SHELL-AND-TUBE HEAT EXCHANGER DESIGN

MET 440 – Heat Transfer

Robert Hoke, Louis-Khanh Le

Mechanical Engineering Technology

Old Dominion University

Dr. Orlando M. Ayala H.

Department of Mechanical Engineering Technology

Old Dominion University

Submitted December 13, 2018

TABLE OF CONTENTS

LIST OF FIGURES AND TABLES

ABSTRACT

Newport News Shipbuilding hired our manufacturing company to design and build a shell-and-tube heat exchanger to cool liquid ammonia with liquid water. An engineering process was utilized to research a solution in agreement with their initial design constraints. Short-term considerations such as cost, manufacturability, and environmental aspects, were all considered while future circumstances such as expected maintenance schedules and product lifespan were also assessed into our decision making. The end result was a single shell, four-pass shell-and-tube heat exchanger design that cools liquid ammonia in copper-nickel alloy tubes by flowing water inside a galvanized steel shell.

PROJECT BREAKDOWN

INITIAL DESIGN CRITERIA AND PHILOSOPHY

The shell-and-tube heat exchanger must be adequately designed to cool 320,000 lb/hr of liquid ammonia from 122°F to 86°F with liquid water at 50°F. In addition, the water can only reach a maximum temperature of 80°F. The design must utilize commercial tubing with a BWG gauge of 16, and the maximum velocity inside these tubes could not exceed 10 ft/s to limit erosion. Due to space limitations, the maximum length was required to be 24 feet and fins cannot be used. Lastly, the pressure drop on both sides of the heat exchanger could not exceed 10 psi with a baffle cut of 25%.

JOB SITE LOCATION

The heat exchanger will be operating at the shipyards of Newport News Shipbuilding. Special design considerations were made for environmental factors such as saltwater conditions and frigid climate during the winter.

HEAT EXCHANGER MATERIALS

Galvanized steel was selected for our shell material. The metal is dipped in a protective coating of zinc to prevent corrosion due to rust. This easy to weld, low-cost option is ideal for ease of manufacturing by welding, minimal maintenance, and an expected lifespan of up to 75 years before replacement is necessary.

Copper-nickel alloy was selected for our tubing material. Pure copper by itself is easily susceptible to corrosion, particularly from ammonia. However, by combining copper's excellent thermal conductivity with the durability and corrosion resistance of nickel, this alloy is an excellent cost-effective and low-maintenance option for the internal workings of our system.

FLUIDS CHARACTERISTICS

Aside from ambient air, there are two fluids in the heat exchanger system: ammonia (NH_3) and water (H_2O). The ammonia used in this system is the liquid form of pure ammonia gas.

Ammonia was selected to flow within the tubes while water flows in the heat exchanger shell and cools the tubes. This was done because while ammonia poses no immediate fire danger due to its high ignition temperature (1,204°F), the gas is very pungent and a respiratory irritant. Exposure to high concentrations causes burning of the nose, throat, and lungs, with prolonged exposure leading to difficulty in breathing and death due to respiratory failure.

This design consideration was also done in preparation for servicing. Should the shell need basic maintenance, the only fluid technicians would be in contact with is water inside the shell. Only for more severe repairs, such as replacing copper tubing, will technicians encounter ammonia.

SOURCES

Bayazıtoğlu Yıldız, and Özışık M. Necati. A Textbook for Heat Transfer Fundamentals. Begell House, Inc., 2012.

