



NUMERICAL SIMULATION OF A STRATIFIED THERMAL STORAGE SYSTEM, EFFECT OF THE REYNOLDS NUMBER ON THE THICKNESS OF THE THERMOCLINE ZONE

Yassine Sougtan, Hamid El Qarnia, Abdelhamid Kheiri, Mohammed Khalij

► To cite this version:

Yassine Sougtan, Hamid El Qarnia, Abdelhamid Kheiri, Mohammed Khalij. NUMERICAL SIMULATION OF A STRATIFIED THERMAL STORAGE SYSTEM, EFFECT OF THE REYNOLDS NUMBER ON THE THICKNESS OF THE THERMOCLINE ZONE. International Conference on MATERIALS & ENERGY, Jun 2021, Metz, France. hal-03531863

HAL Id: hal-03531863

<https://hal.science/hal-03531863>

Submitted on 18 Jan 2022

HAL is a multi-disciplinary open access archive for the deposit and dissemination of scientific research documents, whether they are published or not. The documents may come from teaching and research institutions in France or abroad, or from public or private research centers.

L'archive ouverte pluridisciplinaire **HAL**, est destinée au dépôt et à la diffusion de documents scientifiques de niveau recherche, publiés ou non, émanant des établissements d'enseignement et de recherche français ou étrangers, des laboratoires publics ou privés.

NUMERICAL SIMULATION OF A STRATIFIED THERMAL STORAGE SYSTEM, EFFECT OF THE REYNOLDS NUMBER ON THE THICKNESS OF THE THERMOCLINE ZONE

Yassine Sougtan^{a,b,*}, Hamid El Qarnia^a, Abdelhmaid Kheiri^b, Mohammed Khalij^b

^a Cadi Ayyad University, Faculty of Sciences Semlalia, Fluid Mechanics and Energetic Laboratory, Affiliate to CNRST, URAC 27, Department of Physics, P.O. 2390, Marrakesh, Morocco

^b Université de Lorraine, CNRS, LEMTA, Nancy F-54000, France

*Corresponding author: Email: yassine.sougtan@ced.uca.ma / yassine.sougtan@univ-lorraine.fr

ABSTRACT

Numerical simulations are carried out on a thermocline thermal storage system with thermal oil as the storing medium. The main objective was to study the effect of the Reynolds number on the thermocline zone thickness during the charging period. Results show that Reynolds number has a significant influence on the thermocline thickness and the thermocline zone near the wall is slightly affected by increasing the Reynolds number. The findings shows also that the temperature of the fluid varies significantly in the axial direction as compared to the radial direction

NOMENCLATURE

ρ	density (kg/m ³)
u	axial velocity (m/s)
v	radial velocity (m/s)
μ	dynamic viscosity (Pa·s)
p	pressure (Pa)
g	gravitational acceleration (m/s ²)
T	temperature (K)
cp	specific heat capacity of fluid (J/kg·K)
λ	conductivity (W/m·K)
m	mass flow rate (kg/s)
Re	Reynold number
t	time (s)
D	diameter (m)
H	height (m)

INTRODUCTION

The non-continuous nature of renewable energy requires the adoption of an energy storage system to ensure the continuity of the production. For solar

thermal systems, many storage technologies are studied, some are profitable and have been commercialized while many of them require more exploration enhance their competitiveness.

This study is focused on the stratified thermal storage technology, so-called thermocline storage, which is a promising technology of heat storing. The basic principle of this technology lies in the creation of a zone with a strong temperature gradient between the hot and cold fluid during the charging or discharging phases. This zone, called the thermocline zone, acts as a thermal barrier between hot and cold fluid and prevent hydraulic and thermal mixing. This storage system type may replace advantageously the well-known and high-cost two-tank thermal storage.

In order to achieve an optimal thermal stratification and improve the efficiency of the storage system, its key parameters have to be first found. These parameters are a priori the tank size, the aspect ratio, the mass flow rate, the inlet and initial tank fluid temperatures and the fluid properties.

