EFFECTS OF HEAT EXCHANGER CONFIGURATION ON SOLAR DOMESTIC HOT WATER SYSTEM PERFORMANCE

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Abstract

This paper presents an experimental comparison of two common heat exchanger configurations used in residential-sized solar water heating systems to evaluate the relative merits and effects of heat exchanger placement (in-tank immersed coil versus external thermosyphon heat exchanger) on storage tank temperature profiles. Experiments were conducted on a specialized test apparatus that allowed controlled charge sequences to be studied for the two configurations considered. The results of this study provide a significant insight into the operation of indirect solar systems configured with immersed coil, or external, heat exchangers and their effect on tank stratification levels. Test results were evaluated on both a First and Second Law Thermodynamic basis, illustrating differences in temperature profiles, energy and exergy levels occurring after identical charge sequences. Results of these tests indicate that, for the configurations studied, higher tank temperatures, stratification and exergy levels were achieved earlier in the charge sequence with the external thermosyphon heat exchanger than with the immersed coil heat exchanger. In addition, it was observed that charging of a partially stratified storage with an immersed coil can result in tank mixing, resulting in a reduction in the temperature and exergy level of the stratification layer.

1. Introduction

Solar hot water heating systems designed for year round use in many locations must include features that will prevent freezing of the collector fluid. A common configuration uses an anti-freeze fluid (e.g., a propylene glycol/water solution) to transfer heat from a roof-top solar array to a solar storage located in a heated space. A heat exchanger is used to transfer the solar heat to the potable water in the storage tank. Referred to as indirect, closed loop systems, two heat exchanger configurations are common. One configuration uses an immersed coil heat exchanger located at the bottom of the solar storage (SERI/TR-253-2866, 1986) and relies on buoyancy driven natural convection to distribute the solar heat through the thermal energy storage (TES) tank; the other, uses a heat exchanger external to the storage tank and is connected through a secondary circulation loop. Both of these configurations are illustrated in Fig. 1. Potable water circulation through the secondary loop may be achieved by thermosyphon flow (Cruickshank and Harrison, 2009) or by use of a second pump.

Each of these configurations will produce differing temperature profiles in the storage tank over a charge cycle. In particular, simple immersed heat exchangers will have a tendency to mix upper portions of the storage tank whereas external natural convection heat exchangers (NCHEs) will promote stratification in the storage tank during charging (Hollands et al, 1989).

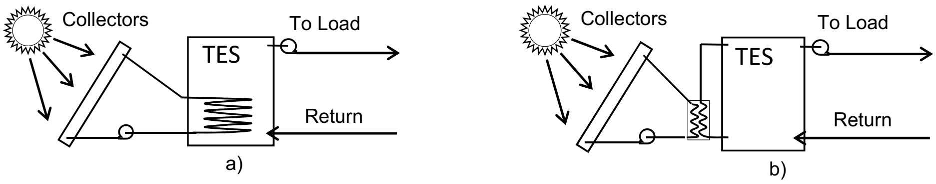


Fig. 1: Representation of an indirect system with a) an immersed coil heat exchanger, and b) with an external heat exchanger

Previous studies have indicated that there is a significant benefit (theoretically up to 37%) in annual solar energy delivery, due to the establishment and maintenance of thermal stratification in solar storage tanks

(Hollands and Lightstone, 1989; Haller et al., 2009). However, within the design categories mentioned above, differing levels of stratification may be achieved as a result of mixing of the storage tanks due to adverse temperature gradients and high circulation velocities due to buoyancy and momentum effects. These phenomena were investigated in this paper for two typical heat exchanger configurations through experimental measurements.

1.1 Study Objectives and Methodology

The primary objective of this study was to investigate the effects of heat exchanger configurations on heat transfer and temperature profiles in a typical thermal storage. To accomplish this, an experimental program was undertaken in a controlled laboratory environment. Two heat exchanger configurations were studied: 1) a spiral heat exchanger located in the storage tank approximately 15 cm from the bottom of the tank; and 2) an external, braised-plate side-arm heat exchanger connected from the bottom of the storage to the top through a thermosyphon natural convection loop. Both of these configurations have been studied previously by various authors and are typically used in commercially available solar domestic hot water heating systems in Europe and North America.

A secondary objective of this study was to quantify the relative benefits of the storage tank and heat exchanger configurations. To accomplish this, values of exergy stored versus time were determined for the test sequences studied. While the application of the First Law of Thermodynamics enables the determination of energy stored during a process (and the amount lost to the surroundings), the Second Law of Thermodynamics provides a mechanism for quantifying any degradation in the "usefulness" of the energy that occurs during the storing process. To accomplish this, both exergy level and exergy efficiency have been widely used to evaluate the performance of thermal energy storage systems (Dincer and Rosen, 2002).

1.2 Second Law Performance of a TES

Traditionally, exergy is considered as a measure of the "quality" of energy or its potential to do work relative to a reference or dead state, usually representative by the surrounding conditions. Applying the First and Second Laws of Thermodynamics to a control volume with uniform properties, the specific exergy of a substance, *Ex*, can be defined as,

$$Ex = (h - h_o) - T_o \cdot (s - s_o) \qquad \text{(eq. 1)}$$

where h and s are the specific enthalpies and entropies of the substance at its current temperature and pressure, and h_o and s_o are its enthalpies and entropies at a reference state. T_o is the temperature of the reference state.

