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Computer-based Program for Analyzing the Thermo-hydraulic Design of a Shell and Tube Heat Exchanger: Parametric Study

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ABSTRACT.

A preliminary design of a shell and tube heat exchanger (STHE) was conducted using a computer code developed in Engineering Equation Solver (EES). This code, based on the Kern method, systematically varied exchanger parameters, specifically the tube length, to identify configurations that met specified heat transfer coefficients and pressure drops within the Tubular Exchanger Manufacturers Association (TEMA) standards. The program considered shell-and-tube side flow heat exchangers with one shell pass and two tube passes, featuring segmental baffles and fixed tube sheets, with single-phase fluid flow on both sides. It was used to compute the overall dimensions of the design, ensuring conformance with TEMA requirements regarding allowable pressure drop and heat transfer coefficient. To enhance the fidelity of the work, the mathematical model was validated against literature. The study considered five different tube lengths, ranging from 6 feet to 14 feet. Observations indicated that the 14-foot tube length met all the criteria prescribed in the TEMA standard and was therefore selected as the core dimension for the preliminary design of the current STHE. Finally, a parametric study capturing the effect of tube length on the heat transfer coefficient and pressure drop was also performed.

Keyword: Baffle spacing, friction factor, heat transfer coefficient, pressure drop, tube bundle.

I. INTRODUCTION

Shell-and-tube heat exchangers (STHEs) are widely used in industries such as oil and gas, refineries, food preservation electric power generation etc. This is due to their ease of maintenance, high heat transfer capability, and potential for upgrades, and they account for approximately 35-40% of heat exchangers used in global heat transfer processes [1]. Over the years, these types of heat exchangers have been developed and several improved configurations have been proposed for increasing heat transfer capability while maintaining the allowable pressure [2, 3].

The design of shell and tube heat exchangers can involve several challenges and potential problems that engineers need to address. Among the problems encountered in the design of the shell and tube heat exchanger are the low heat transfer coefficient and high-pressure drop. Low heat transfer coefficient can occur due to improper selection of tube size, length, or configuration, resulting in reduced performance and inefficient heat exchange. Excessive pressure drop can lead to increased energy consumption and decreased overall system efficiency. This can happen if the flow path is not optimized or if the design variables are not appropriately selected. To overcome these challenges, it is pertinent to conduct a parametric study using computer-based tools to optimize the design. Analyzing different design variables such as baffle spacing, cut, or number of passes, tube diameter, length, and layout to maximize the heat transfer coefficient, pressure drop, and overall performance.

A heat exchanger's design is an iterative process that heavily relies on the designer's prior knowledge and expertise. Typically, a reference geometric configuration of the equipment is selected first, and a maximum allowable pressure drop value is set. Then the values of the design variables are then determined based on the design specifications. Assumptions of several mechanical and thermodynamic parameters are made in order to achieve a satisfactory heat transfer coefficient that results in a suitable utilization of the heat exchange surface for the particular application. The designer's selections are then validated iteratively through numerous trials until a reasonable design is obtained that meets design specifications while maintaining a satisfying balance between pressure drops and thermal exchange performance [4, 5] . Various parametric and optimization studies have been reported in the literature using different programming techniques to design shell and tube heat exchangers [3-5].

Mizutani et al. [6] in their work used mathematical programming technique to design and optimized the design of their STHE adopting the Bell–Delaware method for the calculation of shell-side pressure drop and heat transfer coefficient. Kara and Gũraras [7] created a computer based design for the purpose of modelling a shell and tube heat exchangers with single-phase fluid flow on both the shell and tube sides. By computing the lowest or permissible shell-side pressure drop, the program calculates the overall dimensions of the shell, the tube bundle, and the best heat transfer surface area required to satisfy the specified heat transfer duty. It is better to place the stream with lower mass flow rate on the shell side due to the baffled space. They also concluded that the circulating cold fluid in the shell-side has some advantages over hot fluid as shell stream because the former causes lower shell-side pressure-drop and requires smaller heat transfer area than the latter. Bhatt and Javhar [8] conducted an analysis on the performance of heat exchangers, and it was noted that various results can be obtained by altering the value of one variable while maintaining the constant value of the other variables, thus improving the design of the shell and tube-type heat exchanger based on that outcome.

