TRANSIENT HEAT STORAGE SYSTEMS

Final Design Report

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Abstract

Phase change materials (PCM) have many applications in transient cooling systems, including those with high transient heat loads and low duty cycles. These materials allow a system to remain within a narrow temperature range with a relatively low weight compared to conventional heat sinks or high-power cooling systems. This senior capstone project includes the design of a PCM based thermal energy storage system to integrate into an existing cooling loop, as well as a determination of viable PCM's for the application. This report contains the necessary information to build the test apparatus.

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1.0 INTRODUCTION

The sponsor and main stakeholder of the project is Pacific Design Technologies (PDT), a firm that designs liquid cooling systems for aviation electronics. The other stakeholder in the project is the class advisor, Professor Rossman, who will monitor the progression of the project and provide feedback and advice through major project milestones. PDT wants to have an optimized system ready for an upcoming technology in the aerospace and defense industries. This technology contains electronics with high power and low duty cycle, the power ratio is up to 25:1 between high and low demand. The current solution to this problem is a cooling system designed for the maximum heat load of the application. This solution is not optimal as it requires a large, heavy cooling loop that is only in use for a small portion of the operation time. More of PDT's requirements and information regarding the current cooling system is contained in Table 2 and Appendix A.

A thermal energy storage system, or "heat flux capacitor," could absorb the dynamic loads that a standard size steady state system could then slowly relieve. The goal of the project is to implement a phase change material to act as the thermal storage system without raising the operating temperature of the electronics.

The rest of this document contains a section on background information, objectives, the design process, manufacturing plan, test plan, and project management. The background section contains a repertoire of existing products, technologies, and patents concerning our project. The objectives section contains the problem statement, a summary of the goals of the project, and a set of engineering specifications by which we will measure the success of the project. The first design section illustrates the ideation process and analysis of the uncompensated and a hypothetically compensated system. The second design section discusses the design for the testing apparatus and introduces cost analysis of the system. The manufacturing section discusses how to build the design and challenges in operation. The test plan goes over the two main phases of testing: PCM properties and overall system behavior. The project management section includes the key deliverables and a timeline for the project, which sets dates to important milestones to determine if the project is on schedule.

2.0 BACKGROUND

Our sponsor would like to see if a phase change material (PCM) is an optimal way to solve this problem. Although every material is a phase change material, a suitable PCM will change phases at a specific temperature so that the material absorbs the heat without a large increase in system temperature. A typical good PCM has a high latent heat, and for our relatively quick application we also desire high thermal conductivity. Latent heat (Btu/lb) is the energy required to change a material's phase. High latent heat is ideal for PCMs as it reduces the amount of material required to absorb the heat load of the system. Thermal conductivity (Btu/ft/°F) defines a material's ability to transfer heat with its environment. High thermal conductivity allows a PCM to respond rapidly to temperature changes in the system. Because of these factors, typical PCMs include: paraffin waxes, salts, and most commonly, water. A summary of these materials can be found in Figure 1.

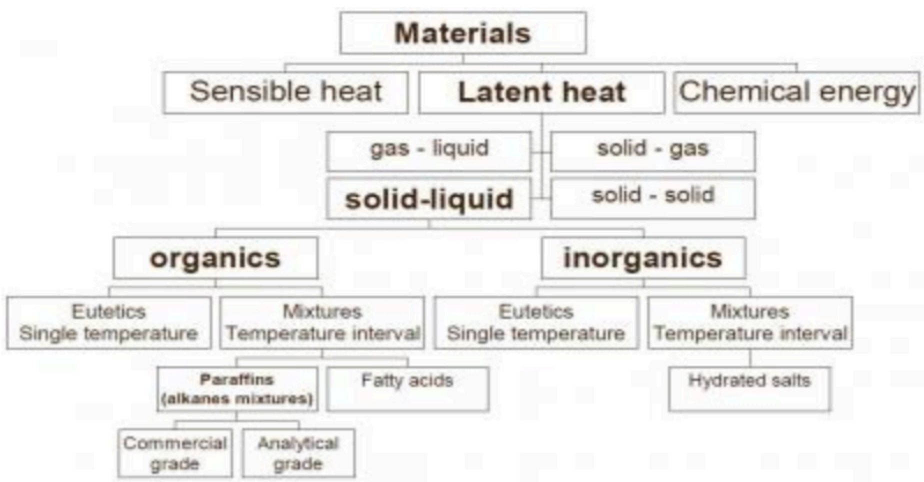


FIGURE 1. CHART OF POSSIBLE PHASE CHANGE MATERIALS (ABHAT)

As can be seen in Figure 1, there are two main types of solid-liquid transition PCMs. These categories are organic and inorganic. They are separated in this way because each type has specific advantages and disadvantages. For example, organic PCM's have chemical stability, but lower thermal conductivity and latent heat (or phase change enthalpy), while inorganic PCM's have large latent heats but can be corrosive or have phase separation. A summary of the main advantages and disadvantages for each type can be found in Figure 2.

A comparison between organic and inorganic materials for heat storage is shown in Table 1.

Table 1. Comparison of organic and inorganic PCM for heat storage. Source: IEA, 2005

Organics	Inorganics
Advantages - No corrosiveness - Low or no undercooling - Chemical and thermal stability	Advantages - Greater phase change enthalpy - Subcooling
Disadvantages - Lower phase change enthalpy - Low thermal conductivity - Inflammability	Disadvantages - Subcooling - Corrosion - Phase separation - Phase segragation, lack of thermal stability

FIGURE 2. SUMMARY OF PROS AND CONS OF EACH TYPE OF PCM (CLIMATETECHWIKI)

Pacific Design Technologies wants to evaluate existing PCMs for use in transient heat storage applications with a high heat load for short duty cycles. In our research, we were unable to find any existing commercial products that fit with PDT's application. A few industrial applications for PCMs were found with thermal profiles that are similar enough to be useful in the design process. Most of these applications were in air conditioning or solar power. For example, a common application is in passive solar temperature regulation of buildings or houses. In this application, the walls of the building contain PCMs, which melt throughout the day, absorbing some of the heat flux from the environment. These PCMs then solidify at night, releasing their heat back into the building. This has the effect of stabilizing temperature fluctuations that would normally occur in a building. PCMs are also used in solar thermal power farms. The towers that all the mirrors are pointing at have molten salt pumped through them to carry heat to the water in the turbine room. After the sun goes down, the molten salt releases latent heat as it slowly solidifies, which keeps the turbines running through the peak evening electrical demand. While none of these applications are the same as our product, they all demonstrate that for systems with dynamic cooling needs, PCMs can provide stabilization and power savings to the system.

Although all of these systems contain PCMs, they can be in thermal contact with their system in a few ways. One way that the material can be introduced is through macroencapsulation, where the reservoir for the PCM is a tank that has a high thermal conductivity. The coolant is then piped through this tank to affect heat transfer between them. The PCM could also be stored in a sleeve, which would then be wrapped around piping in the cooling loop. Alternatively, the PCM can be introduced into the system directly with microencapsulation. In this method, the PCM is contained in small capsules on the scale of tens of microns, and the coolant is in direct contact with the capsule, increasing thermal conductivity. The material changes phase within the capsules, absorbing or dissipating heat, while the capsules appear as a powdery substance. This process can increase temperature reduction rate by up to 23.5% (Mehravar). The image on the next page shows an example of PCM containment in microcapsules. The PCM is the colored substance inside of the grey pill that absorbs heat when hot and releases it once it is cool. The PCM in the capsule changes phase and transfers heat without affecting the capsule.

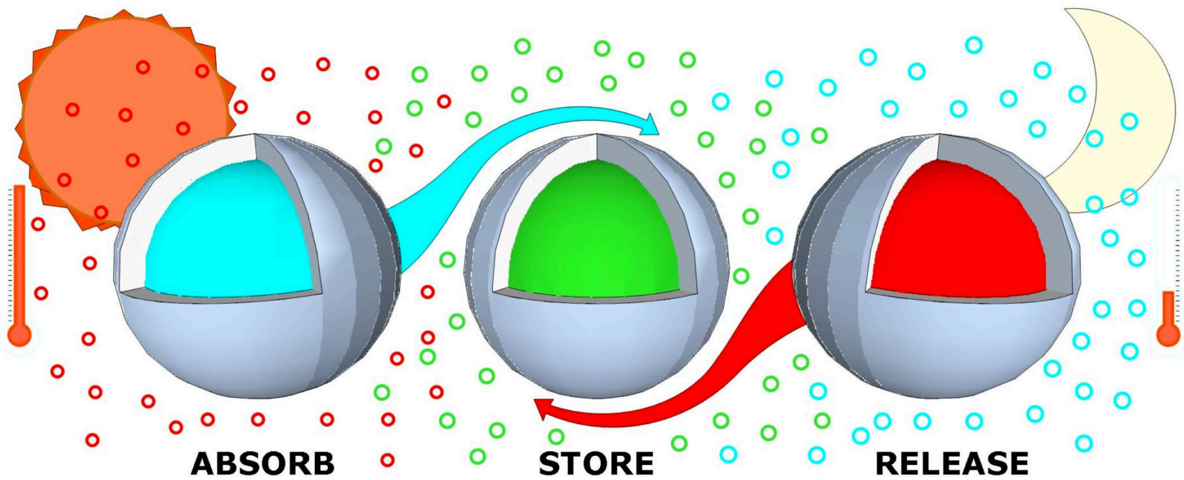


FIGURE 3. EXAMPLE OF MICROENCAPSULATED PCM CONTAINMENT (NEXUS)

Phase change is not the only way to add thermal storage, however. Water has a large sensible as well as a large latent heat capacity. In large building complexes, such as Cal Poly, huge water tanks are chilled at night but the water circulates through multiple buildings' air conditioning systems in the day. This reduces the electrical load during expensive daytime hours, and utilizes the air conditioning systems at night, when the electricity is cheaper and the refrigeration units run more efficiently.

2.1 Patent Search

There are several existing patents related to PCM's that need to be considered. The patents do not directly affect the project because it is a study of the viability of these systems, but they should be checked before this type of product is marketed. This includes patents by Airbus and Northrop Grumman relating to the absorption of dynamic heat loads using PCM's. The patent assigned to Rolls Royce includes PCM cooling for directed energy weapons. The patent numbers are included in the Table 1 for review.

TABLE 1. SUMMARY OF APPLICABLE PATENTS

Patent	Original Assignee	URL	Citation
Aircraft electronics PCM cooling system	Airbus Deutschland Gmbh	https://www.google.ch/patents/US20090180250	Holling, Mark, and Wilson Noriega. Peak-Load Cooling of Electronic Components by Phase-Change Materials. 16 July 2009.
Laser PCM cooling system	Northrop Grumman Corporation	https://www.google.com/patents/US20080000613	Harpole, George. System for Delivering Coolant to a Laser System. 3 Jan. 2008.
Vehicle Cooling with PCMs	Rolls Royce North American Technologies	https://www.google.com.gi/patents/US9534537	Gagne, Steven, Rigoberto Rodriguez, William Siegel, and John Arvin. Phase Change Material Cooling System for a Vehicle. Patent US9534537 B2. 3 Jan. 2017.

3.0 OBJECTIVES

Pacific Design Technologies (PDT) provides cooling systems for many military and aerospace contractors. These systems are designed to cool at the maximum heat load of the application. For high heat load, low duty cycle applications, this solution is not ideal. As such, PDT wishes to evaluate and prototype a transient heat storage system, which utilizes phase change materials (PCMs) in cooling systems that fit these criteria. Our final physical product will not be used in a real system, but it will give PDT confidence in a future design.

3.1 BOUNDARY DIAGRAM

The product must be integrated into a larger system, but we will restrict ourselves to designing only the portion of the system specifically to add "thermal inertia". This restriction is depicted in Figure 4, a boundary diagram. The boundary diagram shows the pieces of the project that are within our scope inside of the dashed lines. Objects outside the dashed line directly influence the project but are not up to us to design, as they are outside of the project scope.

3.2 CUSTOMER NEEDS

There are four main objectives of this project. The first goal is to determine the overall architecture of the system. This will entail the examination and analysis of existing cooling loops to determine an optimal location for the introduction of a thermal energy storage component into the loop. We will include a block diagram of the system, to represent the chosen architecture. Second will be to examine and profile existing phase change materials for use as the thermal energy storage component. This will involve creating a matrix of existing materials to determine the best materials for the application, then testing a few of these materials. The third portion of this project will be to design a storage method for our phase change material, as well as a method of heat transfer between the existing cooling loop and our thermal storage component. This process will involve analysis of the heat transfer occurring between the storage component and the cooling loop, during the peak and idle load portions of the power cycle. Finally, we will create a small (5%-10%) scale prototype using off the shelf components to replicate a typical cooling loop. Our thermal storage device will then be integrated into this loop and tested under a typical transient heat load application. This prototype will be designed to serve as a proof of concept for the technology.

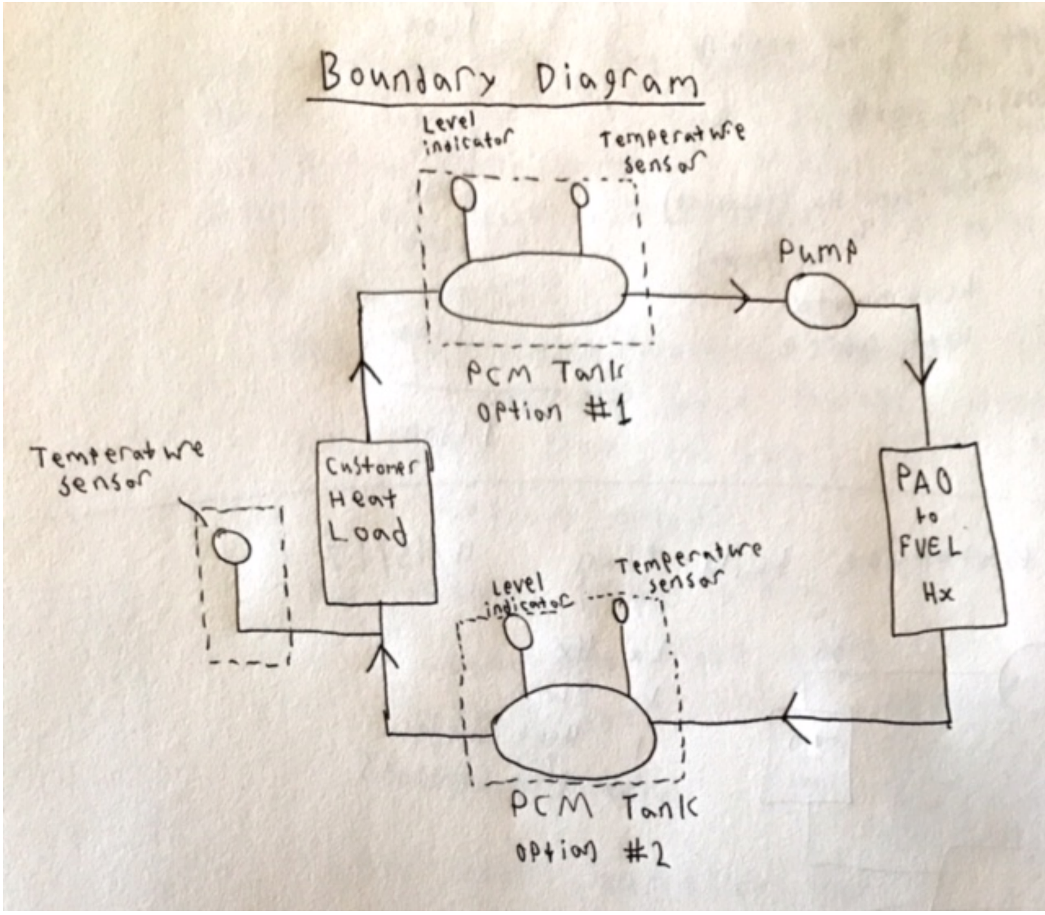


FIGURE 4. BOUNDARY DIAGRAM, SHOWING THE PHYSICAL EXTENT OF THIS DESIGN.

The coolant used for the system will be polyalphaolefin (PAO). This coolant will pass through a pump and pass through a liquid to liquid heat exchanger which uses the fuel of the plane as a heat sink. After the heat exchanger, the coolant would absorb heat from the mission equipment. The PCM reservoir needs to be integrated such that the coolant reduces to a temperature below 20-30°C before the next cycle. Whether this requires an existing reservoir or direct integration will be explored through the early phases of the project.

3.3 Quality Function Deployment

The Quality Function Deployment (QFD) House of Quality was developed based upon the engineering specifications and needs of the customer (PDT). The final system design also affects PDT's customers (Northrop, Lockheed, etc.), so their needs were also considered. The design needed to maintain the low coolant temperature delivered to the customer electronics through with the high load and low duty cycle as the highest priority. The other critical requirements for the QFD were that the device could operate in the conditions that it would be subject to as part of an aircraft. The other specifications in the QFD were less critical and these had to do with system life (beyond a few flights) and optimizing the system for power use, size, and cost. Comparing the design to existing products is difficult, because the related designs are company and/or government intellectual property. The full QFD quantifies the needs and expectations of the customers, PDT and defense contractors, and is shown in Appendix A.

3.4 Design Specifications

The design specifications shown below are for the integrated PCM reservoir design that PDT has requested. Most of these specifications can only be verified through analysis, and since we are not making a full scaled system prototype, all the test specifications will be completed with the subscale prototype that we will develop. Table 2 lists the specifications for the system that they require. In the table, H, M, and L stand for High, Medium, and Low levels of risk. The risks indicate the relative importance of meeting each design requirement. For the compliance column, A, T, and I stand for Analysis, Testing, and Inspection respectively. These indicate the way that we show the design meets the overall specifications.

TABLE 2. FULL DESIGN SPECIFICATIONS

Spec #	Engineering Specification	Dimensions	Risk	Compliance
1	Duty Cycle	20% Duty Cycle (30 seconds High Load)	H	A, T*
2	High Thermal Load (Low Load)	50kW (3kW) of heat added to the system	H	A, T*
3	Coolant delivery temperature to customer equipment	20-30 (C)	H	A, T*
4	Ambient air temperature operation range	-84 F to 131 F	H	A
5	Pressure drop across PCM equipment	25 psid max (at 9.5 gpm and 20 C)	M	A,T*
6	Altitude Operation Range	Sea level to 65,000 ft	H	A
7	Mean Time Between Failure (MTBF)	At least 5,000 hours	M	A, I
8	Safety (Flammability, Corrosivity, Toxicity)	Not Corrosive, flammable, or toxic	H	I
9	Power Requirement	Minimize power use	L	I
10	Weight/Volume	Minimize	L	I
11	Cost	Minimize	L	I

* Tested on sub-scale model

Most of the specifications for the design are critical to the functionality of the system. The system must function at the specified thermal profile otherwise the system will overheat and fail within several cycles. The coolant delivery to the system is also critical so that the customer's electronics will function properly. The device will also have to operate at high altitudes and a wide temperature range because the system cools airplane electronics. Failure to design a viable system for the supplied conditions would render the device useless to PDT's main customers. Finally, the material must be safe. This means that it is cannot be flammable, corrosive, or toxic. All of the aforementioned specifications are critical to the design functionality so they are listed as high risk.

The pressure drop across the PCM equipment determines how much more head the pump has to supply for the system. A larger pressure drop than given is not a critical failure, but it could require a new pump which would require coordination with PDT beyond the original scope. The Mean Time Between Failure (MTBF) of 5,000 hours is also non-critical for the design. A lower than expected MTBF would require the PCM to be switched out more often during ground operation. As long as the MTBF is beyond the range of a few flights this specification is important, but not critical. Limiting the pressure drop and MTBF are considered medium risk because they do not cause the system to fail, just a redesign of the existing loop and more frequent maintenance.

Some other specifications less critical to the functionality of the design are that the size, weight, cost, and power usage of our final design should all be minimized. Although these criteria are low risk, they are still important to the final design, and aided us during the concept development process.

PDT has specified additional requirements for the full design. The PCM reservoir must have equipment to monitor the PCM's temperature and fill level. The device may not become hot enough to cause ignition in an explosive environment. The reservoir must have enough space to account for the thermal expansion of the system and will not vent to the surrounding compartment during operation.

The subscale model for the system needs to operate on a 5-10% heat load of the system load in order to demonstrate the feasibility of the concept. The model is not expected to be operated or tested in the environmental conditions shown.

4.0 CONCEPT DEVELOPMENT

The goal of this project is to test multiple sets of PCMs and system architectures with a recommendation for the final design and a prototype demonstrating the concept. This made the goal of concept development to generate multiple viable systems for testing in the winter of 2018. The three variations between systems can be the PCM used, the type of reservoir for the PCM (if applicable), and the location of the reservoir (if applicable).

4.1 IDEATION

Design concepts were generated through several ideation sessions aimed at finding solutions to each of the system functions. These solutions were then evaluated and iterated upon using Pugh matrices for each of the functions. The feasible solutions extracted from this process were expressed through a set of foam core models as a demonstration, and to get a sense of what each concept would entail. The solutions ranged in complexity from a conventional, tube and shell heat exchanger between the PCM and coolant to a set of solid-solid PCM fins directly inside of the coolant loop. The full solutions list can be found in Appendix B.1.

4.2 IDEA EVALUATION

To determine the best solutions to pursue, each solution was evaluated against one another using a weighted decision matrix. The criteria used in this matrix were mostly the design specifications, with a few more overall considerations added. Several concepts were ruled out due to foreseen problems in their designs. For instance, introducing microencapsulated PCMs directly into the cooling loop, or using solid-solid PCMs within the coolant, has the potential to cause fouling in the pump. This, as well as the high cost of these types of PCMs, caused us to rule out these possibilities. The full decision matrix can also be found in Appendix B.2. The second way in which we evaluated our solution choices was by modeling the system. First the original system was modeled to quantify the heat load dissipation required by the design solution. The PCM reservoirs were represented as isothermal heat exchangers (assuming heat transfer is only occurring between phases) placed just before and after the mission equipment. The addition of the PCM reservoir, optimally designed, can drastically reduce the temperature variation of coolant going to the mission equipment, and there was significant difference between reservoir designs by placing the reservoir before or after the mission equipment, for the same system response.

The two concepts that we would like to test moving forward both involve a PCM reservoir that resembles a heat exchanger in the coolant loop. The reservoir can be located before or after the mission equipment. The first reservoir will contain a pure PCM material that will change phase with fluctuations in temperature in the system. Several specific solid-liquid

PCMs will be tested in this design. The second reservoir will use a slurry to cool the PAO. The slurry will consist of microencapsulated PCM contained in a solution (e.g. water-glycol) that will change phase within the capsule but not affect the overall state of the slurry. Both concepts will be tested to compare their effectiveness and ability to meet the design specifications.

4.3 Analysis

Wanting to find the difference between "architectures" meant first finding a baseline of what happens if we run the system without modification. Then the damping effects of our addition could be added in various places to find the relative benefits. The two main locations would be before the extant heat exchanger or after; the heat rise from the pump and viscous heating throughout the piping is assumed to be negligible for these analyses.

4.3.1 Building An Equation

For simplification, we assume the PAO in circulation to have no mixing in the direction of flow. Rather, the heated fluid follows and leads cold fluid with a distinct discontinuous boundary, as if a pipeline sabot was keeping the two separate. This helps the model's output mimic the on/off mode of the heat input, and eases calculations. The result will be very blocky temperature profiles; the reality will be smoother, with rounded edges, but the same maximum and minimum temperatures should be met.

From the drawings of the extant system, it appears the accumulator tank does not have all of the flow circulate through it, but is a side path for excess and deficit PAO. This helps validate our no mixing assumption, and we should recognize this volume is separate from the volume of PAO in circulation.

These assumptions let us model the temperature flowing through the system as a periodic function, with all the convenient familiarities such as a time period that defines how long it takes for fluid to circulate through the entire loop once. It is a small jump from here to say that the temperature reaching the mission equipment depends on the temperature that same fluid was at the last time around. So, in part,

$$T(t) = T(t - \tau) \tag{4.3.1}$$

where T is the temperature of the fluid coming into the mission equipment, t is the independent variable time, and τ is the "time constant" representing how long the fluid takes to pass through the system once.

Of course, eq. 4.3.1 represents a trivial system with no heat transfer. The temperature will rise by a function of power input, specific heat of PAO, and flowrate of PAO. For convenience, we'll use the volumetric heat capacity from Appendix M: PaoPropertiesSheet because the quantity is more stable with temperature than mass based heat capacity, and is aesthetically better to work with because the number is greater than one but not large. It is a simple conjecture that the temperature rises by the power input per amount of fluid and divided by the specific heat

$$T(t) = T(t-\tau) + \frac{P(t-\tau)}{\dot{V} * C_V} \tag{4.3.2}$$

where P is the power input, C_V is the volumetric heat capacity, and $\dot{V}$ is the volumetric flowrate. Both C_V and $\dot{V}$ vary with temperature, but not much, so a value corresponding to an average temperature is acceptable.

To model the effect of the PAO-to-fuel heat exchanger, it is a simple matter of using the hot-side effectiveness. If the flowrate of either the PAO or fuel changed, this would not be valid, but we are given testing data for these flowrates and are assuming nothing changes here. Different heat exchangers may be used assuming that the hot-side effectiveness is known.

To make the effectiveness portion of the equation more concise, we use a new temperature scale, θ, such that the base incoming fuel temperature is zero degrees.

$$\theta(t) = T(t) - T(cold\ fuel) \tag{4.3.3}$$

Factoring in our PAO-to-fuel heat exchanger effectiveness, E,

$$\theta(t) = \left[\theta(t-\tau) + \frac{P(t-\tau)}{\dot{V} * C_V}\right](1-E) \tag{4.3.4}$$

where we note that $\dot{V}$ is 9.5 gpm, T(*cold fuel*) is 50 °F, E is .66, P cycles between 50 kW and 3 kW, C_V ranges from 3.41 to 3.55 Btu/gal/°F, and θ + 50 °F is the temperature in Fahrenheit of PAO reaching the mission equipment.

Now modeling a second heat exchanger can be fairly straightforward. It will not be as pretty, because the base temperature term will not drop out unless it has the same value as the incoming fuel temperature. We can either model an existing HX or, since we do not have one yet, how one will have to perform in order to get the results we require. If we designate E_{PCM} to be the effectiveness of the second heat exchanger to bring the coolant temp to T_{PCM}, then

$$T_{out} = (T_{in} - T_{PCM})(1 - E_{PCM}) + T_{PCM} \tag{4.3.5}$$

determines the temperature of PAO leaving our additional heat exchanger. This is not exact, because the phase change process for many PCM's is not a strictly isothermal process. In such cases a couple degrees difference will exist in the direction of lower heat transfer. Rearranging,

$$T_{out} = T_{in}(1 - E_{PCM}) + E_{PCM}T_{PCM} \tag{4.3.6}$$

gives an expression easy to incorporate into equation 4.3.4. If the PCM container is put after the PAO-to-fuel HX in the system loop, then eq. 4.3.4 represents the incoming temperature in 4.3.6, so

$$\theta(t) = \left[\theta(t-\tau) + \frac{P(t-\tau)}{\dot{V} * C_V}\right](1-E)(1-E_{PCM}) + E_{PCM}T_{PCM} \tag{4.3.7}$$

is our equation to determine the incoming PAO temperature if the PCM addition is *after* the HX.

To model the PCM addition being put in place *ahead* of the HX, the bracketed portion of eq. 4.3.4 is the incoming temperature in 4.3.6. This result is now the incoming temperature to the HX, so the result is multiplied by 1 – E, so

$$\theta(t) = \left[\left[\theta(t-\tau) + \frac{P(t-\tau)}{\dot{V} * C_V}\right](1-E_{PCM}) + E_{PCM}T_{PCM}\right](1-E) \tag{4.3.8}$$

is our equation to determine the incoming PAO temperature if the PCM addition is *before* the HX.

4.3.2 Optional Refinements

We note that some mixing of fluid up and down the pipes can be simulated by taking an average value of temperature around θ(t-τ) term, or better, a weighted average such as

$$\theta(t-\tau) \approx \frac{1}{13}[\theta(t-1.2\tau) + 3\theta(t-1.1\tau) + 5\theta(t-\tau) + 3\theta(t-.9\tau) + \theta(t-.8\tau)] \tag{4.3.9}$$

where explicit numbers (such as 1/13 and 1.2) are to be determined by guessing and checking against experimental data. However, for this problem, almost the entire volume of coolant passes through any given heat exchanger during one high power portion of the cycle, so mixing the fluid will not alter the highest and lowest temperatures reached, it would only smooth out the sharp edges.

The function for power absorbed by PAO fluid is taken to mimic the load function for the mission equipment, which is a high and low load with a specified duty cycle, and constant pumping power. The easiest way to model transient loads is with the unit, or Heaviside, step function, u(t). When the term inside the brackets of u(t) is less than zero, the switch is off; zero or larger flips the switch on:

$$u(t) = \begin{Bmatrix} 0 \; if \; t < 0 \\ 1 \; if \; t \geq 0 \end{Bmatrix} \tag{4.3.10}$$

$$\left(u(t) - u(t-a)\right) = \begin{Bmatrix} 0\ if\ t < 0 \\ 1\ if\ 0 < t < a \\ 0\ if\ t\ \geq a \end{Bmatrix} \tag{4.3.11}$$

It is a simple but cumbersome task to assemble a large summation of step functions to activate and deactivate the high and low power modes at any time. Such work in MATLAB® appears in Appendix C, and has the advantage of being able to manipulate individual power cycles. But if one wants to model more than a few cycles it is more elegant to take the step function of a complex polar variable (e^{ix}).

```
(heaviside(k-Tao)                  -heaviside(k-Tao-TimeON))              *PowerH
(heaviside(k-Tao-TimeON)           -heaviside(k-Tao-TimeON-TimeOFF))      *PowerL
(heaviside(k-Tao-TimeON-TimeOFF)   -heaviside(k-Tao-2*TimeON-TimeOFF))    *PowerH
(heaviside(k-Tao-2*TimeON-TimeOFF)   -heaviside(k-Tao-2*TimeON-2*TimeOFF))*PowerL
(heaviside(k-Tao-2*TimeON-2*TimeOFF)-heaviside(k-Tao-3*TimeON-2*TimeOFF))*PowerH
(heaviside(k-Tao-3*TimeON-2*TimeOFF)-heaviside(k-Tao-3*TimeON-3*TimeOFF))*PowerL
(heaviside(k-Tao-3*TimeON-3*TimeOFF)-heaviside(k-Tao-4*TimeON-3*TimeOFF))*PowerH
(heaviside(k-Tao-4*TimeON-3*TimeOFF)-heaviside(k-Tao-4*TimeON-4*TimeOFF))*PowerL
```

FIGURE 5. MODEL SOLVING PROCESS

Using e^{it} makes it difficult to make particular changes in the cycle behavior during the same simulation. But the cycle can run continuously, so visualizing PCM degradation will be relatively simple.

4.3.3 Extracting Additional Information

If one wanted to find the temperature of the PAO after it runs through the mission equipment, or just before any heat exchanger, negating a portion of the established equation is required. For example, the temperature after the mission equipment is the same as what is entering the HX, so for that case just take eq. 4.3.4 and divide by (1-E) if you already have run that equation, but if you delete (1-E) outright the next cycle around will not have cooled down from last time.

Using the information of both the PAO temperature before our PCM addition and the properties E_{PCM} and T_{PCM} we can verify the design parameters of our heat storage tank. Note that there has been no check on total heat absorbed by the PCM. This amount of heat can be determined by an integration of the difference the PCM has on the system. This can be quickly and easily approximated from the rectangular areas presented on the Figures 6-10. The two main things that have yet to be analyzed are, that there is enough PCM to absorb the influx of heat, and that there is at least that same amount of heat flux rejected during the cooldown period. If there is more heat flux rejected than absorbed, then the real operating temperature is slightly less and is inconsequential. If there is less heat rejected, then the PCM has not fully cooled before the new cycle, and a different PCM may be necessary.

To choose a PCM, you first need an initial guess at the phase change temperature you required. To make this initial guess, first chose a temperature between the highest and lowest temperatures that will be in thermal contact with the material. Further, you want this temperature to be a weighted average between those temperatures; for a 20% duty cycle, you want the high temperature to influence only 20% of the chosen temperature. Therefore,

$$T_{pcm} = \frac{T_{hot} + 4T_{cold}}{5} \tag{4.3.12}$$

is a sufficient guess. In reality, the contending solid-liquid PCM's have a small range of temperature where it changes phase, because they are mixtures of different molecules. A way to account for this is to put a penalty, say, two degrees, closer to whichever temperature is in contact. Now when some integration is done on the figures to make sure the same amount of heat is absorbed as is dissipated, the longer cooldown time will be more impacted, and a higher temperature PCM will be selected. This agrees with the previous concern, which concluded that is to put a hotter PCM in is to err on the side of safety.

Another option, although it may be outside of the project scope, would be to use a liquid-vapor PCM mixture, which can use multiple gases to utilize latent heats of fusion at a variety of temperatures. All coolant temperatures and results depend upon the cold fuel side, and we are unsure if this value remains below 50°F throughout the cycle.

More information on theory and analysis can be read about in Appendix L. Further Investigations in Theory.

4.4 Model Results

To start, let us examine Figure 6, which shows how the LCSU would respond if installed without purposeful heat storage. For clarity, the system is set up with enough PAO coolant that no heated coolant returns to the system while the high power mode is active. There is a small gap in time after the thirty second mark, when the equipment switches to low power mode, where cold PAO enters and only slightly warmer coolant exits. Then heated PAO enters, but colder since it went through the PAO-to-Fuel heat exchanger first, and slightly warmer PAO exits. This repeats until the high power mode comes on again. For this illustration, we assume 5.9 gallons of PAO in circulation.

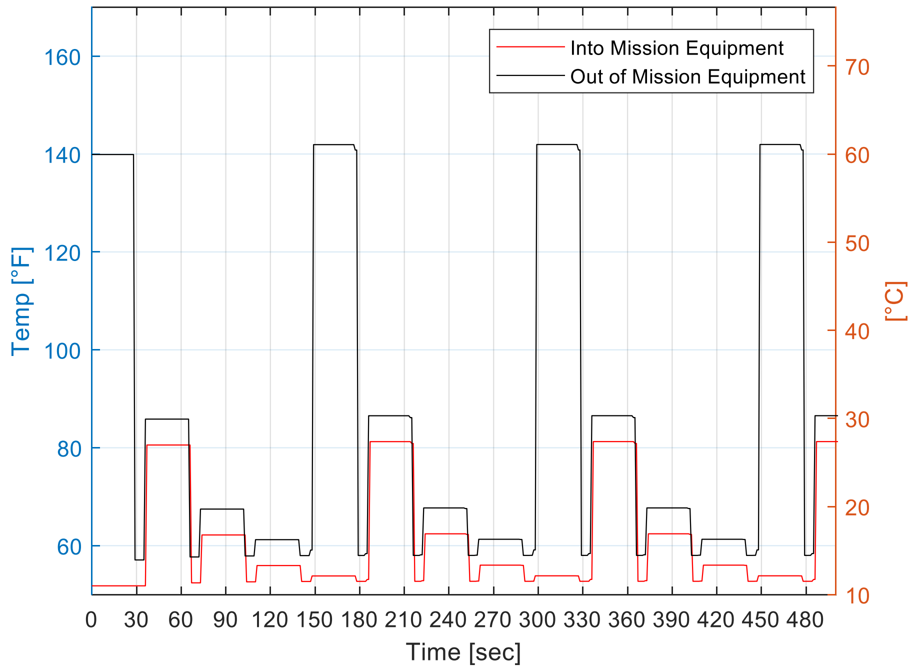


FIGURE 6. TEMPERATURES AT MISSION EQUIPMENT WITHOUT HEAT STORAGE

A major takeaway from this figure is that the incoming PAO temperature is already within the design specifications of 30 °C. Because of this, we have a few options moving forward. One option would be to design for a larger max heat load, such as 75 kW. Another option would be to design for a smaller, less powerful cooling loop. The third option would be to set the target maximum coolant temperature at the low end of the specification, at 20-24 °C.

Some of the biggest factors in the effectiveness of the system are coolant volume and flowrate. For example, the system preforms much worse when the coolant volume and flowrate combine to circulate more than once during one high power cycle, as can be seen on Figure 7. For this illustration, there is 4.5 gallons of PAO in circulation. There is a significant spike in temperature at the mission equipment, enough to fail to meet the customer's specified max incoming temperature.

