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How to Calculate and Design Shell and Tube-type Heat Exchanger with a Single Heat Transfer

*Risti Ragadhita, Asep Bayu Dani Nandiyanto**

Universitas Pendidikan Indonesia, Indonesia
Correspondence: E-mail: nandiyanto@upi.edu

ABSTRACT

The process in the industry requires equipment with high efficiency, especially energy. This is closely related to the need to measure the heat-transfer process in heat exchangers. A study of the effectiveness of a heat exchanger is needed to increase the effectiveness of the heat exchanger because each heat exchanger product created has different effectiveness. This study aims to discuss and explain the calculations in the design of shell and tube-type heat exchangers. A step-by-step method for calculating heat exchanger designs is presented, including reviewing simple cases regarding heat exchanger design calculations. This research has the potential to be used as standard information on how to do step-by-step calculations for heat exchanger design.

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1. INTRODUCTION

The transfer of heat from one fluid to another is a critical activity in most chemical industries (Pordanjani *et al*., 2019; Muthukrishman *et al*., 2020). Heat transfer is most commonly used in the construction of heat transfer equipment for exchanging heat from one fluid to another. Heat exchangers are often used equipment for efficient heat transfer (Thanikodi *et al*., 2020; Patel, 2023). The application, available floor area, available resources, connections in the field, cost, and many other considerations all influence the selection of a heat exchanger for a certain application. In a highly competitive context, the heat exchanger must provide the required heat transfer while taking up less space, weighing less, and being priced competitively (Douadi *et al*., 2022).

Shell and tube-type heat exchangers are the most commonly used heat exchangers in industries such as chemical processing, power production, oil refining, refrigeration, and air conditioning (Gupta *et al*., 2022; Gürses *et al*., 2022). More than 35-40% of heat exchangers utilized in modern engineering sectors are shell and tube type due to their sturdy construction geometry, ease of maintenance, and upgradeability. Shell and tube heat exchangers transmit significant amounts of heat at a low cost, give a cost-effective tube surface while limiting floor area, liquid volume, and weight requirements, and come in a variety of diameters and lengths (Mohammadi *et al*., 2020). Heat exchangers with shell and tube construction have been around for over 150 years. The current manufacturer's thermal technology and manufacturing techniques are thoroughly defined and utilized.

Because of their widespread use, shell and tube heat exchangers must be improved in performance. These heat exchangers are studied in two parts: shell side and tube side. The research on the tube side is simple with few characteristics; on the other hand, the study of the shell side is complicated with numerous parameters, the most essential of which are the baffle and tube layout. Many reports explain the theory of heat exchangers (Vasiliev, 2005). However, studies documenting how to calculate shell and tube-type heat exchanger designs have not been well documented. This report is to discuss and explain the calculation of a shell and tube-type heat exchanger. Here, a step-by-step method on how to calculate a shell and tube-type heat exchanger design is presented, including a review of simple case examples of heat exchanger calculations.

2. LITERATURE REVIEW

2.1. Shell and Tube Type Heat Exchanger

Amongst all types of exchangers, shell and tube exchangers are the most commonly used heat exchange equipment. This type is used for higher flow rates, thus frequently used in industry (Gupta *et al*., 2022; Gürses *et al*., 2022). Tubes are installed in parallel and many are in one shell. Cold fluid enters the tube. Hot fluid enters from different ends to counter the current flow in the shell (Bahiraei *et al*., 2021). The advantage of this type of heat exchanger is that it has a larger heat transfer surface per unit volume, has a good mechanical arrangement with a shape good enough for operation pressurized, is available in a variety of construction materials, operating procedures easier, better design methods are available, and clean-up can be done easily (Li *et al*., 2020). Therefore, the most common type of heat exchanger profitable to use is a shell and tube-type heat exchanger. The common types of shell and tube exchangers are (Saffarian *et al*., 2019; Saffiudeen *et al*., 2020):

(i) Fixed tube sheet exchanger

Fixed tube-sheet exchanger called a non-removable tube bundle. The fixed tube sheet design is the most basic and least expensive type of shell and tube exchanger. The tube sheet

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is welded to the shell in this form of exchanger, and there is no relative movement between the shell and tube bundle. An illustration of a fixed tube-sheet exchanger is shown in **Figure 1**.

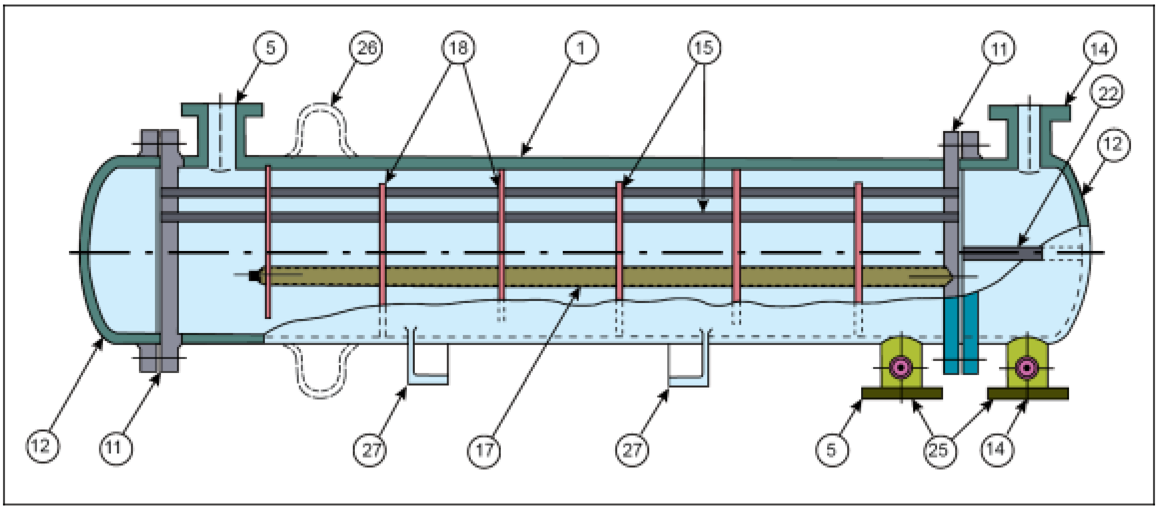

Figure 1. Fied tube-sheet exchanger.

(ii) Removable tube bundle

To facilitate cleaning and replacement, the tube bundle can be detached. Floating-head and U-tube exchangers are two types of removable tube bundle exchangers. The floating-head exchanger is made out of a stationary tube sheet that is fastened to the shell flange. The tubes may expand into a freely riding floating head or floating tube sheet at the other end of the bundle. The tube sheet is affixed to a floating head cover, and the entire bundle can be removed for cleaning and inspection of the interior. **Figure 2** depicts the floating-head exchanger. The U-tube exchanger is made up of tubes bent in the shape of a "U" and rolled back into the tube sheet seen in **Figure 3**. This means that depending on the tube configuration, it will omit some tubes in the center of the tube bundle. The tubes can freely grow towards the "U bend end".

