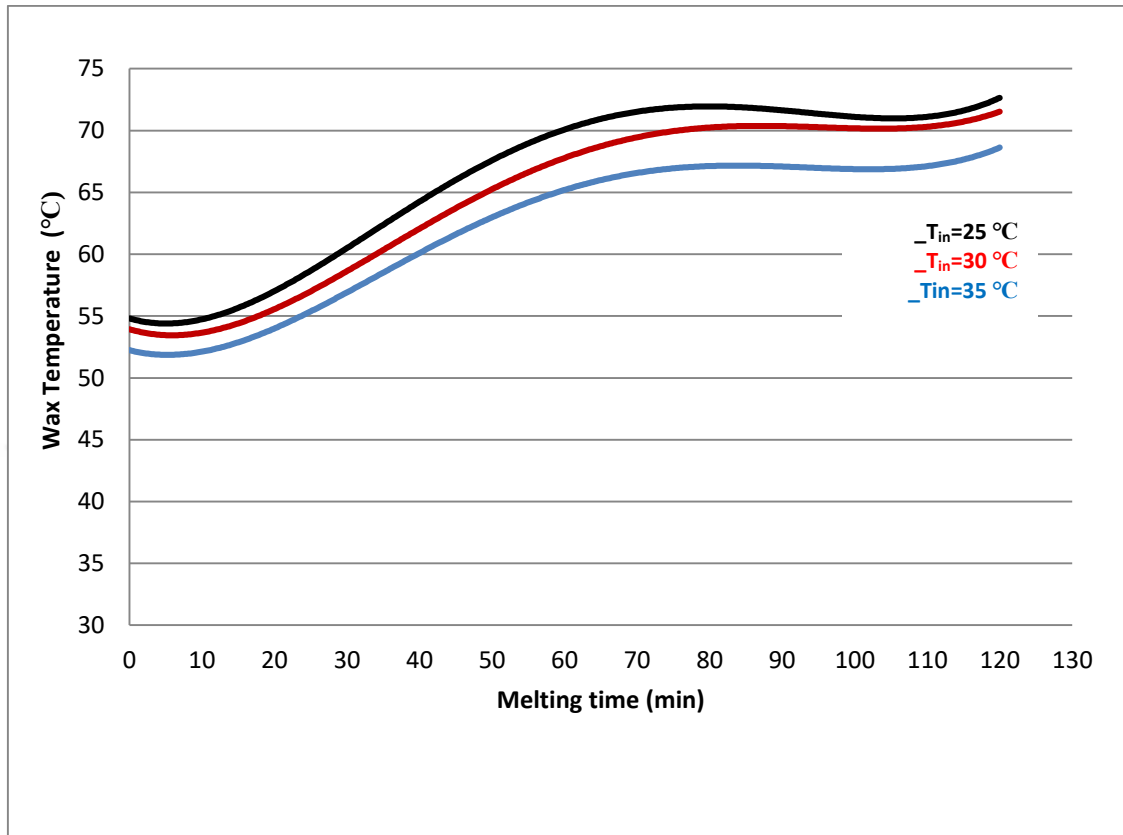


Figure 5.31: Storage tank wax temperature time with different inlet water temperatures ($V_{in} = 2$ L/min), (May 2022).

For all inlet water temperatures, the melting time begins at 54 °C and completes melting at 70 °C with little average temperature difference. Also, for all inlet water temperatures, the melting time was kept constant at 70 min.

The experimental comparison results between wax behavior under the same conditions have been presented in Figure 5.32.

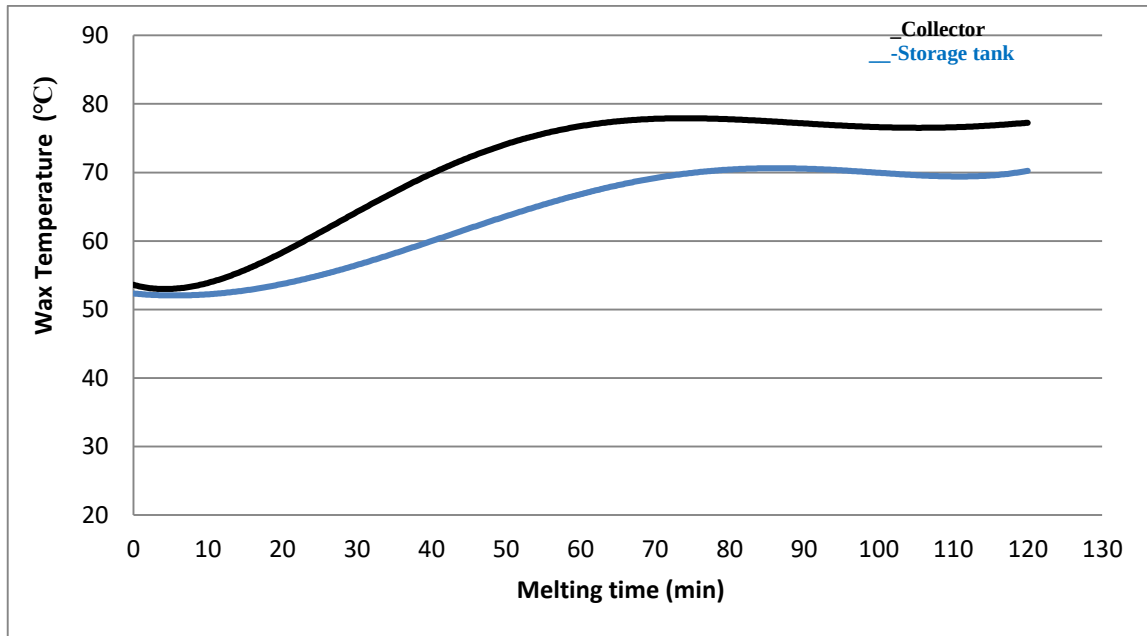


Figure 5.32: Wax average temperature at melting time (1st May2022), ($V_{in} = 2$ L/min), ($T_{in} = 25^{\circ}\text{C}$).

The results show the same behavior of the wax in the tank and wax in the collector. Due to high solar exposure to collector wax the average temperature high value than storage tank wax. The results show the identical behavior of melting PCM after 70 min.

5.3 NUMERICAL ANALYSIS

Numerical simulation has been obtained by using the models and solving continuity, momentum, and energy equations with 2-D turbulent forced convection flow. In this thesis study, the collector inclination angle of 30 degrees, changing the water volumetric flow rate and inlet water temperature with daytime and month are used in thermal simulations.

