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MSc. in Mechanical Engineering

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REPUBLIC OF TÜRKİYE

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INVESTIGATION OF THERMAL PERFORMANCE OF A GROUND COUPLED CARBONATED SOFT DRINK COOLING SYSTEM WITH AN UNDERGROUND THERMAL ENERGY STORAGE TANK

M.Sc. THESIS

IN

MECHANICAL ENGINEERING

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June 2023

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Ahmed Wshyar RASOOL

ABSTRACT

INVESTIGATION OF THERMAL PERFORMANCE OF A GROUND COUPLED CARBONATED SOFT DRINK COOLING SYSTEM WITH AN UNDERGROUND THERMAL ENERGY STORAGE TANK

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This study investigates the thermal performance of a ground coupled carbonated soft drink cooling system with an underground thermal energy storage tank. The system is composed of three main components: a chiller unit, a thermal energy storage (TES) tank, and a syrup room to be cooled. An analytical model was developed to determine the thermal performance parameters of the system using Duhamel's superposition and dimensionless variables to solve the transient heat transfer problem around the TES tank. The analytical model includes energy expressions for the main components as well as a solution to the heat transfer problem surrounding the TES tank. The hourly temperature variation of water in the TES tank and other performance parameters, such as Coefficient of Performance (COP), are calculated. To do this, a simulation model was developed using MATLAB software. The study investigated the effect of various system parameters such as the size of the underground thermal energy storage tank, soil type, and Carnot factor on the thermal performance of the system. The findings reveal that granite is the most efficient type of soil for the cooling process and that COP increases with increasing Carnot Factor and tank size. The results indicate that the ground coupled carbonated soft drink cooling system with an underground thermal energy storage tank is a feasible and energy-efficient solution for providing cooling for carbonated soft drinks. The system has the potential to reduce energy consumption and greenhouse gas emissions associated with traditional cooling systems.

Key Words: Energy, Energy storage, Ground coupled chiller, Syrup room, TES

ÖZET

YERALTI TERMAL ENERJI DEPOLAMA TANKI ILE BIRLEŞTIRILMIŞ BIR TOPRAK BAĞLANTILI KARBONATLI İÇECEK SOĞUTMA SISTEMININ TERMAL PERFORMANSININ İNCELENMESI

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57 sayfa

Bu çalışma, yer altı termal enerji depolama tankına sahip, yere bağlı gazlı meşrubat soğutma sisteminin termal performansını araştırmaktadır. Sistem üç ana bileşenden oluşmaktadır: chiller ünitesi, termal enerji depolama (TES) tankı ve soğutulacak şurup odası. TES tankı etrafındaki geçici ısı transferi problemini çözmek için Duhamel'in süperpozisyon ve boyutsuz değişkenleri kullanılarak sistemin ısıl performans parametrelerini belirlemek için analitik bir model geliştirilmiştir. Analitik model, ana bileşenler için enerji ifadelerinin yanı sıra TES tankını çevreleyen ısı transferi sorununa bir çözüm içermektedir. TES tankındaki suyun saatlik sıcaklık değişimi ve Performans Katsayısı (COP) gibi diğer performans parametreleri hesaplanır. Bunun için MATLAB yazılımı kullanılarak bir benzetim modeli geliştirilmiştir. Çalışma, yeraltı termal enerji depolama tankının boyutu, toprak ve Carnot faktörü gibi çeşitli sistem parametrelerinin sistemin termal performansı üzerindeki etkisini araştırmıştır. Bulgular, granitin soğutma işlemi için en verimli toprak türü olduğunu ve artan Carnot Faktörü ve tank boyutu ile COP'nin arttığını ortaya koymaktadır. Sonuçlar, bir yeraltı termal enerji depolama tankına sahip yere bağlı gazlı meşrubat soğutma sisteminin gazlı alkolsüz içecekler için soğutma sağlamak için uygun ve enerji açısından verimli bir çözüm olduğunu göstermektedir. Sistem, geleneksel soğutma sistemleriyle ilişkili enerji tüketimini ve sera gazı emisyonlarını azaltma potansiyeline sahiptir.

Anahtar Kelimeler: Enerji, Enerji depolama, Zemin bağlantılı soğutma grubu, şurup odası, TES

"Dedicated to my family"

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c	Specific heat of earth (kJ kg^{-1} K^{-1})
$\mathbf{c_{pa}}$	Specific heat of air (kJ kg^{-1} K^{-1})
$\mathbf{c_w}$	Specific heat of water (kJ kg^{-1} K^{-1})
k	Thermal conductivity of earth (W m^{-1} K^{-1})
P	Dimensionless, $P = \frac{\rho_w\, c_w}{3\rho\, c}$
q	Dimensionless heat transfer to the tank
Q	Heat transfer to the tank (W)
$\dot{Q}_L$	Heat gain to the syrup room (W)
r	Radial distance from the tank centre (m)
R	Tank radius (m)
x	Dimensionless radial distance
t	Time (s)
T	Earth temperature (K)
$\mathbf{T_a}$	Ambient air temperature (K)
$\mathbf{T_o}$	Outside design air temperature (K)
$\mathbf{T_w}$	Water temperature in the tank (K)
$\mathbf{T_c}$	Water temperature in fan coil unit (K)
$\mathbf{T_\infty}$	Deep ground temperature (K)
u	Dimensionless parameter for design condition
$\mathbf{(UA)_H}$	Product of heat transfer coefficient and area of syrup room (W K^{-1})
$\mathbf{(UA)_{FC}}$	Product of heat transfer coefficient and area for fan coil (W K^{-1})
V	Volume of the tank (m^3)
w	Dimensionless compressor work
W	Compressor input power (W)
$\boldsymbol{\alpha}$	Thermal diffusivity of the soil (m^2 s^{-1})

$\boldsymbol{\beta}$	Carnot factor
$\boldsymbol{\eta_c}$	Carnot efficiency
$\boldsymbol{\phi}$	Dimensionless temperature for earth, $\frac{T-T_\infty}{T_\infty}$
$\boldsymbol{\phi_a}$	Dimensionless ambient air temperature
$\boldsymbol{\phi_w}$	Dimensionless water temperature in the tank
$\boldsymbol{\psi}$	Dimensionless temperature
$\boldsymbol{\gamma}$	Dimensionless parameter, $4\pi Rk/(UA)_H$
$\boldsymbol{\rho}$	Density of soil (kg m^{-3})
$\boldsymbol{\rho_w}$	Water Density (kg m^{-3})
$\boldsymbol{\tau}$	Dimensionless time, $\tau = \frac{\alpha t}{R^2}$

LIST OFABBREVIATIONS

COP	Coefficient of Performance
COPs	Coefficient of Performance of the whole system
CF	Carnot Factor
TES	Thermal Energy Storage
CSD	Carbonated Soft Drink
CO_2	Carbon Dioxide
PVGIS	Photovoltaic Geographical Infromation System

CHAPTER 1
INTRODUCTION

Energy, which is characterized as the capacity to perform work, is a fundamental idea in science. There are many different types of it, including thermal, kinetic, chemical, and nuclear energy. As it drives everything from the devices we use to the cars we drive, energy is a crucial component of our daily life. One of the most frequent forms of energy is thermal energy, which is the energy of heat. Conduction, convection, or radiation are three ways that this form of energy can pass from one substance to another. It is created by the motion of atoms and molecules within a substance. Several processes depend on thermal energy as a source of power, and it can even be used to create electricity. In addition, it is employed in numerous regular activities like food preparation, building heating and cooling, and manufacturing. In order to lessen our reliance on fossil fuels and lessen the environmental impact of our energy use, it is crucial to take into account the sustainability and efficiency of the thermal energy sources we use. We should also investigate renewable alternatives, such as solar thermal energy, as well as energy conservation techniques.

Global warming has become one of the most important and urgent issues in the world due to the increasing demand for energy and the current dominance of fossil fuels as a source of energy. Although the use of renewable energy sources like wind and solar has increased recently, their availability can be erratic and time-based. As a result, it's critical to develop more efficient methods of using energy, especially in energy storage. Long used as a method of energy conservation, thermal energy storage entails temporarily storing thermal energy at either high or low temperatures [1]. There exist several different kinds of thermal energy storage systems, including: Latent heat storage, which includes the storing of thermal energy in the form of latent heat, which is discharged when a substance changes state from solid to liquid or vice versa, is one sort of thermal energy storage system.

Sensible heat storage, which involves preserving thermal energy as the heat that is absorbed or released when a substance is heated or cooled, and the practice of storing thermal energy in materials that go through a phase change at a certain temperature, such as melting or evaporating, is known as thermal energy storage in phase change materials. There are several uses for thermal energy storage, including producing electricity, heating and cooling buildings, and industrial activities. It can be combined with renewable energy sources like geothermal and solar power to boost their dependability and lessen the need for fossil fuels. The ability to store excess energy and use it when needed rather than wasting it is one of the significant advantages of thermal energy storage. This may aid in lowering greenhouse gas emissions and energy expenses. Systems for storing thermal energy recover enough energy and reduce thermal energy and temperature depreciation losses to maximize energy efficiency. They may also be tailored to meet the special requirements of a given application, making them a versatile and effective solution for energy efficiency. Overall, thermal energy storage is a crucial element in the shift to a more sustainable energy system and the conservation of energy. It is a useful supplement to any energy saving strategy due to its adaptability and capacity to retain surplus energy.

In order to accumulate thermal energy, thermal energy storage (TES) tanks, which are essentially big tanks that are insulated to preserve the temperature of the stored thermal energy, are employed in conjunction with heat pump and refrigeration systems. TES tanks are in two major varieties: those that store cold water and those that store hot water. In order to store extra heat produced by heat pump systems, TES tanks for hot water are frequently utilized in conjunction with them. When necessary, the extra heat might be used to heat structures or other areas. In order to store extra cooling produced by refrigeration systems, TES tanks that hold chilled water are frequently utilized in conjunction with them. Buildings or other areas can then be cooled using the extra cooling. TES tanks can help save energy and money on energy bills by reducing the need for energy-guzzling heating and cooling systems. Installing TES tanks underground is one technique to increase their efficiency. The dirt around the tank serves as a natural insulator, assisting in maintaining the temperature of the thermal energy stored and reducing temperature changes brought on by outside air temperatures.

A highly effective and affordable option for heating and cooling areas and processes is a ground-coupled heat pump and refrigeration systems. These systems make use of the heat capacity of the ground to regulate the temperature of the spaces. The high energy efficiency of ground-coupled heat pump systems is one of their main benefits. Compared to traditional methods of heating and cooling, these systems can operate at substantially better efficiency thanks to the thermal mass of the ground. Moreover, these systems can frequently run at lower compressor rates, which lowers energy usage and increases equipment longevity. These systems may also be made to function at a variety of temperatures, rendering them appropriate for usage in a range of climates. An example of a refrigeration system that uses ground as a heat sink is the earth linked cooling system. With these systems, heat is transferred from the refrigeration system to the earth through an underground pipe network. In comparison to conventional air-cooled refrigeration systems, ground coupled cooling systems provide a number of advantages, including greater performance in hot regions and increased energy efficiency.

Chillers are heat transfer devices used to extract heat from a fluid or process by a vapor-compression refrigeration cycle and exhaust that heat into the atmosphere or surrounding environment. There are two conventional types of chillers, based on their condensation processes, used in the various industrial and residential applications. Air-cooled chillers exchange heat with the ambience during the condensation process, and water-cooled chillers use cooling towers to which heat is rejected. An important application in which both types of chillers have been used is the cooling of Carbonated Soft Drinks (CSD). Carbonated Soft Drinks have a specific temperature range to which they need to be cooled before the filling process of Carbon Dioxide (CO_2) into the beverage. Air-cooled and water-cooled chillers operate with reasonable coefficients of performances (COP) in these systems. However, since the primary parameter affecting the system performance is the inlet temperature into the condenser, when the ambient temperature of air is very high, especially during summer days when the cooling load requirement is highest, there is an overload in the compressor work and a decrease in the COP of the system. A viable potential solution to this problem is the application of ground coupled cooling systems. Due to the fact that heat is dissipated to an adequate ground depth, in which there is a lower and more stable temperature, ground coupled cooling systems are capable of functioning at greater values of COP throughout the summer. The similar idea underlies heat pump systems that are ground coupled, which exploit the earth's greater heat sink temperatures in winter to achieve greater values of

COP than air source systems. Accordingly, ground coupled cooling systems have high potential to prove to be energy efficient systems in the industry and could provide an alternative to conventional chiller systems, especially in the food and beverage production sectors.

The purpose of the study is to create a mathematical model for the Ground Coupled Carbonated Soft Drink Cooling System with a spherical TES tank. The hourly (unsteady) heat transfer issue for the underground thermal tank is resolved by applying Duhamel's superposition and nondimensional variables. To create the model, energy equations for each section of the refrigeration system were used. The model is created in MATLAB, and after that, performance metrics and the system's periodic performance over a ten-year period is to be determined.

The remaining chapters that compose this thesis are as follows:

Chapter Two, literature review and relevant research are presented in detailed and comprehensive summaries.

Chapter Three, analytical modeling of the system with a spherical TES tank is performed. The spherical storage differential equation, boundary conditions, and energy balance are specified along with the problem. The issue is then resolved using Duhamel's superposition method and similarity transformation. Using the deduced formulas for the refrigeration unit, the tank's water temperature is then determined hourly.

Chapter Four, the validation results of two systems in literature with TES tank are illustrated. One is for a space heating application of heating system for a home with a ground coupled heat pump using a spherical TES tank using a similarity transformation technique. The second one is a system for ground-coupled space cooling with a TES tank that applied complex finite Fourier transform (CFFT) technique.

Chapter Five, the results and discussions are presented using the models shown in chapter three.

Chapter Six, Conclusions of this research along with propositions for additional research are offered.

CHAPTER 2
LITERATURE REVIEW

2.1 Introduction

Extensive amounts of energy are used in the industrial cooling processes, which is growing more expensive as the price of fossil fuels rises. Concerns have also been raised about the effects on the environment and human health. Alternative and renewable energy sources should be used to address these problems. For cooling systems to operate efficiently and affordably, thermal energy ought to be stored in underground TES tanks. As a result, less energy is needed to complete the cooling process. The use of storage of thermal energy in refrigeration and heat pump systems for heating and cooling implementations is thoroughly reviewed in this chapter.

