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Performance Assessment of Shell and Tube Heat Exchanger: A Case Study

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Abstract- This paper evaluated the performance of a counter flow heat exchanger in a pasta processing plant. The data obtained from the plant were analysed by applying various energy equations. A logarithmic mean temperature difference method was used to assess the effect of the performance parameters (heat duty, overall heat transfer coefficient, capacity ratio, temperature ratio, temperature range, pressure drop and logarithmic mean temperature difference) on the effectiveness and the effect on the fouling factor. It was found that the calculated value for the effectiveness deviated from the design value by 60%. The calculated values for heat duty, overall heat transfer coefficient, capacity ratio, effectiveness, temperature ratio, logarithmic mean temperature difference and pressure drop are 31838.96kW, 615.85W/m^{2}K, 1.87, 0.18, 0.27, 47.93K and 240.39kPa while for the design values are 52500kW, 680.50W/m^{2}K, 1.08, 0.3, 1.5, 35.65K and 170kPa respectively. The heat duty and overall heat transfer coefficient deviated from the design values by 60.65% and 87.92% respectively. The capacity ratio and temperature ratio also deviated from their design values by 57.75% and 18% respectively. The logarithmic mean temperature difference and the pressure drop in the tube side were found to be 75.64% and 70.72% less than the design values. These deviations are indication of fouling in the walls of the heat exchanger as the fouling factor also showed an increase of 78.26%. This work therefore justifies that the effectiveness of the heat exchanger depend on the overall heat transfer coefficient, heat duty and temperature ranges of the heat exchanger.

Keywords- *Performance Evaluation, Effectiveness, Fouling Factor, Dirty Heat Transfer Coefficient, Temperature Ratio*

I. INTRODUCTION

Heat transfer is the physics of thermal energy that deals with the rate of exchange of heat between two bodies. Heat transfer is commonly linked with fluid dynamics and it also supplements the law of thermodynamics by providing additional rules to establish energy transfer rates (Hasu & Rao, 2017). Energy utilization is the main stay of the world economy and it plays a key role in the nation's growth. The aim of every engineer is to improve processes and increase efficiency so as to further develop renewable source of energy (Kevin, 1998).

Heat exchangers are devices used to transport heat between two or more fluid streams at different temperatures. Heat exchangers find extensive use in power generation, chemical processing, electronics cooling, air-conditioning, refrigeration, and automotive applications etc.

Heat exchanger is an instrument that is used to transport heat energy between two fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperature and in thermal contact (Kaplesh & Chopra, 2013).

Due to the large number of heat exchanger designs, a classification set-up was devised based upon the basic operation, construction, heat transfer, and flow arrangements (Kakac & Liu 1998). Heat exchangers are considered to be one of the most useful devices of mechanical systems in modern society where sophisticated machines are used in almost all industrial processes. Almost all industrial processes involve the transport of heat and more often, it is required that the heat transport process be controlled. For that, the device that is used to achieve this controlling action is the heat exchanger.

The objectives of the heat exchanger is to recover or turn-down heat or sterilize, pasteurize, fractionate, distill, concentrate, crystallize or control a process fluid(Rao & Savsaric, 2012). Heat exchanger comes in different types and specifications depending on the specified heating or cooling output. Some common types of heat exchangers are finned and unfinned tubular, plate and frame, pate-fin, microchannel, cooling towers and shell and tube heat exchangers

Material coating on the surfaces of the heat exchanger tubes may add more thermal resistances to heat transport. Such accumulations, which are detrimental to the heat exchange process, are known as fouling. Fouling can be caused by a wide range of reasons and may significantly influence heat exchanger performance.

This work analysed the performance of shell and tube heat ex-changer (STHE), which are most commonly used type in oil refineries, chemical processing and also for process industries. The STHE dispenses a relatively huge quantity of heat per unit volume and also it can be designed to match nearly all heat transport service (Serth, 2007). Shell and tube heat exchangers can either be horizontal or vertical, for which the horizontal ones can be single or stacked in multi-units. The STHE can be further subdivided in three categories: U-tube, floating head (single pass straight), and fixed head (two pass straight) heat exchangers.

Shell and Tube heat exchanger on Fig: 1 does the same job like any other heat exchanger by passing heat from one fluid to another. Fig. 1 shows two inlets and two outlets pipes where the fluid flows in, in their respective inlet and exits the device at their outlets.

