Thermohydraulic Analysis of Heat Exchanger Cleaning

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Abstract
Considerable research efforts have been made to diminish fouling problems in industrial plants. Fouling effects in heat exchanger comprehends energy losses which cause more fuel consumption with, consequently, more operational costs and an increase in carbon dioxide emissions. Additionally, if the hydraulic resistance increases too much due to fouling, the thermal equipment might be partially bypassed in order to keep throughput. The careful analysis of both effects can indicate an optimal schedule which can improve process profitability. Few works concerning the optimization of cleaning actions focus on the possible hydraulic problems. Hydraulic limitations can affect significantly the plant operational condition and should be considered including the presence of bypasses which avoids a throughput decrease. This paper presents an optimization analysis of cleaning schedule of critical heat exchangers in an industrial plant, exploring thermal and fluid dynamic aspects. The optimization problem is solved using a simulated annealing algorithm where the decision variables are the cleaning instants and the control valve opening fractions.

Keywords: Fouling, Heat Exchangers, Optimization, Cleaning Schedules.

1. Introduction
Many industrial plants are concerned in how to process their raw material with minimum energy consumption once it can significantly affect industrial profitability and carbon dioxide emissions. The world is going to an age where carbon dioxide emissions not only should be reduced but must be reduced. For example, analyzing the operations of an oil refinery, it can be seen that the distillation unit consumes a considerable fraction of total energy of the plant. In order to diminishes this consumption a preheat train is used, so hot product streams and pumparounds from the distillation column preheats the crude oil stream. However, the final heat is still done in a fired heater with fuel burning. During operation, the effectiveness of the crude preheat train decreases as a consequence of fouling in the heat exchangers and there is a significant increase of energy consumption in the furnace. Assuming fuel gas burning at the furnace the impact of fouling in a five years operation of a large refinery (360,000 bpd) can be of about 6 million US$ just considering the fuel gas price. The average emission of CO₂ per barrel of crude oil due to fouling might be about 70,000 t/year. Because of its importance and deleterious effects (diminishes the overall heat transfer coefficient and heat transfer rate, also increasing the magnitude of flow resist), fouling is considered the major unsolved problem in heat transfer technology.

The effects on an industrial plant might be not only energy loss with consequent more fuel consumption (and carbon emissions), but also an increase of cleaning and maintenance costs. Additionally, if a throughput cannot be sustained due to the increase of hydraulic resistance, it is common in industrial practice to partially bypass the thermal equipment in order to avoid a production loss. Nevertheless, in this case, the energy integration is compromised.

Because of these aspects, in order to mitigate fouling, the optimization of the cleaning schedule started to be studied ([1-5]). The literature approaches, however, only focus on thermal analysis, despite of possible hydraulic effects. In reality, hydraulic aspects can affect significantly the plant operational condition and should be considered ([6-9]). In several process plant problems, the stream flow rates may be constrained by the hydraulic facility limitations and the conventional HEN analysis applied to these systems may imply considerable prediction errors.

Previous works ([7-8]) showed that ignoring the effect of fouling on the hydraulic system the best cleaning instant might be mispredicted and an increase of energy consumption equal to about 28% might occur. This paper presents an optimization analysis of cleaning schedule of critical heat exchangers in an industrial plant exploring thermal and fluid dynamic aspects. Using simulation routines capable to predict the heat exchanger performance subjected to fouling, during a certain time horizon, an optimization algorithm is applied to the
problem to determine the future instants when the exchanger must be cleaned. The optimization problem is solved using both a simulated annealing algorithm and a simplex algorithm where the decision variables are the cleaning instants and the control valve opening fractions.

2. Methodology
In literature, the basic formulation for the heat exchanger network simulation in stationary state comprehends the determination of network temperatures based on algebraic models with mass and energy balances associated with heat exchanger equation. However, in many problems the flow might be restricted by hydraulic limitation. The mathematical structural of the system is composed by two interconnected layers: optimization and simulation. This structure can imply slower optimization convergence rates but has the important advantage to provide robustness for the computational search.

