

DOKUZ EYLÜL UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

**NUMERICAL INVESTIGATION OF SOLAR
ASSISTED THERMAL ENERGY STORAGE
SYSTEMS**

by
Ersin ALPTEKİN

September, 2021
İZMİR

NUMERICAL INVESTIGATION OF SOLAR ASSISTED THERMAL ENERGY STORAGE SYSTEMS

**A Thesis Submitted to the
Graduate School of Natural and Applied Sciences of Dokuz Eylül University
In Partial Fulfillment of the Requirements for the Degree of Doctor of
Philosophy in Mechanical Engineering, Thermodynamics Program**

**by
Ersin ALPTEKİN**

September, 2021

İZMİR

Ph.D. THESIS EXAMINATION RESULT FORM

We have read the thesis entitled “**NUMERICAL INVESTIGATION OF SOLAR ASSISTED THERMAL ENERGY STORAGE SYSTEMS**” completed by **ERSİN ALPTEKİN** under supervision of **ASSOC. PROF. DR. MEHMET AKİF EZAN** and we certify that in our opinion it is fully adequate, in scope and in quality, as a thesis for the degree of Doctor of Philosophy.

.....
Assoc. Prof. Dr. Mehmet Akif EZAN

Supervisor

.....
Prof. Dr. Aytunç EREK

Thesis Committee Member

.....
Prof. Dr. Tahsin BAŞARAN

Thesis Committee Member

.....
Assoc. Prof. Dr. Alpaslan TURGUT

Examining Committee Member

.....
Prof. Dr. Arif HEPBAŞLI

Examining Committee Member

.....
Prof. Dr. Özgür ÖZÇELİK

Director

Graduate School of Natural and Applied Sciences

ACKNOWLEDGEMENTS

I would like to thank Assoc. Prof. Dr. Mehmet Akif EZAN, the supervisor of this dissertation. What can I say, it has been almost 10 years since I met you and there has not been a single moment that I regret. You are one of the very few people in this world whose opinions I respect and value.

I would like to thank Prof. Dr. Aytunç EREK, the committee member and the head of the Thermodynamics program, not only for his contribution to the dissertation, but also for the support of his students for their scientific self-improvements.

I want to thank Prof. Dr. Tahsin BAŞARAN, the committee member, for his advices on the scientific progress of the dissertation at periodical thesis meetings.

I want to thank Assoc. Prof. Dr. Alpaslan TURGUT and Prof. Dr. Arif HEPBAŞLI, examining committee members, for their valuable comments and recommendations.

I also want to thank MSc. Mehmet UZUN (aka Prof.), my wingman, for his friendship since the beginning of my undergraduate education. As Nazım said “*we will see beautiful days, we will see sunny days*”.

I want to thank Kenan ÖZTÜRK, Nurhan VATANSEVER, Kazım YILDIZ, Ayça Seda BİNCAN YILDIZ, my friends, for being patient with me for so many years despite my unbearable behaviors. I want to thank Alim ZORLUOL, the Technician of the Thermodynamics & Heat Transfer Laboratory, for the long tea talks and for sincerely sharing his life experiences during these conversations.

This dissertation is dedicated to my beloved family, my father Hüseyin ALPTEKİN (requiescat in pace), my mother Nesrin ALPTEKİN and my brother Aydın ALPTEKİN. I feel so privileged to have your sustained love, support and encouragement. I love you.

To my father who never saw this adventure.

Ersin ALPTEKİN



NUMERICAL INVESTIGATION OF SOLAR ASSISTED THERMAL ENERGY STORAGE SYSTEMS

ABSTRACT

Due to the intermittent nature of solar energy and the mismatch between the instantaneous demand and supply, solar energy could not be used effectively or continuously in solar-assisted applications such as heating a building. As a common approach, it is possible to store the available solar energy when it is abundant and use it later to operate solar-assisted applications sustainably. Thermal energy storage (TES) applications provide critical solutions for ensuring the sustainability of solar energy. This dissertation focuses on numerically evaluating the performance indicators of the sensible and latent heat packed-bed TES tanks under various working and design parameters. An in-house transient mathematical model is developed, which consists of a flat plate solar collector, TES tank, and heat pump to simulate the TES systems' heat transfer phenomena under variable weather conditions for the heating season months of Izmir City, Turkey. The heat transfer characteristics and the second law aspect of the TES tank are examined to provide a thorough understanding. The integrated TES systems' optimum design parameters and working conditions are proposed for heating season months (November, December, January, and February). Consequently, it is observed that increasing the mass flow rate of the heat transfer fluid and decreasing storage material diameter enhances system indicators. Moreover, comparative results reveal that a well-designed TES tank integrated with a heat pump unit could have up to 48% higher COP values and provide 30% less energy consumption when it is compared with the traditional air-source heat pump.

Keywords: Thermal energy storage, energy analysis; exergy analysis, integrated storage systems, packed bed, solar energy

GÜNEŞ DESTEKLİ ISIL ENERJİ DEPOLAMA SİSTEMLERİNİN SAYISAL İNCELENMESİ

ÖZ

Güneş enerjisinin düzensiz doğal yapısından ve enerjinin anlık arz ve talep dengesindeki uyumsuzlukları nedeniyle bir binanın ısıtılması gibi güneş enerjisi uygulamalarında güneş enerjisi verimli veya devamlı bir şekilde kullanılamayabilmektedir. Yaygın bir yaklaşım olarak, mevcut güneş enerjisini bol olduğunda depolamak ve daha sonra güneş destekli uygulamaları sürdürülebilir bir şekilde çalıştırmak için kullanmak mümkündür. Isıl enerji depolama (IED) uygulamaları güneş enerjisinin sürdürülebilirliğini sağlamak için can alıcı çözümler sunar. Bu tez çalışması, dolgulu yataklı duyulur ve gizli ısı IED tanklarının performans indikatörlerinin çeşitli çalışma ve dizayn parametreleri için sayısal olarak değerlendirilmesine odaklanmaktadır. Düz plakalı güneş kolektörü, IED tankı ve ısı pompası içeren zamana bağlı matematiksel model IED sistemlerinin ısı transferi olgularını değişken gerçek hava koşulları altında İzmir ili, Türkiye için ısıtma ihtiyacının olduğu aylarda elde edebilme amacıyla geliştirilmiştir. Kapsamlı bir inceleme gerçekleştirebilmek için IED tanklarının sadece ısı transferi karakteristikleri değil ayrıca ikinci yasa yönü de incelenmiştir. Isıtma sezonu ayları (Kasım, Aralık, Ocak, Şubat) için entegre IED sistemlerinin optimum dizayn parametreleri ve çalışma koşulları önerilmiştir. Sonuç olarak ısı transferi akışkanının kütledebisinin artması ve depolama malzemesi çapının azalması ile birlikte sistem indikatörlerinde iyileşme gözlemlenmiştir. Ayrıca, karşılaştırmalı sonuçlar göstermiştir ki ısı pompasına entegre edilmiş iyi tasarlanmış bir IED tankı sayesinde hava kaynaklı ısı pompası ile karşılaştırıldığında %48 daha yüksek COP değeri ve %30 daha az enerji tüketimi elde edilebildiği ortaya konulmuştur.

Anahtar kelimeler: Isıl enerji depolama, enerji analizi, ekserji analizi, entegre depolama sistemleri, dolgulu yatak, güneş enerjisi

CONTENTS

	Page
Ph.D. THESIS EXAMINATION RESULT FORM	ii
ACKNOWLEDGEMENTS	iii
ABSTRACT	v
ÖZ	vi
LIST OF FIGURES	x
LIST OF TABLES	xiv
 CHAPTER ONE – INTRODUCTION	 1
1.1 Literature Review	8
1.1.1 Sensible Heat Packed Bed TES Systems	10
1.1.2 Latent Heat Packed Bed TES Systems	13
1.2 Motivation and Objectives of the Dissertation	18
 CHAPTER TWO – MATERIAL & METHOD	 20
2.1 Definition of the Problem	20
2.2 Numerical Models	22
2.2.1 Flat Plate Solar Collector Model	22
2.2.2 Sensible Heat Packed Bed TES Tank Model	28
2.2.3 Latent Heat Packed Bed TES Tank Model	32
2.2.3.1 Enhanced Thermal Conductivity Approach	38
2.2.4 Cyclic Performance Evaluation and Heat Pump Model Integration ..	39
2.3 Solution Procedure	41
 CHAPTER THREE – RESULTS & DISCUSSION	 48
3.1 Results of the Numerical Model of Flat Plate Solar Collector	48
3.1.1 Comparison of Top Heat Loss Coefficient of Solar Collector	48
3.1.2 Flat Plate Solar Collector Dynamic Performance Evaluation	49

3.2 Charging Performance of Sensible Heat Packed Bed TES System	51
3.2.1 Preliminary Analyses of Sensible Heat TES Tank	51
3.2.2 Validation of the Mathematical Model of Sensible Heat TES System	53
3.2.3 Parametric Results of Sensible Heat TES System	54
3.2.3.1 Influence of Mass Flow Rate on System Charging Performance	54
3.2.3.2 Influence of Storage Material Diameter on System Charging Performance	58
3.2.3.3 Influence of Storage Tank Height on System Charging Performance	60
3.2.3.4 Assessment of Integrated System Charging Performance	63
3.2.3.5 Influence of Operating Month on System Charging Performance	66
3.3 Charging Performance of Latent Heat Packed Bed TES System	68
3.3.1 Validity of Enhanced Thermal Conductivity Approach	69
3.3.2 Mesh and Time Step Size Independency Study	71
3.3.3 Validity of the Mathematical Model of Packed Bed Latent Heat TES System	72
3.3.4 Parametric Results of Latent Heat Packed Bed TES System	73
3.3.4.1 Effect of Mass Flow Rate on Integrated System Charging Performance	74
3.3.4.2 Effect of Capsule Diameter on Integrated System Charging Performance	80
3.3.4.3 Overall Assessment of Integrated Latent Heat TES System Charging Performance	84
3.4 Cyclic Performance of Latent Heat Packed Bed TES System	89
3.4.1 Preliminary Analysis of Cyclic Latent Heat TES System	89
3.4.2 Parametric Cyclic Results of Latent Heat TES System	91
3.4.2.1 Effect of Mass Flow Rate on System Cyclic Performance	92
3.4.2.2 Effect of Capsule Diameter on System Cyclic Performance	97
3.4.2.3 Overall Cyclic Performance of Integrated Latent Heat TES System	102

CHAPTER FOUR – CONCLUSIONS & RECOMMENDATIONS	109
REFERENCES.....	113
APPENDICES	124
APPENDIX–1: Nomenclature	124



LIST OF FIGURES

	Page
Figure 1.1 The annual variation of the primary energy consumption by years	2
Figure 1.2 The median amount of CO ₂ emissions per kWh electricity produced by energy production resources.....	3
Figure 1.3 The trajectory of primary energy demand by fuel type	4
Figure 1.4 The trajectory of global energy demand and CO ₂ emissions	5
Figure 1.5 Electricity final consumption by sectors in Turkey.....	5
Figure 1.6 The variation of energy consumption in residential areas by utilization	6
Figure 1.7 The supply methods of domestic heating and hot water needs in newly built houses.....	7
Figure 1.8 The volumetric TES capacities of different types of TES methods according to the storage materials	8
Figure 2.1 Schematic representation of the integrated TES system.....	21
Figure 2.2 The meteorological data for İzmir, Turkey (a) solar radiation, (b) ambient temperature and wind speed.....	22
Figure 2.3 Schematic representation of the flat plate solar collector	23
Figure 2.4 Thermal network of the flat plate solar collector.....	24
Figure 2.5 Schematic representation of the sensible heat packed bed TES tank ..	29
Figure 2.6 Schematic representation of the computational grid structure of the sensible heat TES tank	30
Figure 2.7 Schematic representation of the integrated latent heat TES system	33
Figure 2.8 The reduced computational domain of the latent heat TES tank.....	35
Figure 2.9 The representation of the interface position and heat flux	36
Figure 2.10 The representation of temperature transformation method	37
Figure 2.11 Actual and idealized the interface variations for inward melting.....	39
Figure 3.1 The comparison of the top heat loss coefficient	49
Figure 3.2 A typical time-wise variation of the solar collector parameters integrated with sensible heat TES tank	51
Figure 3.3 A typical time-wise variation of the solar collector parameters integrated with latent heat TES tank	51

Figure 3.4 The variation of the temperature of the HTF along with the storage tank height at some particular instances.....	54
Figure 3.5 The effect of mass flow rate on the (a) energy and exergy efficiency of the tank and collector efficiency and (b) stratification number and the tank outlet temperature – $H_T = 0.8$ m & <i>January</i>	56
Figure 3.6 The effect of mass flow rate on the (a) energy and exergy efficiency of the tank and collector efficiency and (b) stratification number and the tank outlet temperature – $H_T = 1.6$ m & <i>January</i>	58
Figure 3.7 The effect of diameter of the sphere on the (a) energy and exergy efficiency of the tank and collector efficiency (b) stratification number and the tank outlet temperature – $H_T = 0.8$ m & <i>January</i>	59
Figure 3.8 The effect of diameter of the sphere on the (a) energy and exergy efficiency of the tank and collector efficiency (b) stratification number and the tank outlet temperature – $H_T = 1.6$ m & <i>January</i>	60
Figure 3.9 The effect of tank height on the (a) energy and exergy efficiency of the tank and collector efficiency (b) stratification number and the tank outlet temperature – <i>mass flow rate of 0.1 kg/s & January</i>	61
Figure 3.10 The effect of <i>tank height</i> on the (a) energy and exergy efficiency of the tank and collector efficiency (b) stratification number and the tank outlet temperature – <i>mass flow rate of 0.4 kg/s & January</i>	62
Figure 3.11 An overview of the system performance indicators	65
Figure 3.12 The effect of <i>operating month</i> on the (a) energy and exergy efficiency of the tank and collector efficiency (b) stratification number and the tank outlet temperature.....	67
Figure 3.13 Time-wise variations in liquid fractions with different Nusselt correlations	70
Figure 3.14 The comparison of the time-wise variations of PCM temperature at three positions along the flow direction	73
Figure 3.15 The effect of mass flow rate on the mean PCM temperature and liquid fraction for different PCM arrangements – $D_s = 20$ mm & <i>January</i> .	75
Figure 3.16 Spatial liquid fraction variations of the different PCM arrangements (Case #1 & Case #3) at certain instances for four different mass flow rates of HTF – $D_s = 20$ mm & <i>January</i>	77

Figure 3.17 Spatial liquid fraction variations of the different PCM arrangements (Case #5 & Case #7) at certain instances for four different mass flow rate of HTF – $D_s = 20$ mm & <i>January</i>	77
Figure 3.18 The effect of mass flow rate on the exergy efficiency of the storage tank, energy and exergy efficiencies of the integrated system, and collector efficiency – $D_s = 20$ mm & <i>January</i>	79
Figure 3.19 The effect of storage material diameter on the mean PCM temperature and liquid fraction for different PCM arrangements– <i>mass flow rate of 0.8 kg/s & January</i>	81
Figure 3.20 Spatial liquid fraction variations of the different PCM arrangements (Case #1 & Case #3) at certain instances for different sphere diameters – <i>mass flow rate of 0.8 kg/s & January</i>	82
Figure 3.21 Spatial liquid fraction variations of the different PCM arrangements (Case #5 & Case #7) at certain instances for different sphere diameters – <i>mass flow rate of 0.8 kg/s & January</i>	82
Figure 3.22 The effect of storage material diameter on the exergy efficiency of the storage tank, energy and exergy efficiencies of the integrated system, and collector efficiency – <i>mass flow rate of 0.8 kg/s & January</i>	84
Figure 3.23 An overview of the effects of design and working parameters on charging performance of the integrated storage system	87
Figure 3.24 The daily time-wise variations of the mean PCM temperature, liquid fraction TES tank inlet temperature, and COP value of the heat pump	90
Figure 3.25 The effect of mass flow rate on the mean PCM temperature and liquid fraction – $D_s = 20$ mm & <i>January</i>	94
Figure 3.26 The effect of mass flow rate on the liquid fraction, charged and discharged energies – $D_s = 20$ mm & <i>January</i>	95
Figure 3.27 The effect of mass flow rate on the inlet temperature of the TES tank and COP of the heat pump – $D_s = 20$ mm & <i>January</i>	97
Figure 3.28 The effect of capsule diameter on the mean PCM temperature and liquid fraction – <i>mass flow rate of 0.8 kg/s & January</i>	99
Figure 3.29 The effect of capsule diameter on the liquid fraction, charged and discharged energies – $D_s = 20$ mm & <i>January</i>	100

Figure 3.30 The effect of capsule diameter on the inlet temperature of the TES tank and COP of the heat pump – <i>mass flow rate of 0.8 kg/s & January</i>	101
Figure 3.31 An overview of the effects of design and working parameters on the cyclic performance of the integrated latent heat TES system.....	106



LIST OF TABLES

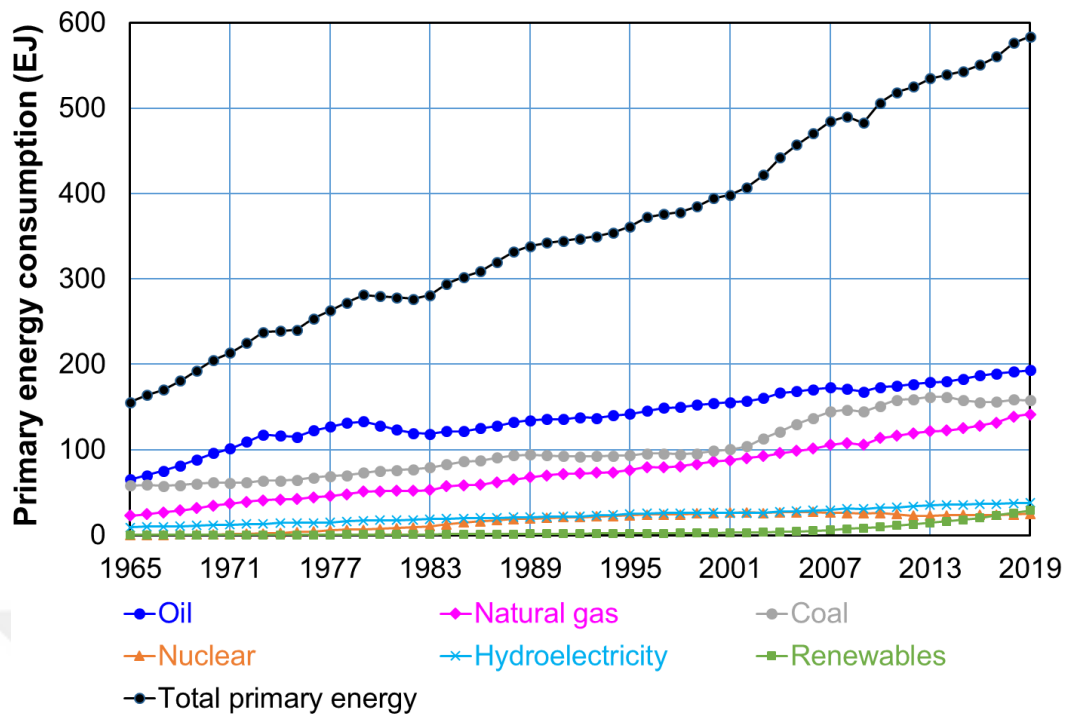
	Page
Table 1.1 A comparative review on recent works on packed bed sensible heat TES systems	16
Table 1.2 A comparative review on recent works on packed bed latent heat TES systems	16
Table 2.1 Design parameters of the flat plate solar collector.....	23
Table 2.2 The thermo-physical properties of the HTF and storage material	29
Table 2.3 The thermo-physical properties of the HTF, shell material, and PCM.	33
Table 3.1 The values of the Biot number for some studied scenarios	52
Table 3.2 The parameters used in the validation case (Meier et al., 1991).....	53
Table 3.3 Effect of source temperature on heat-pump performance.....	66
Table 3.4 The integrated system performance for different operating months.....	68
Table 3.5 Single, double and triple PCM Arrangements inside the TES tank.....	69
Table 3.6 Nusselt correlations for natural convection inside the spherical annulus	70
Table 3.7 The results of the mesh independency study	71
Table 3.8 The results of the time step independency study	72
Table 3.9 The optimum design parameters and working conditions for the charging performance of the integrated TES system.....	86
Table 3.10 The comparison of the variation of the primary outcomes at the end of the cycle days	91
Table 3.11 The optimum design parameters and working conditions for the cyclic performance of the integrated TES system	108
Table 3.12 The mean COP and electrical consumption of the air source heat pump for heating season months	108

CHAPTER ONE

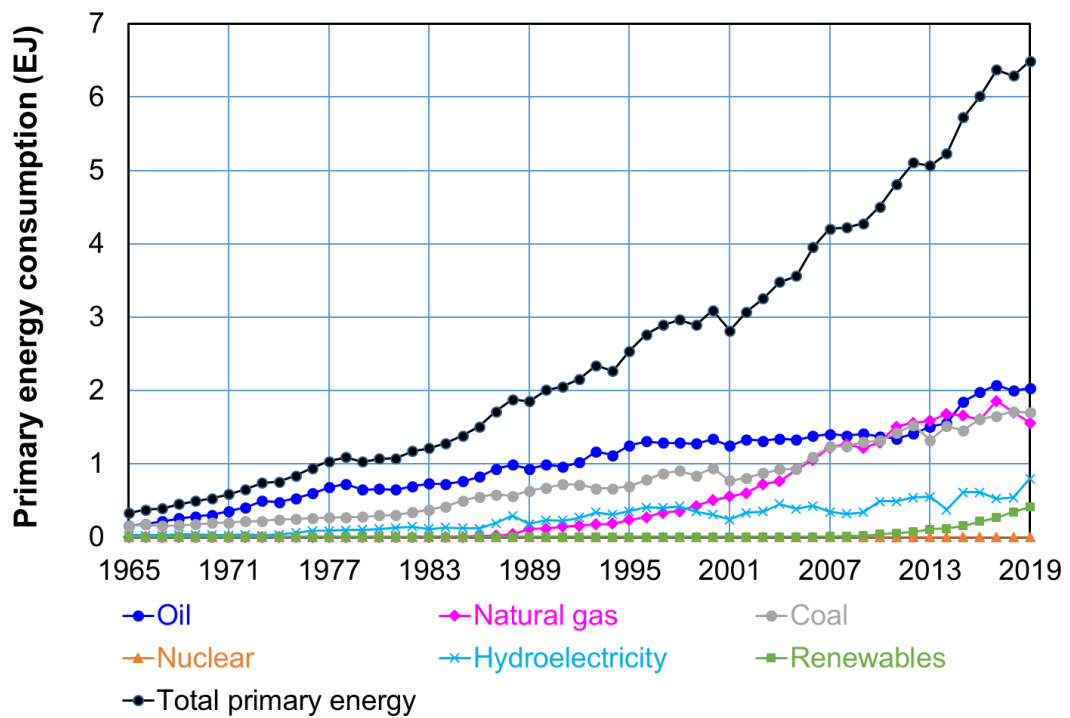
INTRODUCTION

Energy consumption is gradually increasing, and the rapid depletion of energy resources causes a severe energy shortage (Dincer & Ezan, 2018). The variation of the primary energy consumption in terms of fuel type in the world and Turkey is illustrated in Figure 1.1. Since 1965, the fossil-based (oil, coal, and natural gas) produced energy covers ever-increasing energy demand. As can be seen from the figure, although the share of renewable energy resources in energy production has increased significantly since the end of the 2000s, this enhancement can not compensate for the increasing energy demand, and fossil-based energy production is also growing. While the energy supply with renewable energy resources is 4.96% of the total energy demand in the world, it is indicated that this rate becomes 6.33% in Turkey (BP, 2021).

Energy production methods have different effects on global climate change. One of them is the amount of CO₂ emissions to the Earth's atmosphere during the energy production process. According to Intergovernmental Panel on Climate Change (IPCC, 2014), the increase in fossil fuel-based energy production is directly related to the rise in the amount of CO₂ emissions, and this situation is one of the leading causes of the global climate change issue (IPCC, 2014). The amount of CO₂ emissions per kWh electricity produced by energy production resources is presented in Figure 1.2. It can be observed that fossil or biomass-based energy production methods cause 5 to 75 times more damage to nature than renewable energy resources. Furthermore, considering that three-quarters of human-induced CO₂ emissions are due to burning fossil fuels for energy production, the main focus should be on innovative energy generation technologies that cause less CO₂ emissions per unit of energy.



(a) World



(b) Turkey

Figure 1.1 The annual variation of the primary energy consumption by years (modified from BP, 2021)

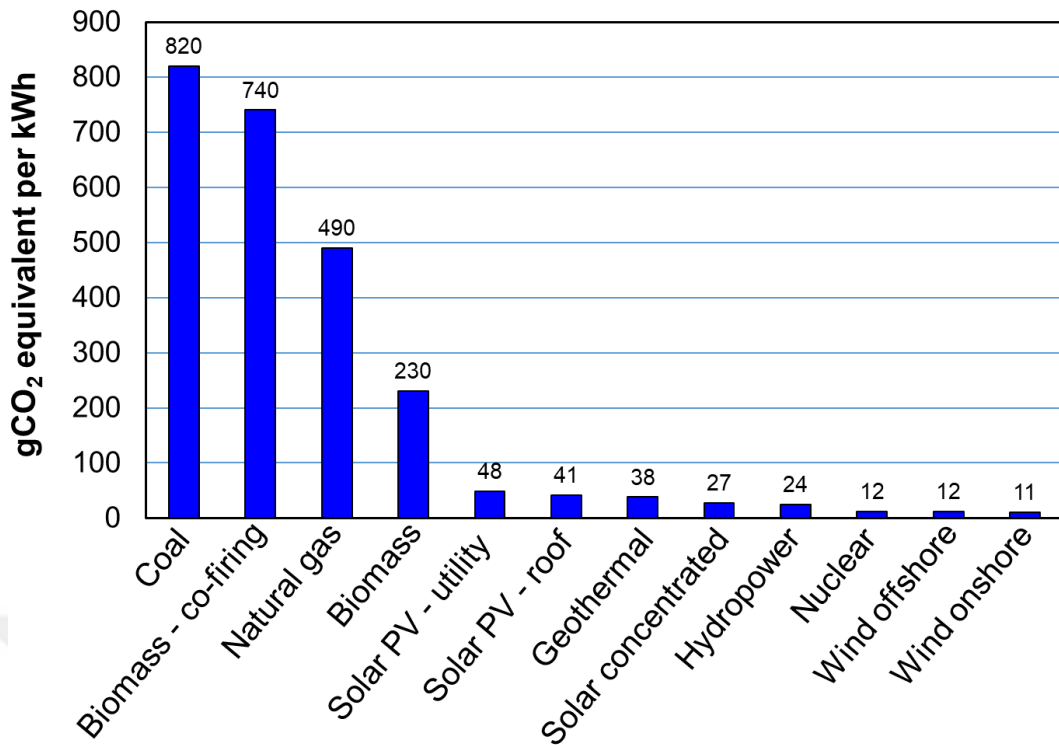


Figure 1.2 The median amount of CO₂ emissions per kWh electricity produced by energy production resources (modified from IPCC, 2014)

Under the name of Stated Policies Scenario (STEPS), the International Energy Agency (IEA) tries to anticipate the problems which the future generation of humanity may encounter according to the current energy policy objectives. In the light of the considerations made for this purpose, the primary energy demand variations by fuel type are presented in Figure 1.3. Here, the impact of the global crisis we have faced since the beginning of 2020 on energy demand is also included. It can be seen from the figure that for the short-term effect of the crisis, only a loss of momentum in an increasing trend of renewable energy resources is foreseen while it is expected that there will be a decrease in fossil-based, i.e., coal and oil, and nuclear energy fuel types. Additionally, it is also estimated that there will be a 7% decrease in 2030 compared to 2019 for coal which has a high amount of CO₂ emissions per unit of energy. On the other hand, an estimated 60% increase in energy supply from renewable energy resources is expected.

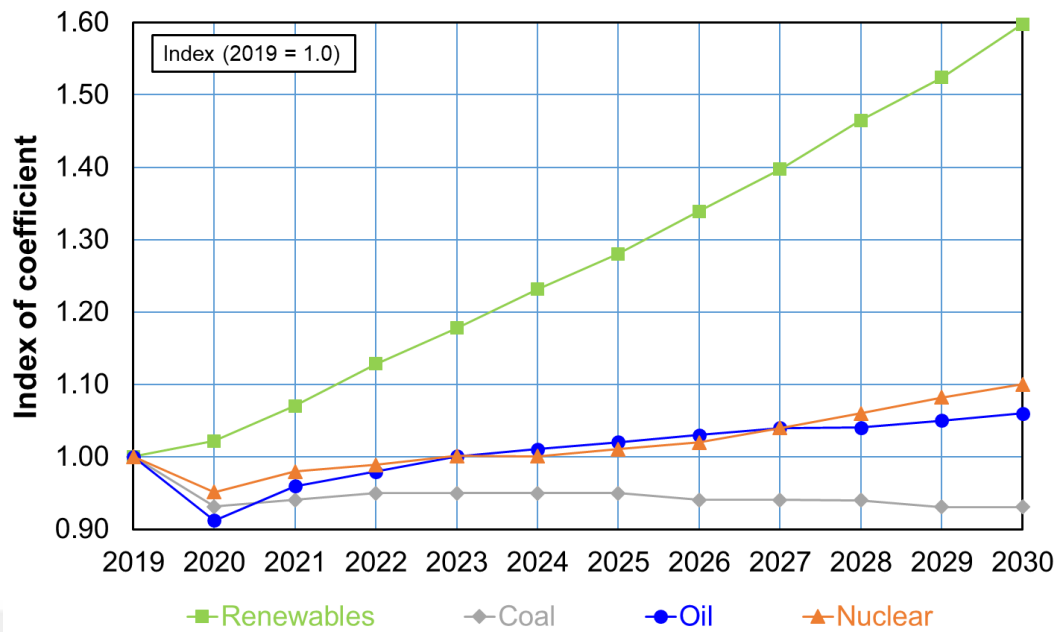


Figure 1.3 The trajectory of primary energy demand by fuel type (modified from IEA, 2020a)

In addition, in STEPS, the amount of global CO₂ emissions and energy demand until 2030 are tried to be estimated based on 2019 values are represented in Figure 1.4. It can be noted that substantial decrement is anticipated for both CO₂ emissions and energy demand due to the global crisis. With the effect of this crisis, it is estimated that the energy demand will barely reach the level of 2019 in 2023. Moreover, while energy demand is expected to increase by 12% until 2030 compared to 2019 in the pre-crisis scenario, the prediction is revised as a 9% increment, according to the new expectation. On the other hand, CO₂ emissions are expected to be lower than the level of 2019 until 2026 in the post-crisis period. Additionally, CO₂ emissions are expected to increase by 1% by 2030 compared to 2019.

The sector-based variation in the final energy consumption of electricity in Turkey, according to 2018 data, is represented in Figure 1.5. As indicated in the figure, 23.02% of the consumed energy is utilized in the field of residential and services. When the details of this usage are examined, it is seen that 22% of the electricity is consumed for residential hot water needs and 48% is consumed for residential space heating needs, so in total, 70% of the electricity is utilized for living in comfort conditions (Figure 1.6). In this case, the huge electricity

consumption rate for these needs creates the possibility of high savings with the integration of the appropriate renewable energy source systems into domestic application areas.

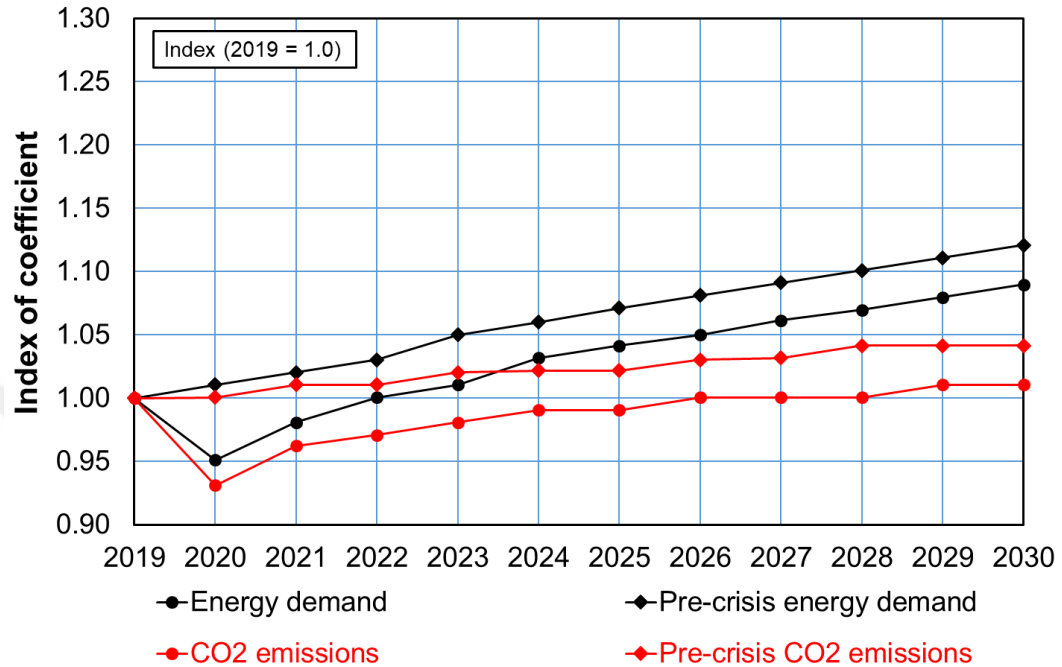


Figure 1.4 The trajectory of global energy demand and CO₂ emissions (modified from IEA, 2020a)

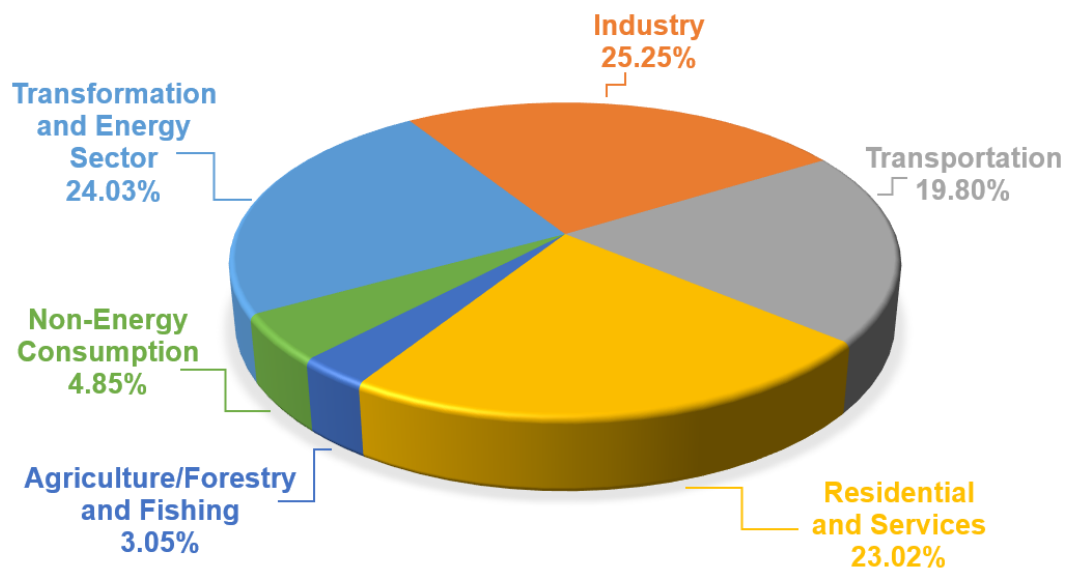


Figure 1.5 Electricity final consumption by sectors in Turkey (modified from IEA, 2021)

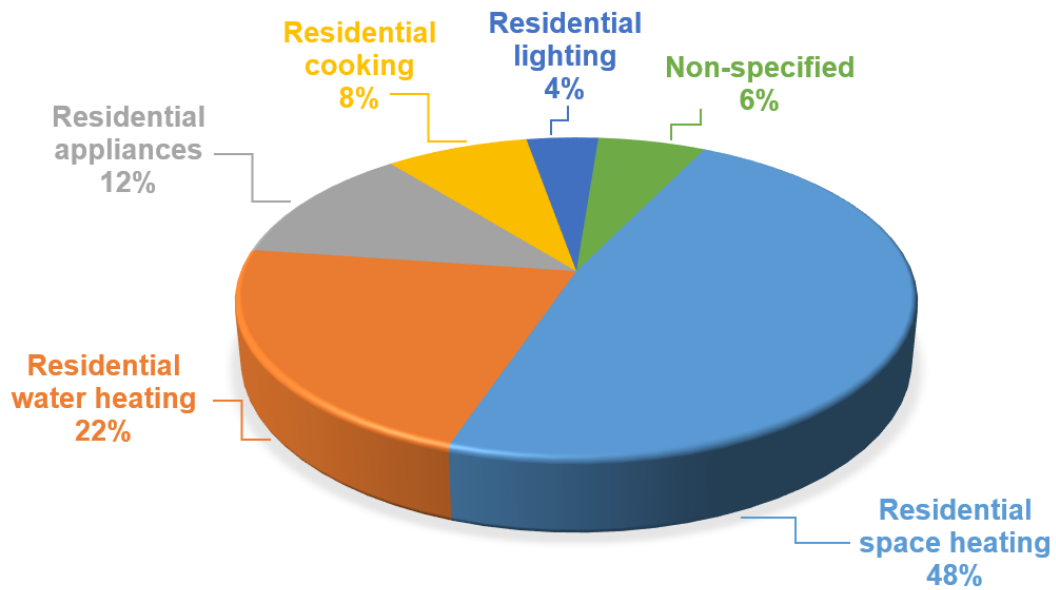


Figure 1.6 The variation of energy consumption in residential areas by utilization (modified from IEA, 2021)

In addition to STEPS, the IEA developed the Sustainable Development Scenario (SDS). According to the SDS, the milestones to be achieved in accordance with the essential alteration that the global energy system has to go through are determined. Additionally, the SDS is in full compliance with the Paris agreement aimed at mitigating climate change. The supply methods of domestic heating and hot water supply in newly built houses are represented in Figure 1.7. It can be seen from the figure that the most frequently utilized method is fossil-based, and it was preferred by 56.5% in 2019. However, according to the SDS, this rate should be 40.2% in 2025 and 35.3% in 2030. It is stated that the rate of utilization of fossil-based resources should decrease 21.2% within ten years. In addition, the heat pump usage was 5% in 2019; on the other hand, its usage rate should increase to 15.4% in 2025 and 22.1% in 2030.

As mentioned in detail above, recently, the given importance to renewable energy resources, such as solar and wind energy, has also been increasing (see Figure 1.1) because of the well-known irreversible side-effects of the fossil-based energy production systems on the environment. One of the most significant problems of utilizing renewable energy resources is the sustainability of these resources (Dincer & Ezan, 2018).

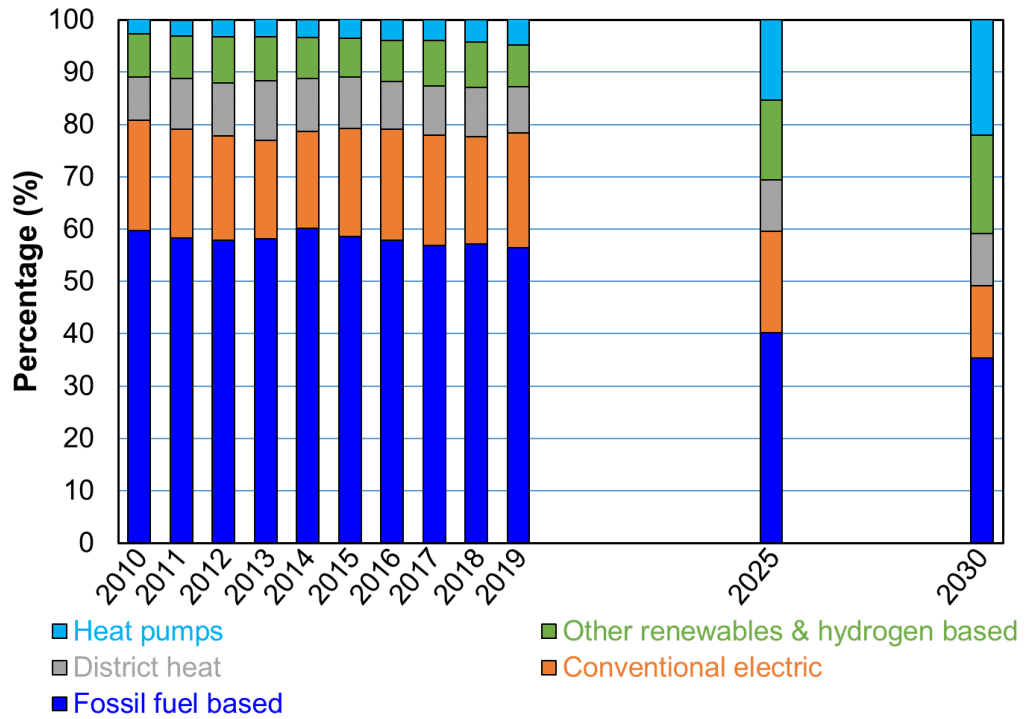


Figure 1.7 The supply methods of domestic heating and hot water needs in newly built houses (modified from IEA, 2020b)

Energy storage techniques maintain the sustainability of intermittent and unsteady renewable energy sources and ensure an enhancement of efficient and effective usage of existing energy conversion systems (Dincer & Ezan, 2018). Sustainable, efficient, economical, and environmentally friendly applications can be achieved by designing qualified thermal energy storage (TES) systems and integrating them into existing energy systems (Dincer & Ezan, 2018). Energy could be stored in many forms, such as mechanical, thermal, chemical, electrical, or electrochemical (Dincer & Rosen 2010). In TES methods, energy is charged or discharged by (i) the variation of the temperature (*sensible*), (ii) the phase change (*latent*) of the storage material, or (iii) thermochemical reactions. In the sensible heat TES method, thermal energy, either at low or high temperature, is stored in a storage medium or regained by varying the solid or liquid storage medium's temperature. On the other hand, in the latent heat TES method, released or absorbed energy is utilized during liquid-solid or liquid-gas phase-change processes. In the thermochemical TES method, a higher amount of thermal energy can be stored with reversible endothermic or exothermic reactions (Dincer & Ezan, 2018). The volumetric TES capacity of different types of TES materials is illustrated in Figure

1.8 (Lizana, Chacartegui, Barrios-Padura, & Valverde, 2017). Despite having the highest energy density, the thermochemical TES technique is still in the laboratory stage and does not have practical applications like sensible or latent heat TES techniques (Yan, Wang, Clough, & Anthony, 2020). Although the sensible heat TES method has is the cheapest and the most mature one, the main handicap of this method is temperature stability during the regain process; however, in the latent heat TES method, the discharge temperature of the system remains almost constant for a certain period of time (Alva, Liu, Huang, & Fang, 2018).