Figure 1. ABC shell and tube heat exchanger (Olam)

Pasta is a universal food mainly produced from wheat, rice and other cereals. Pasta has been in existence for century and has its roots from China and Italy, however, the industrial revolution did not start before the 1950s (De Vita, 2009). The eating of pasta for children and adults are now everyday meal. Pasta production is by three steps: Semolina is copolymerize during mixing, the material is condense and mould by means of sheeting or extruding and this shape is make stable by drying (Kratzer, 2007).

Pasta production is produced by blending milled wheat, eggs, water (for egg spaghetti or noodles) and sometime alternative ingredients. These ingredients are regularly added to an uninterrupted, high capacity anger extruder (the extrusion barrels are fitted with water cooling jacket to evaporate the heat during the removal process. The cooling jacket also helps to maintain a continuous evaporated temperature, which should be approximately 51°C (124°f). If the pasta is not too hot above 74°C (165°f), the pasta will be deface) which can be prepared with a type of dies that regulate the shape of the pasta. The pasta is then dried and packaged for market (Food and Agricultural Industry 2017).

II. Materials and Methods

A. *Thermal Analysis*

In thermal analysis of a shell and tube heat exchanger (STHE), these will involve the determination of the coefficient of heat transfer. A simplified method is use for evaluating the thermal performance of a STHE, which is the fouling factor.

B. *Fouling Factor*

This is the process that makes known to what extent the heat exchanger is working with recommendation to its original pattern. Accumulation and formation of unwanted material in the ex-changer decreases the heat capacity which also increases the drop in pressure of the heat ex-changer.

Fouling factor (R_f) can be determined by the equation (1):

$$R_f = \frac{1}{U_d} - \frac{1}{U_c} \tag{1}$$

Where

R_f = fouling factor or unit thermal resistance of deposit, W/m^{2}°C

U_d = overall heat transfer coefficient, dirty W/m^2 °C

U_c = overall heat transfer coefficient, clean W/m^2 °C

(The Engineering ToolBox, 2010)

C. *Dirty Overall Heat Transfer Coefficient, (U_d)*

The dirty overall heat transfer coefficient is defined as the calculated heat transfer coefficient based on the outside surface area of a tube of the heat exchanger and can be calculated from the following equation.

$$U_d = \frac{Q_{hd}}{A_{ht} F_c \Delta T_{lm}} = \frac{Q_{hd}}{A_{ht} \Delta T_{lc}} \tag{2}$$

Where

Q_{hd} = the actual heat duty, W

A_{ht} = heat transfer surface area, m^2

F_c = correction factor

ΔT_{lm} = Logarithmic Mean Temperature Difference, °C.

ΔT_{lc} = Corrected Logarithmic Mean Temperature Difference, °C

(Sulaiman *et al*, 2016)

D. *Logarithmic Mean Temperature Difference (ΔT_{lm})*

Logarithmic mean temperature difference represents the average temperature different between the two heat transfer fluids over the heat exchanger length.

It is perhaps the most outstanding method used to evaluate heat transfer in heat exchangers and is described in the Tubular Exchanger Manufacturers Association (TEMA) Standards.

$$\Delta T_{lm} = \frac{\Delta T_1 + \Delta T_2}{In(\frac{\Delta T_1}{\Delta T_2})} \quad (3a)$$

The equation (3a) is for parallel flow heat exchanger while the equation (3b) is for counter flow heat exchanger.

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{In(\frac{\Delta T_1}{\Delta T_2})} \quad (3b)$$

$$\Delta T_1 = T_{h,i} - T_{c,o} \quad (4)$$

$$\Delta T_2 = T_{h,o} - T_{c,i} \quad (5)$$

Where

$T_{h,i}$ = inlet temperature of hot fluid, °C.

$T_{h,o}$ = outlet temperature of hot fluid, °C.

(Cuneyt, 2017)

E. Heat Duty

Heat duty is define as the capacity of the heat exchanger equipment expressed in terms of heat transfer per unit time. It shows that the heat exchanger is capable of performing within this capacity in particular system. Heat duty can be defined as the heat gained by cold fluid which is equal to the heat loss of the hot fluid (Sulaiman *et al,* 2016).