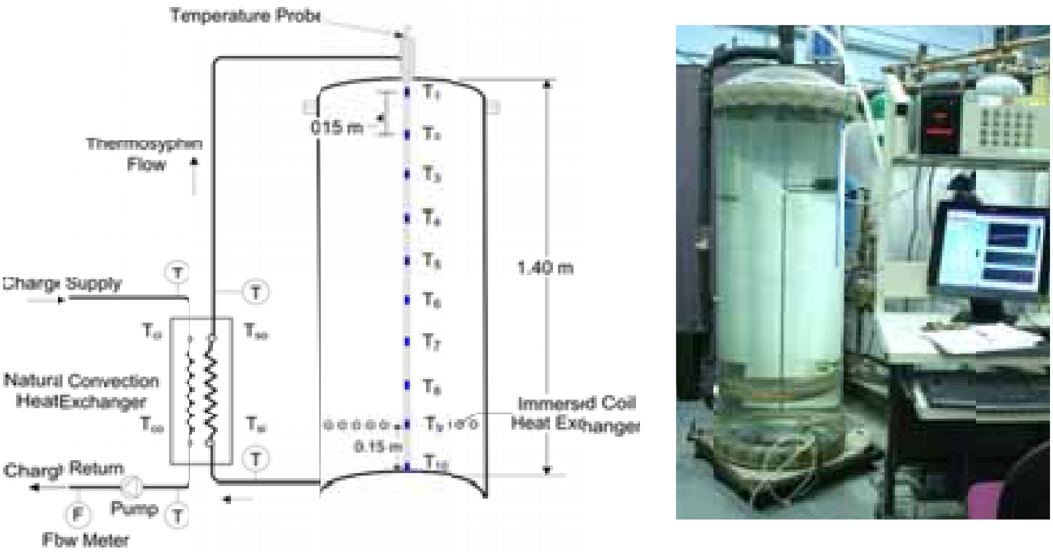
It is highly desirable to develop thermal energy storages (TES) that can store energy at its highest exergy level and to minimize the destruction of exergy associated with irreversible processes, (i.e., entropy production). In a thermal storage, consisting of an effectively incompressible fluid (i.e., water), exergy destruction will primarily occur due to mixing and diffusion occurring during the charging, storage and discharging processes. Exergy destruction during the storage of energy, over a period of time, occurs due to heat losses to the surroundings and the diffusion of heat through the fluid and the storage vessel. Exergy destruction also occurs during the charging and discharging of a thermal storage, however, as we are primarily concerned with the charging of the thermal storage, we will focus our analysis on that process.

Many indices have been proposed or are under development to quantify the Second Law performance of a TES system (Haller et al., 2008), however, the performance of a TES can be studied by observing the exergy level in the storage tank during the charging process. The exergy level of a stored fluid at any time is primarily related to its temperature relative to a reference state as determined through the equation described above. Simply put, to maximize the exergy quantity in a TES, it is important to maintain the fluid at as high a temperature as possible relative to the reference state. For solar heating systems, the temperature of the reference state may be the temperature of the surroundings or, in the case of an SDHW system, the temperature of the mains water supply. In addition, to avoid violating the First and Second Laws of

Thermodynamics, the maximum temperature in the thermal energy storage will be determined by the maximum temperature occurring during the charge sequence. As such, high exergy will be achieved during charging if the bulk of the volume of the thermal storage can be brought, as close as possible, to the temperature of the charge fluid. In addition, higher degrees of temperature stratification in a storage should reduce exergy destruction associated with mixing and diffusion and are highly desirable in a storage.

2. System Description

The experimental rig consisted of three parts: a heater (to simulate the solar collector array), a charge flow loop, and the storage unit and heat exchanger under study. A schematic of the heating-loop used to charge the storage system is shown in Fig. 2. The apparatus was instrumented, with a temperature probe inserted into the storage tank, allowing the tank temperature profile to be determined during simulated charge tests. The temperature profile in the storage tank was recorded at 0.15 m intervals with type "T" thermocouples. A computer based data acquisition system and a custom *National Instruments LabVIEW* routine was used to record and display storage and heat exchanger temperatures in real time. The temperature of the charge loop was controlled by a PID controller that adjusted the heat input and a positive-displacement pump was used to deliver hot fluid (50/50% by volume propylene glycol/water mixture) to the heat exchanger.



a) b)

Fig. 2: a) Schematic of test apparatus used to charge the storage system, and b) photo of the storage test rig

Two common heat exchanger configurations were chosen for this study, i.e., immersed coil heat exchangers and external side-arm natural convection heat exchangers. It is important to note that there are an infinite range of heat exchanger sizes and shapes that can be used for both configurations.

