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Mary Bess Wilson

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Design and analysis of an increased thermal capacitance and thermal storage
management (ITC/TSM) system

By

Mary Bess Wilson

A Dissertation
Submitted to the Faculty of
Mississippi State University
in Partial Fulfillment of the Requirements
for the Degree of Doctor of Philosophy
in Mechanical Engineering
in the Department of Mechanical Engineering

Mississippi State, Mississippi

May 2019

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2019

Design and analysis of an increased thermal capacitance and thermal storage
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In this dissertation, an increased thermal capacitance (ITC) and thermal storage management (TSM) system was simulated to reduce building energy consumption, specifically energy related to heating, cooling and domestic hot water. An increased thermal capacitance allows phase shift and amplitude reduction of heat flow fluctuations associated with the building's internal temperature response due to weather. An adaptive allocation and control of the added capacitance through TSM significantly improves the benefits of the extra capacitance. This dissertation was conducted in three parts: (1) a first-order analysis of the ITC/TSM applied to a micro-building; (2) a transient simulation of the ITC/TSM with PCM implementation for tank volume control; and (3) a parameter study on the ITC/TSM system with added complexities such as the inclusion of DHW and a multiple story residential building. The first-order analysis was used for transient simulation comparison, as simple models are much more suitable for real time implementation in actual control systems. A first order study on a small residential building is also used to establish the merit of the ITC/TSM concept before integrating into a more complex analysis. This study determined that the ITC/TSM could potentially provide savings but required a very large thermal mass.

The ITC/TSM system was then coupled with phase change materials (PCMs), which enable thermal energy storage volume reduction. The transient energy modeling software, TRNSYS, is used to simulate the building's thermal response and energy consumption, as well as the ITC/TSM system and controls. Four temperature-controlled operating regimes are used for the ITC/TSM operations: building shell circulation, heat exchanger circulation, solar panel circulation, and storage. After this, 125 simulations were conducted to design and optimize the ITC/TSM. The three parameters of interest were: tank volume size, solar panel size, and mass flowrate. Domestic hot water usage was also included as another energy savings opportunity. Results for the parameter study showed that savings are optimized when the solar panel and the hot water tank are sized together. If they are not sized simultaneously, the temperature of the large thermal capacitance is not adequately controlled. For all simulations conducted in the parameter study, the building energy usage was reduced significantly.

DEDICATION

To my parents who taught me first.

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TABLE OF CONTENTS

DEDICATION.....	ii
ACKNOWLEDGEMENTS.....	iii
LIST OF TABLES	vi
LIST OF FIGURES	viii
CHAPTER	
I. INTRODUCTION.....	1
1.1 Current State of Energy Consumption.....	1
1.2 Research Objectives.....	2
1.3 Literature Review	3
1.3.1 First Order Approximations used for Building Energy Calculations	3
1.3.2 Passive Methods for Energy Consumption Reduction	4
1.3.3 Active Thermal Mass for Energy Consumption Reduction	5
1.3.4 Phase Change Materials and Solar Energy for DHW Energy Reduction	7
II. A FIRST-ORDER STUDY OF THE ITC/TSM SYSTEM IN A MICRO-BUILDING...9	
2.1 Models.....	9
2.1.1 Reference Model.....	10
2.1.2 Increased Thermal Capacitance and Thermal Storage Management Model.16	
2.1.2.1 Verification of the Time Constants Using TRNSYS	18
2.1.2.2 Operation of ITC/TSM System.....	20
2.1.3 Cooling and Heating	25
2.2 Results and Discussion	25
2.2.1 Sensitivity Analysis	28
2.2.2 Economic Analysis	29
2.3 Conclusions	31
III. TRANSIENT CASE STUDY OF THE ITC/TSM SYSTEM	33
3.1 Reference Model	34
3.1.1 Building Specifications and Location.....	34
3.1.2 Heat Pump Parameters	37
3.2 ITC/TSM Model with PCM	38

3.2.1	Verification of PCM Tank Operation	45
3.2.2	TRNSYS Solar Panel Selection.....	46
3.2.3	Verification of Building Thermal Time Constant Increment	47
3.2.4	Energy Balance	48
3.2.4.1	Water Tank	49
3.2.4.2	Solar Panel.....	49
3.3	Results.....	50
3.4	Conclusions	53
IV.	DESIGN AND OPERATION OF THE ITC/TSM SYSTEM.....	55
4.1	Model Description	56
4.1.1	Reference Building Specifications and Location	56
4.1.2	Proposed ITC/TSM System.....	63
4.2	Results and Discussion	72
4.3	Conclusions	83
V.	SUMMARY AND CONCLUSIONS.....	85
	REFERENCES	89

LIST OF TABLES

Table 2.1	Building properties for a single room with brick cavity brick walls.	12
Table 2.2	Constant operating temperature for the temperature node T_{AC}	12
Table 2.3	Upper and lower dead-band operation temperatures for $T_{Building}$	12
Table 2.4	Sensitivity analysis results.	29
Table 2.5	Average retail price of electricity to residential sector, July 2014–2010 [54].	30
Table 3.1	Building Materials and Properties.	35
Table 3.2	Ground Coupling Properties.	36
Table 3.3	Heat Pump Parameters.	37
Table 3.4	Heat Pump Nominal Thermostat Settings.	38
Table 3.5	Tank Circulation Regime. HEX: heat exchanger.	40
Table 3.6	PCM Parameters.	45
Table 3.7	Solar Panel Parameters for Solar Rating and Certification Corporation (SRCC) Data Sheet.	47
Table 4.1	Building Shell Materials [63].	57
Table 4.2	Soil Temperature Properties [65]	59
Table 4.3	Heat Pump Parameters [66]	61
Table 4.4	Heat Pump Thermostat Settings.	62
Table 4.5	Solar Panel Parameters	66
Table 4.6	PCM Parameters [61]	67
Table 4.7	Percentage Flowrate for Surfaces.	69
Table 4.8	ITC/TSM Valve Operating Regime	71

Table 4.9	Parametric Study Range	71
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LIST OF FIGURES

Figure 2.1	Reference building circuit.....	13
Figure 2.2	Air conditioning example from July 21st.	15
Figure 2.3	ITC/TSM building circuit	17
Figure 2.4	Trendline used to calculate the time constant for the TRNSYS simulation. .	20
Figure 2.5	Flowchart for state variable equations for the ITC/TSM system.	24
Figure 2.6	Energy required for reference building and ITC/TSM building	26
Figure 2.7	The difference in required energy and the percent energy reduction.	27
Figure 2.8	Initial investment amount (USD) per payback period.....	31
Figure 3.1	Transient System Simulation Tool (TRNSYS) Schematic for Reference Building	34
Figure 3.2	TRNSYS schematic for the increased thermal capacitance (ITC)/ thermal storage management (TSM) model.	39
Figure 3.3	Numerical Approximation of Specific Heat for Phase Change Materials (PCM).	42
Figure 3.4	Graphical Representation of Energy Flows into a Node.	44
Figure 3.5	PCM Phase Change Operation.....	46
Figure 3.6	Building Free-Response.....	48
Figure 3.7	Energy Consumption.	51
Figure 3.8	Energy Reduction.	52
Figure 4.1	Building Model in SketchUp.	56
Figure 4.2	Reference TRNSYS Schematic.....	59
Figure 4.3	DHW Draw Profiles	60

Figure 4.4	Lighting and Occupancy Schedule.....	61
Figure 4.5	ITC/TSM TRNSYS Schematic	64
Figure 4.6	Flowchart for ITC/TSM Operation	65
Figure 4.7	Pipe Dimensions in Building Shell	69
Figure 4.8	Constant Flowrate: Energy Consumption vs Tank Volume and Solar Panel Area	73
Figure 4.9	Flowrate and Solar Panel Size vs Percent Savings for Constant Tank Volume	75
Figure 4.10	Building Temperature vs. Flowrates on January 21 st	76
Figure 4.11	Building Temperature vs. Flowrates on July 21 st	76
Figure 4.12	Constant Tank Volume: Energy Consumption vs. Mass Flowrate and Solar Panel Area.....	77
Figure 4.13	Tank Volume and Solar Panel Size vs Percent Savings for Constant Mass Flowrate	79
Figure 4.14	Constant Solar Panel Area: Energy Consumption vs. Mass Flowrate and Tank Volume.....	80
Figure 4.15	Solar Panel Size and Mass Flowrate vs Percent Savings for Constant Tank Volume	82

CHAPTER I

INTRODUCTION

1.1 Current State of Energy Consumption

Currently, in the U.S. residential sector, 46% of electricity consumption is allocated to space cooling (17%), space heating (15%) and water heating (14%) [1]. These values also do not include the energy required to operate air handlers, ceiling fans, humidifiers, and dehumidifiers, which are also implemented for building comfort control. Maintaining thermal comfort of a building is not only a financial burden to residents, but also a burden for power producing plants to deliver the required load during peak operating times. This dissertation is focused on regions of high impact, therefore the South Atlantic portion of the United States (US) is of interest. In this region energy consumption associated with space cooling, space heating and water heating is 13% higher than the national average. Reducing energy consumption not only has a positive effect on the resident through decreased operating costs, it also has a possible environmental impact by decreasing dependence on natural resources such as coal and natural gas. Many government incentives provide tax credits to reduce the carbon footprints of residential homes. As an example, consider California's Zero Net Energy initiative, whose goal is that all new residential buildings produce the same amount of energy consumed per year [3]. This dissertation aims to assist in the goal of reducing carbon emissions and reducing the dependency on fossil fuels specifically in the residential sector.

1.2 Research Objectives

The main objective of this research is to reduced building energy consumption associated with heating, cooling and DHW systems by simulating and implementing an ITC/TSM subsystem. To achieve this goal the following objectives were examined:

1. Determine merit of concept for the ITC/TSM system
 - a. Develop a first order model of the ITC/TSM system to determine the governing aspects of a building's thermal response
 - b. Determine savings associated with the ITC/TSM model by comparing to a first order model without the ITC/TSM system.
 - c. Determine initial investment to implement the ITC/TSM system for different payback periods.
 - d. Determine of the influence of location on the ITC/TSM system
2. Develop a transient model of the ITC/TSM system
 - a. Develop model for small/medium residential building using materials common in the southeastern part of the US.
 - b. Inclusion of additional weather data such a radiation and humidity
 - c. Develop and verify a tank module to include phase change materials for tank volume reduction.
 - d. Include a solar panel loop for tank temperature control
 - e. Determine savings associated with the ITC/TSM model by comparing to a transient model without the ITC/TSM system.
3. Design Improvement of the ITC/TSM system
 - a. Inclusion of the single family residential prototype developed by PNNL as the building model
 - b. Inclusion of DHW and heat gains associated with lighting and occupancy
 - c. Redesign of ITC/TSM to include a cold water storage tank and a ground heat exchange for tank temperature control
 - d. Perform a parameter study over the solar panel size, tank volume and mass flowrate to maximize savings and reduce initial investment cost.

1.3 Literature Review

The purpose of this section is to discuss relevant research pertaining to increased thermal mass systems and the obstacles of these systems.

1.3.1 First Order Approximations used for Building Energy Calculations

Chapter one of this dissertation contains a first order study of the ITC/TSM in a micro-building. Micro homes are very small houses, typically less than 18.56 m^2 (200 ft^2) and can be as small as 9.29 m^2 (100 ft^2) [4], [5]. The building selected for the study in this chapter is comparable to such micro homes. Although the size of the building in this study is small in comparison with standard residential dwellings, it serves as a good start to establish the merit of the ITC/TSM concept. Once the results for the ITC/TCM were established for a micro home, and lessons were learned regarding implementation of the ITC/TCM, this research was adapted and extended to residential dwellings of standard sizes. Furthermore, a micro home lends itself well to first-order transient modeling, using a simple thermal lumped capacitor/resistance model. This helped lead to a better understanding of how to design the TSM without dwelling into the many complexities of a comprehensive and exhaustive building simulation [6], [7]. Although a first-order thermal model did not account for all features of a standard residential building, first-order transient approximations are commonly used to provide improved assessment of the dynamic performance of a building over pure static models [6], and are often used to understand the main aspects governing the response of a system that are not necessarily apparent from detailed and complex computational models. In addition to the above, simple models are much more suitable for real time implementation in actual control systems.