2.1. Problem investigated
The proposed procedure concerns the cleaning schedule of two critical heat exchangers operating in parallel, responsible for a heating process in an industrial plan. Figure 1 illustrates the corresponding network, where a cold stream flows from a previous section of the plant through a pump. This stream receives energy in the heat exchangers and goes to a demand unit where the temperature is adjusted at expense of fuel consumption. It is considered that the hot stream fouling is not relevant.

![Network digraph with two heat exchangers in parallel - Cold stream: continuous line - Hot stream: dashed line](image)

The system flow rate is kept at a set point by a control valve. An additional control valve manipulates a bypass stream around the heat exchangers (not represented in Figure 1). This valve opens in order to maintain the total flow when the head loss in the heat exchangers is excessive due to fouling. It occurs if the main valve is already fully open but the flow rate is still below the set point. As there is not heating in the bypass circuit, this measure penalizes the heat exchange. In order to find the best cleaning schedule it is necessary a hydraulic analysis of the network followed by a thermal simulation of the heat exchangers. This model must be complemented by a relation able to describe the dynamic fouling behavior. The equations and considerations used in this work is described as follows.

2.2. Thermal model
The heat transfer in heat exchangers is modeled by the $\varepsilon$-NTU method. In this method it is present three interrelated adimensional groups: thermal effectiveness ($\varepsilon$), number of transfer units (NTU) and heat capacities rates ratio ($C_R$).

The number of transfer units (NTU) is defined in Eq. (1) as the ratio between the heat transfer area ($A$) multiplied by the overall heat transfer coefficient ($U$) and the lower heat capacity rate ($C_{min}$).

$$ NTU = \frac{UA}{C_{min}} $$

The heat capacity rates ratio ($C_R$) is determined by Eq. (2), where $C_{max}$ is the larger heat capacity flow rate, and $C_{min}$ the minimum heat capacity flow rate.

$$ C_R = \frac{C_{min}}{C_{max}} $$

The thermal effectiveness can be calculated for countercurrent heat exchangers by the ratio of the actual heat transferred between the streams ($Q$) and the maximum heat which could be transferred between the fluids ($Q_{max}$):

$$ \varepsilon = \frac{Q}{Q_{max}} $$
\[ \varepsilon = \frac{Q}{Q_{\text{max}}} \]  

(3)

For each heat exchanger configuration is possible to establish a relation as Eq. (4):

\[ \varepsilon = \varepsilon(U_{\text{NUT}}, C_{R}) \]  

(4)

The maximum heat which could be transferred between the fluids \((Q_{\text{max}})\) might be directly determined by Eq. (5), where \(T_{h,i}\) is the hot stream inlet temperature and \(T_{c,i}\) is the cold stream inlet temperature. That way, the heat transferred between the streams \((Q)\) might be calculated by Eq. (3).

\[ Q_{\text{max}} = C_{\text{min}} \left(T_{h,i} - T_{c,i}\right) \]  

(5)

The overall heat transfer coefficient in clean condition \((U_{\text{cl}})\) is represented by Eq. (6), where \(h_{i}\) is the film coefficient in the tube side, \(h_{e}\) is the film coefficient in the shell side, \(k_{\text{w}}\) is the thermal conductivity, \(D_{ti}\) and \(D_{to}\) are inner and outer diameters.

\[ U_{\text{cl}} = \frac{1}{\left[D_{ti}/(D_{ti}+D_{to})\right]+\left[1/h_{i}\right]+\left[D_{to}\log(D_{to}/D_{ti})/2k_{\text{w}}\right]} \]  

(6)

Eq. (7) gives the overall heat transfer coefficient in the operational condition \((U_{\text{op}})\), where \(R_{f}\) is the fouling factor:

\[ U_{\text{op}} = U_{\text{cl}} \left(1/R_{f}\right) \]  

(7)

The film coefficient of each stream in the heat exchanger is calculated using Eq. (8), considering a base case and a correction factor to account for the variation of the stream flow rate.

\[ h = h_{\text{base}} \left(m/m_{\text{base}}\right)^{n} \]  

(8)

where \(h\) is the film coefficient, \(m\) is the mass flow rate and the exponent \(n\) is 0.8 for tube side flow and 0.66 for shell side flow, considering turbulent regime.

