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Design and Construction of a Shell and Tube Heat Exchanger

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ABSTRACT

Heat exchanger is necessary laboratory equipment in a thermo-fluid laboratory, and lack of this equipment makes learning ineffective. Hence, this study aims to design and construct a shell and tube heat exchanger for laboratory use. To achieve this aim, mechanical and thermal factors were considered. The thermal design was done using the Bell Delaware method. The heat exchanger was designed based on the assumption that there is no phase change, while water at a cold inlet temperature of 15 °C enters the heat exchanger through the tube and hot water at 100 °C enters the heat exchanger through the shell. Results show that the geometry of the heat exchanger favours turbulent flow which enhances heat transfer. This causes a heat load of 107.973 kW to be transferred from the hot fluid to the cold fluid through the tube wall when hot fluid of 0.5 kg/sec flows at a velocity of 0.3 m/s and cold fluid of 2.58 kg/sec flows with a velocity of 1 m/sec. This heat transfer caused the cold fluid temperature to increase by 10 °C as it exits the tube, while the temperature of the hot fluid fell to 45 °C as it exits the shell. The pressure drops in both the shell and tubes were within the allowable range, and hence, accepted. With the overall heat coefficient at 134.23 W/m^{2}K and the efficiency of the system at 73.3%, the study is said to have achieved its objective.

Introduction

A heat exchanger is a thermal device that enables the exchange of heat between a solid and a fluid or between two or more different fluids. Heat exchangers are used in car radiators, air conditioners, refrigerators, chemical industries for processes involving distillation, fractionation, etc. Heat exchangers are also used in pulp and paper, power generation, pharmaceuticals and food and beverages processing industries. Generally, heat exchangers are needed wherever heat is generated in the system needs to be removed. Heat exchangers can be classified based on the direction of flow viz: parallel flow, counter flow, and cross flow. In parallel flow, the fluids enter the heat exchanger at the same end, and travel in parallel direction to each other to the other side. In counter flow, the fluids enter the heat exchanger from opposite ends, while in the cross flow, the fluids travels roughly perpendicular to each other.

However, there are eight main types of heat exchangers these are: shell and tube heat exchanger, plate heat exchanger, plate and shell heat exchanger, adiabatic wheel heat exchanger, plate fin heat exchanger, pillow plate heat exchanger, fluid heat exchanger, waste heat recovery unit heat exchanger. Among these types of heat exchangers, the shell and tube heat exchanger is the most versatile.

The shell and tube heat exchanger consists of tubes which could also be cylindrical or rectangular; mounted in a shell. The shells could also be cylindrical, rectangular or arbitrary in shape. Shell and tube heat exchangers are mainly used in process industries conventional and nuclear power stations, steam generators etc. its relative large ratios of heat transfer area to volume and ease of cleaning gives it an edge over the other types of heat exchangers. Due to its wide range of application, thermal engineering programmes include its study in courses such as heat and mass transfers. In order to carry out the necessary experiments involved in its study, most universities in Nigeria prefer to import this equipment. This action neither favours the growth of the economy nor local content inclusion.

Hence the aim of this study is to design and construct a shell and tube exchanger for laboratory demonstrations.

Research Elaboration

A shell and tube heat exchanger is made up of a tube bundle mounted in a shell. The tube bundle contains fluid that could be either heated or cooled, while the fluid flowing over the tube bundles contain fluid that could either be cooled or heated. Usually, two fluids of different starting temperature flow through the heat exchanger; these fluids could either be liquid or gas. Heat is transferred from one fluid to the other through the tube walls. Besides the shell and tube bundle, other parts that make up the heat exchanger include: tube sheet, baffles, tie rods and spacers, and headers. The tube sheets are plates or forgings drilled to provide holes through which tubes are inserted and it is located at both ends of the shell. It also ensures that the fluids in the shell and tube do not mix. The baffles main functions are: to provide support for structural rigidity, prevent tube vibration and sagging; divert the flow across bundle to obtain a higher heat transfer coefficient; maintain tube spacing. According to [1] baffles could be segmental-cut type, disc and ring type or orifice type. The segmental-cut type are efficient in diverting flow across the tubes. The disc and ring baffles consist of alternating outer rings and inner discs, which direct flow radially across the tube field. It is effective in pressure drop to heat transfer conversion. The orifice baffle shell side fluid flows through the clearance between outside diameter and baffle-hole diameter.