1.3.2 Passive Methods for Energy Consumption Reduction

Methods for providing thermal comfort with no or little additional cost are often referred to as passive methods. Passive energy saving methods comprise a large variety of implementations; examples of such are building orientation and shading[8], and indirect and direct heat storage. In indirect heat storage, gains can come from concepts such as a Tromba wall, water wall, or solarium, while in direct heat storage, heat enters the building through windows and walls [9]. These methods are limited to natural modes of heat transfer such as free convection and conduction. Passive methods also use heavy wall constructions, or large interior thermal mass to achieve an increased thermal capacitance for the building [10], [11]. Shaviv et al. [12] studied the passive influence of thermal mass in hot humid locations by performing simulations using four variations of building materials. They found that buildings with a large thermal mass were able to reduce the internal building temperature by 3–6 °C without air conditioning (AC) operation. In the early 60s, increased capacitance was part of a solar building plan through the study of geodesic domes [13], [14]. However, the use of thermal management was not combined with thermal capacitance. Passive methods such as earth coupling, where buildings are constructed to benefit from consistent temperatures below the surface of the earth [15], can have a negative influence on building operating costs. This is because of the permanent additional capacitance during seasons when minimal capacitance is optimal. An increased time constant associated with the added thermal capacitance can be detrimental to the operation of an air conditioning (AC)/heating system in situations when the temperature of the fixed capacitance remains above or below the thermal comfort specifications for a prolonged period of time.

Research using building thermal mass for energy reduction has been well documented [19]–[30]. The thermal inertial of a building is one of the controlling factors for the phase shift

and amplitude reduction of “heat flow fluctuations” associated with the building’s internal temperature response due to weather [24]. One often sought benefit of increased thermal mass is the ability to shift peak usage to a time of off peak power consumption and thus purchase the require building electricity at a reduced rate [18], [19]. The research presented in this dissertation does not account for peak shift savings, since the focus of this study is the reduction of energy dependence not just a shift of usage time. Though this could be an additional benefit, this study does not implement it as the primary method of cost savings. Aste et al. [25] found that buildings with external walls of a high thermal capacitance could results in lower heating and cooling demand, with savings of around 10% for heating and 20% for cooling. However, Reilly et al.[21] found that for cold climates, a large thermal mass could be a hindrance, and should be coupled with preheating strategies. Simulations by Slee et al. [26] showed an exponential relationship between thermal capacitance and the fluctuation of the daily internal building temperature. This illustrates that there is point at which increasing the thermal mass for a specific system would not provide any addition in benefits. The study presented in this dissertation uses an active mode of increased thermal capacitance along with temperature-driven controllers to manage the thermal capacitance.

1.3.3 Active Thermal Mass for Energy Consumption Reduction

Active thermal energy storage methods, such as the one described in this dissertation, are those that require additional energy input to operate. There are numerous ways to implement an active thermal mass [28], such as integration with the building shell [29]–[31], the heating, ventilation and air conditioning (HVAC) system [11], [30], and through water storage tanks [29], [30]. Using transient simulations for a one-room enclosure incorporating an increased thermal capacitance (ITC) system, Carpenter et al. [32] determined that the most beneficial placement of

the additional capacitance is the ceiling, i.e., to circulate water from a tank through piping embedded in the ceiling. However, the energy savings due to their ITC methodology were rather modest, not taking full advantage of the weather conditions. In order to improve on the work of Carpenter et al.[32], a more advanced thermal management system (TSM) is proposed in this dissertation. The proposed ITC/TSM takes advantage of daily maximum or minimum ambient temperatures, radiation, and heat exchangers to control the temperature of the water used in the ITC. The idea is to operate the ITC in such a way that the additional capacitance can be disengaged from the building envelope when the operation is unfavorable to the energy consumption of the building. A review performed by Navarro et al. [28] found benefits associated with active methods for thermal energy storage (TES), but also stated that thermal water tanks have limited potential due to the impractical amount of space required. This study aims to address this issue by introducing phase change materials (PCM) to a water storage tank to reduce tank size with only small reductions to heat storage capacity. Kong et al. [33] and Whiffen et al. [34] both conducted experiments showing the validity of integrating PCMs for increased thermal mass, Kong et al. through passive PCM wallboards, and Whiffen et al. through an active hollow slab with embedded PCM. Whiffen et al. were able to show a delay in AC onset by 1.2 h. The impact of PCM on building energy consumption has also been studied using numerical simulations using software such as the Transient System Simulation Tool (TRNSYS [35])[36]–[39], which was also used in this paper’s study. Chen et al. [31] coupled thermal storage, through a ventilated concrete slab, with a building-integrated photovoltaic-thermal (BIPV/T) system to increase the overall efficiency of the system. Often phase change materials (PCMs) are used in these applications for their latent heat storage capabilities. Navarro et al. [30] conducted experimentations using PCM enclosed in tubes in a hollow concrete slab for thermal

storage and found the potential for the design. However, they noted that a control system is needed to manage the charging and discharging of the latent heat storage system. The study contained in this dissertation aims to provide a control scheme for utilizing the thermal storage through temperature and seasonal controllers. Jin et al. [29] conducted simulations in which two PCM layers are implemented in a radiant floor system. One layer has a melting point used during cooling applications and the other has a melting point for heating applications. They found a significant increase in “energy released by the floor” (41% during heating and 38% during cooling) compared to a radiant floor system without PCM.

1.3.4 Phase Change Materials and Solar Energy for DHW Energy Reduction

The coupling of domestic hot water (DHW) systems with PCMs for the use of latent heat energy storage and tank volume reduction is another point of interest for research [40]–[43]. Because a large thermal capacitance is needed for the building in this study, PCMs offer a unique opportunity to reduce the volume of the water tank without losing energy storage capability. Kousksou et al. [43] found that without proper selection of PCM parameters such as melting point, integration of PCM into DHW system are not significantly beneficial. Preliminary studies for the selection of the melting point of PCMs to use in the cold water storage and the hot water storage are considered to increase benefits. Solar domestic hot water systems is another common focus of research [44]–[47]. In this study, solar panels are implemented for temperature control of the increased thermal capacitance, however incorporating DHW into the simulation is a natural step to increase the overall savings of the system.

Three different ITC/TSM configurations are discussed in Chapter II-IV. Chapter II discusses a first-order analysis of an ITC/TSM system, Chapter III discuss a transient model of

the ITC/TSM with added complexities, and Chapter IV discusses the design and operation of the ITC/TSM with integrated domestic hot water.

CHAPTER II

A FIRST-ORDER STUDY OF THE ITC/TSM SYSTEM IN A MICRO-BUILDING

This chapter uses a first-order approximation of a micro-building to investigate the major factors determining how increased thermal capacitance (ITC) with thermal storage management (TSM) can reduce energy consumption in locations with relatively mild weather conditions such as the southeastern part of the United States of America. ITC is achieved through water circulation between a large storage tank and pipes embedded within the building envelope. Although ITC results in a larger dominant time constant for the thermal response of a building, an adaptive allocation and control of the added capacitance through TSM significantly improves the benefits of the extra capacitance. This chapter compares two first-order models for a micro-building: a reference case model with a single lumped thermal capacitance associated with the building, and another model, with the building's capacitance plus the capacitance of the water system. Results showed that the ITC/TSM system reduced the cost of conditioning the building by reducing the operating time of both the cooling and the heating systems. May through September, the air conditioning operating time was reduced by an average of 70%, and October through April, the operation of the heating system was reduced by an average of 25%..

2.1 Models

Two simple thermal circuits were developed for comparison purposes: a reference model with the capacitance of the building, and a second model featuring the ITC/TSM. Each of these circuits is correspondingly modified to account for warm season operation and for cold season

operation. During the warm season, a simulated AC system is used, while during the cold season a simulated heating system is used.

2.1.1 Reference Model

The RC thermal circuit for the reference model takes into account the lumped thermal capacitance for an 11 m² (118.4 square feet) single room dwelling with 2.46 m high ceiling, interior and exterior brick walls, concrete flooring, and a metal roof with insulation. Using a small sized building offers opportunities for construction and experimentation after all computational analyses are completed. Constructing a typical residential home for experimentation would not be as feasible as the building mentioned above. Once the experimental building is constructed and any discrepancies in the proposed model are rectified, the simulation can be expanded to apply to typical residential homes. The first-order differential equation seen in Equation 2.1 was used to determine the relationship between the time constant of the building, $\tau_{Building}$, and the building properties as follows:

$$\tau_{Building} \times \frac{dT_{Building}}{dt} + T_{Building} = T_{amb}(t) \quad (2.1)$$

where $T_{amb}(t)$ is the ambient temperature, which is the hourly dry bulb temperature collected by the National Renewable Energy Laboratory for Jackson, MS [48], and $T_{Building}$ is the internal temperature of the building.

The time constant of the building seen in Equation 2.2 is found by multiplying the envelope resistance, $R_{Building}$, by the thermal capacitance of the building, $C_{Building}$. The envelope resistance and thermal capacitance values were obtained from Lombard and Mathews [6].

$$\tau_{Building} = R_{Building} \times C_{Building} \quad (2.2)$$

Preliminary simulations that were run in this study show that ITC is more beneficial when the piping for the ITC is placed close to the interior of the room. Following this idea, the thermal resistance of the building envelope is divided in two, resulting in an interior envelope and exterior envelope with piping for the ITC between these two regions. The exterior envelope, Equation 2.3, was selected to account for 90% of the thermal resistance, and the interior envelope, Equation 2.4, accounted for 10%, indicating that the water piping is closer to the interior side of the building.

$$R_1 = R_{Exterior\ Envelope} = \frac{9}{10} \times R_{Building} \quad (2.3)$$

$$R_2 = R_{Interior\ Envelope} = \frac{1}{10} \times R_{Building} \quad (2.4)$$

Although the reference model does not take advantage of the ITC, for comparison purposes, resistances for the exterior and interior envelopes are used in the reference model. The thermal circuit for the reference model is shown in Figure 1. R_1 and R_2 account for exterior and interior lumped thermal resistances, respectively. The AC system is modeled as room air circulating through constant-temperature cooling or heating coils. The resistance R_{AC} accounts for the net thermal resistance between the coil temperature and the temperature of the room air circulating through the coils. R_{AC} was selected by sizing an AC unit for an 11 m² building. A 2.64 kW (9000 Btu/h) unit was selected for this building [49]. Multiple air conditioner sizes were considered when calculating the AC resistance value, however larger units did not provide any considerable benefits. A 4.4 kW unit only affected the required energy of the AC subsystem by 3% and an 800 W unit increased the required energy by only 4%. R_{AC} resistance variations are explored in the sensitivity analysis discussed later in Section 2.2.2. The building properties used in the simulations are presented in Table 2.1. Details for the AC/heating subsystem are given in

Section 2.1.3, and operating conditions and temperatures for the AC/heating subsystem can be found in Tables 2.2 and 2.3.

Table 2.1 Building properties for a single room with brick cavity brick walls.

Building Properties	Values
Outer envelope resistance (R_1)	2.6×10^{-2} K/W
Inner envelope resistance (R_2)	2.9×10^{-3} K/W
HEX resistance (R_{HEX})	5.4×10^{-4} K/W
AC resistance (R_{AC})	2.7×10^{-3} K/W
Building capacitance ($C_{Building}$)	4×10^6 J/K
Water tank capacitance (C_{Tank})	2.4×10^7 J/K
Building time constant	32 h

Table 2.2 Constant operating temperature for the temperature node T_{AC} .

Operating Temperature	(°C)
Cooling	13
Heating	82

Table 2.3 Upper and lower dead-band operation temperatures for $T_{Building}$.

Operating Temperature	Cooling (May–September)	Heating (October–April)
8 am–8 pm on temperature	23 °C	22 °C
8 am–8 pm off temperature	21 °C	24 °C
8 pm–8 am on temperature	22 °C	22 °C
8 pm–8 am off temperature	20 °C	21 °C

Figure 2.1 Reference building circuit

The transient model consists of a state variable equation for a first-order lumped parameter system. The state variable, $T_{Building}$, represents the temperature of the lumped thermal capacitance of the building. A variable S is used as a flag to determine the operation of the AC/heating system. When S is equal to 1, the AC/heating subsystem is active, and when S is equal to 0, the AC/heating subsystem is inactive. Equation 2.5 shows the state variable equation for the internal temperature of the building, $T_{Building}$.

$$\frac{dT_{Building}}{dt} = \frac{1}{C_{Building}} \times Q_{Building} \quad (2.5)$$

where $Q_{Building}$ is the heat transfer through the building.

