HEAT EXCHANGER DESIGN

Prepared by

Name: Spyros Ploussiou (sp872) College: Wolfson College Date: 02/05/2019

Executive summary

The main objective of this project was to design a shell and tube heat exchanger to perform a particular duty. Its estimated capital cost based on tube material of construction was then compared to that of a gasketed plate heat exchanger of the same material make-up, performing the same duty.

The ethylene glycol process fluid was allocated to the shell-side while the available poor-quality cooling water was allocated to the tube-side of the heat exchanger. Maintenance, cleaning, fouling considerations and fluid velocity comparisons were the critical decision factors behind the fluid allocation scenario chosen. A split-ring floating head heat exchanger with an oval shaped end and a flat end was selected to perform the required duty.

A series of iterations executed by computational means led to the selection of a reasonable design incorporating 4 tube fluid passes and 1 shell fluid pass. The estimated overall heat transfer coefficient based on outer-tube diameter was 396.5 W m^{-2} K^{-1} which was 13% higher than the required value and the total area available for heat transfer was 50.7 m^2. The selected tube arrangement was triangular with a 27 mm pitch. In addition, the tube outer and inner diameters of choice were 21.6 mm and 16 mm respectively with a tube length of 5.2 m. The total number of tubes estimated was 144 with 36 tubes per fluid pass. The velocities of the fluids in the tubes and shell were calculated at 0.916 m s^{-1} and 0.151 m s^{-1} respectively. The tube side heat transfer coefficient was calculated at 4928 W m^{-2} K^{-1} while that of the shell side was calculated at 656 W m^{-2} K^{-1}. Furthermore, the tube bundle diameter was calculated at 0.407 m and after allowing for shell clearance, that of the shell was estimated at 0.458 m. The baffle spacing was estimated at 0.279 m giving rise to an equivalent hydraulic mean diameter of 0.0153 m. Segmental baffles with a 25% cut were employed in the calculations.

With regard to pressure drop calculations, nozzles with outer and inner diameters of 49.5 mm and 46.5 mm respectively were employed, giving rise to an overall shell side pressure drop of 0.097 bar and a tube side pressure drop of 0.333 bar, both of which were below the design specification of 0.5 bar. In terms of mechanical aspects of the design, the shell wall thickness was estimated at 3.2 mm and that of the tube sheet was estimated at 16.2 mm. For the shell ends, the oval shaped end thickness was estimated at 0.48 mm whereas that for the flat end was estimated at 6.65 mm.

Concerning the gasketed plate heat exchanger for which a total number of 51 plates was estimated in order to perform the same duty with grade 304 stainless steel as the main material design component, a total material mass of 612 kg was calculated. An equivalent total mass of 991 kg was calculated for the shell and tube heat exchanger. Grade 304 stainless steel was also the principal material component make-up of both the shell and tubes.

Based on the analysis conducted, the gasketed plate heat exchanger designed to perform the same duty seems to be the more economical option, but other factors must also be considered such as installation costs, cleaning and maintenance costs, approximate operating costs and an overall total initial capital expenditure approximation.

1 Introduction

1.1 Outline of project

The main target of this project was to design in detail, including thermal, mechanical, hydraulic calculations and simple engineering drawings, a simple shell and tube heat exchanger to perform the cooldown of 15 000 kg h^{-1} of ethylene glycol from 90 °C to 40 °C using poor quality cooling water available at 20 °C and limited to a maximum outlet temperature of 45 °C. The ethylene glycol is available for processing at a pressure of 10 bar and the cooling water is available at 8 bar. Design specification limits the maximum allowable pressure drop on either side of the heat exchanger to 0.5 bar.

Physical properties of either fluid were to be appropriately obtained from credible scientific literature. Suitable fouling resistance estimates were to be also made for either fluid. Furthermore, simple calculations were to be performed for a gasketed plate heat exchanger carrying out the same duty and of the same material make up as the shell and tube heat exchanger, for capital cost comparison purposes.

1.2 Properties and common applications of Ethylene glycol

Ethylene glycol is an organic compound with a saturated hydrocarbon chain length consisting of 2 hydroxyl side groups which account for its moderately high solubility in water. It's classified as an irritant that is colourless, odourless and significantly more viscous than water, especially in the temperature range of 20-40 °C **[1]**. Figure 1 depicts its structure in detail in terms of expected physical and chemical properties.

Figure 1: Skeletal formula of ethylene glycol

Ethylene glycol is one of the most fundamental glycols available commercially and is synthesised on a large scale particularly in the U.S **[1]**. It's mainly used as a coolant in automobiles due to its low molecular weight and low volatility. It also has good antifreeze properties for de-icing fluids and heat transfer agents. For example, it is employed in the aircraft industry for de-frosting the wings and fuselage of an aircraft. It's also used widely as a raw material in the production of polyester resins used in the manufacture of films and fibres. Moreover, it acts as a stabiliser in terms of preventing gel formation in water dispersions. In addition, when in its high purity form, ethylene glycol can be utilised as a solvent for ammonium perborate which is a commonly used conductor in electrolytic capacitors **[2]**.

Ethylene glycol has viscosity that exhibits high temperature dependence. For instance, its viscosity at 90 °C is approximately 0.00244 Pa s whereas at 40 °C it is approximately 0.00910 Pa s **[3]**. It has lower fouling properties than poor quality cooling water but is relatively corrosive, a property which can be minimised by use of corrosion inhibitors **[2]**. Having a higher viscosity signifies that ethylene glycol tends to flow in the laminar flow regime which tends to reduce its film heat transfer coefficient. As a result, higher hydraulic diameters based on the geometry allowed for flow and higher fluid velocities are required to ensure that ethylene glycol exhibits turbulent flow. Pressure drops in the direction of ethylene glycol flow may therefore be higher than expected.

1.3 Common types of heat exchangers

A shell and tube heat exchanger, which is one of the most common types encountered in petrochemical plants, is simply a bundle of tubes mounted in a cylindrical shell enclosure. It's a typical unit of process equipment in the chemical industry which carries two fluids individually allocated on either the tube or shell side of the heat exchanger which are separated by the ends of the tubes. The ends of the tubes are contoured into tube sheets. In order to provide rigid support for the tubes, baffles are installed in the unit. This also aids in directing the flow of the shell-fluid across the tubes evenly.

Another common type of heat exchanger unit is that of a gasketed-plate one. It's a specialised kind of heat exchanger made of a stack of uniformly spaced, narrow plates braced together in a stand. A gasket is used to seal each individual narrow plate around the edges.

The gasketed-plate heat exchanger is mostly used in operations that require easy, quick and flexible cleaning. It's preferred in industrial settings where flexibility is paramount and when highly viscous fluids are involved (Coulson & Richardson Chemical Engineering Design, Vol.6, p.756-757). Shell and tube heat exchangers on the other hand, are used to process fluids at considerably high pressures in applications requiring high surface area to volume ratio (Coulson & Richardson Chemical Engineering Design, Vol.6, p.640-641) **[4]**. Table 1 lists some of the advantages and disadvantages of both shell and tube and gasketed plate heat exchangers.

Table 1: List of advantages and disadvantages of gasketed-plate and shell and tube heat exchangers respectively.

Gasketed-plate heat exchanger	
Advantages	**Disadvantages**
• Small pressure drops across either side thus saving on energy costs. • Relatively cheap and simple to design. • Can be constructed from a wide range of materials. • Supports relatively high operating pressures and temperatures.	• More difficult to disassemble and clean than a gasketed-plate heat exchanger. • Not as suitable for processing highly viscous fluids as gasketed-plate heat exchangers.
Shell and tube heat exchanger	
Advantages	**Disadvantages**
• Plates are preferable to shell and tube when material costs are high. • Easier to maintain, clean and service. More flexible in this sense. • Less prone to fouling effects than shell and tube heat exchangers. • Generally, offer a higher log-mean temperature difference correction factor F_t.	• More susceptible to failure under high pressure and high temperature than shell and tube heat exchangers. Maximum operating temperature limited to about 250 °C and definite failure above operating pressures of 30 bar (Coulson & Richardson Chemical Engineering Design, Vol.6, p.756-757) **[4]**.