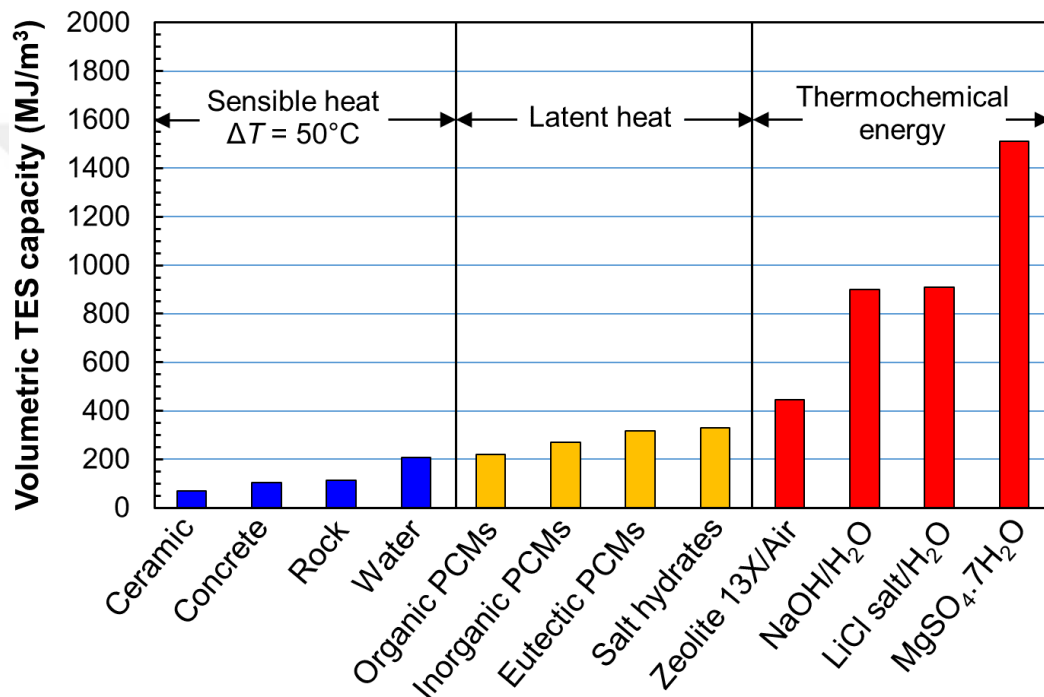


Figure 1.8 The volumetric TES capacities of different types of TES methods according to the storage materials (modified from Lizana et al., 2017)

1.1 Literature Review

In the literature, the storage of thermal energy is generally studied as sensible and latent heat storage. In the latent heat TES technique, energy is stored in phase change materials. However, it has limited applications, and selecting the appropriate phase change material is crucial. Sensible heat TES is a mature and economically feasible technique and has more practical applications compared to latent heat TES. Sensible heat TES systems store the thermal energy only via

temperature variations; that is, to store high amounts of thermal energy, the mass of the storage medium should also be significant, and the storage medium should undergo high-temperature variations. Recently, sensible and latent heat storage, hybrid, in a packed bed is the most tempting thermal energy storage method (Dincer & Ezan, 2018).

The studies related to the TES systems mostly deal with energy-based performance indicators such as energy efficiency, spatial and temporal temperature distributions, and stored energy. Besides, exergy analysis, or the 2nd law analysis, deals with both the quantity of the energy and the quality of the stored thermal within the TES unit (Dincer & Ezan, 2018). Thus, exergy-based analyses represent an in-depth understanding to evaluate and compare system performance indicators more adequately. Additionally, exergy analysis can help to link the economic and environmental aspects of the energy systems and allows to optimize the design and working parameters from a more comprehensive perspective (Dincer & Ezan, 2018). Verma & Singal (2008) mentioned that exergy analysis provides a good understanding of the efficiency of the thermal energy storage system. Bindra, Bueno, Morris, & Shinnar (2013) compared the discharging performances of two different packed-beds as sensible heat TES and latent heat TES unit. Exergy-based considerations showed that the sensible heat TES system could provide much exergy recovery when it is compared with the latent heat TES. It is indicated that the rate of axial dispersion and heat loss through the ambient significantly reduces the exergy efficiency. Thermal stratification within the storage tank is another critical parameter that significantly affects the exergy efficiency (Rosen, 2001). An adequately controlled thermal stratification can provide an enhanced exergetic storage capacity of the TES system even though the amount of energy remains unchanged.

Singh, R. P. Saini, & J. S. Saini (2010) stated that evaluating a packed bed TES system's dynamic performance integrated with a solar heating system could only be assessed by considering the actual meteorological data as boundary conditions. TES methods are widely incorporated with solar collectors to eliminate the intermittency of solar energy. Additionally, Yang, Zhang, & Xu (2014) proposed a

numerical model of a packed-bed latent heat TES tank integrated with a flat plate solar collector. Results showed that instantaneous and average collector efficiencies vary in a wide range throughout the day due to the variable meteorological data.

Latent heat TES techniques have been used in many applications for lots of purposes, such as for solar energy storage systems to eliminate the imbalance between the energy source and the demand (Badiei, Eslami, & Jafarpur, 2020; Palacio, Rincón, & Carmona, 2020) and enhancing system performance by waste heat recovery of power generation systems (White & Sayma, 2020; D. Li, Hu, D. Li, & Wang, 2019) or industrial furnaces (Ortega-Fernández & Rodríguez-Aseguinolaza, 2019; Royo et al., 2019). Rectangular and cylindrical type tanks are generally used as geometric configurations of the TES unit in literature for solar applications. The rectangular tanks are typically used in sensible heat storage applications, and the cylindrical tanks are commonly preferred in latent heat storage systems (Mao, 2016). In cylindrical tanks, two popular configurations are mostly used as shell-and-tube (Liu, Riahi, Jacob, Belusko, & Bruno, 2020; Yang, Yu, Xiao, Hu, & He, 2020) and packed beds (Afshan, Selvakumar, Velraj, & Rajaraman, 2020; Ahmed, Elfeky, Lu, & Wang, 2020). In a recent review study by de Gracia & Cabeza (2017), it is stated that using spherical capsules in packed-bed latent heat TES systems is worth investigating, as it provides the highest heat transfer area between the working fluid and the phase change material (PCM).

1.1.1 Sensible Heat Packed Bed TES Systems

The recent numerical/experimental studies on sensible heat TES units filled with spherical or cylindrical filler materials are summarized in Table 1.1. Here, the selected works' assessments are represented by considering some primary design/working conditions, the data reduction techniques, and the main conclusions. The selected items are (i) Boundary conditions, (ii) Working parameters, (iii) Exergetic aspects, (iv) Analyses mode, and (v) Major findings.

Many studies on sensible heat packed beds are available in the literature. The number of power plants that include TES systems increases in recent years as TES

provides a steady thermal energy output in an extended duration (Liao et al., 2018; Fasquella, Falcoz, Neveu, & Hoffmann, 2018; Li, Xu, Song, Wang, & Yuan, 2017). Li et al. (2017) investigated the two-tank indirect thermal energy storage system integrated with the solar field and power plant. However, the constant inlet temperature is defined as a boundary condition for both the charging and discharging processes. The variation in solar radiation is included in the model by varying the mass flow rate of HTF. Liao et al. (2018) examined complete charging/discharging processes and the cyclic performances of two different packed-bed TES systems that include (i) rock only and (ii) rock/PCM. A capacity ratio is defined as the ratio of the maximum stored thermal energy in the tank during the charging process to the theoretical storage capacity of the storage tank. The charging/discharging efficiencies and capacity ratios are evaluated for different cut-off temperature values. A packed bed TES system with alumina storage materials is numerically modeled by Fasquella et al. (2018) for a power plant with parabolic trough solar collectors. While the cyclic system performance is examined in energetic and exergetic aspects, variable nature effects are also included in the model. The monthly-averaged electricity productivity of the power plant is obtained throughout the year. Abdulla & Reddy (2017) numerically studied the charging and discharging performances of a rock-bed TES system integrated with concentrating solar power plant. Nevertheless, the variation of solar radiation, which is a variable nature effect, has been neglected in the parametric analyses. Agrawal, Gautam, Kunwar, Kumar, & Chamoli (2018) experimentally investigate the impact of storage material with different grooved structures on the only energetic charging performance of the packed bed TES system. Nusselt number and friction factor coefficients correlate with void fraction, storage material aspect ratio, and Reynolds number as a performance indicator. Gautam & Saini (2020) experimentally examined the effect of pore structures in the spherical storage materials on the thermal and hydraulic performances of the solar TES system. Despite the variable nature effect is considered in the charging mode, second-law analyses are not performed. The authors indicated that the storage materials with pores enhance integrated thermo-hydraulic parameters of the packed bed TES system compared to the TES systems with having smooth surface storage material. Elouali et al. (2019) compared the four numerical methods used in modeling packed bed TES tanks,

such as the single-phase model, Schumann's model, continuous solid phase model, and model with the thermal gradient in the solid particles. The authors expressed that the most significant parameter in determining the appropriate model is the Biot number. When the Biot number is smaller than 0.1, the storage material's thermal gradient needs to be taken into account. Yang, Bai, Y. Wang, & Z. Wang (2019) substantially focused on the storing period of the packed bed TES system. To do so, not only storing process is investigated, cyclic behaviour of the TES system integrated with CSP and Brayton cycle is analyzed, including variable solar radiation. The decay of thermal stratification on the storing period is parametrically studied. Lugolole, Mawire, Lentswe, Okello, & Nyeinga (2018) conducted experimental analyses to compare three sensible heat TES systems. The charging performance of these systems is compared on energetic and exergetic aspects. On the other hand, the systems to be proposed as solar TES system technology have been studied with constant HTF inlet temperature. Elfeky, Mohammed, & Wang (2021) numerically examined the packed bed TES tank for CSP with constant HTF inlet temperature at the charging process. Spherical storage materials are randomly distributed inside the TES tank. The effect of airflow distributor at the inlet section is studied on the wall effect, and it is found that TES tank design has a critical mass flow rate of HTF to maximize the performance indicators.

Bruch, Fourmigué, & Couturier (2014) and Bruch, Molina, Esence, Fourmigué, & Couturier (2017) used a mixture of silica-gravel and silica-sand as a filler material to minimize the porosity in packed-bed. Such an approach could significantly reduce the cost of the storage unit since the HTF is often more expensive than sensible storage material. The use of rock as a storage medium has some advantages, such as low cost, non-toxic, non-flammable, and rocks do not require additional shell material that provides improved heat transfer (Alva et al., 2017). Grirate et al. (2016) stated that quartzite rock has a high potential to be used as storage material in direct contact with HTFs such as thermal oils. Mertens, Alobaid, Frigge, & Eppe (2014) also studied the quartzite-rock bed with a silica sand refinement thermal energy storage method for solar tower power plants application. The authors developed a numerical model and validated it against the results of the experimental work of Meier, Winkler, & Willemin (1991). However,

Mertens et al. (2014) defined constant inlet conditions in both charging and discharging processes.

Silicon liquids, *e.g.*, Syltherm 800, are commonly used as HTF for solar heating and sensible heat TES applications (Peiró, Gasia, Miró, Prieto, & Cabeza, 2017; Arena, Casti, Gasia, Cabeza, & Gau, 2018; Ouagued, Khellaf, & Loukarfi, 2013). Recently, Ouagued et al. (2013) investigated the potential of solar radiation in Algeria and the performance of the solar parabolic trough collector under climatic conditions using different thermal oils. It is concluded that Syltherm 800 represents the best annual averaged thermal capacity compared to the rest of the silicon HTFs studied in Ouagued et al. (2013).

1.1.2 Latent Heat Packed Bed TES Systems

The recent numerical/experimental studies on latent heat TES units filled with spherical encapsulated PCMs are summarized in Table 1.2. Here, the selected works' assessments are represented by considering some primary design/working conditions, the data reduction techniques, and the main conclusions. The selected items are (i) PCM arrangements, (ii) Boundary conditions, (iii) Volume fractions, (iv) Working parameters, (v) Physical properties, (vi) Influence of natural convection, (vii) Exergetic aspects, and (viii) Major findings. In Table 1.2, Li, Jin, Ma, & Yuan (2018) and Afshan et al. (2020) only deal with single-type PCM usage, and Mao & Zhang (2020) and Khor, Sze, Li, & Romagnoli (2020) consider only the cascaded-type PCM arrangement. Wu, Xu, & He (2016), Elfeky, Ahmed, & Wang (2018), and Ahmed et al. (2020) consider both single and cascaded types of PCM arrangements and compare the performances.

Li et al. (2018) evaluated the cyclic energetic performance of a packed bed TES unit filled with a single-type PCM ($T_m = 395.1^\circ\text{C}$) both experimentally and numerically. Parametric numerical analyses are carried out to investigate the effects of mass flow rate and inlet temperature of the heat transfer fluid (HTF) and the storage material's diameter on the TES system's charging/discharging performance and time-wise variation of PCM center temperature. The thermal performance of

three different geometric configurations of TES units, packed bed, shell-and-tube, and finned-tube types, are also compared. In another study, Afshan et al. (2020) experimentally analyzed the effect of storage tank aspect ratio on the latent heat TES system's charging performance for solar applications. The variable nature of solar power is ignored in this work, and a constant heating load is defined during the charging experiments. Mao & Zhang (2020) considered only cascade-type PCM in packed bed TES systems. The authors parametrically investigated the cyclic thermal performance of the three-cascaded PCM TES tank. The effects of packed bed void fraction, the storage material's diameter, and the tank aspect ratio are numerically analyzed. In another study, Khor et al. (2020) conducted numerical analyses to optimize the packed bed TES system. The effect of the diameter of the storage material and the storage capacity of cascade layers was parametrically obtained. In the design of cascade PCM arrangement configurations, the equal distribution of volume fraction and the effect of volume fraction of each layer are examined. In addition to the melting temperatures, the other thermophysical properties, *i.e.*, latent heat and specific heat, of PCMs that using different cascade layers are also varied in the study. The TES system's cyclic energy efficiency can be improved by increasing the PCM's thermal storage capacity with the lowest melting temperature. Some studies compared the using PCMs single-type and cascade types in the packed bed TES systems in the literature. Wu et al. (2016) numerically compared the cyclic performance of the single and cascaded types of the latent heat TES unit integrated with concentrating solar power (CSP). The authors concluded that using a single type PCM arrangement is not appropriate due to the long charging time and low charging ratio. However, in the simulations of single-type configuration, only the PCM with the highest melting temperature is used, and a constant inlet temperature of the HTF is defined for the charging period. Elfeky et al. (2018) numerically compared the charging and discharging performances of single and cascaded-type PCM arrangements in a latent heat TES system for CSP. The authors claimed that despite the effect of usage of single-type or cascaded-type PCMs inside the packed bed LHTES system on system performance is studied in the literature, the difference in the temperature profiles and phase change processes has not been extensively investigated. The authors commented that using cascaded-type PCMs has higher charging and discharging

energy efficiencies than using single-type ones. However, in the numerical analyses, the HTF's inlet temperature is kept constant during the charging and discharging periods. The performed study does not seem very convenient to compare the effect of PCM arrangement on system performance. In the analyses, specific heat, density, latent heat of fusion, and thermal conductivity values are varied for each PCM layer. Ahmed et al. (2020) numerically investigated the sensible/latent hybrid TES unit's performance filled by encapsulated PCMs for medium-temperature applications. In addition to single-type configuration, cascaded types that have three different volume fractions are also examined.



Table 1.1 A comparative review on recent works on packed bed sensible heat TES systems

References	Boundary Conditions	Parameters	Operating Mode	Exergetic Aspect	Major Findings
Mertens et al. (2014)	Variable nature effects neglected	Storage material diameter, tank height to diameter ratio, alternative operating scenarios	Cyclic (Charging, storing, and discharging)	Not included	The rock-bed TES technique is considered much more economically viable than the molten salt TES technique. Increasing the ratio of H_T/D_T and decreasing storage material diameter increases thermal efficiency.
Abdulla et al. (2017)	Variable nature effects neglected	Storage material diameter, the mass flow rate of HTF, operating temperature range	Charging and discharging	Not included	Smaller storage material diameter provides sufficient hot water for a longer period. Increasing the mass flow rate of HTF decreases discharging efficiency. The operating temperature range is found as the most dominant parameter of the TES system.
Agrawal et al. (2018)	Variable nature effects neglected	Storage material structure, void fraction, mass flow rate, and inlet temperature of HTF	Only charging	Not included	With the increasing Re number, heat transfer increases. Maximum heat transfer occurs when the aspect ratio is 1.0. Heat transfer reduces when the aspect ratio is less than or greater than this value. The Nusselt number and friction factor coefficients are proposed.
Fasquelle et al. (2018)	Variable nature effects included	Operating month, TES system type	Cyclic (Charging, storing, and discharging)	Included	For thermocline design, 11% of the stored energy cannot be used due to outlet temperature drops rapidly and reaches the cut-off value. However, it is concluded that thermocline technology is suitable to decrease the cost of storage in concentrated solar power plants.
Lugolole et al. (2018)	Variable nature effects neglected	Mass flow rate of HTF, the diameter of storage material	Only charging	Included	Decreasing the diameter of the mass flow rate increases the heat transfer, energy, and exergy rates. No significant effect of mass flow rate variation on exergy rate is observed.
Elouali et al. (2019)	Variable nature effects neglected	Mathematical models, storage material properties and diameter, porosity, the mass flow rate of HTF	Only charging	Not included	Mass flow rate of HTF is the dominant parameter affecting thermal performance. Increasing porosity increases charging time. Decreasing the diameter of the storage material increases thermal performance.
Yang et al. (2019)	Variable nature effects included	Effective thermal conductivity of solid, natural convective heat transfer coefficient between HTF and sphere, wall heat loss coefficient	Cyclic (Charging, storing, and discharging)	Not included	Thermal diffusion and heat loss have a greater share in worsening thermal stratification than natural convection during storing process.
Gautam & Saini (2020)	Variable nature effects included	Mass flow rate of HTF, storage material structure	Only charging	Not included	Using spherical storage materials with pores shows better thermal and hydraulic performance. With the increasing mass flow rate of HTF, thermal and hydraulic parameters increase. The effects of different pore parameters are extensively investigated.
Elfeky et al. (2021)	Variable nature effects neglected	Mass flow rate of HTF, inlet section flow distributor	Charging and discharging	Not included	Increasing the mass flow after a specific mass flow rate has a negative effect on the total efficiency due to the pressure drop for the air rock-bed TES system. An optimally designed rock-bed TES unit can be a proper choice for CSP plants.

Table 1.2 A comparative review on recent works on packed bed latent heat TES systems

References	Boundary Condition	Parameters	PCM arrangement	Volume fraction	Physical properties	Natural convection	Operating Mode	Exergetic aspect	Major findings
Wu et al. (2016)	Variable nature effects neglected	Only PCM arrangement is considered as a design parameter	Single & 3- and 5-cascaded types tanks are analyzed. For a single type, only the highest melting temperature PCM is studied	Equal distribution of system volume in cascaded types	Only melting temperature is varied in cascade analysis	Considered	Charging and discharging	Not included	5 cascaded TES system shows very promising performance since it has a high charging/discharging rate
Elfeky et al. (2018)	Variable nature effects neglected	Only PCM arrangement is considered as a design parameter	Single & 3-cascaded types TES tanks are analyzed	Equal distribution of system volume in cascaded type	PCMs with different thermal properties	NA	Charging and discharging	Included	3-cascaded type has the highest heat transfer rate, energy, and exergy efficiencies for charging and discharging processes
Li et al. (2018)	Variable nature effects neglected	Mass flow rate and inlet temperature of HTF, sphere diameter of PCM	Only a single type of PCM is analyzed	–	–	Neglected	Charging and discharging	Not included	Three different TES systems are analyzed, and the packed bed TES system shows better thermal performance. A small diameter is found optimum for the packed bed TES system because increasing the diameter increases charging time
Afshan et al. (2020)	Variable nature effects neglected	Only the effect of tank aspect ratio is analyzed	Only a single type of PCM is analyzed	–	–	Only experimental studies are conducted	Only charging	Not included	Energy storage performance decreases with increasing the aspect ratio of the TES tank
Ahmed et al. (2020)	Variable nature effects neglected	Volume fraction of cascade layers	Single & 3-cascaded types of TES tanks are analyzed	The volume fraction of each cascade layer is a parameter	PCMs with different thermal properties	NA	Charging and discharging	Included	Cascade types shows better performance. Determining optimum melting temperature and cascade layer thickness is critical in obtaining an efficient and cost-effective TES system
Khor et al. (2020)	Variable nature effects neglected	Capsule diameter of PCM and TES capacity of cascade layers	3-cascaded type TES tanks are analyzed	The volume fraction of each cascade layer is a parameter	PCMs with different latent heats	Considered	Charging and discharging	Not included	The highest cyclic efficiency can be achieved by increasing the TES capacity of the PCM that has the lowest melting temperature.
Mao & Zhang (2020)	Variable nature effects neglected	TES tank aspect ratio, sphere diameter of PCM, porosity	3-cascaded type TES tanks are analyzed	Equal distribution of system volume	PCMs with different thermal properties	NA	Charging and discharging	Included	For a fixed tank volume, tank shape has no effect on storage capacity. As the diameter of the storage material decreases, energy storage capacity increases

1.2 Motivation and Objectives of the Dissertation

The main objectives of the present study and contributions to the literature are as below:

- ***Integrated storage system:*** In the literature, the TES tanks are mostly separated from the load side and simulated with defining a presumed boundary condition. Herein, the storage tank model is integrated with a zero-dimensional flat plate solar collector to consider the charging process more realistically. Additionally, for the discharging period, the promising method of integrating the TES tank with a heat pump can enhance the system performance compared to a conventional air-source heat pump and reduce the power consumption adopted into the mathematical model.
- ***Variable boundary conditions:*** Solar energy has a dynamic nature during the day time. The incoming solar radiation, wind speed, and ambient temperature excessively affect the charging performance throughout the working period. Even though the variable boundary conditions significantly impact evaluating the realistic performance, literature review on the packed-bed sensible and latent heat TES systems reveals that the variable boundary conditions are not taken into account, and constant heat flux or constant inlet temperatures are mostly preferred in simulations. Herein, actual meteorological data of Izmir City, Turkey, is adopted to the integrated mathematical model in evaluating charging performance as boundary conditions for four heating season months.
- ***Exergetic analysis:*** In the literature, the exergetic aspects are generally not included when determining the performance of the TES system and comparing parametric results. Additionally, in some studies, exergetic aspects are considered for stand-alone TES tanks, disregarding the solar power's variable nature throughout the day. However, herein, the TES unit's charging performance integrated with a flat plate solar collector, storage performance and discharging performance integrated with a heat pump are investigated

under dynamic boundary conditions by considering the first-law and second-law aspects.

- ***Comprehensive parametric analysis:*** The influences of (i) the mass flow rate of the HTF, (ii) the diameter of storage material, and (iii) the height of the storage tank are investigated to obtain the dynamic charging performance of the sensible heat TES system in January. For January, a total of 36 parametric runs are conducted to determine the optimum working and design parameters of the flat plate solar collector integrated sensible heat TES tank. After that, energetic and exergetic performances are also evaluated for the other heating season months. In the literature, detailed parametric analyses are performed for single and cascaded-types PCM arrangements in separate studies. The investigated parameters in studies related to using single- and multi-types PCM arrangements are limited and insufficient to withdraw comparisons. For each of the four selected months, 84 parametric analyses are carried out in the current work, *i.e.*, a total of 336 runs are conducted. The effects of (i) the mass flow rate of HTF, (ii) the diameter of the storage material, and (iii) the PCM arrangements are evaluated on the heat transfer characteristics and the integrated system's charging and cyclic performances.

CHAPTER TWO

MATERIAL & METHOD

In this chapter, at first, the definition of the problem of this dissertation is explained. Then, the numerical models of the flat plate solar collector, packed bed TES tanks, and heat pump are given in detail. After that, the description of the integration of these models is clarified in the solution procedure section. Lastly, the evaluation of the system indicators represented in the Results & Discussion chapter, *i.e.*, Chapter 3, is explained in the data reduction section.

2.1 Definition of the Problem

In this dissertation, an in-house 1D transient code is developed to numerically evaluate the performance of the packed bed TES tank, which is integrated with a flat plate solar collector and a heat pump. In order to do this, firstly, the dynamic charging performance of the sensible heat and latent heat packed-bed TES tanks are analyzed in charging mode only, and finally, the cyclic charging and discharging performance of the latent heat packed bed TES tank is examined. The modeled TES tanks are cylindrical and consist of aligned spherical storage materials. The charging is provided with a solar collector during the periods when solar energy is effective. The stored thermal energy is then utilized as a heat source for the evaporator of the heat pump in the discharging process when residential heating is needed. A schematic description of the charging and discharging periods of the TES system consisting of a flat plate solar collector, a packed bed TES tank, and a heat pump is illustrated in Figure 2.1. Both charging and discharging periods are closed-loop, and the same HTF is utilized. In the charging period of the TES system, hot HTF leaving the solar collector enters the TES tank from the top surface, and relatively cold HTF exiting the TES tank from the bottom surface enters the solar collector. However, due to obtaining benefit from the inhomogeneous temperature distribution that may occur in the TES tank, the inlet and outlet sections of the closed-loop are reversed in the discharging period. In that case, the warm HTF leaves the tank from the top surface, and after the heat removal process that takes place at the evaporator component of the heat pump, the cold HTF enters the TES

tank from the bottom surface. The charging process is active when the solar radiation is high enough to increase the thermal energy of the TES tank. On the other hand, when the residential heating demand arises, the closed-loop discharging process becomes active. Additionally, in all other situations, the TES system is in storage period, *i.e.*, idle period, which means pumps are not operating, and the TES tank is subjected to heat loss/gain only due to the interactions with the ambient air.

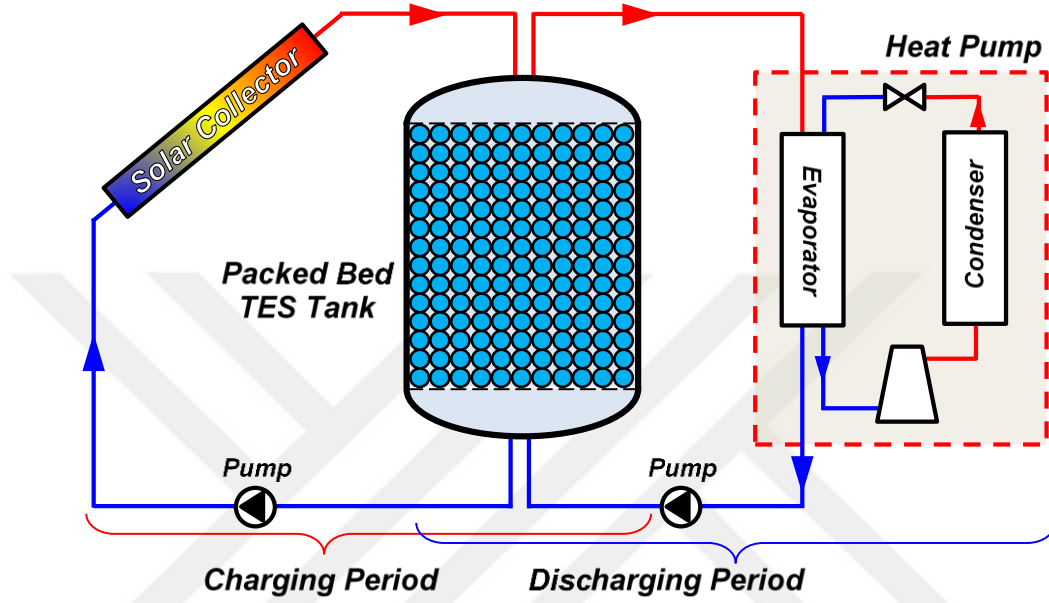


Figure 2.1 Schematic representation of the integrated TES system

Real meteorological data are adopted in the numerical models due to consider the dynamic nature effects for İzmir province, Turkey. The time-wise variations of the monthly-averaged solar radiation, I_{solar} , wind speed, u_w , and ambient temperature T_{amb} , are illustrated in Figure 2.2 for the heating season months of November, December, January, and February. The flat plate solar collector faces south at an angle of 30.3° (tilt angle) to get the maximum benefit from the solar radiation (Ülgen, 2006).

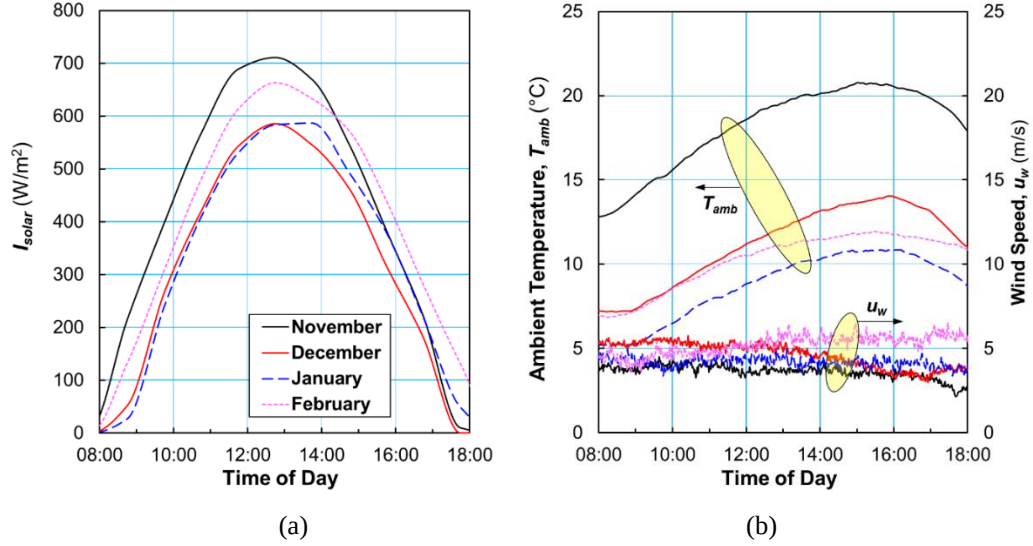


Figure 2.2 The meteorological data for İzmir, Turkey (a) solar radiation, (b) ambient temperature and wind speed (Alptekin & Ezan, 2020)

2.2 Numerical Models

In this dissertation, there are mainly three mathematical models. These are the flat plate solar collector, TES tank, and heat pump models. While the flat plate solar collector and TES tank models are operated collectively in the charging process, the integration of the TES tank model and heat pump model functions in the discharging process. Besides, only the TES tank model is active in the storage process.

The two different TES mechanisms, sensible heat and latent heat, are examined. Therefore, the TES tank model consists of two independent parts. These two particular TES systems can be analyzed by changing the type of storage material. In the sensible heat TES system, spherical quartzite-rock is used as a storage material. Besides, in the latent heat TES system, spherical encapsulated paraffin-based PCM is utilized.

2.2.1 Flat Plate Solar Collector Model

According to the literature review in Section 1.1, it can be realized that the dynamic nature effect of meteorological conditions is generally neglected in

designing TES systems. In this dissertation, a packed bed TES tank is integrated with a flat plate solar collector to include these dynamic effects in the charging period. The schematic representation of the liquid-type solar collector with a single glass cover is illustrated in Figure 2.3. The flow distribution inside the collector is parallel Z-type. The absorber plate is made of copper with a thickness of 0.5 mm. A total of 10 solar collectors is connected as a serial arrangement. The design parameters of the flat plate solar collector are given in Table 2.1.

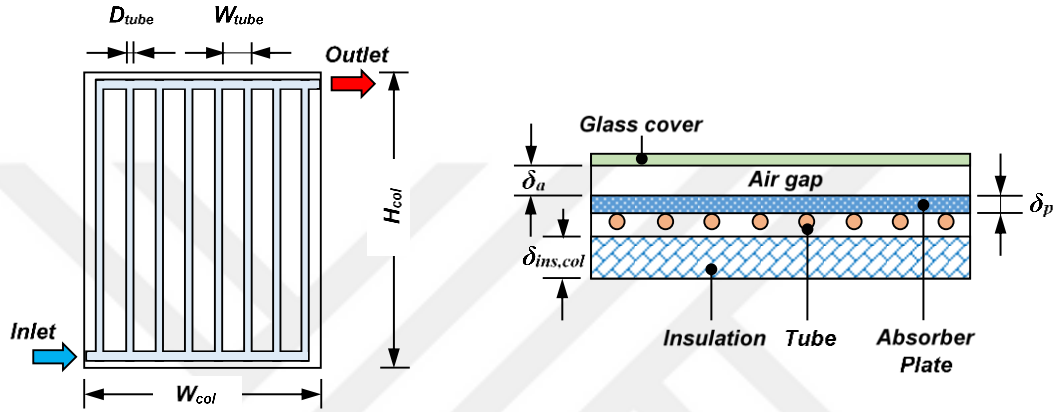


Figure 2.3 Schematic representation of the flat plate solar collector (Alptekin & Ezan, 2020)

Table 2.1 Design parameters of the flat plate solar collector

Parameter	Value	Parameter	Value
N_{col}	10	H_{col} (m)	2
N_c	1	W_{col} (m)	1
δ_p (m)	0.5E-3	ε_p (-)	0.98
k_p (W/m K)	385	ε_c (-)	0.88
D_{tube} (m)	0.0127	τ (-)	0.9
W_{tube} (m)	0.12	α (-)	0.97
ϕ (°)	30.3	$\delta_{ins,col}$ (m)	0.05
δ_a (m)	0.025	$k_{ins,col}$ (W/m K)	0.045

The mathematical model of the flat plate solar collector is developed based on the procedure that is represented by Duffie & Beckman (2013). The model neglects merely the radiative and convective losses from the lateral surfaces of the collector,

and the energy equation for the flat plate solar collector is reduced in the following form

$$\frac{d}{dt}(mc\bar{T})_{HTF} = \dot{Q}_u + [\dot{m}c(T_{col,in} - T_{col,out})]_{HTF} \quad (2.1)$$

where $\dot{Q}_u$ is the useful heat that is transferred to the working fluid (or HTF). The useful heat can be determined as

$$\dot{Q}_u = A_{col} F_R [(I_{solar} \tau \alpha) - U_L (T_{col,in} - T_{amb})] \quad (2.2)$$

where U_L and F_R represent the overall heat loss coefficient of the collector and the collector heat removal factor, respectively. As the heat losses through lateral surfaces of the collector are neglected, the overall heat loss coefficient of the collector (U_L) includes the top and bottom components as followings

$$U_L = U_t + U_b \quad (2.3)$$

where U_t and U_b represent the heat loss coefficients of the collector from the top and bottom surfaces, respectively. The electrical resistance analogy, *i.e.*, the thermal network method, is used to calculate U_t and U_b . The thermal network of the single-cover flat plate solar collector consists of top and bottom components (Figure 2.4). On the top side of the collector, the radiative and convective heat losses occur from the absorber plate through the glass cover. Besides, solely the conductive heat loss is taken into account at the bottom of the collector.

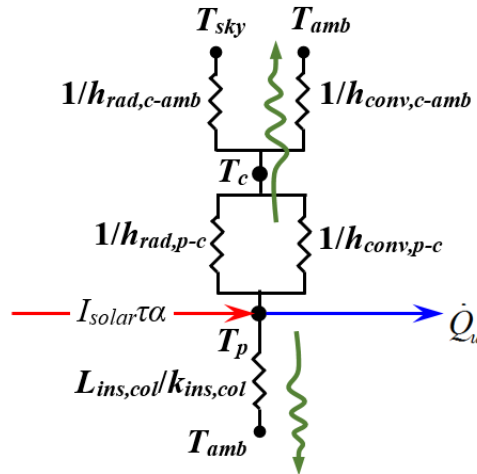


Figure 2.4 Thermal network of the flat plate solar collector

Here, T_p and T_c represent the temperature of the absorber plate and glass cover, respectively. The absorber plate absorbs solar energy of $I_{solar} \tau \alpha$. A portion of the absorbed energy is lost through the top and bottom surface, and the remaining is transferred into HTF as useful energy gain, $\dot{Q}_u$. The energy loss through the top is the result of convection and radiation heat transfer mechanisms between parallel plates. Since the internal energy changes of the absorber plate and glass cover are neglected, the steady-state thermal energy transfer between the absorber plate and glass cover is also equal to the energy loss to the surroundings from the glass cover. The heat loss through the top surface per unit area is equal to the heat transfer from the absorber plate to the glass cover. The heat loss coefficient of the top surface of the collector, U_t , can be calculated as

$$U_t = 1/R_{tot,t} \quad (2.4)$$

The top thermal resistance, $R_{tot,t}$ is defined as

$$R_{tot,t} = (h_{rad,p-c} + h_{conv,p-c})^{-1} + (h_{rad,c-amb} + h_{conv,c-amb})^{-1} \quad (2.5)$$

The radiation heat transfer coefficient between glass cover and ambient can be defined as

$$h_{rad,c-amb} = \sigma \varepsilon_c (T_c^2 + T_{sky}^2)(T_c + T_{sky}) \quad (2.6)$$

where ε_c is the emissivity of the glass cover and T_{sky} is the sky temperature and can be calculated from the ambient temperature, T_{amb} ($^{\circ}\text{C}$), by using the correlation that is suggested by Fuentes (1987)

$$T_{sky} = 0.037536 T_{amb}^{1.5} + 0.32 T_{amb} \quad (2.7)$$

The convective heat transfer coefficient between glass cover and ambient depends on the wind speed, u_w . Watmuff, Charters & Proctor (1977) expressed the relationship between the wind speed and wind convection coefficient, h_w , as

$$h_w = 2.8 + 3.0 u_w \quad (2.8)$$

The radiative heat transfer coefficient between the absorber plate and glass cover is defined by (Incropera, DeWitt, Bergman & Lavine, 2006)

$$h_{rad,p-c} = \frac{\sigma (T_p^2 + T_c^2)(T_p + T_c)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} - 1} \quad (2.9)$$

The convection heat transfer coefficient between absorber plate and glass cover can be defined as

$$h_{conv,p-c} = \frac{Nuk_{air}}{\delta_a} \quad (2.10)$$

where δ_a is the air gap thickness, and k_{air} is the thermal conductivity of the air. Free convection occurs in the air gap, and for tilted cavities, Nusselt number can be calculated from the correlation that is proposed by Hollands, Unny, Raithby & Konicek (1976)

$$Nu = 1 + 1.44 \left[1 - \frac{1708}{Ra_L \cos \phi} \right]^+ \left[1 - \frac{1708(\sin 1.8\phi)^{1.6}}{Ra_L \cos \phi} \right] + \left[\left(\frac{Ra_L \cos \phi}{5830} \right)^{1/3} - 1 \right]^+ \quad (2.11)$$

where ϕ is the collector tilt, and according to Ülgen (2006), the annual optimum collector tilt angle of Izmir City is 30.3° . $[]^+$ is defined as

$$[X]^+ = (|X| + X) / 2 \quad (2.12)$$

where X is any quantity. Ra_L represents the Rayleigh number depends on air gap thickness, and Ra_L can be expressed as

$$Ra = \frac{g\beta\Delta T\delta_a Pr}{\nu^2} \quad (2.13)$$

where g , β , and ΔT are the acceleration of gravity, coefficient of the thermal expansion of air, and temperature difference between absorber plate and glass cover, respectively. Additionally, Pr and ν represent the Prandtl number and kinematic viscosity of air, respectively. The heat loss coefficient from the bottom surface can be defined as

$$U_b = 1/R_{tot,b} = k_{ins,col} / \delta_{ins,col} \quad (2.14)$$

where $k_{ins,col}$ and $L_{ins,col}$ represents the thermal conductivity of the insulation material and its thickness, respectively. On the other hand, the collector heat removal factor, F_R , is defined as

$$F_R = \frac{\dot{m}c}{A_{col}U_L} \left[1 - \exp\left(-\frac{A_{col}U_L F'}{\dot{m}c}\right) \right] \quad (2.15)$$

where F' is the collector efficiency factor and is expressed as

$$F' = \frac{1/U_L}{W_{tube} \left[\frac{1}{U_L [D_{tube} + (W_{tube} - D_{tube}) F]} + \frac{1}{\pi D_{tube} h_{i,col}} \right]} \quad (2.16)$$

where W_{tube} and D_{tube} are the distance between the tubes and the inner tube diameter, respectively. $h_{i,col}$ the heat transfer coefficient between the HTF and the tube wall and calculated by Dittus-Boelter equation (Incropera et al., 2006). F represents the standard fin efficiency for straight fins with rectangular profile as

$$F = \frac{\tanh[m(W_{tube} - D_{tube})/2]}{m(W_{tube} - D_{tube})/2} \quad (2.17)$$

where $m = \sqrt{U_L/k_p \delta_p}$. Also, k_p is the thermal conductivity of the absorber plate, and δ_p is the plate thickness. The absorber plate temperature, T_p , can be calculated depending on the collector inlet and outlet temperature of the HTF, $T_{col,in}$ and $T_{col,out}$, as

$$T_p = \frac{T_{col,in} + T_{col,out}}{2} + \frac{\dot{Q}_u/A_{col}}{F_R U_L} (1 - F_R) \quad (2.18)$$

The glass cover temperature, T_c , can be determined by using the plate temperature, T_p , as

$$T_c = T_p - \frac{U_t (T_p - T_{amb})}{h_{conv,p-c}} \quad (2.19)$$

The unknown parameters, plate and glass temperatures in the flat plate solar collector mathematical model, are interdependent in evaluating for the next time step. Therefore, Equations 2.1 to 2.19 are solved iteratively to obtain independent results from the initial guesses.

