Design, Analysis, and Optimization of a Dual Tank Solar-Assisted Heat Pump System

by

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Author’s Declaration

I hereby declare that I am the sole author of this thesis. This is a true copy of the thesis, including any required final revisions, as accepted by my examiners. I understand that my thesis may be made electronically available to the public.

Abstract

This work investigates the performance of a dual tank solar-assisted heat pump (SAHP) system. These systems combine solar thermal collectors (STCs) and a heat pump (HP) into a hybrid system to meet building thermal loads. The main goals of a system of this type are to reduce energy consumption and provide a method to deliver thermal energy with sustainable sources. This research project undertook the design, testing, model creation and tuning, performance evaluation, and optimization of a dual tank SAHP system for domestic hot water (DHW). The novel SAHP configuration developed uses a second thermal storage tank to create more modes of operation with the intent of decreasing the amount of purchased energy.

A key aspect of the work was a test apparatus built at the University of Waterloo (UW) to facilitate equipment testing and mapping, identify operational problems such as HP short-cycling, and tune mode components.

Alternatives were modelled in TRNSYS to develop design recommendations and assess economic justification for the dual tank SAHP system. Despite noticeable electricity savings by these systems, an analysis of tolerable capital cost versus electricity cost shows that SAHP systems are not justifiable for single family dwellings at the current time.

The impact of electricity time-of-use charges was investigated, which revealed that the DHW, SDHW, and single tank SAHP systems consume a similar amount of electricity during off-peak hours. The performance of the dual tank SAHP system in many global climate types was assessed to identify locations best suited for such a system. Many recommendations are provided to improve economic justification of the system.

Overall, the performance benefits of a dual tank SAHP system over the alternatives was demonstrated. Due to the high capital costs of the additional equipment required, economic justification is difficult for a single-family residential application. There are promising applications and system augmentations that should be investigated which can lead to a successful demonstration project of a dual tank solar-assisted heat pump system.

Dedication

To my parents, Wendy and Jeff, who have always offered their support and encouragement.

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Table of Contents

List of Figures

List of Tables

List of Abbreviations

COP	Coefficient of performance
CSA	Canadian Standards Association
DAQ	Data acquisition system
db	Deadband
DHW	Domestic hot water
Div	Diverter valve
$DSAHP$	Direct solar-assisted heat pump
GB	Gigabytes
$GSHP$	Ground source heat pump
HP	Heat pump
HX	Heat exchanger
I/O	Input/output
$ISAHP$	Indirect solar-assisted heat pump
RAM	Random access memory
$SAHP$	Solar-assisted heat pump
$SDHW$	Solar domestic hot water
STC	Solar thermal collector
TMY	Typical meteorological year
TOU	Time-of-use
$TRNSYS$	Transient System Simulation Tool
TTL	Transistor-transistor logic
VAC	Alternating current voltage

Nomenclature

$a, b, \dots$	Curve fit coefficients	
c	Specific heat	kJ/kgK
COP	Coefficient of performance	$-$
E	Energy	kJ
$\%E$	Percent energy	$\%$
G	Irradiation, energy flux	W/m^2
m	Mass	kg
$ScaleHP$	Scaling factor in heat pump model	
SF	Solar fraction	$-$
T	Temperature	$°C\ or\ K$
Q	Amount of heat transfer	kJ
$\dot{Q}$	Rate of heat transfer	$W\ or\ kJ/h$
$\dot{W}$	Rate of work input to compressor	W

Greek Symbols

η	Efficiency	$\%$
θ	Incidence angle	$degrees$

Subscripts

a	Ambient
aux	Auxiliary
c	Collector
c	Cold
$Carnot$	Best-case system performance based on the ideal Carnot cycle
$comp$	Compressor
D	Direct
h	Hot
$l, load$	Load
m	Mean
ND	Normal direct
$s, source$	Source
$scale$	Scaled energy transfer rate for heat pump model
T	Total
u	Useful

Chapter 1 Introduction

The diminishing abundance and rising cost of energy are resulting in more effort directed towards energy efficiency and conservation. Further, the impact of emissions from fossil fuel consumption is creating awareness and concern for clean energy. Buildings represent a substantial portion of energy demand in Canada at about 31% of all energy used in the country [1].

Providing space heating and hot water to buildings is a necessity in many parts of the world. These two activities represent the majority of energy used by residential sector in Canada: for 2009, space heating required 893 PJ (8.46×10^{14} Btu) and water heating 246 PJ (2.33×10^{14} Btu) [1]. These figures represent 63% and 17% of the total residential secondary energy use in Canada, respectively. In the US, space heating consumed 5,518 PJ (5.23×10^{15} Btu) and water heating 2,026 PJ (1.92×10^{15} Btu) of secondary energy for the same year [2].

The current methods of space and water heating typically use hydrocarbon fuel or electricity via resistance heating. Electric resistance heating supplies 27.7% of the space heating demand in Canada, with the balance coming from the combustion of natural gas, wood, heating oil, propane, and coal [1]. There are many drawbacks associated with burning hydrocarbon fuels, with the most immediate being greenhouse gas emissions and air pollution. Greenhouse gas emissions can be reduced by replacing combustion energy sources with alternatives.

There is a potential for large electricity savings by replacing electric resistance heaters with heat pumps (HPs). Heat pumps use a thermodynamic refrigeration cycle to move heat from a cold source to a hot sink with input of mechanical work. Common heat pump sources are ambient air and ground. The mechanical work is typically supplied by electricity powering a motor. A heat pump can deliver several units of heat to the sink, which is the load for which the HP is designed. This ratio is defined as coefficient of performance (COP).

$$COP_{HP} = \frac{\text{heat delivered to load}}{\text{electricity consumption}} \tag{1.1}$$

An example schematic of an HP is shown below in Figure 1.1. Note that the temperatures listed are only for illustration and that HP design temperatures can vary considerably depending on the application. Heat is extracted from the relatively cold source. The temperature is boosted as the pressure is raised through the compressor. Heat is then delivered to the load at a significantly higher temperature than the source. To complete the cycle the refrigerant passes through an expansion valve, lowering the pressure and temperature for heat extraction.

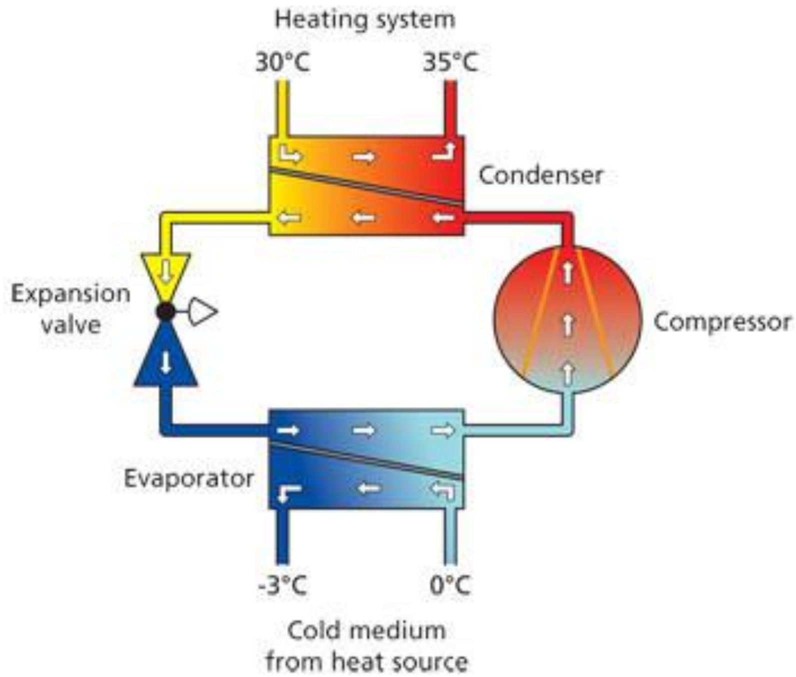

Figure 1.1 - Typical heat pump schematic [3]

If a heat pump replaces an electrical resistance heating system, it can be expected that electricity usage will drop by about 1/2 to 1/3 of the historic heating energy consumption for a given building. This corresponds to a typical COP of about 2-3 achievable by current technology. The COP of a HP system depends heavily on the type of technology used and the temperatures at which the cycle operates.

Another option for replacing conventional heating sources is solar thermal energy. There are two main types of residential solar thermal collection devices: air- and water-based. This research focused on the water-based type. Solar thermal collectors (STCs) are an alternative energy source commonly used for supplying domestic hot water (DHW). Significant energy and emission savings can be achieved by using STCs in lieu of combustion fuels. A sample schematic of a solar domestic hot water (SDHW) system is shown below in Figure 1.2.

STCs consist of a surface with high solar absorption in contact with tubes through which a working fluid is circulated. Heat is transferred to the fluid as it flows through the tubes. The hot fluid typically flows into a water tank for storage. In all but the hottest climates, STCs are typically glazed to reduce heat loss. STCs are often supplemented with a backup heat source for periods of insufficient incident solar radiation.

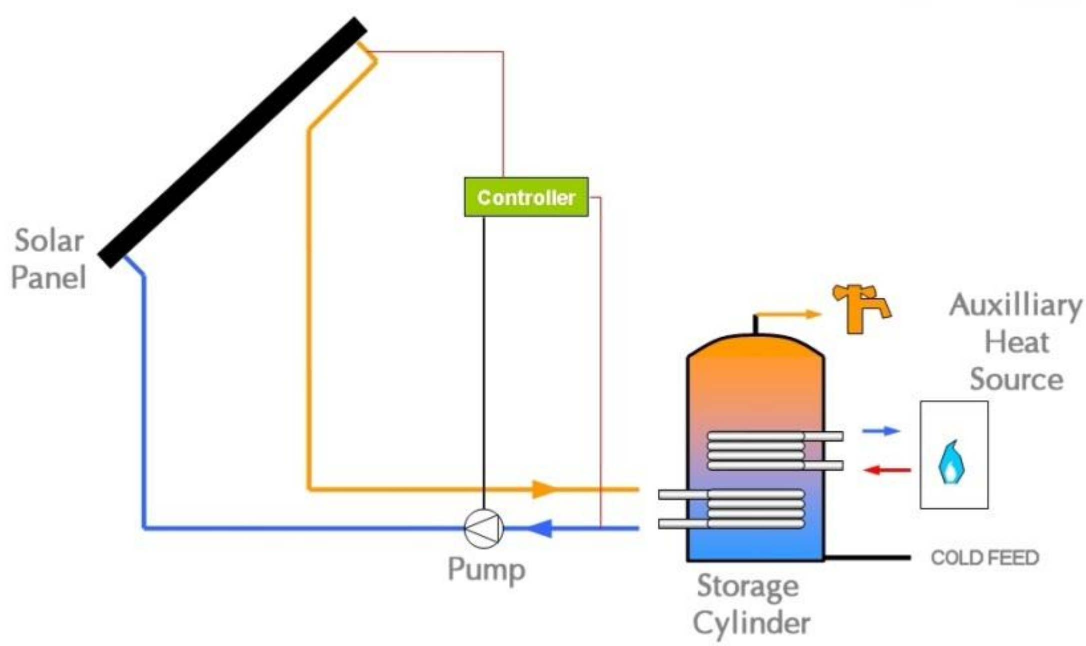

Figure 1.2 - SDHW example schematic [4]

The performance characteristics of HPs and STCs suggest that it may be possible to combine them into a synergistic system, i.e. a solar-assisted heat pump (SAHP). STCs perform better when the fluid temperature entering the collector is lower. This can be accomplished by using a heat pump to extract heat from the fluid leaving the STC. The removed heat is delivered to the storage tank. Figure 1.3 is a typical performance curve for a STC. The chart illustrates that the operating point of a STC can be shifted to the left by reducing the fluid temperature sent to the solar collector, resulting in a higher efficiency. In this chart, the efficiency (η) of the solar collector is the fraction of incoming solar energy delivered to the working fluid at a given time, based on environmental conditions, calculated as:

$$\eta = \frac{Q_u}{A_c G_T} \tag{1.2}$$

Where Q_u is the rate of useful energy collection, A_c is the collector area, and G_T is the total irradiance on the plane of the collector. The efficiency is plotted against a value proportional to the thermal losses of the collector:

$$\frac{T_m - T_a}{G_T} \tag{1.3}$$

Where T_m is the mean temperature of the fluid within the collector and T_a is the ambient air temperature.

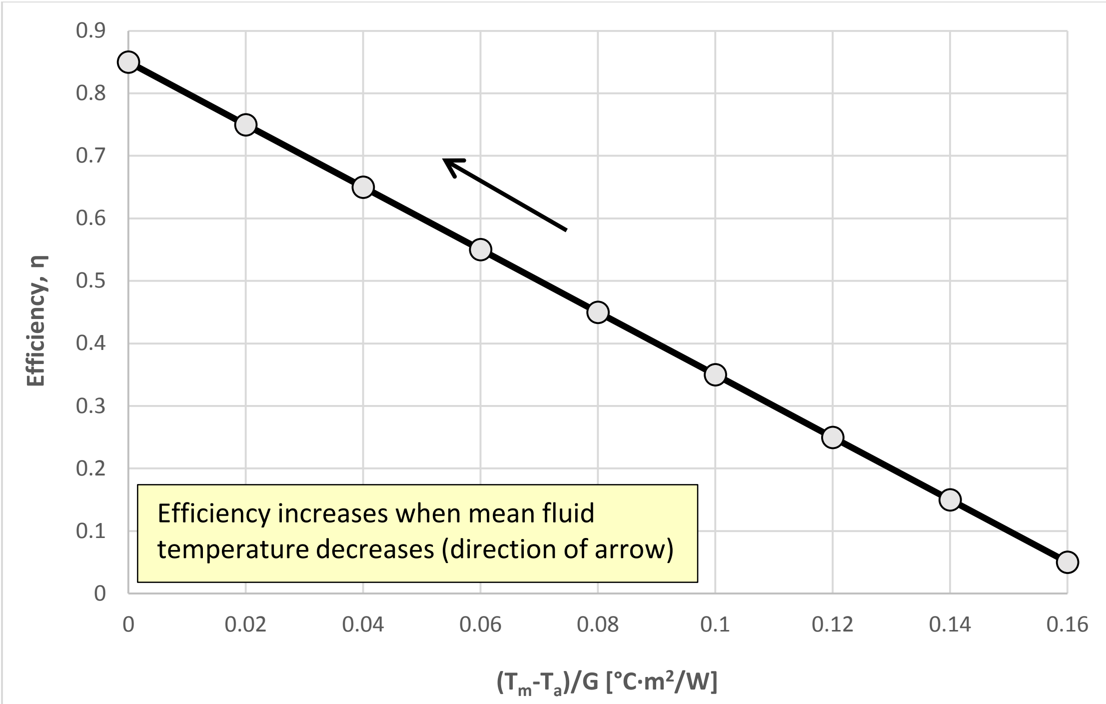

Figure 1.3 – Glazed flat plate solar thermal collector sample performance

The goal of the HP is to extract additional energy from the loop, reducing the mean fluid temperature within the collector. This shifts the operating point of the STC up and to the left, leading to increased STC efficiency. This makes it possible to collect more energy over the course of a given day in comparison to a traditional SDHW system. To further illustrate the approach, sample comparisons of SAHP performance to SDHW are shown for summer and winter in Figure 1.4 and Figure 1.5.

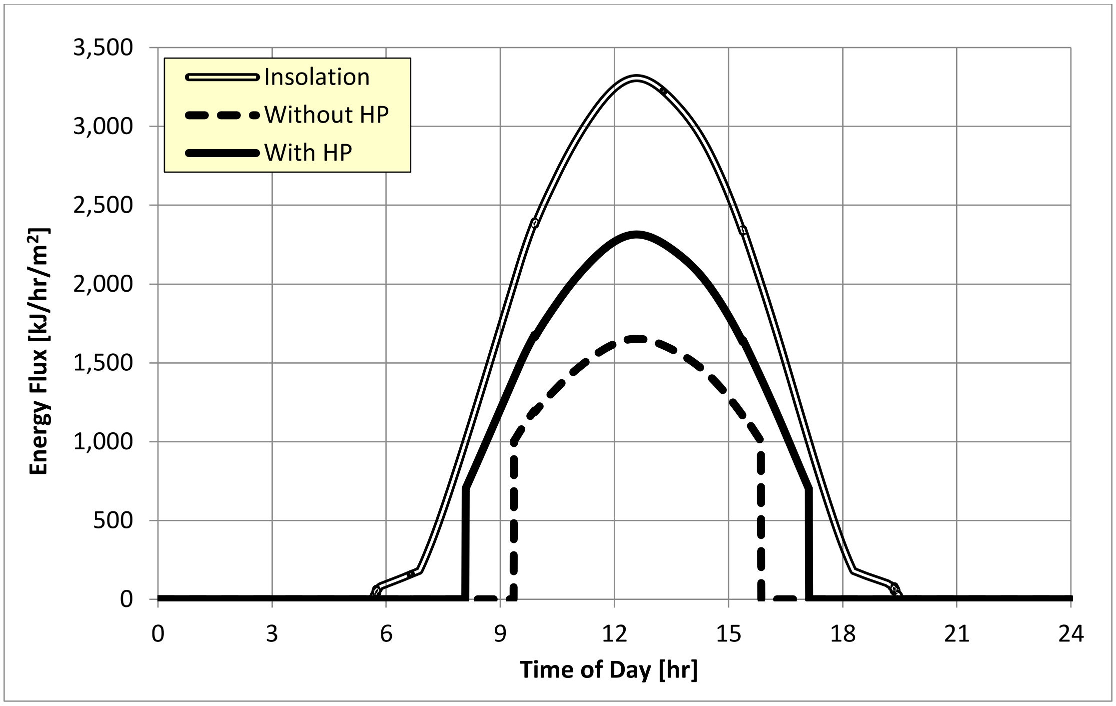

Figure 1.4 - Sample SDHW and SAHP performance during summer

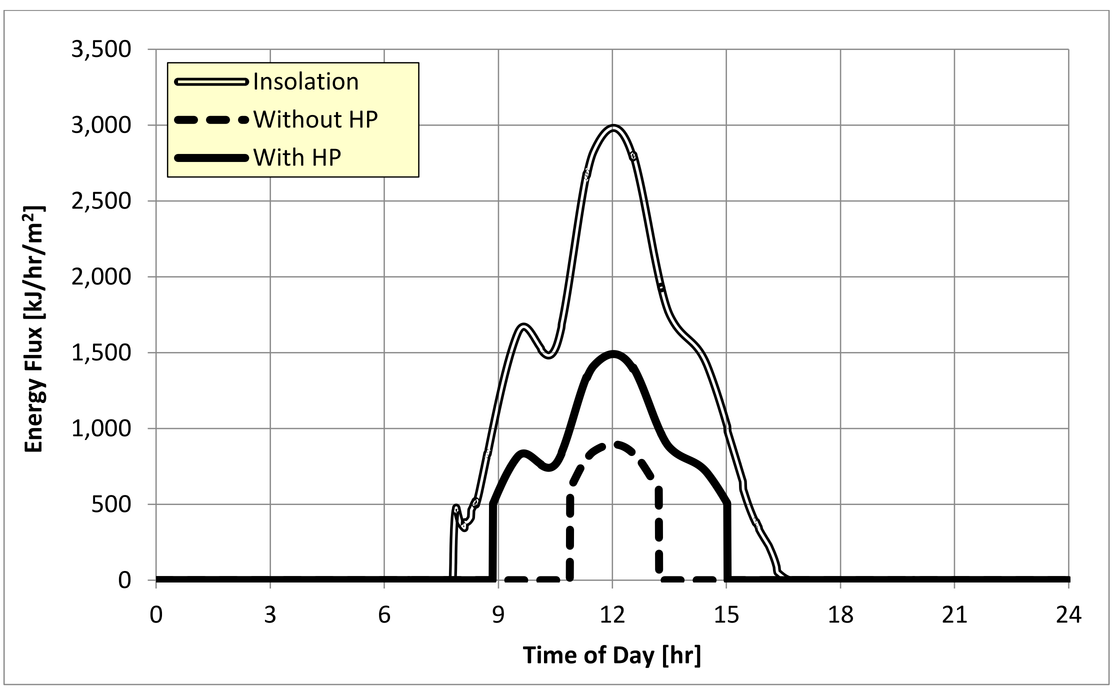

Figure 1.5 - Sample SDHW and SAHP performance during winter

The above figures illustrate two advantages of an SAHP over a traditional STC system. First, the efficiency of the system throughout the day is improved with the inclusion of a HP. Second, the SAHP system is able to operate for a longer period of the day, resulting in more energy collected. The drawback is that electrical power must be supplied to the HP, but this is more efficient than supplying power to a resistance heater. However, the increased efficiency comes with the drawback of additional capital costs and complexity. In addition to the HP benefitting the STC, the HP also benefits *from* the STC.

HPs have a higher efficiency when the difference between the source and sink temperature is smaller. The theoretical maximum COP of a HP is only dependent on the temperatures of the hot load and cold source. This relationship is shown in Equation (1.4).

$$COP_{HP,Carnot} = \frac{T_h}{T_h - T_l} \tag{1.4}$$

Where T_h is the hot side inlet temperature and T_L is the cold side inlet temperature. Considering the hot side temperature of the HP to be fixed, the COP can be improved by increasing the temperature of the cold side. On sunny days the STC will raise the fluid temperature above ambient air conditions, increasing the HP efficiency. The mutual benefits that HPs and STCs offer each other make it possible for a combined system to operate using less energy than either of the systems would individually. A sample configuration of a SAHP system is shown below in Figure 1.6. Two liquid storage tanks are used in this configuration: a DHW tank to store water ready for distribution throughout the building and a large tank to accumulate larger amounts of lower quality solar energy. The DHW tank can be charged using either the heat exchanger or heat pump, depending on operating conditions.

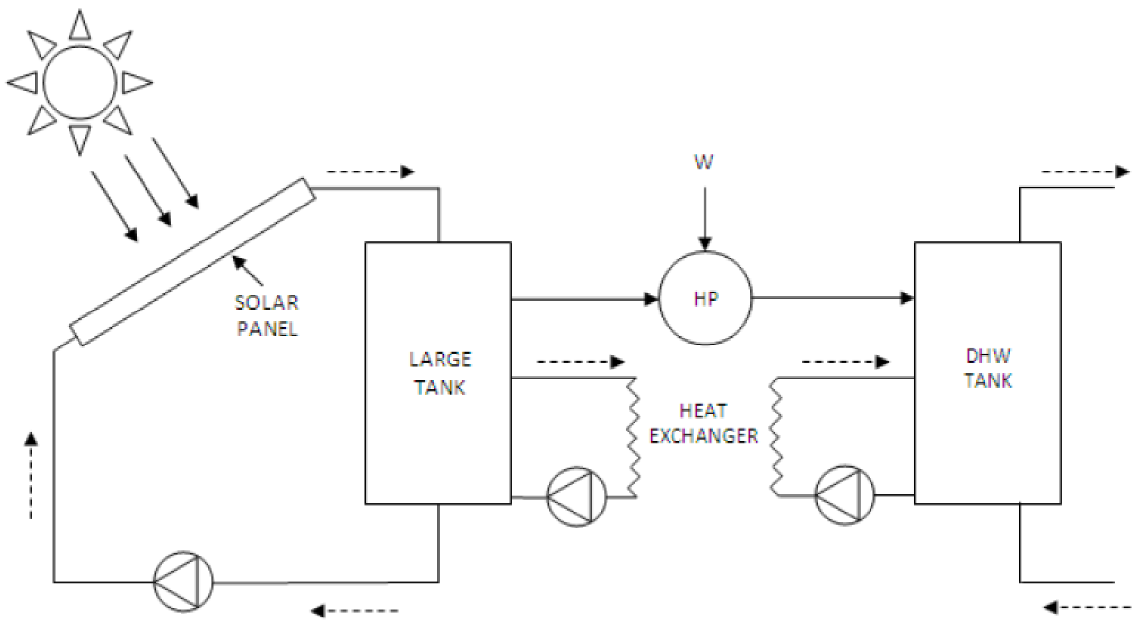

Figure 1.6 - SAHP sample schematic, dual tank system [5]

An overall strategy of using multiple conservation and efficiency technologies in combination with one another is usually good practice for low-energy buildings. The type of heating system proposed by this work would benefit from other energy conservation efforts, for example: efficient building envelopes, low flow fixtures, and drain water heat recovery. Reducing the energy demand of a building allows for smaller equipment, thus providing savings on capital and operational costs.

1.1 Objective, Scope, Approach

The objective of this thesis is to investigate the performance of a dual tank SAHP system in comparison to alternatives. The overall goal is to analyze the feasibility of a system of this type and identify methods for strengthening economic justifiability. The scope of this project is limited to analyzing system performance for a single family residential home with occupancy of approximately 5 people. The dual tank SAHP system is applied to the DHW load of the building.

The proposed system was modelled in Transient System Simulation Tool (TRNSYS) to allow for detailed analysis of performance under a variety of conditions. Model components were tuned in comparison to real equipment that are part of a purpose-built test apparatus. Realistic component models are established from tuning data to improve confidence in the model results.

The tuned model is used to conduct a variety of parametric studies to investigate the impact on annual system performance. Recommendations are formed from the parametric studies to assist in the design process of a dual tank SAHP system. The time-of-use electricity consumption of the system is analyzed to identify if potential exists for further savings and peak electricity shaving. The system is compared in many global climates to identify locations which would be suitable for installation.

Chapter 2 Literature Review

2.1 Solar Space Heating and Domestic Hot Water Systems

Humans have been aware of the immense power of the sun for millennia. Ancient architecture of many civilizations often made passive use of the sun for lighting or heating. Accounts of primitive active solar collectors being used for heating applications date back to the 19th century [6]. One of the first documented systems consisted of a cylindrical tank which was painted black and mounted on a rooftop. Since then, solar heating systems have been separated into multiple components, including an exterior mounted collector, interior storage tank, and circulation pumps. System configurations are numerous and depend strongly on the application and location.

Storage techniques for solar heating applications have advanced significantly over the past hundred years. In addition to the segregation of the collection and storage system into separate components, storage methods and materials have been investigated. A water-glycol mixture is commonly used in place of plain water to address the freeze up issues of temperatures below 0°C.

A stratified storage tank holds the hot water entering from the heat source and supplied to the load at the top and the cold water entering from mains and supplied to the collector at the bottom. It has been demonstrated that the use of stratified tanks increases storage and collection efficiency [7]. Researchers have also investigated the use of phase change materials in solar heating systems in order to increase the thermal density of the storage system and provide an isothermal storage medium [8]. Other storage methods that have been investigated include ground-coupling [9] and rock beds [10].

2.2 Indirect Solar Thermal Assisted Heat Pump Systems

The combination of solar thermal and heat pump technologies into a single system has potential for energy savings. An early prototype system by Terrel [11] in the 1970s compared the effectiveness of a solar-assisted heat pump with a conventional solar system. He argued that the combined system is not much more effective than the conventional. However, his system only used a single source heat pump, with natural gas for backup. In addition, heat pump technology has improved over the past 30 years, increasing performance and reliability significantly. These improvements warrant reinvestigation of SAHP systems.

There are many possible configurations for SAHP systems and determining optimal configurations has become a major part of the research effort. Indirect SAHP (ISAHP) systems refer to configurations in which the heat pump is connected to the solar source through a heat exchanger; this is in contrast to Direct SAHP (DSAHP) where the solar collector becomes the evaporator of the refrigeration cycle. It is worth noting that the literature has used ISAHP in contradiction to refer to both “Indirect” and “Integral-Type”. In this work, the terms “Direct” (also known as “Integral-Type”) and “Indirect” will be used to describe the two main classes of combination systems.

In the early 1980s, Chandrashekar et al. [12] used computer models to explore the economic feasibility of 6 systems for 7 Canadian cities. At the time, it was determined that some of the configurations were economically justifiable over electric resistance heating for multiplex dwellings, but not for single family homes. This finding suggests that an effective way to justify SAHP systems is by increasing the scale to service multiple units or a community. However, these findings should be reinvestigated since many factors affecting the viability of SAHP systems have changed drastically in the past 30 years, including: utility costs, equipment costs, equipment efficiency, and building loads.

Bridgeman et al. [13] explored the performance of a system with the solar and storage loops connected only by a heat pump. Computer simulations validated with laboratory tests in Kingston, Ontario, Canada resulted in COP values from 2.8 to 3.3. Nuntaphan et al. [14] tested a similar system to Bridgeman et al., with the heat pump connecting the solar collector to the storage tank. The study, conducted in Thailand, determined that the addition of a heat pump to a solar thermal system significantly increased the hot water temperatures in the storage tank.

The two aforementioned studies require that the heat pump be run at all times when collecting solar energy. There are many occasions where the heat pump is not needed to boost temperatures, therefore excess electricity is being used. The incorporation of a heat pump bypass to a heat exchanger, which allows the system to operate as a conventional solar setup, has the potential to reduce electricity consumption. Operating the

HP only when required results in longer equipment life. This is an important consideration for future work in this area.

Scott Sterling [5] performed a feasibility analysis of two ISAHP systems using Transient System Simulation Tool (TRNSYS) for his Master's Thesis at the University of Waterloo, Canada. The two SAHP systems were compared to a traditional SDHW system and an electric DHW tank. The solar fraction of the SDHW system was found to be 0.58. The SAHP systems improved on this by roughly 0.1, resulting in solar fractions of 0.66 and 0.67. This work demonstrates that the solar fraction of solar heating system can be improved by the inclusion of a heat pump.

There are some aspects of the problem that were not addressed by Sterling. First, Sterling identified that the TRNSYS simulations were not validated, but should be in the future. This was an important issue that is addressed by this research project. Second, he recommended that a lifecycle cost analysis be performed to evaluate the systems' economic feasibility. The increased equipment costs and service life of the SAHP system must be compared to conventional alternatives.

It was also noted that system performance was sensitive to the timing of water draws. This is to be expected since the heat source of solar energy is abundant during daytime hours and completely absent during the night hours. As a result, water draws in the early morning are not as easily met with solar energy as water draws in the evening. Some test standards, such as "Packaged solar domestic hot water systems" from Canadian

Standards Association (CSA) specify water draws throughout the day, even though the majority of households have no water draws in-between 9 AM and 5 PM for weekdays [15].

The two SAHP configurations examined by Sterling required HPs of different sizes. The SAHP with two storage tanks (shown previously in Figure 1.6) was able to operate with a larger capacity HP since the float tank acted as a thermal buffer, protecting the system from freeze-up. The solar side system, pictured below in Figure 2.1, was limited to a smaller HP capacity since it operates within the solar loop and has no significant thermal buffer. The reduced size of the HP consumes less electricity; however, the ability to meet the heating demand is decreased when compared to the dual tank ISAHP.

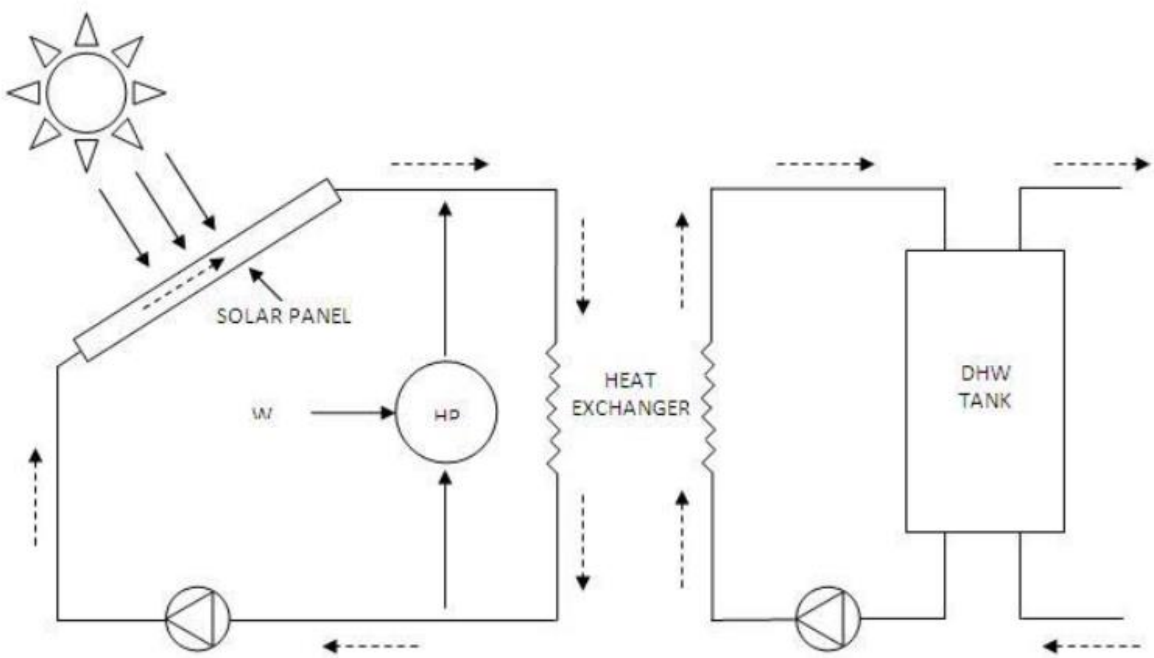

Figure 2.1 - Solar-side ISAHP

Two graduate students, William Wagar and the author, continued the work of Sterling following the completion of his Master's thesis. A brief summary of Wagar's progress is outlined in the following paragraphs. Wagar focused on a single tank SAHP system, investigating the performance of a derivative system of that shown in Figure 2.1.

Wagar built a model of a single tank SAHP system in TRNSYS and performed preliminary simulations. Developing an effective control strategy was a major part of the modeling effort. Early simulations identify that oscillating control signals could be an issue due to the complexity of the system. It was also identified that the single tank SAHP system may require an HP of small capacity, confirming a suggestion from Sterling.

Addressing the recommendations of Sterling is a major aspect for the continuation of his work. Wagar and the author designed and built a test rig that is able to validate the performance of SAHP components and an overall system. The test rig is controlled using a series of programs written using National Instruments' LabVIEW graphical development environment. Details of test rig design and operation are listed in Section 3.2.

An extension of the ISAHP concept is a dual-source configuration. The heat pump cycle can receive heat from either the solar collector *or* a secondary source, such as air or ground. Kaygusuz [16] investigated the performance of a dual-source ISAHP using computer simulations and a test apparatus in Trabzon, Turkey. It was shown that the dual-source SAHP had higher energy savings than both the series and parallel systems. In order to meet heating loads and conserve energy, it is recommended to select an appropriate supply water control temperature to switch between the solar and secondary source. Finally, it is concluded that heat storage type and size is a critical part of a successful SAHP and must be chosen based on climatic conditions.

In 2007, a solar community of 52 homes was built in Okotoks, Alberta, Canada, about 40 km south of Calgary. The solar energy project consists of R-2000 energy efficient homes, garages with rooftop solar thermal collectors, a borehole thermal energy storage field, and short term thermal storage tanks. Ongoing monitoring of the site shows that a solar fraction over 90% is achieved each year. The impressive performance of this project highlights the benefit of large scale storage. The borehole field storage is effective because its losses are low since it is sized for 52 homes [17]. The addition of a heat pump to this district system would allow the natural gas backup to be eliminated. A heat pump could be used to boost the temperature of the heat extracted from the ground to meet the demand that is currently not met by solar energy.

Recent work at Ryerson University in Toronto, Ontario, Canada explored the combination of solar thermal and ground source heat pumps (GSHP) [18]. STCs were used to supplement the space heating supply of a GSHP for a heating dominated house. A similar system was designed for an office building in Tianjin, China in collaboration with Ryerson University [19]. Two GSHP systems with two dedicated borehole heat exchangers were used to mitigate the negative impact of heating/cooling load imbalance. One GSHP system supplies the cooling for the whole building and approximately 75% of the heating load. The second GSHP has STCs included and only supplies the remaining 25% of the heating load.

Building on the above projects, studies have explored the feasibility of a SAHP system with ground source in a cold climate [20]. Historical loads from a real house were used to specify energy demands, but the performance of the model was not validated experimentally. Although ground heat exchanger length could be reduced, the study concluded that adding solar thermal collectors to a ground source heat pump system provided small economic benefit.

Much research on these systems has investigated the feasibility and energy savings using models alone. Although these studies do have use for preliminary analysis of a system, improving confidence in models requires experimental validation. Yumrutas et al. developed an analytical model to predict the long term performance of an SAHP system utilizing an underground storage tank. The average water storage temperature over several years of operation was calculated and varying ground properties were considered [21]. Using an unvalidated analytic and computational model, it was found that 5 years would be required for the ground source to reach annual periodic operating conditions and that course graveled earth provides the best thermal performance.

Carbonell et al. analyzed the potential energy savings of an SAHP for space heating and DHW. This work developed a numerical model to compare SAHP performance to air and ground source heat pump systems without solar collectors. In both cases electricity input was reduced, and it was concluded that locations with cold temperatures and high irradiation benefit the most [22]. For Davos, Switzerland, it was found that the two systems had fractional electricity savings of 46-49%.

Loose et al. identified that many field tests exist for the separate technologies of solar thermal collectors and heat pumps, but not for systems combining the two technologies. Their work reported the results of in-situ monitoring of a brine to water heat pump using shallow geothermal and solar thermal energy. It was found that the system performed well given that it was not optimized, having a seasonal COP of about 3.5 [23]. Loose et al. also share the view that experimental validation of system operation and models is lacking.

Some research projects have undertaken experimental model validation, but there is still a large gap in knowledge and findings in this area due to the wide variety of system configurations possible. Experimental validation has commonly been done using in-situ monitoring of operating systems. Chow et al. created and validated a model for an SAHP system used to heat a swimming pool [24]. Although the study was successful in validation and performance, it is difficult to extend the results to other system types and thermal load characteristics.

