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"FOULING MODELS AND THEIR APPLICATIONS IN CRUDE OIL PRE-HEAT TRAINS AND COOLING WATER NETWORKS"

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El presente trabajo se realizó en el Departamento de Ingeniería Química de la División de Ciencias, Naturales y Exactas de Campus Guanajuato de la Universidad de Guanajuato, bajo la dirección del Dr. Graham Thomas Polley. Para la realización de dicho trabajo se contó con el apoyo del Consejo Nacional de Ciencia y Tecnología, CONACYT.

Por medio de lo presente, yo, **Elvis Koku Tamakloe** me responsabilizo de la originalidad y autenticad de este trabajo bajo el título:

FOULING MODELS AND THEIR APPLICATIONS IN CRUDE OIL PRE-HEAT TRAINS AND COOLING WATER NETWORKS

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DEDICATION

To my baby, Klenam … Tamakloe Castro

ABSTRACT

The fouling process is complex. Different mechanisms may interact within a process to generate the deposits that are costing refineries and other processing industries millions of dollars in energy costs, capital cost, cleaning cost, mitigation strategy costs amongst others. These mechanisms may include particulate fouling, crystallization fouling and chemical reaction fouling. Fouling in the crude oil pre-heat train and the cooling water network is investigated in this work.

Different fouling models characterize fouling in heat exchangers. Since the late 1950s when Kern and Seaton proposed a model for particulate fouling, other fouling models in recent years have emerged from this analogy. The analogy and the idea of a "sticking probability" are applied in the development of a fouling model called the Asphaltene Precipitation model (AP model) for chemical reaction fouling. The model is successfully used to analyze fouling in shell and tube heat exchangers and Compabloc units that gave very good predictions. The results suggest that the same model can be used for analysis irrespective of the high differences in fluid shear stress.

The crude oil pre-heat train serves as a heat recovery network for the crude oil on its way to the distillation column. A thermo-hydraulic simulation based on a path analysis served as an invaluable tool for analyzing the pre-heat train. This results in major heat recovery actions that can help reduce the external energy demands. In light with the model developed some mitigation strategies are also presented for use in controlling fouling.

Advances in heat exchanger technology for applications in pre-heat trains are invaluable in this age when fouling has been accepted as a major problem in crude oil processing. The Compabloc unit is presented with an in depth approach into its thermal behavior. The algorithm developed can be applicable in every field where the Compabloc unit is used. Maintaining the Compabloc unit at high fluid shear stress is a great tool when fouling needs to be controlled.

An earlier model developed for the analysis of crystallization fouling of carbonate and sulfate salts are used in the analysis of a revamp process for a cooling water network where a new heat exchanger needs to be installed. The integration of the model in a thermo-hydraulic simulation proved well in helping to choose the position of the new heat exchanger such that as little interruption is felt by the other heat exchangers in the network in terms of fouling, temperature profiles and flow rates. Emphasis is placed on the need to incorporate fouling analysis in the design process.

RESUMEN

El proceso de ensuciamiento es compleja. Diferentes mecanismos pueden interactuar dentro de un proceso para generar los depósitos que están costando las refinerías y otras industrias de procesamiento millones de dólares en costos de energía, costos de capital, gastos de limpieza y estrategia de mitigación, entre otros. Estos mecanismos pueden incluir ensuciamiento por partículas, el ensuciamiento por cristalización y el ensuciamiento por reacción química. Por lo tanto se investiga el ensuciamiento en el tren de precalentamiento de petróleo crudo y la red de agua de enfriamiento.

Diferentes modelos de ensuciamiento caracterizan el ensuciamiento en intercambiadores de calor. Desde finales de la década de 1950 cuando Kern y Seaton propusieron un modelo de ensuciamiento por partículas, otros modelos de ensuciamiento han surgido de esta analogía en los últimos años. La analogía y la idea de una "probabilidad de adhesión" fueron aplicadas en el desarrollo de un modelo de ensuciamiento llamado el modelo de "Asphaltene Precipitation" (modelo AP) ensuciamiento por reacción química. Se utilizó el modelo con éxito para analizar y predecir el ensuciamiento en los intercambiadores de calor de tubo y coraza y las unidades Compabloc. Los resultados sugieren que el mismo modelo puede ser utilizado para análisis independientemente de las altas diferencias en esfuerzo cortante.

El tren de precalentamiento de petróleo crudo sirve como una red de recuperación de calor para el petróleo crudo en su camino hacia la columna de destilación. Una simulación termo-hidráulico basada en un análisis de trayectoria sirvió como una herramienta muy valiosa para el análisis del tren de precalentamiento. Esto se traduce en importantes acciones de recuperación de calor que pueden ayudar a reducir la demanda de energía externa. A la luz del modelo desarrollado también se presentan algunas de las estrategias de mitigación para su uso en el control de ensuciamiento.

Los avances en la tecnología de intercambiador de calor para aplicaciones en trenes de precalentamiento son de gran valor en esta época cuando ensuciamiento ha sido aceptada como un problema inevitable en el procesamiento de petróleo crudo. La unidad Compabloc se presenta con un enfoque en su comportamiento térmico. El algoritmo desarrollado puede ser aplicable en todos los campos donde se utiliza la unidad Compabloc. La operación de la unidad de Compabloc a alta valores de esfuerzo cortante es una gran herramienta cuando ensuciamiento tiene que ser controlada.

Un modelo anterior desarrollado para el análisis de ensuciamiento por cristalización de sales de carbonato y sulfato se utilizan en el análisis de un proceso de readecuación de una red de agua de enfriamiento en donde un nuevo intercambiador de calor necesita ser instalado. La integración del modelo en un análisis termo-hidráulico demostró ser muy útil en elegir la posición de la nueva intercambiador de calor de tal manera que la menor interrupción es sentida por los otros intercambiadores de calor en la red en términos de ensuciamiento, perfiles de temperatura y velocidades de flujo. Se hace nota en la necesidad de incorporar el análisis de ensuciamiento en el proceso de diseño.

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NOMENCLATURE

Names

a, b, c, d, e, f, g, h	Parameter coefficients

R	Fouling resistance, m^2K/W
U	Overall heat transfer coefficient, W/m^2K
E	Activation energy, *kJ/mol*
Rg	Gas constant, *kJ/mol K*
T, t	Temperature, ^oC
Re	Reynolds number
A	Deposition constant
t	Time
F	Shear factor
P	Sticking probability
m	Mass flow rate, *kg/s*
Cp	Specific Heat capacity, *kJ/kgK*
CP	Heat capacity rates, *kJ/K*
A	Heat transfer area, m^2
N_{TU}	Number of transfer units
k_R	Arrhenius rate constant
c	Concentration of salts, *ppm*
V	Volumetric flow rate, m^3/s
d	Tube diameter, *m*
K	K value, $Pa.s^2/m^6$

Greek Letters

α	*Heat transfer coefficient,* W/m^2K
β	*Mass transfer coefficient*
γ	*Gamma constant*
δ	*Thickness of deposit, mm*
ε	*Effectiveness*
μ	*Viscosity, Pa.s*
τ	*Fluid Shear stress, Pa*
ϕ	
κ	
$\mathfrak{D}$	*Diffusion coefficient*
λ	*Thermal conductivity, W/mK*
ρ	*Density of fluid,* kg/m^3
ψ	

Subscripts

b	*Bulk*

c	Clean condition
d	Deposit
f	film
i	Counter
in	Inlet
out	Outlet
min	Minimum
r	Removal
s	Surface
w	wall/water
H	Hot stream
1,2,3,..	Different sections/bases

Superscripts

*	*Saturated/boiling point*

INTRODUCTION

Fouling is the deposition of an unwanted phase on heat transfer surfaces that interrupts the heat transfer process. Fouling is one of the greatest problems affecting the operation of heat transfer equipment in the manufacturing, petroleum refinery and power generation industries. The fouling process is complex. The deposition of an unwanted phase during processing may also result in mal flow distribution. The fouling mechanisms including chemical reaction fouling, crystallization fouling and particulate fouling have been studied both academically and on an industrial scale over the years and different theories have been presented about their occurrence and the mechanisms involved. These mechanisms according to Epstein [3] can be represented in a 5x5 matrix emphasizing the depth of information or investigation that has been carried out on the different fouling stages recognized in the fouling process alongside the different fouling mechanisms.

In the operation of heat exchangers, so many factors contribute to the formation of fouling deposits on the heat transfer surfaces. Some of these include the presence of water and dissolved salts, particles, crude oil mixture, the chemistry of the stream and the operational conditions amongst others. Investigations into fouling are not a novice process for the analysis of the pre-heat train or the cooling water network; in recent studies emphasis is being placed on the study of the mechanisms involved and the thermal transformation of the foulants on the heat transfer surface. Chemical reaction fouling is attributed to asphaltene precipitation and deposition in petroleum refinery which is influenced by oil incompatibilities and the concentration of the other components of petroleum crude [4-6]. This precipitation and deposition mechanisms is influenced by the behavior of the boundary or sub layer close to the heat transfer surface [7, 8]. Crystallization fouling in related to the ions present in the chemistry of the stream.

The fouling mechanism

Fouling is defined by Wiehe [5] as the formation and accumulation of an unexpected phase that interferes with processing. Fouling is a very expensive industrial problem and without attention can lead to millions of losses in output and energy. Mostly the development of fouling in a heat exchanger can be linked to two operating or design parameters; the local temperature and the local velocity in the heat exchanger. Research into the development of fouling models and mitigation strategies have arrived at the following conclusions:

- The rate of fouling increases with wall temperature and
- At a fixed wall temperature, fouling rates decreased with increasing flow velocity.

The availability of a fouling model for the prediction of fouling will help developers, designers and analysts define appropriate design parameters for use in heat exchanger designs whiles postulating adequate operating conditions to minimize this phenomenon.

Fouling resistance is also defined by a measure of the effect of its thermal conductivity, λ_f on the heat transfer process and expressed as (δ is the thickness of the deposit on the heat transfer surface):

$$R_f = \frac{\delta}{\lambda_f}$$

There is also the fouling resistance being defined by the relation between the overall heat transfer coefficients of a clean, U_c and a dirty, U heat transfer surface. This is given by:

$$R_f = \frac{1}{U} - \frac{1}{U_c}$$

Fouling is characterized into so many different types [9] and different fouling models have been developed to be able to analyze, predict and help control or mitigate the fouling problem. Most of the models are attributed to chemical reaction of organic molecules in the bulk of the fluid or close to the heat transfer surface. The difficulty in being able to predict fouling in most equipment is attributed to the complexity and overlapping nature of the mechanisms of fouling [10] and the scarcity and uncertainties in plant data for the analysis of fouling. The mechanisms that lead to the formation of deposits do change along the flow path, in its development with time as well as changing operating conditions of the heat exchangers involved. In the case of cooling water systems the composition of the water is a great factor to the type of fouling mechanism and its development. Another difficult in the modeling of fouling in heat exchangers is the interaction between the mechanisms of fouling where one type of fouling results in the incorporation of other mechanisms of fouling resulting in a complex process, however the individual mechanisms can be successful modeled and isolated in the fouling process for analysis.

There are three ways by which fouling affects heat exchanger operation [11]:

- Fouling layer has a low thermal conductivity resulting in an overall high resistance to the heat transfer in the heat exchanger. The thermal conductivity of the deposits lie between 0.1 and 0.5 W/mK [10] and may increase to 1.0 W/mK as a result of coke formation. This is mostly higher for deposits in precipitation fouling.
- With the development of the fouling deposits, the cross sectional flow area of the tubes (in tubular heat exchangers) decreases causing an increase in the pressure drop across the heat exchanger and mal distribution of the flow.
- Fouling can also lead to substrate corrosion and subsequent erosion of the heat transfer surface.

Fouling types

The mechanisms involved in the formation of fouling are defined by the precursors available in the bulk flow. These precursors may include organic molecules, inorganic salts or particulates or a combination of these. The different mechanisms are outlined below:

- *Chemical reaction fouling*: This type of fouling is as a result of a chemical reaction or thermal decomposition process of components in the bulk of the fluid. The reactions involved are very complex and significant research has been conducted to really understand the mechanisms [10, 12].
- *Crystallization or precipitation fouling*: This is associated with the presence of dissolved salts in the bulk of the flow. It is the inorganic component of the fouling process. These salts are mostly carbonates and sulfates whose solubility decreases with an increase in temperature: reverse soluble salts. The mechanism and some important factors are presented in other chapters.
- *Particulate fouling*: This type of fouling is associated with the presence of particles and other solid materials in the bulk flow, which settles on the heat transfer surface or contribute to the formation of crystals of the salts. The deposition process is largely due to transport to the surface by Brownian motion and/or gravity.
- *Biological fouling*: This results in the formation of microbiological deposits on the surface as a result of the presence of bacteria, fungi or algae in the bulk flow mainly cooling water. It is a major problem for cooling water systems that use sea water as a medium for cooling [13].

- *Corrosion and fouling*: This is *in situ* corrosion, a reaction on the heat transfer surface [11]. There exist a strong relationship between fouling and corrosion. The subsequent deposition of materials onto the surface and other particles present in the bulk of the fluid may result in reactions with the heat transfer surface in the presence of oxygen. Corrosion may result, contributing greatly to the degradation of the heat transfer surface material and fouling at these sites.

Generally there are five stages in the fouling process: initiation of fouling, mass transport to the heat transfer surface, formation and attachment of deposits to the surfaces, removal from the surfaces and aging of the deposits [3, 11, 14]. A schematic representation of the deposition process is depicted below:

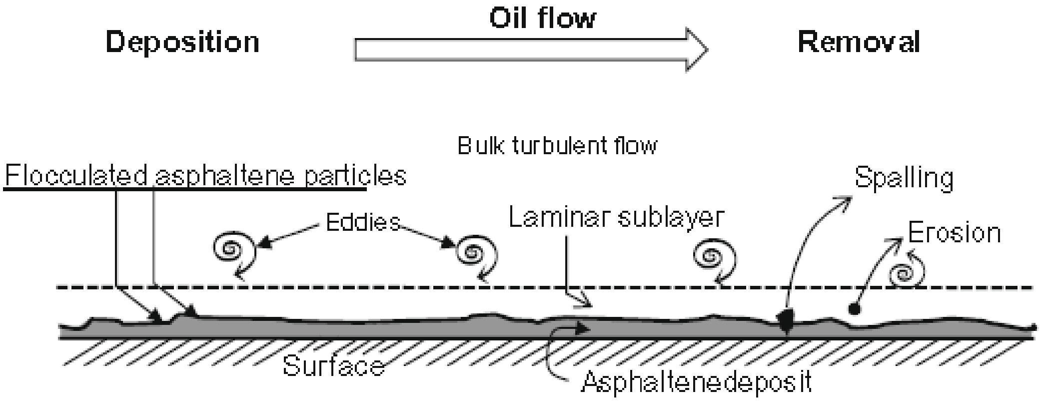


Figure 1: Schematic representation of the deposition and removal process [1]

Fouling Causes

The causes of fouling can be linked directly to the following sources:

- Inorganic: Dissolved salts, acids in water present in the bulk flow.
- Organic: Suspended insoluble particles, emulsions, gums and insoluble asphaltenes.
- Chemical reaction processes that may include thermal decomposition, polymerization, autoxidation or coking.
- Chemical incompatibilities [5]: As a result of mixing the incoming streams, a typical case in the crude oil refinery process.
- Oil water emulsions [5]
- Polymerization of olefins after thermal conversion [5]

Practicalities of the crude oil Pre-heat train

The refinery of crude oil involves a system of heat recovery units that help to make use of the heat content of the products and pump around streams from the distillation column to heat up the crude oil on its way to the distillation column. The network of heat exchangers that integrate the heat recovery section of the refinery process is the pre-heat train. Crude oil flowing through these heat exchangers is heated from a temperature as low as the prevailing weather temperature conditions of the area to a temperature in the range of 250 – 320 ºC. The pre-heat train also consists of a desalter and a furnace located at different points in the train. A flash drum may also be incorporated in the train to prevent vaporization of the crude oil in the heat exchangers.

The pre-heat train is very important in the crude oil distillation process and no two pre-heat trains processing the same crude oil give the same heat recovery results. Crude oil market trends indicate that the industry is shifting to the use of heavier and more acidic crude oil, and thus fouling has become a severe process. The fouling mechanism varies with location along the pre-heat train. Prior to the desalter, crystallization and particulate fouling are the mechanisms describing the deposition process whiles downstream from the desalter, chemical reaction fouling is prominent.

So many problems can be associated with the operation of the crude oil pre-heat train. Notable amongst them is the carry-over of water and sediments to the heat exchangers immediately after the desalter. These problems are practical and are associated with operational conditions of the desalter and heat exchangers that make-up the pre-heat train. The desalter is mostly operated at an average temperature of 120 ºC and serves to remove sediments and water-soluble salts from the crude oil. Due to changes in operational conditions, these sediments and water-soluble salts may be carried-over to the heat exchangers downstream from the desalter leading to water hydrolysis and subsequent corrosion of these heat exchangers. These carry-overs may also reduce heat recovery and malfunction of these heat exchangers.

Corrosion is a function of the crude chemistry and coupled with other operating conditions can occur throughout the pre-heat train. Corrosion may be *in-situ* where the heat transfer surface initiates and is involved with the corrosion process or *ex-situ* where sediments in the bulk of the fluids leads to the formation and subsequent development of the corrosion process. Because corrosion can occur at all parts of the crude oil pre-heat train, care has to be taken in the operation of the heat exchangers and the desalter since the consequences may lead to high capital and maintenance costs of the pre-heat train.

Another practicality in the operation of pre-heat train is the inability to obtain the same results from running the same crude oil type on different pre-heat trains. The difficulty and complex nature in the heat recovery process does not permit the same result to be obtained from running the same type of crude oil. These uncertainties in results occur from the varying operating conditions and blends that are associated with processing crude oils.

Operational problems are also associated with the management of the crude oil pre-heat train. For example the lack of adequate attention to the fouling mechanisms or foulants in the heat exchanger immediately downstream from desalter may pass out for the proper operation of the desalter but that may be false. Since the desalter is limited by temperature operation and helps in the elimination of water-soluble substances in the crude oil feed stock, the carry-over of for example salts may affect the performance of the heat exchangers downstream of the desalter, producing more complex foulants and problems with heat recovery.

The behavior of temperature instruments deteriorates with time and a great source of worry for the operator of the pre-heat train. The functionality of these instruments is important if data is to be gathered for further analysis especially in modeling fouling. In a lot of cases during the operation of the pre-heat trains, these errors in instrument measurements are not detected until the instrument fails. There would therefore be the need to understand how this missing data can be recuperated for analysis.

The cooling water network

Fresh water is natural water that has very low content of dissolved salts compared to seawater. Crystallization fouling is the major cause of fouling in cooling water networks but other mechanisms such as biological fouling, particulate fouling and corrosion fouling may overlap with the crystallization process presenting a more complex situation. Crystallization fouling may include deposit of sulfates and carbonates of calcium amongst other types of salts. These dissolved salts are often called reverse soluble salts because their solubility is inversely proportional to the surface temperature. Najibi [15] has successfully studied and modeled fouling in systems containing calcium carbonate and calcium sulfate salts relating the effect of the operating parameters on the fouling processes. Other studies of crystallization fouling has also been done on the activation and delay periods for fouling [16], two component fouling of calcium carbonate and sulfate deposits [17]. The primary causes of fresh water fouling are associated to the chemical composition of the water, that is, the high concentrations of salts, the presence of microbes (like bacteria, algae or fungi) and sludge or dirt.

Fouling in cooling water networks is also not a new feature in the cooling water networks. Recent design considerations are based more on the experience and operation of the heat exchangers involved rather than the use of the TEMA fouling factors that are available in heat exchanger design literatures. Moreover fouling resistances depend critically on the operating conditions of the heat exchangers involved [18, 19], their location in the network and also elapsed time. More so the cost of fouling in coolers is estimated to be between 6 million and 12 million dollars (US) for a single 600MW power plant and an average of 0.25% of the gross national product in the highly industrialized countries [18]. The extent of fresh water fouling is therefore dependent on the composition of the water, operating conditions and maintenance strategies.

There are research works and published guides on sea water fouling and fresh water fouling [11] but little is seen in the consideration of fouling in the design of cooling water networks. Like fouling in the petroleum industry, cooling water fouling has major consequences on heat recovery and the management of the cooling water networks. This is because apart from the fouling in the individual coolers in the network, the fouling at the long run affects the return temperature of the water to the cooling tower since there is a limitation on the maximum water return temperature [20]. Especially in the open recirculation system where a cooling tower is employed for the rejection of heat to the environment, a high water inlet temperature to the tower will also cause salt deposition in these towers.

The design and simulation of cooling water networks has also been studied along the years [21, 22]. A thermo-hydraulic simulation of these networks is important and necessary in understanding the performance of the coolers and how these can be modified in the case of a revamp situation. The integration of fouling in the simulation of cooling water networks is new in the simulation process and a great tool for design and revamp analysis. The use of a short cut design procedure [20] as the initial heat exchanger design decision in the revamp of cooling water networks will help not only in making quick and accurate revamp decisions but also ease on the design procedure.

Fouling in Compabloc units

Compabloc units are welded plate heat exchangers that have seen a lot of application in the process of heat recuperation. The design and performance analysis of Compabloc units for use in crude oil service has been for their enhanced heat recovery measures. The following can be summarized about Compabloc units:

- Compabloc units are compact, welded plate heat exchangers with an effective cross flow in the flow distribution. They are small in size compared to a shell and tube heat exchanger of the same heat load.
- Compabloc units have high heat transfer coefficients on both sides of fluid flow giving it a high overall heat transfer coefficient for enhanced heat recovery.
- Compabloc units mostly have high wall temperatures because of the enhanced heat recovery compared to shell and tube heat exchangers.
- So many flow passes can be employed in the design of Compabloc units. Increasing the number of passes on a flow side results in an increase in the velocity of flow of that particular stream. A case is known where there were 12 passes on the cold fluid side and 9 passes on the hot fluid side.

Philosophy and Application of a fouling model

Fouling models are mathematical expressions that help to quantify the fouling resistances in a heat exchanger. These models help to relate the operation of the heat exchanger with parameters such as the temperature and velocity and the physical properties of the stream being analyzed.

The fouling resistances give great insight into the analysis and understanding of the operation of the heat exchangers that make up a network of heat exchangers. These operational conditions can be related to parameters that can be readily measured and provide other solutions to the already existing mitigation strategies. For example, the available models have given insights on the role of temperature and the shear stress in the fouling process. The generalized Ebert and Panchal model [23] defines a threshold for the operation of the heat exchangers and the Yeap et al [24] model shows the influence of velocity on the fouling rates. The thermal decomposition of the deposits to form coke, has also been proposed and studied by Wilson et al [25], an invaluable tool to understand the physical properties of the deposit with time.

A number of fouling models have been used to quantify chemical reaction fouling including but not all, the original Ebert–Panchal model [26] and the generalized version [23], Epstein model [27], Panchal model [23] and Polley et al model [24]. These models give good predictions for fouling rates in heat exchangers. This must be as a result of so many parameters involved in the calculation and a failure of which will not be accepted.

However, there are some difficulties in applying these models in the analysis of pre-heat trains.

- Most of the information or data available for analysis and prediction of pre-heat train are not accurate and filled with a lot of uncertainties. Quite often the data needs to be first reconciled before it can be applied for the analysis. If these uncertainties are not removed from the data then the prediction abilities of these models will be put in doubt due to their dependence on a lot of the physical properties of the bulk flow.
- Another difficult is the lack of more practical and simple analysis that needs to be undertaken. The fouling process is so complex that the models developed are equally complex which does not give a straightforward approach or a simple way of viewing the phenomenon and understanding the analysis process.
- In addition, the full dependence of the engineer or analyst on the use of the fouling models in predictive situations is becoming more important and is preventing the use of independent knowledge and experience about the operational conditions of the pre-heat train. This does not only make one solely dependent on the model but it reduces the ingenuity and independent brainstorming of the problem at hand.

The philosophy of a fouling model from the author’s point of view is that the model serves as a guide in the analysis of data and prediction of fouling in heat exchangers. This philosophy of application is therefore outlined below:

1. A model should also be practical and simple in its application, helping the user apply it when possible without complications.
2. The model should not only explain the chemical phenomenon but also give a practical description to the physical process as well.
3. The model is developed principally to serve as a guide to analyze and give good predictions comparable if not better than other models earlier developed.

4. A model being a guide implies that the user will have the opportunity to make the necessary changes to the operational conditions and brainstorm for possible changes in analysis.

Uses of fouling models

Understanding the mechanisms of fouling and subsequent development of a fouling model will go a long way to save the petroleum refinery, manufacturing and power generation industries millions of dollars in their efforts to reduce the problems associated with the fouling process. The availability of a fouling model will serve as a design, diagnostic and a mitigation tool.

Design tool

- A comprehensive fouling model will help in the design of heat exchangers and subsequently provide a design range for the selection of a geometry that can operate under acceptable fouling instead of the TEMA fouling factors for the design. A design example is shown in the work of Polley et al [28-30].
- A fouling model will help in the retrofit of pre-heat trains in an effort to reduce the fouling in the heat exchangers and analysis of the fouling effect on the fired heaters. This may also involve the modification of the whole structure or some heat exchangers in the train. In some cases it may also serve in the revamping of the process [29].

Diagnostic tool

- The fouling models will help in the management of fouling in the heat exchangers and identify where a heat exchanger is not operating properly.
- The efficiency of the desalter in a crude oil pre-heat train can also be checked with a fouling model. It is believed that the fouling mechanisms after the desalter are mostly chemical reaction fouling so a chemical reaction model can help diagnose if the desalter is working well in preventing for example crystallization fouling in the heat exchangers downstream.
- A fouling model as an online diagnostic tool can help in the management of the fired heater in the crude pre-heat train [31] helping the crude processing industry make real time analysis of fuel consumption for the fired heaters.

Mitigation tool

- A fouling model will help define if a tube insert is necessary for the operation of a heat exchanger. The model will give information and predictions about performance of the heat exchanger that will then call for this mitigation strategy.
- Scheduling for cleaning of heat exchanger especially in pre-heat trains of crude processing industry is now being incorporated into in the performance of these trains. The availability of a fouling model will predict the performance of a heat exchanger found in the train which will then make the operators plan for its cleaning and thus cut down on losses during shutdowns [32].

BACKGROUND

Fouling and its effect on the crude oil pre-heat trains and cooling water systems has been a headache for the process industries over the years. Since the introduction of a fouling rate model by Kern and Seaton [33] in the late 50’s to first describe particulate fouling, many other fouling models have been developed to help understand the mechanism and provide a mitigation strategy; the Ebert–Panchal model [23, 26], Epstein model [27], Panchal model [23] and Polley et al model [24] amongst others. The cost of fouling has been amounted to billions of dollars of losses in plant shut-down and mitigation measures [34] related to the cost of energy and other capital costs needed to run the process lines.

Watkinson and Wilson [10] and Wiehe [5, 35] have both done extensive work on the mechanism of chemical reaction fouling, the basis for asphaltene deposition and other deposits of organic component base and concludes that the fouling mechanism at the hot end of the crude oil pre-heat train is predominantly a chemical reaction that takes place close to the surface involving asphaltene molecules. This is emphasized by the model developed by Panchal and Watkinson [36]. What the model lacks in this case is the simplicity and ease of use for analysis. Even though different fouling models are available for analysis and prediction purposes, the complexity of the fouling process coupled with the complexity of these models makes its use a difficult one. There is the need to develop a simple model that will serve the purpose of practicality, simplicity and that provides good predictions.

Other authors including Yung [8], Cleaver and Yates [7] have modeled the behavior of asphaltene molecules in the deposition process and their attachment to the heat transfer surface. They have related the complexity of the fouling process to the instability and the difficulty in the analysis of the viscous sub layer of fluid flow over a surface. These investigations point out that there is just not the option of subtracting a fraction of fluid behavior on the heat transfer surface based on the fluid shear stress but the fouling process is influenced differently. For example, the movement of foulants into the viscous sub-layer may be modeled as a Brownian motion or by eddy diffusivity but in the viscous sub-layer, a well-defined path is established for the movement of deposits.

Wilson et al [23] provided the progress on the development and use of the generalized form of the Ebert-Panchal model that has seen many applications especially in commercial software in the analysis of fouling in heat exchangers processing crude oil. The model provides a basis for understanding the mechanism of fouling in these heat exchangers; it associates the removal of deposits to a negative term that on occasions may counteract the deposition resulting in no fouling being recorded. No solid explanation is given to the removal term only that it is a linear relationship between the fluid shear stress and a constant whose value is about 3.8×10^{-9} $m^2K/WhrPa$.

Advances in the application of heat exchanger technologies to crude oil pre-heat trains have helped a lot in the mitigation strategies and the improvement of heat recovery. Andersson et al [37] presented a paper on the experience in the application of Compabloc units to a crude oil pre-heat train, pointing out that Compabloc units are now being accepted for use in the crude oil pre-heat trains with the a similar behavior of fouling as compared to shell and tube heat exchangers. No indication of the thermal or fouling analysis of these Compabloc units have been given, only the effects of fouling and its comparison with shell and tube heat exchangers in the crude oil processing.

Najibi [15] applied a model for the analysis of crystal deposit of calcium carbonate and calcium sulfate in the cooling water industry. These models have been extensively applied to understand crystallization fouling and its effect on process industry parameters [38]. No evidence of application to Compabloc units is noted.

The modification or revamp of a heat exchanger network aims at helping in heat recovery and/or meet the production demand [22, 39, 40]. Tavares [21] and Kim &Smith [22] have studied the simulation and design of cooling water networks and the dangers in the revamp process like the position of the new heat exchanger in the network. Fouling is rarely an integral part of this modification process and with the availability of a fouling model this will help to understand the effect of fouling in the heat exchangers during the management and modification of the network.

GENERAL OBJECTIVES

1. Describe chemical reaction fouling and available models with the aim of developing a new fouling model: the Asphaltene Precipitation model, for chemical reaction fouling in heat exchangers processing crude oil.
2. Present a heat recovery measure and how this can be applied to the crude oil pre-heat train.
3. Introduce one of the important advances in technology for the crude oil pre-heat train and its application.
4. Present a fouling model for the crystallization fouling and its application in shell and tube heat exchangers and Compabloc units.
5. Present the effect of fouling on heat exchangers in a cooling network and some strategies for the modification of the network.

STRUCTURE OF THE THESIS

The development of this work is based on investigations conducted on both plant and laboratory data for the analysis of fouling in crude oil pre-heat trains and a network of heat exchangers for cooling. This thesis is thus divided into two distinctive but integrated parts:

Part 1: The crude oil pre-heat train
Part 2: The cooling water network

The investigations leading to this thesis were conducted in the following order:

1. Studies were conducted on different fouling models and how they have been applied in the analysis and prediction of fouling in heat exchangers.
2. A fouling model is then proposed and applied to data from a laboratory study and plant data obtained from the operation of shell and tube heat exchangers and Compabloc exchangers.
3. The effect of fouling on crude oil pre-heat trains is taken into consideration and a new heat recovery measure is applied for revamp purposes.
4. The fouling model proposed is then used to analyze the performance of Compabloc units used on pre-heat trains.
5. A model for the analysis of crystallization fouling in heat exchangers is then elaborated for its application in heat exchangers.
6. The crystallization fouling model is also used to analyze the effect of fouling during the modification and installation of a heat exchanger in a cooling water network.

Different case studies are presented for the application of the fouling models in both the crude oil pre-heat train and the cooling water network.

A summary of the proposed chapters and contents are presented below.

PART 1: THE CRUDE OIL PRE-HEAT TRAIN

Part 1 of this project describes the fouling processes in the crude oil pre-heat. It also looks at the development of a fouling model for the analysis and prediction of fouling in the heat exchangers that make-up the pre-heat train. The performance and analysis of pre-heat trains through a thermo-hydraulic process is also presented to appreciate the effect of fouling on the pre-heat trains. Part 1 is further divided into the following chapters:

Chapter 1: Fouling and the fouling models

This chapter focuses on the mechanisms of fouling that affects the hot end of the crude oil pre-heat train and how different fouling models have been used to describe these mechanisms. A general overview is given on what the fouling process is, the causes and the operation parameters that are most affected by fouling heat exchangers processing crude oil.

Fouling models are then presented and how they have been used to describe the fouling process and predict the fouling mechanisms. These fouling models span from the late 50's until the year 2010. Some of these models described include the Kern and Seaton model, Ebert-Panchal model with the generalized version (the most widely applied in industries), Panchal and Watkinson model and the Ageing model from the University of Cambridge.

Chapter 2: Development of a new fouling model for fouling in heat exchangers handling crude oil – the Asphaltene Precipitation model

This new model is the heart of this research project. The model is based on the concepts of the analysis of the fluid close to the heat transfer surface, the reaction mechanism in asphaltene molecules and the foulant deposition process. Information is gathered about the fluid behavior in the sub-layer and how the asphaltene reaction and deposition process occurs.

The model is made up of a deposition term that is defined by the film temperature in an Arrhenius relationship and coupled with a probability term that defines the behavior of the deposit based on the fluid shear stress.

The AP model is applied in the analysis of fouling in heat exchangers including the shell and tube heat exchangers and the Compabloc units processing crude oil. Different operational data are used to verify the model for both cases as well as on data from the laboratory experiment from Knudsen et al [41].

Chapter 3: Fouling and the thermo-hydraulic simulation in the operation of the crude oil processing pre-heat trains (PHT)

The crude oil pre-heat train is the heart of heat integration in the crude oil refinery process. The structure of the pre-heat train is presented as an introduction to the chapter with the effects of fouling on the different sections of the train. This helps understand where chemical reaction fouling is likely to occur and the implications in the operation of the pre-heat train.

A thermo-hydraulic simulation strategy is presented for the analysis of the pre-heat train. This simulation process takes into consideration some of the changes that can be made to the pre-heat train for energy recovery purposes. A case study is presented to address this process with suggestions on simple effective heat recovery measures.

Some of the technologies available for the mitigation of fouling are presented to conclude the chapter. These technologies are available on a more commercial basis and applied both as an online process and during the design process of heat exchangers for use in crude oil processing.

Chapter 4: Advanced heat exchanger technology in crude oil pre-heat trains – the Compabloc unit

Advanced heat exchanger technologies are available for use in the crude oil pre-heat train. The main purposes of these technologies are in the mitigation of fouling and the enhancement of heat transfer in the heat exchangers. One of these important technologies is the use of Compabloc units.

The structure and mechanism of operation of the Compabloc unit is explored in this chapter. The Compabloc unit is a compact heat exchanger that presents higher heat transfer coefficient values compared to that of shell and tube heat exchangers operating at the same temperatures and with the same flow. In this chapter a mechanism is developed for the thermal analysis of Compabloc units with the integration of the fouling model from *Chapter 2*. The mechanism includes an iterative procedure to determine the effect of fouling on the efficiency of the Compabloc unit.

The chapter also consists of different case studies for the analysis of accepted structures in their operation on crude oil pre-heat trains.

PART 2: THE COOLING WATER NETWORK

Part 2 of this work talks about the investigations associated with the cooling water network. First a method for the analysis of fouling in heat exchangers for the cooling process is introduced with its application in shell and tube heat exchangers and Compabloc units. Later this method is used in the thermo-hydraulic analysis of a cooling water network that is to be revamped.

Chapter 5: Fouling in cooling water networks

Cooling water systems are not exempt from the fouling process. Fouling in cooling water networks can be severe and drastically affect the performance of the network involved and the cooling tower being used. This chapter summarizes the crystallization fouling in cooling water systems and how they affect the heat transfer process.

The chapter is introduced with a model for the analysis of crystallization fouling in heat exchangers for the cooling systems. This model is further modified for use in crystallization fouling in Compabloc units. For crystallization fouling in shell and tube heat exchangers, design strategies are presented for future incorporation into the design algorithms for the cooling water networks. These strategies are a range of design parameters for the heat exchanger.

Case studies are presented to affirm the design options.

Chapter 6: Consideration of fouling in the design and modification of cooling water systems

As a follow up to *Chapter 5*, a cooling water network simulation is presented for the case where modifications are to be made to an existing cooling water network. As part of the modification process, the design steps of a cooling network is briefly presented.

A thermo-hydraulic simulation is used to determine where changes can be made in the cooling water network. The process involved the identification of the best position of a heat exchanger in the network that will not strongly affect the performance of the existing heat exchangers in the network and at the same time obtain the best efficiency of the new heat exchanger.

A case study is presented to support this thermo-hydraulic simulation.

PART 1

THE CRUDE OIL PRE-HEAT TRAIN

CHAPTER 1

FOULING AND FOULING MODELS

1.0. CHEMICAL REACTION FOULING

Chemical reaction fouling occurs downstream from the desalter but most importantly in the heat exchangers located at the hottest end of the pre-heat train and also in the fired heaters in the process line to the distillation column. Watkinson[9]defines chemical reaction fouling as a deposition process whereby a chemical reaction results directly in the formation of the deposit material or is involved in the formation of the deposit precursors which subsequently results in the deposition on the heat transfer surface and where the heat transfer surface is not directly involved in the chemical reaction. These precursors can form in the bulk of the fluid, in the boundary layer close to the surface or on the surface itself [4, 42]. The kinetics of chemical reaction fouling has been associated with three different reactions which may interact in the formation of the deposits: autoxidation, thermal decomposition and polymerization [5, 9, 10]. The mechanisms of these reactions are attributed to the reactions of the hydrocarbon species present in the fluid bulk under certain conditions during the operation of the heat exchangers. For example, while autoxidation is as a result of the presence of oxygen in the bulk flow, thermal decomposition occurs in the absence or presence of very little oxygen and at the hottest parts of the crude processing process. Agglomeration of the precursors and their solubility limits in the bulk of the fluid play very important roles in chemical reaction fouling.