Several studies have been conducted to optimize these parameters to get a better stratification.

Several studies have been conducted to optimize these parameters to get a better stratification.

Nelson et al. [1] have carried out a parametric study to analyze the stratification decay in thermally stratified vertical cylindrical cool storage systems. They study, in static and dynamic modes of operation, the effect of the length to diameter ratio, the wall thickness to length ratio, the thermo-physical properties of the material of the storage tank, the type and thickness of the insulation. They found that the stratification is not significantly improved beyond an aspect ratio of 3. Besides, few advantages in thermal stratification are obtained beyond the tank length/wall thickness ratio of 200

for a storage tank regardless of the aspect ratio. They also observed that the increase of Peclet number generates a degradation of the thermal stratification due to mixing at the inlet and outlet. The material of the storage tank has small effect on the thermocline during charging and discharging phases. In dynamic operating mode, the effects of mixing override the influence of the other parameters. However, in static storage systems, the effect of the wall material should be considered.

Advaith et al. [2] have shown experimentally that the increase of the Atwood number leads to the a more efficient stratification which could last longer, i.e up to 6 hours. However, sufficient insulation of the storage tank should be used to reduce heat loss by axial conduction through the wall of the storage tank. On the other hand, the radial diffusers are devices widely used in large scale stratified thermal energy storage tanks. They ensure a better distribution of the fluid over any section of the storage tank. The shape and dimensions of the diffusers also influence the stratification. Yajundeng et al [3], studied these effects and proposed a novel nonequal-diameter radial diffuser. The results of their study showed that the thermal stratification of the stratified storage tank is optimal when the ratio of the long baffle and the tank diameters is $1/3$ and the distance between the two baffles is as small as possible.

Vanneremet al [4] studied the impact of the interstitial fluid velocity on the storage performance and they found an optimal velocity of $4,104 \text{ m/s}$ that maximizes the storage utilization rate (80.6%) for ideal charges between 293°C and 393°C .

Karim [5] studied an air-conditioning system with naturally stratified water storage tank with a new type of diffuser. Factors that influence the performance of chilled-water storage tanks were investigated. The results indicated that stratified storage tank consistently stratified well without any physical barrier. Investigation also showed that storage efficiency is decreased with increasing flow rate due to the increased mixing of warm and chilled water. Diffuser design and layout primarily affected the mixing near the inlet diffuser and the extent of this mixing had primary influence on the shape of the thermocline. Numerical simulations have been carried out by Shaikh et al [6] on single media thermal storage system with and without flow distributor for different fluid inlet configurations in order to investigate the behavior of storage system at different configurations and under various operating conditions. Simulations have been performed for two working fluids, water and solar salt.

The aim of the present work is to investigate the nature of the temperature profiles in a single media tank, for different Reynolds number, through a complete three-dimensional numerical simulation. The mass, momentum and energy conservation equations are solved for the single media tank using the commercial software ANSYS Fluent ©. The model developed is validated with analytical and experimental results existing in the literature. The effect of the Reynolds number on the thickness of the thermocline zone is studied.

THERMOCLINE TANK CFD MODEL

1. COMPUTATIONAL DOMAIN

The 3D schematic of the thermal energy storage tank considered in the present study is illustrated by the Figure 1a.

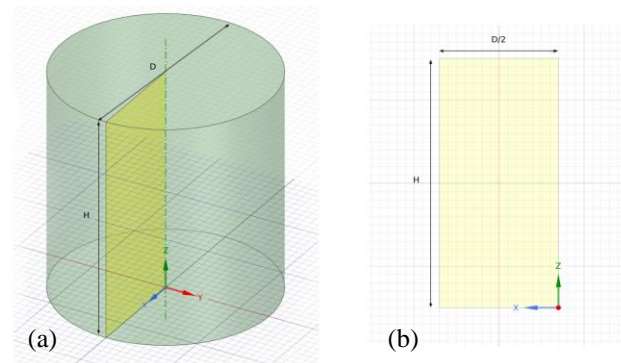


Figure 1. Geometry of the thermocline tank (a) in 3D (b) computational domain (2D)

