Fouling Model for Optimization of Cleaning Schedule of Industrial Heat Exchanger Networks

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Abstract — **Availability of reliable fouling models is important for the accurate simulation of heat exchanger networks (HENs) influenced by fouling. The fouling models play an important role in the optimization of cleaning schedule of HEN. A model with the least number of parameters and that requires minimum computational load is preferred as compared to the theoretical or semi-theoretical models in the cleaning schedule optimization problem. This paper presents the development of empirical models for an industrial HEN using the plant operating data. The fouling models are developed and shown to describe the fouling resistance over time in a crude preheat train with reasonable accuracy.**

Keywords:
Cleaning schedule optimization, Crude preheat train, Fouling, Heat exchanger network, Modeling.

I. Introduction

Fouling in heat exchangers is an undesirable process that reduces the realization of the maximum benefits of heat integration in process industries. Heat exchangers operating under fouled conditions experience reduced thermal efficiencies and increased pressure drops [1]. Fouling, especially, in large heat exchanger networks (HENs) affects the delicate balance of heat integration and reduces the heat recovery considerably.

Fouling occurs in many process industries. Crude oil fouling in crude preheat train (CPT) of a petroleum refinery is one of the major problems encountered by the refineries with considerable economic losses. For a refinery processing 100,000 bbl/day, a drop of 1 °C in the coil inlet temperature (CIT) due to fouling results in approximately £25,000 of additional fuel cost and 750te of additional carbon dioxide emission each year [2]. On both economic and environmental basis, process industries are highly motivated to minimize fouling. Fouling cannot be avoided, yet it can be mitigated through the implementation of effective fouling mitigation techniques [1].

Major fouling mitigation techniques include the use of antifoulant chemicals, designing of more efficient heat exchangers and periodic cleaning of heat exchangers. Besides their capability in mitigating the fouling to certain extent, the techniques have their own drawbacks. Adding antifoulant chemicals may lessen the fouling, yet it increases the operating cost of the plant. Retrofitting heat exchangers, with greater size and efficiency can also overcome the problem of fouling; nevertheless the capital cost of the HEN also increases [1, 3, 4].

In general, periodic cleaning of heat exchangers is required to recover the lost heat transfer efficiencies. HENs with no fouling mitigation efforts may require frequent cleanings. It may be noted that the HENs which employ fouling mitigation techniques such as the use of antifoulant chemicals also require periodic cleaning but at lesser frequencies. Periodic cleaning affects the plant economy as some penalties have to be incurred when the heat exchangers are cleaned. Higher cleaning cost would be incurred when the heat exchanger is cleaned too frequently while less frequent cleaning lead to increased heat loss. An optimum cleaning schedule is, therefore, necessary for each heat exchanger or HEN in order to optimize the economic loss due to the effects of fouling.

The HEN cleaning schedule optimization problem involves three major components, *viz.*, a HEN simulation model, reasonably accurate fouling models and an efficient optimization tool.

The task of developing the simulation models for any given configuration of HEN has been made simple with the availability of several process simulation software packages such as Aspen, Hysys, Petrosim, etc. Several theoretical, semi-empirical and empirical models have been proposed to describe heat exchanger fouling in the literature [5]. The use of theoretical fouling models in the HEN simulation has been rather very restrictive due to the many unknown constants in the models [6].

Following the Ebert-Panchal threshold fouling model [7], several semi-empirical fouling models have been proposed in the literature [8-10]. These models are mainly employed in determining the operating conditions that result in zero or minimum fouling rates, especially during the design stage or during retrofits. For example, the software package, EXPRESS™ by ESDU uses the threshold fouling models for design, rating, revamping or retrofitting shell and tube heat exchangers [11].

Typically, empirical fouling models were developed and used in the optimization of cleaning schedules for the HENs [1, 3, 12]. In literature, linear, asymptotic (exponential) and falling rate fouling models were utilized for the purpose of optimization of heat exchanger cleaning schedule [1, 3, 12]. The use of threshold fouling models for the optimal cleaning schedule determination has been very limited [13].

In many heat exchangers, often fouling occurs after a certain period of time which is called the initiation period for fouling. Most of the empirical models ignore the initiation period because it is difficult to predict [5]. Models which include the initiation period may lead to a better estimate of optimal cleaning periods. An attempt has been made in this paper to develop empirical fouling models which include the initiation period where necessary and thereby more accurate fouling models.

