

**ARTICLE****Performance Assessment of Heat Exchangers for Process Heat Integration****Fenwicks Shombe Musonye<sup>1,\*</sup>, Hiram Ndiritu<sup>2</sup> and Robert Kinyua<sup>3</sup>**<sup>1</sup>Institute of Energy and Environmental Technology, Jomo Kenyatta University of Agriculture and Technology, Nairobi, Kenya<sup>2</sup>College of Engineering and Technology, Jomo Kenyatta University of Agriculture and Technology, Nairobi, Kenya<sup>3</sup>Office of the Deputy Vice Chancellor (Academic Affairs), Jomo Kenyatta University of Agriculture and Technology, Nairobi, Kenya

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**ABSTRACT**

Pinch Analysis is an attractive solution for reduction of thermal energy costs in thermo-chemical industries. In this approach, maximum internally recoverable heat is determined and a heat exchange network is designed to meet the recovery targets. The thermal performance of a heat exchanger over its lifetime is however a concern to industries. Thermal performance of a heat exchanger is affected by many factors which include the physical properties of the shell and tube materials, and the chemical properties of the heat transfer fluid. In this study, thermal performance of shell and tube heat exchangers designed to meet heat recovery targets in a Pinch Analysis study is simulated. The aim of this paper is to present predictions of thermal performances of shell and tube heat exchangers with different heat transfer fluids and geometries as they undergo fouling degradation. Engineering approaches based on thermodynamic analysis, heat balance and Kern Design equations, as well as what-if simulation modeling are used in this work. Shell and tube heat exchangers were designed to meet internal heat recovery targets for three process plants, A, B and C. These targets were published in a separate paper. The effects of degradation of the tubes-due to incremental growth of fouling resistance - on thermal performance of the exchanger were simulated using Visual Basic Analysis (VBA). Overall, it was found that growth in fouling reduces thermal efficiency of shell and tube heat exchangers with an exponential relationship. An increase of 100% of fouling resistance leads to an average reduction of 0.37% heat transfer. Higher values of logarithmic mean temperature difference (LMTD) and higher ratios of external diameter to internal diameter of the exchanger tubes amplify the effect of fouling growth on thermal performance of the exchangers. The results of this work can be applied in pinch analysis, during design of heat exchangers to meet the internal heat recovery targets, especially in predicting how fouling growth can affect these targets. This can also be useful in helping operators of shell and tube heat exchangers to determine cleaning intervals of the exchangers to avoid heat transfer loss.

**KEYWORDS**

Pinch analysis; internal heat recovery; thermal performance; fouling resistance; fouling growth; what if simulation; shell and tube heat exchangers



## Nomenclature

Symbol	Property	Units
T <sub>ci</sub>	Initial temperature of cold stream	°C
T <sub>co</sub>	Target temperature of cold stream	°C
T <sub>hi</sub>	Initial temperature of hot stream	°C
T <sub>ho</sub>	Target temperature of hot stream	°C
m <sub>ic</sub>	Mass flow rate of the cold stream	kg/s
m <sub>ih</sub>	Mass flow rate of the hot stream	kg/s
ρ <sub>c</sub>	Density of the cold stream	kg/m <sup>3</sup>
ρ <sub>h</sub>	Density of the hot stream	kg/m <sup>3</sup>
C <sub>pc</sub>	Specific heat capacity of the cold stream	kJ/kg.°C
C <sub>ph</sub>	Specific heat capacity of the hot stream	kJ/kg.°C
S <sub>c</sub>	Specific gravity of cold stream	Dimensionless
S <sub>h</sub>	Specific gravity of hot stream	Dimensionless
μ <sub>c</sub>	Dynamic viscosity of cold stream	kg·m <sup>-1</sup> ·s <sup>-1</sup>
μ <sub>h</sub>	Dynamic viscosity of hot stream	kg·m <sup>-1</sup> ·s <sup>-1</sup>
U <sub>ass</sub>	Assumed overall coefficient of heat transfer	kW/m <sup>2</sup> ·°C
K <sub>s</sub>	Thermal conductivity of shell side fluid	kW/m·°C
K <sub>t</sub>	Thermal conductivity of tube side fluid	kW/m·°C
K <sub>w</sub>	Thermal conductivity of tube material	kW/m·°C
R <sub>t</sub>	Tube side fouling factor	m <sup>2</sup> ·K/W
R <sub>s</sub>	Shell side fouling factor	m <sup>2</sup> ·K/W

## 1 Introduction

Costs of production of goods and services have been increasing, partly due to increase in the energy costs. The energy cost is associated both with the fuel and electricity purchase prices and the environmental degradation [1]. Pinch analysis is one of the process integration tools used to reduce energy consumption in thermo-chemical plants. The tool reduces costs through minimization of energy required in the processes [2]. It involves process energy balance, energy targeting and design of heat exchanger network to meet the computed energy targets [3].

