

Evaluation of a Series-Configuration Solar-Assisted Heat Pump with Performance-Predictive Controls

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.

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Abstract

This thesis evaluates a configuration of a 3-mode solar-assisted heat pump (SAHP), which features a heat exchanger and heat pump which may operate independently or in series to provide heat for a domestic hot water system. The system is designed to deliver heat through the heat exchanger, heat pump, or both simultaneously, with the operating mode dictated by a system controller. The configuration is evaluated in comparison to established system designs, such as electric-only, solar domestic hot water (SDHW) and 2-mode SAHP (heat exchanger or heat pump) systems. TRNSYS 17 simulation software is used to perform year-long simulations of the systems for performance comparison purposes. Subjecting each system to the same parameters, weather conditions and draw patterns ensures that differences in performance are due to the available modes of heat delivery.

The systems simulated in TRNSYS use mathematical models of physical components to model the behaviour of a real system. Most of the models used were previously validated by former student William Wagar, supporting their use here. The system includes a heat exchanger and heat pump that are connected to operate independently or in series, depending on the settings of several flow diverter valves. The inclusion of the series configuration is intended to maximize heat gains in high-demand scenarios and to extend heat pump operating hours in low-availability scenarios.

A highly-detailed decision-making scheme is developed to control the system. The scheme uses measurements of available sunlight and system temperatures to predict the performance of individual components, using validated equations and specified system settings. The predicted performance values are used, in conjunction with an evaluation of the current system demands for heat, to determine the optimal mode of operation at that time. The “optimal” mode may be the

mode which delivers the most heat, consumes the least electricity, or has the best heat delivery to electricity consumption ratio (COP), depending on the demand; the highly effective control strategy balances these priorities to achieve high solar gains and energy savings, while maintaining high temperatures in the storage tank.

The behaviour of multi-mode SAHP systems, when contrasted with single-mode designs, shows that having the option to use a heat exchanger and a heat pump substantially improves solar gains and electricity savings. Such systems have the flexibility to adapt to changing weather conditions to ensure strong performance at all times. The behaviour study shows that under the current control settings, the series configuration is rarely utilized by the system. Additionally, system behaviour is dominated by the maximum heat delivery mode, with rare consideration of power consumption. A parametric study indicates that SAHP systems operate most effectively with a constant thermostat setting of 60 C. A location/climate study indicates that SAHP systems are best-suited to settings in which SDHW systems fare poorly, such as the sub-arctic climate of Whitehorse, Yukon, Canada.

A highly detailed control scheme with the ability to apply several definitions of “optimal performance” is shown to allow excellent system performance in comparison to single-definition designs. The results of this study indicate that the series mode is a beneficial but rarely-used option, though use of this mode may be increased if a smaller heat pump is installed. It is recommended to focus on further development of the control scheme by optimizing system settings. It is also recommended to perform tests of the system, using an appropriately-sized variable-speed heat pump, to validate the controller’s feasibility and further advance the field of SAHP study.

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Chapter 1 - Introduction

The ability to convert fuel-stored energy into useful forms is at the core of modern civilization; it is the means by which all people experience motorized transport, long-distance communication and the benefits of electricity today. As more of the world's societies grow, the collective energy consumption of mankind grows as well, a need which is met mostly using fossil fuels. BP [1] reports that in 2016, over 13 250 million tons of oil equivalent (over 154 PWh) were consumed worldwide, a 1% growth over 2015, with most of the growth taking place in Asia and the Middle East. More than 85% of this energy was provided by fossil fuels such as oil, coal and natural gas.

Economic growth, quality-of-life improvements and increased energy consumption are inextricably linked. However, a profound detrimental effect of this growth has been established, related to the environmental impact of increased fossil fuel consumption. The emission of Greenhouse Gases (GHGs) – gases such as carbon dioxide (CO₂) which trap heat in the lower atmosphere – has brought the levels of atmospheric carbon above 400 ppm. 2016 is the first full year in measurable history for which CO₂ levels stayed above 400 ppm for the entire year [2]. The consequences of this astoundingly high concentration has manifested itself in climate change and the phenomenon known as global warming. The shifting of extreme and damaging climate conditions worldwide poses a major threat to mankind today and in the future, when the consequences of global warming will be even more apparent than they are today.

1.1 – Renewable Energy

The world is taking remedial and preventative action as Earth becomes hotter. In 2015, the United Nations collectively signed the Paris Agreement [3], collectively agreeing to make changes that will put a hard-stop on rising temperatures by encouraging development of low-emissions energy

sources. Renewable energy sources are growing rapidly – though the total share of worldwide energy supplied by renewables is low, sources such as wind, geothermal and solar energy have grown at a rate of 16% per year since 2005, with China and the USA contributing the most as growth began to stall in Europe, as shown in Figure 1.1 [1, 4]. In particular, solar power is gaining capacity quite rapidly – while virtually no solar energy was harnessed in 2000, there was 50 GW of capacity worldwide in 2010 and well over 300 GW are available today [4]. The explosive growth of solar energy, compared to wind and nuclear energy, in the countries included in BP’s Statistical Review is also illustrated in Figure 1.1 [1].

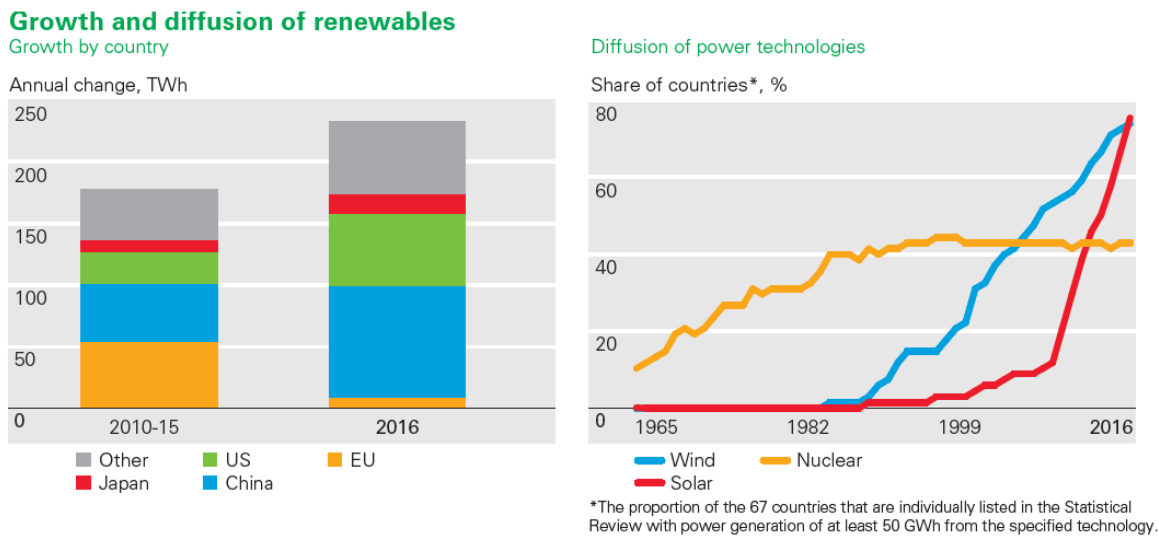


Figure 1.1 - Growth and diffusion of renewable energy sources as per BP's Statistical Review [1]

Even so, more needs to be done to replace fossil fuels with renewable energy sources. Far more renewable energy is available than is consumed for human purposes, even from individual sources such as the Sun. The Sun itself emits incredible amounts of heat and light energy – even though Earth is a relatively small planet, more than 100 million kilometres away from the Sun, about 1.7×10^{17} W of solar energy strikes the atmosphere. A lot of this energy is reflected or absorbed by

the atmosphere, following the paths illustrated in Figure 1.2, but about half makes it down to the surface where it can be harnessed for use [5].

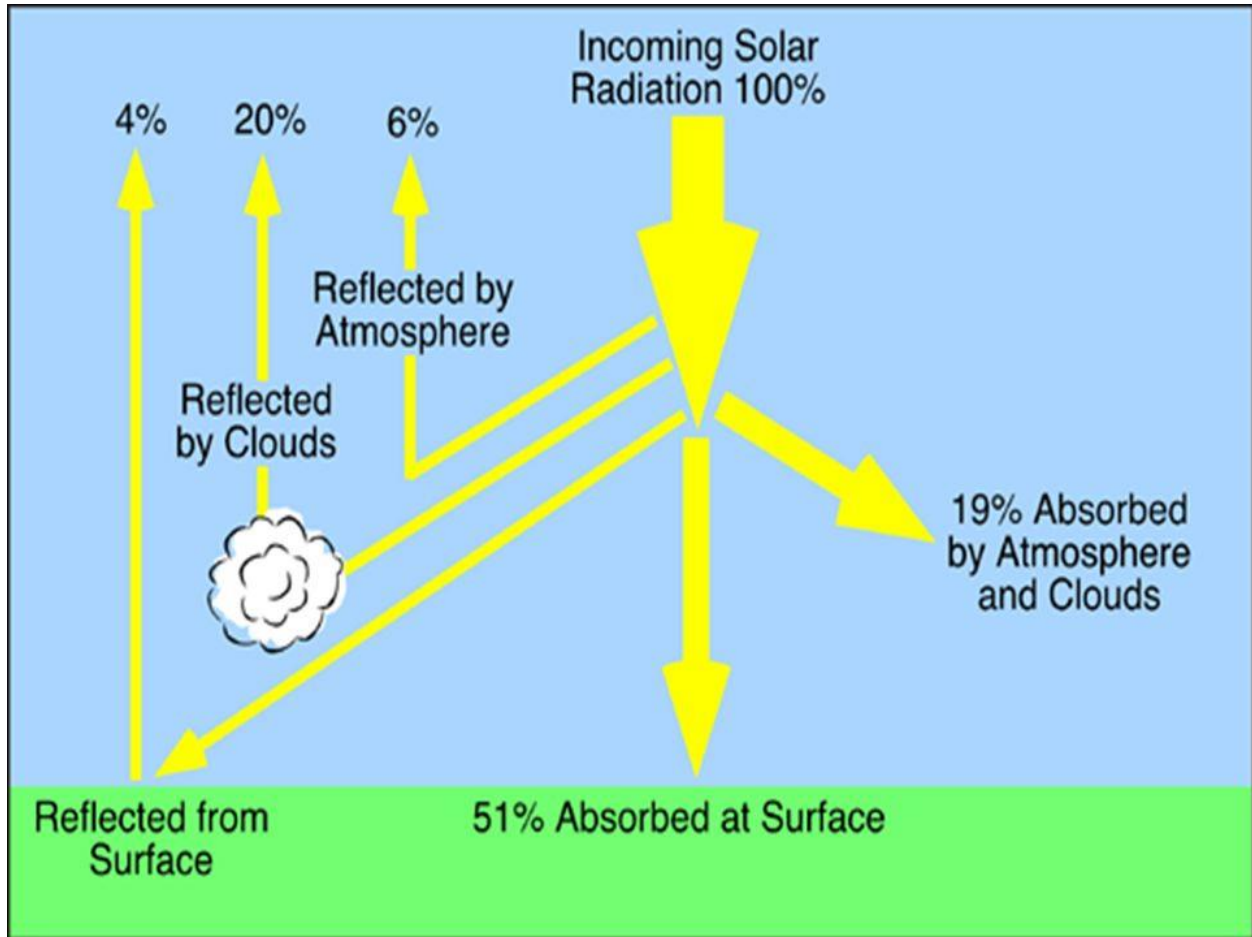


Figure 1.2 - Solar heat budget of the Earth [5]

The amount of solar energy that reaches the surface of the Earth each year is on the order of magnitude of 10^9 PJ (10^{24} J), several thousand times greater than the annual energy consumption of the world. The only costs and emissions associated with harnessing this energy come in the start-up phase – collection is entirely emissions-free. Given the vast availability of the resource and the urgent need to dramatically reduce GHG emissions, the Sun is a key target for renewable energy technologies.

1.2 – Energy Use in Canada

Energy use in Canada has stabilized since 2010, peaking in 2013; in 2016, the nation owned a 2.5% share of the total worldwide energy consumption, ranking 6th worldwide [1]. Factoring in the populations of the top 6 energy-consuming nations (the top 5 all have populations at least 5 times greater than Canada) indicates that Canadians are the greatest energy consumers in the world. Despite a large use of hydroelectric energy sources (almost 10% of all hydro used worldwide), 63% of Canada’s energy still comes from fossil fuels, making Canadians some of the world’s greatest GHG emitters as well [1, 6]. Of the energy used in Canada, the greatest portions are used for industry, transportation and residential purposes, illustrated in Figure 1.3.

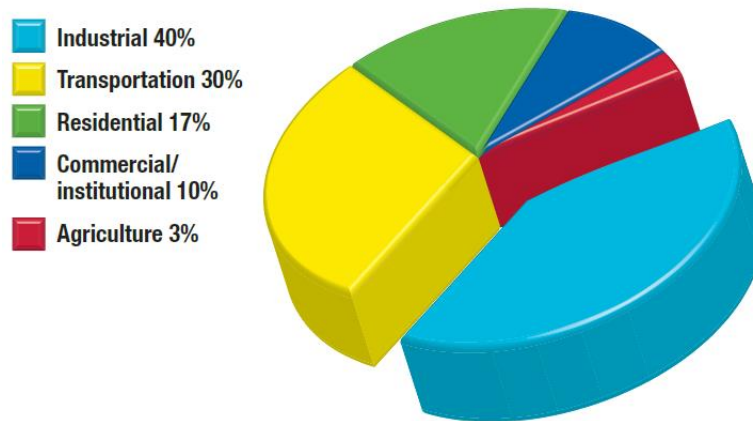


Figure 1.3 - Energy use in Canada by sector [7]

Though more energy is spent in industry and transportation, there is little that an average Canadian can do to affect their contributions here beyond choosing efficient personal vehicles. However, every Canadian can dictate their residential consumption through habits and design choices, making residential energy use an attractive field of study. In the residential sector, most energy consumption is electrical or natural gas – alternative energy sources account for less than 16% of all Canadian consumption [8]. As shown in Figure 1.4, the vast majority of residential consumption

goes towards space heating, accounting for 63% of all usage [7]. Water heating uses another 19%, for a total of 82% of residential use being dedicated to heating [7].

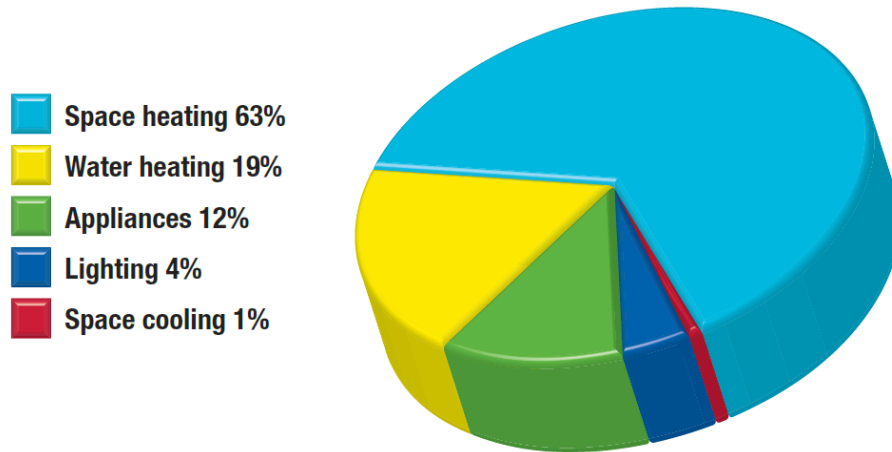


Figure 1.4 - End-use of residential energy [7]

Natural gas and electricity are the primary energy sources for space and water heating (above 80%), with heating oil being a distant third source [8]. Though heating equipment continues to improve, the growth of the market has exceeded gains in efficiency, resulting in continuing increases in energy consumption that signal a need for still better technology [6]. With such an abundant pure-heat resource as sunlight so readily available, there is little question that solar heating technology provides the greatest opportunity for energy use and GHG emissions reductions for Canadians.

1.3 – Production and Storage of Hot Water

Water heating systems can achieve very high energy efficiency levels, even without solar resource input. Natural gas-fired tanks, though inexpensive to operate, lose efficiency whenever incomplete combustion occurs or any time the gas exhausts at a temperature higher than the injected fuel. These events represent energy that is being exhausted through the flue instead of being transferred to the water in the tank. Additionally, the storage tank will lose heat to its surroundings, further

reducing efficiency. While the heat source for an electric tank is much more efficient than natural gas (100% of purchased energy is converted into heat that gets stored in the tank), such tanks are still subject to losses, and so more energy must be purchased than is used.

Tank energy losses can be reduced by allowing the stored water to stratify itself according to density. Since hot water is slightly less dense than cold water, heated water will naturally rise and cold water will naturally settle at the bottom of the tank. A tank design can save energy by encouraging stratification of the water – the result is a concentration of hot water near the top of the tank, reducing the surface area through which heat can be lost to the environment. It also ensures that no more water is held at a high temperature than is needed, further reducing losses; in fact, the cold water at the bottom of the tank is likely to gain energy from the environment, thereby offsetting even more of the energy lost at the top. A stratified storage tank design is illustrated in

Figure 1.5.

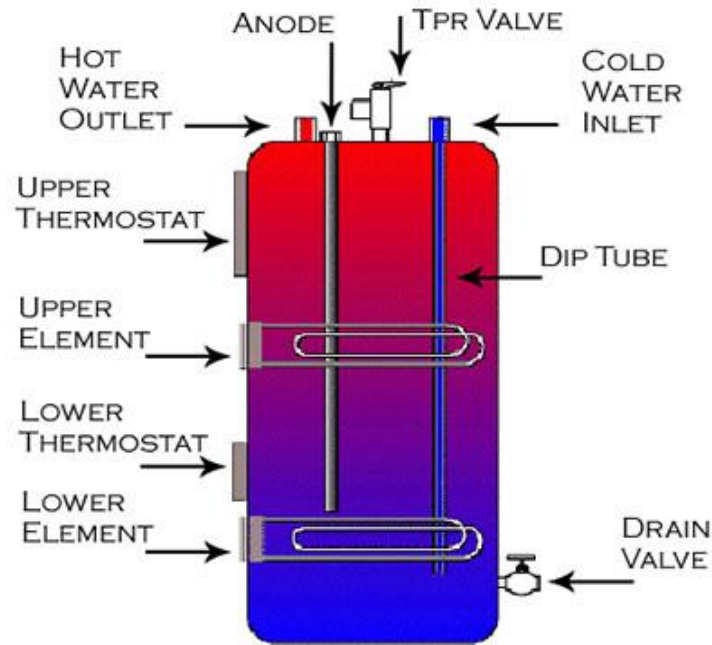


Figure 1.5 - Electric Domestic Hot Water tank design [9]

Natural gas is much less expensive as an energy source - in Ottawa, the cost of electricity is more than four times the cost of natural gas, per kWh equivalent. However, natural gas is not available in all parts of the country and its consumption results in high GHG emissions, exacerbating the problem of global warming. Electricity is a much cleaner energy source, especially in Ontario and Quebec where most energy is nuclear or hydro-source, but coal or oil-fired power plants in western Canada negate this benefit. Electricity is also much more expensive than gas in most of the country, placing an economic burden on users in exchange for cleaner energy. Unfortunately, a simple electric tank design is still limited to an energy efficiency of 100% or less, hence the amount of purchased energy is almost the same as with a natural gas design, reducing the attractiveness of electric tanks as a solution to residential emissions.

The inclusion of a Solar Thermal Collector (STC) can significantly alleviate these problems. Solar energy is “free” (after accounting for the capital and maintenance costs of the equipment) and has no emissions – integrating this with a natural gas tank can substantially reduce emissions, while integration with an electric tank can significantly reduce costs. Such a Solar Domestic Hot Water (SDHW) system is illustrated in Figure 1.6.

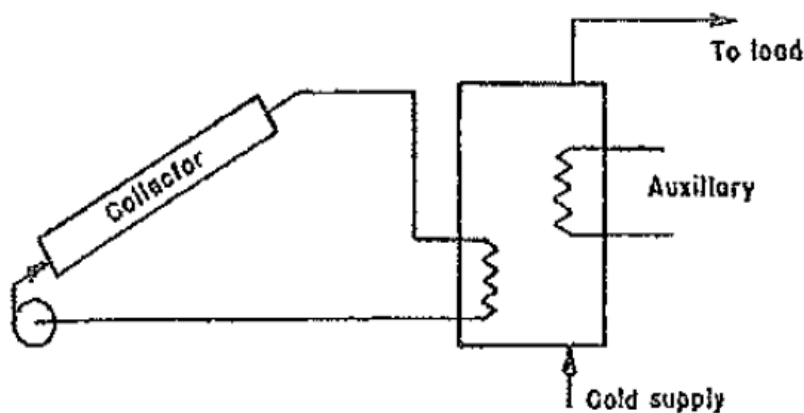


Figure 1.6 - Solar Domestic Hot Water system designs [10]

Design options are highly variable, but the core components are consistent. Fluid is typically circulated at a low flow rate with a pump and heated in a solar thermal collector. Especially in Canada, where anti-freezing protection is required, it is common to separate the stored water and the heat-collecting fluid; heat is transferred through a Heat Exchanger (HX) instead. A drawback of the SDHW system in cold climates is that the attainable heat gains are low. If the temperature of the fluid leaving the solar collector is low, it may not be possible to transfer any energy to the tank water. The result is very poor performance in the winter season, a time when the demand for hot water is at its highest and solar availability is lowest.

However, technology for “pumping” thermal energy from a cold source to a hot sink is well-established, allowing for energy collection even during the winter. The vapour compression refrigeration cycle is an extremely common thermodynamic process, used to move heat against its natural transfer direction just as a pump can be used to move water against the force of gravity. There are four stages to the cycle, illustrated in Figure 1.7 [11]: evaporation, compression, condensation and expansion.

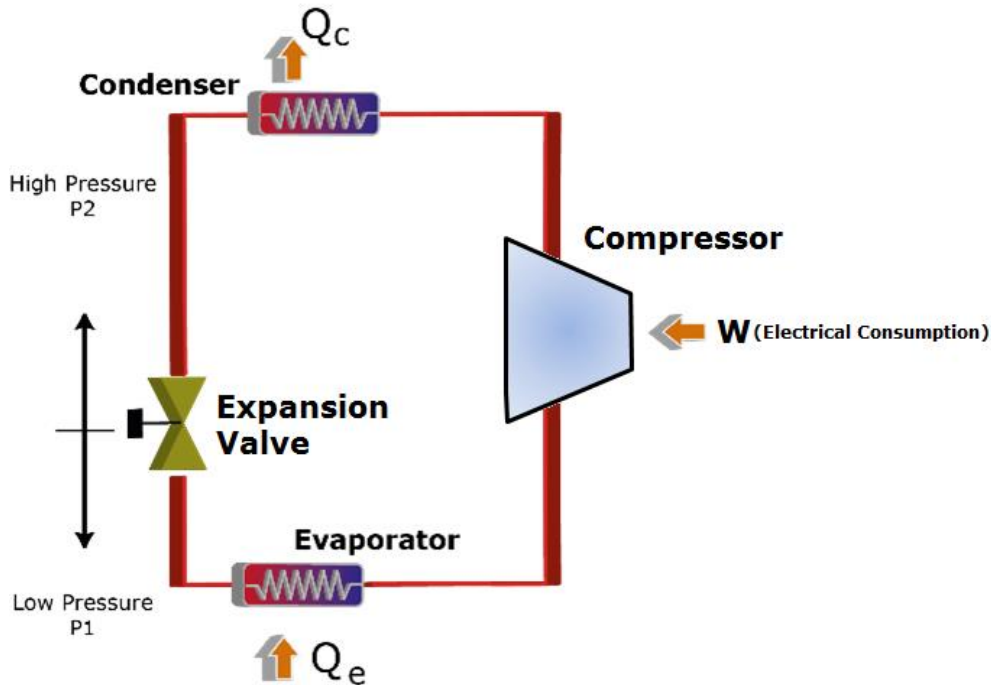


Figure 1.7 - Vapour compression refrigeration cycle [11]

In the evaporation stage, the refrigerant fluid is a low-pressure liquid, with a very low boiling point. The fluid passes through a heat exchanger, where it boils, collecting heat from a low-temperature energy source (thereby cooling the energy source). The boiled vapour then enters a compressor, where it becomes highly pressurized. The high pressure causes the temperature of the vapour to rise substantially as well. This hot, high-pressure gas then enters a condensing heat exchanger, in which the gas condenses to a liquid and releases heat into a hot energy sink. When the fluid has fully condensed into a liquid, it finally passes through an expansion device, which substantially reduces the fluid pressure and reduces the boiling point. It is at this point that the fluid re-enters the evaporator, completing the cycle.

This technology is used around the world as refrigerators to chill a cool space; however, it is also used for heating purposes, in which case the device is called a Heat Pump (HP) instead of a refrigerator. As a heater, the vapour compression cycle is far more energy efficient than an electric

resistance heater. Not only is 100% of the consumed electricity eventually converted into heat, but most (if not all) of the energy drawn out of the cool source is delivered to the hot sink as well. While delivering 1 kWh of energy requires 1 kWh of electricity with a resistance heater, a heat pump delivering the same amount of energy can do so while only consuming 0.1-0.5 kWh of electricity. The consumed electricity makes up some of the 1 kWh of heat, while the remaining 0.5-0.9 kWh is drawn from the low-temperature heat source.

The question of using a heat pump for a domestic hot water system is primarily focused on the source of the energy. Air-Source Heat Pumps (ASHPs) are inexpensive, but their effectiveness is often limited by the ambient temperature, since efficiency drops substantially in sub-zero conditions. Ground-Source Heat Pumps (GSHPs) are more stable, since the deep-ground temperature changes very slowly over time, but the installation of such a system is expensive and labour-intensive. A Solar-Assisted Heat Pump (SAHP) design offers a highly effective compromise between the two, with notable advantages over air and ground-source systems for residential heating applications. Except for installing a solar thermal collector, the required equipment is no different than that of an ASHP, keeping the initial capital costs low compared to GSHPs. Then, during operation, the STC provides a warmer-than-ambient heat source for the heat pump by collecting solar energy, boosting the SAHP's performance compared to ASHP. This is because increasing the cold-side source temperature improves the efficiency of the vapour compression cycle. The definition of a heat pump allows the maximum Coefficient of Performance (COP), which describes the "efficiency" of the device, to be defined as follows:

$$COP_{HP,Carnot} = \frac{T_H}{T_H - T_C} \quad (1)$$

Where T_H is the hot-side inlet temperature (from the tank) and T_C is the cold-side inlet temperature (from the collector). For a fixed tank temperature, then, the inclusion of an STC will boost T_C above the ambient temperature, increasing the theoretical maximum (and consequently the actual) performance compared to an air-source design during daylight hours.

An additional benefit of this design is the reciprocal performance boost that the heat pump gives to the STC by reducing the temperature of the STC inlet fluid. A standard HX-based SDHW can only deliver STC inlet temperatures as low as the storage tank outlet – this is the minimum possible outlet temperature for the source-side of the HX. Under the same conditions, a heat pump can bring the source-side temperature much lower, since it sheds the limitations of unassisted natural heat transfer. The colder STC inlet fluid is more receptive to heat gains, from the sun and from the surrounding environment, thereby increasing the amount of solar energy collected in the STC. Figure 1.8 [12] shows efficiency curves for various STC designs, plotted as a function of $(T_{\text{Fluid}} - T_{\text{Ambient}}) / G_{\text{T,total}}$. Observe the loss of efficiency as T_{Ambient} drops (increasing the temperature gradient) and the efficiency gained as T_{Fluid} drops (decreasing the temperature gradient). This illustrates why winter-time performance is poor and why HP-assisted performance is improved.

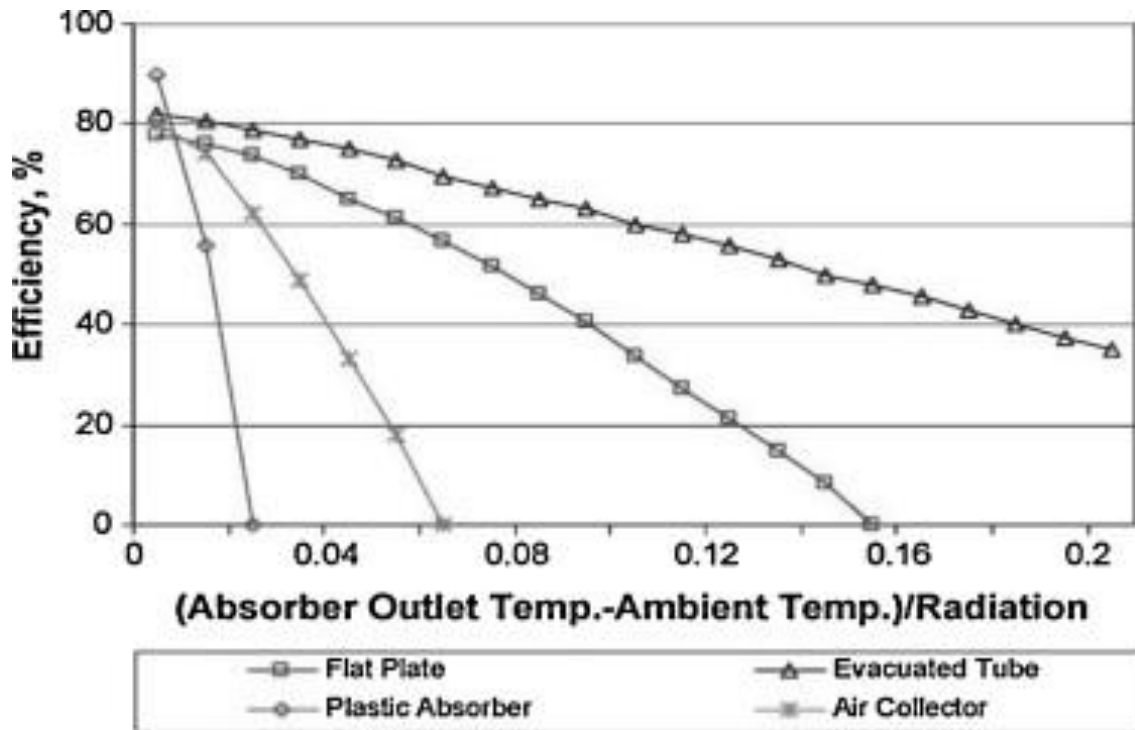


Figure 1.8 - Efficiency curves for several solar thermal collector designs [12]

A wide field of study exists for SAHP systems as a result of this mutually-beneficial partnership, including this thesis.

1.4 – Objective and Outline of Thesis

The objective of this thesis is to investigate the performance of a three-mode SAHP system with a sophisticated control scheme, using scenario-based decision making, in comparison to alternatives. The potential of simultaneous HX and HP operation in a series configuration to increase solar gain and reduce electricity consumption is to be analyzed. A detailed control scheme for a multi-mode SAHP is to be developed for the purpose of minimizing energy consumption. The system considered, and those to which it is compared, is intended to meet the DHW load for a detached, single-family dwelling. The studies of this thesis are to be performed using TRNSYS 17 simulation software, with system components modelled based on the work of Wagar and Bannister [13, 14], which included experimental validation of most of the models used.

A literature review is provided, reviewing methods of thermal energy storage, solar domestic hot water systems and the various fields of study in SAHP technology. The history of SAHP study is reviewed, and branches of study are discussed with tabulated sources; topics include Direct Expansion (DX) systems, Photovoltaic-Thermal (PVT) systems, combined-function designs and Indirect Expansion (IDX) systems. Multi-mode IDX SAHP systems, which have been studied by the Solar Research Laboratory at the University of Waterloo in the past, are discussed, as these studies provide the background for this thesis.

Chapter 3 presents the systems to be compared in the studies of this thesis. Electric DHW, SDHW, single-mode SAHP, series-configuration SAHP, 2-mode SAHP and 3-mode SAHP systems are chosen for study and described. TRNSYS software is introduced, and the component models required for simulating the aforementioned systems are described, with relevant parameter choices listed. The sophisticated control scheme employed by the multi-mode SAHP systems is described in detail.

Chapter 4 discusses a behaviour and performance comparison study of the described systems. Two systems are introduced which are schematically identical to the 3-mode SAHP but which use modified controls; the three control plans used in separate simulations of the 3-mode SAHP are described as “Minimum Power Consumption (Min P)”, “Maximum Rate of Heat Gain (Max Q)” and “Balanced”. Behaviour of the systems is studied for representative summer and winter days, with advantages and disadvantages discussed throughout. Following this, year-long simulation results are presented for analysis and comparison in categories of solar, energy and thermal performance.

A parametric study is presented in Chapter 5 to analyze the effects of implementing non-constant thermostat settings on the performance of the balanced 3-mode SAHP. It is possible that by

allowing a reduced thermostat setting in times of low demand (such as overnight or mid-day), the system will be encouraged to use auxiliary backup heat less frequently and to use the HX during peak solar hours instead of the HP. The study considers the effect of a thermostat setting profile that is “low” for two periods every day and “high” for two periods every day. The “low” periods are centered about noon and midnight; the “high” periods are of equal duration and are centered about 7 AM and 5 PM. The parameters studied are the temperature level of the “high” setting, the step size between “high” and “low” settings and the duration of the “high” periods.

Also presented in Chapter 5 is a location effect study. The performance, and associated costs, of operating electric DHW, SDHW, 2-mode SAHP and balanced 3-mode SAHP systems is analyzed for 4 locations in unique climates. The cities of Ottawa (Canada), Rome (Italy), Aswan (Egypt) and Whitehorse (Canada) are chosen for the broad range of climate conditions they represent and for comparability to the work of Bannister [14], which considered the same locations.

Finally, conclusions and recommendations are summarized and presented in Chapter 6. Conclusions regarding the behaviour and performance of the various systems are discussed. The viability of the series configuration and the effectiveness of the sophisticated control scheme are summarized. Results of the thermostat set-point and location studies are repeated in brief. Recommendations consider design decisions that should be made based on the work of this thesis, the furthering of said work and the further broadening of the field of SAHP study.

Chapter 2 – Literature Review

Solar heating is a natural phenomenon that mankind has been using to its own devices for centuries. In the modern era, solar energy is one of the first sources considered for renewable or clean energy, prompting rapid growth in the solar energy industry. One area of this industry that is experiencing this high level of interest is solar water heating, through which the electrical/natural gas hot water loads of a building can be offset and potentially even eliminated. Duffie and Beckman present system configurations which can achieve the goal of applying available sunlight to hot water loads by collecting and then storing solar energy in a water tank [10], with an auxiliary heat source attached to ensure that water is delivered at an acceptable temperature. These systems can come in a multitude of configurations such as those shown in Figure 2.1, with design considerations including freeze protection, active versus passive collection, collector type, auxiliary source and location, energy delivery method to the storage and the storage medium.

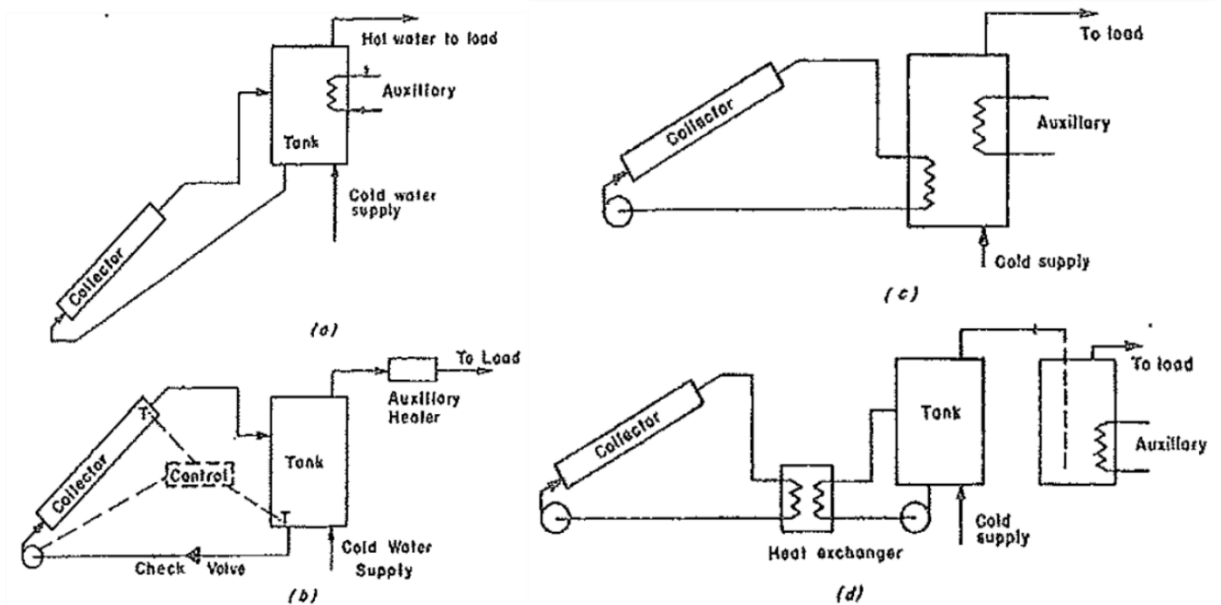


Figure 2.1 - Common SDHW system configurations [10]

Many Thermal Energy Storage (TES) media are available – one can store heat energy in the ground, in a contained rock bed, in a water tank or even in a Phase-Change Material (PCM), which takes advantage of latent energy storage to perform constant-temperature heat exchange; Table 2-1 lists studies relevant to these topics [15-18].

Table 2-1 - Sources examining various TES media

Author(s)	Title	Ref. #
Shitzer, M. L. A.	Transient Behavior of a Rock-Bed Thermal Storage System Subjected to Variable Inlet Air Temperatures: Analysis and Experimentation	[15]
Wang and Qi	Performance study of underground thermal storage in a solar-ground coupled heat pump system for residential buildings	[16]
Pinel et al	A review of available methods for seasonal storage of solar thermal energy in residential applications	[17]
Kaygusuz	Performance of solar-assisted heat-pump systems	[18]

Ground-style storage tends to be very large scale and expensive, whereas PCM storage is high maintenance and potentially volatile, hence water storage is the method-of-choice for many Canadian homes [17]. While different designs may be used, it is common for the energy to be stored in the actual water used by the residence, rather than storing energy elsewhere and using it to heat fresh mains water on demand.

SDHW, while useful, has weaknesses. A diurnal disparity exists between supply and demand – the most sunlight is available when residences tend to be empty, while late night or early morning hot water draws occur when the sun is down. Using thermal energy storage media resolves this problem by collecting solar energy and storing it for later use. An even greater disparity exists, however, between the seasonal supply and demand [17]. Residences generally use more hot water during the winter, especially if water is used for space heating; this coincides with the lowest levels of solar irradiation and the poorest solar collector performance of the year. There are few solutions

to this disparity, all of which involve extremely high-capital cost TES applications that are highly impractical for single-family residences [17]. As a result, researchers have instead looked at improving the ways in which solar energy is collected so that seasonal effects are not so strong. A leading branch of this work studies SAHP technology, in which a heat pump is coupled to the solar hot water system.

A SAHP system is characterized by the use of a vapour compression refrigeration cycle as well as a solar collector in the water-heating system. The refrigeration cycle, which is performed by the heat pump, allows the system to collect energy at times when the solar collector would be otherwise ineffective, and delivers more heat energy per kWh of electricity purchased than an electric water heater can. The applications are not limited to water heating either – Terrell studied a system that used a heat pump for space heating [19], drawing heat energy from a low-temperature water tank that stored energy collected from the sun.

Chandrashekar performed an elaborate study of six potential configurations of SAHP systems (shown in Figure 2.2) [20], with heat pumps installed in parallel or series configurations, drawing energy from water or air sources (and sometimes both), in seven major Canadian cities. After identifying that dual-source heat pumps, which could receive heat from a solar collector and ambient air, were consistently more effective than other designs, the effects of varying system parameters such as collector size and type were studied in several cities. The study concluded that SAHP systems were promising and delivered substantial reductions in energy costs, but at high initial capital investments that made their viability uncertain.

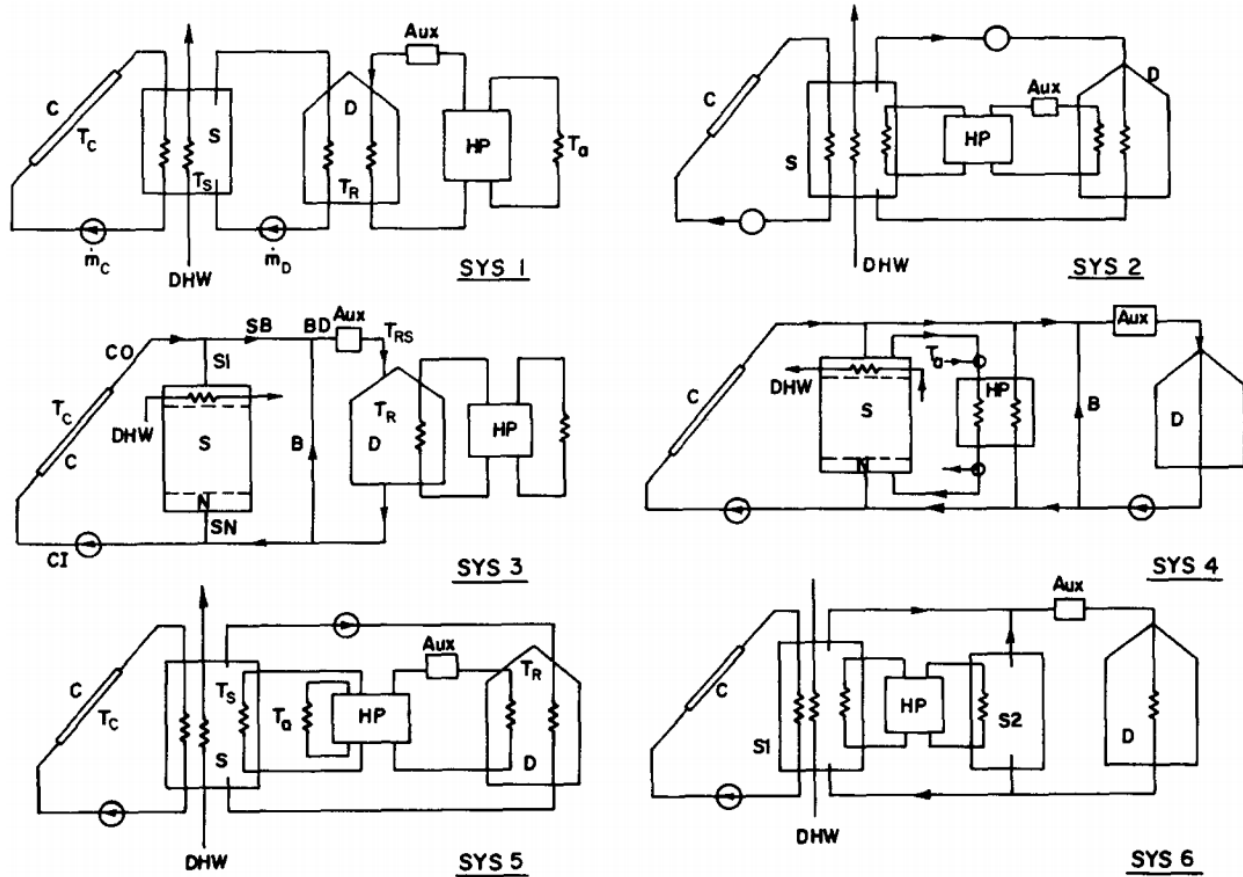


Figure 2.2 - Schematics for systems studied by Chandrashekar et al [20]

Kaygusuz later studied the ability of a dual-source SAHP system that used phase-change material (PCM) storage to provide space heating in Turkey [18]. This system could use a heat exchanger or a heat pump to deliver energy, depending on the conditions available; a simple control scheme used a temperature reading to determine the appropriate operating mode. An important observation noted that air-source HPs did not perform as well as solar-heated liquid-source HPs, and a recommendation suggested varying the control temperature based on heating load to improve performance further.

As development of SAHP technology moves away from using air as the primary heat source for the heat pump and becomes more focused on liquid-source design, two primary design concepts

have emerged – direct expansion and indirect expansion. In direct expansion (DX) systems, the system is configured such that the solar collector is the evaporator heat exchanger of the heat pump.

A schematic of such a system is shown in Figure 2.3.

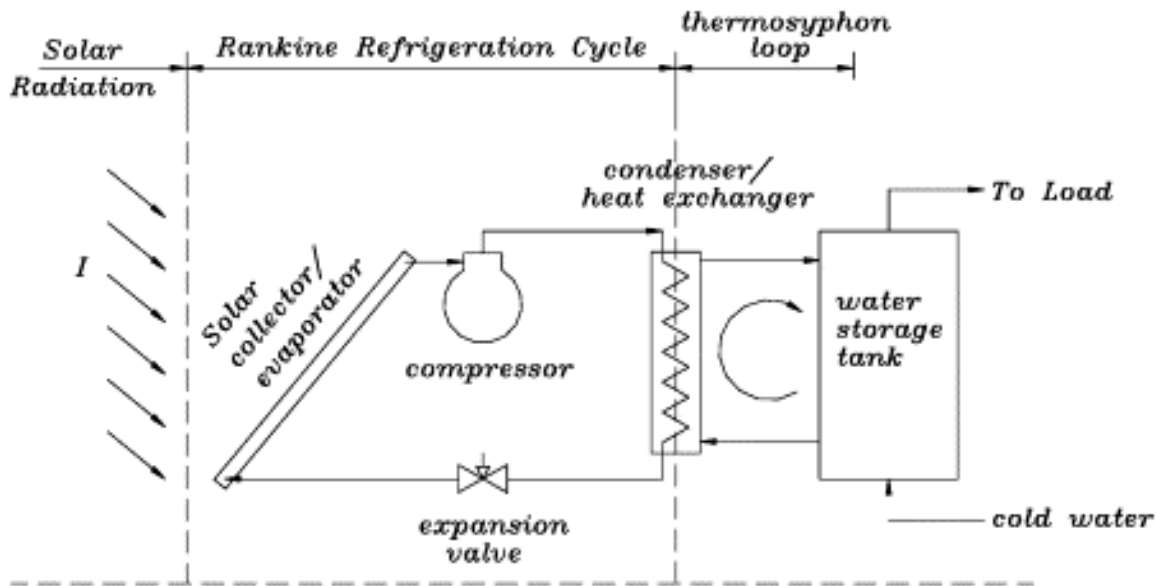


Figure 2.3 - Direct-expansion solar-assisted heat pump schematic [21]

These systems have the benefit of reducing the number of heat exchanges that take place between the collector and the load. In climates where high heat source temperatures are consistently available, DX has been shown to be highly effective, especially in comparison to air-source systems [22, 23]. Huang et al. have performed extensive multi-year studies of experimental DX systems [21, 24, 25], while Kong et al. have developed and validated numerical models for investigating the effects of system parameters on performance [26-28]. Table 2-2 lists a variety of sources relevant to the study of direct-expansion SAHP systems.

