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Experimental Study of Heat Losses in Thermal Energy Storage System

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Abstract: Thermal energy storage (TES) is an important method to store the energy in the form of heat by passing hot heat transfer fluid into the storage domain, where single or double storage tank is usually used to store the heat energy. Thermal energy storage systems could help in integrating the energy sources (solar and wind energy) of intermittent nature and thus providing continuous energy based on the demand. In the present study, a lab scale storage vessel is used with 8" diameter and 23.63" height. Air is used as working fluid with inlet temperature of 95℃ and gravel is used as storage materials. A numerical model was built using finite element method to solve governing equations. COMSOL Multiphysics software is used to solve the problem numerically and validate the experimental results. The focus of the present study is on the analysis of heat losses to the ambient and their influence on the thermal performance and storage efficiency. The amount of heat loss rate is calculated which a significant parameter to quantify the system performance. Therefore, the main studied parameters are temperature, energy and efficiency of the storage system. The amount of energy stored into the bed is low due to the heat losses based on the experimental and numerical results. Charging efficiency is 43.3% at 0.0004 kg/s and 55% at 0.006 kg/s. The heat loss rate per unit area increases with the increase the time, leading to a reduction in thermal performance

Keywords: Thermal energy storage; heat losses; storage efficiency

1. Introduction

Demand on energy is increasing worldwide. Fossil fuels are the largest supplier for the needed energy which leads to significant CO_2 emissions [1]. Renewable energy sources could reduce the usage of traditional energy sources (coal, oil and natural gas) and thus reduce the impact of carbon dioxide emissions and global warming [2]. However, renewable energy sources are not efficient during the cloudy days and night due to the intermittent nature [3]. Therefore, energy storage systems can play a significant role in providing continuous energy for homes, industry and other sectors. Several methods have been used to store the energy such as mechanical, magnetic, biological, chemical and heat [4]. Concentrated solar thermal applications (CST) have several applications such as electricity production, desalination, and enhanced oil recovery. The thermal energy from CST plants can be used to generate electricity, which is known as concentrated solar power (CSP). Thermal energy storage in packed bed systems can be used to store the energy when the production is high for later use [5], leading to continuous energy supply. TES is the way of storing the energy in the form of heat either in sensible heat [6], where the materials that have high thermal conductivity and high volumetric heat capacity are the best candidate or latent heat [7] by using phase change materials that have high energy density. A combination of sensible and latent heats is considered a promising technique [8, 9].

Sensible heat method raises the temperature of the storage media such as rocks [10], alpha-alumina [11], concrete [12], recycled materials (ceramics) [13] molten salt [14], etc [15], leading to store the energy in the solid media.

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Latent heat storage occurs by changing the phase of storage media (phase change materials PCMs) from solid to liquid [16]. Some studies proposed combining sensible/latent heats to improve the performance [17, 18], where the thermal performance is improved by combining solid materials with PCMs. The cost, availability, range of temperature, and thermophysical properties are the most important parameters that should be considered when select a storage material [19, 20]. Packed beds (single tank or two tanks) are usually used in storing the heat energy either in sensible heat form (filler materials) or in latent heat form (PCMs), where heat transfer fluid (HTF) heats and cools the bed domain during the storage and recovery processes. Air [21, 22], molten salt [23, 24], oil [24], and supercritical carbon dioxide [25] are the most common heat transfer fluids that have been used. Two-tank was widely used where the first tank for cold fluid and second tank for hot fluid. However, it is expensive [26]. Single-tank is cheaper and could provide more flexibility in installation in vertical or horizontal orientation. The temperature gradient inside the single-tank storage is known as thermocline, where half of the bed domain at high temperature and the second half at low temperature [13, 27]. Designing of the storage vessel is another important component that should be considered. Cylindrical packed bed is the most used storage tank (vertical layout [28] and horizontal layout [10, 29]), both layout support thermocline in the bed domain. Zanganeh et al. [30] used conical storage vessel to analyze the thermocline performance. The results showed that energy storage in conical storage is higher than that in cylindrical vessel. In addition, heat losses to the surrounding from the inlet cover is high because of the big diameter at the top of the vessel. Another design which is rectangular cross section vertical tank (412.75 mm square x 1219.2 mm height) was studied too [31] using five packing arrangements. The results showed that energy stored is improved due to short packing. Axial flow is the common flow in the mentioned storage vessel designs, where the flow passes from top to bottom (vertical layout) or right to left (horizontal layout).

Recently, it was found that dividing the bed domain into several segments could improve the thermal front and thus thermal efficiency [31-35]. Heat losses and thermal axial dispersion were reduced by dividing the length/height of the storage domain into two/three segments [32] showed that heat losses. In addition, segmentation with axial flow in large reservoirs is considered [36, 37], leading to a reduction in conductive losses and the exergy losses. Recently, some studies proposed a new flow scheme which is radial flow, aiming to improve energy stored; reduce pressure drop losses, thermal axial dispersion and heat losses [11, 21, 38-41]. These studies showed that the energy stored in the storage domain increases. However, pressure drop still an issue that influence the efficiency of the storage system. The thermal performance of the packed bed systems is affected by several parameters (design of the packed bed, type of HTF, storage materials, pressure drop, dispersion, and heat losses). Several research studies investigated these parameters. The impact of heat losses on thermal performance of the storage systems is significant which is considered in the present study.

Heat losses are an important component that should be mitigated. In packed bed systems, heat losses happen by two modes, conductive losses through the bed wall (stainless/carbon steel) and insulation and by convective losses from the wall/insulation surface to surrounding due to the temperature difference [42]. Big surface to volume ratio increases heat losses. The small surface to volume ratio (storage tanks with large diameters) reduces heat losses [42]. However, the cost of designing packed beds with big diameters is high. In addition, beds with large diameters exhibit radial temperature gradient from the bed center toward the bed wall [43-45] which should be considered. The thickness, and type of insulation could reduce the heat losses as discussed in [46] where thermal losses decrease with the increase of insulation thickness. Another significant parameter that influences heat losses is the location of insulation (internal or external insulation). Cascetta et al. [45] recommended internal insulation to reduce the radial temperature gradient and wall effects and thus improve thermal performance and efficiency. Radial temperature gradient in case of external insulation is higher than that of internal insulation by approximately 3 times as was investigated in a high temperature storage system [13]. However, designing an internal insulation is expensive especially for storage tanks with commercial scales.