Pinch analysis was first used in 1978 [4] and since then, various improvements have been suggested, with some of them successfully implemented. Some of these improvements and suggestions were highlighted in work published by [5], which include the shift from process to site wide pinch analysis, the use of stream specific values of temperature difference and a combination of principles of mathematical programming and pinch analysis. Significant improvements resulting from these suggestions have been demonstrated to work in energy targeting phases of pinch analysis. These include a combination of energy and pinch analysis [6,7], total site targeting and the use of stream specific minimum temperature difference [8].

Published work on design and simulation of heat recovery network, for example, Becker [9] and Harkin et al. [10] demonstrated that there is potential improvement in combination of mathematical programming and pinch analysis. However, as observed by reference [5], these designs in pinch analysis have not considered all attendant state variables of heat exchanger systems. Various algorithms have been

developed to make the designing and incorporation of these variables easier. Notable efforts can be found in [11], for alternative Mixed Integer Linear Programming (MILP) formulations and [12], for basic MILP formulations. The authors used MILP equations and executed them using General Algebraic Modelling System (GAMS). This is perhaps one of the important steps in improving the shell and tube heat exchanger design in pinch analysis.

From the available published works in pinch analysis [6–12], little attempt has been made to investigate the ability of shell and tube heat exchangers to meet the computed heat recovery targets, over their lifetime. Thermal performance of the exchangers is bound to change with time, especially in light of growth of fouling, as this reduces the heat transfer rate [13].

Some aspects of fouling growth and performance of heat exchangers have been investigated in studies unrelated to pinch analysis. They however have not centered on thermal performance of shell and tube heat exchanger and neither have such studies elaborated how tube thickness and temperature gradient contribute to the fouling resistance effects severity on thermal performance. In [14] for example, investigation of the effects of sea water fouling on thermal efficiency of a nuclear power plant was reported. This revealed that a change in fouling factors from 0.00015 to 0.00035  $\text{m}^2\cdot\text{K}/\text{W}$  resulted in decrease of the output power and thermal efficiency of the plant by 1.36% and 0.448%, respectively. Effects of fouling of fills in a cooling tower have been studied by [15], where the cooling tower effectiveness and the outlet water temperatures were modelled. This study used experimental data to validate the model. It was concluded that fill fouling had a bigger impact in medium sized as opposed to small sized cooling towers, with temperature decreases of 18.3 per cent and 23.5 per cent. Other studies have focused on fouling, but they were concerned with computational fluid dynamics modelling of fouling growth [16] and the effects of flow velocity on rate of fouling [17]. Effects of fouling factor on performance of shell and tube heat exchangers-in pinch analysis-can be visualized through the use of Kern Design Equations. The published studies of pinch analysis discussed here do not take into account these equations, thus missing out on capturing both the actual exchanger area and the fouling factor effects on thermal performance of the same.

This paper seeks to contribute to the research on fouling by assessing the effects of tube side fouling growth on thermal performance of shell and tube heat exchangers designed using Kern Design Equations. This assessment includes visual representation of the thermal performance curve and analysis of the effects of tube thickness and fluid temperature difference on the severity of fouling resistance in heat transfer. The assessment is based on selected 5 shell and tube heat exchangers for three plants, A, B and C. Pinch analysis was carried out in these plants and heat recovery targets were determined. The targets were published in a separate paper [18].

### ***1.1 Pinch Analysis and Shell and Tube Heat Exchangers***

Pinch analysis is a heat integration approach used to internally recover heat in thermo-chemical process plants. The analysis involves heat balancing, where the total heating loads for each process stream are determined, using the First Law of Thermodynamics. The second stage involves determination of maximum possible internally recoverable heat, using the heat cascading method. A heat exchange network is then developed to meet the computed recovery targets. Detailed description of pinch analysis can be found in [3] and [4].

A heat exchange network is made up of heat exchangers which recover heat from hot streams to cold streams. Shell and tube heat exchangers are the common types of exchangers used in thermo-chemical industries. They are described in [19]. During heat recovery, the fluids in the hot and cold streams in a processing plant are conveyed in the tube and the shell sides of the exchanger. Allocation of the fluids to either sides of the exchanger is based on factors that vary from process to process [20]. In some processes, hot fluid can be conveyed in the tube side while in others, it can be conveyed in the shell side.

