Analysis And Optimization Of A Solar Thermal Collector With Integrated Storage

2010

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ANALYSIS AND OPTIMIZATION OF A SOLAR THERMAL COLLECTOR WITH INTEGRATED STORAGE

by

MONICA FRANCES BONADIES
B.S. University of Central Florida, 2009

A thesis submitted in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering, Thermofluids in the Department of Mechanical, Materials, and Aerospace Engineering in the College of Engineering and Computer Science at the University of Central Florida Orlando, Florida

Summer Term
2010
ABSTRACT

Solar energy, a topic popular in the United States during the oil embargo of the 1970’s, has become a relevant topic once more with the current focus on reducing greenhouse emissions. Solar thermal energy in particular has become popular as it uses existing steam turbine technology to produce electricity, with the benefit of using solar energy to produce steam rather than coal or nuclear heat sources. Solar thermal can also be used at lower temperatures to heat water for pools or for residential use. While this energy source has its benefits, it has the problem of being opportunistic – the energy must be used as it is captured. With the integration of storage, a solar thermal system becomes more viable for use. In this work, a low temperature (50-70°C) thermal storage unit with a solar thermal collector is experimentally run then studied using both analytical and numerical methods. With these methods, suggestions for future developments of the storage unit are made. The prototype collector and storage combination tested worked best during the winter months, when there was low humidity. Furthermore, the heat exchanger design within the storage unit was found to work well for charging (heating) the unit, but not for discharging the storage to heat water. The best modeling method for the storage unit was the use of FLUENT, which would allow for the suggested changes to the prototype to be simulated before the next prototype was constructed.
This work is dedicated to my family and extended family, who have been a source of unconditional love and support through all the times of my life, whether dark or light. We have had many adventures, from living in California to traveling to England, and I am sure we will have many more. I love you all very much.

I also dedicate it to Chase Hansel - I love you. We’ve come quite a ways from making meatball subs in the measurements lab.

I dedicate it to the memory of my grandparents, Frank and Fran Bonadies, Peter and Esme Gehoski, who taught me to enjoy life and appreciate the gifts of nature – the song of a bird, frogs singing after a rainstorm, and snow in the wintertime. One day we will all meet again. I also dedicate it to the memory of my great-uncle Geoff Ware, a man whose many adventures and sense of humor were something to be admired.
ACKNOWLEDGMENTS

I want to especially thank Dr. Jay Kapat for his guidance and patience through my bachelor’s and master’s degrees. Dr. Kapat, thank you for helping me to follow my interests in engineering, encouraging me through my frustrations, and challenging me to explore new ideas. You are an incredible teacher and adviser, and UCF is fortunate to have you.

I also want to thank my committee members Dr. Louis Chow, Dr. Ali Gordon, Dr. Nicoleta Hickman, Dr. Muthusamy Swami, and Dr. Ratan Guha for their insights and support of my thesis work. I also must thank Dr. Guha for his management of the Ariel cluster, on which I ran my CFD cases.

I want to also thank the members of the Center for Advanced Turbines and Energy Research for their friendship and advice. Dr. Mark, Dr. Son, Carson, Jeff, Jason, Bryan, Michelle, Lucky, Greg, Nghia, Perry, Roberto, Tim, and Mark, I wish you the very best as you continue your work at CATER and beyond. I would especially like to thank Carson and Nate, who worked with me on senior design.

I want to thank my mentors from industry: Mike Gabriel, Chris Chandler, Chris Soileau, and Travis Kaltenbach of Gas Turbine Efficiency and Peter Szedlaczek, Ray Laster, and Scott Martin of Siemens Energy.

I also owe special thanks to Mrs. Waheeda Illsarie for her academic guidance throughout my undergraduate and graduate years. I also would like to thank the MMAE office staff for their help in ordering supplies and Mr. Steven Dick for his technical expertise.

Finally, I want to thank Dr. Carl Sagan for his inspiration and his care for our “pale blue dot.” I hope to continue in the spirit of his work.
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>LIST OF FIGURES</td>
<td>ix</td>
</tr>
<tr>
<td>LIST OF TABLES</td>
<td>xii</td>
</tr>
<tr>
<td>NOMENCLATURE</td>
<td>xiii</td>
</tr>
<tr>
<td>LIST OF PUBLICATIONS</td>
<td>xvii</td>
</tr>
<tr>
<td>CHAPTER ONE: INTRODUCTION</td>
<td>1</td>
</tr>
<tr>
<td>CHAPTER TWO: LITERATURE REVIEW</td>
<td>11</td>
</tr>
<tr>
<td>Solar Energy Progress</td>
<td>11</td>
</tr>
<tr>
<td>The Necessity of Storage</td>
<td>13</td>
</tr>
<tr>
<td>Photovoltaic Storage</td>
<td>16</td>
</tr>
<tr>
<td>Phase Change Materials</td>
<td>18</td>
</tr>
<tr>
<td>Inorganic Phase Change Materials</td>
<td>21</td>
</tr>
<tr>
<td>Organic Phase Change Materials</td>
<td>26</td>
</tr>
<tr>
<td>Thermal Conductivity Enhancement of Organic PCMs</td>
<td>31</td>
</tr>
<tr>
<td>Analytical and Numerical Simulation of Thermal Storage Unit</td>
<td>39</td>
</tr>
<tr>
<td>Computational Fluid Dynamics Simulations</td>
<td>49</td>
</tr>
<tr>
<td>CHAPTER THREE: PROBLEM DEFINITION</td>
<td>55</td>
</tr>
<tr>
<td>CHAPTER FOUR: METHODOLOGY</td>
<td>57</td>
</tr>
<tr>
<td>Experimental Procedures</td>
<td>57</td>
</tr>
<tr>
<td>Test Stand</td>
<td>57</td>
</tr>
<tr>
<td>Phase Change Material</td>
<td>64</td>
</tr>
<tr>
<td>Data Acquisition</td>
<td>64</td>
</tr>
<tr>
<td>Testing Procedure</td>
<td>67</td>
</tr>
</tbody>
</table>
Data Reduction...................................................................................................................... 69
Analytical and Numerical Procedures .................................................................................... 70
Analytical Model .................................................................................................................. 70
Numerical Model - Fluent..................................................................................................... 81
CHAPTER FIVE: RESULTS AND DISCUSSION..................................................................... 87
Experimental Processes ............................................................................................................ 87
Analytical Model ...................................................................................................................... 92
Numerical Model (FLUENT) ................................................................................................... 95
Comparisons of Results .......................................................................................................... 129
Validation of Analytical Model .......................................................................................... 129
Analytical vs. Numerical Model ......................................................................................... 130
Analytical Model vs. Experimental Data............................................................................. 131
Numerical Model vs. Experimental Data............................................................................. 133
CHAPTER SIX: CONCLUSIONS AND FUTURE WORK ..................................................... 138
Experimental Rig .................................................................................................................... 138
Analytical Model .................................................................................................................... 140
Numerical Model .................................................................................................................... 141
APPENDIX A: PUMP HEAD LOSS CALCULATIONS – HEAT LOAD SIDE ..................... 142
APPENDIX B: PUMP HEAD LOSS THROUGH HEAT EXCHANGER ............................... 145
APPENDIX C: EXPERIMENTAL DATA: CHARGING PROCESS ....................................... 149
APPENDIX D: EXPERIMENTAL DATA: DISCHARGING PROCESS ................................. 150
APPENDIX E: MATLAB CODE FOR DISCHARGING PROCESS ....................................... 151
APPENDIX F: MATLAB CODE FOR MELTING PROCESS............................................... 157
LIST OF FIGURES

Figure 1: Current Energy Use [1] ................................................................................................... 2
Figure 2: Worldwide Solar Availability [4] .................................................................................... 4
Figure 3: Family Hot Water Use Guide [5] .................................................................................... 7
Figure 4: Concept of Load Shift (adapted from[14]) ................................................................. 14
Figure 5: Phase Change Material Typology (Sharma et. al) [15] ................................................. 20
Figure 6: Multiple Phase Change Material Layout [47] ............................................................... 34
Figure 7: Schematic of Storage Unit Simulated by CFD [22] ...................................................... 36
Figure 8: Fin Regions [51] ............................................................................................................ 42
Figure 9: Control Volume [51] ..................................................................................................... 43
Figure 10: Cell Definition [51] ................................................................................................... 47
Figure 11: Energy Balance Across Fin [54] ................................................................................. 47
Figure 12: Computational Domain ............................................................................................... 51
Figure 13: Molten Salt Storage Tank Schematic .......................................................................... 52
Figure 14: Solar Thermal Rig ....................................................................................................... 57
Figure 15: Cross Section of Rig .................................................................................................... 58
Figure 16: Cross Section of Solar Thermal Collector Tube (Courtesy GEAR Solar) .............. 59
Figure 17: Heat Transfer Loops .................................................................................................. 61
Figure 18: Wax and Heat Exchanger Within Storage Unit ........................................................... 62
Figure 19: Heat Exchanger Assembly .......................................................................................... 63
Figure 20: Thermocouple Locations .......................................................................................... 66
Figure 21: Thermocouple Depths ............................................................................................... 67
Figure 22: Wax Curvature Around Fins [61] ............................................................................. 71
Figure 46: Melting Process (22 Fins): Temperatures ................................................................. 124
Figure 47: Freezing Process (22 Fins): Melt Fractions............................................................... 126
Figure 48: Freezing Process (22 Fins): Temperatures ............................................................... 128
Figure 49: Resistance Value Trends ........................................................................................... 130
Figure 50: Actual and Calculated Fluid Temperatures ............................................................... 132
Figure 51: Wax Temperature Predicted by CFD versus Experimental Results ....................... 134
Figure 52: Glycol Outlet Temperature Comparison ................................................................... 135
Figure 53: Freezing Temperature Changes ................................................................................. 137
# LIST OF TABLES

Table 1: United States Concentrating Solar Power Plants [4] ........................................................ 6

Table 2: Insolation Data for Orlando, Florida [6] ........................................................................... 8

Table 3: Thermal Storage Units .................................................................................................... 21

Table 4: Inorganic Storage for Solar Power Plants [33] ............................................................... 22

Table 5: Inorganic PCM Advantages and Disadvantages ............................................................. 26

Table 6: Organic PCM Advantages and Disadvantages [15] ........................................................ 31

Table 7: Heat Exchangers Studied by Medrano et al [24] .............................................................. 38

Table 8: Significant Parameters .................................................................................................... 54

Table 9: Central Florida Insolation Data ...................................................................................... 68
NOMENCLATURE

A  Area
E  Energy
g  Gravity
h  Heat transfer coefficient, enthalpy
H  Latent heat
I  Bessel function of the first kind
K  Bessel function of the second kind
L, l  Length
ℓ  Latent heat
m  Fin heat transfer parameter
.\textit{m}  Mass flow rate
q  Heat in Watts
q''  Heat flux
.\textit{Q}  Heat flow in Joules
R  Resistance
r  Radius
S  Melting or freezing wax thickness
s  Entropy
t  Fin thickness
T  Temperature
x  Horizontal distance
y  Vertical distance

*Acronyms*

<table>
<thead>
<tr>
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<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics</td>
</tr>
<tr>
<td>CNT</td>
<td>Carbon Fiber Nanotube</td>
</tr>
<tr>
<td>DSC</td>
<td>Differential Scanning Calorimeter</td>
</tr>
<tr>
<td>EG</td>
<td>Exfoliated Graphite</td>
</tr>
<tr>
<td>HTF</td>
<td>Heat Transfer Fluid</td>
</tr>
<tr>
<td>MWNT</td>
<td>Medium Weight Nanotube</td>
</tr>
<tr>
<td>PCM</td>
<td>Phase Change Material</td>
</tr>
<tr>
<td>SEM</td>
<td>Scanning Electron Microscope</td>
</tr>
<tr>
<td>TC</td>
<td>Thermocouple</td>
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*Greek*

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>α</td>
<td>Thermal diffusivity</td>
</tr>
<tr>
<td>β</td>
<td></td>
</tr>
<tr>
<td>δ</td>
<td>Thickness of freezing or melting front</td>
</tr>
<tr>
<td>ε</td>
<td>Effectiveness</td>
</tr>
<tr>
<td>η</td>
<td>Efficiency</td>
</tr>
<tr>
<td>μ</td>
<td>Kinematic viscosity, chemical potential</td>
</tr>
<tr>
<td>ρ</td>
<td>Density</td>
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<tr>
<td>τ</td>
<td>Time</td>
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*Subscript*

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<td>0,1</td>
<td>Order of Bessel function</td>
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1,2 Radius level
c Critical
cond Pertaining to conduction
conv Pertaining to convection
f, fin Pertaining to a fin
freeze Pertaining to the freezing process
fusion Pertaining to latent heat of fusion
inf Pertaining to mainstream temperature
in Pertaining to entrance of unit
l Liquid
m, mean Pertaining to mean temperature
melt Pertaining to the melting process
old Pertaining to previous time iteration
out Pertaining to exit of unit
pipe Pertaining to pipe portion of heat exchanger
s Solid
total Pertaining to a sum
unit Pertaining to storage unit
w, wall Pertaining to wall
wax Pertaining to wax
x Horizontal direction
y Vertical direction
LIST OF PUBLICATIONS

Conferences:


CHAPTER ONE: INTRODUCTION

From the beginning of the use of electricity to power lights and heat homes, it has found numerous other applications, particularly as society has advanced in terms of technology and industry. Now, electricity is used to power the computers required for a typical office to conduct business, power servers used to deliver internet access, and power the multitude of household appliances that have been developed to cook meals, do laundry, and heat water. First world countries use the bulk of this power, but nations such as China and India are industrializing quickly and so are beginning to require more electricity. The bulk of the electricity produced in the world is produced from fossil fuels, as seen in Figure 1. However, the renewable resource availability is several orders of magnitude larger.
The price of fossil fuels fluctuates with the world economy, but the price of their extraction and processing is definitely significant. To prepare oil or natural gas for consumption requires exploration to first find the oil or gas reserves in the earth’s crust. This is not always in hospitable areas; oil can be brought up from beneath the ocean or from beneath the ice in Siberia. The next step is bringing oil up from these places through wells, which require a sophisticated system of valves and pipes to keep the flow of crude oil at an acceptable pace. Offshore drilling requires an even more complicated rigging system to not only draw up oil from the earth’s crust, but maintain the oil in tanks so that it does not spill out into the ocean. Once the oil or gas is extracted from the earth, it must then be refined to remove impurities, which limit the performance of the machine they are used in, whether it is a gas turbine or a vehicle. This
process has been refined over time, but still takes quite a bit of time and money to accomplish [2].

As seen in Figure 1, much of the electricity produced worldwide is from nonrenewable sources. This is the small purple circle on the upper right. More remarkable is the larger orange and yellow circle, representing the potential energy generation from renewable sources. This greatly surpasses the nonrenewable resources currently used. As countries such as India and China become more industrialized, further electricity production is necessary to allow for new offices, industrial plants, and residences to be constructed. Furthermore, the expansion of the internet requires an expansion of electricity throughout these countries.

However, the conventional methods of producing electricity with fossil fuels produce carbon monoxide, nitrous oxides, and other chemicals that can produce smog in the atmosphere. This health concern along with the greenhouse effect produced from excess carbon dioxide in the atmosphere has lead to controls on the amounts of such emissions produced. But, a plan is necessary to prevent permanent damage to the earth. It must involve working with what power plants we have and planning for the future. The noted astrophysicist Carl Sagan suggested four such ideas in the television series Cosmos for energy solutions and for the betterment of mankind [3]:

1. Effective use of our existing resources – instead of cars that get 20 miles a gallon, how about cars that get 70 miles per gallon?

2. Development of alternative energy sources, specifically solar and wind

3. Reforestation on a grand scale

4. Work with our brothers and sisters, 1 billion of them living in poverty, to bring them up to better economic conditions.
The focus of this thesis is point two – the development of alternative energy sources, specifically in solar energy. There are many areas in the world which are underdeveloped and happen to also be areas where the sun is very present. Development of solar technology for these areas would allow for better quality of life as they would have access to electricity. These areas can be seen in Figure 2.

![Figure 2: Worldwide Solar Availability](image)

Solar energy has been a topic of interest over the past few decades, particularly with the fuel shortages found in the 1970’s and, more recently, public interest in global warming and possible climate change. The most visible sign of solar energy generation has been fields upon fields of photovoltaic cells, but another important collection method converts the radiation energy of the sun to thermal energy. This is known as solar thermal power, and can be used to
generate heat for domestic or industrial applications. On the large scale, solar thermal collectors transfer their heat to water to produce steam for steam turbines, which are then used to generate electrical power. During the oil embargo of the 1970’s, solar plants were constructed in the United States due to concerns about the availability of future oil and gas supplies from the Middle East. Plant construction slowed during the 1980’s when oil became more available again, but has made a comeback in recent times. The existing concentrating solar power plants in the United States are shown in Table 1.
Table 1: United States Concentrating Solar Power Plants [5]

<table>
<thead>
<tr>
<th>Plant Name</th>
<th>MW</th>
<th>Collection Method</th>
<th>Status</th>
</tr>
</thead>
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<tr>
<td>Maricopa Solar Project</td>
<td>1.5</td>
<td>Dish / Stirling Engine</td>
<td>Starts 2010</td>
</tr>
<tr>
<td>Saguaro Power Plant</td>
<td>1</td>
<td>Parabolic Trough</td>
<td>Started operation in 2006</td>
</tr>
<tr>
<td>Solana</td>
<td>280</td>
<td>Parabolic Trough</td>
<td>Under development</td>
</tr>
<tr>
<td>Abengoa Mojave Solar Project</td>
<td>250</td>
<td>Parabolic Trough</td>
<td>Starts in 2013</td>
</tr>
<tr>
<td>Apline SunTower</td>
<td>92</td>
<td>Power Tower</td>
<td>Starts in 2012</td>
</tr>
<tr>
<td>Blythe Solar Power Project</td>
<td>250</td>
<td>Parabolic Trough</td>
<td>Starts in 2013</td>
</tr>
<tr>
<td>Calico–Solar One</td>
<td>850</td>
<td>Dish / Stirling Engine</td>
<td>Starts in 2011</td>
</tr>
<tr>
<td>Genesis Solar Energy Project</td>
<td>250</td>
<td>Parabolic Trough</td>
<td>Under development</td>
</tr>
<tr>
<td>Imperial Valley–Solar Two</td>
<td>750</td>
<td>Dish / Stirling Engine</td>
<td>Starts in 2014</td>
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<tr>
<td>Kimberlina Solar Thermal Power Plant</td>
<td>5</td>
<td>Compact Linear Fresnel Reflector</td>
<td>Started operation in 2008</td>
</tr>
<tr>
<td>Nevada Solar One</td>
<td>72</td>
<td>Parabolic Trough</td>
<td>Started operation in 2007</td>
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<tr>
<td>Palen Solar Power Project</td>
<td>250</td>
<td>Parabolic Trough</td>
<td>Starts in 2013</td>
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<td>Rice Solar Energy Project</td>
<td>150</td>
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<td>Starts in 2013</td>
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<td>Sierra SunTower</td>
<td>5</td>
<td>Power Tower</td>
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<tr>
<td>Solar Electric Generating Station I</td>
<td>13.8</td>
<td>Parabolic Trough</td>
<td>Started operation in 1984</td>
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On the small scale, solar thermal collectors can help to generate hot water for domestic use. The underlying goal in the study is to supplement the hot water use for an average family. There are numerous ways in which hot water is used, from cooking to bathing, as seen in Figure 3.

![Average Hot Water Use](image)

From this table, estimation can be made as to how often a family of four will use hot water. With an estimation of preparing two meals per day, one shower per person per day, four loads of laundry per month, four loads of dishes washed per month, and four loads of dishes handwashed per month, the family’s hot water usage per month is 3000 gallons, or approximately 100 gallons per day. So, this gives a guideline as to the amount of energy required to supplement a conventional home hot water heating system.

The focus of this work is the optimization of an integrated collection and storage system for a single family home. The experimental portion of this work will be the construction and testing of a solar thermal energy storage unit with integrated collector. The collection system is commercially available from GEAR Solar and will be used to heat up the heat transfer fluid (in this study, glycol), then pump it within the heat exchangers. These heat exchangers are
embedded within the paraffin wax used as the PCM. This setup will be explained in more detail in the methodology chapter. Once this unit is successfully built and tested, its data will be used to inform the analytical portion of the work. When testing this setup, it is vital to know what tilt on the collector will give the maximum insolation. This data was provided by the Florida Solar Energy Center, and can be seen in Table 2. Using this data, the optimal tilt of the collector will be known for the months in which the collector is operated. Although the most insolation occurs during the summer months, the radiation is scattered by the humidity in the air. The angles from 25 – 30 degrees provide the best insolation regardless of time of year. An angle of 30 degrees is used for testing, as it is the easiest of the angles provided to implement on the unit.

