EVALUATION OF A STRATIFIED MULTI-TANK THERMAL STORAGE FOR SOLAR HEATING APPLICATIONS

by

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A thesis submitted to the Department of Mechanical and Materials Engineering
In conformity with the requirements for the degree of Doctor of Philosophy

Queen’s University
Kingston, Ontario, Canada
June, 2009

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Abstract

A novel multi-tank thermal energy storage (TES) was evaluated experimentally and numerically. The multi-tank storage is based on the interconnection of standard hot water storage tanks by a single charge flow loop. Each tank is charged through a thermosyphon loop and natural convection heat exchanger (NCHE). Both series- and parallel-connected configurations were investigated and results show that high degrees of stratification can occur.

To predict the performance of the series- and parallel-connected multi-tank TES, a numerical model was developed and implemented in the TRNSYS simulation environment. Laboratory tests were also conducted to measure the unit’s performance under charge conditions representative of combinations of clear and overcast days. The effects of rising and falling charge loop temperatures and power levels on storage temperatures and heat transfer rates were studied and indicated that sequential stratification was achieved in the series-connected storage.

Under certain conditions, reverse flow through the thermosyphon loops was identified, leading to destratification and carry-over of heat to the downstream storage tanks. Consequently, a new model was developed and showed to model reverse thermosyphon operation. A subsequent analysis showed that these effects could be minimized by careful system design.

To quantify the relative benefits of the sequentially stratified TES, values of exergy stored versus time were determined and compared against fully stratified and fully mixed storages. Results show that the series configuration closely matches the exergy level attained by a perfectly stratified storage.
Finally, annual simulations conducted for a typical multi-family installation showed that the multi-tank storage performed at a level comparable to a single, fully stratified, storage.
Acknowledgements

I wish to express the utmost gratitude to my supervisor, Dr. Stephen J. Harrison of the Department of Mechanical and Materials Engineering at Queen’s University, for his support, guidance and friendship. His enthusiasm and expertise in this area have provided me with a wealth of knowledge and experiences that are invaluable. I will always remember his encouraging words, “If we knew the answer, we wouldn’t call it research.” It was truly a pleasure to work with him and learn from him.

I would also like to express my appreciation to the Canadian Solar Buildings Research Network for funding this work. Without their support, this project would not have been possible. In addition, I want to acknowledge EnerWorks Inc. for providing the necessary equipment to conduct this study.

To my colleagues at the Solar Calorimetry Lab, past and present, I would like to say thank you for your friendship, assistance and encouragement. I wish you good luck with your studies and your future projects.

To my fiancé, thank you for your love, patience and understanding. Your encouraging words and ongoing support were always there at times when I needed them the most.

Last but not least, I thank my parents, sister and brother for their unconditional love, kindness, support and encouragement. From the start until the accomplishment of this manuscript, they have been my source of strength and courage. To them, I dedicate this thesis.
Statement of Originality

I hereby certify that all of the work described within this thesis is the original work of the author. Any published (or unpublished) ideas and/or techniques from the work of others are fully acknowledged in accordance with the standard referencing practices.

Cynthia Ann Cruickshank

June, 2009
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<th>Definition</th>
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<tr>
<td>$A_{c,\text{fluid}}$</td>
<td>Cross-sectional area of tank fluid (m$^2$)</td>
</tr>
<tr>
<td>$A_{c,\text{wall}}$</td>
<td>Cross-sectional area of tank wall (m$^2$)</td>
</tr>
<tr>
<td>$A_{c,i}$</td>
<td>Cross-sectional area of $i$-th node (m$^2$)</td>
</tr>
<tr>
<td>$A_{s,i}$</td>
<td>Surface area of $i$-th node (m$^2$)</td>
</tr>
<tr>
<td>$C_P$</td>
<td>Specific heat capacity (kJ/kg-K)</td>
</tr>
<tr>
<td>$C_{P_l}$</td>
<td>Specific heat of latent phase (kJ/kg-K)</td>
</tr>
<tr>
<td>$C_{P_s}$</td>
<td>Specific heat of sensible phase (kJ/kg-K)</td>
</tr>
<tr>
<td>$r_C$</td>
<td>Capacity ratio (-)</td>
</tr>
<tr>
<td>$C_{r_{\text{mod}}}$</td>
<td>Modified capacity ratio (-)</td>
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<tr>
<td>$F_S$</td>
<td>Fraction of the load met by solar energy (-)</td>
</tr>
<tr>
<td>$\Delta H_f$</td>
<td>Enthalpies of formation associated with phase change (kJ/mol)</td>
</tr>
<tr>
<td>$\Delta k$</td>
<td>Additional thermal conductivity term (W/m-K)</td>
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<tr>
<td>$\Delta P$</td>
<td>Pressure difference (Pa)</td>
</tr>
<tr>
<td>$\Delta T$</td>
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<tr>
<td>$\Delta x$</td>
<td>Center-to-center distance between nodes (m)</td>
</tr>
<tr>
<td>$\eta$</td>
<td>Annual Efficiency (-)</td>
</tr>
<tr>
<td>$E$</td>
<td>Energy (kJ)</td>
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<td>$E_x$</td>
<td>Specific Exergy (kJ/kg)</td>
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<td>$\varepsilon$</td>
<td>Heat exchanger effectiveness (-)</td>
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<tr>
<td>$\varepsilon_{\text{mod}}$</td>
<td>Modified heat exchanger effectiveness (-)</td>
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<tr>
<td>$h$</td>
<td>Enthalpy (kJ/kg)</td>
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<tr>
<td>$H_{\text{tank}}$</td>
<td>Height of Tank, (m)</td>
</tr>
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<td>$H_x$</td>
<td>Height of heat exchanger, (m)</td>
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<tr>
<td>$k_{\text{eff}}$</td>
<td>Effective thermal conductivity of tank fluid and wall (W/m-K)</td>
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<tr>
<td>$k_{\text{fluid}}$</td>
<td>Thermal conductivity of tank fluid (W/m-K)</td>
</tr>
</tbody>
</table>
\( k_{\text{wall}} \)  Thermal conductivity of tank material (W/m-K)
\( m \)  Mass of fluid in storage tank (kg)
\( m_i \)  Mass of node \( i \) (kg)
\( \dot{m} \)  Mass flow rate (kg/s)
\( N \)  Number of nodes ( - )
\( NTU \)  Number of heat transfer units ( - )
\( NTU_{\text{mod}} \)  Number of heat transfer units (modified) ( - )
\( Q_{\text{actual}} \)  Measured heat transfer across the heat exchanger (W)
\( Q_{\text{aux},i} \)  Rate of auxiliary energy input into node \( i \) (W)
\( Q_{\text{Load}} \)  Rate of energy removal to supply the load (W)
\( Q_{\text{Loss}} \)  Rate of standby heat loss from the storage tank (W)
\( Q_{\text{max}} \)  Maximum theoretical heat transfer across the heat exchanger (W)
\( q_{\text{fluid}} \)  Heat transfer by conduction through fluid (W)
\( q_{\text{total}} \)  Total heat transfer by conduction (W)
\( q_{\text{wall}} \)  Heat transfer by conduction through wall (W)
\( s \)  Specific entropy (kJ/kg-K)
\( t \)  Time (s)
\( T_{\text{env}} \)  Temperature of the environment (°C)
\( T_i \)  Temperature of the \( i \)-th node (°C)
\( T_{\text{mains}} \)  Temperature of mains water (°C)
\( T_1 \)  Temperature of inlet fluid to heat exchanger, collector-side, (°C)
\( T_2 \)  Temperature of outlet fluid to heat exchanger, storage-side, (°C)
\( T_3 \)  Temperature of inlet fluid to heat exchanger, storage-side, (°C)
\( T_4 \)  Temperature of outlet fluid to heat exchanger, collector-side, (°C)
\( U_i \)  Heat loss coefficient of node \( i \) (W/m² °C)
\( U_{\text{Loss}} \)  Tank heat loss coefficient to the environment (W/m² °C)
Chapter 1

Introduction

1.1 Background

Building energy use in the residential and commercial sectors accounts for 30% of Canada’s energy consumption; 70% of which is for space heating/cooling and water heating applications (Natural Resources Canada, 2005). Roughly the same percentage of end-use greenhouse gas (GHG) production can be attributed to these demands.

The use of solar energy to offset these loads will reduce the consumption of conventional fossil fuels and thereby reduce Canada’s GHG emissions. When considering solar energy, both photovoltaic (PV) and solar thermal conversion devices are readily available. While PV systems represent a promising source of electrical energy, solar thermal systems are ideally suited to offset heating loads. In fact, these latter systems are typically two to four times as efficient as photovoltaic systems and are available at approximately 1/3 to 1/2 of the cost. As well, with the recent concerns over global warming and the environment, there has been a substantial increase in the use of solar energy systems, as shown in Figure 1. Currently solar thermal technology represents
one of the most cost-effective ways to use renewable energy and to offset the use of fossil fuels.

However, as a time-dependent source of energy, solar energy is not always available when there is an energy demand. To make efficient use of the available resource, it is often necessary to store solar energy until it can be used to supply a particular load. Consequently, a storage system is particularly important for solar thermal systems as the availability of the solar resource varies over the day and season.

![Figure 1. Worldwide use of solar thermal energy (Weiss et al., 2009)](image)

To reduce cost and space requirements, thermal storages must have sufficient energy density, low energy losses and efficient charge and discharge characteristics. To arrive at an efficient storage configuration for a particular application, it is necessary to conduct a detailed analysis of the thermodynamics, heat transfer and fluid dynamics associated with that application.
1.2 Solar Thermal Energy Systems

The primary function of a solar thermal energy system is to convert solar energy directly into heat in an efficient manner for a specific application. Conceptually, a solar thermal system consists of a “solar collector” to capture solar energy and convert it directly to heat and a mechanism for transporting the collected heat to a location where it is needed to offset a thermal load. Typically, energy is removed from a solar collector by a heat transfer fluid that is heated as it is pumped through the device. The heated fluid is then transported to a thermal storage for use at another time or circulated directly to supply the thermal load. A simple conceptual diagram of a solar heating system is shown in Figure 2.

![Figure 2. Basic components of a solar thermal heating system.](image)

The efficiency\(^1\) of a solar heating system typically drops as the temperature of the system increases. This is largely due to the increasing heat losses from a system to the surroundings that result in less energy being delivered to the load.

---

\(^1\) Efficiency is defined as the solar energy delivered to the load divided by the solar energy incident on a solar collector’s surface over a time period.
As a consequence, the majority of solar thermal applications are for the production of low to medium temperature heat (e.g., 40 to 120 °C). One of the most widespread global applications is to heat potable water for domestic use. These systems tend to be small and relatively inexpensive. A solar hot water heating system for a typical household consisting of four individuals would have a solar collector array of 4 to 6 m² and a hot water storage of 150 to 300 L. A variety of different configurations have been developed worldwide. The most common configurations are described in Section 1.2.1.

1.2.1 Solar Domestic Hot Water Systems

Solar thermal energy systems have existed in a variety of forms for hundreds of years. The widespread use of solar energy to heat water for domestic consumption started in the early part of the 19th century, however, competition from low-cost fossil fuels reduced its popularity. It was not until the mid-1970s that it was reconsidered as an alternative to fossil or nuclear generated energy. During that period, a number of configurations were developed in an effort to lower costs and improve performance. The developments that followed were often driven by local climatic, regulatory and market conditions. As such, simple passive systems were developed for non-freezing, hot climates and were widely used in the Mediterranean, Middle East and Asia-Pacific regions. Different system approaches were used in Europe, Japan, North America and Australia to produce systems that were freeze protected. However, virtually all systems include: one or more solar collectors to capture and convert the sun’s energy into heat; a storage tank to store the available energy until it is required; and a circulation system to move a heat transfer fluid between the collectors and the storage tank.
In general, solar domestic hot water (SDHW) designs can be classified as passive or active, and as direct (also called open loop) or indirect (also called closed loop). Passive systems are self-pumping and regulating and require no power input to drive pumps or fans; while active systems usually incorporate one or more circulation pumps and an electronic controller. In a direct system, the potable water is circulated through the solar collectors to the storage, as opposed to, an indirect system where an intermediate fluid transports heat from the solar collectors to the storage.

Passive systems typically use thermally induced buoyancy forces to transfer hot fluid from the collectors to the storage tank or directly to the load, eliminating the need for pumps, controls or moving parts. The simplest of these designs is a “thermosyphon” system, Figure 3a), where the hot water tank is located above, or integrated into, the collector. Passive systems are generally more reliable and less expensive to operate than active systems since they do not need external power or electronic controls to function. Storage water is circulated through the solar collectors by natural convection when the sun shines. Fluid heated in the solar collectors becomes less dense and is displaced by cold, denser fluid coming in from the thermal storage. This buoyancy-induced flow continues until the sun sets or the storage becomes fully charged with warm fluid. In effect, a correctly configured thermosyphon system is self-pumping and self-controlling. Simple thermosyphon systems are widely used in warm climates and are particularly popular where low cost is important and reliable external power is not available. These systems, however, are not suitable for climatic regions where there is a potential of freezing temperatures. As well, in cooler climates if the storage is placed on a roof and
exposed to low outdoor air temperatures, it may be subject to high standby losses overnight.

Consequently, in North America and Europe, the majority of systems are the “active” or forced circulation type. In this configuration, a small pump is used to circulate a heat transfer fluid through the solar collectors thereby allowing the storage tank to be installed indoors in a heated space (i.e., reducing standby heat losses) and located below the collectors (e.g., in the basement). In a “direct” configuration, potable water is pumped directly through the solar collectors (Figure 3b) and drained back to the storage or down the drain (i.e., a “drain-back” or “drain-down” system) in the advent of freezing conditions. In a closed loop, indirect system, an antifreeze solution is circulated through the solar collectors to the heat exchanger where heat is transferred to potable water in a storage tank, Figure 4(a) and Figure 4(b).

Figure 3. Schematic of: (a) typical passive thermosyphon SDHW system and (b) typical active (i.e., pumped circulation) direct solar hot water system.
The heat exchanger at the storage tank can take many forms but can generally be classified into three categories, e.g., immersed coils, external (side-arm) or mantle types, as shown in Figure 5(a, b, c), respectively (Han et al., 2008). Immersed coil heat exchangers are generally located at the bottom of thermal storage tanks to take advantage of the greatest temperature differences between the solar heated fluid and the incoming potable water. These designs tend to produce uniform temperatures in the storage tank (i.e., unstratified temperature distributions; see Section 1.3.2) which is undesirable. SDHW systems configured in this way often use specialized (and more costly) thermal storage tanks fitted with special internal heat exchangers. Other elements (e.g., baffles, diffusers, etc.) may be added to the interior of the storage tank to promote stratification but these further increase complexity and cost (Davidson and Adams 1994a, Shaw and Furbo 2003, Jordon and Furbo 2004, Altuntop et al. 2005, Anderson et al. 2007, Chung et al. 2008).

Figure 4. Schematic of: (a) typical indirect “pumped” SDHW system and (b) an indirect SDHW system utilizing a natural convection heat exchanger (NCHE).
External heat exchangers represent flexible options for an indirect system as they allow standard (i.e., low cost) storage tanks and heat exchangers to be used. Most traditional solar water heaters used in North America are indirect systems with external heat exchangers and two pumps, one to circulate an anti-freeze solution (typically a 50/50% by volume propylene glycol/water mixture) through the solar collectors and the other to circulate potable water from the storage tank through the heat exchanger, Figure 4(a). These systems are often easier to retrofit than passive systems as the storage tanks do not need to be installed above or near the collectors and are usually located in a heated space. Although active systems are usually more expensive than passive systems, they are generally more efficient due to their reduced standby losses.

Mantle heat exchangers consist of a double walled storage tank that allows the heat transfer fluid to be circulated through the storage mantle (i.e., a cavity formed by the two walls) transferring heat to the stored water. Mantle tank storage systems tend to have a large heat transfer surface area that increases performance but requires specialized tanks that may be more costly. Used in Denmark and Australia, mantle tank SDHW systems have been extensively studied (e.g., Rosengarten et al. 1999, Andersen and Furbo 1999, Knudsen 2002).

Figure 5. Different heat exchanger concepts used in solar hot water heating systems (Han et al., 2008).
A configuration that has been promoted over the last decade in North America combines the features of the external heat exchanger with the simplicity of the thermosyphon system, Figure 4(b). In this indirect, closed-loop configuration only one pump is used to circulate fluid in the solar collector closed loop. On the other side of the heat exchanger, the circulation of potable water through the heat exchanger is driven by natural convection in a matter similar to the traditional thermosyphon system. This hybrid configuration offers a number of unique features in that the solar collectors can be located anywhere relative to the storage tank, and the fact that the system only requires a single pump, rather than two, lowers cost and increases reliability (by eliminating a potential failure point). The buoyancy driven flow is also self-regulating and controlling in a similar fashion to the traditional thermosyphon system.

Under certain conditions, however, there is the potential for adverse temperature gradients to occur such that the flow direction may reverse resulting in partial discharging of the thermal storage. This condition, referred to as “reverse thermosyphoning”, may occur during standby periods (i.e., non-solar input periods) due to heat loss from the heat exchange loop, or during periods when cooler fluid is circulated through the heat exchanger thereby removing heat from the heat exchange loop.

Numerous studies have been completed in the past to optimize the performance of this type of system and, in particular, to characterize the performance of the natural convection heat exchangers (NCHEs) (Purdy et al. 1998, Lin et al. 2000, Cruickshank and Harrison 2006a). Commercially available for a number of years, this configuration has led to increased reliability, cost-effectiveness and improved thermal performance due to its potential for increasing thermal stratification in the storage tank. Thermal
stratification allows cool fluid to circulate to the solar collectors while maintaining hot water at the top of the storage tank for distribution to the load.

Studies conducted over the past two decades have also led to the development of the “micro-flow” concept (Hollands and Lightstone, 1989). These studies have shown that lowering the collector loop fluid flow rate further improves the thermal performance and the cost effectiveness of the overall system. A micro-flow system requires less pumping power and a smaller pump which reduces the cost of the system and parasitic power consumption. Lower flow rates result in smaller diameter piping and insulation levels which reduce installation cost and time. More importantly however, a low flow system further enhances thermal stratification in the storage tank and improves the performance of the entire system. That is, in a correctly designed micro-flow system, hot water will exit the heat exchanger at a low flow rate and enter the storage tank with a low velocity, allowing the hot water to remain in a layer at the top of the tank; the low flow, in turn, promotes a steeper thermocline (this topic will be discussed in more detail in the Section 1.3.2) making more of the energy in the tank available at a temperature closer to the desired load temperature (i.e., with a higher exergy) and allowing cool fluid to circulate back to the solar collectors. As a result, more solar energy is delivered to the load and a higher solar fraction\(^2\) is achieved.

1.3 Solar Thermal Energy Storage

Thermal energy storage (TES) is an integral component of a solar hot water system that may significantly improve its efficiency and cost effectiveness by allowing

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\(^2\) Solar fraction is described as the ratio of solar heat yield to the total energy needed for hot water heating.
better utilization of the solar hardware and the matching of the solar resource to the load. For example, consider an idealized solar process with storage over a 3-day period, Figure 6 (Duffie and Beckman 2006). As indicated by the vertical shaded area, the collector useful gain (available energy) exceeds the load on the 1\textsuperscript{st} and 3\textsuperscript{rd} day. This excess energy is then held in the storage subsystem for use at a later time. When required, the excess energy is withdrawn from the storage to meet the load (as represented by the horizontal shaded area). If there is insufficient thermal energy in the storage, the load is met by an auxiliary energy source.

The optimum choice and sizing of a thermal storage depends on many factors, including the distribution and temperature of the energy supply; the temperature requirements, magnitude and distribution of the load throughout the day or the season; the required charge and discharge rates; and the spatial limitations related to the installation and placement of the storage.

![Figure 6. Application of thermal storage in a solar heating system.](image)

Most small to medium sized solar installations use diurnal storage, where energy is typically stored for one or two days, however weekly and seasonal storage is also used in certain applications. Primarily used to offset space heating loads, seasonal storage systems are designed to collect solar energy during the summer months and retain the
heat in the storage for use during the winter months (Fisch et al., 1998). Characterized by their large capacity requirement (in the order of a hundred times the capacity of a daily storage) (Dincer and Rosen, 2002), these systems typically run at a much higher cost and require a larger storage volume than short term storages. Seasonal storages have been pilot-tested and used in a number of countries for district space and water heating (Fisch et al., 1998). Existing seasonal storage systems have been shown to meet close to 100% of annual building heating needs by solar (Dincer and Rosen, 2002). Seasonal thermal storage systems may take on several physical configurations including underground aquifers or large pools (Carotenuto et al. 1990, Kangas and Lund 1994, Paksoy et al. 2000) or bore-hole buried earth storage (Breger et al. 1996, Reuss et al. 1997, Bernier 2001). Because of their physical magnitude and cost, they are often custom engineered and constructed accounting for the specific load, climatic locations and soil conditions. Large scale thermal storage of this type is not the focus of the current study.

Diurnal or short term storage is often more suitable for small-scale domestic applications. Diurnal storage is designed to store heat for up to a few days and generally consists of smaller devices that are typically manufactured and assembled off-site and subsequently installed within a building. Although short term TES systems will seldom contribute a solar fraction in excess of 60%, they can operate on a competitive cost basis with conventional fuels (Dincer and Rosen, 2002).

1.3.1 Energy Storage Processes

Sensible Heat Storage. Thermal energy can be stored in a variety of ways. The most common method is to heat a substance, increasing its temperature, thereby storing the heat as internal energy within the material. If there is no change of phase or chemical
composition associated with the heating process then the process is considered to be one of sensible heat storage. Aside from the heat transfer processes associated with the addition or removal of heat from the substance, the amount of heat, $E$, that can be stored in a sensible heat storage is directly proportional to the specific heat, $C_p$, and mass, $m$, of the material and the temperature range associated with the process, $\Delta T$, i.e.,

$$ E = mC_p\Delta T $$

For this reason, to increase energy density, solids (e.g., rock, concrete, etc.) and liquids (e.g., water, glycol, etc.) of high mass and specific heat are usually considered for sensible heat storage. A high energy density (i.e., storage capacity per unit volume) is important as it allows the storage to be compact thereby reducing the cost of the storage vessel and its installation. A smaller surface area will also result in lower standby thermal losses from the storage to the surroundings, reducing insulation requirements.

The choice of storage medium is often influenced by the working fluid in a solar heating system. Thus, if air is the heat transfer fluid used in a solar system, then a rock bed thermal storage is an obvious choice (Duffie and Beckman, 2006). If the primary working fluid is a liquid, then a liquid is usually used as a storage medium. It has been noted that water is an excellent storage medium for the low-to-medium temperature range because of its high volumetric heat capacity, low cost and widespread availability. Consequently, thermal storages consisting of tanks of water are widely used. Water-based thermal storages are described in Section 1.3.2.

**Latent or Phase Change Heat Storage.** Latent heat storages primarily use the energy absorbed or released during a change in phase (e.g., water/ice, salt hydrates, etc.) that occurs at a particular temperature. For each storage medium, there is a wide variety of
choices depending on the temperature range and application. When utilizing a material that undergoes a change of phase, the total energy stored over a particular temperature range is related to the specific heats of the material in the various phases and latent heat (i.e., transition enthalpies, $\Delta H_f$) associated with the phase change (NATO, 1976). Thus the energy stored in a material undergoing a temperature change from $T_1$ to $T_2$, involving a change of phase is given by

$$E = m \left[ \int_{T_1}^{T_f} C_p(s) (\Delta T) dT + \int_{T_f}^{T_2} C_p(l) (\Delta T) dT + \Delta H_f \right]$$

(1.2)

where $T_f$ is the temperature at which the phase change occurs, $\Delta H_f$ is the enthalpy associated with the change of phase, and $C_p(s)$ and $C_p(l)$ are the specific heats associated with the two phases. The energy associated with a change of phase can be several orders of magnitude greater than that associated with a change of internal energy associated with sensible heat storage.

Consequently, the feasibility of storage systems using phase change materials (PCM) has been investigated by many researchers (Abhat 1983, Kaygusuz 1995) because of their potential to increase energy density, allowing large amounts of energy to be stored in small volumes or for their use as an additive to building materials to enhance the performance of passive solar buildings (Pasupathy et al. 2008, Kuznik et al. 2008).

To develop a reliable PCM storage, the latent heat effect must be reversible over a large number of cycles without degradation of the material or its container. A suitable PCM material must maintain good thermal properties in both phases while meeting the requirements of low corrosion, toxicity and volume change. To date very few latent heat
storages have been widely used or are commercially available, however they are the source of considerable research (Zalba et al. 2003, Sharma et al. 2008).

**Chemical Storage.** The concept of chemical storage makes use of chemical reactions in substances to store energy. As with phase change materials, few practical, commercially viable, chemical storages have been developed. A suitable thermo-chemical reaction for an energy storage would be an endothermic reaction resulting in reaction products that are easily separated and do not undergo further reactions. Ideally, the products of the chemical reaction could be recombined and the reaction reversed to permit recovery of the stored energy (NATO 1976, Duffie and Beckman 2006). Unfortunately, there are no obvious candidate materials for this type of reaction at low temperature and they will not be considered further as they are outside the scope of this investigation.

1.3.2 Water-based Sensible Heat Thermal Storage

Water-based storage systems have been widely studied and applied in a variety of configurations. As a working fluid, water is readily available at little or no expense. Much effort has been put into maximizing the performance of water storages and minimizing the cost of the storage vessels. As well, small hot water heaters and storages (i.e., 180 L and 270 L) have been produced in large quantity for the North American Market and are readily available at low cost. Coupled to an external heat exchanger as shown is Figure 4(b) they represent a cost effective storage option for residential solar hot water heaters. However, larger storage volumes in the range of 500 to 1500 L, are often required for multi-family residential units and small to medium-sized commercial applications. Unfortunately, suitable storage vessels of this size are only produced in
limited quantities, resulting in significantly higher costs per unit of storage volume (Cruickshank and Harrison, 2006b). In addition, these larger storage vessels are not well suited to retrofit situations where the storage vessel must be moved into a building space through existing door openings. Consequently, larger storages are often constructed on site, and maintained at low pressure and vented to the atmosphere. This may pose a performance and health risk to the occupants.

**Thermal Stratification in Liquid Storage Tanks.** An important aspect related to the performance of a TES, and solar thermal systems, is thermal stratification. It is the existence of a temperature gradient in the storage that allows the separation of fluid at different temperatures. When observing the temperature distribution in a real tank, one concept used to characterize the level of stratification within a storage is to quantify the temperature gradient \( \frac{dT}{dx} \) and thickness of the thermocline (intermediate region) that separates the hot and cold regions within the storage (Dincer and Rosen, 2002). This concept is illustrated in Figure 7(a-c) where three storage tanks with differing stratification levels, but containing equivalent energy, are illustrated.

![Figure 7. Differing levels of stratification within a storage tank with equivalent stored energy (a) left, highly stratified, (b) center, moderately stratified and (c) right, showing a fully mixed, unstratified storage.](image-url)
In Figure 7(a), the temperature gradient between the hot and cold regions of the storage is observed to be large and the thickness of the thermocline small. In Figure 7(b), the temperature gradient is smaller (i.e., \( \frac{dT}{dx}_b < \frac{dT}{dx}_a \)) and the thickness of the thermocline is larger than the storage shown in Fig. 7(a). In effect, the storage shown in Figure 7(a) is more stratified than the storage shown in Figure 7(b). Finally, in Figure 7(c), the storage is at a uniform temperature and is observed to be unstratified.

Numerous studies have been conducted to quantify the effects of stratification on the performance of solar hot water systems (Lavan and Thompson 1977, Phillips and Dave 1982, Hollands and Lightstone 1989, Cristofari et al. 2003) and have reported performance improvements of up to 37 percent depending on the nature of the load (Hollands and Lightstone, 1989). Besides the obvious thermodynamic benefit of maximizing the temperature and exergy (van Berkel et al. 1997, Rosengarten et al. 1999, Rosen et al. 1999, Shah and Furbo 2003) in a thermal storage of a solar hot water system, a stratified storage has the benefit of delivering hot water to the load early in the day (thereby reducing auxiliary heater input) and returning the coldest fluid back to the solar collectors. As the efficiency of solar collectors reduces at higher temperatures, this latter feature contributes to improved energy collection and solar system efficiency. As such, it is highly desirable to develop storage systems that promote high levels of thermal stratification. The quantification of stratification level and the determination of the factors that promote or destroy stratification (i.e., mixing) in storage tanks are also of principal interest.
1.3.3 Multi-tank Thermal Storage

As the popularity of solar domestic hot water systems increases, they are increasingly being considered for use in multi-family residential and small commercial applications (e.g., motels, restaurants, laundromats, etc.). These larger systems (e.g., 500 to 1500 L) normally require larger storage vessels that are significantly more expensive than standard residential units and that must be constructed on site, increasing installation costs.

As an alternative to a large single tank, storage systems consisting of interconnected single small tanks, Figure 8, have been investigated (Bejan 1982, Taylor and Krane 1991, Arata and de Winter 1991, Sekulic and Krane 1992a,b, Mather et al. 2002, Tacchi 2003). Constructed of small, prefabricated, inexpensive tanks, they may be easily transported into a building for interconnection. Highly modular, the multiple storage tanks can be interconnected in a variety of ways to achieve the desired flow characteristics and storage capacity. In addition, researchers have shown that by connecting individual storage tanks in series it is possible to achieve high levels of stratification in the storage system, reducing entropy production and improving overall system performance (Bejan 1982, Taylor and Krane 1991, Sekulic and Krane 1992a,b). When connected, these tanks form a single hydraulic unit that can be configured in a variety of arrangements (e.g., series vs. parallel).
Figure 8. Multi-tank thermal storage concept shown with three individual storage tanks connected in series.

Although the storage configuration shown in Figure 8 is an open loop, direct type, other multi-tank arrangements have been studied including closed loop, indirect, with and without immersed heat exchangers (Mather 2002, Cruickshank and Harrison 2006b). Previous studies have also shown that it is possible to achieve sequential stratification between storage tanks even if the individual storages are themselves mixed. Relevant research on multi-tank thermal storage is discussed in the literature review.

Of particular interest and the focus of this thesis is a specific configuration of multi-tank thermal storage that incorporates natural convection heat exchangers on each of the individual storage tanks, Figure 9 (Cruickshank and Harrison, 2006b). This indirect arrangement, described in Chapter 4, has many unique features that are of potential benefit to the operation and performance of medium-sized solar systems.
1.4 Problem Definition

To maximize the benefits of solar energy systems, effective energy storage devices are required. Their detailed design and operation significantly affects the operation of both solar heating and cooling systems. In particular, storage devices must minimize energy losses, facilitate charging and discharging, and ensure the optimum operation of the solar energy collection and distribution systems (Dincer and Rosen, 2002).

Worldwide, the solar domestic hot water market is well developed, however a significant but yet unexploited market segment exists for small to medium sized systems to supply multi-family or small commercial applications. Consequently, there is increasing interest in developing cost-effective configurations for thermal storage.
particular there is a need to develop an effective storage system that is modular, flexible, cost-effective and easy to install, especially in retrofit situations. The topic of this thesis addresses the design, operation and evaluation of such a system.

1.4.1 Objectives

The objectives of this study were to:

1. design an innovative, low-cost, multi-tank thermal storage for multi-family residential or small commercial use in the capacity range from 500 to 1500 L based on commercially available hot-water heater tanks and standard natural convection heat exchanger units;

2. construct, instrument and commission a full-scale, multi-tank test apparatus (810 L) to allow experimental data to be obtained under a variety of operational conditions;

3. develop data analysis routines to process the experimental data in a form suitable for presentation and comparison with modeled results;

4. develop a computer model of the operation of the multi-tank storage unit using the TRNSYS simulation software package and develop new purpose-specific component routines as required;

5. compare experimental and modeled results in order to refine and verify the accuracy of the numerical model to adequately model the performance of the multi-tank storage system under a range of test conditions;

6. assess the stratification potential of a multi-tank thermal storage unit;

7. analyze the impact of operational conditions (e.g., charge conditions) on the thermal stratification and system performance; and

8. illustrate the use of the computer model (item 4) by applying it to a typical installation.
1.4.2 Contribution of Research

This work has:

1. resulted in significant improvements in the cost performance, ease of installation and thermal performance of storage systems for solar heating applications;
2. increased the technical and economic feasibility of using solar energy for multi-family residential and small commercial applications;
3. furthered the widespread use of renewable, solar energy that will reduce fossil fuel consumption and reduce harmful greenhouse gas emissions;
4. demonstrated the operation of a unique multi-tank thermal storage design through testing and simulation;
5. determined the key design and operational factors that affect the performance of the investigated multi-tank storage system;
6. developed and verified the accuracy of a computer model implemented in the TRNSYS simulation environment for the multi-tank storage investigated;
7. produced a preliminary assessment of the feasibility of the multi-tank TES in a typical installation; and
8. illustrated the effects of load and climatic conditions on the performance of the multi-tank TES (using the validated computer model).

1.5 Experimental and Computational Methodology

The performance of a modular thermal storage for use in solar heating systems was experimentally and numerically studied (Cruickshank and Harrison, 2006c). To conduct this evaluation, an experimental rig was constructed and instrumented allowing the time/temperature history within the storage to be recorded (Cruickshank and Harrison, 2006b). The apparatus allows prototype storage systems to be evaluated under a range of charge and discharge conditions and is plumbed such that a variety of physical configurations can be evaluated (i.e., storage tanks may be connected in series or parallel
configurations for both charge and discharge cycles). Solar input is simulated through the use of electric heaters that allow the power output to be varied according to prescribed sequences. The apparatus is fully instrumented, with temperature probes inserted into each storage tank, allowing stratification levels to be determined. A computer based data acquisition (DA) system records the storage temperature profiles and heat exchanger temperatures in real time. Experimental test sequences were modeled using TRNSYS and the results compared.

1.5.1 Parameters Studied

As previously described, a number of parameters were studied under this project. Although the work of this project focuses around a particular configuration of multi-tank thermal storage, the test apparatus was designed to allow for re-configuration of the system. For example, through an arrangement of valves and piping, the storage system can be operated in both series and parallel configurations for both charge and discharge. The addition of the computer-based data acquisition and control system allows various charge profiles to be investigated. In particular, this included investigations and sensitivity analyses on:

1. the magnitude and distribution of the heat input to the thermal storage, which is representative of the number of solar collectors, and the weather conditions;
2. the flow rate and temperature level of the charge sequences;
3. the magnitude of the effect of reverse thermosyphon flow and carryover on the energy stored during typical daylong charge periods; and
4. the magnitude of heat loss from the energy storage tanks.
A test program was undertaken to collect data to verify the accuracy of the numerical model used to describe the performance of the thermal storage system. Experimental data was also analyzed to characterize: the performance of the natural convection heat exchangers; the level of thermal stratification in the storage tanks; and the charge and discharge characteristics of the storage (i.e., thermal diode aspects).

The development and refinement of a numerical model for this type of multi-tank thermal storage within the TRNSYS simulation platform will allow a variety of aspects to be investigated in the future, including the capacity and placement of heat exchangers, the level of thermal insulation, and the magnitude of the heat input and extraction from a multi-tank thermal storage.