Depending on the season, i.e., warm season vs. cool season, the equation for $Q_{Building}$ varies; Equation 2.6 is used in the warm season requiring cooling, while Equation 2.7 is used in the cool season requiring heating. In both equations $T_{cooling}$ and $T_{heating}$ are constant set point temperatures and the rest of the temperatures are time-varying.

$$Q_{Building} = \begin{cases} \frac{T_{amb}-T_{Building}}{R_1+R_2} - \frac{T_{Building}-T_{cooling}}{R_{AC}} & \text{if } S = 1 \\ \frac{T_{amb}-T_{Building}}{R_1+R_2} & \text{otherwise} \end{cases} \quad (2.6)$$

$$Q_{Building} = \begin{cases} \frac{T_{amb}-T_{Building}}{R_1+R_2} - \frac{T_{Building}-T_{heating}}{R_{AC}} & \text{if } S = 1 \\ \frac{T_{amb}-T_{Building}}{R_1+R_2} & \text{otherwise} \end{cases} \quad (2.7)$$

The flag S was used in simulating the more realistic scenario of dead-bands in the AC and heating systems. The AC system begins to operate once the building temperature reaches the upper dead-band limit, T_{Coolon} . Once the temperature decreases below the upper dead-band limit, the AC system continues to operate while inside the dead-band region. Once the temperature reaches the lower limit, $T_{Cooloff}$, the AC will shut off and allow the ambient temperature to increase the building temperature until the upper limit is reached again. Conversely, when using heating during the cool season, the simulation will allow ambient temperatures to lower the temperature of the building until the lower dead-band temperature limit, T_{Heaton} , is reached, then the heating will turn on and increase the building temperature until it reaches the high dead-band temperature limit, $T_{Heatoff}$, and the heating turns off. An example of the AC operating with a dead-band for 21 July can be seen in Figure 2.2. In addition, the ambient, building, cool on, and cool off temperatures are also illustrated in this figure. The dead-band shifting at 8 am and 8 pm is to simulate a typical residential building where the occupant desires to save energy and allows the building to reach a higher temperature while it is unoccupied. Some differential equation solvers do not allow to track time-varying variables other than state variables. For this reason, in order to track the value of S , a state variable equation for S was developed assuming that the simulation time step is constant. The state variable equation for S during the warm season and the cool season can be seen below in Equations 2.8 and 2.9, respectively. The equations were created such that the value of S can change between 0 and 1 within a single time step of Δt .

Figure 2.2 Air conditioning example from July 21st.

$$\frac{dS}{dt} = \begin{cases} \frac{-S+1}{\Delta t} & \text{if } T_{Building} \geq T_{Coolon} \\ \frac{-S}{\Delta t} & \text{if } T_{Building} \leq T_{Cooloff} \\ 0 & \text{otherwise} \end{cases} \quad (2.8)$$

$$\frac{dS}{dt} = \begin{cases} \frac{-S+1}{\Delta t} & \text{if } T_{Building} \leq T_{Heaton} \\ \frac{-S}{\Delta t} & \text{if } T_{Building} \geq T_{Heatoff} \\ 0 & \text{otherwise} \end{cases} \quad (2.9)$$

For all the simulations presented in this paper, Δt is 0.2 minutes, i.e., there are 5 data points of the building temperature and the state of the AC taken every minute.

A third state variable equation was added to integrate the heat rate from the AC, Q_{AC} , such as to determine the total energy extracted, E_{AC} , by the AC subsystem. Similarly, during the cool season, the total energy added, E_{Heat} , by the heating subsystem is obtained by integrating the

heat rate from the heater, Q_{Heat} . The state variable equations for each case are shown below in Equations 2.10 and 2.11.

$$\frac{d}{dt} E_{AC} = Q_{AC} = \begin{cases} \frac{T_{Building} - T_{cooling}}{R_{AC}} & \text{if } S = 1 \\ 0 & \text{otherwise} \end{cases} \quad (2.10)$$

$$\frac{d}{dt} E_{Heat} = Q_{Heat} = \begin{cases} \frac{T_{heating} - T_{Building}}{R_{AC}} & \text{if } S = 1 \\ 0 & \text{otherwise} \end{cases} \quad (2.11)$$

In summary, the system of first order ordinary differential equations, i.e., state variable equations, solved during the cooling season is composed of Equations 2.6, 2.8, and 2.10. Conversely, the state variable equations solved during the heating season are composed of Equations 2.7, 2.9, and 2.11. To avoid errors in selecting the initial building temperature, numerical data was collected only after the first 24 h of simulation, once the effects of the initial conditions subsided.

2.1.2 Increased Thermal Capacitance and Thermal Storage Management Model

A second circuit was developed to study how the ITC/TSM system can reduce the operating time for the AC system and the heating system. The ITC/TSM as described in Figure 2.3, has the same lumped thermal capacitance of the building presented in Section 2.1.1, but, in addition, copper pipes carrying water are embedded between the inner and outer building envelopes. A heat exchanger exposed to ambient temperatures was also added to improve the temperature of the water in the tank whenever favorable. For instance, during warm seasons, the water will circulate through the external heat exchanger if the ambient temperature is lower than the temperature in the water tank, and during the cold seasons the water will circulate through the heat exchanger when the ambient temperature is higher than the water tank temperature.

Figure 2.3 ITC/TSM building circuit

This thermal storage system saves energy for the entire system by taking advantage of the climate. Using the properties of copper piping found in [32] and the building envelope properties found in [6], it was found that the envelope resistance is 200 times greater than the resistance of the copper pipe; therefore, the copper piping's contribution to the net resistance will be omitted in this first order analysis. As with the reference model, the thermal building envelope resistance is also split into two sections, R_1 and R_2 . The pipes allow water to circulate through the building and back to a storage tank. This tank has a volume of approximately 5.7 m^3 (1500 gallons). The total water in the pipes and in the storage tank increases the thermal capacitance of the building presented in Section 2.1.1 by a factor of six. The water tank capacitance was calculated using the specific heat, density and volume of the water in the tank. The sensitivity of the results to the tank size are presented later in Section 2.2.2.

The time constant for the building was significantly increased by the addition of the water tank. The time constant was also found from the simulation results by disconnecting the AC system and setting the ambient temperature to a constant value that will decay the building temperature to zero i.e., $T_{Amb}=0$. Equation 2.12 below was used to solve for the time constant obtained from the lumped parameter models.

$$T_{Building}(t) = T_{Building}(0) \times e^{-\frac{t}{\tau}} \quad (2.12)$$

where $T_{Building}$ is the temperature of the building at a given time; and $T_{Building}(0)$ is the initial temperature of the building. Equation (2.12) was rearranged to solve for the time constant as follows:

$$\tau = \frac{-t}{\ln\left(\frac{T_{Building}(t)}{T_{Building}(0)}\right)} \quad (2.13)$$

2.1.2.1 Verification of the Time Constants Using TRNSYS

Transient simulations using the software TRNSYS 17 [35] were conducted to verify the magnitude of the time constants used in the lumped parameter system presented in this paper. A detailed model was constructed using the software TRNSYS 17. The reference building was simulated using predefined standard ASHRAE wall, roof and flooring materials provided by the TRNSYS library. The walls were double brick as defined in Lombard and Mathews and have a standard thickness of 92 mm [50].

A constant weather input temperature of 10 °C was used when determining the time constant. The time constant was obtained from the response from the TRNSYS 17 assuming that after some time, transients with smaller time constant will decay and the dynamics of the response will follow the dominant time constant. Equation 2.14 was used to model the response

of the dominant time constant of the TRNSYS 17 model, assuming the response to faster time constants has already decayed.

$$T_{Building}(t) = A \times e^{\frac{-t}{\tau}} + C \quad (2.14)$$

where C is the constant weather temperature; and A is a constant of integration. For purely first order systems $A = T_{Building}(0)$ is the initial temperature of the building, but for a higher order model consisting of several time constants, the value of A associated with the dominant time constant will not necessarily be equal to $T_{Building}(0)$.

In order to extract the time constant information from the data obtained from TRNSYS 17 model, the above equation was rearranged as shown in Equation 2.15 where it is shown that the slope of the curve obtained by plotting the data in a semilog plot, i.e., $\ln(y-C)$ vs. t , is the inverse of the time constant. Figure 2.4 shows this plot and the trendline used to calculate the time constant. It was found that the time constant was 29 h, which is consistent with the initial calculated time constant for the reference building of 32 h and provides verification for the seemingly large value.

$$\ln(T_{Building} - C) = \ln(A) - \frac{1}{\tau} \times t \quad (2.15)$$

Figure 2.4 Trendline used to calculate the time constant for the TRNSYS simulation.

The second TRNSYS simulation used the same building that was constructed for the time constant verification, but TMY2 weather data was used during the simulation rather than the constant temperature input. Only the dry bulb temperature of Jackson, MS was used, and it was applied to the ambient temperature. It should be noted that all effects from radiative heat transfer have been neglected in both the TRNSYS simulation and the circuit analysis. The heat extracted by the building was collected for the months of February and July for both methods of solving the system. For the month of February, the TRNSYS simulation required 1246 MJ of heating and the circuit analysis required 1171 MJ, a 6% difference between the two simulations. During the month of July the TRNSYS simulation required 480 MJ of cooling and the circuit analysis required 446 MJ, a 7.1% difference between the two simulations.

2.1.2.2 Operation of ITC/TSM System

The ITC/TSM system is based on three modes of operation. In Mode 1 water is circulated through the water tank and the building. In Mode 2, water is only circulated between the water

tank and the heat exchanger. Mode 3 corresponds to the inactive state, where water is not circulated through the building or the heat exchanger. Mode 3 is used exclusively for storage. In all modes, the original capacitance of the building is used when calculating the room temperature. In Figure 3, the heat exchanger is modeled as a single resistance in series with the ambient temperature. The value of this resistance, seen in Table 2.1, was calculated by sizing a water-to-air heat exchanger [51] and by using the ambient and water temperatures.

T_{Emb} is the temperature at the location where the water pipes are embedded within the envelope thermal resistance. The requirements for circulating water through the building depend on T_{Emb} , the temperature of the water in the water tank, and the season. During warm months, May through September, the water circulates if T_{Emb} is greater than the temperature in the water tank. When water circulation from the tank capacitance is disconnected from the building envelope, the resulting T_{Emb} in Equation 2.16 is found using a “voltage divider” equation across the inner and outer envelope resistances, R_1 and R_2 , between the ambient temperature and the temperature of the building. Otherwise, the T_{Emb} is assumed to be at the same temperature as the water tank temperature, T_{Water} . All temperatures in Equation 2.16 are time-varying.

$$T_{Emb} = \begin{cases} T_{Water} & \text{if } \left(T_{amb} - \frac{T_{amb} - T_{Building}}{R_1 + R_2} \times R_1 \right) \geq T_{Water} \\ \left(T_{amb} - \frac{T_{amb} - T_{Building}}{R_1 + R_2} \times R_1 \right) & \text{otherwise} \end{cases} \quad (2.16)$$

During the cool months, October through April, the operation of the ITC/TSM is opposite of that in the warm months, resulting in Equation 2.17.

$$T_{Emb} = \begin{cases} T_{Water} & \text{if } \left(T_{amb} - \frac{T_{amb} - T_{Building}}{R_1 + R_2} \times R_1 \right) \leq T_{Water} \\ \left(T_{amb} - \frac{T_{amb} - T_{Building}}{R_1 + R_2} \times R_1 \right) & \text{otherwise} \end{cases} \quad (2.17)$$

The temperature of the water in the water tank, T_{Water} , is an additional state variable described in Equation 2.18 where Q_{Water} represents the heat transferred into the water.

$$\frac{dT_{Water}}{dt} = \frac{1}{C_{Water}} \times Q_{Water} \quad (2.18)$$

There are two equations associated with Q_{Water} , one for the warm season and one for the cool, shown in Equations 2.19 and 2.20, respectively.

$$Q_{Water} = \begin{cases} \frac{T_{Amb}-T_{Water}}{R_1} - \frac{T_{Water}-T_{Building}}{R_2} & \text{if } T_{Emb} \geq T_{Water} \\ \frac{T_{Amb}-T_{Water}}{R_{HEX}} & \text{if } (T_{Emb} \leq T_{Water}) \text{ and } (T_{Water} \geq T_{Amb}) \\ 0 & \text{otherwise} \end{cases} \quad (2.19)$$

$$Q_{Water} = \begin{cases} \frac{T_{Amb}-T_{Water}}{R_1} - \frac{T_{Water}-T_{Building}}{R_2} & \text{if } T_{Emb} \leq T_{Water} \\ \frac{T_{Amb}-T_{Water}}{R_{HEX}} & \text{if } (T_{Emb} \geq T_{Water}) \text{ and } (T_{Water} \leq T_{Amb}) \\ 0 & \text{otherwise} \end{cases} \quad (2.20)$$

Equations (21) and (22) describe the heat rate to the building, $Q_{Building}$, during warm and cool seasons, respectively.

$$Q_{Building} = \begin{cases} \frac{T_{Emb}-T_{Building}}{R_2} - \frac{T_{Building}-T_{cooling}}{R_{AC}} & \text{if } S = 1 \\ \frac{T_{Emb}-T_{Building}}{R_2} & \text{otherwise} \end{cases} \quad (2.21)$$

$$Q_{Building} = \begin{cases} \frac{T_{Emb}-T_{Building}}{R_2} - \frac{T_{Building}-T_{heating}}{R_{AC}} & \text{if } S = 1 \\ \frac{T_{Emb}-T_{Building}}{R_2} & \text{otherwise} \end{cases} \quad (2.22)$$

To avoid errors in selecting the initial building temperature, numerical data was collected only after the first 8 days of simulation, once the effects of the initial conditions have subsided. This is greater than the reference case because of the increased time constant. Note that this is only one time constant, not the typical 5τ that is required for a simulation to fully reach steady-state. Reducing the computational time only affected the collected data by 0.17%. This was

found by comparing the extracted energy on 21 July with 8 days of simulation and 50 days of simulation prior to collecting data.

