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Parametric analysis of shell and tube heat exchanger for a simple cycle gas turbine power plant

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Abstract:
Optimising the performance of shell and tube heat exchangers in simple cycle gas turbine power plant can lead to improved efficiency and reduced operating costs. The aim of this study is to conduct a parametric analysis of a shell and tube heat exchanger for a simple cycle gas turbine power plant. The objectives are to evaluate the effects of various design parameters, such as shell diameter, baffle spacing, tube bundle diameter, and mass flow rate on the performance of the heat exchanger. The results of the analysis show that increasing the shell diameter, baffle spacing, and mass flow rate leads to an increase in the heat transfer coefficient and pressure drop. The overall heat transfer coefficient is found to be 16.00 W/m^{2}K, while the pressure drops on the shell and tube sides are found to be 0.22 kPa and 5.34 kPa respectively. The results indicate the effectiveness of heat transfer between the hot and cold fluids, a moderate heat transfer performance and relatively low and moderate shell and tube sides pressure drop. The study provides valuable insights into the design and optimization of shell and tube heat exchangers for gas turbine power plants for increased power output production and general overall cycle performance.

Keywords: *Gas turbine, Heat exchanger, Performance, Power plant, Pressure drop*

1. Introduction

Shell and tube heat exchanger is a type of heat exchanger that consists of a shell with a series of tubes inside it. It is one of the most common types of heat exchangers widely used in the industrial processes. It allows flow of fluid through the tubes, while another fluid flows over the tubes, allowing heat transfer from one fluid to the other. Ease of manufacture and layout of flow configurations are some of the important features of shell and tube heat exchanger over other types in the process industry for waste heat recovery and other applications [1]. There have been several studies on parametric analysis of shell and tube heat exchangers for a simple cycle gas turbine power plant. However, more investigations need to be conducted especially on the Siemens gas turbine model SGT5-2000E located in Geregu power plant Kogi State, Nigeria for optimization and efficiency.

Patel [2] presented paper on advancements in heat exchanger design for waste heat recovery in industrial processes. Ali and Zaki [3] analyze study of shell and tube heat exchanger for clough company with different parameters to improve the design. Ali, *et al.* [1] carried out performance analysis of shell and tube heat exchanger, considering the parametric study to investigate effect of shell diameter and tube length on heat transfer coefficient and pressure drop for shell side with both triangular and square pitches.

Chen, *et al.* [4] presented paper on shell and tube heat exchanger flexible design strategy for process operability and considered flexibility index in the optimization procedure can improve the STHE design operability under the expected range of disturbance factors and lower the total cost. Natesan and Karinka [5] conducted comprehensive review of heat transfer enhancement of heat exchanger, heat pipe and electronic components using graphene, and found out various applications of graphene in different types of heat exchangers and electronic devices. Pontevedra [6] studied the shell and tube heat exchanger with the effect of types of baffles and discovered that helical baffles have advantages over the other ones. Silaipillayarputhur and Khurshid [7] reviewed the design of shell and tube heat exchangers and used some analytical techniques such as log mean temperature difference (LMTD) and effectiveness-number of transfer units (ε-NTU) which gave same results. Franti and Martin [8] presented paper on shell and tube heat exchanger considering the heat transfer area design process using various design calculations. Perumal, *et al.* [9] presented heat transfer analysis in counter flow shell and tube heat exchanger using design of experiments via Taguchi L9 orthogonal array showing baffle plate thickness to be highly significant factors amongst three parameters considered for the experiment. Mulyana, *et al.* [10] identified model nonlinear of the four heat exchanger types using mathematical models of dynamic system to measure the input and output signal of the real system.

Baina, *et al.* [11] analyzed a high-temperature heat exchanger for a micro gas turbine MATLAB and Aspen Plus, finding that the effectiveness of a corrugated plate heat exchanger is more affected by deposit material thickness than a shell and tube heat exchanger. Pina, *et al.* [12] presented paper on the heat exchanger design and optimization by using generic algorithm for externally fired micro-turbine. MATLAB and FRONTIER were used to build code and optimization through multi-objective maximizes the overall heat transfer coefficient and minimizes both costs and pressure drops across the equipment. Wang, *et al.* [13] investigated the effects of shell diameter on heat transfer performance in the shell-and-tube heat exchangers. Computational Fluid dynamics (CFD) software was used to simulate the flow and heat transfer in the heat exchanger. Results from the study show that the heat transfer coefficient increases with increasing shell diameter, the pressure drop decreases with increasing shell diameter and the overall heat transfer performance is optimized at a shell diameter of 0.3 to 0.4 m [13]. Mishra and Das [14] investigated the effect of mass flow rate on the performance of a shell and tube heat exchanger using numerical model for simulating the heat transfer and flow of fluid in the heat exchanger. The simulated results show that increasing the mass flow rate enhances the heat transfer coefficient, but also increases the pressure drop [14].

