INTERACTION OF HEAT TRANSFER ENHANCEMENT AND FOULING IN OPERATING HEAT EXCHANGERS

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ABSTRACT

Recent increased focus on energy efficiency has contributed to ongoing research and industrial applications of heat transfer enhancement. Early work on efficiency of enhanced surfaces under clean conditions, Rabas 1989 [1], and performance under fouling conditions, Panchal and Rabas 1999 and Watkinson 1991, [2,3] focused on interactions of enhancement and fouling. Interactions arise related to higher clean heat transfer coefficients, and higher wall shear stress of the enhanced surfaces, both of which should give rise to lower initial fouling rates according to popular models of the fouling process. On the Lab scale, enhancement was often achieved by static elements, such as wire wrapping on a heated rod in an annular flow (HTRI probe). Lab scale tests of necessity suffered from effects of relatively short duration of runs, and re-circulation of a batch of liquid with the danger of changing the fluid composition during experiment. Nevertheless, initial fouling rate results consistent with expected trends could be achieved for hydrocarbon fouling, and particulate fouling. There is a continued need to better represent and compare performance of plain and enhanced surfaces over prolonged operating periods in industrial heat exchangers, using commercial enhancement technologies. Example case studies are discussed which demand different methods of comparing performance of the enhanced unit compared to the plain tube unit. A daily averaged cost analysis is identified as a possible systematic approach on the evaluation of the economic benefit of the use of enhancement devices. This is illustrated for a case study example where tube insert is used as an enhancement option.

INTRODUCTION

Heat transfer enhancement techniques has existed even before the introduction of the tubular heat exchangers, where an early reference appears in the use of twisted tapes in steam boilers [4]. However, even nearly a century later, interaction between fouling dynamics and heat transfer enhancements were still poorly defined as evident in a quote by Starner [5], ‘Heat exchanger design innovation utilizing enhanced surfaces is severely influenced and perhaps limited by the lack of detailed knowledge in this field’. Heat transfer enhancement continues to be an important research area with frequent reviews in the last half century [6–12]. This was followed by independent developments in the areas of research on heat transfer enhancement and fouling which gradually combined in the late 1980’s [1,11]. Since, the topic has become a key area of discussions at heat transfer conferences (e.g. 28th National Heat Transfer Conference in Minnesota [13]). Somerscales and Bergles [14,15] provided a comprehensive review in this area.

Techniques for heat transfer enhancement can be divided into four categories, use of passive enhancement, active enhancement, use of nano-fluids and compound techniques. Passive enhancement devices do not require an external power source (e.g. rough surfaces, tube inserts, extended surfaces, etc.). Active devices require external power sources to enhance heat transfer (electric field, acoustic field, surface vibration, etc.). Nanofluids are solid-liquid composite material which has the ability to transfer heat across a small temperature difference. Compound techniques include combination of different enhancement techniques.

Depending on the enhancement technique used, there will be impact on transport coefficient, surface shear stresses and temperature fields which will interact with fouling depending on the fouling mechanism associated with the stream. Examples of the impact of fouling with heat transfer enhancement for both laboratory experiments and industrial observations are summarized in Table 1 and Table 2. The list shows a combination of increase or decrease in fouling rate on the use of enhancement devices for a variety of systems under a range of operating conditions. Some literature shows a decrease in fouling rate with both decrease and increase in surface temperature for crystallization fouling [16,17] indicating the interpretation of the fouling dynamics would require an assessment of the interaction between mass transfer, precipitation and removal. Additionally the effects of augmentation (surface area effects) versus enhancement (increase in heat transfer coefficient) are not always separated.

In this manuscript the interaction on chemical reaction fouling is mainly discussed for a shell-and-tube exchanger.