5.3.1 Grid Independence

Grid independence is one of the most important tests that should be performed throughout any numerical analysis of an issue. The number of grids should have no bearing on the grid independence test's goal of validating the results. The number of grids generated is always a factor in the correctness of the findings in numerical problems. As a result, changing the number of grids affected both the results and the convergence time. The smallest number of grids at

which the results remain unchanged when the number of grids is altered is known as the optimal grid size. The collector hot outlet water temperature and the outlet water temperature from the storage tank that enters the solar collector are the characteristics that are used to test the grid independence and are depicted in Figures 5.33 and 5.34.

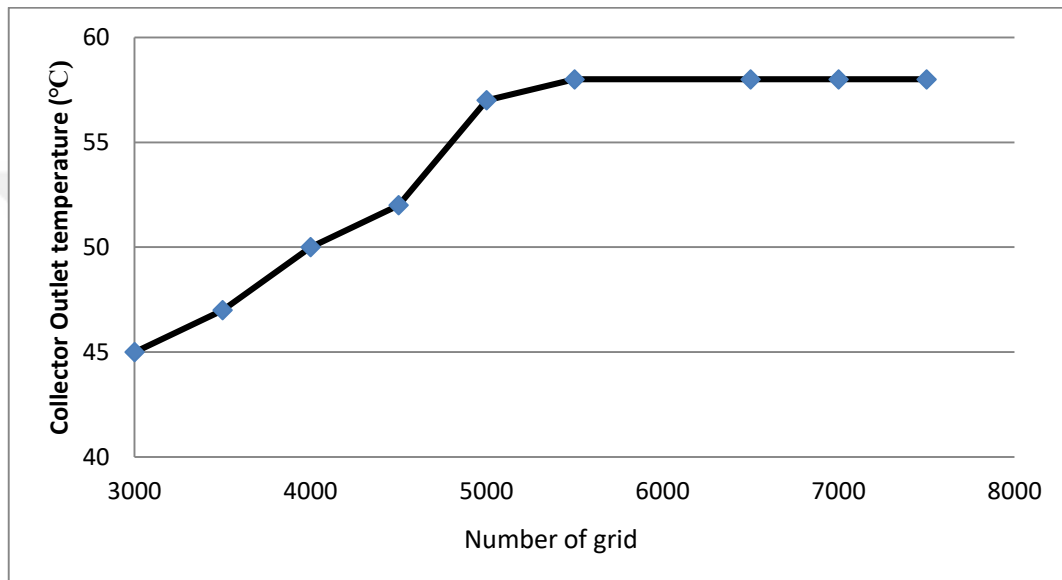


Figure 5.33: Grid independence test based on the hot water temperature (outlet from collector) ,1st May 2022.

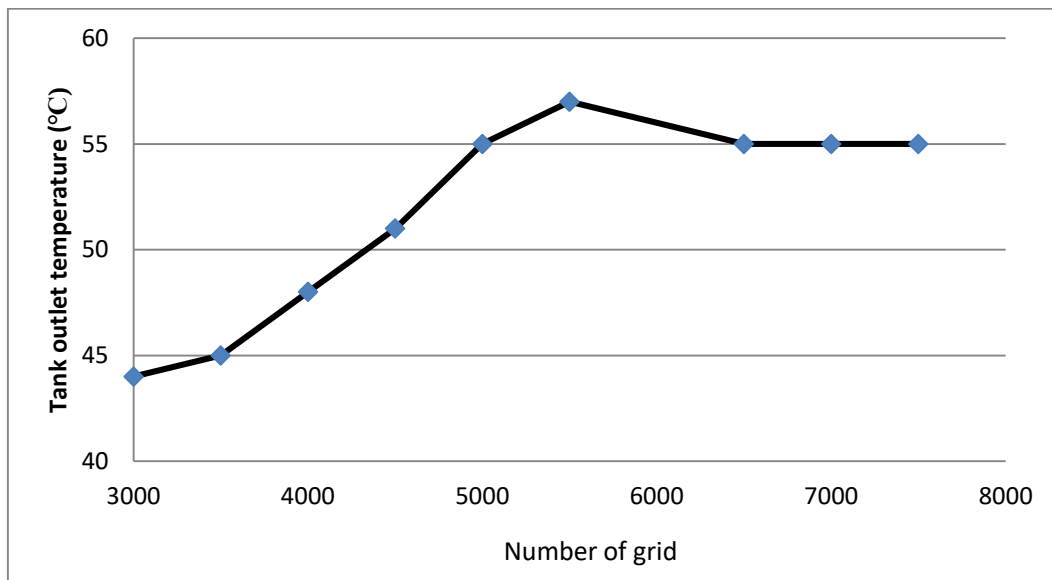


Figure 5.34: Grid independence test based on the of tank outlet temperature ,1st May 2022.

In Figures, 5.33 and 5.34 two graphs for water temperature were given on the Y axis and the number of grids on the X axis. The grid divisions were increased and the corresponding temperature of the outlet hot water in Figure 5.33 and the outlet of tank outlet water temperature in Figure 5.34 were shown. After 6000 grid numbers of collector and 6500 grid numbers of storage tank divisions the results do not depend on grids, so the results are grid independent and can be used for all numerical simulations.

5.3.2 Collector Performance

The 2D-transient model forced convection is simulated numerically by solving the Navier Stokes, energy. Controlling a numerical simulation is used to by the following:

- The mass of PW (10 kg).
- The water flow rate ($V_{in}=2$ to 8/min).
- Water supply temperature (25,30 &35 °C).

The aforementioned information is chosen to analyze its significant impact on the performance of a thermal storage water tank that holds 80 liters of cold water with an initial temperature of 25 °C.

Figure 5.35 presents the numerical results of collector glass and water average temperature variation on 1st May 2022 in the daytime.

