Quantifying the 'implementation gap' for antifouling coatings

Short title: Implementation gap for antifouling coating

Ole Mathis Magens[a], Edward M. Ishiyama[b], D. Ian Wilson[a,1]

[a] Department of Chemical Engineering and Biotechnology, New Museums Site, Pembroke Street, Cambridge, CB2 3RA, UK

[b] IHS Downstream Research, 133 Houndsditch, London, EC3A 7BX, UK

Abstract

Fouling is a chronic problem in many heat transfer systems and leads to regular cleaning of heat exchangers. Antifouling coatings are one mitigation option: the financial attractiveness of installing a coated exchanger depends on trade-offs between capital and operating costs over the lifetime of the unit. Such considerations effectively set bounds on the price of coatings, bounded by manufacturing costs and the maximum saving that can be achieved from fouling mitigation, in a 'value pricing' calculation. The 'value pricing' concept is considered here, for the first time, for heat exchangers subject to asymptotic fouling. An explicit solution to the cleaning scheduling optimisation problem is presented for the case of equal heat capacity flow rates in a co- or counter-current single phase exchanger. A case study is used to illustrate the concepts and key learnings. A sensitivity analysis identifies scenarios where the use of antifouling coatings may be attractive, and also where there is no financial benefit in cleaning a fouled exchanger.

Keywords Fouling, cleaning, asymptotic, antifouling coating, techno-economic analysis

1 Introduction

Fouling is a chronic problem in many process heat transfer systems. The presence of unwanted deposit layers cause increased resistance to heat transfer and can cause blockage. The associated losses in thermal and hydraulic performance over time directly impact the sustainability of systems affected by fouling. It also introduces the need to clean heat exchangers on a regular basis. Cleaning is rarely instantaneous, requiring the unit to be taken out of service. This incurs further energy losses, or capital expenditure in order to maintain a backup facility to cover the absent unit. Cleaning operations also introduce further, non-thermal environmental impacts and wider sustainability considerations associated with consumption and disposal of cleaning chemicals and wasted product.

[1]Corresponding author. Email: diw11@cam.ac.uk; Tel. +44(0) 1223 334 777; Fax. + 44(0) 1223 334 796

The decision when and how to clean an exchanger is an optimisation problem, considering the cost of energy losses due to fouling over an operating period of length t and those incurred as a result of cleaning (taking time τ). Figure 1 illustrates the problem for a single heat exchanger. This 'fouling-cleaning cycle' problem was first described by Ma and Epstein [1] and a practical example and further analysis was presented by Cosado [2]. A dimensional analysis of the problem, including the effects of ageing, was given in [3].

Methods for identifying the optimal fouling-cleaning cycle period, *i.e.*, $t + \tau$, have been identified for different operating scenarios [3], as well as cases where there is a choice of cleaning method [4]. The objective function to be optimised for scenarios involving a single heat exchanger is the time-averaged operating cost, ϕ_{op}, given by

$$\phi_{op} = \frac{c_E \left[\int_0^t (Q_{cl} - Q) dt' + Q_{cl}\tau \right] + C_{cl}}{t + \tau} \quad , \tag{1}$$

where Q is the heat duty, c_E is the cost of energy and C_{cl} is the cost of a cleaning operation. The calculations require knowledge of the fouling behaviour over time, $Q(t')$. If this is available, it allows the operator and the designer to determine the optimal configuration and operating strategy for the unit. This is a classical example of a trade-off between capital investment, linked to the design of the unit, and operating costs, linked to time in service. Different designs can then be compared.

The use of antifouling coatings to delay the onset of fouling or to hinder fouling (maintain Q near Q_{cl}), as well as to enhance cleaning (reduce τ and/or C_{cl}), has been actively pursued in several industrial sectors. Such 'non-stick' coatings often incur additional capital spend related to the cost and manufacture of coated surfaces. There can also be a reduction in heat transfer coefficient when the layer has a relatively low thermal conductivity. The financial attractiveness (*i.e.* the economic sustainability) of installing a coated heat exchanger then depends on the trade-off between capital and operating costs over the lifetime of the unit. In practice the lifetime of the unit is likely to be determined by the effectiveness of the layer, as the layer is likely to degrade or otherwise suffer reduced performance over time. The balance between these costs will differ between a new system and a revamped or retrofitted one. In the latter case, an existing exchanger is replaced and the extra capital outlay needs to be recovered from improved operation.

These financial considerations – which can include CO_2 taxes – effectively set bounds on the price of antifouling coatings, determined by comparing manufacturing costs and the maximum saving that can be achieved from fouling mitigation, in a 'value pricing' calculation. Order of magnitude estimates for different applications can establish the potential attractiveness of antifouling coatings for a given scenario. This concept was outlined by Gomes da Cruz *et al*. [5], who applied it to three cases with different operating and cost bases. They assumed simple fouling behaviour, *i.e*. where the fouling resistance, R_f, increased linearly with time at constant fouling rate b, *viz*.

$$R_\mathrm{f} = \frac{1}{U} - \frac{1}{U_\mathrm{cl}} = b \cdot t' \quad , \tag{2}$$

where U is the overall heat transfer coefficient and U_cl the value after cleaning. Linear R_f -t' behaviour, as described by (2), is often not observed in practice as (*i*) there may be an induction time, t_ind, before noticeable effects of fouling appear, and (*ii*) the rate of increase in R_f varies with time owing to changes in surface temperature, deposit strength *etc*. Asymptotic fouling behaviour is often reported, wherein R_f approaches a limit at long times. This is often described mathematically by the Kern-Seaton model [6]:

$$R_\mathrm{f} = \begin{cases} 0, & t' < t_{ind} \\ R_f^\infty\left(1-\exp\left((t_{ind}-t')/t_f\right)\right) = R_f^\infty\left(1-\exp\left(-t^*/t_f\right)\right), & t' \geq t_{ind} \end{cases} . \tag{3}$$

Here t'-t_ind is written as t^* for convenience: t_ind is the induction period where there is negligible deposition, t_f is the characteristic timescale (the kinetic parameter), and R_f^∞ is the asymptotic fouling resistance. The latter parameter is frequently employed in overdesigning heat exchangers subject to fouling, even though this approach tends to promote fouling in a 'self-fulfilling prophecy' [7]. The Kern-Seaton model is employed here: other expressions may also be used, but the results obtained in Section 2.3 may not apply.

This paper develops the 'value pricing' concept for heat exchangers subject to asymptotic fouling, extending the numerical analysis in [5] to one type of fouling behaviour which is of direct relevance to industrial practice. Criteria determining when an exchanger should be cleaned are identified. We have identified once case, that of equal heat capacity flow rates, where a semi-analytical result can be obtained which does not require tedious calculatoin. Its use is illustrated using a case study based on data reported by Oldani *et al*. [8], comparing water crystallisation fouling on a stainless steel tube and one coated with a perfluoropolyether (PFPE) coating.