2.2 Applications of TES Tanks in Cooling and Heating Processes

There have been many analytical and experimental investigations regarding the development, evaluation, and improvement of TES systems for use in heating and cooling applications. Yang et al. [2] made a comprehensive review of cold thermal energy storage methods at sub-zero temperatures, including storage material properties and their criteria, numerical and experimental examples, and their current and potential applications. Al Shaqsi et al. [3] made an overview of several forms of energy storage and compared the costs, limitations, advantages, environmental considerations, and characteristics. Selvnes et al. [4] presented a thorough review on researches on the use of phase change materials (PCM) in refrigeration systems as cold thermal energy storages (CTES). Their results illustrate that given water's extensive use and inexpensive cost, as PCM for CTES, it is the most frequently studied system. Mehari et al. [5] reviewed the application of absorption thermal energy storage cycles and their incorporation with conventional absorption chiller/heat pump systems. Bott et al. [6] reviewed and selected tank thermal energy storages, pit thermal energy storages, and water-gravel thermal energy storages to

be the most promising water-based closed seasonal thermal energy storage systems. Wang et al. [7] reviewed the research progress on the application of CO_2 hydrates in cold thermal energy storages by summarizing models, characterization methods, and system simulations. Lizana et al. [8] reviewed recent developments in the creation of thermal energy storage materials for use in construction applications toward zero-energy buildings. They concluded from their analysis that latent and sensible energy storage systems have the highest potential for competitive energy efficiency measures.

There have also been several research and experimental work done to improve the COP of cooling systems in industries and buildings as well as investigations on ground coupled heat pump systems for thermal energy storage systems' optimization. Rosen et al. [9] evaluated cold thermal storage systems in terms of energy and exergy analyses. Yumrutas et al. [10] determined the annual periodic performance of a space cooling system employing a spherical underground thermal energy storage tank and a ground coupled chiller system by developing a computational model. De Swardt and Meyer [11] conducted an experimental and simulation based comparison between the performances of a conventional air source heat pump system and a reversible ground coupled heat pump connected to a municipality water reticulation system for both cooling and heating applications. Their results indicated the validity of using ground source heat pumps to optimize the consumption of energy in the sector of air conditioning. Kuyumcu et al. [12] developed an analytical model to utilize waste or excess energy that is rejected from a chiller unit of an ice rink in the form of heat and is then kept in a thermal energy storage (TES) tank underground in order to obtain and improve the long-lasting performance of a swimming pool heating system using Duhamel's superposition and similarity techniques. They found that it takes the heating system for swimming pool five to seven years to reach regular operation. Also, the outcomes demonstrated that the heating system's effectiveness is at its peak when the ice rink surface area is 475 m^2 and the swimming pool area is 625 m^2.

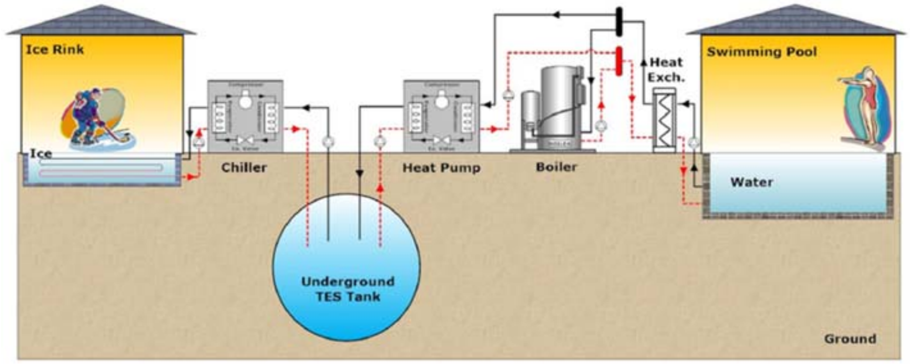

Figure 2.1 A Swimming Pool heating System with Ice Rink and TES Tank [12]

Zogou and Stamatelos [13] discovered that many of the applications for heating and cooling benefit from the warmer Mediterranean climates after studying the effects of climatic conditions on their performance. Yumrutas and Unsal [14] developed an analytical and computational model based on hourly operation to establish the performance parameters of a heat pump-based home heating system using a spherical subsurface TES tank. In order to solve the TES tank issue, Tutumlu et al. [15] designed a computational and analytical model for a cooling system with a spherical underground TES tank for an ice rink. Duhamel superposition and similarity transformation techniques were used to solve the problem. Ismaeel and Yumrutas [16] investigated the performance of solar-assisted heat pump system for drying wheat with an underground thermal energy storage tank by developing a mathematical model to find performance parameters on an hourly basis. The results indicate that the system begins to operate on a regular basis from the fifth year onward and that earth type has substantial effect on the performance of the system. In order to examine the effectiveness of a heating system for space heating that made use of a hemispherical tank, Yumrutas and Unsal [17] created a computer model. The kind of soil surrounding the storage, the type of collector, the angle of the collector, the contact area of the collector, and the size of the storage were all studied for their effects on the system. According to their research, the highest water temperatures were recorded in September and November, whilst the lowest temperatures were recorded in March and April. They also found that granite provided the lowest water temperature, while sand soil produced the uppermost storage temperature but the least efficiency for the solar collectors.

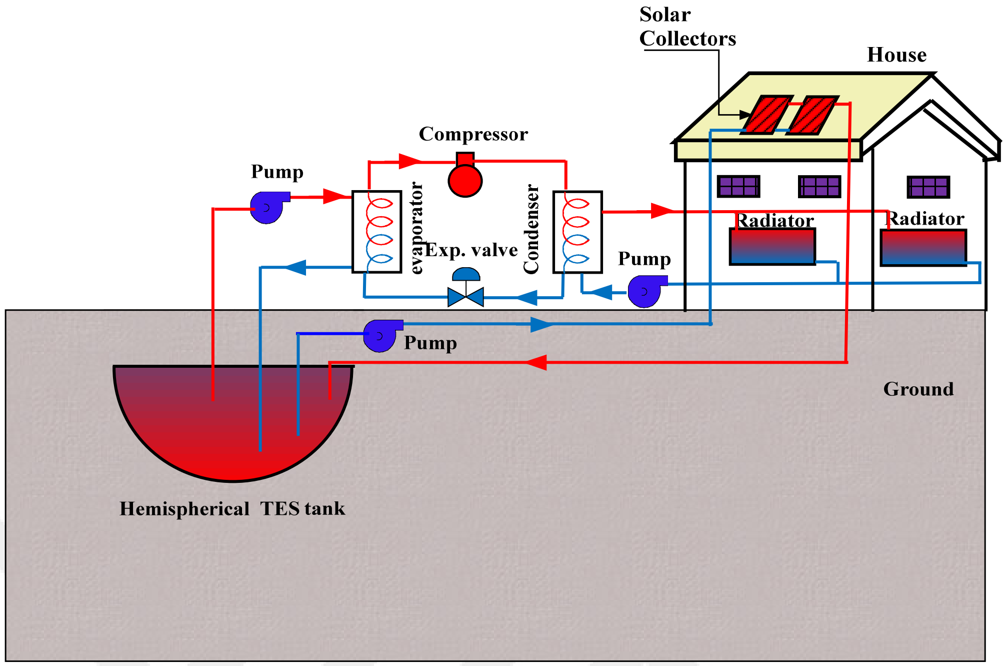

Figure 2.2 Heat Pump System with a Hemispherical Tank [17]

Sarbu and Sebarchievici [18] made an assessment of ground source heat pump (GSHP) systems, which are employed for both cooling and heating in buildings. They talked about the various hybrid ground-coupled heat pump (GCHP) systems that are frequently utilized in buildings for heating and cooling. Their analysis showed that adopting GSHP systems in both hot and cold climate regions has a large potential for energy savings. Yanshun Yu et al. [19] created a new combined system that comprises a ground-coupled heat pump and cooling storage in the soil. In the system, pipes are employed in both geothermal heat exchangers and cooling storage devices. Humid soil is used as the cooling storage medium. Energy is kept in the soil during after peak hours during cooling seasons and utilized for indoor cooling during peak hours. The system operates as a ground-coupled heat pump to provide air conditioning during other seasons. They found that key variables that affect the cycle performance of the integrated system include soil moisture content, the charging inlet temperature, pipe length, and tube diameter.

An analytical model for a solar-assisted heat pump (SAHP) system that includes a thermal energy storage (TES) tank for space heating was developed by Yumrutas and

Unsal [20]. They found that it takes this kind of heating system 5-7 years to achieve a periodic state. They also discovered that the collector area and tank volume were directly correlated with the water temperature.

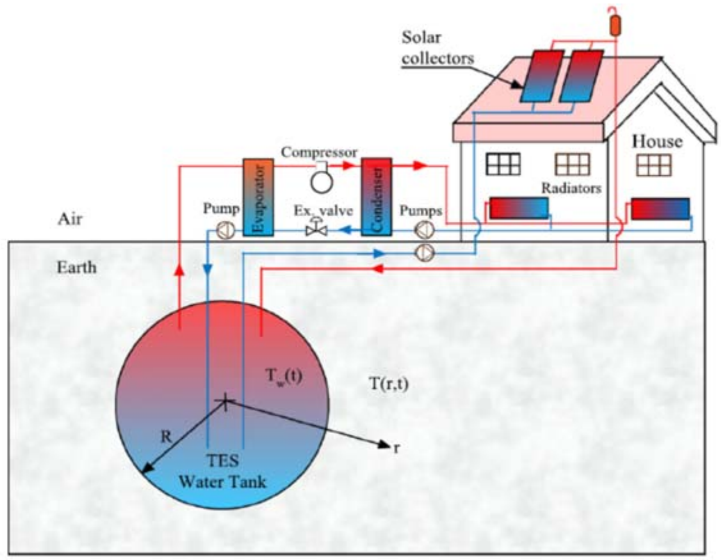

Figure 2.3 SAHP and TES Tank for House Heating [20]

Tutumlu and colleagues [21] researched the thermal efficiency of an ice rink cooling system that makes use of an underground TES tank. According to their research, the system must develop periodic functioning over a period of five to seven years.

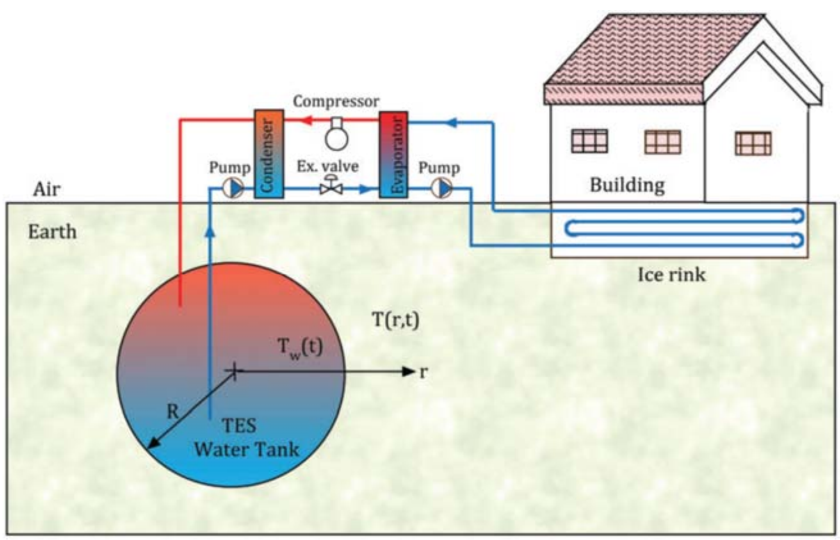

Figure 2.4 Cooling System with TES Tank for Ice Rink [21]

Khan et al. [22] discussed the use of TES tanks for storing thermal energy for cooling applications and provided an overview of the various types of TES systems that can be used. M. Ali and S. Riffat [23] presented a study on the performance of a hybrid cooling system that utilizes a TES tank and compared it to a conventional cooling system. According to their study, they found that the TES tank allowed the system to shift its cooling load from peak demand periods to off-peak periods, when electricity prices are lower, thus reducing energy consumptions. C. E. Uraz and B. Yildiz [24] described the use of a genetic algorithm to optimize the operation of a cooling system with a TES tank. M. Y. Ghazvini et al. [25] performed a numerical and experimental investigation of a cooling system with a TES tank and analyzed the effects of various parameters on the system's performance. The cooling load profile, the type and dimensions of the TES tank, as well as the control method employed, are only a few of the variables that the authors observed affect the optimal layout and operation of a cooling system with a TES tank. Al-Abidi et al. [26] presented a review of thermal energy storage for air conditioning systems and the applications of phase change materials.

Soylu et al. [27] investigated the effects and effectiveness of evaporative cooling on air-cooled chiller systems; their findings illustrate that the COP of the system increased by reducing the temperature of the air-cooled condenser's inlet air by cooling the air before it enters the condenser. However, it is also evident that it was more difficult to cool the air when relative humidity of the outside air increased, leading to a decrease in the COP of the system. Thus, evaporative cooling is most effective in hot and dry climates when the effectiveness of the evaporative cooler is highest to provide a useful solution to decrease condenser inlet air temperature. Yang et. al. [28] performed an analytical and experimental investigation on the application of evaporative pre-cooling with water mist for air-cooled chillers to enhance the chiller plant effectiveness. The inlet condenser air temperature using water mist pre-cooling could decrease by up to 9.4 K from the ambient air temperature, according to their experimental findings. The condensing temperature could be reduced by up to 7.2 K thanks to the pre-cooled condenser air, and the chiller coefficient of performance (COP) could be enhanced to varied degrees by up to 18.6%.

Most if not all of the previous studies done in literature on cooling systems, especially those integrating underground TES tanks, have been applied to air-conditioning of houses and building sectors. There seems to be a gap or a lack of study on the application of ground coupled cooling systems in food and beverage industries, especially in the CSD industry. Accordingly, this paper aims to investigate such a system so that more energy efficient and environmentally friendly Carbonated Soft Drink Cooling System is developed. In this paper, using a ground connected chiller and a spherical underground thermal energy storage tank, this study's computational model estimates the annual periodic performance of a carbonated soft drink (CSD) cooling system for a real case. Firstly, to determine the temperature distribution outside the tank, the transient heat transfer expressions will be formulated and solved. The cooling unit's formulation will then be created. For various soil, chiller, and storage tank properties, the suggested model will be created and utilized to calculate variations in the water temperature in the tank over the course of a full year. Investigations will also be conducted into how different design and operational parameters affect ground temperature and system COP.

CHAPTER 3
MATHEMATICAL MODELING

3.1 Introduction

The focus of this chapter is the construction of a mathematical model for the ground-coupled cooling system that includes a thermal energy storage (TES) tank. The performance of the system over the long run is predicted by the model.

To find the performance parameters of the cooling system over a period of ten years, including hourly, monthly, and yearly parameters, a model incorporating Duhamel's superposition technique is developed for the TES tank as well as several expressions for other components of the system.