In the present study, the thermal behavior in a lab-scale storage packed bed system is investigated, focusing on heat losses from the wall and its effects on storage efficiency. A storage tank with 8" diameter and 80 cm height is used, the surface to volume ratio is 20.5. Air is used as HTF with inlet temperature of 95°C. A commercial gravel is used as storage material with size of 9 mm. External insulation; fiberglass is used to prevent heat losses from the bed wall. A flow rate of 0.0004 kg/s is used in the experiment. Heat losses to the ambient have been investigated, but with limited details as explained above. To the best of the author' knowledge, a few studies analyze heat losses. The present study analyzes and quantifies the amount of heat loss rate to the ambient. Future work will consider high flow rates and full charge/discharge cycle. Specific objective is to analyze the thermal losses from the bed domain and its impact on thermal performance and storage efficiency. A CFD model in COMSOL Multiphysics is built to validate the experimental results. The numerical results agree with the experimental results. A second flow rate of 0.006 kg/s is considered in the modeling efforts. The results show that charging efficiency and energy storage increase with increase of flow rate. The heat losses decrease with the increase of flow rate. Future work will consider thermal performance during recovery period, high temperature, pressure drop effects, different storage media and heat transfer fluids.

2. Methodology

2.1 Research Methodology

The flowchart in Fig. 1 represents the methodology used in the study. The main methodology of the present study is investigating experimentally the impact of heat losses using air-gravel TES packed bed system. Air and gravel were selected as working fluid and storage material respectively based on their thermophysical properties and availability. A CFD model was built in COMSOL Multiphysics and run for a few cases. The experimental results match the model results. Heat losses to the ambient, energy storage and charging efficiency were analyzed and discussed.

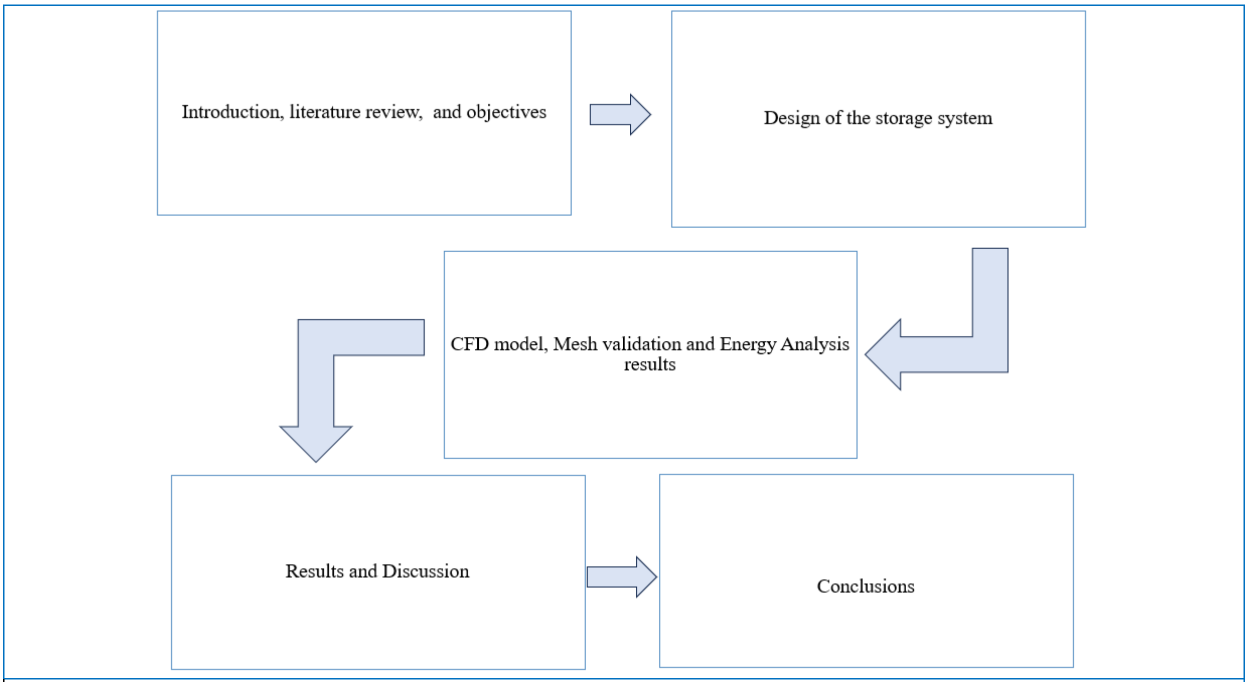


Fig. 1 Flowchart showing the general methodology of the study.

2.2 Experimental Set-up

Fig. 2 shows the apparatus used in the present work which was designed and built in the heat transfer laboratory (College of Engineering-University of Misan). The storage packed bed is a vertical cylindrical vessel made of stainless steel 304 which is characterized by its high durability and resistance to high and low temperatures. The outer diameter is 8 inch (20.32cm) and the inner diameter is 7.68 inch (19.52cm) (8″ NPS schedule 10 with 0.157″wall thickness). The height of the cylindrical tank is 80 cm.