PRELIMINARY DRAWINGS AND SKETCHES

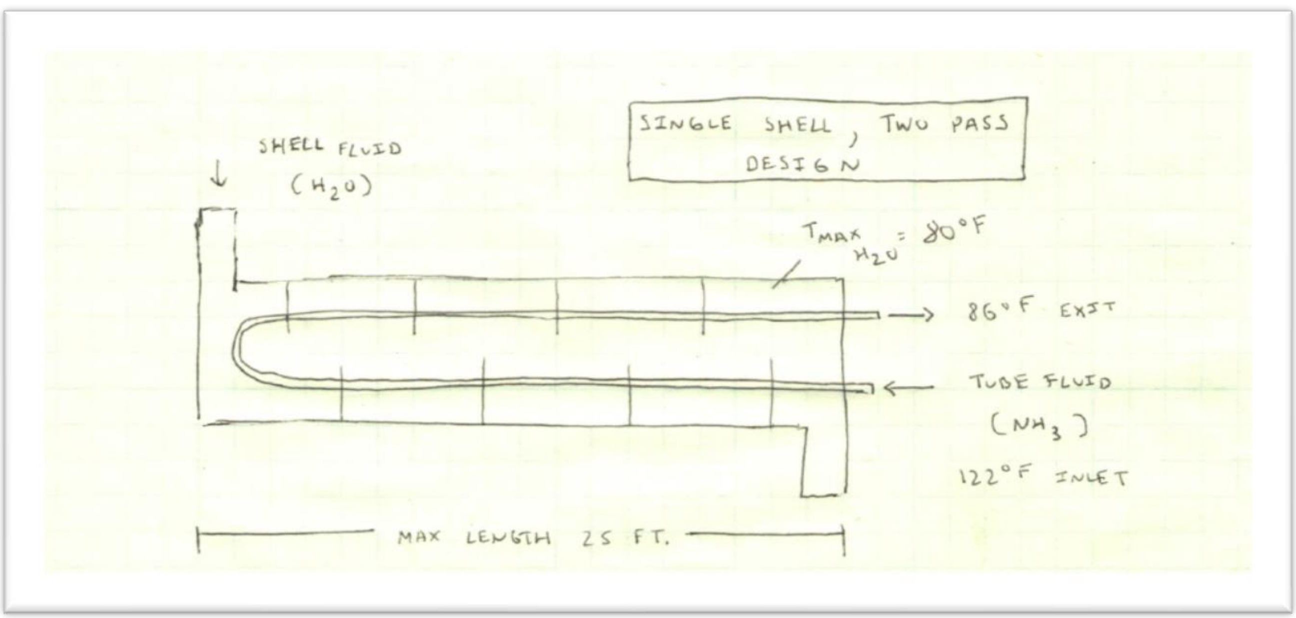


Figure 1. *Preliminary Heat Exchanger Drawing*

METHODOLOGY

All initial design criteria were given in U.S. customary units. For ease of calculations and consistency, all units were converted to SI and will be converted back to customary for the heat exchanger data sheet.

Our first task was to calculate the heat exchanger area using the following equation:

(Eq. 1)

$$\boxed{Q = U_O A_O (MLDT) \times F}$$

Since the overall heat transfer coefficient (U_O) is dependent on the design of the heat exchanger, that value was assumed and checked later with another equation.

The area of the heat exchanger was found using the Eq. 2, where N_t is the number of tubes.

(Eq. 2)

$$\boxed{A_O = \pi D_O L \times N_t}$$

By fixing the values for length (L) and tube outer diameter (D_O), the number of tubes was approximated using Eq. 1.

The shell diameter (D_S) was selected using Table 8.3 from the appendix. The pitch (P_T), number of passes (N_P), and tube configuration were fixed in order to more easily adjust the number of tubes in the system.

The spacing between each baffle (B) was determined using the following relationship:

(Eq. 3)

$$\boxed{0.4D_s < B < 0.6D_s}$$

The next check was to ensure the velocity (v_{tube}) in each tube was less than 10 ft/s (3.048 m/s). This was accomplished by use of Eq. 4:

(Eq. 4)

$$\boxed{\dot{m} = \rho v_{tube} A_{total}}$$

Using the known inner diameter (D_i), the area for each tube is:

(Eq. 5)

$$\boxed{A_{tubes} = \frac{\pi}{4} {D_i}^2}$$

Using the area from each tube found above, the total area of the tube network was calculated using:

(Eq. 6)

$$\boxed{A_{total} = \frac{N_f}{N_p} \frac{\pi}{4} {D_i}^2}$$

Now, the overall heat transfer coefficient is recomputed using the Eq. 7 below and compared with Eq. 1:

(Eq. 7)

$$\boxed{\frac{1}{U_O} = \frac{A_O}{A_i}\left(\frac{1}{h_i} + R_{fi}\right) + A_O R_w + R_{fo} + \frac{1}{h_o}}$$

Most of these values, like areas and fouling factors, are given or can be easily computed. However, some values, such as the heat transfer coefficient inside the shell (h_o) and the conductive resistance of the tube (R_w), are much more complex and required a closer look, as shown below.

To calculate the heat transfer coefficient inside the shell (h_o), the following Nusselt Number correlation was used:

(Eq. 8)

$$\boxed{Nu = 0.023 Re^{0.8} Pr^{0.3}}$$

To calculate the conductive resistance of the tube (R_w), the following equation was utilized:

(Eq. 9)

$$\boxed{R_w = \frac{\ln\left(\frac{r_o}{r_i}\right)}{2\pi k_{tube} N_t L}}$$

The equations were compared to each other using an iteration process. Once the two overall heat transfer coefficients were found to be within 10% of each other, the iteration process was stopped.