**Table 2: Insolation Data for Orlando, Florida [7]**

<table>
<thead>
<tr>
<th>Tilt</th>
<th>Jan</th>
<th>Feb</th>
<th>Mar</th>
<th>Apr</th>
<th>May</th>
<th>Jun</th>
<th>July</th>
<th>Aug</th>
<th>Sept</th>
<th>Oct</th>
<th>Nov</th>
<th>Dec</th>
<th>Annual</th>
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<tr>
<td>0°</td>
<td>3.14</td>
<td>3.92</td>
<td>4.99</td>
<td>5.99</td>
<td>6.27</td>
<td>5.78</td>
<td>5.68</td>
<td>5.28</td>
<td>4.72</td>
<td>4.11</td>
<td>3.46</td>
<td>2.92</td>
<td>4.69</td>
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<tr>
<td>15°</td>
<td>3.75</td>
<td>4.43</td>
<td>5.30</td>
<td>6.05</td>
<td>6.10</td>
<td>5.54</td>
<td>5.49</td>
<td>5.24</td>
<td>4.89</td>
<td>4.53</td>
<td>4.06</td>
<td>3.56</td>
<td>4.91</td>
</tr>
<tr>
<td>20°</td>
<td>3.92</td>
<td>4.56</td>
<td>5.36</td>
<td>6.01</td>
<td>5.99</td>
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<td>5.18</td>
<td>4.90</td>
<td>4.63</td>
<td>4.23</td>
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<td>25°</td>
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<td>4.67</td>
<td>5.39</td>
<td>5.95</td>
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<td>4.75</td>
<td>5.39</td>
<td>5.85</td>
<td>5.67</td>
<td>5.07</td>
<td>5.06</td>
<td>4.99</td>
<td>4.86</td>
<td>4.75</td>
<td>4.49</td>
<td>4.04</td>
<td>4.93</td>
</tr>
<tr>
<td>35°</td>
<td>4.29</td>
<td>4.80</td>
<td>5.36</td>
<td>5.72</td>
<td>5.47</td>
<td>4.87</td>
<td>4.87</td>
<td>4.85</td>
<td>4.79</td>
<td>4.77</td>
<td>4.58</td>
<td>4.15</td>
<td>4.88</td>
</tr>
<tr>
<td>40°</td>
<td>4.37</td>
<td>4.82</td>
<td>5.31</td>
<td>5.56</td>
<td>5.24</td>
<td>4.63</td>
<td>4.66</td>
<td>4.69</td>
<td>4.71</td>
<td>4.76</td>
<td>4.64</td>
<td>4.24</td>
<td>4.80</td>
</tr>
</tbody>
</table>

The concept of thermal storage is not entirely new; beds of rocks or tanks of water have been used in the past to collect waste heat and use it to heat water or air. While these low temperature storage systems have been well developed, low temperature solar thermal systems to heat water have gone widely unused in the United States since the 1970’s and 1980’s, as the price of fossil fuels has decreased and until recently, some sort of compensation to the installer had not been available. However, there are many types of solar water heaters available for use, so there is no reason not to implement them and improve their designs. A recent discussion from
Jane Davidson [8] has several recommendations to increase the use of low temperature solar energy systems in the United States:

- Identify cost efficient materials for construction of the unit and energy storage
- From the start of building construction, include solar thermal units and / or storage
- Develop systems that provide heat as well as electricity
- Create combined photovoltaic and solar thermal units for domestic use
- Encourage the use of renewable energy at various levels of education, especially with funding at the university level.

The most widely used type of storage, as mentioned, is the use of a rock bed or large water tank to absorb waste heat for later use or to cool a building. This type of heat storage uses the sensible heat of the collection material; therefore the material does not change state during the storage process. Another method of thermal storage utilizes the latent heat of a material, in other words, its phase change. For this study, it is the latent heat of fusion (the melting point) of the material. Since the material requires energy beyond the sensible heat to change phase, this extra energy can be utilized to store thermal energy. This fact makes a phase change material more economical for thermal energy storage; the same volume of a phase change material can store more than the same volume of a material using only sensible heat storage. Therefore, this study will focus on the use of a phase change material for latent storage.

There are several goals to the study, the main goal being to analyze and optimize the collector and storage combination for replacing or supplementing a domestic hot water heating system. Under this goal fall the objectives of numerically modeling the temperature values in the wax at different locations during charging and discharging and the time to charge and discharge. These will be the most relevant values during the operation of the unit. The unit will be
optimized in terms of determining the most effective number of fins in the heat exchanger. The number of fins will affect the heat transfer from the heated fluid to the wax during the melting process and from the melted wax to the fluid during the solidification process. This efficiency will be defined as the amount of energy stored over the amount of energy received from the solar collector.

The design tested in this study uses a candle-grade paraffin wax and a simple heat exchanger design involving the use of copper tubing with copper fins brazed to the tubes. The design will be experimentally tested based upon various mass flow rates of heat transfer fluid and the extraction of heat from the storage unit. Furthermore, an analytical model will be developed and verified to check other parameters without the need for testing. A numerical model using FLUENT will be run to examine the location of the liquid-solid interface within the PCM, and this will be compared to the temperatures found by experimental work to determine the validity of the model.
CHAPTER TWO: LITERATURE REVIEW

Solar Energy Progress

While the concept of using the sun as an energy source is not new, many advances have been made in both photovoltaic and solar thermal power plants. This includes more efficient and thinner cells as well as advanced materials in the case of photovoltaic and better mirrors and heat transfer fluids in the case of solar thermal. According to a recent article by National Geographic [1], these and other advances are making solar power more cost-effective and leading to novel storage methods and solar solutions across the globe. Even Germany, which is cloudy much of the year, has become one of the world leaders in solar development with 2,220 Giga-Watt hours generated in 2006 (the United States produced 565 Giga-Watt hours) [1]. In sunnier Spain, several large solar thermal plants have been built, using both parabolic mirrors with tubes inside, and mirrors aimed at a “power tower” which collects the heat energy and then exchanges it to a steam turbine. The United States, meanwhile, experienced a boom in solar power plant construction during the oil embargo of the 70’s, however, this declined afterwards due to inexpensive fossil fuels as well as a change in political climate. Presently, as the price of oil and gas rises once again, solar technology has become more popular again. This rise in popularity will likely be more permanent with the financial support of the current administration.

The progress in solar energy and effective storage can be divided into two main areas: photovoltaic and solar thermal. In the photovoltaic field, the main improvements have been to create more efficient cells as well as thinner and lighter cells that could be implemented easily. However, a balance must be found between cost and deployability – the more efficient a cell, the
greater its cost. So, cells could be nearly painted on to any flat surface exposed to the sun, but they may not attain the average photovoltaic cell efficiency of 20% [8]. Batteries have also been utilized for storage, since rechargeable batteries have been available for several decades and can be easily applied here. However, the chemicals within batteries must be properly disposed of, so this poses an additional impact on the environment.

Since solar thermal power utilizes different collection systems based upon the desired outlet temperature, various contributions have been made to its use from better evacuated tubes for solar hot water heaters to better designs and materials for energy collection and storage for parabolic dish collectors. Another aspect of the solar thermal power system has been the use of existing power turbine cycles. A Sterling cycle can be utilized with a concentrating solar collector compressing air, but this does require some cooling. A Rankine cycle, on the other hand, can be used with a parabolic dish power plant, using the heat transfer fluid within the collector tubes to generate steam and run a turbine. Both these turbines are not new technology, so obtaining parts and repairing them is not a difficult task. This lends an advantage to the use of solar thermal over photovoltaic power for the near future.

There are also combinations of photovoltaic and solar thermal systems available, in which the excess heat from the photovoltaic cells is transferred by heat exchanger to a heat transfer fluid. This excess heat may then be used to heat water or supplement the heating needs of a building. This combination has been utilized in both industrial [10] and domestic [11] applications. The benefit of these systems is that more of the solar collector’s area is utilized even in a location that does not even face the sun. Furthermore, the combination produces a higher efficiency than photovoltaic cells alone – one domestic unit had an overall conversion
efficiency of up to 51% [11], with its photovoltaic cells having a conversion efficiency of only about 10%.

The Necessity of Storage

Several studies have revealed reasons to include storage as part of the solar collection system. These reasons include:

- Collected thermal energy available at night or with cloud cover
- No interruption in thermal energy supply [12]
- Melting temperature occurs over a range; system is relatively isothermal [13]
- Waste heat recovery [14]

Furthermore, solar energy is opportunistic. When the sun is out, its radiant energy must be taken advantage of for a solar collection system to have any value. With the proper storage unit, the load received mid-day, when the sun is at its peak, can be shifted for use at a later hour. This decoupling of the availability of solar energy and the load used by a residence or business increases the value of a solar energy power plant or hot water heater. This concept can be seen in Figure 4.
Storage is possible in both the solar thermal and photovoltaic systems, and both have been developed since the beginning of use of solar power. Storage for solar thermal systems will be covered in detail, and generally is done using wax, water, and other such organic materials. Traditionally, rocks beneath homes were also used for storing heat during the wintertime. However, photovoltaic storage stores electrical energy rather than thermal energy. Electrical energy is largely stored in batteries using chemical energy. This will be further explored in the literature review on storage.

Although this study will focus on the use of storage in solar thermal systems, it is beneficial to examine its use in other applications, for instance, the work of Asano et al [14] proposed such a system to collect the waste heat from a generator using paraffin wax. A small heat storage unit would be located in each home served by the generator, and these units would be connected in a loop. While this storage system does not use a solar collector, it does use a phase change material, which shows the benefits of storing thermal energy, no matter how it is produced.

Storage is applicable to an air conditioning or cooling system as well, not just a heating system. For example, a refrigeration system would be operated during off peak hours (usually at night, when the temperature is lower) to cool air or freeze water. The off peak electricity used by
the system is cheaper than peak period electricity in some areas [16], so the system makes economical sense to implement. Cold energy storage is best used when the air conditioning in a building is tied to a chiller plant – the chilled water is routed through the building through large pipes to cool it. An example of such a chilled water storage plant can be found in the recently built thermal storage tank at the University of Central Florida [17].

With the fact that storage is necessary argued, the type of storage material must be considered. Storage materials can be categorized into two main types – sensible heat and latent heat storage. The sensible heat makes use of a material’s sensible heat without a phase change occurring. A latent heat storage material, on the other hand, changes phase during the storage charging process. This phase change allows for more storage capacity per unit volume. Therefore, less mass of the latent heat storage material is required. During a phase change process, the temperature remains constant and the energy received by the medium melts it. Extra energy is needed to melt the medium, and this energy is released when the medium freezes. This extra energy is stored using the latent heat of the material. Latent heat is defined in (1).

\[ \ell = T \Delta s \]  

(1)

Enthalpy can be defined as shown in (2), and also corresponds to the change in energy of a system at constant pressure.

\[ h = Ts + \mu \]  

(2)

Since during a first order phase change, the chemical potential remains constant, the change in enthalpy then becomes:

\[ \Delta h = T \Delta s \]  

(3)
Photovoltaic Storage

Photovoltaic storage is done primarily using the chemical energy of batteries to store the energy. Studies of such a storage method focus upon the shifting of load from peak solar energy collection hours and the control system necessary to do so.

Palomino et. al [19] attached a 2.4 kW photovoltaic array with a 25.2kWh battery energy storage to a typical 1500 square foot house in Phoenix, Arizona to test the economic feasibility of load shifting of electrical power. Also attached to the house was an inverter to convert the DC voltage received from the PV array to AC voltage for residential use, as well as a control system, data acquisition system, and the necessary wiring for such a setup. Outside of the peak energy use period by the house, the photovoltaic energy was stored by the battery. This storage time was set by the local utility company. This load shift requires that the customer shift their energy use to take advantage of this power saving. For instance, if electricity costs more during the evening time in summer, it would behoove the homeowner to reduce their electrical use during this time. Furthermore, it would also be best if stored electrical energy was used during this time rather than drawing it from the grid. The main results of the study were that a battery storage system could be feasible for residential use and that load management is necessary for maximum money saving. However, the multiple wiring connections cause some loss through the resistance of the wires, and the battery itself was not 100% efficient. If a typical flooded lead-acid battery is used, the efficiency is around 70-80%. Additionally, during the collection of energy using the photovoltaic array, some of it will go to waste because of the losses within the electrical system. Finally there is the issue of battery recycling when the battery reaches the end of its useful life. This requires the treatment and disposal of such toxic materials as lead and lithium. The
materials must be considered not only for their environmental effects but also their long-term effects on the people who deal with their disposal.

Chiang et. al [20] proposed a control system for a residence with four distinct modes. During the night time, the electrical grid is used to charge the batteries in the storage system, and to supply energy use for the residence. In early morning hours, the photovoltaic array begins to charge the battery, and the excess is made up by the grid. During the daytime, the photovoltaic power is larger than the load consumed by the residence, so it is stored up to the maximum capacity of the battery and sent to the grid. Finally, during the evening, the stored energy is used to supply the energy needs of the residence. The components of this system include a PV array, a dc-dc converter, a single-phase battery storage system, and the power grid of the local utility company. The various modes are optimized using a maximum power-point tracking algorithm. The proposed system works well in terms of integrating with the grid, but would require an increase in energy costs to be economically feasible.

One of the major problems with providing power to remote islands is the difficulty of providing power transport lines from mainland areas. The solution has been to use small oil or gas power plants to generate electricity for the islands. Kaldellis et. al [21] propose the use of a photovoltaic system with energy storage to replace some of the fossil fuel based systems for islands. There are several dimensions to the development of this system, including the type of storage used, the sizing of the plant based on the irradiance of the island, the modes of operation, and the economic feasibility. The types of storage considered are fuel cells, lead acid batteries, sodium-sulphur batteries, Regenesys brand batteries, pumped-hydro storage, and flywheels. The main components of the system include photovoltaic arrays, the storage system, and the existing fossil fuel power station. Various operating modes are considered, but this study takes into
account several situations as they determine the use of the existing fossil fuel plant. During the daytime, the PV array and the storage may supply the electricity needs of the community. If there is excess electrical energy from the PV array, it is stored. If it is a cloudy day or the storage is empty, the fossil fuel plant will supply electricity. During the night time, the energy may be supplied by the charged storage or by the fossil fuel plant if the storage is empty. Furthermore, the event of an emergency is considered, during which all available electricity generation sources are put to use. The system sizing is calculated by first selecting the power produced by the PV / storage system and calculating the necessary dimensions. The energy produced by the PV is estimated, and from this the surplus electricity produced on an average day is calculated. The entire installation is then economically evaluated. As far as storage concerned, the storage technologies requiring the least amount of power were flywheels, lead acid and sodium-sulphur batteries. The most amount of power is required by fuel cells, and the pumped-hydro and Regenesys batteries fall in the middle. Overall, it was found that this storage proposal was economically feasible for a remote island.

**Phase Change Materials**

Reliable and cost effective materials must be chosen for the storage needs of residential solar thermal collectors and solar thermal power plants. Phase change materials are inherently superior to sensible heat storage materials because they can absorb heat energy without a significant change in temperature, as exhibited in equations (1) to (3). Furthermore, they have a larger energy density than sensible heat storage materials, so a smaller amount of material is needed to store the same amount of energy [18]. Another benefit is the range of melting temperatures offered by phase change materials. They can range from a range of 5-15 C for cool
energy storage to 100-175 C for storing concentrated solar energy from a focusing collector [18]. They also are used for the high storage temperature applications of approximately 300-500 C for solar thermal power plants.

Phase change materials have been used since the 1800’s and earlier [22] to store energy for later use. In earlier times, a typical application of a phase change material was using ice to keep food cool, a forerunner to refrigerators. Since then, leaps and bounds have been made in the development of new phase change materials to meet energy storage needs for domestic and industrial applications. Figure 5 summarizes the types of phase change materials by composition. These range from paraffin wax (organic) for domestic applications to molten salts (inorganic) for parabolic dish power plants. Eutectic mixtures may also be used for domestic applications, provided they are sufficiently nontoxic. Usually, to increase the effective thermal conductivity of an organic PCM, metal chips or brushes are added, and are generally stainless steel or a similar substance. Fins, whether embedded within the unit or attached to a heat exchange tube within the unit, can also serve this purpose.
Various types of PCMs have been used in previous studies to examine the best combination of PCM and storage system. These uses are summarized below in Table 3 and include different applications and PCMs to inform the state of the current technology.
Although this comparison provides useful information in terms of the capacity of the various forms of phase change energy storage, the two main branches of these materials should be explored in more detail with their pros and cons outlined. This will give more insight into the material and configuration decisions made in the current study.

### Inorganic Phase Change Materials

A summary of common inorganic storage materials can be seen in Table 4. These materials are used for the high temperature and high demand of power plants.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>kJ/kg</th>
<th>MJ/m³</th>
<th>Collector Location</th>
<th>Application</th>
<th>PCM</th>
<th>Thermal Conductivity Enhancement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reddy [23]</td>
<td>194</td>
<td>18.5</td>
<td>Integrated</td>
<td>Building Hot Water</td>
<td>Paraffin</td>
<td>Fins</td>
</tr>
<tr>
<td>Agyenim [24]</td>
<td>338</td>
<td>465</td>
<td>None</td>
<td>Experimental</td>
<td>Erythritol</td>
<td>Fins</td>
</tr>
<tr>
<td>Velraj [26]</td>
<td>336</td>
<td>400</td>
<td>Separate</td>
<td>Building Hot Water</td>
<td>Paraffin (RT-58)</td>
<td>Internal fins, lessing rings</td>
</tr>
<tr>
<td>Hamada [27]</td>
<td>362</td>
<td>298</td>
<td>Separate</td>
<td>Experimental</td>
<td>Paraffin</td>
<td>Graphite additives</td>
</tr>
<tr>
<td>Stritih [28]</td>
<td>6587</td>
<td>7604</td>
<td>Separate</td>
<td>Building Thermal Storage</td>
<td>Paraffin</td>
<td>Fins</td>
</tr>
<tr>
<td>Balikowski [29]</td>
<td>191</td>
<td>378</td>
<td>Separate</td>
<td>Experimental</td>
<td>Paraffin</td>
<td>Spiny Fins</td>
</tr>
<tr>
<td>Alva [30]</td>
<td>3529</td>
<td>4646</td>
<td>Integrated</td>
<td>Building Hot Water</td>
<td>Paraffin</td>
<td>Graphite additives</td>
</tr>
<tr>
<td>Erek [31]</td>
<td>334</td>
<td>336</td>
<td>Separate</td>
<td>Experimental</td>
<td>Water</td>
<td>Circular Fins</td>
</tr>
<tr>
<td>Pincemin [32]</td>
<td>Varies</td>
<td>Varies</td>
<td>Separate</td>
<td>Power Generation</td>
<td>Molten Salt</td>
<td>None</td>
</tr>
<tr>
<td>Steinmann [33]</td>
<td>Varies</td>
<td>Varies</td>
<td>Separate</td>
<td>Power Generation</td>
<td>Molten Salt</td>
<td>None</td>
</tr>
</tbody>
</table>
The inorganic phase change materials are divided mainly into molten salts and metallics. Metallics have not been explored in depth as the molten salts due to their weight. However, they have the highest thermal conductivity of any phase change material and a relatively low vapor pressure [16]. Molten salts, on the other hand, are widely used for a range of solar thermal power plants. They have a high latent heat of fusion and a small volume change when they melt. Also, they have higher thermal conductivity than the organic phase change materials. These salts melt in a range of 300C to 500C and are used to produce superheated steam in a large scale solar thermal power plant [34]. Although advances have been made towards using the molten salts for the heat transfer fluid, they are mainly used as storage materials for these plants. One of the main
reasons for requiring storage for this scale of an operation is that it assures a predictable run of the plant – power generation cannot be stopped by a cloudy day or the nighttime. Numerous studies have been conducted on the suitability of inorganic phase change materials for storage, including methods to enhance their thermal conductivity as well as the development of filler materials to reduce the cost of storage.

The benefits of storage for a power plant have been identified by Herrmann, with the main one being that storage will help with even power production [35]. This is particularly important for a solar thermal plant since the load may change due to weather conditions. The ideal storage medium for a solar thermal plant is cost effective, able to transfer heat effectively from the HTF, interacts favorably with common engineering materials, particularly those used to construct a parabolic dish solar thermal plant. It is noted that latent heat storage is preferable due to a higher energy density. With this higher energy density, the cost of latent heat phase change materials is less than comparable sensible heat storage materials. Much of the work in creating efficient solar thermal power plants was done before 1990, with not many changes being made in the past decade. However, the development of storage materials has formed the focus of recent work, and will help to make the construction of solar thermal power plants cost-effective even when compared to their fossil fuel counterparts. This is especially important for increased acceptance by the public of solar thermal power plants, as they will not have a steep increase in cost of their electricity.

Since the inorganic phase change materials used in solar steam power plants have an excellent heat of fusion value but a poor thermal conductivity value, devices such as an embedded tube heat exchanger can be implemented. This helps the storage system achieve a storage time and discharge time of several hours, which is necessary for a solar thermal power
Three ideas were tested by Steinmann – placing the molten salt between graphite sheets, infiltrating a graphite matrix with molten salt, and encapsulating the molten salt with a higher conductivity material [33]. At the time of publication, these prospective storage solutions had not been tested in the laboratory but based upon their individual heat capacities, they showed promise for application in a solar thermal power plant. Upon successful laboratory testing, the most successful of these three methods would be implemented at a solar thermal power plant to see how it would behave under actual plant operating conditions.

Graphite is the material of choice for thermal conductivity enhancement for molten salts, and it can be utilized in several different ways. Both chloride salts and nitrates were examined by Pincemin et. al, though nitrates were chosen because they were less corrosive and less toxic than the chloride salts [32]. These salts were impregnated into a graphite matrix, and several types of graphite were used here. The resulting materials were evaluated based on their capacity to store heat. The routes to produce these materials were compounding (mixing the graphite with PCM at or above its melt temperature), infiltration (PCM is injected into expanded natural graphite), and cold compression (PCM and graphite powder are mixed at the PCM’s melt point, then they are cooled and molded. These three methods involved different forms of graphite. These include graphite powder, expanded natural graphite, and naturally occurring graphite flakes.

Molten salts are the material of choice for a solar thermal power plant’s storage system. However, a conventional storage system uses two tanks, both filled with the molten salt materials (usually a mix of Sodium Nitrate and Potassium Nitrate salts). These materials can get fairly costly, so a filler material could help reduce the cost while maintaining the storage capacity of the tank. Brosseau et. al examined quartzite rock and silica sand as filler materials [37]. These
two materials were tested isothermally and cycled with the molten salts. After these tests, the filler materials were studied both qualitatively and quantitatively. The quantitative studies were carried out using SEM and EDS. At the end of the testing and analysis, it was found that both materials withstood the molten salt satisfactorily, but field testing would be necessary to successfully utilize them in a future parabolic solar plant and better determine their behavior at plant conditions.

An earlier study concerning the use of fillers in thermal energy storage was done by Pacheco et. al [38]. About 15 materials were chosen for testing, including bauxite, marble, and limestone from two different states. These materials would be placed in a thermocline tank along with the existing molten salt and used to store thermal energy for a parabolic dish solar power plant. The materials were thermally cycled and analyzed at different times as well as analyzed isothermally. The actual thermocline tank for use within the parabolic plant was also tested at comparable plant conditions. The single tank design tested was found to be about 2/3 of the cost of a typical two tank storage system, using quartzite as the filler.