The height of storage packed bed domain is 60 cm. Tori-spherical flanges are used as connections at the inlet/outlet of the storage tank. The flow inlet at the top side and the flow outlet at the bottom side, the inlet and outlet of the tank is ½ inch (1.27 cm) FNPT (Female National Pipe Thread). Two screens (distributors) were used to get uniform flow through the vessel. The axial location of the screens is $x = 10\ cm\ and\ 70\ cm$ (one at the inlet side and the second one at the outlet side). The second advantage of the screens is to hold the storage materials prevent them from falling in the pipes at the bottom side. At the surface of the cylindrical tank seven holes were drilled of ¼ inch (0.635 cm) FNPT along the axial length to insert a K-type thermocouples. Those thermocouples help to get the temperature variation inside the storage domain. Seven thermocouples were used starting at $x = 0\ cm\ to\ x = 60\ cm$. The sensing points are distributed axially with even spaces (10 cm, TI1− TI7). The radial location of the seven sensing points is $r = 0\ cm$. Two sensors are used to get the temperature at the inlet and outlet of the storage tank. The thermocouples have errors of ± 2.2°C as specified by the producer. The nine thermocouples are connected to a data acquisition system to record the temperature data points which is then exported to excel sheets for analysis. To get indication about the heat losses from the surface, Digital Thermal Camera was used to get the surface temperature at the insulation surface. During the charging process, hot air is entered the storage packed bed from the top of the vessel. After depositing the hot energy in the gravel materials, the air leaves as cold air from the bottom of the vessel and is then vented to outside of the lab room. Open gate valves are installed on the direction of the hot air to control the flow and let the air goes in the right direction during the charging process. Cold air is provided using available commercial blower. On flow is used in the experimental work which is 0.0004 kg/s (0.7 standard cubic feet per minute, SCFM). In the numerical model another flow rate was tested which is 0.006 kg/s (10 SCFM). To heat up the air coming from the blower and thus provides the required heat energy, an electric heater is used. The maximum outlet temperature that the heater can provide is approximately 750°C. However, the inlet temperature used in the current study is 95℃ due to some limitations in the lab. Temperature controller is used to control the heater outlet temperature and thus provides the desired temperature. In order to reduce the heat losses to the environment, external insulation was used which is fiberglass with thermal conductivity of 0.03 (W/m. K). The thickness of the insulation is 1 cm. Gravel was used as storage material with one size (9 mm) which can be noticed in Fig. 2. The thermophysical properties of the storage tank, insulation, air and storage materials are shown in Table 1. The flow in axial direction is the traditional flow in single packed bed where the hot HTF passes from top to bottom during the storage process and from bottom to top during the recovery process.

Table 1. Thermophysical properties of the storage tank, storage materials and insulation [47-49]

Property	Value	Unit
Stainless steel 304		
Density	7930	kg/m^3
Thermal conductivity	16.2	W/m K
Heat capacity at constant pressure	500	J/kg K
Gravel		
Density	1460-1920	kg/m^3
Thermal conductivity	0.4	W/m K
Heat capacity at constant pressure	840	J/kg K
Insulation		
Density	20	kg/m^3
Thermal conductivity	0.03	W/m K
Specific heat capacity at constant pressure	840	J/kg K

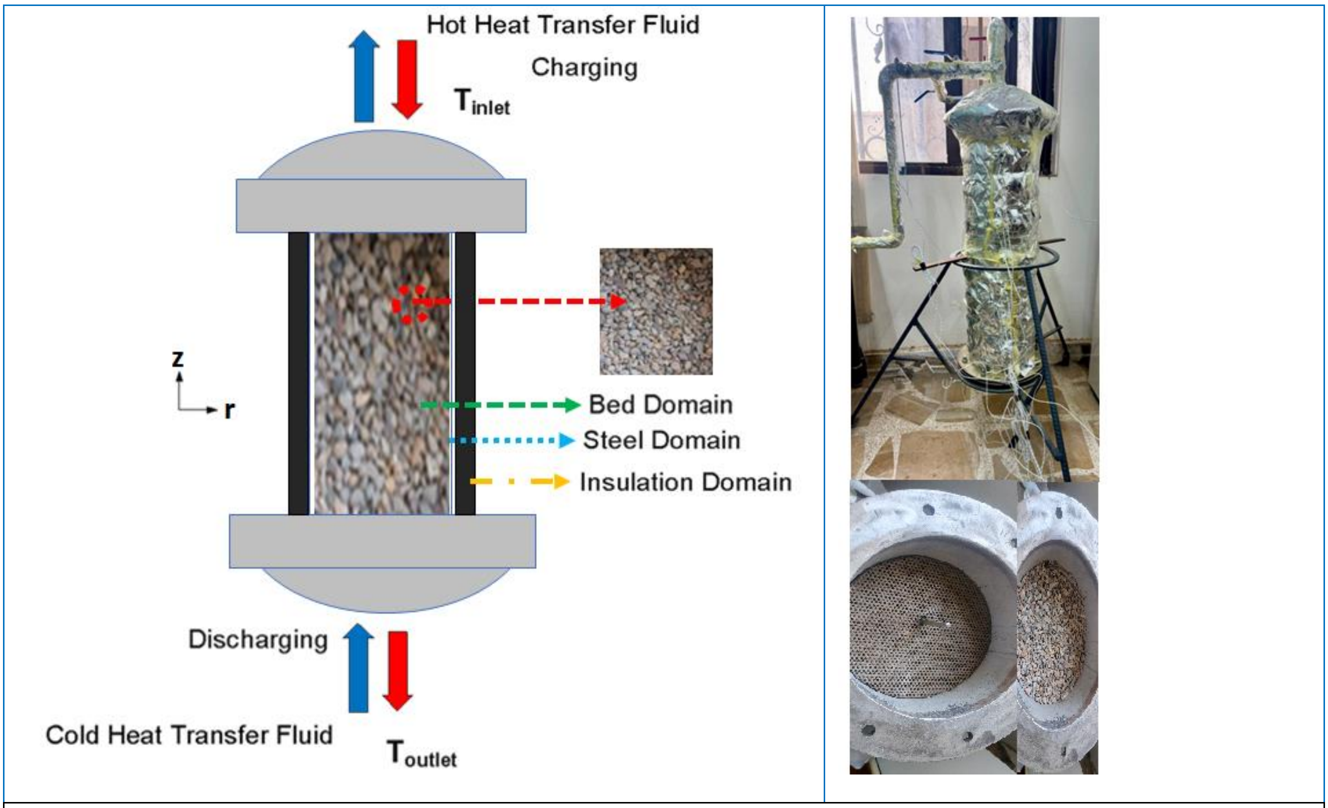


Fig. 2 Left: A schematic of the storage system showing the three domains of the storage tank (bed, steel, and insulation) with inlet and outlet directions. Right: An image of the storage tank installed in the lab.

