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Performance Characterization of Packed Bed Storage System

A.A. Adeyanju
Department of Mechanical and Manufacturing Engineering,
University of the West Indies, St. Augustine, Trinidad and Tobago, Nigeria

Abstract: Modeling of packed-bed heat transfer can be a problem of immense complexity in some cases requiring the use of 3-D finite-element techniques to understand the dynamics of stratification and fluid-solid interaction during different modes of operation. The analytical solution to the packed bed heat transfer was identified in this study and a set of governing equations that economically and accurately characterize the dominant energy transfer mechanisms in a charging or discharging packed bed storage tank over long time periods that include multiple cycles were analyzed.

Key words: Packed-bed, analytical, heat transfer, characterization, stratification

INTRODUCTION

The ability to store large amounts of high temperature thermal energy both efficiently and at low-cost has the potential to increase the economic viability of solar power generation. Without storage or conventional fuel sources as backup, solar power generation is subject to the regular and irregular variations in insolation. This variation in insolation results in a generated energy profile that may or may not be consistent with end-user demands. Thermal energy storage enables the delivery of solar-thermal power to be tailored to meet end-user demands.

Most energy storage systems proposed for solar electricity generation systems seek to accomplish one of three tasks:

- Compensate for the normal diurnal variation in solar radiation
- Shift electricity output to match the utility peak period
- Extend power plant operation past sunset

Many different energy storage concepts have been proposed. Energy can be stored readily in many forms: kinetic, potential, electric, thermal and chemical. Thermal energy storage is the most common choice for solar-thermal power generation. In solar thermal power generation solar radiation is captured as thermal energy in a heat transfer fluid before conversion to electricity. Using thermal energy storage in solar-thermal applications prevents conversion losses associated with using another form of energy storage as well as eliminating added system complexity. Of the thermal storage concepts, sensible (vs. phase-change or chemical reaction) thermal energy storage is accepted as the near-term option for parabolic trough solar power plants (Herrmann and Kearney, 2002). The storage systems of principal interest in this category are two-tank storage, single-tank stratified (thermocline) storage and concrete (packed bed) storage. When thermal storage systems are discussed herein, direct indicates the solar field heat transfer fluid is also used as the storage medium. In an indirect storage system the solar field heat transfer fluid is separated from the storage medium via a heat exchanger.

Direct storage systems eliminate losses associated with the heat exchanger used in indirect systems. However, some solar field heat transfer fluids have pressurization requirements that would make direct storage systems prohibitively expensive. In addition, indirect systems have the added design flexibility of using different fluids in the solar field and storage system.

This study consider a tank with diameter D, filled with coarsely-packed solid material and fluid flowing through the free space, characterized by a void fraction ε:

$$\varepsilon = V_f/V_f+V_s \qquad (1)$$

where, V_f and V_s are the solid and fluid volumes.

Now consider a differential segment of this tank of length dx, shown in Fig. 1. Figure 2 shows one-dimensional fluid and solid energy balances corresponding to the differential tank segment shown in Fig. 1. Axial conduction, viscous dissipation and losses to the environment are neglected. The energy balances are written in one dimension assuming that significant temperature variations occur only in the axial (x) direction. This formulation of the one-dimensional packed-bed heat transfer problem is popular and credited to Schumann (1929).

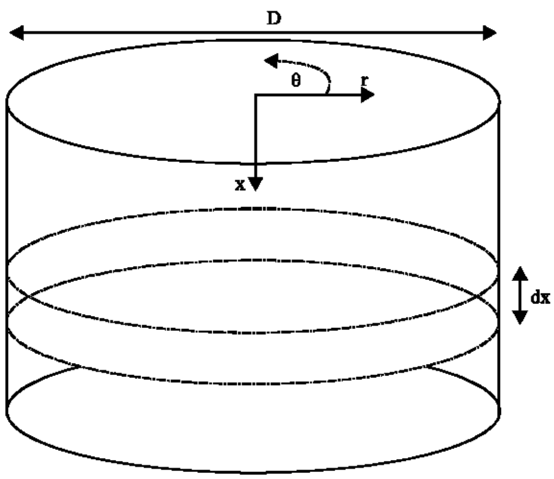

Fig. 1: Storage tank coordinate system and differential control volume

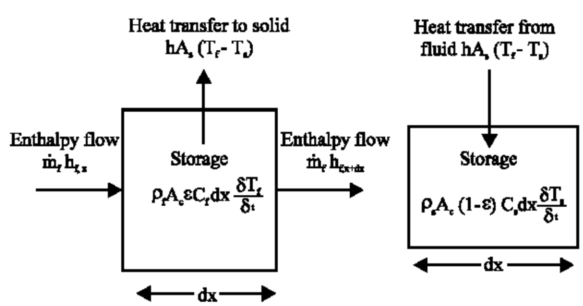

Fig. 2: Energy balance on the fluid and solid components of the differential element of the storage tank