Polymerization of precursors in the bulk of the fluid results in the formation of agglomerates of foulants for the deposition process. Epstein [27] has modeled the fouling process based on chemical reaction by polymerization whiles Wiehe [5] has studied fouling caused by polymerization of olefins. The availability of precursors in the bulk of the fluid may serve as a basis for the polymerization reaction.

Autoxidation often referred to as autocatalytic oxidation of hydrocarbons consist of a set of radical hydrocarbon reactions that result in the formation of hydro-peroxides and/o poly-peroxide gums and an extensive review is presented in Watkinson [9]. The products of these reactions are not mainly the cause of fouling but they control the formation of the fouling precursors [10]. Chemical reaction fouling by autoxidation is a result of the formation of precursors and their transport to the heat transfer surfaces where they settle, forming deposits. The deposition processes attributed to transport, reaction, mass transfer and attachment to the surface is greatly dependent on the precursor's concentration in the bulk of the fluid [1, 10].

Thermal decomposition is the degradation of hydrocarbons in the bulk of the fluid at high temperatures of operation of the heat exchanger. Generally it is associated with the production of precursors from asphaltene molecules that subsequently react to form substances of higher molecular weight called "asphaltene cores" in a decomposition reaction. The presence of precipitated asphaltenes in the bulk of the fluid lead to the formation of fouling deposits on the heat transfer surfaces of heat exchangers. The insolubility of asphaltenes and subsequent formation of deposits is influenced by the other constituents of the bulk of the crude: resins, aromatics and saturates. Further decomposition of the deposits result in the formation of coke on the heat transfer surface in the pre-heat train, a process coupled with deposit ageing [4, 10, 25]. The process for the formation of the deposits has also been related to a phase separation process [5]whiles a reaction scheme is also suggested [4] for the transformation of deposits from asphaltenes to coke.

Chemical reaction fouling in crude oil processing heat exchangers is a very complex process because the reactions involved may overlap in the formation of the deposits but thermal decomposition appears to be the dominant process at the hot end of the crude oil pre-heat train.

The factors that affect chemical reaction fouling include:

- Temperature – The rate of fouling increases with an increase in temperature. The modeling of chemical reaction fouling has been disputed between the use of the film temperature [26] of the viscous sub-layer where the deposition occurs or the wall and subsequently surface temperature [24].
- Velocity – The rate of fouling is inversely proportional to the velocity of the bulk flow of the fluid. Velocities have also been used in the modeling of chemical reaction fouling [23, 24, 27].
- The proportion of saturates, asphaltenes, resins and aromatics present in the bulk of the fluid. The proportions of these constituents of the crude oil entering the heat exchanger may cause asphaltene molecules to precipitate out leading to fouling [5].

1.1. CHEMICAL REACTION FOULING MODELS

1.1.1. Fouling factors

Fouling factors have been the basis of integration of a fouling component for the design of heat exchanger for operation over an "acceptable" operating period. These fouling factors are incorporated in the early design stages of heat exchangers and are provided for by the Tubular Exchangers Manufacturers Association (TEMA) [14]. TEMA data are available for different fluids at different temperatures of operation as well as for various heat exchangers. The dependence of the process industry on the use of the TEMA fouling factors in the design of heat exchangers is decreasing, this is because research indicate that fouling resistances depend critically on the operating conditions of the heat exchangers [18, 19, 43] and the use of TEMA fouling factors in the design of heat exchangers may result in an over specified or underspecified design.

1.1.2. Available models

1.1.2.1. Kern-Seaton Model

Most fouling models are developed based on the Kern-Seaton [11, 33] approach. The approach used defines the rate of development or growth of fouling as the net difference between the rates of deposition on the heat transfer surface and the rate of removal of the deposit from the surface. This model was developed for the prediction of particulate fouling in aqueous systems and is mathematically represented as:

$$\frac{dR_f}{dt} = \dot{m}_d - \dot{m}_r \qquad 1.0$$

Hewitt and Müller-Steinhagen [11] explained the analogy behind this modeling: material transport to the heat transfer surface is modeled as a diffusion process (mass transfer process) and adhesion to the surface as a chemical reaction of the key components in the fluid bulk whose kinetic rate constant is calculated from an Arrhenius-type relationship.

1.1.2.2. The Epstein Model

The Epstein model proposed by Epstein [27] is as a result of the analysis of the experimental data for the formation of styrene polymerization in kerosene from the work of Crittenden et al (1987) cited in [44]. The model focuses on a chemical reaction rate that occurs on the heat transfer surface as a result of the mass transfer of the reactants to the heat transfer surface. This model follows an earlier model developed by Crittenden et al cited in [44] from the same experiment and was developed to predict fouling in cases where the rate is mass transfer and reaction controlled.

Yeap et al [24] presents the Epstein model for a single order chemical reaction kinetic as

$$\frac{dR_f}{dt} = \frac{1}{\lambda_f \rho_f}\left[\frac{k_m C_b}{1 + \kappa u^2 \left(\frac{C_f}{2}\right)\left(\frac{\mu}{\rho}\right) k_m exp^{\left(\frac{E}{RT_s}\right)}}\right] \quad 1.1$$

This model was used in the determination of polystyrene deposition from a multicomponent solution of styrene in kerosene and with the reaction at the heat transfer wall. The model showed an excellent fit to the data used by Crittenden in their analysis.

1.1.2.3. The Panchal-Watkinson Model

The Panchal – Watkinson model was developed in 1993 [36]with further work on the model a year later [45]. This was based on the reaction steps on the generation of the precursors and then the formation of the foulants. The aim of the work was to investigate the governing mechanisms in the fouling process using a chemical system of olefin-based fouling which produces poly-peroxides in the presence of dissolved oxygen and indene as the impurity.

The research modeled the generation of the precursors and formation of the foulants at different sections of the bulk of the flow including the boundary layer, the bulk of the fluid and the heat transfer wall. The results for the generation of the precursors at the boundary layer and subsequent formation of foulants at the heat transfer wall gave the best predictions with experimental data.

1.1.2.4. The Ebert-Panchal Model

The Ebert-Panchal (EP) model [26], given by Equation 1.2 was developed to model the "threshold" fouling levels from the analysis of an Exxon crude oil.

$$\frac{dR_f}{dt} = a_1 Re^{a_2} exp^{\left(\frac{-E}{RT_f}\right)} - a_3 \tau_w \tag{1.2}$$

Where the $a_i^{\prime}s$ are constants that the user had to find experimentally or by regression analysis.

The Ebert Panchal model is one of the earliest works in defining a fouling threshold for the design and operation of heat exchangers. The basis of the model is the development of a correlation to be able to predict conditions above which fouling cannot be observed. The model consist of an integrated reaction term involving the film temperature present in the boundary layer [26]. In this same work the authors suggested that the removal of foulant from the boundary layer is by some transport mechanism attributed to the wall shear stress and may be different from the removal of deposits from the heat transfer surface as explained in the Kern and Seaton model. Wilson et al [23] explains the significance of this model in the investigations into fouling of heat exchangers.

A generalized form has been developed, Equation 1.3, which has seen a lot of applications in the development of software for fouling analysis [23, 46, 47]. This consists of two opposing mechanisms: first is a deposition mechanism in which the thermal film is defined as the volume of a "chemical reactor" in which the reaction follows an Arrhenius relationship.

$$Depostion\ term = \frac{A}{\alpha} exp^{\left(\frac{-E}{RT_f}\right)} \tag{1.3}$$

The deposition term is based on the film temperature rather than the wall temperature as used in other models [30].

$$T_f = 0.55T_w + 0.45T_b \tag{1.4}$$

The second, is a suppression term postulated as a linear function of the wall shear stress. This was a result of the observation that an increase in the velocity of the flow results in a decrease in the rate of fouling.

$$Suppression\ term = \gamma\tau_w \qquad 1.5$$

The generalized Ebert Panchal model is thus given by

$$\frac{dR}{dt} = \frac{A}{\alpha} exp^{\left(\frac{-E}{RT_f}\right)} - \gamma\tau_w \qquad 1.6$$

The three parameters (A, E and γ) are derived from the analysis of fouling data. There is no physical basis that the effect of wall shear stress on the fouling rate should be a linear relationship or be defined as a suppression term subtracted from the deposition term.

The EP Model has been found to fit industrial data quite well. Polley et al [48, 49] outlines the use of the generalized Ebert Panchal model in advance applications for heat recovery and fouling mitigation. The extensions to the application of the model is suggested for the passing of the crude oil in the shell side of heat exchangers and when tube inserts are applied in the mitigation process. Very good predictions were obtained in these analyses with the model, however, this is to be expected with the model that has three parameters that need to be evaluated and adjusted for prediction purposes. Of particular note is that in order to fit data for exchangers fitted with tube inserts the suppression constant has to have quite a low value, typically 3.8×10^{-9} m^2K/WhrPa [49].

1.1.2.5. The Ageing model – Work at University of Cambridge

Ageing is the process of transformation of a fouling layer following its initial formation, whereby the fouling deposit converts into another, usually more cohesive form[25]. This is because the deposits, which are static layers on the heat transfer surface, are continually exposed to the operational conditions. The different deposits resulting from the fouling mechanisms are suggested to experience ageing on the heat transfer surface. Ageing is noticed when the thermal properties of the deposits is different from that of the initial deposits and thus affecting the thermal conductivity property of the deposit, an integral part in fouling analysis.

Wilson et al [25] developed a quantitative model based on a simple reaction principle of the work done by Fan and Watkinson. The overall fouling deposit over a period of time is defined by a sum of resistances of layers of deposits over the individual periods of the deposition process. The thickness of a fouling deposit is characterized by the thermal thickness (Equation 1.7) over each time interval given by

$$\delta_i = \left.\frac{d\delta}{dt}\right|_i . \Delta t = \lambda_{f,0} . \left(\frac{dR_{f,s}}{dt}\right) . \Delta t \qquad 1.7$$

A fouling model is then used to express the rate of deposition on the heat transfer surface. The overall resistance over the period defined will thus be a sum of the individual resistances of each layer as depicted in Equation 1.8.

$$R_f = \sum_{i=1}^{n} \frac{\delta_i}{\lambda_{f,i}} \qquad 1.8$$

CHAPTER 2

DEVELOPMENT OF A NEW MODEL FOR FOULING IN HEAT EXCHANGERS HANDLING CRUDE OIL – ASPHALTENE PRECIPITATION (AP) MODEL

2.0. CONSTRUCTION OF FOULING MODELS

A fouling model is a mathematical expression that takes into consideration the physical properties of the system to be analyzed to be able to search for a pattern in measured data for heat exchangers and make prognostics about their future occurrence. The purposes of a fouling model is to provide a means of relating the rate of fouling encountered within a heat exchanger to its operating conditions and subsequently predict the behavior of the fouling process. Different fouling models have been applied for the analysis of the fouling rates in heat exchangers.

The practicability of a fouling model depends on the simplicity and the ability to be able to apply it over varying situations. Simplicity is not only got to do with the easy representation of the model but also the need to be able to provide simple logic to explain the parameters involved. This helps to emphasize the physical basis of the parameters used in the model. Moreover a model does not only have to be applicable mathematically but also applicable in the explanation of the physical and chemical process with which it represents. In addition a fouling model must allow the designer and operator of a heat exchanger to take quantitative decisions in maintaining fouling levels at manageable levels for the smooth running of the plant [23].

2.1. ASPHALTENE DESPOSITION AND THE SUB-LAYER THEORY

There has not been a concrete model to explain the asphaltene precipitation and deposition in chemical reaction fouling and the models that are available look so complex and complicated that simple application of the model becomes very difficult. This may be partly due to the lack of available information on the mechanism of the operation, which is linked to a lack of understanding to this complex process. The Asphaltene Precipitation model seeks to improve on the predictions of the already developed models in use and also present a very simple but logical explanation to what is probably going on in the film layer of the heat transfer surface from a combination of analysis of independent investigations into the mechanisms of fouling over the years.

2.1.1. Fluid behavior close to the heat exchanger wall

The fluid behavior close to the wall is described as unstable, unpredictable and very difficult to analyze [7, 8]. The flow regime through a tube is defined by a region of turbulent flow where the flow is at its mean velocity, a buffer region where the flow velocity is smaller than then mean flow velocity due to the interaction between the turbulent phase and the laminar phase and finally the flow close to the wall called the boundary flow. The viscous or laminar sub-layer is a thin film within the boundary layer and closer to the wall where the viscous forces of the fluid is suggested to overcome the inertia force of the bulk flow of the fluid. It is also the region close to the wall or surface where the fluid velocity is most retarded. It is in this flow region that fluid behavior is unsteady partly due to the turbulent fluctuations above the layer causing a periodic inflow and outflow of fluid eddies and also to the interactions between the wall and the bulk of the flow. This flow region is an important factor in predicting the magnitude of the deposition and the removal process [7].

The deposition process is also suggested to occur in two steps: transport to the edge of the sub-layer by turbulent eddies and then transport across the sub-layer [50]. The movement of particles into the sub-layer is generally attributed to Brownian diffusion of fluid particles related to the free flight model [7] until the particles reach the edge of the sub-layer. Recent studies also relate it to eddy diffusivity which vanishes when the particles gets close to the wall [1, 7] and other investigators to the turbulent burst model [8] where high speed fluid perturbations from the region above the sub-layer causes an inflow and outflow of fluid [7, 51, 52].

An inference can then be drawn of the fact that transport of particles or agglomerates of molecules in the sub-layer is different and needs to be well defined if we need to understand the flow in this region. The inflow to the sub-layer is considered a more gently flow with the scoops following a defined streamline while the outflow is more abrupt. Yung et al [8] postulates that particles carried along with the scoops get settled onto the surface and remain there because of the adhesive forces from the particle – particle, particle - wall interactions and gravity. The particles arriving at the surface of the sub-layer are postulated to stick and become part of the deposit layer residing on the surface [1].

Cleaver and Yates [7] developed a sub-layer model for the deposition of particles. They suggest from research and experiments that a particle entrained in a downward sweep will continue in its trajectory with little interruption but its contact with the surface will depend on what position the particle gets entrained in the sweep and its inertia. They further point out that only particles between bursts will have the opportunity to approach the surface since a particle trapped in a burst will eventually be carried away from the surface. They conclude that the particle relaxation time has a direct relation with deposition rate. The inertia phenomenon in the deposition process is also confirmed by Yung et al [8] and Wu et al [50] in their work to develop a model for particle deposition and bounce off.

On the contrary, there exist the removal process which is predominantly controlled by a turbulent bursts. Turbulent bursts, the major cause of particle re-entrainment is believed to happen in two stages, first an ejection of the deposit from the surface and secondly a sweep from the point of detachment [52]. The burst rates increase with fluid velocity and hence heat transfer [8, 53] but the rate of re-entrainment is not directly proportional to the frequencies of the bursts. With each fluid of the burst away from the surface inflow fluid will bring depositing materials to the surface and this will result in a competition between the inflows and the bursts. The turbulent bursts occur periodically over a small fraction of the control area but may eventually cover the whole area prompting for more fluid inflow into the sub-layer.

The fluid shear has a significant impact on deposition and removal rates of asphaltene molecules along with the particle – wall interactions [54]. The initial stage of particle re-entrainment is attributed to the horizontal force, implying that a critical shear has to exist before re-entrainment occurs [8]. Considering that in the viscous sub-layer the only force acting on particles will be the viscous drag force, the work by Yung et al [8] and Akbarzadeh et al [54] confirm the importance of the shear stress on the attachment and detachment of deposits from the heat transfer surface. Chen [53] in the investigation of a mechanistic insight into the convective heat transfer from the wall to the fluid in turbulent flows found out that the average time between bursts is given by:

$$\Delta t = \psi \frac{\mu}{\tau_w} \tag{2.0}$$

This equation (2.0) implies that an increase in the wall shear stress will decrease the average time that a particle or deposit stays on the heat transfer surface decreasing the rate of fouling in the heat exchanger.

2.1.2. Asphaltene reactions, precipitation and deposition

Asphaltenes are defined as the fraction of petroleum that is insoluble in non-polar organic compounds such as heptane (or hexane) but soluble in solvents of high surface tension such as benzene or toluene or carbon tetrachloride. Early investigations like Eaton and Lux [4] as well as Dickakian and Seay [6] have explained the mechanism of fouling in heat exchangers to be a result of asphaltene molecules present in the bulk flow. The asphaltenes present in petroleum both as non-reacted and asphaltene cores constitute the greatest in the deposition of asphaltenes. They exist in petroleum in the form of colloidal suspensions [5, 9, 44, 55]. Small particles of asphaltenes present in a petroleum mixture are soluble but may form coagulates when there is a change in the composition or level of the other fractions in petroleum [6] and at certain temperatures [12, 44] resulting in their deposition.

The formation of deposits by asphaltenes is seen as a complex process involving physical, chemical and thermodynamic processes [54]. The precipitation of the asphaltenes molecules from the bulk of the flow alone does not constitute the formation of a deposit but certain oil incompatibilities in oil blends as well as other physical properties of the oil like the bulk temperature, presence of insoluble particles and the amount of the precipitated asphaltene account for the deposition of asphaltenes [5, 10, 54, 56, 57].

In chemical reaction fouling which occurs at the hotter end of the pre-heat train, asphaltenes are the main contributors in the fouling process. Hong and Watkinson [12] cited a work by Wiehe where asphaltene molecules dissolved in the bulk of the fluid when the temperature was raised above 200 ^{o}C and they further confirmed this over the temperature range of 60 ^{o}C to 300 ^{o}C. Very high temperatures are conditions for the formation of coke. The formation of coke from asphaltene cores does not re-dissolve in the bulk flow. Therefore even though solubility is enhanced at these high temperatures in the heat exchanger, the thermal decomposition of asphaltene molecules to form coke prevents the dissolution of the precursors formed encouraging fouling. This is further confirmed by Asomaning [44] on his investigation that asphaltenes are not very active species at temperature conditions below 200 ºC but raising the temperature above this level present different forms of thermal reaction processes for the asphaltene molecules, the presences of asphaltene cores.

Wiehe [5] continues that the thermal decomposition involving asphaltene molecules consist of a two stage reaction: firstly, there is the decomposition of maltenes and other saturates to form asphaltenes, increasing the composition of asphaltene in the bulk of the flow and secondly the thermal decomposition of the asphaltene and its core molecule to form coke. These two reactions occur in series. The first reaction the conversion of maltenes into asphaltenes is the rate-controlling step characterized by an Activation energy of 44.3 kJ/mol. The physical process associated with the precipitation and transport of asphaltene molecules to the heat transfer surface has been investigated extensively and various postulations have also been made about the mechanisms involved [5, 44, 56, 58].

Asphaltene deposition has also been related to the changes in the asphaltene on-set pressure [54, 59]. In these investigations some parameters were defined as important in the modeling of asphaltene deposition: a shear term, a particle – wall collision efficiency and a critical particle size. Akbarzadeh et al [54] introduced the depletion effect of the deposition of asphaltene molecules. Depletion is defined as the reduction in the concentration of asphaltene molecules in the bulk of the fluid as a result of the growth of asphaltene agglomerates to a critical size such that deposition on the heat transfer surface is not favored. The effect of the particle size on the sticking probability is also investigated [51].

Kern and Seaton (cited in Yung [8]) relates re-entrainment to the shear stress due to velocity gradient whiles Chen [53] explains that the higher the shear stress the less time a particle will be in contact with the surface. Eskin [58] gives values for sticking probability in their analysis of particle deposition, concluding that the degree of the deposition rate strongly depends on the particle relaxation time. This clearly explains that whiles the unsteadiness of the sub-layer allow for the transport of the asphaltene particles to the surface, the particle-particle interactions, coupled with the particle-wall interaction and the effect of the shear stress on these particles contribute to how much time the deposited asphaltene molecule will stick to the surface and stay there or be carried away by the next burst.

Moreover from the continuous inward sweep of pockets of fluid from outside the sub-layer and a subsequent burst away from the layer, the rate of these sweeps is found to increase with increasing shear. This suggests that this phenomenon will eventually affect the probability of the particle attaching to the heat transfer surface to form a deposit. The following can therefore be concluded:

1. While at higher shear stresses a particle will spend less time attached to the heat transfer surface, the contrary is also true at low shear stress.
2. The solubility of asphaltene molecules has a limit above which foulants are formed leading to deposition. Other factors also affect this solubility including the presence of resins, aromatics or saturates.
3. The size of the deposit does not only depend on the asphaltene core but its ability to form agglomerates.
4. The transport of particles to the boundary layer may take different forms but entering the viscous sub-layer, the particle – particle interaction and particle – wall interactions also play a vital role in reaching the heat transfer surface.
5. Turbulent bursts occur taking scoops of fluid and particles away from the heat transfer surface but then it also contribute to the introduction of “fresh” deposits to the surface. These bursts are related to the fluid velocity and thus the fluid shear stress.
6. Re-entrainment may occur periodically over a small heat transfer area but they are not directly proportional to the turbulent bursts.

2.2. THE ASPHALTENE PRECIPITATION MODEL

A mathematical model needs to have the tendency to not only predict and explain the system under consideration but it should also be able to explain the physical phenomenon behind its application. While so many different models have been developed to predict and analyze fouling in heat exchangers, some of these models have been complex in their applications [30].

With the discussion given above about the behavior of the sub-layer, the asphaltene precipitation and deposition, the Asphaltene Precipitation model is thus presented to consist of a deposition term and a factor for the probability of attachment.

2.2.1. The Deposition term

The Asphaltene precipitation model like the generalized Ebert-Panchal model is modeled to consist primarily of a deposition process for the rate of fouling of the petroleum macromolecules characterized by the Arrhenius relationship for chemical reactions. The use of the Arrhenius relationship is justified in a work about the kinetic model for coke formation in thermal conversion processes from asphaltene molecules [5, 26, 33]. The deposition term is modeled as a reaction occurring in a "reactor volume" equal to the thickness of the "thermal film" next to the hot surface. This is represented below in Equation 2.1:

$$Deposition = \frac{A}{\alpha} exp\left(\frac{-E}{RT_f}\right) \qquad 2.1$$

T_f is the film reaction temperature, which is defined as a relationship between the bulk temperature T_b and the wall temperature T_w and given in Equation 2.2:

$$T_f = 0.55T_w + 0.45T_b \qquad 2.2$$

2.2.1.1. The Activation Energy

The two step reactions postulated by Wiehe [5]is applied here in the deposition term. In his investigation he found the activation energies for both reactions. The asphaltene cores from the first reaction were subject to a solubility limit. If the limit is exceeded, the molecules precipitate. The activation energy for the rate-controlling step is reported as 44.3 kJ/mol.

This limit in precipitation implies this value can be used as the Activation Energy in the model for chemical reaction fouling in heat exchangers processing crude oil.

2.2.1.2. The Deposition constant

The deposition constant *A* in the deposition term of the model plays a significant role in fitting the model. There are still unanswered questions about the choice of value for this constant but in the case of the AP model it becomes the only constant that needs to be determined and its choice becomes one of simple analysis. Variations in the deposition constant in the different analysis presented are worth noting. The important thing here is the deposition constant has remained the same when fitting the model to different parts of the operating period under investigation.

It is postulated that the deposition constant may be linked to the composition of the crude oil or linked to the temperatures at which the crude oil is processed. Analysis of data from different sources and operating conditions will be necessary to be able to conclude on this observation.

2.2.2. The sticking probability term

There is no physical basis for assuming that the effect of fluid shear stress on fouling rates is either a linear function of shear stress or that it involves a suppression term that should be subtracted from the deposition rate. Moreover activities in the sub-layer as explained earlier indicate that there is a sweep of the precipitates towards the heat transfer surface which tend to rather “stick” to the surface because of the strong particle – wall and particle – particle interactions and the influence of the fluid shear stress. If we assume the likelihood of a precipitated particle attaching to a surface will be affected by the time in which it is close to the surface or deposit we can see the effect of shear can be interpreted in terms of a “sticking probability”, P. So, we have a new model defined as in Equation 2.3.

$$\frac{dR}{dt} = \frac{A}{\alpha} exp\left(\frac{-E}{RT_f}\right) P \qquad 2.3$$

The author suggests this sticking probability to be related to a shear factor. The shear factor has limits of zero (occurring at lower shear stress) and unity (occurring at higher shear stress). If we assume an upper limit of 100 Pa for shear stresses encountered in some welded plate heat exchangers like Compabloc units and a lower shear stress limit of 2 Pa, the Shear Factor, F can be defined as:

$$F = \frac{\tau_w - 2}{98} \qquad 2.4$$

The sticking probability, P is then assumed to be given by the function

$$P = 1 - F^n \qquad 2.5$$

For values of the shear stress less than 2 *Pa*, assume a shear factor of zero and subsequently a sticking probability of one. In the case where the shear stress is greater than 100 *Pa*, assume a shear factor of one and finally a sticking probability of zero.

This equation can give a range of shapes for the sticking probability and this is explored below graphically in Figure 2.0 with a range of values for *n* from 0.25 to 2.

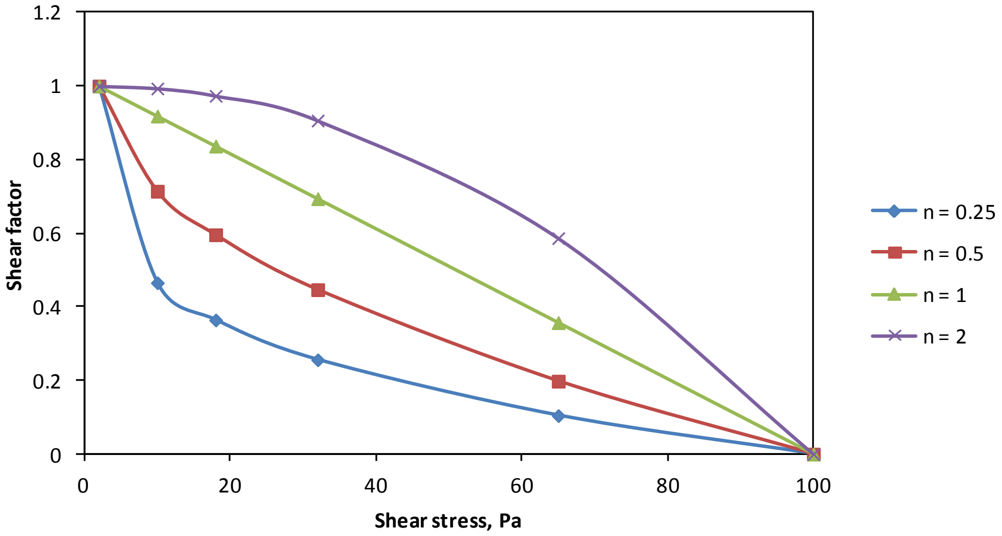


Figure 2.0: Sticking factor profiles showing the effect of the exponent *n* on the shear factor

A value of 0.5 has been found to fit experimental data well.

The Asphaltene precipitation model is thus given by

$$\frac{dR}{dt} = \frac{A}{\alpha} exp\left(\frac{-E}{RT_f}\right)\left[1 - \left(\frac{\tau_w - 2}{98}\right)^{0.5}\right] \qquad 2.6$$

2.2.3. Advantages to other models

The AP model has the following advantages to other models:

1. The model is simple and easy to use preventing dependence on most of the physical properties of the crude oil.
2. The model only present the need to predict and vary the deposition constant compared to other cases like the EP model where values for the deposition constant, suppression constant and the activation energy needs to be predicted.

3. The model agrees with current research that influences of a particle – wall interaction and critical shear factor causes the precipitates to rather stick to the heat transfer surface as long as conditions will allow it.
4. The model has a physical explanation to the interactions on the heat transfer surface and represents this in the mathematical model.

2.3. APPLICATIONS

The AP model is tested with the results of the laboratory study by Knudsen et al [41] at the Oregon State University and plant data from the use of a shell and tube heat exchanger and a Compabloc units on different crude oil pre-heat trains.

2.3.1. Predictions from laboratory data

The work at Oregon State University by Knudsen et al [41] focused on the investigation of a threshold fouling curve for crude oil. The experiments were conducted with velocity ranges from 0.91 m/s to 3.1 m/s. The results confirmed the existence of a fouling threshold while an increase in the velocity in the test section resulted in an increase in the surface temperature but this tended to reach an asymptotic level at temperatures greater than 330 ºC. Also the fouling rate increased with an increase in the surface temperature at a fixed bulk temperature.

The AP model is applied to the results from this experiment. The activation energy defined earlier is applied and a deposition constant of 225 gave good predictions of the fouling rate.

Table 2.0
An extract of the results from the experiment at Oregon State University with predicted fouling rates

Run No	**Velocity, m/s**	**T film, K**	**Measured rate, $\times 10^{-6}$m^{2}W/Khr**	**Predicted rate, $\times 10^{-6}$m^{2}W/Khr**	**% Error**
41	0.91	219.4	8.0	5.63	-29.6
41A	0.91	219.4	5.63	5.63	0.0
42	0.91	227.1	8.64	6.62	-23.4
50	1.68	257.9	4.7	6.33	35.7
32	2.44	265.6	4.12	4.73	14.8
38	2.44	265.6	4.0	4.74	18.5
38A	2.44	265.6	5.31	4.74	-10.7
56	3.05	265.6	3.06	3.57	16.7

Runs 41, 41A and 42 are repeat experiments at a velocity of 0.91 m/s whiles runs 41 and 41A are at the same film temperature of 219.4 ºC. Runs 32, 38 and 38A are repeat experiments at a velocity of 2.44 m/s and film temperature of 265.6 ºC.

As it can be seen from Table 2.0, the AP model gives predictions with errors less than forty percent from the measured fouling rates and that are well within prediction range. The average difference between prediction and measurement was just 2.6% and the root mean square error was just 20%.

2.3.2. Predictions from plant data

The application of the AP model to predict fouling from plant data is of a great advantage to its use as a tool for design, diagnosis and mitigation. For this work, the model is applied to a shell and tube heat exchanger and a welded plate heat exchanger of the type Compabloc (designed and built by Alfa Laval Inc.). Firstly, results of the performance of a shell and tube heat exchanger (part of a refinery pre-heat train) are presented and then that of the performance of Compabloc exchanger.

One of the ways for fitting the fouling model and make predictions of fouling rates is by analyzing the inlet conditions of the streams under investigation. This is because the operation of for example heat exchangers in the crude processing industry is characterized by non-uniform data recording. Results of changes in plant operating conditions are closely related to the fouling rates. These plots need to be examined and a “slice” of the data is taken in the range where no abrupt changes occurred for analysis. This methodology is employed here in the analysis of the plant data.

2.3.2.1. Shell and tube heat exchanger

This case is a shell and tube heat exchanger located at the hot end of a pre-heat train. The hot stream is a pump around stream from the distillation unit with an average inlet temperature of 280 ºC transferring heat to the crude oil flowing through the tubes of this exchanger with an average inlet temperature over the operating period of 224 ºC.

The fouling analysis of the heat exchanger is presented. The measured fouling resistances increased over different operating periods as shown in Figure 2.1 and the crude oil outlet temperature is presented in Figure 2.2. A deposition constant of 150 fitted well the predictions with the measured fouling resistances.

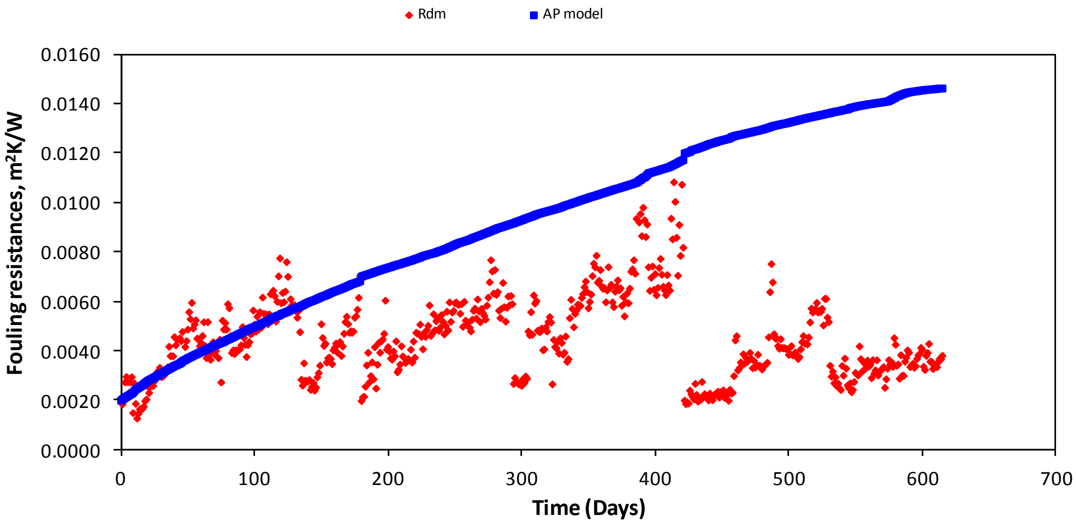


Figure 2.1: Fouling resistance plot for the shell and tube heat exchanger (Whole operating period). Rdm – measured fouling resistances and AP model – predicted fouling resistances.

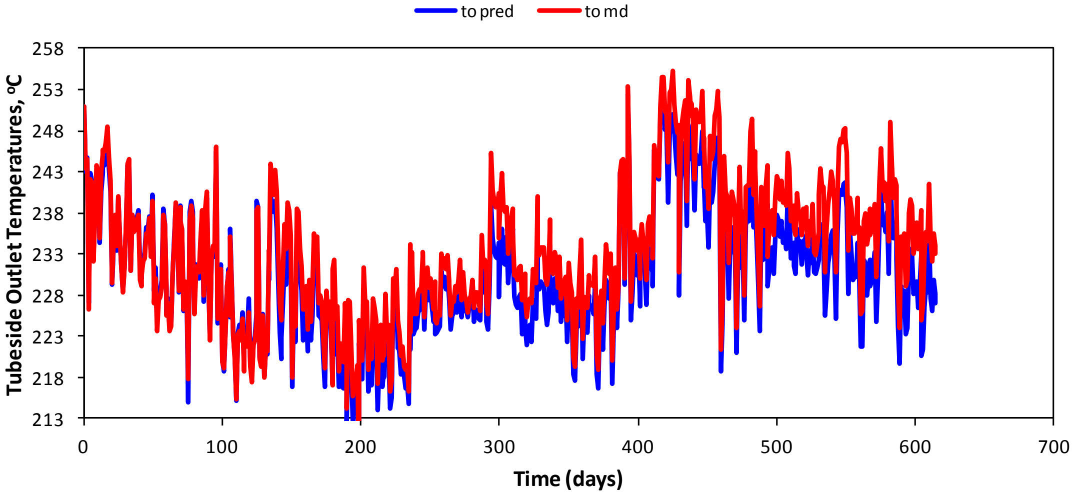


Figure 2.2: Crude oil temperature predictions compared to the measured outlet temperatures in the operation of the heat exchanger (to pred – predicted outlet temperature, to med – measured outlet temperature)

Figure 2.1 shows that there were no consistencies in the development of fouling during the operating period under investigation. To further analyze the choice of deposition constant, data in the operating period 140 – 300 days are selected. The result of this analysis is presented in Figure 2.3. Comparing the results for the chosen range (Figure 2.3) and the data for the whole operating period (Figure 2.1), a well-fitted model is observed for the measured data and predictions.

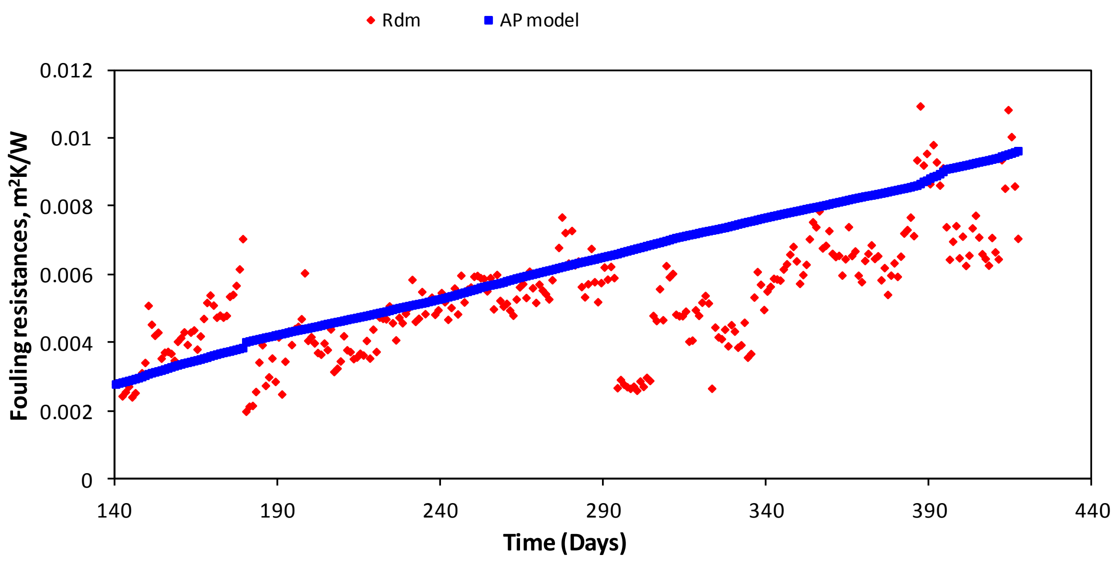


Figure 2.3: Fouling resistance plot for the shell and tube heat exchanger (selected period of operation). Rdm – measured fouling resistances and AP model – predicted fouling resistances.