2.3 Physical Model, Governing Equations and Boundary Conditions

In the numerical model, heat transfer in heat transfer in porous media physics is selected, where two phases are available (solid and fluid). Local thermal equilibrium between solid and fluid phases is assumed in the current model [50]. COMSOL Multiphysics 6.1 version is used to solve the governing equation, energy. The single phase model (the general governing energy equation) was derived by Nield and Bejan [51]. The single phase equation, Eqn. 1 assumes a homogenous medium in the storage tank [50] and uses effective (effe) volume-averaged properties.

$$(\rho C_p)_{effe}\frac{\partial T}{\partial t} + (\rho C_p)_f V \cdot \nabla T = \nabla \cdot (k_{effe}\nabla T) - Q_{loss}, \quad (1)$$

with

$$(\rho C_p)_{effe} = \varepsilon \rho_f C_{p,f} + (1-\varepsilon)\rho_s C_{p,s} \quad (2)$$

and

$$k_{effe} = \varepsilon k_f + (1-\varepsilon)k_s \quad (3)$$

Where Q_{loss} represents heat losses to the surrounding. The equation for the insulation and steel domains is:

$$(\rho C_p)_j\frac{\partial T}{\partial t} = \nabla \cdot (k_j \nabla T) \quad (4)$$

The numerical model calculates the transient temperature profile in all domains (radial and axial directions). 2-D axisymmetric modeling was used; Properties of the storage materials stainless steel, insulation and heat transfer fluid are inserted directly in the model. The time step is the default option of COMSOL solver which is automatic.

Boundary conditions:
The following assumption are considered in the present study:

1- Fluid flow in the porous media is two-dimensional.
2- The study is time-dependent.
3- Constant thermophysical properties.
4- The porous medium is homogenous, rigid, isotropic, and fully saturated with fluid.
5- The local thermal equilibrium is considered between the solid phase and the fluid phase inside the porous media.
6- The influence of radiation heat transfer is neglected
7- The radial temperature distribution in the storage bed domain is very small, so buoyancy effects were neglected.

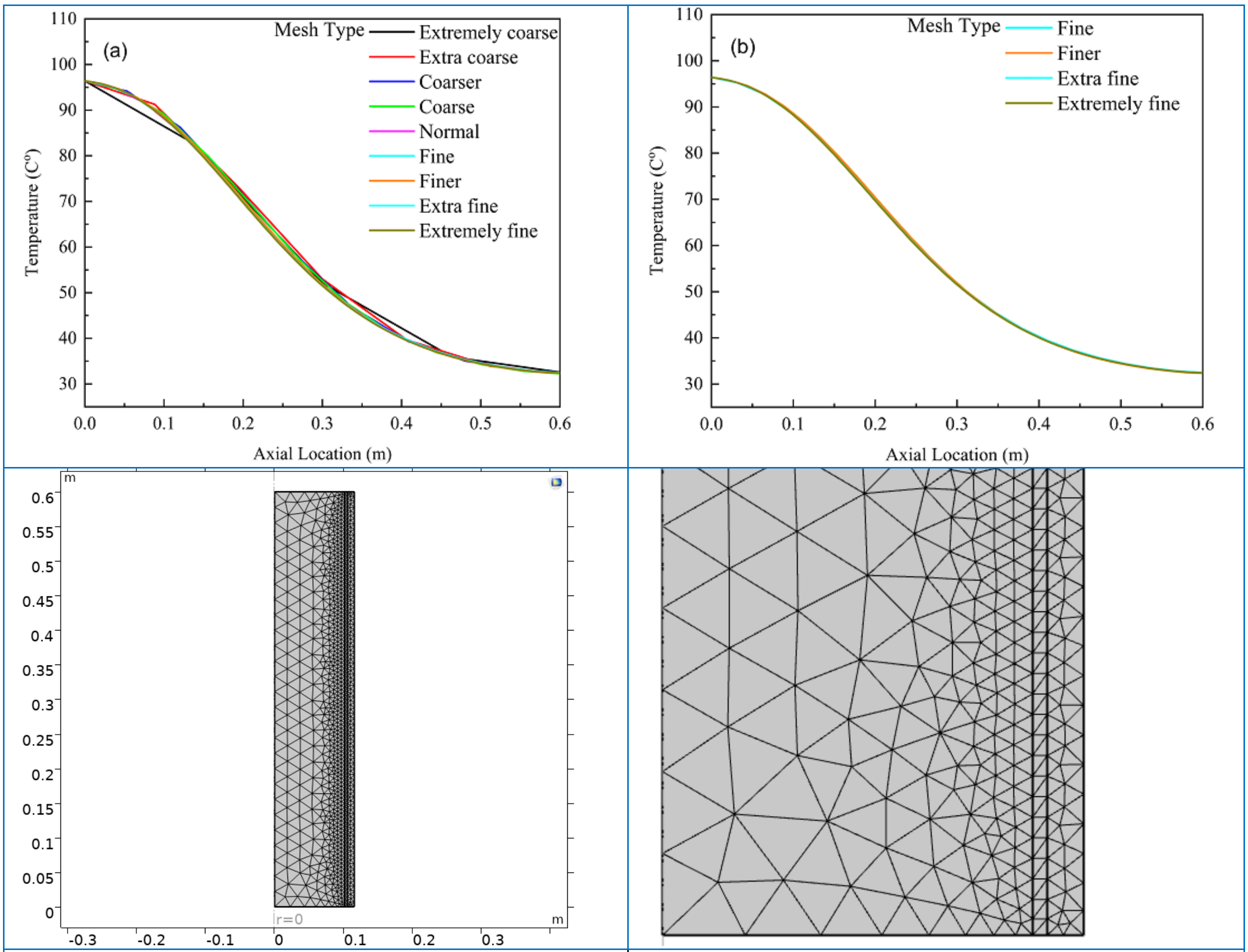
The below boundary conditions were used in the current model based on the experimental inlet/outlet conditions:
$T = T_{inlet}\ at\ x = 0$
$p = 0\ Pa\ (gauge)\ at\ outlet$
Convection heat transfer coefficient at the wall (surface of the insulation), is 3.76 (W/m^2 K) based on wall temperatures and heat transfer correlations [52]. Table 2 shows the key parameters used in the modeling efforts.