Panaras et al. investigated the performance of a domestic hot water SAHP in comparison to a numerical model and found good agreement for high radiation conditions, but poor agreement for low radiation conditions. Overall, it was found that the SAHP could save about 70% of auxiliary energy usage in comparison to an electric hot water tank. Once again, the variety of configuration possibilities limits extending these results. The heat pump and solar thermal collectors in this study can only be used independently [25].

Tamasauskas et al. investigated the performance of an SAHP system with a latent storage medium of an ice slurry. Experimental validation confirmed the ice mass and tank fluid temperature predictions. This simulations compared the performance of an SAHP with sensible storage against the same system with latent storage, and found that operating energy savings in comparison to an electrical resistance heating system was increased from 81% to 86%, respectively [26].

SAHP systems can also be used to deliver process heat for industrial processes. Sevik et al. developed and installed an experimental ISAHP system for drying mushrooms. They found that the COP was improved to about 3.0 when using solar thermal collectors with a heat pump. In contrast, the COP without solar energy was about 2.2 [27]. This result confirms that significant performance gains are achievable with SAHP systems. Again, the inability to extend these specific results to more general scenarios limits the applicability of the findings.

The other major class of SAHP systems, direct, will be discussed briefly, followed by concluding remarks stemming from the literature review. Only a select portion of the literature regarding direct systems will be discussed, since these systems differ greatly from indirect systems in configuration and operation.

2.3 Direct Solar Thermal Assisted Heat Pump Systems

In contrast to indirect systems, Direct SAHP (DSAHP) systems replace the traditional evaporators of heat pump cycles with solar thermal panels. The operating refrigerant flows through the panel and supplies heat directly to the refrigeration cycle. Therefore, DSAHP systems eliminate a heat exchange loop that would be necessary in an ISAHP system. This theoretically improves performance [28]. A major drawback of DSAHP systems is the need to design a solar collector which must also satisfy the requirements of a heat pump evaporator: for example, containing high pressure gas refrigerant.

Chaturvedi et al. investigated the effects of a variable speed drive compressor on the performance of a DSAHP system. Their results indicate that COP can be improved by reducing the compressor rotation speed during warm months. Theoretical predictions from computational results agreed with their experiments [28].

Chyng et al. developed a model for a DSAHP system and simulated the performance considering an operational period of one year. It was found that the COP of the system ranged from 1.7 to 2.5 throughout the year, but was generally higher than 2.0. Experimental results agreed with the model very well [29].

Huang et al. built and evaluated a DSAHP consisting of a small reciprocating compressor and an unglazed solar collector. The measured COP of the system was between 2.5 and 3.7. It was found that above a solar insolation of about 400 [W/m^2] the COP of the system remained relatively constant. Below this threshold, the COP dropped off significantly as solar insolation decreased [30]. The performance of this system could be improved by using a more efficient compressor than the reciprocating type, such as a scroll-type compressor.

Li et al. experimentally analyzed the performance of a DSAHP specifically designed for the Chinese market. Their system had a very high COP of 6.6 during high intensity incident solar radiation and 3.1 during an overcast day [31]. The relatively high COP during minimal incident solar radiation was achievable due to the use of unglazed collectors. The uninsulated solar thermal collectors act as heat exchangers with ambient air during periods of low solar energy availability.

Many more studies of DSAHP systems have been conducted; however, since the focus of this research project is indirect systems, the studies of direct systems are not summarized at length. The configuration and operating characteristics of direct systems are far different than indirect systems since the solar thermal collector is adopted as the refrigerant cycle's evaporator.

2.4 Summary

The literature consistently concludes that SAHP systems outperform their individual system counterparts. However, there is a lack of knowledge regarding whether or not the lifecycle cost of an SAHP system is an improvement on alternatives. Details of the control strategy used for the SAHP systems mentioned above was commonly lacking throughout the literature. The strategies that were discussed have room for improvement, for example those that have no heat pump bypass.

Despite the lack of focus on control strategies, there were some important findings identified in the previous work. The concept of minimum solar insolation threshold suggested by Huang et al. [30] is logical. Energy delivered by the heat pump to the load is limited to the energy flow from the solar panels to the heat pump. For the dual-source system investigated by Kaygusuz, a suggestion is made to establish an appropriate switch-over temperature from the solar to secondary source. Control ideas beyond the traditional differential thermostat approach, such as energy flux and temperature limits, will be explored in the present work.

A large gap exists between modelling and experimental investigations of SAHP systems. The majority of studies target one or the other, with model validation being undertaken only rarely. Furthermore, the validated models of SAHP systems are rare relative to the wide variety of configuration types possible. In addition, detailed performance analysis and parametric studies are lacking in the literature. This makes it difficult for the reader to qualitatively assess performance of a given system in a different setting.

In addition, the impact of parameter variation on other key performance metrics, such as thermal stagnation of the system, is not commonly addressed. Direct comparison of multiple system configurations using validated models is an important area of SAHP research that has not yet been addressed effectively. The work of this thesis is built on successful model validation for a novel SAHP system, which is then used to carry out many performance investigations to address deficiencies in the literature.

Chapter 3 Apparatus

3.1 Overview

An SAHP experimental test platform has been built at the University of Waterloo, Waterloo, Ontario, Canada. The design and construction of the apparatus was completed collaboratively with Will Wagar. Any further details on the apparatus not included here can be found in his M.A.Sc. thesis [32]. The apparatus replicates the proposed systems of investigation, except that the solar collector has been replaced with an electrical resistance heater to facilitate laboratory testing. The apparatus is capable of characterizing equipment, identifying and solving true operating issues, and testing control strategies. A system schematic is presented in Figure 3.1 and screenshot of the main LabVIEW control screen is included in Appendix A – Test Apparatus.

The purpose of the experimental apparatus is to tune component models and replicate the behavior of a system installed in a building for domestic hot water production. Therefore, the main goal of the system's operation is to deliver heat to the DHW tank. An electrically controlled floating point valve draws water from the DHW tank to represent a demand from the building. The floating point function of the valve allows the flow rate to be controlled in accordance with the specifications set out by CSA for solar water heating system testing by providing a variable valve opening [15].

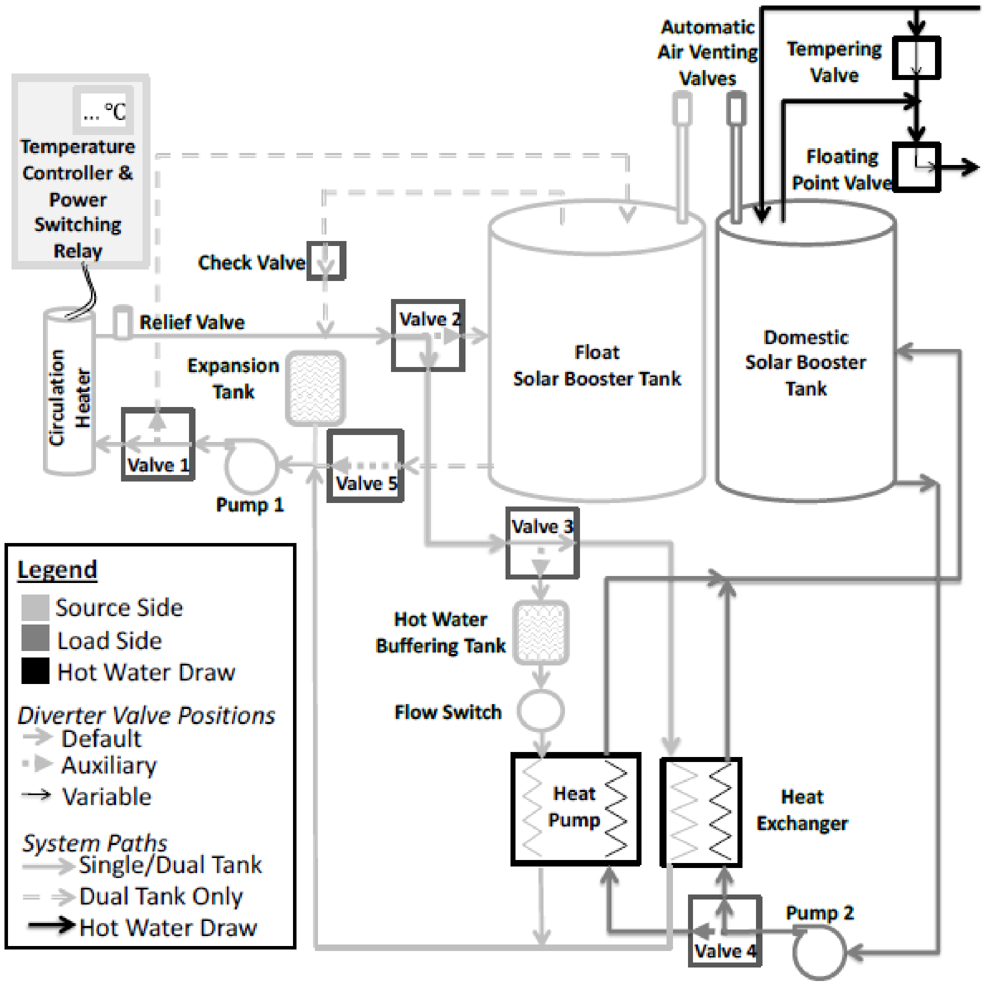

Figure 3.1 – SAHP experimental apparatus system schematic [32]

In order to meet the hot water demand, the test apparatus chooses from a variety of energy sources and sinks. The simplest mode of operation resembles a SDHW system, where energy is collected with the STCs and delivered to the DHW tank via the heat exchanger. If certain incident radiation conditions are met, the heat transfer can instead be done using the HP, which is able to extract additional heat from the solar-side loop.

The second thermal storage tank, labelled float solar booster tank in the schematic above, adds operation abilities. Heat due to excess solar energy can be stored in this tank when the DHW tank does not require heat and there is solar energy available. The float tank can later be used as a thermal source to heat the DHW tank via either the HX or HP. All of these modes are made possible through the use of 3-way diverter valves and variable speed pumps. Variable flow rates are needed to meet the differing heat transfer characteristics of the HX and HP.

The solar input is represented using an electric heater, with the energy input being metered using a controller connected to a PC. All the other equipment is controlled via the PC, through use of custom electronics designed specifically for the apparatus.

3.2 Detailed Design

The components of the test rig are broken down into two categories: system equipment and measurement equipment. System equipment consists of the components which would be required to install a SAHP system. These components are listed below in Table 3.1. The remaining test rig components fall into the category of measurement equipment, meaning they allow for the monitoring and recording of experimental results. These components are then broken down further into instruments and data acquisition, and are listed in Table 3.2 and Table 3.3, respectively. Data acquisition is performed using a National Instruments CompactDAQ system.

Instrumentation used includes: thermocouples, flow meters, and power meters. Brief descriptions of the components are provided following the tables below. The errors associated with these instruments are very small in comparison to the uncertainty of numerical simulations. Further details of the test apparatus and uncertainty calculations relating to measurements from the experimental apparatus are included in the M.A.Sc. thesis of Wagar [32].

Table 3.1 - Apparatus equipment specifications

Equipment	Make and Model	Size	Details
Domestic thermal storage tank	AOSmith SUN 80 110	302.8 L	With 4.5 kW electric heater element
Float thermal storage tank	AOSmith SUN 120 110	450.4 L	With 4.5 kW electric heater element
Heat pump	Ecologix GX-W2W36	3.8 kW	Water-to-water
Heat exchanger	Packless SL15-25	44 kW	25 plate
Circulation pump	TACO 008-VVSF6-IFC	1/25 HP	Variable speed control
Circulation heater	WATLOW CBEC27J10	6 kW	Electric power source

Table 3.2 – Test apparatus instrument specifications

Instrument	Make and Model	Accuracy	Range
Liquid flow meter (Qty. 3)	Omega FTB4607	±1.5% of reading	0.22 to 20.0 GPM
Type T thermocouples (Qty. 3)	Omega TG-T-30-SLE	±0.3 °C	-200 to 350 °C
Current transducer (Qty. 1 ea.)	Magnelab SCT-0750-010 and -025	±1% of reading at 10-130% of rated current	0 to 10 A 0 to 25 A
Potential voltage transformer (Qty. 1)	Magnelab SPT-0375-300	±1% of reading at 10-130% of rated voltage	0 to 230 VAC

The water tanks are standard solar thermal storage units and both include an auxiliary heating element slightly above the midpoint. Each tank includes 4 ports to connect to load and source, as well as 4 other ports for safety pressure release, air release, tank drainage, and thermocouple placement. Although the internal electric heater elements are rated for 4.5 kW at 240 VAC, the elements are supplied with a 208 VAC source, reducing their output to 3.38 kW.

Table 3.3 – Test apparatus data acquisition specifications

Component	Make and Model	Quantity	Purpose
Chassis	NI cDAQ-9178	1	Computer interface
Thermocouple module	NI 9213	2	Measure and record temperatures
Digital IO module	NI 9401	2	Flow meter readings, valve control
Analogue input module	NI 9205	1	Power meter readings
Analogue output module	NI 9265	1	Variable speed pump control

The heat pump is a water-to-water unit custom built by Ecologix Heating Technologies Inc., Cambridge, Ontario, Canada. The compressor technology used is a fixed speed scroll type with a nominal output capacity of 3.8 kW. R123-a refrigerant is used because it is most appropriate for the main operating conditions expected for the SAHP system. A flow switch is included to ensure that the HP does not run if no flow is being supplied to the source side. The load side of the HP has integrated overheating protection, whereby electrical power is disconnected from the compressor if the condenser pressure becomes too high.

An electrical resistance circulation heater is used in lieu of solar thermal collectors to facilitate laboratory testing of the system. The output power of the heater is reduced to

4.5 kW due to connection with a 208 V electrical supply, which allows the system to simulate up to nearly 4 m^2 of STC area. Heater controls were developed in LabVIEW to provide two methods of operation. The power output of the heater can be controlled to simulate the performance of a STC over the course of a day. It is also possible to control the output temperature of the heater to test and characterize equipment.

Variable speed pumps are used to supply different flow rates depending on the mode of operation. HP operation requires much higher flow rates than HX operation. The pumps are controlled using an analog current output. A feedback loop utilizing the flow meters adjusts the pump power to reach the desired flow rate.

A standard tempering valve on the supply side of the system ensures that water delivered to the load does not exceed 55°C. Three way valves are used to control flow direction for the various modes of operation. Water mains supply in the laboratory building is plumbed to the test rig as a cold water source. A floating point valve allows water draws to be simulated. The three way and floating point valves operate on a 120 VAC supply which is controlled by digital transistor-transistor logic (TTL) outputs through an appropriate relay.

Three flow meters monitor the flow rates on the source side, load side, and DHW draw. They provide a method for calculating heat transfer rates when the temperature change is also measured. Voltage and current are measured on the HP and DHW heating element. The HP can be characterized using the measured power consumption and the heat transfer rate. DHW heating element power consumption is monitored to measure the amount of energy consumed by auxiliary heating.

The test rig includes approximately 32 thermocouples to monitor temperatures throughout the system. Typical placements are on the inlet and outlet of devices to provide a method for calculating heat transfer rate. For example, 4 thermocouples are assigned to the heat exchanger to measure the inlet and outlet temperatures on the source and load side. In addition to measuring temperature differences across devices, 5 thermocouples are placed in each water tank to measure the storage temperature. These thermocouples are distributed vertically with equal spacing to monitor stratification within the tanks. Thermocouples are also installed at all inlets and outlets of the tanks. The ambient air temperature is measured for the purpose of evaluating heat losses from the equipment.

All the components of the test rig are interfaced through the Data Acquisition system (DAQ). Various inputs and outputs within the DAQ allow for monitoring, data collection, and operational control. Two dedicated thermocouple modules with built-in cold-junction compensation are used to record temperatures throughout the system. A digital I/O card is used in input mode to count pulses from the flow meters which are converted to flow rates. A second digital I/O card is used in output mode to control all the electronic valves in the system and signal the HP on and off. An analog output card controls the speeds of the circulation pumps. An analog input card is used to measure the voltages from the power metering equipment, from which the power consumption of a device can be calculated. For more information on the design, construction, and operation of the experimental apparatus the reader should refer to Wagar’s M.A.Sc. thesis, as this was a primary focus of his project [32].

3.3 Limitations of Test Apparatus

3.3.1 HP Capacity

The HP used in the test apparatus was the smallest scroll compressor HP that could be acquired, and it was a custom build. Through the experimental work of Wagar, it was identified that the HP cannot be operated continuously because its nominal capacity of 3.8 kW is too large relative to the solar energy input. This forced Wagar to adjust the control strategy for the test apparatus and his TRNSYS simulation, whereby the HP is operated on a duty cycle. The HP is cycled on and off throughout solar energy collection via Solar-HP-DHW to prevent the source side temperature from reaching the freezing point of water [32]. This is not a desirable way to operate a SAHP system. This limitation of the experimental apparatus forced Wagar to adapt a control strategy for his TRNSYS simulation that is not desirable for an installed system.

3.3.2 Solar Heater Capacity

At 4.5 kW, the power output of the heater used to simulate solar energy input restricts the capabilities of the test rig due to its low output relative to the nominal heat pump capacity. It restricts the amount of STC area that can be simulated in experiment to about 4 m^2. This creates further limitations of heat pump operation, since it is the relative capacity of the HP and STC that determines whether or not system operation can be stable in Solar-HP-DHW mode. The need to cycle the heat pump during Solar-HP-DHW is compounded by this limitation.

3.3.3 Pump Flow Control

The flow meters in the apparatus do not allow accurate measurements below approximately 2 L/min. Below this flow rate, the measured flow rate oscillates between 0 L/min and a measured value. This limitation negatively impacts the flow control routine in the apparatus control software, since the variable speed pumps cannot be controlled accurately without feedback from the flow meters. This limitation prevents the tuning of components at flow rates lower than approximately 2.5 L/min, which is used as a minimum to prevent nearing the flow meter instability at 2 L/min. For the simulated heat exchanger, it was desirable to operate at a flow rate lower than this minimum of the apparatus. In this case, the results from heat exchanger tuning are extrapolated to the desired flow rate.

3.4 Summary

The test rig designed and constructed through this work can successfully support the tuning of component models for SAHP systems. Limitations imposed by the apparatus design and equipment selection restrict direct validation of daily results of overall system performance from the TRNSYS model. This was identified by the work of Wagar [32] and required undesirable control adaptations for his modeled system.

Despite these limitations, the test apparatus is able to characterize equipment accurately. Experimental investigations on component performance facilitate tuning of TRNSYS models to build confidence in realistic performance. The test apparatus is a valuable and effective tool for evaluating the quality of component models, in particular thermal storage tank models.

Chapter 4 Investigated System

There are many different ways to design a SAHP system, which is a benefit, but this potential creativity leads to the need to answer many questions. The exploration of system performance and optimization has resulted in the evolution of system design. The alternatives discussed in the literature review varied from direct systems, where the solar panel becomes the evaporator in the refrigeration cycle, to indirect systems, where a heat exchanger connects the STCs to the HP.

Beyond choice in overall system type is the possibility of using an additional 'float' storage tank to boost collection of solar energy in the summer due to extra volume and a lower temperature than the DHW tank. The float tank is allowed to fall in temperature during fall and winter months, making it possible to collect energy that a SDHW tank cannot due to the DHW tank being at a higher temperature. The addition of this extra tank, termed a 'float tank' because its temperature is permitted to fluctuate, makes several modes of system operation possible. The model developed, its modes of operation, and the details of its control strategy are presented in this chapter. A simplified schematic of the system is shown below in Figure 4.1.

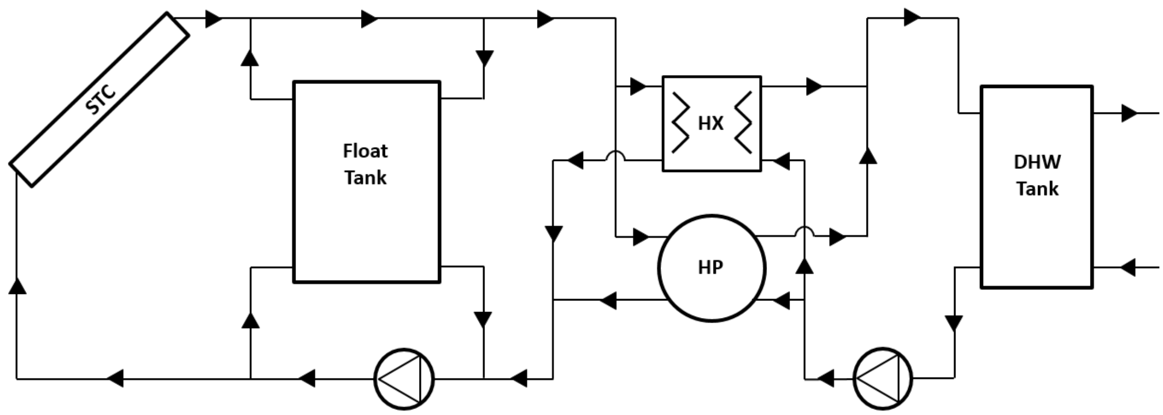

Figure 4.1 – Simplified schematic of dual tank SAHP system studied

The ideal heat source for the system is the STC, which can be directed to either HX, HP, or float tank. All of these options collect solar energy for immediate of future delivery to the DHW tank. The float tank is used as an energy source when solar energy cannot be collected using the STC. Energy can be transferred from the float tank to the DHW tank via either the HX or HP.

4.1 Detailed Model Description

The construction of an experimental apparatus allows for the testing of components and modes of operation, but is limited by the main constraint of operating only in real time. Predicting the performance of an SAHP system over the course of a full year would require 8760 hours of operation for the test apparatus. This would be an energy-, time-, and cost-intensive process, and can be done far more effectively using accurate models. The approach taken in this work is to tune component models using the test apparatus to establish confidence in realistic performance. This section describes the TRNSYS model developed for simulation in detail and the following chapter reports the model tuning results.

The model includes storage tanks, pipes, solar thermal collectors, a heat exchanger, and heat pump, and hydronic components to direct fluid flow. A simplified schematic of the TRNSYS model for the investigated system is illustrated in Figure 4.2. In reality, there are many more background components included in the model to facilitate data collection, analysis, and presentation. The components used for these purposes are listed below in Table 4.1.

Table 4.1 – TRNSYS output components used

Component Name	TRNSYS Type
Calculator	-
Integrator	24
Printer	25c
Online Plotter with File	65a
Online Plotter without File	65d

The majority of components used in the model are standard, with the types listed below in Table 4.2. All the standard components are well-documented within TRNSYS and are open source to TRNSYS users [33].

Table 4.2 – Standard TRNSYS model component names and types [33]

Component Name	TRNSYS Type
Solar (Solar thermal collector)	1b
Pipe	31
Tee	11h
Div (Flow diverter)	11f
Tank (Thermal storage)	4b
Pump (Hydronic)	110
HX (Heat exchanger)	5b
Temper (Tempering valve)	11b
TMY2 (Weather)	109
CSA-C (Water draw)	14b

The final component, HP, is a custom type programmed by Wagar, making the type number arbitrary. This component completes various practical checks, such as ensuring the inlet flow conditions are within the bounds deemed acceptable. The inlet source temperature of the heat pump is not permitted below 0 °C since freeze-up would occur in water. The energy transfer rates are calculated using the correlations established in the previous chapter for the heat pump being used. Refer to the preceding chapter and Wagar for these details [32]. The heat pump used in simulations is scaled to 0.6 of the size used experimentally.

Conventional SDHW heating systems use a thermostat with deadbands to control the operation of the circulation pump, collecting solar energy when it is available and feasible to do so. The typical setup has two temperature sensors as inputs to the thermostat. One temperature sensor is located at the bottom of the tank, where the fluid exits and is conveyed to the solar panel or heat exchanger. The second temperature sensor is located at the outlet of the solar collector, where the fluid is conveyed back to the storage tank or heat exchanger. This conventional thermostat control strategy is inadequate for SAHP systems where multiple modes of system operation exist.

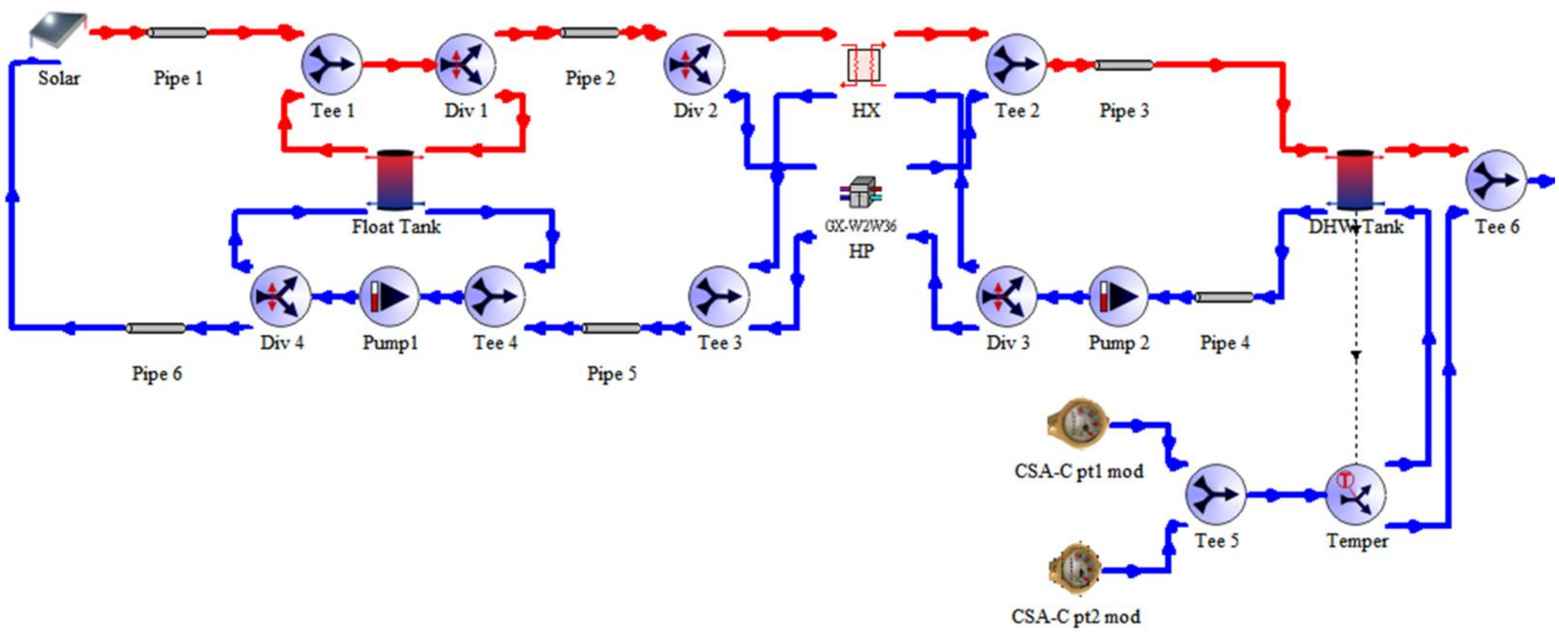

Figure 4.2 – TRNSYS model schematic

4.2 Modes of Operation

The additional system capabilities make system control more complicated than a simple 'on' or 'off' decision. The addition of the heat pump results in the need to more carefully consider operation because the heat pump makes it possible to collect energy when a SDHW cannot, and is able to boost temperatures when operating. The addition of a second storage tank creates further complexity, as priorities must be established which preserve the ability of the DHW tank to meet demands, but also maximize energy collection by using the float tank when appropriate.

These extra control decision requirements posed by the dual tank SAHP necessitated the creation of a custom controller. The controller must determine which of the following possible system operation modes listed in Table 4.3 is most appropriate given the circumstances encountered.

The priorities of the possible operating modes reflect a combination of a need to meet the DHW demands, minimize the electrical consumption of the system, and maximize the collection of solar energy. Although the collection of solar energy is generally beneficial, collecting that energy and not putting it to effective use is wasteful. This is the reason that the direct use of solar energy to heat the DHW tank is first priority and using it to heat the float tank has least priority. This is also a reason why it would not be desirable to have an operation mode of Solar-HP-Float. Although it may be possible to collect extra solar energy, it does not pose a clear net benefit since any electricity input to do so could be outweighed by tank losses by the time the energy is delivered to the DHW tank.

Table 4.3 – SAHP system modes of operation (Note: single tank SAHP system only uses mode 3; dual tank SAHP system uses all other modes)

Mode	Name	Description
1	Solar – DHW	Solar energy is captured using the STCs, circulates to the HX where is exchanged to the DHW loop and into the DHW tank.
2	Float – DHW	Previously collected solar energy that was stored in the float tank is circulated to the HX where it can be transferred to the DHW tank.
3	Solar – HP – DHW *(only used for single tank SAHP system)*	Solar energy is collected in a similar way to mode 1, but is directed to the HP to increase the rate of heat transfer to the DHW tank.
4	Float – HP – DHW	Heat is extracted similar to mode 3, but the HP is used, allowing energy deposit to the DHW tank even when the float tank temperature is lower.
5	Solar – Float	If DHW tank does not require heat input, direct any available solar energy to the float tank for future use towards charging DHW tank.

The next best option after using the mode 1, Solar – DHW, is to meet DHW demand via mode 2, Float-DHW. This mode uses previously collected solar energy in the float tank to meet DHW demand. This mode is useful during non-daylight hours if the DHW tank and float tank were sufficiently charged during the preceding day. It does not negatively impact the operation of Solar – DHW, as this mode would be prioritized if it can operate.

Mode 3, Solar – HP – DHW, is used to boost output from the STCs within a range of incident radiation thresholds. This mode improves the amount of solar energy collection by reducing the thermal losses of the STCs. This mode runs if the DHW tank needs heat during the periods of the day when incident radiation at the STCs falls within the threshold range.

Similar to Float – DHW, Float – HP – DHW is intended to be used when solar energy availability is low. This occurs during overcast or cold days, or outside of daylight hours. If the float tank was charged sufficiently high enough during previous days, it may be possible to transfer heat to the DHW tank without using the HP. Otherwise, the HP is used to boost the output temperature to the DHW tank. The Float – HP – DHW mode is important for reducing operating costs outside of the summer months.

Regardless of which mode of operation above is possible, the final option for meeting the DHW load is to use the auxiliary heat source. Since this mode does not use any solar energy, it is the least cost effective method for heating. It operates based on a traditional DHW tank thermostat with deadband, maintaining the water temperature available to load by the DHW at a minimum of 55°C. The final mode of operation, Solar – Float (mode 5), is used when the DHW tank does not require heat, or when the auxiliary energy source is operated. Operation of modes 2 and 4 relies on stored energy from operating mode 5.

The interrelatedness of these modes and the process of decision making is illustrated in a flow diagram in Figure 4.3. With the operation modes of the system and their priorities established, the next step is to determine what modes of operation are feasible to operate at a given time.

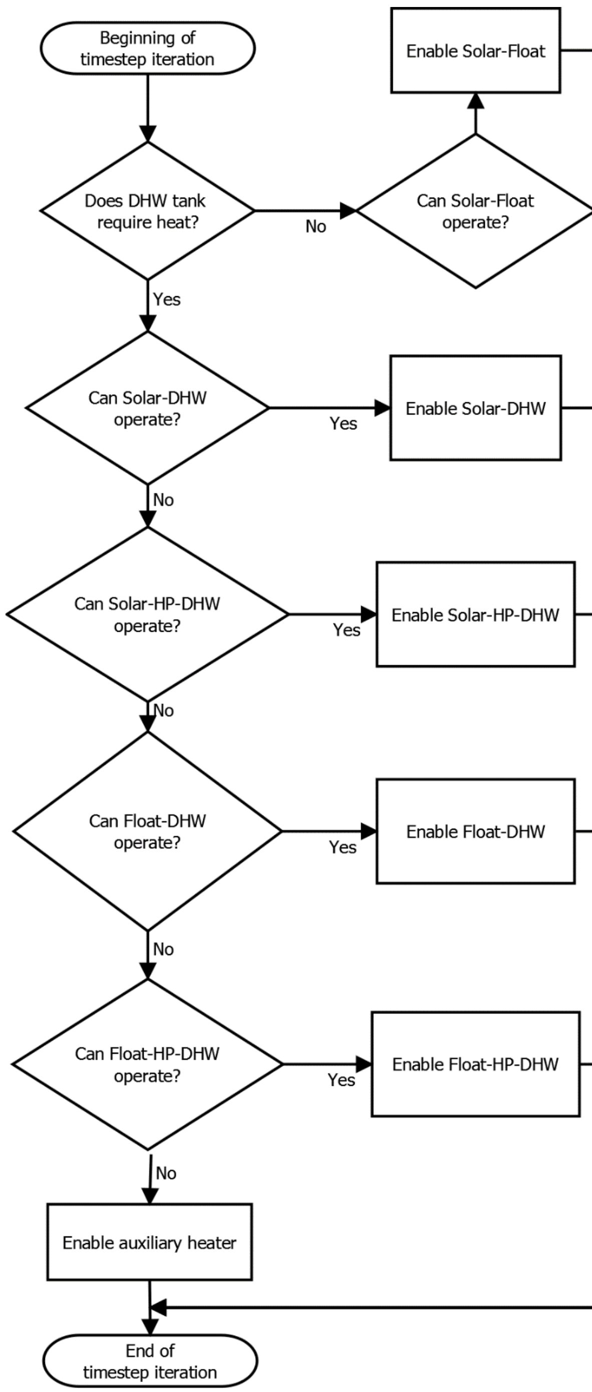

Figure 4.3 – Flow chart of dual tank SAHP system controller logic

4.3 Control Strategy

A custom controller was programmed in Fortran 95 for use in TRNSYS to control the dual tank SAHP system outlined in section 4.1 Detailed Model Description. The following section details the design of and decision making process followed by the controller. The parameters, inputs, and output for the custom controller are described in Table 4.4,

Table 4.5, and Table 4.6, respectively. The source code for this custom component is provided in Appendix D – Custom Component Source Code.

Table 4.4 - Parameters of the custom controller component

Parameter	Description
Solar-DHW db upper	Upper deadband for Solar – HX – DHW operating mode. The relevant temperature difference must be greater than this value to begin operating this mode.
Solar-DHW db lower	Lower deadband for Solar – HX – DHW operating mode. This mode continues operating until the relevant temperature difference falls lower than this value.
Solar-Float db upper	Upper deadband for Solar – Float operating mode. The relevant temperature difference must be greater than this value to begin operating this mode.

Parameter	Description
Solar-Float db lower	Lower deadband for Solar – Float operating mode. This mode continues operating until the relevant temperature difference falls lower than this value.
DHW setpoint db	Used when DHW tank previously did not require heating to determine if temperature difference between it and the setpoint has increased beyond the deadband
# of oscillations	Limits the maximum number of control decision oscillations per time step. Required to ensure control stability during iterations on a given timestep.
Short-cycle time	Minimum amount of time that system must remain in a given operating mode. Prevents rapidly turning equipment on and off to prolong life.
HX flow rate	A scaling value used to determine the fraction of full speed flow rate by pumps to be used when operating the heat exchanger.
HP flow rate	A scaling value used to determine the fraction of full speed flow rate by pumps to be used when operating the heat pump.
OFF to HP threshold	The threshold of horizontal incident solar radiation above which the HP can begin to operate in conjunction with the STCs.

Parameter	Description
HP to OFF threshold	The threshold of horizontal incident solar radiation below which the HP can no longer operate in conjunction with the STCs.
HX to HP threshold	The threshold of horizontal incident solar radiation above which the HX is utilized instead of the HP.
HP to HX threshold	The threshold of horizontal incident solar radiation below which the HP is utilized instead of the HX.
Fluid specific heat	The specific heat of the working fluid of the system.

Table 4.5 – Inputs of the custom controller component

Input	Description
DHW source	Monitors the temperature within the DHW tank that is fed to the source – the temperature of the bottom node in the tank.
DHW load	Monitors the temperature within the DHW tank that is fed to the load – the temperature of the top node in the tank.
DHW setpoint	Temperature value DHW tank is maintained at by auxiliary element.

Input	Description
Float source	Monitors the temperature within the float tank that is fed to the source – the temperature of the bottom node in the tank.
Float load	Monitors the temperature within the float tank that is fed to the load – the temperature of the top node in the tank.
Solar out	Monitors the temperature at the outlet of the STC array.
Solar in	Monitors the temperature at the inlet of the STC array.
Ambient	Monitors the outdoor ambient air temperature.
Allow Float-DHW	A binary value that specifies whether the mode of Float – DHW tank can operate. (Note: 'Allow Solar-Float' must also be enabled.)
Allow Solar-HP-DHW	A binary value that specifies whether the mode of Solar – HP – DHW tank can operate.
Allow Float-HP-DHW	A binary value that specifies whether the mode of Float – HP – DHW tank can operate. (Note: 'Allow Solar-Float' must also be enabled.)
Allow Solar-Float	A binary value that specifies whether the mode of Solar – Float can operate. (Note: Must be used in conjunction with either or both of 'Allow Float-DHW' and 'Allow Float-HP-DHW'.)
Radiation tilted	Monitors the energy flux of solar radiation reaching the tilted surface of the STC array.

Table 4.6 – Outputs of the custom controller component

Output	Description
System Mode	An integer value from 1 to 5 indicating the system's current mode of operation.
Div 1	A binary control signal indicating the position of diverter 1, switching flow from the float tank to either the HX or HP.
Div 2	A binary control signal indicating the position of diverter 2, switching flow between the HX and HP on the source side.
Div 3	A binary control signal indicating the position of diverter 3, switching flow between the HX and HP on the load side.
Div 4	A binary control signal indicating the position of diverter 4, switching flow between the float tank and the STC array.
Pump 1	A pump control signal for the source side between 0 and 1 indicating the fraction of full speed flow rate.
Pump 2	A pump control signal for the load side between 0 and 1 indicating the fraction of full speed flow rate.
Heat Pump	A binary control signal indicating whether or not the HP should be operational.
DHW element	A binary control signal indicating whether or not the DHW tank auxiliary element should be operational.

