Design And Construction of Equipment

Part Two: Design Specifications of the Heat Exchanger

Almaz Khalilov

Table of Contents

Executive Summary

This report outlines the specifications, selection criteria and calculations of a shell and tube heat exchanger to cool a 182kmol/hr 1% ethanol, 99% P-xylene stream from 128.84 to 30 Degrees C. With water as our process fluid, our inlet temperature was set at 15 degrees C from an evaporative cooling tower, and exited the heat exchanger at 50 degrees C.

An AEL type heat exchanger, with 3 in series was chosen, as the particular combination yields the highest temperature correction factor of 0.95, without drawbacks such as thermal leakage. Our corrected log-mean temperature was thus calculated to be 36.55 degrees C.

A total heat transfer coefficient of 500 W/m^2K was used, and verified to be within 10% tolerance via iterative calculations. Thus, yielding a heat transfer area of 54.71 m^2.

A tube diameter of 0.75 inches was chosen, and a tube length of 96 inches was chosen, yielding an appropriate aspect ratio of 7.25. For our tube layout, a rotated triangular pattern was used, as it provides the most compact design. Additionally, our pitch was calculated to be 0.9375 inches. An internal diameter of 13.25 inches for the shell was chosen from the standard tube layout count table. Hence the number of real tubes used per shell was 124, or 62 per tube pass.

We allocated water as our tube-side fluid as it was felt that the corrosivity and dirtiness of water trumped other factors, and that easy cleaning of the tubes was a priority.

A baffle spacing of 0.14343m was used, which resulted in equal spacing in all areas. A baffle cut of 25% was used, and a baffle thickness of 1.6mm was used.

A tube wall thickness of 16 gauge was selected, translating to 1.651mm, and a tube side pressure drop of 2.6 kPa was calculated. The shell side pressure drop was calculated as 28.77 kPa, and thus the vessel is not classified as a pressure vessel. Turbulent flow was ensured for both the shell and tube side.

DN40 and DN50 was used for the tube side and shell side nozzles respectively. Schedule 40 pipe was used for both. Additionally, 4 tie rods of 0.25-inch diameter was used, and an outer shell thickness of 7.9mm was calculated. A 25.4mm tubesheet thickness was used at both ends.

The tube side and shell side fouling factors used were 3000 and 2500 W/m^2 K respectively.

This report also sets out the managerial aspects of operation such as the importance of checking for leaks, keeping adequate documentation and protocols, as well as safety factors such as overpressure, and loss of containment that must be considered.

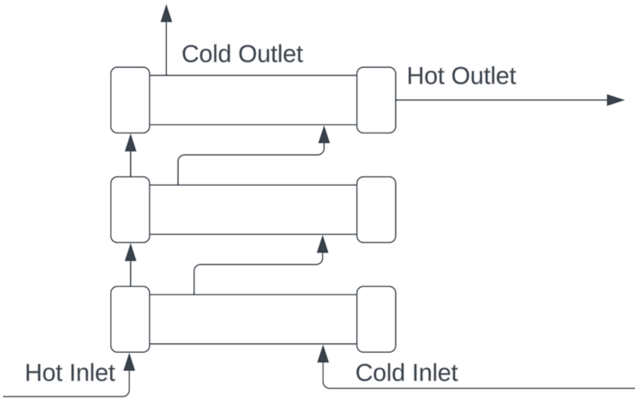
EQUIPMENT SPECIFICATION SHEET

Shell and Tube Heat Exchanger Data Sheet

GENERAL	1	Service of Unit: 20 years		
	2	Type of Unit: AEL	Orientation: Horizontal	TEMA Class: C
	3	No of units: 3	No. of Shell passes: 1	No of tube passes: 2
	4			
SERVICE SPECS	5			
	6		Shell Side	Tube Side
	7	Fluid Circulated	p-xylene/ethanol	Water
	8	Total Fluid Entering	182.00 kmol/hr	1365.77 kmol/hr
	9	Vapour	0	0
	10	Liquid	182.00 kmol/hr	365.77 kmol/hr
	11	Steam	0	0
	12	Non-Condensables	0	0
	13	Fluid Vapourized or Condensed	Condensed	Condensed
	14	Steam Condensed	N/A	Y
	15	Liquids: Specific Gravity (Operating Temperature) (kg/m^3)	806.8167	994.78
	16	Liquids: Viscosity (mPa.s)	0.348485	0.7514
	17	Liquids: Specific Heat (kJ/molK)	200.0937	75.3
	18	Liquids: Thermal Conductivity (W /m K)	0.114718	0.6145
	19	Fouling Factor (W/m^2 K)	3000	2500
	20	Vapours: Molecular Weight	N/A	N/A
	21	Vapours: Latent Heat (kJ/kg)	N/A	N/A
	22	Vapours: Viscosity (mPa.s)	N/A	N/A
	23	Vapours: Specific Heat (kJ/kgK)	N/A	N/A
	24	Vapours: Thermal Conductivity (kJ/m.s.K)	N/A	N/A
	25	Fouling Factor	N/A	N/A
	26	Non-Condensables – Mol. Wgt	0	0
	27	Temperature In (°C)	128.84	15
	28	Temperature Out (°C)	30	50
	29	Operating Pressure (kPa)	101.325	101.325
	30	Pressure Drop Allowed (kPa)	28.77	2.60
	31	Max. Operating Temp. (°C)	130	130

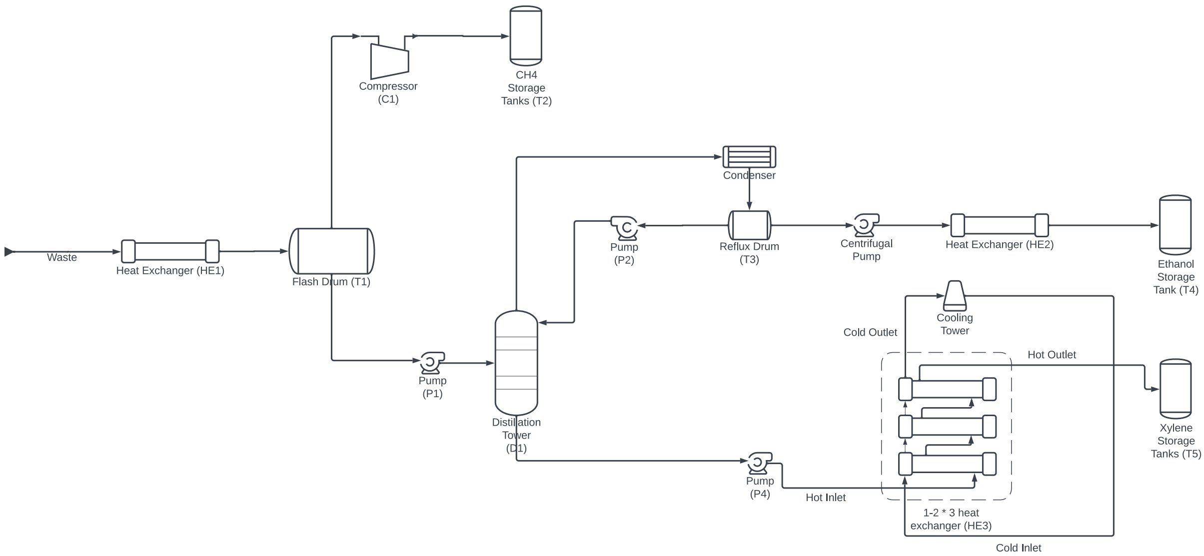
	32	Heat Exchanged (kW)		999.85		999.85
	33	Overall Coefficient (W/m^{2}K)		507.75		
	34	Calculated LMTD (°C)		36.55		
	35	Calculated Heat Transfer Area (m^2)		54.71		
CONST RUCT.	36					
	37	Tubes:	No.: 124	O.D.: ¾ inch	Thickness:1.651	Length (max): 96 inch
	38	Pitch: 0.9375”		Type: Rotated Triangular		Spacing:
	39	Shell side baffles-cross	Type: Segmental	% Cut (Diam.Area): 25		Spacing: 0.14343m
	40	Shell side baffles-long: N/A		Seal Type: N/A		
	41	Impingement Protection: None				
	42					

GENERAL ARRANGEMENT SKETCH



Initiator			Almaz Khalilov							
Checked			Project: DCE Part 2							
Approved			Specification No.1							
Date			Version	1						Page1.of 1

Process flow diagram



Calculations and discussion

Mass Balance

We need to perform a mass balance around D1 to find the flowrate and temperature of the inlet to HE3 heat exchanger.

We start with the flowrate to D1. We know that only p-xylene and ethanol travels to the distillation column, as all of the methane travels to the storage tanks from the flash drum.

As in part 1, the average molecular weight was calculated via

$$M_{\text{avg}} = n\%_1 \cdot M_1 + n\%_2 \cdot M_2 + n\%_3 \cdot M_3$$

With the average molar mass is $80.2\ kg/kmol$, we divide the total mass flowrate of each component in the waste stream by the average molar mass, to get the molar flow to the distillation tower.

Hence, we know that the molar flowrate to D1 is 181.04 kmol/hr, and the molar flowrate of ethanol is 43.89 kmol/hr.

Due to the reflux stream from the distillate, there is additional input via reflux, which may affect the following calculations. However, we do not know the reflux ratio, which should be set at a later point. Therefore, we assume that the effects of the reflux stream in negligible in determining the flowrate of the bottoms stream, and encompass our balance to include the recycle stream.

Now to perform the mass balance around D1 distillation tower, we use the equation for the total mass flow (Feed = Distillate + Bottoms):

$$F = D + B$$

We perform the mass balance for ethanol, where z is the fraction of ethanol in the feed stream, y is the fraction of ethanol in the distillate, and x is the fraction of ethanol in the bottoms:

$$F \cdot z = B \cdot x + D \cdot y$$

We also know that this is a binary mixture, so the fraction of p-xylene is 1 – the fraction of ethanol, so we can perform the mass balance for p-xylene:

$$F \cdot (1 - z) = B \cdot (1 - x) + D \cdot (1 - y)$$

From the project brief, we know that the maximum percent of ethanol in the bottoms is 1%, and the distillate percent of ethanol must contain at least 98%. We will set those values as our component fractions, due to the fact that additional separation would require a much larger distillation tower which requires additional distillation plates, leading to increased capital expenditure and complexity.