2.3. Hydraulic model

The following hydraulic elements are present in the system: pump, pipe sections, heat exchanger and control valve. The centrifugal pump model presented in Eq. (9) corresponds to the representation of its characteristic curve, where \(a_{i}\) are coefficients of the corresponding polynomial equation, \(\rho\) is the specific mass, \(q\) is the volumetric flow rate and \(g\) is the gravity acceleration.

\[ \Delta P_{\text{pump}} = -\rho g \sum_{i} a_{i} (q)^{i} \]  

(9)

The hydraulic model for a pipe section corresponds to the Bernoulli equation with the head loss described by the Darcy equation, where \(P\) is the stream pressure, \(L\) is the pipe length, \(D\) is the inner pipe diameter and \(f\) is the Fanning friction factor.

\[ \Delta P_{\text{pipe}} = \frac{8 \rho L q^{2} f}{\pi^{2} D^{5}} \]  

(10)

The pressure drop for heat exchangers is expressed in Eq. (11), where, \(f\) is the Darcy friction factor, \(L_{s}\) is the tube length, \(\delta\) is the fouling layer \(\nu t\) is the tube velocity, \(N_{pt}\) is the number of tubes per passes and \(D_{t}\) is the tube inner diameter:

\[ \Delta P_{\text{HE}} = \frac{f}{(D_{t} - 2\delta)} \frac{N_{pt} \nu^{2}}{2g} \rho g \]  

(11)

The pressure drop for control valve is described by its characteristic equation:

\[ \Delta P_{\text{valve}} = \alpha d_{\text{g}} \left(q / \left(C_{v} F(l)\right)\right)^{2} \]  

(12)

where \(\alpha\) is a conversion factor for the SI system, \(d_{\text{g}}\) is the specific gravity, \(C_{v}\) is the flow coefficient, \(F(l)\) is a function of the flow characteristic (e.g. linear, equal percentage, etc).

2.4. Dynamic aspects

Fouling decreases the heat transfer rate and interferes in the hydraulic behavior of the heat exchanger. The dynamic simulation of a fouled network during a certain period may be executed using a pseudo stationary approach, since fouling is much slower than the response of the process variables. Thus, a time series of the network behavior due to fouling can be generated through a sequence of steady-state simulations, where fouling factors are updated after each individual run.

Fouling rates can be computed in a simplified manner using linear or asymptotic models. More complex alternatives are able to describe the influence of fluid temperature and velocity in fouling.

As discussed in [7], fouling in the tube side can be hydraulically described as a reduction of the heat exchanger tube diameter, diminishing the cross-sectional area available for flow. In this case, it is important to update the pressure drop in the heat exchanger due to the area reduction as expressed in Eq. (11). It is necessary for this evaluation to calculate the fouling layer thickness \((\delta)\), which rises with the increase of the fouling coefficient.
depending on the tube inner diameter \((D_{ti})\) and the thermal conductivity of the deposits \((k_f)\):

\[
\delta = 0.5 \left[ D_n - D_n \exp\left(-2R_i k_f / D_n\right) \right]
\]  

(13)

2.5. Simulation

Initially, for a given flow rate related to the desired set-point, the hydraulic model determines the pump head and the head loss in the pipe sections. The remainder head is employed to evaluate the potential flow rate along the thermal equipment. If this value is larger than the set-point, it means that the control valve will be partially closed in order to keep the specified set-point flow rate. Otherwise, a fraction of stream will be deviated to the by-pass, such that its flow rate corresponds to the difference between the set-point and the exchanger flow rate.

In previous work [8] it was presented a case where an additional condition was introduced to keep the flow rates in the heat exchangers always with the same values, according to the split suggestion of [7]. However, now this condition is not employed respecting the hydraulic balance of the network.

The hot stream split division respects the cold stream balance in all cases.