3.2 Description of the Cooling System

A refrigeration unit and an underground thermal energy storage tank are used in the cooling system of the carbonated soft drink facility to chill the syrup room. The refrigeration unit is in charge of taking heat from the syrup room and rejecting it to the thermal energy storage tank. It operates on a refrigeration cycle based on vapor compression. The underground thermal energy storage tank serves as a heat sink and is used to store sensible heat. These assumptions are made for the tank: spherical in shape, water-filled, and situated far below the earth's surface. The deep ground temperature, which is supposed to remain constant throughout the year, is used as the tank's initial temperature. An image of the syrup room is shown in figure 3.1 below:

Figure 3.1 Photograph of Syrup Storage Room

The system under study is represented schematically in figure 3.2 and consists of a chiller that uses a vapor compression refrigeration cycle, a spherical underground thermal energy storage (TES) tank, and the syrup room that needs to be cooled. The refrigeration cycle uses the TES tank as a heat sink. Heat is transferred to water in the TES tank by the refrigerant as it captivates heat from the cooled space in the evaporator, compresses in the compressor, and releases heat. It is assumed that the chiller is a water-to-water type, in which, heat is exhausted to the water in the condenser while water circulates in the evaporator inside its fan-coil unit. Because the TES tank's water temperature is lower than the surrounding air's temperature and experiences fewer temperature changes than an air source system, the current ground-coupled cooling system is anticipated to have greater values of COP than air source systems. The system is considered to be installed in Erbil-Iraq.

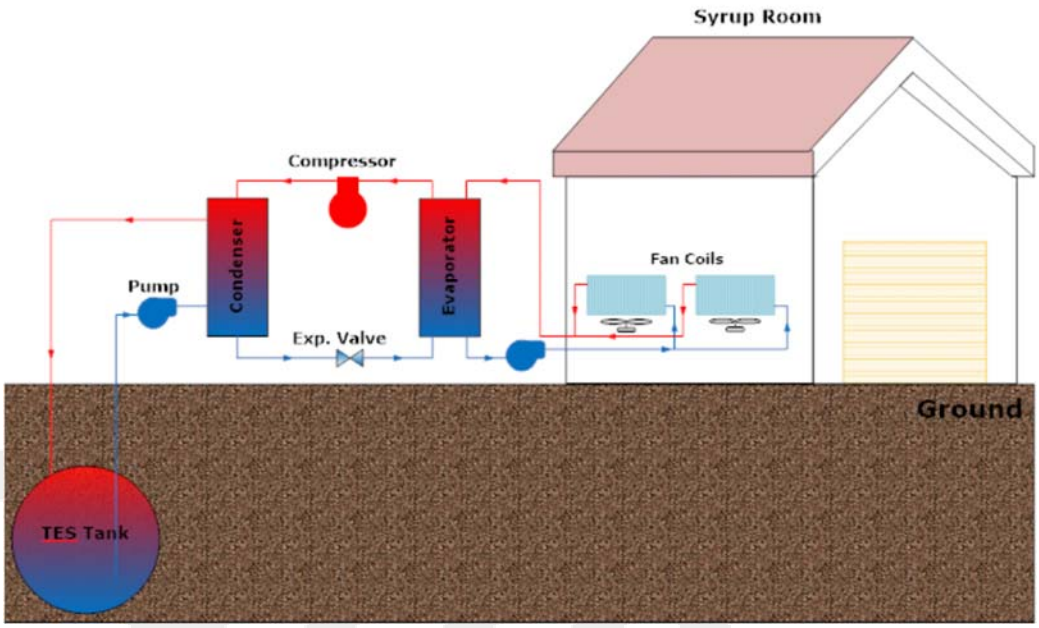

Figure 3.2 Schematic of the Cooling System with Underground Spherical TES Tank

3.3 Mathematical Modeling of The Cooling System with TES Tank

The parameters of the cooling system are input into a computer program to calculate the coefficient of performance of the chiller, the temperature distribution in the earth surrounding the tank, and the transient temperature of the water inside the thermal energy storage tank. The MATLAB-written application is built on the analytical model of the system. The system documentation includes a list of the physical and thermal characteristics of the materials utilized in the system, including the ground around the tank. The volume of the TES tank, the combination of the heat overall transfer coefficient and the area of the syrup room, and the Carnot factor are some of the design factors for the cooling system. During operation, the system is intended to keep the syrup room at a set temperature of 25°C, and the regulation system modifies the cooling rate in accordance with temperature measurements. Overall, it is anticipated that the cooling system will deliver effective and dependable cooling for the syrup room, ensuring the nutritional soft drink product's quality.

3.3.1 Solution of the Transient Temperature Distribution Outside the TES Tank

The TES tank is presumed to be spherical, water filled, fully mixed, and buried, at a spatially aggregated time-varying temperature $T_w(t)$. At the beginning, the tank is at the deep ground temperature T_∞. The soil is thought to have a uniform composition and stable thermal characteristics. The periodic transient heat transfer issue outside the spherical TES tank can be written as a one-dimensional partial differential equation with boundary and initial conditions using the spherical coordinate system:

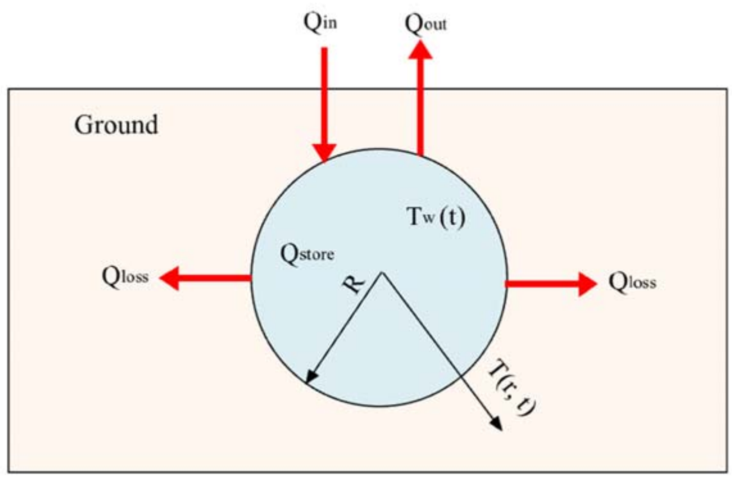

Figure 3.3 Schematic of the Energy Balance around the TES tank [16]

$$\frac{\partial^2 T}{\partial r^2} + \frac{2}{r}\frac{\partial T}{\partial r} = \frac{1}{\alpha}\frac{\partial T}{\partial t} \tag{3.1}$$

$$T(R.t) = T_w(t) \tag{3.2}$$

$$T(\infty.t) = T_\infty \tag{3.3}$$

$$T(r.0) = T_\infty \tag{3.4}$$

The heat that is lost from the tank by conduction to the earth covering it and the energy gained in the tank are equal to the energy supplied to the tank. The equation below can be used to express this relationship:

$$Q = \rho_w c_w V_w \frac{dT_w}{dt} - kA\frac{\partial T}{\partial r}(R.t) \tag{3.5}$$

where ρ_w, c_w, and V_w stand for the density, specific heat, and volume of the water in the tank, respectively, and k, R, and A stand for stand for the thermal conductivity of the soil, radius, and surface area of the tank, respectively.

The following variables can be used to define the transient heat transport issue outlined by equations (1) through (5) in a dimensionless form:

$$x = \frac{r}{R} \qquad \tau = \frac{\alpha t}{R^2} \qquad q = \frac{Q}{4\pi R k T_\infty} \qquad p = \frac{\rho_w\, c_w}{3\rho c}$$

$$\emptyset = \frac{T - T_\infty}{T_\infty} \qquad \emptyset_w = \frac{T_{w-}T_\infty}{T_\infty} \qquad \emptyset_a = \frac{T_a - T_\infty}{T_\infty} \tag{3.6}$$

the variables x, τ, $\emptyset$, and q in the equations denote the dimensionless parameters of radial distance, time duration, temperature, and net energy transmitted to the tank, respectively. Moreover, r and c stand for the density and the specific heat of the ground, correspondingly. The letters "w" and "a" denote water and surrounding air.

The following expressions can be generated by substituting these equations into the expressions given in Equs. (3.1) through (3.6):

$$\frac{\partial^2 \emptyset}{\partial x^2} + \frac{2}{x}\frac{\partial \emptyset}{\partial x} = \frac{\partial \emptyset}{\emptyset \tau} \tag{3.7}$$

$$\emptyset(x.\tau) = \emptyset_w(\tau) \tag{3.8}$$

$$\emptyset(\infty.\tau) = 0 \tag{3.9}$$

$$T(x.0) = 0 \tag{3.10}$$

$$q = P\frac{d\emptyset_w}{d\tau} - \frac{\partial \emptyset}{\partial x}(1.\tau) \tag{3.11}$$

Using the transformation shown below:

$$\psi(x.\tau) = x\emptyset(x.\tau) \tag{3.12}$$

This problem may be simplified as:

$$\frac{\partial^2 \psi}{\partial x^2} = \frac{\partial \psi}{\partial \tau} \tag{3.13}$$

$$\psi(1.\tau) = \emptyset_w(\tau) \tag{3.14}$$

$$\psi(\infty.\tau) = 0 \tag{3.15}$$

$$\psi(x.0) = 0 \tag{3.16}$$

$$q = p\frac{d\emptyset_w}{d\tau} - \frac{\partial\emptyset}{\partial x}(1.\tau) \qquad (3.17)$$

Implementing this transformation $\boldsymbol{\eta}$

$$\eta = \frac{x-1}{2\sqrt{\tau}} \qquad (3.18)$$

A solution for constant $\emptyset$ can be achieved:

$$\psi(x.\tau) = \emptyset_0\left[1 - erf\left(\frac{x-1}{2\sqrt{\tau}}\right)\right] \qquad (3.19)$$

Using Duhamel's superposition approach, the following unsteady temperature variation in the soil surrounding the TES tank is obtained:

$$\psi(x.\tau) = \emptyset_w(0)\left\{1 - erf\left(\frac{x-1}{2\sqrt{\tau}}\right)\right\} + \int_0^\tau \frac{d\emptyset_w(\xi)}{d\xi}\left\{1 - erf\left(\frac{x-1}{2\sqrt{\tau}}\right)\right\} d\xi \quad (3.20)$$

Using Equ. (3.17) and differentiating $\emptyset$ with regards to x, at x=1, we obtain the following equation:

$$q = P\frac{d\emptyset_w}{d\tau} + \emptyset_w(\tau) + \int_0^\tau \frac{d\emptyset_w(\xi)}{d\xi}\frac{d\xi}{\sqrt{\pi(\tau-\xi)}} \qquad (3.21)$$

By discretizing Equ. (3.21) and solving in dimensionless form for the tank's water temperature during the nth time interval, we get:

$$\phi_w(\tau_n) = \frac{q(\tau_n) + \left[\frac{P}{\Delta\tau} + \frac{1}{\sqrt{\pi\Delta\tau}}\right]\emptyset_w(\tau_{n-1}) - \sum_{i=1}^{n-2}\frac{\emptyset_w(\tau_{i+1}) - \emptyset_w(\tau_i)}{\sqrt{\pi\Delta\tau(n-i)}}}{1 + \frac{P}{\Delta\tau} + \frac{1}{\sqrt{\pi\Delta\tau}}} \qquad (3.22)$$

To determine the hourly dimensionless water temperature, Equ. (3.22) is used, where q(τ) is the tank's total dimensionless heat flux rate. The dimensionless compressor work and cooling load, represented as q(τ), determine the amount of heat gain to the TES tank.

$$q(\tau) = \frac{w(\tau)}{\gamma} + q_L(\tau) \qquad (3.23)$$

where $q_L(\tau)$ is the dimensionless heat extracted by the evaporator for the cooling unit, $w(\tau)$ is the dimensionless compressor work input, and γ is a dimensionless parameter.

3.3.2 Modeling of the Chiller Unit

The evaporator, compressor, condenser, and expansion valve are the four parts that make up the refrigeration process that powers the cooling system. The condenser transfers heat to the TES tank, while the evaporator takes heat from the syrup room to be cooled. The chiller must completely absorb all of the heat the storage generates in order for the cooling load to be represented as:

$$Q_L(t) = (UA)_H(T_a - T_i) \tag{3.24}$$

where Ta is the mean of the highest design temperature in summer, and Ti is the interior design temperature, and (UA_H) is the product of the total heat transfer coefficient for the storage and the heat transfer area.

The fan-coil unit's specifications can also be used to express the cooling load as follows:

$$Q_L(t) = (UA)_{FC}(T_i - T_c) \tag{3.25}$$

Tc is the average water temperature inside the fan-coil unit, and (UA_{FC}) is the multiplication of the cooling system's overall heat transfer coefficient by the heat transfer area.

The chiller's COP is expressed by:

$$COP = \frac{Q_L(t)}{W(t)} = \frac{Q_L(t)}{Q_H(t) - Q_L(t)} \tag{3.26}$$

where $Q_H(t)$ is the heat discharged to the TES tank and W is the work required by the compressor.

$$COP_{carnot} = \frac{T_L}{T_H - T_L} \tag{3.27}$$

$$\beta = \frac{COP}{COP_c} \tag{3.28}$$

As a function of Carnot factor, COP may be given as:

$$COP = \beta \frac{T_c}{T_w - T_c} \tag{3.29}$$

where β is the Carnot factor, which ranges from 0 to 1. Tc can be solved for by combining Equ. (3.24) and Equ. (3.25) as follows:

$$(UA)_H(T_a(t) - T_i) = (UA)_{HE}(T_i - T_c) \tag{3.30}$$

$$\frac{(UA)_H}{(UA)_{HE}} = \frac{T_i - T_c}{T_a - T_i} = u \tag{3.31}$$

$$T_c = u(T_i - T_a) + T_i \tag{3.32}$$

Substituting Equ. (3.32) into Equ. (3.29) results in:

$$COP = \beta \left\{ \frac{u(T_i - T_a) + T_i}{u(T_a - T_i) + T_w - T_i} \right\} \tag{3.33}$$

Using the dimensionless parameters provided in Equ. (3.6), we get:

$$COP = \beta \left[\frac{u(\phi_i - \phi_a(\tau)) + \phi_i + 1}{u(\phi_a(\tau) - \phi_i) + \phi_w(\tau) - \phi_i} \right] \tag{3.34}$$

The compressor's work may be expressed as:

$$W(t) = \frac{Q_L(t)}{COP} = \frac{(UA)_H(T_a(t) - T_i)}{COP} \tag{3.35}$$

By plugging Equ. (3.33) into Equ. (3.35) and converting the temperatures into dimensionless form, we get:

$$w(t) = \frac{W(t)}{(UA)_H T_\infty} = \frac{(\phi_a(t) - \phi_i)\,.\,[u(\phi_a(t) - \phi_i) + \phi_w(t) - \phi_i]}{\beta\left[u\left(\phi_i - \phi_a(t)\right) + \phi_i + 1\right]} \tag{3.36}$$

3.3.3 Modeling of the Syrup Room

To obtain the overall heat transfer coefficient and calculate the cooling load for the syrup storage room, the following equations are used:

$$\dot{Q}_{(\tau)} = U * A * (T_o(t) - T_i) \tag{3.37}$$

Where $\dot{Q}_{(\tau)}$, U, A, $T_o(t)$, and T_i are the heat gain (W/h), overall heat transfer coefficient (W/m^2.°C), surface area of the cold storage room (m^2), outside temperature of atmosphere air (°C), and inside temperature of cold storage air (°C), respectively.