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Tie rod and spacers hold the baffles and ensures that the baffle spacing is unchanged. The headers contain the fluid introduced to the tubes and provide a flow path for the fluid exit. It is installed at both sides of the heat exchanger.

Design Method

In designing the shell and tube heat exchanger, the following factors were considered.

- Tube geometrics which include: tube outside diameter, tube thickness, length, pitch, corrugation, passes, and layout.
- Material selection for the various parts viz: shell, tubes, baffles, tie rod and spacers, headers and tube shell.
- Selection criteria for fluid allocation, shell type, front and rear end header.
- Mechanical design factors such as diameter of shell, pressure of shell, pressure of fluid, shell thickness, thickness of tube plates, thickness of headers, and total number of tubes.
- Thermal design factors such as tube side heat transfer coefficient, heat balance, shell side heat transfer coefficient and overall heat transfer coefficient.

Tube geometrics

Tube outside diameter: An arbitrary value of 1” (25.4 mm) was chosen with recourse to ease of mechanical cleaning and fouling.

Tube thickness: Major points of consideration are provision of a greater area to volume density and the hoop stress developed as a result of the fluid passing through it.

Tube length: In determining the tube length, a compromise was made between cost and ease of anti-fouling mechanical cleaning while considering the installation space. Hence, a length of 12.5 cm was opted for.

Tube pitch: The tube pitch is the centre to centre distance between adjoining tubes. Care is taken to ensure that the tubes in the bundle are not closely packed as this would cause leaky joints in the tube sheet. According to [1], the range for the pitch ratio is $1.25 < P_T/d_o < 1.5$.

Tube passes: The tube layout characterized the angle between the pitches of the tubes. A 60° layout was chosen as it favours greater heat transfer [2].

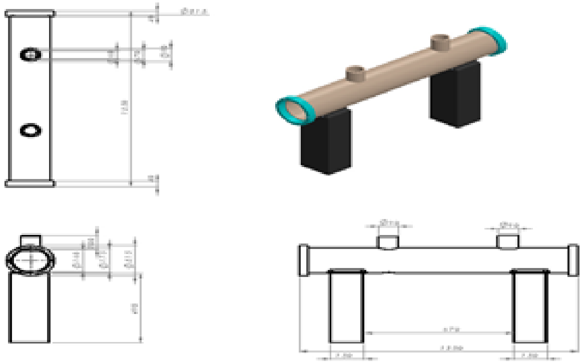

Figure 1. Engineering drawing of heat exchanger shell.

Material selection

Shell: The shell of the heat exchanger is made up of AISI type 304 stainless steel. This is to overcome the challenge of corrosion associated with mild steel, and also because of the good strength quality it possesses.

Tubes: The tubes are made up of Aluminum in order to enhance effective heat transfer due to its high thermal conductivity and also because they are corrosion resistant.

Baffles: The baffles are made of stainless steel and because of its strength, durability and corrosion resistance.

Tube sheets: The tube sheets are made up of Aluminum in order to enhance effective heat transfer due to its high thermal conductivity and also because they are corrosion resistant.

Headers: The headers contain the tube side fluid and are made up of AISI type 304 stainless steel for reasons of strength and corrosion resistance.

Selection criteria

Fluid allocation: Viscous fluids are allocated to the shell side in order to increase the rate of heat transfer, while fluids prone to fouling are passed through the tubes because tubes favour anti-fouling mechanical cleaning more than the shell.

Shell type: A one pass shell type is used in the design. This was chosen due to the experiment demonstration purposes the equipment is designed for.

Front and Rear end headers: considering the intended application of heat exchanger, a bonnet type header is used for the design since the heat exchanger will not require frequent cleaning.