The final step in the development of molten salts as storage materials is their implementation in a solar thermal power plant. Most recently, the construction of Solar TRES in Spain has been a flashpoint for research and development of a suitable system including molten salts for not only storage but heat transfer fluids as well [36]. The starting point of Solar TRES was based upon Solar TWO, which made some very important contributions toward the field of solar thermal power (the main one being a 97% storage efficiency). One interesting point is that in Europe, these power plants are developed as semi-commerical interests, i.e. companies must show that the plants are commercially viable and will create just as much power as existing fossil fuel power plants. A big part of Solar TRES is utilizing molten salt both as the HTF and as the
storage fluid. This increases the plant capacity and ensures that the plant can convert energy just as well as conventional fossil fuel power plants. Another environmental benefit is that the plant will require less CO2 mitigation than a comparably sized coal power plant. For this project, nitrate salts are used. From the sensitivity analysis done by the authors, the storage capacity of the plant will range from 10-20 hours, which would nearly guarantee operation overnight and on cloudy days, particularly in the sunny climate of Spain. Some of the other innovations in the plant include a more efficient steam turbine as well as improved control of the field of mirrors, a better pumping system to circulate the salt, and most importantly a larger thermal storage configuration. As for the cost of such a plant, it is estimated that it will be slightly more cost per kWh than an existing comparable coal power plant.

With these studies in mind, the advantages and disadvantages of inorganic phase change materials can be summarized. These are presented in Table 5.

<table>
<thead>
<tr>
<th>Table 5: Inorganic PCM Advantages and Disadvantages</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Advantages</strong></td>
</tr>
<tr>
<td>Low volume change</td>
</tr>
<tr>
<td>Temperature range</td>
</tr>
</tbody>
</table>

*Organic Phase Change Materials*

Organic phase change materials are mainly classified into two groups – paraffins and non-paraffins [16]. The paraffins exhibit similar physical properties, since they are increasingly large chains of alkanes. Longer chains of alkanes mean a higher melting point. Furthermore, paraffins of different melting points can be mixed together to create different melting temperatures. Although paraffins do not exhibit supercooling, they do have a low thermal conductivity and are flammable. Numerous studies of thermal storage units have used paraffin
for the phase change material due to its availability and relatively inexpensive cost.

Furthermore, one of the benefits of using a phase change material such as paraffin in the first place is that it continues to absorb heat after it has melted. This provides the benefits of both the latent and sensible heats. Paraffin provides a range of melting temperatures, which must be considered when designing the storage unit. To accurately determine the temperature range, some researchers have used a differential scanning calorimeter (DSC) to analyze the material in their work [39]. The output of a DSC will be useful in the design of the specific storage unit, since melting may begin earlier than expected if one relies only on the paraffin manufacturer’s given data.

The use of a DSC has been found to not only give data on the range of melting points, but also the latent heat of fusion. However, depending upon the speed of scanning, different results may be derived. He et. al [40] recommend that a low speed be used for better results. Furthermore, this work talks about how to accurately analyze a DSC curve when dealing with a binary mixture of two different paraffins – in this case, tetradecane and hexadecane. Another consideration when using a DSC is the difference between the melting process and the freezing process, and the fact that the output curves of watts per mass versus time or temperature will be steeper for a pure substance. DSC measurements can also be used to determine the storage density of the material by using the overall change in enthalpy and integrating it over the amount of time the peak change in enthalpy occurs during. Furthermore, the DSC gives five significant temperatures along the curve designating relevant temperatures and the beginning of the melting and freezing zones, but the authors recommend that three of these five temperatures be used for the most accurate results. These three temperatures are located towards the center of the curve,
not at the extrapolated ends of it. This will give more accurate data when considering the critical
temperatures and latent heat of fusion of the material being tested.

Paraffin has come into use for thermal storage units within the past two decades, and it is
important to identify other portions of the storage system that affect the speed of melting and
freezing. A study of a typical scientific-grade paraffin was undertaken by Akgun et al [41] to
examine these parameters. A differential scanning calorimeter was used to find the heat storage
capacity, melting point temperature, and density of the paraffin since the grade used was
commercial. This grade of paraffin generally has more impurities than laboratory grade paraffin,
and this affects the latent heat of fusion values as well as the melting and freezing points. For
this study, water is circulated through a copper tube, outside which is the PCM, and the PCM is
enclosed radially by a stainless steel tube. The unit was studied for charging and discharging for
different temperatures and flow rates. As would be expected, with a higher temperature when
charging the storage, it takes the PCM less time to melt. The same is also true for the increase of
HTF mass flow rate – it reduces melting time – but the higher mass flow rate is not
recommended by the authors as it requires more pumping power. This initial analysis shows the
benefits of the paraffin (low toxicity, non-corrosive) but identifies the fact that more surface area
is needed for heat transfer due to its low conductivity.

Paraffins have been the focus of the organic phase change materials due to their wide use
in candles and other commercial products, but they are certainly not the only one. Non-paraffin
phase change materials include glycols, acids, and esters. Some of these materials may vary only
slightly from each other in chemical formula, but have very different physical properties. Of the
organic non-paraffin phase change materials, fatty acids are the most similar to paraffins and
thus show the most promise for future application. Another aspect of fatty acids is that they can
be derived from renewable plant and animal sources [42], which adds environmental sustainability.

Fatty acids can be mixed together to achieve different melting points, just as two paraffins can be mixed as well. Tuncbilek et al [42] considered palmitic acid and lauric acid for use in solar thermal energy storage as their melting temperatures are around the 50°C minimum for home hot water heating systems. Furthermore, they can be produced from animal or vegetable oils, hence they are environmentally viable. In this study, a mixture of about 70% lauric acid and 30% palmitic acid was mixed and tested as the phase change material for a thermal energy storage system. The mixture was found to be viable for a melting temperature range of 34-37°C, lower than what would be required for a home hot water heating system. However, the system did display congruent melting and the other good characteristics of fatty acids for use in thermal energy storage. It was found that in the mixture of lauric and palmitic acid that more palmitic acid resulted in a higher latent heat of fusion as well as a higher melting temperature. So, pure palmitic acid or a mixture high in palmitic acid would be more feasible for heating water for domestic use.

It is important to compare the usefulness over time of paraffin to other organic phase change materials, particularly fatty acids. Sharma et al [39] tested three different materials for 1500 cycles to determine the whether their melting temperature and the latent heat of fusion degraded over time. This amount of time was meant to simulate approximately five years of use, with one cycle corresponding to one day. One cycle here refers to complete melting and complete freezing of the material. These were commercial grade materials, so the performance over time needed to be evaluated based upon the fact that they had some degree of impurity. As well as the paraffin, acetamide and stearic acid were tested. Of these three, the acetamide
experienced the lowest drop in latent heat of fusion. However, the acetamide absorbed moisture over time, so this would affect the melting and freezing points of the material.

Another cycling study of organic phase change materials was done by Shukla et al [42]. In addition to paraffins at three different melting temperatures, the material erythritol was tested as well. Erythritol is a sugar alcohol used for flavoring in sugarless gum, and is widely available. It melts at approximately 120°C, which makes it suitable for use in medium temperature range latent heat storage. Alternatively, with an appropriate design, it could be used for home heating, exchanging heat to water to be run through pipes in the floor of a house. This could help augment the use of a typical heating system in the fall.

Along with other organic materials, fatty acids show promise as phase change materials for low temperature latent heat storage. They have the benefit of low environmental impact as well as a high latent heat of fusion, different melting temperatures to suit different applications, no noticeable decrease in properties after many cycles of melting and freezing, and have a low corrosion. Sari et al [44] impregnated different types of fatty acids into exfoliated graphite (EG) to increase their thermal conductivity. Before any differential scanning calorimeter (DSC) testing was completed, it was found that the density of the mixture of fatty acid and EG was less than that of the pure fatty acid. This would be a benefit in the fact that a storage unit with the mixture of fatty acid and EG would weigh less than pure fatty acids. Next, a DSC was used to determine the latent heat of fusion and melting point range for the mixture of EG and fatty acid as well as the pure fatty acid. Examining the DSC curves, the mixture of EG and fatty acid maintained nearly the same properties (latent heat of fusion, melting temperature range) as that of the pure fatty acid. The mixture of EG and fatty acid was able to melt faster than that of the
pure fatty acid by about 20%. However, the freezing process time decreased by about 50%, as the EG served as nucleation sites for freezing of the fatty acid.

Since paraffins and other organic PCMs are used for domestic hot water units based upon their wide availability and low toxicity level, their specific detriments must be dealt with appropriately. The flammability of an organic phase change material will not be an issue if the storage unit is designed with this in mind and the appropriate safety factors integrated into the unit’s design. The volume change must also be considered when constructing the unit, i.e. about 10% should be allowed for as recommended by Farid et. al [39]. The low thermal conductivity, on the other hand, has numerous solutions, one of which [42] has been discussed. The focus of the present work will be on the improvement of thermal conductivity by addition of a finned heat pipe.

As indicated in Table 6 below, one of the main weaknesses of organic PCMs is their low thermal conductivity, which must be addressed to make them viable as efficient thermal energy storage. This will be discussed in depth in the following section.

<table>
<thead>
<tr>
<th>Advantages</th>
<th>Disadvantages</th>
</tr>
</thead>
<tbody>
<tr>
<td>Non-corrosive</td>
<td>Flammable</td>
</tr>
<tr>
<td>Chemically stable</td>
<td>Low thermal conductivity</td>
</tr>
<tr>
<td>Negligible subcooling</td>
<td>Large phase change volume</td>
</tr>
<tr>
<td>Does not react with most building materials</td>
<td></td>
</tr>
<tr>
<td>Low vapor pressure</td>
<td></td>
</tr>
<tr>
<td>Low environmental impact</td>
<td></td>
</tr>
</tbody>
</table>

Thermal Conductivity Enhancement of Organic PCMs

As mentioned previously, various efforts have been made towards the improvement of the thermal conductivity of phase change materials. These include a eutectic mix of PCM and a
metal, a finned heat exchanger with the PCM on the outside and the heat transfer fluid (HTF) on the inside, and encapsulation. A eutectic mix of a PCM and metal may be achieved in different ways, although the most common is to mix metal chips into the PCM when it reaches its melting point. Jegadheeswaran et al [46] review the most common enhancement methods for thermal conductivity enhancement. The various methods of increasing the conductivity of a phase change material are explored in depth. In particular, the use of fins and the addition of more conductive materials are explored. The use of encapsulation and multiple phase change materials is touched upon along with the methods of analysis to estimate the temperatures within the phase change material structure. Latent heat storage units are mentioned as superior for the reasons of high storage density and phase change at a constant temperature, as specified in equations (1) to (3). The fins are located on the phase change material side of the shell and tube arrangement, since the fins have more efficiency with a decrease in heat transfer coefficient. The fins have most effect during the melting process, since they increase the heat transfer area and allow for more natural convection in the system. This results in more movement of the melting fluid, which helps to move the melting front more quickly throughout the system, resulting in a shorter charging time. The use of fins can be described in either a rectangular cavity or in a shell and tube heat exchanger style arrangement. In the freezing stage of the material, fins are most effective at the beginning because there is some amount of natural convection. However, the bulk of the heat transfer is by conduction in the freezing stage, so the main effect of the fins is as a nucleation site for the freezing. In this case, fewer fins help the process, so a fin arrangement must be optimized for both melting and freezing processes. Fins can also be implemented on separate loops for the melting and freezing processes, as the optimal number may be different for these two processes. So, the number and dimensions of the fins must be optimized to determine
the best operating conditions of the unit. The other main methods of enhancement come in the impregnation of a porous material (such as exfoliated graphite), mixing high conductivity particles in the PCM, adding metal structures, and using high conductivity particles specifically chosen for their low density. The low density stuff is most likely a nanofiber or nanoparticle. The uses besides fins will be explained to some extent but the finned enhancement will be explained in depth as this is the focus of this study.

Another idea is the use of nanomaterials to enhance the conductivity of an organic phase change material. For instance, Zeng et al [47] proposed the use of multi-walled carbon nanotubes in palmitic acid to increase the thermal conductivity. For the effective thermal conductivity enhancement, it is important to of course increase the conductivity but also maintain the high latent heat of fusion of the material. Also, the melting temperature of the material should be affected as little as possible. In this study, two different types of multi-walled carbon nanotubes are used with three different types of surface treatments. The surface treatments enable the carbon nanotubes to integrate into the palmitic acid mixture. At a 5% volume fraction of carbon nanotubes, the surface of the carbon reacts with certain branches of the palmitic acid that contain easily reacted carbon components. This leads to the absorption of the palmitic acid by the nanotubes and results in a lower latent heat of fusion. So, the amount of carbon nanotubes must be optimized to overcome this effect. With all levels of carbon nanotubes tested, there is not a significant difference in the DSC curves, which indicate the melting point temperature as well as the latent heat of the material. This indicates that the addition of nanotubes does not significantly affect the melting point and latent heat of fusion of the palmitic acid. The best combination was with the longer nanotubes and the surface treatment with nitric acid, as this optimized the surface of the tubes, and the tubes were longer, which provides more carbon
material to enhance the conductivity of the palmitic acid. Although the thermal conductivity of the palmitic acid was increased at maximum by 25%, it still is not as effective as the addition of a finned tube heat exchanger to the system.

Another idea explored to enhance the phase change thermal storage is the use of multiple phase change materials along with a finned tube, as studied by Seeniraj et al [48]. The main difference of this study as opposed to others with fins is the fact that it has multiple phase change materials. The PCMs were placed in the tube with their melt temperatures decreasing from the entrance of the tube, so that melting will occur as soon as possible. This configuration can be seen in Figure 6.

![Figure 6: Multiple Phase Change Material Layout [48]](image)

The governing differential equations and enthalpy relations were set up for a 3-D system, considering the melt fraction in the PCMs. The total energy was calculated using the sum of the sensible and latent heats. All energy equations were coupled and numerically solved, and the results compared to previous experimental work by other researchers. The results with 5 PCMs were compared to the results with just one PCM. The PCMs used were molten salt – LiF & CaF2, and a eutectic mixture of LiF2-MgF2. The melt temperatures varied depending upon the composition of the mixture, that is, how much of each salt was contained in the mixture. The
interface location, as well as the temperature of the HTF and tube wall, is reported for 1 PCM and 5 PCMs. The main result is that 5 PCM’s are better than one. The usefulness of this study is the fact that the PCMs are arranged in order of decreasing melt temperature; therefore the PCM at the end may still extract energy out of the heated fluid within the heat exchanger.

The finned heat exchanger method involves attaching fins to the pipes through which the heat transfer fluid flows. The fin shape, quantity, and material can be varied to examine the effects on the operation of the system. Finally, encapsulation is the placement of the organic PCM in capsules made of a material with a higher thermal conductivity. For the capacity of finned thermal storage units, refer back to the studies compared in Table 3. To determine the effectiveness of these phase change configurations, they should be compared to the latent heat of water – 333 kJ/kg, and its volumetric heat capacity 334 MJ/m³.

Thermal conductivity enhancement of organic phase change materials is necessary to make the storage of solar thermal energy viable for domestic and industrial applications. In the case of hot water production, the finned heat exchanger is most commonly employed. A finned tube has more heat transfer area than an unfinned tube, so the low thermal conductivity of the organic phase change material is not so much an issue. Three recent studies in particular can be examined in terms of their fin configurations, results, and analysis.

One of the simplest forms of enhancing thermal conductivity is the addition of rectangular fins within the phase change material. Of course, the configuration and number of fins must be optimized so that natural convection, which speeds the melting process, can take place. Also, the fins cannot be too far apart, as this would create too big an area for them to transfer heat across. Reddy [48] studied the configuration of rectangular fins within an integrated collector system with paraffin wax with embedded fins set beneath a specially coated
piece of glass. Solar radiation would be trapped beneath this glass and the thermal energy transferred to the paraffin wax. The wax in turn would transfer heat to the water bath beneath it, which would be pumped in during the night time. The entire unit was tilted to optimize the sun’s position in the sky as well. Once the boundary conditions and heat loss values were determined for this configuration, FLUENT was used to numerically analyze the temperature distributions for 0, 4, 9, and 19 fins within the paraffin wax. A schematic of the unit studied can be seen in Figure 7.

![Figure 7: Schematic of Storage Unit Simulated by CFD](image)

FLUENT can be used for this sort of problem, but it does not directly give the location of the melting front. Instead, a mushy region can be tracked, with 0 being completely solid material and 1 being a completely melted material. This modeling method was used along with a sinusoidal heat transfer to represent the daily variation of the sun. Results were given in terms of temperature contours and liquid fraction variations, which could be given after a certain amount of time. The most important result was that the nine-fin configuration maintained the best heat level, allowed for the maximum amount of natural convection, and had the highest average water temperature and lowest night time temperature drop. This study shows that although placing fins
in the phase change material does increase its thermal conductivity, there is a maximum amount of fins one can use to optimize the performance of the storage unit.

The next step up from just embedding the fins in the wax is to create a finned-tube system with fins brazed or otherwise manufactured around the pipe. In this design, the heat transfer fluid would flow through the tube and exchange heat to the wax by the fins attached to the outside. This method has been identified as the simplest to construct of the thermal conductivity enhancement schemes. Agyenim et al [24] compared a tube without fins to a tube with longitudinal fins and circular fins. The circular fins were placed at a uniform distance along the tube, while the longitudinal fins were rectangular shaped and placed at various angles surrounding the tube. They were used to increase the thermal conductivity of Erthritol for a medium temperature thermal energy storage unit.

The study is purely experimental and results are presented in terms of average temperatures within the phase change material. The temperature values are taken by strategically placed thermocouples within the Erythritol. The mass flow rate of the heat transfer fluid was varied during the experiment, which occurred over 8 hours as it was determined to be the maximum amount of solar availability in most of Europe. It was found that the longitudinal fins were the best at melting the PCM within the least amount of time; most of the PCM was melted within about four hours of charging. Qualitative analysis of convection within the system was done by examining the temperature contours displayed by the thermocouples at each time step. In addition to the valuable design recommendation provided by the study, the assumption that thermal conductivity in the axial direction can be considered negligible was verified by the experimental data. This can be considered true for similar models, and so simplifies their
analysis. This fact will be extended to the present study, as axial direction can also be considered negligible.

While new heat exchanger designs can be created for a thermal energy storage unit, prefabricated heat exchangers have been on the market for a number of years and can be also considered to help increase the thermal conductivity of an organic phase change material. Medrano et al [25] examined five different commercially available heat exchangers with paraffin wax (RT-35 by Rubitherm) to see which ones were best suited for thermal conductivity enhancement. Each heat exchanger was composed as shown in Table 7 below.

<table>
<thead>
<tr>
<th>Designation</th>
<th>Materials</th>
</tr>
</thead>
<tbody>
<tr>
<td>Double copper tube heat exchanger</td>
<td>Copper tubes</td>
</tr>
<tr>
<td>Double copper tube heat exchanger</td>
<td>Copper, PCM embedded in graphite matrix</td>
</tr>
<tr>
<td>Finned double copper tube heat exchanger</td>
<td>Copper tubes and fins</td>
</tr>
<tr>
<td>Compact heat exchanger</td>
<td>Copper tubes, aluminum fins</td>
</tr>
<tr>
<td>Plate and frame heat exchanger</td>
<td>Stainless steel</td>
</tr>
</tbody>
</table>

It was found that the compact fin tube and plate heat exchangers offered the best ratios of heat transfer area to external volume before any experiments were executed. The experimental setup included a thermostatic bath to ensure uniform temperature distribution in the incoming heat transfer fluid (in this case water), a flowmeter, and thermocouples to take the water inlet and outlet temperature. Thermocouples were also placed within the PCM in the heat exchanger to determine its temperature at several critical points. The main results were that the turbulent regime on the water side accentuated the heat transfer between water and PCM, and thus made for a faster charge and discharge process. Also, it was found that the compact heat exchanger had the highest average thermal power, while the storage with a PCM matrix and no fins had the best average power per unit area.
Analytical and Numerical Simulation of Thermal Storage Unit

The benefit of a numerical model is the ability to quickly and easily simulate and optimize a system – in this case, parameters such as the number of fins and phase change material used can be easily changed. Experimental data is necessary alongside the model to validate its predictions. Many numerical simulations of thermal storage units have been created for various types of units, whether they have thermal conductivity enhancement or not. The numerical models differ in their underlying methods – usually they are based upon the energy equation, combining it with the Stefan number to predict the location of the boundary between the frozen and melted phase change material. FLUENT can also used in some cases to help predict the temperature distribution and velocity associated with natural convection. However, as will be seen in the existing literature, FLUENT must be carefully utilized as its phase change indicators are not as evident as those implemented in analytical models.

Guo et al [49] developed a model based upon a high temperature latent heat thermal energy storage system utilizing a mixture of potassium nitrate and sodium nitrate as well as aluminum foils to help the heat transfer to the material. It is identified that the challenge of these materials is their low thermal conductivity. So, developing proper heat transfer techniques to make up for this is vital. The domain formed a portion of a series of tubes running through the phase change material. The tube wall temperature was assumed as constant as the water inside was steam due to the high temperature of the system. The system was numerically simulated using a simplified version of the conduction energy equation, with assumptions made based upon the geometry of the system. This energy equation would be solved using FLUENT, and since FLUENT cannot explicitly track the moving liquid-solid melting front, the enthalpy-porosity method is used to numerically solve the equations, much in the same way that Reddy [48]
reached a solution for his work. To validate the numerical solution, a cavity of the same size without any foils was used, with a constant temperature applied to all sides in two dimensions. When these results matched known solutions to this two dimensional problem, the model was considered to be validated.

Results were parametrically determined, with the foil pitch, distance between tubes, foil thickness, tube radius, tube wall temperature, and thermal conductivity of PCM being changed. It was found that the foils within the PCM helped it to discharge more quickly into cold fluid during solidification, but this also added temperature variation in the axial as well as radial direction. The larger foil pitch leads to a slower decrease in liquid fraction during discharge. A smaller distance between tubes results in a decrease of liquid fraction during discharge. A speedier discharge can be affected by an increase in foil thickness up to 2 mm, and an increase in radius of the tube. Finally, with a lower wall temperature, a more rapid decrease of the liquid fraction occurs.

Another work with a parametric study and a numerical model was done by El Qarnia et al [50]. The first objective of the study was to derive a numerical model to predict the behavior of the phase change material latent heat storage system. The second objective was to find the parameters that provide the best performance, such as PCM mass, the amount of tubes, and mass flow rate of the water through the tubes. In the mathematical model, natural convection is assumed to play a role and an effective thermal conductivity is used to show its effects. The performance of the system was evaluated based upon its efficiency – the latent heat stored in the PCM divided by the solar radiation received by the collector over a certain time period. The numerical model was verified based on experiments with three different PCM’s – n-octodecane, paraffin, and stearic acid, and then used with the local radiation and temperature conditions on an
average day to determine the best storage configuration. The numerical model was based upon the energy equations for conduction with natural convection being accounted for with an effective conduction, based upon the Rayleigh number and geometry of the unit. The solution of the energy equations was completed using a finite-difference method, and solved using a tri-diagonal matrix.