Further comparison of the temperature predictions with measured values for the sliced region as presented in Figure 2.4 is in great agreement for the model applied. It should be noted that the model also fitted well the data range from day 290 to 420 with the same deposition constant.

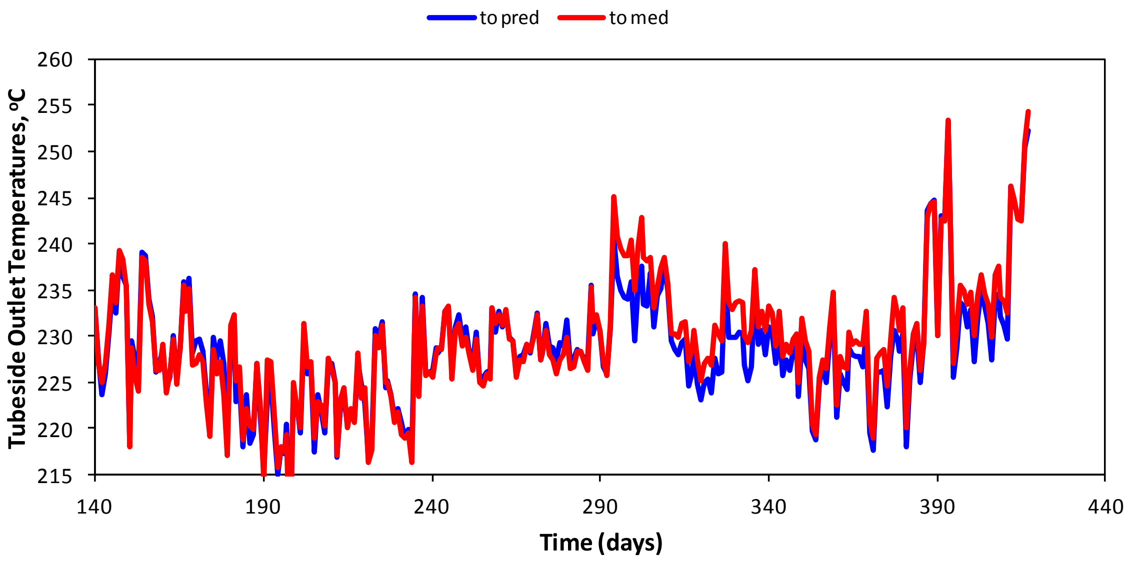


Figure 2.4: Crude oil temperature predictions compared to the measured outlet temperatures in the selected operating period of the heat exchanger (to pred – predicted outlet temperature, to med – measured outlet temperature)

Another shell and tube heat exchanger analyzed from a different pre-heat train in a refinery in Europe is also presented below. This heat exchanger has an average crude inlet temperature of 170 ºC and a shell side flow (Heavy gas oil) inlet temperature of 242 ºC. A deposition constant of 200 fitted both operating periods very well. Figure 2.5 shows the predictions of the model for the first operating period and Figure 2.6 for the second operating period. An average wall shear stress of 9 Pa was calculated from the data.

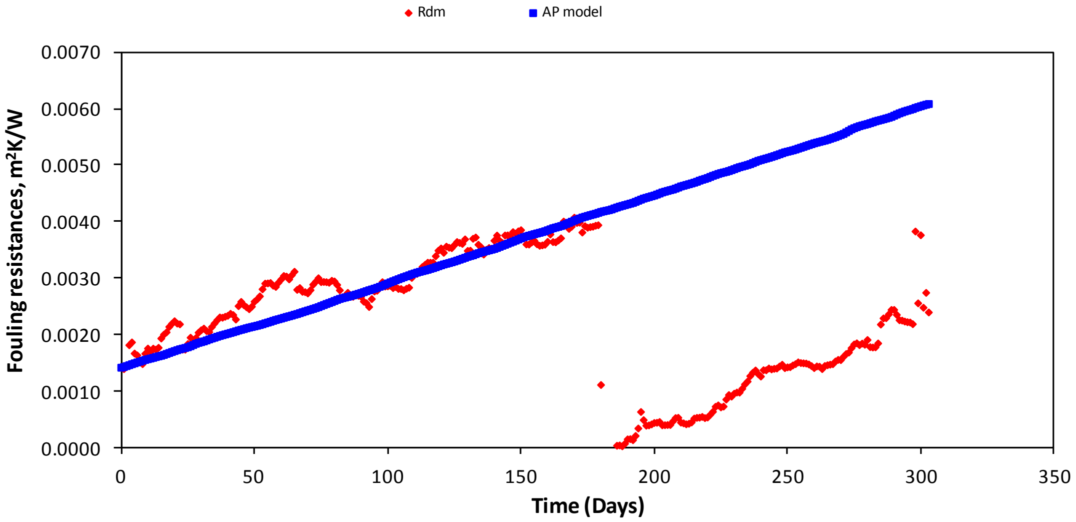


Figure 2.5: Fouling resistance plot for the shell and tube heat exchanger (total operating period). Rdm – measured fouling resistances and AP model – predicted fouling resistances.

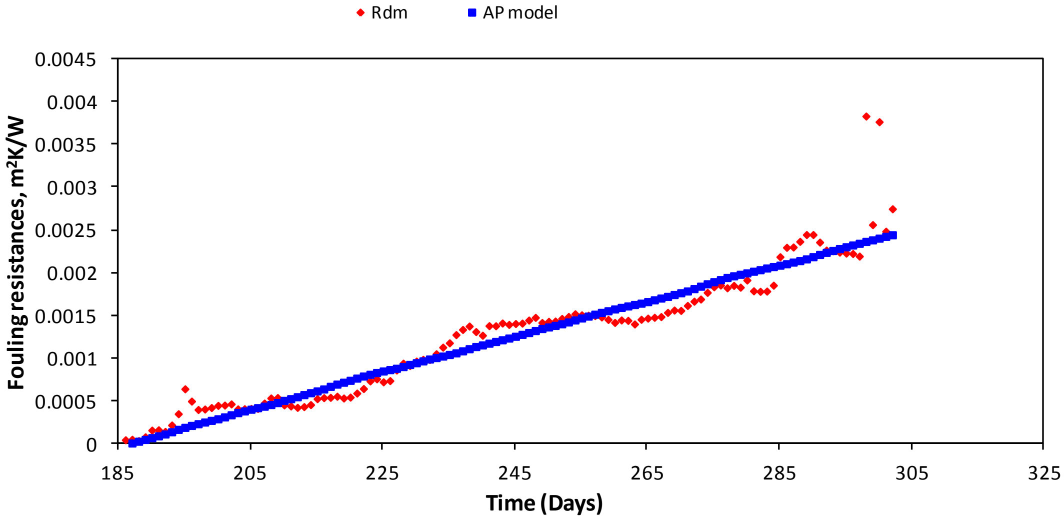


Figure 2.6: Fouling resistance plot for the shell and tube heat exchanger (second operating period). Rdm – measured fouling resistances and AP model – predicted fouling resistances.

The measured fouling data in Figure 2.5 shows that the operation of the heat exchanger was probably interrupted with a scheduled cleaning operation causing the measure of fouling resistances to start from zero and increasing linearly.

The temperature predictions of the heat exchanger treating independently the two identified operating periods (first operating period and second operating period) gave very good fits for the chosen deposition constant. This is presented in Figures 2.7 and Figure 2.8 respectively.

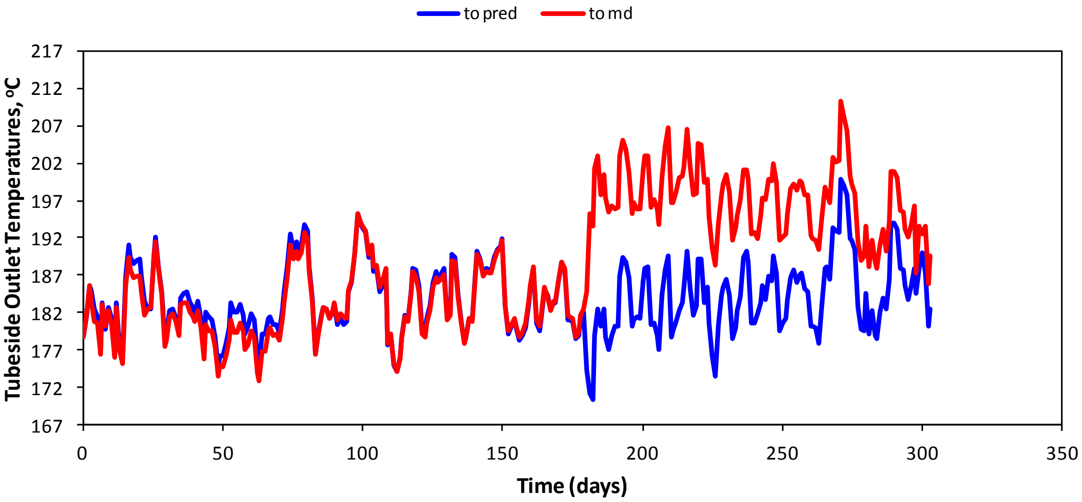


Figure 2.7: Crude oil temperature predictions compared to the measured outlet temperatures in the operation of the heat exchanger (to pred – predicted outlet temperature, to med – measured outlet temperature)

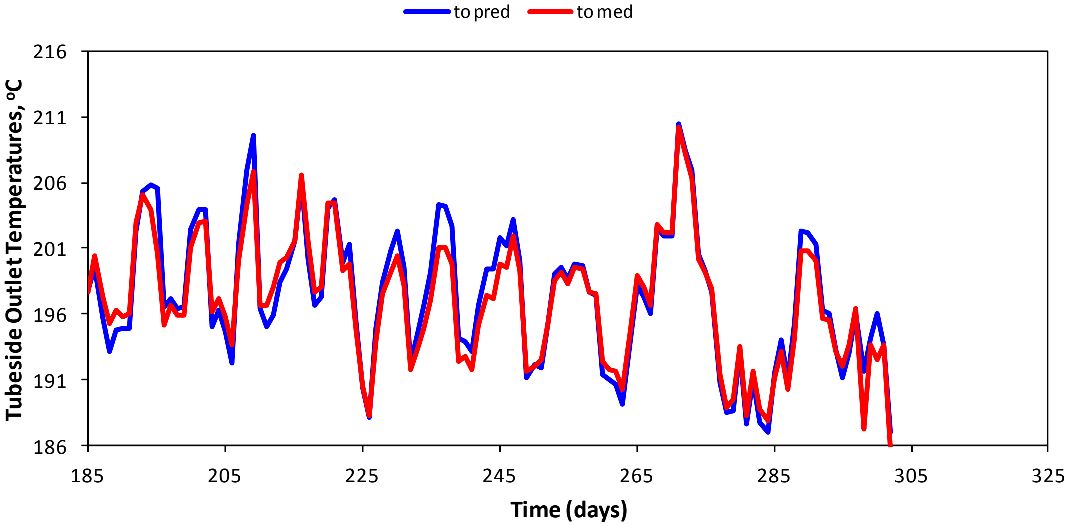


Figure 2.8: Crude oil temperature predictions compared to the measured outlet temperatures in the (second operating period) of the heat exchanger (to pred – predicted outlet temperature, to med – measured outlet temperature)

2.3.2.2. Compabloc exchangers

Evidence of the great importance of this fouling model came to light when it was applied in the performance analysis of a Compabloc unit. Compabloc units are compact, welded plate heat exchangers and present great advantage in space and in heat recovery (*refer to Chapter 4*). Compabloc units due to its design have high fluid shear stress values and so do not foul as much as heat exchangers but during a period of bad operation may foul greatly due to the high wall temperature and heat transfer coefficients that it presents.

The basis for the analysis of the Compabloc units was an observation made in the operation of one of these units in a known pre-heat train. Initially the units were operating at a high fluid shear stress (70 Pa, Figure 2.9) but in the second operating period the shear stress was reduced to about one-fifth the original value (Figure 2.10) and severe fouling was observed. The predictions and measured fouling resistances in this Compabloc unit for the two operating period are represented below. In both cases a deposition constant of 80 was found appropriate.

At a wall shear of 70 Pa:

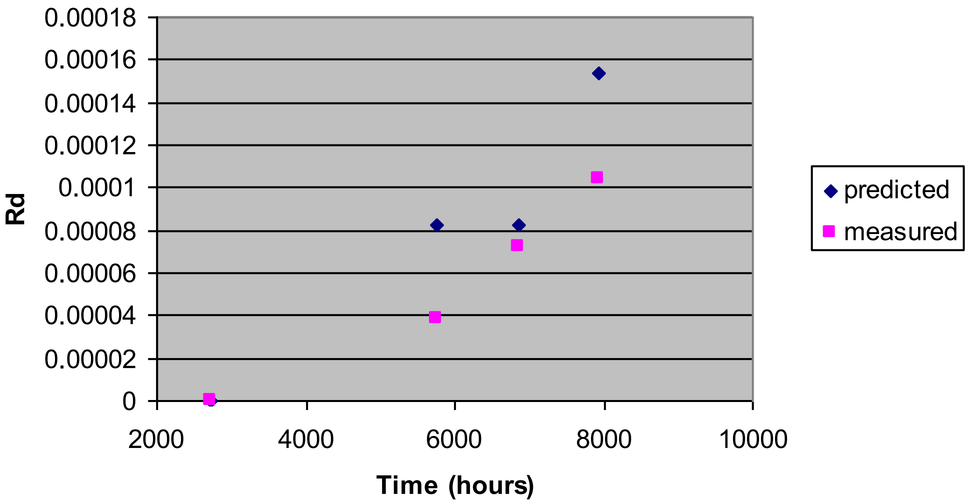


Figure 2.9: Fouling resistance measured in the first operating period with a shear stress of 70Pa of the Compabloc exchanger

And around 17 Pa:

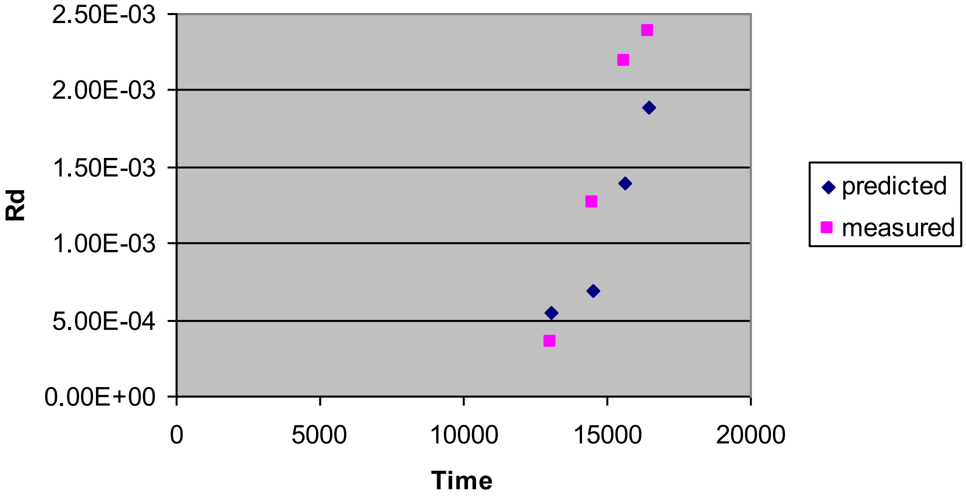


Figure 2.10: Fouling resistance measured in the first operating period with a shear stress of 17Pa of the Compabloc exchanger

The high fouling resistances registered in Figure 2.10 are due to the low fluid shear stress values encountered in the Compabloc exchanger. Even though the fluid shear stress value of 17 Pa may be equal to or greater than that found in shell and tube exchangers, the fouling resistances encountered are substantially high compared to the data in Figure 2.9. This enhanced heat transfer in Compabloc exchangers account for this high level of fouling rates.

Other Compabloc exchangers (EX1 and EX2) are therefore.

The Compabloc units EX1 and EX2 are located on a pre-heat train of a crude processing unit in the USA. These units are placed in parallel and transferring heat between the crude oil and hot asphalt leaving the distillation column. It is noted that there was a great variation in the recorded viscosities of the hot asphalt at the inlet and outlet of the Compabloc unit. The data for the streams is presented in Table 2.1.

Table 2.1
Stream data for Compabloc units

Stream	Flow rate, kg/s	Inlet Temperature, C	Number of passes
Crude oil	28	234	9
Hot asphalt	13	341	12

A deposition constant of 50 was used for Compabloc unit EX1 and a deposition constant of 5 was used for Compabloc unit EX2. The predicted fouling resistances of the units are produced in Figure 2.11 for EX1 and Figure 2.12 for EX2. An average wall shear stress of 30 Pa was encountered in both units.

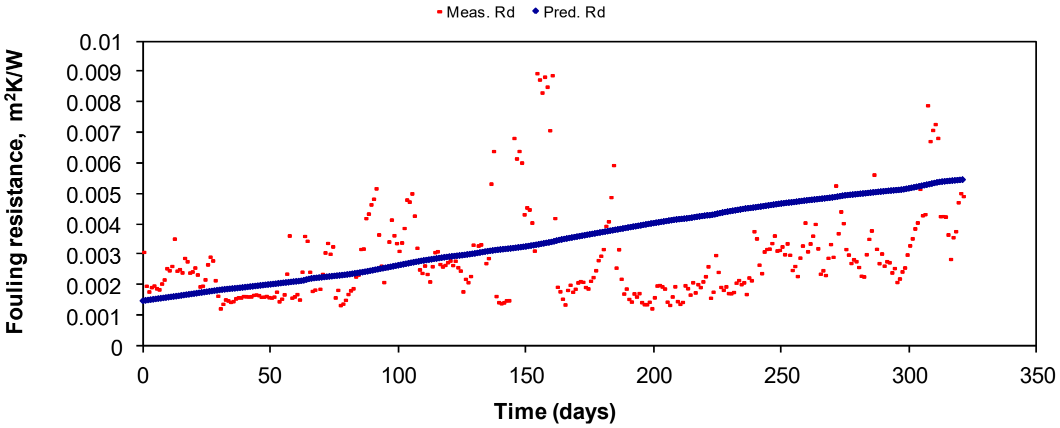


Figure 2.11: Predicted and measured fouling resistances for Compabloc unit EX1

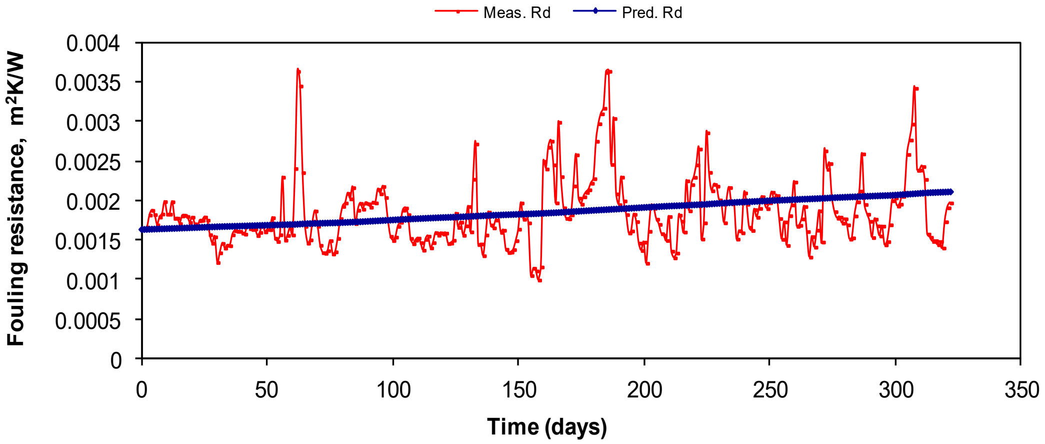


Figure 2.12: Predicted and measured fouling resistances for Compabloc unit EX2

The predicted temperature plots for the crude oil are presented in Figure 2.13 for unit EX1 and Figure 2.14 for unit EX2.

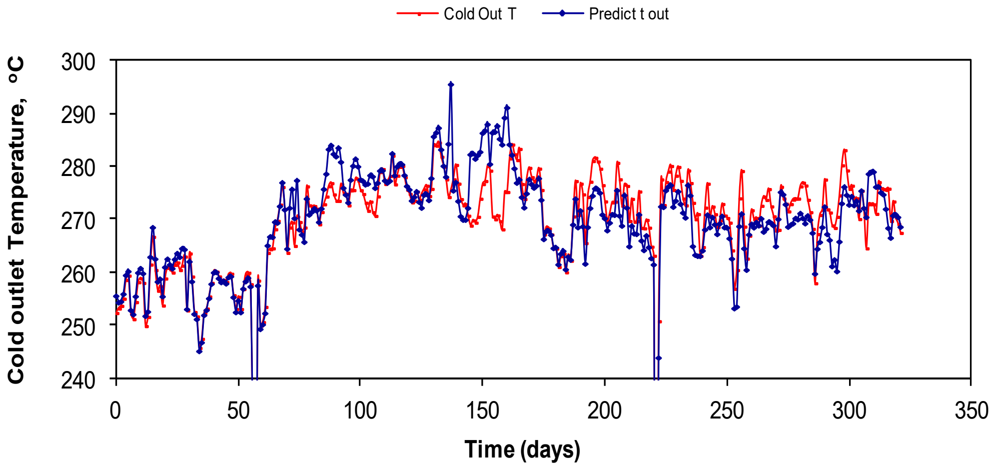


Figure 2.13: Predicted and measured temperature for Compabloc unit EX1

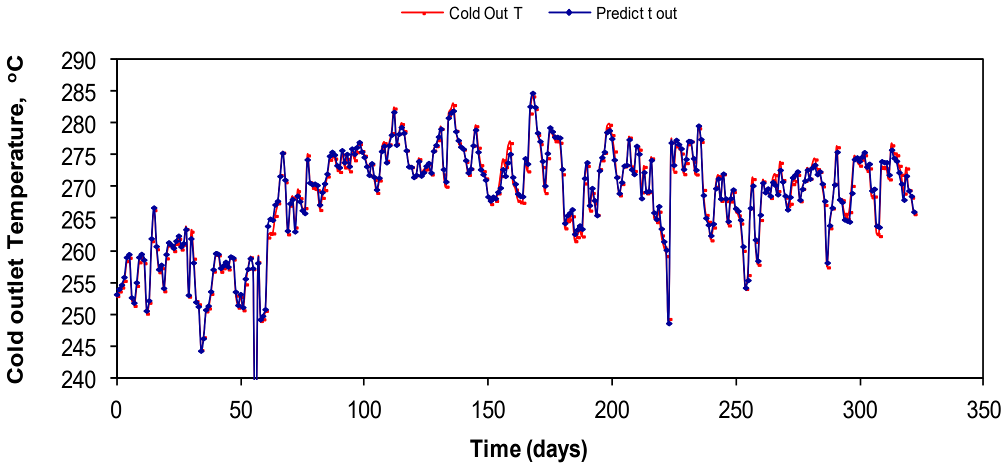


Figure 2.14: Predicted and measured temperature for Compabloc unit EX2

For the Compabloc unit EX1, the temperature predictions for the first half of the data agrees better than the second half so the model is further fitted with the second half of the data where the differences in the measured and predicted fouling resistances as shown in Figure 2.11 is now well predicted in Figure 2.15.

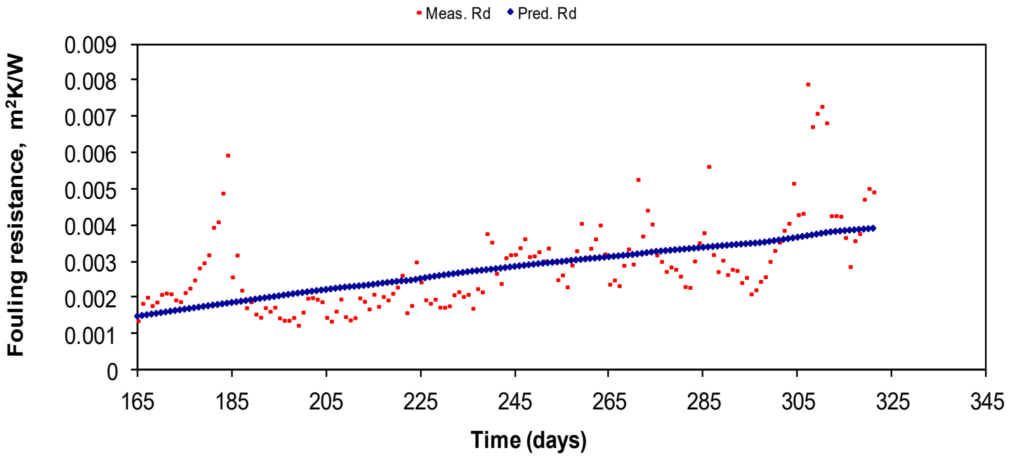


Figure 2.15: Predicted and measured fouling resistances for Compabloc unit EX1 for the second half of the data

The temperatures for this second half of the data for the exchanger EX1 is also very well predicted as illustrated in Figure 2.16 (comparing that to Figure 2.13).

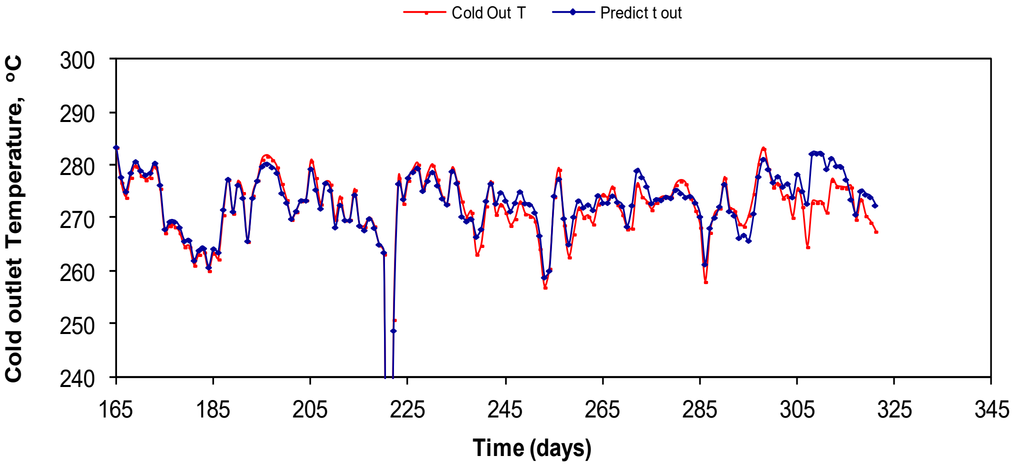


Figure 2.16: Predicted and measured temperature for Compabloc unit EX1 for the second half of the data

CHAPTER 3

FOULING AND THE THERMO-HYDRAULIC SIMULATION IN THE OPERATION OF THE CRUDE OIL PROCESSING PRE-HEAT TRAIN (PHT)

3.0. STRUCTURE OF THE PRE-HEAT TRAIN

Refineries are a major part of the economy of many nations. Their presence and ability to exploit and process crude oil is seen as a boost for the economy. Mexico alone has six refineries across the country with plans underway to build a seventh one whiles the UK has nine major refineries [46]. Ghana on the other hand has one refinery. The pre-heat train is an integral part of the crude oil processing plants. The heart of heat recovery and integration is located at this part of the processing plant and the slightest problem in its management results in losses amounting to millions of US dollars. The pre-heat train is made up of a network of heat exchangers transferring heat between the products leaving the distillation column and the crude oil on its way to the distillation column, matching up the least hot products with the coldest part of the crude oil and so on until the fired heater. It contributes to more than about 50% of the heat recovery process in refinery plants.

A typical pre-heat train may be made up of different types of heat exchangers from Compabloc units to shell and tube heat exchangers and all interacting in a systematic manner for heat recovery. The energy recovered from the distillation products on their way to further processing or storage and the pump-around is used to heat the crude oil for separation in the distillation column. The network of heat exchangers that make up the pre-heat train can be divided into the cold end, the intermediate hot end and the hot end. The train is also integrated with a desalter, a flash drum and a fired heater which all perform various roles in conditioning the crude oil for distillation.

The cold end of the pre-heat train like the name suggest is that part of the train that is coldest (below an average temperature of 130 ºC) and is mostly the network of heat exchangers at the beginning of the train upstream to the desalter. This part of the pre-heat train heats up the crude oil to temperatures adequate for the desalting process.

The intermediate and the hot end of the pre-heat train is the network of heat exchangers downstream from the desalter but before the fired heater. This part of the pre-heat train also has the flash drum as part of it to separate the gas phase from the flashed crude which is transported directly to the distillation column whiles the rest of the crude oil goes through the heat exchangers and the fired heater. The crude oil leaving the network of heat exchangers is mostly at temperatures greater than300 ^{o}C before entering the distillation column for separation. Sometimes a fired heater is necessary to supply the rest of the energy needed to reach this temperature [31].

The heat exchangers found in the different sections of the pre-heat train suffer different fouling mechanisms in their operation. These mechanisms range from particulate and crystallization fouling to chemical reaction fouling depending on the operating conditions and location of the heat exchanger.

3.1. FOULING IN PRE-HEAT TRAINS

The performance of crude oil PHTs has a lot of impact on the operation of the distillation column and the profitability of the whole plant. Fouling has been a major problem in the operation of the PHTs causing billions of losses in US dollars to the crude oil processing industry. Moreover it is a complex problem whose cause and mitigation continue to be a major topic in the design of heat exchangers and the revamp of the pre-heat trains. The major effects of fouling in crude pre-heat trains are both hydraulic and thermal. They include:

- The blockage of tubes as a result of fouling causing mal distribution of flow in the heat exchangers and leads to loss of capacity and in most severe cases the removal of the heat exchanger and subsequent shutdown of the process line.
- The presence of deposits in the heat exchangers increases the frictional resistances whiles reducing the flow area. This increases the pressure drop across the heat exchanger and the network thus the operational cost of the process increases.
- The presence of foulant on the heat transfer surfaces increases the resistance of the heat exchanger to heat transfer, reducing their performance with time.

- Apart from the fact that fouling is location dependent, it is also seen to be time dependent. With age, the chemistry of the deposits on the heat transfer surfaces change and pose greater problems in heat transfer and maintenance of the heat exchangers.
- The temperature profiles of heat exchangers change as a result of fouling and these changes prevent the proper recuperation of energy causing a ripple of low performance of heat exchangers in the network.
- Fouling also causes the mal functioning of major units in the pre-heat train like the desalter. Since this operates at fixed temperature the effect on changes in the temperature profile of the train affects its performance.

Fouling at the cold end of the PHT is mostly related to particulate and crystallization fouling as a result of the presence of inorganic salts, wax and particles in the flowing crude oil [46]. These precursors do not mostly end up at the hot end of the pre-heat train because of the functionality of the desalter. The desalter process helps to remove any other inorganic precursor that might not have formed deposits in the heat exchangers upstream to it. Unlike the deposits at the hot end of the pre-heat train that are complex and hard to remove, these deposits are easily identified and removed by methods outlined in Wiehe [5]. Several models have been developed for the prediction and measuring of inorganic fouling at the cold end of the pre-heat train. These fouling models have been based on a competing mechanism between the deposition process and the removing process.

At the hot end of the pre-heat train, formation of deposits is mostly by a chemical reaction from the precursors in the crude oil identified as asphaltene macromolecules. These reactions resulting in the deposits compared to the inorganic precursors are more complex and may involve different reaction parameters. These reactions are basically made up of a set of decomposition reactions involving the asphaltene molecules which if not attended to will result in the formation of coke [25], a substance more difficult to remove and very resistant to the transfer of heat. The fouling model mostly used for the analysis and prediction of chemical reaction fouling include the model developed by Ebert and Panchal [26]. Improvements have been made to these models and a lot more other models have been proposed for the analysis of chemical reaction fouling [23, 60]. Some models have been described in *Chapter 2* of this work.

3.2. THERMO-HYDRAULIC SIMULATION OF THE PHT

The performance of a crude oil pre-heat train is often different from its initial design targets. These are mainly due to unforeseen changes that occur during the operation of the pre-heat train like fluctuations in operational parameters and the use of different combinations of crude oil. Sometimes these unforeseen changes are as a result of incomplete instrumentation or inability to access certain areas of the pre-heat train leading to the lack of measured data for assessment. These uncertainties need to be tackled with the constant monitor of parameters like flow and temperature in the operation of the pre-heat train. The operational behavior of the crude oil pre-heat train is closely linked to market demands and importantly to fouling in the heat exchangers that make up the train. The inability of the engineer to identify and assess these fluctuations will lead to future consequences in operation.

The identification of how an existing pre-heat train structure can be modified in order to improve its energy efficiency can be undertaken easily and systematically. It is necessary to also identify the economics of saving in the process and a thermo-hydraulic simulation is a great methodology for this. The thermo hydraulic simulation of crude oil processing PHT involves the understanding of the thermal efficiency and the distribution of the bulk of the flow in the heat exchangers giving information on the performance of a network and at the same time presenting avenues for improvement.

In recent years, a thermo-hydraulic analysis of a network of heat exchangers has been the basis for both retrofits and revamp actions. Understanding the results from a thermo-hydraulic simulation presents ideas on the management of the performance of the network involved and for any changes that need to be undertaken. Models for the prediction of fouling in heat exchangers can be integrated into simulation tools for the design and revamp of pre-heat trains. These models can help give predictions on both the hydraulic behavior of the heat exchangers in the train and the thermal effects of fouling on their performance. The idea of heat recovery in a pre-heat train is a search for ways of making efficient use of heat whiles minimizing the changes including piping that need to be done to the network.

There are various ways by which heat can be recovered for use in a pre-heat train but a more practical way is the use of path analysis. The concept of transferring heat along paths has been suggested and further investigations have been carried out by Adiarso [39] and Van Reisen [40]. Asante and Zhu [61] also outlined a mathematical approach in retrofit analysis for the efficient use of energy. Path analysis identifies how heat duty can be moved along the path between heat exchangers in the same network for heat recovery. It is necessary for the determination of how much energy can be saved in the network. In some cases a detailed analysis is required to identify areas where heat is being transferred across the pinch point so that changes may be made to avoid this phenomenon.

Thermo-hydraulic simulation of a pre-heat train through the transfer of heat along paths between heat exchangers in the network needs to be coupled with the evaluation of various options for improving the heat integration and increase the performance of the heat exchangers involved. These options may include the use of heat transfer enhancement tools like tube inserts, replacement of a tube bundle for one with higher efficiency or the investment of a new heat exchanger with a larger heat transfer area. A complete replacement of a heat exchanger in a network is preferred to the installation of a second heat exchanger in parallel to an existing one in the network. Apart from the additional cost in piping and structural changes for the parallel installation, both heat exchangers in parallel may be subjected to severe fouling.

In a path analysis, there are three requisites for thermal integration of heat exchangers in a heat recovery network:

1. The availability of a heat source that can be recovered: Suitable sources are points in the network where heat is being rejected to the environment like coolers, points where it is used to provide utility that may be better generated elsewhere like steam raisers.
2. The presence of a heater currently providing heat to the network: Like a fired heater in a crude oil processing pre-heat train.
3. A “path” between the heater and the heat source.

3.3. A CASE STUDY: HEAT RECOVERY IN A PRE-HEAT TRAIN

A typical structure of a pre-heat train is presented below in Figure 3.0. The residue stream leaving the distillation column have the highest supply temperature and may have enough heat duty to heat up the incoming crude oil at different sections of the pre-heat train. First the residue stream is an important source of heat for crude oil at the hottest end of the pre-heat train (just before the fired heater) and the remaining heat duty also available to heat up the crude oil at the cold end of the pre-heat train (before the desalter). As depicted in the Figure 3.0, there exist a heat source below the desalter in the form of a cooler, C2, linked to heat exchanger E1 and a heater represented by the fired heater and located after heat exchanger E3.

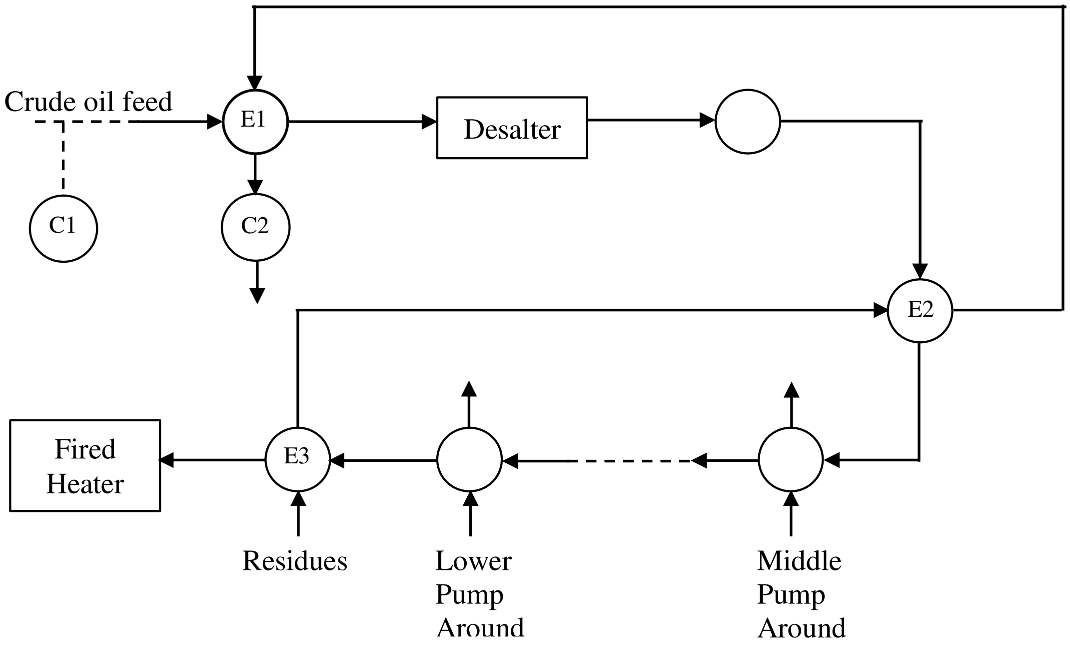


Figure 3.0: A typical pre-heat train structure.
The circles denote the heat exchangers with the letter “C” representing coolers.