Figure 2. Engineering drawing of heat exchanger tubes with header and baffles.

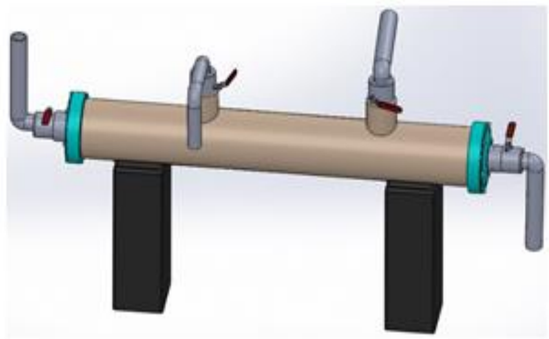

Figure 3. 3D model of Shell and Tube Heat Exchanger assembly.

Mechanical design

Shell diameter, (D_s): The shell diameter is evaluated considering the tube bundle equivalent diameter the formula expressed in equation one was used.

$$D_S = D_b + \text{Shell bundle clearance} \quad (1)$$

Where D_b = bundle equivalent diameter.

$$D_b = d_o\left(\frac{N_t}{K_1}\right)^{1/n_1} \quad (2)$$

$n_1 = 2.142$, $K_1 = 0.319$ [2]

Shell thickness, (t_s): The shell thickness of the heat exchanger is evaluated with equation 3.

$$t_s = \frac{P_s D_s}{2\sigma\eta} \quad (3)$$

Where P_s is the pressure within the shell, σ is the yield strength of the shell material (AISI type 304) = 215 Mpa [3], η is the joint efficiency of the shell = 1. This equation gives the minimum thickness required to withstand the operating fluid pressure within the shell.

Tube thickness, (t_t): The expression for the tube thickness is based on the assumption that the force it experiences is uniformly distributed. The tube thickness is evaluated using the formula given below.

$$t_t = K_1 d_0 \sqrt{\frac{P_t}{\sigma_t}} \quad [4] \qquad (4)$$

Where: K_1= A constant which depends upon the material of the plate and the method of holding the edges. For mild steel with fixed edges, $K_1 = 0.35$.

Pt = pressure inside the tube as a result of the operating fluid.

σt = Allowable design stress of aluminum = 79 Mpa [5]

Thickness of dished head, (t_d): Equation 5 gives an expression for determining thickness of the header. This expression gives the permissible minimum thickness the headers should be.

$$t_{dh} = \frac{4.8 P_s r}{\sigma_u} \qquad (5)$$

Where: P_s is the pressure inside the shell,

r is the inside radius of the plate curvature = 50 mm,

σ_u is the ultimate strength for the AISI type 304 stainless steel material of the plate = 505 Mpa [3]

Tube pitch (t_p): The tube pitch is expressed using equation 6

$$t_p = 1.25 d_o \qquad (6)$$

Number of tubes, (N_t): The expression put forward by [1] is used to determine the number of tubes.

$$N_t = \frac{CTP \pi D_s^2}{4 A_1} \qquad (7)$$

According to [1] CTP is the tube count pass which accounts for the incomplete coverage of the shell diameter by the tubes due to necessary clearances between the shell and the outer tube circle. CTP = 0.49 for a six tubes pass

$$A_1 = (CL) t_p^2 \qquad (8)$$

Where CL is the tube layout constant = 0.87 for a 60^o tube layout.

Tube outside surface area, (A_o): it is determined using the equation expressed below.

$$A_o = \pi d_o N_t L \qquad (9)$$

Where d_o is the outside tube diameter, N_t is the number of tubes, L is the length of tubes.

Baffle spacing: The expression proposed by [2] is used to determine the baffle spacing.

$$B = 74 d_o^{0.75} \qquad (10)$$

Thermal design

Bell Delaware method was used to carry out the thermal design. Factors such as the tube transfer coefficient, log mean temperature difference, true temperature, shell side heat transfer coefficient, and the overall heat transfer coefficient were considered.