To find the overall heat transfer coefficient for the syrup room, the following equations and data are used:

$$U = \frac{1}{R_{total}} \tag{3.38}$$

$$R_{total} = \frac{1}{h_i} + \frac{X_1}{K_{Al}} + \frac{X_2}{K_{Foam}} + \frac{X_3}{K_{Al}} + \frac{1}{h_o} \tag{3.39}$$

R_{total}, h_i, X_1, K_{Al}, X_2, K_{Foam}, X_3, and h_o are overall thermal resistance, convective heat transfer coefficient of inside air, aluminum plate thickness, thermal conductivity of aluminum, polyisocyanurate foam thickness, thermal conductivity of the foam, aluminum plate thickness, and convective heat transfer coefficient of outside air, respectively.

Table 3.1 Thickness of the materials used in the Syrup Room wall and Roof

Material	Thickness (m)
X_1= Aluminium Plate	0.0004
X_2= Polyisocyanurate Foam	0.0492
X_3= Aluminium Plate	0.0004

Table 3.2 Thermal conductivity of the materials used in the Syrup Room wall

Material	Thermal Conductivity ($W/m.K$)
Aluminium	237
Polyisocyanurate Foam	0.023

Table 3.3 Convection heat transfer coefficient of air

Position	Convection heat transfer coefficient (W$/m^2.K$)
h_i = Inside air	9.37
h_o = Outside air	22.4

It is assumed in this study that the solar air temperature is neglected because the syrup room walls are not exposed to sunlight (solar radiations), and the roof is covered with a tent to prevent solar heat gain. The heat that enters the refrigerated compartment through its surface is known as the transmission load. The size of the syrup room is 41 m length x 9.75 m width x 12 m height.

The overall area of the walls and roof is equal to: 1617.75 m^2

CHAPTER 4
VALIDATION AND COMPUTATIONAL PROCEDURES

4.1 Introduction

After investigating of the literature review, it is apparent that there were no available research pertaining to a ground coupled cooling system with an underground storage tank using Duhamel's superposition and nondimensional correlations. Hence, a new MATLAB program was developed, and the research by [14] and [10] were selected for validation. Figures in section 5.2 of the report include the results of the new program, which are contrasted with those from [14] and [10].

4.2 Validation Procedure for the Study [14]

An important component of any theoretical work is validation. If there are any experimental study that is comparable to the theoretical study, the outcomes of the experimental research should be compared to those from theory. When no similar theoretical or practical research are available in the literature, [14] can still be used to validate particular parts for space cooling. As a result, the following analyses were carried out to confirm the conclusions drawn from a MATLAB program that used the same model as [14].

The hourly changes in water temperature within the TES tank were calculated using the MATLAB algorithm used to determine the outcomes of the modeling described in [14]. A similarity transformation technique was utilized to produce the model in [14], and the same model was replicated in this investigation using the MATLAB program to validate the same components. The following figures, alongside the corresponding figures from [14], show some of the outcomes from running the MATLAB program with similar input data:

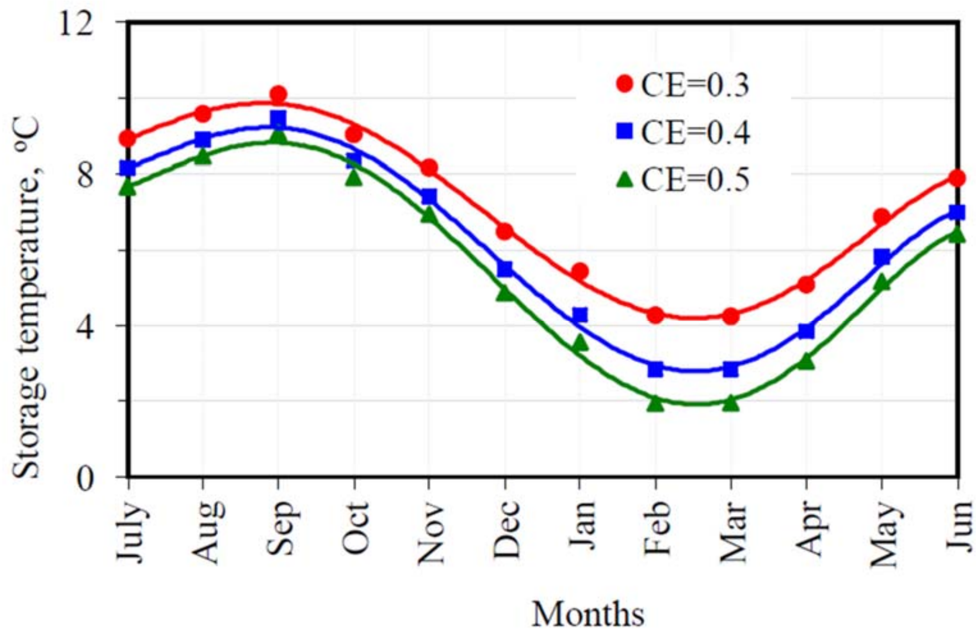

Figure 4.1 Effect of CE on the Storage Temperature during the Tenth Year [14]

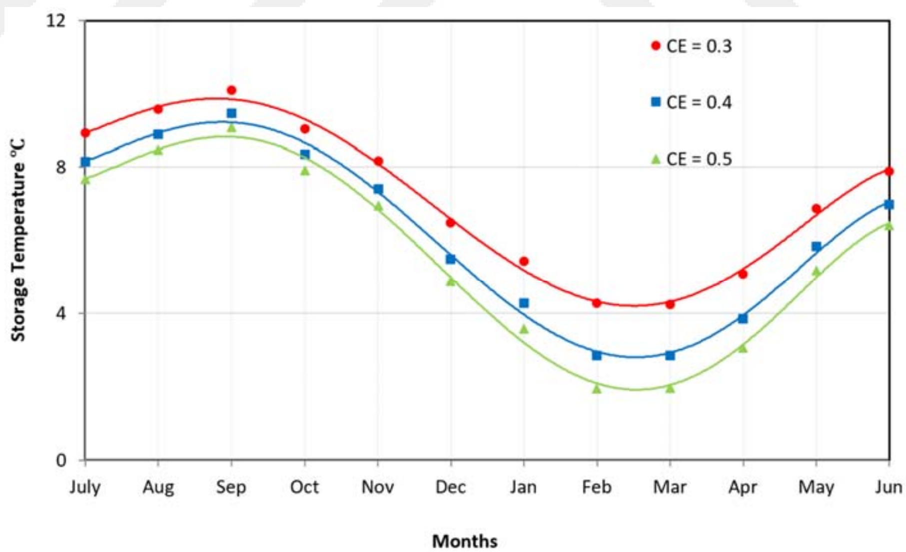

Figure 4.2 Effect of CE on the Storage Temperature during the Tenth Year [Validation]

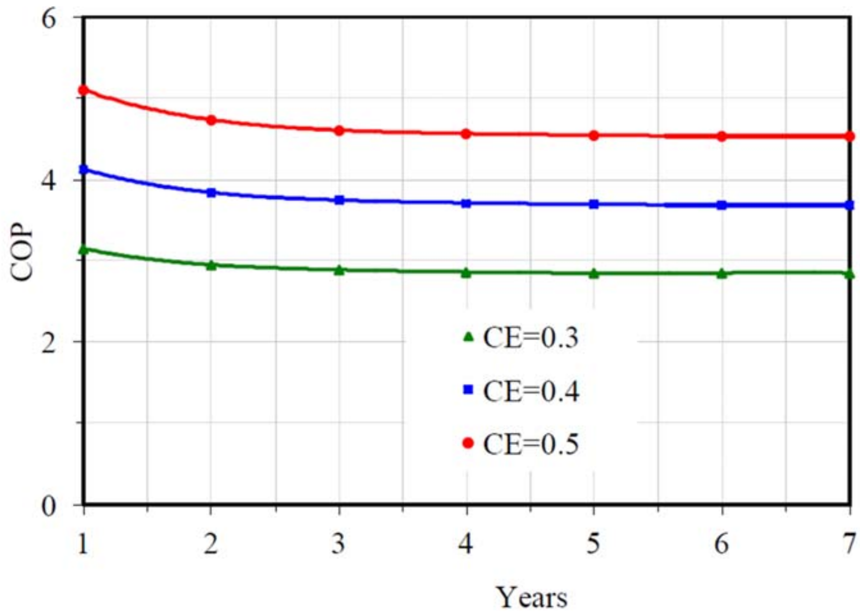

Figure 4.3 Effect on CE on the Heat Pump COP during the Tenth Year [14]

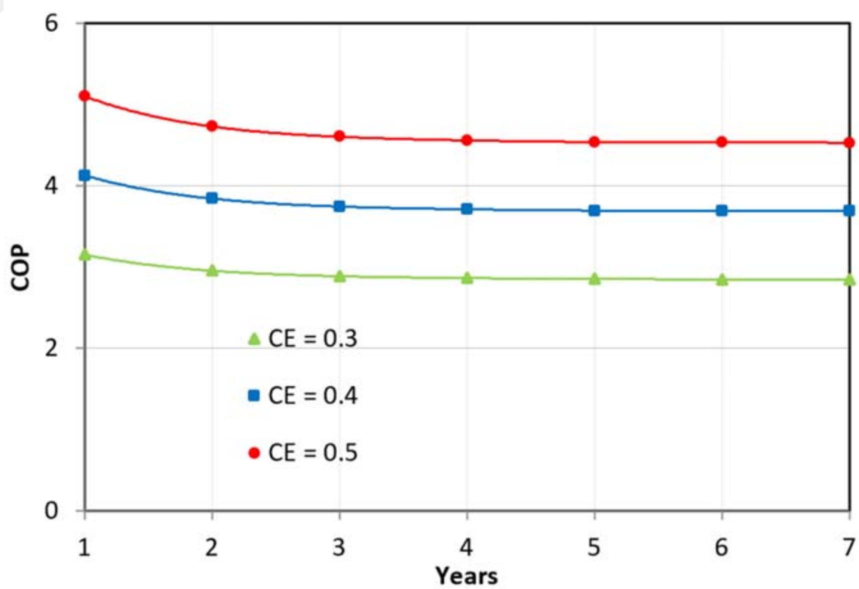

Figure 4.4 Effect on CE on the Heat Pump COP during the Tenth Year [Validation]

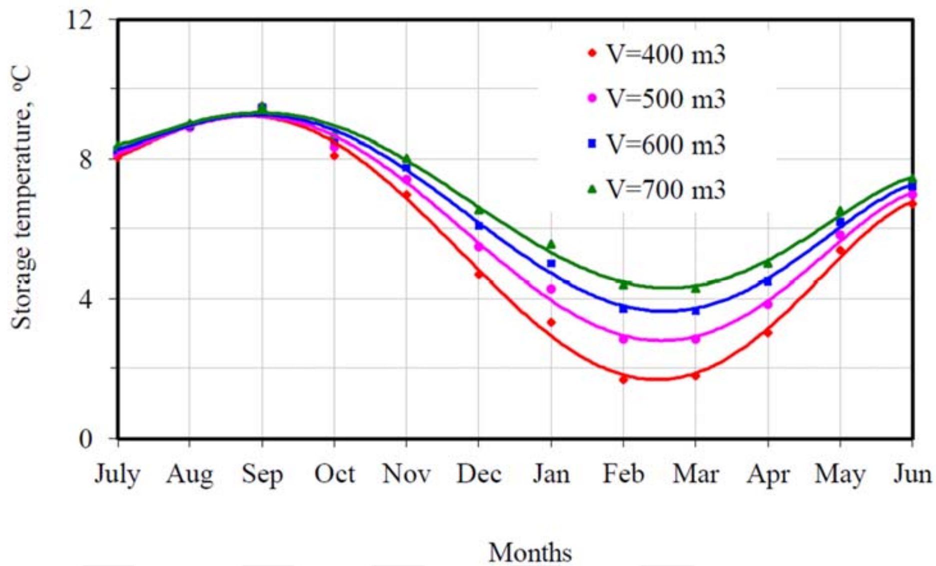

Figure 4.5 Variation of Annual Storage Temperature with Tank Volume during the Tenth Year [14]

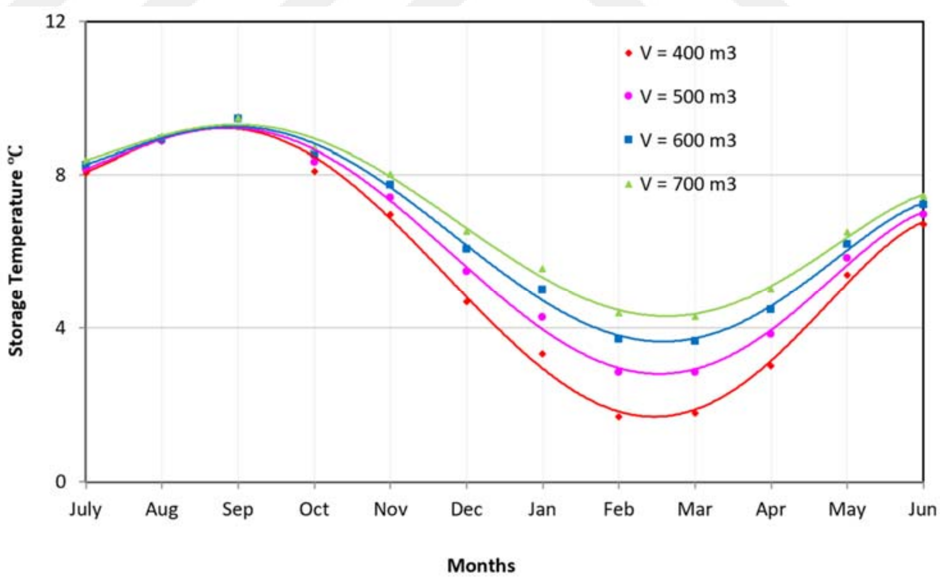

Figure 4.6 Variation of Annual Storage Temperature with Tank Volume during the Tenth Year [Validation]

Figures (4.1-4.6) indicate that the MATLAB program's validation results for the model stated in [14] are reliable. As a result, the models for the other components and calculations from the MATLAB program can be used in the cooling system of this research. The results of the study will be shown in chapter 6.

