USING TRNSYS TYPES 4, 60 AND 534 TO MODEL RESIDENTIAL COLD THERMAL STORAGE USING WATER AND WATER/GLYCOL SOLUTIONS

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ABSTRACT

As the peak load placed on the electrical grid increases, and with the introduction of time of use billing, a greater emphasis must be placed on when energy is consumed. This paper presents the validation of three standard TRNSYS tank models (Type 60, Type 534 and Type 4) when used to model a cold thermal storage systems using water and a 50/50 glycol/water solution (by volume) as the storage medium. An experimental set-up was designed and constructed to measure the inlet conditions and the temperature distribution within a thermal storage tank. It was determined that all three models predict almost identical temperature distributions within the cold thermal storage tank using both water and a water/glycol solution. When modelling cold storage using water, the simulation provides an accurate representation at low flow rates and at higher temperatures, while the model loses its accuracy as flow rates increase and the temperature approaches freezing. When modelling the glycol/water solution, the standard tank models accurately predict the temperature distribution across all temperatures and flow rates tested.

INTRODUCTION

As the demand for residential space cooling continues to increase in Canada, an every growing peak load is being placed on the electrical grid to meet this cooling load [1]. This increased peak load has many detrimental effects including increasing the required capacity of the electrical grid, as the transmission and generation capacity must be sized to meet the annual maximum load [2]. Additionally, during peak loading periods, in many jurisdictions, including Ontario, the base electrical demand is met using low greenhouse gas emitting methods including nuclear and hydroelectricity, while peak loads must be met using greenhouse gases intensive methods that typically burn fossil fuels [2,3]. As a result, energy used during peak periods generates significantly more greenhouse gases per unit energy compared to non-peak periods.

To combat the increase in peak loading, policies have been put in place to encourage consumers to move their electricity consumption to off-peak periods. This includes time of use billing, where consumers are charged much higher rates (commonly more than twice the price) during peak periods compared to off periods [4]. Additionally, rebates are being provided to home owners to replace heavy consuming devices, and to install smart thermostats that can be controlled by the utility provider, cycling air conditioners on and off systematically to reduce the overall peak load consumption.

In addition to the policy changes that have a positive effect on the reduction of peak loads, new technologies could play a pivotal role in the long term reduction of peak electricity consumption. One such system that has significant potential is pairing a heat pump with thermal storage to generate cooling potential in off-peak periods and then store this potential until cooling is required during peak periods. Before these systems can experience widespread implementation within the residential market place, extensive research and modelling must be conducted to determine the feasibility and to optimize the systems to ensure they can realize a reduction in peak loading, and when implementing time of use billing, can realize costs savings for the consumer.

One possible system configuration that could be used to reduce the peak loading consists of a liquid-to-liquid heat pump with a cold thermal storage on the source side and a hot thermal storage system on the load side. The air conditioning is then provided as needed from the cold thermal storage while the heating is provided by the hot thermal storage when needed. One of the simplest and most common methods for thermal storage is using insulated cylindrical storage tanks, and use water as the storage medium. Additionally, when being used as a cold thermal storage, a water/glycol solution can be used to allow lower temperatures and to prevent freezing. Theses tanks can either be completely mixed, or if lower flow rates are used, these tanks can be stratified, creating a temperature gradient from the warmest fluid at the top, and coldest fluid at the bottom of the tank. The use of a stratified tank can offer many benefits and has been extensively studied within hot thermal storage tanks [5-9].

To date, almost all research on stratified thermal storage tanks has been completed using hot thermal storage tanks, with little work completed on cold thermal storage tanks. Stratification within a cold thermal storage tank provides a number of benefits, including providing chilled water quicker, as the entire tank does not need to be completely chilled before use. Additionally, a typical deterrent to using thermal storage systems as the storage medium decreases in temperature, the entering temperature to the evaporator decreases, and subsequently the performance of the heat pump decreases [10]. Using a stratified tank keeps the fluid temperature leaving the top of the tank and entering the heat pump constant and at a higher temperature compared to a fully mixed tank. This is valid until the complete tank is cycled through the heat pump, at which point the entering temperature steps down to the exiting temperature of the first pass. This can provide an increase in the overall performance of the system.

Before a complete system can be modelled, it is imperative that the individual components of the system be validated. Stratified hot tank models have been extensively studied and validated using TRNSYS, however there is little literature on using these cylindrical tank models for small scale, stratified thermal storage systems [11]. This paper presents the steps that were undertaken to model and validate a stratified thermal storage tank to store cooling potential using both water and a 50/50 water/glycol solution (by volume) as the storage medium.