The pressure drop on both sides of the heat exchangers was then checked so that it would not exceed 10 psi. First up was the pressure drop inside the shell, shown in Eq. 10:

(Eq. 10)

$$\boxed{\Delta P_{shell} = \frac{f(G_s)(N_b + 1)D_s}{2\rho D_e \theta_s}}$$

Where:

(Eq. 11)

$$\boxed{\theta_s = \frac{\mu_b}{\mu_w}^{0.14}}$$

And the friction factor could be found using:

(Eq. 12)

$$\boxed{f = \exp[0.576 - 0.19\ln(Re)]}$$

The pressure drop inside the tube was then checked using Eq. 13:

(Eq. 13)

$$\boxed{\Delta P_{tube} = [4f\left(\frac{LN_p}{d_i}\right) + 4N_p]\frac{\rho U_m}{2}}$$

The final step was calculating effectiveness. This can be found using the following few equations, the first of which calculates our heat capacity rate (C), where little “c” signifies the cold fluid, which for us was water:

(Eq. 14)

$$\boxed{C_c = \dot{m}_c C_{p,c}}$$

And in Eq. 15, “h” signifies the hot fluid, which for us was ammonia:

(Eq. 15)

$$\boxed{C_h = \dot{m}_h C_{p,h}}$$

This can be used to find the minimum heat capacity, as shown in Eq. 16:

(Eq. 16)

$$\boxed{C_{min} = \min(C_h, C_c)}$$

From here, we can calculate the maximum heat flux (q) value using the inlet temperature of our hot fluid, ammonia, and the inlet temperature of our cold fluid, water:

(Eq. 17)

$$\boxed{q_{max} = C_{min}(T_{hi} - T_{ci})}$$

In order to find our expected heat flux value, using the specific heat and temperature inlet and outlet of our hot fluid, ammonia:

(Eq. 18)

$$\boxed{q = \dot{\mathrm{m}}_h C_{p,h}(T_{hi} - T_{ho})}$$

Using Eq. 16 and 17, we can finally use the following to calculate our effeciveness:

(Eq. 18)

$$\boxed{\varepsilon = \frac{q}{q_{max}}}$$

DESIGN CALCULATIONS

The initial step required selecting an initial U value:

$$Q = U_O A_O (MLDT) \times F$$

$$4031.122\ kW = U_O (206.940\ m^2)(19.676℃) \times 0.9$$

$$\boxed{U_O = 1076.889 \frac{W}{m^2.K}}$$

The initial U value was compared to the second calculated U value found in the following. Once these two U values were within 10% of each other, the iteration process was concluded. We were able to come within 0.05% after two iterations.

$$\frac{1}{U_O} = \frac{A_O}{A_i}\left(\frac{1}{h_i} + R_{fi}\right) + A_O R_w + R_{fo} + \frac{1}{h_o}$$

$$\frac{1}{U_O} = \frac{(210.141\ m^2)}{(171.717\ m^2)}\left(\frac{1}{5218.892} + 1.76 \times 10^{-4}\right) + (210.141\ m^2)(3.80 \times 10^{-7}) + 3.522 \times 10^{-4} + \frac{1}{19061.33}$$

$$\frac{1}{U_O} = 0.000929$$

$$\boxed{U_O = 1075.986\ \frac{W}{m^2.K}}$$

The pressure drop in the shell needed to be evaluated to not exceed 10 psi:

$$\Delta P_{shell} = \frac{f(G_s)(N_b + 1)D_s}{2\rho D_e \theta_s}$$

$$\Delta P_{shell} = \frac{(0.226)(1015.933)(13 + 1)(0.787\ m)}{2(1000.775 \frac{kg}{m^3})(0.024\ m)(1.005)}$$

$$\boxed{\Delta P_{shell} = 68.911\ kPa\ or\ 7.716\ psi}$$

The pressure drop in the tube also needed to be calculated to not exceed 10 psi:

$$\Delta P_{tube} = [4f\left(\frac{LN_p}{d_i}\right) + 4N_p]\frac{\rho U_m}{2}$$

$$\Delta P_{tube} = \left[4(0.02)\left(\frac{(5.486\,m)(2)}{0.016\,m}\right) + 4(2)\right]\frac{\left(580.99\,\frac{kg}{m^3}\right)(1.113\,\frac{m}{s})}{2}$$

$$\boxed{\Delta P_{tube} = \ 22.954 \text{ kPa or } 3.331 \text{ psi}}$$

Finally, the effectiveness of the heat exchanger system was considered:

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

$$\varepsilon = \frac{4031.122\,kW}{8062.245\,kW}$$

$$\boxed{\varepsilon = 0.5}$$

FINAL DRAWINGS

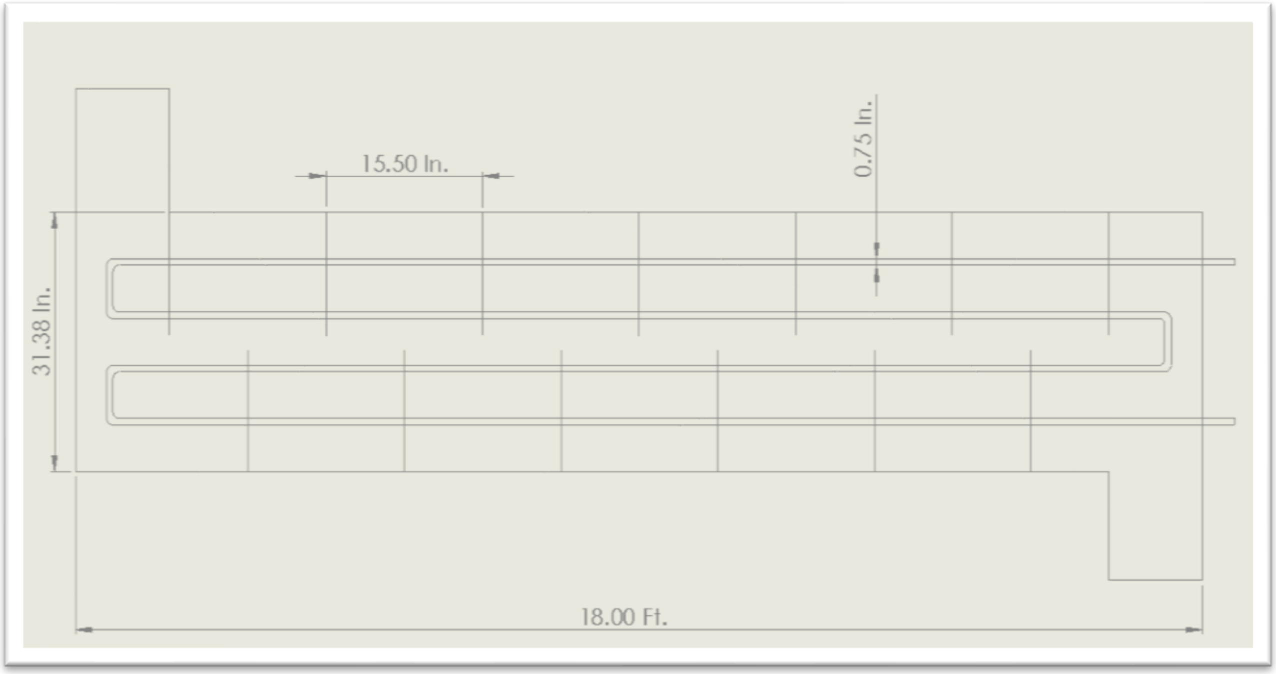
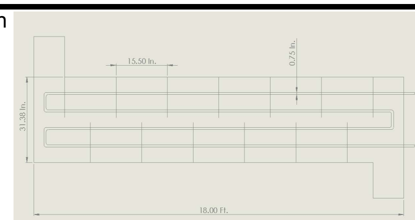


Figure 2. *Final Heat Exchanger Drawing in SolidWorks (Single Shell, Four Pass Heat Exchanger)*

SHELL-AND-TUBE HEAT EXCHANGER

				CLIENT Newport News Shipbuilding	EQUIP. NO	PAGE 1
REV	PREPARED BY	DATE	APPROVAL	W.O.	REQUISITION NO.	SPECIFICATION NO.
0	Robert Hoke and Khanh Le	12/12/18				
1				UNIT AREA	PROCURED BY	INSTALLED BY
2						