Table 2-2 - Studies evaluating the performance of DX-SAHP systems

Author(s)	Title	Ref. #
Sun et al.	Performance Comparison of Direct Expansion Solar-assisted Heat Pump and Conventional Air Source Heat Pump for Domestic Hot Water	[22]
Li et al.	Experimental performance analysis on a direct-expansion solar-assisted heat pump water heater	[23]
Huang and Lee	Long-term performance of solar-assisted heat pump water heater	[21]
Huang and Chyng	Performance characteristics of integral type solar-assisted heat pump	[24]
Chyng, Lee and Huang	Performance analysis of a solar-assisted heat pump water heater	[25]
Chow et al.	Modeling and application of direct-expansion solar-assisted heat pump for water heating in subtropical Hong Kong	[26]
Kong et al.	Thermal performance analysis of a direct-expansion solar-assisted heat pump water heater	[27]
Kong et al.	Modeling evaluation of a direct-expansion solar-assisted heat pump water heater using R410A	[28]

Indirect expansion (IDX) systems are any systems that do not use the solar thermal collector as an evaporator for the heat pump, and as such are incredibly more diverse in potential configurations than DX systems. Since the evaporator's heat source is not the ambient environment, indirect systems can outperform DX systems when outdoor temperatures are low, making them attractive for study in cooler climates. This thesis will therefore concern itself with IDX systems, leaving further study of DX systems to the reader.

Much has been done using IDX systems, especially in Canada. Rad et al. incorporated solar thermal collectors to a ground-source heat pump system and found that the borehole heat exchangers could be substantially shortened thanks to the boost provided by the collectors, but that the initial capital cost was relatively unaffected [29].

Freeman simulated an IDX-SAHP for comparison to electric water heating, traditional SDHW and to an air-source heat pump water heating system and found general improvements in all areas of performance evaluation [30]. Bridgeman later built this IDX system to validate Freeman’s results using the experimental system to validate and adjust the model [31], confirming Freeman’s conclusions. The system studied by Freeman, Bridgeman and Harrison is shown in Figure 2.4.

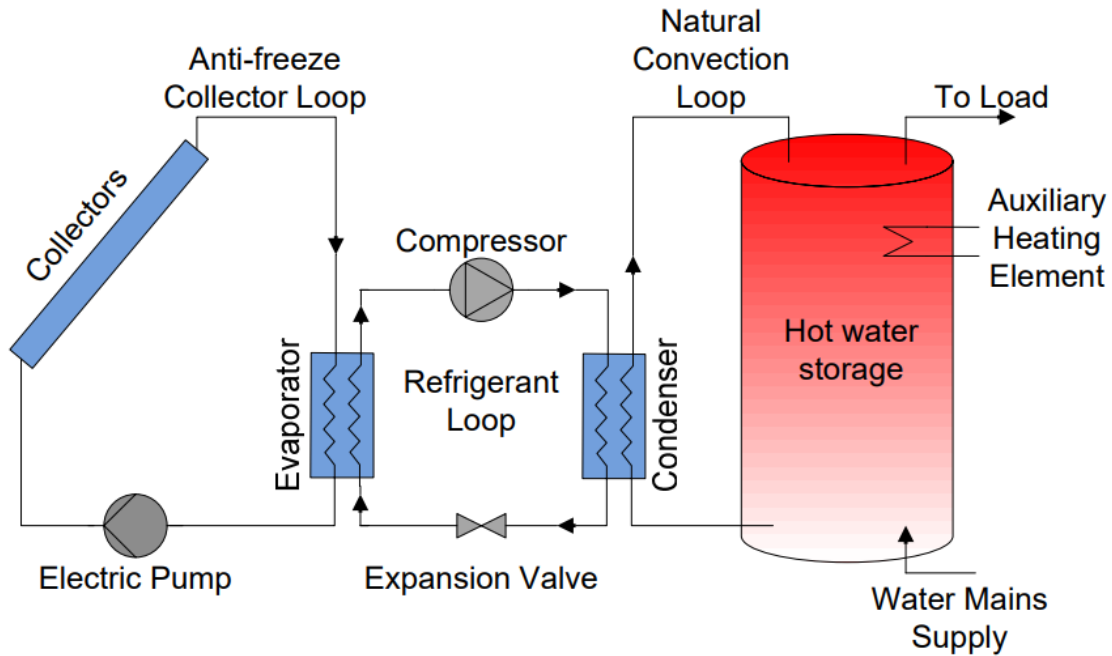


Figure 2.4 - System schematic for an indirect-expansion SAHP [31]

The conclusions from these studies include a note that low compressor speeds seem to improve system performance in general, results which have been found for DX systems as well [31, 32]. Nuntaphan studied the effects of refrigerant, water flow rates and storage volume [33], after performing a study to select a high-performance mixture of several refrigerants for the heat pump. Recent work with IDX systems is detailed and diverse. While beyond the scope of this thesis, one technology of rising interest is solar photovoltaic/thermal (PVT) collector integration. These SAHP systems use PV cells to generate the electricity needed for heat pump operation to further

increase the amount of free, renewable energy used. What to do with excess electricity generated is a particularly common subject for experiments using PVT technology, with some researchers immediately sending it to an electric water heater [34], storing it in batteries for later use [35], or even modifying the heat pump design so that it can increase its heat delivery capacity when extra solar electricity is available [36]. These sources are presented in Table 2-3.

Table 2-3 - Studies of PVT SAHP systems

Author(s)	Title	Ref. #
Aguilar, Aledo and Quiles	Experimental study of the solar photovoltaic contribution for the domestic hot water production with heat pumps in dwellings	[34]
Calise, Figaj and Vanoli	A novel polygeneration system integrating photovoltaic/thermal collectors, solar assisted heat pump, adsorption chiller and electrical energy storage: Dynamic and energy-economic analysis	[35]
Fine, Friedman and Dworkin	Detailed modeling of a novel photovoltaic thermal cascade heat pump domestic water heating system	[36]

Other interesting technologies include chemical and latent storage techniques, such as phase-change material (PCM) tanks [17]. While such storage techniques show promise for seasonal storage applications (due to an absence of self-discharge) [17], they can also be used for diurnal applications, storing excess heat for use by the heat pump when collector gains would otherwise be insufficient for system operation [37]. Cost and material longevity persist as barriers for such systems, but the study of such materials may eventually lead to storage solutions that render hot water tanks irrelevant [17].

The aforementioned systems are advanced both in technologies employed and system complexity - despite the interest in incorporating these cutting-edge technologies to SAHP systems, validation of such systems is difficult and prohibitively expensive. There is much left to explore in the field of indirect-expansion SAHPs without tying in additional capital costs. Many studies, such as those

highlighted in Table 2-4, consider the use of solar heat energy, and thereby the heat pump, for hot water and space heating simultaneously in an attempt to maximize solar gains and minimize electricity/fuel costs [16, 38-40].

Table 2-4 - Some notable studies of combined-function SAHP systems

Author(s)	Title	Ref. #
Wang and Qi	Performance study of underground thermal storage in a solar-ground coupled heat pump system for residential buildings	[16]
Calise, Figaj and Vanoli	A novel polygeneration system integrating photovoltaic/thermal collectors, solar assisted heat pump, adsorption chiller and electrical energy storage: Dynamic and energy-economic analysis	[35]
Fraga et al.	Solar assisted heat pump system for multifamily buildings: Towards a seasonal performance factor of 5? Numerical sensitivity analysis based on a monitored case study	[38]
Chu	Evaluation of a Dual Tank Indirect Solar-Assisted Heat Pump System for a High Performance House	[39]
Chu et al.	Modeling of an Indirect Solar Assisted Heat Pump System for a High Performance Residential House	[40]
Eicher et al.	Solar Assisted Heat Pump for Domestic Hot Water Production	[41]

Even these systems are subject to countless design decisions that impact performance, and though the technologies used are well-understood, the complex interactions of various building energy systems using a single SAHP unit makes them very difficult to validate. Full-scale operational prototypes are often needed for experimental studies which can take years to carry out.

While this literature review recognizes several branches of study in the SAHP development field, the possibilities for such systems are endless, as is the variety of work performed to study them. Several helpful papers have been written to summarize what has been or is being done, to help researchers guide their work. Chu and Cruickshank identify the variety and inconsistency of this field, while suggesting that a series configuration of components would be best suited for Canadian applications [42]. Buker and Riffat later provide a broad collection of work [43], attempting to

standardize the classification of systems and recommending a visualization method put forward by Frank et al. [44]. Most recently, Wang et al. provided a review which categorized the types of studies performed and pointedly highlighted current opportunities for growth and improvement in this field [45].

An observational reader might note that many IDX systems in these studies are fully dependent on the heat pump – that is, heat delivery does not occur unless the heat pump is operational, such as that shown in Figure 2.4. Some systems, such as those studied by Terrell in 1979 or Çağlar and Yamali in 2012 [19, 46], even go so far as to use the heat pump as the mode of energy delivery from storage to the load for space heating applications. Rather than rely on the heat pump, the Solar Thermal Research Lab at the University of Waterloo has pursued a course of design that simply augments the performance of an otherwise traditional SDHW system. In this system, the heat pump is only used when traditional SDHW collection is ineffective or non-ideal, greatly reducing the annual runtime of the heat pump. The sizing of a heat pump for such a system would be much smaller than for a heat pump-dependent system since it is only needed for times when heat exchanger use is in-viable.

The origins of this thesis are found in the work by Sterling, which simulated two different IDX systems in comparison to an SDHW and an electrically heated system [47]. One of Sterling's SAHP systems, called the solar-side system, used a heat pump to lower the collector inlet temperature and boost the collector outlet temperature (see Figure 2.5 for a schematic representation) [47]. This would enhance the efficiency of the collector without negatively impacting collector outlet temperatures. Energy was then transferred via heat exchanger to a storage tank. Such a system, while energy effective, is very difficult to validate experimentally due to the relatively minute heat pump capacity used.

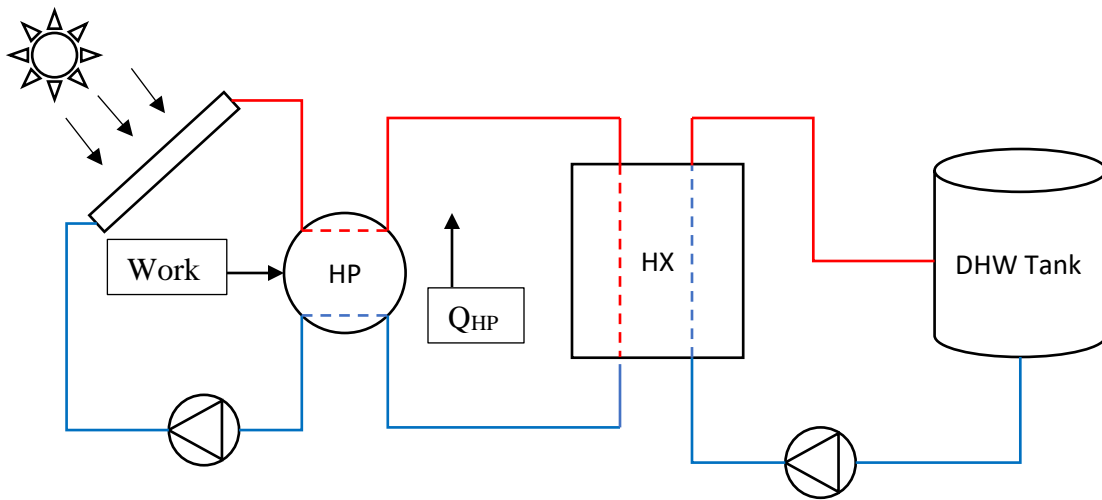


Figure 2.5 - Solar-side SAHP system [47]

The second system studied was a dual-tank system that transferred solar energy directly into a “float tank” of uncontrolled temperature [47]. When the temperature-controlled domestic water tank needed energy, the system could use a heat exchanger or a heat pump to transfer energy from the float tank, with the configuration illustrated in Figure 2.6 [47].

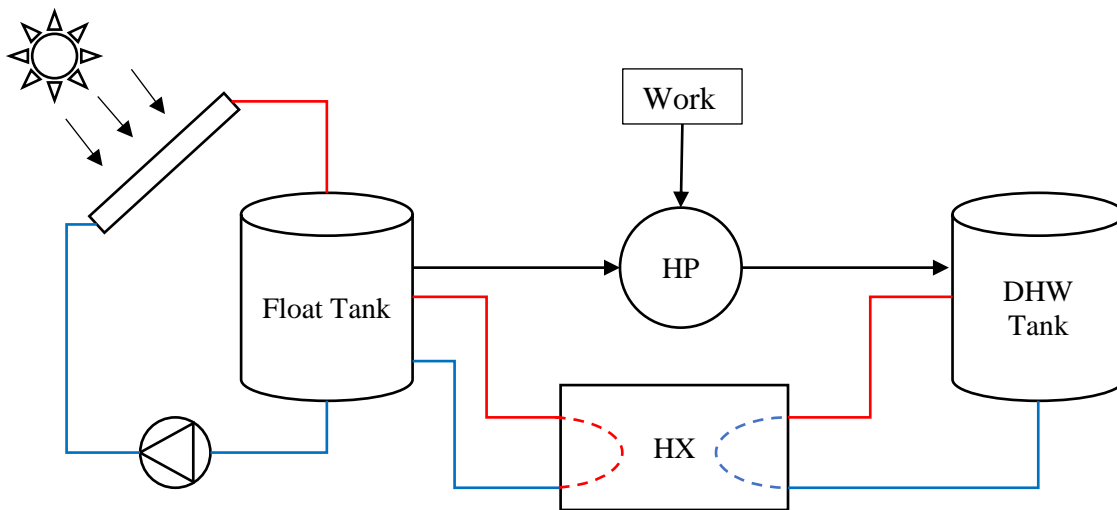


Figure 2.6 - Dual-tank SAHP system as studied by Sterling and Bannister [47]

The ability to store low-quality energy in the float tank, which was then available on-demand, greatly enhanced system performance versus traditional SDHW systems [47]. This dual-tank system was later built in a laboratory setting by Wagar and Bannister; the control and detailed study of it was the subject of Bannister's PhD thesis [13, 14]. While the system continued to prove highly effective, the capital costs involved and high floor space requirements rendered it economically unjustifiable for single-family dwellings in Canada [14]. A later study dedicated solely to the techno-economic feasibility of retrofitting this system confirmed this conclusion, recommending that incentive measures be used to encourage homeowners to install it [48].

Wagar studied a single-tank system, called a dual-side system since it had two distinct fluid flow loops between the collector and storage tank [13]. The system used either a heat pump or a heat exchanger to transfer heat from one loop to the other, thereby delivering it from the collector to the storage tank [13]. Figure 2.7 shows a schematic of this system.

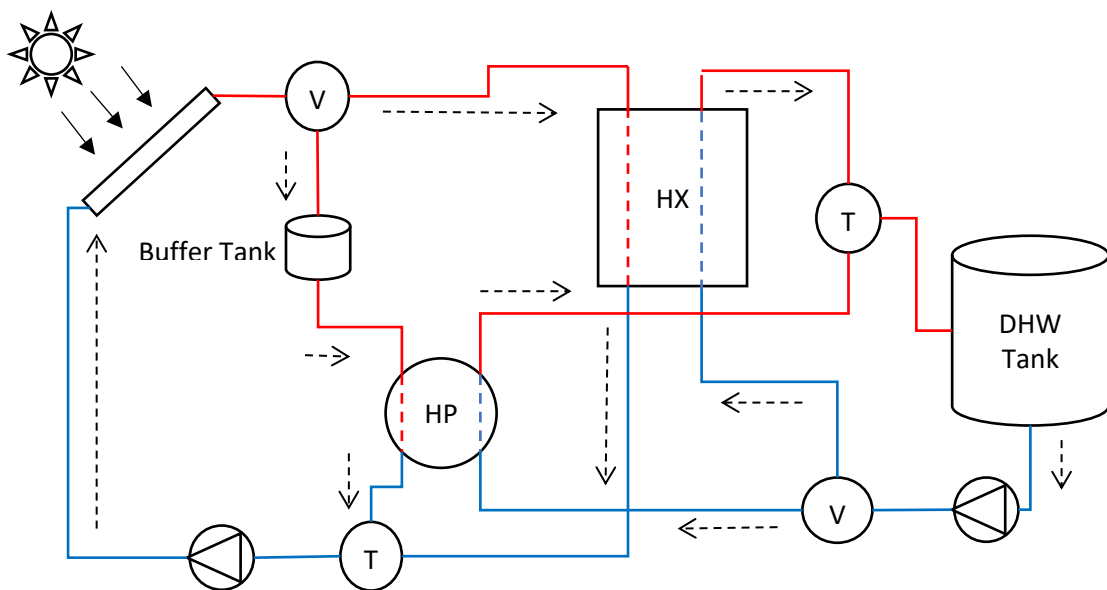


Figure 2.7 - Dual-side SAHP system as studied by Wagar [13]

While it is expected that solar gains and annual energy costs would be negatively affected by eliminating the float tank, the capital cost reduction may offset this enough that the system could be viable. In Wagar's thesis, a TRNSYS simulation of the system is validated using a physical construction of the system in a lab. However, the control scheme used is deliberately simple so that system behaviour is predictable and validation is readily achievable, and it is recommended that a more involved control scheme be employed [13]. Such a control scheme is featured at the core of this Master's thesis.

Another observation of Wagar was that due to high temperatures present at the bottom of the storage tank near the end of the day, late afternoon sunlight would often go to waste [13]. The control scheme would prevent heat exchanger use, while the high temperature triggered the heat pump's safety cut-off, resulting in a static system that is later forced to use auxiliary heaters after sundown [13]. Therefore, in addition to the operating modes of delivery via heat exchanger and delivery via heat pump, a novel system modification is employed here. In the dual interest of capitalizing on peak sunlight hours and increasing the operating window of the heat pump, a third operational mode is introduced by which the heat exchanger outlets are directed to the heat pump inlets. In this thesis, this will be called "series" operating mode, since the HX and HP are used in series with each other. During peak hours, when solar gain may exceed the heat transfer abilities of the HX, the HP can then be used to further increase the temperature lift of the system. During shoulder hours, especially in the late-day, when high inlet temperatures would prevent operation, this series mode can be used to temper the HP inlet flows if the solar-side loop is cooler than the load-side loop. The HX rejects heat in this case. However, if this heat rejection then provides conditions which are operable for the heat pump, the system would experience a net heat gain since the heat pump's delivery capacity far exceeds that of the heat exchanger.

While the system studied in this thesis makes no use of PCM or PVT technology, and there is no dual-purpose integration with space heating or cooling considered, the array of operating modes possible with this system is not found in any other studies at this time. Additionally, a focus on system control builds upon the work of Wagar, which re-emphasized the need for an intricate or otherwise interactive control scheme. The goal of this thesis, then, is to develop a superior control scheme, to demonstrate the performance advantage of a three-mode IDX-SAHP versus other dual-tank systems and to identify appropriate settings for key design characteristics of the system.

Chapter 3 – System Descriptions and TRNSYS Models

3.1 - Systems to be Compared - Descriptions

This thesis is primarily concerned with the energy and temperature performance of a specific SAHP system, configured so that it can operate as SDHW, SAHP or a series combination of both. To evaluate the quality of its performance, some benchmark systems are required for comparison. The several systems to be examined are described here.

3.1.1 - System 1 – Electric Hot Water (EHW)

The basis for all comparisons is to be an electric hot water system, one which depends entirely upon electric auxiliary heating for energy, such as that shown in Figure 3.1. Another non-solar design that could be considered uses natural gas as a heat source. Natural gas is substantially cheaper than electricity; no solar-assisted system studied here can financially compete with a gas system. However, natural gas is a finite resource, is not available in all locations, and produces more greenhouse gases than most electricity sources by an order of magnitude. For these reasons, natural gas systems are not considered in this thesis.

The electric system consists of a hot water tank, with a mains inlet near the bottom and a hot water draw outlet at the top, and an auxiliary heater mounted near the middle of the tank. The thermostat to control the heater is near the top of the tank to prevent excessive heating. Due to the nature of the system, a strong and consistent stratification pattern emerges in the hot water tank, which is used as the initial condition for all simulations.

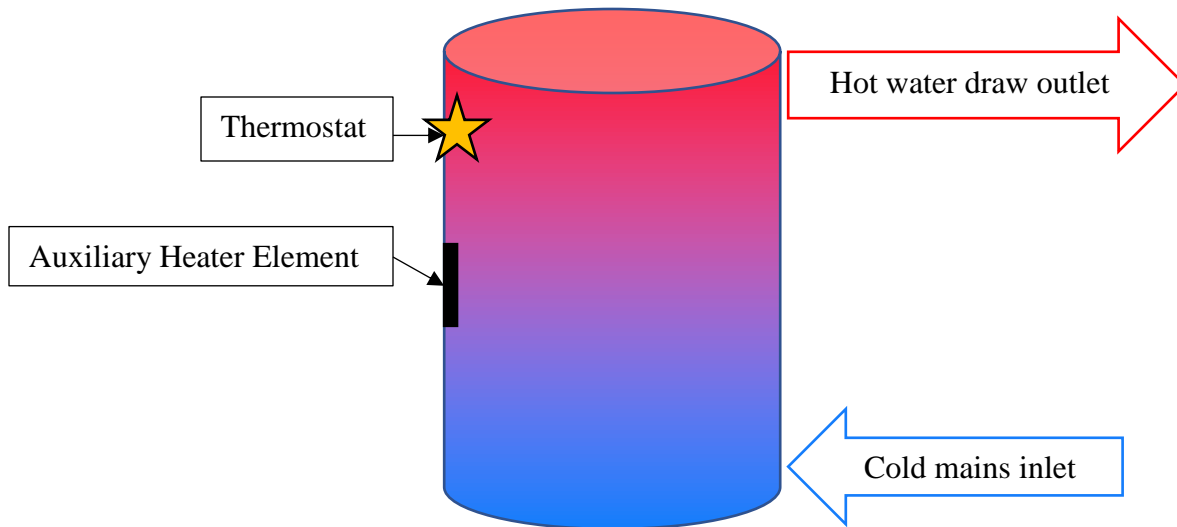


Figure 3.1 - Single-element electric hot water tank system

There are very few influences of weather on this system; the seasonal change in mains water temperature, which is one such influence, is accounted for in the simulations. There are no other physical components required. The energy consumed by this system throughout a year will provide the benchmark for all comparisons and evaluations of energy performance, and the tank model, its configuration and the draw pattern used here will be used for all other systems as well. The only exception will be the addition of inlet and outlet ports for the solar system.

3.1.2 - System 2 – Solar Domestic Hot Water (SDHW)

The second benchmark system is a classic SDHW-style system. An extra set of inlet/outlet ports is added to the tank, with the outlet at the very bottom and the return inlet near the middle of the tank. The system, illustrated in Figure 3.2, consists of a solar-side loop and a load-side loop. The solar side has a flat-plate solar collector and a circulation pump; the load side only needs a second pump. The sides are connected by a counter-flow heat exchanger. This system will operate in solar mode when collection is possible, based on irradiation, ambient temperature and tank temperature

conditions. When solar collection is not possible or insufficient, the backup electrical auxiliary heater will be activated.

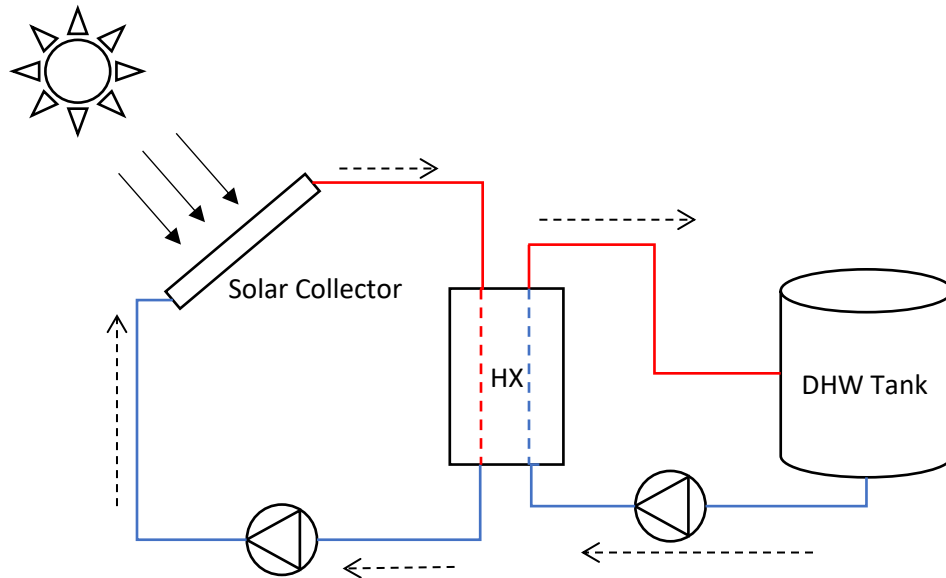


Figure 3.2 - Traditional SDHW system

This system will serve as another energy performance benchmark for the SAHP system. While comparison to the electric system describes the overall benefits, one must also compare the SAHP system to its primary competition. Due to additional capital costs required for an SAHP system, justifying its use demands that it outperforms the traditional SDHW well enough to recoup the extra investment.

3.1.3 - Systems 3 and 4 – Heat Pump-Only and Series Configuration-Only (HP and SRS)

The traditional SDHW system may also be referred to as a HX-only system – its only available method for transferring solar energy from the collector to the storage medium is through a heat exchanger. The HP-only and series-only (SRS-only) systems are defined in the same way.

The HP-only system replaces the HX with a HP, and adds a buffer tank to the solar side of the system to increase its thermal mass and reduce HP cycling frequency. This system cannot collect

solar energy unless the heat pump is operating, which is a common trait among IDX-SAHP systems studied today. However, most systems of this type use a dual-tank design – essentially replacing the buffer tank with a full-size water storage tank [30, 31]. Figure 3.3 illustrates the system; note the absence of a full-size tank on the solar side.

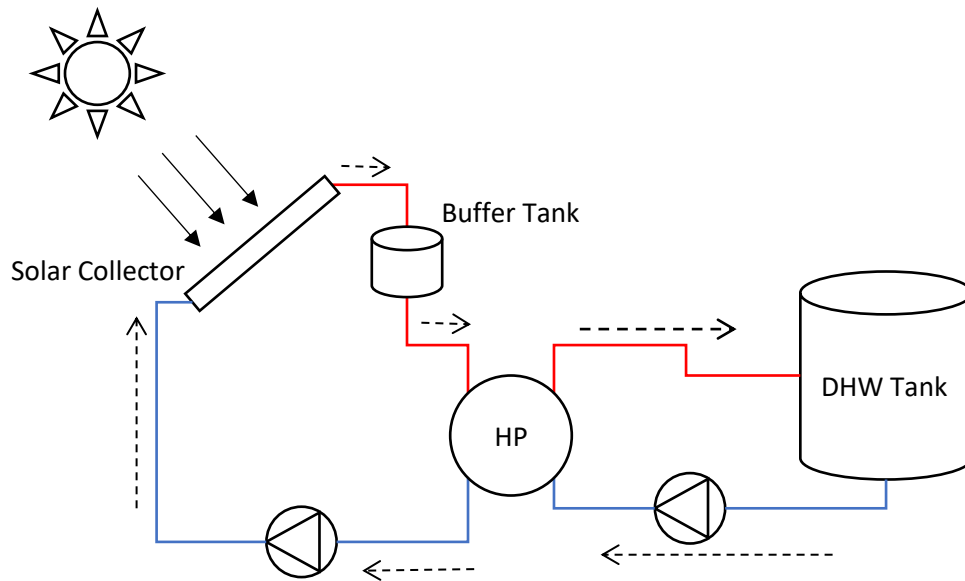


Figure 3.3 - HP-only system (single-tank IDX SAHP)

Here, HP-only is presented as an alternative, lower-cost single-tank design for comparison to the 3-mode system. It is not presented here as a comparison to other “HP-only” SAHP systems studied by others. Its control will be based on the heat pump’s safe operating limits and a prediction of its ability to operate for a given minimum cycle time without freezing out the solar loop.

The SRS-only system provides the same critical hardware as the 3-mode system without any of the intricate controls. Shown in Figure 3.4, this design passes bottom tank water and heated solar collector water through the same counter-flow heat exchanger and then through the heat pump, which is operational and actively moving heat to the load side. In this version of the system, there is no option for running HX-only or HP-only.

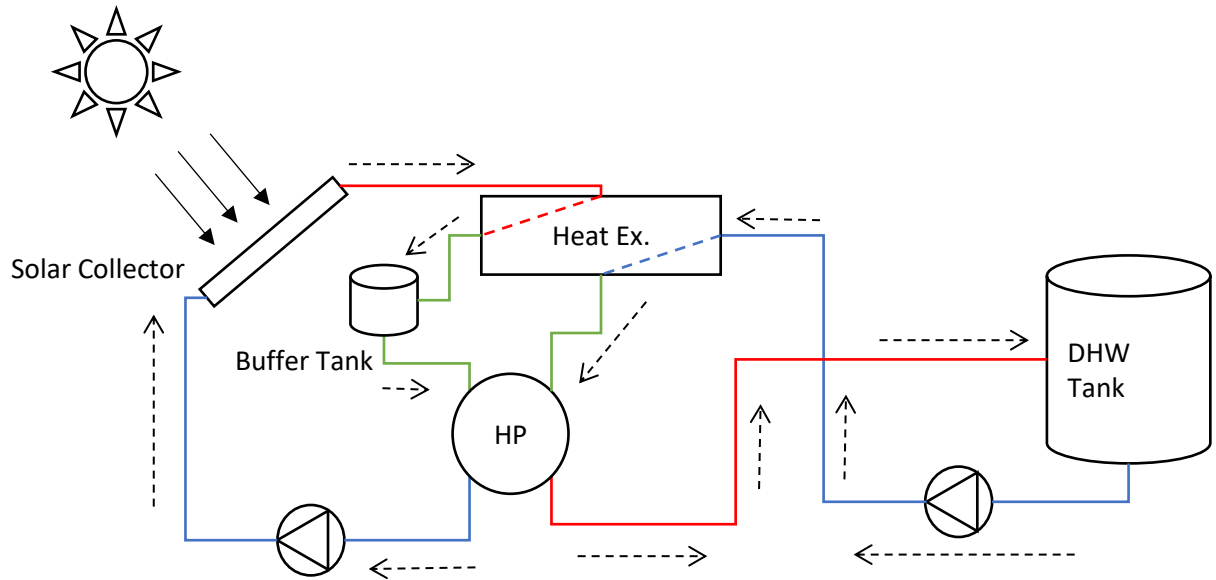


Figure 3.4 - SRS-configuration IDX SAHP, with a heat pump connected in series to an upstream heat exchanger

During times of high solar gain, this system is expected to enhance solar collection efficiency by driving down the temperature of the solar loop and to maximize the heat transfer into the tank by augmenting the HX's performance. During periods of low gain or high temperatures in the bottom of the tank (45 C or greater), this system is expected to extend operating hours and enhance collection compared to the other single-mode systems. At such times, HX potential is low – it may be the case that the load-side water is hotter than the solar side, resulting in a “loss” through the HX. Additionally, sufficiently high tank temperatures can prevent heat pump operation if the temperature exceeds the safe operation limit at the condenser inlet. With this configuration, however, any “loss” through the HX can be regained by the HP, which has much higher heat transfer capacity. Additionally, if heat is “lost” through the HX, the condenser inlet fluid may be tempered enough that HP operation is safe, allowing the system to operate when it would be shut down otherwise. Finally, in cases where the HX “loses” energy, the performance of the heat pump is dually improved by higher solar-side and lower load-side temperatures, resulting in net heat gains higher than the HP could have achieved on its own.

The system presents advantages at certain times, but these same features can work against it at others. It is not expected to necessarily out-perform the HX-only or HP-only systems, but it will be useful for comparison purposes, and it is expected to enhance the performance of the 3-mode dual-side SAHP system relative to the 2-mode system studied by Wagar. The system control will be based on safe heat pump operating limits, anticipated operating cycle time and the expected net COP of operation.

3.1.4 - System 5 – Dual-Side Solar-Assisted Heat Pump

The dual-side system was classified by Wagar to refer to the unique solar and load side flow loops outside of the storage tank; most systems only have a solar loop, with a heat exchanger inside the tank in place of a load-side flow loop [13]. In this system, solar energy can be delivered to the tank through a heat exchanger as if it were a traditional SDHW design when conditions allow [13]. When the heat exchanger cannot be effective, the system can switch into heat pump mode, operating the heat pump to deliver solar energy to the storage tank [13]. Figure 2.7 illustrated this system, with dotted lines to indicate flow paths only used in heat pump mode.

The 3-mode dual-side IDX system studied in this thesis is based on this 2-mode dual-side system, with one goal being to improve the quality of the control scheme for thorough evaluation of the system's potential. Therefore the 2-mode system that was studied by Wagar is to be used to quantify the system improvement provided by introducing a third operating mode. The control of this system will be similar to that used for the 3-mode system, except without the option to consider the series mode performance. This will enhance the value of the comparison made as opposed to using Wagar's simplified control scheme for the dual-side system.

3.1.5 - System 6 – 3-Mode Dual-Side SAHP

All the systems described so far – electric, SDHW, SAHP, SRS-configuration SDHW/SAHP and HPA-SDHW – will be simulated and evaluated as benchmarks for comparison to the system that is the focus of this thesis, the series-configuration indirect expansion solar-assisted heat pump system (3-mode IDX SAHP). Thus far it has been referred to as the 3-mode system; this notation will continue to be used. Figure 3.5 illustrates this system. At flow junctions, either a mechanized flow diverter valve (circled V) or a two-inlet tee piece (circled T) is placed.

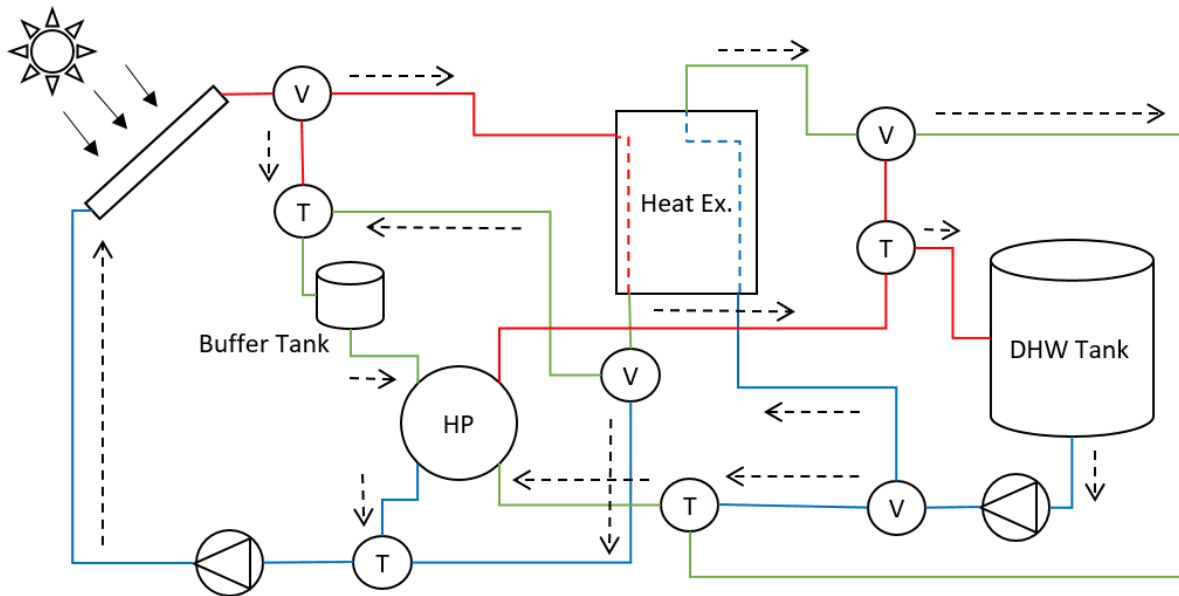


Figure 3.5 - 3-mode SRS-configured dual-side IDX SAHP system

In HX mode, the valves direct flow directly towards the heat exchanger, and from there directly towards the tank or collector. Likewise, in HP mode the valves direct flow directly towards the heat pump, from which flow naturally returns directly to the tank or collector. In SRS mode, flow is directed towards the heat exchanger. HX outlet flow is then directed towards the HP, from which flow naturally returns directly to the tank or collector.

The control of this system is to be carried out by a complex scheme, designed to follow Wagar's recommendation for flexible decision-making, to consider Kaygusuz's recommendation for a non-rigid set point temperature and to incorporate elements of predictive control, a philosophy pursued far more rigorously by Wanjiru [13, 18, 49].

3.2 - TRNSYS Model and Components

The data used in this thesis for evaluating and comparing these hot water system designs is generated using the Transient System Simulation Tool (TRNSYS) 17 simulation program. In TRNSYS, physical components and influential forces are represented by mathematical models called "Types". Each "Type" (model) contains the relevant mathematical equations for processing the given inputs and parameters into outputs; TRNSYS performs the operations required to obtain the outputs, then delivers them as inputs to whichever other models might need them. The program iterates through solutions in each timestep until convergence is reached, at which point the outputs are recorded and the timestep advances.

The 3-mode system that is the focus of this thesis, as it appears in TRNSYS, is shown in Figure 3.6. Note that utility components, which input forcing functions, perform data management and output results, are not shown for the sake of clarity. From left to right, the components represent the solar or source loop, the load side loop and the draw side. Notable components in the solar loop are the solar thermal collector and buffer tank. The heat pump and heat exchanger serve as a bridge between the solar and load sides and as such are placed in-between the two loops. The load side loop features the domestic hot water storage tank, the most important feature of the model. Finally, to the right of the tank are components which act on the draw side of the system, including a thermostatic mixing valve for limiting the maximum drawn water temperature. The following sections identify the models used in these simulations, most of which are available in the standard

TRNSYS package or in the expanded Thermal Energy System Specialists (TESS) library and are open source to TRNSYS users [50, 51].

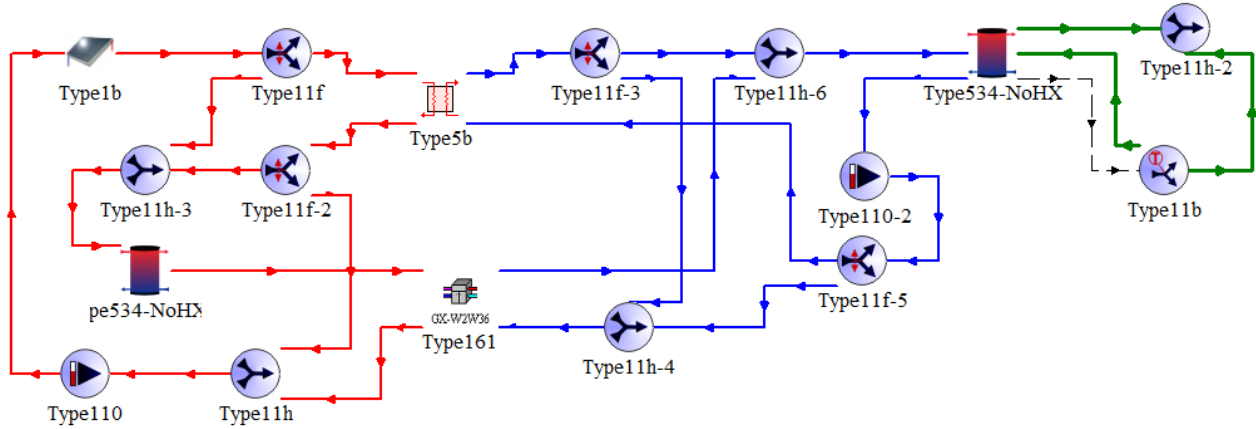


Figure 3.6 - TRNSYS simulation model for the 3-mode SAHP system

3.2.1 – Solar Thermal Collector (Type 1b)

Type 1b represents a solar thermal collector. It takes inputs regarding the fluid and weather conditions, and uses a quadratic efficiency model with a 1st-order incident angle modifier to produce outputs such as fluid temperature and useful solar heat gain. Wagar used test data from the Solar Rating & Certification Corporation (SRCC) for the Viessmann Vitosol 100-F flat plate collector for their validation simulations; Bannister used the same data [13, 14, 52]. For consistency, the same performance parameters are therefore used here. Table 3-1 lists these as seen in TRNSYS. The simulations in this work use a single solar collector, though the option to connect several in series is available. Note that the fluid is assumed to be water; while a real, practical system would likely use a water/glycol antifreeze mixture, the experimental setup at the Solar Thermal Research Lab uses water – validation will require that the simulations use the same working fluid.

Table 3-1 - Type 1b - Solar thermal collector parameters

Parameter	Value	Unit	Source
Number in series	1	-	
Collector area	2.494	m ²	[13, 52]
Fluid specific heat	4.19	kJ/kg.K	Water (approx.)
Tested flow rate	72.17	kg/hr.m ²	[13, 52]
Intercept efficiency	0.769	-	[13, 52]
Efficiency slope	3.614	W/m ² .K	[13, 52]
Efficiency curvature	0.01358	W/m ² .K ²	[13, 52]
1st-order IAM	0.32	-	[13, 14]
2nd-order IAM	0	-	[13, 14]

3.2.2 – Variable Speed Circulation Pump (Type 110)

The Type 110 model represents a variable-speed pump. These models use rated conditions, a 0-1 analog control signal and a polynomial curve to determine the output flow rate and power consumption. Wagar experimentally determined the rated power and linear power correlation coefficients that would match the simulated pumps to the real models [13]. These values are used in this work for consistency, but should be re-evaluated when re-construction of the ETU is complete. Table 3-2 lists the parameters used in the TRNSYS simulation.

Table 3-2 - Type 110b - Variable speed circulation pump parameters [13]

Parameter	Value	Unit
Rated flow rate [53]	2840	kg/hr
Rated power (solar-side)	76.13	W
Rated power (load-side)	51.47	W
Motor heat loss fraction	1	-
Number of power coefficients	2	-
Power coefficient-1 (solar-side)	.2322	-
Power coefficient-2 (solar-side)	.7678	-
Power coefficient-1 (load-side)	.7288	-
Power coefficient-2 (load-side)	.2712	-
Total pump efficiency	.07	-
Motor efficiency	.25	-

3.2.3 – Tee-Fittings (Type 11)

The Type 11 models represent various flow diverting/converging pieces. Three kinds are used in this work: 11b, a tempering valve, 11f, a motorized flow diverter valve, and 11h, a tee piece. The tempering valve is used to limit all draw temperatures to be within a safe upper limit. The parameters are set as per Bannister, shown in Table 3-3 [14].

Table 3-3 - Type 11b - Thermostatic mixing valve parameters

Parameter	Value	Unit
Set point temperature	60	C
Tempering valve mode	4	
Number of oscillations allowed	7	

Type 11f is used to send a user-controlled proportion of the inlet flow to one of the outlets; the remaining flow exits through the second outlet. In this work, the control signal is either 0 or 1 (binary) – the flow is never intended to be split between parallel paths. These are used to direct flow down the appropriate paths as per the operating mode dictated by the controller. These diverted paths are rejoined using Type 11h tee pieces, which fully mix two inlet flows into a single outlet. Once again, in this work, there is only ever inbound flow from a single inlet at a time.

3.2.4 – Hot Water Storage Tank (Type 534 – No HX)

Type 534 represents a multi-node stratified hot water tank, and is unlike the Type 4f or Type 60d models used by Wagar and Bannister [13, 14]. This model is not in the standard TRNSYS library but is instead accessed from the TESS library. A study at Carleton University compared the performance of several available tank models in TRNSYS to the behaviour of a real tank, finding Type 534 to be the most suitable [54]. Type 60 showed mathematical instabilities and Type 4 showed an inability to realistically model a de-stratifying tank; Type 534 performed well in both aspects [54]. Therefore, this thesis deviates from Wagar and Bannister by replacing the Type 60d

Buffer Tank and Type 4f Domestic Hot Water Tank models with Type 534 – No HX Hot Water Tank models to improve the accuracy achievable in the simulation.

De-stratification is especially important for SAHP applications. Wagar and Bannister both note the effects that de-stratification has on system behaviour and overall performance [13, 14]. Due to the nature of the system, when operational there is a downwards flow that establishes within the tank, from the hot solar water return inlet to the cold water outlet. This results in very warm temperatures near the bottom of the tank that are nearly uniform through the bottom half of the tank – unless a draw occurs during operation, which introduces large mixing effects. The Type 534 routine allows adjacent nodes to be temperature-inverted when this happens; the user specifies the rate at which inverted nodes mix and re-balance, whereas other models instantaneously mix the two. This allows the model to more realistically represent a de-stratifying tank than the Type 4f when information about the rate of mixing is available. In this thesis, the value of -100 kg/hr is used to cause instantaneous mixing of adjacent offending nodes, as an opportunity to experimentally verify a realistic inversion flow rate is not available [55].

