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Performance Analysis of Shell and Tube Heat Exchangers: A case study

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Abstract

This paper presents performance analyses of shell and tube heat exchangers. Analytical method was used to develop correlation for the performance analysis. A program was written in MATLAB to check for the thermal and hydraulic suitability of the heat exchangers. The program was tested with dataof five different industrial heat exchangers from Port Harcourt Refinery. The results obtained showed reasonable agreement with the actual field data, thus demonstrating that the program is reliable and can be applied in the performance analysis of shell and tube heat exchangers.

Keywords: heat exchangers, performance, analysis, MATLAB.

INTRODUCTION

Heat exchangers are one of the most important devices of mechanical systems in modern society. Most industrial processes involve the transfer of heat and more often, it is required that the heat transfer process be controlled. According to Oko (2008), a heat exchanger is a device of finite volume in which heat is exchanged between two media, one being cold and the other being hot. There are different types of heat exchangers; but the type widely used in industrial application is the shell and tube. As its name implies, this type of heat exchanger consists of a shell with a bundle of tubes inside it. One fluid runs through the tubes, and another flows over the tubes to transfer heat between the two fluids. The tube bundle may consist of several types of tubes: plain, longitudinally finned, etc. To ensure that the shell-side fluid will flow across the tubes and thus induce higher heat transfer, baffles are installed in the shell to force the shell-side fluid to flow across the tube to enhance heat transfer and to maintain uniform spacing between the tubes (Holman, 2004); schematically, this is shown in Figure 1.

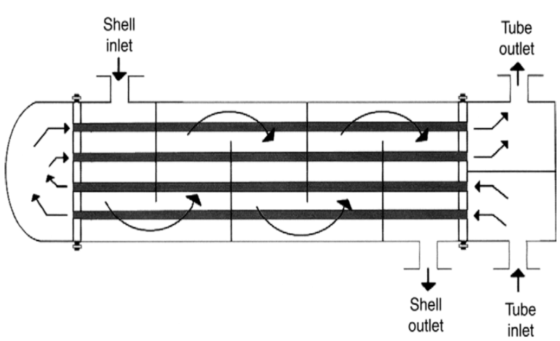

Figure 1: A typical Shell and Tube Heat Exchanger with one shell pass and two tube passes

As the two fluids in the heat exchanger that are at different temperatures, heat exchanger analysis and design therefore involve both convection and conduction. Two important problems in heat exchanger analysis are (1) rating existing heat exchangers and (2) sizing heat exchangers for a particular application. Rating involves the determination of the rate of heat transfer, the change in temperature of the two fluids and the pressure drop across the heat exchanger. Sizing involves selection of a specific heat exchanger from those currently available or determining the dimensions for the design of a new heat exchanger, given the required rate of heat transfer and allowable pressure drops (Thirumarimurugan *et al*, 2008).

There are design charts such as ∈-NTU (Effectiveness- Number of Transfer Unit) curves and LMTD (Logarithm Mean Temperature Difference) correction factor curves for the analysis of simple types of exchangers. Similar design charts do not exist for the analysis of complex heat exchangers with multiple entries on the shell side and complex flow arrangements (Ravikumaur *et al*, 1988).

In the recent past, some experts studied on the design, performance analysis and simulation studies on heat exchangers. Modeling and Simulation of Shell and Tube Heat Exchangers under Milk Fouling were carried out (Thirumarimurugan *et al*, 2008). Dynamic Model for Shell and Tube Heat Exchangers was discussed. Shell and Tube heat exchangers are applied where high temperature and pressure demands are significant and can be employed for a process requiring large quantities of fluid to be heated or cooled. Due to their design, these exchangers offer a large heat transfer area and provide high heat transfer efficiency in comparison with others

Attempts to generate such charts and curves were made by many researches. Saryal (Ravikumaur and others, 1988) suggested electro-analog models like lumped resistance, resistance-capacitance and hybrid model for heat exchanger calculations. He generated the dimensionless temperature distribution curve to simplify the heat exchanger calculations and showed that the use of auxiliary curves eliminated iterations in the process of calculation (Ravikumaur and others, 1988).