Cooling water and ethylene glycol have relatively significant fouling factors. Furthermore, ethylene glycol is of considerably high viscosity. These conditions are more appropriate for a gasketed plate heat exchanger.

Despite the afore-mentioned conditions, the heat exchange duty required for the cooldown of the ethylene glycol process stream is relatively large. In addition, a shell and tube heat exchanger is more reliable and there already exist for it well-documented fabrication techniques and design procedures **[4]**.

2 Design decisions and discussion

2.1 Shell and tube side fluid allocation

Poor quality water consisting of a variety of impurities in solution or suspension is more likely to corrode metallic surfaces than ethylene glycol. In addition, at both inlet and outlet temperatures, ethylene glycol is much more viscous than the available cooling water. By allocating the ethylene glycol to the shell side of the heat exchanger, the chances of obtaining a higher heat transfer coefficient are therefore maximised. The corrosion factor also justifies the choice of allocating the cooling water to the tube side since, this effectively minimises the cost of expensive metal alloy parts used in the heat exchanger unit.

Moreover, despite the fact that ethylene glycol is supplied at a higher pressure of 10 bar than the cooling water at 8 bar, indicating that the cooling water could potentially be re-allocated to the shell-side of the heat exchanger, their difference in fouling resistances suggests otherwise. The cooling water has a fouling factor of approximately 2.5×10^{-4} W^{-1} m^2 K whereas ethylene glycol has a fouling resistance of approximately 2×10^{-4} W^{-1} m^2 K (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 640). The fact that the poor-quality cooling water fouls surfaces more than ethylene glycol is therefore supportive of the allocation of cooling water to the tube side of the heat exchanger. The tubes of the heat exchanger allow higher fluid velocities to be achieved than the shell side and by maximising the cooling water velocity, the possibility of fouling is reduced, simplifying servicing and cleaning operations at the same time **[4]**.

In total, given that the fluid allocation criterion was collaterally examined with respect to both process fluids available, the ethylene glycol was finally allocated to the shell side while the cooling water was finally allocated to the tube side.

2.2 Selection of design materials of construction and head type

Materials of fabrication commonly used in the design of shell and tube heat exchangers include aluminium alloys, copper alloys, carbon steel and stainless steel amongst others **[6]**. Table 2 shows approximate prices in USD per kg of some of these materials.

Table 2: List of prices in for common materials of construction employed in shell and tube heat exchangers **[5]**.

Type of material	Cost / $ kg^{-1}
Copper alloys	2.25
Titanium alloys	16.25
Carbon steel	1.01
Aluminium alloys	1.80
Grade 304 stainless steel	2.53
Grade 316 stainless steel	3.74

Grade 304 stainless steel is the best option covering the design of both the tubes and shell since it offers a good combination of strength, corrosion resistance and affordable pricing in \$ kg^{-1}. Grade 316 stainless steel is much more expensive and is not worth the additional capital expenditure since it offers minimal increase in strength and flexibility compared to grade 304 stainless steel. Carbon steel could also be considered in terms of its pricing as well as the fact that it's relatively lightweight. On the other hand, the difference in strength and corrosion resistance with grade 304 stainless steel makes the latter more attractive and the best option.

For design purposes, a split ring floating-head heat exchanger was employed. This type of shell and tube heat exchanger has the advantage of being easier to service and clean. Differential expansion between the shell and tubes is not significant enough given the design specifications but nevertheless the choice of a split ring floating-head heat exchanger also covers that scenario.

2.3 Synopsis of steps required to finalise the design

The maximum allowable outlet temperature of the cooling water is 45 °C. The relevant temperature employed in performing the calculations was 40 °C, operating close to the maximum one permitted. At this temperature the value of F_t is satisfactory, at approximately 0.95. At higher temperatures it would be considerably lower.

Applying overall energy balances using C_p values extracted from literature for ethylene glycol and water at their respective mean temperatures of 65 °C and 30 °C, allowed calculation of the cooling water mass flowrate $\dot{m}_{H2O}$. All necessary physical properties for both process fluids involved were obtained from literature at inlet, outlet and mean stream temperatures.

An initial estimate of the overall heat transfer coefficient U_o was made in the range of 250-450 W m^{-2} K^{-1} which is typical for heat exchangers of this type and as a result, the required heat transfer area for the specified duty was calculated. The log-mean temperature difference ΔT_{lm} was corrected to ΔT_m based on the estimated F_t value for a heat exchanger design incorporating 1 shell and 4 tube passes. Initial estimates of the design variables L, d_i and d_o were made and the required tube pitch p_t was also calculated using the recommended relationship of $p_t = 1.25d_o$ for the chosen triangular tube arrangement (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 646) **[4]**. As a result, the total number of tubes required N_t and tube side fluid velocity u_t were calculated. The bundle and shell diameters D_b and D_s were also subsequently computed using equations obtainable from literature. The baffle spacing L_B as well as the % baffle cut were also adjusted accordingly.

The Reynolds number Re and Prandtl number Pr were successively calculated for the tube side fluid. By extracting the heat transfer factor j_h from a suitable chart and using an appropriate correlation (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 665-666) **[4]**, the Nusselt number Nu and tube side film heat transfer coefficient h_i were estimated.

L_B was appropriately re-adjusted and the cross-flow area in between the tubes A_{CF} as well as the equivalent hydraulic mean diameter d_e were also computed using equations available in literature. The values of Re, Pr and shell side fluid velocity u_s were successively calculated. The value of j_h for the shell side was obtained from an appropriate chart and by use of a suitable correlation, Nu for the shell fluid and the film heat transfer coefficient of the shell fluid h_o were estimated (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 673, 675) **[4]**.

Using fluid fouling resistance estimates obtained from literature and the thermal conductivity of the material make-up of the tubes, the overall heat transfer coefficient was calculated and compared to that initially estimated. The whole process was repeated until both values converged by using U_o values in between successive estimates.

The entire procedure involved perpetual re-adjustment of the main design variables, putting more emphasis mainly on the re-adjustment of L_B and the % baffle cut. The iterations were repeatedly performed until both the initial estimated U_o and calculated U_o converged and the overall pressure drops on both the tube and shell side ΔP_t and ΔP_s respectively met design specification constraints. When hydraulic and thermal calculations were completed, the mechanical design of the heat exchanger was thoroughly considered using simple thin-walled pressure vessel theory and equations obtainable from literature.

3 Safety health & the environment

3.1 Design safety aspects

Ethylene glycol is poisonous, an irritant and corrosive. The poor-quality cooling water provided is also expected to be considerably corrosive. Nonetheless, the fluids involved are in liquid form and are therefore less hazardous than vapours and gases, given the specified operating pressures of both process streams. The ethylene glycol has entry and exit temperatures of 90 °C and 40 °C respectively at a specified pressure of 10 bar. The pressure of the ethylene glycol stream is therefore considerably high. In case of leakage it may pose a threat, but this can be averted by additionally shielding the shell component of the heat exchanger unit.

Relief or throttling valves could also be installed on either process stream due to the high operating pressures involved but, this would be highly unnecessary given that the process fluids are not highly pressurised vapours.

Moreover, the velocities of the shell and tube side fluids were calculated at 0.916 m s^{-1} and 0.151 m s^{-1} respectively, and these are well below the 3 m s^{-1} critical velocity thresholds for flow-induced tube vibrations (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 654) **[4,7]**. As a result, negligible damage to heat exchanger components is to be expected from this phenomenon.

Furthermore, by using a type floating-head shell and tube heat exchanger, the effects of thermal expansion on the mechanical integrity of the tubes are minimised.