Table 2. Parameters used in the numerical model.

Sizing		
Parameter	**Value**	**Unit**
Inlet temperature-charging	95	ºC
Cut off temperature- charging	30	ºC
Charging time- low flow rate	4.5	hour
Charging time- high flow rate	40	minutes
Porosity	0.4	-
Media size	9	mm

2.4 Mesh Validation

Finer mesh (free Triangular) is in the modelling efforts. To check the accuracy of the numerical results, nine different mesh were checked to solve the numerical model as can be observed in Fig a-d, which show the temperature distributions at steady state for low flow rate, 0.0004 kg/s inside the bed domain. ($at\ z = 0 - 0.6\ and\ r = 0\ m$). Fi a show all meshes, the change in the temperature is small. However, in the first five meshes (extremely coarse, extra coarse, coarser, coarse and normal), the differences in temperature can be observed. Fi b shows the temperature profiles at four meshes (fine, finer, extra fine, and extremely fine) which are considered more accurate and showed an excellent convergence. As can be noticed, the change in temperature is approximately zero. The average bed temperature from the four mesh sizes in Fi b is $49.81℃ \mp 0.287$ at steady state with $z = 0.3\ m$. Finer mesh is selected in the present study to get accurate results as can be seen in Fi c-d.



c- The storage vessel showing the mesh | **d- Zoomed in finer mesh**

Fig. 3 (a) The impact of mesh size on temperature profiles in the storage tank domain at nine meshes. (b) Temperature profiles at the most accurate meshes. (c) The storage vessel showing the generated finer mesh (d) Zoomed-in finer mesh generated by COMSOL Multiphysics model, showing the bed, steel, and insulations domains

2.5 Energy Storage Analysis and heat loss rate

Steady-flow thermal energy equation (1st law of thermodynamics) is used to analyze the energy storage in the storage tank and thus calculate the storage efficiency during the storage process, Eqn. 5 determines the amount of energy stored in the bed domain [53].

$$Q_{bed} = \int_{T_o}^{T_{bed}} mCp_{gravel}(T)dT \quad (5)$$

where T_o and T_{bed} are the initial and averaged bed temperature, m is the mass of gravel, and Cp_{gravel} is the specific heat capacity of the gravel beads. The total energy delivered to the system is defined in Eqn. 6:

$$Q_{supplied} = \dot{m}Cp_{air}\int_{0}^{t_s}(T_{hot} - T_o)dt \quad (6)$$

where $\dot{m}$ is the mass flow rate, Cp_{air} is the heat capacity of the air (constant due to the low inlet temperature), T_{hot} is the inlet temperature ($T_{hot} = 95$°C, T_o is the ambient temperature, and t_s is the storage time.

Based on Eqn. 5 and Eqn. 6 above, the charging efficiency is calculated, which represents the ratio of energy stored in the bed to energy supplied to the storage bed.

$$\zeta_{charging} = \frac{Q_{bed}}{Q_{supplied}} \quad (7)$$

The heat loss rate per unit area is calculated using Eqn. 8 based on Fouriers law [53]

$$q_r'' = -k\frac{dT}{dr} \quad (8)$$

where q_r'' is the heat loss rate per unit area, k is the thermal conductivity, $\frac{dT}{dr}$ is the temperature gradient in radial direction (through the thickness of insulation).

3. Results and Discussion

The result section focuses on the influence of heat losses on the thermal performance and thus the energy stored into the bed.

3.1 Temperature Distribution- Model Validation

Fig. 4 shows the temperature distribution in the storage domain from the experiment and model. as can be noticed, the agreement between the numerical and experiment results is excellent. The bed is at high temperature at $x = 0\ to\ x = 0.2\ m$. Then the temperature drops along the axial height of the storage domain. At the outlet, the bed domain at low temperature 28℃ (the temperature inside the lab room). The charge process stops when the outlet temperature starts to increase and reaches 32℃ (4 degrees above the room temperature), which is considered cut off temperature. The charge time was 4 hrs. and 20 minutes. As can be observed, the temperature stored into the bed is very low. This is because of the heat losses to the surrounding during the entire storage period which will be seen next.

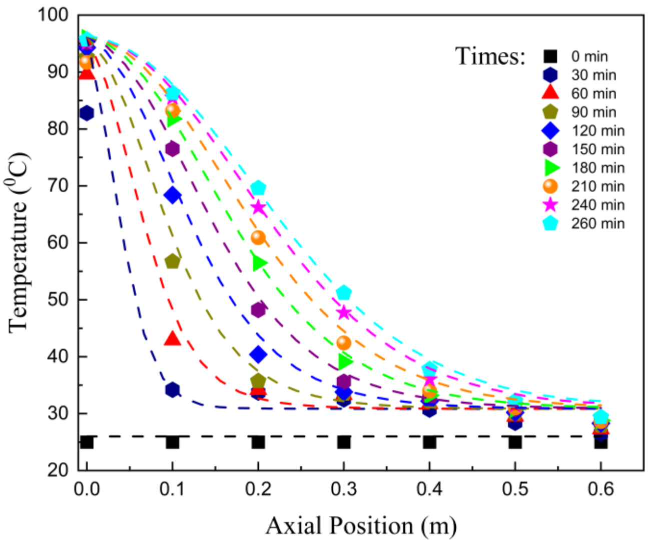


Fig.4 Temperature profiles in the storage domain during storage process. Experimental vs. model results. Dashe d lines represent the model results.