A brief description of the industrial HEN used in this study is included in Section II. Section III describes the data collection, reconciliation, the extraction and interpretation of fouling characteristics. Fouling model development and results & discussions are provided in Sections IV and V, respectively.

II. Crude Preheat Train

The HEN under the present study is a crude preheat train (CPT) in a petroleum refinery. The CPT consists of 11 heat exchangers with each of which containing multiple shells that operate in series and/or parallel modes. The schematic diagram of the CPT is shown in Fig. 1.

The operational data from the plant historian in the Distributed Control System (DCS) is collected for a period of four years following a turn-around operation. The heat exchangers are assumed to be clean after the turn-around operation. The data collected from the plant historian are mainly the daily averaged values of inlet and outlet temperatures and flow rates of crude oil and product and pump-around streams (heating mediums). Physical properties of the crude oil at various temperature conditions and the product streams required for performing energy balance calculations and for the determination of individual film heat transfer coefficients are estimated using the property packages in Petrosim® software. In addition, the design data of all the heat exchangers in the CPT were also collected.

III. Heat Exchanger Fouling Calculations

Generally, the averaged data collected from an operating plant do not satisfy the steady-state mass and energy balance equations. The data contain measurements from various instruments with varying instrumental error and bias.

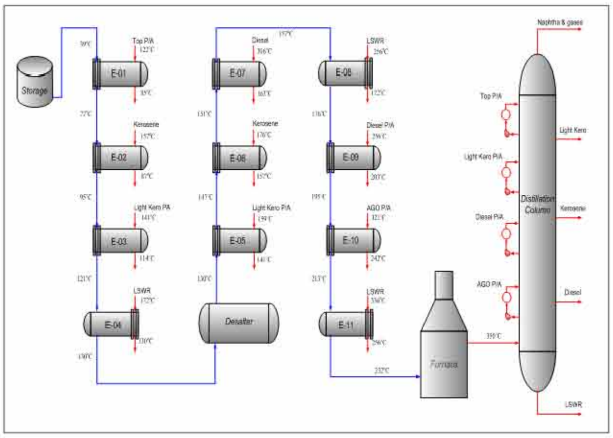

Figure 1. The schematic diagram of the CPT

Prior to utilizing the data for further analysis, the data need to be processed. This step includes recovering missing data; removing outliers and data reconciliation. Outliers were identified using principal component analysis (PCA) and removed. The data reconciliation is accomplished to obey the mass and energy balance equations.

Together with physical properties data, the reconciled data is subsequently used to calculate the required parameters, namely: tube-side heat transfer coefficient, h_i (W/m^2 K), shell-side heat transfer coefficient, h_o (W/m^2 K), overall heat transfer coefficient, U (W/m^2 K), and fouling resistance, R_f (m^{2}K/W). The calculations are carried out using established methods as described below.

The heat exchanger performance is evaluated based on the actual overall heat transfer coefficient. From the measured data, the actual overall heat transfer coefficient is given by:

$$Q_i = U_i A_i F_i \Delta T_{lm,i} \qquad \text{for } i = 1, 2, ..., n \tag{1}$$

where
- A = heat transfer area (m^2)
- F = LMTD correction factor
- i = heat exchanger number
- ΔT_{lm} = Log Mean Temperature Difference (LMTD) (°C)
- n = number of heat exchangers in HEN
- Q = heat duty (W)

Heat duty of the i^{th} heat exchanger can be calculated using either the hot- or cold-side fluid as given in (2) and (3).

$$Q_{c,i} = m_c c_{p,c}(T^c_{out,i} - T^c_{in,i}) \qquad \text{for } i = 1, 2, ..., n \tag{2}$$

$$Q_{h,i} = m_h c_{p,h}(T^h_{in,i} - T^h_{out,i}) \qquad \text{for } i = 1, 2, ..., n \tag{3}$$

where
- $Q_{h,i}$, $Q_{c,i}$ = heat duties of hot and cold fluids the i^{th} heat exchanger for (W), respectively
- $c_{p,h}$, $c_{p,c}$ = specific heat capacities of hot and cold fluids (J/kg K)
- $T^h_{out,i}$, $T^c_{out,i}$ = outlet temperatures of hot and cold fluids the for i^{th} heat exchanger (°C)
- $T^h_{in,i}$, $T^c_{in,i}$ = inlet temperatures of hot and cold fluids the for i^{th} heat exchanger (°C)
- m_h, m_c = mass flow rates of hot and cold fluids (kg/s)

Under the steady-state conditions, the heat duties calculated based on the cold- and hot-side fluids are equal:

$$Q_{c,i} = Q_{h,i} \qquad \text{for } i = 1, 2, ..., n \tag{4}$$

This condition is used while reconciling the measurements around each heat exchanger in the CPT. Once the measurements are reconciled to obey the energy balance equations, the actual overall heat transfer coefficient can be determined for each heat exchanger using (1).