A complication of using the Type 534 model is the loss of a built-in auxiliary backup heater. While other models allow the user to specify a heater location, capacity and set point temperature as parameters, Type 534 accepts auxiliary heat as an input. This places the burden of auxiliary heater control onto the user, which is complex but recommended by Wagar as an alternative to the autonomous Type 4f model [13]. In this work, the controller is designed to output both an auxiliary heater on/off signal and an auxiliary heat rate, which is input to the appropriate tank node. In a real system, the control signal would control a heater placed inside the tank.

This work uses an 80-gallon domestic hot water tank with 20 equal-sized nodes. Edge loss coefficients are interpolated from those established by Wagar [13]. The locations of the inlet and

outlet ports, as well as the auxiliary heater, are determined based on the locations of the associated components on the ETU. A smaller 4-gallon tank called the “buffer tank” is used to provide thermal mass to the solar side of the system, helping extend the heat pump’s operating cycles. The buffer tank is modelled as a single, fully-mixed node. Table 3-4 lists the values used for the DHW tank; Table 3-5 lists those used for the solar-side buffer tank.

Table 3-4 - Type 534-No HX - DHW Tank parameters

Parameter	Value	Unit	Source
Number of nodes	20		Arbitrary
Number of Misc. Heat Flows	1		For auxiliary input
Tank Volume	80	Gal	[13]
Tank Height	60	in	[13]
Top, Bottom Loss Coefficient	0	W/m ² .K	Assume insulated
Edge Loss Coefficient for Nodes 1 - 10	1.18	W/m ² .K	[13]
Edge Loss Coefficient for Node-11	1.34	W/m ² .K	[13]
Edge Loss Coefficient for Node-12	1.66	W/m ² .K	[13]
Edge Loss Coefficient for Node-13	1.66	W/m ² .K	[13]
Edge Loss Coefficient for Nodes 14 - 17	1.935	W/m ² .K	[13]
Edge Loss Coefficient for Node-18	2.027	W/m ² .K	[13]
Edge Loss Coefficient for Node-19	2.21	W/m ² .K	[13]
Edge Loss Coefficient for Node-20	2.21	W/m ² .K	[13]
Additional Thermal Conductivity	0	W/m.K	Assume negligible
Entry Node 1	17		[13]
Exit Node 1	1		[13]
Entry Node 2	8		[13]
Exit Node 2	20		[13]
Node for Miscellaneous Heat Gain (Aux Heater Location)	8		[13]
Edge Loss Temperature for All Nodes	20.0	C	Room conditions
Inversion Mixing Flowrate	-100	kg/hr	[55]

Table 3-5 - Type 534-No HX - Buffer Tank parameters

Parameter	Value	Unit	Source
Number of Tank Nodes	1	-	[13]
Tank Volume	4.05	gal	[13]
Tank Height	15	in	[13]
Top Loss Coefficient	5	W/m ² .K	[13]
Edge Loss Coefficient for Node	5	W/m ² .K	[13]
Bottom Loss Coefficient	5	W/m ² .K	[13]

3.2.5 – Heat Exchanger and Heat Pump (Types 5b and 161)

The simulated system includes two heat transfer devices. Type 5b models a counter-flow heat exchanger; positive heat transfer rate indicates heat transfer from the source to the load. Bannister used an overall heat transfer coefficient of 150 W/K, which is also used here for consistency [14]. The use of this model assumes a constant effectiveness regardless of inlet temperatures; should validation work determine that variable effectiveness needs to be considered, a different model could be substituted in. TRNSYS features HX models that accept effectiveness as an input (so a calculator could be used to estimate effectiveness) or, as done by Wagar, a custom HX model could be prepared with the variable effectiveness built-in [13].

The second heat transfer device is the heat pump, a custom model developed by Wagar as Type 161 (an arbitrary name not used in the standard TRNSYS library) [13]. Safety checks which were originally built in to the model have been disabled and are instead integrated to the controller itself to centralize all system controls to the master control scheme. Thus, the model simply applies the inlet conditions to its heat transfer and power consumption equations to determine the output results without performing safe temperature or short cycle prevention checks. These equations were established over a specific range of inlet temperatures by Wagar using a real heat pump [13]. Included in the model is a scaling feature called ScaleHP, which applies a linear scaling model for

heat pump capacity and power consumption to simulate systems using smaller or larger heat pumps [13]. The validity of this feature is unknown and is to be a subject of future studies.

Heat Pump Sizing

The behaviour and performance of the SAHP systems is extremely sensitive to the heat pump design. The capacity of the heat pump, or the nominal rate of heat delivery provided, must be very carefully determined. A large heat pump will draw more energy from the solar loop than can be replaced by gains in the STC; if the device is oversized, it will drop the solar loop to near-freezing temperatures very rapidly and trigger a low-temperature safety warning. This will lead to frequent cycling of the heat pump, causing premature failure of the equipment. This behaviour was observed by Wagar and Bannister when performing validation experiments with a 3.6 kW heat pump [13, 14]. Ensuring that the heat pump is not oversized is even more critical for systems such as those studied in this thesis, where heat pump operation is intended to improve system performance in poor solar conditions.

While it is essential to avoid oversizing, it is also possible to under-size the heat pump. In an under-sized system, the heat pump will not draw enough energy from the solar loop or operate with high enough COP to justify its inclusion in the system. A well-sized system will have the most powerful heat pump that can operate continuously under the design condition – this will depend on the system configuration, the climate of the installation and the designer's choice of the design condition. In this thesis, the condition of interest is typical morning/evening and winter-time solar flux levels of about 500 W/m².

Another parameter to which the system performance is highly sensitive is the choice of operating flowrate. Typical SDHW systems use very low flowrates, allowing for very large temperature gains in the STC and good HX performance. This also helps to preserve stratification in the hot

water storage tank. However, in a SAHP system, a higher flowrate is required to maintain continuous operation. Low flowrates allow the heat pump to induce a substantial temperature drop across the evaporator, quickly triggering anti-freeze safety checks. Therefore, the best system flowrate is the lowest flowrate that still allows continuous heat pump operation under the design condition.

A parametric study is performed by varying the system flowrate and the ScaleHP parameters to determine the best system flowrate and heat pump size for the configuration studied in this thesis. By incrementally studying the effect of flowrate, then ScaleHP, and then the combined effect on many days of the year, appropriate sizing ranges are identified. It is found that flowrates below 175 kg/hr are too low for consistent heat pump operation, while flowrates above 300 kg/hr lead to increased energy costs overall – 200 kg/hr is chosen to provide a buffer above the low flow cutoff. A heat pump scale of 0.3 is found to be ideal – this is a capacity of approximately 1.0 kW. Larger heat pumps are more expensive to operate, for limited gains, and very small heat pumps are unimpactful. Therefore, the heat pump used throughout this thesis will be sized by $\text{ScaleHP} = 0.3$ and the system will be operated with equal solar and load side flowrates of 200 kg/hr.

3.2.6 – Pipe (Type 31)

The simulation considers some effects of piping as well. Type 31 provides a model for pipe behaviour, specifically introduced by Wagar to account for the thermal losses of pipes in the ETU and to account for the thermal mass of the ETU's circulation heater [13]. In this work, the pipe losses are considered negligible, and so modelling heat loss through piping is not required; likewise, they would be simplified or ignored when developing a new system. They still provide a small quantity of thermal mass to the system, however, and contribute to the realism of the simulation.

3.2.7 – Weather Input File Reader (Type 15-2)

The simulation uses a TMY2-format weather file to provide critical inputs to the solar collector, controller and domestic water tank, including various irradiation and temperature data. In this work the weather file used is for Ottawa, Canada. Several parameters regarding the solar system are prescribed here, as listed in Table 3-6. Two notable parameters are the slope of surface and azimuth of surface; in this work, the collector faces due south at an angle of 45 degrees, matching the latitude of its location for optimal performance. The parameter “Tracking mode” identifies the method by which the solar collector tracks the Sun; a value of 1 indicates that the collector is non-tracking.

Table 3-6 - Type 15-2 - Weather profile parameters

Parameter	Value	Unit
Ground reflectance - no snow	0.2	-
Ground reflectance - snow cover	0.7	-
Number of surfaces	1	-
Tracking mode	1	-
Slope of surface	45	degrees
Azimuth of surface	0	degrees

3.2.8 – Utility Routines and Forcing Functions

The TRNSYS model in Figure 3.6 only showed the icons representing physical components of the system, and the connections between them. About a dozen other components are included in the model, however, to represent the forces which act upon it and to utilize the data that the model produces. Table 3-7 lists these models, which are briefly discussed in the remainder of this section.

Table 3-7 - Utility components used in TRNSYS model

Type Number	Purpose of Model
14	Forcing function – inputs to components
25	Print outputs to external files
57	Unit conversion

24	Integrate outputs for cumulative totalling
1629	Track instances of cool water draws

The Type 14 routine is a forcing function model. These models act as inputs to various components to enforce certain conditions upon the system. Two kinds are used in this work – Type 14b, a forced water draw model, and Type 14h, a generic model. The Type 14b is used to enforce a hot water draw profile on the system. The profile is modelled after CSA-A standards [56], but is modified due to timestep constraints so that the total volume of hot water drawn is correct at every instance. The CSA-A standard profile, and the modified version used in simulations for this thesis, are described using Table 3-8.

Table 3-8 - CSA-A daily water draw profile with modified flow rates [56]

Time of Day	Withdrawal at 10 L/min [L]	Modified Flowrate [L/min]
00hrs to 07hrs	0	--
07hrs to 08hrs	5	1
08hrs to 09hrs	25	5
09hrs to 10hrs	0	--
10hrs to 11hrs	45	9
11hrs to 12hrs	0	--
12hrs to 13hrs	5	1
13hrs to 17hrs	0	--
17hrs to 18hrs	5	1
18hrs to 19hrs	15	3
19hrs to 20hrs	30	6
20hrs to 21hrs	20	4
21hrs to 24hrs	0	--
Total Daily	150	--

The simulation also includes three Type 14h models. These are used as inputs for the controller, representing the thermostat set-point T_{Set} , the temperature required for hot water draws T_{target} (which may differ from T_{Set}) and the maximum length of time allowed for reheating the tank without using the auxiliary backup t_{target} . Having these parameters input as forcing functions allows them to change over periods of time, after the suggestion by Kaygusuz; a major component of this work examines the effect of allowing the system set point to oscillate throughout the day [18]. The timing and level of the oscillations are prescribed using the continuous forcing functions.

Type 25 is a printing routine that accepts numerous inputs and outputs them to a specified file. The results are overwritten every time the simulation is run; therefore, they must be copied and saved in a separate document for further study. The simulation includes several such printers for detailed study of individual component performance.

Type 57 is a unit conversion routine. A difficulty when using TRNSYS is that its components are written to use unexpected standard units. A very common example is the use of kJ/hr in place of Watts for power calculations. However, Wagar developed the Type 161 Heat Pump model to use Watts [13]. Type 57 allows the user to read power and heat transfer outputs from the heat pump alongside power outputs from other components.

Type 24 is an integrator, providing a cumulative sum of various inputs over specified periods of time. The routine can start at user-specified times and can be a total integrator or a periodic one. It is used in this work to convert heat transfer rates and power consumption into energy amounts for evaluating the system's energy performance.

Type 1629 is a custom component developed by the author to monitor the temperature of hot water draws. It is equipped with an internal signal generator, and an internal signal counter, to alert a

user to water draw temperatures in a medium or low temperature band. Though the design of the component could be developed so that there are a variable number of thresholds and so that the thresholds can be positive or negative, it is not considered necessary to do so for this work. Here, temperatures below 50 C are considered the low temperature band and the 50-55 C range defines the medium temperature band. The component helps to illustrate the trade-off between energy performance and temperature performance and helps ensure system parameters are established so that the best energy performance is achieved without experiencing unacceptably frequent low-temperature draws.

All components are connected as they would be in the real system, where applicable. Some components, such as the thermostatic mixing valve, need to be connected in a way that schematically disagrees with a real system, but functionally the operation is the same. The system is operated with a 300 s (5 min) timestep, which (as shown by Bannister) comes with negligible loss of accuracy [14]. This also matches up to the desired minimum heat pump cycle time, preventing the system from unintentionally short-cycling the heat pump. It enables day-long simulations to be completed in seconds, while season-length simulations take 170 s and year-long simulations take about 13 min. The bulk of the calculation time is spent on the hot water tanks, but reducing the number of tank nodes is not considered acceptable, since tank model accuracy is considered of paramount importance based on the results obtained by Wagar and Bannister [13, 14].

3.3 - 3-Mode Dual-Side SAHP System Controller

The last, and most important, component modelled in the TRNSYS simulation is the SAHP system controller, which was arbitrarily named Type 1630 by the author (controllers for the other systems, to which the 3-mode SAHP is compared, are Types 1631-1636). This component is the arbiter

through which all operating decisions are made, and acts as the auxiliary backup heater when required. Its development is inspired by the work of Wagar and Bannister. Wagar, who focused on validating a simulated model of the 2-mode system, employed a simple threshold-based controller. Within a certain solar irradiation band, the HP would be used; under higher irradiation, the HX would be used instead. The controller was designed purely for simplicity, to assist in validating the performance of individual component models – the development of a more sophisticated control plan was recommended. The results of validation testing indicated that the bottom tank temperature output by the simulation was not reliable – a better tank model, and a control scheme which relied on this output as little as possible, were also recommended. Bannister, who studied a 2-mode dual-tank system, employed a more advanced set of controls. Their system used a short series of questions to establish the desired operating mode. Though the specific questions used are not applicable in this thesis (since only a single tank is used), the question-and-answer approach will be used.

The controller design is intended to use current system temperatures and current weather conditions, in conjunction with system settings prescribed by the user, to determine the best possible mode of operation for the timestep and to enforce 100% command over the system. It predicts the performance of all critical components of the system using weather data and accessible temperature measurements. The severity of the need for heat in the tank (if any) is evaluated, and numerous conditions which influence the viability of certain operating choices are analyzed. The controller uses tank conditions to determine if it should try to run with minimal electricity consumption or to maximize solar gains, and then applies predicted component performance values to determine the operating mode of choice. All safety checks required by the system are performed within the controller, to ensure that the selected mode is truly the best considered option.

It considers whether it is acceptable to change the operating mode, if its logic has indicated that a change is desirable, and finally determines whether auxiliary operation is required.

By including performance predictions, which are made using the same equations contained in the appropriate component models, and safety checks in the control design, the controller becomes a simulation within a simulation. Should an experimental version of the controller be developed, it would function by essentially simulating the performance of each available operating mode over a short time period, and then selecting the operating mode with the most desirable performance. This prototypical design could be expanded to include iterative predictive models, using fewer and more reliable input values. It could also be advanced to incorporate organic pattern-learning controls for more accurate load readiness settings. The focus of this thesis, which is to evaluate the potential of the SRS configuration, is also the focus of this prototype controller; as such, these advanced controls options are not explored here.

The process by which the control decisions are made is detailed and lengthy. The following describes the operation of the controller in a step-by-step form, with flowcharts included to help visualize the procedure. Figure 3.7 provides a general overview of the process as 10 stages in 3 phases; this section will proceed to discuss the process at the stage level.

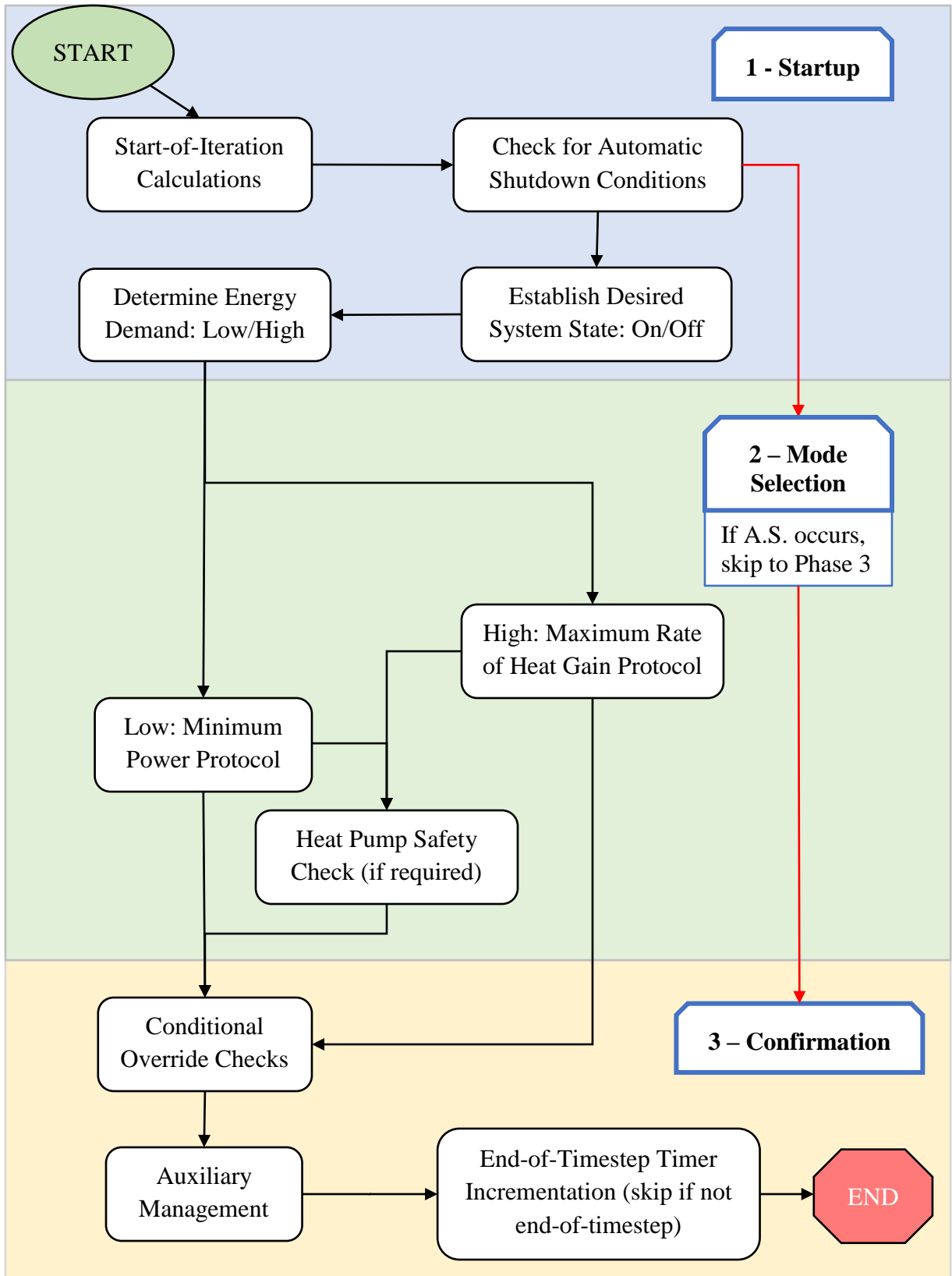


Figure 3.7 - 3-mode controller general flowchart

The end-goal of the controller is to determine which operating mode, if any, is the most beneficial to the system, and if the auxiliary backup heater is needed. The operating modes which can be selected by the controller are:

- Mode 0 – Off. The system does not operate.
- Mode 1 – HX. The system operates as a traditional SDHW design, with heat transfer performed by the heat exchanger alone.
- Mode 2 – HP. The system operates as a single-side SAHP design, with heat transfer performed by the heat pump alone.
- Mode 3 – SRS. The system operates with the HX and HP in series. Heat transfer is performed on a given fluid parcel in each loop in the heat exchanger and then in the heat pump before the fluid is returned to the solar collector or storage tank.
- Mode 4 – WU. The system operates in heat pump warmup mode, an operating mode introduced by Wagar to allow thermal energy to accumulate in the solar loop when the temperature is too low for safe HP operation [13]. The solar-side circulation pump is energized and flow is directed through the heat pump/buffer tank, but the load-side pump and heat pump are not active. Energy gained through the solar collector is temporarily stored in the solar loop through this action. This system uses WU primarily as a safety feature and as a way for the system to properly evaluate and compare the three heat transfer modes in each timestep.

3.3.1 - Phase 1 – Stage 1 – Start-of-Iteration Calculations

In every timestep, the controller performs a series of calculations based on its inputs. It uses input temperatures, weather data and set-point/target temperature profiles to evaluate the needs and capabilities of the system in that timestep. This precedes the decision-making process to ensure

that these amounts are calculated and are available for study in every timestep (for example, if the system decides heat is not required at all and immediately terminates the process, one can still see the potential heat gains for that timestep even though operation was never considered). These calculations are all performed at the start of the code; they will be mentioned or discussed in the following sections at their individual points of application, which occur throughout the control scheme.

3.3.2 - Phase 1 – Stage 2 – Automatic Shutdown Conditions

The first decisions that the controller must make establish if the system should operate at all. If these conditions do or do not exist, the system will skip over the rest of Phase 1 and all of Phase 2, jumping to various stages in the Confirmation Phase as appropriate to enforce the required form of non-operation. The steps of Stage 2 are illustrated in Figure 3.8 and discussed afterwards.

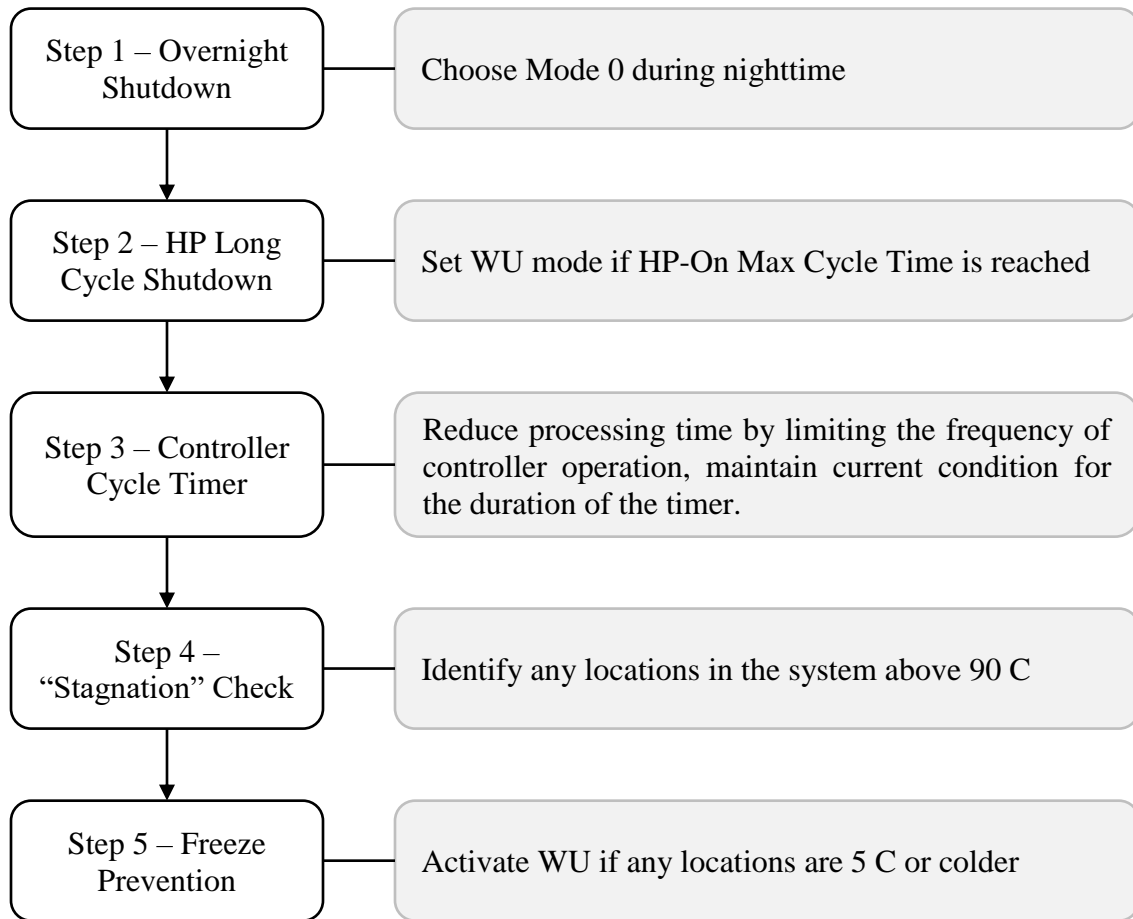


Figure 3.8 - Automatic shutdown checks for Stage 2 of control scheme

During testing, it was found that during particularly warm summer nights, the system would operate in warmup mode for the ambient gains that it could achieve, with little benefit for the system. Therefore, the controller is instructed to keep the system off within a certain range of hours from midnight. The duration of the shutdown is given as a parameter by the user; setting this to 4 hours creates a shutdown from 8 PM to 4 AM. During this time, the controller will not choose modes 1-4 (HX, HP, SRS and WU, respectively), but the auxiliary backup heater may be used if needed.

The controller then considers the effect of long-term heat pump operation. It had been noticed that even during periods of very high solar gain, the system would not switch from HP mode to HX

mode, because the predicted gains via HX mode were too low. This is because of the low-temperature state that HP operation can maintain on the solar side – the controller continues to predict a net loss during HX operation and so continues to use the heat pump. Therefore, the controller now checks to see how long the heat pump has been operating for. An upper limit is prescribed as a parameter; if the heat pump operating time has matched or exceeded this amount, the system enters WU mode and checks to see if backup heat is needed. This forces the heat pump to take a short break, allowing the solar-side to warm up – if HX operation is ideal, the controller will have an easier time identifying this. If not, then it can re-select HP right away.

An intricate control scheme without many dead-bands presents the opportunity for frequent and rapid cycling in-between operating modes, potentially damaging the equipment. Therefore, a timer is introduced to the system to prescribe the minimum amount of time in between decisions. Assuming the timestep of the simulation is smaller than the timer, this line will skip the entire remaining body of code unless the timer has expired. For example, if the timer is 3 minutes and the timestep is 30 seconds, the remaining steps will only be performed every 6th timestep. The controller will continue to perform calculations, and may invoke the previously-mentioned interventions, but it will not make any further decisions and thus cannot make any changes on the 1st-5th timesteps.

The controller includes a feature to indicate times when the solar collector, or another system component, is reaching dangerously high temperatures. The feature is called “stagnation”, though this is a bit of a misnomer – the feature will flag any timestep during which a temperature above 90 C is detected in the system. Future development may wish to introduce a protocol for addressing this, such as running in SRS or WU mode to move heat around or reject it to the environment, but

this is outside the scope of this thesis. In this thesis, it is simply a tool for identifying extended periods of inactivity despite the availability of high sunlight.

Finally, to conclude Stage 2, the controller checks the system for low-temperature conditions. The minimum acceptable temperature is prescribed as a parameter, “ $T_{HP,S,out,min}$ ” – typically set to 5 C, if any point in the system falls below this temperature, the controller chooses WU mode and proceeds to the Conditional Override section. Otherwise, system operation is acceptable and the controller may proceed to Stage 3.

3.3.3 - Phase 1 – Stage 3 – Establishing Desired System State: On/Off

It is at this point that the controller starts making decisions which dictate system behaviour. By now, it is established that the time of day is reasonable for operation and the current time is a good time to evaluate the system’s options. This begins by asking if the tank needs energy. A parameter, T_{Max} , is compared to $T_{tank,top,average}$ (T_{tta}), which is calculated as the average temperature in the top ¼ of the tank. If $T_{tta} > T_{Max}$, the system does not need energy – mode 0 is selected, turning the system off, and the code jumps to Stage 8 – Conditional Overrides.

Next, the controller establishes if solar gain is achievable. The user prescribes the minimum rate of solar gain that warrants operation. This is compared to Q_{STC} , the predicted rate of solar gain if the system is activated. The controller uses the same calculations performed in the Type 1b – Solar Thermal Collector model to estimate the solar gain with high accuracy; parameters entered by the user in the Type 1b model are also entered as parameters here. The calculation, and the full controller code, can be viewed in Appendix A. Solar irradiation levels and inlet temperatures are taken as inputs from the weather file and the appropriate pipe model, respectively, to maximize accuracy. If sufficient solar gain is not achievable, the system turns off. Figure 3.9 illustrates this stage, which concludes Phase 1 – Startup.

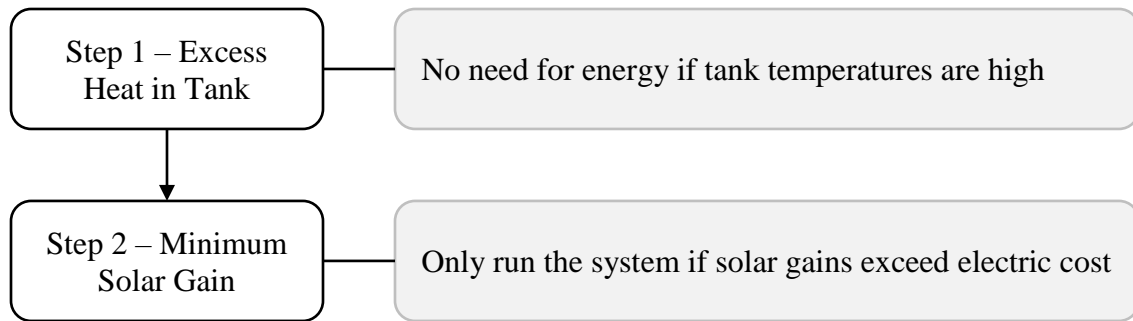


Figure 3.9 - Thresholds checked in Stage 3 of control scheme

3.3.4 - Phase 2 – Stage 4 – Determining Energy Demand: High/Low

Phase 2 – Mode Selection now begins. Here, the controller offers a split philosophy. It can choose to follow a protocol designed to maximize the amount of heat delivered to the tank, or a protocol that minimizes the electricity used to run the system. It has been established by now that the average tank temperature is below T_{Max} . However, the set-point temperature of the system may be much lower than this, and it is necessary to determine if the system is currently below the set-point before proceeding.

The thermostat setting (T_{Set}) is delivered as an input so that the possibility of a variable set-point can be explored. If $T_{Set} < T_{tta}$, the need for energy is considered “low”, and the minimum power consumption (Min P) protocol is applied. Otherwise, the need is “high”; this protocol is passed over and the maximum heat delivery (Max Q) protocol is applied instead.

3.3.5 - Phase 2 – Stage 5 – Minimum Power Protocol

In the calculations step, the controller estimates the power that would be consumed by running in HX, HP and SRS modes, the rate of heat delivery by each mode, and the efficiency of each mode (calculated as a COP, with the rate of heat delivery to the load divided by the power consumption of the operating mode). It also identifies which modes provide the minimum power consumption

(P_{MIN}), the maximum rate of heat delivery (Q_{Max}) and the best COP (COP_{Best}). These identifiers are used extensively in the following protocols.

Q_{HX} is calculated with the same effectiveness calculations used in Type 5b, with the UA value entered as a parameter; calculations for Q_{HP} and Q_{SRS} are slightly more involved. Q_{HP} uses the same calculations established by Wagar, modified to provide the common TRNSYS units of kJ/hr. Based on inlet temperatures that are input from the buffer tank and hot water tank, respectively, the controller predicts the rates of heat transfer and power consumption for the heat pump. The power consumed by an operating mode is estimated using the calculated P_{HP} and equations from the Type 110 circulation pump model. To determine Q_{SRS} and P_{SRS} , the controller first finds Q_{HX} and estimates the HX outlet temperatures based on the system flow rate. These become the inlet temperatures used by the HP calculations. Q_{SRS} is the sum of Q_{HX} and $Q_{\text{HP,Load,Series}}$. The equations used to calculate these parameters is available in Appendix A.

The Min P protocol, which is illustrated in Figure 3.10, opens by identifying the value of P_{MIN}. The value identifies the mode which consumes the least power in this timestep; P_{MIN}=[1, 2 or 3] indicates [HX, HP or SRS]. When the value is identified, the protocol first asks if the associated COP and Q values are acceptable. For example, if P_{MIN}=1, the controller asks if $\text{COP}_{\text{HX}} > 1$ and $Q_{\text{HX}} > 0$ to make sure that HX operation is a) more energy efficient than auxiliary operation and b) net energy positive for the tank. If $\text{COP} < 1$, it is implied that the system delivers less energy to the tank than is required to run it – running the auxiliary backup would be preferable in this case. Likewise, $Q < 0$ indicates a net loss of energy from the tank as a result of system operation. If the controller finds neither of these to be the case, the minimum power mode is selected and the controller exits the Min P protocol. However, if either value is unacceptable, the protocol continues.

The next question then tries to determine the second-lowest power mode. It asks if one of the two remaining choices has lower power consumption than the other. If yes, it asks if that mode has $COP > 1$ and $Q > 0$ – if this is true, then that mode is selected. Otherwise, the third and final mode is checked to see if it is acceptable to run. If no mode is acceptable (no mode has $COP > 1$ and $Q > 0$), the controller chooses mode 0 and jumps to the Conditional section.

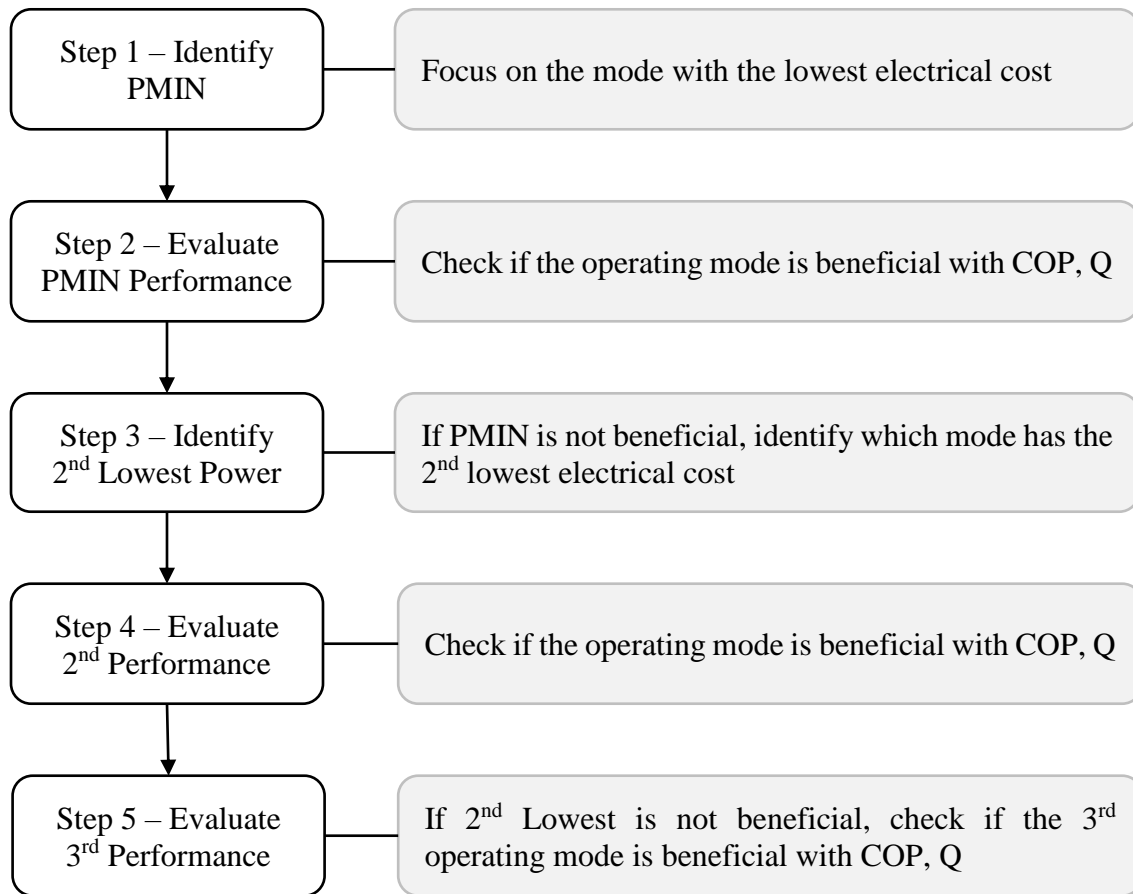


Figure 3.10 - Steps of Stage 5 - Minimum Power Protocol of control scheme

At the end of the Min P protocol, the controller will jump to the Conditional Override stage (for modes 0 and 1) or to the Heat Pump Safety Check stage (for modes 2 and 3). Therefore, the following sequence will only run if the Min P protocol is skipped because $T_{set} > T_{tta}$. This is the Max Q protocol.

3.3.6 - Phase 2 – Stage 6 – Maximum Rate of Heat Gain Protocol

In the Max Q protocol, efficiency remains a priority. This sequence begins by identifying the mode with the best COP, (the best heat-delivery-rate to power-consumption ratio). Once this mode is identified, the controller asks if that mode can deliver heat quickly enough to satisfy the load. This introduces an internally-calculated quantity called Q_{Load} .

Q_{Load} is calculated based on an estimate of how long the system can afford to take to get up to temperature. An input to the controller, t_{target} , describes the maximum permissible time to heat up the tank. A target temperature T_{target} , which is usually slightly higher than the set-point, is used in comparison to T_{tta} , and an arbitrary but large “load” size (in kg of water drawn) is prescribed as a parameter. In this thesis, the load mass is slightly larger than the largest single draw in the CSA-A profile, and approximates the mass of fluid represented by T_{tta} . Using this mass, the difference between T_{target} and T_{tta} , and the prescribed t_{target} , the controller estimates the rate of heat delivery required for an operating mode to meet the load.

As illustrated in Figure 3.11, the Max Q protocol compares Q_{Load} to the estimated heat gain achievable by the mode with the best COP. If the mode which operates with the best COP can also meet the load ($Q > Q_{Load}$), that mode is selected. Failing this – meaning that the most efficient mode cannot meet the needs of the system – the controller then identifies the mode that will deliver heat to the tank most quickly. The code then jumps to Stage 7 - Heat Pump Safety Check or Stage 8 - Conditional Override stage as appropriate.

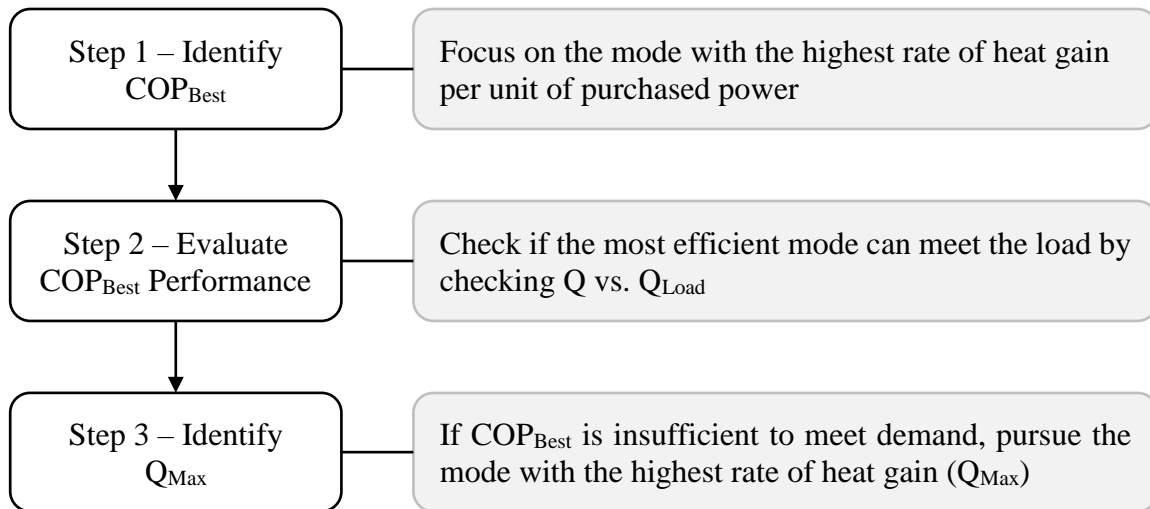


Figure 3.11 - Steps of Stage 6 - Maximum Rate of Heat Gain Protocol in control scheme

If the maximum rate of gain is via the HX, mode 1 is selected. If the maximum rate of gain is via HP or SRS modes, the code proceeds to Stage 7 - HP Safety Check. In the instance that none of the above provides the most gain (implying that all modes result in a net loss, an indication of an error), the system enters WU mode. This concludes the Max Q protocol.

3.3.7 - Phase 2 – Stage 7 – Heat Pump Safety Check

Wagar and Bannister noted several limitations on the heat pump in their studies [13, 14]. The inlet temperatures on either side had to fall below certain limits, and the solar-side outlet had to be above 5 C to avoid freezing. Additionally, due to the oversized nature of the physical heat pump unit they had available, time limitations had to be placed on the system to avoid short cycling. They included a buffer tank for thermal mass on the solar side and introduced WU mode to allow heat accumulation in the solar loop as well [13, 14]. The following protocol is designed to perform checks that ensure safe operation, checks which were previously performed by the heat pump model itself, to maximize the controller’s capabilities [13]. This safety protocol, illustrated in Figure 3.12, begins by identifying whether the controller is trying to select HP-only or SRS mode. It then asks if the heat pump is currently in use.

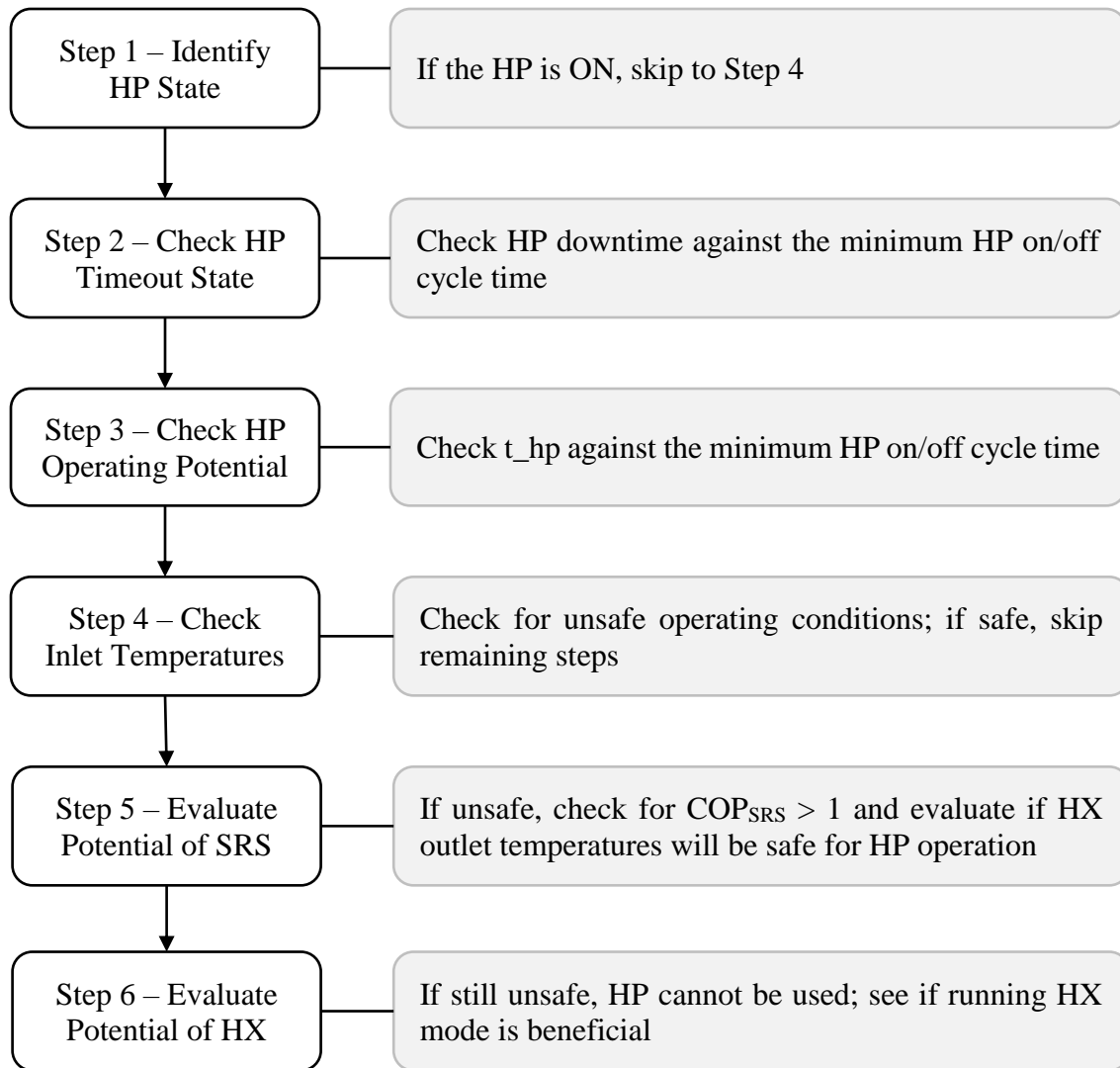


Figure 3.12 - Steps of Stage 7 - Heat Pump Safety Checks for control scheme

If the heat pump is already on, then there is no need to check the start-up criteria. If it is not on, however, the controller asks two questions, both of which are based on the minimum allowable cycle time for the heat pump. This is a parameter set to 5 minutes in this thesis. First, if the heat pump is off, it must stay off for at least the minimum cycle time. If this time has not passed since the HP turned off, the system will choose HX mode (if energy-profitable) or WU mode. Second, if the heat pump is off, it should stay off unless it can operate longer than the minimum cycle time without freezing the solar-side loop. If the predicted run-time of the heat pump does not exceed

the minimum cycle time, the controller chooses HX or WU mode instead, as appropriate. Otherwise, the protocol continues as if the heat pump was already operational.

