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/m²K and the efficiency of the system at 73.3%, the study is said to have achieved its objective.

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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].

**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.

*Tub 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.

**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.

$$D_s = D_b + \text{Shell bundle clearance}$$

Where $D_b$ = bundle equivalent diameter.

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

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

$$t_s = \frac{P_s d_s}{2 \pi \eta}$$

Where $P_s$ is the pressure within the shell, $\sigma$ is the yield strength of the shell material (AISI type 304) $= 215$ Mpa [3], $\eta$ 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.

\[ t_i = K_1 d_o \frac{P_t}{\sqrt{\sigma_t}} \]  

Where: \( K_1 \) is a constant which depends upon the material of the plate and the method of holding the edges. For mild steel with fixed edges, \( K_1 = 0.35 \).

\( P_t \) = pressure inside the tube as a result of the operating fluid.
\( \sigma_t \) = Allowable design stress of aluminum = 79 MPa [5]

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

\[ t_o = \frac{4 P_t r}{\sigma_o} \]  

Where: \( P \) is the pressure inside the shell, \( r \) is the inside radius of the plate curvature = 50 mm, \( \sigma_o \) 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

\[ t_p = 1.25 d_o \]  

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

\[ N_t = \frac{C T P n d_o^2}{A_4} \]  

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

\[ A_1 = (C L) t_p^2 \]  

Where \( C L \) is the tube layout constant = 0.87 for a 60° tube layout.

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

\[ A_o = \pi d_o N_t L \]  

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.

\[ B = 74 d_o^{0.75} \]  

**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.

\[ \frac{1}{u_o} = \frac{1}{u_{in}} + \frac{1}{u_{out}} + \frac{1}{R_i + R_o} \]  

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:

\[ q = (m C_p) (T_{t_o} - T_{t_i}) = (m C_p) h (T_{s_i} - T_{s_o}) \]  

Where \( m C_p = 0.7855 \rho vd_o^2 N_t \)

\( Cp = 4.185 \text{ KJ/kgK} \).

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

\[ Re_t = \frac{p_t v d_i}{\mu} \]  

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

\[ h_t = \frac{Nu}{L} \]  

Where \( Nu \) = Nusselt number

\[ Nu_t = 0.023 \times Re^{0.8} \times Pr^{0.33} \times \left( \frac{\nu}{\mu} \right)^{0.14} \]  

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

\[ Re_s = \frac{\rho v_s D_s}{\mu} \]  

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

\[ h_s = \frac{Nu \times K}{L} \]  

\[ Nu_s = 0.023 \times Re^{0.8} \times Pr^{0.33} \times \left( \frac{\nu}{\mu} \right)^{0.14} \times K \]  

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

\[ \dot{m}_s = \frac{q}{c_p(T_{s_i} - T_{s_o})} \]  

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

\[ A_s = 0.7855(D_s^2 - d_i^2)N_s \]  

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

\[ v_s = \frac{m_h}{A_s \rho} \]  

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

\[ Re_s = \frac{\rho v_s D_s}{\mu} \]  

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

\[ \Delta P_t = N_p [B j_f \left( \frac{L}{N_t} \right) \left( \frac{\mu}{\mu_t} \right)^m + 2.5] \]  

Where \( N_p \) = number of passes, \( j_f \) = friction factor = 0.0026,

\( L \) = length of tube,

\( \mu = \) dynamic viscosity of bulk fluid 15°C,

\( \mu_t = \) 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].

\[ \Delta P_s = 8 j_f \left( \frac{p_s}{D_s} \right) \left( \frac{\mu}{\mu_t} \right)^{-0.14} \left( \frac{L}{N_t} \right) \left( 1/2 \rho v_s^2 \right) \]  

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

\[ \Delta P_t = \left( \frac{\rho v_s^2 D_s}{2} \right) \left( \frac{\mu}{\mu_t} \right)^{-0.14} \left( \frac{L}{N_t} \right) \left( 1/2 \rho v_s^2 \right) \]  

**Shell side fluid side heat transfer coefficient (\( h_s \)):** This is expressed by equation 20.
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:

$$\Delta T_{lm} = \frac{(T_{i1}-T_{o1})-(T_{i2}-T_{o2})}{\ln\left(\frac{T_{i1}-T_{o1}}{T_{i2}-T_{o2}}\right)} \quad [9] \quad (23)$$

True Temperature: this is expressed by the equation

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

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

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<tr>
<th>Parameter</th>
<th>Equation Used</th>
<th>Value</th>
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<tbody>
<tr>
<td>Diameter of shell</td>
<td>1</td>
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<td>Shell thickness</td>
<td>3</td>
<td>5 mm</td>
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<tr>
<td>Tube thickness</td>
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<td>1 mm</td>
</tr>
<tr>
<td>Thickness of dished head</td>
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<td>5 mm</td>
</tr>
<tr>
<td>Tube pitch</td>
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<td>31.75 mm</td>
</tr>
<tr>
<td>Number of tubes</td>
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<tr>
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<td>Baffle spacing</td>
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**Thermal design**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Equation Used</th>
<th>Value</th>
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<tr>
<td>Overall heat transfer coefficient</td>
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<td>134.23 W/m²K</td>
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<tr>
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<tr>
<td>Mass flowrate of cold water</td>
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<td>73.3%</td>
</tr>
</tbody>
</table>

**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 Nusselt number, with the shell side flow having a higher Nusselt 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/m²K, 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.

References