2 Modelling and analysis

2.1 The operating cost

The time to clean, τ, is assumed to be independent of processing history. This assumption is expected to be valid if the exchanger has to be disassembled for cleaning. τ is likely to be reduced if cleaning-in-place is used and the antifouling coating promotes cleaning. Inspection of Equation (1) shows there will be an optimal processing time, t_{opt}, if $d\phi_{op}/dt = 0$, which requires

$$\phi_{op}(t_{opt}) = c_E\left(Q_{cl} - Q(t_{opt})\right) \quad . \tag{4}$$

This statement of the the optimal processing criterion requires the operating cost at t_{opt} to equal the thermal cost penalty due to fouling at that instant. The condition for a minimum in ϕ_{op}, $d^2\phi_{op}/dt^2 > 0$, requires $dQ/dt < 0$, *i.e.* the heat duty has to continue to decline. The optimal processing period will therefore always exceed the induction period. There are two other results of practical interest:

(i) Where t_{opt} is large (i.e. long operating periods), such that $t_{opt} \gg t_f$, asymptotic fouling behaviour results in $dQ/dt = 0$ and there is little benefit in cleaning the exchanger: it should be left to operate in its fouled state, or until another criterion applies.

(ii) If fouling is very fast, such that R_f^∞ is reached quickly, the unit is best left to operate in its fouled state. Only under these conditions the unit should be designed with a U value including R_f^∞, which is the basis of the TEMA approach. If fouling is very fast, however, mitigation should be given stronger consideration.

2.2 Impact of fouling in a simple heat exchanger

We consider an individual heat exchanger, rather than one in a network. Its thermal performance is modelled using a lumped parameter approach for the purposes of illustration: more detailed models could be employed as required. An implementation of a one-dimensional model which incorporates spatial resolution along the exchanger has been reported by Magens *et al.* [9]. Here, the instantaneous heat duty, $Q(t')$ is calculated using the *NTU*-effectiveness method [10]. Equations (2) and (3) are combined to give the overall heat transfer coefficient, U, *viz.*

$$U = \frac{1/R_f^\infty}{1/(U_{cl}R_f^\infty) + 1 - \exp(-t^*/t_f)} = \frac{a_1}{a_2 + 1 - \exp(-t^*/t_f)} \quad . \tag{5}$$

Here, $a_1 = 1/R_f^\infty$ and $a_2 = 1/(U_{cl}R_f^\infty)$. The latter is the reciprocal of an asymptotic fouling Biot number $Bi_f^\infty = U_{cl}R_f^\infty$. The number of heat transfer units of the heat exchanger, NTU, is given by

$$NTU = \frac{UA}{W_{min}} = \frac{a_1 A}{W_{min}\left(a_2 + 1 - \exp(-t^*/t_f)\right)} \quad , \tag{6}$$

where A is the heat transfer area and W is a stream heat capacity flow rate, given by $W = wc_p$. W_{min} is the smaller of the heat capacity flow rates of the two streams entering the exchanger.

Solution of Equation (1) subject to U given by Equation (5) and heat exchanger performance relationships such as the NTU-effectiveness approach usually requires numerical calculation. This is illustrated here by considering one of the simplest practical cases, that of the co- or counter-current heat exchanger with equal heat capacity flow rates ($W_{hot} = W_{cold} = W_{min}$). It will be shown that this yields a tractable semi-analytical solution. Examples where $W_{hot} = W_{cold}$ arise include preheaters (where an outlet stream is used to preheat or precool an inlet stream) and sections of dairy plate heat exchangers.

The effectiveness, ε, is given by the simple relationship

$$\varepsilon = \frac{NTU}{1+NTU} = \frac{a_1 A}{a_1 A + W\left(a_2 + 1 - \exp(-t^*/t_f)\right)} \quad , \tag{7}$$

where ε is the ratio of the actual rate of heat transfer, Q, to the thermodynamically maximum possible duty, Q_{max}, which is related to the maximum heat transfer driving force, ΔT_{max} via

$$Q = \varepsilon Q_{max} = \left[\frac{a_1 A}{a_1 A + W\left(a_2 + 1 - \exp(-t^*/t_f)\right)}\right] W \Delta T_{max} \quad . \tag{8}$$

This can be written as

$$Q = \frac{a_3}{a_4 - \exp(-t^*/t_f)} \quad .$$

It follows from combining (6) – (8) that

$$a_3 = a_1 A \Delta T_{max} = \frac{A \Delta T_{max}}{R_f^\infty} = Q_{cl}\frac{1+NTU}{U_{cl} R_f^\infty} = Q_{cl}\frac{1+NTU}{Bi_f^\infty} \quad , \tag{9}$$

and

$$a_4 = a_2 + \frac{a_1 A}{W} + 1 = \frac{1}{U_{cl} R_f^\infty} + \frac{A}{R_f^\infty W} + 1 = \frac{1}{Bi_f^\infty}\left(1 + NTU + Bi_f^\infty\right) \quad . \tag{10}$$

The clean heat duty at $t^* = 0$ is equal to $Q_{cl} = a_3/(a_4\text{-}1)$, whereas the heat duty after a long period of operation, *i.e.* $t^* \to \infty$, is $Q = a_3/a_4$. Substituting Equation (8) into the operating cost function, Equation (1), and integrating yields

$$\phi_{op}(t) = \frac{c_E\left[\frac{a_3}{a_4 - 1}(t - t_{ind} + \tau) - \frac{a_3}{a_4} t_f \log\left(\frac{1 - a_4 \exp((t - t_{ind})/t_f)}{1 - a_4}\right)\right] + C_{cl}}{t + \tau} \quad . \tag{11}$$

If the unit is never cleaned (*i.e.* $t \rightarrow \infty$), the operating cost will approach the thermal cost penalty asymptotically. This is given by

$$\lim_{t \to \infty} \phi_{op} = c_E Q_{cl} \left[1 - \frac{1+NTU}{1+NTU+Bi_f^{\infty}} \right] . \tag{12}$$

Equation (12) shows that a large exchanger (NTU large) is less sensitive to fouling, since if $1 + NTU >> Bi_f^{\infty}$ the term in the brackets is small.

For an optimal processing period, *i.e.* minimum operating cost, Equation (4) has to hold. This can be solved numerically for t_{opt}. In engineering applications, however, such as scheduling cleaning or as an instrument to quantify the financial attractiveness of heat exchanger coatings, a simplified approach is desirable. This is considered in the next section where a tractable explicit solution is presented.

2.3 An explicit result for cases of equal heat capacity flow rate

2.3.1 Derivation

Ishiyama *et al.* [3] obtained implicit analytical solutions for (1) for the case of linear fouling (Equation (2)). Similarly, Equation (3) does not, to the authors' knowledge, yield explicit analytical solutions for t_{opt}. This section describes an explicit approximation which can be computed without iteration.