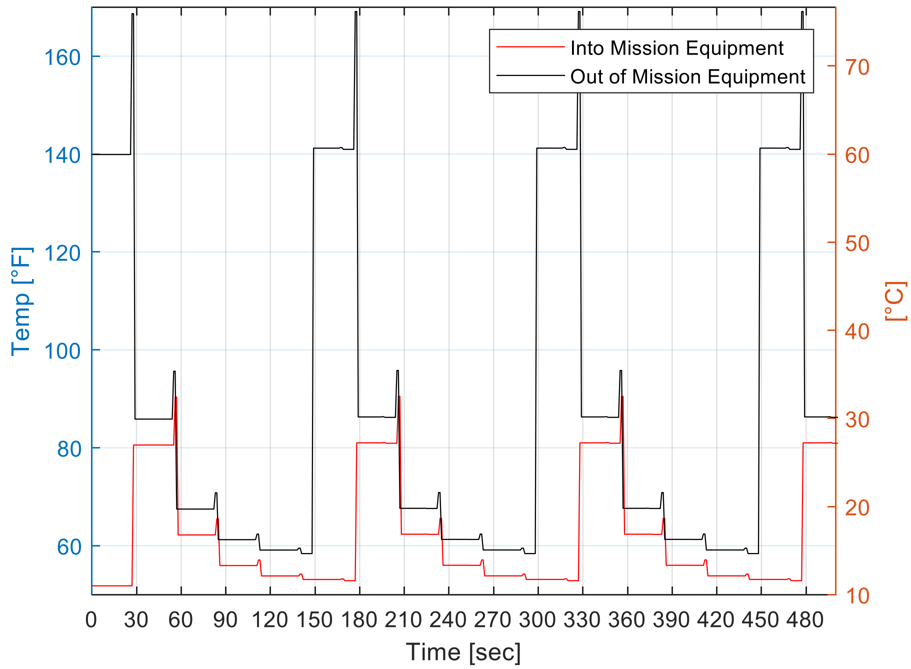
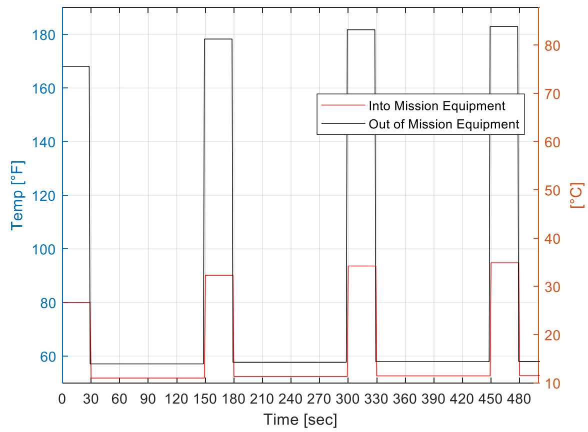


FIGURE 7. PAO TEMPERATURES WITH SLIGHT ADJUSTMENT IN VOLUME, NO PCM

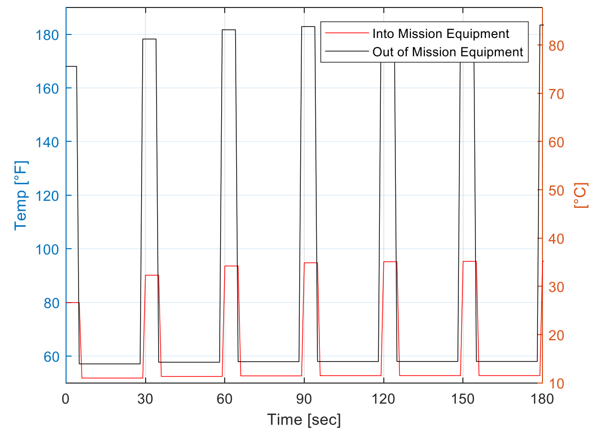
One way to fix this issue is to add more coolant to the system. Note that accumulator volume does not contribute to coolant volume if it is a tangent line to circulation. Also, longer coolant lines may contribute to a harmonic heat input, where the lengths of time for coolant to circulate and total duty cycle time are the same (or some other natural harmonic contribution, such as one half, a third, or twice, three times, etc). This problem is illustrated in Figure 8, which demonstrates two ways the problem could be created. The second way is for the customer to read 20% duty cycle rating to mean any cycling on and off times up to a maximum of 30 seconds on. For instance, they may turn the equipment on for six seconds and off for twenty-four, still 20% duty, but creates harmonic input.

Because of this, our heat storage tank must be able to "dampen out" these harmonic inputs as well as it smooths out the input temperature in the generic case.

Since we know that the outgoing temperature depends on the incoming temperature, we will not continue to graph the outlet temperature, letting us focus on the details of the incoming temperature, where the heat storage tank has an effect. For the following graphs, the situation with 4.5 gallons of circulating PAO is used to include the effect on the temperature spike.



(a) Volume Increases to Harmonic Input



(b) Power On Time Decreases to Harmonic Input

FIGURE 8. HARMONIC HEAT INPUTS

Where the heat storage tank *is* in relation to the customer heat load can now graphically show its effect. Assuming either position has a heat exchanger effectiveness of .6 with its isothermal phase change temperature, the temperature profiles of heat storage tanks placed before and after the extant heat exchanger are shown in Figure 9 and Figure 10, respectively.

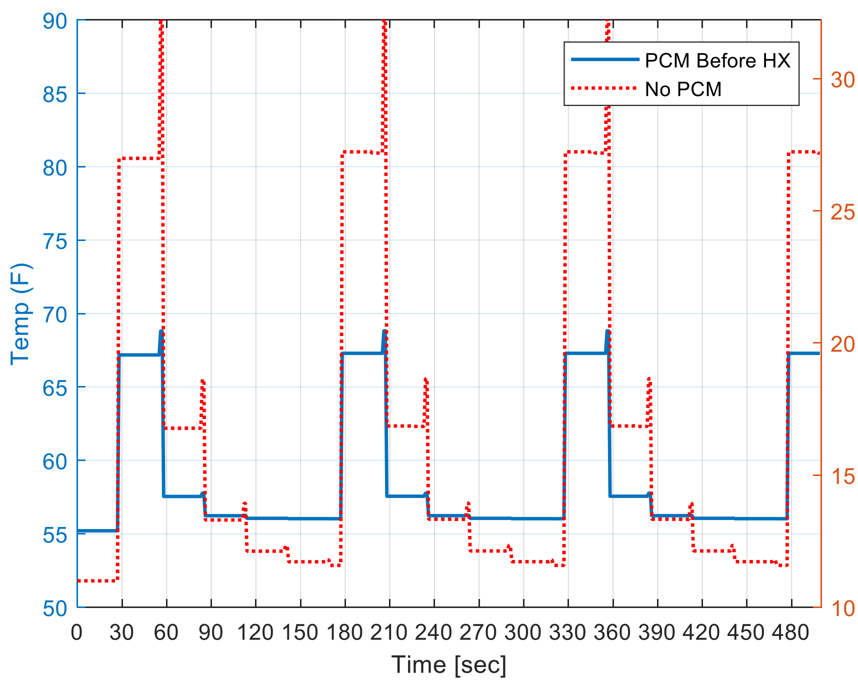


FIGURE 9. INCOMING PAO TEMPERATURES WITH HEAT STORAGE BEFORE HEAT EXCHANGER

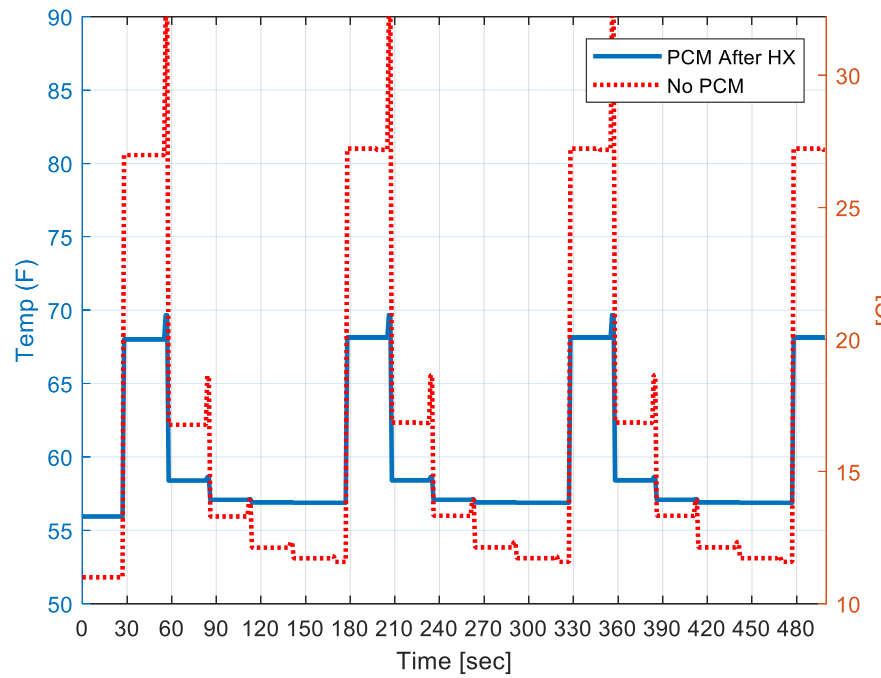


FIGURE 10. INCOMING PAO TEMPERATURES WITH HEAT STORAGE AFTER HEAT EXCHANGER

Subtle differences exist between the two options, for the same effective tank. Putting the tank before the PAO-to-Fuel heat exchanger lowers the resulting temperature to below 20°C, whereas putting the tank after lowers the temperature to 20°C. The spike still protrudes above the line, but this is likely still an acceptable temperature. The second method has a constant borderline period before the spike infringes, but still remains at an acceptable temperature.

What is not evident from these graphs is that the total heat load absorbed by the first option is about 700 Btu, which is only half the dynamic load. This is in stark contrast to preliminary guesses that the heat load would be nearly all of the dynamic load. Even more surprising is that the second option reaches acceptable temperatures by only absorbing about 200 Btu. This is where the second option shines.

In short, placing the reservoir before the electronics regulates the coolant temperature with a lower heat exchange rate, while placing the reservoir after the electronics regulates this temperature using less PCM.

Do not be fooled by the area between the red dotted line and the blue line. That is not the area of integration that determines the total heat absorbed and released by the PCM, that graph is still to come, but it is not yet pretty. Using that graph did tell us that in both cases the phase change temperature had to rise and the effectiveness must go up to dampen the load satisfactorily, because we assumed the wrong temperatures. Score one for the graph.

4.5 Visual Model

Part of the Ideation process entailed creating simple models of possible designs for the project. These initial designs are shown in Figure 11. From left to right they are: the PCM sleeve, a tube and shell heat exchanger, solid-solid PCM fins, and a more in-depth model of another tube and shell heat exchanger. Once our final design, the tube and shell heat exchanger, was chosen we created a CAD model for it in Solidworks. This model can be seen in Appendix D.

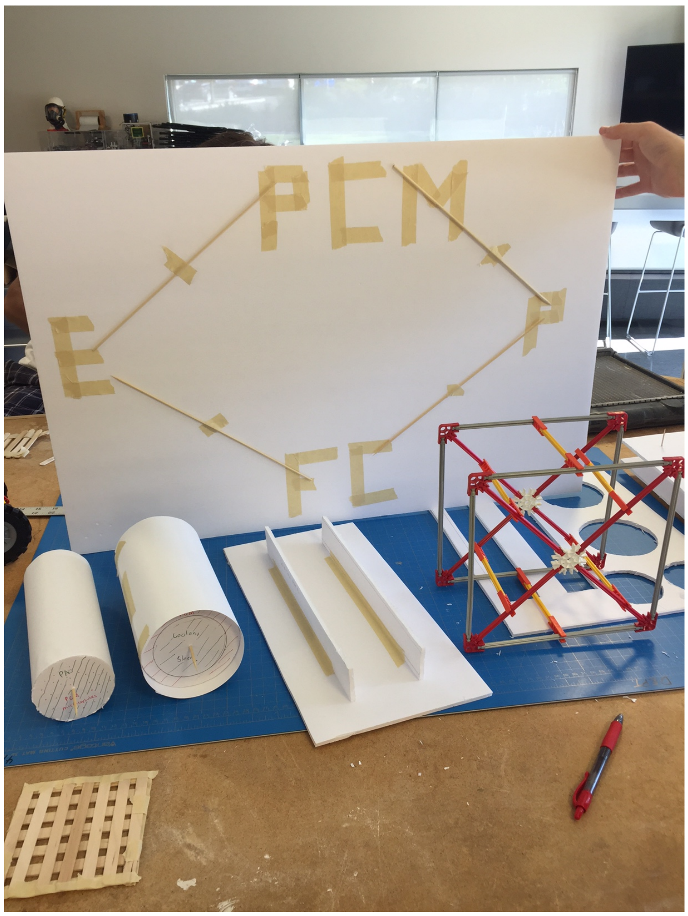


FIGURE 11. INITIAL MODELS OF POSSIBLE DESIGNS

4.6 Design Safety

In terms of risks to the health and safety the users of this product, the most significant is the internal pressure in the system. This risk can be mitigated by installing pressure gauges to the system and using them to regulate the pressure throughout the system to safe levels, as well as designing the system to handle these loads. In terms of risks to the functionality of the product, the biggest concern is not that we will lack capacity for thermal energy storage, but rather that the rate at which we can store thermal energy will be too low. This is because the capacity for energy storage is based mostly on the mass of PCM, while the heat transfer rate is based more on the type of heat exchanger or heat exchanging method than the properties of the specific PCM. To this end, multiple concepts will be tested. A full listing of additional design hazards can be found in Appendix I: Safety Hazard Checklist.

5.0 FINAL TESTING DESIGN

The purpose of this project was to test out a design with a variety of PCMs to determine where the PCM device should be located as well as the type and the operating temperature of the PCM. The major decisions made at this point were to determine the best apparatus for heat transfer between the PCM and the coolant and the design of the test apparatus for the system.

As discussed in section 4.2, we used a decision matrix in Appendix B-2 to compare our PCM integration solutions with one another. We determined that using a modified tube and shell heat exchanger would be the best option because it had no glaring flaws in any major criteria and excellent performance in limiting potential fouling of the system, PCM life, and known thermal conductivity (from supplier data). These factors would make the system more effective overall and meet the goal for device longevity. The testing was not subject to construction delays and craftsmanship shortcomings due to machining the parts ourselves as the majority of the parts were purchased.

The testing apparatus is a heat exchanger that we ordered from American Industrial Heat Transfer Incorporated (AIHTI), with the coolant (water) flowing through the tubes of the heat exchanger. The shell side contained PCM and was capped off at both ports. One of the ends included a threaded sight to determine the state and level of the PCM (to check for effectiveness and leaks respectively). The other port was connected to a thermocouple that was sealed inside of the shell side to determine the PCM temperature. A concept layout of the completed device is available in Appendix D.

To construct the overall system loop, we used existing lab equipment inside of the Cal Poly Fluids lab. The apparatus contained a pump, reservoir for coolant, pressure sensors, a totalizer, PVC piping and valves, and an electric control box for control of two pumps and a heater. We added a few more elements to this system. First, we added the PCM Reservoir to the loop along with an in-line water heater. Next, we installed a brazed plate heat exchanger to interface with the secondary coolant loop. We added a flow meter and several thermocouples throughout the system to measure the overall system performance. We included an additional low flow rate pump and alternate heat exchanger to provide an alternative PCM reservoir architecture for water and microcapsule mixtures. A full outline of the system can be found in Appendix E with the two loops shown in a Solidworks assembly.

5.1 Cost Analysis

Most of this project involved ordering parts and materials and assembling them into our testing apparatus. The components that we were able to get with no cost were either borrowed from Cal Poly or PDT. The components that we had to personally order or PDT had to order are counted towards the overall cost of the system.

The biggest costs of this project came from the AIHTI heat exchanger, the PCMs, and the thermocouple reader. Each of these components cost several hundred dollars, and drove the budget for the system. The exact cost of all components and the status of each of them can be found in our bill of materials in Appendix E, along with data sheets and vendor/budget information in Appendices F and G respectively.

5.2 Safety Considerations

The apparatus had several safety considerations and requirements for operation from Cal Poly. The main safety concerns were the requirement for an emergency shutoff for the system, pressure increases in the PCM heat exchanger, and the high voltage required to operate the apparatus. The apparatus contains a control box for all the electronic devices, which can serve as an emergency shutoff for each of them. The PCM HX also has a pressure relief valve for early stages of testing, while its necessity is determined. The electrical setup had to be signed off on by the campus electrician along with guidelines describing how to operate the apparatus safely. The complete list of all safety concerns and how they were addressed can be found in Appendix I.

Using the system in the Cal Poly fluids lab created additional requirements for the operation of the test apparatus. All members of the senior project team needed to acquire a gold card to work in the lab, and get a team key to the fluids lab itself. While we were allowed to work during regular class times, it was recommended to work during open periods and the weekends to not disrupt lectures. The open periods were Friday mornings and weekday nights. The school also required at least two people to be present whenever the system was in use.

5.3 Future Design Ideas

Based on the results of our testing seen in section 7, the current design of the system was not ideal for testing or as a design for the full-scale system.

The testing apparatus needs to be able to output the expected heater input of 5-6 kW to create a large enough temperature differential and the secondary reservoir must be held at a constant temperature.

For future prototypes of this system, one option to consider would be an outer shell made of aluminum and tube bundles ordered directly from the vendor. An aluminum pipe could be extruded with axial fins (also called a fluted pipe) on both inside and outside surfaces to get the large heat transfer surface area required, and the thickness of the PCM layer reduced. Adding radial fins to an existing pipe is another common method, and probably cheaper to make for smaller production volumes. A single pipe could hold the high PAO pressures with much less weight than the 76 tubes in the tube and shell. The inherent risk associated with this method is that the expansion and contraction of the PCM could cause cracks in the tube.

This construction could make a smaller heat exchanger with less wasted PCM, which could be set inline in a straight portion of existing pipe. Several smaller units could be placed in series or parallel, depending on packaging constraints in the aircraft. Overall, we estimate this design will be lighter and cheaper than the test setup. It may be light and cheap enough that the best solution to phase change expansion is to oversize the product and not fill the tube fully with PCM to allow for expansion.

6.0 MANUFACTURING PLAN

The main phases to building the design were: procurement, manufacturing, and assembly. This project was mostly procuring and assembling off the shelf components. The manufacturing phase consisted of slight modifications to PVC piping and the creation of stands for each of the components to the system. The PVC was cut to the necessary sizing and then set with PVC cement and allowed 1 day to dry before operation. The stands for the system were formed from wooden blocks set to allow the heater, both heat exchangers, and the secondary pump to rest on. The thermocouples were installed by having capped T connections in the PVC pipe that were drilled and sealed with hot glue to keep the thermocouple junction inside the loop. The next two sections will discuss the more intensive portions of the manufacturing plan in detail.

6.1 PROCUREMENT

Procurement was split into two sections: required components for the beaker tests and required components for the test apparatus.

The beaker tests needed each of the PCMs for testing: Crodatherm 29, MicroTek 24, and PureTemp 23 and 37. We also needed a thermocouple reader (which we acquired through PDT), a hot plate, scale, and a set of beakers from the chemistry department.

The system prototype had a few more necessary components to complete the apparatus inside of the fluids lab. The parts of the previous apparatus that we were able use are: the primary loop pump, water reservoir, totalizer, two pressure gauges, several flow valves, some PVC piping, and an electric control box. We ordered the tube and shell heat exchanger with corresponding PVC piping to incorporate it into the system. PDT supplied an in-line water heater rated for 6 kW, and an additional flow meter for the primary loop. The thermocouple reader that was used for the beaker test was also used for the testing apparatus.

The team ordered a few additional components to complete the primary loop of the system. This included two 1 gpm, 12 V pumps, a second brazed plate heat exchanger, and wiring for the electrical system. We also personally supplied a 12 Volt car battery to power both of the small pumps used in the primary loop. For a complete list of purchased parts and pricing please see Appendix G.

6.2 THE ASSEMBLY

The stands for each part were not individually made, instead, a half sheet of OSB was used to cover the bottom of the water tower, serving as a stand for all parts. Scrap pieces of plywood leftover from the previous user of the water tower were used to shim the heater up to the same height level as the shell and tube heat exchanger. Due to the low pressure of the system a sight-glass was not required on the PCM heat exchanger. We were able to simply remove one cap and look into the shell. Either the wax was totally melted and we could check with a stick or it wasn't. The thermowells were also not required, we simply drilled a small hole in a threaded PVC cap and hot-glued both ends. This resulted in sufficiently little leaking and necessarily low latency between water temperature and thermocouple reading temperature.

The second pump ordered was a RV water pump, pushing 1 gpm and hooked up to a 12V battery through a fused line and a light switch. The secondary heat exchanger ordered was a brazed plate marketed towards beer-wort chilling.

The first day of assembly is pictured in Figure 12, minus the big water tank which is above the range of view. The big pump is seen askew from the rectilinear edges of the platform, this should help differentiate primary and ultimate heat sink loops.

FIGURE 12 EARLY STAGE OF ASSEMBLY

The assembly is seen closer in with the PCM tank clearly to the left side, and the heater is center low, with a stripe of blue tape on its face. The heater wiring, for which we hoped to have access to 240V, was connected to 208V because that is what is available in the fluids lab. The resistive heater was, therefore, only able to put out 75% of the rated 6 kW power draw. The school electrician, Ben Johnson, had to hook up the wall outlet with a plug type rated for 30 amps 3 phase, with the pump requiring three phase power.

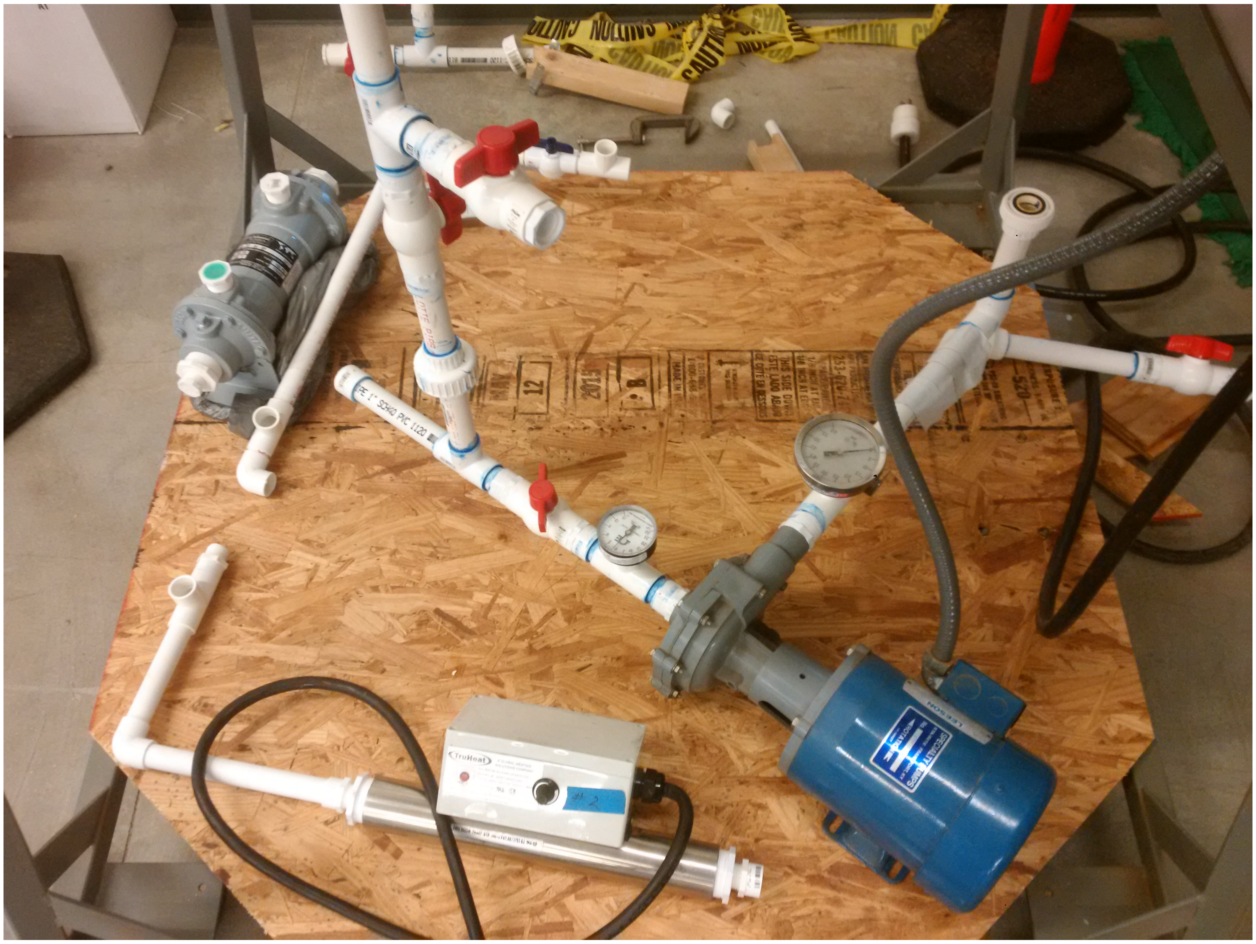


FIGURE 13 THE THREE BIG DEVICES, WHICH WERE THE MAIN HINDERANCE TO STARTING ASSEMBLY

The primary circuit connected the PCM reservoir using two unions, so that the tank could be removed from the line with ease for draining and replacing with alternate heat storage devices. The only other device tested was a brazed plate heat exchanger with microcapsules flowing through. The large union nuts can be seen in Figure 14 and the concept of flowing microcapsules in Figure 15, though this picture shows them being pumped through the shell & tube. Having a flowing PCM required a second 12 volt pump, and the associated fittings and tubing.

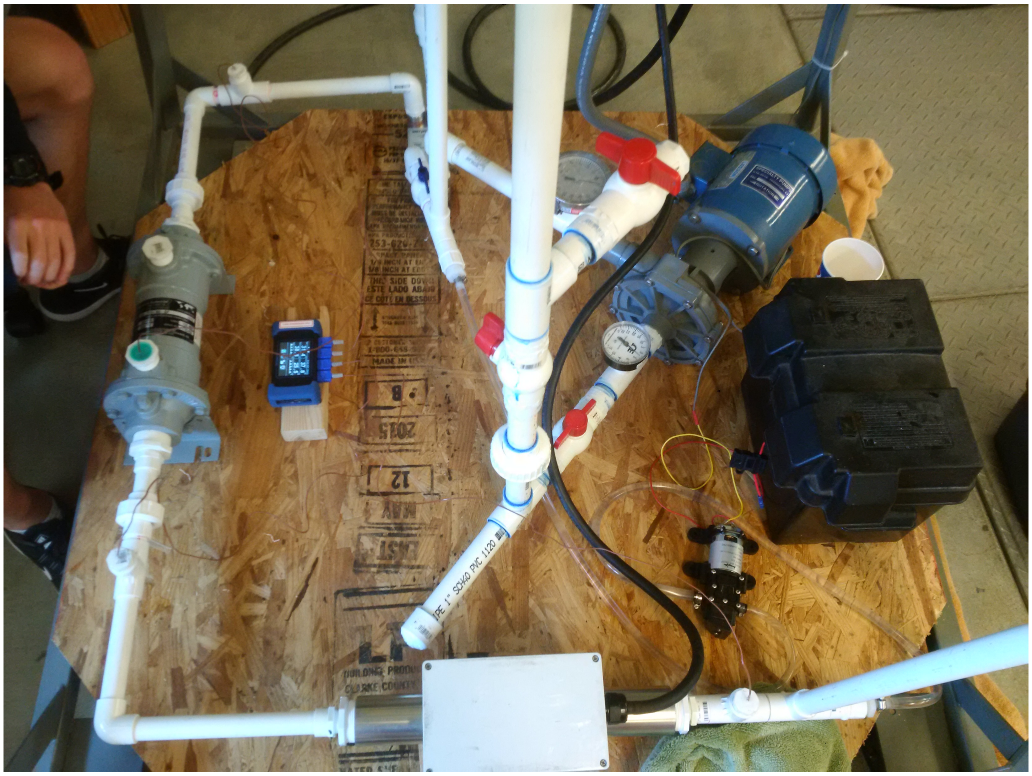


FIGURE 14 COMPLETE ASSEMBLY

FIGURE 15 TEST ASSEMBLY SET UP TO FLOW MICROENCAPSULATED PCM'S THROUGH HX

The second brazed plate was substituting in for the shell & tube on account of its terrific performance for the price. The thought was that the fluid, with PCM, would move much faster between the plates than it does in the shell, so there shouldn't be any stagnant paste sitting around.

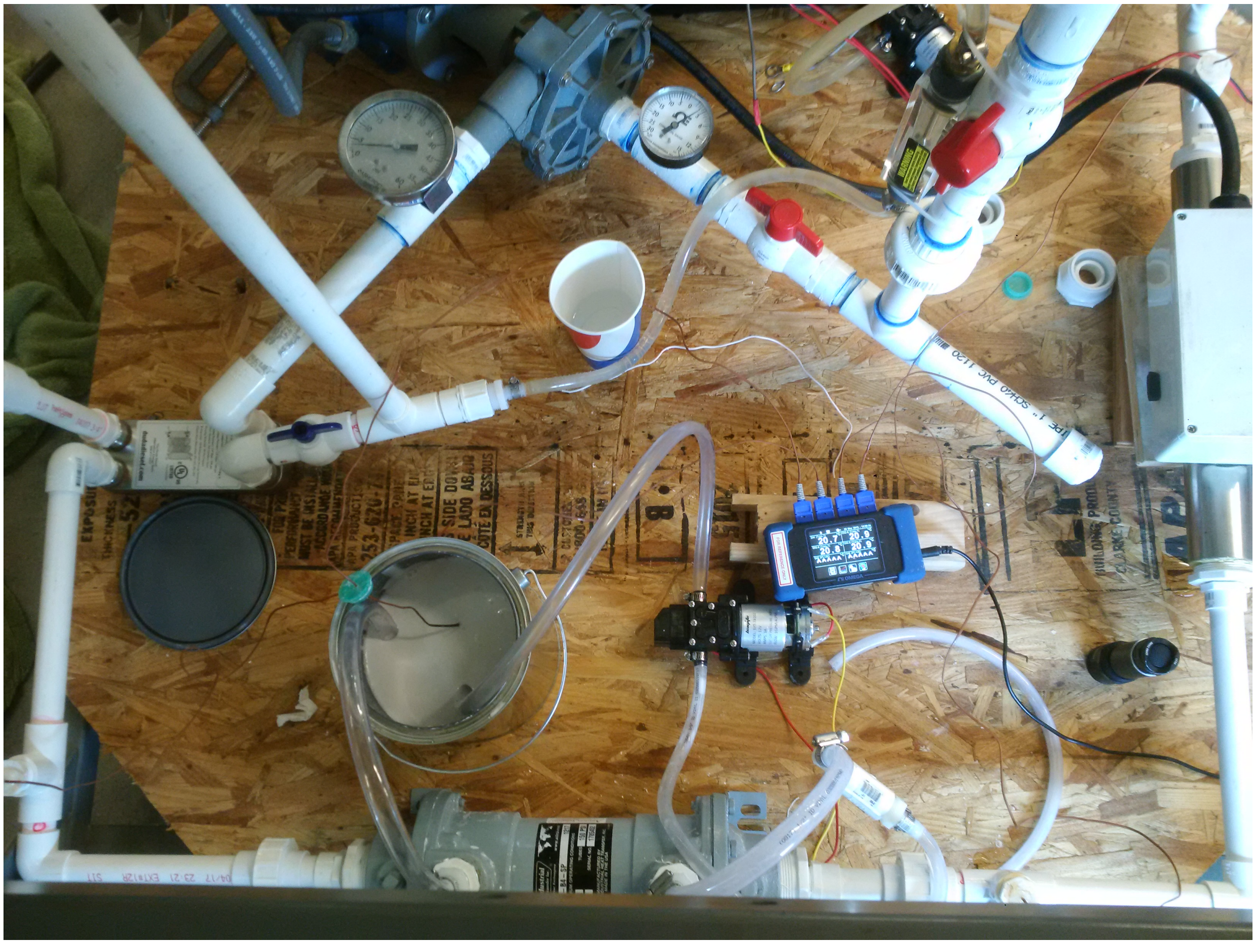


FIGURE 16 TOP DOWN VIEW, SHOWING BRAZED PLATE ON LEFT, TWO PUMPS, EXTERNAL PCM RESERVOIR.

Switching PCMs between tests presents the challenge of completely removing and adding waxy PCMs to the shell side of the heat exchanger without any contamination. We were able to add the correct amount of PCM by heating them until they became liquid and pouring it in through a funnel. Removing them required running hot water through the tubes of the system for an extended period to turn the PCMs completely liquid. The PCM would be poured out and the remaining PCM was flushed out by running hot water through the shell side until there was no trace of the original PCM in the heat exchanger. To confirm that everything had been removed the mass of the heat exchanger was compared to its empty mass. A detailed procedure for this can be found in Appendix J.

7.0 DESIGN VERIFICATION PLAN

There are two main avenues of testing which had to be completed for our project. The first was the thermal profiling of our PCMs. The goal of these tests was to generate accurate data for the heat capacity and melting temperature of our PCMs. The second avenue was the testing of our prototype system. The goal of these tests was to first verify the effectiveness of the system without a PCM, then to add a PCM to the system and analyze its effect on the system. Using this data, we were able to determine if PCMs are a viable method of thermal energy storage for this application. These results can be seen in Section 7.3

7.1 PCM PROFILING TESTS

For the first suite of tests the required equipment was:

1. Scale (2kg range, 1g resolution)
2. Hot Plate
3. Beaker
4. Thermocouple Type T Wire
5. Omega RDXL6SD Thermocouple Reader
6. PCMs to test

The first suite of tests involved the profiling of all of our PCMs. For each material (excluding microcapsules), we heated some known mass of PCM in a beaker, at a constant and known heat transfer rate. For microcapsules, we instead tested many different concentrations in a water solution. At a later stage in the project we attempted to repeat the experiment with antifreeze as the solvent for microcapsules, but this turned out to be ineffective. The formal test procedure can be found in Appendix J. Using temperature data collected during these experiments we were able to generate curves of heat capacity vs. temperature for each material. By testing each material three times we were be able to determine this profile with more confidence.

7.2 SUB-SCALE PROTOTYPE TESTING

For the second suite of tests the required equipment was:

1. Test apparatus
2. Thermocouple Type T wire (5x)
3. Omega RDXL6SD Thermocouple Reader
4. PCMs
5. Power source
6. Water source

The second suite of tests was performed on the fully assembled system. Firstly, the system was checked to ensure safety for the users. Secondly, a control test was performed. This test did not involve PCMs, as it was designed to test the performance of the base system and secondary cooling loop. The results of this experiment were used as a baseline when analyzing the experiment with PCMs included. Thirdly, PCMs (or microcapsules in solution) were added to the reservoir, and the system was tested again. The formal test procedure for these three tests can be found in Appendix J and a listing of the tests performed can be found in Appendix H. The results from these experiments were used to verify or invalidate our prediction of system behavior.

7.3 Beaker Testing Results

Initially it was quite difficult to calibrate our test set up due to the imprecise nature of the hot plate used. Once calibrated, the tests were simple, if tedious, to perform, and yielded mostly expected results. For each wax PCM we were able to generate a curve of specific heat capacity versus temperature. These curves matched the data sheets given to us by the PCM suppliers, but included significant error or irregularities due to the nature of the test setup. Because of this, we opted to use the given data sheets to obtain properties of our PCMs. These data sheets are attached in Appendix F.

The more useful outcome of these tests was the experience we gained in handling PCMs. This experience allowed us to progress much more quickly with our system tests.

7.4 System Testing Results

To determine a baseline (or control) for our test setup we first decided to run the cycle with only air in the PCM heat exchanger, instead of PCM. This test went through several iterations as we sought to eliminate the contribution of as many external variables as possible to the outcome of the test.

7.4.1 Control Tests

The main variable that we had control over was the secondary loop flowrate. There were two constraints that we used to determine its optimal value. The first constraint is that the higher the flow rate, the more cooling that is generated by the secondary loop. If this cooling becomes too high, the PCM heat exchanger becomes unnecessary. The second constraint is that if the flow rate becomes too low, there will not be enough cooling during the low cycle to re-solidify the PCM, or even to bring the coolant temperature back down to standard levels. To determine the optimal secondary loop flowrate, we tested a range of flowrates and examined their effect on the system. An overview of the results of these tests can be seen in Figure 17.

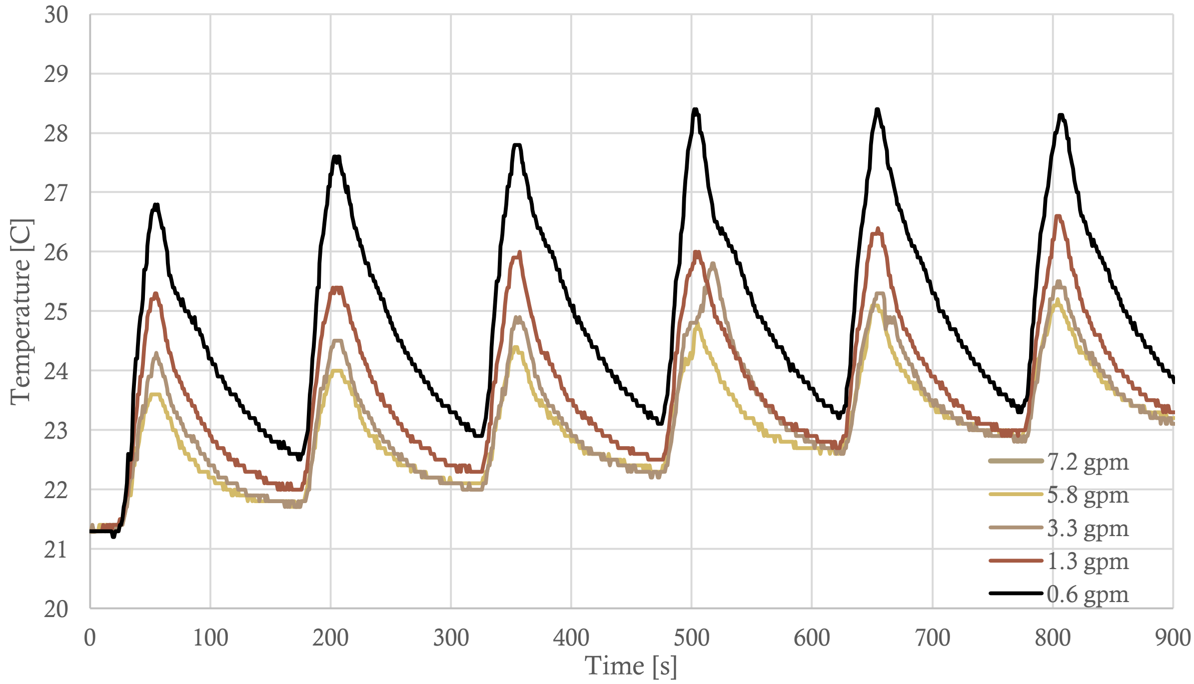


FIGURE 17. TEMPERATURE AT THE INLET TO THE ELECTRONICS FOR PROTOTYPE TESTS AT A RANGE OF FLOWRATES (DATA ADJUSTED FOR DIFFERING INITIAL TEMPERATURES).

