

1 Quantifying the ‘implementation gap’ for antifouling coatings

2 *Short title:* Implementation gap for antifouling coating

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7 **Abstract**

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

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

20 **1 Introduction**

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

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

35

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

$$40 \quad \phi_{op} = \frac{c_E \left[\int_0^t (Q_{cl} - Q) dt' + Q_{cl} \tau \right] + C_{cl}}{t + \tau}, \quad (1)$$

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

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

57 These financial considerations – which can include CO₂ taxes – effectively set bounds on the price of
 58 antifouling coatings, determined by comparing manufacturing costs and the maximum saving that can
 59 be achieved from fouling mitigation, in a ‘value pricing’ calculation. Order of magnitude estimates for
 60 different applications can establish the potential attractiveness of antifouling coatings for a given

61 scenario. This concept was outlined by Gomes da Cruz *et al.* [5], who applied it to three cases with
 62 different operating and cost bases. They assumed simple fouling behaviour, *i.e.* where the fouling
 63 resistance, R_f , increased linearly with time at constant fouling rate b , *viz.*

$$64 \quad R_f = \frac{1}{U} - \frac{1}{U_{cl}} = b \cdot t' \quad , \quad (2)$$

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

$$71 \quad R_f = \begin{cases} 0, & t' < t_{ind} \\ R_f^\infty (1 - \exp(-(t_{ind} - t')/t_f)) & t' \geq t_{ind} \end{cases} = R_f^\infty (1 - \exp(-t^*/t_f)) \quad . \quad (3)$$

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

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

85

86 2 Modelling and analysis

87 2.1 The operating cost

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

$$92 \quad \phi_{op}(t_{opt}) = c_E (Q_{cl} - Q(t_{opt})) . \quad (4)$$

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

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

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

104 2.2 Impact of fouling in a simple heat exchanger

105 We consider an individual heat exchanger, rather than one in a network. Its thermal performance is
106 modelled using a lumped parameter approach for the purposes of illustration: more detailed models
107 could be employed as required. An implementation of a one-dimensional model which incorporates
108 spatial resolution along the exchanger has been reported by Magens *et al.* [9]. Here, the instantaneous
109 heat duty, $Q(t)$ is calculated using the *NTU*-effectiveness method [10]. Equations (2) and (3) are
110 combined to give the overall heat transfer coefficient, U , *viz.*

$$111 \quad U = \frac{1/R_f^\infty}{1/(U_{cl}R_f^\infty) + 1 - \exp(-t^*/t_f)} = \frac{a_1}{a_2 + 1 - \exp(-t^*/t_f)} . \quad (5)$$

112 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
113 $Bi_f^\infty = U_{cl}R_f^\infty$. The number of heat transfer units of the heat exchanger, *NTU*, is given by

$$114 \quad NTU = \frac{UA}{W_{min}} = \frac{a_1 A}{W_{min} (a_2 + 1 - \exp(-t^*/t_f))} , \quad (6)$$

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

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

124

125 The effectiveness, ε , is given by the simple relationship

$$126 \quad \varepsilon = \frac{NTU}{1 + NTU} = \frac{a_1 A}{a_1 A + W(a_2 + 1 - \exp(-t^*/t_f))} \quad (7)$$

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

$$129 \quad Q = \varepsilon Q_{\max} = \left[\frac{a_1 A}{a_1 A + W(a_2 + 1 - \exp(-t^*/t_f))} \right] W \Delta T_{\max} \quad (8)$$

130 This can be written as

$$131 \quad Q = \frac{a_3}{a_4 - \exp(-t^*/t_f)} \quad (9)$$

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

$$133 \quad a_3 = a_1 A \Delta T_{\max} = \frac{A \Delta T_{\max}}{R_f^\infty} = Q_{cl} \frac{1 + NTU}{U_{cl} R_f^\infty} = Q_{cl} \frac{1 + NTU}{Bi_f^\infty} \quad (9)$$

134 and

$$135 \quad a_4 = a_2 + \frac{a_1 A}{W} + 1 = \frac{1}{U_{cl} R_f^\infty} + \frac{A}{R_f^\infty W} + 1 = \frac{1}{Bi_f^\infty} (1 + NTU + Bi_f^\infty) \quad (10)$$

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

$$139 \quad \phi_{op}(t) = \frac{c_E \left[\frac{a_3}{a_4 - 1} (t - t_{ind} + \tau) - \frac{a_3}{a_4} t_f \log \left(\frac{1 - a_4 \exp((t - t_{ind})/t_f)}{1 - a_4} \right) \right] + C_{cl}}{t + \tau} \quad (11)$$

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

$$142 \quad \lim_{t \rightarrow \infty} \phi_{op} = c_E Q_{cl} \left[1 - \frac{1 + NTU}{1 + NTU + Bi_f^\infty} \right]. \quad (12)$$

143 Equation (12) shows that a large exchanger (NTU large) is less sensitive to fouling, since if $1 +$
 144 $NTU \gg Bi_f^\infty$ the term in the brackets is small.