To check the validity of the flat plate solar collector model, the evaluated heat loss coefficient of the top surface of the collector, U_t , is compared with the ones that are provided in Duffie & Beckman (2013) for various ambient temperature, absorber plate temperature and wind convective heat transfer coefficient in Section 3.1.1.

2.2.2 Sensible Heat Packed Bed TES Tank Model

As a first step, a transient numerical model is developed for an integrated TES unit that is composed of a packed bed sensible heat TES tank and a flat plate solar collector to obtain the charging performance of the sensible heat TES system. The schematic of the integrated TES model is provided in Figure 2.5. The details of the mathematical model of the flat plate solar collector are given in Section 2.2.1. TES tank is filled with aligned spherical rigid storage materials. As a common approach in the literature (Zhao, Cheng, Liu & Dai, 2018; Cheng & Zhai, 2018), the flow distribution is assumed to be uniform along the radial direction of the tank. That is, instead of modeling the whole storage tank, a single column of storage material and the surrounding HTF domain along the flow direction is considered in the current mathematical model to evaluate temporal and spatial temperature variations within the storage tank. The representative flow column that is selected from the storage tank is also highlighted in Figure 2.5.

The inner diameter of the storage tank, D_T , is 0.8 m. The tank height, H_T , and the diameter of the spherical storage material, D_S , on the other hand, are varied from $H_T = 0.8$ m to 1.6 m and from $D_S = 20$ mm to 80 mm, respectively. An insulation material covers the outer surface of the storage tank to reduce heat losses through the ambient. The overall heat transfer coefficient of the storage tank is assumed to be $U_T = 0.678$ W/m²K (Hänchen, Brückner & Steinfeld, 2011). A total of 10 flat plate solar collectors with one glass cover (see Figure 2.2) are connected in series and integrated with the storage tank to transfer the solar thermal energy through the storage material throughout the daytime. Spherical quartzite rocks are used as storage material, and SYLTHERM 800 silicone fluid is the heat transfer fluid (HTF). The thermo-physical properties of the materials can be found in Table 2.2.

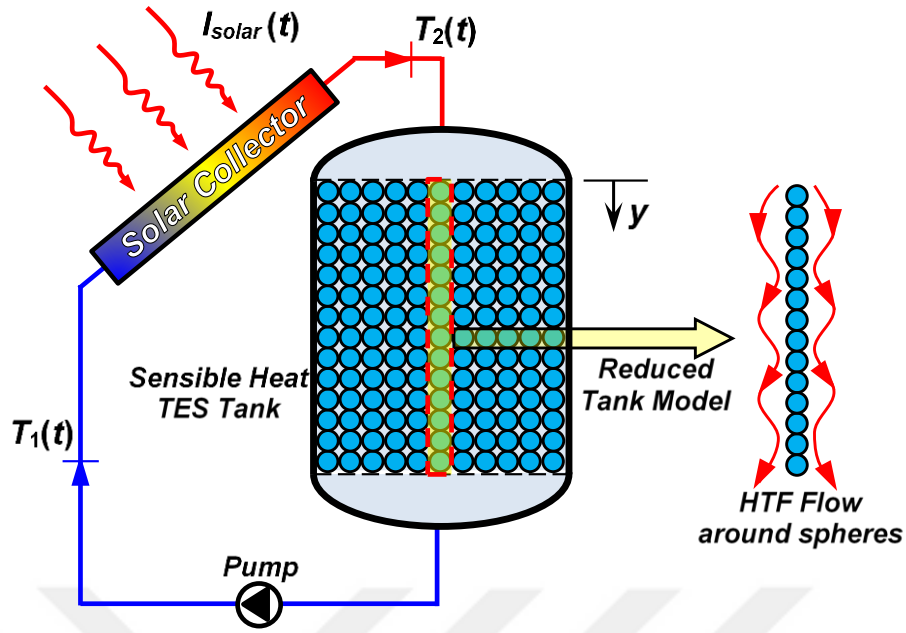


Figure 2.5 Schematic representation of the sensible heat packed bed TES tank (Alptekin & Ezan, 2020)

Table 2.2 The thermo-physical properties of the HTF and storage material

Property	SYLTHERM 800 (Peiró et al., 2017)	Quartzite-rock (Esence et al., 2017)
Density, ρ (kg/m ³)	917.07	2600
Specific heat, c (J/kgK)	1643	850
Thermal conductivity, k (W/mK)	0.1312	5.5
Thermal diffusivity, α (m ² /s)	8.7×10^{-8}	2.5×10^{-6}
Dynamic viscosity, μ (Pas)	7E-3	-

Governing equations of the storage tank are reduced by considering the following simplifying assumptions:

- Flow is fully developed within the storage tank, and empirical correlations are valid to characterize heat exchange between the spheres and HTF (Wang, Zhao, Xu, Wang & Dai, 2019),
- Temperature variations along the radial direction of the tank are neglected (Khor, Sze, Li & Romagnoli, 2020),

- The top and bottom surfaces of the storage tank are taken as adiabatic, and the tank loses heat from the lateral surface (Marti, Geissbühler, Becattini, Haselbacher & Steinfeld, 2018),
- Radiation effect considered as negligible, and no internal heat generation occurs (Elfeky, Ahmed & Wang, 2018),
- Porosities along the radial and axial direction are assumed to be uniform in the storage tank (Peng, Yang, Li & Ling, 2016),
- The thermal properties of storage material and HTF are constant (Niedermeier et al., 2018).

The generated computational grid system of the packed bed sensible heat TES tank with the spherical storage material is illustrated in Figure 2.6.

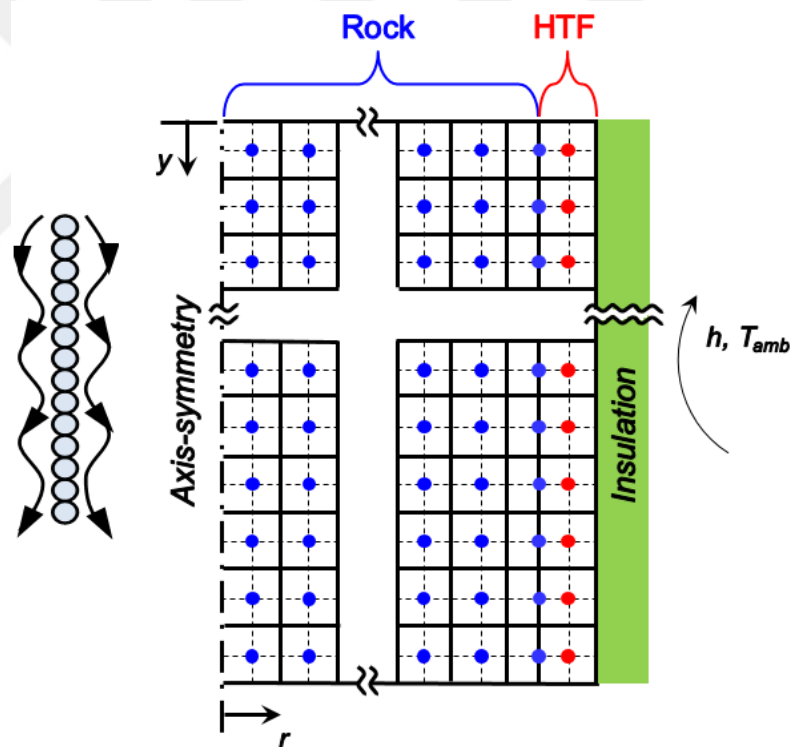


Figure 2.6 Schematic representation of the computational grid structure of the sensible heat TES tank

Consequently, the energy equations for the solid and fluid domains can be reduced as follows

- *for Spherical Solid Domain*

$$\frac{\partial}{\partial t}(\rho c T) = \frac{1}{r^2} \frac{\partial}{\partial r} \left(k r^2 \frac{\partial T}{\partial r} \right) \quad (0 \leq r \leq D_s/2) \quad (2.20)$$

- *for HTF Domain*

$$\begin{aligned} \frac{\partial}{\partial t}(\rho c T) \varepsilon A_T \Delta y + \frac{\partial}{\partial y}(\rho c u T) \varepsilon A_T \Delta y = \\ k \frac{\partial^2 T}{\partial y^2} A_T \varepsilon \Delta y + h (\pi D_s^2) N (T_s(r=r_o) - T) + U_T (\pi D_T \Delta y) (T - T_{amb}) \end{aligned} \quad (2.21)$$

where $T_s(r=r_o)$ is the surface temperature of the sphere. ε is the average porosity of the storage tank and can be calculated as (Beavers, Sparrow & Rodenz, 1973)

$$\varepsilon = 0.368 \left[1 + 2 \frac{D_s}{D_T} \left(\frac{0.476}{0.368} - 1 \right) \right] \quad (2.22)$$

The convective heat transfer coefficient, h , between the spherical storage material and HTF, can be determined from the following equation that is suggested by Coutier & Farber (1982)

$$h = \frac{700}{6(1-\varepsilon)} \left(\frac{\dot{m}}{A_T} \right)^{0.76} D_s^{0.24} \quad (2.23)$$

In the literature, assuming a constant convective heat transfer coefficient along the axial direction is a common approach in modeling packed bed TES systems (Sciacovelli et al., 2017; Zanganeh, Pedretti, Haselbacher & Steinfeld, 2015). The number of spherical storage materials that fill the cross-sectional area of the storage tank is evaluated from the following equation as

$$N_{sphere} = \frac{3D_T^2}{2D_s^3} \Delta y (1-\varepsilon) \quad (2.24)$$

u in Equation 2.21 is the superficial velocity of the HTF and is defined based on the actual cross-sectional area of the fluid as

$$u = \frac{\dot{m}}{\rho A_T \varepsilon} \quad (2.25)$$

It is assumed that the integrated sensible heat TES system is initially at thermal equilibrium with its surroundings. That is, a uniform temperature distribution is

defined for the TES unit, collector, and HTF. The corresponding initial and boundary conditions for each domain of the storage tank are given below:

- *for HTF:*

$$T(y, 0) = T_{init}, \quad T(0, t) = T_{T,in}, \quad -k \frac{\partial T}{\partial y} \Big|_{(H,t)} = 0 \quad (2.26)$$

- *for storage material:*

$$T(r, 0) = T_{init}, \quad \frac{\partial T}{\partial r} \Big|_{(0,t)} = 0, \quad -k \frac{\partial T}{\partial r} \Big|_{(r_o,t)} = h[T_s(r=r_o) - T_{HTF}] \quad (2.27)$$

Here, the TES tank's inlet temperature of the HTF depends on actual meteorological data, so it varies during the charging process and is evaluated from Equation 2.1.

2.2.3 Latent Heat Packed Bed TES Tank Model

As a second step, an in-house 1D transient code is developed to evaluate the latent heat TES tank's charging performance, which is integrated with a flat plate solar collector. A cylindrical TES tank that consists of aligned spherical capsules is filled with phase change material (PCM) is numerically modeled. A schematic description of the integrated latent heat TES system is illustrated in Figure 2.7. The details of the mathematical model of the flat plate solar collector are given in Section 2.2.1. Like the sensible heat TES system, flow distribution is assumed uniform along the TES tank's radial direction throughout the charging process, which is commonly accepted in the literature (Zhao et al., 2018; Cheng & Zhai, 2018). By doing so, a single column of spherical capsules and the surrounding HTF is numerically modeled instead of modeling the whole TES tank.

In the analyses of the determination of the charging performance of the latent heat TES tank, the TES tank diameter, D_T , and height, H_T , are kept constant as 0.8 m and 1.2 m, respectively. The tank is filled with PVC spherical capsules with a shell thickness of 1 mm (Ismail & Henriquez, 2000). The ethylene glycol (EG) mixture is circulated within the system as the HTF. The thermophysical properties of the shell material, HTF, and PCMs are represented in Table 2.3. Even if the

melting temperatures of the PCMs are varied, the thermophysical properties are defined unchanged for each selected PCMs. The TES tank's overall heat transfer coefficient is assumed as $0.678 \text{ W/m}^2\text{K}$ (Hänchen et al. 2011).

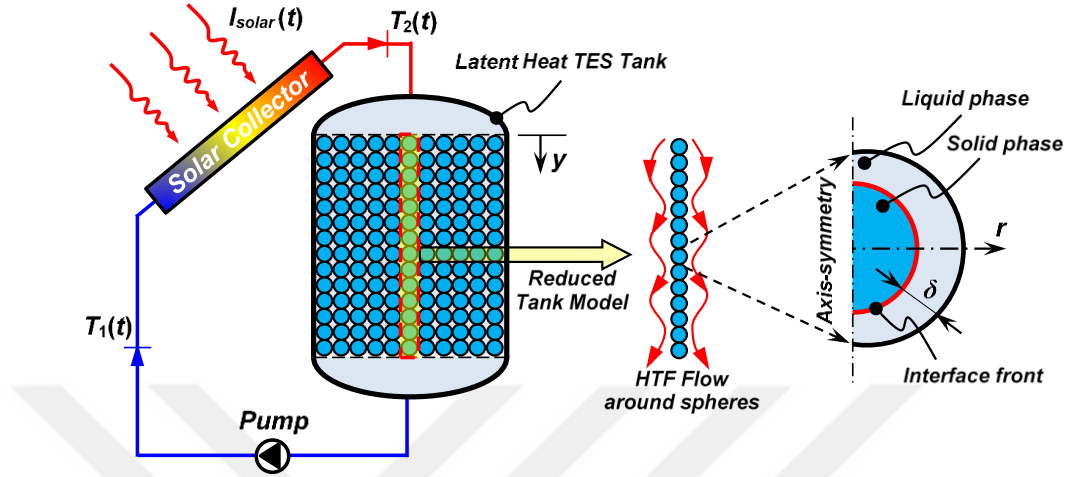


Figure 2.7 Schematic representation of the integrated latent heat TES system (modified from Alptekin & Ezan, 2021)

Table 2.3 The thermo-physical properties of the HTF, shell material, and PCM

Property	PCM (Rubitherm®, 2020)	Shell material (PVC) (Cengel & Ghajar, 2019)	HTF (%60 water - %40 EG by vol.) @30°C (ASHRAE, 2017)
Density, ρ (kg/m ³)	760 – liquid 880 – solid	1470	1055.39
Specific heat, c (J/kgK)	2000	840	3502
Thermal conductivity, k (W/mK)	0.2	0.1	0.412
Latent heat of fusion, h_{sf} (J/kg)	160000	–	–
Thermal expansion coefficient, β (1/K)	9.1E–4	–	–
Dynamic viscosity, μ (Pas)	3.8E–3	–	2.26E–3

A transient numerical code is developed in MATLAB to obtain the spatial and temporal temperature variations of the latent heat TES tank under real meteorological data. The latent heat TES tank model is reduced to a single column of spherical capsules (see Figure 2.7) due to the uniformity of the flow distribution in the radial direction. The computational domain consists of PCM, shell, and HTF

regions, and the reduced domain is represented in Figure 2.8. The enhanced thermal conductivity approach is implemented to include the natural convection effects in the liquid phase of the PCM during melting.

In order to evaluate spatial and temporal temperatures of HTF and storage medium in the storage tank, the governing equations are reduced by considering the following simplifying assumptions:

- The distribution of the HTF inside the storage tank is uniform throughout the charging process (Zhao et al., 2018; Cheng & Zhai, 2018),
- Only the HTF region exchanges heat with the surroundings (Mao & Zhang, 2020),
- The temperature of the HTF varies along the axial direction (*y-direction*) of the storage tank (1D) (Khor et al., 2020),
- The top and bottom surface of the storage tank is taken as adiabatic, and it is assumed that only HTF loses heat from the lateral surface (Marti et al., 2018),
- Radiation effect considered as negligible, and no internal heat generation occurs (Elfeky et al., 2018),
- Porosities along the radial and axial direction are assumed to be uniform in the storage tank (Peng et al., 2016),
- The thermal properties of PCM, shell material, and HTF are constant, except for the thermal conductivity of liquid PCM. The enhanced thermal conductivity approach is used for the liquid phase of PCM (Khor et al., 2020).

Consequently, the energy equations for the solid domains can be reduced as follows

- *for PCM and shell domains:*

$$\frac{\partial H}{\partial t} = \frac{1}{r^2} \frac{\partial}{\partial r} \left(kr^2 \frac{\partial T}{\partial r} \right) \quad (0 \leq r \leq r_o) \quad (2.28)$$

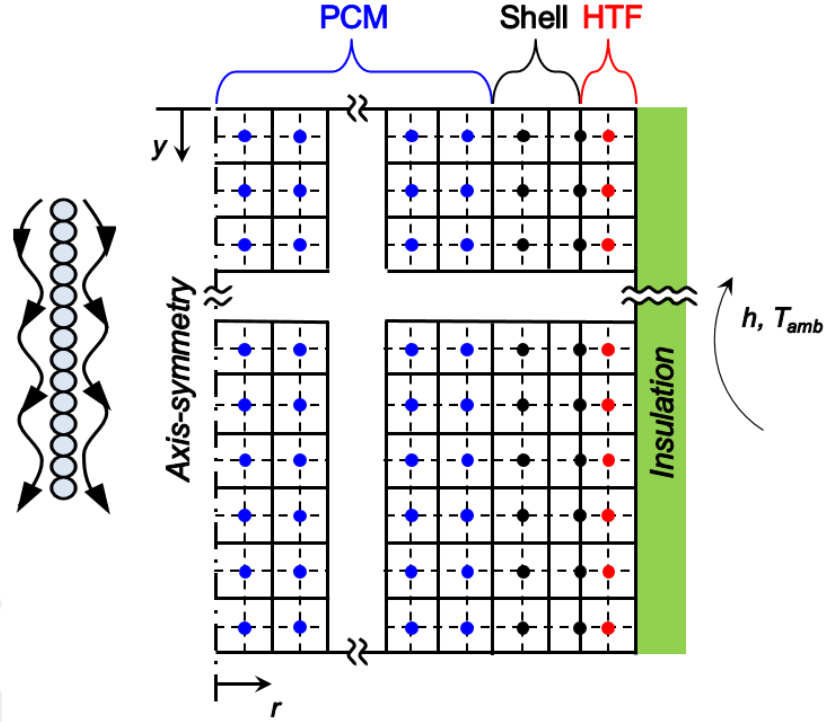


Figure 2.8 The reduced computational domain of the latent heat TES tank (Alptekin & Ezan, 2021)

Additionally, the governing equation for the HTF domain and the initial and boundary conditions of each domain of the storage tank is already defined in the previous section, and the same procedure is followed in the current model.

The temperature transformation method suggested by Cao & Faghri (1990) is applied to simulate the phase change process. According to this method, the relationship between temperature and enthalpy can be defined as,

$$H(T) = C(T)T + S(T) \quad (2.29)$$

where C and S represent the volumetric heat capacity ($= \rho c$) of the material and source term, respectively. Applying Equation 2.29 into Equation 2.28 gives the following energy balance equation for the PCM region (Ezan et al., 2014),

$$\frac{\partial}{\partial t}(CT) = \frac{1}{r^2} \frac{\partial}{\partial r} \left(kr^2 \frac{\partial T}{\partial r} \right) - \frac{\partial S}{\partial t} \quad (2.30)$$

where k represents the harmonic mean of the thermal conductivity at the solid/liquid interface. The heat transfer at the solid/liquid interface position, e , for the spherical coordinates is represented in Figure 2.9.

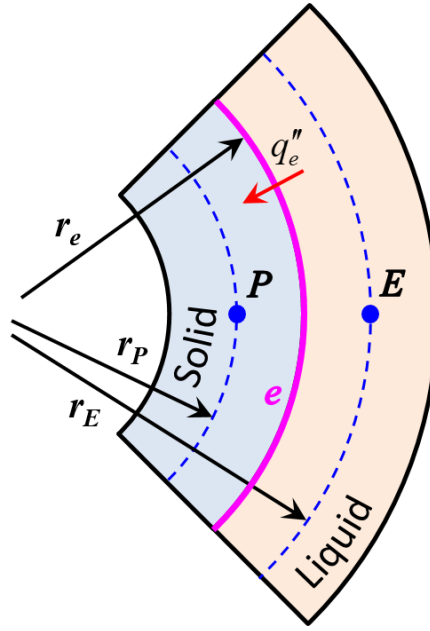


Figure 2.9 The representation of the interface position and heat flux

Heat flux at the interface can be written in the following forms

$$q_e'' = -k_e \frac{T_P - T_E}{r_e^2 \left(\frac{1}{r_E} - \frac{1}{r_P} \right)} \quad (2.31)$$

$$q_e'' = - \frac{T_P - T_E}{r_e^2 \left[\frac{1}{k_E} \left(\frac{1}{r_E} - \frac{1}{r_e} \right) + \frac{1}{k_P} \left(\frac{1}{r_e} - \frac{1}{r_P} \right) \right]} \quad (2.32)$$

Applying Equation 2.31 into Equation 2.32 gives the following form of the harmonic mean thermal conductivity value at the interface position e as

$$k_e = \frac{\frac{1}{r_E} - \frac{1}{r_P}}{\frac{1}{k_E} \left(\frac{1}{r_E} - \frac{1}{r_e} \right) + \frac{1}{k_P} \left(\frac{1}{r_e} - \frac{1}{r_P} \right)} = \frac{r_e (r_P - r_E) k_P k_E}{r_P (r_e - r_E) k_P + r_E (r_P - r_e) k_E} \quad (2.33)$$

The enhanced thermal conductivity approach is adopted to evaluate the thermal conductivity value of the nodes for the liquid phase of the PCM. The details of the enhanced thermal conductivity approach are given in Section 2.2.3.1.

The schematic representation of the temperature transformation method (TTM) is illustrated in Figure 2.10. The TTM assumes that phase change does not occur at a specific temperature but within a temperature range of $2\delta T_m$. There are three

regions as solid, liquid, and mushy. In the solid and liquid regions, only sensible heating occurs; however, in the mushy region, the internal energy of the PCM varies by considering both the sensible and latent heats. The definitions of the volumetric heat capacity, C , and source term, S , for these three regions are defined as (Ezan, Erekan & Dincer, 2013),

$$C(T) = \begin{cases} (\rho c)_s & T \leq T_m - \delta T_m \\ (\rho c)_m + \rho_l h_{sf} / (2\delta T_m) & T_m - \delta T_m < T < T_m + \delta T_m \\ (\rho c)_l & T \geq T_m + \delta T_m \end{cases} \quad (2.34)$$

$$S(T) = \begin{cases} (\rho c_p)_s (\delta T_m - T_m) & T \leq T_m - \delta T_m \\ (\rho c_p)_m (\delta T_m - T_m) + \frac{\rho_l h_{sf}}{2} - \frac{\rho_l h_{sf} T_m}{2\delta T_m} & T_m - \delta T_m < T < T_m + \delta T_m \\ (\rho c_p)_s \delta T_m - (\rho c_p)_l T_m + \rho_l h_{sf} & T \geq T_m + \delta T_m \end{cases} \quad (2.35)$$

where T_m represents the melting temperature, h_{sf} is the latent heat of fusion and $(\rho c)_m$ corresponds to the volumetric heat capacity in the mushy region and is evaluated as

$$(\rho c)_m = ((\rho c)_s + (\rho c)_l) / 2 \quad (2.36)$$

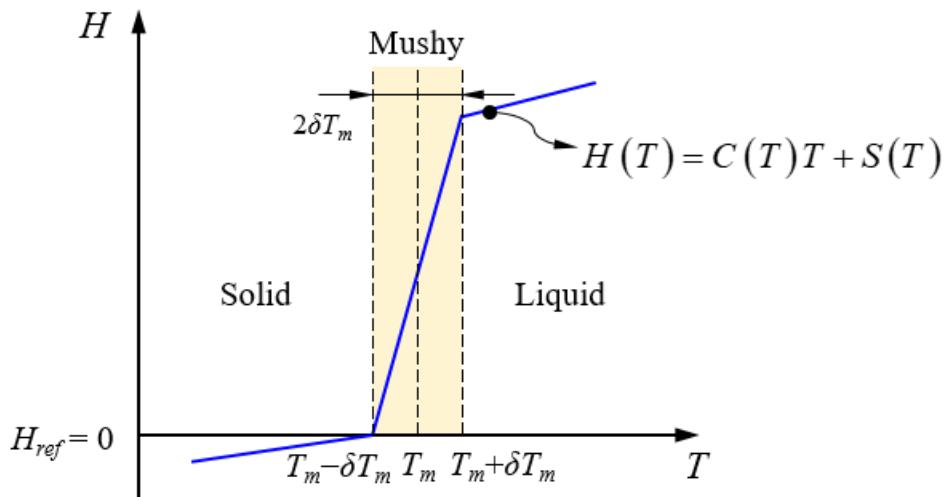


Figure 2.10 The representation of temperature transformation method

2.2.3.1 Enhanced Thermal Conductivity Approach

The enhanced thermal conductivity approach is implemented into the numerical model to take into account the effect of natural convection inside the liquid phase of the PCM during the inward melting process. The enhanced thermal conductivity of a fluid phase can be defined as (Bishop, Mack & Scanlan, 1966)

$$k_{enh} = k_l Nu \quad (2.37)$$

where k_l is the thermal conductivity of the liquid PCM, and the Nusselt number (Nu) is defined for the natural convection inside the spherical annulus. To obtain the enhanced thermal conductivity of the liquid PCM, the inward melting problem is simplified that occurrence of the eccentric annulus shape of solid PCM is idealized as a concentric annulus. Thus, the 2D inward melting model formed by natural convection inside the liquid PCM can be reduced in a 1D model. The schematic description of the actual and idealized process of the inward melting problem inside the spherical capsule is illustrated in Figure 2.11. It can be seen from the figure that the shape of the solid PCM is almost spherical at the early process of the phase change. However, with the advancing time, the strength of natural convection increases, and the eccentric shape is formed. The equivalent gap thickness of the liquid PCM, δ , can be defined as follows

$$\delta = r_i - r_{eq} \quad (2.38)$$

here, r_{eq} refers to the equivalent inner radius of the solid PCM of the idealized process. The Rayleigh number can be expressed as

$$Ra_\delta = \frac{g\beta\delta^3(T_i - T_m)}{\nu\alpha} \quad (2.39)$$

where T_i represents the inner surface temperature of the spherical capsule.

Various enhanced thermal conductivity correlations from the literature are compared with the experimental study, and the enhanced thermal conductivity approach is validated in Section 3.2.1.

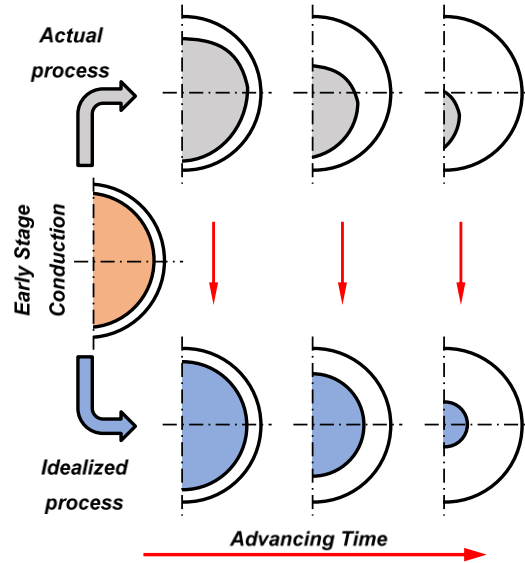


Figure 2.11 Actual and idealized the interface variations for inward melting (modified from Metin et al., 2019)

2.2.4 Cyclic Performance Evaluation and Heat Pump Model Integration

In order to achieve the cyclic performance of the TES tank, the TES tank is integrated with a heat pump to simulate the discharging process when domestic heating is needed. Since the time when solar energy is effective and the time when the heating need occurs does not generally overlap, after the maximum thermal energy is charged in the TES tank, the TES system turns into storage, *i.e.*, idle, mode.

The assumptions given in Section 2.2.3 are also valid for storage mode. In this mode, pumps remain off, and HTF is not circulating. Warm HTF, which leaves the solar collector, enters the TES tank from its top surface during the charging mode. That is, at the beginning of the storage mode, a thermal stratification occurs in the tank where the upper region is hotter than the lower region. So, only the conduction heat transfer mechanism takes place in the HTF domain, and heat loss/gain occurs to/from the environment.

Consequently, the energy equations for the solid and fluid domains can be reduced to evaluate the spatial and temporal temperature distributions for PCM, shell, and HTF domains in the storage period as follow

- for PCM and shell domains:

$$\frac{\partial H}{\partial t} = \frac{1}{r^2} \frac{\partial}{\partial r} \left(kr^2 \frac{\partial T}{\partial r} \right) \quad (0 \leq r \leq r_o) \quad (2.40)$$

- for HTF Domain:

$$\begin{aligned} & \frac{\partial}{\partial t} (\rho c_p T) \varepsilon A_T \Delta y + \\ & = k \frac{\partial^2 T}{\partial y^2} A_T \varepsilon \Delta y + k (\pi D_s^2) (T_s(r = r_o) - T) + U_T (\pi D_T \Delta y) (T - T_{amb}) \end{aligned} \quad (2.41)$$

The details of the terms of Equations 2.40 and 2.41 are already defined in Sections 2.2.2 and 2.2.3. The corresponding boundary conditions for each domain of the storage tank are given below for the storage process

- for HTF:

$$-k \frac{\partial T}{\partial y} \Big|_{(0,t)} = 0, \quad -k \frac{\partial T}{\partial y} \Big|_{(H,t)} = 0 \quad (2.42)$$

- for PCM and shell domains:

$$\frac{\partial T}{\partial r} \Big|_{(0,t)} = 0, \quad -k_{shell} \frac{\partial T}{\partial r} \Big|_{(r_o,t)} = k_{HTF} \frac{[T_s(r = r_o) - T_{HTF}]}{2r_o} \quad (2.43)$$

As soon as the domestic heating need occurs, the discharging process is activated. The energy stored in the TES tank is utilized as a heat source by the heat pump. The assumptions given in the Section 2.2.3 are also valid for the discharging processes. However, natural convection in the PCM region does not occur in the solidification process of the PCM, so the enhanced thermal conductivity approach is not adopted. Additionally, the evaporator component of the heat pump is well insulated, and no thermal losses take place. The energy equations for the PCM, shell, and HTF domains can be evaluated as Equations 2.21 and 2.28. The corresponding boundary conditions for each domain of the TES tank are given below for the storage process

- for the HTF domain:

$$T(0,t) = T_{T,in}(t), \quad -k \frac{\partial T}{\partial y} \Big|_{(0,t)} = 0 \quad (2.44)$$

- for PCM and shell domains:

$$\frac{\partial T}{\partial r} \Big|_{(0,t)} = 0, \quad -k \frac{\partial T}{\partial r} \Big|_{(r_o,t)} = h [T_s(r = r_o) - T_{HTF}] \quad (2.45)$$

The HTF leaving the top surface of the TES tank enters the heat pump's evaporator as a heat source. Then, HTF, which transfers its heat, leaves the evaporator and is directed to the tank from the bottom surface of the TES tank. The inlet temperature of the HTF to the TES tank can be evaluated as,

$$T_{T,in}(t) = T_{T,out}(t) - (\dot{Q}_{evap}(t)/\dot{m}c) \quad (2.46)$$

where $\dot{Q}_{evap}$ represents evaporator capacity and can be determined as follows

$$\dot{Q}_{evap}(t) = \dot{Q}_{load} - \dot{W}_{comp}(t) \quad (2.47)$$

here, $\dot{Q}_{load}$ and $\dot{Q}_{comp}$ represent the heating load (also condenser capacity) and the consumed power from the compressor, respectively. While the heating load is kept constant throughout analyses, the compressor power consumption can be calculated as

$$\dot{W}_{comp}(t) = \dot{Q}_{load} / \text{COP}_{HP}(t) \quad (2.48)$$

where COP_{HP} is the coefficient of performance of the heat pump. Kaygusuz (1995) experimentally evaluated the performances of air-source and water-source heat pumps under the climatic conditions of Turkey. The air-source heat pump was operated at ambient temperature, and the water-source heat pump was integrated with a solar-assisted TES tank. Kaygusuz (1995) proposed the following correlations to calculate the coefficient of performances (COP) of air-source and water-source heat pumps,

- *for air-source heat pump:*

$$\text{COP}_{HP,air} = -27.86 + 0.121T_{air} + 1.601 \times 10^{-4}T_{air}^2 - 7.035 \times 10^{-7}T_{air}^3 \quad (2.49)$$

- *for water-source heat pump:*

$$\text{COP}_{HP,water} = 5.46 + 5.33 \times 10^{-2}T_{HTF} - 5.53 \times 10^{-4}T_{HTF}^2 + 1.20 \times 10^{-6}T_{HTF}^3 \quad (2.50)$$

where T_{HTF} and T_{air} indicate source temperatures in kelvin. The current model assumes that T_{HTF} is equal to the TES tank outlet temperature, $T_{T,out}$.

2.3 Solution Procedure

Equations of flat plate solar collector and sensible or latent heat TES tank models are resolved iteratively and simultaneously, by defining variable meteorological

data in the flat plate solar collector equations, with an in-house code in MATLAB software to evaluate the charging performance of the sensible or latent heat TES system. Additionally, in assessing the cyclic performance of the latent heat TES system, equations of flat plate solar collector, latent heat TES tank, and heat pump models are resolved iteratively. The numerical models yield the spatial and temporal temperatures of HTF and storage material within the storage tank and the average temperatures of the absorber plate and cover glass of the solar collector.

The mathematical models for the solar collector and the storage tank are linked by evaluating the inlet of the storage tank from the outlet temperature of the solar collector, and similarly, the inlet temperature of the solar collector is evaluated from the outlet temperature of the storage tank in the charging process. However, the storage tank and heat pump are connected by evaluating the inlet of the storage tank from the outlet temperature of the evaporator, and similarly, the inlet temperature of the evaporator is evaluated from the outlet temperature of the storage tank in the discharging process. The heat losses through the piping networks are also considered to predict the inlet temperatures of the components more realistically.

A hybrid discretization technique is used to discretize the energy balance equations for the solution domain. The control volume approach is implemented for the spherical storage material domain, and the finite difference method is used for the HTF domain. Each sphere is divided into forty control volumes with equal radial increments for the sensible storage material. However, the PCM domain is divided into forty control volumes with equal radial increments, and the shell domain has two control volumes. Additionally, the grid size is taken as the diameter of the storage material in the axial direction. The time-step size is defined as 5 seconds to capture the sudden variations within the system due to the transient nature of the weather data in obtaining the charging performance of the sensible heat TES system. However, time-step size is determined as 0.5 seconds for the analyses of latent heat TES system to capture the phase change process as well as the dynamic meteorological effects. The results of the grid and time step size independency analyses for the latent heat TES system are represented in Section 3.3.2. The resultant set of algebraic equations is iteratively solved by using SIS (Strongly

Implicit Solver) (Lee, 1990) algorithm. The convergence criterion of the SIS solver is 10^{-15} .

The duration of the evaluation of the charging performance of the sensible and latent heat TES systems' analyses depends on the weather data and the charging characteristics of the storage tank. The analyses are initiated and terminated based on two specific criteria. The calculations start with the sunrise, *i.e.*, $I_{solar} \geq 1 \text{ W/m}^2$, and end when the charging process is completed inside the storage tank, *i.e.*, $T_{T,out} - T_{T,in} \geq 0$ or $\eta_{col} \leq 0$. However, the cyclic analyses of the latent heat TES systems consist of consecutive charging, storage, and discharging periods. The charging process is similarly terminated when the maximum charging occurs. Then, the storage process begins. The temperature distribution inside the TES tank becomes homogeneous, and heat loss occurs from the TES tank to the environment in the storage process. The domestic heating demand is determined as 8 kW for all heating season months. It is assumed that domestic heating demand is needed from 19:00 to 00:00. That is, the process starts at 19:00, and if the TES tank's inlet temperature is above the cut-off temperature of 0°C, the process proceeds until 00:00. If the TES tank inlet temperature reduces below 0°C before 00:00, an auxiliary electric heater meets the remaining heating need. After midnight, the storage period starts again, and it lasts until the charging period begins. The charging process starts when one of the following conditions is satisfied $T_{T,out} - T_{T,in} \leq 0$ or $\eta_{col} \geq 0$. Cyclic analyses are proceeds as described above and are terminated at the end of day-4. The preliminary studies on how many consecutive days the analyses time should be to obtain independent results from the initial boundary conditions are discussed in detail in Section 3.4.1.

2.4 Data Reduction

One of the most critical parameters to evaluate the thermal behavior of the sensible heat TES tank is the thermal stratification (Rosen, 2001; Li, 2016). In the literature, thermal stratification is expressed as the ratio of the mean of the transient temperature gradients to the maximum mean temperature gradient for the dynamic

working mode of a TES tank. Thermal stratification number, Str , is obtained as follows

$$Str(t) = \frac{\overline{(\partial T / \partial y)}}{(\partial T / \partial y)_{\max}} \quad (2.51)$$

Additionally, the mean temperature of the TES tank is defined as a decisive parameter of the charging performance of the sensible heat TES tank and referred to the mass-weighted-average temperature of the storage tank and defined as follows

$$T_{mean} = \frac{[mcT]_{HTF} + [mcT]_S}{[mc]_{HTF} + [mc]_S} \quad (2.52)$$

Besides, the heat transfer rate of the charging process is defined as the ratio of the total stored energy in the TES tank to the charging time as

$$\dot{Q} = \frac{[mc(\bar{T} - T_{ini})]_{HTF} + [mc(\bar{T} - T_{ini})]_S}{t_{\text{charging}}} \quad (2.53)$$

here, it should be noted that while the TES tank consists of HTF and spherical storage material domains for the sensible heat TES, the TES tank consists of HTF, shell material, and PCM domains for the latent heat TES.

The thermodynamic analyses are also conducted to reduce the spatial and temporal variations in system performance indicators such as first and second law efficiencies. The energy-based efficiencies of the storage tank and the solar collector are defined as

$$\eta_T(t) = \frac{\dot{E}_T}{\dot{E}_{HTF}} \times 100 = \frac{\{[mc(T(t) - T_{init})]_{HTF} + [mc(T(t) - T_{init})]_S\}}{\sum_{t=0}^t [\dot{E}_{in} - \dot{E}_{out}]_{HTF} dt} \quad (2.54)$$

$$\eta_{col}(t) = \frac{\dot{Q}_u(t)}{I_{solar}(t) A_{col}} \times 100 \quad (2.55)$$

here, in the TES tank energy efficiency calculation, as in the heat transfer rate evaluation, there are different regions within the TES tank in sensible and latent heat TES systems.