EXPERIMENTAL SET-UP

The first task in this study was to design, construct and instrument a full-scale experimental test set-up in a laboratory setting. Typically in a validation study, a model is first created followed by the experimental test set-up, however in this case, a black box model will be used to model the heat pump being used, and as such, experimental data is required to create the performance maps used as an input to the model. These performance maps must be developed before the complete system can be modelled. As such, a single experimental set-up was designed that could be used to both experimentally evaluate the steady state performance of the heat pump, while also being used to obtain the transient performance of the heat pump and thermal storage tanks. A high level schematic of the experimental test set-up is shown in

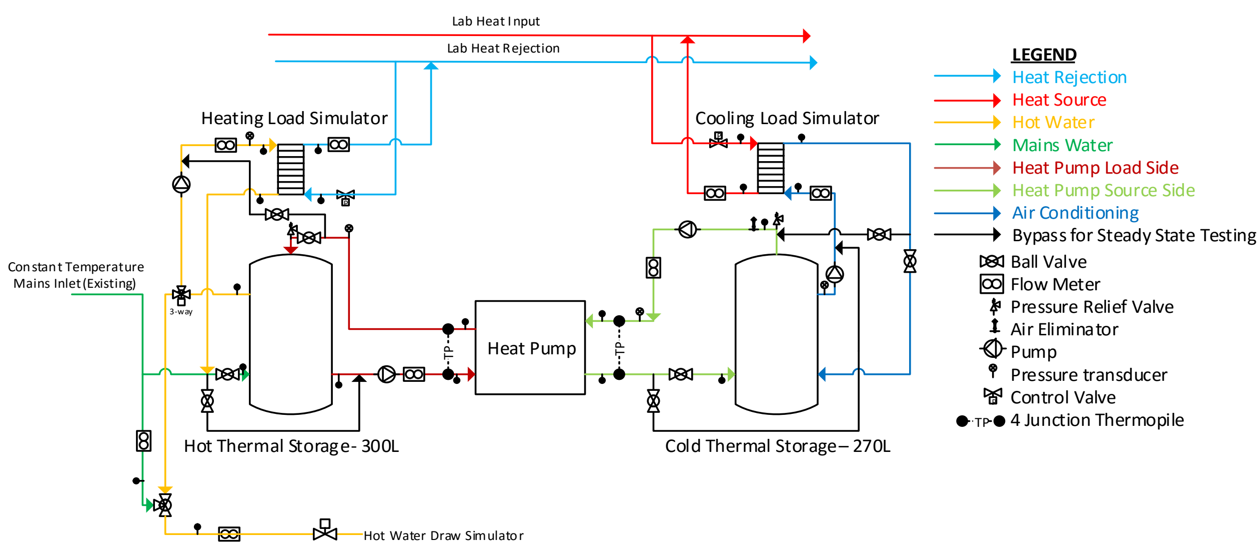

Figure 1: Schematic of experimental test set-up

Figure 1, with an image of the system shown in Figure 2.

The system was constructed in a lab facility at Carleton University with an existing heat input and heat rejection system. As such, this infrastructure formed the backbone of the experimental test set-up. The heat rejection system was connected through a heat exchanger to the hot thermal storage and load side of the heat pump, with a proportional control valve regulating the flow of cold water into the heat exchanger. This allowed both the heat pump to be tested without the thermal storage and for the thermal storage system to be cooled at the end of tests without having to dump copious amounts of water. Additionally, this heat exchanger will be used to mimic the heating load of the building in future tests where transient and long term testing will occur. This same arrangement was repeated with the cold thermal storage and source side of the heat pump with the lab heating system, allowing of steady state source heating temperature and reheating the cold thermal storage system between tests. Finally, this will also be used for future testing to mimic the cooling load of a house or building.

Figure 2: Experimental set-up with the cold thermal storage on the left and hot thermal storage on the right (before insulating pipes)

To control the temperature exiting the heat exchanger, a thermocouple was installed at the exit of heat exchanger on the experimental test set-up side. This temperature is of paramount interest as it also the entry temperature of the heat pump while testing in steady state mode and is the temperature of entering fluid when cooling or heating the thermal storage tanks. A PID loop was programmed into LabView, using the exiting fluid temperature as the control variable, and controlled the flow of hot or cold water entering the heat exchanger using a 2-10 V proportional control valve on the lab side of the heat exchanger. This allowed for the temperature to be kept constant as the heat transfer rates change within the heat pump. In addition to the temperature at the exit, the inlet temperature was also measured as well as the flow rate through the experimental side of the heat exchanger. This allows for future development to control not the exiting temperature but the heat transfer rate across the heat exchanger. This was designed for when long term testing is conducted using the test set-up, to mimic the heating and cooling loads of a building obtained through measurement of an actual house or simulation.

The inlet and outlet on both the source and load side of the heat pump were instrumented with T-type thermocouples to measure the inlet and outlet temperature, while the flow rate through both the source and load side of the heat pump were measured using a turbine flow meter. To measure the stratification in the tank, the sacrificial anode was removed and a 5/8" stainless steel pipe with one end welded closed was inserted. Thermocouples were installed at 10 cm intervals to measure the temperature at each location. The final piece of instrumentation that was installed was a power monitoring system to measure the power draw of the heat pump compressor.