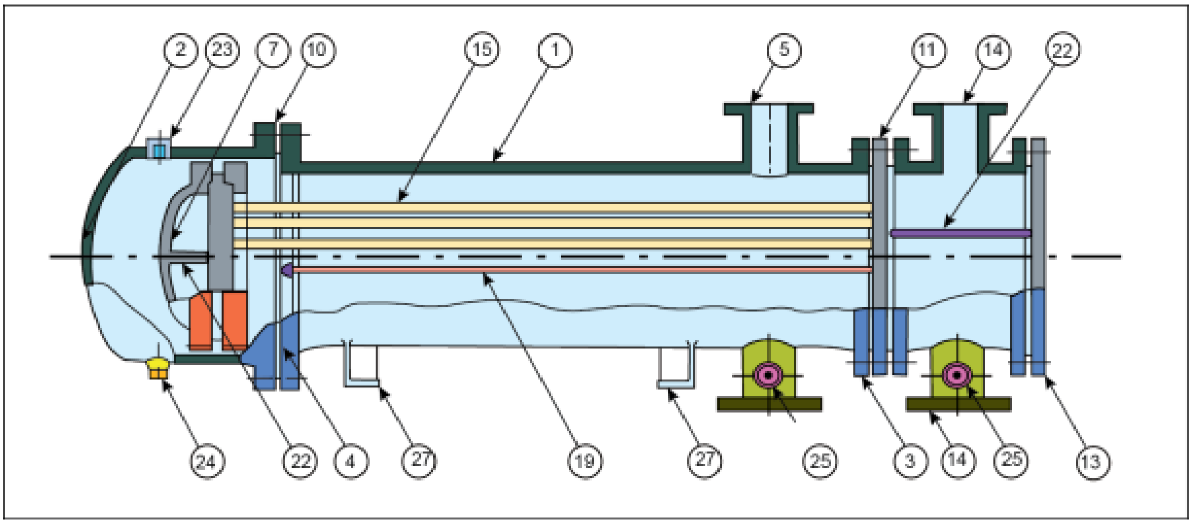

Figure 2. Floating-head heat exchanger.

DOI:
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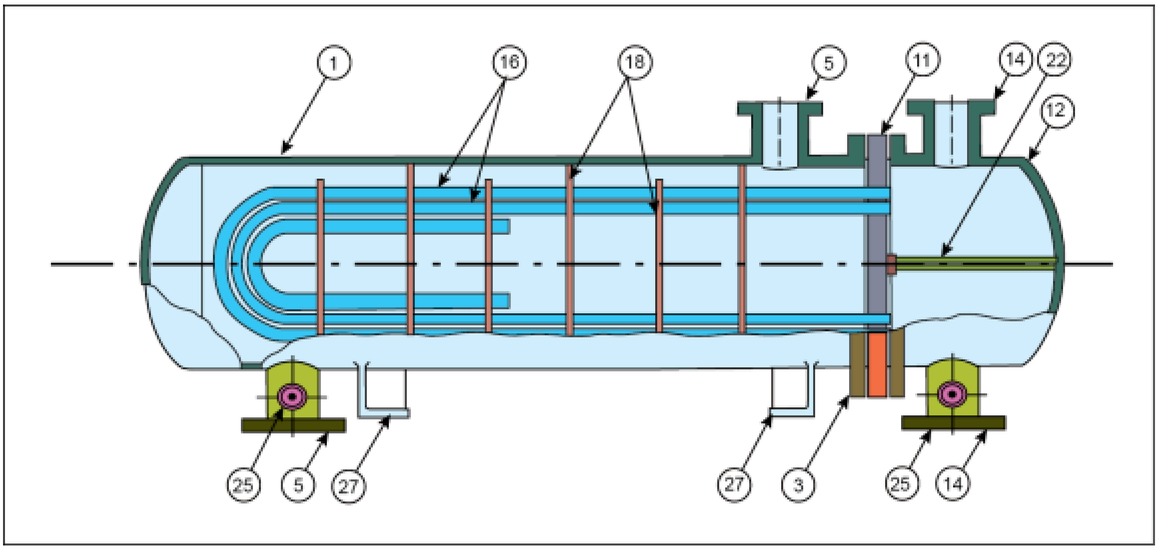

Figure 3. Romovable u-tube heat exchanger.

Figures 1-3 (International Standard: 4503-1967) depict typical parts and connections, which are summarized in **Table 1**.

Table 1. Typical component of shell and tube type heat exchanger.

No	Component	No	Component
1	Shell	16	Tubes (U-type)
2	Shell cover	17	Tie rods and spacers
3	Shell flange (channel end)	18	Transverse (or cross) baffles or support plates
4	Shell flange (cover end)	19	Longitudinal baffles
5	Shell nozzle or branch	20	Impingement baffles
6	Floating tube sheet	21	Floating head support
7	Floating head cover	22	Pass partition
8	Floating head flange	23	Vent connection
9	Floating head gland	24	Drain connection
10	Floating head backing ring	25	Instrument connection
11	Stationary tube sheet	26	Expansion bellows
12	Channel or stationary head	27	Support saddles
13	Channel cover	28	Lifting lugs
14	Channel nozzle or branch	29	Weir
15	Tube (Straight)	30	Liquid-level connection

2.2 Flow Type in Heat Exchanger

Several flow types in the heat exchanger, including counter-current, co-current, and cross-flow. Counter current flow is flow in the opposite direction, where one fluid enters at one end of the heat exchanger, while the other fluid enters at the other end of the heat exchanger, each fluid flows in the opposite direction. This type of counter-current flow provides better heat compared to unidirectional or parallel flow. Meanwhile, the number of passes also influences the effectiveness of the heat exchanger used (Bahiraei *et al.*, 2021; Saeid and Seetharamu, 2006). Parallel flow or co-current is unidirectional flow, where both fluids enter at the same end of the heat exchanger and both fluids flow in the same direction towards the other end of the heat exchanger (Saeid and Seetharamu, 2006). Cross flow often called cross flow is fluids that flow along a surface moving in mutually perpendicular directions (Harris *et al.*, 2002).

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2.3 Standards and Codes for Heat Exchanger

British Standard, BS 3274, governs the mechanical design characteristics, manufacturing, materials of construction, and testing of shell and tube exchangers. The TEMA standards, developed by the American Tubular Heat Exchanger Manufacturers Association, are also widely utilized. The TEMA standards cover three types of exchangers: class R for exchangers used in the petroleum and related industries, class C for exchangers used in commercial and general process applications, and class B for exchangers used in the chemical process industries.

2.4 Thermal Design Consideration

A shell and tube heat exchanger's thermal design typically includes determining the heat transfer area, number of tubes, tube length, and diameter, tube layout, number of shell and tube passes, type of heat exchanger (fixed tube sheet, removable tube bundle, and so on), tube pitch, number of baffles, their type and size, shell and tube side pressure drop, and so on (Sing *et al*., 2019).

2.4.1 Tubes

Tube diameter sizes range from 5/8 in. (16 mm) to 2 in. (50 mm). Smaller diameters (5/8 to 1 in. (16 to 25 mm) are favored for most jobs because they result in more compact, and thus less expensive, exchangers. Larger tubes are easier to clean mechanically and would be chosen for highly fouling fluids. The tube thickness (gauge) is chosen to resist internal pressure while also allowing for corrosion (Aresti *et al*., 2018). Steel tubes for heat exchangers are covered by BS 3606 (metric sizes); other materials are covered by BS 3274. **Table 2** shows the standard sizes and wall thicknesses for steel tubes.