An expression relating the overall heat transfer coefficient with the individual film heat transfer coefficients and fouling resistances is given by:

$$\frac{1}{U}=\frac{d_o}{d_i h_i}+\frac{d_o R_{f,i}}{d_i}+\frac{d_o \ln\left(\frac{d_o}{d_i}\right)}{2k_w}+R_{f,o}+\frac{1}{h_o} \tag{5}$$

where d_i = inside diameter (m)
d_o = outside diameter (m)
$R_{f,i}$ = inside fouling resistance (m^2K/W)
$R_{f,o}$ = outside fouling resistance (m^2K/W)
k_w = thermal conductivity of wall (W/m K)

In this study, it is assumed that fouling occurs only in the tube-side of the heat exchanger and therefore, the $R_{f,o}$ is considered to be constant. If the heat exchanger is very new, the value of $R_{f,o}$ may be zero, but in older heat exchangers, $R_{f,o}$ may take a finite value to account for the loss in thermal efficiency of the heat exchanger due to its aging and repeated cleanings. The values of initial fouling resistances on either side are determined from the data that follows a cleaning operation. The inside fouling resistance which undergoes a change over time due to fouling is determined using the calculated actual overall heat transfer coefficient and the individual heat transfer coefficients determined as explained below.

The individual film heat transfer coefficients depend on the physical properties of the fluid and the operating conditions. In a refinery operation, the plant throughput and the crude blend vary considerably from day to day leading to different values of film heat transfer coefficients. The individual heat transfer coefficients are generally estimated using empirical correlations reported in literature. The following sub sections describe the estimation of film heat transfer coefficients based on the daily operating conditions and crude blend properties.

The values of film heat transfer coefficients given in the heat exchanger data sheet was considered to be the base value for the design operating conditions and the variations in the tube-side heat transfer coefficients due to the changes in the flow rates and fluid properties are estimated using the empirical correlations.

A. *Tube-Side Film Heat Transfer Coefficient*

The film heat transfer coefficient on the tube inside is calculated using the Colburn equation as given by [14, 15]:

$$h_t = 0.023\,\mathrm{Re}_t^{0.8}\,\mathrm{Pr}_t^{0.4}\left(\frac{\mu_s}{\mu_w}\right)^{0.14}\left(\frac{k}{d_i}\right) \tag{6}$$

where h_t = tube-side heat transfer coefficient (W/m^2 K)
Pr_t = tube-side Prandtl number
Re_t = tube-side Reynolds number
μ_s, μ_w = dynamic viscosity at the bulk and tube wall temperatures (Pa s)
k = thermal conductivity of fluid (W/m K)

The tube-side individual heat transfer coefficient is calculated using (6) as a correction to the variations in the operating condition and fluid properties as

$$\frac{h_{ta}}{h_{td}}=\left(\frac{\dot{V}_a\rho_a}{\dot{V}_d\rho_d}\right)^{0.8}\left(\frac{\mu_a}{\mu_d}\right)^{-0.4}\left(\frac{c_{pa}}{c_{pd}}\right)^{0.4}\left(\frac{k_a}{k_d}\right)^{0.6} \tag{7}$$

where c_{pa}, c_{pd} = specific heat capacities of tube side fluid at actual and design conditions (J/kg K)
ρ_a, ρ_d = densities of tube side fluid at actual and design conditions (kg/m^3)
h_{ta}, h_{td} = tube-side heat transfer coefficients at actual and design conditions (W/m^2 K)
k_a, k_d = thermal conductivities of tube side fluid at actual and design conditions (W/m K)
μ_a, μ_d = dynamic viscosities of tube side fluid at actual and design conditions (Pa s)
$\dot{V}_a$, $\dot{V}_d$ = volumetric flow rates of tube side fluid at actual and design conditions (m^3/hr)

B. *Shell-Side Film Heat Transfer Coefficient*

For heat exchangers with segmental baffle, Bell-Delaware method was found to estimate the heat transfer coefficients closer to the values given by the heat exchanger fabricators.