Now we know that the molar fraction of ethanol in the feed is 0.195, x = 0.01, and y = 0.98, and F = 224.93 kmol/hr, hence we can solve the equations simultaneously to find the flowrate of distillate and Bottoms.

Therefore B = 182.00 kmol/hr, and D = 42.93 kmol/hr.

Fluid Specification

Firstly, regarding the fluid of the xylene stream, we use the VLE data and linear interpolate between values.

Assuming the worst-case cooling load, we can assume that no cooling occurs from the distillation tower to the heat exchanger. Since we know that the bottoms contain 1% ethanol, we use the value of 0.01 mole fraction in liquid to find the bubble point of the liquid at that temperature. Linear interpolating from the data between the values of 0.0101, and 0.005. Hence, we find that the hot fluid input is 128.84 degrees C.

In the design brief, the hot fluid output must be 30 degrees C, hence we set that as the output temperature.

For the water, we know that the input temperature is 15 degrees C from the cooling tower, and the maximum allowable output is 50 degrees C, we will use those values to maximise cooling capacity.

However realistically, we must think about where cooling water will not be available at 15 degrees C. In warmer months, an evaporative cooling system will not be able to supply 15-degree temperature water, so make-up water or cooling via refrigeration need to be used, both of which increase operational costs.

It may be possible to use saltwater from the sea, as the sea temperature in Victoria do not rise above 20 degrees C. Future research may be done into the effects of using saltwater, and similar calculations can be done with saltwater as the cooling fluid instead of utility water.

It is important to use the correct values of heat capacity, density, and viscosity of each component, as it affects following calculations. The midpoint temperature of input and output is used to determine these values, which is at 352K for xylene/ethanol, and 306 K for water, we also know that the operation is at 1atm pressure.

From experimental data we determine the approximate heat capacity for p-xylene is 200.63 J/(K*mol) (Garg et al., 1993), 147 J/(K*mol) for ethanol (Ethanol - Specific Heat Vs. Temperature and Pressure, 2023), and 75.3 J/(K*mol) for water (Water - Specific Heat Vs. Temperature, 2023) at the specified temperature

Densities were found in a similar way from experimental or theoretical data to be:

- 807.53 kg/m^3 for p-xylene (Garg et al., 1993)
- 736.2 kg/m^3 for ethanol (Ethanol - Density and Specific Weight Vs. Temperature and Pressure, 2023)
- 994.78 kg/m^3 for water (Water - Density, Specific Weight and Thermal Expansion Coefficients, 2023)

Lastly, the the same was done with viscosity

- 0.3519 mPa s for p-xylene (P-Xylene (Data Page), 2023)
- 0.0104 mPa s for ethanol (Ethanol - Dynamic and Kinematic Viscosity Vs. Temperature and Pressure, 2023)
- 0.7514 mPa s for water (Water - Dynamic (Absolute) and Kinematic Viscosity Vs. Temperature and Pressure, 2023)

The heat capacity, density and viscosity values for the hot stream were averaged out to account for the 1% ethanol in the stream.

Heat Transfer Calculations

From the data, we can calculate the Energy required to cool the hot stream via the equation

$$Q = m_h \cdot c_{p_h} \cdot \Delta T_h$$

From the data, we find that 999.86 kJ/s of energy is required to cool the hot stream.

Now, since the energy required to cool the hot stream is equal to the cooling power required of the cold stream, we can rearrange the equation and solve for the flowrate of the water:

$$m_c = \frac{Q}{c_{p_c} \cdot \Delta T_c}$$

Now when picking the number of shell passes our heat exchanger has, we must consider the flowrate of water.

If the water inlet was 15 degrees and outlet of 20 degrees C, to avoid a temperature pinch, we would require almost 9560.38 kmol/hr of cooling water, equating to 172278 kg/hr. This is obviously unfeasible, and not only would require excess cooling water make up, but a much bigger cooling tower, multiplying costs.

Hence, as states above, we opt to use the maximum water outlet temperature of 50 degrees C. Calculating via the equation above, we require a cooling water usage of 1365.77 kmol/hr. Because we have a temperature cross, we must use multiple shell passes, either in separate shells or in the same shell.

Head selection and Tubesheets

We must select between a fixed Tubesheet exchanger and a floating header exchanger, as a U-tube type exchanger is not considered, as our application does not demand that our tube-side fluids are clean or extremely dangerous.

The advantage of a fixed tubesheet heat exchanger is that it is cheapest, and simplest design due to the welding of the tubesheet to the shell, however it is not advised to use this under a high temperature differential due to the fact that little room for thermal expansion exists. On the other hand, a floating head exchanger allows room for thermal expansion, and performs well under high pressures and temperatures, and allows removal of the tube bundle for cleaning the shell. However, it can be up to 25% more expensive than a fixed tubesheet exchanger.

Because we are operating at atmospheric pressure, and our log-mean temperature difference is 38.47 degrees, we do not anticipate much thermal expansion in our shell, and our pressure is low. Additionally, we use clean and non-corrosive p-xylene and ethanol fluid in our shell side, and therefore we do not anticipate that fouling will form in the shell side. Therefore, we pick a fixed tubesheet.

Now we have a range of front and rear heads to choose from, as shown in this figure:

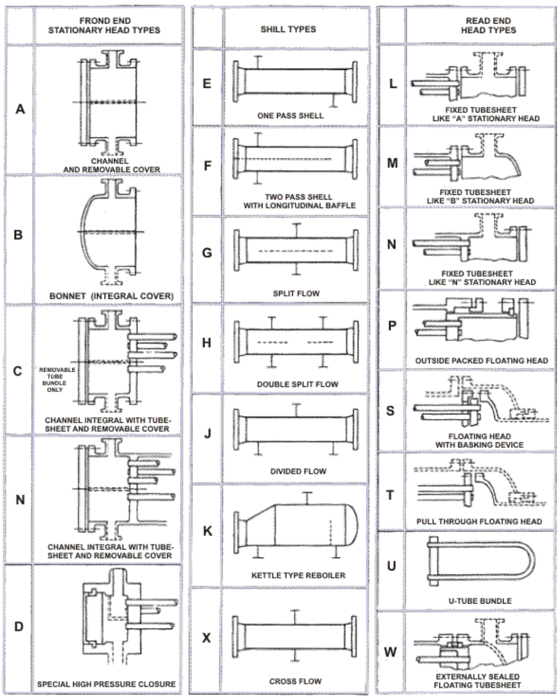


Figure 1 – Heat exchanger type (Standards | TEMA, 2019)

We will pick A front head, due to the fact that it is easy to repair and replace, and gives easy access to the tubes for cleaning or repair. For the rear head, we will pick the L rear head type, as it also gives easy access to cleaning and repair of the tubes, and clearances are small.

Shell selection

Next, we calculate the log-mean temperature difference with T1 as the hot fluid inlet temperature, T2 as hot fluid outlet, t1 as cold fluid inlet, and t2 as cold fluid outlet:

$$\Delta T_{lm} = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln \dfrac{(T_1 - t_2)}{(T_2 - t_1)}}$$

We find that our log-mean temperature difference is 38.47 degrees C.

Now, to find the correction factor, that aids us to find the real temperature difference for multiple shell pass heat exchangers, we must find correction factors, R and S. The equations used for them are:

$$R = \frac{(T_1 - T_2)}{(t_2 - t_1)} \text{ and } S = \frac{(t_2 - t_1)}{(T_1 - t_1)}$$

We find that R = 2.82 and S = 0.31.

Now we find the temperature correction factor (Ft), from graphs to find how many shell passes to use:

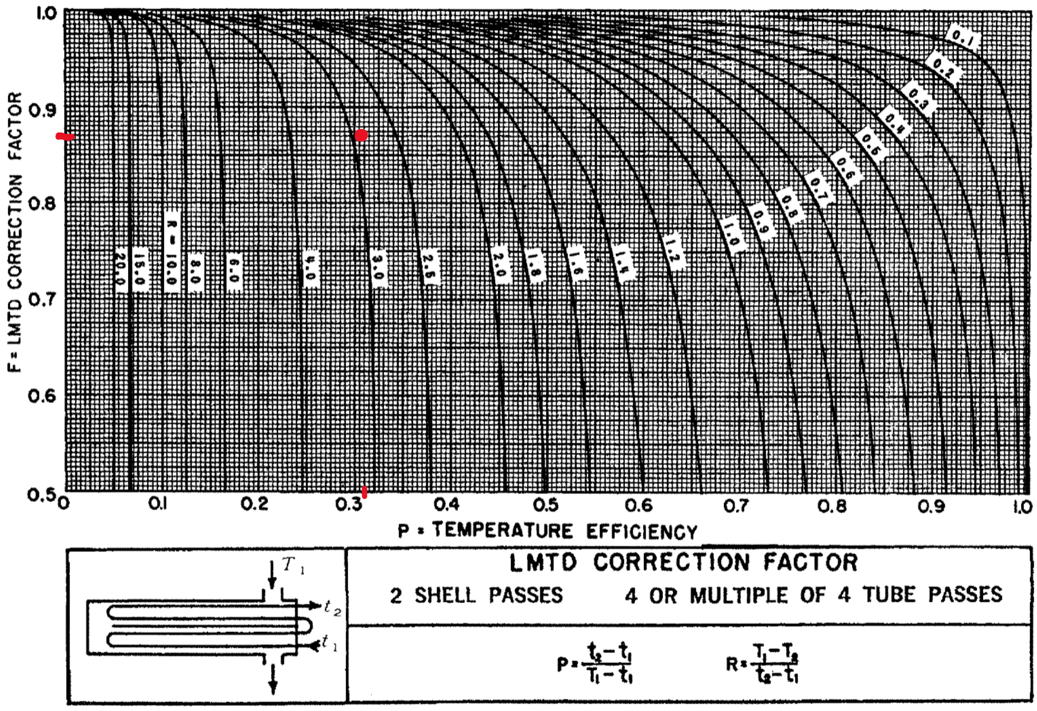


Figure 2 – 2 Shell Pass heat exchanger correction (Standards | TEMA, 2019)