To undertake this study, a custom thermal storage was constructed and fitted with the two heat exchangers, Fig. 2. The geometry of the storage was set such that it closely resembled a standard 270 L North American residential water heater, however, it was completely constructed of clear acrylic plastic so fluid flow visualization studies to be conducted within the storage (however, this was not the focus of the current study). The tank was constructed such that the top could be removed to allow various custom immersed coil heat exchangers to be installed at the bottom of the storage. Specifications of the storage tank are given in Tab. 1.

Tab. 1: Storage tank specifications

Storage Tank	- Custom Fabricated Acrylic Plastic Hot Water Tank - Nominal Height = 1.4 m, Diameter = 0.55 m - Volume = 270 L - Integral Immersed-coil Heat Exchanger plus External Side-arm Heat Exchanger with Thermosyphon Loop

Since a suitable commercial immersed coil heat exchanger was not available at the time of the study, a custom unit was fabricated from refrigeration-grade tubing, Fig. 3. It was wound in a flat spiral and placed 0.15 m from the bottom of the storage tank, Fig. 2. Specifications of the immersed coil heat exchanger are given in Tab. 2.

An external side-arm, natural convection heat exchanger (NCHE) was also installed on the custom storage tank. This type of heat exchanger uses buoyancy-driven natural convection to circulate the water on the storage side, thus eliminating the need for one of the pumps, Fig. 3. The rate of circulation through the NCHE is governed by the state of charge of the storage tank (as indicated by its temperature profile) relative to the water-side, exit temperature of the heat exchanger. Specifications of the side-arm natural convection heat exchanger are given in Tab. 2.

Fig. 3: Immersed coil heat exchanger (left) and external plate heat exchanger (right)

Tab. 2: Heat exchanger specifications

Immersed Coil	- Refrigeration-grade Copper Tube Rolled into Flat Spiral - Length = 5.5 m, Outer Diameter = 0.0127 m - Wall Thickness = 0.001 m - Surface Area = 0.22 m^2
External Side-arm	- Compact, Brazed-plate Heat Exchanger - 20 plates, 0.31 x 0.07 m - Effective Heat Transfer Area = 0.396 m^2

3. Experimental Procedure and Analysis

For this study, only constant-temperature and constant-power input charge sequences were considered. During each of the test sequences, the water in the storage was preconditioned to a uniform start temperature of 20^{o}C. Heating was then initiated through either the side-arm NCHE or the immersed-coil heat exchanger and continued until the storage was effectively fully charged. During this period, the vertical temperature profile in the storage tank was measured, and the charge level of the storage was determined. In addition, to assist in evaluating the influence of temperatures on stratification levels, average storage exergy levels were determined. Thermocouples located at the inlets and outlets of each of the heat exchangers also allowed charge heat transfer rates (across the heat exchangers) to be determined. The use of the artificial thermal input to charge the storage tanks allowed specified test sequences to be repeated for both of the heat exchanger configurations studied, thereby allowing the test results to be directly compared.

The test sequences studied for each of the heat exchanger configurations included:

i) a constant temperature charge at 50^{o}C and heat exchanger flow rate of 1.5 L/min;

ii) a constant power input charge at 4.9 L/minute; and,

iii) a constant power input charge at 4.9 L per minute starting from a half charged, fully stratified storage condition.

The last test sequence (i.e., test sequence iii) was investigated to study a the case of a fully charged storage tank that had been partially discharged such that cold ”mains” water would have occupied the bottom of the storage while the top remained at a high temperature. This situation is of interest as it indicated what happens to the high-temperature portion of the storage as heating through each heat exchanger configuration was resumed.

3.1 Second Law Analysis

To evaluate the potential of a TES to achieve high stored exergy values, the experimental data was analyzed and comparisons were made with recognized limiting cases. Both constant temperature and constant power test cases were considered. In addition, both immersed coil and external heat exchanger configurations were studied under each test sequence. To evaluate the exergy at any point in time during the charge sequence, the value of the stored exergy was calculated, for each node within the tank based on the recorded temperature profiles. Individual node exergy values were determined through a separate routine implemented within EES (2009) using its library of thermophysical property functions for water. To estimate the exergy level in the storage tanks, values of exergy in each of the nodes within the storage tank were summed, throughout the test sequence, i.e.,

$$Ex_{\text{tank}}(t) = \sum_{\text{node}=1}^{9} Ex_{\text{node}}(t) \qquad \text{(eq. 2)}$$

4. Results and Discussion

The experimental results obtained during each of the three test sequences described above were analyzed and the results plotted in the Figs. 4 to 8. The data recorded during the constant temperature and constant power charge sequences were analyzed according to Eqs. 1 and 2. Specifically, the exergy level in the TES was determined at various intervals throughout the charge sequences.

Figures 4-A and 5-A compare the tank temperatures, cumulative energy level and the average value of specific exergy in the storage during the charge sequences. Temperature profiles for the constant temperature and constant power charges, i.e., Figs. 4-A and 5-A, plots (a-1), show that high stratification levels occur with the side-arm heat exchanger. In contrast, the results for the temperature distribution in the tank heated by the immersed coil heat exchanger, Figs. 4-A and 5-A, plots (a-2), show that the storage is effectively, totally mixed and at a uniform temperature throughout the charge. As expected, the temperature of the tank increases gradually during the charge, ultimately reaching higher temperatures at the end of the charge period. In both cases, as the average tank temperature increases, the heat exchangers' heat transfer rates decrease.