Heat Duty (Q_{hd}) can be determined by the equation (6):

$$Q_{hd} = \frac{Q_{hd,h} + Q_{hd,c}}{2} \quad (6)$$

$Q_{hd,h}$ And $Q_{hd,c}$ can be obtain by using the energy balance equation:

$$Q_{hd,h} = \dot{m}_h(h_{h,i} - h_{h,o}) = \dot{m}_h c_{ph}(T_{h,i} - T_{h,o})$$

$$= \text{WC}\ (T_1 - T_2) \quad (7a)$$

$$Q_{hd,c} = \dot{m}_c(h_{c,i} - h_{c,o}) = \dot{m}_c c_{pc}(T_{c,i} - T_{c,o})$$

$$= \text{wc}\ (t_1 - t_2) \quad (7b)$$

Where

$Q_{hd,h}$ and $Q_{hd,c}$ = heat duty of hot stream and cold stream accordingly, kW.

$h_{h,i}$ and $h_{h,o}$ = inlet and outlet specific enthalpy of hot fluid, kJ/kg

$h_{c,i}$ and $h_{c,o}$ = specific enthalpy of inlet cold fluid and specific enthalpy of outlet cold fluid, kJ/kg.

$\dot{m}_h$ and $\dot{m}_c$ = the hot fluid and cold fluid mass flow rate, kg/s.

$c_{p,h}$ and $c_{p,c}$ = the hot fluid and cold fluid specific heat capacity, J/kgK.

$T_{c,i}$ and $T_{c,o}$ = the inlet cold fluid and outlet cold fluid, °C.

WC = mass flow rate and specific heat capacity of hot fluid.

wc = mass flow rate and specific heat capacity of cold fluid.

The logarithmic mean temperature difference (ΔT_{lm}) with the correction factor F_c is determined by using equation (8).

$$\Delta T_{lc} = \frac{\Delta_1 - \Delta_2}{In(\frac{\Delta_1}{\Delta_2})} \text{ x Fc} \quad (8)$$

Where

$\Delta_1 = T_{h,i} - T_{c,o}$ and $\Delta_2 = T_{h,o} - T_{c,i}$

F_c = correction factor

$$F_C = \frac{\sqrt{R^2}+1\ in(\frac{1-X}{1-RX}}{(R-1)\ in\{\frac{\frac{2}{X}-1-R+(\sqrt{R^2}+1)}{\frac{2}{X}-1-R-(\sqrt{R^2}+1}\}} \quad (9)$$

Where $X = \frac{1-(\frac{RP-1}{P-1})^{\frac{1}{NP}}}{R-(\frac{RP-1}{P-1})^{\frac{1}{NP}}}$

Where R = 1

$$F_c = \frac{(\frac{P\sqrt{2}}{1-P})}{In\{\frac{2-P(2-\sqrt{2})}{2-P(2+\sqrt{2})}\}} \quad (10)$$

P = temperature ratio

(Bowman *et al,* 1940)

F. Clean Overall Heat Transfer Coefficient, (U_c)

Clean overall heat transfer coefficient is defined as the calculated heat transfer coefficient based on the properties of the fluid, material thickness and the heat exchanger configuration.

$$U_c = [\frac{d_o}{h_i d_i} + \frac{d_o In({}^{d_o}/_{d_i})}{2k} + \frac{1}{h_o}]^{-1} \quad (11)$$

Where

d_o = outside diameter of tube, m.

d_i = inside diameter of tube, m.

h_o = outside fluid film heat transfer coefficient, W/m^2 °C.

h_i = inside fluid film heat transfer coefficient, W/m^2 °C.

k = thermal conductivity, W/m °C.

(WeBBuster z.org, 2019)

G. Temperature Ratio (P):

This is the temperature rise in cold fluid difference from the temperature of the inlet fluid.

$$P = \frac{t_{c,2} - t_{c,1}}{t_{h,1} - t_{c,1}} = \frac{\Delta T_c}{T_{h,1} - T_{c,1}} \quad (12)$$

(Rajput, 2007)

H. Capacity Ratio (R)

This is the ratio of the products of the mass flow rate multiply by the heat capacity of the fluids.

$$R = \frac{Temperature\ drop\ of\ the\ hot\ fluid}{Temperature\ rise\ in\ the\ cold\ fluid} \quad (13a)$$

$$R = \frac{\dot{m}_c}{\dot{m}_h} \times \frac{C_{p,c}}{C_{p,h}} = \frac{t_{h,1} - t_{h,2}}{t_{c,2} - t_{c,1}} = \frac{C_{min}}{C_{max}} = \frac{\Delta T_h}{\Delta T_c} \qquad (13b)$$

(Rajput, 2007)