HEAT PUMP MODEL

Before the cold tank could be modelled, a model of the heat pump had to first be developed. A nominal 6 $kW_{thermal}$ heat pump was used in this study, and connected to the thermal storage systems. To model the heat pump, TRNSYS Type 927 from the TESS libraries that models a liquid-to-liquid heat pump was utilized [12]. This component utilizes a performance map of the specific heat pump and linearly interpolates between the system's inputs, which includes the entering source and load temperature, as well as the flow rate of the load and source fluid entering the heat pump. The heat transfer rate out of the source fluid stream and the heat transfer rate into the load fluid stream, as well as the electrical consumption of the compressor is supplied for predetermined flow rates and temperatures and the component linearly interpolates for the heat transfer rates and electrical consumption.

The data contained in the performance map can either be taken from manufacture's data, or obtained through experimental evaluation of the performance of the heat pump at each condition. To assess the performance and stratification of the cold thermal storage, low source side flow rates are required. These flow rates are much lower than the design flow rates for the heat pump, which is intended to be used as a ground source heat pump, and as a result, the performance of the heat pump had to be obtained experimentally. Additionally, no performance data was provided with a 50/50 water glycol solution (by volume), and therefore experimental performance data was obtained when a water/glycol solution was used on the source side. All experimental tests and modelling studies were conducted using propylene glycol. A 50/50 water/glycol solution was chosen as the maximum compatibility with different heat sources was desired, with one possible heating source being solar thermal collectors. For the system to function with solar thermal collectors as the heat source, a 50/50 solution must be used to prevent possible freezing in the collectors.

Test were first performed using water on the source (cold) side of the heat pump with source and load flow rates of 3 L/min, 6 L/min, and 9 L/min, and entering source side temperatures of 10°C, 15°C, 20°C, and 25°C, while the load side entering temperatures were evaluated at 20°C, 25°C, 30°C, 35°C, and 40°C. In total, 180 tests were conducted. Each test consisted of setting the inlet temperature and flow rates, and once the inlet conditions on both the load and source side reached steady state, data was collected for 10 minutes at 30 second intervals. The average inlet and outlet source and load temperatures and the flow rates were then averaged and the heat transfer rate into the load fluid stream and out of the source fluid stream was calculated. Additionally, the average power consumption of the compressor was measured and averaged over the 10 minute period. The results of each test were then tabulated and a complete performance map was created.

As water was being used as the test fluid, if the outlet source temperature dropped below 3°C, the heat pump turned off. Additionally, the refrigerant temperature entering and leaving the condenser was monitored, and if the exiting refrigerant temperature was above 56°C, the test was stopped due to high pressure in the refrigerant line (this corresponds to ~515psi for the refrigerant being used, R-410a). As a result, all tests at a source flow rate of 3 L/min and entering temperatures of 10°C and 15°C failed, and many tests at a load entering temperature of 40°C also failed due to the high pressure limit. As the performance map must be complete to function within TRNSYS, these values were extrapolated from the remainder of the data.

This process was repeated using a glycol/water solution on the source side however tests were no conducted with a load side temperature of 40°C as the heat pump will not function, and an additional source side temperature of 5°C was added as much lower temperatures are possible when using a glycol solution. Once the performance map was completed, the heat pump model had to be validated to ensure accuracy and that any stratification discrepancies are a result of the tank model, and not a compounded error caused by the heat pump model.

This was completed by first running a number of experimental tests using the complete experimental test set-up. A constant flow rate on both the source and load side was set and the inlet temperature varied based on the temperature change in the tanks. The inlet and outlet temperature and flow rate of both the source and the load fluid stream, as well as the electrical power consumption were recorded at 30 second intervals. To compare the experimental performance of the heat pump and the results using the performance map in TRNSYS, a separate TRNSYS model that contains only the heat pump was developed. The inlet conditions to the heat pump (both temperatures and flow rates) were supplied from the experimental data as user supplied files. This ensured only the performance of the heat pump was compared and the experimental and modelled results for water on the source side were compared in Figure 3 for three separate flow conditions.

Based on these results it can be seen that the heat pump accurately models the actual performance of the heat pump. As such it can be concluded that any observed discrepancies in tank performance is a direct result of the tank model and not caused by the heat pump model. This procedure was then repeated using the glycol/water

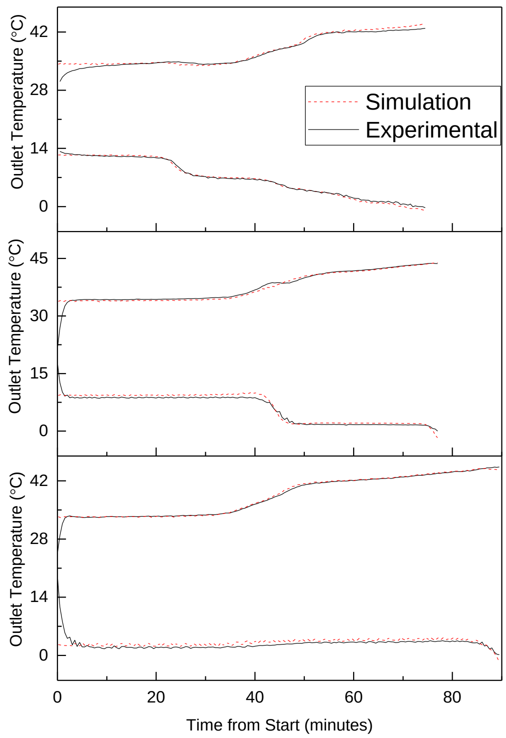

Figure 3: Comparison of simulated and experimental heat pump source and load outlet temperatures under 3 different source and load flow rates with water as the source fluid