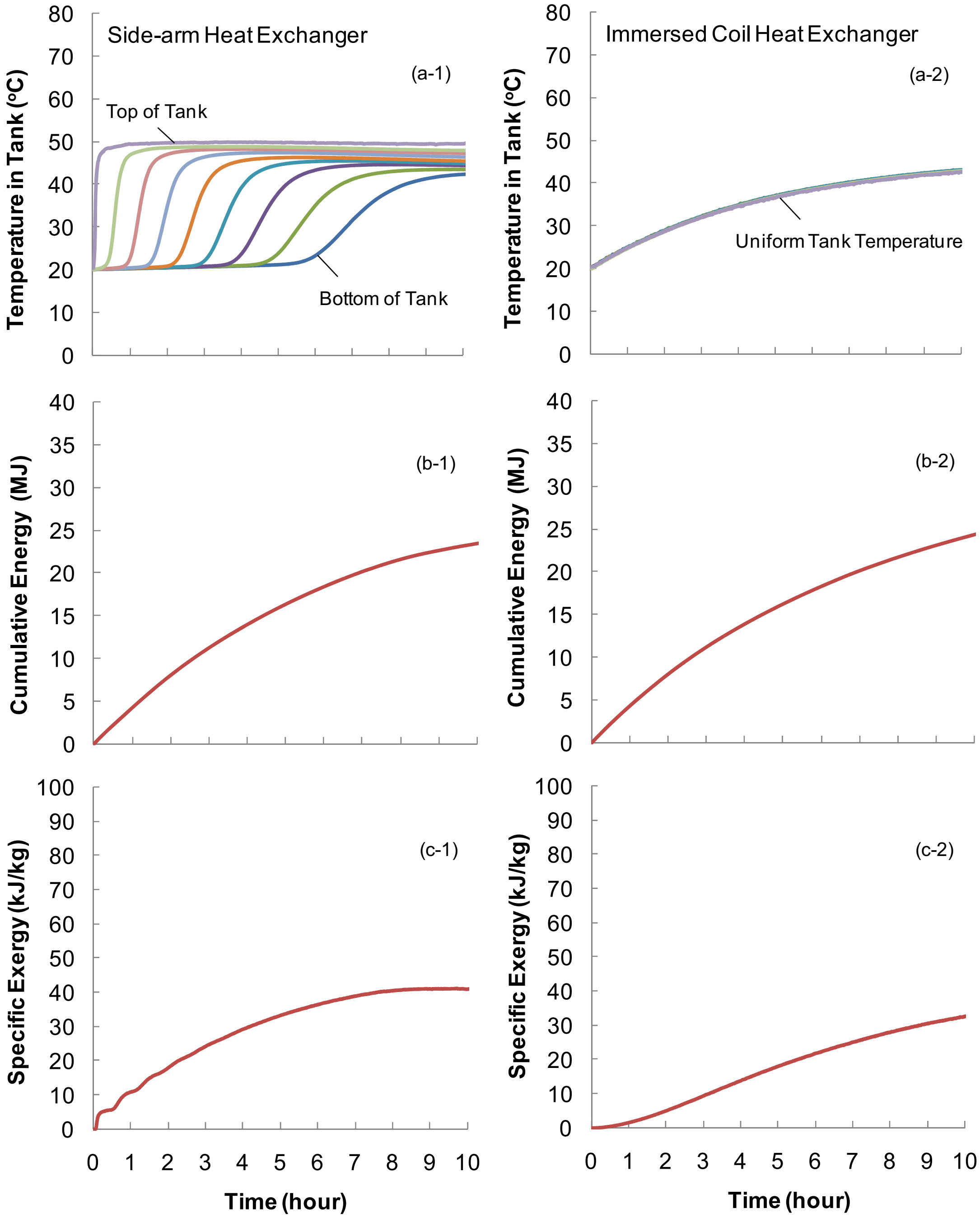


Fig. 4-A: CONSTANT TEMPERATURE TEST SEQUENCE (Set point 50°C, 1.5 L/min). Measured temperature distribution, cumulative energy and specific exergy for the immersed coil and side-arm heat exchanger configurations during a constant temperature charge.