Some observations can be made by contrasting the outcomes of the MATLAB software utilized in this work with those shown in [14]. The following are these remarks:

- The results reported in figures (4.1, 4.3, and 4.5) in reference [14] correspond to the results from the validation figures (4.2, 4.4, and 4.6).
- The results of the validation are reliable and applicable to this study since the disparity between the validation results and the literature [14] is insignificant if there is any.
- This study's model is intended for cooling applications whereas the one in [14] is utilized for purposes of heating space.
- There are some differences in this study's input data and that used in [14].
- The MATLAB program code is given in appendix 1.

4.3 Validation Procedure for the Study [10]

Using the same logic outlined in section 4.2, it is possible to claim that no previous studies have been conducted on the performance of a ground coupled cooling system using Duhamel's superposition and transformation techniques. [10] studied a very similar system but solved the problem by means of complex finite Fourier transform (CFFT) technique. For validation and comparison, a few figures from reference [10] were chosen. The figures below compare the results from [10] with the verified results produced from a MATLAB program created for this investigation.

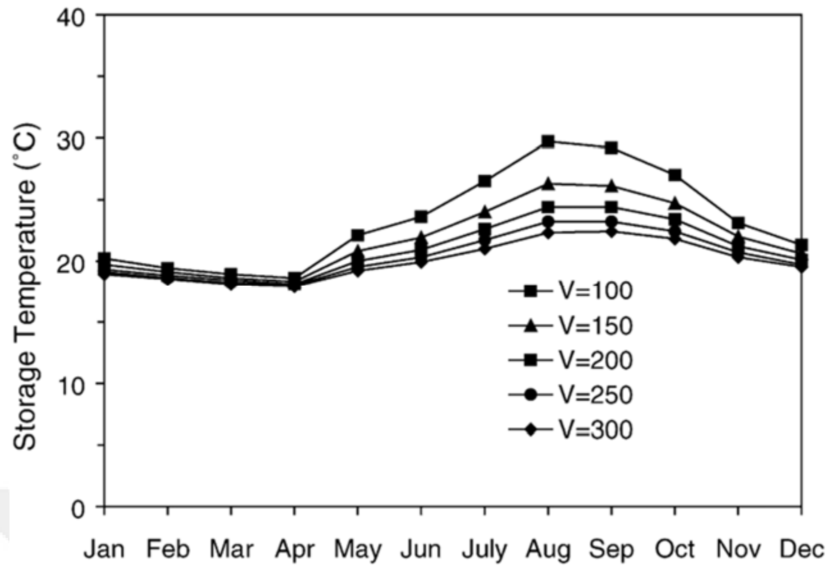

Figure 4.7 Annual Variation of Storage Tank Temperature for Different Tank Volumes. Earth is Granite [10]

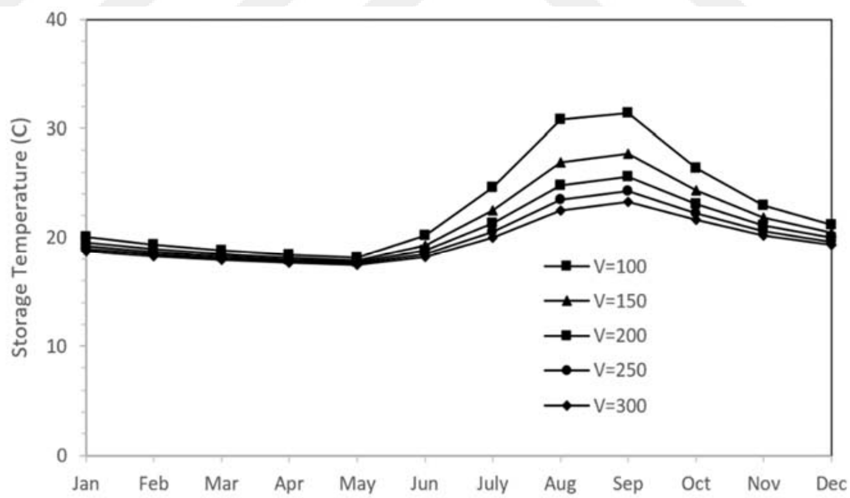

Figure 4.8 Annual Variation of Storage Tank Temperature for Different Tank Volumes. Earth is Granite [Validation]

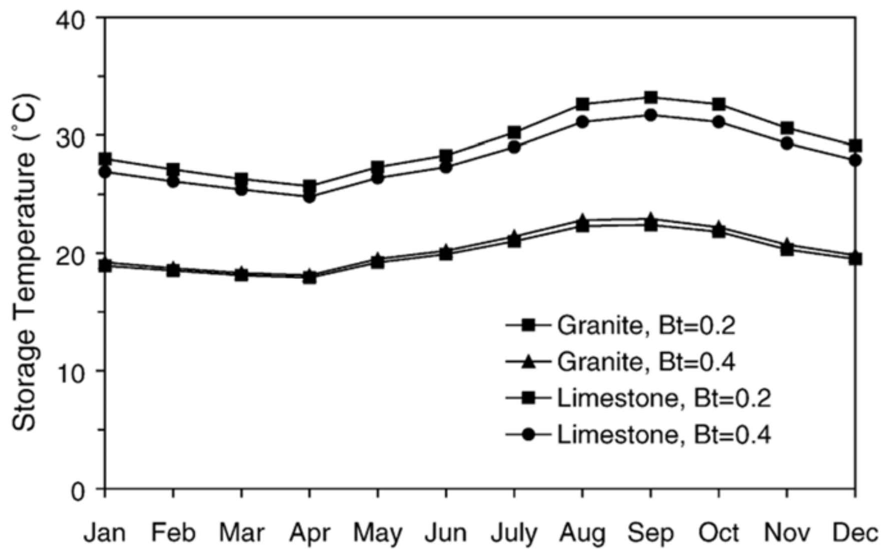

Figure 4.9 Annual Variation of Storage Tank Temperature for Different Earth Types and Carnot Factors [10]

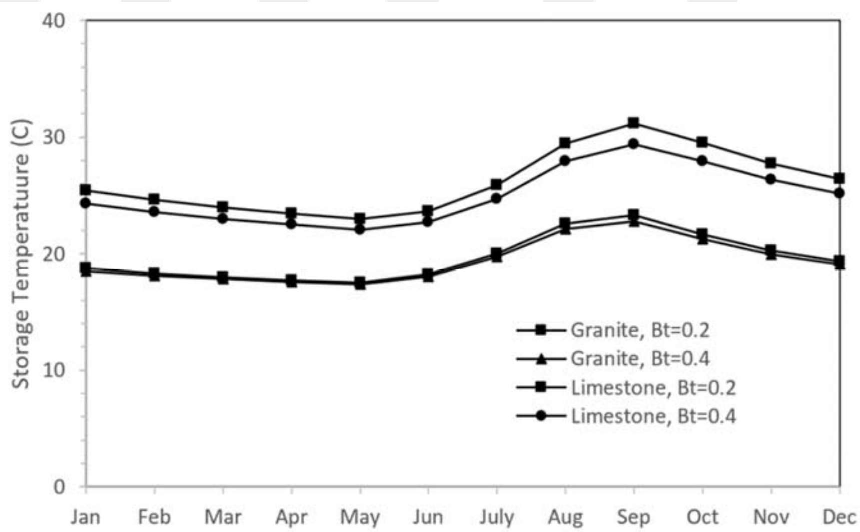

Figure 4.10 Annual Variation of Storage Tank Temperature for Different Earth Types and Carnot Factors [Validation]

The results from the validation compared to the study in [10] are very similar but do not seem exact. While the shapes and behaviors of the curves are the same, the values seem to differ a little. This could be because of two possible reasons:

- While the current study used the method of Duhamel's superposition and transformation techniques, [10] used the complex finite Fourier transform (CFFT) technique.
- It is not specified in [10] which months are considered for operation. Thus, the operating months were assumed in the validation program.
- The MATLAB program code is given in appendix 2.

4.4 Input Data and Computational Procedure

In chapter 3, an analytical model was established, and a MATLAB computer program was made based on this model for numerical computations. The program is used to compute the hourly COP of the chiller as well as the temperature distribution in the soil around the TES tank, and transient temperature of water in the storage tank.

Table 4.1 provides a list of the thermal and physical characteristics of three different types of soil: limestone, clay, and granite. The starting storage temperature is considered to be 15°C, the same as the deep earth temperature at the beginning of the calculations, and the indoor design air temperature during summer is assumed to be 25°C. With the Photovoltaic Geographical Information System (PVGIS) Online Tool, the average hourly outdoor air temperature may be determined. The average value obtained for each month represents the highest ambient air temperature. The sum of the overall heat transfer coefficient and the area of the syrup room (UA_H) is determined to be 705 W/°C, with the summer cooling load of being roughly 15 kW.

Table 4.1 Properties of the Soil surrounding the TES Tank [29]

Earth Type	Thermal Conductivity (W/mK)	Thermal Diffusivity (m^2/s)	Specific Heat (J/kgK)	Density (kg/m^3)
Limestone	1.3	5.75×10^{-7}	900	2500
Clay	1.4	1.10×10^{-6}	880	1500
Granite	3	1.40×10^{-6}	820	2640

Based on design conditions, the u value in Equs. (3.34) and (3.36) is calculated as 1.2. Zogou and Stamatelos [13] noted that small electric heat pumps typically have CF ratings between 0.30 and 0.50. Three CF values (0.30, 0.40, and 0.50) were consequently used in this research. Unless otherwise stated, a CF value of 0.4 and a tank volume of 400 m^3 are used for all computations during the fifth year. The water in the TES tank is considered to be evenly mixed. The water has a specific heat of 4.18 $kJ/kg°C$ and a density of 1000 kg/m^3, respectively. The water is thought to be at a temperature of 15 ℃, the temperature of the deep ground, at the beginning of the system's operation. Three types of soil are taken into account in the study: limestone, clay, and granite, which evaluates the effect of soil type on system performance. It is also important to remember that in this study, the solar air temperature is not considered because the syrup room walls are not exposed to sunlight (solar radiations), and the roof is covered with a tent to prevent solar heat gain.

CHAPTER 5
RESULTS AND DISCUSIONS

5.1 Introduction

This chapter displays and analyzes the outcomes of the simulation for a ground-coupled, underground thermal energy storage tank-equipped carbonated soft drink cooling system. The results of the simulation computations of the modeling covered in chapter three are shown and analyzed in this section. The results are depicted in the figures, and their significance are covered. The goal of this inquiry is to evaluate the system's performance and find any patterns that might develop during a 10 year periodic operation.

5.2 Results

Due to different thermophysical qualities, the performance parameters of each earth surrounding the TES tank change. The TES tank's water temperature variation is one of these performance criteria. Therefore, the temperature of the storage affects every performance parameter. Figure 5.1 displays this concept by showing the variations in water storage temperature during the tenth year for three different earth types.

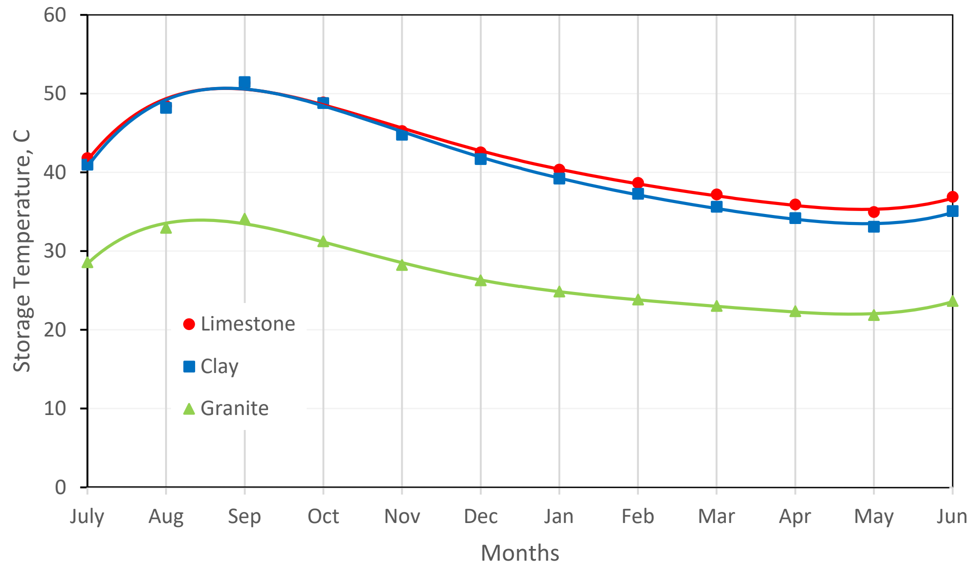

Figure 5.1 Effect of Soil Type on Water Temperature Variation in the TES Tank during Tenth Year

These properties have led to the observation that materials with lesser thermal conductivity and diffusivity, like limestone soil, inhibit the transfer and dispersion of heat from the TES (thermal energy storage) tank to the ground around the tank. Figure 5.2 displays the effect of soil type on the COP for the cooling months. It can be observed from the figure that the COP gradually decreases from June to September for all earth types because the temperature progressively rises to reach the highest values by the end of August or in September. It is also evident from this figure that granite, once again, is the most efficient type of soil mainly due to its higher heat capacity.

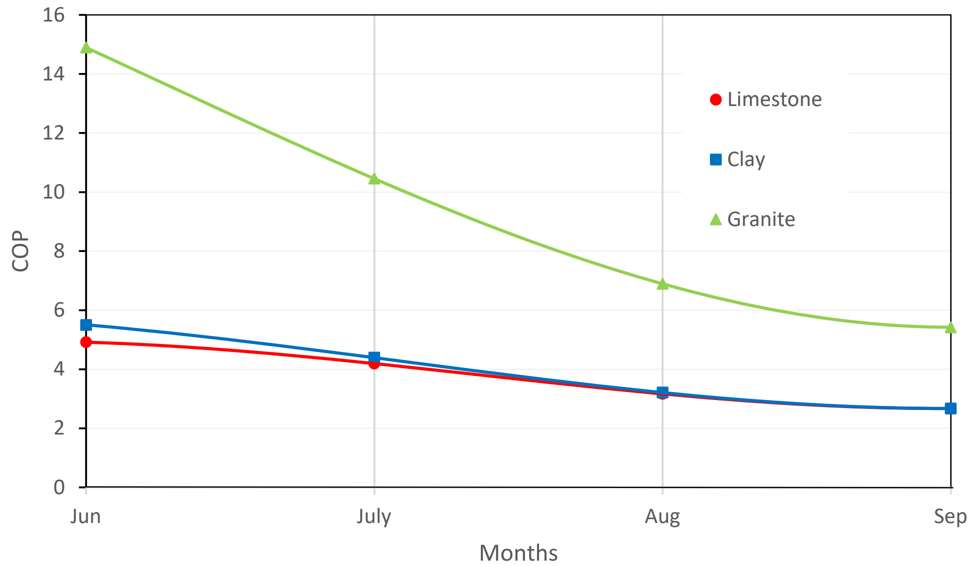

Figure 5.2 Effect of Soil Type on COP during the Fifth Year for Cooling Months

It is evident from these figures that granite is the most efficient type of soil for the cooling system. This is mainly due to its thermos-physical properties of having higher thermal conductivity and thermal diffusivity, allowing faster heat rejection into the surrounding earth. It should be noted, however, that the soil type for the real case at the CSD production plant location is clay, which operates with reasonable values and is used in most of the following figures.

