Boeing Heat Exchanger

Engineering Summary

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Top Level Design Summary

For the Boeing Heat exchanger demonstrator, the main problem statement stems from the requirement of separating a flammable refrigerant by way of a firewall and must be capable of transferring heat out of the system. The team is tasked with designing and manufacturing a liquid-liquid heat exchanger to transfer heat from the coolant to the flammable refrigerant, which will be represented as ice water. The team's solution consists of a 2-pass shell and tube heat exchanger which can be seen below in Figure 1. An exploded view of the heat exchanger is shown in Figure 2

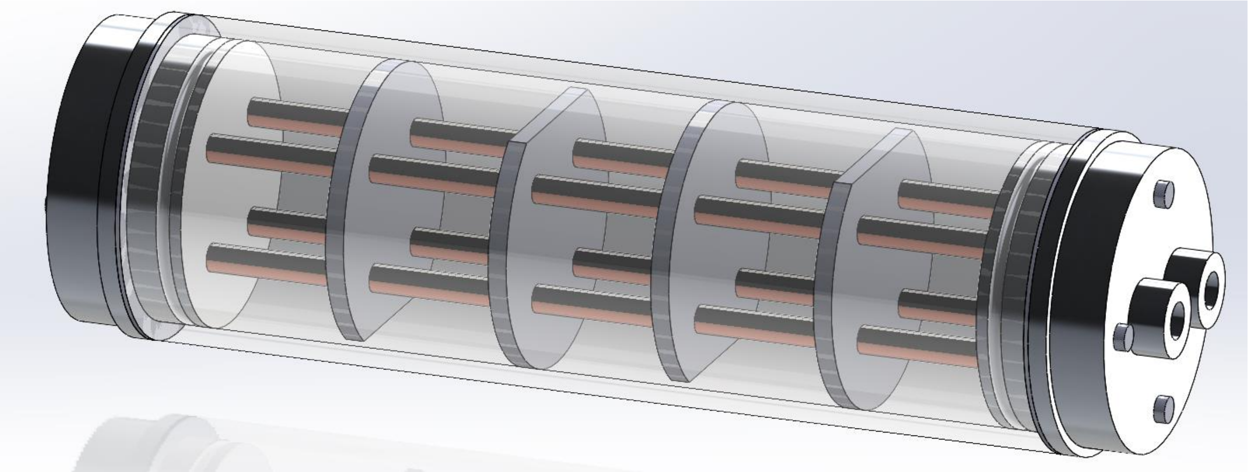


Figure 1: Heat Exchanger CAD

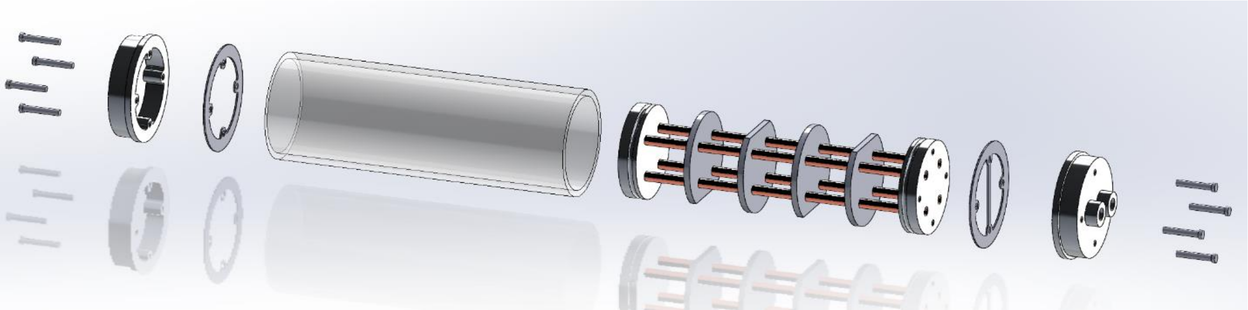
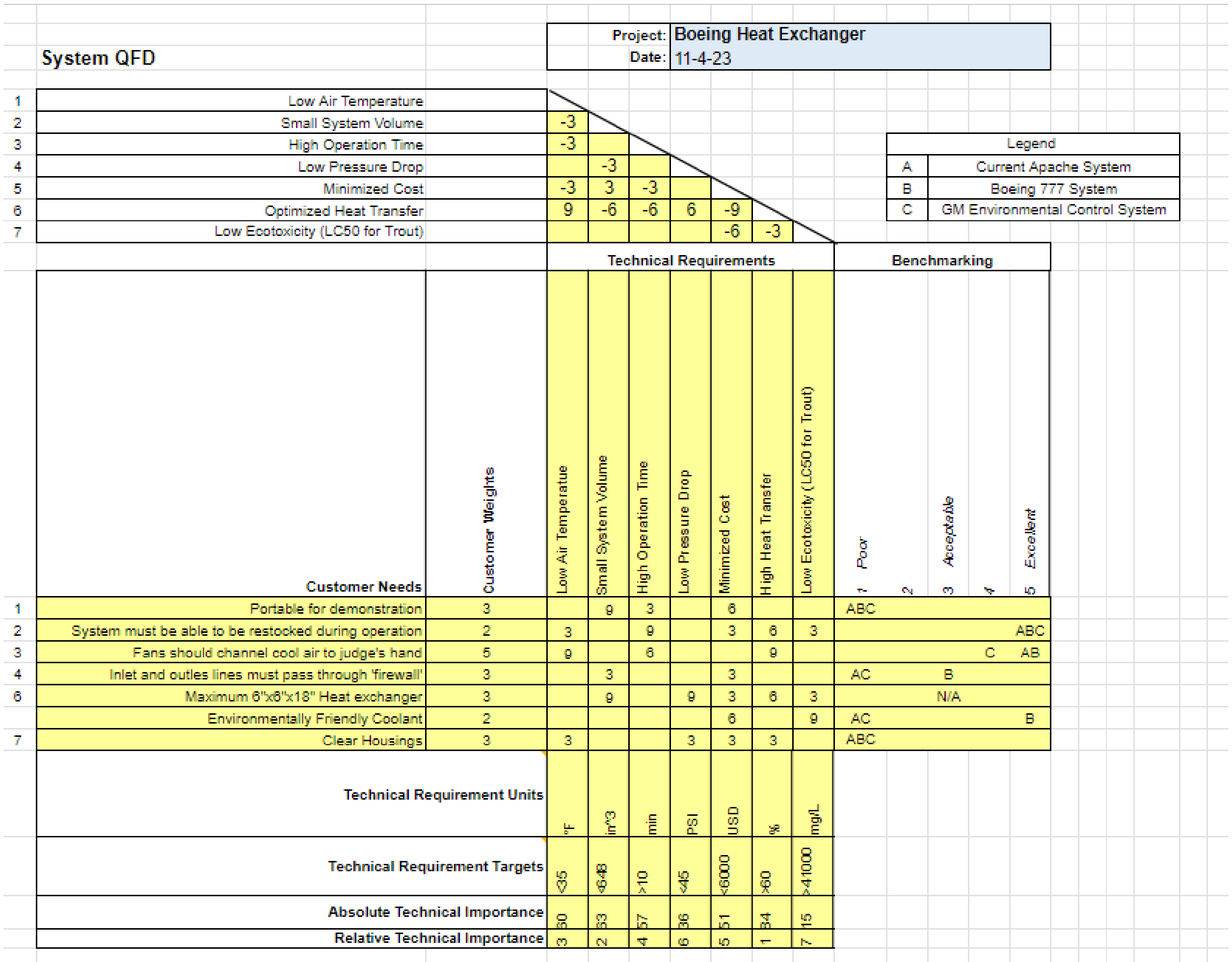


Figure 2: Heat Exchanger Exploded View

The QFD for the project is shown in Figure 3, including the customer and technical requirements identified by the team and client. Additionally, the technical targets for the design is shown at the bottom.



System QFD

Project: Boeing Heat Exchanger
Date: 11-4-23

		Low Air Temperature	Small System Volume	High Operation Time	Low Pressure Drop	Minimized Cost	Optimized Heat Transfer
1	Low Air Temperature						
2	Small System Volume	-3					
3	High Operation Time	-3					
4	Low Pressure Drop		-3				
5	Minimized Cost	-3	3	-3			
6	Optimized Heat Transfer	9	-6	-6	6	-9	
7	Low Ecotoxicity (LC50 for Trout)					-6	-3

Legend	