Gaddis and Schlunder (Ravikumaur and others, 1988) proposed a cell model to predict dimensionless temperature distribution in shell and tube exchangers. They generated ∈-NTU curves for shell and tube heat exchangers of various types. Mikhailov and Ozisik (Ravikumaur and others, 1988) adopted the model and introduced the procedures followed in finite element analysis to obtain the temperature distribution. They extended the model for the analysis of heat exchanger networks. They applied it to cases of assemblies of heat exchangers studied by Domingo and found that their results agreed well with the solution of Domingo (Ravikumaur and others, 1988).

MATERIALS AND METHODS

The thermal analysis of a shell and tube heat exchanger involves the determination of the overall heat-transfer coefficient from the individual film coefficients, and(Kern, 1965). The shell-side coefficient presents the greatest difficulty due to the very complex nature of the flow in the shell. In addition, if the exchanger employs multiple tube passes, then the LMTD correction factor must be used in calculating the mean temperature difference in the exchanger. For the turbulent flow regime $(Re \geq 10^4)$, the following correlation is widely used (Serth, 2007)

$$Nu = 0.027 Re^{0.8} Pr^{1/3} (\mu/\mu_w)^{0.14} \qquad (1)$$

where

$$Nu = NusseitNumber = hD/k$$
$$Re = ReynoldsNumber \equiv DV\rho/\mu$$
$$Pr = PrandtlNumber \equiv c_p\mu/k$$
$$D = insidepipediameter$$
$$V = averagefluidvelocity$$

$$c_p, \mu, \rho, k = fluidpropertiesevaluatedattheaveragebulkfluidtemperature$$
$$\mu_w = fluidviscosityevaluatedataveragewalltemperature$$

Equation (1) is valid for fluids with Prandtl numbers between 0.5 and 17,000, and for pipes with $L/D > 10$. However, for short pipes with $10 < L/D > 60$, the right-hand side of the equation is often multiplied by the factor $[1 + (D/L)^{2/3}]$ to correct for entrance and exit effects (Serth, 2007).

For laminar flow in circular pipes $(Re < 2100)$, the Seider-Tate correlation takes the form;

$$Nu = 1.86 [RePr\, D/L]^{1/3} (\mu/\mu_w)^{0.14}$$

This equation is valid for $0.5 < Pr < 17{,}000$ and $[RePr\, D/L]^{1/3} (\mu/\mu_w)^{0.14} > 2$.

For flow in the transition region $(2100 < Re < 10^4)$, the Hausen correlation is:

$$Nu = 0.116\left(Re^{2/3} - 125\right)Pr^{1/3}(\mu/\mu_w)^{0.14}\left(1 + (D/L)^{2/3}\right) \qquad (3)$$

In computing the tube-side coefficient, h_i, it is assumed that all tubes in the exchanger are exposed to the same thermal and hydraulic conditions. The value of h_i is then the same for all tubes, and the calculation can be made for a single tube. Equations (1), (2), or (3) were used, depending on the flow regime.The tube fluid heat transfer coefficient, h_i, can be calculated using;

$$h_i = \frac{Nu\, k}{D_i} \qquad (4)$$

The Delaware method (Serth, 2007) was used to compute the shell-side heat transfer coefficient, h_o.