The predicted total operation time is calculated as t_{hp} (or t_{hps} in the case of SRS mode). The time is calculated based on the current heat pump solar-side outlet temperature, the minimum acceptable outlet temperature, the thermal mass of the solar-side loop of the system and the predicted net rate of energy loss of the loop. The net energy loss is calculated as $Q_{out} - Q_{STC}$, where Q_{out} is the rate of heat extracted from the solar-side loop by the HX and HP (Q_{HX} and $Q_{HP,Source}$ or $Q_{HP,Source,Srs}$). The difference is coerced to 1 kJ/hr if it falls below this value. Likewise, the temperature difference is coerced to 0 once it falls below the tolerance amount. In the case that $Q_{out} - Q_{STC} < 0$ (indicating that solar gain exceeds heat delivery), the theoretical runtime is infinite, hence the calculated result is coerced to 1000 hours to indicate a very strong ability to run continuously. The equations for calculating t_{hp} are available in Appendix A.

At this point, the controller has addressed concerns of short-cycling the heat pump. Now it addresses the safety concerns of operating within acceptable temperature limits. If the inlet temperatures for the heat pump are acceptable, HP-only mode may be selected. If not (or if the system is trying to select SRS mode), the controller checks if $COP_{SRS} > 1$ and if the HX outlet temperatures would be below the HP inlet temperature limits if SRS mode is activated.

If these temperatures are acceptable, SRS mode is selected. Otherwise, heat pump operation is not possible. The controller will check to see if HX operation is beneficial and will turn the system off if it is not. This completes the Heat Pump Safety Check protocol and concludes Phase 2 of the controller operation.

3.3.8 - Phase 3 – Stage 8 – Conditional Override

Phase 3 – Confirmation is focused on confirming the mode selected by Phases 1 and 2, identifying the need for auxiliary backup heating and performing control-variable management. This begins with Stage 8 – Conditional Override, which compares the selected mode to the mode that is currently in use. The controller may, in this stage, choose to override its selected mode in favour of maintaining the current condition. The conditions for overriding the selection are illustrated in Figure 3.13 and described thereafter.

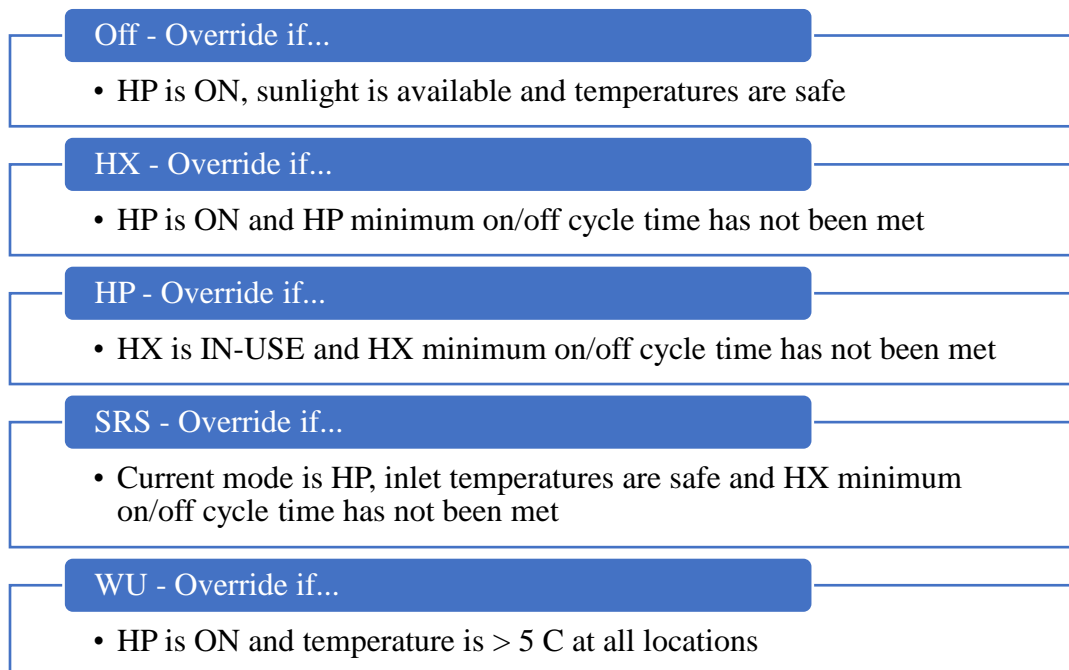


Figure 3.13 - Breakdown of the checks performed in Stage 8 - Conditional Override in control scheme

If the selection is mode 0 (off) and the current mode involves heat pump operation, the reason for shutting down is weighed against the desire to avoid short-cycling the heat pump. The system will turn off if there is no more sunlight available, or if heat pump inlet conditions have become unsafe, but otherwise the system will continue to run in the current mode until the minimum heat pump cycle time has been achieved.

If the selection is HX and the current mode involves HP use, the controller checks to see if the HP cycle time has been met. If not, it will continue to run with the HP until the cycle time is satisfied.

If the selection is HP and the current mode involves HX use, the controller checks to see if the minimum HX cycle time has been met. This is prescribed to prevent over-cycling of the motorized flow control valves; in this thesis, the minimum HX cycle time is less than the timestep of the simulations. Similarly, if the selection is SRS and the current mode is HP, the controller will allow HP-only operation to continue if the inlet temperatures are safe and the HX cycle time has not been met.

Finally, if the selection is WU and the current mode involves heat pump operation, the controller will only allow WU to start if there is a system temperature colder than the minimum allowable source-side outlet temperature. Other causes, such as $t_{hp} < t_{hp,min}$, should not result in WU being selected if the heat pump is already active. This concludes the Conditional Override stage.

3.3.9 - Phase 3 – Stage 9 – Auxiliary Management

The last stage of the iterative controller logic addresses the use of the auxiliary backup heater. The process observed for the Auxiliary Management stage is illustrated in Figure 3.14.

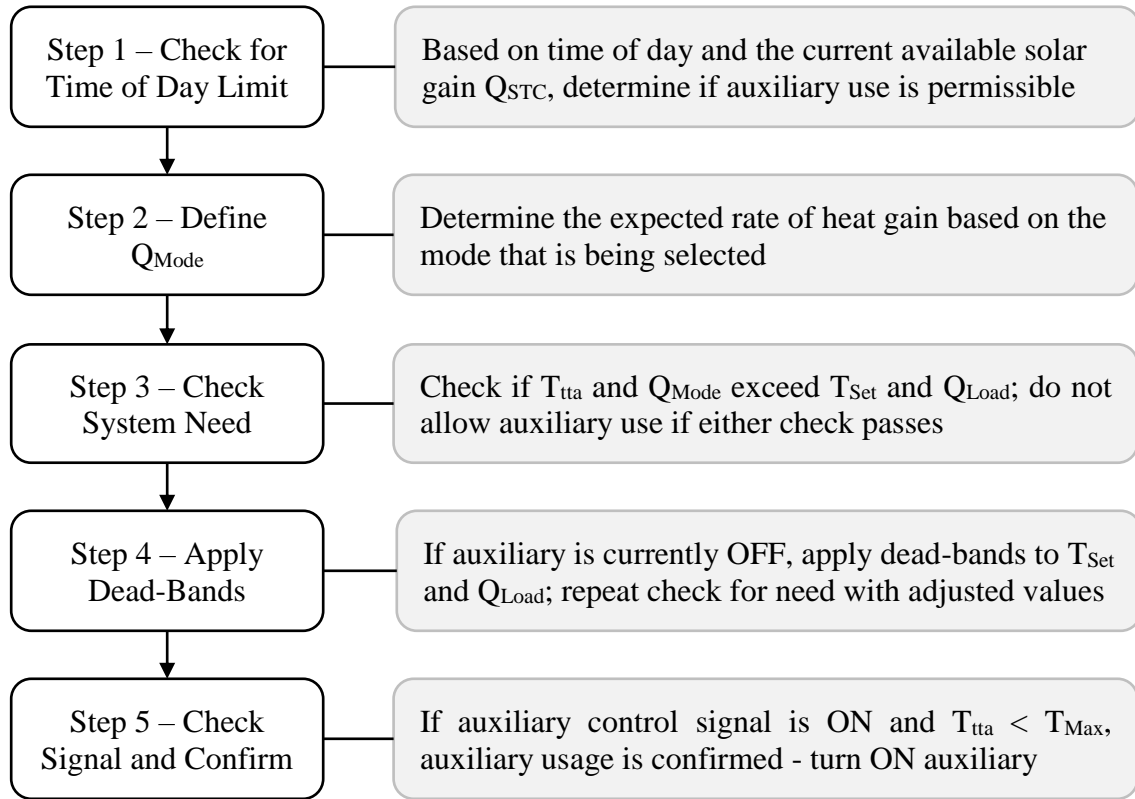


Figure 3.14 - Steps of Stage 9 - Auxiliary Management for control scheme

First, based on the mode selection, a parameter Q_{Mode} is defined for comparison to Q_{Load} . Then the controller determines if auxiliary usage is permitted at all. This is done by forcing the auxiliary control signal to 0 if the time of day is outside of a range that is measured forwards and backwards from midnight. This time is prescribed by the user; this thesis uses 8 hours to disable the auxiliary heater between 8 AM and 4 PM, hours during which the winter solstice (and therefore every day of the year) receives sunlight. A check is included so that the backup heater may still be used during daytime hours, if the available sunlight is unusually low.

If auxiliary use is permissible, either due to time of day or low sun conditions, the auxiliary control signal may be activated. If $T_{tta} < T_{Set}$ and $Q_{Mode} < Q_{Load}$, then auxiliary use is desired – this condition

indicates that the upper part of the tank is colder than acceptable and that the current operating mode will not provide heat fast enough to meet the demands of the system.

Following this, the temperature dead-bands prescribed by the user are applied. If the auxiliary heater is off, it should not turn on unless the tank temperature has fallen below the dead-band-adjusted set point. The rate of heat gain requirement is adjusted and checked in a similar manner. If both checks pass, the auxiliary heater stays off.

Finally, if the control signal is on and $T_{tta} < T_{Max}$, the controller sets the auxiliary power output to the prescribed auxiliary heater capacity value. Otherwise, the output power is set to zero. This code is repeated with every iteration during each timestep. If TRNSYS is still performing iterations, Stage 9 represents the end of the controller program; however, if iterations have converged and the timestep is ending, the code proceeds to Stage 10.

3.3.10 - Phase 3 – Stage 10 – End-of-Timestep Timer Management

A final set of steps is performed at the end of each timestep based on the new mode selection, resetting the timers that keep track of HX and HP cycling times as appropriate. A tracker indicating the system mode at the start of the timestep is set to the new mode value, the HX and HP state timers are incremented and the timestep ends. This concludes Phase 3, confirming all the outputs of the controller and the operating mode that it has selected, and thereby concludes all operation of the control scheme for the timestep.

Chapter 4 – System Comparison

In this chapter, the seven previously described systems are compared so that the performance of the 3-mode system can be evaluated. The comparisons will include evaluations of performance on a single summer day (August 14), a single winter day (December 21) and over a full simulated year. The daily studies will focus on system behaviour, highlighting the timing and duration of selected operating modes and auxiliary operation against the available sunlight, current tank temperatures and the net energy balance on the tank for the day. Behaviour studies were used throughout the development of the controller to highlight problems in the written program logic, or to point out flaws in the design itself, leading to features such as the nighttime shutdown, daytime auxiliary shutdown and the heat pump long cycle check. The annual studies will focus on system performance, using factors such as solar fraction (SF), saved energy (E_{Saved}) and the frequency of “low-temperature” draw temperature occurrences to evaluate and compare the performance of the systems.

The Key Performance Indicators (KPIs) are defined as follows. In an electric-only system, a certain quantity of energy will be purchased to heat the water in the tank. Most is delivered to the load, some is lost to the environment, and all of it is purchased by the consumer. This quantity of energy, E_{Load} , is assumed to be the amount of energy that is required by the load over the time period in question. For a SDHW or SAHP system, much of E_{Load} is acquired through the solar collector, but electricity is used in pumps and compressors where the basic system only uses the auxiliary component. Duffie and Beckman defined solar fraction as the percentage reduction of purchased auxiliary energy [10]:

$$SF = 1 - \frac{E_{\text{Aux}}}{E_{\text{Load}}} \quad (2)$$

This definition neglects the “parasitic” loads of circulation pumps (or, in this case, compressors); it assumes that a reduction in auxiliary use is entirely attributable to solar gain [10]. Another KPI is used in this thesis, then, to help quantify the energy savings realized by the system. E_{Saved} is defined as the difference between E_{Load} and the total electrical consumption of the system being evaluated. E_{Bought} is used to represent this consumption, defined as the amount of electricity used by the auxiliary heater, heat pump and circulation pumps combined:

$$E_{Saved} = E_{Load} - E_{Bought} \quad (3)$$

The other KPIs are the cumulative total of medium-temperature-band and low-temperature-band draws that occur during a simulation. This thesis uses the CSA-A profile that was employed by Wagar and Bannister [13, 14, 56], with the flowrates modified so that the drawn volume of water is accurate for each draw occurrence. The definition of “low temperature” is subjective, with Sterling setting an auxiliary thermostat temperature of 50 C and Bannister recording delivery temperatures above 53, 55 and 57 C to evaluate system performance [14, 47]. Though liquid water at 50 C is still hot enough to burn exposed skin, it is generally a low bar for stored DHW. To prevent the growth of legionella in the storage tank, most systems use a thermostat set-point temperature of 60 C. This will be used as the thermostat setting throughout this thesis, except where stated otherwise; for evaluating thermal performance, then, draws below 50 C will be considered “low temperature”. The temperature range of 50-55 C is considered “medium temperature”. A system with high thermal performance should never have low-temperature draws and rarely have medium-temperature draws; high counts in these KPIs will indicate a system that is not providing the tank with sufficiently high-quality heat energy.

The days to be used for the daily behaviour study were selected for their favourable solar conditions in combination with highly favourable/highly unfavourable temperature conditions.

August 14th is, on average, one of the brightest and hottest days of the year in Ottawa as per data obtained using the TMY2 file; December 21st is also quite bright, but is one of the coldest days of the year and experiences few hours of sunshine. Studying system behaviour on these two days should therefore highlight the system’s sensitivity to ambient temperature and available daylight. The ambient temperature and total tilted solar irradiation profiles for these days are plotted in Figure 4.1.

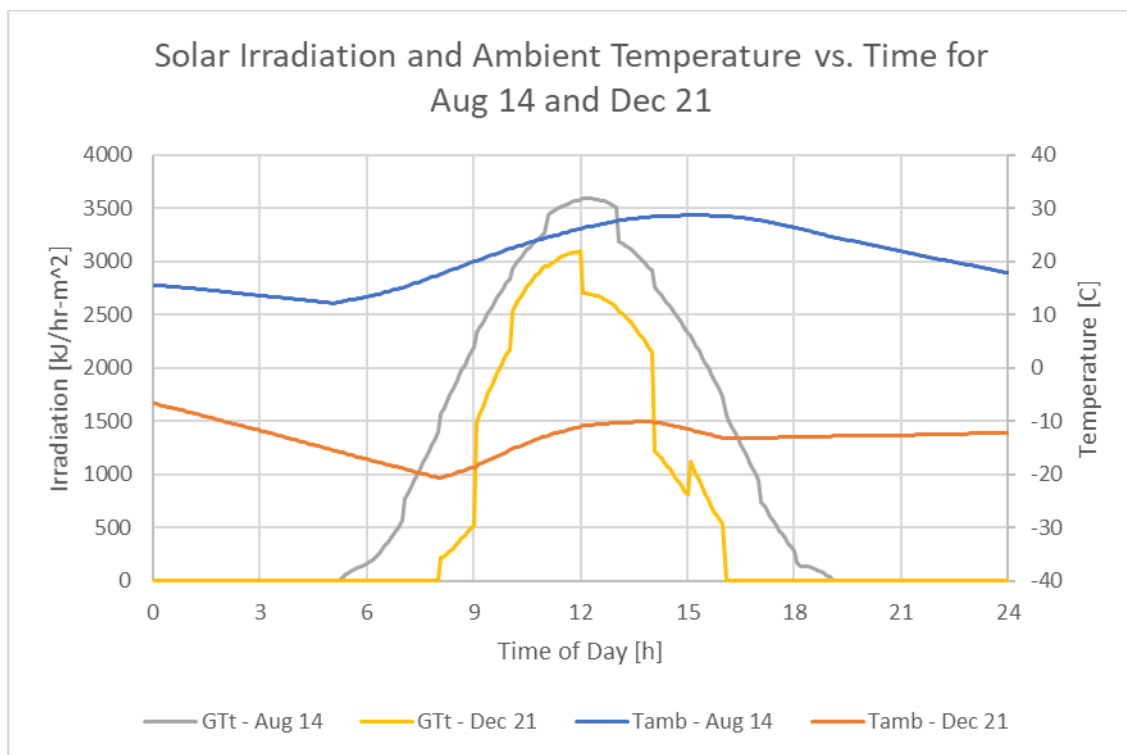


Figure 4.1 - Solar-thermal property profiles for Aug 14 and Dec 21

4.1 – Electric DHW System Behaviour

The electric system is the basis against which all other systems are to be measured. It will not experience any solar gain, and will behave exactly the same on any given day of the year, since the only time-of-year variable in the system is the mains inlet temperature. Figure 4.2 shows the system behaviour for a simulated day with the electric system. The top and bottom tank

temperatures are shown, with the wide separation between them indicating the stratification that occurs in the tank. The energy balance indicates the level of internal energy stored in the water in the tank relative to a reference point; this is often referred to as the tank energy. Since initial conditions are somewhat arbitrary, the simulation runs for several days before the day to be studied, for which data is recorded – this will often result in a non-zero energy balance for midnight on the day to be studied.

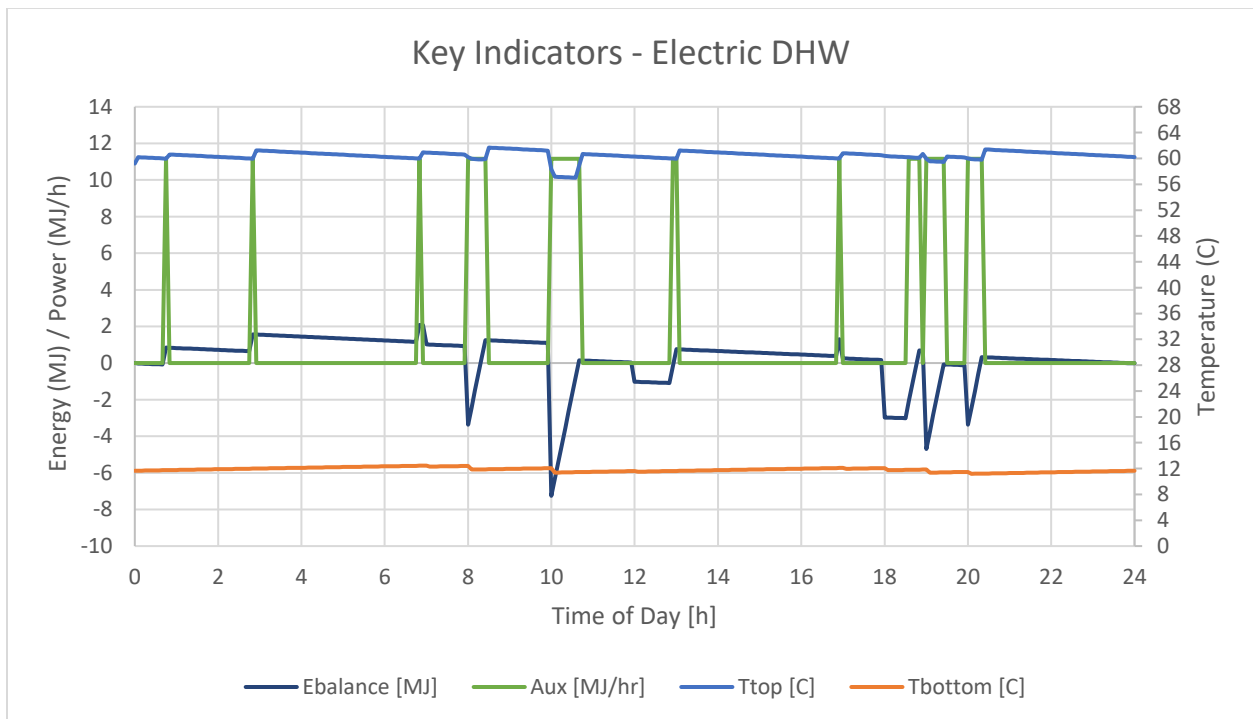


Figure 4.2 - System behaviour chart for Electric DHW system

The tank temperatures remain highly stratified throughout the day, since the only forced flows occur from cold to hot. The daily energy balance is very close to zero, since a daily draw pattern allows the system to achieve a pseudo-steady state daily cycle. Heat losses to the environment occur at a slow and steady rate from the top of the tank, which is observable as the steady downwards slope in the tank energy and top tank temperature lines, periodically triggering the auxiliary heater.

The largest instances of heater usage follow large water draws, when the hottest water at the top of the tank becomes fully replaced by cooler middle-tank water. At the times where the auxiliary heater is active, tank energy rises rapidly, corresponding to the energy consumed by the heater. There are 10 instances of heater activation. The two early-morning activations are the only instances that are entirely independent of draw occurrences. The other 8 instances correlate to the timing of the daily water draws, which occur at 7, 8 and 10 AM, and 12, 5, 6, 7 and 8 PM. The water draws can be identified by a sharp corresponding drop in tank energy.

The larger water draws cause the top tank temperature to drop, immediately activating the auxiliary heater at 8 AM, 10 AM, 7 PM and 8 PM. Two draws are small enough that the top tank temperature is not affected – the auxiliary heater does not turn on until shortly after the draws, after further losses to the environment. Observe how the draws at 12 and 6 PM do not affect the top tank temperature but still cause drops in tank energy, which stays low for some time before the heater is activated, restoring the tank energy levels. The two smallest water draws, at 7 AM and 5 PM, occur immediately after auxiliary heater activation. In these cases, the heater is activated by falling tank temperatures just before the draws occur. The tank energy briefly spikes upwards due to the heater usage, and then snap back down because of the draws. More importantly, the top tank temperature is boosted by the heater but is unaffected by the small water draws, allowing the heater to shut down.

4.2 – Traditional SDHW System Behaviour

With the SDHW system, the standard for solar system performance is established. While all solar systems should provide improvements vs. the electric system, any heat pump-assisted system must be justified against the simpler, less-expensive SDHW system. Figure 4.3 shows the behaviour of a SDHW system for a representative summer day. Note that while Figure 4.2 used Time of Day

on its x-axis, other behaviour charts use Time of Year. The charts are presented on a midnight-to-midnight basis for easy time-of-day comparisons.

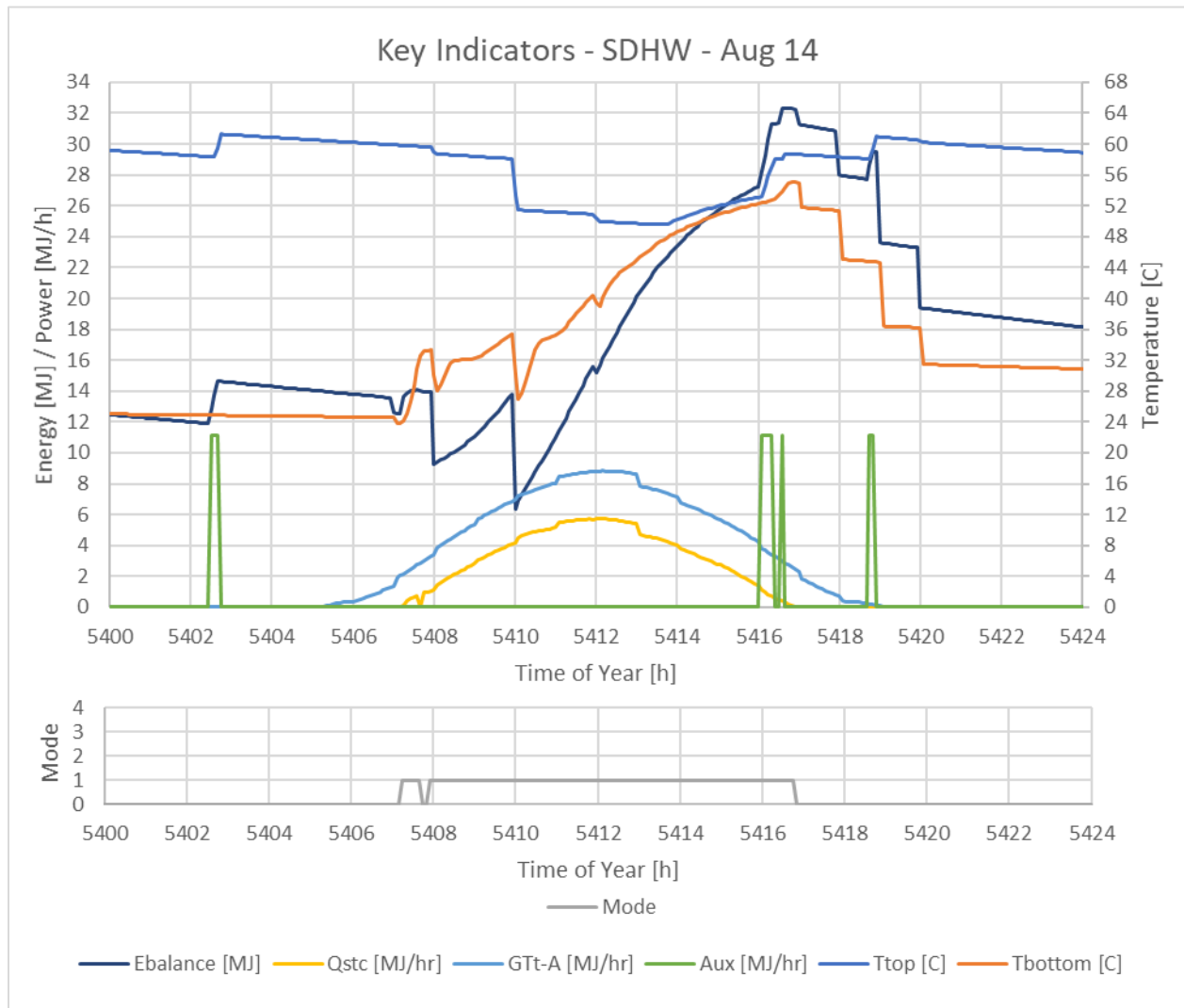


Figure 4.3 - System behaviour chart for SDHW system on Aug 14

The behaviour of this system is substantially different from the electric-only system. The top tank temperature is not consistently held at 60 C, and the bottom tank temperature reaches 50 C by the end of the day. The tank energy path is very dramatic here, as large amounts of low-quality energy are delivered from the solar collector. The 4 instances of heater activation correspond to sharp increases in tank energy and top tank temperature. The 8 water draws are still notable by sharp

drops in tank energy; with this system, the occurrence of a water draw also causes a sharp drop in bottom tank temperature, as cold mains water rushes in to the tank.

The introduction of a solar component corresponds to the addition of three more data sets to the chart. Q_{STC} represents the rate of energy collection by the solar collector, G_{Tt-A} is the total solar irradiation incident on the collector, and Mode indicates the operating mode of the system. Here, the only available operating modes are 0 (off) and 1 (HX-only). The system is only operating when Mode \neq 0 – observe that Q_{STC} is 0 if the system is off. One can estimate the collector effectiveness throughout its operating period by comparing Q_{STC} to G_{Tt-A} .

This summer day shows a very long period of system operation (where Mode = 1). Tank losses force the auxiliary backup to activate at night before sunlight is available. Once the sun rises, the SDHW system is operating almost continuously from about 7 AM (5407 hours) to about 5 PM (5417 hours), during which time $Q_{STC} > 0$ and the tank energy balance becomes increasingly positive. The auxiliary heater kicks on near the end of this cycle to boost the upper tank temperature, since the solar system is not delivering water hotter than 54 C. Note how T_{Bottom} approaches T_{Top} in the late afternoon – continuous operation of the heat exchanger system without any water draw events establishes a downwards current in the tank that causes complete destratification. Nighttime water draws help re-stratify the tank after operation stops, but the degree of stratification is only half as strong as observed in the electric system.

It is somewhat surprising to see the system struggle to deliver water hotter than 55 C to the tank on this day. Observing the raw data shows that the collector outlet temperatures rarely exceed 60 C, limiting the maximum HX outlet temperatures; this may be due to the flowrate used for the simulation, which matches that used on all the systems to be compared. An alternative approach could try to refine system performance by lowering the flowrate, thereby increasing the lift

achieved in the collector. This study uses a flowrate of 200 kg/hr; it is more typical for a SDHW system to operate on flowrates closer to 120 kg/hr. A lower flowrate allows the collector to output much higher temperatures, even if the rate of solar heat gain is decreased, which then allows the system to deliver water in excess of 60 C.

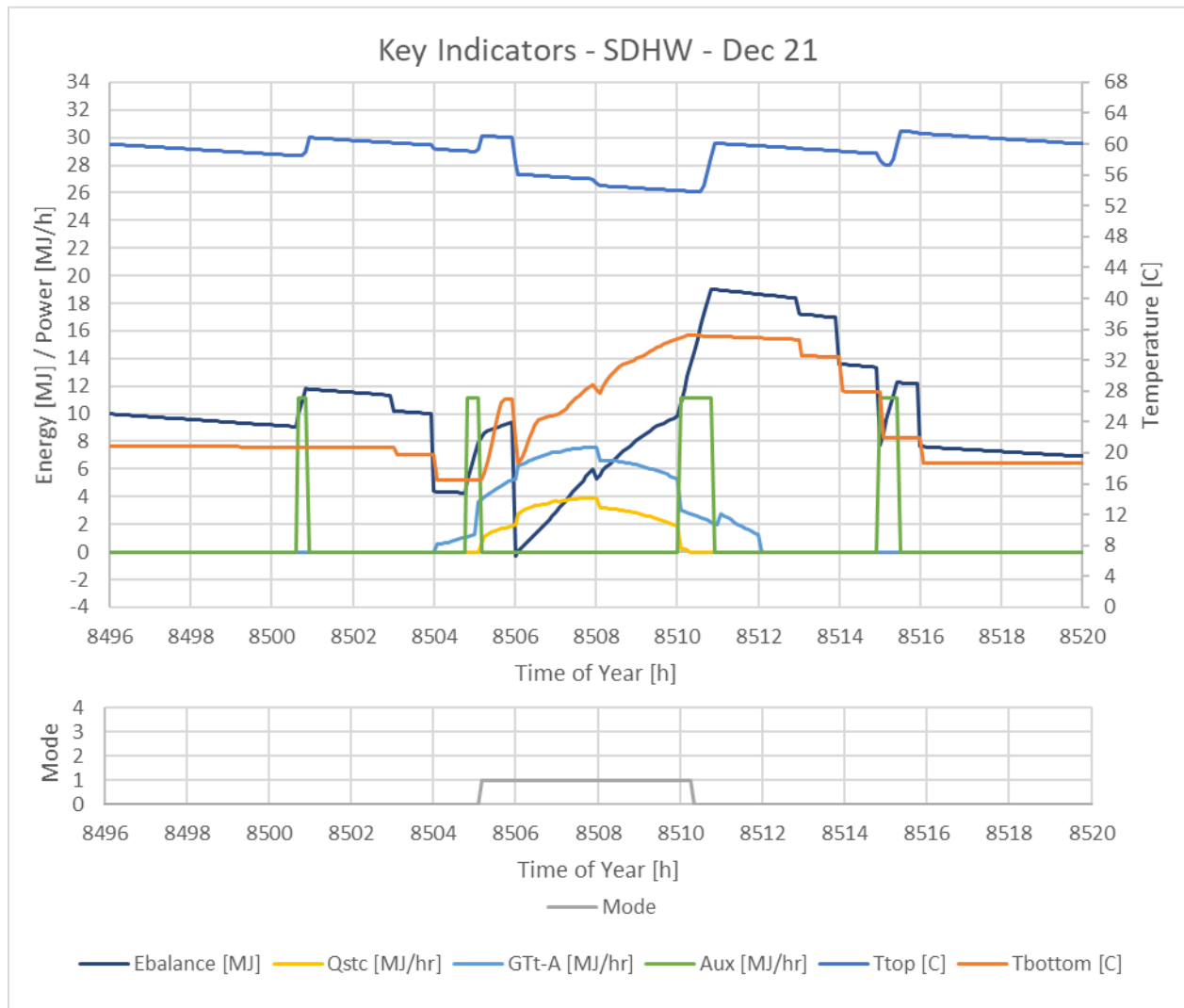


Figure 4.4 - System behaviour chart for SDHW system on Dec 21

Figure 4.4 presents the behaviour of a SDHW system on a representative winter day, showing this system’s inability to maintain effectiveness in adverse conditions. Despite fairly intense solar irradiation during the day, the short daylight hours and cold ambient temperatures reduce the

operating hours and efficiency of the solar collector. The operating hours are reduced to 8505 – 8510 (9 AM to 2 PM), half as long as on the summer day. Therefore, the destratification observed in the summer does not occur to such a strong degree, but the only times that T_{Top} gets boosted are times of auxiliary usage (unlike the summer day, during which the top temperature was boosted slightly by system operation from 5413-5417 hours). One can also observe that despite favourable solar conditions, the SDHW system nets a tank energy reduction of more than 3 MJ over the course of this day, compared to a 5.7 MJ tank energy net increase over the summer day. It is this poor winter performance that the SAHP system is intended to boost.

4.3 – Single-Tank SAHP (HP-only) System

The single-tank SAHP (HP-only) system consists of a solar collector loop, which leads to the HP evaporator, a load-side loop connected to the HP condenser, and the HP and storage tank themselves. This system is fully dependent on the heat pump for delivering solar energy to the tank. Unlike many similar systems, this design does not incorporate a full storage tank on the solar side, instead opting for the use of a buffer tank to increase the thermal mass of the loop. This design is used for direct comparison to the other systems – a HP-only system may exhibit better performance in a dual-tank configuration as studied by Freeman and Bridgeman [30, 31]. However, the optimal configuration for a HP-only system is beyond the scope of this thesis. Figure 4.5 illustrates the behaviour of the HP-only system for August 14th in Ottawa, Canada.

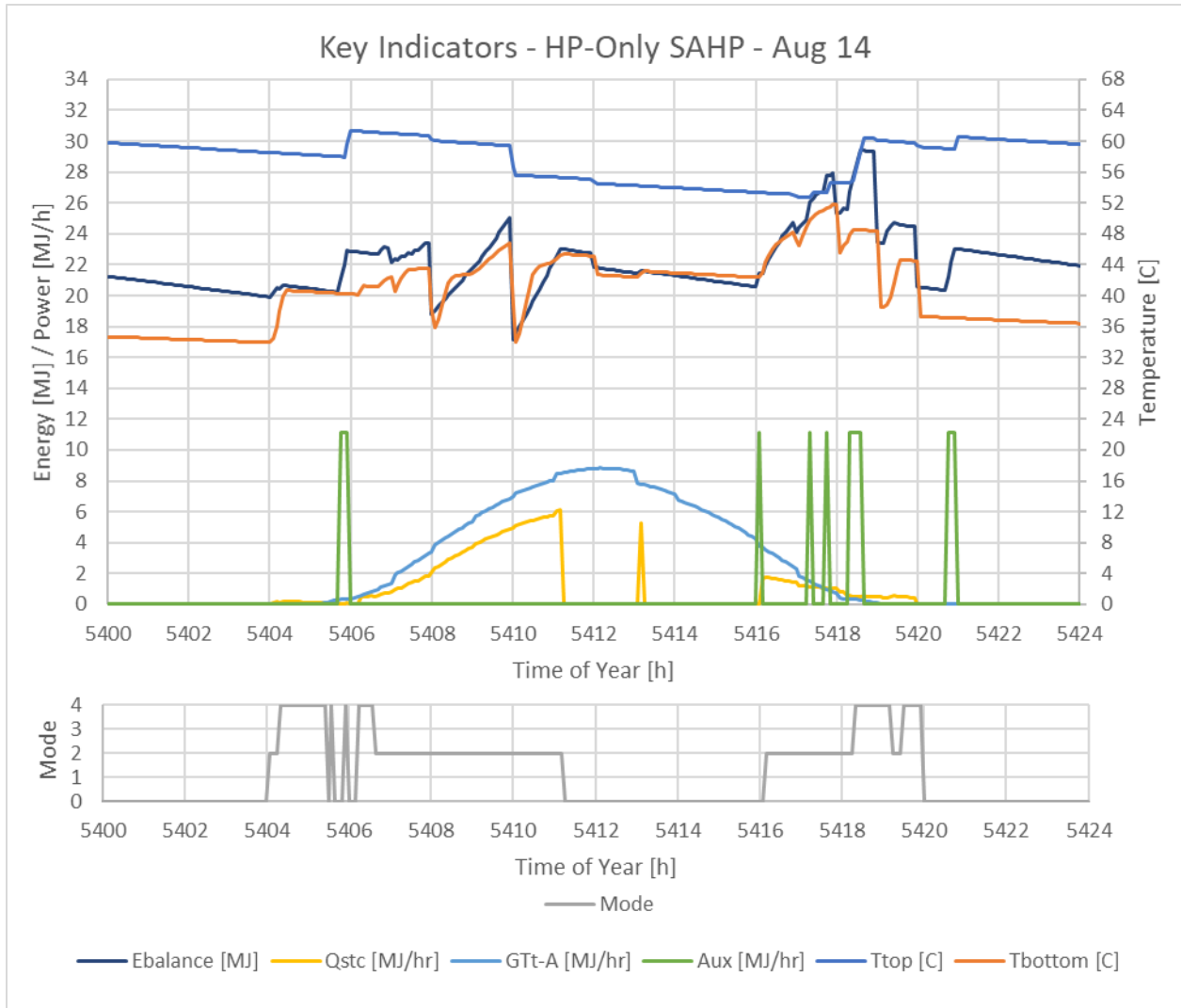


Figure 4.5 - System behaviour chart for HP-Only SAHP system on Aug 14

System behaviour is somewhat inconsistent in the morning and evening hours on this day. Long periods running in the warmup mode are observed just before and after sunrise/sunset, showing the collector’s ability to gather ambient energy. The gains are small, but so is the expense of running in these shoulder hours.

Another prominent characteristic of the behaviour on this day is the mid-day shutdown of the system. Checking the raw data for this simulation shows that the solar collector outlet climbs above the heat pump’s evaporator inlet temperature limit around 11 AM (5411 hours), forcing the heat

pump to shut down. The system stagnates during the day and cannot resume operation until 5416 hours (4 PM). This condition would justify the integration of a system protocol to defend against stagnation damage, such as using a shaded heat exchanger on the solar loop for venting excess heat.

Compared to the SDHW system, the start and end times for operation of the SAHP are extended, but continuous operation is harder to achieve. Destratification does not occur to the same degree, since operation is not continuous through the day, but the level of stratification around midnight hours is not as strong as in the SDHW. Collector efficiency (defined as the rate of energy collection divided by available solar irradiation) is significantly improved by the heat pump, since the fluid temperature entering the collector is lower in the SAHP system than the SDHW. Where the SDHW showed an efficiency in the 55% range, SAHP collector efficiency exceeds 70% with ease. This means that despite the long stagnation period during the day, much more energy is collected during operating hours than the SDHW managed – therefore, the degradation in system performance is not particularly strong. However, the existence of the stagnation period is a strike against the use of a single-tank HP-only system without appropriate stagnation countermeasures, and supports the case for a 2-mode system.

Figure 4.6 illustrates the behaviour for December 21st. The behaviour on December 21st is much more stable for the HP-only system than the SDHW system. Here the system operates almost continuously from sunrise to sunset, with the change in tank temperatures evidenced by the increasing T_{Bottom} . T_{Top} never shows evidence of the heat pump's influence, but the reduction in auxiliary heater usage shows that substantially more medium-quality heat energy is provided to the tank by the SAHP than by the SDHW system. There is more destratification taking place during

the day, since the system can run continuously, but the re-stratification at night is much stronger than in the summer condition, likely thanks to the lower water main temperatures.

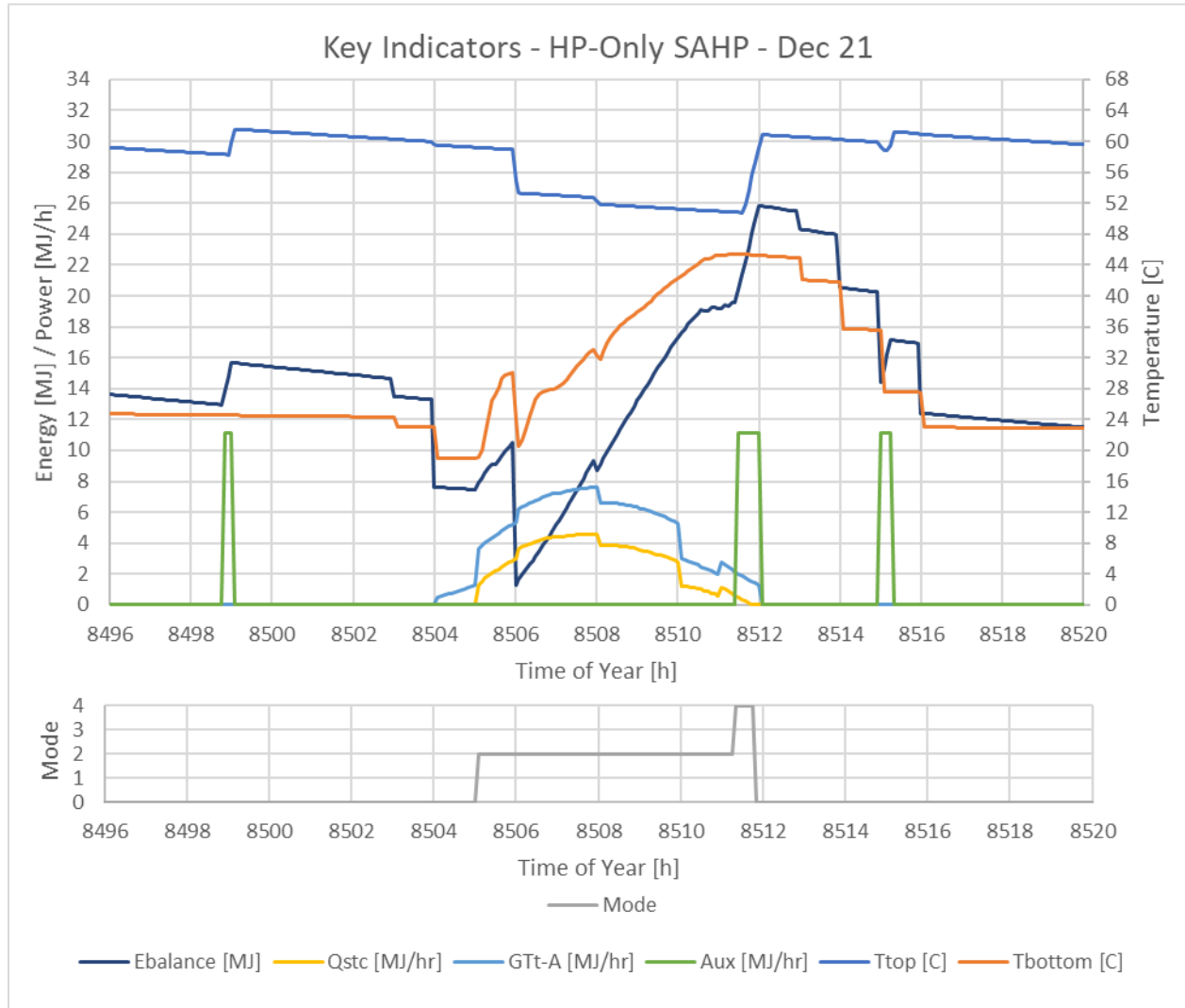


Figure 4.6 - System behaviour chart for HP-Only SAHP system on Dec 21

While the SDHW performed very well in the summer condition, it was nearly useless in the winter condition; in contrast, the SAHP displays strong behaviour and similar performance in either condition. There remains a clear gap between SAHP behaviour and achievable solar fraction in the summer. These sub-sections have provided a strong proof-of-concept for a hybridized multi-mode system which mimics a SDHW in the summer and a SAHP in the winter, examined in later sub-

sections. The study of such a design is the entire purpose of this work; dual-tank systems are outside of this scope.

4.4 – Single-Tank SAHP (SRS Configuration) System

The series configuration considers the effect of placing a heat pump in a traditional SDHW system such that the heat exchanger outlets go directly to the heat pump evaporator and condenser inlets, as appropriate. This should maximize the heat gains achieved during high solar hours, significantly improve collector efficiency and further extend the operating hours of the system by keeping fluid temperatures within the safe operating limits of the heat pump. Figure 4.7 and Figure 4.8 illustrate the behaviour of such a system.