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Table 1: Selected experimental and numerical studies relating to fouling on enhanced surfaces

Fouling fluid	**Enhancement method**	**Description**	**Ref.**
Water scaling	Inner fin and spirally indented tubes	The asymptotic fouling resistance went through a maximum with velocity (in 7 out of 8 tests). For the inner-finned tubes, the fouling resistance exceeded plain tube values by 15 to 35%, and increased weakly with the ratio of inside/nominal area. At velocities above 3 ft/sec. fouling resistances of the spirally indented tubes were some 25 to 60% below plain tube values.	[17]
Precipitation fouling from inverse solubility salt solutions	Annular turbulence promoter in plain tubes	Maximum thickness of scale inside tubes with ring-type turbulence promoters is smaller than in smooth shell-and-tube heat exchanger tubes.	[18]
Hard water scaling	double inclined ribs	Reduced overall thermal resistance compared to the plain tube for the same operating period.	[19]
Water scaling	Helical threads	The experimental results show that calcium salt scaling can be mitigated to some extent by changing the turbulence structure by means of attaching wires or incorporating helical threads into the heat transfer surface.	[20]
Crystallization fouling (Na_2SO_4)	Coiled wire insert impact on tube internal and tube external application	Exhibited reduced asymptotic fouling resistance with enhancement either on the tube internal or external	[16]
$CaSO_4$ fouling	Numerical study on rectangular channel with half cylinder vortex generator	The asymptotic value of fouling resistance increases with the increase of the surface temperature and the concentration, and decreases with the increase of the inlet velocity.	[21], [22]
$CaCO_3$ fouling	Internal helical ridges	Potential for fouling increases as the number of starts and helix angle increases.	[23]
$CaCO_3$ fouling	Titanium coating	Reduced fouling and deposits easier to remove	[24]
Aluminum oxide particles	copper, helically ribbed tubes	Discusses the impact of different geometry arrangement on fouling rate.	[25]
Particulate fouling (Aluminum oxide)	Rib tubes	The fouling resistances of the repeated rib tubes were higher than that of the smooth tube a low Reynolds number (Re = 14,000). At high Reynolds number, (Re = 26,000), they were almost the same.	[26]
Magnesium oxide suspended in distilled water	In-tube enhancement	Corrugated or roped surface showed the most reduction in fouling (*Helical rib roughness, Corrugated or roped, Axial fins Helical fins*)	[27]
Electric utility condenser fouling (Particulate fouling, slit and cray)	Various enhanced tubes	Enhanced tubes foul faster than the plain tube. At very low concentration the enhanced and plain tubes foul at the same rate.	[28,29]
Ferric oxide fouling	Glass bead particle	Control and remove fouling	[30]
Particulate fouling (Aluminum oxide particles)	copper, helically ribbed tubes	Two factors to decide the ratio of the enhanced-to-plain tube fouling factor: a fouling process index and an efficiency index.	[31]
Biofouling	Helical flutes	Delay in fouling formation, increase in fouling rate, changes in cleanability	[32]
Crude oil	Internally finned tubes	Reduced fouling rate with the enhancement device	[33]
Electric utility condenser fouling (Particulate fouling, slit and cray)	titanium corrugated tubes	For certain test conditions corrugated tubes fouled faster than the smooth tube	[24,34]
Boiling two-phase	Ultrasound	Fouling reduction	[35]

Table 2: Selected industrial experience in use of enhanced surfaces

Fouling fluid	Enhancement method	Description	Ref.
River water fouling in power plant condensers	Spirally indented	The fouling rates with tile enhanced tubes ranged from about the same as to about twice that of the plain tubes.	[36]
Particulate fouling of condensers	Corrugated tubing	Higher fouling with enhancement.	[37]
Particulate fouling	Spirally indented or corrugated tubing	Higher fouling with enhancement.	[38]
Refinery and chemical plant fluids	spirelf®, turbotal® and fixotal® systems	Combination of increased turbulence and movements results in reduction of the tube side fouling layer.	[39]
Quench water in an ethylene plant	Dual enhancement tubes	No significant fouling observed over 11 months of operation	[40]
Crude oil	spirelf® and turbotal® systems	Lower fouling observed with enhancement	[41], [42]
Cooling water	Helical baffle	Lower fouling observed with enhancement	[43]