In the Bell-Delaware method, the flow fraction for each stream is found by knowing the corresponding flow areas and flow resistances. The heat transfer coefficient for ideal cross flow is then modified for the presence of each stream through correction factors. The shell-side heat-transfer coefficient, h_s, is given by [16, 17] as

$$h_s = h_i J_c J_l J_s J_b J_r \tag{8}$$

where h_i is the heat transfer coefficient for pure cross flow of an ideal tube bank; J_c is the correction factor for baffle cut and spacing; this correction factor is used to express the effects of the baffle window flow on the shell-side ideal heat-transfer coefficient h_i which is based on cross flow; J_l is the correction factor for baffle leakage effects, including both shell-to-baffle and tube-to-baffle leakage; J_s is the correction factor for the bundle bypass flow; J_b is the correction factor for variable baffle spacing in the inlet and outlet sections; J_r is the correction factor for adverse temperature gradient buildup in laminar flow. In this study, all the correction factors are assumed to be constant.

In order to calculate the ideal heat transfer coefficient, h_i, calculation of the shell-side mass velocity, G_s, shell-side Reynolds number, Re_s, and shell-side Prandtl number, Pr_s, must be performed, and the ideal heat transfer coefficient is calculated using (9).

$$h_i=\frac{j_i C_{p,s} G_s\left(\frac{\mu_s}{\mu_w}\right)^{0.14}}{\mathrm{Pr}_s^{2/3}} \tag{9}$$

In (9), the term j_i is the ideal Colburn j factor for the shell-side and can be determined from the appropriate Bell-Delaware curve for the tube layout and pitch.

Similar to the calculation of variations in the tube-side heat transfer coefficient, the variations in the shell-side heat transfer coefficient for the clean conditions is calculated for the changes in the fluid flow rate and properties as shown in (10).

$$\frac{h_{sa}}{h_{sd}} = \left(\frac{\dot{V}_a \rho_a}{\dot{V}_d \rho_d}\right)^{0.65} \left(\frac{\mu_a}{\mu_d}\right)^{-0.32} \left(\frac{c_{p\,a}}{c_{p\,d}}\right)^{1/3} \left(\frac{k_a}{k_d}\right)^{2/3}$$ **Error! Bookmark not defined.** (10)

where h_{sa}, h_{sd} are the actual and design shell-side heat transfer coefficients (W/m^{2}K).

For heat exchangers with helical baffle the following equation is proposed by Peng [18].

$$h_s = C\ \mathrm{Re}_s^m \mathrm{Pr}_s^{1/3} \frac{k}{d_o} \tag{11}$$

where C and m are constants. The variations in the shell-side heat transfer coefficient are calculated by (12).

$$\frac{h_{sa}}{h_{sd}} = \left(\frac{\dot{V}_a \rho_a}{\dot{V}_d \rho_d}\right)^{0.7} \left(\frac{\mu_a}{\mu_d}\right)^{-0.4} \left(\frac{c_{p\,a}}{c_{p\,d}}\right)^{1/3} \left(\frac{k_a}{k_d}\right)^{2/3} \tag{12}$$

As mentioned earlier that an older heat exchanger is not restored to the conditions of a new heat exchanger after each cleaning. This results in a lower heat transfer performance and the actual condition of the heat exchanger in terms of its thermal performance after cleaning must be captured in the model in order to get the calculated $R_{f,i}$ values to represent only the fouling that occur after the cleaning.

In order to determine the equivalent initial fouling resistances to represent the loss in thermal efficiencies, a rigorous heat exchanger simulation model is developed in Petrosim® software and configured to the actual plant configuration in terms of number of shells and parallel/series operation. The values of initial $R_{f,i}$ and $R_{f,o}$ values are adjusted by trial and error method until the predicted outlet temperatures from each heat exchanger matches with the plant data.