A	Current Apache System
B	Boeing 777 System
C	GM Environmental Control System

			Technical Requirements							Benchmarking				
	Customer Needs	Customer Weights	Low Air Temperatue	Small System Volume	High Operation Time	Low Pressure Drop	Minimized Cost	High Heat Transfer	Low Ecotoxicity (LC50 for Trout)	1 Poor	2	3 Acceptable	4	5 Excellent
1	Portable for demonstration	3		9	3		6			ABC				
2	System must be able to be restocked during operation	2	3		9		3	6	3					ABC
3	Fans should channel cool air to judge's hand	5	9		6			9					C	AB
4	Inlet and outles lines must pass through 'firewall'	3		3			3			AC		B		
6	Maximum 6"x6"x18" Heat exchanger	3		9		9	3	6	3			N/A		
	Environmentally Friendly Coolant	2					6		9	AC				B
7	Clear Housings	3	3			3	3	3		ABC				
	Technical Requirement Units		°F	in^3	min	PSI	USD	%	mg/L					
	Technical Requirement Targets		<35	<648	>10	<45	<6000	>60	>41000					
	Absolute Technical Importance		30	33	57	36	51	34	15					
	Relative Technical Importance		3	2	4	6	5	1	7					

Figure 33: QFD

Customer Requirements

The fans must blow cold air onto a judges hand: The liquid coolant going into the radiator must be cold enough from the heat exchanger that when it conducts heat transfer to the air it will drop its temperature to a felt degree

Inlets and outlets must pass through a firewall: This system models the Apache Helicopter vapor compression system which will use HFO-1234yf which is a combustible refrigerant so a firewall will need to be put in place in order to protect the pilot.