3.2 Cleaning and maintenance

Prior to cleaning the heat exchanger unit, the equipment must be safely shut down. Both ethylene glycol and cooling water flows must cease, and the equipment must be suitably depressurised, drained and purged of any hazardous material (TEMA standards 9^{th} edition section 4 p.5) **[7]**.

When attending to the tubes, blowing hot steam is not recommended for cleaning purposes. Hot steam may result in heating up the tubes, causing severe expansion strains and deformation. On the other hand, hydro-blasting with water is a much better option (TEMA standards 9^{th} edition section 4 p.7) **[7]**.

The heat exchanger may be cleaned either by use of mechanical or chemical means. For instance, commercial cleaning chemical agents may be employed at high washing velocities to remove fouling agent deposits from either the tubes or shell. In addition, high pressure water impact jets may be employed as afore-mentioned, to ensure complete removal of fouling deposits **[7]**.

When mechanical means are employed to clean the tube bundle and tube sheets, care should be taken not to damage any delicate parts **[7]**.

When disconnecting the heat exchanger unit from the main process network, the fluid flow in the pipelines must first be re-directed from the unit. Moreover, with regard to any pumping system already in place, it must be first switched off before shutting off main valve connections to the heat exchanger unit. Pressure conditions in the heat exchanger unit must never exceed specified design conditions and as a result, upon disassembly of the unit for cleaning, suitable shut-off valves must be employed **[7]**.

3.3 Dealing with any possible leaks

In the scenario of shell fluid leakage into the tubes, the pressure inside the tubes will be higher than the normal operating pressure of 8 bar because ethylene glycol is supplied at 10 bar. As a result, upon conducting mechanical design calculations and determining the tube wall thickness, an additional calculation was performed to ensure that the tube walls are able to resist the pressure of the mixed fluids during a leak.

The mixing of ethylene glycol and water results in an aqueous solution instead of 2 separate immiscible liquid systems. Two phase flow upon leaking is therefore highly unlikely. In addition, ethylene glycol and water do not undergo any highly exothermic chemical reactions.

In the case that leakages are detected in the heat exchanger unit, both stream fluid flows should be ceased. This must be performed by cautiously slowing down the ethylene glycol flow and subsequently stopping the cooling water flow (TEMA standards 9th edition section 4 p.5) **[7]**.

4 Thermal calculations

4.1 Step 1: Specification

The ethylene glycol and cooling water are available at 10 bar and 8 bar pressure respectively. The maximum allowable pressure drop on either stream is 0.5 bar. The fouling factor for ethylene glycol $\boldsymbol{R_{f,o}}$ employed in the thermal calculations was 2×10^{-4} W^{-1} m^{2} K whereas that for the cooling water $\boldsymbol{R_{f,i}}$ was 2.5×10^{-4} W^{-1} m^{2} K (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 640) **[4]**.

To begin with, the duty of the heat exchanger $\boldsymbol{Q}$ and water mass flowrate $\boldsymbol{\dot{m}_{H2O}}$ were calculated using equations 1, 2, 3 & 4, where $\boldsymbol{T_1}$ and $\boldsymbol{T_2}$ are the ethylene glycol inlet and outlet temperatures respectively and $\boldsymbol{t_1}$ and $\boldsymbol{t_2}$ are the cooling water inlet and outlet temperatures respectively. Table 3 lists the physical properties for ethylene glycol and cooling water utilised in the thermal calculations **[3]**. The outlet temperature of cooling water $\boldsymbol{t_2}$ was taken as 40 °C, just below the maximum permissible one in order to maximise the value of $\boldsymbol{F_t}$ for the chosen heat exchanger configuration.

$$\text{Ethylene glycol mean temperature} = \frac{T_1+T_2}{2} = 65\ °\text{C} \tag{1}$$

$$Q = \dot{m}_{Eg} \times C_{p,Eg} \times (T_1 - T_2) = \frac{150}{36} \times 2647 \times 50 = 551\,529 \text{ W} \quad (2)$$

$$\text{Cooling water mean temperature} = \frac{t_1+t_2}{2} = 30\ ℃ \quad (3)$$

$$\dot{m}_{H2O} = \frac{Q}{C_{p,H2O}(t_1-t_2)} = \frac{551529}{4186\times10} = 6.59 \text{ kg s}^{-1} \quad (4)$$

Table 3: List of physical properties of ethylene glycol and (cooling) water at inlet, outlet and average temperatures respectively **[3]**.

Fluid	**Cooling water [3]**			**Ethylene Glycol [3]**		
Condition	**Inlet**	**Average**	**Outlet**	**Inlet**	**Average**	**Outlet**
Temperature / °C	20	30	40	90	65	40
***k* / kg m s^{-3} K^{-1}**	0.599	0.616	0.631	0.263	0.261	0.259
***ρ* / kg m^{-3}**	999	996	993	1063	1081	1099
***μ* / 10^{-3} Pa s**	1.001	0.797	0.653	2.440	3.725	9.100

4.2 Step 2: Overall heat transfer coefficient

The overall heat transfer coefficient $\boldsymbol{U_o}$ for such a heat exchanger configuration is expected to lie between 250-450 W m^{-2} K^{-1} (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 639) **[4]**. The successful iteration resulted by use of the initial estimate of 350 W m^{-2} K^{-1}.

4.3 Step 3: Heat exchanger type and dimension criteria

A shell and tube heat exchanger incorporating 1 shell pass and 4 tube passes was selected. Equations 5, 6, 7 & 8 were used in calculating the corrected log-mean temperature difference $\boldsymbol{\Delta T_m}$. The temperature ratios $\boldsymbol{R}$ and $\boldsymbol{S}$ were appropriately made use of in determining $\boldsymbol{F_t}$ at 0.95 from charts obtainable from literature (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 658) **[4]**.

$$\Delta T_{lm} = \frac{(T_1-t_2)-(T_2-t_1)}{\ln\left(\frac{T_1-t_2}{T_2-t_1}\right)} = \frac{50-20}{\ln\left(\frac{50}{20}\right)} = 32.7\ ℃ \quad (5)$$

$$S = \frac{t_2-t_1}{T_1-t_1} = \frac{40-20}{90-20} = 0.286 \quad (6)$$

$$R = \frac{T_1-T_2}{t_2-t_1} = \frac{90-40}{40-20} = 2.50 \quad (7)$$

$$\Delta T_m = \Delta T_{lm} \times F_t = 32.7 \times 0.95 = 31.10\ ℃ \quad (8)$$

Once $\boldsymbol{\Delta T_m}$ was calculated at 31.10 °C, the heat transfer area $\boldsymbol{A}$ was estimated by use of equation 9.

$$A = \frac{Q}{U_o \times \Delta T_m} = \frac{551529}{350\times31.10} = 50.7 \text{ m}^2 \quad (9)$$

4.4 Step 4: Tube layout, size and number

Stainless steel was the material of choice for tube fabrication purposes and as justified previously, the cooling water was allocated to the tubes. An outer tube diameter $\boldsymbol{d_o}$ of 21.6 mm, inner tube diameter d_i of 16 mm and tube length L of 5.2 m were selected with regard to dimensions. The preferred tube arrangement was triangular with pitch p_t of 27 mm based on the relationship $p_t = 1.25d_o$ (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 649) **[4]**.