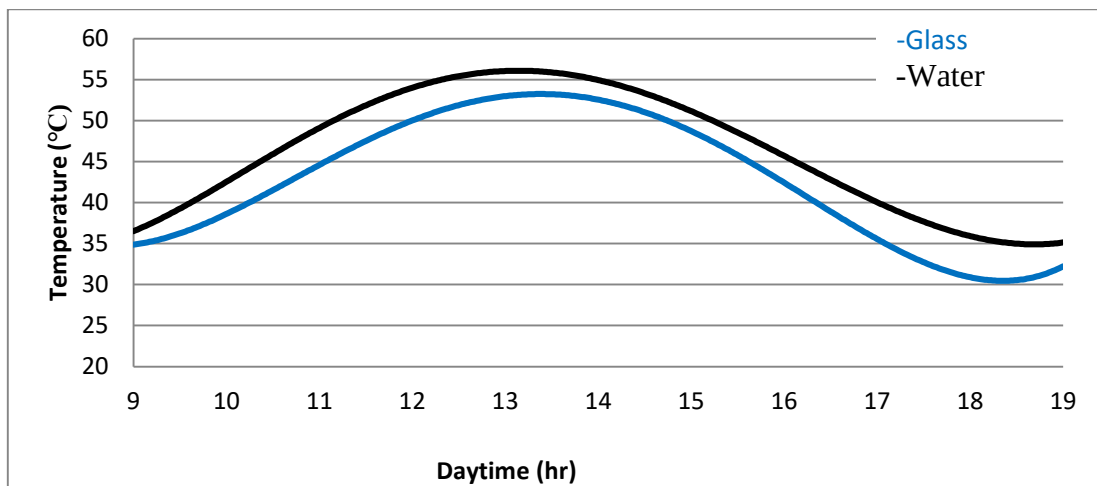


Figure 5.35: Numerical average temperature through collector glass and water at daytime, (1st May 2022), ($T_{in} = 25^{\circ}\text{C}$).

The collector water temperature is high than the cover glass due to the better thermal capacity of water than the glass layer.

Figure 5.36 indicates the comparison results of collector outlet temperature with and without Wax (May 2022), $V_{in} = 2 \text{ L/min}$, $T_{in} = 25^\circ\text{C}$.

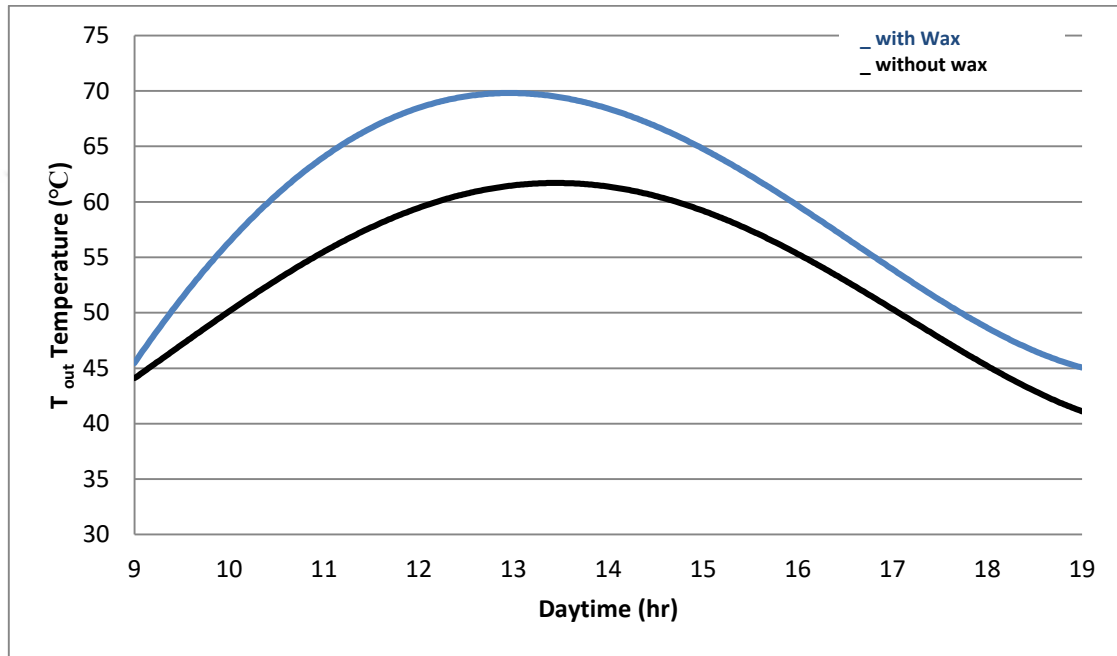


Figure 5.36: Comparison of Collector outlet temperature with and without Wax (May 2022), $V_{in} = 2 \text{ L/min}$, $T_{in} = 25^\circ\text{C}$.

The results show about a 10°C increase at the middy due to the effect of the PCM that surrounded the water pipes.

Numerical collector performance is presented in Figures 5.37 & 5.38 respectively as inlet, outlet and wax temperature. Figures 5.37&5.38 show the collector performance at different storage inlet rate water during the daytime on 1st May 2022.

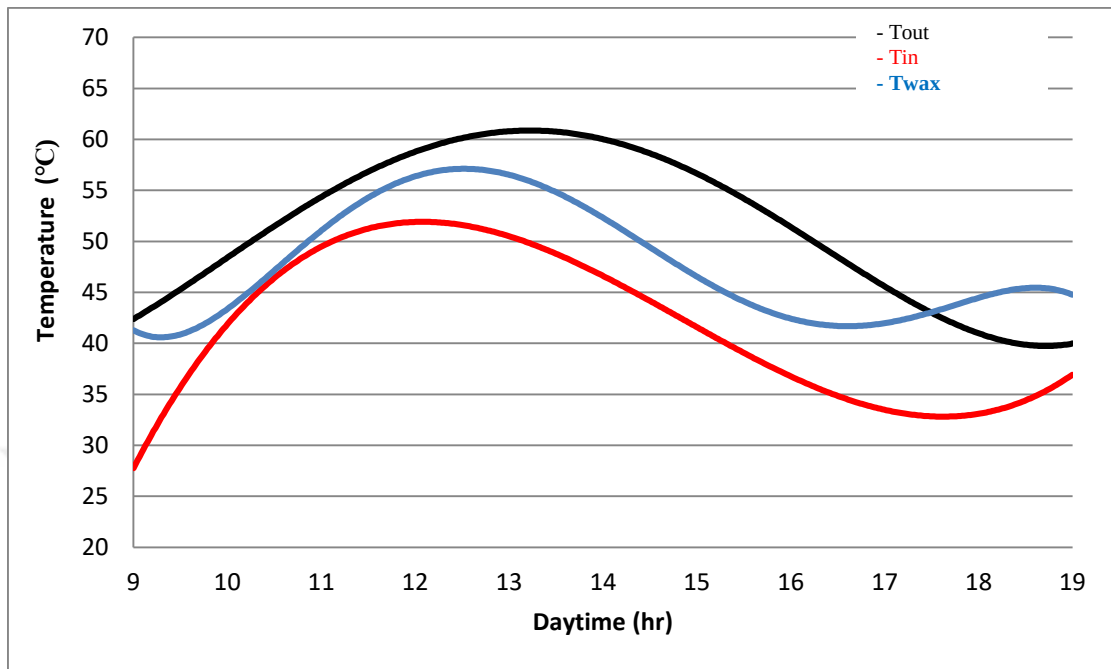


Figure 5.37: Temperature distribution for collector wax and water in May 2022 at daytime $V_{in} = 2$ L/min in tank, $T_{in} = 25^{\circ}\text{C}$, $V_{in} = 4$ L/min in collector.