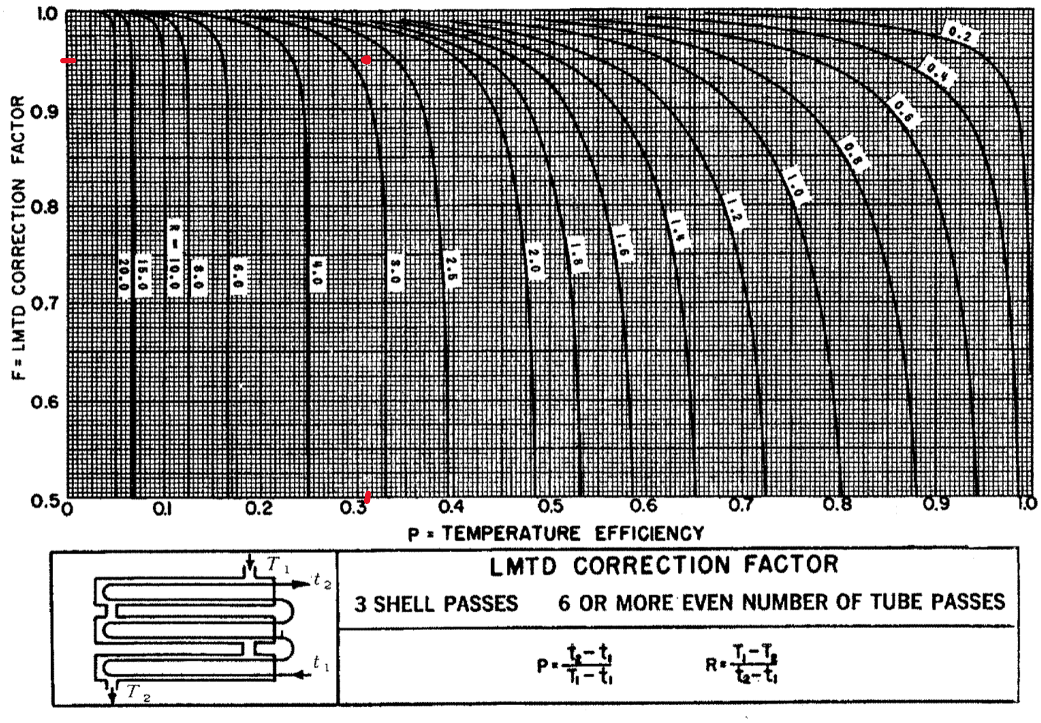


Figure 3 – 3 Shell pass correction (Standards | TEMA, 2019)

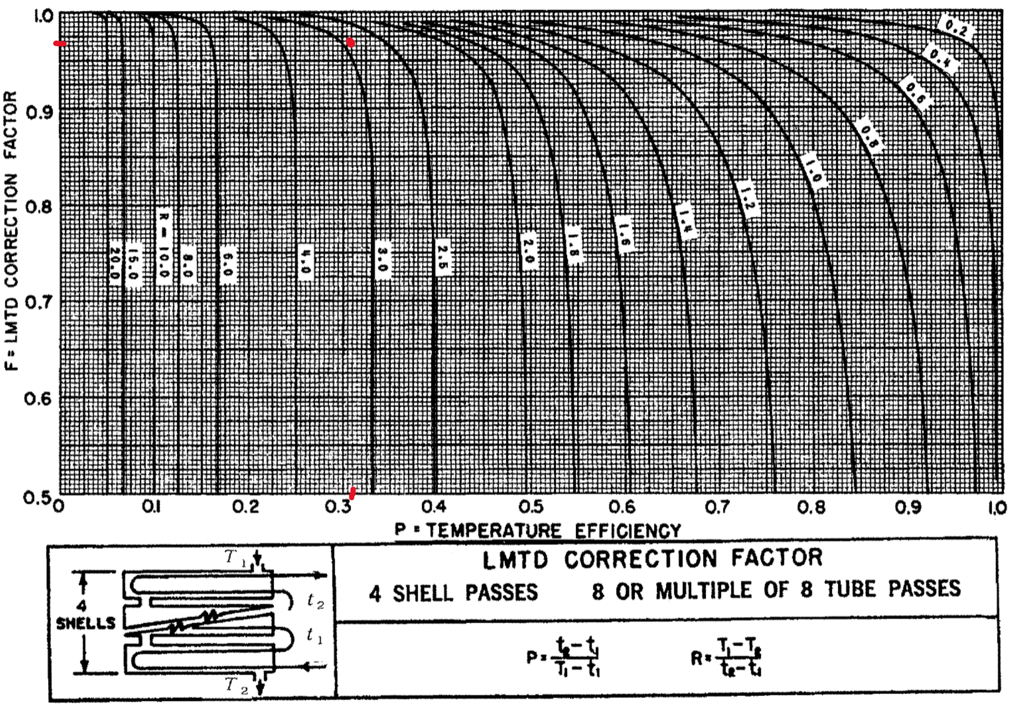


Figure 4 – 4 shell pass correction (Standards | TEMA, 2019)

Hence, we calculate that for 2, 3 and 4 shell pass heat exchanger our Ft values are 0.86, 0.95 and 0.965 respectively.

Now, we pick 3 shell passes, as it provides the highest value. For 2 shell passes the Ft value seems low, and hence require a higher flowrate of cooling water, which may make the operational cost higher than the capital cost in the long run, while the 4-shell pass heat exchanger provides a higher Ft value, but reaches diminishing returns, where the correction value is slightly higher, however other problems may arise such as higher capital expenditure, more maintenance, and a higher pressure drop.

Hence our real temperature difference using the following equation is calculated to be 36.55 degrees C;

$$\Delta T_m = F_t \Delta T_{\mathrm{lm}}$$

We then pick a shell type for our heat exchanger from the following table:

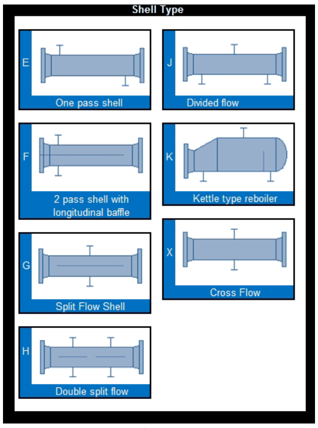


Figure 5 – Shell types (Standards | TEMA, 2019)

We will consider shell types E and F, as they are most commonly used in industry, and hence are the most available and cheapest. Other types such as the Kettle type reboiler and the double split flow, are not applicable to our application.

The advantage of a F shell type, or a 2-pass longitudinal shell is that it saves on costs due to only requiring one shell to use. However, problems arise with this shell such as leakage, and thermal inefficiency, which may reduce the effectiveness of the heat exchanger (Brogan, 2008). Hence, we opt to use 3 separate E shells for our heat exchanger.

Therefore, we have an AEL heat exchanger.

Heat Transfer Area

The heat transfer area, can be calculated via the following equation, where Q is the duty required, Delta T is the real temperature difference calculated above and U is the overall heat transfer coefficient:

$$A = \frac{Q}{U \cdot \Delta T}$$

On initial calculation, we use an approximate value of the overall heat transfer coefficient from typical operations such as in figure 6. Picking a value of 500 W/m^2 K results in an area requirement of 54.71m^2.

HEAT-TRANSFER EQUIPMENT

Table 12.1. Typical overall coefficients

Shell and tube exchangers		
Hot fluid	Cold fluid	U (W/m^2°C)
Heat exchangers		
Water	Water	800–1500
Organic solvents	Organic solvents	100–300
Light oils	Light oils	100–400
Heavy oils	Heavy oils	50–300
Gases	Gases	10–50
Coolers		
Organic solvents	Water	250–750
Light oils	Water	350–900
Heavy oils	Water	60–3[illegible]
Gases	Water	20–300
Organic solvents	Brine	150–500
Water	Brine	600–1200
Gases	Brine	15–250
Heaters		
Steam	Water	1500–4000
Steam	Organic solvents	500–1000
Steam	Light oils	300–900
Steam	Heavy oils	60–450
Steam	Gases	30–300
Dowtherm	Heavy oils	50–300
Dowtherm	Gases	20–200
Flue gases	Steam	30–100
Flue	Hydrocarbon vapours	30–100
Condensers		
Aqueous vapours	Water	1000–1500
Organic vapours	Water	700–1000
Organics (some non-condensables)	Water	500–700
Vacuum condensers	Water	200–500
Vaporisers		
Steam	Aqueous solutions	1000–1500
Steam	Light organics	900–1200
Steam	Heavy organics	600–900

Figure 6 – Typical overall coefficients (Coulson & Richardson, 2017)

However, this is an iterative process, and this value must line up with the final value calculated from further calculation. Upon iteration, it was found that adjusting to 500 W/m^2 K provided the most accurate heat transfer area of 54.71 m^2

Tubing Selection

Firstly, we must set the outside diameter of the tube. Advantages of a high tube diameter are that it allows for higher flowrates, and is easier to clean, however it requires more space, and most importantly the heat transfer rates are lower, especially if the fluid velocity is not sufficiently high.

For the final design, we pick a ¾ inch standard pipe OD, hence we calculate the total length of piping required via division of the total heat transfer area required by the area of tubes per meter, requiring 914.21m of piping in total, or 304.74m per shell

Now we must select the appropriate pipe length. Standard pipe length used are 6, 8, 12, 16, 20, and 24 feet. We will select an 8-foot (92 inch) tube length to minimise the cost of the heat exchanger, and ensure that our aspect ratio of our heat exchanger is between 4 and 10.

From selection, our aspect ratio is 7.25, which is appropriate.