The total number of tubes required N_t was calculated using equations 10 & 11 and the number of tubes per pass for N_{pt} of 4 was calculated using equation 12. Moreover, the cooling water velocity u_t was calculated using equations 13, 14 and the steady state mass balance in equation 15.

$$\text{Single tube area} = \pi \times L \times d_o = \pi \times 0.0216 \times 5.2 = 0.353 \text{ m}^2 \tag{10}$$

$$N_t = \frac{A}{\text{Single tube area}} = \frac{50.7}{0.353} = 143.6 \text{ (approximately 144)} \tag{11}$$

$$\text{Tubes per pass} = \frac{N_t}{N_{pt}} = \frac{144}{4} = 36 \tag{12}$$

$$\text{Cross sectional area of tube} = \frac{\pi}{4} \times {d_i}^2 = \frac{\pi}{4} \times 0.016^2 = 2.01 \times 10^{-4} \tag{13}$$

$$\text{Area per pass} = \text{Tubes per pass} \times \text{Cross sectional area of tube} = 7.22 \times 10^{-3} \text{ m}^2 \tag{14}$$

$$u_t = \frac{\dot{m}_{H2O}}{\text{Area per pass} \times \rho_{H2O}} = \frac{6.59}{996 \times 7.22 \times 10^{-3}} = 0.916 \text{ m s}^{-1} \tag{15}$$

4.5 Step 5: Tube bundle and shell diameters

For the design incorporating 4 tube passes and a triangular tube arrangement, $K_1 = 0.175$ & $n_1 = 2.285$ (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 649). Equation 16 was employed in calculating bundle diameter D_b and after allowing for a shell clearance of 51 mm (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 651) **[4]** the shell diameter D_s was calculated using equation 17.

$$D_b = d_o \times \left(\frac{N_t}{K_1}\right)^{\frac{1}{n_1}} = 0.0216 \times \left(\frac{144}{0.175}\right)^{\frac{1}{2.285}} = 0.407 \text{ m} \tag{16}$$

$$D_s = D_b + 0.051 = 0.407 + 0.051 = 0.458 \text{ m} \tag{17}$$

4.6 Step 6: Tube side heat transfer coefficient

The cooling water Re and Pr were calculated using equations 18 & 19 respectively. The heat transfer factor j_h for use in calculating Nu in equation 20 was estimated at 4×10^{-3} (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 665) **[4]**.

$$Re = \frac{\rho_{H2O} u_t d_i}{\mu_{H2O}} = \frac{996 \times 0.916 \times 0.016}{7.97 \times 10^{-4}} = 18\,325 \quad \text{Turbulent flow} \tag{18}$$

$$Pr = \frac{\mu_{H2O} C_{p,H2O}}{k_{H2O}} = \frac{7.97 \times 10^{-4} \times 4186}{0.616} = 5.42 \tag{19}$$

$$Nu = j_h \times Re \times Pr^{0.33} = 0.004 \times 18325 \times 5.42^{0.33} = 128 \tag{20}$$

The tube side heat transfer coefficient h_i was calculated at 4928 W m^{-2} K^{-1} using equation 21.

$$h_i = \frac{Nu \times k_{H2O}}{d_i} = \frac{128 \times 0.616}{0.016} = 4928 \text{ W m}^{-2} \text{ K}^{-1} \tag{21}$$

4.7 Step 7: Shell side heat transfer coefficient

In estimating the shell side heat transfer coefficient $\boldsymbol{h_o}$, Kern's method was employed (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 671-675) **[4]**. The baffle spacing $\boldsymbol{L_B}$ was estimated using the approximation of $\boldsymbol{L_B} = 0.61\boldsymbol{D_s}$ which gave a value of 0.279 m. The area for cross flow in between the tubes $\boldsymbol{A_{CF}}$ was calculated using equation 22 and the hydraulic mean diameter for shell fluid flow $\boldsymbol{d_e}$ was calculated using equation 23. In addition, the ethylene glycol flow velocity $\boldsymbol{u_s}$ was calculated using equation 24 and its $\boldsymbol{Re}$ & $\boldsymbol{Pr}$ using equations 25 & 26 respectively.

$$A_{CF} = \frac{D_s \times L_B(p_t - d_o)}{p_t} = \frac{0.458 \times 0.279 \times (0.027 - 0.0216)}{0.027} = 0.0256 \text{ m}^2 \tag{22}$$

$$d_e = \frac{1.1}{d_o}\left(p_t^{\,2} - 0.917 d_o^{\,2}\right) = \frac{1.1}{0.0216}(0.027^2 - 0.917 \times 0.0216^2) = 0.0153 \text{ m} \tag{23}$$

$$u_s = \frac{\dot{m}_{Eg}}{\rho_{Eg} \times A_{CF}} = \frac{150/36}{1081 \times 0.0256} = 0.151 \text{ m s}^{-1} \tag{24}$$

$$Re = \frac{\rho_{Eg} u_s d_e}{\mu_{Eg}} = \frac{1081 \times 0.151 \times 0.0153}{3.725 \times 10^{-3}} = 670 \tag{25}$$

$$Pr = \frac{\mu_{Eg} c_{p,Eg}}{k_{Eg}} = \frac{3.725 \times 10^{-3} \times 2647}{0.261} = 37.8 \tag{26}$$

The shell side heat transfer factor $\boldsymbol{j_h}$ for use in calculating $\boldsymbol{Nu}$ in equation 27 was estimated at 1.9×10^{-2} (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 673) **[4]**. Segmental baffles with a 25 % cut were employed for the purposes of conducting the thermal calculations. The value of $\boldsymbol{h_o}$ was calculated at 718 W m^{-2} K^{-1} by use of equation 28.

$$Nu = j_h \times Re \times Pr^{0.33} = 0.019 \times 670 \times 37.8^{0.33} = 42.2 \tag{27}$$

$$h_o = \frac{Nu \times k_{Eg}}{d_e} = \frac{42.2 \times 0.261}{0.0153} = 718 \text{ W m}^{-2} \text{ K}^{-1} \tag{28}$$

4.8 Step 8: Overall heat transfer coefficient

The material selected for the design of the tubes is grade 304 stainless steel. Its thermal conductivity $\boldsymbol{k_s}$ is approximately 16.2 W m^{-1} K^{-1} **[8]**.

The thermal resistance of the tube walls $\boldsymbol{R_{wall}}$ was calculated using equation 29 and by taking $\boldsymbol{R_{f,i}}$ and $\boldsymbol{R_{f,o}}$ as 2.5×10^{-4} W^{-1} m^2 K and 2×10^{-4} W^{-1} m^2 K respectively, equation 30 was employed in order to calculate $\boldsymbol{U_o}$.

$$R_{wall} = \frac{d_o \times \ln\left(\frac{d_o}{d_i}\right)}{2 \times k_s} = \frac{0.0216 \times \ln\left(\frac{0.0216}{0.016}\right)}{2 \times 16.2} = 2.00 \times 10^{-4} \text{ W}^{-1} \text{ m}^2 \text{ K} \tag{29}$$

$$U_o = \left(R_{wall} + \frac{1}{h_o} + \frac{d_o}{d_i}\left(R_{f,i} + \frac{1}{h_i}\right) + R_{f,o}\right)^{-1} \tag{30}$$

$$U_o = \left(2 \times 10^{-4} + \frac{1}{718} + \frac{0.0216}{0.016}\left(2.5 \times 10^{-4} + \frac{1}{4928}\right) + 2 \times 10^{-4}\right)^{-1} = 416 \text{ W m}^{-2} \text{ K}^{-1}$$

The $\boldsymbol{U_o}$ value calculated of 416 W m^{-2} K^{-1} is approximately 18 % greater than the initial estimate of 350 W m^{-2} K^{-1} made.

4.9 Step 9: Implementation of viscosity correction factor

The viscosity correction factor $(\boldsymbol{\mu}/\boldsymbol{\mu_o})^{0.14}$ was initially neglected when performing the calculations with regard to $\boldsymbol{h_i}$ and $\boldsymbol{h_o}$. For the cooling water, viscosity correction gave an estimate for inner wall temperature $\boldsymbol{T_{w,i}}$ of 33 °C which is extremely close to 30 °C. As a result, viscosity correction for the cooling water was ignored. Between 30 and 33 °C the viscosity of water varies negligibly with temperature.

