Hydraulic, adiabatic wheel heat exchanger

Design

From the data given and specific heat capacity researched for ethylene glycol, a Mutual script was constructed to calculate values from a range f formulae for a variety of variables including the initial duty, which was calculated to be 1486. 1 K. It was chosen, that as water is a more corrosive fluid and has a higher operating pressure, the water should pass through tubes made of stainless steel, which are easier to clean, and the ethylene glycol through the shell side. From here the Log Mean Temperature Difference (LIMIT Heat Transfer Coefficient (U = 714) and Total Interracial Heat Transfer Area (A = 40. Mom) and a Reynolds number were all calculated using Mutual. A number of 336 tubes for the exchanger were established and it was concluded that a 1-4 Shell Tube design was most appropriate with tubes of inner diameter 15. Mom and an outer diameter of 19. Mom with a triangular pitch of 23. 81 mm giving a pipe thickness of mm. Using Kernel's method (Collusion & Richardson, 2005). Chemical Engineering Design, a beef cut of 25% was required leading to a tube side pressure drop of 0. 45 bar and a shell side pressure drop of 0. 4 bar. The report investigates further design specifications and other important data as well as calculations of how these specifications were achieved. Introduction The objective of this problem was to design a heat exchanger for the transfer of heat between a relatively warmer Ethylene Glycol fluid stream and a cooler water stream using a shell and tube heat exchanger. Shell and Tube heat exchangers are the most common heat exchangers found in processing and are popular because of a variety of reasons.

A large surface interface ratio to volume and mass of allowable heat exchange within the structure is a very important design consideration for efficient and cost effective heat exchange between the fluids. In terms of construction and trials, there is generally quite a bit of flexibility when it comes to selection. Many different materials can be used to prevent corrosion and dismantling the exchanger for maintenance (such as replacement of tubes) and or cleaning is relatively straightforward.

Due to the fouling of fluids in this problem, high resistance to corrosion was a significant requirement for an efficient design. The design of shell and tube exchangers makes use of turbulence from Increased fluid velocity so heat exchange between the fluids Is maximized by the use of baffles, which also support he structure. Without baffles, fluids may stagnate In particular parts of the shell also considered as a design possibility because of their large surface area of heat exchange possibility between the fluids produced by the large interracial area of the plates, which make up the structure.

This in turn maximizes effectiveness of these unit's as well as possibly allowing for a physical size reduction of the heat exchanger itself. Shell and tube exchangers may not necessarily offer these physically small structures, yet, on the other hand compact designs of plate heat exchangers can be a read expense due to their intricate and elaborate structures with a possibility of being quite fragile. Materials of high expense such as titanium plates are often used only contributing to this problem and not allowing for a great deal of material flexibility like shell and tube structures.

The temperature changes of both fluids are fairly low so thermal and expansionary stresses are not as significant which made the design choice a little easier as both temperature differences were within the allowable range of a shell and tube exchanger's capabilities. That is by less than bout 100 F for most widely used materials (Bell & Mueller, 2001). Obviously this value would change with a change in material use. Shell and Tube exchangers have a variety of problems when it comes to fouling fluids within and their ability to be effectively cleaned.

Plate exchangers on the other hand can be deconstructed a little easier when it comes to cleaning or replacement of damaged plates making maintenance simple and preservation more long term. However interior corrosion is still a major problem, which in this case of ethylene glycol and water both fouling, a late heat exchanger would have not been a particularly appropriate choice of exchanger. The ability of plate heat exchanges to cause quick temperature changes in fluids would have likely been a negative occurrence.

This is because of the fluid properties of the fluids in the system as the water's inability to exceed 40 degrees Celsius suggested a large amount of instability and possible reactivity within the fluid itself and with the interior surface of the tubes and connections. Undesirable pressure drops are also usually higher than Shell and Tube exchangers. Phase hanged heat exchangers were immediately not considered in the design, as the poor quality of water entering the exchanger could not exceed the maximum specified temperate of 40 degrees Celsius, thus not undergoing a phase change (South West Thermal, 2010).

Adiabatic Wheel heat exchangers make use of storing heat on one side of the exchanger and then transferring it to the other by means of wheel rotations through the various temperature fluids (Luaus Inc, 2012). These exchangers are commonly used over shell and tube exchangers in industry such as food processing as well as in situations where mixing of both streams is tolerable. However, as mixing of fluid's in this circumstance is not tolerated; an adiabatic wheel exchanger could not be considered as a design possibility over a shell and tube exchanger.

A shell and tube heat exchanger design was chosen for this problem as its ability to be maintained fairly easily as well as having good properties for heat exchange Fluid Properties Fluid properties are highly dependent on temperature, so we will categories each fluid by its mean temperature, that is, the average temperature between the inlet and outlet. Water at 28. ICC Density _w) 995 keg/mm Viscosity _w) 0. 8 x 10-3 Pa Thermal Conductivity _w) 0. 59 W/ms Ethylene Glycol at ICC Density _s) 1071. 5 keg/mm Specific Heat Capacity _as) 2. 675 k/keg Viscosity _s)3. Xx 10-3 pa Thermal Conductivity _s) 0. 262 W/ms Heat Exchanger Design Our design procedure is an iterative one. By utilizing a simple MUTUAL script to initialize values and substitute them into applicable formulas we are able to describe the process as a function of a number of design choices. It is difficult to express in words the steps undertaken in this design, as variables were tweaked and calculated on-the-fly until the design satisfied optimal requirements. The initial guesses below represent both chosen parameters and calculations one iteration before final design.