Table 2. Standard dimension for steel tubes

Outside diameter (mm)	Wall thickness (mm)				
16	1.2	1.6	2.0	-	-
20	-	1.6	2.0	2.6	-
25	-	1.6	2.0	2.6	3.2
30	-	1.6	2.0	2.6	3.2
38	-	-	2.0	2.6	3.2
50	-	-	2.0	2.6	3.2

Tube lengths for heat exchangers are as follows: 6 ft (1.83 m), 8 ft (2.44 m), 12 ft (3.66 m), 16 ft (4.88 m), 20 ft (6.10 m), and 24 ft (7.32 m). Longer tubes will reduce the shell diameter for a given surface area, resulting in a lower cost exchanger, especially at high shell pressures (Aresti *et al*., 2018). The ideal tube length-to-shell diameter ratio is usually between 5 and 10. When using U-tubes, the tubes on the outside of the bundle will be longer than the tubes on the interior. For usage in the thermal design, the average length must be estimated. U-tubes will be bent and trimmed to size from standard tube lengths. As a starting point for design calculations, 3/4 in. (19 mm) is a good trial diameter.

2.4.2 Tube arrangements

Figure 4 shows how the tubes of an exchanger are typically organized in an equilateral triangular, square, or rotating square arrangement. The triangular and rotated square patterns give higher heat-transfer rates but at the expense of a higher pressure drop than the square pattern. A square, or rotated square arrangement, is used for heavily fouling fluids, where it is necessary to mechanically clean the outside of the tubes. The recommended tube pitch (distance between tube centers) is 1.25 times the tube outside diameter, and this will normally be used unless process requirements dictate otherwise. Where a square pattern is used for ease of cleaning, the recommended minimum clearance between the tubes is 0.25 in. (6.4 mm) (Kallannavar *et al*., 2020).

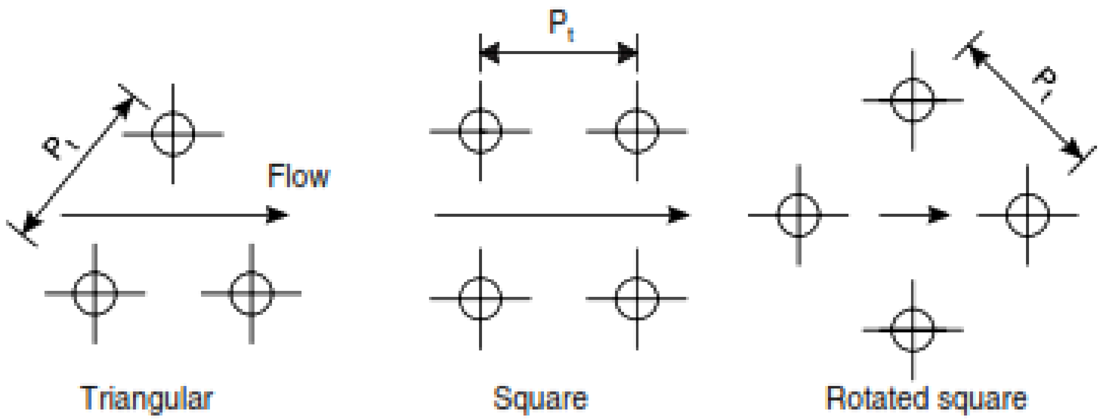

Figure 4. Tube arrangements.

2.4.3 Tube-side passes

To expand the length of the flow channel, the fluid in the tube is frequently directed to flow back and forth in several "passes" via groups of tubes placed in parallel. The number of passes is chosen to provide the necessary tube-side design velocity. Exchangers are constructed with one to sixteen tube passes. By splitting the exchanger headers (channels) with partition plates (pass partitions), the tubes are organized into the number of passes required. **Figure 5** depicts the pass partition layout for 2, 4, and 6 tube passes (Silaipillayarputhur and Khurshid, 2019).

2.4.4 Shell

The British standard BS 3274 covers exchangers with diameters ranging from 6 in. (150 mm) to 42 in. (1067 mm), while the TEMA standards cover exchangers with diameters ranging from 60 in. (1520 mm). Shells up to around 24 in. (610 mm) are typically made of standard, near tolerance tubing; above 24 in. (610 mm), they are rolled from the plate. **Table 3** shows the values, which have been translated to international standard units and rounded (Abd *et al*., 2018). Furthermore, the shell inside diameter is calculated by considering clearance. The clearance between the shell inside diameter and the tube bundle can be determined from **Figure 6**, assuming a pull-through floating head.

2.4.5 Tube-sheet layout

The bundle diameter is determined not only by the number of tubes but also by the number of tube passes because spaces must be allowed in the tube sheet pattern to accommodate the pass partition plates (Boulougouras and Besseris, 2023). **Table 3** contains the constants for use in triangular and square tube-sheet layouts.

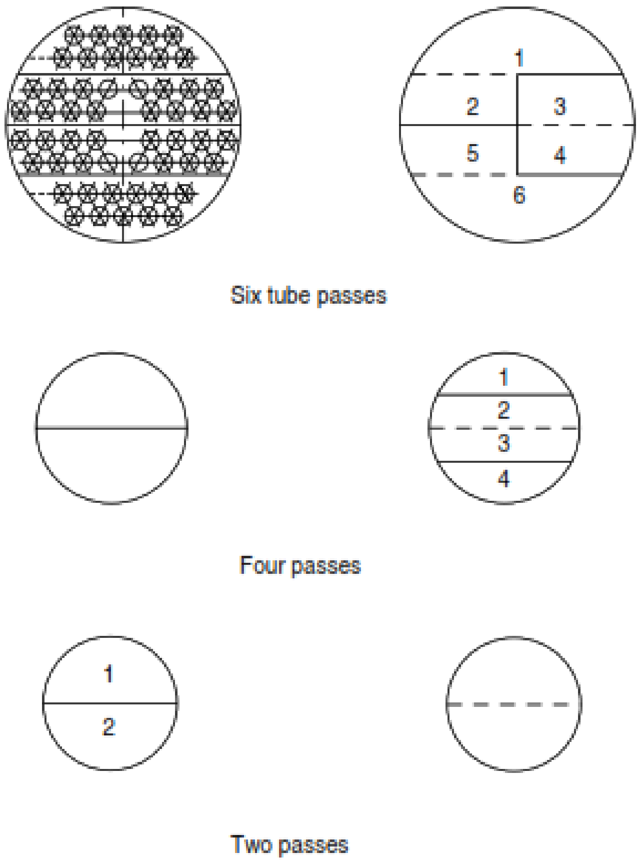

Figure 5. Tube arrangements, showing pass-partitions in the header.

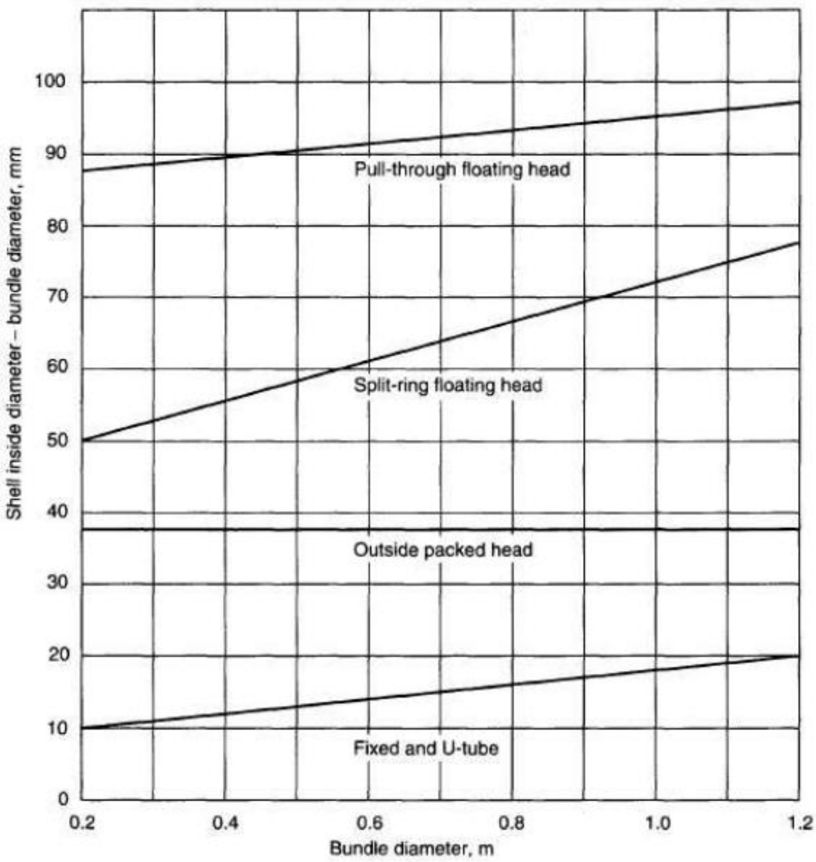

Figure 6. Tube bundle clearance.