A summary of the state variables equations and the process used when solving these equations can be seen in the flowchart shown in Figure 2.5.

Figure 2.5 Flowchart for state variable equations for the ITC/TSM system.

2.1.3 Cooling and Heating

The cooling system operates only for the months of May through September, and the heating system operates from October through April. The cooling and heating system each operate at a constant temperature and were modeled as a single temperature node (T_{AC}) and resistance (R_{AC}) since they would never operate simultaneously. This constant temperature node is intended to represent a working fluid, such as water or a refrigerant, coming in contact with heating or cooling coils. These constant temperatures are listed in Table 2.2. The AC operates on a cycle. During the evening hours, from 8 pm to 8 am, the house is kept at a cooler temperature than during the day hours, i.e., from 8 am to 8 pm. Occupational Safety and Health Administration (OSHA) requires that a workplace temperature must be kept in the range of 20 °C to 24 °C [52]. Thus the average daytime summer temperature was chosen to be 22 °C with an on/off cycle of 1 °C, i.e., the cooling system begins to operate when the building reaches 23 °C and stops operating when the building temperature drops to 21 °C. The heating system operates in a similar fashion, but the off temperatures are set slightly higher to allow the building to reach a higher temperature during the winter. Table 2.3 lists the upper and lower limit building temperatures.

2.2 Results and Discussion

Simulations were performed to provide a month-by-month comparison of the heat rate associated with the AC of each circuit. The total energy transferred each month was calculated by numerically integrating the heat transfer rate, Q_{AC} , over the month. Figure 2.6 displays absolute values and does not distinguish between whether the energy is extracted or added to the building. The energy extracted by the AC system is a result of the cooling system operating, May through September, and the energy added to the building is the result of the heating system

operating between the months of October through April. For the months of March through November, there are significant benefits from using the ITC/TSM system, but the simulation shows that there are only slight benefits from using the system from December through February. One reason the additional capacitance system is not beneficial during the winter months is because the ambient temperature does not reach a high enough temperature to assist with warming up the water. If the water temperature at the water tank, T_{Water} , is lower than the embedded temperature, T_{Emb} , the water will not circulate through the building and the building operates using the heating system in the same manner as the reference case. Though the system is not very beneficial for three months of the year, the benefits that occur during the 9 remaining months compensate for these winter months. May through September, the average reduction of the AC energy requirement is about 70% and May alone has a reduction of nearly 85%.

Figure 2.6 Energy required for reference building and ITC/TSM building

All monthly energy requirements can be seen in Figure 2.6. Though May has a high percentage of energy reduction, the month of October has the greatest reduction of energy required. The month of May does not heavily rely on the use of the AC subsystem and the ITC/TSM system almost eliminates the need for the AC subsystem completely. However, the month of October does rely on the use of the heating system, and the ITC/TSM system is able to provide the greatest reduction in required extracted/added energy. A month-by-month comparison for the difference in energy for the reference system and the ITC/TSM can also be seen in Figure 2.7.

Figure 2.7 The difference in required energy and the percent energy reduction.

The spike in the heat transfer rate through the heating system during the winter months can be explained by the difference between AC/heater temperature and the building temperature for the summer versus the winter. During the summer this temperature difference is roughly 9 °C

and in the winter this temperature difference is 60 °C. Since the AC/heater resistance remains constant, the heat transfer rates depend on the temperature difference between the building and the AC/heater operating temperature. Therefore, it takes more energy to heat the house during the winter than it does to cool the house during the summer.

2.2.1 Sensitivity Analysis

A sensitivity analysis was conducted for the following variables: the heat exchanger resistance, the AC/heating resistance, the building capacitance and the tank capacitance. The sensitivity analysis used 21 January for the simulation day and was conducted only for the circuit with the ITC/TSM system. Table 2.4 shows a summary of the variables which were altered and the resulting effect on the simulation. Each chosen variable was increased and decreased by 3% to show that it did not have a large impact on the calculation of the energy added by the heater. Increasing and decreasing each variable by 3% had less than a 2.3% effect on the energy added by the heater. The largest variance was from the increased building capacitance. This was expected since this effectively increases the building's mass and thus increases the required energy to maintain thermal comfort. All other variables, including the tank size, have less than a 0.6% effect on the energy added to the system.

Table 2.4 Sensitivity analysis results.

Variable	Modified Amount	Percent change from original energy added by heater
Heat exchanger resistance	Increased 3%	0.07%
	Decreased 3%	−0.6%
Heater resistance	Increased 3%	0%
	Decreased 3%	−0.53%
Building capacitance	Increased 3%	2.1%
	Decreased 3%	−2.3%
Water tank capacitance	Increased 3%	0.07%
	Decreased 3%	−0.6%

2.2.2 Economic Analysis

This section addresses the potential economic benefits of the proposed system. Three different cities were considered during the economic analysis: Jackson, MS; Jacksonville, Florida; and Atlanta, Georgia. This was done to show that the proposed system could be beneficial for climates similar to Jackson MS. An initial investment amount to install the proposed system was calculated by selecting the number of years desired for the payback period. To calculate the electricity consumption, the following equation was used [53]:

$$Electricity_{in}[W] = \frac{Cooling_{power}[\frac{BTU}{hr}]}{EER \text{ or } HSPF} \quad (2.23)$$

where EER is the energy efficiency ratio and HSPF is the heating seasonal performance factor.

As can be seen from Equation 2.23, the electricity consumption will strongly depend on the system selected, which will have different EER and HSPF values. For illustration purposes, a system with a typical average EER of 13 and an average HSPF of 10 was selected to estimate the potential savings and the initial investment amount that would be available to install the proposed system. During the cool season the HSPF value was used during calculations and during the warm season the EER value was used to calculate the require electricity.

Both the reference case and the ICT/TSM systems were simulated for a full year and the difference in electricity consumption between the two was multiplied by the electricity cost for each state. Electricity costs were taken from the U.S. Energy Information Administration [21]. To account for the inflation of the electricity cost, the increase in electricity cost was considered for the five previous years. The average of the percent increase for each year was the value used during the calculations, and these values and the electricity cost for each city are presented in Table 5. The ITC/TSM system was able to reduce the operating costs of the Jackson, Atlanta, and Jacksonville buildings by 31%, 23%, and 44%, respectively. Figure 8 shows the initial investment amount available to install the proposed system for payback periods up to 10 years. It can be seen that Jackson does produce the largest initial investment, though Atlanta and Jacksonville provide similar results. Though these initial investments might seem inadequate, they are due to the building being very small, and are limited by the fact that the building does not require much electricity to operate.

Table 2.5 Average retail price of electricity to residential sector, July 2014–2010 [54].

State	2014 (/kWh)	2013 (/kWh)	2012 (/kWh)	2011 (/kWh)	2010 (/kWh)	Average. % increase
Mississippi	11.63	10.88	10.07	10.41	10.29	3%
Georgia	12.55	12.31	11.96	12.07	10.95	4%
Florida	11.96	11.39	11.56	11.82	11.72	1%

Figure 2.8 Initial investment amount (USD) per payback period.

This analysis can be extrapolated to a bigger building in which the proposed system would potentially provide more savings and the initial investment to install the proposed system would be higher. In addition, the same estimation can be done for different systems that would have different values of EER and HSPF.

2.3 Conclusions

The purpose of this chapter was to determine if the proposed ITC/TSM system could potentially reduce the energy required for thermal comfort in a building. This was achieved by comparing two circuits, a reference circuit and a circuit implementing the ITC/TSM system, both simulated for a year to determine the energy required by the AC/heating subsystem. The first-order simulations determined that there were significant reductions in the energy consumption of the AC subsystem when the ITC/TSM system was used in a simple one-room micro building

with climate conditions corresponding to Jackson, Mississippi. However, since the simulations correspond to a first-order model approximation, the benefits would likely not be as large as the 70% reduction seen during summer operation. The largest monthly reductions for the evaluated ITC/TSM system occur in the months of March through November. It is expected that the ITC/TSM system would be able to deliver a similar performance for locations with climate conditions similar to these months, as reference temperatures during these seasons reach as high as 37 °C, and as low as -7 °C in the beginning of March and at the end of October. The ITC/TSM is frequently active in Mode 1 or Mode 2 and is occasionally operating in Mode 3, the inactive state, which is equivalent to the reference case. The greatest benefits are gained from the ITC/TSM system when the building does not operate in Mode 3. The ITC/TSM system is often active from March to October, which suggests that a significant portion of these months consistently take advantage of ITC/TSM system.

CHAPTER III

TRANSIENT CASE STUDY OF THE ITC/TSM SYSTEM

This chapter simulates an increased thermal capacitance (ITC) and thermal storage management (TSM) system to reduce the energy consumed by air conditioning and heating systems. The ITC/TSM is coupled with phase change materials (PCM), which enable tank volume reduction. The transient energy modeling software, the Transient System Simulation Tool (TRNSYS), is used to simulate the buildings' thermal response and energy consumption, as well as the ITC/TSM system and controls. Four temperature-controlled operating regimes are used for the tank: building shell circulation, heat exchanger circulation, solar panel circulation, and storage. This study also explores possible energy-saving benefits from tank volume reduction such as losses associated with the environment temperature due to tank location. Three different tank locations are considered in this paper: outdoor, buried, and indoor. The smallest tank size (five gallons) is used for indoor placement, while the large tank (50 gallons) is used either for outdoor placement or buried at a depth of 1 m. Results for Atlanta, Georgia show an average 48% required energy decrease for cold months (October–April) and a 3% decrease for warm months (May–September) for the ITC/TSM system with PCM when compared with the reference case. A system with PCM reduces the tank size by 90% while maintaining similar energy savings.

3.1 Reference Model

A reference model was simulated to provide a basis to compare the ITC/TSM system. All of the building materials, simulation parameters, and weather data listed in sections 3.1.1 and 3.1.2 were kept constant for all of the simulations. This was done to isolate the effect of the water tank on the building air conditioning load requirement. The heating and cooling system was also unchanged between simulations. Figure 3.1 shows the TRNSYS schematic used for the reference model simulations.

Figure 3.1 Transient System Simulation Tool (TRNSYS) Schematic for Reference Building

3.1.1 Building Specifications and Location

A small single room wood-sided building with a concrete slab was selected for the simulations. It was intended to have a similar size as a small/medium apartment or small office building. The building had a floor plan of 100 m² (1300 ft²) and 3-m (10 ft) high ceiling. Surfaces were chosen from the TRNSYS library, which contains typical wall constructions [55].

Materials were chosen to simulate lightweight buildings that are common in the southeastern part of the US. The wall, ceiling, and floor building materials and thickness are listed in Table 3.1.

Table 3.1 Building Materials and Properties.

Surface Type	Layer (Inside to Outside)	Thickness (mm)	Conductivity (kJ/(h m K))	Capacity (kJ/(kg K))	Density (kg/m ³)
External Wall	Plasterboard	12	1.9	0.84	1200
	Fiberglass Insulation	67	0.144	0.84	12
	Wood Siding	9	0.504	0.9	530
Floor	Tile	25	5.17	1.5	880
	Insulation	76	0.16	0.75	32
	Concrete	100	6.23	0.75	2242
Ceiling/Roof	Plasterboard	10	1.9	0.84	1200
	Fiberglass Insulation	112	0.144	0.84	12
	Roof Decking	19	0.504	0.9	530

Additionally, the building was ground coupled to allow heat transfer through the floor of the building. This was done by calculating the soil temperature at the depth of the slab foundation and allowing this to be the thermal boundary condition for the bottom side of the slab. An approximation for the soil temperature was obtained using Kusuda's equation [56].

$$T_{\text{soil}}(t) = T_{\text{mean}} - T_{\text{amp}} \left(\exp \left(-d \left(\frac{\pi}{365\alpha} \right)^{0.5} \right) \cdot \cos \left(\frac{2\pi}{365} \left(t - t_{\text{shift}} - \frac{d}{2} \left(\frac{365}{\pi\alpha} \right)^{0.5} \right) \right) \right) \quad (3.1)$$

where T_{mean} is the average annual air temperature, T_{amp} is the maximum air temperature minus the average air temperature, d is the depth of the desired temperature, α is the soil thermal diffusivity, t is the day at which the temperature is found, and t_{shift} is the day of year associated with the minimum air temperature. The values used for Atlanta, Georgia are given in Table 3.2.

Table 3.2 Ground Coupling Properties.