The parameters selected above were run through the analytical model and the thermal storage efficiency was evaluated for each. Furthermore, it was investigated how long it took for the PCM to reach liquid conditions for the selected configuration. Both the charging and discharging processes were investigated. It was found that of the three phase change materials, the stearic acid would work best in this case based upon the water temperature it produced in the discharge mode.

Ait Adine et al [51] tested two phase change materials (P116 and n-octodecane) in a shell and tube configuration to store latent heat. A test was also done with the P116 alone. All predictions are based upon an analytical model derived from the beginning energy equations then combined with a control-volume approach – the equations are integrated over the control volume in two dimensions. The energy equations are simplified based upon the geometry and boundary conditions of the storage unit. Then, they are solved using a finite-differencing method. For validation, experimental data from a previous study by Lacroix were used. The main variables to predict were the solid-liquid boundary, storage efficiency, liquid fraction (0 = solid, 1 = totally melted), and heat transfer to the wax at a given time step. The mass flow rate of the heat transfer fluid and the length of the storage unit were two of the variables changed when examining the effect of relevant parameters on the performance of the storage system. Several flow rates and
other conditions were found for the optimal operation of the unit. Interestingly, the unit was found in some cases to operate best when some of the PCM was in the “mushy” zone.

The most relevant analytical model to the current study is the one created by Lamberg et al [52]. It provides a simulation of melting within a semi-infinite finned enclosure, and the following study provides the corresponding solidification simulation. The model in this paper is for a prediction of a solid-liquid interface location and temperature distribution of a fin in the melting process of a PCM storage. The value of the model for the melting process is that it determines when the thickness of the phase change material is such that natural convection takes over from conduction. Furthermore, three different phases of the melting of the wax are identified: conduction and natural convection from the fin, pure conduction from the fin, and solid phase change material with a fin and no phase change.

The first step in the creation of the model was the definition of two distinct regions: the finned and unfinned areas of the thermal storage unit, as shown in Figure 8.

![Figure 8: Fin Regions [52]](image-url)
The next step in the creation of the model was the development of an energy balance of the fin studied, as shown in Figure 9.

![Figure 9: Control Volume][52]

The energy balance and the choice of region defines the area in which the model will take effect. Once this backbone of the model is set up, assumptions are made to guide the calculation of the results. The ones relevant to the present study as well are that the temperature distribution can be taken as one-dimensional, the sensible heat of the liquid PCM is negligible, and that the physical properties of the phase change material and the fin material can be assumed to be constant. The energy balance of the fin starts as:

\[
E_f^* = q_{x}^* - q_{x+dx}^* - q_c^*
\]  

(4)

Next, the boundary conditions are defined that the interface starts at zero at time zero, the temperature of the phase change material around the fin is close to the melting temperature, and at time greater than zero the following two conditions are true:

\[
(\rho c_p)_f \frac{\partial T_f}{\partial t} = k_f \frac{\partial^2 T_f}{\partial x^2} - h(T_f - T_m)
\]  

(5)
\[ T_f(\infty, t) = T_m \] (6)

After these boundary conditions and the energy balance are set up, they are combined as shown in (7) (with the interface \( S_y \) beginning at zero when time starts):

\[
\frac{(\rho H)_y}{\partial t} = \frac{k_f}{x} (T_w - T_m) \frac{\partial S_y}{\partial x} + h(T_f - T_m)
\] (7)

Next, the heat transfer coefficient must be defined. There are a few different phases in the melting process – conduction, melting with only conduction, and melting with natural convection. The Rayleigh number was defined as:

\[
Ra = \frac{g(T_f - T_m) \rho_i^2 c_p S_y^3 \beta}{\mu k_i}
\] (8)

The critical thickness at which the natural convection took over was determined by setting the Rayleigh number equal to 1708, a value taken from the heat transfer concepts, and solved for \( S_y \). This critical thickness was combined with the Nusselt number as defined by Marshall [53], [54] to produce the following equation for heat transfer coefficient:

\[
h = 0.072 \left[ \frac{g \left( \frac{T_w - T_m}{2} \right) \rho_i^2 c_p k_i^2 \beta}{\mu} \right]^{1/3}
\] (9)

The analytical solution derived from the energy balance resulted in two different main equations, each for the non-finned and finned portions of the fin enclosure. These results are as follows:

\[
S_x(t) = 2\lambda \sqrt{\alpha_f t}
\] (10)

where \( \lambda \) is the solution of the transcendental equation:
\[ \lambda e^{-\frac{x^2}{H}} \text{erf}(\lambda) = \frac{c_{pl}(T_w - T_m)}{H\pi} \]  

and

\[ S_y(x,t) = h(T_f - T_m)x \left( \frac{-bx + \sqrt{(bx)^2 + 2abt}}{ab} \right) \]

where \( a \) and \( b \) are constants defined in the literature.

Once these equations were established, boundary conditions were applied to them for the three different conditions defined. The analytical results and numerical results are in agreement with the exception of the speed of the solid-liquid interface during the phase change process.

The main complications in solving phase change material problems are that the liquid and solid states have different properties; there is a nonlinear moving liquid solid interface. Furthermore, the melting front location is not known a priori.

The results are presented for the three different conditions (pure conduction with no melted wax, pure conduction with melted wax, and natural convection with melted wax), and will be explored now. For case one, natural convection within the melting phase change material, a small error was found, as the actual solid-liquid interface was just ahead of the analytical solution found. So, this shows that the convection within the melt can be calculated and justifies the solution. For case two, the solid-liquid interface is greatly underestimated because the natural convection is not adequately taken into consideration. Natural convection will further the melting interface because it strips away the melting wax from the mushy zone immediately adjacent to the still solidified wax. So to conclude, the model is suitable for predicting the wax interface progress of such a defined storage unit. However, it is not suitable for a solidification process because the solidification process works mainly through conduction,
while convection occurs within the melting process at a critical thickness as defined by the Rayleigh number.

Siren and Lamberg worked to also create a model for solidification so that a complete melting and freezing process could be simulated for a given finned thermal storage unit [55]. Two models were analyzed for the simulation of solidification of a PCM storage system with internal fins, but the one dimensional model will be focused on as it most closely resembles the one created in the present work. Most of the heat transfer in the case of solidification is done by conduction, so that is the focus of the driving equations. Also, several assumptions were first made before the model was created. The ones relating to both this model and the one created in this work are that the PCM and fin are just at the solidification temperature of the phase change material, the natural convection of the PCM is negligible, and the properties of the PCM and fin material are constant.

The cell is split into the region with the fin and the region with just the pipe for the analysis, since the Neumann solution can be used for just the pipe. A dimensionless rate of change is developed first for the fin, and then it is multiplied by the length to determine the solid-liquid interface location. A finite-difference solution is developed also for the solidification process for the two dimensional model. The symmetry of the cell is set as shown in Figure 10.
Also, an energy balance similar to the one developed for the melting model is created as shown in Figure 11.

The energy balance is somewhat similar to the one derived for the melting case [52], but in this case the signs are slightly different due to the direction of the heat transfer. With the assumptions made and these energy balances set up, the solution begins as shown in (13):

\[(\rho c_p)_f \frac{D T_f}{Dt} = k_f \frac{\partial^2 T_f}{\partial x^2} - \frac{k_x}{S_y} (T_f - T_m)\]  

(13)
With this energy balance established, it was realized that a non-dimensionalization of the energy equation would allow for simpler manipulation. So, several nondimensional variables were created as follows:

\[
\theta = \frac{T - T_m}{T_w - T_m} \quad (14)
\]

\[
\tau = \frac{k_f t}{(\rho c_p)_f l_f^2} \quad (15)
\]

\[
\gamma = \frac{S_y}{l_c} \quad (16)
\]

\[
\eta = \frac{x}{l_f} \quad (17)
\]

\[
\lambda = \frac{l_f}{l_c} \quad (18)
\]

\[
\Psi = \frac{D}{l_c} \quad (19)
\]

\[
\kappa = \frac{k_s}{k_f} \quad (20)
\]

\[
\xi = \frac{(\rho c_p)_f (T_w - T_m)}{-L \rho_s} \quad (21)
\]

The energy equation then becomes (22):

\[
\frac{\partial \theta}{\partial \tau} = \frac{\partial^2 \theta}{\partial \eta^2} = \frac{\lambda^2 \kappa}{\Psi} \frac{\theta}{\gamma} \quad (22)
\]

When solving the temperature distribution, the solution form becomes a summation of terms representing the energy equation within the cell defined. This solution was used to compare the one dimensional model to test conditions. Ultimately, it was found that the
geometry of the cell dictated the accuracy of the one dimensional model. If $\lambda$ is sufficiently large or small, the one dimensional model is quite accurate, since the heat transfer occurs mostly in the $x$ direction (small $\lambda$ value) or in the $y$ direction (large $\lambda$ value). This fact will be taken into consideration when setting up the analytical model for the present case.

The results from Lamberg and Siren’s study will be discussed further when they are compared to the analytical model in this study to validate it.

*Computational Fluid Dynamics Simulations*

FLUENT has also been analyzed in terms of its method of solving phase change problems. Pinelli and Piva [56] identified the fact that FLUENT’s phase change solving method was meant for a few specific applications, so it must be modified accordingly to solve problems outside that realm of applications. They applied FLUENT in solving a phase change occurring within a cylinder with the top of the cylinder heated at a temperature greater than the melting point of the phase change material. The goal of the study is to determine the usefulness of FLUENT when analyzing phase change materials, in this case n-octodecane. The numerical results are compared with existing experimental data first before using the program to analyze a phase change condition. Thus, some tweaking is necessary to utilize the code for this application to a cylinder of n-octodecane wax heated from the top (temperature is greater than its melting point). The temperature distributions found numerically agreed quite closely with the experimental results previously found by Pinelli and Piva. However, the phase change boundary seemed to move more quickly in the numerical model. This difference can be accounted for by the fact that some properties had to be treated as temperature dependent, due to FLUENT’s method of handling the phase change – the enthalpy-porosity model was used, with 0 being a solid material and 1 being a fully liquid material. The authors emphasize that the methodology
that FLUENT uses should be carefully worked through before using the code, so that the user is aware of the processes. This then allows for changes and assumptions to be made that will help to arrive at a reasonable answer once the code is run.

Besides the use of FLUENT to make predictions for thermal energy storage related applications, it can also be used to predict heat dissipation in electronic devices. With the size of cell phones decreasing but the amount of power they require increasing slightly to keep up with more applications, the cooling of the electronics becomes vital. A passive cooling system must be utilized since active cooling would require a fan or similar instrument, which is not practical for the cell phone user. In a study by Tan et. al [57], a theoretical cell phone is tested under eight cases with varying power loads and with or without a PCM to help cool it. The PCM is placed in a heat storage unit, located near the power amplifier, which is the greatest heat-dissipating component within the cell phone. A three dimensional numerical model using FLUENT was set up to test these cases. The first pair of cases tested were one case with aluminum in the heat sink unit and one case with PCM in the heat sink unit. The following three pairs of cases were the cell phone model with and without PCM at power dissipation rates of 3W, 2W, and 0.88 W. The most important result is that there is a limit at which the PCM will not be as effective. The 0.88 W case showed this; after an hour of operation the PCM did not melt, and it was seen that it would not melt at all at this power level. Thus it was concluded that a PCM is useful only at higher heat dissipation rates.

Another use of CFD to predict temperature distributions was a study by Shatiaikan et. al [58]. A partitioned storage unit using aluminum for the partitions and Rubitherm 25 for the PCM is studied. This particular storage unit is identified as suitable for a few different applications, such as storing solar thermal energy or dissipating heat from electronics. The unit
tested was composed of the partitions and PCM with an air gap left at the top of each partition of PCM to allow for expansion as the PCM melted. A heating or cooling device was placed underneath the base of the unit to conduct either a melting test or a solidification test. FLUENT was used to numerically simulate the melting and solidification process with the unit two dimensionally. The two dimensional slice was taken from the top to bottom of one half of each storage partition and is shown in Figure 12.

Within FLUENT, the solidification / melting model was used along with the “volume-of-fluid” model. Obviously the solidification / melting model accounts for the phase change of the wax whilst the “volume-of-fluid” model accounts for the fact that there are multiple fluids to be solved during the computational time. FLUENT uses a modified version of the energy equation, in which the enthalpy of the solid and liquid portions of the melting wax are considered, to solve for the solidification or melting process. The model was solved using the time-dependent calculations in FLUENT, and the results compared to the experimental data. It was found that melting of the wax or solidification of the wax begins at the base of the storage unit, which is
heated or cooled for each test. This is due to the extra conductivity provided by the partitions and base of the storage unit. In both cases, the heat transfer from the partitions to the PCM or the PCM to the partitions decreases over time due to the decreasing temperature difference between the wax and the partition. Also, heat transfer to or from the air decreases over time in both the solidification and melting cases.

Another application of CFD to predict phase change within a material was done by Schulte-Fischedick et. al [59]. A molten salt storage tank was analyzed during cooldown to determine how long it would take for localized solidification to occur while the power plant was not in operation. Since explicit test data was not available for comparison to the simulations, they were verified using a simple energy balance. The storage tank studied is similar to those of Solar Two, since it is a proven use of the molten salt storage. The layout of the tank studied can be seen in Figure 13.

![Figure 13: Molten Salt Storage Tank Schematic](image)

The main items of interest were the velocity distribution of the salt due to natural convection, the heat loss through the sides and base of the tank, and the effect on the natural convection due to the existence of several rod heaters to prevent total solidification during an emergency. There were three steps undertaken to model the situation: FEM to determine heat loss using Ansys, 2-D CFD using FLUENT, and finally 3-D CFD using FLUENT. In the FEM
simulation, a published correlation was used to account for laminar free convection along a cylindrical wall, which was the case on the inside of the tank. Some approximations were made to account for the amount of insulation around the tank, the convection and conduction on the outside of the tank, and the existence of the rod heaters within the unit. It was also noted for future reference in the study that the tank was considered “empty” when the level of molten salt was just above the impeller of the salt pump. The salt could not go below this level so that it could be pumped at all times. The 2-D and 3-D CFD simulations required that some boundary conditions be adjusted to fit with the available options within FLUENT. For example, the heat flux into the soil from the base of the tank was rearranged so that it could be used with the convection boundary condition in FLUENT. The 2-D model focused on an internal slice of the tank to examine the velocity patterns as well as the effects of heat loss to the sides and base of the tank. The 3-D model was done as a quarter slice of the tank to examine the effect of the rod heaters on the molten salt temperature distribution. Along with the velocity distribution and effect of the rod heaters, the time it took for local solidification was also calculated in the 2D and 3D CFD cases.

The significant parameters found from the experimental and numerical studies are tabulated in Table 8 below. From these studies, it is evident that heat transfer and heat load fluid mass flow rates as well as the overall finned heat exchanger design are the parameters that most often affect the operation of the unit. These factors will inform the creation of a test plan and the basis for analytical and numerical models in the current study, which will be outlined in the Methodology chapter.
### Table 8: Significant Parameters

<table>
<thead>
<tr>
<th>Study</th>
<th>Most Effective Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Zeng et al [47]</td>
<td>Addition of metal to PCM</td>
</tr>
<tr>
<td>Seeniraj et al [48]</td>
<td>PCM melt temperature</td>
</tr>
<tr>
<td>Reddy [23]</td>
<td>Fin Spacing, HTF mass flow rate</td>
</tr>
<tr>
<td>Agyenim et al [24]</td>
<td>Fin configuration, HTF mass flow rate</td>
</tr>
<tr>
<td>Medrano et al [25]</td>
<td>Heat exchanger configuration, HTF and heat load mass flow rate</td>
</tr>
<tr>
<td>Guo et al [49]</td>
<td>Distance between heat exchanger tubes</td>
</tr>
<tr>
<td>Lamberg et al [52]</td>
<td>Fin length</td>
</tr>
<tr>
<td>Lamberg et al [55]</td>
<td>Ratio of length to width of cell</td>
</tr>
</tbody>
</table>
CHAPTER THREE: PROBLEM DEFINITION

The main goal of this study is to analyze the performance of an inexpensive solar thermal energy storage unit to supplement or replace a hot water heater for a single family residence. It is already known that certain parameters play a role in determining the effectiveness of the storage unit, so they will be studied analytically as well as experimentally. These parameters tested in the study will be the mass flow rate of the heat transfer fluid to the solar array and the number of fins in the heat exchanger. It is important to note that while this optimization may work well for a storage system of this scale, it is not necessarily applicable to a larger building such as a hotel or hospital. The mass flow rate can be varied experimentally, however, the number of fins must be tested using computational fluid dynamics since the construction of a new unit would take a considerable amount of time. The uniqueness of this study can be found in the fact that a solar collector is used to increase the temperature of the heat transfer fluid as well as the fact that the heat load and heat exchange loops are both included within the storage unit.

The boundary conditions important to this study affect the operation of the collector and the unit together, since they are bound by a mass flow loop. These include the mass flow rate of the heat transfer fluid in kilograms per second, the radiation incident upon the solar collector in Watts per meter squared, the number of fins within the heat exchanger, and the insulation around the storage unit in Kelvin times meters squared per Watt.

The following chapters detail the methodology, results, and conclusions from this study, which is guided by the preceding boundary conditions and reviewed literature. The methodology is broken up into the experimental methods wherein the rig is tested by altering specific
boundary conditions and the analytical methods wherein simulations of the rig are conducted. These simulations are broken down further into analytical and numerical simulations. Analytical calculations will be completed based upon the one-dimensional resistance analysis from the realm of heat transfer, and a numerical solution considering the full solution of the Navier-Stokes and energy equations will be determined using FLUENT. These models will be used in tandem to determine optimal parameters for a domestic unit. Once they are verified with the experimental results, they will be run to optimize the unit based on various heat transfer fluid mass flow rates and number of fins.

The results will be presented and discussed in three different sections, much in the same way as the methodology description. The experimental results will be presented and discussed in terms of the efficiency of the unit, the results when the mass flow rate and weather conditions vary, and the accuracy of the data acquisition. The analytical results will be compared to the experimental results and evaluated in terms of their accuracy. The simpler one dimensional model will also be compared to the more complex FLUENT model to see if the results have similar trends. The FLUENT model will be evaluated according to its similarity to experimental results, as well as its predictions on the optimization of the unit based upon the addition of fins and different mass flow rates of heat transfer fluid.

Finally, conclusions will be drawn based upon this study as to which parameters are optimal for the charging and discharging of the storage unit, using the current test rig’s materials and design. These conclusions will inform suggestions for future work in the development of an integrated solar thermal collector and storage system for residential hot water.
CHAPTER FOUR: METHODOLOGY

Experimental Procedures

Test Stand

The test stand integrated the storage medium of wax as well as the solar collectors and some form of tracking. Thus, the design implemented placed the storage unit beneath the collector within an insulated box. The end result is shown in Figure 14 and was developed with the help of a senior design team. The structure is a wooden box with casters, which encloses the Styrofoam insulation around the aluminum box, which contains the heat exchanger and the wax used. A schematic of the unit’s cross section follows in Figure 14.

Figure 14: Solar Thermal Rig
The flat panel of collectors seen is a model from GEAR Solar, which manufactures solar water heaters. This collector panel will provide the heat to the heat transfer fluid (glycol) through the heat pipes within the solar thermal tubes. The interior of these tubes can be seen in Figure 16.
Figure 16: Cross Section of Solar Thermal Collector Tube (Courtesy GEAR Solar)
Once the heat is collected using the solar thermal collector tube array and absorbed by the glycol, which is the working fluid, it is transferred to the wax using a copper heat exchanger within the thermal storage unit. Water will be used to extract heat from the melted wax through the heat load loop within the heat exchanger when the unit is in discharge mode. Two pumps are located at the front of the unit, one to circulate the glycol to transfer thermal energy to the storage unit and one to circulate the water to extract heat from the storage unit. The unit can also be hooked up to a city water hose to simulate the use of city water temperature and pressure to extract heat from the storage unit. A pump can also be hooked up to the heat load side of the unit to do a closed loop test during the discharging process.

The novelty of this storage unit comes in the configuration of its heat exchangers. For this unit, the heat load loop and the heat exchanger loop are both included as loops within the storage unit, which has not been tested in the literature cited in this work. A schematic of the different loops can be seen in Figure 17, along with the thermocouple location points, which will be explained in the data acquisition section. The red loop represents the charging piping, the blue loop represents the discharging piping, and the black lines represent the fin locations. The heat transfer loop in this case receives energy from the solar collection manifold, while the heat load loop extracts heat from the thermal storage unit for use. The fins help to increase the effective thermal conductivity of the storage medium, as indicated in the literature reviewed in Chapter 2. The wax maintains its material properties but the fins extend areas of higher conduction into the wax, since it is poured in over the heat exchanger, as shown in Figure 18.
Heat load loop
Heat exchange loop

TC Location

Figure 17: Heat Transfer Loops
There are three heat load loops and four heat transfer loops layered within the unit, as shown in Figure 19.
The heat load loops handle the heat transfer fluid used to draw heat out of the charged storage unit, in this case, water. The heat exchanger loops are filled with glycol and transfer the heat collected from the solar collector to the wax through the fin system. Note that the fins are attached to both the heat exchanger and heat load loops. This design element will be further explored in the analytical and numerical calculations and noted when examining the experimental data, as the number of fins will affect both the amount of extra conduction within the paraffin wax and the amount of natural convection allowed within the wax once it begins to melt. If too many fins are in the container, this will impede the natural convection process as the wax cannot circulate. Whether or not the amount of fins in the heat exchanger design is optimal is something that will be explored in the analytical and numerical modeling.

Furthermore, the task of constructing the test stand necessitated the selection of two pumps – one to pump glycol to the solar collector and through the heat exchanger portion of the
storage unit, and one to pump water through the heat load section of the storage unit. The loss calculations for a sample flow rate in each pump can be seen in Appendix A. From these loss calculations and comparison to pump curves, a model CRE1-5 donated from Grundfos Pump Company was used along with a TACO circulator pump borrowed from the Florida Solar Energy Center with the necessary connections to the hoses that ran to the heat exchanger and heat load loops. An infrared remote control was used to limit the pump motor of the Grundfos pump in conjunction with the variable frequency drive (VFD) to assure the correct amount of mass flow rate as per the head loss calculations done and the pump curves provided by Grundfos.