Nader, *et al.* [15] presented paper on performance of a high-temperature particle-based shell and tube crossflow heat exchanger suitable for CSP power generation application. Experiment was performed through thermal evaluation of a pilot 50 kW moving packed-bed particle-to-air heat exchanger. The results obtained from the work show high values of the overall heat transfer coefficient (up to ~120 W/m^2°C) in accordance with shell-and-tube indirect contact heat exchangers [15]. Kara [16] presented a computer program for designing shell-and-tube heat exchangers. Computer codes for design are organized to vary systematically the exchanger parameters such as, shell diameter, baffle spacing, number of tube-side pass to identify configurations that satisfy the specified heat transfer and pressure drops [16]. Hadidi, *et al.* [17] presented a new design approach for shell-and-tube heat exchangers using imperialist competitive algorithm (ICA) from economic point of view. Chukwudi and Ogunedo [19] presented study on design and construction of a shell and tube heat exchanger for laboratory use. Bell-Delaware method was used for the study and results obtained show heat load was 107.973 kW, temperature changes for cold fluid was +10 °C, hot fluid was -55 °C (to 45 °C), pressure drops within allowable range, overall heat transfer coefficient was 134.23 W/m²K and system efficiency was 73.3%. Thet, *et al.* [20] designed a shell and tube heat exchanger utilizing MATLAB and AutoCAD software for calculations and found out that computer program was useful for designing and modifying shell-and-tube heat exchangers.

The study covers parametric analysis of shell and tube heat exchanger for a simple cycle gas turbine power plant. The heat exchanger parameters considered include tube diameter, shell diameter, number of tubes, and baffle spacing on heat transfer coefficient and pressure drop for shell and tube sides of the heat exchanger.

2. Methodology

The input data for the study was based on assumptions. However, the hot fluid inlet temperature from the gas turbine exhaust waste was based on the exhaust outlet temperature of SGT5-2000E gas turbine model located in Geregu gas turbine power plant in Kogi State. Similarly, the Figure 1 shows the shell and tube heat exchange considered for the study and, various mechanical and thermal design methods were utilized using relevant equations as follows.

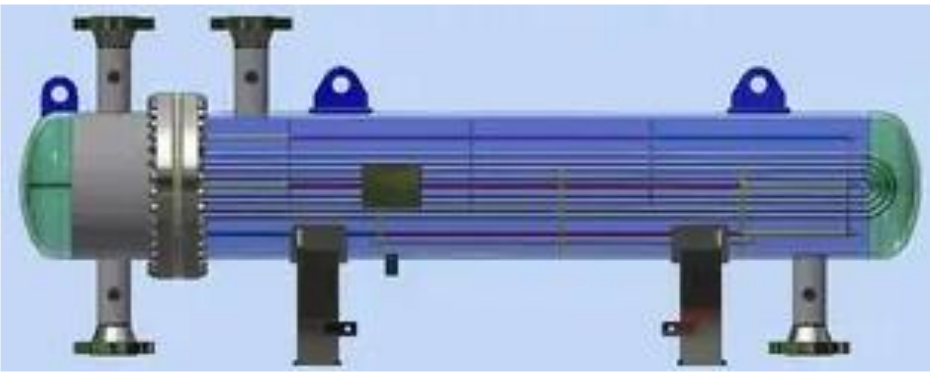

Figure 1: Shell and tube heat exchanger MechStudies [21]

2.1 Mathematical models

2.1.1 Mechanical parameters design

The mechanical parameters used in the design of the heat exchanger include:

Tube Pitch (t_p) [19]

$$t_p = 1.25\, d_o \tag{1}$$

Baffle spacing (B) [19]

$$B = 74 {d_o}^{0.75} \tag{2}$$

Number of Tube (N_t) [22]

$$N_t = \frac{CTP\,\pi\, Ds{,}i^{\;2}}{4\,A_1} \tag{3}$$

Where: N_t = number of tubes, CTP = tube counting factor (assume 0.87 for 60° triangular layout) [23], $D_{s,i}$ = shell inside diameter (m), A_1 = cross sectional area (m²)

Tube surface area (A_o) [19]

$$A_o = \pi\, d_o\, N_t\, L \tag{4}$$

Where A_o = tube surface area (m²), d_o = tube outer diameter (m), N_t = number of tubes, L = tube length (m)

Cross-sectional area of shell (m²) was calculated using equation 5 as reported in ASME [24].

$$A_s = (\pi\, x\, D_{s,i} \times B) - (N_t\, x\, \pi\, \frac{do^{\,2}}{4}) \tag{5}$$

Where A_s = cross-sectional area of shell (m²), $D_{s,i}$ = shell inside diameter (m), B = baffle spacing (m), N_t = number of tubes, d_o = tube outside diameter (m) and N_t = number of tube.