2.6. Optimization

The optimization procedure seeks the minimization of the additional energy due to fouling \((Q_{add})\) that should be added in the network to complete the heating of the cold stream. The objective function is calculated in Eq.(14):

\[
fobj = -\int Q dt
\]

subject to

\[
0.1 \leq \alpha \leq 0.9 \quad 0 \leq t_{HE1}, t_{HE2} \leq N
\]

where \(Q\) is the heat exchanger load during the time horizon, \(\alpha\) is valve 1 position in each month of the time horizon and \(t_{HE1}, t_{HE2}\) are the cleaning instants of heat exchanger 1 and 2. \(N\) represents the total number of months of the analysis.

The decision variables are the time instants when the heat exchangers must be put off-line to be cleaned and the control valve position. In the proposed procedure, the number of cleaning actions during the time horizon must be previously established and the time necessary for cleaning activities are not included in the analysis (these periods are much smaller than the time horizon). The limits of the decision variables are eliminated from the problem through a transformation of variables so that the original valves positions \((\alpha)\) are replaced by new decision variables \((\phi)\):

\[
\alpha = \frac{\exp(\phi)}{\exp(\phi) + 1}
\]

(15)

Thus, the new set of decision variables becomes unbounded and the problem assumes the following form:

\[
fobj = -\int Q dt \\
where Q(\phi, \phi_2, \phi_3, ..., \phi_N, t_{HE1}, t_{HE2}) \\
\phi \in \mathbb{R}
\]  

(14)

The optimization problem is solved using both a simulated annealing algorithm and a simplex algorithm. The optimization methods parameters used are presented in Table 1. It was also used and hybrid method where the simulated annealing gives an initial estimative to the simplex method.

<table>
<thead>
<tr>
<th>Method</th>
<th>Tolerance</th>
<th>Step</th>
<th>Maximum iteration number</th>
<th>Number of moves per temperature level</th>
<th>Temperature cooling parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Simulated annealing</td>
<td>20</td>
<td>2</td>
<td>10000</td>
<td>50</td>
<td>0.7</td>
</tr>
<tr>
<td>Simplex</td>
<td>1*10^-4</td>
<td>3</td>
<td>10000</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

*Number maximum of moves without acceptance  Maximum number of moves
3. Numerical results
The application of the proposed procedure is illustrated by an example with two heat exchangers (Fig. 1). It is considered one cleaning during the total period, but the proposed procedure is versatile and can be employed for any number of cleaning actions.

The cold stream flows into the pump suction at a pressure of 1 bar and the final pressure of the system is also 1 bar. The characteristic curve of the pump is represented by a polynomial with the following parameters: $a_0 = 77.86\ m$, $a_1 = -1085\ s/m^2$ and $a_2 = -3634.5\ s^2/m^5$.

The description of the hot and cold streams can be found in Table 2, the heat exchanger specifications are presented in Table 3 and the pipe and valve parameters are shown in Table 4.

### Table 2: Inlet streams specifications

<table>
<thead>
<tr>
<th>Stream</th>
<th>$m$ (kg/s)</th>
<th>$T_i$ (°C)</th>
<th>$\rho$ (kg/m$^3$)</th>
<th>$\mu$ (Pa.s)</th>
<th>$C$ (J/kgK)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cold</td>
<td>120</td>
<td>155</td>
<td>819</td>
<td>$1.27\times10^{-3}$</td>
<td>1378</td>
</tr>
<tr>
<td>Hot</td>
<td>110</td>
<td>232</td>
<td>750</td>
<td>-</td>
<td>2476</td>
</tr>
</tbody>
</table>

### Table 3: Heat exchanger specifications

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube – Film coefficient (W/m$^2$K)$^a$</td>
<td>2150</td>
</tr>
<tr>
<td>Shell – Film coefficient (W/m$^2$K)$^b$</td>
<td>1238</td>
</tr>
<tr>
<td>Effective area (m$^2$)</td>
<td>543</td>
</tr>
<tr>
<td>Number of passes in tubes</td>
<td>2</td>
</tr>
<tr>
<td>Total number of tubes</td>
<td>1520</td>
</tr>
<tr>
<td>Tube thermal conductivity (W/mK)</td>
<td>55</td>
</tr>
<tr>
<td>Tube length (m)</td>
<td>5.968</td>
</tr>
<tr>
<td>Outer tube diameter (mm)</td>
<td>0.01905</td>
</tr>
<tr>
<td>Inner tube diameter (mm)</td>
<td>0.01485</td>
</tr>
</tbody>
</table>