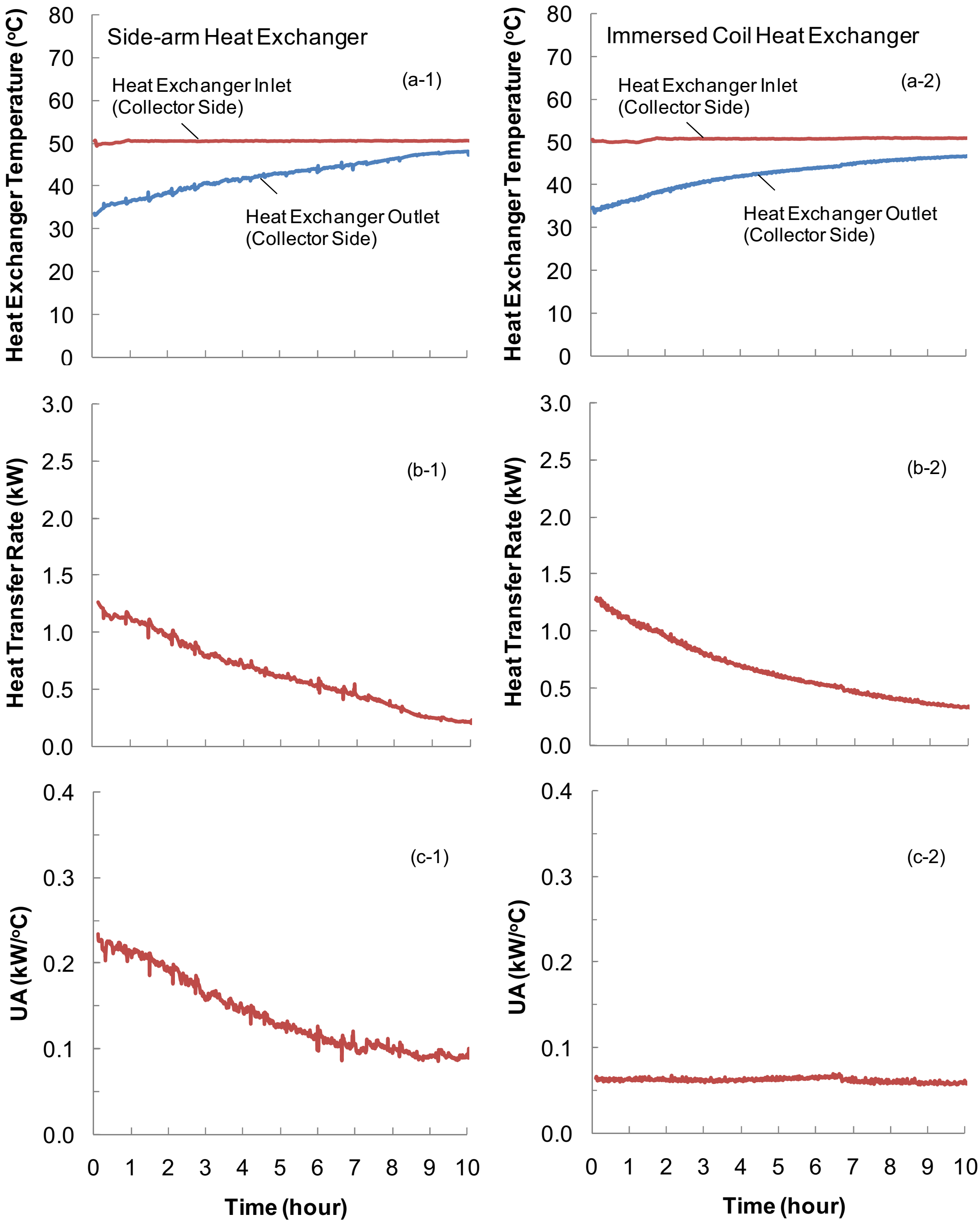


Fig. 4-B: CONSTANT TEMPERATURE TEST SEQUENCE (Set point 50°C, 1.5 L/min). Measured heat exchanger temperatures, heat transfer rates and UA values for the immersed coil and side-arm heat exchanger configurations during constant temperature charge.

Average UA values are also indicated in Figs. 4-B and 5-B, plots (c-1) and (c-2) for the immersed coil and side-arm heat exchangers. UA values for the side-arm heat exchanger are seen to drop over the course of the charge due to the reduction in the thermosyphon flowrate that occurs as the storage becomes fully charged (Cruickshank and Harrison, 2009). Results for the immersed coil heat exchanger indicate that, although the UA value remains constant throughout the charge, Figs. 4-B and 5-B, plots (c-2), the temperature difference across the heat exchanger drops due to the increasing average tank temperature. The result is that the net heat transfer rates are similar for both heat exchanger configurations.

