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Research Article

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Development of a Shell and Tube Heat Exchanger for Experimental Purposes in Tertiary Institutions

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Abstract The design of shell and tube heat exchangers is a very important subject in industrial processes. However, some difficulties were discovered, especially in the shell-side design, because of the complex characteristics of heat transfer and pressure drop. [8]

In this research work, the aim is to Design, Fabricate and Test a small Shell and Tube Heat Exchanger (STHE), suitable for experimental purposes in tertiary institutions.

The design is based on optimal combination of fluid flow parameters using LMTD software, it is laboratory scale, locally available technology and materials were utilized.

Difficulties reported [8] were also put into consideration in the design to come up with a better facility that will be operating at minimal cost under optimized conditions. The design was optimized using LMTD (Log Mean Temperature Difference) software before fabrication to distinguish which fluid flowing parameters will be suitable and economical.

It was discovered from the obtained results that the Bell-Delaware correlation produced the optimal value of shell fluid film coefficient over the other correlation techniques at shell diameter at any parametric iterations. Kern technique was the closest with correction factors between 0.8440 and 0.9850. There was an inverse relationship between the shell fluid film coefficient and heat transfer rate. Irrespective of the shell diameters, the heat transfer rate is constant. There is an inverse relationship between the LMTD and the heat transfer rate under any shell design parameters.

Keywords Heat Exchanger, Shell, Tube, Heat Transfer, Fluids.

1. Introduction

A heat exchanger is a system used to transfer heat between two or more fluids. Heat exchangers are used in both heating and cooling processes. The fluids may be separated a solid wall to prevent the mixing or they may be in direct contact. Heat exchangers are specialized devices used to transfer thermal energy between two or more fluids, at a different temperature. It is a device in which energy is transferred from one fluid to another across a solid surface, every living thing is equipped in some way or the other with heat exchangers. Heat Exchangers are devices used to enhance or facilitate the flow of heat [4]. It can also be defined as a device which provides a flow of thermal energy between two or more fluids at different temperatures [10]. A heat exchanger is equipment that transfers the energy from a hot fluid to a cold fluid, with maximum rate and minimum investment and running cost. [7]

They are used in many engineering applications like power generation, waste heat recovery, manufacturing industry, air-conditioning, refrigeration, space applications, petrochemical industries [10], food industries, space heating, refrigeration, air conditioning, power plants, chemical plants, petrochemical plants, petroleum refineries, natural gas processing, and sewage treatment [4].

In heat exchangers, there are usually no external heat and work interactions. The essence of heat exchanger is to recover or reject heat, or sterilize, pasteurize, fractionate, distill, concentrate, crystallize, or control a process fluid [3]. In a few heat exchangers, the fluids exchanging heat are in direct contact. In most heat exchangers, heat transfer between fluids takes place through a separating wall or into and out of a wall in a transient manner. In many heat exchangers, the fluids are separated by a heat transfer surface, and ideally they do not mix or leak ([9]; as cited by [3]).

Heat exchanger may be classified according to the following main criteria ([10]; [7]).

i. Arrangements: parallel, counter and cross-flows.
ii. Shell and tube Recuperators and Regenerators.
iii. Transfer process: Direct contact and Indirect contact.
iv. Geometry of construction: tubes, plates and extended surfaces.
v. Heat transfer mechanisms: single phase and two phase.

The rate of heat transfer between the two fluids at a location in a heat exchanger depends on the magnitude of the temperature difference at that location, which varies along the heat exchanger. In the analysis of heat exchangers, it is usually convenient to work with the logarithmic mean temperature difference LMTD, which is an equivalent mean temperature difference between the two fluids for the entire heat exchanger. [2]

The Logarithmic Mean Temperature Difference (LMTD)

The log mean temperature difference ΔT_{lm} relation developed earlier is limited to parallel-flow and counter-flow heat exchangers only. Similar relations are also developed for cross-flow and multipass shell-and-tube heat exchangers, but the resulting expressions are too complicated because of the complex flow conditions.