1.6 Organization of Research and Thesis Document.

The information presented in this thesis documents research conducted over a span of four years. Over this period, most of the material was published in peer-reviewed journals and conference papers. The thesis document represents a compilation of results presented in these papers. These papers were referenced throughout the thesis.

The approach taken during the course of this research focused primarily on an experimental evaluation of the full-scale thermal storage apparatus. Considerable effort was undertaken to design, construct, instrument, commission and calibrate the apparatus.

The experimental program was supported by a parallel activity related to the modeling of the multi-tank thermal storage under test. A detailed work plan (flowchart) is shown in Figure 10 that outlines the activities undertaken during the course of this study.
The written thesis document is arranged into the following sections:

Chapter 1 provides a background to solar thermal systems and basic thermal storage concepts as well as outlines the scope and objectives of the thesis;

Chapter 2 presents previous research considered relevant to this study;

Chapter 3 presents the background theory and methodology used to model the thermal storage and describes the numerical model assembled to simulate the performance of the multi-tank thermal storage system;

Chapter 4 describes the experimental apparatus developed to evaluate both the series and parallel configured thermal energy storages;

Chapter 5 presents and summarizes the results of the experimental test program conducted at a range of conditions and compares these to simulated results;

Chapter 6 provides a discussion and analysis of the experimental results and includes an analysis focused on refining the numerical model of the storage systems; and

Chapter 7 presents conclusions and recommendations based on the work described in the previous chapters, and makes recommendations for future areas of study.

Appendices A through J present associated material and studies conducted in support of this research but was considered too lengthy to be included in the main body of the thesis.
Investigate Literature Regarding Factors to Characterize the Level of Thermal Stratification in Storage Tanks

Identify Methods to Quantify Thermal Stratification and Evaluate System Design/Thermal Performance

Run Daylong Charge Input Profile Tests for Series and Parallel Configurations at Low and High Flow Rates

Conduct Annual Simulation of Multi-tank System for various Locations and Conditions

Conclusions/Recommendation

Figure 10. General outline used for study.
Chapter 2

Literature Review

2.1 Introduction

A general introduction to the design of small solar domestic hot water systems, and more specifically, different types of thermal energy storage was presented in Chapter 1. As well, storage systems used for solar heating applications have been described in a number of texts and papers (NATO 1976, Dincer and Rosen 2002, Duffie and Beckman 2006). Of particular relevance is the book published by Dincer and Rosen (2002) that treats many important aspects of thermal storage including fluid flow and heat transfer, energy demand, storage media, environmental issues, thermal stratification, phase change and sensible heat storage, etc. In their book, the authors also outline both energy and exergy analyses related to thermal energy storage.

There are a number of aspects of importance in the design of the thermal energy storage. These include: total capacity; energy density; the size, shape and volume; the heat loss; and the charge and discharge efficiency (Lavan and Thompson 1977, Hahne and Chen 1998). As also mentioned above, the capability of the storage to deliver its
stored energy at as high an exergy as possible is an important aspect. Maintaining a high exergy level in the thermal storage tends to equate to maintaining as high of a temperature as possible in the storage. This is most often and simply accomplished by ensuring that the storage remains thermally stratified with the hot charge fluid stored with as little mixing as possible. The generation and maintenance of a highly stratified storage depends on many factors, including those related to fluid dynamics and heat transfer. Specifically, the inlet fluid streams to the storage and storage construction, including its geometry and material properties may significantly affect the performance of thermal energy storage. Heat losses through the walls of the storage to the surrounding environment may also degrade the storage performance and lead to destratification.

With regard to thermal stratification, there are two approaches that may be followed. The first and most common is referred to as “natural stratification” where the charge fluid is circulated into a storage vessel of simple geometry and the fluid in the storage does not mix appreciably. This is most commonly achieved in vertical storages, where the inlet velocities are very low, thereby not mixing the existing fluid in the storage. As the fluid velocity is increased, the momentum of the inlet fluid stream is seen to “entrain” or mix with the fluid in the tank, increasing entropy and reducing exergy (Lightstone et al., 1988, Newton 1995). It should be noted, as well, that mixing can occur from the inflow of the charging fluid, as well as from the inflow associated with the discharging of the thermal energy storage. These aspects will be discussed further in the following sections.

Numerous designers (especially in Europe) have attempted to increase stratification in storage tanks by modifying the design and geometry of storage vessels.
In particular, they have added various diffusers and baffles to the interior of the storage tank to reduce the velocity and momentum associated with the flow of fluids into and out of the storage. These devices can be successfully deployed, but tend to increase the cost of the storage. Relevant research on this topic is reviewed in the following sections, although a key feature of the multi-tank thermal storage investigated in this study is that it is based on the use of simple, standard water storage tanks.

As described in Chapter 1, two common approaches to solar system design are direct and indirect configurations. As previously stated, a direct system is when storage fluid is circulated through the solar collectors and back to the storage during charging. These types of systems have been subject to failures due to freezing conditions, and consequently most current systems are of the indirect type, and therefore make use of a heat exchanger between the solar collector loop, which is filled with an antifreeze solution, and the potable water in the thermal storage.

A variety of heat exchanger styles are available in an indirect system, although the two most common are: an immersed coil located inside the bottom section of the thermal storage; and an external side-arm heat exchanger.

If the flow through the tank side of an external exchanger is pumped, it may exhibit all of the features of a direct system, including mixing and destratification. As well, it is commonly accepted that if an immersed coil is placed at the bottom of a fluid filled thermal storage, it will promote mixing of the portion of the storage above the heat exchanger, i.e., promoting destratification and entropy production. To promote stratification, some storage designs used in Europe do include extra baffles to direct
heated fluid to the top of a storage from an immersed coil, however, this configuration is expensive to implement.

The indirect concept, based on the use of natural convection heat exchangers, is increasingly being used as a simple means to promote stratification. Successful design of this configuration includes a careful balance between buoyancy induced flow velocity and heat exchanger effectiveness. Literature relevant to the design of natural convection heat exchangers is provided in the following sections.

As such, the following sections summarize the relevant literature on thermal storage. In an effort to address the key issues, the following topics will be considered: (i) single tank studies, (ii) natural convection heat exchangers, (iii) multi-tank studies, and (iv) performance indices.

2.2 Single Tank Studies

**Stratification Effects in Thermal Storage.** The concept of stratification in thermal solar tanks was discussed in Section 1.3.2. It has been shown that the thermal performance of water-based storage devices can be significantly improved by lowering the collector loop fluid flow rate, thus promoting an increase in thermal stratification in the storage tank(s) (Lavan and Thompson 1977, Phillips and Dave 1982, Hollands and Lightstone 1989, Cristofari *et al.* 2003). A study of “micro-flow” systems conducted by Hollands and Lightstone (1989) demonstrated that a stratified storage delivered 37% more energy than a fully mixed storage of corresponding size, Figure 11. This increase in efficiency can be attributed to two reasons: the low flow rate allows the hot water to enter and remain in a layer at the top of the tank thus allowing the energy in the tank to be available at a temperature that is closer to the desired load temperature and secondly, the resultant
stratification allows cool fluid to circulate back to the solar collectors which increases the collector efficiency due to its lower inlet temperature (Duffie and Beckman, 2006). In reality, the advantages of stratification will vary depending on the system configuration and the distribution of the load throughout the day (Rosengarten et al., 1999). A comprehensive survey on thermal stratification and its benefits was conducted by Han et al. (2008).

![Figure 11. Annual solar fraction versus collector flow rate (adapted from Wuestling et al. 1985) (Hollands and Lightstone, 1989).](image)

Some of the factors that affect stratification in a thermal storage including level of insulation, charge and discharge rates, tank inlet and outlet geometry, have been investigated by various authors including Lavan and Thompson (1977), Sharp and Loehrke (1979), Zurigat et al. (1990), Cataford and Harrison (1990), Ghajar and Zurigat (1991), Kleinbach et al. (1993), Andersen and Furbo (1999), Rosen (2001), and Shah and Furbo (2003).
Various schemes have been used to increase stratification and to maintain it over long periods of time, including baffles and diffusers. The use of baffles can effectively decrease mixing at the inlet to the tank by redirecting the incoming fluid. Diffusers are used to decrease the velocity and the kinetic energy associated with the incoming flow. Shah and Furbo (2003) conducted a theoretical and experimental analysis on the impact of different baffle plate designs on the flow patterns in a tank, Figure 12. Using computational fluid dynamics calculations, the authors were able to show how the energy quality in a hot water tank is reduced with a poor inlet design.

Altuntop et al. (2005) performed a similar type of analysis on 12 different baffle plate designs. The authors concluded that placing the baffle plate in the tank provided better stratification compared to the no baffle plate case and that the cone plate provided the best thermal stratification in the tank among all the considered cases. Jordon and Furbo (2004) measured the temperature and velocity fields around buffer plates in the center plane of a clear tank with an optical method called Particle Image Velocimetry (P.I.V.) and computer simulation. The results showed that with fairly high flow rates, the solar fraction increased by 5% when using the large buffer plate compared to the smallest
(marketed) buffer plate. Chung et al. (2008) also investigated the effects of diffuser configuration on thermal stratification in a rectangular storage tank. They presented their results as dimensionless parameters, e.g., Reynolds number, Froude number and dimensionless diffuser diameter. The Reynolds number and Froude number were based on the length scale of the inlet diffuser. Results indicated that the Reynolds number was the most dominant design parameter and that the diffuser shape played a significant role on the degree of stratification in the thermal storage tank.

To further encourage stratification, studies have also been conducted on inlet stratifiers in tanks, Figure 13 and Figure 14. Typically built of rigid material or fabric, stratifiers reduce the momentum of water entering the tank thus allowing buoyancy forces to direct the collector fluid to the location in the tank where the temperature of the two fluids are equal (Davidson and Adams, 1994a). Shah et al. (2005) and Anderson et al. (2007) conducted a series of charge experiments using a rigid stratifier mounted in a clear water tank and illustrated the influence of mounting flaps working as “non-return” valves at the stratifier openings similar to that shown in Figure 13. The authors concluded that the mounting flaps reduced the unwanted flows at the lowest opening of the inlet stratifier and therefore had a significant influence on maintaining thermal stratification in the tank.

Davidson and Adams (1994a) investigated the level of thermal stratification that can be maintained in forced-flow, direct solar heating systems using a fabric manifold, and compared the vertical temperature profiles and height-weighted energy stored in a tank against a rigid manifold and a conventional drop-tube inlet. Results indicated that the fabric manifold was 4% more effective than the rigid manifold and 48% more effective than the drop tube inlet. Although numerous studies have been performed on
stratifications enhancers (as previously stated) for storage tanks, their application is not within the scope of this thesis. Many of these studies contribute to the background on stratification in TES and provide an insight into measurement and qualification of stratification indices.

Figure 13. Advanced stratifier design (Shah et al., 2005).  

Figure 14. A stratified injection with a “charging lance” by SOLVIS, Germany (Peuser et al., 2002)

**Destratification in Storage Tanks.** Stratification of a thermal energy storage (TES) may be destroyed by different physical processes such as mixing caused by: “plume entrainment” of the incoming liquid during charging and discharging (Hollands and Lightstone, 1989), Figure 15 and Figure 16; high conductivity within the working fluid that will tend to promote mixing by transferring heat through the storage medium; and heat loss and conduction in and through the storage vessel walls (Hess and Miller, 1982). These processes are caused by several factors: the kinetic energy of the fluid jet entering the tank; heat conduction in tank components; and inverse temperature gradients that lead to buoyancy induced flow. The mixing introduced during the charge and discharge cycles is generally the major cause of destratification (Kleinbach et al., 1993).
The magnitude and distribution of daily hot water draw profiles also affects the temperature profile within a storage tank and consequently have been the subject of a number of studies. As a result of these previous works, it is common practice to use standard draw volumes and load profiles for comparing the performance of solar systems. As such, standard draw profiles have been established for the evaluation of solar domestic hot water storages (ASHRAE 1981, CAN/CSA-F379.1-88 2004, Ontario Hydro Research Division 1984, SRCC 2006) and subsequently employed in numerous studies to investigate the effects of various load profiles on the storage stratification. Buckles and Klein (1980) investigated the influence of different hot water consumption patterns on a system’s solar fraction and concluded that for recurring draw patterns, the early morning hours were the worst times to draw water from the system while the mid-afternoon hours were the best times. Furthermore, the authors noted that for a non-recurring draw pattern,
where the daily hot water demand changed from day to day, the solar fraction was reduced significantly. McCarthy (1990) undertook a similar investigation and agreed with the findings of Buckles and Klein (1980). His results indicated that the tank exhibited the least amount of stratification during morning hour draws and that an evenly distributed hourly load profile with a large total daily withdrawal was found to provide the highest degree of stratification. Anderson and Furbo (1999) investigated the influence of draw-offs on the annual performance of the system and showed that thermal destratification in solar tanks can cause a decrease of up to 23% of the net utilized solar energy, if the lower 51% of the storage tank is mixed compared to the case with no mixing. Similarly, Knudsen (2002) showed a decrease of 10-16% of the net utilized solar energy, if the lower 40% of the storage tank is mixed. Spur et al. (2006) indicated that for a constant daily draw-off profile, a larger number of small draw-offs resulted in a better storage performance. Other studies have been conducted by Tabarra and Bowman (1985), Bannerot and Wu (1987), Elliott (1994), Cataford (1995), and Jordan and Furbo (2005).

2.3 Natural Convection Heat Exchangers

As discussed earlier, modern SDHW systems often use buoyancy driven natural convection to circulate the water on the storage side. As these systems become more refined, they are usually pre-engineered (i.e., of fixed configuration). To rate and model these systems, heat exchanger/storage performance characteristics are required. The thermal performance of this type of SDHW system is strongly dependent on the operational characteristics of the natural convection heat exchanger (NCHE). Predicting the heat exchanger performance has proven to be challenging as the natural convection flow is intrinsically self-controlling and dependent on the state of charge (i.e., the
temperature profile) of the storage tank. Over the last fifteen years, several authors have proposed empirical and theoretical models for natural convection heat exchangers (Parent et al. 1990, Fraser et al. 1995, Dahl and Davidson 1997, 1998, 1999, Liu and Davidson 1999, Purdy et al. 1998, Lin et al. 2000). These works formed the basis of the procedure used in this study to characterize the NCHE in the proposed multi-tank configuration.

Dahl and Davidson (1997, 1998, 1999) presented empirical correlations for tube-in-shell natural convection heat exchangers. Their results were presented in terms of two correlations: shear pressure drop across the heat exchanger as a function of the thermosyphon flow rate and the overall heat transfer-area product, $UA$, as a function of the Prandtl, Reynolds and Grashof numbers. The authors showed the importance of mixed convection in the heat transfer associated with the tube-in-shell thermosyphon heat exchangers they investigated. Liu and Davidson (1999) later applied this analysis to predict the performance of solar hot water systems utilizing different natural convection heat exchangers and compared these results with a pumped heat exchanger system.

Fraser et al. (1995) proposed an empirical model based on experimentally obtained pressure head and effectiveness data. Their steady-state procedure required that tests be run at a range of thermosyphon flow rates for each forced-side flow rate of interest. This procedure could be applied to a wide range of heat exchangers, but required a complicated test apparatus. Purdy et al. (1998) later modified this procedure by using a quasi-steady-state method based on the transient charging of a thermal storage. Purdy et al. (1998) tested a variety of heat exchangers over a range of typical operating conditions. Testing was conducted on an apparatus constructed at the Solar Calorimetry Lab (SCL) at Queen’s University. This new procedure allowed NCHEs to be quickly characterized but
still required one test sequence for each forced-side flow rate. Lin et al. (2000) applied the procedure introduced by Purdy et al. (1998) to test two commercially available, compact, plate heat exchangers operating at a range of forced-side flow rates and temperatures. The apparatus consisted of three parts: a supply tank (to simulate the collector array), a charge flow-loop, and a thermosyphon water recirculation loop. The results of this study suggested that computational models for NCHEs can be based on empirically obtained data relating the modified effectiveness to the modified capacity ratio, and the heat-exchange-loop pressure head to the thermosyphon flow rate. A set of functional relationships were derived for use in modeling the performance of these heat exchangers under various operating conditions. The approach that will be used for this study is based on the method of Lin et al. (2000). This method evaluates the system in a fixed configuration, representative of the manufactured system as installed (Cruickshank and Harrison, 2006a).

2.4 Multi-tank Studies

The use of divided storage or multi-tank systems was first introduced by Bejan (1982) in an effort to reduce entropy generation by identifying designs (i.e., systems, configurations) in which exergy destruction is at a minimum. The method consisted of dividing a thermal energy system into a set of subsystems that were in local (or internal) thermodynamic equilibrium with each other (Bejan et al., 1999). Taylor and Krane (1991) later investigated Bejan’s proposal in an analytical study and agreed that it was possible to design a distributed storage element of a realistic size without seriously degrading the thermodynamic performance.
Arata and de Winter (1991) presented a comparative study on seven different system configurations for solar water heating, including three configurations using multiple tanks, coupled thermally so heat can only flow in one direction. The multi-tank systems consisted of two or three tanks, a controller and a transfer pump to serve as a thermal diode to transfer heat from the solar tank to the auxiliary storage tank and offer a large resistance to heat flows in the opposite direction. The effects of the SDHW system configurations were examined under fully mixed tank conditions, no heat loss, constant load profile and constant solar input. The authors concluded that it was clearly better to use several successive segregated tanks in order to produce effective stratification than to use a mixed single tank.

Sekulic and Krane (1992a,b) investigated the use of multiple storage units to improve the thermodynamic efficiency of a sensible heat thermal energy storage. Specifically, the study focused on two tanks connected in series, each with an immersed heat exchanger. The authors conducted a First and Second Law analysis on the multi-tank system and determined that the two storage elements in series were shown to have entropy generation numbers up to 38% smaller than those corresponding to single storage element systems, thus increasing the thermodynamic efficiency. The model assumed adiabatic boundaries for each storage tank and a fixed overall heat transfer coefficient for the heat exchanger. Expressions for the transient storage temperature and heat exchanger outlet temperature were also presented.

Mather et al. (2002) investigated the concept of series-connected multiple storage tanks with immersed coil heat exchangers installed in each tank, Figure 17. The heat exchangers for charging the storage were located at the bottom of each tank while those
for discharge were located in the top section of each tank. By charging and discharging in a counter-flow configuration, a thermal diode effect could be achieved such that the individual heat exchangers would only deliver heat to the storages and would not discharge or transfer heat from a hot tank to a cold tank when cooler collector fluid was circulated through the heat exchangers. The proposed configuration was found to have numerous advantages; however it required specialized storage tank/heat exchanger configurations. As such, this concept was not widely utilized commercially.

![Schematic of multi-tank thermal storage system with immersed coil heat exchangers (Mather, 2000).](image)

Tacchi (2003) proposed a new modular tank and piping arrangement for heat storage applications. The storage device consisted of a multiple of superimposed horizontal tanks and featured an innovative network of pipes that allowed the upper and lower parts of each tank to connect with the upper and lower parts of the adjacent tank, respectively. This piping arrangement facilitated natural convection circulation between tanks and suppressed mixing between counter opposed flows.

Ragoonanan et al. (2006) also demonstrated the benefits of dividing an indirect thermal storage into two compartments. The dual compartment storage tank, Figure 18,
consisted of an equally divided rectangular storage vessel and a copper heat exchanger immersed in the storage fluid. The heat exchanger flow path was in series through the two compartments resulting in a staged heating process. It was found that for much of a discharge processes, the heat exchanger outlet temperature was higher than it would be using the same heat exchanger in an undivided storage. The authors compared the cumulative energy and exergy delivery for the entire discharge process to equivalent data for the undivided storage and found that the divided storage delivered a maximum of 11% more energy than the undivided storage.

![Figure 18. Divided indirect Integral Collector Storage with two storage compartments (Ragoonanan et al., 2006).](image)

A horizontally partitioned water tank, Figure 19, was studied by Han et al., (2008) for use in large-scale solar systems in buildings. An investigation on the effect of energy storage and thermal stratification under various operational conditions was conducted. Experimental and numerical results showed that the gap between the insulation plates effectively inhibited transverse heat transfer in the tank.
Boies and Homan (2008) investigated the limiting behaviour of single element storages in integral collector storage systems, Figure 20. The performance of perfectly stratified and fully mixed single element storages was compared to the performance of a multi-element storage of equal size. It was found that dividing the storage volume into separate sections showed significantly improved discharge characteristics as a result of improved elemental area utilization and temperature variation between elements. It was also noted that in many cases, the performance of a multi-element storage is similar to a single perfectly stratified storage element.

Figure 20. Simplified schematic of an unpressurized integrated collector storage system composed of a single perfectly stratified storage element and a counterflow heat exchanger (Boies and Homan, 2008).
2.5 Performance Indices

Although methods to enhance stratification though the addition of baffles and diffusers has been investigated, this thesis focuses on the use of simple storage tanks and natural convection. It is therefore beneficial to investigate indices that allow the performance of the storages (including stratification level) to be quantified (Castell et al. 2006, Haller et al. 2008). Ideally, a stratification index corresponds to a meaningful efficiency value that indicates how far the storage is from a storage with perfect thermal stratification and a storage that is always fully mixed. Many parameters, both dimensional and non-dimensional, have been defined in the past in order to characterize the level of stratification. According to a recent study, the authors (Rosen et al., 2004) concluded that “at present, no valid and generally accepted standards have been established for the comparison of the performance of stratified thermal storage systems”. Castell et al. (2006) further explains: “it is difficult to decide which [parameter] to use, since there is no work comparing their suitability or performance”. Inspired by these conclusions, Cabeza et al. (2006) conducted a recent study investigating and comparing various theoretical parameters in an attempt to identify the most suitable parameters for the evaluation and characterization of stratification in hot water tanks. Consequently, a series of charge tests were run under controlled conditions and results compared. A summary of their conclusions and a brief review of previous work on stratification indices are presented in Appendix A.
Chapter 3

Modeling of the Multi-tank Thermal Energy Storage

3.1 Introduction

Various multi-tank thermal storage concepts were presented in Chapter 2. The focus of the current study, however, is to investigate the operation of an innovative, medium capacity, multi-tank thermal storage conceived at the Solar Calorimetry Laboratory of Queen’s University, Figure 21 and Figure 22. In this arrangement, individual storage tanks are connected to each other in either a series or parallel configuration and charged through an “indirect”, freeze protected, charge loop. In both cases, individual natural convection heat exchangers (NCHEs) mounted on the tanks are used to transfer heat to the storage tanks while isolating the storage fluid from the charge loop heat transfer fluid. The natural convection heat exchangers (Cruickshank and Harrison 2006a, 2009a) operate in a fashion similar to a traditional thermosyphon (Duffie and Beckman, 2006) and, as such, the charging of the individual storage tanks should be

3 Throughout this document, the expressions charge flow rate and collector loop flow rate are used interchangeably.
self-regulating and limit discharge potentials (i.e., the NCHE should act as a thermal diode). In the case of the series-connected arrangement (Figure 21), the energy flow should also be self-distributing (i.e., the energy supplied from lower temperature charge periods is directed to lower temperature, downstream tanks). In the parallel flow configuration, the charge-loop flow is split through a special header arrangement and distributed to each of the natural convection heat exchangers, Figure 22. Assuring a balanced flow distribution, however, has proven difficult in parallel configurations (Cruickshank and Harrison, 2006b).

Figure 21. Schematic of the multi-tank storage plumbed in the series charge and discharge configuration.
Figure 22. Schematic of the multi-tank storage plumbed in the parallel charge and discharge configuration.

3.2 Numerical Modeling of the Multi-tank Thermal Energy Storage

The thermal performance of a solar heating system is usually estimated by computer simulation, accounting for local climatic conditions and energy load. Detailed computer models for solar heating equipment have existed for many years and have evolved over time to account for new design concepts. An important aspect in predicting the performance of a solar heating system is the accurate modeling of the operation of the components within the system and their interaction. The storage system investigated in this study is, in itself, unique as it is also comprised of a number of components. In particular, the configuration studied consisted of three separate storage tanks, each incorporating a natural convection heat exchangers. These were then connected in either the series or parallel configuration as shown in Figure 21 and Figure 22.
To accurately predict the performance of these configurations, individual numerical models of the storage tanks, the natural convection heat exchangers, and their interaction and response to system charge inputs, were required. To accomplish this, a component and system level model of the multi-tank thermal storage was developed in the TRNSYS simulation environment (TRNSYS, 2000). The details of the full multi-tank model as implemented in TRNSYS are given in Section 3.3.

During the course of the experimental investigations (described in Chapter 4), this model was refined based on comparisons with the experimental results (Chapter 5). To fully appreciate the approach used to simulate the performance of the multi-tank system and its component elements, a detailed discussion of both the basis of operation of the components and the formulation of their numerical models is provided in the following sections.

3.2.1 Component Models

Two key components that influence the overall performance of the multi-tank system are the heat exchangers and the thermal energy storage tanks. To complete the analysis of these components and determine their effect on the system performance, it was first necessary to obtain relationships that describe the transfer of mass and energy within them.

Currently, problems of this nature are often solved by numerical means and it is worthwhile at this point to outline the general governing equations associated with similar problems in fluid dynamics and heat transfer. These are described in Appendix B for the case of an incompressible, Newtonian fluid with constant properties and represent the conservation of mass, momentum, and energy associated with a laminar flow through
a control volume. Discretization and solution of these general governing equations, and the associated boundary conditions, through CFD analysis has been attempted by Purdy (1998) and Nizami et al. (2008) for both the natural convection heat exchanger and the thermal storage. However, as this approach is not the focus of the current study, this analysis will not be considered further in this document.

Rather, for this study, simple but efficient models for describing the performance of the components within the thermal storage system were evaluated. Specifically, for the natural convection heat exchangers used in the multi-tank thermal storage, a semi-empirical method similar to that proposed by Purdy et al., (1998) and refined by Lin et al., (2000) was used. For modeling the response of the multiple thermal storages, a simplified numerical model based on the work of Newton (1995) was used and implemented in the simulation software TRNSYS, ver. 15 (2000).

Although these approaches include simplifications of the fundamental fluid dynamics and heat transfer, they are usually applied to lower computational overhead and to allow for efficient simulation of annual system performance. As such, they are described below and their suitability for modeling the multi-tank thermal storage is discussed in regard to their underlying assumptions and their accuracy when compared to experimental results.

**Modeling of Sensible Heat Storage.** Computer models of storage operation have been developed and implemented within various simulation environments including the widely used TRNSYS simulation package (TRNSYS, 2000). As well, it is now possible to model water-based thermal storage with considerable accuracy through detailed multi-dimensional CFD modeling (Nizami et al., 2008). In the case of annual performance
evaluations, however, it is standard practice to use simplified computer algorithms to reduce computational overhead and computing times. The complexity of both component and system models is often weighed against “user convenience”, computing time and resources, and desired accuracy. In many instances, due to the lack of detailed information, a number of simplifying assumptions are usually made in the model. The success of this process relies on the accurate specification of the system’s physical and thermal characteristics and the complexity and underlying assumptions of the computer algorithm.

Current storage algorithms are often based on 1-D finite-volume assumptions which incorporate basic models of tank heat loss, thermal diffusion, flow and buoyancy induced mixing (Newton, 1995). These approaches have been shown to adequately represent the performance of stratified thermal storages in cases when the charge and discharge flow rates into the storage are low and therefore mixing of the tank fluid is minimal (Cruickshank and Harrison, 2006c).

The suitability of the simplified 1-D approach is based on the assumption that the temperature distribution through the thermal storage can be treated as 1-D, implying that temperature gradients exist in the vertical direction but are negligible in the horizontal direction. If heat losses through the tank wall are high or if tank wall thermal conductivity is large then it would be expected that the associated heat transfer at the wall would result in non-uniform temperature distributions and lead to buoyancy induced mixing of the storage tank. In addition, it has been suggested that actual heat loss rates from typical cylindrical thermal storages are higher than would be calculated by a simple 1-D approximate method used to estimate wall heat loss. The discrepancies would most likely
be due to multi-dimensional effects that affect the heat loss rates from the storage and the diffusion of heat through the tank wall and the fluid. In stratified thermal storages, this may also lead to discrepancies in the tank temperature profile and lead to errors in the heat loss prediction. The storage algorithm and the basic assumptions typically used in the computer modeling of solar storage heat losses (e.g., one-dimensional temperature profiles, minimal tank wall conduction, uniform wall heat loss) are described below, particularly in the context of a thermally stratified thermal storage.

3.2.2 Modeling of Stratified Thermal Energy Storage

Previous research has shown that high degrees of stratification are possible in a correctly designed thermal storage system, (Mather et al., 2002). Numerous models have been developed for liquid based, sensible heat thermal storage (Dincer and Rosen, 2002), however most are simplified 1-D models. Some have been refined to account for the effects of mixing or the entrainment of fluid in the storage during charging or load draws.

It is possible to develop a simple and efficient model of a stratified thermal storage by dividing up the tank into N constant volume sections or “nodes”, each assumed to be fully mixed and at a uniform temperature (TRNSYS, 2000), Figure 23. The choice of the number of nodes, N, will determine the resolution to which the vertical temperature distribution can be modeled in the storage tank, i.e., increasing N will allow for significant temperature gradients to be more accurately modeled (Cruickshank and Harrison, 2006c). Section 3.3.1 presents a sensitivity analysis conducted to evaluate these effects for the geometry and conditions investigated in this study. For the special case of N=1, the tank is modeled as a fully mixed tank and no stratification effects are possible (Kleinbach et al., 1993).
The time-temperature history of a node can be predicted by performing an energy balance on each storage section, accounting for thermal losses to the surroundings and the influence of the adjacent nodes (e.g., mass and energy flows, including conduction between the layers and vertical conduction through the tank walls, etc.), Figure 24 and Equation (3.1).

Figure 23. A liquid storage tank divided into sections for the purpose of modeling stratification.
To estimate the temperature distribution and the heat loss characteristics of the vertical tank, the energy and mass flows into and out of each storage node from adjacent nodes are estimated based on the node temperatures that existed at the beginning of the time step, Figure 24. Any temperature inversions that result from these flows are eliminated by mixing appropriate nodes at the end of each time step (Kleinbach et al., 1993). To solve for the temperature distribution in the storage tank, a set of N first-order, ordinary differential equations resulting from each node’s energy balance (Newton, 1995) is assembled, e.g., an energy balance written about the i-th tank node is:

\[
\dot{m}_{\text{in}} C_p (T_i) \quad \text{or} \quad (k + \Delta k) A_{x,i} \frac{T_{i+1} - T_i}{\Delta x_{i+1}} \quad \text{or} \quad (k + \Delta k) A_{y,i} \frac{T_{i+1} - T_i}{\Delta x_{i+1}}
\]
\[
M_i C_p \frac{dT_i}{dt} = \frac{(k + \Delta k)A_{c,i}}{\Delta x_{i+1-i}} (T_{i+1} - T_i) + \frac{(k + \Delta k)A_{s,i}}{\Delta x_{i-1-i}} (T_{i-1} - T_i) + (U_i)A_{s,i} (T_{\text{env}} - T_i) + \\
+ \dot{m}_{\text{down}} C_p (T_{i-1}) - \dot{m}_{\text{up}} C_p (T_i) - \dot{m}_{\text{down}} C_p (T_i) + \dot{m}_{\text{up}} C_p (T_{i+1}) + \\
+ \dot{m}_{1\text{in}} C_p T_{1\text{in}} - \dot{m}_{1\text{out}} C_p T_i + \dot{m}_{2\text{in}} C_p T_{2\text{in}} - \dot{m}_{2\text{out}} C_p T_i
\]  

(3.1)

where \( \dot{m}_{\text{up}} \) and \( \dot{m}_{\text{down}} \) are the fluid flow rates up and down the tank, respectively; \( A_{c,i} \) and \( A_{s,i} \) are the cross-sectional and surface area of node \( i \), respectively; \( k \) and \( \Delta k \) are the tank fluid thermal conductivity and the de-stratification conductivity, Equation (3.5), (used to model destratification due to mixing at node interfaces and conduction along the tank wall), respectively; \( U_i \) is the node heat loss coefficient per unit area; \( \dot{m}_{1\text{in}}, \dot{m}_{1\text{out}}, \dot{m}_{2\text{in}} \) and \( \dot{m}_{2\text{out}} \) are the mass flow rates of the entering and exiting fluids 1 and 2, respectively; \( T_{i+1}, T_i, T_{i-1}, T_{1\text{in}}, T_{2\text{in}} \) and \( T_{\text{env}} \) are the temperatures located below, at and above node \( i \), the temperature of the entering fluid 1 and the entering fluid 2, and the temperature of the environment, respectively; and \( \Delta x_{i+1-i} \) and \( \Delta x_{i-1-i} \) are the center-to-center distance between node \( i \) and the node below and above it, respectively.

As the temperature of each node depends on the temperatures of the adjacent nodes and the temperature of the environment, it is necessary to simultaneously solve the system of equations. Computational efficiency and speed are therefore important, as an annual simulation may involve 200,000 time steps. In computer simulation programs (e.g., TRNSYS, 2000), it is common practice to use standard numerical techniques to solve for the temperature distribution at each time step (Newton, 1995). Newton compared four solution algorithms on the basis of speed and accuracy, and compared them to an analytical approach in his thesis. His worked formed the basis of the algorithm.
used in the current TRNSYS (2000) modeling routine. This routine is used extensively in this study.

It has also been noted that simple storage models rapidly become more complicated as multiple inlet and outlet ports are added or auxiliary elements are placed in a storage tank (Newton 1995, Lightstone et al. 1989). As such, a number of variations have been introduced in current models, to better accommodate issues such as variable storage volumes, plume entrainment (Lightstone et al., 1988), and draw-off mixing (Kleinbach et al., 1993). Using a simple one-dimensional model as described above, it should be noted that the accuracy of the model relies on a number of assumptions being met. These are:

1. the flow of liquid within the tank is one-dimensional;
2. the temperature and density of the fluid in each node is uniform and constant over the time step;
3. the fluid streams from each node are considered fully mixed before they enter an adjacent node;
4. the heat loss to the exterior of the tank and conduction in the tank walls are low enough that two- or three-dimensional temperature gradients do not form, promoting convection and de-stratification; and
5. the fluid velocities entering and exiting the storage tank are low enough that they do not promote extensive mixing within the storage tanks.

Not all of these assumptions are fully met in all real storages. For example, Shyu et al. (1989) found that stratification decayed in a tank more rapidly than predicted for the theoretical rate when using the conductivity of water. The main reason for this is that the thermal conductivity of the wall material is typically higher than the conductivity of the water and this can promote convection motion along the wall and destratification within the tank. Given this, Shyu et al. (1989) computed a table of “effective” conductivity
values for various walls and insulation thicknesses to serve as a starting point for users. Newton (1995) also discussed the effect of conduction between adjacent nodes and along the storage wall on the accuracy of 1-D models, Figure 25.