We hence need 124.97 tubes per shell, and we have 2 tube passes per shell.

Next, the tube pitch controls the spacing between the pitch, where the pitch is normally ¼ used, hence a pitch value of 1.25*d_o is usually used. Our pitch is thus 0.9375 or 15/16”.

Additionally, we select a triangular pitch at 60 degrees, as it provides the most compact construction. A square pitch is also not considered due to the fact that we have a fixed tubesheet, and thus the positive properties of a square pitch such as ease of cleaning do not matter.

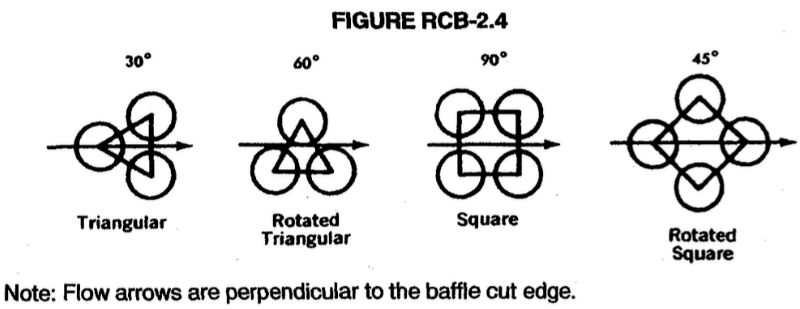


Figure 7 – pitch types (Standards | TEMA, 2019)

Now, to size the shell we lookup ¾” on 15/16” triangular tube, for a fixed tube two-pass heat exchanger. We read the value of the tubes per shell which is 124.97. The closest value from the table is 124 and select that to read the inside diameter of the shell to be 13 & ¼ Inches. From this we know that we use 124 tubes in total, and thus a slightly higher flowrate is required of water.

TABLE 8.13. Tube Counts of Shell-and-Tube Heat Exchangers[a]

Heat Exchanger Tube Sheet Layout Count Table

37	35	33	31	29	27	25	23¼	21¼	19¼	17¼	15¼	13¼	12	10	8	I.D. of Shell (In.)		
1269	1143	1019	881	763	663	553	481	391	307	247	193	135	105	69	33	¾″ on 15/16″ △	Fixed Tubes	One-Pass
1127	1007	889	765	667	577	493	423	343	277	217	157	117	91	57	33	¾″ on 1″ △		
965	865	765	665	587	495	419	355	287	235	183	139	101	85	53	33	¾″ on 1″ □		
699	633	551	481	427	361	307	247	205	163	133	103	73	57	33	15	1″ on 1¼″ △		
595	545	477	413	359	303	255	215	179	139	111	83	65	45	33	17	1″ on 1¼″ □		
1242	1088	964	846	734	626	528	452	370	300	228	166	124	94	58	32	¾″ on 15/16″ △	Fixed Tubes	Two-Pass
1088	972	858	746	646	556	468	398	326	264	208	154	110	90	56	28	¾″ on 1″ △		
946	840	746	644	560	486	408	346	280	222	172	126	94	78	48	26	¾″ on 1″ □		
688	608	530	462	410	346	292	244	204	162	126	92	62	52	32	16	1″ on 1¼″ △		
584	522	460	402	348	298	248	218	172	136	106	76	56	40	26	12	1″ on 1¼″ □		
1126	1008	882	768	648	558	460	398	304	234	180	134	94	64	34	8	¾″ on 15/16″ △	U Tubes[2]	
1000	882	772	674	566	484	406	336	270	212	158	108	72	60	26	8	¾″ on 1″ △		
884	778	688	586	506	436	362	304	242	188	142	100	72	52	30	12	¾″ on 1″ □		
610	532	466	396	340	284	234	192	154	120	84	58	42	26	8	XX	1″ on 1¼″ △		
526	464	406	356	304	256	214	180	134	100	76	58	38	22	12	XX	1″ on 1¼″ □		
1172	1024	904	788	680	576	484	412	332	266	196	154	108	84	48	XX	¾″ on 15/16″ △	Fixed Tubes	Four-Pass
1024	912	802	692	596	508	424	360	292	232	180	134	96	72	44	XX	¾″ on 1″ △		
880	778	688	590	510	440	366	308	242	192	142	126	88	72	48	XX	¾″ on 1″ □		
638	560	486	422	368	308	258	212	176	138	104	78	60	44	24	XX	1″ on 1¼″ △		
534	476	414	360	310	260	214	188	142	110	84	74	48	40	24	XX	1″ on 1¼″ □		
1092	976	852	740	622	534	438	378	286	218	166	122	84	56	28	XX	¾″ on 15/16″ △	U Tubes[2]	
968	852	744	648	542	462	386	318	254	198	146	98	64	52	20	XX	¾″ on 1″ △		
852	748	660	560	482	414	342	286	226	174	130	90	64	44	24	XX	¾″ on 1″ □		
584	508	444	376	322	266	218	178	142	110	74	50	36	20	XX	XX	1″ on 1¼″ △		
500	440	384	336	286	238	198	166	122	90	66	50	32	16	XX	XX	1″ on 1¼″ □		
1106	964	844	732	632	532	440	372	294	230	174	116	80	XX	XX	XX	¾″ on 15/16″ △	Fixed Tubes	Six-Pass
964	852	744	640	548	464	388	322	258	202	156	104	66	XX	XX	XX	¾″ on 1″ △		
818	224	634	536	460	394	324	266	212	158	116	78	54	XX	XX	XX	¾″ on 1″ □		
586	514	442	382	338	274	226	182	150	112	82	56	34	XX	XX	XX	1″ on 1¼″ △		
484	430	368	318	268	226	184	154	116	88	66	44	XX	XX	XX	XX	1″ on 1¼″ □		
1058	944	826	716	596	510	416	358	272	206	156	110	74	XX	XX	XX	¾″ on 15/16″ △	U Tubes[2]	
940	826	720	626	518	440	366	300	238	184	134	88	56	XX	XX	XX	¾″ on 1″ △		
820	718	632	534	458	392	322	268	210	160	118	80	56	XX	XX	XX	¾″ on 1″ □		
562	488	426	356	304	252	206	168	130	100	68	42	30	XX	XX	XX	1″ on 1¼″ △		
478	420	362	316	268	224	182	152	110	80	60	42	XX	XX	XX	XX	1″ on 1¼″ □		
1040	902	790	682	576	484	398	332	258	198	140	94	XX	XX	XX	XX	¾″ on 15/16″ △	Fixed Tubes	Eight-Pass
902	798	694	588	496	422	344	286	224	170	124	82	XX	XX	XX	XX	¾″ on 1″ △		
760	662	576	490	414	352	286	228	174	132	94	XX	XX	XX	XX	XX	¾″ on 1″ □		
542	466	400	342	298	240	190	154	120	90	66	XX	XX	XX	XX	XX	1″ on 1¼″ △		
438	388	334	280	230	192	150	128	94	74	XX	XX	XX	XX	XX	XX	1″ on 1¼″ □		
1032	916	796	688	578	490	398	342	254	190	142	102	68	XX	XX	XX	¾″ on 15/16″ △	U Tubes[2]	
908	796	692	600	498	422	350	286	226	170	122	82	52	XX	XX	XX	¾″ on 1″ △		
792	692	608	512	438	374	306	254	194	146	106	70	48	XX	XX	XX	¾″ on 1″ □		
540	464	404	340	290	238	190	154	118	90	58	38	24	XX	XX	XX	1″ on 1¼″ △		
456	396	344	300	254	206	170	142	98	70	50	34	XX	XX	XX	XX	1″ on 1¼″ □		
37	35	33	31	29	27	25	23¼	21¼	19¼	17¼	15¼	13¼	12	10	8	I.D. of Shell (in.)		

[1] Allowance made for Tie Rods.
[2] R.O.B. = $2\frac{1}{2}$ × Tube Dia. Actual Number of "U" Tubes is one-half the above figures.
[a] A 3/4 in. tube has 0.1963 sqft/ft, a 1 in. OD has 0.2618 sqft/ft. Allowance made for tie rods.
[b] R.O.B. = $2\frac{1}{2}$ × tube dia. Actual number of "U" tubes is one-half the above figures.

Figure 8 – Tube layout selection chart (Walas, 1988)

Bundle Selection

We therefore select the bundle diameter, and hence see the clearance required. Using the following table, we select the constants for a two-pass triangular shell and use the following formula to get the bundle diameter:

$$D_b = d_o \left(\frac{N_t}{K_1}\right)^{1/n_1}$$

Table 12.4. Constants for use in equation 12.3

Triangular pitch, $p_t = 1.25d_o$					
No. passes	1	2	4	6	8
K_1	0.319	0.249	0.175	0.0743	0.0365
n_1	2.142	2.207	2.285	2.499	2.675
Square pitch, $p_t = 1.25d_o$					
No. passes	1	2	4	6	8
K_1	0.215	0.156	0.158	0.0402	0.0331
n_1	2.207	2.291	2.263	2.617	2.643

Figure 9 – constants to determine bundle diameter (Coulson & Richardson, 2017)

Hence, we determine our bundle diameter to be 0.318m. We thus determine the clearance reading the following graph, and picking a clearance of 11mm:

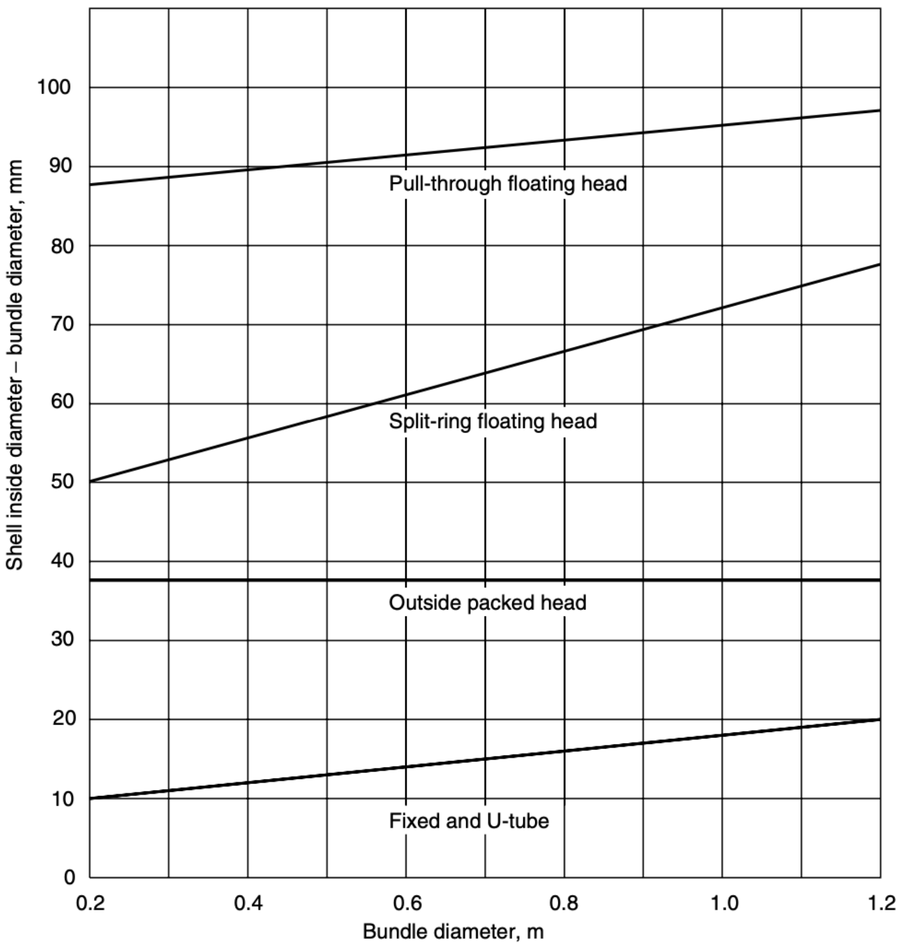


Figure 12.10. Shell-bundle clearance

Figure 10 – shell clearances (Coulson & Richardson, 2017)

Fluid Allocation

We allocate fluid in a systematic fashion, with water being the most corrosive fluid, and has a high potential for fouling, opting for it to be allocated to the tube. Xylene however is the hotter fluid which would opt for it to be allocated to the tube, and water has the higher viscosity opting for it to be allocated to the tube.

Additional factors such as high pressure, low flow rates, and pressure drop are not significant enough to be considered.

Because of the significance of cleaning, the high cost of maintenance of fouling, we put significant weight to minimise those factors. Hence, we allocate water to the tube-side, and the ethanol/p-xylene mixture to the shell side.

Tube-side pressure drop

To find the tube-side and shell-side pressure drop we must go through a lengthy process. Finding the pressure drop throughout, firstly tells us how much pressure builds up inside the heat exchanger. A large delta P means that increased costs are required to pump the liquid through, and result in a larger wear and tear on the heat exchanger. Additionally, heat exchangers with a high pressure may be subject to different regulations than at low pressures, and may be classified as pressure vessels.

Firstly, we find the mass flow per tube per pass to be 0.110265044 kg/s, by dividing the mass flowrate 24611.16 kg/hr by the number of tubes used per pass (62).

We must at this point determine the wall thickness of the tubing. We pick 16 BWG piping due to the fact that it is an adequate wall thickness to be strong and hold up, while also having tolerance for corrosion allowance and a good well-rounded heat transfer coefficient. Hence, we calculate our tubing internal diameter to be 15.7480381mm.

After calculating the cross-sectional area of the tube, we find that the average velocity of fluid is 0.569m/s, which is quite low, but within the ranges a normal liquid velocity. Higher velocities increase the heat transfer efficiency, and we may lower out tube inner diameter, however we decide to keep it at ¾ inches due to the fact that lower velocities reduce erosion and wear, and reduce the fouling inside the tubing.

Now we calculate the Reynolds number via $Re = \frac{\rho v d}{\mu}$. We find the Reynolds number in the tube-side is 11864, which is much larger than 2000, hence we know that we are operating in turbulent flow. Turbulent flow is a good flow regime to be operating in due to the fact that there are increased heat transfer rates due to better mixing of fluids, and less chance of fouling or hotspots that might occur.

We are therefore able to read the Friction factor from the following chart, to yield j_f = 0.0045.

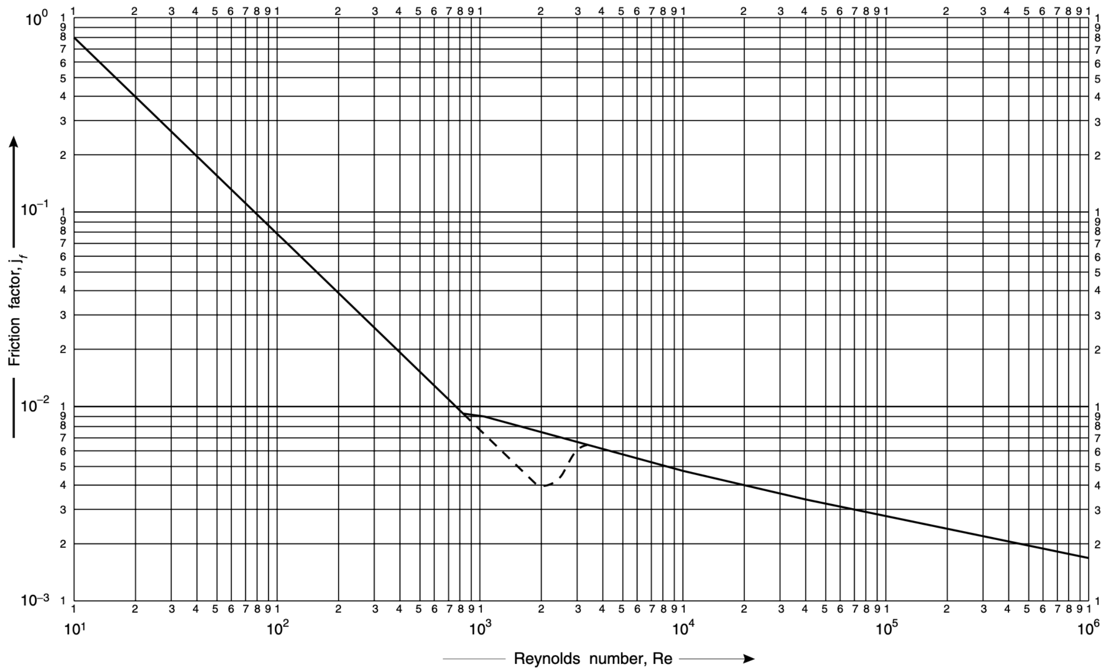


Figure 12.24. Tube-side friction factors
Note: The friction factor j_f is the same as the friction factor for pipes $\phi(= (R/\rho u^2))$, defined in Volume 1 Chapter 3.

Figure 11 - Friction factor determination (Coulson & Richardson, 2017)

Now we can determine the Pressure drop from the tube-side via the following equation:

$$\Delta P_t = N_p \left[8 j_f \left(\frac{L}{d_i}\right)\left(\frac{\mu}{\mu_w}\right)^{-m} + 2.5 \right] \frac{\rho u_t^2}{2}$$

Now we assume that the fluid viscosity at the bulk fluid temperature is roughly equal to the fluid viscosity at the wall, and hence μ/μ_w is 1.

We therefore calculate a tube-side press pressure drop of 2.6 kPa per shell, which is almost negligible. (hence may not be classified as a pressure vessel).

Shell-side pressure drop

Now upon determining the shell-side pressure drop we calculate the volumetric flowrate and velocity in a similar way; however we must pick the baffle spacing, and baffle which, which are the major determinant of the pressure drop in the heat exchanger, due to the way that the geometry is set up.

From best practises, the minimum baffle diameter must be the biggest of 1/5 diameter of the shell, or 50mm. While the maximum baffle spacing is set by how strong the shell as, because the baffles act as support, and the shell may collapse without them.

We chose a baffle spacing of 0.14343 m, or 14.343cm. Using such a value ensures that the length of the tube/baffle spacing is close to an integer value, so that the baffles close to the tubesheet are evenly spaced. Additionally, that number must be odd due to the fact that the inlet and outlet of the heat exchanger must exit to a different side.

Now we must also check that the ratio of baffle spacing over shell ID is between 0.3 and 0.5. It is 0.426, hence our baffle configuration is within the recommended design parameters.

We also choose a baffle segmental cut of 25%, as it is proven to be the most commonly used, and most effective.

Now because fluid travels over the lengths of tubes perpendicularly, we do not have a tube diameter, but we can use an equivalent diameter, and hence calculate the theoretical area, A_s. We hence find that the velocity of fluid is estimated to be 0.68 m/s, which again is slightly low but not out of the ordinary. We thus calculate a Reynolds number of 21234 and find that the flow is well within the turbulent region.