Heat exchange must fit inside a 6''x6''x18'' box: The liquid-to-liquid heat exchanger must meet this maximum sizing requirement. No weight constraint is specified.

Clear housing for the heat exchanger: Since the client requires the design of a demonstrator, they have requested a clear housing to make the internal design of the heat exchanger visible.

Portable: The client would like to see the demonstration of the heat exchanger in person, so the system must be easy to transport and can be set up and operational within 1 hour.

It must work for at least 30 minutes: The heat exchanger must continue to expel cold coolant to the radiator for a duration of 30 minutes.

Engineering Requirements

Low Air Temperature: The system must channel cold air to an observer's hand. As such, the design must be able to bring the fan air temperature as low as possible. At minimum, air temperature should be at least 5 °C below room temperature while the target temperature is 10 °C.

Small System Volume: The entire system must be easily transportable for the purposes of transferring from NAU to the Boeing facility in Mesa, AZ. Thus, all components should fit within a compact volume. When packed, the system should fit within a volume of 1 m^3. Additionally, the client defined a maximum area of 6"x6"x18" for the liquid-to-liquid heat exchanger, which represents an additional volume constraint specific to that subsystem.

High Operation Time: The client specified that the system must operate for at least 30 minutes without melting all the ice within the system. As such, the design should be fully functional for more than half an hour.

Low Head Losses: An important consideration for heat exchanger design is the amount of head losses in the form of pressure drop. Client listed a specified pump that supplied 45 psi of pressure head. Given this limit, the team set the maximum allowable pressure loss to be 40 psi.

Low Cost: The team has a budget of $5,000 provided by Boeing with an additional $1,000 in the form of a VA grant. All testing equipment, prototyping, and material costs must collectively be below the $6,000 of funding available. To appropriately allocate the budget, the material cost of the system should be less than $1,000, excluding expenditures for ice.

High Heat Transfer: The liquid-to-liquid heat exchanger is the main component to be designed. Since the team is unable to control the efficiency of the radiator (coolant-air heat exchanger), the liquid-to-liquid heat exchanger should be as efficient as possible. The team has defined a minimum effectiveness of 50% with a target of 60%.

Environmental Impact: Since the premise of this project is to model a more environmentally friendly ECS system, environmental impact is important. Materials such as coolant should be selected with environmental concerns in mind such as toxicity. Since the coolant has the potential to spread to natural bodies of water, environmental impact is measured in the lethal concentration (LC50) for trout. Thus, coolant should have a high LC50 of at least 41,000 mg/L for aquatic life.

Summary of Standards, Codes, and Regulations

Table 1 below shows several standards that the design adheres to. Each purchased part has at least one standard to ensure quality of manufacturing. Manufactured parts like the baffles have standards for assembly to optimize performance. Standards are listed as applicable to each part.

Table 1: Standards, Codes, and Regulations

Part	Standard	Description
Acrylic Tube	ISO 9001, 13485	Quality Management System
Thermocouple	ISO 17025	Quality Standard for Testing & Calibration Labs
Baffles	TEMA RCB-4.5.1	Minimum Spacing for segmental baffles
Pressure Transducers	ASTM F2070-00	Standard Specification for Transducers, Pressure and Differential, Pressure, Electrical and Fiber-Optic
Aluminum Sheet	ISO 18842	Method For the Determination of Tapped and Untapped Density
O-ring	USA AS568	Standard Sizing for general industrial and aerospace applications
Pipe Threads	ASME B1.20.1	Foundational standards on NPT pipe thread dimensions

In the heat exchanger demonstrator system, the ‘worst case’ load condition that will be referenced in the FoS table will stem from the pump’s maximum pressure which would be caused by some form of blockage or flow impediment. As such, the minimum factors of safety will be evaluated at the maximum pressure of 45 psi. These performance parameters are organized by topic.

Calculations

Ice Amount: To fully calculate the amount of ice the team would need for the system to run for 30 minutes both the amount of energy and heat transfer entering the system needed to be calculated. For easy access there are 2 MATLAB codes located in appendix A that goes into detail on the shape factor conduction used for the heat transfer through the cooler wall and an energy balance to find the amount of ice needed. The team found the amount of ice required to be 160lbs of ice based on the heat gained through the cooler wall and the energy needed to be absorbed from the warm water inlet.

Heat Transfer: The heat transfer from the cold fluid utilizes the NTU method to approximate the heat transfer of the system based on several parameters like hot and cold flow rates, specific heat capacities, temperatures and more. This method is calculation heavy and for ease of prototyping was computed in an excel document to easily vary parameters. The Excel document can be seen in appendix B.