![Figure 25. Destratification between adjacent nodes due to wall conduction (Newton, 1995).](image)

He described the heat flow from node $i$ to node $i+1$ as:

\[ q_{\text{total}} = q_{\text{wall}} + q_{\text{fluid}} \]  

(3.2)

\[ q_{\text{total}} = k_{\text{wall}} \cdot A_{c,\text{wall}} \frac{(T_i - T_{i+1})}{\Delta x} + k_{\text{fluid}} \cdot A_{c,\text{fluid}} \frac{(T_i - T_{i+1})}{\Delta x} \]  

(3.3)

where $A_{c,\text{wall}}$ and $A_{c,\text{fluid}}$ are the cross-sectional areas of the wall and tank fluid, respectively, perpendicular to the direction of heat flow. This equation can be rearranged to give:

\[ q_{\text{total}} = \frac{(k_{\text{fluid}} + \Delta k) A_{c,\text{fluid}}}{\Delta x} (T_i - T_{i+1}) \]  

(3.4)

where $\Delta k$ is an additional thermal conductivity term defined as:

\[ \Delta k = k_{\text{wall}} \frac{A_{c,\text{wall}}}{A_{c,\text{fluid}}} \]  

(3.5)
This simple approach (Equations (3.4) and (3.5)) relies on the following assumptions being met:

1. the wall and fluid are assumed to be at the same temperature in each node;
2. the conductivity of the fluid and wall in each node is uniform and constant over the time step; and
3. the thickness of the tank wall is much less than the radius of the tank.

Newton (1995) concluded that Equation (3.5) can serve as a good approximation to calculate the “effective” conduction between the adjacent nodes however he recommended that it should only be used if no experimental data is available. The reason for this is that although the equation is a good approximation of the conduction between the layers, it does not take into account the additional convection motion occurring in the tank due to this mixing and hence under-predicts the stratification decay. Hence, by using experimental data, the user can obtain an additional conductivity factor that will better reflect the stratification behaviour of the tank.

The storage algorithm and the basic assumptions typically used in the computer modeling of solar storage heat losses (e.g., one-dimensional temperature profiles, minimal tank wall conduction, uniform wall heat loss) were investigated (Cruickshank and Harrison, 2009b) in the context of a thermally stratified thermal storage (Appendix C). This investigation consisted of a cool-down test similar to that presented in the SRCC Document TM-1 (SRCC, 2006) to determine the heat loss characteristics (e.g., the average heat loss rate, the area weighted average U-value, etc.) of the unit under evaluation and a heat diffusion test sequence to determine the heat diffusion rate (e.g., the maximum temperature gradient, thermocline profile, etc.) of the storage unit. In addition, the assumptions relating to the one dimensional temperature distribution, minimal wall
conduction and uniform heat loss were also investigated and are presented in Appendix C.

During the cool-down test, it was observed that a large percentage of the heat loss to the surroundings was through the bottom of the tank. This is most likely due to the fact that less insulation was located at the bottom of the tank which increased heat conduction from the adjacent regions. As expected, the average heat loss rate diminished as the test continued since the temperature difference between the tank and the surroundings was reduced. The U-values for all the nodes remained fairly constant over the test period. The average heat loss rate and area-weighted average U-values derived from the cool-down test sequence were compared to computer predictions based on estimated thermal properties. An average storage wall U-value was derived by computer simulation to best represent the heat loss characteristics of the entire tank. A heat diffusion test sequence was also conducted and results showed that there was close agreement between the measured time-temperature history and the simulated results using the U-values derived for each node and the average storage wall U-value. Finally, it was observed that there were virtually no temperature gradients in the horizontal direction, and it was determined that to adequately produce acceptable predictions, it is necessary to use a heat loss parameter that accurately represents the unit as installed.

3.2.3 Modeling of Natural Convection Heat Exchanger Performance

As described, the multi-tank system studied utilizes an array of natural convection heat exchangers to transfer heat to each of the component storage tanks (Cruickshank and Harrison, 2006a, 2009a).
Heat exchangers operating under forced flow conditions have been extensively studied (Kays, 1984). Conventional performance indices (Appendix D) were developed to characterize systems with forced flow on both sides of the heat exchanger and may be misleading for systems where at least one of the flows is governed by natural convection. In these cases, the flow on the storage side is driven by temperature-dependent buoyancy forces and therefore is influenced by changes in density. These buoyancy forces are controlled by the temperature distribution of the heat exchange loop and the storage tank and can result in a net hydrostatic pressure difference across the heat exchanger loop, i.e.,

\[
\Delta P_{\text{net, hydrostatic}} = \rho_{T_{\text{Tank}}} \cdot g \cdot H_{\text{Tank}} - \rho_{T_2} \cdot g \cdot (H_{\text{Tank}} - H_{\text{HX}}) - \rho_{T_{\text{hx}}} \cdot g \cdot H_{\text{HX}}
\]  

\[
\Delta P_{\text{net, hydrostatic}} = \rho_{T_{\text{Tank}}} \cdot g \cdot H_{\text{Tank}} - \rho_{T_2} \cdot g \cdot \left( H_{\text{Tank}} - \frac{H_{\text{HX}}}{2} \right) - \rho_{T_3} \cdot g \cdot \frac{H_{\text{HX}}}{2}
\]  

where \( \rho_{T_{\text{Tank}}} \) is the mean density of the water in the storage tank, \( \rho_{T_2} \) is the density of the water at the heat exchanger outlet temperature, \( \rho_{T_3} \) is the density of the water at the heat exchanger inlet temperature and \( \rho_{T_{\text{hx}}} \) is the mean density of the water in the heat exchanger. The dimensions \( H_{\text{tank}} \), \( H_{\text{pipe}} \), and \( H_{\text{hx}} \) are shown in Figure 26.

The derivation of Equation (3.7) is based on the assumption that the temperature distribution in the NCHE is linear, which is a reasonable approximation for small heat exchangers.
As well, the natural convection flow rate depends on the pressure-drop characteristics of the heat exchanger and the associated piping (assuming the pressure drop through the storage tank is negligible). During operation, the natural convection flow rate increases until the pressure drop through the heat exchanger and the associated piping, $\Delta P_{\text{shear}}$, is equal in magnitude to $\Delta P_{\text{net hydrostatic}}$. Consequently, as the thermal storage temperature approaches the temperature of the heat exchange loop, both $\Delta P_{\text{net hydrostatic}}$ and natural convection flow rate decrease to maintain equilibrium conditions (Purdy et al., 1998).

To account for the performance dependencies in natural convection heat exchangers, a set of modified equations was proposed to better reflect their operational characteristics (Fraser et al., 1995). The motivation for this is evident in the case of a charging thermal storage, i.e., as the storage tank is heated, the net static pressure head...
will be reduced, causing the flow through the NCHE to reduce. Since the collector-side flow rate is pumped at a predetermined rate, the natural convection flow will become the minimum capacitance rate, \((\dot{m}c_p)_{\text{min}}\). As the capacitance rate approaches zero, the effectiveness value defined in Equation (D.3) will increase to infinity. This is misleading since as the capacitance rate tends to zero, heat transfer diminishes, implying reduced effectiveness rather than increased.

The following “modified” performance parameters are based on the pumped, collector-side capacitance rate rather than the minimum capacitance rate as calculated by the conventional indices. That is, the minimum capacitance rate, \((\dot{m}c_p)_{\text{min}}\), in the denominator of Equations (D.3), (D.4) and (D.5), is replaced by the forced-side capacitance rate, \((\dot{m}c_p)_{c}\). This avoids the situation where the performance indices could be based on the natural convection flow that varies with the state of charge of the thermal storage. Therefore, the modified performance indices are defined as the following:

- **The modified effectiveness**
  \[
  \varepsilon_{\text{mod}} = \frac{Q_{\text{actual}}}{Q_{\text{max}}} = \frac{(\dot{m}c_p)_{s}(T_2 - T_3)}{(\dot{m}c_p)_{c}(T_1 - T_3)}
  \] (3.8)

- **The modified number of transfer units**
  \[
  NTU_{\text{mod}} = \frac{UA}{(\dot{m}c_p)_{c}}, \text{ and}
  \] (3.9)

- **The modified capacity ratio**
  \[
  C_{r\text{mod}} = \frac{(\dot{m}c_p)_{s}}{(\dot{m}c_p)_{c}}
  \] (3.10)

These expressions may be used to predict the performance of natural convection heat exchangers (Appendix E) during normal or “positive” thermosyphon operation and are usually implemented in a general simulation routine that allows overall system performance to be determined for various loads and climatic conditions.
3.3 Implementation of the Multi-tank Model in TRNSYS

To model the thermal storage unit and the natural convection heat exchanger unit, TRNSYS ver. 15 (2000), a transient system simulation package, developed by the Solar Energy Laboratory at the University of Wisconsin, was used. The program consists of linking system components together in a desired manner (i.e., typically as they would occur in a physical system) in order to simulate the transient performance of a thermal energy system. Each component of the energy system is described as a Fortran subroutine, having inputs and outputs, (T.E.S.S., 2006) and is usually referred to as a type in the TRNSYS software. The modular nature of TRNSYS greatly reduces the complexity of the system and gives the program tremendous flexibility (TRNSYS, 2000). Users are also given the capability to run their own mathematical models in addition to the pre-existing component models that are available in the TRNSYS library\(^4\). Once the components are assembled in the main TRNSYS program (referred to as a deck), a set of algebraic and differential equations, representing each component, are simultaneously solved for the system. The performance of each component will normally depend upon fixed characteristic parameters, the performance (or outputs) of other components and time-dependent forcing functions (TRNSYS, 2000), e.g., load and weather data. For example, consider a simple solar domestic hot water system, Figure 27. The inputs to the TRNSYS component “type” algorithm for a storage tank are the outputs from the collector, the electric pump #2, and the controller (e.g., collector mass flow rate, outlet temperature of collector, controller on-off output, etc.).

---

\(^4\) The TRNSYS library includes components for weather data and processing, solar thermal and photovoltaic systems, advanced buildings, HVAC systems, hydrogen and fuel cell systems, etc.
In addition, the user-defined time-step may strongly influence system modeling and simulation time if not selected properly. Hence, choosing the proper time-step is an important decision. If the time step is too large, the solution engine may not be able to converge on a solution. If the time step is too small, the simulation time will be increased.

![Flow diagram for simple solar domestic hot water system.](image)

Figure 27. Flow diagram for simple solar domestic hot water system.

To model the operation of the TES evaluated, a TRNSYS simulation deck was developed consisting of pre-existing and custom TYPE and UNIT numbers for the system components involved. The flow diagrams for the TRNSYS simulation decks used in this study are shown in Figure 28 and Figure 29. Figure 28 illustrates the TRNSYS deck used to model the operation of the TES during charging sequences (e.g., constant temperature charge, varying power input charge, etc.) and Figure 29 illustrates the TRNSYS deck used to predict the annual performance of the complete TES system. The schematics shown are for the series-configured multi-tank arrangement. Typical parameter and input values used in the TYPE 60 subroutine for this study are shown in Appendix F.
Figure 28. Flow diagram of TRNSYS deck used to model the operation of the TES during charging sequences.
Figure 29. Flow diagram of TRNSYS deck used to predict the annual performance of the complete TES system.
3.3.1 Modeling of the Thermal Storage

Of the 5 pre-existing TRNSYS (2000) thermal storage component models built into the program, TYPE 4, 38 and 60 model the behaviour of fixed volume stratified storage tanks. TYPE 4 is the simplest of the models, but only sub-divides the storage volume into a maximum of 15 fully mixed layers. Previous work has shown that it may be difficult to accurately model the temperature profile in a stratified thermal storage if the number of layers is low (Newton 1995, Cruickshank and Harrison 2006c). An alternative is the TYPE 38 routine that is based on a “plug flow” model. It adjusts the number and size of the tank layers to accommodate the flow of liquid into and out of the storage during each time step. Tank temperature profiles are not readily available in its output and, as such, it was not used.

The TYPE 60 routine is based on the work of Newton (1995) and was used to model the behaviour of the thermal energy storage system under evaluation. The routine models the performance of fluid-filled sensible energy storage tanks, subject to thermal stratification, by assuming that the tanks consist of N fully-mixed equal volume segments, Figure 23 (Newton 1995). Energy and mass balance equations are written for each segment while accounting for thermal losses to the surroundings and vertical conduction through the tank walls, Figure 24 and Equation (3.1). In addition, the TYPE 60 routine uses its own internal time steps to minimize errors. This not only increases the model’s speed and accuracy but also allows results to be unaffected by the size of the TRNSYS time step (Newton, 1995).

TYPE 60 offers numerous options including fixed or variable inlets, unequal size nodes, temperature deadbands on heater thermostats, incremental loss coefficients,
internal heat exchangers and non-circular and horizontals tanks (Newton, 1995). TYPE 60 also allows up to 100 layers within a single storage tank and can output storage temperature profiles. A larger number of nodes allows predictions closer to ideal stratification or, equivalently, less mixing (Dincer and Rosen, 2002). The limitations within the default configuration of TRNSYS ver. 15 restrict the total number of storage layers of all tanks being modeled to 100. This limited the modeling of the three tank series configuration, i.e., each storage was configured with 30 equal volume layers. The temperature profile in the parallel configuration was also modeled with the TYPE 60 routine, assuming 60 equal volume layers.

To investigate the resolution to which the vertical temperature distribution can be modeled in the storage tank, a sensitivity analysis was conducted to compare the calculated energy storage values to the number of volume layers selected (Cruickshank and Harrison, 2006c). For this investigation, a constant temperature charge sequence was simulated using the TYPE 60 routine for the parallel flow configuration, varying the number of layers from 1 to 75. Two values of storage heat loss coefficient were studied, i.e., \( U = 5 \text{ kJ/hr m}^2 \text{ C} \) (the base case) and the case of zero storage heat loss, i.e., \( U = 0 \text{ kJ/hr m}^2 \text{ C} \). The simulation was conducted over a 40 hour period and a time-step of 0.05 hours was used for the simulation. The results indicated that the predicted energy transfer depends on the number of layers chosen however the error is less than 1% (relative to the 60 layer case) when more than 10 layers are used, Table 1. This difference is reduced as the heat loss rate \( U \) from the tank is reduced. The results verify previous studies indicating that many storage layers are required to model the temperature profiles in highly stratified storage tanks (Appendix G).
Table 1. Energy transfer to storage versus number of nodes selected.

<table>
<thead>
<tr>
<th># of Nodes</th>
<th>Energy (MJ)</th>
<th>Error (%)</th>
<th># of Nodes</th>
<th>Energy (MJ)</th>
<th>Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>132.74</td>
<td>4.43</td>
<td>1</td>
<td>145.91</td>
<td>7.81</td>
</tr>
<tr>
<td>2</td>
<td>136.58</td>
<td>1.67</td>
<td>2</td>
<td>151.94</td>
<td>4.01</td>
</tr>
<tr>
<td>3</td>
<td>137.79</td>
<td>0.80</td>
<td>3</td>
<td>154.11</td>
<td>2.63</td>
</tr>
<tr>
<td>5</td>
<td>138.60</td>
<td>0.22</td>
<td>5</td>
<td>155.88</td>
<td>1.52</td>
</tr>
<tr>
<td>7</td>
<td>138.86</td>
<td>0.03</td>
<td>7</td>
<td>156.64</td>
<td>1.04</td>
</tr>
<tr>
<td>10</td>
<td>138.99</td>
<td>-0.06</td>
<td>10</td>
<td>157.21</td>
<td>0.68</td>
</tr>
<tr>
<td>15</td>
<td>139.04</td>
<td>-0.10</td>
<td>15</td>
<td>157.64</td>
<td>0.40</td>
</tr>
<tr>
<td>30</td>
<td>139.00</td>
<td>-0.07</td>
<td>30</td>
<td>158.07</td>
<td>0.13</td>
</tr>
<tr>
<td>45</td>
<td>138.95</td>
<td>-0.04</td>
<td>45</td>
<td>158.21</td>
<td>0.04</td>
</tr>
<tr>
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<td>138.90</td>
<td>0</td>
<td>60*</td>
<td>158.28</td>
<td>0</td>
</tr>
<tr>
<td>75</td>
<td>139.00</td>
<td>-0.07</td>
<td>75</td>
<td>158.32</td>
<td>-0.03</td>
</tr>
</tbody>
</table>

* Error relative to the 60 layer case.

As a means of comparison, the top and bottom node temperatures of the storage tank (for all cases) are illustrated in Figure 30 against the temperature profile obtained for a case of 60 nodes. This figure clearly illustrates the improvement in the temperature estimates as the number of simulated layers increases.

![Figure 30. Node sensitivity of storage model.](image-url)
3.3.2 Modeling of Natural Convection Heat Exchanger (NCHE)

Although TRNSYS has a number of pre-existing component models built into the program, no models were readily available for natural convection heat exchangers. As such, two custom component modules were used to allow modeling of the NCHE and the thermosyphon loop (Purdy et al. (1998) and Lin et al. (2000)). These custom types were coupled to a TYPE 60 thermal energy storage routine and used to predict the thermosyphon flow rate, $\dot{m}_s$, and the modified effectiveness, $\varepsilon_{\text{mod}}$, according to the algorithm shown in Figure 31 (Cruickshank and Harrison, 2006a).

![Iterative Solution Procedure Diagram](image)

**Figure 31. Iterative solution procedure for the natural convection heat exchanger.**
The NCHE TRNSYS solution algorithm begins by calculating the operating pressure head, $\Delta P$, for the existing average temperature of the tank, $T_{\text{tank}}$, and the inlet and outlet temperatures, $T_3$ and $T_2$, respectively, of the heat exchanger on the storage side. Since the outlet temperature of the heat exchanger is not initially known, a guess value is entered into the algorithm. The iterative process in the solution algorithm is based on this guess value. Empirically-derived NCHE correlations (Appendix E) are subsequently used to calculate the natural convection mass flow rate, $\dot{m}_s$, for the operating pressure head and using this mass flow rate, the modified effectiveness and capacity ratio can be determined. The modified effectiveness value is then used to calculate the heat transferred through the heat exchanger, $Q_{\text{actual}}$. From this, the outlet temperature of the heat exchanger on the collector side, $T_4$, and on the storage side, $T_2$, can be found.

### 3.4 Preliminary Comparison with Experimental Results

The intended approach for the validation of the numerical model consisted of conducting controlled laboratory experiments on a full scale multi-tank thermal storage. The details of the experimental facility used to conduct the laboratory experiments, including the instrumentation and measurement hardware, are described in Chapter 4, Sections 4.2 and 4.3, and Cruickshank and Harrison (2006b).

Various plumbing arrangements were evaluated, including both series- and parallel-connected configurations. For each configuration, controlled test sequences were run consisting of charge sequences as described in Section 4.4. Each test consisted of pre-conditioning the storage to a uniform temperature, followed by conducting a controlled
charge sequence according to a specified condition (i.e., constant temperature, variable power input, etc.).

During each test sequence, experimental measurements were recorded throughout the duration of the test period. For example, variables to be measured included the temperature profiles of the storage tanks, the temperature and flow rate of the charge fluid, and the temperatures at the inlets and outlets of the heat exchangers (for charge test sequences). To aid in the validation, a variety of test sequences were investigated, including those of constant temperature charge at a range of charge loop flow rates and selected test sequences with varying power input that were representative of realistic charge conditions found in normal operation.

Data obtained from these experiments were analyzed and plotted such that detailed temperature profiles were produced for the duration of the test sequence. Using the initial storage tank temperatures and the charge conditions, the experimental test sequences were simulated using the TRNSYS simulation code. Both experimental and modeled results were compared to evaluate the model’s accuracy in predicting storage tank temperature profile and stratification levels during the course of the test sequence, and to evaluate heat exchanger performance (i.e., effectiveness and temperatures), and total energy flows into and out of the storage system.

In performing this limited validation, it was recognized that not all possible operational scenarios would be experimentally investigated. However, both ideal and realistic operational conditions would be evaluated in an attempt to identify both the accuracy and limitations of the numerical model. For example, a series of charging tests consisting of sinusoidal input power sequences, of varying magnitude and frequency,
were conducted to investigate the complex operation of the storage and its interaction with the surroundings, Chapter 5. Specifically, the sinusoidal shaped charge sequences were designed to identify the potential of destratification occurring in the storage due to adverse temperature gradients.

The accuracy of the numerical model to predict the charge sequences depends on accurate characterization of the natural convection heat exchangers under the conditions experienced during the charge sequences. For this reason, the initial sequence of charge tests were conducted under constant temperature conditions according to the sequence described by Cruickshank and Harrison (2006a) thus allowing the NCHEs used in the experimental apparatus to be characterized in-situ.

An important consideration was an accurate determination of the heat loss characteristics of the thermal storages. This quantity is often difficult to determine because of the complex geometry of the thermal storage and the interaction of the various inlet and outlet ports to and from the storages that act as thermal conduits. To evaluate these effects, a separate test sequence was conducted on the storage system according to a procedure similar to that presented in SRCC DOCUMENT TM-1 (2006) and included a cool-down test and a heat diffusion test sequence (Appendix C). The values derived from these test sequences were then used to formulate the heat loss characteristics in the numerical model.
Chapter 4

Experimental Study

4.1 Introduction

The performance of a modular thermal storage for use in solar heating systems was experimentally and numerically studied (Cruickshank and Harrison, 2006c, 2008b, 2009a). To conduct this evaluation, an experimental rig was constructed and instrumented at the Queen’s University Solar Calorimetry Lab (SCL). The apparatus allowed prototype storage systems to be evaluated under a range of charge and discharge conditions and was plumbed such that a variety of physical configurations could be evaluated (i.e., storage tanks may be connected in series and parallel configurations for both charge and discharge cycles). Solar input was simulated through the use of electric heaters that allowed the power output to be varied according to prescribed sequences. The apparatus was fully instrumented, with temperature probes inserted into each storage tank, allowing stratification levels to be determined. A computer based data acquisition (DA) system recorded the storage temperature profiles and heat exchanger temperatures in real time.
4.2 Description of the Storage System

The apparatus was constructed to evaluate thermal storage systems for heating potable water in large residential or small commercial applications. In particular, the multi-tank storage configurations considered allowed the use of inexpensive, standard, electric hot-water tanks (with the electric elements disabled). The storage tanks were fitted with side-arm, natural convection, heat exchangers (Cruickshank and Harrison, 2006a, 2009b) that allowed each storage tank in the system to be charged individually.

Two configurations were studied, i.e., series-connected and parallel-connected. A photo of the test apparatus is shown in Figure 32. The test rig consisted of three standard electric hot water tanks (270 L each), each equipped with an individual natural convection heat exchange loop.

Figure 32. Photo of the storage test rig (prior to insulating).
The heat exchangers were commercially available, compact, brazed-plate units consisting of 20 plates each (forming 10 channels on the cold side and 9 channels on the hot side). The heat exchangers were located at the bottom of side-arm circulation loops that connect the bottom and the top of each storage tank. A photo of the heat exchangers used to conduct the experiments is shown in Figure 33. The specifications of the storage tanks and heat exchangers used in the multi-tank system are summarized in Table 2.

Figure 33. Photo of one of the heat exchangers used in the apparatus (prior to insulating).
Table 2. Specifications of storage tanks and heat exchangers.

| Storage Tanks | • Three identical (residential) electric hot water heaters (electric elements disabled), 270 L each.  
• Steel (glass lined), insulated with 2 inch (0.05 m) thick fibreglass  
• Height = 1.5 m, diameter = 0.55 m |
|---------------|-------------------------------------------------------------------------------------------|
| Heat Exchangers/Natural Convection Loops | • Three stainless steel, compact, brazed-plate heat exchangers (effective heat transfer area, 0.396 m², 20 plates each)  
• $UA = 160-220$ W/°C each (Cruickshank and Harrison, 2006a).  
• Insulated (3/8” Armaflex foam), height = 0.31 m  
• Natural convection loop of nominal 0.5 inch (0.0125 m) copper pipe from heat exchanger to inlet port on top of tank. |

During charging, hot collector fluid is circulated through each natural convection heat exchanger (NCHE), thereby heating the potable water from the bottom of each storage tank and directing it to the top of each tank. The rate of circulation through each NCHE is governed by the state of charge of the associated storage tank (as indicated by its temperature profile) relative to the water-side, exit temperature of the heat exchanger (Dahl and Davidson 1997, Lin et al. 2000, and Cruickshank and Harrison 2006a). Each storage tank will charge until the storage tank temperature is effectively equal to the charge temperature in the heat exchanger.

**Series- and Parallel-Connected Multi-Tank Storages.** Both series- and parallel-configurations were evaluated. Schematics of the series- and parallel-connected storages are shown in Figure 21 and Figure 22, respectively.

In the series configuration, hot “collector” fluid was supplied to the heat exchanger on the first tank. From the exit of this heat exchanger, it was directed to the downstream heat exchangers on the second and third tanks in a sequential arrangement,
Figure 21. The flow of cold water into each heat exchanger was arranged in a “counter-flow” configuration relative to the charge fluid. To study the parallel multi-tank storage configuration, Figure 22, the experimental apparatus was modified to allow the flow configuration of the supply and return lines on both the water side and the collector-loop side to be reconfigured. The initial plumbing arrangement for this configuration used a Tichelmann-ring (Peuser, F. A et al., 2002). This arrangement is shown in Figure 34(a).

![Diagram of Tichelmann-ring and branched configurations](image)

Figure 34. Schematic of the two parallel plumbing connections configurations tested: a) Tichelmann-ring and b) branched.

Initial testing using the Tichelmann-ring arrangement revealed significant problems associated with balancing the flow through the three heat exchangers. It has been noted in the literature (Peuser, F. A et al., 2002) that this arrangement is extremely sensitive to friction losses through the piping (e.g., surface roughness, solder connections, elbows and other constrictions) that may result in flow imbalance. The non-uniform charging initially observed was also aggravated by the low flow rates through the collector-side charge loop and low pressure drops through the heat exchangers. To remedy this situation, the plumbing arrangement shown in Figure 34(b) was adopted. Subsequent tests conducted with this arrangement produced significantly improved results with respect to flow balance. However, it was discovered that extreme care must be taken to ensure that all air is removed from the charging circuit as air pockets can form.
and become trapped in the heat exchangers, causing the flow to divert to one of the other parallel branches. This has been dealt with by carefully purging the charge loop and placing an in-line air-eliminator directly ahead of the exchangers to minimize this effect.

### 4.3 Experimental Measurements

The experimental rig consisted of three parts: a supply tank and heater (to simulate the solar collector array), a charge flow loop, and the storage unit under investigation (Cruickshank and Harrison, 2006b). A schematic of the heating-loop used to charge the storage system under test is shown in Figure 35. The apparatus was instrumented, with temperature probes inserted into each storage tank, allowing the tank temperature profiles to be determined during simulated charge and discharge tests. The temperature profiles in each of the storage tanks were 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, Figure 36. For the constant temperature tests, the temperature of the charge loop was controlled by a PID controller that adjusted the heat input. This was later modified to allow the computer to directly control variable power tests. Positive-displacement pumps were used to deliver hot fluid (50/50% by volume propylene glycol/water mixture) to each heat exchanger.

Flow schematics for the series and parallel charging configurations are shown in Figure 37 and Figure 38, respectively. Table 3 indicates the valve positions required for the various series and parallel configurations.
Figure 35. Schematic of test apparatus used to charge the storage system (only one storage shown).

Figure 36. LabView display of storage and heat exchanger temperatures in real time.
Figure 37. Flow schematic for the series charging configuration.

Figure 38. Flow schematic for the parallel charging configuration.
Table 3. Valve positions required for test sequences.

<table>
<thead>
<tr>
<th>Series Configuration Charging and Discharging Sequences</th>
</tr>
</thead>
<tbody>
<tr>
<td>LOAD</td>
</tr>
<tr>
<td>NO LOAD</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Parallel Configuration Charging and Discharging Sequences</th>
</tr>
</thead>
<tbody>
<tr>
<td>LOAD</td>
</tr>
<tr>
<td>NO LOAD</td>
</tr>
</tbody>
</table>

*Note: 0 = Closed, 1 = Open

The uncertainty associated with measurement of the key parameters in the thermal energy storage (i.e., temperatures, flow rates and heat transfer rates) were estimated by the propagation of the instrument measurement uncertainties, as shown in Appendix H.

4.3.1 Experimental Setup and Commissioning

Early tests were conducted at fixed inlet temperatures, low flow and low power settings to obtain a complete picture of the operation of the thermal storage. In an effort to investigate a wider range of operational conditions, the existing apparatus was later modified to allow realistic daylong charge profiles to be evaluated. This modification required both the charge fluid flow rate and power input to be increased.

To increase the flow rate capability, which was limited to approximately 2 L per minute, two additional positive displacement pumping stations were added to the rig in a parallel flow configuration, Figure 39. This configuration allowed for test flow rates of up to 6 L per minute.
In addition, to increase power input, an additional in-line electric heater, identical to the original, was added to the charge loop, Figure 40. This allowed the effective power input to be increased to approximately 6 kW. Part of this increase was achieved by rewiring the heater elements from a series to parallel configuration.

An important aspect of running the charge tests was that each test took approximately two to three days to complete. As such, the apparatus often operated unattended (especially late in the evening) and therefore appropriate safety mechanisms were built into the controls to shut down the heater power in the event of a pump failure or loss of coolant in the charge loop. Both passive and active controls were built into the heater circuits through mechanical high temperature limit switches and through software control.
Figure 40. Photo of test apparatus used to charge the storage under test.

To allow for realistic daily charge profiles to be created through the software control of the heater power, an electronic SCR controller was utilized to linearly control the power output of the heaters based on an analog input signal (0 to 5 V) that was provided by an analog proportional control output from the data acquisition system. Modification of the data acquisition routine was required to allow various daily profiles to be followed during the test sequences covering multiple days. Calibration of the power output at various levels was conducted and as a double check of this value, the system calculated the power input to the storage through measurements of the charge loop flow rate and storage inlet and outlet temperatures. This measurement relies on accurate values of fluid specific heat and density which are both functions of temperature. Highly accurate determination of these quantities is difficult as it relies on an accurate mixing of the propylene glycol solution with water, used as primary heat transfer fluid in the charge
loop (to mimic real systems that use non-toxic anti-freeze solution). Slight variations in fluid properties, i.e., specific heat and density, could occur from batch to batch production of this chemical or due to uncertainty during mixing. Initial mixing of the water-glycol solution was accomplished by combining equal volumes of water and glycol. After mixing of the two liquids, measurements of glycol concentration were taken with an optical refractometer. However, to further minimize uncertainty related to the fluid properties, a “reference heat source” apparatus (Harrison and Bernier, 1984) was installed in the charge loop. This apparatus was able to accurately measure the product of mass flow rate and specific heat from a precisely measured electrical power input. This rig, which has a much smaller heater capacity than the main heater, was run occasionally to check the specific heat and mass flow rate calculations. The positive displacement flow meter used for determining the charge loop flow rate was calibrated by a gravimetric analysis as described in Appendix H.

Finally, to calibrate the temperature measurements recorded during the test sequences, individual thermocouple probes were placed in reference temperature baths and calibrated against secondary transfer standards for temperature. The temperature baths and reference temperature standards were available at the Solar Calorimetry Laboratory. The measurements and results of the calibration are given in Appendix H.

4.4 Experimental Procedure

Laboratory tests were initially conducted on the storage prototype in an effort to measure the unit’s thermal performance and temperature profiles under constant temperature charge conditions. Under these ideal conditions, the operational characteristics of the natural convection heat exchanger (NCHE) and the associated
thermosyphon loop were measured. These characteristics allowed performance coefficients for simple empirical expressions to be determined (Appendix E) and used as inputs to a general simulation routine, allowing the overall system performance to be determined for various loads and climatic conditions.

Under constant temperature operation (i.e., ideal conditions), maximum thermal stratification was obtained in the storage tanks and minimal carryover of heat from the high temperature storage to the lower temperature, downstream storages was observed. This favorable operational mode, however, was not typical of a normal operation of the solar heating system. In real systems, non-ideal conditions exist including: a finite number of storage tanks; limited heat transfer rates in the heat exchangers; and non-adiabatic storage tank walls and piping, etc. In addition, under normal operation, the thermal input is seen to rise in the morning and fall in the afternoon as the sun moves across the sky. The falling collector and inlet fluid temperatures associated with the afternoon periods have the potential to destratify the partially charged thermal storage. In addition, sequentially stratified thermal storages have the potential to destratify due to the transfer of heat from a high temperature storage to a storage at lower temperature. The net effect would have been to drive the temperatures in all storages to the same levels. These effects would be undesirable and would lead to lower overall system performance. Since the configuration studied utilized natural convection heat exchange loops on each of the storage tanks, the configuration should have acted as a thermal diode, reducing the potential for the transfer of heat across the various thermal storages. However, in certain circumstances, adverse pressure gradients may form that result in the
flow through the heat exchanger to reverse. This effect could have increased thermal losses from the storage and reduced the thermal performance of the system.

Laboratory tests were conducted on the series- and parallel-connected prototype in an effort to measure the unit’s thermal performance and temperature profiles under specified charge conditions (i.e., constant temperature charge and variable power input charge). These tests were performed to study the interaction of the individual sequentially-connected tanks and to investigate the effects of rising and falling charge loop temperatures on temperature profiles in the storage tanks. In particular, the test sequences evaluated the storages in terms of temperature stratification, heat transfer and energy storage rates.

4.4.1 Constant Temperature Charge Tests

Testing was conducted under constant temperature charge (i.e., ideal conditions) to determine the performance characteristics of the NCHE evaluated in this study, and evaluate the performance of the multi-tank storages at a range of operating conditions. To conduct this analysis, the following test sequence was used (refer to Figure 35).

Prior to testing, the solar storage tank and heat exchange loop were filled with cold water at a uniform temperature. The charge tank and collector-loop were set to the initial “hot-side” charge temperature. A three way valve was used to isolate the collector-loop from the test heat exchanger during preconditioning. The collector-loop flow rate was adjusted to the desired value at this time. At the start of the test, the three way valve was switched such that the hot “collector-loop” fluid was directed through the heat exchanger. During this interval, the computer based data acquisition system was started and test conditions were recorded at 1 minute intervals. An electric PID controller
was used to control heaters in the charge loop during the test period. A temperature probe was used to record the temperature profile in the solar storage tank at 0.15 m (6 in.) intervals. Individual measurements of inlet and outlet temperatures around the heat exchanger and the forced side (collector-loop) flow rate were recorded for the duration of the test. The natural convection flow rate was calculated based on an energy balance performed across the heat exchanger assuming that heat loss from the body of the heat exchanger was negligible. This assumption is supported by the fact that the outer surface area of the NCHE was small (0.08m²) and the heat exchanger was insulated to reduce heat loss. In addition, due to the flow configuration of the heat exchanger, the outer plates were adjacent to the cold side flow channels. Testing was continued until the storage became fully charged as identified by an increase in temperature at the bottom of the tank. To capture a range of operational conditions, tests were conducted at a range of collector loop flow rates and collector supply temperatures, Table 4, for the series and parallel configurations. All tests were conducted with initial tank temperatures of 5 or 15°C.
Table 4. Operational conditions for constant temperature charge tests.