**Phase Change Material**

The phase change wax chosen for the initial analysis was purchased from Candle Science and had a melt point of around 50 C given by the manufacturer. However, the actual melt point of the material is not 50 C, because of the quality of the wax. Since it is not laboratory grade there will be a certain amount of impurities, which change the melting point of the wax. The wax melt temperature was chosen for its closeness in value to the temperature of water for the home, and also as a proof of concept to ensure that the test stand was operating correctly. Another aspect to the choice of paraffin wax was its price – for the purposes of this study, it was less expensive than proprietary phase change material blends.

**Data Acquisition**

A temperature measurement device had to be chosen to measure the heat of the wax only and the temperature at the inlet and outlet of the heat transfer fluid piping. A T-type thermocouple was chosen due to its temperature range (-250 C to 400 C), since temperature measurements would be made within this range. This range was also the smallest one
commercially available. To assure the uniform depth of each thermocouple, it was attached to a wooden dowel with nonconducting tape, as the dowel rod would not affect the conductivity of the wax as much as the heat exchanger components. These thermocouples would be used to determine two things: where the melting front was at a given time, and to determine when the unit was fully charged. The thermocouple set furthest from the fins would determine a fully charged unit, as it would reach melting temperature after those set closer to the fins. This location was 4B, as shown in Figure 17.

To measure the temperature difference at the entrance and exit of the thermal storage unit during the charging process, a thermopile was constructed and placed at these points. This thermopile were created using T-type thermocouples connected in series. All of the thermocouples were welded using a thermocouple welder to ensure a low time constant. This low time constant allowed the welded thermocouple bead to be treated as a lumped capacitance, and therefore the assumption of temperature being the same at the surface and center applies. To calibrate the thermopile, a known temperature difference was established using a thermocouple calibration device from Omega Engineering and an ice bath. This established various temperatures from 30°C to 50°C and a temperature of 0°C respectively. In this way, temperature changes were established and a calibration curve was created. The reasoning behind measuring the temperature difference as opposed to the separate temperatures was the fact that the temperature difference drives the flow of energy from the heat transfer fluid to the wax during the melting process and from the wax to the heat load fluid during the freezing process. Using this temperature difference, the energy flow can be calculated using the following equation:

$$\dot{Q} = \dot{m} C_p (T_{in} - T_{out})$$  \hspace{1cm} (23)
Every 30 minutes, a temperature reading would be taken of the glycol pump’s inlet and outlet temperature and temperatures within the bulk of the wax. Also, the change in temperature would be measured using the thermopile. These temperatures would be then used to verify the results of the mathematical model and to determine when the wax within the storage unit was fully melted. Figure 20 shows the location of the thermocouples within the wax.

The locations were chosen based upon the fin configuration, as well as the fact that the heat load loop and the heat exchange loop are both enclosed within the storage unit. The effect of the depth was also tested, with two different depths tested adjacent to both the heat load loops and the heat exchange loops. These depths can be seen in Figure 21.
**Testing Procedure**

Two main tests were conducted: the charging and discharging of the integrated collector system. The goal of these tests was to determine the time to charge and discharge the unit as well as the temperatures of the fluid and wax during these times. Tests were undertaken in the months of November through June in Orlando, FL, approximately latitude 29 degrees north. Using the irradiance information presented in Table 9, the array was set at 30 degrees to attain the maximum solar insolation. The variable changed for each test was the volumetric flow rate of the heat transfer fluid – it was run at 0.5, 1.0, and 1.5 gpm.
Table 9: Central Florida Insolation Data

<table>
<thead>
<tr>
<th>Tilt</th>
<th>Jan</th>
<th>Feb</th>
<th>Mar</th>
<th>Apr</th>
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<th>July</th>
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<th>Sept</th>
<th>Oct</th>
<th>Nov</th>
<th>Dec</th>
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<tbody>
<tr>
<td>0°</td>
<td>3.14</td>
<td>3.92</td>
<td>4.99</td>
<td>5.99</td>
<td>6.27</td>
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<td>5.68</td>
<td>5.28</td>
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<td>4.11</td>
<td>3.46</td>
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<tr>
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<td>4.43</td>
<td>5.30</td>
<td>6.05</td>
<td>6.10</td>
<td>5.54</td>
<td>5.49</td>
<td>5.24</td>
<td>4.89</td>
<td>4.53</td>
<td>4.06</td>
<td>3.56</td>
<td>4.91</td>
</tr>
<tr>
<td>20°</td>
<td>3.92</td>
<td>4.56</td>
<td>5.36</td>
<td>6.01</td>
<td>5.99</td>
<td>5.41</td>
<td>5.37</td>
<td>5.18</td>
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<td>4.63</td>
<td>4.23</td>
<td>3.74</td>
<td>4.94</td>
</tr>
<tr>
<td>25°</td>
<td>4.07</td>
<td>4.67</td>
<td>5.39</td>
<td>5.95</td>
<td>5.85</td>
<td>5.26</td>
<td>5.23</td>
<td>5.10</td>
<td>4.89</td>
<td>4.70</td>
<td>4.37</td>
<td>3.90</td>
<td>4.95</td>
</tr>
<tr>
<td>30°</td>
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<td>5.39</td>
<td>5.85</td>
<td>5.67</td>
<td>5.07</td>
<td>5.06</td>
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<tr>
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<td>4.76</td>
<td>4.64</td>
<td>4.24</td>
<td>4.80</td>
</tr>
</tbody>
</table>

The testing procedure was simple, but several crucial steps had to be followed so that they followed the assumptions made in the analytical model. Two subprocedures were created, one for the charging process and one for the discharging process.

To begin the charging process, the rig had to be pre-heated during the months of November to February due to the cooler weather conditions. After finding that the antifreeze boiled within the rig during the first week of March, the rig was simply rolled outside and the pump turned on to prevent the fluid from boiling.

For the discharging process, once the wax was all melted, a hose with city water pressure and volumetric flow rate was attached to the heat load loop and was turned on. This circulated the water through the heat load loop, which led to the water receiving heat from the melted wax. This process was allowed to continue until the temperature out of the unit did not significantly change (within 0.1 degrees C). This process was open loop since this would be the case for home hot water use; the water out of the storage unit was not recirculated to the inlet of the storage unit. The ambient temperature during this time was noted because it would affect the withdrawal of the heat from the thermal storage unit. The discharge test was done in the afternoon time, after the unit had received heat.
In addition to the discharge test using one pass through the unit, a discharge test with recirculation within the unit was done. This would provide another option for achieving the minimum temperature requirement of the water. A TACO recirculation pump, model 006-BC4, was used for this process as it came closest to the actual water flow rate expected from city lines. This test was run for one hour, with data taken every 15 minutes, so it would be comparable to the open loop discharge test in that regard and could be compared.

Data Reduction

The two main data sets taken were the temperatures of the wax as well as working fluid temperatures for either the heat transfer or heat load fluid, depending upon the test taken. For the melting, the temperature difference between the inlet and outlet of the storage unit and the temperature at the inlet and outlet of the solar thermal collector’s manifold were measured. On the other hand, the water temperature at the inlet and outlet of the storage unit were more important during the freezing process. These measurements would be used for heat transfer calculations and to determine boundary conditions for the FLUENT solution.

The wax temperature and temperatures of the fluid running into and out of the storage unit and manifold were recorded every 15 minutes during the freezing process and every 30 minutes for the melting process. With this wax temperature data and the corresponding temperatures from the heat load and heat exchange entrance and exit, the overall operation of the test rig could be determined for various conditions and analyzed against other thermal storage units. Furthermore, the temperature variations during the day could indicate the weather patterns of the day, i.e. a cloud passing overhead would change the amount of radiation incident on the solar thermal collection tubes.
Analytical and Numerical Procedures

Analytical Model

The mathematical analysis in this study combined the techniques of analytical and numerical analysis. The analytical portion of this work will involve the solving of the energy equation in one dimension, including provisions for fins and the moving melt boundary. The construction of an analytical model was guided by Patankar’s text on numerical heat transfer [63], following the examples of discretizing points within the system studied. The numerical analysis involves the replication of the testing conditions using FLUENT in two dimensional double precision mode. Both of these analysis methods will be verified using the experimental data before further untested conditions are calculated.

Analytical calculations are conducted for the melting and solidification of the wax, which corresponds to the charging and discharging of the unit. The entire process is transient, and thus a few assumptions needed to be made to simplify the charging and discharging process. In the analytical method, the models are one dimensional in space and are solved using a finite difference method to account for the transient nature of the system. For the startup of the system, it was assumed that the heat transfer fluid was circulating before the rig was rolled outside and the solar collector allowed to begin to pick up heat. From the test data collected, an equation relating time of day with the temperature of the heat transfer fluid was developed. This allowed the temperature of the fluid at the inlet of the heat exchanger to be determined as time went on. Furthermore, it was assumed that all heat transfer from the pipe for charging and discharging went in the radial direction, as the resistance in that direction was the smallest. With these assumptions limitations must also be stated. Firstly, the model was one dimensional and did not, therefore, take into account the movement of melted front due to gravity or the curvature
of the wax as it solidified or melted, as shown in Figure 22, where the white portions represent melted wax and the black portions represent solidified wax. The domain is divided in three sections.

![Figure 22: Wax Curvature Around Fins](image)

Also, the analytical model assumes that the wax melts exactly at 50 C as indicated by the manufacturer, and that the mushy zone does not exist.

At first, the length of pipe was run without fins to ensure the code developed was in good working order. Two regimes were defined: conduction until the outside of the pipe reached 50C, the melting point of the wax, then a combination of conduction and convection until the wax had entirely melted. Once this code was completed correctly, fins were added to the code, as well as an equation to account for natural convection once the wax reached the critical thickness, giving a Rayleigh number over 1900. The fins provided extra conduction to the wax at first, and then when the temperature of the wax around them reached its melting point temperature, the natural
convection process began and the temperature distribution along the fin could be solved for. It also had to be estimated how far along the pipe the fin’s effect could be felt. This distance could then be approximated better with the use of the experimental data to validate the model.

With this general idea in mind, the analytical model will now be explained in detail, beginning with the startup or pure conduction portion of the process. Then, the beginning of the melting will be explored then pure natural convection. This process will be repeated for the solidification process in reverse, but only using conduction heat transfer. This code was written in MatLAB and is presented in its entirety in Appendix E and Appendix F.

The model begins with pure conduction through the pipe and fins to the wax, with the fluid and wax both beginning at room temperature. However, the pumps will be running before the solar collectors are exposed to sunlight, so the flow can be assumed to be fully-developed throughout the entire pipe. The temperature of the fluid increases as the collector begins to warm up, and this temperature is input as a function of time based upon the temperature data from the glycol in the experiment. A finite-difference equation is used to determine the temperature decrease of the glycol as it moves through the heat exchanger. Several approximations are made here since the model is one-dimensional. The wax is treated as a semi-infinite solid with a fluid moving over it with a specified heat transfer coefficient. From the Dittus-Boelter correlation, the heat transfer coefficient is found. A further approximation is made by combining the resistance of the heat transfer coefficient and the resistance resulting from the pipe wall into an effective resistance, which is then used in the transient conduction equation to determine the interface temperature of the wax. The pipe is broken up into sections to solve the heat transfer equations at those respective locations within the storage unit. The
The division chosen was 1/16 inches because this was the thickness of the fin material. Figure 23 shows a schematic of these sections, which are divided into two different types.

Type A takes into account only the heat transfer fluid, the pipe wall, and the wax, while Type B takes into account the heat transfer fluid, the pipe wall, the wax, and the fin.

The fin dimensions are approximated by using a circular fin and the corresponding relationships for these fins. Since the fins are six inches wide and twenty-four inches long, the circular fins are assumed to have a radius of three inches. This will result in four separate circular fins instead of one large plate fin for the heat transfer loop, as shown in Figure 24.
For the heat load loop, on the other hand, there will be three separate circular fins, as shown in Figure 25.

![Figure 25: Fin Approximation Sections (Head Load Side)](image)

The reasoning behind these approximations is not only the simplification of the fin problem, but also the concept of fin effectiveness. At a certain point, the fin will not gain any additional effectiveness from the additional length. In the approximation the radius is the critical length, but in the actual heat exchanger, the critical length is the length of the fin away from the tubes, i.e. how much it extends out into the wax.

At the start of the heat transfer by conduction to the wax, a semi-infinite solid approximation is used due to the dimensions and thermal diffusivity of the wax, as well as the time over which the simulation and operation of the unit is conducted. The heat transfer to the wax is determined using the effective resistance over the area of the control volume along with the temperature difference between the surface temperature of the wax and the mean temperature of the heat transfer fluid. Then, equation 20 is used to find the temperature exiting from the control volume.

\[
\dot{m} C_p (T_{\text{mean,in}} - T_{\text{mean,out}}) = \frac{1}{R_{\text{total}}} (T_{\text{wax}} - T_{\text{mean}})
\]

(24)

The case with convection over the surface of the semi infinite solid is considered here, with the overall heat transfer coefficient being the sum of the heat transfer coefficient for the
convection within the tube and the resistance produced by the tube wall. The resistance of the tube wall is very small, but is included for completeness of the model.

This process is repeated for the entire length of the pipe and over time until the wax interface over the pipe length begins to reach the melting point of the wax, 50 C. After this temperature is reached, different equations are used to account for the resistance of the melting wax and any natural convection occurring within the melted wax. There are two processes simulated using the analytical model: freezing of the wax and melting of the wax. The analytical model has several common points for both models, so they will be noted as the explanation progresses.

To create the melting analytical model, several assumptions were made before the control volume analysis. The first was that the wax was completely frozen and at room temperature (approximately 25 C) at the start of the charging model’s run. For the discharging model’s run, the wax was assumed to be completely melted and at the melting temperature of 50 C. Also, the water or temperature at the inlet was assumed to change in the same way the water or glycol temperature from the data collected changed. This value varied with different test dates; a test date with complete melting of the wax was selected for verification of the analytical model. The input values were established by a curve fit to the data using Microsoft EXCEL.

Next, a control volume type and thickness needed to be defined. This was done based upon the structure of the heat exchanger. There are two different types of control volumes analyzed, so the analytical model will be explained in terms of these two volumes. Control volumes A and B have been specified in Figure 23. The control volume was set to be the thickness of the fin so that a resistance model could be created for each pipe section and fin
The two control volumes share three common elements: the convection condition within the pipe, the pipe wall, and the freezing or melting thickness of the wax. These three elements have corresponding resistance values, which are calculated based on the cylindrical heat transfer resistance model found in Incropera’s text [60]. The convection resistance was determined by the following equation:

\[ R_{\text{conv}} = \frac{1}{hA} \]  

The conduction through the wall is determined using the cylindrical conduction resistance, as in equation 26. This is used to simulate the radial conduction resistance of the copper pipe wall.

\[ R_{\text{cond,wall}} = \frac{\ln \left( \frac{r_2,\text{pipe}}{r_1,\text{pipe}} \right)}{2\pi k_{\text{pipe}} L} \]  

The convection resistance and the conduction resistance through the pipe wall were assumed constant throughout the freezing wax model’s run, however, the frozen wax layer’s resistance changed with time. The same assumption was made for the melting model, but the melting layer instead changed with time and thus the resistance did as well. The solidification or melting distance of the wax was calculated using the Stefan equation:

\[ xe^x \text{erf}(x) = \frac{C_p (T_{\text{wax}} - T_{\text{mean}})}{H \sqrt{\pi}} \]  

In this case, \( x \) represents:
The expression in (27) was solved using a simple transcendental solver for $x$, then (28) was solved for $\delta$ to determine the wax thickness as it froze or melted. Next, the wax thickness was placed in the cylindrical resistance formula to determine the resistance resulting from the increasing wax thickness at each time step. The resistance is calculated using equation (26), but with the wax thicknesses and properties instead that of the copper tube. This wax thickness could be melting or freezing depending upon the model case run.

The first control volume (A) as shown in Figure 23 has a total resistance determined by equation 29.

$$R_{\text{total}} = \frac{1}{h_{\text{mean}}A} + \ln\left(\frac{r_{2,\text{pipe}}}{r_{1,\text{pipe}}}\right) + \ln\left(\frac{r_{\text{freeze}}}{r_{2,\text{pipe}}}\right)$$  \hspace{1cm} (29)$$

However, for control volume type B, the fact that there is heat transfer through the sides of the fin must be taken into consideration. This is done by developing an expression for the equivalent resistance for the fin. In the case of melting, the heat transfer coefficient will be determined by natural convection methods. This was determined by the Rayleigh number value, which was calculated once the wax began to melt. Once the Rayleigh number was greater than 1900, an additional resistance was added to those in the melting process. This resistance was calculated by an equation derived from Lamberg and Siren [52]. This is calculated by (30).

$$h = 0.072 \left[ \frac{g(T_w - T_m)}{2 \mu} \rho_i^2 c_{p,i} \kappa_i^2 \beta \right]^{\frac{1}{3}}$$  \hspace{1cm} (30)$$
For the freezing process, we replace the heat transfer coefficient with the thermal conductivity of the solid wax divided by the distance the wax has frozen at that point. This represents an equivalent heat transfer coefficient in the solidification case. The thermal resistance for a circular fin is given by the following:

\[
R_{\text{fin}} = A_{\text{fin}} \frac{k_{\text{fin}}}{\delta_{\text{freeze}}} \frac{2r_{1,\text{fin}}}{m(r_{2,\text{fin}}^2 - r_{1,\text{fin}}^2)} \frac{K_1(mr_{1,\text{fin}})I_1(mr_{2,\text{fin}}) - I_1(mr_{1,\text{fin}})K_1(mr_{2,\text{fin}})}{K_0(mr_{1,\text{fin}})I_1(mr_{2,\text{fin}}) - I_0(mr_{1,\text{fin}})K_1(mr_{2,\text{fin}})}
\]

(31)

with the term \( m \) defined as in equation (32) for freezing, as in equation (33) for heating up without melting or melting without natural convection, and as in (34) for heating up with melting & natural convection.

\[
m = \sqrt{\frac{2k_{\text{fin}}}{\delta_{\text{freeze}}}} \frac{k_{\text{fin}}t_{\text{fin}}}{k_{\text{fin}}t_{\text{fin}}}
\]

(32)

\[
m = \sqrt{\frac{2k_{\text{fin}}}{\delta_{\text{melt}}}} \frac{k_{\text{fin}}t_{\text{fin}}}{k_{\text{fin}}t_{\text{fin}}}
\]

(33)

\[
m = \sqrt{\frac{h_{\text{wax}}P_{\text{fin}}}{k_{\text{fin}}A_c}}
\]

(34)

So, the total resistance with the fin included is as shown below in (35), and the resistance chain types can be seen in Figure 26.

\[
R_{\text{total}} = \frac{1}{h_{\text{mean}}A} + \frac{\ln \left( \frac{r_{2,\text{pipe}}}{r_{1,\text{pipe}}} \right)}{2\pi Lk_{\text{pipe}}} + R_{\text{fin}}
\]

(35)
Figure 26: Resistance Chain Types

Type A represents the pipe section without the fins, which corresponds to control volume type A, while type B, which represents the pipe section with a fin, corresponds to control volume type B.

Having defined the resistances to be used in the model, the analytical code could be run for the length of the pipe and for the time desired. First, the storage unit was initialized for all lengths during the first time, and for the first portion of the tube length for all times run. The fluid always entered the heat load section at 298 K (27 °C) when simulating the freezing process, since it was assumed that the water coming out from the heat load will be used up. To simulate the melting process, the curve fit to the data collected was used. Furthermore, it was assumed that for the very first section of the pipe at the first second of heat extraction from the melted wax, the wax did not freeze. Once the initialization was complete, the main code was run. First, the mean fluid temperature in the section of interest for calculation was set to be the exit fluid temperature from the previous section.
From the resistance determined from control volume A or B, the heat transfer from the wax to the fluid was determined using:

\[ q_{wall} = \frac{1}{R_{total}} (T_{wax} - T_{mean}) \]  

(36)

Next, the temperature out of the control volume was determined using the fact that the heat transferred from the wax to the water could be set equal to the heat gained by the fluid from the wax as it passed through that control volume. The following equation was then solved for the temperature exiting the small control volume.

\[ m C_p (T_{\text{mean, in}} - T_{\text{mean, out}}) = \frac{1}{R_{total}} (T_{wax} - T_{\text{mean}}) \]  

(37)

Using this equation, the temperature coming out from the small section of pipe was determined. This temperature was then passed on to the next section of pipe, and the process repeated over the entire length of the pipe. It was assumed that there was no heat loss from the fluid due to the bends in the heat exchanger. The entire pipe was modeled so that the temperature at the exit of the heat load loop could be determined. It is also important to note that although the analytical model does take into account the fact that there is solidification occurring on both sides of the fin, it does not take into account the fact that the solidification front will take on a curve shape as it nears the fin. The solidification process is also significantly different from the melting process since the fins act as nucleation sites for the freezing wax [62].

The results from these analytical calculations for solidification of the wax will be compared to the actual conditions of outlet and inlet temperatures of the manifold and storage unit.
Conduction is considered to be the dominant mode of heat transfer up to the point at which the Rayleigh number reaches 1900, where natural convection begins to occur within the wax. In the code used to run the analytical calculations, the critical thickness of the wax will be solved using (38):

\[
S_{cr} = \left(\frac{1900 \mu k_f}{\beta g (T_f - T_w) \rho^2 c_{p,f}}\right)^{\frac{1}{3}}
\]  

(38)

After this critical value, natural convection dominates the melting process, and the fins begin to act as fins with convection around them determined by the density differences within the melting wax. The value of the heat transfer coefficient from the natural convection will be determined by (34) [52]. The natural convection is only used in the melting model.