For the ethylene glycol, equations 31, 32 & 33 were employed in order to estimate the magnitude of the viscosity correction factor. The $\boldsymbol{T_{w,o}}$ was calculated in equation 33 at 49.8 °C which is significantly lower than the mean value of 65 °C .

$$\text{Outer tube area} = N_t \times \pi d_o \times L = 144 \times \pi \times 0.0216 \times 5.2 = 50.7 \text{ m}^2 \tag{31}$$

$$q_o = \frac{Q}{\text{Outer tube area}} = \frac{551529}{50.7} = 10\,886 \text{ W m}^{-2} \tag{32}$$

$$T_{w,o} = \frac{T_1+T_2}{2} - \frac{q_o}{h_o} = 65 - \frac{10886}{718} = 49.8 \text{ °C} \tag{33}$$

With regard to the ethylene glycol process stream, in order to account for viscosity correction, equation 34 was employed with a $\boldsymbol{\mu_o}$ value at 49.8 °C of 7.102×10^{-3} Pa s **[3]**.

$$\left(\frac{\mu_{Eg}}{\mu_o}\right)^{0.14} = \left(\frac{3.725}{7.102}\right)^{0.14} = 0.914 \tag{34}$$

The viscosity correction factor of 0.914 calculated in equation 34 was utilised in order to obtain a corrected $\boldsymbol{U_o}$ value of 396.5 W m^{-2} K^{-1} as opposed to the previous calculated value of 416 W m^{-2} K^{-1} and a corrected $\boldsymbol{h_o}$ value of 656 W m^{-2} K^{-1}. The corrected $\boldsymbol{U_o}$ value is satisfactory since it's only approximately 13 % greater than the initial estimated value of 350 W m^{-2} K^{-1} (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 684) **[4]**.

5 Hydraulic calculations

5.1 Nozzle sizing

For entry and exit of process streams to both the shell and tube sides of the heat exchanger, nozzles of outer and inner diameters of 49.5 mm and 46.5 mm respectively were selected.

The nozzle flow area was calculated using equation 35, which makes use of the nozzle inner diameter.

$$\text{Nozzle flow area} = (46.5 \times 10^{-3})^2 \times \frac{\pi}{4} = 1.70 \times 10^{-3} \text{ m}^2 \tag{35}$$

5.2 Total tube side pressure drop

The nozzle velocity for the tube side of the heat exchanger $\boldsymbol{u_{Nt}}$ was calculated at 3.89 m s^{-1} using equation 36.

$$u_{Nt} = \frac{\dot{m}_{H2O}}{\rho_{H2O} \times \text{Nozzle flow area}} = \frac{6.59}{996 \times 1.70 \times 10^{-3}} = 3.89 \text{ m s}^{-1} \tag{36}$$

The Darcy friction factor $\boldsymbol{j_f}$ for the tube side of the heat exchanger was estimated at 4.1×10^{-3} (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 668). Equation 37 was successively employed in order to calculate the overall tube side pressure drop $\boldsymbol{\Delta P_t}$. For the nozzles on the tube side, a pressure drop equivalent to 1.5 nozzle velocity heads was estimated (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 667) **[4]**.

$$\Delta P_t = N_{pt} \times \left[8 j_f \times \left(\frac{L}{d_i}\right) + 2.5\right] \times \frac{\rho_{H2O} {u_t}^2}{2} + 1.5 \times \frac{\rho_{H2O} {u_{Nt}}^2}{2} \tag{37}$$

$$\Delta P_t = 4\left[8 \times 0.0041 \times \left(\frac{5.2}{0.016}\right) + 2.5\right] \times \frac{996 \times 0.916^2}{2} + 1.5 \times \frac{996 \times 3.89^2}{2} = 0.333 \text{ bar}$$

The $\boldsymbol{\Delta P_t}$ of 0.333 bar calculated is much lower than the maximum specified limit of 0.5 bar.

5.3 Total shell side pressure drop

The nozzle velocity for the shell side of the heat exchanger $\boldsymbol{u_{Ns}}$ was calculated at 2.27 m s^{-1} using equation 38.

$$u_{Ns} = \frac{\dot{m}_{Eg}}{\rho_{Eg} \times \text{Nozzle flow area}} = \frac{\frac{150}{36}}{1081 \times 1.70 \times 10^{-3}} = 2.27 \text{ m s}^{-1} \tag{38}$$

The Darcy friction factor $\boldsymbol{j_f}$ for the shell side of the heat exchanger was estimated at 6.95×10^{-2} (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 674). Equation 39 was consecutively employed in order to calculate the overall shell side pressure drop $\boldsymbol{\Delta P_s}$. For the nozzles on the shell side, a pressure drop equivalent to 2 nozzle velocity heads was estimated (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 675) **[4]**.

$$\Delta P_s = \left[8 j_f \times \left(\frac{D_s}{d_e}\right)\left(\frac{L}{L_B}\right)\left(\frac{\mu_{Eg}}{\mu_o}\right)^{-0.14}\right] \times \frac{\rho_{Eg} {u_s}^2}{2} + 2 \times \frac{\rho_{Eg} {u_{Ns}}^2}{2} \tag{38}$$

$$\Delta P_t = [8 \times 0.0695 \times 608.37] \times \frac{1081 \times 0.151^2}{2} + 2 \times \frac{1081 \times 2.27^2}{2} = 0.097 \text{ bar}$$

The $\boldsymbol{\Delta P_s}$ of 0.097 bar calculated is much lower than the maximum specified limit of 0.5 bar.

6 Mechanical design calculations

6.1 Overview

In conducting the mechanical design calculations for both the shell and tube parts of the heat exchanger, simple thin-walled pressure vessel theory was employed. The main stresses considered in the cylindrical pressure vessel approximation were the longitudinal stress $\boldsymbol{\sigma_l}$ and hoop stress σ_h. The radial stress component was neglected in the calculations performed because it's generally of the order of magnitude of the vessel's internal gauge pressure. When compared to σ_l and σ_h this tends to be negligible in terms of magnitude. Equation 39 is an expression for σ_h in cylindrical coordinate system geometry where $\boldsymbol{P}$ is the internal gauge pressure, $\boldsymbol{d}$ is the internal diameter of the vessel and $\boldsymbol{t}$ is its wall thickness. The relationship between σ_h and σ_l is $\sigma_h = 2\sigma_l$ for a cylindrical pressure vessel.

$$\sigma_h = \frac{Pd}{2t} \tag{39}$$

For the purposes of mechanical calculations, only σ_h was considered since it's larger than σ_l. Moreover, the yield strength of grade 304 stainless steel is approximately 480 MPa **[8]**. The design stress was constrained to approximately 80 % of the grade 304 stainless steel yield stress at 380 MPa.

6.2 Tube walls

Under normal operating conditions the differential pressure sustained by the tube walls is the pressure difference P_{diff} between the ethylene glycol and cooling water fluids, which is 2 bar. The cooling water gauge pressure $P_{g,t}$ of 8 bar is much larger than 2 bar. The minimum tube wall thickness required was calculated using equation 40.

$$\sigma_h = \frac{P_{diff}d_o}{2t_t} = 380 \times 10^6 \Rightarrow t_t = \frac{2\times 0.0216}{2\times 380\times 10} = 5.68 \times 10^{-6} \text{ m} \tag{40}$$

The actual tube wall thickness of 2.8×10^{-3} m is far in excess of 5.68×10^{-6} m.

6.3 Tube sheet

During optimal operating conditions, the differential pressure experienced by the tube sheet is P_{diff} between the ethylene glycol and cooling water fluids, which is 2 bar.

In subsequent calculations, bending and shearing forces were accounted for in order to determine the actual minimum allowable tube sheet thickness. The minimum allowable tube sheet thickness in bending $t_{ts,limB}$ and minimum allowable tube sheet thickness in shearing $t_{ts,limS}$ were calculated using equations 41 & 42 respectively.

$$t_{ts,limB} = \frac{D_s}{3} \times \left[\frac{P}{380\times10^6\left(1-0.907\times\left(\frac{d_o}{p_t}\right)^2\right)}\right]^{0.5} \Rightarrow t_{ts,limB} = 5.41 \times 10^{-3} \text{ m} \tag{41}$$

$$t_{ts,limS} = \frac{P}{380\times10^6} \times \frac{0.31D_s}{\left(1-\frac{d_o}{p_t}\right)} = \frac{2}{380\times10} \times \frac{0.31\times0.458}{\left(1-\frac{0.0216}{0.027}\right)} = 3.74 \times 10^{-4} \text{ m} \tag{42}$$

According to $t_{ts,limB}$ and $t_{ts,limS}$ calculated, the minimum tube sheet thickness should be at least 5.41×10^{-3} m . In addition, the tube sheet thickness should in reality be at least approximately 0.7 times the value of d_o (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 649) **[4]**. As a result, the actual tube sheet thickness should be 16.2 mm.