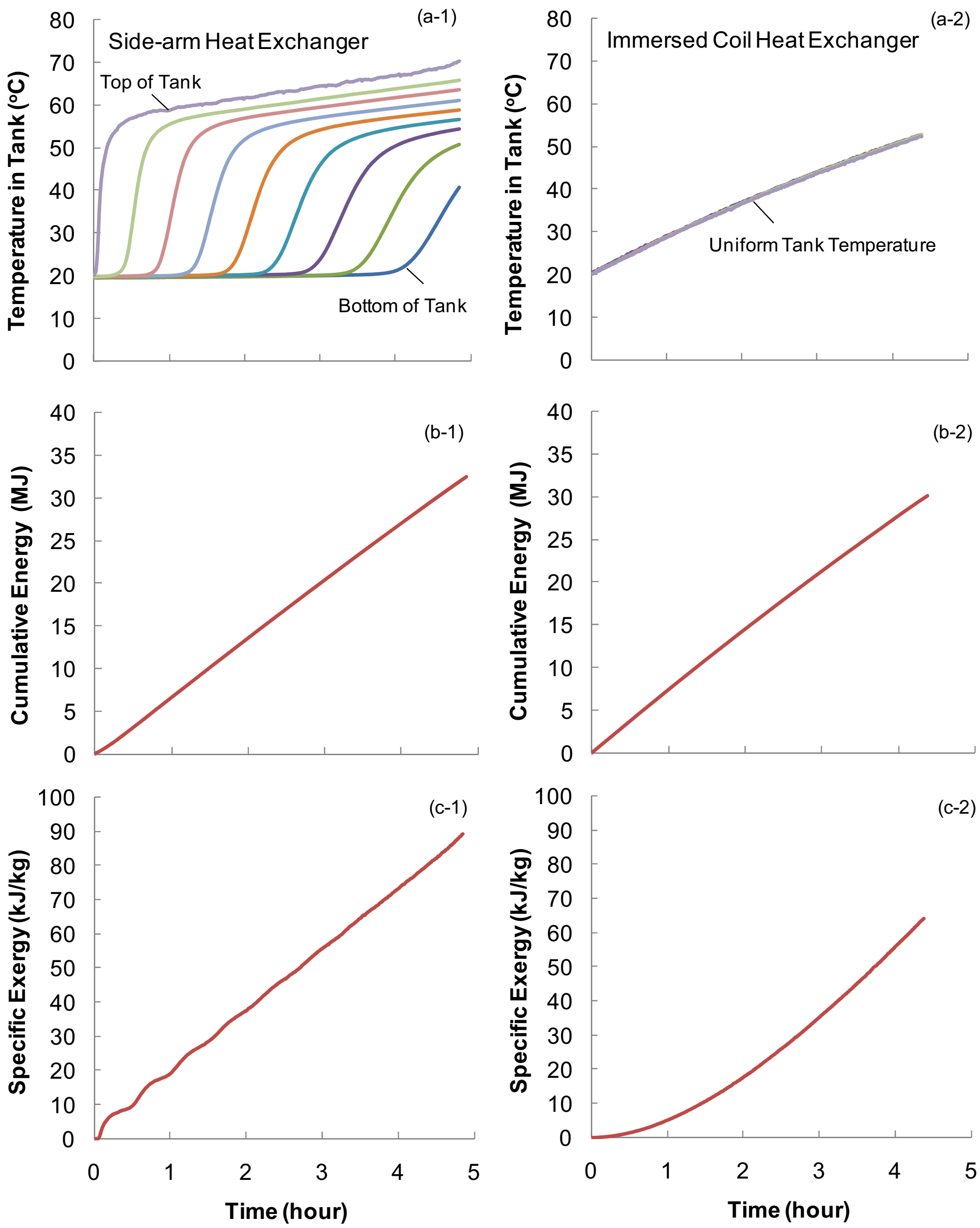


Fig. 5-A: CONSTANT POWER TEST SEQUENCE (4.9 L/min). Measured temperature distribution, cumulative energy and specific exergy for the immersed coil and side-arm heat exchanger configurations during the constant power charge.