1	Size		TEMA Type	AEL		Connected in (series/parallel)	parallel		
2	Surface per Unit		ft²	Shells per Unit	1	Surface per Shell		ft²	
3	**Performance of One Unit**								
4	Fluid Allocation			Shell Side			Tube Side		
6	Fluid Name			Water			Ammonia		
7	Flow Total	lb/h		624888			320000		
8	Vapor	lb/h	(in/out)						
9	Liquid	lb/h	(in/out)						
10	Steam	lb/h	(in/out)						
11	Water	lb/h	(in/out)						
12	Noncondensable	lb/h	(in/out)						
13	Temperature (In/Out)	°F	(in/out)	50	72		122	86	
14	Density	lb/ft3		62.476	62.476		36.269	36.269	
15	Viscosity	cP							
16	Molecular Weight, vapor								
17	Specific Heat	Btu/lb-°F		1.0005	1.0005		1.1939	1.1939	
18	Thermal Conductivity	Btu/h-ft-°F		.339	.339		.285	.285	
19	Latent Heat	Btu/lb							
20	Inlet Pressure	psigg	(inlet)						
21	Velocity	ft/s		.019			3.65		
22	Press Drop Allow/Calc	psig		10	7.72		10	3.33	
23	Fouling Factor	ft²-h-°F/Btu		.002			.001		
24	Heat Exchanged	Btu/hr		13662636.08		LMTD (corrected) °F	35.417		
25	Service Coeff.	Btu/h-ft²-°F	Dirty			Clean			
26	**Construction Data for One Shell**								
27				Shell Side	Tube Side	Sketch			
28	Design/Test Press	psigg							
29	Design Temperature	°F							
30	No. Passes per Shell			2					
31	Corrosion Allowance	in							
32	Connections Size & Rating	In	inches						
33		Out							
34		Intermediate							
35	**Tubes** No.	OD, in	0.75	Gauge	16	Length, ft.	18	Pitch layout, deg.	Square Pitch
36	Type			Material	Copper nickel alloy			Pitch ratio	
37	**Shell**	OD, in	31.375	ID, in	31	Material	Galvanized Steel		
38	Channel or Bonnet	OD, in	31.375	Thick	.375 in	Channel Cover			
39	Tubesheet Type								
40	Floating Heat Cover					Impingement Protection			
41	Baffles Cross (number)	13		% Cut (d)	25%			Spacing C/C, in	15.5
42	Baffles Long			Seal Type No					
43	Supports Tube			U-Bend				Type	
44	Bypass Seal Arrangement					Tube-Tubesheet Joint			
45	Expansion Joint No.					Type			
46	Rho-V2-Inlet Nozzle			Bundle Entrance				Bundle Exit	
47	Gaskets - Shell Side					Tube Side			
48	Floating Heat Cover					Supports			
49	Code Requirements	Follow all applicable codes/standards				TEMA Class			
50	Weight per shell lb			Filled w/water				Bundle	
51									
52	Notes								
53									
54									



DISCUSSION

In this project, the design goal of our heat exchanger was always the same: a simple and basic design. This design route was chosen so that the complex calculations needed to properly design the heat exchanger we not made unnecessarily complicated. These design selections were taken into consideration against others, however, the simple design has similar benefits to the more complex heat exchanger designs that are available. The gain from further complicating the layout of the heat exchanger is relatively negligible compared to the other design considerations, such as the tube size and material.

Throughout the project, several iterations were attempted, however, the best fit case for our heat exchanger came at the smallest available tube diameter. This allows for more heat transfer and a relatively smaller shell diameter, compared to the larger tube and shell diameters. This makes more sense because the larger the tube and shell size, the higher the cost. The materials chosen were relatively inexpensive, however, anything that gets too large becomes more of a custom order and thus increases cost.

FINAL REMARKS

There are many portions of this project where the values are dependent on each other. For example, when changing the initial value for the overall heat transfer coefficient, the number of tubes and the diameter of the shell is directly affected by this change. This phenomenon created early issues in the project as the initial iterations did not seem to converge. However, once the relationship between the different values was found, it was easy to see where our iterations and initial guesses needed to change. This phenomenon of related values is present throughout the design of the heat exchanger. This meant that there was not a large room for error, leading to the large window for percent error. However, with correct parameters, a value that was well within the criteria was able to be calculated. Once the project was into the calculations, most of the data was easily accessible and easy to calculate.