To determine if Solar – DHW mode can operate during a given simulation timestep, one of two decision paths must be taken, depending on whether or not this mode is currently operating. If this mode was running during the previous timestep, and the temperature difference between the DHW source and solar outlet exceeds the lower deadband limit, the mode can continue to run. If the mode was not previously running, the temperature difference must exceed the upper deadband to permit operation. A similar control strategy is used for determining the operation of Float-DHW and Solar-Float.

The control strategy to operate the Solar-HP-DHW mode requires more care because an energy balance between the input and output must be considered. If solar energy input is low, energy extraction by the HP may drop the evaporator temperature too low, risking ice formation. For this purpose, the amount of radiation on the tilted STC array is monitored and thresholds are used to determine if this mode can operate.

The final mode, Float-HP-DHW, operates on the basis of minimum float tank temperatures. The mode is permitted to begin operate if the float tank temperature is above 10°C and can continue to operate until the tank temperature falls to 5°C. This strategy ensures that enough stored energy is available to justify running the mode, and that the tank temperature does not fall too close to the freezing point of water.

The first major decision made by the controller is whether to deliver heat to the DHW tank. If the DHW load requires heat, modes of operation that might meet the need are evaluated in order of priority. Top priority is given to modes that deliver heat to the load with the least input of energy and that utilize currently available solar energy rather than stored energy. The auxiliary heater is the mode with the lowest priority and is used only when no other modes are possible. The overall goal of the control strategy is to minimize electricity consumption while maximizing collection of solar energy.

Below the high level control strategy, there are decisions occurring at a lower level, such as the implementation of deadbands and a short-cycle timer to prevent control decisions from oscillating rapidly. In addition, there are specific comparisons made at the mode level to determine whether the mode is possible given the current conditions. For example, incident radiation thresholds are used to numerically predict if the HP modes can operate in a stable manner. The reader can interpret the finer details of the control strategy by referring to Appendix D – Custom Component Source Code.

4.4 Improvements Made Over Test Apparatus

A key part of establishing an effective system model was to reduce the capacity of the HP to better match the solar energy availability during periods of low solar insolation. To achieve this, a HP scaling factor of 0.6 was used to enable stable operation of the system. This enables the HP to operate continuously during periods of low insolation rather than cycling, as was required for the test apparatus.

The function of the model and controller enable additional modes of operation required for the dual tank SAHP system. The model utilizes experimentally tuned components that were investigated by Wagar [32] and others that are presented in Chapter 5 - Model Tuning. The control strategy was designed to be entirely flexible, enabling the simulation of many system types, including SDHW, single tank SAHP, and dual tank SAHP. These system types can be separately simulated with the controller by enabling or disabling the appropriate modes in

Table 4.5 – Inputs of the custom controller component. For example, the single tank SAHP is simulated by disabling all modes other than ‘Allow Solar-HP-DHW’. This functionality of the controller is a key model feature that enables performance comparison of these systems, which is a major aspect of this research project.

Chapter 5 Model Tuning

The objective of this chapter is to tune components used in the TRNSYS (TRaNsient SYstem Simulation tool) model of the dual tank SAHP system. The TRNSYS component models are tuned using a test apparatus that has been built specifically for this purpose. Experimental results are compared to output from TRNSYS simulations to enable tuning of component parameters to ensure realistic performance. The tuned components are then used in the TRNSYS model of the SAHP system, as presented in Chapter 4 - Investigated System, which enables annual simulations for comparison.

The author collaborated with Wagar in his validation work, starting from the conceptualization and design of the test rig. Will Wagar’s M.A.Sc. thesis focused on design, construction, and commissioning of the test apparatus. The author and Wagar worked together to make the test rig functional and enable it to support equipment characterization and model validation. This included extensive work programming the test rig controls and data acquisition procedures in LabVIEW and planning experiments.

An important contribution of Wagar is the characterization of the heat pump, since this was a major deficiency in the work of Sterling [5]. The findings of Wagar’s heat pump characterization are summarized in section 5.1.5 - Heat Pump. The heat pump model developed through his characterization is used in this work. The details of the heat pump model are contained in the MASc thesis of Wagar [32].

Wagar compared the performance of the experimental apparatus and a TRNSYS model of the single tank SAHP system with a modified controller used to cycle the heat pump. In general, it was confirmed that the experimental and simulated results agreed within a reasonable tolerance. It was found that the error of overall energy consumption predicted by the TRNSYS model was less than 10%. Wagar identified that the major limitation in modelling a system of this type accurately was the inaccuracies of thermal storage tank models [32].

Limitations in the test apparatus prevent full validation of the dual tank SAHP system. The approach taken for ensuring a realistic model is to tune individual components to establish appropriate parameters for the TRNSYS model.

5.1 Components

The test apparatus is able to successfully measure the performance and operation of components used in a SAHP system. Individual components can be characterized for the purpose of determining parameters for computer simulations. Wagar's work focused on mapping system components, validating modes of operation, and validating the performance of modified daily system operation [32].

The following operating modes of the system were used to verify and tune component models for the dual tank SAHP system: (1) thermal storage tank heating via internal electric resistance element; (2) thermal storage tank standby losses; (3) solar source charging thermal storage tank via heat exchanger; (4) solar source directly charging thermal storage tank. The experimental and model results of each of these operating modes is documented in detail below. Full details of the parameters and inputs used in the TRNSYS simulations are documented in Appendix B – Model Component Parameters/Inputs.

5.1.1 Internal Electric Resistance Element

The DHW tank was heated from room temperature to about 65°C using the internal electric resistance element. Thermocouples placed at equally spaced heights within the tank allow for the analysis of stratification and comparison to model results. The TRNSYS type used for modelling the storage tank is Type 4b. This model has inlet and outlets, an auxiliary heater, and models tank losses due to wall conduction. A single element was used and was placed in the second node from the top. Five nodes were used to model the tank, each with a height of 0.3 m, with an overall tank height of 1.5 m. A high level of stratification was expected since there was no forced convection of the fluid within the storage tank. The measured and TRNSYS results are both shown in Figure 5.1 and Figure 5.2 for temperature and electricity consumption, respectively. The parameters for Type4 used to model the DHW tank are summarized in Table B.5.

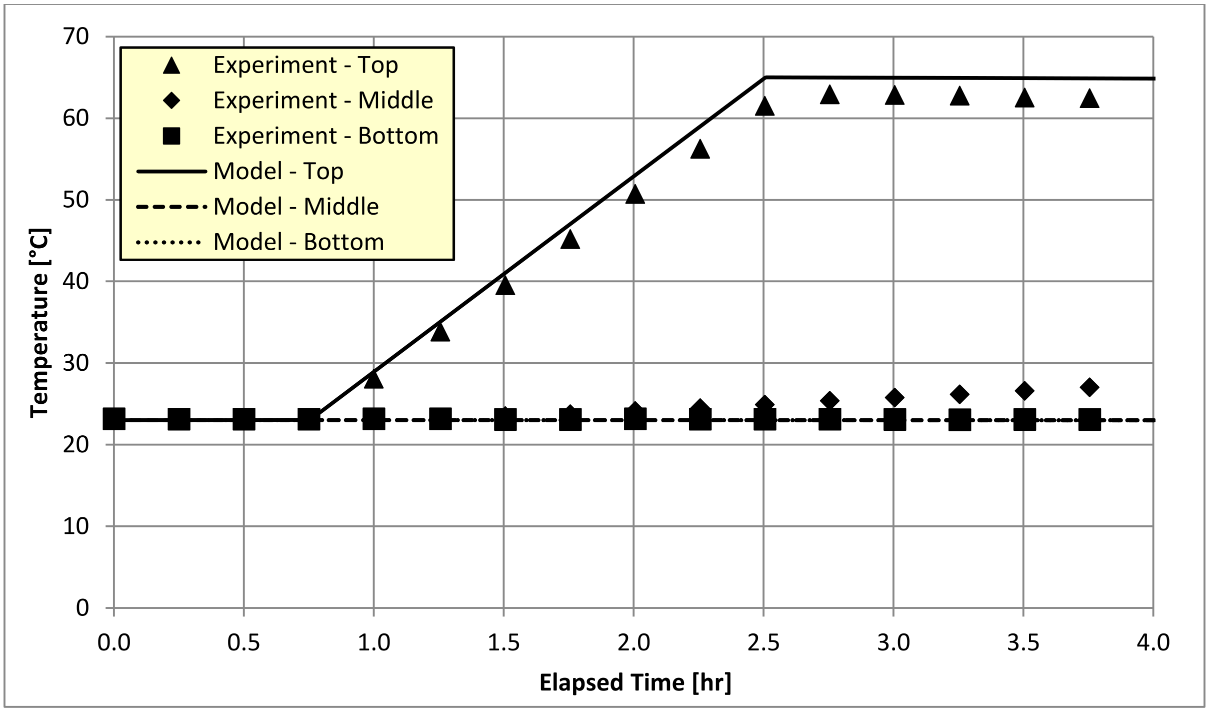

Figure 5.1 – Temperature within DHW tank during charging via internal electrical resistance heater

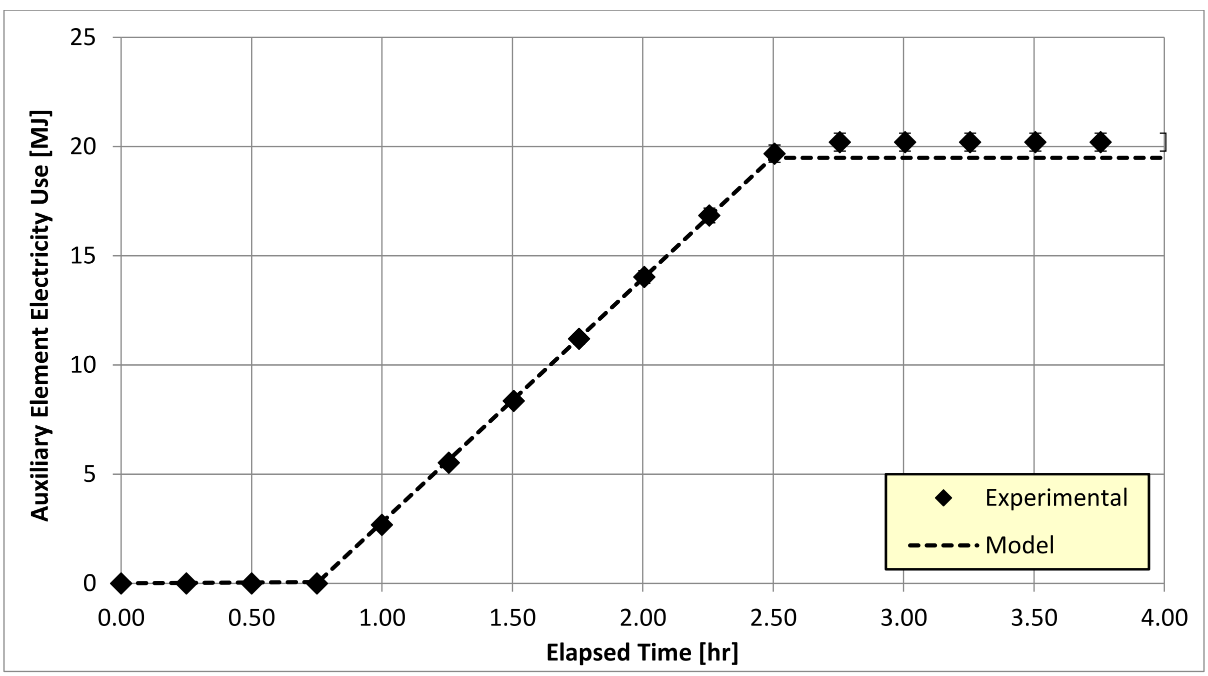

Figure 5.2 – Electricity consumption of internal electrical resistance heater during DHW charging

A high degree of stratification was observed, with the differential across the tank approximately 40°C throughout the test. The tank temperatures at the top rose at very similar rates for both the experiment and model, however the model overestimated this rise by about 2°C. At the bottom of the tank both the model and experiment showed no noticeable temperature rise. Looking finally at the middle of the tank, the model predicted no temperature rise, but the experimental results showed a rise of about 3°C.

This difference in middle tank temperature is due to simplifications made by the model. The model assumes that heat stored within each node does not spread to other nodes via conduction, convection, or radiation. In reality, heat transfer occurs between nodes, which can be seen in the rise of the middle tank temperature for the experimental results. This middle tank temperature continues to rise over time, long after the electrical element has been shut off, due to heat transfer between nodes.

The slope of the energy consumption in Figure 5.2 shows that the electricity consumption rate of the auxiliary heater agrees well for model and experiment. The model was configured at 3.375 kW to match the output rate of the real heating element used in the experiment. The cumulative electricity consumptions differed slightly at the end of the test, with the real heater consuming 20.2 MJ and the model heater consuming 19.5 MJ. This is a difference of 2.4%, which is slightly larger than the experimental uncertainty of the power consumption of 1.4% calculated by Wagar [32]. The main reason for this difference is the modelling approximations made by the stratified tank model. In reality, the experimental setup must supply more heat to reach a certain setpoint temperature since the heat spreads throughout the nodes.

Overall, the agreement between model and experiment is strong, but there are some deficiencies to be noted in the stratified thermal storage tank model. It is difficult to precisely model stratification and heat transfer effects within the tank since they result from natural convection and conduction.

5.1.2 Storage Tank Standby Losses

The thermal storage tank was left sitting for approximately 12 hours to validate standby loss modelling. The average tank temperature at the beginning of the experiment was approximately 65°C, which represents the typical tank temperature during normal operation. The initial conditions for node temperatures in the model were set equal to experiment. Once again, TRNSYS Type 4b was used to model the thermal storage tank. Five nodes were used for the simulation model and five measurement points were installed in the experimental tank to match the middle of the node as closely as possible. The default loss coefficient of -3.0 kJ/hr m^2 K for the TRNSYS thermal storage tank component was maintained as a result of validation. A comparison of model and experimental results for tank standby losses is included below in Figure 5.3.

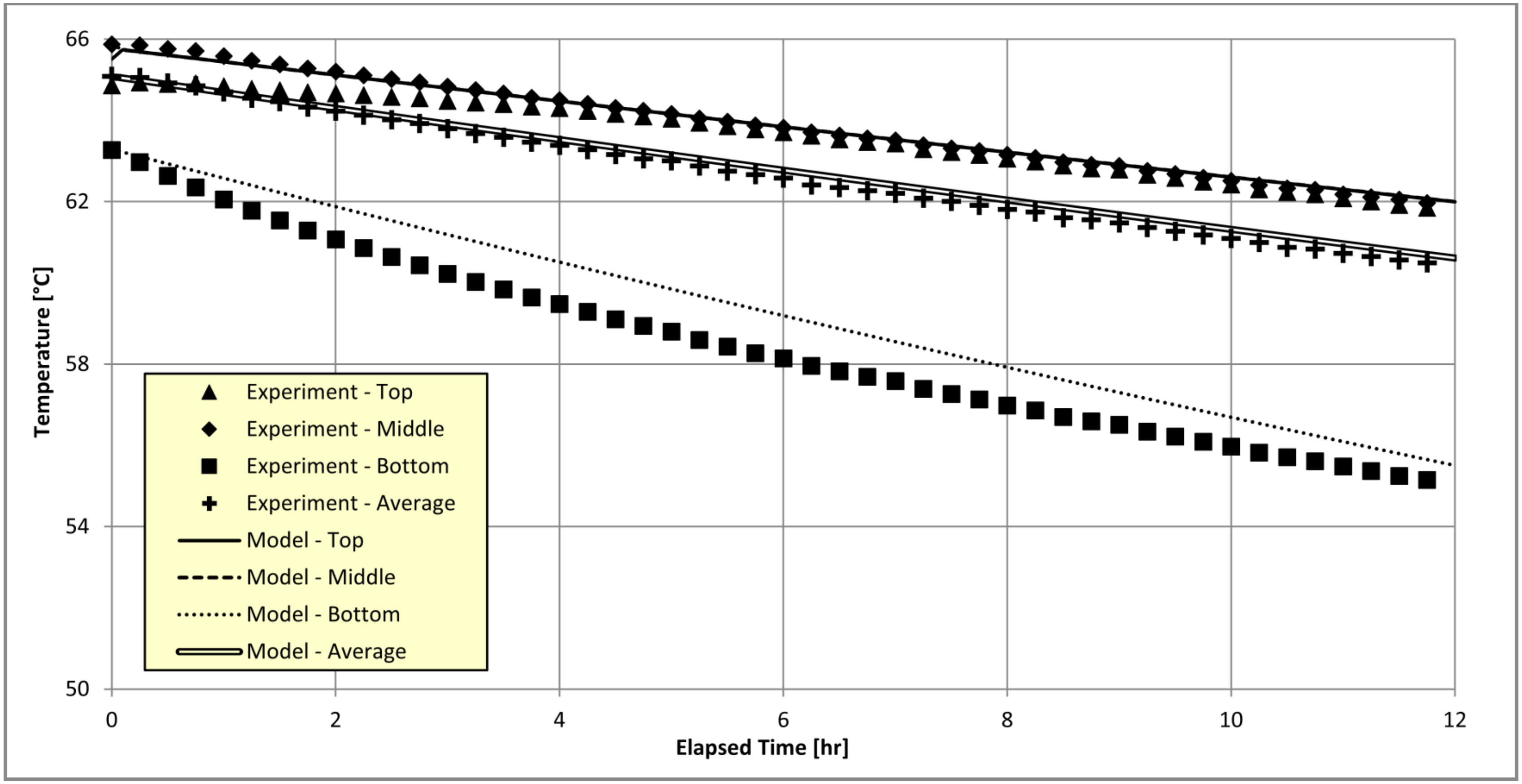

Figure 5.3 – Temperatures within the DHW tank during standby

The loss coefficient of the thermal storage tank in the model was modified until the slope of the temperature decrease between the elapsed time of 6 and 12 hours matched. For both the experiment and model the temperature decrease per time in this time range was 0.34 °C/hr.

Regardless of fine-tuning, there are discrepancies between the model and experiment which are difficult to resolve. TRNSYS models node volumes as a uniform temperature, but the nodes in the apparatus are not uniform temperature and are only measured at one point each. Settling may occur during the standby period, whereby some nodes decrease in temperature more dramatically than others. The TRNSYS model does not account for colder fluid falling to the bottom node from those above. This is a deficiency that is not practical to resolve, therefore it is fortunate that it has a minor effect on the results. It is important to note, but it is expected to be negligible for system operation over time spans of days or longer. In addition, the common operating range of the DHW tank is 50 to 70°C, and the model performs sufficiently well within these temperatures.

5.1.3 Solar Source to Domestic Hot Water Tank via Heat Exchanger

The process of charging the DHW tank with the solar source via the external heat exchanger was modelled and validated experimentally. The following TRNSYS types were used in addition to the aforementioned thermal storage tank: Type 5b – heat exchanger, Type 3d – hydronic pump, and Type 659 – auxiliary heater. The parameter of importance is the heat transfer rate coefficient of the heat exchanger. Although the manufacturer specifies a total heat transfer rate of 44 kW, the conditions for this rating are not provided.

This process also explores the thermal stratification and storage within the DHW tank. In contrast to tank charging with the internal element, it was expected that charging via the heat exchanger would diminish stratification within the tank, since a constant circulation of water causes mixing. The flow rates through both sides of the heat exchanger were kept constant for two separate tests at 5 L/min and 2.5 L/min for several hours. The energy input by the heater in the simulation and experiment was set at 4166 W, with a maximum output temperature of 70°C set to conclude the test. The experimental results are presented below in Figure 5.4 and Figure 5.5.

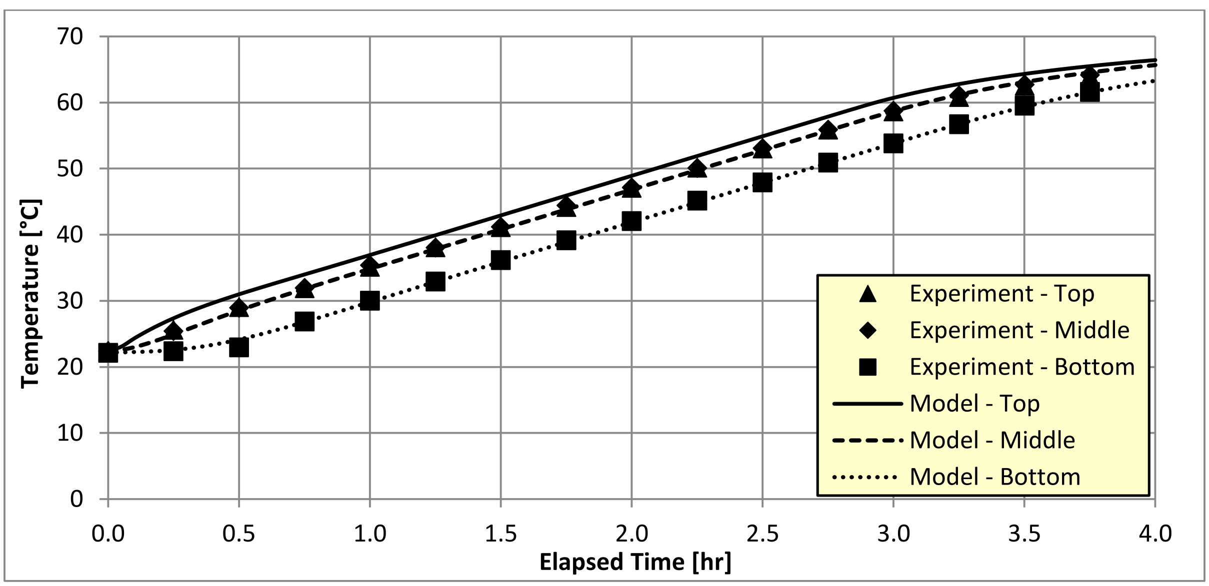

Figure 5.4 – Temperatures within DHW tank during charging via solar source and heat exchanger at 5 L/min

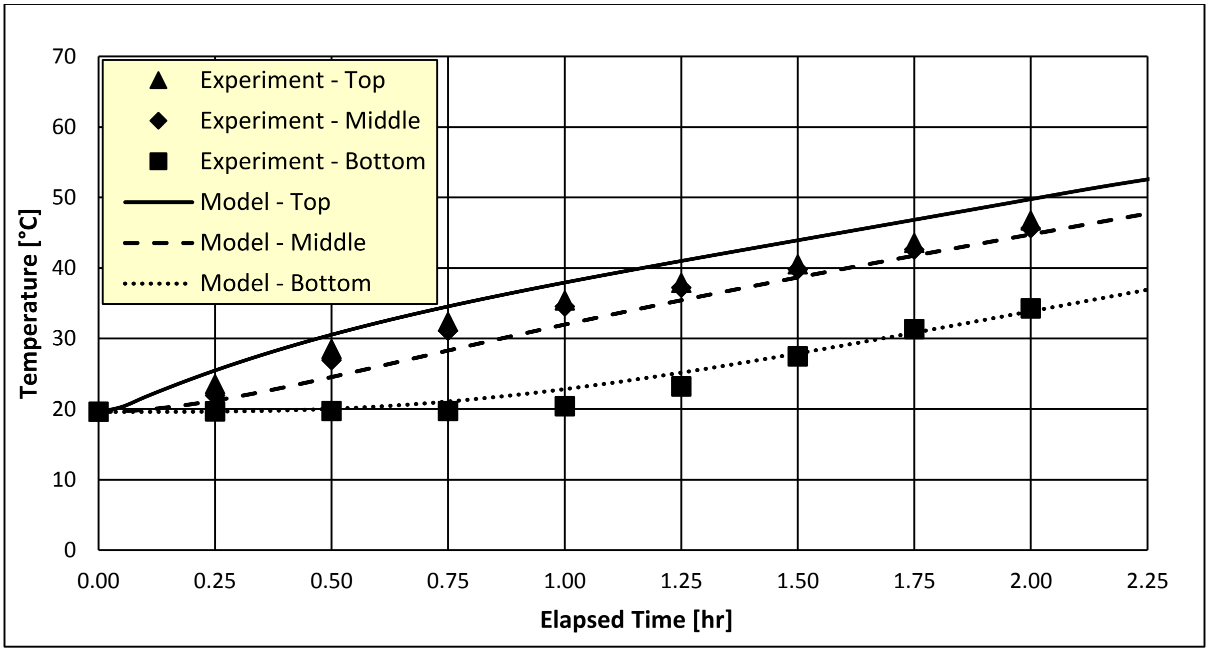

Figure 5.5 – Temperatures within DHW tank during charging via solar source and heat exchanger at 2.5 L/min

In both cases, the model was tuned to match the average temperature rise in the DHW tank per unit time. A summary of the tuning data for the heat exchanger is presented in Table 5.1.

Table 5.1 – Heat exchanger model tuning results

Flow rate [L/min]	Rate of Temperature Rise [°C/hr]	Heat Transfer Coefficient [W/K]
5.0	12.0	800
2.5	12.1	450
1.05 (extrapolated)	-	147

The heat transfer coefficient results at the two tested flow rates were used to estimate the performance at a reduced flow rate of 1.05 L/min. This is the flow rate used with the heat exchanger in the TRNSYS simulation. Linear extrapolation was used to estimate the heat transfer coefficient of 147 W/K. Various heat transfer coefficient values were modeled with a SDHW system to evaluate the sensitivity on the collection of solar energy and use of auxiliary heating. The results from an annual simulation are summarized in Table 5.2.

Table 5.2 – Heat transfer coefficient sensitivity analysis

Heat Transfer Coefficient [W/K]	Solar Energy Collection [GJ]	Auxiliary energy Consumption [GJ]
147	13.07	8.30
294	13.38	7.99
441	13.45	7.90

As expected, the solar energy collection decreases as the heat transfer coefficient decreases due to reduced performance of the heat exchanger. Despite this, the overall change is not very large. The actual performance of the heat exchanger in an installed system will vary. The value chosen for the heat transfer coefficient is rounded to 150 W/K and kept constant for all systems to provide an even basis for comparison.

Note that this testing was completed using water, which has a higher specific heat than a propylene glycol solution that may be needed for freeze protection. A propylene glycol solution would reduce heat transfer rates in comparison to water due to lower thermal conductivity, lower volumetric heat capacity, and higher viscosity. However, if using a drainback solar thermal system, these additional sources of uncertainty would not apply since water could be used.

In contrast to charging the storage tank with the internal element, less significant stratification occurs within the tank while charging with the solar source and heat exchanger. This is due to the large quantity of water that is circulated with a flow rate. In addition, the flow causes mixing to occur where it enters the tank. While using the heat exchanger, the forced convection minimizes the temperature differential throughout the tank to about 5°C. In contrast, the internal element heating results presented earlier showed an overall temperature differential around 40°C. The stratification predicted by the model differs from the experimental results only marginally.

The model over-predicts the tank top temperature and under-predicts the lower temperature. The stratification calculated by the model is ideal and does not account for complex inlet and outlet water flows and diffusion between layers. Typical temperature inaccuracies of the model are 0.5°C, which is not much larger than the temperature measurement uncertainty of 0.3°C.

5.1.4 Solar Source to Thermal Storage Float Tank

Direct charging of the float tank via the solar heat source was investigated to ensure proper performance of the component model used for the float tank. This operation mode is similar to, but simpler than, charging the DHW tank via heat exchanger. The procedure for validating this mode of operation was to circulate water from the float tank directly through the electric resistance heater. The heater's output was set to a maximum temperature of 80°C, which resulted in a constant power input to the fluid until the input temperature rose to about 68°C. Above 68°C inlet temperature, the output of the heater was gradually reduced to maintain a constant output temperature of 80°C.

Once the float tank was sufficiently warm that the resistance heater could achieve an output of 80°C, the resistance heater's power output was scaled down to maintain a constant output until the tank temperature reached 80 °C. This mode was validated from a starting temperature of about 20°C to a finishing temperature of 80°C. Thermostats were used in both the experiment and model to control the final tank temperature. The resulting average float tank temperatures for both cases are presented in Figure 5.6. Note that the temperature variations throughout the height of the float tank are not shown because they do not vary significantly and would therefore be difficult to distinguish in a chart.

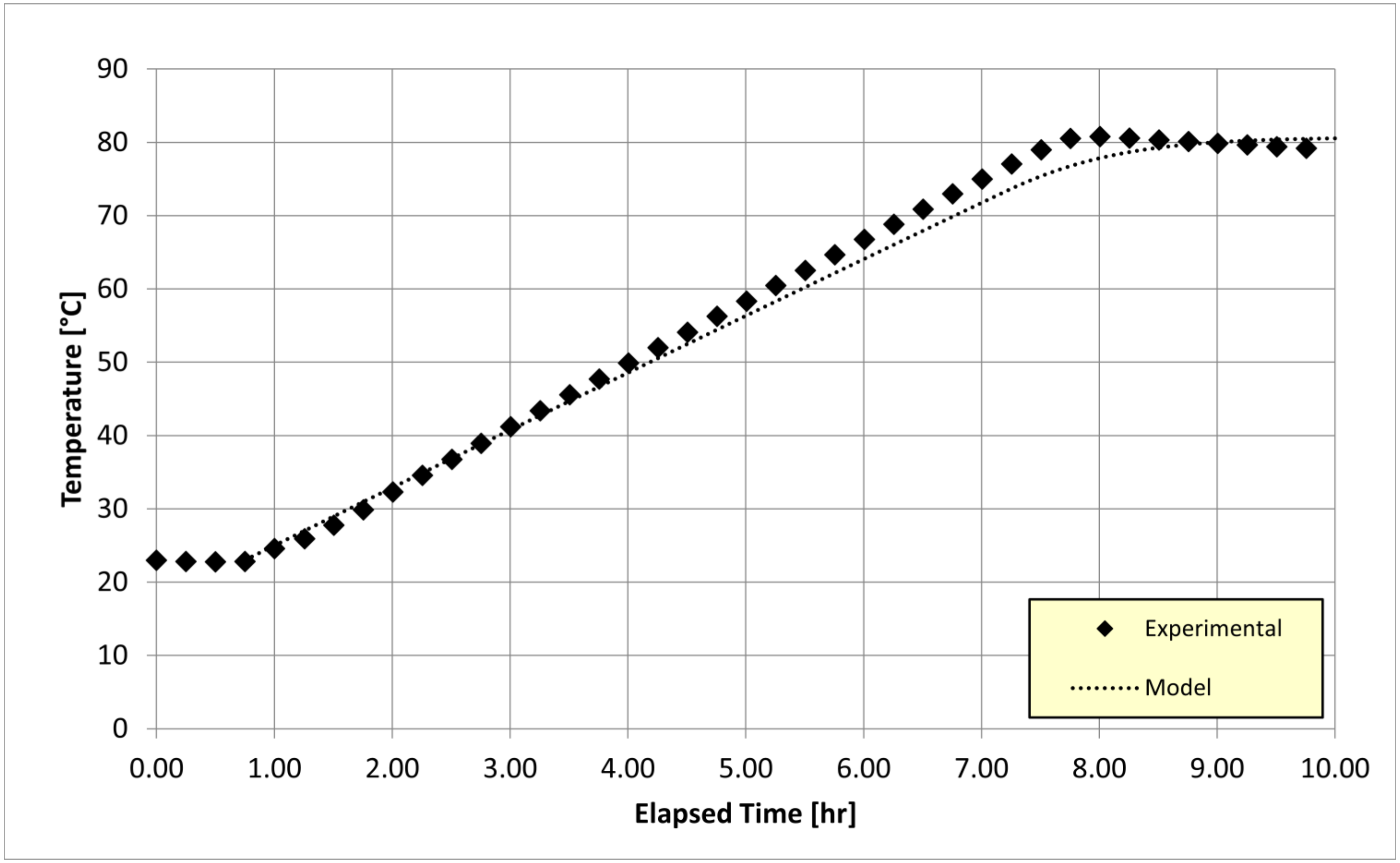

Figure 5.6 – Average temperature within float tank during direct charging via solar source

There is good agreement between the experimental and model results for this mode of operation. Average tank temperatures matched well for both cases, but there were some deviations, most notably towards the end of the heating process. A similar issue arose that existed for other modes of operation, where the experimental temperature variations take time to settle out. The modelled results have no delay, therefore the measured average tank temperature can appear different, but be much closer in reality. This inaccuracy was due to assumptions in the model and the limitations imposed by having a single temperature measurement per node in the experimental tanks.

5.1.5 Heat Pump

The performance of the heat pump was tested over a wide range of source and load temperatures at balanced and constant flow rates of 10.75 kg/min by Wagar [32]. A total of 27 data points were collected from a combination of source and load temperatures. The source temperature was held at values of approximately 10, 15, 20, 25, and 30 °C, while the load temperature was varied from 15 °C to 50 °C, by 5 °C increments. At no time was the load temperature permitted to be lower than the source temperature. The results from the heat pump performance mapping can be found in the work of Wagar [32].

The heat pump measurements were used to establish correlations for compressor work, source heat transfer rate, and load heat transfer rate. The ratio of heat transfer rate to the load and compressor electrical input defines the coefficient of performance (COP).

The following equations correlate accurately to the experimental results within a maximum error of 1.6% and a minimum coefficient of determination equal to 0.998.

$$\dot{Q}_{source} = a_s + b_s y + c_s x + d_s xy + f_s x^2 + g_s x^2 y + h_s x^3 + i_s x^3 y \tag{5.1}$$

$$\dot{Q}_{load} = a_l + b_l y + c_l x + d_l xy + f_l x^2 + g_l x^2 y + h_l x^3 + i_l x^3 y \tag{5.2}$$

$$\dot{W}_{comp} = a + bx + cy + dx^2 + fy^2 + gx^3 + hy^3 + ixy + jx^2 y + kxy^2 \tag{5.3}$$

$$COP = \dot{Q}_{load}/\dot{W}_{comp} \tag{5.4}$$

Where:

- $\dot{Q}_{source}$ is the heat transfer rate to the source in Watts;
- $\dot{Q}_{load}$ is the heat transfer rate to the load in Watts;
- $\dot{W}_{comp}$ is the rate of electricity input to the compressor in Watts;
- COP is the coefficient of performance of the heat pump;
- x is the source inlet temperature supplied to the heat pump in °C;
- y is the load inlet temperature supplied to the heat pump in °C;
- With the remaining parameters defined according to Table 5.3.

Table 5.3 – Heat pump performance coefficients [32]

$\dot{Q}_{source}$	$\dot{Q}_{load}$	$\dot{W}_{comp}$
$\boldsymbol{a_s = 2.2792 \times 10^2}$	$a_l = 6.6388 \times 10^3$	$a = 3.9791 \times 10^2$
$\boldsymbol{b_s = 2.7090 \times 10^1}$	$b_l = -1.0778 \times 10^2$	$b = -5.5335 \times 10^0$
$\boldsymbol{c_s = 5.6909 \times 10^2}$	$c_l = -6.6532 \times 10^2$	$c = 1.5933 \times 10^1$
$\boldsymbol{d_s = -1.0981 \times 10^1}$	$d_l = 1.7795 \times 10^1$	$d = 2.5389 \times 10^{-1}$
$\boldsymbol{f_s = -2.9092 \times 10^1}$	$f_l = 4.7623 \times 10^1$	$f = -2.0851 \times 10^{-1}$
$\boldsymbol{g_s = 6.6651 \times 10^{-1}}$	$g_l = -1.1335 \times 10^0$	$g = -3.0063 \times 10^{-3}$
$\boldsymbol{h_s = 5.8121 \times 10^{-1}}$	$h_l = -8.8722 \times 10^{-1}$	$h = 4.2935 \times 10^{-3}$
$\boldsymbol{i_s = -1.3125 \times 10^{-2}}$	$i_l = 2.1387 \times 10^{-2}$	$i = -1.3560 \times 10^{-1}$
-	-	$j = 9.2727 \times 10^{-5}$
-	-	$k = 3.1464 \times 10^{-3}$

The experimental measurements of heat pump performance for a $T_{source} = 10°C$ varied from a COP of 6.3 for $T_{load} = 15°C$ to a COP of 2.3 with $T_{load} = 50°C$. The results from the heat pump performance mapping were used by Wagar to create a custom model for the component within TRNSYS [32]. To facilitate a remedy of the HP sizing issue encountered in Wagar's experimental work, a HP scaling factor was included in the TRNSYS model to allow simulations with smaller HP capacities. The scaling factor multiplies by the energy transfer rates of each correlation to reduce the simulated size of the HP, according to the following relations:

$$\dot{\mathrm{Q}}_{\mathrm{source,scale}} = \mathrm{ScaleHP} \cdot \dot{\mathrm{Q}}_{\mathrm{source}} \tag{5.5}$$

$$\dot{\mathrm{Q}}_{\mathrm{load,scale}} = \mathrm{ScaleHP} \cdot \dot{\mathrm{Q}}_{\mathrm{load}} \tag{5.6}$$

$$\dot{\mathrm{W}}_{\mathrm{comp,scale}} = \mathrm{ScaleHP} \cdot \dot{\mathrm{W}}_{\mathrm{comp}} \tag{5.7}$$

The component model of the HP created by Wagar is used for the investigations completed in this work. This model was established through rigorous experimental testing of the HP by Wagar and the performance was validated through comparison of daily operation between Wagar's TRNSYS model and experimental tests [32].