Table 3. Constant for use in triangular and square tube-sheet layout.

Triangular pitch = 1.25d_o					
No. Passes	**1**	**2**	**4**	**6**	**8**
K_1	0.319	0.249	0.175	0.0743	0.0365
n_1	2.142	2.207	2.285	2.499	2.675
Square pitch = 1. 25d_o					
K_1	0.215	0.156	0.158	0.0402	0.0331
n_1	2.207	2.291	2.263	2.617	2.643

2.4.6 Shell types (passes)

Figure 7(a-e) depicts the main shell layouts. TEMA standards use the letters E, F, G, H, and J to indicate the various varieties. The E shell is the most typical configuration. When the shell and tube side temperature differences are too great for a single pass, two shell passes (F shell) are employed. However, obtaining a suitable seal with a shell-side baffle is difficult, and the same flow arrangement can be achieved by connecting two shells in series (Costa and Queiroz, 2008). **Figure 7(f)** depicts one technique for sealing the longitudinal shell-side baffle. The divided flow and split-flow arrangements (G and J shells) are utilized to reduce shell-side pressure drop, where pressure drop, not heat transmission, is the controlling factor in the design.

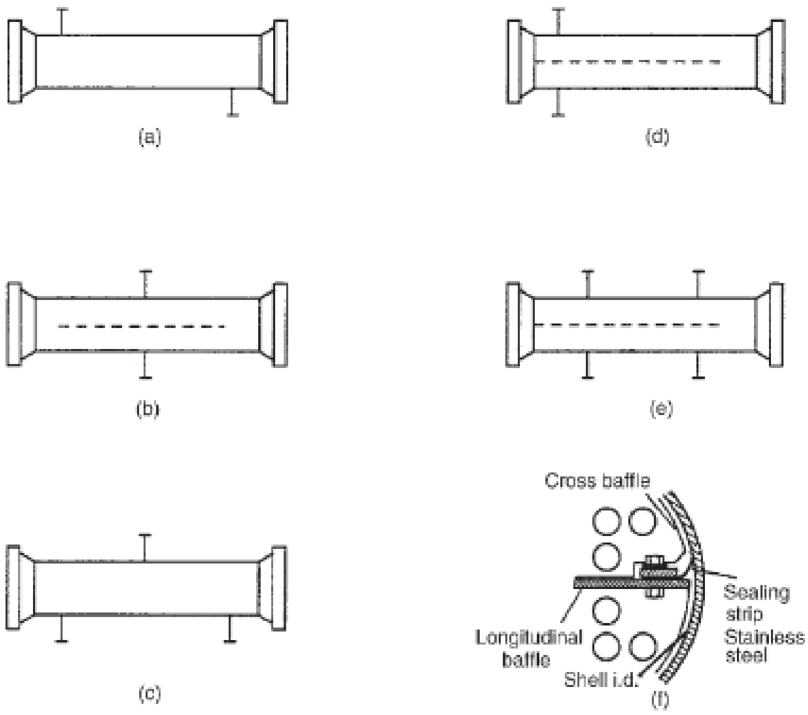

Figure 7. Shell types (pass arrangements) of (a) one-pass shell (E shell); (b) split flow (G shell); (c) divided flow (J shell); (d) two-pass shell with longitudinal baffle (F shell); and (e) double split flow (H shell).

2.4.7 Shell and tube designation

A common method of describing an exchanger is to designate the number of shell and tube passes: m/n; where m is the number of shell passes and n is the number of tube passes. Thus, 1/2 describes an exchanger with 1 shell pass and 2 tube passes, and 2/4 an exchanger with 2 shell passes and 4 four tube passes (Silaipillayarputhur and Khurshid, 2019).

DOI:
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2.4.8 Baffles

In the shell, baffles are employed to direct the fluid stream over the tubes, increasing the fluid velocity and thus the rate of transfer (Abolpour *et al*., 2021). The single segmental baffle, shown in **Figure 8(a)**, is the most often used type of baffle; additional varieties are shown in **Figures 8(b, c, and d)**.

The baffles would restrict condensate flow if the setup depicted in **Figure 8(a)** was utilized with a horizontal condenser. This issue can be solved by either rotating the baffle arrangement through 90° or trimming the baffle's base, as shown in **Figure 8**. The word "baffle cut" refers to the proportions of a segmental baffle. The baffle cut is the height of the segment removed to construct the baffle given as a percentage of the diameter of the baffle disc.

Baffle cuts ranging from 15% to 45% are employed. In general, a baffle cut of 20 to 25% is optimal for good heat transfer rates without excessive drop. As a clearance must be left for assembly, there will be some fluid leakage around the baffle (Kücük *et al*., 2019; Abdelkader and Zubair, 2019). The required clearance is determined by the shell diameter; typical values and tolerances are shown in **Table 4**.

Another leakage path is the space between the tube holes in the baffle and the tubes. Normally, the maximum design clearance is 1/32 in. (0.8 mm). The specifications specify the minimum thickness for baffles and support plates. The baffle spacings employed vary between 0.2 and 1.0 times the shell diameters. Closer baffle spacing results in higher heat transfer coefficients but larger pressure drops. The ideal separation will be between 0.3 and 0.5 times the shell diameter.

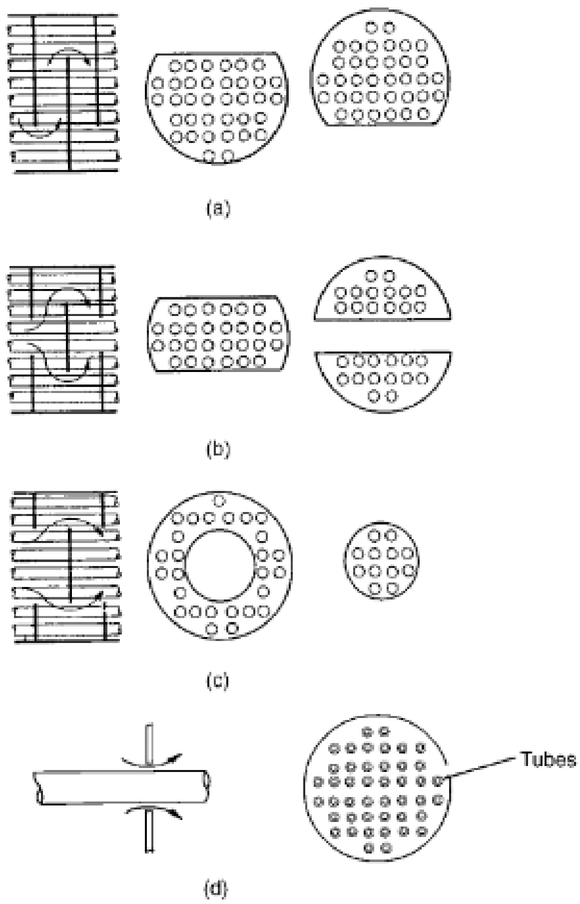

Figure 8. Types of baffle used in shell and tube heat exchangers of (a) segmental; (b) segmental and strip; (c) disc and doughnut; and (d) orifice types.