<table>
<thead>
<tr>
<th>Test</th>
<th>Collector Flow Rate, (m_c) (L/min)</th>
<th>Collector Supply Temperature, (T_1) (°C)</th>
<th>Initial Temperature of Tanks, (T_{ini_Tank}) (°C)</th>
<th>Temperature Difference (T_1 - T_{ini_Tank}) (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.9</td>
<td>25</td>
<td>5</td>
<td>20</td>
</tr>
<tr>
<td>2</td>
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<td>40</td>
<td>5</td>
<td>35</td>
</tr>
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<td>65</td>
</tr>
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<td>5</td>
<td>70</td>
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<td>15</td>
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<td>75</td>
<td>5</td>
<td>70</td>
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</table>

4.4.2 Varying Power Input Charge Tests

For this study, a series of multiple-day radiation profiles were simulated. Using a computer controlled SCR programmable controller, an electric heater was adjusted to provide the desired power output at any time throughout the day. The shape of the radiation profiles was chosen to represent the power output of a fixed solar array oriented directly at zero azimuth. As an approximation of these profiles, a sine function was applied to the output control signal as shown in Figure 41. In particular, three test sequences were studied, Figure 41(a)-(c), corresponding to three hypothetical test cases. Case A consisted of two consecutive clear, sunny days (designated C-C case), Case B consisted of one clear, sunny day followed by one overcast day (designated C-O case) and Case C consisted of one overcast day followed by one clear, sunny day (designated O-C case).
Specifically, the clear days provided a maximum input to the storage of 3000 W, while an overcast day peaked at 1500 W. Sunrise to sunset consisted of 10 hours in both cases. To accelerate testing times, the “nighttime” periods were removed from the profiles, reducing the “two day” test period to 20 hours. The elimination of the “nighttime” periods was considered justifiable based on the expectation that in a real solar heating system, the differential controller would shut off the collector loop circulation pump during this period. In addition, the standby heat losses, during the “nighttime” periods, were not considered in this study.

To capture a range of operational conditions, tests were conducted at a range of collector loop flow rates, e.g., 1.2, 1.5, 2.0, 3.0 and 4.5 L/min, and corresponding charge temperatures for each of the test sequences shown in Figure 41. Both energy storage rates and temperature profiles were experimentally measured during charge periods consisting of two consecutive clear days or combinations of a clear and overcast days. Of particular interest was the effect of rising and falling charge loop temperatures on storage tank stratification levels. A further aspect of the study was to investigate the effect of increasing charge-loop flow rate on the temperature distribution within the storage system during a simulated charge sequence.
Figure 41. Radiation profiles used in the experimental sequence.
Chapter 5

Experimental and Simulation Results

5.1 Introduction

In the previous chapters, the background theory, specifications and analysis associated with the multi-tank thermal storage under investigation were presented. In Chapter 4, the experimental prototype was described, as well as the details of the experimental apparatus and procedure used to evaluate its performance.

In this chapter, the results of a comprehensive range of tests including constant temperature charge sequences, and variable input power test sequences (representative of hypothetical daily solar input profiles) are presented. In analyzing the experimental data obtained from the test sequences described in this chapter, it should be noted that significant analysis of the raw data was required. The analysis and plotting of experimental data was done using Microsoft Excel using custom worksheets.

As a basis of comparison, numerical results produced from the simulation analysis described in Chapter 3 are also presented in Section 5.5 for the test sequences. Comparison of experimental and numerical predictions provides valuable insight into the
accuracy and suitability of the numerical models, and quickly identifies any significant inadequacies. The refinement of the numerical model based on the comparisons presented in this chapter is discussed in Chapter 6. For all tests, a 50% propylene glycol/50% water mixture (by volume) was used in the charge loop. Treated water from the city mains water supply was used as the storage medium for all tests.

5.2 Constant Temperature Charge Tests

Laboratory tests were conducted on the storage prototype to measure the unit’s thermal performance and the temperature profiles in the storage tanks under constant temperature charge conditions. The constant temperature charge tests, although not typical of field operation, were used to characterize the operation of the storage system, and the natural convection heat exchangers (NCHEs) and their associated thermosyphon loops. Temperature probes inserted in each of the storage tanks (see Chapter 4) were used to record the temperature distributions and to indicate the state of charge of the TES during the course of the test sequences. As well, thermocouples installed in temperature wells on each of the heat exchangers recorded heat exchanger temperatures during the tests. The details of the full instrumentation, charge flow loop and heater control are included in Chapter 4. Tests were conducted at charge flow rates between 0.9 and 1.5 L/min and temperatures for both the series and parallel configurations recorded, however, since the constant temperature charge test results were very similar at these flow rates, only the results for 1.5 L/m are shown.

Series-Connected TES. Typical experimental results, consisting of storage tank temperature profiles, are shown in Figure 42 for a charge temperature of 50°C and a
collector-loop flow rate of 1.5 L/min (0.024 kg/s) for the series configuration. It is evident from these results, that the storage tanks were highly stratified indicating that minimal mixing occurred during the charging sequence. The plots indicate that the storages rapidly increased in temperature at the top, and that the thermoclines moved lower in the tanks as charging progressed. The plots for the downstream tanks indicate that the sequential charging of the storage tanks was evident, i.e., Tank 1 initially charged, followed by Tanks 2 and 3. This observation was supported by the corresponding plots (Figure 43 and Figure 44) of heat transfer rate measured across each of the heat exchangers during the charge sequence.
Figure 42. Temperature distribution of the series-connected storage tanks during constant temperature charging at 50°C and charge loop flow rate of 1.5 L/min.
Figure 43. Individual charge rates across each heat exchanger for the series configured thermal storage during a constant temperature (50°C) charge test at a charge loop flow rate of 1.5 L/min.

Figure 44. Stacked area graph showing individual and cumulative charge rates across the heat exchangers for the series configured thermal storage during a constant temperature (50°C) charge test at a charge loop flow rate of 1.5 L/min.
Parallel-Connected TES. Experimental results for the parallel configuration, consisting of storage tank temperature profiles, are shown in Figure 45 for a charge temperature of 50°C and a collector-loop flow rate of 1.5 L/min (0.024 kg/s). Comparing these results to those of the series-connected storage, it is immediately evident that the charging occurred simultaneously across all of the heat exchangers. It is worth noting that although Figure 45 showed that all three of the storages respond at effectively the same time, this result was not easy to achieve. Initially, the parallel system showed high degrees of flow imbalance during the testing that often resulted in only two of the three storages heating. The flow imbalance was corrected by modifying the supply header to the heat exchangers for the parallel configuration as described in Section 4.2.

![Figure 45. Temperature distribution of the parallel-connected storage tanks during a constant 50°C charge test at a charge loop flow rate of 1.5 L/min, (results for all three tanks shown).](image)

The temperature profiles for each of the tanks shown in Figure 45 indicate that parallel charging also led to highly stratified storages, similar to what was evident for the first tank in the series-connected storage. One distinction that shows up in these results is
that charging occurred rapidly up to the point where the top of the storages reached approximately 35°C and then slowed until the bottom of the tank began to heat and recirculate warmer water to heat exchanger. This effect was a result of reaching the limit of the charge fluid’s heat capacitance rate (m*Cp) which was reduced to a third (per tank) in the parallel configuration relative to the series case.

Figure 46. Individual charge rates across each heat exchanger for the parallel configured storage during a constant temperature (50°C) charge test at a charge loop flow rate of 1.5 L/min, (results for all three tanks shown).

Figure 47. Stacked area graph showing individual and cumulative charge rates across the heat exchangers for the parallel configured storage during a constant temperature (50°C) charge test at a flow rate of 1.5 L/min.
The reduced charge loop heat capacitance rate (per tank) meant that sufficient energy was not available to heat the NCHE thermosyphon flow up to the charge temperature in a single pass. It is worth noting, however, that in parallel, the initial charge rate of the three combined heat exchangers exceeded the initial rate measured for the series-connected storage. This observation was confirmed in Figure 46 and Figure 47 where the rate of heat transfer measured across each of the parallel heat exchangers is shown. Figure 46 shows that all three tanks charged at virtually identical rates and, in effect, the system acted very much like a large single tank. In Figure 47, which shows the cumulative charging capacity, it can be seen that the initial charge rate was approximately 3.5 kW for the parallel configured storage, as compared to the series configuration that had an initial charge rate of approximately 2.7 kW, Figure 44.

5.3 Storage Discharge Test Results

Although discharge of the thermal storage was not the primary focus of this study, it was decided in the early phases of this research to verify the operation of the multi-tank storage in discharge mode. This was accomplished by performing constant rate discharge tests in both the series and parallel configurations. To complete these tests, the three storage tanks in the system were raised to a uniform temperature of approximately 45°C. With charging turned off, the drain valve to the storage system was opened allowing cold mains water to enter the storage tanks, pushing out the hot water. Separate discharge plumbing circuits installed on the experimental apparatus, as described in Figure 37 and Figure 38, were used to configure the storage system for the discharge tests. Valve positions were set according to the plumbing diagrams and schedule indicated in Table 3 to obtain either the series or the parallel discharge. As with the parallel charge case, a
specially configured “branched” header arrangement was required to ensure that each of the tanks discharged equally in the parallel flow configuration. Refer to Chapter 4 for further discussion of this issue. During the discharge tests, storage tank temperature distributions were recorded for the series and parallel cases and are presented in Sections 5.3.1 and 5.3.2.

5.3.1 Series Discharge Test Results

Individual tank temperature profiles for the series-connected discharge sequence, conducted at a charge loop flow rate of 8.5 L/min, are shown in Figure 48. In the series configuration, the charge and discharge flow direction was counter-flow to the storage charge direction to encourage the delivery of the hottest water to the load. Consequently, Tank 3 was observed to drop in temperature first, followed by Tank 2 and 1. It can also be seen that significant levels of stratification were maintained even during the discharge cycle and that draw-off mixing was minimal for this case.

Figure 48. Draw temperature profiles for the three series-connected tanks during a series discharge test conducted at a charge flow rate of 8.5 L/min.
5.3.2 Parallel Discharge Test Results

The tank temperature profiles for a discharge sequence conducted on the parallel configured storage system are shown in Figure 49. In the parallel configuration, the charge and discharge flow direction was also counter-flow (i.e., the discharge occurred from the bottom to the top of each storage tank, as opposed to charging that occurs from the top to the bottom of each storage). In the parallel discharge configuration, all tanks were observed to drop in temperature at identical rates with cold water displacing and pushing out hotter water at the bottom of the tanks. It is evident that significant levels of stratification were maintained during the discharge and that minimal draw-off mixing occurred.

![Figure 49. Draw temperature profiles for the three parallel-connected storage tanks recorded during a parallel discharge test (flow rate of 16 L/min total for 3 tanks).]
The combined draw flow rate for the parallel configuration was 16 L/min which was higher than the series case. This is to be expected as the series case exhibited a higher pressure drop.

5.4 Variable Input Power Tests

In an effort to evaluate the operation of the thermal energy storage under more realistic conditions, including the effects of non-uniform and falling charge temperatures and input power, a series of tests were conducted in which the input power to the charge loop was adjusted to represent various daily charge sequences.

As such, laboratory tests were conducted on both the series- and parallel-connected storage configurations in an effort to measure the unit’s thermal performance and temperature profiles under these variable power input charge sequences. Specifically, tests were performed to study the interaction of the individual parallel and series-connected storage tanks and to investigate the effects of rising and falling charge loop temperatures on their temperature profiles.

Hypothetical Daily Charge Profiles. To represent the changing power input resulting from combinations of clear and overcast days, hypothetical input power profiles were constructed in the form of sinusoidal power distributions, peaking at “solar noon” and occurring over two consecutive 10 hour days, Figure 41. To reduce testing time, the theoretical nighttime periods were removed from the test sequence as they were not of primary interest in this study. The energy input to these daily sequences was provided by electric heaters installed in the charge loop and controlled by an analog output from the computer data acquisition system. This voltage output was used to control an SCR-based
power controller that modulated the output of the heaters in proportion to the desired average power output as discussed in Chapter 4.

Desired power output profiles were constructed in computer data files allowing the power output of the charge loop heaters to be adjusted at three-minute intervals. The data acquisition and control system was automated to run and record data for full 20 hour tests. To account for heat losses in the charge loop, actual power input to the storage was determined by measurements of charge loop flow rate and the recorded temperatures at the inlet and outlet of the complete thermal energy storage system. During typical test sequences, approximately 86% of the power input was delivered to the storage, i.e., 14% of the heater input was lost to the surroundings as piping heat losses.

For this analysis all of the input power (except for charge loop heat losses) was assumed to be delivered to the storage. That is to say that the reductions in net power input, due to limits in solar collector conversion efficiencies resulting from solar radiation transmission and absorption losses and thermal heat losses (Duffie and Beckman, 2006), were not considered in this study. This was done to ensure that it was possible to clearly identify the effects of variable input power on thermal storage performance.

Three hypothetical test sequences were constructed for the evaluations, Figure 41, designated Cases A, B, and C. Case A consisted of two consecutive clear, sunny days (i.e., C-C case), Case B consisted of one clear, sunny day followed by one overcast day (i.e., C-O case), and Case C consisted of one overcast day followed by one clear, sunny day (i.e., O-C case).
5.4.1 Test Results - Variable Input Power

The net heater input power and corresponding charge temperatures for each of the test sequences (Cases A, B and C) are shown in Figure 50 to Figure 57. As expected, the temperature of the fluid feeding into the first heat exchanger (i.e., the charge temperature) is seen to follow the input power profile, rising in the morning and falling in the afternoon, for all three cases. In addition, the charge temperature on the second day was higher than that of the first day for Cases A and B (i.e., C-C and C-O), and the net power input was slightly lower. This is due to the fact that on Day 2, the storage tanks were at a higher temperature than on Day 1, therefore requiring the charge loop to reach a higher temperature to facilitate further charging of the storage. This effect was even more evident in Figure 53 to Figure 56 (for Case B, i.e., C-O case) where the charge temperature on the second day was considerably higher than would be expected at lower input powers. In Figure 50 to Figure 57, it is immediately evident that, although the temperature profiles resemble those of the previously recorded constant temperature charge case, Figure 42, subtle differences exist. For example, in Figure 50, the temperature profiles of the first day closely resemble those of Figure 42 during the morning period; however, at hour six of the tests, the temperatures at the top of the Tank 1 converge due to the falling charge temperature occurring at that time. In effect, cooler temperatures in the charge loop lowered the temperature of the fluid entering the top of the hot storage tank resulting in destratification from mixing and lower temperatures at the top of the tank.

Comparing the temperature profiles from Figure 42 (constant temperature charge) and Figure 50, variable power input Case A, (i.e., C-C case), it is evident that charging of
all tanks reduced towards the end of the first day and stopped at the end of the day. Both Figure 42 and Figure 50 show that even in the series configuration, heat transfer occurred in the downstream heat exchangers, (i.e., at Tanks 2 and 3) due to limits in the heat exchanger effectiveness. However, in the case of Figure 50, Tank 1 stopped charging after the charge temperature began to fall below the tank temperature (at the end of the day). At that point the charging transferred to the downstream Tanks 2 and 3 that were at lower temperatures.

In Figure 50, it can be seen that on Day 2, as the input power increased, the situation reverses and Tank 3 initially charged, then Tank 2 and finally Tank 1. As the day progressed and the charge temperatures started to fall after “noon”, charging switched from Tank 1 to Tank 2 and then finally back to Tank 3. It is also evident in all the temperature plots that peak storage temperatures occurred later in the day for the downstream storage tanks, consistent with sequential charging and stratification. This is particularly evident in Figure 53 for Case B, (i.e., C-O case ) where the Tank 1 initially charges on Day 1, followed by Tank 2 and Tank 3 towards the end of the day. On the second day, which has a lower power input and corresponding lower temperatures, it may be observed that charging did not resume on Tank 1, but rather continued on Tank 2 and Tank 3. It can also be seen that at the end of the second day, the hot upper sections of the storage tanks were mixed due to dropping charge temperatures and carryover from the upstream heat exchangers to the downstream ones. This process involved reverse thermosyphoning through the NCHE charge loop as discussed in Chapter 6.

Reviewing temperature plots for the other test sequences, a number of observations can be made. In particular, for charge Case B (i.e., C-O case), mixing at the
top of the tank occurred as the charge loop temperature, and power input, drop in the afternoon from their peak values. Also evident is that due to sequential charging, the peak storage temperatures in each tank occurred later in the day as charging switches to the downstream storages.
Figure 50. Net heater input, charge temperature and temperature distribution of the storage tanks (Case A, series configuration, charge flow rate: 1.5 L/min).
Figure 51. Net heater input, charge temperature and temperature distribution of the storage tanks (Case A, series configuration, charge flow rate: 3.0 L/min).
Figure 52. Net heater input, charge temperature and temperature distribution of the storage tanks (Case A, series configuration, charge flow rate: 4.5 L/min).
Figure 53. Net heater input, charge temperature and temperature distribution of the storage tanks (Case B, series configuration, charge flow rate: 1.5 L/min).
Figure 54. Net heater input, charge temperature and temperature distribution of the storage tanks (Case B, parallel configuration, charge flow rate: 1.5 L/min).
Figure 55. Net heater input, charge temperature and temperature distribution of the storage tanks (Case B, series configuration, charge flow rate: 4.5 L/min).
Figure 56. Net heater input, charge temperature and temperature distribution of the storage tanks (Case B, parallel configuration, charge flow rate: 4.5 L/min).
Figure 57. Net heater input, charge temperature and temperature distribution of the storage tanks (Case C, series configuration, charge flow rate: 1.5 L/min).
Effects of Flow Rate on Storage Tank Temperature Profiles. In addition to the variable input power sequences, the effects of increasing charge-loop flow rate on the temperature distributions within the storage system were also investigated. To capture a range of operational conditions, tests were conducted at collector loop flow rates of 1.2, 1.5, 2.0, 3.0 and 4.5 L/min. Both energy storage rates and temperature profiles were experimentally measured for each radiation profiles i.e., Cases A, B and C.

Referring to Figure 50, Figure 51 and Figure 52, results are shown for Case A (i.e., the C-C case) at charge loop flow rates of 1.5, 3.0 and 4.5 L/min, respectively. These results show that higher tank temperatures and greater stratification were achieved at lower flow rates. Of particular interest is that, at the highest flow rate, 4.5 L/min, all three storage tanks had similar temperature distributions and levels, Figure 53, resembling the profiles expected for the parallel charge case. This result differs from the lower flow cases shown in Figure 50(1.5 L/min) and Figure 51(3.0 L/min) where it is observed that the storage was highly sequentially stratified, i.e., the first tank reached a considerably higher temperature that the two downstream storage tanks.

Heat Transfer Rates During Charge Sequences. The rate of heat transfer measured across each of the heat exchangers during the charge sequences are shown in Figure 58 to Figure 63. As shown in Figure 58 (for a charge flow rate of 1.5 L/min), this sequence of events was consistent with the decrease in heat transfer rate that occurred across the heat exchanger of Tank 1, which went to zero by hour eight.

Also evident in Figure 58 (1.5 L/min) is that due to limited heat exchanger capacity, a portion of the total energy transferred to the storage was transferred across the heat exchangers of Tank 2 and Tank 3. The heat transferred to Tank 2 is seen to increase
during the day as the charge level in Tank 1 increased, finally reaching a maximum at hour eight, as the charge temperature dropped below the temperature of Tank 1. Later in the day, as the charge temperature dropped even further, Tank 3 took up the charge. Although the top section of Tank 1 in Figure 42 was seen to remain at a fairly uniform temperature between hours eight and thirteen (corresponding to the period when the natural convection flow rate went to zero), it is evident that the temperatures within the storage tank were slowly reducing due to standby heat losses or reverse thermosyphoning. Careful examination of Figure 58 (1.5 L/min) for the corresponding period shows that the heat transfer rate across the first heat exchanger was slightly negative indicating that some heat was being removed from Tank 1 and transferred to the downstream heat exchangers. While undesirable, the magnitude of this effect appeared to be small. As well, in a typical solar heating system, the differential temperature controller would have normally shut off the circulation pump at this time.

With the onset of Day 2, the charge temperature increased and heat was initially transferred to Tank 3 which was at the lowest temperature. As the charge temperature rose throughout the day, Tank 2 and Tank 1 accepted further charging. At the midpoint of the second day, the charge temperature was high enough that the majority of the heat input went to Tank 1. The net result of this sequence of events was that the storage system was observed to be directing the energy input to the tank with the closest temperature distribution. In this way, sequential stratification was passively maintained. This process, however, was not perfect, as there was a small degree of carry-over from the high temperature storage to the downstream storage tanks.
The capability of the storage system to direct energy to the storage tank at the appropriate temperature is further illustrated in Figure 51 to Figure 57, for the Case A, B and C charge sequences. In particular, referring to the temperature profiles for Tank 1 in Figure 53 and Figure 57, it may be observed that none of the heat input from Day 2 was directed to Tank 1 for Case B (i.e., C-O case) whereas the majority of heat was directed to Tank 1 for Case C (i.e., O-C case). This interaction was also evident when looking at the individual charge rates in Figure 59, i.e., on the second day, the charge temperatures were too low to charge Tank 1 for Case B (i.e., C-O case) and were above the tank temperature in Case C (i.e., O-C case). Lastly, Table 5 shows the maximum and minimum recorded increase in tank temperature (relative to the initial tank temperature) that occurred at the end of the “two day” charge sequences for each of the tanks. Table 6 summarizes the stored energy in each storage tank at the end of the test sequence, as a percentage of the total charge.
Table 5. Temperature rise of storage tanks during variable input power test sequences.

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Configuration</th>
<th>Collector Flow Rate (L/min)</th>
<th>Temperature Rise, Max / Min (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Tank 1</td>
</tr>
<tr>
<td>1</td>
<td>A Series</td>
<td>1.5</td>
<td>40.2 / 28.1</td>
</tr>
<tr>
<td>2</td>
<td>A Series</td>
<td>3.0</td>
<td>34.0 / 20.9</td>
</tr>
<tr>
<td>3</td>
<td>A Series</td>
<td>4.5</td>
<td>31.7 / 19.5</td>
</tr>
<tr>
<td>4</td>
<td>B Series</td>
<td>1.5</td>
<td>31.5 / 11.8</td>
</tr>
<tr>
<td>5</td>
<td>B Parallel</td>
<td>1.5</td>
<td>25.1 / 18.0</td>
</tr>
<tr>
<td>6</td>
<td>B Series</td>
<td>4.5</td>
<td>25.5 / 14.7</td>
</tr>
<tr>
<td>7</td>
<td>B Parallel</td>
<td>4.5</td>
<td>25.7 / 19.7</td>
</tr>
<tr>
<td>8</td>
<td>C Series</td>
<td>1.5</td>
<td>38.2 / 21.3</td>
</tr>
</tbody>
</table>

Table 6. Percent of stored energy in storage tanks for variable input power tests.

<table>
<thead>
<tr>
<th>Test Case</th>
<th>Configuration</th>
<th>Collector Flow Rate (L/min)</th>
<th>$E_{\text{TANK 1}} / E_{\text{TOTAL}}$ (%)</th>
<th>$E_{\text{TANK 2}} / E_{\text{TOTAL}}$ (%)</th>
<th>$E_{\text{TANK 3}} / E_{\text{TOTAL}}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>A Series</td>
<td>1.5</td>
<td>43.6</td>
<td>34.9</td>
<td>21.5</td>
</tr>
<tr>
<td>2</td>
<td>A Series</td>
<td>3.0</td>
<td>39.3</td>
<td>32.9</td>
<td>27.8</td>
</tr>
<tr>
<td>3</td>
<td>A Series</td>
<td>4.5</td>
<td>35.8</td>
<td>32.1</td>
<td>32.1</td>
</tr>
<tr>
<td>4</td>
<td>B Series</td>
<td>1.5</td>
<td>46.2</td>
<td>33.4</td>
<td>20.4</td>
</tr>
<tr>
<td>5</td>
<td>B Parallel</td>
<td>1.5</td>
<td>33.3</td>
<td>33.3</td>
<td>33.4</td>
</tr>
<tr>
<td>6</td>
<td>B Series</td>
<td>4.5</td>
<td>34.7</td>
<td>32.8</td>
<td>32.5</td>
</tr>
<tr>
<td>7</td>
<td>B Parallel</td>
<td>4.5</td>
<td>33.4</td>
<td>33.2</td>
<td>33.4</td>
</tr>
<tr>
<td>8</td>
<td>C Series</td>
<td>1.5</td>
<td>42.2</td>
<td>32.7</td>
<td>25.1</td>
</tr>
</tbody>
</table>
Figure 58. Individual charge rates across each heat exchanger (Case A, series configuration, charge flow rates: 1.5, 3.0 and 4.5 L/min).
Figure 59. Individual charge rates across each heat exchanger (Cases A, B and C, series configuration, charge flow rate: 1.5 L/min).
Figure 60. Individual charge rates across each heat exchanger (Case B, series configuration, charge flow rate: 1.5 L/min).

Figure 61. Individual charge rates across each heat exchanger (Case B, parallel configuration, charge flow rate: 1.5 L/min).
Figure 62. Individual charge rates across each heat exchanger (Case B, series configuration, charge flow rate: 4.5 L/min).

Figure 63. Individual charge rates across each heat exchanger (Case B, parallel configuration, charge flow rate: 4.5 L/min).
5.5 Comparison of Experimental and Modeling Results

The experimental charge test sequences were simulated using the TRNSYS simulation code as described in Chapter 3. The experimental and modeled results were compared to evaluate the model’s accuracy in predicting storage tank temperature profiles and stratification levels during the course of the test sequences, and to evaluate heat exchanger performance (i.e., effectiveness and temperatures), and total energy flows into and out of the storage system.

Both constant temperature charge and variable input power charge cases were evaluated in an attempt to identify both the accuracy and limitations of the numerical model.

5.5.1 Constant Temperature Charge.

Temperature profiles and energy storage rates were compared by computer simulation, for a charge temperature of 50°C and a charge flow rate of 1.5 L/min, Figure 64 to Figure 66 for the series configuration and Figure 67 to Figure 69 for the parallel configuration. As is evident from the results, both experimental and simulated results compared closely for the constant temperature charge cases. For the case studied, the simulated and measured results showed a high degree of correspondence, with the simulation results correctly predicting the stratification levels, sequential charging and corresponding heat exchanger heat transfer rates for both the series and parallel cases.
Figure 64. Measured and simulated temperature profiles for the series-connected storage tanks during constant temperature charging at 50°C and a charge loop flow rate of 1.5 L/min.
Figure 65. Measured and simulated charge rates across each heat exchanger for the series configured storage during a constant temperature (50°C) charge test at a charge loop flow rate of 1.5 L/min.
Figure 66. Measured and simulated stacked area graph showing individual and cumulative charge rates across the heat exchangers for the parallel configured thermal storage during a constant temperature (50°C) charge test at a charge loop flow rate of 1.5 L/min.
Figure 67. Measured and simulated temperature distribution of the parallel-connected storage tanks during a constant 50°C charge test at a charge loop flow rate of 1.5 L/min, (results for all three tanks shown).
Figure 68. Measured and simulated charge rates across each heat exchanger for the parallel configured storage during a constant temperature (50°C) charge test at a charge loop flow rate of 1.5 L/min, (results for all three tanks).
Figure 69. Measured and simulated stacked area graph showing individual and cumulative charge rates across the heat exchangers for the parallel configured thermal storage during a constant temperature (50°C) charge test at charge loop flow rate of 1.5 L/min.
5.5.2 Variable Input Power Charge.

Experimentally measured temperature profiles, energy storage rates and heat exchanger temperatures were modeled in TRNSYS (as described in Chapter 3) for the identical test conditions and the results are plotted in Figure 70 to Figure 74 for the Case A (i.e., C-C case) sinusoidal input power sequence at a charge flow rate of 1.5 L/min. Comparisons made at the higher flow rates show similar trends and levels of correspondence.

The simulated and measured results shown exhibit a high degree of correspondence, with small discrepancies in temperature profiles and heat exchanger temperatures during cool-down periods. These discrepancies in the tank temperature profiles are most evident in Figure 70 for Tank 1. Referring to the region of the plots indicated as “W”, it may be seen that the measured upper tank temperatures dropped in the afternoon and during the “night” period and did not in the simulated results. As well, by referring to the region of the plots indicated as “X”, it is evident that the measured temperature at the bottom of the storage tank was rising over the same period but not in the modeled results.

These differences would seem to be due to the advent of reverse thermosyphoning in the NCHE heat exchange loop which was not allowed in the algorithm used in the TRNSYS simulation. Reverse thermosyphoning occurred during periods when adverse temperature and pressure gradients formed in the thermosyphon loop associated with the natural convection heat exchangers, causing the flow direction to reverse slightly and heat to be extracted from the storage tank.
The corresponding plots of heat exchanger heat transfer rates and temperatures support this finding. Specifically, referring to region “Y” as shown in Figure 73, the heat transfer rates measured across the heat exchanger were negative during this period. Similarly the corresponding heat exchanger temperatures were not well represented by the simulation during these periods setting these values to zero over region “Y”. Similar limits to the model are shown in Figure 74, where the heat exchanger temperatures in the region designated “Z” are not well represented by simulation.
Figure 70. Temperature distribution of Tank 1 during variable input power test (Case A, series configuration, charge flow rate: 1.5 L/min). Results show discrepancy in tank temperature profiles as indicated by regions “X” and “Y”.
Figure 71. Temperature distribution of Tank 2 during variable input power test (Case A, series configuration, charge flow rate: 1.5 L/min).
Figure 72. Temperature distribution of Tank 3 during variable input power test (Case A, series configuration, charge flow rate: 1.5 L/min).
Figure 73. Individual charge rates across each heat exchanger during variable input power test (Case A, series configuration, charge flow rate: 1.5 L/min).
Figure 74. Heat exchanger temperatures at first heat exchanger during variable input power test (Case A, series configuration, charge flow rate: 1.5 L/min).
6.1 Introduction

In this chapter, the key features of the multi-tank thermal storage are discussed in reference to the experimentally derived test results and numerical simulations presented in Chapter 5. The operation of the natural convection heat exchangers and their associated thermosyphon flow loops is discussed in reference to the interactions of charge-loop temperature and flow rate. In addition, the aspects of the various storage configurations, (i.e., series-connected versus parallel-connected) are discussed with regard to stratification levels and heat transfer rates obtained during simulated daily charge sequences.

Maintaining a high exergy level during the charge and discharge of a thermal storage has been identified as an important aspect in their design (Dincer and Rosen, 2002). Therefore, as a measure of the effectiveness and comparison for the multi-tank thermal storage, a Second Law analysis was performed to quantify the exergy levels associated with the multi-tank configurations. This analysis is presented in Section 6.4.
To further investigate the application of a series-connected, multi-tank thermal energy storage in a realistic application, annual performance simulations were conducted for three North American locations and compared against a simple single tank configuration. Finally, this chapter concludes with a summary discussion of the various aspects, issues and trade-offs associated with the application of a multi-tank thermal energy storage in a typical solar energy application.

6.2 Discussion of Test Results

The performance of a medium-sized, multi-tank thermal storage was investigated by experiment and computer simulation. Results were presented in Chapter 5 for a comprehensive range of tests including constant temperature charge sequences, and variable input power test sequences (representative of hypothetical daily solar input profiles). Both simulation and measured test results show that high degrees of stratification can occur in both series and parallel multi-tank storages.

6.2.1 Constant Temperature Charge Tests

Constant temperature charge tests were run at a charge temperature of 50°C and a charge loop flow rate of 1.5 L/min (0.024 kg/s), for the series and parallel configurations. Experimental and simulated results, consisting of storage tank temperature profiles and heat transfer rates, were compared and shown to correspond well for both charge and discharge cycles.

High levels of stratification occurred in both the series and parallel arrangements during the charging and discharging cycles. However, experience gained while conducting the parallel configured tests showed that a balanced flow was difficult to
achieve. Flow balance between the parallel-connected heat exchangers was finally achieved using a branched header arrangement.

To compare the rates of charge for the series and parallel-connected configurations, the heat transfer rate and total energy delivered to the storage versus time are plotted in Figure 75 and Figure 76, respectively. The results show that for the constant temperature charge case investigated in this study, the experimental and simulated results corresponded well and that the parallel configuration initially stored larger quantities of energy than the series configuration. Over the 40 hour charge period, the parallel configuration stored 10.1% more energy than the series configuration.

As discussed in Chapter 5, the test and simulation results indicated that sequential stratification was achieved in the series-connected multi-tank storage. The benefits of sequential stratification, however, are not readily obvious when comparing the parallel and series arrangement under constant temperature tests because falling charge temperatures, a significant source of destratification, do not occur under these ideal conditions. The advantages of the series configuration with respect to energy charge capacity and exergy level are discussed further in Section 6.2.2 and Section 6.4.

![Figure 75. Charge heat transfer rate measured across all heat exchangers for the constant temperature (50°C) charge test at a charge loop flow rate of 1.5 L/min.](image-url)
6.2.2 Variable Input Power Charge Tests

In the previous chapters, the background to the multi-tank thermal storage was presented. As described in Chapter 5, to evaluate the operation of the thermal energy storage under non-ideal (i.e., realistic) conditions, a number of tests were conducted and the experimental and simulated results presented. These were conducted on the series- and parallel-connected storage in an effort to measure the unit’s thermal performance and tank temperature profiles under variable input power charge conditions. Hypothetical charge profiles were created (Figure 41, Chapter 5) to investigate the effects of rising and falling power levels (and corresponding charging temperatures) on the stratification levels and temperature profiles in the storage tanks, and to study the potential for reverse thermosyphoning in the natural convection loops that could lead to destratification.

A further aspect of this study was to investigate the effect of increasing charge-loop flow rate on the temperature distribution within the storage system during the variable input power charge sequences. Results were shown in Chapter 5 (i.e., energy
storage rates and temperature profiles) for charge loop flow rates of 1.5, 3.0 and 4.5 L/min.

**Sequential Stratification in the Series-Connected TES.** Laboratory tests were conducted on a series-connected modular thermal storage to study the interaction of the individual sequentially-connected tanks. Results of this study show that the series-connected thermal storage reached high levels of temperature stratification in the storage tanks during periods of rising charge temperatures and limits destratification during periods of falling charge temperature. This feature was a consequence of the series-connected configuration that allowed sequential stratification to occur in the component tanks and facilitated energy distribution according to temperature level.

The capability of the storage system to direct energy to the storage tank at the appropriate temperature is illustrated in Figure 77 to Figure 80, for the Case A (i.e., the C-C case) charge sequence (at a charge flow rate of 1.5 L/min). For example, in Figure 77, at hour six of the test (designated period “E” on the graph), the temperatures at the top of the Tank 1 converged due to the falling charge temperature that occurred during that time, (i.e., the charge temperature was lower than the temperature at the top of Tank 1). In addition, as shown in the plot at the bottom of Figure 77, which compares charging rates across the individual tank’s heat exchangers, the heat transfer across Tank 1 was also reduced during period “E” and went to zero by hour eight.

Consistent with sequential charging, during period “F” shown in Figure 78, the temperatures and heat transfer rate of Tank 2 reached their peak values for Day 1. This is expected as the temperatures in Tank 1 exceeded the charge loop temperature at hour 8 and consequently charging switched to Tank 2 that was at a lower temperature. As the
day progressed and the charge temperature dropped, a similar event occurred and charging switched to Tanks 3 as shown for period “G” in Figure 79.

On Day 2 shown in Figure 80, the reverse situation occurred during startup, i.e., period “H”. As illustrated, Tank 3 heated first, (period “I”), followed by Tank 2, (period “J”), and finally by Tank 1, (period “K”). This sequence was evident in the charge rates for the same period.