Since the steady state temperature increase was the same for all except the lowest flow rate, we decided to perform our tests at around 1.3 gpm, as this would minimize the cooling power provided by the secondary cooling loop.

During a preliminary PCM test (with PureTemp 23) we noticed that there was not enough cooling power to the PCM during the low cycles for it to re-solidify or re-cool. We realized that there was not enough temperature difference to create significant levels of heat transfer. Given that the lowest temperature our primary coolant reached was only approximately 21.5 °C and the PCM during phase change is at 23 °C, there is only a 1.5 °C temperature difference. To remedy this problem, we decided to decrease the temperature of the water in the reservoir and repeat the control test. The results of that test can be seen in Figure 18.

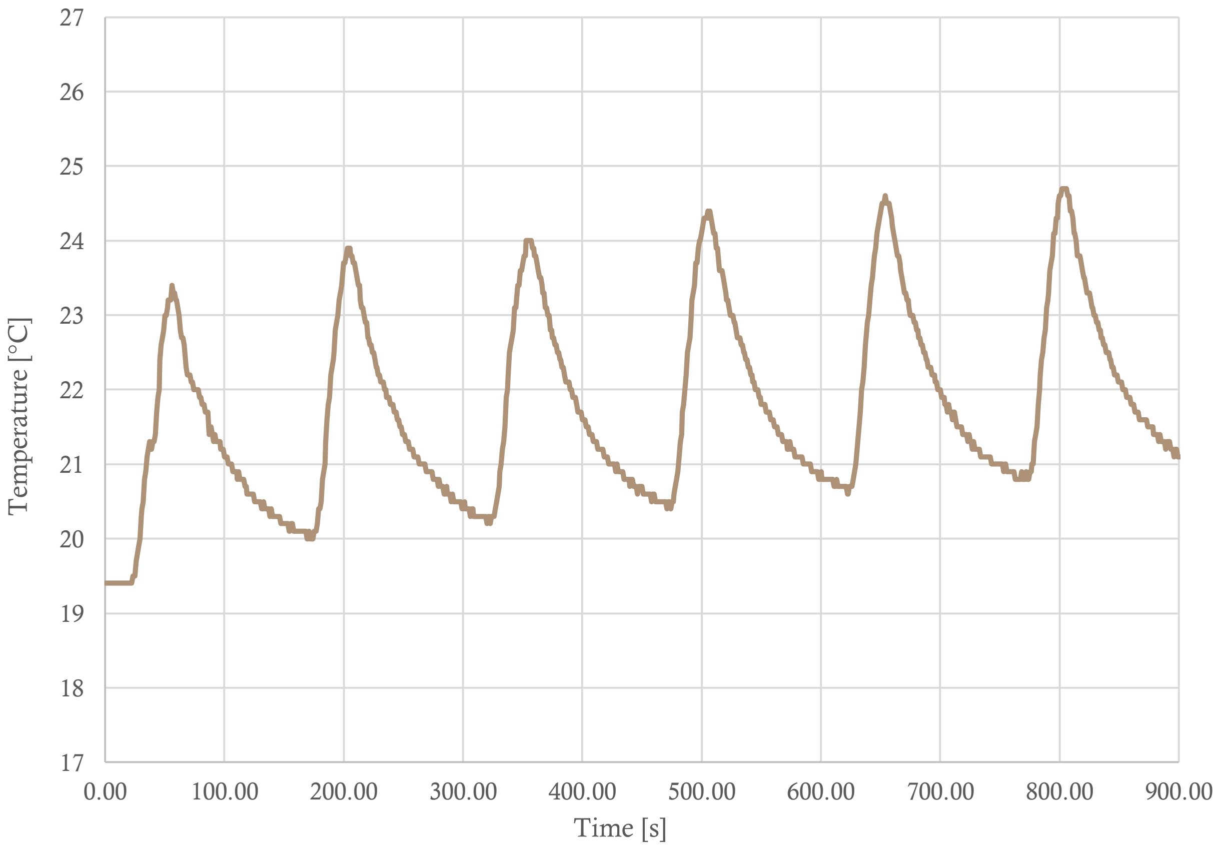


FIGURE 18. TEMPERATURE AT THE INLET TO THE ELECTRONICS FOR CONTROL TEST AT 1.6 GPM WITH TOP TANK WATER AT ~18 °C.

Having decided on a secondary loop flowrate we next tested the effectiveness of water as our thermal storage material, so that we could have a baseline to compare to. The results of this test can be seen in Figure 19.

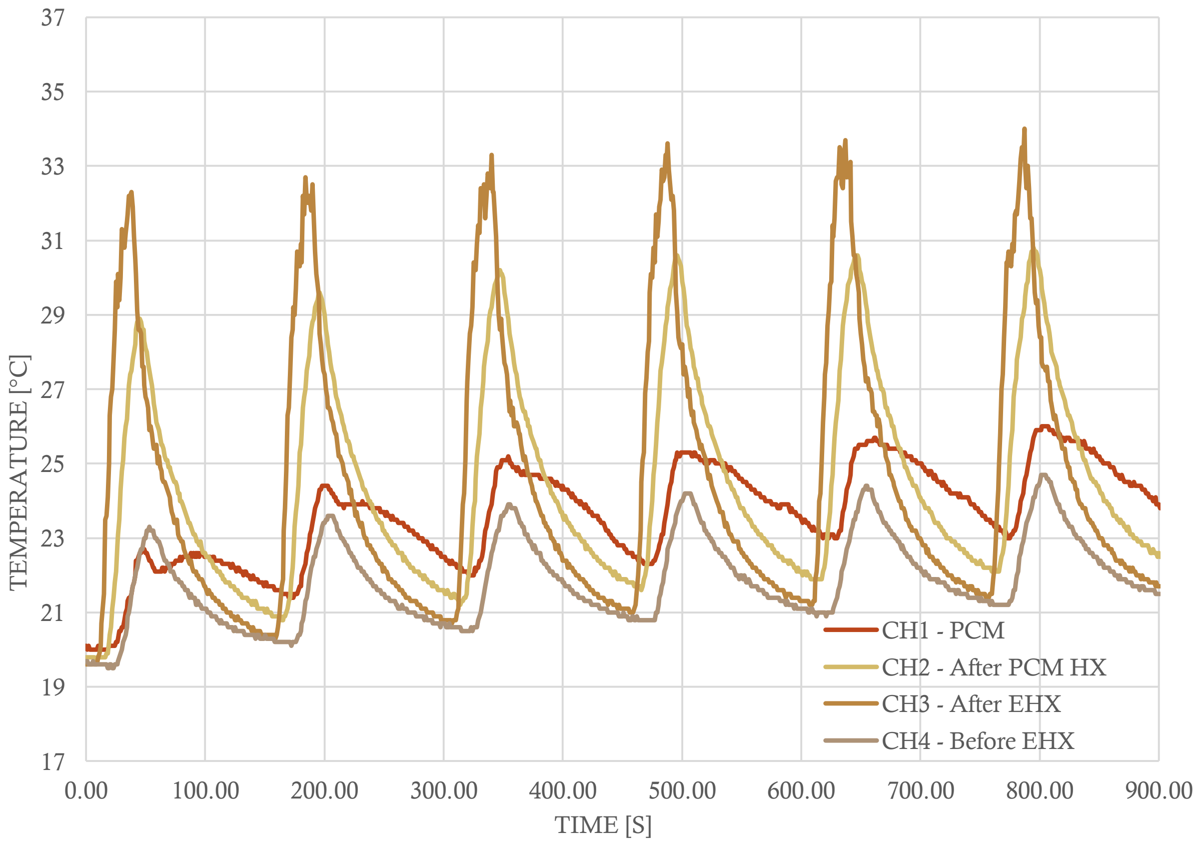


FIGURE 19. TEMPERATURES IN THE PRIMARY LOOP FOR WATER CONTROL TEST AT 1.6 GPM WITH TOP TANK WATER AT ~17 °C.

Over the course of our tests with PCMs there are two pieces that we wanted to examine. First, we examined the peaks of CH4 – The temperature at the inlet to the electronics, as the main consumer requirement stipulates that this temperature remains below 25 °C. Second, we examined the difference between the CH3 and CH2, or the temperature difference across the PCM heat exchanger, as this will show us how effective the PCM was at absorbing heat.

7.4.2 PCM Tests

Over the course of our testing we tested three different PCMs: PureTemp 23, CrodaTherm 29, and MicroTek 24.

First, we tested CrodaTherm 29. The results of this test can be seen in Figure 20. From these results, we determined that CrodaTherm 29 would not be effective for use in this application as it did not reach its phase change temperature over the course of the test, and therefore was unable to take advantage of its latent heat of fusion. This PCM was also invalid because it did not satisfy the primary design requirement; the temperature at the inlet to the electronics peaked above the design value of 25 °C several times over the course of the test.

Although this test was performed before the switch to colder top reservoir water was made, we believe that we can still conclusively say that it is invalid for this application, as its phase change temperature is too high.

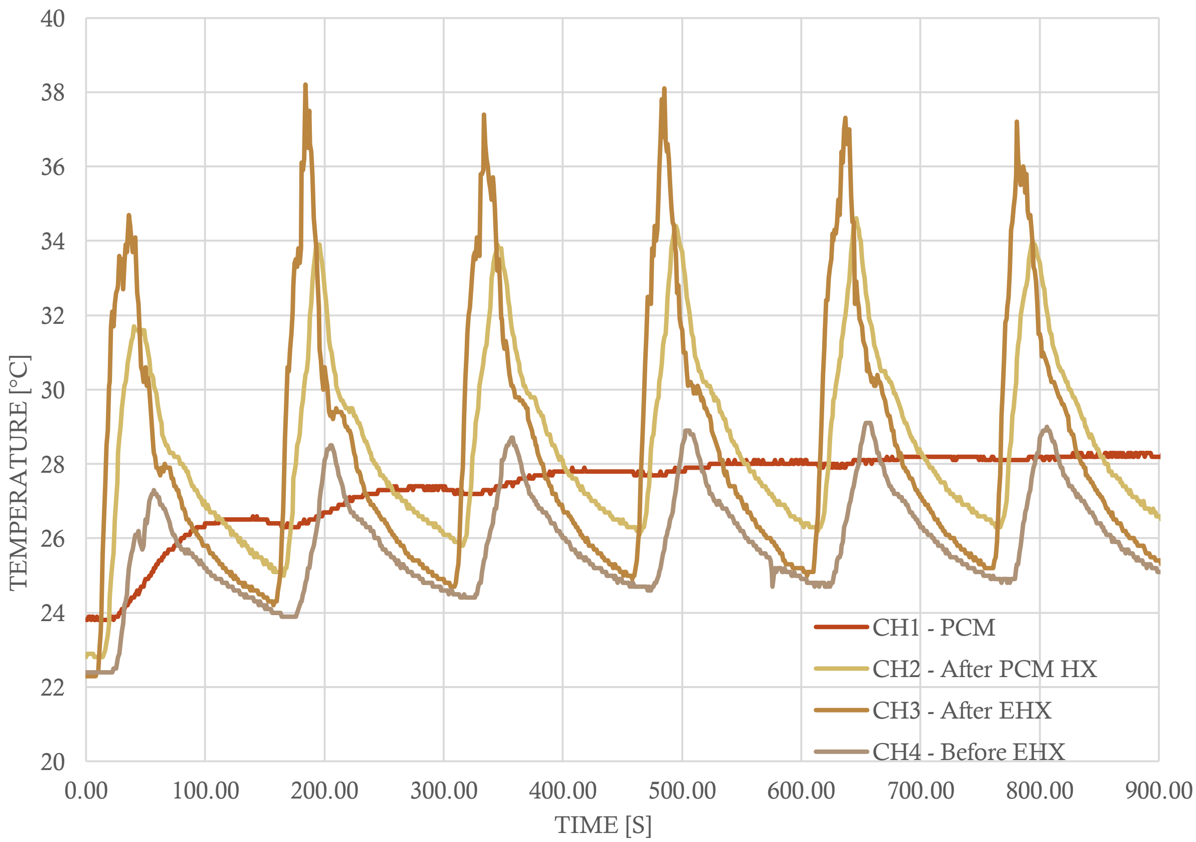


FIGURE 20. CRODATHERM 29 TEST: 1.45 GPM SECONDARY LOOP FLOWRATE.

Next, we tested PureTemp 23. The results from this test can be seen in Figure 21. From these results, we can see that while the PCM is very good at regulating its own temperature (it was never above its melting point), it cannot absorb heat quickly enough to be effective in this system. This result may also be an article of the size of the temperature difference at over the course of the test. Because we could not get very much heat (3.5 kW) out of the in-line heater, the heat transfer rate from the coolant to the PCM was lower than it could have been.

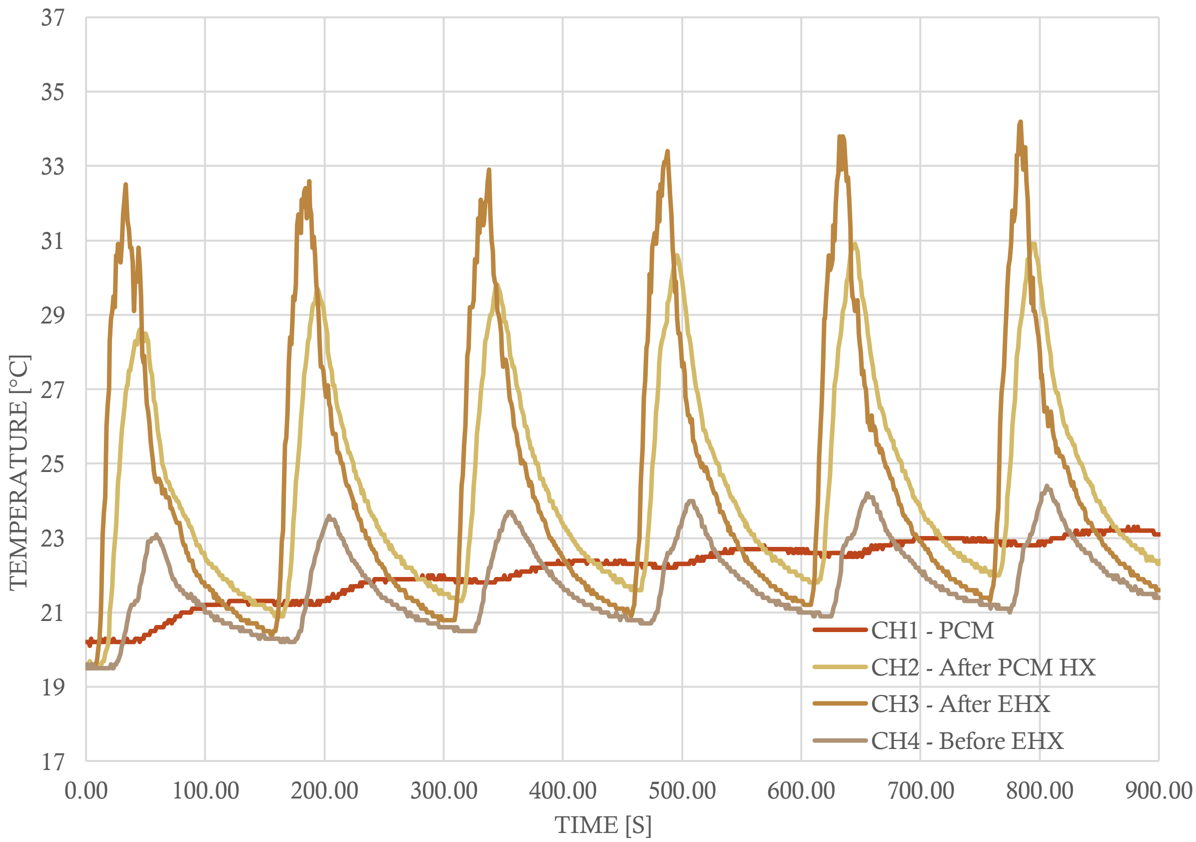


FIGURE 21. PURETEMP 23 TEST: 1.6 GPM SECONDARY LOOP FLOWRATE.

The main surprise was found when comparing the results with PCM to those without. This comparison, which can be found in Figure 22, shows that the addition of PCM had no significant effect on the system.

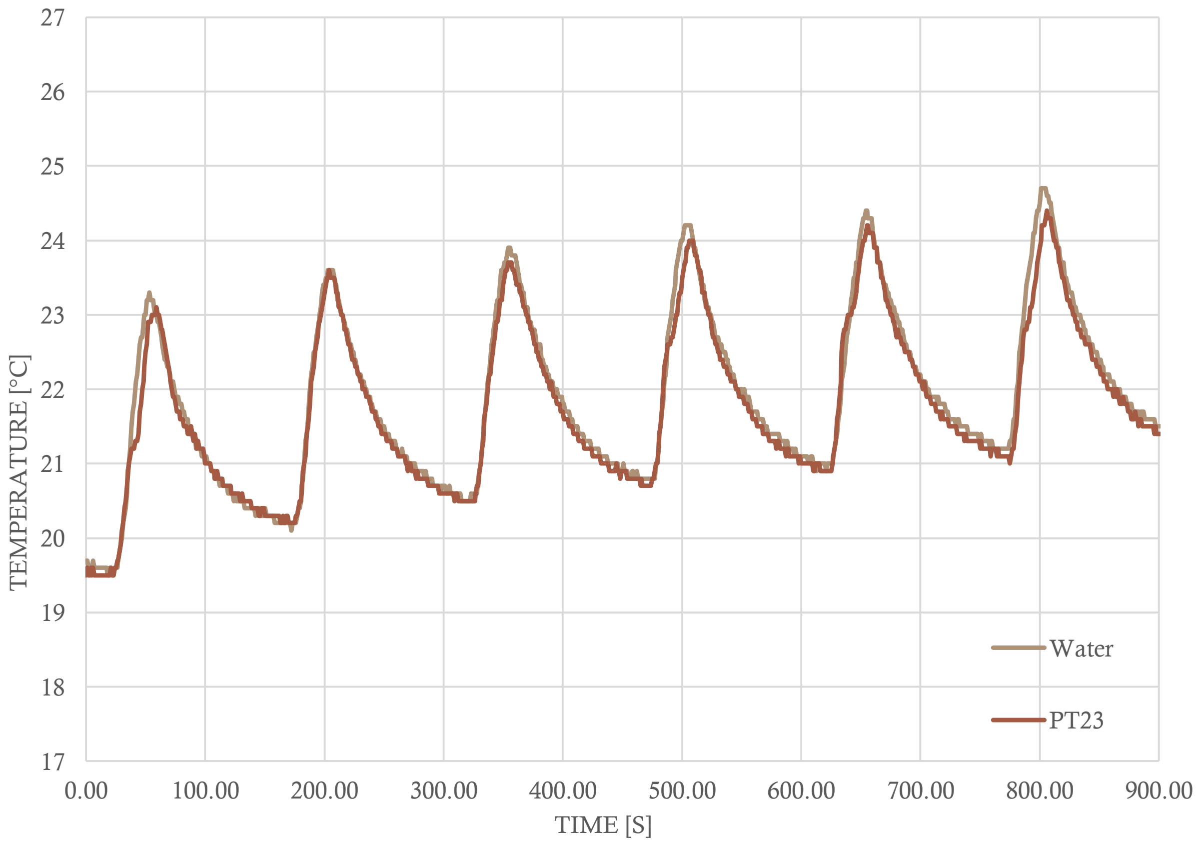


FIGURE 22. COMPARISON: CONTROL VS PURETEMP 23, 1.6 GPM SECONDARY LOOP FLOWRATE.

Another way in which this data can be analyzed is to determine the rate of heat transfer into the PCM. The results of these calculations are shown in Figure 23A. From this figure, we can see that although the PCM seemed to have a negligible effect on the system the PCM heat exchanger was able to remove approximately $1/5^{th}$ of the power added to the system from the heater during the high cycle. The reason that this does not correlate to an effective PCM is that this same amount of heat was absorbed by the PCM heat exchanger during the Control test. This can be attributed to the thermal mass of the brass tubes (~8kg), which are very effective at absorbing heat, due to the high thermal conductivity of brass. Finally, the total heat absorbed by the PCM can be calculated. This data can be seen in Figure 23B. From this chart, we can see that the system does reach a quasi-steady state where the total energy absorbed by the PCM fluctuates from ~35kJ just after the high cycle, to ~25kJ at the end of the low cycle. This steady-state point occurs while the PCM is at its phase change temperature (23 °C), which means that this heat is being used to change the phase of the PCM. Although this does not mean that adding a PCM makes the system effective, it does show that transient thermal energy storage is possible in this sort of system, if not for the scale that we tested.

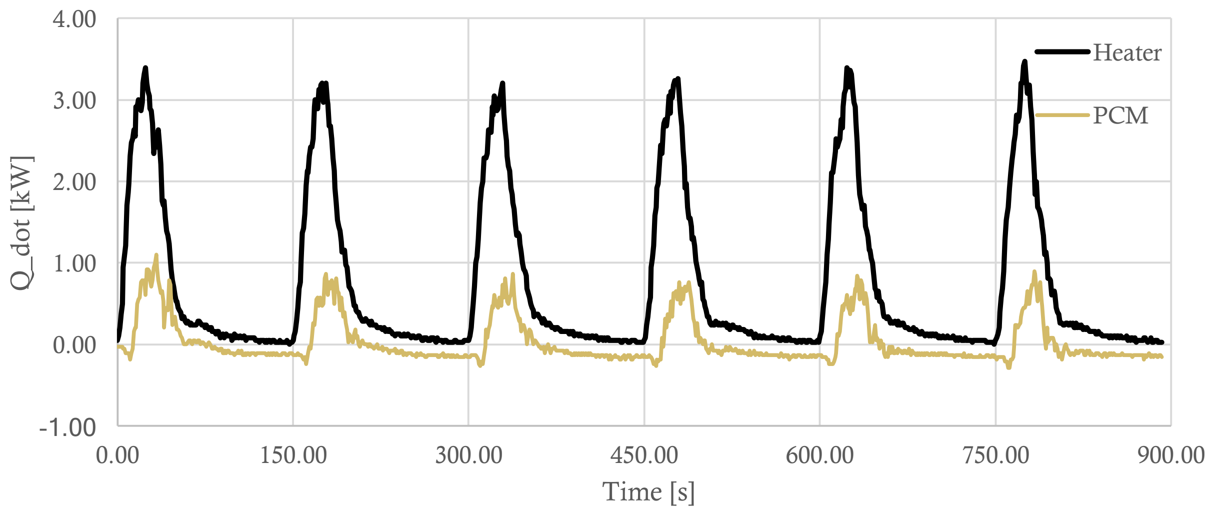


FIGURE 23A. HEAT TRANSFER RATE OF THE HEATER AND PCM HEAT EXCHANGER.

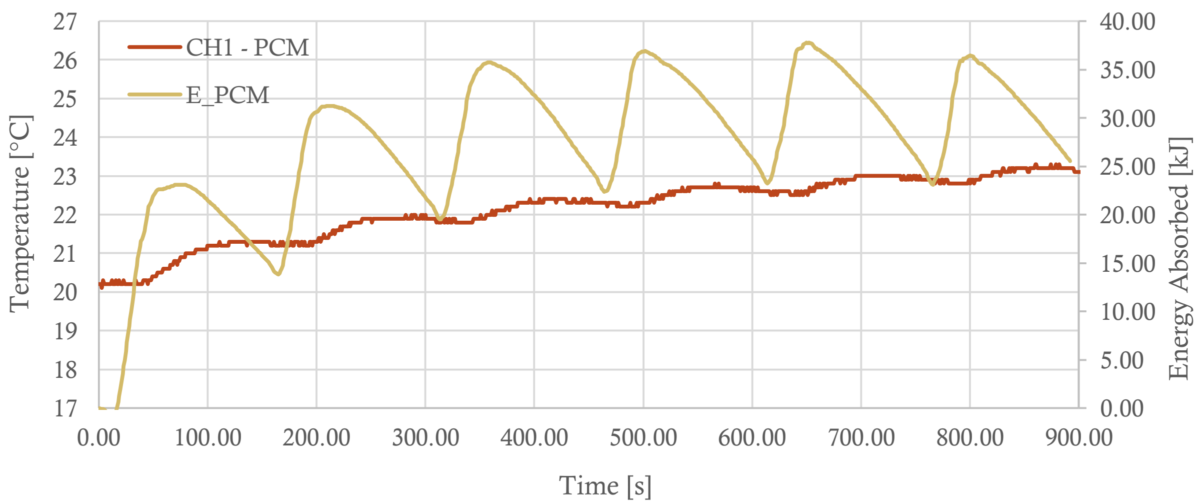


FIGURE 23B. TOTAL ENERGY ABSORBED BY THE PCM.

Finally, we tested MicroTek 24. From our research, we thought that this would be the most ideal PCM because it would have high thermal conductivity relative to the other PCMs and have a better specific heat curve than water for regulating temperature. However, the microcapsules had some issues that inhibited overall performance.

The first test performed quite poorly and at this point we observed the first problem with microcapsules. The microcapsule mixture separated into a paste and blocked any more of the mixture from entering the heat exchanger and preventing it from filling up. The test was re-attempted with the shell side of the tube completely full. This was verified by filling the shell from both inlets pushing the paste through to allow more of the mixture to enter. The results of these tests are seen below in Figure 24 (Microcapsule static and full).

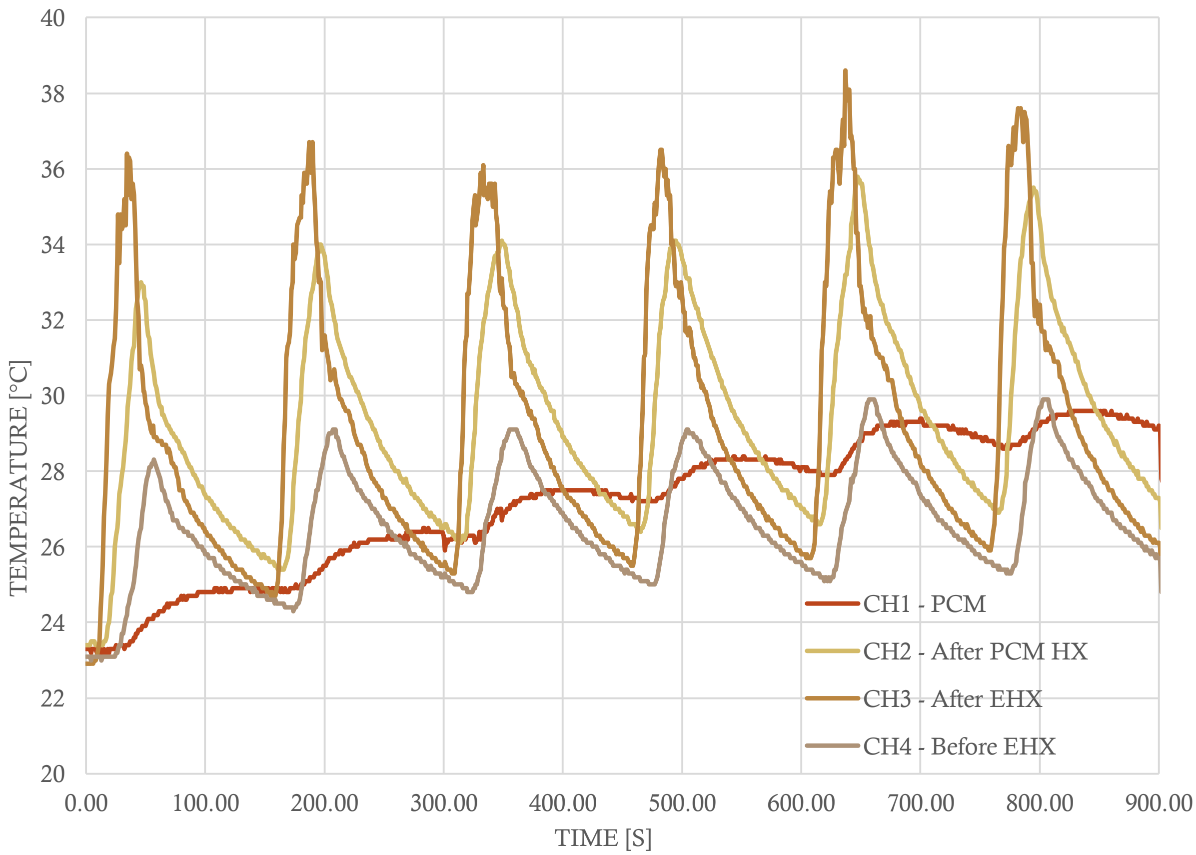


FIGURE 24. STATIC MICROCAPSULES (40% CONCENTRATION).

The results of this test indicated that the microcapsules were still not very effective at regulating the inlet temperature. This is where the second issue of the microcapsules was observed. The bottom of the reservoir was very warm to the touch and the top was cool. This is because the microcapsules separated inside of the reservoir. The water was able to absorb heat very quickly in the low concentration area (incidentally this was where the thermocouple was located), but the paste that formed at the top of the reservoir had very poor thermal conductivity and did not absorb very much heat.

The next step for evaluating microcapsules was to find a way to keep them in solution to remain effective. We tested the microcapsules at a much lower concentration of 25% and ran it through the shell side of the heat exchanger with a pump and reservoir of microcapsules. The results of the test can be seen in Figure 25. Even though we were able to keep the microcapsules in solution with this test, they were not any better at regulating the temperature of the coolant.

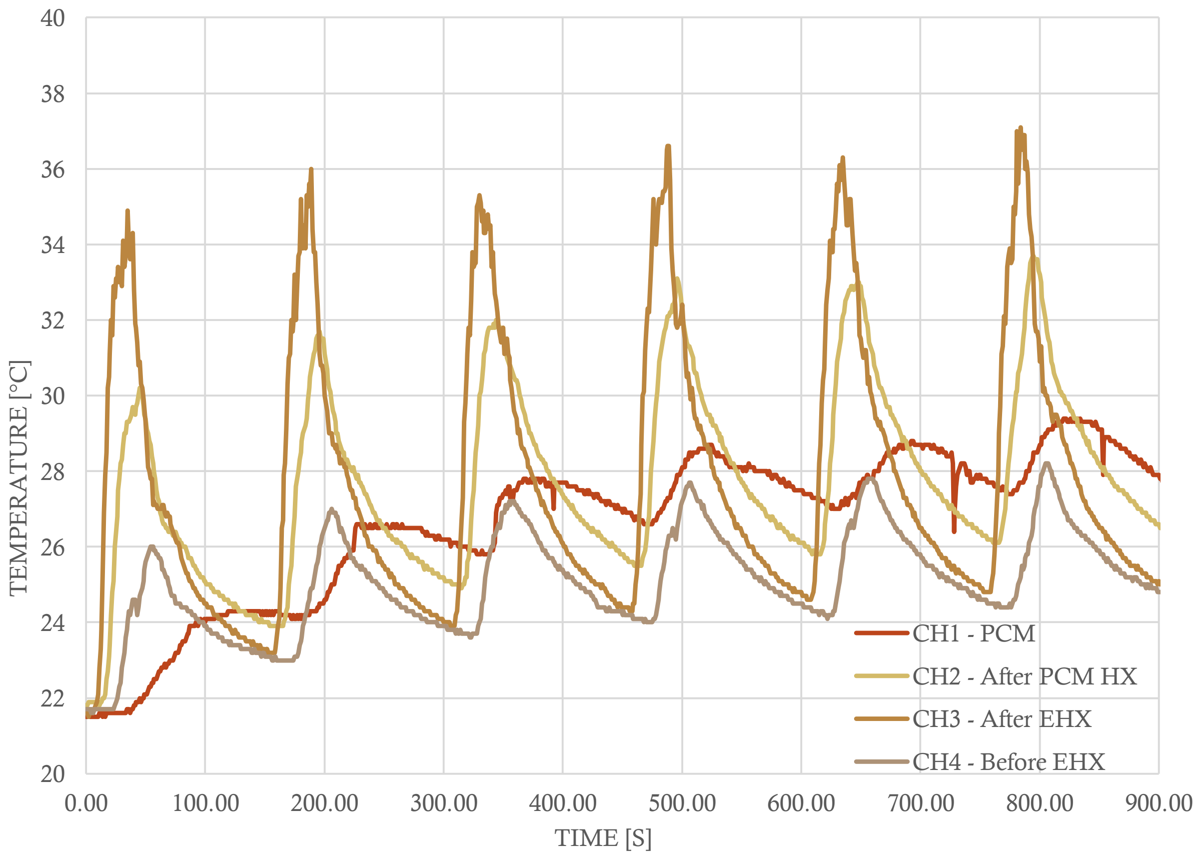


FIGURE 25. FLOWING MICROCAPSULES (25% CONCENTRATION).

To compare the effectiveness of the current architecture to other design options so we replaced the tube and shell heat exchanger with a brazed plate heat exchanger. The results of this test are seen in Figure 26. Although these results show that this configuration would be effective, it would be essentially the same as increasing the cooling power of the secondary cooling loop, thus adding weight and complexity to the design. It also was still unable to show the effect of the PCM, as to create a mixture that could flow the concentration had to be quite low.

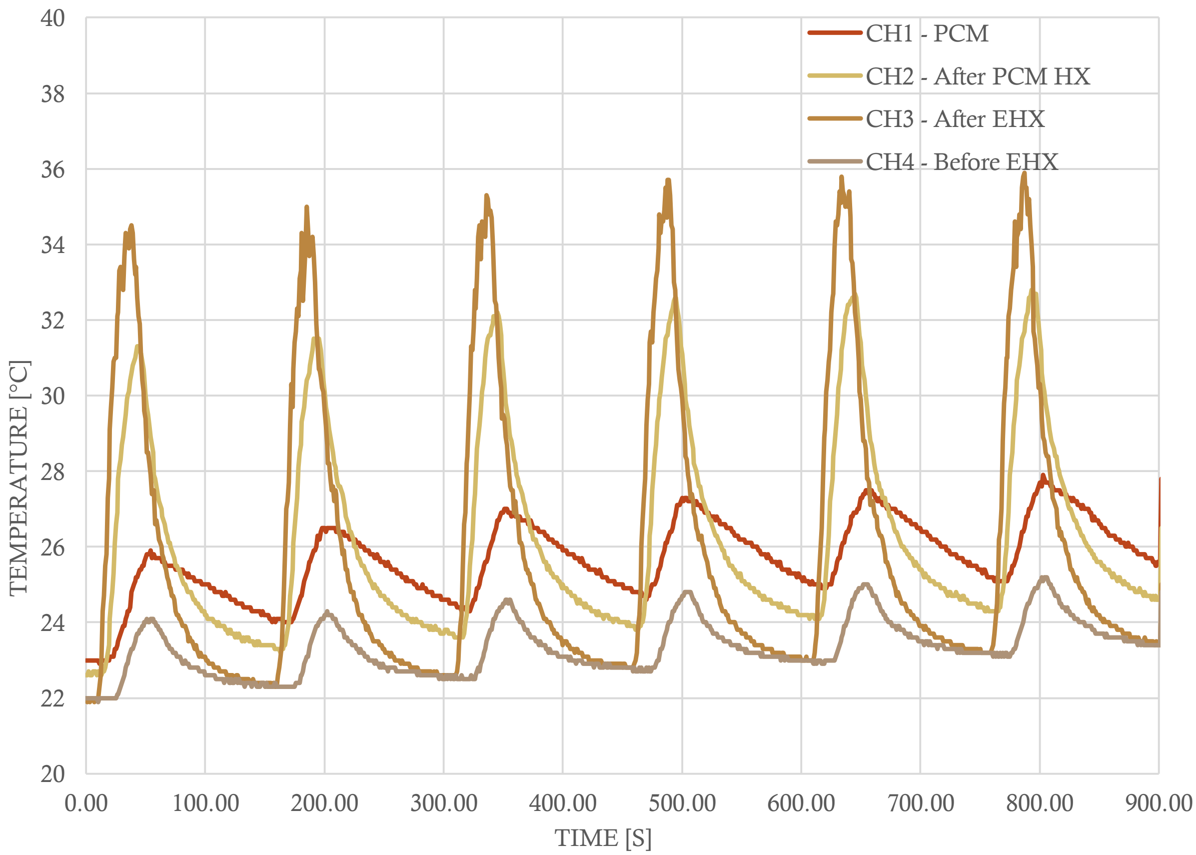


FIGURE 26. FLOWING MICROCAPSULES (25% CONCENTRATION), BRAZED PLATE HEAT EXCHANGER.

The change in the architecture did not have a significant effect on the results of the test, so we decided to consider other options for utilizing microcapsules at a higher concentration. We looked at the microcapsule material and learned that it was a nonpolar compound. So, we tried mixing the microcapsules in nonpolar solvents. The selected solvents were antifreeze and mineral oil. We put a 30% mass concentration of microcapsules in a beaker and attempted to mix them together and let them sit. The antifreeze mixed but eventually had the same separation out of solution as with water. The mineral oil would not even mix with the microcapsules and so was deemed inviable. The only way we determined to keep the microcapsules in solution to remain effective was to continuously stir or circulate a low concentration solution in the system. We attempted to contact MicroTek (the supplier) for any suggestions but they have not responded to us at the time of writing this report.