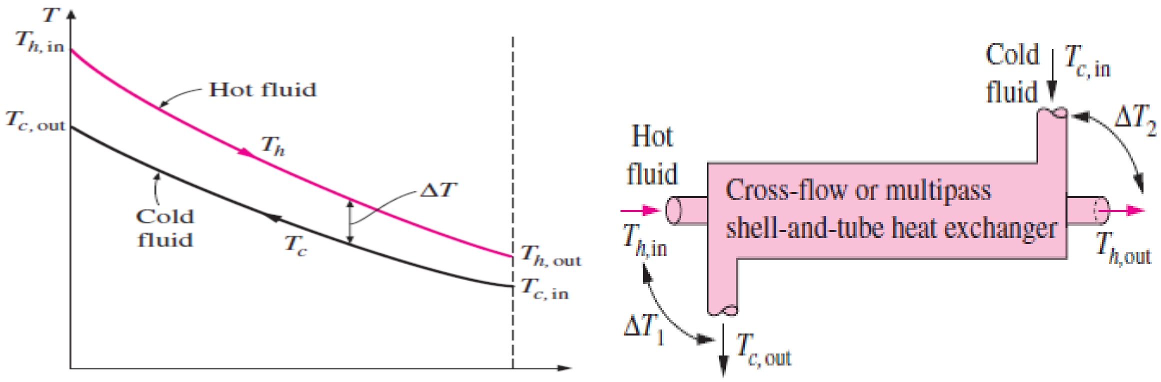

Figure 1: The determination of the heat transfer rate for cross-flow and multipass shell-and-tube heat exchangers using the correction factor

Heat transfer rate:

$$\dot{Q} = UA_sF\Delta T_{lm,CF} \quad (1)$$

Where,

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{In(\Delta T_1 / \Delta T_2)} \quad (2)$$

$$\Delta T_1 = T_{h,in} - T_{c,out} \quad (3)$$

$$\Delta T_2 = T_{h,out} - T_{c,in} \quad (4)$$

In such cases, it is convenient to relate the equivalent temperature difference to the log mean temperature difference relation for the counter-flow case as

$$\Delta T_{lm} = F\Delta T_{lm,CF} \quad (5)$$

Where F is the **correction factor,** which depends on the geometry of the heat exchanger, the inlet and outlet temperatures of the hot and cold fluid streams. The $\Delta T_{lm,CF}$ is the log mean temperature difference.

The correction factor is less than unity for a cross-flow and multipass shell and- tube heat exchanger. That is, $F \leq 1$. The limiting value of $F = 1$ corresponds to the counter-flow heat exchanger. Thus, the correction factor F for a heat exchanger is a measure of deviation of the ΔT_{lm} from the corresponding values for the counter-flow case.

Materials and Methods

Method

In the present project, the methodology used in the design of the heat exchanger is studied and presented. The thermal design involves the calculation of shell side and tube side heat transfer coefficients, heat transfer surface area and pressure drops on the shell side and tube side. The mechanical design involves the calculations of thickness of pressure parts of the heat exchanger such as the shell, channel, tube etc. to evaluate the rigidity of part under design pressures.

Description of the project/Working Principle

i. The principle of operation is simple enough, two fluids of different temperatures are brought into close contact but they are not mixing with each other.
ii. One fluid runs through the tube, and another fluid flows over the tube (through the shell) to transfer heat between the two fluids.
iii. The hot water run through the shell while glycerin run through the tube side. In order to transfer heat efficiently, a large heat transfer area were used, so there are many tubes.
iv. The temperatures of the two fluids tend to be equal. The heat is simply exchanged from one fluid to another and vice versa. No energy is added or removed.

Fabrication of components of shell and tube heat exchanger:

1. The tubes used in this project were welded and made of copper due to its high thermal conductivity. The tubes are 914.4 mm and ϕ 12.7 mm each.
2. **Tube Pitch:** The tube pitch is the shortest distance between two adjacent tubes. In this work a triangular pitch was adopted because it provides a more robust tube sheet construction
3. **Tube Sheet/plates:** They are used to hold the tubes at the ends. The tube sheet used in the construction is made of circular metal (galvanized steel) plate with holes drilled through for the desired tube pattern, holes for the tie rods, grooves for the gaskets and bolt holes for flanges to the shell. The tubes are held in place by being inserted into holes in the tube sheet and there either expanded into grooves cut into the holes or welded to the tube sheet where the tube protrudes from the surface.
4. **Baffles:** In this project, the baffles machined were made from galvanized steel which is compatible with the shell side fluid. In this project cuts areas of 25% was adopted.
5. **Nozzles:** The entrance and exit ports for the shell fluid and tube fluid are referred to as “Nozzles”. These nozzles are pipes of constant cross section welded to the shell and channels.
6. **Front-End And Rear End Covers:** They are containers for tube fluids for every pass. In this project both ends are stationary.
7. **Tie Rods and Spacers:** These were provided to retain all transverse baffles and tube support plates securely in position. They serve two purposes; one to maintain the spacing between the baffles and second function is to reduce the fluid by-passing.
8. **Shell:** In this project the shell which is cylindrical was produced from the rolling machine.
9. **Fluid Chambers:** Both the inlet and outlet and rear end fluid chambers are fabricated from rolled galvanized steel plate and are of adequate proportions to minimize pressure drop and turbulence. The inlet and outlet fluid chamber carries water and inspection cover was divided internally into inlet and outlet chambers, each having a flanged connection. The rear end fluid chamber consists of a simple dished cover or a flat end cover.
10. **Surface Water Pump;** A surface pump has a sealed motor closely coupled to the pump body and pushes water to the surface. It is a centrifugal pump which uses a rotating impeller to increase pressure of a fluid.
11. **Temperature Controller;** A temperature controller was used to monitor and control the temperature at the inlet and exit if the heat exchanger.
12. **Water Heater:** A water heater of 3000W was used to heat the hot fluid.
13. Contact Breaker
14. Water Heater Switch

Design Consideration

The optimum thermal design of a shell and tube heat exchanger involves the consideration of many interacting design parameters which can be summarized as follows:

A. Process

1. Process fluid assignments to shell side or tube side.
2. Selection of stream temperature specifications.
3. Setting shell side and tube side pressure drop design limits.
4. Setting shell side and tube side velocity limits.
5. Selection of heat transfer models and fouling coefficients for shell side and tube side.
6. Leak-tight is an important consideration when toxic or expensive fluids are involved. However, mixing between the two fluids (in the event of leaks where the tube is sealed into the tube sheet) must be avoided [7].

B. Mechanical

1. Selection of heat exchanger layout and number of passes.
2. Specification of tube parameters - size, layout, pitch and material.
3. Setting upper and lower design limits on tube length.
4. Specification of shell side parameters: materials baffle cut, baffle spacing and clearances.
5. Setting upper and lower design limits on shell diameter, baffle cut and baffles spacing.
6. Ease of servicing, low maintenance cost, and safety and reliability are some other important considerations in the selection process.
7. Quietness which is one of the primary considerations in the selection of liquid-to-air heat exchangers used in heating and air-conditioning applications.

Design Calculations

Assumptions

The following are the major assumptions made for the pressure drop analysis;

1) The operating or Flow condition is steady and isothermal, and fluid properties are independents of time.
2) Fluid density is dependent on the local temperature only or is treated as constant.
3) The pressure at a point in the fluid is independent of direction.
4) Body force is caused only by gravity.
5) There are no energy sink or sources along streamline; flow stream mechanical energy dissipation is idealized as zero. i.e. Changes from K.E to P.E of fluid streams are negligible
6) The friction factor is considered as constant with passage flow length.
7) The heat exchanger is well insulated so that heat loss is negligible and thus heat transfers from hot to cold fluids are equal.
8) Heat transfer coefficient and fouling factors are constant and uniform
9) Thermal resistance of the inner tube is negligible, since the tube is thin walled and highly conductive.

Mathematical Modeling

The mathematical model has been constructed considering the principles of heat transfer and fluid mechanics. In this project a simple counter flow shell and tube type heat exchanger was designed to cool the water from 100℃ to 60℃ by using glycerin at room temperature (26℃).