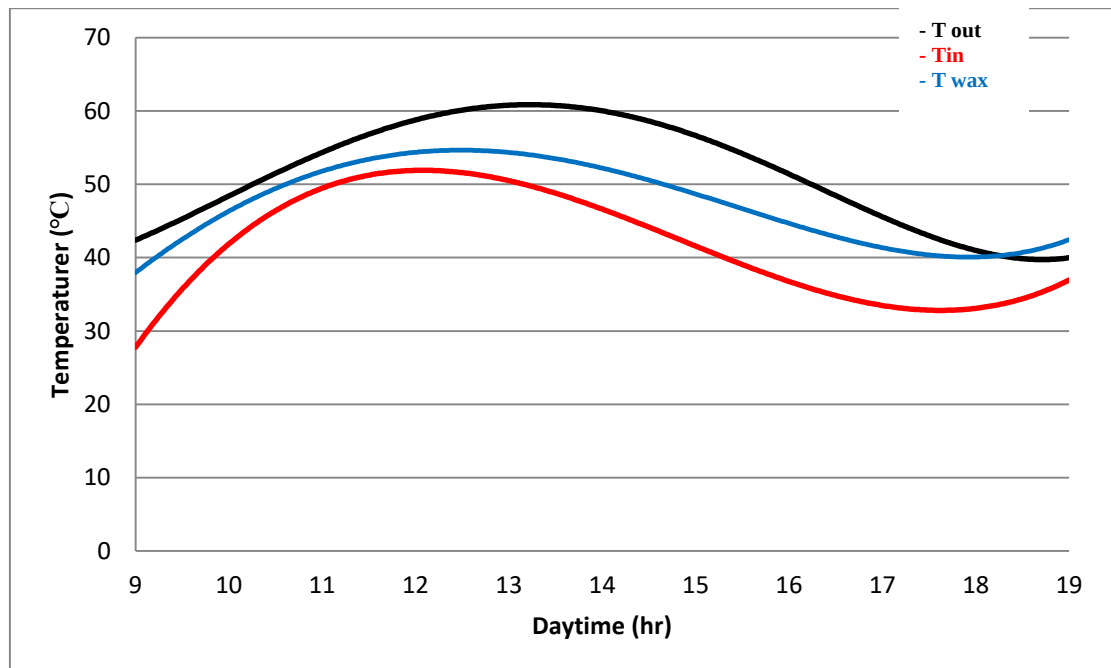


Figure 5.38: Temperature distribution for collector wax and water in May 2022 at daytime, $V_{in} = 4$ L/min, $T_{in} = 25^{\circ}\text{C}$ in tank, $V_{in} = 4$ L/min in collector.

The results show high outlet water collector temperature in the daytime compared with wax temperature at both flow rate 2 & 4 L/min, that's mean no effect of the storage flow rate on the collector performance numerically. At 18.00 PM the PCM began to solidification and release heat to water reaching 45 °C with $V_{in}=2$ L/min. All temperatures decrease at a sunrise as shown in figures 5.38 & 5.39. Also, the behavior of PCM is less than at $V_{in} = 4$ L/min due to less storage energy.

Figure 5.39 presents the collector efficiency at daytime and inlet water flow at constant $T_{in} = 25$ °C, 1st May 2022.

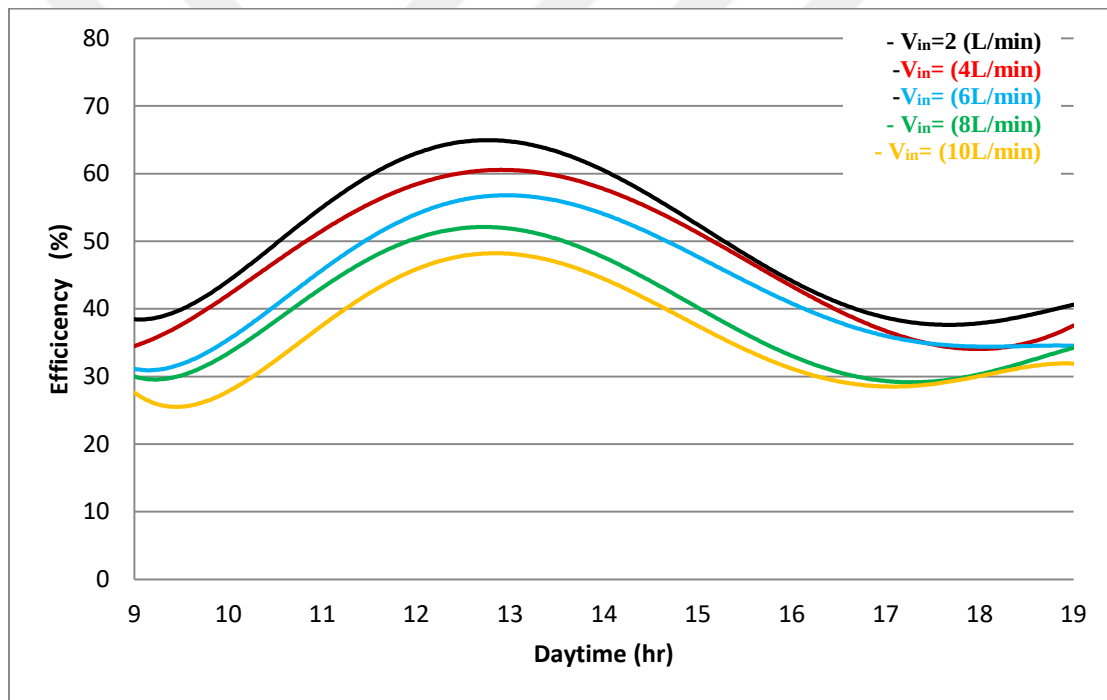


Figure 5.39: Collector efficiency at daytime and inlet water flow at constant $T_{in} = 25$ °C, 1st May 2022.