7.4.3 Other Results

Part of what we were hoping to see was an effective heat exchanger for the duration of the high cycle. Another part was a classic nonlinear graph of temperature vs. heat absorbed, where the temperature increases then levels out during the phase change, then increases again. Neither of these things happened. The closest we came to a nonlinear result is shown in Figure 27, In this test, the heater ran for seven minutes straight to get the PCM temperature from 15 degrees below the phase change temperature to ten degrees above.

The rate of heat absorption is graphed in Figure 28. The line of temperature ascension gradually decreases, but that is because the heat rate slows down with lower temperature differences. Eventually, at 80 degrees, the PCM temperature rises quickly, even with little heat absorption. From this we can see that the PCM does not act much like a PCM as it does not hold temperature during melting. This can be attributed to the low concentration of PCM compared to water.

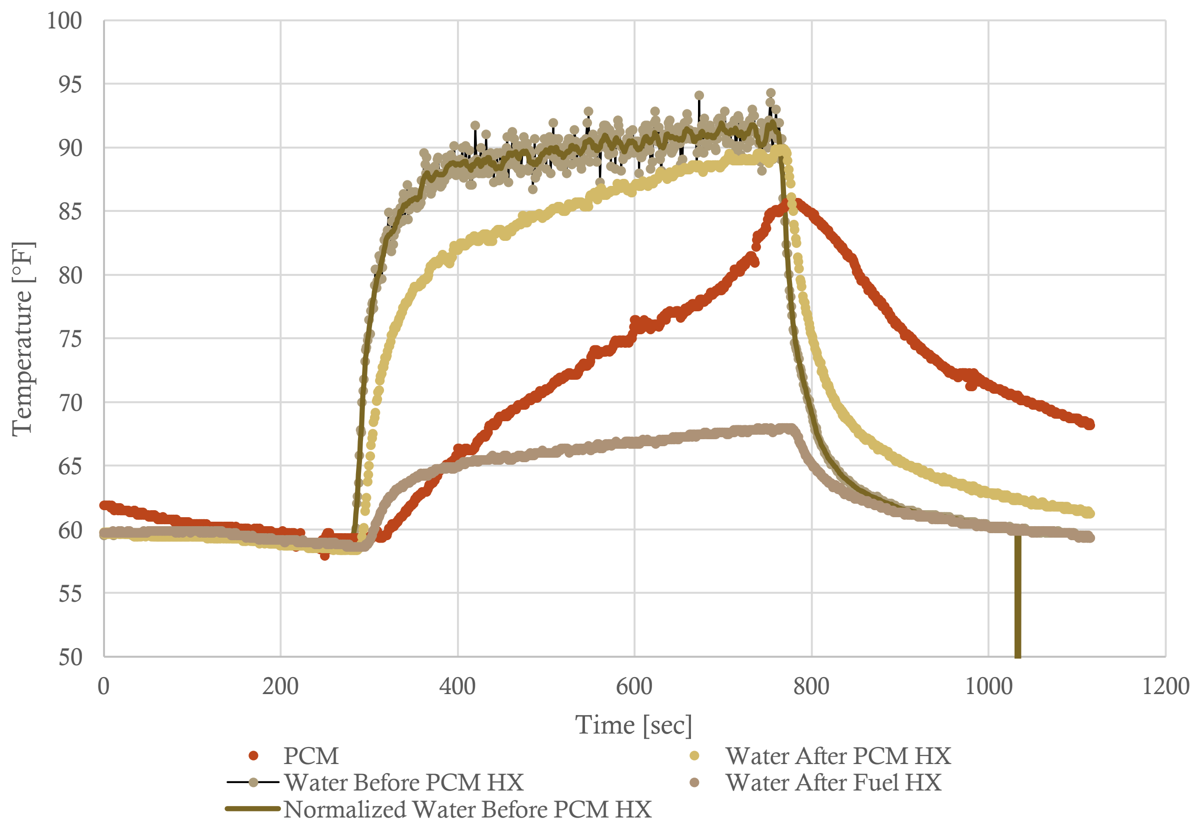


FIGURE 27. FLOWING MICROCAPSULES IN SHELL, WITH EXTRA COLD WATER IN TANK.

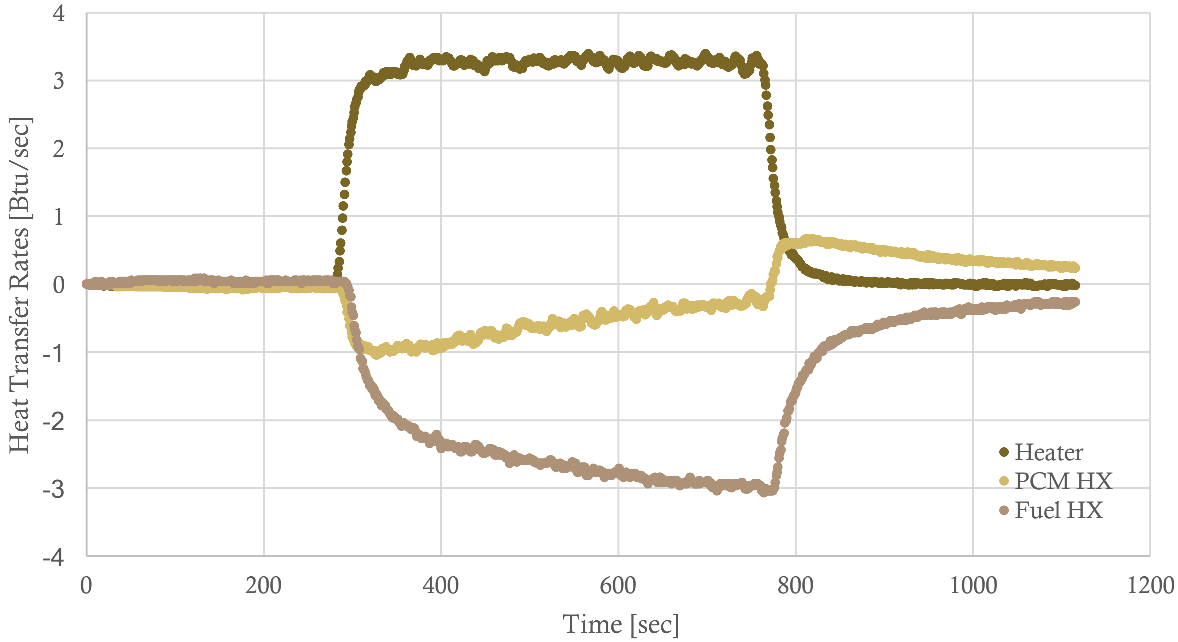


FIGURE 28 RATES OF HEAT TRANSFER IN EACH ELEMENT.

7.5 Scaling up the Results

One of the goals of this project was to use our subscale prototype to model how a full-size system would perform. To do this a Simulink model was created with parameters matching our subscale system. This model was then tuned such that it produced data that matched our experimental results. Using these tuned parameters, the model could then be scaled back up to the full-size system, and some results could be extrapolated. At the conclusion of this study it was determined that it was likely that the effect of the PCM could be seen much more easily on a larger scale, as our small-scale setup did not take full advantage of the thermal properties of our PCM.

7.5.1 Creating the Model

Using the Thermal Liquid library in Simulink, we were able to construct a mathematical model to simulate our system. The block diagram for this model can be seen in Figure 29.

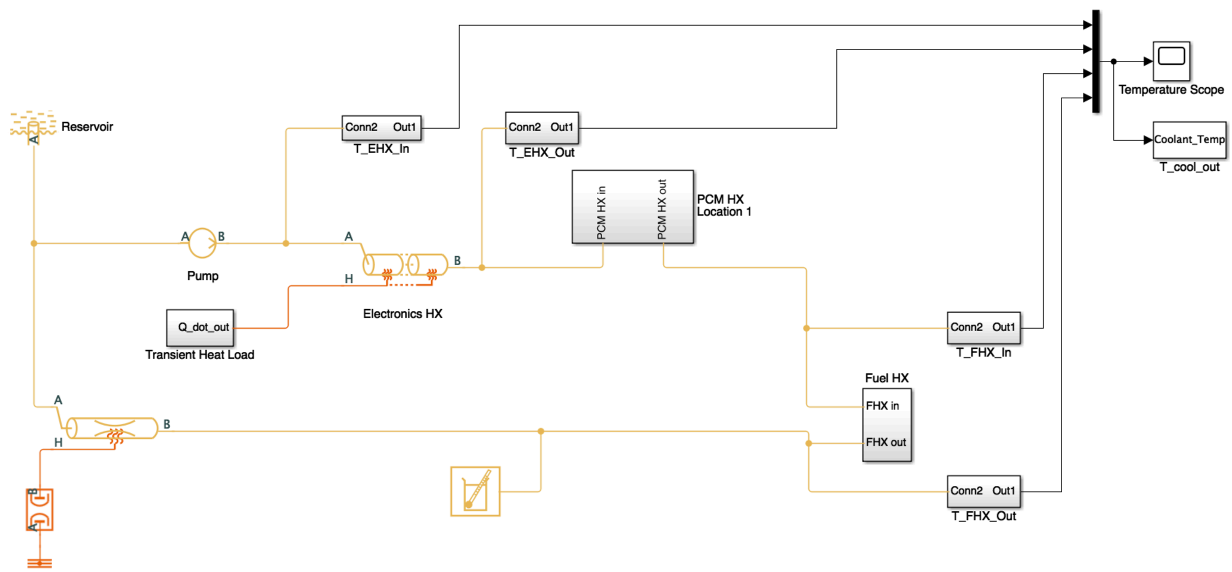


FIGURE 29. BLOCK DIAGRAM OF THE SIMULINK MODEL.

There are three subsystems present in this block diagram. The first, and also the simplest, is the *Transient Heat Load*. This subsystem is composed of a repeating step input such that the correct transient heat profile can be formed. The second is the *Fuel Heat Exchanger.* This subsystem is composed of the secondary cooling loop, which flows low temperature coolant through a heat exchanger to remove heat from the primary loop. The last subsystem, and the most complex, is the *PCM Heat Exchanger*. The complexity for this element arose when attempting to model the PCM. Eventually it was modeled as a variable specific heat thermal mass, where the specific heat at the phase change temperature is the latent heat of fusion.

One drawback to this method is that the quality of the PCM (or %liquid) could not be measured directly. Another drawback is that this method would be ineffective for substances that melt and freeze at different temperatures.

7.5.2 Tuning the Model for the Subscale System

The main variables being tuned for this model were the heat-transfer parameters from the primary loop to the PCM. To tune it, we first would compare the simulation results to experimental data for a control test. The next step was to adjust the heat transfer parameters until these matched. Then we compared the results of the simulation with experimental data of a PCM test. If these matched for many different secondary loop flowrates, as well as for different PCMs, the model was considered tuned. Figure 30 A and B shows the results from the tuned model compared to the experiment.

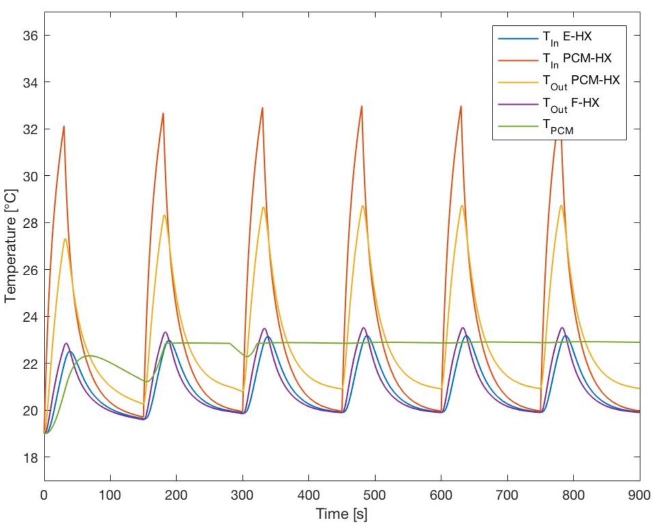


FIGURE 30. RESULTS OF THE SIMULATION MODEL

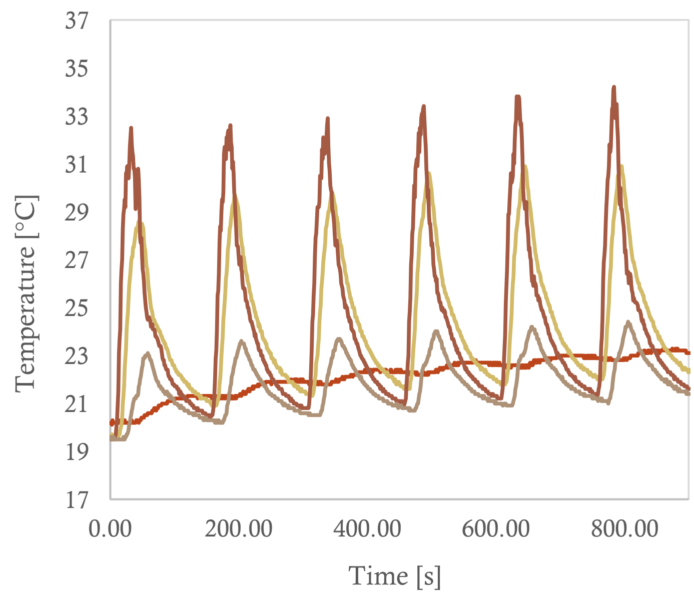


FIGURE 30B. RESULTS FROM THE EXPERIMENTAL TEST

7.5.3 Scaling Up the Model

To scale up the model, the system parameters simply had to be adjusted to match the parameters of the actual system. This included parameters like primary loop flowrate, secondary loop flowrate, and primary and secondary loop fluid properties.

Once the parameters were in place the system was simulated with no PCM to act as a control test. Then the system was simulated with increasing masses of PCM until the system reached quasi-steady-state. These results can be seen in Figure 31.

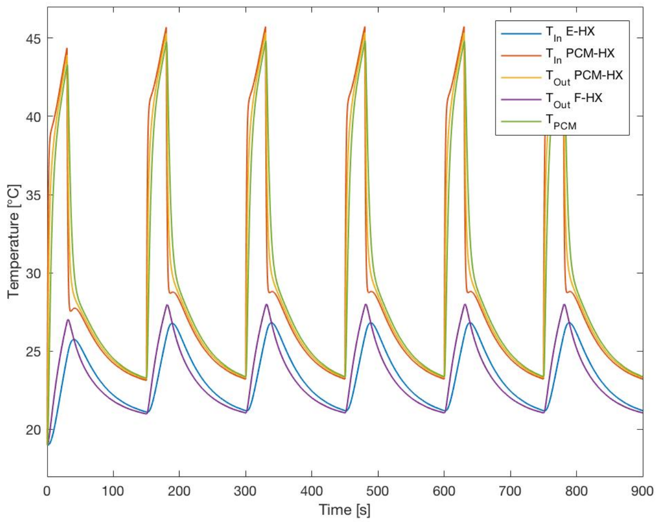


FIGURE 31A. RESULTS OF THE SCALED-UP SIMULATION MODEL (NO PCM)

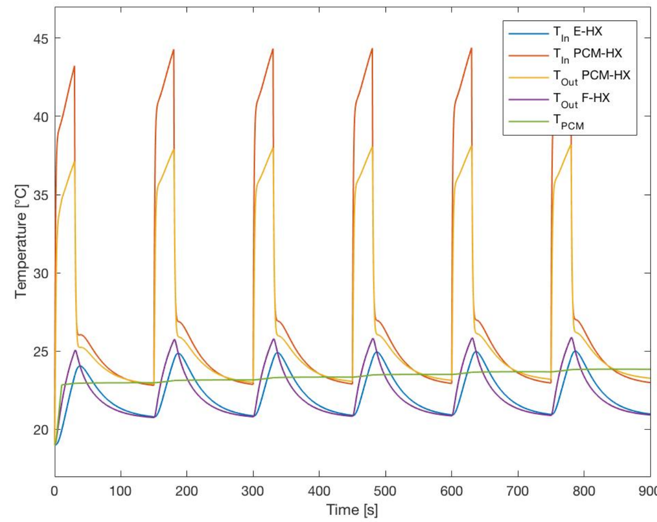


FIGURE 31B. RESULTS OF THE SCALED-UP SIMULATION MODEL (15KG PCM)

Although this simulation was able to show that the system could be made more effective by the addition of PCMs, the 2-3°C of cooling does not justify the ~15kg of PCM required.

7.6 What Would it Take?

If we ignore the problems faced with the test apparatus, and used a wax PCM to store the excess transient heat, then we can hypothesize about the final design that would satisfy the problem statement. Using only a small portion of the theory presented in Appendix L.2, it would take at least 25 ft^2 of surface area of perfectly efficient aluminum fins, spaced about 1/32 of an inch apart, to melt the couple of pounds of wax. This corresponds to some ten pounds of aluminum fins at 5 thousandths of an inch thick. Because it would take so much metal, the PCM benefit of thermal mass for very little real mass is lost. Plus, it requires cold fuel in the ultimate heat sink, or else the cycle times would increase substantially.

Using gallium is another possible PCM to absorb the transient heat. Somewhere around 7 pounds of gallium would be used and half a pound of surface area. Gallium has the property where having some solid gallium makes it much easier to re-solidify. This means that a system implementing gallium requires an excess amount that will remain solid for the whole cycle. Gallium is also fairly expensive at 150 dollars per pound so more cost effective PCMs should be considered first.

A PAO reservoir is probably the most reliable and sellable fix. Section L.6 graphically describes how it may only take ~2.25 gallons of extra PAO to successfully regulate the coolant temperature.

8.0 PROJECT MANAGEMENT

This project was initiated in fall 2017 and reached completion in June of 2018. The project included several milestones for class and sponsor obligations.

The early phase of the project involved individual outside research into PCM's for the systems and environments expected, and possible ways to integrate the materials into the sponsor's system. Following individual research, the team brainstormed several possible systems and discussed them in a series of meetings between the team members and the sponsor. The team organized our findings from the early stage and delivered a presentation to PDT.

The settled test PCM reservoir was a tube and shell heat exchanger. PCMs were tested in a beaker and their thermal properties were compared to the supplier specifications. Results of the PCM testing were organized into a set of summaries about PCM specs and relevant notes on their performance. While the data on PCMs was collected, the team tried to gain access to the fluids lab and apparatus. While permission for fluids lab access was granted early, several obstacles prevented testing with the full apparatus for several weeks. Following the final sign off by safety coordinators testing on the system began. Testing was refined throughout the month of May, and PCM performance was analyzed to determine viability in the full-scale system.

The table below shows the main deliverables for the project. The PCM matrix and functional subscale prototype system were completed and used for testing. The system architecture and full-scale design requires further refinement in order to be viable for the actual system based on the poor results from the PCM testing. The main issue was low thermal conductivity, so an improved system architecture and a PCM optimized for thermal conductivity are recommended in order to implement PCMs in a full-scale design. We have sent the PCM matrix to PDT and we used a modified tube and shell heat exchanger as the system architecture. The functional prototype was completed using the tube and shell heat exchanger and incorporated into an existing apparatus for testing the system on a lower thermal scale.

TABLE 3. KEY DELIVERABLES

Deliverable	Description
PCM matrix	Comparison of commercially available PCM's, include material properties, price, availability, etc. Once samples are obtained, test to determine effectiveness in system
Theoretical System Architecture	Multiple possible designs for integrating the PCM reservoir with the coolant and/or electronic components. Including comparison and ideal design choice.
Design for a full scale thermal cycle	Full design of a system with parameters as stated in Table 2: Design Specifications: 50kW high, 3kW low load for a 20% duty cycle. Includes block diagram of the system and the PCM reservoir architecture.
Functional Subscale Prototype System	Prototype that functions on a 10:1 or 20:1 scale of the full system and demonstrates the viability of the concept.

This report marks the end of the project and there were several milestones that were important for the class and for PDT. The table below shows the important tasks and dates for the project culminating in the completion of the project in June 2018.

TABLE 4. TIMELINE

Name	Description	Final Draft Delivery
Scope of Work	Document describing the nature of the project and expected deliverables to the client from the team.	10/13/17
PCM Matrix	Matrix that compares a variety of PCMs based on supplier specifications, type, and price.	10/27/17
PDR Report and Presentation	A report and presentation for PDT and our class describing the conceptualization process and the initial ideas for the PCM's system architecture.	11/17/17
Interim Design Review	Class presentation discussing manufacturing methods and plans for testing with the apparatus.	1/16/18
CDR Report and Presentation	A report and presentation for PDT and the class detailing our design for the prototype system and our testing plan. Design for full-scale system supposed to be based on test results.	2/08/18
Manufacturing and Test Review	Presentation on manufacturing progress and updated test plan. Updated schedule for project completion.	03/13/18
Hardware/Safety Demonstration	Testing presentation showing that the prototype is safe and functional.	04/26/18
Operations Manual	Document containing the correct way to operate the system. This will point out safety hazards and include pictures.	05/22/18
FDR Report	Working Subscale Prototype, design for a full-scale system and a report on feasible PCM's and system architectures.	06/06/18
Project Expo	Public Presentation on the Project and showcase of the functional prototype.	06/01/18

For the project schedule that outlined the specific tasks of the project see Appendix K for the Gantt chart.

8.1 Project Plan Deviations

There were a few deviations from our timeline and the milestones that were part of the main project. Many of the deviations occurred in the last quarter of the project because of issues with using the lab space and results from the testing data.

The milestones that did not achieve desirable results were the system architecture and design for the full-scale system. The system architectures tested was the tube and shell heat exchanger and the brazed plate (for microcapsule flow), both before and after the electronics in the system. These architectures did not have very much of a meaningful difference between one another, but they each had poor thermal conductivity for the PCM application. We recommend that PDT continues to research other possible system architectures for better performance. The team has recommendations for developing a full-scale design using PCMs to improve the performance of the system, but our testing results did not give promising aspects that we think should be incorporated into the existing system at this time.

There were some delays in the project plan and the timeline for some of the class obligations that were a smaller concern than the missing of the milestones. The report deadline was extended for a little under a week to prepare for the Project Expo and give more time to edit the final report. The main delay during the project was being restricted from testing the subscale model of the system sooner in the spring quarter. The team needed to get permission to have access to the fluids lab, modify an existing apparatus to create the subscale prototype, and to run tests of the system. The main issue was that the system had to have work altering the electrical supply for the prototype and had to jump through several unexpected hoops before changing the system and being able to run them.

9.0 Conclusion

The project studied how PCMs could be implemented in a high power low duty cycle cooling system that is small, lightweight, and low cost. By the conclusion of the project we have successfully generated a PCM matrix, a study of theoretical system architectures, and a functional subscale testing apparatus, and a modeled response for a full-size system. Although PCMs were not effective at regulating the temperature of the coolant in our system, modeling suggests that with a larger temperature differential or a larger area, the performance of the PCMs will improve.

9.1 Recommendations for Future Progress

This project has a few steps to move forward with the implementation of PCMs in the overall system and to properly harness their properties. First, the testing apparatus requires an improvement to the heater output of the system. We would recommend a heater output of 6 kW or above in the subscale system to create a high enough temperature differential to see the effects of the PCM in the system. PDT should note that this requires the fixed tank volume to be regulated at a constant temperature during testing or to use an open source to provide a constant temperature to the recharging heat exchanger.

We also have a few recommendations for improving the overall design of the system. The system architecture could use some improvements to its thermal conductivity by increasing the surface area of PCM relative to its volume. The architecture could also have weight reductions by removing the tube bundle from the heat exchanger and replacing the steel shell with an aluminum one. The PCMs should be reselected focusing on the thermal conductivity of the materials because most of the waxes struggled to transfer while solid. Because of this a small layer of insulation of solid PCM would form during the cooling portion, so the heat exchanger was unable to transfer enough heat the liquid portion to solidify it.

If these improvements were implemented, we believe that the prototype would give a better indication of the potential of PCMs in the prospective system.

9.2 Final Thoughts

While this project was not a complete success for developing a new design of the target system, we believe that there is potential for the overall system and a path forward for PDT or future students to work on. The next steps are to create a large temperature differential to get a better handle on performance, and improve the thermal conductivity in the system architecture and the selected PCMs.

If PDT has any further questions or requests clarification the team can be reached at transientheatstorage.cp@gmail.com.

Thank you to PDT for sponsoring this project, and to Professor Rossman for keeping us on track, we very much enjoyed working with you for this past year.

Sincerely,

Cal Poly Transient Heat Storage Team

10.0 WORKS CITED

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4. *Nexusenergycenter.org*, www.nexusenergycenter.org/use-phase-change-material-thermal-buffer-indoor-comfort/.
5. Shell and Heat Tube exchanger, by Harsh Shah, GrabCAD. https://grabcad.com/library/shell-and-heat-tube-exchanger-1

APPENDICES.

Appendix A. House Of Quality

	WHO: Customers																				
								Column #	1	2	3	4	5	6	7	8	9	10	11	12	13
								Direction of Improvement	▲	▲	◇	▼	▼	▼	▼	▲	▼	▼	▼	▼	▼
Row #	Weight Chart	Relative Weight	PDT	Defense contractors			Maximum Relationship	WHAT: Customer Requirements (explicit & implicit) / HOW: Engineering Specifications	High load cooling	Idle load cooling	High to Low cooling power ratio	Weight	Volume	Cost	Power Requirement	Number of cycles until failure	Flamability	Corosivity	Toxicity	Pressure Loss	Coolant Temp
1		7%	5	5			9	Lightweight				●	○								
2		4%	4	2			9	Low Cost				○		●							
3		3%	2	2			9	Low power usage	○	○	○				●						
4		6%	4	4			9	Small					●								
5		7%	4	6			9	Safe	○								●	●	●		
6		8%	6	6			9	Long material life								●					
7		7%	3	7			3	Works in environment					○		○		○	○			
8		14%	10	10			9	Electronics stay below max temp	●	○											
9		14%	10	10			9	Stays cool during high load period	●												
10		10%	7	8			9	Recovers thermal storage potential during idle period		●	●										
11		11%	8	8			9	minimize pressure loss												●	
12		10%	7	7			9	low coolant temp													●
								HOW MUCH: Target	6 kW	300 W	20:1	min	min	min	0	5000 hrs	Non-Flamable	Non-Corosive	Non-Toxic	25 psi	20-30 ºC
								Max Relationship	9	9	9	9	9	9	9	9	9	9	9	9	9
								Technical Importance Rating	277.43	142.71	101.29	74.714	90.857	37.714	45.286	74.571	82.143	82.143	61.714	99.429	87
								Relative Weight	22%	11%	8%	6%	7%	3%	4%	6%	7%	7%	5%	8%	7%
								Weight Chart													

APPENDIX B. DECISION MATRICES

B-1. SOLUTIONS LIST

1.) PCM Reservoir: PCM is stored in a tank that is in thermal contact with the coolant in the system. The tank may resemble a tube and shell architecture.

2.) Microcapsule PCM Reservoir: The same system as the PCM reservoir, but the material in the tank is a slurry with microencapsulated PCM stored inside of a water glycol mixture.

3.) Direct Microcapsule: The microencapsulated PCM is directly inside the PAO and changes phases along the coolant loop.

4.) Solid-Solid fins: Using a type of PCM that stays solid throughout the phase change (Absorbed in a sponge to retain shape) as a set of fins inside of the coolant loop. The fins regulate temperature from within the loop directly with temperature regulation.

5.) PCM electronic cooling: Placing the electronics in direct thermal contact with the PCM while the PAO cools the PCM separately.

6.) Pipe “sleeve”: Have a sleeve around the piping of the system that contains PCM to regulate the temperature throughout the system. Different materials can be in various sections of the system.

7.) Secondary Coolant Loop: The PCM reservoir has its own cooling loop that is connected to the main loop with a heat exchanger.

B-2. Weighted Decision Matrix

Criteria	Relative Weight	Concept					
		Microcapsule	Sleeve	Separate Loop	Tube/ Shell	Solid-Solid Fins	Intermediary fins
PCM Life	3	3	4	5	4	2	5
Effective k	3	4	3	3	4	5	4
PCM Containment	4	2	4	4	4	2	4
Ease of maintenance	4	2	4	5	3	2	4
Heat Storage	5	3	3	3	3	2	3
Consistent Temperature Regulation	5	5	3	3	4	4	3
Simplicity of system	3	4	3	1	3	2	1
Resistance to ambient conditions	2	3	3	3	3	3	3
Heat transfer rate	4	4	3	3	3	4	4
Weight	3	5	3	1	3	4	1
Volume	3	5	4	1	3	4	1
Cost	2	3	4	2	4	2	3
Fouling	4	2	5	5	5	2	5
	SUM:	**155**	**159**	**141**	**160**	**131**	**146**

B-3. Pugh Matrix

Function: Integration of PCM into the coolant loop

Criteria\Concept	Direct Microcapsule Integration	PCM Sleeve	PCM Macrocapsule Tank	Modified Tube and Shell HX	Solid-Solid Fins	PCM on Electronics
PCM Containment	-	-	D	S	-	-
Corrosivity/Life	-	S	A	S	-	-
Effective K	+	+	T	+	+	+
PCM sustainability	-	S	U	S	-	S
PCM location	-	+	M	+	+	+
Serviceability	-	+		S	-	-
Sum +	1	3	NA	2	2	2
Sum -	5	1	NA	0	4	3
Sum S	0	2	NA	4	0	1

Appendix C. Preliminary Analyses

PDT Thermal model, No PCM

This code approximates the temperature a part of a coolant loop hits as the loop circulates Todd Lundberg 11/7/17 Last modified 11/8/17

```
tic             % Let's see how long this takes to run
% clear all
% close all
% clc
```

Constants

```
Volume = 4.5;                 % total PAO volume in circulation [gallons]
Vdot = 9.5;                 % PAO flowrate [gpm]
Vdots = Vdot/60;            % PAO flowrate [gps]
Tao = Volume/Vdots;         % Period of time it takes to fully circulate ...
                                % PAO through system [seconds]
% Tao = 30;                 % pretend Tao is exactly 30 [seconds]
Cp = .52;                   % specific heat capacity [Btu/lb/F]
PowerH = 50/1.03;                % Power heat [Btu/sec]
PowerL = 3/1.03;                 % Low cycle power [Btu/sec]
Eff = .66;                 % HX effectiveness    [Unitless]
TimeON = 30;                % [seconds] high cycle
TimeOFF = 120;              % [seconds] low cycle
% CpV = Cp*49*(231/1728);     % Volumetric heat capacity [Btu/gal/F]
CpV = 3.48;

endoftime = 1500;           % length of time simulation runs for (seconds)
t = 0:1:endoftime-1;              % make a time scale for plot (seconds)
Pheta = zeros(length(t),1); % initialize Pheta for speed
                            % Pheta is new temperature scale for our
                            % purposes. Zero is set at incoming fuel
                            % temperature, 50 F, so just add fifty to Pheta
                            % to get real temperature
% Tpcm = 18;                  % Temp of PCM (instant change) [deg Pheta]
% Epcm = .5;                  % Effectiveness of PCM HX [Unitless]
```

Run a big 'ol Calc

separate for loops, because Pheta(negative or zero value index) doesn't like to be defined The double heaviside functions, and everything inside them, are just switches to turn on and off the high and low power cycles. This way, there is no limit to the number of cycles than can be run

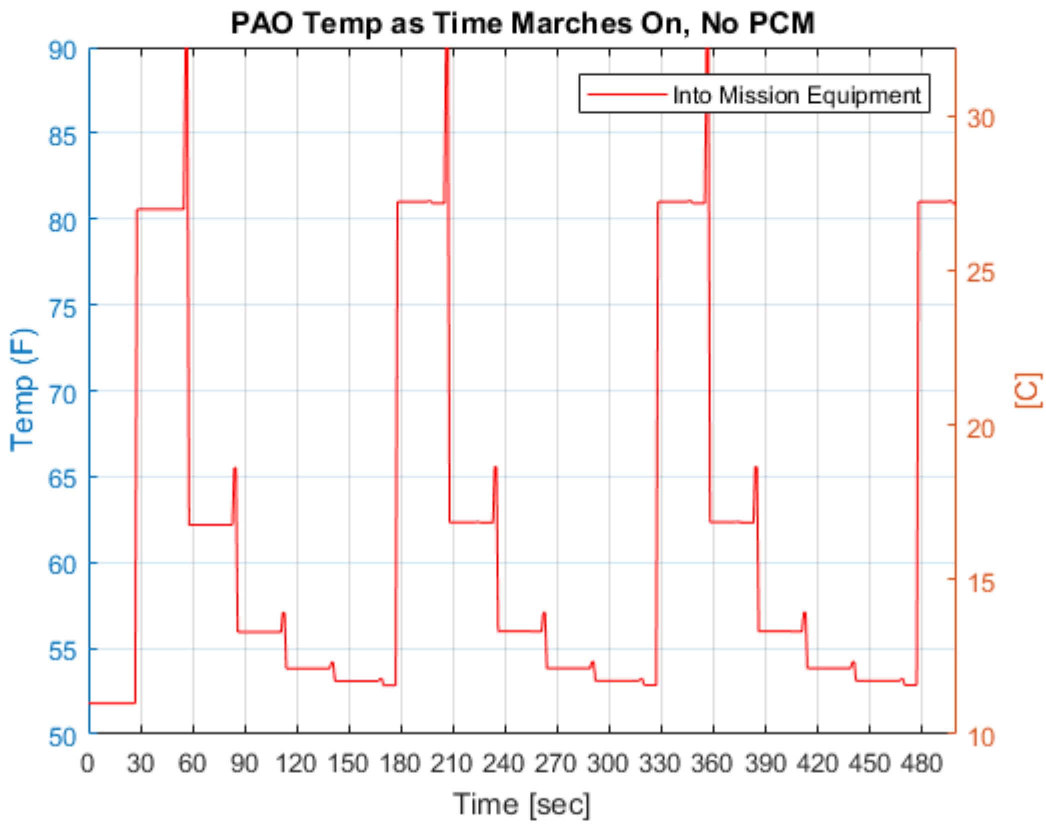
```
for k = 1:floor(Tao)

    Pheta(k) =  (1-Eff)/Vdots/CpV * ...
                (heaviside(heaviside(+real(exp(1i*(2*pi*(k-Tao-(TimeON/2))/(TimeON+TimeOFF))) -
cos(pi*TimeON/(TimeON+TimeOFF))))-.75)*PowerH + ...
                 heaviside(heaviside(-real(exp(1i*(2*pi*(k-Tao-(TimeON/2))/(TimeON+TimeOFF))) -
cos(pi*TimeON/(TimeON+TimeOFF))))-.75)*PowerL);
end


for k = ceil(Tao):endoftime   % ceil(Tao)+2

     Pheta(k) = (1-Eff)*( interp1(Pheta, ceil(k-Tao)) +   1/Vdots/CpV * ...
                (heaviside(heaviside(+real(exp(1i*(2*pi*(k-Tao-(TimeON/2))/(TimeON+TimeOFF))) -
cos(pi*TimeON/(TimeON+TimeOFF))))-.75)*PowerH + ...
                 heaviside(heaviside(-real(exp(1i*(2*pi*(k-Tao-(TimeON/2))/(TimeON+TimeOFF))) -
cos(pi*TimeON/(TimeON+TimeOFF))))-.75)*PowerL));
end

% % this for loop smoothes out curve a little bit, also requires ceil(Tao)+2
% % in above for loop
% for k = ceil(Tao)+3:endoftime
%
%             Pheta(k) = (1-Eff)*( (interp1(Pheta,k-Tao-2) + interp1(Pheta,k-Tao-1) +
interp1(Pheta,k-Tao) + interp1(Pheta,k-Tao+1) + interp1(Pheta,k-Tao+2))/5 + ...
%             1/Vdots/CpV * (heaviside(heaviside(+real(exp(1i*(2*pi*(k-Tao-
(TimeON/2))/(TimeON+TimeOFF))) - cos(pi*TimeON/(TimeON+TimeOFF))))-.75)*PowerH + ...
%                            heaviside(heaviside(-real(exp(1i*(2*pi*(k-Tao-
(TimeON/2))/(TimeON+TimeOFF))) - cos(pi*TimeON/(TimeON+TimeOFF))))-.75)*PowerL));
% end

% Time to figure out other temps
Phethot=Pheta*0;
for k = 1:endoftime-ceil(Tao)
Phethot(k) = Pheta(k+ceil(Tao))/(1-Eff);
end
```

PLOT IT UP

```
figure()
yyaxis left
plot(t,Pheta+50, 'r')%, t,Phethot+50, '-k')
title('PAO Temp as Time Marches On, No PCM')
xlabel('Time [sec]')
ylabel('Temp (F)')
grid on;
axis([0 500 50 90]);
xticks([0 30 60 90 120 150 180 210 240 270 300 330 360 390 420 450 480])
yyaxis right
axis([0 500 10 (90-32)*5/9])
ylabel('[C]')
```