The heat duty can be rewritten as

$$Q = \frac{a_3}{a_4 - \exp(-t^*/t_f)} = \frac{a_3}{a_4 - \exp(-t^*/t_f)} - \frac{a_3}{a_4} + \frac{a_3}{a_4} = \frac{a_3}{a_4}\left[\frac{\exp(-t^*/t_f)}{a_4 - \exp(-t^*/t_f)} + 1 \right]. \tag{13}$$

If $a_4 >> \exp(-t^*/t_f)$ $\forall$ $t^* \geq 0$, Equation (13) can be simplified to give the approximate result

$$Q \approx Q_{approx} = \frac{a_3}{a_4}\left[\frac{\exp(-t^*/t_f)}{a_4 - 1} + 1 \right] . \tag{14}$$

The approximate heat duty gives the exact result for $t^* = 0$ and $t^* \rightarrow \infty$. The relative approximation error at other instances of t^* is calculated from

$$\eta = \frac{Q - Q_{approx}}{Q} = 1 - \left(\frac{\exp(-t^*/t_f))}{a_4 - 1} + 1 \right)\left(1 - \frac{\exp(-t^*/t_f)}{a_4} \right) . \tag{15}$$

To find the maximum error, $d\eta/dt^*$ is set to zero. The maximum relative error for a physically feasible time occurs at $t_{\eta,max}^* = t_f \ln(2)$. This corresponds to a maximum relative error of

$$\eta_{\max} = \eta(t_{\eta,\max}{}^*) = -\frac{1}{4a_4(a_4-1)} \quad . \tag{16}$$

It should be noted that the relative error is negative, since $a_4 > 1$. If the relative error is constrained to lie with $\eta_{\max} > \eta_c$ = -5%, the minimum value of a_4 is (ignoring the physically infeasible negative solution for a_4) given by

$$a_4 = \frac{1}{Bi_f^\infty}\left(1 + NTU + Bi_f^\infty\right) \;>\; \frac{1}{2} + \sqrt{\frac{1}{4} - \frac{1}{4\eta_c}} \approx 2.8 \quad . \tag{17}$$

This holds for many heat exchangers in practice, where the flow is counter-current, since the number of heat transfer units of a thermally well-designed heat exchanger is greater than 3 [10]. To find the optimal processing period, the approximate heat duty is inserted into Equation (4). A series of algebraic transformations gives the approximate optimal processing period

$$t_{opt-approx} = -\tau - t_f\left[1 + \mathrm{W}_{-1}\left(-\frac{\chi \exp(-(\tau + t_{ind})/t_f - 1)}{t_f}\right)\right] \quad . \tag{18}$$

Here, W_{-1} is the negative branch of the Lambert W function, shown in Figure 2, and

$$\chi = t_f + t_{ind} - \frac{C_{cl}}{c_e Q_{cl}} - \frac{1 + NTU}{Bi_f^\infty}\left(\tau + \frac{C_{cl}}{c_e Q_{cl}}\right) \quad . \tag{19}$$

If the heat exchanger subject to fouling can be modelled as described above, a criterion can be derived indicating whether it is financially attractive to clean the unit. Equation (4) is then treated as an inequality and the condition for an optimum in operating cost is relaxed. If no other constraints apply, *e.g.* hygiene considerations in food processing or scheduled mechanical integrity checks, it is financially not attractive to clean the exchanger if the following condition holds:

$$C_{cl} > \lim_{t \to \infty} c_E\left[\int_0^t Q\, dt' - Q(t)(t + \tau)\right] \quad . \tag{20}$$

Integration of the heat duty, Q, using Equation (8), tends to infinity when integrated from zero to infinity: employing the integrated approximate heat duty, Equation (14), gives a finite result. Utilising this results in $\chi < 0$.

Given a countercurrent heat exchanger with identical, constant heat capacity flow rates, static inlet temperatures, the availability of fouling data and $a_4 > 2.8$, application of the approximate method is straightforward. If $\chi < 0$ there is no resultant cost benefit of cleaning and the exchanger should be allowed to operate in the fouled state. In this case, the operating cost can be calculated with Equation (12). Otherwise, Equations (18) and (19) are used to schedule cleaning. The approximate optimal operating cost is then calculated by inserting the approximate solution for the optimal processing period, $t_{\text{opt-approx}}$, in Equation (11). This methodology enables researchers and practitioners to estimate the economical value of anti-fouling coatings in heat exchangers without employing involved numerical techniques such as described in [5].

3 Illustrative case study

A case study, based on data taken from the literature, is used to illustrate the quantifying of financial benefit of anti-fouling coatings. Oldani *et al.* [8] reported the performance of a single-pass counter-current shell-and-tube unit with constant flow rates and approach temperatures. The process and utility streams were both aqueous, and the unit was subject to crystallisation fouling. The R_f-t' data sets in Figure 3 were interpreted to exhibit asymptotic fouling behaviour and were fitted by the least-squares method to Equation (3). The model parameters are reported in Table 1. In this case the PFPE coating reduced the rate of fouling and magnitude of the asymptotic fouling resistance. However, the characteristic fouling timescale remained similar. No induction period was observed for either surface. The design and operating parameters of the exchangers considered in this study are summarised in Table 1. The parameters resemble the conditions in the model heat exchangers of Oldani *et al*. Some parameters were not reported, and these values were taken from the case study by Gomes da Cruz *et al*. [5], where the fouling behaviour was modelled as linear.

The two major differences between this work and [5] are: (*i*) the use of an asymptotic fouling model, and (*ii*) identical heat capacity flow rates. The clean heat duties are comparable. It should be noted that Oldani *et al.*'s experiments employed bench scale units, and their results are assumed to apply at a larger scale. This is expected to be valid if the processing conditions and conditions at the heat transfer interface are comparable [11]. Processing conditions include the nature and source of the foulant, additives, bulk temperature, flow velocity and flow regime. Importance surface factors include the local temperature, surface energy, roughness, topography, and nucleation sites.

An uncoated stainless steel (SS) unit and a coated unit are compared. In addition, because fluoropolymers provide good corrosion resistance, a coated carbon steel (CS) unit will be considered. Carbon steel is generally cheaper and conducts heat better than SS: this could compensate for the additional thermal conductivity associated with the coating [5]. The different conductivities of the wall material and the coating are included in the evaluation of U_{cl} via:

$$U_{cl} = \left(1/h_i + \delta_{coat}/\lambda_{coat} + r_i/\lambda_{wall}\log(r_o/r_i) + r_i/(r_o h_o)\right)^{-1} \quad . \qquad (21)$$

Here h_i is the internal and h_o the external film heat transfer coefficients, δ_{coat} the coating thickness, and λ_{coat} and λ_{wall} are the coating and tube wall thermal conductivities, respectively. The internal and external radii of the tube are r_i and r_o. For the uncoated unit, δ_{coat} is zero. To achieve the specified clean heat duty, the coated unit will require a different heat transfer surface area, which is calculated from the definition of *NTU*, *i.e.* $A_{coat} = AU_{cl}/U_{cl,coat}$.