The steps of designing a typical shell and tube heat exchanger are described as follows:

A. Decision on the type of exchanger to be used (which in this case is shell and tube type)
B. Definition of the duty**:** heat-transfer rate, fluid flow-rates and temperatures: The energy balance was considered first, to find out the values of some unknown temperature values. Certainly some inputs like hot fluid inlet and outlet temperatures, cold fluid inlet temperature, and mass flow rates of the two fluids are needed to serve the purpose. The energy balance equation may be given as:

$$Q_{cold} = Q_{hot}$$
$$M_c CP_c (T_{c2} - T_{c1}) = M_h CP_h (T_{h1} - T_{h2})$$

C. Then we consider the LMTD expression to find its value: [2]

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{In\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

$$\Delta T_{lm} = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{In\left(\frac{T_{h1} - T_{c2}}{T_{h2} - T_{c1}}\right)}$$

D. Collection of the fluid physical properties required like density, viscosity, thermal conductivity etc.
Shell side fluid – Hot water
Tube side fluid – Cold water
Inlet temperature of hot water, $T_{h1} = 100^0C$
Outlet temperature of hot water, $T_{h2} = 75^0C$
Inlet temperature of Cold Water, $T_{c1} = 27C$
Outlet temperature of Cold Water, T_{c2}
Mass flow rate of Cold Water, m_t
Mass flow rate of hot water, $m_s = 0.644\ kg/s$
Thermal conducting of the tube material, k_w
The maximum flow rate through the tube, U_{max}

E. Calculation on the number of tubes, tube bundle diameter, diameter of shell and its thickness. Then, proper baffle dimension viz. its diameter, thickness and baffle spacing.

1. **Number of tubes:** from $\ell_t U_m A_t$

$$N_t = \frac{M_t}{\ell_t U_m A_c}$$

$$A_c = \frac{\pi {d_i}^2}{4}$$

$$N_t = \frac{4M_t}{\ell_t U_m \pi {d_i}^2}$$

2. **Hypothetical Area of the tube**

$$A_t = \frac{\pi {d_i}^2}{4} \times N_t$$

3. **Hypothetical Area of the shell/estimated shell area**

$$A_S = (1 - PR^{-1}) D_S B$$

$$B = 7.48 {d_0}^{0.75}$$

$$P_R = \frac{P_t}{d_0}$$

$$P_R = \frac{d_0 + C}{d_0}$$

$$D_s = \frac{N_t \times CL \times PR^2 \times d_0}{0.785CTP}$$

$$CL = 0.87\ for\ 30^0 \& \ 60^0$$

$$CTP = 0.95\ for\ on\ e\ tube\ pass$$

$$A_S = (1 - PR^{-1}) D_S B$$

F. Determination of heat transfer coefficient on the inner and outer surface of the tubes using the above methods to optimized the design

1. **Shell-side Transfer Coefficient ($h_0 = h_s$)** [1]

i. Kern's Method

$$d_e = \frac{4}{\pi d_{t0}}\left[p_t^2 - \frac{\pi d_{t0}^2}{4}\right]$$

$$Re_s = \frac{M_s}{A_s} \times \frac{D_e}{\mu_s}$$

$$N_u = \frac{h_s d_e}{k_s}, h_s = \frac{N_s k_s}{d_e}$$

$$N_u = 0.36{R_e}^{0.55} Pr^{1/3}$$

$$h_s = \frac{0.36ks}{d_e}\ {R_e}^{0.55} Pr^{1/3}$$

ii. Taborek Method

$$h_s = \frac{0.2ks}{d_{t0}}\ {R_e}^{0.6} Pr^{0.4}$$

$$R_{es} = \frac{m_s}{A_s} \times \frac{d_{t0}}{\mu_s}$$

iii. Bell-Delaware Method

$$h_s = j_i CP_s \left(\frac{M_s}{A_s}\right) \left[\frac{K_s}{CP_s \mu_s}\right]^{2/3} \left[\frac{\mu_t}{\mu_s}\right]^{0.14}$$

Neglecting $\frac{\mu_t}{\mu_s} \approx 1$

$$J_i = 0.37{R_{es}}^{-0.395}$$

$$h_s = j_i c_{ps} \left(\frac{m_s}{A_s}\right) \left[\frac{K_s}{c_{ps}\mu_s}\right]^{2/3}$$

2. Tube side heat transfer coefficient

i. Petikhov – Kiwlov Correlation [1]

$$h_t = h_i = \frac{Nu_t k_t}{d_{ti}}$$

$$Nu_t = \frac{\left(f/2\right) Re_t Pr_t}{(1.07) + (12.7)\left(f/2\right)\left({Pr_t}^{1/2} - 1\right)}$$

$$f = \left[\left[1.58\ ln(R_{et})\right] - (3.28)\right]^{-2}$$

$$R_{et} = \frac{\ell_t U_m d_{ti}}{U_t}$$

G. Calculation on the value of the overall coefficient, U.

$$\frac{1}{\mu_0} = \frac{1}{h_0} + \frac{1}{h_i}\frac{d_0}{d_i} + \frac{r_0 ln\left(r_0/r_i\right)}{k_w}$$