**Numerical Model - Fluent**

The FLUENT model was created to study the effect of adding more fins to the heat exchanger as well as to provide a qualitative and quantitative explanation of the behavior of the wax within the storage unit. Any differences would show up in the melt patterns within the wax, which would be driven by temperature differences. This along with the temperature distribution of the wax can be presented qualitatively as well as quantitatively, which is the benefit of using a commercial CFD code such as FLUENT. This will be exacerbated by the fact that the melting of the wax does not occur at a uniform temperature. The “mushy zone”, where the wax is gradually melting or solidifying, is another area of interest when running the FLUENT simulation. Furthermore, determining the difference additional fins make can provide useful information on future heat exchanger designs.
The most basic model was created in two dimensions with one fin and one half of the pipe wall. The next step was to build a two dimensional model of a “slice” of the interior of the storage unit. This corresponded to a depth measured by thermocouples embedded within the wax. Experimental data taken from the test rig would then be compared to this FLUENT model. Once this model is validated, the process to create it will be used to create models with more fins within the heat exchanger.

The first consideration given to the initial numerical model was appropriate assumptions to create the model to be run in FLUENT. Furthermore, the internal geometry of the heat exchanger was somewhat complex, so the FLUENT analysis began with a two dimensional model of just the first fin and tube section to ensure that the model was realistic in comparison to the physical data and the resistance model.

From the storage unit, the first fin and tube section is modeled in two dimensions. There are several assumptions made here to analyze only two dimensions. First, the effect of gravity is neglected, so there is no natural convection modeled here. Furthermore, the heating up of the fluid as it moves past the melted wax is not modeled, and it is assumed that the flow to all tubes in the heat transfer array is split evenly among each tube. The area modeled is shown enclosed in red in Figure 27.
The boundary condition was set as adiabatic for faces A and D, symmetric for face B, and convection with a heat transfer coefficient corresponding to the Dittus-Boelter Correlation for water flowing at 298K on face C. The interface between the wax and copper became a shadow wall coupled with the copper fin. This enabled conduction from the fluid, to the pipe wall, to the fin and ultimately to the wax. A schematic of the computational domain can be seen in Figure 28. This shows the small portion on side C that is exposed to the convection condition to simulate the water flow through the pipe. A resistance is also added to this small portion to account for the fact that there would be a resistance value from the pipe wall in the actual storage unit.
The equivalent resistance from the length of the pipe wall modeled was added to the fin base, and the length of pipe wall without the fin was set to adiabatic. Thus, there was only convection at the fin base. This was done to determine the resistance across the fin by dividing the heat transfer across the base and to the wax by the temperature difference between the melting temperature of wax and the fluid temperature in the free steam within the pipe.

When creating a base for the model in GAMBIT, the mesh was first constructed on the half-fin section. Once this edge was established, a boundary layer mesh was implemented. GAMBIT’s built in function for the boundary mesh was used to establish a finer mesh around the four sides of the wax. This ensured that a finely structured mesh was in place along the crucial edge portions of the wax next to the copper heat exchanger. Finally, a uniform mesh was set on the remainder of the wax portion. This methodology was also extended to the internal slice of the FLUENT model, showing the entire heat exchanger.

When running the model, the solidification / melting model was selected, which automatically enabled the energy equation calculations. The appropriate properties were entered
for the materials involved in the calculation, which in this case were the paraffin wax and copper. Since the solidification / melting model was used, the latent heat of fusion, solidus temperature, and liquidus temperature were entered for the paraffin wax in addition to its other liquid properties. The boundary conditions were set as indicated in the computational domain, and the model was initialized at 323 Kelvin (the melting point of the paraffin wax) or 300 K (the frozen temperature of the paraffin) before iterations were run. An unsteady calculation was run because the melting front movement had to be observed for various times. To check the model, it was run for a one hour then a two hour calculation, and checked to ensure that the melting front had advanced. Once these models were run, the flux across the portion of the fin exposed to convection was calculated using FLUENT’s Report feature. This gave the flux in watts across the desired surface. The appropriateness of these results will be determined based upon the experimental data found.

The next level of CFD model included the entire heat exchanger in two dimensions at a specified level of the storage unit. There were two levels to choose from, as shown earlier in Figure 21. The bottom level was chosen for analysis since it would be affected less by gravity. Instead of only simulating the melting of the wax, the heat transfer to or from the fluid was also simulated, depending upon the conditions the model was under.

A mass flow rate of one gallon per minute was used in the testing process to verify that the CFD solutions attained were realistic. This would be compared with the experimental data taken using a one gallon per minute flow rate. The works of Reddy [48] and Pinelli [56] were used as guidelines, since FLUENT does not explicitly track the solidification front’s movement through the wax. The glycol temperature was set as the average over the testing date simulated. The water would be supplied as an open loop, so it would be continuously entering the system at
25 C. The glycol, on the other hand, would be circulated throughout the system. The wax melting characteristics were set as described in the simpler model.

To simulate the heat exchanger, a solid tube wall and fin structure was simulated within FLUENT with the appropriate properties of copper. The assumption made here to simplify the FLUENT model was that the plumbing flux used to assemble the heat exchanger was not modeled. A mapped mesh was used wherever possible on the simulation to assure a low skewness value.
CHAPTER FIVE: RESULTS AND DISCUSSION

Experimental Processes

The initial results of the solar thermal storage unit were achieved in November 2009, and provided insight into the success of the unit’s design. Testing continued through June 2010. Tests were conducted during the dry and rainy seasons to be able to compare similar flow rates with different humidity conditions. Humidity in the air diffuses the sun’s radiation and so prevents some of it from reaching the solar thermal collectors. The humidity increases as summer comes in Florida, so this is of interest if the unit is to be used for a residence. The main interest in the solar collection test was the melting of the wax, as this was the cardinal benefit of adding fins to the heat exchanger within the wax. The time that this process took varied based upon the mass flow rate of the heat transfer fluid as well as the weather conditions on the test day. For example, a cloudy day would result in more time required to charge the unit, since the sun’s radiation would be obscured.

Results of this testing are presented with different flow rates as described in the methodology section and also with different times of the year to examine the effect of more humidity in the air. These results are shown in Figure 29 and Figure 30.
Figure 29: Flow Rate Comparison
The maximum amount of heat collected was using a flow rate of 0.5 gpm. This is due to the fact that first, more heat is collected by the solar thermal evacuated tubes in the first place as the humidity is low. Furthermore, the optimal flow rate of the three tested provides enough heat transfer by convection from the manifold of the collector to the working fluid but is also balanced by heat transfer by conduction to the wax through the heat exchanger. Finding the optimal flow rate for the working fluid is crucial because heat transfer by convection is increased with turbulent flow, but this does not allow ample time for conduction to occur to the copper heat exchanger and finally to the wax. Furthermore, although in the summer months there is more humidity in the air, the longer daylight hours can help to make up for this fact since the unit may have more opportunity to collect the sun’s radiation.
Heat extraction from the unit was tested once the thermocouples read above the melting point of the wax, but thermocouple location 4B was used as the main designation in this case, as it lagged behind the others during the solar thermal collection test. Once this was complete, freezing tests were conducted using city water at city temperature and pressure values, as this is the fluid that would be used in a residence. The best open loop discharge test is shown in Figure 31.

The main result from these tests is that the freezing of the unit is not ideal with one pass. The fluid may warm up during the first fifteen minutes of operation, but this does not remain the case for the hour during which the unit is tested. Several passes through the unit would be
needed to achieve an acceptable water temperature (between 120°F and 140°F). Another test was done to examine the closed loop discharge process to see if this would result in the desired temperature of the water.

The closed loop freezing test is shown in Figure 32. After one hour of testing, the water reaches about 54°C (124°F), which is just above the 120°F minimum. This recirculation is then necessary if this unit were to be deployed for domestic use. A solenoid valve at the exit of the storage unit could be used to control the water exiting the unit once it reaches the minimum temperature, then fresh water could be run through the rig to heat it up.

![Figure 32: Freezing Temperatures with Recirculation](image-url)
Analytical Model

The analytical model is validated based upon the experimental results from the rig and the results achieved by Lamberg and Siren, since their conditions were the closest found in literature to the present study. Furthermore, the model will be compared to the one hour of discharging of heat from the rig, since this was the maximum possible for useful water heating. It will also be compared to the various points throughout the charging process. This comparison will show how accurate the model is while pointing out areas of improvement and areas that cannot be simplified or neglected when designing such a model.

First, the freezing model is considered. The one-dimensional freezing analytical model from Lamberg and Siren represents a symmetrical cell with an internal fin, and the analytical model developed for the current study was used with the same conditions. The phase change material properties and fin material properties and the dimensions of the enclosure were placed in the analytical model to see how well it would compare with Siren and Lamberg’s results. The fin material used was aluminum, and the phase change material was n-octodecane, with the unit undergoing the freezing process for 724 seconds. These results are compared in Figure 33. The first evident result is that the developed analytical model grossly overpredicts the distance of the freezing front from the wall and the fin when compared to the results of Lamberg and Siren. This error will also be seen when comparing the analytical model to the experimental rig results – the model overpredicts the amount of frozen wax as well as the temperature exiting the thermal storage unit.
This result points to the fact that there are other aspects to the freezing model that must be taken into consideration. One point brought up by Siren and Lamberg is that the geometry of the finned enclosure has a significant effect on whether or not a one-dimensional model should be used. In the present case, the finned enclosure is defined more loosely than in Siren and Lamberg’s work. Furthermore, the energy equation is defined simply in the one-dimensional control volume sense, not in the full two-dimensional aspect as in Siren and Lamberg’s model (shown in the literature review section, Equations 9-18). The addition of the full energy equation in two dimensions to the present analytical freezing model could lead to a more accurate solution. Furthermore, the parameters of the candle wax used in the current study were assumed
constant regardless of melting or solidification, which may be another factor in the inaccuracy of the freezing process.

Next, the melting analytical model is validated against the melting model developed by Lamberg and Siren. The melting model developed in the present study compares much more closely with Lamberg and Siren’s model than the freezing model. The same materials and parameters were used, and the total time was 3600 seconds. This could be explained by the fact that the same heat transfer coefficient calculation is used once the melted wax reaches a critical thickness and natural convection begins to take hold. Furthermore, the fact that the geometry is not defined as clearly as in Lamberg and Siren’s study can be seen here, as the endwall distance of the enclosure does not have a clear effect on the fin temperature. This reflects the fact that in the model, the fin is collapsed by the resistance method.
Numerical Model (FLUENT)

The FLUENT simulation was run with the boundary conditions and initial conditions described in the Methodology section. To provide initial verification of the simulation method used, a simple two-dimensional model was developed. This included one half of the wax section between fins, a half-width of the fin, and the equivalent wall of the pipe with the convective heat transfer from the liquid going through the pipe. This section was taken at the entrance to the storage unit, for both the melting and solidification cases. FLUENT models were run for one, two, three, and four hours of heat extraction. Also, the contours of liquid fraction were noted, as these indicated the “mushy zone” of the wax during the freezing process. The contours of liquid fraction for the freezing process are shown in sequence in Figure 35.
The process starts at the one hour mark at the top plot in the set, then is shown for hours two through four. The location of the freezing front makes sense; it increases with time and is
slightly curved, based upon the location of the cold fluid entering the fin section at the bottom of each snapshot. Furthermore, there would be less heat transfer moving along the copper wall from the bottom to the top of the figure, because the temperature of the fluid would begin to approach the temperature of the wax. The corresponding melting sequence is shown in Figure 36 for one through four hours, with each picture representing a time step of one hour. This is also a qualitative process done to ensure that the modeling parameters set up make sense before creating a more complex FLUENT simulation.
The other main result from the FLUENT simulation is the heat transfer over change in temperature value at the fin, which corresponds to its resistance. This value is computed by having FLUENT compute the heat transfer through the fin, then dividing it by the temperature difference (in this case, it is between the melting point of the wax, 323 K, and the temperature of the fluid through the pipe, 298 K). These results will be compared to the resistance results achieved in the analytical model to compare the two methods.

Once the simplified model was completed, a two dimensional simulation was created to take a slice of the entire heat exchanger, which would be run and compared against actual experimental data. This slice was taken horizontally through the unit and resembles the cross-section shown in Figure 20. Figure 37 shows the sequence of melt fraction of the system for the simulated experimental case over one hour, four hours, seven hours, and thirteen hours. The melt fraction ranges from 0 to 1, with 0 being completely frozen material and 1 being completely
melted material. The water and ethylene glycol used do not freeze or melt at any time in the
process; they remain at 1 as far as the liquid fraction is concerned. The lower limit of the
melting fraction is shown in blue, while the higher limit is shown in red.
From the results presented one sees that the unit is approximately 95% in the melted state after 13 hours. As will be seen when comparing this model to the experimental data, this FLUENT simulation can predict temperatures of both the wax and the heat transfer fluid within 10%, which indicates the simulation is a reasonable one. This is important because the melting process of the wax cannot be seen while the unit is operating, so with a simulation such as FLUENT different designs can be accurately tested.

Figure 38 shows the temperature contours of the unit during this process. As expected, the heat transfer fluid is cooler when it exits the storage unit, as it has lost heat to the storage unit. It also indicates, along with the melt fraction results, the fact that the solid-liquid boundary of the wax is not a perfect line, rather, there is a mushy zone.
Figure 38: Melting Process Temperatures
As for the freezing case, the CFD and experimental results compare within 10% as with the melting case. At most, the exit temperature of the water was approximately 20°C, which occurred after 15 minutes. A slower mass flow rate would be required to extract heat if nothing else on the unit was changed. Alternatively, the water could make more than one pass through the unit to achieve the minimum safe hot water temperature of approximately 50°C (120°F) [9]. This fact was also demonstrated using the experimental closed loop run of the discharge side of the unit.

The wax solidification process during the extraction of heat from the wax can be seen in Figure 39 and Figure 40. The results are presented in a similar way to those of the melting process, with the liquid fraction presented then the temperature contour results. It is evident in these simulation results that the fins of the heat exchanger serve as nucleation sites for the freezing process as well as the tubes. It is also interesting to note how the connection of the heat exchange and heat load loops are connected by the fins in this case; the fins allow for some freezing to take place along the heat exchange loop. This calls attention to the fact that it might be useful at some points during the discharge cycle of the unit to run the charging cycle for some time to keep the wax around the heat exchange loop in the melted or mushy state.
Figure 39: Freezing Process: Liquid Fractions
Figure 40: Freezing Process: Temperatures
To examine the effect of adding more fins to the heat exchanger and maintaining the same configuration otherwise, two more sets of simulations were run, using otherwise the same conditions as the simulation run for validation against the experimental results. The first results are those found using 18 fins in the heat exchanger instead of 14. The melting results are presented, then the freezing results. Figure 41 shows the liquid fraction contours, while Figure 42 shows the temperature contours. The main interest of this simulation is to see if the additional fins decrease the time required to melt the unit completely.
From the last time step shown in Figure 41, it is evident that the 18 fins provide approximately the same results as the 14 fins when charging the unit. However, the distribution of the melted wax is spread out more due to the fact that there are more fins within the unit. It also is very obvious that at the top of the figure, there is very little melted wax. This is due to the fact that there are no fins on this loop. This indicates that the storage unit might benefit from the addition of fins around the bend on both the head load and heat exchange loops to extend the benefit of the fin to that region.
Figure 42 and Figure 44 show the corresponding freezing process over one hour using 18 fins. The melt fraction is shown but is of less interest for this case since the temperature of the
water at the exit of the unit is the crucial value when running the heat extraction process.

However, the melting fractions also show one drawback of adding more fins to the storage unit. With 18 fins, there is more freezing along the heat exchange tubes, more so than with only 14 fins. This again illustrates the need for more fins at the top of the unit, as shown in the simulation results.
Figure 43: Freezing Process (18 Fins): Melt Fractions
Now, the melting and freezing cases with 22 fins are considered. The melting process is shown in Figure 45 and Figure 46, while the freezing process is shown in Figure 47 and Figure 48. With the melting process, it is evident that once again, the wax is 95% melted at the end of the 13 hour simulation. However, an even more even distribution of the melted wax can be seen in this simulation. This also indicates that the unit would benefit from fins on the horizontal section (as seen in the images, near the top of the simulation).

Furthermore, the trend observed from the 18 fin freezing process can be also seen here in the 22 fin freezing process. Freezing of the wax occurs more over the heat exchange tubes since there are more avenues for the heat to move along. This will be an issue that must be addressed if the amount of fins within the storage unit are increased – the heat load loop may have to be disconnected from the heat exchange loop via the fins, otherwise freezing will occur too quickly.
This would require the heat load loop to be identical to the heat exchange loop in length and shape so that it would extend through most of the storage unit, as well as have its own set of fins.
Figure 45: Melting Process (22 Fins): Liquid Fraction
Figure 46: Melting Process (22 Fins): Temperatures
Figure 47: Freezing Process (22 Fins): Melt Fractions
Figure 48: Freezing Process (22 Fins): Temperatures
Comparisons of Results

With the variety of procedures used, a comparison of the results is necessary to evaluate the accuracy of each process and the usefulness of the results achieved. In comparing these results, the best ways to optimize the operation of the unit can be determined. Furthermore, the assumptions made in each model can be evaluated based upon the differences between the experimental data and the model.

Validation of Analytical Model

The analytical model was validated using the work of Lamberg and Siren [52],[55], which taken into consideration different cases of semi-infinite finned storage units. The cases closest to the current model are used for validation. The results of the validation show much the same trends that occur when comparing the analytical model to the experimental model, so from this validation several conclusions can be drawn about what must be taken into consideration when developing such a model. First, the geometry should be considered more in detail than has been done in the current analytical model. Non-dimensionalizing the equations used would help to take this into account. Furthermore, this step could be combined with the energy equation, taken from its most complex form and gradually narrowed down using assumptions based upon the storage unit’s operation. Also, the natural convection may be an issue in the freezing process as well as in the melting process. This would impede the freezing process, as the movement of the melted wax from lower points to higher points would strip away the wax in the mushy zone. This may also explain the analytical results versus the experimental results, which will be explained shortly.
Analytical vs. Numerical Model

Figure 49 compares the analytical and experimental resistance values for the first finned portion of the heat exchanger. This includes the pipe wall and half of the first fin, as shown in Figure 12.

These results show the same trends, but it is also evident that there is a certain amount of error in the analytical model’s method of calculation for the freezing process when compared to the numerical techniques used by FLUENT. Another consideration is the fact that while the melting resistance increases in the analytical model, it decreases in the FLUENT model. This is due to the fact that the melting resistance depends upon natural convection resistance as well as the conduction resistance from the molten wax. Since the thickness of the molten wax will
increase, that corresponding conduction resistance will increase over time. Once the natural convection begins, however, its resistance will decrease over time due to the larger heat transfer coefficient.

The decrease in value is not seen in the analytical model due to the fact that, until the melted wax reaches a critical thickness, the melted wax offers an extra resistance. Once the wax reaches this critical thickness, the resistance shifts from conduction to convection. This convection heat transfer may result in an increased resistance as seen in the trends. Although the melting model accurately predicts the temperature output of the storage unit, as will be seen in the next section, the resistance values do not follow the trends of the more reliable FLUENT model.

Analytical Model vs. Experimental Data

For a selected testing date using a flow rate of one gallon per minute, the analytical model was compared to the experimental data taken for both the charging and discharging processes. The specific value compared was the temperature of the glycol or water at the exit of the storage unit. This would indicate how much energy was being stored in the charging process of the unit and the temperature the water would reach in the discharging process. These values are compared in Figure 50.
The melting process compares much more accurately than the freezing process does – the values are within ten degrees away from the actual fluid exit temperatures in the experimental data. This can be explained by the fact that the analytical model takes into consideration the natural convection from the melted wax during this process once the melted wax reaches a critical thickness, as defined by the Rayleigh number. With the freezing process, any natural convection occurring is not taken into consideration, and this would limit the progress of the freezing of the wax over time. Also, as pointed out by Lamberg and Siren [55], the geometry of the unit has an effect upon whether or not one may use a one-dimensional model. The enclosure is not clearly defined in the one-dimensional model due to the fact that the resistance method collapses the fins that create the enclosure of the space. Furthermore, the wax freezing on the
sides of the storage unit enclosure was not included in the calculations. This may have occurred in the experimental rig due to the fact that, although there was one foot of Styrofoam insulation, the rig was not perfectly insulated.

Also, in the analytical model, the freezing process was assumed to occur only on the tubes containing the water. As shown by the FLUENT processes, the freezing of the wax also occurs along the glycol tubes, since the two sets of tubes are connected by the fins. This leads to the glycol tubes acting as another nucleation site for the frozen wax besides the fins and the water tubes. The model would have to take into consideration the fact that the freezing front occurs along the glycol tubes as well as the water tubes to achieve better accuracy.

Numerical Model vs. Experimental Data

The numerical model accurately predicts the melting of the wax and the fluid temperatures within a few degrees. This fact is shown in Figure 51 and in Figure 52. The experimental melting process and the CFD melting process are within ten degrees of each other for the first seven hours of unit operation, but at the 13 hour mark, the CFD is significantly higher at the 1B and 6B locations. Interestingly, the 4B location is relatively accurate, and it the closest during the entire simulation. This may be due to the fact that location 4 experiences the least bit of temperature change during the entire process, while locations 1 and 6 experience the greatest and least heat transfer due to their location within the storage unit. Location 1 is considered to be at the exit of the heat transfer loop whilst location 6 is at the entrance of the heat transfer loop.
The glycol temperature results are also reasonably predicted during the melting process by the CFD. These results are compared in Figure 52, and are realistic as the temperature should be rising over the time presented. Furthermore, the temperatures predicted by the CFD simulation are higher than those experimentally found, due to the fact that the CFD process is ideal heat transfer (the storage unit is perfectly insulated, there is no contact resistance due to the solder attaching the copper fins to the copper tubing), but the experimental process experiences these two things.
The outlet temperature of the water in the discharging CFD model is six degrees higher than the inlet temperature of the water at 900 seconds, which is approximately twice the increase noted in the experimental data. The fact that the heat transfer to the water in the CFD model is higher than reality makes sense, as the CFD calculates ideal heat transfer. In the experiment, there was heat loss through the aluminum box the wax was encased in, as well as loss to bubbles forming in the city water supply. Considering the wax temperature, the wax was, on average, near the melting point at all data points taken within the storage unit. Comparing this to the same location in the CFD model, the model also has this feature. However, it must also be noted that the wax used within the storage unit was not laboratory grade purity, therefore it has a larger melting point range than the storage unit modeled.
Finally, the change of temperature of the wax during the discharging process is predicted and the temperature rises of the wax in the experimental process and numerical process are compared in Figure 53. The temperature change is roughly three degrees higher in the FLUENT process over the hour of data compared. This is due to the fact that the outside wall of the storage unit is perfectly insulated in the FLUENT calculations, however, it is not perfectly insulated in reality. Furthermore, there could be contact resistance at the soldered joints where the fins meet the pipes of the heat exchanger, which is not taken into consideration in the FLUENT model.