3.2 Heat Losses to the Ambient

Heat losses to the surrounding have significant influence on the stored energy in the storage tank and thermal efficiency. Heat losses occur due to conduction through steel wall and insulation and by convection to the ambient [54]. Commercial fiberglass was used to insulate the packed bed from the ambient and prevent heat losses. However, significant impact of heat losses was noticed during the storage period as can be observed in Fig. 5, where a thermal digital camera was used to take images during the storage process at several time intervals. As can be seen the amount of heat losses to the surrounding is very high because of the high temperature at the surface. High heat losses are due to the low flow rate [54]. In addition, small thickness of the insulation is another reason for high temperature at the surface of the insulation. At the inlet of the bed domain $x = 0\ cm$ the temperature of the surface reaches about 44.7℃ at time interval $t = 180\ min.$ which indicates that heat losses to surrounding is very high which reduces the energy stored in the storage tank and charging efficiency. At the middle of the bed ($x = 30\ cm$), the maximum temperature at the surface is 35.9°C after 3 hours of charging. At the outlet ($x = 60\ cm$), the surface temperature is 32.1°C. Therefore, the maximum heat losses to the ambient occurs at ($x = 0 - 20\ cm$).

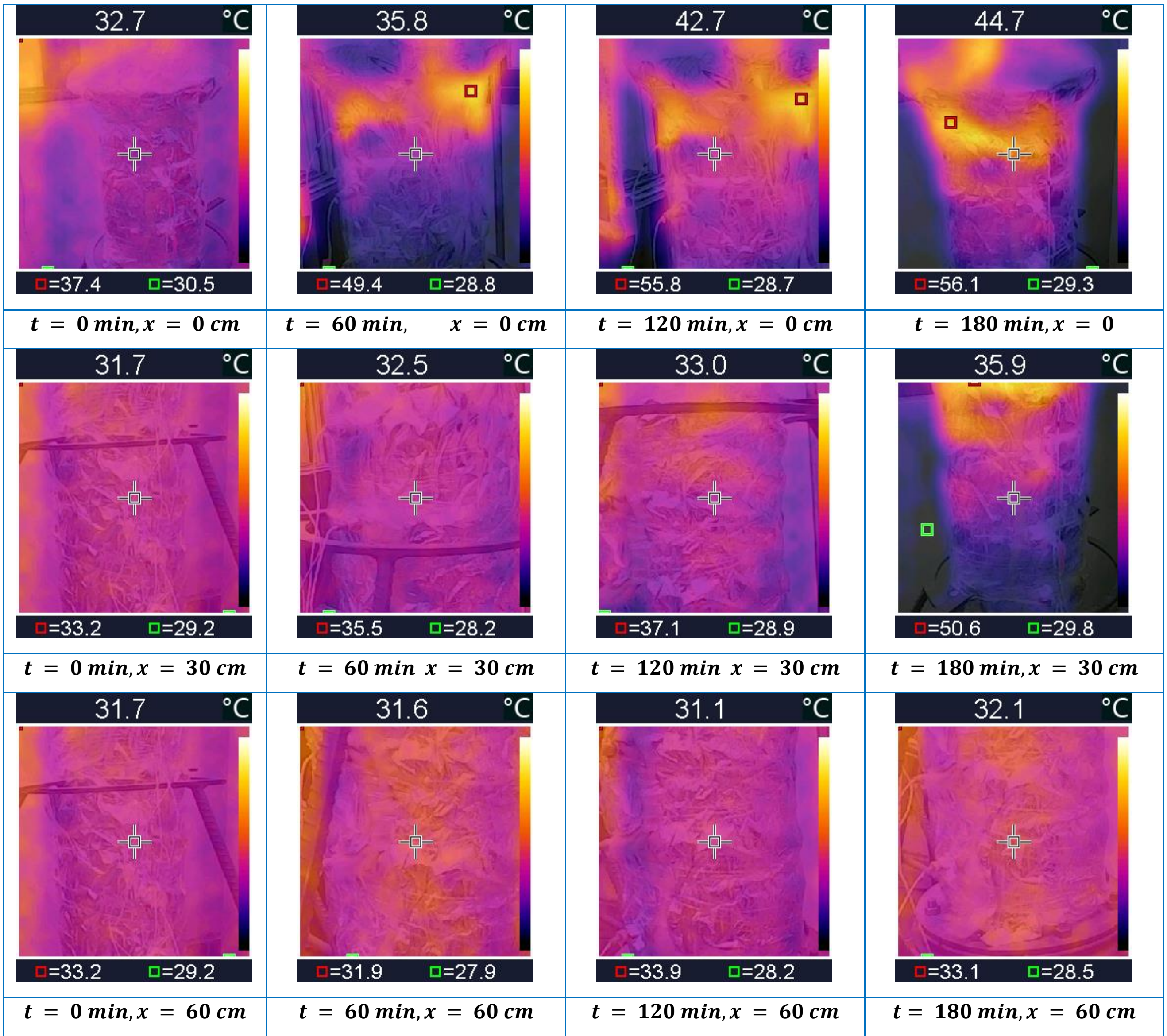


Fig. 3 Temperature of the surface of the insulation wall (images from digital camera) showing the surface temperature at different time intervals.

Fig shows the radial temperature inside the steel wall and insulation. As can be seen, the temperatures at the surface agree with surface temperatures that have been taken by the digital camera. For example, at x = 0, t = 180 min, the surface temperature is ~ 44.7°C which can be seen in Fig. 3. In Fig, at x = 0, t = 180 min, the surface temperature from the model is 45.8°C. High surface temperature can be seen at the inlet ($x = 0$) of storage tank which leads to high heat losses to the ambient.