6.4 Shell wall

In the duration of normal operating conditions, the differential pressure sustained by the shell wall is the gauge pressure of the ethylene glycol $P_{g,s}$, which is 10 bar.

The minimum allowable shell wall thickness was calculated using equation 43 relating σ_h for the shell to t_s.

$$\sigma_h = \frac{P_{g,s}D_s}{2t_s} = 380 \times 10^6 \Longrightarrow t_s = \frac{10\times0.458}{2\times380\times10} = 6.03 \times 10^{-4} \text{ m} \quad (43)$$

According to the calculation performed, the shell wall thickness should be no less than 6.03×10-4 m. Nonetheless, the actual shell wall thickness should be 3.2 mm (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 647) **[4]**.

6.4 Shell ends

In the duration of normal operating conditions, the differential pressure sustained by the shell ends is the gauge pressure of the ethylene glycol $P_{g,s}$, which is 10 bar.

The oval shell end thickness t_{oval} and flat end shell thickness t_{flat} were calculated using equations 44 & 45 respectively (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 818-819) **[4]**.

$$t_{oval} = \frac{P_{g,s}D_s}{(2\times380\times10^6 - 0.2\times P_{g,s})} = \frac{10^6\times0.458}{(2\times380\times10^6 - 0.2\times10^6)} = 4.82 \times 10^{-4} \text{ m} \quad (44)$$

$$t_{flat} = D_s \times \sqrt{\frac{0.1\times P_{g,s}}{380\times10^6}} = 0.458 \times \sqrt{\frac{0.1\times10^6}{380\times10^6}} = 6.65 \times 10^{-3} \text{ m} \quad (45)$$

With regard to the calculations performed, as far as the shell ends are concerned, the oval shaped end should be 0.482 mm thick and the flat end should be 6.65 mm thick.

7 Gasketed-plate heat exchanger comparative analysis

7.1 Plate number and dimensions

The overall plate heat transfer coefficient U for such a heat exchanger configuration is expected to lie between 250-450 W m^{-2} K (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 638) **[4]**. The successful iteration resulted by use of the initial estimate of 450 W m^{-2} K^{-1}. Equation 46 was used in order to calculate the number of transfer units NTU for the plate heat exchanger. The value of ΔT_{lm} is still exactly the same value as that employed in the shell and tube heat exchanger thermal calculations.

$$NTU = \frac{t_{out} - t_{in}}{\Delta T_{lm}} = \frac{90-40}{32.7} = 1.53 \quad (46)$$

By using the configuration of a 1:1 pass gasketed plate heat exchanger, the value of F_t was estimated at 0.968 (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 760) **[4]**.

By specifying a plate length $\boldsymbol{L_{plate}}$ of 1.5 m, a plate width $\boldsymbol{w_{plate}}$ of 0.5 m and a plate thickness $\boldsymbol{t_{plate}}$ of 2mm, equations 47, 48 & 49 were used in order to calculate the number of plates required at approximately 51. Moreover, equation 50 was used so as to estimate the number of plates per pass. In equation 47, $\boldsymbol{A_{GP}}$ stands for the available heat transfer area for the gasketed plate heat exchanger.

$$A_{GP} = \frac{Q}{U \times F_t \times \Delta T_{lm}} = \frac{551529}{450 \times 0.968 \times 32.7} = 38.7 \text{ m}^2 \tag{47}$$

$$\text{Plate area} = L_{plate} \times w_{plate} = 1.5 \times 0.5 = 0.75 \text{ m}^2 \tag{48}$$

$$\text{Number of plates} = \frac{A_{GP}}{\text{Plate area}} = \frac{38.7}{0.75} = 51.5 \text{ (approximately 51)} \tag{49}$$

$$\text{Plate number per pass} = \frac{\text{Number of plates} - 1}{2} = \frac{51-1}{2} = 25 \tag{50}$$

7.2 Plate spacing and flow velocities

By assigning a plate spacing of 5.5×10^{-3} m, equation 51 was used in order to calculate the plate channel cross sectional area $\boldsymbol{A_{Ch}}$ and equation 52 was used in order to calculate the gasketed plate hydraulic mean diameter $\boldsymbol{d_{GP}}$. Furthermore, equations 53 & 54 were used so as to calculate the ethylene glycol channel velocity $\boldsymbol{u_{c,Eg}}$ and cooling water channel velocity $\boldsymbol{u_{c,H2O}}$ respectively.

$$A_{Ch} = w_{plate} \times \text{Plate spacing} = 0.5 \times 5.5 \times 10^{-3} = 2.75 \times 10^{-3} \text{ m}^2 \tag{51}$$

$$d_{GP} = \text{Plate spacing} \times 2 = 5.5 \times 10^{-3} \times 2 = 0.011 \text{ m} \tag{52}$$

$$u_{c,Eg} = \frac{\dot{m}_{Eg}}{\rho_{Eg} A_{Ch} \times \text{Plate number per pass}} = \frac{\frac{150}{36}}{1081 \times 0.00275 \times 25} = 0.0554 \text{ m s}^{-1} \tag{53}$$

$$u_{c,H2O} = \frac{\dot{m}_{H2O}}{\rho_{H2O} A_{Ch} \times \text{Plate number per pass}} = \frac{6.59}{996 \times 0.00275 \times 25} = 0.0951 \text{ m s}^{-1} \tag{54}$$

7.3 Thermal calculations

Equation 55 was employed in order to calculate $\boldsymbol{Nu}$ for both ethylene glycol and cooling water in the plate heat exchanger.

$$Nu = 0.26 \times Re^{0.65} \times Pr^{0.4} \tag{55}$$

For ethylene glycol, equation 56 was used in order to calculate its film transfer coefficient $\boldsymbol{h_{GP,Eg}}$ at 762.6 W m^{-2} K^{-1}.

$$h_{GP,Eg} = 0.26 \times \frac{k_{Eg}}{d_{GP}} \times \left(\frac{\rho_{Eg} \times u_{c,Eg} \times d_{GP}}{\mu_{Eg}}\right)^{0.65} \times \left(\frac{\mu_{Eg} \times C_{p,Eg}}{k_{Eg}}\right)^{0.4} \tag{56}$$

$$h_{GP,Eg} = 6.169 \times \left(\frac{1081 \times 0.0006094}{3.725 \times 10^{-3}}\right)^{0.65} \times \left(\frac{0.003725 \times 2647}{0.261}\right)^{0.4} = 762.6 \text{ W m}^{-2}\text{K}^{-1}$$

For cooling water, equation 57 was used in order to calculate its film transfer coefficient $\boldsymbol{h_{GP,H2O}}$ at 3036.5 W m^{-2} K^{-1}.

$$h_{GP,H2O} = 0.26 \times \frac{k_{H2O}}{d_{GP}} \times \left(\frac{\rho_{H2O} \times u_{c,H2O} \times d_{GP}}{\mu_{H2O}}\right)^{0.65} \times \left(\frac{\mu_{H2O} \times C_{p,H2O}}{k_{H2O}}\right)^{0.4} \tag{57}$$

$$h_{GP,H2O} = 14.56 \times \left(\frac{996 \times 0.0010461}{7.97 \times 10^{-4}}\right)^{0.65} \times \left(\frac{0.000797 \times 4186}{0.616}\right)^{0.4} = 3036.5 \text{ W m}^{-2}\text{K}^{-1}$$