The passive distribution of heat to each of the storage tanks according to temperature was also evident for the other input charge profiles. For example, results for Case B (i.e., the C-O case) taken at a collector flow rate of 1.5 L/min are shown in Figure 81. During the first “sunny” day, charging was similar to Case A, with down-stream charging occurring during periods “L”, “M” and part of “N”. However on the following overcast day (Day 2 of Case B), the lower temperature charging by-passed the hotter Tank 1, instead charging Tanks 2 and 3 (i.e., periods “N” and “O”). From these results it is evident that the series configuration promoted sequential stratification of the storage system by directing heat to the tank at the closest lower temperature.
Figure 77. Temperature profiles and charge rates for Case A: 1.5 L/min. Period “E” shows converging temperature at the top of Tank 1 and transfer of charge to Tank 2.
Figure 78. Temperature profiles and charge rates for Case A: 1.5 L/min. Period “F” shows the Tank 2 charge period.
Figure 79. Temperature profiles and charge rates for Case A: 1.5 L/min. Period “G” shows charging in Tank 3 only (i.e., charging of Tank 1 and Tank 2 have stopped).
Figure 80. Temperature profiles and charge rates for Case A: 1.5 L/min. Periods “I”, “J” and “K” shows the sequential charging of the storage.
Figure 81. Temperature profiles and charge rates for Case B: 1.5 L/min. Periods “L”, “M”, “N” and “O” show sequential charging of Tanks 1, 2 and 3.
6.2.3 Effects of Charge Loop (i.e., Collector) Flow Rate

The effects of charge flow rate on the tank temperature distribution and energy storage rates were summarized in Table 5 and Table 6 in Chapter 5. Figure 82, below, shows the experimental results obtained for the variable power sequence Case B, (i.e., the C-O case) at a range of charge loop flow rates. Flow rates across the first NCHE are shown for charge loop flow rates of 1.2, 1.5 and 2.0 L/min and show the dependence of the natural convection (thermosyphon) flow rate on the magnitude of the charge loop flow rate. In the configuration tested, lower charge loop flow rates will increase supply temperatures to the NCHE (for a fixed heater input power), and lead to increased fluid temperatures in the NCHE loop. This situation results in increased higher tank temperatures, higher stratification and higher exergy levels (as discussed in Section 6.4). In addition, Figure 82 also illustrates that the natural convection flow rate was negative between hour 8 and hour 13, allowing heat to be transferred from the first storage tank to the circulating charge fluid. Consequently, this energy was carried to the downstream storage tanks.

![Figure 82. Natural convection flow rates through the NCHE as measured during the variable input power tests, Case B, shown for various charge flow rates.](image-url)
Figure 83 also shows the effect of the collector loop flow rate on the magnitude of the heat transfer rate for Day 1 of Case B, (i.e., the C-O case). This figure illustrates that at lower charge flow rates, higher degrees of stratification were obtained in the storage tanks. In particular, the higher charge temperatures resulting from the lower charge loop flow rate led to higher temperatures in the first storage tank and increased stratification levels. Conversely, at higher charge loop flow rates, less energy was transferred to Tank 1 and more was carried over to Tanks 2 and 3. This result was expected as the temperatures in the storages tended towards a uniform temperature at higher flow rates. In the extreme case, the storages behaved as though the tanks were connected in parallel. This conclusion was supported by the results shown in Figure 83 and Figure 55 and Figure 56 shown for the series and parallel cases at 4.5 L/min.

![Figure 83. Individual charge rates across each heat exchanger for variable input power tests at various charge flow rates.](image)

The effects of flow rate on the performance of the multi-tank thermal storage are discussed further in Section 6.4 with regard to the storage of exergy for the series- and parallel-connected configurations.
6.3 Comparison and Refinement of the Numerical Model

Simulation and experimental results, consisting of storage tank temperature profiles, heat transfer rates and heat exchanger temperatures, were shown in Chapter 5, for the series and parallel, multi-tank configurations.

6.3.1 Comparison with Experimental Results

For the constant temperature charge case studied, the simulated and measured results show a high degree of correspondence. Good agreement was also attained for the variable input power test sequences; however, small discrepancies were identified in the plots of the tank temperature distributions particularly during the cool-down periods, Figure 70. Differences were also identified in the heat transfer rates and heat exchanger temperatures predicted by the TRNSYS simulations when compared to the experimentally recorded values, Figure 73 and Figure 74. Most of the differences between the simulated and measured results were attributed to the inability of the numerical model to correctly simulate the onset of reverse thermosyphon flow and its associated carryover of heat to the downstream storage tanks. The onset of reverse flow through the NCHEs is visible in the results presented in Figure 82.

6.3.2 Model Limitations and Refinement

The storage tanks were seen to stratify in a sequential manner (for the series configuration), as well as, individually during the charge sequences. During periods of declining charge temperatures, however, there was some evidence of mixing, resulting in a slight temperature drop in the upper sections of the storage tanks. As well, it was noted that, during the intervals corresponding to low charge temperatures, the storage tanks
appear to be slowly dropping in temperature as shown by the red dotted line in region “W” of Figure 70. This effect was a result of a number of factors including: standby heat losses from the tank walls to the surrounding environment; heat losses from the natural convection loop; and reverse thermosyphoning caused by discharge or carry-over of heat from a high temperature storage to a lower temperature storage. As well, careful inspection of the dotted blue line in region “X” of Figure 70 shows that the bottoms of the tanks were actually increasing in temperature as the upper sections were dropping. This is also indicative of reverse thermosyphoning and carry-over of heat to downstream storages.

**Modeling of Reverse Thermosyphoning.** Reverse thermosyphoning can be particularly significant in the case of a multi-tank thermal storage, and may contribute to the carryover of energy from an upstream storage to a downstream storage. This effect can degrade the performance of the storage system by limiting stratification levels.

During normal operation, the heat exchanger in a thermosyphon charge loop normally operates in a counter-flow configuration, Figure 84a). The modeling of the thermosyphon heat exchange loop in this mode has been described in Sections 3.2.3 and 3.3.2. To model the reverse flow condition, however, it was necessary to modify the previous approach, as the TRNSYS TYPE 60 thermal energy storage routine does not allow negative flow values. To accommodate the negative flow, it was decided to utilize the secondary inlet and outlet ports available within the TYPE 60 model. These are often used to accommodate water draws through the thermal storage, but for this investigation, the reverse flow was modeled as entering the secondary ports shown in Figure 84b). The net result was that during positive flow conditions, the normal inlets and outlets of the
tank were utilized and during periods of reverse flow, the secondary inlets and outlets were used to model the flow through the tank.

A variety of approaches were investigated to model the heat exchanger in the reverse flow condition. Experimental data obtained during controlled reverse flow tests were compared against the simulation data in order to arrive at an appropriate model. One approach that proved particularly successful was to model the heat exchanger as a small thermal reservoir consisting of four nodes, Figure 84b). The heat transfer to the heat exchanger was modeled as a heat addition or removal to/from the two interior nodes. Temperatures associated with the small nodes located on the top and bottom were used to indicate inlet and outlet temperatures to the heat exchanger and were placed in an attempt to correspond with the measurement locations used during the experimental portion of the study. A TRNSYS TYPE 4 thermal energy storage model was used for the small tank. The volume of the small tank was chosen to be representative of the effective capacitance of the heat exchanger body and the fluid resident in it. The small tank (i.e., the simulated heat exchanger) was connected to the storage tank by two pipes modeled with a TRNSYS TYPE 31. The fluid volumes in the connecting piping, and the heat exchanger were adjusted to best match the thermal response of the system as measured.

**NCHE Performance during Reverse Thermosyphon Periods.** During periods when reverse thermosyphon flows occurs, the NCHE operates in a parallel flow configuration rather than the normal counter flow arrangement. The solution algorithm for this case is similar to that of the positive arrangement described in Figure 31, however it was necessary to obtain new coefficients for the correlations to describe the operation of the NCHE in parallel flow. This required setting up special tests as described in Appendix I.
In parallel flow, the temperatures $\overline{T}_{\text{tank}}$, $T_1$ and $T_2$ (i.e., the mean tank temperature and the inlet temperatures of the heat exchanger on the collector side and the storage side, respectively) are initially known and by setting a guess value for $T_3$, the new empirically-derived NCHE correlations (for the reverse flow case) can be numerically solved to determine $T_3$ and $T_4$ (i.e., the outlet temperatures of the heat exchanger on the storage side and collector side, respectively). These can then be used to predict the performance of the heat exchanger during reverse thermosyphon operation and be implemented in a general simulation routine.

Using this approach, it was possible to estimate the net hydrostatic head during the test sequence and to predict the magnitude of the natural convection flow rate in the negative case. However, in completing this analysis, it was observed that the model was not able to accurately predict the reverse thermosyphon flow at very low values of the net pressure head, especially at start-up. In particular, for $|\Delta P| < 10$ Pa, experimentally measured values of thermosyphon flow did not correspond well to the values calculated (in accordance with Equation I.2). The reasons for this discrepancy have yet to be fully investigated but are most likely due to the relative magnitude of the fluid viscous and momentum forces experienced during the initiation of natural convection flow, in relation to the extremely low hydrostatic pressure head. As such, in order to quantify the magnitude of the effects of the reverse thermosyphon flow for the case studies, it was decided to use the experimentally measured values of thermosyphon flow as an input to the numerical model. As described in Appendix I, the simulation was then used to predict the net hydrostatic head (using Equation I.2), the corresponding values of heat exchanger effectiveness and the associated heat transfer rates across the heat exchangers.
Figure 84. A liquid storage tank divided into sections representative of a) the physical apparatus, b) the numerical model.

The natural convection flow rate and hydrostatic pressure head for the Case A (i.e., the C-C case) charge sequence at a charge loop flow rate of 1.5 L/min (i.e., Figure 85) are shown in Figure 86 and Figure 87 for Tank 1. The thermosyphon flow rate was calculated based on an energy balance performed across the heat exchanger assuming that heat loss from the body of the heat exchanger was negligible (Cruickshank and Harrison, 2009c). As shown in Figure 88, and described in Appendix I, the thermosyphon flow through the NCHE, both positive and negative, can be described by a simple empirical correlation based on net hydrostatic pressure head.
Figure 85. Net heater input power and charge temperature for 2-day high input power test (Case A, charge flow rate of 1.5 L/min).

Figure 86. Variation in thermosyphon flow rate during test sequence (Case A, charge flow rate of 1.5 L/min).

Figure 87. Variation in hydrostatic pressure during test sequence (Case A, charge flow rate of 1.5 L/min).
6.3.3 Comparison of Model A and Model B

Based on the analysis presented above, the original TRNSYS simulation model (Model A) was revised and the experimental sequences, shown in Figure 70 to Figure 74, were re-simulated with the revised numerical model (Model B). The results of this analysis are shown in Figure 89 to Figure 93. The results show that the refined model very closely matches the experimental result, even in the areas where discrepancies previously existed, i.e., regions W and X in Figure 89, and Y and Z in Figure 92 and Figure 93, respectively.
Figure 89. Temperature distribution of Tank 1 for (Case A, series configuration, 1.5 L/min). Model B show improvements in regions “X” and “Y”.
Figure 90. Temperature distribution of Tank 2 during variable input power test (Case A, series configuration, charge flow rate: 1.5 L/min).
Figure 91. Temperature distribution of Tank 3 during variable input power test (Case A, series configuration, charge flow rate: 1.5 L/min).
Figure 92. Charge rates across heat exchangers during variable input power test (Case A, series configuration, 1.5 L/min). Improvement in Model B results shown, e.g., Region Y.
Figure 93. Temperatures of first heat exchanger during variable input power test (Case A, series configuration, 1.5 L/min). Improvement in Model B results shown, e.g., Region Z.
6.4 Second Law Analysis of the Multi-tank Thermal Energy Storage

6.4.1 Exergy Analysis of Series and Parallel Configured Multi-tank TES

To quantify the relative benefits of the sequentially stratified thermal energy storage, values of exergy stored versus time were determined for the test sequences studied. While the application of 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 (Moran and Shapiro, 2004). 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).

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, $E_x$, can be defined as

$$E_x = (h - h_a) - T_a \cdot (s - s_a)$$  \hspace{1cm} (6.1)

where $h$ and $s$ are the specific enthalpies and entropies of the substance at its current temperature and pressure, and $h_a$ and $s_a$ are its enthalpies and entropies at a reference state. $T_a$ 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 multi-tank 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) and Appendix A, 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 Equation (6.1). 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.
Therefore, to evaluate the potential of a multi-tank TES to achieve high stored exergy values, the experimental data was analyzed and comparisons were made with recognized limiting cases. Both constant temperature and variable input power test cases were considered. In addition, both series- and parallel-connected multi-tank 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 according to Equation (6.1), for each node within the tank(s) 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 stored exergy values at any time \( t \), (i.e., \( E_{\text{stored}(t)} \)), within either the series- or parallel-connected multi-tank TES, values of exergy in each of the nodes within each of the storage tanks were summed, i.e.,

\[
E_{\text{stored}(t)} = \sum_{\text{tank}=1}^{3} E_{\text{tank}}(t) \tag{6.2}
\]

where

\[
E_{\text{tank}}(t) = \sum_{\text{node}=1}^{9} E_{\text{node}}(t) \tag{6.3}
\]

### 6.4.1 Exergy Analysis of the Constant Temperature Charge Sequence

The data obtained during the constant temperature charge sequence performed on the multi-tank thermal storage (as presented in Chapter 5) was analyzed according to Equations (6.1) to (6.3). Specifically, the sum of the stored exergy in the TES was determined at various intervals throughout the charge sequence. Values are plotted for the series and parallel constant temperature charge cases, for a charge loop flow rate of 1.5 L/min, in Figure 94 and Figure 95. When these results are compared, it is apparent that
both the series and parallel configurations performed very closely. This result is not unexpected for the constant temperature charge test sequence as, under these conditions (i.e., constant charge temperature) adverse temperature gradients that would result in significant destratification in the parallel case do not occur. It is informative, however, to compare these experimental values to ones calculated for two reference cases representative of minimum exergy and maximum exergy, Figure 94. For thermal energy storage, it is normal to assume that a perfectly stratified (i.e., plug flow) scenario is representative of the configuration that would produce the maximum achievable exergy level and, that a fully mixed storage is representative of the configuration producing the minimum exergy level, for a prescribed charge sequence (Dincer and Rosen, 2002). The exergy levels associated with these hypothetical cases are shown in Figure 94, and are seen to bracket the experimental results. In addition, these results support the use of the multi-tank system, and its natural convection heat exchanger, in that it may be observed that high exergy levels are achieved at an earlier stage in the test sequence than the case for the fully mixed storage.

The exergy values for the individual storage tanks are also shown in Figure 95 and illustrate that even over the limited charge sequence, high exergy levels are achieved in the first tank for the series-connected configuration. This result is significant, in that it shows that useful energy will be available to meet a distributed load at an earlier time in the test sequence, or in a real application, useful energy will be available at a point earlier in the day.
Figure 94. Measured cumulative exergy for the series and parallel configurations during constant temperature charge (set point temperature at 50°C and charge flow rate of 1.5 L/min). Results are compared to a simulated fully stratified single tank and fully mixed single tank with zero heat loss condition.

Figure 95. Measured cumulative exergy for the individual tanks in the series and parallel configurations during constant temperature charge (set point temperature at 50°C and charge flow rate of 1.5 L/min).
Cumulative exergy values for the series, parallel and theoretical single tank storages during the constant temperature charge test are shown in Table 7.

Table 7. Cumulative exergy values for the series, parallel and theoretical single tank storages, for a constant temperature charge test at 50°C and a collector flow rate = 1.5 L/min.

<table>
<thead>
<tr>
<th>Tank Configuration</th>
<th>Cumulative Exergy (MJ)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Series TES (Measured)</strong></td>
<td></td>
</tr>
<tr>
<td>Tank 1</td>
<td>2.56</td>
</tr>
<tr>
<td>Tank 2</td>
<td>2.39</td>
</tr>
<tr>
<td>Tank 3</td>
<td>2.04</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>6.99</strong></td>
</tr>
<tr>
<td><strong>Parallel TES (Measured)</strong></td>
<td></td>
</tr>
<tr>
<td>Tank 1</td>
<td>2.56</td>
</tr>
<tr>
<td>Tank 2</td>
<td>2.50</td>
</tr>
<tr>
<td>Tank 3</td>
<td>2.53</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>7.59</strong></td>
</tr>
<tr>
<td><strong>Theoretical TES</strong></td>
<td></td>
</tr>
<tr>
<td>Fully Stratified Single Tank</td>
<td>9.07</td>
</tr>
<tr>
<td>Fully Mixed Single Tank</td>
<td>6.50</td>
</tr>
</tbody>
</table>

6.4.2 Exergy Analysis of the Variable Input Power Charge Sequence

To further investigate the operation of the multi-tank thermal storage under more realistic operational conditions, the test data obtained during the variable input power charge sequence was analyzed according to Equations (6.1) to (6.3). Experimental test results for the series-connected storage are shown in Figure 96 to Figure 98 and Table 8 as determined for charge rates of 1.2, 2 and 4.5 L/min. It is apparent from these results that as the collector loop flow rate is increased, the rate at which exergy is stored is reduced.
These results are also shown in Figure 98 where the normalized exergy value is plotted against the test duration. In this case, the exergy level in the storage tank at any time is divided by the exergy level obtained at the end of the test. It is also interesting, to compare these results based on the specific exergy level where the exergy in the storage at any time, is divided by the collector loop mass flow rate, Figure 99. This allows results obtained at various flow rates to be directly compared.
Figure 98. Measured normalized exergy value for the series configuration during variable input power charge (charge flow rate: 1.2 - 4.5 L/min).

Figure 99. Measured specific exergy for the series configuration during variable input power charge (charge flow rate: 1.2 - 4.5 L/min).
Table 8. Measured cumulative exergy for the series configuration during variable input power charge (charge flow rate: 1.2 - 4.5 L/min).

<table>
<thead>
<tr>
<th>Tank Configuration</th>
<th>Flow Rate (L/min)</th>
<th>Cumulative Exergy (MJ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Series TES (Measured)</td>
<td>1.2</td>
<td>3.06</td>
</tr>
<tr>
<td></td>
<td>2.0</td>
<td>2.93</td>
</tr>
<tr>
<td></td>
<td>4.5</td>
<td>2.61</td>
</tr>
</tbody>
</table>

As a basis of comparison, the series and parallel configurations were also modeled and compared against two theoretical multi-tank configurations, i.e., fully stratified and fully mixed multi-tank storages. Simulations were run for a charge flow rate of 1.2 L/min and 4.5 L/min for a zero heat loss case and an area-weighted heat loss case. Results show (Figure 100 to Figure 103, and Table 9 and Table 10) that the series-connected configuration closely matched the exergy level of the fully stratified multi-tank case. At the higher collector flow rate, the exergy levels in both the series and parallel cases were reduced compared to the low flow case. At this high flow rate, the series configuration operated in a similar manner to the parallel configuration, however, the increased heat exchanger capacity of the parallel system resulted in higher exergy levels at the end of the test.
Figure 100. Simulated cumulative exergy for the series and parallel configurations during variable input power charge (charge flow rate of 1.2 L/min). Results are compared to a fully stratified multi-tank and fully mixed multi-tank. All tanks have a zero heat loss condition.

Figure 101. Simulated cumulative exergy for the series and parallel configurations during variable input power charge (charge flow rate of 1.2 L/min). Results are compared to a fully stratified multi-tank and fully mixed multi-tank. All tanks have an area-weighed heat loss condition.
Figure 102. Simulated cumulative exergy for the series and parallel configurations during variable input power charge (charge flow rate of 4.5 L/min). Results are compared to a fully stratified multi-tank and fully mixed multi-tank. All tanks have a zero heat loss condition.

Figure 103. Simulated cumulative exergy for the series and parallel configurations during variable input power charge (charge flow rate of 4.5 L/min). Results are compared to a fully stratified multi-tank and fully mixed multi-tank. All tanks have an area-weighed heat loss condition.
Table 9. Cumulative exergy values for the series, parallel and theoretical fully stratified and fully mixed multi-tank storages, for a constant temperature charge test with a setpoint at 50°C and a collector flow rate of 1.5 L/min.

<table>
<thead>
<tr>
<th>Tank Configuration (U = 0)</th>
<th>Cumulative Exergy (MJ)</th>
<th>Tank Configuration, (Uarea-weighted)</th>
<th>Cumulative Exergy (MJ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Series TES</td>
<td>3.27</td>
<td>Series TES</td>
<td>2.61</td>
</tr>
<tr>
<td>Parallel TES</td>
<td>3.10</td>
<td>Parallel TES</td>
<td>2.49</td>
</tr>
<tr>
<td>Fully Stratified Multi-tank</td>
<td>3.28</td>
<td>Fully Stratified Multi-tank</td>
<td>2.61</td>
</tr>
<tr>
<td>Fully Mixed Multi-tank</td>
<td>2.98</td>
<td>Fully Mixed Multi-tank</td>
<td>2.40</td>
</tr>
</tbody>
</table>

Table 10. Cumulative exergy values for the series, parallel and theoretical fully stratified and fully mixed multi-tank storages, for a constant temperature charge test with a setpoint at 50°C and a collector flow rate of 4.5 L/min.

<table>
<thead>
<tr>
<th>Tank Configuration (U = 0)</th>
<th>Cumulative Exergy (MJ)</th>
<th>Tank Configuration, (Uarea-weighted)</th>
<th>Cumulative Exergy (MJ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Series TES</td>
<td>2.48</td>
<td>Series TES</td>
<td>2.00</td>
</tr>
<tr>
<td>Parallel TES</td>
<td>2.79</td>
<td>Parallel TES</td>
<td>2.26</td>
</tr>
<tr>
<td>Fully Stratified Multi-tank</td>
<td>2.48</td>
<td>Fully Stratified Multi-tank</td>
<td>2.01</td>
</tr>
<tr>
<td>Fully Mixed Multi-tank</td>
<td>2.82</td>
<td>Fully Mixed Multi-tank</td>
<td>2.28</td>
</tr>
</tbody>
</table>

6.5 Summary Discussion on Multi-tank Thermal Energy Storage

6.5.1 Design and Modeling Aspects

Series versus Parallel. The performance of a medium sized, multi-tank thermal storage was investigated by experiment and computer simulation. Results were presented in Chapter 5 for a comprehensive range of tests including constant temperature charge sequences, and variable input power test sequences (representative of hypothetical daily solar input profiles).
In the case of the series-connected tanks, the results also indicate the feasibility of using side-arm, natural convection heat exchangers in a multi-tank storage system. This arrangement has the advantage of allowing the use of low cost, conventional hot water storage tanks.

**Effects of Reverse Thermosyphon Flow on TES Performance.** The experimental results show that during periods of adverse temperature gradients, reverse flow occurred in the natural convection heat exchange loops. The magnitude of the reverse flow was small however, particularly since the natural convection heat exchanger used in this study were located close to the bottom of the storage tank and had a low height. The results show that during typical daily charge sequences, the relative magnitude of the positive and negative (i.e., reverse) flow were approximately 10:1. Similarly, the energy loss due to reverse operation also represents only a small fraction of the daily energy collected, approximately 1% of the energy input as described in Table 11.

The results also show that the correspondence between the measured and simulated energy transfer rates to the first heat exchanger and the downstream heat exchangers was in close agreement. Also of note is the fact that, even when the solar collector (i.e., charge) outlet temperature dropped below the temperature at the top of the storage tank (e.g., in a cool-down period following peak solar “noon”), it was observed that de-stratification and mixing occurs at the top of the thermal storage even if an adverse pressure gradient is not present.

To form an adverse pressure gradient and establish reverse thermosyphon flow, the natural convection loop temperature would have to be significantly cooler than the average storage tank temperature. This worst-case scenario was evaluated during the
reverse thermosyphon characterization tests that were performed as part of this study (Appendix I). During these tests, a cool charge loop fluid at 25°C was circulated at three test flow rates to a fully charged TES at a uniform high temperature of 60°C. Results show that a reverse flow rate of 0.45 L/min was obtained for a collector loop flow rate of 4.5 L/min, which is considerably higher than the design flow rate for this system.

Table 11. Measured and Simulated Energy Transfer.

<table>
<thead>
<tr>
<th>EXPERIMENTAL</th>
<th>Tank 1</th>
<th>Tank 2</th>
<th>Tank 3</th>
<th>$E_{\text{TOTAL}}$ (kJ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_{\text{TANK,NET}}$ (kJ)</td>
<td>50225</td>
<td>39483</td>
<td>24232</td>
<td>113939</td>
</tr>
<tr>
<td>$E_{\text{POSITIVE}}$ (kJ)</td>
<td>51213</td>
<td>39577</td>
<td>24241</td>
<td>115031</td>
</tr>
<tr>
<td>$E_{\text{REVERSE}}$ (kJ)</td>
<td>988</td>
<td>94</td>
<td>9</td>
<td>1090</td>
</tr>
<tr>
<td>$E_{\text{REVERSE}} / E_{\text{POSITIVE}}$</td>
<td>0.0193</td>
<td>0.0024</td>
<td>0.0004</td>
<td>0.0095</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>SIMULATION</th>
<th>Tank 1</th>
<th>Tank 2</th>
<th>Tank 3</th>
<th>$E_{\text{TOTAL}}$ (kJ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_{\text{TANK,NET}}$ (kJ)</td>
<td>49665</td>
<td>38406</td>
<td>26616</td>
<td>114687</td>
</tr>
<tr>
<td>$E_{\text{POSITIVE}}$ (kJ)</td>
<td>50792</td>
<td>38509</td>
<td>26616</td>
<td>115917</td>
</tr>
<tr>
<td>$E_{\text{REVERSE}}$ (kJ)</td>
<td>1127</td>
<td>103</td>
<td>0</td>
<td>1230</td>
</tr>
<tr>
<td>$E_{\text{REVERSE}} / E_{\text{POSITIVE}}$</td>
<td>0.0222</td>
<td>0.0027</td>
<td>0</td>
<td>0.0106</td>
</tr>
</tbody>
</table>

It is also worth noting the effect of heat exchanger height on the magnitude of the reverse thermosyphon flow rate, Figure 104, for the case of a uniform high temperature tank and a significantly cooler charge temperature. For this graph, the thermosyphon flow rate is plotted against the non-dimensional height of the heat exchanger relative to the height of the tank for a temperature difference of $T_{\text{tank,ave}}$ (i.e., the mean temperature of the tank) and $T_1$ (the inlet temperature of the heat exchanger on the collector side). From these results, we see that it is important to place the heat exchanger as low as possible on
the thermal energy storage and to ensure that it is compact (i.e., as low as possible) relative to the TES. However, most solar hot water systems employ differential thermostats to control the operation of the circulating pump in the solar collector loop, and these would normally stop the charge flow in the heat exchanger if the collector outlet temperature was below the temperature at the bottom of the TES, significantly minimizing the occurrence of reverse thermosyphoning due to adverse temperature gradients. However, in a highly stratified TES, it is possible that the bottom of the storage will be at a temperature lower than the collector outlet even when the collector outlet is below the temperature at the top of the storage. In these cases, it will be important to introduce design features and control the storage geometry to limit reverse thermosyphoning.

Figure 104. Effect of heat exchanger height on the magnitude of the reverse thermosyphon flow rate.

Accounting for reverse flow conditions in system simulations presents a number of challenges. In particular, during most reverse-flow situations, the associated pressure drop and driving force for the reverse flow is exceedingly small, resulting in the flow...
being very low and often unstable. These factors make predictions of the flow conditions particularly difficult during periods of flow reversal. In addition, as flow direction is reversing and during periods of low flow, it is difficult to predict the temperature distribution through the charge loop. This increases the uncertainty in the calculation of the net hydrostatic pressure used in the method proposed by Purdy et al. (1998) and Lin et al. (2000) which was based on density variations due to temperature changes.

**Carryover of Energy in Multi-tank TES.** Another important aspect in the operation of the series-connected multi-tank TES is the effect of carryover. Figure 105 and Table 12 illustrate the effect of carryover from one storage tank to a subsequent downstream storage tank in the multi-tank sequential storage. The simulation was run for the actual series-connected multi-tank case and a multi-tank system with an ideal heat exchanger. In particular, for comparable charge temperatures and flow conditions, values calculated show that slightly less than half the total energy input was transferred to the first storage and that the subsequent energy was transferred to Tank 2 and Tank 3. The proportion of the charge within a multi-tank storage also depends on the heat exchanger effectiveness values as shown in Table 2, where the charge rates are compared against a storage with an ideal heat exchanger of fixed effectiveness equal to 1. It may also be observed that as the heat exchanger effectiveness increased, a larger quantity of the charge energy was stored in the first tank and less was carried over to the downstream storage tanks. Lower heat exchanger effectiveness also resulted in higher temperatures exiting the storage which resulted in higher heat losses from the solar collector array, thereby lowering overall system performance.
Figure 105. Simulated effect of carryover for a multi-tank storage for a $\varepsilon_{\text{mod}}$ value a) equal to measured effectiveness results and b) equal to 1.

Table 12. Simulated energy transfer rates.

<table>
<thead>
<tr>
<th>SIMULATION</th>
<th>Tank 1</th>
<th>Tank 2</th>
<th>Tank 3</th>
<th>$E_{\text{TOTAL}}$ (kJ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\varepsilon_{\text{mod}}$ (Fig. J.3)</td>
<td>$E_{\text{TANK,NET}}$ (kJ)</td>
<td>49665</td>
<td>38406</td>
<td>26616</td>
</tr>
<tr>
<td></td>
<td>$E_{\text{TANK,NET}} / E_{\text{TOTAL}}$</td>
<td>0.43</td>
<td>0.34</td>
<td>0.23</td>
</tr>
<tr>
<td>$\varepsilon_{\text{mod}}=1$</td>
<td>$E_{\text{TANK,NET}}$ (kJ)</td>
<td>100053</td>
<td>61639</td>
<td>14945</td>
</tr>
<tr>
<td></td>
<td>$E_{\text{TANK,NET}} / E_{\text{TOTAL}}$</td>
<td>0.57</td>
<td>0.35</td>
<td>0.08</td>
</tr>
</tbody>
</table>
In a series-connected, multi-tank TES, reverse thermosyphoning in the upstream storage tank will result in heat being removed from the upstream tank and transferred to the lower temperature downstream tanks. While much of the energy removed from the upstream storage is not lost but stored in the downstream tanks, there is a tendency to contribute to destratification in the upstream TES. As with the single tank, this effect can be minimized by using a heat exchanger that has a low elevation relative to the TES.

6.5.2 Annual System Performance

The overall system performance of a SDHW system, as well as the amount of solar energy ultimately delivered to the load, depends on a number of factors including: the climatic region, the mains water temperature, the tank insulation level and the location of the storage tanks (i.e., in a heated basement or an unheated garage, etc.). The magnitude and distribution of daily hot water draw profiles also affects the temperature profile within a storage tank and consequently have been the subject of a number of studies. As a result of these previous works, it is common practice to use standard draw volumes and load profiles for comparing the performance of solar systems. Standard draw profiles have been established for the evaluation of solar domestic hot water storages by various groups (e.g., ASHRAE 1981, CAN/CSA-F379.1-88 2004, Ontario Hydro Research Division 1984, SRCC 2006) and subsequently employed in numerous studies to investigate the effects of various load profiles on the storage stratification.

A potential negative aspect of a multi-tank TES is the increased tank surface area when compared with a single tank storage of similar volume. Depending on the level of thermal insulation on the tanks, overall heat losses from the storage system will increase in proportion to the increased surface area. Although both series and parallel configured
multi-tank TES have been studied by experiment and computer simulation, their operation relative to a single tank configuration has not been quantified for various climatic regions.

Consequently, the annual performance of the multi-tank TES was evaluated for a typical multi-family, domestic hot water application using a TRNSYS model of the complete solar water heating system including solar collectors, controller and storage unit. The system modeled was a series-connected multi-tank system consisting of three 270 L tanks, similar to the unit tested. As a basis for comparison, results were compared to a similar system equipped with a single large tank of equal volume (equipped with a single larger capacity heat exchanger). In addition, a typical small single-tank unit (270 L), operating under the same load and environmental conditions was modeled.

For this investigation, simulations were run for Montreal, QC (lat. N. 45.68°), assuming the mains water temperature varied from 1.5 to 21°C over the course of a year, Table 13, and a load set-point temperature of 50°C.

Table 13. Estimated monthly mains temperatures for Montreal, QC.

<table>
<thead>
<tr>
<th></th>
<th>Mains Water Temperatures for Montreal, °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Jan</td>
<td>3</td>
</tr>
<tr>
<td>Feb</td>
<td>2</td>
</tr>
<tr>
<td>Mar</td>
<td>2</td>
</tr>
<tr>
<td>Apr</td>
<td>1.5</td>
</tr>
<tr>
<td>May</td>
<td>5.5</td>
</tr>
<tr>
<td>Jun</td>
<td>12.5</td>
</tr>
<tr>
<td>Jul</td>
<td>18</td>
</tr>
<tr>
<td>Aug</td>
<td>21</td>
</tr>
<tr>
<td>Sept</td>
<td>20.5</td>
</tr>
<tr>
<td>Oct</td>
<td>18</td>
</tr>
<tr>
<td>Nov</td>
<td>11.5</td>
</tr>
<tr>
<td>Dec</td>
<td>6.5</td>
</tr>
</tbody>
</table>

Monthly average mains water temperatures were estimated from data provided by Bernier (2006). The required hourly meteorological data (solar radiation and dry bulb temperature) were taken from the Typical Meteorological Year (TMY) data files for Montreal. A daily hot water draw profile consistent with the CSA standard-day recommendations (CAN/CSA-F379.1-88, 2004) for a 300 L/day draw was assumed for all cases, Figure 106.
For both the large multi-tank and large single tank systems, the daily draw volume and collector area were increased to 3 times that of the small SDHW system, i.e., draw volume was 900 L/day and the collector area was 17.5 m². Similarly, for comparison, the collector loop flow rates increased to 3 times that of the small SDHW, in order to maintain a similar collector array efficiency and temperature distribution through the collectors. The full system computer models were then used to predict the operation of the cases considered. The specifications of the three configurations compared are given in Table 14 and Table 15.
Table 14. System specifications for the 3 systems studied, Montreal, QC.

<table>
<thead>
<tr>
<th></th>
<th>MONTREAL</th>
</tr>
</thead>
<tbody>
<tr>
<td>Collector Orientation</td>
<td>Facing south at a tilt of 30°</td>
</tr>
<tr>
<td>Distance From Collector</td>
<td>10 meters pipe length (each way)</td>
</tr>
<tr>
<td>to Tank</td>
<td></td>
</tr>
<tr>
<td>Auxiliary Heater Set</td>
<td>55°C</td>
</tr>
<tr>
<td>Point</td>
<td></td>
</tr>
<tr>
<td>Weather Data</td>
<td>Montreal TMY data [TRNSYS]</td>
</tr>
<tr>
<td>Air Temperature</td>
<td>$T_{\text{surroundings}}=20^\circ\text{C}$, this estimates the temperature in a basement</td>
</tr>
<tr>
<td>Around Indoor Tanks</td>
<td></td>
</tr>
</tbody>
</table>

Table 15. Simulation parameters for the 3 systems studied, Montreal, QC.