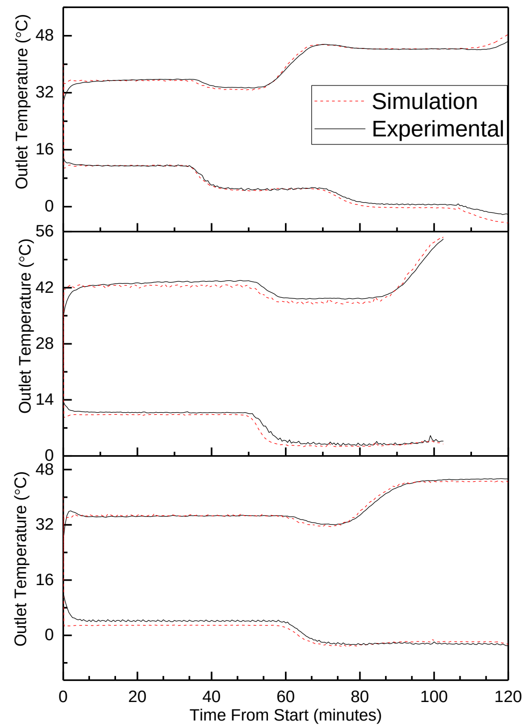

Figure 4: Comparison of simulated and experimental heat pump source and load outlet temperatures under 3 different source and load flow rates with a glycol solution as the source fluid

solution. The experimental and model results of three separate tests for the outlet temperatures and the power consumption are graphed in Figure 4.

Based on these results, it was determined that the heat pump model accurately predicts the performance of the heat pump and will provide the required accuracy to determine that any discrepancies in the simulation results of the thermal storage tank is as a result of the tank model, and not based on discrepancies between the inlet conditions when comparing experimental and modelled data.

THERMAL STORAGE TANK MODEL

TRNSYS contains two separate vertical thermal storage tanks (Type 4 and 60) in the standard package and an additional tank model is available through the TESS component library (Type 534) [11,13]. All three models use a similar method for determining the temperature profile into the tank, in which all heat transfer is assumed to be in the vertical axis of the tank. The tank itself is then broken down into equal height (and volume) segments. Each of this segments, commonly referred to as nodes, are considered fully mixed (constant temperature) and an energy balanced is performed on each node, at each time step to determine the temperature of the node. The stratification can be determined by plotting the temperature at each node over time.

After building and validating the performance maps to model the heat pump using both water and a glycol/water solution on the source side, a new model was constructed to simulate the heat pump and cold thermal storage tank. To ensure that the cold thermal storage tank can be properly validated, only the cold storage was modelled with the inlet conditions of the load side of the heat pump are supplied using a file input from experimental data. Additionally, the source side flow rate was obtained from experimental results and inputted to ensure no errors are caused as a result of small fluctuations in flow rate. The primary reason for this was the performance, and subsequently the source outlet temperature, is directly related to the inlet temperature and flow rate on the load side. As such, any discrepancy in the hot thermal storage tank model would carry through and cause discrepancies in the cold thermal storage tank temperature profile. Additionally, TRNSYS will only accept 100 differential equations, and as a result can only model a total of 100 temperature nodes across all tanks. As such, if two tanks are modelled, the number of nodes possible in the cold thermal storage tank is reduced.

Three separate models were constructed in TRNSYS using each of the three tank models available using the fluid properties of water, and the properties of 50/50 water/glycol solution (by volume). All other components were kept the same in each of the models to ensure no discrepancies can propagate as a result of different settings or fluid properties. Table 1 outlines the fluid properties used for both water and water/glycol solution, while Table outlines the parameters used within each tank model where applicable. Figure 5 shows the TRNSYS model as built using Type 534 as the cold thermal storage tank.