The residue stream in this case provides a path between the heat source and the heater as shown in Figure 3.1. The major advantage of such a path is that it bypasses a lot of the heat exchangers in the network that implies that not a lot of heat exchangers will require modification in the process of heat recovery.

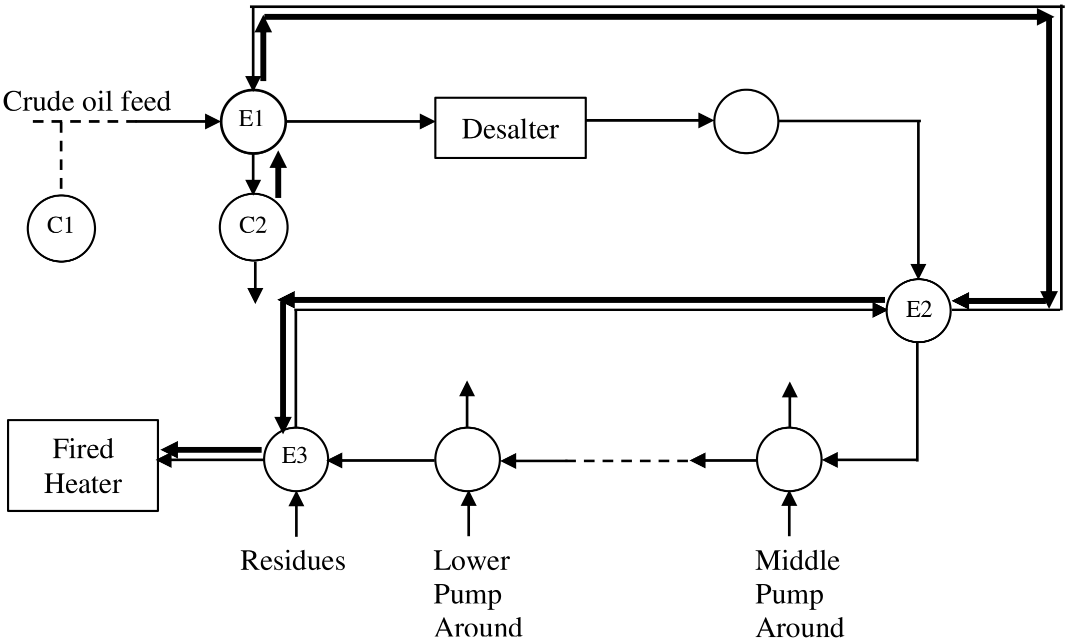


Figure 3.1: A representation of the path for heat recovery.

For the path depicted in Figure 3.1, if the heat recovery is increased in heat exchanger E3 by increasing the heat duty of the exchanger, it will lead to a reduction in the supply temperature of the residue stream to heat exchanger E2 downstream of the desalter. This will subsequently lower the performance of the heat exchanger E1 immediately before the desalter. A change at one end of the path feeds on unto the last heat exchanger involved in the recovery process and to maintain the heat duty or requirements of the other heat exchangers a retrofit strategy would need to be undertaken.

An effective retrofit action will involve the following:

1. Increase heat extraction from the residue stream at the hot end of the pre-heat train in exchanger E3.
2. Restore the performance of the heat exchanger E2 midway in the pre-heat train so as to reduce disturbances in heat exchangers placed downstream from this heat exchanger. These heat exchangers downstream from this point are mostly fed with pump around streams and as little disturbance as possible is necessary in these heat exchangers.
3. Improve heat recovery below the desalter. For example the desalter inlet temperature would have to be maintained. This can be achieved by either improving the performance of the heat exchanger immediately below the desalter or modify another heat exchanger below the desalter to compensate for the performance losses.

These retrofit actions are presented in Figure 3.2 below.

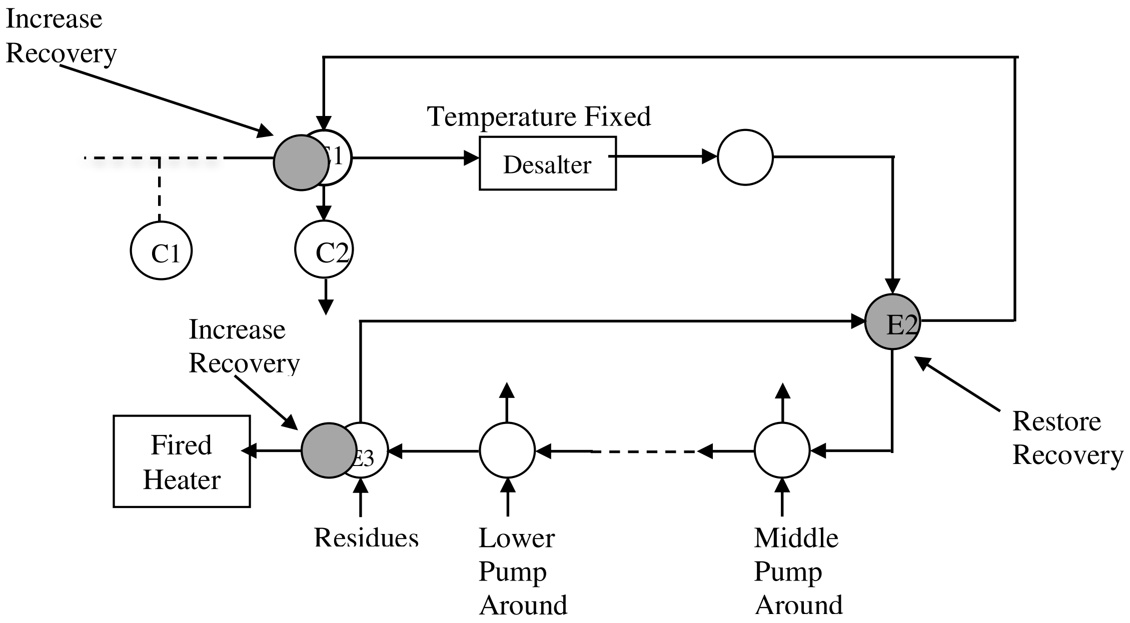


Figure 3.2: Retrofit proposals for the pre-heat train depicted in Figure 3.0

3.3.1. Features to be observed

Applying a thermo-hydraulic simulation to a pre-heat train through a path analysis presents three important features that need to be observed if the best heat recovery decisions would have to be taken. These features include:

- The need for bridges
- The presence of blocks in the pre-heat trains
- Process operations with fixed heat loads

3.3.1.1. Need for bridges

Bridges are paths between heat exchangers on the retrofit paths and located at different ends of an identified barrier. Process operations are an integral part of the performance of a network structure and are difficult to separate during a simulation analysis. For example, the operation of the pre-flash column at the hot end of a pre-heat train may need to operate within a temperature range and positioned such that crude oil flashing does not occur in the heat exchangers. The same is true for the desalter. In the case of Figure 3.0, improvements in heat recovery in the heat exchanger below the desalter will put a constraint on the performance of the desalter increasing the crude oil supply temperature to its upper limit. Thus heat will have to be transferred along the residue path from heat exchanger E1 to heat exchanger E2. For this case a “bridge” will be required.

3.3.1.2. Presence of blocks

The presence of blocks within the pre-heat train may be due to restrictions on the structure of the network or fouling considerations from the modification of heat exchangers in a retrofit process. Consider the structure of the network in Figure 3.3 below:

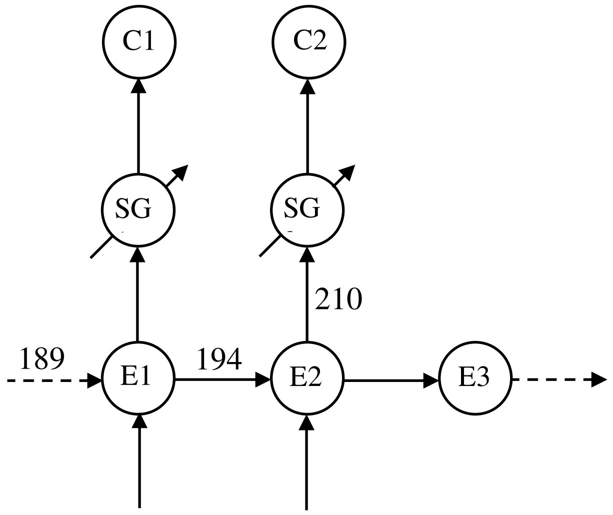


Figure 3.3: Presentation of two potential heat recovery opportunities.
SG – steam generator, C – coolers and E – Heat exchangers.
Horizontal and vertical lines represent cold and hot streams respectively.

Opportunities for heat recovery include improving the performance of heat exchanger E1 by transferring heat from cooler C1 or steam generator SG1 to the cold stream for the heat exchanger. In a likewise manner, heat can be transferred from C2 or SG2 to cold stream in E2. Examination of the local temperatures indicates that the temperature at which the cold stream leaves E1 (194 ºC) is close to that at which the hot stream leaves heat exchanger E2 (210 ºC). The temperature cannot be raised significantly before there is a decline in the heat recovery from E2: E2 is therefore providing a ‘block’ on the path of C1.

The block can be removed by splitting the streams to E1 and E2 as shown in Figure 3.4 preventing heat exchanger E2 from serving as a block on the heat recovery path. The temperature at which the cold stream enters E2 is reduced and revamping or replacing the existing heat exchanger can also improve its performance. The revamping process of heat exchanger E2 needs to be validated with a fouling analysis to prevent severe fouling both in the already existing heat exchanger and the revamped heat exchanger. Another block is encountered from a fouling analysis in terms of the film temperature in the heat exchanger and this must be avoided. This is because high film temperatures correspond to high fouling rates. These fouling blocks can only be removed by an analysis of plant data

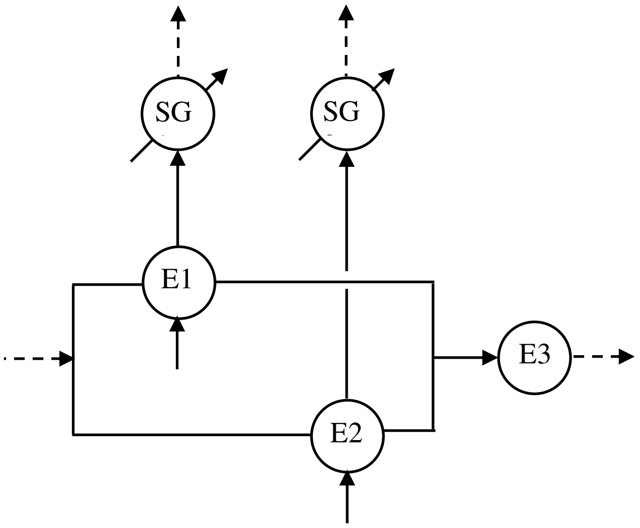


Figure 3.4: Removal of block on the heat recovery path from Figure 3.3.

3.3.1.3. Fixed heat loads

Some process streams entering the pre-heat train need as little disturbance as possible and care needs to be taken during heat recovery actions involving these process streams. This is because these heat exchangers perform under a fixed heat load and involve streams such as pump-around streams from the distillation column. Like the example shown in Figure 3.0, paths in the recovery process that bypass these heat exchangers (heat exchangers between E3 and E2) are preferred. In cases where these heat exchangers cannot be bypassed, a heat exchanger simulation program would be needed to analyze the performance of these heat exchangers and find best ways to restore their heat loads.

3.4. HEAT EXCHANGER TECHNOLOGIES AVAILABLE FOR FOULING MITIGATION

Fouling has been a major industrial problem over the years and the mitigation of fouling by the manipulation of the process parameters like velocity and temperature, the main parameters that affect formation of deposits has been the subject of many investigations [23, 24, 29, 62]. Fouling has thus resulted to be one of the major considerations in the design and operation of heat exchangers and pre-heat trains processing crude oil with the sole aim of reducing the formation of deposits in the heat exchangers.

The development of heat exchanger technologies have moved alongside research in the development of fouling models for fouling analysis. These technologies identify ways of controlling the thermo-hydraulic behavior of heat exchangers and the network as a whole. It also looks at favoring heat transfer on the crude oil side of the heat exchanger. These heat exchanger technologies work on reducing the wall temperature and increasing the shear stress of the crude oil in the heat exchanger. Polley et al [63] have identified three ways where these exchanger technologies are being applied: helical tubes, twisted tubes and tube inserts. These and other technologies are examined.

3.4.1. Tube inserts

Tube inserts are tools that are used to enhance flow by promoting turbulence in the heat exchanger tubes and thereby enhancing the heat transfer rate. These tube inserts have been used over years and have been accepted as valuable tools for enhancing heat transfer and mitigation of fouling. Different tube inserts are available for use and under various commercial names. The range include: wire mesh (hiTRAN), rotating coils (TURBOTAL), vibrating springs (SPIRELF) [48], wire coil (FIXOTAL) [64], twisted tapes and helical coils.

3.4.1.1. hiTRAN

This is a wire mesh mainly designed to disrupt the laminar fluid layer on the tube surface and increase fluid shear on the surface of the plain tubes. With these conditions established there is an enhancement in heat transfer on the tube side. Petitjean et al [65] published the effects of the use of hiTRAN inserts in a laboratory study in the UK concluding that while heat transfer is enhanced at low Reynolds numbers, the effect of fouling was not fully suppressed. This can be explained by the nature of the deposits that are formed. Should fouling not be totally suppressed the loops will become embedded in the deposits, accelerating fouling at the same time making it difficult to remove the mesh.

HiTRAN inserts can be used to provide a cost effective revamp for heat recovery situations between vacuum residues and crude oil match placed prior to the desalter in the pre-heat train. The traditional approach for placement of the residue is on the shell side of the heat exchanger. Switching the vacuum residue which has relatively high viscosity to the tube side and installing a wire mesh insert will not only increase significantly the heat transfer from the vacuum residue to the crude oil on the shell side but it will also help reduce fouling in the tubes. Petitjean et al [65] presented a case study to support this enhancement of heat transfer when the crude oil is switched to the shell side and a wire mesh is used in the tubes.

3.4.1.2. Turbotal

Turbotal inserts consists of a rotating wire coil whose action is to suppress fouling in tubes by the rotating action of the coils as a result of the fluid flow. Initially as the coil rotates, it causes high turbulence increasing shear on the tube surface and thereby enhancing the heat transfer in the tubes. Later as fouling occurs, the rotating coil also mitigates fouling by removing fresh deposits from the fouling layer and with this action prevent further growth of the deposit [48, 64]. This will lead to an asymptotic fouling level in the analysis of the heat exchangers fitted with Turbotal inserts. These tube inserts operate over a certain velocity range for a more effective result in heat transfer and fouling mitigation.

3.4.1.3. Spirelf

Spirelf inserts consists of springs fitted at both ends of the tube that can move forward and backwards by the action of the stream flow causing vibrations both axially and radially [64]. This action also results in generating more shear stress in the flowing stream resulting in the reduction of fouling and the enhancement of heat transfer. Like the Turbotal inserts, the Spirelf inserts also results in the maintenance of a fouling deposit level in the tubes of heat exchangers but unlike the Turbotal operates over a wider velocity range [48].

3.4.2. Design specifications

Traditional design of shell and tube heat exchangers has been the use of plain tubes and straight baffle plates to direct the fluid flow around the tube bundle. These designs have helped the crude oil industry but with an increased need in the conservation of energy and protection of the environment there has been the need to re-examine the structure of the heat exchangers as a means of making them more energy efficient. Some of these structural changes are the use of helical baffles and twisted tubes in the design of shell and tube heat exchangers.

Helical baffles have the advantage of increasing the turbulence in the shell of the shell and tube heat exchangers. These baffles are placed at an angle to the tube bundle axis and arranged in a manner such that the flow through the shell side forms a helical pattern. This helical pattern of flow helps to increase the thermal effectiveness of the heat exchanger while at the same time making good use of the heat exchanger area and thus reducing the heat exchanger size. These helical baffles have an advantage over segmental baffles as they have very low shell diameters (0.3m or less) and large baffle angles [48].

Twisted tubes have also been used to enhance heat transfer and reduce fouling in the crude oil processing industry [63].

3.4.3. Compact heat exchangers

Shell and tube heat exchangers used in the refinery industry have been mostly described as a big limitation on plant floor space. Compact heat exchangers on the other hand not only occupy just a fraction of these spaces but also improve on the effectiveness of these heat exchangers. An example of a compact heat exchanger that is gaining a lot of use in the crude oil industry is the Compabloc unit.

The Compabloc units are welded plate heat exchangers made from high resistant alloys. The flow within the unit is cross flow and high shear rates have been registered in their applications leading to significant heat transfer rates as well. The units can be set up with multiple pass arrangements. They are designed such that there are improvements on the heat transfer on both the cold and hot fluid. In terms of operation, care must be taken to increase and maintain high fluid shear stress under all conditions to prevent severe fouling, making the consideration of fouling an integral part in its design.

3.4.4. Heat exchanger cleaning

The first of the mitigation measures considered is the cleaning of heat exchangers after they foul[32]. Mechanical and/or chemical cleaning mechanisms have been the most applied in the removal of deposits from the surface of tubes in the shell and tube heat exchangers used in the crude oil industry. Industry experiences have observed that the rate of formation of deposits on an existing deposit or a cleaned surface is greater than when a new heat exchanger is applied. This makes the cleaning mechanism applied very important on the subsequent performance of the heat exchanger.

The best option for the removal of deposits from the tube side of a heat exchanger is hydro-drilling [48] and for the cleaning of heat exchanger shells and Compabloc units, chemical cleaning enhanced with ultrasound stands out as best option [66].

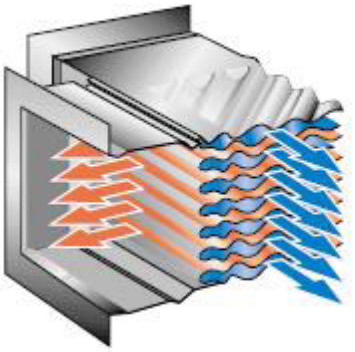
CHAPTER 4

ADVANCED HEAT EXCHANGER TECHNOLOGY– THE COMPABLOC UNIT

4.0 THE COMPABLOC UNIT

A Compabloc unit is a welded plate heat exchanger that is made up of a stack of corrugated heat transfer plates made from materials that are able to withstand aggressive temperature and pressure conditions. These plates are welded alternately in a manner such that channels are formed for the passage of the streams, thus two streams exchange heat whiles passing through alternate channels.

These heat exchangers can be fabricated in a number of configurations. The greater the number of passes used the higher the velocity within the exchanger and the greater the film heat transfer coefficient. The elements of a Compabloc unit include a pack of plates, heads (Upper and lower), gasket, panels, support, girder and nozzles (inlet and outlet of streams). The flow field within the individual channels is complex. Individual passes are in “cross-flow”, Figure 4.0. A typical construction is shown in Figure 4.1. In this example each of the streams flows across the stack four times. So, the unit has a 4-4 configuration.



The two media flow in cross-flow
in alternately welded channels

Figure 4.0: The flow configuration in a Compabloc unit (Courtesy Alfa Laval)

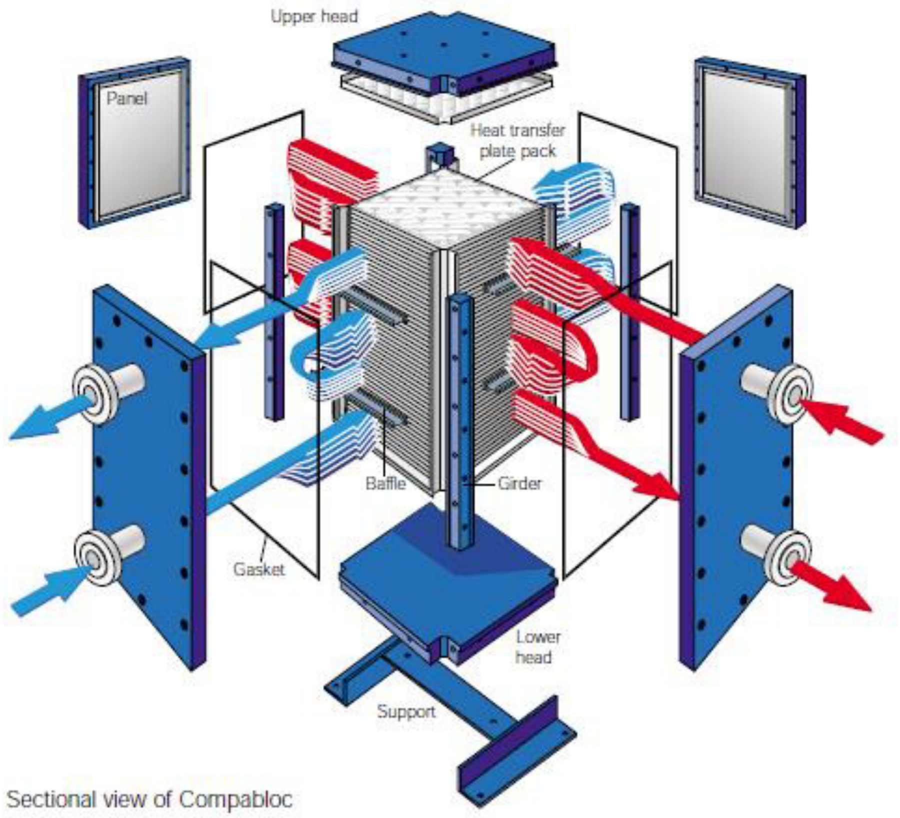


Figure 4.1: The parts of a welded plate Compabloc unit (Courtesy Alfa Laval)

As the pairs of passes increases the overall unit makes a closer approach to a pure "counter-current flow" arrangement. Therefore, the pass arrangement can have a strong influence on the performance of the unit. Several arrangements are possible. Permitted pass arrangements are those in which the ratio of passes is integer, integer plus one half and integer plus one third.

4.1. CONSTRUCTION OF A MULTI-PASS UNIT

A stream is said to have made a pass through a heat exchanger if it flows through one full length of the flow path in the heat exchanger. A multi pass unit is the configuration of a heat exchanger such that one or both of the streams flowing through the exchanger has to do so a number of times over the full or part length of the exchanger before finally exiting the heat exchanger. This number of turns or passes associated with the presence of a turn or a baffle in the path of the stream defines the configuration or flow arrangement and is related to its efficiency of the heat exchanger.

The construction of a multi pass heat exchanger takes into consideration the terminal temperatures of the fluids entering the exchanger. The most efficient transfer of heat is when the outlet of the cold stream is located in the same zone or end as the inlet of the hot stream and likewise the exit of the hot stream matched with the inlet of the cold stream. This allows the exchanger to have an overall hot end (the inlet of the hot stream and outlet of the cold stream zone) and an overall cold end (the outlet of the hot stream and the inlet of the cold stream zone). This flow arrangement presents the two fluids flowing parallel to each other but in the opposite direction and is classified as a counter-current flow. On the contrary, a parallel flow is when both streams flow in the same direction along the length of the heat exchanger. A change in the direction of the flow of one stream or creation of a pass in a heat exchanger as a result of the baffle or the turn in its path together with the other stream present in the exchanger may be isolated out of the heat exchanger configuration and classified as an independent counter-current flow. Therefore a series of counter-current flows or arrays of minute heat exchangers can be treated independently but integrated for their analysis.

4.1.1. The Multi-pass Algorithm

A multi-pass algorithm is necessary for the determination of the temperature profiles within a Compabloc unit. This distribution permits both the analysis of the performance and further prediction of the development and the distribution of fouling along the length of these units. The performance of the Compabloc unit is strongly influenced by the pass arrangements of the streams flowing within the unit. These pass arrangements pave way for the division of the Compabloc unit into sections for its analysis taking into consideration the changes in the properties along the length of the unit.

4.1.2. Effectiveness-NTU Method

The effectiveness (ε) of a heat exchanger is defined as the actual rate of heat transfer in a given heat exchanger to the maximum possible rate of heat transfer. This is dependent on the configuration of the exchanger, the minimum heat transfer rate, number of transfer units and the ratio of the heat capacity flow rates.

The Number of Transfer Units,N_{TU}, for a given heat exchanger is defined by the ratio of the overall thermal conductance to the smaller of the heat capacity rates (Equation 4.0) [67]. It is given in Equation 4.1:

$$CP = mass\ flowrate\ \times\ heat\ capacity \tag{4.0}$$

$$N_{TU} = \frac{UA}{CP_{min}} \tag{4.1}$$

So, the first step in any simulation is the determination of which of the two streams has the lower heat capacity flow-rate. This leads immediately to the calculation of the ratio of the heat capacity flow-rates:

$$R = \frac{CP_{min}}{CP_{other}} \tag{4.2}$$

The effectiveness of the exchanger is then calculated. The flow pattern in a Compabloc unit is complex and can be equated to a cross flow with both streams fully mixed. The equation for the effectiveness applied in the analysis is defined by [14].

$$\varepsilon = \frac{1}{\left[\frac{R\text{ø}^{R}}{\text{ø}^{R}-1} + \frac{\text{ø}}{\text{ø}-1} - \frac{1}{\ln \text{ø}}\right]} \tag{4.3}$$

Where$\text{ø} = Exp(N_{TU})$

This allows the temperature responses to be calculated for any type of heat exchanger. For the stream having the lower heat capacity flow-rate the temperature response will be given by Equation 4.4 whiles Equation 4.5 gives that for the stream having the higher heat capacity flow rate.

$$\delta T = \varepsilon(T_{in} - t_{in}) \tag{4.4}$$

$$\delta T = \varepsilon R(T_{in} - t_{in}) \tag{4.5}$$

4.1.3. The caloric temperature approach

Unfortunately, because the streams flow through each section in a cross-flow manner some approximation has to be introduced into the calculation of the performance of a given section.

Three approaches are possible. In the first approach, heat transfer coefficients at the ends of the section are determined and a mean value assumed to apply for the section. The second approach is to calculate physical properties at the end of each section and use mean values in the calculation of the heat transfer coefficients. The third approach is to calculate the "caloric temperatures" for each of the streams flowing through the section and use physical properties calculated at these temperatures in the determination of the heat transfer coefficients.

The determination of caloric temperatures is described by Kern [68]. They are based on the work of Colburn [69] who analyzed the performance of a counter-current flow heat exchanger and concluded that there is a linear variation in the overall heat transfer coefficient over its length. The approach could be extended to multi-pass heat exchangers without large loss in accuracy.

The caloric temperature approach has been used in this work.

4.2. GENERAL ALGORITHM FOR DETERMINING THE PERFORMANCE OF THE COMPABLOC UNIT

4.2.1. Preliminary calculations

1. Set the parameters for physical property equations from specified reference properties at reference temperatures. The following equations apply to liquids:

   Density ρ

   $$\rho = a + b * t \tag{4.6}$$

   Viscosity μ

   $$\mu = c * exp\left(\frac{d}{t}\right) \tag{4.7}$$

   Heat capacity C_p

   $$C_p = e + f * t \tag{4.8}$$

   Thermal conductivity k

   $$k = g + h * t \tag{4.9}$$

Where a, b, c, d, e, f, g and h are the parameter coefficients obtained from the physical properties at two reference temperatures and t is the temperature at which the physical properties are required. Liquid viscosity exhibits an exponential relationship whiles the other properties exhibit a linear relationship with regard to temperature.

2. Determine which stream has the larger number of passes and set the maximum and the minimum pass counts. Determine the pass ratio (maximum flow pass to minimum flow pass) in order to check that that arrangement is acceptable and identify the exchanger model to be used.
3. Determine the geometry of the heat exchanger. The geometry is dependent on the number of channels available for each of the flow passes and how these channels are distributed for the streams in the different sections. It involves:
   a. Determine the total number of channels for the unit
   b. Determine the channels per stream
   c. Determine the overall heat transfer area of the unit
4. Determine the mass fluxes within the unit.

4.2.2. Determination of the overall performance for the Compabloc unit

The true performance of a unit involves taking careful account of the changes to the physical properties across the unit. These changes are not known at the outset of the analysis, therefore a two-stage procedure is used.

The first stage is to determine an approximate temperature profile for the unit based upon fixed physical properties. The second stage is to use this temperature profile as a starting point for analysis that involves consideration of the changes in the physical properties across the unit. This analysis uses an iterative procedure in order to determine the actual temperature profile within the unit.

4.2.2.1. Stage 1: Determine the approximate temperature profile of the unit

At the outset we only know the inlet temperatures of the two streams to the unit. Approximate values of the outlet temperatures are determined in the following manner.

5. Determine the physical properties of the streams at the inlet temperatures from Equations 4.6 to 4.9.
6. Calculate the Reynolds, Prandtl numbers and the heat transfer coefficients for each stream at the inlet temperatures of the unit.

7. Set the initial overall fouling resistances of the unit.
8. Determine the overall heat transfer coefficient using the film heat transfer coefficients determined for the inlet conditions of the unit. Assume this value apply across all the different sections of the unit.
9. Set the stream having the maximum number of passes as reference for the calculations. This helps to maintain symmetry in the calculations irrespective of whether the cold or the hot stream has the highest number of passes.

With the physical properties and the overall heat transfer coefficient calculated the objective would be to apply the ε-NTU method to determine the approximate values of the inlet and outlet temperatures of the two streams for the sections. The following procedure is then adopted.

10. Undertake an analysis of the structure (Section 4.3) of the Compabloc unit with the aim of determining the combinations of the mass flow rates of both streams and the heat transfer area for each section within the structure (Section 4.4).
11. Determine the heat capacity flow rate ratios of the streams and the effectiveness for each section. This involves:
    a. Determine the heat transfer area available for the section
    b. Determine the mass flow rate of the streams through the section and finally the heat capacity flow rate for both streams and the ratio of the minimum heat capacity flow rate to the other heat capacity flow rate
    c. Determine the number of transfer units for the section.
    d. Determine the effectiveness (ε) of the section from Equation 4.3.
    e. Repeat *steps a* to *e* for all the sections

The above calculations give estimates of the effectiveness for each section within the model. The temperature profile across the model is now determined using these values.

12. For a particular section, setting the known temperatures at the inlets as $t_{in}$and T_{in}, the unknown temperatures at the outlets T_{out} and t_{out} can be calculated from the δTexpressions defined in Equations 4.4 and 4.5.

$$t_{out} = t_{in} + \delta T \tag{4.10}$$

$$T_{out} = T_{in} - \delta T \tag{4.11}$$

13. Apply Equations 4.10 and 4.11 above to all the sections to determine the outlet temperatures from the respective inlet temperatures, heat capacity ratios and effectiveness. In cases where an inlet temperature is not known then
    a. Set a temperature T_{new} as the initial inlet temperature to the section being analyzed for the stream with the minimum number of stream passes.
    b. Determine the outlet temperature T_{out} for that section using the effectiveness and the ratio of the heat capacity rates as calculated for the section.
    c. Determine the outlet temperatures for the rest of the sections. The outlet temperature(s) of a section would define the inlet temperatures to another section.
    d. Compare the calculated temperatures with initial set values reassign the set inlet temperature with the calculated.
    e. Loop through *steps a* through *d* until the difference between the set temperatures T_{new} and T_{out} is less than 0.001.

4.2.2.2. Stage 2: Determine the real temperature profile of the unit

At this stage, we have an approximate prediction of the temperature profile and this is based on constant properties. So the following iterative procedure is applied to calculate the “real” temperature profile of the unit.

14. Set the individual fouling resistances for all the sections of the unit.
15. Choose the first section (within the identified structure).
16. With the approximate temperatures calculated for the section, recalculate the actual physical properties for the inlet conditions to the section.
17. Determine the Reynolds and Prandtl numbers and the local heat transfer coefficients at the inlet conditions for the section.
18. Re-determine the overall heat transfer coefficient for the section U_o.
19. Repeat steps *11* to *13* above to determine the exit temperatures for the each section. This is the new temperature profile of the Compabloc unit based on the property changes at each section.
20. Re-determine the physical properties of the streams at the calculated inlet and outlet temperatures for the section for both streams.

21. Recalculate the dimensionless properties (Reynolds and Prandtl numbers) and the local heat transfer coefficients at both ends of the section for the inlet and outlet conditions.
22. Determine the overall heat transfer coefficients at both ends of the section from the film heat transfer coefficients calculated at the inlet and outlet conditions of the streams.
23. If the difference between the calculated inlet overall heat transfer coefficient and the outlet overall heat transfer coefficient is less than 0.1 then the overall heat transfer coefficient is equal to the arithmetic mean else calculate the mean overall heat transfer coefficient for the section from the Colburn methodology [68, 69] of "caloric temperature". It includes:
    a. Set a "caloric condition".
    b. Determine the cold stream and the hot stream temperatures at this condition.
    c. Calculate the physical properties of the streams and determine the local heat transfer coefficients at this condition.
    d. Determine the overall heat transfer coefficient U_{new} at this caloric condition.
24. Compare the calculated overall heat transfer coefficient U_{new} and U_o . If $Abs(U_o - U_{new}) < 0.001$ then move to the next step. Else repeat steps *14* to *24* reassigning $U_o = U_{new}$ in step *18* until the given criterion above is met.
25. Assign the overall outlet temperatures of the Compabloc unit.
26. Determine the physical properties of the streams at the overall inlet and outlet conditions to the Compabloc unit.
27. Determine the film heat transfer coefficients at the inlet and outlet conditions to the Compabloc unit.
28. Determine the overall heat transfer coefficient at both ends of the unit.
29. If the difference between the overall heat transfer coefficient at both ends of the unit is < 0.1 then determine the arithmetic mean else determine the mean overall heat transfer coefficient of the unit as follows (from the Colburn methodology above):
    a. Set a "caloric condition"
    b. Determine the cold stream and the hot stream temperatures at this caloric condition.
    c. Calculate the physical properties of the streams and the local heat transfer coefficients at this caloric condition.
    d. Determine the overall heat transfer coefficient. This is the mean overall heat transfer coefficient for the Compabloc unit.
30. Determine the log mean temperature difference across the unit.

31. Determine the heat transferred across the unit from $Q = UA\Delta T_{\mathrm{LMTD}}$

4.3. COMPABLOC UNIT STRUCTURES

The different pass arrangements can be classified into a number of similar units in series for the performance analysis. These similar units in series are defined as bases or the basic models in this investigation. For any configuration of a Compabloc unit there are *n* bases or basic models. In other words the base is that structural part of the Compabloc unit that keeps recurring.

A base of a Compabloc unit can be further divided into sections for ease in the performance analysis. These sections can be "isolated" alongside their respective bases in the analysis. The total analysis of a base will mean the individual analysis of the sections of the base in question. Consider a Compabloc unit with the configuration 2-6, this is made up of two*1-3* base models (2 bases) and three sections in each base. This configuration thus has 2 bases and 6 sections in total. Take Figure 4.2 below for example, if we want to determine the outlet temperature of the streams leaving the first base the following methodology can be applied.

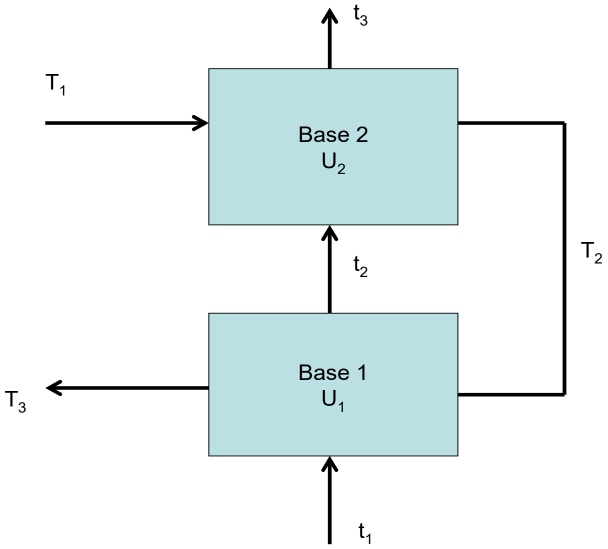


Figure 4.2: Representation of a base in the Compabloc structure (2-bases)

For the analysis of this structure the only known temperatures are T_1 and t_1. Again the other temperatures can be determined using an iterative approach. In this case the variable used in the iteration is the hot stream that leaves base 2 (T_2). Knowledge of this temperature allows calculation of the performance of base 1. This yields the cold stream inlet temperature to base

2 thereby allowing the performance of base 2 to be calculated. Temperature T_2 is then adjusted until assumed and calculated values coincide. This same analogy is applied to all the sections in the base to determine the temperature profiles.