<table>
<thead>
<tr>
<th></th>
<th>MONTREAL</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Small Single Tank</td>
</tr>
<tr>
<td></td>
<td>Series Multi-Tank</td>
</tr>
<tr>
<td></td>
<td>Large Single Tank</td>
</tr>
<tr>
<td>Storage Volume</td>
<td>270 L</td>
</tr>
<tr>
<td></td>
<td>3 tanks x 270 L</td>
</tr>
<tr>
<td></td>
<td>810 L</td>
</tr>
<tr>
<td>Area of Collector, $A_c$</td>
<td>5.716 m$^2$</td>
</tr>
<tr>
<td></td>
<td>17.148 m$^2$</td>
</tr>
<tr>
<td></td>
<td>17.148 m$^2$</td>
</tr>
<tr>
<td>“a” Value in Eq. (E.1)</td>
<td>2.388</td>
</tr>
<tr>
<td></td>
<td>2.388</td>
</tr>
<tr>
<td></td>
<td>4.522</td>
</tr>
<tr>
<td>Collector Flow rate, $m_c$</td>
<td>74.16 kg/hr (1.2 L/min)</td>
</tr>
<tr>
<td></td>
<td>222.48 kg/hr (3.6 L/min)</td>
</tr>
<tr>
<td></td>
<td>222.48 kg/hr (3.6 L/min)</td>
</tr>
<tr>
<td>Heat Loss Coefficient, $U$</td>
<td>$\frac{kJ}{hr\ m^2\ ^\circ C}$</td>
</tr>
<tr>
<td></td>
<td>$\frac{kJ}{hr\ m^2\ ^\circ C}$</td>
</tr>
<tr>
<td></td>
<td>$\frac{kJ}{hr\ m^2\ ^\circ C}$</td>
</tr>
<tr>
<td>Load Volume</td>
<td>300 L</td>
</tr>
<tr>
<td></td>
<td>900 L</td>
</tr>
<tr>
<td></td>
<td>900 L</td>
</tr>
</tbody>
</table>

To compensate for the required increase in heat exchanger size for the large single tank, a comparison was done on both large systems (large single tank and multi-tank) with a zero heat loss condition. The pressure drop and effectiveness of the large single tank NCHE were normalized to provide a similar solar fraction as in the series multi-tank case. This was accomplished by modifying the “a” coefficient in Equation (E.1). Once a
suitable “a” value was obtained, the system simulation was run with the normal heat loss condition. The results are shown in Table 16.

The annual performance of a solar domestic hot water system was characterized by the annual efficiency and annual solar fraction, which are defined as

\[
\eta = \frac{Q_{\text{del}} - Q_{\text{par}}}{G \cdot A} \quad \text{and} \quad F_s = \frac{Q_{\text{del}} - Q_{\text{par}}}{Q_{\text{load}}}
\]

respectively, (6.4)

where \( Q_{\text{del}} \) is the solar energy delivered to the load, \( Q_{\text{par}} \) is the energy consumption due to the pump and controller, \( G \) is the incident solar radiation per m², \( A \) is the area of the collector, and \( Q_{\text{load}} \) is determined from

\[
Q_{\text{load}} = \sum_{n=1}^{365} Q_n \quad \text{for} \quad Q_n = \sum_{h=1}^{24} c_p \rho V_h (T_d - T_{\text{mains}})
\]

where \( T_d \) is the auxiliary setpoint temperature, (e.g., 50 or 55°C) and \( V_h \) is the hourly draw volume. \( T_{\text{mains}} \) is the monthly average mains water temperature.

Table 16. Annual simulation results for the 3 systems studied.

<table>
<thead>
<tr>
<th></th>
<th>Small Single Tank</th>
<th>Large Single Tank</th>
<th>Series Multi-Tank</th>
</tr>
</thead>
<tbody>
<tr>
<td>Collected Solar Energy (GJ)</td>
<td>11.04</td>
<td>29.33</td>
<td>31.60</td>
</tr>
<tr>
<td>Solar Energy Delivered to Load (GJ)</td>
<td>10.56</td>
<td>29.30</td>
<td>29.15</td>
</tr>
<tr>
<td>No Solar Load (GJ)</td>
<td>18.29</td>
<td>54.86</td>
<td>54.86</td>
</tr>
<tr>
<td>Storage Losses (GJ)</td>
<td>0.038</td>
<td>-0.613</td>
<td>0.861</td>
</tr>
<tr>
<td>Parasitic Energy* (GJ)</td>
<td>0.387</td>
<td>0.377</td>
<td>0.392</td>
</tr>
<tr>
<td>Solar Fraction</td>
<td>0.556</td>
<td>0.527</td>
<td>0.524</td>
</tr>
<tr>
<td>System Efficiency</td>
<td>0.353</td>
<td>0.312</td>
<td>0.337</td>
</tr>
</tbody>
</table>

* Energy consumption for controller and circulation pump.
Figure 107 illustrates the monthly solar energy delivered to the load for the three systems.

Figure 107. Solar energy delivered to the load.

In the process of completing this comparison, it became apparent that the solar fraction and the storage heat losses were dependant on the mains water temperature and the air temperature adjacent to the storage. Results showed that during the winter period, the cold portions of the storage tanks were heated from the surrounding air (assuming a 20°C environment adjacent to the storages). During the summer period, this situation is reversed and heat is lost from the storages to the surroundings.

This makes the selection of the optimum insulation level for these thermal storages a fairly complex issue that depends on the storage volume to surface area ratio, and the mains and room air temperature, etc. As well, if the positive gains to the thermal
storages, during the winter period, add to the building space-heating load, there may be little benefit to the building owner.

For the cases studied, the insulation level in the storage tanks was assumed to be equal \((U = 5 \text{ kJ/m}^2 \text{°C})\) which resulted in an annual net heat loss from the storage of 0.861 GJ for the series multi-tank system. At a similar insulation level, the large single storage showed a slightly negative net annual heat loss (i.e., corresponding to a net heat gain).

In both cases, the heat losses or gains from the surroundings to the storage units were less than 2% of the annual load energy requirement. It is also worth noting in Table 16 that the solar fractions and system efficiencies of both the large single tank and the series, multi-tank systems were very close. For the same insulation levels and similar heat exchanger capacities, solar fractions of 52.7 and 52.4 % were obtained, respectively, for the large single-tank and multi-tank configurations. An example of a TRNSYS deck, representative of the one used for the annual system performance analysis for Montreal, is shown in Appendix J.

A similar analysis was conducted for two US locations (i.e., Los Angeles, CA and Madison, WI) which were representative of different climatic regions. To estimate the performance of solar SDHWs in the USA, the Solar Rating and Certification Corporation (SRCC, 2006) publishes simulation results determined under specified environmental conditions, representative of different climatic zones within the USA. The operational and climatic conditions specified by SRCC were used as the basis for these simulations.

The results of this analysis conclude that the solar energy delivered to the load for the multi-tank system is very close to the large single tank, e.g., in Madison, the multi-tank system delivered 98.7% of the annual energy of the large single tank. In Los
Angeles, this value was 95.9%. Simulation results for the modeling of a multi-family SDHW system using the series-connected, multi-tank storage indicate that the annual performance is comparable to a system incorporating a single storage of equal volume. In Madison, for the same insulation levels and similar heat exchanger capacities, solar fractions of 62.9% and 62.0% were obtained for the large single-tank and multi-tank configurations, respectively. The corresponding solar fractions for Los Angeles were 68.2% and 65.4%. During the comparison, it became apparent that the solar fraction and the storage heat losses were dependant on the mains water temperature and the air temperature adjacent to the storage. Results showed that during the winter period, the cold portions of the storage tanks were heated from the surrounding air (assuming a 20°C environment adjacent to the storages). During the summer period, this situation is reversed and heat was lost from the storages to the surroundings.

The full description of this analysis and results are given by Cruickshank and Harrison (2007).
Chapter 7

Conclusions and Future Work

As a result of the research undertaken and described in this document, the following conclusions may be made.

7.1 Multi-tank Thermal Storage Concept

A novel design for a multi-tank thermal energy storage (TES) was described for both parallel and series configurations. The concept for the multi-tank storage is based on the inter-connection of standard hot water storage tanks through a single indirect charge flow loop. Each tank is fitted with a natural convection heat exchanger (NCHE) that is connected to a storage tank by a buoyancy driven thermosyphon loop. The flow-through each thermosyphon loop is governed by its temperature distribution and the average temperature in the storage tank. As such, each component storage tank is self-regulating and controlling, and heat transfer rates are dependent on the state of charge in the component thermal storage. As well, the array of storage tanks, when plumbed in the series configuration, was shown to sequentially stratify under a range of operational conditions. This arrangement has the advantage of allowing the use of low cost,
conventional hot water storage tanks, and its modular construction and incremental sizing make it ideally suited for retrofit situations where a large single tank thermal storage cannot be easily installed.

7.2 Numerical Modeling of Series- and Parallel-connected Multi-tank TES

To predict the performance of the series- and parallel-connected multi-tank thermal energy storage, a component and system level model of the multi-tank TES was developed in the TRNSYS simulation environment (TRNSYS, 2000). The model included individual numerical models of the storage tanks, the natural convection heat exchangers, the charge loop and the different charge scenarios.

To arrive at a suitable simulation model, various component models (i.e., TRNSYS types) were evaluated for modeling stratified liquid-based thermal storage, and natural convection heat exchangers. In the case of the stratified thermal storage, the resolution of the model, as set through the number of nodes (or constant temperature layers), was investigated relative to the accuracy of the results.

Validity of the 1-D Approach. The basic assumptions typically used in the computer modeling of solar storage (e.g., one dimensional temperature profiles, minimal wall conduction, uniform heat loss, etc.) were investigated in context to the thermal storages used in the multi-tank storage and their operation. Separate experimental tests were devised and conducted to obtain data on the suitability of these model assumptions. Results of the experiments indicted that, at the tank insulation levels and inlet flow rates studied, virtually no horizontal temperature gradients were identified.
Storage Overall “U-value”. To better quantify and investigate the heat loss characteristics of the storage tanks used in the multi-tank system (e.g., the average heat loss rate, the area weighted average U-value, etc.), a storage cool-down test was configured and conducted on a single tank. As a result of this evaluation, both individual node and tank average heat loss rates, and the area-weighted average U-value were determined. Based on these results, better estimates of the tank heat loss characteristics were made and introduced into the storage model.

Tank Wall Conductivity and Heat Diffusion. A test sequence was also conducted to determine the heat diffusion rate, the maximum temperature gradient and the thermocline profile within the storage tanks. The results of this testing indicated that by using an effective thermal conductivity based on an area-weighted parallel heat flow model for the axial conduction of the water and the tank wall (Newton, 1995), accurate temperature profiles could be modeled for the cool down tests.

Modeling of Natural Convection Heat Exchanger (NCHE). To model the operation of the natural convection heat exchanger, the original semi-empirical method was refined and new performance characteristics were derived by experimental testing and regression. A simple algorithm to predict NCHE performance was described and the new performance characteristics obtained for the natural convection heat exchangers used in the study were implemented in the TRNSYS model. The validity of the model was checked by comparison with experimental results.
7.3 Evaluation of Multi-tank Thermal Energy Storage

The performance of a medium sized, multi-tank thermal storage was investigated by experiment and computer simulation. Both simulation and measured test results show that high degrees of stratification can occur in both series and parallel multi-tank storages. In the case of the series-connected tanks, the results also indicate the feasibility of using side-arm, natural convection heat exchangers in a multi-tank storage system. Preliminary draw tests conducted for both the series and parallel configured TES indicated that mixing during draw-offs was not an issue and that the tank temperature profiles were well represented by simulation (Cruickshank and Harrison, 2006e).

7.3.1 Constant Temperature Charge Evaluation

Testing and simulation also indicated that under a constant temperature charge scenario, slightly higher storage rates were achieved with the parallel storage configuration relative to the series case. However, considerable difficulty was experienced to achieve a balanced flow distribution in the parallel configuration. Consequently, system designers may choose to utilize the series configuration. As well, the constant temperature charge results are not indicative of normal operational conditions and do not account for Second Law (e.g., exergy storage) factors. Constant charge test sequences are useful in identifying the functional operation and validity of the numerical model.


7.3.2 Variable Input Power “Daylong” Charge Sequences

Laboratory tests were conducted on the series-connected, multi-tank thermal storage to measure the unit’s performance and the tank temperature profiles resulting from varying charge conditions representative of hypothetical combinations of clear and overcast days. In particular, tests were performed to study the interaction of the individual tanks, and to investigate the effects of rising and falling charge loop temperatures and power levels on temperature profiles and heat transfer rates in the storage tanks. The results of this testing indicated that individual and sequential tank stratification occurred for the charge profiles studied.

Cool-down Mixing and Destatification. During the tests, a small amount of mixing was observed in the upper section of the storage tanks due to falling charge temperatures that occurred in the “afternoon” and “morning” periods. In addition, results indicated that a small amount of heat was also carried over from the high temperature storage to the lower temperature, downstream storages during charging. Test results also indicated that reverse thermosyphoning occurred when cooler fluid was circulated through the heat exchanger, on the charge loop (i.e., solar collector) side of the NCHE, cooling the water in the thermosyphon loop on the tank side.

Effects of Charge Loop (i.e., Collector) Flow Rate. Furthermore, the dependence of the heat transfer rate on the magnitude of the charge loop flow rate was also shown. For fixed charge power, reducing the charge loop flow rate resulted in higher charge supply temperatures that, in-turn, resulted in increased temperatures and pressure heads in the thermosyphon heat exchanger loop. The net result was that the thermal storage attained
higher temperatures and higher stratification levels at lower charge loop flow rates. Increasing charge loop flow rates resulted in lower storage temperatures initially, lower stratification and lower exergy levels when compared to the low-flow cases. Finally as predicted by theory, as collector loop flow rate was increased, the distinction between the series- and parallel-connected storages disappeared and heat exchanger charge rates and tank temperature profiles coincided.

**Reverse Thermosyphon and Carryover.** Results indicate that sequential stratification was observed for the charge profiles studied, however a small amount of heat was also carried over from the high temperature storages to the lower temperature storages. The carryover of heat from an upstream heat exchanger to a downstream heat exchanger cannot always be eliminated and is due to the limit of the heat exchanger effectiveness that results in higher charge temperatures being supplied to the downstream heat exchangers.

To investigate these effects, a numerical study was conducted to compare a TES with an ideal heat exchanger \((\varepsilon = 1)\) to the typical effectiveness values associated with the NCHE studied. The results illustrated that higher effectiveness heat exchanger values resulted in less carryover of energy to the downstream tanks and resulted in lower supply temperatures to the solar collector, thereby increasing the solar collector and overall system performance. The optimum sizing of heat exchangers is a trade-off between overall solar system performance and cost.

Carryover of energy also occurred due to adverse temperature and pressure gradients caused by low charge fluid temperatures. In a series-connected, multi-tank TES, reverse thermosyphoning in the upstream storage tank resulted in heat being removed
from the upstream tank and transferred to lower temperature downstream tanks. While much of the energy removed from the upstream storage was not lost but stored in the downstream tanks, there was a tendency to contribute to destratification in the upstream TES.

7.4 Refined Numerical Modeling of a Series-Connected Multi-tank TES

A comparison of experimental and simulated results for the series-connected multi-tank thermal storage indicated that its operation during periods of reverse thermosyphoning was not well modeled. As a consequence, an experimental and numerical study was conducted on a full scale experimental apparatus to investigate the effects of reverse thermosyphoning in the multi-tank TES. Results identified that reverse thermosyphoning can occur in storages using side-arm NCHEs, however the magnitude of the reverse flow depends on the height of the heat exchanger relative to its associated storage tank and the value of the net hydrostatic pressure head which depends on the difference in the average fluid temperatures (and the densities) of the fluid in the heat exchange loop and the storage tank.

Revised Numerical Model. To model the reverse flow through the heat exchanger loop, a new model was developed within the TRNSYS simulation environment that allowed energy quantities, heat exchanger and storage temperatures to be predicted with reasonable accuracy during periods of reverse thermosyphon operation. Revised values of stored energy predicted by the numerical model compared very closely to the experimentally measured values.
Revised Parallel Flow Characterization and Model for NCHEs. In order to develop the numerical model of reverse thermosyphon operation, it was first necessary to modify the test method and algorithm used to characterize the NCHE while operating in parallel flow mode rather than the typical counter flow case. This was done and new coefficients were determined for the parallel flow case and implemented in the new TRNSYS model described above.

Design Guidelines to Minimize Reverse Thermosyphoning. The revised numerical model was used to illustrate the effect of heat exchanger height, relative to the difference in average heat exchanger and TES temperatures, on reverse flow rate. It was observed that if the heat exchanger height was small relative to the storage tank, the reverse thermosyphoning is very low and can be considered negligible.

7.5 Second Law Analysis of the Multi-tank TES

To quantify the relative benefits of the sequentially stratified thermal energy storage, values of exergy stored versus time were determined for the test sequences studied. The Second Law of Thermodynamics provides a mechanism for quantifying any degradation in the “usefulness” of the energy that occurs during the storing process.

As a basis of comparison, the series and parallel configurations were also modeled and compared against two theoretical multi-tank configurations, i.e., fully stratified and fully mixed multi-tank storages. Results show that the series-connected configuration closely matched the exergy level of the fully stratified multi-tank case. At the higher collector flow rate, the exergy levels in both the series and parallel cases are reduced compared to the low flow case.
7.6 Annual Performance Prediction for a Multi-tank TES

To demonstrate the operation and feasibility of the multi-tank TES, annual performance simulations were conducted to model a multi-family solar domestic hot water system equipped with a series-connected, multi-tank storage. Results indicated that the annual performance was comparable to a system using a fully stratified single storage of equal volume. For the same insulation levels and similar heat exchanger capacities, solar fractions of 52.7 and 52.4 % were obtained, respectively, for the large single-tank and multi-tank configurations. The ultimate choice of whether to use a multi-tank thermal storage will depend on local climatic conditions, thermal load and building constraints.

7.7 Future Work

The focus of this study was to investigate the operation of an innovative, medium capacity, multi-tank thermal storage through testing and simulation, and to determine the key design and operational factors that affect the thermal stratification and system performance of the unit. Although this work assessed the stratification potential of a multi-tank thermal storage unit and produced a preliminary assessment of the feasibility of the multi-tank TES in a typical installation, a number of additional studies would add to the accuracy of the numerical model to predict the performance of the multi-tank storage system under a range of test conditions.

Laboratory tests were conducted on the series- and parallel-connected modular thermal storage to measure the unit’s thermal performance and temperature profiles under specified charge conditions. Due to resources and time constraints however, only limited
discharge tests were conducted. The potential for mixing exists during hot water draw-offs, promoting thermal destratification in the storage tank. Therefore, further charge and discharge scenarios and their interaction should be investigated.

In addition, stratification of a TES may also be destroyed by mixing caused by “plume entrainment” of the incoming liquid during charging. Plume entrainment occurs when cooler water is inserted into the top of the tank which contains warmer water. The resultant falling plume of cool water causes mixing. Preliminary results have indicated that increasing the collector loop flow rate leads to lower tank temperatures, lower stratification and lower exergy levels. Therefore it would be of interest to study the effects of high flow rates in the NCHE flow loop and compare strategies by which plume entrainment is minimized.

The experimental and simulation results also indicated that a high degree of stratification was achieved in the series-connected storage tanks, however, Tanks 2 and 3 stratify to a lesser degree than Tank 1. This was due to the initial heating of the “downstream” storage tanks during charging of the first tank. This effect could be reduced by increasing the size of the heat exchangers but the associated cost penalty may not be justified. Therefore, additional design modifications should be considered to optimize the heat exchanger size and increase overall system performance. Furthermore, it would be of great value to refine the multi-tank configuration to optimize its operation both on a First Law and Second Law basis. This would allow more useful energy to be available to meet a distributed load at a point earlier in the day.

Also, a challenge during modeling was to simulate the operation of the heat transfer loop in reverse thermosyphon operation, and in particular, it would be useful to
determine the temperature distribution in the circulation loop so that the density and net pressure head can be accurately determined. As such, it would be very relevant to this work to investigate the temperature distribution in the natural convection heat exchange loop in order to improve the accuracy of the model. Development of a new, robust TRNSYS model should be completed that has the capability to model positive and reverse thermosyphon flows.

It would be of great benefit to extend laboratory testing to cover longer periods that include weather and load variability. This should include the study of nighttime standby heat losses and their effect on the operation and stratification level of the TES. The potential for increased heat losses from the TES due to reverse thermosyphoning in the NCHE loop needs further study. Finally, it would be of great benefit to monitor the performance of a multi-tank TES in a realistic field installation.
References


Cruickshank, C.A. and Harrison, S.J.: “Investigation of Reverse Thermosyphoning in an Indirect SDHW System”, Accepted for Publication in the Proceedings of the ES 2009, Annual Meeting of the American Society of Mechanical Engineers (ASME), San Francisco, California (2009c)


EES: Engineering Equation Solver, University of Wisconsin Solar Energy Laboratory, Madison, Wisconsin (2009)


LabVIEW: *National Instruments LabVIEW*. Austin, TX, Version 8.0 (2005)


NATO Science Committee: Thermal Energy Storage, NATO Science Committee Conference, Turnberry, Scotland (1976).

Natural Resources Canada: National Energy Use Database. NRCAN, Office of Energy Efficiency, Ottawa, ON. (2005)


Appendix A

Previous Work on Stratification Indices
According to Panthalookaran *et al.* (2007), the methods currently used to characterize a TES may be broadly categorized as “(1) those based on the general dimensionless numbers of heat transfer and fluid dynamics, (2) those based on the First Law of Thermodynamics (energy based characterizations), (3) those based on the Second Law of Thermodynamics (entropy/exergy-based characterizations) and (4) those based on a combination of methods”. This approach will be followed in presenting a brief summary of the previous work on stratification indices.

**Stratification Indices Based on Dimensionless Numbers.** In the analyses of stratified tank problems, several dimensionless numbers arise (Dincer and Rosen, 2002). These include the Reynolds number, Grashof number, Richardson number, Froude number, Peclet number, Biot number and Fourier number. The reference quantities used in these numbers need to be established when interpreting the results in the literature as each author takes liberty in defining length, velocity, temperature and turbulence scales. Since a typical thermal energy storage (TES) process may involve a multitude of length, velocity, temperature and turbulence scales, it is difficult to keep perspective of the influence of one or the other design parameter on the global behavior of the TES (Panthalookaran *et al.*, 2007). Lavan and Thompson (1977) attempted to experimentally correlate the extraction efficiency of a TES in terms of an inlet Reynolds number, a tank Grashof number and the tank length to diameter ratio, Figure A.1. Lavan and Thompson concluded that stratification improved with increasing: length to diameter ratio; temperature difference between the incoming cold water and the stored hot water; and inlet and outlet port diameter. The authors further noted that stratification decreased with increasing flow rate.
The Richardson number has also found application in many correlations for TES efficiency. Given that the Richardson number is a measure of the ratio of buoyancy forces to mixing forces, a small Richardson number indicates a mixed storage tank and a large Richardson number is characteristic of a stratified storage tank. Sliwinski et al. (1978) found that the position and sharpness of the thermocline was a function of the Richardson and Peclet numbers and concluded that stratification did not occur below a critical Richardson value of 0.244. However, Cole and Bellinger (1982) observed a sharp decrease in stratification when the Richardson number was between 0.31 and 0.48. Zurigat et al. (1990) showed that inlet geometry had a strong influence on the thermal stratification for Richardson numbers below 3.6. Ghajar and Zurigat (1991) later stated that inlet geometry has no effect over a Richardson number of 10. van Berkel et al. (1999) concluded that the mixing in a two layer stratified storage was insignificant for
Richardson values of 10 to 20. Yee and Lai (2001) investigated the effect of thermal buoyancy by keeping the Reynolds number fixed and varying the Richardson number. The authors concluded that at low Richardson numbers, the mixing is due to jet entrainment and that stratification was only more evident at a Richardson number of 100. Kleinbach et al. (1993) and Hahne and Chen (1998) developed the modified Richardson number as the ratio between the Grashof number and the square of the Reynolds number. Hahne and Chen (1998) concluded that the thermal stratification in the storage depends mainly on the modified charging Richardson number and the Peclet number, Figure A.2. Cabeza et al. (2006) recently concluded that the Richardson number represented the behaviour of the water tank correctly when stratification was present and that the results obtained when using a charge flow rate of 3.5 and 4L/min were very similar suggesting that stratification would not be improved if the flow rate was further increased. A similar conclusion was drawn for the discharge test. Cabeza et al. (2006) also noted that the Peclet and Reynolds numbers were not good parameters to evaluate the stratification of a water tank but could be useful terms if used with other parameters that do not take into account the experimental flow rate.

Although the use of non-dimensional parameters is interesting, and these methods have been applied successfully for the evaluation of results from particular experiments, their application is limited to cases with only one thermocline (Dincer and Rosen, 2002).

**Stratification Indices Based on the First Law of Thermodynamics.** Stratification indices using the energy approach are based on the First Law of Thermodynamics and generally account for the heat transfer interactions between the storage and its surroundings. Energy-based indices can be classified as those based on the fraction of
recovered heat and those based on thermal storage efficiencies and charging/discharging efficiencies. Most notable, is the work by Lavan and Thompson (1977) who defined an extraction efficiency as the proportion of the tank that can be drawn off before the difference between the exit and inlet temperature decreased 10% from its original value. Similarly, Abdoly and Rapp (1982) introduced a parameter representing recoverable heat, which only considered heat that had not been degraded more than 20% of its original temperature value towards ambient temperature. Nelson et al. (1999) analyzed cold storage and calculated a percentage of cold recoverable in a similar fashion to the recoverable heat of Abdoly and Rapp (1982). Chan et al. (1983) analyzed thermal storage efficiencies of charging and discharging processes for a TES used for solar heating and cooling in buildings. The authors defined two charging and discharging efficiencies representing the actual energy change at time $t$ divided by the maximum energy change after ideal plug flow replacement of the entire storage volume. Therefore, a high charging or discharging efficiency represents very little mixing and low heat losses to the surroundings. Similar efficiency approaches were used by Yoo and Pak (1993), Mavros et al. (1994), Hahne and Chen (1998), Bouhdjar and Harhad (2002) and Shah et al. (2005). Davidson and Adams (1994b) defined a dimensionless number (i.e., the MIX number) that is based on the energy and temperature distribution level in the tank. The number is calculated as the momentum of energy difference between a perfectly stratified store and the actual store under investigation divided by the momentum of energy difference between a stratified storage and a fully mixed store (i.e., the MIX number is equal to 0 if the storage is fully stratified and equal to 1 if the storage is fully mixed). Both “theoretical” storage temperature profiles, stratified and fully mixed, are calculated
assuming full stratification and mixing, respectively, from the beginning of the experiment, and including heat losses to the surroundings with the heat loss coefficient for the experimental storage. Andersen et al. 2007 later adapted the “moment of energy” method, however in contrast to Davidson and Adams (1994b), the MIX number was calculated assuming the same energy content for all three storages (stratified, actual and fully mixed) at each time step, thus not requiring the heat loss coefficient for the experimental storage. Cabeza et al. (2006) concluded that the MIX number was not sensitive to the mass flow rate of the inlet water, resulting in similar MIX number values for seven different flow rates. In addition, for higher flow rates, the MIX number suggested high stratification during the discharge experiments which is contradictory to what was expected (i.e., high flow rates destroy stratification). Cabeza et al. (2006) also stated that the MIX number was very sensitive to small changes in inlet and outlet temperatures, producing variable results for small variations in temperature.

**Stratification Indices Based on the Second Law of Thermodynamics.** Stratification indices using the entropy/exergy-approach are based on the Second Law of Thermodynamics and generally useful when the energy stored will be used to produce work. Hence, a system with a high level of thermal stratification would be characterized by high exergy content. Entropy/exergy-based characterizations have found application in many correlations for TES efficiency. van Berkel et al. (1997) introduced a non-dimensional exergy term called the stratification efficiency based on the comparison of an actual experimental storage with the optimal case of a fully stratified storage and the worst case of a fully mixed storage. Rosengarten et al. (1999) also defined a non-dimensional exergy term called the stratification efficiency which corresponds to the
percentage of energy that is available for discharge from the storage in an isentropic process. Rosen et al. (1999) defined the exergy efficiency of a storage cycle as the percentage of exergy input that can be recovered in the output of the storage during a storage cycle. Shortly after, Rosen (2001) calculated the exergy content of a storage with reference ambient (dead-point) temperature of 283 K (10 °C) and compared this value to the energy content in the storage. Similarly to van Berkel et al. (1997), Homan (2003) defined the Entropy generation number as the ratio of entropy generated in stratified charging over the entropy generated in fully mixed charging. Inspired by van Berkel et al. (1997) and Rosengarten et al. (1999), Shah and Furbo (2003) defined a new entropy efficiency and an exergy efficiency for storage tank discharging based on the exergy of an actual storage and the exergy of an ideally stratified storage.

**Stratification Indices Based on the Combined Approach.** Panthalookaran et al. (2007) have recently presented a new method for the characterization of a stratified thermal energy store based on efficiency definitions that include terms for both the First Law and Second Law of Thermodynamics. These are: the energy response factor and the entropy generation ratio, respectively. Panthalookaran et al. (2007) concluded that “a TES characterization scheme based purely on the First Law of Thermodynamics either makes the real quality of the stored energy obsolete or treats it only partially. An accurate characterization of a TES, therefore, requires a strategy based also on the Second Law of Thermodynamics. A product of these two quantities is at the heart of the TES efficiency definitions.”

As shown, numerous authors utilize stratification indices and figures of merit to characterize the key performance elements of a thermal storage. Specifically, the exergy
analysis is of particular interest to this work as it accounts not only for energy stored but also for the temperature at which energy is stored (Rosen 2001, Rosen and Dincer 2003). The exergy stored in the tank, during charging and discharging periods, is calculated from the exergy of each discrete water layer considered in the analysis (Han et al. 2008). This is of importance when optimizing the design and configuration of a solar storage in a SDHW system.

The studies presented and discussed in this literature are not all directly relevant to the study, however, the results and methodologies of these studies all add significantly to the knowledge and background associated with the factors that influence stratification and the characterization of it.

For this study, it was found most useful to determine both the increase in exergy and final exergy level that exists in the storage relative to a reference state, e.g., the mains water temperature. Clearly, high exergy levels occur from higher temperatures and stratification levels, and indicate that the “quality” of the thermal energy has not been lowered by mixing and diffusion during the charging process.

With regard to the case of the non-dimensional parameters, Re and Ri numbers appear to be the most widely used. Calculations of the Re number show that, at typical flow rates experienced in the NCHE loop, the flow can be considered laminar. In addition, calculations of Ri result in high values, indicating that buoyancy forces dominate and high stratification levels should occur.
Appendix B

Governing Equations
B.1 Governing Equations

To determine heat transfer and fluid flow in a control volume it is necessary to know the velocity, temperature and pressure distribution. The classical equations governing fluid flow and heat transfer for three-dimensional, steady-state, incompressible flow with constant thermal properties, for laminar forced convective flow are: the conservation of mass; the conservation of momentum; and the conservation of energy (Turns, 2006).

**Conservation of Mass.** The rate at which energy leaves a control volume must equal the rate at which it enters, as described by the continuity equation.

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \tag{B.1}
\]

**Conservation of Momentum.** The net force acting on a control volume in any direction is the difference between the rate at which momentum leaves and enters in this direction. This is described by the so called Navier-Stokes equations for the \( x \)-, \( y \)- and \( z \)-directions.

\[
u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \nu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \tag{B.2}
\]

\[
u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial y} + \nu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) \tag{B.3}
\]

\[
u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial z} + \nu \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) \tag{B.4}
\]
Conservation of Energy. The conservation of energy within the control volume is described by

\[
\frac{u}{\partial x} + \frac{v}{\partial y} + \frac{w}{\partial z} = k \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \tag{B.5}
\]

The continuity and Navier-Stokes equations are used to calculate the velocity field. These equations are then used with the energy equation to obtain the temperature field. For most cases, these equations are usually solved numerically i.e., through a process of discretization, these governing equations are transformed into algebraic equations that, with the specifications of the appropriate boundary conditions, are solved numerically for a particular region.
Appendix C

Heat Loss Characteristics for a Typical SDHW System
For solar heating systems, the accurate estimation of storage heat losses is important in the prediction of annual energy performance and product performance ratings. This process relies on the accurate specification of the system’s physical and thermal characteristics, and is often based on a number of simplifying assumptions. In computer simulation programs, it is common practice to represent the rate of heat loss from a thermal energy storage with an overall or average U-value, however, this value does not account for the complex geometry of the thermal storage or the interaction of the various inlet and outlet ports that may act as thermal conduits. In addition, most solar storage models assume that the tank temperature profile is one-dimensional and that conduction within the tank wall is negligible. These assumptions may lead to errors in heat loss calculations and errors in the estimation of the temperature distributions within the storage tank.

To investigate these effects, an experimental program was undertaken to measure the rate of heat loss from a typical thermal energy storage used in a solar domestic hot water system. During the course of this investigation, measurements were taken to verify the predicted temperature field within the storage, including the vertical temperature distribution and the horizontal temperature distribution taken at two locations within the thermal energy storage.

Specifically, testing was conducted to investigate if the temperature profile in the storage tank was effectively 1-D; if the heat losses from the thermal storage can be modeled by an overall or average U-value; and if the thermal diffusion in the storage tanks can be modeled by a simple 1-D conduction algorithm between tank layers. Tests were conducted on an isolated storage tank and the experimental apparatus and analysis
was configured to only evaluate the above effects. Specifically, mixing of the storage fluid due to flow into, or out of, the tank was not considered in this study and is the focus of other ongoing investigations.

C.1 Experimental Analysis

Testing was conducted on a commercially available, glass lined steel, 270 L standard electric water heater storage tank (GSW 6E2270SC) which is routinely used as a solar storage in SDHW packages, Figure C.1. The storage was insulated with 0.047 m of fiberglass insulation with an estimated conductivity, \( k = 0.036 \text{ W/m°C} \) (ASHRAE, 1997). For use as a solar storage, the electric heater elements in the storage tank are disabled and the tank is only used to store solar heated, mains water. Normally this solar preheated water is supplied to an existing auxiliary heater.

![Figure C.1. Schematic of experimental rig showing the a) vertical temperature probe and b) horizontal temperature probes.](image)
The apparatus used for this study consists of a disabled standard electric hot water heater (typically used as a solar preheat storage) plumbed to a temperature controlled charge loop, Figure 35. Temperature probes were installed in the storage tanks to measure the vertical temperature distribution at 0.15 m intervals and horizontally at 0.05 m intervals. The horizontal probes we inserted in the storage tank through the opening normally used for the electric immersion heaters, Figure C.2. The temperature probes and data system were calibrated and determined to be repeatable (i.e., precise) to within ±0.02°C over the test range. A computer based data acquisition system was used to record tank fluid and ambient air temperatures at approximately 3 minute intervals during the test periods. Testing was completed to determine both local and the average, effective heat loss coefficients by conducting a “cool-down” test similar to the “Heat Loss Test (Standard Decay Method)” described in Section 7.4 of the SRCC Document TM-1 (SRCC, 2006). In addition a thermal diffusion test sequence was completed to assess the effects of fluid and tank wall conduction on the temperature distribution in the storage tank over an extended period.
C.1.1 Measurement of Heat Loss Coefficient (Cool-down Test)

Standard methods have been proposed to evaluate standby heat loss coefficients in conventional domestic hot water heaters but these methods are not directly applicable for solar storages that do not have integral auxiliary heaters. Therefore, a cool down test or “Heat Loss Test” (SRCC, 2006) was carried out to investigate the storage under evaluation. The test was performed over a 48 hour period on the fully charged storage to determine its heat loss characteristics (e.g., the average heat loss rate, the area-weighted average U-value). The water was initially pre-heated and mixed to a uniform temperature of 54.0°C by an electric-heated charge loop and circulating pump. At the start of the test, the heater and pump were turned off, and tank temperatures were subsequently recorded at 3 minute time intervals for a 48 hour period. The ambient temperature was also recorded during the test sequence and remained constant at approximately 20.0 ± 0.3°C.
At the end of the 48 hour period, the storage was once again mixed to within ± 0.03°C over a ten minute period and the tank temperatures were recorded. Average rates of energy loss from the overall storage and from different segments (or nodes) were calculated on a six hour basis over the duration of the 48 hour test. From this data, the area-weighted average U-value for each node and the overall tank were calculated.