Maximum efficiency indicates a low flow rate and 12.30 PM, reaches 67%. Also, the effect of storage energy appears clearly in the evening day when low flow rate.

5.3.3 Storage Tank Performance

The numerical model is developed to give a complete description of heat transfer in solar heating water integrating by phase change material PCM by applying it for the CFD programs. The

purpose of solution has been to find the thermal performance of the stored energy, melting time of PCM, at different temperature supplies, and water flow rate in the storage tank.

Figure 5.40 shows the heat exchanger's overall heat transfer coefficient at different months and daytime, ($V_{in} = 2 \text{ L/min}$), ($T_{in} = 25^\circ\text{C}$).

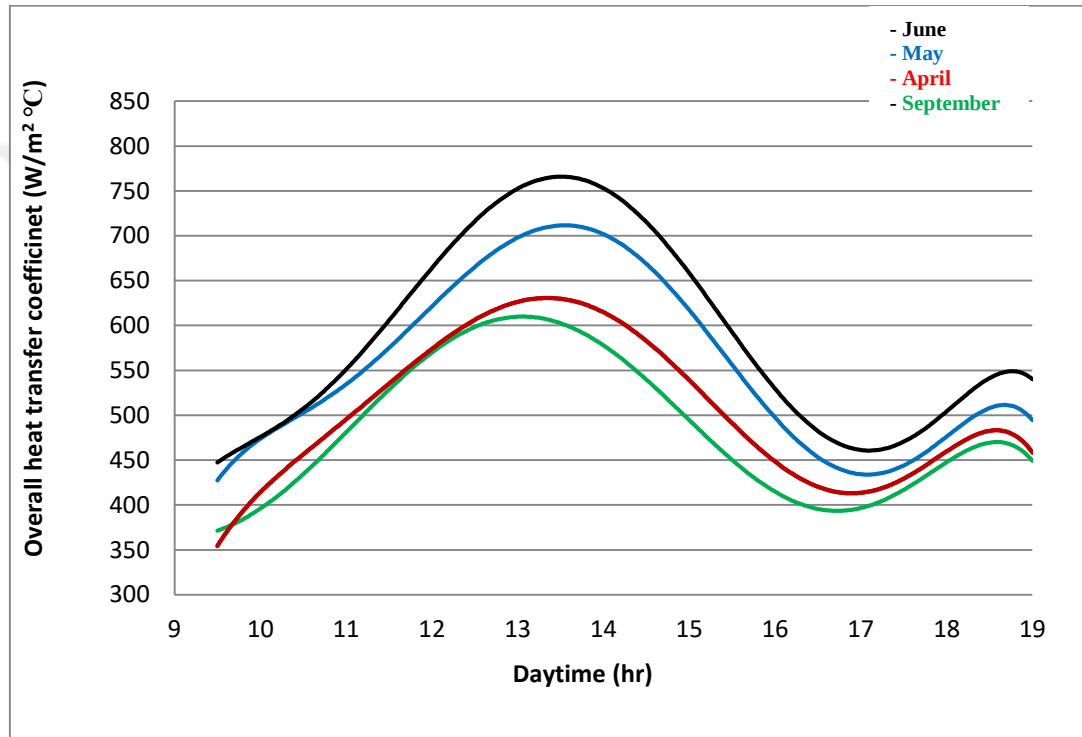


Figure 5.40: Heat exchanger overall heat transfer coefficient at different months and daytime, $V_{in} = 2 \text{ L/min}$, $T_{in} = 25^\circ\text{C}$.

The heat exchanger is represented in the shell(tank) and helical coil tube with counter flow. The results show a high overall heat transfer coefficient with high solar intensity by June month. This value at midday of June increases by about $150 \text{ W/m}^2\text{C}$ from 1st Sep. Also, the results show the effect of PCM on the exchanger's performance in the evening time.

Figure 5.41 shows the heat exchanger effectiveness at different months and daytime, ($V_{in} = 2 \text{ L/min}$, $T_{in} = 25^\circ\text{C}$).

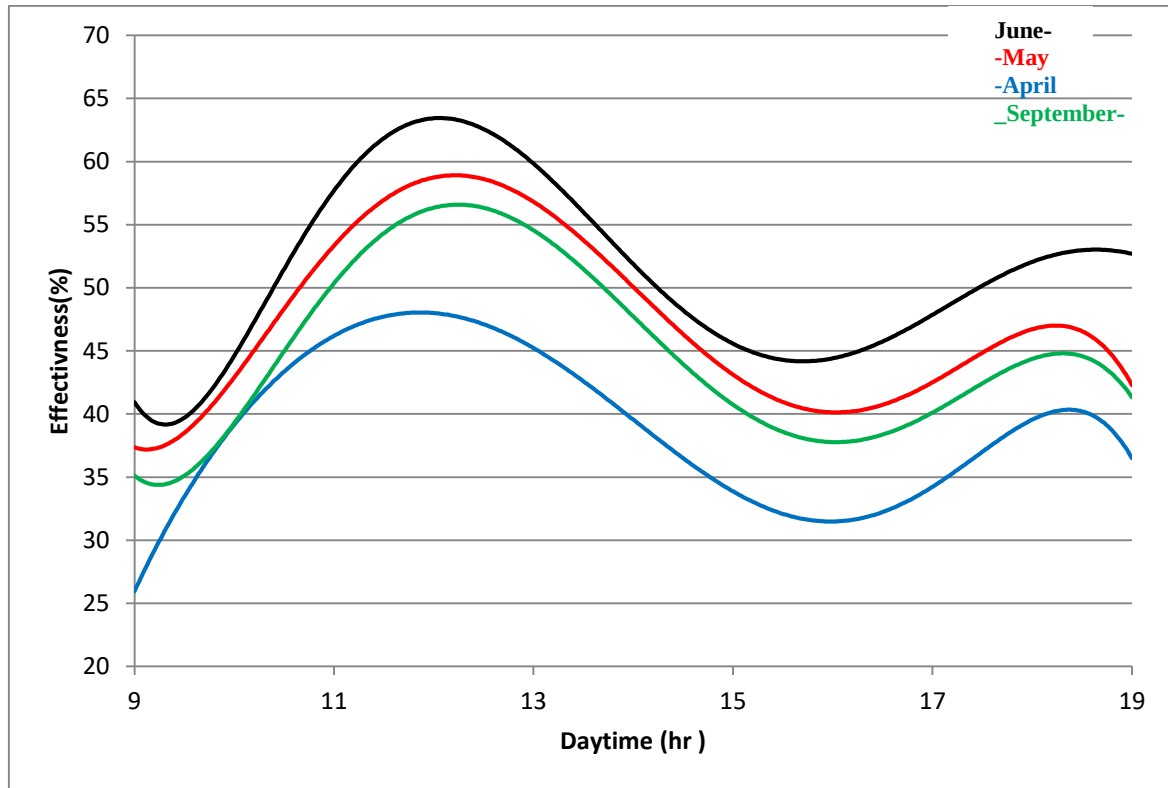


Figure 5.41: Heat exchanger effectiveness with different month and daytime, $V_{in} = 2 \text{ m}$

The results show high effectiveness with high solar intensity by June month. This value at midday of June increases about 10% from 1st September. Also, the results show the effect of PCM on the exchanger performance in the evening time increases by about 15%.