APPENDIX

ADDITIONAL RELATED MATERIALS

Table 1. *Physical Properties of Selected Saturated Liquids*

Table B-2 Physical properties of selected saturated liquids

T, °C	ρ, kg/m^3	c_p, kJ/(kg · °C)	ν, m^2/s	k, W/(m · K)	α, m^2/s	Pr	β, K^{-1}
Ammonia NH_3							
−50	703.69	4.463	0.435×10^{-6}	0.547	1.742×10^{-7}	2.60	
−40	691.68	4.467	0.406×10^{-6}	0.547	1.775×10^{-7}	2.28	
−30	679.34	4.476	0.387×10^{-6}	0.549	1.801×10^{-7}	2.15	
−20	666.69	4.509	0.381×10^{-6}	0.547	1.819×10^{-7}	2.09	
−10	653.55	4.564	0.378×10^{-6}	0.543	1.825×10^{-7}	2.07	
0	640.10	4.635	0.373×10^{-6}	0.540	1.819×10^{-7}	2.05	
10	626.16	4.714	0.368×10^{-6}	0.531	1.801×10^{-7}	2.04	
20	611.75	4.798	0.359×10^{-6}	0.521	1.775×10^{-7}	2.02	2.45×10^{-3}
30	596.37	4.890	0.349×10^{-6}	0.507	1.742×10^{-7}	2.01	
40	580.99	4.999	0.340×10^{-6}	0.493	1.701×10^{-7}	2.00	
50	564.33	5.116	0.330×10^{-6}	0.476	1.654×10^{-7}	1.99	
Carbon dioxide, CO_2							
−50	1,156.34	1.84	0.119×10^{-6}	0.0855	0.4021×10^{-7}	2.96	
−40	1,117.77	1.88	0.118×10^{-6}	0.1011	0.4810×10^{-7}	2.46	
−30	1,076.76	1.97	0.113×10^{-6}	0.1099	0.5133×10^{-7}	2.20	
0	926.99	2.47	0.108×10^{-6}	0.1045	0.4578×10^{-7}	2.38	
10	860.03	3.14	0.101×10^{-6}	0.0971	0.3608×10^{-7}	2.80	
20	772.57	5.0	0.091×10^{-6}	0.0872	0.2219×10^{-7}	4.10	14.00×10^{-3}
30	597.81	36.4	0.080×10^{-6}	0.0703	0.0279×10^{-7}	28.7	
Dichlorodifluoromethane (Freon-12), CCl_2F_2							
−50	1,546.75	0.8750	0.310×10^{-6}	0.067	0.501×10^{-7}	6.2	2.63×10^{-3}
−40	1,518.71	0.8847	0.279×10^{-6}	0.069	0.514×10^{-7}	5.4	
−30	1,489.56	0.8956	0.253×10^{-6}	0.069	0.526×10^{-7}	4.8	
−20	1,460.57	0.9073	0.235×10^{-6}	0.071	0.539×10^{-7}	4.4	
−10	1,429.49	0.9203	0.221×10^{-6}	0.073	0.550×10^{-7}	4.0	
0	1397.45	0.9345	0.214×10^{-6}	0.073	0.557×10^{-7}	3.8	
10	1,364.30	0.9496	0.203×10^{-6}	0.073	0.560×10^{-7}	3.6	
20	1,330.18	0.9659	0.198×10^{-6}	0.073	0.560×10^{-7}	3.5	
30	1,295.10	0.9835	0.194×10^{-6}	0.071	0.560×10^{-7}	3.5	
40	1,257.13	1.0019	0.191×10^{-6}	0.069	0.555×10^{-7}	3.5	
50	1,215.96	1.0216	0.190×10^{-6}	0.067	0.545×10^{-7}	3.5	
Engine oil (unused)							
0	899.12	1.796	0.00428	0.147	0.911×10^{-7}	47,100	
20	888.23	1.880	0.00090	0.145	0.872×10^{-7}	10,400	0.70×10^{-3}
40	876.05	1.964	0.00024	0.144	0.834×10^{-7}	2,870	
60	864.04	2.047	0.839×10^{-4}	0.140	0.800×10^{-7}	1,050	
80	852.02	2.131	0.375×10^{-4}	0.138	0.769×10^{-7}	490	
100	840.01	2.219	0.203×10^{-4}	0.137	0.738×10^{-7}	276	
120	828.96	2.307	0.124×10^{-4}	0.135	0.710×10^{-7}	175	
140	816.94	2.395	0.080×10^{-4}	0.133	0.686×10^{-7}	116	
160	805.89	2.483	0.056×10^{-4}	0.132	0.663×10^{-7}	84	