As it is mentioned in the figure, the height of the storage tank is H and its diameter is D . The storage tank is considered adiabatic and the flow and the thermal fields are assumed to be axisymmetric. A 2D analysis is then adopted to simplify the mathematical model and gain in terms of calculation time. Figure 1b shows the 2D schematic of the computational domain adopted for the present numerical investigations.

2. GOVERNING EQUATIONS

The thermal behavior of the fluid inside the thermocline tank is governed by the time-dependent mass, momentum and energy conservation equations.

$$\frac{\partial \rho}{\partial t} + \text{div}(\rho \vec{u}) = 0 \quad (1)$$

$$\frac{\partial(\rho \vec{u})}{\partial t} + (\vec{u} \cdot \text{grad}) \rho \vec{u} = -\text{grad} p + \mu \Delta \vec{u} + \rho \vec{g} \quad (2)$$

$$\frac{\partial(\rho c_p T)}{\partial t} + \vec{u} \cdot \text{grad}(\rho c_p T) = \lambda \Delta T \quad (3)$$

where, ρ is the fluid density, u is the velocity, μ is its dynamic viscosity, p is the pressure, c_p is specific heat capacity of the fluid, T is the temperature of fluid and λ is its thermal conductivity. The above equations are based on the following assumptions:

- The fluid is incompressible, Newtonian, isotropic and homogenous;
- The flow is laminar;
- The thermo-physical properties of the fluid are independent of the temperature;
- The viscous dissipation in the fluid is neglected.

3. BOUNDARY AND INITIAL CONDITIONS

The wall is assumed to be perfectly adiabatic, thus a zero-heat flux boundary condition type is imposed on the wall. During all the numerical simulations, the fluid is supposed to enter the tank at constant temperature and velocity. The following boundary conditions are used:

$$u|_{z=H} = u_{in} \text{ (normal inlet velocity)}$$

$$v|_{z=H} = 0 \text{ (radial inlet velocity)}$$

$$T|_{z=H} = T_{in} \text{ (inlet fluid temperature)}$$

At outlet ($z=0$),

$$P_{out} = P_{atm} \text{ (Pressure outlet)}$$

At the wall ($r=R$),

$$u|_{r=R} = 0 \text{ (No slip condition)}$$

$$\frac{\partial T}{\partial r}|_{r=R} = 0 \text{ (Adiabatic wall)}$$

At axis ($r=0$),

$$\frac{\partial u}{\partial r}|_{r=0} = 0$$

$$v(r=0) = 0$$

$$\frac{\partial T}{\partial r}|_{r=0} = 0$$

During the charging phase, the fluid, that is assumed to be a hot thermal oil, enters from the top of the storage tank, therefore $T_{in} = T_h$; while during the discharging phase, cold thermal oil enters from the bottom and $T_{in} = T_c$. During the charging phases, the initial temperature of the thermal oil inside the

storage tank is set equal to T_c while it's set to T_h at the discharging phase.

NUMERICAL PROCEDURE

ANSYS Fluent © (2020 R1) software is used to perform numerical simulations of fluid flow and heat transfer in the thermal energy storage tank. It implements the finite element method, in order to solve the discretized equations associated to the previous model.

VALIDATION OF THE RESULTS

The results of the experimental work of Mawire [7] were chosen to validate our CFD model. Simulations were made on the same geometry and under the same operating conditions.

The predicted and experimental axial temperature profiles of the average temperature (between the axis and the wall) are compared. Figure 2 presents the temperature profiles at four different time.