5.2 Comparison to Test Apparatus

The test process for producing experimental results for a given day first starts with conditioning the system to the initial conditions that are desired. This may involve performing water draws to remove heat or operating equipment to add heat to thermal storage tanks. The initial conditions in the system are duplicated in TRNSYS by setting the initial timestep values accordingly. The whole experimental procedure is controlled within LabVIEW, where the weather file is selected and the detailed system parameters can be adjusted, for example pump flow rates, STC scaling, and controlled characteristics.

The process of comparing a full day of operation builds upon previous work, where SAHP system modes of operation were validated using the test rig in Chapter 3. TMY data is used both to simulate system performance and compare results experimentally for a representative day. Weather and water draw data from the TRNSYS simulation are supplied to the test rig, allowing for direct comparison between model and experiment. The water draw profile used for experimental testing is CSA-A from the Canadian Standards Association Packaged solar domestic hot water systems (liquid-to-liquid heat transfer) and is listed in Table 5.4 [15].

The weather data included with TRNSYS was used for the simulations and was duplicated for the experimental apparatus. The experimental apparatus cannot operate identically to the system modelled in TRNSYS due to limitations of the test apparatus identified in section 3.3 Limitations of Test Apparatus. The control strategy and heat pump sizing cannot be made identical for both cases. However, realistic operation of the simulation can be verified through comparison to the experimental results. This comparison is primarily a verification that the heat added by the solar collector and heat removed by the water draw provides realistic results.

Table 5.4 – Daily water draw profile, CSA-A [15]

Time of Day	Withdrawal at 10 L/min [L]
00hrs to 07hrs	0
07hrs to 08hrs	5
08hrs to 09hrs	25
09hrs to 10hrs	0
10hrs to 11hrs	45
11hrs to 12hrs	0
12hrs to 13hrs	5
13hrs to 14hrs	0
14hrs to 15hrs	0
15hrs to 16hrs	0
16hrs to 17hrs	0
17hrs to 18hrs	5
18hrs to 19hrs	15
19hrs to 20hrs	30
20hrs to 21hrs	20
21hrs to 24hrs	0
Total Daily	150

The comparison process comprises of running of both the experimental apparatus and TRNSYS model for a particular day of the TMY data and comparing the results. The weather file used is for Ottawa, Canada and is found in the Meteonorm folder of TRNSYS16. The simulation time step used is 0.08333 hours, which is 5 minutes. Since temperature is a good indication of how much energy transfer has taken place, this is the method of comparison used to verify that the model is functioning realistically. The volume and substance within the system are constant, meaning that changes in temperature are directly proportional to the amount of energy exchanged if density and specific heat variations with temperature are neglected. In addition, temperature is the primary concern when delivering domestic hot water. Note that the auxiliary heating element is disabled for this validation so that it does not dominate the simulation and experimental process.

Since the control strategies are not identical, the goal of this experimental validation is to compare the general function of the two systems over the course of test days. Two representative days, August 14 and October 29, were used from the TMY to verify proper function of the model developed. The former day is typical summer weather, whereas the latter is typical fall weather. The solar radiation intensity for each day is included in the results.

5.2.1 Day 1 – August 14

This particular day is one with clear skies and very high amounts of solar irradiation. The horizontal irradiation peaks around 3600 kJ/h per m^2, providing a high amount of solar energy into the system. The resulting DHW tank average temperatures for both the experiment and simulation are including in Figure 5.7. The solar irradiation throughout the course of the day is also included for reference.

The DHW tank temperature starts around room temperature at 20 °C and is heated to just over 45 °C by the end of the day. Although the DHW tank will typically not experience temperatures as low as 20°C, these initial conditions were selected to ensure that the system ran for as much time as possible. A comparison of the average tank temperatures for both the model and experiment are presented in Figure 5.7.

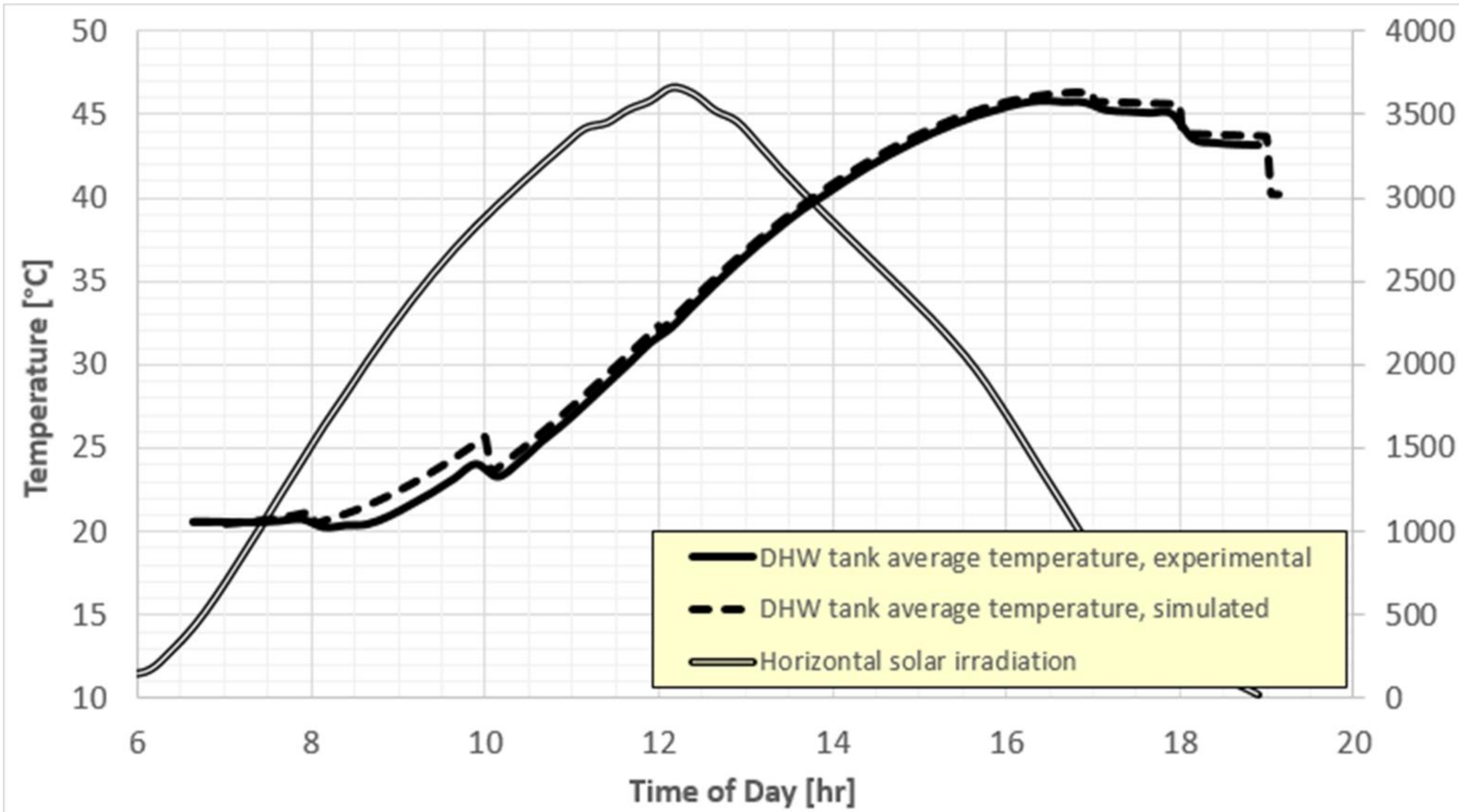

Figure 5.7 – Results comparison for TMY weather data for August 14 in Ottawa, Ontario, Canada

5.2.2 Day 2 – October 29

The second test day, October 29, represents a fall day where the solar irradiation is not as high as the summer day and experiences more variation due to cloud cover. This type of day is one that the SAHP synergy targets to improve. Since irradiation levels are quite lower than summer ideals for the location, the heat pump operates to extract extra heat from the solar thermal loop. Corresponding results are included in Figure 5.8 below.

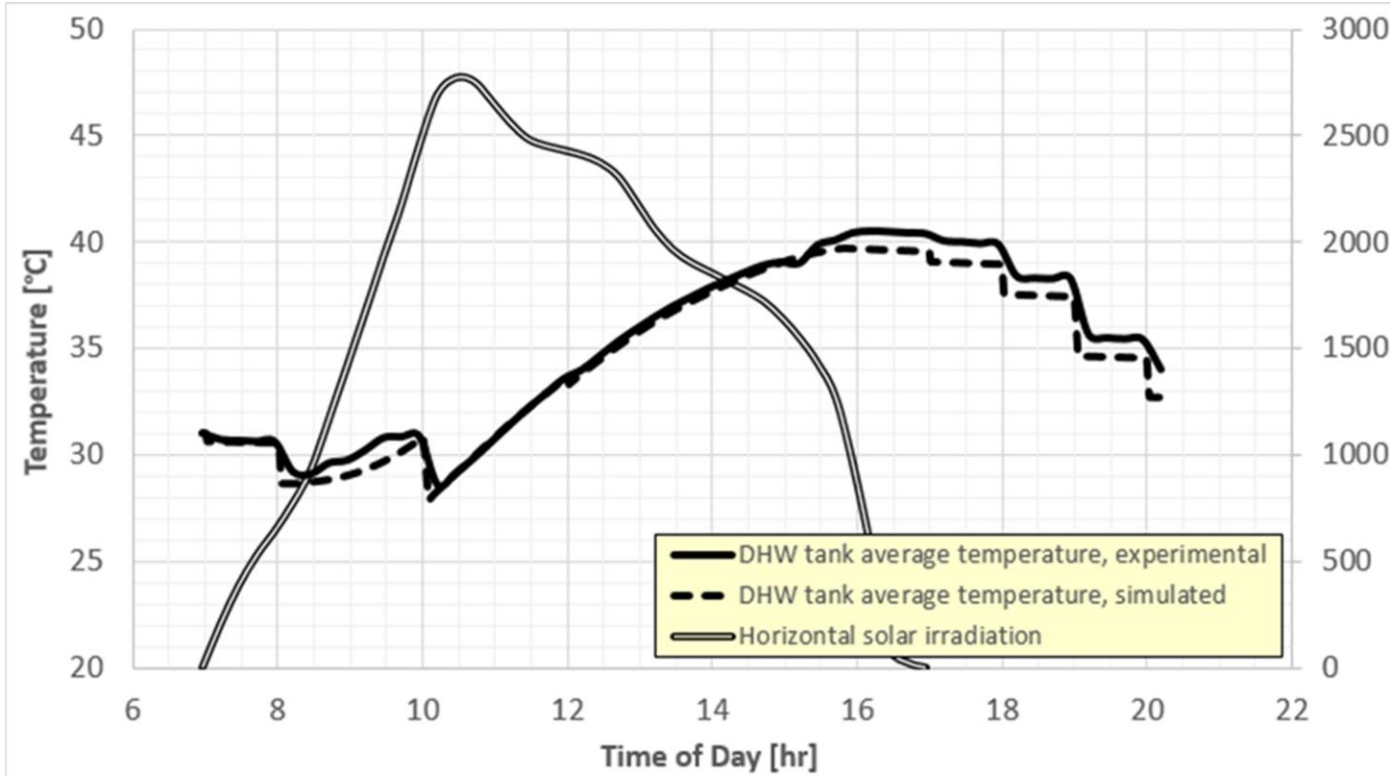

Figure 5.8 – Results comparison for TMY weather data for October 29 in Ottawa, Ontario, Canada

Energy collection during this day was less than that for the summer day, as would be expected because of the lower solar energy availability. For this case the DHW tank temperatures began around 30°C and were only able to reach about 40°C at the peak. In both cases, the sharp drops seen in tank temperature is due to water being drawn through the system. The rising temperatures are due to heat being added to the tank via the heat pump and solar thermal collector.

5.3 Analysis of Experimental Comparison

Overall, the two test days confirm realistic operation of the model. Throughout the course of either test day, the temperature difference between the model and the experimental results never exceeds 1°C. These differences are not cumulative, meaning that they diminish over time. The time required for heat to diffuse throughout the experimental thermal storage tank contributes to the variations seen in the validation process. In addition, the slight differences in HP control strategy also contribute to temperature variations.

A cause for differences between the experiment and model is the response rate of temperature changes. The model calculates the temperatures within the system, therefore it always has accurate "measurements" of conditions within the system. However, the experimental apparatus has a delay in reading temperatures since heat addition or removal must spread within the system via modes of heat transfer. For heat input or a water draw, the experimental apparatus is slow to respond. The abrupt changes seen in the simulation results are smoothed out during experimental measurements.

The addition of pipes to the model was necessary to provide numerical stability. Note that TRNSYS documentation recommends this technique, since many of the connected components do not include any fluid storage in their models (e.g. pumps, heat exchangers, diverters, tees).

The tuning of model components described in section 5.1 Components and the equipment characterization of Wagar [32] allowed for the specification of realistic parameters and inputs for TRNSYS components. In this previous work, it was noted that the biggest discrepancies between model and experiment are due the difficulties in modelling thermal storage tank stratification. The process by which stratification occurs is fairly complex, and modelling approaches simplify it considerably. Despite these limitations, the realistic performance of the TRNSYS model developed in this work makes it worthy to be used for further studies involving the modelling of this SAHP system.

In reality, there are many other factors that contribute to uncertainty in modelling the performance of this system, such as occupant energy use patterns and actual weather encountered. This model verification serves to illustrate that the system components and control strategy are operating as intended. Ultimately, the annual energy use of such a system is difficult to predict, but the model developed is excellent for further investigation of the relative performance in comparison to alternatives.

5.4 Verification of Control Strategy

After tuning of the model components, it was necessary to debug and verify the control strategy developed for the proposed system. Limitations of the test apparatus, as detailed in section 3.3 Limitations of Test Apparatus, prevent the test apparatus from performing realistically as an installed system. Primarily, the large size of the heat pump and the relatively small size of the heater used to simulate solar input cause limitations on the control strategy used. In experiment, it was necessary to cycle the heat pump on and off during energy collection via Solar-HX-DHW to prevent freeze up of the source-side fluid. This is not desirable in real operation because it puts extra wear on the compressor, reduces the amount of time the HP operates at pseudo steady state, and limits the solar collection improvements over a SDHW system by increasing the average fluid temperature in the STCs when the HP is not operating.

The control strategy presented in section 4.3 Control Strategy allows the system to be modelled as either a SDHW, single tank SAHP system, or dual tank SAHP system. This is achieved by allowing or disallowing modes via inputs to the controller defined in

Table 4.5 – Inputs of the custom controller component. Throughout the next sections, the operation of the three solar systems of concern is compared over simulated days to establish confidence that the controller and model are functioning as intended. Weather data for Ottawa, ON is used with a STC array size of 7.5 m^2.

5.4.1 Single Day Comparison of All Systems

The three solar systems, SDHW, single tank SAHP, and dual tank SAHP were simulated for a span of days containing October 1 to allow comparison of their operation. In order to allow an effective comparison, an individual day of operation is presented below in Figure 5.9. The simulation was begun several days before the day of interest to minimize the impact of the initial simulation conditions on the outcome.

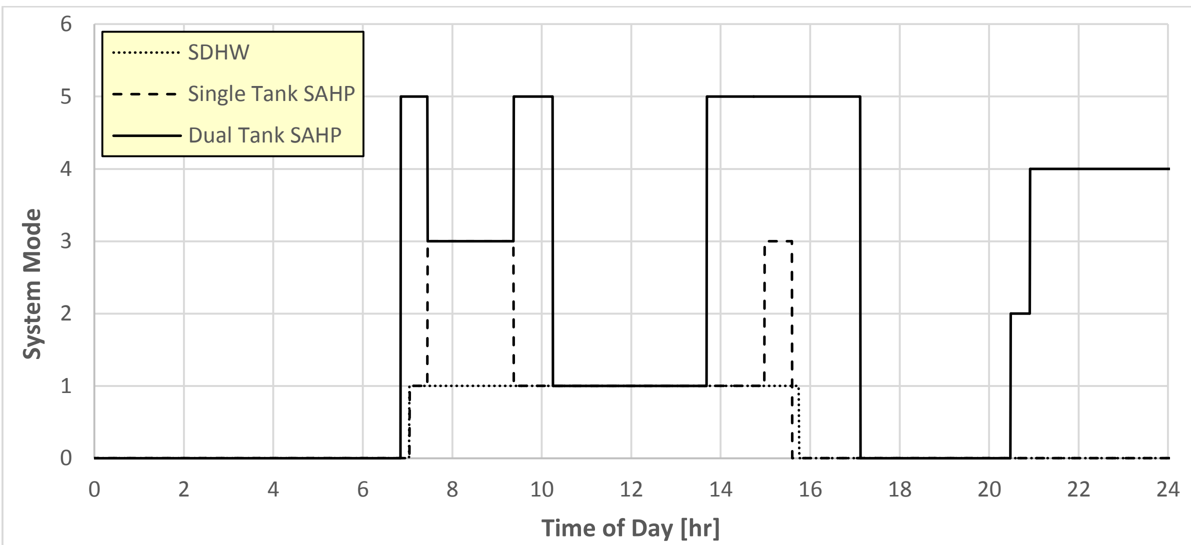

Figure 5.9 – Operation mode comparison of 3 solar systems with custom controller for October 5 in Ottawa, Ontario, Canada

These simulation results enable the verification that the custom control strategy is operating as intended. With all modes disabled other than Solar-HX-DHW, the system operates as a SDHW system. This system operates in mode 1 – Solar-HX-DHW from the hour of 7 to 16, after which solar energy collection cannot continue due to a fully charged DHW tank. For the single tank SAHP system, mode 3 is enabled during periods of moderate insolation in the morning and afternoon hours. This occurs in accordance with the radiation thresholds outlined in Table 6.6.

The dual tank SAHP adds three more modes of operation. The DHW tank can be charged via the HX (mode 2), which occurs for about 30 minutes in the evening hours. The DHW tank can also be charged via the HP (mode 4), which occurs later in the evening hours to recover the DHW tank. The final additional mode allows the Float tank to be charged via the STCs (mode 5), which occurs during a portion of the morning and evening hours when the DHW demand is met.

Substantial investigation and debugging of the control strategy programmed in Fortran resulted in a custom controller that operates the SDHW, single tank SAHP, and dual tank SAHP systems as intended.

Chapter 6 Design, Performance, and Justification

The establishment of tuned model components opens up many opportunities to explore the design and performance of SAHP systems. Confidence that all components of the model perform well permits the use of the model for further system development and optimization. Using the core control strategy developed, SAHP configurations can be compared to one another, to traditional SDHW systems, and to yet more conventional systems, such as an electric resistance DHW system and the theoretical minimum heat input required.

This work utilizes a detailed model of a SAHP to determine the feasibility, sizing, and configuration for such a system using weather data for Ottawa, Ontario, Canada. Performance of the candidate systems is evaluated on the basis of solar fraction, which indicates the extent of utilization of solar energy, and energy savings, which gives a more appropriate measure of the cost savings expected.

In addition to benchmarking the performance of SAHP systems, this chapter explores the impact of varying system parameters, such as STC area, DHW and float tank sizes, and STC array tilt angle. The frequency of system stagnation and the ability of the system to meet the load are evaluated. The main outcome of the chapter is to establish recommendations for sizing components and determining key parameters. These design guidelines are concluded with an evaluation of the dual tank SAHP system in general.

6.1 Model Modifications

An important consideration when using models to perform investigations is the computational resources and time required to solve the governing equations. Constraints with TRNSYS and physical time prompted a need to modify the model to expedite the numerical solution process.

In order to ensure that water draw mass flows occur correctly, it was necessary to ensure that the time step of the simulation was equal to the minimum time interval of the water draws. If this requirement is not met, the simulated water draw mass flow over the course of a day or year will not match the desired draw. For the CSA draw profiles, this means a time step of 0.5 minutes must be used since the flow rate during all water draws is 10 L/min and the minimum draw volume was 5 L/min. However, using such a small time step for a simulation duration of one year presented some difficulties.

The simulation time required to calculate results for 8760 hours, the equivalent of one year, is on the order of 20 minutes. Therefore, to perform only three simulations of one year would require an hour of real time. Although this may not appear excessive, the number of simulations to complete this work is on the order of hundreds. Many model variations were explored, and difficulties arise during simulations that may require runs to be repeated. For example, the run may crash partway through or it may be determined after the run that some settings were not correct.

In addition to the large time requirement for simulations when using a 0.5 minute time step, the performance of the TRNSYS software also suffers. The online plotter that displays results during the simulation has a difficulty drawing such a high quantity of points, and if the plot is moved or refocused, all the points must be redrawn. This is a prohibitive process since it causes the simulation progress to be paused and is time consuming. It can easily take a minute or two to redraw the graphs and if they are moved again the process must be repeated. The benefit of using a modified time step is illustrated below in Table 6.1.

Table 6.1 – Simulation times for true-to-life time step and modified

Draw Profile	Simulation Time Step [minutes]	Minimum Water Draw Time Interval [minutes]	Simulation Computational Time [mm:ss]
True-to-life	0.5	0.5	19:22
Modified	5	5	1:56

As expected, increasing the time step by a factor of 10 reduced the simulation time by approximately the same factor. With a time step of 5 minutes it is feasible to complete about 30 simulations per hour, significantly boosting productivity.

Increasing the time step for simulations has clear benefits, but it also must provide accurate results. Simulations with all factors identical other than the simulation time step and the water draw profiles were performed to evaluate the impact of draw modifications on accuracy. Note that the total volume of water drawn by the load remains equal at

109,500 L, which is 300 L/day for a full year of 365 days. A comparison of these results is summarized below in Table 6.2.

Table 6.2 – Comparison of results from true-to-life and modified draws/timesteps

Draw profile	Solar [GJ]	HP electricity [GJ]	Auxiliary [GJ]	Tank losses [GJ]	Solar fraction	Energy savings [GJ]	Float losses [GJ]
True-to-life	16.38	2.259	4.027	2.090 (8.4%)	72.3%	14.31 (69.5%)	0.1395
Modified	16.61	2.277	4.134	2.070 (8.6%)	72.2%	14.19 (68.9%)	0.1598
Difference	+0.23 +1.4 %	+0.018 +0.8%	+0.107 +2.7%	-0.02 -0.95%	**-0.1%**	**-0.12 -0.8%**	+0.0203 +14.6%

Inevitably, there are differences between the results for the true-to-life water draws and the modified version. The amount of solar energy collection, heat pump electricity, and the losses from the DHW tank remained quite accurate, differing by about a percent. The auxiliary energy consumption varied more, at around 3%. One of the reasons for these differences is that each timestep is taken as pseudo steady state, where the inputs and outputs of all components are constant during a given timestep. This means that the resolution for turning equipment on or off is reduced when increasing the timestep. The heat pump or auxiliary heater can remain on for more time with the longer timesteps, consuming more energy.

A very surprising anomaly with the simulations is the difference between the float tank losses for the two cases. Although the energy difference is not very high, at around 20.3 MJ, the relative change was about 15%. The float tank losses are very sensitive to the simulation inputs and parameters because it is not a very large cumulative value for an annual simulation.

Overall, the values of primary concern are the metrics used to evaluate system configurations and alternatives, the solar fraction and energy savings. For these values the results from the two simulations differed by less than 1%. The modified draw simulation underestimated solar fraction by 0.1% and underestimated energy savings by 0.8%. These low variations indicate that using the modified draws does not impact accuracy significantly and does not affect the conclusions drawn from the key indices of merit. The advantages of computational and time savings outweigh the small inaccuracies introduced.

6.2 Comparison to Alternatives

The performance of the SAHP model is compared to a traditional domestic hot water (DHW) tank system, which is simply a water tank with an electric heat source. A further comparison is made to a traditional solar domestic hot water (SDHW) system, which includes a thermal storage tank, solar thermal collectors, a pump, and a heat exchanger.

The base parameters and inputs used for the TRNSYS simulations are detailed in Appendix B – Model Component Parameters/Inputs.

The indices of merit selected for the analysis are solar fraction, SF, and the energy saving percentage compared to the base case of a tradition DHW system. The solar fraction is defined as:

$$SF = \frac{\text{energy supplied by solar source}}{\text{total energy supplied}} = \frac{Q_{solar}}{Q_{solar} + Q_{aux} + Q_{HP}} \tag{6.1}$$

Where:

- Q_{solar} is the total useful energy gain from the solar thermal collectors;
- Q_{aux} is the total energy input from the auxiliary heater; and
- Q_{HP} is the total electricity input to the compressor.

The energy savings, $E_{savings}$, are defined as:

$$E_{savings} = Q_{DHW} - Q_{aux} - Q_{HP} \tag{6.2}$$

And the energy saving percentage is defined as:

$$\%E_{savings} = \frac{Q_{DHW} - Q_{aux} - Q_{HP}}{Q_{DHW}} \times 100\% \tag{6.3}$$

Without time-of-use (TOU) energy charges, the energy savings represent the cost savings percentage, since electricity is purchased at the same price at any time of day.

6.2.1 Baseline Calculations

The starting point for analyzing the performance of a hot water heating system is to consider a baseline case for comparison. This can be done using the equation for sensible heating:

$$Q = mc(T_h - T_c) \tag{6.4}$$

Where m is the mass of fluid, c is the specific heat of the fluid, and $T_h - T_c$ is the temperature rise of the fluid. Assuming the mains water source is at a constant temperature of $T_c = 12°C$ and hot water is delivered to loads at a constant $T_h = 55°C$, the minimum amount of required energy input can be calculated.

Assuming a constant specific heat of $c = 4.19\ kJ/kg \cdot K$, the energy input per unit mass of water is:

$$\frac{Q}{m} = 179.74 \frac{kJ}{kg} \tag{6.5}$$

The water draw profile examined for the performance study is CSA-C, which is a total of $300\ L/day$ [15]. The details of this water draw are provided in Table 6.3. Assuming the density of water to be constant at $1\ kg/L$, the total minimum annual energy required to meet these loads is:

$$Q_{CSA-C} = 19.72\ GJ \tag{6.6}$$

Table 6.3 – Daily water draw profile, CSA-C [15]

Time of Day	Withdrawal at 10 L/min [L]
00hrs to 07hrs	0
07hrs to 08hrs	10
08hrs to 09hrs	25
09hrs to 10hrs	25
10hrs to 11hrs	45
11hrs to 12hrs	25
12hrs to 13hrs	10
13hrs to 14hrs	5
14hrs to 15hrs	0
15hrs to 16hrs	0
16hrs to 17hrs	15
17hrs to 18hrs	25
18hrs to 19hrs	45
19hrs to 20hrs	25
20hrs to 21hrs	30
21hrs to 22hrs	10
22hrs to 23hrs	5
23 to 24hrs	0

6.2.2 Electric Water Heating Tank

In reality, it takes more energy than the calculated baseline amount to provide a prescribed amount of hot water to a household. All water heaters have losses, whether they are supplied by electricity, natural gas, or other heat sources and whether they are tankless or include a storage tank. For the case of a water tank with an electric element, the losses to the environment cause increased energy use. A TRNSYS simulation was used to estimate the amount of energy required for this scenario. The tank model used is Type 4b and was modelled with the parameters listed in Table B.5.

As predicted, simulation of a DHW tank consumes more than the calculated minimum due to storage losses, as shown in Table 6.4. However, since the lower nodes of the tank are often below the ambient environment temperature of 20°C, small amounts of heat gain slightly offset the losses. These gains are minimal due to a very small temperature gradient when compared to the upper, hot nodes.

Table 6.4 - Results of electric DHW tank annual simulation with CSA-C draw profile

Energy consumed	**20.60 GJ**
Tank losses	0.857 GJ (4.2%)

These results serve as a baseline to which DHW systems including solar collectors and heat pumps can be compared.

6.2.3 Solar Domestic Hot Water System

The first alternate system to be considered is a simple SDHW arrangement including glazed flat plate collectors connected to the storage tank via a heat exchanger and a single storage tank. The CSA-C draw profile represents a household occupied by approximately 5 people, so 2-3 collectors, each with an area of 2.494 m^2, gives 1-2 m^2 of collector area per occupant. The collectors modelled correspond to the Viessmann Vitosol 200-F, Type SV2; component parameters used in simulation are specified in Table B.2. Detailed results from simulations using 1, 2, 3, and 4 STCs are presented below in Table 6.5.

The first collector provides the greatest energy savings, with each successive collector providing less benefit than the previous. Tank losses are increased due to temperatures above 60°C occurring inside the tank during parts of the year and due to less stratification being maintained in the storage tank. Circulation of water through the solar collector reduces the amount of stratification achieved. As the number of collectors increases, a diminishing performance increase is encountered since the storage size and intermittency of solar energy stays the same.

Table 6.5 - Traditional SDHW heating system performance for several collector quantities. Modelled in Ottawa, ON using CSA-C draw profile.

Collector Quantity	Solar [GJ]	Auxilliary [GJ]	Tank losses [GJ]	Solar fraction	Energy savings [GJ]
1 collector (2.494 m^2)	7.004	13.85	1.151 (5.2%)	31.8%	6.75 (32.8%)
2 collectors (4.988 m^2)	11.2	9.924	1.389 (6.2%)	49.7%	10.68 (51.8%)
3 collectors (7.482 m^2)	12.89	8.306	1.467 (6.5%)	56.9%	12.29 (59.7%)
4 collectors (9.976 m^2)	13.47	7.735	1.471 (6.5%)	59.4%	12.87 (62.5%)

6.2.4 Single Tank SAHP System

Modelling the single tank system brings forth unique control considerations. A SDHW system is typically operated using a differential thermostat: when the fluid temperature at the tank's outlet to the collector is less than the fluid temperature at the collector outlet, the circulating pump(s) should run. Temperature deadbands are included to avoid oscillating control signals. Adding a heat pump to the system requires the incident solar flux to also be included in the control strategy.

Considering incident solar flux for system control is important because, when considering pseudo-steady state operation, the amount of energy input by the solar collector must match the energy extracted by the heat pump. Therefore, the operation of the heat pump depends on the incident solar flux and the quantity of solar panels available. If the energy addition rate by the collectors is too low, the heat pump will be starved of energy; if the energy addition rate by the collectors is too high, the heat pump could overheat and is likely not benefitting the system. Table 6.6 summarizes the threshold ranges used for simulations to match realistic operating scenarios and also to maintain numerical stability.

Table 6.6 - Thresholds of incident radiation on tilted surface for heat pump control

Collector Quantity	Lower Threshold		Upper Threshold	
	[kJ/m²hr]	[W/m²]	[kJ/m²hr]	[W/m²]
1 collector	2000	555	4800	1333
2 collectors	1500	417	3600	1000
3 collectors	1000	278	2400	667
4 collectors	1000	278	1800	500

*(*this insolation level does not ever occur terrestrially at ground level, so the HP runs as long as the insolation is above the lower threshold)*

Modelling results for the single tank SAHP system are summarized below in Table 6.7. The SAHP system with a single tank shows an insignificant performance in comparison to the traditional SDHW system. Total solar energy collection is increased due to the reduced fluid temperature entering the collectors. A lower inlet temperature increases collector efficiency by reducing thermal losses. However, collecting this additional solar energy requires electrical input to the HP compressor.

Table 6.7 - Single tank SAHP system performance for several collector quantities. Modelled in Ottawa, ON using CSA-C draw profile.

Collector Quantity	Solar [GJ]	HP electricity [GJ]	Auxiliary [GJ]	Tank losses [GJ]	Solar fraction	Energy savings [GJ]
1 collector ($2.494\ m^2$)	8.025	2.523	11.28	1.364 (5.9%)	36.8%	6.797 (33.0%)
2 collectors ($4.988\ m^2$)	12.42	2.312	7.542	1.642 (6.6%)	55.8%	10.75 (52.2%)
3 collectors ($7.482\ m^2$)	14.04	2.13	6.209	1.672 (7.0%)	62.7%	12.26 (59.5%)
4 collectors ($9.976\ m^2$)	14.28	1.803	5.829	1.600 (6.8%)	65.2%	12.97 (63.0%)

The major difficulty with the single tank SAHP system is the challenge in control strategy. Without accurate weather prediction, it is difficult to determine whether or not operating the HP will be beneficial for a given day of the year. There are many days of the year, particularly during the summer months, where the STCs would collect adequate energy without any assistance from the HP. Therefore, operating the HP on these days is a waste of energy.

With a predictive control strategy, operating the Solar-HP-DHW mode could pose overall energy benefits. However, the complexity of operating such a mode outweighs the small energy benefits. Due to the poor performance of this operating mode, it will not be used when investigating the dual tank SAHP system. Major limitations of the single tank arrangement are that the system can still only deliver solar energy to the DHW tank when the sun is present and it is difficult to predict when extra solar energy will be required.

6.2.5 Dual Tank SAHP System

Hot water demand does not always coincide with the presence of sufficient insolation, so it is important to have thermal storage. A second storage tank creates an energy source that can be used when the sun is not present. In addition, it provides the ability to have the storage material, in this case water, at different temperatures in each tank. The addition of a second storage tank, referred to as a ‘float tank’, was modelled and is compared to the performance of the above systems. The results from the model are summarized below in Table 6.8. Note that the mode of Solar-HP-DHW is disabled for the dual tank SAHP system.

Table 6.8 - Dual tank SAHP system performance for several collector quantities. Modelled in Ottawa, ON using CSA-C draw profile.

Collector Quantity	Solar [GJ]	HP electricity [GJ]	Auxiliary [GJ]	Tank losses [GJ]	Solar fraction	Energy savings [GJ]	Float losses [GJ]
1 collector (2.494 m^2)	7.730	1.175	11.34	1.421 (6.6%)	38.2%	8.085 (39.1%)	-1.001
2 collectors (4.988 m^2)	12.86	2.118	6.459	1.827 (7.9%)	60.0%	12.02 (58.4%)	-0.8224
3 collectors (7.482 m^2)	16.61	2.277	4.134	2.070 (8.2%)	72.2%	14.19 (68.9%)	0.1598
4 collectors (9.976 m^2)	18.91	2.03	3.063	2.216 (8.5%)	78.8%	15.51 (75.3%)	1.021

For modest collector quantities, adding a second storage tank is similar to adding an additional solar panel. This could be advantageous because it adds more possibilities of system operation modes and functions. For example, time-of-use electricity pricing could be used as an advantage and the system could be applied to additional thermal energy loads.

At very high collector areas, losses from the DHW tank increase due to the elevated temperatures for an increased amount of time. It is important to ensure tanks are adequately insulated. In reality, commercial tanks are available at set insulation levels and additional insulation may be prohibited by lack of space. During the cooling season of the summer it may be beneficial to use the float tank for a purpose other than storing heat. For example, the float tank could be used to store chilled water for air conditioning.

6.3 Comparison of Alternatives

The solar fractions and energy savings listed in earlier tables are graphed below in Figure 6.1 and Figure 6.2. The diminishing returns of additional STC panels is apparent. Each additional panel has a reduced potential to collect energy. As collector area is added, the average fluid temperature in the collector goes up, reducing efficiency due to increased losses.

It is important to consider the most appropriate metrics when analyzing solar heating systems. Solar fraction (SF) is typically used to compare performance, but in reality the energy or cost savings is often a more appropriate indication of merit. SF is not necessarily the best indicator because it measures how much solar energy is obtained in comparison to other sources, rather than what fraction of the load is offset by solar energy. The performance of the single tank SAHP system demonstrates this distinction well, as solar fraction was increased for all collector quantities, but the energy savings were not improved in comparison to the SDHW system.

The dual tank system boosts solar fraction more than energy savings due to the increased tank losses from additional heat transfer surface area and higher average temperatures within the tanks. In this case, some of the extra solar energy collected is not useful to the load and ends up as waste. For these reasons, energy savings is a more appropriate measure of system performance.

The single tank SAHP system increased solar energy collection as expected by the interaction between the STCs and the HP, however the additional energy input to the HP outweighed this increase. It was found that the single tank SAHP system only increased energy savings marginally. Due to the increased complexity of operating the system with radiation thresholds and this lack of performance increase, a single tank SAHP system is not justifiable. For this reason, the Solar-HP-DHW mode was not used with the dual tank SAHP system.

In terms of energy savings, the dual tank system improves performance by 6-13% depending on the quantity of solar collectors. The energy savings increase as the quantity of solar collectors increases due to an improved match with the float tank storage capacity. The dual tank SAHP system demonstrates that increased STC area can be used if a larger storage medium is provided. It is important that this additional storage volume is in a separate tank to avoid reaching the temperature limit of the DHW tank.

Switching focus to the impact of adding STC area, the biggest gain comes from adding a second collector, regardless of the system. This saves an additional 19% of energy compared to the base case. Adding a third panel has about half the benefit, at 8-10% additional annual energy saved. Adding a fourth panel has an even smaller benefit, only saving an additional 3-7% of annual energy consumption.