Fouling in heat exchangers contributes to heat transfer resistance. Growth of fouling depends on the chemical characteristics of the heat transfer fluid, the velocity profile and the temperature [21].

## 2 Methods

This study was executed in three stages:

- i) Extraction of design from the previous pinch analysis study
- ii) Determination of shell and tube heat exchanger areas using Kern Design Equations
- iii) What-if simulation of effects of fouling on quantity of heat transferred

### 2.1 Data from Previous Study

This work is a continuation of a pinch analysis study carried out on a sulphonation plant, a dairy processing plant and an alcohol distillery plant. The plants were named A, B and C, respectively, and they were described in [18]. The data included the chemical and physical properties of the process stream fluids from the three plants, the heat exchanger material properties and the quantity of heat to be recovered by each exchanger.

### 2.2 Determination of Heat Exchanger Areas

Kern Design Equations were used to take into account all the variables attendant to shell and tube exchanger heat transfer process. The starting point in design of these exchangers is expressed by:

$$A = \frac{Q}{U_{ass} \times LMTD} \quad (1)$$

where  $U_{ass}$  is the estimated overall coefficient of heat transfer. Estimates of  $U$  have been published for different fluids.

LMTD is the logarithmic mean temperature difference between the medium exchanging heat, computed as:

$$LMTD = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln \left\{ \frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}} \right\}} \quad (2)$$

Here,

$T_{hi}$  is the inlet temperature of the hot fluid

$T_{co}$  is the outlet temperature of the cold fluid

$T_{ho}$  is the outlet temperature of the hot fluid

$T_{ci}$  is the inlet temperature of the cold fluid

$Q$ , the heat flux to be exchanged, is determined by the process streams variables. The hot or the cold stream can determine this value.

$$Q = \dot{m}(T_{ho} - T_{hi})C_p \quad (3)$$

$\dot{m}$  is the mass flow rate of the hot fluid

$C_p$  is the specific heat capacity of the hot fluid

Even though Eqs. (1–3) are useful in estimation of the required heat exchanger area, they do not represent the actual thermo-fluid characteristics of flow in a heat exchanger. The actual characteristics are considered by Kern equations of heat exchanger design [22]. These equations modify the LMTD and

$U_{ass}$ , and then impose some design constraints on the fluid velocity, the Reynold's number and the pressure drops. They as well allow selection of discrete geometrical variables.

Eq. (1) is therefore modified to become;

$$A = \frac{Q}{U_c \times F_t \times LMTD} \tag{4}$$

where  $U_c$  is the calculated overall coefficient of heat transfer and  $F_t$  is the geometric correction factor. This factor corrects for true counter flow characteristics and can be determined analytically or graphically.  $U_c$  is determined by:

$$U_c = 1 \div \left[ \left( \frac{1}{h_s} \right) + R_s + \left( \frac{A_o}{A_i} \right) \left\{ \frac{d_{to} - d_{ti}}{2k_m} \right\} + \frac{A_o}{A_i} \left( \frac{1}{h_t} \right) + \frac{A_o}{A_i} R_t \right] \tag{5}$$

In Eq. (5), all the true thermo-fluid characteristics of a heat exchanger have been captured. The variables are as follows:

$h_s$	Shell side coefficient of heat transfer	$A_i$	Tube internal area	$R_t$	Tube side fouling factor
$h_t$	Tube side coefficient of heat transfer	$d_{to}$	Tube external diameter	$K_m$	Thermal conductivity of the tube material
$R_s$	Shell side fouling factor	$d_{ti}$	Tube internal diameter	$A_o$	Tube external area

$d_{to}$  and  $d_{ti}$  are discrete variables, determined by Tubular Exchanger Manufacturers Association (TEMA) standards [23]. From these,  $A_o$  and  $A_i$  are derived. Thermal conductivity  $k_m$  is also fixed, according to the material used. Fouling factors  $R_s$  and  $R_t$  are ranges of values, provided in literature and the designer has the latitude of varying them. They are dependent of the heat transfer fluid under consideration, the surface of the tube and shell and the flow characteristics [24].

Tube side heat transfer coefficient  $h_t$  is determined by:

$$h_t = \frac{Nu_t \cdot k_t}{d_{ti}} \tag{6}$$

where  $k_t$  is the thermal conductivity of the tube side fluid and  $Nu_t$ , the tube side Nusselt number is computed using the Dittus-Boelter correlation:

$$Nu_t = 0.023 Re_t^{0.8} \cdot Pr_t^{0.3} \tag{7}$$

$Pr_t$ , the Prandtl number, is given by:

$$Pr_t = \frac{C_{pc} \cdot \mu_c}{k_t} \tag{8}$$

where  $k_t$  is thermal conductivity of the tube side fluid,  $C_{pc}$  is the constant pressure specific heat capacity of the tube side fluid and  $\mu_t$  is the tube side dynamic viscosity.