Table 1: Fluid properties for water and a 50/50 water/glycol solution by volume [14]

Fluid Property	Water	Water/ Glycol
Density (kg/m^3)	1000	1046.7
Thermal Conductivity (W/m·K)	0.594	0.378
Specific Heat (kJ/kg·K)	4.19	3.32
Viscosity (cP)	1	5
Thermal Expansion (1/K)	0.00026	0.000495

Table 2: Physical tank parameters used for all tank models

Tank Parameter	Water
Volume	270 L
Height	1.3 m
Number of Nodes	50
Entry Node	1 (bottom)
Exit Node	50 (top)
Auxiliary Heater	0 kW

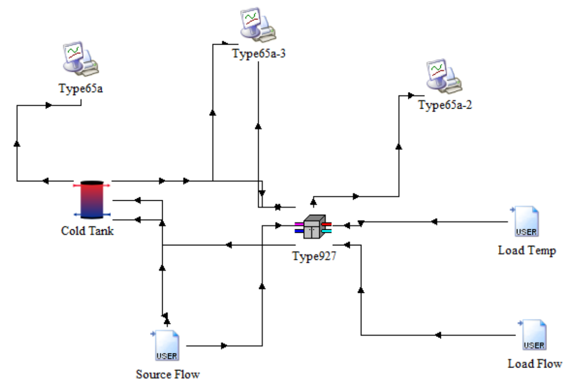

Figure 5: TRNSYS model containing data readers for the load conditions and source flow rate for tank Type 534

COLD THERMAL STORAGE RESULTS

To validate the applicability of the standard TRNSYS thermal storage tank models for use as a cold thermal storage tank with both water and a water/glycol solution, the results using Type 534 were first compared to experimental results. The primary purpose of the comparison was to determine whether the tank model could accurately predict the stratification present during a charging cycle of the cold storage. As such, the temperatures were measured at 13 equal increments through the tank, while the temperature at that level was also determined through simulation. The results were then plotted allowing the temperature profiles to be compared. This process was conducted three separate times, at three different source side flow rates, allowing the effect of flow rate on both stratification and the ability for the model to predict the temperature profile to be determined. As such, a low (3 L/min), medium (6 L/min) and high (9 L/min) source side flow rates were compared, with the results in Figure 6, Figure 7, and Figure 8, respectively. In all of these graphs, the left most line represents the bottom most measurement point moving up the tank to the right to the graph with the right most line representing the top measurement point. The first measured point is located 5 cm from the bottom of the tank, and in 20 cm increments with the top measurement being 125 cm from the bottom of the tank.

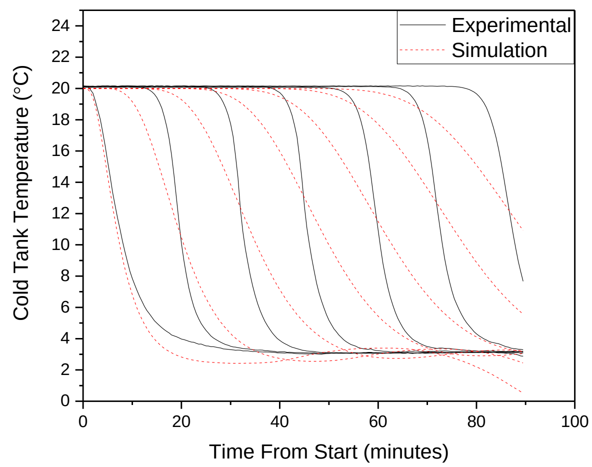

Figure 6: Tank temperature profile at a source flow rate of 3 L/min and a load flow rate of 3 L/min

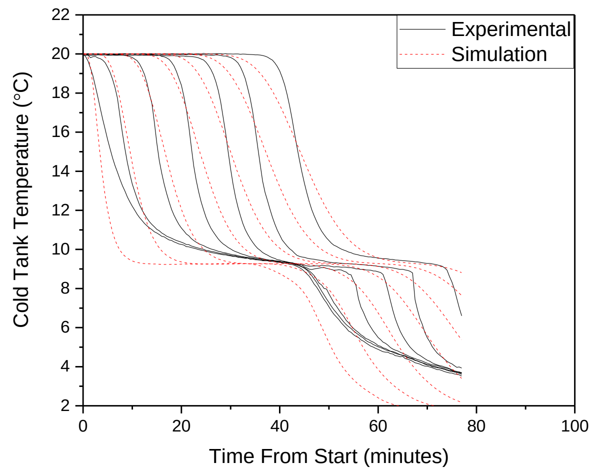

Figure 7: Tank temperature profile at a source flow rate of 6 L/min and a load flow rate of 6 L/min

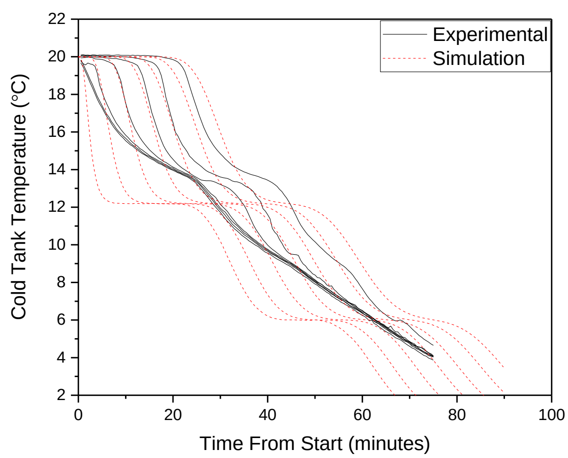

Figure 8: Tank temperature profile at a source flow rate of 9 L/min and a load flow rate of 6 L/min

From these figures, it can be see that as the flow rate increases, the experimental results show a rapid destruction of stratification starting at the bottom of the tank (where the inlet is located). Additionally, as the temperature of the tank storage medium decreases, the stratification seen in the tank also decreases and as this decrease in temperature occurs, the more the tank resembles a fully mixed tank. As a result, it was determined that the model accurately represents the temperature profile of the tank at low flow rates, and at higher temperatures, however the models fail to accurately predict at high flow rates and as the temperature approaches freezing.