The fouling resistance of the operating shell and tube unit can be represented as:

$$\frac{1}{U_o} = \frac{1}{h_o} + R_{f,o} + \frac{d_o \ln\left|\frac{d_o}{d_i}\right|}{2k_w} + \frac{d_o}{d_i} \times R_{f,i} + \frac{d_o}{d_i} \times \frac{1}{h_i} \qquad (1)$$

$$R_f = R_{f,o}\frac{d_o}{d_i} + R_{f,i} \qquad (2)$$

Here, U_o is the overall coefficient based on outside area, h_o is the shell-side fluid film coefficient, h_i is the tube-side fluid film coefficient, k_w is the thermal conductivity of the tube material, d_i is the tube internal diameter, d_o is the tube external diameter, $R_{f,i}$ is the tube-side fouling resistance, $R_{f,o}$ is the shell-side fouling resistance.

If T_w is the mean tube wall temperature for a shell and tube unit with cold fluid on the tube-side and hot fluid on the shell side, T_w can be estimated by solving

$$h_o A_o \left(T_h - T_w\right) = h_i A_i \left(T_w - T_c\right) \qquad (3)$$

or

$$T_w = T_h - \frac{\left(T_h - T_c\right)}{\left(1 + \frac{h_o A_o}{h_i A_i}\right)} \qquad (4)$$

Here A_o and A_i are the tube external and internal heat transfer areas, respectively. Changes in T_w due to enhancement is likely to impact the fouling dynamics, in addition to the other changes attributed to the enhancement (e.g. change in heat transfer area, shear stress and geometrical arrangements that facilitate or impede foulant trapping) which will impact the R_f term in equation (2) at a given time instance.

Depending on the type of enhancement devices, numerous approaches for performance evaluation of enhanced surfaces are discussed in the literature (e.g. [44–49]); however it is necessary to account for the performance comparison over a prolonged period of operation. In this manuscript an illustration is provided on the performance impact of a use of tube-side enhancement device on a shell-and-tube unit with tube-side ~~chemical reaction~~ fouling. An average daily operating cost assessment is proposed to perform the performance comparison.

Enhancement interaction with chemical reaction fouling

Performances of heat exchangers installed with commercial tube-inserts, Turbotal® and Spirelf®, were reported by Ishiyama *et al*. [50] for two of the crude preheat trains at the Normandy and Grandpuits TotalEnergies refineries. On one of the preheat trains (section of the hot end of the preheat train shown in Figure 1), a comparison was done on units labelled E31AB and E31CD. Due to the arrangement E31AB was colder than E31CD though both consisted of very similar geometries, with the only difference being that E31CD is equipped with Turbotal® tube inserts. Observed fouling performance showed about an order of magnitude reduction in the observed fouling behavior (Figure 2). Ishiyama *et al*. [50] discusses a fouling model that was fitted to historical plant data taking the form:

$$\frac{dR_f}{dt} = \frac{\alpha}{h}\exp\left(-\frac{E}{RT_f}\right)P \qquad (5)$$

Here, dR_f/dt is the rate of change in thermal resistance, h is the film transfer coefficient, a is the fouling propensity factor which depends on the crude chemistry and the fouling surface, R is the gas constant, E is the activation energy fixed to 44.3 kJ mol^{-1} and T_f is the film temperature. P is the probability of attachment given by Eq'n 6 which allows for decreased attachment at higher wall shear stress.

$$P = 1 - \left(\frac{\tau_s - 2}{100 - 2}\right)^{0.5} \text{ when } \tau_s > 2 \tag{6}$$

$$P = 1 \text{ when } \tau_s \leq 2$$

Here, τ_s is the surface shear stress.

If subscripts e and p represent performance of enhanced tubes and plain tubes, respectively, equation (5) can be written as

$$\beta = \frac{\left(\frac{dR_f}{dt}\right)_e}{\left(\frac{dR_f}{dt}\right)_p} = \frac{\phi_{D,e}}{\phi_{D,p}} = \frac{1}{\left(\frac{h_e}{h_p}\right)} exp\left[-\frac{E}{R}\left(\frac{1}{T_{f,e}} - \frac{1}{T_{f,p}}\right)\right]\left(\frac{P_e}{P_p}\right) \tag{7}$$

Here ϕ_D is the deposition flux, based on the plain tube surface area. Enhancement method discussed in this manuscript are the use of tube-inserts and no change in heat transfer area will occur following this enhancement method.