Sensor Error Propagation: All sensors come with a resolution uncertainty, accuracy and other uncertainties that can contribute to incorrect data. This section will deal with the heat transfer and actual effectiveness seen below.

$$q = \rho \dot{V} C_p (T_2 - T_1) \tag{1}$$

$$\varepsilon = \frac{q}{q_{max}} \tag{2}$$

All components in equation 1 will have some design stage uncertainty. The design stage uncertainty equation can be seen below with u_o being the uncertainty due to accuracy and u_c being the resolution uncertainty.

$$u_d = (u_o^2 + u_c^2)^{1/2} \tag{3}$$

With the design stage uncertainty now the error propagation can be found below. The d in the subscript changed to denote the design stage uncertainty of that particular variable.

$$u_t = \left[\left(\frac{\partial q}{\partial \rho} u_\rho\right)^2 + \left(\frac{\partial q}{\partial \dot{V}} u_V\right)^2 + \left(\frac{\partial q}{\partial \rho} u_\rho\right)^2 + \left(\frac{\partial q}{\partial T_1} u_{T_1}\right)^2 + \left(\frac{\partial q}{\partial T_2} u_{T_2}\right)^2\right]^{1/2} \tag{4}$$

solving the partial derivatives in equation 4 results in the following uncertainty in q (i.e ∓ the resultant value).

$$u_t = \left[\left(\dot{V} C_P \left(T_2 - T_{(1)}\right) \cdot u_\rho\right)^2 + \left(\rho C_p (T_2 - T_1) \cdot u_V\right)^2 + \left(\rho C_p (T_2 - T_1) \cdot u_{C_p}\right)^2 + \left(\rho \dot{V} C_p \cdot u_{T_2}\right)^2 + \left(-\rho \dot{V} C_p \cdot u_{T_1}\right)^2\right]^{1/2} \tag{5}$$

since q_{max} is theoretical and no error propagation will be seen the effectiveness uncertainty can be simple found with

$$Uncertainty = u_t / q_{max} \tag{7}$$

Determining the propagated error due to sensors is a critical step in selecting sensors that will reach the teams accuracy goals.

Pressure Losses: Head loss analysis informed the design by determining the drop in pressure across both sides of the heat exchanger. A CFD analysis was performed in ANSYS Fluent to find the pressure drop in both the shell and the tubes. The CAD model was imported and meshed using Fluent. Calculations were done using the pressure-based solver, absolute velocities, and steady state flow. The energy model and k-epsilon viscous model were used in the simulation as well. The tube side and shell side pressure change relative to 45 psi can be seen in Figures 4 and 5 respectively.