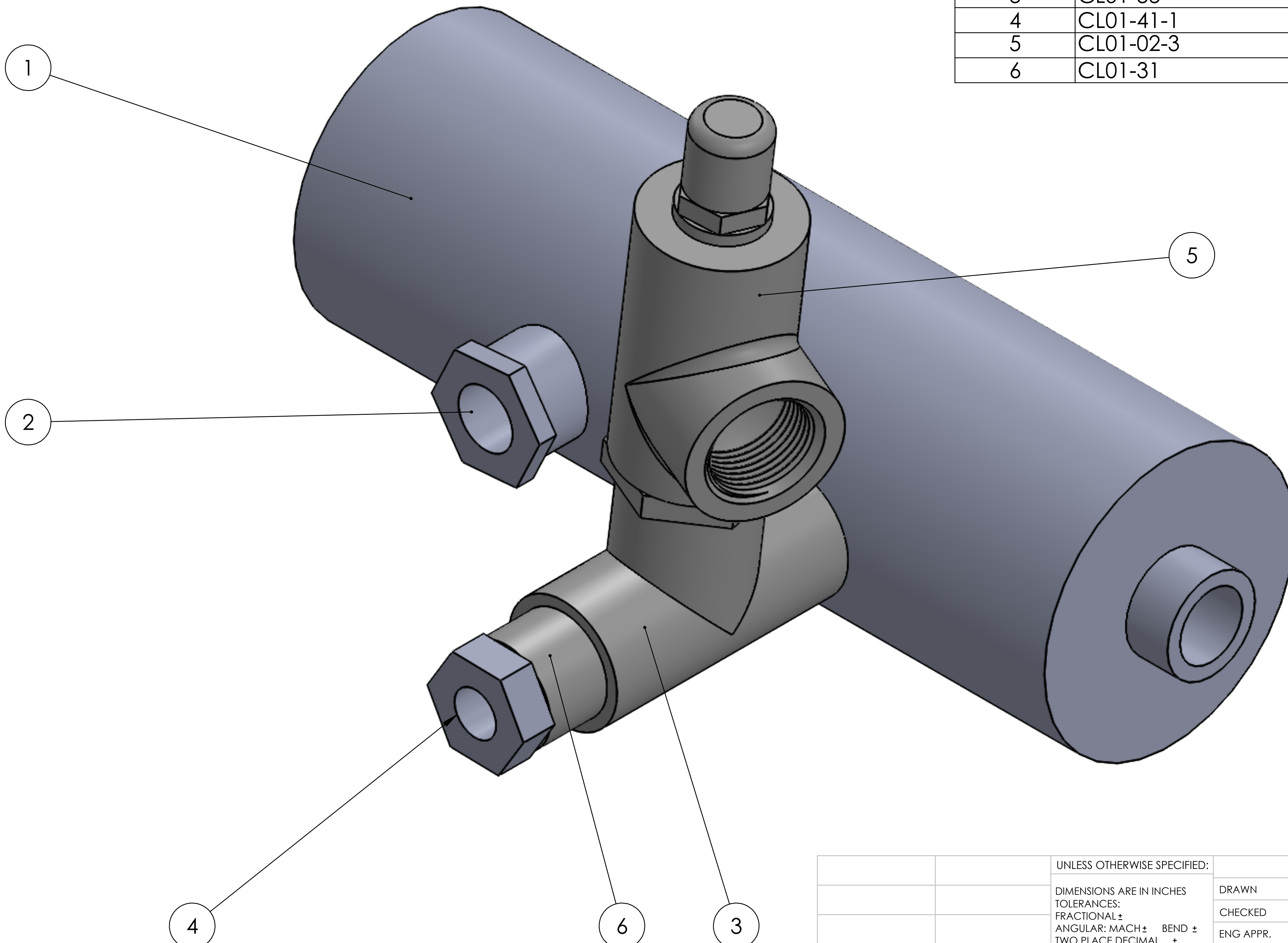
```
legend('Into Mission Equipment', 'Out of Mission Equipment')

toc                     %Let's see how long this took


% figure()
% plot(t,Phethot)
```

```
Warning: Ignoring extra legend entries.
Elapsed time is 5.717202 seconds.
```



APPENDIX D. CONCEPT LAYOUT DRAWINGS

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	CL01-02	PCM Reservoir	1
2	CL01-02-1	Sight	1
3	CL01-33	Tee	1
4	CL01-41-1	Thermowell	1
5	CL01-02-3	Relief Valve (75 psi)	1
6	CL01-31	Connector	1



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APPLICATION

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DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	I.S.	2/9/18
CHECKED	B.J.	2/10/18
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

PCMReservoir

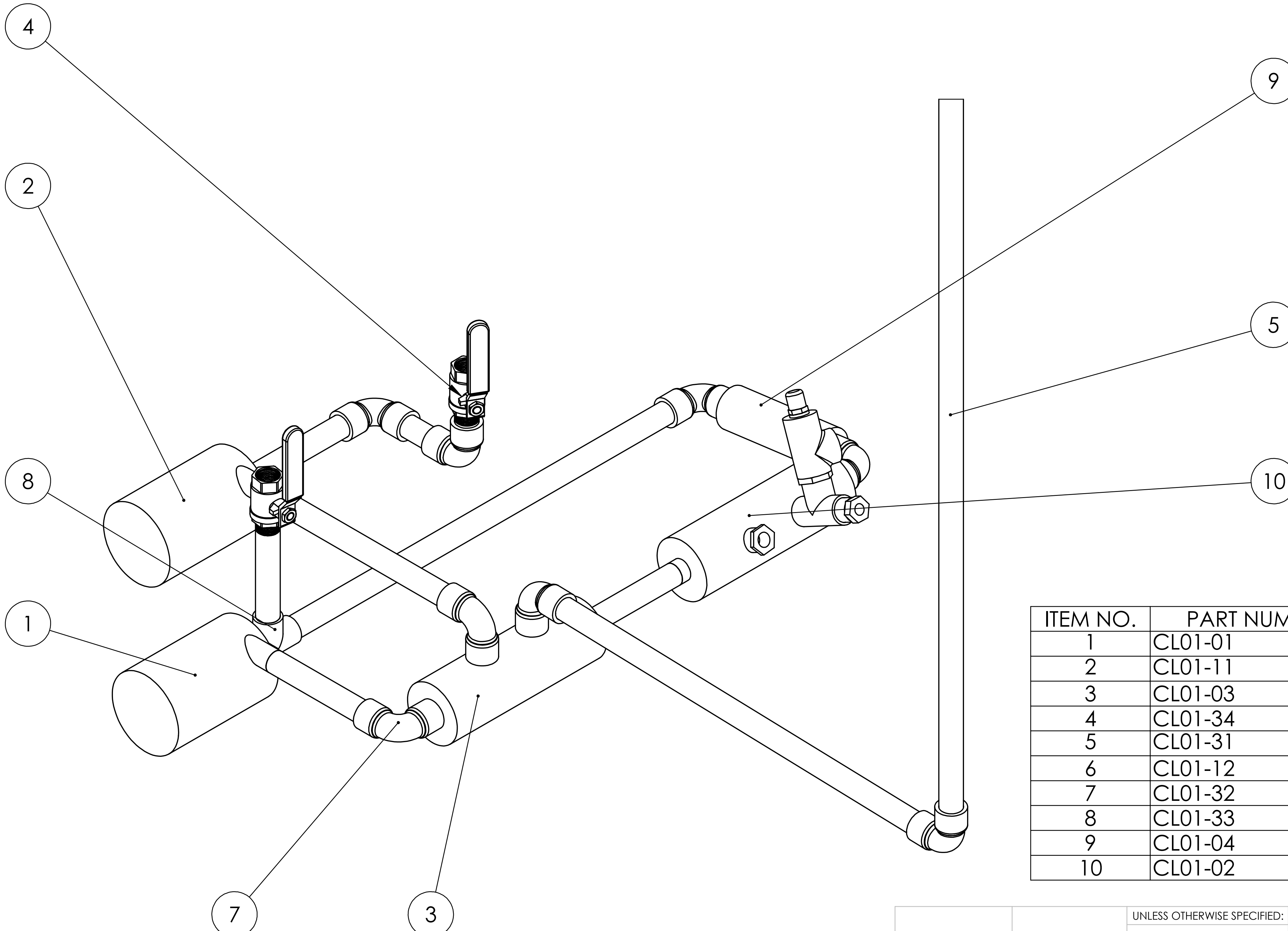
SIZE B | DWG. NO. | REV

SCALE: 1:5 | WEIGHT: | SHEET 1 OF 1

APPENDIX E. COMPLETED DRAWING PACKAGE

Indented Bill of Material (BOM): Transient Heat Cooling Loop Assembly

Assy Level	Part Number	Description	Matl	Vendor	Qty	Cost	Ttl Cost	Status
0	CL01-00	**Sub-scale Model**						
1	CL01-01	Main Pump	------	------	1	$ -	$ -	Received
1	CL01-31	Piping	PVC	McMaster-Carr	2	$ 5.50	$ 11.00	Purchased
1	CL01-32	90° Elbow	PVC	McMaster-Carr	10	$ 3.40	$ 34.00	Purchased
1	CL01-33	Tee	PVC	McMaster-Carr	10	$ 0.44	$ 4.40	Purchased
1	CL01-34	Ball Valve	Brass	McMaster-Carr	2	$ 3.21	$ 6.42	Purchased
1	CL01-03	Heat Exchanger	------	PDT	2	$ 40.00	$ 80.00	Purchased
1	CL01-04	Heater	------	PDT	1	$ -	$ -	Purchased
1	CL01-02	PCM Reservoir						
2	CL01-02-0	PCM Heat Exchanger	------	AIHTI	1	$ 565.75	$ 565.75	Received
2	CL01-02-1	Sight	PVC/ glass	McMaster-Carr	1	$ 7.30	$ 7.30	Canceled
2	CL01-02-2	Cap	PVC	McMaster-Carr	1	$ 0.32	$ 0.32	Purchased
2	CL01-02-3	Relief Valve	Brass	McMaster-Carr	1	$ 81.75	$ 81.75	Canceled
1	CL01-41	Temperatuere Sensors						
2	CL01-41-1	Thermowell	SST 304	McMaster-Carr	4	$ 24.75	$ 99.00	Canceled
2	CL01-41-2	Thermocouple	Type T	McMaster-Carr	1	$ 83.00	$ 83.00	Purchased
2	CL01-41-3	Thermocouple Reader (Omega RDXL6SD)	------	Omega	1	$ 620.00	$ 620.00	Purchased
1	CL01-42	Flow Meter	------	------	2	$ -	$ -	Received
1	CL01-43	Pressure Sensor	------	------	2	$ -	$ -	Received
1	CL01-11	Secondary Pump	------	------	2	$ 33.49	$ 66.98	Purchased
1	CL01-12	Reservoir	Plastic	------	1	$ -	$ -	Received
1	CL01-51	PVC Cement	------	McMaster-Carr	1	$ 4.77	$ 4.77	Purchased
1	CL01-52	PVC Cleaner	------	McMaster-Carr	1	$ 4.87	$ 4.87	Purchased
0	CL01-53	Structural Supports	Wood	Home Depot	1	$ -	$ -	Received
1		**PCMs**						
1	PCM01	Microtek 24	------	Microtek	2	$ 110.00	$ 220.00	Received
1	PCM02	PureTemp 37	------	PureTemp	2	$ 140.00	$ 280.00	Received
1	PCM03	PureTemp 23	------	PureTemp	2	$ 140.00	$ 280.00	Received
1	PCM04	CrodaTherm 29	------	Croda	1	$ -	$ -	Received
						Purchased Parts Total:	**$ 2,449.56**	



NOTES:

1. All piping sections will be cut to length on site for fit.

2. Reservoir is hidden to show component. It can be seen in the isometric view attached

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	CL01-01	Primary Pump	1
2	CL01-11	Secondary Pump	1
3	CL01-03	Tube and Shell HX	1
4	CL01-34	Ball Valve	2
5	CL01-31	PVC Piping	10
6	CL01-12	Reservoir	1
7	CL01-32	Elbow (90)	10
8	CL01-33	Tee	1
9	CL01-04	In-Line Heater	1
10	CL01-02	PCM Reservoir	1

NEXT ASSY | USED ON

APPLICATION

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	I.S.	2/8/18
CHECKED	B.J.	2/9/18
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

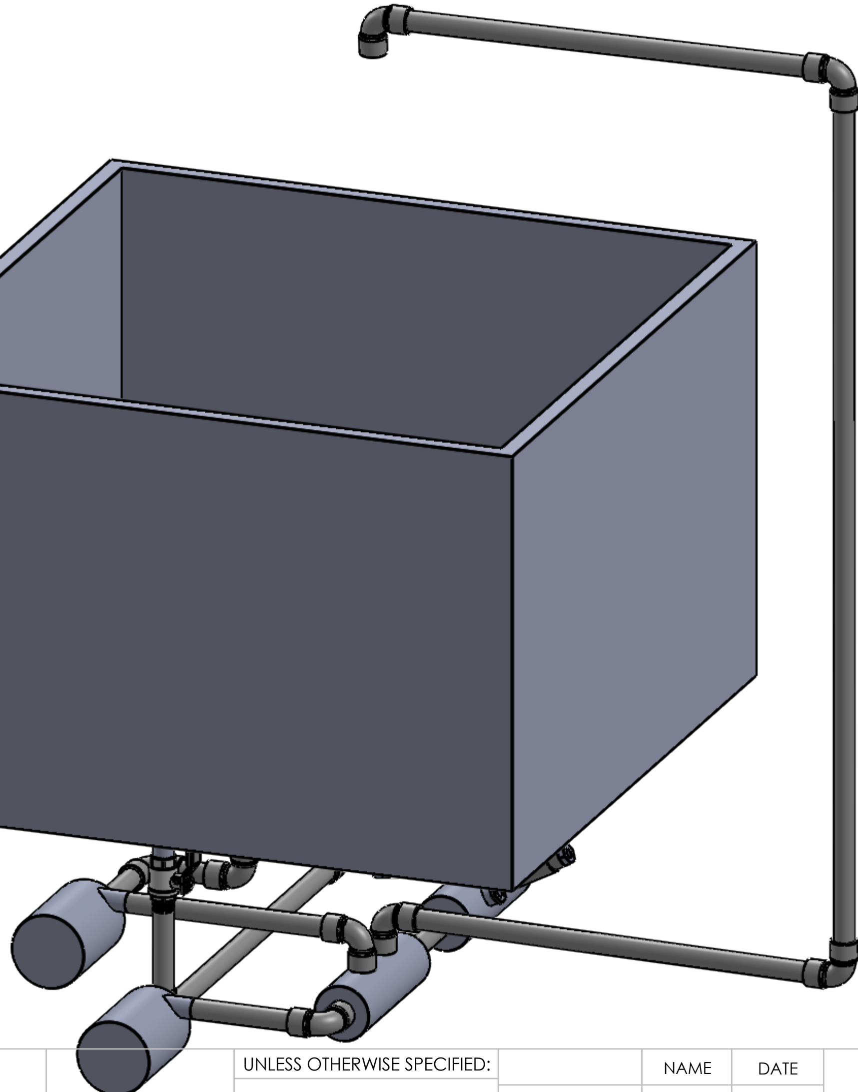
Scale prototype

SIZE **B** | DWG. NO. | REV

SCALE: 1:20 | WEIGHT: | SHEET 1 OF 1

2 1

B B



A A

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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

Scale Prototype

SIZE	DWG. NO.	REV
A	Overview	

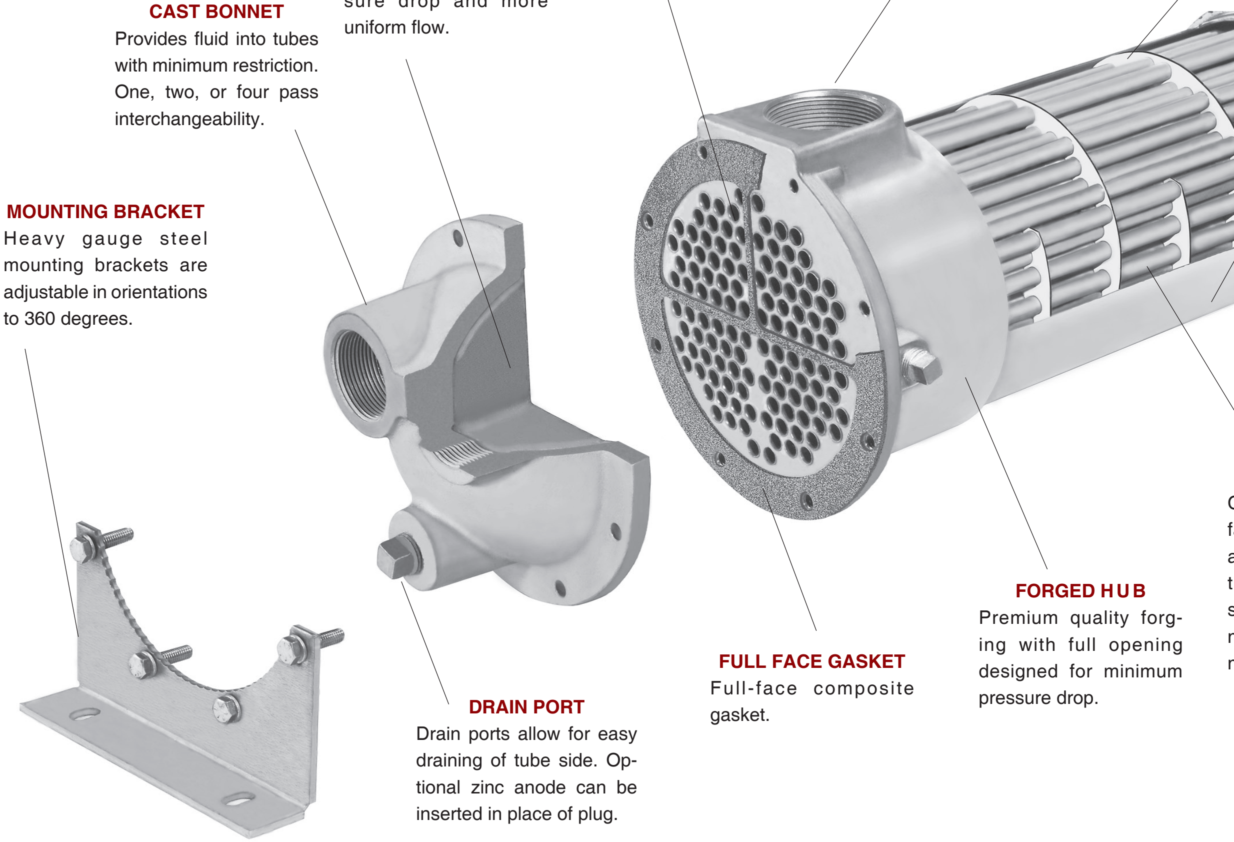
SCALE: 1:50 WEIGHT: SHEET 1 OF 1

2 1

APPENDIX F. PURCHASED PARTS DETAILS

Purchased Parts Details
Transient Heat Cooling Loop Assembly

Assy Level	Part Number	Description (Lvl0)	Description (Lvl1)	Description (Lvl2)	Vendor	Specifications Link
0	CL01-00	**Sub-scale Model**				
1	CL01-01		Main Pump		------	n/a
1	CL01-31		Piping		McMaster-Carr	https://www.mcmaster.com/#48925k92/=1bf0ijv
1	CL01-32		90° Elbow		McMaster-Carr	https://www.mcmaster.com/#4880k589/=1bf1mjm
1	CL01-33		Tee		McMaster-Carr	https://www.mcmaster.com/#4880k42/=1bf1n3j
1	CL01-34		Ball Valve		McMaster-Carr	https://www.mcmaster.com/#47865k44/=1bf0cmu
1	CL01-03		Heat Exchanger		PDT	Data Sheet #1
1	CL01-04		Heater		PDT	n/a
1	CL01-02		PCM Reservoir			
2	CL01-02-0			PCM Heat Exchanger	AIHTI	Data Sheet #1
2	CL01-02-1			Sight	McMaster-Carr	https://www.mcmaster.com/#1210k24/=1bj500o
2	CL01-02-2			Cap	McMaster-Carr	https://www.mcmaster.com/#4880k52/=1bfxsnl
2	CL01-02-3			Relief Valve	McMaster-Carr	https://www.mcmaster.com/#4780k12/=1bj1yvs
1	CL01-41		Temperatuere Sensors			
2	CL01-41-1			Thermowell	McMaster-Carr	https://www.mcmaster.com/#3957k11/=1biotqo
2	CL01-41-2			Thermocouple	McMaster-Carr	https://www.omega.com/techref/colorcodes.html
2	CL01-41-3			Thermocouple Reader (Omega RDXL6SD)	Omega	Data Sheet #2
1	CL01-42		Flow Meter		------	n/a
1	CL01-43		Pressure Sensor		------	n/a
1	CL01-11		Secondary Pump		------	http://www.everflopump.com/pumps/e2gpm.php
1	CL01-12		Reservoir		------	n/a
1	CL01-51		PVC Cement		McMaster-Carr	https://www.mcmaster.com/#74605a14/=1bj56wv
1	CL01-52		PVC Cleaner		McMaster-Carr	https://www.mcmaster.com/#74605a44/=1bj574y
0	CL01-53		Structural Supports		Home Depot	n/a
1		**PCMs**				
1	PCM01		Microtek 24		Microtek	Data Sheet #3
1	PCM02		PureTemp 37		PureTemp	Data Sheet #4
1	PCM03		PureTemp 23		PureTemp	Data Sheet #5
1	PCM04		CrodaTherm 29		Croda	Data Sheet #6



TUBE JOINT
Roller expanded tube joint to integral forged hub.

THREAD
CNC precision threading to provide accurate leak-proof connections.

BAFFLES
CNC manufactured baffles to provide maximum turbulence and heat transfer with a minimum fluid pressure drop.

FLOW CAVITY
Generously sized to allow for minimum pressure drop and more uniform flow.

CAST BONNET
Provides fluid into tubes with minimum restriction. One, two, or four pass interchangeability.

FINISH
Gray semigloss enamel. Can be used as a base for additional coats.

MOUNTING BRACKET
Heavy gauge steel mounting brackets are adjustable in orientations to 360 degrees.

BUNDLE ASSEMBLY
CNC precision manufactured parts to guarantee a close fit between the baffles, tubes, and shell. Clearances are minimized to provide for maximum heat transfer.

FORGED HUB
Premium quality forging with full opening designed for minimum pressure drop.

FULL FACE GASKET
Full-face composite gasket.

DRAIN PORT
Drain ports allow for easy draining of tube side. Optional zinc anode can be inserted in place of plug.

UNIT CODING

Example Model

Options

AB - 1204 - C 4 - TP - CNT - B - Z

Model
AB
SAB
SAE
SSAE
STS
EAB

Shell Diameter
400 = 2.13"
700 = 3.65"
1000 = 5.13"
1200 = 6.13"
1600 = 8.00"

Effective Tube Length
(9" increments)

Baffle Spacing Code
A = 1 1/8"
B = 2 1/4"
C = 4 1/2"
D = 9 "
E = 18 "

Cooling Tube Diamenter
4 = 1/4"
6 = 3/8"
10 = 5/8"

Tube Side Passes
SP = 1 pass
TP = 2 pass
FP = 4 pass

Tubing
Blank = Copper
Options
CNT= 90/10 Cu Ni
STS = Stainless Steel
C = Carbon Steel

End Bonnets
Blank = Cast Iron
Options
B = Bronze

Zinc Anode
Z = 1 Zinc Anode
2Z = 2 Zinc Anode etc.

STANDARD CONSTRUCTION MATERIALS & RATINGS

Standard Model	AB Series	SAB & SSAE Series*	SAE Series	STS Series	EAB Series	Standard Unit Ratings
Shell	Brass	Steel	Brass	316 Stainless Steel	Steel	Operating Pressure
Tubes	Copper	Copper	Copper	316 Stainless Steel	90/10 Copper Nickel	Tubes......150 psig
Baffle	Aluminum	Aluminum	Aluminum	316 Stainless Steel	Aluminum	
Integral End Hub	Forged Brass	Forged Brass	Forged Brass	316 Stainless Steel	Forged Brass	Operating Pressure
End Bonnets	Cast Iron	Cast Iron	Cast Iron	316 Stainless Steel	Cast Iron	Shell.......300 psig
Mounting Brackets	Steel	Steel	Steel	Steel	Steel	
Gasket	Hypalon Composite	Hypalon Composite	Hypalon Composite	Hypalon Composite	High Temp Gasket	Operating Temperature
Expansion Bellows	–	–	–	–	Stainless Steel	300 °F

*Offered in 5" through 8" shell diameter.

note: AIHTI reserves the right to make reasonable design changes without notice.

AB, SAE, STS, & EAB Series *selection*

STEP 1: Calculate the heat load

The heat load in BTU/HR or (Q) can be derived by using several methods. To simplify things, we will consider general specifications for hydraulic system oils and other fluids that are commonly used with shell & tube heat exchangers.

Terms

GPM = Gallons Per Minute
CN = Constant Number for a given fluid
$\triangle T$ = Temperature differential across the potential
PSI = Pounds per Square Inch (pressure) of the operating side of the system
MHP = Horsepower of the electric motor driving the hydraulic pump
Kw = Kilowatt (watts x 1000)
T_{in} = Hot fluid entering temperature in °F
T_{out} = Hot fluid exiting temperature in °F
t_{in} = Cold fluid temperature entering in °F
t_{out} = Cold fluid temperature exiting in °F
Q = BTU / HR

For example purposes, a hydraulic system has a 125 HP (93Kw) electric motor installed coupled to a pump that produces a flow of 80 GPM @ 2500 PSIG. The temperature differential of the oil entering the pump *vs* exiting the system is about 5.3°F. Even though our return line pressure operates below 100 psi, we must calculate the system heat load potential (Q) based upon the prime movers (pump) capability. We can use one of the following equations to accomplish this:

To derive the required heat load (Q) to be removed by the heat exchanger, apply ONE of the following. Note: The calculated heat loads may differ slightly from one formula to the next. This is due to assumptions made when estimating heat removal requirements. The factor *(v)* represents the percentage of the overall input energy to be rejected by the heat exchanger. The *(v)* factor is generally about 30% for most hydraulic systems, however it can range from 20%-70% depending upon the installed system components and heat being generated (ie. servo valves, proportional valves, etc…will increase the percentage required).

Formula	Example
A) Q = GPM x CN x actual $\triangle T$	A) Q = 80 x 210 x 5.3°F = 89,040 BTU/HR
B) Q = [(PSI x GPM) / 1714] x (*v*) x 2545	B) Q = [(2500x80)/1714] x .30 x 2545 = 89,090 BTU/HR
C) Q = MHP x (*v*) x 2545	C) Q = 125 x .30 x 2545 = 95,347 BTU/HR
D) Q = Kw to be removed x 3415	D) Q = 28 x 3415 = 95,620 BTU/HR
E) Q = HP to be removed x 2545	E) Q = 37.5 x 2545 = 95,437 BTU/HR

Constant for a given fluid (CN)

1) Oil ....CN = 210
2) Water.........................CN = 500
3) 50% E. Glycol...........CN = 450

STEP 2: Calculate the Mean Temperature Difference

When calculating the MTD you will be required to choose a liquid flow rate to derive the cold side $\triangle T$. If your water flow is unknown you may need to assume a number based on what is available. As a normal rule of thumb, for oil to water cooling a 2:1 oil to water ratio is used. For applications of water to water or 50 % Ethylene Glycol to water, a 1:1 ratio is common.

Formula

$$\textbf{HOT FLUID } \triangle T \textbf{ (Oil)} = \frac{Q}{CN \times GPM}$$

$$\textbf{COLD FLUID } \triangle t \textbf{ (Water)} = \frac{BTU / hr}{CN \times GPM}$$

Example

$$\triangle T = \frac{89{,}090 \text{ BTU/hr (from step 1, example B)}}{210 \text{ CN} \times 80\text{GPM}} = 5.3°F = \triangle T \text{ Rejected}$$

$$\triangle t = \frac{89{,}090 \text{ BTU/hr}}{500 \text{ CN} \times 40\text{GPM (for a 2:1 ratio)}} = 4.5°F = \triangle t \text{ Absorbed}$$

T_{in} = Hot Fluid entering temperature in degrees F
T_{out} = Hot Fluid exiting temperature in degrees F
t_{in} = Cold Fluid entering temperature in degrees F
t_{out} = Cold Fluid exiting temperature in degrees F

T_{in} = 125.3 °F
T_{out} = 120.0 °F
t_{in} = 70.0 °F
t_{out} = 74.5 °F

$$\frac{T_{out} - t_{in}}{T_{in} - t_{out}} = \frac{S \text{[smaller temperature difference]}}{L \text{ [larger temperature difference]}} = \left(\frac{S}{L}\right)$$

$$\frac{120.0°F - 70.0°F = 50.0°F}{125.3°F - 74.5°F = 50.8°F} = \frac{50.0°F}{50.8°F} = .984$$

STEP 3: Calculate Log Mean Temperature Difference (LMTD)

To calculate the LMTD please use the following method;

L = Larger temperature difference from step 2.
M = S/L number (LOCATED IN TABLE A).

$LMTD_i$ = L x M $LMTD_i$ = 50.8 x .992 (FROM TABLE A) = 50.39

To correct the $LMTD_i$ for a multipass heat exchangers calculate **R** & **K** as follows:

Formula

$$R = \frac{T_{in} - T_{out}}{t_{out} - t_{in}} \qquad K = \frac{t_{out} - t_{in}}{T_{in} - t_{in}}$$

Example

$$R = \frac{125.3°F - 120°F}{74.5°F - 70°F} = \frac{5.3°F}{4.5°F} = \{1.17=R\}$$

$$K = \frac{74.5°F - 70°F}{124.5°F - 70°F} = \frac{4.5°F}{55.4°F} = \{0.081=K\}$$

Locate the correction factor CF_B (FROM TABLE B)
$LMTD_c = LMTD_i \times CF_B$
$LMTD_c$ = 50.39 x 1 = **50.39**

note: AIHTI reserves the right to make reasonable design changes without notice.

355 American Industrial Drive LaCrosse, VA 23950 tel: 434-757-1800 fax: 434-757-1810 email: sales@aihti.com

STEP 4: Calculate the area required

$$\textbf{Required Area sq.ft.} = \frac{Q\ (BTU/HR)}{LMTD_c \times U\ (\text{FROM TABLE C})} \qquad \frac{89{,}090}{50.39 \times 100} = \textbf{17.68 sq.ft.}$$

STEP 5: Selection

a) From TABLE E choose the correct series size, baffle spacing, and number of passes that best fits your flow rates for both shell and tube side. Note that the tables suggest minimum and maximum information. Try to stay within the 20-80 percent range of the indicated numbers.

Example

Oil Flow Rate = 80 GPM = Series Required from Table E = **1200 Series**
Baffle Spacing from Table E = **C baffle**
Water Flow Rate = 40 GPM = Passes required in 1200 series = **4 (FP)**

b) From TABLE D choose the heat exchanger model size based upon the sq.ft. or surface area in the series size that will accommodate your flow rate.

Example

Required Area = 17.68sq.ft Closest model required based upon sq.ft. & series= **AB-1202-C6-FP**

If you require a computer generated data sheet for the application, or if the information that you are trying to apply does not match the corresponding information, please contact our engineering services department for further assistance.

TABLE A- FACTOR M/LMTD = L x M

S/L	M	S/L	M	S/L	M	S/L	M
		.25	.541	.50	.721	.75	.870
.01	.215	.26	.549	.51	.728	.76	.864
.02	.251	.27	.558	.52	.734	.77	.879
.03	.277	.28	.566	.53	.740	.78	.886
.04	.298	.29	.574	.54	.746	.79	.890
.05	.317	.30	.582	.55	.753	.80	.896
.06	.334	.31	.589	.56	.759	.81	.902
.07	.350	.32	.597	.57	.765	.82	.907
.08	.364	.33	.604	.58	.771	.83	.913
.09	.378	.34	.612	.59	.777	.84	.918
.10	.391	.35	.619	.60	.783	.85	.923
.11	.403	.36	.626	.61	.789	.86	.928
.12	.415	.37	.634	.62	.795	.87	.934
.13	.427	.38	.641	.63	.801	.88	.939
.14	.438	.39	.648	.64	.806	.89	.944
.15	.448	.40	.655	.65	.813	.90	.949
.16	.458	.41	.662	.66	.818	.91	.955
.17	.469	.42	.669	.67	.823	.92	.959
.18	.478	.43	.675	.68	.829	.93	.964
.19	.488	.44	.682	.69	.836	.94	.970
.20	.497	.45	.689	.70	.840	.95	.975
.21	.506	.46	.695	.71	.848	.96	.979
.22	.515	.47	.702	.72	.852	.97	.986
.23	.524	.48	.709	.73	.658	.98	.991
.24	.533	.49	.715	.74	.864	.99	.995

TABLE D- Surface Area

Model Number	Surface Area in Sq.ft. 1/4" O.D Tubing	3/8" O.D Tubing	5/8 O.D Tubing	Model Number	Surface Area in Sq.ft. 1/4" O.D Tubing	3/8" O.D Tubing	5/8 O.D Tubing
AB-401	1.4	–	–	AB-1602	44.4	30.3	17.6
AB-402	3.0	–	–	AB-1603	66.3	45.3	26.5
AB-403	4.6	–	–	AB-1604	88.3	60.3	35.3
				AB-1605	110.3	75.6	44.1
AB-701	3.6	2.6	–	AB-1606	132.3	90.4	53.0
AB-702	7.3	5.2	–	AB-1607	154.3	105.4	61.8
AB-703	11.1	7.9	–	AB-1608	176.3	120.4	70.6
AB-704	14.9	10.6	–	AB-1609	197.9	135.2	79.5
AB-705	18.7	13.3	–	AB-1610	219.9	150.2	88.3
				AB-1611	241.9	165.2	97.1
AB-1002	17.7	11.2	5.9	AB-1612	263.9	180.2	105.9
AB-1003	26.5	16.8	8.8	AB-1613	285.9	195.2	114.7
AB-1004	35.4	22.4	11.8				
AB-1005	44.3	28.0	14.7	AB-2004	155.1	110.7	60.8
AB-1006	53.2	33.6	17.6	AB-2005	193.8	138.4	76.1
				AB-2006	232.6	166.1	91.3
AB-1202	25.5	17.9	8.8	AB-2007	271.4	193.8	106.5
AB-1203	38.0	26.7	13.2	AB-2008	310.2	221.4	121.7
AB-1204	50.3	35.4	17.6	AB-2009	349.0	249.1	137.0
AB-1205	63.0	44.2	22.1	AB-2010	387.7	276.8	152.2
AB-1206	75.6	53.2	26.5	AB-2011	426.5	304.5	167.4
AB-1207	88.2	62.0	30.9	AB-2012	465.3	332.2	182.7
AB-1208	100.6	70.7	35.3	AB-2013	504.1	359.9	197.9
AB-1209	113.0	79.4	39.6	AB-2014	542.9	387.6	213.2
AB-1210	125.4	88.1	44.1	AB-2015	581.7	415.3	228.4

LMTD correction factor for Multipass Exchangers

R

	.05	.1	.15	.2	.25	.3	.35	.4	.45	.5	.6	.7	.8	.9	1.0
.2	1	1	1	1	1	1	1	.999	.993	.984	.972	.942	.908	.845	.71
.4	1	1	1	1	1	1	.994	.983	.971	.959	.922	.855	.70		
.6	1	1	1	1	1	.992	.980	.965	.948	.923	.840				
.8	1	1	1	1	.995	.981	.965	.945	.916	.872					
1.0	1	1	1	1	.988	.970	.949	.918	.867	.770					
2.0	1	1	.977	.973	.940	.845	.740								
3.0	1	1	.997	.933	.835										
4.0	1	.993	.950	.850											
5.0	1	.982	.917												
6.0	1	.968	.885												
8.0	1	.930													
10.0	.996	.880													
12.0	.985	.720													
14.0	.972														
16.0	.958														
18.0	.940														
20.0	.915														

K

TABLE E- Flow Rate for Shell & Tube

Shell dia . Code	Max. Liquid Flow - Shell Side Baffle Spacing A	B	C	D	E	Liquid Flow - Tube Side SP Min.	SP Max.	TP Min.	TP Max.	FP Min.	FP Max.
400	10	15	20	–	–	3.5	21	–	–	–	–
700	17	29	30	35	–	9	61	4.5	30	2.2	15
1000	24	48	68	70	–	20	120	10	70	5.0	37
1200	29	56	105	115	120	30	250	15	112	7.5	56
1600	38	70	150	200	220	57	460	29	180	14	90
2000	–	–	190	370	550	90	650	45	320	25	160

TABLE C

U	TUBE FLUID	SHELL FLUID
400	Water	Water
350	Water	50% E. Glycol
100	Water	Oil
300	50% E. Glycol	50% E. Glycol
90	50% E. Glycol	Oil

note: AIHTI reserves the right to make reasonable design changes without notice.

355 American Industrial Drive LaCrosse, VA 23950 tel: 434-757-1800 fax: 434-757-1810 email: sales@aihti.com

AB, SAE, STS, & EAB Series *performance*

Instructions

The selection chart provided contains an array of popular sizes for quick sizing. It does not provide curves for all models available. Refer to page 4 & 5 for detailed calculation information.

Computer selection data sheets for standard or special models are available through the engineering department of American Industrial. To use the followings graphs correctly, refer to the instruction notes "1-5".

1) HP Curves are based upon a 40°F approach temperature; for example: oil leaving a cooler at 125°F, using 85°F cooling water (125°F – 85°F = 40°F).

2) The oil to water ratio of 1:1 or 2:1 means that for every 1 gallon of oil circulated, a minimum of 1 or 1/2 gallon (respectively) of 85°F water must be circulated to match the curve results.