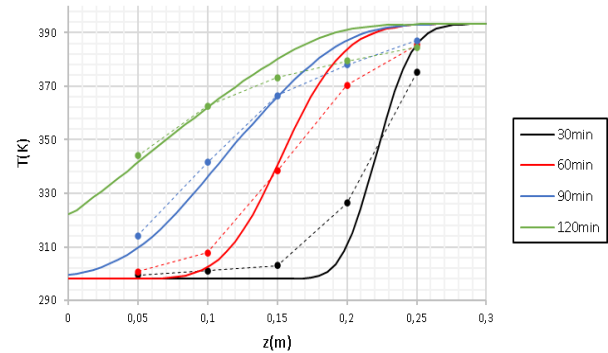


Figure 2. Experimental ---- and simulated — temperature profiles along the height of the storage at different time intervals

As can be seen from the figure, the numerical results follow the same trend and behavior as the experimental results. However, they show relative small difference especially at the inlet (relative error does not exceed 12%). The reason of such a difference is due to that in numerical simulations the axial wall conduction and the radial and axial heat losses towards the exterior are not taking into consideration.

RESULTS AND DISCUSSIONS

Thermal oil is used for this study. Its thermo-physicals properties are listed in Table 1. The operating conditions are summarized in table 2

Table 1: Thermo-physicals properties of the thermal oil

Properties	Thermal oil (shell)
Density ρ (Kg.m ⁻³)	876
Specific heat C_p (J.Kg ⁻¹ .K ⁻¹)	1809
Thermal conductivity λ (W.m ⁻¹ .K ⁻¹)	0.136
Dynamic viscosity μ (Kg.m ⁻¹ .s ⁻¹)	0.253731

Table 2: Operating conditions

Operating conditions	First case study
Hot temperature T_h (K)	393.15
Cold temperature T_c (K)	298.15
Inlet velocity (m/s)	3.573×10^{-5}

1. MESH CONVERGENCE

The computational domain is discretized using the adaptive mesh technique with inflation near the wall where there are large gradients (Figure 3a). The time step is adjusted in an adaptive way such that the Courant number is always maintained less than unity. A relative tolerance of 10^{-4} is used as the convergence criterion for mass and momentum equations and 10^{-6} for energy equation.

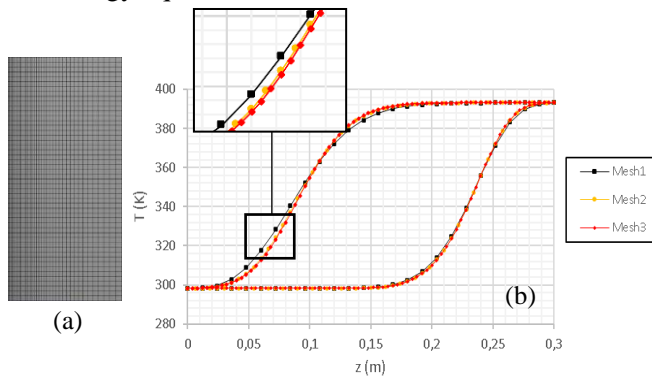


Figure 3. (a) Mesh method for simulation geometry
(b) Axial temperature for $r=0$ (Axis) and for $r=D/2$ (Wall) at $t= 60$ min for different meshing size

Figure 3b shows the results of the mesh convergence study. Table 3 shows the details of each mesh used in this study. The axial temperature profiles for $r = 0$ and $r = R$, at $t = 1h$ for the different mesh sizes is plotted in Figure 3b. The figure shows that after the

M2 mesh, the temperature profiles remain unchanged (the difference in terms of relative error does not exceed 0.5%). The simulations were made on an Intel core i7-8750 CPU. 2.20 Ghz processor.