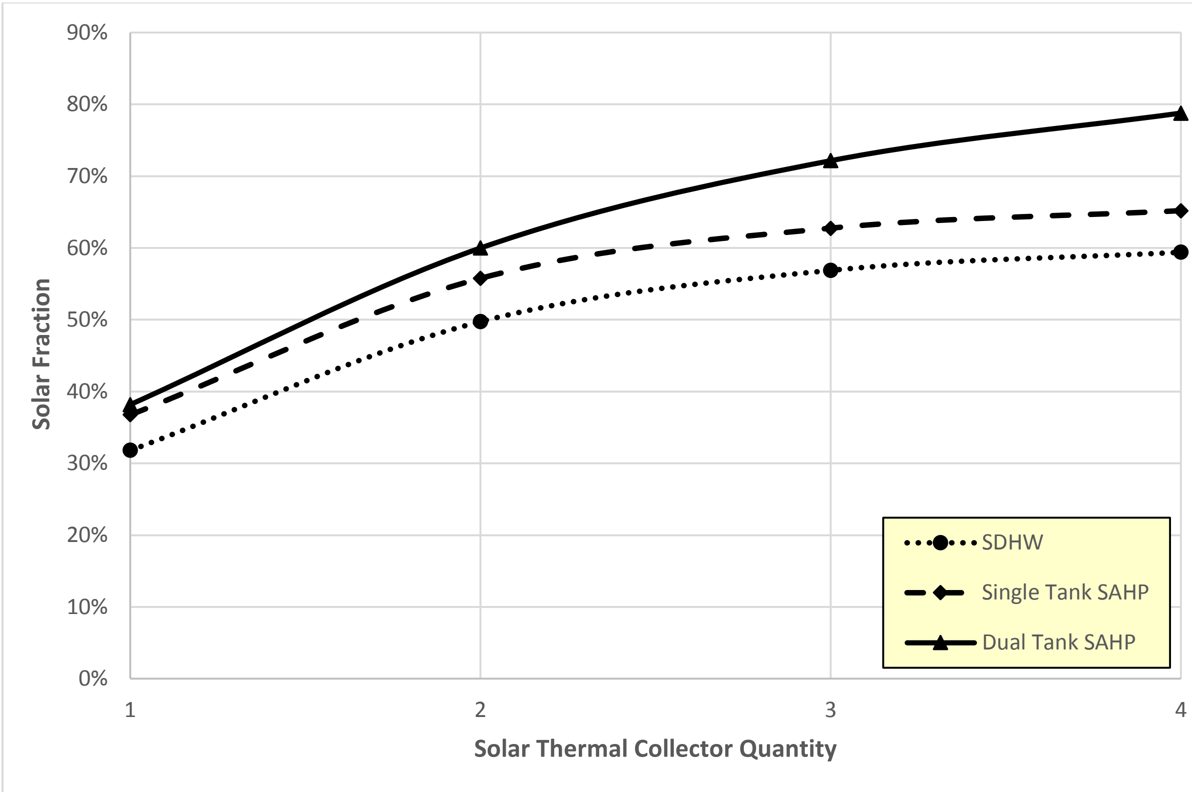

Figure 6.1 - Comparison of solar fraction for systems investigated with various solar thermal collector quantities

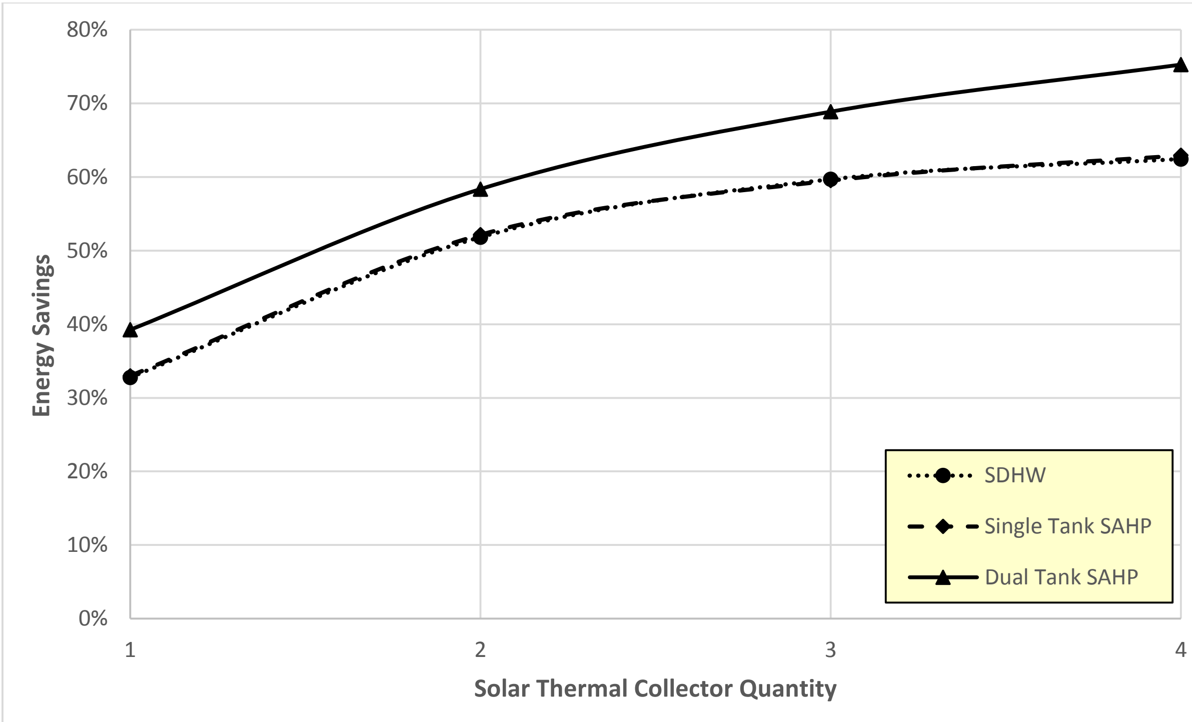

Figure 6.2 – Comparison of energy savings for systems investigated with various solar thermal collector quantities

6.4 System Stagnation

Stagnation is a condition of concern for solar thermal systems. It occurs when storage temperatures reach a high cut-off and no more heat can be added. This means that heat cannot be removed from the STCs because there is nowhere to transfer it to. The dangers of stagnation are primarily: excessively high pressures within the system, steam blowout from pressure safety valves, damage to equipment, and degradation of glycol antifreeze.

Stagnation is an issue in SDHW systems, so it is important to evaluate the impact that a SAHP system will have on this condition. The weather conditions, quantity of solar panels, panel orientation, and storage size all contribute to the frequency of stagnation. Table 6.9 summarizes the results of simulations investigating the impact of the system type and quantity of solar panels on stagnation. The number of hours of the year above a prescribed temperature are presented for comparison and analysis.

A particularly bad scenario for thermal stagnation is when occupants are not present for extended periods of time, for example during a holiday. During these periods, any solar thermal system would be susceptible to stagnation once the storage medium is charged. The investigation performed studies how each system responds over a simulated year with regular daily water draws.

Table 6.9 – Frequency of stagnation for systems investigated

System Type and Configuration		Annual Hours Collector Temperature Above			
		90 °C	95 °C	100 °C	105 °C
SDHW	2 STCs	44	30	25	21
	3 STCs	402	307	234	180
Single Tank SAHP	2 STCs	212	174	147	124
	3 STCs	517	389	298	240
Dual Tank SAHP	2 STCs	9	1	0	0
	3 STCs	177	93	29	4

Overall, on the basis of frequency of stagnation experienced by the system, an STC panel quantity of 2 would be most appropriate for the modelled conditions. Adding a third panel to the system results in considerably more stagnation time for the system, as shown in Table 6.9.

In all cases, the single tank SAHP system experiences stagnation more often than a SDHW system with the same amount of solar collector area. The addition of a heat pump allows the SAHP system to collect more energy, resulting in the DHW tank temperatures being higher on average. The higher DHW tank temperatures mean that more occasions arise where the STCs cannot be used due to the high temperature cut-off, and therefore the panels experience more stagnation.

The addition of a float tank to the SAHP system mitigates the issue of stagnation effectively when the building is occupied. In comparison to both the SDHW and single tank SAHP systems, the dual tank configuration reduces the time above 100 °C by about an order of magnitude when using 3 STC panels. Temperatures above 100 °C are completely avoided when 2 STC panels are used. STC outlet temperature, sorted by hour, is presented in Figure 6.3 for the SDHW and dual tank SAHP systems. This chart shows the reduction of stagnation occurrence by the dual tank SAHP more clearly.

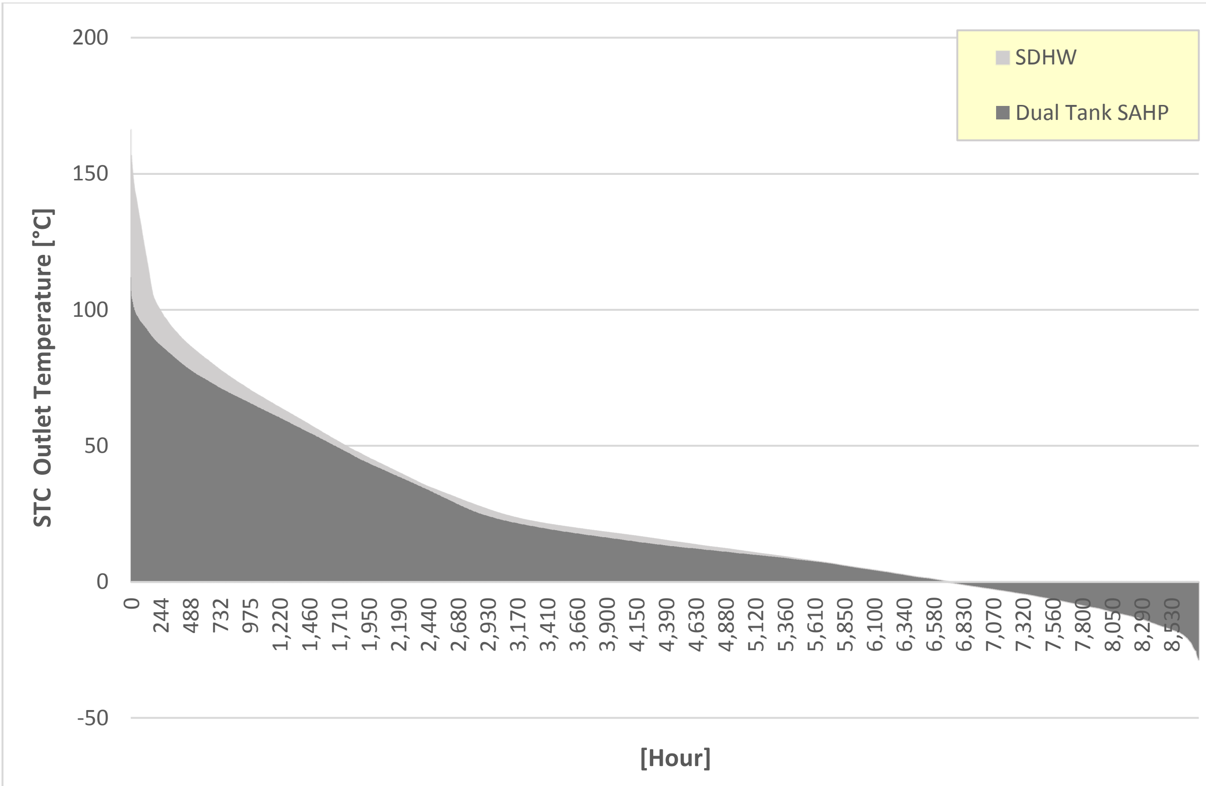

Figure 6.3 – STC outlet temperature sorted by hour of year for SDHW and Dual Tank SAHP systems with 3 collectors, 45° tilt in Ottawa, ON

From the above chart, it is clear that the dual tank-SAHP system can increase the lifespan of the equipment and working fluid. This will help reduce lifecycle costs in comparison to a SDHW system. Figure 6.4 provides a zoomed in view of the high temperature data and more clearly shows the stagnation reduction by the dual tank SAHP. From this figure, it is seen that the dual tank SAHP system reduces the number of hours above 100°C from approximately 230 to about 30, a decrease of 200 hours. In addition, the peak temperature was reduced from 166°C for the SDHW to 112°C for the dual tank SAHP system.

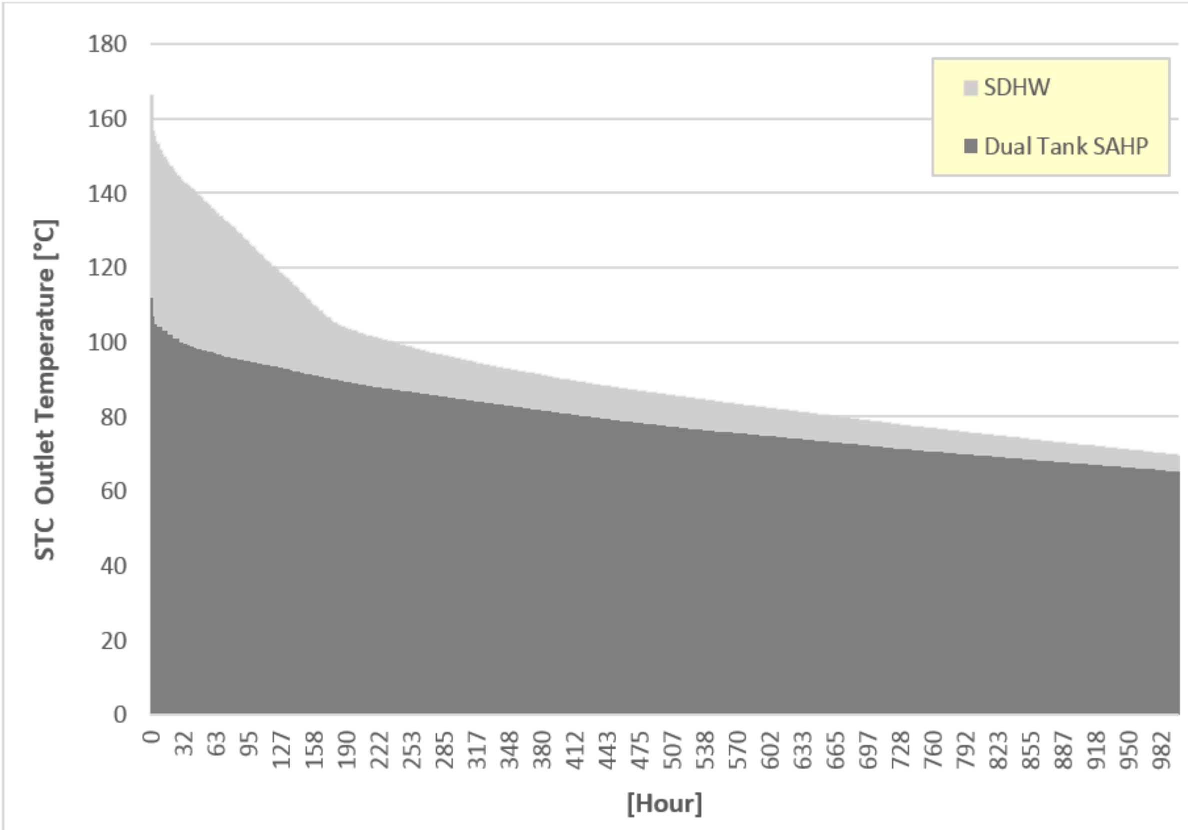

Figure 6.4 – STC outlet temperature for hottest 1000 hours of year for SDHW and Dual Tank SAHP systems with 3 collectors, 45° tilt in Ottawa, ON

The second tank provides an additional place for heat to be stored, reducing issues inherent in traditional SDHW systems and handling the extra thermal energy capabilities provided by the heat pump. If not using a float tank, it may be necessary to modify the design and configuration of a SAHP system to reduce stagnation to an acceptable level. This could be achieved by using less STC area, installing a larger DHW tank to store heat at lower temperature, or modifying the tilt of the collector array to reduce summer solar energy collection.

6.5 Ability to Meet Load

An important measure of performance for any hot water system is its ability to meet the load placed on it by the occupants. Regardless of the system being a conventional gas or electric DHW tank, a tankless heating system, or any variety of solar thermal DHW systems, the load must be met or the occupants will be dissatisfied and inconvenienced.

With solar thermal systems, an auxiliary heating element is typically required to provide heat input when an insufficient amount of solar energy is available. This type of system is used to ensure that the water delivered to the load reaches the expected temperature. For the case of these tests and simulations, the SDHW standards published by CSA are used, requiring hot water to be delivered to the load at 55 ± 2°C, after passing through a tempering valve [15]. The water draw profile used from the standard is CSA-C, which is a total of 300 L/day. Table 6.10 summarizes the ability of each system to meet the load.

Table 6.10 – Ability to meet load with 3 collectors, 45° tilt in Ottawa, ON

System Type and Configuration	Annual Frequency Temperature to Load Above		
	53 °C	55 °C	57 °C
SDHW	99.9%	16.9%	0%
Single Tank SAHP	99.9%	17.6%	0%
Dual Tank SAHP	99.9%	38.7%	0%

From Table 6.10, it is clear that all systems are meeting the load at all times of the year, keeping the temperature within the bounds of 53°C and 57°C. This indicates that the system and custom controller are operating as intended. Note that for all systems, the temperature is below 53°C for a brief period in time during initial system startup at the beginning of the simulation.

6.6 DHW Tank Temperature

The average temperature of the DHW tank is important because it determines how often the load is met and how much stored energy is available. A chart of average DHW tank temperature by hour over the course of a year for the SDHW and dual tank SAHP systems is shown in Figure 6.5. Through this data representation, the main benefits of the dual tank SAHP over a SDHW system can be visualized.

The chart provides confirmation that the dual tank SAHP system meets the load for the full duration of the year, since the DHW tank temperature never falls below 53°C. This confirms that the control strategy operates correctly from the perspective of meeting the CSA test standard for water delivery temperature [15].

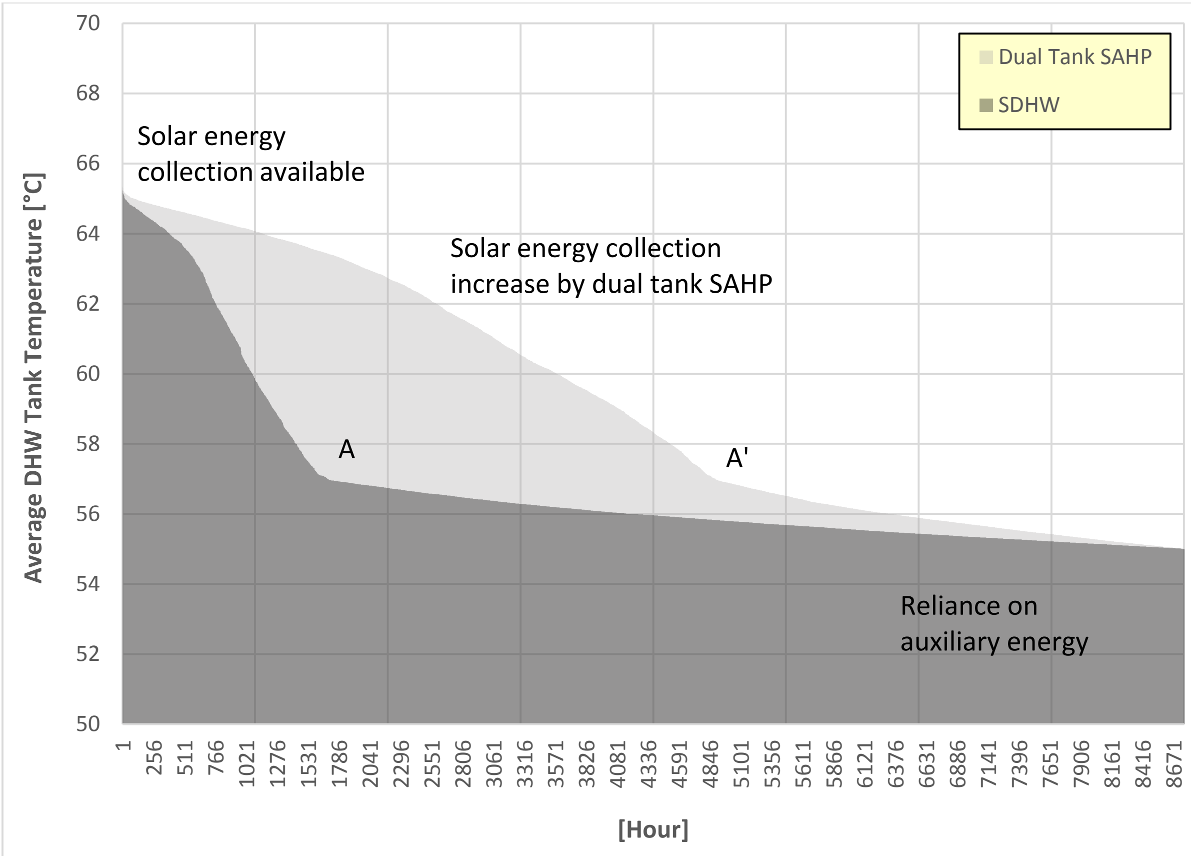

Figure 6.5 – Average DHW tank temperature by hours of year for SDHW and Dual Tank SAHP systems with 3 collectors, 45° tilt in Ottawa, ON

The energy savings resulting from the dual tank SAHP occur due to reduced reliance on auxiliary heating. Point A in Figure 6.5 divides the hours of operating in reliance on auxiliary heating (right side) from the hours where the STC and HP are contributing strongly to meeting the load (left side). The location of this point within the total hours of the year shows that the SDHW system relies on auxiliary heating for significantly more hours of the year.

For both systems, the hours to the left of point A are when solar energy collection can occur. The solar input raises the DHW tank temperature above the setpoint maintained by the auxiliary element, and additional solar energy is stored. Even when relying on auxiliary energy, the dual tank SAHP system has higher average DHW tank temperatures due to the additional solar energy that is stored in the lower nodes of the tank.

6.7 Float Tank Capacity

The presence of a second tank allows for increased solar fraction and energy savings. The main reason is that the overall volume of storage is increased, allowing the storage of more thermal energy without causing unreasonably high losses due to high tank temperatures. In addition, the float tank is allowed to reach higher temperatures than the maximum allowed in the DHW tank of 65°C.

Another benefit of a second tank is that the two tanks can have distinct temperatures. Although stratification is possible in the DHW tank during solar energy collection, the use of a HP dictates higher flow rates, which compromises stratification. A final benefit of the additional tank is that it allows for many more modes of system operation, opening up further possibilities to save energy and money.

It is important to identify an appropriate float tank size for a given situation. Although tanks in the range of 300 to 500 L are available at roughly the same cost, they do differ substantially in their overall dimensions. Since these tanks are typically located within the thermal envelope of the building, occupants or owners may be concerned with losing floor space. Therefore, it is beneficial to develop insight into the selection of a suitable float tank size. Table 6.11 outlines the results of simulations exploring the performance of a dual tank SAHP system with varying float tank sizes. The loss coefficient was kept constant for all tanks.

Table 6.11 – Impact of float tank size on performance of dual tank SAHP system with 3 collectors, 45° tilt in Ottawa, ON

Float Tank Size	Solar energy collected [GJ]	HP electricity [GJ]	Auxiliary electricity [GJ]	DHW tank losses [GJ]	Solar fraction	Energy savings [GJ]	Float tank losses [GJ]
One-third (150.1 L)	15.61	1.687	4.958	1.967 (8.1%)	70.1%	13.96 (67.7%)	0.06587
Half (225.2 L)	16.04	1.817	4.663	1.998 (8.1%)	71.2%	14.12 (68.5%)	0.1108
Two-thirds (300.2 L)	16.28	1.948	4.491	2.019 (8.2%)	71.7%	14.16 (68.7%)	0.1274
Baseline (450.4 L)	16.61	2.277	4.134	2.070 (8.3%)	72.2%	14.19 (68.9%)	0.1598
Double (900.8 L)	16.89	2.858	3.580	2.155 (8.5%)	72.4%	14.162 (68.7%)	0.07114

Solar energy collection increases modestly as float tank size is increased, however the amount of solar energy available is ultimately constrained by the amount of STC area installed. The DHW tank size remained constant in all cases, providing the main storage location for solar energy.

Auxiliary energy consumption decreases with larger float tank sizes, as the extra solar energy collected reduces the need for the backup source. However, DHW tank losses increase because the average tank temperature is increased due to the higher availability of stored thermal energy. Stratification is increased in the DHW tank for the smaller float tank sizes, since the heating element is used in lieu of the HP more often. This reduces the circulation of water which would cause destratification of the tanks, leading to a reduction in overall losses from DHW tank.

As expected, solar fraction grows with increasing float tank size since more storage exists for energy capture from the STCs. Although this is a good indication of performance, the energy savings better illustrate the range of ideal float tank size for the given conditions because they are proportional to the cost savings. Energy savings top out around a float tank size of 450 L, and bottom out for the smallest float tank size. These results highlight that increasing the float tank size beyond a certain point provides little or no benefit. Oversized float tanks have the downside of costing more, consuming more raw materials, having a larger surface area for heat loss, and taking up extra floor area. Overall, system performance is not significantly impacted by float tank sizes in the range of 225-450 L.

6.8 Tilt Angle

Solar panel tilt angle is an important design parameter of a solar thermal system because it dictates the orientation between incident radiation and the energy collection components. A common guideline for tilt is to have the panel at an angle roughly equal to the latitude at a location. This means that STCs should be installed horizontally at locations near the equator (0 degrees latitude) and at 45° from horizontal in southern Canada, northern USA, and continental Europe. This tilt angle orientation corresponds closely to the average solar altitude during periods of feasible solar energy collection.

Table 6.12 presents results for the performance of the dual tank SAHP system with 3 collectors for array tilt angles varying from 35° to 55°.

The performance of this SAHP is not heavily influenced by tilt angle in the range of 35° to 55°. The is mainly because the direct incident radiation on the collector plane, G_D, is based on the product of the direct radiation normal to the earth's surface, G_{ND}, multiplied by the cosine of the incidence angle, θ. Due to the shape of the cosine function, this means that the direct incident radiation does not drop steeply as the tilt angle varies. Equation (6.7) highlights this relationship [34].

Table 6.12 – Impact of STC array tilt angle on performance of dual tank SAHP system 3 collectors, 45° tilt in Ottawa, ON

STC Tilt Angle	Solar energy collected [GJ]	HP electricity [GJ]	Auxiliary electricity [GJ]	DHW tank losses [GJ]	Solar fraction	Energy savings [GJ]	Float tank losses [GJ]
35°	16.35	2.073	4.568	2.032 (8.1%)	71.1%	13.959 (67.8%)	0.2525
40°	16.55	2.180	4.315	2.057 (8.2%)	71.8%	14.11 (68.5%)	0.2373
45°	16.61	2.270	4.134	2.070 (8.3%)	72.2%	14.20 (68.9%)	0.1598
50°	16.59	2.417	3.993	2.086 (8.3%)	72.1%	14.19 (68.9%)	0.0459
55°	16.37	2.597	3.903	2.089 (8.4%)	71.6%	14.1 (68.4%)	-0.1323

$$G_D = G_{ND} \cos \theta \tag{6.7}$$

Table 6.13 shows that the optimal angle is indeed roughly the latitude of the location, which is 45°15’ N for Ottawa. Since the energy savings resulting from the system are not heavily influenced by tilt angles in the ranges explored, other factors may highlight potential benefits of modifying angle away from 45°, such as reducing the frequency of system stagnation and integrating with building roof slope.

It is worthwhile to revisit the issue of system stagnation explored for the variety of system configurations to look at the impact of collector tilt angle. Frequency of time above temperatures ranging from 90 °C to 105 °C are shown below in Table 6.13 for various STC tilt angles.

Table 6.13 – Impact of tilt angle on stagnation of dual tank system with 3 collectors, 45° tilt in Ottawa, ON

STC Tilt Angle	Annual Hours Collector Temperature Above			
	90 °C	**95 °C**	**100 °C**	**105 °C**
35°	206	117	46	10
40°	195	109	37	8
45°	177	93	29	4
50°	161	70	21	3
55°	138	54	15	3

These results confirm the expectation that increasing the tilt from horizontal reduces the frequency of stagnation. This occurs because the direct radiation flux on the panels is decreased during the summer months, but increased during the winter months. Since solar heating systems operate far better during the summer months and experience higher temperatures, the risk of stagnation is reduced. For the case of 55° tilt from horizontal, the system only experiences temperatures above 100 °C for 15 hours of the year.

Adjusting the collector tilt angle to 55° from 45° barely reduces the energy savings of the dual tank system for the location of Ottawa, Canada, but reduces the annual stagnation time above 100°C by about 14 hours. Stagnation is heavily influenced by the behavior of occupants: if they do not use hot water because they are absent, the system will be at an increased risk of stagnating. With this in mind, the higher tilt angle could have increased benefit since there are often times in the summer when occupants are away from their homes due to vacation or holidays.

6.9 System Justification

The results presented in this work show that the addition of a heat pump to a SDHW system did not significantly improve system performance. Although solar fraction was increased, this addition free energy collection was offset by the electricity input to the heat pump. Due to the lack of performance increase by the single tank SAHP system, the operating mode of Solar-HP-DHW was not used when investigating the performance of the dual tank SAHP system.

The real measure of justification is how much energy and money the additional equipment saves compared to its cost. For either 2 or 3 collectors, the dual-tank SAHP system saves about 1.5-2 GJ of energy per year. Energy sources and costs vary widely depending on the location and local markets, so it is difficult to estimate a payback period that would be relevant to people in all locations.

A more useful approach is to assume a payback period based on the expected life of the equipment and determine a ratio of capital expenses to energy cost. For example, if the payback period were assumed to be 20 years and the system saves 2 GJ per year, the ratio of the costs would have to equal the cost of 40 GJ. If electricity costs $0.10 per kWh, which is equivalent to $27.8 / GJ, the tolerable capital cost would be $1,112. Tolerable capital costs for many cases of payback periods and electricity rates are summarized below in Figure 6.6.

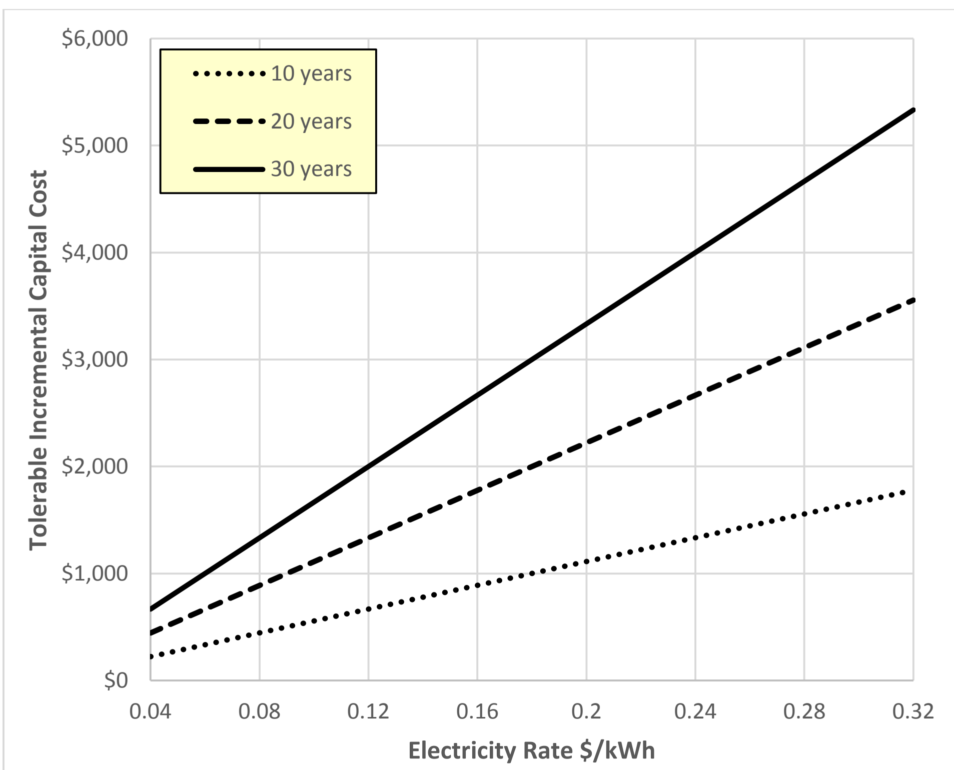

Figure 6.6 - Tolerable capital cost for dual tank SAHP system based on three payback periods

This sample calculation shows that SAHP systems in the context of single family homes and under the conditions investigated here are likely not easy to justify with current electricity and gas rates. This is because an additional storage tank alone would cost around $1000 at current prices. In addition, the HP, extra plumbing components, and added complexity make justification more difficult. Despite these drawbacks, there are many more applications and possibilities to explore which can improve justifiability.

One way to improve the justifiability of SAHP systems is to scale up their application to include multi-residential units. Although STC cost would provide constant returns to scale, the heat pump, storage, hydronic pumps, plumbing, and installation would provide increasing returns to scale. Another avenue to boost justifiability of SAHP systems is to apply them to additional loads such as air conditioning and space heating. Increased equipment utilization will lead to a higher return on investment. In addition, the implementation of time-of-use charges for electricity consumption prompts the investigation of intelligent controls to minimize the use of energy during ‘on-peak’ hours.

This chapter serves as a starting point in an ongoing development and evaluation process for SAHPs. This conclusion provides motivation to find other uses and benefits that the addition of a heat pump and second thermal storage tank can provide to a SDHW system.

6.10 Conclusions

For the water draw profile investigated, it was found that the addition of a heat pump to a SDHW system did not significantly reduce electricity consumption in comparison to the base case of an electric hot water tank. Configuring the SAHP system to include a second storage tank reduced energy consumption by 6-13% depending on the quantity of collectors used. Following a diminishing returns trend similar to a conventional SDHW system, additional solar panels also provide diminishing return. The second, third, and fourth panels added save an additional 19.2%, 10.5%, and 6.4% of energy, respectively, compared to only using one panel. System design recommendations resulting from the investigations in this chapter are summarized in the following section.

Overall, the current market conditions and electricity prices in Ontario and much of Canada may make it difficult to justify the installation of SAHP systems for single family residential homes. System implementation in multifamily residential settings would benefit from economies of scale. Applying the design to additional loads, such as space heating and cooling, would also make the dual tank SAHP system more attractive.

6.11 Design Recommendations

The following section is a culmination of the simulation results presented in this chapter. Based on the information produced and the inputs of the simulation, recommendations for STC area, HP capacity, thermal storage tank sizes, and collector tilt angle are discussed.

6.11.1 STC Area

The most suitable collector area to include with a SAHP system closely follows the typical practice for a SDHW installation. For a household of 5-6 people located in an area with similar weather to Ottawa, Canada, it would be appropriate to install either 2 or 3 collectors of the size investigated in this work, or about 1 to 1.25 m^2 area per occupant. This recommendation depends on the climate where the system is located. Areas with more favorable solar energy resources will require less collection area to achieve comparable performance.

Since the addition of a heat pump and second thermal storage tank improves system performance, one may be tempted to reduce the collector area. This is not recommended because the equipment that would be replacing some STCs is typically more costly and takes up floor space. A heat pump will typically cost more than a solar thermal panel and a storage tank requires interior floor space, whereas a solar panel does not require floor area.

A general scenario that could benefit from using a heat pump and additional thermal storage while reducing solar collection area is when access to direct solar radiation is limited. A common example is in urban environments where buildings have high energy use relative to their available collection area. Another common problem is shading from nearby structures or objects, such as buildings or trees. In cases where access to solar energy is restricted, maximizing the output per-unit-area of the installation may be more important than optimizing installation costs. Project constraints may have a significant impact on system design.

A logical next step would be to evaluate the performance of this SAHP system for a variety of locations throughout Canada and globally. Ottawa has relatively low solar resources due to a high occurrence of overcast skies. Calgary, although further north than Ottawa, has a higher design insolation, in part due to clearer skies on average [35].

6.11.2 HP Capacity

The nominal capacity of the heat pump for a SAHP system should roughly match the heating rate of an electric DHW heater that would be selected for the anticipated load. This will ensure that the system recovery will be similar to that of a traditional DHW tank, meeting the load requirements dictated by the occupants. Using a scaling factor of 0.6 for the simulations corresponds to a nominal HP output of 2.3 kW.

The size of the heat pump must be carefully selected if it is desired to operate the mode of Solar-HP-DHW since the operation mode of using the HP to extract heat directly from the solar loop is an important function. In this mode, a steady state energy balance must be achievable between the STCs and the HP, such that the HP extracts an amount of heat equal to the STC rate of delivery. If this energy balance cannot be achieved, the HP will turn off when the evaporator temperature reaches the low temperature cut-off, leading to cycling of the equipment. Frequent cycling of the compressor motor may lead to premature failure and will cause poor performance. In addition, operating at excessively low evaporator temperatures results in low COP values.

HP capacity selection is less constrained for the dual tank SAHP system operating without Solar-HP-DHW due to the large thermal buffer provided by the float tank. The additional modes of operation provided by this arrangement means that this system does not need to rely on being able to operate using the energy path of Solar–HP–DHW. Solar energy can be stored in the float tank when available and transferred to the DHW tank as required. For this system, the primary concern is to ensure the HP has enough capacity to recover the DHW tank temperature at an acceptable rate. It is still worthwhile to avoid oversizing equipment to minimize capital installation costs, and to promote sufficiently long run times so that peak efficiency is achieved.

6.11.3 DHW Tank Size

The size of the DHW tank for a SAHP system should follow sizing techniques that would be used for a standard SDHW system. This approach will help ensure a suitable quantity of stored thermal energy to meet the hot water demands for a building. Since water tank cost does not scale up with increasing capacity, it could be beneficial to be conservative by selecting a tank one size larger than required. This will also enable the system to be better prepared to meet unexpected loads, such as a change in occupancy patterns.

Currently in North America, common tank sizes for conventional electric DHW systems are 40, 50, and 60 gallons (151, 189, 227 Litres, respectively). Although a 60 gallon tank has 50% more capacity than a 40 gallon tank, the price increase is under 20%. Therefore, to promote acceptable system performance, the tank size should at least match the size of the DHW tank that would be selected for a SDHW system, and should use a larger size when it is reasonable to consume additional floor space. For 5 person occupancy, a DHW tank size of approximately 300 L is appropriate.

6.11.4 Float Tank Size

The size of the float tank impacts the solar fraction, energy savings, and stagnation frequency of the dual tank SAHP system. It provides additional storage volume to the DHW tank and can serve as a lower temperature source and sink. Although the lower temperature heat often cannot be used directly by the building's demands, the HP is able to upgrade the quality of this heat and deliver it to the DHW tank. The benefit of a lower temperature storage material is an increased ability to collect energy from the STCs.