Also, as seen in the FLUENT simulation, the glycol tubes as well as the water tubes serve as nucleation sites for the freezing wax. The FLUENT model takes this into consideration, which is why the temperature change predicted is closer than the change predicted by the analytical model.
Figure 53: Freezing Temperature Changes
CHAPTER SIX: CONCLUSIONS AND FUTURE WORK

With the analytical model and numerical models developed based upon the experimental results of the thermal storage unit, there are several suggestions that can be made for future work based upon the utility of the storage system. The experimental rig will be addressed first, then the analytical and numerical models.

Experimental Rig

The experimental rig proved to be successful in the melting of the wax; a total melt could be achieved over approximately two days with the optimal volumetric flow rate of 1 gallon per minute of heat transfer fluid. However, the main problem was the extraction of heat from the unit. At the start, the extraction allowed for an acceptable hot water temperature, however, this temperature rapidly decreased with time. The simplest improvement would be the increasing of the wax melt temperature. Alternatively, the phase change material could be encapsulated instead, leading to a more direct transfer of heat to the fluid, as it would be passing over many thousands of capsules rather than through a heat exchanger. This would also take advantage of the known correlations for flow over a packed bed. It would also be beneficial to run the rig in charging mode if discharging is done during the day, so that the wax remains close to the melted state during the discharging process. It also must be borne in mind that this storage unit was a very preliminary prototype; the most crucial components were limited by cost. This lead to an inexpensive and impure wax being used, and only fourteen copper fins due to their cost. With these two facts taken into consideration the experimental rig produced acceptable results.

One good aspect of the experimental rig was its insulation properties. With the layers of insulation provided, the unit’s temperature dropped two to three degrees overnight, which aids in
keeping a fully charged storage unit. Even if the rig is discharged during the day and the wax begins to freeze, this helps because the next day the rig can be charged from a higher wax temperature, and so requires less energy to be charged. This design element should be kept in any future iterations of the rig – particularly since the Styrofoam insulation used was inexpensive and would not add significant cost to the storage unit.

For the future, another change in the rig would be the disconnection between the charging and discharging loops of the system. Currently, the fins are brazed to both the charging and discharging loops, and so there is heat transfer by conduction along the fins during both the charging and discharging processes. This disconnection, as well as the addition of fins along the shortest portion of the heat exchanger (shown at the top of the FLUENT simulation results), has the potential to improve the storage unit’s performance. This could be studied using a FLUENT simulation before the construction of a rig.

Furthermore, the use of a commercially developed heat exchanger within the rig could be another option. This idea was tested by Medrano et. al with some success [25], and other similar studies could be used as the basis for such a design. The use of a commercial heat exchanger would make use of an existing design, which would save time in the design process of the storage unit as a whole. Even if the commercial heat exchanger is not an exact solution to the problem, it would be beneficial to do an epsilon-NTU analysis of the heat exchanger within the storage unit. This would improve the performance of the heat exchanger during the charging of the storage unit. However, the freezing process would require the opposite heat transfer process, so this must also be carefully implemented with the epsilon-NTU method. Therefore, a separate heat exchanger design for the two different processes would be most appropriate if this storage unit is optimized further.
Analytical Model

One main weakness of the analytical model is that it does not fully take into consideration gravity’s effect on the melted wax. The melting wax will proceed to the bottom of the rig, while the frozen wax remains at the top since it is lighter. Another crucial element that the model neglects is the curvature of the melting and freezing fronts. Since the model is only one dimension, it cannot predict the two dimensional nature of the curved front. A way to remedy this would be to use the energy equation in two dimensions in place of the resistance model. However, the gravity is somewhat taken into consideration with the fact that the Rayleigh number is calculated and a heat transfer coefficient is determined for the molten wax when it reaches a critical thickness. These two conditions should be maintained in any future analytical modeling. The resistance method used within the model is best for simple shapes, such as the tube portions of the heat exchanger. This portion could be left alone in the model, but the fins require special consideration, as seen in the comparison between the experimental and analytical results.

Overall, the model achieved relatively good accuracy when predicting the melting performance of the rig, i.e. the temperatures into and out of the storage during the melting process. The other benefit of the model is its simplicity; there are no Navier-Stokes equations to be solved. The entire process is a simple resistance model, which is a basic heat transfer concept. The melting model could potentially be used going forward, but changes must be made to the freezing model to successfully apply it in future applications.
Numerical Model

As identified, the main issue with the numerical model is the fact that different measures of density, viscosity and other relevant fluid parameters cannot be changed from liquid to solid as the FLUENT model proceeds in calculation. A future simulation should make use of the user-defined functions in FLUENT to address the change in these parameters. Another user-defined function should be used to define the change in temperature over the course of the day, which would lead to a more accurate solution of the FLUENT model.

Another consideration that must be made is the natural convection throughout the unit. A full three-dimensional simulation of the storage unit would be beneficial in examining this phenomenon. Then, it would be possible to accurately predict the time necessary to melt the entire unit. This would be useful not only for the current study but also for future designs. To begin this three-dimensional simulation, a two-dimensional simulation should be done of the entrance portion of the storage unit. The plane would be in the vertical direction and so would take into consideration the gravity.

This work represents a starting point for a storage unit of this size and capacity. The next steps in the development of such a unit would be running FLUENT simulations for the improvements suggested here. This would provide a reliable source for construction of another prototype and, ultimately, the commercialization of this storage unit.
APPENDIX A: PUMP HEAD LOSS CALCULATIONS – HEAT LOAD SIDE
\[
Q := \frac{8 \text{ gal}}{\text{min}} \quad d := 0.5 \text{ in} \quad \text{Area} := \frac{\pi}{4} d^2 \quad \text{Vel} := \frac{Q}{\text{Area}} \quad \text{Vel} = 3.984 \frac{\text{m}}{s}
\]

\[
\mu_{\text{H}_2\text{O}} := 1 \cdot 10^{-3} \text{Pa}\cdot\text{s} \quad \rho_{\text{H}_2\text{O}} := 1000 \frac{\text{kg}}{\text{m}^3} \quad \text{Reynolds} := \frac{\text{Vel} \cdot d}{\mu_{\text{H}_2\text{O}}} \cdot \rho_{\text{H}_2\text{O}}
\]

\[
\text{Reynolds} = 5.06 \times 10^4
\]

Standard elbow at 90 degrees:

\[
\text{LebyD} := 30 \quad \text{This represents the equivalent length divided by diameter for the standard elbow.}
\]

\[
\text{rbyD} := \frac{1.5\text{ in}}{0.5\text{ in}} \quad \text{rbyD} = 3
\]

Note: \text{L sub e} was taken from the 6th edition of Introduction to Fluid Mechanics by Fox and McDonald, page 346. Value of 12 corresponds to the radius of curvature divided by the diameter of the pipe, which was then compared to the chart on that page.

\[
\epsilon_{\text{Cu}} := 5 \cdot 10^{-6} \text{ ft} \quad \frac{\epsilon_{\text{Cu}}}{d} = 1.2 \times 10^{-4}
\]

So from the Moody Chart:

\[
\text{f}_{\text{pipe}} := 0.0375
\]

\[
\text{length}_1 := 2 \cdot 19.5\text{ in} + 8\text{ in}
\]

\[
\text{h}_{\text{loss}_1} := \frac{\text{f}_{\text{pipe}} \cdot \text{length}_1 \cdot \text{Vel}^2}{2 \cdot d \cdot g} \quad \text{h}_{\text{loss}_1} = 9.361\text{ ft}
\]

Elbow losses:

\[
\text{h}_{\text{elbow}} := \frac{\text{f}_{\text{pipe}} \cdot 30 \cdot \text{Vel}^2}{2g} \quad \text{h}_{\text{elbow}} = 2.987\text{ ft}
\]

Loss based on the 90 degree bend:

Note: the calculations above would correspond to having 24 gpm total from the pump, because the flow rate would have to be divided by three to approximate the amount of water flowing through each branch of the heat load tubing.

\[
d_{\text{down}} := 0.75\text{ in} \quad L_1 := \frac{5}{8}\text{ in} \quad L_2 := 6\text{ in} \quad L_3 := L_2
\]

\[
\text{Reynolds}_{\text{down}} := \frac{\text{Vel} \cdot d_{\text{down}}}{\mu_{\text{H}_2\text{O}}} \cdot \rho_{\text{H}_2\text{O}}
\]

\[
\epsilon_{\text{Cu}} \quad \frac{d_{\text{down}}}{d_{\text{down}}} = 8 \times 10^{-5}
\]

\[
\text{h}_{\text{loss}_{\text{d}_1}} := \frac{\text{f}_{\text{pipe}} \cdot L_1 \cdot \text{Vel}^2}{2 \cdot d_{\text{down}} \cdot g}
\]

\[
\text{h}_{\text{loss}_{\text{d}_2}} := \frac{\text{f}_{\text{pipe}} \cdot L_2 \cdot \text{Vel}^2}{2 \cdot d_{\text{down}} \cdot g}
\]

\[
\text{h}_{\text{loss}_{\text{d}_3}} := \frac{\text{f}_{\text{pipe}} \cdot L_3 \cdot \text{Vel}^2}{2 \cdot d_{\text{down}} \cdot g}
\]
Three more losses due to the elbow at the bottom as well as the two tees on the copper piping.

\[
h_{\text{loss, tee}} := \frac{f_{\text{pipe}} \cdot 60 \cdot \text{Vel}^2}{2 \cdot g} \quad \quad h_{\text{loss, elbow}} := \frac{f_{\text{pipe}} \cdot 30 \cdot \text{Vel}^2}{2 \cdot g}
\]

\[
\text{loss, connect} := 2 \cdot h_{\text{loss, tee}} + h_{\text{loss, elbow}}
\]

\[
\text{total_loss} := h_{\text{loss, 1}} + h_{\text{loss, d, 1}} + 2 \cdot h_{\text{loss, d, 2}} + \text{loss, connect} + h_{\text{elbow}}
\]

\[
\text{total_loss} = 28.961 \text{ ft}
\]

This represents the loss in one branch of the heat load side exchanger, thus it must be multiplied by 3 to represent the total loss in the heat load pipe assembly:

\[
3 \cdot \text{total_loss} = 86.884 \text{ ft}
\]
APPENDIX B: PUMP HEAD LOSS THROUGH HEAT EXCHANGER
Fluid properties are glycol at 100 C.

\[ \text{vol}_{\text{dot}} := 20 \text{ gal/min} \quad \mu := 2.75 \text{-mPa·s} \quad c_{\text{copper}} := 5 \times 10^{-6} \text{-ft/\mu\Omega} \quad \text{den} := 1.036 \text{-g/cm}^3 \quad c_{\text{silicon}} := 75 \times 10^{-6} \text{-in/\mu\Omega} \]

**Manifold**

39 5/8 long
5 wide
3 15/16 thick

Assume 1/8 thickness of manifold inner reservoir

\[ l_{\text{man}} := \frac{1}{8} \text{in} \quad t_{\text{man}} := \left( 39 + \frac{5}{8} \right) \text{in} \]

\[ A_{\text{mancavity}} := \left( 5 - 2 \cdot t_{\text{man}} \right) \text{in} \left( 3 + \frac{15}{16} - 2 \cdot t_{\text{man}} \right) \text{in} \]

\[ A_{\text{tubeinset}} := \left( \frac{5}{8} \right) \text{in} \left( 2 + \frac{3}{16} \right) \text{in} + \left( 2 + \frac{15}{32} \right) \text{in} \left( 3 + \frac{5}{8} \right) - \left( 2 + \frac{3}{16} \right) \text{in} \]

\[ A_{\text{manifoldequiv}} := A_{\text{mancavity}} - A_{\text{tubeinset}} \]

\[ D_{\text{hydraulicMan}} := \frac{4 \cdot A_{\text{manifoldequiv}}}{\text{peri}_{\text{man}}} \quad D_{\text{hydraulicMan}} = 2.776 \text{ in} \]

\[ v_{\text{man}} := \frac{\text{vol}_{\text{dot}}}{\pi \left( \frac{D_{\text{hydraulicMan}}}{2} \right)^2} \quad v_{\text{man}} = 0.323 \text{-m/s} \]

\[ \text{Re}_{\text{man}} := \frac{\text{den} \cdot v_{\text{man}} \cdot l_{\text{man}}}{\mu} \quad \text{Re}_{\text{man}} = 1.225 \times 10^5 \]

Turbulent Flow Through Manifold

\[ f_{\text{man}} := 0.024 \]

\[ B_{\text{manifold}} := f_{\text{man}} \frac{l_{\text{man}} v_{\text{man}}^2}{D_{\text{hydraulicMan}}^2} \quad B_{\text{manifold}} = 0.018 \frac{\text{m}^2}{\text{s}^2} \]

\[ \frac{c_{\text{copper}}}{D_{\text{hydraulicMan}}} = 0.000021615 \]
**Exit manifold copper pipe**

\[ l_{\text{exitcopperpipe}} := \left( 1 + \frac{3}{4} \right) \text{in} \]

\[ d_{\text{exitcopperpipe}} := \frac{3}{4} \text{in} \]

\[ v_{\text{exitcopperpipe}} := \frac{\text{vol}_{\text{dot}}}{\pi \left( \frac{d_{\text{exitcopperpipe}}}{2} \right)^2} \]

\[ \text{Re}_{\text{exitcopperpipe}} := \frac{\text{den} \cdot v_{\text{exitcopperpipe}} \cdot l_{\text{exitcopperpipe}}}{\mu} \]

\[ v_{\text{exitcopperpipe}} = 4.427 \frac{\text{m}}{\text{s}} \]

\[ \text{Re}_{\text{exitcopperpipe}} = 7.413 \times 10^4 \]

**Turbulent flow through Exit Copper Pipe**

\[ f_{\text{exitcopperpipe}} := 0.026 \]

\[ \frac{e_{\text{copper}}}{d_{\text{exitcopperpipe}}} = 0.00008 \]

\[ h_{\text{exitcopperpipe}} := f_{\text{exitcopperpipe}} \left( \frac{l_{\text{exitcopperpipe}} \cdot v_{\text{exitcopperpipe}}}{d_{\text{exitcopperpipe}}^2} \right)^{\frac{2}{3}} \]

\[ h_{\text{exitcopperpipe}} = 0.594 \frac{\text{m}^2}{\text{s}} \]

\[ h_{\text{manifoldexitconn}} := K_{\text{manifoldexitconn}} \frac{(v_{\text{exitcopperpipe}} + v_{\text{man}})^2}{2} \]

\[ h_{\text{manifoldexitconn}} = 1.41 \frac{\text{m}^2}{\text{s}} \]

**Silicon Piping**

\[ \text{OD} := 1 \text{-in} \]

\[ d_{\text{silicon}} := \frac{3}{4} \text{-in} \]

\[ l_{\text{silicon}} := 1.50 \text{-in} \]
\[ v_{\text{silicon}} := \frac{\text{vol}\dot{\text{dot}}}{\pi \left( \frac{d_{\text{silicon}}}{2} \right)^2} \]

\[ v_{\text{silicon}} = 4.427 \frac{\text{m}}{\text{s}} \]

\[ \text{Re}_{\text{silicon}} := \frac{\text{den} \cdot v_{\text{silicon}} \cdot l_{\text{silicon}}}{\mu} \]

\[ \text{Re}_{\text{silicon}} = 6.354 \times 10^6 \]

Turbulent flow through Silicon Pipe

\[ f_{\text{silicon}} := 0.0116 \]

\[ \frac{c_{\text{silicon}}}{d_{\text{silicon}}} = 0.0001 \]

\[ h_{\text{silicon}} := f_{\text{silicon}} \frac{l_{\text{silicon}} \cdot v_{\text{silicon}}}{d_{\text{silicon}}^2} \]

\[ h_{\text{silicon}} = 22.734 \frac{\text{m}^2}{\text{s}^2} \]

Silicon Piping Turns

Assume heat collection at 52 deg

90 deg turn

\[ L_{\text{equiv}90\text{turn}} := 12 \]

\[ h_{90\text{degturn}} := f_{90\text{degturn}} \frac{L_{\text{equiv}90\text{turn}} \cdot v_{\text{silicon}}^2}{2} \]

\[ h_{90\text{degturn}} = 1.364 \frac{\text{m}^2}{\text{s}^2} \]

38 deg turn

\[ L_{\text{equiv}40\text{turn}} := 10 \]

\[ h_{40\text{degturn}} := f_{40\text{degturn}} \frac{L_{\text{equiv}40\text{turn}} \cdot v_{\text{silicon}}^2}{2} \]

\[ h_{40\text{degturn}} = 1.137 \frac{\text{m}^2}{\text{s}^2} \]
APPENDIX C: EXPERIMENTAL DATA: CHARGING PROCESS
APPENDIX D: EXPERIMENTAL DATA: DISCHARGING PROCESS
APPENDIX E: MATLAB CODE FOR DISCHARGING PROCESS
clc;
clear;
clear all;

k_Cu = 400; %W/m*K
rho_gly = 1.113*10^3; %kg/m^3
mu_gly = 1.61*(10^(-2)); %Pa-s
nu_gly = mu_gly/rho_gly;
Cp_gly = 2.409*(10^3); %J/kg*Ks
k_gly = 0.25; % W/m*K
alpha_gly = k_gly / (rho_gly*Cp_gly); %Units
Pr_gly = nu_gly/alpha_gly; %unitless
D_pipe_inner = 0.011; %m
D_pipe_outer = 0.013; %m
P_inner = D_pipe_inner*pi;
P_outer = D_pipe_outer*pi;
A_pipe_inner = (pi*(D_pipe_inner)^2)/4; %m^2
L_pipe = 2.425; %m
k_wax = 0.25; %W/m*K
rho_wax = 800; %kg/m^3
Cp_wax = 2.13*(10^3); %J/kg*K
alpha_wax = k_wax/(rho_wax*Cp_wax);
latent_heat = 200*10^3; %J/kg
beta_wax = 0.00011;
mu_wax = 1e-04 / rho_wax;
tol=0.01; %set error tolerance
inc=0.01; %set solver increment
vol_wall = (0.61^2)*6*(1.587*10^-3); %m^3
rho_alum = 2.7*10^3; %kg/m^3
vol_pipes = 2.425*((0.013^2)/4)*pi;
vol_wax = (0.61^3) - vol_pipes;
Cp_alum = 906; %J/kg*K
t_fin = (1/16)*0.0254;
r_1 = 0.5*0.0254;
r_2 = 3*0.0254;
r2_c = r_2 + t_fin/2;
A_fin = 2*pi*(r2_c^2 - r_1^2);

time = 0;
L_wax = 0.1; %m
while L_wax > sqrt(alpha_wax*time)
    time = time + 1; %time in seconds
end
%The wax remains as semi-infinite.
total = time/3600;

dx = 2.667 / 1667; %m
SA_inner = P_inner*dx; %m^2
SA_outer = P_outer*dx; %m^2

%Volumetric flow rate entered in gallons per minute then converted to
%meters cubed per second. Code will determine appropriate Nusselt number
%based upon the Reynolds number and other fluid conditions.
vol_flow_gpm = 3;
\[ \text{vol\_flow} = \text{vol\_flow\_gpm} \times (6.30901964 \times 10^{-5}); \quad \text{m}^3 / \text{s} \]
\[ \text{mdot} = \rho_{\text{gly}} \times \text{vol\_flow}; \quad \text{kg} / \text{s} \]
\[ \text{vel\_ht\_fluid} = \text{vol\_flow} / A_{\text{pipe\_inner}}; \quad \text{m/s} \]
\[ \text{Re\_gly} = (\rho_{\text{gly}} \times \text{vel\_ht\_fluid} \times D_{\text{pipe\_inner}}) / \mu_{\text{gly}}; \quad \text{unitless} \]
if \( \text{Re\_gly} > 2300 \)
\[ \text{Nu} = 0.023 \times ((\text{Re\_gly})^{0.8}) \times ((\text{Pr\_gly})^{0.4}); \quad \text{unitless} \]
else if \( \text{Re\_gly} < 2300 \)
\[ \text{Nu} = 4.36; \]
end
\[ h_{\text{gly}} = (\text{Nu} \times k_{\text{gly}}) / D_{\text{pipe\_inner}}; \quad \text{W/m}^2\text{K} \]

Wall resistance of copper piping and of the fin.
\[ \text{Res\_wall} = 0.035; \quad \text{K/W} \]
\[ \text{fin\_res} = \log(r_{2}/r_{1})/(2\times\pi\times k_{\text{Cu}}\times t_{\text{fin}}); \]

Initialize all arrays becoming part of the finite difference loops with zeros.
\[ T_{\text{wax\_int}} = 323; \]
\[ q_{\text{wall}} = \text{zeros}(1667,10); \]
\[ T_{\text{out}} = \text{zeros}(1667,10); \]
\[ T_{\text{mean}} = \text{zeros}(1667,10); \]
\[ \text{entropy} = \text{zeros}(1667,10); \]
\[ \text{freeze\_dist} = \text{zeros}(1667,10); \]
\[ \text{wax\_cond\_fin} = \text{zeros}(1667,10); \]
\[ \text{wax\_cond} = \text{zeros}(1667,10); \]
\[ \text{res\_total} = \text{zeros}(1667,10); \]
% Based upon above total pipe length, with 1/16 inch spacing.
\[ h_{\text{eff}} = \text{Res\_wall} + 1/(h_{\text{gly}}\times SA_{\text{inner}}); \]

Initialize \( q \) at start from the temperature of the input glycol, interface temp of the wax (assume it starts at 298 K), heat transfer at the surface of the wax, and the temperature out of the first dx section.