Second-law based thermodynamic assessment is considered by defining the exergy balance equation for the storage tank as

$$\dot{E}x_{HTF_{net,in}} - \dot{E}x_{Q_{loss}} - \dot{E}x_{dest} = \dot{E}x_T \quad (2.56)$$

The entropy balance equation is also resolved to verify the rate of exergy destruction that is evaluated from the exergy balance equation (Equation 2.56). The entropy balance equation is written as follows

$$\dot{S}_{HTF_{net,in}} - \dot{S}_{Q_{loss}} + \dot{S}_{gen} = \dot{S}_T \quad (2.57)$$

The expressions of each term in Equations 2.56 and 2.57, the rate of exergy and entropy transfer by flow and heat loss, and exergy and entropy variation of the system, are separately represented for sensible and latent heat TES systems as

- *exergy balance equation terms for sensible heat TES*

$$\dot{E}x_{HTF_{net,in}} = \dot{m}c \left[(T_{T,in} - T_{T,out}) - T_{amb} \ln(T_{T,in}/T_{T,out}) \right] \quad (2.58a)$$

$$\dot{E}x_{Q_{loss}} = (1 - T_{amb}/T_{HTF}) \dot{Q}_{loss} \quad (2.58b)$$

$$\begin{aligned} \dot{E}x_T = & \left\{ mc \left[(T - T^0) - T_{amb} \ln(T/T^0) \right] / \Delta t \right\}_{HTF} \\ & + \left\{ mc \left[(T - T^0) - T_{amb} \ln(T/T^0) \right] / \Delta t \right\}_S \end{aligned} \quad (2.58c)$$

- *entropy balance equation terms for sensible heat TES*

$$\dot{S}_{HTF_{net,in}} = \dot{m}c \ln \left(\frac{T_{T,in}}{T_{T,out}} \right) \quad (2.59a)$$

$$\dot{S}_{Q_{loss}} = \dot{Q}_{loss} / T_{HTF} \quad (2.59b)$$

$$\dot{S}_T = \frac{\left[mc \ln \left(\frac{T}{T^0} \right) \right]_{HTF} + \left[mc \ln \left(\frac{T}{T^0} \right) \right]_S}{\Delta t} \quad (2.59c)$$

- *exergy balance equation terms for latent heat TES*

$$\dot{E}x_{HTF_{net,in}} = \dot{m}c \left[(T_{T,in} - T_{T,out}) - T_{amb} \ln(T_{T,in}/T_{T,out}) \right] \quad (2.60a)$$

$$\dot{E}x_{Q_{loss}} = (1 - T_{amb}/T_{HTF}) \dot{Q}_{loss} \quad (2.60b)$$

$$\begin{aligned} \dot{E}x_T = & \left\{ V \left[(H - H^0)(1 - T_{amb}/T) \right] / \Delta t \right\}_{PCM} \\ & + \left\{ mc \left[(T - T^0) - T_{amb} \ln(T/T^0) \right] / \Delta t \right\}_{Shell} \\ & + \left\{ mc \left[(T - T^0) - T_{amb} \ln(T/T^0) \right] / \Delta t \right\}_{HTF} \end{aligned} \quad (2.60c)$$

- *entropy balance equation terms for latent heat TES*

$$\dot{S}_{HTF_{net,in}} = \dot{m}c \ln(T_{T,in}/T_{T,out}) \quad (2.61a)$$

$$\dot{S}_{Q_{loss}} = \dot{Q}_{loss}/T_{HTF} \quad (2.61b)$$

$$\dot{S}_T = \frac{\left[V(H - H^0)/T \right]_{PCM} + \left[mc \ln(T/T^0) \right]_{Shell} + \left[mc \ln(T/T^0) \right]_{HTF}}{\Delta t} \quad (2.61c)$$

here T^0 represents the temperature at the previous time step. The rate of exergy destruction from the entropy generation is evaluated as

$$\dot{E}x_{dest} = T_{amb} \dot{S}_{gen} \quad (2.62)$$

The energy-based efficiency of the latent heat TES system in the charging process can be defined as the ratio of the amount of total stored energy in the TES tank to the amount of total incoming solar energy as,

$$\eta_{system}(t) = \frac{\sum_{i=1}^N (H(t) - H^0) V}{\int_{t=0}^t I_{solar}(t) A_{col} dt} \times 100 \quad (2.63)$$

where N represents different regions in the computational domain as PCM, shell, and HTF. Besides, H^0 and V represent the enthalpy at the previous time step and the volume of these regions, respectively.

The exergy efficiency of the storage tank in the charging process is defined as (Dincer & Rosen 2010),

$$\psi_T(t) = \frac{\sum_{t=0}^t \dot{E}x_T dt}{\sum_{t=0}^t \dot{E}x_{HTF} dt} \times 100 \quad (2.64)$$

Additionally, the exergy-based charging efficiency of the integrated TES system can be evaluated as the fraction of the amount of total stored exergy to the amount of total exergy of incoming solar energy (Dincer & Rosen 2010),

$$\psi_{system}(t) = \frac{\sum_{t=0}^t \dot{E}x_T dt}{\int_0^t I_{solar}(t) A_{col} \left(1 - \frac{T_{amb}(t)}{T^*} \right) dt} \times 100 \quad (2.65)$$

where T^* is the apparent sun temperature, and according to Petela (1964), it approximately equals to $(3/4)T_{sun}$, where T_{sun} is the apparent black body temperature of the sun (≈ 6000 K).

The cyclic energy efficiency of the integrated latent heat TES system is defined as the ratio of the total energy supplied to the heat pump to the total incoming solar energy as follows

$$\eta_{cyclic}(t) = \frac{\sum_{t=0}^{t_{dis}} \dot{Q}_{evap}(t) dt}{\sum_{t=0}^{t_{char}} I_{solar}(t) A_{col} dt} \quad (2.66)$$

The cyclic exergy-based efficiency of the integrated latent heat TES system is defined as the ratio of the total exergy supplied to the heat pump to the total exergy of the incoming solar energy as follows

$$\psi_{cyclic}(t) = \frac{\sum_{t=0}^{t_{dis}} \dot{E}x_{HTF}(t) dt}{\sum_{t=0}^{t_{char}} I_{solar}(t) A_{col} \left(1 - \frac{T_{amb}(t)}{T^*}\right) dt} \quad (2.67)$$

One should notice that the pumping power due to the pressure drop is neglected in the definition of the exergy efficiency as it does not significantly affect the results (Geissbühler, Mathur, Mularczyk & Haselbacher, 2019). Following the recent work of Bindra, Bueno & Morris, 2014, for the current range of parameters, *i.e.*, $0.8 \text{ m} \leq H_T \leq 1.6 \text{ m}$ and $20 \text{ mm} \leq D_s \leq 80 \text{ mm}$, exergy destruction due to pressure drop is negligible.

As is mentioned before, when the stored energy in the proposed latent heat TES system cannot be sufficiently sourced for the heat pump throughout discharging period, the auxiliary electric heater will be activated to compensate for the remaining heating need. Electricity fraction is defined as the ratio of the total electricity consumption of the latent heat TES tank source heat pump to the total electricity consumption of the air-source heat pump as follows

$$Electricity \text{ Fraction} = \frac{W_{comp,HP,HTF} + W_{aux}}{W_{comp,HP,air}} \quad (2.68)$$

CHAPTER THREE

RESULTS & DISCUSSION

This chapter of the dissertation represents the results of each mathematical model in a systematic manner. At first, the top heat loss coefficient of the flat plate solar collector, which directly affects the performance of the TES system, is compared against the data from the literature. In addition, the time-wise variations of the solar collector parameters are given that occurs thanks to the integration of the solar collector providing dynamic effects of meteorological data to the sensible heat and latent heat TES systems. Additionally, the numerical models of enhanced thermal conductivity approach, charging performance of the sensible heat, and latent heat TES tanks are validated from the literature. Moreover, the temporal, spatial, and overall results of the charging performance of the sensible and latent heat TES tanks and cyclic performance of the latent heat TES system have been interpreted in detail. Consequently, the optimum design parameters and working conditions for the TES systems are proposed for heating season months.

3.1 Results of the Numerical Model of Flat Plate Solar Collector

In the process of developing a mathematical model of the flat plate solar collector, the top surface heat loss coefficient, which has a direct effect on useful heat, Q_u , and hence the collector efficiency, is compared with the literature for several constant ambient parameters. Additionally, the time-wise variations of the effective parameters of the solar collector in determining the overall heat transfer coefficient are represented as a matter of example during the charging process of the sensible and latent heat TES.

3.1.1 Comparison of Top Heat Loss Coefficient of Solar Collector

The validation of the numerical model of the flat plate solar collector is provided by comparing the obtained top surface heat loss coefficients, U_t , with the results represented by Duffie & Beckman (2013). The variation of the top heat loss coefficient depends on absorber plate temperature is illustrated in Figure 3.1.

Additionally, as part of the comparison, the ambient temperature and wind convection coefficients are varied, and the effects of these parameters on the top heat loss coefficient are evaluated. Figure 3.1 shows that the top heat loss coefficient increases as the absorber plate, ambient temperature, and wind convection coefficient increase. Also, it can be deduced that the effect of wind-based heat transfer coefficient, *i.e.*, h_w , is more influential than ambient temperature. The results show that obtained top heat loss coefficient values from the developed model have a good harmony with the literature.

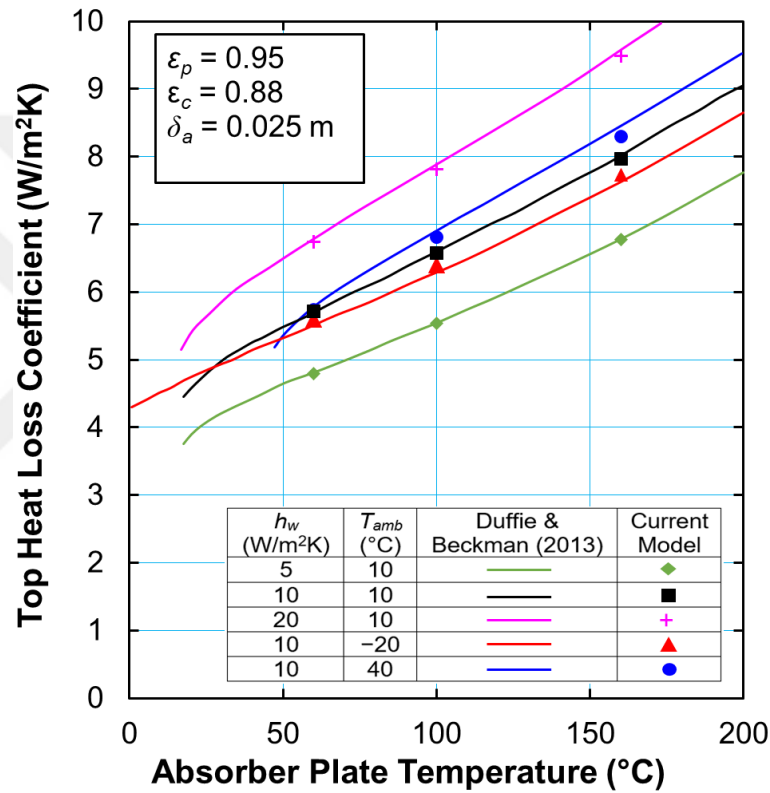


Figure 3.1 The comparison of the top heat loss coefficient

3.1.2 Flat Plate Solar Collector Dynamic Performance Evaluation

The flat plate solar collector is modeled to include the dynamic effects of solar energy. The details of the numerical model can be found in Chapter 2.2.1. In this section, the time-wise variations of the parameters that affect the efficiency of the solar collector are represented throughout the charging process of typical sensible and latent heat TES systems in Figures 3.2 and 3.3, respectively. In the graphs,

time-wise variations of the radiation and convective heat transfer coefficients between absorber plate and glass cover and between the glass cover and ambient and top surface and overall heat loss coefficients are given in Figures 3.2a and 3.3a. Besides, the time-wise variations of the solar radiation and the ambient air temperature, absorber plate and glass cover, and the inlet and outlet temperature of the HTF are illustrated in Figures 3.2b and 3.3b. The operating month, the mass flow rate of HTF, the diameter of the storage material, the height of the storage tank are taken identical to represent the parameters of solar collector integrated with sensible and latent heat TES tanks. The melting temperature of the PCM is taken as 30°C for the latent heat TES tank. The convective heat transfer coefficient between glass cover and ambient is only dependent on wind speed.

Convective heat transfer coefficient between absorber plate and glass cover and radiative heat transfer coefficients slightly increase at the early stages of the charging process of the sensible heat TES system. However, as the charging continues, the increment trend of the absorber plate and glass cover temperatures diminish, so heat transfer coefficients remain constant until the end of the charging process. On the other hand, for the latent heat TES system, as expected, collector inlet temperature increases until it reaches the melting temperature of the PCM with increasing solar radiation for the latent heat TES system. In addition to this, unlike the sensible heat TES system, due to the high volumetric heat capacity of the PCM inside the TES tank, a certain horizontalization is observed at the collector inlet and outlet and absorber plate temperatures during the phase change process. Therefore, it is seen that the increment trends of the convective and radiative heat transfer coefficients have ended and remain constant while the phase change process occurs. After, heat transfer coefficients again narrowly increase.

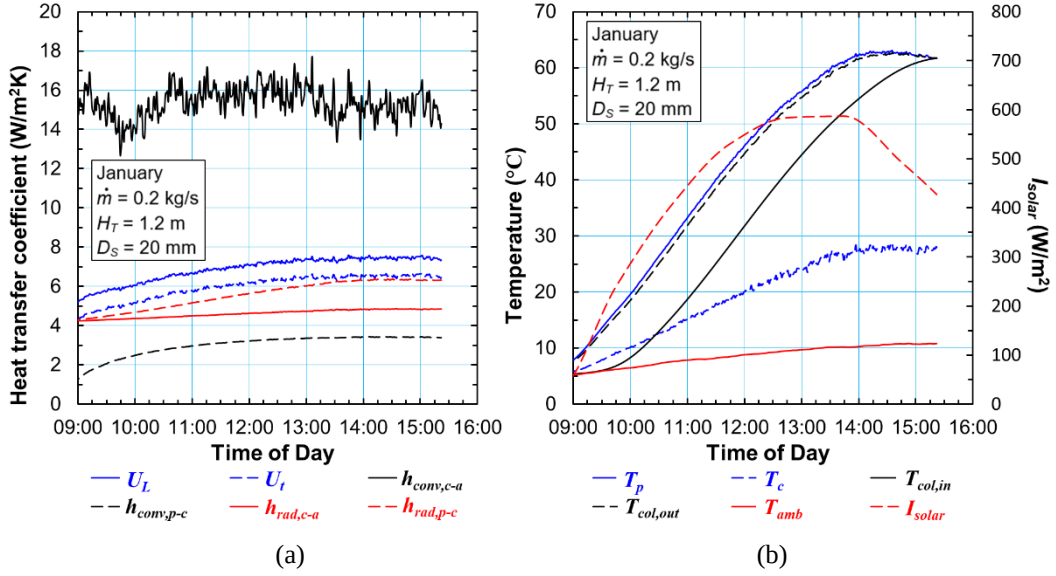


Figure 3.2 A typical time-wise variation of the solar collector parameters integrated with sensible heat TES tank

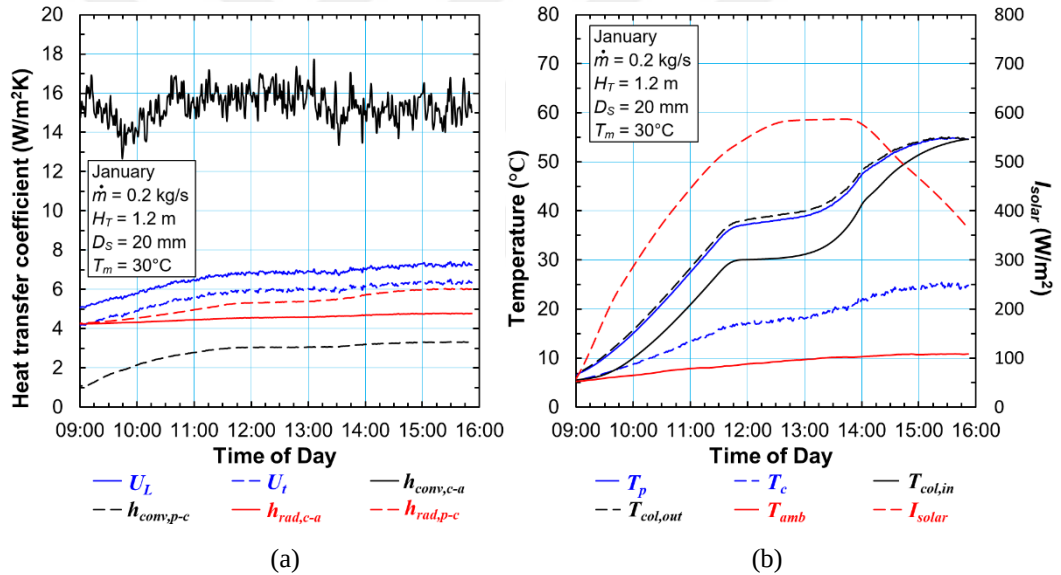


Figure 3.3 A typical time-wise variation of the solar collector parameters integrated with latent heat TES tank

3.2 Charging Performance of Sensible Heat Packed Bed TES System

3.2.1 Preliminary Analyses of Sensible Heat TES Tank

In the literature, the packed-bed TES systems could be analyzed under certain conditions by implementing a simplified lumped transient model without considering the spatial variations within the storage materials. That is, before going

through the discussions on the detailed spatial and temporal variations within the tank, a preliminary survey is conducted to understand whether it is appropriate to use a lumped model in the current system. To do so, the limit of the validity of the lumped capacitance method is assessed. Biot number, Bi , is the ratio of the conduction resistance in the material to convective resistance at the surface of the material and can be defined as

$$Bi = \frac{hL_c}{k} \quad (3.1)$$

where L_c represents the characteristic length, and it is defined as the ratio of the volume to the surface area of the solid. Hence, characteristic length becomes as $r_o/3$ for a spherical material, and Biot number can be expressed as

$$Bi = \frac{hr}{3k} \quad (3.2)$$

The mass flow rate of the HTF and the diameter of the spherical storage material are varied over an extensive range from 0.2 to 1.6 kg/s and 20 to 60 mm, respectively. The variation range of the Biot number is illustrated in Table 3.1 for some studied extreme design parameters and working conditions. The calculations show that the Biot number is varied between 0.02 and 0.42. Since the criterion of $Bi < 0.1$ does not meet for all scenarios, lumped capacitance method does not apply to the mathematical model of the packed bed sensible heat TES tank. Therefore, spatial temperature variations inside the storage material have been obtained in detail in addition to temporal temperature variations.

Table 3.1 The values of the Biot number for some studied scenarios

Diameter of the storage material (mm)	The mass flow rate of the HTF (kg/s)	Convective heat transfer coefficient (W/m ² K) (Equation 2.23)	Biot Number
20	0.2	36.14	0.02
20	1.6	175.54	0.11
60	0.2	47.87	0.09
60	1.6	232.50	0.42

3.2.2 Validation of the Mathematical Model of Sensible Heat TES System

In the literature, to validate numerical models of packed bed sensible heat TES units, the variation of the temperature of the HTF along the axial direction is mostly compared with the experimental data. In this study, the experimental work of Meier, Winkler & Wuillemin (1991) is reproduced to validate the solution method described above. Meier et al. (1991) investigated laboratory-scaled sensible heat packed-bed thermal energy storage system. The design and working parameters of the reference work are summarized in Table 3.2. Rock is used as the storage material, and air flows around the rocks as the HTF. The height and the inner diameter of the storage tank are 1.2 m and 0.148 m, respectively. A total of 6 thermocouples were placed at specific locations along the flow direction of the storage tank. The inlet temperature of the HTF is 550°C and is maintained constant throughout the experiments.

Table 3.2 The parameters used in the validation case (Meier et al., 1991)

Parameter	Value	Parameter	Value
T_{HTF} (°C)	550	ε (-)	0.4
T_{amb} (°C)	20	c_s (J/kgK)	1068
$\dot{m}_{HTF}$ (kg/s)	0.004	ρ_s (kg/m ³)	2680
H_T (m)	1.2	k_s (W/mK)	2.5
D_T (m)	0.148	U_T (W/m ² K) (Hänchen et al., 2011)	0.678
D_s (m)	0.02		

In Figure 3.4, variations in HTF temperature along the height of the storage tank are represented at six different flow times. The solid lines correspond to the predicted temperature variations in the current work. The symbols and the dashed lines, on the other hand, represent the experimental measurements of Meier et al. (1991) and the numerical results of Mertens, Alobaid, Frigge & Eppele (2014), respectively. Comparative results reveal that the predicted temperature variations are in good harmony with experimental and numerical studies in the literature. The mean deviations between the numerical and experimental works are 0.02% and 27.5%, respectively.

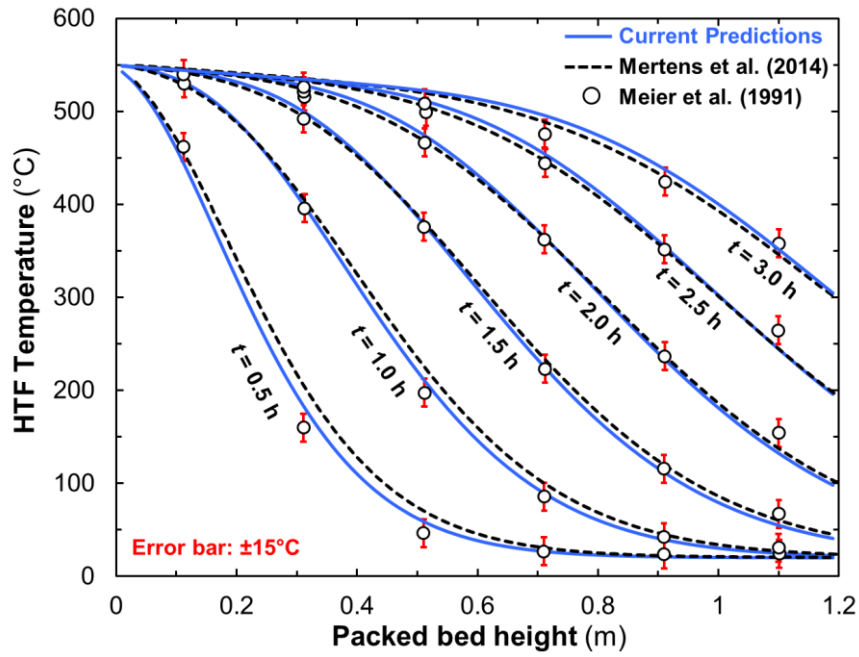


Figure 3.4 The variation of the temperature of the HTF along with the storage tank height at some particular instances

3.2.3 Parametric Results of Sensible Heat TES System

In the first part of the parametric study, the influences of (i) the mass flow rate of the HTF, (ii) the diameter of storage material, and (iii) the height of the storage tank are investigated to obtain the dynamic storage performance of the sensible heat TES tank in January. Time-wise variations of the exergy and energy efficiencies of the tank, collector efficiency, stratification number, and tank outlet temperature are examined to analyze the effects of design and working parameters. The influence of each parameter is discussed in detail in the following sub-sections. Analyses are then extended for additional months, *i.e.*, November, December, and February, to assess the performance of the charging period of the integrated sensible heat TES system for the cold season in which there is a heating demand.

3.2.3.1 Influence of Mass Flow Rate on System Charging Performance

Figures 3.5 and 3.6 show the daily variations of the performance indicators at four mass flow rates of 0.05 kg/s, 0.10 kg/s, 0.20 kg/s and 0.40 kg/s. In Figure 3.5, the results are given for a tank design with $H_T = 0.8$ m and $D_S = 20$ mm. One should

notice that the analyses are terminated when the mean temperature of the storage tank reaches its maximum to achieve the highest charging performance, *i.e.*, useful heat, from the sensible heat TES unit. Increasing the mass flow rate provides to reach the maximum storage temperature earlier. Regarding the required time for complete charging, the difference is approximately 56 min between the highest and lowest mass flow rates. HTF tank outlet temperature substantially depends on mass flow rate. The outlet temperature of the tank increases as the mass flow rate increases. At the end of the charging process, regarding the outlet temperature of the tank, the difference between the highest and the lowest mass flow rates is 9.6°C.

Mass flow rate also significantly improves the exergy efficiency of the storage tank. Maximum exergy efficiencies are evaluated as 96.1% and 90.7% for the highest and lowest mass flow rates, respectively. It is also interesting to note that exergy efficiency dramatically increases before midday since the high-temperature gradients within the tank at the early periods of the charging process disappear. Then it remains almost constant for a while, and the exergy efficiency slightly reduces at the end of the charging period.

Even though the mass flow rate of the HTF does not significantly influence the tank energy efficiency, it should be noted that the highest energy efficiency is obtained at the lowest mass flow rate. Moreover, the tank energy efficiency gradually decreases throughout the charging process because of increasing heat loss through the ambient. For the early periods of the process, the collector efficiency increases by increasing the working fluid's mass flow rate. The maximum collector efficiencies are obtained as 40%, 54%, 65.5% and 73.5%, at mass flow rates of 0.05 kg/s, 0.10 kg/s, 0.20 kg/s, and 0.40 kg/s, respectively. However, at a certain time, for the current simulations beyond 12:35, the curves are inverted, and the highest collector efficiency is achieved at the lowest flow rate. This is due to the dynamic effects during the charging period. As mentioned earlier, the tank outlet temperature increases rapidly at high mass flow rates, but the slopes of the tank outlet temperatures suddenly change beyond 12:35, as the solar irradiance tends to reduce beyond this moment. Consequently, one should notice that both the mass flow rate

and the outlet temperature of the storage tank have a significant influence on collector efficiency.

Stratification number, which is a measure of temperature non-uniformity along the flow direction, dramatically increases at the beginning of the charging process in all cases. Beyond the mid-day, in parallel to the reduction in solar irradiance and the collector efficiency, the tank inlet temperature approaches approximately to a constant inlet temperature. Consequently, the stratification number decreases. At high mass flow rates, stratification number affects more quickly; it means that the thermal uniformity of the storage tank decays and restores faster. Although the time-wise variation of the stratification number shows a similar tendency with the solar irradiance, it can be noticed that at high mass flow rates, the instantaneous variations in the solar irradiance become more traceable on the stratification number. At low mass flow rates, the instant variations in stratification numbers have weak fluctuations. The stratification number reaches its maxima, *i.e.*, unity, earlier at high mass rates, the difference is approximately 42 min between the highest and lowest mass flow rates. Moreover, at the end of the charging process, the stratification numbers are 0.26, 0.15, 0.08 and 0.04 for the mass flow rates of 0.05 kg/s, 0.10 kg/s, 0.20 kg/s, and 0.40 kg/s, respectively.

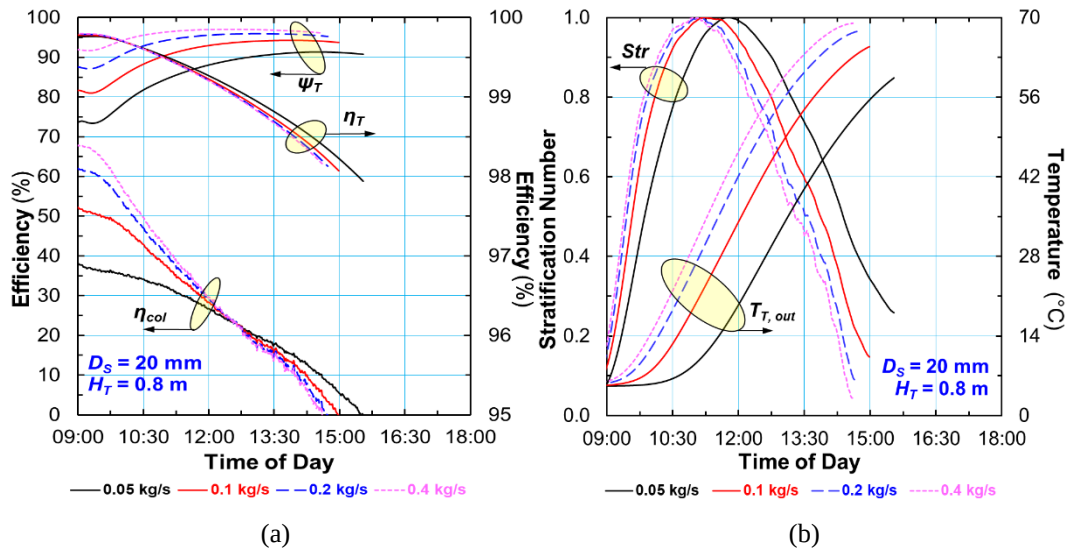


Figure 3.5 The effect of mass flow rate on the (a) energy and exergy efficiency of the tank and collector efficiency and (b) stratification number and the tank outlet temperature – $H_T = 0.8$ m & January

In Figure 3.6, on the other hand, the influence of the mass flow rate of HTF is given for $H_T = 1.6$ m. Increasing the tank height from 0.8 m to 1.6 m mainly shifts outlet temperature and collector efficiency curves. Regarding the outlet temperature of the tank, the curves are translated horizontally through the right, and the temperature rise is significantly delayed. Considering the lowest mass flow rate of the HTF, one can infer that no change is observed regarding the tank outlet temperature for the first 3 hours of the charging period. At the end of the charging period, regarding the outlet temperature of the HTF, the difference between the highest and the lowest mass flow rates is nearly 17°C . Similar to the previous tank design, *i.e.*, $H_T = 1.6$ m, in this case, inversion points in the collector efficiency variations are observed at approximately 13:40. Before this point, the collector efficiency increases with increasing the mass flow rate; however, the relationship is inverse beyond this point. Variations in the exergy and energy efficiencies of the storage tank show similar tendencies as in the previous design. At the end of the charging period, for the highest and lowest mass flow rates of the HTF, the exergy efficiencies are obtained as 95.84% and 88.81%, respectively. Besides, no considerable change is observed in energy efficiency by varying the mass flow rate of HTF. The energy efficiency is approximately 97.9%. The time for complete charging significantly differs depending on the mass flow rate of the HTF, and the difference between the highest and lowest mass flow rates is almost 1 hour.

The stratification number, the non-dimensional thermal inhomogeneity within the tank, is higher in the high mass flow rates. The stratification number reaches unity later than the others for the lowest mass flow rate, *i.e.*, 0.05 kg/s. At the end of the charging process, the numbers of 0.51, 0.22, 0.10 and 0.06 are obtained for 0.05 kg/s, 0.10 kg/s, 0.20 kg/s and 0.40 kg/s, respectively.

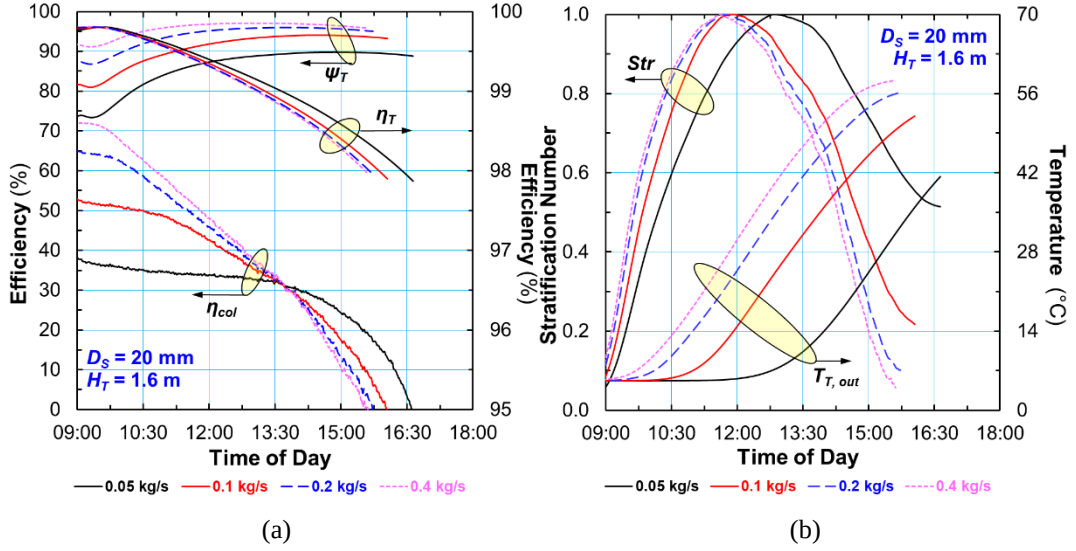


Figure 3.6 The effect of mass flow rate on the (a) energy and exergy efficiency of the tank and collector efficiency and (b) stratification number and the tank outlet temperature – $H_T = 1.6$ m & January

3.2.3.2 Influence of Storage Material Diameter on System Charging Performance

The transient response of the flat plate solar collector integrated sensible heat TES system is evaluated under three different storage material diameters of 20 mm, 40 mm, and 80 mm. In Figures 3.7 and 3.8, the time-wise variations of the dimensional and dimensionless performance indicators, *i.e.*, exergy and energy efficiencies of the tank, collector energy efficiency, tank outlet temperature of the HTF, and stratification number, are represented for $H_T = 0.8$ m and 1.6 m, respectively. Here the mass flow rate is kept constant as 0.2 kg/s in each analysis. Figures 3.7 and 3.8 depict that the effect of the storage material diameter on charging time is not considerable; the maximum difference is approximately 7 min between the smallest and biggest storage sphere diameter.

The exergy efficiency of the energy storage tank increases regardless of the design or working condition at the early stages of the charging process, as the temperature gradients gradually reduce. Beyond the middle of the process, no significant change is observed in the variation of the exergy efficiency since the rate of stored energy reduces in time. It is interesting to note that the exergy efficiency

increases with decreasing the diameter of the storage material. Exergy efficiencies are obtained at the end of the charging process as 95.2% and 91.4% for the designs with 20 mm and 80 mm of sphere diameter, respectively. One should also notice that the time-wise variations of the storage tank energy efficiency, flat plate collector efficiency, and storage tank outlet temperature do not significantly differ by varying the storage material diameter from 20 mm to 80 mm. Regarding the collector efficiency, the maximum difference between the designs with 20 mm and 80 mm of sphere diameters is less than 4%. At the end of the charging process, the tank energy efficiency and the outlet temperature of the HTF from the tank are evaluated as 98.1% and 67.4°C, respectively.

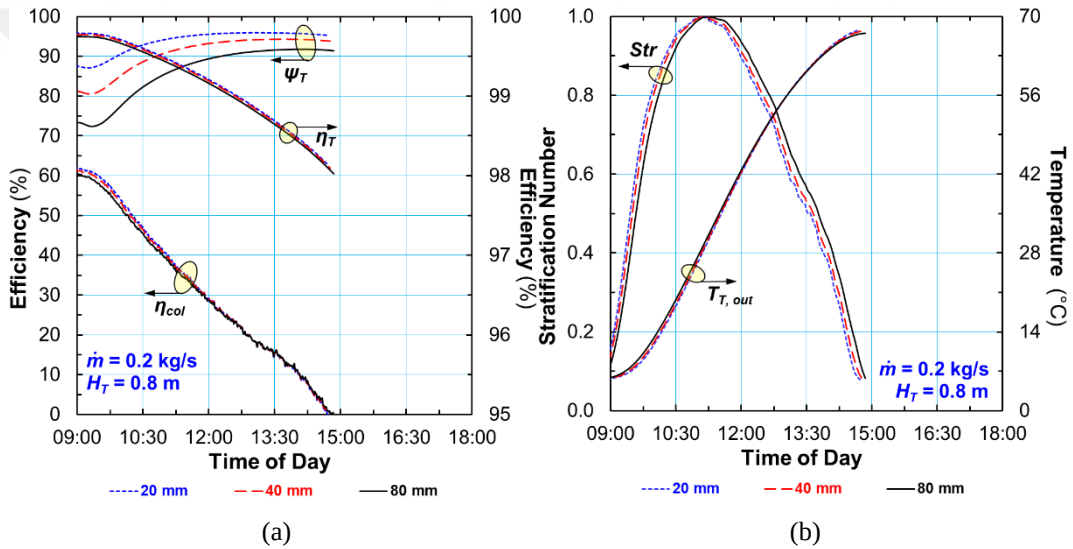


Figure 3.7 The effect of diameter of the sphere on the (a) energy and exergy efficiency of the tank and collector efficiency (b) stratification number and the tank outlet temperature – $H_T = 0.8 \text{ m}$ & January

On the other hand, regarding the stratification number, the curves reach their peak values nearly at the same time for each sphere diameter. The curves slightly shift beyond the peak point, and the rate of change in the stratification is higher for the design with the smallest diameter, i.e., 20 mm. One should also notice that the fluctuations in the weather data are directly observed for the designs with small diameters. The fluctuations are diminished for the design with 80 mm of sphere diameter. At the end of the charging process, stratification number values are almost identical, $Str = 0.09$, for each sphere diameter. Regardless of the tank height, $H_T =$

0.8 m and 1.6 m, the time-wise variations of performance measures have similar trends in Figures 3.7 and 3.8. Besides, the time for complete charging and the outlet temperature of the HTF from the tank differ remarkably as the tank height is increased from 0.8 m to 1.6 m. The HTF tank outlet temperature is approximately 56.05°C for the tank with 1.6 m, which is 11.35°C lower than the small tank. On the contrary, the time for complete charging increases by 57 min, as the tank height is increased from 0.8 m to 1.6 m.

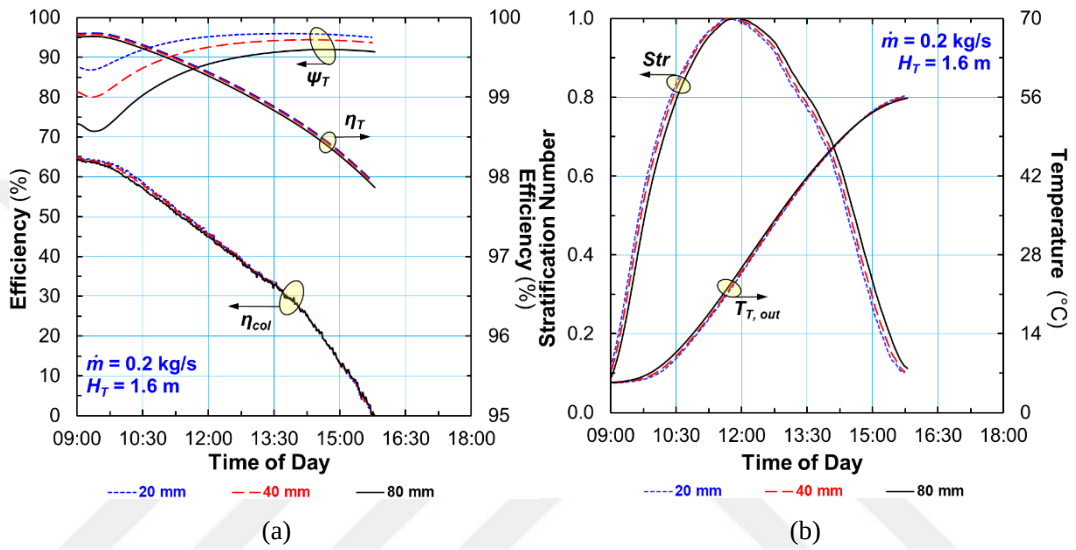


Figure 3.8 The effect of diameter of the sphere on the (a) energy and exergy efficiency of the tank and collector efficiency (b) stratification number and the tank outlet temperature – $H_T = 1.6 \text{ m}$ & January

3.2.3.3 Influence of Storage Tank Height on System Charging Performance

Three different tank heights, *i.e.*, 0.8 m, 1.2 m, and 1.6 m, are considered to evaluate thermal characteristics of the integrated sensible heat TES unit with a flat plate collector. Figures 3.9 and 3.10 represent the results at mass flow rates of 0.1 kg/s and 0.4 kg/s, respectively. In this set of analyses, the sphere diameter is selected to be 40 mm. Figure 3.9 clearly shows that the time for complete charging is significantly affected by the height of the storage tank. Increasing the height of the storage tank also increases the total mass of the storage medium, that is, the time to reach the maximum temperature, *i.e.*, complete charging, increases. Regarding the time for complete charging, the difference between the storage tanks with $H_T = 0.8$

m and $H_T = 1.2$ m is nearly 38 min. The further increment of the tank height from $H_T = 1.2$ m to $H_T = 1.6$ m delays the charging process by 25 min. At the early and final periods of the charging process, the time-wise variations of exergy efficiencies are nearly overlapping. Besides, exergy efficiency slightly differs between 10:30 and 14:15. Instantaneous exergy efficiency reduces as the tank height increases. Figure 3.9(a) depicts that the flat plate collector efficiencies are nearly identical for three different tank heights at the beginning of the charging process. The collector efficiencies are initially 52% and gradually reduces in time. One should notice that the time-wise variation of the collector efficiency hardly depends on the height of the storage tank. The collector efficiency drops more rapidly as the height of the storage tank reduces. A similar trend is also observed in the energy efficiency of the storage tank. The tank efficiency decays more rapidly as the height of the storage tank reduces. Besides, it is interesting to note that, regardless of the height of the storage tank, the energy efficiencies are almost identical at the end of the complete charging process.

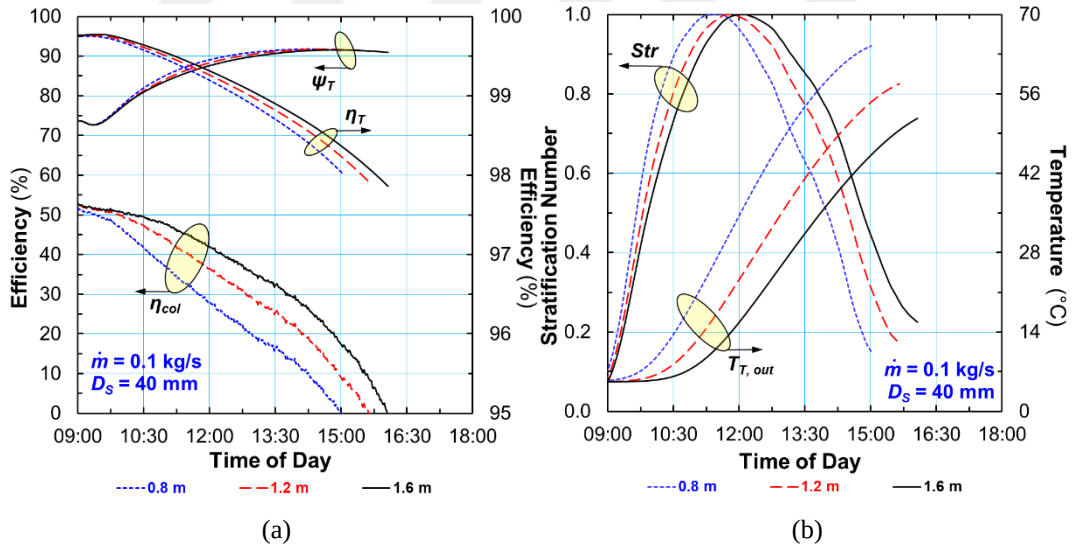


Figure 3.9 The effect of tank height on the (a) energy and exergy efficiency of the tank and collector efficiency (b) stratification number and the tank outlet temperature – mass flow rate of 0.1 kg/s & January

Further results are provided in Figure 3.9b to assess the variations in tank stratification number and the HTF outlet temperature of the tank. The stratification number reaches the unity increment of 20 min with increasing storage tank height,

and it is clear that transient responses differ depending on the tank height. The thermal uniformity of the storage tank rapidly changes for the small storage tank. At the end of the charging process, stratification numbers are achieved as 0.15, 0.17, and 0.22 for $H_T = 0.8$ m, 1.2 m, and 1.6 m, respectively. Moreover, the outlet temperature of the HTF from the tank also significantly varies regarding the tank height. Increasing the tank height also improves the thermal mass of the storage tank. That is the transient response of the tank outlet temperature of the HTF delays as the tank height increases. At the end of the charging process, regarding the tank outlet temperature of HTF, the difference between the storage tanks with $H_T = 0.8$ m and 1.6 m is 12.8°C .