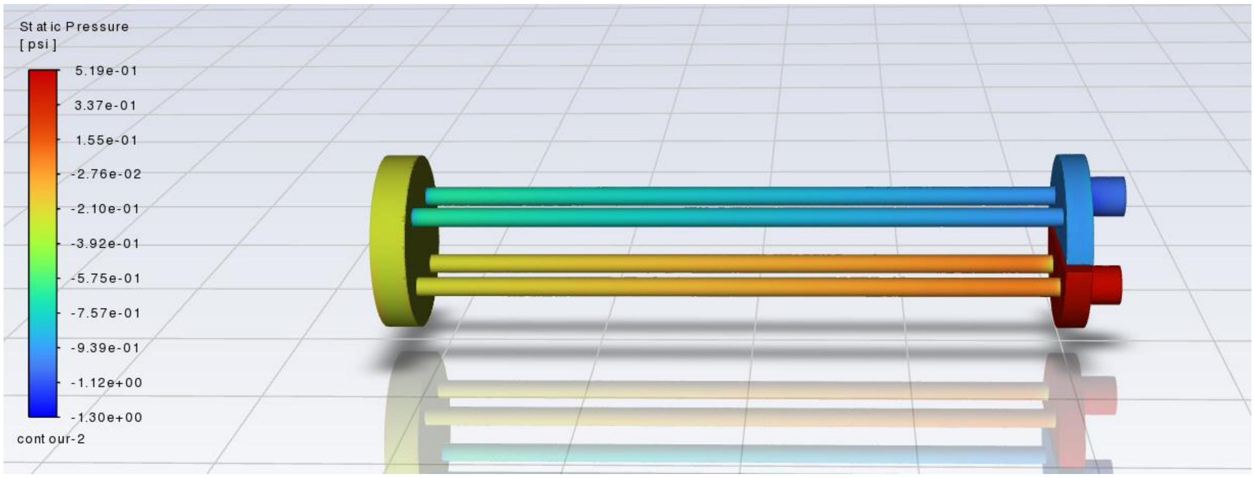


Figure 44: Tube Side Static Pressure

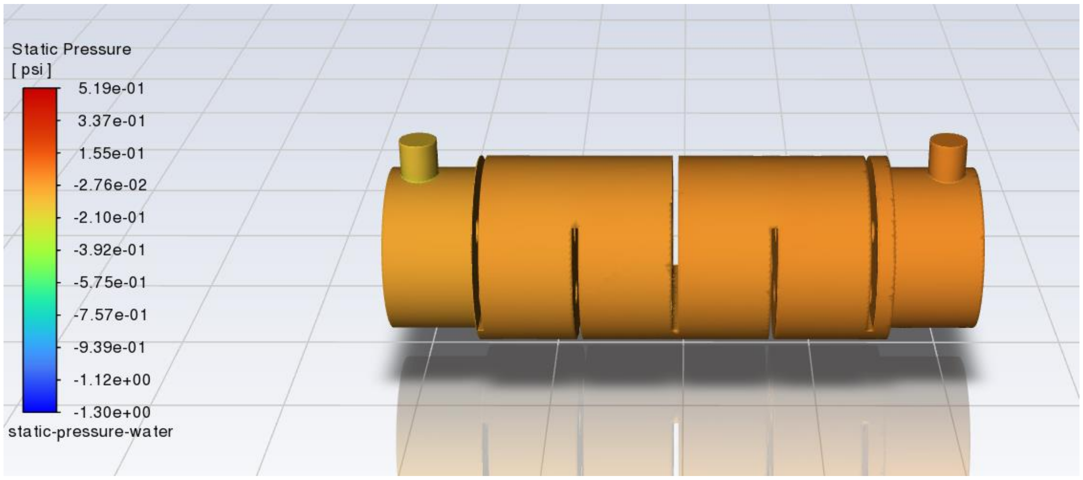


Figure 55: Shell Side Static Pressure

The pressure drops for the tube and shell were 1.05 psi and .134 psi. The stated maximum allowable pressure drop from the QFD is 30 psi, so the heat exchanger design meets all pressure requirements.

Factors of Safety

Table 2 categorizes the factor of safeties by each load case and sub system. Since the pressures are limited based on the pump selection the team focused its attention on calculating all types of pressure cases to make sure the pump would not shut off due to too much pressure exertion.

Table 2: Factor of Safety

Sub-System	Part	Load Case	Material	FoS
Heat transfer	Assembly	N/A	Aluminum Copper	7.4
Pressure losses	Hot Side	45 PSI	Propylene Glycol Solution	28.6
	Cold Side	45 PSI	Ice Water	224
System Flow	Clear tubing	45 PSI	PVC Clear	1.2
	Brass Bung	45 PSI	Brass	26.6
	Sensor Mount	45 PSI	CPVC	3.8
	Hose Clamp	45 PSI	Stainless Steel	4.4

The smallest factors of safety in the design are in the clear tubing between sections due to the tubing being rated up to 55 psi. However, this is not a concern since the pump cannot provide enough head to reach that pressure. All other parts that have fluid flowing through them can sufficiently withstand much higher pressures than the load case. Additionally, the heat exchanger geometry provides more than enough NTU to reach maximum efficiency and achieves a pressure drop within the target range. Thus, this design is unlikely to fail at these points.

Flow Charts and other Diagrams

Figures 6 and 7 below demonstrate the fluid flow for the coolant and ice water respectively. The coolant loop flows through the pump and into the tube side of the heat exchanger. The cooled fluid then enters the radiation before returning to a holding tank. The ice water loop flows through the pump and into the shell of the heat exchanger before returning to the holding tank.

Pressure transducers are located at the inlets and outlets of the heat exchanger to measure the pressure drop across both sides. Thermocouples are located at the inlets and outlets of both the heat exchanger and the holding tanks to measure the temperature of both fluids as they flow through the system.