Ground Coupling Properties	Value
T_{mean}	16.33 °C
T_{amp}	4.945 °C
d	0.12 m
α	0.84 kJ/(kg K))
t_{shift}	0 day

Weather information was selected from Typical Meteorological Year 2 (TMY2) data for Atlanta, Georgia. The following is the weather information used from TMY2 data: dry bulb temperature, percent relative humidity, and atmospheric pressure. TRNSYS Type 15 was used in conjunction with TMY2 data to calculate total radiation, beam radiation, and angle of incidence of beam solar radiation each surface as well as the effective sky temperature, humidity ratio, and ground reflectance. The walls of the building were identical and were oriented at azimuth angles of 0°, 90°, 180°, and 270°. Each external wall of the building had a 0.5 m² insulated window. These windows had a u-value of 1.3 and a g-value of 0.591.

Natural convection was assumed as the boundary condition of the internal walls, and an averaged forced convection was assumed for external walls. A standard value was used for the convective heat transfer coefficient, which was 3 W/m² K for the internal wall side, and 24 W/m² K for the external wall side [55]. For increased thermal capacitance (ITC)/thermal storage management (TSM) cases, automatic calculation was allowed to determine the inside heat transfer coefficient. This was because the internal heating or cooling of the wall can impact the surface temperature on which the natural convection heat transfer coefficient is dependent.

3.1.2 Heat Pump Parameters

A small air source heat pump was chosen to provide the heating and cooling requirements for the building. It was modeled using TRNSYS TESS (Thermal Energy System Specialists) Library component 954 with the percent relative humidity mode for moist air calculations. This component uses normalized capacity, sensible capacity, and power tables to calculate both the heating and cooling rates. The parameters for the selected heat pump were chosen from a commercially available split system heat pump data sheet, and are shown in Table 3.3.

Table 3.3 Heat Pump Parameters.

Heat Pump Parameter	Value
Rated Cooling Capacity	10 kW (~3 ton)
Rated Heating Capacity	10 kW
Rated Air Flowrate	36 m ³ /min (1200 CFM)

From ASHRAE Standards 62.1 and 62.2 [57], the required fresh air CFM (cubic foot per minute) per person for an office building is 17, and for an apartment building is 7.5. Therefore, 17 CFM/person was used in all of the simulations. Using this value, an air flow rate of 1200 CFM, and an outside damper position of 30% allows for up to 21 people to occupy the building at a single time.

Two different climate seasons were considered: a heating season and a cooling season. The heating season was set from 1 November to 31 March, and the cooling season was set from 1 April to 31 October. The thermostat setting included a setting for daytime and evening operation, and a 2 °C deadband around the nominal setting. Evening was considered to last from 8 pm to 8 am, and daytime was considered to last from 8 am to 8 pm. For the cooling season, a

step-up temperature was applied during the daytime, and for the heating season, a step-back temperature was applied during the daytime. This was done to simulate how an individual might control the AC/heat when the building was not occupied to reduce operating costs. Values for the thermostat are listed in Table 3.4.

Table 3.4 Heat Pump Nominal Thermostat Settings.

Setpoint	Cooling (1 April to 31 October)	Heating (1 November to 31 March)
Daytime (8 am–8 pm)	22 °C	20 °C
Evening (8 pm–8 am)	21 °C	22 °C

During the heating season, the air condition operated if the indoor temperature was above 24 °C (75.5 °F), and during the cooling season, the heater operated if the indoor temperature was below 18.5 °C (65.5 °F). Due to the nature of the southeast US weather, the conditions that allowed the heat pump to operate during the cooling season and the AC to operate during the heating season only occurred during the mild months of the year, which were April, May, and September.

3.2 ITC/TSM Model with PCM

The increased thermal capacitance and thermal storage management (ITC/TSM) model expanded upon the reference case by addition of a water loop system. Copper pipes embedded in the walls and ceiling of the building used water circulation to increase the thermal capacitance of the building, and thus increased the overall time constant of the building. Variations in the building's thermal time constant for each model are discussed in Section 3.2.3.

the water temperature when climate conditions were beneficial. A seasonal switch was also implemented to only allow HEX circulation during the cooling season and solar panel circulation during the heating season. The purpose of this method was to add a non-temperature restriction on circulation. This approach kept the water from attempting to cool during the winter and heat during the summer. The switch operated on a monthly basis, which was associated with the heating/cooling seasons. Building circulation was always a priority; when the building could not benefit from water circulation and the outdoor climate was not favorable, the pump shut off and the water was stored in the tank (which was subject to losses to the surroundings). The requirements for these valves are listed in Table 3.5.

Table 3.5 Tank Circulation Regime. HEX: heat exchanger..

Season	Building Circulation	HEX Circulation	Solar Panel Circulation
Cooling Season	$T_{\text{Tank}} \leq T_{\text{Building}}$	$T_{\text{Tank}} > T_{\text{Building}}$ and $T_{\text{Tank}} > T_{\text{Ambient}}$	N/A
Heating Season	$T_{\text{Tank}} \geq T_{\text{Building}}$	N/A	$T_{\text{Tank}} < T_{\text{Building}}$ and Collector is gaining energy ¹

where T_{Tank} is the average tank temperature, T_{Building} is the average building temperature, and T_{Ambient} is the outdoor dry bulb temperature of Atlanta, Georgia.

A large water tank was used to significantly increase the time constant of the building [58]. To retain a large thermal time constant and reduce the size of the tank, a phase change material (PCM) was added. Only water was circulated through the loop; the PCM remained in the tank. For simulation purposes, it was assumed that the phase change material was perfectly mixed in the water tank. The HEX and solar panel were used to heat or cool the water that cycles

back to the PCM. This process kept the water temperature oscillating within the small temperature band close to the melting point of the PCM.

To implement the PCM in TRNSYS, a vertical cylinder fluid storage tank with a single inlet and outlet was modified. Seven user input parameters were added: percent of tank volume that is PCM, PCM solid and liquid density, PCM solid and liquid specific heat, melting temperature, and heat of fusion.

The program for the vertical thermal storage was modified in various ways. A crucial addition was the numerical approximation for the PCM's properties. This was done using a method demonstrated by Voller [59], which proposed that the phase change happens over a small temperature range instead of isothermally. This allowed the specific heat of a phase change material to be approximated by a piecewise function as seen in Equation 3.2:

$$C_{p_{pcm}}(T) = \begin{cases} C_{p_{solid}}; & T \leq T_{melt} - \epsilon \\ \frac{C_{p_{solid}} + C_{p_{liquid}}}{2} + \frac{LHF}{2\epsilon}; & T_{melt} - \epsilon < T < T_{melt} + \epsilon \\ C_{p_{liquid}}; & T \geq T_{melt} + \epsilon \end{cases} \quad (3.2)$$

where $C_{p_{solid}}$ is the specific heat for the solid PCM with units of J/kg K, $C_{p_{liquid}}$ is the specific heat of the liquid PCM with units of J/kg K, LHF is the latent heat of fusion with units of J/kg, and ϵ is the small temperature interval over which phase change occurs with units of K. Selection of the temperature interval ϵ is discussed by Voller [59]. A graphical representation of the specific heat's relationship to temperature and enthalpy is shown in Figure 3.3.

Other modifications included the reduction of water mass when PCM was added to the tank.

Equation 3.2 is the water mass on a nodal basis:

$$m_{water}(i) = A_C(i) \cdot H(i) \cdot \rho \cdot (1 - Per_{pcm}) \quad (3.3)$$

where $A_C(i)$ is the cross-sectional area of the tank for each node with units of m^2 , $H(i)$ is the tank height of each node with units of m , ρ is the density of water with units of kg/m^3 , and Per_{pcm} is the percentage of the tank volume occupied by the PCM.

Figure 3.3 Numerical Approximation of Specific Heat for Phase Change Materials (PCM).

Equation 3.3 represents the mass of PCM on a nodal basis:

$$m_{pcm}(i) = A_C(i) \cdot H(i) \cdot \rho_{pcm} \cdot Per_{pcm} \quad (3.4)$$

To determine the mass of PCM, the density of the solid state was used. It was assumed that the PCM was solid when calculating the volume occupied in the tank.

Equation 3.4 is the modified nodal stored internal energy to include the PCM:

$$e(i) = m_{water}(i) \cdot T(i) \cdot C_p + m_{pcm}(i) \cdot T(i) \cdot C_{p_{pcm}}(T(i)) \quad (3.5)$$

where $T(i)$ is the temperature at each node with units of K, C_p is the specific heat of water with units of J/kg K, and $C_{p_{pcm}}(T(i))$ is the temperature-dependent specific heat of the PCM. This value is only reported when calculating the internal energy change i.e., $\Delta e(i) = e(i) - e_0(i)$ where e_0 is the internal energy at the beginning of the simulation.

The heat equation (Equation 3.5) for the i th tank segment was also altered to include the phase change mass and specific heat. The specific heat for the PCM is evaluated at the previous timestep.

$$\begin{aligned} \left(m_{\text{water}}(i) \cdot C_p + m_{\text{pcm}}(i) \cdot C_{p_{\text{pcm}}}(T(i)) \right) \frac{dT(i)}{dt} = & \frac{(k + \Delta k) \cdot A_c(i)}{\Delta x_{i+1 \rightarrow i}} (T(i+1) - T(i)) \\ & + \frac{(k) \cdot A_c(i)}{\Delta x_{i-1 \rightarrow i}} (T(i-1) - T(i)) + (U_{\text{tank}}) A_s(i) (T_{\text{env}} - T(i)) + \dot{m}_{\text{down}} C_p (T(i-1)) T(i-1) \\ & - \dot{m}_{\text{up}} C_p (T(i)) T(i) - \dot{m}_{\text{down}} C_p (T(i)) T(i) + \dot{m}_{\text{up}} C_p (T(i+1)) T(i+1) \\ & + \dot{m}_{\text{in}} C_p (T_{\text{in}}) T_{\text{in}} - \dot{m}_{\text{out}} C_p (T(i)) T(i) \end{aligned} \quad (3.6)$$

where, $\Delta x_{(i+1 \rightarrow i)}$ is the center-to-center distance between node i and the node below it ($i+1$) in m, $\Delta x_{(i-1 \rightarrow i)}$ is the center-to-center distance between node i and the node above it ($i-1$) in m, k is the thermal conductivity of water with units of W/mK, U_{tank} is the loss coefficient (per unit area) of the tank with units of W/m² K, $A_s(i)$ is the surface area of node i in m², T_{env} is the environment temperature in K, $\dot{m}_{\text{down}}$ and $\dot{m}_{\text{up}}$ are the bulk fluid flowrate down/up the tank in kg/s, and $\dot{m}_{\text{in}}$ and $\dot{m}_{\text{out}}$ are the mass flowrate entering at the inlet and leaving at the outlet in kg/s. This can be seen in a graphical representation of the energy flow into each node in Figure 3.4, which was taken from the mathematical reference [60] provided by TRNSYS for Type 60.

Figure 3.4 Graphical Representation of Energy Flows into a Node.

Only the internal energy of the tank's i th segment was altered to include the PCM, since the PCM is not part of the flow energy transfer. The conduction of the phase change materials in each segment is included in the numerical approximation of the specific heat.

A generic paraffin PCM was chosen for simulations because of the stability in thermal cycling and the high heat of fusion [61]. Typical values of heat of fusions range from 200 kJ/kg to 280 kJ/kg; the specific heat is around 2 kJ/kg K, solid density is around 900 kg/m³, and melting temperatures range from -20 °C to 100 °C [61]. Preliminary simulations found benefits peaked for a melting temperature of 21.5 °C.

Values were chosen for the PCM using the generic paraffin ranges, and are presented in Table 3.6.

Table 3.6 PCM Parameters.

PCM Parameter	Value
Heat of Fusion	230 kJ/kg
Specific Heat	2 kJ/kg K
Density (solid)	900 kg/m ³
Melting Temperature	21.5 °C

3.2.1 Verification of PCM Tank Operation

To verify that the specific heat modifications were working correctly, the tank was isolated with only a sinusoidal temperature input and constant flowrate. Figure 3.5 shows the average tank temperature, the PCM melting temperature, and the upper and lower bounds for phase change. It can easily be seen how the temperature slowly decreased over 0.016 °C ($\epsilon = 0.008$ °C) during phase change. This simplified mode is only shown for verification that the tank specific heat piecewise equation (i.e., Equation (3.1)) was operating as intended.

Figure 3.5 PCM Phase Change Operation.

3.2.2 TRNSYS Solar Panel Selection

A 2 m² flat plate solar panel was selected and the parameters were found from a Solar Rating and Certification Corporation (SRCC) data sheet [62]. The TRNSYS TESS component 539 was used to model the solar panel in simulations. This was chosen for the ability to internally modulate the flowrate for outlet temperature control. If the collector was gaining energy, the flowrate varied between a user-defined maximum and minimum flowrate to attempt to deliver a desired outlet temperature. If this temperature could not be maintained, the collector would run at the minimum flowrate, and would automatically shut off if the collector was losing energy.