C.1.2 Verification of 1-D Assumption (Temperature Field Measurements)

To investigate the horizontal variation in temperature across the storage, horizontal temperature data was recorded at 3 minute intervals during the cool down test described above. Data was analyzed to determine its uniformity and the magnitude of the variation in the horizontal direction for both the upper and lower sections of the storage tank.

C.1.3 Measurement of Heat Diffusion

A heat diffusion test was conducted over a 40 hour period on the storage to determine its heat diffusion characteristics (e.g., the maximum temperature gradient, thermocline profile, etc.). To prepare for this test, the top half of the tank was initially pre-heated to a temperature of 56.0°C by an electric-heated charge loop and the bottom half of the tank was set to 4°C. Ambient air and storage tank temperatures were recorded at 3 minute time intervals over the 40 hour test interval. The ambient air temperature remained constant at approximately 20.3 ± 0.3°C over this period.

C.2 Computer Modeling

The average heat loss rate was determined using the energy balance method described in Chapter 3 (Figure 23, Figure 24, and Equation (3.1)), for each node and the
overall tank. Since there are no flows entering or exiting the tank, Equation (3.1) reduces to:

\[
M \cdot \frac{dT}{dt} = \frac{(k + \Delta k)A}{\Delta x_{i+1-i}} (T_{i+1} - T_i) + \frac{(k + \Delta k)A}{\Delta x_{i-1-i}} (T_{i-1} - T_i) + (U_i)A_{s,i}(T_{env} - T_i) \quad (C.1)
\]

The area-weighted average U-value for each node and the overall tank are calculated as:

\[
U_i = \frac{Q_{loss,i}}{A_{s,i}(T_i - T_{env})} \quad \text{and} \quad \bar{U}_{tank} = \frac{Q_{loss,tank}}{A_{s,tank}(T_{mean,tank} - T_{env})}, \quad \text{respectively.} \quad (C.2)
\]

C.3 Results and Discussion

Tests were conducted on a single tank storage system according to procedures similar to that presented in the SRCC Document TM-1 (SRCC, 2006) and included a cool-down test and a heat diffusion test sequence. The values derived from these test sequences were then compared to computer predictions based on typical manufacturer’s data. In addition, the basic assumptions typically used in the computer modeling of solar storage heat losses (e.g., one dimensional temperature profiles, minimal wall conduction, uniform wall heat loss, etc.) were investigated in the context of a thermally stratified storage.

C.3.1 Cool Down Test Results

The results of the cool down test are shown in Figure C.3, where vertical temperature measurements are plotted over the duration of the test sequence. These results show that the bottom of the storage dropped in temperature rapidly followed by the two temperature sensors located in the region directly above. The remaining sections of the tank remained at a uniform temperatures that was observed to drop at a fairly
constant rate during the test period. This separation of thermal layers (i.e., thermal stratification) exists as a result of density variations within an unmixed tank. The measurements recorded by the horizontal probes were consistent with these observations, as shown in Figure C.4 and C.5. As shown in Figure C.5a) and b), it is evident that there are virtually no temperature gradients in the horizontal direction within the measurement precision of the temperature data system estimated to be ± 0.02°C. This strongly supports the 1-D approach used in the numerical analysis. These results, however, should be viewed with some caution as the temperature field within the tank can be highly affected by high velocity incoming fluid streams (Hollands and Lightstone, 1989).

Figure C.3. Vertical temperature profile measured during cool down test as a function of time.

Figure C.4. Horizontal temperature profile measured during cool down test as a function of time.

Figure C.5. Horizontal probe temperatures measured during cool down test as a function of distance from the tip of a) the upper probe and b) the lower probe.
The average heat loss rate was determined using the energy balance method for each node and for the overall tank, and is shown in Figure C.6, at 6 hour increments. It is apparent that the average heat loss rate is greater at the beginning of the test since there is a high temperature difference between the tank and the surrounding air temperature. Also of interest is the average heat loss rate for the bottom node of the tank (e.g., Node 9). It is observed that a large percentage of the heat loss to the surroundings is through the bottom of the tank. This is most likely a result of the fact that less insulation was located at the bottom of the tank and resulted in heat conduction from the adjacent regions of the tank to the bottom. As expected, the average heat loss rate diminished as the test continued since the temperature difference between the tank and the surroundings was reduced. The remaining heat loss rates for Nodes 1 to 8 appear to be reasonably constant over the 48 hour sequence. The area-weighted average U-values for each node and the overall tank were also determined for the test sequence, Figure C.7. The bottom node has the highest U-value since this section had the highest heat loss rate per area during the test sequence. All the U-values remained fairly constant over the test period. Table C.1 shows the average heat loss rates and U-values for each node and the overall tank as calculated from the cool-down test sequence.

To arrive at an overall effective U-value for the storage, two approaches where taken: first, an area weighted average U-value \( (U_{\text{EFF \_EXP}}) \) was determined from the node values as listed in Table C.1; secondly, an effective U-value based on a log-mean temperature difference \( (U_{\text{LMTD \_EXP}}) \) was calculated (SRCC, 2006) as

\[
U_{\text{LMTD \_EXP}} = \frac{m_{\text{tank}} C_p}{A_{\text{tank}} \cdot \text{time}_{\text{decay}}} \ln \left[ \frac{(\bar{T}_{\text{tank \_initial}} - T_{\text{env}})}{(\bar{T}_{\text{tank \_final}} - T_{\text{env}})} \right] \tag{C.3}
\]
where the heat loss from the storage tank $E_{\text{loss,tank}}$ is given by

$$E_{\text{loss,tank}} = m_{\text{tank}} \cdot C_p \cdot (\bar{T}_{\text{tank initial}} - \bar{T}_{\text{tank final}}) \quad (C.4)$$

where $\bar{T}_{\text{tank initial}}$ is the mean temperature of the mixed storage at the start of the cool-down test and $\bar{T}_{\text{tank final}}$ is the mean temperature of the mixed storage at the end of the test.

Tabulated values of these quantities are given in Table C.1.

![Figure C.6. Average heat loss rate determined using the energy balance method, for each node and the overall tank.](image)

![Figure C.7. Area-weighted U-value determined from the average heat loss rate, for each node and the overall tank.](image)
Table C.1. Summary of measured average heat loss rates and area-weighted U-values for each node and the overall tank.

<table>
<thead>
<tr>
<th>Cool-down Test</th>
<th>Surface Area (m²)</th>
<th>Average Heat Loss Rate (W)</th>
<th>Area-weighted Average U-value (W/m²°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Node 1</td>
<td>0.34</td>
<td>48.91</td>
<td>0.66</td>
</tr>
<tr>
<td>Node 2</td>
<td>0.25</td>
<td>48.70</td>
<td>1.13</td>
</tr>
<tr>
<td>Node 3</td>
<td>0.25</td>
<td>48.75</td>
<td>1.13</td>
</tr>
<tr>
<td>Node 4</td>
<td>0.25</td>
<td>48.74</td>
<td>1.13</td>
</tr>
<tr>
<td>Node 5</td>
<td>0.25</td>
<td>47.90</td>
<td>1.11</td>
</tr>
<tr>
<td>Node 6</td>
<td>0.25</td>
<td>41.53</td>
<td>0.95</td>
</tr>
<tr>
<td>Node 7</td>
<td>0.25</td>
<td>33.54</td>
<td>0.79</td>
</tr>
<tr>
<td>Node 8</td>
<td>0.25</td>
<td>44.36</td>
<td>1.16</td>
</tr>
<tr>
<td>Node 9</td>
<td>0.34</td>
<td>143.83</td>
<td>2.54</td>
</tr>
<tr>
<td>Tank</td>
<td>2.43</td>
<td>56.25</td>
<td>U_{EFF, EXP} = 1.21</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>U_{LMTD, EXP} = 1.26</td>
</tr>
</tbody>
</table>

The results of the testing indicate that the measured total heat loss, \( E_{loss, tank} \), was 14580 kJ over the cool-down test, and \( U_{EFF, EXP} \) and \( U_{LMTD, EXP} \) were 1.21 W/m² °C and 1.26 W/m² °C, respectively. In analyzing the results of both the experiment and the modeling it was considered worthwhile to compare the overall U-values as measured during the cool-down test with values that would normally be used as an input to the computer simulation. The simulation program uses the conductance of the tank’s thermal insulation as an input on which the tank losses are estimated. It is common practice in computer programs to apply a simple one-dimensional analysis for heat loss through the sides of the storage tank, assuming the U-value is similar on all surfaces. For example, the storage tank evaluated was insulated with 0.05 m of fiberglass batt insulation on the sides and top but the bottom of the storage was un-insulated and was resting on an elevated 19 mm plywood floor. A typical conductance for the insulation is estimated at 0.036 W/m °C (ASHRAE, 1997). As shown in Table C.2, assuming the thermal
resistance on the water side to be negligible, and assuming that the heat transfer coefficients on the exterior of the storage were 9.26, 8.29 and 6.13 W/m²°C for the top, sides and bottom of the tank, respectively (ASHRAE, 1997), a simple area-weighted estimated overall conductance through the tank walls would be approximately 0.86 W/m²°C. It is immediately evident that this value is lower than the overall U-value measured during the cool-down test; which was estimated at 1.21 W/m²°C or 1.26 W/m²°C based on a log-mean temperature difference. This result is not unexpected as the overall heat loss rate and effective U-value should be higher than the value predicted from the thermal insulation alone. This is due to the fact that the tank also has a number of inlet and outlet ports that act as thermal conduits or short-circuits through the insulation, increasing the overall heat loss from a storage tank.

If the previously determined U-values are used as inputs to the computer simulation of the cool-down test sequence, the simulated heat loss values given in Table C.3 result. From these results, it is evident that when the individual node U-values derived from the cool-down test are used, the correspondence between measurement and simulation is very close. However when the tank’s average U-values, as determined from the both the area-weighted average of the node U-values and the overall effective U-value based on the log-mean temperature difference, are used, the discrepancy between measured and simulated heat loss increases.

All of these values are much closer than the estimated heat loss obtained by using the U-value estimated based on the storage tank’s thermal properties, (i.e., $U_{\text{EFF.TH}} = 0.86$) which underpredicts the storage heat losses by almost 20%. To investigate the magnitude of the underestimation of the wall heat loss properties, the simulation was run with
various average wall U-values until the estimated heat loss corresponded to the experimental value. This process predicted that the average storage wall U-value, $U_{WALL\_ESTIMATE}$, would need to be 1.13 W/m²°C, an increase in U-value of approximately 30%.

Table C.2. Summary of average heat loss rates and area-weighted U-values calculated for each node and the overall tank using tank and insulation properties.

<table>
<thead>
<tr>
<th>Tank and Insulation Properties</th>
<th>Surface Area (m²)</th>
<th>Film Coefficient, $h_o$ (W/m²°C)</th>
<th>U-value Insulation (W/m²°C)</th>
<th>Area-weighted Average U-value (W/m²°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Top Surface</td>
<td>0.18</td>
<td>9.26</td>
<td>0.766</td>
<td>$U_{TOP} = 0.71$</td>
</tr>
<tr>
<td>Side Surface</td>
<td>2.25</td>
<td>8.29</td>
<td>0.766</td>
<td>$U_{SIDE} = 0.70$</td>
</tr>
<tr>
<td>Bottom Surface</td>
<td>0.18</td>
<td>6.13</td>
<td>0.766</td>
<td>$U_{BOTTOM} = 3.06$</td>
</tr>
<tr>
<td>Tank</td>
<td>2.43</td>
<td></td>
<td></td>
<td>$U_{EFF_TH} = 0.86$</td>
</tr>
</tbody>
</table>

Table C.3. Summary of measured and simulated average heat loss rates for various area-weighted U-values.

<table>
<thead>
<tr>
<th>Measured U-Values (Cool-down Test)</th>
<th>Area-weighted Average U-value (W/m²°C)</th>
<th>Measured Total Heat Loss (kJ)</th>
<th>Simulated Total Heat Loss (kJ)</th>
<th>(Simulated-Measured) / Measured</th>
</tr>
</thead>
<tbody>
<tr>
<td>$U_{NODGES_EXP}$</td>
<td>$U_{EFF_EXP} = 1.21$</td>
<td>14580</td>
<td>14755</td>
<td>0.01</td>
</tr>
<tr>
<td>$U_{MTD_EXP} = 1.26$</td>
<td>$U_{EFF_EXP} = 1.21$</td>
<td>14580</td>
<td>15406</td>
<td>0.06</td>
</tr>
</tbody>
</table>

C.3.2 Heat Diffusion Test Results

A test was conducted over a 40 hour period to determine the rate of the heat diffusion within the storage. For this test, the top half of the tank was pre-heated to a temperature of 56.0°C and the bottom half was initially set to 4.0°C. The ambient air temperature remained constant at approximately 20.3 ± 0.3°C over this period. The storage was then isolated and allowed to equalize over a 40 hour test interval. During this
time, storage tank temperatures were recorded at 3 minute intervals and are shown in Figure C.8. It is evident from this result that as time progresses, the tank tends toward a uniform temperature and that the temperature gradient within the tank reduces over time.

To simulate the diffusion results, it was necessary to include an incremental $\Delta k$ term to the water conductivity value to account for heat conduction within the steel wall of the storage tank that enhanced heat transfer from the top of the storage to the bottom (Newton, 1995). This approximate method increases the effective conductivity, $k_{\text{water}}$, of the storage tank fluid in the vertical direction by treating the heat transfer between nodes, and through the wall, as parallel conduction paths (TRNSYS, 2000). Within the simulation, the conductivity of the fluid is increased by the cross-sectional, area-weighted conductivity of the wall material, i.e.,

$$k_{\text{effective fluid}} = k_{\text{water}} + k_{\text{steel}} \frac{A_{\text{cs_wall}}}{A_{\text{cs_water}}}$$

(C.5)

For this calculation the cross sectional area of the steel tank was taken as the perimeter of the tank (i.e., 1.5 m) multiplied by a wall thickness of 0.004 m, giving $A_{\text{cs_wall}} = 0.006$ m$^2$. For an $A_{\text{cs_water}}$ (the horizontal cross-sectional area of the tank interior) equal to 0.18 m$^2$ and $k_{\text{steel}} = 50$ W/m $^\circ$C, the value of $k_{\text{effective fluid}}$ was estimated at 8.2 W/m $^\circ$C.

Using this effective conductivity, the vertical temperature profile was simulated over a 40 hour period and the results are shown in the center and right hand side of Figure C.8, based on the nodal and storage wall estimate U-values determined from the cooldown test. In these cases, the experimental and simulated results agree closely. As can be seen from the results, vertical conduction within the storage fluid and the tanks walls will
reduce stratification within the storage and would ultimately result in the storage equalizing or de-stratifying to a single uniform temperature.

![Figure C.8. Vertical temperature profile of tank as a function of time.](image)

De-stratification is generally undesirable in a solar heating system where it is beneficial to supply water at the highest temperature to the load. One measure of the degree of stratification is the magnitude and thickness of the temperature thermocline between hot and cold regions of a stratified thermal storage (Al-Najem, 1993), Figure C.9. This effect is shown in Figure C.10 where the tank temperature profiles are shown as function of test time for both the measured and simulated cases. Values of this parameter (i.e., $dT/dy$), derived from the experimental and simulated data, are also shown as a function of test time in Figure C.11 and it can be clearly observed that the temperature gradient decreases and tends toward a zero value, consistent with a uniform temperature. Here, measured and simulated values correspond well.
Figure C.9. Illustration of the thermocline in a stratified storage tank.

Figure C.10. Thermocline profile of the storage tank as a function of position in the tank.

Figure C.11. Maximum temperature gradient within the storage tank.
Appendix D

Conventional Heat Exchanger Systems
The performance of heat exchangers operating under forced flow conditions is defined by the amount of heat transferred between the two fluid streams and is characterized by the various factors: the effectiveness, $\varepsilon$, the $UA$ value or number of transfer units ($NTU$'s), and the capacity ratio, $C_r$, (Turns, 2006).

The performance of heat exchangers can be calculated by defining a control volume, incorporating the inlet and outlet of the heat exchanger and applying an energy balance on the system, Figure D.1.

$$Q = \dot{m}_c (T_2 - T_3) = \dot{m}_c (T_1 - T_4)$$

(D.1)

where $\dot{m}c_p$ is the heat capacity rate of one of the fluid streams.
The heat exchanger effectiveness, $\varepsilon$, is defined as the ratio of the rate of heat transfer in the exchanger, to the maximum theoretical rate of heat transfer, $Q_{\text{max}}$, i.e.,

$$
\varepsilon = \frac{Q}{Q_{\text{max}}}
$$

(D.2)

The maximum theoretical rate of heat transfer is limited by the fluid stream with the smallest heat capacity rate, i.e.,

$$
\varepsilon = \frac{(\dot{m}c_p)_s(T_2 - T_3)}{(\dot{m}c_p)_{\text{min}}(T_1 - T_3)}
$$

(D.3)

where the $(\dot{m}c_p)_{\text{min}}$ is the smaller of $(\dot{m}c_p)_s$ or $(\dot{m}c_p)_c$.

The unit’s $UA$ value is defined as the product of the overall heat transfer coefficient and the heat transfer area. For counter-flow applications, the heat transfer rate is defined as the product of the $UA$ value and the log-mean temperature difference, $LMTD$, i.e.,

$$
Q = UA \cdot LMTD
$$

(D.4)

where the log-mean temperature difference is equal to,

$$
LMTD = \frac{\Delta T_{\text{out}} - \Delta T_{\text{in}}}{\ln\left(\frac{\Delta T_{\text{out}}}{\Delta T_{\text{in}}}\right)}
$$

(D.5)

for counter flow units, $\Delta T_{\text{out}} = (T_1 - T_2)$ and $\Delta T_{\text{in}} = (T_2 - T_4)$ (Elliot, 1994).

The number of transfer units ($NTU$) is an indicator of the actual heat-transfer area or physical size of the heat exchanger. The larger the value of $NTU$, the closer the unit is to its thermodynamic limit. It is defined as,
The capacity ratio, $C_r$, is representative of the operational condition of a given heat exchanger and will vary depending on the geometry and flow configuration (parallel flow, counterflow, cross flow, etc.) of the exchanger. This value is defined as the minimum heat capacity rate divided by the maximum heat capacity rate, i.e.,

$$C_r = \frac{(mc_p)_{\min}}{(mc_p)_{\max}}$$  \hspace{1cm} (D.7)

It is important to note that the capacity ratio will be directly proportional to the ratio of the mass flow rates if the specific heats of the flows are equivalent.
Appendix E

Empirical Correlation of NCHE Performance Characteristics
Under ideal charge conditions, the operational characteristics of a natural convection heat exchanger (NCHE) and the associated thermosyphon loop can be measured. It was shown by Lin et al. (2000) that the performance of a NCHE could be described by relating the pressure head to the thermosyphon mass flow rate and the modified effectiveness to the modified capacity ratio, i.e.,

\[ \dot{m}_s = a \cdot (\Delta P)^b \]  
\[ \varepsilon = c \cdot C_{r_{mod}}^2 + d \cdot C_{r_{mod}} \]  

(E.1)  
(E.2)

where \( \dot{m}_s \) is the thermosyphon or natural convection flow rate, and \( a, b, c \) and \( d \) are constants empirically derived from a regression through experimental data. These constants will differ for each individual heat exchanger tested.

These relations allow performance coefficients for simple empirical expressions to be determined (e.g., \( a, b, c \) and \( d \)) and used as inputs to a general simulation routine, allowing the overall system performance to be determined for various loads and climatic conditions.

As such, laboratory tests were conducted on one of the NCHE/storage tank combinations used in the multi-tank storage prototype in an effort to measure the unit’s thermal performance and temperature profiles under constant temperature charge conditions. Using the experimental data, values of thermosyphon flow rate are plotted versus pressure head (Figure E.1 and Figure E.2), calculated according to Equation (3.6) as proposed by Lin et al. (2000), (Cruickshank and Harrison, 2006a, 2009a). Data is shown for tests conducted at three collector loop flow rates. To increase the clarity of the plots, test data (derived during a specified charge sequence) were averaged and plotted.
Figure E.1 and Figure E.2 illustrate that the data falls on a single characteristic curve. Only the data corresponding to the initial charge region (shown as Section A in Figure E.3) was used in the analysis. Section B shown in Figure E.3, represents the period of the test when the solar storage becomes fully charged. During this period, the heat transfer rate, the $UA$ value, and the thermosyphon flow rate were observed to drop dramatically (Cruickshank and Harrison, 2006a, 2009a). Further heating of the storage at this point would result in an increase in collector-loop temperature and a further increase in storage tank temperature. In effect, thermal storage would be seen to go through a second charge cycle.

To gain an insight into the repeatability of the test procedure, one of the test conditions was repeated 5 times. The mean of these tests produced a heat exchanger effectiveness of 0.629 with a standard deviation of ± 0.002.

Nonlinear regression curves were fitted (by the method of least squares) to the experimental data as shown in Figure E.1 and Figure E.2 according to the recommendations of Lin et al. (2000). The corresponding curve fits are

$$\dot{m}_s = 0.0396 \Delta p^{0.6508} \quad \text{and} \quad \varepsilon_{\text{mod}} = -0.3225 C_{r_{\text{mod}}}^2 + 1.1215 C_{r_{\text{mod}}}.$$ 

It should be noted that the coefficients obtained are specific to the particular heat exchanger under test and depend on the geometry and pressure characteristics of the thermosyphon loop and heat exchanger. As such, a new test sequence should be undertaken if the configuration is changed.

The data shown, however, does cover a range of operating conditions and should represent the performance of the heat exchanger under a variety of storage tank charge conditions and collector-loop flow rates and temperatures.
Figure E.1. Plot of experimental results showing the dependence of the thermosyphon flow rate on the pressure head.

Figure E.2. Experimentally derived modified effectiveness as a function of modified capacity ratio.
A typical data set for constant temperature charging is shown in Figure E.3, showing the storage tank temperature profile.

Figure E.3. Typical temperature profile for constant temperature charge.
Appendix F

TRNSYS Input and Parameters
As described in Chapter 3, the operation of the multi-tank storage was modeled in TRNSYS, ver.15 (2000). The simulation “deck” was developed from pre-existing and custom component types. Typical parameter and input values used in this study are shown in Tables F.1 and F.2 for the TYPE 60 subroutine, and in Tables F.3 to F.6 for the custom heat exchanger subroutines (i.e., TYPE 76 and 77).

Table F.1. Simulation parameter values for the TYPE 60 TRNSYS subroutine.

<table>
<thead>
<tr>
<th>Parameter No.</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Mode</td>
<td>2</td>
</tr>
<tr>
<td>2</td>
<td>$V_{\text{TANK}}$</td>
<td>270</td>
</tr>
<tr>
<td>3</td>
<td>$H_{\text{TANK}}$</td>
<td>1.5</td>
</tr>
<tr>
<td>4</td>
<td>per</td>
<td>-1</td>
</tr>
<tr>
<td>5</td>
<td>$H_{\text{in}}$</td>
<td>1.5</td>
</tr>
<tr>
<td>6</td>
<td>$H_{\text{out}}$</td>
<td>0</td>
</tr>
<tr>
<td>7</td>
<td>$H_{\text{in}}$</td>
<td>0</td>
</tr>
<tr>
<td>8</td>
<td>$H_{\text{out}}$</td>
<td>1.5</td>
</tr>
<tr>
<td>9</td>
<td>$C_p$</td>
<td>4.184</td>
</tr>
<tr>
<td>10</td>
<td>$\rho$</td>
<td>1000</td>
</tr>
<tr>
<td>11</td>
<td>$U_{\text{TANK}}$</td>
<td>5</td>
</tr>
<tr>
<td>12</td>
<td>$k$</td>
<td>2.1996</td>
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<tr>
<td>13</td>
<td>$\Delta k$</td>
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<td>14</td>
<td>$T_{\text{Boil}}$</td>
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<td>$\Delta T_{\text{db1}}$</td>
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<td>$Q_{\text{aux1}}$</td>
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<td>21</td>
<td>$H_{\text{aux2}}$</td>
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<td>1.2</td>
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<td>23</td>
<td>$T_{\text{set2}}$</td>
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<td>24</td>
<td>$\Delta T_{\text{db2}}$</td>
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<td>25</td>
<td>$Q_{\text{aux2}}$</td>
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<td>$U_{\text{Flue}}$</td>
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<tr>
<td>32</td>
<td>UMode</td>
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Table F.2. Simulation input values for the TYPE 60 TRNSYS subroutine.

<table>
<thead>
<tr>
<th>Input No.</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$m_{1\text{in}}$ Mass flow</td>
<td>[10, 2] Mass flow rate value from routine Pipe to Tank (TYPE 31)</td>
</tr>
<tr>
<td></td>
<td>rate of entering fluid 1</td>
<td>(kg/hr)</td>
</tr>
<tr>
<td>2</td>
<td>$m_{1\text{out}}$ Mass flow</td>
<td>Value same as Input 1</td>
</tr>
<tr>
<td></td>
<td>rate of exiting fluid 1</td>
<td>(kg/hr)</td>
</tr>
<tr>
<td>3</td>
<td>$m_{2\text{in}}$ Mass flow</td>
<td>Value same as Input 4</td>
</tr>
<tr>
<td></td>
<td>rate of entering fluid 2</td>
<td>(kg/hr)</td>
</tr>
<tr>
<td>4</td>
<td>$m_{2\text{out}}$ Mass flow</td>
<td>[11, 2] Mass flow rate value from routine Mixture for Hot Water</td>
</tr>
<tr>
<td></td>
<td>rate of exiting fluid 2</td>
<td>(TYPE 78, custom TYPE)</td>
</tr>
<tr>
<td>5</td>
<td>$T_{1\text{in}}$ Temperature</td>
<td>[10,1] Outlet temperature value from routine</td>
</tr>
<tr>
<td></td>
<td>of entering fluid 1 (°C)</td>
<td>Pipe to Tank (TYPE 31)</td>
</tr>
<tr>
<td>6</td>
<td>$T_{2\text{in}}$ Temperature</td>
<td>[132,22] Temperature at the top of next downstream tank</td>
</tr>
<tr>
<td></td>
<td>of entering fluid 2 (°C)</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>$T_A$ Ambient air temperature</td>
<td>Constant (obtained from experimental results)</td>
</tr>
<tr>
<td></td>
<td>(°C)</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>$\gamma_{\text{htr},1}$ Enable</td>
<td>Constant (e.g., 1)</td>
</tr>
<tr>
<td></td>
<td>signal for first heating</td>
<td></td>
</tr>
<tr>
<td></td>
<td>element</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>$\gamma_{\text{htr},2}$ Enable</td>
<td>Constant (e.g., 1)</td>
</tr>
<tr>
<td></td>
<td>signal for second heating</td>
<td></td>
</tr>
<tr>
<td></td>
<td>element</td>
<td></td>
</tr>
</tbody>
</table>

Table F.3. Simulation parameter values for the custom TYPE 76 TRNSYS subroutine.

<table>
<thead>
<tr>
<th>Parameter No.</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$H_T$ Height of storage tank (m)</td>
<td>1.5</td>
</tr>
<tr>
<td>2</td>
<td>$a$ Heat exchanger coefficient determined from characterization test ($m_{NC}^* \text{ vs. } \Delta P$ plot) $\text{m}_{NC}$ in terms of kg/hr</td>
<td>2.376</td>
</tr>
<tr>
<td>3</td>
<td>$b$ Heat exchanger coefficient determined from characterization test ($m_{NC}^* \text{ vs. } \Delta P$ plot)</td>
<td>0.6508</td>
</tr>
<tr>
<td>4</td>
<td>$H_{HX}$ Height of heat exchanger (m)</td>
<td>0.31</td>
</tr>
</tbody>
</table>

Table F.4. Simulation input values for the custom TYPE 76 TRNSYS subroutine.

<table>
<thead>
<tr>
<th>Input No.</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$T_3$ Temperature at heat exchanger inlet (tank side) (°C)</td>
<td>[12,5] Outlet temperature of bottom of tank (TYPE 60)</td>
</tr>
<tr>
<td>2</td>
<td>$T_2$ Temperature at heat exchanger outlet (tank side) (°C)</td>
<td>[9,3] Outlet temperature of heat exchanger (TYPE 77)</td>
</tr>
<tr>
<td>3</td>
<td>$T_{T_{AVG}}$ Average temperature of solar storage tank (°C)</td>
<td>[12,17] Average tank temperature (TYPE 60)</td>
</tr>
</tbody>
</table>
Table F.5. Simulation parameter values for the custom TYPE 77 TRNSYS subroutine.

<table>
<thead>
<tr>
<th>Parameter No.</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$C_{P_C}$</td>
<td>3.648</td>
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<tr>
<td>2</td>
<td>$C_{P_{NC}}$</td>
<td>4.184</td>
</tr>
<tr>
<td>3</td>
<td>$c$</td>
<td>-0.3225</td>
</tr>
<tr>
<td>4</td>
<td>$d$</td>
<td>1.1215</td>
</tr>
</tbody>
</table>

Table F.6. Simulation input values for the custom TYPE 77 TRNSYS subroutine.

<table>
<thead>
<tr>
<th>Input No.</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$T_1$</td>
<td>[7,1] Outlet temperature of routine Pipe from Collector (TYPE 31)</td>
</tr>
<tr>
<td>2</td>
<td>$m_C$</td>
<td>[7,2] Outlet flow rate from routine Pipe from Collector (TYPE 31)</td>
</tr>
<tr>
<td>3</td>
<td>$T_3$</td>
<td>[12,5] Outlet temperature from bottom of tank (TYPE 60)</td>
</tr>
<tr>
<td>4</td>
<td>$m_{NC}$</td>
<td>[8,1] Natural convection flow rate from routine Thermosyphon Flow Rate (TYPE 76)</td>
</tr>
<tr>
<td>5</td>
<td>$T_A$</td>
<td>Constant (obtained from experimental results)</td>
</tr>
</tbody>
</table>
Appendix G

Node Sensitivity of Storage Model
An investigation of node sensitivity was conducted on the TYPE 60 storage model to evaluate its accuracy in predicting the temperature profile and stratification levels. The number of nodes was varied from 1 to 75 for a typical constant charge condition (i.e., a collector flow rate of 1.5 L/min and a charge temperature of 50°C). As a means of comparison, the top and bottom node temperatures of the storage tank (for all cases) are illustrated against the temperature profile obtained for a case of 60 nodes. The results of this analysis are shown in Figure G.1. This figure clearly illustrates the improvement in the temperature estimates as the number of simulated layers increases.

These results verify previous studies indicating that a large number of storage nodes is required to accurately model the temperature profiles in highly stratified storage tanks (Cruickshank and Harrison, 2006c).
Figure G.1. Node sensitivity of storage model.
Appendix H

Calibration of Sensors and Uncertainty Analysis
To increase the accuracy of experimental measurements, key instrumentation was calibrated by comparison with available reference instruments or by comparison against fundamental measurements. In particular, the thermocouples used to measure energy flows through the heat exchangers and the flowmeter used in the primary charge loop were calibrated as described below.

**H.1 Calibration of Flowmeter**

A Micro-oval-Mark II positive displacement flowmeter (model LNC 45C2-000-EO) was used to determine the energy delivered to the thermal energy storage through the primary charge loop. The flowmeter produced a switch closure (voltage pulse) for each 2.4875 cc of fluid volume flow. A custom fabricated frequency counter was used to transmit the pulse rate to the computer data acquisition system (Hewlett-Packard, model 3497A). LabVIEW, ver. 7, was used to capture and display measured flow rates in real time. The unit's original calibration was based on water rather than propylene glycol so it was calibrated by direct comparison with gravimetric flow rate measurements covering the range of flow rates used for the experimental investigation.

To conduct the calibration, the flow from a primary loop was directed into a reservoir such that the flow could be diverted into a measurement beaker, Figure H.1. The average flow rate was determined by measuring the time interval required to accumulate a fixed volume of fluid. This procedure was repeated five times for flow rates ranging from 0.5 to 5 L per minute and the standard error associated with each sequence of flow measurements was determined. Between each test, the tare weight of the container was recorded to minimize systematic errors.
The results of this calibration are shown below (Figure H.2), where the flow rate indicated by the computer data acquisition (micro-oval flowmeter) is plotted against the values measured by the gravimetric method. A linear regression analysis performed on the data shows that the instrument’s original calibration was approximately 9% higher when compared to the gravimetric measurements performed on the propylene glycol water mixture. A residual plot of the uncorrected errors is shown in Figure H.3.

**Figure H.1.** Gravimetric flow rate measurement set up.

**Figure H.2.** Measured micro-oval and gravimetric flow rates (before calibration).

**Figure H.3.** Residual plot of uncorrected errors.
To correct for this discrepancy, flow measurements recorded by the data acquisition system (based on the micro-oval flowmeter) were then adjusted by a new calibration factor, (i.e., the actual flow equaled the measured flow multiplied by 0.916). The bias error for the measurements was assumed equal to zero (i.e., the regression curve was assumed to go through 0). For comparison, the corrected flow values are plotted against the values determined by the gravimetric analysis in Figure H.4, and the residuals (i.e, the corrected flow minus measured flow) are shown in Figure H.5. The error bars shown in Figure H.4 are representative of the error associated with the measurement of mass for each gravimetric flow rate test values. The magnitude of the error bars associated with the measured gravimetric flow rate was determined by calculating the standard deviation, $\sigma$, of repeated tests conducted at each flow rate and are representative of an uncertainty corresponding to $2\sigma$ (i.e., 95% confidence interval). Also shown in Figure H.5 as dotted lines, is the 95% confidence interval (i.e., 20 to 1 odds or $2\sigma$). Based on this analysis, it was assumed that flow rate could be determined to within ±0.029 L/min for $1\sigma$ and ±0.059 L/min for $2\sigma$. Consequently, a value of ±0.06 L/min was chosen as the uncertainty associated with the flow measurements (see Section H.3).

![Figure H.4](image1.png)  
Figure H.4. Measured micro-oval and gravimetric flow rates (after calibration).  

![Figure H.5](image2.png)  
Figure H.5. Residual plot of corrected errors.
H.2 Calibration of Temperature Sensors

Temperature measurements were made with Type T thermocouples and recorded on the computer-based data acquisition system. The Hewlett-Packard model 3497A data acquisition and control unit were connected to a dedicated PC computer through a USB interface. LabVIEW, ver. 7, was used to capture and display measured temperatures in real time. The software temperature compensation was used to provide a simulated ice-point reference for the thermocouple measurements.