In the equation for the overall heat transfer coefficient, the temperature difference, ΔT_m, is the mean temperature difference between the two fluid streams. Since U is independent of position along the exchanger, ΔT_m is the logarithmic mean temperature difference (Serth, 2007);

$$\Delta T_m = \Delta T_{ln} = LMTD = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln\left(\frac{T_1 - t_2}{T_2 - t_1}\right)} \qquad (5)$$

Equation (5) is valid regardless of whether counter flow or parallel flow is employed.In multi-pass shell-and-tube exchangers, the flow pattern is a mixture of co-current and countercurrent flow. For this reason, the mean temperature difference is derivedby introducing a correction factor, F, which is termed the LMTD correction factor;

$$\Delta T_m = F\left(\Delta T_{ln}\right)_{cf} \qquad (6)$$

The correction factor is a function of the shell and tube fluid temperatures, and the number of tube and shell passes. This is corrected using two dimensionless temperature ratios (Serth, 2007); let $N = Numberofshell - sidepasses$, then

$$P = \frac{t_2 - t_1}{T_1 - t_1}; \qquad R = \frac{T_1 - T_2}{t_2 - t_1} \qquad (7)$$

For $R \neq 1$, (8)

$$F = \frac{\sqrt{R^2+1}\ln\left(\frac{1-S}{1-RS}\right)}{(R-1)\ln\left[\frac{2-S\left(R+1-\sqrt{R^2+1}\right)}{2-S\left(R+1+\sqrt{R^2+1}\right)}\right]} \quad (9)$$

For $R = 1$,

$$F = \frac{S\sqrt{2}}{(1-S)\ln\left[\frac{2-S\left(2-\sqrt{2}\right)}{2-S\left(2+\sqrt{2}\right)}\right]} \quad (10)$$

The required overall heat transfer coefficient is given as;

$$U_{req} = \frac{q}{AF(\Delta T_{ln})_{cf}} \quad (11)$$

The clean overall heat transfer coefficient is given as;

$$U_c = \left[\frac{D_o}{h_i D_i} + \frac{D_o \ln(D_o/D_i)}{2k} + \frac{1}{h_o}\right]^{-1} \quad (12)$$

And the design overall heat transfer coefficient is given as;

$$U_D = (1/U_C + R_D)^{-1} \quad (13)$$

The effect of fouling is allowed for in the design by including the inside and outside fouling coefficients. Kern (1965) presented typical values for the fouling factors for common process service fluids used in plane tubes (not finned tubes). The fouling factor for the exchanger is given as (Serth, 2007);

$$R_D = R_{Di}(D_o/D_i) + R_{Do}$$

Design problems frequently include specifications of the maximum allowable pressure drops in the two streams. In that case, pressure drops for both streams would have to be calculated in order to determine the hydraulic suitability of the heat exchanger.

The pressure drop due to fluid friction in the tubes is given by Equation (15) with the length of the flow path set to the tube length times the number of tube passes (Serth, 2007).

$$\Delta P_f = \frac{f n_p L G^2}{2\rho D_i s \phi} \quad (15)$$

Where

ΔP_f =Pressure drop (Pa)

f =Darcy friction factor (dimensionless)

n_p =Number of tube passes (dimensionless)

L =Tube length *(m)*

G =Mass flux*(kg/s. m^2)*

D_i =Tubeinsidediameter*(m)*

ρ =Density of water (*kg/m^3*)

s =Fluid specific gravity (dimensionless)

ϕ =Viscosity correction factor (dimensionless)

$\phi = (\mu/\mu_w)^{0.14}$ for turbulent or transition flow

$\phi = (\mu/\mu_w)^{0.25}$ for laminar flow

For laminar flow, the friction factor is given by

$$f = \frac{64}{Re} \quad (16)$$

For turbulent flow, the following equation can be used for Re≥3000:

$$f = 0.4137 Re^{-0.2585} \quad (17)$$

The minor losses on the tube side are estimated using the following equation:

$$\Delta P_r = 5.0 \times 10^{-4} \, \alpha_r G^2/s$$

where α_r is the number of velocity heads allocated for minor losses

Serth (2007) proposed the following expression for computing the shell-side pressure drop:

$$\Delta P_f = \frac{f G^2 d_s (n_b + 1)}{2\rho d_e s \phi} \quad (19)$$

The shell-side friction factor is given by the formula:

$$f = 144\{f_1 - 1.25(1 - B/d_s)(f_1 - f_2)\} \quad (20)$$

An approximate equation for f_1 and f_2 are as follows.