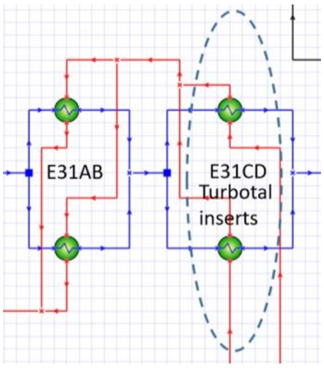

Figure 1: Section of the TOTAL refinery crude preheat train (extracted from [41]).

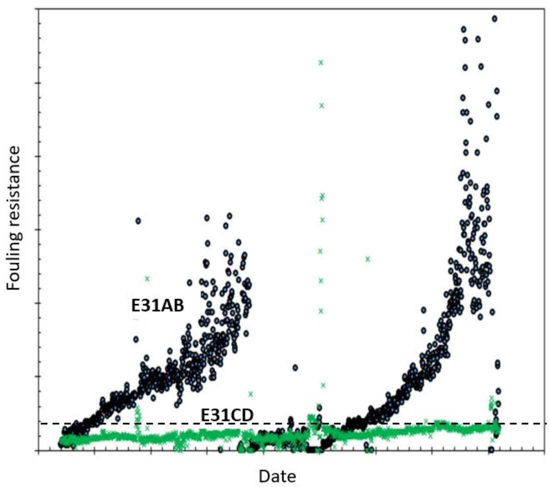

Figure 2: Comparison of thermal resistance performance of plain tube (E31AB) and unit with tube insert (E31CD) at TOTAL Grandpuits preheat train (image based on [41]). The dashed horizontal line represent possible observed asymptote with the use of tube-side enhancement in this application. The absolute values of the axis are hidden due to confidential information. The fouling resistance noted here is the overall fouling resistance (based on the tube-and shell sides), however, the product stream on the shell-side is relatively clean and the dominant fouling is likely on the crude-side for this unit.

Equation (7) shows that the ratio of fouling deposition flux between enhanced tubes and plain tubes is a function of the change in film transfer coefficients, film temperature and wall shear stress. Taking a hypothetical case, for the plain tube, the fluid bulk temperature, $T_{b,p}$, of 100 °C and a tube wall temperature, $T_{w,p}$, of 210 °C (film temperature, $T_{f,p}$, taken as the average of $T_{b,p}$ and $T_{w,p}$). Figure 3 is a plot of , β (ratio of fouling rates of enhanced and plain tube units given by equation (7)), for a range of h_e/h_p and a range of P_e/P_p. The plot provides an indication of the extent of fouling rate change compared to a plain tube with the use of the enhancement method which changes the wall temperature, film transfer coefficients and shear stress. For example, if an enhancement method offered a 1.5 times increase in the film transfer coefficient with a reduction in the shear dependent attachment of 0.9 (P_e/P_p), then the initial fouling rate with this enhancement would be about 35 % of the plain tube case. However, with progress in fouling, the operating conditions do change and the fouling dynamics will be affected and a need exists to predict the performance over a prolonged period.

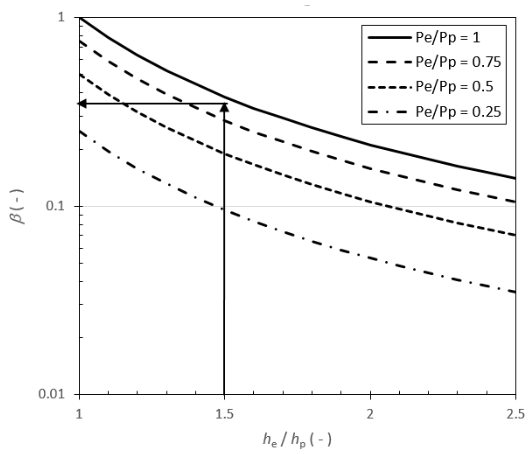

Figure 3: Projected ratio of fouling rates vs. enhancement factors generated using equation (7).