There are disadvantages of a large float tank, such as more use of space and increased cost. A massive tank may require special equipment to install and may not fit through building doorways. Therefore, for single family residential installations, there is a definite upper limit on the size of tank that can and should be used. Larger scale projects are not subject to these limitations.

Simulation results show that doubling the tank size from 450 L to 900 L results in decreased performance by reducing energy savings by 0.2%. This highlights the necessity of balancing the size of the energy source with the storage. Beyond a certain point, additional storage cannot benefit the system unless the energy source capacity is increased.

In the opposite direction, the system's energy savings drop off as float tank size is decreased. Based on this trend, it is advisable to have a float tank size approximately 75-100% of the size of the DHW tank size. A capacity from 225 L to 300 L is a good float tank choice for the investigated system with a DHW capacity of 303 L. If additional space is available, the larger end of the size range is preferred since it will increase energy savings and is at worst marginally more expensive than a smaller tank. The larger tank also provides increased storage if other energy loads or sources were added to the system.

6.11.5 STC Tilt Angle

The tilt angle of the STC array impacts the amount energy collected by the system and the frequency and severity of system stagnation. The general recommendation of matching the tilt angle to the latitude of the location holds, but stagnation can be reduced by increasing the tilt angle. This does not decrease the energy collection of the system significantly, but could prolong the life of system components and the operating fluid. A tilt angle ranging from the latitude of the location to 10° greater than the latitude is recommended. Smaller tilt angles from horizontal than this are not recommended due to increased stagnation and decreased energy collection.

Chapter 7 Further Investigations

7.1 Time-of-Use Electricity

A large portion of the world's energy is consumed as electricity, and due to behavioral and scheduling patterns, there are peaks and troughs in this consumption which are predictable on large scales. Unlike thermal or chemical energy, electricity has very few implementations of large-scale storage, meaning that the load must be met in real-time by the generation. This has the unfortunate consequence that the source must be sized to the maximum anticipated load, even if this load is only rarely encountered.

Fortunately, intertied electricity grids exist in many parts of the world, allowing different electricity authorities to buy and sell as necessary. The unfortunate aspect of this is that wholesale electricity prices vary greatly and prices during peak demand are very high. The overall price of electricity is eventually passed down to the consumer, who is generally the person in charge of choosing their appliances and heating systems. Thus, it makes sense to reward consumers who reduce their power consumption during peak periods, and charge users more when wholesale electricity is more expensive.

Thanks to the advancement of electronics and communications hardware, time-of-use (TOU) metering is now implemented in many jurisdictions. Many U.S. states are currently upgrading to smart meters, along with most European countries and some Asian locations. In 2013, about 30% of U.S. homes had smart meters installed, with more installations to come, illustrating that smart meters are becoming more prevalent [36]. A technology shift should occur to follow this trend, with electricity consuming devices taking

TOU charges into account. For some appliances, such as dishwashers and clothes washers, users can manually take advantage of lower electricity rates.

Water heaters provide an energy storage medium that the electricity grid does not have, through a tank holding hot water. By intent, tankless heaters have negligible storage, but for the SAHP system under consideration there are many storage options. In addition to the ubiquitous DHW tank, the dual tank SAHP system has a second tank that can serve as thermal storage. The float tank could be used to store additional hot water, which can be prepared when electricity rates are low. Due to thermal losses to the environment, it may not be beneficial to store extra hot water during the cooling season. Therefore, supplemental hot water storage would be most suited to the heating season, where losses are not entirely disadvantageous. In addition, solar energy is at its lowest availability during these months, therefore auxiliary energy use would be at its highest, making this the most advantageous time to utilize the extra storage.

During the summer months, when extra thermal dissipation to the indoor environment is to be avoided, it would be beneficial to have a storage medium for chilled water. In Ontario during the summer months, the peak load is dictated by the cooling load throughout the province. Being able to shift the cooling load away from the peak would have great benefits for the province and its taxpayers.

This peak load generally occurs in the early afternoon on the hottest days of the year, when the ambient temperature, humidity, and solar gains are at their highest. The performance of a vapor compression refrigeration cycle using the ambient air as its heat sink is at its worst, since the COP decreases in line with the Carnot efficiency for the given temperatures. This means that not only is electricity more expensive, but the system is less efficient, further driving up operating costs. The separate storage provided by the float tank would allow chilled water for air conditioning to be produced during off-peak electricity rates and during colder times of the day, such as night or early morning. In this chapter, the TOU performance of the system previously described and the potential for energy and cost savings are evaluated.

7.1.1 Performance with Base Controller in Ontario

All system types previously investigated in this work were simulated once again, but with the electricity consumption being metered as TOU. A custom TRNSYS component was developed to allow the quantity of electricity in each time range to be tallied over the course of simulation. The days of the year and time of day which govern the TOU schedule can be fully customized to meet the needs of any jurisdiction. The source code for the TOU component is included in Appendix D – Custom Component Source Code.

For this work, Ontario's current time-of-use schedule is used, displayed in Figure 7.1 below.

Figure 7.1 – Current (2014) time-of-use electricity pricing in Ontario, Canada [37]

The resulting TOU totals for each system and time price category are presented in Table 7.1. From the overall results of system energy consumption, it can be seen that off-peak consumption does not vary greatly between the systems other than the dual tank SAHP. All systems consume about 42-50% of their power during off-peak hours. The baseline DHW system had the lowest on-peak consumption of all systems other than the dual tank SAHP, which was roughly equal to its mid-peak consumption, both being at about 27%. Since this system's performance is not impacted by weather conditions, on-peak and mid-peak rates are applicable an equal amount of time. In both summer and winter, each is applicable for 6 hours of each weekday, and their positions in the day switch from summer to winter.

Table 7.1 – Time-of-use results for system alternatives with 3 collectors, 45° tilt in Ottawa, ON

System Description	Electricity Consumption [GJ]			Blended Rate [¢/kWh]
	Off-Peak	Mid-Peak	On-Peak	
Baseline DHW	9.49 (46.1%)	5.59 (27.1%)	5.52 (26.8%)	9.73
SDHW	4.15 (49.9%)	1.85 (22.2%)	2.32 (27.9%)	10.00
Single Tank SAHP	3.54 (42.4%)	2.37 (28.4%)	2.45 (29.3%)	10.31
HP portion	0.628 (29.4%)	0.876 (41.0%)	0.633 (29.6%)	10.79
Dual Tank SAHP	3.80 (59.1%)	0.975 (15.2%)	1.65 (25.8%)	9.61
HP portion	1.65 (72.3%)	0.353 (15.5%)	0.278 (12.2%)	8.80

The off-peak charges are applicable every weekday between 7pm and 7am, or 50% of the time during each weekday. However, since the majority of hot water is used outside of this time range, it would be expected that off-peak consumption would be much less than 50% of the total. Since all weekends are off-peak for the full day, the proportion of energy consumed during off-peak is shifted back close to 50% for the DHW and SDHW systems.

It has been made clear in previous sections that the overall energy consumption is reduced when progressing through the investigated systems. In reality, the cost savings realized are not directly proportional to the energy savings due to TOU price variations. For the single tank SAHP system, on-peak consumption rose from the base case of 27% to 29%. For the single-tank system, there was a small shift of electricity consumption to both mid- and on-peak, which results from how the system operates.

The potential advantage the single-tank SAHP offers over a SDHW system is the ability to boost solar input, particularly during periods of low insolation levels. Returning to the TOU pricing diagram for Ontario, it is evident that nearly all of the weekday solar energy coincides with mid- and on-peak rates – 7 am to 7 pm. Only weekday solar energy during the early morning and late evening evades these high electricity rates, but the insolation levels at these times are so low they are generally not very beneficial as thermal input. This is a disadvantage to SAHP systems with controllers that do not take TOU charges into account.

Another important result to consider is that the dual tank system simultaneously reduces overall energy consumption, has higher HP usage and energy consumption, but the HP uses a very high proportion of its electricity during off-peak. For the single-tank system, the heat pump consumed only 29% of its total electricity during off-peak, whereas the HP for the dual tank SAHP consumed 59% as off-peak. The dual tank SAHP system is able to run the HP frequently during off peak hours due to the second storage tank. The dual tank SAHP system presents opportunities to further shift this consumption to off-peak times due to the additional thermal storage tank. For example, the DHW tank could be pre-charged in the early morning hours via the HP and float tank.

7.1.2 Potential Performance Gains and Cost Savings

In an ideal situation, all electricity would be consumed during off-peak hours to minimize the effective rate. In reality, this is unlikely to be possible since the majority of the hot water is consumed during mid- and on-peak hours. In order to investigate the limiting case, 100% off-peak usage will be considered for comparison. In this situation, the blended rate would be 7.5 cents per kWh for electricity consumption. It is important to note that there are many other charges on an electricity bill, for example, Waterloo North Hydro's current additional charges are: delivery ($17.90 per month and $0.0191 per kWh), regulatory ($0.25 per month and $0.007 per kWh), and debt retirement ($0.007 per kWh) charges [38]. While these additional per kWh charges factor in to the total operational cost, and therefore operational savings, they do not change based on time-of-use.

The dual-tank SAHP system used a total of 6.43 GJ, or 1,785 kWh, as shown in Table 7.1. Based on its blended rate of 9.61 cents per kWh, this equates to $172 of electricity *usage* charges per year. Note that this does not include all the other charges involved in an electricity bill. This is in contrast to the 5722 kWh or approximately $557 per year consumed for the base case of a traditional DHW tank modelled in Section 6.2.2, assuming a blended rate of 9.73 cents/kWh from Table 7.1.

Once again considering the dual tank system, the best possible blended rate would be 7.5 cents per kWh, obtained from achieving 100% off-peak usage. At this rate, the yearly cost to operate the system would be $134, or a savings of about $38. While this is a further 22% reduction in cost, it is not a substantial dollar value for a single-family home. This mentality of not benefitting much from existing time-of-use rates can be extended to general electricity consumption within a home. With an estimated 63% of consumption already naturally occurring during off-peak hours [39], there is not much ability to reduce overall charges significantly, unless a very large amount of power is consumed.

Until a more rewarding off-peak rate is seen, the benefits to consumers will be quite small and unattractive. Shifting electricity usage to off-peak is only noticeably beneficial for high energy draws, such as traditional water heating systems that use electricity. The dual tank SAHP system uses such a small amount of electricity for a single family home that further optimizations offer small cost savings. At the current time, optimizing time of use performance for the dual-tank SAHP system developed for DHW delivery will not be pursued.

Electricity-based space heating or cooling systems could offer significant savings since they consume a high amount of energy. This can only be possible with thermal storage, since heating can be required at any time of the day and cooling is generally in highest demand during daylight hours. Therefore, the chilled or heated water must be prepared during off-peak hours and stored for later use.

Another example where taking advantage of electricity time-of-use charges for a SAHP system would be beneficial is if the system was used as a district heating system, as the load could easily be multiplied by servicing many residences. Although it is good practice as a region, province, or country to reduce peak loads to save money purchasing expensive power or building additional power plants, the savings must be realized by the customer in order to motivate them to make the effort to reduce peak usage.

7.2 Climate

When considering overall building design and performance, local weather variations are of great importance. Up to this point, only Ottawa, Ontario, Canada was considered in order to provide a consistent basis for comparison during system configuration and optimization. Although the baseline calculation and the DHW simulation are not dependent on weather, the SDHW and both SAHP systems are greatly affected by weather.

Solar space and water heating is not prevalent in Canada due to less than ideal solar conditions. Contrast this with highly solar equipped countries such as Austria, Germany, or Greece, where economic pressure and/or solar resources fuel the market. Even though the majority of population centres in Canada have better annual solar insolation levels than Germany, acceptance of the technology is much lower in Canada [35].

Many global locations are investigated in this chapter, with an attempt made to represent the different climate and solar regions that exist in the world. The Köppen climate classification was used in order to make the best attempt to represent each major climatic region. The classifications that were considered are: Hot Desert, Hot Summer Mediterranean, Humid Subtropical, Humid Continental, and Subarctic [40].

The Köppen classification system was used in conjunction with a world map of solar design insolation in order to choose particular locations of interest [35]. These locations, their latitudes, and their design insolations are summarized in Table 7.2.

Table 7.2 – Global locations used for climate study, their climate type, latitude, and design insolation

Location	Climate Type	Latitude [°]	Design Insolation [35] [kWh/m^2/day]
Aswan, Egypt	Hot Desert	24°N	6.4
Rome (Fiumicino), Italy	Hot Summer Mediterranean	42°N	3.0
Sydney, Australia	Humid Subtropical	34°S	4.1
Shanghai, China	Humid Subtropical	31°N	3.1
Beijing, China	Humid Continental	40°N	4.2
Salzburg, Austria	Humid Continental	48°N	2.2
Ottawa, Canada	Humid Continental	45°N	2.0
Whitehorse, Canada	Subarctic	61°N	1.5

The performance of a SDHW system and a dual tank SAHP system with 3 collectors was modelled for these locations. The tilt of the STCs was set to match the latitude of each location. DHW tank size used is 300 L and float tank size used is 450 L. The results of annual simulation for the systems and locations selected are presented in Table 7.3.

Table 7.3 – SDHW and dual tank SAHP electricity consumption for global locations

Location	Total Electrical Energy Input [GJ]		
	SDHW	Dual Tank SAHP	SAHP Savings
Aswan, Egypt	4.72	1.28	3.44
Rome (Fiumicino), Italy	6.43	4.31	2.12
Sydney, Australia	6.61	4.26	2.35
Shanghai, China	9.05	6.96	2.09
Beijing, China	8.51	6.17	2.34
Salzburg, Austria	11.0	8.92	2.08
Ottawa, Canada	8.30	6.41	1.89
Whitehorse, Canada	11.7	9.41	2.29

7.2.1 Hot Desert Climate

It should be expected that any solar thermal energy system will perform well in any hot desert climate due to frequent clear skies, high amounts of insolation, and high ambient temperatures. The additional storage volume of the float tank improves the solar fraction since the maximum temperature permitted within the DHW tank is 65°C, which limits solar energy storage in this tank. The float tank provides storage for additional solar energy available during the day and can be discharged at night via the heat exchanger or heat pump.

Aswan, Egypt was investigated as a representative of a hot desert climate, and generally experiences clear and hot days throughout the course of a year. This results in high levels of insolation year-round, producing a very repetitive pattern of daily temperature oscillations throughout the whole year. Looking at the temperature within the DHW tank over the course of a year in Figure 7.2, it is very difficult to identify a particular season, weather pattern, or time of year.

The simulation results in Table 7.3 for Aswan show that both systems performed best in this location out of those considered. The dual tank SAHP system significantly reduced the annual electricity input due to the extra storage tank. For both systems, the DHW temperature oscillates daily throughout the year, as shown in Figure 7.2 and Figure 7.3. There are no days during the year where the average tank temperature is unable to reach 65°C, which shows that solar energy from either the STCs or the float tank is readily available.

The grey line labeled ‘Final Outlet’ in both figures indicates the temperature delivered to the load after tempering, which is centered around 55°C with a variation of less than two degrees. The setpoint for the DHW heating element is 60°C to ensure that water is not delivered below 55°C. Therefore, any instances when ‘DHW to Load’ in Figure 7.2 is able to reach 65°C indicates that solar energy was used.

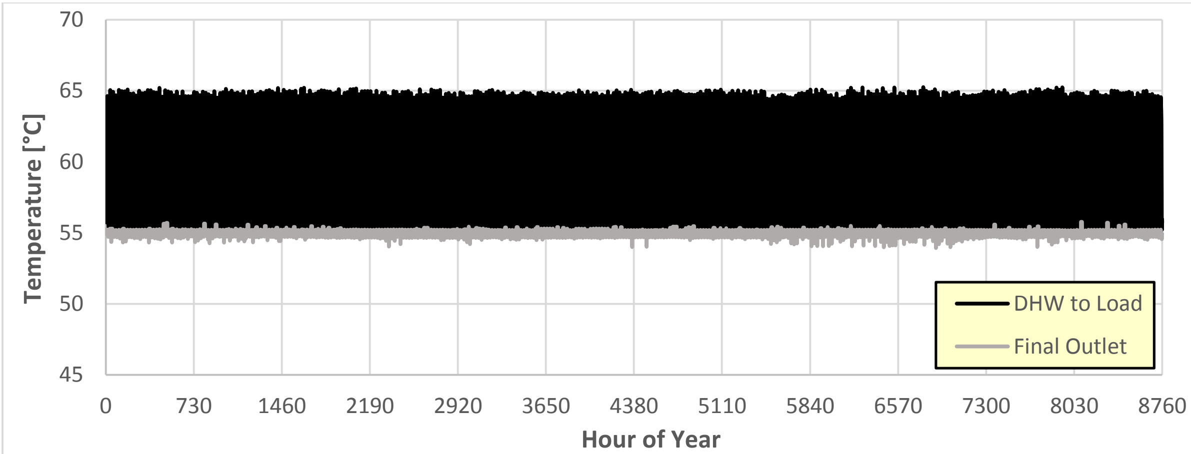

Figure 7.2 – DHW tank and final outlet temperature for Aswan, Egypt (SDHW)

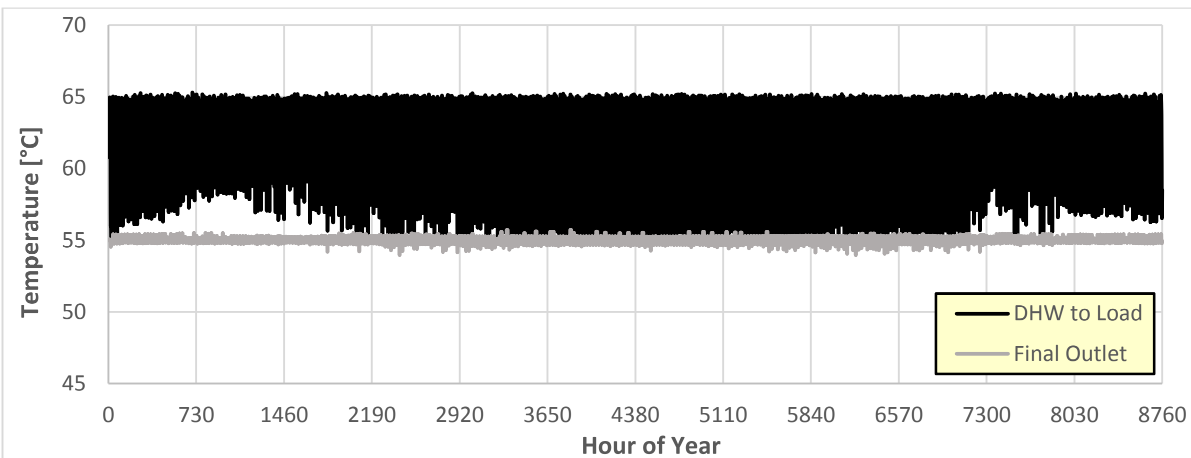

Figure 7.3 – DHW tank and final outlet temperature for Aswan, Egypt (dual tank SAHP)

Keeping the results for the SDHW system in mind, now consider the dual tank SAHP results in Figure 7.3. In contrast to the SDHW system temperatures, the SAHP system is able to maintain higher temperatures in the DHW tank during portions of the year.

Overall, it is clear that for hot and dry climates a dual tank SAHP increases energy savings in comparison to a SDHW system. Due to the high availability of solar resources year-round, the STCs are able to meet a high proportion of the annual hot water demand without the second tank. However, due to maximum temperature constraints of the DHW tank, the float tank provides additional thermal storage to improve the amount of solar energy collected.

The HP may have additional use in this climate type is for space heating. Although temperatures below freezing are not seen very often in hot desert climates, the temperature does fall a substantial amount at night due to radiative cooling. In this situation, the additional storage and HP would be able to meet any nighttime heating requirements.

Another potential application of a HP in this climate would be for simultaneous space cooling and DHW heating. Depending on the magnitudes of the respective loads, this system would be able to reduce the quantity of panels required for water heating.

7.2.2 Hot Summer Mediterranean

Hot summer Mediterranean is similar to hot desert climate type due to the generally high temperatures year round, but this climate type is slightly cooler and not as dry as hot desert. It is expected that solar systems will perform quite well in hot summer Mediterranean, but with noticeably higher electrical energy usage in comparison to a hot desert climate.

The SDHW temperature results for Rome illustrate this drop in solar collection and increase in electrical consumption. In Figure 7.4, it can be seen that the DHW temperature for the SDHW system does not sit close to the temperature limit of 65°C for as often as it did in a hot desert climate. This is also reflected in the increased electricity usage reported in Table 7.3. The dual tank SAHP results for this location are shown in Figure 7.5.

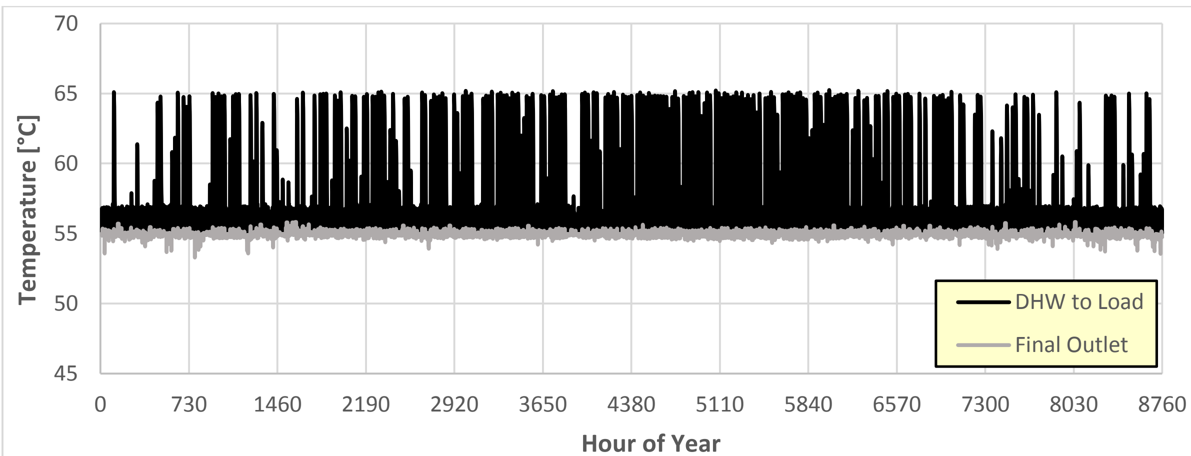

Figure 7.4 – DHW tank and final outlet temperature for Rome, Italy (SDHW)

Comparing the SDHW results (Figure 7.4) to the SAHP results (Figure 7.5), it is evident that the second tank and heat pump provides a significant boost to the DHW tank temperatures from approximately mid-February (hour 1000) to mid-November (hour 7500). This effect is what results in the reduced electricity consumption, as the system is able to collect more solar energy using the second thermal storage tank and the heat pump. The total electricity reduction was about 2.12 GJ, in comparison to the reduction for Ottawa of 1.89 GJ.

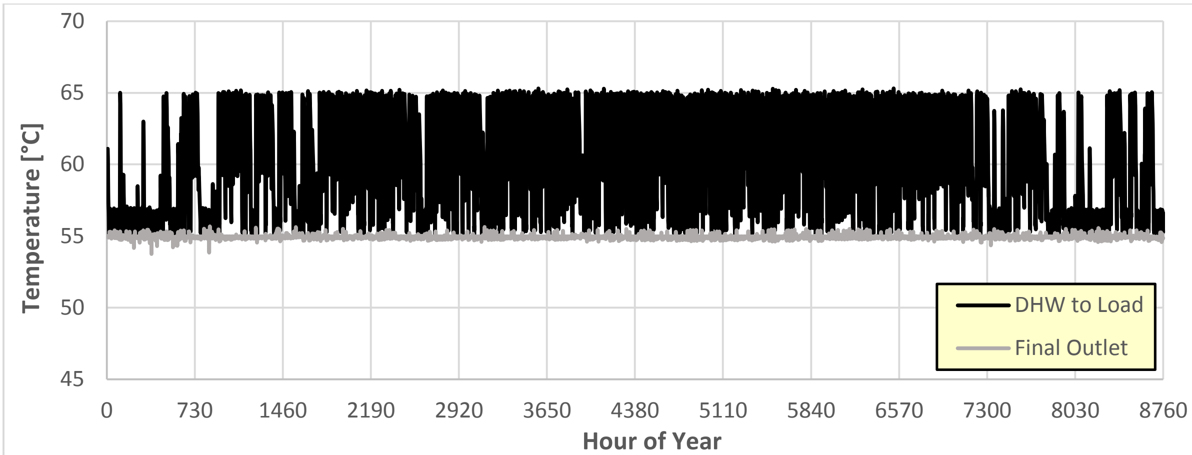

Figure 7.5 – DHW tank and final outlet temperature for Rome, Italy (dual tank SAHP)

The SAHP system delivers tangible energy savings in a hot summer Mediterranean climate. If the HP and extra storage tank could be put to use for space cooling, the system justifiability could be much more feasible.

7.2.3 Humid Subtropical

The next climate considered is humid subtropical, which is similar to Mediterranean, but hotter and more humid during the summer months. In general, locations of this climate type will have slightly higher design insolation values, predicting that they are likely to use less auxiliary energy than Hot Summer Mediterranean locations. In reality, the two Humid Subtropical locations investigated used *more* auxiliary energy for the SDHW system than the Hot Desert and Hot Summer Mediterranean Climates, which can be explained by looking at further weather metrics.

The design insolation for Rome is lower than Shanghai and Sydney in part due to being further from the equator. Therefore, during cloudy periods in winter, Rome is more likely to experience the lowest insolation levels of the group. However, due to the high humidity for Beijing and Sydney, these two locations experience significantly lower mean daily sunshine hours than Rome. For example, during the 3 month period beginning from the month with the summer solstice, Rome receives 10.2 mean daily sunshine hours, whereas for Shanghai the mean is 5.6 hours per day, and for Sydney 7.1 hours per day [41,42,43]. This result highlights the necessity of considering parameters beyond latitude and ambient air temperature when identifying exceptional locations for solar energy systems.

Despite using more auxiliary energy than Rome for SDHW, Sydney used less energy for the SAHP system. Part of this effect can be explained by the generally higher air temperatures in Sydney, allowing the SAHP system to collect solar energy with reduced losses to the ambient environment. Overall, an SAHP system in a humid subtropical climate will see significant performance improvements over an SDHW system. The energy savings are greater than the initial location, Ottawa, and the Hot Summer Mediterranean location.

7.2.4 Humid Continental

The penultimate climate type considered revisits the original location of Ottawa, Canada, and compares system performance to Salzburg, Austria, and Beijing, China. Locations of this climate type have high humidity, similar to Humid Subtropical locations, but experience lower ambient air temperatures due to increased latitude. Electricity usage for both SDHW and SAHP was very similar for Ottawa and Beijing. Although Ottawa experiences colder ambient temperatures, Beijing receives less sunshine during the months of June, July, and August at 7.7 mean hours per day, whereas Ottawa receives 9.0 per day. Salzburg receives even less sunshine at 6.9 mean hours per day during the same period [44,45,43,46].

Overall, favorable solar conditions vary more widely over the course of a year in Ottawa, but the conditions achieved during crucial summer months give it an overall comparable solar performance to Beijing's more steady solar energy conditions. Performance of both SDHW and SAHP struggles in Salzburg due to very low sunshine hours all year round, with the winter months being exceptionally poor. During the 3 month period beginning with the winter solstice month, Salzburg receives 2.5 mean hours of sunshine, Ottawa 3.4, and Beijing 6.3 [44,45,43,46].

Among these locations of the same climate type, energy savings varied noticeably. The high ambient humidity in these regions contributes to this variability, since the formation of clouds dictates the availability of solar energy at the earth's surface over the course of a year. An interesting comparison can be made between Ottawa and Beijing and yet another by considering Salzburg.

Beijing's insolation levels over the course of a year are less variable, with the summer and winter averages being 7.7 and 6.3 mean hours of sunshine, respectively. This contrasts Ottawa, which receives 9.0 and 3.4, respectively. Insolation levels in Ottawa vary significantly from summer to winter, primarily due to many days of overcast skies. The comparison between these two locations highlights the strength of a SAHP system: performance is significantly improved during periods of moderate sunlight.

The SDHW in Ottawa outperforms Beijing marginally, by using 8.3 GJ of electricity in comparison to 8.5 GJ. However, the SAHP performed moderately better in Beijing, dropping the electricity usage to 6.2 GJ versus Ottawa's usage of 6.4 GJ. This contrasting result can be explained by the connection between the weather, float tank, and heat pump operation. An overcast winter day in Ottawa does not provide any opportunity to collect solar energy. In contrast, a moderate winter day in Beijing with nearly double the mean hours of sunshine provides more opportunities for the SAHP system to collect solar energy than in Ottawa.

Beijing's better access to solar energy in the winter explains the higher performance gains of the SAHP system; the higher performance in Ottawa of the SDHW system can be explained by considering the summer weather. Ottawa's summer is not as hot as Beijing's, but it does have clearer skies on average. Since the solar energy input is driven more by the incident radiation than the ambient temperature, this results in Ottawa's SDHW performance being superior during the summer.

Overall, these findings show that locations with less variation of mean hours of sunshine from summer to winter benefit more from the use of a SAHP system. This is because the weather in these locations more frequently aligns with the conditions under which the SAHP system thrives.

7.2.5 Subarctic

Subarctic locations are of particular interest for energy conservation and new energy sources due to the isolation from sources in more populated areas. For example, in Canada, the subarctic region is not connected to the main electric grids of North America. In a similar way, pipeline infrastructure is also lacking in this region in comparison to southerly locations. Due to the higher cost of power, capital costs could be more easily justified if SAHP system performance in subarctic regions is acceptable.

Whitehorse, Canada was selected for a representative subarctic location. The summers days are long and sunny with about 8.2 mean daily sunshine hours, but the winter days are short and dark with only about 2.0 mean daily hours. The average ambient temperature is very low, at just -0.1°C on an annual basis. These factors make Whitehorse a very challenging location for solar energy, in particular because of the long portion of the year when such a system would be dormant. Not surprisingly, SDHW performance in Whitehorse was the worst out of all the locations investigated. The challenging conditions force the system to rely heavily on auxiliary energy input. Despite this, there were some surprising results relating to Whitehorse.

First, Whitehorse's SDHW performance at 11.7 GJ total energy use was not much worse that Salzburg's at 11.0 GJ. This is surprising because Whitehorse is at a far higher latitude, has a worse design insolation, and has a much colder climate, particularly in the winter months [47]. Second, the electricity savings that the dual tank SAHP could provide over the SDHW system were comparable to the savings seen for other locations. It is encouraging that even a subarctic location sees a similar performance increase with a HP addition, as these locations typically have higher energy costs due to lengthy and challenging supply routes.

Figure 7.6 and Figure 7.7 plot the temperature of water reaching the tempering valve. These two figures highlight the impact a SAHP has over SDHW: the load is met more consistently by solar energy during the summer months due to the temperature and energy boost given by the float tank and HP.

Although special challenges arise when using HPs in extremely cold climates, the performance increase obtained by a SAHP in this region make it an intriguing candidate for a DHW system. In conjunction with an additional heat source, for example ground heat, a SAHP system could offer considerable cost savings in a subarctic location.

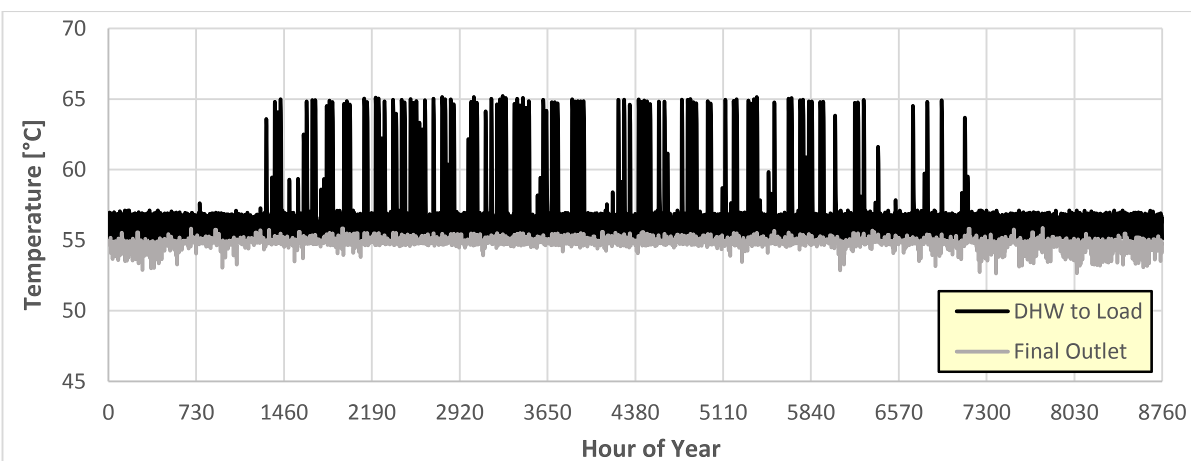

Figure 7.6 – DHW tank and final outlet temperature for Whitehorse, Canada (SDHW)

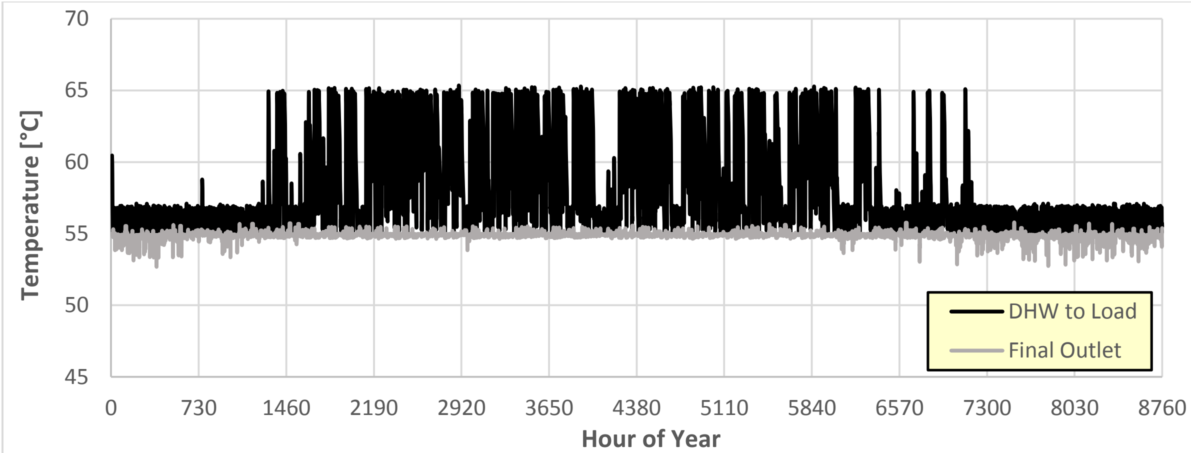

Figure 7.7 – DHW tank and final outlet temperature for Whitehorse, Canada (SDHW)

7.2.6 Overall Recommendations

The majority of the locations benefitted noticeably from the addition of a heat pump and second storage tank to the DHW system, as depicted in Figure 7.8. If space cooling is to be used in a hot desert climate, the HP could be used for DHW heating as well, reducing the quantity of STC area required. However, a HP should not be used exclusively to replace a STC, since the capital cost and operational complexity is higher.

For the majority of locations, the dual tank SAHP reduced electricity consumption by approximately 2 GJ. This is an important result because the SDHW electricity usage of these locations ranged very widely, from 6.4 to 11.7 GJ. The hot desert climate had savings of over 3 GJ due to the combination of a high amount of solar energy available for the location and the additional thermal storage tank.

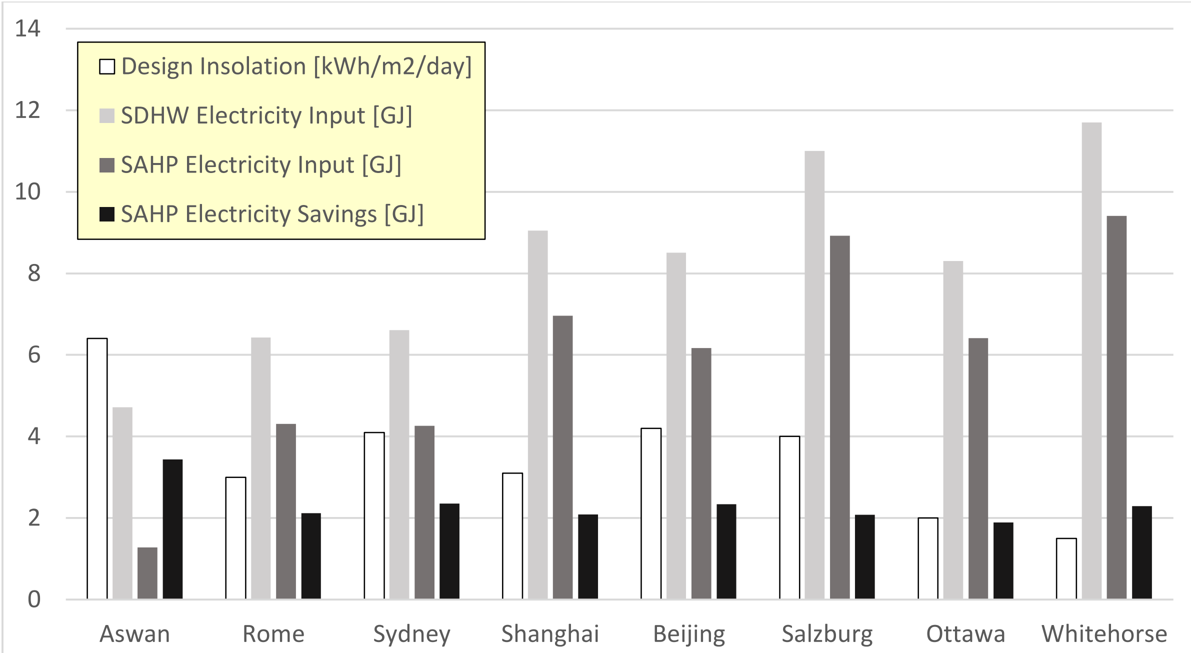

Figure 7.8 – SDHW vs. SAHP performance for select global locations

In comparison to the electricity consumption of the base case DHW system, all locations benefitted from the additional functions enabled by the dual tank SAHP system. On the high end of this range excluding Aswan, the dual tank SAHP decreased electricity consumption by 36% for Sydney. On the low side, the savings for Salzburg were only about 19%.