The heat transfer rate also increases as the tube metallurgy's thermal conductivity increases while less baffle spacing and more shell side passes result in greater heat transfer at the expense of pressure loss. Nyong et al. [9] also carried out a parametric analysis on the effects of mass flow rate on pressure drop, heat transfer coefficient, and turbulence intensity in STHEs. The studies have shown that increasing mass flow rates significantly impact these parameters [9]. Sanaye et al. [10] worked on optimization problem where they focused on maximizing heat transfer while minimizing pressure drop. The Bell-Delaware procedure and NTU method were used to calculate the shell side pressure drop and heat transfer coefficient of STHE with segmental baffle (STHE-SB), respectively. A genetic algorithm was used to solve the optimization problem.

Nyong, et al. [11] explore the effects of baffle design on shell and tube heat exchanger (STHE) performance. They carried out computational fluid dynamics (CFD) simulations which revealed that increasing baffle cut ratios from 15% to 45% leads to decreased heat transfer coefficients and pressure gradients.

Awan et al. [12] in their studies demonstrated that optimizing baffle spacing (20-35%) and cut percentage (20-35%) can improve heat exchanger efficiency by up to 110%, with the addition of seals further enhancing performance by 10.9%. Muhammad et al.[13] studied the economics of a shell-and-tube heat exchanger using Kern, Bell-Delaware, and Wills-Johnston. They developed a numerical code for hydraulic, exergy, thermal, and economic analysis of shell and tube heat exchangers. The Wills-Johnston and Bell-Delaware methods correlated well, whereas the Kern approach indicated modest differences in shell side computations due to many assumptions made in the kern technique. The pressure loss increased with mass flow rate and baffle number, according to the parametric study. The problem associated with sizing when designing a shell and tube heat exchanger is finding the right balance between the heat transfer coefficient and pressure drop while determining the precise dimensions of the heat exchanger. The heat exchanger size must be designed to meet specific requirements, such as the heat transfer rate and temperature distribution, and must also take into account factors such as flow rates, fluid properties, fouling, and corrosion. Some other necessary requirements include ease of cleaning and maintenance, tube vibration, and thermal expansion.

This work is basically to produce a preliminary design of a shell and tube heat exchanger for future development using computer-based tools and software.

II. MATHEMATICAL MODEL.

There are several techniques for calculating the shell-side heat transfer coefficient and pressure drop; however, the correlations were derived based on experimental data for a typical STHE to provide a more accurate result. In particular, the Kern's technique, which was an attempt to correlate data for conventional SHTE using a simple equation to equations for flow in tubes, was one of the ways used. However, because this technique is constrained to a set baffle cut of 25 percent, it is unable to effectively account for leakages from the baffle to the shell and from the tube to the baffle. The Kern's method, on the other hand, is not totally exact, but it does allow for the computation of the rate of heat transfer as well as the pressure drop.

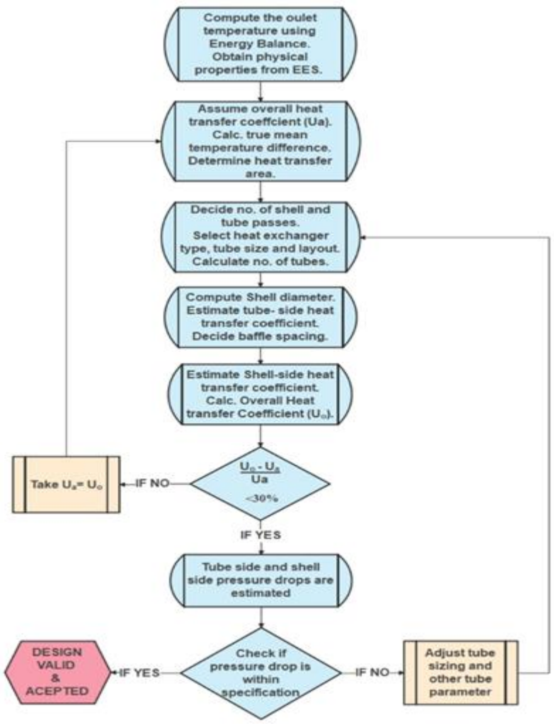

Figure 1.0 Flow chart showing design procedures

Figure. 1 shows the flow chart of the design procedures which starts with determining the heat duty of the SHTE. This is defined by calculating the unknown parameters. The choice of assuming three known temperatures and determining the fourth, or assuming four temperature values and determining one of the hot or cold side flow rates. One of the parameters could be calculated by solving the following heat duty equation [14].