5.4 COMPARISON RESULTS

Comparisons between numerical and experimental results should be carried out to check the solution's accuracy.

Figure 5.42 demonstrates the contrast between the numerical and experimental results for the collector water outlet temperature, May 2022, $V_{in} = 2 \text{ L/min}$. There is a perfect agreement in the results during daylight hours and the maximum error is recorded at 10% at 12.30 PM.

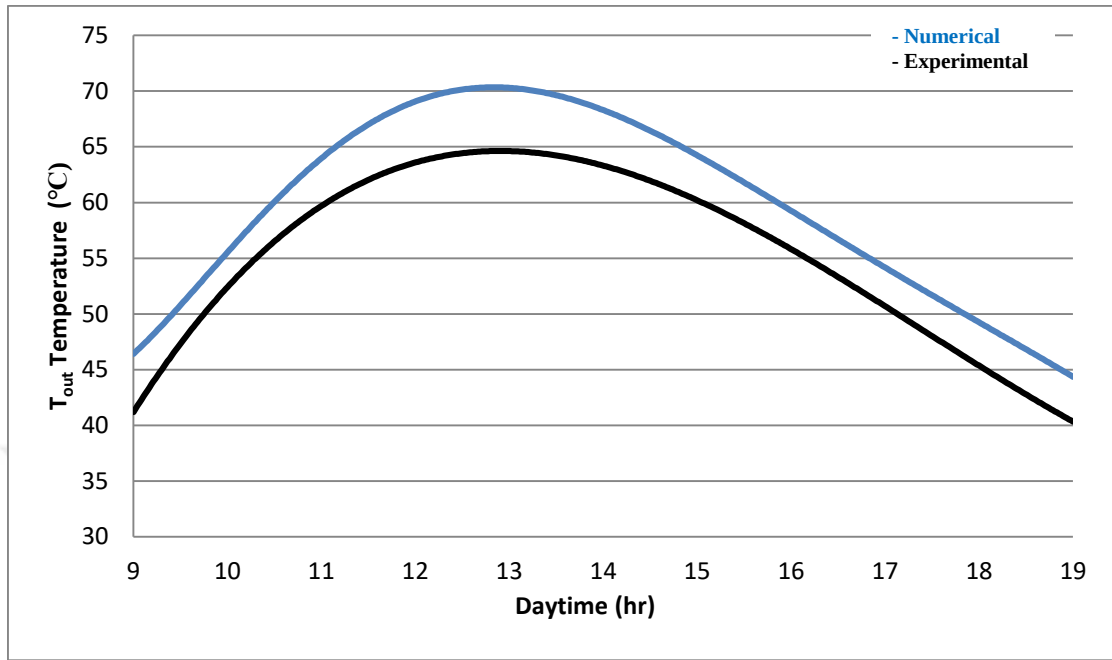


Figure 5.42: Numerical and experimental results comparison for the collector water outlet temperature, May 2022, $V_{in} = 2$ L/min.

Figure 5.43 shows the comparison results of the average collector wax temperature. May 2022, $V_{in} = 2$ L/min, $T_{in} = 25$ °C.

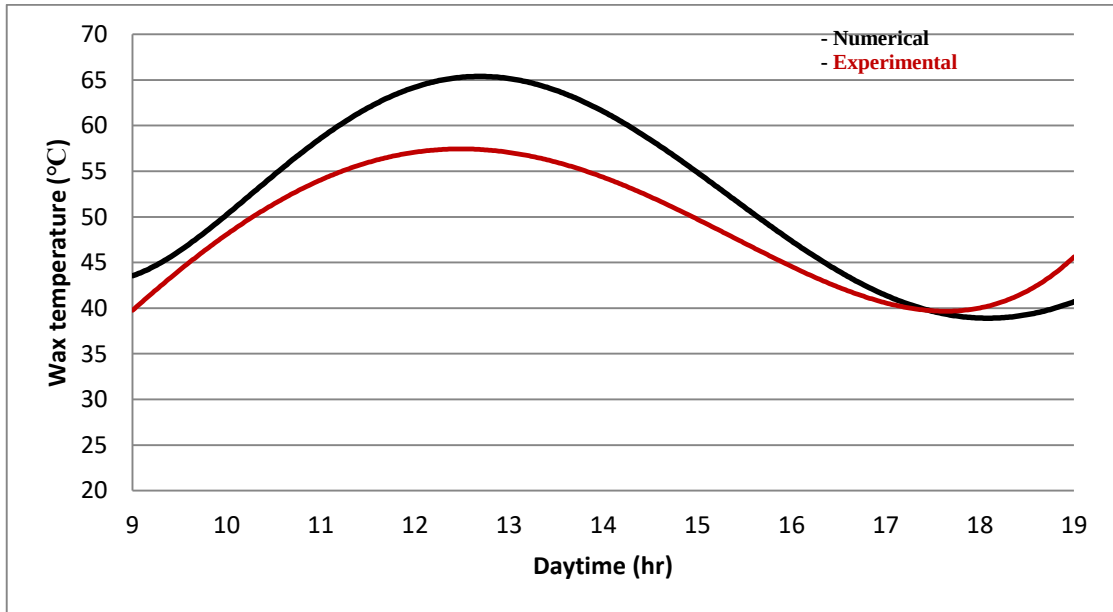


Figure 5.43: Comparison of average collector wax temperature. May 2022, $V_{in} = 2$ L/min, $T_{in} = 25$ °C.

A high difference appears at the maximum solar intensity reaches 8°C. Overall results indicate good agreement in the results during daylight hours and the maximum error is recorded at 8% at 12.30 PM compared to 3% at the other daytime.