H. Determination of the area required of the heat exchanger (on the basis of assumed U_0)

Using $Q = UA\Delta TL_m$, where

$$A = \frac{Q}{\Delta TL_m}$$

$$\therefore L = \frac{A_s}{\pi \Delta_s}$$

I. Pressure Drop in STHE

a. Tube Side Pressure Drop: The tube side pressure drop can be calculated by knowing the number of tube passes (Np) and length (L) of heat exchanger; the pressure drop for the tube side fluid is given by equation

$$\Delta P_t = 4f\frac{LN_p}{d_{ti}}\rho\frac{u_m^2}{2}$$

The change of direction in the passes introduction in the passes introduction an additional pressure drop due to sudden expansions and contractions that the tube fluid experiences during a return that is accounted for allowing four velocity head per pass

$$\Delta P_t = 4N_p \frac{\rho u_m^2}{2}$$

The total pressure drop of the side becomes:

$$\Delta P_t = \left(4f \frac{LN_p}{d_{ti}} + 4N_p\right) \rho \frac{u_m^2}{2}$$

b. Shell Side Pressure Drop: The shell side pressure drop depends on the number of tubes, the number of times the fluid passes the tube bundle between the baffles and the length of each crossing. The pressure drop on the shell side is calculated by the following expression [10]:

$$\Delta P_s = f \frac{\rho u_m^2 (N_b + 1) d_s}{2 d_{to} \phi_s}$$

Where,

$$\phi_s = (\mu_t + \mu_s)0.14$$

N_b = Number of baffles

$(N_b + 1)$ = Number of times fluid passes to the tube bundle

Friction factor (f) calculated from:

$$f = exp(0.576 - 0.19 \ln Re_s)$$

Where

$$400 < Re_s = \frac{\rho_s u_m d_s}{\mu_s} \leq 1 \times 10^6$$

The correlation has been tested based on data obtained on actual exchangers. The friction coefficient also takes entrance and exit losses into account [10].

J. Determination of the values of parameters of interest after rigorous mathematical calculations.

TDC Method - Temperature Design Constant:

When the data sets are not available and the heat exchanger is already installed in service, TDC can be calculated by observing the steam pressure (and finding the steam temperature from steam tables) and the corresponding secondary inlet and outlet temperatures at any load.

Once the exchanger size is fixed and the design temperatures are known, it easier to predict operating temperatures using what could be termed a heat exchanger Temperature Design Constant (TDC).

The TDC method does not require logarithmic calculations.

$TDC = \frac{T_s - T_1}{T_s - T_2}$ [6]

Where:

TDC = Temperature Design Constant;

T_s = Steam temperature;

T_1 = Secondary fluid inlet temperature;

T_2 = Secondary fluid outlet temperature.

Note: The TDC equation can be transposed to find any one variable as long as the other three variables are known. The following equations are derived from the TDC equation.

a) To find the steam temperature at any load:

$$T_s = \frac{(T_2 \times TDC) - T_1}{TDC - T_1}$$

b) To find the secondary fluid inlet temperature at any load:

$$T_1 = T_s - [(T_s - T_2)TDC]$$

c) To find the secondary fluid outlet temperature at any load:

$$T_2 = T_s - \left(\frac{T_s - T_1}{TDC}\right)$$

Observation: For any heat exchanger with a constant secondary flow rate, the operating steam temperature can be calculated for any combination of inlet temperature and outlet temperature.

Results and Discussions

Field Test

STHE was subjected to field test after fabrication. Flow arrangement adopted for the design is cross flow due to greater log mean temperature it offers. PVC pipe was used to convey fluid at tube inlet and it was well connected to ensure the pipe is firmly fitted to heat exchanger inlet port. The pipe connected to inlet of tube side is in turn connected to outlet of surface pump connected into cold water container for easy flow of the fluid (cold water) into heat exchanger, while pipe connected to outlet of shell side fluid (hot water) was connected to retaining container.