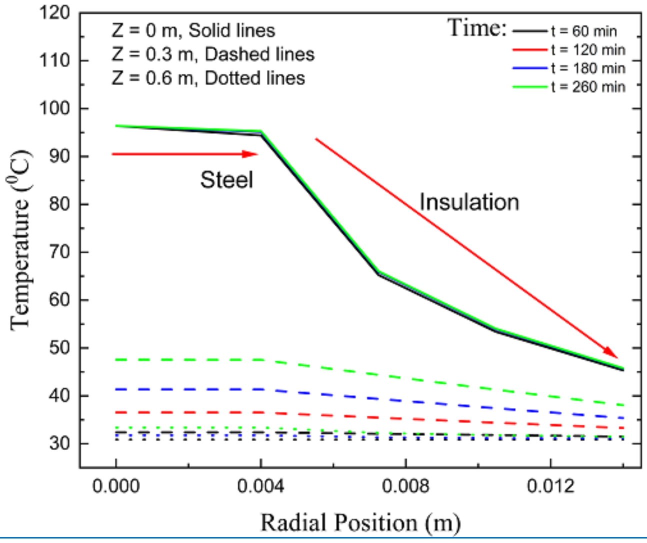


Fig. 6 Radial temperature profiles inside steel and insulation domains showing the temperature gradient due to the heat transfer by conduction and convection.

To quantify the systems performance based on heat loss from the storage packed bed, heat loss rate per unit area is calculated using Eqn. 8. Table 3 shows the heat loss per unit area at low flow rate which is determined based on temperature gradient through the insulation. As can be observed, the heat loss rate increases with the increase of time, leading to a reduction in the storage energy. The agreement between the experimental and numerical results are reasonable. At early times and outlet of the bed ($z = 0.6\ m$), the heat loss rate is zero due to no temperature gradient.

Table 3: Experimental and numerical heat loss rate per unit area from the storage packed bed system.

Axial section (m)	Heat loss rate (W/m^2)			
	Experiment			
	t = 60 min	t = 120 min	t = 180 min	t = 260 min
0	172.5	185.1	189.6	191.1
0.3	15.6	16.2	35.7	73.8
0.6	0	0	1.8	3.9
	Model			
0	191.1	191.1	190.8	190.8
0.3	2.1	14.4	29.1	47.4
0.4	0	0	0	5.1

3.3 Energy Stored in the Storage Tank

In thermal energy storage packed bed systems, the amount of energy stored is an important parameter that should be considered due to its importance in determining storage/recovery efficiency from the storage system. Fig. 7 shows the amount of energy storage in the tank during the storage process from the experiment and CFD model. The experimental energy results match the model result with some error. The total energy was calculated by volume averaging the bed domain temperature at each time interval during the storage process. The final total energy is 962.64 and 883.6 kJ from the experimental work and model respectively at 0.0004 kg/s. At flow of 0.006 kg/s the total energy is 990 kJ from the model. As can be observed, the amount of energy stored into the bed domain is low due to the influence of heat losses during the entire storage process, leading to low thermal performance. The charging efficiency is 43.3% and 55% for 0.0004 and 0.006 kg/s respectively. Fig. 8 a-b shows the 2-D surface temperature in the three domains (bed, stainless steel, and insulation). As can be seen, the bed has more energy to store in case of 0.006 kg/s flow rate based on the high and low temperatures in the bed domain.