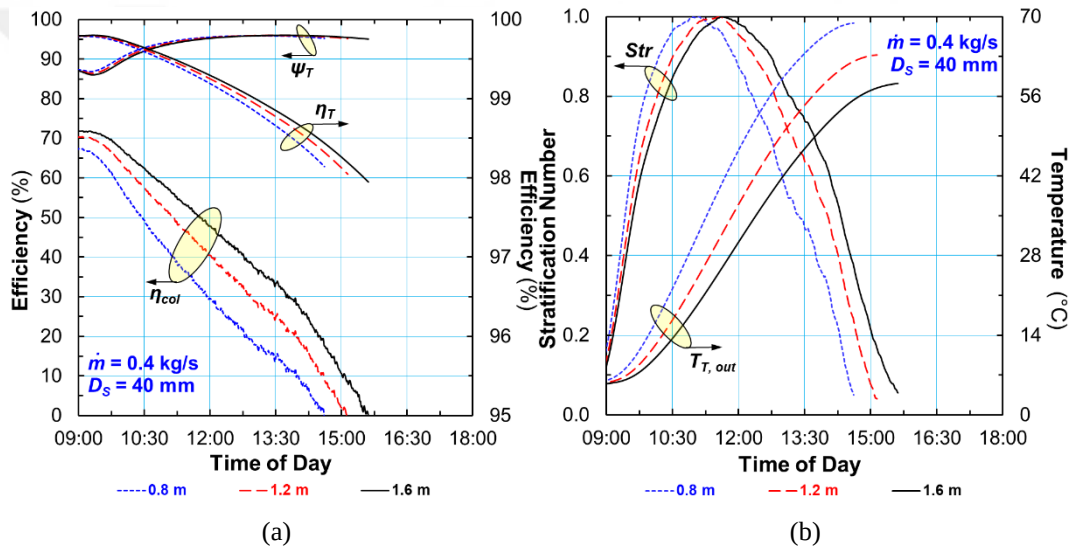


Figure 3.10 The effect of tank height on the (a) energy and exergy efficiency of the tank and collector efficiency (b) stratification number and the tank outlet temperature – mass flow rate of 0.4 kg/s & January

In Figure 3.10, the influence of storage height is given for the mass flow rate of 0.4 kg/s. It is evident that increasing the mass flow rate alters the time for complete charging, exergy efficiencies, and stratification number. Increasing the mass flow rate of the HTF from 0.1 kg/s to 0.4 kg/s reduces the time for complete charging by nearly 24 min, 31 min, and 27 min for the storage tanks with the heights of $H_T = 0.8$ m, 1.2 m, and 1.6 m, respectively. Besides, the maximum exergy efficiency of the tank increases by 4% for the tank with 0.8 m height. It is also interesting to note that the stratification number at the end of the charging process drops significantly

as the mass flow rate of the HTF increases. Stratification numbers are evaluated respectively as 0.05, 0.04, and 0.06 for the storage tanks with the heights of $H_T = 0.8$ m, 1.2 m, and 1.6 m.

3.2.3.4 Assessment of Integrated System Charging Performance

A total of thirty-six transient analyses are conducted for January, and the results are consolidated in Figure 3.11 to assess the performance of the flat plate solar collector integrated sensible heat TES system. Average values are represented for exergy efficiency, energy efficiency, collector efficiency, and heat transfer rate. The mean temperature and stratification number, on the other hand, corresponds to the values that are obtained at the end of the complete charging process.

The influence of mass flow rate on the mean temperature and the heat transfer rate seems to have similar trends for each sphere diameter and the tank height. Increasing the mass flow rate enhances the mean temperature and the heat transfer rate. Mass flow rate, on the other hand, adversely affects the stratification number. Increasing the mass flow rate diminishes the temperature non-uniformities along the flow direction; that is, the stratification, which is the dimensionless measure of uniformity, reduces significantly. The mean exergy efficiency of the storage tank significantly depends on the mass flow rate of the storage tank. Increasing the mass flow rate enhances the exergy efficiency. Notice that the height of the tank does not cause any remarkable variation in the exergy efficiency. The sphere diameter, on the other hand, has a significant influence on the exergetic efficiency. As an instance, for the highest mass flowrate, increasing the sphere diameter from 20 mm to 80 mm reduces the exergy efficiency from 94.1% to 87.5%. It is also interesting to note that the influence of mass flow rate on the exergy efficiency becomes remarkable when the sphere diameter is increased from 20 mm to 80 mm. The exergy efficiency varies in the range of 82.8% to 94.1% when the sphere diameter is 20 mm. Besides, for the largest sphere diameter, *i.e.*, 80 mm, exergy efficiency is between 69.1% and 87.5%. The tank's highest exergy efficiency of 94.1% is achieved with a height of 1.6 m, a sphere diameter of 20 mm, and a mass flow rate of 0.4 kg/s.

Collector efficiency varies from 24.5% to 47.3%, depending on the working and design parameters of the storage tank. Increasing the mass flow rate of the HTF increases collector efficiency. On the other hand, the tank height has a slight effect at low mass flow rates, and the influence of tank height becomes apparent at high flow rates. It should also be noted that increasing the diameter of the spherical capsules slightly reduces the collector efficiency. For instance, increasing the sphere diameter from 20 mm to 80 mm reduces the collector efficiency between 0.5% and 1.8%. The maximum collector efficiency of 47.3% is obtained for the tank with a height of 1.6 m, a sphere diameter of 20 mm, and a mass flow rate of 0.4 kg/s.

The average heat transfer rate improves as the mass flow rate increases. Increasing the mass flow rate from 0.05 kg/s to 0.40 kg/s, the rate of heat transfer increases by 27.5% for the tank with height and sphere diameter of 0.8 m and 20 mm, respectively. On the other hand, when the tank height and sphere diameter are increased to 1.6 m and 80 mm, respectively, the difference between the lowest and highest mass flow rates becomes 41.9%. The highest average heat transfer rate is obtained as 3.02 kW for the configuration with a tank height of 1.6 m, the mass flow rate of 0.4 kg/s, and the sphere diameter of 20 mm.

The mean temperature of the energy storage tank increases with increasing mass flow rate. On the contrary, the mean temperature reduces when the diameter of the storage material and tank height increase. Increasing tank height from 0.8 m to 1.6 m, the mean temperature of the storage tank reduces by 20.5-percent when the mass flow rate is 0.05 kg/s. At the highest flow rate, *i.e.*, 0.40 kg/s, the reduction becomes approximately 15.3-percent. Additionally, the maximum mean temperature becomes 69.1°C when the tank height is 0.8 m, the mass flow rate is 0.4 kg/s, and the sphere diameter is 20 mm.

The thermal stratification number gives information about the degree of homogeneity within the storage during the charging process. Notice that increasing mass flow rate reduces the stratification number. The stratification number slightly increases as the tank height is increased. The worst scenario occurs when tank

height is 1.6 m, the mass flow rate is 0.05 kg/s, and the sphere diameter is 40 mm. On the other hand, regarding stratification number, the best scenario is evaluated when tank height is 1.2 m, the mass flow rate is 0.4 kg/s, and the sphere diameter is 20 mm.

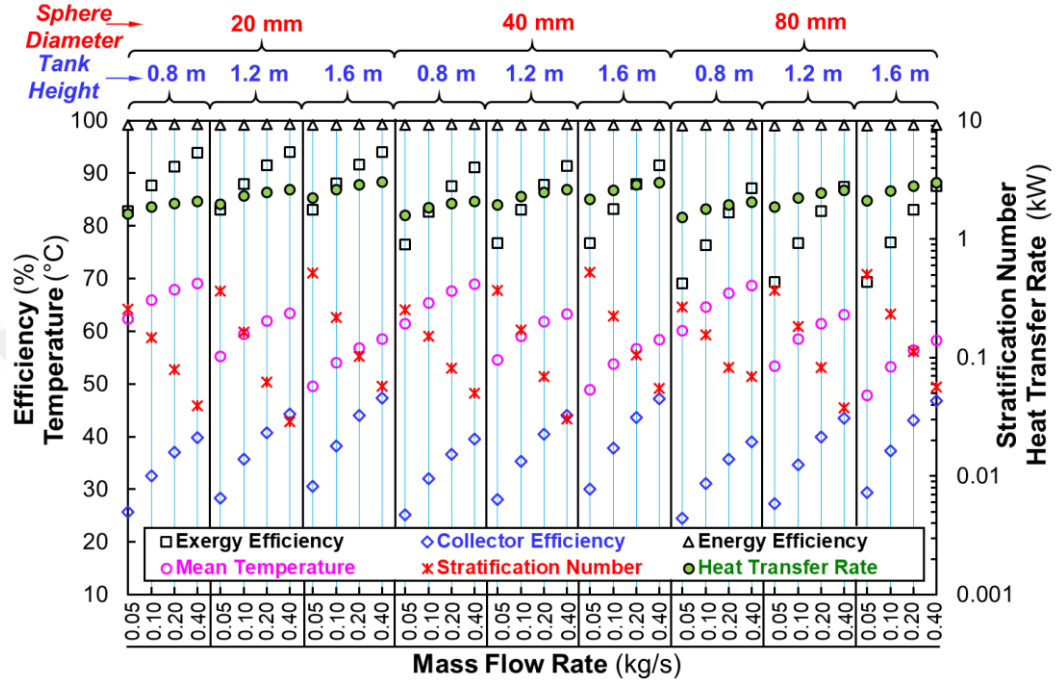


Figure 3.11 An overview of the system performance indicators

Response surface methodology technique is used to evaluate the optimum design and working parameters for the charging process of the flat plate solar collector integrated sensible heat TES tank. As an optimization analysis software, Minitab 18.1 is utilized. The heat transfer rate and exergy efficiency are primarily selected to represent both first and second law aspects. The quadratic regression method is used with $R^2 = 95.5\%$. The mass flow rate of 0.4 kg/s, tank height of 1.6 m, and sphere diameter of 20 mm are evaluated as optimum design and working parameters that maximize the rate of heat transfer and exergy efficiency in January.

Thanks to the integration of the TES tank with the heat pump, a heating system that is more efficient than a conventional air-source heat pump can be obtained during the discharging process. In this integration, the stored thermal energy is directly used as a heat source from the evaporator component of the heat pump.

According to Equations 2.40 and 2.41, the heat pump's performance coefficient (COP) is only dependent on the ambient temperature and the temperature of the HTF leaving the TES tank, respectively. According to Figure 2.2, the ambient temperature at night-time varies from 5°C to 11°C. Besides, the mean temperature of the TES tank at the end of the charging period roughly varies from 47.8°C to 69.1°C. The corresponding COP values for these temperatures could be simply evaluated from Equations 2.49 and 2.50 as listed in Table 3.3. The water-source heat pump provides up to 8% higher COP values when it is compared with the air-source heat pump. It is worth noting that supplying a heat pump with a high temperature significantly enhances the COP. As a baseline case, an air-source heat pump which is working at an ambient temperature of 10°C, the COP is evaluated as 3.27. Besides, it is clear that supplying the heat pump with a high-temperature reservoir considerably improves the performance. Current results reveal that integrating a sensible heat TES to a water-source heat pump could allow high-grade thermal energy from 35°C to 50°C, and the corresponding COPs are 4.49 and 5.43, respectively. This means that the sensible heat TES integrated heat pump system could improve the COP up to 66% when it is compared to the baseline case.

Table 3.3 Effect of source temperature on heat-pump performance

Temperature (°C)	COP _{air}	COP _{water}
5	3.04	3.32
10	3.27	3.46
15	3.47	3.61
35	—	4.49
50	—	5.43
70	—	7.12

3.2.3.5 Influence of Operating Month on System Charging Performance

After determining the optimum working and design parameters of the flat plate solar collector integrated sensible heat TES tank, energetic and exergetic performances are also evaluated for other months when the heating needs. The effect of operating month on the time-wise variation of the exergy efficiency,

energy efficiency, collector efficiency, stratification number, and tank outlet temperature are illustrated in Figure 3.12. Figure 2.2 indicates that the solar radiation variations do not significantly differ for December and January. However, for November, solar radiation becomes 20% higher than in December and January. However, regarding the ambient temperature, the maximum deviation is more than 10°C. The difference between the sunrise and sunset times in the selected months affects the charging duration. The most extended period of the storage process takes place in November with 8.0 hours. The charging duration reduces to 7.7 h for January and February. It is observed that variations in exergy efficiency are significantly affected by weather data in the early period of the charging process. Beyond noon the variations in exergy efficiency are almost identical. Regarding the tank outlet temperature of HTF, the highest and lowest values are evaluated in November, 76.8°C, and January, 58.3°C, respectively. While February has almost similar tank outlet temperature to January, tank outlet temperature becomes 64.6°C for February at the end of the charging process. Collector efficiency and energy efficiency of the tank are only affected by the sunrise time. The efficiencies gradually decrease due to the increasing heat loss through the ambient in the proceeding time. Additionally, the stratification number is almost identical for all months and reaches 0.06 at the end of the charging process.

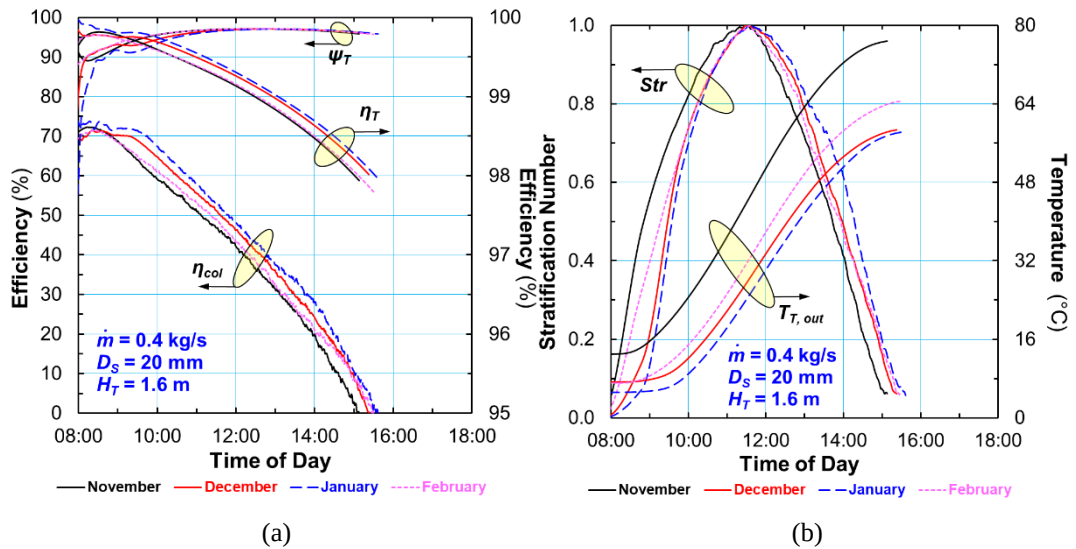


Figure 3.12 The effect of operating month on the (a) energy and exergy efficiency of the tank and collector efficiency (b) stratification number and the tank outlet temperature

Average values of exergy efficiency, energy efficiency, collector efficiency, and heat transfer rate are given in Table 3.4, together with the stratification number and mean temperature at the end of the charging process. The maximum exergy efficiency becomes 95.43% for February. However, no significant variation is observed regarding maximum exergy efficiency. The difference between the minimum (in January) and maximum (in February) values is 1.36%. For the selected months, no significant change in energy efficiency is reported. The minimum stratification number occurs as 0.057 at the end of the charging process for January; however, the values for the other months are close to this range. The maximum mean heat transfer rate is obtained in November as 3.48 kW. Regarding the rate of heat transfer, the difference between November and December is 14.1%. However, depending on the solar radiation, the maximum mean temperature of the storage tank becomes 76.95°C for November. The mean temperature values for December and January are almost 18°C lower than in November.

Table 3.4 The integrated system performance for different operating months

Operating Month	Energy Efficiency (%)	Collector Efficiency (%)	Exergy Efficiency (%)	Mean Temperature (°C)	Heat Transfer Rate (kW)	Stratification Number (-)
November	99.19	46.13	94.65	76.95	3.48	0.062
December	99.22	46.47	94.92	58.85	2.99	0.062
January	99.23	47.29	94.07	58.48	3.02	0.057
February	99.10	43.99	95.43	64.75	3.24	0.058

3.3 Charging Performance of Latent Heat Packed Bed TES System

As design parameters, spherical capsule diameter and type of PCM arrangements are investigated to evaluate the latent heat TES system's energetic and exergetic performance. Besides, the effect of the HTF's mass flow rate is also examined as an operating parameter. Capsule diameter and mass flow rate are varied as 20, 40, and 60 mm and 0.2, 0.4, 0.8, and 1.6 kg/s, respectively. Seven type PCM arrangements include cascade (*double/triple-type*) and non-cascade (*single-*

type) systems, which are studied, and the details are represented in Table 3.5. A total of 84 parametric runs is conducted for each heating season month.

Table 3.5 Single, double and triple PCM Arrangements inside the TES tank

Arrangement type	Case number	Melting temperature of PCMs (T_m) along the flow direction (°C)		
Single (non-cascaded)	#1			30
	#2			40
	#3			50
Double (cascaded)	#4	40		30
	#5	50		30
	#6	50		40
Triple (cascaded)	#7	50	40	30

The dynamic response and the performance of the solar-collector integrated latent heat TES system are evaluated by monitoring the time-wise variation of the mean PCM temperature and liquid fraction, collector energy efficiency, storage tank exergy efficiency, and exergy/energy efficiencies of the integrated storage system. Additionally, in some instances, liquid fraction variation along the flow direction is also represented to understand the heat transfer process for different PCM arrangements. The independent variables are selected as weather conditions, PCM arrangements, the mass flow rate of HTF, and the spherical capsules' diameter. The analyses are terminated when the maximum charging occurs. Time-wise variations of charging performance analyses are given only for January to be more concise. In the last part of this section, the comparisons of all alternative designs and working conditions of the system are given for four selected months to evaluate the optimum charging performance of the latent heat TES system.

3.3.1 Validity of Enhanced Thermal Conductivity Approach

Various Nusselt correlations are available in the literature for the natural convection inside the spherical annular gap. Four different Nusselt correlations

(Table 3.6) presented in the recent review of Liao et al. (2018) are implemented in the current code.

Table 3.6 Nusselt correlations for natural convection inside the spherical annulus

Correlation	Reference	
$Nu = 0.21(Ra_\delta)^{0.24}$	Chiu & Chen (1996)	(Equation 3.5)
$Nu = (3.8475 \times 10^{-8}) Ra_{r_i} + 1.5859$	Amin et al. (2014)	(Equation 3.6)
$Nu = 0.228 \left(Ra_\delta \frac{\delta}{r_i} \right)^{0.226}$	Scanlan et al. (1970) – C1	(Equation 3.7)
$Nu = 0.202 (Ra_\delta)^{0.228} \left(\frac{\delta}{r_i} \right)^{0.252} Pr^{0.029}$	Scanlan et al. (1970) – C2	(Equation 3.8)

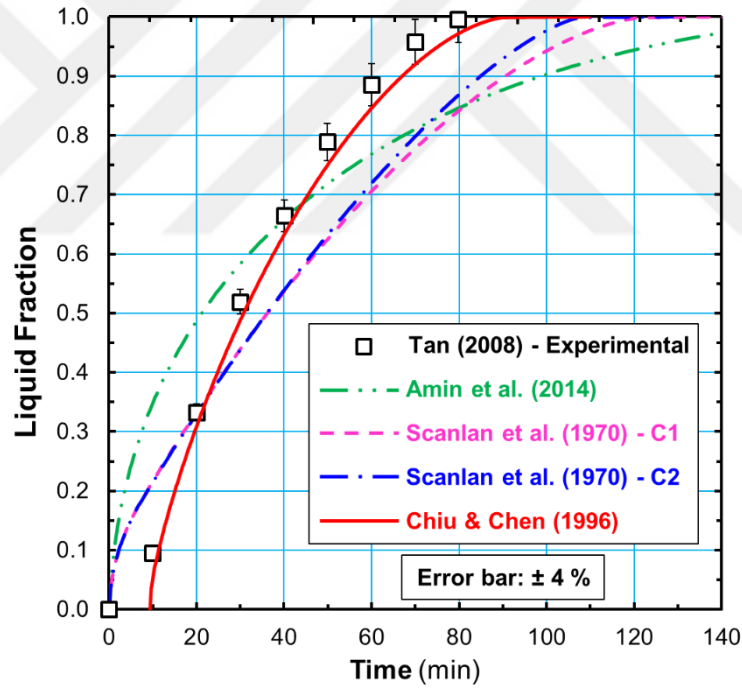


Figure 3.13 Time-wise variations in liquid fractions with different Nusselt correlations

In order to ensure the validity of the enhanced thermal conductivity approach that is convenient for simulating the inward melting of the PCM, the problem of Tan (2008) is reproduced. The authors experimentally studied the inward melting of the PCM, n-octadecane, in a spherical capsule with a 101.66 mm diameter and

shell thickness is 1.5 mm. In this reference work, PCM is initially at 26.5°C, and then suddenly, capsule surface temperature increased to 45°C. The time-wise variation of the liquid fraction is observed throughout the process by using a novel image capturing technique. The numerical results that are obtained by defining different Nusselt correlations are compared with the experimental findings of Tan (2008). As shown in Figure 3.13, the correlation proposed by Chiu & Chen (1996) gives the closest predictions to the experimental findings of reference work with less than 4 %, and this correlation is adopted in the parametric analyses.

3.3.2 Mesh and Time Step Size Independency Study

Mesh independency analyses are performed to obtain the optimum number of mesh for the PCM domain. Mesh number is varied between 20 and 100 when the time-step size and the diameter of the spherical storage material are kept constant as 0.5 s and 60 mm, respectively. The effect of mesh number on energy and exergy efficiencies of the integrated system at the end of the charging process is chosen as a decisive criterion, and the results are represented in Table 3.7. Regarding the energy and exergy efficiencies, the percentage differences between 40 and 100 nodes are found as 0.01 and 0.02, respectively. Besides, the computing time is reduced by 6.2 times. That is, the optimum mesh number for the PCM domain is determined as 40.

Table 3.7 The results of the mesh independency study

PCM domain mesh number	20	40	60	80	100
System exergy efficiency (%)	3.745	3.746	3.747	3.747	3.747
System energy efficiency (%)	46.597	46.604	46.606	46.607	46.608
Computing time (s)	2572	5458	6793	8097	34004

Additionally, time-step-size independency analyses are also carried out when the mesh number for the PCM domain is 40, and the diameter of the spherical

storage material is 60 mm. The time-step size is varied between 0.1 and 2 s, and the results are represented in Table 3.8. The percentage difference for system energy and exergy efficiencies with 0.5 s and smallest time-step sizes are evaluated as 0.03 and 0.07, respectively. Besides, the computing time of 0.5 s time-step size is 3.7 times faster than 0.1 s. That is, for parametric analyses, the optimum time-step size is found as 0.5 s.

Table 3.8 The results of the time step independency study

Time step size (s)	0.1	0.2	0.5	1.0	2.0
System exergy efficiency (%)	3.749	3.748	3.746	3.741	3.736
System energy efficiency (%)	46.618	46.615	46.604	46.568	46.546
Computing time (s)	20102	11728	5458	1606	721

3.3.3 Validity of the Mathematical Model of Packed Bed Latent Heat TES System

The mathematical model of the packed bed TES system is validated with the numerical and experimental results of Li et al. (2018). In the reference work, the inlet temperature of the HTF is kept constant at 465°C throughout the analyses, while the initial temperature of the TES tank is 325°C in the charging mode. The inner diameter of the PCM capsules is 30 mm. The time-wise variations of the center temperature of the PCM capsules at three different heights are illustrated in Figure 3.14 with the numerical and experimental results of Li et al. (2018). It can be seen from the figure that the current model accurately predicts the numerical results of the reference work at the sensible heating process of the charging. However, with the starting of the phase change process, the difference between the current work results and the experimental results of the reference study becomes higher. This inconsistency can occur for several reasons, such as the assumptions related to the TES system's heat loss or experimental uncertainties (Li et al., 2018).

The results show that predicted time-wise PCM center temperature variations have a good harmony with reference work's numerical findings.

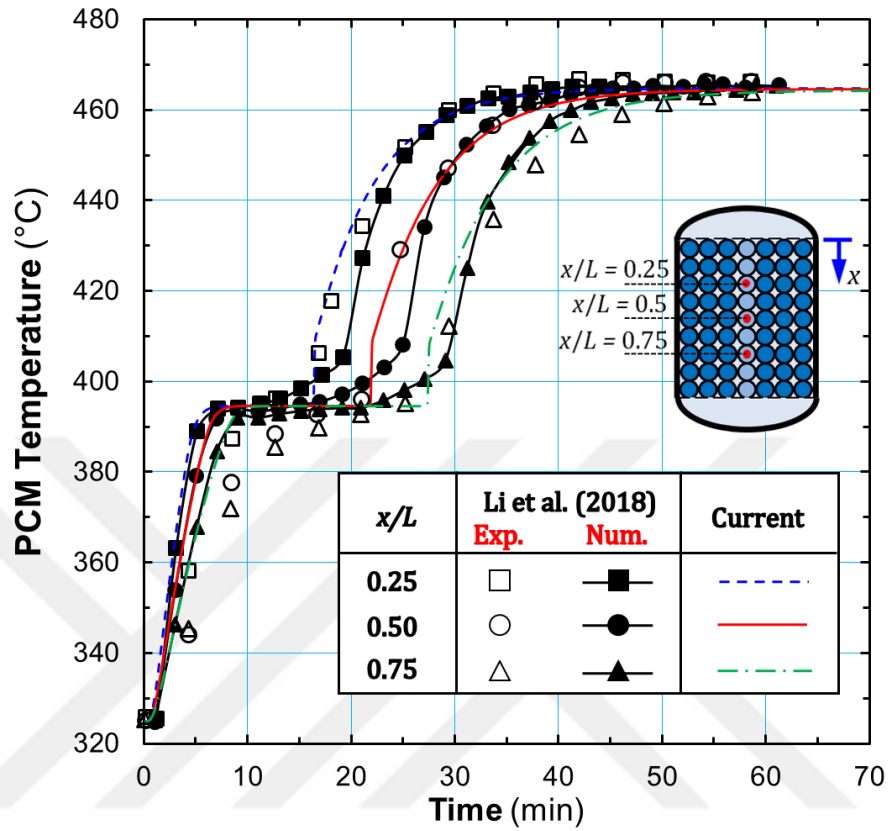


Figure 3.14 The comparison of the time-wise variations of PCM temperature at three positions along the flow direction

3.3.4 Parametric Results of Latent Heat Packed Bed TES System

The dynamic response and the charging performance of the solar collector integrated latent heat TES system is evaluated by monitoring the time-wise variation of the mean PCM temperature and liquid fraction, collector energy efficiency, storage tank exergy efficiency, and exergy/energy efficiencies of the integrated storage system. Additionally, liquid fraction variation along the flow direction is also represented in certain instances to provide a better understanding of the heat transfer process for different PCM arrangements. The independent variables are selected as weather conditions, PCM arrangements, the mass flow rate of HTF, and the spherical capsules' diameter. The analyses are terminated when the maximum charging occurs. Time-wise variations of charging performance analyses

are given only for January to be more concise. In the last part of this section, the comparisons of all alternative design and working conditions the system are given for four selected months to evaluate the optimum charging performance of the latent heat TES system

3.3.4.1 Effect of Mass Flow Rate on Integrated System Charging Performance

The effect of the mass flow rate of the HTF is studied to characterize the performance of the charging process of the storage system and is varied from 0.2 kg/s to 1.6 kg/s. Even though the flow rate is varied for each design alternative and working conditions, the time-wise variations are presented for a specific condition due to the limited space. As a design parameter, the storage material diameter is kept constant as 20 mm, and the coldest month is selected, *i.e.*, January. The time-wise variation of the mean PCM temperature and the liquid fraction is illustrated in Figure 3.15 for four different PCM arrangement cases. The variation of time-wise mean PCM temperature shows an almost independent behavior from the mass flow rate of the HTF in the sensible heating part of the storage. In single type PCM arrangement cases, Case #1 and Case #3, with the start of the phase change, higher mean PCM temperatures are observed when the mass flow rate has a lower value. However, when the PCM completely turns into the liquid phase, this fact reverses, and the highest mean PCM temperatures occur at the highest mass flow rate, except for Case #3 PCM arrangement ($T_m = 50^\circ\text{C}$). Notice that, in Case #3, complete melting could not be obtained, and the maximum liquid fraction occurs as 0.913 for the mass flow rate of 1.6 kg/s. The mean PCM temperature is approximately 50.3°C for each of the selected mass flow rates. Similarly, for the double (Case #5) and triple (Case #7) cascaded PCM arrangements, systems with lower mass flow rates perform better at the start of the latent heat TES process. This behavior is reversed as the complete melting is achieved, and the maximum PCM temperature values are observed for the higher mass flow rate of HTF. Consequently, with the increasing mass flow rate, the heat transfer rate increases; as a result of this, the liquid fraction reaches its maximum value earlier for all PCM arrangement cases. Additionally, overall charging time slightly decreases with the increase of the mass flow rate.

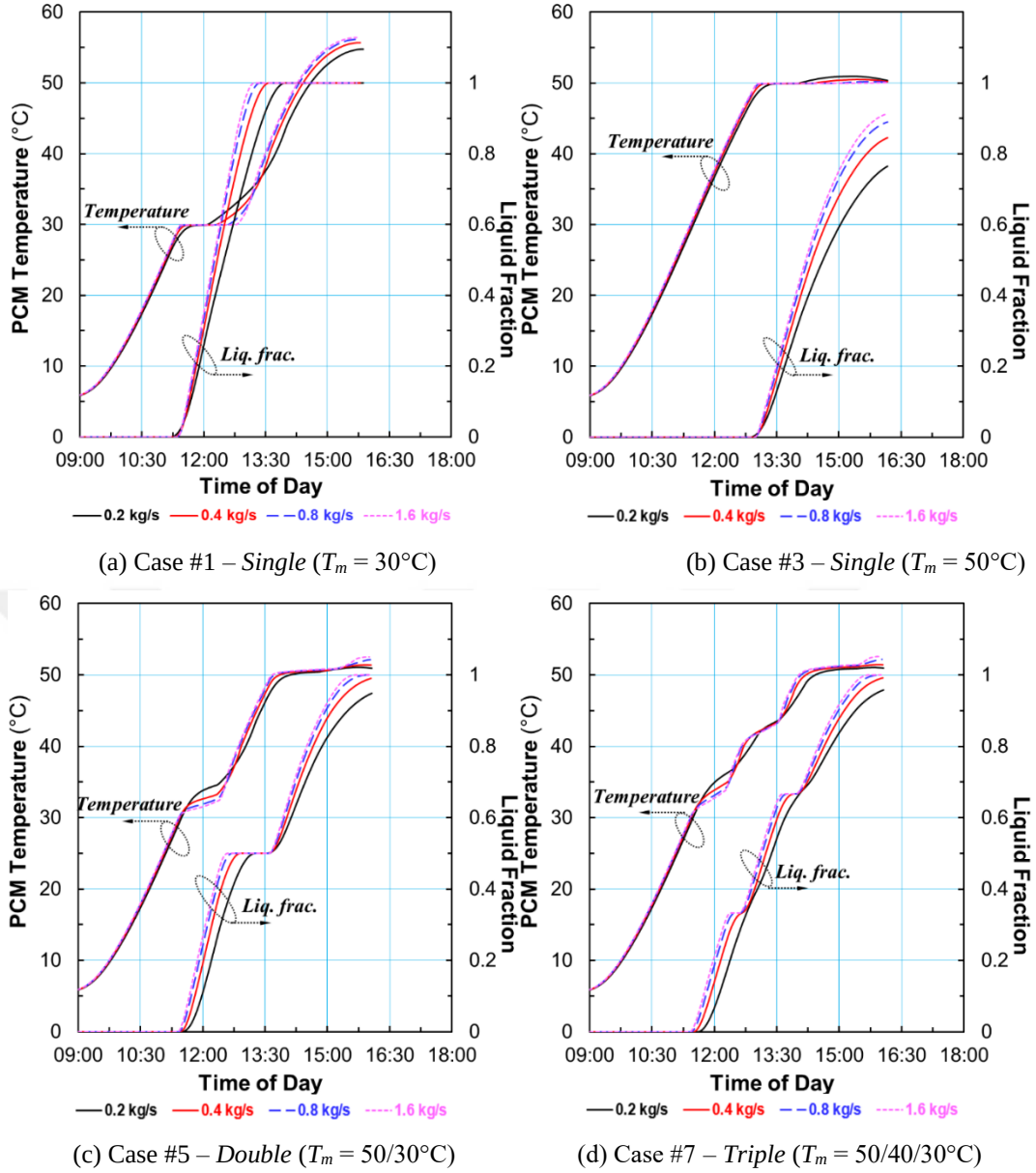


Figure 3.15 The effect of mass flow rate on the mean PCM temperature and liquid fraction for different PCM arrangements – $D_s = 20$ mm & January

Figure 3.16 represents the liquid fraction variation inside the TES tank at selected instances for two single-type PCM arrangements, *i.e.*, Case #1 & Case #3, for four different mass flow rates of the HTF. The solid and dashed lines are for spatial liquid fractions of Case #1 and Case #3, respectively. The charging rate of the latent heat TES system increases with the increasing mass flow rate. Additionally, it is observed that the gradient of the liquid fraction along the flow direction becomes smaller at higher mass flow rates, *i.e.*, the stored energy in the TES tank becomes homogeneous. For Case #1 ($T_m = 30^\circ\text{C}$), the PCM is completely

the liquid phase at 16:00 becomes for each of the selected mass flow rates. Besides, for Case #3 ($T_m = 50^\circ\text{C}$), the mean liquid fraction inside the TES tank occurs as 0.59 for the mass flow rate of 0.2 kg/s at 16:00. As the mass flow rate increases by eight times to 1.6 kg/s, the mean liquid fraction becomes 0.75, corresponding to a 27% enhancement in the charging rate. One should notice that at 0.2 kg/s, 27% of the PCM capsules completely turn into a liquid phase from the tank's inlet. However, at 1.6 kg/s, complete melting could not be achieved inside the capsules. The maximum liquid fraction is observed to be 0.9 at the inlet of the tank.

In Figure 3.17, the spatial liquid fraction variations are compared for the cascaded-type PCM arrangements, *i.e.*, Case #5 & Case #7. Similar to the single-type PCM arrangements, the heat transfer rate increases with the increasing mass flow rate, and as a result, gradients of the liquid fraction curves along with the tank height decreases. For Case #5 ($T_m = 50/30^\circ\text{C}$), it is seen that PCM capsules with low phase melting temperature, *i.e.*, $T_m = 30^\circ\text{C}$, are completely turned into the liquid phase for each of the selected mass flow rates at 16:00. It is also observed that for PCM capsules with high melting temperature, *i.e.*, $T_m = 50^\circ\text{C}$, the lowest liquid fraction value occurs as 0.27 in the storage tank for the mass flow rate of HTF is 0.2 kg/s. As the mass flow rate is increased to 1.6 kg/s, the lowest liquid fraction value rises to 0.77. Additionally, it is found that the mean liquid fraction value is increased from 0.83 to 0.92 by 11% with increasing the mass flow rate from 0.2 kg/s to 1.6 kg/s. For Case #7 ($T_m = 50^\circ\text{C}-40^\circ\text{C}-30^\circ\text{C}$), it is clear that PCM capsules which are filled with moderate, *i.e.*, $T_m = 40^\circ\text{C}$, and low, *i.e.*, $T_m = 30^\circ\text{C}$, melting temperatures are completely turned into the liquid phase. For the capsules with PCM at the highest melting temperature, *i.e.*, $T_m = 50^\circ\text{C}$, complete melting does not occur for selected working conditions at the end of the charging process. Similar to Case #5, in Case #7, the variation of the mean liquid fraction inside the storage tank increased from 0.84 to 0.92, with the increasing mass flow rate from 0.2 kg/s to 1.6 kg/s.

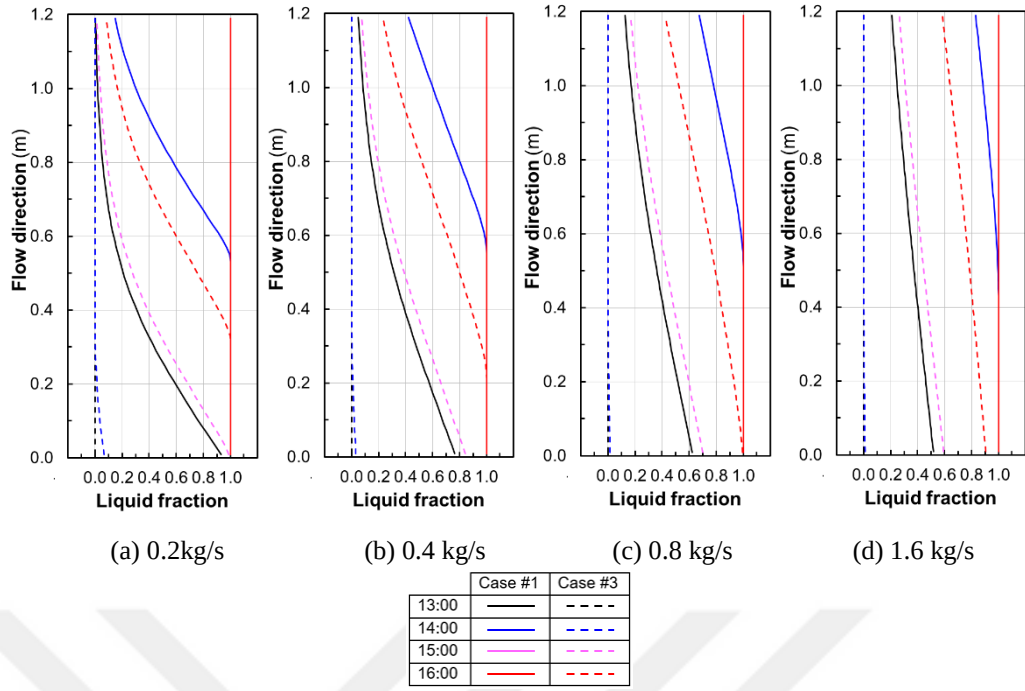


Figure 3.16 Spatial liquid fraction variations of the different PCM arrangements (Case #1 & Case #3) at certain instances for four different mass flow rates of HTF – $D_S = 20$ mm & *January*

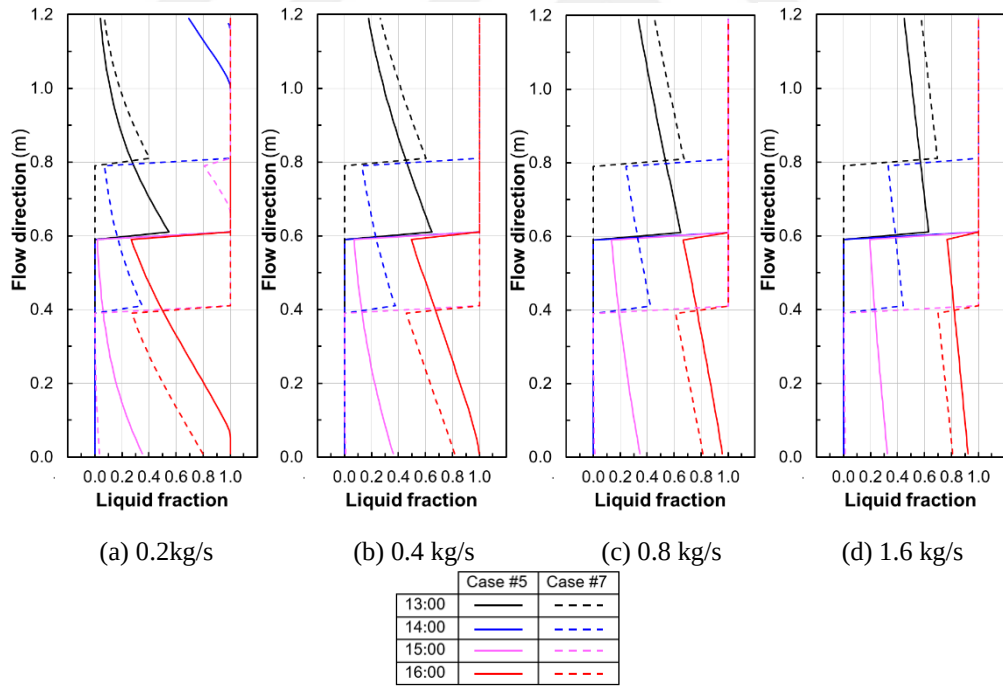


Figure 3.17 Spatial liquid fraction variations of the different PCM arrangements (Case #5 & Case #7) at certain instances for four different mass flow rate of HTF – $D_S = 20$ mm & *January*

The time-wise variations of the energy efficiencies of the integrated system and collector and exergy efficiencies of the integrated system and storage tank are

illustrated in Figure 3.18 for four different PCM arrangements. The effect of the mass flow rate of the HTF on the efficiencies has a similar pattern, as discussed before. Increasing the mass flow rate generally increases all presented efficiencies. For Case #1 ($T_m = 30^\circ\text{C}$), the integrated system energy efficiency increases by 2.11%, 1.70%, and 0.36% in doubling the mass flow rate from 0.2 to 0.4 kg/s, from 0.4 to 0.8 kg/s and from 0.8 to 1.6 kg/s, respectively. Similarly, integrated exergy efficiency increases by 3.59%, 2.48%, and 0.73% for similar variations. Therefore, it can be mentioned that the effect of increasing mass flow rate decreases gradually, and this tendency is also observed in the other PCM arrangements. Mean collector efficiencies for the mass flow rate of 1.6 kg/s become as 55.12%, 49.52%, 51.99%, and 52.01% for Case #1, Case #3, Case #5, and Case #7, respectively. It can be concluded that increasing the melting temperature of the PCMs decreases mean collector efficiency.