The approach is easily extended to cover exchangers involving higher number of base configurations in series. Depending on the different pass ratio, the following conclusion can be drawn:

1. For a maximum to minimum pass ratio equal to an integer, the following structures can be obtained from a basic model:
    a. For example, a 2-2 unit is a two*1-1* unit in series and a 3-3 unit is a three*1-1* unit in series. This gives the ratio of the maximum to minimum flow passes as one. The basic model for this ratio is a *1-1* unit. Different arrangements giving an *n(1-1)* structure can be presented.
    b. For example, a 1-2 unit or a 1-3 unit falls into a category of *1-n* configuration of passes for a Compabloc unit. This has a pass ratio of *n* with only one pass for one of the streams.
    c. For example, a 2-6 unit is a two*1-3* unit in series and subsequently 2-8 unit is a two*1-4* units in series. This also has a pass ratio of *n* with the same basic model for the structure defined in step *2* above but this is a structure with the *2(1-n)* configurations of passes. One of the streams will always have two passes for this structure.
    d. For example, a 2-4 unit is a two*1-2* unit in series. Likewise a 3-6 unit is three*1-2* units in series. This arrangement of stream passes has a pass ratio of two with a basic model of a *1-2* arrangement. Different arrangement of passes can be written to satisfy the structure with an *n(1-2)* configuration of passes for the Compabloc unit.
2. For a ratio of maximum to minimum flow passes equal to an integer *n* plus one half = *n.5*,the following arrangements of flow passes can be made: a 4-6 unit represents two*2-3* units in series whiles a 6-9 unit is a three*2-3* unit in series. These structures have a basic model of a *2*-3 unit representing an *n(2-3)* configuration.
3. For a ratio of maximum to minimum flow passes equal to an integer plus one third (*n.333…*), the following pass arrangements can be obtained with a base structure of a *3-4* unit. A 9-12 unit is a three*3-4* unit in series and likewise a 6-8 unit is a two*3-4* unit in series. An *n(3-4)* combination of stream passes is represented.

In the structure analysis n can be any whole number. Table 4.0 below gives examples of the configurations and its relation to the base configuration and the sections.

Table 4.0
Relationship between Compabloc configurations, bases and sections

Configuration	Base / Basic model	Number of bases	Sections per base	Total number of sections
2-4	1-2	2	2	4
6-9	2-3	3	4	12
3-3	1-1	3	1	3
4-8	1-2	4	2	8
9-12	3-4	3	6	18

4.4. STRUCTURAL ANALYSIS

4.4.1. For a 1-2 pass Compabloc unit

Conditions:

1 – Pass stream 1 with temperature t and mass flow rate m_1

2 – Pass stream 2 with temperature T and mass flow rate m_2

Minimum pass

$$minp = 1$$

Maximum pass

$$maxp = 2$$

$$Ratio = \frac{maxp}{minp} = \frac{2}{1} = 2$$

Number sections

$$Number\ of\ sections = maxp = 2$$

The 1-2 pass unit can be represented in Figure 4.3 below:

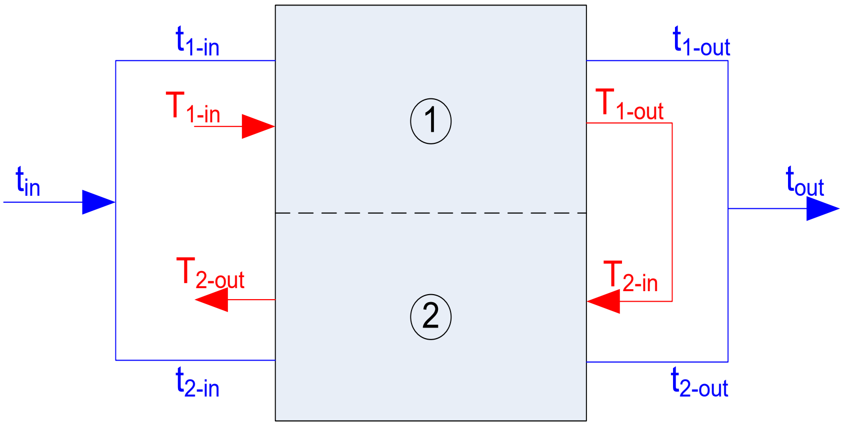


Figure 4.3: The 1-2 configuration

Breaking it down into the individual sections, we obtain the following independent heat exchangers in Figure 4.4:

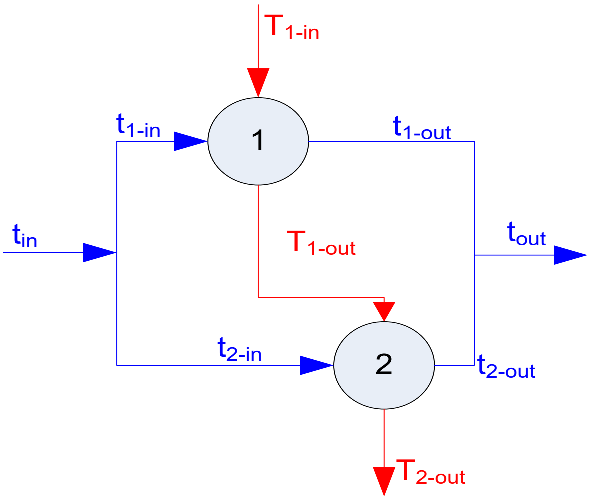


Figure 4.4: The 1-2 configuration defined by individual exchangers

Known: T_{in} and t_{in}

Unknown: T_{out} and t_{out}

Observations:

For stream 1

$$t_{in} = t_{1-in} = t_{2-in}$$

t_{1-out} and t_{2-out} are fractions of t_{out}

For stream 2

$$T_{1-in} = T_{in}$$

$$T_{1-0ut} = T_{2-in}$$

The summary of the properties of the sections is presented in the Table 4.1.

Table 4.1
Properties of a 1-2 configuration

Section	**Area** (Fraction of total heat transfer area)	**Mass flow rate** (fraction of m) **Stream 1**	 **Stream 2**
1	½	½	1
2	½	½	1

For

Section 1

1. Determine the heat transfer area for the section

$$A_1 = \frac{1}{2}A$$

2. Calculate the heat capacity flow rate from the following mass flow rates

$$m_1 = \frac{1}{2}m_1$$

$$m_2 = m_2$$

$$CP1 = m_1 h_1$$

$$CP2 = m_2 h_2$$

3. Determine the minimum heat capacity flow rate $CP_{min}$from the equations above, the ratio of the heat capacities and the effectiveness as given in *step 11* above.

$$N_{TU} = \frac{U_1 A_1}{CP_{min}}$$

$$ø = Exp(N_{TU})$$

$$\varepsilon_1 = \frac{1}{\left[\frac{Rø^R}{ø^R-1} + \frac{ø}{ø-1} - \frac{1}{\ln ø}\right]}$$

4. Calculate the outlet temperatures from the following equations

$$t_{1-out} = t_{1-in} + \varepsilon_1 \left(\frac{CP_{min}}{CP1}\right)(T_{1-in} - t_{1-in})$$

$$T_{1-out} = T_{1-in} - \varepsilon_1 \left(\frac{CP_{min}}{CP2}\right)(T_{1-in} - t_{1-in})$$

Section 2

5. Determine the heat transfer area for the section

$$A_2 = \frac{1}{2}A$$

6. Determine the heat capacity rates from the following mass flow rates

$$m_1 = \frac{1}{2}m_1$$

$$m_2 = m_2$$

$$CP1 = m_1 h_1$$

$$CP2 = m_2 h_2$$

7. Determine the minimum heat capacity flow rate $CP_{min}$from the equations above, the ratio of the heat capacities and the effectiveness as given in *step 11* above.

$$N_{TU} = \frac{U_2 A_2}{CP_{min}}$$

$$ø = Exp(N_{TU})$$

$$\varepsilon_2 = \frac{1}{\left[\frac{R\varnothing^R}{\varnothing^R-1} + \frac{\varnothing}{\varnothing-1} - \frac{1}{\ln \varnothing}\right]}$$

8. Calculate the outlet temperatures from the following equations

$$t_{2-out} = t_{2-in} + \varepsilon_2 \left(\frac{CP_{min}}{CP1}\right)(T_{2-in} - t_{2-in})$$

$$T_{2-out} = T_{2-in} - \varepsilon_2 \left(\frac{CP_{min}}{CP2}\right)(T_{2-in} - t_{2-in})$$

9. Determine the outlet temperature for both streams for the section.
    a. For stream 1 which has just one pass

$$t_{out} = \frac{1}{2} t_{1-out} + \frac{1}{2} t_{2-out}$$

    b. For stream 2

$$T_{out} = T_{2-out}$$

4.4.2. For a 3-2 pass unit

Conditions:

2 – Pass stream 1 with temperature t and mass flow rate m_1

3 – Pass stream 2 with temperature T and mass flow rate m_2

Minimum pass

$$minp = 2$$

Maximum pass

$$maxp = 3$$

$$Ratio = \frac{maxp}{minp} = \frac{3}{2} = 1.5$$

Number sections

$$Number\ of\ sections = 2 \times minp = 2 \times 2 = 4$$

The 3-2 pass unit can be represented below in Figure 4.7:

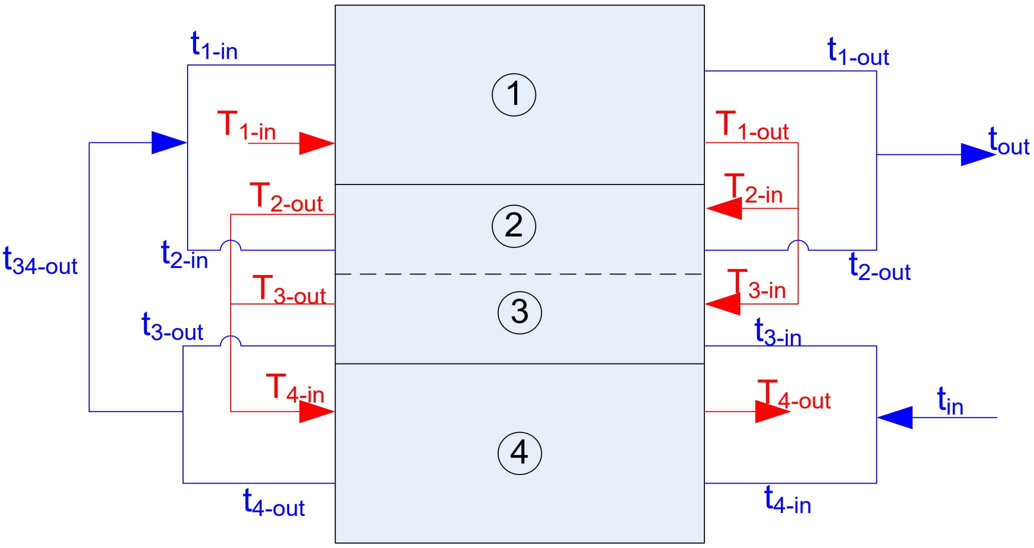


Figure 4.7: The 3-2 configuration

The individual sections represented in Figure 4.8:

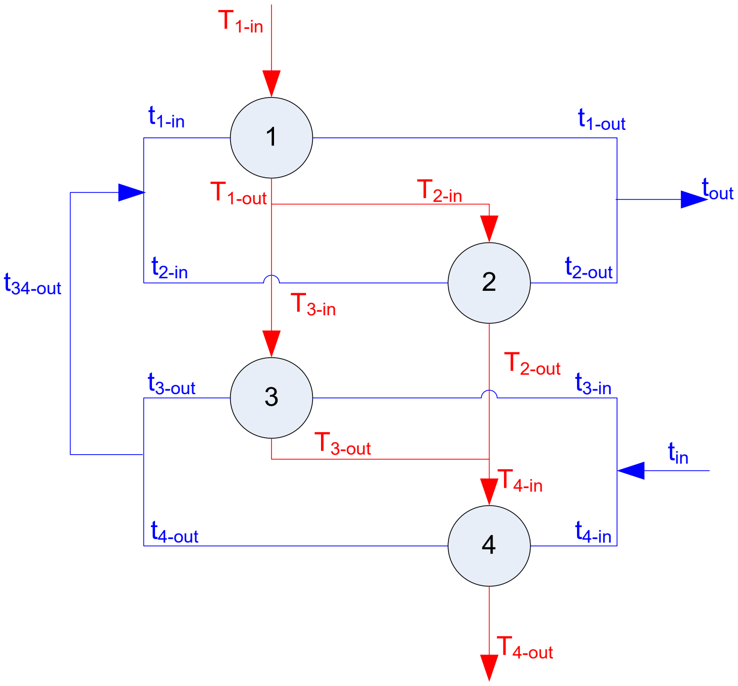


Figure 4.8: The 3-2 configuration defined by individual exchangers

Known: T_{in} and t_{in}

Unknown: T_{out} andt_{out}

Observations:

For stream 1

$$t_{in} = t_{3-in} = t_{4-in}$$

$$t_{34-out} = t_{2-in} = t_{1-in}$$

t_{3-out} and t_{4-out} are fractions of t_{34-out}

t_{1-out} and t_{2-out} are fractions of t_{out}

For stream 2

$$T_{1-in} = T_{in}$$

$$T_{4-out} = T_{out}$$

$$T_{1-0ut} = T_{2-in} = T_{3-in}$$

T_{3-out} and T_{2-out} are fractions of T_{4-in}

The summary of the properties of the sections is presented in the Table 4.3.

Table 4.3
Properties of a 3-2 configuration

Section	**Area** (Fraction of total heat transfer area)	**Mass flow** (fraction of m)	
		Stream 1	**Stream 2**
1	1/3	2/3	1
2	1/6	1/3	½
3	1/6	1/3	½
4	1/3	2/3	1

For

Section 1

1. Determine the heat transfer area for the section

$$A_1 = \frac{1}{3}A$$

2. Calculate the heat capacity flow rate from the following mass flow rates

$$m_1 = \frac{2}{3}m_1$$

$$m_2 = m_2$$

$$CP1 = m_1 h_1$$

$$CP2 = m_2 h_2$$

3. Determine the minimum heat capacity flow rate $CP_{min}$from the above equations, the ratio of the heat capacities and the effectiveness as given in *step 11* above in stage 1.

$$N_{TU} = \frac{U_1 A_1}{CP_{min}}$$

$$ø = Exp(N_{TU})$$

$$\varepsilon_1 = \frac{1}{\left[\frac{Rø^R}{ø^R - 1} + \frac{ø}{ø - 1} - \frac{1}{\ln ø}\right]}$$

4. Set the value of t_{34-out} (Observations above)
5. Calculate the outlet temperatures from the following equations

$$t_{1-out} = t_{1-in} + \varepsilon_1 \left(\frac{CP_{min}}{CP1}\right)(T_{1-in} - t_{1-in})$$

$$T_{1-out} = T_{1-in} - \varepsilon_1 \left(\frac{CP_{min}}{CP2}\right)(T_{1-in} - t_{1-in})$$

Section 2

6. Determine the heat transfer area for the section

$$A_2 = \frac{1}{6}A$$

7. Determine the heat capacity rates from the following mass flow rates

$$m_1 = \frac{1}{3}m_1$$

$$m_2 = \frac{1}{2}m_2$$

$$CP1 = m_1 h_1$$

$$CP2 = m_2 h_2$$

8. Determine the minimum heat capacity flow rate $CP_{min}$from the above equations, the ratio of the heat capacities and the effectiveness as given in *step 11* above in stage 1.

$$N_{TU}e = \frac{U_2 A_2}{CP_{min}}$$

$$ø = Exp(N_{TU})$$

$$\varepsilon_2 = \frac{1}{\left[\frac{Rø^R}{ø^R - 1} + \frac{ø}{ø - 1} - \frac{1}{\ln ø}\right]}$$

9. Calculate the outlet temperatures from the following equations

$$t_{2-out} = t_{2-in} + \varepsilon_2 \left(\frac{CP_{min}}{CP1}\right)(T_{2-in} - t_{2-in})$$

$$T_{2-out} = T_{2-in} - \varepsilon_2 \left(\frac{CP_{min}}{CP2}\right)(T_{2-in} - t_{2-in})$$

10. Calculate the outlet temperature for stream 1

$$t_{out} = \frac{2}{3} t_{1-out} + \frac{1}{3} t_{2-out}$$

Section 3

11. Determine the heat transfer area for the section

$$A_3 = A_2 = \frac{1}{6} A$$

12. Calculate the heat capacity flow rate from the following mass flow rates

$$m_1 = \frac{1}{3} m_1$$

$$m_2 = \frac{1}{2} m_2$$

$$CP1 = m_1 h_1$$

$$CP2 = m_2 h_2$$

13. Determine the minimum heat capacity flow rate $CP_{min}$from the above equations, the ratio of the heat capacities and the effectiveness as given in *step 11* above in stage 1.

$$N_{TU} = \frac{U_3 A_3}{CP_{min}}$$

$$ø = Exp(N_{TU})$$

$$\varepsilon_3 = \frac{1}{\left[\frac{Rø^R}{ø^R - 1} + \frac{ø}{ø - 1} - \frac{1}{\ln ø}\right]}$$

14. Calculate the outlet temperatures from the following equations

$$t_{3-out} = t_{3-in} + \varepsilon_3 \left(\frac{CP_{min}}{CP1}\right) (T_{3-in} - t_{3-in})$$

$$T_{3-out} = T_{3-in} - \varepsilon_3 \left(\frac{CP_{min}}{CP2}\right)(T_{3-in} - t_{3-in})$$

Section 4

15. Determine the heat transfer area for the section

$$A_4 = A_1 = \frac{1}{3}A$$

16. Determine the heat capacity rates from the following mass flow rates

$$m_1 = \frac{2}{3}m_1$$

$$m_2 = m_2$$

$$CP1 = m_1 h_1$$

$$CP2 = m_2 h_2$$

17. Determine the minimum heat capacity flow rate $CP_{min}$from the above equations, the ratio of the heat capacities and the effectiveness as given in *steps 11* above in stage 1.

$$N_{TU} = \frac{U_4 A_4}{CP_{min}}$$

$$ø = Exp(N_{TU})$$

$$\varepsilon_4 = \frac{1}{\left[\frac{Rø^R}{ø^R - 1} + \frac{ø}{ø - 1} - \frac{1}{\ln ø}\right]}$$

18. Determine the mixed temperature of stream 2 T_{4-in} that serves as the inlet temperature for section 4

$$T_{4-in} = \frac{1}{2}T_{2-out} + \frac{1}{2}T_{3-out}$$

19. Calculate the outlet temperatures from the following equations

$$t_{4-out} = t_{4-in} + \varepsilon_4 \left(\frac{CP_{min}}{CP1}\right)(T_{4-in} - t_{4-in})$$

$$T_{4-out} = T_{4-in} - \varepsilon_4 \left(\frac{CP_{min}}{CP2}\right)(T_{4-in} - t_{4-in})$$

20. The outlet temperature of stream 2 is

$$T_{out} = T_{4-out}$$

21. Calculate the mixed temperature t_{34-out} for stream 1 from the outlet temperatures of sections 3 and 4 as

$$t_{34-out} = \frac{1}{3} t_{3-out} + \frac{2}{3} t_{4-out}$$

22. Compare the calculated value of t_{34-out} with the value set in *step4* under section 1; if the absolute difference between these temperatures is less than 0.001 then end iterations else reassign the set temperature in *step4* with the calculated temperature in *step21*.
23. Repeat the calculations for the outlet temperatures using *steps1* through *steps22* until the set criterion above is met.

4.4.3. For a 2-4 Compabloc unit

Conditions:

2 – Pass stream 1 with temperature t and mass flow rate m_1
4 – Pass stream 2 with temperature T and mass flow rate m_2

Minimum pass

$$minp = 2$$

Maximum pass

$$maxp = 4$$

$$Ratio = \frac{maxp}{minp} = \frac{4}{2} = 2$$

Number sections

$$Number\ of\ sections = maxp = 4$$

The 2-4 pass unit can be represented below in Figure 4.9 and individual sections in Figure 4.10:

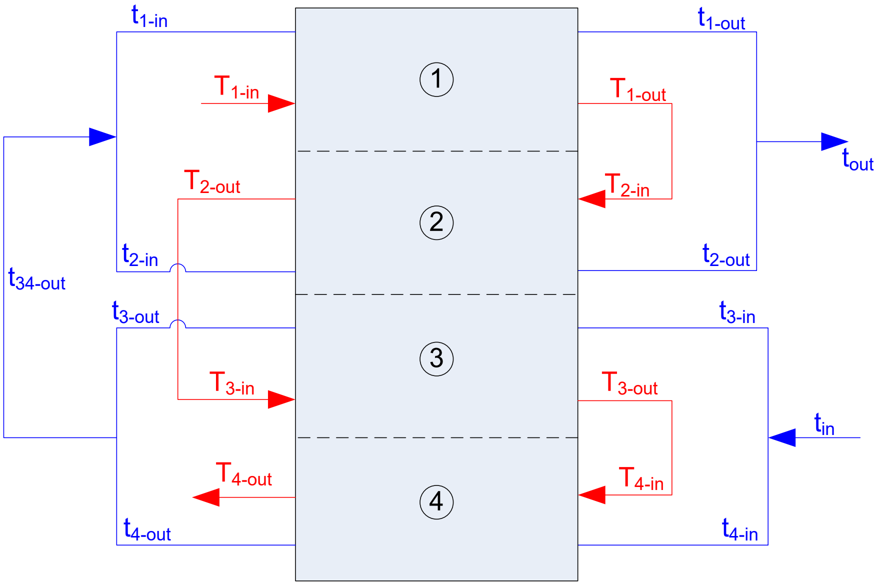


Figure 4.9: The 2-4 configuration

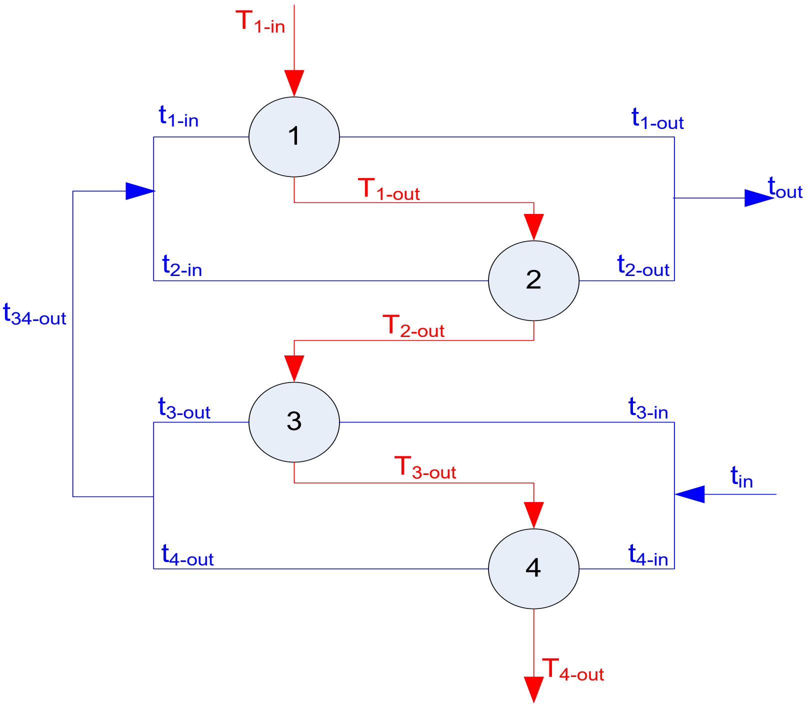


Figure 4.10: The 2-4 configuration defined by individual exchangers

Known: T_{in} and t_{in}

Unknown: T_{out} and t_{out}

Observations:

For stream 1

$$t_{in} = t_{3-in} = t_{4-in}$$

t_{3-out} and t_{4-out} are fractions of t_{34-out}

$$t_{34-out} = t_{2-in} = t_{1-in}$$

t_{1-out} and t_{2-out} are fractions of t_{out}

For stream 2

$$T_{1-in} = T_{in}$$

$$T_{4-out} = T_{out}$$

$$T_{1-0ut} = T_{2-in}$$

$$T_{2-0ut} = T_{3-in}$$

$$T_{3-0ut} = T_{4-in}$$

The summary of the properties of the sections is presented in the Table 4.4 below.

Table 4.4
Properties of a 2-4 configuration

Section	**Area** (Fraction of total heat transfer area)	**Mass flow** (fraction of m)	
		Stream 1	**Stream 2**
1	¼	½	1
2	¼	½	1
3	¼	½	1
4	¼	½	1

This is a 2 *1*-2 base configuration. Base 1 is made up of individual exchangers *1* and *2* and represented in Figure 4.11 and Base 2 in Figure 4.12 made up of sections *3* and *4*.

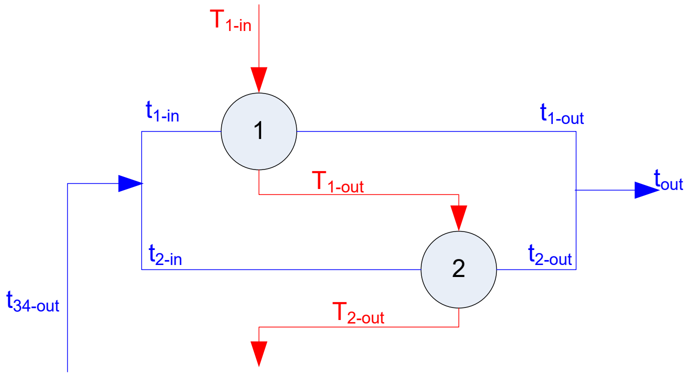


Figure 4.11: Base 1 of the 2-4 configuration

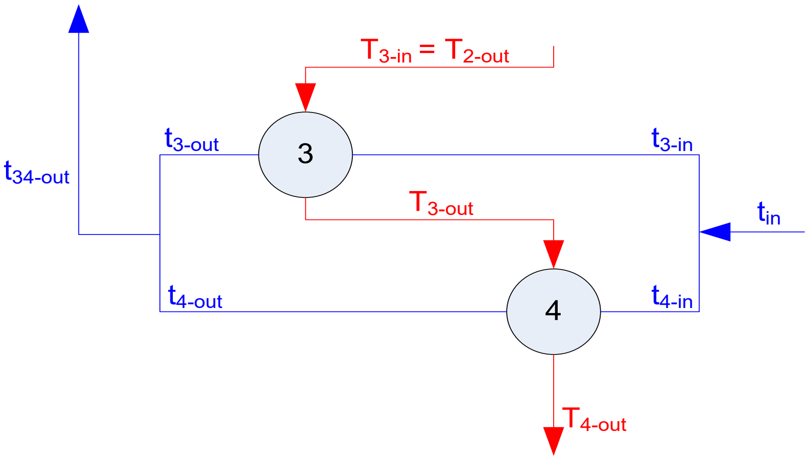


Figure 4.12: Base 2 of the 2-4 configuration

A look at the individual heat exchanger structure and observations of the inlet and outlet temperatures show that individual exchangers *1* and *2* can be separated from individual exchangers *3* and *4*. These two separations both have structures of a *1-2* unit making the *2-4* unit a two*1-2* unit in series.

The exchangers presented above can be analyzed individually in the same way as the *1-2* unit above.

4.5. FOULING ANALYSIS IN COMPABLOC UNITS

The performance analysis of heat exchangers will not be complete without the integration of a fouling model. Fouling in Compabloc units can be severe if the wrong operating practices are used in their operation because of the high wall temperatures and heat transfer coefficients that they exhibit. The wall shear stresses calculated in Compabloc units can be very high as compared to values obtained in the operation of shell and tube heat exchangers, an advantage in minimizing fouling but a good model will be necessary in the fouling analysis.

The model developed in *Chapter 2* of this work, the Asphaltene Precipitation model is also applied in the fouling analysis of Compabloc units and have provided very good results as can be seen from the case studies presented in *Chapter 2*.

The following methodology is used to incorporate the fouling model into the performance analysis of the Compabloc units. The use of the fouling model includes the determination of the fouling factors over given operational periods. The calculated resistances will then be used to determine the fouling resistance for the next operation period. In most analysis an operating period of twenty-four hours is chosen. For application in the performance analysis of the Compabloc unit, the fouling model is applied after the steps 1 to 31 detailed above in section 4.2to be able to predict the fouling resistance for the period.

For the fouling analysis, first determine the fouling rate (refer to model in *Chapter 2*) at the inlet and the outlet of the Compabloc unit. The fouling rate can also be applied to each section of the Compabloc unit to study the distribution of fouling across the sections. The following steps are then followed to determine the fouling resistances for the chosen operating period:

a. Determine the film temperature of the stream from the wall and bulk temperatures.

$$T_f = 0.55T_{wall} + 0.45T_{bulk} \qquad 4.12$$

b. Calculate the fouling rate

$$\frac{dR}{dt} = \frac{A}{\alpha} exp^{\left(\frac{-E}{R_g T_f}\right)} \times P \qquad 4.13$$

If the shear stress $\tau_w > 100$ then$P = 0$

Else if $\tau_w > 2$ then$P = 1 - \left(\frac{\tau_w - 2}{98}\right)^{0.5}$ 4.14

Else$P = 1$

The hydraulic diameter is recalculated based on the new fouling resistances and this serves as initial data for the performance analysis of the next operating period. Steps 1 to 31 (Section 4.2) as described above are then repeated.

PART 1: CONCLUSIONS

Fouling is a key industrial problem especially in the crude oil processing industry. A thermo-hydraulic simulation of the pre-heat train is the first step in the identification and resolution of problems that are associated with fouling. Two decades down the lane from the development of the threshold-fouling model, a lot more work is being done to come up with a more efficient and cost effective way to combat fouling in heat exchangers.

The availability of a fouling model like the ones for chemical reaction fouling will help simulate heat exchangers based on the experience of their operations to obtain the fouling factors for the designs and also make available relevant and more target driven solutions for the fouling problems. The Asphaltene Precipitation model developed in this work is directed to that effect. The importance of considering and integration of a fouling model in the design process of a heat exchanger is encouraged since a good geometry chosen only on the basis of a clean condition design might not be the best decision for operation.

The thermo-hydraulic simulation of pre-heat trains will require the availability of plant data and this is always not the case. Plant data if available are either inconsistent or contain so much uncertainties that their use in simulation analysis is questionable. But the path analysis opens the way for a more systematic and practical approach to heat recovery.

It is recommended that a series of simulations be undertaken when a revamp or retrofit exercise becomes necessary in a PHT. These simulations should take step changes based on temperature fields and heat saving actions in the heaters to achieve the required heat recovery. Several heat exchanger analysis software are available for use in conjunction with the process described above. This software can help to develop the design options necessary to achieve the required heat recovery.

PART 2

THE COOLING WATER NETWORK

CHAPTER 5

FOULING IN COOLING WATER NETWORKS

5.0 CRYSTALLIZATION FOULING

Crystallization fouling is the formation of deposits of salts, mostly called scales, on the heat transfer surface as a result of crystallization or the precipitation of the dissolved salts in the bulk of the flow [11, 17, 70]. These salts mainly include sulfates, phosphates and carbonates of calcium or magnesium with the most common being calcium sulfate and calcium carbonate [11, 17]. The major cause of crystallization fouling is associated with the super saturation level of these salts in the bulk of the water as a result of the diminishing effect of their solubility with temperature. The depositions on the heat transfer surface may be tenacious substances difficult to remove called scales or soft deposits termed sludge [11, 42].

The factors that affect crystallization fouling include:

- Surface temperature – This is the temperature of the layer between the heat transfer surface and the bulk of the flow where fouling is likely to form. Its influence depends on the type of fouling; whiles in biological fouling an increase in temperature results in the decrease in fouling the opposite is true for crystallization fouling.
- Bulk temperature – the temperature of the bulk fluid also contribute to the deposition of salts in heat exchangers.
- Water velocity – Generally an increase in the velocity for heat transfer equipment results in an increase in the heat transfer coefficient and also in the decrease in fouling. But specifying a high velocity for a heat exchanger will also result in unacceptable high pressures in their operation.
- Solid particles – The presence of solid particles may aid in the formation of crystals of salts or a combining effect of particulate fouling with crystallization fouling.
- pH – pH tends to affect the crystallization of the dissolved salts. For instance Hewitt and Müller-Steinhagen [11] explains that the deposition of calcium carbonate in coolers can be controlled by making the cooling water slightly acidic. Hofling et al [17] also emphasis the effect of pH on deposition for a multicomponent system of calcium carbonate and calcium sulfate.

A measure of the tendency of scaling or crystallization in cooling water systems is dictated by the solubility index; Ryznar stability index and the Langelier Saturation index [11, 17]. Different methodologies have been developed for both sulfate and carbonate salts of dissolved in water. These calcium sulfates have high solubility values compared with calcium carbonate solutions at the same temperature and the most difficult to tackle in the cooling water industry.

In the crude oil processing industry, crystallization fouling is suggested to occur upstream of the desalter because of prevailing conditions and in the desalter where the majority of the salt crystallize out from the crude before it gets into the heat exchangers at the hot end of the pre-heat train. The presence of salt deposits downstream of the desalter questions the efficiency of the desalter unit and increases the effect of corrosion in the heat exchangers downstream of the desalter.

Fresh water employed in cooling processes contain dissolved salts that precipitate in heat exchangers as a result of their solubility levels and the temperature of operation of the heat exchanger. For a salt to crystallize out of solution it means that it should be supersaturated in the solution in which it is found. The solubility of the reverse solubility salts like calcium sulfate decreases with an increase in temperature. Therefore at elevated temperatures of the operation of a particular heat exchanger, the concentration of the dissolved salts reaches its super saturation point and crystallizes out of the solution forming deposits on the heat transfer surfaces in the process. These deposits may be soft or hard depending on the type of crystal deposits that are formed on the surface [11, 70]. An overlapping of particulate fouling and crystallization fouling may result in deposits that are softer and thus easy to remove.

Depending on the temperature of operation of the heat exchangers different forms of these aforementioned salts in their stable and unstable forms may be present. For example in the case of calcium carbonate deposits, the most stable form at moderate temperatures is calcite with the other forms being vaterite and aragonite [11, 70]. For calcium sulfate deposits, gypsum, the dehydrate form is the most common type of deposit occurring in heat exchangers whiles the other forms include calcium hemihydrate and calcium sulfate anhydride [11]. In depth review of calcium sulfate and calcium carbonate crystallization fouling in the use of fresh water for cooling has been done by Hewitt and Müller-Steinhagen [11].

5.1. MODELLING OF CRYSTALLIZATION FOULING IN HEAT EXCHANGERS

The primary cause of fouling in coolers is super saturation: the concentration of the dissolved salts in the bulk of the solution at a given temperature. Following the analogy of Kern and Seaton [33] and Najibi [15], the rate of deposition of precipitated salts on the heat transfer surface can be modeled from two consecutive steps:

- First, the mass transport of the ions in solution to the heat transfer surface by convection defined by the mass transfer coefficient and the concentration gradients in the solution and the heat transfer surface, and
- Secondly, the reaction and precipitation of the ions on the heat transfer surface to form the deposits. This is characterized by an Arrhenius relationship.

Najibi [15]focused on the steps given above and the Bohnet and Krause model cited in Najibi [15]and came up with the rate of deposition defined by the bulk concentration of the salt c_b, the saturated concentrationc^* of the salt on the hot surface, the mass transfer coefficient β and the reaction rate constant k_R. This is also presented in Hewitt and Müller-Steinhagen [11]. Equation 5.0 represents the deposition term for the prediction of crystallization fouling.

$$\dot{m}_d = \beta\left[\frac{1}{2}\left(\frac{\beta}{k_R}\right) + (c_b - c^*) - \sqrt{\frac{1}{4}\left(\frac{\beta}{k_R}\right)^2 + \left(\frac{\beta}{k_R}\right)(c_b - c^*)}\right] \qquad 5.0$$

The Arrhenius relationship for the reaction rate constant is characterized by the surface temperature of the deposit and given by

$$k_R = k_{R1} exp^{\frac{-E}{RT_s}} \qquad 5.1$$

He further considered that the rate of removal is negligible and defines the dependency of the rate of fouling on the deposition rate $\dot{m}_d$, the thermal conductivity $\lambda_d$and density ρ_d of the foulant.

$$\frac{dR}{dt} = \frac{\dot{m}_d}{\lambda_d \rho_d} \tag{5.2}$$

The mass transfer coefficient β in Equation 5.0 is calculated from the Sherwood number defined in Equation 5.3,using the Dittus–Boetler equation (Equation 5.4) and the mass and heat transfer analogy from the Colburn equation [71] (Equation 5.5).

$$Sh = \frac{\beta d}{\mathfrak{D}} \tag{5.3}$$

$$Nu = 0.0023 Re^{0.8} Pr^{1/3} \tag{5.4}$$

$$\frac{Nu}{Sh} = \left(\frac{Pr}{Sc}\right)^{1/3} \tag{5.5}$$

The Schmidt number *Sc,* is related to the diffusion coefficient for the depositing species which is defined by a temperature dependence relationship given in Equation 5.6. Knowing the diffusion coefficient at one temperature, the diffusion coefficient can be calculated for the surface temperature for the deposition process.

$$\mathfrak{D}_{T_2} = \mathfrak{D}_{T_1}\left(\frac{T_2}{T_1}\right) exp\left[3.8T^*\left(\frac{1}{T_1} - \frac{1}{T_2}\right)^2\right] \tag{5.6}$$

Given T^* as the boiling temperature of the cooling medium. Information for the physical properties for calcium sulfate is reproduced in this work [11].

Table 5.0
Physical properties and constants for the $CaSO_4$ deposition

k_{R1}, m^4/kgs	E, J/kmol	ρ_d, kg/m^3	λ_d, W/mK	$\mathfrak{D}$(For Ca^{++} and SO_4^{-2} at 355.5K), m^2/s
5.7 x 10^{11}	112500	2160	2.2	1.0633 x 10^{-9}

Equation 5.2 will be applied in the determination of the fouling rates in Compabloc units and shell and tube heat exchangers.