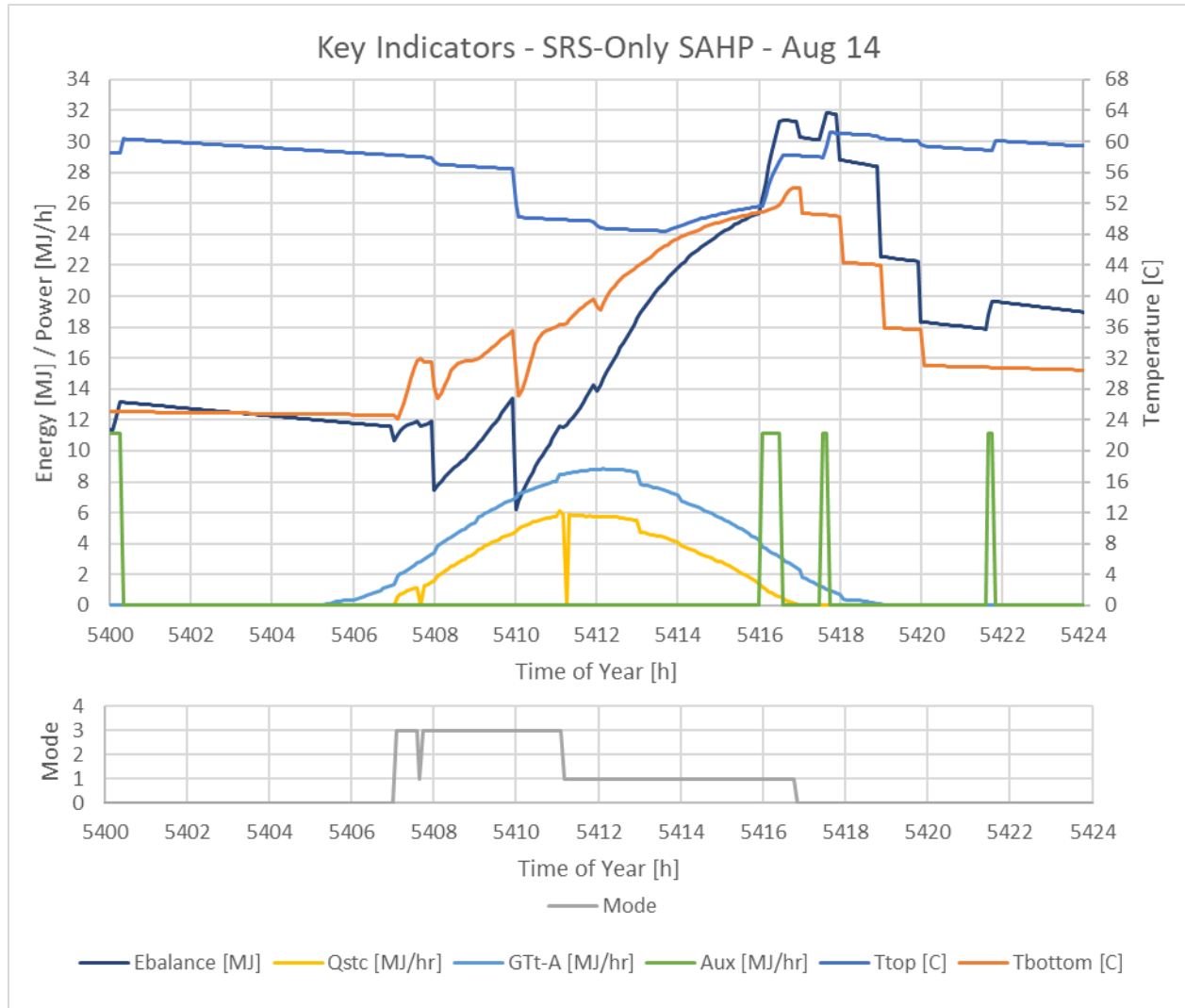


Figure 4.7 - System behaviour chart for SRS-Only SAHP system on Aug 14

On August 14th, a hybrid version of the SDHW and HP-Only SAHP systems is observed. The system operates with the heat pump on, in SRS mode, through the morning until about 11 AM, when the collector outlet temperature – and shortly thereafter, heat exchanger source-side outlet temperature – rises above the heat pump evaporator’s safe operating limit. Unlike the HP-only system, which is forced to shut down at this point, the SRS-only system has the option to de-energize the compressor and continue operating through the heat exchanger. At this time, water is flowing across the evaporator and condenser of the heat pump, but no heat transfer occurs as the

compressor is inactive and the refrigerant is not flowing. The system therefore behaves as a SDHW design through the afternoon. This avoids stagnation and greatly enhances the daytime gains of the system, even boosting T_{Top} directly by the day's end. The tank shows high destratification during the continuous operation period, but fairly strong stratification re-establishes overnight. The behaviour in general is a very interesting split of the SDHW and HP-only systems.

It is worth observing at this point that for both the HP-only and SRS-only systems, the heat pump was allowed to run continuously for several hours. The controller was designed with a feature that periodically stops heat pump operation to allow the solar loop to warm up – however, this feature was designed for use with the two and three-mode systems, which actively choose whether to use the HP or HX. The single-mode systems do not make this choice, and so in these systems, the heat pump is allowed to run continuously as long as it is safe to do so.

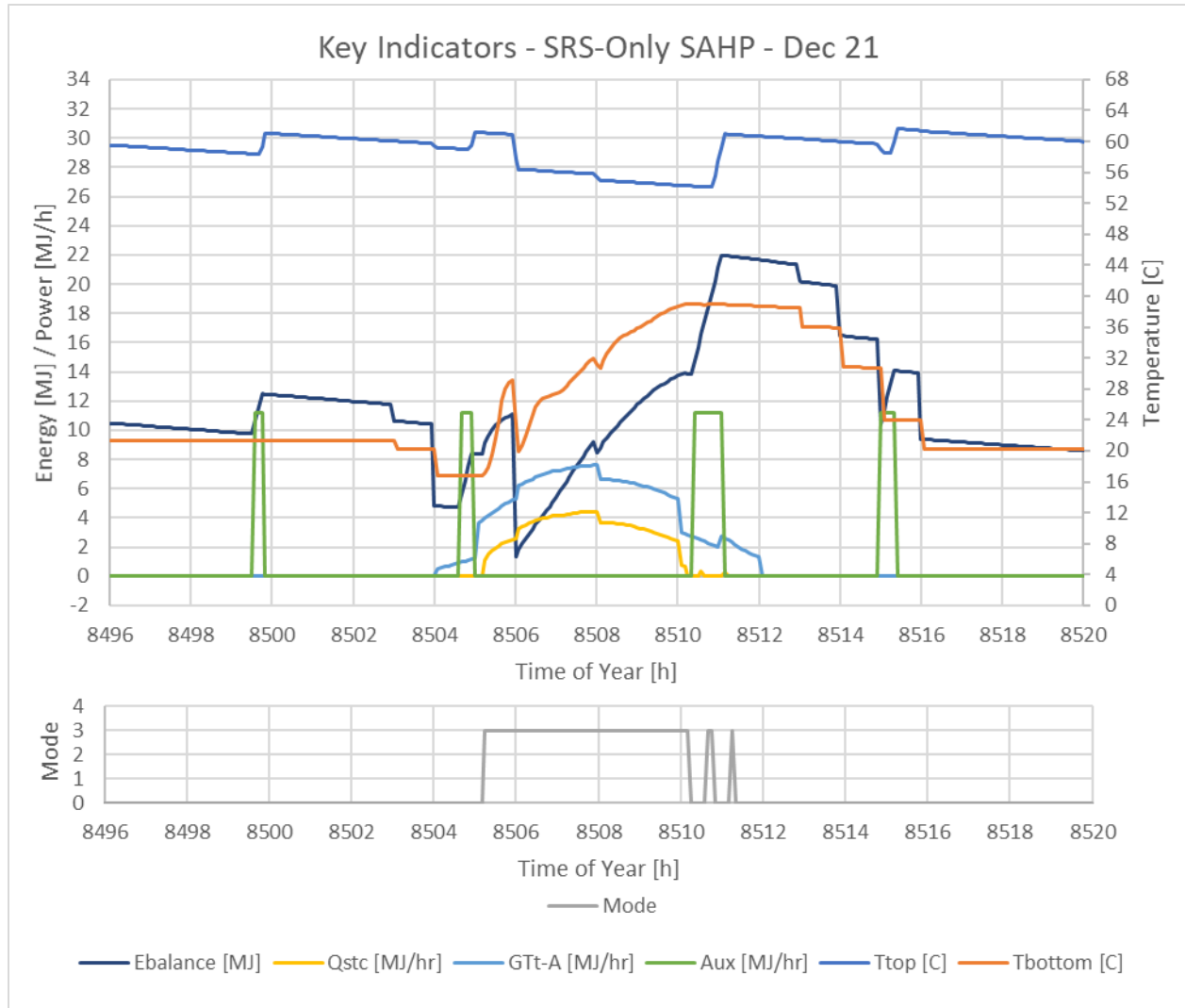


Figure 4.8 - System behaviour chart for SRS-only SAHP system on Dec 21

System behaviour on this day appears to be favourable, with operation in SRS mode for most of the daylight hours. However, due to the low ambient temperatures in December, the solar collector outlet temperatures rarely exceed 25 C. In the SDHW system, the collector can warm up a bit more since the heat exchanger can only withdraw a small amount of energy. However, with a heat pump operating, the collector temperatures stay low. Since the system is forced to use the heat exchanger before sending fluid to the heat pump, the system is constantly using the heat pump to counteract losses experienced in the heat exchanger, where warm tank water loses heat to the cool collector

water. The result is strong behaviour with poor performance – boosted STC and HP performance over the other designs is insufficient to overcome the effect of constant losses in the HX. Yet again, a single-mode single-tank system proves to be insufficient for year-round use, as the characteristics which make it favourable for one season cause it to perform poorly in the other.

4.5 – Two-Mode Dual-Side SAHP (HX or HP) System

The two-mode system applies the same control scheme as described in the previous chapter, but without providing the option to select SRS mode. This simplifies things greatly, since many fewer lines of code are needed to establish the better of only two options. The intent of this system is to provide the best of both SDHW and HP-only systems, without the drawbacks of either. This system is an excellent comparison for the 3-mode design, since the capital cost for equipment is the same in either design; the 3-mode system requires some additional plumbing, but no large equipment investments such as an additional tank or heat transfer device.

This system is designed based on the work of Wagar – as noted in the previous chapter, many of the component and parameter selections are the same, and the same heat pump model is used [13]. However, there are unknowns regarding Wagar’s system (such as pipe model dimensions), and substantial differences such as system flowrate, heat pump scale and tank model selection [13]. Therefore, any differences between the results of these simulations and those performed by Wagar cannot be strictly attributed to the control scheme used in this thesis, and any comparisons to Wagar’s work should take these differences and unknowns into account. Figure 4.9 and Figure 4.10 illustrate the behaviour of the 2-mode system simulated for this thesis.

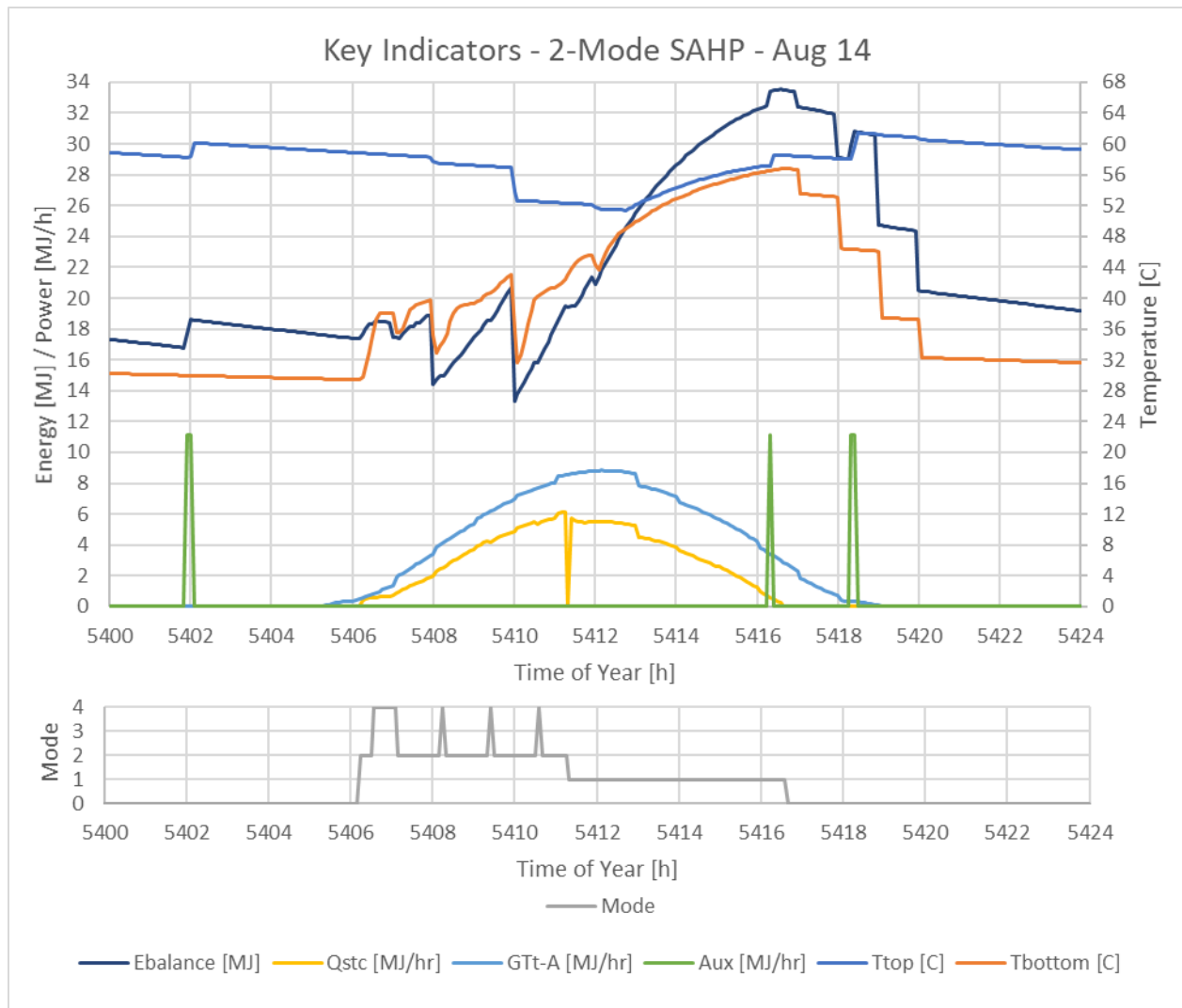


Figure 4.9 - System behaviour chart for 2-Mode SAHP system on Aug 14

The system shows similar behaviour to the SRS-only system, using the HP throughout the morning and swapping to HX mode for the afternoon. The system automatically switches to HX mode around 11 AM, when the collector outlet temperature exceeds the safe operating limit of the heat pump evaporator. The energy balance is very good for the day, but tank stratification is relatively low – a large amount of low-quality thermal energy is in the tank, but this may be beneficial by allowing the auxiliary heater to more easily boost T_{Top} . Collector efficiency is notably higher in

the morning, when the heat pump is in use, than the afternoon, but the heat exchanger is still able to deliver heated water above 55 C.

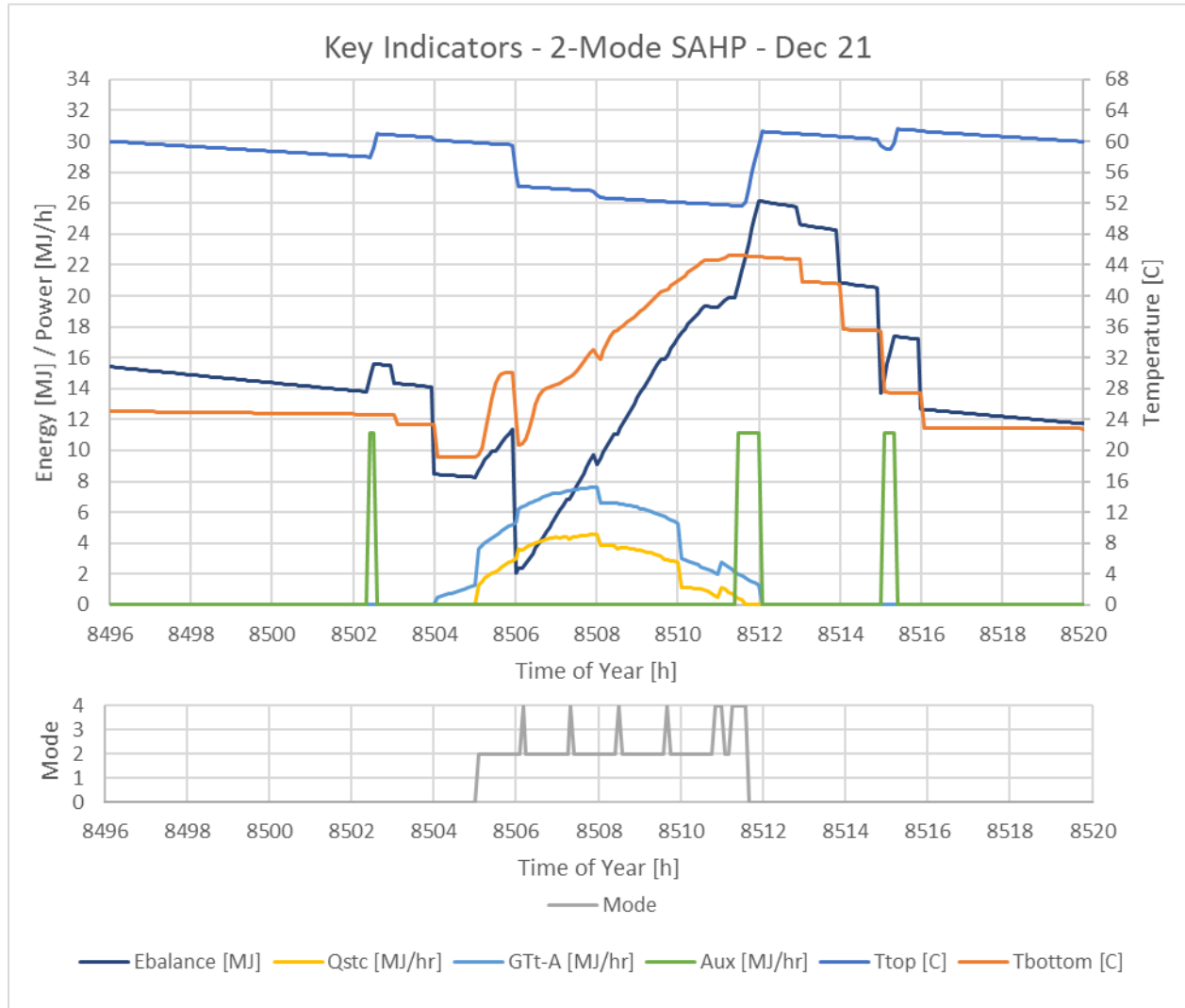


Figure 4.10 - System behaviour chart for 2-mode SAHP system on Dec 21

On December 21st, the system is dominated by the use of the heat pump. Though operation temporarily pauses to let the solar loop warm up, the controller never selects HX as the optimal operating mode, always reverting to HP mode; this is consistent with the results of previous simulations, which showed HX mode to be largely ineffective on this day. Operation extends almost until sundown, when the auxiliary heater switches on to boost T_{Top} . It is interesting to

compare this to the Dec 21 behaviour of the HP-only system. The HP-only system exhibits very similar behaviour, but does not enforce a periodic shutdown of the heat pump – this is because the shutdown is only used in multi-mode systems. Otherwise, the number of warmup hours at the end of the day – and the operating hours in general – are very similar.

This unusually similar behaviour and similar initial conditions allows for a comparison of system performance on this day. The 2-mode system demonstrates notably improved performance compared to the single-mode SAHP in terms of SF and E_{Saved} . This must be attributed to the periodic warmup cycles, which boost the temperature of the solar-side fluid every hour. Because the heat pump COP improves with higher solar-side temperatures, the short warmup periods boost heat pump performance, resulting in lower energy consumption and higher solar heat collection for some time afterwards. The combined effect more than compensates for the productivity lost due to the warmup cycle. The periodic warmup cycles therefore lead to better performance for this system overall, cementing the necessity for a multi-mode system in temperate climates.

4.6 – 3-Mode Dual-Side SAHP (HX, HP or SRS) System

The final system to be simulated, and the subject of this thesis, is the 3-mode system. This system builds upon the 2-mode model by allowing the controller to send heat exchanger outlet fluid to the heat pump inlet for further heat transfer. An additional function is to temper incoming fluid in the heat exchanger so that the heat pump can operate safely for more hours. The behaviour of this system is demonstrated in Figure 4.11 and Figure 4.12 for a summer and winter day, respectively.

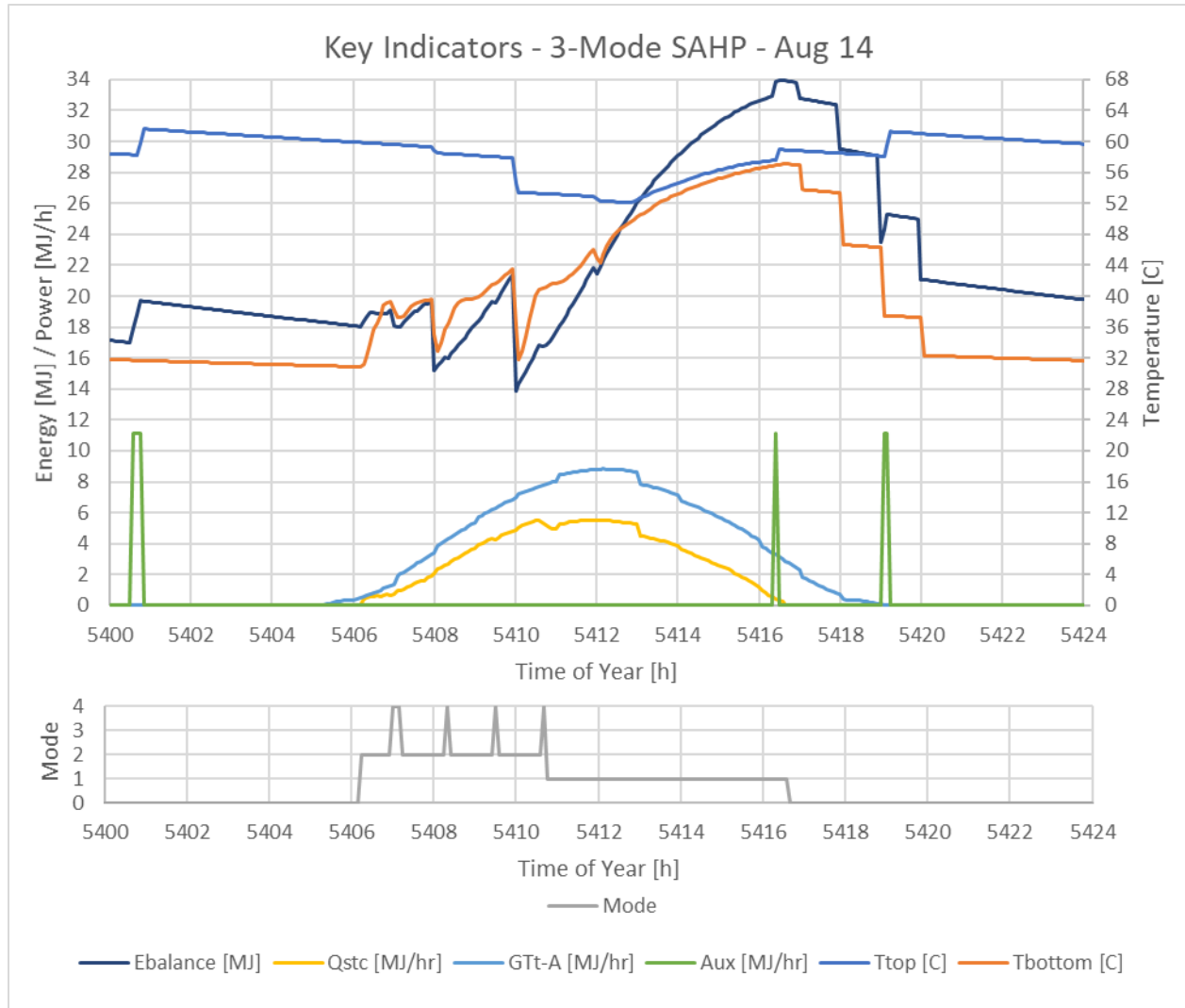


Figure 4.11 - System behaviour chart for 3-Mode SAHP system on Aug 14

The behaviour of the 3-mode system varies only slightly from that of the 2-mode system. Behaviour changes are more likely the result of subtle, long-term differences in tank stratification than of changes in controller logic. Due to higher temperatures in the solar collector, this system is forced to switch into HX mode a little bit earlier than the 2-mode and SRS-only systems, resulting in slightly lower solar gains for this day. This is not expected to be a symptom of a systematic difference, but is perhaps a consequence of relatively high energy storage levels and the slightly different timing of the warmup-cycles experienced on this day.

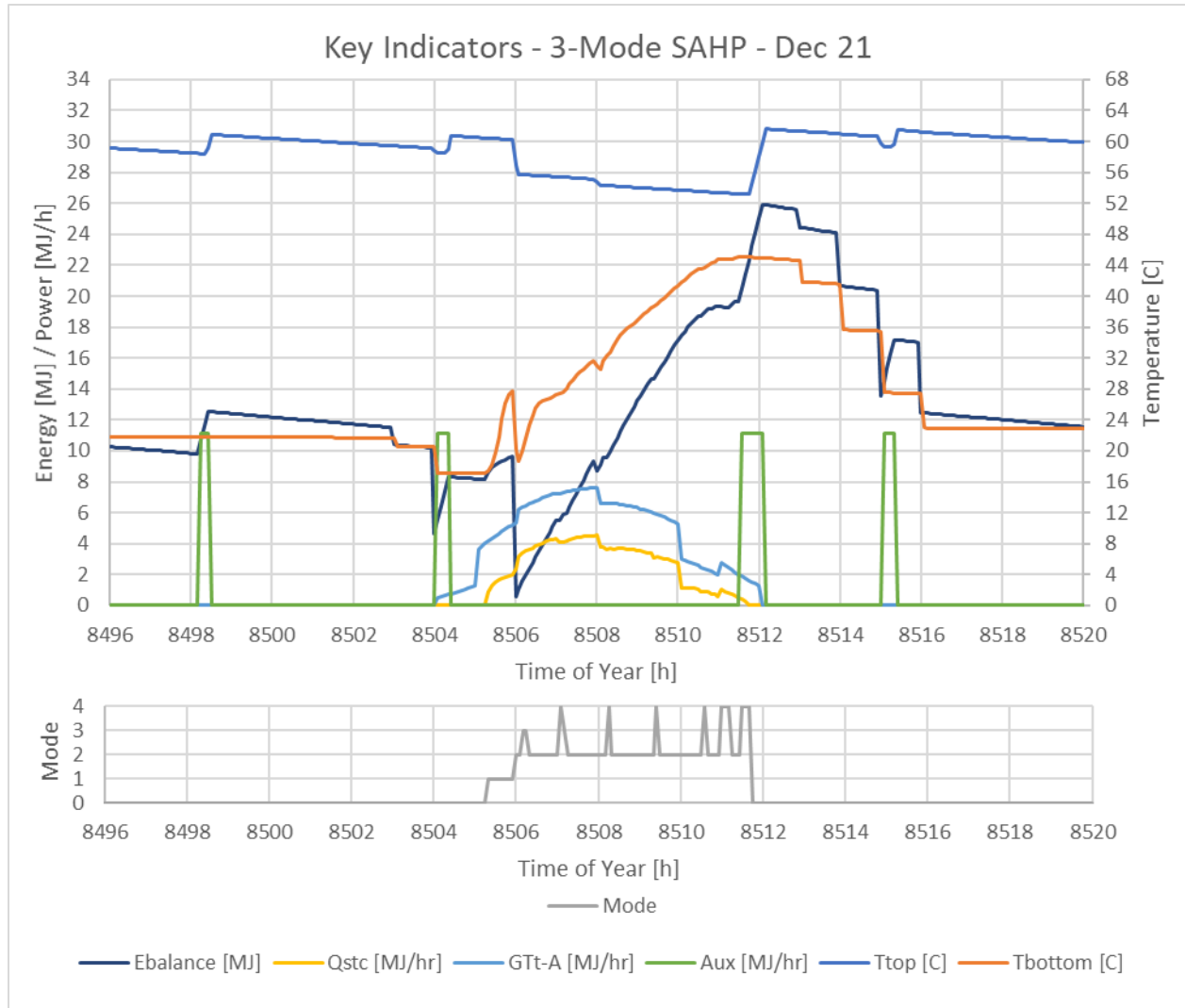


Figure 4.12 - System behaviour chart for 3-Mode SAHP system on Dec 21

Operation on this day commences shortly after the available sunlight intensifies, showing a bit of a time lag in responding to this increased solar availability. This is attributed to a prediction made by the controller where it is estimated that $Q_{STC} < 0$ at that time, meaning that operating would reject heat to the environment. The system initiates itself in HX mode due to high T_{Top} , but uses HP mode for most of the rest of the day. The behaviour is not as energy-conservative as observed in the 2-mode system; however, the 2-mode system ran a tank energy balance of -3.6 MJ on the day, while the 3-mode system ran a 1.3 MJ surplus. The conservative behaviour of the 2-mode

system resulted in a loss of stored energy that will have to be replaced later. This difference in energy balance makes up for the liberal use of electricity, and makes the 3-mode system the only system in the study to run a positive tank energy balance for December 21st.

4.7 – 3-Mode SAHP System – Minimum Power Consumption Design

A prominent feature in the control scheme for the three-mode system is the ability to follow one of two decision-making protocols when selecting an operating mode. Based on the current energy demand in the storage tank, the controller will try to operate using as little electrical energy as possible or it will try to maximize the rate of heat delivery to the tank. These are the Minimum Power (Min P) and Maximum Rate of Heat Gain (Max Q) sequences. The value of including both in a single control plan is to be studied by evaluating the performance and behaviour of the standard “balanced” 3-mode controller against that of a Min P-only and Max Q-only controller design. The balanced version has already been examined in the previous section; the following, including Figure 4.13 and Figure 4.14, evaluates the behaviour of the Min P controller in a 3-mode system.

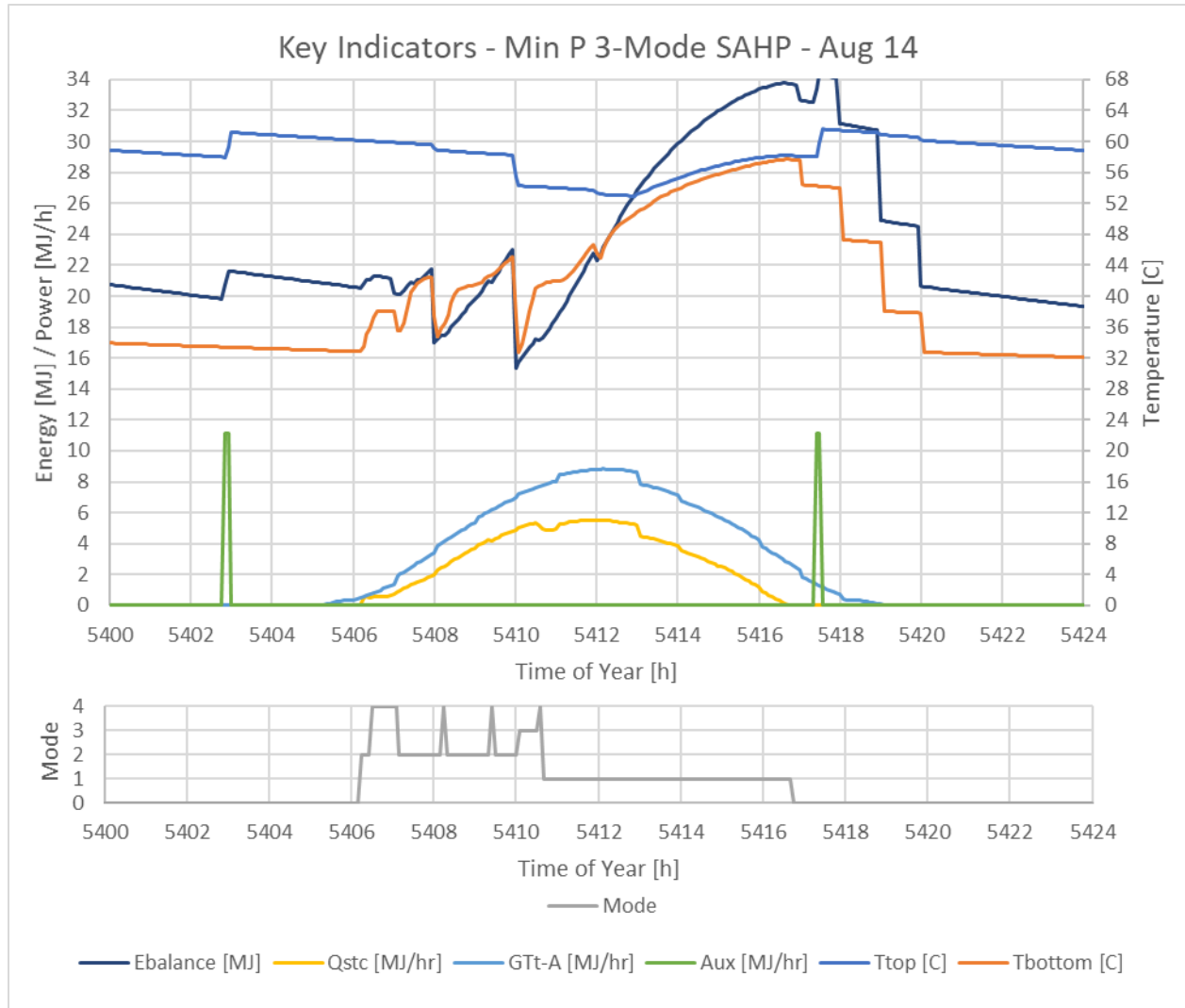


Figure 4.13 - System behaviour chart for 3-Mode SAHP (Min P Design) system on Aug 14

On August 14th, the system performs very well. Delivered temperatures exceed 58 C by the end of operation, and the energy levels in the tank are the highest seen for any system thus far. Stratification is low, as is auxiliary energy use. The system uses the heat pump through most of the morning, but switches to use SRS mode at 10 AM (5410 hours) before changing to HX mode around 11 AM like the other systems do. The behaviour, including the morning warmup cycle, is similar to the 2-mode system. It is worth noting that this is the only system to run a negative energy

balance on the tank for this summer day; however, the high energy storage levels justify the net reduction in stored energy for the day.

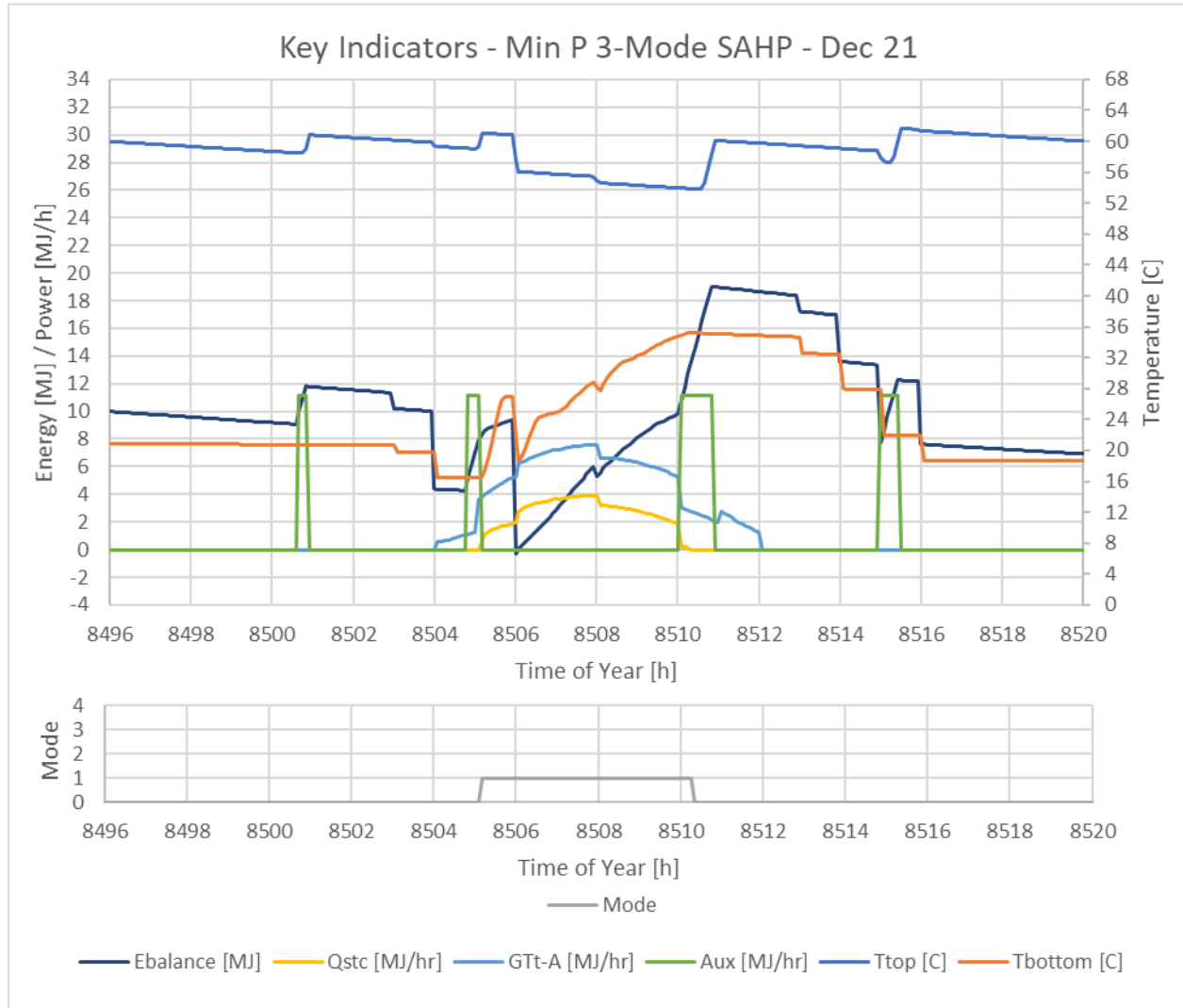


Figure 4.14 - System behaviour chart for 3-Mode SAHP (Min P Design) system on Dec 21

Studying the Min P system on December 21st yields an interesting result. The controller never activates the heat pump on this day, opting instead for HX mode throughout, resulting in performance data that perfectly match the SDHW system. Since the intent of the SAHP is to improve performance on days such as this, the Min P protocol clearly fails to satisfy the objectives of the system.

4.8 – 3-Mode SAHP System – Maximum Rate of Heat Gain Design

The Max Q protocol is primarily focused on heat gains. In the sequence, the controller first identifies the mode that operates with the greatest rate of heat delivery to power consumption ratio (the COP). The controller then evaluates the “most efficient” mode’s ability to meet the demands of the tank; if the most efficient operating mode can also satisfy the heating load, then it should be used. If it cannot satisfy the load, however, the protocol identifies the mode which provides heat the most quickly and attempts to select that as the ideal operating mode. The behaviour of a 3-mode controller designed to always follow this protocol is illustrated in Figure 4.15 and Figure 4.16.

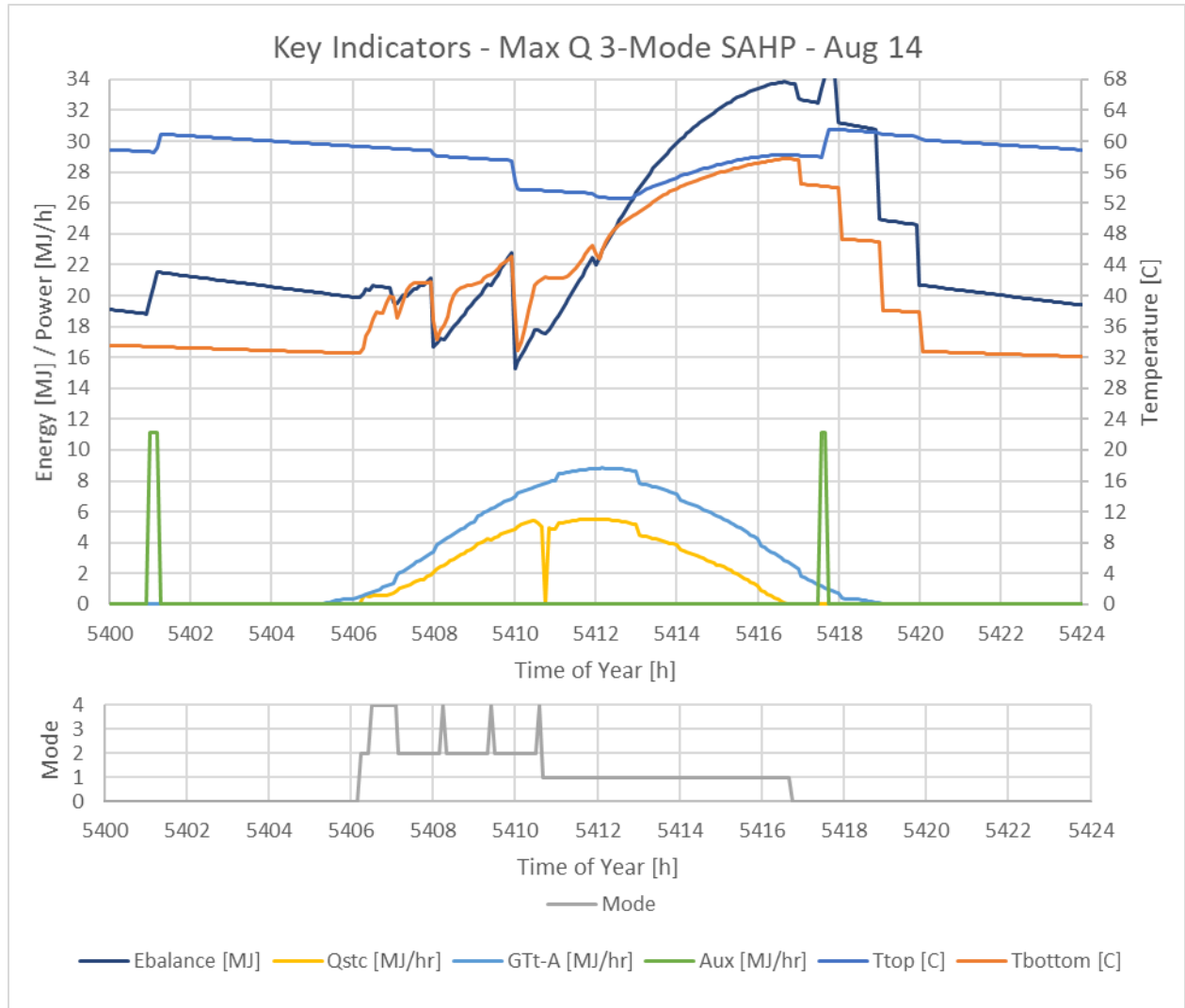


Figure 4.15 - System behaviour chart for 3-Mode SAHP (Max Q Design) system on Aug 14

As observed in the Min P system, the energy storage levels are very high, as is the delivered water temperature (compared to other designs studied here). System behaviour again mimics that of the 2-mode system, as SRS mode is never selected during this day.

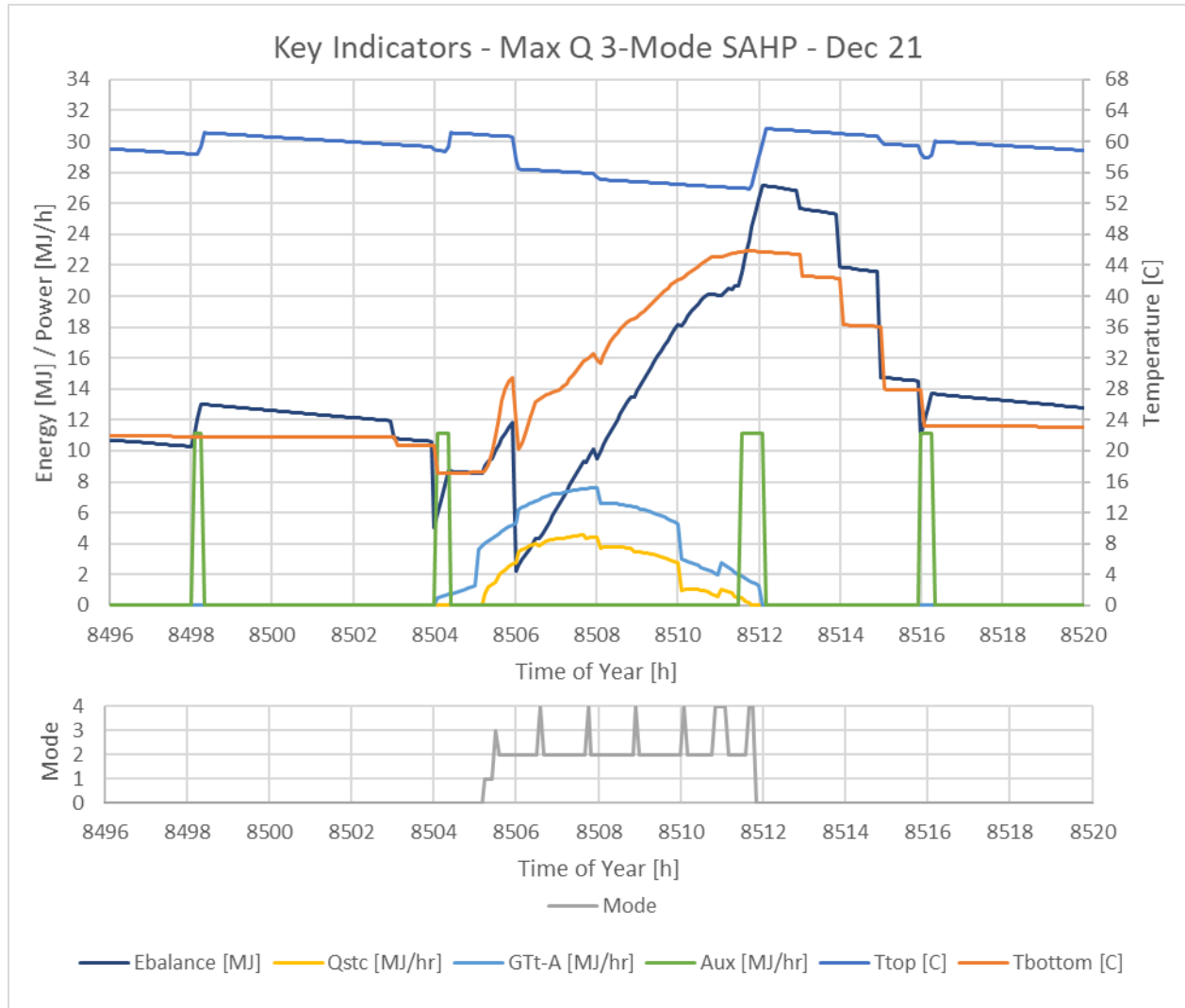


Figure 4.16 - System behaviour chart for 3-Mode SAHP (Max Q Design) system on Dec 21

The winter day behaviour for this system is highly comparable to the balanced design. In fact, the charts are almost identical except for a shorter HX mode cycle at the start of operation for the day. As discussed previously, this system makes improvements on the SDHW design, but comes up short in a direct comparison to the 2-mode design (which runs a high energy deficit for the day, while this 3-mode design runs a large energy profit). Thus, there is little evidence to suggest that the choice or availability of the different decision-making protocols has a high impact on the system's daily behaviour.