It is appropriate to neglect environmental losses due to the large volume to surface area ratio for storage units designed for power production system. The thermal losses to the environment will be negligible compared to energy transfer in the tank during a charge or discharge cycle in a well-designed unit. In the event, the tank lies idle for long periods shell losses become significant. For well designed storage systems axial conduction will be negligible. Limiting axial conduction is essential to maintaining stratification in the fluid. One way to mitigate axial conduction effects is the presence of low-conductivity plates at various heights in the tank, which allow fluid flow but limit dynamic conductive effects as well as any conduction through the solid material. Maintaining stratification requires prevention of large-scale mixing, accomplished by limiting fluid velocity in the tank. Low fluid velocities correspond to a negligible viscous dissipation effect.

A one-dimensional energy balance performed on the control volumes shown in Fig. 2 results in governing partial differential equations for the fluid and solid, respectively:

$$m_f C_f\, dT_f/dx + \rho_f\, \varepsilon\, A_0 C_f d \times dT_f/dt = -hA_s\,(T_f - T_s) \quad (2)$$

$$\rho_s\,(1 - \varepsilon)\, A_0 C_s d \times dT_s/dt = hA_s\,(T_f - T_s) \quad (3)$$

In Eq. 2 and 3, as is the solid-fluid interface surface area available for heat transfer. A_s can be determined in terms of average particle diameter (dp) for the differential control volume if the particles are assumed to be spheres of uniform size:

$$A_s = 6\,A_C\,(1 - \varepsilon)\, dx/dp \quad (4)$$

A_c is tank cross-sectional area. If packing material particle size varies dramatically a technique that considered particle size variation (Eq. 4 does not) would be required to determine A_s.

It is important to make note of several additional assumptions implicit in the formulation of Eq. 2 and 3:

- Material and transport properties are constant
- Solid material is thermally lumped (internal conductivity is ~infinite, no temperature gradients within solid particles)

MATERIALS AND METHODS

The Schmuann equations have general applicability to packed bed heat transfer problems. It is useful to introduce a test case in order to provide a realistic framework for the thermal energy storage systems being considered.

The materials for the packed-bed stratified tank test consist of oil and a rock-sand. Approximate system parameters are listed in Table 1. Note that the bed material is treated as spheres with average diameter dp. The fluid-solid heat transfer coefficient, h is calculated using a correlation developed by Ranz and Marshall (Rohsenow *et al.*, 1998).

$$Nu_p = 2 + 1.8 Re_p^{1/2}\, Pr^{1/3} \quad (5)$$

where, Re_p and Nu_p are defined in terms of the particle diameter and axial fluid velocity (fluid velocity within the packed bed). This correlation was developed for a multi particle system incorporating heat transfer enhancement related to dynamic particle-fluid interaction. As bed void fraction decreases, linear velocity increases thus, increasing dynamic heat transfer enhancement.

Table 1: Storage system test case parameters

Parameters	Values
C_f	2400 J kg^{-1} K
C_s	1000 J kg^{-1} K
ρ_f	1000 kg m^{-3}
ρ_s	2400 kg m^{-3}
h	183 w m^{-2}K
T_o	300°C
T_s	400°C
d_p	0.01 m
ε	0.23
A_c	729 m^2
L	14 m

Faas (1986)

The test case considers a single charge period of 1 h with a HTF mass flow rate of 720 kg sec^{-1} and the following temperature boundary conditions:

$$T_f\,(x = 0, t) = 400°C \tag{6}$$

$$T_S\,(x, t = 0) = 300°C \tag{7}$$

Using the analytical solution to the Schumann model and the test conditions, it is possible to explore the parameters that govern packed-bed storage system performance.

Understanding, the dominant parameters will provide design information for future storage systems for solar power applications in addition to providing a framework for interpreting the results of more complex simulation and modeling efforts.