5.1.1. Shell and tube heat exchangers

Shell and tube heat exchangers are mostly incorporated in cooling water networks and they have seen so many design specifications and modifications to suit the needs of the process industry. Even though they do not present the greatest heat transfer coefficients, there are so many mitigating strategies to counter fouling, which applaud for its varied application. For the cooling water network, it is not a novel case and the analysis of fouling in these industries is presented with subsequent considerations for the design and revamp of a cooling water network (*Chapter 6)*.

Picón Núñez et al [20] have presented a shortcut method for the design of heat exchanger networks for the cooling water industry where the analysis is based on the K values for heat exchangers. The K values are resistant coefficients of valves and fittings which are related to their type or size. The pressures losses due to fiction in networks can be related to the k value and this pressure drop across a pipe is directly proportional to the square of the velocity of the flow and can be represented by the equation

$$\Delta P = k_i v^2 \tag{5.7}$$

$k_i$values are available in literature for valves, pipes and other accessories in the design of cooling water networks [72].

Research works from the 90s until recently have developed a relationship between the pressure drop and the volumetric flow rates applicable to valves, tubes and heat exchangers [73] and the relationships between pressure drop and heat transfer coefficients on both the tube side and shell side for shell and tube heat exchangers [74, 75]. This implies that the pressure drop across a tube, a valve, an accessory or heat exchanger can be represented by

$$\Delta P = KV^2 \tag{5.8}$$

Jegede and Polley [74] have shown that pressure drop consideration in a heat exchanger can be related to the heat transfer coefficient α, the surface area of the exchanger A, and the volumetric throughput V. For turbulent flow through a shell and tube heat exchanger, this can be given by

$$\Delta P = \frac{J}{V} A\alpha^{3.5} \qquad 5.9$$

$J$is a function of the physical properties of the liquid flowing through the tubes and the inside and outside tube diameters. Note that the tube diameter is a non-process parameter and is generally fixed in the early stages of the design of heat exchangers. Relating Equations 5.8 and Equation 5.9 above, the following can be derived

$$\alpha = \left[\frac{KV^3}{JA}\right]^{0.286} \qquad 5.10$$

Assuming typical values for the heat transfer coefficient and the fouling factors the following steps can then be taken for sizing heat exchangers based on a standard length.

1. Determine the thermo-physical properties of both streams.
2. Assume a value for the water side heat transfer coefficient α_w
3. Determine the heat capacity flow rate of the water.
4. Determine the log mean temperature difference across the exchanger.
5. Determine the clean overall heat transfer coefficient from

$$\frac{1}{U} = \frac{1}{\alpha_o} + \frac{1}{\alpha_w}\left(\frac{d_o}{d_i}\right) + R_w \qquad 5.11$$

6. Calculate the heat transfer area from

$$A = \frac{Q}{U\Delta T_{LMTD}}$$

7. Recalculate the waterside heat transfer coefficient α from Equation 5.10 above.
8. Calculate the difference between the heat transfer coefficients α calculated with the assumed α_w. Reassign $\alpha_w = \alpha$ until and repeat steps 3 to 8 above until the difference is less than 1.
9. Determine the Reynolds and Prandtl numbers, the mass flux and the velocity of water through the exchanger.

10. With the velocity and volumetric flow, determine the number of tubes for the heat exchanger and the length of the tubes.
11. If the length calculated above is less than the standard length above then increase the assumed K value of the exchanger and repeat the steps 2-10 above until the desired length is obtained else decrease the K value and repeat steps 2-10 above to obtain the standard length.

Thus, values of heat transfer coefficients, K values or other design parameters used in the design of heat exchangers can be incorporated in the analysis and prediction of fouling rates for each design.

With known values of the inlet and outlet temperatures of the streams to the heat exchanger, the surface temperature can be approximated by

$$T_w = \frac{U}{\alpha}(sT_b - T_b) + T_b \qquad 5.12$$

where

$$T_b = \frac{(t_i + t_o)}{2}$$

$$sT_b = \frac{(T_i + T_o)}{2}$$

To calculate the fouling rates, we will consider first the Dittus – Boetler equation (Equation 5.4) for the determination of the heat transfer coefficient during the process of cooling given by [14]

$$Nu = 0.023Re^{0.8}Pr^{1/3}$$

$$\alpha = 0.023\left(\frac{\rho d}{\mu}\right)^{0.8}(Pr)^{\frac{1}{3}}\left(\frac{k}{d}\right)v^{0.8}$$

$$\alpha = k_5 v^{0.8} \qquad 5.13$$

$$k_5 = 0.023\left(\frac{\rho d}{\mu}\right)^{0.8}(Pr)^{1/3}\left(\frac{k}{d}\right)$$

Using heat and mass transfer analogy presented earlier in Equation 5.5, the mass transfer coefficient for the application of shell and tube heat exchangers for cooling process is given by

$$Sh = \frac{\beta d}{\mathfrak{D}} = 0.023Re^{0.8}Sc^{\frac{1}{3}}$$

$$\beta = 0.023\left(\frac{\rho d}{\mu}\right)^{0.8}\left(\frac{1}{d}\right)Sc^{\frac{1}{3}}\mathfrak{D}v^{0.8}$$

$$\beta = k_6\mathfrak{D}v^{0.8} \qquad 5.14$$

$$k_6 = 0.023\left(\frac{\rho d}{\mu}\right)^{0.8}\left(\frac{1}{d}\right)Sc^{\frac{1}{3}}$$

Divide Equation 5.14 by Equation 5.13

$$\beta = k_7\alpha\mathfrak{D}^{2/3} \qquad 5.15$$

$$k_7 = \left(\frac{k}{\rho C_p}\right)^{\frac{1}{3}}\left(\frac{1}{k}\right)$$

A case study can be set up to verify how changes in the heat transfer coefficient affect the wall temperature and eventually the fouling rate. This can also be related to the changes in the K value for a given heat exchanger for early design cases.

In a similar manner, a threshold fouling case can be considered from Equation 5.2 by equating to zero.

$$\frac{dR_f}{dt} = \frac{\dot{m}_d}{\lambda_d \rho_d} = 0 \qquad 5.16$$

$$\frac{\beta}{\lambda_d \rho_d}\left[\frac{1}{2}\left(\frac{\beta}{k_R}\right) + (c_b - c^*) - \sqrt{\frac{1}{4}\left(\frac{\beta}{k_R}\right)^2 + \left(\frac{\beta}{k_R}\right)(c_b - c^*)}\right] = 0$$

From Equation 5.16 above the fouling rate is equal to zero when parameters related to heat transfer coefficient or the diffusion coefficient are equal to zero or when the saturation concentration is equal to the bulk concentration of the flowing water. In addition, the diffusion coefficient and the saturation concentration are all dependent on the surface temperature of the deposits while the heat transfer coefficient is strongly dependent on the velocity of the flowing stream. Thus it can be concluded that the fouling rate is a function of the bulk temperature and the velocity of the flow.

$$R_f = f(T, v) \qquad 5.17$$

A range of bulk temperatures with varying flow velocities will be a great tool in the design process of heat exchangers for cooling processes.

A design case study

Consider a design process where a 2MW shell and tube heat exchanger is available for cooling a process stream from 95°C to 70°C with an assumed heat transfer coefficient of 800 W/m^{2}K for the process stream. The physical properties of the process stream and water are presented in Table 5.1.

Table 5.1
Physical properties of the streams for the cooling process

Physical property	Process stream	Cooling water
Density, kg/m^3	1000	1000
Viscosity, Pa.s	1x10^{-3}	1x10^{-3}
Thermal conductivity, W/mK	0.7	0.7
Heat capacity, kJ/kg.K	4.2	4.2

Considering a calcium sulfate concentration of 2000ppm dissolved in the flow, an algorithm was developed for the analysis of the effect of the bulk temperature and velocity of the water on the fouling rate in the heat exchanger. This algorithm is based on Equation 5.17 developed for the analysis of crystallization fouling in shell and tube heat exchangers. The result of this analysis is presented in Figure 5.0.

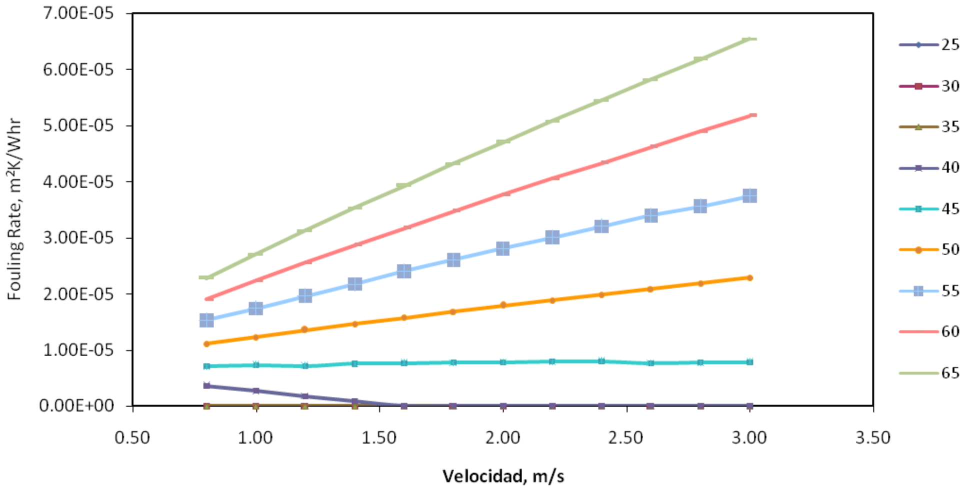


Figure 5.0: The effect of bulk water temperature and velocity on the fouling rate

This result is a great tool in the selection of a heat exchanger for a particular duty in the process industry. In the case depicted in Figure 5.0, it is deduced that at a bulk water temperature of 45ºC, the fouling rate is almost independent of the flow velocity whiles below this temperature increasing the velocity at a fixed temperature reduced the fouling rate in the heat exchanger. At high bulk temperature (above45ºC), increasing the velocity at a fixed temperature has no effect on the fouling rate indicating that the temperature dominates the fouling process. The results for moderate water bulk temperature for the case above is further presented in the graph below.

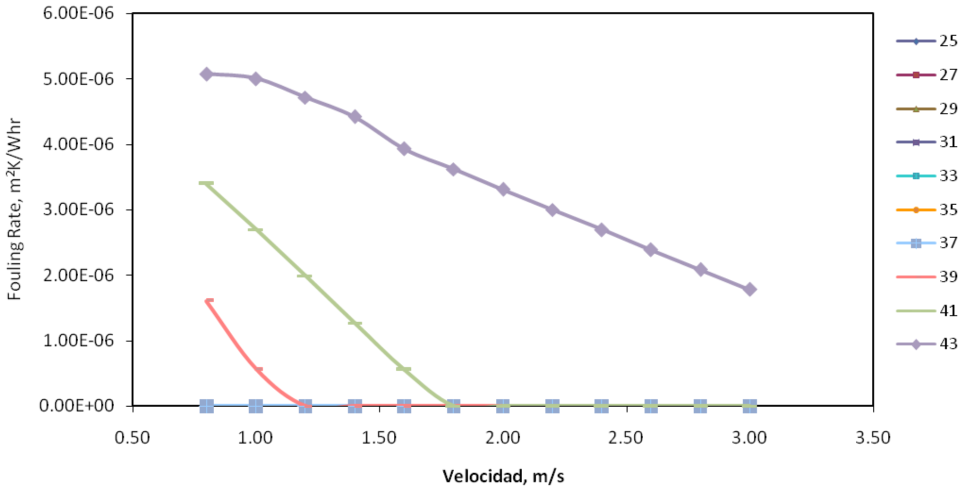


Figure 5.1: The effect of moderate bulk water temperature and velocity on the fouling rate

The fouling resistance at1000 hours of operation is presented below in Table 5.2 for a temperature and velocity range of 20 – 65 ºC and 0.8 – 3.0 m/s respectively.

Table5.2
Fouling factors at1000 hours of operation of the heat exchanger

Velocity, m/s	**Water Bulk Temperature, ºC**								
	25	*30*	*35*	*40*	*45*	*50*	*55*	*60*	*65*
0.8	0	0	0	0.0036	0.0071	0.0111	0.0153	0.0190	0.0228
1.0	0	0	0	0.0027	0.0073	0.0123	0.0174	0.0224	0.0271
1.2	0	0	0	0.0018	0.0071	0.0135	0.0196	0.0256	0.0313
1.4	0	0	0	0.0009	0.0075	0.0146	0.0217	0.0287	0.0354
1.6	0	0	0	0	0.0076	0.0158	0.0240	0.0317	0.0393
1.8	0	0	0	0	0.0077	0.0168	0.0261	0.0347	0.0432
2.0	0	0	0	0	0.0078	0.0179	0.0281	0.0377	0.0471
2.2	0	0	0	0	0.0079	0.0188	0.0301	0.0405	0.0508
2.4	0	0	0	0	0.0080	0.0198	0.0320	0.0434	0.0545
2.6	0	0	0	0	0.0077	0.0209	0.0339	0.0462	0.0582
2.8	0	0	0	0	0.0077	0.0219	0.0356	0.0490	0.0619
3.0	0	0	0	0	0.0078	0.0228	0.0375	0.0517	0.0655

The data clearly indicates the effect of operating the heat exchangers with a water inlet temperature greater than 40ºC for the case under investigation.

5.1.2. Compabloc heat exchangers

The analysis for the use of Compabloc exchangers in cooling water networks is made taking into consideration the algorithm developed in chapter 4 of this work. This algorithm should be integrated with the fouling deposition model for calcium sulfate deposits explained in the preceding section. Applying the heat transfer – mass transfer analogy from Equation 5.4, the mass transfer coefficient for a Compabloc unit will be given by

$$\beta = 0.023Re^{x}Pr^{y}Sc^{1/3}\left(\frac{\mathfrak{D}}{d}\right) \qquad 5.18$$

x and *y* depend on the Reynolds numbers encountered in the Compabloc unit.

Using Equation 5.18 the mass transfer coefficient and hence the fouling rate can be determined.

CHAPTER 6

CONSIDERATION OF FOULING IN THE DESIGN AND MODIFICATION OF COOLING WATER NETWORKS

6.0. FOULING IN COOLING WATER NETWORKS

Cooling water networks used in the industries can be divided into 3 different types: the open recirculation system where the water is circulated through a cooling tower to reject heat to flowing air while some of the water is replaced before it returns back to the cooling network, the closed system, where the water to be cooled is passed through a system of heat exchangers to reject heat to other process streams before it is returned to the cooling network and the once-through system where a river or lake serves as the source of water for cooling. In this system the water goes through the cooling network once and is rejected to the source.

Hewitt and Müller-Steinhagen [11] have extensively written about fresh water fouling in cooling water networks and propose that the deposition of the crystal salts on the heat transfer surface can be modeled by a mass and heat transfer analysis. They explain that the rate of deposition is dependent on the concentration of salt in the water flowing through the heat exchanger among other factors as discussed in *Chapter 5*.

Fouling in the cooling tower tend to pose more problems than fouling in the individual coolers making up the network. Fouling in cooling towers is greatly affected by the temperature of the water in the return pipe to the tower. Since the cooling tower is built based on specific ratings, the return temperature of the water must be controlled as a form of mitigation strategy for fouling in the tower.

Fouling considerations in the coolers would have to be analyzed individually and jointly with the other coolers emphasizing on its overall effect on adjacent coolers in the network. Fouling considerations in the design of heat exchanger mainly focus on the type of fluids in the exchanger, the material of construction of the heat exchanger and the effect of the pressure drop across the heat exchanger. What these design considerations lack is the integration of a fouling analysis in the design process. The availability of information relating to fouling of individual heat exchangers is important if a revamp process is to be undertaken. This is demonstrated in this part of the thesis, managing the effect of fouling in a network when a new heat exchanger is introduced.

6.1. DESIGN OF COOLING WATER NETWORKS

The general steps in the design of a cooling water network include

- Selection of the cooling tower
- Design of feed and return pipes to the cooling tower
- Selection of the circulation pump
- Design of the internal structure of the cooling network which include the selection of the branch parts and components
- Design of the individual coolers

6.1.1. Selection of the Cooling Water Tower

Cooling towers are units that are used to remove heat from water leaving a cooling water network. The water is made-up for and returned to the cooling network of heat exchangers. A Cooling tower is the principal component of a cooling water network and is provided by specialist companies. The effectiveness of a cooling tower and its capacity is a major concern in its ability to help in the cooling network. The water inlet and outlet temperature, the wet bulb temperature and the capacity define the specifications of a cooling tower. Generally it will be more convenient to obtain an over-sized cooling tower duty to make due for expansions in the company than to have a cooling tower specifically designed for the network. Cooling towers are designed to allow for maximum wetting of the tower for the loss of heat and the control of the inlet temperatures to the tower because it helps to prevent fouling in the tower.

6.1.2. Design of Feed and Return Pipes

The feed and return pipes define the amount of cooling water available and the pressure drop necessary for the network of heat exchangers. It is more convenient to design the feed and return pipes based on the economic diameter and then linking this diameter to the nominal pipe diameters available for design purposes. Peters and Timmerhaus [76] define the economic diameter as that diameter that gives a minimum cost for the sum of pumping cost and fixed cost for the pipeline. The pressure drop defined for the network of heat exchangers can then be linked to the piping specifications for the full design of the feed and return pipes.

6.1.3. Selection of a Pump

Pumps like cooling towers are provided by specialist companies and they have to be selected such that throughputs of the network can be increased at any time without necessarily changing the pump. They are designed to provide a specified throughput with considerations on low energy cost and flexibility of use. Picón-Núñez et al [20] points out that the selection of a pump is based on finding a relationship between the pumping curve, the pressure drop and delivery losses that the network would be subjected to. Some of the considerations for the selection of a pump include the pumping head, the fluid type, presence of suspensions in fluids, fluid throughput, material of construction and energy costs [77].

6.1.4. Design of the Internal Network

The internal network of the cooling network is made up of the coolers, connecting pipes, valves and other accessories for the direction and distribution of water in the network. Picón-Núñez et al [20] concludes that the basic structure of the cooling network is controlled by the piping cost and all these add up to the total pressure drop within the network. The hydraulic analysis and subsequent calculation of the pressure drops across the valves, pipes and other accessories in the cooling network can be done from data provided for in literature.

6.2. THE DESIGN PROCESS

The pressure drop across a valve, an accessory or a heat exchanger can be expressed by

$$\Delta P = KV^2 \tag{6.1}$$

The advantages of using the volumetric flow rate rather than the velocities in calculating the pressure drop across the cooler is explained in Picón-Núñez et al [20].

Consider a cooling network as depicted below.

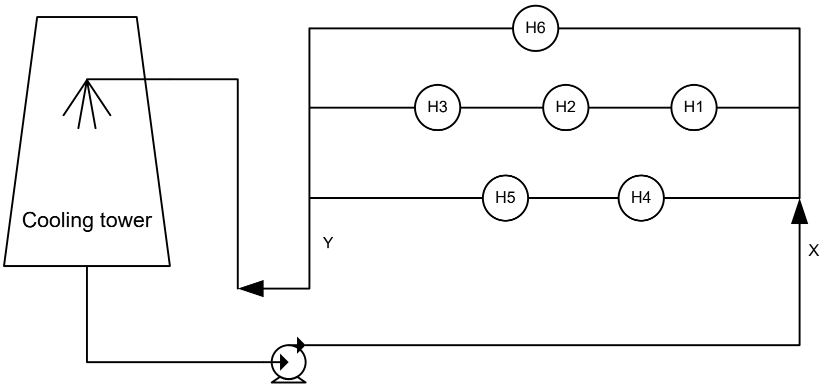


Figure 6.0: Network of six coolers (Adapted from Picón-Núñez et al [2])

The cooling network is made of three branches: A, B and C. Branch A is made up of two coolers H4 and H5, branch B is made up of three coolers H1, H2 and H3 and branch C has one cooler H6. The network is represented such that branch C is the most elevated above the ground, followed by branch B and then finally branch A which is closest to the factory floor. Water coming in from the cooling tower, and through the feed pipe to the network, enters the network at point X and divides into the three branches and then join together after the cooling process into the return pipe at point Y for the water to be returned to the cooling tower.

6.2.1. Thermo-hydraulic simulation

The aim of a hydraulic analysis of the cooling water network is to get a well-established flow in the network that will eventually define the distribution of water in each branch and the respective coolers. An encouraging phenomenon about the design and management of most cooling networks is that at any point in time the flow rate of water in the network of coolers does not change considering that there are no phase changes as the water exchanges heat with the other stream in the heat exchanger.

The hydraulic simulation of a cooling water network has been explained by Canizalez Davalos [73]. First, a static head distribution will have to be undertaken to determine the flow rates in the various branches. The simplest way of doing this is to solve the momentum and mass continuity equations. This includes considering frictional losses, changes in velocities and changes in gravitational head and once the flow is established the flow distribution through the network will be purely dependent on the frictional losses [20]. With an established flow and known flow rates through the coolers, a heat exchanger *K* value can be assumed which will help define the pressure drop across that cooler for its design. Using the analogy of resistors in an electric circuit, calculations of overall K values can be made for combinations of components in series and in parallel in a branch of the cooling water network and finally a K value for the whole network [20, 73].

The thermal simulation involves the determination of the temperature profiles of each heat exchanger and the feed and return temperatures of the cooling water to and from the cooling tower. For a fixed design of heat exchangers, different methods are available for the determination of the temperature profiles. In the ε-NTU method, the effectiveness can be expressed as a function of the ratio of the minimum to the maximum heat capacity flow rates of the streams transferring heat, the number of transfer units and the heat exchanger configuration. Standard equations for the calculation of the effectiveness of an exchanger based on different geometries can be obtained from literature [67, 72].

With known values of the inlet temperatures of the hot stream and the water to a cooler, the outlet temperatures of both streams can be calculated from the effectiveness and the ratio of the heat capacity flow rates. For example consider a heat exchanger with known inlet temperatures for both streams with the aim of finding the outlet temperatures. Take the water inlet and outlet temperature as t_{in} and t_{out} and a heat capacity flow rates as CP_w and hot stream temperatures as T_{in} and T_{out} and a heat capacity flow rate as CP_H. With knowledge of the minimum heat capacity flow rates the outlet temperatures can be calculated as thus:

For water

$$t_{out} = t_{in} + \varepsilon \frac{CP_{min}}{CP_w} (T_{in} - t_{in}) \qquad 6.2$$

For hot stream

$$T_{out} = T_{in} - \varepsilon \frac{CP_{min}}{CP_H} (T_{in} - t_{in}) \qquad 6.3$$

6.2.2. Considerations for a new design

In the cases where a new heat exchanger will have to be incorporated into an already designed network of coolers, the position and propensity to fouling of the new heat exchanger as well as that of the other heat exchangers in the network with respect to the thermo-hydraulic behavior need to be considered. For example the dangers of not considering where to locate a new heat exchanger in an existing network have been identified by Tavares [21]. Moreover the introduction of a new cooler into an existing network will result in changes in the inlet and/or outlet temperatures of the coolers in the network causing them to exhibit different wall temperatures. Crystallization fouling in fresh water systems is dependent on the wall temperatures encountered in these coolers and an increase in these values will result in an increase in the fouling rates. It should also be noted that the introduction of a new heat exchanger into an existing network would also disrupt the flow distributions, which may subsequently lead to low or high fouling rates in the heat exchangers making up the network.

It is in the light of this that an integral approach should be used when a new exchanger needs to be installed into an existing network. This approach will involve applying the flexible design method for heat exchangers as discussed in [20]. Fouling rate effects in the new heat exchanger to be installed should also be paramount in the design process as well as the effect of fouling in the already existent heat exchangers in the network. This approach is applied to the case study for the design of a new exchanger for the network of coolers given.

6.3. CASE STUDY

A new cooler needs to be installed in the network depicted in Figure 6.0. The cooling tower has an effectiveness of 0.65 and a wet bulb temperature of 24 °C. Fouling is due to crystallization fouling and a concentration of calcium sulfate in the cooling water is taken as 1800 ppm. The crystallization model described in the preceding chapter is integrated in the analysis.

The pump curve is given by the polynomial $\Delta P = 50 - 20V - 74V^2$ and is shown in Figure 6.1.

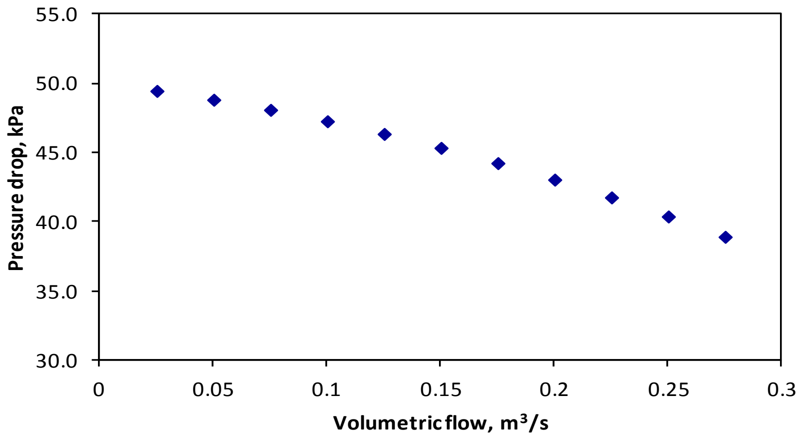


Figure 6.1: Curve for the existing pump for the cooling water network

The piping details and the stream and heat exchanger properties in all the branches are presented in Table 6.0 and Table 6.1 respectively.

Table 6.0
The piping details for the heat exchanger network

Branch	**Diameter, m**	**Length, m**	**Elevation, m**	***k* valve, Pa.s/m^3**
Feed	0.2	200	0	0.5
A	0.1	120	3	0.5
B	0.1	170	3	0.5
C	0.1	180	3	0.5
Return	0.2	200	3	12

Table 6.1
The stream and heat exchanger properties with branch indications

	H1 – B	**H2 – B**	**H3 – B**	**H4 – A**	**H5 – A**	**H6 – C**
Heat Exchanger properties						
K value	0.5	0.5	0.5	0.5	0.5	0.5
Number of tubes	800	820	450	800	400	850
Number of passes	2	2	2	2	2	2
Length, m	6.1	6.1	6.1	6.1	6.1	6.1
Tube id, m	0.0148	0.0148	0.0148	0.0148	0.0148	0.0148
Tube od, m	0.01905	0.01905	0.01905	0.01905	0.01905	0.01905
Fouling factor, m^{2}K/W	0.00224	0.0005	0.0005	0.0005	0.0005	0.001
Properties of the water						
Ref. flow rate, kg/s	85	85	85	85	85	85
htc @ ref. flow, W/m^{2}K	6570	6700	10830	6570	11900	6500
Bypass	0	0	0	0	0	0
Properties of the hot stream						
Ref. flow rate, kg/s	14	20	12	23	14	18
htc @ ref. flow, W/m^{2}K	1035	495	510	680	650	375
Bypass	0	0	0	0	0	0
Heat capacity, J/kgK	2600	2250	2100	1800	2800	2200

6.3.1. Simulation of the original network

6.4.1.1. Hydraulic simulation

The hydraulic simulation includes the determination of the total water throughput for the cooling water network through an iteration process. Details of this analysis is presented in Canizalez Davalos [73]. Knowing the flow rate through the different branches and the whole network, a thermal simulation can be carried out. The total water throughput for the simulation of the original network is 0.079 m^3/s and the water flow rates through the different branches is presented in Table 6.2.

Table 6.2
Water flow rates through the different branches in the existing network

Branch	**Flow rate, m^3/s**
A	0.025
B	0.022
C	0.032

6.4.1.2. Thermal simulation

Applying the effectiveness-NTU relationships the outlet temperature for the cooling water and the hot stream can be calculated. The cooling water is fed at a temperature of 33.7 ºC to the network of coolers and returns to the cooling tower at a temperature of 51.6 ºC. The load on the cooling tower is thus 5.95 MW and this is represented by a 17.9 ºC rise in the cooling water temperature to the tower. The thermal performance of the various coolers in the existing network is given in Table 6.3.

Table 6.3
Thermal performance of the existing network

Exchanger	**Hot stream Temperatures, ºC**		**Water Temperatures, ºC**		**Heat Load, MW**
	Inlet	Outlet	Inlet	Outlet	
H1	85	48.7	33.7	51.5	1.635
H2	70	55.3	51.5	55.5	0.371
H3	75	60.3	55.5	62.1	0.667
H4	75	46.6	33.7	44.2	1.113
H5	75	52.9	44.2	52.6	0.876
H6	80	43.5	33.7	43.6	1.327

The fouling analysis of the existing coolers in the network is presented in Table 6.4. The data represents the fouling resistances at the inlet and outlet of the individual coolers at 100 days of operation.

Table 6.4
Fouling resistances in the existing heat exchangers

Exchanger	Rd @ Inlet, m^2K/W	Rd @ outlet, m^2K/W
H1	0	0.0007951
H2	0.0001915	0.0022711
H3	0.0013475	0.0034218
H4	0	0
H5	0	0.0008895
H6	0	0

Results from Table 6.4 shows that the coolers H4 and H6 do not foul at all during the operational period whiles coolers H2 and H3 foul at both inlet and outlet to the unit but with the heaviest fouling in cooler H3. There is only fouling at the outlet of the coolers H1 and H5, whose values are smaller than those encountered in the cooler H3.

6.3.2. Revamping the existing network

From a performance analysis of the network, a new heat exchanger (HEX) is to be installed in the cooling network close to cooler H5 on branch A. The duty of the new heat exchanger to be installed is given as:

Heat Load = 0.5 MW
Hot stream Inlet Temperatures = 100 °C
Hot stream Outlet temperature = 80 °C
Heat transfer coefficient on the hot side = 800 W/m^2K
Fouling resistance = 0.005 m^2K/W

The new unit has a heat load that is less than 10% of the current cooling tower heat load.

An analysis of the location for the installation of the new heat exchanger indicates that a number of revamp options can be considered. The options include:

- Option 1: Installation of the new heat exchanger in a new branch
- Option 2: Installation of the new heat exchanger in series with heat exchanger H5 in branch A.
- Option 3: Installation of the new heat exchanger parallel to heat exchanger H5 in branch A.

These revamp options will be analyzed based on the cooling water distribution, effect on temperature and heat load on the coolers and the cooling tower and the fouling resistances to be encountered at 100 days of operation as a result of changes to the existing cooling water network.

6.3.2.1. Design Option 1: A new parallel branch

The new parallel branch D is installed in the existing network with the same branch dimensions as branch B as indicated in Figure 6.2 below.

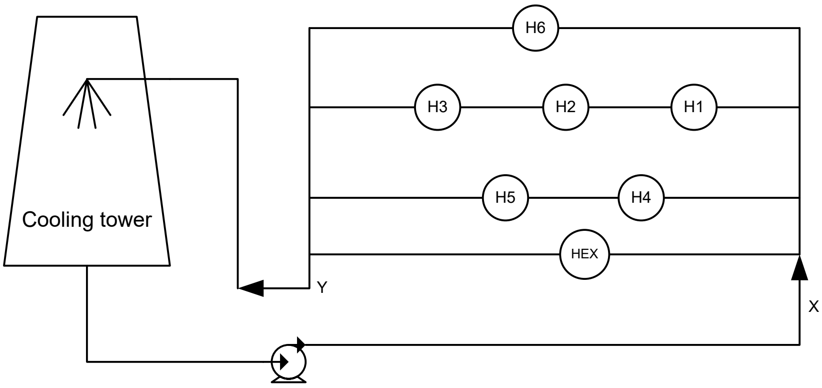


Figure 6.2: Revamp option 1 – A new parallel branch installed in the network

Option 1A: A short cut design method is used to look for a *K* value that will provide the heat load and properties of the new heat exchanger for a standard length of 6.1 m. The new heat exchanger consists of 220 tubes and a *K* value of 3200 Pa.s^2/m^6 to be installed in the new branch for the existing pump performance. A thermo-hydraulic simulation is carried out with the new heat exchanger installed and the results are presented in Table 6.6 to Table 6.8.

The effect of this revamp strategy indicates that less heat is being extracted from the coolers because the heat load on the tower has decreased by 0.05 MW as indicated in Table 6.5. This is as a result of the decrease in the flow distribution in the branches from the increased flow resistances encountered as a result of the new branch (Table 6.6). This is the case even though an overall increase in water flow rate is observed in the whole network. In a likewise manner, the outlet temperature of the hot streams in this design option registered high temperature changes up to 3.8 ^{o}C rise in cooler H3 (Table 6.7). This may seem a small value in temperature rise but the effect is seen in the fouling rates of the heat exchangers in the network (Table 6.8).

Fouling increased substantially for the coolers H1, H2, H3 and H5 as a result of the increase in the outlet hot stream temperatures. The greatest impact is in H2 where the increase in fouling rates are 642% and 35% at the inlet and outlet of the cooler respectively (Table 6.8). This is just as a result of a 3.2 °C rise in the hot stream outlet temperature; the effect with time will be great with such a fouling rate level. It is noted that the new installed cooler did not foul over the operating period.

Restoring flow distribution in the branches may present a different case and this is also investigated by changing the pump performance. This is design option 1B.

Design Option 1B: The existing pump is replaced with a new design pump with the following performance equation $\Delta P = 80 - 20V - 74V^2$. The performance curve is shown in Figure 6.3.

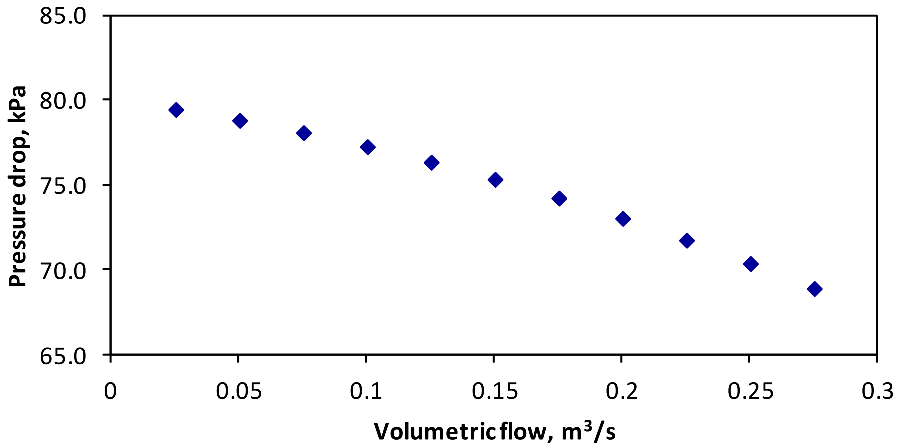


Figure 6.3: Performance curve for the new pump

This will provide a higher flow rate for the network. The new heat exchanger with the duty mentioned above to be installed in the network would have 210 tubes and a K value of 3300 Pa.s^2/m^6. The results from the thermo-hydraulic simulation are also presented in Table 6.5 to Table 6.8.

The flow distribution in the network is regulated back to the original network design with just branch C experiencing a decrease in the water flow (Table 6.6). A total increase of 33% is seen in the total water flow to the network. This contributed to a decrease in the water feed and return temperatures from and to the tower, with an increase in the tower load of 11% as indicated in Table 6.5, an increase greater than the sum of the original load on the network and the new heat exchanger installed. There is more heat recovery from this design.

Fouling rates in the coolers have decreased in all cases, Table 6.8, as a result of the decrease in the water temperatures to the network and the hot stream outlet temperatures (Table 6.7). The new heat exchanger to be installed did not also foul in this operating period.

6.3.2.2. Design option 2: Placed in series with H5

Design option 2 involves the installation of the new HEX in series with cooler H5. It will be prudent to install the new HEX after cooler H5 so as to avoid a lot of disturbances in the network. The specifications of this new heat exchanger at the heat load indicated are 378 tubes and a K value of 1200 Pa.s^2/m^6. The position of the new cooler in the network is indicated in Figure 6.4. Tables 6.5 to 6.8 indicate the results for the thermo-hydraulic simulation of the new cooling network.

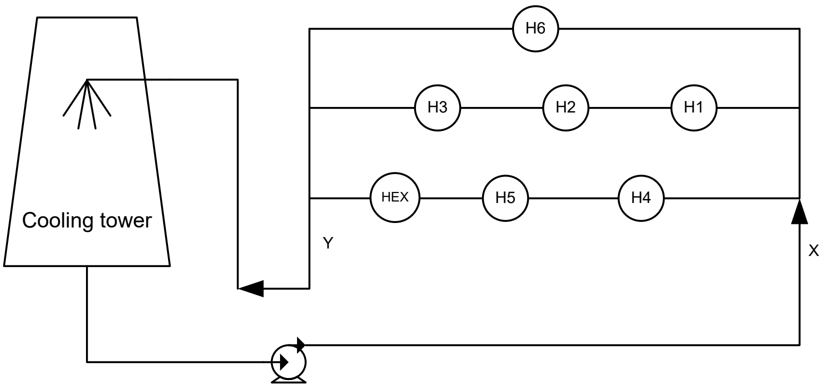


Figure 6.4: The position of the new HEX for design option 2 in the cooling network

The design option 2 provides very little resistance to flow in the network. A decrease of 1.3% in overall water flow to the network is simulated but with branch A experiencing the greatest decrease of 12% of water flow whiles branch C experiences an increase of 3% in water flow alone in the branch (Table 6.6). The water feed and return temperatures from and to the tower registered increases as well as the load on the cooling tower as indicated in Table 6.5. The thermal performance of the coolers is not greatly affected. Whiles coolers H4 and H5 experience the greatest increase in the hot stream outlet temperatures of 1.2 ^{o}C and 1.7 ^{o}C respectively, the coolers (H2 and H3) experience a decrease in hot stream outlet temperature of 0.2 ^{o}C (Table 6.7). The heat load of 6.31 MW on the tower is less than the sum of the original load and the added heat load from the new cooler.