Turbotal® inserts are rotating devices hooked on a fixed head set on the tubesheet on the inlet side. This system converts the energy of the fluid flow in the tubes into rotation. For the device under the correct operating regime, the deposit thickness should not exceed the gap between the tube surface and the insert. The mechanical motion is likely to mitigate the deposit build up. That is, once the deposit thickness, δ, reaches the gap between the tube-insert and the tube internal diameter, g_{insert}, it is assumed that further deposition will be essentially zero (assuming the insert is in the intended operating regime). This assumption may further explain the observation of the fouling resistance reaching an asymptote in Figure 2 and may be modelled as:

$$\left(\frac{dR_f}{dt}\right)_e = 0, \text{ when } \delta = g_{insert} \quad (8)$$

The following case study (based on [51]) is taken to explore the impact of the operation described by equation (8). As the Turbotal® and Spirelf® insert geometries and thermo-hydraulic models are proprietary and are only available through licensed commercial tools (such as HTRI SmartPM™ [52]), its performance is not modeled here. Instead the impact on the possible long term operation is assessed via the assumption that with the use of a hypothetical tube insert, the initial h_e/h_p is 1.5 and the P_e/P_p is 0.9 (i.e. the impact of the enhanced tube in this instance is calculated via application of scaling factors for the plain tube f and j factors. A hypothetical geometry of the example exchanger is summarized in Table 3. The assumed stream thermo-physical properties and operating conditions are summarized in Table 4.

Table 3: Summary of a hypothetical exchanger geometry for case study 1

Parameter	Value
Shell internal diameter	508 mm
Number of tube-side passes	2
Total number of tubes	220
Shell type	AES
Tube length	4877 mm
Tube outside diameter/ inside diameter	19.05/14.83 mm
Tube layout	90°
Baffle cut	18 %
Number of baffles	30

Table 4: Stream thermo-physical properties and operating conditions

	Cold stream	Hot stream
Inlet temperature	146.1 °C	268.5 °C
Inlet flow rate	21.67 kg/s	14.36 kg/s
Density	764.6 kg/m³ (120 °C)	754.9 kg/m³ (150 °C)
	648.9 kg/m³ (250 °C)	632 kg/m³ (300 °C)
Sp. heat capacity	2.36 kJ/kg K (120 °C)	2.42 kJ/kg K (150 °C)
	2.81 kJ/kg K (250 °C)	2.94 kJ/kg K (300 °C)
Conductivity	0.121 W/m K (120 °C)	0.132 W/m K (150 °C)
	0.074 W/m K (250 °C)	0.095 W/m K (300 °C)
Viscosity	0.9591 cP (120 °C)	0.6144 cP (150 °C)
	0.2786 cP (250 °C)	0.158 cP (300 °C)

In this study a constant deposit thermal conductivity, λ_f, of 0.2 W/m K was used and the deposit thickness calculated using the approximation:

$$\delta = R_{f,i}\lambda_f \quad (9)$$

Equation (5) represent the fouling rate model used for the crude stream with a fouling propensity factor, a of 250 h^{-1} (*i.e.* assuming the fouling propensity is approximately half of the value reported in [51]). Figure 4 is a plot of the tube (cold)-side skin temperature variation with fouling for plain tube and with the enhancement devise. Initially the tube with the tube-insert will have a lower skin temperature (see equation (4)). Once the gap between the tube internal and the tube insert is fouled and the remaining fouling is suppressed *via* the tube-insert motion, no further changes in operation will be visible (horizontal dashed line in Period B).