The tube side Reynold's number,  $Re_t$ , is:

$$Re_t = \frac{4\dot{m}_c \left( \frac{n_p}{n_t} \right)}{\pi d_{ti} \mu_c} \quad (9)$$

Here,  $\dot{m}_c$  is the mass flow rate of the tube side fluid,  $n_t$  is the number of tubes and  $n_p$  is the number of shell passes. They are discrete too, determined by TEMA standards. Initial value of  $n_t$  can be determined using the formula:

$$n_t = \frac{A}{\pi \cdot d_{to} \cdot L} \quad (10)$$

$L$ , the length of the exchange tubes, is discrete and obtained from the TEMA standards.

$Re_t$  is a design constraint. For design to be acceptable, this value should be greater than 10000. Turbulent flow is desired for optimal convective transfer of heat [15]. This value can be manipulated only through selection of  $d_{ti}$ ,  $n_p$  and  $n_t$ . Once this condition is satisfied, the tube side coefficient of heat transfer is determined.

The shell side heat transfer coefficient,  $h_s$ , is determined by:

$$h_s = \frac{Nu_s \cdot k_s}{d_{to}} \quad (11)$$

Here,  $k_s$  is the thermal conductivity of the shell side fluid and  $Nu_s$  is the shell side Nusselt number.  $Nu_s$  is determined by the Nusselt number equation for turbulent sensible flow:

$$Nu_s = 0.36 \times Re_s^{0.5} \cdot Pr_s^{0.33} \quad (12)$$

Shell side Prandtl number  $Pr_s$  is calculated using:

$$Pr_s = \frac{C_{ph} \mu_h}{k_s} \quad (13)$$

where  $C_{ph}$  and  $\mu_h$  are the isobaric specific heat capacity and the dynamic viscosity of the shell side fluid, in that order.

Shell side Reynold's number is given by:

$$Re_s = \frac{D_e \cdot G_s}{\mu_h} \quad (14)$$

$G_s$  is the shell side fluid ratio of mass flow rate. It is a function of  $\dot{m}_h$  the shell side mass flow rate and cross-sectional flow area  $a_s$ .

$$G_s = \frac{\dot{m}_h}{a_s} \quad (15)$$

The cross-sectional flow area  $a_s$  is:

$$a_s = \frac{C \cdot B \cdot D_s}{PT} \quad (16)$$

$$C = PT - d_{to} \quad (17)$$

Here,  $d_{i0}$  is the external diameter of the exchanger tubes.

$$B = 0.5D_s \tag{18}$$

The equivalent shell diameter  $D_e$ , for square pitch, is calculated using:

$$D_e = \frac{4 \left( PT^2 - \left( \frac{\pi}{4} \right) d_{i0}^2 \right)}{\pi d_{i0}} \tag{19}$$

Eq. (1) was used to determine the required area of a heat exchanger, without considering the TEMA standards and the Kern Heat Exchanger design considerations. Eqs. (4) and (5) were used to determine the effects of fouling on thermal performance of heat exchangers.

### 2.3 Simulation of Effects of Fouling on Thermal Performance

For each heat exchanger, tube side fluid fouling factors were gradually increased, from the maximum design recommended value, in incremental steps of 5%, to a maximum increase of 100%. The values were obtained from [24]. The factors were varied in Eq. (5). The result was applied in Eq. (4) to get the change in the required heat exchanger area. To get the heat loss due to increased fouling resistance, a difference in the designed maximum allowable fouling resistance and the required area under fouling resistance that exceeds the design limit was used in Eq. (4), and applied as follows:

$$Q_{loss} = A_{additional} \times U_{cf} \times F_t \times LMTD \tag{20}$$

where  $A_{additional}$  is the difference between required area with designed maximum allowable resistance and the required area due to fouling resistance that exceeds design limit, and  $U_{cf}$  is the coefficient of heat transfer after fouling growth.  $Q_{loss}$  was plotted against the changes in fouling factor for each exchanger. At the maximum allowable design fouling factor, performance was assumed to be 100% (0 heat loss), because the exchanger meets the targets it was designed to recover. These equations were executed through VBA. Simulation was carried out on 5 heat exchangers. Selection of the exchangers was based on the type of the fluid on the tube side. The exchangers had air (Exchanger 1, Plant A), air (Exchanger 2, Plant A), water (Exchanger 3, Plant A), milk (Exchanger 1, Plant B) and wash (Exchanger 1, Plant C).