$$Q = \dot{m}_h Cp_h \left(T_{h,i} - T_{h,o}\right) \tag{1}$$

$$Q = \dot{m}_c Cp_c \left(T_{c,o} - T_{c,i}\right) \tag{2}$$

where Q is the heat duty, m is the mass flow rate, C_p is the specific heat capacity, T is the temperature, the subscripts h and c stand for the hot and cold stream respectively, and the subscripts i and o stand for the inlet and outlet of the hot and cold stream respectively. The shell inner diameter is given as

$$D_s = \left(\frac{n}{k_1}\right)^{1/n_1} .d_o \tag{3}$$

where n is the tube number, d_o is the tube outer diameter, K_1 and n_1 are coefficients taken according to the arrangement and number of tube passes whose data is shown in Table 1.

TABLE 1
SPECIFICATION FOR NUMBER OF PASSES [14]

Number of passes	Triangle tube pitch		Square tube pitch	
	K_1	n_1	K_1	n_1
1	0.319	2.142	0.215	2.207
2	0.249	2.207	0.156	2.291
4	0.175	2.285	0.158	2.263
6	0.0743	2.499	0.0402	2.617
8	0.0365	2.675	0.0331	2.643

The tube pitch and inner diameter are computed as

$$P_t = 1.25 d_o \tag{4}$$

$$d_i = d_o - 2t \tag{5}$$

The heat transfer surface is calculated as:

$$S_A = \frac{Q}{UF\Delta T_{lm}} \tag{6}$$

where S_A is the heat transfer surface area, U is the overall heat transfer coefficient. ΔT_{lm} is the logarithmic mean temperature difference (LMTD). While F is the correction factor for the flow configuration, which is found as a function of dimensionless temperature ratio for most flow configuration [15]. The correction factor for ΔT_{lm} in a multi pass STHE is estimated by [16, 17]. The logarithmic mean temperature difference (LMTD) and F are derived from Eqs. (4) and (5) [2].

$$\Delta T_{lm} = \frac{\left(T_{h,i} - T_{c,o}\right) - \left(T_{h,o} - T_{c,i}\right)}{\ln\left(\dfrac{T_{h,i} - T_{c,o}}{T_{h,o} - T_{c,i}}\right)} \tag{7}$$

$$F = \frac{\left(\sqrt{1+R^2}\ln\left(\dfrac{1-P}{1-RP}\right)\right)}{(R-1)\ln\left(\dfrac{2-P\left(R+1-\sqrt{1+R^2}\right)}{2-P\left(R+1+\sqrt{1+R^2}\right)}\right)} \tag{8}$$

Where;

$$R = \frac{T_{h,i} - T_{h,o}}{T_{c,o} - T_{c,i}} \tag{9}$$

and

$$P = \frac{T_{c,o} - T_{c,i}}{T_{c,o} - T_{c,i}} \tag{10}$$

The overall heat transfer coefficient is calculated as[17]:

$$U = \left(\frac{1}{h_s} + R_{f,s} + \frac{d_o \ln\left(d_o / d_i \right)}{2k_w} + R_{f,t} \frac{d_o}{di} + \frac{d_o}{h_t d_i} \right)^{-1} \tag{11}$$

where h_s and h_i are the heat transfer coefficient whereas $R_{f,s}$ and $R_{f,t}$ are the fouling coefficient of the shell and tube side respectively.

k_w is the thermal conductivity for tube wall at the bulk mean temperature of the fluid. The tube is obtained from Eqs. (12)

$$n = \frac{A}{\pi d_o L} \tag{12}$$

where d_o is the outer tube diameter, L is the tube length. The tube side heat transfer coefficient is obtained based on an assumption of turbulent fully developed flow [15].

$$h_t = 0.023 \frac{k_t}{d_i} \mathrm{Re}_t^{0.8} \mathrm{Pr}_t^{1/3} \left(\frac{\mu_t}{\mu_{tw}} \right)^{0.14} \tag{13}$$

where μ_t is fluid dynamics viscosity at the bulk temperature at the tube side, $T_{b,t}$ and μ_{tw} is the fluid dynamics viscosity at the inner tube wall temperature, $T_{t,w}$ which is derived from Eqs. (14)

$$h_t \left(T_{t,w} - T_{b,t} \right) = U \left(T_{b,s} - T_{b,t} \right) \tag{14}$$