Shell outside diameter ($D_{s,o}$) was calculated using equation 6 as reported in Chukwudi and Ogunedo [19].

$$D_{s,o} = D_b + SBC \tag{6}$$

Where $D_{s,o}$ = shell outside diameter (m), D_b = bundle equivalent diameter (m) and SBC = shell bundle clearance

$$D_b = d_o \left(\frac{N_t}{K_1}\right)^{\frac{1}{n_1}} \tag{7}$$

Where n_1 = 2.141 and K_1 = 0.319 according to Chukwudi and Ogunedo [19]

Shell bundle clearance was calculated using equation 8

$$SBC = \frac{(D_{s,i} - d_o\, tb)}{2} \tag{8}$$

Where $d_o\, tb$ = tube bundle outer diameter (m) which was calculated using equation 9.

$$d_o\, tb = \frac{d_o + 2\,\text{x}\,(\text{S} - d_o)}{2} \tag{9}$$

Where S = tube pitch (t_p)

2.2 Thermodynamic design parameters

2.2.1 Overall heat transfer coefficient (U_o)

The overall heat transfer coefficient is the function of all the fouling resistance, individual heat coefficient and surface efficiency of the tubes. The fouling resistance are caused do to dirt's build-up on the heat exchanger surfaces and the equation to determine overall heat transfer coefficient is given in equation 10.

$$\frac{1}{U} = \frac{1}{\frac{1}{h_t} + R_{di} + R_w + R_{do} + \frac{1}{h_s}} \tag{10}$$

Where U = overall heat transfer coefficient (W/m²K), h_t = tube-side heat transfer coefficient (W/m²K), h_i = inside fouling resistance = 0.0001[19] (m²K/W), h_w = wall resistance (negligible because thin tube wall was considered) (m²K/W), h_o = outside fouling resistance = 0.0002 [19] (m²K/W), h_s = shell-side heat transfer coefficient (W/m²K).

2.2.2 Heat transfer rate (q)

Heat transfer rate in the heat exchanger according to [12] is given as shown in equation 11:

$$\text{q} = m_h c_{p,h}(T_{h,i} - T_{h,o}) = m_c c_{p,c}(T_{c,i} - T_{c,o}) \tag{11}$$

Where m_h = mass flow of hot fluid, $c_{p,h}$= specific heat capacity of hot fluid, $T_{h,i}$ = Temperature inlet of hot fluid, $T_{h,o}$ = Temperature outlet of hot fluid, m_c = mass flow of cold fluid, $c_{p,c}$ = specific heat capacity of cold fluid, $T_{c,i}$ = Temperature inlet of cold fluid and $T_{c,o}$ = Temperature outlet of cold fluid.

2.2.3 Energy balance in the heat exchanger

Heat loss by hot fluid = Heat gain by cold fluid which was calculated using equation 12.

$$\text{q} = m_h c_{p,h}(T_{h,i} - T_{h,o}) \tag{12}$$

2.2.4 Heat transfer area (A)

The heat transfer area (A) was determined using equation 13 as reported in [25]

$$A = \frac{Q}{U\Delta T_{LMTD}} \tag{13}$$

Where A = heat transfer area (m²), Q = heat transfer rate (kW), U = overall heat transfer coefficient (W/m²K) and ΔT_{LMTD} = logarithm mean temperature difference (K)

Logarithm mean temperature difference (ΔT_{LMTD}) was calculated using equation 14

$$\Delta T_{LMTD} = \frac{(T_{h,i} - T_{c,o}) - (T_{h,o} - T_{c,i})}{ln(\frac{T_{h,i} - T_{c,o}}{T_{h,o} - T_{c,i}})} \tag{14}$$

The mass flow rate of liquid can also be obtained from equation 15 to 17

$$\text{q} = m_h c_{p,h}(T_{h,i} \text{-} T_{h,o}) = m_c c_{p,c}(T_{c,i} \text{-} T_{c,o}) \tag{15}$$

$$\text{q} = m_h c_{p,h}(T_{h,i} \text{-} T_{h,o}) = m_l(h_{c,o} \text{-} h_{c,i}) \tag{16}$$

$$m_l = \frac{m_h c_{p,h}(T_{h,i} - T_{h,o})}{(h_{c,o} - h_{c,i})} \tag{17}$$

2.2.5 Tube-side heat transfer coefficient

The tube-side Reynolds number depends on fluid properties such as fluid velocity, viscosity and characteristics diameter. Therefore, to calculate the tube–side heat transfer coefficient of the shell and tube heat exchanger, Dittus-Boelter correlation was used as shown in equation 18:

$$h_t = 0.023 \times (\frac{\text{K}}{\text{d}_o}) \times Re_t^{\ 0.8} \times Pr_t^{\ 0.3} \tag{18}$$

Where K = thermal conductivity of tube side fluid (W/m²K), d_o = diameter of cold fluid (m), Re_t = tube-side Reynolds number (dimensionless) and Pr_t = tube-side Prandtl number (dimensionless)

2.2.6 Shell-side heat transfer coefficient (h_s)

The shell-side heat transfer coefficient (h_s) for the heat exchanger can be calculated using the Bell-Delaware method in equation 19.