Also, another pipe was connected to inlet of shell side fluid (hot water) and outlet of cold side fluid which are gripped with a hose clip. The inlet of the shell side fluid (hot water) was connected to outlet of surface pump while inlet of pump was dipped in the hot water.

Hot water used for experiment was obtained by boiling 40 litres of water by a 3000w water heater. The observed temperature of shell side fluid was 100^0c, while that of tube fluid side (cold fluid) was 26^0c before been allowed to flow into heat exchanger. Temperature measurements were taken with aid of a temperature controller. At the end of experiment, heat transferred to cold fluid increased by 16^0c while that of the hot fluid reduced to 75^0c.

Laboratory Test

The sheet and tube heat exchanger (STHE) was also subjected to laboratory test after field test. The essence of this is to test the STHE on a large scale.

Experimental Results

The results obtained from the experimental results are presented below:

Shell side fluid – Cold water
Tube side fluid – Hot water
Inlet temperature of hot water, $T_{h1} = 100^0C$
Outlet temperature of hot water, $T_{h2} = 75^0C$
Inlet temperature of Cold water, $T_{c1} = 27^0C$
Outlet temperature of Cold water, $T_{C1} = 43^0C$
Mass flow rate of Cold water, $m_t = 0.664kg/s \quad m_c$
Mass flow rate of hot water, $m_s = 0.524kg/s \quad m_h$
Thermal conductivity of the tube material, k_w = 16.26 W/m^{0}C
The maximum flow rate through the tube, $U_{max} = 0.82\ mm/s$

Table 1: Thermal and Physical Properties of Shell Side Fluid @ 27^0C

S/No.	Property	Symbol	Value
1	Spec. heat cap.	Cp	4.18kJ/kg/k
2	Thermal conductivity	K	0.6100 w/m/k
3	Pradtl No.	Pr	5.34
4	Density	ℓ	996 Kg/m^3
5	Viscosity	μ	0.7797 E-3 Kg/m.s

Table 2: Thermal and Physical Properties of Tube-Side Fluid @ Average Temp.= $\frac{T_{h1}+T_{h2}}{2}$

S/ No	Property	Symbol	Unit	Values				
				95 (^{0}C)	**87.5 (^{0}C)**	**75 (^{0}C)**	**50 (^{0}C)**	**27 (^{0}C)**
1	Spec. heat cap.	Cp	kJ/kg/k	4.22	4.21	4.19	4.18	4.18
2	Thermal conductivity	K	j/kgk	0.6791	0.6730	0.6668	0.6435	0.6100
3	Pradtl No.	Pr		1.85	2.05	2.38	3.55	5.34
4	Density	ℓ	Kg/m^3	958.4	967.3	975.1	987.8	997.7
5	Viscosity	μ	Kg/m.s	0.298 E-3	0.328 E-3	0.378 E-3	0.547 E-3	0.779 E-3

Table 3: Parametric Studies

S/N	T_{h1}	T_{h2}	T_{c1}	T_{c2}	Tube diameters (mm)		Shell Dia. (mm)	Baffle spacing (mm)	fluid film coeff. Tube side	fluid film coeff. Shell side	Overall Heat coeff.	LMTD	Heat transfer rate kW
	(^{0}C)	(^{0}C)	(^{0}C)	(^{0}C)	d_{t0}	d_{ti}	d_s	B	h_i	h_o	U_o	ΔT_{lm}	Q
1	100	90	27	30.35	12.7	10.7	152.4	220	42577.6	2614.3	2463.06	66.26	27.014
2	100	80	27	34.68	12.7	10.7	152.4	220	42577.6	2533.7	2391.39	58.96	54.304
3	100	75	27	42.00	12.7	10.7	152.4	220	42577.6	2412.6	2283.27	52.85	67.886
4	100	60	27	43.34	12.7	10.7	152.4	220	42577.6	2216.4	2106.73	42.54	108.62
5	100	50	27	47.64	12.7	10.7	152.4	220	42577.6	2154.7	2050.91	35.69	135.77