The performances of plain and enhanced tube can be made by comparing the relative impact on thermal and hydraulic operation. This approach is made via considering two dimensionless parameters, h_e/h_p and $\Delta P_e/\Delta P_p$. Here h is the film transfer coefficient, ΔP is the pressure drop and subscripts e and p denote the enhanced and the plain tube conditions, respectively. h_e/h_p is a representation of the enhancement in the heat transfer and $\Delta P_e/\Delta P_p$ is a representation of the hydraulic penalty. This dimensionless values are generated for case study 1 and plotted in Figure 5 over a dynamic variation accounting to two year period. The time scale is common and hence not plotted. At the start (clean condition denoted by point A), an enhancement of 1.5 times in the film transfer coefficient is observed with a hydraulic penalty of about 1.8 (the pressure drop has increased by 1.8 times due to the hydraulic restrictions imposed by the insert). Fouling progresses differently in plain and enhanced tubes due to the different temperature fields, film coefficients and associated hydraulic resistance. Based on equation 7, the enhance tube has a reduced initial fouling rate due to the enhanced film coefficient and increased hydraulic resistance. With progress in fouling, for the enhanced tube, the deposit thickness reaches the gap between the enhancement device and the tube internal surface (point B). Beyond this point, no fouling is assumed (based on equation 8). However for the plain tube, the fouling will continue (even with diminishing rate) and increases the plain tube hydraulic penalty from Point B to C. At one point the plain tube pressure drop will exceeding the pressure drop of the enhancement device installed (when $\Delta P_e/\Delta P_p$ goes below 1). The film transfer coefficient will also continue to increase (in a diminishing rate) for the plain tube case as fouling leads to flow area occlusion hence a reduction in h_e/h_p is observed over B to C.

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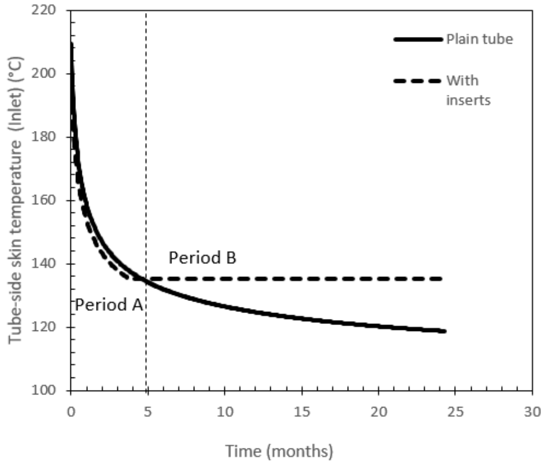

Figure 4: Tube-side skin temperature (at inlet). Solid line, plain tube; dashed line with inserts.

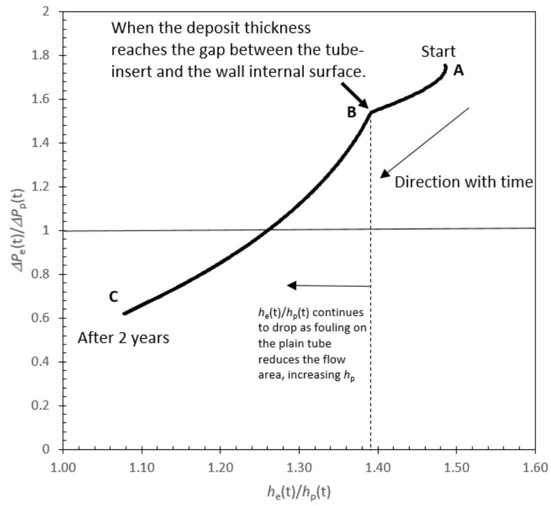

Figure 5: Plot of $\Delta P_e/\Delta P_p$ over h_e/h_p for a period of 2 years starting with the clean condition (point A) till the end of prediction study (point C).