$$h_s = \frac{\mathrm{K}}{\mathrm{D}_h} \times 0.36 \times {Re_s}^{0.55} \times {Pr_s}^{0.333} \quad (19)$$

Where K = thermal conductivity, D_h = hydraulic diameter of shell (m), Re_s = Reynolds number of shell (dimensionless) and Pr_s = Prandtl number of shell (dimensionless)

2.2.7 Tube side pressure drop (ΔPt)

The tube side pressure drop calculated using equation 20 as reported in [19]

$$\Delta P_t = f_t \, x \, (\frac{L}{d}) \, x \, (\rho \, x \, \frac{V^2}{2}) \quad (20)$$

Where, ΔP_t= tube side pressure drop (kPa), f_t = tube side friction factor (dimensionless), L = tube length (m), d = tube inner diameter (m), ρ = density of fluid (kg/m^3) and V = fluid velocity (m/s)

Friction factor f_t is a dimensionless quantity which represent ratio of shear stress at the tube wall to kinetic energy of the fluid per unit volume. It is therefore the fluid resistance to flow due to friction between fluid and the tube. Friction factor is calculated using the equation 21as reported in [26].

$$f_t = 0.023 \, x \, {Re_t}^{-0.2} \quad (21)$$

2.2.8 Shell side pressure drop (ΔPs)

The shell side pressure drop was calculated using equation 22 as reported in [19]

$$\Delta P_s = f_s \times (\frac{D_s}{D_e}) \times (\rho \times \frac{{V_s}^2}{2}) \quad (22)$$

Where, ΔP_s = shell side pressure drop (Pa), f_s = shell side friction factor (dimensionless), D_s = shell diameter (m), D_e = equivalent or hydraulic diameter (D_h) (m), ρ = density of fluid (kg/m^3) and V_s = shell side fluid velocity (m/s)

The design data of the SGT5-2000E gas turbine are shown in Table 1

3. Results and Discussion

3.1 Assumptions and basic parameters

The Table 2 outlines the assumptions made for the heat exchanger analysis. They include fluid properties such as flue gas and water inlet temperatures, enthalpy values, materials and diameter of the shell and tube side respectively. Similarly, the Table 3 presents the heat exchanger basic parameters, which include mass flow rates, velocities, heat transfer rates, geometric parameters, heat transfer coefficient and pressure drops.

Table 1: Design data of SGT5-2000E gas turbine

S/No	Parameter	Value	Unit
1	Ambient temperature	288	K
2	Compressor outlet temperature	623	K
3	Turbine inlet temperature	1333	K
4	Exhaust temperature	813	K
7	Compressor pressure ratio	11:1	
8	Turbine pressure ratio	1:11	
9	Power output	145	MW
10	Mass flow of air	500	kg/s
11	Mass flow of gas	8	kg/s
12	Specific heat ratio	1.4	
13	Specific heat capacity of air	1.005	kJ/kgK
14	Specific heat capacity of air	1.14	kJ/kgK
15	LHV of NG	47,976.5[27]	kJ/kg

Table 2: Assumptions for heat exchanger analysis

Parameters	Shell side	Tube side	Units
Fluid	Flue gas	Water	--
Inlet temperature	813	288	K
Outlet temperature	373	423	K
Enthalpy of fluid at temperature of 423 K	--	2776.2	kJ/kg
Enthalpy of fluid at temperature of 373 K	417.46	--	kJ/kg
Material	ASTM A387 Gr. 22 (2.25Cr-1Mo)	ASTM A213 Gr. T22 (2.25Cr-1Mo)	--
Diameter	0.3	0.02	m

The Table 2 presents a comprehensive analysis of the effects of varying different parameters on heat transfer and pressure drop in the heat exchanger. The table examines the impact of varying tube side and shell side parameters on the heat transfer and pressure drop. The varied parameters are tube inside diameter, tube length, tube side mass flow rate, tube side pressure drop, baffle spacing, tube bundle diameter, shell inside diameter and shell side pressure drop respectively. The data indicates that increasing the shell inside diameter, baffle spacing, and mass flow rate results in an increase in the heat transfer coefficient and pressure drop.

3.2 Effect of design parameters on performance

There is a significant impact of various design parameters on the performance of a shell and tube heat exchanger as can be seen in Figure 1 to 7. As shown in Figure 1, increasing the shell diameter leads to a higher heat transfer rate, with a linear relationship observed between the two parameters.