Figure 5.3 shows how the storage temperature in the TES tank is affected by the size or the volume of the tank. The annual oscillations in the TES tank's water temperature with respect to tank volume is illustrated. The highest storage temperatures are found at the end of the summer season in the smallest tank sizes. By the end of August, the storage temperature rises to about 45 °C when the storage capacity is 200 m^3.

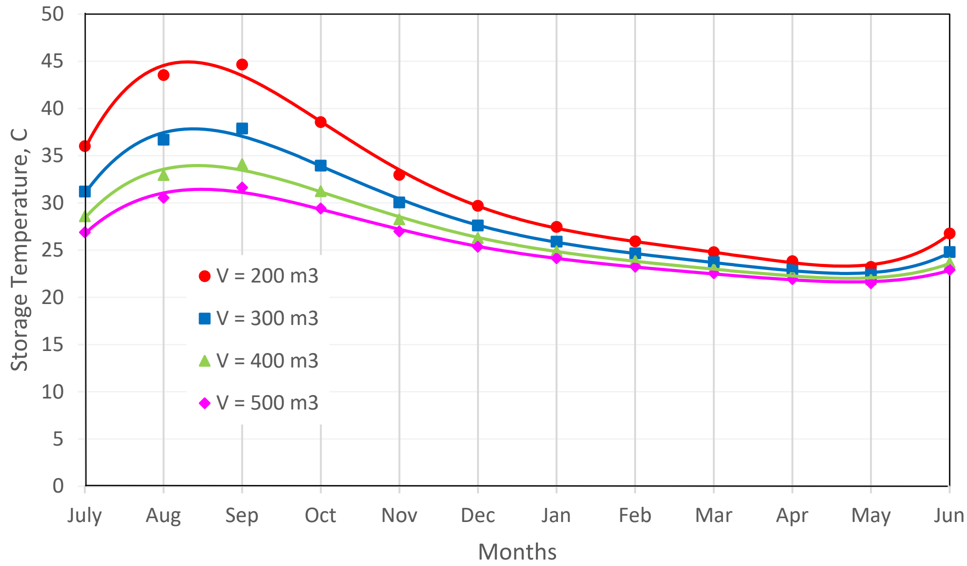

Figure 5.3 Annual Temperature Variation of Storage Water with Tank Volume in the Fifth Year

Figure 5.4 shows the effects of CF values on the storage temperature on a monthly basis. Three different values for CF are considered based on literature.

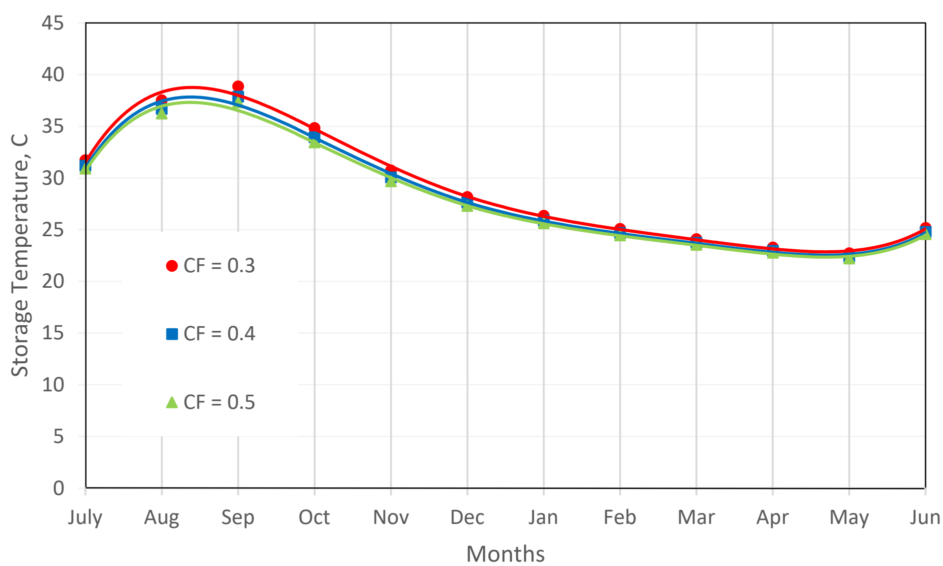

Figure 5.4 Temperature Variations in TES Tank for Different CF Values

Although the CF values do not have a huge impact on water temperature variations in the storage, the COP is significantly influenced by CE, as can be seen in the following figures, with greater CE values translating into a higher COP. Figures 5.5 and 5.6 show the effects of CF on the COP on a monthly and yearly basis, respectively.

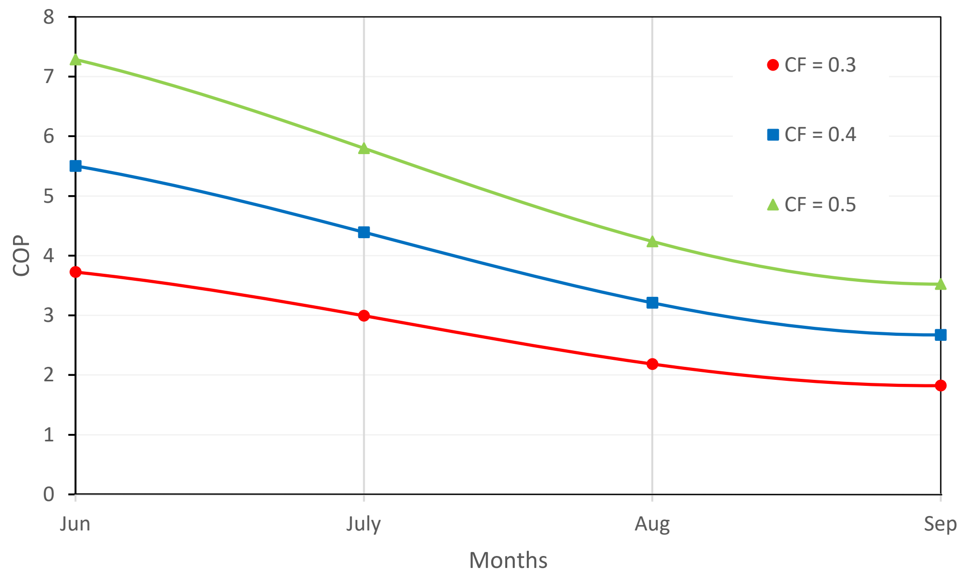

Figure 5.5 Effect of Carnot Factor (CF) on the Refrigeration System COP for Cooling Months

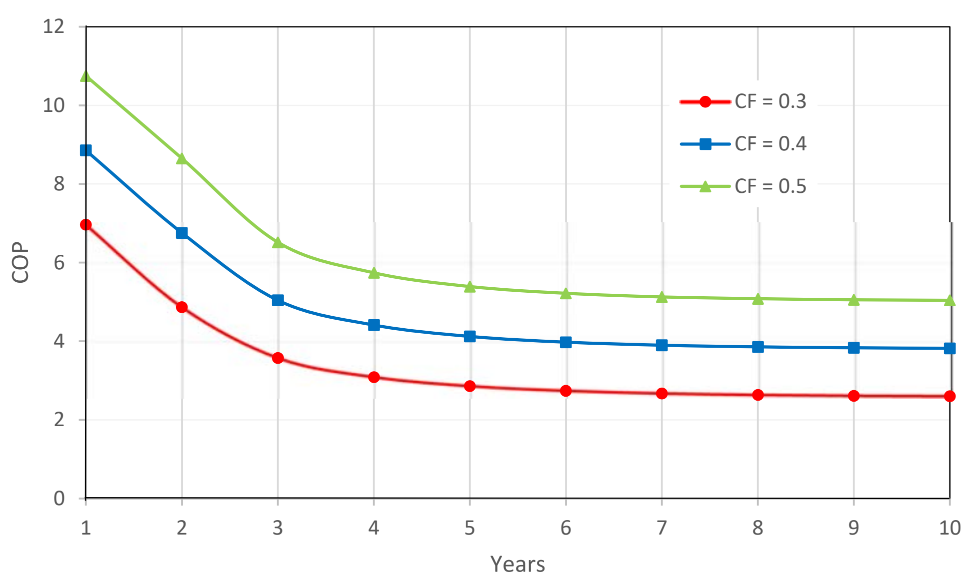

Figure 5.6 Effect of CF on COP with Operation Time

For a given cooling load, a higher Carnot factor results in less heat rejection to the tank (i.e. a higher COP), allowing for quicker heat dissipation into the soil and maintaining a lower storage temperature as a result. But with time, the COP falls until the refrigeration unit switches to an annual cycle of operation.

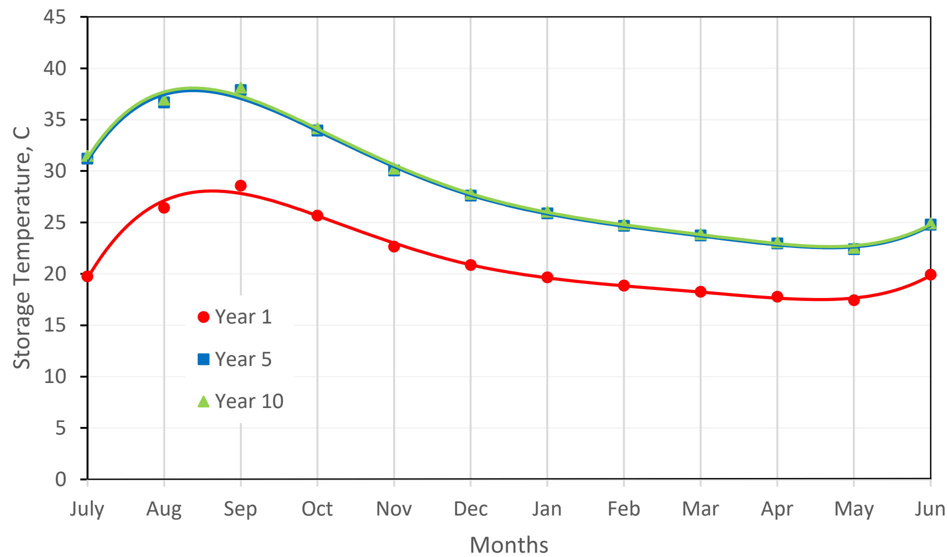

Figure 5.7 TES Tank Temperature Variations with Operation Time

The annual variations in the tank's water temperature are shown in Figure 5.7 throughout time. Rapid temperature swings occur in the tank, especially from the first year to fifth year of operation. Although the temperature steadily rises after the beginning years of operation until the fifth year of operation, it then stays almost constant. As a result, it may be said that the system transitions to an annually periodic regime from the fifth year onwards. As it can be seen, the values of years 5 and 10 are almost identical, meaning that the temperature does not undergo much change from the fifth year onward.

The effects of earth types are also depicted in figures 5.8 and 5.9 for the entire duration of operation on both storage temperature and COP of the refrigeration system. It can be seen from both figures that granite is superior to the other earth types, resulting in lower temperatures and higher COP. It can also be observed that although limestone starts off with lower temperature and higher COP than clay, its values go down with operation time when they stabilize after the fifth year, where clay is evidently more efficient.

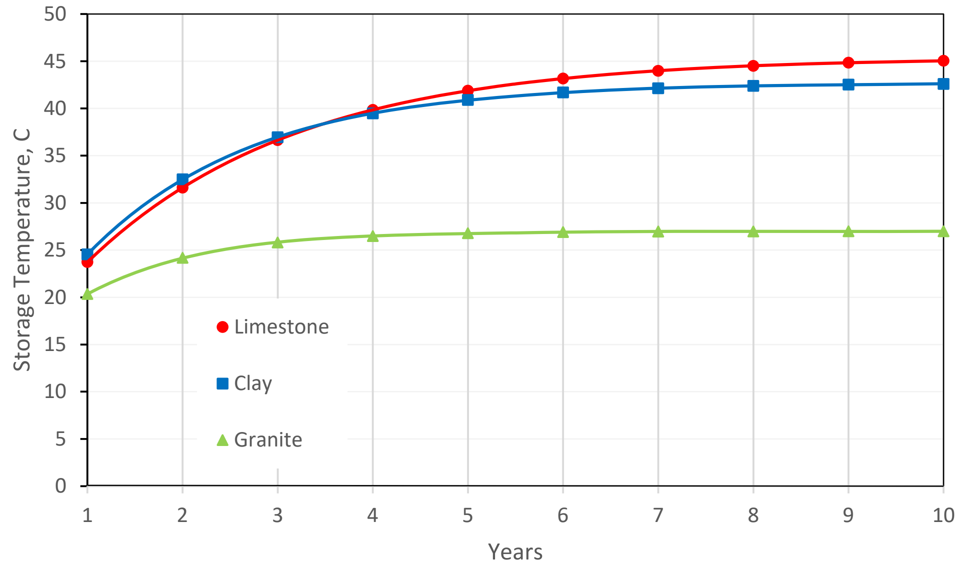

Figure 5.8 TES Tank Temperature Variations with Operation Time for Earth Types

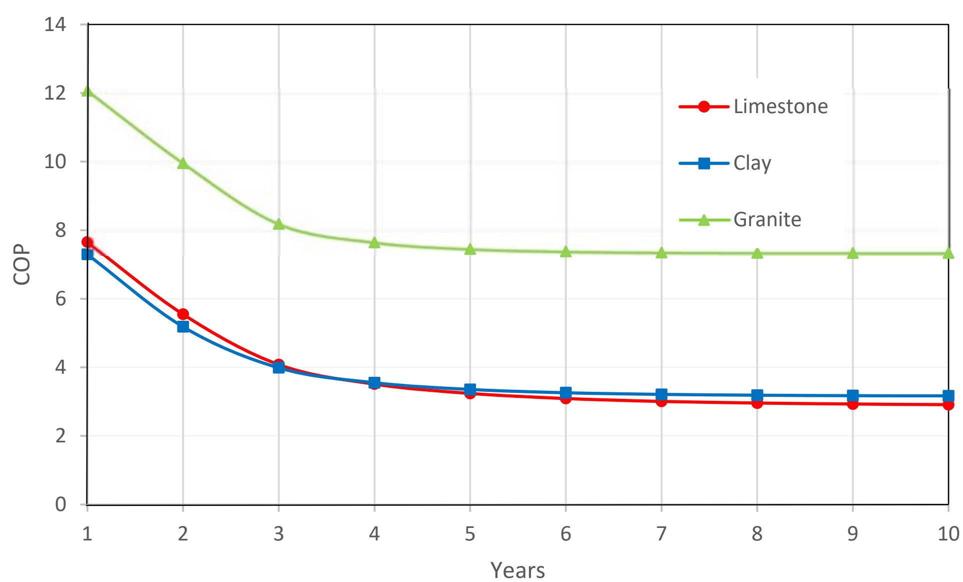

Figure 5.9 Effects of Earth Type on COP Values with Operation Time

Figure 5.10 has been supplied to show how the values of COP change relative to the values in the tank's storage temperature for the entire operation time. As can be seen from the graph, as temperature increases with operation time, COP decreases relatively until they both reach the annually period phase.