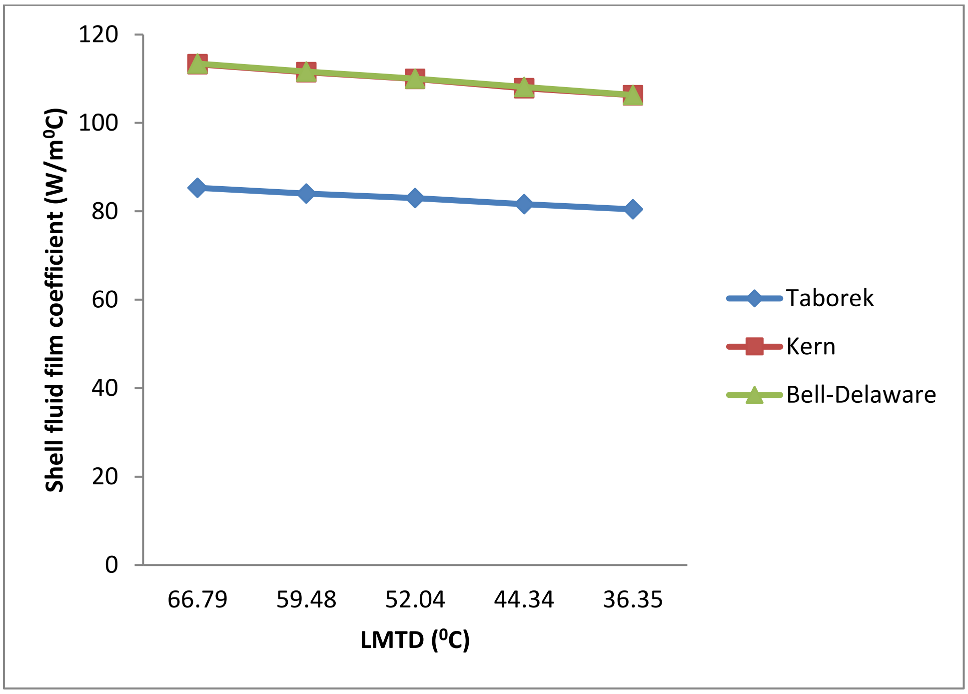

Figure 2: Fluid film variation in shell @ D_S = 152.4MM

Fig. 2 showed the result obtained from the plot of shell fluid film coefficient against LMTD under different correlation techniques. It was observed that Bell-Delaware correlation produced the optimal value of shell fluid film coefficient over the other correlation techniques at shell diameter of 152.4 mm and other parametric iterations. Kern technique was the closest with correction factors between 0.0481 and 0.3942.

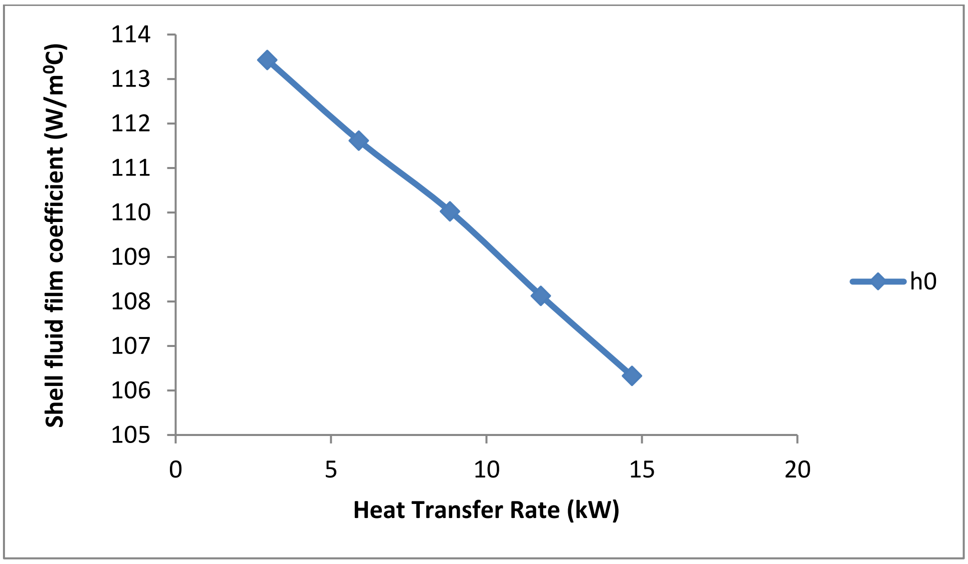

Figure 3: Plot of H_0 against Heat Transfer Rate @ D_S = 152.4 MM

Fig. 3 showed the result obtained from the plot of shell fluid film coefficient against heat transfer rate. It was observed that there is an inverse relationship between the two parameters. That is, the higher the heat transfer rate, the lower the shell fluid film coefficient at shell diameter of 152.4 mm and other parametric iterations.