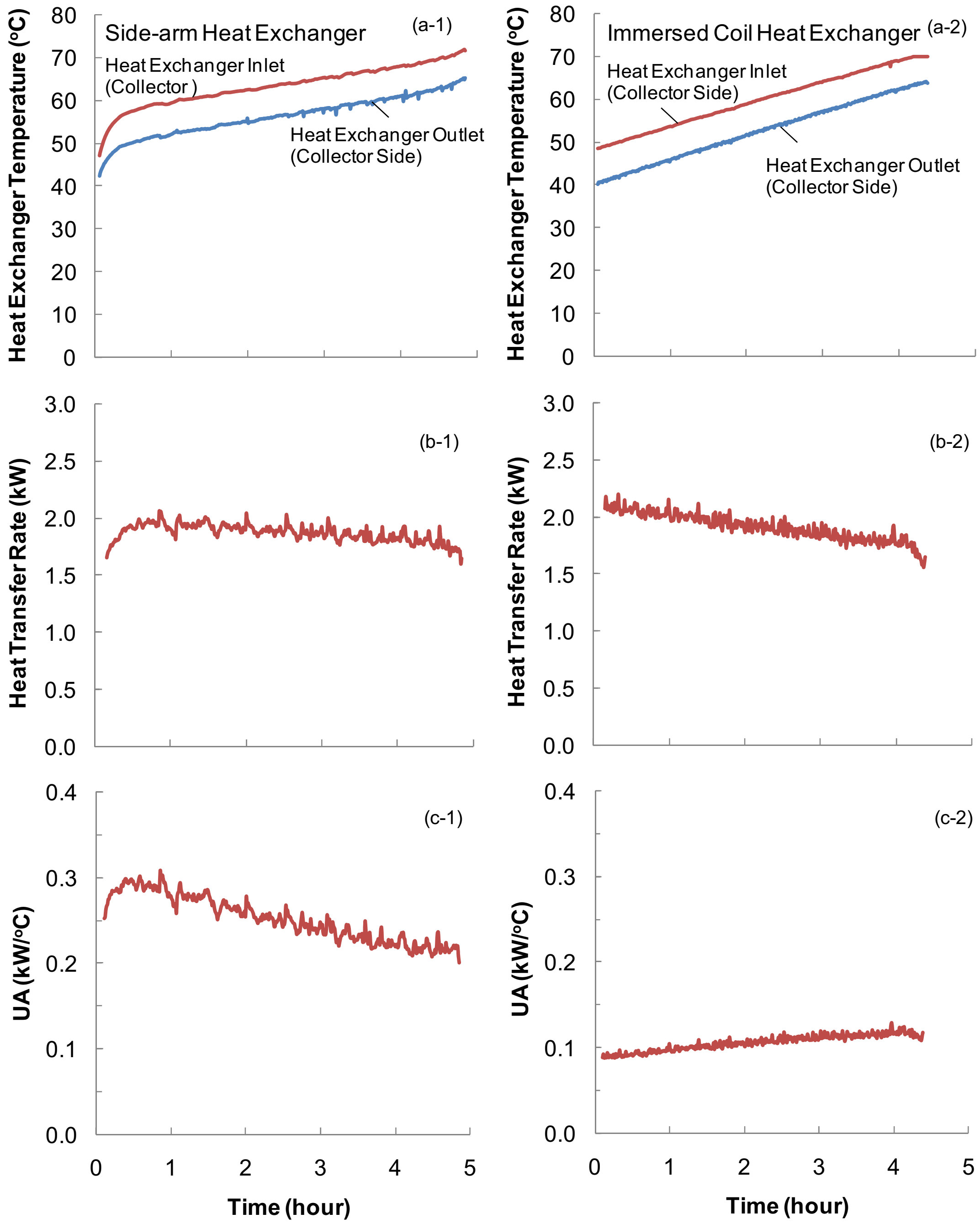


Fig. 5-B: CONSTANT POWER TEST SEQUENCE (4.9 L/min). Measured heat exchanger temperatures, heat transfer rates and UA values for the immersed coil and side-arm heat exchanger configurations during the constant power charge.

In Fig. 6, plots (a-1) and (a-2), the average, specific exergy in the storage tank is plotted throughout the charge sequence for each of the heat exchanger configurations studied. Figure 6, plots (b-1) and (b-2), indicate the difference in average specific exergy during the test sequences for each of the configurations. When these results are compared, it is apparent that, for the configurations studied, higher tank temperatures, stratification and exergy levels were achieved earlier in the charge sequence with the external thermosyphon side-arm heat exchanger than with the immersed coil heat exchanger.