For $Re \geq 1000$,

$$f_1 = (0.0076 + 0.000166 d_s) Re^{-0.125} \quad (8 \leq d_s \leq 42) \quad (21)$$

$$f_2 = (0.0016 + 5.8 \times 10^{-5} d_s) Re^{-0.157} \quad (8 \leq d_s \leq 23.25) \quad (22)$$

For $Re < 1000$,

$$f_1 = exp[0.092(\ln Re)^2 - 1.48 \ln Re - 0.000526 d_s^2 + 0.047 d_s - 0.338] \quad (8 \leq d_s \leq 42) \quad (23)$$

$$f_2 = exp[0.123(\ln Re)^2 - 1.78 \ln Re - 0.00132 d_s^2 + 0.0678 d_s - 1.34] \quad (8 \leq d_s \leq 23.25) \quad (25)$$

RESULTS AND DISCUSSION

The program was evaluated with data obtained from five industrial heat exchangers. The program was written and implemented in MATLAB. It is required to cool one fluid (either in the shell or tube side) by exchanging heat with the other. Table 1 shows the working fluids, fluid physical properties and the fluid performance data for the five heat exchangers. It was required to evaluate the heat exchangers if thermally and hydraulically suitable for this service they are being used for.

The results obtained after implementation of the program for the five heat exchangers are shown in Table 2.

Table 1: Input data for rating of the Heat Exchangers

INPUT DATA												
			Ex. 1		**Ex. 2**		**Ex. 3**		**Ex. 4**		**Ex. 5**	
S/N	**Parameters**	**Units**	**Shell Side Fluid**	**Tube Side Fluid**	**Shell Side Fluid**	**Tube Side Fluid**	**Shell Side Fluid**	**Tube Side Fluid**	**Shell Side Fluid**	**Tube Side Fluid**	**Shell Side Fluid**	**Tube Side Fluid**
A	**Physical Properties**											
1	Specific Gravity	-	0.73	0.82	0.82	0.73	0.85	0.82	0.85	0.82	0.73	0.82
2	Viscosity	Ns/m^2	0.43	3.2	3.2	0.43	0.17	3.2	0.17	3.2	0.43	3.2
3	Heat Capacity	kJ/kgK	2.47	2.05	2.05	2.47	2.28	2.05	2.28	2.05	2.47	2.05
4	Thermal Conductivity	W/mK	0.132	0.134	0.134	0.132	0.125	0.134	0.125	0.134	0.132	0.134
5	Thermal Conductivity at the wall temperature	W/mK	55		55		45		45		55	
B	**Performance Data**											
1	Fluid Stream	-	KERO	CRUDE	CRUDE	KEROPA	HDO	CRUDE	LDO	CRUDE	KERO	CRUDE
2	Inlet Temperature	K	441	365	414	457	581	382	480	365	506	431
3	Outlet Temperature	K	388	382	431	428	406	402	395	411	441	441
4	Mass Flow Rate	kg/s	32.23	116.14	232.29	125.5	10.9	116.14	56	116.14	32.23	232.29
5	Length of Tube	m		5.56		5.59		5.56		5.56		5.56
6	Outside Diameter	m		0.01905		0.01905		0.01905		0.01905		0.01905
7	Inside Diameter	m	1.1		1.05		1.15		1.1		1.1	
8	Baffle Space	m	0.22		0.32		0.23		0.22		0.22	
9	Passes	-	1	2	1	2	1	2	1	2	1	2
10	Number of Tubes	-		1256		1132		1388		1256		1256
11	Pitch	m		0.025		0.025		0.025		0.025		0.025
12	Fouling Factor	-	0.0002	0.0008	0.0006	0.0002	0.0006	0.0008	0.0004	0.0008	0.0002	0.0004
13	Pressure Drop	kpa	19.620	34.335	137.340	29.430	9.810	34.335	58.860	63.765	9.810	29.430
14	Baffle Cut	-	25		25		25		25		25	
15	Birmingham Wire Gage (BWG)	-	16		16		16		16		16	