4. Results and Discussion

4.1 Case study: Quantifying the financial attractiveness of a PFPE coating

The effect of operating period length on the time-averaged operating cost, calculated using Equation (11), as well as the thermal cost penalties, for the uncoated and coated SS units are presented in Figure 4. The optimal time to operate the uncoated unit before cleaning is 64 days and for the coated SS unit it is 100 days. It can be seen that the minimum in ϕ_{op} is not symmetrical, so that the penalty for cleaning early is slightly larger than that for cleaning later. The optimised operating costs and other performance indices are summarised in Table 2.

The difference in ϕ_{op} values (285 \$ day^{-1} *cf.* 191 \$ day^{-1}) indicates that the antifouling coating gives an appreciable benefit in operating cost. A holistic approach means that the capital cost has also to be considered. Excluding the capital cost of the coating, the capital cost of the base unit, C_{cap}, is calculated and expressed as an amortised cost, ϕ_{cap}, by assuming straight line depreciation over the unit (or coating) lifetime. The heat exchangers differ in base material, heat transfer area and coating. According to Hewitt *et al.* [12], a 500 m^2 CS heat exchanger cost approximately 80 GBP m^{-2} in 1994. Conversion into US\$ and updating it with the chemical engineering plant index to December 2013 yields an installed cost of 193 US\$ m^{-2}. A SS heat exchanger with this area is roughly twice as expensive [12].

We now consider a coated SS and a coated CS unit. Both of the coated exchangers have a smaller clean heat transfer coefficient and require a larger heat transfer area. The higher thermal conductivity of CS almost compensates for the heat transfer resistance introduced by the coating. The reduction in fouling resistance of the coated units results in longer processing periods, by up to 56%. This is desirable, because cleaning of the heat exchanger leads to a reduction in product throughput, consumption and disposal of chemicals and waste product. The total averaged costs of the coated SS and coated CS heat exchangers are 28% and 35% lower than the reference (uncoated) case, respectively. Both of the coated units transfer about 32% more heat over a year than the uncoated unit. This heat does not need to be provided elsewhere, *e.g.* in a furnace. Assuming straight line depreciation period of ten years, the maximum price of the coating per unit area ranges from (*i*) 642 to 844 US$ m^{-2}, for a greenfield application, where the unit is new, and (*ii*) 272 to 464 US$ m^{-2} for a revamp. This sum is the 'value price' and represents the maximum benefit which needs to be shared between the operator and the coating vendor. If the coating cannot be provided at this price or less there would be no incentive for the operator to install such a unit.

It is important to note that ϕ_{cap} is inversely proportional to the unit (or coating) lifetime, t_{lf}. In practice, the coating lifetime is likely to set t_{lf} and it can be seen that this parameter affects the techno-economic calculation strongly. A simple finding is that $t_{\mathrm{lf}} \geq t_{\mathrm{opt}}$: the coating should last at least as long as the operating period. When $t_{\mathrm{lf}} \approx t_{\mathrm{opt}}$, one can be considering renewable coatings, *i.e.* ones which are applied regularly, such as at the completion of a cleaning operation before the unit is put back on-line.

The *NTU* value for this case study unit is about 1: it is close to the *NTU* of the exchangers used to generate the fouling data, but it is not well designed. With this low *NTU*, the a_4 parameter for the uncoated unit is < 2.8, which was the criterion for accurate estimation using the approximate scheduling approach. However, Table 2 reports that scheduling the exchanger with the simplified method results in a difference in operating costs of only 1%. Considering the potential error in the parameters involved, particularly in predicting the fouling rate, this is not a significant difference. The approximate analysis is more readily calculable and suitable for initial estimates.

4.2 Sensitivity analysis

This techno-economic model allows one to determine whether the cost of the performance improvement provided by surface coating is justified by satisfactory financial returns via the reduction in operating cost. The main source of uncertainty in these calculations lies in the fouling kinetics. A sensitivity analysis was conducted on the two Kern-Seaton parameters, t_{f} and R_{f}^{∞}, based on the uncoated unit in Table 1.

Figure 5 shows the impact of t_{f} and R_{f}^{∞}, on the optimal operating cost. Increasing R_{f}^{∞} (more severe fouling) and reducing t_{f} (faster fouling) both increase ϕ_{op}. The dash-dot and dashed lines indicate the special cases when the characteristic fouling time constant approaches zero and infinity, respectively. If t_{f} is small, fouling occurs very quickly and cleaning is not attractive. The dash-dot line shows where the criterion $\chi < 0$ holds. In this region, the corresponding minimum in Figure 4 is shallow or practically non-existent. Figure 5 shows that the operating cost is then only dependent on four variables, given by Equation (12). In contrast, if fouling is slow, *i.e.* t_{f} tends towards infinity, cleaning is not attractive either. The figure confirms what might be an obvious result, that cleaning is therefore only economically sensible if fouling is neither too fast nor too slow. The value of this analysis is that it allows the terms 'too fast' and 'too slow' to be quantified.

A favourable effect of a coating, compared to an uncoated surface, might be that the corresponding locus in Figure 5 is moved down and to the left, close to the upper constraint ($\chi < 0$). If the anti-fouling performance is sustainable and there are no other restrictions (for instance, hygienic considerations in the food and biotechnology sectors; product changeover in the FMCG sector), cleaning facilities would not be needed for this HEX. This in turn would release capital.

5 Conclusions

The attractiveness of using anti-fouling coatings to mitigate fouling in a heat exchanger subject to asymptotic fouling behaviour has been assessed using a techno-economic analysis of the performance of the exchanger over a fouling and cleaning cycle. The methodology allows the financial attractiveness of an anti-fouling coating and the associated optimal cleaning strategy to be quantified.

For the special case of a co- or counter-current exchanger with equal heat capacity flow rates, a standard approximation allows solutions to be calculated explicitly which are very close to those obtained by numerical methods. Demonstration of the 'value pricing' concept for such an exchanger is presented for a case study on water scaling where fouling data were extracted from a recent study on PFPE coatings.

A sensitivity analysis was applied to this simple heat exchanger model, which identified regions for further investigation, and scenarios where cleaning is either not required or not justified financially.

The tools presented in Section 2.3 is limited to the configuration and flow conditions mentioned above. It also assumes that Kern-Seaton fouling behaviour applies, with constant coefficients, and that the operating conditions do not change over time. A further requirement is that $a_4 > 2.8$, which is reasonable as most practical exchanger designs feature $NTU > 1$ and $Bi_f^{\infty} < 1$. Detailed numerical simulations are required for other cases, and are discussed in [9].