Table 3: Heat exchanger basic parameters

Parameters	Values	Units
Heat transfer surface area	0.60	m^2
Tube length	3.5	m
Tube diameter	0.02	m
Number of tubes	130	
Shell diameter	0.3	m
Tube layout	Triangular	
Tube pitch	0.025	m
Baffle spacing, (B)	3.9	m
Tube surface area, (A_o)	28.59	m^2
Cross-sectional area of shell (m^2)	3.64	m^2
Shell outside diameter (D_(s,o))	0.35	m
Overall heat transfer coefficient (U_o)	16.00	W/m^{2}K
Tube-side heat transfer coefficient (h_t)	1986	W/m^{2}K
Shell-side heat transfer coefficient (h_s)	24.28	W/m^{2}K
Tube side pressure drop (ΔPt)	5.34	kPa
Shell side pressure drop (ΔPs)	0.22	kPa

Table 4: Parametric analysis of heat exchanger performance

q (kW)	$\dot{m}_t$ (kg/s)	ΔP_t (kPa)	d_{tb} (m)	L (m)	D (m)	ΔP_s (kPa)	U_o (W/m^{2}K)	B (m)
3600	1.4	4.34	0.013	3	0.28	0.19	18.92	3.5
3800	1.5	4.84	0.014	3.2	0.29	0.2	20.92	3.6
4012.8	1.7	5.34	0.015	3.5	0.3	0.21	22.92	3.9
4225.6	1.9	5.84	0.016	3.8	0.31	0.22	23.92	4.2
4438.4	2.1	6.34	0.017	4.1	0.32	0.23	24.92	4.5
4646.08	2.26	6.84	0.018	4.36	0.33	0.24	26.82	4.72
4856.32	2.44	7.34	0.019	4.64	0.34	0.25	28.32	4.98
5066.56	2.62	7.84	0.02	4.92	0.35	0.26	29.82	5.24
5276.8	2.8	8.34	0.021	5.2	0.36	0.27	31.32	5.5
5487.04	2.98	8.84	0.022	5.48	0.37	0.28	32.82	5.76

It can be observed from Figure 1 that as the shell diameter increases, the heat transfer rate also increases. The linear relationships of the two parameters suggest that increasing the shell diameter can lead to a higher heat transfer rate which can be beneficial in certain applications. The value of heat transfer rate in the heat exchanger is 4012.8 kW with the heat exchanger shell diameter of 0.3 m. However, it is very essential to consider other factors such as pressure drop, fluid flow, and heat exchanger design to ensure its performance is optimal. This result is comparable to the reported work of Wang, *et al.* [13], numerical study to investigate the effects of shell diameter on heat transfer performance in the shell-and-tube heat exchangers. However, this increase in shell diameter also results in a higher pressure drop on the shell side, as depicted in Figure 2.

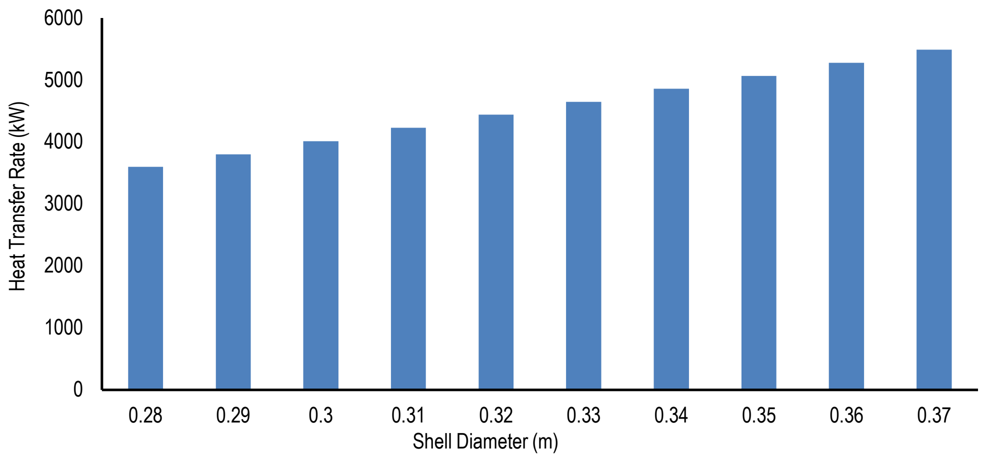

Figure 1: Effect of shell diameter on heat transfer rate

Increase in the length of heat exchanger lead to increase in pressure drop on the shell side. This is because the shell side fluid has to travel a longer distance resulting in more frictional losses and a higher pressure drop. Also, a longer heat exchanger means more resistance to fluid flow resulting in higher pressure drop, which lead to reduced overall efficiency of the heat exchanger. The value of pressure drop on the shell side is 0.22 kPa while the heat exchange has tube length of 3.5 m respectively. The discussion on the effect of length on shell-side pressure drop is consistent with the studies of Wang, *et al.* [13]. The tube length is also found to have a significant impact on the pressure drop on both the shell and tube sides, with longer tubes resulting in higher pressure drops as shown in Figure 2 and Figure 3 respectively.