## 3 Results and Discussion

The fluid flow parameters and the Heat Exchanger Design for Plant C are presented in Tabs. 1 and 2, respectively. The parameters and design for Plants A and B are presented in the appendix section.

**Table 1:** Heat transfer parameters from Plant C

Symbol	Exchanger	1	2	3
	Shell	Ethanol Liquid	Ethanol Vapor	Ethanol Vapor
	Tube	Wash	Wash	Wash
Tci	K	333.15	301.15	334.15
Tco	K	334.15	333.15	339.9
Thi	K	383.15	384.15	384.15
Tho	K	344.15	383.15	383.15
mt	kg/s	0.766	0.766	0.766
ms	kg/s	0.766	0.0986	0.766
Pt	kg/m <sup>3</sup>	1000	1000	1000

(Continued)

**Table 1 (continued).**

Symbol	Exchanger	1	2	3
		Ethanol Liquid	Ethanol Vapor	Ethanol Vapor
		Wash	Wash	Wash
Ps	kg/m <sup>3</sup>	789	3.181	3.181
Cpt	kJ/kg.°C	586.69	3.514	3.514
Cps	kJ/kg.°C	2.43	837.85	837.85
St	NA	1	1	1
Ss	NA	0.789	2.49	2.49
μt	Pa.s	0.00089	0.00089	0.00089
μs	Pa.s	0.001095	0.0000124	0.0000124
Uass	kW/m <sup>2</sup> .°C	0.015	0.015	0.015
Ks	W/m.°C	0.171	0.0144	0.0144
Kt	W/m.°C	0.677	0.677	0.677
Kw	kW/m.°C	111	111	111
Rt	m <sup>2</sup> .K/W	0.00003	0.00004	0.00004
Rs	m <sup>2</sup> .K/W	0.00012	0.00012	0.00012

**Table 2:** Heat exchanger design for plant C

Design Variable	Heat Exchanger Number		
	1	2	3
Quantity of Heat (kW)	72.5938	86.135168	15.477413
Tube length (m)	6.096	6.096	6.096
Tube outer diameter (m)	0.012875	0.012875	0.012875
Tube inner diameter (m)	0.000635	0.000635	0.000635
Birmingham Wire Gauge	1	1	1
Tube thickness (m)	0.00762	0.00762	0.00762
Selected number of tubes	838	408	108
Number of tube passes	8	8	8
Shell inside diameter (m)	0.9398	0.38735	0.38735
Tube pitch (m)	0.0254	0.0254	0.0254
Number of baffles	20	20	20

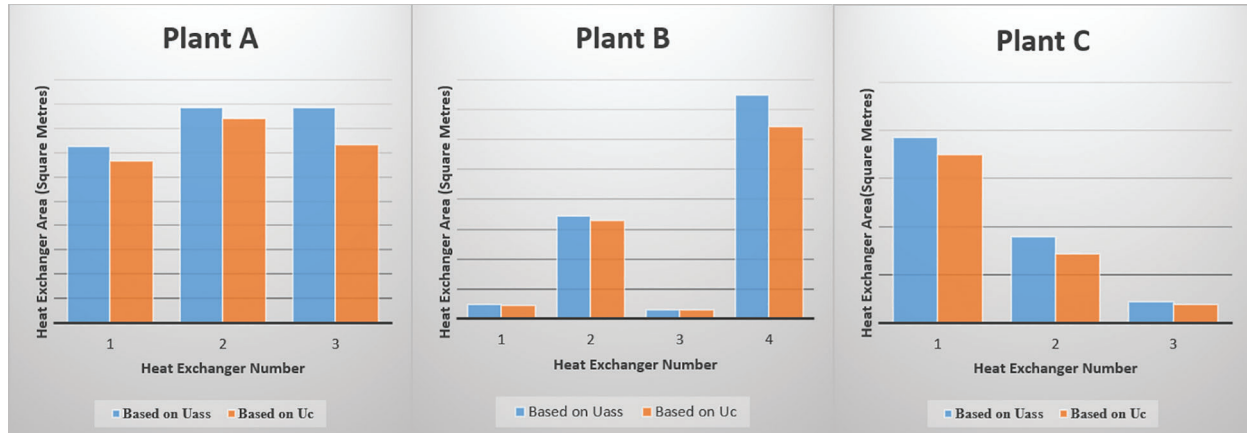
In plant C, as is for A and B, the hot fluids were allocated the shell side and the cold fluids on the tube side. The parameters in [Tab. 1](#) were used for design of heat exchangers for Plant C, shown in [Tab. 2](#).