To assess the accuracy and precision of the thermocouple temperature measurements, all sensors used for heat exchanger inlet and outlet temperature measurement were calibrated by comparison against a precision reference thermometer (PRT), (Guildline model 9535). The Guildline thermometer was independently calibrated to an accuracy of ±0.012°C for 20 to 1 odds (Guildine Certificate of Calibration, 2007). For the calibration, the thermocouple leads were removed from the test apparatus and bundled to the Guildline thermometer and immersed in a calibration temperature bath (EXTECH, Model 7312), Figure H.6, that was stepped through temperatures ranging from 0 to 60°C.

Figure H.6. Thermocouple leads from the test apparatus bundled to the Guildline thermometer before immersion into the calibration temperature bath.
The results of this calibration are shown in Figure H.7 and Figure H.8. A third order polynomial equation was fit to the data by the method of least-squares.

\[ y = -7 \times 10^{-7}x^3 + 0.0003x^2 + 0.9157x + 3.1557 \]

\[ R^2 = 1 \]

![Figure H.7](image1.png)  
**Figure H.7.** Measured PRT and heat exchanger temperatures (before calibration).  

![Figure H.8](image2.png)  
**Figure H.8.** Residual plot of uncorrected errors.

After correction of the thermocouple readings recorded by the data acquisition system, the accuracy of all the temperature sensors was increased to ±0.08°C for 1σ or ±0.16°C for 20 to 1 odds, i.e., 95% confidence interval or 2σ, Figure H.9 and Figure H.10. The dotted lines in Figure H.10, is the 95% confidence interval (i.e., 20 to 1 odds or 2σ). Consequently, a value of ±0.16°C was chosen as the uncertainty associated with the temperature measurements (see Section H.3).

![Figure H.9](image3.png)  
**Figure H.9.** Measured PRT and heat exchanger temperatures (after calibration).  

![Figure H.10](image4.png)  
**Figure H.10.** Residual plot of corrected errors.
H.2.1 Precision of Temperature Sensors

To ensure repeatable measurements, the precision of the temperature sensors located in the storage tank was evaluated. To accomplish this, a cool down test (with mixing) similar to the “Heat Loss Test (Standard Decay Method)” described in Section 7.4 of the SRCC Document TM-1 (SRCC, 2006) and a room temperature steady state test were performed on the single tank system to determine the repeatability and uniformity of the temperature data set during transient and non-transient conditions, respectively.

H.2.1.1 Cool-down Test (With Mixing)

A cool down test was conducted over a 48 hour period on the single tank storage to measure the spread of the temperature data during transient conditions, Figure H.11 and Figure H.12. The water was initially pre-heated and mixed to a uniform temperature of 53.5°C by an electric-heated charge loop and a small, circulating pump. At the beginning of the test, the heater was turned off, and tank temperatures were subsequently recorded at 3 minute intervals. For the duration of the cool down test, the tank was continuously mixed at a rate of approximately 10 L/min. The ambient temperature was also recorded during the test sequence and remained constant at approximately 21.2 ± 0.3°C.

Results show that the tank was at a uniform temperature for the duration of the test, Figure H.11, and that the sampling distribution closely approximates a normal distribution, Figure H.12 to Figure H.14, with a normalized slope of y=0.9858x. The standard deviation of the upper temperature probes was determined to be ±0.018°C. To calculate the root mean square deviation, a set of sample data, consisting of 11560 data
points, was used. In addition, 72.1% of the values were found to lie within one standard deviation of the mean value.

![Graph of Horizontal Probe Temperatures](image1)

**Figure H.11.** Horizontal temperature profile measured during cool down test (with mixing) as a function of time.

![Graph of Sampling Frequency](image2)

**Figure H.12.** Spread of temperature data during transient conditions.

![Graph of Normal and Sampling Distribution](image3)

**Figure H.13.** Normalized slope of the probability density function.

**Figure H.14.** Probability density function for a normal and sampling distribution with a σ=0.018°C.

### H.2.1.2 Steady State Test

A steady state test was conducted over a 1 hour period on the single tank storage to measure the spread of the temperature data during zero heat loss conditions, with the tank stabilized at room temperature. The steady state test consisted of recording the temperature values of the horizontal sensors during the non-charging, non-mixing period.
Results for the steady state test show that the temperature data of the upper and lower horizontal probes tend to be very close to the mean temperature of the probes, Figure H.15 and Figure H.16. The standard deviations of the upper and lower temperature probes were found to be ±0.0239°C and ±0.0316°C, respectively. For the set of sample data evaluated (consisting of 110 data points for each probe), 77.2% of the values were found to lie within one standard deviation of the mean value. The mean temperatures of the upper and lower probes were 20.3°C and 19.7°C, respectively, and the ambient room temperature was 19.8°C.

Figure H.15. Horizontal temperatures measured during steady state test as a function of distance from the tip of the probes.

Figure H.16. Spread of horizontal temperature data as a function of distance from tip of a) the upper probe and b) the lower probe.
H.3 Propagation of Measurement Uncertainty

The propagation of measurement uncertainty is defined as the way in which uncertainties in individual variables affect the uncertainty in the results (The Engineering Handbook, 1995). Single-sample uncertainty has been described in the engineering literature by the works of Kline and McClintock (1953) and Moffat (1985). The proposed techniques are based on an estimated uncertainty in each variable assuming a 95% confidence interval, i.e., 20 to 1 odds that a measured value will fall within the uncertainty interval. Therefore, the result \( R \) of an experiment is assumed to be calculated from a set of \( n \) independent variables, e.g., \( X_1, X_2, ..., X_n \), i.e.,

\[
R = R(X_1, X_2, X_3, ..., X_n)
\]

each with uncertainties, \( \omega_1, \omega_2, ..., \omega_n \).

Assuming the same uncertainty estimate for all component variables (i.e., the uncertainties of the variables are based on the same odds), the estimated uncertainty in the result, \( \omega_R \), based on the root-sum-square method proposed by Kline and McClintock (1953) is given as

\[
\omega_R = \pm \left[ \left( \frac{\partial R}{\partial X_1} \cdot \omega_1 \right)^2 + \left( \frac{\partial R}{\partial X_2} \cdot \omega_2 \right)^2 + \left( \frac{\partial R}{\partial X_3} \cdot \omega_3 \right)^2 + ... + \left( \frac{\partial R}{\partial X_n} \cdot \omega_n \right)^2 \right]^{1/2}
\]

H.3.1 Uncertainty in the Heat Transfer Rate

For a steady-flow device such as a heat exchanger where the mass flow rate and fluid properties are assumed to remain constant, the heat transfer rate, \( Q \), across a heat exchanger (assuming no heat transfer to the ambient) can be written as
\[ Q = \dot{m}c_p(T_1 - T_2) \]

where \( \dot{m} \) is the mass flow rate of the heat transfer fluid, \( c_p \) is the specific heat of the heat transfer fluid, and \( T_1 \) and \( T_2 \) are the inlet and outlet temperatures of the heat exchanger, respectively.

If \( \omega_Q \) is the uncertainty in a calculated variable \( Q \) such that \( Q = \dot{m}c_p(T_1 - T_2) \)

then

\[ \omega_Q = \pm \left[ \left( \frac{\partial Q}{\partial \dot{m}} \cdot \omega_{\dot{m}} \right)^2 + \left( \frac{\partial Q}{\partial c_p} \cdot \omega_{c_p} \right)^2 + \left( \frac{\partial Q}{\partial T_1} \cdot \omega_T \right)^2 + \left( \frac{\partial Q}{\partial T_2} \cdot \omega_T \right)^2 \right]^{\gamma / 2} \]

where

\[ \frac{\partial Q}{\partial \dot{m}} = c_p(T_1 - T_2) \quad \frac{\partial Q}{\partial c_p} = \dot{m}(T_1 - T_2) \quad \frac{\partial Q}{\partial T_1} = \dot{m}c_p \quad \frac{\partial Q}{\partial T_2} = -\dot{m}c_p. \]

The above expression can be simplified by expressing the relative uncertainty in the heat transfer rate, i.e.,

\[ \frac{\omega_Q}{Q} = \pm \left[ \left( \frac{c_p(T_1 - T_2)}{\dot{m}c_p(T_1 - T_2)} \cdot \omega_{\dot{m}} \right)^2 + \left( \frac{\dot{m}(T_1 - T_2)}{\dot{m}c_p(T_1 - T_2)} \cdot \omega_{c_p} \right)^2 + \left( \frac{\dot{m}c_p}{\dot{m}c_p(T_1 - T_2)} \cdot \omega_T \right)^2 + \left( \frac{-\dot{m}c_p}{\dot{m}c_p(T_1 - T_2)} \cdot \omega_T \right)^2 \right]^{\gamma / 2} \]

and

\[ \frac{\omega_Q}{Q} = \pm \left[ \left( \frac{\omega_{\dot{m}}}{\dot{m}} \right)^2 + \left( \frac{\omega_{c_p}}{c_p} \right)^2 + \left( \frac{\omega_T}{(T_1 - T_2)} \right)^2 + \left( \frac{-\omega_T}{(T_1 - T_2)} \right)^2 \right]^{\gamma / 2}. \]

As such, the relative uncertainty in the heat transfer rate measurement can be calculated using a sample set of measurements, Table H.1. The uncertainty values of flow and temperature (i.e., \( \omega_{\dot{m}} \) and \( \omega_T \) respectively) used in this analysis are based on values obtained from the sensor calibration analysis described in Sections H.1 and H.2. The
relative uncertainty of the specific heat value, \( \omega_{c_p}/c_p \), was estimated based on a ±5°C range of tabulated specific heat values.

The measured total heat transfer rate and the individual charge rates (across each heat exchanger) are illustrated for a variable input power sequence (Test 3) in Figure H.17 and Figure H.18. The results show that when the mass flow rate, \( \dot{m} \), is low, the relative uncertainty in the heat transfer rate is dominated by the relative uncertainty in the mass flow rate.

Table H.1. Uncertainty in heat transfer rate measurements.

<table>
<thead>
<tr>
<th>Test</th>
<th>( \dot{m} ) (L/min)</th>
<th>( \omega_{\dot{m}} ) (L/min)</th>
<th>( \omega_{c_p}/c_p )</th>
<th>( T_1 ) (°C)</th>
<th>( T_2 ) (°C)</th>
<th>( \Delta T ) (°C)</th>
<th>( \omega_T ) (°C)</th>
<th>( \omega_Q/Q ) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 Series, Case B, max. 3000 W</td>
<td>1.2 ±0.06</td>
<td>±0.025</td>
<td>39.05</td>
<td>18.36</td>
<td>20.69</td>
<td>±0.16</td>
<td>±6.11</td>
<td></td>
</tr>
<tr>
<td>2 Parallel, Case B, max. 3000 W</td>
<td>1.2 ±0.06</td>
<td>±0.025</td>
<td>30.58</td>
<td>18.40</td>
<td>12.18</td>
<td>±0.16</td>
<td>±6.26</td>
<td></td>
</tr>
<tr>
<td>3 Series, Case A, max. 3000 W</td>
<td>1.5 ±0.06</td>
<td>±0.025</td>
<td>46.29</td>
<td>27.32</td>
<td>18.96</td>
<td>±0.16</td>
<td>±5.22</td>
<td></td>
</tr>
<tr>
<td>4 Series, Case B, max. 3000 W</td>
<td>1.5 ±0.06</td>
<td>±0.025</td>
<td>37.94</td>
<td>30.07</td>
<td>7.86</td>
<td>±0.16</td>
<td>±5.86</td>
<td></td>
</tr>
<tr>
<td>5 Series, Case B, max 3000 W</td>
<td>1.8 ±0.06</td>
<td>±0.025</td>
<td>38.03</td>
<td>28.53</td>
<td>9.49</td>
<td>±0.16</td>
<td>±5.07</td>
<td></td>
</tr>
<tr>
<td>6 Series, Case B, max 3000 W</td>
<td>2.0 ±0.06</td>
<td>±0.025</td>
<td>43.45</td>
<td>23.10</td>
<td>20.34</td>
<td>±0.16</td>
<td>±4.29</td>
<td></td>
</tr>
<tr>
<td>7 Series, Case A, max 3000 W</td>
<td>3.0 ±0.06</td>
<td>±0.025</td>
<td>55.66</td>
<td>41.77</td>
<td>13.89</td>
<td>±0.16</td>
<td>±3.70</td>
<td></td>
</tr>
<tr>
<td>8 Series, Case A, max 3000 W</td>
<td>4.5 ±0.06</td>
<td>±0.025</td>
<td>47.76</td>
<td>39.45</td>
<td>8.30</td>
<td>±0.16</td>
<td>±3.98</td>
<td></td>
</tr>
<tr>
<td>9 Series, Case B, max 3000 W</td>
<td>4.5 ±0.06</td>
<td>±0.025</td>
<td>47.07</td>
<td>37.21</td>
<td>9.86</td>
<td>±0.16</td>
<td>±3.69</td>
<td></td>
</tr>
<tr>
<td>10 Series, Case A, max 6000 W</td>
<td>4.5 ±0.06</td>
<td>±0.025</td>
<td>68.59</td>
<td>48.81</td>
<td>19.79</td>
<td>±0.16</td>
<td>±3.12</td>
<td></td>
</tr>
</tbody>
</table>
Figure H.17. Total heat transfer rate across the heat exchangers (Case A, series configuration, collector flow rate of 1.5 L/min). The dotted lines represent the estimated uncertainty in the heat transfer rate assuming a 95% confidence interval.

Figure H.18. Individual charge rates across each heat exchanger (Case A, series configuration, collector flow rate of 1.5 L/min). The dotted lines represent the estimated uncertainty in the heat transfer rate assuming a 95% confidence interval.
Appendix I

Heat Exchanger Characterization under Reverse Flow Operation
I.1 Heat Exchanger Characterization under Reverse Flow Operation

During normal operation, the heat exchanger in a thermosyphon charge loop normally operates in a counter-flow configuration, where cold water from the bottom of the storage is heated and transported by temperature induced buoyancy forces to the top of the storage. During reverse operation, however, the heat exchanger will be operating in a parallel flow configuration, and its performance will differ from the normal counter-flow case. Consequently, it is necessary to change the previously established test method (proposed by Lin et al. (2000)) and redefine the relationship for modified effectiveness to allow new performance indices to be measured. Therefore in the parallel flow configuration, the modified effectiveness may be represented as:

$$
\varepsilon_{\text{mod:\ REV}} = \frac{Q_{\text{actual}}}{Q_{\text{max}}} = \frac{(\dot{m}_c p_s)(T_2 - T_3)}{(\dot{m}_c p_c)(T_2 - T_1)}
$$

During testing, values of $\varepsilon_{\text{mod:\ REV}}$ were measured at a range of capacitance ratios, $C_r$, and temperatures. This was accomplished by conducting controlled experiments as described below.

Specifically, the apparatus described in Reference Cruickshank and Harrison (2006b) was used, Fig. I.1, however, system operational temperatures were modified to reproduce conditions that resulted in reverse flow through the heat exchange loop. To do this, a single thermal storage was heated and preconditioned to a uniform set point temperature. The temperature of the collector loop supply to the heat exchanger was then set to a predetermined lower temperature and circulation through the heat exchanger started. Under these conditions, heat was removed from the heat exchange loop, lowering the density of the fluid within it, thereby lowering the positive pressure
gradient. Continuing under these conditions, the positive pressure gradient would be reduced and even become negative. This adverse pressure gradient would cause the flow through the heat exchange loop to stop and reverse direction.

![Diagram of heat exchanger setup](image)

**Figure I.1.** Apparatus used to conduct heat exchanger characterization tests.

During testing, the fluid in the solar collector loop was externally conditioned to ensure that the supply temperature to the heat exchanger remained fixed at the predetermined value. To capture a range of operational conditions, tests were conducted at storage tank temperatures of 40°C and 60°C, and at a range of collector loop flow rates, e.g., 1.5, 3.0, and 4.5 L/min. All tests were conducted at a charge loop temperature of 25°C.
I.2 Modeling Approach

The operation of the thermosyphon loop and storage was modeled in TRNSYS ver. 15 (2000) using routines previously developed to describe its operation in positive “charging” mode (Lin et al. 2000, Cruickshank and Harrison 2006a, 2006c, 2009b). Although these studies have shown good correspondence between measured and simulated results, they did not consider conditions in which the flow through the heat exchange loop was in the reverse direction. During periods of reverse flow, the NCHEs operate in a parallel-flow configuration rather than the usually counter-flow arrangement and for this case, Equation I.1 for the modified effectiveness should be used. In addition, a new empirical correlation for NCHE effectiveness as a function of $C_r$ was derived based on the experimental data obtained during the reverse flow characterization tests. The coefficients were derived from temperature and heat transfer data obtained during a controlled discharge experiment. Consistent with the previous analysis (Cruickshank and Harrison, 2006a, 2009a), the measured performance of the NCHE was fit to the following relationships:

\[ \dot{m}_s = 0.0396 \times (\Delta P)^{0.6508} \]  

(I.2)

\[ \varepsilon_{\text{mod}_\text{REV}} = -0.8228 \times C_{r_{\text{mod}}}^2 + 0.9683 \times C_{r_{\text{mod}}} \]  

(I.3)

The experimental data and associated curve fits are shown in Figure I.2 and Figure I.3. In addition to the experimental data shown in Figure I.2, the characteristic curve for thermosyphon flow in the positive direction versus hydrostatic pressure head is shown for comparison. It is important to note that the values of $\Delta P$, $\dot{m}_s$ and $C_{r_{\text{mod}}}$ are plotted as absolute values for ease of comparison between the positive and reverse flow.
curve fits. It is apparent that the reverse flow data closely matches the data for the positive flow condition. This correspondence was only achieved after the procedure for estimating the temperature distribution in the heat exchanger charge loop was modified to better represent the temperature distribution occurring during reverse flow operation. In particular, it was found that the best results were achieved when the heat exchanger side hydrostatic head was calculated as

$$\Delta P_{\text{HX hydrostatic}} = \rho_{T_2} \cdot g \cdot (H_{\text{Tank}} - H_{\text{HX}}) + \rho_{T_3} \cdot g \cdot H_{\text{HX}}$$  \hspace{1cm} (I.4)

resulting in the following relationship for net hydrostatic pressure head:

$$\Delta P_{\text{net hydrostatic}} = \rho_{T_{\text{Tank}}} \cdot g \cdot H_{\text{Tank}} - \rho_{T_2} \cdot g \cdot (H_{\text{Tank}} - H_{\text{HX}}) - \rho_{T_3} \cdot g \cdot H_{\text{HX}}$$  \hspace{1cm} (I.5)

![Figure I.2. Plot of experimental results showing the dependence of the reverse thermosyphon flow rate on the pressure head.](image-url)
Figure I.3. Experimentally derived modified effectiveness as a function of modified capacity ratio.

Measured heat exchanger effectiveness values are shown in Figure I.3 as determined according to the procedure described above. As expected, measured values of modified effectiveness for the reverse flow case are lower than those of the counter-flow case as previously determined (Cruickshank and Harrison, 2006a, 2009a).

Using this approach, it was possible to establish the net pressure head during the test sequence and to predict the magnitude of the natural convection flow rate in both positive and negative cases.
Appendix J

Sample TRNSYS Code
This simulation was produced by Cynthia A. Cruickshank, at the Queen's Solar Lab in Kingston, ON

* Simulation of the Enerworks Inc. Model EWRA2 Solar Domestic Hot Water System

* Simulation of a typical multi-family, domestic hot water application using a TRNSYS model of the complete solar system including solar collectors, controller and storage unit. The system modeled is a series-connected multi-tank system consisting of three 270 L tanks, similar to the unit tested.

* Simulations were conducted using standard TMY data files for Montreal.

* A daily hot water draw profile consistent with the CSA standard-day recommendations (CAN/CSA-F379.1-88, 2004) for a 300 L/day draw was assumed.

Version 15.1

ASSIGN "C:\SeriesmultitankCSACyn2006\test.LST" 6
ASSIGN "C:\SeriesmultitankCSACyn2006\Mont.TRN" 10
ASSIGN "C:\SeriesmultitankCSACyn2006\DATA_Montreal.txt" 13
ASSIGN "C:\SeriesmultitankCSACyn2006\TEMPDATA_Tank1_Montreal.txt" 16
ASSIGN "C:\SeriesmultitankCSACyn2006\TEMPDATA_Tank2_Montreal.txt" 17
ASSIGN "C:\SeriesmultitankCSACyn2006\TEMPDATA_Tank3_Montreal.txt" 18
ASSIGN "C:\SeriesmultitankCSACyn2006\report_Montreal.txt" 12
ASSIGN "C:\SeriesmultitankCSACyn2006\intergratedata_Montreal.txt" 15

CONSTANTS 43

Tmax=70
LUA=10
LUC=13
LUD=12
LUE=16

TD=50
TTset=55
TTST=20
*TTST=8.1
*TM=15
*TM=8.1
TA=20
DON = 1
TON=1
TOF=8760
*TOF=40
THO=TA
*Tcollout=TM

FRTA=0.650
FRUL=15.9912
FRUL2=0.0
DT=0.05
*FL=74.16
FL=222.48
*FL=92.7
* 1.5 L/min is equal to 92.7 kg/hr.... 1.2 L/min is equal to 74.16kg/hr

CPG=3.648
CP=4.184
DEN=1000
DENG=1030
k=2.1996
NC=1
*AC= 5.716
AC=17.148
ANG=30
*set to zero for Standard Day Test
AZ=0.0

VOL1=0.270
BURNER = 45337
TH1=-1.385
HT=1.07
HE=0.29
US1=5

TST=TON
STP=TOF

GR=0.3
*GR set to zero for Std Day
*GR=0.2
LAT=45.5

*LAT=51.083 for Calgary
*LAT=43 for London
*LAT=44.23 for Kingston
*LAT=45.42 for Ottawa
*LAT=45.5 for Montreal
*LAT=43.7 for Toronto

A=2.388
B=0.6505
C=-0.3488
D=1.1402
E=0.00

HWLoad=900
*Draw= 0.0

* seasonal hour values 2160 4344 6552 8760
* A is a coefficient of correlation for pressure to mfr
* B is a coefficient of correlation for pressure to mfr
* C is a coefficient of correlation for mfr to eff
* D is a coefficient of correlation for mfr to eff

273
* DON is the day of the year of the start of the simulation
* TM is average mains temperature, C
* TA is indoor ambient temperature, C
* TD is the delivery temperature, C
* Tcollout is the temperature at the collector outlet, C
* THO is the temp. at the cold side outlet of H.E., C
* TOF is simulation end time, hrs
* TON is simulation start time, hrs
* TTST is the temperature at the start of the test, C
* STP is the stop time for output file, hrs
* TST is start time for output file, hrs
* LUA is the file unit number
* DT is timestep size in hours for simulation, hrs
* US is store overall loss coefficient
* FL is collector flow rate, kg/hr
* AC is the collector area, m2
* TH1 is the solar tank height, m
* VOL1 is the volume of the preheat tank, m3
* VOL2 is the volume of the auxiliary tank, m3
* HT is the height of the storage tank height, m
* NC is the number of collectors in series
* GR is the ground reflectance
* ANG is the collector tilt angle, degrees
* AZ is the collector azimuth, degrees
* LAT is the site latitude, degrees
* CP is the specific heat of water @ 25 C (kJ/kg-C)
* CPG is the specific heat of 50-50 glycol mix @ 25 C
* CPGE is the effective specific heat of the glycol loop (Guess) (kJ/kg-C)
* DEN is the density of the water @ 25 C (kg/m3)
* FRTA is the intercept efficiency
* FRUL is the negative slope of the efficiency curve
* 
EQUATIONS 3
EQ_QTap = (([12,6]-[13,1])*CP*DEN*[11,2])/1000
EQ_QLoad = ((TD-[13,1])*CP*DEN*[14,1])/1000
FLOW=FL*1000/(60*DEN)

WIDTH 132
TOLERANCES 0.05 0.05
SIMULATION TON TOF DT
LIMITS 1000 500

UNIT 1 TYPE 9 Data Reader (Formatted)
PARAMETERS 12
1 0 2 1 -1 1 0 -2 1 0 LUA 1
(5X,F7.2,2X,F6.2)

UNIT 2 TYPE 16 Solar Radiation Processor
PARAMETERS 8
1 1 3 DON LAT 4921 0 1
INPUTS 7
1,1 1,99 1,100 0,0 0,0 0,0 1,101
0 0 1 GR ANG AZ 0
UNIT 3 TYPE 1 Solar Collectors, No Heat Exchanger
PARAMETERS 11
NC AC CPG 1 12.360 FRTA FRUL FRUL2 2 0.202 0
INPUTS 9
6.1 6.2 1.2 2.7 2.4 2.5 0.0 2.10 0.0
TA 0.0 TA 0.0 0.0 0.0 GR 0.0 ANG
*Tci mc, Ta, It I Id gr lang slope

UNIT 4 TYPE 2 Differential Temperature Flow Controller
PARAMETERS 2
5 70
INPUTS 6
*Tmax 12.5 12.22 4.1 0.0 0.0
3.1 142.5 142.22 4.1 0.0 0.0
TA TM TA 0 10 3
*Tcollout, Ttank-bottom, Tmax, Y0(control function), Upper deadband temp diff, Lower deadband temp diff

UNIT 5 TYPE 3 pump in collector loop
PARAMETERS 4
FL CPG 111.5 0.0
INPUTS 3
139.1 139.2 4.1
0 0.0 0.0
*Tipump, mc, Yo(control function)

UNIT 6 TYPE 31 PIPE TO COLLECTOR
PARAMETERS 6
0.01 10 8 DENG CPG 15
INPUTS 3
5.1 5.2 0.0
15 0.0 15
*3rd input is for indoor pipes (currently)

UNIT 7 TYPE 31 PIPE FROM COLLECTOR
PARAMETERS 6
0.01 10 8 DENG CPG 15
INPUTS 3
3.1 3.2 0.0
15 0.0 15
*3rd input is for indoor pipes (currently)

*************** STORAGE 1 ***************

UNIT 8 TYPE 76 THERMOSYPHON FLOW RATE
PARAMETERS 4
HT A B HE
INPUTS 4
12.5 9.3 12.17 9.3
*12.1 9.3 12.12 9.3
TA TA 20 20
UNIT 9 TYPE 77 NCHE Heat Exchanger
PARAMETERS 5
CPG    CP    C   D   E
INPUTS 5
7,1   7,2   12,5   8,1   0,0
TA   0.0   20.0   TA

UNIT 10 TYPE 31 PIPE TO TANK
PARAMETERS 6
0.018  1.55  5 DEN CP  20
INPUTS 3
9,3   9,4   0,0
TA   0.0   TA

UNIT 11 TYPE 78 MIXTURE FOR HOT WATER
PARAMETERS 1
TD
INPUTS 3
13,1   12,6   14,1
TA   TA   0.0

UNIT 13 TYPE 14 Monthly Mains Water Temperatures
PARAMETERS 26
0      3    744    2    1416   2
2160   1.5   2880   5.5   3624  12.5
4344   18   5088   21   5832  20.5
6552  18   7296  11.5   8016  6.5
760    0
*used for annual simulation for Montreal from Michel Bernier

*UNIT 13 TYPE 14 Monthly Mains Water Temperatures
*PARAMETERS 26
*0      6.58   744    5.78    1416   6.58
*2160   8.77   2880   11.76   3624  14.75
*4344  16.97   5088   17.75  5832  16.95
*6552  14.75   7296  11.76   8016  14.75
*760    0
*used for annual simulation for Montreal (data from NRCan)

*UNIT 13 TYPE 14 Monthly Mains Water Temperatures
*PARAMETERS 26
*0      5.05   744    3.65    1416   5.05
*2160  8.875   2880   14.1   3624  19.325
*4344  23.15   5088   24.55  5832  23.15
*6552  19.325  7296  14.1   8016  8.875
*8760    0
*used for annual simulation for Calgary

UNIT 12 TYPE 60 Stratified Fluid Storage Tank 1
PARAMETERS 32
2   0.270   1.5   -1   1.5
0   0   1.5    CP   DEN
US1   k   0   105  1
0.4 0.4 TTset 3 0
1.2 1.2 TTset 3 0
0 20 6 0 0
0 0
INPUTS 9
10,2 10,2 0,0 11,2 10,1 132,22 0,0 0,0 0,0
0.0 0.0 -2 0.0 6 15 20 1 1

DERIVATIVES 30
TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST

*************** STORAGE 2 ******************

UNIT 127 TYPE 31 PIPE FROM FIRST H.X. TO SECOND H.X.
PARAMETERS 6
0.0144 1.285 5 DENG CPG TA
INPUTS 3
9,1 9,2 TA
15 0.0 15

UNIT 128 TYPE 76 THERMOSYPHON FLOW RATE
PARAMETERS 4
HT A B HE
INPUTS 4
132,5 129,3 132,17 129,3
TA TA 20 20

UNIT 129 TYPE 77 NCHE Heat Exchanger
PARAMETERS 5
CPG CP C D E
INPUTS 5
127,1 127,2 132,5 128,1 0,0
TA 0.0 20 0.0 TA

UNIT 130 TYPE 31 PIPE TO TANK
PARAMETERS 6
0.018 1.55 5 DEN CP 20
INPUTS 3
129,3 129,4 0,0
TA 0.0 TA

UNIT 132 TYPE 60 Stratified Fluid Storage Tank 2
PARAMETERS 32
2 0.270 1.5 -1 1.5
0 0 1.5 CP DEN
US1 k 0 105 1
0.4 0.4 TTset 3 0
1.2 1.2 TTset 3 0
0 20 6 0 0
0 0

277
INPUTS 9
130,2  130,2  0,0  11,2  130,1  142,22  0,0  0,0  0,0
0,0  0,0  -2  0,0  15  8  20  1  1

DERIVATIVES 30
TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST TTST

******************** STORAGE 3 *******************

UNIT 137 TYPE 31 PIPE FROM SECOND H.X. TO THIRD H.X.
PARAMETERS 6
0.0144  1.285   5  DENG  CPG  TA
INPUTS 3
129,1   129,2   TA
15   0.0   15

UNIT 138 TYPE 76 THERMOSYPHON FLOW RATE
PARAMETERS 4
HT  A  B  HE
INPUTS 4
142,5  139,3   142,17   139,3
TA   TA   20   20

UNIT 139 TYPE 77 NCHE Heat Exchanger
PARAMETERS 5
CPG  CP  C  D  E
INPUTS 5
137,1   137,2   142,5   138,1   0,0
TA   0.0   20   0.0   TA

UNIT 140 TYPE 31 PIPE TO TANK
PARAMETERS 6
0.018  1.55  5  DEN  CP  20
INPUTS 3
139,3   139,4   0,0
TA   0.0   TA

UNIT 142 TYPE 60 30 Node Stratified Fluid Storage Tank 3
PARAMETERS 32
2   0.270   1.5  -1  1.5
0   0   1.5  CP  DEN
USI  k  0  105  1
0.4   0.4  TTset  3  0
1.2   1.2  TTset  3  0
0   20   6   0   0
0   0
INPUTS 9
140,2   140,2   0,0   11,2   140,1   13,1   0,0   0,0   0,0
0,0   0,0   -2   0,0   8   15   20   1   1
UNIT 25  TYPE 25  Print Data to File
PARAMETERS 4
1  0  8760  13
*was 1 1 8760 13
INPUTS 8
2,7  3,3  12,3  12,7  EQ_QTap EQ_QLoad  5,3  12,6
QRAD  Qell  m2in  Qlos  QTAP  QLOAD  Qpar  Tdel
*Changed 2,7 to 1.1 and input 5 & 6 from 11.4 and 11.3 to EQ_QTap & EQ_QLoad

UNIT 31  TYPE 25  Print Data to File Tank 1
PARAMETERS 4
0.2  0  8760  16
INPUTS 22
12,22  12,31  12,34  12,37  12,40  12,43  12,46  12,49  12,23  3,1  9,1  12,5  9,3  12,6  12,7  12,8  12,9
12,10  12,11  12,16  12,17  12,18
T1  T9  T12  T15  T18  T21  T24  T27  T30  Tci  Tco  Tsi  Tso  Tdraw  Qloss  Qso  Qsi
Qm  Qd  deltaE  Tave  staticpressure

UNIT 32  TYPE 25  Print Data to File Tank 2
PARAMETERS 4
0.2  0  8760  17
INPUTS 22
132,22  132,31  132,34  132,37  132,40  132,43  132,46  132,49  132,23  9,1  129,1  132,5  129,3
132,6  132,7  132,8  132,9  132,10  132,11  132,16  132,17  132,18
T1  T9  T12  T15  T18  T21  T24  T27  T30  Tci  Tco  Tsi  Tso  Tdraw  Qloss  Qso  Qsi
Qm  Qd  deltaE  Tave  staticpressure

UNIT 33  TYPE 25  Print Data to File Tank 3
PARAMETERS 4
0.2  0  8760  18
INPUTS 23
142,22  142,31  142,34  142,37  142,40  142,43  142,46  142,49  142,23  129,1  139,1  142,5  139,3
142,6  142,7  142,8  142,9  142,10  142,11  142,16  142,17  142,18  FLOW
T1  T9  T12  T15  T18  T21  T24  T27  T30  Tci  Tco  Tsi  Tso  Tdraw  Qloss  Qso  Qsi  Qm  Qd  deltaE  Tave  staticpressure  FLOW(L/min)

*Standard draw percentages for exact CSA 300L
Equations 24
L1 = HWLoad*0.0
L2 = HWLoad*0.0
L3 = HWLoad*0.0
L4 = HWLoad*0.0
L5 = HWLoad*0.0
L6 = HWLoad*0.0
L7 = HWLoad*0.0
L8 = HWLoad*0.03333333
L9 = HWLoad*0.08333333
L10 = HWLoad*0.08333333
L11 = HWLoad*0.15
L12 = HWLoad*0.08333333
L13 = HWLoad*0.03333333
L14 = HWLoad*0.01666666
L15 = HWLoad*0.0
L16 = HWLoad*0.0
L17 = HWLoad*0.05
L18 = HWLoad*0.08333333
L19 = HWLoad*0.15
L20 = HWLoad*0.08333333
L21 = HWLoad*0.1
L22 = HWLoad*0.03333333
L23 = HWLoad*0.01666666
L24 = HWLoad*0.0

UNIT 14 TYPE 14 (Draw profile)
PARAMETERS 146
  0  0   0 L1  1 L1
  1  0   1 L2  2 L2
  2  0   2 L3  3 L3
  3  0   3 L4  4 L4
  4  0   4 L5  5 L5
  5  0   5 L6  6 L6
  6  0   6 L7  7 L7
  7  0   7 L8  8 L8
  8  0   8 L9  9 L9
  9  0   9 L10 10 L10
 10  0  10 L11 11 L11
 11  0  11 L12 12 L12
 12  0  12 L13 13 L13
 13  0  13 L14 14 L14
 14  0  14 L15 15 L15
 15  0  15 L16 16 L16
 16  0  16 L17 17 L17
 17  0  17 L18 18 L18
 18  0  18 L19 19 L19
 19  0  19 L20 20 L20
 20  0  20 L21 21 L21
 21  0  21 L22 22 L22
 22  0  22 L23 23 L23
 23  0  23 L24 24 L24
 24  0

END