From the graph, we calculate the friction factor for a 25% baffle cut to be j_f = 0.042:

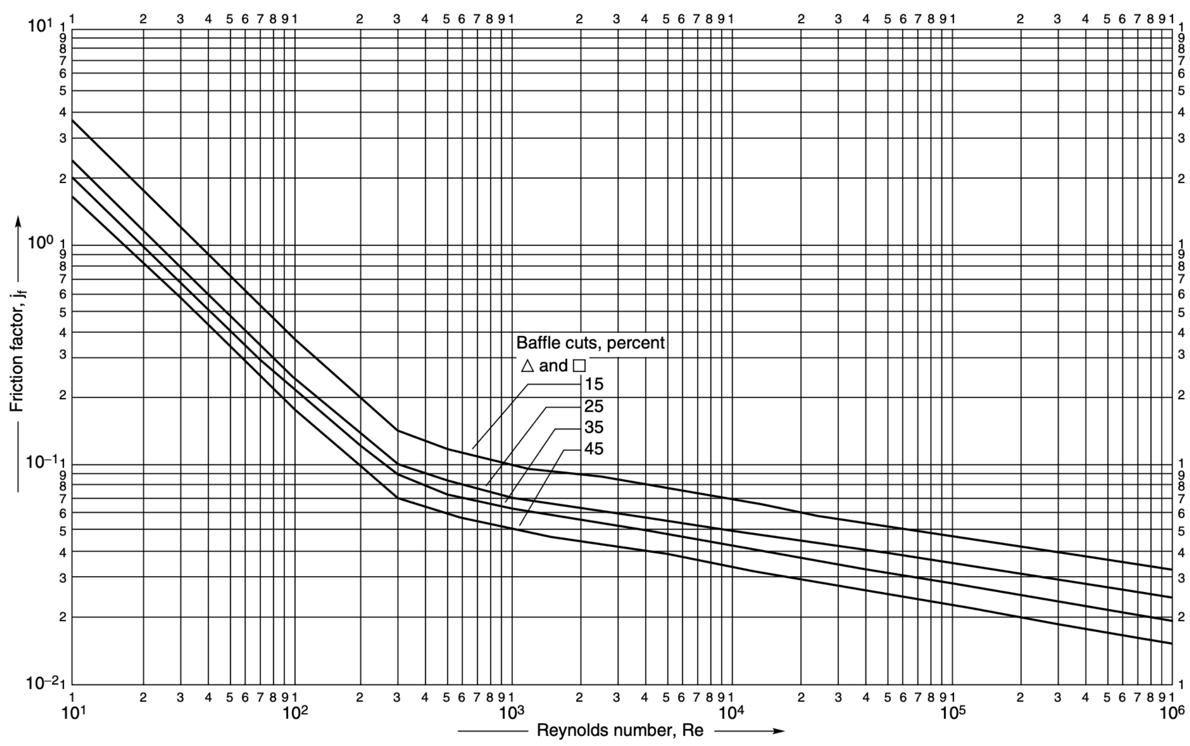


Figure 12.30. Shell-side friction factors, segmental baffles

Figure 11 – shell Friction factor determination (Coulson & Richardson, 2017)

Therefore, when using Kern's method via the following equation:

$$\Delta P_s = 8j_f\left(\frac{D_s}{d_e}\right)\left(\frac{L}{l_B}\right)\frac{\rho u_s^2}{2}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

We find that the shell side pressure drop is 26.86 kPa. According to AS1210, it is not classified as a pressure vessel at the vessel diameter, and thus we may proceed.

Nozzles

Firstly, our pressure vessel fits into TEMA class "C", as a general heat exchanger, hence strict R classification does not apply to us.

For our tube-side diameter, we pick a velocity in the nozzle such as 1.2 m/s, and calculate the DN diameter and schedule from there. A 1.2m/s velocity seems appropriate, as it is the standard range seen in literature. We calculate the nozzle diameter via area and velocity to be 48.17mm, hence we pick the closest DN value of 40, and using the standard schedule of 40, we find that the outside nozzle diameter is 48.3mm, and 3.68mm wall thickness.

For our shell-side diameter, we follow the same procedure, and using a velocity at nozzle of 0.7. We set the nozzle velocity lower due to flow-induced tube vibrations. As we are not using an impingement plate, we want to completely remove the possibility of vibrations, and allow for a higher flowrate in the future without completely changing the heat exchanger.

We thus choose a DN50 outside shell diameter, with a standard schedule of 40.

Tie-Rods

We must use tie-rods to hold the shell bundles together and ensure they adequately support the shell. Looking at the following TEMA guidelines for C class exchangers, we see that we must use 4 tie rods, at a diameter of ¼ inches:

TABLE CB-4.71

TIE ROD STANDARDS
Dimensions in Inches (mm)

Nominal Shell Diameter		Tie Rod Diameter		Minimum Number of Tie Rods
6 – 15	(152-381)	1/4	(6.4)	4
16 – 27	(406-686)	3/8	(9.5)	6
28 – 33	(711-838)	1/2	(12.7)	6
34 – 48	(864-1219)	1/2	(12.7)	8
49 – 60	(1245-1524)	1/2	(12.7)	10
61 – 100	(1549-2540)	5/8	(15.9)	12

Figure 12 – Tie rod determination (Standards | TEMA, 2019)

Baffle thickness and diameter

Based on the TEMA guidelines for our class, shell ID, and unsupported tube lengths (baffle spacing), we select a baffle thickness of 0.0625 inches, as per the following table:

TABLE CB-4.41

BAFFLE OR SUPPORT PLATE THICKNESS
Dimensions in Inches (mm)

Nominal Shell ID	Plate Thickness					
	Unsupported tube length between central baffles. End spaces between tubesheets and baffles are not a consideration.					
	12 (305) and Under	Over 12 (305) to 24 (610) Inclusive	Over 24 (610) to 36 (914) Inclusive	Over 36 (914) to 48 (1219) Inclusive	Over 48 (1219) to 60 (1524) Inclusive	Over 60 (1524)
6-14 (152-356)	1/16 (1.6)	1/8 (3.2)	3/16 (4.8)	1/4 (6.4)	3/8 (9.5)	3/8 (9.5)
15-28 (381-711)	1/8 (3.2)	3/16 (4.8)	1/4 (6.4)	3/8 (9.5)	3/8 (9.5)	1/2 (12.7)
29-38 (737-965)	3/16 (4.8)	1/4 (6.4)	5/16 (7.5)	3/8 (9.5)	1/2 (12.7)	5/8 (15.9)
39-60 (991-1524)	1/4 (6.4)	1/4 (6.4)	3/8 (9.5)	1/2 (12.7)	5/8 (15.9)	5/8 (15.9)
61-100 (1549-2540)	1/4 (6.4)	3/8 (9.5)	1/2 (12.7)	5/8 (12.7)	3/4 (19.1)	3/4 (19.1)

Figure 13 – baffle plate thickness (Standards | TEMA, 2019)

We also calculate our baffle thickness from the following table, and find that our baffle outside diameter is 0.333m.

TABLE RCB-4.3

Standard Cross Baffle and Support Plate Clearances
Dimensions In Inches (mm)

Nominal Shell ID	Design ID of Shell Minus Baffle OD
6-17 (152-432)	1/8 (3.2)
18-39 (457-991)	3/16 (4.8)
40-54 (1016-1372)	1/4 (6.4)
55-69 (1397-1753)	5/16 (7.9)
70-84 (1778-2134)	3/8 (9.5)
85-100 (2159-2540)	7/16 (11.1)

Figure 14 – standard baffle clearances (Standards | TEMA, 2019)

Material and shell thickness

For our shell, we opt to use carbon steel, as p-xylene and ethanol are not corrosive to it, and are compatible (Chemical Compatibility Guide, 2023), as well as being cheap. For our tubes, we opt to use copper tubing, as it provides the highest heat transfer coefficient, while having a reasonable price.

We check the following table to ensure that our baffle spacing and temperature falls within the allowable maximums for our material, and find that it is adequate:

TABLE RCB-4.52

MAXIMUM UNSUPPORTED STRAIGHT TUBE SPANS
Dimensions in Inches (mm)

Tube OD	Tube Materials and Temperature Limits ° F (° C)	
	Carbon Steel & High Alloy Steel, 750 (399) Low Alloy Steel, 850 (454) Nickel-Copper, 600 (316) Nickel, 850 (454) Nickel-Chromium-Iron, 1000 (538)	Aluminum & Aluminum Alloys, Copper & Copper Alloys, Titanium Alloys At Code Maximum Allowable Temperature
1/4 (6.4)	26 (660)	22 (559)
3/8 (9.5)	35 (889)	30 (762)
1/2 (12.7)	44 (1118)	38 (965)
5/8 (15.9)	52 (1321)	45 (1143)
3/4 (19.1)	60 (1524)	52 (1321)
7/8 (22.2)	69 (1753)	60 (1524)
1 (25.4)	74 (1880)	64 (1626)
1-1/4 (31.8)	88 (2235)	76 (1930)
1-1/2 (38.1)	100 (2540)	87 (2210)
2 (50.8)	125 (3175)	110 (2794)
2-1/2 (63.5)	125 (3175)	110 (2794)
3 (76.2)	125 (3175)	110 (2794)

Figure 15 – max unsupported tubes (Standards | TEMA, 2019)

For our shell, we find the minimum shell thickness via the following table, we find that we may use either a pipe of schedule 20, or plate of 7.9mm thickness. We opt to use the plates at this time, hence our thickness of our shell is 7.9mm, and therefore our outer shell OD is 0.343m:

TABLE CB-3.13
MINIMUM SHELL THICKNESS
Dimensions in Inches (mm)

Nominal Shell Diameter		Minimum Thickness				
		Carbon Steel			Alloy *	
		Pipe	Plate			
6	(152)	SCH. 40	-		1/8	(3.2)
8-12	(203-205)	SCH. 30	-		1/8	(3.2)
13-23	(330-584)	SCH. 20	5/16	(7.9)	1/8	(3.2)
24-29	(610-737)	-	5/16	(7.9)	3/16	(4.8)
30-39	(762-991)	-	3/8	(9.5)	1/4	(6.4)
40-60	(1016-1524)	-	7/16	(11.1)	1/4	(6.4)
61-80	(1549-2032)	-	1/2	(12.7)	5/16	(7.9)
81-100	(2057-2540)	-	1/2	(12.7)	3/8	(9.5)

*Schedule 5S is permissible for 6 inch (152 mm) and 8 inch (203 mm) shell diameters.

Figure 16 – minimum shell thickness (Standards | TEMA, 2019)

Tubesheet and bolts

Now we estimate the thickness required of the tubesheet via the following table, hence for our initial design we use a tubesheet thickness of 25.4mm:

C-7.11 MINIMUM TUBESHEET THICKNESS WITH EXPANDED TUBE JOINTS

In no case shall the total thickness minus corrosion allowance, in the areas into which tubes are to be expanded, of any tubesheet be less than three-fourths of the tube outside diameter for tubes of 1" (25.4 mm) OD and smaller, 7/8" (22.2 mm) for 1 1/4" (31.8 mm) OD, 1" (25.4mm) for 1 1/2" (38.1 mm) OD, or 1 1/4" (31.8 mm) for 2" (50.8 mm) OD.