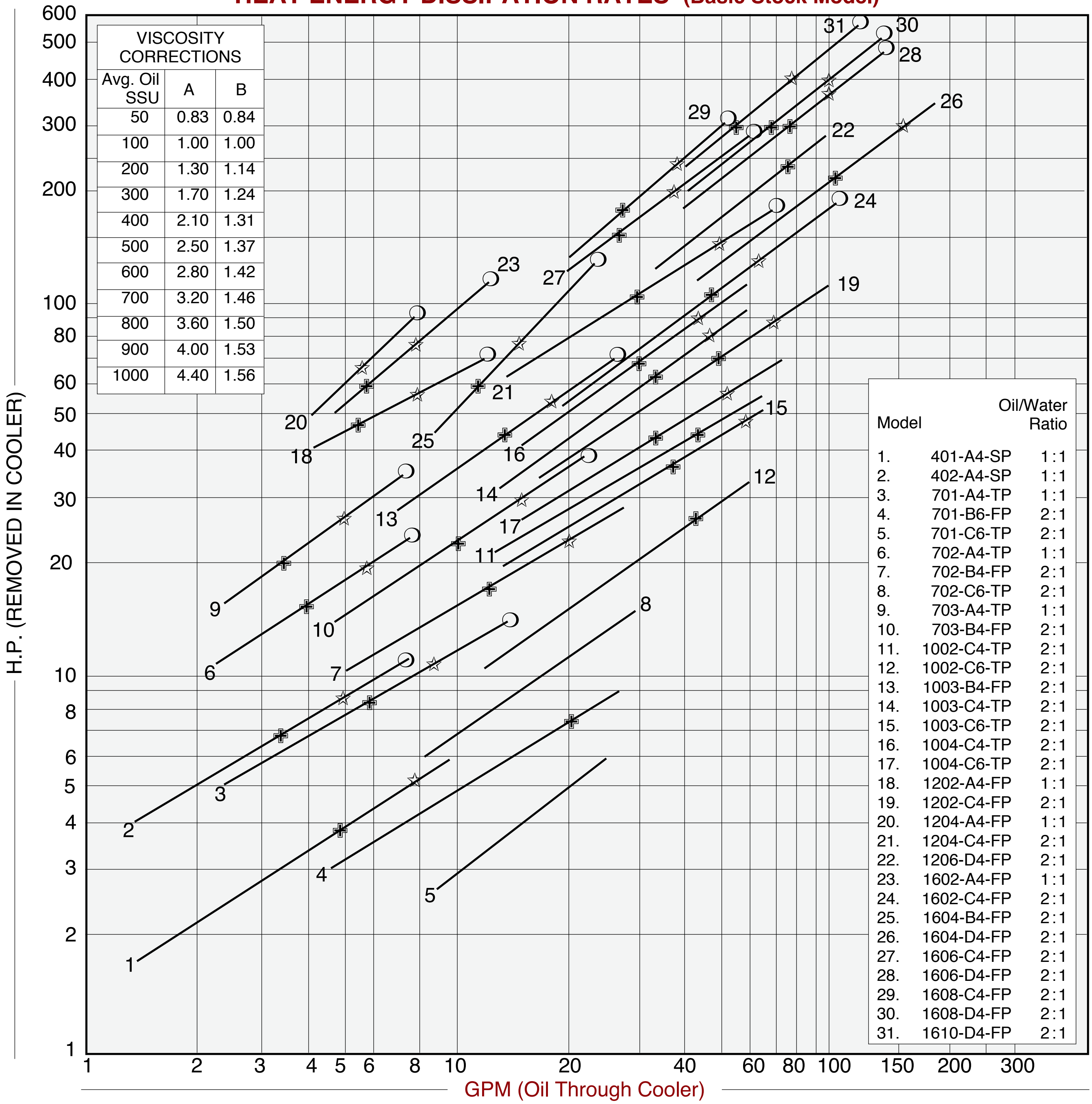
3) OIL PRESSURE DROP CODING: ✚ = 5 psi; ☆= 10 psi; ❍ = 20 psi; △ = 50psi. Curves that have no pressure drop code symbols indicate that the oil pressure drop is less than 5 psi for the flow rate shown.

4) Pressure Drop is based upon oil with an average viscosity of 100 SSU. If the average oil viscosity is other than 100 SSU, then multiply the indicated Pressure Drop by the corresponding value from corrections table A.

5) Corrections for approach temperature and oil viscosity are as follows:

$$\text{H.P.}\left(^{\text{Removed}}_{\text{In Cooler}}\right) = \text{H.P.}\left(^{\text{Actual}}_{\text{Heat Load}}\right) \times \left(\frac{40}{\text{Actual Approach}}\right) \times \text{B.}$$

HEAT ENERGY DISSIPATION RATES (Basic Stock Model)



VISCOSITY CORRECTIONS		
Avg. Oil SSU	A	B
50	0.83	0.84
100	1.00	1.00
200	1.30	1.14
300	1.70	1.24
400	2.10	1.31
500	2.50	1.37
600	2.80	1.42
700	3.20	1.46
800	3.60	1.50
900	4.00	1.53
1000	4.40	1.56

Model		Oil/Water Ratio
1.	401-A4-SP	1:1
2.	402-A4-SP	1:1
3.	701-A4-TP	1:1
4.	701-B6-FP	2:1
5.	701-C6-TP	2:1
6.	702-A4-TP	1:1
7.	702-B4-FP	2:1
8.	702-C6-TP	2:1
9.	703-A4-TP	1:1
10.	703-B4-FP	2:1
11.	1002-C4-TP	2:1
12.	1002-C6-TP	2:1
13.	1003-B4-FP	2:1
14.	1003-C4-TP	2:1
15.	1003-C6-TP	2:1
16.	1004-C4-TP	2:1
17.	1004-C6-TP	2:1
18.	1202-A4-FP	1:1
19.	1202-C4-FP	2:1
20.	1204-A4-FP	1:1
21.	1204-C4-FP	2:1
22.	1206-D4-FP	2:1
23.	1602-A4-FP	1:1
24.	1602-C4-FP	2:1
25.	1604-B4-FP	2:1
26.	1604-D4-FP	2:1
27.	1606-C4-FP	2:1
28.	1606-D4-FP	2:1
29.	1608-C4-FP	2:1
30.	1608-D4-FP	2:1
31.	1610-D4-FP	2:1

note: AIHTI reserves the right to make reasonable design changes without notice.

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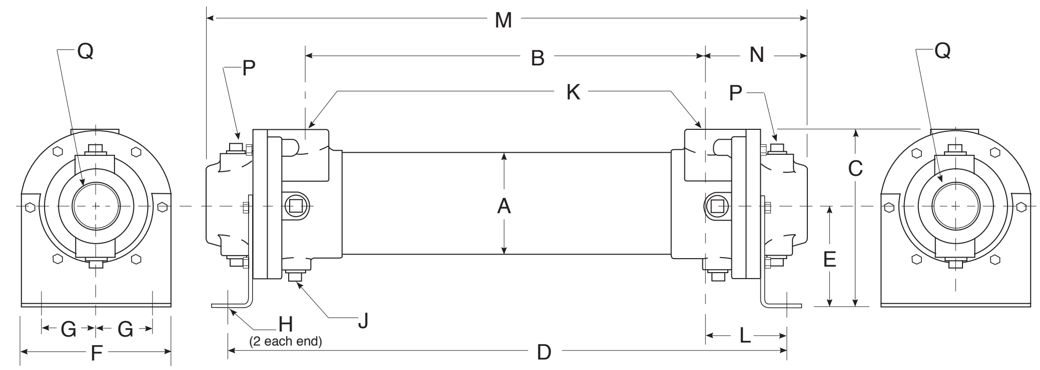
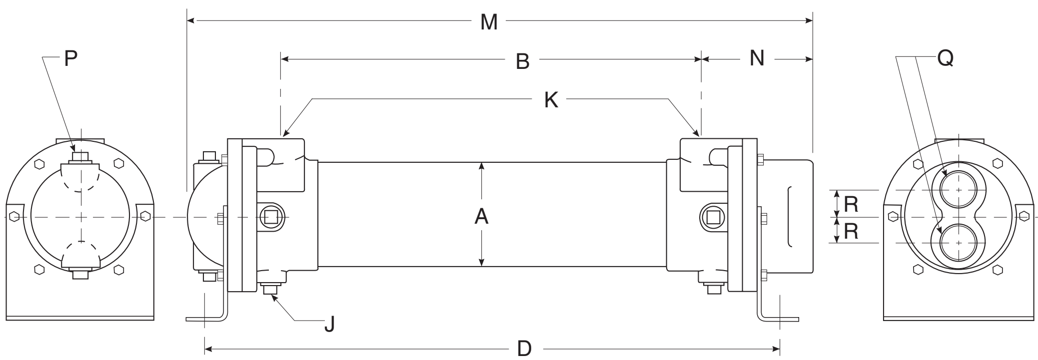


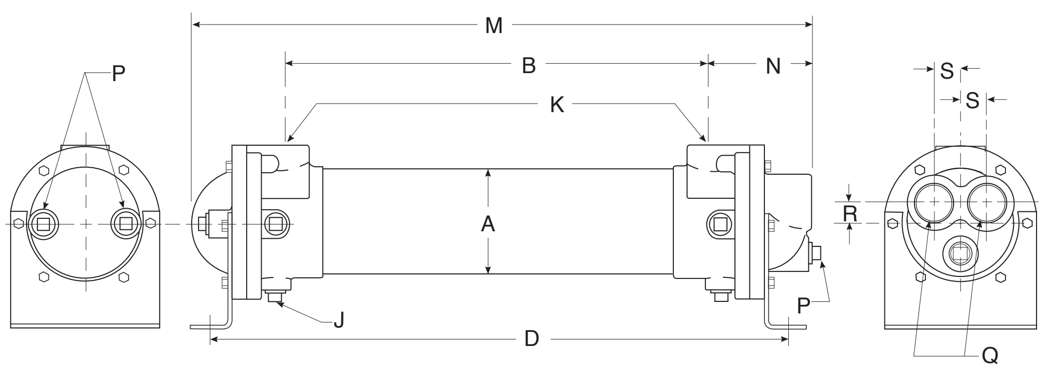
SINGLE PASS (SP)

Model	M	N	P NPT	Q NPT
AB-401	11.24	1.81	-	1.00
AB-402	20.24			
AB-701	13.47	3.24	(4) .38	1.50
AB-702	22.47			
AB-703	31.47			
AB-704	40.47			
AB-1002	23.60	4.05	(4) .38	2.00
AB-1003	32.60			
AB-1004	41.60			
AB-1202	24.38	4.88	(4) .50	3.00
AB-1203	33.25			
AB-1204	42.12			
AB-1205	51.12			
AB-1206	60.25			
AB-1207	69.25			
AB-1208	78.12			
AB-1209	87.12			
AB-1210	96.12			
AB-1602	26.62	6.52	(4) .50	4.00
AB-1603	35.62			
AB-1604	44.62			
AB-1605	53.62			
AB-1606	62.62			
AB-1607	71.62			
AB-1608	80.62			
AB-1609	89.62			
AB-1610	98.62			



TWO PASS (TP)

Model	M	N	P NPT	Q NPT	R
AB-701	13.28	3.30	(2) .38	1.00	.88
AB-702	22.28				
AB-703	31.28				
AB-704	40.28				
AB-1002	23.29	3.80	(2) .38	1.50	1.19
AB-1003	32.29				
AB-1004	41.29				
AB-1202	23.94	4.56	(2) .50	2.00	1.44
AB-1203	32.81				
AB-1204	41.69				
AB-1205	50.69				
AB-1206	59.81				
AB-1207	68.81				
AB-1208	77.69				
AB-1209	86.69				
AB-1210	95.69				
AB-1602	25.10	6.08	(2) .50	2.50	1.88
AB-1603	34.10				
AB-1604	43.10				
AB-1605	52.10				
AB-1606	61.10				
AB-1607	70.10				
AB-1608	79.10				
AB-1609	88.10				
AB-1610	97.10				



FOUR PASS (FP)

Model	M	N	P NPT	Q NPT	R	S
AB-701	13.42	3.24	(3) .38	.75	.62	.88
AB-702	22.42					
AB-703	31.42					
AB-704	40.42					
AB-1002	23.55	4.06	(3) .38	1.00	.75	1.19
AB-1003	32.55					
AB-1004	41.55					
AB-1202	24.44	4.90	(3) .50	1.50	1.06	1.44
AB-1203	33.31					
AB-1204	42.19					
AB-1205	51.19					
AB-1206	60.31					
AB-1207	69.31					
AB-1208	78.19					
AB-1209	87.19					
AB-1210	96.19					
AB-1602	26.72	6.48	(3) .50	2.00	1.38	1.88
AB-1603	35.72					
AB-1604	44.72					
AB-1605	53.72					
AB-1606	62.72					
AB-1607	71.72					
AB-1608	80.72					
AB-1609	89.72					
AB-1610	98.72					

COMMON DIMENSIONS & WEIGHTS

Model	A	B	C	D	E	F	G	H	J NPT	K NPT	L	Approx. Weight	Model
AB-401	2.13	7.62	3.50	10.91	1.94	2.62	.88	.41φ	-	.50	1.72	7	AB-401
AB-402		16.62		20.91								10	AB-402
AB-701	3.66	7.00	6.25	12.38	3.62	5.25	1.50	.44φ x 1.00	(2) .38	1.00	2.69	23	AB-701
AB-702		16.00		21.38								29	AB-702
AB-703		25.00		30.38								33	AB-703
AB-704		34.00		39.38								49	AB-704
AB-1002	5.13	15.50	7.38	21.62	4.00	6.75	2.00	.44φ x 1.00	(6) .38	1.50	3.06	54	AB-1002
AB-1003		24.50		30.62								76	AB-1003
AB-1004		33.50		39.62								82	AB-1004
AB-1202	6.13	14.62	8.81	21.50	4.75	7.50	2.50	.44φ x 1.00	(6) .38	2.00	3.44	79	AB-1202
AB-1203		23.50		30.38								98	AB-1203
AB-1204		32.38		39.25								115	AB-1204
AB-1205		41.38		48.25								130	AB-1205
AB-1206		50.50		57.38								150	AB-1206
AB-1207		59.50		66.38								170	AB-1207
AB-1208		68.38		75.25								190	AB-1208
AB-1209		77.38		84.25								210	AB-1209
AB-1210		86.38		93.25								230	AB-1210
AB-1602	8.00	13.60	12.13	22.38	6.50	10.00	3.50	.44φ x 1.00	(6) .38	3.00	4.39	145	AB-1602
AB-1603		22.60		31.38								170	AB-1603
AB-1604		31.60		40.38								200	AB-1604
AB-1605		40.60		49.38								225	AB-1605
AB-1606		49.60		58.38								250	AB-1606
AB-1607		58.60		67.38								275	AB-1607
AB-1608		67.60		76.38								315	AB-1608
AB-1609		76.60		85.38								350	AB-1609
AB-1610		85.60		94.38								390	AB-1610

note: AIHTI reserves the right to make reasonable design changes without notice.

Six Channel Handheld Temperature Data Logger

With Touch Screen

RDXL6SD

Optional

- **Four Type J, K, T, E, R, S or N Thermocouple Inputs and Two 3-Wire Pt100 RTD Inputs (RTD Connectors are Included)**
- **Display and Log All 6 Channels Simultaneously or Individually**
- **Scheduled and Manual Logging Start/Stop**
- **4 GB SD Card Stores Up to 1 Year of Data**
- **Display Maximum, Minimum, Average and Standard Deviation**
- **Display Temperature Difference Between Any 2 Channels**
- **Battery or Mains Powered (With Adaptor)**
- **Touch Screen with Backlight**
- **2-Channel Temperature Chart**
- **Alarm Indication for Each Channel**

Four Type K Thermocouples included.

RDXL6SD shown smaller than actual size.

The RDXL6SD is a six-channel touch screen data logger that accepts four thermocouple and two RTD inputs. Each channel has individually configurable HI/LO alarms. The intuitive touch screen interface allows each channel to be configured separately for input type, math, alarm and display options. All six channels can be displayed and logged simultaneously or individually. Data can be logged as fast as once a second and is strored on the included 4 GB SD card in Excel® compatible format.

Specifications

Inputs: 4x thermocouple inputs (any of the following types), for use with miniature thermocouple connectors, and 2 x 3-wire Pt100 inputs, for use with 3-pole stereo 3.5 mm (0.14") connectors (see below)

Input Temperature Range:
- **Type J:** -200 to 1200°C
- **Type K:** -200 to 1372°C
- **Type T:** -200 to 400°C
- **Type R:** 0 to 1768°C
- **Type S:** 0 to 1768°C
- **Type N:** 0 to 1300°C
- **Type E:** -200 to 1000°C
- **RTD:** -200 to 850°C

Temperature Accuracy:
- **For Thermocouples:** ±0.1% or 0.8°C
- **For RTD:** ±1.0% or 1.0°C

Temperature Resolution: 0.1° for temperatures below 1000°C or °F, 1° for temperatures above 1000°C or °F

Display: 72 mm (2.83") resistive touch TFT, 320 x 240 pixels, backlit

Configurable Parameters: Temperature units, alarms, signal processing, date and time, data logging, power options, graph channels

Temperature Units: °F or °C

Alarm Configuration: 12 x alarms (2 per channel) with adjustable level, individually configurable as HI or LO

Signal Processing: Average, minimum, maximum, standard deviation, 2-channel temperature difference

Display Response Time: 1 second

Operating Temperature: 0 to 50°C (32 to 122°F)

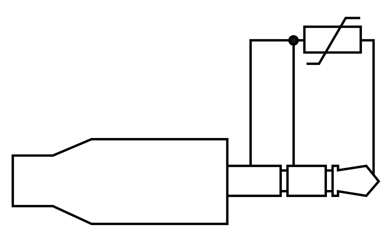
Power Supply: Two "AA" batteries, 5 Vdc adaptor (included)

Current Draw: 250 mA while logging with full display brightness; 60 mA while logging in power saving mode

Weight: 180 g (6.3 oz) without batteries

Dimensions: 136 W x 71 H x 32 D mm (5.35 x 2.8 x 1.25")

Pt100 Connectors (RTD): Use stereo (3-pole) 3.5 mm (0.14") connectors, wired as follows:



Data Logging Specifications

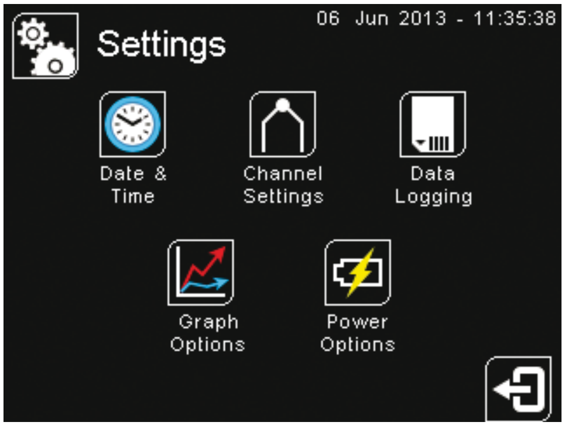
Data Logging Interval: 1 to 86,400 seconds (1 day)

Maximum SD Card Capacity: 4 GB SD Card (included)

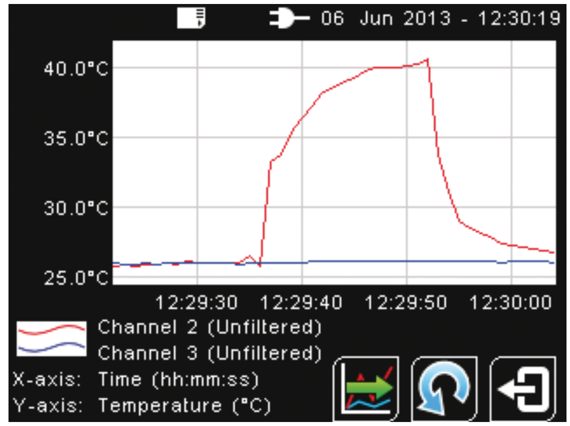
Variables Logged: Measured temperature, cold junction temperature, alarm events

File Format: .csv (can be imported to Excel)

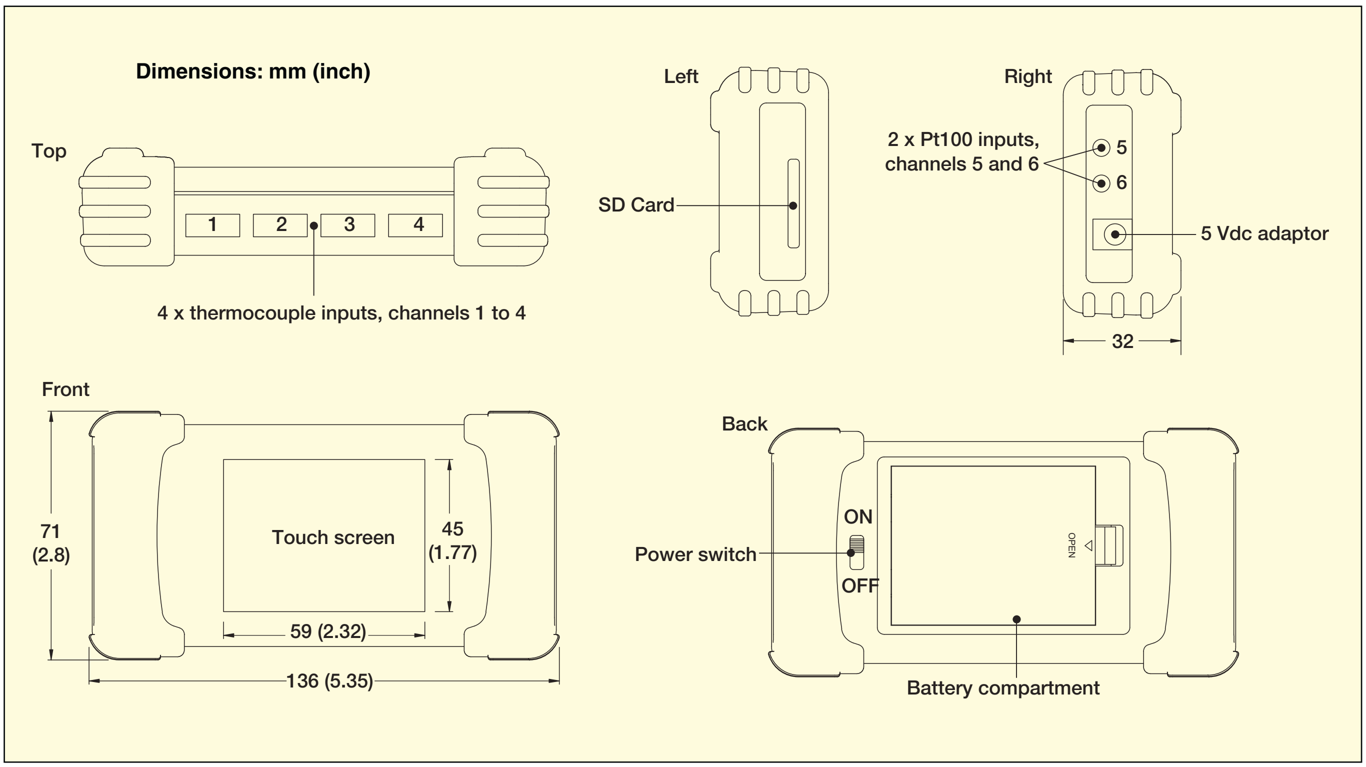
Configurable Parameters: Sample period, number of samples, scheduled start date and time



Intuitive Touch Screen Interface
Fully configurable, no additional hardware is required for configuration. Each channel is configurable separately for input type, maths, alarms and display options.



Temperature Graph
Two selectable channels are displayed. Scroll back in time to view chart history and data logging points.



To Order

Model No.	Description
RDXL6SD	6-channel handheld temperature data logger
RDXL6SD-CAL-3-HH	6-channel handheld temperature data logger with NIST traceable calibration certificate with points

Accessories

Model No.	Description
RDXL6-VLMA	Spare 5 Vdc power adaptor
RDXL6SD-CONN	Spare RTD connector

Comes complete with operator's manual, 5 Vdc adaptor, two "AA" batteries, four beaded wire Type K thermocouples, two RTD connectors and 4 GB SD card.

2400 E. River Rd.
Dayton, Ohio 45439

Tel: 937.236.2213 • Fax: 937.236.2217
e-mail: microtek@microteklabs.com
www.microteklabs.com

MPCM 24D
Microencapsulated Phase Change Material
Phase Change: 24°C, 75.2°F

DESCRIPTION
Microencapsulated phase change materials (MicroPCMs) are very small bi-component particles consisting of a core material, the PCM, and an outer shell or capsule wall. PCMs are low melting materials with melt points in the range of -30°C to 70°C that can absorb and release large amounts of heat. The capsule wall is an inert, stable polymer or plastic.

APPLICATIONS
Microencapsulated PCMs are used to regulate temperatures and for heat storage in a variety of applications.

A primary use of the microPCM products is in the coating of fabrics and foams for the textile industry. The coated materials have broad applications for use in various wearing apparel such as inner and outer garments, gloves and footwear. These end-use products containing microPCMs work by absorbing the body's excess heat, storing that heat, and releasing it back to the body as needed.

Microencapsulated PCMs are also finding widespread use in several other applicaton areas, including in:

- **Electronics** - for cooling electrical components in computers, increasing duty cycles in lasers, and helping maintain constant temperatures for scientific instrumentation and military equipment used in the field.
- **Building Materials** – to increase the energy efficiency of residential and commercial buildings. The materials are being used in combination with radiant heat and solar energy to extend the heating and cooling efficiencies of these systems. PCMs are also being incorporated in plasters, fiberboards, tiles, and insulation.
- **Storage Solutions** – to protect food, beverages, medical products, and temperature-sensitive chemicals in transit.

PROPERTIES
The MPCM 24D product exhibits the following general properties:

Typical Properties	
Appearance	White to slightly off-white color
Form	Dry Powder (≥97% Solids)
Capsule composition	85-90 wt.% PCM 10-15 wt.% polymer shell
Core material	Paraffin
Particle size (mean)	15 - 30micron
Melting Point	24°C (75.2°F)
Heat of Fusion	154 - 164 J/g
Specific Gravity	0.9
Temperature Stability	Extremely stable – less than 1% leakage when heated to 250°C
Thermal Cycling	Multiple

PACKAGING
This product is generally shipped in 50-gallon fiber drums of 140 pounds net weight. Sample quantities may be ordered for customers requiring smaller amounts of product.

HEALTH AND SAFETY
Please refer to the Safety Data Sheet (SDS) for necessary safety and handling precautions for this product.

PureTemp® Thermal Energy Storage Materials

PureTemp thermal energy storage materials offer new levels of performance in storing or releasing large quantities of thermal energy at any given temperature. Our proprietary formulations and patented manufacturing processes yield superior quality biobased phase change materials at cost effective prices.

Some key properties:

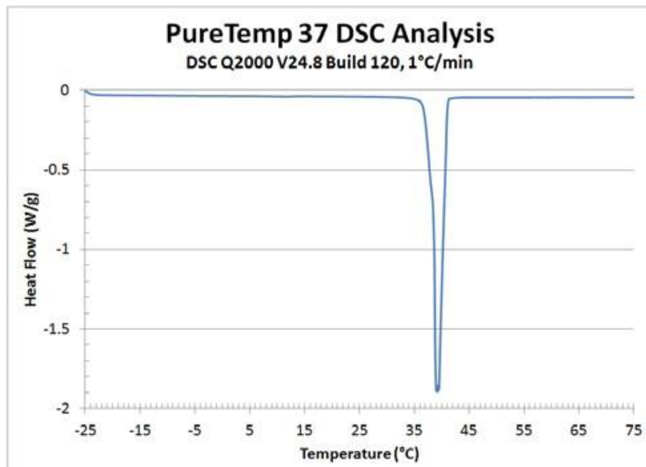
- Thermal energy storage capacities which average 200 J/g
- Over 200 unique, engineered phase change transition temperatures between -40˚C and 150˚C
- Consistent, repeatable performance over thousands of thermal (melt/solidify) cycles
- 100% renewable and readily biodegradable – produced from agricultural sources, not petroleum

PureTemp 37 Technical Information

PureTemp 37 is a USDA Certified Biobased product

Appearance	Clear liquid, waxy solid
Melting point	37 ºC
Heat storage capacity	210 J/g
Thermal conductivity (liquid)	0.15 W/m°C
Thermal conductivity (solid)	0.25 W/m°C
Density (liquid)	0.84 g/ml
Density (solid)	0.92 g/ml
Specific heat (liquid)	2.63 J/g˚C
Specific heat (solid)	2.21 J/g˚C

Typical physical properties are listed in the table above.



Thermal Cycle Stability

A thermal cycle stability study was performed on PureTemp 37 in which samples underwent a series of freeze and thaw cycles. The two year study completed ten thousand thermal cycles, with performance analyses performed on the samples at various time points. The study for PureTemp 37 found that:

- The average latent heat for PureTemp 37, over the course of 10,000 cycles, passes the product specification
- PureTemp 37 maintained a peak melting point of 38.1 ± 0.2 ºC

PureTemp 37 is stable through ten thousand thermal cycles, which is approximately 27.4 years of continuous daily usage.

Entropy Solutions, LLC.
151 Cheshire Lane N Suite 400, Plymouth, MN 55441
Tel: +1-952-941-0306 Email: info@puretemp.com
www.puretemp.com

USDA CERTIFIED BIOBASED PRODUCT
PRODUCT 100%

PureTemp® Thermal Energy Storage Materials

PureTemp thermal energy storage materials offer new levels of performance in storing or releasing large quantities of thermal energy at any given temperature. Our proprietary formulations and patented manufacturing processes yield superior quality biobased phase change materials at cost effective prices.

Some key properties:

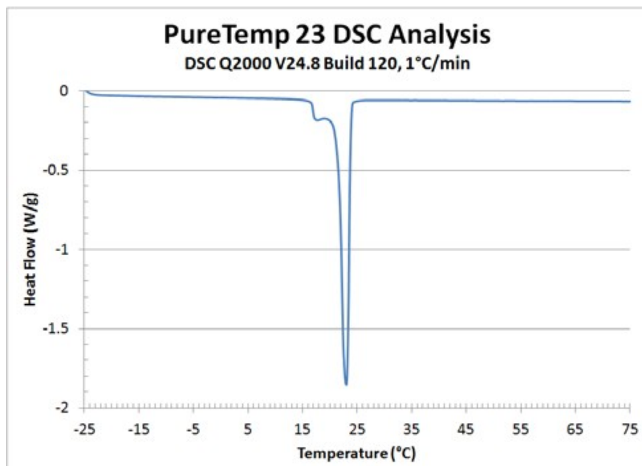
- Thermal energy storage capacities which average 200 J/g
- Over 200 unique, engineered phase change transition temperatures between -40 ˚C and 151 ˚C
- Consistent, repeatable performance over thousands of thermal (melt/solidify) cycles
- 100% renewable and readily biodegradable – produced from agricultural sources, not petroleum

PureTemp 23 Technical Information

PureTemp 23 is a USDA Certified Biobased product

Appearance	Clear liquid, waxy solid
Melting point	23 °C
Heat storage capacity	227 J/g
Thermal conductivity (liquid)	0.15 W/m°C
Thermal conductivity (solid)	0.25 W/m°C
Density (liquid)	0.83 g/ml
Density (solid)	0.91 g/ml
Specific heat (liquid)	1.99 J/g˚C
Specific heat (solid)	1.84 J/g˚C

Typical physical properties are listed in the table above.



Thermal Cycle Stability

A thermal cycle stability study was performed on PureTemp 23 in which samples underwent a series of freeze and thaw cycles. The two year study completed 10,000 thermal cycles, with performance analyses performed on the samples at various time points. The study for PureTemp 23 found that:

- The average latent heat for PureTemp 23, over the course of 10,000 cycles, passes the product specification.
- PureTemp 23 maintained a peak melting point of 23.4 ± 0.2 °C.

PureTemp 23 is stable through 10,000 thermal cycles, which is approximately 27.4 years of continuous daily usage.

Entropy Solutions, Inc.
151 Cheshire Lane N. Suite 400, Plymouth, MN 55441
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Inquiry: www.puretemp.com/contact
Website: www.puretemp.com

USDA CERTIFIED BIOBASED PRODUCT
PRODUCT 100%

CRODATHERM
Bio-Based Phase Change Material

Innovation you can build on™

CrodaTherm™ 29

Ambient temperature phase change material
Phase Change: 29°C, 84°F

CrodaTherm 29 is a water insoluble organic phase change material derived from plant-based feedstocks and has the form of a crystalline wax or oily liquid (depending on temperature).

CrodaTherm 29 is ideal for use in textiles, HVAC and applications requiring above room temperature thermal management. CrodaTherm 29 has low flammability, is readily biodegradable and non-toxic.

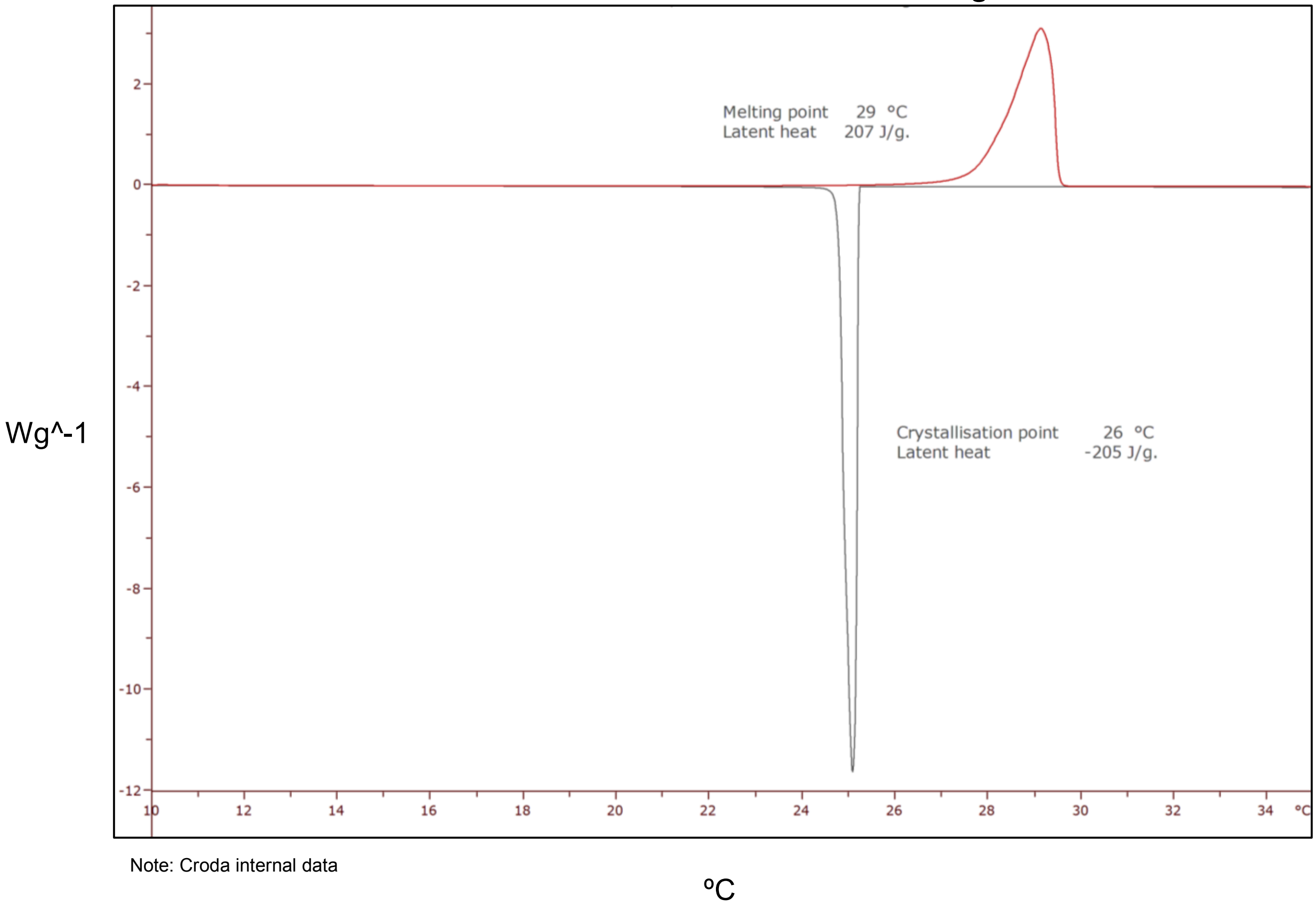
Typical properties

Test	Typical Value	Units
Melting temperature	29	°C
Latent heat, melting	207	kJ/kg
Crystallisation temperature	26	°C
Latent heat, crystallisation	- 205	kJ/kg
Specific heat capacity, solid	1.7	kJ/(kg·°C)
Specific heat capacity, liquid	2.1	kJ/(kg·°C)
Volumetric heat capacity, solid	1.56	$MJ/(m^3 \cdot °C)$
Volumetric heat capacity, liquid	1.79	$MJ/(m^3 \cdot °C)$
Thermal conductivity, solid	0.22	W/(m·°C)
Thermal conductivity, liquid	0.15	W/(m·°C)
Flash point	215	°C
Density at 6ºC (solid)	917	kg/m^3
Density at 30ºC (liquid)	851	kg/m^3
Thermal expansion (solid to liquid)	7.8	% Volume