Overall heat transfer coefficient: This is a function of all the fouling resistances, individual heat coefficient and surface efficiency of the tubes. The fouling resistances are as a result of the build-up of dirt film on the heat exchanger surfaces. The tube overall heat transfer coefficient is determined as follows.

$$\frac{1}{U_o} = \frac{1}{\frac{1}{h_T} + \frac{1}{h_s} + \frac{L}{K} + R_i + R_o} \qquad (11)$$

Where R_i, R_o, are the inside, and outside fouling resistances factors.

$R_i = 0.0001$, $R_o = 0.0002$, [6]

K = Thermal conductivity of tube wall material.

L = Length of heat exchanger.

Heat Load: Assuming no phase change exists, the heat duty of the heat exchanger is expressed as:

$$q = (\dot{m} C_p)_c (T_{t,o} - T_{t,i}) = (\dot{m} C_p)_h (T_{s,i} - T_{s,o}) \qquad (12)$$

Where $\dot{m}_c = 0.7855 \rho v d_i^2 N_t$ (13)

Cp = 4.185 kJ/kgK,

$\mathbf{T_{t,o}} =$ **outlet tube side fluid temperature = 25°C**,

$\mathbf{T_{t,i}} =$ **inlet tube side fluid temperature = 15°C**,

$\mathbf{T_{s,i}} =$ **inlet shell side fluid temperature = 100°C**,

$\mathbf{T_{s,o}} =$ **outlet shell side fluid temperature = 45°C**

Tube side Reynolds number: The tube side Reynold number classifies the flow regime of the fluid in the tube. It is quantified using the equation 14.

$$Re_t = \frac{\rho_t v_t d_i}{\mu_t} \qquad (14)$$

Tube side heat transfer coefficient (h_t): This is expressed by equation 15.

$$h_t = \frac{Nu \times K}{L} \qquad (15a)$$

Where Nu = Nusselts number

$$Nu_t = 0.023 \times Re^{0.8} \times Pr^{0.33} \times \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad (15b)$$

K = thermal conductivity of water = 0.606 W/mK [7],

μ = dynamic viscosity of water at 15 °C = 1.1373×10^{-3} Pa.s [8],

μ_w = dynamic viscosity of water at 25 °C = 8.89×10^{-4} Pa.s,

$$Pr = Prandtl\ number = \frac{\mu \times C_p}{K}$$

Shell side heat transfer coefficient (h_s): The heat transfer coefficient of the shell side is expressed using the equation put forward by [2].

$$h_s = \frac{Nu \times K}{L} \qquad (16a)$$

$$Nu_s = 0.023 \times Re^{0.8} \times Pr^{0.33} \times \left(\frac{\mu}{\mu_w}\right)^{0.14} \times K \qquad (16b)$$

Where μ = dynamic viscosity of water at bulk fluid temperature of 100°C = 2.814×10^{-4} Pa.s [8],

μ_w = dynamic viscosity of water at shell temperature of 90°C = 3.15×10^{-4} Pa.s [8]

Shell side fluid mass flow rate: the mass flow rate of the hot fluid can be expressed using equation 17.

$$\dot{m}_s = \frac{q}{C_p (T_{s,i} - T_{s,o})} \qquad (17)$$

Shell side cross sectional flow area: The shell side cross sectional area is expressed as:

$$A_s = 0.7855 (D_s^2 - d_0^2 N_t) \qquad (18)$$

Shell side linear velocity: Equation 18 gives an expression for the velocity at which the hot fluid flows.

$$v_s = \frac{\dot{m}_h}{A_s \rho_s} \qquad (19)$$

Shell side Reynolds number: The shell side Reynold number classifies the flow regime of the fluid in the shell. It is quantified using the equation 20.

$$Re_s = \frac{\rho_s v_s D_s}{\mu_s} \qquad (20)$$

Tube side pressure drop: The pressure drop in the tube side is evaluated using the expression given by [2].

$$\Delta P_t = N_p \left[8 j_f \left(\frac{L}{d_i}\right) \left(\frac{\mu}{\mu_w}\right)^{-m} + 2.5 \right] \qquad (21)$$