The quantity of heat specified for each exchanger should be exchanged to meet the specified internal recoverable heat, in pinch analysis.



### 3.1 Heat Exchanger Areas

Two types of heat exchanger areas required, designed to meet the quantified heat recovery are presented in this section. One area has been designed using  $U_{\text{ass}}$ , the assumed value of coefficient of heat transfer, while the other area has been designed using Kern Equations. The results are shown in Fig. 1.



**Figure 1:** Comparison of heat exchanger areas

For all the plants, the areas computed by assuming the value of overall coefficient of heat transfer are more than those computed using the one calculated using Kern Equations. The average percentage difference for the plants is 10.1%, 6.9% and 14.9% for A, B and C, respectively. The design approach that uses assumed value of overall coefficient of heat transfer does not consider the fouling factor. The  $U_{\text{ass}}$  value from the literature is assumed to cater for the fluid fouling factor. However, this assumption leads to overstating the required heat transfer area, thus overstating the cost of heat recovery, by these computed percentages. This confirms findings of a simulation study by [25], which revealed that the design method that obviates the use of fouling factors during design of fin and plate heat exchangers increased the costs by a variation of 0.34% to 21.65%.

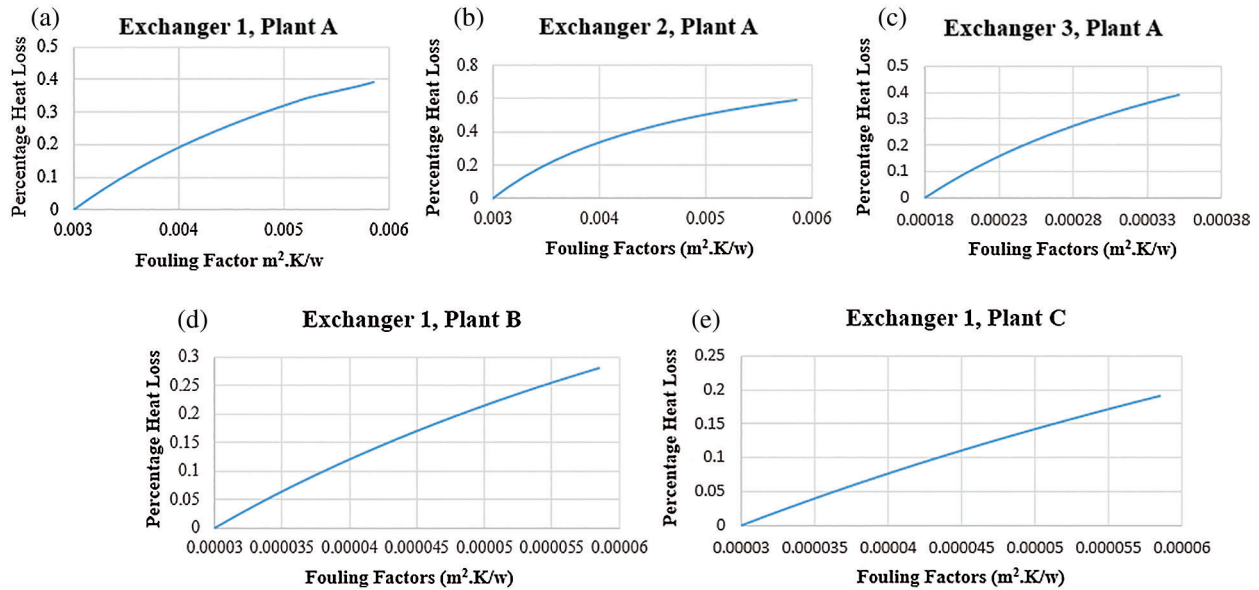
### 3.2 Effects of Fouling Growth on Thermal Performance of Heat Exchangers

The percentage heat transfer loss due to growth in fouling resistance for the 5 selected heat exchangers is presented in Fig. 2. The simulation assumed that at the optimal design fouling factor, for example, for Exchanger 1, Plant A, at  $0.003 \text{ m}^2 \cdot \text{K/W}$ , the heat exchanger will meet the targeted quantity of heat transfer. As fouling increases beyond the designed value, heat transfer reduces.