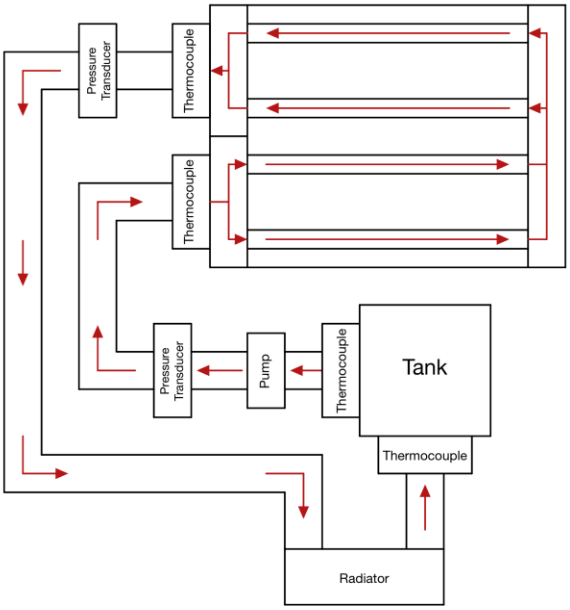


Figure 6: Coolant Loop

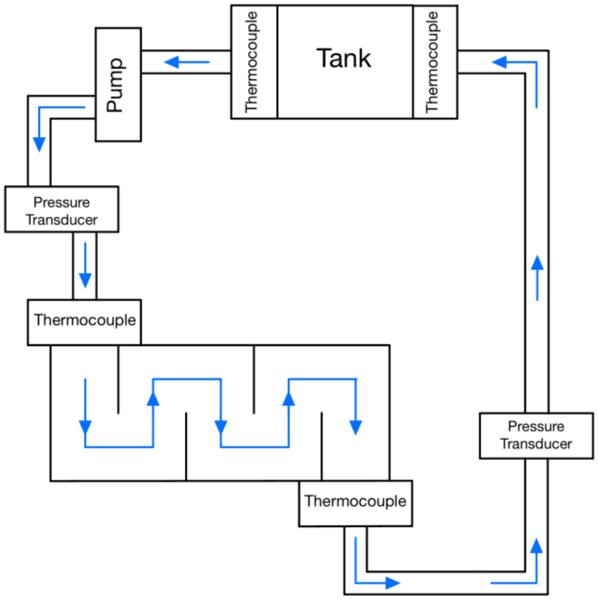


Figure 7: Ice Water Loop

Figure 8 shows the functional decomposition of the whole system. This model shows the material and energy flows as they move through each component of the system. The model also shows the signals that will be used to perform measurements on the fluid such as temperature and pressure readings.