$^a$ Base values for 138 kg/s  
$^b$ Base values for 110 kg/s

### Table 4: Pipeline and valve specifications

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pipe diameter (mm)</td>
<td>0.2032</td>
</tr>
<tr>
<td>Total pipe length (m)</td>
<td>20</td>
</tr>
<tr>
<td>Pipe roughness (m)</td>
<td>$46\times10^{-6}$</td>
</tr>
<tr>
<td>Valve behavior (R) – equal percentage</td>
<td>20</td>
</tr>
<tr>
<td>Valve flow coefficient</td>
<td>700</td>
</tr>
</tbody>
</table>

The fouling growth involves a linear model with a constant rate of $c = 5\times10^{-11}\ m^2/\text{JK}$, associated to a deposit with thermal conductivity equal to 1.5 W/mK. The total horizon of the analysis is 36 months, where one heat exchanger is clean at the startup and the other is already fouled ($R_f = 0.0007\ m^2/\text{K/W}$).

The best result obtained with the optimizations was an optimal solution corresponding to 18.6 months for the first heat exchanger and 16.83 for the second heat exchanger, i.e., approximately the middle of the span time.

The nature of the problem and its solution can be illustrated by the profiles of the main variables. Figure 2 presents the fouling growth during the time horizon.
Figure 2: Fouling growth (a) in the first heat exchanger – clean (b) in the second heat exchanger - dirty

The flow rates in the heat exchangers and in the bypass are shown in Figure 3. In this figure, it is possible to observe the fouling impact on the hydraulic behavior of the system. Due to the valves positions the bypass opens even when the fouling is still low, however it can be observed that there is a upgrading tendency of the bypass flow rate until the 18\textsuperscript{th} month (cleaning of the heat exchangers). After the cleanings the bypass flow rate diminishes significantly until a period where the fouling is again high (final period).

Figure 3: Flow rates in the heat exchangers and bypass

Figure 4 shows the valves fraction opening during the time horizon. The significantly variation of valve position causes the bypass opening even with cleaning heat exchangers.
The outlet temperature of the heat exchangers and the final temperature (resultant from the mixture of bypass and heat exchanger streams) are displayed in Figure 5. The modification of the values during the simulation depends on the impact of fouling growth on the thermal and hydraulic behavior of the system, associated to the two cleaning actions and the valve positions.

The increase of the energy demand due to fouling is displayed in Figure 6, following a trend coherent with the final temperature profile and with the bypass flow rate profile.
Table 5 presents a comparison between results obtained with different optimization methods. The three best results are obtained with the simulated annealing method. It can be observed that in those cases, the total number of moves (and consequently, also total number of evaluation of the objective function) is considerably lower than in the simplex method.

Comparing case 3 with case 12 it can be seen that the same result was achieved with a lesser computational work using the hybrid method.