Figure 17 – (Standards | TEMA, 2019)

TEMA standards outline the bolt size and spacing. We will use a bolt size of M12, as per the minimum, and a bolt spacing of over 31.7mm, as recommended from the following charts:

C-11.1 MINIMUM BOLT SIZE

The minimum recommended bolt diameter is 1/2" (M12). If bolting smaller than 1/2" (M12) is used, precautions shall be taken to avoid overstressing the bolting. Dimensional standards are included in Section 9, Table D-5. Metric bolting is shown in Section 9, Table D-5M.

Figure 18 – Bolt sizing (Standards | TEMA, 2019)

TABLE D-5M

METRIC BOLTING DATA - RECOMMENDED MINIMUM

(All Dimensions in Millimeters Unless Noted)

Bolt Size d_B	Threads		Nut Dimensions		Bolt Spacing B	Radial Distance R_h	Radial Distance R_r	Edge Distance E	Bolt Size d_B
	Pitch	Root Area (mm^2)	Across Flats	Across Corners					
M12	1.75	72.398	21.00	24.25	31.75	20.64	15.88	15.88	M12
M16	2.00	138.324	27.00	31.18	44.45	28.58	20.64	20.64	M16
M20	2.50	217.051	34.00	39.26	52.39	31.75	23.81	23.81	M20
M22	2.50	272.419	36.00	41.57	53.98	33.34	25.40	25.40	M22
M24	3.00	312.748	41.00	47.34	58.74	36.51	28.58	28.58	M24
M27	3.00	413.852	46.00	53.12	63.50	38.10	29.00	29.00	M27
M30	3.50	502.965	50.00	57.74	73.03	46.04	33.34	33.34	M30
M36	4.00	738.015	60.00	69.28	84.14	53.97	39.69	39.69	M36
M42	4.50	1018.218	70.00	80.83	100.00	61.91		49.21	M42
M48	5.00	1342.959	80.00	92.38	112.71	68.26		55.56	M48
M56	5.50	1862.725	90.00	103.92	127.00	76.20		63.50	M56
M64	6.00	2467.150	100.00	115.47	139.70	84.14		66.68	M64
M72	6.00	3221.775	110.00	127.02	155.58	88.90		69.85	M72
M80	6.00	4076.831	120.00	138.56	166.69	93.66		74.61	M80
M90	6.00	5287.085	135.00	155.88	188.91	107.95		84.14	M90
M100	6.00	6651.528	150.00	173.21	207.96	119.06		93.66	M100

Figure 19 – Metric bolt minimums (Standards | TEMA, 2019)

Verification

We must verify the Initial estimate for the overall heat capacity, and if need be, iterate.

We start by determining the tube-side heat transfer coefficient. We know that the length of the shell divided by pipe OD is 128. We find that the thermal conductivity of water is 0.6145 W/m K (Water - Thermal Conductivity Vs. Temperature, 2023), and hence we can determine the Prandtl Number via the following equation:

$$\left(C_p \mu / k_f\right)$$

We find it to be 5.11, and we use it in our calculation of j_h factor in the following chart:

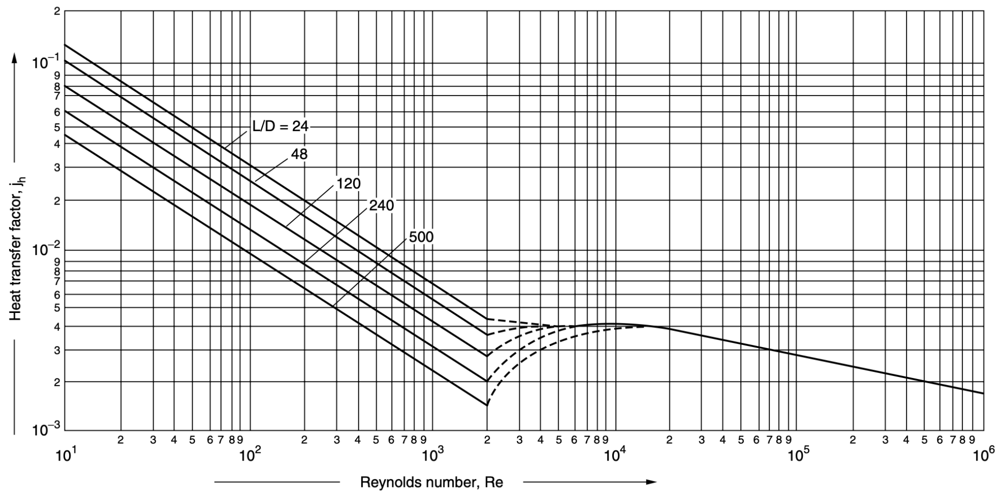


Figure 12.23. Tube-side heat-transfer factor

Figure 20 – tube heat transfer coefficient (Coulson & Richardson, 2017)

We find it to be 0.004, and we use it in our tube-side heat transfer coefficient equation:

$$h_i = \frac{k_f j_h Re Pr^{0.33}}{d_i}$$

To yield a value of 3172.30 W/m^2 K as our tubeside heat transfer coefficient.

Now for our shell side heat transfer coefficient, we consult literature to find the thermal conductivity of p-xylene and ethanol at close to the average temperature inlet and outlet to be 0.1142 W/m K (Mylona et al., 2014) and 0.166 W/m K respectively.

We thus calculate the shell-side prantyl number and use our baffle cut of 25% to read the j_h value of 0.004 from the following chart:

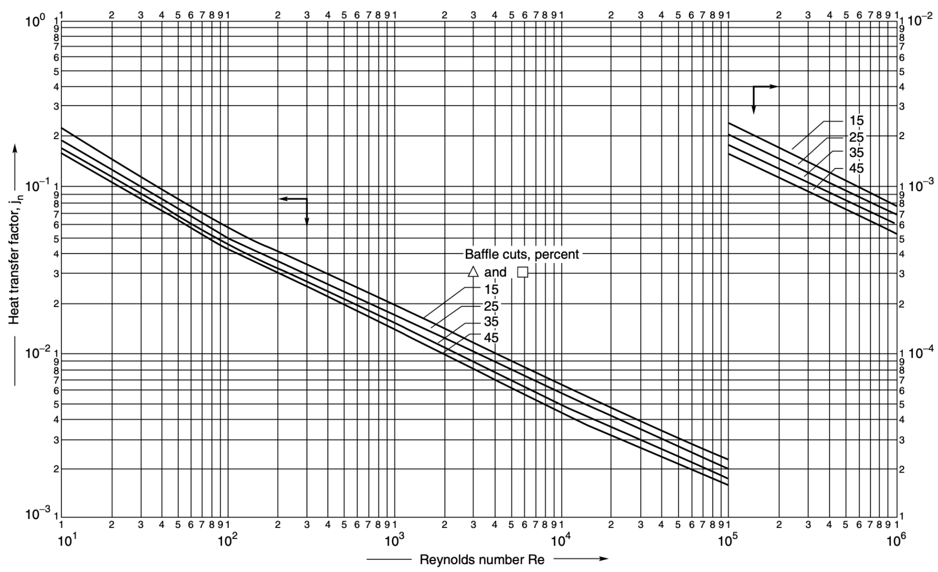


Figure 12.29. Shell-side heat-transfer factors, segmental baffles

Figure 21 – shell side heat transfer (Coulson & Richardson, 2017)

We then calculate the shell side heat transfer coefficient to be 1281.87 W/m K via the following equation:

$$h_s = \frac{k_f j_h Re Pr^{1/3}}{d_e}$$

To calculate the overall heat transfer coefficient, we must first find the thermal conductivity of the piping material. Our piping material is copper, as determined in earlier sections and thus from the following table we estimate it as 378 W/m K at the elevated temperature:

Table 12.6. Conductivity of metals

Metal	Temperature (°C)	k_w (W/m°C)
Aluminium	0	202
	100	206
Brass	0	97
(70 Cu, 30 Zn)	100	104
	400	116
Copper	0	388
	100	378
Nickel	0	62
	212	59
Cupro-nickel (10 per cent Ni)	0–100	45
Monel	0–100	30
Stainless steel (18/8)	0–100	16
Steel	0	45
	100	45
	600	36
Titanium	0–100	16

Figure 22 – Metal Conductivity (Coulson & Richardson, 2017)

Fouling factors (h_fi and h_fo) in the tube and shell side respectively can be estimated via the following table:

Table 12.2. Fouling factors (coefficients), typical values

Fluid	Coefficient (W/m²°C)	Factor (resistance) (m²°C/W)
River water	3000–12,000	0.0003–0.0001
Sea water	1000–3000	0.001–0.0003
Cooling water (towers)	3000–6000	0.0003–0.00017
Towns water (soft)	3000–5000	0.0003–0.0002
Towns water (hard)	1000–2000	0.001–0.0005
Steam condensate	1500–5000	0.00067–0.0002
Steam (oil free)	4000–10,000	0.0025–0.0001
Steam (oil traces)	2000–5000	0.0005–0.0002
Refrigerated brine	3000–5000	0.0003–0.0002
Air and industrial gases	5000–10,000	0.0002–0.0001
Flue gases	2000–5000	0.0005–0.0002
Organic vapours	5000	0.0002
Organic liquids	5000	0.0002
Light hydrocarbons	5000	0.0002
Heavy hydrocarbons	2000	0.0005
Boiling organics	2500	0.0004
Condensing organics	5000	0.0002
Heat transfer fluids	5000	0.0002
Aqueous salt solutions	3000–5000	0.0003–0.0002

Figure 23 – fouling factors (Coulson & Richardson, 2017)

We Use Cooling water or towns water for the inner tube, and hence we can estimate the factor to be 3000 W/m^2 K, and p-xylene is between a heavy and a light hydrocarbon, hence we pick a value of between those of 2500 W/m^2 K.