Where $T_{b,s}$ is the fluid bulk temperature of shell-side flow. The Reynolds number for the tube-side flow is computed using

$$\mathrm{Re}_t = \frac{\rho_t V_t d_i}{\mu_t} \tag{15}$$

The tube-side fluid velocity is calculated as

$$V_t = \frac{N_p}{n} = \frac{m_t}{\pi \left(d_i^2 / 4 \right) \rho_t} \tag{16}$$

Where N_p is the tube pass number, then the shell side heat transfer coefficient is expressed in Eqs. (17)[5] as

$$h_s = 0.36 \frac{k_s}{D_e} \mathrm{Re}_s^{0.55} \mathrm{Pr}_s^{\frac{1}{3}} \left(\frac{\mu_s}{\mu_{sw}} \right)^{0.14} \tag{17}$$

Where D_e is the shell hydraulic diameter, μ_s is the fluid dynamics viscosity at the bulk temperature at the shell side and μ_{sw} is the dynamic viscosity coefficient at outer tube wall temperature which is estimated through Eqs.

$$h_s \left(T_{b,s} - T_{t,w} \right) = U \left(T_{b,s} - T_{b,t} \right) \tag{18}$$

The shell side hydraulic diameter and the Reynolds number are computed by Eqs. (19) and (20)

$$\begin{cases} \dfrac{4\left(P_t^2 - \pi d_o^2 / 4 \right)}{\pi d_o} \text{For square arrangement} \\ \left(\dfrac{4\left(P_t^2 / 2 * 0.87 P_t - \left(0.5 \pi d_o / 4 \right) \right)}{\pi d_o} \right) \text{For triangular arrangement} \end{cases} \tag{19}$$

According to the flow regime in the tube side, the friction factor and convective heat transfer coefficient are calculated centered on the Reynolds number as [16, 17].

$$\mathrm{Re}_s = \frac{m_s \times D_e}{\mu_s \times A_s} \tag{20}$$

Where A, is the cross area of fluid flow which is given by

$$A_s = \frac{D_s . B \left(P_t - d_o \right)}{P_t} \tag{21}$$

Where the B is the baffle spacing and P_t is the tube side pressure drop which can be obtained as in Eq. (20)[14]

Parameter	**Value**
Shell side flow rate (kg/s)	6.5
Tube side flow rate (kg/s)	50.5
Hot fluid inlet temperature (℃)	90
Cold fluid inlet temperature (℃)	25
Hot fluid outlet temperature (℃)	40
Cold fluid outlet temperature (℃)	?
Allowable operating pressure (tube side (kpa))	100
Allowable operating pressure (shell side (kpa))	100
Tube inner diameter (m)	0.0189
Tube outer diameter (m)	0.0222

$$\Delta P_t = N_p\left(4 f_t \frac{L}{d_i} + 2.5\right)\frac{\rho_t V_t^2}{2} \quad (22)$$

f_t is the friction factor for turbulent tube flow which is stated in Eq, (23)

$$f_t = 0.046\left(\mathrm{Re}_t\right)^{-0.2} \quad (23)$$

Then the shell side pressure drop is calculated as in Eq, (24)[18]

$$\Delta P_s = f_s \cdot \left(\frac{\rho_s V_s^2}{2}\right) \cdot \left(\frac{L}{B}\right) \cdot \left(\frac{D_s}{D_e}\right) \quad (24)$$

where the friction factor for shell-side which is expressed in Eq. (25) as

$$f_s = 2 b_o \mathrm{Re}_s^{-0.15} \quad (25)$$

Where b_o = 0.72 [18] and its valid for Re_s < 40,000. The total power consumption is obtained through Eq. (26)[19]

$$P = \frac{1}{\eta}\left(\frac{m_s}{\rho_s}\Delta P_s + \frac{m_t}{\rho_t}\Delta P_t\right) \quad (26)$$

Where η is the pump efficiency which has a constant value of 0.7.

The physical properties of the fluid are collated at the average temperature of the hot and cold stream respectively. Table 2 shows the initial parameter for the design with the allowable operating pressure at the tube and shell side at 100 KPa respectively. The flow rate for the tube and shell side were provided with three temperatures, while the fourth temperature for the cold outlet was calculated using the energy balance equation.