Increasing the mass flow rate of the HTF enhances the exergy efficiency of the storage tank significantly for the PCM that has a lower melting temperature. However, for the PCMs with higher melting temperatures, the effect of the mass flow rate of HTF on the exergetic efficiency of the tank becomes insignificant. It is also interesting to note that, in all simulations, exergy efficiencies of the storage tank have an upward tendency from the beginning of the simulations to particular instances when the phase change process initiates. As the phase change process continues, exergy efficiencies decrease until complete melting is achieved. Beyond this moment, exergy efficiencies tend to increase again. The integrated system exergy efficiency shows a similar increasing tendency with the storage tank exergy efficiency at a sensible heating process. However, the effect of the mass flow rate of the HTF on system exergy efficiency seems insignificant for system exergy efficiency at sensible heating. With the beginning of the latent heat storage process, the increasing trend of the system exergy efficiency decreases, and the effect of the mass flow rate of the HTF becomes apparent. The system exergy efficiency reaches a peak value when the solar irradiation starts to decrease. When the time-wise variation of the integrated system energy efficiency is examined, it can be said that with the starting of the phase change process, the slope of the decreasing trend of

system energy efficiency becomes smoother because of decreasing the increment rate of the temperature difference between HTF and ambient air.

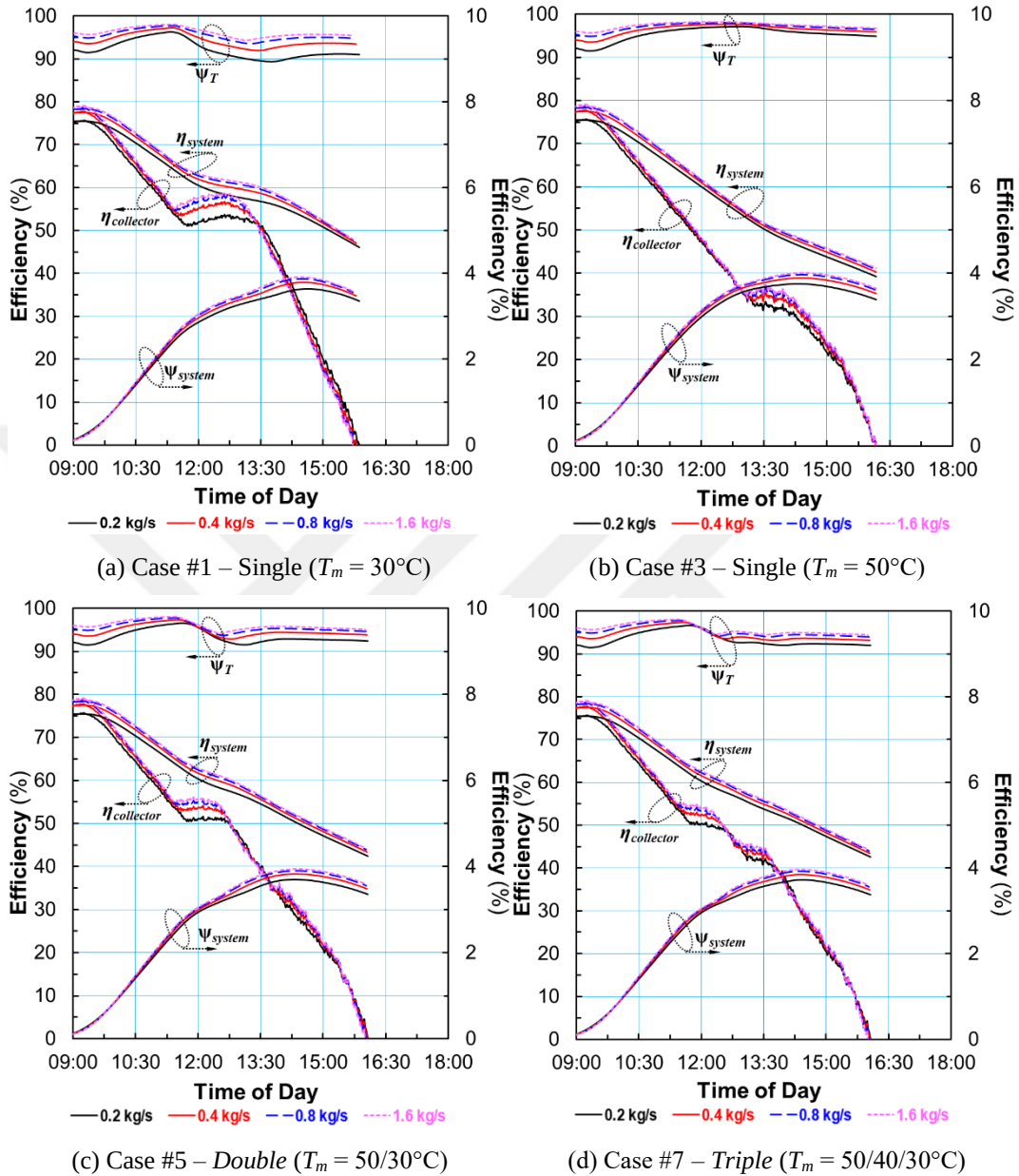


Figure 3.18 The effect of mass flow rate on the exergy efficiency of the storage tank, energy and exergy efficiencies of the integrated system, and collector efficiency – $D_s = 20$ mm & January

Additionally, the effect of mass flow rate on system energy efficiency occurs more apparent, especially for Case #1. Regarding the energy efficiency of the integrated system, at the end of the charging process, the effect of the mass flow rate gains more importance when the phase change temperature increases. When investigating the relationship between melting temperature of PCM and system

exergy efficiency, it can be mentioned that increasing phase change temperature from 30°C (Case #1) to 50°C (Case #3) increases system exergy efficiency, and the maximum enhancement is achieved as 1.77% for the mass flow rate of 1.6 kg/s.

3.3.4.2 Effect of Capsule Diameter on Integrated System Charging Performance

Parametric analyses are conducted for three different capsule diameters, *i.e.*, 20 mm, 40 mm, and 60 mm. In this set of analyses, the thickness of the shell and the mass flow rate of the HTF are kept constant as 1 mm and 0.8 kg/s, respectively. The analyses are conducted for January. Figure 3.19 shows the development of mean PCM temperature and liquid fraction for four different PCM arrangement cases of the charging process. Altering the capsule diameter does not significantly affect the starting time of the phase change process for all PCM arrangements. Besides, the time for completing melting is affected by the amount of PCM within the storage tank, as expected. For Case #1 ($T_m = 30^\circ\text{C}$), the charging process ends 10.2 minutes late when the capsules' diameter is increased from 20 mm to 40 mm. Similarly, charging occurs 7.7 minutes late when the diameter increases from 40 mm to 60 mm for Case #1.

The PCM temperatures overlap during the initial sensible heat storage period. At the early periods of the melting process, the PCM temperatures are higher for the larger spheres. However, as complete melting is achieved, the temperatures reverse, and higher temperature values are observed for the smaller capsules. For Case #3 ($T_m = 50^\circ\text{C}$), complete melting is not achieved for the selected capsule diameters. For Cases #5 and #7, step-like variations are observed in the temperature and liquid fraction values. Notice that, regardless of the capsule diameter, PCM with $T_m = 30^\circ\text{C}$ completely turns into the liquid phase both in Case #5 and Case #7. However, in a double PCM arrangement, *i.e.*, Case #5, the complete melting for the PCM with $T_m = 50^\circ\text{C}$ is achieved for only $D_s = 20$ mm. On the other hand, for the triple PCM arrangement, *i.e.*, Case #7, even though the complete melting for the PCM with $T_m = 40^\circ\text{C}$ is observed in each capsule diameter, the complete melting for the PCM with $T_m = 50^\circ\text{C}$ is obtained only for the smallest capsule, *i.e.*, $D_s = 20$ mm.

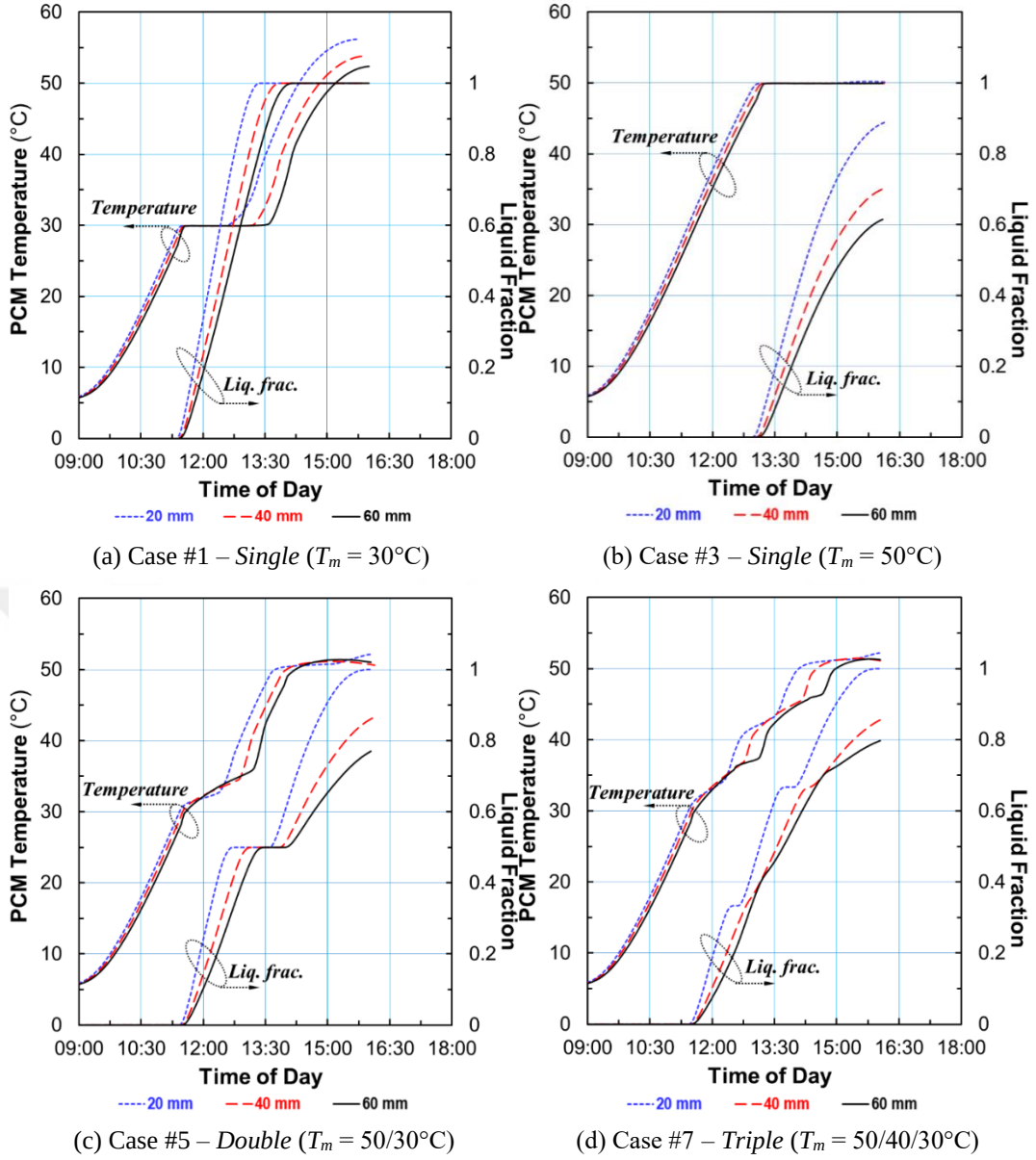


Figure 3.19 The effect of storage material diameter on the mean PCM temperature and liquid fraction for different PCM arrangements– mass flow rate of 0.8 kg/s & January

The effect of capsule diameter on liquid fraction variation along the TES tank's height is illustrated in Figures 3.20 and 3.21 at four selected instances. The liquid fraction curves become steeper with the increasing capsule diameter. Additionally, the mean liquid fraction decreases with increasing the diameter of the capsules. Complete melting only occurs in Case #1 ($T_m = 30^\circ\text{C}$) for all capsule diameters for the selected cases. When the diameter increases from 20 mm to 60 mm, the liquid fraction value decreases from 0.73 to 0.48 for Case #3 ($T_m = 50^\circ\text{C}$).

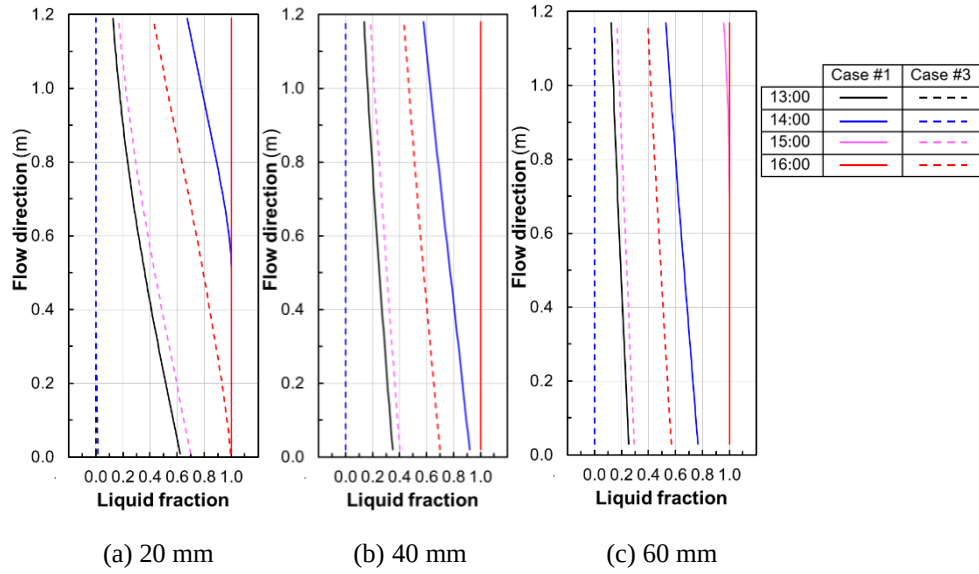


Figure 3.20 Spatial liquid fraction variations of the different PCM arrangements (Case #1 & Case #3) at certain instances for different sphere diameters – mass flow rate of 0.8 kg/s & January

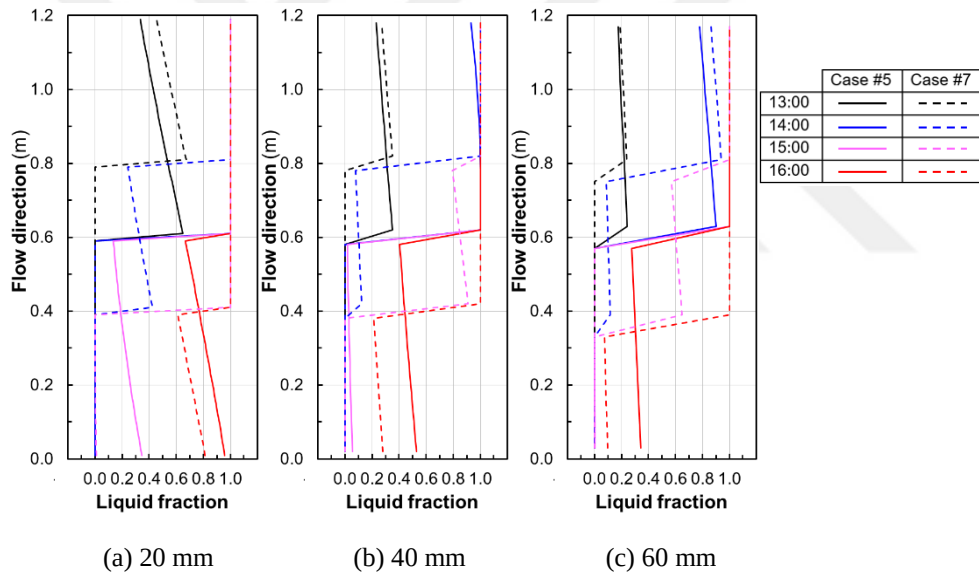


Figure 3.21 Spatial liquid fraction variations of the different PCM arrangements (Case #5 & Case #7) at certain instances for different sphere diameters – mass flow rate of 0.8 kg/s & January

For the cascaded-type PCM arrangements (Case #5 & Case #7), it is seen that complete melting occurs at the end of the charging process for the PCMs with low or medium melting temperatures. Besides, for the high-temperature PCM ($T_m = 50^\circ\text{C}$), when the diameter of the capsule is 20 mm, the mean liquid fraction of PCM capsules in this region becomes 0.82 and 0.71 for Case #5 and Case #7, respectively

at 16.00. However, considering the mean liquid fraction values inside the TES tank, no significant change is observed between the two cases.

The effect of capsule diameter on the time-wise variations in the energy efficiency of flat plate solar collector, exergy efficiency of the storage tank, and energy and exergy efficiencies of the integrated system is illustrated in Figure 3.22. Here the mass flow rate is defined as 0.8 kg/s, and simulations are carried out for January. Until the phase change process in the storage tank initiates, energy efficiencies of the flat plate solar collector and integrated system act independently from the diameter of the capsules. With the beginning of the phase change process, the integrated system and collector energy efficiencies of the TES systems with capsule diameters of 20 mm become higher than the other scenarios. As the amount of PCM is lower in that configuration, capsule, it has the highest phase change rate. Additionally, at the end of the charging period, the effect of capsule diameter in terms of the integrated system and the collector energy efficiencies becomes insignificant. For Case #1 with a 20 mm capsule diameter, after the complete melting is achieved, the HTF temperature tends to rise dramatically; as a result of this, the collector efficiency decreases rapidly with the increasing heat loss through the surroundings. At the end of the charging period, for capsule diameter of 20 mm, integrated system energy efficiencies of Case #1, Case #3, Case #5, and Case #7 become 47.85%, 40.90%, 43.94%, and 44.00%, respectively. The maximum difference in terms of the system operation time, or the charging time, between the systems with capsule diameters of 20 mm and 60 mm, is obtained as 30 mins for Case #1. However, for the PCMs with higher melting temperatures, the effect of the capsule diameter on charging time becomes insignificant.

As a general trend, with the increasing the capsule diameter, the exergetic efficiency of the integrated system decreases. It is important to note that, during the sensible heat storage period, the exergy efficiency of the storage tank increases independently from the diameter; in contrast, exergy efficiency tends to reduce when the melting becomes the dominant heat transfer mechanism within the capsules. As a general trend, with increasing the diameter of the capsules, the exergy efficiency of the storage tank decreases, like system exergy efficiency.

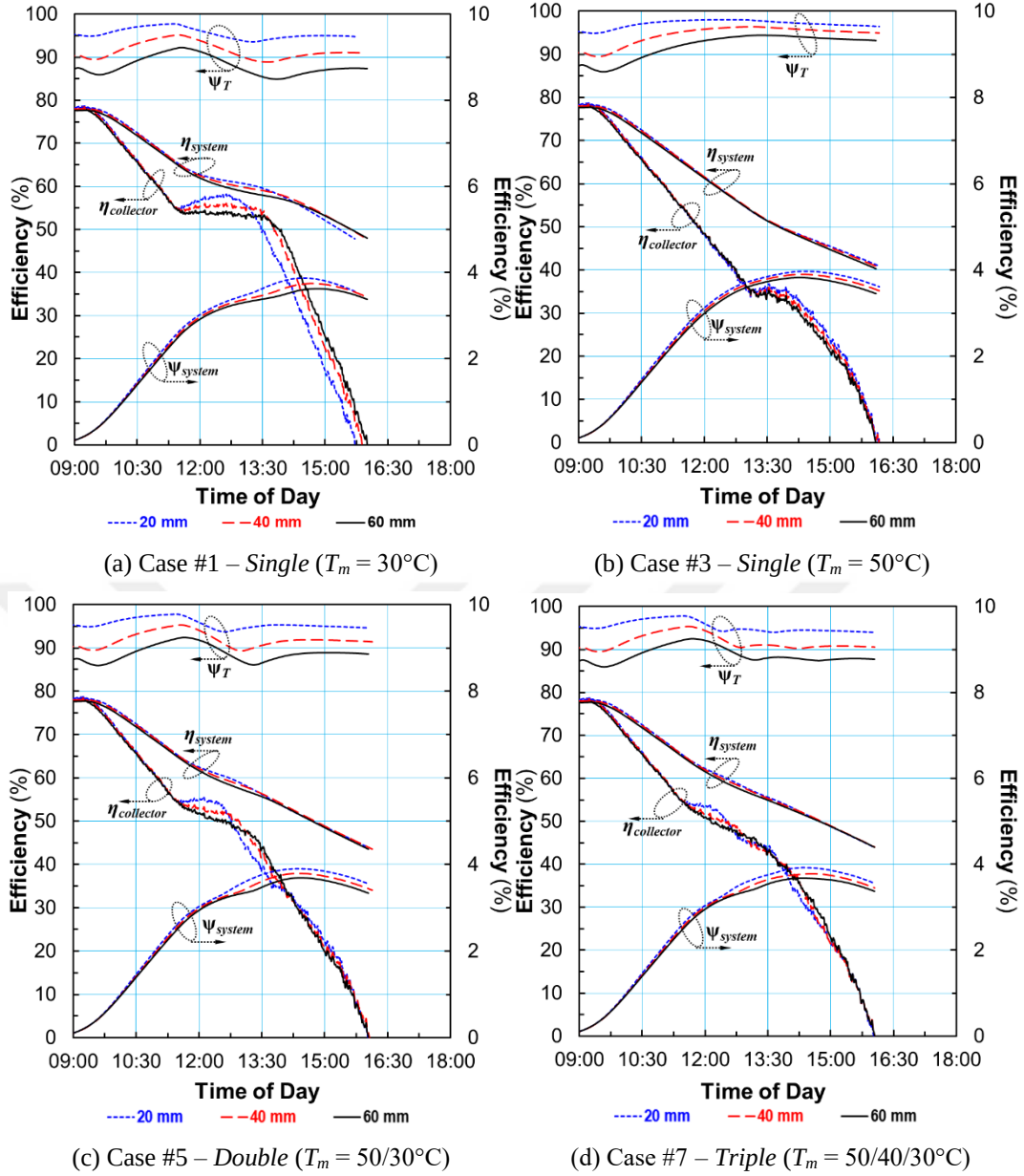


Figure 3.22 The effect of storage material diameter on the exergy efficiency of the storage tank, energy and exergy efficiencies of the integrated system, and collector efficiency – mass flow rate of 0.8 kg/s & January

3.3.4.3 Overall Assessment of Integrated Latent Heat TES System Charging Performance

A summary of the analyses is represented in Figure 3.23 for the heating season months to assess the charging performance of the latent heat TES tank integrated with a flat plate solar collector. For each month, 84 parametric runs are conducted. Here, the effect of the design and working parameters of the latent heat TES system

on charging performance is outlined in terms of the mean liquid fraction, system energy and exergy efficiencies, the mean PCM temperature, and the heat transfer rate. The heat transfer rate is simply defined as the ratio of the total stored energy in the energy storage tank to the charging time.

It is worth noting that increasing the mass flow rate of the HTF enhances all of the system indicators. The effect of mass flow rate becomes apparent as the capsule diameter increases. Additionally, increasing mass flow rate is generally more efficient with high phase change temperature PCMs. More storage performance enhancement is observed in December and January when solar radiation is relatively low. Complete melting occurs for all scenarios for November since the solar radiation falling on the collector is highest this month. At the end of the charging process, the mean PCM temperatures are obtained between 68°C and 78°C for November. On the other hand, this range becomes 46°C to 57°C for December and January. Besides, it is shifted towards 53°C to 64°C for February. The highest exergy efficiency of the integrated energy storage system is obtained in November for Case #3 arrangement. For 1.6 kg/s mass flow rate and 20 mm capsule diameter, the highest exergy efficiency is obtained as 4.10%. Besides, the lowest exergy efficiency is observed in December for the Case #1 PCM arrangement. For 0.2 kg/s mass flow rate and 60 mm capsule diameter, the lowest exergy efficiency is obtained as 2.96%. On the other hand, the highest and lowest system energy efficiencies are 49.47% and 37.74% and are observed in January and February, respectively. It is also interesting to note that the effects of design and working parameters on integrated system energy efficiency become more significant when the incident solar radiation falling on the solar collector is lower. While the difference between the highest and lowest system energy efficiencies is about 5% for November, the gap becomes 10% for December and January. Additionally, the heat transfer rate shows a similar behavior with integrated system energy and exergy efficiencies. It should also be noted that increasing the melting temperature of the PCMs enhances system exergy efficiency and reduces the system energy efficiency.

Since the various design and working parameters are studied, it is crucial to conduct an optimization analysis to suggest the appropriate conditions. The

response surface methodology (Box and Wilson, 1951) is utilized to evaluate the latent heat TES system's optimum design parameters and working conditions for the heating season months. Minitab 18.1 software is used in optimization analyses. The type of PCM arrangements (Case #1 to Case #7), the mass flow rate of the HTF (0.2, 0.4, 0.8, and 1.6 kg/s), and the diameter of the capsules (20, 40, and 60 mm) are selected as independent variables. In order to consider both first and second law aspects, the aim of the optimization is determined as maximizing the integrated system's energy and exergy efficiencies. For each of the selected months, the optimum design and working parameters are summarized in Table 3.9 with the results of the integrated system energy and exergy efficiencies. Case #2 ($T_m = 40^\circ\text{C}$) with a mass flow rate of 1.6 kg/s is found to be the optimum PCM configuration and the working condition throughout the heating season. The optimum capsule diameter is observed as 20 mm, except for November. For November, $D_S = 40$ mm is found as the optimal capsule diameter.

Table 3.9 The optimum design parameters and working conditions for the charging performance of the integrated TES system

Operation month	PCM arrangement	The diameter of the storage material (mm)	The mass flow rate of the HTF (kg/s)	System energy efficiency (%)	System exergy efficiency (%)
November	Case #2	40	1.6	47.00	3.94
December	Case #2	20	1.6	46.30	3.60
January	Case #2	20	1.6	45.81	3.78
February	Case #2	20	1.6	43.64	3.86

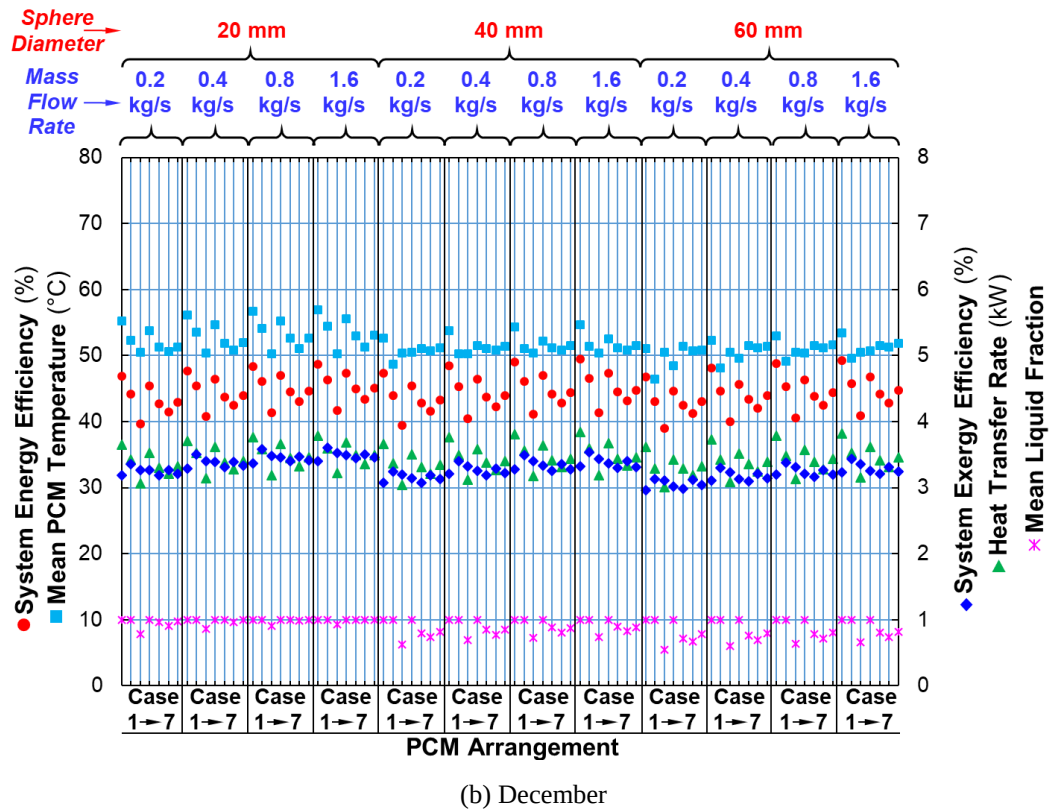
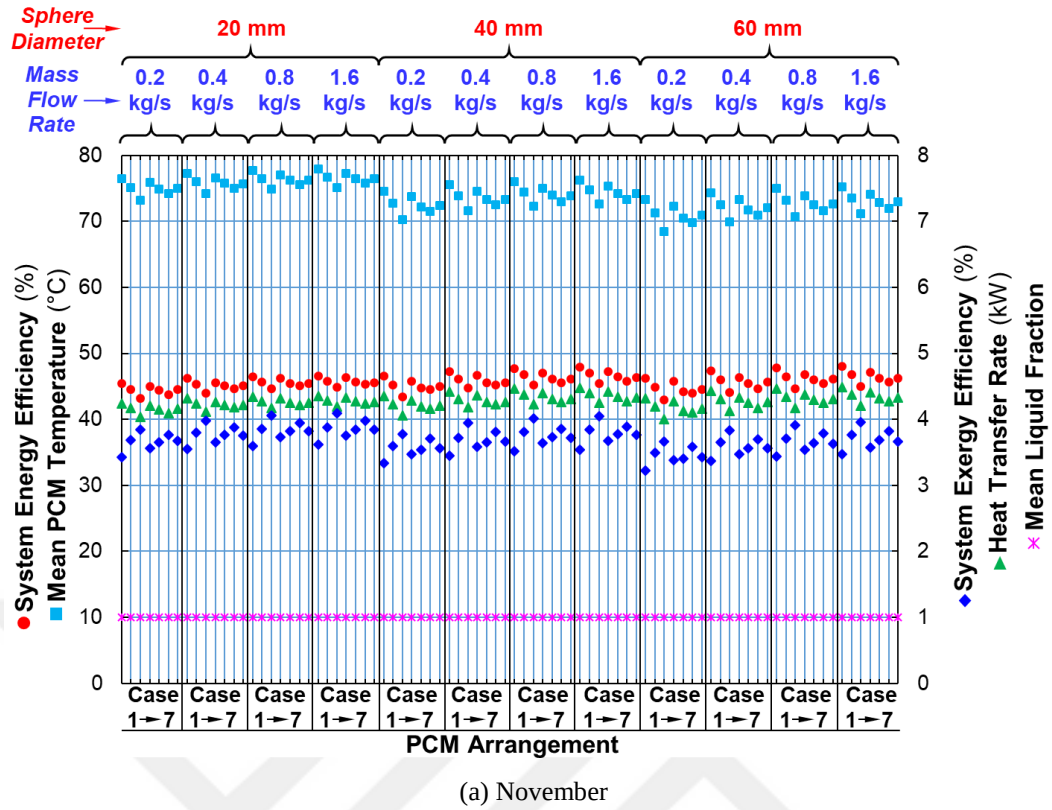


Figure 3.23 An overview of the effects of design and working parameters on charging performance of the integrated storage system

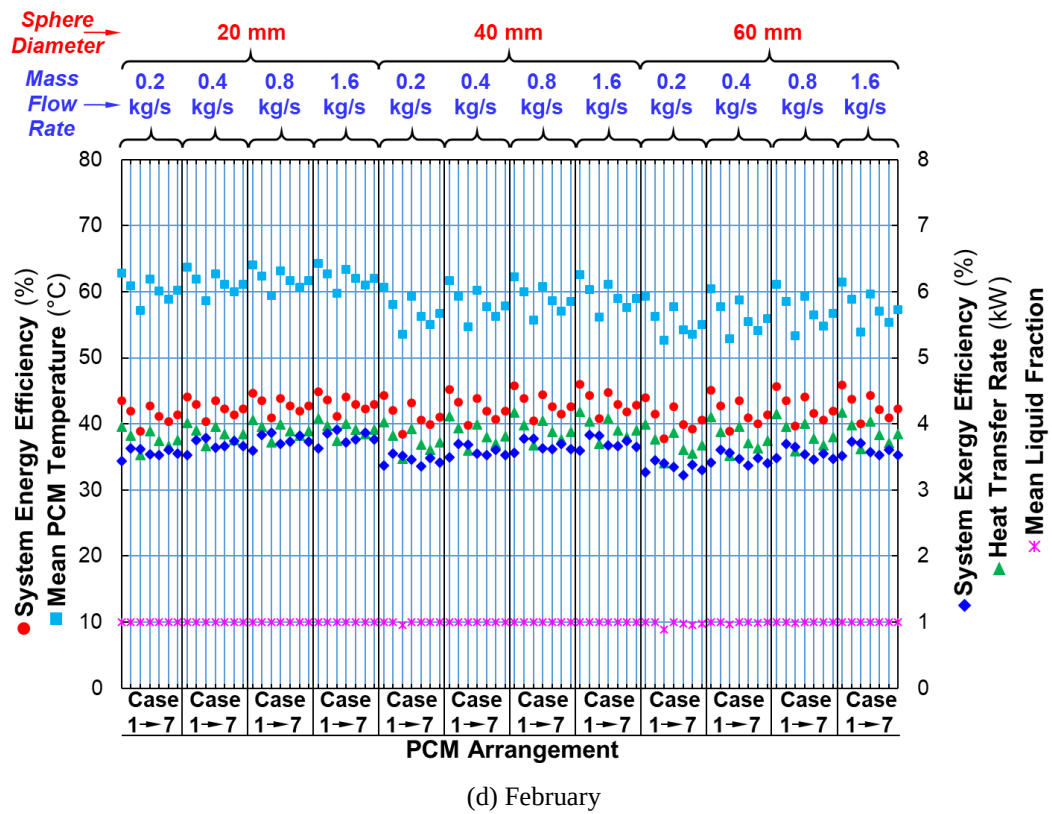
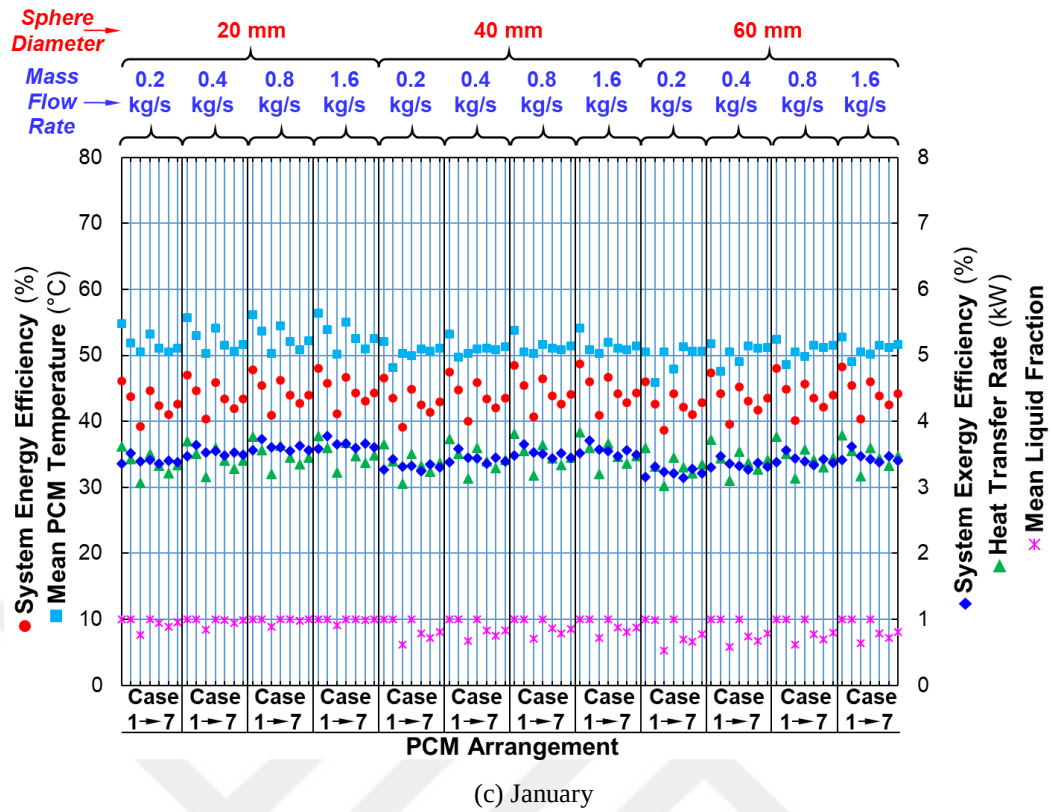


Figure 3.23 continues

3.4 Cyclic Performance of Latent Heat Packed Bed TES System

Like obtaining charging performance of the latent heat TES system in Section 3.3, as design parameters, spherical capsule diameter, and type of PCM arrangements are investigated to evaluate the latent heat TES system's cyclic energetic and exergetic performance. Besides, the effect of the HTF's mass flow rate is also examined as an operating parameter. Capsule diameter and mass flow rate are varied as 20, 40, and 60 mm and 0.2, 0.4, 0.8, and 1.6 kg/s, respectively. Seven type PCM arrangements include cascade (*double/triple-type*) and non-cascade (*single-type*) systems are studied, and the details are represented in Table 3.5. A total of 84 parametric runs is conducted for each heating season month. Before performing the parametric analyses, it was checked that the cyclic steady-state condition has occurred or not so that the initial boundary conditions did not affect the main outputs. Then, the effect of the mass flow rate of the HTF and capsule diameter on time-wise system indicators and the overall system cyclic performance for all heating season months are represented in the subsections. Finally, the optimum design and working conditions have been proposed for an integrated solar-aided heating system consisting of a flat plate solar collector, a latent heat TES tank, and a heat pump.

3.4.1 Preliminary Analysis of Cyclic Latent Heat TES System

Preliminary analysis was carried out to determine how many consecutive days are required for obtaining independent results from the initial boundary conditions. The cyclic analysis proceeded seven consecutive days in the preliminary run to examine the variations between each successive day. As a result, it is found that a 3-day period is adequate to reach a cyclic steady-state condition. Still, the total number of days that the analyses proceed is selected as 4-day to ensure that the cyclic steady-state for each design and working parameters will be achieved.

Further, the variations in the primary outcomes are observed for the lowest and highest mass flow rate of the HTF, storage material diameter, and the melting temperature of the PCM. It is seen that the highest difference between the 3rd day

and 4th has occurred for 0.2 kg/s mass flow rate of the HTF, 20 mm capsule diameter and PCM with 30°C melting temperature for each heating season months. So, the scenario with the listed design and working parameters has been determined as critical. The cyclic performance of the latent heat TES system for this critical scenario is conducted for seven consecutive days.