This process was repeated, but the storage medium was replaced with a 50/50 water/glycol solution (by volume). The simulations were run using Type 534, however all the fluid properties were changed to those of the glycol solution. The simulations were run at a low (4 L/min), medium (5 L/min) and a high (7.5 L/min) source side flow rate. A slight change in flow rates from the water tests were used to further validate the models at different flow rates. The simulation and experimental results were plotted together for each set of test conditions and the results are show in Figure 9, Figure 10, and Figure 11, respectively.

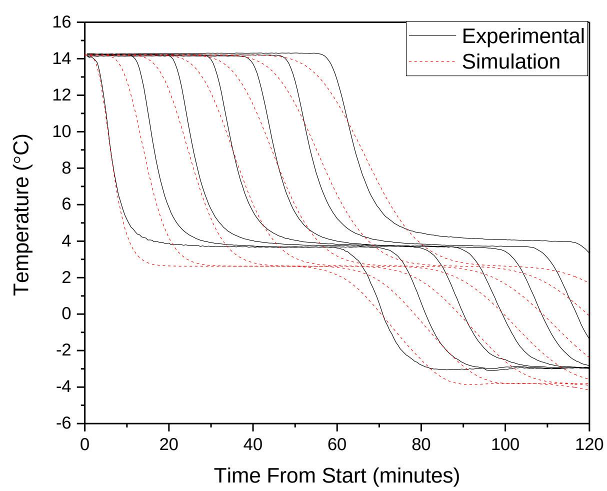

Figure 9: Tank temperature profile at a source flow rate of 4 L/min and a load flow rate of 4 L/min

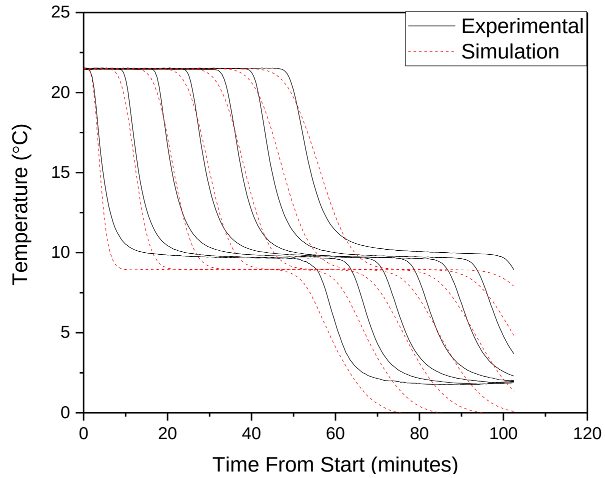

Figure 10: Tank temperature profile at a source flow rate of 5 L/min and a load flow rate of 3.5 L/min

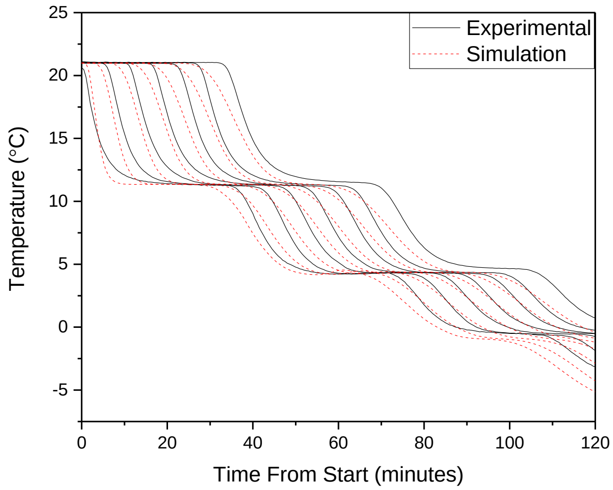

Figure 11: Tank temperature profile at a source flow rate of 7.5 L/min and a load flow rate of 5.5 L/min

Based on these results, it is shown that TRNSYS Type 534 correctly predicts the temperature distribution within a cold thermal storage system that uses a glycol/water solution as the storage medium. A small discrepancy is noted in the end temperatures, with this most likely resulting from small discrepancies seen in the source outlet temperatures of the heat pump. There is no change in the predictive capabilities of the model at higher flow rates or at lower tank temperatures as seen in the experimental results of the tank temperature profiles when using water as the storage medium.