DOI:
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Table 4. Typical baffle clearances and tolerances.

Shell Diameter, D_s	Baffle Diameter	Tolerance
Pipe Shells		
6 to 25 in. (152 to 635 mm)	$D_s - \frac{1}{16}\,in.$ (1.6 mm)	$+\frac{1}{32}\,in.$ (0.8 mm)
Plate Shell		
6 to 25 in. (152 to 635 mm)	$D_s - \frac{1}{8}\,in.$ (3.2 mm)	+0, $-\frac{1}{32}\,in.$ (0.8 mm)
27 to 42 in. (686 to 1067 mm)	$D_s - \frac{3}{16}\,in.$ (4.8 mm)	+0, $-\frac{1}{16}\,in.$ (1.6 mm)

3. METHOD

In the research, the heat exchanger was designed to obtain optimal performance results. The standard used in designing shell and tube-type heat exchangers is TEMA (Tubular Exchanger Manufactures Association). In this study, a series of calculations were carried out to analyze the heat transfer correction coefficient values, pressure drops on the tube and shell sides, as well as design effectiveness. The first step in designing a shell and tube type heat exchanger is to determine several thermal design specifications and dimensional specifications of the shell and tube heat exchanger. Several initial specifications that must be determined and collected are presented in **Tables 5** and **6**. **Table 5**, which is a thermal specification, must be adapted to the literature. Meanwhile, the tool dimension specifications in Table 6 must be adjusted to TEMA standards.

Table 5. Heat exchanger operating data.

Spesifications	Hot Fluida	Cold Fluida
Mass flow rate (m; kg/s)	…..	…..
Dynamic viscosity (μ; Kg/m.s)	…..	…..
Inlet Temperature (°C)	…..	…..
Outlet Temperature (°C)	…..	…..
Heat Capacity (Cp; J/Kg.K)	…..	…..
Tube and Shell material	…..	…..
Thermal conductivity of fluid materials (K; W/m.K)	…..	…..

Table 6. Initial specifications of the tool.

Dimensions	Standard Tube and Shell Design	International Standard
Tube outer diameter (do_t)	…. inchi	…. m
Tube inner diameter (di_t)	…. inchi	…. m
BWG	….	….
Tube length (L)	…. inchi	…. m
Tube Layout	….	….
Pass Flow (N_p)	….	….
Inner diameter (di_s)	…. inchi	…. m
Outer diameter (do_s)	…. inchi	…. m
Baffle	…. inchi	…. m
Tube pitch (P_t)	…. inchi	…. m
Clearance (C; Pt-do_t)	…. inchi	…. m

The next step to determine the effectiveness of heat exchanger design is to carry out calculation steps according to equations 1-27 as presented in **Table 7**. The equations in **Table 7** have been organized sequentially to finally calculate the effectiveness of the heat exchanger.

Table 7. Heat exchanger parameter calculation.

Section	Parameter	Equation	Eq.
Basic parameters	The energy transferred (Q)	$Q_c = Q_h$ $m_c \times Cp_c \times \Delta T_c = m_h \times Cp_h \times \Delta T_h$ Where, Q = the energy transferred (Wt) *m* = the mass flow rate of the fluid (Kg/s) *Cp* = the specific heat ΔT = the fluid temperature difference (°C).	(1)
	Logarithmic mean temperature differenced (LMTD)	$LMTD = \frac{(T_{hi} - Tc_i) - (T_{ho} - Tc_o)}{ln\frac{(T_{hi} - Tc_i)}{(T_{ho} - Tc_o)}}$ Where, T_{hi} = temperature of hot fluid inlet (°C) T_{ho} = temperature of hot fluid outlet (°C) Tc_i = temperature of cold fluid inlet (°C) Tc_o= temperature of cold fluid outlet (°C)	(2)
	Correction factor	$P = \frac{T_{co} - T_{ci}}{T_{hi} - T_{ci}}$ $R = \frac{T_{hi} - T_{ho}}{T_{co} - T_{ci}}$ $F = \frac{\sqrt{R^2+1}\ \ln[\frac{1-P}{1-PR}]}{(R-1)\ln\left(\frac{2-P(R+1-\sqrt{R^2+1}}{2-P(R+1+\sqrt{R^2+1}}\right)}$ Thus, the value of the temperature change is $\Delta t = F \times LMTD$ Where, T_{hi} = temperature of hot fluid inlet (°C) T_{ho} = temperature of hot fluid outlet (°C) Tc_i = temperature of cold fluid inlet (°C) Tc_o= temperature of cold fluid outlet (°C) *P* = temperature efficiency of the heat exchanger *R* = ratio of the product of fluid flow in the shell with specific heat to fluid flow in the tube *F* = correction factor Δt = temperature change LMTD = Logarithmic mean temperature differenced (calculated using Eq. 2)	(3) (4) (5) (6)
	Heat Transfer Field Area (A)	$A = \frac{Q}{U \times (LTMD \times F)}$ Where, *Q* = the energy transferred (W) *U* = the overall heat transfer coefficient *LMTD* = the logarithmic mean temperature difference.	(7)

DOI:
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Table 7 (Continue). Heat exchanger parameter calculation.

Section	Parameter	Equation	Eq.
	Number of Tubes (N)	$Nt = \frac{A}{\pi \times D_o \times l}$ Where, *N* = the number of tubes *A* = the area of the heat transfer area (m^2), π = 3.14 D_o= tube diameter (m) l = tube diameter (m).	(8)
Tube	Surface Area of Total Heat Transfer in Tube (a_t)	$a_t = N_t \frac{a'_t}{n}$ Where, a_t = the total heat transfer surface area in the tube (m^2) N_t = the number of tubes a'_t = the flow area in the tube (m^2) *n* = the number of passes. And,	(9)
		$a'_t = \frac{\pi}{4} \times (D_{i,t})^2$ Where, $D_{i,t}$ = inner diameter of tube	(10)
	Mass Flow Rate of Fluid in Tube (Gt)	$Gt = \frac{m_{th}}{a_t}$ Where, Gt = the mass flow of water in the tube (kg/m^{2}s) m_h = the mass flow rate of the hot fluid (Kg/s) a_t = the flow area tube (m^2	(11)
	Reynold number (Re,t)	$Re_t = \frac{di_t \times Gt}{\mu}$ Where, Re_t = the Reynolds number in tube di_t = the inner tube diameter (m), Gt = the mass flow of water in the tube (m^2) μ = the dynamic viscosity (Kg/ms).	(12)
	Prandtl Number (Pr,t)	$Pr = (\frac{C_p \times \mu}{K})^{\frac{1}{2}}$ Where, *Pr* = Prandtl number *Cp* = the specific heat of the fluid in the tube μ = the dynamic viscosity of the fluid in the tube (Kg/ms) *K* = the thermal conductivity of the tube material (W/m°C).	(13)
	Nusselt number (Nu,t)	$Nu = 0.023 \times {Re_t}^{0.6} \times Pr^{0.33}$ Where, Re_t = the Reynolds number in tube *Pr* = Prandtl number	(14)

DOI:
p-ISSN : 2828-920X e-ISSN: 2828-9951

Table 7 (Continue). Heat exchanger parameter calculation.