The overall heat transfer coefficient under clean conditions, U_c, is calculated with the corrected values of film heat transfer coefficients, as given by (7) and (10) or (12) and is given by

$$\frac{1}{U_c} = \frac{d_o}{d_i h_i} + \frac{d_o R_{f,i}}{d_i} + \frac{d_o \ln\left(\frac{d_o}{d_i}\right)}{2k_w} + R_{f,o} + \frac{1}{h_o} \tag{13}$$

Please note that the $R_{f,i}$ and $R_{f,o}$ values used in (13) are the initial values as determined before and kept constant. The fouling resistance at any given instant in time, t, is then determined by

$$R_{f,i}(t) = \frac{1}{U(t)} - \frac{1}{U_c(t)} \tag{14}$$

The fouling resistance profile with time for each heat exchanger in the HEN is determined through the calculations explained in (13) and (14).

IV. Fouling Modeling

Refineries are generally operated with varying crude oils and crude blends which varies in their properties very much. Ignoring the physical and chemical properties of the crude oils, the fouling characteristics are modeled through simple empirical models such as linear, falling rate and exponential models (15-17) [1,3,9]. In this research, sigmoidal equation (18) is proposed and utilized to characterize the fouling growth behavior as observed from the characteristics curves for certain heat exchangers, as seen in Fig. 4.

Linear [1]: $$R_f(t) = at \tag{15}$$

Falling rate [13]: $$R_f(t) = a \cdot \ln(t) - b \tag{16}$$

Exponential [3]: $$R_f(t) = a(t - e^{-bt}) \tag{17}$$

Sigmoidal: $$R_f(t) = \frac{a}{1 + e^{-\left(\frac{t - t_o}{b}\right)}} \tag{18}$$

Appropriate relationship is chosen based on the observed fouling characteristics for each heat exchanger in the CPT and the model parameters are estimated through nonlinear regression. The model form used and the respective model parameters are summarized in Table I.