Table 3: Mesh details

	Mesh1	Mesh2	Mesh3
Elements	900	2000	4500
Nodes	962	2091	4636
Element type	Quadrilateral	Quadrilateral	Quadrilateral
Quality Min	1	1	1

2. TEMPERATURE VARIATION INSIDE THE STORAGE TANK

2.1. TEMPERATURE VARIATION IN RADIAL AND AXIAL DIRECTIONS

First, the temperature variation in the storage tank during the charging phase is studied. Figure 4 plots the temperature contours in the computational domain for different instants. The results indicate that the temperature varies in the axial direction as well as in the radial direction. This observation is consistent with the experimental results reported by Mawire [7]. As time progresses, the heat diffuses in the oil and the temperature increases.

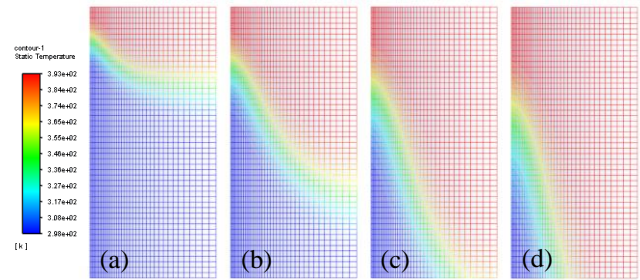


Figure 4. Temperature contours in the storage tank at (a) $t = 30$ min, (b) $t=60$ min, (c) $t=90$ min and (d) $t=120$ min

2.2. TEMPERATURE VARIATION ALONG THE TANK HEIGHT

Figure 5 shows the axial profile of the area mixed temperature of the oil as a function of the height of the storage tank during the charging phase. Area mixed temperature is calculated defined as:

$$T_{mix} = \int_0^D \frac{2\pi\rho T(r,z)U(r,z)r}{\dot{m}} dr$$
 where \dot{m} is the mass flow rate of the oil. We observe the formation of the

thermocline layer and its subsequent propagation over time. As the charging time progresses, the thermocline zone is moved to the bottom of the storage tank. At the end of the charging process, the thermal oil in the whole storage tank is at the hot temperature.

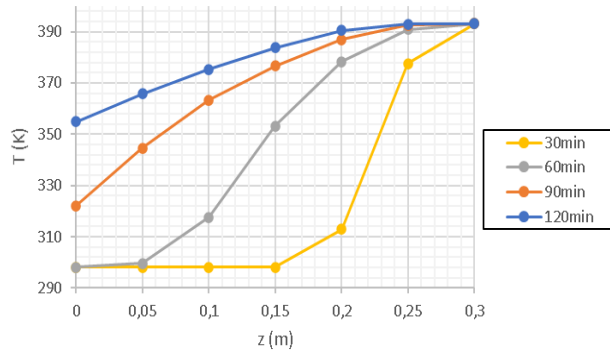


Figure 5. Axial profile of the area mixed temperature as a function of the height of the storage tank

3. EFFECT OF REYNOLDS NUMBER

The heat transfer in the storage tank is governed by two phenomena: advection and thermal diffusion. The thermocline zone is then dependent on these two phenomena. Numerical simulations are achieved for three different values of the Reynolds number. The variation of the Reynolds number is ensured by the variation of the inlet velocity of the thermal oil. The results are shown in Figures 6 (a) and (b) for the same times. As it is shown in these figures, by increasing the Reynolds number, keeping the same fluid and dimensions of the storage tank, advection is enhanced and consequently the thermocline zone is increased. However, the impact of increasing Reynolds number on the thermocline zone is not the same for axial position ($r=0$) and near the wall ($r=D/2$). Indeed, Figure 6, the thermocline zone near the wall is slightly affected increasing the Reynolds number, while it is increased in the axial position. This is due to the effect of advection which is more important in the axial position when increasing the Reynolds number.