A Comparison of experimental & numerical outlet temperature collector with and without Wax material during the daytime, May 2022, $V_{in} = 2 \text{ L/min}$, $T_{in} = 25^\circ\text{C}$ is presented in figure 5.44.

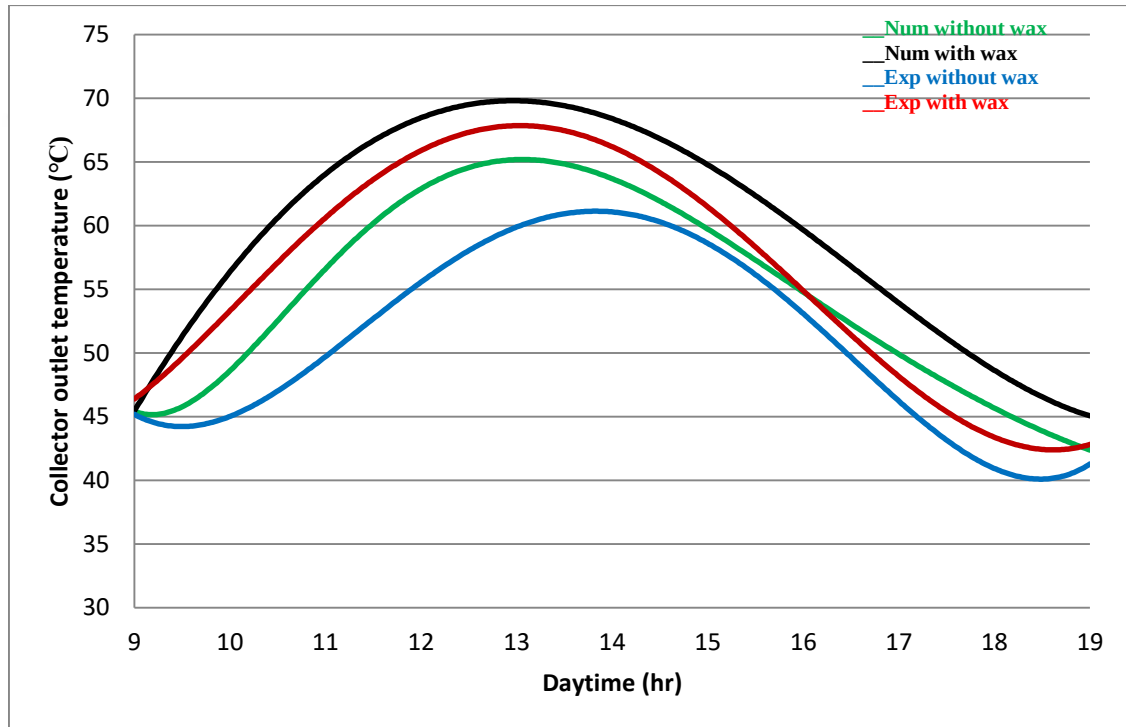


Figure 5.44: Comparison collector outlet temperature at daytime, May 2022, $V_{in} = 2 \text{ L/min}$, $T_{in} = 25^\circ\text{C}$.

Numerical results with and without Wax give high value than experimental results at the beginning of daytime. Maximum error is recorded without wax, reaches 4.7% at mid-daytime.

An important comparison is represented by the figures 5.45&5.46 between collector wax melting time, months, and different water inlet temperatures at constant $V_{in} = 2 \text{ L/min}$.

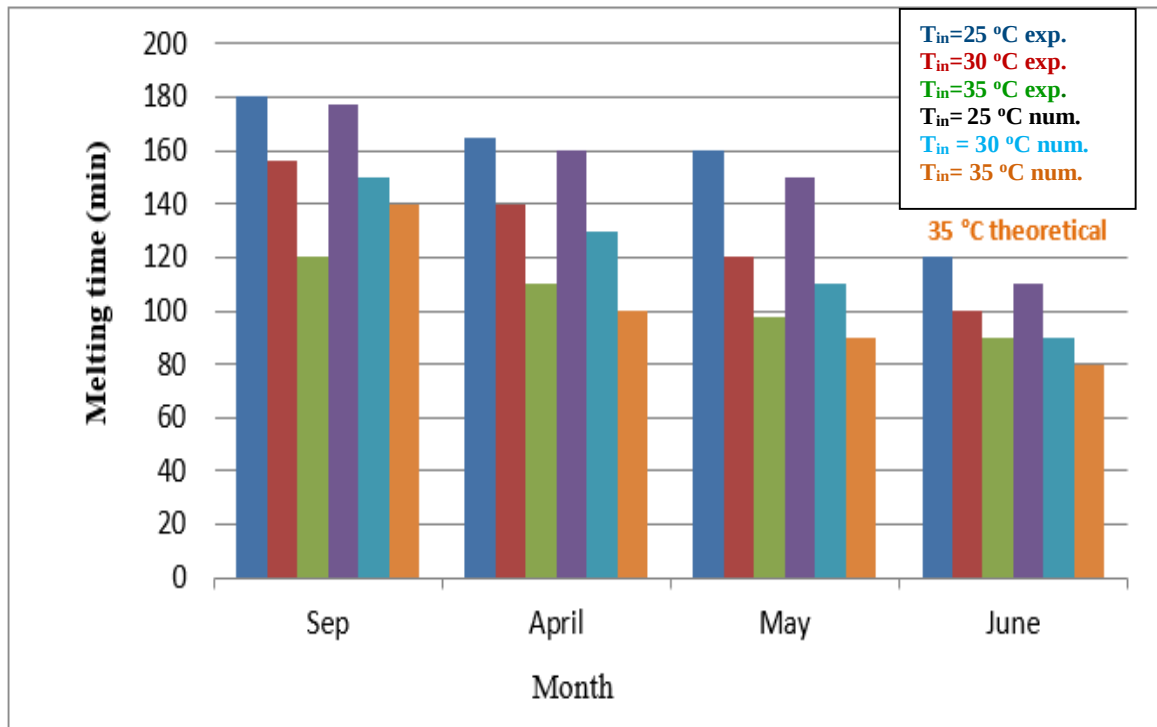


Figure 5.45: Comparison of collector wax melting time at months with different water inlet temperatures and constant $V_{in} = 2\text{ L/min}$.