We thus calculate 1/U via the following equation:

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{1}{h_{od}} + \frac{d_o \ln\left(\frac{d_o}{d_i}\right)}{2k_w} + \frac{d_o}{d_i} \times \frac{1}{h_{id}} + \frac{d_o}{d_i} \times \frac{1}{h_i}$$

Taking the inverse, we get a final value for the overall heat transfer coefficient of 507.75 W/m^2 K. It is within 10% of our initial estimation, hence we will accept our design.

Managerial Aspects

Safety issues

The main safety issue is a loss of containment of fluid. Ethanol and p-xylene are both flammable liquids and hence must be handled with care, and ensured that no sources of ignition are nearly. Additionally, leakage monitoring, either digital or physical must be put in place to ensure that leaks from nozzles or piping are detected.

Overpressure monitoring should be put in place in the upstream and downstream processes to ensure that the piping network does not burst, and that fluids are safely contained. As the heat exchanger is not classified as a pressure vessel, overpressure must not be allowable to ensure that it complies with regulations and design specifications. Vapour release valves could be looked into, to release any unwanted vapour that may arise.

Furthermore, corrosion on the tube side poses a significant risk, potentially causing the feed fluid to leak into the water. Active monitoring should be established to ensure the process fluid remains uncontaminated by the feed fluid, with appropriate disposal strategies ready to be deployed if contamination occurs. Moreover, the potential for fouling and sediment build-up on the water side could diminish the water's cooling capacity, thereby inadequately cooling the feed fluid. Continuous monitoring of the heat exchanger's output temperature is essential to prevent downstream processes, like p-xylene storage tanks, from exceeding allowable design temperatures.

Before operation, a thorough analysis of risks should take place via a HAZOP to ensure that all potential hazards are mitigated or extinguished.

Operational and managerial issues

A systematic inspection and maintenance schedule should be instituted to prevent extensive fouling or sedimentation on either side of the heat exchanger. This includes routine checks for corrosion, verifying adequate pressure drops, assessing heat transfer coefficients, and monitoring output temperatures.

Records of inspections and repairs conducted on the heat exchanger should be kept to ensure adequate documentation, to demonstrate compliance with regulations and provide historical records into the condition of the heat exchanger for the future.

Standard operating procedures should be followed for the safe operation of the heat exchanger, and clearly implemented. Operators should have clearly defined roles and responsibilities, outlines of start-up and shutdown procedures, and emergency response protocols to ensure that the heat exchanger is not subject to extreme conditions outside of specifications.

Compliance with regulations

The design, fabrication and operation of the heat exchanger should comply with the most recent TEMA regulations and guidelines. Additional regulations, such as AS4343 for piping and AS1200 for non-pressure vessels, and AS3857 should be complied with to ensure safe operation. Additional regulations such as local council regulations should also be adhered to.

Operators should stay updated with any changes in regulations that arise to ensure that the heat exchanger stays in complains throughout its lifecycle. Coordination with regulatory bodies and authorised inspection agencies should occur to obtain necessary certifications. Including for design and fabrication, and for ongoing inspections.

Documentation, including design calculations, certificates, inspection reports, and permits should be kept. An organised document control system should be put in place to ensure quick retrieval of documents.

References

S, W. (2023). *Heat Transfer by Shell and Tube Heat Exchangers*. Retrieved September 17, 2023, from https://www.wermac.org/equipment/heatexchanger_part5.html

FUNDAMENTALS OF VAPOR–LIQUID PHASE EQUILIBRIUM. (2023). Retrieved September 17, 2023, from https://catalogimages.wiley.com/images/db/pdf/0471778885.c01.pdf

Porter, K. E. (2008, August 4). *DISTILLATION*. Begellhouse eBooks. https://doi.org/10.1615/atoz.d.distillation

Bell, K. J. (2008, August 4). *TEMA STANDARDS*. Begellhouse eBooks. https://doi.org/10.1615/atoz.t.tema_standards

Prices and charges for business. (2023). South East Water. Retrieved September 17, 2023, from https://southeastwater.com.au/business/accounts-and-billing/prices-and-charges-business/

p-Xylene. (2023). Retrieved September 17, 2023, from https://webbook.nist.gov/cgi/cbook.cgi?ID=C106423&Mask=1A8F#ref-12

P-Xylene Compound. (2023). PubChem. Retrieved September 17, 2023, from https://pubchem.ncbi.nlm.nih.gov/compound/P-Xylene#section=Decomposition

Tubular Heat Exchanger. (2023). Retrieved September 17, 2023, from https://www.svlele.com/he_info.htm#:~:text=Tube%20OD%3A%20Most%20important%20parameter,%2F4%22%20)%20or%20higher.

Water - Dynamic (Absolute) and Kinematic Viscosity vs. Temperature and Pressure. (2023). Retrieved September 17, 2023, from https://www.engineeringtoolbox.com/water-dynamic-kinematic-viscosity-d_596.html

Source A Wholesale u tube heat exchanger price - Alibaba.com. (2023). Retrieved September 17, 2023, from https://www.alibaba.com/showroom/u-tube-heat-exchanger-price.html

Vengateson, U. (2010, May). Design of multiple shell and tube heat exchangers in series: E shell and F shell. *Chemical Engineering Research and Design*, *88*(5–6), 725–736. https://doi.org/10.1016/j.cherd.2009.10.005

Gulyani, B. B. (2000, February 14). Estimating Number of Shells in Shell and Tube Heat Exchangers: A New Approach Based on Temperature Cross. *Journal of Heat Transfer*, *122*(3), 566–571. https://doi.org/10.1115/1.1287159

MULTIPLE PASS AND CROSS FLOW HEAT EXCHANGERS. (2023). Retrieved September 17, 2023, from https://paragchaware.files.wordpress.com/2016/08/ntu.pdf

Magazoni, F. C., Cabezas-Gómez, L., Alvariño, P. F., & Saiz-Jabardo, J. M. (2019, June). THERMAL PERFORMANCE OF ONE-PASS SHELL-AND-TUBE HEAT EXCHANGERS IN COUNTER-FLOW. *Brazilian Journal of Chemical Engineering*, *36*(2), 869–883. https://doi.org/10.1590/0104-6632.20190362s20180424

Garg, S., Banipal, T., & Ahluwalia, J. (1993, January). Heat capacities and densities of liquid o-xylene, m-xylene, p-xylene, and ethylbenzene, at temperatures from 318.15 K to 373.15 K and at pressures up to 10 MPa. *The Journal of Chemical Thermodynamics*, *25*(1), 57–62. https://doi.org/10.1006/jcht.1993.1007

Chemical Compatibility Guide. (2023). Retrieved September 17, 2023, from https://www.graco.com/content/dam/graco/ipd/literature/misc/chemical-compatibility-guide/Graco_ChemCompGuideEN-B.pdf

Components of Shell-and-Tube Heat Exchangers. (2013). Retrieved September 17, 2023, from https://www.codesignengg.com/new/pvnewsletter/PV%20Newsletter%20-%20March%202013.pdf

Carbon Steel Chemical Compatibility Chart. (2023). Retrieved September 17, 2023, from https://www.calpaclab.com/carbon-steel-chemical-compatibility-chart/

Water - Thermal Conductivity vs. Temperature. (2023). Retrieved September 17, 2023, from https://www.engineeringtoolbox.com/water-liquid-gas-thermal-conductivity-temperature-pressure-d_2012.html

Mylona, S. K., Antoniadis, K. D., Assael, M. J., Huber, M. L., & Perkins, R. A. (2014, November 21). Reference Correlations of the Thermal Conductivity of o-Xylene, m-Xylene, p-Xylene, and Ethylbenzene from the Triple Point to 700 K and Moderate Pressures. *Journal of Physical and Chemical Reference Data*, *43*(4). https://doi.org/10.1063/1.4901166

Ethanol - Specific Heat vs. Temperature and Pressure. (2023). Retrieved September 17, 2023, from https://www.engineeringtoolbox.com/specific-heat-capacity-ethanol-Cp-Cv-isobaric-isochoric-ethyl-alcohol-d_2030.html#:~:text=At%20ambient%20pressure%20and%20temperature,%5Bcal%2Fg%20K%5D

Water - Specific Heat vs. Temperature. (2023). Retrieved September 17, 2023, from https://www.engineeringtoolbox.com/specific-heat-capacity-water-d_660.html?vA=306&units=K

Ethanol - Density and Specific Weight vs. Temperature and Pressure. (2023). Retrieved September 17, 2023, from https://www.engineeringtoolbox.com/ethanol-ethyl-alcohol-density-specific-weight-temperature-pressure-d_2028.html

Water - Density, Specific Weight and Thermal Expansion Coefficients. (2023). Retrieved September 17, 2023, from https://www.engineeringtoolbox.com/water-density-specific-weight-d_595.html

P-Xylene (data page). (2023, April 11). en.wikipedia.org. https://en.wikipedia.org/wiki/P-Xylene_(data_page)#cite_note-lange1669-2

Ethanol - Dynamic and Kinematic Viscosity vs. Temperature and Pressure. (2023). Retrieved September 17, 2023, from https://www.engineeringtoolbox.com/ethanol-dynamic-kinematic-viscosity-temperature-pressure-d_2071.html

Water - Dynamic (Absolute) and Kinematic Viscosity vs. Temperature and Pressure. (2023). Retrieved September 17, 2023, from https://www.engineeringtoolbox.com/water-dynamic-kinematic-viscosity-d_596.html

Brogan, R. (2008, August 4). *SHELL AND TUBE HEAT EXCHANGERS*. Begellhouse eBooks. https://doi.org/10.1615/atoz.s.shell_and_tube_heat_exchangers

Standards | TEMA. (2019). Retrieved September 17, 2023, from https://tema.org/standards/

Chhabra, R. P., & Shankar, V. (2017, November 28). *Coulson and Richardson's Chemical Engineering*. Butterworth-Heinemann.

Walas, S. M. (1988, January 1). *Chemical Process Equipment*. Butterworth-Heinemann.