\[ \text{for b} = 1:10 \]
\[ \quad T_{\text{mean}}(1,b) = 298; \]
end

% \( q_{\text{wall}} \) needs to be initialized for all first times and first lengths!
% This is ok because it is the first one, no fin here!
\[ q_{\text{wall}}(1,1) = (1/h_{\text{eff}}) \times (T_{\text{wax\_int}} - T_{\text{mean}}(1,1)); \]
\[ T_{\text{out}}(1,1) = T_{\text{mean}}(1,1) + q_{\text{wall}}(1,1)/(\text{mdot}\times C_{\text{p\_gly}}); \]

\[ \text{for g} = 2:10 \]
\[ \quad \% \text{Put in} \ q_{\text{wall}} \text{ here first, based upon the wax advancing at each time} \]
\[ \text{% step.} \]
\[ \text{for k= 1:1000} \text{ \% set range accordingly} \]
\[ x=k\times\text{inc}; \]
\[ \text{if abs}(x\times\exp(x)\times\text{erf}(x) - (C_{\text{p\_wax}}\times(323 - 298))/\text{latent\_heat}\times\sqrt{\pi})\leq\text{tol} \]
\[ \text{break} \]
\[ \text{else} \]
\[ x=x+1; \]
end
if x==1000
    disp('No solution found, increase range')
end

root = x; % Comes from the previous solver.
freeze_dist(1,g) = 2*root*sqrt(alpha_wax*g); % From Stefan #.
% If the fin exists, the wax resistance happens beyond the fin.
res_wax(1,g) = log((freeze_dist(1,g)+r_1)/r_1)/(2*pi*t_fin*k_wax);
res_total(1,g) = h_eff + res_wax(1,g);
q_wall(1,g) = (1/res_total(1,g))*(Twax_int - 298);
% Then, calculate the T_out based upon this. Then, pray that it % works!!!
T_out(1,g) = T_mean(1,g) + q_wall(1,g)/(mdot*Cp_gly);
end

% Need different initialization process for Tmean and Tout.
for d = 2:1667
    if i == 95
        res_total(d,1) = h_eff + fin_res;
        % Kelvin times kelvin per watt is kelvin squared!
    elseif i == 191
        res_total(d,1) = h_eff + fin_res;
    elseif i == 286
        res_total(d,1) = h_eff + fin_res;
    elseif i == 476
        res_total(d,1) = h_eff + fin_res;
    elseif i == 572
        res_total(d,1) = h_eff + fin_res;
    elseif i == 667
        res_total(d,1) = h_eff + fin_res;
    elseif i == 762
        res_total(d,1) = h_eff + fin_res;
    elseif i == 953
        res_total(d,1) = h_eff + fin_res;
    elseif i == 1048
        res_total(d,1) = h_eff + fin_res;
    elseif i == 1143
        res_total(d,1) = h_eff + fin_res;
    elseif i == 1334
        res_total(d,1) = h_eff +
    elseif i == 1429
        res_total(d,1) = h_eff +
    elseif i == 1524
        res_total(d,1) = h_eff +
    elseif i == 1619
        res_total(d,1) = h_eff +
    else res_total(d,1) = h_eff;
    end
end
\[ T_{\text{mean}}(d,1) = T_{\text{mean}}(d-1,1) + \frac{1}{\text{res}_\text{total}(d,1)}(T_{\text{wax int}} - T_{\text{mean}}(d-1,1)) \]

end

\% T_out is based on the T_means.
for e = 1:1666
    T_out(e,1) = T_{\text{mean}}(e+1,1);
end

\% Stop here coding 11/8/09

for j = 2:10
    for i = 2:1667
        T_{\text{mean}}(i,j) = T_{\text{out}}(i-1,j); \% Mean temp from previous pipe value.
        for k= 1:1000 \% Set range accordingly
            x=k*inc;
            if abs(x*exp(x)*erf(x) - (Cp_{\text{wax}}(323 - T_{\text{mean}}(i,j)))/\text{latent heat\_sqrt}(pi))<=tol
                break
            else
                x=x+1;
            end
            if x==1000
                disp('No solution found, increase range')
            end
        end
        root = x; \% Comes from the previous solver.
        freeze_{\text{dist}}(i,j) = 2*root*sqrt(\text{alpha\_wax}*j); \% From Stefan #.
        \% If the fin exists, the wax resistance happens beyond the fin.
        wax_{\text{cond\_fin}}(i,j) = \text{log}((\text{freeze\_dist}(i,j)+r_2)/r_2)/(2*\pi*t_{\text{fin}}*k_{\text{wax}});
        \% If there is no fin, the freeze distance just happens with the pipe.
        wax_{\text{cond}}(i,j) = \text{log}((\text{freeze\_dist}(i,j)+r_1)/r_1)/(2*\pi*t_{\text{fin}}*k_{\text{wax}});
        if i == 95
            \text{res\_total}(i,j) = h_{\text{eff}} + wax_{\text{cond\_fin}}(i,j);
        else if i == 191
            \text{res\_total}(i,j) = h_{\text{eff}} + wax_{\text{cond\_fin}}(i,j);
        else if i == 286
            \text{res\_total}(i,j) = h_{\text{eff}} + wax_{\text{cond\_fin}}(i,j);
        else if i == 476
            \text{res\_total}(i,j) = h_{\text{eff}} + wax_{\text{cond\_fin}}(i,j);
        else if i == 572
            \text{res\_total}(i,j) = h_{\text{eff}} + wax_{\text{cond\_fin}}(i,j);
        end
    end
end

end
res_total(i,j) = h_eff + wax_cond_fin(i,j);
else if i == 667
    res_total(i,j) = h_eff + wax_cond_fin(i,j);
else if i == 762
    res_total(i,j) = h_eff +
    wax_cond_fin(i,j);
else if i == 953
    res_total(i,j) = h_eff +
    wax_cond_fin(i,j);
else if i == 1048
    res_total(i,j) = h_eff +
    wax_cond_fin(i,j);
else if i == 1143
    res_total(i,j) = h_eff +
    wax_cond_fin(i,j);
else if i == 1334
    res_total(i,j) = h_eff +
    wax_cond_fin(i,j);
else if i == 1429
    res_total(i,j) = h_eff +
    wax_cond_fin(i,j);
else if i == 1524
    res_total(i,j) = h_eff +
    wax_cond_fin(i,j);
else if i == 1619
    res_total(i,j) = h_eff + wax_cond_fin(i,j);
else
    res_total(i,j) = h_eff + wax_cond(i,j);
end
end
end
end
end
end
end
end

% Wall heat transfer is calculated from the mean temperature and
% the wax interface temperature.
q_wall(i,j) = (1/res_total(i,j))*(T_wax_int - T_mean(i,j));
% Temperature coming out is the mean temperature, plus the wall
% heat flux,
T_out(i,j) = T_mean(i,j) + (q_wall(i,j)/(mdot*Cp_gly));
APPENDIX F: MATLAB CODE FOR MELTING PROCESS
clc;
clear;
clear all;

% Initial model of the pipe with heat transfer as the unit begins to heat up. Different temperatures for different times. A relation is assumed % such that the temperature increases 1 degree every 5 minutes. This will % be fixed when we have data from the rig.

k_Cu = 400; %W/m*K
rho_gly = 1.113*10^3; %kg/m^3
mu_gly = 1.61*(10^-2); %Pa-s
nu_gly = mu_gly/rho_gly;
Cp_gly = 2.409*(10^3); %J/kg*Ks
k_gly = 0.25; % W/m*K
alpha_gly = k_gly / (rho_gly*Cp_gly); %Units

D_pipe_inner = 0.011; %m
D_pipe_outer = 0.013; %m
P_inner = D_pipe_inner*pi;
P_outer = D_pipe_outer*pi;
A_pipe_inner = (pi*(D_pipe_inner)^2)/4; %m^2
Pr_gly = nu_gly/alpha_gly; %unitless
L_pipe = 2.425; %m
fin_sp = L_pipe/14; %m
k_wax = 0.25; %W/m*K
rho_wax = 800; %kg/m^3
Cp_wax = 2.13*(10^3); %J/kg*K
mdot_gly = 0.351;
alpha_wax = k_wax/(rho_wax*Cp_wax);
latent_heat = 200*10^3; %J/kg

tol=0.01; %set error tolerance
inc=0.01; %set solver increment
vol_wall = (0.61^2)*6*(1.587*10^-3); %m^3
rho_alum = 2.7*10^3; %kg/m^3

vol_pipes = 2.425*((0.013^2)/4)*pi;
vol_wax = (0.61^3) - vol_pipes;
Cp_alum = 906; %J/kg*K
Beta_wax = 0.00011;
mu_wax = 1e-04 / rho_wax;
t_fin = (1/16)*0.0254;
r_1 = 0.5*0.0254;
r_0 = r_1 - 0.0254*0.028;
r_2 = 3*0.0254;
r2_c = r_2 + t_fin/2;
A_fin = 2*pi*(r2_c^2 - r_1^2);

dx = 2.667 / 1667; %m
SA_inner = P_inner*dx; %m^2
SA_outer = P_outer*dx; %m^2
% Divides the pipe into 210 sections, 0.5 inch divisions.

vol_flow_gpm = 1;
vol_flow = vol_flow_gpm * (6.30901964 * (10^-5)); %m^3 / s
\text{mdot} = \text{rho}_{\text{gly}} \times \text{vol\_flow}; \text{kg / s}
\text{vel} = \frac{\text{vol\_flow}}{A_{\text{pipe\_inner}}}; \text{m/s}
\text{Re}_{\text{gly}} = \frac{(\text{rho}_{\text{gly}} \times \text{vel}) \times D_{\text{pipe\_inner}}}{\mu_{\text{gly}}}; \text{unitless}
\text{if } \text{Re}_{\text{gly}} > 2300
    \text{Nu} = 0.023 \times ((\text{Re}_{\text{gly}})^{0.8}) \times ((\text{Pr}_{\text{gly}})^{0.3}); \text{unitless}
\text{else if } \text{Re}_{\text{gly}} < 2300
    \text{Nu} = 4.36;
\text{end}
\text{end}
\text{h}_{\text{gly}} = \frac{(\text{Nu}\times k_{\text{gly}})}{D_{\text{pipe\_inner}}}; \text{W/m}^2\text{K}

\% Wall resistance of copper piping, and crucial information for going forward - effective resistances.
\text{Res\_wall} = 0.035; \text{K/W}
\text{h\_eff} = \text{Res\_wall} + \frac{1}{(\text{h\_gly}\times SA_{\text{inner}})};
\text{fin\_res} = \frac{\log(r_{2}/r_{1})}{(2\times pi\times k_{\text{Cu}}\times t_{\text{fin}})};

\% Initialize all arrays becoming part of the finite difference loops with zeros.
\text{Twax\_int} = \text{zeros}(95,14400);
\text{q\_wall} = \text{zeros}(95,14400);
\text{T\_out} = \text{zeros}(95,14400);
\text{T\_mean} = \text{zeros}(95,14400);
\text{frac} = \text{zeros}(95,14400);
\text{h} = \text{zeros}(95,14400);
\text{One\_by\_R\_fin} = \text{zeros}(95,14400);
\text{m} = \text{zeros}(95,14400);
\text{res\_total} = \text{zeros}(95,14400);
\text{melt\_dist} = \text{zeros}(95,14400);
\text{wax\_res} = \text{zeros}(95,14400);
\text{T\_l4} = \text{zeros}(1,14400);

\text{for } g = 1:14400
    \text{T\_l4}(g) = 338;
\text{end}
\text{for } w = 1:95
    \text{Twax\_int}(w,1) = 298;
\text{end}

\% Change this depending upon the date required.
\text{for } z = 1:14400
    \text{T\_mean}(1,z) = \text{T\_l4}(1,z);
\text{end}
\% First heat transfer thing
\text{q\_wall}(1,1) = \left(\frac{1}{\text{h\_eff}}\right)\times(\text{T\_mean}(1,1) - \text{Twax\_int}(1,1));
\text{for } d = 2:95
    \text{if } d == 95
        \text{res\_total}(d,1) = \text{h\_eff} + \text{fin\_res};
        \% Kelvin times kelvin per watt is kelvin squared!
        \% else if d == 191

159
res_total(d,1) = h_eff + fin_res;

else if d == 286
    res_total(d,1) = h_eff + fin_res;
else if d == 476
    res_total(d,1) = h_eff + fin_res;
else if d == 572
    res_total(d,1) = h_eff + fin_res;
else if d == 667
    res_total(d,1) = h_eff + fin_res;
else if d == 762
    res_total(d,1) = h_eff + fin_res;
else if d == 953
    res_total(d,1) = h_eff + fin_res;
else if d == 1048
    res_total(d,1) = h_eff + fin_res;
else if d == 1143
    res_total(d,1) = h_eff + fin_res;
else if d == 1334
    res_total(d,1) = h_eff + fin_res;
else if d == 1429
    res_total(d,1) = h_eff + fin_res;
else if d == 1524
    res_total(d,1) = h_eff + fin_res;
else if d == 1619
    res_total(d,1) = h_eff + fin_res;
else res_total(d,1) = h_eff;
end

q_wall(d,1) = (1/res_total(d,1))*(Twax_int(d,1)-T_mean(d-1,1));

T_mean(d,1) = T_mean(d-1,1) + q_wall(d,1)/(mdot*Cp_gly);
end

% Tout should go with corresponding q, for first times and first lengths
for y = 1:14400
    T_out(1,y) = T_mean(1,y) - (q_wall(1,y)/mdot*Cp_gly);
end
%T_out is based on the T_means for first time and all lengths.
for e = 2:95
    T_out(e,1) = T_mean(e-1,1);
end

for f = 2:14400
    frac(1,f) = 1 - exp(((h_eff)^2 * alpha_wax^2 * f) / k_wax^2) * erfc(((h_eff*sqrt(alpha_wax*f))/k_wax);
    Twax_int(1,f) = (T_mean(1,f-1)-298)*frac(1,f) + 298;
    if Twax_int(1,f) > 323
        for k= 1:1000 %set range accordingly
            x=k*inc;
            if abs(x*exp(x)*erf(x) - (Cp_wax*(T_mean(1,f) - 323))/latent_heat*sqrt(pi))<=tol
                break
            else
                x=x+1;
            end
        if x==1000
            disp('No solution found, increase range')
        end
    end
    root = x;
    melt_dist(1,f) = 2*root*sqrt(alpha_wax*f);
    wax_res(1,f) = log((r_1 + melt_dist(1,f))/r_1)/(2*pi*k_wax*melt_dist(1,f));
    res_total(1,f) = h_eff + 1/wax_res(1,f);
    if melt_dist(1,f) > nthroot((1900*alpha_wax*1E-04)/(9.81*Beta_wax*(Twax_int(1,f)-323)), 3)
        h(1,f) = 0.072*nthroot((9.81*(Twax_int(1,f) - 323))/(2*(rho_wax^2)*Cp_wax*k_wax^2*Beta_wax/mu_wax), 3);
        res_total(1,f) = h_eff + 1/wax_res(1,f) + 1/(h(1,f)*SA_inner);
    else
        res_total(1,f) = h_eff + 1/wax_res(1,f);
    end
    q_wall(1,f) = (1/res_total(1,f))*(T_mean(1,f) - Twax_int(1,f));
    T_out(1,f) = T_mean(1,f) - q_wall(1,f)/(mdot_gly*Cp_gly);
end

% for j = 2:1491
%     for i = 2:95
%         T_mean(i,j) = T_out(i-1,j);
%         q_wall(i,j) = (1/h_eff)*(T_mean(i,j) - Twax_int(i,j));
%         T_out(i,j) = T_mean(i,j) - (q_wall(i,j)/(mdot_gly*Cp_gly));
%         frac(i,j) = 1 - exp(((h_eff)^2 * alpha_wax^2 *j^2) / k_wax^2) * erfc(((h_eff*sqrt(alpha_wax*j))/k_wax);
%         Twax_int(i,j) = (T_mean(i,j)-298)*frac(i,j) + 298;
%     end
% end
% j
% end
% Begin new loop here, coming off of the loop for the pure conduction stuff.
% Start at 262 for 1 gal per minute flow rate, since fins make it happen
% Start at 504 with new conditions.

% earlier.

% One loop here from 2 to 5, without the melting process.

for q = 2:5
    for p = 2:95
        T_mean(p,q) = T_out(p-1,q);
        if p == 95
            res_total(p,q) = h_eff + fin_res;
            else if p == 191
                res_total(p,q) = h_eff + fin_res;
            else if p == 286
                res_total(p,q) = h_eff + fin_res;
            else if p == 476
                res_total(p,q) = h_eff + fin_res;
            else if p == 572
                res_total(p,q) = h_eff + fin_res;
            else if p == 667
                res_total(p,q) = h_eff + fin_res;
            else if p == 762
                res_total(p,q) = h_eff + fin_res;
            else if p == 953
                res_total(p,q) = h_eff + fin_res;
            else if p == 1048
                res_total(p,q) = h_eff + res_total(p,q) = h_eff + fin_res;
            else if p == 1143
                res_total(p,q) = h_eff + fin_res;
            else if p == 1334
                res_total(p,q) = h_eff + fin_res;
            else if p == 1429
                res_total(p,q) = h_eff + fin_res;
            else if p == 1524
                res_total(p,q) = h_eff + fin_res;
            else if p == 1619
                res_total(p,q) = h_eff + fin_res;
        end
        end
    end
end
end
end
end
end
end
end
end

162
frac(p,q) = 1 - \exp((h_{\text{eff}})^2 * \alpha_{\text{wax}}^2 * q^2) / k_{\text{wax}}^2) * \text{erfc}((h_{\text{eff}}*\sqrt{\alpha_{\text{wax}}*q})/k_{\text{wax}});

Twax_int(p,q) = (T_{\text{mean}}(p,q)-298)*frac(p,q) + 298;
q_{\text{wall}}(p,q) = res_{\text{Total}}(p,q)*(T_{\text{mean}}(p,q) - Twax_int(p,q));
T_{\text{out}}(p,q) = T_{\text{mean}}(p,q) - (q_{\text{wall}}(p,q)/(\dot{m}_{\text{gly}}*C_{p_{\text{gly}}}));

end

q

end

for q = 6:14400
    for p = 2:95
        T_{\text{mean}}(p,q) = T_{\text{out}}(p-1,q);
        frac(p,q) = 1 - \exp((h_{\text{eff}})^2 * \alpha_{\text{wax}}^2 * q^2) / k_{\text{wax}}^2) * \text{erfc}((h_{\text{eff}}*\sqrt{\alpha_{\text{wax}}*q})/k_{\text{wax}});
        Twax_int(p,q) = (T_{\text{mean}}(p,q)-298)*frac(p,q) + 298;
        for k = 1:1000
            x = k*\text{inc};
            if abs(x*\exp(x)*\text{erf}(x) - (C_{p_{\text{wax}}}(T_{\text{mean}}(p,q) - 323))/\text{latent}_{\text{heat}}*\sqrt{\pi}) <= tol
                break
            end
        end
        if x = 1000
            disp('No solution found, increase range')
        end
    end
end
oot = x;
melt_dist(p,q) = 2*\sqrt{\alpha_{\text{wax}}*q};
wax_{res}(p,q) = \log((r_{1} + melt_dist(p,q))/r_{1})/(2*\pi*k_{\text{wax}}*melt_dist(p,q));
res_{total}(p,q) = h_{eff} + 1/wax_{res}(p,q);
if melt_dist(p,q) > \text{n}\text{th}\text{root}((1900*\alpha_{\text{wax}}*1E-04)/(9.81*\beta_{\text{wax}}*(Twax_{int}(p,q)-323)), 3)
h(p,q) = 0.072*n\text{th}\text{root}((9.81*(Twax_{int}(p,q)-323))/(2*(\rho_{\text{wax}}*2)*C_{p_{\text{wax}}}/k_{\text{wax}}), 3);
if h(p,q) < 0
    h(p,q) = k_{wax}/melt_dist(p,q);
else h(p,q) = k_{wax}/melt_dist(p,q);
end
res_{total}(p,q) = h_{eff} + 1/wax_{res}(p,q) + 1/(h(p,q)*SA_{\text{outer}});
m(p,q) = \sqrt{(h(p,q)*(r_{2}-r_{1})*2*\pi)/(k_{\text{Cu}}*((r_{2}-r_{1})*2*\pi*t_{\text{fin}}))};
One_{by}_{R_{\text{fin}}}(p,q) = \text{real}(h(p,q)*A_{\text{fin}}*2*r_{1}*\text{besselk}(l,m(p,q)*r_{1})*\text{bessel}(l,m(p,q)*r_{2})-bessel(1,m(p,q)*r_{1})*\text{besselk}(l,m(p,q)*r_{2})/(m(p,q)*(r_{2}^2-r_{1}^2)*bessel(0,m(p,q)*r_{1})*\text{bessel}(1,m(p,q)*r_{2})+bessel(0,m(p,q)*r_{1})*\text{bessel}(1,m(p,q)*r_{2}))));
if p <= 95
    res_{total}(p,q) = res_{total}(p,q) + One_{by}_{R_{\text{fin}}}(p,q);
else if p == 24
    res_{total}(p,q) = res_{total}(p,q) + One_{by}_{R_{\text{fin}}}(p,q);
end
\begin{verbatim}
\% else if p == 36
\%     res_total(p,q) = res_total(p,q) + One_by_R_fin(p,q);
\% else if p == 60
\%     res_total(p,q) = res_total(p,q) +
\%     One_by_R_fin(p,q);
\% else if p == 72
\%     res_total(p,q) = res_total(p,q) +
\%     One_by_R_fin(p,q);
\% else if p == 84
\%     res_total(p,q) = res_total(p,q) +
\%     One_by_R_fin(p,q);
\% else if p == 96
\%     res_total(p,q) = res_total(p,q) +
\%     One_by_R_fin(p,q);
\% else if p == 120
\%     res_total(p,q) = res_total(p,q) +
\%     One_by_R_fin(p,q);
\% else if p == 132
\%     res_total(p,q) = res_total(p,q) +
\%     One_by_R_fin(p,q);
\% else if p == 144
\%     res_total(p,q) = res_total(p,q) +
\%     One_by_R_fin(p,q);
\% else if p == 168
\%     res_total(p,q) = res_total(p,q) +
\%     One_by_R_fin(p,q);
\% else if p == 180
\%     res_total(p,q) = res_total(p,q) +
\%     One_by_R_fin(p,q);
\% else if p == 192
\%     res_total(p,q) = res_total(p,q) +
\%     One_by_R_fin(p,q);
\% else if p == 204
\%     res_total(p,q) = res_total(p,q) + 0;
\% end
\end{verbatim}

q_wall(p,q) = res_total(p,q)*(T_mean(p,q) - Twax_int(p,q));
T_out(p,q) = T_mean(p,q) - (q_wall(p,q)/(mdot_gly*Cp_gly));
end
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