Minor modifications were made to the TRNSYS TESS component; the maximum flowrate was redefined as an input variable instead of a parameter. This allowed the maximum flowrate to change at each timestep. The maximum flowrate was set to the inlet flowrate to the

solar panel; this method kept the solar panel from operating when the input flowrate was zero, and allowed it to stay on as long as the collector was gaining energy. The parameters for the solar panel are listed in Table 3.7.

Table 3.7 Solar Panel Parameters for Solar Rating and Certification Corporation (SRCC) Data Sheet.

Solar Panel Parameter	Value
Collector Area	2 m ²
Intercept Efficiency (a_0)	0.775
First Order Efficiency Coefficient (a_1)	18.37 kJ/(h m ² K)
Tested Flowrate per Unit Area	70 kg/(h m ²)
First Order Incidence Angle Modifier (IAM) Coefficient	0.0664

3.2.3 Verification of Building Thermal Time Constant Increment

To confirm that the dominant time constant of the building increased when implementing the ITC/TSM system, a free response simulation was conducted. Equation 3.6 was used to model the response of each case.

$$T_{\text{Building}} = A \cdot e^{\frac{-t}{\tau}} + C \quad (3.7)$$

where τ is the time constant, C is the constant weather temperature, and A is a constant of integration.

The above equation was rearranged to solve for the time constant τ .

$$\ln(T_{\text{Building}} - C) = \ln(A) - \frac{1}{\tau} \cdot t \quad (3.8)$$

The inverse of the slope of Equation 3.7 yields the dominant time constant for the building. For each case, the building's free response was found, and $\ln(T_{\text{Building}} - C)$ was plotted versus time.

These slopes and their respective linear trendline equations can be seen in Figure 3.6.

Figure 3.6 Building Free-Response.

The time constant of the reference case was 18 h, for the 50-gallon tank ITC/TSM case was 34.4 h, and for the five-gallon ITC/TSM with 95% phase change materials was 31.8 h. Thus, the PCM allowed for a 90% reduction in tank volume size with a very small reduction in the dominant time constant of the building.

3.2.4 Energy Balance

An energy balance was conducted for each major component in the simulation to ensure that the modifications to the water tank and the solar panel did not result in any unbalanced

energy equations. A certain amount of error is expected for numerical simulations; therefore, the percentage of unbalanced energy is reported in the results section.

3.2.4.1 Water Tank

When examining the water tank, all of the thermal losses to the ambient conditions were neglected. Equation 3.8 was used to calculate the unbalanced energy rate for the water tank.

$$UBE_{\text{tank}} = (E_{\text{out}} - E_{\text{in}}) - C_{p_{\text{water}}}(\dot{m}_{\text{out}}T_{\text{out}} - \dot{m}_{\text{in}}T_{\text{in}}) \quad (3.9)$$

where UBE_{tank} is the unbalanced energy rate of the water tank, E_{out} and E_{in} are the energy removed by the outlet and energy supplied by the inlet as calculated internally by TRNSYS with units of kJ/h, $C_{p_{\text{water}}}$ is the specific heat of water with units of kJ/kgK, $\dot{m}_{\text{out}}$ and $\dot{m}_{\text{in}}$ are the mass flowrate associated with the outlet and inlet with units of kg/h, and T_{out} and T_{in} are the inlet and outlet temperatures in K.

For the water tank, 0.2% of timesteps did not satisfy the energy balance; however, after numerical integration of the energy rate, the energy associated with these timesteps only accounted for 0.00012% of the total energy transferred into or out of the tank subsystem.

3.2.4.2 Solar Panel

The solar panel energy balance was checked using Equation 3.9.

$$EB_{\text{Solar.Panel}} = \frac{\text{Gained}_{\text{Energy}} - C_{p_{\text{water}}}\dot{m}_{\text{in}}(T_{\text{out}} - T_{\text{in}})}{\text{Gained}_{\text{Energy}}} \quad (3.10)$$

where $\text{Gained}_{\text{Energy}}$ is the TRNSYS internal calculation of energy gained by the solar collector with units of kJ/h, $C_{p_{\text{water}}}$ is the specific heat of water with units of kJ/kgK, $\dot{m}_{\text{in}}$ is the mass flowrate for the inlet with units of kg/h, and T_{out} and T_{in} are the inlet and outlet temperatures in K, respectively.

The energy balance for the solar panel showed that all of the timesteps balanced.

3.3 Results

Simulations based on the equations presented in Section 3 were used to obtain the monthly energy requirements for each case. The total cooling and heating loads were numerically integrated using TRNSYS Type 46. This was the primary metric that was used to determine the energy reduction benefits of the ITC/TSM system. A month-by-month case was used for showing seasonal benefits as opposed to an overall yearly benefit. Although energy was added to the building during the heating months and extracted during the cooling months, energy is shown only as positive values in figures 7 and 8. April, May, and September had the magnitudes of the added and extracted energy summed together. This was because both the heating and cooling modes are likely to be active during these months.

The main objective of this study is to show the viability of tank volume reduction without loss of energy savings by implementing PCM. Simulations were completed for a 50-gallon tank without PCM, and a five-gallon tank with 95% of the volume occupied by PCM. Due to the size restrictions, the 50-gallon tank was required to be either buried at 1-m depth or located outside the building exposed to ambient conditions, and the five-gallon tank was located inside the building.

Figure 3.7 Energy Consumption.

Figure 3.7 shows the required energy for the reference case and two ITC/TSM cases. It illustrates how the addition of PCM allows for similar results with a volume reduction of 90%. It needs to be noted that the heat transfer losses to the ambient conditions are different for each case. Since the five-gallon tank is kept inside the building, heat transfer losses are less compared to the 50-gallon tank. Figure 7 also shows results for the 50-gallon tank when it is located outdoors.

Another benefit of using the small PCM tank is the ability for the tank to be located in a temperature environment that is close to the desired tank temperature, allowing for heat transfer to the ambient to be reduced.

The thermal losses for the indoor five-gallon tank is 99.58% less than the outdoor 50-gallon tank, and 99.54% less than the buried 50-gallon tank. For all of the cases, the tank was assumed to be well-insulated, and has the same tank loss coefficient of 3 kJ/h m² K.

Figure 3.8 Energy Reduction.

Figure 8 shows the monthly percent reduction in energy required by the heat pump. Energy reduction is not consistent, with an average heating reduction of 48% and an average cooling reduction of 3%. The month of May has no significant benefits, and September has moderate losses (24%). However, these losses are offset by the considerable savings found between November–March. The results indicate that it is possible that the valve operating conditions and the PCM parameters are more beneficial for cold months. Using different PCM and adjusting operating regimens for a specific season could show an increase in benefits.

Specifically, water circulation currently operates on a monthly schedule when switching between the HEX and the solar panel for temperature regulation. More benefits could be found, or the losses in September could be reduced if a semi-predictive method was used on a weekly basis instead of using historical averages on a monthly basis. Although a 24% increase in consumption for September sounds significant, it is only 46 kWh, and is around 2% of the total required energy for the reference model during the cooling months. These new operating conditions could likely benefit other transitional months, as mild temperatures showed the least benefits from the system. Again, it must be noted that these mild months consume the least amount of energy. The overall yearly savings of the ITC/TSM model was about 35%, and the ITC/TSM with the PCM model was about 35.2%.

3.4 Conclusions

This chapter showed the capability of reducing the energy consumption of a small, lightweight, free-standing building due to air conditioning and heating units through a water loop system designed for ITC/TSM. PCM was added for the purpose of reducing the tank volume without a significant loss of energy savings. Tank volume reduction also allowed for reduced heat transfer loss as a result of indoor storage, although it was not the only factor for energy savings. Three different models were constructed for simulation: a reference case, an ITC/TSM case, and an ITC/TSM with PCM case. A 50-gallon tank was used in simulations for the ITC/TSM case, and a five-gallon tank was used for the ITC/TSM with PCM case. Although there was a 90% reduction of tank size, the PCM allowed for similar energy savings. The overall yearly savings of the ITC/TSM model were about 35%, and the savings of the ITC/TSM with PCM model were about 35.2%.

Although there was a slight loss in savings using the smaller tank, further development can possibly minimize this loss or provide additional benefits. A complete parameter study of the PCM could allow further energy savings, as was seen in preliminary simulations for the melting temperature. Future simulations will also include complex buildings with interior walls, doors, and additional windows to allow simulations that are a better representative of physical applications. Studies will also be conducted to determine the benefits associated with split flow operation and semi-predictive methods for heating and cooling the water. This would allow portions of the flow to circulate through the building, while the remaining flow could be heated or cooled for the building benefit. These new operating regimes could be found using Bayesian classification techniques to minimize air conditioning and heating operating times.

CHAPTER IV

DESIGN AND OPERATION OF THE ITC/TSM SYSTEM

This chapter explores the design and operation of an increased thermal capacitance (ITC) and thermal storage management (TSM) system for reducing building energy consumption associated with heating, cooling, and domestic hot water (DHW) usage. Thermal capacitance is a controlling factor for phase shift and amplitude reductions of cyclical heat transfer due to weather, and control of the ITC through TSM improves upon the benefits offered by additional capacitance. Using the transient energy modeling software TRNSYS, ITC is achieved by water circulating in copper pipes embedded in the walls of a three-story residential building. Temperature and season-controlled valves divert circulation to either the building shell, ground heat exchanger, solar panel, cold storage tank or hot storage tank. A phase change material is thoroughly mixed into the water storage tanks to allow a tank volume reduction without loss of thermal storage. Proper design and operation of the ITC/TSM system are investigated through a parametric study of the water tank size, water mass flowrate and solar panel size. These analyses are focused on reducing the overall building energy requirement associated with the heat pump and DHW usage. Results range from 24% to 35% energy savings for the evaluated parameter ranges.

4.1 Model Description

The reference simulation was designed to isolate benefits found from implementing the ITC/TSM system. The ability to compare the same building with and without the ITC/TSM system easily show the savings associated with the system. Both the reference case and the ITC/TSM case were modeled using the building modeling software, Transient System Simulation Tool (TRNSYS) [35]. The primary building energy objective was maintaining thermal comfort and providing domestic hot water (DHW). Building materials, location, heat pump size, and schedules associated with DHW, occupancy, and lighting were unchanged between the reference and the ITC/TSM case.

4.1.1 Reference Building Specifications and Location

Building materials and the floor plan were taken from the single family residential prototype developed by the Pacific Northwest Laboratory (PNNL) [63] and follows the 2012 International Energy Conservation Code (IECC). The fully electric residential building had a 111.57 m^2 (1200 ft²) slab foundation, 3 stories which includes an unconditioned attic. A model of the building using SketchUp [64] can be seen below in Figure 4.1.

Figure 4.1 Building Model in SketchUp.

Material used for the external walls, floor and roof are listed in Table 4.1.

Table 4.1 Building Shell Materials [63]

Building Construct	Layer (internal to external)	Thickness (m)	Conductivity (kJ/hr m K)	Capacity (kJ/kg K)	Density (kg/m³)
External Walls	Dry Wall	0.013	0.259	0.768	512.64
	Wall Consol	0.140	0.215	1.037	120.801
	OSB	0.016	0.419	1.214	544.628
	Sheathing Consol	0.013	0.339	1.173	685.008
	Paper	Massless	-	-	-
	Stucco	0.025	5.037	0.879	1922
External Floor	Carpet and Pad	0.025	0.217	0.837	32.037
	Plywood	0.019	0.416	0.675	544.68
	Floor Consol	0.014	0.175	0.918	55.074
	Slab	0.10	3.241	0.65	2300
Interior Ceiling	Dry Wall	0.013	0.259	0.768	512.64
	Ceiling Consol	0.318	0.222	0.777	41.929
End Gable	Dry Wall	0.013	0.259	0.768	512.64
	Air Gap	Massless	-	-	-
	OSB	0.016	0.419	1.214	544.628
	Paper Felt	Massless	-	-	-
	Stucco	0.025	5.037	0.879	1922
Ceiling and Roof	Ceiling Consol	0.318	0.222	0.777	41.929
	Plywood	0.019	0.416	0.675	544.68
	Asphalt Shingle	0.006	0.295	1.25	1120

Each external wall had one window with a u-value of 1.4, a g-value of 0.589, and internal blinds with shade factor of 0.8675. Windows on the east side of the building also included a shading overhang. All weather data was taken from Typical Meteorological year 2 (TMY2) data for the city of Atlanta Georgia. Dry bulb temperature, effective sky temperature, and percent relative humidity are used as weather inputs to the building and TRNSYS Type 15 internally calculated the following weather information for each of the six building surfaces: total tilted surface radiation, beam radiation, and angle of incidence. The vertical walls of the building were orientated with azimuth angles of 0°, 90°, 180°, and 270°, and the pitched roof had a 20° slope with azimuth angles of 0° and 180°.