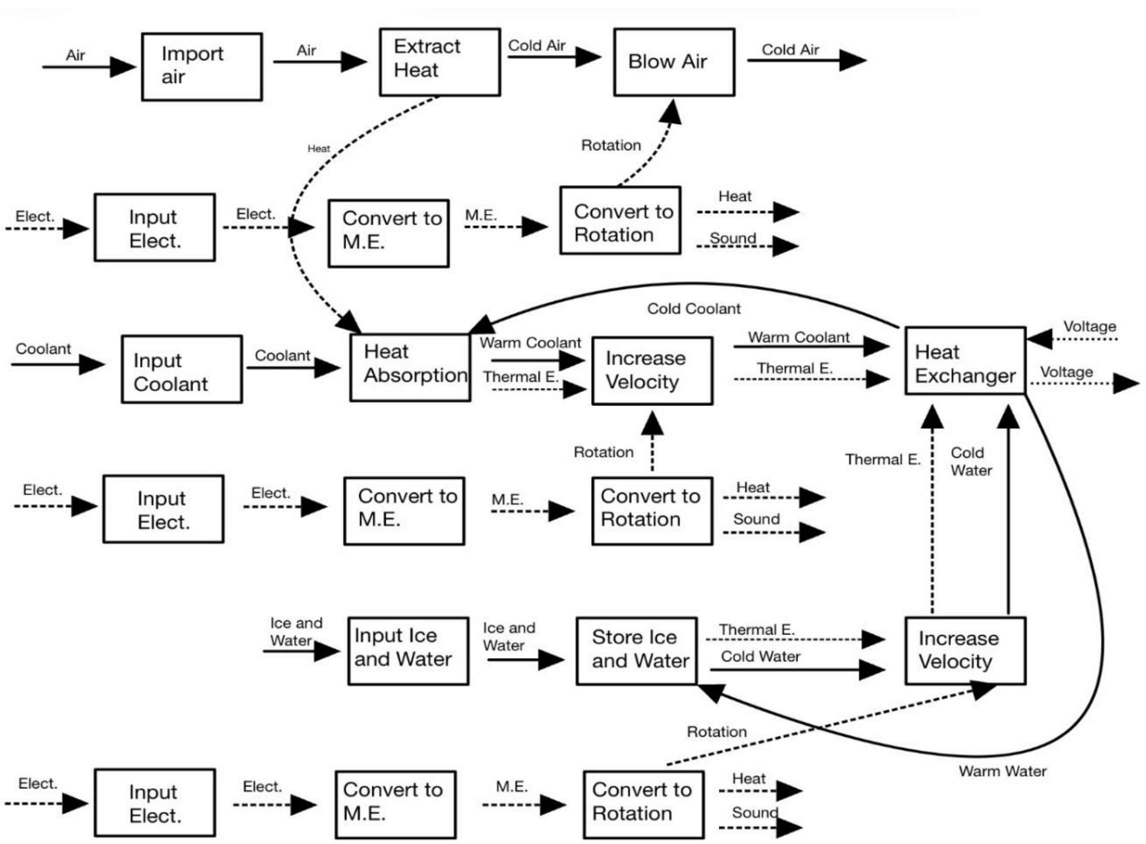


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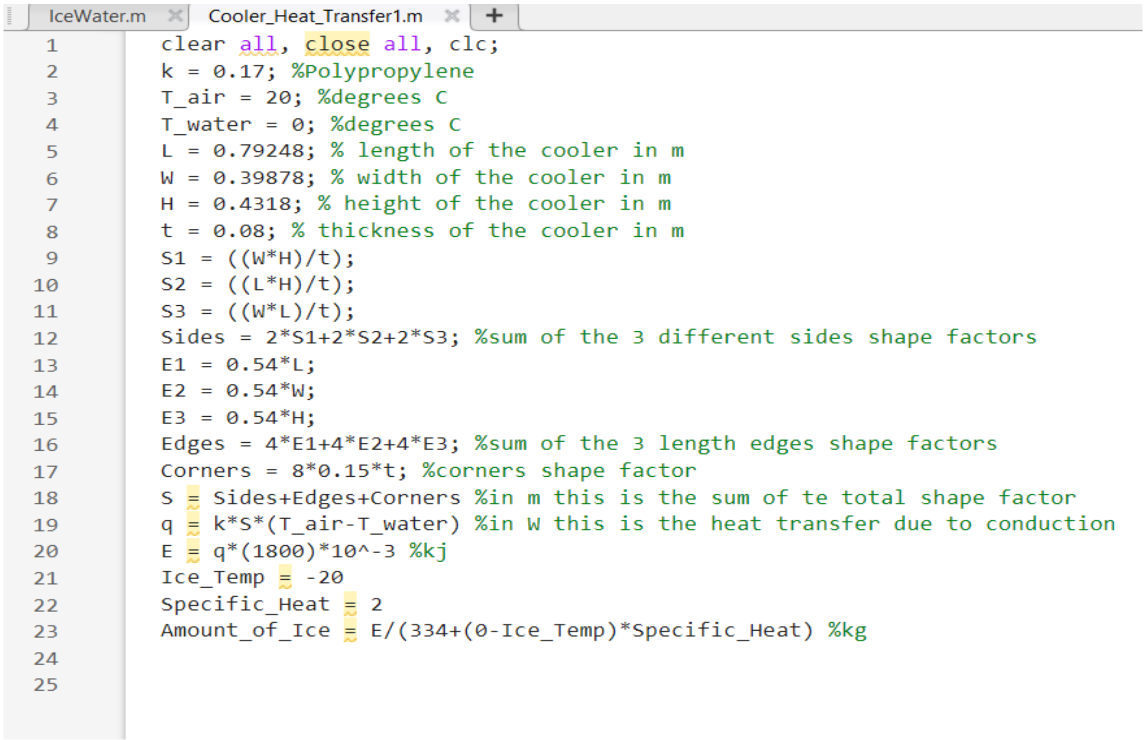
Figure 8: Functional Decomposition

Future Plans

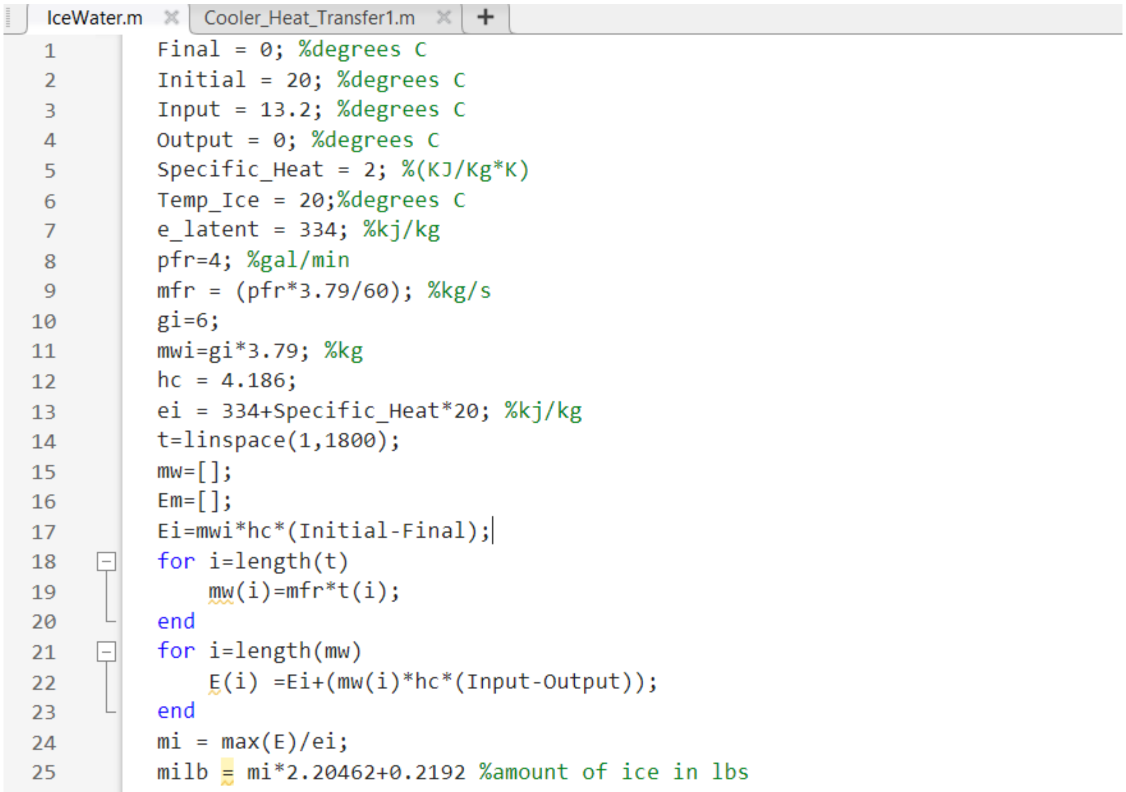
Going forward, the team will focus on manufacturing all the required components and feels no additional analysis is required for this process. Additionally, one more stage of prototyping will be started to determine the gasketing material/number of clamping locations (screws used) and the gasket thickness needed to work at our desired pressure. The testing will likely be done at a pressure of 45 psi and will be held for the desired 30-minute operation time to ensure no failure will occur.