To begin we have to determine the heat exchanger's duty and a corrected log-mean temperature difference (LIMIT). Before we begin we must assign each fluid to the shell or tube side of the exchanger. A number of factors for determining side assignment have been considered (Collusion & Richardson, 2005). First and foremost, water is a more corrosive fluid and should be placed in the tube side. This gives us better control over the tube fluid velocity, and higher allowable velocities can reduce fouling. Moreover, the tubes are easier to clean.

It is also suggested the stream with a higher operating pressure (in our case, water, at brag) should be placed in the tube side. We will assign ethylene glycol to the shell side. As specified in our assignment, the maximum allowable temperature for water is limited to a maximum of ICC. For this reason we have chosen to design our exchanger to provide a water outlet temperature of ICC. This consideration can be shown in our design program to have a great effect on final design. It influences fluid properties that are functions of temperature, as well as the LIMIT.

This will require a larger throughput of water necessary to reach the required exchange of heat. By performing an energy balance a standard shell and tube heat exchanger we can calculate the required water mass flow rate and the overall duty required. Using the relationship Q= m' AT, for the ethylene glycol portion, we obtain Q= 40000/3600 k. This duty must be the same for water. Hence we can solve the water mass flow rate as 27. 35 [kvass] which equivalent to 98, 478 [keg/her]. Log- Mean Temperature Difference The governing equation for heat exchanger design is expressed as Q= THAT_ELM

Where U is the overall heat transfer coefficient for the process [W/make], A is the total area across which all heat transfer occurs, and a log-mean temperature difference between the two streams. It is communicated mathematically assuming a linear temperature gradient in both stream is counter-current flow, as, AT_ELM= ((T_1-t_2 (T_2-t_1))/(Len((T_1-t_2))/((T_2-t_1))) Where TTL and TO represent the hotter fluid (ethylene glycol) at the inlet and outlet respectively; TTL and to are inlet and outlet 60, TTL = 22 and to = 35 degrees Celsius.

When dealing with multi-pass exchangers, as n our 1-4 design, a correction factor must be applied to our calculated value. This is achieved be finding two temperature ratios R and S. R=(T_1-T_2)/(t_2-t_1) and S=(t_2- Thusly we calculate R = 3. 85 and S = 0. 15. By reading from the graph to the left for 2-4 shell-and-tube exchangers, we can see our correction factor is simply one (1). Substituting our values we calculate our LIMIT to be 54. ICC. Design Process At this stage we must assume a U value and calculate the required area. Many tables guide us to expected values between 250 and 750 [W/make].

Let us start be assuming a value of 650 [W/make]. Provisionally our area A, urn, NO We must now decide on appropriate tube lengths, outer diameter and inner diameter. After testing a number of parameters in our program, the best results we obtained were a pipe length of mm, a standard % inch or 19. Mom ODD at a thickness of mm, hence a tube ID of 15. Mom. It is suggested that a triangular pitch 1. 25 times that of our ODD, or 23. 81 mm, will be optimal for design. Triangular pitches mean we can fit more tubes across the same area and can also lessen the pressure drop across the exchanger.

Our shell-side fluid is relatively non-fouling; cleaning our arranging pitch will not pose a problem. Using a few simple geometric relationships, we can calculate the number of tubes required by taking the area of one tube, dividing into our A to get the even number of tubes. (1119. 05. (10) Bundle diameter Db can be found through a correlative method, such that D_b= D_o ((N/K_I ((352/0. 175)) mm Where SKI and in can be found in Table 12. 4 (Collusion & Richardson, 2005) and are specific to the number of shell and tube passes.

Selecting a pull-through floating head (TEAM T), our total shell diameter DSL becomes 531 + 90 = 621. Mm courtesy of Figure 12. 0 (Collusion & Richardson, 2005). Tube-side Coefficient Taking all fluid properties at mean temperatures and accounting for 4 tube passes we can determine the velocity of water. The total flow area is given sagas/u' 4 (15. 05) AH (10) 0. Mamma, together with mass flow rate and density we can obtain This value checks with appropriate values for water in a tube-side? the velocity is high enough to circumvent fouling but still maintain an allowable pressure drop.

The heat transfer coefficient when dealing with water has been correlated quite precisely. Its formula: h_ii = (v_w) AH. 8/(Dido. 2 K) Shell-side Coefficient An important design consideration for all shell and tube heat exchangers is the spacing between the baffles and hence the number of baffles needed. We will space ours appropriately such that the spacing is equal to a quarter of the shell diameter, hence l_b, our spacing becomes 155. Mm. By taking the ratio of the clearance between the tubes and the total distance between the tube centers or (1. 25-1)/ 1. 5= 0. 2 we can obtain the relevant area for cross flow on the shell-side, A s, as O. AD_s (10)

For a triangular pitch arrangement, the hydraulic diameter required for Reynolds umber calculations for the shell side d_e can be realized through geometric calculations. It can be shown, where p_t refers to the tube pitch, that d_e can be found as Again using mean temperatures as the basis of our fluid properties, we can now calculate the Prenatal and Reynolds numbers. It is through these dimensionless parameters that heat transfer coefficients can be estimated.