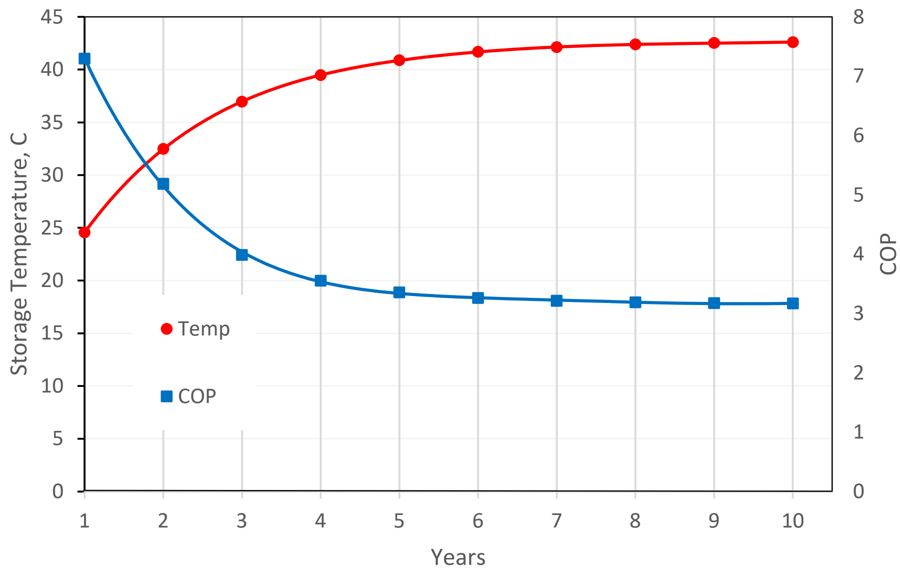

Figure 5.10 Changes in COP Relative to Storage Temperature for Clay Earth Type with Operation Time

Figures 5.11 and 5.12 have been supplied to examine how tank volume affects storage temperature and COP. Figure 5.11 shows yearly variation in storage temperature for tanks with sizes of 200, 300, 400, and 500 m^3. The graph shows that the storage temperature and the water temperature amplitude both decrease as tank volume grows.

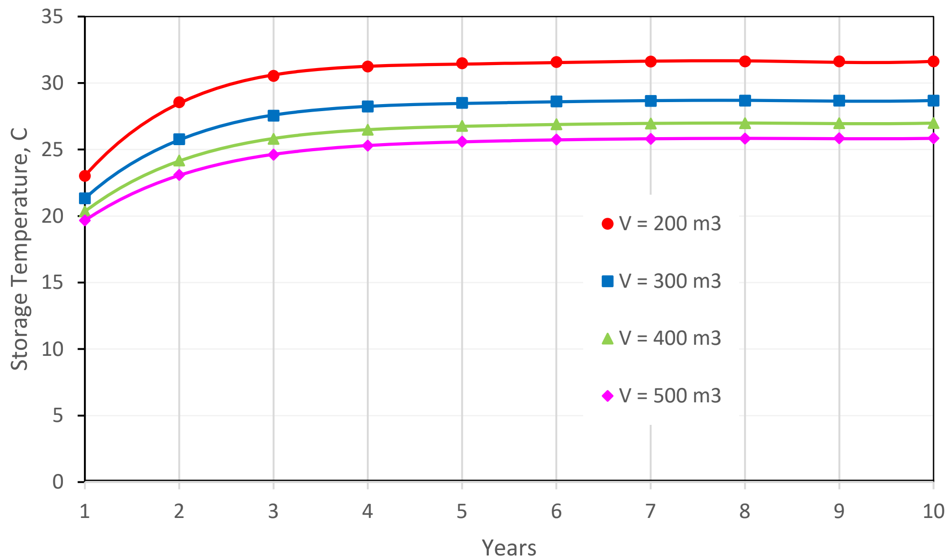

Figure 5.11 Variation of TES Tank Temperature with Volume

The graph illustrates that the temperature is relatively lower in the first few years but increases afterward and stabilizes beyond the fifth year onward.

Figure 5.12 illustrates how the size of the tank affects the heat pump's annual COP for various storage sizes. According to the graph, while increasing tank capacity improves performance, the amount of COP increase diminishes as tank size increases. After achieving the annually recurring operational regime, the COP stays constant for the following years. It is observed that the size of the TES tank has a good impact on the refrigeration system's annual COP.

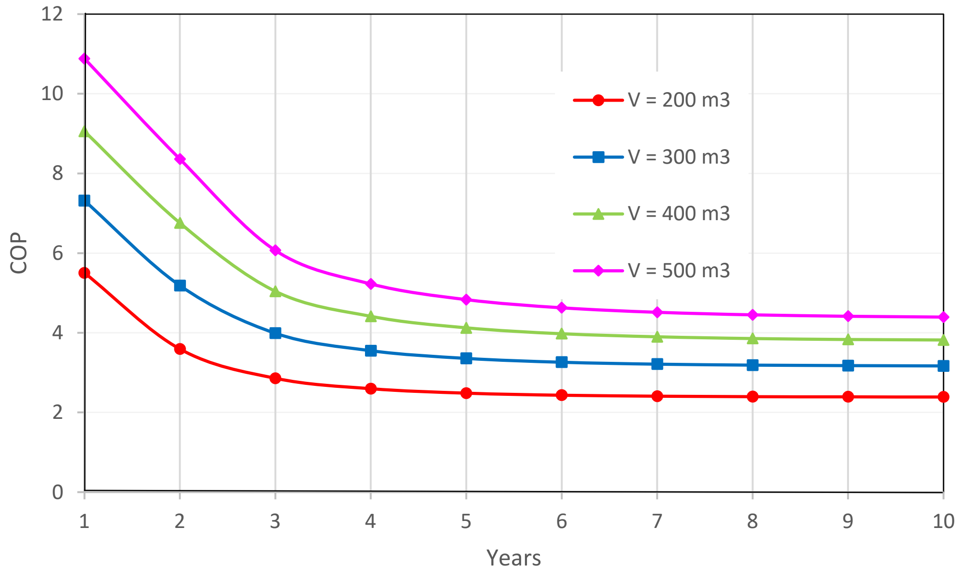

Figure 5.12 Effects of Tank Volume on COP

Larger tanks are preferable because they allow for higher heat dissipation from TES tank to surrounding earth, which keeps the storage temperature lower. It is obvious that the storage temperature decreases with tank size. While choosing the right tank size, however, economic factors should be taken into account because choosing a larger tank implies a higher initial cost. A TES tank with a volume of 400 m^3 is appropriate because less storage space is needed and a lower initial investment is preferred.

The progression toward an annual periodic regime is also shown in Figure 5.13. The temperature varies notably, especially in the first and second years, but after the fifth onward, when the storage temperatures are similar to the ones of the sixth year, it stabilizes. These numbers provide crucial information about how long the transient period lasts until we get to the annually periodic phase.

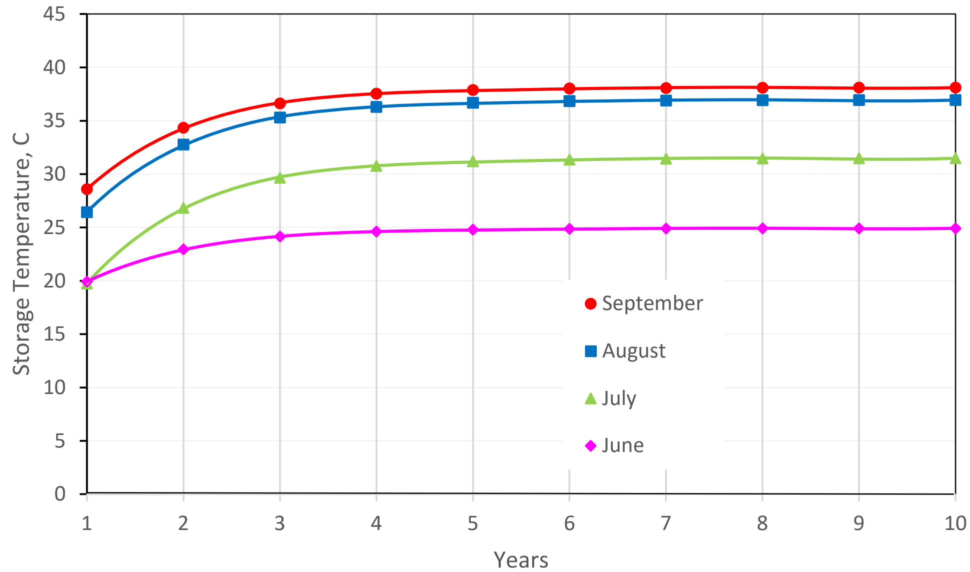

Figure 5.13 Storage Temperature Variations with Operating Time for Cooling Months

The lowest temperatures are recorded in June and the highest ones are in September, which makes sense as we saw from figures 5.2 and 5.5 that the higher COP values occurred in the months of June and the lowest ones were in September. These values only consider the heat from ambient air because it is assumed that the solar air temperature is neglected since the syrup room walls are not exposed to sunlight (solar radiations), and the roof is covered with a tent to prevent solar heat gain.

CHAPTER 6
CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

The thermal performance of a Carbonated Soft Drink (CSD) cooling system that uses an underground TES tank was examined in this study. The system's components were expressed theoretically, and the performance characteristics of the system were constructed using a mathematical model. To evaluate the effects of input parameters on the system performance, a MATLAB program was developed. The analytical model discussed in this study is useful for forecasting the long-term performance of a refrigeration system for industrial applications in CSD cooling of a storage, that includes an underground spherical TES tank. In order to estimate temperatures in the storage tanks and the heat pump's COP, the model takes into account a variety of system factors, including Carnot efficiency and storage volume. According to the findings, this system operates at a higher thermodynamic efficiency than conventional air source refrigeration systems. The following is a summary of the study's main conclusions:

1. Performance indicators like the temperature of water in the TES tank and the COP of the refrigeration system gradually rise from the beginning of the cooling operation until the fifth year, when it runs on a periodic basis. According to the study, it takes about five years for the system parameters taken into account in the research to establish an annually periodic operating regime.
2. The type of earth or thermos-physical properties of the geographical composition surrounding the subversive TES tank have a significant impact on the performance if the cooling system. As a result, it is suggested that an appropriate location is carefully picked for the installation of such cooling systems.
3. The TES water temperature changes very little with an increase in the CE value, while the COP increases noticeably.

4. With a cooling load of approximately 15 kW, higher refrigeration system COP, and a clay earth type, a storage tank volume of 400 m^3 is deemed to be an appropriate amount for chilling the syrup room for the climatic circumstances of Erbil's Coca-Cola Plant.

6.2 Recommendations for Future Research

1. It is recommended to perform a detailed thermos-economic study while taking into account the soil surrounding the tank, the tank size, and the rest of the system's features. This will make it easier to choose the optimal design for the system under consideration.
2. Further developments can be made for energy savings if an extensive exergy and energy analyses are undertaken for the components of the system.
3. Because it is outside the scope of this study, cost analysis for the system was not addressed. Hence, one can economically analyze the cost of such system and compare it to conventional heat pump or refrigeration systems to observe the long term economic viability.