Appendix A: Ice Amount MATLAB Codes



```
clear all, close all, clc;
k = 0.17; %Polypropylene
T_air = 20; %degrees C
T_water = 0; %degrees C
L = 0.79248; % length of the cooler in m
W = 0.39878; % width of the cooler in m
H = 0.4318; % height of the cooler in m
t = 0.08; % thickness of the cooler in m
S1 = ((W*H)/t);
S2 = ((L*H)/t);
S3 = ((W*L)/t);
Sides = 2*S1+2*S2+2*S3; %sum of the 3 different sides shape factors
E1 = 0.54*L;
E2 = 0.54*W;
E3 = 0.54*H;
Edges = 4*E1+4*E2+4*E3; %sum of the 3 length edges shape factors
Corners = 8*0.15*t; %corners shape factor
S = Sides+Edges+Corners %in m this is the sum of te total shape factor
q = k*S*(T_air-T_water) %in W this is the heat transfer due to conduction
E = q*(1800)*10^-3 %kj
Ice_Temp = -20
Specific_Heat = 2
Amount_of_Ice = E/(334+(0-Ice_Temp)*Specific_Heat) %kg
```

Figure 9: Cooler Heat Transfer Code



```
Final = 0; %degrees C
Initial = 20; %degrees C
Input = 13.2; %degrees C
Output = 0; %degrees C
Specific_Heat = 2; %(KJ/Kg*K)
Temp_Ice = 20;%degrees C
e_latent = 334; %kj/kg
pfr=4; %gal/min
mfr = (pfr*3.79/60); %kg/s
gi=6;
mwi=gi*3.79; %kg
hc = 4.186;
ei = 334+Specific_Heat*20; %kj/kg
t=linspace(1,1800);
mw=[];
Em=[];
Ei=mwi*hc*(Initial-Final);
for i=length(t)
    mw(i)=mfr*t(i);
end
for i=length(mw)
    E(i) =Ei+(mw(i)*hc*(Input-Output));
end
mi = max(E)/ei;
milb = mi*2.20462+0.2192 %amount of ice in lbs
```

Figure 10: Amount of Ice Code

Appendix B: Heat Transfer Excel Sheet

	A	B	C	D	E	F	G	H	I	J	K	L	M	N
5	OD(in,m)	0.5	0.0127											
6	Shell Diameter (in,m)	3.75	0.09525											
7	Baffle Spacing (in,m)	2	0.0508											
8	Clearance (in,m)	0.75	0.01905											
9														
10	# of Tubes	2												
11	# of Passes	2			Rows 1-12 for CAD									
12	L (Length per tube) (in,m)	10	0.254		chambers .5 in for coolant									
13	L (total) (in,m)	40	1.016		baffel width 1/8 in									
14	L (entrance region) (in,m)	3.75	0.09525											
15	L (Fully developed) (in,m)	32.5	0.8255											
16														
17	Ice Water Properties (Shell)									Effectiveness-NTU Relations				
18	Density (kg/m^3)	998		V (m/s)	0.05215450431		h (W/m^2K)	2607.661973		U Term 1	0.0005113137159			
19	Viscocity (N*s/m^2)	1.76E-03		Vmax (m/s)	0.16					U Term 2	0.000004601463879			
20	Cp (kJ/kgK)	4.211		Re (ext)	1,126.77					U Term 3	0.000382035393			
21	kf (W/mK)	0.598		C	0.27					U	1113.647043			
22	Pr (T∞)	13.6		m	0.63					NTU	41.73791256			
23	Pr (Ts)	6.62		Nu	55.38011214					ε	0.6230403232			
24	C cold	1.060562584												
25														
26	Propylene Glycol Properties (Tube)									Heat Transfer				
27	Density(m^3/kg)	1038		V (m/s)	1.770804729		h (W/m^2K)	2617.558525		qmax	18640.33923			
28	Viscocity (N*s/m^2)	0.00505		Re (int)	3466.902537					q	11613.68298			
29	Cp (kJ/kgK)	3.558		f	0.0654					ΔT,h (C)	12.46080646	**Th,o (C)**	**7.539193537**	
30	k (W/mK)	0.393		Nu	63.44082686					ΔT,c (C)	10.95049283	**Tc,o (C)**	**10.95049283**	
31	Pr	45.71984733												
32	C hot	0.9320169614												
33														