Performance Evaluation

Performance evaluation criteria defined at point A appears no longer relevant in evaluating the overall performance of the unit over its' operating cycle. Defining a performance of inserts in an operating exchanger over a prolonged period requires performing a techno-economic evaluation on the following

- Cost of inserts, transportation and installation
- Average life time: Some inserts have a life time limited due to mechanical fatigue. E.g. Turbotal® inserts would have a defined lifetime due to the wearing of the bearings
- Immediate heat transfer enhancement and overall thermal performance over the cycle
- Overall hydraulic performance

For a unit with falling rate process, a cyclically averaged daily cost of operation, φ, can be defined [53,54] for the original unit as

$$\varphi_p = \frac{C_E\left[\int_0^t \left(Q_{cl,p} - Q\right)dt + Q_{cl,p}\mathrm{X}_{cl,p}\right] + f\left(\Delta P - \Delta P_{cl,p}\right) + C_{m,p}}{t + \mathrm{X}_p} \qquad (10)$$

Here C_E is the energy cost, Q_{cl} is the clean heat duty, Q is the operating heat duty, X_{cl} is the time taken for cleaning the unit, C_m is the maintenance cost including cost of cleaning and any other expenditure (for an enhanced tube, this may include the cost of replacing the enhancement device if advised by the manufacturer), t is the operating time. The function $f(\Delta P - \Delta P_{cl})$ defines the cost associated with the pressure drop penalty which can be related to the increased pumping cost or lost opportunity cost. Subscript p refers to the plain tube unit. When comparing φ between a plain and an enhanced unit (subscript e), φ_e can be presented as

$$\varphi_e = \frac{C_E\left[-\int_0^{t_p} \left(Q_{cl,e} - Q\right)dt + \int_{t_p}^{t} \left(Q_{cl,p} - Q\right)dt + Q_{cl,e}\mathrm{X}_{cl,e}\right] + f\left(\Delta P - \Delta P_{cl,e}\right) + C_{m,e}}{t + \mathrm{X}_{cl,e}} \qquad (11)$$

Here t_p is the time instance when the enhanced unit reaches the original clean unit duty before enhancement. The excess duty while the unit is operating above the original clean duty is a 'negative cost' in equation (11).

Table 5 shows an assumed hypothetical economic parameter, the maintenance cost with the insert is assumed double that of the original case in this example with an assumption that the enhancement device may need replacement due to mechanical fatigue. Figure 6 is obtained *via* solving equations (10) and (11) with example hypothetical economic parameters in Table 5.

Table 5: Example parameters used to assess φ_p and φ_e.

	Original unit	Enhanced unit
C_m (US$)	20,000	40,000
C_E (US$/GJ)	4	4
X_{cl} (days)	7	7

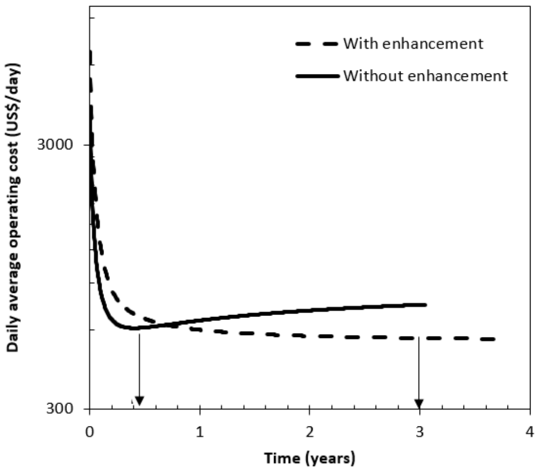

Figure 6: Daily average operating cost over time for original unit (solid line) and enhanced unit (dashed line). Daily average cost is plotted in logarithmic scale.

For the plain tube case, a week minimum is observed around 6 months operation (appr. 610 US$/day) while for the enhanced tube, the minimum has not reached in the considered time frame. Though the cleaning time is constrained by the timing where the enhancement device requires replacement (e.g. 3 years is assumed in this example). In 3 years, φ_e is appr. 550 US$/day. If the original unit was operated for the same time period without any cleaning event (as the enhanced unit), φ_p is appr. 750 US$/day. The result can be interpreted as:

- If both the original unit and the enhanced unit are operated for 3 years followed by a cleaning, the economic performance $\{(\varphi_e - \varphi_p) \times \text{period}\}$ of the enhanced unit will be superior to that of the original unit. In the hypothetical example (Figure 6), this is {(750-550) × 365 × 3 = US$ 153,300}.
- If the original unit followed an optimum cleaning cycle schedule, then the superiority of the enhanced device is {(610-550) × 365 × 3 = US$ 65,700}. However this implies the original unit is cleaned twice a year (i.e. a total of 5 additional cleans compared to the enhanced device) before the end of the 3 year cycle. Safety considerations with additional cleaning needs to be accounted for in such decisions.