Section	Parameter	Equation	Eq.
	Inside coefficient (h_i)	$$hi = \frac{Nu \times K}{d_i, t}$$ Where, hi = the convection heat transfer coefficient in the tube (W/m^2°C) K = the thermal conductivity of the material (W/m°C) d_i, t = the inner tube diameter (m).	(15)
Shell	Shell flow area (A_s)	$$A_s = \frac{d_s \times C \times B}{P_t}$$ Where, d_s = shell diameter (m) C = clearance (P_t-d_o) B = Baffle spacing P_t = tube pitch (1.25× d_o) (m).	(16)
	Mass Flow Rate of Water in Shell (Gs)	$$Gs = \frac{m_c}{A_s}$$ m_c = the mass flow rate of the cold fluid (Kg/s) A_s = the shell flow area (m^2).	(17)
	Equivalent diameter (d_e)	$$d_e = \frac{4(\frac{Pt}{2} \times 0.87\, Pt - \frac{1}{2}\pi \frac{{d_{o,t}}^2}{4})}{\frac{1}{2}\pi d_{o,t}}$$ Where, P_t = tube pitch (1.25× d_o) (m) π = 3.14 $d_{o,t}$ = tube outside diameter (m).	(18)
	Reynold number (Re,s)	$$Re_s = \frac{d_e \times Gs}{\mu}$$ Re_s = Reynold number di_s = inner tube diameter (m) Gs = the mass flow of water in the shell (Kg/m^{2}s) μ = the dynamic viscosity (Kg/ms).	(19)
	Prandtl Number (Pr,s)	$$Pr = (\frac{C_p \times \mu}{K})^{\frac{1}{2}}$$ Pr_s = Prandtl number Cp = specific heat capacity (kJ/kg°C) μ = dynamic fluid viscosity (Kg/ms) K = thermal conductivity (W/m°C).	(20)
	Nusselt number (Nu,s)	$$Nu_s = 0.023 \times {Re_s}^{0.6} \times Pr^{0.33}$$ Re_s = Reynold number Pr = Prandtl number	(21)
	Convection Heat Transfer Coefficient (hs)	$$hs = \frac{Nu \times K}{d_e}$$ ho = convection heat transfer coefficient (W/m^2°C) K = thermal conductivity (W/m°C) d_e = shell diameter (m).	(22)

DOI:
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Table 7 (Continue). Heat exchanger parameter calculation.

Section	Parameter	Equation	Eq.
Heat rate	Hot Fluid Rate (tube) (C_h)	$C_h = m_h.Cp_h$ Where, C_h = hot fluid rate (W/°C) Cp_h = specific heat capacity (J/Kg°C) m_h = mass flow rate of hot fluid (Kg/s).	(23)
	Cold Fluid Rate (shell) (C_C)	$C_C = m_c.Cp_c$ C_c = cold fluid rate (W/°C), Cp_h = specific heat capacity (J/Kg°C), m_c = mass flow rate of cold fluid (Kg/s)	(24)
Maximum Heat Transfer Rate (Qmax)		$Qmax = C_h(T_{hi,t} - T_{ci,s})$ Q_{max} = maximum heat transfer (W) C_{min} = minimum heat capacity rate (W/°C) $T_{h,i}$ = temperature of hot fluid inlet (°C) $T_{c,i}$ = temperature of cold fluid inlet (°C).	(25)
Effectiveness	Heat Exchanger Effectiveness (ε)	$\varepsilon = \frac{Qact}{Qmax} \times 100\%$ Where, Q_{act} = actual energy transferred (W) Q_{max} = maximum heat transfer (W)	(26)
	Number of Transfer Unit (NTU)	$NTU = \frac{U \times A}{C_{min}}$ Where, U = overall heat transfer coefficient (W/m^2°C A = heat transfer area (m^2) C_{min} = minimum heat capacity rate (W/°C).	(27)
Tube Length	Tube Length (Lt)	$Lt = \frac{NTU \times C_{min}}{U \times \pi \times do_{,t} \times Nt \times 2}$ Where Lt = tube length (m) NTU = Number of transfer unit C_{min} =Hot fluid rate (W/K) U = Overall heat transfer ($\frac{W}{m^2}K$) $do_{,t}$ = Outer tube diameter (m) Nt = number of tube	(28)

4. RESULTS AND DISCUSSION

Here, an example of a case of designing a shell and tube-type heat exchanger is given. The input for thermal design calculations before modeling is shown in **Table 8**.

After the input thermal design data is determined, calculations are carried out according to **equations (1)-(27)** as follows.

(i) Determining the energy transferred (Q)

Determination of heat transfer for hot fluids and cold fluids is calculated using **Eq. (1)**. Here, for example, the calculation of heat transfer determination is carried out to calculate heat transfer for hot fluids. The variable value of $m_h, Cp_h, and\ \Delta T_h$ ware taken from **Table 8**.

$Q_c = Q_h$ (1)

$$Qh = m_h \times Cp_h \times \Delta T_h$$

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$Qh = 1.5\frac{kg}{s} \times \frac{2161J}{Kg.K} \times (380 - 340)K = 129{,}660J/s$

Table 8. Heat exchanger operating data.

Specification	Input Fluid
Tube Side (Hot Fluid)	
Fluid Material	Oil
Mass flow rate (m_{th}; kg/s)	1.5
Dynamic viscosity (μ; Kg/m.s)	6.16×10^{-4}
Inlet Temperature in tube side (T_{hi}; °C)	107
Inlet Temperature in tube side (T_{hi}; K)	380
Outlet Temperature in tube side (T_{ho}; °C)	27
Outlet Temperature in tube side (T_{ho}; K)	300
Heat Capacity (*Cp*; J/Kg.K)	2161
Thermal conductivity of fluid materials (*K*; W/m.K)	0.138
Tube and Shell material	Copper
Shell Side (Cold Fluid)	
Fluid Material	Water
Mass flow rate (m_{sh}; kg/s)	1.72
Dynamic viscosity (μ; Kg/m.s)	2.54×10^{-3}
Inlet Temperature in tube side (Tc_i; °C)	27
Inlet Temperature in tube side (Tc_i; K)	300
Outlet Temperature in tube side (Tc_o; °C)	45
Outlet Temperature in tube side (Tc_o; K)	318
Heat Capacity (*Cp*; J/Kg.K)	4174
Thermal conductivity of fluid materials (*K*; W/m.K)	0.640
Tube and Shell material	Aluminium

(ii) Determining LMTD value, using input values in **Table 8**.