Email: CrodaIndustrialChemicals@Croda.com
www.CrodaTherm.com

CRODA
Industrial Chemicals

CrodaTherm 29 DSC at 1K/ min scanning rate

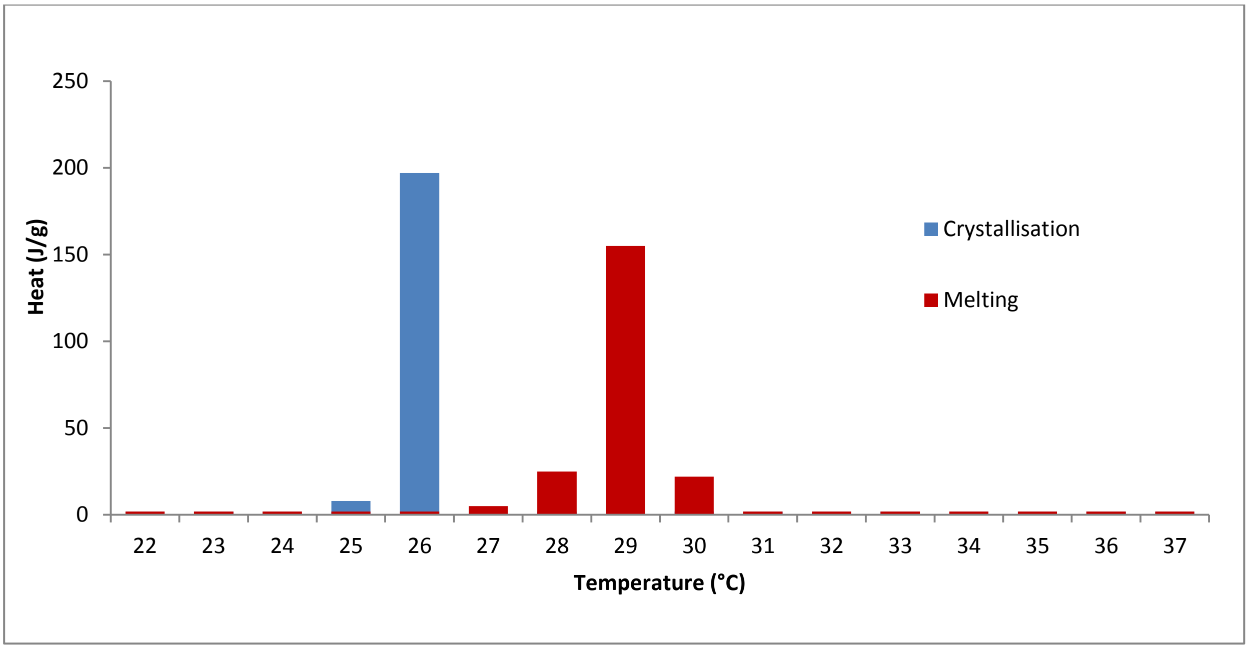


Note: Croda internal data

Enthalpy Distribution

Temperature interval (°C)		Crystallisation Heat (J/g)	Crystallisation Heat (J/mL)	Melting Heat (J/g)	Melting Heat (J/mL)
21.5 22.5	22	2	2	2	2
22.5 23.5	23	2	2	2	2
23.5 24.5	24	2	2	2	2
24.5 25.5	25	8	7	2	2
25.5 26.5	26	197	169	2	2
26.5 27.5	27	2	2	5	4
27.5 28.5	28	2	2	25	22
28.5 29.5	29	2	2	155	133
29.5 30.5	30	2	2	22	19
30.5 31.5	31	2	2	2	2
31.5 32.5	32	2	2	2	2
32.5 33.5	33	2	2	2	2
33.5 34.5	34	2	2	2	2
34.5 35.5	35	2	2	2	2
35.5 36.5	36	2	2	2	2
36.5 37.5	37	2	2	2	2
	Total	**233**	**200**	**231**	**199**

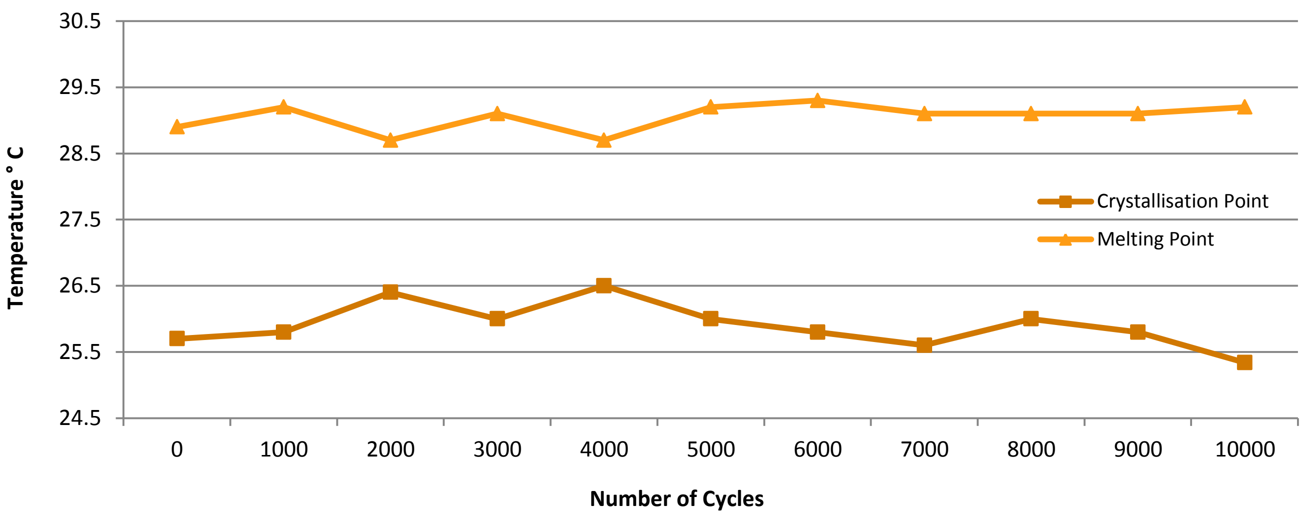
CrodaTherm 29 Heat Diagram

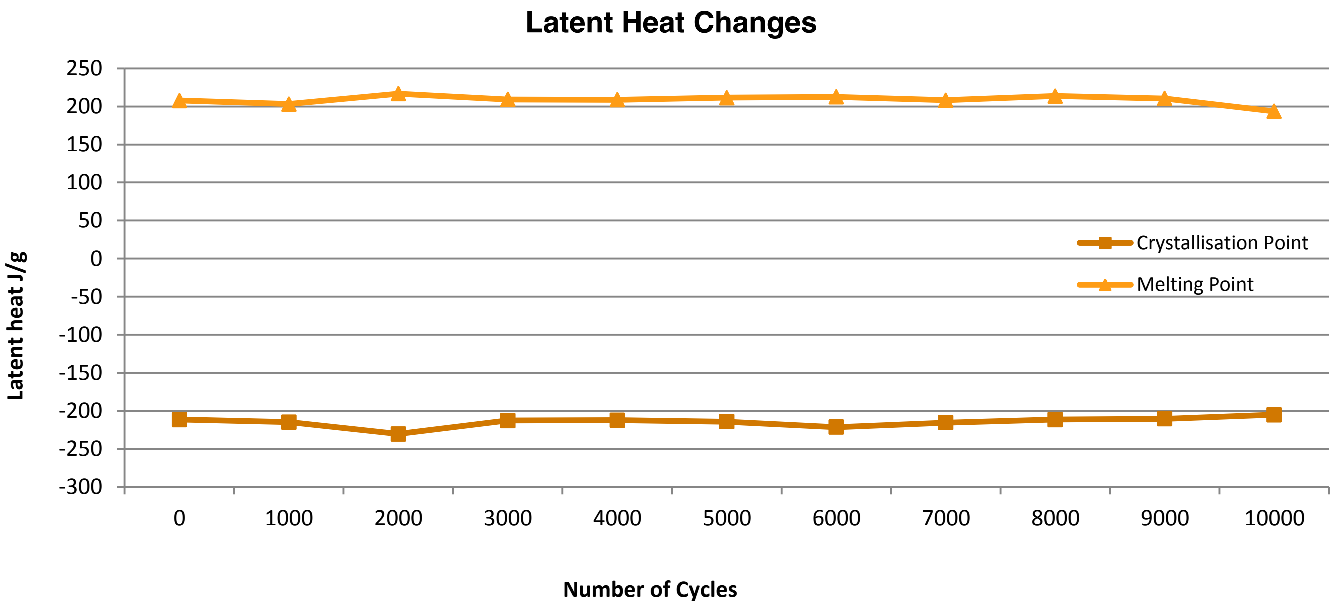


Cycle Stability Testing

CrodaTherm 29 has been tested to RAL defined specifications and achieved category "A" performance, reaching 10,000 thermal (melt/solidify) cycles. For example purposes only, 10,000 cycles would represent approximately 27 years use if the PCM undergoes one melt/freeze cycle per 24 hour period, demonstrating long term stability.

Melting and Crystallisation Points





Latent heat change data

Cycle	% Change in Latent heat of Melting	% Change in Latent heat of Crystallisation
0	0	0
1,000	-2.21	1.61
2,000	4.33	8.94
3,000	0.72	0.61
4,000	0.53	0.43
5,000	1.78	1.32
6,000	2.26	4.63
7,000	0.19	1.89
8,000	2.84	0.05
9,000	1.30	-0.52
10,000	-6.79	-2.93

Evaporative Loss

CrodaTherm PCMs have a lower evaporation rate than paraffinic alternatives. In testing, CrodaTherm 29 had an evaporative loss rate approximately **19 times lower** than an alternative octadecane PCM. The table below displays comparisons between CrodaTherm PCMs and their paraffinic alternatives.

Material	**Octadecane**	**CrodaTherm 25**	**CrodaTherm29**
Sample mass (mg)		**9**	
Weight loss (%)	15	8	0.8
Evaporates X times slower than equivalent alkane	-	2	19
Testing temperature / °C		80	

Note: samples tested over a 12 hour period

Handling CrodaTherm 29

For ease of handling and optimum performance, please ensure CrodaTherm 29 is completely molten and homogenous before transferring from the container. CrodaTherm 29 may need heating before use, to ensure that the product has completely melted. Avoid excessive temperatures (>100°C) and prolonged exposure to open air.

Email: CrodaIndustrialChemicals@Croda.com
www.CrodaTherm.com

Email: CrodaIndustrialChemicals@Croda.com

www.CrodaTherm.com

Appendix G. Budget/Procurement List

Since there are no manufactured parts in our design the budget/procurement list is the same as the indented BOM. However, it is attached here as well for convenience.

Indented Bill of Material (BOM): Transient Heat Cooling Loop Assembly

Assy Level	Part Number	Description	Matl	Vendor	Qty	Cost	Ttl Cost	Status
0	CL01-00	**Sub-scale Model**						
1	CL01-01	Main Pump	------	------	1	$ -	$ -	Received
1	CL01-31	Piping	PVC	McMaster-Carr	2	$ 5.50	$ 11.00	Purchased
1	CL01-32	90° Elbow	PVC	McMaster-Carr	10	$ 3.40	$ 34.00	Purchased
1	CL01-33	Tee	PVC	McMaster-Carr	10	$ 0.44	$ 4.40	Purchased
1	CL01-34	Ball Valve	Brass	McMaster-Carr	2	$ 3.21	$ 6.42	Purchased
1	CL01-03	Heat Exchanger	------	PDT	2	$ 40.00	$ 80.00	Purchased
1	CL01-04	Heater	------	PDT	1	$ -	$ -	Purchased
1	CL01-02	PCM Reservoir						
2	CL01-02-0	PCM Heat Exchanger	------	AIHTI	1	$ 565.75	$ 565.75	Received
2	CL01-02-1	Sight	PVC/ glass	McMaster-Carr	1	$ 7.30	$ 7.30	Canceled
2	CL01-02-2	Cap	PVC	McMaster-Carr	1	$ 0.32	$ 0.32	Purchased
2	CL01-02-3	Relief Valve	Brass	McMaster-Carr	1	$ 81.75	$ 81.75	Canceled
1	CL01-41	Temperatuere Sensors						
2	CL01-41-1	Thermowell	SST 304	McMaster-Carr	4	$ 24.75	$ 99.00	Canceled
2	CL01-41-2	Thermocouple	Type T	McMaster-Carr	1	$ 83.00	$ 83.00	Purchased
2	CL01-41-3	Thermocouple Reader (Omega RDXL6SD)	------	Omega	1	$ 620.00	$ 620.00	Purchased
1	CL01-42	Flow Meter	------	------	2	$ -	$ -	Received
1	CL01-43	Pressure Sensor	------	------	2	$ -	$ -	Received
1	CL01-11	Secondary Pump	------	------	2	$ 33.49	$ 66.98	Purchased
1	CL01-12	Reservoir	Plastic	------	1	$ -	$ -	Received
1	CL01-51	PVC Cement	------	McMaster-Carr	1	$ 4.77	$ 4.77	Purchased
1	CL01-52	PVC Cleaner	------	McMaster-Carr	1	$ 4.87	$ 4.87	Purchased
0	CL01-53	Structural Supports	Wood	Home Depot	1	$ -	$ -	Received
1		**PCMs**						
1	PCM01	Microtek 24	------	Microtek	2	$ 110.00	$ 220.00	Received
1	PCM02	PureTemp 37	------	PureTemp	2	$ 140.00	$ 280.00	Received
1	PCM03	PureTemp 23	------	PureTemp	2	$ 140.00	$ 280.00	Received
1	PCM04	CrodaTherm 29	------	Croda	1	$ -	$ -	Received
						Purchased Parts Total:	**$ 2,449.56**	

APPENDIX H. FINAL ANALYSES AND TESTING DETAILS

Test Title	Date	Flowrate	T_initial	T_final	T_peak	T_PCM_peak
Control 1-1	5/9/18	7.2	21.3	23.0	24.5	24.6
Control 1-2	5/9/18	5.8	22.1	24.7	26.0	26.6
Control 1-3	5/9/18	3.3	23.9	26.5	28.4	28.3
Control 1-4	5/9/18	1.3	25.6	28.7	30.9	29.8
Control 2-1	5/16/18	4.8	20.4	22.5	24.1	24.5
Control 2-2	5/16/18	4.8	20.9	23.1	23.8	27.2
Control 3-1	5/17/18	0.6	22.0	26.4	28.8	30.4
Control 3-2	5/17/18	0.6	21.3	25.4	28.4	27.5
Control Cold - Air	5/28/18	1.6	19.4	21.9	24.7	29.7
Control Cold - Water	5/28/18	1.6	19.6	22.5	24.7	26.0
Control Cold-Reverse - Water	5/28/18	1.6	19.8	21.6	25.6	23.4
Control Cold-Reverse - Air	5/28/18	1.6	19.5	21.3	25.7	22.7
Croda29	5/19/18	1.5	21.6	26.7	29.3	28.0
Croda29 - Cooling	5/19/18	1.5	22.4	22.8	29.1	28.3
MicroTek 24 - Static	5/19/18	0.6	23.1	25.7	29.9	29.6
MicroTek 24 - Flowing	5/19/18	0.6	21.7	24.8	28.2	29.4
MicroTek 24 - Flowing Brazed Plate	5/22/18	0.6	22.0	23.4	25.2	27.9
PT23 COLD	5/28/18	1.6	19.5	21.6	24.4	23.3
PT23 COLD_Reverse	5/28/18	1.6	18.8	21.3	24.7	21.5

Appendix I. Safety Hazard Checklist and FMEA

Team: Transient Heat Storage Advisor: Dr. Rossman Date: 02/04/08

Y	N	
☐	☒	1. Will the system include hazardous revolving, running, rolling, or mixing actions?
☐	☒	2. Will the system include hazardous reciprocating, shearing, punching, pressing, squeezing, drawing, or cutting actions?
☐	☒	3. Will any part of the design undergo high accelerations/decelerations?
☐	☒	4. Will the system have any large (>5 kg) moving masses or large (>250 N) forces?
☐	☒	5. Could the system produce a projectile?
☒	☐	6. Could the system fall (due to gravity), creating injury?
☒	☐	7. Will a user be exposed to overhanging weights as part of the design?
☐	☒	8. Will the system have any burrs, sharp edges, shear points, or pinch points?
☐	☒	9. Will any part of the electrical systems not be grounded?
☐	☒	10. Will there be any large batteries (over 30 V)?
☐	☒	11. Will there be any exposed electrical connections in the system (over 40 V)?
☐	☒	12. Will there be any stored energy in the system such as flywheels, hanging weights or pressurized fluids/gases?
☐	☒	13. Will there be any explosive or flammable liquids, gases, or small particle fuel as part of the system?
☐	☒	14. Will the user be required to exert any abnormal effort or experience any abnormal physical posture during the use of the design?
☐	☒	15. Will there be any materials known to be hazardous to humans involved in either the design or its manufacturing?
☐	☒	16. Could the system generate high levels (>90 dBA) of noise?
☐	☒	17. Will the device/system be exposed to extreme environmental conditions such as fog, humidity, or cold/high temperatures, during normal use?
☒	☐	18. Is it possible for the system to be used in an unsafe manner?
☒	☐	19. For powered systems, is there an emergency stop button?
☐	☒	20. Will there be any other potential hazards not listed above? If yes, please explain on reverse.

For any “Y” responses, add (1) a complete description, (2) a list of corrective actions to be taken, and (3) date to be completed on the reverse side.

Description of Hazard	Planned Corrective Action	Planned Date	Actual Date
6- The system has a water reservoir that when filled, makes the system top heavy. This leaves the potential for the device to tip if knocked.	Keep the caution tape that currently surrounds the existing apparatus and only have the top reservoir filled when the apparatus is in use.	2/26/18	
7- The water reservoir will be above the operator's head when adjustments are made to the apparatus.	Fill the reservoir after the required adjustments are made to the apparatus and keep the time spent under the reservoir during operation to a minimum (turning two valves near the start of operation).	2/26/18	
18- The system can be used in an unsafe manner by allowing the heater to run continuously, which it is not designed to handle, or adjusting the flow valves incorrectly.	Have operators stand with the apparatus to adjust the heater during operation. Only adjust the flow valves during the start up and shutdown of operation.	2/26/18	
19- The system contains an electrical shutoff for our two pumps and the heater that we are planning to use.	Confirm that the electric components of the system are functioning properly before operation, and have one operator ready to shutoff the system at any time.	2/21/18	

Appendix J. Operator's Manual

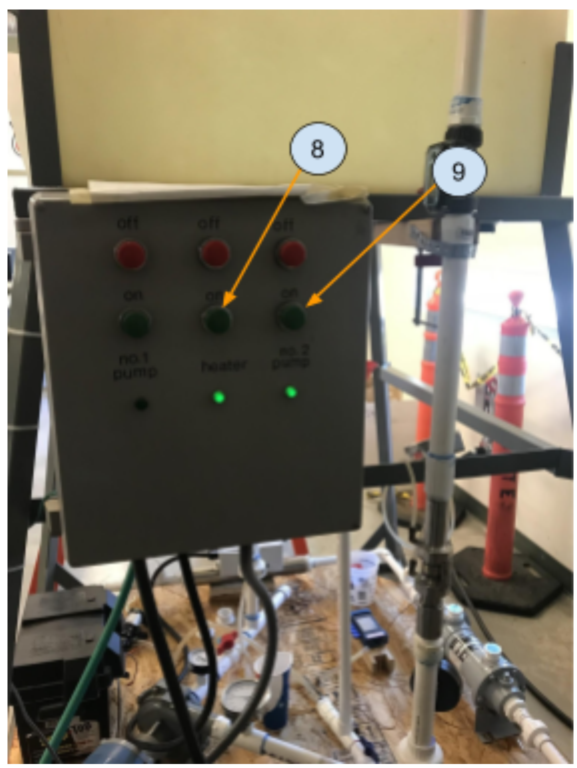
Transient Heat Storage Test Apparatus

Equipment Required:

1. Main loop pump
2. In line heater
3. PCM heat exchanger
4. Secondary loop heat exchanger
5. Secondary loop pump
6. Omega thermocouple reader
7. Main loop pump switch
8. Heater on/off buttons
9. Secondary loop pump on/off buttons

Auxiliary Equipment/Materials:

- PCM's to test
- Coolant for the both loops
- Thermocouples (5x recommended)
- Hot plate to melt PCMs



Test Procedure:

1. Safety/Preparation
    a. **Close** all valves and make sure all powered elements are off.
    b. **Fill** the reservoir to the desired level.
    c. **Fill** the main loop with water.
        i. Watch for leaks.
    d. **Turn on** the Main Loop Pump.
        i. Watch for leaks, if there are leaks turn off the pump.
        ii. Check the pressure gauges around the system. They should all read approximately 14.7 psi.
    e. Continue to add water to the primary loop until no more air bubbles can be seen in the loop, then **turn off** the pump.
    f. **Open** the secondary loop valve to fill the loop.
        i. Watch for leaks, if there are leaks close the valve.
    g. With the valve still open, **turn on** the Secondary Loop Pump.
        i. Watch for leaks, if there are leaks close the valve and turn off the pump.
    h. After the pump has run for around a minute, **close** Valve #2 and **turn off** the pump.

2. Control Test (No PCM)
    a. Make sure there is nothing (or water) in the shell of the PCM heat exchanger.
        i. If there is, see section 4, Emptying the shell.
    b. **Turn on** both pumps and wait until the temperature reading for thermocouple #1 has reached steady state.
    c. Begin recording temperature data.
    d. **Turn the heater** with the heater button on the control panel.
        i. Make sure that the dial on the heater is turned to maximum.
        ii. If any of the temperature readings go above 60°C, Turn off the heater.
            1. Leave the pumps running until the temperature is safe again.
    e. After 30 seconds, **turn off** the heater.
    f. After 2 minutes, **turn on** the heater.
    g. Repeat steps d and e for 6 cycles (or 15 minutes).
    h. Save the temperature vs time data for the run by removing the sd card from the thermocouple reader and copying its contents.
    i. Wait for the temperature to go back down to its initial value, then turn off both pumps

.

3. PCM Effectiveness Test
    a. For wax based PCMs

i. If the shell is not empty, see section 5, Emptying the shell.
ii. Melt the desired amount of PCM by placing its storage bucket onto the hot plate
iii. Fill the shell of the PCM Reservoir with the melted PCM.

b. For microcapsules
   i. If the shell is not empty, see section 5, Emptying the shell.
   ii. Fill the shell with microcapsule-water solution.
      1. Concentration will be based on previous testing.
c. **Turn on** both pumps and wait until the temperature reading for thermocouple #1 has reached steady state (below the melting point of the chosen PCM).
d. Begin recording temperature data.
e. **Turn the heater** with the heater button on the control panel.
   i. Make sure that the dial on the heater is turned to maximum.
   ii. If any of the temperature readings go above 60°C, Turn off the heater.
      1. Leave the pumps running until the temperature is safe again.
f. After 30 seconds, **turn off** the heater.
g. After 2 minutes, **turn on** the heater.
h. Repeat steps d and e for 6 cycles (or 15 minutes).
i. Save the temperature vs time data for the run by removing the sd card from the thermocouple reader and copying its contents.
j. Wait for the temperature to go back down to its initial value, then turn off both pumps.

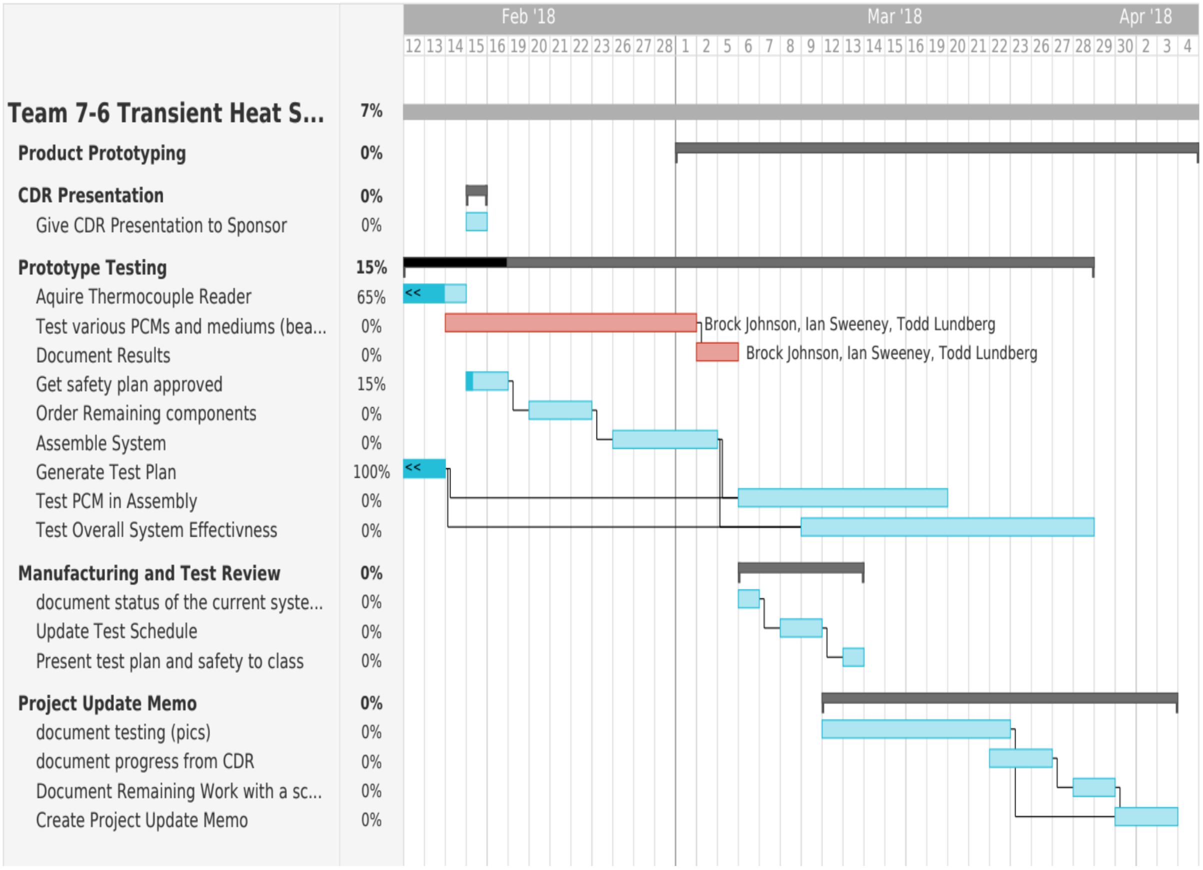
4. Clean-up
   a. Open valve at the bottom of the tank to begin draining the top tank.
   b. Remove Cap from the vertical tube in front of the pump.
   c. Use the shop vac to remove water from the reservoir.
      i. Turn on the pump after 20 seconds to improve suction.
      ii. Turn off the pump when there is no more water being removed.
   d. Remove the vertical tube in front of the pump with the shop vac at the opening.
   e. Use the shop vac to remove water from the system.
      i. Turn on the pump after 20 seconds to improve suction.
      ii. Turn off the pump when there is no more water being removed.
   f. Remove both couplings on either side of the PCM heat exchanger.
   g. Vacuum water from both parts of the main loop.
   h. Vacuum the water out of the heat exchanger.
      i. Hold the O-ring in place on the PVC piping.
      ii. If the O-ring is pulled out, get it out of the shop vac and place it back.
   i. Pour any remaining water out from PCM heat exchanger.

    i. Be careful to not pour out liquid PCM if there is any in the shell.

j. Return any equipment borrowed from the lab back to its original location.
k. Once the tank has stopped draining, shut off the tank valve and remove the hose.
    i. Make sure to remove all water in the hose before coiling it back up.
l. With the shop vacuum on slowly unthread the coupling in the secondary loop and drain the water as it comes out.
    i. Check for any spills and vacuum them up.
    ii. Rethread the coupling when complete
m. Vacuum up any remaining water on the floor or apparatus.
n. Pour the water from the shop vacuum down the drain.
o. **Turn off** the electrical supply.
p. Pull out the plug from the outlet and coil the wire around the control box.
q. Move the apparatus into its designated location and surround it with caution tape.

5. Emptying the shell
    a. If the shell contains a solid PCM, run the apparatus with the heater on until the contents are entirely liquid.
    b. **Remove** the tube and shell heat exchanger removed from the loop.
        i. See section 4 if unclear how to do this.
    c. **Cap one shell outlet** and pour liquid from the other side into the PCM container.
    d. **Switch the cap** to the other side and repeat. This is necessary because of the baffles in the heat exchanger.
    e. Pour hot water into the shell to melt any residual material that may have clung to the walls.
    f. Then pour this into a waste container or drain.
    g. Repeat steps e and f until the water comes out clear.

APPENDIX K. GANTT CHART



Tasks in the near future are expanded, this includes Product Testing the Interim Design Review and Product Prototyping. The next major milestone for the project is the Critical Design Review in early February.

APPENDIX L. FURTHER INVESTIGATIONS IN THEORY

What are important parameters in the design of a Transient Heat Storage System? Preliminary guesses point out:

Duty Cycle

Duty Time

Power Ratio? Maybe more specifically:

Additional Power of high power stage to low power stage. Power difference?

Steady state temperatures at each power stage.

Total energy to store → derived from Duty Time

Thermal Mass

Ratio of allowable temperature change to steady state temperatures?

In trying to pinpoint exactly what drives the design, some of these ought to describe the problem very succinctly and others will be superfluous. Hopefully we can get some dimensionless numbers like the Jakob number, the ratio of sensible and latent heats, that can direct us right to the best solution from the problem statement. Turbomachinery design does the same thing with specific speed, which relates the problem to the most efficient pump/turbine type.

L.1 START DESIGNING

Let us look at an extreme case of thermal storage: lunchbox. An insulated cooler with an icepack inside.

Duty time ~6 – 12 hours of cooling, same with recharge time.

Low thermal mass of contents besides the ice.

Low allowable temperature change

Low power → insulated box, low heat absorption.

natural convection, low circulation power available.

no auxiliary power available.

Medium amount of energy to store.

1 cycle per "flight"

During use, the ratio of cooler wall thermal resistance to icepack thermal resistance is high. In addition, ice floats to the high point of pack, corresponding to the hottest part in contact with pack. This reduces the thickness of the liquid layer that heat must be conducted through. In the end, the air temperature stays closer to the icepack phase change temperature than to the outside air temperature. I'd guess the phase change temperature of icepacks is a decent fifteen degrees below pure ice, to weight the air temp further down, close to fridge temperatures. Regular home freezers are at around 0°F, so the icepack can still recharge overnight. The results are terrific, and my lunch is always refreshingly cold so long as the icepack is above the food.

Now let's look at another familiar heat problem, computer processor cooling. Simple, easy to analyze (power consumed all goes to heat, conduction, essentially one dimensional), could have a very high transient load for the space, possibly limited coolant (air or liquid), and probably very applicable to PDT's problem. If the processor just got an upgrade, very possible, and now needs to deal with huge power transients in an existing system, there are a variety of solutions.

Bigger heatsink

+ more convection possible

+ more thermal mass

- more mass

Bigger fan/ add a fan

+ more convection

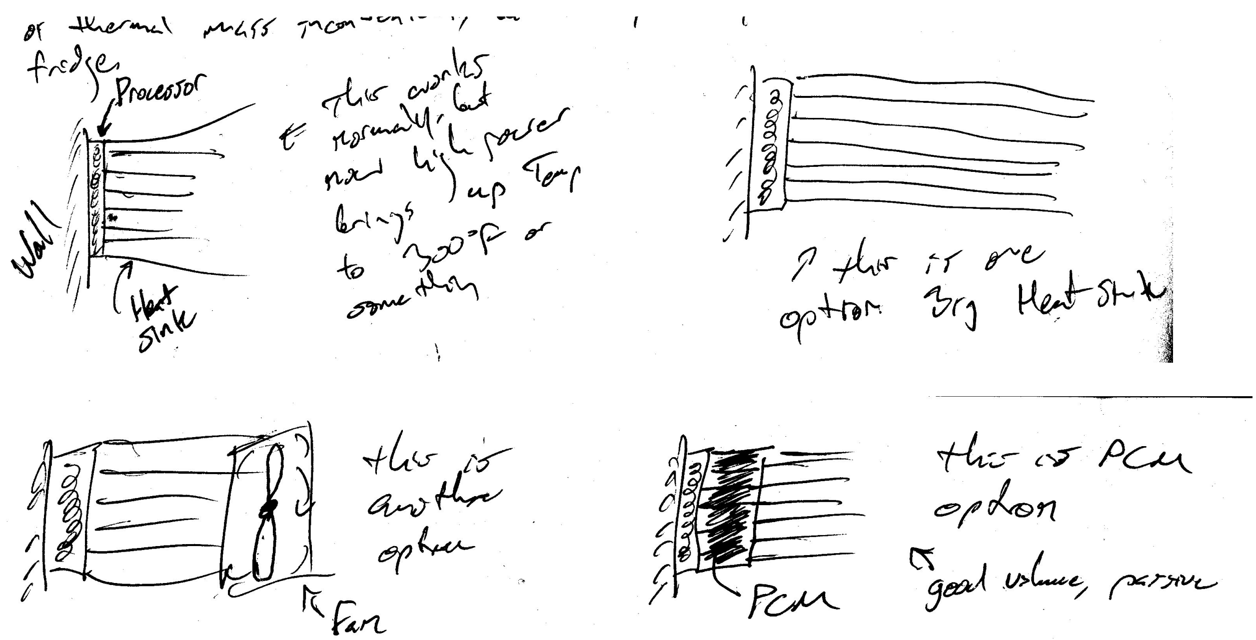
- more power

Add PCM

+ more thermal mass

- more mass

- expensive (both design and material costs)



More convection wins in steady state power level. More thermal mass is primarily a transient benefit. Higher frequency transients lead to more benefit per pound of real mass. As long as thermal conductivity is high enough, lumped capacitance is valid and all thermal mass has equal benefit. If lumped capacitance is not valid, then geometry affects mass efficiency. More mass is bad at all times (weight, cost). PCMs can have a good thermal mass/weight mass ratio, if frequency is low enough to penetrate the geometry, but high enough to not need too much weight (compared to more convection).

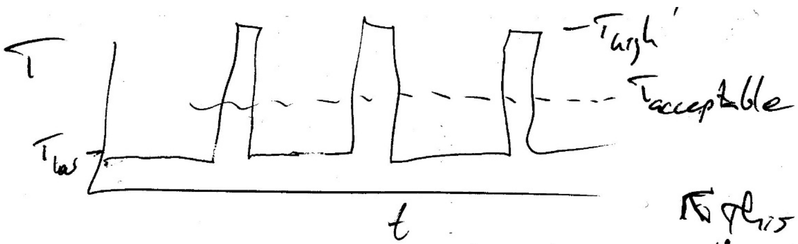
Let's say this processor is inconveniently located, where adding more convection is a pain, (Either due to geometrical constraints like a crowded area, coolant routing, or power routing, or coolant and power availability is maxed out) then adding PCM into heatsink may be a good plan. This correlates well with the lunchbox idea, adding a fan, power, or thermal mass (without pcm, just a block of cool metal) would be inconvenient, but the steady state problem (very low frequency) would take a fridge.

To design this PCM solution I'd want to know:

Power of high stage

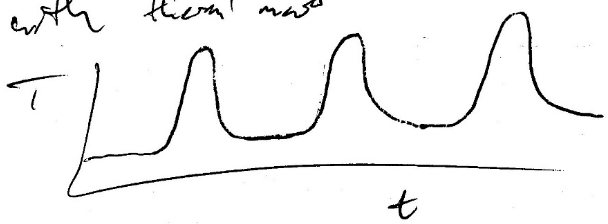
Power dissipated at allowable raised temperature during high stage (this just lets me know how much power I can ignore)

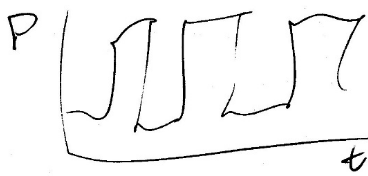
Probably, the profile of heat transfer rates with temperature as the system cools down, probably a constant with respect to temperature difference from steady state. This is a confusing way to state an obvious and well-known Fourier's law of conduction or Newton's law of convection.



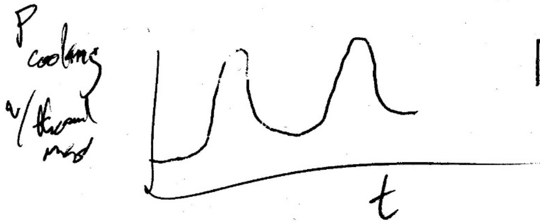
This graph assumes no thermal mass of heat sink, but higher convection at higher temp.

with thermal mass





Just like T vs t no thermal mass
Also, Power cooling no th mass is the same



P cooling w/ thermal mass looks just like T vs t with th mass.

We want to add thermal mass so that the temperature stays below the acceptable maximum temperature. In order to do this, the cooling power integral must equal the incoming power integral per cycle. In other words, either enough time in between high stages must be put in, or either power in or power out must change. To maximize power out each low stage, with PCM, the PCM phase change temperature must be as high as we can allow. To maximize power absorbed by the PCM, the phase change temperature must be as low as possible. The compromise that allows minimum geometry, with a known duty cycle, is a phase change temperature of

$$T_{PC} = (duty\ cycle) * T_{allowable} + (1 - duty\ cycle) * T_{low} \tag{L.1}$$

This minimum geometry design will take the entire low stage to cool down, assuming T_{PC} is constant, meaning no subcooling or superheating is required to make the phase change happen. If there was both, the equation would be modified to

$$\begin{aligned} T_{PC} = (duty\ cycle) * (T_{allowable} - superheat) \\ + (1 - duty\ cycle) * (T_{low} + subcooling) \end{aligned} \tag{L.2}$$

For example, if T_{low} = 70°F, $T_{allowable}$ = 120°F, duty cycle = 30%, L.1 would find the optimum temperature to be 85°F, but tests showed that it took five degrees of superheating and seven of subcooling, then

$$T_{PC} = (.3) * (120 - 5) + (1 - .3) * (70 + 7)$$

And the optimum temperature to pick would be 88.4°F. As a note, our tests were not good enough to point out degrees of superheating or subcooling, but this is a known issue with gallium, where amazon reviews have people complaining about their gallium samples not solidifying. This formula assumes cooling and heating are equal and opposite events in the PCM, even though some Nusselt number correlations indicate that it isn't, even in single phase convection, and our PCM data shows very different conductivity in solid and liquid phases of the same PCM.