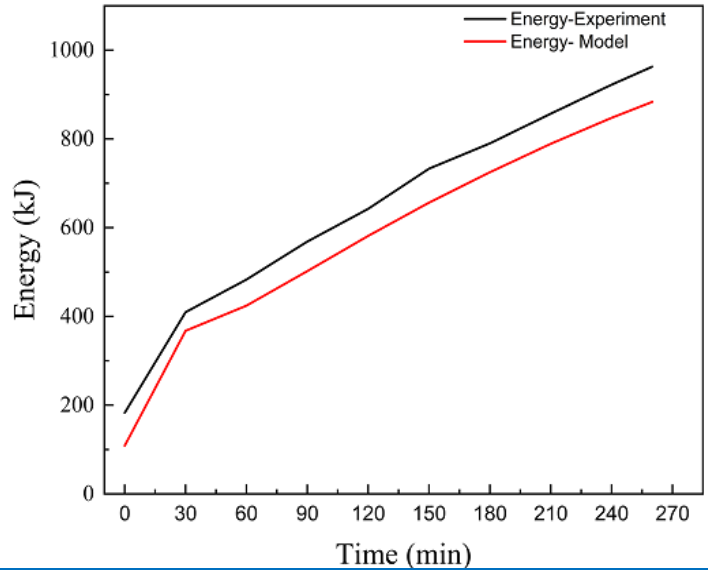
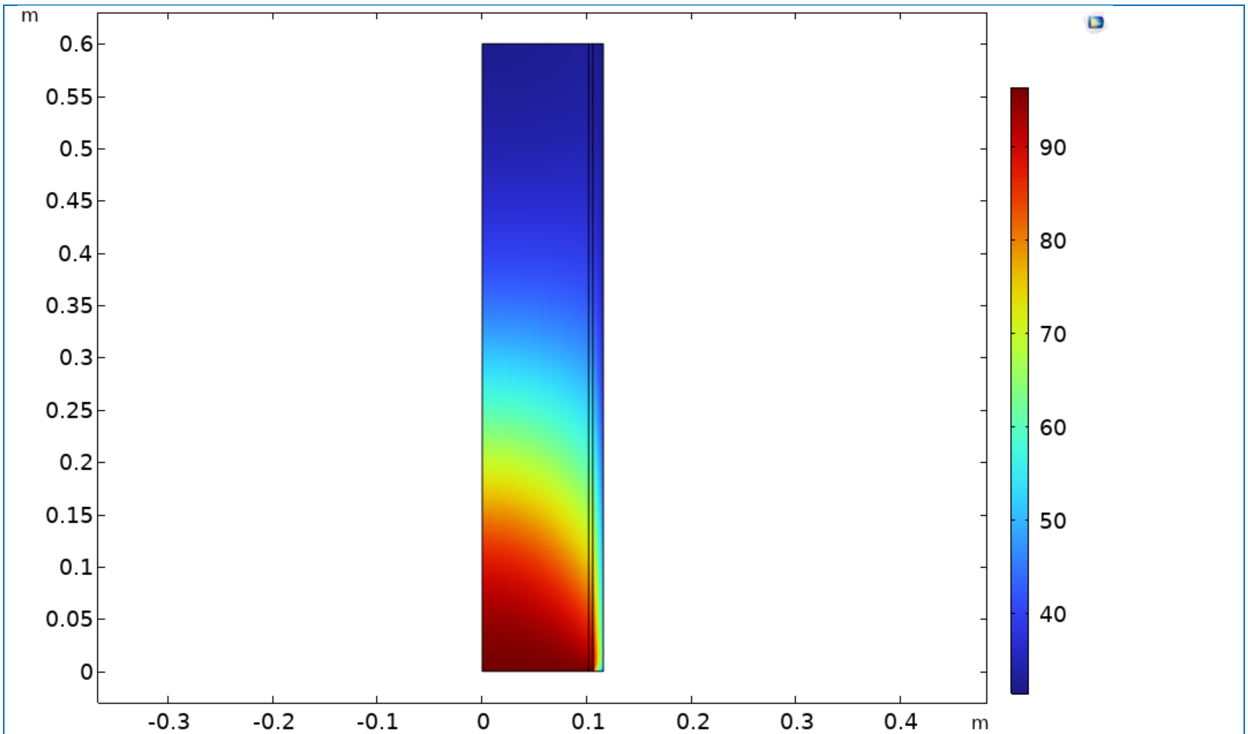
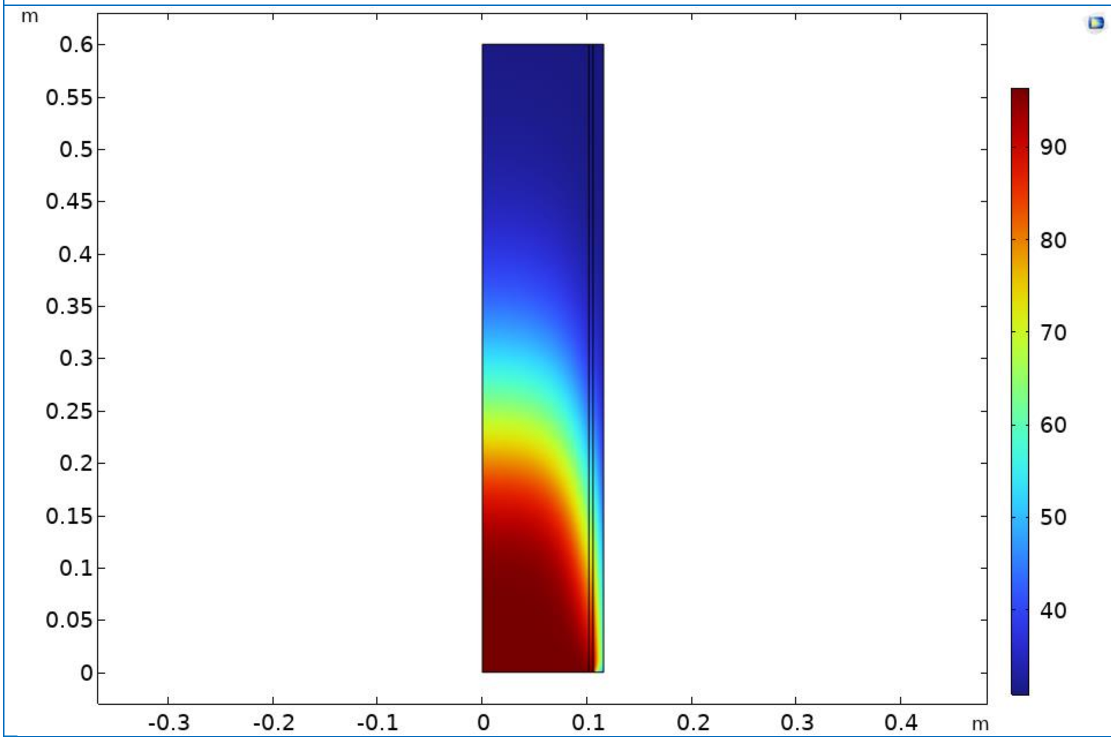


Fig. 7 Thermal energy stored in the storage domain over time during the storage period.



(a) Low flow rate 0.0004 kg/s



(b) High flow rate 0.006 kg/s

Fig. 4 Two-Dimensional surface temperature from the model at low and high flow rates showing the hot/cold regions in the bed, stainless steel, and insulation domains.

4. Conclusion

The thermocline of a lab-scale storage vessel system was analyzed experimentally and numerically to study the impact of heat losses on the thermal performance and efficiency. Air was used as working fluid with an inlet temperature of 95℃. Gravel was used as storage materials at 9 mm size with porosity of 0.4. One flow rate (0.0004 kg/s) was tested experimentally and numerically. A CFD model was built to make sure that the present storage tank performs as it should be. The agreement between the numerical results and the experimental results is very well. Therefore, another flow rate (0.006 kg/s) was tested numerically. The results shows that heat transfer losses to the surrounding are very high which affects the thermal behavior and thus thermal efficiency. Heat losses are high due to the low flow rate and small thickness of the insulation. High flow rate shows better performance. The charging thermal efficiency was 43.3% at 0.0004 kgs and 55% at 0.006 kg/s and thus the charging efficiency increases with flow rate due to the increase of energy stored into the bed. Future work will focus on storage/recovery performance and high inlet temperature in axial and radial flow configurations.

Nomenclature

ε	packed bed porosity (-)
ρ	density (kg/m^3)
p	pressure (Pa)
C_p	heat capacity at constant pressure (J/kg K)
h	height of the storage tank (m)

k	thermal conductivity (W/m K)
$\dot{m}$	mass flow rate (kg/s)
m	mass of gravel (kg)
Q_{loss}	energy losses to ambient (W/m^3)
t	time (s)
T	temperature (K)
V	velocity (m/s)
ζ	efficiency

Subscripts

f	fluid
s	solid
hot	hot temperature
$effe$	effective
j	solid domains (stainless steel/insulation)
0	initial
t_s	storage time
∇	del operator

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Conflicts of Interest

The author declares no conflicts of interest.

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