Table 2: Performance Result

			Ex. 1	**Ex. 2**	**Ex. 3**	**Ex. 4**	**Ex. 5**
S/N	**Quantity**	**Units**	**Values**				
1	Heat load	kW	4200	7460	4349	10853	5174.5
2	Log mean temp. difference	K	38.21	19.39	77.14	46.82	29.38
3	Heat transfer area	m^2	418	378.76	461.92	417.99	417.99
4	Shell side film coefficient	kW/(m^{2}K)	14.20	70.51	8.95	18.08	14.2
5	Tube side film coefficient	kW/(m^2.K)	13.70	11.83	13.25	13.69	17.25
6	Design fouling factor	-	0.001	0.0009	0.0015	0.0013	0.0006
7	Required overall coefficient	kW/(m^{2}K)	0.3033	1.4328	0.1415	1.1878	0.5128
8	Clean overall coefficient	kW/(m^{2}K)	6.297	6.622	4.925	6.960	7.113
9	Design overall coefficient	kW/(m^{2}K)	6.26	6.58	4.89	6.90	7.081
10	Allowable shell side pressure drop	kPa	19.620	137.340	9.810	58.860	9.810
11	Allowable tube side pressure drop	kPa	34.335	29.430	34.335	63.765	29.430
12	Shell side pressure drop	kPa	0.6823	13.237	0.0718	0.5568	0.6822
13	Tube side pressure drop	kPa	4.2345	0.3962	3.4672	4.2345	8.4737

The results in Table 2 shows that the clean and design overall coefficients are greater than the required overall coefficient for the five heat exchangers. This implies that the heat exchangers are thermally suitable for the service they are being used for. Also, since the shell and tube side pressure drops are greater than the allowable pressure drop, the heat exchangers are hydraulically suitable for the service they are being used for

Table 3a: Evaluation of the Performance Analysis

Exchanger Number	**Ex. 1**			**Ex. 2**			**Ex. 3**		
Parameters	**Actual Value**	**Calc. Value**	**% Error**	**Actual Value**	**Calc. Value**	**% Error**	**Actual Value**	**Calc. Value**	**% Error**
Shell Side Fluid	KERO	-	-	CRUDE	-	-	HDO	-	-
Tube Side Fluid	CRUDE	-	-	KERO PA	-	-	CRUDE	-	-
Required Overall Heat Transfer Coefficient kW/(m^2.K)	0.3033	0.2999	1.121	0.3916	1.4328	72.669	0.1737	0.1415	18.5377
Heat Load (Duty) kW	4140.28	4200		9536.6	7460		5070.68	4349.1	
Heat Transfer Area (m^2)	418	417.990	0.00239	379	378.756	0.06438	460	461.918	0.4152

Table 3b: Evaluation of the Performance Analysis

Exchanger Number	**Ex. 4**			**Ex. 5**		
Parameters	**Actual Value**	**Calc. Value**	**% Error**	**Actual Value**	**Calc. Value**	**% Error**
Shell Side Fluid	LDO	-	-	KERO	-	-
Tube Side Fluid	CRUDE	-	-	CRUDE	-	-
Required Overall Heat Transfer Coefficient kW/(m^2.K)	0.3341	1.1878	71.8724	0.2928	0.2565	12.3975
Heat Load (Duty) kW	11734.67	10853.0		5628.92		
Heat Transfer Area (m^2)	418	417.990	0.00239	418	417.99	0.00239

CONCLUSION

A computer program in MATLAB was developed to evaluate the performance of shell and tubeheat exchangers for their thermal and hydraulic suitability. The Program tested with field data from five industrial exchangers (Ex. 1, Ex. 2, Ex. 3, Ex. 4 and Ex. 5) showed that the result obtained, compares reasonably with the actual performance data, thus, demonstrating that the program is reliable and can be applied in the performance analysis of shell and tube heat exchangers.

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