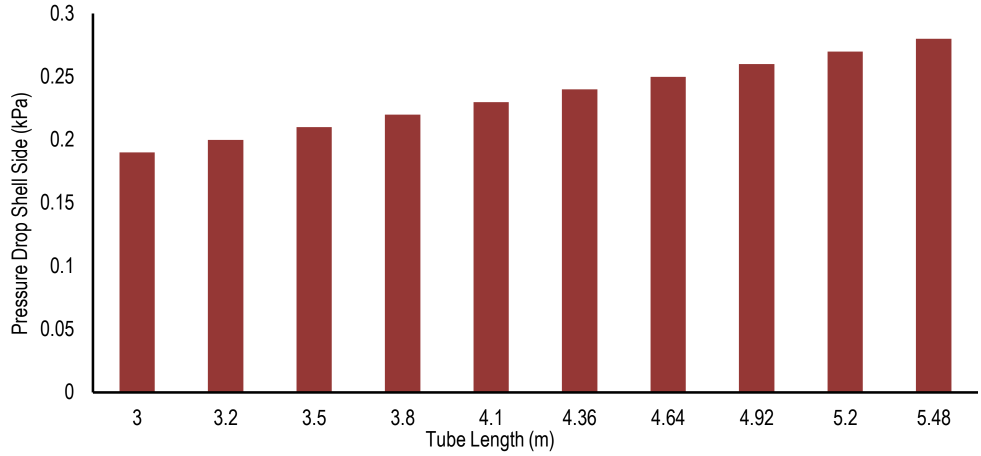

Figure 2: Effect of tube length on pressure drop on shell side

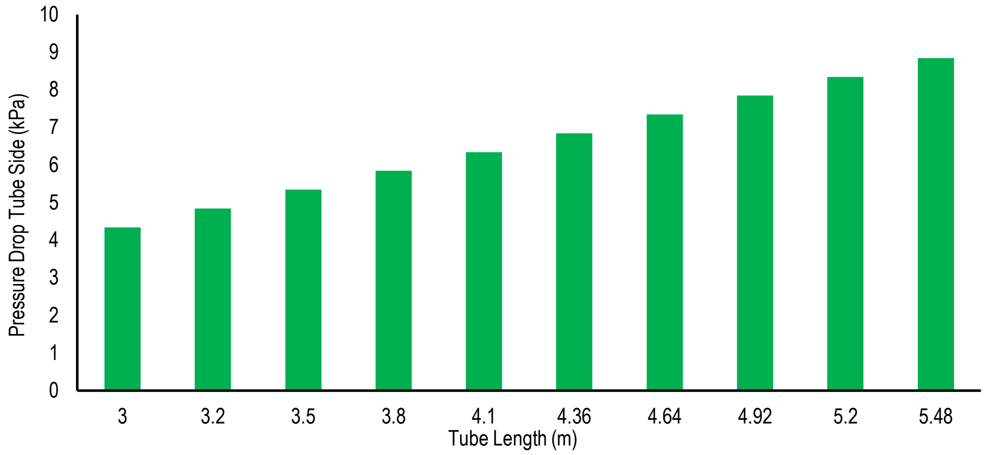

Figure 3: Effect of tube length on pressure drop on tube side

It can be spotted from Figure 3 that increase in the length of the heat exchanger leads to pressure drop on the tube side. The reason for this is that the fluid has to travel a longer distance as in the case with the shell side resulting in more frictional losses and a higher pressure drop on tube side. The heat exchanger tube length is 3.5 m and the pressure drop on the tube side is 5.34 kPa. The scenario is same with that of Figure 2. Conversely, the overall heat transfer coefficient is found to increase with increasing tube length, as shown in Figure 4.