4.9 – Annual Performance Evaluation of Compared Systems

The previous sections in this chapter have presented charts that show system behaviour characteristics for a single summer and winter day. Those charts were produced from the last day of 3-week simulations, with the lead time provided so that the systems could stabilize and be more appropriately represented by the behaviour charts. Also discussed were performance data for the days in question; these data, however, only represent performance on individual days and do not encompass broader seasonal or annual patterns. This was alluded to when studying the Min P and Max Q systems and how they out-performed the 2-mode SAHP in terms of E_{Saved} . The following, then, presents annual performance data for the system designs and control schemes studied in this chapter, to appropriately compare and evaluate the performance of the systems. The KPIs for the annual simulations are generally summarized in Table 4-1.

Table 4-1 - KPI results for annual simulations of all systems

System	Solar Fraction	E_{Bought} [MJ]	E_{Saved} [MJ]	$E_{\text{Delivered,solar}}$ [MJ]	E_{Lost} (%)	Med-T Draws	Low-T Draws
Electric	0.0%	12445	0	0	5.2%	0	0
SDHW	38.2%	8226	4219	5296	12.7%	270	6
SAHP	60.9%	7822	4624	6200	14.8%	322	3
SRS-only	44.1%	8910	3535	4739	12.4%	274	6
2-mode	58.8%	7397	5048	6526	14.3%	248	3
3-mode	56.6%	7455	4990	6433	14.2%	213	3
Min P	44.8%	8051	4394	5658	13.5%	257	6
Max Q	57.6%	7450	4995	6491	14.4%	250	3

It is observed that the solar fraction for SAHP systems far exceeds that of the SDHW design. Energy savings are increased by over 15% by introducing a heat pump, to a total reduction in energy use of 40% compared to an electric system. All solar systems exhibit higher losses to the environment than the electric tank due to destratification; there seems to be a direct trade-off between tank losses and energy savings. Likewise, the temperature performance of all systems is

fairly good, with very rare instances of draw temperatures below 50 C and less-than-daily occurrences of draw temperatures below 55 C; systems with better energy performance tend to exhibit cooler draw temperatures.

4.9.1 – System Evaluation by Solar Performance

Solar performance of the systems can be evaluated by a combination of results. The total solar gain (E_{STC}) indicates how effectively the collector was used. Solar fraction is applied to indicate how much of the original tank heater load has been displaced by the solar system. $E_{Delivered,solar}$ identifies how much of the energy delivered by the solar system to the tank can be attributed to the sun itself, while the solar efficiency η_{Solar} helps quantify the efficiency of the solar system itself by dividing $E_{Del.,sol.}$ by E_{STC} . Both values give conservative estimates by including the assumption that all power consumed by the pumps and compressor is delivered to the storage tank as heat. Values for these indicators from annual simulations are summarized in Table 4-2.

Table 4-2 - Solar performance indicators for annual simulations

System	E_{STC} [MJ]	Solar Fraction	$E_{Delivered,solar}$ [MJ]	η_{Solar}
SDHW	6075	38.2%	5296	87.2%
Min P	6610	44.8%	5658	85.6%
SRS-only	6799	44.1%	4739	69.7%
SAHP	7350	60.9%	6200	84.4%
3-mode	7414	56.6%	6433	86.8%
Max Q	7526	57.6%	6491	86.2%
2-mode	7613	58.8%	6526	85.7%

In Table 4-2, the electric system is not shown since it has no solar aspect; the other systems have been ordered by the amount of solar gain they achieved from lowest to highest. Generally speaking, SF increases with solar gain, though this is not always the case; this is due to the electric loads due to heat pump usage not being accounted for in solar fraction. There is a consistent positive

relationship between solar gain and $E_{Del.,sol.}$, with the exception being the SRS-only system due to frequent heat losses via HX during operation. Interestingly, the solar efficiency decreases consistently as solar gain increases – this is most likely due to increased losses in the solar system equipment as hotter fluid cycles through it.

There is little difference in the performance of the 1-mode, 2-mode and 3-mode SAHP systems, and the data suggests that the inclusion of SRS mode is, at best, not useful on a large scale. E_{STC} varies by less than 5%; the other indicators behave similarly. These systems collect 20% more sunlight, displace almost 50% more auxiliary load and deliver 17% more solar energy than the SDHW system, and the efficiency of the solar system is reduced by no more than 2% as a result.

4.9.2 – System Evaluation by Energy Performance

The energy performance of the system is evaluated using additional KPIs. $E_{Delivered}$ is a measurement of the quantity of energy that enters the tank at the hot solar water return port, minus the quantity of energy that leaves at the cold solar water outlet; therefore, $E_{Delivered}$ shows how much energy the solar system has delivered to the tank. This plus $E_{Auxiliary}$ represents the total energy consumption of the system from all sources, including E_{Loss} (the quantity of energy lost from the tank to the environment). The percentage of energy retained and available to the load, after accounting for tank energy losses, is represented by η_{Tank} (the storage tank efficiency). Finally, E_{Saved} compares the amount of electricity purchased to operate a system to the amount required for the electric DHW system – the difference is the energy savings that are ultimately achieved by the solar-assisted hot water system. The results for the studied systems in these categories is presented in Table 4-3.

Table 4-3 - Energy performance indicators for annual simulations

System	E _{Delivered} [MJ]	E _{Auxiliary} [MJ]	E _{Loss} [MJ]	η _{Tank}	E _{Saved} [MJ]	E _{HPC,Compressor} [MJ]
Electric	0	12445	687	94.5%	0	0
SDHW	5836	7686	2004	85.2%	4219	0
SRS-only	6697	6953	2137	84.3%	3535	1403
Min P	6838	6871	2195	84.0%	4394	582
3-mode	8485	5403	2358	83.0%	4990	1483
Max Q	8667	5273	2424	82.6%	4995	1586
2-mode	8795	5128	2407	82.7%	5048	1688
SAHP	9154	4868	2533	81.9%	4624	2351

The systems are arranged by order of E_{Delivered}. A nearly-linear inverse relationship between E_{Delivered} and E_{Auxiliary} is immediately apparent, indicating that the energy returned from the solar loop is directly useful. This relationship is charted in Figure 4.17 below:

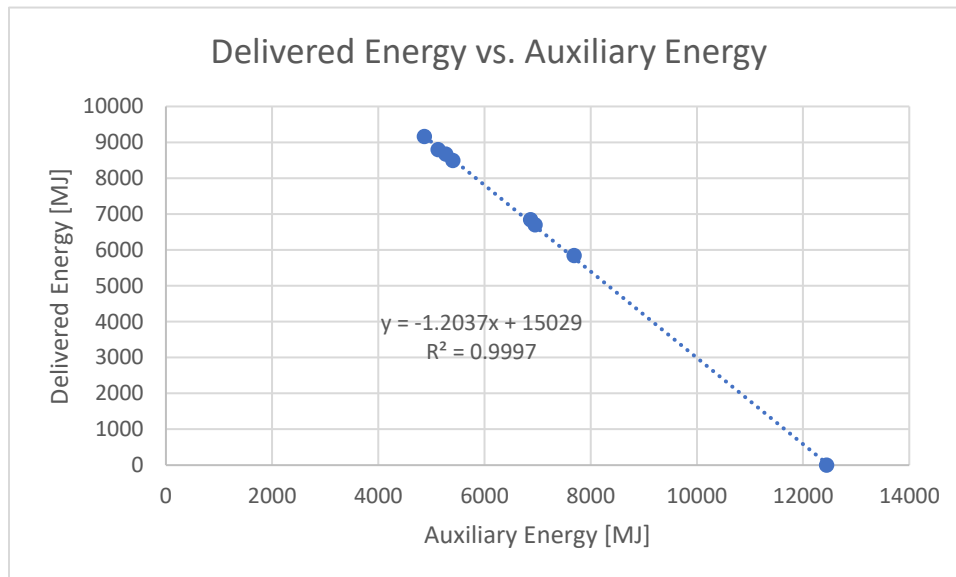


Figure 4.17 - Effect of delivered energy on auxiliary usage

The y-intercept of the linear trendline indicates that the solar system would have to deliver about 15 000 MJ to reduce auxiliary usage to 0 MJ. This implies that every 1.2 MJ gained through the solar system offsets 1 MJ of auxiliary heat, indicating that the delivered energy is of fairly high quality.

Tank losses increase (and tank storage efficiency decreases) as the amount of delivered energy rises, but there is not such a strong linear relationship between the two variables. The behaviour charts from previous sections showed that the SAHP systems exhibited far weaker stratification, which would contribute directly to increased tank losses. Figure 4.18 and Figure 4.19 show the stratification observed in the storage tank for the electric and 3-mode SAHP system on August 14th, respectively.

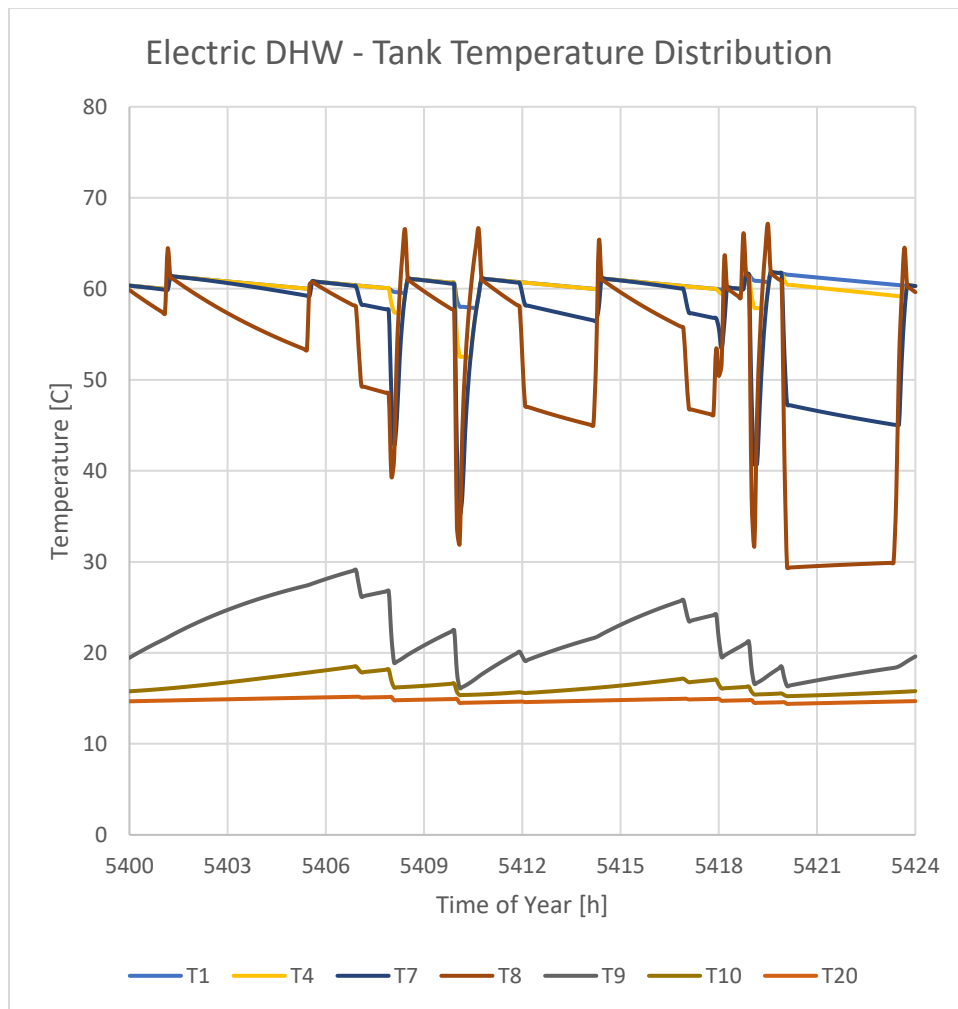


Figure 4.18 - Electric DHW storage tank temperature distribution on Aug 14

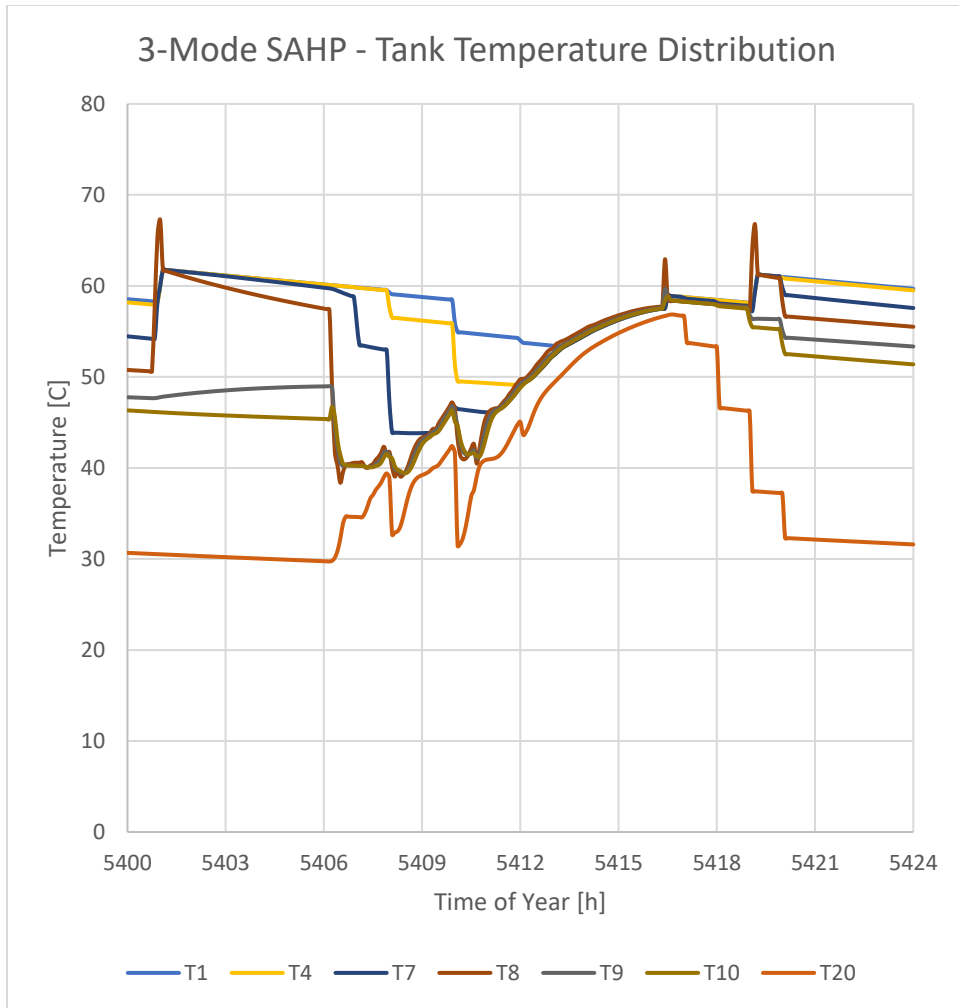


Figure 4.19 - 3-Mode SAHP storage tank temperature distribution on Aug 14

The SAHP's storage tank is shown to be entirely above 30 C, at all tank nodes, for the entire day; the top half of the tank (T10 and above) never falls below 40 C. In contrast, not a single node in the entire bottom half of the electric tank rises above 20 C at any point during the day. While the top quarter of the electric tank is much warmer than the SAHP tank, the losses caused as a result are relatively small. Because half of the electric tank is below room temperature, this part of the tank will experience heat gains that offset some of the losses at the top. The SAHP tank, on the other hand, is losing heat to the environment from every single tank node, hence the increased losses for the more active, less stratified systems. Not only is the tank storing more energy, but it

is storing the energy in such a way that it can be lost to the environment. The use of chemical or latent storage is intended to counter this effect, but it comes at high cost and with fairly high risk, as discussed by Pinel et al [17, 37].

E_{Saved} is the last energy performance indicator to be evaluated. Used frequently throughout this thesis, it represents the net reduction in purchased energy caused by the solar system. Since the reference is an electric system here, E_{Saved} is measured directly in MJ. There is a general upward trend in E_{Saved} as $E_{\text{Delivered}}$ rises, but it is not perfect. Varying heat pump operation times will influence the amount of energy saved; both outliers in the upward E_{Saved} trend can be explained by their relatively high heat pump usage compared to adjacent designs. While the single-mode SAHP system delivers the most energy and uses the least auxiliary backup, it spends 40% more energy on heat pump operation than the next-highest HP-user, allowing the 2-mode and 3-mode systems to achieve greater energy savings with better storage tank efficiency. This is because they can operate without using the heat pump when sufficient sunlight is available, greatly improving their operating efficiency (both systems fully out-perform the single-mode SAHP in the solar category as well).

The $E_{\text{HP,Compressor}}$ data yields another interesting piece of information. The 3-mode (balanced), Max Q and 2-mode systems show very similar energy savings (5019 +/- 29 MJ), but their HP usage varies by nearly 200 MJ. This may indicate a performance ceiling, whereby different control plans with the same equipment are unable to surpass a certain performance level.

Overall, the SAHP designs all perform to a similar level. They deliver 45% more energy to the tank and use 30% less auxiliary energy than the SDHW system, but suffer a 25% increase in tank losses. The net result is a total energy savings at least 10% greater than with the SDHW, with multi-mode systems providing the best results (achieving 40% overall energy savings). The

inclusion of SRS mode once again seems to hurt performance overall, with the more aggressive 3-mode control schemes substantially out-performing the Min P version just as seen in the solar performance category.

4.9.3 – System Evaluation by Thermal Performance

The thermal performance of the system is primarily focused on delivered water temperatures. It has been observed that the solar heating systems rarely, if ever, delivered water hotter than 60 C to the storage tank, requiring a boost from the auxiliary heater to reach the set point temperature. Table 4-4 summarizes the thermal performance data, with the systems organized by the number of medium-temperature (50-55 C) draws.

Table 4-4 - Thermal performance indicators for annual simulations

System	Med-T Draws	Low-T Draws
Electric	0	0
3-mode	213	3
2-mode	248	3
Max Q	250	3
Min P	257	6
SDHW	270	6
SRS-only	274	6
SAHP	322	3

The first piece of information in this table shows that the electric system has perfect thermal performance, with no draws below 55 C. The solar-assisted systems all experience about 5 medium-temperature draws per week. Generally, systems with more frequent medium-temperature draws experienced more low-temperature draws as well – however, no system experienced low-temperature draws more than 6 times in a year, which may be considered negligible.

Interestingly, the SDHW system is not the worst system in terms of thermal performance – this dubious honour is held by the HP-only SAHP system, with almost 1 draw per day. While most designs showed medium-temperature draw counts close to 260 (averaging almost 5 times per week), two systems stand out – the HP-only SAHP (322 occurrences or at least 6 times per week) and the 3-mode SAHP (213 occurrences or about 4 times per week). While the solar and energy performance of the 3-mode SAHP is directly comparable to the HP-only and 2-mode SAHP systems, here it stands clearly ahead of the alternatives.

In this thesis, draws are performed 8 times per day, making the annual number of draw events 2920. Thus, the probability of a draw temperature above 50 C is at least 99.8% for all systems, and at least 89% for draws above 55 C. This substantially exceeds the hot draw rates observed by Bannister, where draws above 55 C occurred for no more than 40% of all instances [14]. Worth noting is Bannister’s target condition of 55 C to the load (this thesis uses 60 C) [14].

4.9.4 – System Evaluation – Overall

The previous sub-sections evaluated the studied systems and control plans based on solar, energy and thermal performance; the overall findings are repeated in summary here.

The inclusion of a heat pump substantially improves solar performance compared to a traditional SDHW, but the control scheme of choice has little effect. More “free” solar energy, then, can be captured by a given system through the addition of a heat pump. Such a system also delivers far more energy to the tank than a SDHW system, but increases tank losses. The result is a relatively small improvement in energy savings compared to the SDHW. The solar and energy performance indicators show that an aggressive control scheme will outperform a more passive design, and that the inclusion of a SRS mode does not enhance the solar collection or energy delivery/savings rates

of an SAHP system. Finally, the thermal performance indicators show that including a SRS mode improves the thermal performance of the system, helping it meet its draw temperature targets more frequently than any other design. The importance of a balanced control scheme is highlighted here as well – while the Max Q design showed similar-quality performance to the balanced 3-mode design in the solar and energy fields, it was decisively out-performed in the thermal category.

The conclusion of the study of annual performance data, then, is that the 3-mode balanced controller is the best of the studied system/controller designs for the given set of system parameters. Considering the minimal additional investment required to use a 3-mode system compared to a 2-mode system, the option to operate in SRS mode is absolutely recommended for a single-tank, dual-side SAHP design. The economic viability of a SAHP system over a SDHW system is outside the scope of this thesis, but given the substantial improvements in the solar, energy and thermal fields of system evaluation provided by the HP, its inclusion in a solar hot water system is recommended.

Chapter 5 – Set-Point Temperature and Climate Effect Studies

Beyond simply comparing the performance of a 3-mode system against various established designs, this thesis project has considered questions regarding system parameters as well. Some parameters, such as the system flow rate used or the heat pump sizing, are selected using small parametric studies that attempted to find peak performance conditions. Other values or settings are simply carried over from the work of Wagar and Bannister (such as pump, solar collector and heat exchanger parameters), or set in a default state for a lack of validation testing (such as the inversion mixing flowrate for the domestic hot water storage tank). An experimental test setup is under construction, with a new heat pump, for the development of a new heat pump model and validation of the 3-mode system simulations. With such plans in place, studies for determining “optimal” controller settings are left for after the completion of the experimental setup. As such, this thesis has focused heavily on comparing different designs under the same settings to this point.

This chapter will detail the results of further studies to answer two major questions about the system. The first challenges the use of a rigid 60 C set-point temperature by asking, “What if the set-point temperature is lowered during periods of low demand?” The concept is that if the set-point is lower, the controller is less likely to use auxiliary backup heaters and more likely to select a low-power operating mode through the Min P protocol. By designing the controller with T_{Set} as an input, a forcing function can be modified to deliver a T_{Set} pattern of any kind to test this hypothesis; the study will evaluate system performance at various rigid set-point temperatures and with various T_{Set} profiles.

The second question considers the effect of climate on performance. Bannister [14] performed a thorough study of system performance in 8 locations across 5 climate zones for the dual-tank

SAHP; this thesis includes a similar, but scaled-down, study for the 3-mode system. However, the results of these studies showed that the inclusion of SRS mode did little to improve system performance; the 3-mode and 2-mode SAHP systems were essentially equivalent, with small trade-offs in the performance categories. Therefore, the study will not only include the 3-mode system and the bases for comparison (the electric and SDHW systems), but will also include the 2-mode system. This inclusion will help to either confirm or refute the conclusions of Chapter 4 which determined that the 3-mode system was not measurably better than the 2-mode design, while also establishing the heat pump's ability (or inability) to enhance system performance in various climate zones.

5.1 – Set-Point Temperature Effect Study

Kaygusuz [18] posited that using a variable set-point temperature may enhance the performance of a SAHP system. This thesis considers the possibility of employing a daily set-point cycle to reduce the load on the system. Since the controller is designed to choose between the Min P and Max Q protocols based on demand, a variable set-point can be employed to influence which protocol the controller will tend to choose at certain times in the day. For example, lowering T_{Set} should cause the controller to tend towards the Min P protocol. This protocol naturally favours HX mode; thus, lowering the set-point during mid-day hours influences the system to rely on HX-only mode at the times when solar irradiation is at its peak. Raising T_{Set} near hours of peak demand, on the other hand, is likely to influence the controller to choose Max Q more often, ensuring that the load is met.

The TRNSYS simulation is constructed with T_{Set} , T_{Target} and t_{Target} as forcing-function inputs to the controller, to allow the user to program them to be variable throughout the day. T_{Set} is input as the set-point temperature of the system and is used for making control decisions. T_{Target} is input as

the temperature required by the load; it is used to this purpose in the calculation of Q_{Load} . It is set to 62 C in this thesis so that the controller always makes a conservative estimate of Q_{Load} . Finally, t_{Target} is entered as the estimated time until the next draw, thereby defining the required rate of heat delivery Q_{Load} . In this thesis, it is held to a constant 0.5 h to avoid being caught off-guard by a draw, but in a more advanced control scheme with pattern-learning capabilities, t_{Target} could be used to reduce Q_{Load} if the next water draw is not expected for several hours.

The following study will examine the effects of various fixed set-point temperatures and numerous variable set-point temperature profiles. Studies of complex profiles, which included stepped T_{Set} variations from 45 C to 60 C, indicated that auxiliary usage was increased by frequent and extreme changes in T_{Set} , and so the variable T_{Set} profiles to be studied will be simple and more moderate. The variable profiles studied will consider T_{Set} step sizes of 2, 5 and 10 C between the highest and lowest T_{Set} values used. The profiles will consist of two “high T_{Set} ” periods and two “low T_{Set} ” periods, with the lows centred around midnight and noon, and highs of equal duration centred around 7 AM and 5 PM. The total daily duration of the lows/highs will be 8/16, 12/12 or 16/8 hours per day to show the effect of the balance between the high and low settings.

Finally, the temperature levels to be studied must be established. System performance with fixed set-point temperatures will be evaluated for integer values from 55 to 62 C. Stepped profiles will be studied with high T_{Set} values of 56, 58, 59, 60 and 62 C. The studies with high $T_{\text{Set}} = 59$ C are included when, after performing the fixed set-point study, it becomes evident that the system is highly sensitive to T_{Set} . In addition to this sensitivity, it is noted in the study that better performance in the solar/energy categories leads to worse performance in the thermal category, and vice versa; therefore, improved sample resolution is desired in the middle of the range instead of at the extrema.

The result is a fairly large number of simulations. One simulation is needed to provide the performance of the electric DHW system, 8 are needed for the fixed T_{Set} study and the effect study with 5 high T_{Set} s x 3 T_{Set} steps x 3 low/high ratios adds 45 further simulations for a total of 54. At a rate of about 3 minutes per simulated year, it would take more than 2.5 hours just to run the simulations, before processing the data and adjusting the parameters. Therefore, to save time, these simulations will consider the months of April-June, covering most of the Spring season; the resulting performance values are expected to be close to the annual average, allowing the data to be extrapolated if desired.

5.1.1 – Fixed Set-Point Temperature Study

The fixed T_{Set} study consists of 9 datasets – one for an electric DHW system, and 8 for the 3-mode SAHP system with set-point temperatures from 55 C to 62 C. T_{Target} is held at 62 C and t_{Target} is held at 0.5 h for all simulations in the study. The results of the Spring simulations are summarized in Table 5-1:

Table 5-1 - KPI results for seasonal simulations of fixed T_{Set} conditions

Condition	Solar Fraction	E_{STC} [MJ]	$E_{Auxiliary}$ [MJ]	E_{Saved} [MJ]	Med-T Draws	Low-T Draws
Electric	0.0%	0	3047	0	0	0
55 C	72.5%	2260	839	1600	208	12
56 C	70.5%	2259	897	1536	167	6
57 C	68.1%	2256	973	1462	137	2
58 C	65.8%	2256	1041	1385	110	1
59 C	65.7%	2259	1044	1376	91	1
60 C	64.3%	2249	1087	1341	56	1
61 C	63.3%	2262	1117	1294	40	0
62 C	60.7%	2228	1197	1244	21	0

There are clear and opposite trends in this data. Solar fraction and E_{Saved} increase as T_{Set} decreases, while the number of medium-temperature and low-temperature draws increases rapidly at lower

set-points. This indicates that an ideal choice of set-point must be somewhere in-between the extremes presented. Most KPIs show a roughly quadratic reduction as T_{Set} increases (auxiliary use increases in the same manner), highlighting the increasingly extreme performance results at lower set-point temperatures. In terms of solar performance, the amount of energy collected shows little variance – solar fraction increases at low T_{Set} because of reduced auxiliary loads, not because of improved solar collection. The data shows slightly increased heat pump use at higher T_{Set} ; this and the auxiliary use reductions make up most of E_{Saved} . Table 5-2 indicates this by comparing the rise of heat pump and auxiliary use to the drop in E_{Saved} as T_{Set} rises; Δ_{Net} shows just how closely the change in the amount of energy saved corresponds to the change in the amount of heat pump and auxiliary consumption.

Table 5-2 - Change in heat pump and auxiliary consumption vs. change in energy savings

Condition	$E_{HP,Compressor}$ [MJ]	Δ_{HP} [MJ]	$E_{Auxiliary}$ [MJ]	Δ_{Aux} [MJ]	E_{Saved} [MJ]	Δ_{Saved} [MJ]	Δ_{Net} [MJ]
55	437.3		838.9		1600.0		
56	444.5	7.2	897.4	58.6	1536.1	-63.9	1.8
57	441.3	-3.2	972.8	75.3	1462.3	-73.8	-1.7
58	451.0	9.7	1040.7	67.9	1385.5	-76.8	0.8
59	458.1	7.1	1044.4	3.7	1376.0	-9.5	1.3
60	450.1	-8.0	1087.2	42.8	1341.0	-35.0	-0.2
61	468.0	17.9	1116.9	29.8	1293.6	-47.4	0.3
62	439.6	-28.4	1196.9	80.0	1244.2	-49.3	2.3

From a thermal perspective, the results are clear – with $T_{Set} < 57$ C there is a high and constant risk of delivering water below 50 C. The number of medium-temperature and low-temperature draws in this range are four times higher than at higher set-points. For fixed set-point systems, there is little encouragement to use a temperature other than 60 C. Setting T_{Set} any lower results in far more frequent low-temperature draws, with little improvement in energy savings; raising T_{Set} above 60

C provides rapidly diminishing returns on thermal performance but substantially lowers the energy savings achieved.

5.1.2 – Variable Set-Point Temperature Study

The variable T_{Set} study consists of 45 datasets. The study considers the effect of T_{Set} itself at 5 levels, the T_{Set} step size at 3 levels and the duration of the low and high steps at 3 levels. High T_{Set} values studied will be 56, 58, 59, 60 and 62 C; the step sizes will be 2, 5 and 10 C, and the duration low/high ratios will be 16/8, 12/12 and 8/16 hours per day (or 1/3 high, 1/2 high and 2/3 high). T_{Target} is held at 62 C and t_{Target} is held at 0.5 h as before. This experimental design gives substantial statistical weight to the results.

High T_{Set} Effects

Figure 5.1 and Figure 5.2 summarize the effect of high T_{Set} values on the solar, energy and thermal KPIs for the 3-mode SAHP. Each data point is the average value of all data points for which the appropriate high T_{Set} was used, including data from simulations which used a fixed set-point temperature (10 values each). The data from the fixed T_{Set} study are used by considering those simulations as tests with a 0/24 low/high ratio, with a T_{Set} step size of 0 C.

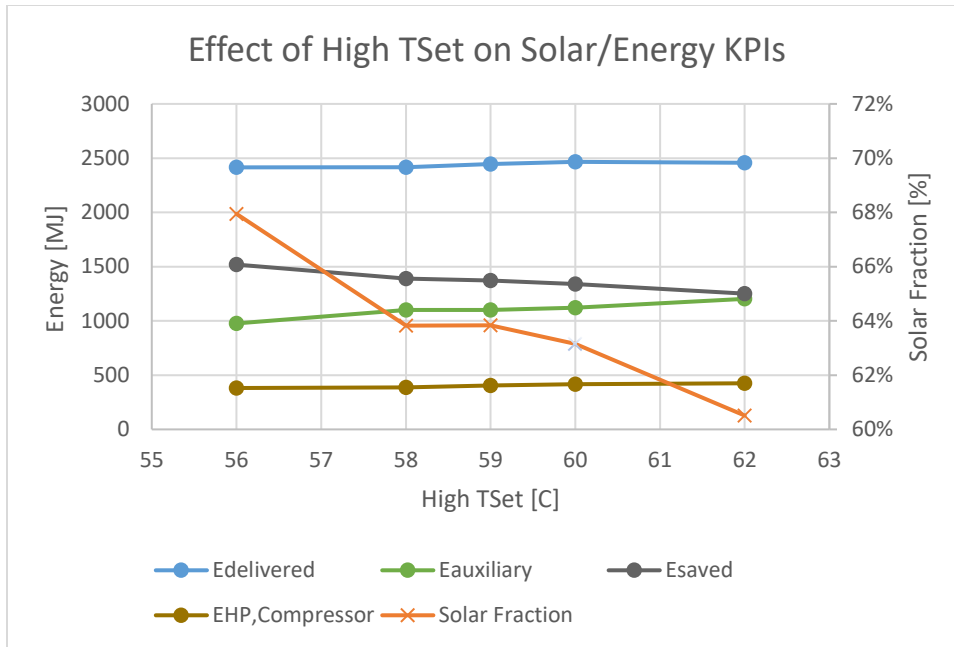


Figure 5.1 - Influence of high T_{Set} on solar and energy KPIs

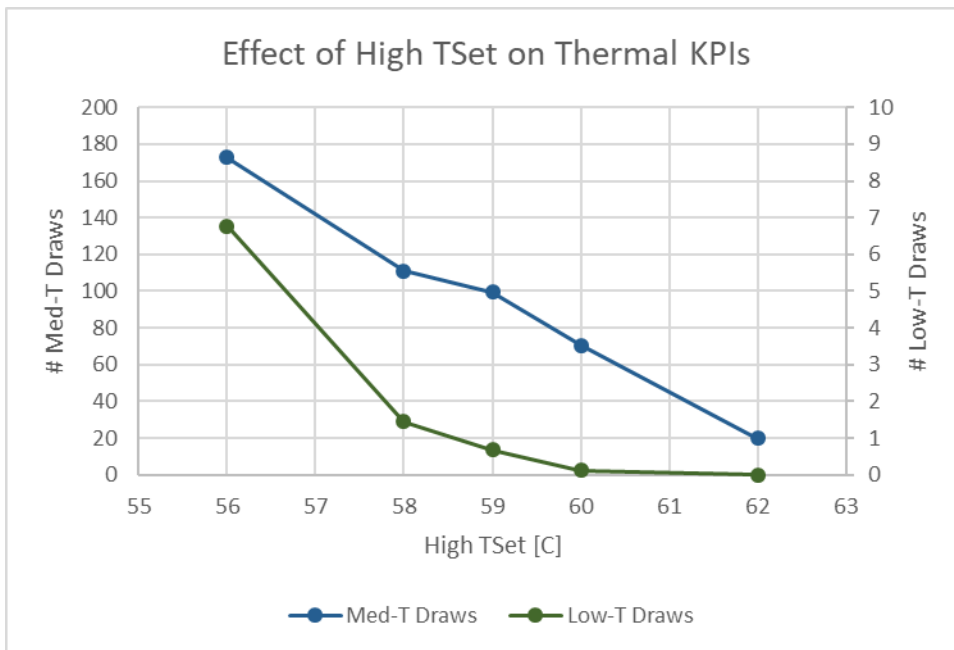


Figure 5.2 - Influence of high T_{Set} on thermal KPIs

$E_{Delivered}$ and $E_{H\&P,Compressor}$ both increase slightly with high T_{Set} . More notable is the increase in $E_{Auxiliary}$ and the corresponding loss of E_{Saved} – the effect is mirrored almost perfectly, as should be expected. Solar fraction drops drastically as temperature increases, but the data shows that this is

almost entirely due to the rising auxiliary use. Finally, very frequent low-temperature draws are associated with low T_{Set} , as noted in the previous section. The results re-affirm the conclusion that the “high” set-point temperature should be 60 C; to change in either direction leads to high costs in either thermal or energy performance, costs which outweigh the gains made in the other category.

T_{Set} Step Effects

Figure 5.3 and Figure 5.4 summarize the effect of the chosen T_{Set} step on performance KPIs. Data from the fixed set-point study are used, with a T_{Set} step of 0 C (averaged from 8 data points). Data from the variable set-point study are averages of 15 data points, measured from simulations with the appropriate step size.

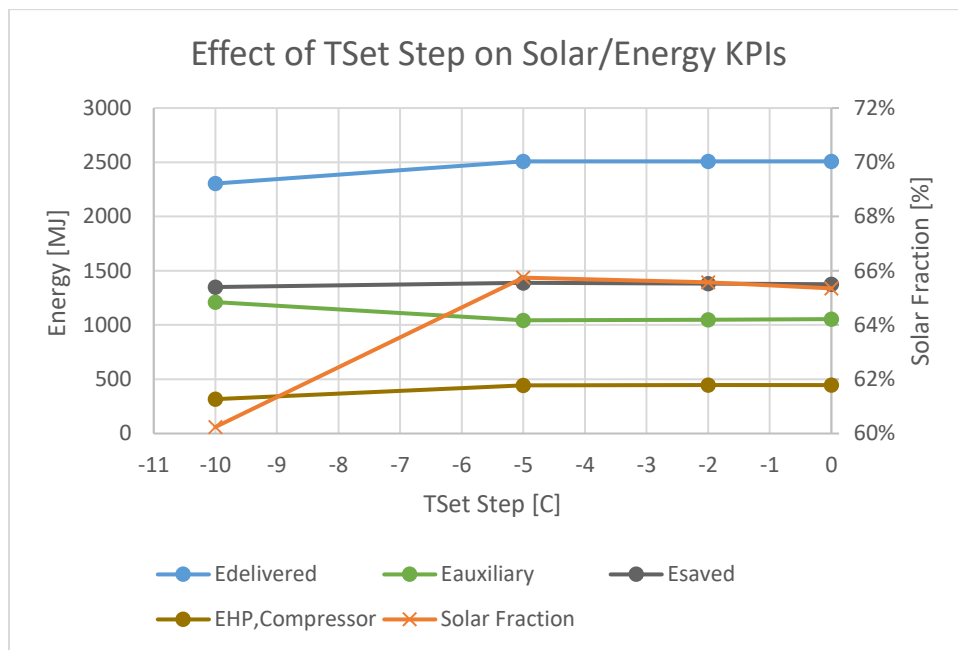


Figure 5.3 - Influence of T_{Set} step size on solar and energy KPIs

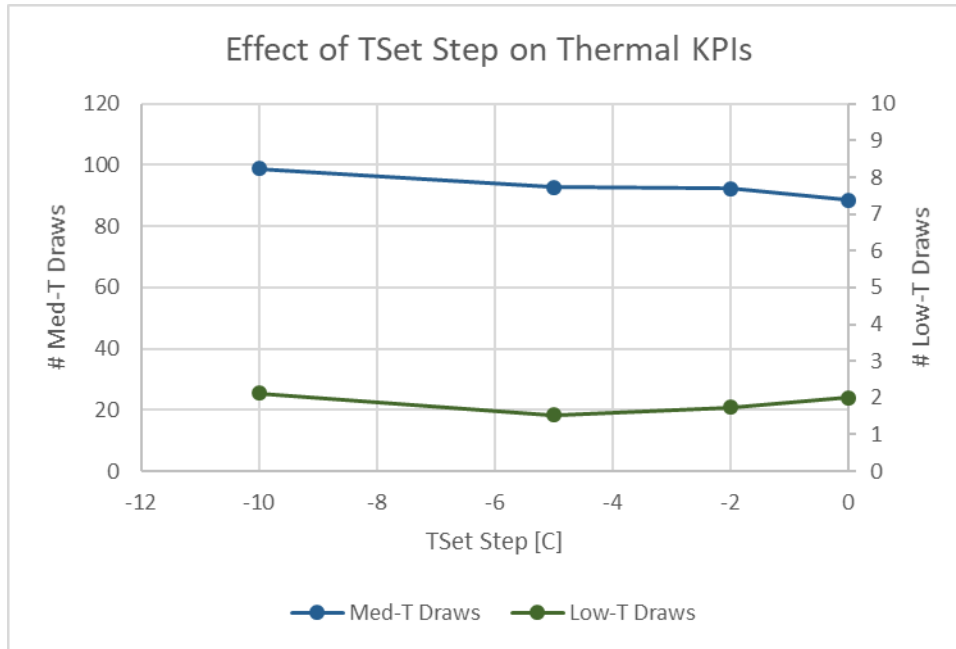


Figure 5.4 - Influence of T_{Set} step size on thermal KPIs

Most of the KPIs appear to be stable as the step size varies, except for a universal drop in performance at a 10 C step size. There is little variation of significance across the smaller steps, but solar and energy performance seem to improve when using a 2-5 C step, with little negative impact to thermal performance. Auxiliary use drops slightly thanks to the relaxed control scheme that a small T_{Set} step provides – it is likely that the overnight use of auxiliary heat is reduced since the upper tank temperatures are allowed to slip a few degrees. However, there does not appear to be any substantial reduction in E_{Bought} as a result of using a variable profile.

Low/High Duration Ratio Effects

The ratios studied are defined by the number of hours in a day spent in either condition. The fixed set-point data are included as a 0/24 low/high ratio for consistency with the previous effect studies. The data points presented in Figure 5.5 and Figure 5.6 are average values taken from 15 simulations which applied the relevant duration ratio.

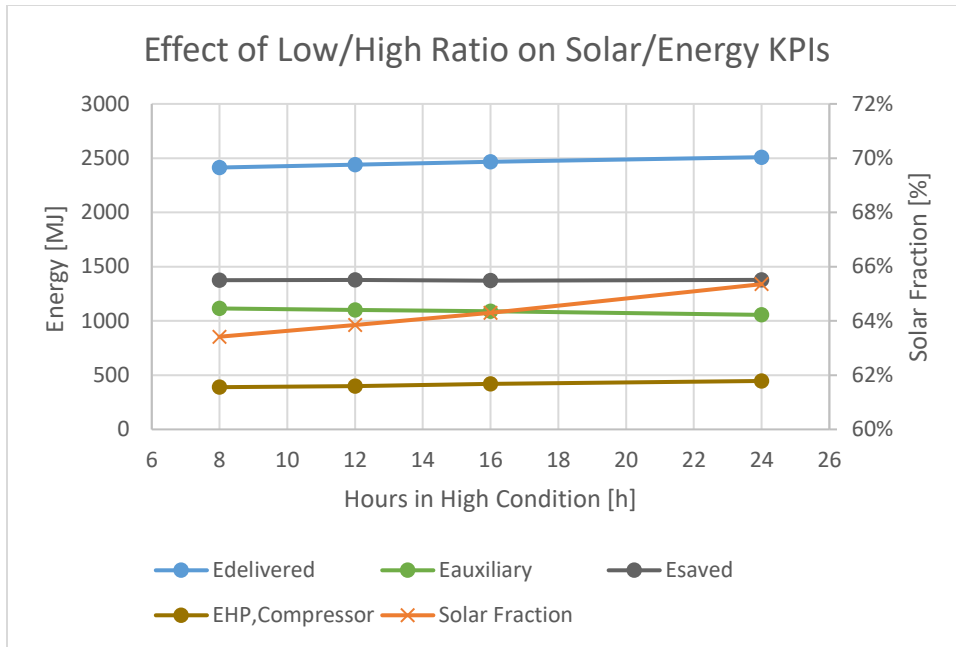


Figure 5.5 - Influence of low/high duration ratio on solar and energy KPIs

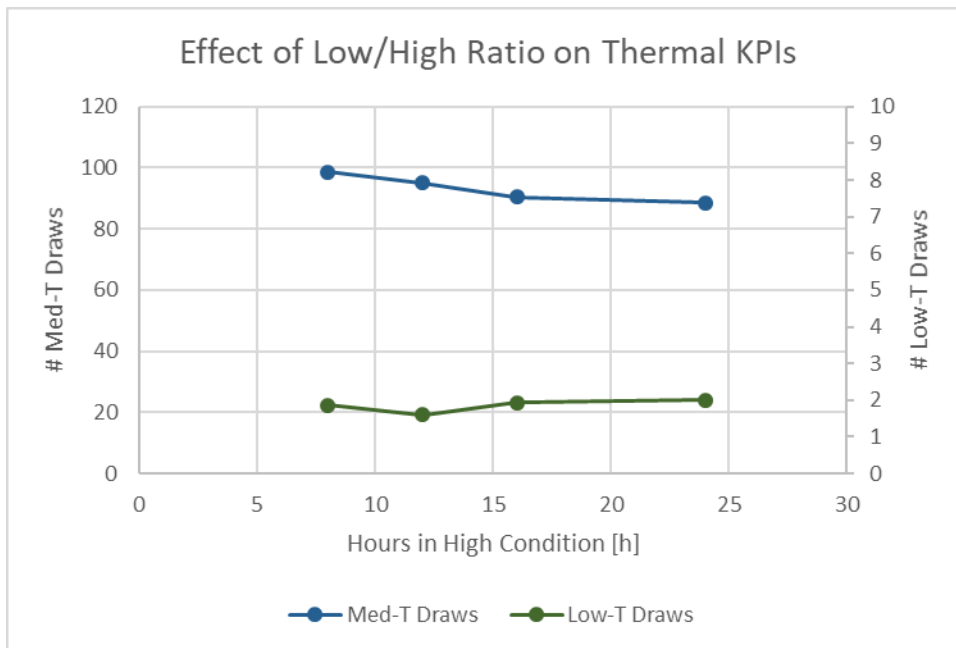


Figure 5.6 - Influence of low/high duration ratio on thermal KPIs

Interestingly, having a large number of hours in the high condition (or, low/high ratios favoring the high T_{Set} condition) seems to positively influence the system. While the total energy savings are virtually unaffected, the total delivered energy increases and the auxiliary usage decreases as

hours in the high condition rise. These are likely related to the increase in heat pump operation than corresponds with the increase in high condition hours. The solar fraction improves notably as well, thanks to the increased heat pump use, and the number of medium-temperature draws is reduced (with little effect on the number of low-temperature draws). It appears, then, that it is more beneficial to make use of the heat pump when it is available than to discourage its use by encouraging Min P or HX-only behaviour.