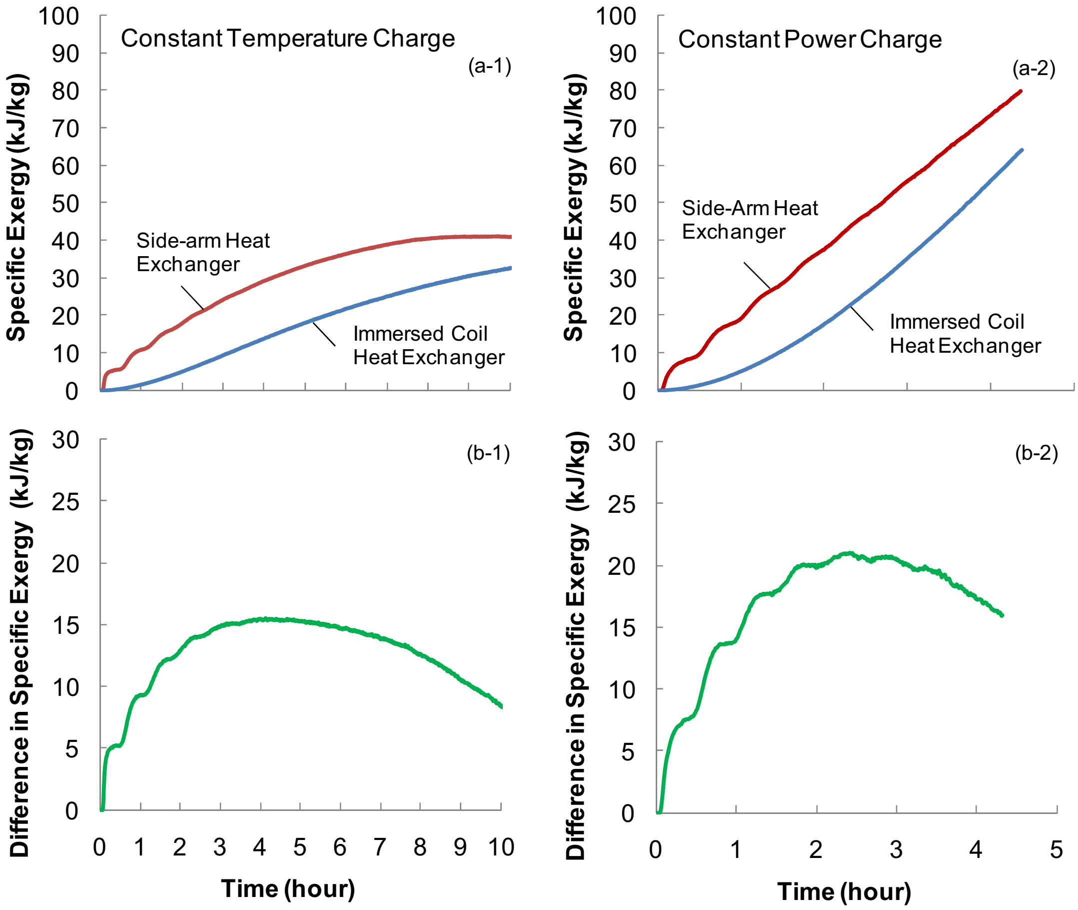


Fig. 6: CONSTANT TEMPERATURE AND CONSTANT POWER SPECIFIC EXERGY COMPARISON: (constant Temperature at low flow rate 1.5L/min (left) and constant Power at high flow rate 4.9 L/min (right).

Figure 7 shows the results for the last case studied that consisted of re-charging a half charged (or half discharged) storage tank (i.e., the top of the storage tank was initially at a temperature of 50° C and the bottom half was at 20° C. These results show that for the immersed coil heat exchanger, mixing of the top portion of the storage tank occurs during the heating process due to the momentum and velocity of the fluid plume created during the healing process. The observation figure 7 shows that the temperature level of the upper layers is briefly reduced prior to heating. In addition, Fig. 8 shows that the exergy level of the tank with the immersed coil heat exchanger is significantly lower than that of the side-arm heat exchanger.

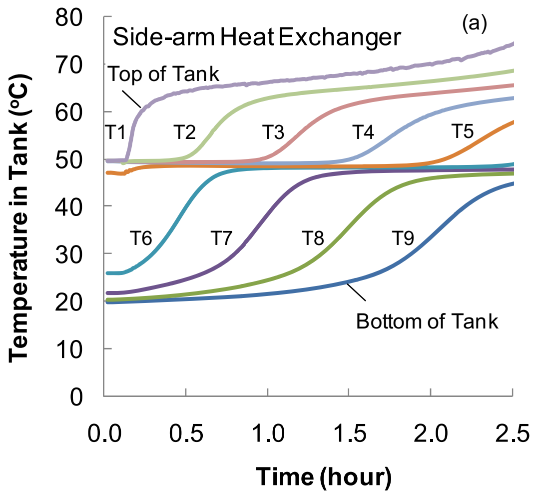


Fig. 7: CONSTANT TEMPERATURE AND CONSTANT POWER – HALF CHARGE TEST: a constant power input charge at 4.9 L per minute starting from a half charged, fully stratified storage condition

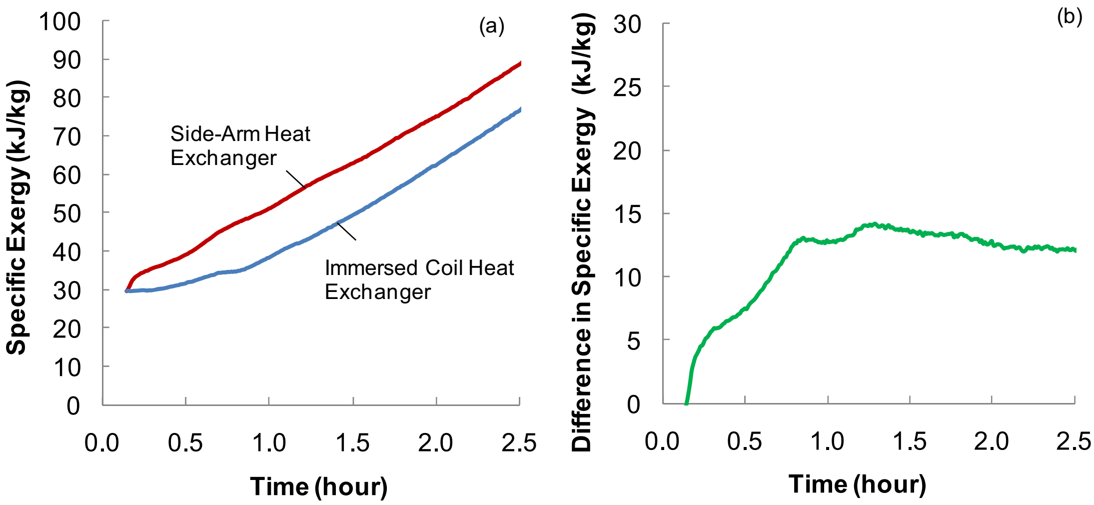


Fig. 8: Average exergy levels for the constant power recharge of a partially discharged storage, (4.9 L per minute) starting from a half charged, fully stratified storage condition

5. Conclusions

The results of these experiments show that differing levels of stratification may be achieved as a result of mixing of the storage tanks due to adverse temperature gradients and high circulation velocities due to buoyancy and momentum effects. For the configurations studied, higher tank temperatures, stratification and exergy levels were achieved earlier in the charge sequence with the external thermosyphon heat exchanger than with the immersed coil heat exchanger. In addition, it was observed that charging of a partially stratified storage with an immersed coil can result in tank mixing, resulting in a reduction in the temperature and exergy level of the existing higher temperature portion of the storage tank.

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