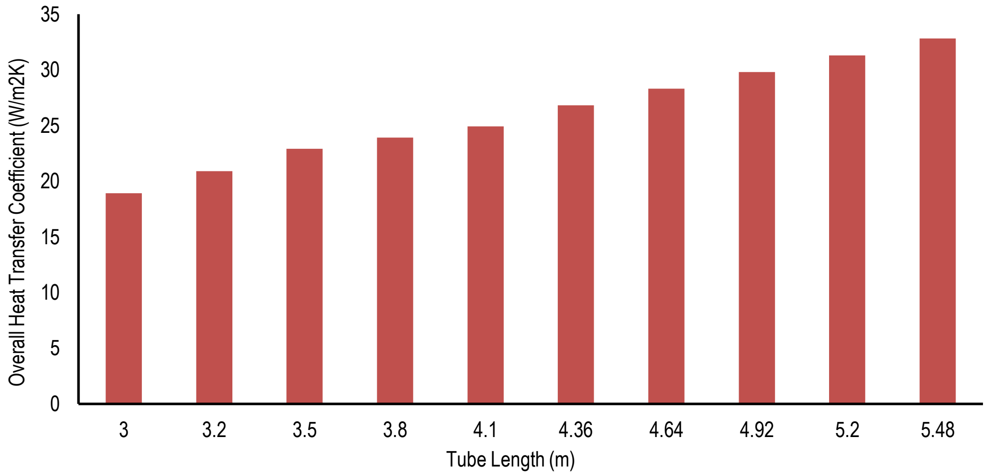

Figure 4: Effect of tube length on overall heat transfer coefficient

It can be observed from Figure 4 that the length of the heat exchanger has an impact on the overall heat transfer coefficient of the heat exchanger. As the length of the heat exchanger increases, the heat transfer area also increases thereby causing higher overall heat transfer coefficient. The length of the heat exchange tube is 3.5 m while the overall heat transfer coefficient is 16.00 W/m^{2}K. Similarly, there is an improved heat transfer performance, as a longer heat exchanger provides more opportunities for heat transfer between the two fluids, resulting in a higher overall heat transfer coefficient. The decreasing effect of tube length on overall heat transfer coefficient caused decrease in heat transfer area and reduced heat transfer performance. The compared literature to validate this result is same with the one referenced earlier Wang, *et al.* [13].

The mass flow rate on the shell side is also observed to have a significant impact on the pressure drop, with higher flow rates resulting in higher pressure drops as shown in Figure 5.

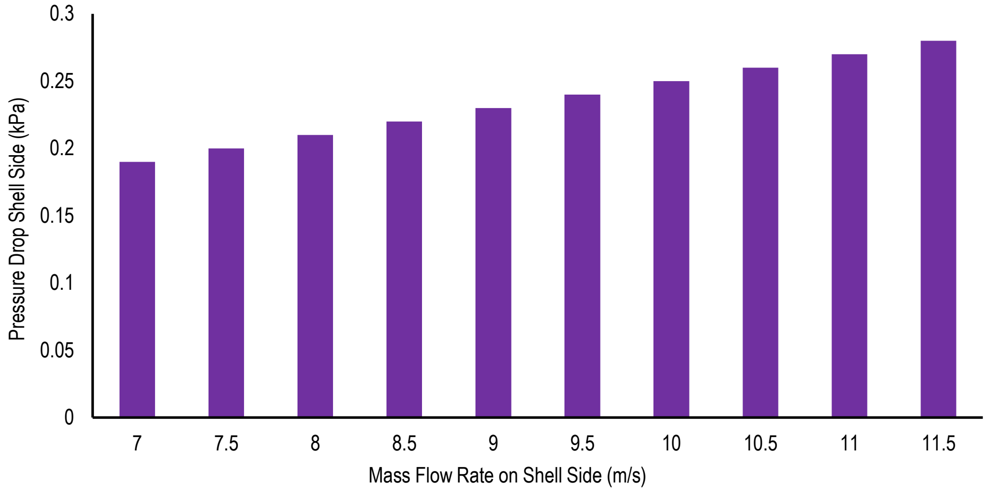

Figure 5: Effect of mass flow rate on pressure drop shell side

It can be identified from Figure 5 that as the mass flow rate of the shell side fluid increases, the pressure drop on the shell side also increases. This is because the higher flow rate results in more frictional losses and a higher pressure drop. A higher mass flow rate also means a higher fluid velocity, which leads to more turbulence and a higher pressure drop. The increases pressure drop results in higher energy losses, which can lead to reduction in overall efficiency of the heat exchanger. The value of the mass flow rate on the shell side is 8 kg/s while the corresponding pressure drop on shell side is 0.22 kPa. Similarly, effect of decreasing mass flow rate on pressure drop include decrease in pressure drop on shell side as lower flow rate results in less frictional losses, lower fluid velocity which leads to less turbulence and lower pressure drop, suggesting improved overall efficiency of the heat exchanger. The present results are in line with the findings reported by Mishra and Das [14]. Conclusively, mass flow rate of the shell side fluid has significant impact on the shell side pressure drop as increasing the mass flow rate increases the pressure drop, while decreases the mass flow rate reduces the pressure drop. Furthermore, the tube bundle diameter is found to have a direct relationship with the shell inside diameter, with larger tube bundles requiring larger shell diameters as shown in Figure 6.