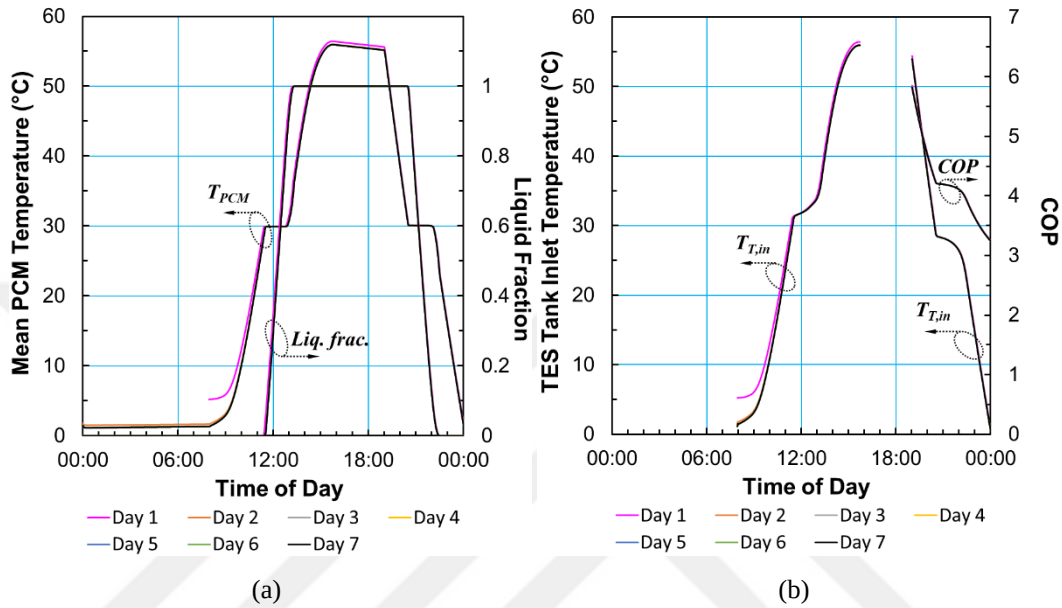


Figure 3.24 The daily time-wise variations of the mean PCM temperature, liquid fraction TES tank inlet temperature, and COP value of the heat pump

The time-wise variation of the mean PCM temperature, liquid fraction, TES tank inlet temperature, and COP value of the heat pump are illustrated in Figure 3.24. It can be seen that the effect of assuming the whole system temperature at the ambient temperature as an initial boundary condition is significantly eliminated at the end of day-two. Beyond the second day, no remarkable variations are observed between each consecutive day in terms of the temperatures, COP, and liquid fraction. Additionally, the effect of the duration of the analyses on the primary outcomes of the integrated TES system, cyclic system energy and exergy efficiencies, mean COP value of the heat pump, and electrical energy (El) consumed for heating needs are presented in Table 3.10 for each heating season months. When the variations in values are examined, it can be concluded that no remarkable difference is observed after the 4th day. In the light of these results, it is determined that performing the

parametric analyses for four consecutive days is sufficient to obtain independent results from the initial boundary conditions.

Table 3.10 The comparison of the variation of the primary outcomes at the end of the cycle days

# of day	November				December			
	COP_{mean}	η_{sys}	ψ_{sys}	El (MJ)	COP_{mean}	η_{sys}	ψ_{sys}	El (MJ)
1	4.999	42.730	7.549	30.088	4.234	51.407	6.875	34.720
2	5.023	43.348	7.750	29.971	4.216	50.911	6.684	34.854
3	5.025	43.367	7.761	29.961	4.214	50.811	6.661	34.866
4	5.025	43.369	7.762	29.960	4.214	50.809	6.659	34.867
5	5.025	43.369	7.762	29.960	4.214	50.808	6.659	34.867
6	5.025	43.369	7.762	29.960	4.214	50.808	6.659	34.867
7	5.025	43.369	7.762	29.960	4.214	50.808	6.659	34.867

# of day	January				February			
	COP_{mean}	η_{sys}	ψ_{sys}	El (MJ)	COP_{mean}	η_{sys}	ψ_{sys}	El (MJ)
1	4.218	49.593	7.353	34.838	4.469	43.731	7.211	33.083
2	4.206	49.494	7.255	34.931	4.471	43.749	7.222	33.073
3	4.205	49.483	7.246	34.939	4.471	43.750	7.223	33.072
4	4.205	49.483	7.245	34.940	4.471	43.750	7.223	33.072
5	4.205	49.483	7.245	34.940	4.471	43.750	7.223	33.072
6	4.205	49.483	7.245	34.940	4.471	43.750	7.223	33.072
7	4.205	49.483	7.245	34.940	4.471	43.750	7.223	33.072

3.4.2 Parametric Cyclic Results of Latent Heat TES System

The dynamic response and the cyclic performance of the TES system are evaluated by monitoring the time-wise variation of the mean PCM temperature and liquid fraction, the inlet temperature of the TES tank, and the COP value of the heat pump. The independent variables are specified by weather conditions, PCM arrangements, the mass flow rate of HTF, and the diameter of the spherical capsules. Time-wise variations of the cycling performance analyses are given only for January to be more concise. In the last part of this section, the comparisons of all alternative designs and working conditions of the system are given for four heating

season months to evaluate the optimum cyclic performance of the latent heat TES system.

3.4.2.1 Effect of Mass Flow Rate on System Cyclic Performance

The mass flow rate of the HTF is varied from 0.2 kg/s to 1.6 kg/s to characterize the performance of the charging process of the storage system. Even though the flow rate is varied for each design alternative and working conditions, the time-wise variations are presented for a specific condition. As a design parameter, the storage material diameter is kept constant as 20 mm, and the coldest month is selected, *i.e.*, January. The daily time-wise variation of the mean PCM temperature and the liquid fraction is illustrated in Figure 3.25 for four different PCM arrangement cases.

As a general trend, decreasing mass flow rate increases the mean temperature of the PCM in the storage mode after the discharging process is over. Additionally, the variation of time-wise mean PCM temperature shows an almost independent behavior from the mass flow rate of the HTF in the sensible heating part of the charging process. For all PCM arrangement cases, with the start of the phase change, higher mean PCM temperatures are observed when the mass flow rate has a lower value. However, as can be seen for Case 1 ($T_m = 30^\circ\text{C}$), when the PCM completely turns into liquid phase, this fact reverses, and the highest mean PCM temperatures occur at the highest mass flow rate. Additionally, in cascaded type PCM arrangement cases, *i.e.*, Case #5 ($T_m = 50/30^\circ\text{C}$) and Case #7 ($T_m = 50/40/30^\circ\text{C}$), complete melting occurs for the mass flow rates of 0.8 and 1.6 kg/s, and the effect of mass flow rate on the mean PCM temperature reverses compared to the above-mentioned arrangement. On the other hand, one should notice that, in Case #3, complete melting could not be obtained, and the maximum liquid fraction occurs as 0.883 for the mass flow rate of 1.6 kg/s. The mean PCM temperature becomes approximately 50.1°C for each of the selected mass flow rates at the end of the charging process.

The storage period begins after the maximum charging is obtained. Around 1% of the stored thermal energy is lost through the ambient due to the heat transfer between the TES tank and the environment during the storage period. After 19:00, the discharging process starts. At the initial period of the discharging process, mean PCM temperature begins gradually decreasing. When the mean PCM temperature reaches the melting temperature of the PCM, during the solidification process, the mean PCM temperature remains nearly constant over a while. After the storage mode starts, PCM, shell, and HTF regions tend to reach a thermal equilibrium state. Although mean PCM temperature slightly reduces at the beginning of the process, a minor increase occurs in advancing times due to the heat gain from the environment.

The mass flow rate of the HTF directly affects the outlet temperature of the flat plate solar collector during the charging process. Therefore, the difference between starting times of the latent heat TES process for lowest and highest mass flow rates becomes 15.5 min for Case #1 and 12.5 min for Case #3. However, in the cascaded types, low melting temperature PCMs are located further in the flow direction in the TES tank. So, the effect of TES tank inlet temperature or solar collector outlet temperature on the beginning of latent heat TES process extensively reduces due to the high heat capacity of the latent heat TES tank. Time differences between the highest and lowest mass flow rates drop as low as 1 min for Case #5 and 4.5 min for Case #7. The working parameter with the lowest mass flow rate initiates an earlier latent heat storage process for single PCM arrangements. The phase change process starts early in the case with the highest mass flow rate for cascaded PCM arrangements. Furthermore, the heat transfer rate increases with increasing the mass flow rate; therefore, the mean liquid fraction reaches its maximum value earlier for all PCM arrangement cases. In the discharging process, the reuse of stored latent heat, *i.e.*, solidification, starts late for the high mass flow rates. However, this behavior is reversed in the ongoing process, and complete solidification firstly occurs for the highest mass flow rate for Case #1 that complete melting occurs. Although the lowest mean liquid fraction takes place for the working condition of the lowest mass flow rate, the reuse of the latent heat process concludes later for all other PCM arrangement cases.

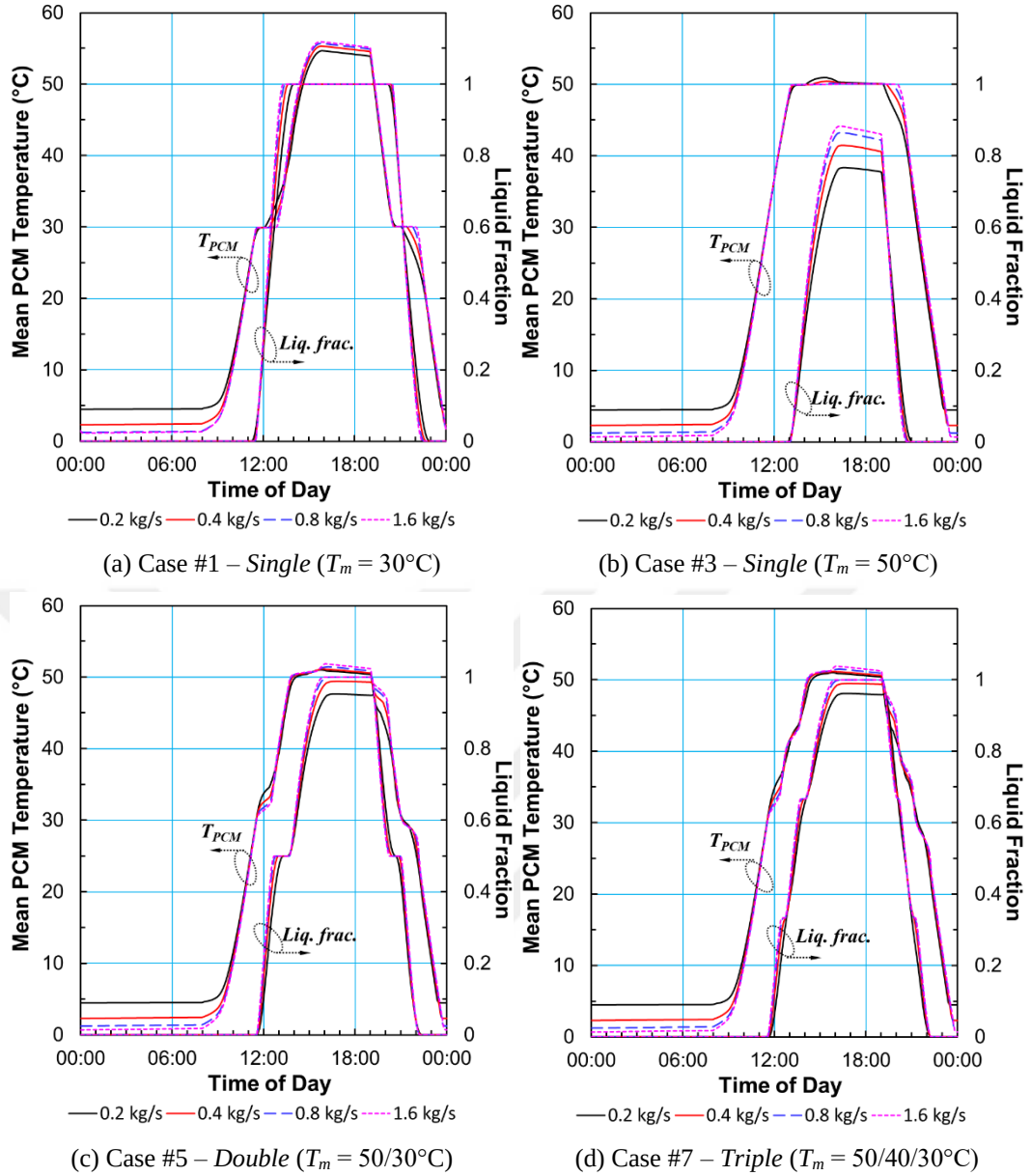


Figure 3.25 The effect of mass flow rate on the mean PCM temperature and liquid fraction – $D_s = 20 \text{ mm}$ & January

Even though the increased mass flow rate of HTF does not seem to affect the mean PCM temperature significantly, the amount of charged and discharged energies are notably affected due to the difference of mean liquid fraction variation. As can be seen in Figure 3.26, the mean liquid fraction at the end of the charging period, charged and discharged energies show an improvement with the increasing mass flow rate of the HTF. For Case #3 ($T_m = 50^\circ\text{C}$), which complete melting does not occur, the charged energy increases by 5.31%, 2.67%, and 1.31% in doubling the mass flow rate from 0.2 to 0.4 kg/s, from 0.4 to 0.8 kg/s and from 0.8 to 1.6

kg/s, respectively. Similarly, discharged energy increases by 5.50%, 2.75%, and 1.33% for similar variations in mass flow rate. It can be said that although complete melting has not been achieved yet, the effect of increasing the mass flow rate of HTF on the system energy performance gradually decreases. Additionally, it is found that about 99.29% of charged energy is discharged.

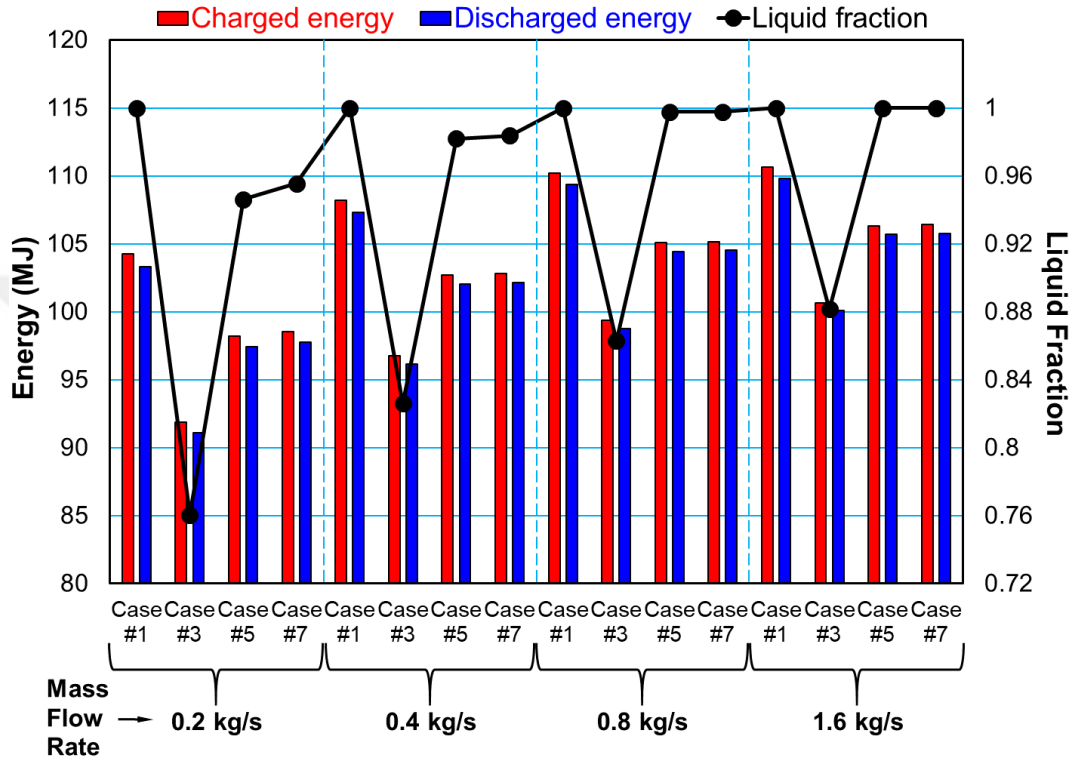


Figure 3.26 The effect of mass flow rate on the liquid fraction, charged and discharged energies – $D_s = 20$ mm & January

The time-wise variation of the TES tank inlet temperature and COP value of the heat pump are illustrated in Figure 3.27 for four different PCM arrangements. At the beginning of the charging process, notice that the inlet temperature of the TES tank starts dramatically increasing for the mass flow rate of 0.2 kg/s for all PCM arrangement cases. However, a substantial variation is observed after a certain time in the tendency, even if a slight decrease occurs at the inlet temperature. At the lowest mass flow rate, the maximum thermal imbalance occurs due to the temperature differences between the top and bottom regions of the storage tank at the end of the discharging process. Additionally, the inlet and outlet sections of the TES tank are positioned upside down during the charging and discharging periods.

As a result, in the early charging process, HTF enters from the top of the surface. The top region of the tank was already warmer than the inlet temperature, so when the HTF enters the storage tank during the charging process reaches the tank outlet, a slight decrease is observed in the inlet temperature of the TES tank. After a while, this effect disappears, and the increasing trend starts again as the solar radiation increases. When the latent TES process, *i.e.*, melting, begins, a slowdown in the increment rate occurs in the inlet temperature of the TES tank like the mean PCM temperature. This change takes place at the lowest mass flow rates at the latest. It is also observed that the increase in the inlet temperature reverses and begins to decrease towards the end of the charging process. The charging process is continued as long as the condition of $T_{T,out} - T_{T,in} < 0$ or $\eta_{col} > 0$ is satisfied. Additionally, overall charging time slightly decreases with the increase of the mass flow rate.

After 19:00, the discharging process is operative. HTF is utilized as the heat source of the heat pump. HTF leaves from the top surface and enters to the bottom surface of the TES tank at the outlet temperature of the evaporator in the discharging process. However, the COP of the heat pump depends only outlet temperature of the TES tank. During the reuse of the stored latent heat thermal energy, an almost horizontal behavior has been observed in the COP due to the fact that the tank outlet temperature remains constant for a certain period of time. The inlet temperature of the TES tank falls below 0°C in the discharging process for all mass flow rates and PCM arrangements except for Case #1 ($T_m = 30^\circ\text{C}$) with a mass flow rate of 1.6 kg/s before 00:00. Thus, the discharging period takes less than it should be, and an auxiliary electrical heater supplies the remaining heating need. It can be said that the COP varies between 5.81 and 3.25 when the heat pump is functioning.

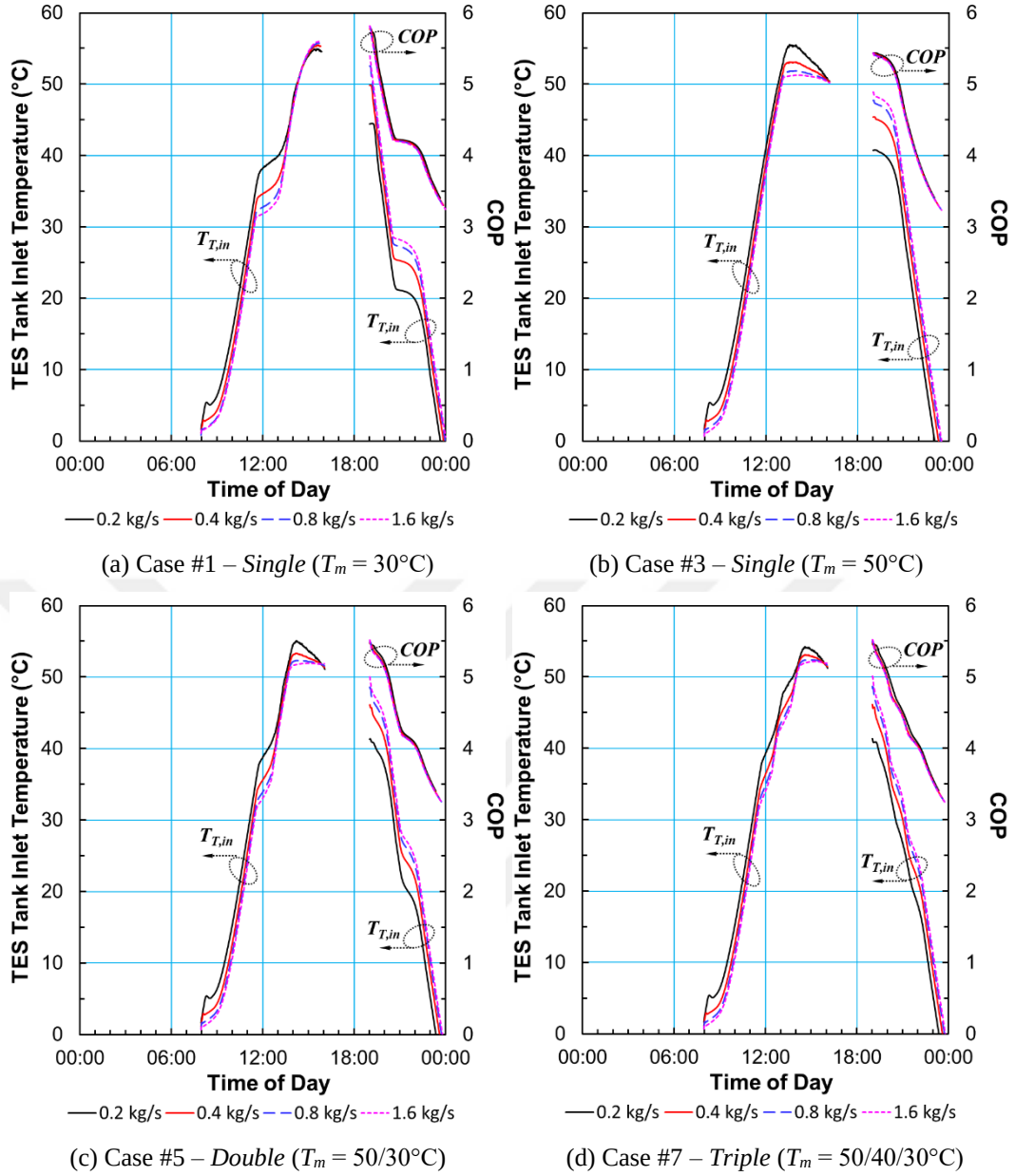


Figure 3.27 The effect of mass flow rate on the inlet temperature of the TES tank and COP of the heat pump – $D_s = 20$ mm & January

3.4.2.2 Effect of Capsule Diameter on System Cyclic Performance

Similarly, like obtaining charging performance of the latent heat TES system, parametric analyses are conducted for three different capsule diameters, *i.e.*, 20 mm, 40 mm, and 60 mm. For each of the selected capsule diameters, the corresponding PCM masses are 242.33 kg, 282.16 kg, and 294.88 kg. In this set of studies, the shell's thickness and the HTF's mass flow rate are kept constant as 1 mm and 0.8 kg/s, respectively. The analyses are conducted for January. Figure 3.28

shows the development of mean PCM temperature and liquid fraction for four different PCM arrangement cases of the cyclic working process of the integrated TES system.

Since January is the coldest month of the Izmir region, the energy required for heating purposes cannot be charged in the storage tank in almost all design and working conditions during the charging process. As a result, the discharging process is not terminated at a time limit of 00:00; the inlet temperature of the TES tank falls below the cut-off temperature, which is 0°C. Consequentially, the temperature distribution in the storage tank becomes lower than the ambient temperature and the starting time of the charging process is the same of all analyses, at 07:56. Additionally, the variation of time-wise mean PCM temperature shows an almost independent behavior from the capsule diameter in the sensible heating part of the charging process, as the mass flow rate of the HTF. Moreover, altering the capsule diameters does not significantly affect the starting time of the phase change process for all PCM arrangements. However, the heat transfer mechanism inside the TES tank occurs faster when the capsule diameter is small and the mean PCM temperatures are higher for the smaller spheres. In addition, the mean liquid fraction for the 20 mm capsule diameter reaches its maximum value 47.5 minutes earlier than the 60 mm capsule diameter for Case #1. On the other hand, the mean liquid fraction between the smallest and highest capsule diameters is found to be 41.0%, 29.9%, and 26.6% for Case #3, Case #5, and Case #7, respectively.

Similarly, with the decreasing capsule diameter, a more efficient discharging process occurs, and the temperature distribution inside the TES tank becomes more homogeneous and relatively low. This behavior is more apparent when the phase change temperature is low; in other words, complete melting and solidification occur. One can also notice that mean PCM temperature remains in the phase change region longer for the higher capsule diameters in the charging and discharging process. On the other hand, it should be noted that there is a 21.68% difference in the amount of PCM in the latent heat TES tank between the smallest and largest capsule diameter.

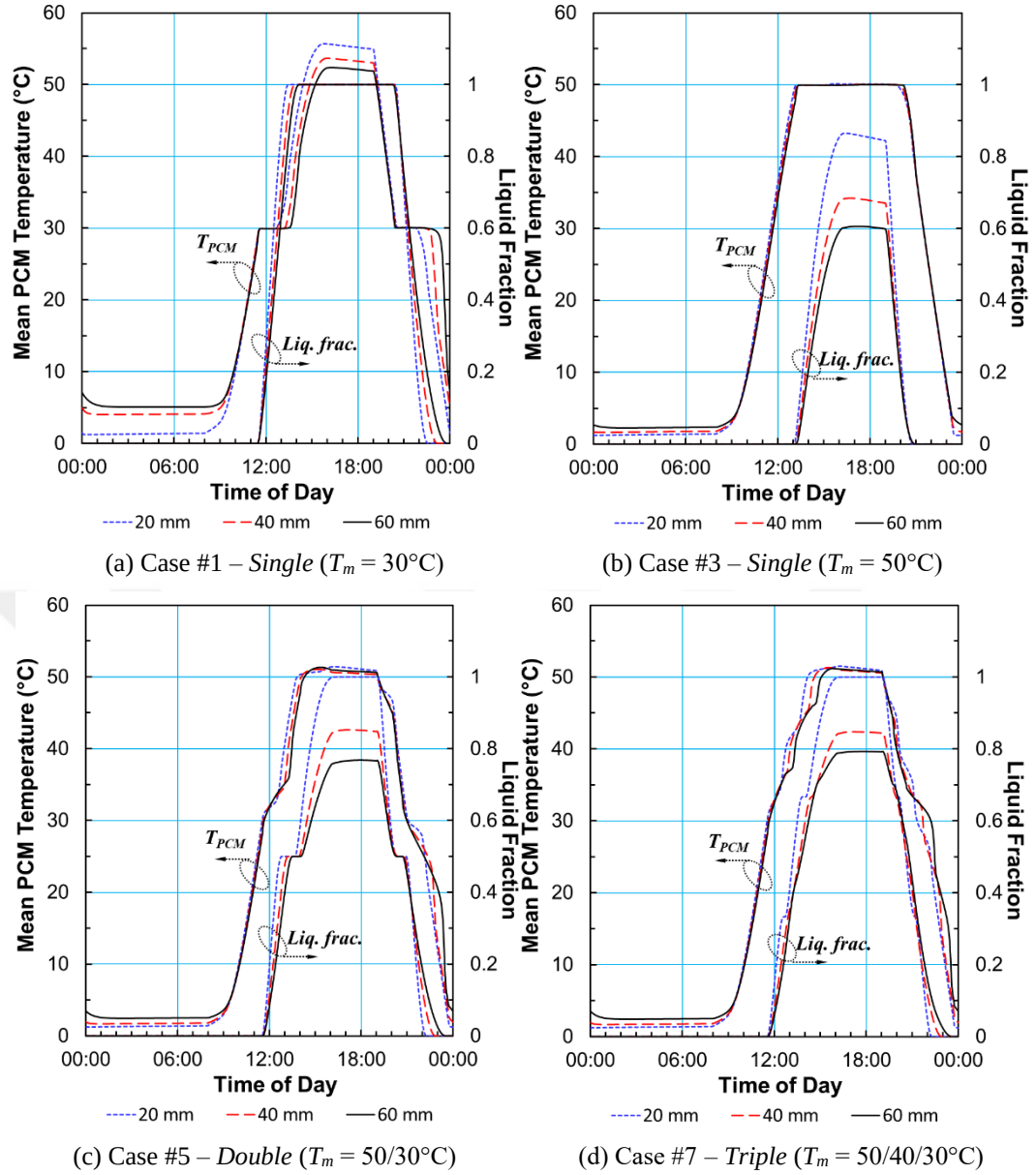


Figure 3.28 The effect of capsule diameter on the mean PCM temperature and liquid fraction – mass flow rate of 0.8 kg/s & January

The effect of capsule diameter on the liquid fraction, charged, and discharged energies are illustrated in Figure 3.29. As a general behavior, with the increasing capsule diameter, charged and discharged energies decrease. Case #3 ($T_m = 50^\circ\text{C}$) shows the worst performance in terms of the mean liquid fraction at the end of the charging period and the charged energy decreases by 1.21%, and 1.99% increasing the capsule diameter from 20 to 40 mm and 40 to 60 mm, respectively. Similarly, discharged energy decreases by 1.25% and 2.04% for similar variations. Additionally, it can be said that in the worst PCM arrangement scenario, the effect

of increasing capsule diameter on the system energy performance gradually increases. However, as an exceptional design condition, there is no remarkable worsening of the system energy performance in increasing capsule diameter from 20 to 40 mm for Case 1 ($T_m = 30^\circ\text{C}$) PCM arrangement.

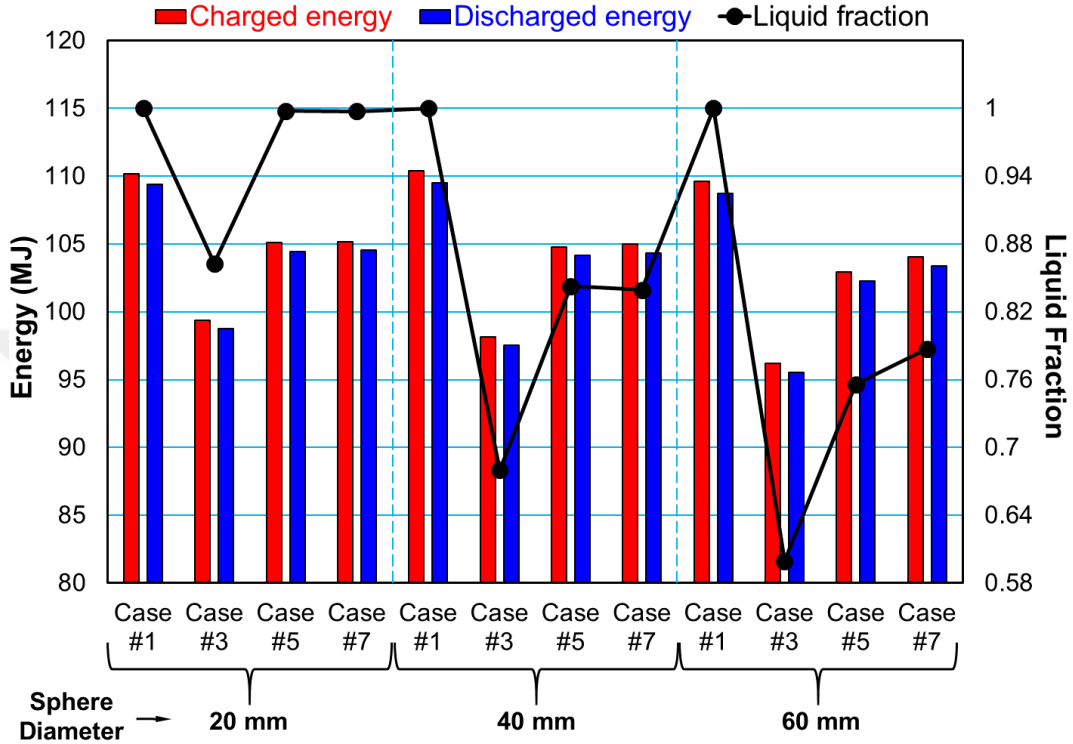


Figure 3.29 The effect of capsule diameter on the liquid fraction, charged and discharged energies – $D_s = 20$ mm & January

The effect of capsule diameter on the time-wise variations in the TES tank inlet temperature and COP of the heat pump is represented in Figure 3.30 for four different PCM arrangements. In these comparisons, the mass flow rate of the HTF is kept constant as 0.8 kg/s, and the analyses are carried out for January, the coldest heating season month.

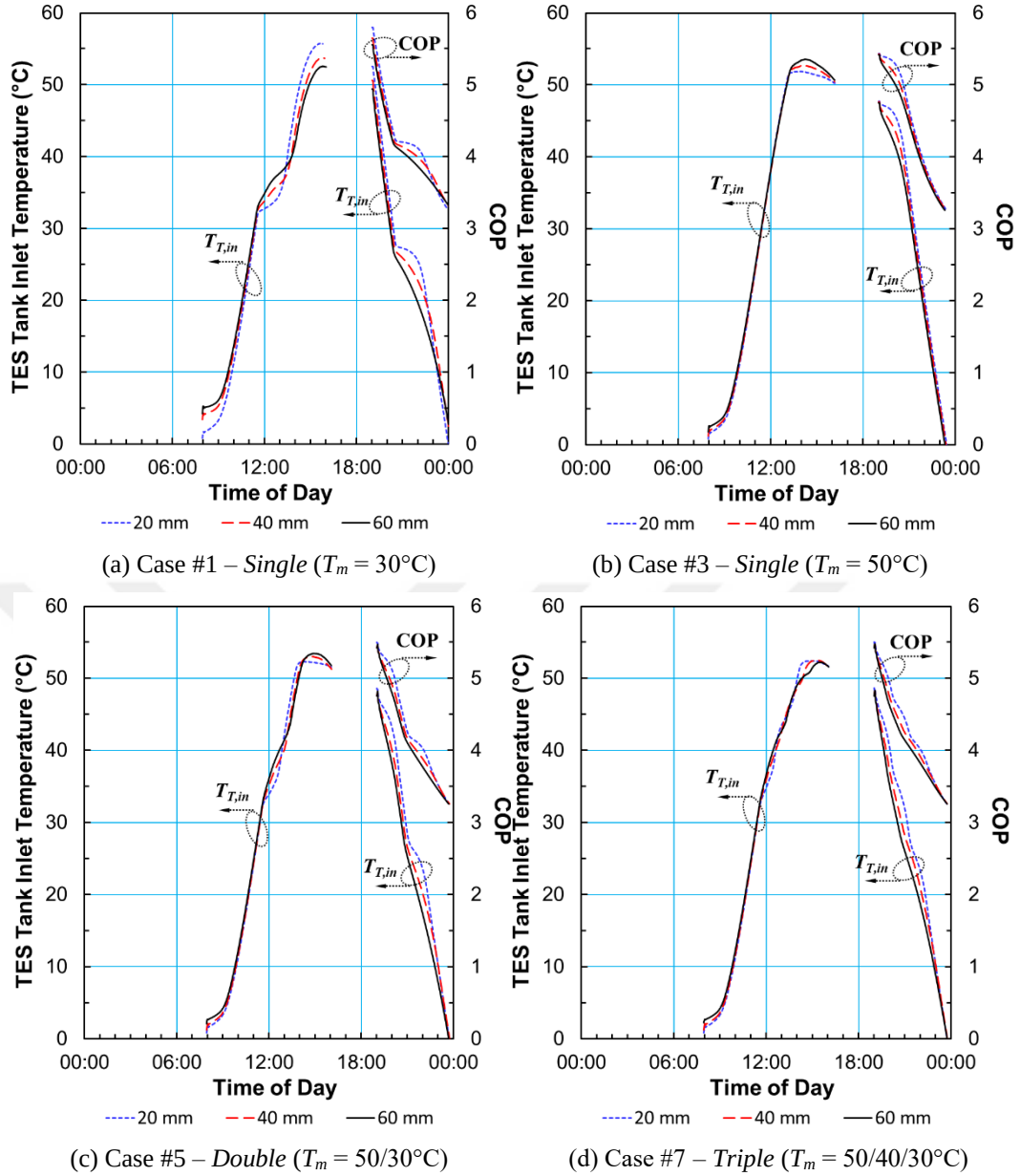


Figure 3.30 The effect of capsule diameter on the inlet temperature of the TES tank and COP of the heat pump – mass flow rate of 0.8 kg/s & January

It is seen that the lowest TES tank inlet temperature occurs for the smallest capsule diameter at the beginning of the charging process; this is because decreasing capsule diameter increases discharging performance. This behavior is most evident in Case #1, which has the lowest phase change temperature and where complete melting and solidification is achieved for all capsule diameters. As phase change rate is higher for smaller capsules, the time-wise variation of the inlet temperature is more affected like the mean PCM temperature at the start of the latent heat process, and the tendency becomes more horizontal for a certain period of time

except for Case #3, which has the highest phase change temperature. While it can be said that the ideal charging process is realized for Case #1 ($T_m = 30^\circ\text{C}$), where complete melting is achieved, the increase in the TES tank inlet temperature towards the end of the charging process not only slowed down but also decreased for a significant period of time for other PCM arrangements.

Like in the charging process, the inlet temperature of the TES tank shows an almost constant time-wise variation for the longest time with the smallest capsule diameter scenario when the phase change process starts in discharging process. The visibility of this tendency decreases as the amount of PCM with a phase change temperature of 30°C decreases in the latent heat TES tank. As a result, the reduction trend in COP also decreases, and the latent heat TES system may have operated more efficiently for a longer period of time. As long as the TES tank inlet temperature is higher than the cut-off temperature ($T_{T,in} > 0^\circ\text{C}$), the heat pump provides the domestic heating need. While almost all of the heating need is supplied by the heat pump for Case #1, it has been observed that the 8 kW electrical heater should be activated as a secondary heater from 15 to 40 minutes for other PCM arrangement cases.

3.4.2.3 Overall Cyclic Performance of Integrated Latent Heat TES System

To obtain the cyclic performance of the numerically modeled TES system, the main outcomes of the conducted parametric runs are represented in Figure 3.31. For each heating season month, the effects of design parameters and working conditions on cyclic energy and exergy efficiencies, mean COP of heat pump, and the electricity fraction are compared for 84 parametric analyses. Here, electricity fraction is defined as the ratio of the electricity consumption of the latent heat TES tank source heat pump to the air-source heat pump. As mentioned before, the discharge period has a cut-off temperature limitation. Therefore, if a sufficient amount of thermal energy could not be stored in the charging process, the discharging process terminates early. As a result, in some cases, the electricity fraction may occur more than 1. In these cases, it means that the proposed TES system has worse performance than the conventional heat pump in meeting the

domestic heating demand. The reason for this situation is when the TES tank inlet temperature becomes lower than the cut-off temperature, and an auxiliary electrical heater meets the heating requirement.

The discharge process terminates at 00:00, which is the end of the heating need only for November for all design parameters and working conditions. As a result of this, the electricity fraction has always been evaluated below 1. The minimum electricity fraction is obtained as 0.671 with 20 mm capsule diameter, mass flow rate of 0.2 kg/s, and Case #3 ($T_m = 50^\circ\text{C}$) PCM arrangement for November. Even though no major effect of the mass flow rate of HTF, capsule diameter, and PCM arrangements on electricity fraction is observed, as a general impression, it is found that electricity fraction decreases with decreasing capsule diameter and mass flow rate when PCM with high phase change temperature is adopted. Similarly, the electric fraction has occurred below 1 for all design parameters and working conditions for February. However, especially when the mass flow rate is 0.2 kg/s and high-temperature PCM is utilized, the discharging process ends early because sufficient thermal energy cannot be stored in the charging process. Consequently, a higher electricity fraction is observed due to the activation of an auxiliary electric heater. In the coldest months, December and January, more efficient results are evaluated when small capsule diameter, high mass flow rate, and low-temperature PCMs are used.

It is worth noting that the cyclic energy efficiency has been observed to oscillate in a small range for November and February when solar radiation is high. The difference between the minimum and maximum cyclic energy efficiency occurs as 2.81% and 6.00% for November and February, respectively. On the other hand, the oscillation becomes approximately 11.6% in December and January, when solar radiation is low. To sum up, the main reason for this oscillation is that adequate thermal energy cannot be stored in the charging period. One should notice that increasing the melting/solidification temperature of the PCM decreases cyclic efficiency for all mass flow rates, capsule diameters, and heating season months. Additionally, increasing the mass flow rate of HTF and decreasing capsule diameter enhances cyclic energy efficiency.

The general behavior of the cyclic exergy efficiency is the same for heating season months. With the increasing mass flow rate and decreasing capsule diameter, cyclic exergy efficiency increases. However, the effect of the PCM arrangement shows a different tendency depending on the operating month. It is seen that the highest exergy efficiency is observed for Case #3 ($T_m = 50^\circ\text{C}$) PCM arrangement for November, the hottest month. On the other hand, in the colder months, the highest exergy efficiency is achieved for Case #2 ($T_m = 40^\circ\text{C}$) since sufficient thermal energy cannot be stored in Case #3 design parameter because of having a high melting temperature. It should also be noted that the highest exergetic performance is obtained in single-type PCM arrangements for all design and working conditions.

Although higher mean COP of the heat pump is obtained for smaller capsule diameter such as cyclic exergy efficiency, unlike cyclic exergy efficiency, with the decreasing mass flow rate of the HTF, mean COP increases. Additionally, since the mean COP is only calculated when the heat pump is operative, independent of the amount of stored thermal energy during the charging process, the mean COP increases with PCM's increasing phase change temperature for all heating season months.

Similar to the determination of charging performance, the most suitable design parameter and working condition of the cyclic latent heat TES system is proposed for each heating season month. The response surface methodology (Box and Wilson, 1951) is conducted as an optimization analysis utilizing Minitab 18.1 software. The type of PCM arrangements (Case #1 to Case #7), the mass flow rate of the HTF (0.2, 0.4, 0.8, and 1.6 kg/s), and the diameter of the capsules (20, 40, and 60 mm) are determined as independent variables. In the optimization analyses, the effects of the first law aspect and the second law aspect are investigated. Cyclic system energy and exergy efficiencies have been determined as the main determinants of the latent heat TES system. The primary goal of the optimization is to maximize these determinants. The optimum design parameters and working conditions are represented in Table 3.11 with the results of the cyclic energy and exergy efficiencies. As a result, a minimum capsule diameter of 20 mm and

maximum mass flow rate of 1.6 kg/s are found as optimum parameters for all heating season months. For PCM arrangements, the most efficient results are obtained for Case #2 ($T_m = 40^\circ\text{C}$) except for January, the coldest month, the better efficiency results are achieved for Case #1 ($T_m = 30^\circ\text{C}$), which has lower phase change temperature, for January.