The soil temperature was also calculated using Kusuda's equation (Eq. 4.1) to provide the proper boundary conditions for heat transfer from the building slab foundation.

$$T_{\text{soil}}(t) = T_{\text{mean}} - T_{\text{amp}} \left(\exp \left(-d \left(\frac{\pi}{365\alpha} \right)^{0.5} \right) \cdot \cos \left(\frac{2\pi}{365} \left(t - t_{\text{shift}} - \frac{d}{2} \left(\frac{365}{\pi\alpha} \right)^{0.5} \right) \right) \right) \quad (4.1)$$

where T_{mean} is the average annual air temperature, T_{amp} is the amplitude of the surface temperature, d is depth at which the temperature is approximated, α is the soil thermal diffusivity, t is the time at which the temperature is calculated, and t_{shift} is the different between the first day of the year and the minimum surface temperature of the year. Parameters used to approximate the soil temperature for Atlanta Georgia at a dept of 0.12 meters are listed in Table 4.2.

Table 4.2 Soil Temperature Properties [65]

Parameter	Value
T_{mean}	16.33°C
T_{amp}	4.945°C
d	0.12 m
α	8.72 kJ/(hr m K)
t_{shift}	0 days

Figure 4.2 below shows the TRNSYS schematic for the reference model.

Figure 4.2 Reference TRNSYS Schematic.

Domestic hot water (DHW) scheduling and flowrates were also taken from the PNNL residential prototype. These schedules and flowrates include usage associated with the dishwasher, washing machine, sinks, baths and showers. Two water draw profiles were created to simulate weekday usage and weekend usage. These profiles are shown below in Figure 4.3.

Figure 4.3 DHW Draw Profiles

For the reference case, a tank volume of 0.2m^3 (53gal) was used with a 5.5 kW heating element, a setpoint of 60°C with a 5°C deadband.

Additional heat gains for the building include 3 person occupancy, and internal lighting. The schedule for both comes from the PNNL residential prototype and can be seen below in Figure 4.4.

Figure 4.4 Lighting and Occupancy Schedule

Three different thermal zones were specified in the building, they were designated as downstairs, upstairs and attic. The upstairs and downstairs zone temperatures were monitored and controlled with a commercially available split system heat pump. Parameters used to model the heat pump can be seen below in Table 4.3.

Table 4.3 Heat Pump Parameters [66]

Heat Pump Parameter	Value
Rated Cooling Capacity	10 kW
Rated Heating Capacity	7 kW
Rated Air Flowrate	36 m ³ /min (1200 CFM)

Normalized cooling and heating performance data, along with the rated cooling capacity, rated heating capacity and the air flowrate, were used to calculate the required heating and cooling rates using the TRNSYS TESS (Thermal Energy System Specialists) Component 954. The third zone, the attic, did not have a heating or cooling system, but operated as an insulator from the environment temperature.

ANSI/ASHRAE Standard 62.2-2016 states that a minimum ventilation air requirement for a 2500 ft² building with 4 bedrooms is 128 cfm. Therefore, the heat pump chosen for each floor delivers 600 cfm along with a 30% damper position to allow fresh air to be mixed with the return. Where the return air is the mixture of the return air from both the upstairs air node and the downstairs air node.

A monthly schedule was used to denote the months which required AC operation (cooling months) and the months which required heat operation (heating months). April 1st to October 31st was considered the cooling season, and November 1st to March 31st was considered the heating season. Table 4.4 contains the temperature setpoint during the daytime (8am-8pm) and evening (8pm-8am) for both the heating and cooling season.

Table 4.4 Heat Pump Thermostat Settings

Setpoint	Cooling (1 April to 31 October)	Heating (1 November to 31 March)
Daytime (8am-8pm)	22°C	20°C
Evening (8pm-8am)	21°C	22°C

To employ a hysteresis effect, each setpoint temperature had a 2°C deadband. Months with mild temperatures, such as April and October, could require both the cooling and the

heating system operation to maintain thermal comfort. During the heating season, the AC would operate if the building reaches 26°C and during the cooling season, the heat would operate if the building drops to 16°C. Controllers do not allow both the heat and the AC to operate simultaneously.

4.1.2 Proposed ITC/TSM System

The ITC/TSM system was designed to reduce the operating requirement of the heat pump system and the DHW storage tank heating element. To increase the building shell thermal capacitance, water was circulated through the building in copper pipes embedded in the walls. Control of the water temperature was vital for building circulation. The temperature of the water was required to be above building room temperature during the heating season and below the building room temperature during the cooling season. This specification kept the additional thermal capacitance from becoming a hinderance to the heat pump system. Whenever the temperature of the water fell outside the compatible temperature requirement for each season the circulation would bypass the building. In addition to the hot water storage tank used for DHW, three devices were used to control the water temperature: a solar panel, a ground heat exchanger, and a cold water storage tank. The TRNSYS schematic for the ITC/TSM case can be seen in Figure 4.5.

Figure 4.5 ITC/TSM TRNSYS Schematic

A flowchart that shows the summer and winter operation of the ITC/TSM system can be seen in Figure 4.6. Though operating requirements, such as the temperature requirement for building circulation, differ per season, the ground heat exchanger is only component not implemented in both seasons.

Figure 4.6 Flowchart for ITC/TSM Operation

Previous research by the authors determined that implementing a PCM into the thermal storage tank for an ITC/TSM system could maintain savings with a 90% reduction in tank volume. A PCM was chosen with a melting temperature near the desired temperature for each tank. The solar panel and the ground heat exchanger are then used to ensure the water is oscillating the melting point of the PCM. Information on the TRNSYS component modification, the numerical method used for phase change approximation and model verification can be found in [67].

The following section details the TRNSYS components that were used to control the water temperature and flowrate operating regime.

Thermal Energy System Specialist (TESS) Component 539, a glazed flat plate solar collector, was implemented to heat the water when conditions were favorable. Parameters for the solar panel were chosen using experimental data from a Solar Rating and Certification Corporation (SRCC) data sheet [62] and can be seen in Table 4.5.

Table 4.5 Solar Panel Parameters

Solar Panel Parameter	Value
Intercept Efficiency (a_0)	0.751
First Order Efficiency Coefficient (a_1)	19.868 kJ/(h m ² K)
Tested Flowrate per Unit Area	304 kg/(h m ²)
First Order Incidence Angle Modifier (IAM) Coefficient	0.156

This TESS component was selected for the internal ability to control the outlet temperature using flowrate variability, and to cease flow if the collector was losing energy. These stipulations kept temperatures below 60°C from entering the hot water storage tank. In preliminary simulations, the hot water storage tank struggled to maintain the required temperature for DHW usage when water circulated to the tank without any regulation on inlet tank temperature. This led to the inclusion of the cold water storage tank. When the total flowrate was not used by the solar panel, the remainder was diverted to the cold water storage tank. This is shown as the first tee intersection in Figure 4.6. If the solar panel is losing energy (i.e. evening), the total flowrate was diverted to the cold water storage tank. During either of these conditions, the hot water tank would supplement flow from the main water line to support DHW usage. Both the hot water storage and the cold water storage do not accumulate mass in their

respective tanks. The exit flowrate for each tank is equal to the inlet flowrate. Therefore, if the hot water tank was not receiving the necessary flowrate from the solar panel, the main water line must provide the remainder flowrate to satisfy DHW usage. This method ensures the minimum hot water exit flowrate was the flowrate required by DHW usage. However, because the main water line was used, the tank auxiliary heating element must operate until the water reached the required 60°C. DHW was a required deliverable and primary before building circulation.

Both the hot water storage tank and the cold water storage tank both used PCM to reduce tank volume without reduction of energy storage. Paraffin wax was chosen as the PCM for both tanks since they have a large latent heat, a wide temperature range and are stable for thermal cycling [61] The melting temperature for paraffins range from -20°C to 100°C, the latent heat of fusion range from 200kJ/kg to 280 kJ/kg, the specific heat is near 2 kJ/kg K and the solid density is near 900kg/m³. The parameters for the phase change materials for the hot water storage and the cold water storage are listed in Table 5.6.

Table 4.6 PCM Parameters [61]

PCM Parameters	Value
Heat of Fusion	230 kJ/(kg)
Specific Heat	2 kJ/(kg K)
Density (Solid)	900 kg/m ³
Melting Temperature (Hot Water Tank)	57°C
Melting Temperature (Cold Water Tank)	15°C

After the required DHW had been diverted from the water loop, the flow was then cooled using the cold water storage outlet flow, shown as the second tee intersection in Figure 4.6. During the heating season, water circulating through the building at 60°C could have an impact on the building air temperature. If this impact resulted in the building air temperature begin higher than 26°C, the AC system would be required to cool the building. To reduce unnecessary loads on the AC system, the water is cooled to a moderate temperature using the cold storage tank. However, it still must be above room temperature to be eligible for building circulation. During the cooling season, the cold water storage tank mixing acted as a precooling process before circulation through the ground heat exchanger.

During the heating season, after the temperature of the water had been reduced, building circulation would occur if the temperature of the water was above the temperature of the building. If this temperature criterion was not met, the water would bypass the building and circulate back to the thermal storage tanks, shown as the return with building bypass in Figure 6. During the cooling season, the water was first passed through a ground heat exchanger to further cool the water. The ground heat exchanger had 2 horizontal pipes buried at 8 meters and runs around the perimeter of the building. If the ground heat exchanger exit temperature was below the temperature of the building, the water would circulate through the building shell. Otherwise, the water would bypass the building and return to the thermal storage tanks.

When water was circulating through the building, it was diverted to 11 different pipes, one for each external surface of the building. The flowrate through each of these surfaces was designed to have the same flowrate per unit area, therefore walls with a larger area receive a larger portion of the total flowrate. Each wall flowrate is listed in Table 4.7 as a percentage of the total inlet flowrate.

Table 4.7 Percentage Flowrate for Surfaces

Surface	Percent of Inlet flow
Large External Wall	6.2%
Small External Wall	4.8%
Roof	14.4%
Floor	27.2%

Preliminary studies found energy savings were maximized when the pipe distance to the internal side of the wall was minimized. Future studies could include an extensive study on the relationship of pipe placement and pipe dimensions to system savings. However, that is outside the scope of this study. Pipe spacing, outside diameter, and wall placement can be seen in Figure 4.7

Figure 4.7 Pipe Dimensions in Building Shell

In summary, there are four season and temperature dependent valves that controlled water circulation. The first valve controlled the flowrate to the cold water storage tank and solar panel, the second valve controlled the DHW outlet and remainder flow used for building circulation, the third valve controlled the ground heat exchanger circulation, and the fourth valve controlled building circulation. Table 4.8 shows the controller logic for both seasons. The DHW valve is excluded since it is neither temperature nor season dependent.

Table 4.8 ITC/TSM Valve Operating Regime

Valve	Component	Heating Season	Cooling Season
Splitter 1	Solar Panel	Collector is gaining energy	Collector is gaining energy
	Cold Water Storage	Else	Else
Splitter 2	Ground HEX	Never	Always
	Ground HEX bypass	Always	Never
Splitter 3	Building	$T_{\text{Water}} \geq T_{\text{Building}}$	$T_{\text{Water}} \leq T_{\text{Building}}$
	Building bypass	Else	Else

A parametric study was conducted over the solar panel size, water tank size and the total flowrate. The purpose of this study is to determine the parameter value for each component in a reasonable range that provides the greatest savings. Ranges for each variable are listed in Table 4.9. The upper limit of the solar panel is constrained by the size of the south facing roof which is 60m².

Table 4.9 Parametric Study Range

Parameters	Range
Solar Panel Size	14m ² -58m ²
Water Tank Size	0.8m ³ -2.4m ³
Flowrate	700 kg/hr-3500 kg/hr

Five values were chosen for each parameter for a total of 125 simulations.

4.2 Results and Discussion

The required heating, cooling, and the DHW heating element rates are used in this section to find the annual consumed energy for the reference case and the parametric study cases with the proposed ITC/TSM system. Instead of discussing the energy requirements for each component, the total energy is presented. This is done not only for conciseness, but also to represent the MW-hrs a resident would need to purchase to satisfy the annual heating, cooling and DHW needs. The total required energy is the addition of the integrated heating rate, cooling rate, and DHW heating element rate. The absolute values of the energy extracted or added are used, despite heat being added to the building system during the winter season and removed from the system during the cooling season.

All simulations in this parametric study were able to outperform the results for the reference case. The required energy for the reference case was 27.5 MW-hrs, while the simulation with the most modest results only required 20.7 MW-hrs, a 24% savings. Alternatively, the most beneficial case required 17.9 MW-hrs, a 35% savings. Using the electricity cost in Atlanta Georgia of 10.96 cents/KW-hr [68], the monetary savings for the best case scenario is \$1320 per year and the worst case \$1010.

To visually show the relation of each parameter to consumed energy, three surface graphs were constructed for a low, intermediate, and high values of each parameter. Figure 4.8 shows three different flowrate values, 700 kg/hr, 2100 kg/hr and 3500 kg/hr.