The heat loss has an exponential relationship with growth of fouling, for the five exchangers. The modelled average losses, for the 100% increase in fouling factors, was 0.37%. The findings of this study fortify the findings of the work by [14], which revealed that growth in fouling factors affected thermal efficiency of a power plant by 0.45%. Exchanger 1 and 2 for Plant A have similar increase in the fouling factors, but their heat transfer losses are different, at 0.39% and 0.59%. The LMTD for the former is  $28.51^\circ\text{C}$  while the latter is  $55.97^\circ\text{C}$ . This implies that heat exchange processes with a higher LMTD are bound to be affected more in thermal performance deterioration due to growth in fouling.

The modelled maximum fouling growth for Exchanger 3 for Plant A is almost half that of Exchangers 1 and 2 for Plant A, at  $0.000351 \text{ m}^2 \cdot \text{K/W}$ , compared to  $0.00585 \text{ m}^2 \cdot \text{K/W}$ . However, the percentage heat loss is 0.39%. Its LMTD is higher, at  $52.8^\circ\text{C}$ , compared to Exchanger 1, at  $28.51^\circ\text{C}$ . Similarly, the ratio of the external to internal diameter of the tube for Exchanger 3 is higher compared to Exchanger 1, at 11.11 (Birmingham Wire Gage of 7.62 mm), compared to 3.03 (Birmingham Wire Gage of 4.572 mm). This is

illustrated in the design data in the appendices. Fouling growth thus has a higher effect on exchangers with tubes that have more thickness than those that have lower thickness.



**Figure 2:** Effect of fouling growth on heat loss

The modelled heat transfer losses for Exchanger 1 Plant B and Exchanger 1 Plant C are 0.28% and 0.19%, respectively. The same fouling growth notwithstanding, the two have different percentage losses. The LMTD for Exchanger 1, Plant B is 31.8°C while that of Exchanger 1 Plant C is 25.4°C. The tube thickness of the two is the same, with Birmingham Wire Gage of 7.62 mm. The effect of fouling growth on the heat loss in the heat exchangers was thus moderated by the LMTD of the two exchangers.

#### 4 Conclusions

This paper has analyzed the effects of growth of fouling in shell and tube heat exchangers, by use of thermodynamic analysis, heat balance and Kern Design equations. This study has revealed that estimation of overall heat transfer coefficient, which uses an estimated fouling factor for design of heat exchangers, leads to an overestimate of the required shell and tube exchanger area. The overestimation was by an average of 10.1%, 6.9% and 14.9% for plants A, B and C, respectively, compared to computation of the same using Kern Design Equations. Increase in the design fouling factor by 100% leads to increase in heat losses by an average of 0.37%, for the three plants.

This work has led to four conclusions concerning heat exchanger design, fouling resistance and thermal performance of shell and tube heat exchanger. The relationship between increase in fouling resistance and heat transfer losses in shell and tube heat exchangers is exponential. In design problems that estimate the overall heat transfer coefficient, thus not factoring in the fouling factors unique to the heat transport fluid, the exchange areas are overestimated, thus overestimating the costs of heat recovery. The LMTD determines the severity of fouling resistance in shell and tube heat exchangers. Higher values of LMTD exacerbate the heat losses, compared to lower values, because of the exponential relationship between exchanger heat loss and fouling growth. Similarly, the ratio of external diameter to internal diameter of the exchanger tubes amplifies the effect of fouling growth on thermal performance of the exchangers. The higher the ratio, the higher the heat loss due to growth in fouling. There is need for more work to be

carried out on the effect of fouling growth in shell and tube heat exchangers conveying different fluids, especially in prediction of fouling growth using computation fluid dynamics and linking the predicted growth to thermal performance.

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## Appendices

### Appendix 1: Data for design of heat exchangers for Plant A

Symbol	Exchanger	1	2	3
	Shell	SO <sub>3</sub>	Air	Air
	Tube	Air	Air	H2O (l)
Tci	K	389.15	388.15	373.15
Tco	K	414.12	389.15	374.15
Thi	K	476.15	474.15	418.18
Tho	K	399.15	418.18	383.15
mt	kg/s	1.303	1.303	0.417
ms	kg/s	0.435	1.235	1.235
Pt	kg/m <sup>3</sup>	1.274	1.274	1000
Ps	kg/m <sup>3</sup>	3.37	1.225	1.225
Cpt	kJ/kg.°C	1.013	1.013	2260
Cps	kJ/kg.°C	0.9	1.013	1.013
St		2	2	1
Ss		2.75	1	1
μt	Pa.s	2.26E-05	2.61E-5	89E-5