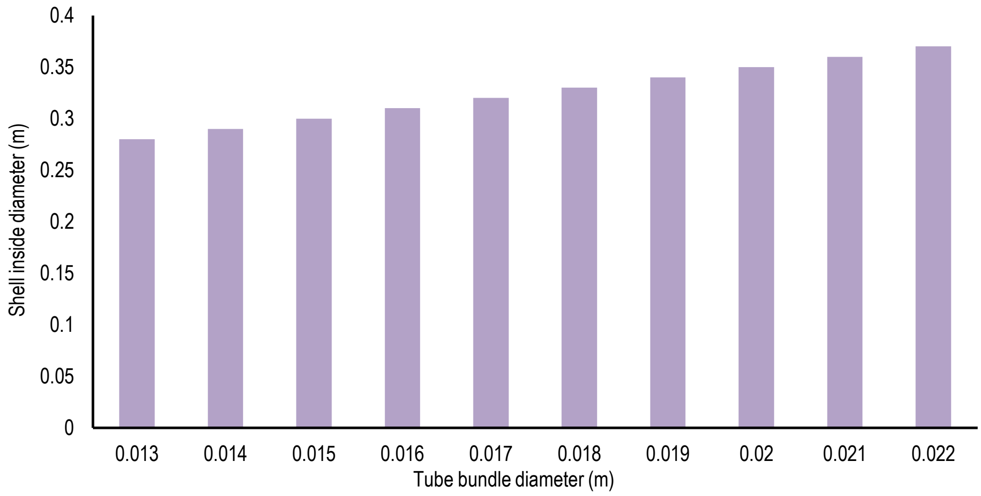

Figure 6: Effect of tube bundle diameter on shell inside diameter

It can be seen from Figure 6 that as the tube bundle diameter increases, the shell diameter also increases in other to accommodate the larger tube bundle. A larger tube bundle diameter allows for more tubes to be packed within the shell, thereby enhancing heat transfer performance. The increase in shell diameter also increases the shell volume, which can lead to higher costs and larger space requirements. While the analysed tube bundle diameter is 0.015 m, the shell inside diameter is 0.3 m respectively. Similarly, as the bundle diameter decreases, the shell diameter also decreases, resulting in more compact design. Fewer tubes can be packed within the shell due to smaller tube bundle diameter thereby compromising heat transfer performance and reduction in shell volume results in a smaller shell volume leading to lower costs and reduced space requirements. Finally, the shell inside diameter is observed to have a significant impact on the overall heat transfer coefficient, with larger diameters resulting in higher coefficients shown in Figure 7.

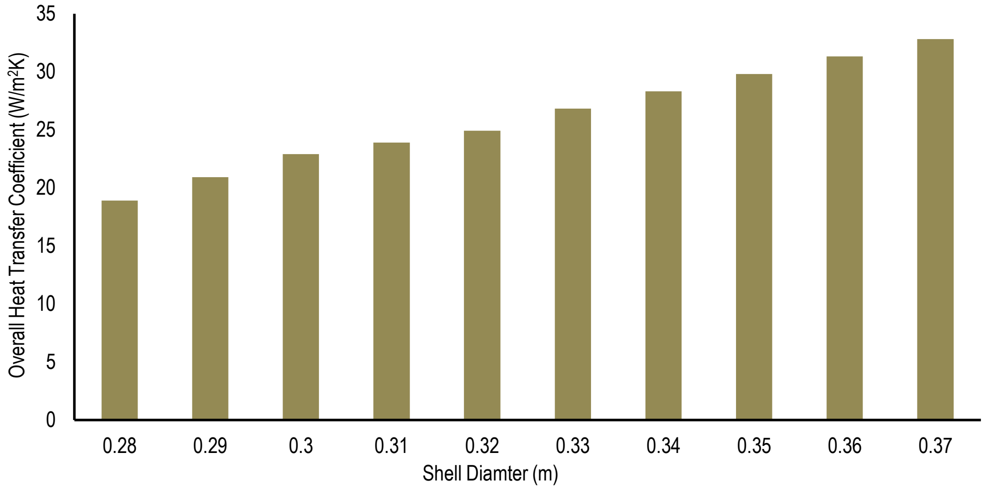

Figure 7: Effect of shell diameter on overall heat transfer coefficient

It can be seen from Figure 7 that as the shell diameter increases, the heat transfer area also increases, resulting in higher overall heat transfer coefficient. A larger shell diameter provides more space for fluid to flow in other to reduce pressure drops and improving heat transfer performance. Turbulence can also be enhanced as a result of increase shell diameter thereby improving heat transfer by increasing convective heat transfer coefficients. The calculated overall heat transfer coefficient is 16.00 W/m^{2}K while the shell diameter is 0.3 m. Similarly, as the shell diameter decreases, the heat transfer area also decreases leading to a lower overall heat transfer coefficient. A smaller diameter provides less space for flowing of fluid, increasing pressure drops and compromising heat transfer performance. Turbulence can also be reduced due to decrease in shell diameter thereby compromising heat transfer by decreasing convective heat transfer coefficient. The discussion on the effect of shell diameter on the overall heat transfer coefficient is consistent with result reported in Mishra and Das [14]. All these findings highlighted the complex interplay between various design parameters and the performance of a shell and tube heat exchanger.

4. Conclusion

The parametric analysis of shell and tube heat exchanger was conducted to investigate the effects of various parameters on heat transfer and pressure drop. The results showed that increasing the shell diameter, baffle spacing, and mass flow rate result in an increase in the heat transfer coefficient and pressure drop. The overall heat transfer coefficient was found to be 16.00 W/m^{2}K, and the pressure drop on the shell side and tube side were found to be 0.22 kPa and 5.34 kPa, respectively. The study provides valuable insights into the design and optimization of shell and tube heat exchangers.

Recommendation

It is therefore recommended from the research that further studies should be conducted to optimize shell diameter, adjust baffle spacing and also to investigate the effects of other parameters, such as tube material, tube arrangement, and operating conditions, on the heat exchanger performance.

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