The simulation and experimental results have almost a perfect agreement at the beginning of the test, however as the test continues, the level agreement between the experimental and simulation results slowly decreases. This is caused by the small errors introduced through each component and time step slowly building on each other and compounding, as a small difference in entering heat pump temperature can cause a larger error on the outlet temperature and then compounds within the thermal storage tank temperature profile. Although this slight disagreement is extremely minor in a 1.5-2 hour charging test run, this could become significant when looking at charging and discharging, or over longer test periods. Of particular importance is how this might impact the overall tank temperature profile and heat pump performance over an annual simulation, and this needs to be investigated further as part of future work.

NODE SENSITIVITY

Although both the water and the glycol/water storage tank models represent the overall tank distributions to varying degree of accuracies, all of the models have discrepancies in the rate of temperature change at any given point, with the simulation results typically changing at a slower rate than the experimental results. When modelling hot thermal storage tanks, this same phenomenon is also typically observed, and is the usually the result of the experimental measurement being a spot measurement while the simulation is the temperature of the node. This node temperature is over a larger volume of water (at 50 nodes for 270L, each node represents 5.4 L of storage medium), and as a result, can take a longer time to change temperatures. A node sensitivity study was conducted for one case with water as the storage medium and once case with the glycol/water solution as the storage medium. This node sensitivity served two purposes, being to determine what level of increased accuracy is observed with higher nodes, and whether the simulations can be conducted with fewer nodes, with little compromise to accuracy to increase the computational efficiency of the simulation. The results when using water are shown in Figure 12, while the results for the water/glycol solution are shown in Figure 13.

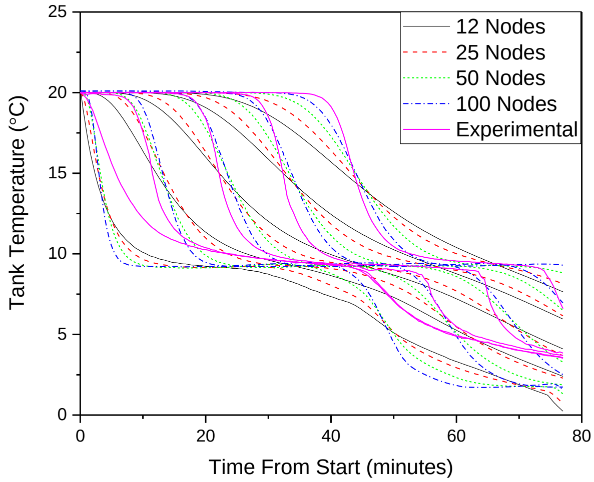

Figure 12: Node sensitivity analysis using a cold thermal storage tank with water as the storage medium

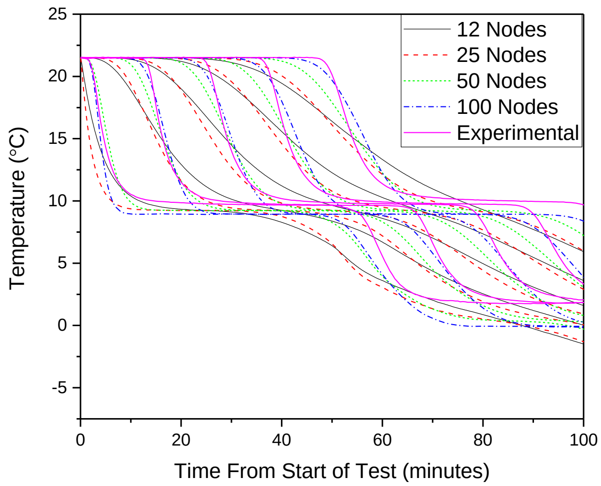

Figure 13: Node sensitivity analysis using a cold thermal storage tank with a 50/50 water/glycol solution as the storage medium

These graphs show the expected result where the more nodes in the tank, the quicker the rate of temperature change experienced at any given point. This is most evident when comparing the extremes of the simulation results using 12 nodes when compared to that using 100 nodes. There is also a significant volume per node difference, where a node represents 22.5 L when using 12 nodes, to 2.7 L when using 100 nodes. This confirmed that there is a significant improvement in the tank temperature profile as the number of tank nodes increased.

Although it was expected to see a significant difference between the two extremes, of more interest was the difference between 50 and 100 node cases. In both the water and water/glycol cases, there is an improvement going to 100 nodes from 50 nodes, however this improvement is small (particularly when looking at the glycol/water solution). As a result, it can be said that the 50 node thermal storage tank model has the necessary accuracy to model the cold thermal storage tank in the system of interest. This was an extremely important determination, as TRNSYS can only handle a total of 100 nodes across all tanks in the model. When the complete system will be modelled with both a hot and cold thermal storage tank, this allows each tank to be modelled with 50 nodes.