<b>Appendix 1 (continued).</b>				
Symbol	Exchanger	1	2	3
	Shell	SO <sub>3</sub>	Air	Air
	Tube	Air	Air	H2O (l)
$\mu s$	Pa.s	3.26E-05	2.61E-5	2.61E-5
$U_{ass}$	kW/m <sup>2</sup> .°C	0.015	0.015	0.015
$K_s$	W/m.°C	0.75	0.0262	0.0262
$K_t$	W/m.°C	0.0262	0.0262	0.677
$K_w$	kW/m.°C	0.111	0.111	0.111
$R_t$	m <sup>2</sup> .K/W	0.003	0.003	0.00018
$R_s$	m <sup>2</sup> .K/W	0.0012	0.0012	0.0012

**Appendix 2: Data for design of heat exchangers for Plant B**

Symbol	Exchanger	1	2	3	4
	Shell	Milk	Milk	Milk	Milk
	Tube	Milk	Water	Steam	Water
$T_{ci}$	K	298.15	298.15	374.15	373.15
$T_{co}$	K	340.0	340.0	415.15	374.15
$T_{hi}$	K	358.15	349.1	420.15	418.02
$T_{ho}$	K	349.1	348.16	418.02	376.15
$\dot{m}_t$	kg/s	1.87	0.194	0.194	0.194
$\dot{m}_s$	kg/s	8.61	8.61	1.87	1.87
$P_t$	kg/m <sup>3</sup>	1026	1000	0.6	1000
$P_s$	kg/m <sup>3</sup>	1026	1026	1026	1026
$C_{pt}$	kJ/kg.°C	4.18	4.18	2.09	2260
$C_{ps}$	kJ/kg.°C	4.18	4.18	4.18	4.18
$St$		1.026	1	0.49	1
$S_s$		1.026	1.026	1.026	1.026
$\mu_t$	Pa.s	0.003	0.00089	0.0000162	0.00089
$\mu_s$	Pa.s	0.003	0.003	0.003	0.003
$U_{ass}$	kW/m <sup>2</sup> .°C	0.015	0.015	0.015	0.015
$K_s$	W/m.°C	0.637	0.637	0.637	0.637
$K_t$	W/m.°C	0.637	0.677	0.0288	0.677
$K_w$	kW/m.°C	111	111	111	111
$R_t$	m <sup>2</sup> .K/W	0.00003	0.00003	0.00003	0.00003
$R_s$	m <sup>2</sup> .K/W	0.00009	0.00009	0.00009	0.00009

**Appendix 3:** Exchanger parameters for Plant A

Design Variable	Heat Exchanger Number		
	1	2	3
Quantity of Heat (kW)	30.1455	70.02	70.02
Tube length (m)	4.877	4.877	4.877
Tube outer diameter (m)	0.00635	0.0127	0.0127
Tube inner diameter (m)	0.002133	0.003556	0.001143
Birmingham Wire Gauge	7	7	1
Tube thickness (m)	0.004572	0.004572	0.00762
Selected number of tubes	886	308	480
Number of tube passes	4	4	4
Shell inside diameter (m)	0.9398	0.59055	0.7366
Tube pitch (m)	0.0254	0.0254	0.0254
Number of baffles	20	20	20
Estimated Area (m <sup>2</sup> )	72.3305	88.4270	88.4270
Multi objective criteria-based Area (m <sup>2</sup> )	66.5392	84.0248	73.2285
Percentage (%) difference in Areas computed	8	4.98	17.19

**Appendix 4:** Exchanger parameters for Plant B

Design Variable	Heat Exchanger Number			
	1	2	3	4
Quantity of Heat (kW)	33.830412	325.70769	16.649358	1506.893
Tube length (m)	4.877	15.24	15.24	40
Tube outer diameter (m)	0.015875	0.015875	0.015875	0.0127
Tube inner diameter (m)	0.000635	0.000635	0.000635	0.0006096
Birmingham Wire Gauge	1	10	10	4
Tube thickness (m)	0.00762	0.0034036	0.0034036	0.0060452
Selected number of tubes	346	948	116	948
Number of tube passes	8	8	8	8
Shell inside diameter (m)	0.635	0.9906	0.38735	0.9906
Tube pitch (m)	0.0254	0.0254	0.0254	0.0254
Number of baffles	20	15	15	20
Estimated Area (m <sup>2</sup> )	95.20459	685.5007	62.216	1499.932
Multi objective criteria-based Area (m <sup>2</sup> )	90.24414	659.8252	59.38413	1288.354
Percentage (%) difference in Areas computed	5.21	3.74	4.55	14.12