Table 2. *Physical Properties of Selected Saturated Liquids (continued)*

Table B-2 *(continued)*

T,°C	ρ, kg/m^3	c_p, kJ/(kg · °C)	v, m^2/s	k, W/(m · K)	α, m^2/s	Pr	β, K^{-1}
Sulfur dioxide. SO_2							
−50	1,560.84	1.3595	0.484×10^{-6}	0.242	1.141×10^{-7}	4.24	
−40	1,536.81	1.3607	0.424×10^{-6}	0.235	1.130×10^{-7}	3.74	
−30	1,520.64	1.3616	0.371×10^{-6}	0.230	1.117×10^{-7}	3.31	
−20	1,488.60	1.3624	0.324×10^{-6}	0.225	1.107×10^{-7}	2.93	
−10	1,463.61	1.3628	0.288×10^{-6}	0.218	1.097×10^{-7}	2.62	
0	1,438.46	1.3636	0.257×10^{-6}	0.211	1.081×10^{-7}	2.38	
10	1,412.51	1.3645	0.232×10^{-6}	0.204	1.066×10^{-7}	2.18	
20	1,386.40	1.3653	0.210×10^{-6}	0.199	1.050×10^{-7}	2.00	1.94×10^{-3}
30	1,359.33	1.3662	0.190×10^{-6}	0.192	1.035×10^{-7}	1.83	
40	1,329.22	1.3674	0.173×10^{-6}	0.185	1.019×10^{-7}	1.70	
50	1,299.10	1.3683	0.162×10^{-6}	0.177	0.999×10^{-7}	1.61	
Water, H_2O							
0	1,002.28	4.2178	1.788×10^{-6}	0.552	1.308×10^{-7}	13.6	
20	1,000.52	4.1818	1.006×10^{-6}	0.597	1.430×10^{-7}	7.02	0.18×10^{-3}
40	994.59	4.1784	0.658×10^{-6}	0.628	1.512×10^{-7}	4.34	
60	985.46	4.1843	0.478×10^{-6}	0.651	1.554×10^{-7}	3.02	
80	974.08	4.1964	0.364×10^{-6}	0.668	1.636×10^{-7}	2.22	
100	960.63	4.2161	0.294×10^{-6}	0.680	1.680×10^{-7}	1.74	
120	945.25	4.250	0.247×10^{-6}	0.685	1.708×10^{-7}	1.446	
140	928.27	4.283	0.214×10^{-6}	0.684	1.724×10^{-7}	1.241	
160	909.69	4.342	0.190×10^{-6}	0.680	1.729×10^{-7}	1.099	
180	889.03	4.417	0.173×10^{-6}	0.675	1.724×10^{-7}	1.004	
200	866.76	4.505	0.160×10^{-6}	0.665	1.706×10^{-7}	0.937	
220	842.41	4.610	0.150×10^{-6}	0.652	1.680×10^{-7}	0.891	
240	815.66	4.756	0.143×10^{-6}	0.635	1.639×10^{-7}	0.871	
260	785.87	4.949	0.137×10^{-6}	0.611	1.577×10^{-7}	0.874	
280.6	752.55	5.208	0.135×10^{-6}	0.580	1.481×10^{-7}	0.910	
300	714.26	5.728	0.135×10^{-6}	0.540	1.324×10^{-7}	1.019	

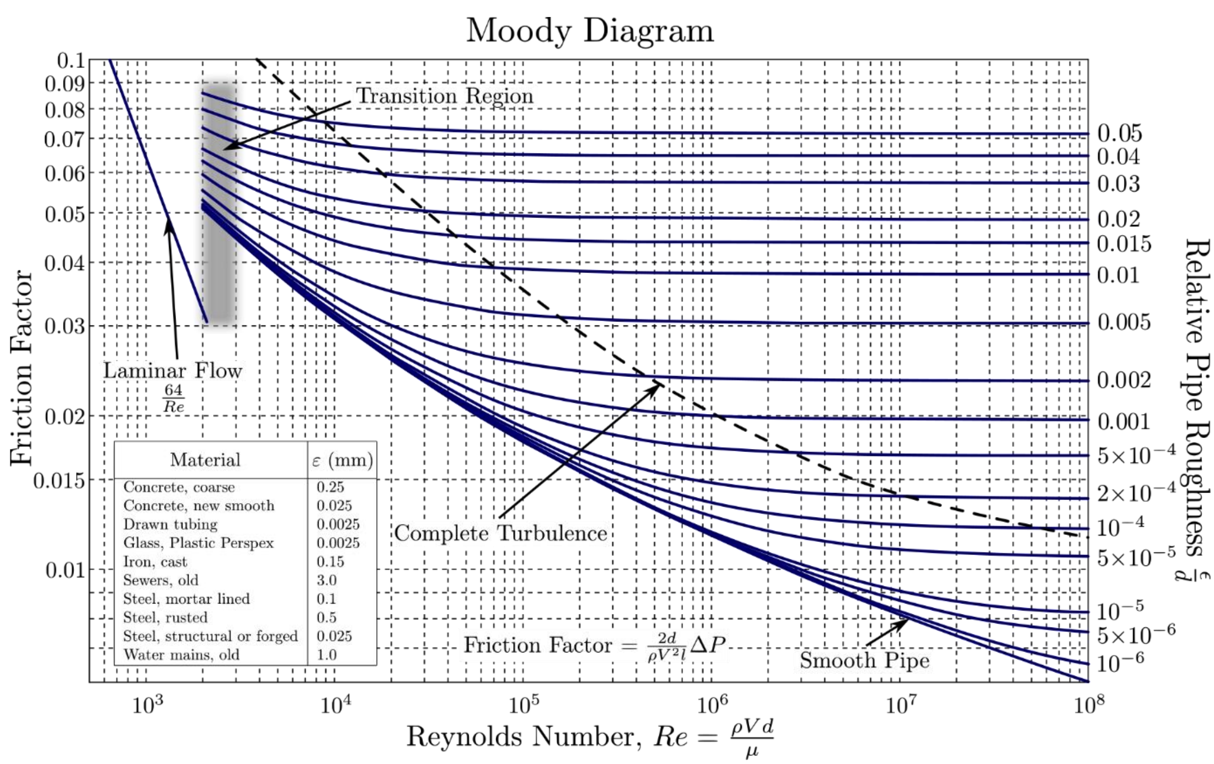
Compiled from Eckert and Drake (1972).