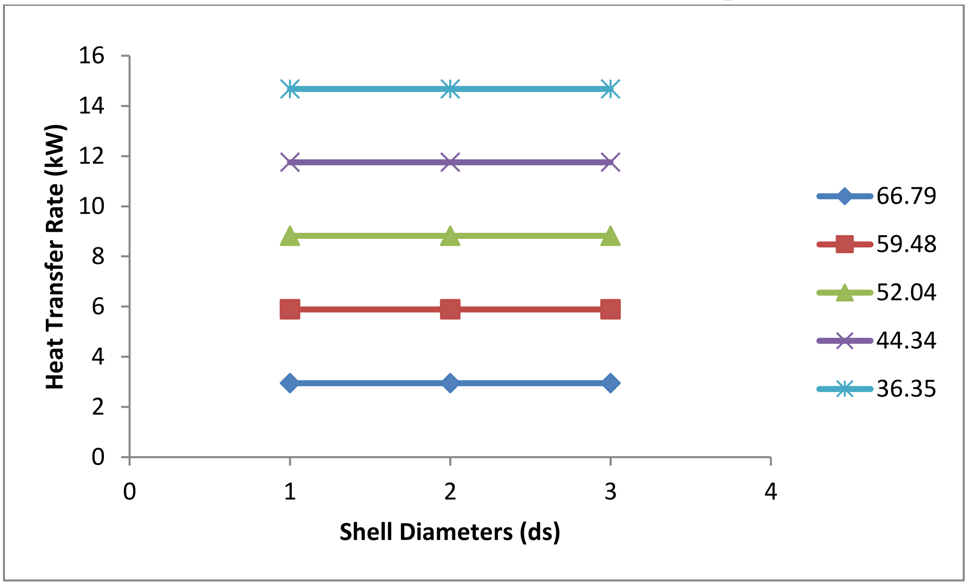

Figure 4: Plot of Heat Transfer Rate against Shell Diameters

Fig. 4 showed the result obtained from the plot of heat transfer rate against shell diameters. It was observed that the heat transfer rate is constant irrespective of the shell diameters. But there is an inverse relationship between the LMTD and the heat transfer rate. That is, the higher the heat transfer rate, the lower the LMTD under any shell design parameters.

Conclusion

The importance of mini shell and tube heat exchangers in industrial and other engineering applications cannot be underestimated. Hence in this project, a mini and portable shell and tube heat exchanger was developed on the laboratory scale using an optimized LMTD technique before embarking on the design and fabrication. The performance of the heat exchanger was assessed and evaluated to determine the optimum combination of design parameters for the transfer of heat between the two fluids involved in the STHE without mixing.

It was discovered from the obtained results that the Bell-Delaware correlation produced the optimal value of shell fluid film coefficient over the other correlation techniques at shell diameter at any parametric iterations. Kern technique was the closest with correction factors between 0.8440 and 0.9850. There was also an inverse relationship between the shell fluid film coefficient and heat transfer rate. Though, irrespective of the shell diameters, the heat transfer rate is constant. But there is an inverse relationship between the LMTD and the heat transfer rate under any shell design parameters.

In addition, there was a direct relationship between h_o and the LMTD. While h_o decreases as the shell diameters decreases. It was observed that as the shell diameter (d_s) increases, both the overall heat transfer coefficient (U_o) and tube fluid film coefficient (h_i) decreases. The LMTD increases, the correction factor, irrespective of the tube sizes decreases. It was also discovered that the correction factor increases as the tube diameters increases. In conclusion, the Values of parameters of interest were also presented after rigorous mathematical calculations at optimal level.

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APPENDIX 1

PLATE 1: SHELL

APPENDIX 2

PLATE 2: TUBE

APPENDIX 3

PLATE 3

APPENDIX 4

PLATE 4

PLATE 5