The fouling factor used for ethylene glycol was 1×10^{-4} W^{-1} m^{2} K and that for cooling water was 1.3×10^{-4} W^{-1} m^{2} K (Coulson & Richardson Chemical Engineering Design, Vol.6, p. 757) **[4]**. Equation 58 was used in order to calculate ***U*** at 501.5 W m^{-2} K^{-1}. This value is satisfactory since it's only approximately 11 % greater than the initial estimate of 450 W m^{-2} K^{-1}.

$$U = \left(\frac{1}{h_{GP,H2O}} + \frac{1}{h_{GP,Eg}} + R_{fp,Eg} + R_{fp,H20} + \frac{t_{plate}}{k_s}\right)^{-1} \tag{58}$$

$$U = \left(\frac{1}{3036.5} + \frac{1}{762.6} + 1 \times 10^{-4} + 1.3 \times 10^{-4} + \frac{2 \times 10^{-3}}{16.2}\right)^{-1} = 501.5 \text{ W m}^{-2}\text{K}^{-1}$$

7.4 Pressure drop calculations

The port diameter was taken as 0.1 m and equation 59 was used in order to calculate the port area.

$$\text{Port area} = 0.1^2 \times \frac{\pi}{4} = 7.85 \times 10^{-3} \text{ m}^2 \tag{59}$$

Equation 60 was employed in order to calculate the pressure drops ***ΔP*** on both ethylene glycol and cooling water streams in the gasketed plate heat exchanger.

$$\Delta P = 4.8 \times Re^{-0.3} \times \frac{\rho {u_c}^2}{2} \times \frac{L_{plate}}{d_{GP}} + 1.3 \times \frac{\rho \times {u_p}^2}{2} \tag{60}$$

For ethylene glycol, ***ΔP*** was calculated at 0.0040 bar, which is much lower than the maximum specified limit of 0.5 bar.

$$\Delta P = 4.8(176.98)^{-0.3} \times \frac{3.318}{2} \times \frac{1.5}{0.011} + 702.65\left(\frac{4.17}{1081 \times 0.00785}\right)^2 = 0.0040 \text{ bar}$$

For the poor-quality cooling water, ***ΔP*** was calculated at 0.0080 bar, which is much lower than the maximum specified limit of 0.5 bar.

$$\Delta P = 4.8(1308)^{-0.3} \times \frac{9.008}{2} \times \frac{1.5}{0.011} + 647.4\left(\frac{6.59}{996 \times 0.00785}\right)^2 = 0.0080 \text{ bar}$$

7.5 Capital cost comparisons

The density of grade 304 stainless steel is approximately 8000 kg m^{-3} **[8]**. Moreover, the cost of grade 304 stainless steel is about 2.53 $ kg^{-1}**[9]**.

For the shell and tube heat exchanger design, the approximate mass of stainless steel required for the tube side was estimated at 991 kg using equation 61. In addition, its capital cost based on tube mass was estimated at $ 2500.

$$\text{Steel mass for tubes} = N_t \times \frac{\pi}{4}\left({d_o}^2 - {d_i}^2\right) \times L \times 8000 \tag{61}$$

$$\text{Steel mass for tubes} = \frac{144\pi}{4}(0.0216^2 - 0.016^2)5.2 \times 8000 = 991 \text{ kg}$$

$$\text{Cost requirement} = 991 \times 2.53 = \$\,2500$$

For the gasketed plate heat exchanger, the approximate mass of stainless steel required for the 51 plates was estimated at 612 kg using equation 62. In addition, its capital cost based on plate mass was estimated at $ 1548.

$$\text{Steel mass for plates} = \text{Number of plates} \times L_{plate} \times w_{plate} \times t_{plate} \times 8000 \tag{62}$$

$$\text{Steel mass for plates} = 51 \times 1.5 \times 0.5 \times 2 \times 10^{-3} \times 8000 = 612 \text{ kg}$$

$$\text{Cost requirement} = 612 \times 2.53 = \$\,1548$$

The mass of metal required for the plates is approximately 38 % less than that required for the tubes of the shell and tube heat exchanger. On this basis, the gasketed plate heat exchanger seems to be the more economical option. On the other hand, other factors such as equipment lifetime and operating costs should be considered prior to deciding on which piece of process equipment is indeed best suited for the specified duty.

8 Nomenclature

A	Heat transfer area	(m^2)
A_{Ch}	Channel cross-sectional area	(m^2)
A_{CF}	Area for cross-flow in between tubes	(m^2)
A_{GP}	Gasketed plate heat exchanger heat transfer area	(m^2)
A_{tube}	Total outer-tube area	(m^2)
$C_{p,Eg}$	Specific heat capacity of ethylene glycol	($J\ kg^{-1}\ K^{-1}$)
$C_{p,H2O}$	Specific heat capacity of (cooling) water	($J\ kg^{-1}\ K^{-1}$)
D_b	Tube bundle diameter	(m)
d_e	Hydraulic mean diameter for fluid flow in shell	(m)
d_{GP}	Hydraulic mean diameter for gasketed plate heat exchanger	(m)
d_i	Inner-tube diameter	(m)
d_o	Outer-tube diameter	(m)
D_s	Diameter of shell	(m)
F_t	Log-mean temperature difference correction factor	(-)
h_i	Film heat transfer coefficient on tube side	($kg\ s^{-3}\ K^{-1}$)
h_o	Film heat transfer coefficient on shell side	($kg\ s^{-3}\ K^{-1}$)
$h_{GP,Eg}$	Film heat transfer coefficient of ethylene glycol in gasketed plate heat exchanger	($kg\ s^{-3}\ K^{-1}$)
$h_{GP,H2O}$	Film heat transfer coefficient of (cooling) water in gasketed plate heat exchanger	($kg\ s^{-3}\ K^{-1}$)
j_f	Darcy friction factor	(-)

j_h	Heat transfer factor	(-)
K_1	Constant parameter value in equation with D_b, d_o, N_t (number of tubes) as variables	(-)
k_{Eg}	Thermal conductivity of ethylene glycol at the mean bulk temperature of the fluid of 65 °C	(kg m s^{-3} K^{-1})
k_s	Thermal conductivity of grade 304 stainless steel	(kg m s^{-3} K^{-1})
k_{H2O}	Thermal conductivity of (cooling) water at the mean bulk temperature of the fluid of 30 °C	(kg m s^{-3} K^{-1})
L	Tube length	(m)
L_B	Baffle spacing length scale	(m)
L_{plate}	Gasketed plate heat exchanger plate length	(m)
$\dot{m}_{Eg}$	Mass flowrate of ethylene glycol	(kg s^{-1})
$\dot{m}_{H2O}$	Mass flowrate of (cooling) water	(kg s^{-1})
n_1	Constant parameter value in equation with D_b, d_o, N_t (number of tubes) as variables	(-)
N_{pt}	Number of tube passes	(-)
N_t	Total number of tubes	(-)
NTU	Number of transfer units	(-)
Nu	Nusselt number	(-)
ΔP	Pressure drops in gasketed plate heat exchanger for both fluid streams	(Pa)
ΔP_s	Shell-side overall pressure drop	(Pa)
ΔP_t	Tube-side overall pressure drop	(Pa)
Pr	Prandtl number	(-)
P	Pressure	(Pa)
P_{diff}	Pressure differential experienced by tube walls	(Pa)
$P_{g,s}$	Shell-side gauge pressure	(Pa)
$P_{g,t}$	Tube-side gauge pressure	(Pa)
p_t	Tube pitch length scale	(m)
Q	Heat exchanger duty	(W)
q_o	Outer-tube surface heat flux	(W m^{-2})
R	Temperature ratio used in estimating F_t	(-)
Re	Reynolds number	(-)
$R_{f,i}$	Fouling factor inside the tubes	(kg^{-1} s^3 K)
$R_{f,o}$	Fouling factor outside the tubes	(kg^{-1} s^3 K)
R_{wall}	Tube-wall thermal resistance	(kg^{-1} s^3 K)
$R_{fp,Eg}$	Fouling factor of ethylene glycol in gasketed plate heat exchanger	(kg^{-1} s^3 K)
$R_{fp,H2O}$	Fouling factor of cooling water in gasketed plate heat exchanger	(kg^{-1} s^3 K)
S	Temperature ratio used in estimating F_t	(-)
ΔT_{lm}	Log-mean temperature difference	(°C)
ΔT_m	Corrected log-mean temperature difference	(°C)
t_1	Inlet temperature of cold process stream of (cooling) water	(°C)
T_1	Inlet temperature of hot process stream of ethylene glycol	(°C)
t_2	Outlet temperature of cold process stream of (cooling) water	(°C)
T_2	Outlet temperature of hot process stream of ethylene glycol	(°C)
t_{flat}	Flat shell end thickness requirement	(m)
t_{in}	Gasketed plate heat exchanger inlet stream temperature	(°C)
t_{oval}	Oval shaped shell end thickness requirement	(m)
t_{out}	Gasketed plate heat exchanger outlet stream temperature	(°C)
t_{plate}	Gasketed plate heat exchanger plate thickness	(m)
$t_{ts,limB}$	Minimum tube sheet thickness for bending	(m)
$t_{ts,limS}$	Minimum tube sheet thickness for shearing	(m)
t_s	Shell-wall thickness	(m)