The comparison of the proposed TES tank-source and air-source heat pumps for the optimum design and working conditions in terms of mean COP and electricity consumption for the heating season months are represented in Table 3.12. Here, the instantaneous COP depends only on the ambient temperature (see Equation 2.50). The air-source heat pump operates continuously, so there is a complete diverse dependency between mean COP and electricity consumption. Additionally, in parallel with the ambient temperature, the lowest mean COP and highest electricity consumption are obtained in January, which is the coldest month, while the highest mean COP and lowest electricity consumption are acquired in November, which is the relatively warmest month for the air-source heat pump. Moreover, there is a 10.92% difference in electricity consumption between the months. However, the energy consumption difference becomes 25.98% between the months for the proposed TES tank source heat pump system. Thus, it can be said that the difference in the meteorological data between the heating season months becomes more apparent and significant for the latent heat TES system. Consequently, thanks to the proposed latent heat TES systems, while a 30.27% improvement in reducing electricity consumption for November, which has the most appropriate meteorological data, electricity consumption decreases 18.26% in December.

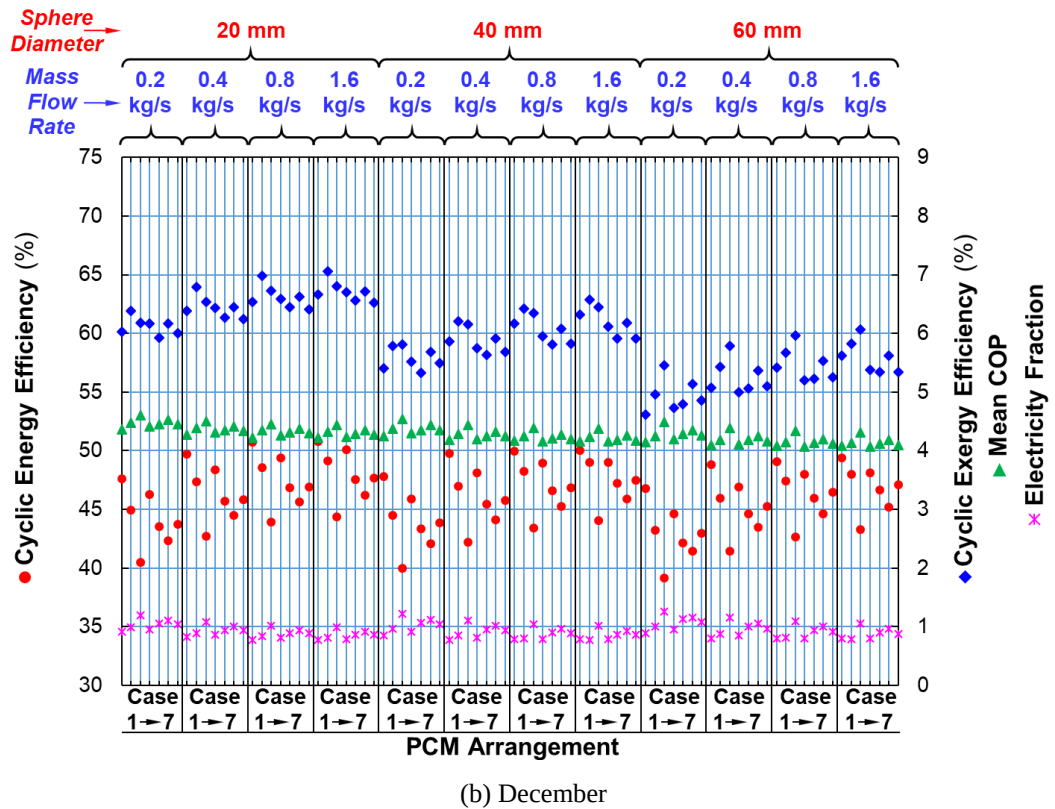
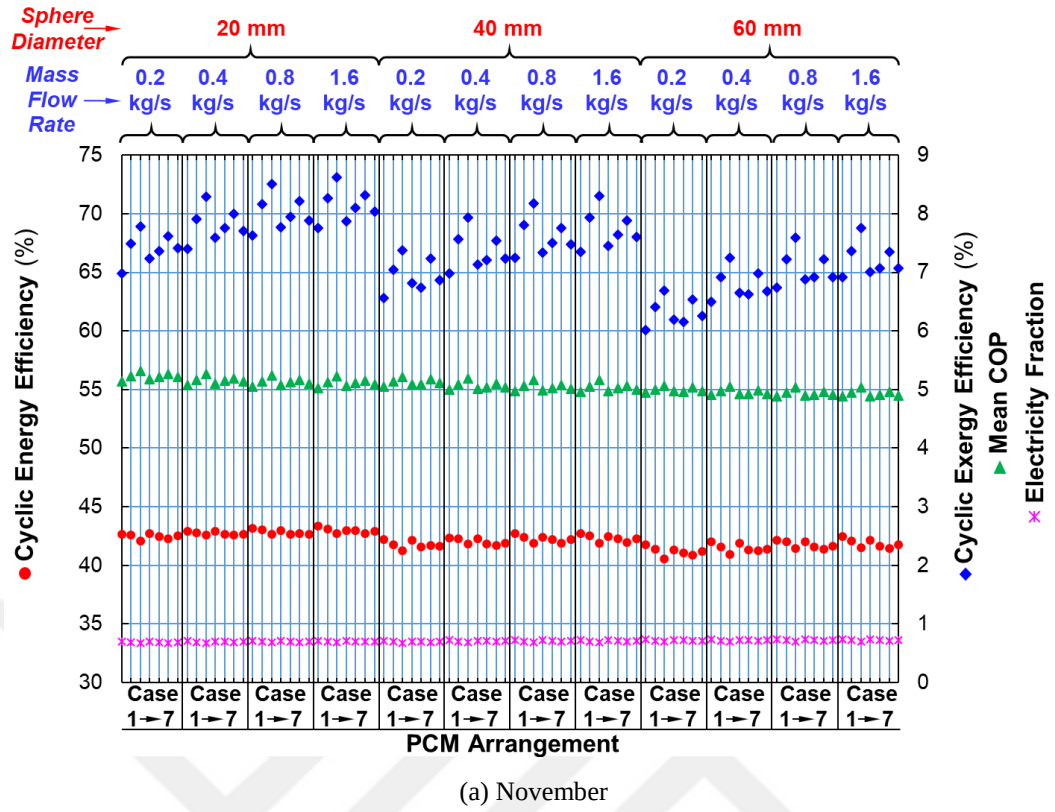


Figure 3.31 An overview of the effects of design and working parameters on the cyclic performance of the integrated latent heat TES system

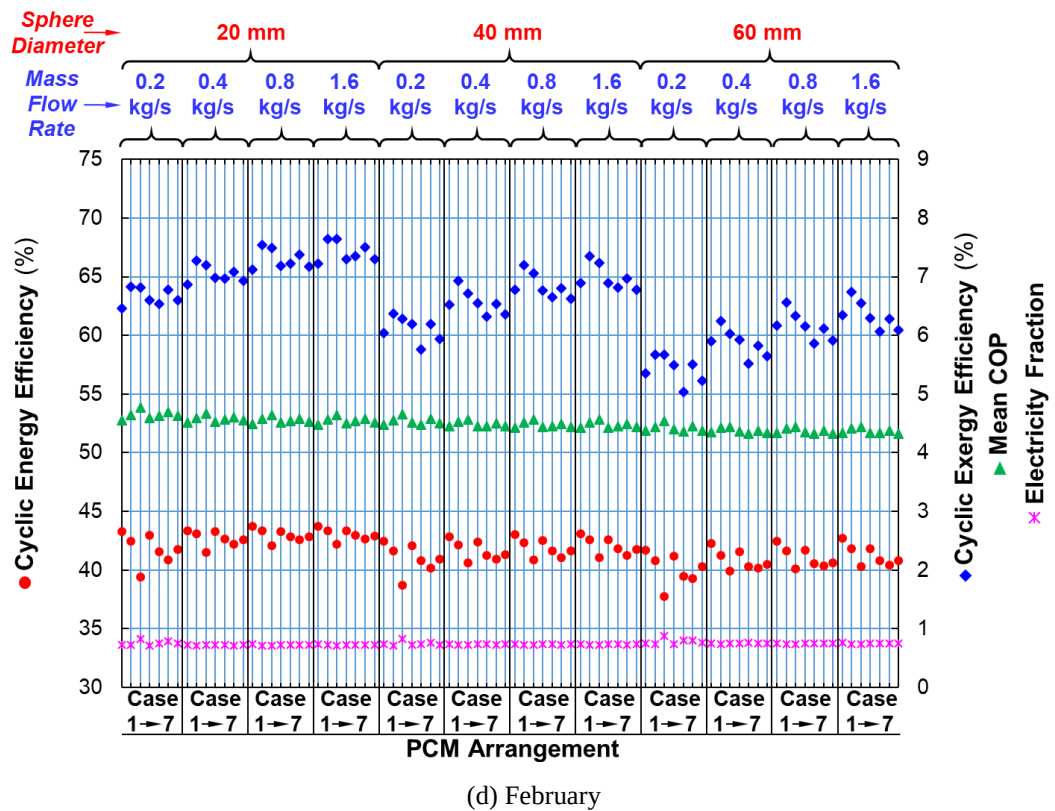
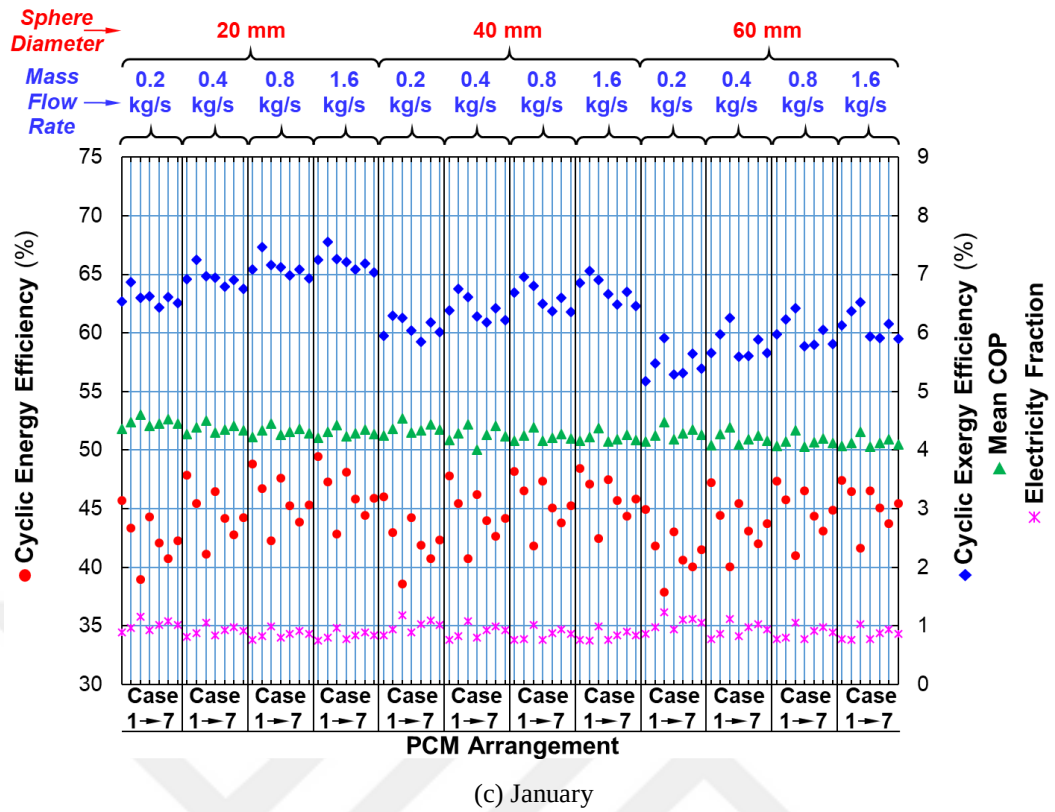


Figure 3.31 continues

Table 3.11 The optimum design parameters and working conditions for the cyclic performance of the integrated TES system

Operation month	PCM arrangement	The diameter of the storage material (mm)	The mass flow rate of the HTF (kg/s)	Cyclic system energy efficiency (%)	Cyclic system exergy efficiency (%)
November	Case #2	20	1.6	43.085	8.263
December	Case #2	20	1.6	49.115	7.062
January	Case #1	20	1.6	49.483	7.245
February	Case #2	20	1.6	43.378	7.647

Table 3.12 The mean COP and electrical consumption of the air source heat pump for heating season months

Operating month	TES tank-source heat pump		Air-source heat pump		Percentage difference	
	Mean COP	Electricity consumption (MJ)	Mean COP	Electricity consumption (MJ)	Mean COP (%)	Electricity consumption (%)
November	5.124	29.118	3.454	41.758	48.35	30.27
December	4.320	36.627	3.219	44.811	34.20	18.26
January	4.207	34.940	3.114	46.317	35.10	24.56
February	4.565	32.364	3.207	44.975	42.34	28.04

CHAPTER FOUR

CONCLUSIONS & RECOMMENDATIONS

In this dissertation, the performance indicators of the sensible and latent heat TES tanks are numerically determined under various working and design parameters. Unlike the related works in the literature, actual meteorological data, *i.e.*, solar radiation, wind speed, and ambient temperature, is implemented in the numerical model. Moreover, an integrated system model consisting of a flat plate solar collector, TES tank, and heat pump is developed rather than considering a standalone storage tank. The heat transfer characteristics and the second law aspect of the TES tank are examined to provide a thorough understanding. Firstly, in evaluating the charging performance of the sensible heat TES tank, the effect of the mass flow rate of HTF, the diameter of the storage material and height of the TES tank on stratification number, energy and exergy efficiencies of the storage tank, and integrated system are obtained for January. After determining the optimum design and working parameters, the performance of the TES tank integrated with a flat plate solar collector is obtained for several other months when the domestic heating needs in Izmir City, Turkey. A total of 39 analyses are conducted to obtain the charging performance of the sensible heat TES system. Additionally, in evaluating the charging and cyclic performances of the latent heat TES tank, the effect of the mass flow rate of HTF, PCM arrangements, and capsule diameter are varied in broad ranges. Heat transfer characteristics and the system's first and second law aspects are evaluated for the heating season months (November, December, January, February). A total of 334 analyses are conducted to determine each of the charging and cyclic characteristics of the latent heat TES system, and the design parameters and working conditions are examined in detail. In addition, the comparison of the proposed latent heat TES tank-source and air-source heat pumps in terms of mean COP and electricity consumption is carried out. Finally, the integrated TES systems' optimum design parameters and working conditions are proposed for the heating season. The following concluding remarks can be drawn,

For the charging performance of the sensible heat TES system:

- Increasing the mass flow rate of the HTF increases the mean temperature of the storage medium and reduces the charging time; that is, the average rate of heat transfer increases. However, the homogeneity of the storage tank decreases with increasing mass flow rate. Additionally, increasing the mass flow rate increases the exergy efficiency of the charging process and the collector efficiency. However, due to the increasing tank outlet temperature so rapidly, collector efficiency tends to reduce after a certain period of the charging process.
- The charging duration and the tank outlet temperature are not significantly affected by the diameter of the storage material. On the other hand, increasing the sphere diameter reduces the exergy efficiency.
- Increasing the tank height increases the collector efficiency, heat transfer rate, and stratification number but reduces the mean temperature. However, the effect of tank height has no significant impact on exergy efficiency. The optimum working and design parameters concerning first and second law aspects for the charging process under actual meteorological data are found for the highest mass flow rate (0.4 kg/s), high tank height (1.6 m), and small sphere diameter (20 mm).
- The tank outlet temperature of the storage tank is significantly affected by the weather data. The highest and lowest outlet temperatures are achieved in November and January, respectively.

For the charging performance of the latent heat TES system:

- Increasing the mass flow rate of the HTF generally enhances all of the system performance indicators; also, charging duration decreases. In addition, the effect of increasing mass flow rate becomes more efficient when solar radiation is low such as in December and January. Increasing mass flow rate from 0.2 to 1.6 kg/s, integrated system energy and exergy efficiencies improves approximately between 4.5-9% and 2.5-5%, respectively.
- Increasing the diameter of the storage material decreases the integrated system's exergy efficiency. However, increasing the diameter from 20 mm

to 60 mm enhances system energy efficiency for November. Additionally, system energy efficiency decreases with the using PCMs having high melting temperature for the other months. Moreover, the heat transfer rate shows similar behavior with system energy efficiency. On the other hand, mean PCM temperature decreases with the increasing diameter of the storage material for the analyses complete melting occurs, for instance, November. Besides, using PCMs having high melting temperatures, the mean PCM temperature increases when complete melting does not occur.

- The effect of single usage and cascaded types (double or triple) of the PCM arrangements on TES system indicators is also investigated. It has been observed that the use of PCMs with a single melting temperature (Case #2) in an application such as solar energy that can generate an energy source in a limited time gives more optimum performance for the studied design parameters and working conditions.

For the cyclic performance of the latent heat TES system:

- With the increasing mass flow rate of the HTF, it is observed that the cyclic energy efficiency increases for all heating season months. In addition, in cases in which complete melting cannot be achieved during the charging process, this improvement is more apparent in cases with high phase change temperatures. However, this increment gradually decreases for high mass flow rates. Moreover, the cyclic exergy efficiency shows similar behavior to the cyclic energy efficiency with increasing mass flow rate. On the other hand, it is seen that there is no general trend in the effect of increasing mass flow rate on cyclic exergy efficiency due to the PCM arrangement. The electricity fraction worsens with the increasing mass flow rate for November, the hottest month. However, it is observed that electricity fraction improves, that is, decreases with the growing mass flow rate when complete melting is not achieved in the charging period.
- The cyclic energy and exergy efficiencies decrease with the increase in the capsule diameter. In cases which complete melting occurs in the charging process, the negative effect of increasing the capsule diameter on the cyclic energy and exergy efficiencies decreases for high capsule diameters, while

in the opposite cases, the worsening in the cyclic energy and exergy efficiencies becomes more apparent with the continuing the capsule diameter increment. Electricity fraction increases, that is, it worsens as the capsule diameter increases. In cases where complete melting is not achieved, an improvement is observed in the electricity fraction when it is increased from 20 mm to 40 mm, while a worsening occurs when it is increased from 40 mm to 60 mm.

- As a general, lower cyclic energy efficiency and higher cyclic exergy efficiency are obtained when PCMs with lower phase change temperature are adopted in the storage tank. In addition, as expected, the highest electricity fraction is obtained in the Case #3 PCM arrangement, in which PCM with the highest phase change temperature is utilized, in January, the coldest month in which the maximum latent heat TES in the storage tank cannot be achieved.

In conclusion, the storage medium temperature of the proposed TES systems roughly varies from 50°C to 77°C. The temperature range looks suitable to adequately fulfill the low-temperature heating demands of a building, such as water heating and underfloor space heating. Additionally, the TES tanks can also assist a district heating system with a heat pump to increase the evaporation temperature. Instead of using an air source heat pump, TES tank integrated heat pump systems will have an improved COP as the storage tank can provide a high quality of thermal energy, *i.e.*, high temperature when it is compared with the ambient conditions. Comparative results reveal that the well-designed TES tank integrated with the heat pump unit could have up to 48% higher COP values and provide 30% less energy consumption when it is compared with the traditional air-source heat pump.

In this dissertation, an economic analysis has not been performed. It is recommended that an economic analysis considers not only equipment and maintenance costs but also environmental effects should be carried out. Additionally, a more comprehensive analysis can be conducted by making analogies with different cities in the world with similar climates through the Thornthwaite climate classification index.

REFERENCES

- Abdulla, A., & Reddy, K. S. (2017). Effect of operating parameters on thermal performance of molten salt packed-bed thermocline thermal energy storage system for concentrating solar power plants. *International Journal of Thermal Sciences*, 121, 30-44.
- Afshan, M. E., Selvakumar, A. S., Velraj, R., & Rajaraman, R. (2020). Effect of aspect ratio and dispersed PCM balls on the charging performance of a latent heat thermal storage unit for solar thermal applications. *Renewable Energy*, 148, 876-888.
- Agrawal, P., Gautam, A., Kunwar, A., Kumar, M., & Chamoli, S. (2018). Performance assessment of heat transfer and friction characteristics of a packed bed heat storage system embedded with internal grooved cylinders. *Solar Energy*, 161, 148-158.
- Ahmed, N., Elfeky, K. E., Lu, L., & Wang, Q. W. (2020). Thermal performance analysis of thermocline combined sensible-latent heat storage system using cascaded-layered PCM designs for medium temperature applications. *Renewable Energy*, 152, 684-697.
- Alptekin, E., & Ezan, M. A. (2020). Performance investigations on a sensible heat thermal energy storage tank with a solar collector under variable climatic conditions. *Applied Thermal Engineering*, 164, 114423.
- Alptekin, E., & Ezan, M. A. (2021). A systematic assessment on a solar collector integrated packed-bed single/multi-layered latent heat thermal energy storage system. *Journal of Energy Storage*, 37, 102410.
- Alva, G., Lin, Y., & Fang, G. (2018). An overview of thermal energy storage systems. *Energy*, 144, 341-378.

- Alva, G., Liu, L., Huang, X., & Fang, G. (2017). Thermal energy storage materials and systems for solar energy applications. *Renewable and Sustainable Energy Reviews*, 68, 693-706.
- Amin, N. A. M., Bruno, F., & Belusko, M. (2014). Effective thermal conductivity for melting in PCM encapsulated in a sphere. *Applied Energy*, 122, 280-287.
- Arena, S., Casti, E., Gasia, J., Cabeza, L. F., & Cau, G. (2018). Numerical analysis of a latent heat thermal energy storage system under partial load operating conditions. *Renewable Energy*, 128, 350-361.
- American Society of Heating, Refrigerating and Air-Conditioning Engineers, (2017). ASHRAE Fundamentals Handbook, *Chapter 31: Physical Properties of Secondary Coolants (Brines)*.
- Badiei, Z., Eslami, M., & Jafarpur, K. (2020). Performance improvements in solar flat plate collectors by integrating with phase change materials and fins: A CFD modeling. *Energy*, 192, 116719.
- Beavers, G. S., Sparrow, E. M., & Rodenz, D. E. (1973). Influence of bed size on the flow characteristics and porosity of randomly packed beds of spheres. *Journal of Applied Mechanics*, 40(3), 655-660.
- Bindra, H., Bueno, P., Morris, J. F., & Shinnar, R. (2013). Thermal analysis and exergy evaluation of packed bed thermal storage systems. *Applied Thermal Engineering*, 52(2), 255-263.
- Bindra, H., Bueno, P., & Morris, J. F. (2014). Sliding flow method for exergetically efficient packed bed thermal storage. *Applied Thermal Engineering*, 64(1-2), 201-208.

- Bishop, E. H., Mack, L. R., & Scanlan, J. A. (1966). Heat transfer by natural convection between concentric spheres. *International Journal of Heat and Mass Transfer*, 9(7), 649-662.
- BP (2021). *Energy Economics, Energy Charting Tool*. Retrieved June 1, 2021, from <https://www.bp.com/en/global/corporate/energyeconomics/energy-charting-tool-desktop.html>.
- Bruch, A., Fourmigué, J. F., & Couturier, R. (2014). Experimental and numerical investigation of a pilot-scale thermal oil packed bed thermal storage system for CSP power plant. *Solar Energy*, 105, 116-125.
- Bruch, A., Molina, S., Esence, T., Fourmigué, J. F., & Couturier, R. (2017). Experimental investigation of cycling behaviour of pilot-scale thermal oil packed-bed thermal storage system. *Renewable Energy*, 103, 277-285.
- Cao, Y., & Faghri, A. (1990). A numerical analysis of phase-change problems including natural convection. *Journal of Heat Transfer*, 112(3), 812-816.
- Cengel, Y., & Ghajar, A. (2015). *Heat and mass transfer: fundamentals and applications* (5th ed.). New York: McGraw-Hill Education.
- Cheng, X., & Zhai, X. (2018). Thermal performance analysis and optimization of a cascaded packed bed cool thermal energy storage unit using multiple phase change materials. *Applied Energy*, 215, 566-576.
- Chiu, C. P., & Chen, W. R. (1996). Transient natural convection heat transfer between concentric and vertically eccentric spheres. *International Journal of Heat and Mass Transfer*, 39(7), 1439-1452.
- Coutier, J. P., & Farber, E. A. (1982). Two applications of a numerical approach of heat transfer process within rock beds. *Solar Energy*, 29(6), 451-462.

- de Gracia, A., & Cabeza, L. F. (2017). Numerical simulation of a PCM packed bed system: a review. *Renewable and Sustainable Energy Reviews*, 69, 1055-1063.
- Dincer I., & Ezan M.A. (2018) *Thermal Energy Storage Methods. In: Heat Storage: A Unique Solution for Energy Systems. Green Energy and Technology* (1st ed.). Switzerland: Springer, Cham.
- Dincer, I., & Rosen, M. A. (2010). *Thermal Energy Storage: Systems and Applications* (2nd ed.). West Sussex: John Wiley & Sons.
- Duffie, J. A., & Beckman, W. A. (2013). *Solar engineering of thermal processes* (4th ed.). New York: Wiley.
- Elfeky, K. E., Ahmed, N., & Wang, Q. (2018). Numerical comparison between single PCM and multi-stage PCM based high temperature thermal energy storage for CSP tower plants. *Applied Thermal Engineering*, 139, 609-622.
- Elfeky, K. E., Mohammed, A. G., & Wang, W. Q. (2021). Performance analysis of an air rock thermocline TES tank for concentrated solar power plants using the coupled DEM–CFD approach. *Clean Technologies and Environmental Policy*, 1-17.
- Elouali, A., Kousksou, T., El Rhafiki, T., Hamdaoui, S., Mahdaoui, M., Allouhi, A., & Zeraouli, Y. (2019). Physical models for packed bed: Sensible heat storage systems. *Journal of Energy Storage*, 23, 69-78.
- Esence, T., Bruch, A., Molina, S., Stutz, B., & Fourmigué, J. F. (2017). A review on experience feedback and numerical modeling of packed-bed thermal energy storage systems. *Solar Energy*, 153, 628-654.
- Ezan, M. A., Ereke, A., & Dincer, I. (2013). Numerical study on solidification process inside annulus in presence of natural convection. *International Journal of Exergy*, 12(4), 423-450.

- Fasquelle, T., Falcoz, Q., Neveu, P., & Hoffmann, J. F. (2018). Numerical simulation of a 50 MWe parabolic trough power plant integrating a thermocline storage tank. *Energy Conversion and Management*, 172, 9-20.
- Fuentes, M. K. (1987). A simplified thermal model for flat-plate photovoltaic arrays (No. SAND-85-0330). *Sandia National Labs.*, Albuquerque, NM (USA).
- Gautam, A., & Saini, R. P. (2020). Thermal and hydraulic characteristics of packed bed solar energy storage system having spheres as packing element with pores. *Journal of Energy Storage*, 30, 101414.
- Geissbühler, L., Mathur, A., Mularczyk, A., & Haselbacher, A. (2019). An assessment of thermocline-control methods for packed-bed thermal-energy storage in CSP plants, Part 2: Assessment strategy and results. *Solar Energy*, 178, 351-364.
- Grirate, H., Agalit, H., Zari, N., Elmchaouri, A., Molina, S., & Couturier, R. (2016). Experimental and numerical investigation of potential filler materials for thermal oil thermocline storage. *Solar Energy*, 131, 260-274.
- Hänchen, M., Brückner, S., & Steinfeld, A. (2011). High-temperature thermal storage using a packed bed of rocks—heat transfer analysis and experimental validation. *Applied Thermal Engineering*, 31(10), 1798-1806.
- Hollands, K. G. T., Unny, T. E., Raithby, G. D., & Konicek, L. (1976). Free convective heat transfer across inclined air layers. *Journal of Heat Transfer*, 98(2), 189-193.
- Incropera, F. P., DeWitt, D. P., Bergman, T. L., & Lavine, A. S. (2006). *Fundamentals of heat and mass transfer* (6th ed.). New Jersey: John Wiley & Sons.

- International Energy Agency (2020a). *World Energy Outlook*. Retrieved June 1, 2021, from <https://www.iea.org/reports/world-energy-outlook-2020/outlook-for-energy-demand>.
- International Energy Agency (2020b), *Heat Pumps*. Retrieved June 1, 2021, from <https://www.iea.org/reports/heat-pumps>.
- International Energy Agency (2021). *Turkey 2021*. Retrieved June 1, 2021, from <https://www.iea.org/reports/turkey-2021>.
- Intergovernmental Panel on Climate Change (2014). *Annex III: Technology-specific cost and performance parameters*. Retrieved June 1, 2021, from https://www.ipcc.ch/site/assets/uploads/2018/02/ipcc_wg3_ar5_annex-iii.pdf.
- Ismail, K. A. R., & Henríquez, J. R. (2000). Solidification of PCM inside a spherical capsule. *Energy Conversion and Management*, 41(2), 173-187.
- Kaygusuz, K. (1995). Performance of solar-assisted heat-pump systems. *Applied Energy*, 51(2), 93-109.
- Khor, J. O., Sze, J. Y., Li, Y., & Romagnoli, A. (2020). Overcharging of a cascaded packed bed thermal energy storage: Effects and solutions. *Renewable and Sustainable Energy Reviews*, 117, 109421.
- Lee, S. L. (1990). A strongly implicit solver for two-dimensional elliptic differential equations. *Numerical Heat Transfer, Part B Fundamentals*, 16(2), 161-178.
- Li, D., Hu, Y., Li, D., & Wang, J. (2019). Combined-cycle gas turbine power plant integration with cascaded latent heat thermal storage for fast dynamic responses. *Energy Conversion and Management*, 183, 1-13.
- Li, G. (2016). Sensible heat thermal storage energy and exergy performance evaluations. *Renewable and Sustainable Energy Reviews*, 53, 897-923.

- Li, M. J., Jin, B., Ma, Z., & Yuan, F. (2018). Experimental and numerical study on the performance of a new high-temperature packed-bed thermal energy storage system with macroencapsulation of molten salt phase change material. *Applied Energy*, 221, 1-15.
- Li, X., Xu, E., Song, S., Wang, X., & Yuan, G. (2017). Dynamic simulation of two-tank indirect thermal energy storage system with molten salt. *Renewable Energy*, 113, 1311-1319.
- Liao, Z., Xu, C., Ren, Y., Gao, F., Ju, X., & Du, X. (2018). A novel effective thermal conductivity correlation of the PCM melting in spherical PCM encapsulation for the packed bed TES system. *Applied Thermal Engineering*, 135, 116-122.
- Liu, M., Riahi, S., Jacob, R., Belusko, M., & Bruno, F. (2020). Design of sensible and latent heat thermal energy storage systems for concentrated solar power plants: Thermal performance analysis. *Renewable Energy*, 151, 1286-1297.
- Lugolole, R., Mawire, A., Lentswe, K. A., Okello, D., & Nyeinga, K. (2018). Thermal performance comparison of three sensible heat thermal energy storage systems during charging cycles. *Sustainable Energy Technologies and Assessments*, 30, 37-51.
- Mao, Q. (2016). Recent developments in geometrical configurations of thermal energy storage for concentrating solar power plant. *Renewable and Sustainable Energy Reviews*, 59, 320-327.
- Mao, Q., & Zhang, Y. (2020). Thermal energy storage performance of a three-PCM cascade tank in a high-temperature packed bed system. *Renewable Energy*, 152, 110-119.
- Marti, J., Geissbühler, L., Becattini, V., Haselbacher, A., & Steinfeld, A. (2018). Constrained multi-objective optimization of thermocline packed-bed thermal-energy storage. *Applied Energy*, 216, 694-708.

- Meier, A., Winkler, C., & Wuillemin, D. (1991). Experiment for modelling high temperature rock bed storage. *Solar Energy Materials*, 24(1-4), 255-264.
- Mertens, N., Alobaid, F., Frigge, L., & Epple, B. (2014). Dynamic simulation of integrated rock-bed thermocline storage for concentrated solar power. *Solar Energy*, 110, 830-842.
- Metin, C., Hacipasaoglu, S. G., Alptekin, E., & Ezan, M. A. (2019). Implementation of enhanced thermal conductivity approach to an LHTES system with in-line spherical capsules. *Energy Storage*, 1(1), e39, 1-13.
- Niedermeier, K., Marocco, L., Flesch, J., Mohan, G., Coventry, J., & Wetzel, T. (2018). Performance of molten sodium vs. molten salts in a packed bed thermal energy storage. *Applied Thermal Engineering*, 141, 368-377.
- Ortega-Fernández, I., & Rodríguez-Aseguinolaza, J. (2019). Thermal energy storage for waste heat recovery in the steelworks: The case study of the REslag project. *Applied Energy*, 237, 708-719.
- Ouagued, M., Khellaf, A., & Loukarfi, L. (2013). Estimation of the temperature, heat gain and heat loss by solar parabolic trough collector under Algerian climate using different thermal oils. *Energy Conversion and management*, 75, 191-201.
- Palacio, M., Rincón, A., & Carmona, M. (2020). Experimental comparative analysis of a flat plate solar collector with and without PCM. *Solar Energy*, 206, 708-721.
- Peiró, G., Gasia, J., Miró, L., Prieto, C., & Cabeza, L. F. (2017). Influence of the heat transfer fluid in a CSP plant molten salts charging process. *Renewable Energy*, 113, 148-158.

- Peng, H., Yang, Y., Li, R., & Ling, X. (2016). Thermodynamic analysis of an improved adiabatic compressed air energy storage system. *Applied Energy*, 183, 1361-1373.
- Petela, R. (1964). Exergy of Heat Radiation. *ASME. J. Heat Transfer*, 86(2), 187–192.
- Rosen, M. A. (2001). The exergy of stratified thermal energy storages. *Solar Energy*, 71(3), 173-185.
- Royo, P., Acevedo, L., Ferreira, V. J., García-Armingol, T., López-Sabirón, A. M., & Ferreira, G. (2019). High-temperature PCM-based thermal energy storage for industrial furnaces installed in energy-intensive industries. *Energy*, 173, 1030-1040.
- Rubitherm Technologies GmbH (2021). *PCM RT-LINE*. Retrieved June 1, 2021, from <https://www.rubitherm.eu/en/index.php/productcategory/organische-pcm-rt>.
- Scanlan, J. A., Bishop, E. H., & Powe, R. E. (1970). Natural convection heat transfer between concentric spheres. *International Journal of Heat and Mass Transfer*, 13(12), 1857-1872.
- Sciacovelli, A., Li, Y., Chen, H., Wu, Y., Wang, J., Garvey, S., & Ding, Y. (2017). Dynamic simulation of Adiabatic Compressed Air Energy Storage (A-CAES) plant with integrated thermal storage—Link between components performance and plant performance. *Applied Energy*, 185, 16-28.
- Singh, H., Saini, R. P., & Saini, J. S. (2010). A review on packed bed solar energy storage systems. *Renewable and Sustainable Energy Reviews*, 14(3), 1059-1069.
- Tan, F. L. (2008). Constrained and unconstrained melting inside a sphere. *International Communications in Heat and Mass Transfer*, 35(4), 466-475.

- Ulgen, K. (2006). Optimum tilt angle for solar collectors. *Energy Sources, Part A: Recovery, Utilization, and Environmental Effects*, 28(13), 1171-1180.
- Verma, P., & Singal, S. K. (2008). Review of mathematical modeling on latent heat thermal energy storage systems using phase-change material. *Renewable and Sustainable Energy Reviews*, 12(4), 999-1031.
- Wang, P., Zhao, P., Xu, W., Wang, J., & Dai, Y. (2019). Performance analysis of a combined heat and compressed air energy storage system with packed bed unit and electrical heater. *Applied Thermal Engineering*, 162, 114321.
- Watmuff, J. H., Charters, W. W. S., & Proctor, D. (1977). Solar and wind induced external coefficients for solar collectors. *Cooperation Mediterraneenne pour l'Energie Solaire*, 56.
- White, M. T., & Sayma, A. I. (2020). A new method to identify the optimal temperature of latent-heat thermal-energy storage systems for power generation from waste heat. *International Journal of Heat and Mass Transfer*, 149, 119111
- Wu, M., Xu, C., & He, Y. (2016). Cyclic behaviors of the molten-salt packed-bed thermal storage system filled with cascaded phase change material capsules. *Applied Thermal Engineering*, 93, 1061-1073.
- Yan, Y., Wang, K., Clough, P. T., & Anthony, E. J. (2020). Developments in calcium/chemical looping and metal oxide redox cycles for high-temperature thermochemical energy storage: A review. *Fuel Processing Technology*, 199, 106280.
- Yang, B., Bai, F., Wang, Y., & Wang, Z. (2019). Study on standby process of an air-based solid packed bed for flexible high-temperature heat storage: Experimental results and modelling. *Applied Energy*, 238, 135-146.

Yang, L., Zhang, X., & Xu, G. (2014). Thermal performance of a solar storage packed bed using spherical capsules filled with PCM having different melting points. *Energy and Buildings*, 68, 639-646.

Yang, X., Yu, J., Xiao, T., Hu, Z., & He, Y. L. (2020). Design and operating evaluation of a finned shell-and-tube thermal energy storage unit filled with metal foam. *Applied Energy*, 261, 114385.

Zanganeh, G., Pedretti, A., Haselbacher, A., & Steinfeld, A. (2015). Design of packed bed thermal energy storage systems for high-temperature industrial process heat. *Applied Energy*, 137, 812-822.

Zhao, B. C., Cheng, M. S., Liu, C., & Dai, Z. M. (2018). System-level performance optimization of molten-salt packed-bed thermal energy storage for concentrating solar power. *Applied Energy*, 226, 225-239.

APPENDICES

APPENDIX–1: Nomenclature

A	area (m^2)
Bi	Biot number
C	volumetric heat capacity ($\text{J}/\text{m}^3\text{K}$)
c	specific heat capacity(J/kgK)
D	diameter (m)
E	east neighbor of P
$\dot{E}$	rate of energy (W)
Ex	stored exergy (J)
$\dot{E}_x$	rate of exergy (W)
El	consumed electric energy (MJ)
F_R	collector heat removal factor (-)
F'	collector efficiency factor (-)
g	gravitational acceleration (m/s^2)
H	volumetric enthalpy (J/m^3) or height (m)
h	convective heat transfer coefficient ($\text{W}/\text{m}^2\text{K}$)
h_{sf}	latent heat of fusion (J/kg)
I_{solar}	incident solar radiation (W/m^2)
k	thermal conductivity (W/mK)
m	mass (kg)
$\dot{m}$	mass flow rate of the HTF (kg/s)
N	number (-)
Nu	Nusselt number (-)
P	control volume
$\dot{Q}$	heat rate (W)
R	thermal resistance ($\text{m}^2\text{K}/\text{W}$)
Ra	Rayleigh number (-)
r	radius or radial coordinate (mm)
S	source term (J/m^3)

$\dot{S}$	rate of entropy (W/K)
T	temperature (°C or K)
T^*	apparent sun temperature (K)
$\bar{T}$	mean temperature (°C)
t	time (s)
Str	stratification number (-)
U	overall heat transfer coefficient (W/m ² K)
u	velocity (m/s)
V	volume (m ³)
W	width (m) or work (kJ)
y	cartesian coordinate (m)

Greek letters

α	absorptivity of glass (-) or thermal diffusivity (m ² /s)
β	thermal expansion coefficient (1/K)
Δy	distance between spheres along the flow direction (m)
δ	thickness (m) or annular gap of the liquid PCM (m)
δT_m	half of the mushy range temperature (°C)
ε	void fraction or emissivity (-)
η	energy efficiency (-)
μ	dynamic viscosity (Pas)
ν	kinematic viscosity (m ² /s)
ρ	density (kg/m ³)
σ	Stefan-Boltzmann constant (W/m ² K ⁴)
τ	transmissivity of glass (-)
ϕ	collector tilt angle (°)
ψ	exergy efficiency (-)

Subscripts

a	air gap
amb	ambient
aux	auxiliary
b	bottom

<i>c</i>	glass cover
<i>col</i>	collector
<i>comp</i>	compressor
<i>conv</i>	convective
<i>dest</i>	destruction
<i>dis</i>	discharging
<i>e</i>	interface
<i>enh</i>	enhanced
<i>eq</i>	equivalent
<i>evap</i>	evap
<i>i</i>	inner
<i>in</i>	inlet
<i>init</i>	initial
<i>ins</i>	insulation
<i>L</i>	solar collector
<i>l</i>	liquid
<i>m</i>	melting or mushy
<i>max</i>	maximum
<i>o</i>	outer
<i>out</i>	outlet
<i>p</i>	absorber plate
<i>rad</i>	radiation
<i>S</i>	storage material
<i>s</i>	solid
<i>sys</i>	system
<i>T</i>	storage tank
<i>t</i>	top
<i>tot</i>	total
<i>tube</i>	collector tube
<i>w</i>	wind

Superscripts

⁰	previous time step
--------------	--------------------

Abbreviations

COP	coefficient of performance
CSP	concentrating solar power
EG	ethylene glycol
HP	heat pump
HTF	heat transfer fluid
IEA	International Energy Agency
IPCC	Intergovernmental Panel on Climate Change
SIS	strongly implicit solver
PCM	phase change material
PVC	polyvinyl chloride
SDS	sustainable development scenario
STEPS	stated policies scenario
TES	thermal energy storage
TTM	temperature transformation method