Applying the HP to additional loads, such as a cooling load, would allow the HP to further reduce site energy consumption. Connecting the HP to additional sources, such as air or ground, would allow the HP to meet demand when using the solar source is not viable. Enabling the HP to be used more frequently, in lieu of auxiliary energy input, would lead to further substantial electricity usage reductions. Overall, the merit of the system is tangible and could be further increased through these measures.

Chapter 8 Conclusions

A novel configuration of a dual tank solar assisted heat pump system was investigated from initial conception to detailed performance analysis. The system configuration was designed and modelled in TRNSYS, and the need to tune the model was identified. Custom controller logic was developed to support the variety of possible operation modes. Over the course of the research project, an experimental apparatus was built collaboratively with a M.A.Sc. project by Will Wagar. This apparatus was used to support the development and tuning of an accurate system model, which was further used for many investigations.

The performance of the SAHP model was compared to the results from the experimental work and showed excellent agreement. The custom control strategy was analyzed and debugged in detail to confirm proper operation. Parametric studies were completed to optimize system performance and form design recommendations. Further investigations were completed using the tuned model, first studying the potential impact of applying control strategies to take advantage of electricity time-of-use pricing. Second, the performance of the dual tank SAHP system was modelled in many global locations to support climate recommendations. Finally, recommendations are offered in the final chapter to identify directions for continuing research on the subject of solar assisted heat pump systems.

Solar assisted heat pumps have a variety of possible system configurations and design, of which some options include: direct, indirect, storage techniques, energy sources, STC selection, compressor type, and plumbing connection design. For the investigations completed in this work, the major components of the SAHP system were STCs, a HP, a HX and two thermal storage tanks. Hydronic pumps and electronically actuated valves allowed these components to be interconnected and controlled. The nature of the configuration allowed for the possibility of several modes of operation, which are selected using a custom control algorithm.

The test apparatus successfully supported the overall project goal of modelling, designing, and optimizing the dual tank SAHP system. The apparatus performed as intended and enabled the development of accurate component models. The temperature, water flow, and electrical power measurement techniques provided accurate methods for characterizing components, in particular the heat pump, which required many points to create an accurate performance map.

After individual mode validation, a unique control strategy was developed to handle the nuances of the configuration and select the appropriate mode of operation based on weather and system conditions. This strategy first identifies which modes are possible, based on specific logic for each, and then selects the mode to operate based on highest priority. The priority order favours modes of operation that minimize electricity usage. Therefore, when collecting solar energy, the controller would prefer to operate without the HP if possible. After model component tuning and controller debugging, the SAHP was used to support design and optimization investigations.

Many parameters were varied during simulations to optimize the overall system design. The simulations resulted in recommendations for system parameter selection, such as STC area and tilt angle, HP capacity, and storage tank sizes. It was identified that the energy savings from the use of a HP was similar in magnitude to the final STC panel addition for a properly sized system. This might suggest that the HP can replace STC area, but this strategy is not recommended. At the time of writing, HPs of the size used in this work are at least double the cost of a 2.5 m^2 glazed STC. In addition, HPs have moving parts that could fail easier than STCs. Therefore, it is recommended that SDHW collector area sizing techniques are used for a SAHP without any STC reduction, unless physical constraints exist. Additional area beyond the area specified by SDHW system design practices would not be very cost-effective, as diminishing returns in system performance do not justify the added expense.

In addition to assessing the electricity savings of the system as parameters are varied, two key performance indicators were assessed: system stagnation and ability to meet load. It was identified that a properly sized dual tank SAHP system does not cause excessive system stagnation. It is important to minimize stagnation in order to prolong the life of the water-glycol mixture used as an antifreeze thermal storage medium. It was found that a single tank SAHP system increases stagnation severity and frequency by a noticeable margin in comparison to a SDHW system. The addition of a float tank reduced stagnation substantially, by approximately an order of magnitude in comparison to the SDHW system. The dual tank SAHP makes better use of solar energy and increases system longevity through stagnation reduction.

Since the purpose of the system is to meet the DHW demand of occupants, it was crucial that the ability of the system and control strategy to meet the load was confirmed. This was completed by totaling the hours that the outlet to the load was above 53, 55, and 57°C. It was found that the final outlet to load is above 53°C for 99.9% of the year, which meets the standard for SDHW systems specified by CSA. These findings indicate that the system and controller were operating as designed and intended. It was found that the outlet temperature is never above 57°C, which means that energy is not being wasted and occupants would not experience abnormally high water temperatures.

For selecting a suitable DHW tank capacity and nominal HP output, a simple SDHW system should be referenced. SDHW systems are tested for recovery time and their ability to meet hot water demand, which is impacted by the tank size and element power output. Therefore, for a given occupant load, typical SDHW sizing techniques should be used to ensure the availability of hot water. The element size can be used for heat pump sizing because its output represents the amount of thermal power that should be available for acceptable recovery. However, it may be desirable to use a smaller capacity heat pump to save costs, but this may cause additional reliance on the auxiliary element.

No analogy to a SDHW can be made for sizing the float tank, since the energy sources and storage temperatures are highly variable. Many float tank sizes were investigated, and it was found that a float tank size of approximately 75-100% of the DHW tank capacity is a reasonable choice. Selecting a specific size within this range is dependent on the installation and local pricing. If space is a concern, a smaller tank can be used and will only reduce electricity savings by about 1-2%. If space is not a big concern and a larger tank is marginally more expensive, then the opportunity to have more thermal storage should be taken.

The impact of collector tilt angle on electricity savings and key performance indicators was assessed. Through this analysis, it was found that an ideal tilt angle for a SAHP system is in the range from the location latitude to 10 degrees extra tilt from horizontal. Therefore, for a location at 45° latitude, the tilt from horizontal could be selected as 55° to provide beneficial performance outcomes. Increased tilt would improve winter and shoulder season collection, and decrease summer energy collection. This results in a minor increase in energy savings and a reduced system stagnation occurrence over the course of a year.

The dual tank SAHP system was compared to alternatives to form a basis for economic justification. The comparison was done with a range of electricity rates and payback rates to make the work more accessible to readers throughout the world and because electricity rates change over time. The general conclusion for the scenario investigated was that system justification is difficult for single family residential applications at current electricity rates. Many suggestions for improving justification were identified, such as applying the system to a larger load, more loads, or adding more energy sources for the HP. Underutilization of the HP relative to its cost is a major cause for the lack of justifiability.

Further investigations were completed which analyzed the impact of electricity time-of-use (TOU) charges on controller design and categorized the performance of the system for a variety of global climates. Since TOU electricity rates for on- and off-peak vary by approximately a factor of 2, the possibility for substantial savings exists. A custom TRNSYS component was developed to track electricity consumption during different time intervals during the day. The capability of switching the charging scheme throughout the year was also created to account for the difference between summer and winter TOU rates in Ontario.

The benefit of TOU electricity was investigated to see if the control strategy should be modified to behave differently depending on the time of day. It was found that all solar DHW systems other than the dual tank SAHP increased the blended electricity rate by using proportionally more electricity during mid- and on-peak hours in comparison to a traditional DHW system. For the dual tank system, the HP operated during off-peak hours nearly twice as much as the single tank system because the second storage tank increases the possibility of operating the HP earlier in the morning and later in the evening.

Current electricity rates were used along with the TOU predictions for the model to estimate an annual electricity usage charge, which excludes fixed charges. Due to the large reductions in electricity usage by the dual tank SAHP system, not much opportunity for further savings exists. If a larger system were installed, it would be beneficial to design a TOU controller since it would be possible to reduce the usage charge by about 22%. It would also be beneficial to implement TOU-based control if the system were used for space cooling, since the summer electrical peak in Ontario results from an air conditioning load. In this way, cooling could be obtained at a far lower cost due to lower electricity prices and more favorable conditions for the HP during nighttime hours.

The final part of the work was to investigate many global climate types to analyze system performance across a broad spectrum of relevant locations. Extremely warm and sunny locations were considered, as well as temperate climates and the subarctic. The dual tank SAHP system improved performance in all locations due to the extra thermal storage of the float tank. Annual energy savings were approximately 2 GJ for all locations, with the exception of Aswan, Egypt saving about 3.4 GJ. Aswan saved more energy due to the abundance of solar energy available throughout the year.

A HP would be easier to justify in the warm climates if it was also used for space cooling. Incremental capital costs can be decreased by combining the functions of space cooling and water heating. A motivating outcome from the climate studies was the performance of the SAHP system in Whitehorse, Canada, the Subarctic location considered. SDHW performance is quite poor due to the long, cold winters, however, the SAHP system was able to provide energy savings similar to most other locations. This is encouraging, as resources are more expensive in isolated regions, resulting in higher energy costs.

This research project started with experimental work to confidently establish a model suitable for system design and development. The experimental work aided in developing an effective control strategy and identifying operational issues that can arise with a system of this type. Recommendations were developed to assist in the design of SAHP systems, with emphasis on the development of location-independent guidelines.

Further research was completed on the optimized system, first considering the potential impact TOU strategies could have on further energy savings. Finally, the dual tank SAHP system was simulated in many global climate types to gain a perspective on what locations are most suitable. Throughout the investigation, many recommendations for new directions were identified, and are described in detail in the final chapter.

Chapter 9 Recommendations

Overall, the dual tank SAHP system created through this work performed well and reduced electricity consumption significantly in comparison to alternatives. Use of a second storage tank improves performance by increasing thermal storage and providing a storage location for water at a distinct temperature than the DHW tank. Due to current electricity rates, in combination with the load expected for a single family residential application, it was found that current equipment costs likely prevent economic justification for installing a dual tank SAHP system in small applications. Recommendations for further areas of investigation result from identifying why economic justification does not exist for the cases studied in this work.

Since the cost of grid electricity is predetermined by market conditions, the approach in improving justification is to reduce the capital costs in proportion to the energy savings. This can be done in two main ways: applying the system to more loads or a larger load and/or supplying the system with more sources.

The simplest approach would be to increase the scale of the load by applying the system to a district residential load or by application to a larger building. Further studies on this subject should investigate the sizing and performance of a system for a multi-residential low-rise apartment building and also to a district heating system. Many existing low-rise apartments could serve as the modelled building, with the main challenge being to appropriately model the water draws. Drake Landing Solar Community presents an excellent modelling opportunity for further investigation, as the replacement of the auxiliary gas-fired heat source with a heat pump could be investigated [17].

Aside from increasing the scale of the load, applying the SAHP system to additional loads is another important area of further study. In order to do so effectively, it may also be necessary to supply the HP with additional energy sources. Options for additional sources will be described after possible additional loads are discussed.

A natural additional load to meet with a dual tank SAHP is space cooling. In Canada, about 52% of homes utilize air conditioning systems. For Ontario, this figure jumps to 80% due to a warm, humid summer [48]. The rates are even higher for homes built after 2000, meaning that most new residential construction in Ontario includes air conditioning. The installed residential air conditioning equipment in the province is underutilized, as it sits dormant for a large portion of the year. Air conditioning is generally only required in Ontario for the months of June, July, and August, meaning the compressor could be used as part of a SAHP system for the remainder of the year [49].

During the cooling season, the float tank could store chilled water rather than warm, since solar resources are at the highest availability. This means that less storage is required to obtain a particular solar fraction for the system. With a thermal storage medium for chilled water, air conditioning systems could operate during off-peak hours, when electricity rates are reduced and higher COP can be achieved due to lower ambient temperature. Transitioning a substantial quantity of homes in Ontario to such a system could allow the summer midday electricity consumption peak to be noticeably decreased.

A further option of an SAHP system which meets the cooling load of a building is rejecting heat from cooling to the DHW tank rather than air. Although at times the COP could be lower than exchanging to air due to higher temperature of DHW water, it could use less energy than separately heating water with an auxiliary element and space cooling with an air conditioner.

A SAHP system can also be used to meet a space heating load; however, the conditions that prompt the need for space heating are generally unfavorable for solar energy collection. Therefore, it would be necessary to provide a large storage medium, for example ground storage, or to provide additional heat sources for the HP to meet the heating load in the winter months. During the shoulder months, a dual tank SAHP system with increased STC capacity might be able to meet a noticeable portion of the space heating requirement.

With a similar mentality of maximizing the efficient use of equipment, the HP could be further utilized by making energy sources other than solar available. For the space cooling or heating arrangements described above, it would be necessary to install an air source or ground source heat exchanger to allow the system to meet the cooling load when the DHW tank does not require any heat and the heating load when solar energy is not available. These additional sources would also allow for further configuration changes, such as reduced float tank size or decreased STC quantity.

After any of these above changes which attempt to deliver more energy to more loads with a SAHP system, it is important to reconsider the impact TOU changes has on the operating costs. On a large application scale, controls implementing TOU strategy could have large savings potential. On the broader regional scale, TOU savings in aggregate would reduce peak electricity and help reduce the need for additional electricity generation facilities, or expensive electricity imports from other geographic regions.

The overall theme of the aforementioned recommendations is to consider energy delivery within a building comprehensively. A shift from considering discrete pieces of equipment for each load, to interconnecting equipment to meet multiple loads should be adopted to better utilize installed equipment.

Further modelling investigations that could be undertaken to increase justifiability without modifying energy sources or loads are: the use of an external auxiliary element, configuring the solar panels in parallel, and utilizing unglazed collectors.

Using an external auxiliary element to allow tank temperature to be lower than the required delivery temperature would save energy by reducing losses and improving the ability to collect solar energy. In the summer months, the reduced losses would decrease the amount of unwanted heat gain to the space. More solar energy could be collected by reducing the average temperature of the fluid going to the STCs over the course of a year.

The investigations completed in this work considered a series STC arrangement. A parallel configuration should be considered in comparison to investigate the impact on overall energy savings. In addition, unglazed STC should be compared to the glazed collectors used for this project to evaluate opportunities for cost reduction. An SAHP system with unglazed collectors may be more easily justified for warmer climates, where a large price decrease can be made by accepting a slight performance decrease. Using unglazed collectors also provides the opportunity for them to be used as an air source heat exchanger for the heat pump, since losses, and therefore gains, exchanged with the ambient air are increased.

Many further options of investigation exist due to the flexible nature of the dual tank SAHP system. Several recommendations have been made for further investigation through this work by experimentally tuning the numerical model and investigating important design parameters. The potential capabilities of a dual tank SAHP system make it an excellent candidate for continued investigation.

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Appendix A – Test Apparatus

Figure A.1 – Partial view of experimental apparatus before pipe insulation

Figure A.2 – DHW tank (left) and float tank (right) with thermocouple instrumentation (centre)

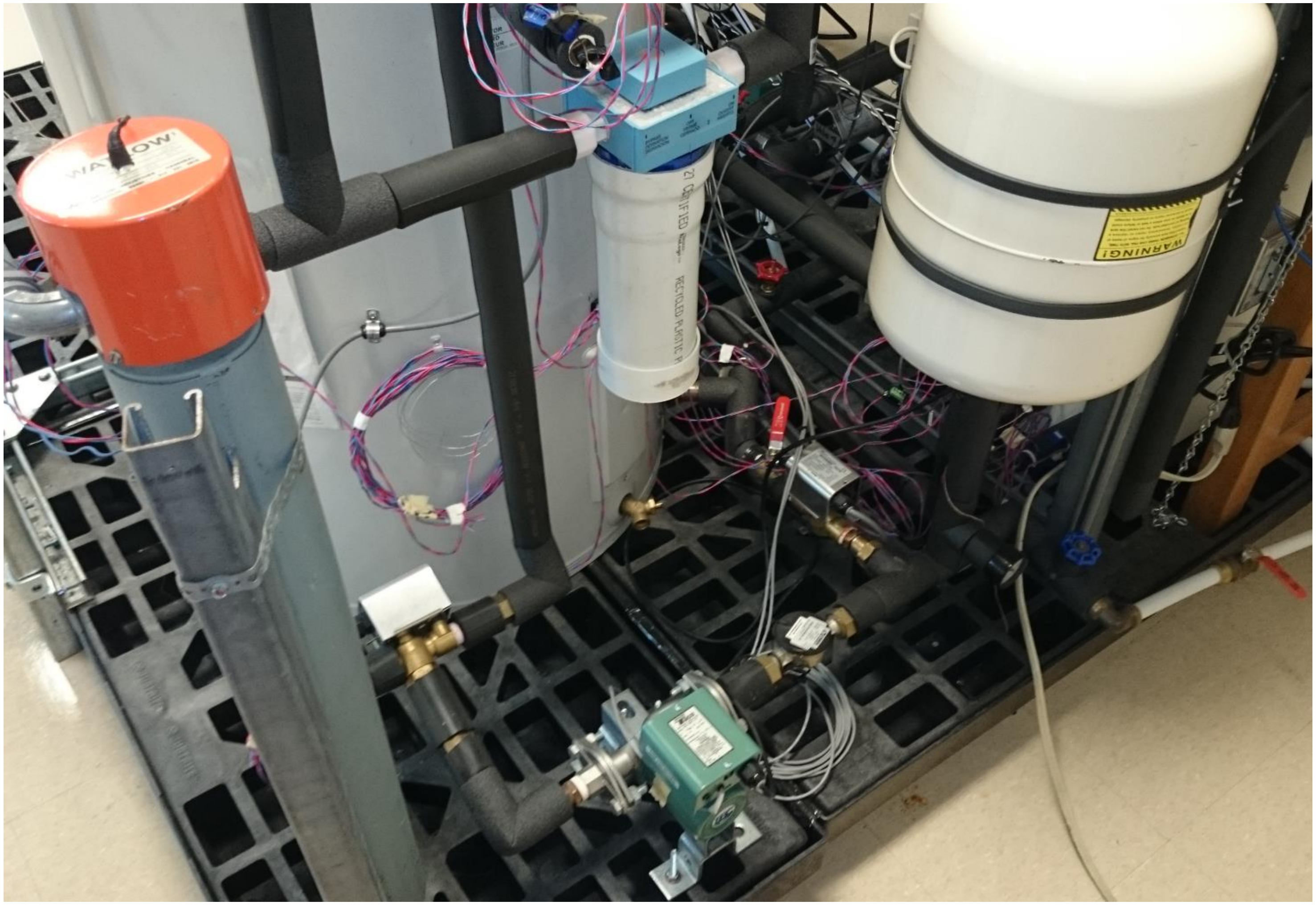

Figure A.3 – Electrical resistance heater (left), expansion tank (right), source-side hydronic pump (bottom), water filter (top)

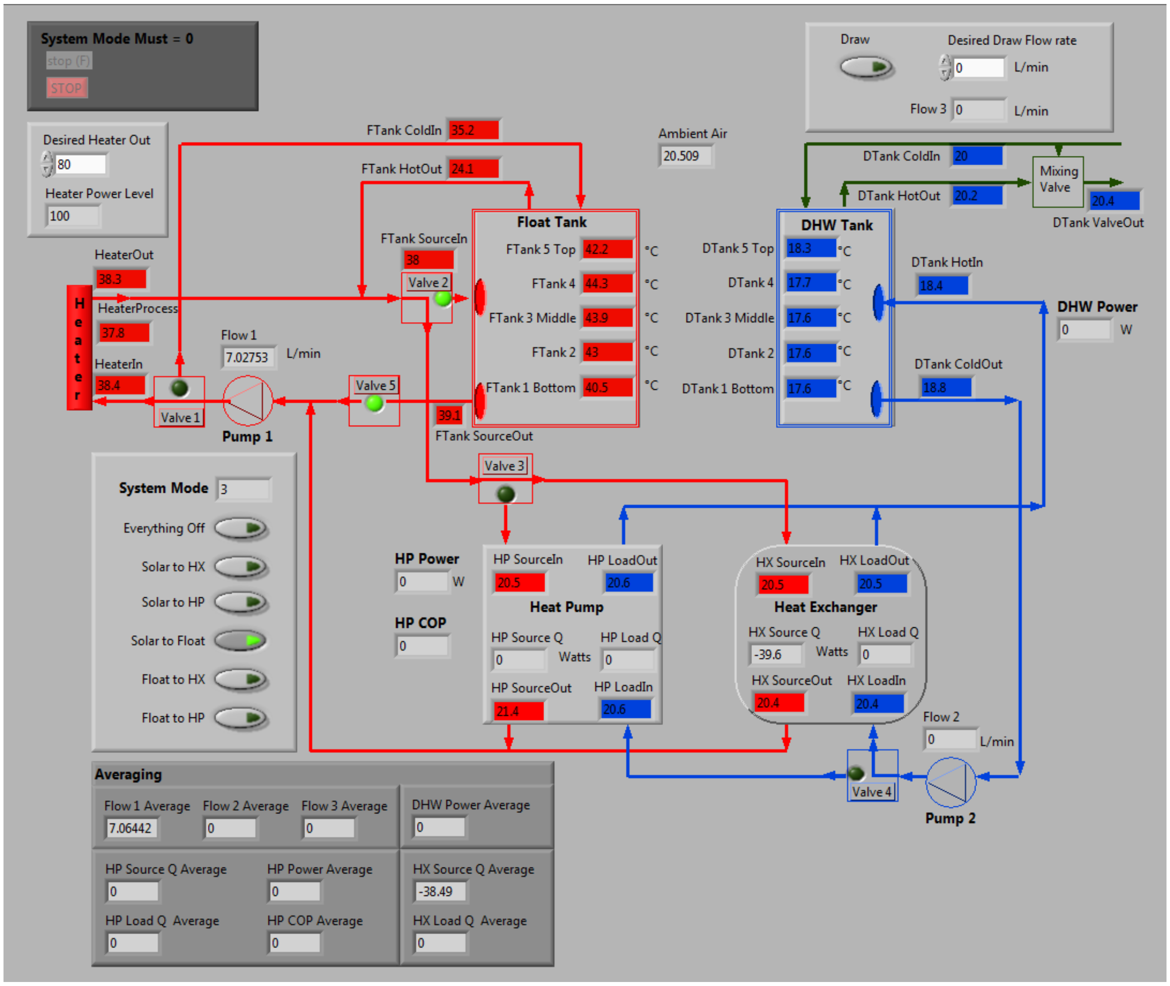

Figure A.4 – LabVIEW control screen for experimental apparatus

Appendix B – Model Component Parameters/Inputs

The following tables specify the parameters and input values used for the TRNSYS components in the model. Note that in TRNSYS parameters are constant during a simulation and inputs remain constant at their initial values if they are not connected to any other components. In other words, inputs that are specified in the following sections act as parameters in the TRSNSYS simulations.

Custom Controller

Table B.1 – Parameters/Inputs for custom controller

Parameter/Input	Value	Units
Solar-DHW db upper	5	deltaC
Solar-DHW db lower	2	deltaC
Solar-Float db upper	5	deltaC
Solar-Float db lower	2	deltaC
DHW setpoint db	5	deltaC
# of oscillations	5	-
Short-cycle time	5	Min
HX flow rate	0.15	-
HP flow rate	1	-
OFF to HP threshold	1000	kJ/hr·m^2
HP to OFF threshold	1000	kJ/hr·m^2

Parameter/Input	Value	Units
HX to HP threshold	1800	kJ/hr·m^2
HP to HX threshold	1800	kJ/hr·m^2
Fluid specific heat	4.19	kJ/kg·K
DHW setpoint	65	C

Solar Thermal Collector (Type 1b)

Table B.2 – Parameters/Inputs for solar thermal collector

Parameter/Input	Value	Units
Collector area	7.482	m^2
Fluid specific heat	4.190	kJ/kg·K
Efficiency mode	1	-
Tested flow rate	71.21	$kg/hr \cdot m^2$
Intercept efficiency	0.769	-
Efficiency slope	13.0104	$kJ/hr \cdot m^2 \cdot K$
Efficiency curvature	0.04889	$kJ/hr \cdot m^2 \cdot K^2$
Optical mode	2	-
1st-order IAM	0.32	-
2nd order IAM	0.0	-
Collector slope	45	degrees

Heat Exchanger (Type 5b)

Table B.3 – Parameters/Inputs for heat exchanger

Parameter/Input	Value	Units
Counter flow mode	2	-
Specific heat of hot side fluid	4.19	kJ/kg·K
Specific heat of cold side fluid	4.19	kJ/kg·K
Overall heat transfer coefficient of exchanger	150	W/K

Heat Pump (Custom Type)

Table B.4 – Parameters/Inputs for heat pump

Parameter/Input	Value	Units
ScaleHP	0.6	-
CpSource	4.19	kJ/kg·K
CpLoad	4.19	kJ/kg·K

DHW Tank (Type 4b)

Table B.5 – Parameters/Inputs for DHW tank

Parameter/Input	Value	Units
Fixed inlet positions	1	-
Tank volume	0.3028	m^3
Fluid specific heat	4.19	kJ/kg·K
Fluid density	1000	kg/m^3
Tank loss coefficient (-)	-3.0	$kJ/hr \cdot m^2 \cdot K$
Height of node-1,2,3,4,5	0.3	m
Auxiliary heater mode	2	-
Node containing heating element 1	2	-
Node containing thermostat 1	2	-
Set point temperature for element 1	60	C
Deadband for heating element 1	5	deltaC
Maximum heating rate of element 1	12,150	kJ/hr
Incremental loss coefficient for node-1,2,3,4,5	0	$kJ/hr \cdot m^2 \cdot K$
Environment temperature	20	C
Control signal for element-1	1	-
Control signal for element 2	0	-

Float Tank (Type 4b)

Table B.6 – Parameters/Inputs for float tank

Parameter/Input	Value	Units
Fixed inlet positions	1	-
Tank volume	0.4504	m^3
Fluid specific heat	4.19	kJ/kg·K
Fluid density	1000	kg/m^3
Tank loss coefficient (-)	-3.0	$kJ/hr \cdot m^2 \cdot K$
Height of node-1,2,3,4,5	0.3	m
Incremental loss coefficient for node-1,2,3,4,5	0	$kJ/hr \cdot m^2 \cdot K$
Environment temperature	20	C
Control signal for element-1	0	-
Control signal for element 2	0	-

Pump (Type 110)

Table B.7 – Parameters/Inputs for pump

Parameter/Input	Value	Units
Rated flow rate	420	kg/hr
Fluid specific heat	4.19	kJ/kg·K
Rated power	0	kJ/hr
Motor heat loss fraction	0	-
Number of power coefficients	1	-
Power coefficient	0	kJ/hr
Total pump efficiency	0.6	-
Motor efficiency	0.9	-

Tempering Valve (Type 11b)

Table B.8 – Parameters/Inputs for tempering valve

Parameter/Input	Value	Units
Tempering valve mode	4	-
Nb. of oscillations allowed	7	-
Set point temperature	55	C

Appendix C – Problems Encountered

TRNSYS is generally well-behaved and fairly easy to use. There are, however, subtle quirks that exist in the simulation software and some component models. A limitation that can cause significant discrepancies between intent and simulation is that the forcing function component can only have one value during a given timestep. TRNSYS is unable to calculate a weighted average of the forcing function value over the duration of a timestep. Therefore, the simulation time step must be a multiple of and less than or equal to the forcing function resolution. Otherwise the forcing function will not behave as intended.

For the simulations completed, it was desirable to reduce the computation time for simulating a full year, which required increasing the simulation time step. The forcing function had to be adjusted accordingly to draw the correct amount of water from the simulated system. All flow rates were divided by 10 and all flow durations were multiplied by 10 to enable an order of magnitude reduction in computational time. This process enabled the simulation time for a year to be reduced by an order of magnitude to around 2 minutes. This time reduction was of great practical benefit because it allowed parametric studies to be performed in a reasonable timeframe. Accuracy was not impacted significantly, primarily because the amount of energy removed from the system via a water draw remained unchanged.