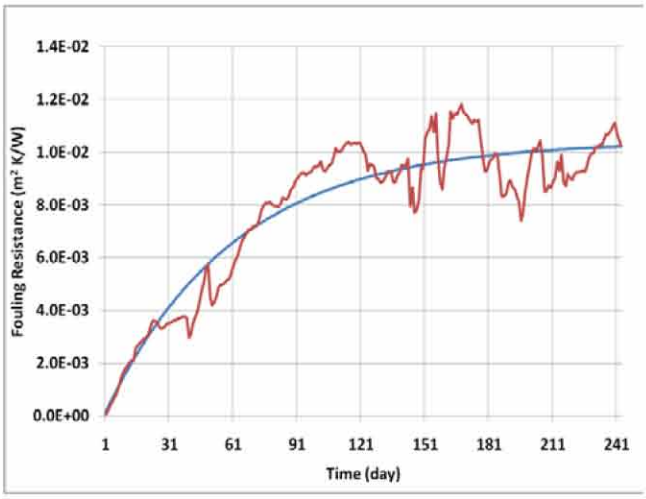

Figure 2. Fouling characteristics of heat exchanger E-03

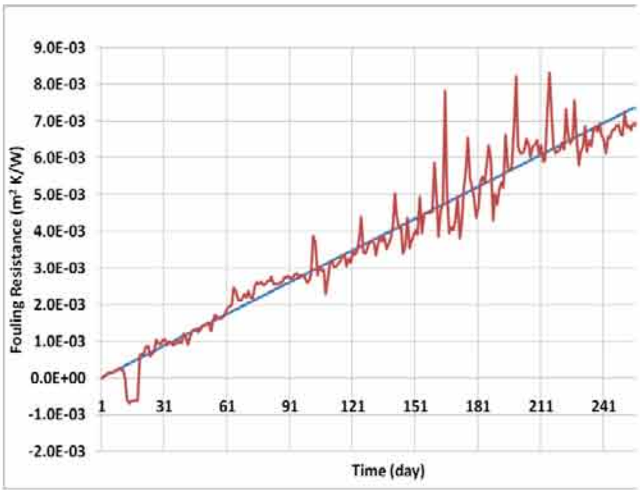

Figure 3. Fouling characteristics of heat exchanger E-06

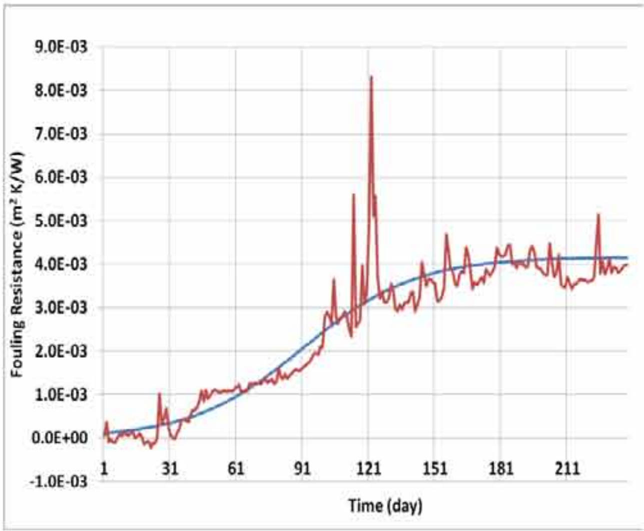

Figure 4. Fouling characteristics of heat exchanger E-07

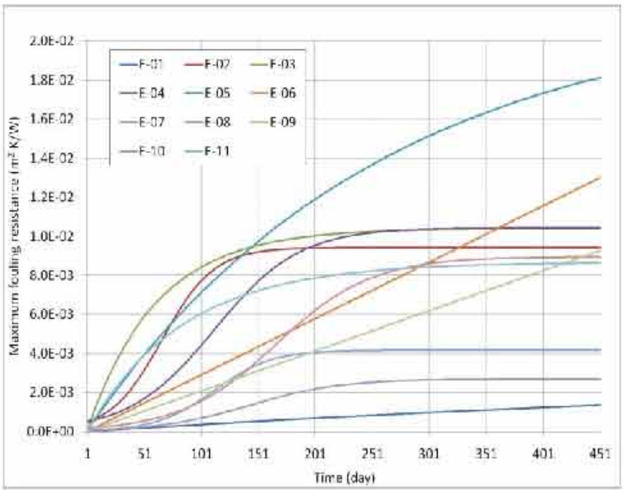

Figure 5. Long range fouling prediction for the heat exchangers in the CPT

TABLE I
FOULING MODEL TYPE AND VALUES OF THE MODEL PARAMETERS OF HEAT EXCHANGERS IN THE CPT

Heat Exc. No.	Model type	Model Parameters			R^2
		A	B	t_0	
E-01	Exp.	3.69E-03	1.01E-03	---	0.882
E-02	Sig.	9.41E-03	22.63	66	0.980
E-03	Exp.	1.04E-02	1.65E-02	---	0.942
E-04	Sig.	1.04E-02	37.26	112	0.965
E-05	Exp.	2.19E-02	3.90E-03	---	0.981
E-06	Linear	2.89E-05	---	---	0.973
E-07	Sig.	4.17E-03	25.40	111	0.899
E-08	Sig.	8.94E-03	42.11	166	0.981
E-09	Linear	2.05E-05	---	---	0.986
E-10	Sig.	2.69E-03	39.68	143.5	0.993
E-11	Exp.	8.67E-03	1.19E-02	---	0.926

V. RESULTS AND DISCUSSIONS

Fouling characteristics of the individual heat exchangers in the CPT differs very much from each other. It is observed that heat exchangers E-06 and E-09 follow linear fouling characteristics. The fouling rate in heat exchanger E-06 is slightly larger than that of E-09. Heat exchangers E-01, E-03, E-05 and E-11 follow exponential fouling growth behavior. The other heat exchangers follow sigmoidal fouling growth behavior. The sigmoid fouling growth behavior indicates that there is an initiation period for fouling. Different fouling behavior among the heat exchangers processing the same crude oil indicates that the operating conditions, namely, the velocity and the bulk and surface temperatures play a major role in the fouling behavior.

From Table I, it is observed that the chosen model forms closely describe the actual fouling characteristics which are indicated by the R^2 values of 0.88 to 0.99. It may be noted that fluctuations in industrial data are quite normal and values of R^2 of 0.8 and above can be considered to be very good fits.

Using the developed fouling models, long range fouling prediction for all heat exchangers in the CPT is obtained as illustrated in Fig. 5.

VI. CONCLUSIONS

An industrial heat exchanger network, namely, the crude preheat train in a petroleum refinery was chosen. The fouling characteristics for each heat exchanger has been investigated and extracted from the operating plant data. Appropriate empirical fouling models were chosen and the model parameters are estimated and reported. High values of coefficients of regression, R2, indicate that the developed models closely follow the actual fouling behavior in the plant. These empirical models will become part of a cleaning schedule optimization problem.

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