Table 3. *Physical Properties of Selected Solid Metals*

Table B-4 Physical properties of selected solid metals

Metal	Melting point, °C	Properties at 20°C: ρ, kg/m³	c_p, kJ/(kg · °C)	k, W/(m · °C)	α, m²/s	Thermal conductivity k, W/(m · °C): −100°C	0°C	100°C	300°C	600°C	1000°C
Aluminum											
Pure	660	2,707	0.896	204	8.418×10^{-5}	215	202	206	228		
Beryllium	1277	1,850	1.825	200	5.92×10^{-5}						
Bismuth	272	9,780	0.122	7.86	0.66×10^{-5}						
Cadmium	321	8,650	0.231	96.8	4.84×10^{-5}						
Copper											
Pure	1085	8,954	0.3831	386	11.234×10^{-5}	407	386	379	369	353	
Aluminum bronze, 95% Cu. 5% Al		8,666	0.410	83	2.330×10^{-5}						
Bronze, 75% Cu. 25% Sn		8.666	0.343	26	0.859×10^{-5}						
Red brass, 85% Cu. 9% Sn, 6% Zn		8.714	0.385	61	1.804×10^{-5}		59	71			
Brass, 70% Cu. 30% Zn		8,522	0.385	111	3.412×10^{-5}	88		128	147		
Constantan, 60% Cu. 40% Ni		8,922	0.410	22.7	0.612×10^{-5}	21		22.2			
Iron											
Pure	1537	7,897	0.452	73	2.034×10^{-5}	87	73	67	55	40	35
Wrought iron, 0.5% C		7,849	0.46	59	1.626×10^{-5}		59	57	48	36	33
Steel (C max ≈ 1.5%): Carbon steel											
C ≈ 0.5%		7,833	0.465	54	1.474×10^{-5}		55	52	45	35	29
1.0%		7,801	0.473	43	1.172×10^{-5}		43	43	40	33	28
1.5%		7,753	0.486	36	0.970×10^{-5}		36	36	35	31	28

Table 4. *Moody Diagram*



PROJECT REFLECTIONS

There were two critical skills I learned from this project that I will utilize for the rest of my life: high-level problem solving and technical writing. This project forces you to work within design restrictions while solving for several unknowns, vastly different from a simple "plug and chug" problem. This kind of decision making, and systematic analysis is vital for a career In engineering design, especially in the research and development sectors. Companies in the automotive and motorsports fields are constantly using technology to push the boundaries; as such, they must design will little to no precedence and starting reference point. Radiators are essentially heat exchangers, and they will continue to be used in the automotive world as long as cars produce heat. Even with the current shift to electric propulsion, vehicles require heat exchangers to keep their motor, motor controller, and accumulator temperatures in check.

Robert and I excelled as a team; I'll explain. As soon as we started this project, our strengths and weaknesses were very apparent; Robert was a go-getter as far as calculations go and I started by preparing an outline of our design criteria and process. As such, we shined with parallel development. I blew through the design and project report while Robert set up the Excel spreadsheet for calculations. These differences in weaknesses are not necessarily a bad thing: it just took me longer to understand the problem and calculations at hand, and it took Robert longer to get organized and formulate his ideas into a readable format.

As always, if I could start the class over, I would have started the project earlier. I think anyone would say they wish they had more time, but what's life without a little pressure? Under pressure has always been where I'm at my best.

-Khanh Le

This project was daunting at first, however, once all the design criteria were taken into account, the project seemed to complete itself. This project was incredibly helpful for my professional career since I plan on working at the shipyard. There will definitely be times where a heat exchanger will be used at the shipyard, and I may be in charge of designing and ensuring the safety of the heat exchanger. This is a great project to explain to a potential employer as it shows that not only can you perform design considerations, but also work as a team to complete the tasks given to you, and properly delegate work between group partners.

This project went very well because Khanh and I work very well as a team. There are many benefits to working with someone who has the same objectives in mind, and we were constantly on the same page throughout the design phase of this project. We were able to delegate the work so that neither of us was over worked or felt that they were doing the majority of the project.

If I could start this class over again, I would definitely create a more organized Excel sheet so that Professor Ayala does not need to critique my Excel layout choices. I would focus more time to working the homework problems and studying the material in class. Speaking of the class, it is hard to predict the future, but it would have been nice to attend more in class lectures. I always have learned better when I am in the physical class setting with the professor, and I think that would have helped me achieve a higher grade in the class.

-Robert Hoke