I. Effectiveness (ε)

This is the ratio of cold fluid temperature range to that of the inlet temperature difference of the hot and cold fluid (Sulaiman *et al*, 2016). It can also be defined as the 'ratio of actual heat transfer to the maximum possible heat transfer' (Rajput, 2007)

Effectiveness (ε) of the heat exchanger can be determined by equation (14):

$$\varepsilon = \frac{Actual\ heat\ transfer}{maximum\ possible\ heat\ transfer} = \frac{Q_{ht}}{Q_{max}} \qquad (14)$$

Where

$$Q_{ht} = \dot{m}_c c_{pc}(T_{c,o} - T_{c,i}) \qquad (15)$$

$$Q_{max} = c_{min}(T_{h,i} - T_{c,i}) \qquad (16)$$

III. Results and Discussion

A. Input Data for Thermal Analysis

The input data for the heat exchanger are tabulated in Tables A and B. The results of the heat exchanger units are gathered by inputting the measured values into various equations in chapter two.

Heat Exchanger Input Data

TABLE I. Input Data for Rating of the Heat Exchanger

INPUT DATA				
			PASTA pre dryer	
S/No.	Parameters	Units	Shell side fluid	Tube side fluid
A	PHYSICAL PROPERTIES			
1	Density of fluid	Kg/m^3	2.208	978.53
2	Viscosity	Ns/m^2	1.55E-05	4.26E-04
3	Specific Heat Capacity	kJ/kgK	0.871	4.183
4	Thermal Conductivity	W/mK	0.0216	0.544
5	Thermal Conductivity at the wall temperature	W/mK	59	
B	PERFORMANCE DATA			
1	Fluid Stream	-----	WET CO_2	COOLING WATER
2	Inlet Temperature	K	375.54	310.46
3	Outlet Temperature	K	350.58	332.51
4	Mass Flow Rate	kg/s	17.16	615.85
5	Length of Tube	m	10.73	
6	Outside Diameter	m	---------	0.019
7	Inside Diameter	m	1.956	0.0159
8	Baffle Space	m	0.716	
9	Passes	----	1	2
10	Number of Tubes	----	--------	250
11	Pitch	m		0.0318
12	Fouling Factor	----		
13	Pressure Drop	kpa		
14	Baffle Cut	%	25	

Monthly Average Temperature

TABLE II. MONTHLY AVERAGE TEMPERATURE FOR PASTA PRE DRYER

S/NO.	Month (2016-2018)	Shell - side		Tube -side	
		Inlet T_{h1} (K)	Outlet T_{h2}(K)	Inlet T_{c1}(K)	Outlet T_{c2}(K)
1	January	401.5	360.8	315.1	343.9
2	February	399.01	360.4	320.2	342.4
3	March	398.71	360	323.3	343.9
4	April	401.24	365.5	321.5	340.5
5	May	402.1	360.8	321.5	343.4
6	June	401.8	360.4	321.1	341.9
7	July	399	361.6	318.5	343.4
8	August	400	360.5	318.5	341.9
9	September	400.2	360	320.2	343.5
10	October	401.79	359.4	325.3	343.4
11	November	402.4	358.6	321.5	343.9
12	December	400.3	358.5	321.5	342.4
13	January	401.2	350.8	318.1	345.9
14	February	400.3	350.4	318.5	340.5
15	March	402.3	348.6	320.5	342.4
16	April	404.3	370.8	318.1	341.9
17	May	400.1	370.4	320.2	342.4
18	June	400.4	370	325.3	340.5
19	July	404.3	365.5	323.5	342.4
20	August	396.2	360.8	325.5	341.9
21	September	397.4	360.4	318.1	342.4
22	October	400.2	361.6	318.5	341.9
23	November	401.3	360.5	318.1	340.5
24	December	400.2	360	320.2	343.4
25	January	400.2	359.4	321.3	341.9
26	February	400.5	358.6	321.5	342.4
27	March	401.6	358.5	321.5	345.9
28	April	400.2	350.8	318.1	340.5
29	May	400.1	350.4	318.5	342.4
30	June	401.3	348.6	320.5	342.15
31	July	400.2	360.42	320.46	341.9
AVE.		400.66	359.45	320.47	342.51