700kg/hr Flowrate

2100kg/hr Flowrate

3500kg/hr Flowrate

Figure 4.8 Constant Flowrate: Energy Consumption vs Tank Volume and Solar Panel Area

The consumed energy shows similar trends for the evaluated flowrates. While there are some differences, such as the placement and value of the minimum point, the graphs trends are consistent. All three graphs have the maximum consumed energy at the smallest solar panel size and the largest tank value. This is an expected result since the 14m² solar panel limits the hot water input to the storage tank. The tank heating element is used in cases when the DHW requirements cannot be satisfied. When these conditions are coupled with the large thermal capacitance associated with the 2.4m³ tank, benefits are reduced.

An unexpected result is the relationship between the minimum point and tank volume. As the flowrate decreases, the minimum value shifts to lower tank volumes. This could be attributed to the large thermal mass of the tank requiring greater circulation through the solar panel to continually heat the water. The lower circulations might not be able to deliver the required load, and be more dependent on the DHW heating element. Therefore, a smaller tank could be beneficial in the low circulation application. Because of the consistency between the flowrate graphs, it appears that flowrate contributes less to overall energy savings than the solar panel size and the tank volume. This is further illustrated in Figure 4.9, which shows the percent savings for each of the three flow rates for an intermediate tank volume and all solar panel sizes.

Figure 4.9 Flowrate and Solar Panel Size vs Percent Savings for Constant Tank Volume

Results indicate that the flowrate has some impact on savings, but it is not as significant as other parameters. This graph can be compared to those presented for the tank volume (Figure 4.13) and the solar panel size (Figure 4.15) later in this section. To further illustrate the lack of dependence on flowrate, Figure 4.10 and Figure 4.11 show the relationship between the building temperature and flowrate for a heating and cooling day respectively. The solar panel size is held constant at 36m² and the tank volume is held constant at 1.6m³. Both figures shows that the building temperature does not vary significantly for different flowrates.

Figure 4.10 Building Temperature vs. Flowrates on January 21st

Figure 4.11 Building Temperature vs. Flowrates on July 21st

The next parameter to discuss is the tank volume. Three tank volumes, 0.8m^3 , 1.6m^3 , and 2.4m^3 , are shown in Figure 4.12

0.8m³ Tank Volume

1.6m³ Tank Volume

2.4m³ Tank Volume

Figure 4.12 Constant Tank Volume: Energy Consumption vs. Mass Flowrate and Solar Panel Area

The parameter values associated with the maximum energy consumed are consistent with those shown for the mass flowrates above. The overall maximum of these three graphs occurs

when the solar panel is 14m^2 , the tank size is 2.4m^3 , and the mass flowrate is 700kg/hr , i.e. when the solar panel and the mass flowrate are at their minimum and the tank size is at its maximum. However, the maximum point for the 1.6m^2 is unexpectedly at the second lowest flowrate. This value is not significantly different than the value found at 700kg/hr . It is only 1% greater from the value shown as the maximum. Therefore, the maximum value is essentially consistent for the flowrates of 1600kg/hr and 700 kg/hr . This same logic can be applied to both the 0.8m^3 and 1.6m^3 graphs concerning the minimum value. The minimum value is expected to occur at the mass flowrate and solar panel maximum, but neither of these two points occur at this location. However, the value at the expected minimum location is 0.07% and 0.3% greater than the minimum value shown in the graph. Therefore, it is reasonable to say that the minimum also occurs where it would be expected at the largest solar panel area and the largest flowrate.

Figure 4.13 shows the relationship of the tank volume and solar panel size to the percent savings. Values shown in the graph have a constant flowrate of 2800kg/hr .

Figure 4.13 Tank Volume and Solar Panel Size vs Percent Savings for Constant Mass Flowrate

Figure 4.13 emphasize the fact that the solar panel and the water tank must be sized together. The combination of the small solar panel with the large water tank provide the least amount of savings. If a 14m² of solar panel area is implemented, a small water tank is favorable over a large storage tank. A large thermal mass is not as beneficial unless the temperature of it can be adequately controlled. From the figure above it can be seen that even when the maximum area of solar panel is used, the largest tank is not preferred.

The three surface plots for 14m², 36m², and 58m² solar panel surface area are shown in Figure 4.14.

14m² Solar Panel

36m² Solar Panel

58m² Solar Panel

Figure 4.14 Constant Solar Panel Area: Energy Consumption vs. Mass Flowrate and Tank Volume

Unlike the two previous parameters, the three graphs for the solar panel do not have similar trends. This is due to the high influence the solar panel size has on the results of the

simulation. In the 14m^2 graph, the maximum point occurs at the tank volume maximum and flowrate minimum. The required energy is shown to be reduced as the tank volume decreases and the flowrate increases. It should be noted again that the expected minimum point, located at maximum flowrate and minimum volume, is only 0.4% greater than the minimum point shown. The 58m^2 graph mirrors the trend seen in the 14m^2 graph, however the maximum point occurs at the tank volume and the flowrate minimum as expect from previous results. Again, the expected minimum point, located at the maximum flowrate and maximum volume, is only 0.03% greater than the minimum point shown. On the other hand, the 36m^2 graph is of interest because a shift can be seen in the location of the maximum and minimum point. The minimum is now located at an intermediate tank volume. Though the maximum is still located at 2.4m^3 , the magnitude has been reduced significantly compared to the 14m^2 case. The maximum point on the 36m^2 graph is nearly the same value as the minimum point for the 14m^2 graph.

Figure 4.15 shows the relationship of solar panel size and mass flowrate to the percent energy savings for a constant tank volume of 1.6m^3 .

Figure 4.15 Solar Panel Size and Mass Flowrate vs Percent Savings for Constant Tank Volume

Figure 4.15 reiterates that the impact of flowrate is less significant than other parameters for overall savings. All five flowrates shown have similar results for the three solar panel sizes. Alternatively, it can be seen that the size of the solar panel has a large effect on savings.

Though the largest solar panel provides the maximum savings in nearly all cases, the capital investment required would be large. Currently the cost of a hot water solar panel used in pool water heating applications is around \$150/m² [69], this can be a significant initial cost compared to the other components. Implementing the total area of the roof for solar panels provides the largest savings, but the initial cost of that system could be reduced by nearly \$3000 if the 36m² area is used instead. The average savings for the 56m² panel (across all flowrates and tank sizes) is \$1000 annually, and the average savings for the 36m² is \$950 annually. It would

take 60 years to make up the additional investment of \$3000 when only saving \$50 extra per year. Therefore, the maximum solar panel size is not the optimum choice. If the 36m² solar panel is chosen, referencing Figure 4.13 shows that a tank size of 1.2m³ is ideal and provides the largest savings which is \$1000 annually when the flowrate is maximum. The final and least significant contributor is the flowrate, reducing the flowrate to its minimum reducing savings by \$50 annually. Therefore, the pump cost could determine which flowrate would be the ideal condition.

4.3 Conclusions

This chapter focused on establishing the benefits associated with an ITC/TSM system with respect to energy savings. Increase thermal capacitance was achieved through water circulating in copper pipes embedded in the walls of the building. Thermal storage was controlled through a PCM embedded water tank and three seasonal/temperature controlled valves. Four devices were used to control the temperature of the circulating water: a solar panel, a ground heat exchanger, and hot water and cold water storage tanks. Benefits for the system are defined as reduced energy consumption required by the heat pump and DHW heating element compared to the reference case. To maximize benefits, a parameter study was conducted for the solar panel size, tank volume, and mass flowrate. Over 125 simulations were conducted, and results found that the mass flowrate of the system has the least effect on the annual savings of the system. It was also found that the tank volume and the solar panel size must be size simultaneously to maximize results. It cannot be assumed that the maximum tank volume will provide the greatest saving, in the case of a medium sized solar panel the medium tank outperforms the largest tank. Additional capacitance without temperature control does not maximize savings for the system. The use of ITC/TSM system provides energy savings between

from 24% to 35% compared to the reference case. Savings were maximized at the tank, solar panel, and flowrate maximum values, and provided an annual savings of around \$1,320. Future studies of interest include nocturnal radiation for precooling purposes, an extensive parameter study on the PCM melting temperature, specific heats and density, and a parameter study on pipe placement in the walls of the building.

CHAPTER V

SUMMARY AND CONCLUSIONS

This dissertation first presented the current state of energy required for building thermal comfort and noted that approximately half of the energy required by a building is allocated to heating, cooling or domestic hot water. Chapter I primarily contains a literature review of thermal mass implementations (active and passive), thermal mass with PCM integration, and DHW integrated with PCMs. Two of the main obstacles for implementing increased thermal mass are the impractical size of additional thermal mass, and the hinderance a large thermal capacitance can be if preheating/cooling strategies are not incorporated. These were the two main objectives determined for this dissertation:

1. Thermally control the additional thermal capacitance.
2. Reduce the required size of the thermal capacitance.

Objective 1 is the primary focus of Chapter II, and objective 2 is attempted in Chapters III and IV. Chapter III builds upon the ITC/TSM design in Chapter II, while incorporating a more complex model and including a PCM. The inclusion of PCM was done for its unique ability to reduce volume of the thermal capacitance without reducing the energy storage capability of the thermal capacitance. Chapter IV further builds upon Chapter III by integrating the ITC/TSM system into the DHW system and investigating the design of the system through a parameter study.

Chapter II focused on establishing merit of the ITC/TSM concept, and determining the main contributing factors associated with building energy savings. This was accomplished through the development of a first order model for a micro building. Increased thermal capacitance was achieved through water circulating in highly conductive pipes embedded in the shell of the building. Thermal storage management was implemented through the adaptive allocation and control of the additional thermal capacitance. Results of the first order study showed significant savings during the cooling season, however the winter only had limited savings. This was due to the systems inability to heat the water adequately for building circulation during the heating season. Three different locations were used to implement this system, Jackson, Mississippi; Atlanta, Georgia; and Jacksonville, Florida. Atlanta showed the least amount of savings as it has a lower average winter temperature than the other two locations. This further recognized the limited potential for implementations without heating schemes during the winter. This opportunity for improvement was pursued while transitioning from the first order study to the transient model contained in Chapter III.

Chapter III investigated the potential savings associated with the ITC/TSM system while using the modeling software TRNSYS. This transient model implemented a more realistic building than the one modeling in Chapter II, and additional weather data such as radiation, humidity, and soil temperature. Objective 1 and Objective 2 were both the focus of this study. The results in Chapter II showed that there could be an improvement regarding Objective 1. This was addressed by the inclusion of a solar panel for preheating the additional capacitance during the heating season. The precooling process used during the cooling season was still a water to air heat exchanger as modeled in Chapter II. Objective 2 was addressed by the inclusion of PCM in the water tank. This inclusion allowed a 90% reduction of the water tank volume without losing

energy storage capability. Results of this study showed a significant jump in savings pertaining to the heating season, but saw a loss of savings in the cooling season compared to the study in Chapter II. This was likely due to additional complexities included in the transient model, specifically those associated with solar radiation. Chapter II does not include radiation, therefore the building had significantly less external heat input than the study in Chapter III. Therefore, additional measures needed to be implemented to cool the water during the summer. Complete thermal control was not achieved in Chapter II or Chapter III, therefore Objective 1 was still not satisfied. However, Objective 2 had been satisfied by the inclusion of PCM and the 90% volume reduction potential of the water tank.

Chapter IV focused on the thermal control of the ITC/TSM system during the cooling season, the inclusion of DHW for additional savings, and the design and operation of the ITC/TSM system. Two components were added to the ITC/TSM system to help accomplish Objective 1. The first was a ground heat exchanger to reduce the temperature of the water during the cooling season. The second was a cold water storage tank to precool the water before circulation through the ground heat exchanger. The cold water storage tank was also a requirement for DHW implementation, without a separate cold water storage tank and hot water tank could not maintain the required temperature for DHW usage. Tank volume, solar panel area and mass flowrate were chosen for the parameter study to see their influence on total energy saving. Total energy savings is the savings associated with the heating, cooling and DHW electric resistance heater. All 125 simulations showed benefits from the ITC/TSM system. As expected, the simulation with the highest initial investment (largest water tank and largest solar panel) provided the greatest energy savings. Results also showed that the mass flowrate had minimal impact on savings, and that the area of the solar panel was the largest contributing

factor. This was expected since the DHW, not just the ITC/TSM system, was dependent upon the solar panel. In Chapter III, the solar panel only operated during the heating season for preheating the added capacitance, but Chapter IV included DHW, which is not seasonal, that also required solar panel operation. Therefore, even when the ITC/TSM system was not operating, savings could still be found from the solar panel integration with DHW. The volume of the water tank was also an integral factor for energy savings. However, this parameter was still coupled to the area of the solar panel. The additional capacitance (tank volume) temperature must be able to be controlled by the solar panel. If the size of the solar panel could not regulate the temperature of the water tank, i.e. if there was a large water tank and a small solar panel, savings would be reduced. Over all savings ranged from 24% to 35% when compared to the same building without the ITC/TSM system.

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