TABLE 2
INITIAL DESIGN DATA

III. RESULTS AND DISCUSSIONS

A. Model Validation

To enhance the fidelity of the work, the current models was validated against available data in literature. The working fluid was water to water on the tube and shell side respectively. The computed results were compared with results in literature as shown in Table 3**, and** it **i**s observed that the deviation of current model results with data in literature is within an acceptable ±5%.

B. Parametric Study

The parametric study was carried out to investigate the effect of varying the tube length of certain parameters of heat exchanger design such as number of baffle, number of tube, the ratio of L_t/D_s, the flow velocity at the shell and tube side, the pressure drop and heat transfer coefficient. The flow rate of the shell and tube side fluid was kept constant while varying the tube length. The model was written in EES code taking into account all the mathematical model using the kern's method. The procedures on the flow chart as depicts in Figure 1 was applied and taking into cognizant three criteria which the design must fulfill before the design must be accepted and these conditions are within TEMA standard for the design of shell and tube heat exchanger.
These criteria are:

1. The ratio of the length of tube (L_t) to the shell diameter that is L_t/D_s must be in the range of 5 and 10.

2. The velocity of flow in the tube should be in the range of 1.50 to 2.5 m/s while that of the shell should be in the range of 0.3 to 1 m/s.

3. The value of $(U_{ass} - U_{cal})/U_{ass}$ *100% should be less than 30%.

With these conditions, the mathematical model was conducted using the kern's method. The tube length was varied from 6 (1.83 m) to 14 feet (4.26 m), while observing the three conditions. Keeping all other parameters constant, the tube length was varied until the all the criteria were meet. Once a calculation is done and the corresponding nding output of the result fall below the specified L_t/D_s, ratio the next tube length is considered until all the conditions are simultaneously meet.

The only tube length that met these conditions were observed to be the 14 feet (4.26 m). This tube length was then taken as the preliminary tube length for the design of the shell and tube heat exchanger.

Table 4 summaries the results obtained from the parametric study. We can see from the Table 4, that the L_t/D_s ratio for 14 feet tube length had a value of 8.27 which is within the range of 5 and 10. The velocity of the tube and shell side were 1.922 m/s and 0.3123 m/s respectively. The expected flow velocity within the TEMA standard specified the ranges to be observed in the design of any heat exchangers.

Lastly the $(U_{ass} - U_{cal})/U_{ass}$ *100% less than 30% was also observed for the 14 feet tube length which has a value 17.28 % less than the stipulated value of 30% for heat exchanger design.

TABLE 3
MODEL VALIDATION

PARAMETERS	Literature [1] Water to Water STHE	Present work Water to Water STHX	Error %
Kern method			
S (m^2)	278.6	278.8	±0.07%
Re_t	15779	15769	±0.06%
h_t (W/m^2K)	4053	4097	±1.08%
Re_S	15095	14841	±1.67%
h_S (W/m^2K)	4300	4378	±1.81%
ΔP_t, (Pa)	6098	6085	±0.210%
ΔP_S, (Pa)	27786	28307	±1.87%

TABLE 4
RESULTS OBTAINED FROM THE PARAMETRIC STUDY

S/N	Length of Tube (L_t) in m	Number of Baffles (N_b)	Number of tubes (N_t)	L_t/D_s ratio ranges between 5 and 10	Tube Reynolds number (Re_t)	Shell Reynolds number (Re_s)	Tube flow velocity, V_t (m/s)	Shell flow velocity, V_s(m/s)	$(U_{ass} - U_{cal})/U_{ass}$ *100% > 30%
1.	1.83	5	439	2.53	18709	5638	0.823	0.1579	infinity
2.	2.44	8	329	3.78	24964	7134	1.098	0.1997	1.49%
3.	3.05	11	292	4.96	26931	7848	1.135	0.2194	7.71%
4.	3.66	16	219	6.68	37503	9889	1.65	0.2769	13.17%
5.	4.26	20	188	8.27	43687	11157	1.922	0.3123	17.28%