TABLE III. COMPARISON OF DATA BETWEEN CALCULATED VALUES AND DESIGN VALUES

S/No	Parameters	Units	Calculated Values	Design Values	Deviations Of Values From The Design Values (%)
1	Heat duty (Q_{hd})	kW	31838.96	52500	60.65
2	Overall heat transfer coeff. (U_o)	W/m^2K	615.85	680.50	87.92
3	Dirty heat transfer coeff.(U_d)	W/m^2K	419.90	550	76.31
4	Capacity ratio (R)	------	1.87	1.08	57.75
5	Effectiveness (ε)	------	0.18	0.3	60
6	Temperature Ratio (p)	----	0.27	1.5	18
7	Temperature range of hot fluid (ΔT_h)	K	40.20	88	45.68
8	Temperature range of cold fluid(ΔTc)	K	22.04	66	33.39
9	Pressure drop of shell side (ΔPs)	kPa	6.37	2.35	36.89
10	Pressure drop of tube side (ΔP_t)	kPa	240.39	170	70.72
11	LMTD (ΔT_{Lm})	K	47.93	35.65	75.64
12	Fouling factor (R_f)	m^2k/W	0.0023	0.0005	21.74

It is seen that the calculated values for the heat duty, overall heat transfer coefficient and temperature range of both hot and cold fluid reduces while the capacity ratio and logarithmic mean temperature difference increase. The calculated value of the heat duty deviated from the design value by 60.65% while that of the overall heat transfer deviates from the design value by 87.92%. These deviations will lead to low effectiveness and may be due to increase in fouling factor.

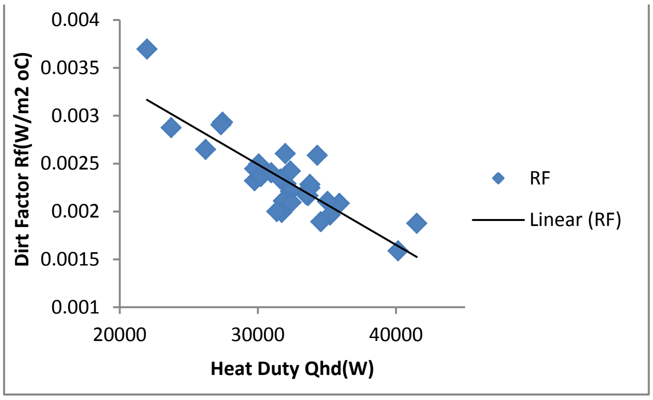

Figure 2. Variation of Dirt Factor with Heat Duty

From Fig. 2, the dirt factor increases as the heat duty decreases. This increase in dirt factor could be as a result of the decrease in temperature and also the drop in pressure in the inner walls of the exchanger. This shows that a pro-active maintenance and monitoring should be sustain to improve the performance as evaluated.

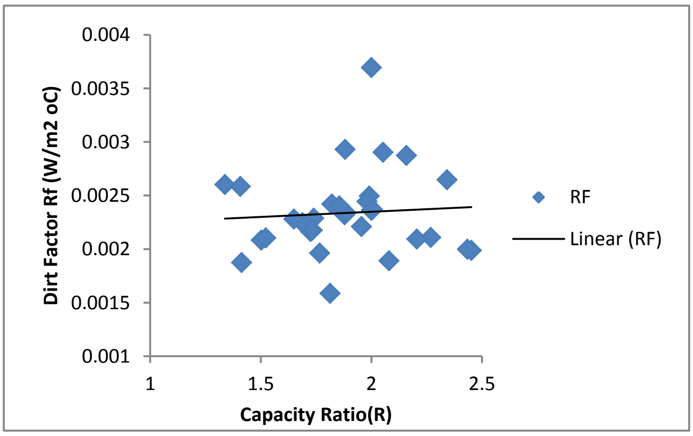

Figure 3. Variation of Dirt Factor with Capacity Ratio

From Fig. 3, it is observed that an increase in dirt factor increases the capacity ratio of the heat exchanger. This could be as a result of unwanted deposit in the walls of the heat exchanger.