COMPARISON OF TANK TYPES

The final step in validating TRNSYS for modelling cold thermal storage tanks was to compare how the three standard tank models in TRNSYS compare. To completed this, the TRNSYS model using Type 534 was adopted and the tank replaced first with Type 60 and then with Type 4. All three tank models have the same basic mathematical principles, in which an energy balance is conducted over each tank node, factoring in both fluid flow into and out of the node, natural convection between node and the conductive heat transfer between the nodes. Although, all three have the same basic principles, slight variations in calculation methods are present between models. As such, each of the three models with each different tank were run for using both water and a water/glycol solution as the storage medium. The results were then plotted to compare the three different models, while the experimental results were also plotted to see if a certain Type better predicted the tank temperature profile. The results when using water as a storage medium are presented in Figure 14, while the results using a water/ glycol solution as the storage medium are presented in Figure 15.

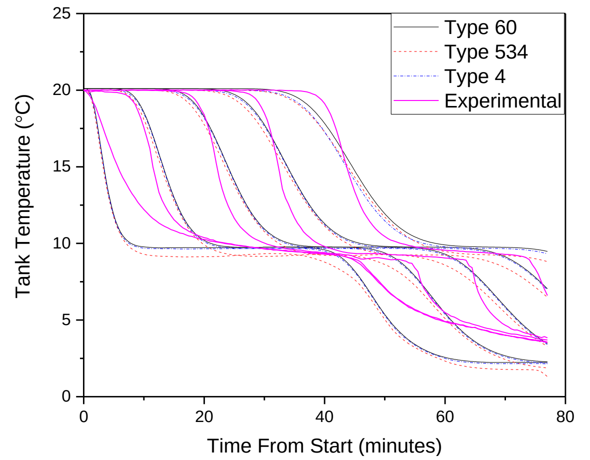

Figure 14: Comparison of TRNSYS Type 60, 534 and 4 when modelling a cold thermal storage tank using water as the storage medium

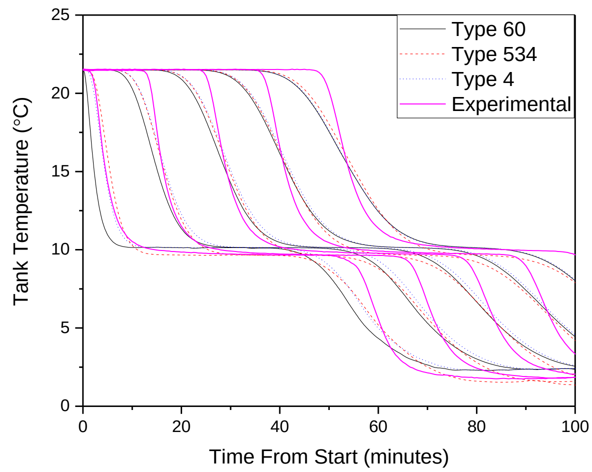

Figure 15: Comparison of TRNSYS Type 60, 534 and 4 when modelling a cold thermal storage tank using a water/glycol solution as the storage medium

Based on these results, for both storage mediums, all three tank models produce basically identical results, with little to no temperature deviation between each other. When compared to the experimentally obtained data, all three models did an adequate job predicting temperatures obtained through experimental measurement. As such, there is no performance benefit between any of the three models, and selecting which to use comes down to personal preference and the model in which the modelling of inlet and outlet, heating coils, and internal heat exchangers best represent the physical system.

CONCLUSIONS

This study developed a number of key outcomes pertaining to the modelling of a small scale, cold thermal storage tank using both water and water/glycol solution as the thermal storage medium. It was determined that the standard TRNSYS tank models have limited accuracy when modelling higher flow rates (>6 L/min) at low temperatures. The experimental results show that significant mixing occurs under these conditions, however the simulation results predicted a high degree of stratification. When modelling water based cold thermal storage tanks at low flow rates, an accurate prediction of thermal stratification can be obtained through simulation. When modelling a cold thermal storage tank using a water/glycol solution, it was found that the TRNSYS tank models provide an accurate prediction of the temperature distribution through the storage tank. This is valid over the complete range of flow rates and temperatures tested within this study.

The influence of the number of nodes used in the single tank was studied. It was determined that although the more nodes present, the better the simulation results agreed with the experimental data, the improvement in agreement from 50 to 100 nodes was minimal and that 50 nodes provide the required accuracy for this system. Finally the three different standard TRNSYS tank models were tested and compared to determine if one provided more accurate results when modelling the results. It was determined that all three models predict almost identical performance of the tank and there is almost not deviation in tank temperature from one type to another. As a result it was determined that Type 4, Type 60, and Type 534 have the same limitations as described above, and can be used interchangeably.

The results obtained through this study will form the backbone to modelling a cold thermal storage system paired with a liquid-to-liquid heat pump. This system will be further examined and modelled on an annual basis to optimize the cold thermal storage to allow for the reduction in peak loading, which in turn could lead to lower utility bills for the homeowner, and reduce the required transmission and generation capacity of the electrical grid.

This study was the first phase in a multi-step project to develop residential scale cold thermal storage systems for use in peak load reduction and time shifting. The next step is to implement the hot thermal storage system and model and experimentally evaluate the complete system over longer periods of time, under both charging and discharging conditions. From there the complete system will be implemented in a complete house model and the system will be optimized and control strategies developed to significantly reduce peak electrical consumption.

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