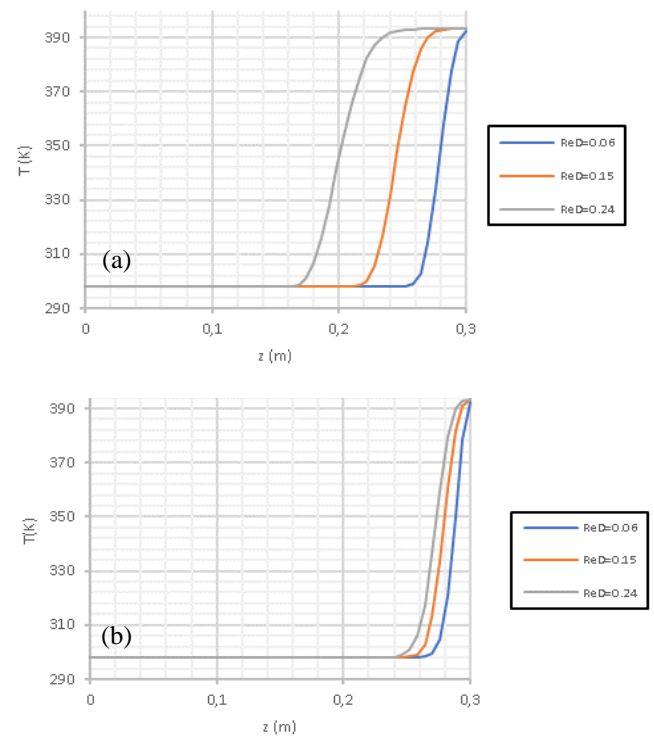


Figure 6. Axial temperature profile at $t = 5\text{min}$ for different Reynolds number (a) axial position ($r = 0$), (b) near wall position ($r = D/2$).

CONCLUSION

Numerical simulations are carried out on a thermocline thermal storage system with thermal oil as the storing medium.

The simulations are done to achieve two main objectives: to study the thermal behavior of the thermocline storage tank and the effect of the Reynolds number on the thickness of the thermocline zone.

The following conclusions are made from the present analysis:

- The temperature of the fluid varies significantly in the axial direction as compared to the radial direction.
- Reynolds number has a significant influence on the thermocline thickness.
- The thermocline zone near the wall is slightly affected increasing the Reynolds number

KEYWORDS

Storage, Thermocline, Stratification, Reynolds number, CFD, ANSYS Fluent ©.

REFERENCES

1. Nelson, J. E. B., Balakrishnan, A. R., & Srinivasa Murthy, S. (1999). Parametric studies on thermally stratified chilled water storage systems. *Applied Thermal Engineering*, 19(1), 89–115. doi:10.1016/s1359-4311(98)00014-3
2. Advait, S., Parida, D. R., Aswathi, K. T., Dani, N., Chetia, U. K., Chattopadhyay, K., & Basu, S. (2021). Experimental investigation on single-medium stratified thermal energy storage system. *Renewable Energy*, 164, 146–155. <https://doi.org/10.1016/j.renene.2020.09.092>
3. Deng, Y., Sun, D., Niu, M., Yu, B., & Bian, R. (2021). Performance assessment of a novel diffuser for stratified thermal energy storage tanks – The nonequal-diameter radial diffuser. *Journal of Energy Storage*, 35(January). <https://doi.org/10.1016/j.est.2021.102276>
4. Vannerem, S., Neveu, P., & Falcoz, Q. (2021). Experimental and numerical Investigation of the impact of operating conditions on thermocline storage performance. *Renewable Energy*, 168, 234–246. <https://doi.org/10.1016/j.renene.2020.12.061>
5. Karim, M. A. (2011). Experimental investigation of a stratified chilled-water thermal storage system. *Applied Thermal Engineering*, 31(11–12), 1853–1860. <https://doi.org/10.1016/j.applthermaleng.2010.12.019>
6. Shaikh, W., Wadegaonkar, A., Kedare, S. B., & Bose, M. (2018). Numerical simulation of single media thermocline based storage system. *Solar Energy*, 174(February), 207–217. <https://doi.org/10.1016/j.solener.2018.08.084>
7. Mawire, A. (2013). Experimental and simulated thermal stratification evaluation of an oil storage tank subjected to heat losses during charging, 108, 459–465.