The logarithmic average of the temperature difference between the hot and cold feeds at each end of the heat exchanger is used to calculate the LMTD. The greater the LMTD, the greater the heat transfer. LMTD was calculated using **Eq. (2)**. Where, the value of T_{hi}, $T_{ho}, T_{ci}, and\ T_{ci}$ were taken from **Table 8**.

$$LMTD = \frac{(T_{hi}-Tc_O)-(T_{ho}-Tc_i)}{ln\frac{(T_{hi}-Tc_O)}{(T_{ho}-Tc_i)}} \quad (2)$$

$$LMTD = \frac{(380-318)-(340-300)}{ln\frac{(380-318)}{(340-300)}} = 50.22\ K$$

(iii) Determine correction factor

Correction factors are carried out by determining the values of several variables, including:

- Determining the temperature efficiency of the heat exchanger (*P*) using **Eq. (3).** Where, the value of $T_{hi}, T_{ho}, T_{ci}, and\ T_{ci}$ were taken from **Table 8**.

$$P = \frac{T_{co}-T_{ci}}{T_{hi}-T_{ci}} \quad (3)$$

$$P = \frac{318-300}{38-300} = 0.225$$

- Determining the ratio of the product of fluid flow in the shell with specific heat to fluid flow in the tube (*R*) with specific heat using **Eq. (4)**. Where, the value of T_{hi}, $T_{ho}, T_{ci}, and\ T_{ci}$ were taken from **Table 8**.

$$R = \frac{T_{hi}-T_{ho}}{T_{co}-T_{ci}} \quad (4)$$

$$R = \frac{380-340}{318-300} = 2.22$$

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- Determining correction factor (F) using **Eq. (5)**. Where, *P* and *R* values were taken using calculation from **Eqs. (3)** and **(4)**, respectively.

$$F = \frac{\sqrt{R^2+1}\,\ln[\frac{1-P}{1-PR}]}{(R-1)\ln(\frac{2-P(R+1-\sqrt{R^2+1}}{2-P(R+1+\sqrt{R^2+1}})} \quad (5)$$

$$F = \frac{\sqrt{2.22^2+1}\,\ln[\frac{1-0.225}{1-0.225\times 2.22}]}{(2.22-1)\ln(\frac{2-0.225(2.22+1-\sqrt{2.22^2+1}}{2-0.225(2.22+1+\sqrt{2.22^2+1}})} = 0.94$$

Thus, the value of the temperature change is calculated using **Eq. 6**. Where, *F* and *LMTD* values were taken using calculation from **Eqs. (5)** and (**2**), respectively.

$$\Delta t = F \times LMTD \quad (6)$$

$$\Delta t = 0.94 \times 50.22K = 47.2K$$

After calculating the thermal design calculations, the next step is to carry out calculations on the tube side. However, before carrying out calculations on the tube side, the design and sizes of the tube must first be determined. The data determined are dimensions such as tube length, inner and outer diameter of the tube, and number of flow paths in the tube concerning standard specifications for tube dimensions such as the TEMA standard. In this study, tube dimension specifications assumtions are summarized in Table 9. The tube side calculation was carried out from point (iv) to point (xi).

Table 9. Tube dimensions.

Dimensions	Standard Tube Design	International Standard
Tube outer diameter (do_t)	¾ inchi	0.01905 m
Tube inner diameter (di_t)	0.652	0.01656 m
BWG	18	-
Tube length (*L*)		2.15 m
Tube Layout	Triangular pitch	Triangular pitch
Pass Flow (n)	2	2

(iv) Determining the heat transfer area

To calculate the heat transfer area using **Eq. (7)**, several parameters must be used as references in the calculation, including the energy balance value (*Q*) that calculated using Eq. 1, the average actual temperature difference or *LMTD* (calculated using **Eq. (6)**), and the overall heat transfer coefficient value (*U*). For the *U* value, first use the assumed heat transfer coefficient. Based on the literature, the heat transfer coefficient value for hot oil and cold-water fluids is 800 $\frac{W}{m^2}$ (Kern, 1965).

$$A = \frac{Q}{U \times LTMD \times F} \quad (7)$$

$$A = \frac{129{,}660\,W}{800\frac{W}{m^2}\times 47.2\,K} = 3{,}43\,m^2$$

(v) Determining the number of tubes

Number of tube was calculated using **Eq. (8)**. Where, heat transfer field area (A) was calculated using **Eq. (7)**. Meanwhile, D_o and l tube were taken from **Table 9**.

$$Nt = \frac{A}{\pi \times D_o \times l} \quad (8)$$

$$Nt = \frac{3.43\ m^2}{3.14\times 1.905\times 10^{-2} m \times 2.15\,m} = 26{,}7 = 26$$

(vi) Determining surface area of total heat transfer in tube

Surface area of total heat transfer in tube was calculated using **Eqs. (9)** and **(10)**.

To calculate this variable, it is necessary to have data on the inner diameter of the tube, number of passes, and number of tubes taken from **Table 9**.

$$a_t = N_t \frac{a'_t}{n} \tag{9}$$

To calculate a'_t, **Eq. (10(** was substituted in **Eq. (9)** as below.

$$a_t = N_t \frac{\frac{\pi}{4} \times (D_{i,t})^2}{n}$$

$$a_t = 26 \frac{\frac{3.14}{4} \times (0.01656m)^2}{2} = 2.798 \times 10^{-3} m^2$$

(vii) Determining mass flow rate of fluid in tube

Mass flow rate of fluid in tube was calculated using **Eq. (11)**. Where, the m_{th} and a_t were taken from **Table 8** and **Eqs. (9-10)**, respectively.

$$Gt = \frac{m_{th}}{a_t} \tag{11}$$

$$Gt = \frac{1.5 \frac{kg}{s}}{2.798 \times 10^{-3} m^2} = 536.097 \frac{Kg}{m} s$$

(viii) Determining the reynolds number on the tube side

Reynolds number on the tube side was calculated using **Eq. (12)**. Where, the μ and di_t were taken from **Tables 8** and **9**. Meanwhile, Gt was calculated using **Eq. (11)**.

$$Re_t = \frac{di_t \times Gt}{\mu} \tag{12}$$

$$Re_t = \frac{0.01656 \text{ m} \times 536.097 \frac{Kg}{m} s}{6.16 \times 10^{-4} \frac{Kg}{m} s} = 14411.95$$

(ix) Determining the prandtl number of the tube side

The Prandtl number is used to determine the temperature distribution in each flow tube. Prandtl number of tube side was calculated using **Eq. (13)**. Where, the value of C_p, μ, and K were taken from **Table 8**.

$$Pr = (\frac{C_p \times \mu}{K})^{\frac{1}{2}} \tag{13}$$

$$Pr = (\frac{2161 \frac{J}{Kg}.K \times 6.16 \times 10^{-4} \frac{Kg}{m}.s}{0.138 \left(\frac{W}{m}.K\right)})^{\frac{1}{2}} = 9.646$$

(x) Determining nusselt number on tube side

Nusselt number on the tube side was calculated using **Eq. (14)**. Where, Re_t and Pr from tube side calaculation were calculated using **Eqs. (12)** and (**13)**, respectively.

$$Nu = 0.023 \times {Re_t}^{0.6} \times Pr^{0.33} \tag{14}$$

$$Nu = 0.023 \times 14411.95^{0.6} \times 9.646^{0.33} = 15.196$$

(xi) Determining heat transfer on the tube side

Heat transfer on the tube side was calculated using **Eq. (15)**. Where, the Nu, K, and d_i, t were obtained from **Eq. (13)**, **Tables 8** and **9**, respectively.

$$hi = \frac{Nu \times K}{d_i, t} \tag{15}$$

$$hi = \frac{15.196 \times 0.138 \left(\frac{W}{m}.K\right)}{0.01656 \, m} = 126.63 \frac{W}{m^2 K}$$

The next stage is the calculation of the shell side. Before calculating the shell side, some initial dimensional specifications can be used to simplify the calculations determined from the standards referenced from the TEMA standard. The shell side dimension values are shown in **Table 10**.

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Table 10. Shell dimensions.