Acknowledgement

A PhD studentship for OMM from Du Pont/Chemours is gratefully acknowledged.

References

[1] R.S.T. Ma, N. Epstein, Optimum cycles for falling rate processes, Can. J. Chem. Eng., 59(5), (1981) 631-633, doi: 10.1002/cjce.5450590512.

[2] E. Cosado, Model optimizes exchanger cleaning, Hydrocarbon Proc., 69, (1990) 71-76.

[3] E.M. Ishiyama, W.R. Paterson, D.I. Wilson, Ageing is important: closing the fouling-cleaning loop, Heat Transfer Eng, 35(3), (2014) 311-326 doi: 10.1080/01457632.2013.825192.

[4] T. Pogiatzis, F.J. Mergulhão, V.S. Vassiliadis, D.I. Wilson, Choosing when to clean and how to clean biofilms in heat exchangers, Heat Transfer Eng., 36, (2015) 676-684, doi: 10.1080/01457632.2015.954940.

[5] L. Gomes da Cruz, E.M. Ishiyama, C. Boxler, W.A. Augustin, S. Scholl, D.I. Wilson, Value pricing of surface coatings for mitigating heat exchanger fouling, Food Bioprod. Proc., 93, (2015) 343-363, doi: 10.1016/j.fbp.2014.05.003.

[6] D.Q. Kern, R.E. Seaton, A theoretical analysis of thermal surface fouling, Brit. Chem. Eng. 14 (5) (1959) 258.

[7] T.R. Bott, Fouling of Heat Exchangers, publ Elsevier, Amsterdam, 1995.

[8] V. Oldani, C.L.M. Bianchi, S. Biella, C. Pirola, G. Cattaneo, Use of perfluoropolyether coatings to prevent fouling on heat exchanger metal surfaces, Proc. Heat Exchanger Fouling and Cleaning Conference, Budapest, Hungary, 2013.

[9] O.M. Magens, J. Hofmans, M. Pabon and D.I. Wilson, Value pricing of antifouling coatings in heat exchangers, Proc. Heat Exchanger Fouling and Cleaning Conference, Enfield, Ireland, 2015.

[10] T. Bergman, F. Incropera, Fundamentals of Heat and Mass Transfer, Wiley, New York, 2011.

[11] T. Geddert, I. Bialuch, W. Augustin, S. Scholl, Extending the induction period of crystallization fouling through surface coating, Heat Trans Eng, 30(10-11), (2009) 868–875, doi: 10.1080/01457630902753789.

[12] G.F. Hewitt, S.J. Pugh, Approximate design and costing methods for heat exchangers, Heat Transfer Eng, 28(2), (2007) 76–86, doi: 10.1080/01457630601023229.

Nomenclature

Roman

a	group of parameters	/
A, A_{coat}	heat transfer area, coated unit	m^2
b	linear fouling rate, Equation (2)	m^2 K/J
Bi_f, Bi_f^{∞}	fouling Biot number, asymptotic value	-
C_{cap}	capital cost of the base unit	US$
C_{cl}	cleaning cost per heat exchanger unit	US$
c_{coat}	coating price per unit area	US$/$m^2$
c_E	energy cost, per unit heat transferred	US$/J
h	film heat transfer coefficient	W/m^2 K
NTU	number of transfer units	-
Q, Q_{cl}	heat duty, clean value	W
Q_{max}	maximum feasible heat duty	W
R_f	fouling resistance	m^2 K/W
R_f^{∞}	asymptotic fouling resistance	m^2 K/W
r_i, r_o	internal, outer tube radius	m
T	temperature	K
ΔT	temperature difference	K
t	time	s
t_{ind}	induction period	s
t^*	time elapsed since induction time	s
t_f	characteristic fouling timescale	s
U, U_{cl}	overall heat transfer coefficient, clean value	W/m^2 K
v_i, v_o	internal (tube), outer (shell) stream velocity	m/s
W_{min}	lower heat capacity flow rate	W/K
W_{max}	larger heat capacity flow rate	W/K
W_{-1}	lambert function	-

Greek

δ_{coat}	coating thickness	m
ε	effectiveness	-
ϕ_{op}	annualised operating cost	US$/day
ϕ_{cap}	amortised capital cost	US$/day
ϕ_{T}	total annualised cost	US$/day
λ_{coat}	thermal conductivity, coating	W/m K
λ_{wall}	thermal conductivity, wall	W/m K
η	relative approximation error	-
τ	time taken for cleaning	day
χ	group of terms, Equation (19)	s

Tables

Table 1: Case Study Parameters. Source: experiments [8]; *case study II in* [5]

Design	A	*Heat transfer surface area*[1]	*500* m^2
	r_i	*Tube external radius*[2]	*5* mm
	r_i	*Tube internal radius*[2]	*3* mm
Thermal properties	c_p	*Cold and hot stream heat capacity*[2]	*4180* J kg^{-1} K^{-1}
	λ_{ss}	*Stainless steel thermal conductivity*[1]	*16* W m^{-1} K^{-1}
	λ_{cs}	*Carbon steel thermal conductivity*[1]	*54* W m^{-1} K^{-1}
	Q_{cl}	*Clean heat duty*	*2.29* MW
Fouling performance	t_{ind}	*Induction period*[2]	*0* day
	$t_{ind,coat}$	*Induction period, coated unit*[2]	*0* day
	R_f^{∞}	*Asymptotic fouling resistance*[2*]	$6.70 \cdot 10^{-3}$ m^2 K W^{-1}
	$R_{f,coat}^{\infty}$	*Asympt. fouling resistance of coated unit*[2*]	$2.94 \cdot 10^{-3}$ m^2 K W^{-1}
	t_f	*Characteristic fouling timescale*[2*]	*159.4* day
	$t_{f,coat}$	*Charac. fouling timescale, coated unit*[2*]	*156.4* day
	τ	*Time taken for cleaning*[1]	*4* day
	τ_{coat}	*Time taken for cleaning, coated unit*[1]	*4* day
Coating properties	δ_{coat}	*Coating thickness*[1]	*10.5* µm
	λ_{coat}	*Coating thermal conductivity*[1]	*0.1* W m^{-1} K^{-1}
Operation	w	*Cold and hot stream mass flow*[1*]	*30* kg s^{-1}
	v_o	*Cold (shell) side stream velocity*[2]	*0.0029* m s^{-1}
	v_i	*Hot (tube) side stream velocity*[2]	*0.61* m s^{-1}
	h_o	*Cold (shell) side heat transfer coefficient*[2]	*500* W m^{-2} K^{-1}
	h_i	*Hot (tube) side heat transfer coefficient*[2]	*800* W m^{-2} K^{-1}
	U	*Clean overall heat transfer coefficient*	*?* W m^{-2} K^{-1}
	T_{cin}	*Cold stream inlet temperature*[2]	*20* °C
	T_{hin}	*Hot stream inlet temperature*[2]	*50* °C
Costs	c_E	*Cost per unit heat*[1]	$0.0057 \cdot 10^{-6}$ US$ J^{-1}
	C_{cl}	*Cleaning cost per heat exchanger unit*[1]	*4200 US$*
	t_{lf}	*Asset lifetime (depreciation)*[1]	*10* years