Appendix D – Custom Component Source Code

SAHP Controller

```
                      SUBROUTINE TYPE153 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C*************************************************************************
C Object: SAHP Controller
C Simulation Studio Model: Type153
C
C Author: Carsen Banister
C Editor:
C Date: May 23, 2012 last modified: June 11, 2013
C
C
C ***
C *** Model Parameters
C ***
C                     Solar-DHW db upper    deltaC [-Inf;+Inf]
C                     Solar-DHW db lower    deltaC [-Inf;+Inf]
C                     Solar-Float db upper  deltaC [-Inf;+Inf]
C                     Solar-Float db lower  deltaC [-Inf;+Inf]
C                     DHW setpoint db              deltaC [-Inf;+Inf]
C                     # of oscillations     - [0;+Inf]
C                     Short-cycle time  - [0;+Inf]
C                     HX Flow Rate  - [0;1]
C                     HP Flow Rate   - [0;1]
C                     OFF to HP threshold kJ/hr.m2 [0;+Inf]
C                     HP to OFF threshold  kJ/hr.m2 [0;+Inf]
C                     HX to HP threshold  kJ/hr.m2 [0;+Inf]
C                     HP to HX threshold  kJ/hr.m2 [0;+Inf]
C                     Fluid specific heat kJ/kg.K [0;+Inf]

C ***
C *** Model Inputs
C ***
C                     DHW source    C [-Inf;+Inf]
C                     DHW load      C [-Inf;+Inf]
C                     DHW setpoint  C [-Inf;+Inf]
C                     Float source  C [-Inf;+Inf]
C                     Float load    C [-Inf;+Inf]
C                     Float switch-over     C [-Inf;+Inf]
C                     Solar out  C [-Inf;+Inf]
C                     Solar in  C [-Inf;+Inf]
C                     Ambient C [-Inf;+Inf]
C                     Allow Float-DHW - [0;1]
C                     Allow Solar-HP-DHW - [0;1]
C                     Allow Float-HP-DHW - [0;1]
C                     Allow Solar-Float - [0;1]
C                     Radiation tilted - [0;+Inf]

C ***
C *** Model Outputs
C ***
C                     System Mode - [0;10]
C                     Div 1 - [0;1]
C                     Div 2 - [0;1]
C                     Div 3 - [0;1]
C                     Div 4 - [0;1]
C                     Pump 1 - [0;1]
C                     Pump 2 - [0;1]
C                     Heat Pump - [0;1]
```

```
C                     DHW element - [0;1]
C                     Mode 1 - [0;1]
C                     Mode 2 - [0;1]
C                     Mode 3 - [0;1]
C                     Mode 4 - [0;1]
C                     Mode 5 - [0;1]
C ***
C *** Model Derivatives
C ***

C (Comments and routine interface generated by TRNSYS Studio)
C************************************************************************

C    TRNSYS access functions (allow to access TIME etc.)
      USE TrnsysConstants
      USE TrnsysFunctions

C-----------------------------------------------------------------------------------------------------
-------------------
C    REQUIRED BY THE MULTI-DLL VERSION OF TRNSYS
      !DEC$ATTRIBUTES DLLEXPORT :: TYPE153                       !SET THE CORRECT TYPE NUMBER HERE
C-----------------------------------------------------------------------------------------------------
-------------------
C-----------------------------------------------------------------------------------------------------
-------------------
C    TRNSYS DECLARATIONS
      IMPLICIT NONE                 !REQUIRES THE USER TO DEFINE ALL VARIABLES BEFORE USING THEM

       DOUBLE PRECISION XIN  !THE ARRAY FROM WHICH THE INPUTS TO THIS TYPE WILL BE RETRIEVED
       DOUBLE PRECISION OUT  !THE ARRAY WHICH WILL BE USED TO STORE THE OUTPUTS FROM THIS TYPE
       DOUBLE PRECISION TIME !THE CURRENT SIMULATION TIME - YOU MAY USE THIS VARIABLE BUT DO NOT SET
IT!
       DOUBLE PRECISION PAR  !THE ARRAY FROM WHICH THE PARAMETERS FOR THIS TYPE WILL BE RETRIEVED
       DOUBLE PRECISION STORED !THE STORAGE ARRAY FOR HOLDING VARIABLES FROM TIMESTEP TO TIMESTEP
       DOUBLE PRECISION T           !AN ARRAY CONTAINING THE RESULTS FROM THE DIFFERENTIAL EQUATION
SOLVER
       DOUBLE PRECISION DTDT !AN ARRAY CONTAINING THE DERIVATIVES TO BE PASSED TO THE DIFF.EQ. SOLVER
       INTEGER*4 INFO(15)           !THE INFO ARRAY STORES AND PASSES VALUABLE INFORMATION TO AND
FROM THIS TYPE
       INTEGER*4 NP,NI,NOUT,ND      !VARIABLES FOR THE MAXIMUM NUMBER OF PARAMETERS,INPUTS,OUTPUTS
AND DERIVATIVES
       INTEGER*4 NPAR,NIN,NDER      !VARIABLES FOR THE CORRECT NUMBER OF PARAMETERS,INPUTS,OUTPUTS
AND DERIVATIVES
       INTEGER*4 IUNIT,ITYPE !THE UNIT NUMBER AND TYPE NUMBER FOR THIS COMPONENT
       INTEGER*4 ICNTRL             !AN ARRAY FOR HOLDING VALUES OF CONTROL FUNCTIONS WITH THE NEW
SOLVER
       INTEGER*4 NSTORED            !THE NUMBER OF VARIABLES THAT WILL BE PASSED INTO AND OUT OF
STORAGE
       CHARACTER*3 OCHECK           !AN ARRAY TO BE FILLED WITH THE CORRECT VARIABLE TYPES FOR THE
OUTPUTS
       CHARACTER*3 YCHECK           !AN ARRAY TO BE FILLED WITH THE CORRECT VARIABLE TYPES FOR THE
INPUTS
C-----------------------------------------------------------------------------------------------------
-------------------

C-----------------------------------------------------------------------------------------------------
-------------------
C    USER DECLARATIONS - SET THE MAXIMUM NUMBER OF PARAMETERS (NP), INPUTS (NI),
C    OUTPUTS (NOUT), AND DERIVATIVES (ND) THAT MAY BE SUPPLIED FOR THIS TYPE
      PARAMETER (NP=14,NI=14,NOUT=14,ND=0,NSTORED=5)
```

```
C-----------------------------------------------------------------------------------------------------
-------------------

C-----------------------------------------------------------------------------------------------------
-------------------
C    REQUIRED TRNSYS DIMENSIONS
      DIMENSION XIN(NI),OUT(NOUT),PAR(NP),YCHECK(NI),OCHECK(NOUT),
       1   STORED(NSTORED),T(ND),DTDT(ND)
      INTEGER NITEMS
C-----------------------------------------------------------------------------------------------------
-------------------
C-----------------------------------------------------------------------------------------------------
-------------------
C    ADD DECLARATIONS AND DEFINITIONS FOR THE USER-VARIABLES HERE

C    PARAMETERS
    DOUBLE PRECISION Solar_DHW_db_upper
    DOUBLE PRECISION Solar_DHW_db_lower
    DOUBLE PRECISION Solar_Float_db_upper
    DOUBLE PRECISION Solar_Float_db_lower
    DOUBLE PRECISION DHW_setpoint_db
       DOUBLE PRECISION No_of_oscillations
       DOUBLE PRECISION Short_cycle_time
       DOUBLE PRECISION HX_flow_rate
       DOUBLE PRECISION HP_flow_rate
       DOUBLE PRECISION OFF_to_HP_threshold
       DOUBLE PRECISION HP_to_OFF_threshold
       DOUBLE PRECISION HX_to_HP_threshold
       DOUBLE PRECISION HP_to_HX_threshold
       DOUBLE PRECISION Fluid_specific_heat

C    INPUTS
    DOUBLE PRECISION DHW_source
    DOUBLE PRECISION DHW_load
       DOUBLE PRECISION DHW_setpoint
    DOUBLE PRECISION Float_source
    DOUBLE PRECISION Float_load
    DOUBLE PRECISION Float_switch_over
       DOUBLE PRECISION Solar_in
       DOUBLE PRECISION Solar_out
    DOUBLE PRECISION Ambient
       INTEGER Allow_F_D
       INTEGER Allow_S_H_D
       INTEGER Allow_F_H_D
       INTEGER Allow_S_F
       DOUBLE PRECISION Radiation_tilted

C    OUTPUTS
       DOUBLE PRECISION System_Mode
       DOUBLE PRECISION Div_1
       DOUBLE PRECISION Div_2
       DOUBLE PRECISION Div_3
       DOUBLE PRECISION Div_4
       DOUBLE PRECISION Pump_1
       DOUBLE PRECISION Pump_2
       DOUBLE PRECISION Heat_Pump
       INTEGER DHW_element
       INTEGER Mode1
       INTEGER Mode2
       INTEGER Mode3
       INTEGER Mode4
```

```
       INTEGER Mode5

C    LOCAL VARIABLES
       INTEGER Stick_Count
       LOGICAL Mode_1 !.TRUE. if the mode can run, .FALSE. if the mode cannot run
       LOGICAL Mode_2
       LOGICAL Mode_3
       LOGICAL Mode_4
       LOGICAL Mode_5
       LOGICAL DHW_demand

C-----------------------------------------------------------------------------------------------------
-------------------
C       READ IN THE VALUES OF THE PARAMETERS IN SEQUENTIAL ORDER
    Solar_DHW_db_upper=PAR(1)
    Solar_DHW_db_lower=PAR(2)
    Solar_Float_db_upper=PAR(3)
    Solar_Float_db_lower=PAR(4)
       DHW_setpoint_db=PAR(5)
    No_of_oscillations=PAR(6)
       Short_cycle_time=PAR(7)
       HX_flow_rate=PAR(8)
       HP_flow_rate=PAR(9)
       OFF_to_HP_threshold=PAR(10)
       HP_to_OFF_threshold=PAR(11)
       HX_to_HP_threshold=PAR(12)
       HP_to_HX_threshold=PAR(13)
       Fluid_specific_heat=PAR(14)

C-----------------------------------------------------------------------------------------------------
-------------------
C    RETRIEVE THE CURRENT VALUES OF THE INPUTS TO THIS MODEL FROM THE XIN ARRAY IN SEQUENTIAL ORDER

    DHW_source=XIN(1)
    DHW_load=XIN(2)
       DHW_setpoint=XIN(3)
    Float_source=XIN(4)
    Float_load=XIN(5)
    Float_switch_over=XIN(6)
       Solar_out=XIN(7)
       Solar_in=XIN(8)
    Ambient=XIN(9)
       Allow_F_D=XIN(10)
       Allow_S_H_D=XIN(11)
       Allow_F_H_D=XIN(12)
       Allow_S_F=XIN(13)
       Radiation_tilted=XIN(14)
    IUNIT=INFO(1)
       ITYPE=INFO(2)

C-----------------------------------------------------------------------------------------------------
-------------------
C    SET THE VERSION INFORMATION FOR TRNSYS
      IF(INFO(7).EQ.-2) THEN
          INFO(12)=16
          RETURN 1
       ENDIF
C-----------------------------------------------------------------------------------------------------
-------------------
```

```
C-----------------------------------------------------------------------------------------------
-------------------
C    DO ALL THE VERY LAST CALL OF THE SIMULATION MANIPULATIONS HERE
      IF (INFO(8).EQ.-1) THEN
          RETURN 1
       ENDIF
C-----------------------------------------------------------------------------------------------
-------------------

C-----------------------------------------------------------------------------------------------
-------------------
C    PERFORM ANY 'AFTER-ITERATION' MANIPULATIONS THAT ARE REQUIRED HERE (end of timestep)
C    e.g. save variables to storage array for the next timestep
      IF (INFO(13).GT.0) THEN
          NITEMS=5
              IF ((System_mode.GT.STORED(1)).OR.((System_mode.EQ.0).AND.
       1       (STORED(1).NE.6).AND.(STORED(1).NE.0))) THEN !System has shut off after running
                     STORED(2)=TIME !reset short-cycle counter
              END IF
           STORED(1)=System_mode !keeps track of System_mode for next iteration at this time step

!             STORED(5) keeps track of DHW demand. 1 if DHW previously needed heat, 0 if not
              IF (DHW_demand) THEN
                     STORED(5)=1
              ELSE
                     STORED(5)=0
              END IF

              CALL setStorageVars(STORED,NITEMS,INFO)
          RETURN 1
       ENDIF
C
C-----------------------------------------------------------------------------------------------
-------------------

C-----------------------------------------------------------------------------------------------
-------------------
C    DO ALL THE VERY FIRST CALL OF THE SIMULATION MANIPULATIONS HERE
       IF (INFO(7).EQ.-1) THEN

C       SET SOME INFO ARRAY VARIABLES TO TELL THE TRNSYS ENGINE HOW THIS TYPE IS TO WORK
        INFO(6)=NOUT
        INFO(9)=1
           INFO(10)=0 !STORAGE FOR VERSION 16 HAS BEEN CHANGED

C       SET THE REQUIRED NUMBER OF INPUTS, PARAMETERS AND DERIVATIVES THAT THE USER SHOULD SUPPLY IN
THE INPUT FILE
C       IN SOME CASES, THE NUMBER OF VARIABLES MAY DEPEND ON THE VALUE OF PARAMETERS TO THIS
MODEL....
        NIN=NI
           NPAR=NP
           NDER=ND

C       CALL THE TYPE CHECK SUBROUTINE TO COMPARE WHAT THIS COMPONENT REQUIRES TO WHAT IS SUPPLIED IN
C       THE TRNSYS INPUT FILE
           CALL TYPECK(1,INFO,NIN,NPAR,NDER)

C       SET THE NUMBER OF STORAGE SPOTS NEEDED FOR THIS COMPONENT
        NITEMS=5
           CALL setStorageSize(NITEMS,INFO)
```

```
C       RETURN TO THE CALLING PROGRAM
        RETURN 1

    ENDIF
C-----------------------------------------------------------------------------------------------------
-------------------

C-----------------------------------------------------------------------------------------------------
-------------------
C    DO ALL OF THE INITIAL TIMESTEP MANIPULATIONS HERE - THERE ARE NO ITERATIONS AT THE INITIAL TIME
              IF (TIME .LT. (getSimulationStartTime() +
              . getSimulationTimeStep()/2.D0)) THEN

C       SET THE UNIT NUMBER FOR FUTURE CALLS
                     IUNIT=INFO(1)
                     ITYPE=INFO(2)

C   CHECK THE PARAMETERS FOR PROBLEMS AND RETURN FROM THE SUBROUTINE IF AN ERROR IS FOUND
C   IF(...) CALL TYPECK(-4,INFO,0,"BAD PARAMETER #",0)

C   PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL VALUES OF THE OUTPUTS HERE
C      System Mode - [0;10]
                     OUT(1)=0
C      Div 1 - [0;1]
                     OUT(2)=0
C      Div 2 - [0;1]
                     OUT(3)=0
C      Div 3 - [0;1]
                     OUT(4)=0
C      Div 4 - [0;1]
                     OUT(5)=0
C      Pump 1 - [0;1]
                     OUT(6)=0
C      Pump 2 - [0;1]
                     OUT(7)=0
C      Heat Pump - [0;1]
                     OUT(8)=0
C      DHW element - [0;1]
                     OUT(9)=0

C       PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL STORAGE VARIABLES HERE
                     NITEMS=5
                     STORED(1)=0 !system mode at last time step
                     STORED(2)=getSimulationStartTime() !time system last shut off
                     STORED(3)=0 !Stick_Count
                     STORED(4)=0 !system mode at last iteration
                     STORED(5)=0 !DHW demand at last iteration (1 - yes, 0 - no)

C       PUT THE STORED ARRAY IN THE GLOBAL STORED ARRAY
                     CALL setStorageVars(STORED,NITEMS,INFO)

C       RETURN TO THE CALLING PROGRAM
                     RETURN 1

              ENDIF
C-----------------------------------------------------------------------------------------------------
-------------------

C-----------------------------------------------------------------------------------------------------
-------------------
```

```
C    *** ITS AN ITERATIVE CALL TO THIS COMPONENT ***
C-----------------------------------------------------------------------------------------
-------------------

C-----------------------------------------------------------------------------------------
-------------------
C    RETRIEVE THE VALUES IN THE STORAGE ARRAY FOR THIS ITERATION
              NITEMS=5
              CALL getStorageVars(STORED,NITEMS,INFO)
              Stick_Count = STORED(3)
C
C-----------------------------------------------------------------------------------------
-------------------
C-----------------------------------------------------------------------------------------
-------------------
C    CHECK THE INPUTS FOR PROBLEMS
C      IF(...) CALL TYPECK(-3,INFO,'BAD INPUT #',0,0)
C      IF(IERROR.GT.0) RETURN 1
C-----------------------------------------------------------------------------------------
-------------------
C-----------------------------------------------------------------------------------------
-------------------
C    *** PERFORM ALL THE CALCULATION HERE FOR THIS MODEL. ***
C-----------------------------------------------------------------------------------------
-------------------

C             ADD YOUR COMPONENT EQUATIONS HERE; BASICALLY THE EQUATIONS THAT WILL
C             CALCULATE THE OUTPUTS BASED ON THE PARAMETERS AND THE INPUTS.    REFER TO
C             CHAPTER 3 OF THE TRNSYS VOLUME 1 MANUAL FOR DETAILED INFORMATION ON
C             WRITING TRNSYS COMPONENTS.

              IF(INFO(7).EQ.0) Stick_Count=0 !reset stick counter at 1st call of timestep
              IF(Stick_count.GE.No_of_oscillations) GOTO 1000 !stick control if max. no. of
oscillations reached

              Mode1=.FALSE.
              Mode2=.FALSE.
              Mode3=.FALSE.
              Mode4=.FALSE.
              Mode5=.FALSE.

C***   FIRST, CHECK WHICH MODES CAN POSSIBLY RUN

C       1. SOLAR-DHW

100           IF (STORED(1).EQ.1) GOTO 110 !if last system mode was 1 check lower deadband

              !SOLAR-DHW was not previously running
              IF ((Solar_out-DHW_source).GT.Solar_DHW_db_upper) THEN
                     Mode1=.TRUE. !SOLAR-DHW can run
              END IF
              GOTO 200

110           IF ((Solar_out-DHW_source).GT.Solar_DHW_db_lower) THEN
                     Mode1=.TRUE. !SOLAR-DHW can continue running
              ELSE
                     Mode1=.FALSE.!SOLAR-DHW cannot continue running
              END IF

C       2. FLOAT-DHW
200           IF (Allow_F_D.EQ.1) THEN
```

```
                    IF (STORED(1).EQ.2) GOTO 210 !if last system mode was 2 check lower deadband

                    !FLOAT-DHW was not previously running
                    IF ((Float_load-DHW_source).GT.Solar_DHW_db_upper) THEN
                           Mode2=.TRUE. !FLOAT-DHW can run
                    END IF

210                 IF ((Float_load-DHW_source).GT.Solar_DHW_db_lower) THEN
                           Mode2=.TRUE. !FLOAT-DHW can continue running
                    ELSE
                           Mode2=.FALSE. !FLOAT-DHW cannot continue running
                    END IF
             END IF

C       3. SOLAR-HP-DHW
300          IF (Allow_S_H_D.EQ.1) THEN
                    IF (Radiation_tilted.GE.OFF_to_HP_threshold) THEN !SOLAR-HP can run
                           Mode3=.TRUE.
                    ELSE !SOLAR-HP cannot run
                           Mode3=.FALSE.
                    END IF

                    IF (Radiation_tilted.GE.HP_to_HX_threshold) THEN !SOLAR-HX should run instead
                           Mode3=.FALSE.
                    END IF
             END IF

C       4. FLOAT-HP-DHW
400          IF (Allow_F_H_D.EQ.1) THEN
                    IF (STORED(1).EQ.4) THEN !F-H-D was running during previous time step
                           IF (Float_source.GT.5) Mode4=.TRUE. !only allow FLOAT to continue if
above 5 degC
                    ELSE
                           IF (Float_source.GT.10) Mode4=.TRUE. !only allow FLOAT to start if above
10 degC
                    END IF
             END IF

C       5. SOLAR-FLOAT
500          IF (Allow_S_F.EQ.1) THEN
                    IF ((STORED(1).EQ.5).OR.(STORED(1).EQ.1)) GOTO 510 !if SOLAR-DHW or SOLAR-FLOAT
was last system mode, check lower deadband

                    IF ((Solar_out-Float_source).GT.Solar_Float_db_upper) THEN
                           Mode5=.TRUE. !SOLAR-FLOAT can run
                    END IF
                    GOTO 600

510                 IF ((Solar_out-Float_source).GT.Solar_Float_db_lower) THEN
                           Mode5=.TRUE. !SOLAR-FLOAT can continue running
                    ELSE
                           Mode5=.FALSE. !SOLAR-FLOAT cannot continue running
                    END IF
             END IF

C       6. DHW ELEMENT
600          System_mode=System_mode

900          IF ((STORED(5)).GT.(0.5)) THEN
                    GOTO 920
             END IF
```

```
C***    NEXT, CHECK IF DHW NEEDS HEAT

C       CHECK IF DHW NEEDS HEAT (PREVIOUSLY DIDN'T)
910             IF ((DHW_setpoint-DHW_load).GE.(DHW_setpoint_db)) THEN !needs heat
                        DHW_demand=.TRUE.
                END IF
                GOTO 1000

C       CHECK IF DHW STILL NEEDS HEAT (PREVIOUSLY DID)
920             IF ((DHW_load).GE.(DHW_setpoint)) THEN !no longer needs heat
                        DHW_demand=.FALSE.
                ELSE !still needs heat
                        DHW_demand=.TRUE.
                END IF

C***    FINALLY, DETERMINE WHICH SYSTEM MODE TO RUN
1000    IF (DHW_demand) THEN
                        IF (Mode3) THEN
                                System_mode=3
                        ELSE IF (Mode1) THEN
                                System_mode=1
                        ELSE IF (Mode2) THEN !FLOAT-DHW
                                System_mode=2
                        ELSE IF (Mode4) THEN !FLOAT-HP-DHW
                                System_mode=4
                        ELSE
                                IF (Mode5) THEN
                                        System_mode=5
                                ELSE
                                        System_mode=0
                                END IF
                        END IF

                ELSE !no DHW demand
                        IF (Mode5) THEN
                                System_mode=5
                        ELSE
                                System_mode=0 !run nothing

                        END IF

                END IF

2000OUT(8)=TIME-STORED(2)
        IF (System_mode.NE.STORED(4)) Stick_Count=Stick_Count+1

C       SHORT-CYCLING PROTECTION
                IF ((TIME-STORED(2)).LT.(Short_cycle_time/3600)) THEN !short-cycle time has not passed
                        IF ((System_mode.NE.0).AND.(System_mode.NE.6)) THEN !if system mode is 1,2,3,4,5
(anything but 0 or 6)
                                IF (.NOT.DHW_demand) THEN
                                        System_mode=Stored(1)
                                        GOTO 3000
                                END IF
                                IF ((System_mode.LT.Stored(1)).OR.(Stored(1).EQ.0)) THEN
                                        System_mode=STORED(1) !stay in previous state
                                END IF
                        END IF
                END IF
```

```
C       OUTPUT MODES
3000    IF (Mode1) THEN
                        Mode_1=1
                ELSE
                        Mode_1=0
                END IF

                IF (Mode2) THEN
                        Mode_2=2
                ELSE
                        Mode_2=0
                END IF

                IF (Mode3) THEN
                        Mode_3=3
                ELSE
                        Mode_3=0
                END IF

                IF (Mode4) THEN
                        Mode_4=4
                ELSE
                        Mode_4=0
                END IF

                IF (Mode5) THEN
                        Mode_5=5
                ELSE
                        Mode_5=0
                END IF

                IF ((System_mode.NE.0).AND.(System_mode.NE.6)) THEN
                        !OUT(13)=(Solar_out-Solar_in)*Fluid_specific_heat*7/60
                ELSE
                        !OUT(13)=0
                END IF

C-----------------------------------------------------------------------------------------------------
-------------------

C-----------------------------------------------------------------------------------------------------
-------------------
C-----------------------------------------------------------------------------------------------------
-------------------
C    SET THE STORAGE ARRAY AT THE END OF THIS ITERATION IF NECESSARY
      NITEMS=4
C      STORED(1)=
       STORED(3)=Stick_Count
       STORED(4)=System_mode
       CALL setStorageVars(STORED,NITEMS,INFO)
C-----------------------------------------------------------------------------------------------------
-------------------
C-----------------------------------------------------------------------------------------------------
-------------------
C    REPORT ANY PROBLEMS THAT HAVE BEEN FOUND USING CALLS LIKE THIS:
C      CALL MESSAGES(-1,'put your message here','MESSAGE',IUNIT,ITYPE)
C      CALL MESSAGES(-1,'put your message here','WARNING',IUNIT,ITYPE)
C      CALL MESSAGES(-1,'put your message here','SEVERE',IUNIT,ITYPE)
C      CALL MESSAGES(-1,'put your message here','FATAL',IUNIT,ITYPE)
C-----------------------------------------------------------------------------------------------------
-------------------
```

```
C-----------------------------------------------------------------------------------------------------
-------------------
C    SET THE OUTPUTS FROM THIS MODEL IN SEQUENTIAL ORDER AND GET OUT

       IF (System_mode.EQ.0) THEN !MODE 0 - OFF
                     Div_1=0
                     Div_2=0
                     Div_3=0
                     Div_4=0
                     Pump_1=0
                     Pump_2=0
                     Heat_Pump=0
                     DHW_element=0
       END IF
C
              IF (System_Mode.EQ.1) THEN !MODE 1 - SOLAR-DHW
                     Div_1=0
                     Div_2=0
                     Div_3=0
                     Div_4=1
                     Pump_1=HX_flow_rate
                     Pump_2=HX_flow_rate
                     Heat_Pump=0
                     DHW_element=0
              END IF


              IF (System_Mode.EQ.2) THEN !MODE 2 - FLOAT-DHW
                     Div_1=0
                     Div_2=0
                     Div_3=0
                     Div_4=0
                     Pump_1=HX_flow_rate
                     Pump_2=HX_flow_rate
                     Heat_Pump=0
                     DHW_element=0
              END IF


              IF (System_Mode.EQ.3) THEN !MODE 3 - SOLAR-HP-DHW
                     Div_1=0
                     Div_2=1
                     Div_3=1
                     Div_4=1
                     Pump_1=HP_flow_rate
                     Pump_2=HP_flow_rate
                     Heat_Pump=1
                     DHW_element=0
              END IF

              IF (System_Mode.EQ.4) THEN !MODE 4 - FLOAT-HP-DHW
                     Div_1=0
                     Div_2=1
                     Div_3=1
                     Div_4=0
                     Pump_1=HP_flow_rate
                     Pump_2=HP_flow_rate
                     Heat_Pump=1
                     DHW_element=0
              END IF
```

```
               IF (System_Mode.EQ.5) THEN !MODE 5 - SOLAR-FLOAT
                      Div_1=1
                      Div_2=0
                      Div_3=0
                      Div_4=1
                      Pump_1=HX_flow_rate
                      Pump_2=0
                      Heat_Pump=0
                      DHW_element=0
               END IF

               IF (System_Mode.EQ.6) THEN !MODE 6 - DHW AUX HEATER
                      Div_1=0
                      Div_2=0
                      Div_3=0
                      Div_4=0
                      Pump_1=0
                      Pump_2=0
                      Heat_Pump=0
                      DHW_element=1
               END IF

C                     System Mode - [0;10]
                      OUT(1)=System_Mode
C                     Div 1 - [0;1]
                      OUT(2)=Div_1
C                     Div 2 - [0;1]
                      OUT(3)=Div_2
C                     Div 3 - [0;1]
                      OUT(4)=Div_3
C                     Div 4 - [0;1]
                      OUT(5)=Div_4
C                     Pump 1 - [0;1]
                      OUT(6)=Pump_1
C                     Pump 2 - [0;1]
                      OUT(7)=Pump_2
C                     Heat Pump - [0;1]
                      !OUT(8)=Heat_Pump
C                     DHW element - [0;1]
                      OUT(9)=DHW_element
C                     Mode 1 - [0;1]
                      OUT(10)=Mode_1
C                     Mode 2 - [0;1]
                      OUT(11)=Mode_2
C                     Mode 3 - [0;1]
                      OUT(12)=Mode_3
C                     Mode 4 - [0;1]
                      OUT(13)=Mode_4
C                     Mode 5 - [0;1]
                      OUT(14)=Mode_5
C-----------------------------------------------------------------------------------------------------
-------------------
C    EVERYTHING IS DONE - RETURN FROM THIS SUBROUTINE AND MOVE ON
      RETURN 1
      END
C-----------------------------------------------------------------------------------------------------------------
```

Time-of-Use Calendar and Calculator

```
                      SUBROUTINE TYPE155 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C*************************************************************************
C Object: Time-of-use
C Simulation Studio Model: Type155
C
C Author: Carsen Banister
C Editor:
C Date:  January 28, 2103 last modified: January 30, 2103
C
C
C ***
C *** Model Parameters
C ***
C                     Summer start   - [1;365]
C                     Winter start   - [1;365]
C                     Monday day number     - [1;7]

C ***
C *** Model Inputs
C ***
C                     Morning start  hr [0;24]
C                     Midday start   hr [0;24]
C                     Evening start  hr [0;24]
C                     Overnight start       hr [0;24]
C                     Load 1  - [-Inf;+Inf]
C                     Load 2  - [-Inf;+Inf]

C ***
C *** Model Outputs
C ***
C                     L1 Off-p usage - [-Inf;+Inf]
C                     L1 Mid-p usage - [-Inf;+Inf]
C                     L1 On-p usage  - [-Inf;+Inf]
C                     L2 Off-p usage - [-Inf;+Inf]
C                     L2 Mid-p usage - [-Inf;+Inf]
C                     L2 On-p usage  - [-Inf;+Inf]

C ***
C *** Model Derivatives
C ***

C (Comments and routine interface generated by TRNSYS Studio)
C*************************************************************************

C    TRNSYS acess functions (allow to acess TIME etc.)
      USE TrnsysConstants
      USE TrnsysFunctions

C-----------------------------------------------------------------------------------------------------
-------------------
C    REQUIRED BY THE MULTI-DLL VERSION OF TRNSYS
      !DEC$ATTRIBUTES DLLEXPORT :: TYPE155                       !SET THE CORRECT TYPE NUMBER HERE
C-----------------------------------------------------------------------------------------------------
-------------------
C-----------------------------------------------------------------------------------------------------
-------------------
C    TRNSYS DECLARATIONS
```

```
      IMPLICIT NONE                  !REQUIRES THE USER TO DEFINE ALL VARIABLES BEFORE USING THEM

       DOUBLE PRECISION XIN  !THE ARRAY FROM WHICH THE INPUTS TO THIS TYPE WILL BE RETRIEVED
       DOUBLE PRECISION OUT  !THE ARRAY WHICH WILL BE USED TO STORE THE OUTPUTS FROM THIS TYPE
       DOUBLE PRECISION TIME !THE CURRENT SIMULATION TIME - YOU MAY USE THIS VARIABLE BUT DO NOT SET
IT!
       DOUBLE PRECISION PAR  !THE ARRAY FROM WHICH THE PARAMETERS FOR THIS TYPE WILL BE RETRIEVED
       DOUBLE PRECISION STORED !THE STORAGE ARRAY FOR HOLDING VARIABLES FROM TIMESTEP TO TIMESTEP
       DOUBLE PRECISION T           !AN ARRAY CONTAINING THE RESULTS FROM THE DIFFERENTIAL EQUATION
SOLVER
       DOUBLE PRECISION DTDT !AN ARRAY CONTAINING THE DERIVATIVES TO BE PASSED TO THE DIFF.EQ. SOLVER
       INTEGER*4 INFO(15)           !THE INFO ARRAY STORES AND PASSES VALUABLE INFORMATION TO AND
FROM THIS TYPE
       INTEGER*4 NP,NI,NOUT,ND      !VARIABLES FOR THE MAXIMUM NUMBER OF PARAMETERS,INPUTS,OUTPUTS
AND DERIVATIVES
       INTEGER*4 NPAR,NIN,NDER      !VARIABLES FOR THE CORRECT NUMBER OF PARAMETERS,INPUTS,OUTPUTS
AND DERIVATIVES
       INTEGER*4 IUNIT,ITYPE !THE UNIT NUMBER AND TYPE NUMBER FOR THIS COMPONENT
       INTEGER*4 ICNTRL             !AN ARRAY FOR HOLDING VALUES OF CONTROL FUNCTIONS WITH THE NEW
SOLVER
       INTEGER*4 NSTORED            !THE NUMBER OF VARIABLES THAT WILL BE PASSED INTO AND OUT OF
STORAGE
       CHARACTER*3 OCHECK           !AN ARRAY TO BE FILLED WITH THE CORRECT VARIABLE TYPES FOR THE
OUTPUTS
       CHARACTER*3 YCHECK           !AN ARRAY TO BE FILLED WITH THE CORRECT VARIABLE TYPES FOR THE
INPUTS
C----------------------------------------------------------------------------------------------------
-------------------

C----------------------------------------------------------------------------------------------------
-------------------
C    USER DECLARATIONS - SET THE MAXIMUM NUMBER OF PARAMETERS (NP), INPUTS (NI),
C    OUTPUTS (NOUT), AND DERIVATIVES (ND) THAT MAY BE SUPPLIED FOR THIS TYPE
      PARAMETER (NP=3,NI=6,NOUT=6,ND=0,NSTORED=0)
C----------------------------------------------------------------------------------------------------
-------------------

C----------------------------------------------------------------------------------------------------
-------------------
C    REQUIRED TRNSYS DIMENSIONS
      DIMENSION XIN(NI),OUT(NOUT),PAR(NP),YCHECK(NI),OCHECK(NOUT),
       1   STORED(NSTORED),T(ND),DTDT(ND)
      INTEGER NITEMS
C----------------------------------------------------------------------------------------------------
-------------------
C----------------------------------------------------------------------------------------------------
-------------------
C    ADD DECLARATIONS AND DEFINITIONS FOR THE USER-VARIABLES HERE
      INTEGER ITIME,Days_before_monday,NDAYYR,NWKYR,NDAYWK
      LOGICAL morning,midday,evening,overnight,summer,winter,
       1   on_peak,mid_peak,off_peak,weekday,weekend
      DOUBLE PRECISION DELT

C    PARAMETERS
      DOUBLE PRECISION S_start
      DOUBLE PRECISION W_start
      DOUBLE PRECISION Monday_day_number

C    INPUTS
      DOUBLE PRECISION Mor_start
      DOUBLE PRECISION Mid_start
```

```
      DOUBLE PRECISION Eve_start
      DOUBLE PRECISION Ove_start
      DOUBLE PRECISION Load_1
      DOUBLE PRECISION Load_2

C----------------------------------------------------------------------------------------------------
-------------------
C        READ IN THE VALUES OF THE PARAMETERS IN SEQUENTIAL ORDER
      S_start=PAR(1)
      W_start=PAR(2)
      Monday_day_number=PAR(3)

C----------------------------------------------------------------------------------------------------
-------------------
C    RETRIEVE THE CURRENT VALUES OF THE INPUTS TO THIS MODEL FROM THE XIN ARRAY IN SEQUENTIAL ORDER

      Mor_start=XIN(1)
      Mid_start=XIN(2)
      Eve_start=XIN(3)
      Ove_start=XIN(4)
      Load_1=XIN(5)
      Load_2=XIN(6)
          IUNIT=INFO(1)
          ITYPE=INFO(2)

C----------------------------------------------------------------------------------------------------
-------------------
C    SET THE VERSION INFORMATION FOR TRNSYS
      IF(INFO(7).EQ.-2) THEN
          INFO(12)=16
          RETURN 1
       ENDIF
C----------------------------------------------------------------------------------------------------
-------------------

C       TRNSYS Functions
       DELT=getSimulationTimeStep()

C----------------------------------------------------------------------------------------------------
-------------------
C    DO ALL THE VERY LAST CALL OF THE SIMULATION MANIPULATIONS HERE
      IF (INFO(8).EQ.-1) THEN
          RETURN 1
       ENDIF
C----------------------------------------------------------------------------------------------------
-------------------

C----------------------------------------------------------------------------------------------------
-------------------
C    PERFORM ANY 'AFTER-ITERATION' MANIPULATIONS THAT ARE REQUIRED HERE
C    e.g. save variables to storage array for the next timestep
      IF (INFO(13).GT.0) THEN
          NITEMS=0
C         STORED(1)=... (if NITEMS > 0)
C        CALL setStorageVars(STORED,NITEMS,INFO)
          RETURN 1
       ENDIF
C
C----------------------------------------------------------------------------------------------------
-------------------
```

```
C-----------------------------------------------------------------------------------------------------
-------------------
C    DO ALL THE VERY FIRST CALL OF THE SIMULATION MANIPULATIONS HERE
      IF (INFO(7).EQ.-1) THEN

C       SET SOME INFO ARRAY VARIABLES TO TELL THE TRNSYS ENGINE HOW THIS TYPE IS TO WORK
         INFO(6)=NOUT
         INFO(9)=1
          INFO(10)=0  !STORAGE FOR VERSION 16 HAS BEEN CHANGED

C       SET THE REQUIRED NUMBER OF INPUTS, PARAMETERS AND DERIVATIVES THAT THE USER SHOULD SUPPLY IN
THE INPUT FILE
C       IN SOME CASES, THE NUMBER OF VARIABLES MAY DEPEND ON THE VALUE OF PARAMETERS TO THIS
MODEL....
         NIN=NI
          NPAR=NP
          NDER=ND

C       CALL THE TYPE CHECK SUBROUTINE TO COMPARE WHAT THIS COMPONENT REQUIRES TO WHAT IS SUPPLIED IN
C       THE TRNSYS INPUT FILE
          CALL TYPECK(1,INFO,NIN,NPAR,NDER)

C       SET THE NUMBER OF STORAGE SPOTS NEEDED FOR THIS COMPONENT
         NITEMS=0
C         CALL setStorageSize(NITEMS,INFO)

C       RETURN TO THE CALLING PROGRAM
         RETURN 1

      ENDIF
C-----------------------------------------------------------------------------------------------------
-------------------

C-----------------------------------------------------------------------------------------------------
-------------------
C    DO ALL OF THE INITIAL TIMESTEP MANIPULATIONS HERE - THERE ARE NO ITERATIONS AT THE INTIAL TIME
      IF (TIME .LT. (getSimulationStartTime() +
     . getSimulationTimeStep()/2.D0)) THEN

C       SET THE UNIT NUMBER FOR FUTURE CALLS
         IUNIT=INFO(1)
         ITYPE=INFO(2)

C       CHECK THE PARAMETERS FOR PROBLEMS AND RETURN FROM THE SUBROUTINE IF AN ERROR IS FOUND
C         IF(...) CALL TYPECK(-4,INFO,0,"BAD PARAMETER #",0)

C       PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL VALUES OF THE OUTPUTS HERE
C               L1 Off-p usage
                     OUT(1)=0
C               L1 Mid-p usage
                     OUT(2)=0
C               L1 On-p usage
                     OUT(3)=0
C               L2 Off-p usage
                     OUT(4)=0
C               L2 Mid-p usage
                     OUT(5)=0
C               L2 On-p usage
                     OUT(6)=0

C       PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL STORAGE VARIABLES HERE
```

```
         NITEMS=0
C          STORED(1)=...

C        PUT THE STORED ARRAY IN THE GLOBAL STORED ARRAY
C          CALL setStorageVars(STORED,NITEMS,INFO)

C        RETURN TO THE CALLING PROGRAM
         RETURN 1

      ENDIF
C-----------------------------------------------------------------------------------------------------
-------------------

C-----------------------------------------------------------------------------------------------------
-------------------
C    *** ITS AN ITERATIVE CALL TO THIS COMPONENT ***
C-----------------------------------------------------------------------------------------------------
-------------------

C-----------------------------------------------------------------------------------------------------
-------------------
C    RETRIEVE THE VALUES IN THE STORAGE ARRAY FOR THIS ITERATION
C      NITEMS=
C      CALL getStorageVars(STORED,NITEMS,INFO)
C      STORED(1)=
C-----------------------------------------------------------------------------------------------------
-------------------
C-----------------------------------------------------------------------------------------------------
-------------------
C    CHECK THE INPUTS FOR PROBLEMS
C      IF(...) CALL TYPECK(-3,INFO,'BAD INPUT #',0,0)
C      IF(IERROR.GT.0) RETURN 1
C-----------------------------------------------------------------------------------------------------
-------------------
C-----------------------------------------------------------------------------------------------------
-------------------
C    *** PERFORM ALL THE CALCULATION HERE FOR THIS MODEL. ***
C-----------------------------------------------------------------------------------------------------
-------------------

C              ADD YOUR COMPONENT EQUATIONS HERE; BASICALLY THE EQUATIONS THAT WILL
C              CALCULATE THE OUTPUTS BASED ON THE PARAMETERS AND THE INPUTS.    REFER TO
C              CHAPTER 3 OF THE TRNSYS VOLUME 1 MANUAL FOR DETAILED INFORMATION ON
C              WRITING TRNSYS COMPONENTS.

C           Calculate day of year, week of the year, day of week
      ITIME = JFIX(TIME - DELT/2.)
      NDAYYR = ITIME/24 + 1  !1 = Jan 1
      NWKYR = ITIME / 168
      NDAYWK = NDAYYR - (NWKYR * 7) !1 thru 7

C           Determine if it is a weekday or weekend
      Days_before_monday = Monday_day_number - NDAYWK
      IF (Days_before_monday .LE. 0) THEN
            Days_before_monday = Days_before_monday + 7
      ENDIF

      IF (Days_before_monday .LE. 2) THEN !it's a weekend
            weekday = .FALSE.
            weekend = .TRUE.
      ELSE !it's a weekday
```

```
            weekday = .TRUE.
            weekend = .FALSE.
      ENDIF

      IF (weekend) THEN
            !rate is off-peak
            on_peak = .FALSE.
            mid_peak = .FALSE.
            off_peak = .TRUE.
      ELSE

C           Determine if it is morning, midday, evening, or overnight
      IF ((ITIME .GE. Mor_start) .AND. (ITIME .LT. Mid_start)) THEN
            morning = .TRUE.
            midday = .FALSE.
            evening = .FALSE.
            overnight = .FALSE.
      ELSEIF ((ITIME .GE. Mid_start) .AND. (ITIME .LT. Eve_start)) THEN
            morning = .FALSE.
            midday = .TRUE.
            evening = .FALSE.
            overnight = .FALSE.
      ELSEIF ((ITIME .GE. Eve_start) .AND. (ITIME .LT. Ove_start)) THEN
            morning = .FALSE.
            midday = .FALSE.
            evening = .TRUE.
            overnight = .FALSE.
      ELSE
            morning = .FALSE.
            midday = .FALSE.
            evening = .FALSE.
            overnight = .TRUE.
      END IF

C           Determine if it is summer or winter and assign load to appropriate rate: off-peak, mid-
peak, on-peak
      IF ((NDAYYR .GE. S_start) .AND. (NDAYYR .LT. W_start)) THEN
C           It is summer
            summer = .TRUE.
            winter = .FALSE.
            IF (morning .OR. evening) THEN !mid-peak
                  on_peak = .FALSE.
                  mid_peak = .TRUE.
                  off_peak = .FALSE.
            ELSEIF (midday) THEN !on-peak
                  on_peak = .TRUE.
                  mid_peak = .FALSE.
                  off_peak = .FALSE.
            ELSE !off-peak
                  on_peak = .FALSE.
                  mid_peak = .FALSE.
                  off_peak = .TRUE.
               END IF
      ELSE
C           It is winter
            summer = .FALSE.
            winter = .TRUE.
            IF (morning .OR. evening) THEN !on-peak
                  on_peak = .TRUE.
                  mid_peak = .FALSE.
                  off_peak = .FALSE.
```

```
            ELSEIF (midday) THEN !mid-peak
                  on_peak = .FALSE.
                  mid_peak = .TRUE.
                  off_peak = .FALSE.
            ELSE !off-peak
                  on_peak = .FALSE.
                  mid_peak = .FALSE.
                  off_peak = .TRUE.
      END IF
      END IF
       END IF

C-----------------------------------------------------------------------------------------------------
-------------------

C-----------------------------------------------------------------------------------------------------
-------------------
C-----------------------------------------------------------------------------------------------------
-------------------
C    SET THE STORAGE ARRAY AT THE END OF THIS ITERATION IF NECESSARY
C      NITEMS=
C      STORED(1)=
C      CALL setStorageVars(STORED,NITEMS,INFO)
C-----------------------------------------------------------------------------------------------------
-------------------
C-----------------------------------------------------------------------------------------------------
-------------------
C    REPORT ANY PROBLEMS THAT HAVE BEEN FOUND USING CALLS LIKE THIS:
C      CALL MESSAGES(-1,'put your message here','MESSAGE',IUNIT,ITYPE)
C      CALL MESSAGES(-1,'put your message here','WARNING',IUNIT,ITYPE)
C      CALL MESSAGES(-1,'put your message here','SEVERE',IUNIT,ITYPE)
C      CALL MESSAGES(-1,'put your message here','FATAL',IUNIT,ITYPE)
C-----------------------------------------------------------------------------------------------------
-------------------
C-----------------------------------------------------------------------------------------------------
-------------------
C    SET THE OUTPUTS FROM THIS MODEL IN SEQUENTIAL ORDER AND GET OUT
      IF (off_peak) THEN
C               L1 Off-p usage
                      OUT(1)=Load_1
C               L1 Mid-p usage
                      OUT(2)=0
C               L1 On-p usage
                      OUT(3)=0
C               L2 Off-p usage
                      OUT(4)=Load_2
C               L2 Mid-p usage
                      OUT(5)=0
C               L2 On-p usage
                      OUT(6)=0
      ELSEIF (mid_peak) THEN
C               L1 Off-p usage
                      OUT(1)=0
C               L1 Mid-p usage
                      OUT(2)=Load_1
C               L1 On-p usage
                      OUT(3)=0
C               L2 Off-p usage
                      OUT(4)=0
C               L2 Mid-p usage
                      OUT(5)=Load_2
```

```
C               L2 On-p usage
                      OUT(6)=0
      ELSE
C               L1 Off-p usage
                      OUT(1)=0
C               L1 Mid-p usage
                      OUT(2)=0
C               L1 On-p usage
                      OUT(3)=Load_1
C               L2 Off-p usage
                      OUT(4)=0
C               L2 Mid-p usage
                      OUT(5)=0
C               L2 On-p usage
                      OUT(6)=Load_2
      END IF

C-----------------------------------------------------------------------------------------------------
-------------------
C    EVERYTHING IS DONE - RETURN FROM THIS SUBROUTINE AND MOVE ON
      RETURN 1
      END
C------------------------------------------------------------------------------------------------------------
```