Where N_p = number of passes,

j_f = friction factor = 0.0026,

L = length of tube,

μ = dynamic viscosity of bulk fluid 15°C,

μ_w = dynamic viscosity of the fluid at a wall temperature of 25°C = 0.0011373 Pa.s, [8]

m = fluid flow regime. For turbulent flow, m = 0.14 [2]

Shell side pressure drop: The pressure drop in the shell side is evaluated using the expression given by [2].

$$\Delta P_s = 8 \left[j_f \left(\frac{D_s}{D_b}\right) \left(\frac{\mu}{\mu_w}\right)^{-0.14} \left(\frac{L}{B}\right) (1/2\, \rho v_s^2) \right] \qquad (22)$$

Log Mean Temperature Difference (LMTD): This quantified the effective temperature difference between the two heat transfer fluids over the length of the heat exchanger for a counter flow, it is expressed as:

$$\Delta T_{lm} = \frac{(T_{s,i}-T_{t,o})-(T_{s,o}-T_{t,i})}{\ln\left(\frac{T_{s,i}-T_{t,0}}{T_{s,o}-T_{t,i}}\right)} \quad [9] \qquad (23)$$

True Temperature: this is expressed by the equation

$$\Delta T = F_t \Delta T_{lm} \qquad (24)$$

Where F_t = correction factor for a 1-1 pass. From graph of correction factor against temperature efficiency, as shown by [9]

$F_t = 1$.

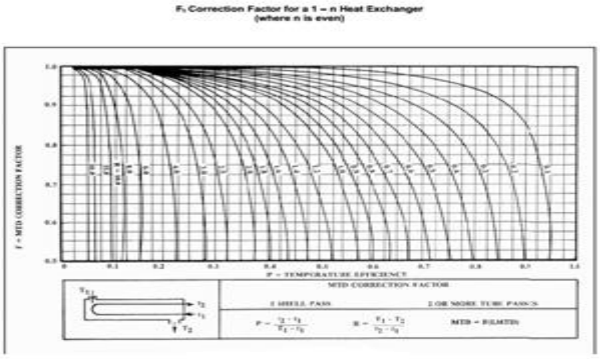

Figure 4. Graph of correction factor against temperature efficiency. Source [9].

Size of heat exchanger: This is determined based on the heat load of the heat exchanger. It is expressed as:

$$A = \frac{q}{U_o \Delta T_m} \qquad (25)$$

Efficiency of the heat exchanger: The efficiency of the heat exchanger is evaluated using equation 27.

$$\eta = \frac{T_{s.i}-T_{s,o}}{T_{s,i}-T_{t,o}} \qquad (26)$$

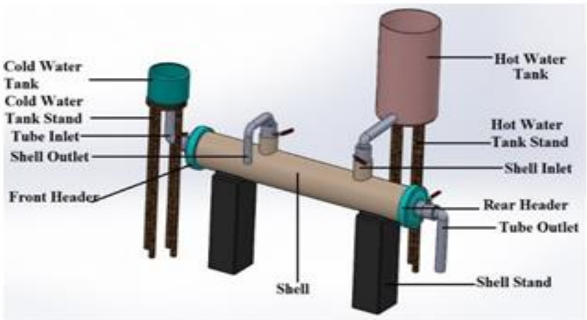

Figure 5. 3D model of assembled view of the Shell and Tube Heat Exchanger laboratory setup.

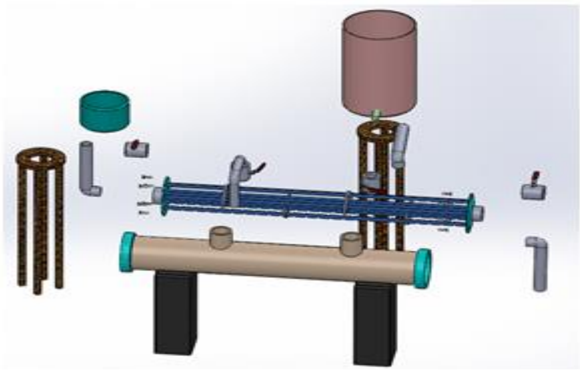

Figure 6. Exploded view of the Shell and Tube Heat Exchanger laboratory setup.