Fouling in the coolers is not significantly high. Coolers H2 and H3 experienced decreases in fouling rates as a result of the decrease in the hot stream outlet temperatures whiles cooler H5 experienced a rise in fouling at the outlet section of the heat exchanger. The greatest effect of fouling is in the new HEX, which registered high values (Table 6.8).

6.3.2.3. Design option 3: Placed in parallel to H5

The new heat exchanger is to be placed in parallel with cooler H5 in branch A. The heat exchanger specifications include a tube count of 304 and a *K* value of 2000 Pa.s^2/m^6 for a standard length of 6.1m. Figure 6.5 shows the position of the new cooler in the cooling network.

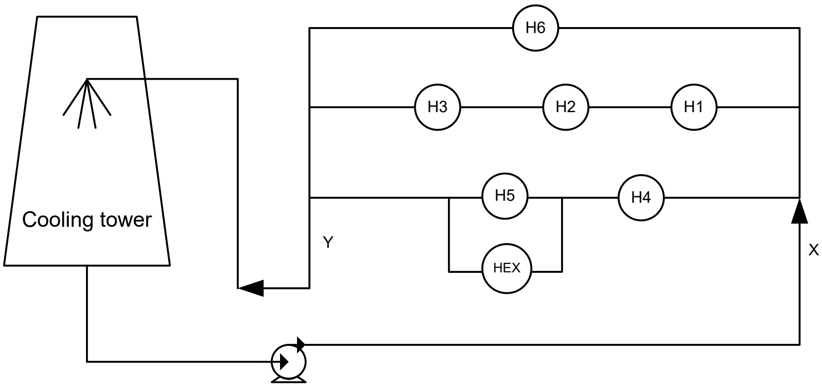


Figure 6.5: The position of the new cooler in design option 3 in the cooling network

The results for the thermo-hydraulic simulation are presented in Tables 6.5 to 6.8.

A heat load of 6.25 MW is registered on the tower, an increase less than the sum of the original load and the load of the new HEX. There is also an increases in the water feed and return temperatures to and from the network (Table 6.5). There is a slight overall increase of water flow to the network of 1.3% with branch A experiencing an increase of 12% whiles the other branches experience decreases in water flow to the network as a result of the design decision (Table 6.6). The thermal performances of the coolers decreased except for cooler H4 that experienced a slight increase in heat recovery. All the coolers but H4, registered a decrease on the hot stream outlet temperature of around 0.2°C. The hot stream leaves cooler H5 at a temperature rise of 2.3 °C (Table 6.7).

The revamp decision results in high fouling rates in all the coolers including the new HEX except for coolers H4 and H6 in the network. Cooler H5 fouled substantially at the outlet.

Table 6.5
The heat load and cooling water temperatures to and from the tower

		Tower Heat Load, MW	Water feed Temp.,ºC	Water return Temp.,ºC	% change Heat load
Original Network		5.95	33.7	51.6	
Revamp options					
Design Option 1	A	5.90	33.2	50.4	– 0.8
	B	6.60	32.1	47.2	+ 10.9
Design Option 2		6.31	34.4	53.6	+ 6.1
Design Option 3		6.25	34.1	52.9	+ 5.0

Table 6.6
Distribution of cooling water in the network

Branch	Flow rate, m3/s	% Change
Design Option 1A		
A	0.019	–24.0
B	0.017	–22.7
C	0.025	–21.9
D	0.021	
Total	0.082	+ 3.8
Design Option 1B		
A	0.025	0.0
B	0.022	0.0
C	0.031	–3.1
D	0.027	
Total	0.105	+ 32.9
Design Option 2		
A	0.022	– 12.0
B	0.022	0.0
C	0.033	+ 3.1
Total	0.078	– 1.3
Design Option 3		
A	0.028	+ 12.0
B	0.021	– 4.5
C	0.031	– 3.1
Total	0.080	+ 1.3

Table 6.7
Thermal performance of the heat exchangers in the network

Exchanger	**Hot stream Temperatures, °C**		**Water Temperatures, °C**		**Heat Load, MW**	**Change Hot Outlet T, °C**	**Change Heat Load, MW**
	Inlet	Outlet	Inlet	Outlet			
Design Option 1A							
H1	85	51.0	33.2	54.9	1.530	+ 2.3	– 0.105
H2	70	58.5	54.9	59.0	0.290	+ 3.2	– 0.081
H3	75	63.8	59.0	65.5	0.464	+ 3.8	– 0.203
H4	75	47.8	33.2	46.4	1.066	+ 1.2	– 0.047
H5	75	55.5	46.4	55.9	0.771	+ 2.6	– 0.105
H6	80	45.6	33.2	45.6	1.283	+ 2.1	– 0.044
New HEX			33.2	38.5	0.500		
Design Option 1B							
H1	85	47.8	32.1	50.7	1.675	– 0.9	+ 0.040
H2	70	54.7	50.7	55.0	0.385	– 0.6	+ 0.014
H3	75	60.0	55.0	61.9	0.620	– 0.3	– 0.047
H4	75	45.7	32.1	43.3	1.151	– 0.9	+ 0.038
H5	75	52.3	43.3	52.0	0.901	– 0.6	+ 0.025
H6	80	42.5	32.1	42.5	1.367	– 1.0	+ 0.040
New HEX			32.1	36.6	0.500		
Design Option 2							
H1	85	48.8	34.4	51.4	1.629	+ 0.1	– 0.006
H2	70	55.1	51.3	55.3	0.374	– 0.2	+ 0.003
H3	75	60.1	55.3	61.8	0.619	– 0.2	– 0.048
H4	75	47.8	34.4	45.9	1.068	+ 1.2	– 0.045
H5	75	54.6	46.0	54.6	0.809	+ 1.7	– 0.067
H6	80	43.8	34.4	43.8	1.314	+ 0.3	– 0.013
New HEX			54.6	60.0	0.500		
Design Option 3							
H1	85	49.3	34.1	52.2	1.605	+ 0.6	– 0.030
H2	70	56.0	52.2	56.2	0.353	+ 0.7	– 0.018
H3	75	61.0	56.2	62.8	0.579	+ 0.7	– 0.088
H4	75	46.4	34.1	43.8	1.120	– 0.2	+ 0.007
H5	75	55.2	43.8	56.3	0.784	+ 2.3	– 0.092
H6	80	44.1	34.1	44.2	1.307	+ 0.6	– 0.020
New HEX			43.8	53.2	0.500		

Table 6.8
Fouling resistances in the heat exchangers after 100 days of operation

Heat Exchanger	Inlet Rd, m^2K/W	Outlet Rd, m^2K/W	%Δ Inlet Rd	%Δ Outlet Rd
Design Option 1A				
H1	0	0.00145	0	+ 81
H2	0.00141	0.00307	+ 642	+ 35
H3	0.00190	0.00350	+ 42	+ 2
H4	0	0	0	0
H5	0	0.00154	0	+73
H6	0	0	0	0
New HEX	0	0		
Design Option 1B				
H1	0	0.00061	0	– 24
H2	0	0.00203	– 100	– 11
H3	0.00119	0.00332	– 11	– 3
H4	0	0	0	0
H5	0	0.00073	0	– 18
H6	0	0	0	0
New HEX	0	0		
Design Option 2				
H1	0	0.00079	0	– 1
H2	0.00016	0.00226	– 16	0
H3	0.00133	0.00345	– 1	+ 1
H4	0	0	0	0
H5	0	0.00136	0	+ 53
H6	0	0	0	0
New HEX	0.00279	0.00750		
Design Option 3				
H1	0	0.00098	0	+ 23
H2	0.00053	0.00251	+ 179	+ 11
H3	0.00151	0.00349	+ 13	+ 2
H4	0	0	0	0
H5	0	0.00231	0	+ 160
H6	0	0	0	0
New HEX	0	0.00131		

The specifications of heat exchangers used in this case study is presented in Table 6.9 below. These are only the initial design specifications and for the real engineering project a commercial design software tool will be necessary for the detailed design specifications for the coolers.

Table 6.9
The specifications for the new cooler

Revamp options		Number of tubes	K value, $Pa.s^2/m^6$	Flow rate, m^3/s	Pressure drop, kPa	Water Temp. rise°C	Fouling @ 100 days
Design Option	1A	220	3200	0.0213	1.5	5.3	0
	1B	210	3300	0.0269	2.4	4.5	0
Design Option 2		378	1200	0.0221	0.6	5.4	0.00750
Design Option 3		304	2000	0.0127	0.13	9.4	0.00131

Installing a new branch in the existing network, design option 1A, with the existing pump results in a decrease in heat recovery in the whole network. This is seen in the decrease in the heat load on the cooling tower as well. Moreover, there is an increase in the temperatures at which the hot stream leaves the coolers contributing to high fouling levels in most of the heat exchangers. All these problems are mitigated for with a change in the pump performance for the network as in design option 1B. Fouling decreased as a result and the flow distribution in all branches were almost restored to the original values.

Design option 2 with the new HEX installed in series with cooler H5 resulted in a slight decrease in the total flow to the network. Cooler H5 experienced the greatest effect in both changes in outlet temperatures, heat load and fouling levels. The new HEX registered high fouling levels as a result of its position in branch A. Being placed after H5 means high water inlet temperatures and thus the high fouling levels.

Placing the new HEX in parallel with cooler H5 in branch A resulted in a slight increase in water flow to the network but resulted in slight reductions in the loads of the existing coolers. Fouling also increased greatly in the existing coolers with the new HEX also fouling. This is because the water flow rate to these heat exchangers will be less than that in the branch.

6.4. CONSIDERATIONS FOR MODIFICATION OF COOLING WATER NETWORKS

The results of this case study above would serve as a guideline for the modification and design of a cooling water network. Different scenarios have been considered in this analysis: installation of a new branch, installation in series and installations in parallel. From the results presented the following recommendations can be considered in the design and modification of a cooling water network:

- A thermo-hydraulic simulation of how a network will perform with the installation of a new unit is important not only on the analysis of existing coolers in the network but also in the effects of the new cooler on the network. This simulation must include consideration of pump performance and cooling water performance. This is because both the hydraulic performance and the thermal performance of all the heat exchangers in the network are altered in a revamp situation.

- Heat exchangers will have to be designed with great consideration to the flow resistance it will present in the network. An increased resistance to flow distribution results in a reduction in the heat load on the individual coolers and the cooling tower. This will result in high wall temperatures a requisite for high fouling levels in the individual coolers and subsequently the cooling tower. An increase in fouling rates will decrease the performance of the cooling network in heat recovery. A simple design consideration has been incorporated for this purpose based on the *K* value of the heat exchanger.

- The installation of a new cooler into an existing network presents very little thermo-hydraulic disturbance to flow distribution when it is installed in series with the other heat exchangers than when installed in parallel as has been demonstrated here in this case study. Moreover, installing in series reduces the piping cost and no consideration for the installation of a new pump. Placing a new unit in a new branch in a network is worth considering, if there is very little restriction on available space and the need to invest in a new pump.

- A method for fouling analysis has to be integrated in the thermo-hydraulic analysis of the cooling network as illustrated in this work to help in the consideration of the performance of the new heat exchanger and how this performance is related to the heat exchangers in the network. The fouling analysis helped monitor the performance of the coolers with time and thus a great tool for operational and mitigation decisions.

PART 2: CONCLUSION

The application of a crystallization fouling model for the analysis of fouling in heat exchangers; both shell and tube heat exchangers and Compabloc units, has been shown in case studies on how they may be used in taking design decisions or analysis and prediction of fouling in heat exchangers. Effectively, crystallization fouling depends on the bulk water temperature and the velocity but depending on the conditions of operation one of these may dominate helping to control the effect of fouling in the heat exchangers under investigation.

Importantly is the fact that this methodology only considers a simple method from the combinations of earlier investigations to select heat exchanger design parameters for a cooling process. There will be the need to use a more detailed heat exchanger design tool to fine tune the initial design decisions from the algorithm described above.

Revamp operations in processing plants are necessary as a result of increased production or the need to increase energy recovery. Thermo-hydraulic simulation of an existing network and subsequent simulation with the installation of a new heat exchanger is paramount in deciding the best revamp option.

Fouling analysis is seen as a great tool in taking the revamp decision and must be made an integral part of the design and revamp strategy.

GENERAL CONCLUSION

The pre-heat train and the cooling water network are innovative areas in the process industry where heat recovery is highly recommended and helps in reducing the external energy cost of processing. In the case of the pre-heat train the maximum amount of energy needs to be recuperated albeit the economic or technical limitations whereas in the cooling water network the amount of heat lost to the environment needs to be as low as possible. The fouling process can be complex but their analysis and effect in these two-heat recovery areas have been studied based on the design and operation of the heat exchangers involved. The control and application of simple mitigation strategies to the fouling mechanism is always necessary when heat recovery is at the highest priority of the processing plant.

The following are the conclusions of this investigation:

1. A new chemical reaction fouling model, The Asphaltene precipitation (AP) model, has been developed and demonstrated with applications in shell and tube heat exchangers and Compabloc units. The simplicity and practicality of the model is demonstrated with very good predictions to the cases presented.
2. The pre-heat train in its complex nature may be tackled from a simple point of view like the use of the path analysis to analyze and increase the heat recovery. This has been successfully demonstrated with a worked example.
3. Advanced heat exchanger technologies are necessary when the mitigation of fouling becomes very important. A Compabloc unit is one of these advances that overcome the shortcomings of the shell and tube heat exchanger. The Compabloc unit can operate at higher fluid shear stresses gaining advantage over fouling and at the same time providing higher heat recovery solutions.
4. The use of fouling models in the design process and revamp situations is important when the real effect of fouling needs to be identified. A fouling model is successfully applied in the revamping of a cooling water network. The use of the model did not only provide information on the new heat exchanger to be installed but it also showed the effect of the new installation on the fouling rates of the already existing heat exchangers.

RECOMMENDATIONS FOR FUTURE WORK

In line with this project and the various investigations that have been made, the following are suggested for future work and should be considered in line with this research:

1. A further exploration of the application of the Asphaltene Precipitation model developed in *Chapter 2* of this work is recommended. It has been faithfully applied to shell and tube heat exchangers and Compabloc units where the analysis and predictions have been very favorable. Applications to other types of heat exchangers will be a valuable tool in fine-tuning the model. Further verification with plant data and real time refinery scenarios for cases where the shell and tube heat exchanger and the Compabloc unit are located on the same pre-heat train will be a great step to optimize the model.
2. The fouling mechanism is complex and an experimental procedure may be necessary in some cases to ascertain the models developed for analysis. An experimental work to be carried out alongside this research is recommended.
3. Compabloc units are becoming of great use not only in the crude oil pre-heat trains but also in the power generation industries. In this work, the thermal analysis is presented and further work is recommended at the design and further applications of these heat exchangers.
4. The relation developed for crystallization fouling in *Chapter 5* will also need further exploring for optimization purposes and its use in the analysis and prediction of fouling in heat exchangers for the cooling industry. Further work is recommended in the development of a fouling threshold for the design of coolers. This will change the direction of design of heat exchangers for the cooling water industry.

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APPENDIX

APPENDIX A – PUBLICATIONS AND CONFERENCE PAPERS

PUBLICATIONS

Polley G. T., Tamakloe E. K., Picón-Núñez M., Ishiyama E. M. And Wilson D. I. (2013), **Applied thermo-hydraulic simulation and heat exchanger analysis to the retrofit of heat recovery systems**, Applied Thermal Engineering 51(1-2), 137-143.

Tamakloe E. K., Polley G. T. and Picón-Núñez M. (2012) **Design of Compabloc exchangers to mitigate refinery fouling**, Applied Thermal Engineering, *In press – Corrected proof* (Available online 26 September, 2012)

Picón-Núñez M., Polley G. T., Canizalez-Davalos L. and Tamakloe E. K., (2012) **Design of coolers for use in an existing cooling water network**, Applied Thermal Engineering 43, 51-59.

CONFERENCE AND CONGRESS PAPERS

2013

Tamakloe E. K., Polley G. T and Picón Núñez M, (2013). **Modeling crystallization fouling for design purposes in a cooling water network.** XXXIV National Meeting and 3rd International Congress AMIDIQ-2013, Mazatlan, Sinaloa (Mexico). May 7 – 10. Abstract No. 380.

G. T. Polley, Elvis K Tamakloe, Picón Núñez M, (2013). **The Design and Revamp of Cooling water networks.** 13thAIChE Spring Meeting and 9th global Congress on Process Safety, San Antonio, TX (USA). April 28- May 2. Abstract No. 61a.

G. T. Polley, Elvis K Tamakloe, Picón Núñez M, (2013). **Identifying Heat Recovery Paths to be used in improving the Energy Efficiency of Pre-Heat Train.** 13thAIChE Spring Meeting and 9th global Congress on Process Safety, San Antonio, TX (USA). April 28- May 2. Abstract No. 80d.

2012

Tamakloe E. and Polley G.T. (2012). **Design of Compabloc Exchangers to Mitigate Refinery Fouling**.12th AIChE Spring Meeting and 8th global Congress on Process Safety, Houston, TX (USA). April 1-5. Abstract No. 31c

Tamakloe E. K., Polley G. T. and Picón Núñez Martín, (2012), **Análisis de rendimiento de un intercambiador de calor de placas soldadas - el caso de la unidad Compabloc**. XXXIII National Meeting and 2nd International Congress AMIDIQ-2012, San Jose de Los Cabos, Baja California Sur (Mexico). May 1 – 4. Abstract No. PRO-C-336.

2011

Tamakloe E. K., Polley G. T. and Picón Núñez Martín, (2011), **Development of a Model for the Prediction of Fouling in Heat Exchangers Processing Crude Oil**. XXXII National Meeting and 1st International Congress AMIDIQ-2011, Riviera Maya, Quintana Roo (Mexico). May 3 – 6. Abstract No. PROC-134.

Polley G.T., Tamakloe E. and Picón Núñez M (2011). **Analysis of the Performance of Multi-Pass Welded Plate Heat Exchangers**. The Twelfth UK National Heat Transfer Conference, Leeds, UK.30/31st August & 1stSeptember.Abstract No. 0025.

Polley G.T., Tamakloe E. and Picón Núñez Martín (2011). **Models for Chemical Reaction Fouling**.11th AIChE Spring Meeting, Chicago, IL (USA). March 13-17. Abstract No. 94c.

Polley G.T., Tamakloe E., Wilson D. I., and Ishiyama E. (2011). **Thermal Hydraulic Simulation of Pre-Heat Train Behavior**.11th AIChE Spring Meeting, Chicago, IL (USA). March 13-17. Abstract No. 94f.

Polley G.T., Tamakloe E. and Picón Núñez Martín (2011). **Evaluating the Heat Exchanger Technologies available for use in refinery pre-heat trains.**11th AIChE Spring Meeting, Chicago, IL (USA). March 13-17. Abstract No. 113a.

Tamakloe E. K., Polley G. T. and Picón Núñez Martín, (2011), **Development of a Model for the Prediction of Fouling in Heat Exchangers Processing Crude Oil**. XXXII National Meeting and 1st International Congress AMIDIQ-2011, Riviera Maya, Quintana Roo (Mexico). May 3 – 6. Abstract No. PROC-134.

APPENDIX B – COMPABLOC ANALYSIS

A couple of examples are presented to explain the algorithm used to analyze the thermo-hydraulic performance of the Compabloc units.

Analysis of a 1-3 pass Compabloc unit

Conditions:

1 – Pass stream 1 with temperature t and mass flow rate m_1

3 – Pass stream 2 with temperature T and mass flow rate m_2

Minimum pass

$$minp = 1$$

Maximum pass

$$maxp = 3$$

$$Ratio = \frac{maxp}{minp} = \frac{3}{1} = 3$$

Number sections

$$Number\ of\ sections = maxp = 3$$

The 1-3 pass unit can be represented in Figure B0:

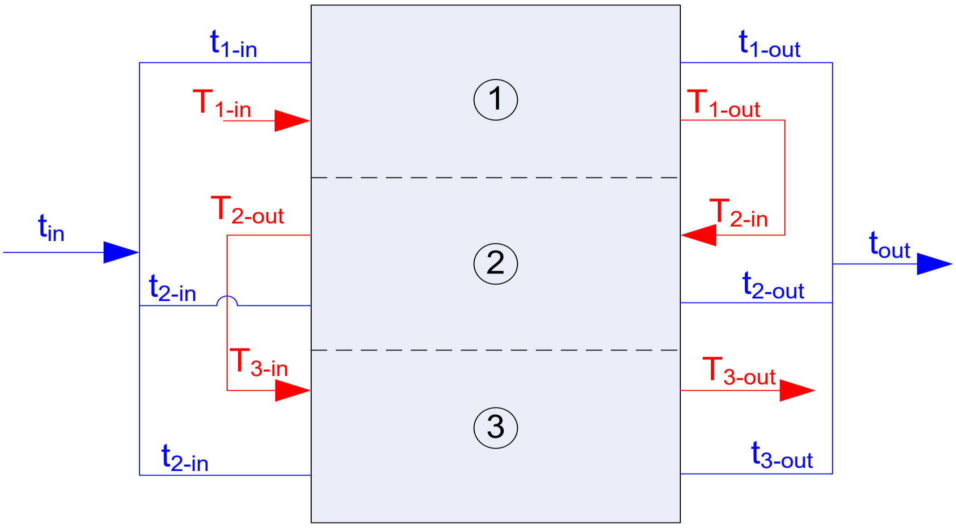


Figure B0: The 1-3 base configuration

Breaking it down into the individual sections, we obtain in Figure B1 the following independent heat exchangers:

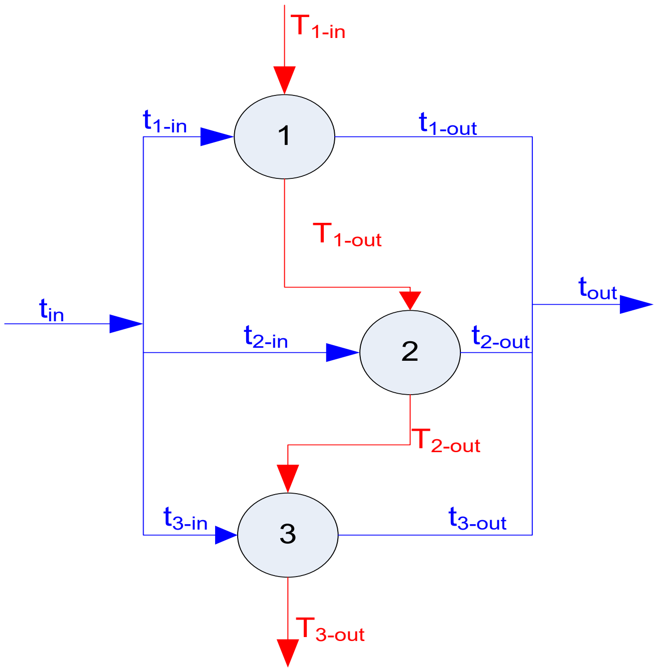


Figure B1: The 1-3 configuration defined by individual exchangers

Known: T_{in} and t_{in}

Unknown: T_{out} and t_{out}

Observations:

For stream 1

$$t_{in} = t_{1-in} = t_{2-in} = t_{3-in}$$

t_{1-out}, t_{2-out} and t_{3-out} are fractions of t_{out}

For stream 2

$$T_{1-in} = T_{in}$$

$$T_{1-0ut} = T_{2-in}$$

$$T_{2-0ut} = T_{3-in}$$

The summary of the properties of the sections is presented in the Table 4.2 below.

Table B0
Properties of a 1-3 configuration

Section	**Area** (Fraction of total heat transfer area)	**Mass flow rate** (fraction of m) **Stream 1**	**Stream 2**
1	1/3	1/3	1
2	1/3	1/3	1
3	1/3	1/3	1

For

Section 1

1. Determine the heat transfer area for the section

$$A_1 = \frac{1}{3}A$$

2. Calculate the heat capacity flow rate from the following mass flow rates

$$m_1 = \frac{1}{3}m_1$$

$$m_2 = m_2$$

$$CP1 = m_1 h_1$$

$$CP2 = m_2 h_2$$

3. Determine the minimum heat capacity flow rate $CP_{min}$from the equations above, the ratio of the heat capacities and the effectiveness as given in *step 11* above.

$$N_{TU} = \frac{U_1 A_1}{CP_{min}}$$

$$ø = Exp(N_{TU})$$

$$\varepsilon_1 = \frac{1}{\left[\frac{Rø^R}{ø^R - 1} + \frac{ø}{ø - 1} - \frac{1}{\ln ø}\right]}$$

4. Calculate the outlet temperatures from the following equations

$$t_{1-out} = t_{1-in} + \varepsilon_1 \left(\frac{CP_{min}}{CP1}\right)(T_{1-in} - t_{1-in})$$

$$T_{1-out} = T_{1-in} - \varepsilon_1 \left(\frac{CP_{min}}{CP2}\right)(T_{1-in} - t_{1-in})$$

Section 2

5. Determine the heat transfer area for the section

$$A_2 = \frac{1}{3}A$$

6. Determine the heat capacity rates from the following mass flow rates

$$m_1 = \frac{1}{3}m_1$$

$$m_2 = m_2$$

$$CP1 = m_1 h_1$$

$$CP2 = m_2 h_2$$

7. Determine the minimum heat capacity flow rate $CP_{min}$from the equations above, the ratio of the heat capacities and the effectiveness as given in *step 11* above.

$$N_{TU} = \frac{U_2 A_2}{CP_{min}}$$

$$ø = Exp(N_{TU})$$

$$\varepsilon_2 = \frac{1}{\left[\frac{Rø^R}{ø^R - 1} + \frac{ø}{ø - 1} - \frac{1}{\ln ø}\right]}$$

8. Calculate the outlet temperatures from the following equations

$$t_{2-out} = t_{2-in} + \varepsilon_2 \left(\frac{CP_{min}}{CP1}\right)(T_{2-in} - t_{2-in})$$

$$T_{2-out} = T_{2-in} - \varepsilon_2 \left(\frac{CP_{min}}{CP2}\right)(T_{2-in} - t_{2-in})$$

Section 3

9. Determine the heat transfer area for the section

$$A_3 = \frac{1}{3}A$$

10. Determine the heat capacity rates from the following mass flow rates

$$m_1 = \frac{1}{3}m_1$$

$$m_2 = m_2$$

$$CP1 = m_1 h_1$$

$$CP2 = m_2 h_2$$

11. Determine the minimum heat capacity flow rate $CP_{min}$from the equations above, the ratio of the heat capacities and the effectiveness as given in *step 11* above.

$$N_{TU} = \frac{U_3 A_3}{CP_{min}}$$

$$ø = Exp(N_{TU})$$

$$\varepsilon_3 = \frac{1}{\left[\frac{Rø^R}{ø^R - 1} + \frac{ø}{ø - 1} - \frac{1}{\ln ø}\right]}$$

12. Calculate the outlet temperatures from the following equations

$$t_{3-out} = t_{3-in} + \varepsilon_3 \left(\frac{CP_{min}}{CP1}\right)(T_{3-in} - t_{3-in})$$

$$T_{3-out} = T_{3-in} - \varepsilon_3 \left(\frac{CP_{min}}{CP2}\right)(T_{3-in} - t_{3-in})$$

13. Determine the outlet temperature for both streams for the section.
    a. For stream 1 which has just one pass

$$t_{out} = \frac{1}{3}(t_{1-out} + t_{2-out} + t_{3-out})$$

    b. For stream 2

$$T_{out} = T_{2-out}$$

Example of a 1-2 Compabloc unit

Consider a Compabloc unit of 2 hot stream passes and 1 cold stream pass. The objective is to determine the outlet temperatures at the end of each section.

Data:

Total heat transfer area $A = 255.67\ m^2$

The inlet conditions for the exchanger are presented below:

Table B1
Exchanger duty for a 1-2 configuration

	Mass flow rate, *kg/s*	**Heat capacity, *J/kg. K***	**Inlet temperature, *C***
Stream 1 (Cold stream)	30.77	2499.9	209.6
Stream 2 (Hot stream)	14.34	2645.1	341.5

Approximate overall heat transfer coefficient $U = 195.823 \frac{W}{m^2K}$

Section overall heat transfer coefficients:

Table B2
Overall heat transfer coefficients for each section for a 1-2 configuration

Sections	Initial U	Iterated U
1	195.8	193.9
2	195.3	193.9

Section 1

1. Determine the heat transfer area of the section

$$A_1 = \frac{1}{2}A = \frac{1}{2} \times 255.67 = 127.835\ m^2$$

2. Determine the mass flow rates for the section

$$m_1 = \frac{1}{2}m = \frac{1}{2} \times 30.77 = 15.39\,\frac{kg}{s}$$

$$CP_1 = m_1 h_1 = 15.4 \times 2499.89 = 38473.307\,\frac{J}{s.K}$$

$$m_2 = m = 14.34\,\frac{kg}{s}$$

$$CP_2 = m_2 h_2 = 14.34 \times 2645.13 = 37931.164\,\frac{J}{s.K}$$

3. Determine the minimum heat capacity rate and the ratio of the heat capacity rates

$$CP_2 < CP_1$$

$$CP_{min} = CP_2 = 37931.164\,\frac{J}{s.K}$$

$$R = \frac{CP_2}{CP_1} = \frac{37931.164}{38473.307} = 0.9859$$

Effectiveness

$$N_{TU} = \frac{U_1 A_1}{CP_{min}}$$

$$N_{TU} = \frac{193.929 * 127.835}{37931.164} = 0.6536$$

$$ø = Exp(N_{TU}) = e^{0.6536} = 1.9224$$

$$\varepsilon = \frac{1}{\left[\frac{Rø^R}{ø^R-1} + \frac{ø}{ø-1} - \frac{1}{\ln ø}\right]}$$

$$\varepsilon_1 = \frac{1}{\left[\frac{0.9859\times(1.9224)^{0.9859}}{(1.9224)^{0.9859}-1} + \frac{1.9224}{1.9224-1} - \frac{1}{\ln(1.9224)}\right]}$$

$$\varepsilon_1 = 0.3803$$

4. Calculate the outlet temperatures

$$\frac{CP_{min}}{CP_1} = R = 0.9859$$

$$\frac{CP_{min}}{CP_2} = 1$$

For stream 1

$$t_{1-out} = t_{1-in} + \varepsilon_1\left(\frac{CP_{min}}{CP1}\right)(T_{1-in} - t_{1-in})$$

$$t_{1-out} = 209.6 + 0.3803 \times 0.9859(341.5 - 209.6)$$

$$t_{1-out} = 259.054\ C$$

For stream 2

$$T_{1-out} = T_{1-in} - \varepsilon_1\left(\frac{CP_{min}}{CP2}\right)(T_{1-in} - t_{1-in})$$

$$T_{1-out} = 341.5 - 0.3803(341.5 - 209.6)$$

$$T_{1-out} = 291.338\ C$$

Section 2

5. Determine the heat transfer area of the section

$$A_2 = \frac{1}{2}A = \frac{1}{2} \times 255.67 = 127.835\ m^2$$

6. Determine the mass flow rates for the sections

$$m_1 = \frac{1}{2}m = \frac{1}{2} \times 30.77 = 15.39\frac{kg}{s}$$

$$CP_1 = m_1 h_1 = 15.4 \times 2499.89 = 38473.307\frac{J}{s.K}$$

$$m_2 = m = 14.34\frac{kg}{s}$$

$$CP_2 = m_2 h_2 = 14.34 \times 2645.13 = 37931.164\frac{J}{s.K}$$

7. Determine the minimum heat capacity rate and the ratio of the heat capacity rates

$$CP_{min} = CP_2 = 37931.164\frac{J}{s.K}$$

$$R = \frac{CP_2}{CP_1} = \frac{37931.164}{38473.307} = 0.9859$$

Effectiveness

$$N_{TU} = \frac{U_2 A_2}{CP_{min}}$$

$$N_{TU} = \frac{193.921 * 127.835}{37931.164} = 0.6535$$

$$ø = Exp(N_{TU}) = e^{0.6535} = 1.9223$$

$$\varepsilon = \frac{1}{\left[\frac{Rø^R}{ø^R - 1} + \frac{ø}{ø - 1} - \frac{1}{\ln ø}\right]}$$

$$\varepsilon_2 = \frac{1}{\left[\frac{0.9859\times(1.9223)^{0.9859}}{(1.9223)^{0.9859}-1} + \frac{1.9223}{1.9223-1} - \frac{1}{\ln(1.9223)}\right]}$$

$$\varepsilon_2 = 0.3803$$

8. Calculate the outlet temperatures

$$\frac{CP_{min}}{CP_1} = R = 0.7489$$

$$\frac{CP_{min}}{CP_2} = 1$$

For stream 1

$$t_{2-out} = t_{2-in} + \varepsilon_2\left(\frac{CP_{min}}{CP1}\right)(T_{2-in} - t_{2-in})$$

$$t_{2-out} = 209.6 + 0.3803 \times 0.9859(291.338 - 209.6)$$

$$t_{2-out} = 240.247\ C$$

For stream 2

$$T_{2-out} = T_{2-in} - \varepsilon_2\left(\frac{CP_{min}}{CP2}\right)(T_{2-in} - t_{2-in})$$

$$T_{2-out} = 291.338 - 0.3803(291.338 - 209.6)$$

$$T_{2-out} = 260.253\ C$$

14. Determine the outlet temperature for both streams for the section.
    c. For stream 1 which has just one pass

$$t_{out} = \frac{1}{2}t_{1-out} + \frac{1}{2}t_{2-out}$$

$$t_{out} = \frac{1}{2}(259.054) + \frac{1}{2}(240.247) = 249.65\ C$$

d. For stream 2

$$T_{out} = T_{2-out} = 260.25\ C$$

Distribution of the temperatures across the Compabloc unit

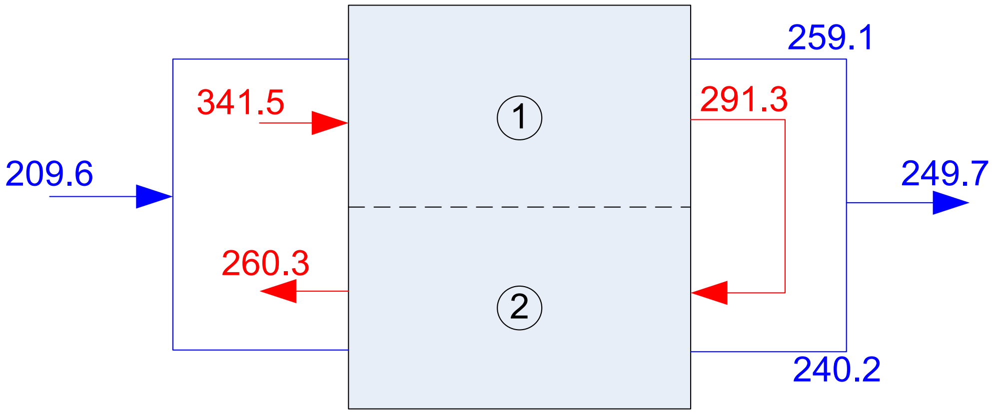


Figure B2: Temperature profile of the 1-2 configuration

Summary of the temperatures for each section

Table B3
Temperature profile of a 1-2 configuration

Section	Stream 1		Stream 2	
	Inlet	*Outlet*	*Inlet*	*Outlet*
1	209.6	259.1	341.5	291.3
2	209.6	240.2	291.3	260.3

Example of a 3-2 Compabloc unit

Consider a Compabloc unit of a 3 hot stream passes and 2 cold stream pass. The objective is to determine the outlet temperatures at the end of each section.