5.1.3 – Set-Point Temperature Study – Summary

The set-point temperature study was performed to evaluate the changes in system performance that would result from using different set-points, from varying said set-points throughout the day, and by expanding the amplitude of the variation. While the system is clearly very sensitive to the set-point temperature, it is demonstrated that a set-point of 60 C provides a strong balance between solar/energy and thermal performance, and so deviating from this is not recommended. Introducing variability by stepping the set-point temperature down at times of low demand was expected to improve system performance by reducing electricity consumption; however, little improvement was realized by varying the set-point. Large temperature steps caused significant harm to the performance values, indicating that the result of allowing temperatures to slip is that the auxiliary heater must work that much harder to boost the stored water temperature during peak hours. Additionally, system performance is quite clearly at its best when the number of hours in the high T_{Set} condition are maximized. As a result, it is not recommended to proceed with a variable set-point temperature; a constant value of $T_{Set} = 60\text{ C}$ will continue to be applied in the remainder of this thesis.

5.2 – Location Effect Study

Solar domestic hot water system performance is inextricably linked to the climate in which the system is used. The use of the system may be more beneficial in some climates than others; for example, the sub-tropical climate of Hong Kong seems to be well-suited for direct-expansion SAHP systems from the host of DX-SAHP studies performed in that region [22, 26, 28]. Bannister [14] considered the performance of a dual-tank design in 8 cities worldwide, in climates described as “Hot Desert”, “Hot Summer Mediterranean”, “Humid Subtropical”, “Humid Continental” and “Subarctic” based on the Köppen classification system [57], with the intent of representing each major climatic region of the world. This thesis chooses 4 of the same 8 locations for the following location effect study. The selection should provide insight into the usefulness of this system in different regions; hence, 4 of the 5 aforementioned climates are covered, with “Humid Subtropical” neglected as a DX-preferred region due to the limited scope of this thesis. The cities to be used are Ottawa, Canada; Whitehorse, Canada; Rome, Italy and Aswan, Egypt. The use of the same cities as Bannister may allow for cross-comparison to their results.

When evaluating the performance of a system, it is worthwhile to consider the cost of electricity and the effect that the system will have on energy costs and GHG emissions. For all-electric source systems, the cost savings can be solved easily as $\text{Cost/kWh} \times E_{\text{Saved}}$; emissions reductions can likewise be calculated simply based on the local tonnes CO₂-equivalent per kWh produced (tonnes CO_{2e}/kWh) amounts. The cost of electricity for each location is estimated from online sources, with transmission, delivery and other associated fees included (when available) to account for the full cost per unit energy consumed. The sources for electricity are also considered so that the associated GHG emissions can be estimated as described. The evaluation will therefore proceed as follows:

1. The total energy purchased and auxiliary energy used by the systems will be presented.
2. The cost of operation will be determined for the system based on the energy usage results from the simulations.
3. The GHG emissions associated with operation will be determined in the same manner.
4. The cost and GHG emissions savings of the system will be evaluated for electric-source auxiliary heat in comparison to the electric-only DHW system.

Finally, the selection of systems to be included in this study must be made. The electric-only system, as the ultimate basis for system evaluation, is the first design to be included. The focus of this thesis is the 3-mode SAHP; based on the results of Chapter 4, which showed that exclusive use of the Min P or Max Q protocols is detrimental to the performance of the 3-mode system, the “balanced” controller for the 3-mode SAHP is also included in this study. Additionally, the SDHW system provides the basis for evaluating the benefits provided by the heat pump, hence the SDHW design will be included as well. Finally, the results of Chapter 4 indicated that the inclusion of SRS mode provided little benefit for solar or energy performance KPIs, with a small benefit noted in the thermal KPIs. The quality of the performance of the 3-mode system was therefore directly comparable to that of the 2-mode SAHP studied by Wagar, when applied with controls written by the author. Therefore, the 2-mode SAHP as is included as the 4th and final system for the location effect study.

5.2.1 – Humid Continental Climate (Ottawa, Canada)

Annual performance data has already been presented for the systems in question in the location of Ottawa, Canada. The notable KPIs are re-presented in Table 5-3; please refer to Section 4.9 – Annual Performance Evaluation of Compared Systems for an analysis of these performance values.

Table 5-3 - KPIs for solar, energy and thermal performance evaluation in Ottawa

System	Solar Fraction	E _{Saved} [MJ]	E _{Delivered,solar} [MJ]	Med-T Draws	Low-T Draws
Electric	0.0%	0	0	0	0
SDHW	38.2%	4219	5296	270	6
2-mode	58.8%	5048	6526	248	3
3-mode	56.6%	4990	6433	213	3

The SAHP systems performed well in this environment, increasing the total solar collection and reducing the auxiliary energy usage compared to the SDHW design; however, heat pump consumption negated much of the gains, and the net energy savings only increased by 20%. To add to this analysis, energy consumption data is presented in Table 5-4. This allows for the evaluation of the cost and greenhouse gas emissions performance aspects of the systems.

Table 5-4 – Consumption and savings data for systems studied in Ottawa

System	E _{Bought} [MJ]	Operating Cost [CAD]	GHG Emissions [tonnes CO ₂ e]	Cost Savings [CAD]	GHG Reductions [tonnes CO ₂ e]
Electric	12445	\$470.15	0.417		
SDHW	8226	\$310.77	0.275	\$159.39	0.14
2-mode	7397	\$279.45	0.248	\$190.70	0.17
3-mode	7455	\$281.64	0.250	\$188.51	0.17

Electricity costs and associated greenhouse gas emissions are approximated using data shared by Hydro Ottawa and the World Nuclear Association [58, 59]. The results of the simulations show that the SDHW system is reasonably effective in this region, reducing costs and emissions by 34% each. Introducing the heat pump, with or without SRS mode, increases these savings to just over 40% each. In comparison, Bannister [14] found similar SDHW consumption but observed substantially greater energy savings with the use of a dual-tank SAHP in this region, indicating the performance boost that comes from adding a second storage tank.

5.2.2 – Hot Summer Mediterranean Climate (Rome, Italy)

Annual performance data is acquired with the location of the system changed to Rome, Italy. Rome has a similar latitude to Ottawa (42° vs 45° N) but a substantially different climate thanks to geographical effects. The KPIs for the system are shown in Table 5-5.

Table 5-5 - KPIs for solar, energy and thermal performance of systems in Rome

System	Solar Fraction	E _{Saved} [MJ]	E _{Delivered,solar} [MJ]	Med-T Draws	Low-T Draws
Electric	0.0%	0	0	0	0
SDHW	50.1%	4798	5933	152	0
2-mode	68.9%	5315	6764	90	0
3-mode	65.2%	5258	6679	87	1

The solar performance of the system is notably improved compared to the Ottawa-based simulations. The SAHP systems again show large improvements over the SDHW system, but the consumption by the heat pump limits the improvement in E_{Saved} that can be achieved. The 2-mode SAHP performs better than the 3-mode system in solar and energy categories, as observed in Ottawa; a change in the relative performance of the systems is that the 3-mode design no longer out-performs the 2-mode design in the thermal category.

Table 5-6 shows the economical and environmental performance of each system as simulated in Rome, using data acquired online [60]. Here the improved solar performance compared to Ottawa is highlighted again. The SDHW system provides energy and emissions savings of 45%, while the SAHPs boost each to 50%. Once more, the inclusion of a heat pump improves performance, but the gains relative to a SDHW system are not very strong.

Table 5-6 - Consumption and savings data for systems studied in Rome

System	E_{Bought} [MJ]	Operating Cost [CAD]	GHG Emissions [tonnes CO ₂ e]	Cost Savings [CAD]	GHG Reductions [tonnes CO ₂ e]
Electric	10732	\$851.57	1.128		
SDHW	5934	\$470.87	0.623	\$380.70	0.50
2-mode	5417	\$429.85	0.569	\$421.72	0.56
3-mode	5475	\$434.40	0.575	\$417.17	0.55

Bannister [14] observed that the SDHW system consumed 6.4 GJ and the dual-tank SAHP consumed 4.3 GJ; the relative gains were therefore much higher in that study. Here, the SDHW system is much more effective than Bannister’s, while the SAHP systems studied are not quite as effective as the dual-tank design, making the upgrade appear to be unjustified [14].

5.2.3 – Hot Desert Climate (Aswan, Egypt)

The hot desert climate showed very promising results in Bannister’s thesis [14]. High ambient temperatures and solar irradiation levels help solar collectors reach very high collection totals. In a region where SDHW performance is expected to already be very good, it is worth investigating how effective a heat pump can be at boosting the system. The solar collector angle is adjusted to Aswan’s latitude of 24° N; this close to the equator, the collector is expected to see more consistent sunlight, as seasonal changes in solar position are much less extreme than for Rome or Ottawa.

Table 5-7 summarizes the KPIs for the systems simulated in Aswan.

Table 5-7 - KPIs for solar, energy and thermal performance of systems in Aswan

System	Solar Fraction	E_{Saved} [MJ]	$E_{\text{Delivered,solar}}$ [MJ]	Med-T Draws	Low-T Draws
Electric	0.0%	0	0	0	0
SDHW	95.5%	7403	9168	6	1
2-mode	95.4%	7405	9170	4	0
3-mode	95.3%	7398	9169	4	0

This data indicates that the SDHW system can provide virtually all the energy required by the system in this region, with energy savings in excess of 85%. There are no energy or emissions savings associated with switching to a SAHP system here, since the performance is virtually unaffected due to very low heat pump use. In contrast, even the dual-tank SAHP studied by Bannister [14] showed greater electrical consumption than the solar systems simulated here, and the SDHW system used in that study used substantially more electricity. In this study, the available conditions allow the system to run in HX mode for most of the year with high efficiency. In such a climate, there appears to be little need to improve upon the capabilities of the traditional SDHW system.

5.2.4 – Subarctic Climate (Whitehorse, Canada)

The last climate type that this thesis will consider is the subarctic climate of Whitehorse, Canada. This region experiences very long hours of irradiation in the summer, and very low hours in the winter. Temperatures are cooler than in other regions and the intensity of the available sunlight is never particularly high, compared to other regions. At 61° N, Whitehorse is as far north of Ottawa and Rome as Aswan is south of them. The collector angle is changed to match the latitude again for optimal performance – Table 5-8 shows the results.

Table 5-8 – KPIs for solar, energy and thermal performance of systems in Whitehorse

System	Solar Fraction	E _{Saved} [MJ]	E _{Delivered,solar} [MJ]	Med-T Draws	Low-T Draws
Electric	0.0%	0	0	0	0
SDHW	4.8%	665	17	223	253
2-mode	40.0%	3994	5230	252	5
3-mode	38.6%	3880	5152	177	4

This climate showcases the strengths of the heat pump. While the solar and energy performance of the 2-mode system is slightly better than that of the 3-mode SAHP, the 3-mode has substantially

better thermal performance. In a region where the SDHW system itself does not offer enough “free” energy to justify its installation, the relatively high savings from the SAHP design make an incredible difference. Table 5-9 shows the cost and emissions performance to further highlight the significance of this change, using information obtained from Yukon Energy and the Yukon Electric Company [61, 62].

Table 5-9 - Consumption and savings data for systems studied in Whitehorse

System	E _{Bought} [MJ]	Operating Cost [CAD]	GHG Emissions [tonnes CO ₂ e]	Cost Savings [CAD]	GHG Reductions [tonnes CO ₂ e]
Electric	13788	\$464.97	0.100		
SDHW	13124	\$442.56	0.095	\$22.41	0.00
2-mode	9794	\$330.29	0.071	\$134.68	0.03
3-mode	9908	\$334.11	0.072	\$130.85	0.03

The savings associated with SDHW use are negligible for this climate in this study (although Bannister’s [14] SDHW system only consumed 11.7 GJ in Whitehorse, compared to 13.1 GJ here). However, introducing a heat pump delivers annual savings of 28%, with annual consumption comparable to Bannister’s dual-tank SAHP [14]. The savings are not as substantial, by percentage or magnitude, as observed in Ottawa or Rome, but in those regions the SDHW system appeared to be viable in comparison. Only in Whitehorse is the SAHP unquestionably justified in comparison to the SDHW, based on these results. However, electricity comes almost entirely from hydroelectric dams in this region, with very low greenhouse gas impact and relatively low cost, limiting the potential for savings.

5.2.5 – Location Study – Summary

The 2-mode and 3-mode SAHP systems perform well in all climates, providing improvements in solar gain and energy savings compared to the SDHW design. These savings are translated into cost and GHG emissions reductions for all locations. However, the magnitude of the improvements

is small. The simulated SAHPs only increase savings by an additional 5-10% at best when compared to the SDHW system, which is not enough to justify the capital cost of the equipment required. The only climate where the SAHP would be justifiable is the Subarctic climate; however, for the location studied (Whitehorse, Canada), the low cost of energy (and low associated emissions) makes it difficult to justify using the system even there.

The results of this study are compared to results obtained by Bannister [14] for tests in the same locations. Bannister [14] observed very different SDHW performance than the author, with the system consuming much less electricity in most locations for the author (while Bannister's SDHW was more effective than the author's in Whitehorse [14]). Additionally, the dual-tank SAHP used less energy than the single-tank designs in all locations, though this was anticipated [14]. The result is that while Bannister observed a consistent consumption reduction of about 2.3 GJ for all locations [14], the author observed completely inconsistent reductions ranging from 0 to 4 GJ. One reason for the discrepancy between the author's results and Bannister's may be due to the thermostat location in the storage tank. Wagar [13] observed that with the thermostat located close to the solar return port, the auxiliary heater would very often be triggered by the operation of the solar system. The result was excessive auxiliary usage, specifically when HX mode was used. The author chose to use a different location higher up in the tank for the thermostat reading instead, to ensure that auxiliary operation would only occur when required by the system; this difference may explain the increased consumption of Bannister's SDHW system, which had its thermostat only slightly higher than Wagar's [13, 14].

This study concludes that the system is more useful when less direct sunlight is available. Climates with high levels of direct solar irradiation are very well suited to the use of STCs, and SDHW system designs perform well in those locations. In such climates, the relative number of hours

where the use of a heat pump would be better than a heat exchanger is low, and the benefits of using a SAHP design are muted. However, for climates with cloudier skies or less sunlight in general, where there is relatively low direct solar irradiation, the SDHW system is not very effective. In these regions, there are many more times at which the heat pump is the only effective choice, and the SAHP design substantially outperforms SDHW. Justifying the extra capital cost required for the heat pump depends heavily on the local cost of energy and the sources used to obtain electricity; if electricity is inexpensive and from a clean source, the magnitude of the gains made will be low. However, in regions with high electrical costs and/or high-emissions energy sources, it can become relatively easy to justify the installation of a SAHP system.

Chapter 6 - Conclusions and Recommendations

6.1 - Conclusions

This thesis has evaluated the performance of a 3-mode single-tank indirect-expansion solar-assisted heat pump in simulations, with comparisons to traditional hot water system designs and other SAHP designs. Traditional solar domestic hot water systems and methods for storing the collected heat until it is needed are reviewed. The incredibly variable field of SAHP design is introduced, addressing styles such as direct-expansion systems, photovoltaic-thermal integrated designs and multi-purpose or multi-source systems. Out of this vast array of design options, the author pursues an indirect-expansion system with multiple operating modes. Expanding on a previously studied design, which alternated between HX-only and HP-only operating modes [13], this thesis considered the same system with the option to route HX outlet flows to the HP inlets, thereby operating them in series (SRS mode).

6.1.1 – Regarding the Comparison of System Designs

Six systems are designed and modelled using TRNSYS 17 simulation software for behaviour and performance comparison purposes. The systems studied are as follows:

- Electric Hot Water
- Solar Domestic Hot Water
- HP-Only SAHP
- SRS-Only SAHP
- 2-mode SAHP
- 3-mode SAHP

The electric system is the baseline for evaluating system performance. The strong stratification which occurs in the electric-only tank is used to provide initial conditions for the tank model in all simulations. A SDHW system is included for justifying the inclusion of the heat pump. The design features a two-loop or dual-side solar collection system, with a solar/source side loop that collects heat from the sun and a load side loop that delivers this heat to the storage tank. The two loops, which are a feature of all solar designs in this thesis, are connected by heat transfer devices (in this case a heat exchanger). The HP-only and 2-mode SAHP systems are included as reference points, since these configurations have been studied by others under different system settings. Both systems are single-tank configurations, as are all systems in this thesis; a small buffer tank is added to the solar side loop for thermal mass instead of using a full-size storage tank. The 2-mode system introduces the first instance of complex decision-making by choosing between HX or HP operating modes. The SRS-only SAHP system is included so that the performance capabilities of that operating mode can be prominently displayed. This system connects the HX outlets to the HP inlets with the intent of enhancing performance overall. Finally, the 3-mode SAHP system combines elements of all the above. Physically, it is constructed like the SRS-only system, but with the ability to operate using only the HX or only the HP as well. The control of this system is a core component of the thesis, aiming to identify and use the “best” operating mode under all circumstances.

TRNSYS 17 [50] is used to simulate the systems as interconnected mathematical models. The TRNSYS model constructed contains elements of those used by Wagar [13] and Bannister [14] in their respective theses. Some models, especially the solar thermal collector and circulation pumps, are the same as used previously, while other significant models are changed. Notable changes include modifications to the heat pump model, a different and more detailed tank model, and a

brand-new controller design. The heat pump model developed by Wagar [13] is used, but it is modified so that it no longer performs its own operational safety checks. Responsibility for these checks is shifted to the controller model. The tank model used (Type 534 – No HX) is selected for improved mathematical stability and improved destratification modelling over the Type 4f model used in previous work [54]. Finally, the new controller is implemented to maximize the utility of the system.

The controller is designed as complex regime of threshold-checking and performance predictions, with the intent being to use the “best” mode at all times. The definition of the “best” mode depends on current conditions such as tank temperatures and targeted recharging rates, and the evaluation is performed by predicting the solar gains, power consumption and heat transfer that would be achieved for each mode in a timestep. The current weather and temperature conditions are used as inputs to the controller to achieve this. The result of the decision-making process is a dictation of the operating mode that the system will use for the next timestep. The modes available are “Off”, “HX-only”, “HP-only”, “SRS-only” and “Warmup”, a mode used to build up solar energy in the solar side loop.

Following the calculations for predicting system performance, the controller checks to see if it should shut down automatically or if it is desired to use the solar loop. The system is prevented from operating overnight – without this control, ambient conditions may allow the system to run inefficiently to collect air-source energy. The heat pump, if operational, may be forced to shut down to allow the solar loop to heat up so that the controller can more accurately evaluate HX mode. The controller also checks for near-freezing temperatures to ensure system safety.

The energy demand is evaluated as low or high, based on the set-point temperature. Based on the result, one of two mode selection protocols is followed. The Minimum Power protocol will select

whichever mode that is effective has the lowest power consumption; the Maximum Rate of Heat Gain protocol will select a mode that delivers the most heat, with a preference for modes with high COP. The controller will use these protocols to decide which mode is desired for the next timestep; if required, it includes heat pump safety checks in the decision. The safety checks include an enforced minimum “off” cycle time, a check of the predicted HP run time if activated and a check for safe operating temperatures. If temperatures are not safe as-is, the controller checks to see if SRS or HX operation is viable as an alternative before confirming its identification of the “best” mode.

If the controller is attempting to change from the current operating mode, it evaluates whether it should maintain the current operating mode instead. These conditions are primarily focused on avoiding equipment short-cycling, whenever it is safe to do so. The controller next determines whether auxiliary backup heat is required. This process starts with a daytime auxiliary embargo, which is overridden in poor solar conditions. The controller then checks tank temperatures and the anticipated rate of heat delivery from the solar system against the set-point demands, with a dead-band as applicable, before deciding if the auxiliary heater should be used. Finally, the controller manages timers that are used to assist in decision-making at the end of each timestep.

Four versions of this controller are used in the thesis. The version described above is applied to the 3-mode system and is described as the “balanced” controller. One version is modified so that it only uses the Min P protocol; another only uses Max Q. Finally, one version is created without the SRS mode for use with the 2-mode SAHP system. The single-mode systems only need to decide whether they should be on or off, and so do not require such involved decision making.

The behaviour of the 8 systems (6 systems, plus 2 alternative control schemes for the 3-mode SAHP) is studied on representative summer and winter days. The study shows the top and bottom

tank temperatures, the rate of solar energy collection, the mode selection and the current energy balance within the storage tank in charts for comparison purposes. Following this, annual simulations are run to evaluate and compare the performance of the systems. The systems are evaluated using three groups of key performance indicators: solar KPIs (concerned with the solar collection abilities of the system), energy KPIs (focused on actual energy savings achieved) and thermal KPIs (tracking the temperature of each water draw event). Essential KPIs include solar fraction (which represents the reduced dependence on auxiliary heat), energy savings (compared to an all-electric system) and the number of times that water draw temperatures fall below the certain thresholds (55 C and 50 C in this thesis).

The electric system is observed to reach a pseudo-steady state, where every day is essentially identical to the last, thanks to its limited dependence on weather. The SDHW system's behaviour varied significantly depending on the weather, as did that for all SAHP systems. Another shared characteristic of the solar systems was a substantial loss in stratification due to the flow patterns established in the tank and lukewarm solar return temperatures.

The SDHW system operated well in the summer but poorly in the winter, as expected. The other single-mode options all performed poorly in one of the two seasons – summer for the HP-only, since the STC outlet temperatures were too high, and winter for the other two due to poor HX performance. The inclusion of a heat pump causes a noticeable improvement in collector efficiency, but a single-mode system is demonstrably incapable of supporting the load year-round.

The 2-mode system behaves well in both seasons. Similarly to SRS-only, it operates with the heat pump in the morning and the HX all afternoon for summer days, while in the winter it uses the heat pump exclusively. This system provides the first instance of the heat pump long-cycle pause being enforced, which offers a comparison to the HP-only system (which did not use the long-

cycle pause). It is observed that the 2-mode system saves more energy than the HP-only on this day, despite having the same operating conditions – this is likely due to the periodic warmup cycles, which allowed the HP to run with higher COP after each pause.

Finally, the 3-mode systems all exhibited a more conservative approach to mode selection than the 2-mode design. The Min P-Only design was ineffective, as it preferred HX operation even in the winter (thereby negating the purpose of having a heat pump). The Max Q system behaved well, and was slightly more aggressive than the balanced version, but the differences were small. All systems behaved similarly to the 2-mode system, indicating that the inclusion of SRS mode had a small impact on behaviour; in fact, for the balanced 3-mode controller, only 1% of all timesteps (3% of all operational timesteps) were performed in SRS mode, or 86 hours/year.

In an annual simulation comparison of the systems, it is observed that SAHP designs provide high solar fraction, increased E_{Saved} and consistent thermal performance. SAHP systems showed 50% better SF and 17% greater E_{Solar} than the SDHW design. For every 1.2 MJ of energy delivered to the tank by a SAHP design, 1 MJ of auxiliary demand was displaced – if the solar system operates with a total COP > 1.2, then, the system will provide energy savings over a non-solar system. Stratification of solar systems is poor, leading to greater tank losses, but this is only a small cost compared to the gains achieved. E_{Saved} is, in general, muted compared to $E_{\text{Delivered}}$, because a substantial portion of the delivered energy is traceable to the heat pump compressor. An interesting observation of the 3-mode, 2-mode and Max Q systems shows that despite a +/- 6% variance in heat pump use, the achieved energy savings of the three systems were nearly identical, indicating that there may be a ceiling for system performance. Overall, the SAHP designs exhibited 16% greater energy savings than SDHW, reducing the overall electricity load by 40%. Finally, the study of thermal performance shows that all systems provide water above 50 C with excellent

consistency, and the multi-mode systems do a better job delivering water above 55 C. The 3-mode balanced controller SAHP performed best of all, with 15% fewer “medium-temperature” draws than any other system.

The conclusions of this behaviour and performance comparison are that compared to a SDHW system, the gains for including a heat pump are relatively small; the inclusion of a SRS mode is not substantially beneficial with the control schemes used here; and the control scheme in general should be designed to be aggressive, but well-balanced, in its decision-making.

6.1.2 – Regarding the Parametric Studies of 3-Mode System Performance

Finally, two parametric studies are conducted on the 3-mode system. The first study examines the effect of different system set-point temperatures, and the effect of allowing the set-point to vary throughout the day, on the performance of the balanced-control system. The amount of variance and the amount of time spent in the high or low set-point condition is varied in the study for a full evaluation of the effect. Changing the high value for T_{Set} showed a strong effect on the system. Solar and energy performance was slightly better at lower T_{Set} values, but thermal performance was exponentially worse; the reverse was true for high T_{Set} values. As a result, 60 C is determined to be the best high T_{Set} value as a compromise between solar/energy and thermal performance categories. Another facet of the study considered the size of the step between low T_{Set} and high T_{Set} , finding that there was little effect on performance at all until the step exceeded 5 C. At larger step sizes, performance in all categories dropped significantly, suggesting that the low set-point may have been allowing the system to behave too passively under these conditions, leading to excessive auxiliary use. Finally, the ratio of time spent in the low and high conditions is varied, showing that system performance is at its best, in general, when there is no time spent in the ‘low’

condition, leading to the conclusion that the practice used thus far – having a fixed 60 C set-point – is the best set-point profile for this system.

The second study examines the effect of climate, considering system performance in 4 locations across the globe, covering Humid Continental, Hot Summer Mediterranean, Hot Desert and Subarctic climate types. In addition to the 3-mode balanced system, the electric and SDHW systems are included for baseline comparisons, as well as the 2-mode system for further evaluation of its performance in relation to the 3-mode design. Comparisons are drawn throughout the study to results obtained by Bannister [14], who performed simulations of SDHW and dual-tank 2-mode SAHP systems in the same locations but with different system parameters. In general, the single-tank SAHP systems do not perform as well as Bannister's [14] dual-tank design; it is also observed that the SDHW systems in either study perform very differently. In Ottawa, the SAHP improved savings from 34% to 40% over the SDHW design. In Rome, a city with similar latitude but a different climate, savings were raised from 45% to 50%. Aswan, Egypt, with a desert climate, had such good SDHW performance (85% reduction in electricity use) that the inclusion of a heat pump had no effect; the SDHW studied here performed much better than Bannister's [14] SDHW simulation, and used less electricity than even the dual-tank system. The SDHW studied here also out-performed Bannister's in Ottawa and Rome. However, in Whitehorse, Canada, the reverse was observed – the SDHW had virtually no impact on performance, whereas Bannister's [14] achieved measurable savings. The SAHP systems achieved 28% energy savings in this location, definitively out-performing the SDHW design.

It is difficult to say why Bannister's [14] SDHW system behaved so differently. Parameter choices, such as panel tilt, number of panels and system flow rate, may be the primary cause of different behaviour. It should be noted, however, that the author used a different tank model than Bannister

[14], and placed the thermostat higher in the tank, away from the hot solar return. Wagar [13] observed substantial negative effects of having the thermostat located near the return port; in Bannister's [14] thesis, the thermostat is raised, but only slightly, perhaps contributing to the different performance results. In any case, the conclusion of the location study is that the SAHP system is more suited to locations that experience less direct sunlight than average, since the heat pump gets more opportunities to improve performance. The inclusion of SRS mode continues to have little to no effect, and the use of a SAHP in general seems difficult to justify unless electricity is very expensive or associated with high GHG emissions.

6.2 - Recommendations

The field of SAHP design is incredibly broad and diverse, with enormous potential for further development in simulated and experimental systems. Narrowing the scope of the field to only include 3-mode collection designs still leaves a wide range of topics for study, especially given the new controller design presented in this thesis.

6.2.1 – Experimental Validation of TRNSYS Simulation

The first and most pressing concern is that of experimental validation of the system. Many of the components used in this thesis have been experimentally validated by Wagar [13] and Bannister [14]; however, models such as the Type 534 – No HX and scaled-down heat pump have not been experimentally refined. Validation tests should focus on pump power curves, HX coefficients, tank stratification and heat loss properties, and heat pump characterization.

Worth noting is that the heat pump model used has been fully validated by Wagar [13] for a full-scale heat pump (1.0) under a given range of inlet conditions ($T_{\text{Source,In}} = [10, 30]$ and $T_{\text{Load,In}} = [15, 50]$) [13]. Outside of these temperature ranges, performance of the heat pump is unknown; if any

input exceeds the limits, the model calculates energy flow rates using the limit instead of the actual value [13]. Additionally, the scaling feature assumes that performance (heat transfer rates and power consumption rates) scale linearly, and cannot be fully validated using the single-speed heat pump that was used to produce the model. The need for heat pump validation, then, does not apply to the Type 161 model with $\text{ScaleHP} = 1$. However, it is recommended that a new heat pump be installed and modelled for use in the system. Wagar [13] and Bannister [14] acknowledged that the heat pump used for Type 161 was excessively powerful and was substantially non-ideal for the system. A new heat pump has been ordered; upon arrival, it should be installed and tested for modelling purposes, to replace the Type 161 in simulations. The heat pump was ordered to the following specifications:

- Capacity is 2.3 kW.
- Design is for water-to-water heat transfer at a flow rate of 5 kg/min (or 5 L/min) on the load and source side.
- Maximum load-side inlet temperature should be at least 55 C.
- Maximum source-side inlet temperature should be at least 30 C.
- Capacity of the HP is to be variable if possible through the use of a variable-speed compressor.

Validation should also include the controller design. The current design of the controller uses numerous mathematical models of system components internally, in conjunction with inputs from several sources, to predict performance. The viability of using current irradiation levels and temperatures at locations including the ambient air, the solar collector outlet, the heat transfer device inlets and outlets and the storage tank ports as controller inputs should be examined by validation studies. The use of temperature measurements as inputs should be especially closely

examined, as Wagar [13] cautioned that wherever unreliable temperature measurements are used to make control decisions, there will be precipitative effects that completely change system behaviour for simulated systems.

The buffer tank is included as a means to increase the thermal mass of the solar side loop; Wagar [13] found that heat pump operation was not sustainable without it. A smaller, less-powerful heat pump may be able to function without this thermal mass. The necessity of this component should be investigated in the interest of reducing the physical size and capital cost of the system, making it easier to justify the upgrade.

The study of variable set-point temperatures found that there is little encouragement to deviate from a standard 60 C set-point. Experimental studies should seek to validate this result before abandoning the concept.

6.2.2 – Further Development of 3-Mode SRS SAHP

As the model is being fully validated, it is also worth considering the use of SRS mode. Despite its predicted ability to enhance performance, the simulated system rarely selected it, using it for only 3% of operational timesteps (compared to 24% HX mode and 58% HP mode). While this indicates that SRS mode may not, in fact, be useful, it is worthwhile to study the mode further. It is recommended to consider a control scheme which prioritizes SRS mode at all times. Such a scheme may start by checking SRS mode for its heat delivery capabilities – it may be desirable to select it whenever its rate of heat delivery is the highest of the three modes. This may result in more frequent use of the mode, which will help to definitively evaluate its potential.

Other controller modifications or additions should include a protocol to follow when stagnation conditions are present. The system may be able to use the heat exchanger, or a reversible heat

pump, to protect the solar collection fluid from stagnation damage, which occurs during long periods of inactivity while the system is exposed to sunlight. While this is not directly related to system performance, its inclusion may help justify upgrading to a SAHP design. It may also be worthwhile to upgrade the model to have different operating conditions for different modes – irradiation thresholds, operating temperature limits and system flowrates may all be different for HX, HP and SRS modes. Such a system would be more complex, but it may also perform much better, as each mode could then operate under its own unique ideal conditions.

Expanding upon this, further work should include parametric studies to find the best operating points for controller parameters such as cycle times, timers, irradiation thresholds, temperature thresholds and flowrates. A single broad-scope experiment simulation could readily identify conditions which maximize the system's operating potential and establish the influence of certain parameters and control features on system performance. This in turn can help simplify and strengthen the controller. Other parametric studies should examine the collector array to be used – additional collectors, connected in parallel and/or in series, will inevitably increase the solar collection totals. There may be a configuration considered optimal for a residential system, one which may clarify the case for or against upgrading to a SAHP design.

It is also recommended to invest some time in developing a realistic water draw profile. The current work uses an arbitrary profile from CSA, modified to work within the simulation timestep used, which does not take into consideration various effects such as weekend or seasonal changes in usage patterns. Methods are available for including such considerations in the development of a draw profile [63]; using a more realistic pattern, one which may include “vacation weeks” without water draws, weekend days with large mid-day draws, and seasonal changes such as increased hot water use during winter, should provide improved estimates of the benefits of these systems.

The TRNSYS model introduced a component not used by Wagar or Bannister, the Type 534 – No HX storage tank. This model includes an input which dictates the rate at which temperature-inverted nodes should mix (most models perform instantaneous mixing, a practice which was also applied in this thesis). The opportunity to improve the realism of the tank model should not be neglected, as this component in particular is a source of uncertainty in most SAHP simulations. Research or experimental validation should be done to establish a realistic value or model for this input.

Further work is recommended for studying deviations from a fixed 60 C set-point temperature. The primary concern regarding low-temperature draws, which increase in frequency as T_{Set} is lowered, is the indication that conditions exist for legionella to thrive; a low-temperature control plan may be considered, then, if the control includes periodic “purging” cycles which sustain very high temperatures for brief periods to ensure the tank remains clean. Further studies of the variable/lowered set-point concept should also consider expanding the timeframe of the study to cover year-long time periods.

The performance of the system in various locations should continue to be considered. A system that works well in Ottawa may not be suitable in Aswan, and parameters for operating the system in Whitehorse will differ from those that should be used in Rome. Other climate types exist, of course, with more than 30 unique classifications available in the Koppen system [57]. Further studies should ensure that enough climates are considered for results to apply to a majority of population centers, and multiple locations should be considered per climate type to ensure accurate estimations of performance are obtained.

6.2.3 – Expanding the Field of Study

It may also be worthwhile to introduce pump speed variations to the controls and to validate them using variable-speed pumps. At the moment, all modes operate with a single pump speed that is equal on the source and load loops. Upgraded controls could not only allow for different speeds in different modes, but they could also continuously vary pump speed according to the current conditions, and could allow unequal flow to occur to improve performance. Examples may include using low flowrates for HX mode, using high flow for low temperature change on the solar side and low flow on the load side for high lift in HP mode, or modifying flowrates as water draws or irradiation changes influence the system's performance.

The new heat pump will improve the operational abilities of the experimental test unit and should provide a more accurate or appropriate model for the heat pump in TRNSYS. It will also introduce new operational abilities to the system which should be accounted for in a modified controller design. Updated controls should include a method to change compressor speed using a variable frequency drive (VFD) to change the capacity of the heat pump in a predictable manner. This behaviour will need to be validated and then modelled in TRNSYS.

Configurations besides the 3-mode SRS configuration are worthy of consideration. Wagar's [13] 2-mode system performed very well in simulations when applied with the author's control scheme; as a simpler alternative to 3-mode designs, it is not worth forgetting as an option. Another 3-mode design worth considering would use HX, HP and Parallel (PLL) modes. PLL mode would see the HX and HP operated in parallel, with STC flow split between the two devices. The inclusion of this mode would open the possibility of a 4-mode system with both SRS and PLL operating options.

Other ways to expand the use of the system could include replacing the STC with a PVT collector. Such a collector could further enhance the energy savings achieved, but at additional capital cost. Adding alternative heat sources for the heat pump, such as an air-source heat exchanger, could also improve performance. Integrating the heat pump to space heating and cooling applications would allow it to further increase residential energy savings, improving its justifiability.

Organic or adaptive control schemes could play a substantial role in the design of futuristic systems [49]. When the user of a system establishes a pattern of behaviour (such as taking showers every morning at 7 AM, laundry every Saturday morning, annual vacations in late December etc.), an organic control scheme can learn and remember the patterns to adapt itself to the user's needs. If the user rarely demands hot water from 7 AM to 3 PM, except on weekends, the controller could use that pattern to adjust t_{target} (thereby adjusting Q_{Load}) so that the demand is low in the morning and high in the afternoon. Such behaviour could allow the system to make control decisions that accurately reflect the user's needs.

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Appendix – Type 1630 – 3-Mode SAHP Controller

The following is the entire FORTRAN code used for the Balanced 3-Mode SAHP Controller. The phases, stages and steps are identified in the manner applied in Chapter 3 where applicable.

Subroutine Type1630

! Object: ETU Controller

! Simulation Studio Model: Type1630

!Variable notation format: Variable Unit [lower limit, upper limit]

! ***

! *** Model Parameters

! ***

!	UAhx	kJ/hr.K [-Inf;+Inf]
!	flow_sys	kg/hr [-Inf;+Inf]
!	Cp	kJ/kg.K [-Inf;+Inf]
!	mass_load	kg [-Inf;+Inf]
!	mass_solar	kg [-Inf;+Inf]
!	t_hpmin	hr [-Inf;+Inf]
!	t_hxmin	hr [-Inf;+Inf]
!	Thplinmax	C [-Inf;+Inf]
!	Thpsinmax	C [-Inf;+Inf]
!	Thpsoutmin	C [-Inf;+Inf]
!	ScaleHP	any [-Inf;+Inf]
!	Deadbands	C [-Inf;+Inf]
!		
!	Timer	hr [-Inf;+Inf]
!	Collector Area	m ² [-Inf;+Inf]
!	Col. Test Rate	kg/hr.m ² [-Inf;+Inf]
!	Eff Intercept	- [-Inf;+Inf]
!	Eff Slope	kJ/hr.m ² .K [-Inf;+Inf]
!	Eff Curve	kJ/hr.m ² .K ² [-Inf;+Inf]
!	Pump Rated Flow Rate	kg/hr [-Inf;+Inf]
!	Pump1 Rated Power	kJ/hr [-Inf;+Inf]
!	Pump1 Power Coefficient 1	- [-Inf;+Inf]

```

!           Pump1 Power Coefficient 2  - [-Inf;+Inf]
!           Pump2 Rated Power  kJ/hr  [-Inf;+Inf]
!           Pump2 Power Coefficient 1  - [-Inf;+Inf]
!           Pump2 Power Coefficient 2  - [-Inf;+Inf]
!           AuxCapacity          kJ/hr [-Inf;+Inf]

```

```
! ***
```

```
! *** Model Inputs
```

```
! ***
```

```

!           auxset          C [-Inf;+Inf]
!           Ttank1         C [-Inf;+Inf]
!           Ttank2         C [-Inf;+Inf]
!           Ttank3         C [-Inf;+Inf]
!           GTt            kJ/hr.m^2 [-Inf;+Inf]
!           Tstcin         C [-Inf;+Inf]
!           Tstcout        C [-Inf;+Inf]
!           Thxsin         C [-Inf;+Inf]
!           Thxsout        C [-Inf;+Inf]
!           Thxlin         C [-Inf;+Inf]
!           Thxlout        C [-Inf;+Inf]
!           Thpsin         C [-Inf;+Inf]
!           Thpsout        C [-Inf;+Inf]
!           Thplin         C [-Inf;+Inf]
!           Thplout        C [-Inf;+Inf]
!           t_target       hr [-Inf;+Inf]
!           Ttarget        C [-Inf;+Inf]
!           GH,d           kJ/hr.m^2 [-Inf;+Inf]
!           GH,t           kJ/hr.m^2 [-Inf;+Inf]
!           theta          degrees [-Inf;+Inf]
!           Tamb           C [-Inf;+Inf]
!           Ground Reflectance  - [-Inf;+Inf]
!           Slope          degrees [-Inf;+Inf]
!           Ttank_therm    C [-Inf;+Inf]

```

```
! ***
```

```
! *** Model Outputs
```

```
! ***
```

```

!           Stagnation     any [-Inf;+Inf]
!           Mode           any [-Inf;+Inf]
!           auxCS          any [-Inf;+Inf]

```

!	CSp1	any [-Inf;+Inf]
!	CSp2	any [-Inf;+Inf]
!	CShp	any [-Inf;+Inf]
!	CSv1	any [-Inf;+Inf]
!	CSv2	any [-Inf;+Inf]
!	CSv3	any [-Inf;+Inf]
!	CSv4	any [-Inf;+Inf]
!	Qstc	kJ/hr [-Inf;+Inf]
!	Qhx	kJ/hr [-Inf;+Inf]
!	Qhps	kJ/hr [-Inf;+Inf]
!	Qhpl	kJ/hr [-Inf;+Inf]
!	Qsrs	kJ/hr [-Inf;+Inf]
!	Qmax	any [-Inf;+Inf]
!	Powerhp	kJ/hr [-Inf;+Inf]
!	COPhx	- [-Inf;+Inf]
!	COPhp	- [-Inf;+Inf]
!	COPsrs	- [-Inf;+Inf]
!	COPbest	any [-Inf;+Inf]
!	t_hp	hr [-Inf;+Inf]
!	t_hps	hr [-Inf;+Inf]
!	HP_to	any [-Inf;+Inf]
!	current_mode	any [-Inf;+Inf]
!	Ttanktopavg	C [-Inf;+Inf]
!	Thxsoutsrs	C [-Inf;+Inf]
!	Thxloutsrs	C [-Inf;+Inf]
!	t_hpstate	hr [-Inf;+Inf]
!	t_hxstate	hr [-Inf;+Inf]
!	HP_off	hr [-Inf;+Inf]
!	7 + 9 tracers	any [-Inf;+Inf]
!	PowerHP_srs	kJ/hr [-Inf;+Inf]
!	Qhps_srs	kJ/hr [-Inf;+Inf]
!	Qhpl_srs	kJ/hr [-Inf;+Inf]
!	AuxPower	kJ/hr [-Inf;+Inf]

!-----

Use TrnsysConstants

Use TrnsysFunctions

!-----

!DEC\$Attributes DLLexport :: Type1630

!-----

!Trnsys Declarations

Implicit None

Integer CurrentUnit,CurrentType

!The following are all Double Precision variables

Double Precision Timestep,Time,rdconv

!rdconv for trigonometry used in STC calcs

a,b,c,d,f,g,h,i,j,k,as,bs,cs,ds,fs,gs,hs,is,al,bl,cl,dl,fl,gl,hl,il,x,y

!coefficients for HP calcs

TauAlf,effsky,effgnd,cosslope,fsky,fgnd,gdsky,gdgnd,xkat,xkatb,xkatds,xkatdg,frulp,fpul,ftest,
rtest,ratio

!STC variables

Ppump1,Ppump2,Qhplsrs,Powerhpsrs,Qhpsrs,Qstc_hp,Qmode,Qload

!internal energy flow rate values

PumpCS,eta,Tsourceout,Tsourceoutsrs,Stored(4),prev_auxCS

!misc (Pump CS, HX efficiency, solar temperature management, timer storage)

Stagnation,mode,auxCS,CSp1,CSp2,CShp,CSv1,CSv2,CSv3,CSv4,Qstc,Qhx,Qhps,Qhpl,Qsrs,

Qmax,Powerhp,COPhx,COPhp,COPsrs,COPbest,t_hp,t_hps,t_hpstate,t_hxstate,Ttanktopavg,

Thxsoutsrs,Thxloutsrs

!model outputs, which must be declared if they are not also used as inputs or parameters

HP_to,HP_off,current_mode,AuxPower

!Heat Pump Status/timeout data, system data

PHX,PHP,PSRS,PMIN

!Power-consumption predictor variables

! PARAMETERS

DOUBLE PRECISION UA _{hx}	!From the HX
DOUBLE PRECISION flow _{sys}	!The desired mass flow rate during operation
DOUBLE PRECISION C _p	!specific heat
DOUBLE PRECISION mass _{load}	!The mass of the water that is expected to be drawn (based on 45 kg max draw)
DOUBLE PRECISION mass _{solar}	!The mass of the water in the solar loop (depends mostly on the buffer tank)
DOUBLE PRECISION t _{hpmin}	!Minimum hp run time = minimum hp off time
DOUBLE PRECISION t _{hxmin}	!Minimum hx state time - to limit flow control valve actuation
DOUBLE PRECISION Th _{plinmax}	!Maximum load-side (condenser) inlet temperature
DOUBLE PRECISION Th _{psinmax}	!Maximum solar-side (evaporator) inlet temperature
DOUBLE PRECISION Th _{psoutmin}	!Minimum solar-side (evaporator) outlet temperature
DOUBLE PRECISION Scale _{HP}	!Set to 0.6 based on CB
DOUBLE PRECISION Deadbands	!The temperature dead-band that the controller should use for signal management
DOUBLE PRECISION Timer	!Defines the frequency at which the controller should actually run
DOUBLE PRECISION area	!Collector_Area, STC Area
DOUBLE PRECISION g _{test}	!Tested flow-rate for the solar collector
DOUBLE PRECISION eta _{int}	!Eff_Intercept, Solar collector efficiency parameters
DOUBLE PRECISION eta _{slope}	!Eff_Slope
DOUBLE PRECISION eta _{curve}	!Eff_Curve
DOUBLE PRECISION rated _{pump_flow}	!Pump_Rated_Flow_Rate, the rated flow rate of the pumps
DOUBLE PRECISION P _{pump1_rated}	!Pump1_Rated_Power, Rated power of the pump
DOUBLE PRECISION P _{pump1_C1}	!Pump1_Power_Coefficient_1, Coeffs 1 and 2
DOUBLE PRECISION P _{pump1_C2}	!Pump1_Power_Coefficient_2
DOUBLE PRECISION P _{pump2_rated}	!Pump2_Rated_Power
DOUBLE PRECISION P _{pump2_C1}	!Pump2_Power_Coefficient_1, Pump1 is solar-side, Pump2 is load-side. Parameter values are from WW.
DOUBLE PRECISION P _{pump2_C2}	!Pump2_Power_Coefficient_2
DOUBLE PRECISION AuxCapacity	!Power capacity of the auxiliary heating element
DOUBLE PRECISION tol	!Tolerance amount to be used
DOUBLE PRECISION b ₀	!Effectiveness parameter for STC
DOUBLE PRECISION half _{night}	!Defines the nighttime shut-off period
DOUBLE PRECISION T _{max}	!Defines the aux setpoint and the system shut-off point
DOUBLE PRECISION Q _{stc_min}	!Defines the minimum gain required to justify operation
DOUBLE PRECISION aux _{off_time}	!Defines the daytime aux shut-down. Enter the number of morning hours for which the aux should be available.