The result of the design formulas employed in the study is shown in table 1.

Table 1. Design parameters and their values.

Parameter	Equation Used	Value
Mechanical design		
Diameter of shell	1	120 mm
Shell thickness	3	5 mm
Tube thickness	4	1 mm
Thickness of dished head	5	5 mm
Tube pitch	6	31.75 mm
Number of tubes	7	6
Tube outside surface area	9	0.6 m^2
Baffle spacing	10	840 mm
Thermal design		
Overall heat transfer coefficient	11	134.23 W/m^2K
Heat load	12	107.973 kW
Mass flowrate of cold water	13	2.58 kg/sec
Tube side Reynolds number	14	20,575
Tube side heat transfer coefficient	15a	1225.72 W/m^2K
Tube side flow Nusselts number	15b	252.83
Shell side heat transfer coefficient	16a	4032.81 W/m^2K
Shell side flow Nusselts number	16b	831.85
Shell side mass flow rate	17	0.5 kg/sec
Shell side cross section flow area	18	8270 mm^2
Shell side linear velocity	19	0.1 m/sec
Shell side Reynold number	20	24182
Tube side pressure drop	21	6.65 Pa
Shell side pressure drop	22	1.88 Pa
Log mean temperature difference	23	49.11 °C
True temperature	24	49.11 °C
Size of heat exchanger	25	16 m^2
Efficiency	26	73.3%

Discussion

Information from table 1 reveals that the heat exchanger is designed to transfer heat from a hot fluid (water) with a mass flow rate of 0.5 kg/sec at a temperature of 100 °C to a cold fluid (water) flowing at 2.58 kg/sec at a temperature of 15 °C. This will cause the temperature of the hot fluid to reduce to 45 °C while the cold fluid increases to 25 °C as both fluid exits the exchanger in a cross flow arrangement. The flow in both the tube and shell compartments of the heat exchanger is turbulent as both Reynolds number is above 4000. This observation is responsible for the high values of Nusselts number, with the shell side flow having a higher Nusselts number. This is an indication that convective heat flow is higher in the shell side as a result of the higher turbulence experienced. This turbulence according to [1] is necessary for effective heat transfer and a way to increase turbulence is by increasing the number of tubes, tube length, shell diameter and employing multiple shells either in series or parallel connection. This would also increase the surface area and subsequently the heat transfer ability of the heat exchanger.

Conclusion

The study carried out a design of a cross flow shell and tube heat exchanger. The design covered both mechanical and thermal considerations. Factors considered in the mechanical design are mainly geometrical factors which include: shell diameter. Shell thickness, tube thickness, thickness of dished head, tube pitch, number of tubes, tube outside surface area, and baffle spacing. In carrying out a thermal design, the Bell Delaware method was employed.

The heat exchanger was designed based on the assumption that there is no phase change, while water at a cold inlet temperature of 15 °C enters the heat exchanger through the tube and hot water at 100 °C enters the heat exchanger through the shell. As a result of the geometry of the tubes and shell, the flow through the heat exchanger is expected to be turbulent thereby enhancing heat transfer between the hot fluid and the cold fluid. With an overall heat transfer coefficient of 134.23 W/m^2K, the heat load to be transferred from the hot fluid to the cold fluid when the hot fluid flows at the rate of 0.5 kg/sec at a velocity of 0.3 m/s in the shell side and cold fluid in the tubes flows at the rate of 2.58 kg/sec with a velocity of 1 m/sec is 107.973 kW. This results to an increase in the cold fluid temperature by 10 °C as it exits the tube, while the temperature of the hot fluid falls to 45 °C. The pressure drops in both the shell and tube were within the allowable ranges, and hence, accepted. With the efficiency of the system at 73.3%, the study is said to have achieved its objective.

This study advocates that reliance on imported laboratory equipment which can be manufactured locally should be discouraged as it neither favours the economy nor the notion of imbibing local content manufacturing in the nation's engineering space.

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