Dimension	Standard Shell Design	International Standard
Inner diameter (di_s)	12 inchi	0.3048 m
Outer diameter (do_s)	12.787 inchi	0.3248 m
Baffle	0.315 inchi	0.008 m
Tube pitch (P_t)	15/16 inchi	0.0238 m
Clearance (C; Pt-do_t)	0.187 inchi	0.475

(xii) Determining the flow area in the shell

Flow area in the shell was calculated using Eq. 16. Where, d_s and B were obtained from **Table 10**. Where, C and P_t values were calculated using embedded equation in **Eq. (16)**.

$$A_s = \frac{d_s \times C \times B}{P_t} \quad (16)$$

$$A_s = \frac{30.48\times10^{-2} m \times 0.475\times10^{-2} m \times 0.06096\, m}{0.0238} = 3.7088 \times 10^{-3}\, m^2$$

(xiii) Determining mass flow velocity

Mass flow velocity was determined using **Eq. (17)**. Where, m_c was obtained from **Table 8** and A_s was calculated first using the **Eq. (16)**.

$$Gs = \frac{m_c}{A_s} \quad (17)$$

$$Gs = \frac{1.72}{3.7088\times10^{-3}} = 463.76\,\frac{Kg}{m^2}.s$$

(xiv) Determining the equivalent diameter of shell

Equivalent diameter of shell was calculated using **Eq. (18)**. Where, the $d_{o,t}$ and Pt were obtained from **Tables 9** and **10**, respectively.

$$d_e = \frac{4\left(\frac{Pt}{2}\times 0.87\, Pt - \frac{1}{2}\pi\frac{{d_{o,t}}^2}{4}\right)}{\frac{1}{2}\pi d_{o,t}} \quad (18)$$

$$d_e = \frac{4\left(\frac{0.0238}{2}\times 0.87\times 0.0238 - \frac{1}{2}\times 3.14\times\frac{0.01905^2}{4}\right)}{\frac{1}{2}\times 3.14\times 0.01905} = 0.0139\, m$$

(xv) Determining the reynolds number on the shell side

Reynolds number on the shell side was calculated using **Eq. (19)**. Where, μ was obtained from **Table 8** and Gs was calculated first using **Eq. (17)**.

$$Re_s = \frac{d_e \times Gs}{\mu} \quad (19)$$

$$Re_s = \frac{0.0139\, m \times 463.76\frac{Kg}{m^2}.s}{2.54\times10^{-3}\frac{Kg}{m}.s} = 2537.89$$

(xvi) Determining the prandtl number coefficient on the shell side

Prandtl number on the shell side was calculated using **Eq. (20)**. Where, the value of C_p, μ, and K were obtained from **Table 8**.

$$Pr = \left(\frac{C_p \times \mu}{K}\right)^{\frac{1}{2}} \quad (20)$$

$$Pr = \left(\frac{4174\frac{J}{Kg}.K \times 2.54\times10^{-3}\frac{Kg}{m}.s}{0.640\frac{W}{m}.K}\right)^{\frac{1}{2}} = 4.070$$

(xvii) Determining nusselt number on the shell side

Nusselt number on the shell side was calculated using **Eq. (21)**. Where, Re_t and Pr from shell side calaculation were calculated using **Eqs. (19)** and **(20)**, respectively.

$$Nu = 0.023 \times {Re_t}^{0.6} \times Pr^{0.33} \quad (21)$$

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$$Nu = 0.023 \times 2537.89^{0.6} \times 4.070^{0.33} = 4.032$$

(xviii) Determining heat transfer on the shell side

Heat transfer on the shell side was obtained from **Eq. (22)**. The d_e and *Nu* values were calculated first using **Eqs. (18)** and (**19)**. Meanwhile, K was obtained from **Table 8**.

$$hs = \frac{Nu \times K}{d_e} \tag{22}$$

$$hs = \frac{4.032 \times 0.640 \frac{W}{m}.K}{0.0139\, m} = 185.646 \frac{W}{m^2K}$$

(xix) Determining the hot fluid rate on tube side

Hot fluid rate for tube side was calculated using **Eq. (23)**. Where, the m_h and Cp_h for tube side were obtained from **Tabel 8**.

$$C_h = m_h.Cp_h \tag{23}$$

$$C_h = 1.5 \frac{kg}{s}.\frac{2161J}{Kg}.K = 3241\, W/K$$

(xx) Determining the cold fluid rate on the shell side

Hot fluid rate for tube side was calculated using **Eq. (24)**. Where, the m_h and Cp_h for shell side were obtained from **Tabel 8**.

$$C_C = m_c.Cp_c \tag{24}$$

$$C_C = 1.72 \frac{Kg}{s}.4174 \frac{J}{kg}.K = 7179.28\, W/K$$

(xxi) Determining the maximum heat transfer rate (Q_{max})

Maximum heat transfer rate was calculated using **Eq. (25)**. Where, $T_{hi,t}$ and $T_{ci,s}$ values were obtained from **Table 8**. Meanwhile, C_h was calculated using **Eq. (23)**.

$$Qmax = C_h(T_{hi,t} - T_{ci,s}) \tag{25}$$

$$Qmax = 3241 \frac{W}{K(380-300)} = 259{,}280\, W$$

(xxii) Determining the value of heat exchanger effectiveness (ε)

Heat exchanger apparatus effectiveness was calculated using **Eq. (26)**. Where, $Qact$ dan $Qmax$ values were calculated using **Eq. (1)**.

$$\varepsilon = \frac{Qact}{Qmax} \times 100\% \tag{26}$$

$$\varepsilon = \frac{129{,}660\, W}{259{,}280} \times 100\% = 50{,}01\%$$

(xxiii) Determining NTU value

The NTU value was calculated using **Eq. (27)**. Where, *U* value was obtained from literature. Meanwhile, *A* and C_{min} were calculated using **Eqs. (7)** and (**23)**.

$$NTU = \frac{U \times A}{C_{min}} \tag{27}$$

$$NTU = \frac{800 \times 3.43}{3241} = 0.85$$

(xxiv) Determining tube length

Tube length was calculated using **Eq. (28)**. Where, *U* value was obtained from literature. The $do_{,t}$ value obtained from **Table 8**. And also, Nt and C_{min} values were calculated using **Eqs. (8)** and **(23)**.

$$Lt = \frac{NTU \times C_{min}}{U \times \pi \times do_{,t} \times Nt \times 2} \tag{28}$$

$$Lt = \frac{0.85 \times 3241 \mathrm{W/K}}{800 \frac{W}{m^2} K \times 3.14 \times 0.01905\, m \times 26 \times 2} = 1.107\, m$$

In this study, the fluid used in the heat exchanger is water with a temperature of 27°C to reduce the oil temperature to 107°C so that the calculation result of the heat transfer coefficient that occurs on the tube side is 126.63 $\frac{W}{m^2K}$ with a Reynolds value of 14409.11. The

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Reynolds value for flow on the tube side shows a value greater than 2300 which indicates that the flow on the tube side is turbulent. Then, the water flowing inside the shell has a heat transfer coefficient of $185.646\,\frac{W}{m^2K}$ with a Reynolds value greater than 2300, namely 2537.89, which indicates that the fluid flow on the shell side is also turbulent. The heat transfer value of oil is quite large because high oil temperatures cause large heat transfer. Meanwhile, the maximum heat transfer rate value from the design is 259,280 W. Meanwhile, the actual heat transfer rate value from the design is 129,660 W, thus the effectiveness of the apparatus obtained based on calculations of the effectiveness of the heat exchanger and NTU is 50.01% and 0.85 respectively.

5. CONCLUSION

Calculation results in the case example of a shell and tube type heat exchanger design using copper tubes and aluminum shell for a triangular pitch shape, with a capacity of 129,660 W, showed that the number of tubes was 26 and tube length was 1.107 m with the effectiveness of the designed device being 50.01%.

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