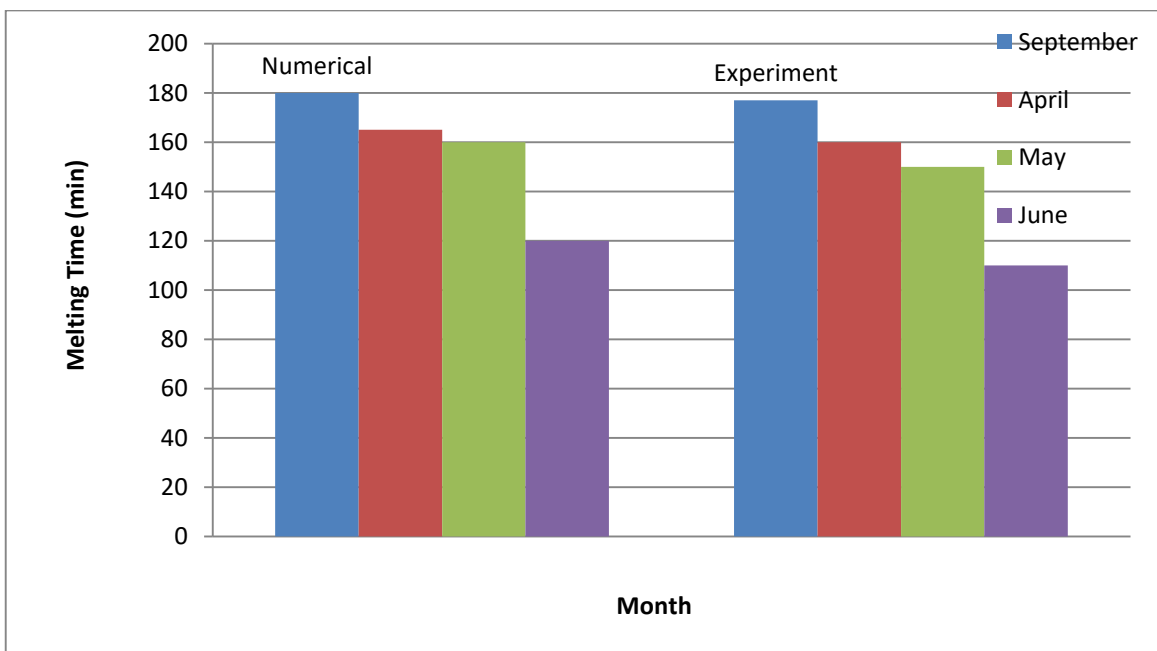


Figure 5.46: Comparison of collector wax melting time at months and constant $V_{in} = 2\text{ L/min}$. $T_{in} = 25^{\circ}\text{C}$.

The inlet temperature plays an important factor to ensure the effectiveness of energy storage. All results show a good agreement at all months and inlet water temperature, maximum error has been recorded at about 1%. The results are close to the fact that there is some difference in the measurement of the practical melting time.



6. CONCLUSIONS AND RECOMMENDATIONS

6.1 CONCLUSIONS

Following have been given the major conclusions reached by this thesis study. It is anticipated that the observed temperature has an influence on the rate of convective heat transfer with solar intensity applications based on the observed experiment temperature distribution and numerical calculations.

6.2 Collector Performance

TES is critical for maintaining the available energy and is a key factor in improving solar energy applications. From the present work make sure by improving energy storage, the balance between demand and consumption can be improved.

-The results showed a significant increase in the temperature of the collector outlet water when filling the collector with PCM. The maximum temperature was reached 68°C compared to 58 °C without PCM as shown in Figure 5.9.

- The HTF rate has a significant impact on both the collector's efficiency and the temperature of the water entering and leaving the collection. The peak output water temperatures increase with a decrease in mass flow rate and integration with the PCM layer.

-The results have shown low inlet HTF and inlet temperature give high Wax melting time compare to others. At a low flow rate, $V_{in} = 2$ L/min melting time reached 225 min at 25 °C as shown in Figure 5.11.

The effect of entry temperature in a helical coil heat exchanger almost has little impact on a solar collector's performance, except in the peak hours of the day, while the greatest effect is PCM release energy at sunset of the day due to the stored energy as shown in Figures 5.12&5.15.

The high value of collector efficiency is recorded at low water inlet flow. At $V_{in} = 2$ L/min the results show an average collector efficiency was increased of 20% at $V_{in} = 2$ L/ than the other flow rate at the mid daytime.

6.3 STORAGE TANK PERFORMANCE

Experiments have shown that utilizing a PCM to maintain hot water for a longer period of time after the system has been turned off or the source of heat energy has stopped produces the optimum thermal performance for the TES system.

- a- In every month that the testing is performed, it was observed that the paraffin's temperature at sunset was higher than the tank's hot water inlet temperature.
- b- The HTF flow rate has a significant influence on the increase in the temperature of the water into the storage tank. Due to an increase in the temperature difference between the input and output, efficiency increases as flow rate declines, reaching at 50% at 2 L/min as shown in Figure 5.28.
- c- The HTF flow rate also has a direct impact on how well the TES unit performs in terms of charging time and PCM melting time. Charging and PCM melting take a long time when the flow rate is raised. Furthermore, the performance of the TES unit in terms of charging time and PCM melting time is clearly impacted by the HTF flow rate. So, at the flow rate $V_{in} = 2$ L/min, Charging and PCM melting take less time.
- d- Inlet water temperature had a significant effect on the PCM melting time. The highest temperatures were in June and the high inlet temperature give the lowest melting time to be observed. At a high inlet temperature of 35°C and water flow rate of 2 L/min presents time of 90 min to melt compared to the other as shown in Figure 5.30. So, when the PCM melting rate increases at the HTF inlet temperature supply increases the melting time falls by 60 percent and its efficiency is reached 67%.
- e- Through the results, it is clear, when HTF flow rate decreases as the PCM melting rate rises by (23%) at 2L/min.
- f- The change in the HTF flow rate at the latent heat region through the collector and storage tank is clearly affecting the thermal storage energy of the storage tank.
- g- The thermal storage energy of the storage tank is significantly influenced by the HTF intake temperature supply. The thermal storage energy rises by when the inlet temperature is raised (60 %).

- h- The effectiveness of the heat exchanger is enhanced by about 10% at sunset time from sunrise time, and the enhancement of PCM on the heat exchanger effectiveness reaches 18% at sunset of May 2022 when compared with storage tanks without PCM.
- i- The study confirms that the presence of energy storage materials is a beneficial factor for solar energy systems in the permanence of the night hours.

6.4 RECOMMENDATIONS

- i. Experiment with more than one type of storage material.
- ii. Addition of nanomaterials to water and energy storage materials.
- iii. The same study can also be done utilizing different types of solar collectors

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