Data:

Total heat transfer area $A = 255.67\ m^2$

The inlet conditions for the exchanger are presented below:

Table B4
Exchanger duty for a 3-2 configuration

	Mass flow rate, *kg/s*	Heat capacity, *J/kg. K*	Inlet temperature, *C*
Stream 1 (Cold stream)	30.77	2499.89	209.6
Stream 2 (Hot stream)	14.34	2645.13	341.5

Approximate overall heat transfer coefficient $U = 275.913 \frac{W}{m^2K}$

Section overall heat transfer coefficients:

Table B5
Overall heat transfer coefficients for each section for a 3-2 configuration

Section	Initial U	Iterated U
1	277.651	270.611
2	272.165	270.600
3	270.495	270.609
4	265.933	270.596

Section 1

1. Determine the heat transfer area of the section

$$A_1 = \frac{1}{3}A = \frac{1}{3} \times 255.67 = 85.223\ m^2$$

2. Determine the mass flow rates for the sections

$$m_1 = \frac{2}{3}m = \frac{2}{3} \times 30.77 = 20.51 \frac{kg}{s}$$

$$CP_1 = m_1 h_1 = 20.51 \times 2499.89 = 51272.744 \frac{J}{s.K}$$

$$m_2 = m = 14.34 \frac{kg}{s}$$

$$CP_2 = m_2 h_2 = 14.34 \times 2645.13 = 37931.164 \frac{J}{s.K}$$

3. Determine the minimum heat capacity rate and the ratio of the heat capacity rates

$$CP_2 < CP_1$$

$$CP_{min} = CP_2 = 37931.164 \frac{J}{s.K}$$

$$R = \frac{CP_2}{CP_1} = \frac{37931.164}{51272.744} = 0.7398$$

Effectiveness

$$N_{TU} = \frac{U_1 A_1}{CP_{min}}$$

$$N_{TU} = \frac{270.611 * 85.223}{37931.164} = 0.6080$$

$$ø = Exp(N_{TU}) = e^{0.6080} = 1.8368$$

$$\varepsilon = \frac{1}{\left[\frac{Rø^R}{ø^R - 1} + \frac{ø}{ø - 1} - \frac{1}{\ln ø}\right]}$$

$$\varepsilon_1 = \frac{1}{\left[\frac{0.7398 \times (1.8368)^{0.7398}}{(1.8368)^{0.7398} - 1} + \frac{1.8368}{1.8368 - 1} - \frac{1}{\ln(1.8368)}\right]}$$

$$\varepsilon_1 = 0.3857$$

4. Set the value of mid temperature t_{34-out}

$$t_{34-out} = t_{in} + \frac{1}{2}\varepsilon_1 (T_{in} - t_{in})$$

$$t_{34-out} = 209.6 + \frac{1}{2} \times 0.3857(341.5 - 209.6)$$

$$t_{34-out} = 235.037\ C$$

5. Calculate the outlet temperatures

$$\frac{CP_{min}}{CP_1} = R = 0.7398$$

$$\frac{CP_{min}}{CP_2} = 1$$

$$t_{1-in} = t_{34-out} = 235.037 \;\; C$$

For stream 1

$$t_{1-out} = t_{1-in} + \varepsilon_1 \left(\frac{CP_{min}}{CP1}\right)(T_{1-in} - t_{1-in})$$

$$t_{1-out} = 235.037 + 0.3857 \times 0.7398(341.5 - 235.037\;)$$

$$t_{1-out} = 265.415\; C$$

For stream 2

$$T_{1-out} = T_{1-in} - \varepsilon_1 \left(\frac{CP_{min}}{CP2}\right)(T_{1-in} - t_{1-in})$$

$$T_{1-out} = 341.5 - 0.3857(341.5 - 235.037)$$

$$T_{1-out} = 300.437\; C$$

Section 2

6. Determine the heat transfer area of the section

$$A_2 = \frac{1}{6}A = \frac{1}{6} \times 255.67 = 42.612\; m^2$$

7. Determine the mass flow rates for the sections

$$m_1 = \frac{1}{3}m = \frac{1}{3} \times 30.77 = 10.26\frac{kg}{s}$$

$$CP_1 = m_1 h_1 = 10.26 \times 2499.89 = 25648.871\frac{J}{s.K}$$

$$m_2 = \frac{1}{2}m = \frac{1}{2} \times 14.34 = 7.17\ \frac{kg}{s}$$

$$CP_2 = m_2 h_2 = 7.17 \times 2645.13 = 18965.582\frac{J}{s.K}$$

8. Determine the minimum heat capacity rate and the ratio of the heat capacity rates

$$CP_2 < CP_1$$

$$CP_{min} = CP_2 = 18965.582\frac{J}{s.K}$$

$$R = \frac{CP_2}{CP_1} = \frac{18965.582}{25648.871} = 0.7394$$

Effectiveness

$$N_{TU} = \frac{U_2 A_2}{CP_{min}}$$

$$N_{TU} = \frac{270.6 * 42.612}{18965.582} = 0.6080$$

$$ø = Exp(N_{TU}) = e^{0.6080} = 1.8368$$

$$\varepsilon = \frac{1}{\left[\frac{Rø^R}{ø^R-1} + \frac{ø}{ø-1} - \frac{1}{\ln ø}\right]}$$

$$\varepsilon_2 = \frac{1}{\left[\frac{0.7394\times(1.8368)^{0.7394}}{(1.8368)^{0.7394}-1} + \frac{1.8368}{1.8368-1} - \frac{1}{\ln(1.8368)}\right]}$$

$$\varepsilon_2 = 0.3858$$

9. Calculate the outlet temperatures

$$\frac{CP_{min}}{CP_1} = R = 0.7394$$

$$\frac{CP_{min}}{CP_2} = 1$$

$$t_{2-in} = t_{34-out} = 265.415 \ C$$

$$T_{2-in} = T_{1-out} = 300.437 \ C$$

For stream 1

$$t_{2-out} = t_{2-in} + \varepsilon_2 \left(\frac{CP_{min}}{CP1}\right)(T_{2-in} - t_{2-in})$$

$$t_{2-out} = 235.037 \ + 0.3858 \times 0.7394(300.437 - 235.037\ )$$

$$t_{2-out} = 253.693\ C$$

For stream 2

$$T_{2-out} = T_{2-in} - \varepsilon_2 \left(\frac{CP_{min}}{CP2}\right)(T_{2-in} - t_{2-in})$$

$$T_{2-out} = 300.437 - 0.3858(300.437 - 235.037)$$

$$T_{2-out} = 275.206\ C$$

15. Determine the final mixed outlet temperature for stream 1

$$t_{out} = \frac{2}{3} t_{1-out} + \frac{1}{3} t_{2-out}$$

$$t_{out} = \frac{2}{3}(265.415) + \frac{1}{3}(253.693) = 261.508\ C$$

Section 3

10. The configuration is the same as for *section 2*. Repeat *steps6* and *7* above

11. Calculate the effectiveness

$$N_{TU} = \frac{U_3 A_3}{CP_{min}}$$

$$N_{TU} = \frac{270.609 * 42.612}{18965.582} = 0.6080$$

$$ø = Exp(N_{TU}) = e^{0.6080} = 1.8368$$

$$\varepsilon = \frac{1}{\left[\frac{Rø^R}{ø^R - 1} + \frac{ø}{ø - 1} - \frac{1}{\ln ø}\right]}$$

$$\varepsilon_2 = \frac{1}{\left[\frac{0.7394 \times (1.8368)^{0.7394}}{(1.8368)^{0.7394} - 1} + \frac{1.8368}{1.8368 - 1} - \frac{1}{\ln(1.8368)}\right]}$$

$$\varepsilon_2 = 0.3858$$

12. Calculate the outlet temperatures

$$\frac{CP_{min}}{CP_1} = R = 0.7394$$

$$\frac{CP_{min}}{CP_2} = 1$$

$$t_{3-in} = t_{in} = 209.6\ C$$

$$T_{3-in} = T_{1-out} = 300.437\ C$$

For stream 1

$$t_{3-out} = t_{3-in} + \varepsilon_3 \left(\frac{CP_{min}}{CP1}\right)(T_{3-in} - t_{3-in})$$

$$t_{3-out} = 209.6\ + 0.3858 \times 0.7394(300.437 - 209.6\ )$$

$$t_{3-out} = 235.512\ C$$

For stream 2

$$T_{3-out} = T_{3-in} - \varepsilon_3 \left(\frac{CP_{min}}{CP2}\right)(T_{3-in} - t_{3-in})$$

$$T_{3-out} = 300.437 - 0.3858(300.437 - 209.6)$$

$$T_{3-out} = 265.392\ C$$

Section 4

13. The configuration is the same as for *section 1*. Repeat *steps1* and *2* above
14. Calculate the effectiveness

$$N_{TU} = \frac{U_4 A_4}{CP_{min}}$$

$$N_{TU} = \frac{270.596 * 85.223}{37931.164} = 0.6080$$

$$\emptyset = Exp(N_{TU}) = e^{0.6080} = 1.8368$$

$$\varepsilon = \frac{1}{\left[\frac{R\emptyset^R}{\emptyset^R - 1} + \frac{\emptyset}{\emptyset - 1} - \frac{1}{\ln \emptyset}\right]}$$

$$\varepsilon_2 = \frac{1}{\left[\frac{0.7398 \times (1.8368)^{0.7398}}{(1.8368)^{0.7398} - 1} + \frac{1.8368}{1.8368 - 1} - \frac{1}{\ln(1.8368)}\right]}$$

$$\varepsilon_2 = 0.3857$$

15. Determine the inlet temperature of stream 2 for *section 4* from the outlet temperatures from *section 2* and *section 3*.

$$T_{4-in} = \frac{1}{2} T_{3-out} + \frac{1}{2} T_{2-out}$$

$$T_{4-in} = \frac{1}{2}(265.392) + \frac{1}{2}(275.206)$$

$$T_{4-in} = 270.299\ C$$

16. Calculate the outlet temperatures

$$\frac{CP_{min}}{CP_1} = R = 0.7398$$

$$\frac{CP_{min}}{CP_2} = 1$$

$$t_{4-in} = t_{in} = 209.6\ C$$

For stream 1

$$t_{4-out} = t_{4-in} + \varepsilon_4 \left(\frac{CP_{min}}{CP1}\right)(T_{4-in} - t_{4-in})$$

$$t_{4-out} = 209.6\ + 0.3858 \times 0.7398(270.299 - 209.6\ )$$

$$t_{4-out} = 226.924\ C$$

For stream 2

$$T_{4-out} = T_{4-in} - \varepsilon_4 \left(\frac{CP_{min}}{CP2}\right)(T_{4-in} - t_{4-in})$$

$$T_{4-out} = 270.299\ - 0.3858(270.299 - 209.6\ )$$

$$T_{4-out} = 246.881\ C$$

17. Determine the final outlet temperature of stream 2

$$T_{out} = T_{4-out} = 246.9\ C$$

18. Determine the mixed temperature from *section 3* and *section 4* and compare with the set value of t_{34-out} under *step* 4 of *section 1*.

Set value of $t_{34-out} = 235.037\ C$

Newly calculated value of t_{34-out}

$$t_{34-out} = \frac{2}{3}t_{4-out} + \frac{1}{3}t_{3-out}$$

$$t_{34-out} = \frac{2}{3}(226.924) + \frac{1}{3}(235.512) = 229.787\ C$$

Newly calculated value $t_{34-out} = 229.787\ C$ is far less than the set value of$t_{34-out} = 235.037\ C$.

19. Reassign the value of $t_{34-out} = 229.787\ \ C$ with the new value calculated and repeat *steps1* to *18* until

$$Abs(Set\ t_{34-out} - Calculated\ t_{34-out}) < 0.001$$

Below are the values obtained from the iterations

Table B6

Iterations for the temperature profile for a 3-2 configuration

Runs	t_{34-out}	t_{in}	t_{out}	T_{in}	T_{out}
1	235.037	209.6	261.508	341.5	246.881
2	229.787	209.6	257.544	341.5	245.487
3	229.144	209.6	257.077	341.5	245.323
4	229.071	209.6	257.021	341.5	245.303
5	229.062	209.6	257.015	341.5	245.301
6	229.061	209.6	257.014	341.5	245.301
7	229.061				

Summary of the temperatures for each section after the iterations

Table B7
Temperature profile for a 3-2 configuration

Section	**Stream 1**		**Stream 2**	
	Inlet	*Outlet*	*Inlet*	*Outlet*
1	229.1	261.1	341.5	298.1
2	229.1	248.8	298.1	271.5
3	209.6	234.8	298.1	264.0
4	209.6	226.2	267.7	245.3

Unit stream final outlet temperatures

$$t_{out} = 257.0\ C$$

$$T_{out} = 245.3\ C$$

Diagram of the temperature distributions

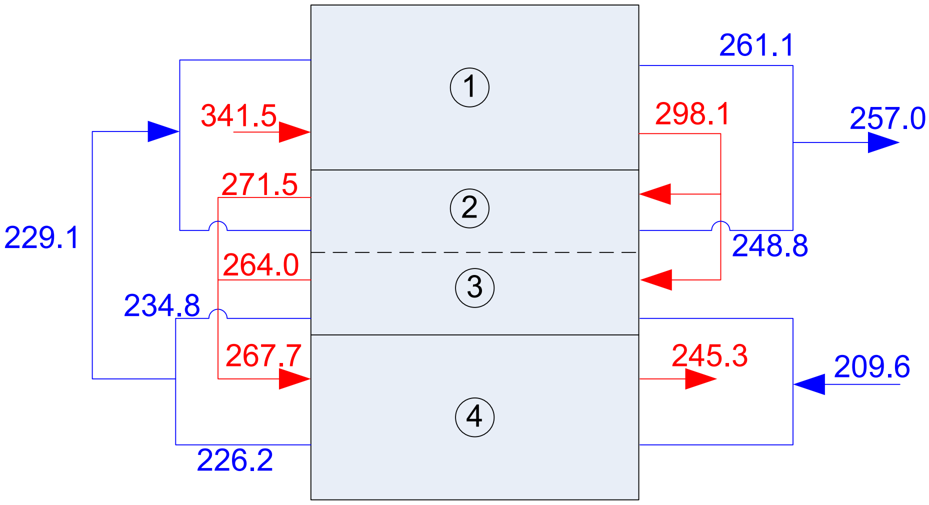
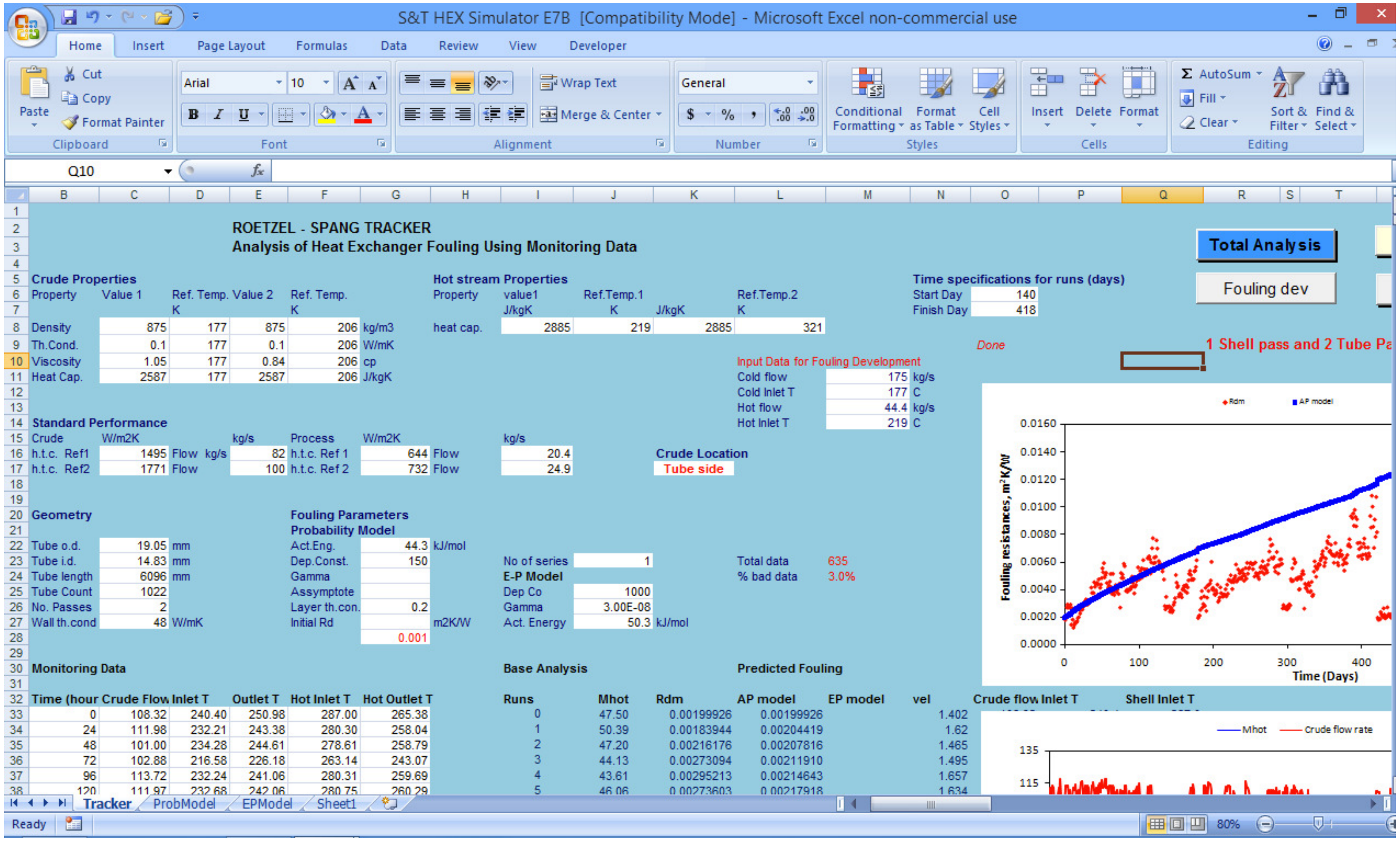


Figure B3: Temperature profile of the 3-2 configuration

APPENDIX C – INTERFACE OF PROGRAMS DEVELOPED

All the programs used for the analysis were developed in Macros – Visual Basic Applications (VBA) in Microsoft Excel 2003.

1. Program interface for shell and tube heat exchanger analysis – *Chapter 2*



ROETZEL - SPANG TRACKER
Analysis of Heat Exchanger Fouling Using Monitoring Data

Crude Properties

Property	Value 1	Ref. Temp. K	Value 2	Ref. Temp. K	
Density	875	177	875	206	kg/m3
Th.Cond.	0.1	177	0.1	206	W/mK
Viscosity	1.05	177	0.84	206	cp
Heat Cap.	2587	177	2587	206	J/kgK

Hot stream Properties

Property	value1 J/kgK	Ref.Temp.1 K	J/kgK	Ref.Temp.2 K
heat cap.	2885	219	2885	321

Time specifications for runs (days)

Start Day 140
Finish Day 418

Done

1 Shell pass and 2 Tube Pa

Input Data for Fouling Development
Cold flow 175 kg/s
Cold Inlet T 177 C
Hot flow 44.4 kg/s
Hot Inlet T 219 C

Standard Performance

Crude	W/m2K		kg/s	Process	W/m2K		kg/s
h.t.c. Ref1	1495	Flow kg/s	82	h.t.c. Ref 1	644	Flow	20.4
h.t.c. Ref2	1771	Flow	100	h.t.c. Ref 2	732	Flow	24.9

Crude Location
Tube side

Geometry

Tube o.d. 19.05 mm
Tube i.d. 14.83 mm
Tube length 6096 mm
Tube Count 1022
No. Passes 2
Wall th.cond 48 W/mK

Fouling Parameters
Probability Model
Act.Eng. 44.3 kJ/mol
Dep.Const. 150
Gamma
Assymptote
Layer th.con. 0.2
Initial Rd m2K/W
0.001

No of series 1
E-P Model
Dep Co 1000
Gamma 3.00E-08
Act. Energy 50.3 kJ/mol

Total data 635
% bad data 3.0%

Monitoring Data

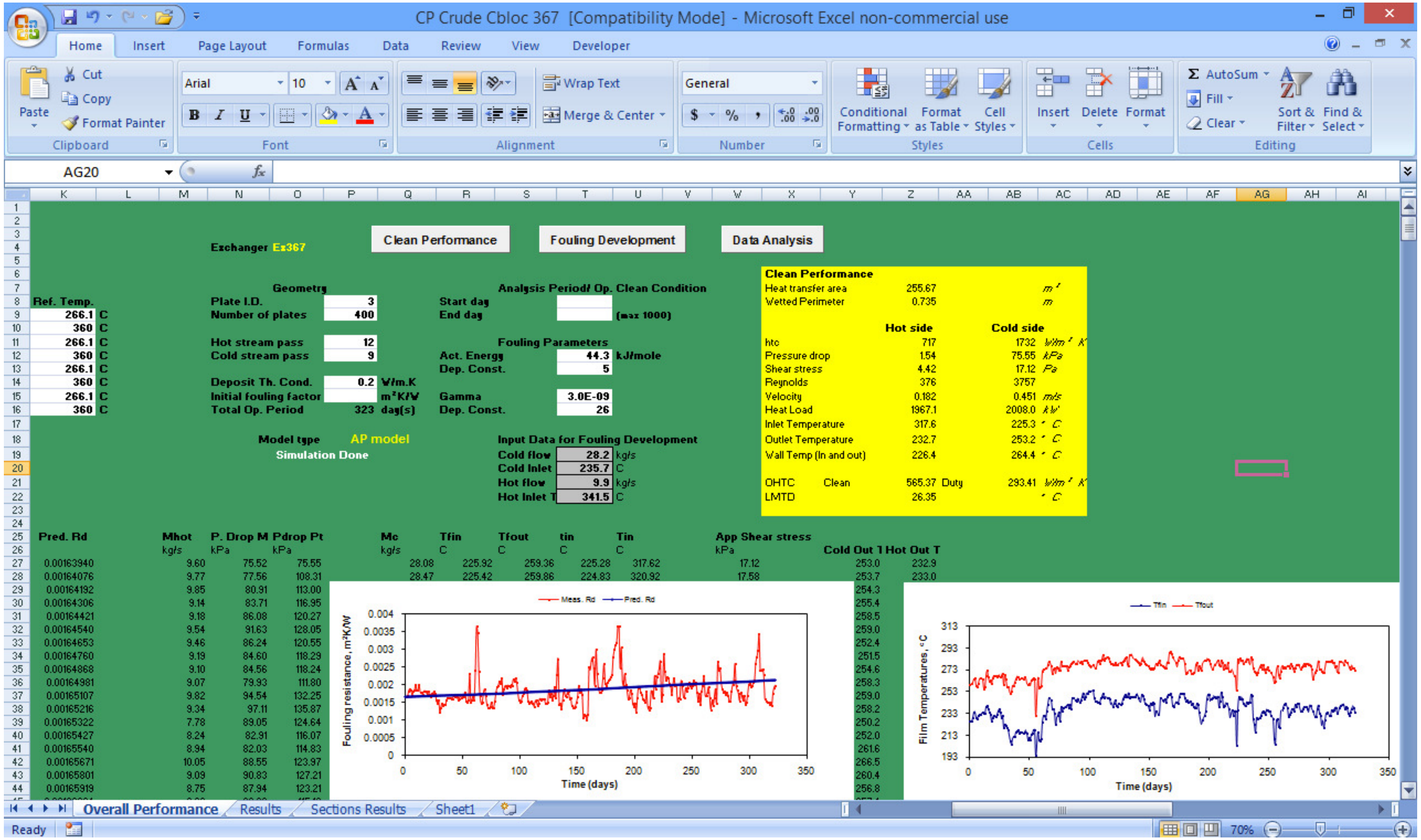
Time (hour	Crude Flow	Inlet T	Outlet T	Hot Inlet T	Hot Outlet T
0	108.32	240.40	250.98	287.00	265.38
24	111.98	232.21	243.38	280.30	258.04
48	101.00	234.28	244.61	278.61	258.79
72	102.88	216.58	226.18	263.14	243.07
96	113.72	232.24	241.06	280.31	259.69
120	111.97	232.68	242.06	280.75	260.29

Base Analysis **Predicted Fouling**

Runs	Mhot	Rdm	AP model	EP model	vel
0	47.50	0.00199926	0.00199926		1.402
1	50.39	0.00183944	0.00204419		1.62
2	47.20	0.00216176	0.00207816		1.465
3	44.13	0.00273094	0.00211910		1.495
4	43.61	0.00295213	0.00214643		1.657
5	46.06	0.00273603	0.00217918		1.634

Crude flow Inlet T Shell Inlet T

2. Program interface for Compabloc Performance analysis – *Chapter* 2 and *Chapter 4*



Exchanger Ex367

Ref. Temp.

266.1 C
360 C
266.1 C
360 C
266.1 C
360 C
266.1 C
360 C

Geometry

Plate I.D. 3
Number of plates 400

Hot stream pass 12
Cold stream pass 9

Deposit Th. Cond. 0.2 W/m.K
Initial fouling factor m²K/W
Total Op. Period 323 day(s)

Model type AP model
Simulation Done

Analysis Period/ Op. Clean Condition

Start day
End day (max 1000)

Fouling Parameters

Act. Energy 44.3 kJ/mole
Dep. Const. 5

Gamma 3.0E-09
Dep. Const. 26

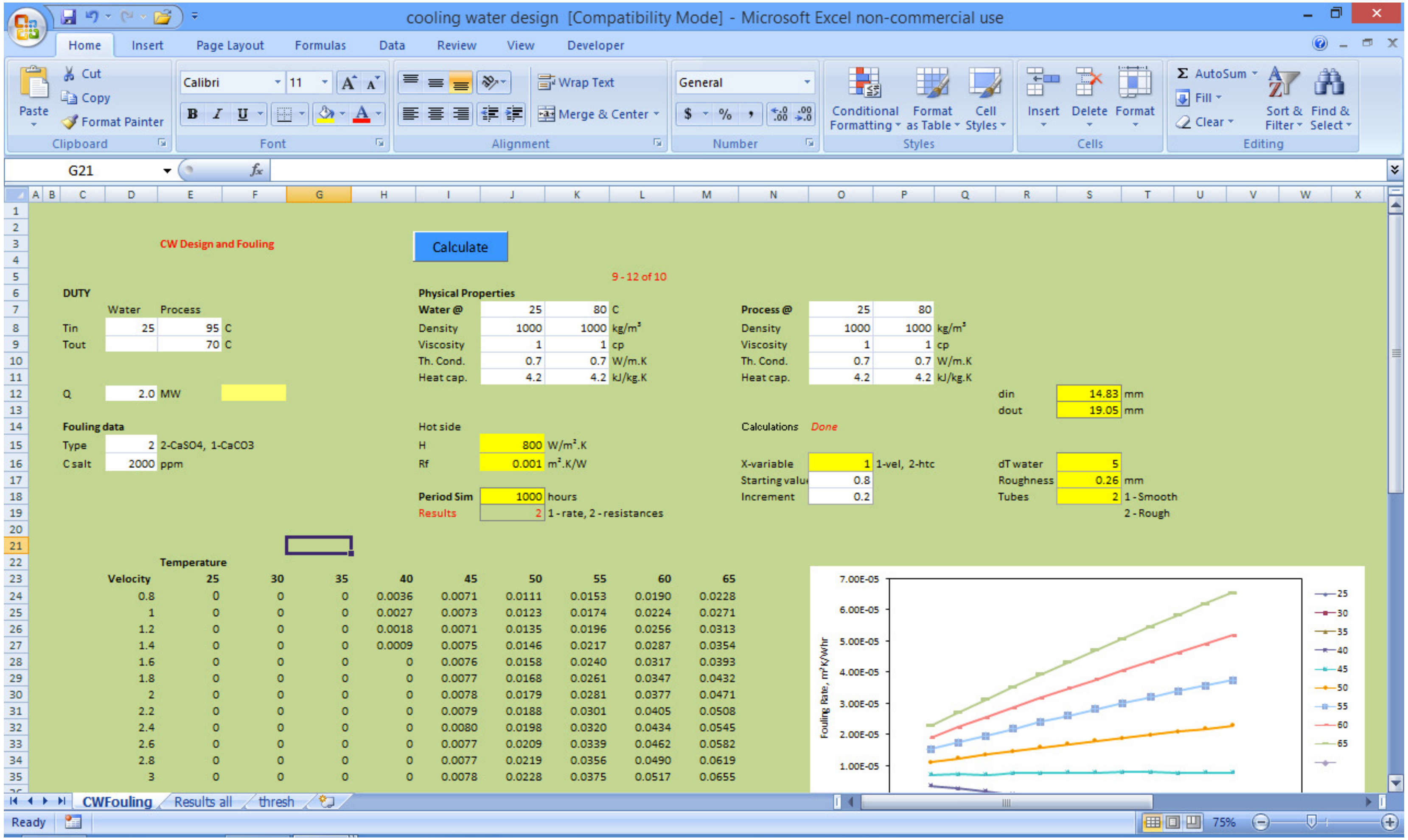
Input Data for Fouling Development

Cold flow 28.2 kg/s
Cold Inlet 235.7 C
Hot flow 9.9 kg/s
Hot Inlet T 341.5 C

Clean Performance			
Heat transfer area	255.67		m²
Wetted Perimeter	0.735		m
	Hot side	**Cold side**	
htc	717	1732	W/m² K
Pressure drop	1.54	75.55	kPa
Shear stress	4.42	17.12	Pa
Reynolds	376	3757	
Velocity	0.182	0.451	m/s
Heat Load	1967.1	2008.0	kW
Inlet Temperature	317.6	225.3	°C
Outlet Temperature	232.7	253.2	°C
Wall Temp (In and out)	226.4	264.4	°C
OHTC Clean	565.37	Duty 293.41	W/m² K
LMTD	26.35		°C

Pred. Rd	**Mhot** kg/s	**P. Drop M** kPa	**Pdrop Pt** kPa	**Mc** kg/s	**Tfin** C	**Tfout** C	**tin** C	**Tin** C	**App Shear stress** kPa	**Cold Out T**	**Hot Out T**
0.00163940	9.60	75.52	75.55	28.08	225.92	259.36	225.28	317.62	17.12	253.0	232.9
0.00164076	9.77	77.56	108.31	28.47	225.42	259.86	224.83	320.92	17.58	253.7	233.0
0.00164192	9.85	80.91	113.00							254.3	
0.00164306	9.14	83.71	116.95							255.4	
0.00164421	9.18	86.08	120.27							258.5	
0.00164540	9.54	91.63	128.05							259.0	
0.00164653	9.46	86.24	120.55							252.4	
0.00164760	9.19	84.60	118.29							251.5	
0.00164868	9.10	84.56	118.24							254.6	
0.00164981	9.07	79.93	111.80							258.3	
0.00165107	9.82	94.54	132.25							259.0	
0.00165216	9.34	97.11	135.87							258.2	
0.00165322	7.78	89.05	124.64							250.2	
0.00165427	8.24	82.91	116.07							252.0	
0.00165540	8.94	82.03	114.83							261.6	
0.00165671	10.05	88.55	123.97							266.5	
0.00165801	9.09	90.83	127.21							260.4	
0.00165919	8.75	87.94	123.21							256.8	

3. Program interface for Heat exchanger Design considerations – *Chapter 5*



CW Design and Fouling

DUTY

	Water	Process	
Tin	25	95	C
Tout		70	C

Q 2.0 MW

Fouling data

Type 2 2-CaSO4, 1-CaCO3

C salt 2000 ppm

9 - 12 of 10

Physical Properties

Water @	25	80	C
Density	1000	1000	kg/m³
Viscosity	1	1	cp
Th. Cond.	0.7	0.7	W/m.K
Heat cap.	4.2	4.2	kJ/kg.K

Process @	25	80	
Density	1000	1000	kg/m³
Viscosity	1	1	cp
Th. Cond.	0.7	0.7	W/m.K
Heat cap.	4.2	4.2	kJ/kg.K

din 14.83 mm

dout 19.05 mm

Hot side

H 800 W/m².K

Rf 0.001 m².K/W

Calculations Done

X-variable 1 1-vel, 2-htc

Starting valu 0.8

Increment 0.2

dT water 5

Roughness 0.26 mm

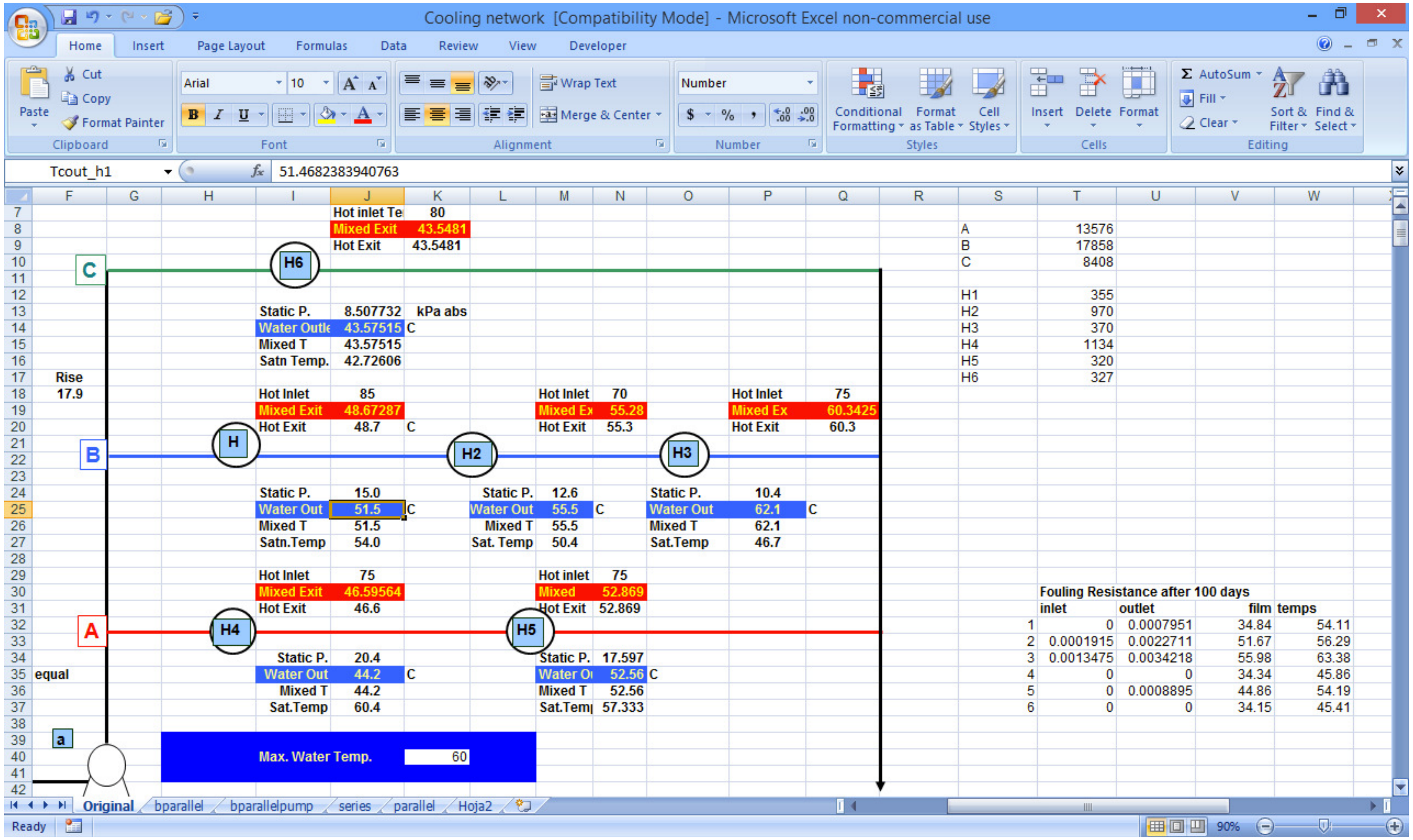
Tubes 2 1 - Smooth
2 - Rough

Period Sim 1000 hours

Results 2 1 - rate, 2 - resistances

	Temperature								
Velocity	25	30	35	40	45	50	55	60	65
0.8	0	0	0	0.0036	0.0071	0.0111	0.0153	0.0190	0.0228
1	0	0	0	0.0027	0.0073	0.0123	0.0174	0.0224	0.0271
1.2	0	0	0	0.0018	0.0071	0.0135	0.0196	0.0256	0.0313
1.4	0	0	0	0.0009	0.0075	0.0146	0.0217	0.0287	0.0354
1.6	0	0	0	0	0.0076	0.0158	0.0240	0.0317	0.0393
1.8	0	0	0	0	0.0077	0.0168	0.0261	0.0347	0.0432
2	0	0	0	0	0.0078	0.0179	0.0281	0.0377	0.0471
2.2	0	0	0	0	0.0079	0.0188	0.0301	0.0405	0.0508
2.4	0	0	0	0	0.0080	0.0198	0.0320	0.0434	0.0545
2.6	0	0	0	0	0.0077	0.0209	0.0339	0.0462	0.0582
2.8	0	0	0	0	0.0077	0.0219	0.0356	0.0490	0.0619
3	0	0	0	0	0.0078	0.0228	0.0375	0.0517	0.0655

4. Program Interface for Cooling water network revamp analysis [73] – *Chapter 6*



Tcout_h1 51.4682383940763

Hot inlet Te 80
Mixed Exit 43.5481
Hot Exit 43.5481

Static P. 8.507732 kPa abs
Water Outle 43.57515 C
Mixed T 43.57515
Satn Temp. 42.72606

Rise
17.9

Hot Inlet 85
Mixed Exit 48.67287
Hot Exit 48.7 C

Hot Inlet 70
Mixed Ex 55.28
Hot Exit 55.3

Hot Inlet 75
Mixed Ex 60.3425
Hot Exit 60.3

Static P. 15.0
Water Out 51.5 C
Mixed T 51.5
Satn.Temp 54.0

Static P. 12.6
Water Out 55.5 C
Mixed T 55.5
Sat. Temp 50.4

Static P. 10.4
Water Out 62.1 C
Mixed T 62.1
Sat.Temp 46.7

Hot Inlet 75
Mixed Exit 46.59564
Hot Exit 46.6

Hot inlet 75
Mixed 52.869
Hot Exit 52.869

Static P. 20.4
Water Out 44.2 C
Mixed T 44.2
Sat.Temp 60.4

Static P. 17.597
Water Ou 52.56 C
Mixed T 52.56
Sat.Temp 57.333

equal

Max. Water Temp. 60

A	13576
B	17858
C	8408
H1	355
H2	970
H3	370
H4	1134
H5	320
H6	327

Fouling Resistance after 100 days

	inlet	outlet	film temps	
1	0	0.0007951	34.84	54.11
2	0.0001915	0.0022711	51.67	56.29
3	0.0013475	0.0034218	55.98	63.38
4	0	0	34.34	45.86
5	0	0.0008895	44.86	54.19
6	0	0	34.15	45.41