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APPENDICES
MATLAB PROGRAMS FOR VALIDATIONS AND STUDY

Appendix 1 MATLAB Program for First Validation [14]

```
clc
clear
close all
load('IITO','to');
rhow = 1000;    %rho water density
cpw = 4.187;    %water heat capacity
tinf = 288;     %tinf soil tempreture
fi = 0.01736;
u=1.2;
uah = 345;

%%%%carnot efficiency%%%%
% ce=0.3;
 ce=0.4;
% ce=0.5;
nohome = 1;
%%%%%%%%%%%%

%%%%tank radius%%%%
% r=2.8794;    %v100
% r=3.6278;    %v200
% r = 4.1528;    %tank radius at v=300m3
% r=4.570;     %v400
% r=4.924;     %v500
% r=5.232;     %v600
 r=5.508;     %v700
v = 4/3 * pi * r^3;

%%%%%%%%soil types%%%%%%%%
%%%%%limstone propetities
%  rhos = 2500;      %rhos is soil density
%  cps = 0.9;        %soil specific heat
%  con = 1.3;        %soil conductivity
%  alpha = 5.75e-7;  %diffusivity

%%%%%Coarse
% % % rhos = 2050;
% % % cps = 1.842;
% % % con = 0.519;
% % % alpha =1.39E-7;
```

```
%%%%%Granite
 rhos = 2640;
 cps = 0.82;
 con = 3;
 alpha =1.4E-6;
%%%%%%%%%%%%%%%%%%%%%%%%%

p= rhow * cpw / (3 * rhos *cps);
dt = 3600 * alpha / r^2;
p1=p/dt;
rt=1/(sqrt(pi*dt));
gama = 4 * pi *r* con / (nohome * uah);

sbt2=(nohome * uah) / (4 * pi * r * con * 288);
crp = [0 0 0 1 1 1 1 1 1 1 0 0];
ido = [31 31 30 31 30 31 31 28 31 30 31 30];
day=0;
ihr = zeros(1,12);
tay=zeros(10,8760);

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
lg=zeros(1,12);
n=zeros(1,12);
n(1)=ido(1);
lg(1)=ido(1);
for i=2:12
    lg(i)=n(i-1)+ido(i);
    n(i)=lg(i);
end

n=182;
nn=n-182;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
for i = 1:12
    day = day + ido(i);
    ihr(i)= day * 24;
end
k=1;
fa = zeros(1,8760);
c1 = zeros(1,8760);
h  = zeros(1,8760);
for i = 1: 8760
    h(i) = crp(k) * sbt2*(20 - to(i));
    fa(i) = ((to(i) + 273) /288)-1;
    c1(i) = (u * (fi - fa(i)))+fi;
    if (h(i) <= 0)
        h(i) =0;
    end
    if (i == ihr(k)), k=k+1; end
end
```

```
fw= zeros(10,8760);
tdg= zeros(10,8760);
fw(1,1) = 0;
tdg(1,1) = 15; %TES water temperature at starting point
k=1;
cop=zeros(1,10);

for m=1:10

    qly=0;  %qly is an annual dimensionless energy
required of house
    wy=0;   %wy is an annual dimensionless compressor
work
    qy=0;   %qy is heat input to TES tank
    fw(m,1) = fw(1,1);
    tdg(m,1) = ((fw(m,1) + 1) * 288) - 273;
    wo(1) = (fi - fa(1)) * (c1(1) - fw(m,1)) / (ce*
(c1(1) +1) * gama);

    if (fw(m,1) >= c1(1)), wo(1) =0; end

    qn(1) = - h(1) + wo(1);
    fw(m,2)=(qn(1)+((p1+rt)*fw(m,1)))/(1+p1+rt);
    tdg(m,2)=((fw(m,2)+1)*288)-273;
    wo(2)=(fi-fa(2))*(c1(2)-fw(m,1))/(ce*(c1(2)+1)*gama);
    if (fw(m,2)>=c1(2)),wo(2)=0;end
    qn(2) = - h(2) + wo(2);

    for i=3:8760
        s1=0;
        for n=1:i-2;
            s1=s1+(fw(m,n+1)-fw(m,n))/(sqrt(pi*dt*(i-
n)));
        end
        wo(i)=(fi-fa(i))*(c1(i)-fw(m,i-
1))/(ce*(c1(i)+1)*gama);
        if (fw(m,i-1)>=c1(i)),wo(i)=0;end

        qn(i) = - h(i) + wo(i);
        fw(m,i)=(qn(i)+((p1+rt)*fw(m,i-1))-s1)/(1+p1+rt);
        tdg(m,i)=((fw(m,i)+1)*288)-273;

        wy=wy+wo(i)*dt;
        qly=qly+h(i)*dt;
        qy=qy+qn(i)*dt;
        if (i~=ihr(k)), continue; end

        tay(m,k)=tdg(m,i);
        k=k+1;
        if (i==8760),fw(1,1)=fw(m,8760); end
    end
```

```
    if (m~=1)
        qst=p*(fw(m,8760)-fw(m,1));
    else
        qst=p*(fw(m,8760)-0);
    end

    cop(m)=qly/wy;
    k=1;
end
```

Appendix 2 MATLAB Program for Second Validation [10]

```
clc
clear
close all
load('IITO','to');
rhow = 1000;    %rho water density
cpw = 4.187;    %water heat capacity
tinf = 288;     %tinf soil tempreture
fi = 0.03472;   % fi=(298-288)/288
u=1.2;
uah = 690;

%%%%carnot efficiency%%%%
 ce=0.2;
% ce=0.3;
% ce=0.4;
% ce=0.5;
nohome = 1;
%%%%%%%%%%%%

%%%%tank radius%%%%
 r=2.8794;    %v100
% r=3.2961;    %v150
% r=3.6278;    %v200
% r=3.9080;    %v250
% r=4.1528;    %tank radius at v=300m3
% r=4.570;     %v400
% r=4.924;     %v500
% r=5.232;     %v600
% r=5.508;     %v700
v = 4/3 * pi * r^3;

%%%%%%%%soil types%%%%%%%%
%%%%%limstone propetities
% rhos = 2500;      %rhos is soil density
% cps = 0.9;        %soil specific heat
% con = 1.3;        %soil conductivity
% alpha = 5.75e-7;  %diffusivity
%%%%%Coarse
```

```
% rhos = 2050;
% cps = 1.842;
% con = 0.519;
% alpha =1.39E-7;

%%%%%Granite
  rhos = 2640;
  cps = 0.82;
  con = 3;
  alpha =1.4E-6;
%%%%%%%%%%%%%%%%%%%%%%%%%

p= rhow * cpw / (3 * rhos *cps);
dt = 3600 * alpha / r^2;
p1=p/dt;
rt=1/(sqrt(pi*dt));
gama = 4 * pi *r* con / (nohome * uah);

sbt2=(nohome * uah) / (4 * pi * r * con * 288);
crp = [1 1 1 0 0 0 0 0 0 0 0 1];
ido = [31 31 30 31 30 31 31 28 31 30 31 30];
day=0;
ihr = zeros(1,12);
tay=zeros(10,8760);

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
lg=zeros(1,12);
n=zeros(1,12);
n(1)=ido(1);
lg(1)=ido(1);
for i=2:12
    lg(i)=n(i-1)+ido(i);
    n(i)=lg(i);
end

n=182;
nn=n-182;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
for i = 1:12
    day = day + ido(i);
    ihr(i)= day * 24;
end
k=1;
fa = zeros(1,8760);
c1 = zeros(1,8760);
c2 = zeros(1,8760);
h  = zeros(1,8760);
for i = 1: 8760
    h(i) = crp(k) * sbt2*(to(i)-25);
    fa(i) = ((to(i) + 273) /288)-1;
```

```
    c1(i) = (u * (fa(i)-fi))-fi;
    c2(i)=(u*(fi-fa(i)))+fi;
    if (h(i) <= 0)
        h(i) =0;
    end
    if (i == ihr(k)), k=k+1; end
end
fw= zeros(10,8760);
tdg= zeros(10,8760);
fw(1,1) = 0;
tdg(1,1) = 15; %TES water temperature at starting point
k=1;
cop=zeros(1,10);

for m=1:10

    qly=0;  %qly is an annual dimensionless energy
required of house
    wy=0;   %wy is an annual dimensionless compressor
work
    qy=0;   %qy is heat input to TES tank
    fw(m,1) = fw(1,1);
    tdg(m,1) = ((fw(m,1) + 1) * 288) - 273;
    wo(1) = (fa(1)-fi) * (c1(1) + fw(m,1)) / (ce* (c2(1)
+1) * gama);

    % if (fw(m,1) >= c1(1)), wo(1) =0; end

    if (wo(1) < 0 ), wo(1) =0; end

    qn(1) =  h(1) + wo(1);
    fw(m,2)=(qn(1)+((p1+rt)*fw(m,1)))/(1+p1+rt);
    tdg(m,2)=((fw(m,2)+1)*288)-273;
    wo(2)=(fi-fa(2))*(c1(2)-fw(m,1))/(ce*(c2(2)+1)*gama);
   % if (fw(m,2)>=c1(2)),wo(2)=0;end
   if (wo(2) < 0 ), wo(2) =0; end

    qn(2) =  h(2) + wo(2);

    for i=3:8760
        s1=0;
        for n=1:i-2;
            s1=s1+(fw(m,n+1)-fw(m,n))/(sqrt(pi*dt*(i-
n)));
        end
        wo(i)=(fi-fa(i))*(c1(i)-fw(m,i-
1))/(ce*(c2(i)+1)*gama);
       % if (fw(m,i-1)>=c1(i)),wo(i)=0;end
        if (wo(i) < 0 ), wo(i) =0; end

        qn(i) =  h(i) + wo(i);
```

```
        fw(m,i)=(qn(i)+((p1+rt)*fw(m,i-1))-s1)/(1+p1+rt);
        tdg(m,i)=((fw(m,i)+1)*288)-273;

        wy=wy+wo(i)*dt;
        qly=qly+h(i)*dt;
        qy=qy+qn(i)*dt;
        if (i~=ihr(k)), continue; end

        tay(m,k)=tdg(m,i);
        k=k+1;
        if (i==8760),fw(1,1)=fw(m,8760); end
    end
    if (m~=1)
        qst=p*(fw(m,8760)-fw(m,1));
    else
        qst=p*(fw(m,8760)-0);
    end

    cop(m)=qly/wy;
    k=1;
end
```

Appendix 3 MATLAB Program for the Study

```
clc
clear
close all
load('Temp','to');
rhow = 1000;    %rho water density
cpw = 4.187;    %water heat capacity
tinf = 288;     %tinf soil tempreture
fi = 0.03472;   % fi=(298-288)/288
u= 1.2;
uah = 705.34;

%%%%carnot efficiency%%%%
% ce=0.2;
% ce=0.3;
 ce=0.4;
% ce=0.5;
nohome = 1;
%%%%%%%%%%%%

%%%%tank radius%%%%
% r=2.8794;    %v100
% r=3.2961;    %v150
% r=3.6278;    %v200
% r=3.9080;    %v250
% r=4.1528;    %tank radius at v=300m3
 r=4.570;     %v400
```

```
% r=4.924;     %v500
% r=5.232;     %v600
% r=5.508;     %v700
v = 4/3 * pi * r^3;

%%%%%%%%soil types%%%%%%%%
%%%%%limstone propetities
% rhos = 2500;      %rhos is soil density
% cps = 0.9;        %soil specific heat
% con = 1.3;        %soil conductivity
% alpha = 5.75e-7;  %diffusivity

%%%%%Coarse
% rhos = 2050;
% cps = 1.842;
% con = 0.519;
% alpha =1.39E-7;

%%%%%%Clay
 rhos = 1500;
 cps = 0.88;
 con = 1.4;
 alpha =1.1E-6;

%%%%%Granite
% rhos = 2640;
% cps = 0.82;
% con = 3;
% alpha =1.4E-6;
%%%%%%%%%%%%%%%%%%%%%%%%%

p= rhow * cpw / (3 * rhos *cps);
dt = 3600 * alpha / r^2;
p1=p/dt;
rt=1/(sqrt(pi*dt));
gama = 4 * pi *r* con / (nohome * uah);

sbt2=(nohome * uah) / (4 * pi * r * con * 288);
crp = [1 1 1 0 0 0 0 0 0 0 0 1];
ido = [31 31 30 31 30 31 31 28 31 30 31 30];
day=0;
ihr = zeros(1,12);
tay=zeros(10,8760);

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
lg=zeros(1,12);
n=zeros(1,12);
n(1)=ido(1);
lg(1)=ido(1);
for i=2:12
    lg(i)=n(i-1)+ido(i);
```

```
    n(i)=lg(i);
end

n=182;
nn=n-182;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
for i = 1:12
    day = day + ido(i);
    ihr(i)= day * 24;
end
k=1;
fa = zeros(1,8760);
c1 = zeros(1,8760);
c2 = zeros(1,8760);
h  = zeros(1,8760);
for i = 1: 8760
    h(i) = crp(k) * sbt2*(to(i)-25);
    fa(i) = ((to(i) + 273) /288)-1;
    c1(i) = (u * (fa(i)-fi))-fi;
    c2(i)=(u*(fi-fa(i)))+fi;
    if (h(i) <= 0)
        h(i) =0;
    end
    if (i == ihr(k)), k=k+1; end
end
fw= zeros(10,8760);
tdg= zeros(10,8760);
fw(1,1) = 0;
tdg(1,1) = 15; %TES water temperature at starting point
k=1;
cop=zeros(1,10);
% Monthly COP variable
monthly_cop = zeros(10, 12);

for m=1:10

    qly=0;  %qly is an annual dimensionless energy
required of house
    wy=0;   %wy is an annual dimensionless compressor
work
    qy=0;   %qy is heat input to TES tank

    qly_monthly = zeros(1, 12);
    wy_monthly = zeros(1, 12);
    qy_monthly = zeros(1, 12);

    fw(m,1) = fw(1,1);
    tdg(m,1) = ((fw(m,1) + 1) * 288) - 273;
    wo(1) = (fa(1)-fi) * (c1(1) + fw(m,1)) / (ce* (c2(1)
+1) * gama);
```

```
    % if (fw(m,1) >= c1(1)), wo(1) =0; end

    if (wo(1) < 0 ), wo(1) =0; end

    qn(1) =  h(1) + wo(1);
    fw(m,2)=(qn(1)+((p1+rt)*fw(m,1)))/(1+p1+rt);
    tdg(m,2)=((fw(m,2)+1)*288)-273;
    wo(2)=(fi-fa(2))*(c1(2)-fw(m,1))/(ce*(c2(2)+1)*gama);
   % if (fw(m,2)>=c1(2)),wo(2)=0;end
   if (wo(2) < 0 ), wo(2) =0; end

    qn(2) =  h(2) + wo(2);

    month_idx = 1;

    for i=3:8760
        s1=0;
        for n=1:i-2
            s1=s1+(fw(m,n+1)-fw(m,n))/(sqrt(pi*dt*(i-
n)));
        end
        wo(i)=(fi-fa(i))*(c1(i)-fw(m,i-
1))/(ce*(c2(i)+1)*gama);
       % if (fw(m,i-1)>=c1(i)),wo(i)=0;end
        if (wo(i) < 0 ), wo(i) =0; end

        qn(i) =  h(i) + wo(i);
        fw(m,i)=(qn(i)+((p1+rt)*fw(m,i-1))-s1)/(1+p1+rt);
        tdg(m,i)=((fw(m,i)+1)*288)-273;

        wy=wy+wo(i)*dt;
        qly=qly+h(i)*dt;
        qy=qy+qn(i)*dt;

        wy_monthly(month_idx) = wy_monthly(month_idx) +
wo(i) * dt;
        qly_monthly(month_idx) = qly_monthly(month_idx) +
h(i) * dt;
        qy_monthly(month_idx) = qy_monthly(month_idx) +
qn(i) * dt;

        if (i~=ihr(k)), continue; end

        tay(m,k)=tdg(m,i);
        k=k+1;
        if (i==8760),fw(1,1)=fw(m,8760); end

         monthly_cop(m, month_idx) =
qly_monthly(month_idx) / wy_monthly(month_idx);
        month_idx = month_idx + 1;
```

```
    end


    if (m~=1)
        qst=p*(fw(m,8760)-fw(m,1));
    else
        qst=p*(fw(m,8760)-0);
    end

    cop(m)=qly/wy;
    k=1;

end
```

CURRICULUM VITAE

PERSONAL DETAILS

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EDUCATION

February 2020 – June 2023	**Gaziantep University, Turkey** Master’s Degree in Mechanical Engineering – Energy Division
September 2014 - August 2018	**The American University of Iraq, Sulaimani** Bachelor's Degree in Energy Engineering
September 2013 - August 2014	**The American University of Iraq, Sulaimani** Certificate of Accomplishment, Academic Preparatory Program (APP)
September 2010 - August 2013	**Halkawt High School, Sulaimani** High School Diploma