If both the hot and cold streams are prone to fouling the enhancement on one side (example cold-side) will also impact the fouling on the hot-side. This can be illustrated by taking an example of a feed-effluent heat exchanger discussed in Ishiyama [55], where the feed stream was subject to chemical reaction fouling and the effluent stream was subject to salt deposition. If a heat transfer enhancement device was applied on the feed stream, this will result in wall temperature cooling. On the effluent stream, if the wall temperature is cooled such that the fluid is exposed to a deposition regime (e.g. moving from point A to B in Figure 7) then it is possible that the effluent stream will also start to foul. These dynamics need to be accounted for when evaluating the performance analysis over a prolonged period which will be part of defining the fouling model for the associated streams.

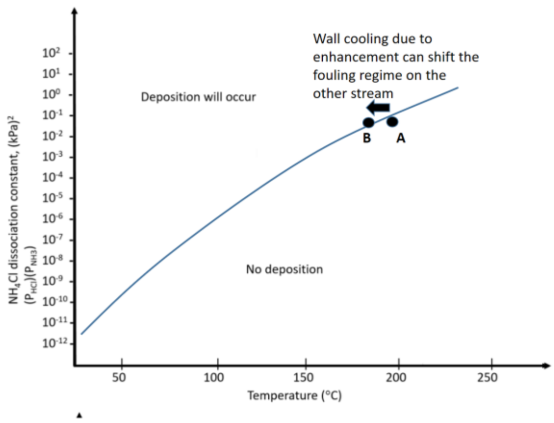

Figure 7: Dissociation curve for ammonium chloride (based on [56]). The abscissa denote wall temperature.

Conclusions

A survey of the literature on fouling and enhancement interactions showed a complex mixture of results, due in part to confounding of augmentation and enhancement effects. There was a lack of examples in industrial heat exchangers.

Representation of the figure of merit of a heat transfer enhancement device requires understanding of the fouling dynamics of the streams associated over a prolonged operation. Enhancement on one side of the stream will impact the wall/skin temperature on the other stream and its fouling dynamics.

Fouling of a shell and tube industrial heat exchanger equipped with Turbotal® inserts was studied. Fouling is eventually limited by the geometry of the Turbota®l element.

A daily averaged operating cost comparison method is proposed to compare the performance benefits between the original unit and the unit that has gone enhancement. The comparison can be used to assess the changes in the cleaning frequency (less cleaning preferable due to operational safety reasons) and also the overall operating cost benefit in the operating campaign.

Nomenclature

A heat transfer area, m^2
C_E cost of energy, US$/J

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C_m cost of maintenance including cleaning and any replacement of enhancement device if applicable, US$
d diameter of the tube, m
E activation energy, J mol^{-1}
g_{insert} gap between the tube insert and the tube wall, m
h film transfer coefficient, W m^{-2}K^{-1}
k_w tube wall thermal conductivity, W m^{-1}K^{-1}
P probability of attachment, -
Q heat duty, W
R gas constant, J mol^{-1} K^{-1}
R_f fouling resistance, m^{2}K W^{-1}
t time, s
t_p time taken for the enhanced unit to reach the original unit duty before enhancement under fouling, s
T fluid temperature, K
U overall heat transfer coefficient, W m^{-2} K^{-1}

Symbols
α fouling propensity factor, s^{-1}
β ratio of fouling rates of enhanced and plain tube units, -
X time taken for cleaning and maintenance, s
φ cyclically averaged daily cost, US$ s^{-1} or US$ day^{-1}
ϕ_D deposition flux, kg m^{-2}
ΔP pressure drop, Pa
τ_s surface shear stress, Pa

Subscripts
c cold side
cl clean condition
e enhanced condition
f film
h hot side
i inner surface / tube-side
o outer surface / shell-side
p plain condition (condition before enhancement)
w wall

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