Table 5: Results

<table>
<thead>
<tr>
<th>Case</th>
<th>Method</th>
<th>Initial estimation</th>
<th>Final results</th>
<th>Number of evaluations of the objective function</th>
<th>Total number of moves</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Simplex</td>
<td>HE1: 10, HE2: 10</td>
<td>HE1: 22.85, HE2: 15.83</td>
<td>1.496*10^{14}</td>
<td>22950</td>
</tr>
<tr>
<td>2</td>
<td>Simplex</td>
<td>HE1: 10, HE2: 10</td>
<td>HE1: 22.78, HE2: 15.96</td>
<td>1.496*10^{14}</td>
<td>23219</td>
</tr>
<tr>
<td>3</td>
<td>Simplex</td>
<td>HE1: 10, HE2: 10</td>
<td>HE1: 14.64, HE2: 22.83</td>
<td>1.522*10^{14}</td>
<td>23253</td>
</tr>
<tr>
<td>4</td>
<td>Simplex</td>
<td>HE1: 10, HE2: 10</td>
<td>HE1: 23.17, HE2: 15.84</td>
<td>1.519*10^{14}</td>
<td>22775</td>
</tr>
<tr>
<td>5</td>
<td>Annealing</td>
<td>HE1: 10, HE2: 10</td>
<td>HE1: 18.60, HE2: 16.83</td>
<td>1.991*10^{13}</td>
<td>-</td>
</tr>
<tr>
<td>6</td>
<td>Annealing</td>
<td>HE1: 10, HE2: 10</td>
<td>HE1: 20.64, HE2: 10.76</td>
<td>1.449*10^{14}</td>
<td>-</td>
</tr>
<tr>
<td>7</td>
<td>Annealing</td>
<td>HE1: 10, HE2: 10</td>
<td>HE1: 24.87, HE2: 14.61</td>
<td>2.514*10^{13}</td>
<td>-</td>
</tr>
<tr>
<td>9</td>
<td>Hybrid</td>
<td>HE1: 10, HE2: 10</td>
<td>HE1: 13.66, HE2: 18.99</td>
<td>1.538*10^{14}</td>
<td>22510</td>
</tr>
<tr>
<td>10</td>
<td>Hybrid</td>
<td>HE1: 10, HE2: 10</td>
<td>HE1: 22.95, HE2: 14.84</td>
<td>1.474*10^{14}</td>
<td>22347</td>
</tr>
<tr>
<td>11</td>
<td>Hybrid</td>
<td>HE1: 10, HE2: 10</td>
<td>HE1: 14.29, HE2: 22.98</td>
<td>1.762*10^{14}</td>
<td>2446</td>
</tr>
<tr>
<td>12</td>
<td>Hybrid</td>
<td>HE1: 10, HE2: 10</td>
<td>HE1: 21.93, HE2: 16.82</td>
<td>1.522*10^{14}</td>
<td>2378</td>
</tr>
</tbody>
</table>

Many results obtained specially using the simplex method showed an optimum where the bypass would only open
a few times as exemplified in Figure 7. In this case after an initial period with full flow along the heat exchangers, fouling implies an increase of the flow resistance, thus imposing the need to open the bypass progressively, until both heat exchangers are cleaned.

![Figure 7](image)

**Figure 7**: Flow rates in the heat exchangers and bypass in case 1

Figure 8 presents the valve opening fraction for case 1. Valve 1 is maintained almost fully opened until the second heat exchanger is cleaned, when the fraction diminishes to allow a great flow rate through the clean heat exchanger and is progressively regressing to the fully opened state. The second valve opens progressively until reaching the maximum opening fraction. The oscillation in that period represents the bypass flow rate variation. When the first heat exchanger is cleaned, the second valve opening fraction is reduced allowing a greater flow rate through the other heat exchanger.

![Figure 4](image)

**Figure 4**: (a) first valve opening fraction, (b) second valve opening fraction

A comparison of results of case 1 and case 5 (the best result) shows that the cleaning instant do not affect significantly the bypass opening, because a later cleaning of the first heat exchanger (case 1) do not imply more bypass valve opening than case 5. It can be concluded, then, that the valves positions have a strong affect in the bypass opening and in the economy of heat added.

### 3. Conclusions

This work presents an algorithm for the optimization of the cleaning schedule of critical heat exchanger in an industrial plant. The approach allows a thermohydraulic analysis of the impact of fouling, which has many advantages. An analysis concentrated only on the thermal behavior may result in significant deviations that can compromise the optimization results.

It could be seen that the simulated annealing method has a better performance than the simplex method for the optimization of the system. The simplex method is dependent of the initial estimative and might be stuck in local
optimums, besides the greater number of evaluations of the objective functions which represents an unnecessary and very high computational time. Future investigations should extend the procedure for complete heat exchanger networks, considering a large set of potential cleaning actions simultaneously.

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