So, let's take the conductivity into account, but not convection. I assume the liquid layer is too thin to start advection, so only conduction applies. For waxes, $k_{solid} \approx 2x\ k_{liquid}$, for instance one wax had a solid conductivity of .144 Btu/hr·ft·°F while the liquid had .088. Let's rederive the optimum phase change temperature equation. The total heat absorbed/released on both stages are equal

$$Q_{high} = Q_{low} \tag{L.3}$$

The heat absorbed is the heat rate times the time of the stage

$$Q_{high} = q_{high} * t_{high} \tag{L.4}$$

Pure conduction, assuming constant temperatures

$$t_{high}\, k_{liquid} A \frac{\Delta T_{high}}{\Delta x} = t_{low}\, k_{solid} A \frac{\Delta T_{low}}{\Delta x} \tag{L.5}$$

Geometry doesn't change between stages, so Area and depth are equal

$$t_{high}\, k_{liquid}(T_{allow} - T_{PC}) = t_{low}\, k_{solid}(T_{PC} - T_{low}) \tag{L.6}$$

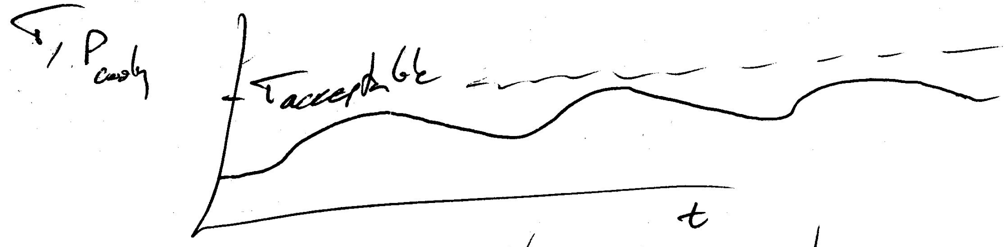
$$\begin{aligned} t_{high}\, k_{liquid} T_{allow} + t_{low}\, k_{solid} T_{low} \\ = (t_{high}\, k_{liquid} + t_{low}\, k_{solid}) T_{PC} \end{aligned} \tag{L.7}$$

$$T_{PC} = \frac{t_{high}\, k_{liquid} T_{allow} + t_{low}\, k_{solid} T_{low}}{(t_{high}\, k_{liquid} + t_{low}\, k_{solid})} \tag{L.8}$$

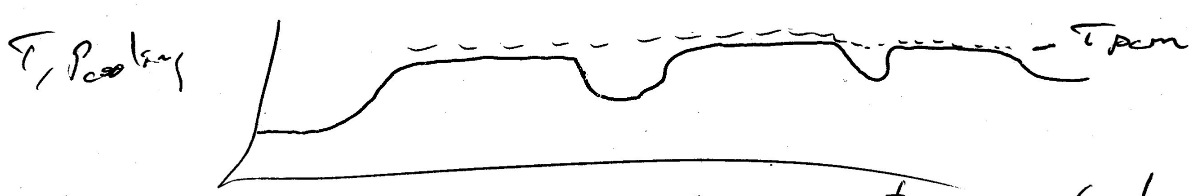
Putting in our degrees of superheat and subcooling

$$T_{PC} = \frac{t_{high}\,k_{liquid}(T_{allow} - {}^{\circ}Super) + t_{low}\,k_{solid}(T_{low} + {}^{\circ}Sub)}{(t_{high}\,k_{liquid} + t_{low}\,k_{solid})} \tag{L.9}$$

We want to add thermal mass so that

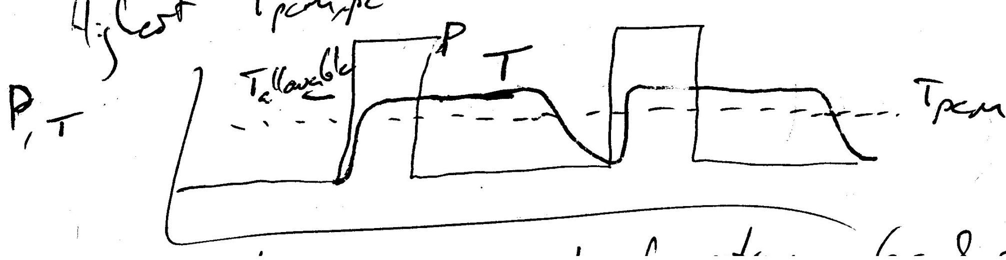


In order to do this, the cooling power integral must equal the incoming power integral high cycles must l
→ either enough time in between
put in, or power in or power out must change.
to maximize power out, with pcm



PCM phase change temperature must be as high as we can allow

Highest Tpcm,pc we can allow



[illegible] minim

L.2 Let's give it a try

Let's try to design PCM reservoir for the processor. We want to now the power absorbed the PCM and time. Really, the total heat absorbed (Btu's in melt) and the power to absorb at the end of the melt. We assume the power level is constant during the high-power stage, but it really could vary quite a bit as long as we take the integral properly, and the power at the end of the high stage is still relatively high. The least effective the PCM will ever be is at the end of its cycle, when the melted layer is thickest, and only ever so much remains solid (for a solid/liquid phase change; for a liquid/vapor phase change, this is still true, but much less so). The power at the end of the cycle will need the most ΔT to get the same heat flux. For a uniform geometry, such as a rectilinear heatsink with square everything, all the melt layer is uniform thickness Δx. ΔT is $T_{allow} - T_{PC}$, k is thermal conductivity of melt layer, and q is the power to absorb. Q_L is latent heat of phase change, and Q_{st} is total heat absorbed in stage, and ρ is the density (I like to just pretend the liquid density and solid density are the same)

$$Q_{st} = \rho Q_L A \Delta x \tag{L.10}$$

$$q = -kA\frac{\Delta T}{\Delta x} \tag{L.11}$$

Q_{st}, T_{allow}, T_{low} are situation dependent

Q_L, ρ , T_{PC}, k are material properties that are fixed once a material is picked.

A, Δx are design variables → 2 unknowns for 2 equations

Material is the third unknown that must be picked, but then represents four constants. Material is picked to minimize required geometry, mass. Typically, A is large, Δx is small, but A determines how much aluminum fin area we need, Δx determines performance.

For our processor example assume wax pcm

q = 5 Btu/sec (~5kW)

Q_{st} = 150 Btu (30 sec stage)

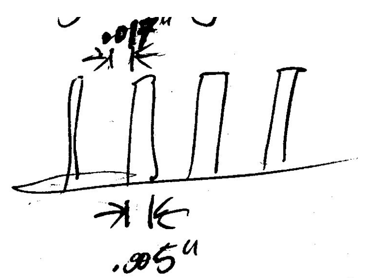
ρ = 55 lb/ft^3

Q_L = 100 Btu/lb

k_{liq} = .088 Btu/hr· ft· °F

Pick ΔT = 15° arbitrarily

Results: A = 19.28 sq ft, Δx = .017"



You gotta cram 20 square feet into 47 cubic inches of wax. I guess that radiator fins are ~ 5 thou thick, and there is area on both sides, so 20 sq ft takes .042 ft^3 of aluminum → 7 lb aluminum to hold 1 ½ lb of wax.

Now try pure gallium

ρ = .2 lb/in^3

Q_L = 30 Btu/lb

k_{liq} = 10 Btu/hr· ft· °F (this is a guess, can't find data that is definitely for *liquid* gallium)

same ΔT = 15° arbitrarily

Results: A = 189 in^2 (1.3 sq ft), Δx = .131"

5 lb gallium but still need ½ lb of aluminum fins

Δx is still small compared to A

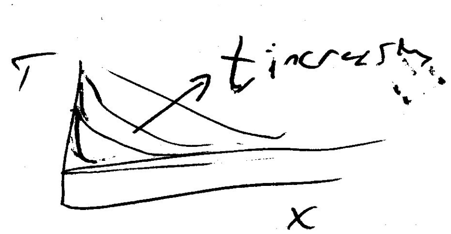
Gallium rocks, probably. Gallium is ~$150/lb. And gallium is like water, it expands upon freezing, so it will try to break its container. The liquid may also separate form solid as it melts, this is pure speculation, but the density difference may cause a void to form, which may lessen the surface contact with the solid layer, and slightly underperform.

Let's adjust the cycle with wax. Keep the same total energy stored, but adjust the power to 1 Btu/sec, so it has a 2 ½ minute high power stage. A*Δx is still the same, but A goes down to 8.6 sq ft, and Δx = .038", so the aluminum fins weigh only 3 pounds for the pound and a half of wax.

These examples all neglected the sensible heat capacity, both of aluminum fins and melted pcm. If ΔT is low, Q_L is high, $mass_{PCM}/mass_{metal}$ is high, then it is very valid to neglect. In this example Q_L is high, but ΔT is medium, and $mass_{PCM}/mass_{metal}$ is low, so let's make a check.

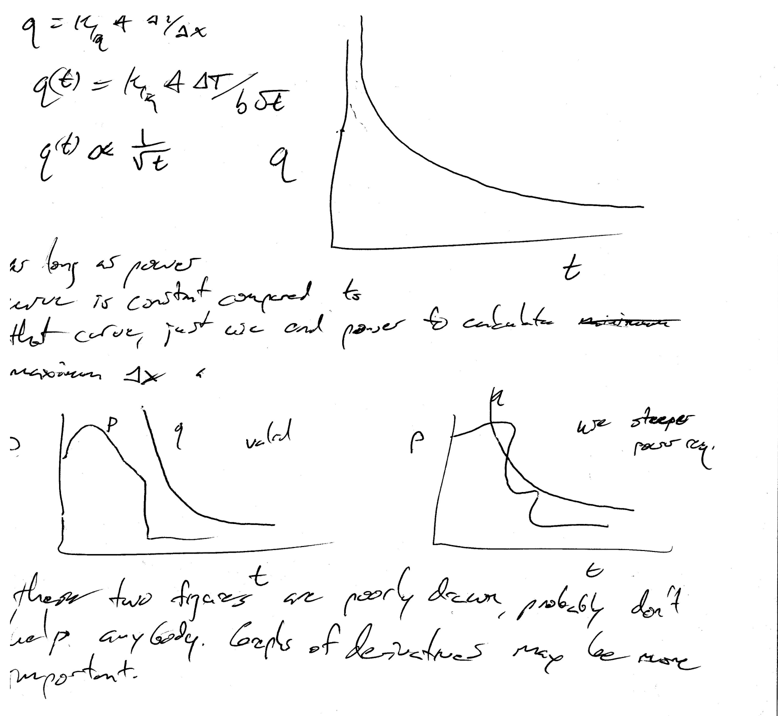
Say we wanted to do the same thing with aluminum, just aluminum, a big block of sweet conduction. Say the total ΔT we can allow is 20°F, $c_{p,\,Al}$ = .22 Btu/lb°F, to absorb 150 Btu it would take 34 lb aluminum. So, at 3 lb of metal we are taking care of about 10% of the problem without accounting for it in the PCM analysis.

Transient conduction through the length of the fins is also important. For aluminum, with a sudden change in surface temperature (corresponding to infinite heat transfer coefficient) it takes two seconds for the temperature an inch deep to reach 75% of the difference, if the aluminum is only an inch thick. So, if fins are propagating 4 inches in a direction, then it would take 32 seconds for the temperature at the end to hit the three quarter mark. This seems like it may not be negligible, but usually heat transfer is greater in the transients, because the temperature gradient is steeper, so one may not have to worry about it so much. Anyways, lesson is to keep it short.



It may be noted that if one multiplies the conductivities of wax, one can get memorizable numbers. The conductivity of the liquid is approximately 1 Btu· in/hr· ft^2· °F, and the solid has a conductivity of 1.728 Btu· in/hr· ft^2· °F (as opposed to .088 and .144 Btu· /hr· ft· °F).

Important parameter of phase change moving boundaries. If the wax starts as all solid, and the surface temperature (metal temperature) suddenly goes up, say 15 degrees, and stays at that 15 degree increase, then the liquid thickness starts at zero and grows at a rate proportional to the square root of time. $\Delta x = b\sqrt{t}$, where b is some constant. As long as the high-power stage thermal load is more constant than this curve (lower slope) then we can use the final power level to drive the design. Otherwise, the geometry may be driven by some intermediate power level.



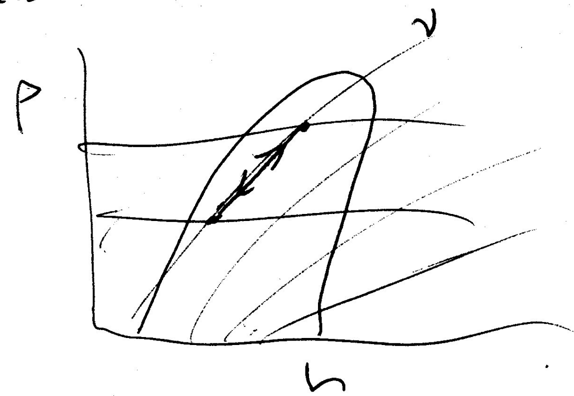
L.3 Liquid Vapor PCMs

Now let's take a loop at liquid vapor PCMs. Everybody's favorite: water. At STP, both phase change temperatures are unusable, too high or too low. The vaporization temperature can be lowered by lowering pressure, but at lower pressures, the bulkiness of water vapor is extreme. Even at standard pressures the bulkiness is extreme. This is one that would have to be dumped to the atmosphere every time it is used. There would be no recovery of water, but that also means no recovery time would be required. Unfortunately, it would usually take power to lower the pressure, unless the aircraft was set to the right altitude. The benefits would be a huge enthalpy of vaporization, cheap to replace, huge boiling heat transfer coefficients, and worldwide availability, necessary because it would have to be refilled every flight.

Another option is ammonia. Still fairly cheap, you could dump it to the atmosphere, and you could set the temperature of vaporization by pressurizing the container on land during refills. Carbon dioxide is a similar possibility, but has lower enthalpies, much higher pressures, and is cheaper.

If one was going to try to recover the vapor, then in a passive system (one big sealed tank) the evaluation is easy. There will be a constant mass in a constant volume, so the pressure and temperature will move along a line of constant bulkiness (specific volume) within the vapor dome. You would need a P-h diagram or similar diagram that shows lines of constant density within the vapor dome, something can be made from the NIST-PROP database to your own specification. Typically, boiling and condensing heat transfer coefficients are very large, so the area of conduction can be very small. The downside is the slipping temperature of evaporation, and the volume of vapor. Once one finds their respectable limits on high and low temperature, then a difference in enthalpies between the two temperature extremes can be read off the chart. The density picked should have the high temperature near the saturated vapor line, this usually gets a curve with the widest enthalpy difference within those two temperatures, and you get the most bang for each pound of refrigerant. I haven't tried it, but it may be possible to get more energy density picking a denser bulkiness curve, so one could balance between a large vapor box and a heavy vapor box.

'liquid-vapor mix " " "
the same volume → moves along lines of constant bulkiness
the vapor dome.
Not constant T
Not constant P



D,

Based on memory, the PDT design problem of 50 kW for 30 sec would take a carbon dioxide container about 2 ft^3 in size, but at 1000 psi, would probably be too heavy. I haven't looked at ammonia, but with the lower pressures and higher enthalpy, should be more doable.

If you already have a vapor compression cycle cooling system, heat pipe, or heat pump, this governs your transient response of the system. So, if your existing can handle the transients, good. If it cannot, then add a bigger accumulator tank until Δ h (Btu/lb) * mass of freon in system (lb) absorbs the transients.

L.4 THE LIQUID COOLING LOOP

Say we have a cooling loop with negligible upstream and downstream mixing of the fluid in the pipes. The fluid travels as distinct slugs of temperatures. All mixing happens in the mission equipment bath, where all electronics are.

Figure L1

Assuming the fluid in the mission equipment bath is very well mixed, then we can do a little thermodynamic analysis with ease.

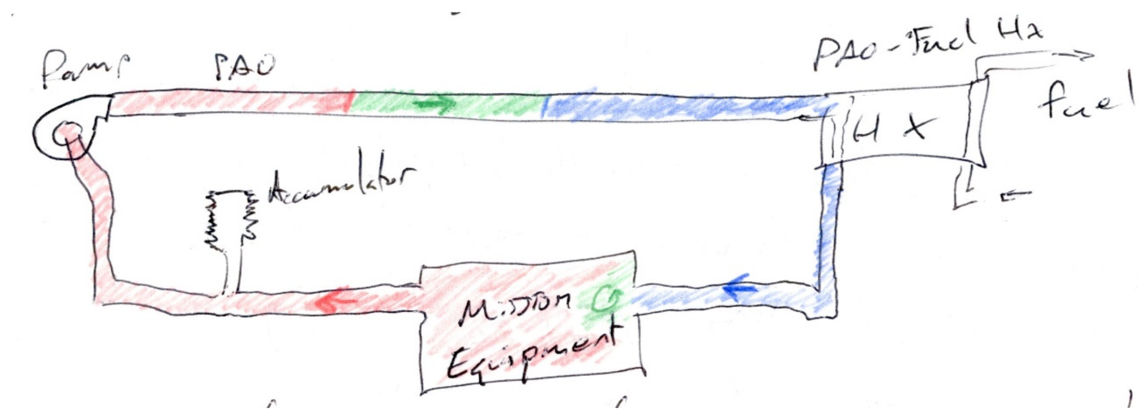


FIGURE 32

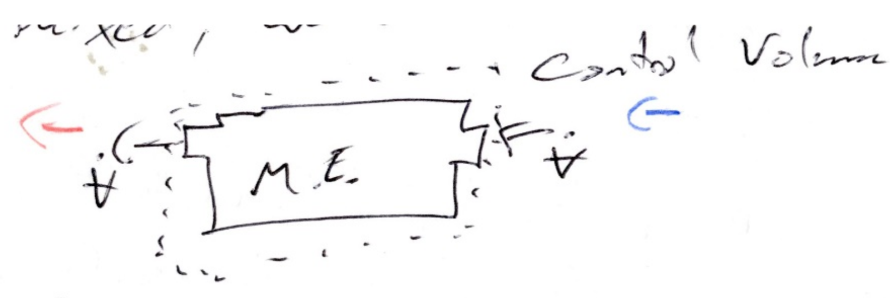


FIGURE 33

Figure L2 Control Volume of bath

Conservation of Energy

$$\dot{E}_{in} - \dot{E}_{out} + \dot{E}_{gen} = \dot{E}_{stored} \tag{L.12}$$

$$\dot{E}_{in} = \dot{\forall}_{in} c_p T_{in} \tag{L.13}$$

$$\dot{E}_{out} = \dot{\forall}_{out} c_p T_{out} \tag{L.14}$$

The heat load from electronics

$$\dot{E}_{gen} = q_{elec} \tag{L.15}$$

$$\dot{E}_{stored} = \forall_{bath} c_p \dot{T}_{bath} \tag{L.16}$$

If the bath is well mixed, then

$$T_{out} = T_{bath} \tag{L.17}$$

And with constant volume, density

$$\dot{\forall}_{out} = \dot{\forall}_{in} \tag{L.18}$$

And we take c_p to be the volumetric heat capacity (Btu/gal°F) because it works better, on account of it being more constant than mass specific heat.

The bath volume itself helps dampen transients, so very quick cycles may be sufficiently quelled, and no action would be required.

Previously, in section 4, I had assumed no mixing because it was easier and I had no idea how much happened in a real system. I got the familiar graphs of section 4, with very blocky temperature profiles. Adding a mixing bath will ease the edges, to varying degrees depending on the sizes in the system.

Most likely it will be preferable that the bath temperature and outgoing temperature are the same. So, substituting in all variables

$$\dot{\forall}_{in} c_p T_{in} - \dot{\forall}_{out} c_p T_{out} + q_{elec} = \forall_{bath} c_p \dot{T}_{bath} \tag{L.19}$$

$$\dot{\forall}_{in} c_p T_{in}(t) - \dot{\forall}_{in} c_p T_{out} + q_{elec}(t) = \forall_{bath} c_p \frac{d}{dt} T_{out} \tag{L.20}$$

q_{elec}, T_{in} are both independent functions of time. Dividing by c_p, $\forall_{bath}$

$$\frac{\dot{\forall}_{in}}{\forall_{bath}} T_{in}(t) - \frac{\dot{\forall}_{in}}{\forall_{bath}} T_{out} + \frac{q_{elec}(t)}{\forall_{bath} c_p} = \frac{d}{dt} T_{out} \tag{L.21}$$

Rearranging,

$$\frac{d}{dt}T_{out} + \frac{\dot{\forall}_{in}}{\forall_{bath}}T_{out} = \frac{\dot{\forall}_{in}}{\forall_{bath}}T_{in}(t) + \frac{q_{elec}(t)}{\forall_{bath}c_p} \tag{L.22}$$

And we see that we have a nonhomogeneous linear differential equation with constant coefficients.

So, let's quickly get through this, I don't know if anyone else cares. Let

$$P(t) = \frac{\dot{\forall}_{in}}{\forall_{bath}} \tag{L.23}$$

$$q(t) = \frac{\dot{\forall}_{in}}{\forall_{bath}}T_{in}(t) + \frac{q_{elec}(t)}{\forall_{bath}c_p} \tag{L.24}$$

$$\int P(t)dt = \int \frac{\dot{\forall}_{in}}{\forall_{bath}}dt = \frac{\dot{\forall}_{in}}{\forall_{bath}}t \tag{L.25}$$

let

$$\mu(t) = exp\,(\frac{\dot{\forall}_{in}}{\forall_{bath}}t) \tag{L.26}$$

And so, finally,

$$T_{out} = \frac{1}{exp\left(\frac{\dot{\forall}_{in}}{\forall_{bath}}t\right)}\left[\int exp\left(\frac{\dot{\forall}_{in}}{\forall_{bath}}t\right)\left(\frac{\dot{\forall}_{in}}{\forall_{bath}}T_{in}(t) + \frac{q_{elec}(t)}{\forall_{bath}c_p}\right)dt + C\right] \tag{L.27}$$

Where C is a constant determined by initial conditions, which should just be the initial temperature of the bath. Since the design goal is to have T_{in} be a constant low temperature, it should be a good approximation to make it a constant until you have the real or simulated data that can be put into the equation piecewise. Until then, it is just fine for back of the envelope calculations. q_{elec} is probably usually a step input for the short period high power stage, so it can be modeled with… The Heaviside Step Function! Represented as either h(t) or u(t)

$$\begin{aligned} q_{elec}(t) = \; & q_{low} + (q_{high} - q_{low})(u(t-0) - u(t-30) \\ & + u(t-150) - u(t-180) + \cdots\,) \end{aligned} \tag{L.28}$$

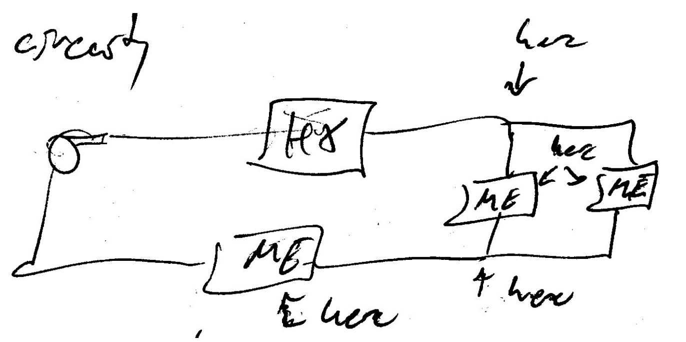
and integrated stepwise. Eventually T_{in} can be found from T_{in} from a while back in time and everything that happens to it, as seen in section 4. A word of caution, those equations may be valid if handled carefully, but work with constant heat exchanger effectiveness. I like that one equation so much I'll write it again!

$$T_{out} = exp\left(\frac{-\dot{\forall}_{in}}{\forall_{bath}}t\right)\left[\int exp\left(\frac{\dot{\forall}_{in}}{\forall_{bath}}t\right)\left(\frac{\dot{\forall}_{in}}{\forall_{bath}}T_{in}(t) + \frac{q_{elec}(t)}{\forall_{bath}c_p}\right)dt + C\right] \quad (L.29)$$

It's so beautiful. It could also be extrapolated to multiple inlets at a tee in a piping network, but those probably have negligible bath volume. So, for a Tee it would just be the weighted average

$$T_{out} = \frac{\dot{\forall}_{1in}T_{1in} + \dot{\forall}_{2in}T_{2in}}{\dot{\forall}_{out}} \quad (L.30)$$

But the beautiful equation can be applied to several systems in any series-parallel network of cooling loops.



From the combination of all uses of these two equations one could still get the temperature profile that will be hitting a pcm reservoir as a function of time for the first "loop cycle" or time to circulate around the loop, which may get confusing if there are parallel loops.

L.5 Modeling the Solid/Liquid Reservoir

We are already familiar with UA values in heat exchangers, and effectiveness for steady state heat transfer. But if we fill one side of a heat exchanger with wax, things will change. Starting from cold conditions, as hot incoming fluid will keep its Nusselt # correlation and so the same heat transfer coefficient that should be previously known for these flow conditions. For heat transfer across a plate (equal areas throughout)

$$\frac{1}{UA} = \frac{1}{hA} + \frac{t_m}{k_m A} + \frac{\Delta x}{k_{pcm} A} \tag{L.31}$$

$$\frac{1}{U} = \frac{1}{h} + \frac{t_m}{k_m} + \frac{\Delta x}{k_{pcm}} \tag{L.32}$$

$$\frac{1}{U} = \frac{k_m \cdot k_{pcm} + t_m \cdot h \cdot k_{pcm} + \Delta x \cdot h \cdot k_m}{h \cdot k_m \cdot k_{pcm}} \tag{L.33}$$

$$U = \frac{h \cdot k_m \cdot k_{pcm}}{k_m \cdot k_{pcm} + t_m \cdot h \cdot k_{pcm} + \Delta x \cdot h \cdot k_m} \tag{L.34}$$

At the start, $\Delta x = 0$, so this reduces to

$$U = \frac{h \cdot k_m}{k_m + t_m \cdot h} \tag{L.35}$$

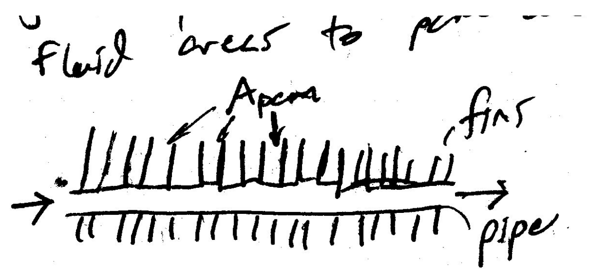
And the far side is at a constant temperature.

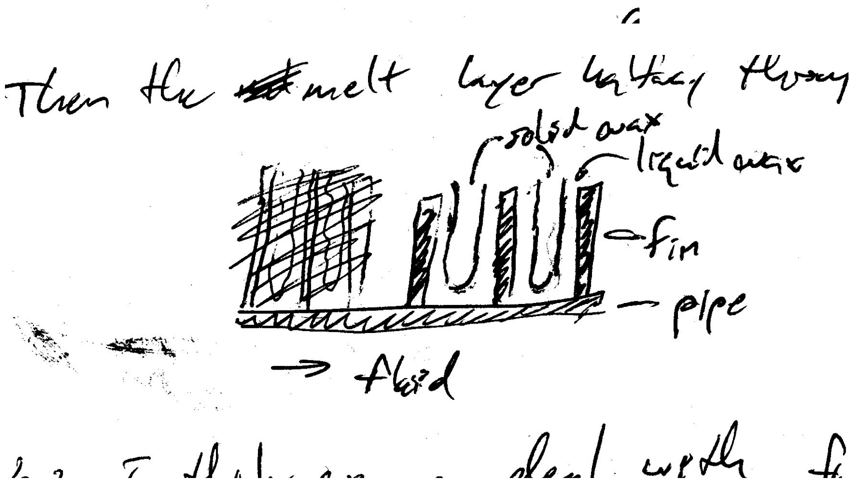
After enough time, when a pcm volume is probably worthless, the coolant wall will be at a constant temperature. This will happen when Δx is very large, so

$$U = \frac{k_{pcm}}{\Delta x} \tag{L.36}$$

Note: this plane wall ignores 2nd dimensional effects, such as heat transfer in the y-direction and cooled fluid hitting the downstream part of the wall. I don’t want to come up with an analytical solution to that mess. That will have to be done using Finite Element methods, speaking of which, mine on MATLAB haven’t panned out yet. I don’t think I have enough time to get that running all the way, even though I have data from the tests to input the incoming temperatures with.

Now we can deal with finned surfaces protruding in the PCM. If the finned meal has relatively low resistance, say, even from the base to the full tip of the fin, such that almost no temperature difference exists in order to push that much heat through, I'd say we could treat the entire metal surface as… *ISOTHERMAL!* For example, a fin .005" thick, 3" long, 3" wide with 1°F across it, would actually be terrible at pushing all the heat through to the tip. We probably can't get away with isothermal, 100% efficient fins.





If we could, however, we could ignore specific geometry effects and straight up compare areas.

$$\frac{1}{UA} = \frac{1}{hA_f} + \frac{t_m}{k_m A_m} + \frac{\Delta x}{k_{pcm} A_{pcm}} \tag{L.37}$$

I believe t_m, A_m are not straightforward, but change as the liquid layer grows. We would hope that as the liquid layer grows it would be the dominant resistance, but that would mean our PCM heat exchanger is probably well past its useful effectiveness. We could probably replace them with a single term that changes as a function of heat absorbed, which would have to be determined through experiment or modeling. Probably has a maximum and minimum value at the beginning and end of the melt, with a smooth function between. Anyways, at the beginning convective resistance dominates until conduction gets lousy enough.

$$\frac{1}{UA} = \frac{k_m A_m \cdot k_{pcm} A_{pcm} + t_m \cdot hA_f \cdot k_{pcm} A_{pcm} + \Delta x \cdot hA_f \cdot k_m A_m}{hA_f \cdot k_m A_m \cdot k_{pcm} A_{pcm}} \tag{L.38}$$

$$UA = \frac{hA_f \cdot k_m A_m \cdot k_{pcm} A_{pcm}}{k_m A_m \cdot k_{pcm} A_{pcm} + t_m \cdot hA_f \cdot k_{pcm} A_{pcm} + \Delta x \cdot hA_f \cdot k_m A_m} \tag{L.39}$$

Note that t_m, A_m counteract each other, the ratio is what matters. As $t_m \rightarrow 0$ or $A_m \rightarrow \infty$, the middle term in the denominator drops out and A_m can be neglected, as well as k_m. As t_m gets large or A_m gets small, or k_m is small, the resistance would have to be accounted for, though A_m is probably A_f at first and A_{pcm} at last, so is bounded, oh, who knows. Probably can brush it under the rug, unless gallium is used, then it ought to matter a lot. Say it is small, then

$$UA = \frac{hA_f \cdot k_{pcm} A_{pcm}}{k_{pcm} A_{pcm} + \Delta x \cdot hA_f} \tag{L.40}$$

Our Biot #, ratio of two terms in the denominator

$$Bi = \frac{\Delta x h A_f}{k_{pcm} A_{pcm}} \tag{L.41}$$

relates the magnitudes of the convection and conduction. Big magnitude means all resistance exists with the other guy, and you can ignore the original guy. If Bi > say, 12, then same q invokes minimal ΔT across convection, and all resistance could be considered to be in conduction. The Biot # at the end of the melt tells you which area should be increased to get more heat flux. So, in this respect it would have been nice to have the metal thickness and area as well, as it would be possible to get a large portion of resistance in metal, and this Biot number would not tell you that. To beat a dead horse, if Bi = 12, then

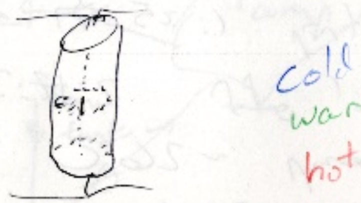
$$\frac{12}{13} \Delta T_{total} = \Delta T_{conduction} \tag{L.42}$$

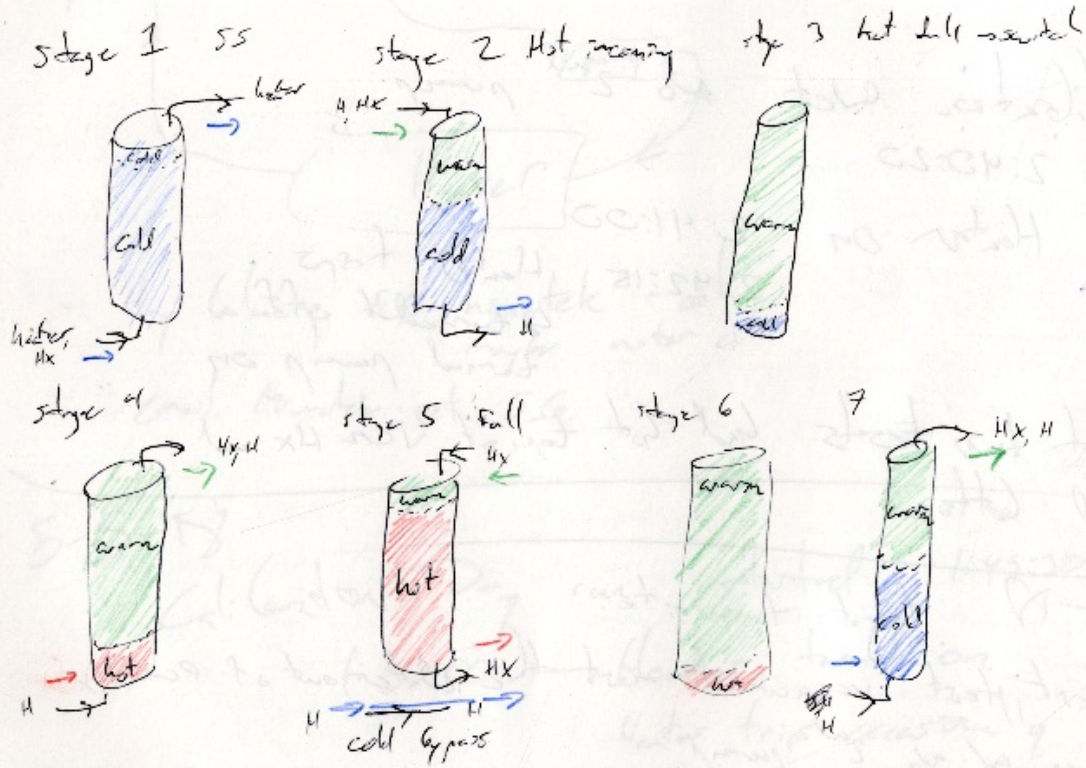
So far, we have assumed the PCM has negligible specific heat, this ought to make the model underperform reality, and so design conservatively. But if we need a PAO-PCM HX that is, say, 70% effective at the end of the heating stage, it will be more effective up until that point. It may have to start at, say, 90% effective and cruise on down to 70%. This means the PCM may absorb more that than planned for, and pushes a non-conservative design. Absorbing more heat grows Δx faster, so by the time the last amount of hot fluid passes through, the PCM may have melted enough such that either: Δx is so large the effectiveness has gone down to, say, 50%, and the test fails, *or*, may have melted completely and the effectiveness takes a sharp drop to near zero.

L.6 PAO Storage

→ Competing thermal storage idea.

PAO storage tank → like accumulator w/ PAO both sides





stage 1 3kW cooling → low temps throughout

heater → pump → HX → tank → heater

Stage 2 50kW

heater → p → HX → tank top, tank bottom → heater stage 3 50 kW switch

heater → pump → tank bottom, tank top → HX → h.

stage 4 same, maybe 3 is redundant

H → P → ~~TT~~ TB ; TT ~~TB~~ → HX → H

stage 5 bypass

H → P → H ; TB → HX → TT

6 TB empty → switch

stage 7

H → P → TB ; TT → HX → H

Once separation plate reaches end, there shall
of gap around edges to continue flow

May require second pump, or spring pulling sepa
bottom. → the restriction in HX amounts to some
spring may not work well

for our example problem, $T_{\infty} = 50°F$ → T_{cold}

$T_{hot} \approx 140°F$

→ tank volume is minimum ~~9gpm~~ 9gpm/hr

$= \frac{9}{4}$ gallons $2\frac{1}{4}$ gallon ~ 17 lbs of PAO

+ weight of tank,
+ weight of extra
keep flow going while

Alternative Option is feasible,

APPENDIX M. PAO PROPERTIES

Temp	Density	K	Specific Heat	Dynamic Viscosity	Kinematic Viscosity	Kinematic Viscosity	Volumetric Specific Heat
°F	lb/ft³	Btu/hr/ft/°F	Btu/lb°F	lb_f*sec/ft^2	ft^2/hr	cST	Btu/gal°F
-65	52.58	0.087	0.456	0.015	47.40	1223.11	3.20
-55	52.35	0.087	0.460	0.011	19.89	513.18	3.22
-45	52.11	0.086	0.465	0.006			3.24
-35	51.87	0.086	0.470	0.004	8.40	216.69	3.26
-25	51.64	0.086	0.475	0.003			3.28
-15	51.40	0.086	0.480	0.002	3.67	94.6	3.29
-5	51.17	0.085	0.484	0.001			3.31
0	51.05	0.085	0.487	0.001	2.12	54.6	3.32
5	50.93	0.085	0.489	0.001			3.33
15	50.70	0.084	0.494	0.001	1.30	33.67	3.35
25	50.46	0.084	0.499	0.000			3.36
35	50.22	0.084	0.503	0.000	0.77	19.87	3.38
45	49.99	0.083	0.508	0.000			3.40
55	49.76	0.083	0.513	0.000		12.32	3.41
75	49.28	0.083	0.522	0.000			3.44
95	48.80	0.082	0.532	0.000	0.22	5.78	3.47
115	48.33	0.081	0.542	0.000			3.50
135	47.85	0.081	0.551	0.000	0.13	3.48	3.53
155	47.39	0.080	0.561	0.000			3.55
175	46.92	0.080	0.570	0.000	0.09	2.31	3.58
195	46.45	0.079	0.580	0.000			3.60
210	46.10	0.079	0.587	0.000	0.07	1.68	3.62