[1] set by the author [2] data taken from [8] [*] differs from case study in [5]

Table 2: Comparison of optimal performance indices (parameters in Table 1). t_{opt} *and* $t_{\mathrm{opt\text{-}approx}}$ *are calculated with Equation (4) using the exact and approximate heat duty, respectively.* $\phi_{\mathrm{op,opt\text{-}approx}}$ *is calculated using* $t_{\mathrm{opt\text{-}approx}}$ *in Equation (11).*

		Uncoated HEX	Coated SS HEX	Coated CS HEX	Units
Design	A	500.0	520.6	507.4	m^2
	U_{cl}	392.8	377.2	387.1	$\mathrm{W\ m^{-2}\ K^{-1}}$
	a_4	1.97	3.32	3.27	-
Investment cost	C_{cap}	192,800	200,750	97,823	US $
Optimised schedule	$t_{\mathrm{opt}} + \tau$	64 + 4	100 + 4	98 + 4	day
	$t_{\mathrm{opt\text{-}approx}} + \tau$	77 + 4	107 + 4	106 + 4	day
	$t_{\mathrm{opt}}^{*} = t_{\mathrm{opt,coat}} / t_{\mathrm{opt}}$	-	1.56	1.53	-
Time averaged cost	$\phi_{\mathrm{op,opt}}$	285.1	191.3	193.8	$\mathrm{US\$\ day^{-1}}$
	$\phi_{\mathrm{op,opt\text{-}approx}}$	288.0	191.6	194.1	$\mathrm{US\$\ day^{-1}}$
	$\phi_{\mathrm{cap}} = C_{\mathrm{cap}}/t_{\mathrm{lf}}$	52.8	55.0	26.8	$\mathrm{US\$\ day^{-1}}$
	$\phi_{\mathrm{opt}} = \phi_{\mathrm{op,opt}} + \phi_{\mathrm{cap}}$	337.9	246.3	220.6	$\mathrm{US\$\ day^{-1}}$
	$\phi_{\mathrm{opt}}^{*} = \phi_{\mathrm{opt,coat}}/\phi_{\mathrm{opt}}$	-	0.73	0.65	-
Max. coat. cost / area: new, greenfield unit	$c_{\mathrm{coat,max\text{-}new}} = (\phi_{\mathrm{opt}} - \phi_{\mathrm{opt,coat}})\, t_{\mathrm{lf}} / A_{\mathrm{coat}}$	-	642	844	$\mathrm{US\$\ m^{-2}}$
Max. coat. cost / area: revamped unit	$c_{\mathrm{coat,max\text{-}rev}} = (\phi_{\mathrm{op,opt}} - \phi_{\mathrm{opt,coat}})\, t_{\mathrm{lf}} / A_{\mathrm{coat}}$	-	272	464	$\mathrm{US\$\ m^{-2}}$
Averaged annual energy loss	E_{loss}	14.3	9.6	9.7	$\mathrm{TJ\ year^{-1}}$
	$E^{*} = E_{\mathrm{loss,coat}} / E_{\mathrm{loss}}$	-	0.67	0.68	-

List of Figure Captions

Figure 1: Schematic of the fouling-cleaning cycle in a single heat exchanger subject to fouling over time t'. Following an induction period of length t_{ind}, the duty Q falls from the clean value Q_{cl}. After operating for length t the unit is cleaned, taking time τ, and performance is restored to Q_{cl}. Grey shaded area represents energy lost.

Figure 2: Negative branch of the Lambert W function, with term from Equation (18) as argument.

Figure 3: Fouling resistance-time data for uncoated (○) and coated (●) heat exchanger reported in [8]. Loci show the fit of the Kern-Seaton model (Equation (3)) to the data, with parameters given in Table 1. Dashed lines show simultaneous 95% confidence bounds. R^2 for the coated and uncoated fits were 0.319 and 0.378, respectively.

Figure 4: Effect of processing period length on the annualised operating cost, ϕ_{op}, and the thermal cost of fouling for the uncoated and coated SS heat exchanger.

Figure 5: Impact of fouling model parameters t_{f} and R_{f}^{∞} on the optimal operating cost for the uncoated SS unit in Table 1. Dash-dot line shows where the criterion $\chi < 0$ applies. Dashed line indicates the no-fouling case. Case study units: (○) uncoated SS unit, (●) coated SS unit.

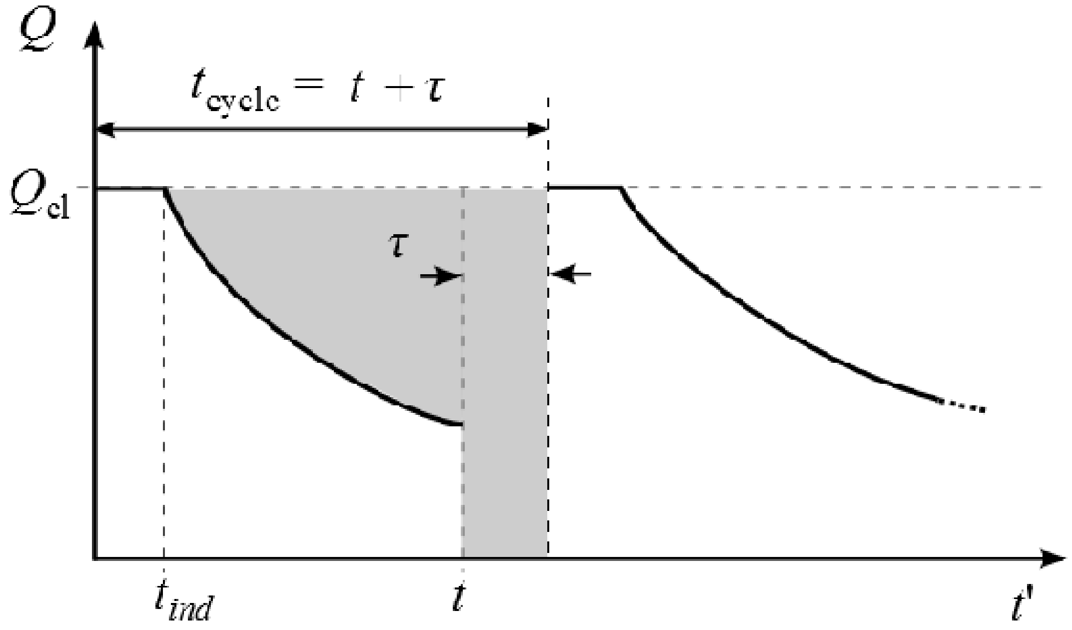

Figure 1: Schematic of the fouling-cleaning cycle in a single heat exchanger subject to fouling over time t'. Following an induction period of length t_{ind}, the duty Q falls from the clean value Q_{cl}. After operating for length t the unit is cleaned, taking time τ, and performance is restored to Q_{cl}. Grey shaded area represents energy lost.