DOUBLE PRECISION Qaux_on !Defines the minimum gain at which the shut-down
is upheld. Lower Qstc will allow the aux to activate if needed.
DOUBLE PRECISION t_hpduration !Defines the maximum continuous runtime of the
heat pump

! INPUTS

DOUBLE PRECISION auxset !The setpoint temperature input to the controller
DOUBLE PRECISION Ttank1 !Top tank temperature
DOUBLE PRECISION Ttank2 !Temperature at the center of m_load (node 3) – calculations
show that m_load occupies the top 4 nodes
DOUBLE PRECISION Ttank3 !Temperature at the bottom of m_load (node 4)
DOUBLE PRECISION GTt !Total tilted radiation from weather file
DOUBLE PRECISION Tstcin !Measured inlet temperature to the STC
DOUBLE PRECISION Tstcout !Measured STC Outlet Temp
DOUBLE PRECISION Thxsin
DOUBLE PRECISION Thxsout
DOUBLE PRECISION Thxlin
DOUBLE PRECISION Thxlout
DOUBLE PRECISION Thpsin
DOUBLE PRECISION Thpsout
DOUBLE PRECISION Thplin
DOUBLE PRECISION Thplout
DOUBLE PRECISION t_target !The calculated time until the set-point temperature reaches
the target temperature
DOUBLE PRECISION Ttarget !Target temperature for the controller (usually auxset+5)
DOUBLE PRECISION gd !Horizontal diffuse irradiation (weather)
DOUBLE PRECISION gh !Horizontal total irradiation (weather)
DOUBLE PRECISION theta !Incident angle (weather)
DOUBLE PRECISION Tamb !Ambient temperature
DOUBLE PRECISION rho !Ground_Reflectance
DOUBLE PRECISION slope !Collector slope
DOUBLE PRECISION Ttank_therm !Temperature of the tank at the thermostat (the specific
node may vary but the height is approximately 37" from the bottom)

!A function from Type 1 for tau-alpha is applied here:

$$\text{TauAlf}(\theta) = 1.0 - b_0 * (1.0 / \text{DMAX1}(0.5, \text{DCOS}(\theta * \text{rdconv})) - 1.0) - (1.0 - b_0) * (\text{DMAX1}(60.0, \theta) - 60.0) / 30.0$$

!-----


```
!-----  
!Get the Global Trnsys Simulation Variables  
  Time=getSimulationTime()  
  Timestep=getSimulationTimeStep()  
  CurrentUnit = getCurrentUnit()  
  CurrentType = getCurrentType()
```

```
!-----  
!Set the Version Number for This Type  
  If(getIsVersionSigningTime()) Then  
    Call SetTypeVersion(17)  
    Return  
  EndIf
```

```
!-----  
!Do Any Last Call Manipulations Here  
  If(getIsLastCallofSimulation()) Then  
    Return  
  EndIf
```

```
!-----  
!Perform Any "After Convergence" Manipulations That May Be Required at the End of Each  
Timestep
```

!Here, the controller resets timers that record how long the system state has been upheld

!Notes:

!Increment t_hpstate, t_hxstate and HP_off at the very end. The reset parameters will be 0 after this increment.

!Because comparisons of t_hpstate etc. are used to t_hpmin etc. later on, they must be calculated and managed in seconds.

!Current_mode is updated in this sequence. This is useful for tracking last-step-status (critical for conditionals and the end-of-timestep work)

!STAGE 10 – END-OF-TIMESTEP TIMER MANAGEMENT

```
  If(getIsEndOfTimestep()) Then  
  
    Stored(4)=Stored(4)+1    !Increment nsteps
```

```

If (mode==0) Then
  If (current_mode==2) Then
    t_hpstate=-Timestep
    HP_off=-Timestep
  Elseif (current_mode==3) Then
    t_hpstate=-Timestep
    t_hxstate=-Timestep
    HP_off=-Timestep
  Endif
Elseif (mode==1) Then
  If (current_mode==0) Then
    t_hxstate=-Timestep
  Elseif (current_mode==2) Then
    t_hpstate=-Timestep
    t_hxstate=-Timestep
    HP_off=-Timestep
  Elseif (current_mode==3) Then
    t_hpstate=-Timestep
    HP_off=-Timestep
  Endif
Elseif (mode==2) Then
  If (current_mode==0) Then
    t_hpstate=-Timestep
  Elseif (current_mode==1 .OR. current_mode==4) Then
    t_hpstate=-Timestep
    t_hxstate=-Timestep
  Elseif (current_mode==3) Then
    t_hxstate=-Timestep
  Endif
Elseif (mode==3) Then
  If (current_mode==0) Then
    t_hpstate=-Timestep
    t_hxstate=-Timestep
  Elseif (current_mode==1 .OR. current_mode==4) Then
    t_hpstate=-Timestep
  Elseif (current_mode==2) Then
    t_hxstate=-Timestep
  Endif
Elseif (mode==4) Then
  If (current_mode==0) Then

```

```

    t_hxstate=-Timestep
Elseif (current_mode==2) Then
    t_hpstate=-Timestep
    t_hxstate=-Timestep
    HP_off=-Timestep
Elseif (current_mode==3) Then
    t_hpstate=-Timestep
    HP_off=-Timestep
Endif
Else
    If (mode > 4 .OR. mode < 0) Then
        Call SetOutputValue(2, Mode)
        call FoundBadParameter(1,'Mode','Unrecognized Mode')
        Return
    Else
        mode=0
    Endif
Endif

```

```

t_hxstate=t_hxstate+Timestep
t_hpstate=t_hpstate+Timestep
HP_off=HP_off+Timestep

```

!If the heat pump was on last step and the heat pump is not on anymore, set Stored(1) to the current time and turn off the HP status

!If the heat pump was not on last step, but it IS on now, turn on the HP status

```

If(current_mode==2 .OR. current_mode==3) Then
    If (mode /= 2 .AND. mode /= 3) Then
        Stored(1)=Time
        Stored(2)=0
    Endif
Elseif (mode==2 .OR. mode==3) Then
    Stored(2)=1
Endif

```

```

current_mode=mode

```

```

    Return
EndIf

```

!-----

!Do All of the "Very First Call of the Simulation Manipulations" Here

 If(getIsFirstCallofSimulation()) Then

 !Tell the TRNSYS Engine How This Type Works

 Call SetNumberOfParameters(34) !The number of parameters that the the model wants

 Call SetNumberOfInputs(24) !The number of inputs that the the model wants

 Call SetNumberOfDerivatives(0) !The number of derivatives that the the model wants

 Call SetNumberOfOutputs(37) !The number of outputs that the the model produces

 Call SetIterationMode(1) !An indicator for the iteration mode (default=1). Refer to section 8.4.3.5 of the documentation for more details.

 Call SetNumberStoredVariables(0,0) !The number of static variables that the model wants stored in the global storage array and the number of dynamic variables that the model wants stored in the global storage array

 Call SetNumberOfDiscreteControls(0) !The number of discrete control functions set by this model (a value greater than zero requires the user to use Solver 1: Powell's method)

 !Set correct units for the inputs and outputs

 Call SetInputUnits(1, 'TE1')

 Call SetInputUnits(2, 'TE1')

 Call SetInputUnits(3, 'TE1')

 Call SetInputUnits(4, 'TE1')

 Call SetInputUnits(5, 'IR1')

 Call SetInputUnits(6, 'TE1')

 Call SetInputUnits(7, 'TE1')

 Call SetInputUnits(8, 'TE1')

 Call SetInputUnits(9, 'TE1')

 Call SetInputUnits(10, 'TE1')

 Call SetInputUnits(11, 'TE1')

 Call SetInputUnits(12, 'TE1')

 Call SetInputUnits(13, 'TE1')

 Call SetInputUnits(14, 'TE1')

 Call SetInputUnits(15, 'TE1')

 Call SetInputUnits(16, 'TD1')

 Call SetInputUnits(17, 'TE1')

 Call SetInputUnits(18, 'IR1')

 Call SetInputUnits(19, 'IR1')

 Call SetInputUnits(20, 'DG1')

 Call SetInputUnits(21, 'TE1')

```
Call SetInputUnits(22, 'DM1')
Call SetInputUnits(23, 'DG1')
Call SetInputUnits(24, 'TE1')
```

!only set units for outputs for which this is appropriate

```
Call SetOutputUnits(11, 'PW1')
Call SetOutputUnits(12, 'PW1')
Call SetOutputUnits(13, 'PW1')
Call SetOutputUnits(14, 'PW1')
Call SetOutputUnits(15, 'PW1')
Call SetOutputUnits(17, 'PW1')
Call SetOutputUnits(18, 'DM1')
Call SetOutputUnits(19, 'DM1')
Call SetOutputUnits(20, 'DM1')
Call SetOutputUnits(22, 'TD1')
Call SetOutputUnits(23, 'TD1')
Call SetOutputUnits(26, 'TE1')
Call SetOutputUnits(27, 'TE1')
Call SetOutputUnits(28, 'TE1')
Call SetOutputUnits(29, 'TD1')
Call SetOutputUnits(30, 'TD1')
Call SetOutputUnits(31, 'TD1')
Call SetOutputUnits(32, 'PW1')
Call SetOutputUnits(33, 'PW1')
Call SetOutputUnits(34, 'PW1')
Call SetOutputUnits(35, 'PW1')
Call SetOutputUnits(36, 'PW1')
Call SetOutputUnits(37, 'PW1')
```

Return

EndIf

!-----

!Do All of the First Timestep Manipulations Here - There Are No Iterations at the Initial Time

```
If (getIsStartTime()) Then
AuxCapacity = getParameterValue(1)
Qstc_min = getParameterValue(2)
Qaux_on = getParameterValue(3)
aux_off_time = getParameterValue(4)
half_night = getParameterValue(5)
```

Tmax = getParameterValue(6)
Deadbands = getParameterValue(7)
Timer = getParameterValue(8)
tol = getParameterValue(9)
flow_sys = getParameterValue(10)
Cp = getParameterValue(11)
mass_load = getParameterValue(12)
mass_solar = getParameterValue(13)
UAhx = getParameterValue(14)
t_hxmin = getParameterValue(15)
t_hpmin = getParameterValue(16)
ScaleHP = getParameterValue(17)
Thplinmax = getParameterValue(18)
Thpsinmax = getParameterValue(19)
Thpsoutmin = getParameterValue(20)
t_hpduration = getParameterValue(21)
area = getParameterValue(22)
gtest = getParameterValue(23)
eta_int = getParameterValue(24)
eta_slope = getParameterValue(25)
eta_curve = getParameterValue(26)
b0 = getParameterValue(27)
rated_pump_flow = getParameterValue(28)
Ppump1_rated = getParameterValue(29)
Ppump1_C1 = getParameterValue(30)
Ppump1_C2 = getParameterValue(31)
Ppump2_rated = getParameterValue(32)
Ppump2_C1 = getParameterValue(33)
Ppump2_C2 = getParameterValue(34)

auxset = GetInputValue(1)
Ttank1 = GetInputValue(2)
Ttank2 = GetInputValue(3)
Ttank3 = GetInputValue(4)
GTt = GetInputValue(5)
Tstcin = GetInputValue(6)
Tstcout = GetInputValue(7)
Thxsin = GetInputValue(8)
Thxsout = GetInputValue(9)
Thxlin = GetInputValue(10)

```
Thxlout = GetInputValue(11)
Thpsin = GetInputValue(12)
Thpsout = GetInputValue(13)
Thplin = GetInputValue(14)
Thplout = GetInputValue(15)
t_target = GetInputValue(16)
Ttarget = GetInputValue(17)
gd = GetInputValue(18)
gh = GetInputValue(19)
theta = GetInputValue(20)
Tamb = GetInputValue(21)
rho = GetInputValue(22)
slope = GetInputValue(23)
Ttank_therm = GetInputValue(24)
```

```
!Set the radians conversion factor here
rdconv=0.017453
```

```
!Preset storage variables
Stored(1)=0
Stored(2)=0
Stored(3)=1
Stored(4)=1
```

```
!Pump calculations - only needed once since there's no inputs
```

```
PumpCS=flow_sys/rated_pump_flow
Ppump1=Ppump1_rated*(Ppump1_C1+Ppump1_C2*PumpCS)
Ppump2=Ppump2_rated*(Ppump2_C1+Ppump2_C2*PumpCS)
```

```
!Set the Initial Values of the Outputs (#,Value)
    Call SetOutputValue(1, 0) ! Stagnation
    Call SetOutputValue(2, 0) ! Mode
    Call SetOutputValue(3, 0) ! auxCS
    Call SetOutputValue(4, 0) ! CSp1
    Call SetOutputValue(5, 0) ! CSp2
    Call SetOutputValue(6, 0) ! CShp
    Call SetOutputValue(7, 0) ! CSv1
    Call SetOutputValue(8, 0) ! CSv2
    Call SetOutputValue(9, 0) ! CSv3
```

```

Call SetOutputValue(10, 0) ! CSv4
Call SetOutputValue(11, 0) ! Qstc
Call SetOutputValue(12, 0) ! Qhx
Call SetOutputValue(13, 0) ! Qhps
Call SetOutputValue(14, 0) ! Qhpl
Call SetOutputValue(15, 0) ! Qsrs
Call SetOutputValue(16, 0) ! Qmax
Call SetOutputValue(17, 0) ! Powerhp
Call SetOutputValue(18, 0) ! COPhx
Call SetOutputValue(19, 0) ! COPhp
Call SetOutputValue(20, 0) ! COPsrs
Call SetOutputValue(21, 0) ! COPbest
Call SetOutputValue(22, 0) ! t_hp
Call SetOutputValue(23, 0) ! t_hps
Call SetOutputValue(24, 0) ! HP_to
Call SetOutputValue(25, 0) ! current_mode
Call SetOutputValue(26, 0) ! Ttanktopavg
Call SetOutputValue(27, 0) ! Thxsoutsrs
Call SetOutputValue(28, 0) ! Thxloutsrs
Call SetOutputValue(29, 0) ! t_hpstate
Call SetOutputValue(30, 0) ! t_hxstate
Call SetOutputValue(31, 0) ! HP_off
Call SetOutputValue(32, 0) ! Power HP Srs
Call SetOutputValue(33, 0) ! Qhp,s srs
Call SetOutputValue(34, 0) ! Qhp,l srs
Call SetOutputValue(35, Ppump1)
Call SetOutputValue(36, Ppump2)
Call SetOutputValue(37, AuxPower)

```

```
Return
```

```
EndIf
```

```
!-----
```

```
!ReRead the Parameters if Another Unit of This Type Has Been Called Last
```

```
  If(getIsReReadParameters()) Then
```

```
    !Read in the Values of the Parameters from the Input File
```

```
    AuxCapacity = getParameterValue(1)
```

```
    Qstc_min = getParameterValue(2)
```

```
    Qaux_on = getParameterValue(3)
```

```
    aux_off_time = getParameterValue(4)
```



```
half_night = getParameterValue(5)
Tmax = getParameterValue(6)
Deadbands = getParameterValue(7)
Timer = getParameterValue(8)
tol = getParameterValue(9)
flow_sys = getParameterValue(10)
Cp = getParameterValue(11)
mass_load = getParameterValue(12)
mass_solar = getParameterValue(13)
UAhx = getParameterValue(14)
t_hxmin = getParameterValue(15)
t_hpmin = getParameterValue(16)
ScaleHP = getParameterValue(17)
Thplinmax = getParameterValue(18)
Thpsinmax = getParameterValue(19)
Thpsoutmin = getParameterValue(20)
t_hpduration = getParameterValue(21)
area = getParameterValue(22)
gtest = getParameterValue(23)
eta_int = getParameterValue(24)
eta_slope = getParameterValue(25)
eta_curve = getParameterValue(26)
b0 = getParameterValue(27)
rated_pump_flow = getParameterValue(28)
Ppump1_rated = getParameterValue(29)
Ppump1_C1 = getParameterValue(30)
Ppump1_C2 = getParameterValue(31)
Ppump2_rated = getParameterValue(32)
Ppump2_C1 = getParameterValue(33)
Ppump2_C2 = getParameterValue(34)
```

```
EndIf
```

!-----

```
!Read the Inputs
```

```
auxset = GetInputValue(1)
Ttank1 = GetInputValue(2)
Ttank2 = GetInputValue(3)
Ttank3 = GetInputValue(4)
GTt = GetInputValue(5)
```

```

Tstcin = GetInputValue(6)
Tstcout = GetInputValue(7)
Thxsin = GetInputValue(8)
Thxsout = GetInputValue(9)
Thxlin = GetInputValue(10)
Thxlout = GetInputValue(11)
Thpsin = GetInputValue(12)
Thpsout = GetInputValue(13)
Thplin = GetInputValue(14)
Thplout = GetInputValue(15)
t_target = GetInputValue(16)
Ttarget = GetInputValue(17)
gd = GetInputValue(18)
gh = GetInputValue(19)
theta = GetInputValue(20)
Tamb = GetInputValue(21)
rho = GetInputValue(22)
slope = GetInputValue(23)
Ttank_therm = GetInputValue(24)

```

!Check the Inputs for Problems (#,ErrorType,Text)

!Sample Code: If(IN1 <= 0.) Call FoundBadInput(1,'Fatal','The first input provided to this model is not acceptable.')

If(ErrorFound()) Return

```

!-----
! *** PERFORM ALL THE CALCULATION HERE FOR THIS MODEL. ***
!-----

```

!PHASE 1 - STARTUP

!STAGE 1 – START-OF-ITERATION CALCULATIONS

!STAGE 1 STEP 1: Calculate the average upper-tank temperature

$$T_{\text{tanktopavg}} = (T_{\text{tank1}} + T_{\text{tank2}} + T_{\text{tank3}}) / 3$$

! STAGE 1 STEP 2: Calculations taken from Type 1b - the Solar Thermal Collector to estimate solar gain.

!Assumes i_optmode 2 and emode 1, applies xns=1. b1=0 in the STC so it is assumed 0.

!fratio=1 so it is ignored. r3=1 so ignored. xns=1 results in r2=1, eliminating the need for r1,r2,xk.

```
If((GTt<=0.) .OR. (theta>=90.)) Then
```

```
  xkat=0.
```

```
  goto 5
```

```
EndIf
```

```
effsky=59.68-0.1388*slope+0.001497*slope*slope
```

```
effgnd=90.-0.5788*slope+0.002693*slope*slope
```

```
cosslope=DCOS(slope*rdconv)
```

```
fsky=(1.+cosslope)/2.
```

```
fgnd=(1.-cosslope)/2.
```

```
gdsky=fsky*gd
```

```
gdgnd=rho*fgnd*gh
```

```
xkatb=TauAlf(theta)
```

```
xkatds=TauAlf(effsky)
```

```
xkatdg=TauAlf(effgnd)
```

```
xkat=(xkatb*(GTt-gdsky-gdgnd)+xkatds*gdsky+xkatdg*gdgnd)/GTt
```

```
If (xkat<=0) xkat=0.
```

```
5  frulp=eta_slope+eta_curve*(Tstcin-Tamb)
```

```
ftest=frulp/gtest/Cp
```

```
If (ftest>=1) Then
```

```
  fpul=frulp
```

```
Else
```

```
  fpul=-gtest*Cp*DLOG(1.-frulp/gtest/Cp)
```

```
EndIf
```

```
rtest=gtest*Cp*(1.-DEXP(-fpul/gtest/Cp))
```

```
ratio=flow_sys*Cp/area*(1.-DEXP(-fpul*area/1/flow_sys/Cp))/rtest
```

```
Qstc=ratio*area*(eta_int*xkat*GTt-frulp*(Tstcin-Tamb))
```

! STAGE 1 STEP 3: Run HP parameter calculations using the performance map developed by Wagar

```
a = 3.9790858669966377E+02
```

b = -5.5334702242327545E+00
 c = 1.5933153236699374E+01
 d = 2.5389274937948403E-01
 f = -2.0850956315706226E-01
 g = -3.0062593306920261E-03
 h = 4.2934663209162971E-03
 i = -1.3560045271329768E-01
 j = 9.2726503781772164E-05
 k = 3.1463637147147051E-03

as = 2.2792422799816873E+02
 bs = 2.7089521676786966E+01
 cs = 5.6909204208573010E+02
 ds = -1.0980819858415735E+01
 fs = -2.9092386404767815E+01
 gs = 6.6651413998470577E-01
 hs = 5.8121232503911524E-01
 is = -1.3124628397107709E-02

al = 6.6387779902539087E+03
 bl = -1.0777549071025348E+02
 cl = -6.6531594240976426E+02
 dl = 1.7794980114293022E+01
 fl = 4.7622518320339196E+01
 gl = -1.1335033919170665E+00
 hl = -8.8722168639507348E-01
 il = 2.1386903667490804E-02

x=Thpsin
 y=Thplin

!heat pump calculations were written in Watts by Wagar; a conversion is offered so that the controller may use the traditional kJ/hr for TRNSYS components

$$\text{Powerhp} = \text{ScaleHP} * (a + b*x + c*y + d*(x**2) + f*(y**2) + g*(x**3) + h*(y**3) + i*x*y + j*(x**2)*y + k*x*(y**2))$$

$$\text{Powerhp} = \text{Powerhp} * 3.6$$

$$\text{Qhps} = \text{ScaleHP} * (as + bs*y + cs*x + ds*x*y + fs*(x**2) + gs*(x**2)*y + hs*(x**3) + is*(x**3)*y)$$

$$Q_{hps} = Q_{hps} * 3.6$$

$$Q_{hpl} = \text{ScaleHP} * (a_l + b_l * y + c_l * x + d_l * x * y + f_l * (x^{**2}) + g_l * (x^{**2}) * y + h_l * (x^{**3}) + i_l * (x^{**3}) * y)$$

$$Q_{hpl} = Q_{hpl} * 3.6$$

$$T_{\text{sourceout}} = x - Q_{hps} / (\text{flow_sys} * C_p)$$

!Trigger low-temperature shutdown if temperatures are close to freezing

If (DABS(T_{sourceout}-T_{hpsoutmin})<=tol) Then

 T_{sourceout}=T_{hpsoutmin}

Endif

If (DABS(Q_{hps}-Q_{stc})<=(tol*10)) Then

 t_{hp}=mass_solar*C_p*(T_{sourceout}-T_{hpsoutmin})/(1) !coerce the difference to 1 kJ/hr

Elseif (Q_{stc} > Q_{hps}) Then

 t_{hp}=1000

Else

 t_{hp}=mass_solar*C_p*(T_{sourceout}-T_{hpsoutmin})/(Q_{hps}-Q_{stc})

Endif

! STAGE 1 STEP 4: HX calculations from Type 5.

!It is assumed that C_{min} = C_{max} since identical fluids, similar temperatures and identical flowrates are used on both sides.

$$\eta = (U A_{hx} / (\text{flow_sys} * C_p)) / (1 + (U A_{hx} / (\text{flow_sys} * C_p)))$$

$$Q_{hx} = \eta * \text{flow_sys} * C_p * (T_{hxsin} - T_{hxlin})$$

$$T_{hxoutsrs} = T_{hxsin} - Q_{hx} / (\text{flow_sys} * C_p)$$

$$T_{hxloutsrs} = T_{hxlin} + Q_{hx} / (\text{flow_sys} * C_p)$$

! STAGE 1 STEP 5: Calculate HP performance based on HX outlets, for SRS mode predictions

$$x = T_{hxoutsrs}$$

$$y = T_{hxloutsrs}$$

$$\text{Power}_{hpsrs} = \text{ScaleHP} * (a + b * x + c * y + d * (x^{**2}) + f * (y^{**2}) + g * (x^{**3}) + h * (y^{**3}) + i * x * y + j * (x^{**2}) * y + k * x * (y^{**2}))$$

$$\text{Power}_{hpsrs} = \text{Power}_{hpsrs} * 3.6$$

$Q_{hpssrs} = \text{ScaleHP} * (a_s + b_s * y + c_s * x + d_s * x * y + f_s * (x^{**2}) + g_s * (x^{**2}) * y + h_s * (x^{**3}) + i_s * (x^{**3}) * y)$
 $Q_{hpsrs} = Q_{hpssrs} * 3.6$

$Q_{hplsrs} = \text{ScaleHP} * (a_l + b_l * y + c_l * x + d_l * x * y + f_l * (x^{**2}) + g_l * (x^{**2}) * y + h_l * (x^{**3}) + i_l * (x^{**3}) * y)$
 $Q_{hplsrs} = Q_{hplsrs} * 3.6$

$Q_{srs} = Q_{hx} + Q_{hplsrs}$

$T_{sourceoutsrs} = x - Q_{hpssrs} / (\text{flow_sys} * C_p)$

If (DABS($T_{sourceoutsrs} - T_{hpsoutmin}$) <= tol) Then
 $T_{sourceoutsrs} = T_{hpsoutmin}$
 Endif

If (DABS($Q_{hx} + Q_{hpssrs} - Q_{stc}$) <= (tol * 10)) Then
 $t_{hps} = \text{mass_solar} * C_p * (T_{sourceoutsrs} - T_{hpsoutmin}) / (1)$
 Elseif ($Q_{stc} > (Q_{hx} + Q_{hpssrs})$) Then
 $t_{hps} = 1000$
 Else
 $t_{hps} = \text{mass_solar} * C_p * (T_{sourceoutsrs} - T_{hpsoutmin}) / (Q_{hx} + Q_{hpssrs} - Q_{stc})$
 Endif

! STAGE 1 STEP 6: Determine Qmax

If ($Q_{hx} \geq Q_{hpl}$.AND. $Q_{hx} \geq Q_{srs}$.AND. $Q_{hx} > 0$) then
 $Q_{max} = 1$
 Else if ($Q_{hpl} > Q_{hx}$.AND. $Q_{hpl} > Q_{srs}$.AND. $Q_{hpl} > 0$) then
 $Q_{max} = 2$
 Else if ($Q_{srs} > Q_{hx}$.AND. $Q_{srs} \geq Q_{hpl}$.AND. $Q_{srs} > 0$) then
 $Q_{max} = 3$
 Else
 $Q_{max} = 4$
 Endif

! STAGE 1 STEP 7: Calculate Pmode, PMIN, COPs and determine COPbest

$PHX = P_{pump1} + P_{pump2}$
 $PHP = P_{pump1} + P_{pump2} + \text{Power}_{hp}$
 $PSRS = P_{pump1} + P_{pump2} + \text{Power}_{hpsrs}$

```

If (PHX <= PHP .AND. PHX <= PSRS) Then
  PMIN=1
Elseif (PHP < PSRS .AND. PHP < PHX) Then
  PMIN=2
Elseif (PSRS <= PHP .AND. PSRS < PHX) Then
  PMIN=3
Else
  PMIN=4
Endif

```

```

COPhx = Qhx/PHX
COPhp = Qhpl/PHP
COPsrs = Qsrs/PSRS

```

```

If (COPhx >= COPhp .AND. COPhx >= COPsrs .AND. COPhx > 0) then
  COPbest=1
Else if (COPhp > COPhx .AND. COPhp > COPsrs .AND. COPhp > 0) then
  COPbest=2
Else if (COPsrs > COPhx .AND. COPsrs >= COPhp .AND. COPsrs > 0) then
  COPbest=3
Else
  COPBest=4
Endif

```

! STAGE 1 STEP 8: Determine the load

```

If(t_target<=(tol/0.4)) Then
  t_target = tol/0.4    !designed to be 15 minutes, in units of hours
Endif

```

```

Qload = mass_load*Cp*(Ttarget - Ttanktopavg)/t_target

```

!STAGE 2 – AUTOMATIC SHUTDOWN CONDITIONS

!STAGE 2 STEP 1 – OVERNIGHT SHUTDOWN

!Disable from 8 PM to 4 AM - skip straight to auxiliary management if appropriate

```

If (ABS((Time/24) - NINT(Time/24))*24 <= half_night) Then
  mode=0
  GOTO 125

```

!STAGE 2 STEP 2 – HP LONG CYCLE SHUTDOWN

!Set to WU mode if the maximum HP cycle time is reached

```
Elseif (t_hpstate >= t_hpduration .AND. Stored(2)==1) Then
  mode=4
  GOTO 125
Endif
```

!STAGE 2 STEP 3 – CONTROLLER CYCLE TIMER

!The timer should only allow controller operation on specific intervals.

!If Timer > Timestep, it can't run every time. This logic applies whether or not the timestep is a factor of Timer.

!Don't change anything if nstep*timestep < nruns*timer. But if it's >=, increment the # of runs counter and proceed through the code.

!If the timestep is greater than the timer, it will run every timestep. It can't run twice. Likewise, if they are equal, it will run every time.

!Thus the only case where the controller shouldn't run, is if Timer > Timestep and the overflow doesn't run Timer out.

!If Timer <= Timestep, the counters are not needed

```
If(Timer>Timestep) Then
  If((Stored(4)*Timestep) < (Stored(3)*Timer)) Then
    GOTO 200
  Else
    Stored(3)=Stored(3)+1
  Endif
Endif
```

!STAGE 2 STEP 4 – “STAGNATION” CHECK

```
If (Ttank1 >= 90 .OR. Tstcout >= 90 .OR. Thxsout >= 90 .OR. Thxlout >= 90) Then
  Stagnation=.TRUE.
Else
  Stagnation=.FALSE.
Endif
```

!STAGE 2 STEP 5 – FREEZE PREVENTION

```
If (Thxsout <= Thpsoutmin .OR. Thpsout <= Thpsoutmin .OR. Tstcin <= Thpsoutmin) Then
  mode=4      !too cold - activate warmup mode
```



```
GOTO 100
Endif          !if not, proceed
```

!STAGE 3 – ESTABLISHING DESIRED SYSTEM STATE: ON/OFF

!STAGE 3 STEP 1 – CHECK FOR EXCESS HEAT IN TANK

```
If (Ttanktopavg > Tmax) Then
  mode=0          !energy is not needed, turn OFF
  GOTO 100
Endif            !energy may be needed; continue
```

!STAGE 3 STEP 2 – CHECK FOR MINIMUM SOLAR GAIN CONDITIONS

```
If (Qstc <= Qstc_min) Then
  mode=0          !energy is not available - turn off
  GOTO 100
Endif            !yes, energy is available – proceed
```

!PHASE 2 – MODE SELECTION

!STAGE 4 – DETERMINE ENERGY DEMAND: HIGH/LOW

!STAGE 4 STEP 1 – ASSESS TANK TEMPERATURE CONDITION

!Is the tank at a moderate condition? If so, try to use the lowest energy mode available. If the tank is at a low condition, try to use the most effective mode instead.

```
If (auxset <= Ttanktopavg) Then
```

!STAGE 5 – MINIMUM POWER PROTOCOL

!This sequence checks if PHX is the least - if not, it checks PSRS and PHP. When it finds PMIN, it asks if that P is associated with good COP and positive Q.

!If it is, it chooses that mode. If not, it checks the 2nd priority mode, and then the 3rd. If no mode is both efficient and beneficial, it does nothing.

!In general, first priority is given to HX, then SRS, then HP. No mode will run unless it is both efficient and energy-positive.

!PERFORM STAGE 5 STEPS 1-5 WHERE APPROPRIATE

<pre> If (PMIN==1) Then If (COPhx >= 1 .AND. Qhx > 0) Then mode=1 GOTO 100 ElseIf (PHP < PSRS) Then If (COPhp >=1 .AND. Qhpl > 0) Then mode=2 GOTO 30 Endif ElseIf (COPsrs >=1 .AND. Qsrs > 0) Then mode=3 GOTO 30 ElseIf (COPhp >=1 .AND. Qhpl > 0) Then mode=2 GOTO 30 Else mode=0 GOTO 100 Endif Elseif (PMIN==2) Then If (COPhp >=1 .AND. Qhpl > 0) Then mode=2 GOTO 30 ElseIf (PHX <= PSRS) Then If (COPhx >= 1 .AND. Qhx > 0) Then mode=1 GOTO 100 Endif ElseIf (COPsrs >=1 .AND. Qsrs > 0) Then mode=3 GOTO 30 ElseIf (COPhx >= 1 .AND. Qhx > 0) Then mode=1 GOTO 100 Else mode=0 GOTO 100 Endif Elseif (PMIN==3) Then If (COPsrs >=1 .AND. Qsrs > 0) Then </pre>	<pre> !STEP 1: IDENTIFY PMIN !STEP 2: EVALUATE PMIN !STEP 3: ID NEXT-BEST !STEP 4: EVALUATE N-B !STEP 5: EVALUATE LAST </pre>
--	--

```

mode=3
GOTO 30
Elseif (PHP <= PHX) Then
  If (COPhp >= 1 .AND. Qhpl > 0) Then
    mode=2
    GOTO 30
  Endif
Elseif (COPhx >=1 .AND. Qhx > 0) Then
  mode=1
  GOTO 100
ElseIf (COPhp >= 1 .AND. Qhpl > 0) Then
  mode=2
  GOTO 30
Else
  mode=0
  GOTO 100
Endif
Else
  If (COPhx > 1 .AND. Qhx > 0) then
    mode=1
    GOTO 100
  Else
    mode=0
    GOTO 100
  Endif
Endif
Endif

```

!STAGE 6 – MAXIMUM RATE OF HEAT GAIN PROTOCOL

!If the tank is hot, it's already turned off; if the need is moderate, a low-power mode has been selected. If it's cold - can the best COP mode meet the load?

!This sequence checks, one by one, if the best COP is HX, HP or SRS. When the COPbest is found, the sequence asks if the associated Q meets the load.

!Used sporadically through the Max Q protocol, mode 9 is a dummy number so that the controller will not override COPbest unless the load is not met.

!PERFORM STAGE 6 STEPS 1-2 WHERE APPROPRIATE

```
If (COPbest==1) Then
```

!STEP 1: IDENTIFY COPBEST

!STEP 2: EVALUATE COPBEST

```
If (Qhx >= Qload) Then
  mode=1
  GOTO 100
Else
  mode=9
  GOTO 25
Endif
Elseif (COPbest==2) Then
  If (Qhpl >= Qload) Then
    mode=2
    GOTO 30
  Else
    mode=9
    GOTO 25
  Endif
Elseif (COPbest==3) Then
  If (Qsrs >= Qload) Then
    mode=3
    GOTO 30
  Else
    mode=9
    GOTO 25
  Endif
Else
  mode=4
  GOTO 100
Endif
```

!If best COP cannot meet the load, the most effective mode (that with the highest heat transfer rate) should run instead.

!This sequence determines which mode is associated with Qmax

!STAGE 6 STEP 3 – IDENTIFY QMAX

```
25 If (mode==9) then
  If (Qmax==1) Then
    mode=1
    GOTO 100
  Elseif (Qmax==2 .OR. Qmax==3) Then
    GOTO 30
```

```
Else
  mode=4
  GOTO 100
Endif
Endif
```

!STAGE 7 – HEAT PUMP SAFETY CHECK

!PERFORM STAGE 7 STEPS 1-6 WHERE APPROPRIATE

!PERFORM STEPS 2-6 ONLY AS REQUIRED

30 If (mode==2 .OR. (Qmax==2 .AND. mode==9)) then

!STAGE 7 STEP 1 – IDENTIFY HP STATE

!Don't ask these questions if the heat pump is already on – skip to Step 4.

```
If (current_mode/=2 .AND. current_mode/=3) then
  If ((Time-Stored(1)) <= t_hpmin) then      !STEP 2: CHECK TIMEOUT STATE
    mode=4
    GOTO 100 Else
  If (t_hp < t_hpmin) then      !STEP 3: CHECK HP OPERATING POTENTIAL
    mode=4
    GOTO 100
Endif
Endif
```

!STAGE 7 STEP 4 – CHECK INLET TEMPERATURES

```
If (Thplin <= Thplinmax .AND. Thpsin <= Thpsinmax) then
  mode=2
  GOTO 100
```

!STAGE 7 STEP 5 – EVALUATE POTENTIAL OF SRS MODE

```
Else
  If (Thxsoutsrs<=Thpsinmax .AND. Thxloutsrs<=Thplinmax .AND. COPsrs>=1) then
    mode=3
    GOTO 100
```

!STAGE 7 STEP 6 – EVALUATE POTENTIAL OF HX MODE

```

Else
  If (COPhx > 1 .AND. Qhx > 0) then
    mode=1
    GOTO 100
  Else
    mode=0
    GOTO 100
  Endif
Endif
Endif

Elseif (mode==3 .OR. (Qmax==3 .AND. mode==9)) then
  If (current_mode/=2 .AND. current_mode/=3) then
    If ((Time-Stored(1)) <= t_hpmin) then           !check timeout
      If (COPhx >= 1) then
        mode=1                                     !just run HX instead until the timeout is off
        GOTO 100
      Else                                         !if HX is not acceptable, run warmup
        mode=4
        GOTO 100
      Endif Else
      If (t_hps < t_hpmin) then                   !if timeout is off, check predicted run time
        If (COPhx >= 1) then
          mode=1                                   !just run HX instead until the circuit is warm enough
          GOTO 100
        Else
          mode=4
          GOTO 100                               !If HX is not profitable, run warmup
        Endif
      Endif
    Endif
  Endif
Endif

If (Thxsoutsrs <= Thpsinmax .AND. Thxloutsrs <= Thplinmax .AND. COPsrs >= 1) then
  mode=3
  GOTO 100
Else
  If (COPhx > 1 .AND. Qhx > 0) then
    mode=1
    GOTO 100
  
```

```

    Else
        mode=0
        GOTO 100
    Endif
Endif
Else
    If (COPhx > 1 .AND. Qhx > 0) then
        mode=1
        GOTO 100
    Elseif (Tstcin < 90) then
        mode=4
        GOTO 100
    Else
        mode=0
        GOTO 100
    Endif
Endif

```

!PHASE 3 - CONFIRMATION

!STAGE 8 – CONDITIONAL OVERRIDE

100 If (mode==0 .AND. (current_mode==2 .OR. current_mode==3)) Then

!If there is no energy available, or if the HX cannot produce safe HP inlet temperatures, turn off.
 !If energy is available, and HP inlet temperatures are safe, then it's OFF because $T > 60$. Stay on for now.

```

        If(Qstc <= Qstc_min .OR. (Thxsoutsrs > Thpsinmax .OR. Thxloutsrs > Thplinmax .OR.
COPsrs < 1)) Then
            mode=0
            Elseif (t_hpstate < t_hpmin) Then
                mode=current_mode
            Endif

```

Elseif (mode==1 .AND. (current_mode==2 .OR. current_mode==3)) Then

!Don't change if HP hasn't been on long enough
 !If the system wants to do HX, it should, as long as it isn't cycling too often.

```
If (t_hpstate < t_hpmin) Then
    mode=current_mode
Endif
```

```
Elseif (mode==2 .AND. (current_mode==1 .OR. current_mode==3)) Then
    If (t_hxstate < t_hxmin) Then
        mode=current_mode
    Endif
```

```
Elseif (mode==3 .AND. current_mode==2) Then
    If (Thplin <= Thplinmax .AND. Thpsin <= Thpsinmax .AND. t_hxstate < t_hxmin) Then
        mode=2
    Endif
```

```
Elseif (mode==4 .AND. (current_mode==2 .OR. current_mode==3)) Then
    If (Thxsout <= Thpsoutmin .OR. Thpsout <= Thpsoutmin .OR. Tstcin <= Thpsoutmin) Then
        mode=4
    Else
```

!t_hp falling low can trigger warmup if the system is off, but it shouldn't if the system is on. This guards against that occurrence.

```
        mode=current_mode
    Endif
Endif
```

!STAGE 9 – AUXILIARY MANAGEMENT

!STAGE 9 STEP 1 – CHECK FOR TIME-OF-DAY LIMIT

!Disable the auxiliary if the time is more than 8 hours away from midnight (8 AM to 4 PM)

!However, do not disable the auxiliary if the amount of sun available is low. Limit is based on Jan 1 levels but is deliberately imprecise.

```
If ((ABS((Time/24) - NINT(Time/24))*24) >= aux_off_time .AND. Qstc >= Qaux_on) Then
    auxCS=0
    GOTO 150
Endif
```

!STAGE 9 STEP 2 – DEFINE QMODE

!Establish what Qmode is, based on the mode selection


```

125 If (mode==1) Then
    Qmode=Qhx
Elseif (mode==2) Then
    Qmode=Qhpl
Elseif (mode==3) Then
    Qmode=Qsrs
Else
    Qmode=0
Endif

```

!STAGE 9 STEP 3 – CHECK SYSTEM NEED

!Allow the tank to help if it is known that the current mode will not suffice and the tank is cool.

!If the current operating mode will meet the load or the tank is appropriately warm, don't allow the auxiliary heater to activate.

```

    If (Qmode < Qload .AND. Ttanktopavg < auxset) Then
        auxCS=1
    Else
        auxCS=0
    Endif

```

!STAGE 9 STEP 4 – APPLY DEAD-BANDS

!Let auxiliary turn on if Ttank is lower than aux-db, or if Qmode is lower than Qload - Qdb

```

    If (auxCS==1 .AND. prev_auxCS==0) Then
        If (Ttanktopavg < (auxset-Deadbands))Then
            auxCS=1
        Elseif (Qmode < (Qload-flow_sys*Cp*Deadbands)) Then
            auxCS=1
        Else
            auxCS=0
        Endif
    Endif

```

!STAGE 9 STEP 5 – CHECK SIGNAL AND CONFIRM

!Set the output auxiliary power (required for Type 534 tank)

```

150 If (auxCS==1 .AND. Ttank_therm <= Tmax) Then
    AuxPower=AuxCapacity

```

```
Else
  AuxPower=0
Endif
```

```
prev_auxCS=auxCS
```

!Now send the appropriate signals

```
If (mode==0) Then
```

```
  CSp1=0
```

```
  CSp2=0
```

```
  CShp=0
```

```
  CSv1=0
```

```
  CSv2=0
```

```
  CSv3=0
```

```
  CSv4=0
```

```
Elseif (mode==1) Then
```

```
  CSp1=PumpCS
```

```
  CSp2=PumpCS
```

```
  CShp=0
```

```
  CSv1=0
```

```
  CSv2=0
```

```
  CSv3=0
```

```
  CSv4=0
```

```
Elseif (mode==2) Then
```

```
  CSp1=PumpCS
```

```
  CSp2=PumpCS
```

```
  CShp=1
```

```
  CSv1=1
```

```
  CSv2=0
```

```
  CSv3=1
```

```
  CSv4=0
```

```
Elseif (mode==3) Then
```

```
  CSp1=PumpCS
```

```
  CSp2=PumpCS
```

```
  CShp=1
```

```
  CSv1=0
```

```
  CSv2=1
```

```
  CSv3=0
```

```
  CSv4=1
```

```

Elseif (mode==4) Then
  CSp1=PumpCS
  CSp2=0
  CShp=0
  CSv1=1
  CSv2=0
  CSv3=0
  CSv4=0
Endif

```

!-----

```

!Set the Outputs from this Model (#,Value)
  Call SetOutputValue(1, Stagnation) ! Stagnation
  Call SetOutputValue(2, mode) ! Mode
  Call SetOutputValue(3, auxCS) ! auxCS
  Call SetOutputValue(4, CSp1) ! CSp1
  Call SetOutputValue(5, CSp2) ! CSp2
  Call SetOutputValue(6, CShp) ! CShp
  Call SetOutputValue(7, CSv1) ! CSv1
  Call SetOutputValue(8, CSv2) ! CSv2
  Call SetOutputValue(9, CSv3) ! CSv3
  Call SetOutputValue(10, CSv4) ! CSv4
  Call SetOutputValue(11, Qstc) ! Qstc
  Call SetOutputValue(12, Qhx) ! Qhx
  Call SetOutputValue(13, Qhps) ! Qhps
  Call SetOutputValue(14, Qhpl) ! Qhpl
  Call SetOutputValue(15, Qsrs) ! Qsrs
  Call SetOutputValue(16, Qmax) ! Qmax
  Call SetOutputValue(17, Powerhp) ! Powerhp
  Call SetOutputValue(18, COPhx) ! COPhx
  Call SetOutputValue(19, COPhp) ! COPhp
  Call SetOutputValue(20, COPsrs) ! COPsrs
  Call SetOutputValue(21, COPbest) ! COPbest
  Call SetOutputValue(22, t_hp) ! t_hp
  Call SetOutputValue(23, t_hps) ! t_hps
  Call SetOutputValue(24, Stored(2)) ! HP_to state
  Call SetOutputValue(25, current_mode) ! current_mode
  Call SetOutputValue(26, Ttanktopavg) ! Ttanktopavg
  Call SetOutputValue(27, Thxsoutsrs) ! Thxsoutsrs
  Call SetOutputValue(28, Thxloutsrs) ! Thxloutsrs

```

```
Call SetOutputValue(29, t_hpstate) ! t_hpstate
Call SetOutputValue(30, t_hxstate) ! t_hxstate
Call SetOutputValue(31, Stored(1)) ! HP_off time
Call SetOutputValue(32, Powerhpsrs)
Call SetOutputValue(33, Qhpsrs)
Call SetOutputValue(34, Qhplsrs)
Call SetOutputValue(35, Ppump1)
Call SetOutputValue(36, Ppump2)
Call SetOutputValue(37, AuxPower)
```

!-----

!-----

```
!If Needed, Store the Desired Discret Control Signal Values for this Iteration (#,State)
!Sample Code: Call SetDesiredDiscreteControlState(1,1)
```

!-----

!-----

```
!If Needed, Store the Final value of the Dynamic Variables in the Global Storage Array (#,Value)
!Sample Code: Call SetDynamicArrayValueThisIteration(1,T_FINAL_1)
```

!-----

```
200 Return
    End
```

!-----