RESULTS AND DISCUSSION

It is helpful to characterize system performance in terms of dimensionless quantities. The first group examined here is defined as:

$$U = \frac{C_f \int m_f dt}{m_{s,\,total}\,Cs + m_{f\,total}\,C_f} \tag{8}$$

U is the ratio of the thermal capacitance of the fluid that flows through the system during some period of time, t and the total static thermal capacitance of the system (solid + entrained fluid). U is sometimes referred to as the utilization of the storage system and can be usefully thought of as a dimensionless thermocline penetration depth (Nellis and Klein, 2006). Figure 3 shows the relationship between the location of the fluid thermocline during a charge cycle and U. U can also be thought of an average normalized penetration depth (δ_{conv} L^{-1}). That is if U = 1 and the heat transfer coefficient were infinite then the solid and fluid would be uniformly at the fluid inlet temperature.

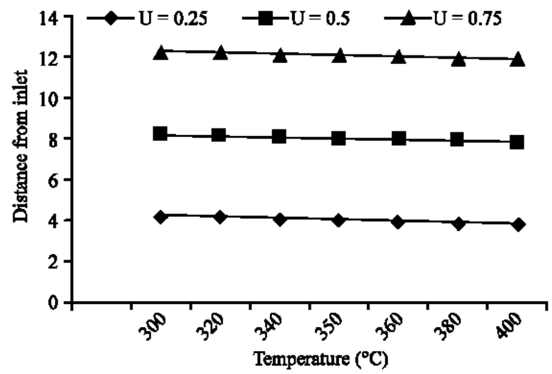

Fig. 3: The relationship between the Utilization (U) and thermocline penetration depth

There is a related dimensionless group that can be defined, R as the static thermal capacitance ratio of the solid and entrained fluid:

$$R = m_{f,total}C_f\,/m_{s,total}C_s \tag{9}$$

R provides an indication of how significant the thermal capacitance of the fluid entrained in the system is relative to the thermal capacitance of the solid material. The value of R depends on the bed void fraction as well as fluid and solid heat capacity and density. In the test case, R = 0.32. This value indicates the fluid capacitance is a significant component of the total system capacitance. This is an important observation, because it is common in the heat transfer community to neglect the thermal capacitance of the fluid when modeling gas-solid systems (Nellis and Klein, 2006). The majority of packed-bed literature concerns gas-solid systems, including the models designed for TRNSYS (Klein, 2006), thus, the results that are applied to packed-bed thermal storage systems must be carefully examined. Equation 2 and 3 consider the thermal capacitance of both the fluid and solid components of the bed.

The performance of a packed-bed thermal storage system is related to the heat transfer performance between the solid and fluid tank constituents. To describe the fluid-solid heat transfer performance NTU_{fluid} is defined in a manner analogous to the NTU used in heat exchanger analysis:

$$NTU_{fluid} = hA_{total}/m_fC_f \tag{10}$$

NTU_{fluid} = Related to the shape of the thermocline (temperature gradient) in the fluid

The effectiveness of a storage system for power generation requires the maintenance of a sharp (large NTU_{fluid}) thermocline. As the thermocline becomes poorly defined (smears) the average outlet temperature delivered to the power cycle will decrease limiting powerplant performance and thus overall storage system efficiency. The relationship between storage system effectiveness and NTU_{fluid} can be quantified specifically for power-cycle applications by using a second law efficiency defined as follows:

$$\eta_{2nd\ law} = \frac{\int (\Psi_{out,discharge} - \Psi_{in,discharge})\, dt}{\int (\Psi_{in,charge} - \Psi_{out,charge})\, dt} \quad (11)$$

where, Ψ is the specific availability (energy) of the fluid defined as:

$$\Psi = m_f (C_f (T_f - T_o) - T_o C_f LN (T_f/T_o)) \quad (12)$$

where, T_o represents the dead-state temperature.

The second law efficiency here represents the ratio of the availability retrieved and stored from the system during charge and discharge cycles of equivalent length. Table 2 shows the relationship between system second Law efficiency and NTU_{fluid} for the test case with a sufficient number of charge and discharge cycles to reach a cyclic steady state.

Cyclic steady-state is defined when, for some periodic forcing functions, the second law efficiency approaches a constant value for consecutive charge discharge cycles. The cyclic steady-state second law efficiency is not obtained from the initial cycles because of transient behavior imposed by the system's initial conditions.

The NTU_{fluid} values encountered in thermal storage applications here are much larger than NTU_{fluid} values typically seen in heat exchanger because of the massive amount of fluid solid interfacial surface area. The second Law efficiency approaches one as NTU_{fluid} becomes large provided the storage tank capacity is not exceeded.

Table 2: Storage system energetic (2nd law) efficiency as defined in Eq. 11

Second law efficiency	NTU_{fluid}
0.70	0
0.75	0
0.80	10
0.85	20
0.90	40
0.95	100
1.00	900

CONCLUSION

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