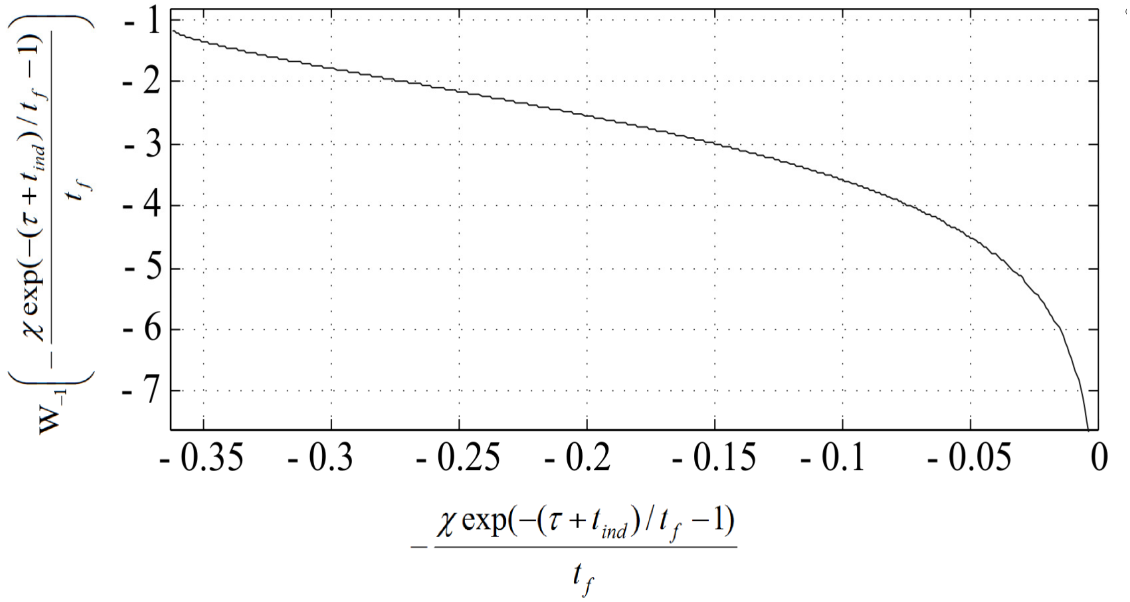

Figure 2: Negative branch of the Lambert W function, with term from Equation (18) as argument.

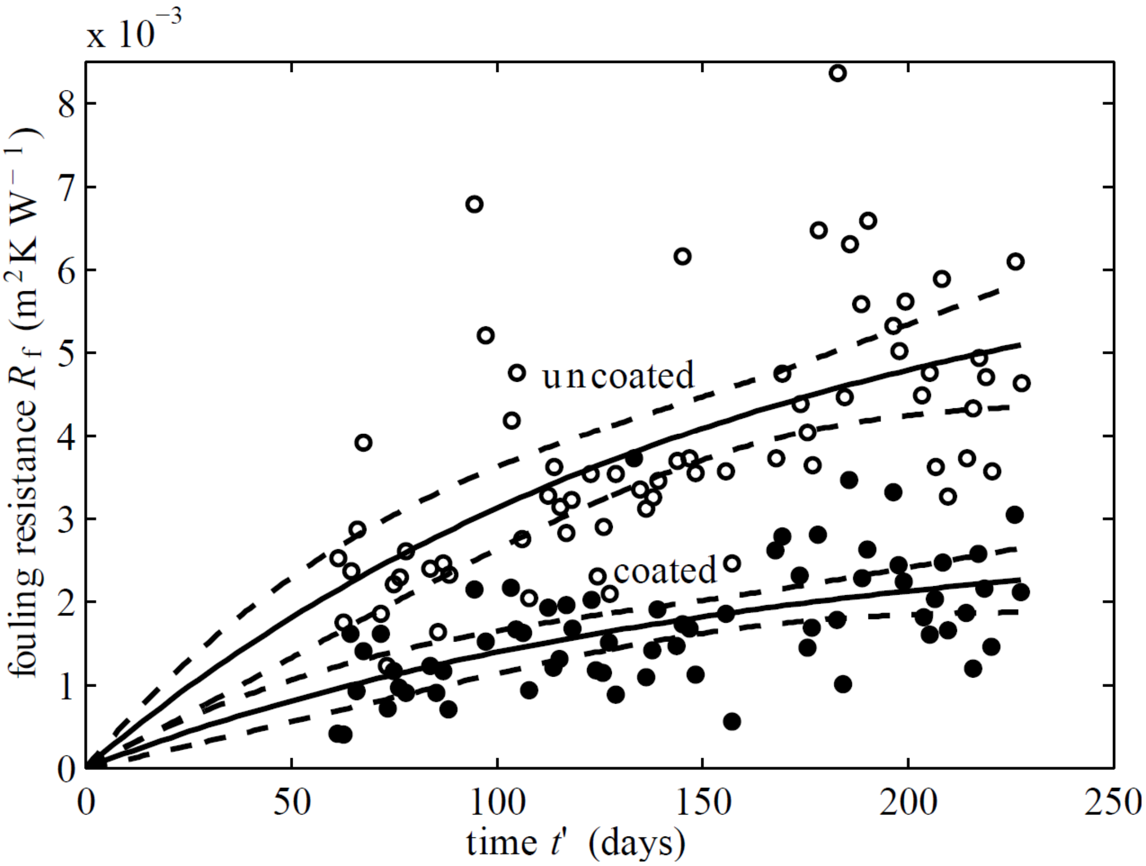

Figure 3: Fouling resistance-time data for uncoated (○) and coated (●) heat exchanger reported in [8]. Loci show the fit of the Kern-Seaton model (Equation (3)) to the data, with parameters given in Table 1. Dashed lines show simultaneous 95% confidence bounds. R^2 for the coated and uncoated fits were 0.319 and 0.378, respectively.