t_t	Tube-wall thickness	(m)
$T_{w,i}$	Inner-tube surface temperature	(°C)
$T_{w,o}$	Outer-tube surface temperature	(°C)
U	Overall heat transfer coefficient for gasketed plate heat exchanger	(kg s^{-3} K^{-1})
U_o	Overall heat transfer coefficient with respect to outer-tube diameter (area)	(kg s^{-3} K^{-1})
$u_{c,Eg}$	Channel velocity of ethylene glycol in gasketed plate-heat exchanger	(m s^{-1})
$u_{c,H2O}$	Channel velocity of (cooling) water in gasketed-plate heat exchanger	(m s^{-1})
u_{Ns}	Nozzle velocity for shell side of heat exchanger	(m s^{-1})
u_{Nt}	Nozzle velocity for tube side of heat exchanger	(m s^{-1})
u_p	Port velocity for gasketed-plate heat exchanger	(m s^{-1})
u_s	Shell-fluid velocity corresponding to ethylene glycol	(m s^{-1})
u_t	Tube-fluid velocity corresponding to (cooling) water	(m s^{-1})
w_{plate}	Gasketed plate heat exchanger plate width	(m)
μ_o	Ethylene glycol viscosity at wall temperature	(Pa s)
μ_{Eg}	Ethylene glycol viscosity at mean bulk temperature of the fluid of 65 °C	(Pa s)
μ_{H2O}	Cooling water viscosity at mean bulk temperature of the fluid of 30 °C	(Pa s)
ρ_{Eg}	Ethylene glycol density at mean bulk temperature of the fluid of 65 °C	(kg m^{-3})
ρ_{H2O}	Cooling water density at mean bulk temperature of the fluid of 30 °C	(kg m^{-3})
σ_h	Pressure vessel hoop stress	(Pa)
σ_l	Pressure vessel longitudinal stress	(Pa)

9 References

[1] National Centre for Biotechnology Information. PubChem Database. 1,2-Ethanediol, CID=174, https://pubchem.ncbi.nlm.nih.gov/compound/174 (accessed on April 27, 2019).

[2] INEOS Oxide, Ethylene Glycol, In: 'Technical Data Sheet', IOX TDEG 2004.01, https://www.ineos.com (accessed on April 27, 2019).

[3] Perry, Robert H, and Don W. Green, Perry's Chemical Engineers' Handbook, New York: McGraw-Hill, 2008, 7th edition, chapter 3 p.325-326, chapter 2 p.91-92, 306.

[4] Sinnott, R. K., Coulson, J. M., & Richardson, J. F. (2005), Coulson & Richardson's chemical engineering, Vol. 6. Oxford, Elsevier Butterworth-Heinemann, Vol. 6 p. 638-641, 646-654, 656-659, 665-675, 684, 756-764, 818-819.

[5] Material type, cost, density, Young's modulus. MIT, http://web.mit.edu/course/3/3.11 (accessed on April 27, 2019).

[6] Puyear, R.B.; Materials Selection Criteria for Shell and Tube Heat Exchangers for Use in the Process industries, Shell and Tube Heat Exchanger, Edited by William R. Apblett, 1981.

[7] Standards of the Tubular Exchangers Manufacturers Association (TEMA), 9th edition, section 6 p.15-18, section 4 p.4-7.

[8] ASM Aerospace Specification Metals Inc., In: 'AISI Type 304 Stainless Steel' Ferrous Metal, Heat Resisting; Metal; Stainless Steel; T 300 Series Stainless Steel, 800 398-4345, http://asm.matweb.com (accessed on April 29, 2019).

[9] MEPS international Ltd., November 2018 U.S stainless steel pricing, http://meps.co.uk./N.Amer%20Price.htm, (accessed on April 29, 2019).

10 Heat Exchanger Data Sheet

Client CEB	**Area** 1	**Item no.** 1

Service Ethylene glycol cooldown from 90 °C to 40 °C		
No of units 1	**Type** AES	**Orientation** Horizontal

	Units	Shell Fluid (Ethylene glycol)		Tube Fluid (Cooling water)	
		In	Out	In	Out
Total fluid flow	**kg s^{-1}**	4.17	4.17	6.59	6.59
Liquid	**kg s^{-1}**	4.17	4.17	6.59	6.59
Vapour	**kg s^{-1}**	0	0	0	0
Non-condensable	**kg s^{-1}**	0	0	0	0
Steam	**kg s^{-1}**	0	0	0	0
Water	**kg s^{-1}**	0	0	6.59	6.59
Temperature	**°C**	90	40	20	40
Mol. Wt of vapour + NC		NA	NA	NA	NA
Liquid density	**kg m^{-3}**	1063	1099	999	993
Liquid viscosity	**Pa s**	0.00244	0.0091	0.001001	0.000653
Vapour + NC viscosity	**Pa s**	NA	NA	NA	NA
Liquid sp. heat	**kJ kg^{-1} K^{-1}**	2.75	2.53	4.18	4.18
Vapour + NC sp. heat	**kJ kg^{-1} K^{-1}**	NA	NA	NA	NA
Liquid thermal conductivity	**W m^{-1} K^{-1}**	0.263	0.259	0.599	0.631
Vap. + NC thermal conduct.	**W m^{-1} K^{-1}**	NA	NA	NA	NA
Latent heat	**kJ kg^{-1}**	999	1005	2260	2260
Surface tension	**N m^{-1}**	0.0425	0.0472	0.0723	0.0697
Dew pt/bubble pt	**°C**	294	294	170.4	170.4
Freeze pt/pour pt	**°C**	-13	-13	0	0
Inlet pressure	**bar**	10		8	
Allowable press. drop	**bar**	0.5		0.5	
Fouling resistance	**m^2 K W^{-1}**	0.0002		0.00025	
Heat exchanged	**MW**	0.55			
Min./max. operating temp	**°C**	35	95	15	45
Max. operating pressure	**bar**	11		9	
Min. design pressure	**bar**	10		8	
Relief valve set pressure	**bar**	11		9	
Design pressure	**bar**	12		10	
Cyclic service		☒		☒	
Materials		Grade 304 stainless steel		Grade 304 stainless steel	
Corrosion allowance	**mm**	0.95		0.95	
Line size	**mm**	46.5	46.5	46.5	46.5
Heat treatment		☒		☒	
Hazards		Hot upon physical contact and highly pressurised		NA	
Insulation		NA		NA	
Cleaning		Once every 2 months		Once every 2 months	
Over design multiplier		1.14		2.23	

Remarks				
Rev no				
Eng/date				
Checked/date				
Approved/date				