TABLE 5
THERMAL DESIGN RESULTS

Parameter	Symbol	Value	Unit
Cold fluid outlet temperature	$T_{c,o}$	31.45	[ºC]
Cold fluid inlet temperature	$T_{c,i}$	25	[ºC]
Hot fluid inlet temperature	$T_{h,i}$	90	[ºC]
Hot fluid outlet temperature	$T_{h,o}$	40	[ºC]
Specific heat capacity of water	C_p	4183.5	J/Kg.ºC
Density of water	ρ	988.25	Kg/m^3
LMTD	ΔT_{lm}	31.98	[ºC]
Assumed heat transfer coefficient	U_a	810	$W/m^2.$ºC
Heat transfer area	A	56	m^2
Tube side velocity	U_t	1.922	m/s
Shell side velocity	U_s	0.3123	m/s
Tube side Reynolds number	Re_t	43687	-
Shell side Reynolds number	Re_s	11157	-
Shell side pressure drop	ΔP_s	11485	Pa
Tube Side Pressure Drop	ΔP_t	27117	Pa

Shell Side Heat Transfer Coefficient	h_s	3393	W/m^2.ºC
Tube Side Heat Transfer Coefficient	h_t	8029	W/m^2.ºC
Nusselt Number	Nu	85.07	-
Overall heat transfer coefficient	U_{cal}	950.9	W/m^2. ºC
Tube inner diameter	T_{ID}	0.0189	m
Tube outer diameter	T_{OD}	0.0222	m
Length of tubes	L_T	4.26	m
Number of tubes	N_T	188	-
Tube Pitch	T_p	0.02775	m
Bundle diameter	B_D	0.4471	m
Shell diameter	S_D	0.5151	mm
Baffle spacing	L_B	0.206	m
Thermal conductivity of copper	K_C	395	W/mºC

Effect of varying tube length on the shell and tube heat exchanger: The parametric study was carried out to investigate the effect of varying the tube length on certain parameters of heat exchanger design such as number of baffles, number of tube, the ratio of L_t/D_s, the flow velocity at the shell and tube side, the pressure drop and heat transfer coefficient. The flow rate of the shell and tube side fluid was kept constant while varying the tube length.

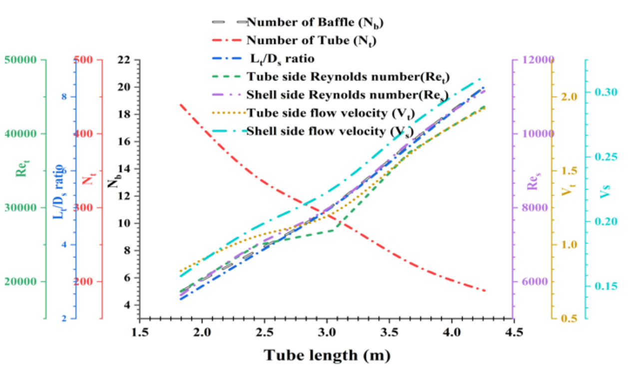

Figure 2: the effect of tube length on the shell and tube heat exchanger design.

The tube length is one imperative parameter that governs the design and performance of every heat exchanger. It is observed from Figure 2 shows the effect of tube length on the output parameters of the shell and tube heat exchanger such as number of battles, number of tubes, L_t/D_s, Reynolds number and the flow velocities at both shell and tube side. From figure 2 increasing the tube length leads to a decrease in the number of tube while other parameters maintain a tremendous increase. An increase in the tube length simultaneously causes an increase in the pressure drop both in the tube and shell side. As the tube length increases, the fluid has to travel a longer distance. This results in a greater frictional resistance, leading to a higher pressure drop. Essentially, longer tubes mean more surface area for the fluid to encounter friction.

IV. CONCLUSION

The preliminary design of shell and tube heat exchangers using a computer code developed in engineering equation solver (EES) based on the Kern method was used to vary heat exchanger parameter such as the tube length in a systematic manner to identify the configurations that meets specified heat transfer coefficients and pressure drops within the TEMA standards. The study highlighted the importance of considering one design variable, the tube length and the corresponding impacts on the output parameters listed above in Table 4, while maintaining TEMA standards. The parametric study showed that the tube length with 14 feet fulfilled all the requirement or conditions stated in TEAM standards. On this point the computed tube length of 16 feet was accepted for the design of the current shell and tube heat exchanger.

The results show that increasing tube length generally leads to an increase in pressure drop and can have a varying effect on the heat transfer coefficient, depending on specific design and operational conditions.

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