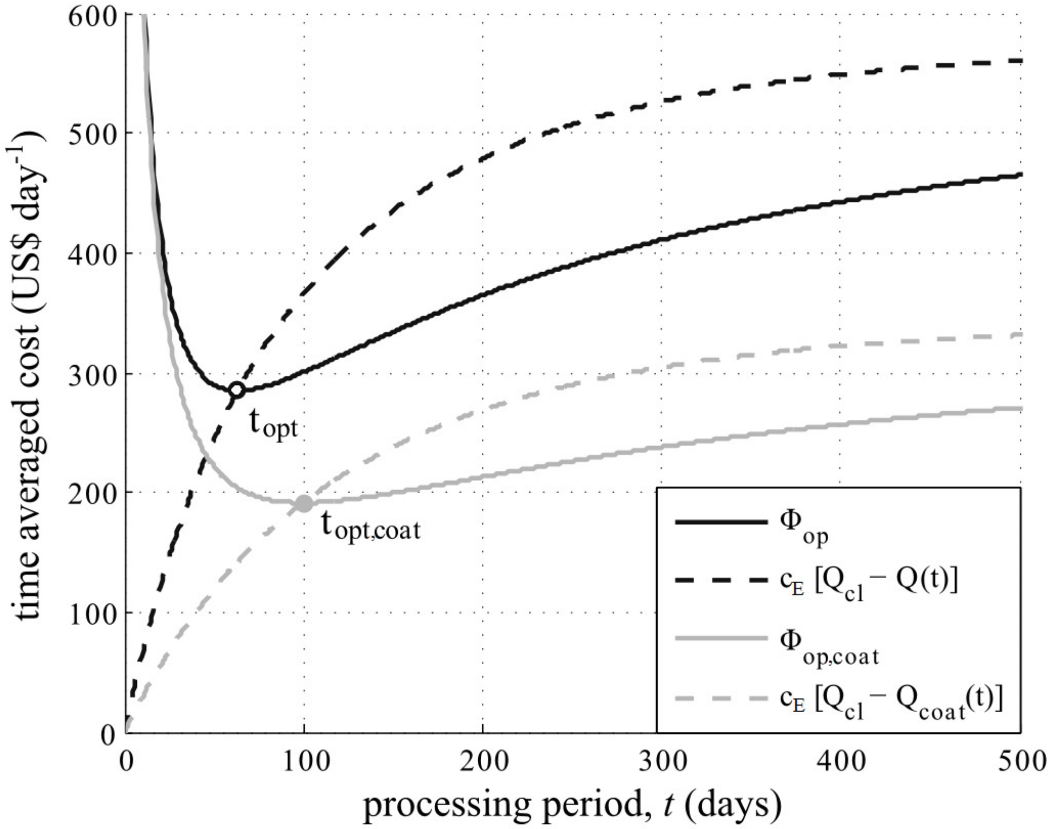

Figure 4: Effect of processing period length on the annualised operating cost, ϕ_{op}, and the thermal cost of fouling for the uncoated and coated SS heat exchanger.

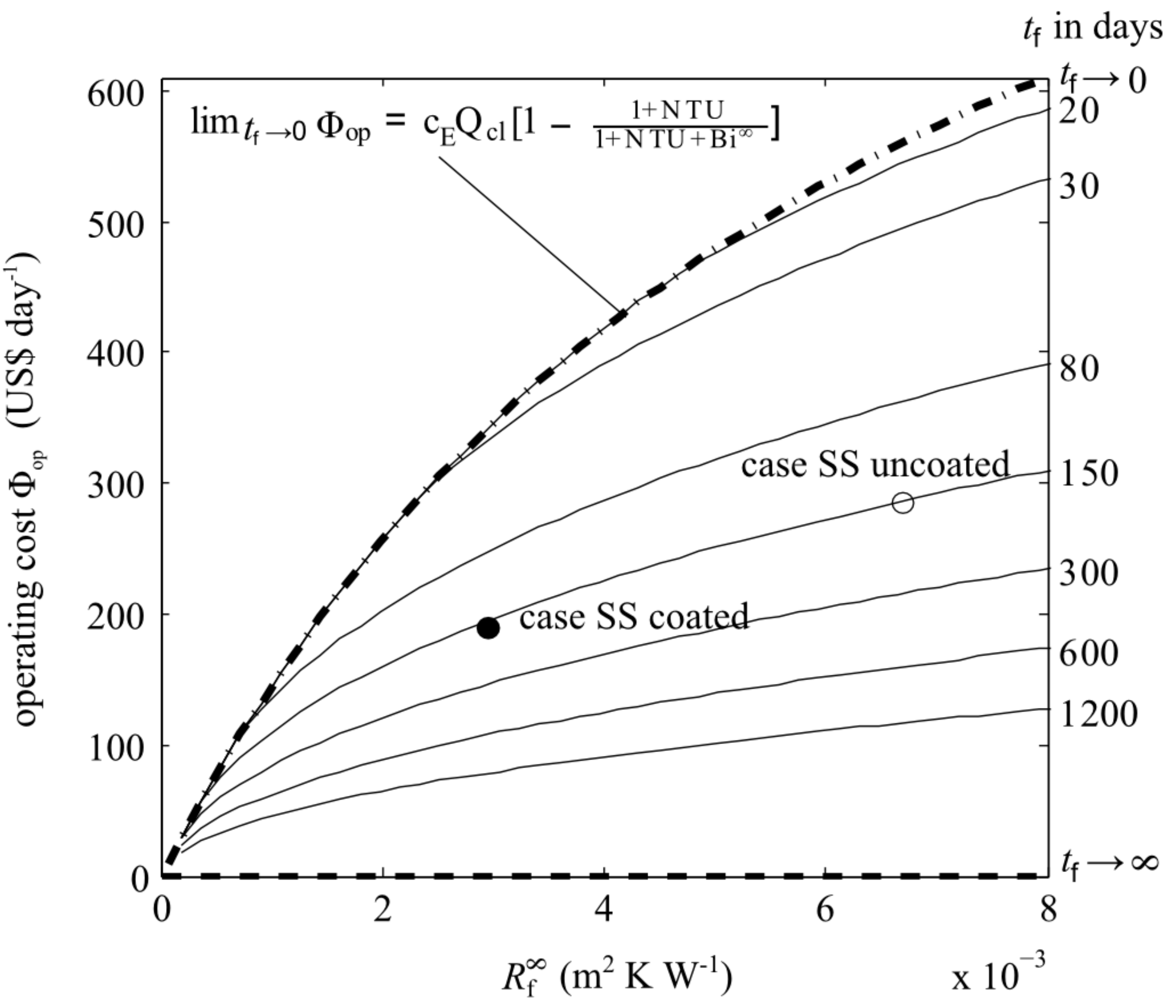

Figure 5: Impact of fouling model parameters t_f and R_f^{∞} on the optimal operating cost for the uncoated SS unit in Table 1. Dash-dot line shows where the criterion $\chi < 0$ applies. Dashed line indicates the no-fouling case. Case study units: (○) uncoated SS unit, (●) coated SS unit.