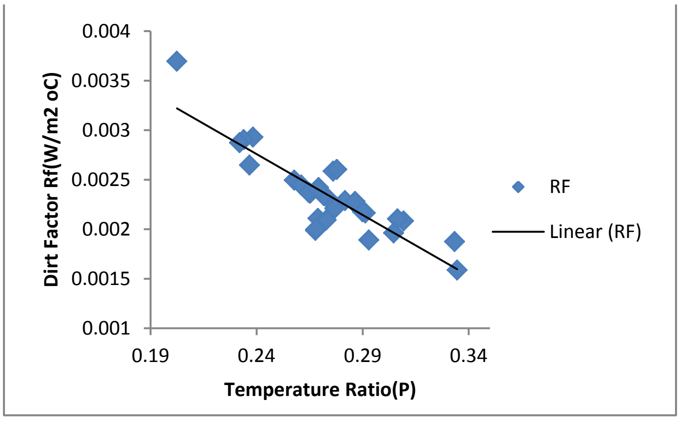

Figure 4. Variation of Dirt Factor with Temperature Ratio

From Fig. 4, it is observed that as the dirt factor increases the temperature ratio reduces and vise-versa. The increase in fouling factor could be as a result of dirt and unwanted deposit at the inner walls of the exchanger.

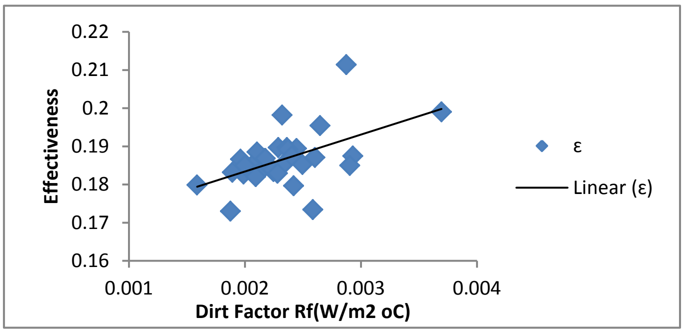

Figure 5. Variation of Effectiveness with Dirt Factor

From Fig. 5, it is observed that as the dirt factor increases the effectiveness reduces. This may be due to an unwanted deposit at the inner walls of the heat exchanger. This will reduce the effectiveness and increases the energy use and also the running costs.

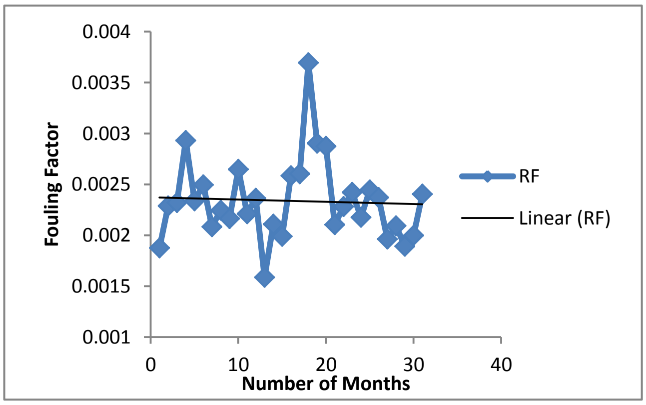

Figure 6. Variation of Dirt Factor with Number of Months

From Fig. 6, it is observed that there is a sharp increase in the fouling factor from the 18th months of operation and it is seen that the fouling factor have affected the heat exchanger. These can be controlled or even avoided by means of monitoring the flow velocity of the fluid so that dirt don't settle out.

IV. Conclusion

In this paper dirt factor trend method has been used to assess the performance of pasta pre dryer heat exchanger. The control parameters are calculated using various energy equations and their calculated values compared with the design values. The deviation in values may be due to the development of fouling in the walls of the exchanger.

From table C, it is noticeable that the fouling factor calculated deviated from its design value by 21.74% indicating that the heat exchanger device is considered fouled and need to be overhauled. From the above observations, it shows that a pro-active maintenance and monitoring should be carried out in every interval of 4 months, as to sustain and enhance the performance as calculated.

The performance of a single counter-current flow shell and tube heat exchanger in a pasta plant has been evaluated using logarithmic mean temperature difference (LMTD) method to monitor the dirt factor trend. From the performance of the heat exchanger unit analysis, it is appropriate to explain that effectiveness emanate from the following parameters (overall heat transfer coefficient, heat duty and temperature) of the exchanger.

Further recommendations are made as follows:

i. Statistical method and quality control approach should be employed to assess the performance of the exchanger unit, inorder to control changes that occur in the course of production and also detect when fouling begins.

ii. Additional investigation should be carried out to regulate the heat transport behavior of Nano-particles as well as how it can be used to boost the heat transfer coefficient, inorder to enhance the performance of the heat exchanger.

iii. More research should be carried out to control the baffle orientation and tube-pitch layout analysis, inorder to control the flow of fluid and the effect of vibration in the tube of the exchanger.

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