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

149 **2.3 An explicit result for cases of equal heat capacity flow rate**

150 **2.3.1 Derivation**

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

154 The heat duty can be rewritten as

$$155 \quad Q = \frac{a_3}{a_4 - \exp(-t^*/t_f)} = \frac{a_3}{a_4 - \exp(-t^*/t_f)} - \frac{a_3}{a_4} + \frac{a_3}{a_4} = \frac{a_3}{a_4} \left[\frac{\exp(-t^*/t_f)}{a_4 - \exp(-t^*/t_f)} + 1 \right]. \quad (13)$$

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

$$157 \quad Q \approx Q_{approx} = \frac{a_3}{a_4} \left[\frac{\exp(-t^*/t_f)}{a_4 - 1} + 1 \right]. \quad (14)$$

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

$$160 \quad \eta = \frac{Q - Q_{approx}}{Q} = 1 - \left(\frac{\exp(-t^*/t_f)}{a_4 - 1} + 1 \right) \left(1 - \frac{\exp(-t^*/t_f)}{a_4} \right). \quad (15)$$

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

163
$$\eta_{\max} = \eta(t_{\eta, \max}^*) = -\frac{1}{4a_4(a_4 - 1)} . \quad (16)$$

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

167
$$a_4 = \frac{1}{Bi_f^\infty} (1 + NTU + Bi_f^\infty) > \frac{1}{2} + \sqrt{\frac{1}{4} - \frac{1}{4\eta_c}} \approx 2.8 . \quad (17)$$

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

172
$$t_{opt-approx} = -\tau - t_f \left[1 + W_{-1} \left(-\frac{\chi \exp(-(\tau + t_{ind})/t_f - 1)}{t_f} \right) \right] . \quad (18)$$

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

174
$$\chi = t_f + t_{ind} - \frac{C_{cl}}{c_e Q_{cl}} - \frac{1 + NTU}{Bi_f^\infty} \left(\tau + \frac{C_{cl}}{c_e Q_{cl}} \right) . \quad (19)$$

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

180
$$C_{cl} > \lim_{t \rightarrow \infty} c_E \left[\int_0^t Q dt' - Q(t)(t + \tau) \right] . \quad (20)$$

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

184

185 Given a countercurrent heat exchanger with identical, constant heat capacity flow rates, static inlet
 186 temperatures, the availability of fouling data and $a_4 > 2.8$, application of the approximate method is

187 straightforward. If $\chi < 0$ there is no resultant cost benefit of cleaning and the exchanger should be
 188 allowed to operate in the fouled state. In this case, the operating cost can be calculated with Equation
 189 (12). Otherwise, Equations (18) and (19) are used to schedule cleaning. The approximate optimal
 190 operating cost is then calculated by inserting the approximate solution for the optimal processing
 191 period, $t_{\text{opt-approx}}$, in Equation (11). This methodology enables researchers and practitioners to estimate
 192 the economical value of anti-fouling coatings in heat exchangers without employing involved
 193 numerical techniques such as described in [5].

194 **3 Illustrative case study**

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

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

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

$$219 \quad U_{cl} = (1/h_i + \delta_{coat} / \lambda_{coat} + r_i / \lambda_{wall} \log(r_o / r_i) + r_i / (r_o h_o))^{-1} . \quad (21)$$

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

225 4. Results and Discussion

226 4.1 Case study: Quantifying the financial attractiveness of a PFPE coating

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

233 The difference in ϕ_{op} values (285 \$ day⁻¹ *cf.* 191 \$ day⁻¹) indicates that the antifouling coating gives an
234 appreciable benefit in operating cost. A holistic approach means that the capital cost has also to be
235 considered. Excluding the capital cost of the coating, the capital cost of the base unit, C_{cap} , is
236 calculated and expressed as an amortised cost, ϕ_{cap} , by assuming straight line depreciation over the
237 unit (or coating) lifetime. The heat exchangers differ in base material, heat transfer area and coating.
238 According to Hewitt *et al.* [12], a 500 m² CS heat exchanger cost approximately 80 GBP m⁻² in 1994.
239 Conversion into US\$ and updating it with the chemical engineering plant index to December 2013
240 yields an installed cost of 193 US\$ m⁻². A SS heat exchanger with this area is roughly twice as
241 expensive [12].

242 We now consider a coated SS and a coated CS unit. Both of the coated exchangers have a smaller
243 clean heat transfer coefficient and require a larger heat transfer area. The higher thermal conductivity
244 of CS almost compensates for the heat transfer resistance introduced by the coating. The reduction in
245 fouling resistance of the coated units results in longer processing periods, by up to 56%. This is
246 desirable, because cleaning of the heat exchanger leads to a reduction in product throughput,
247 consumption and disposal of chemicals and waste product. The total averaged costs of the coated SS
248 and coated CS heat exchangers are 28% and 35% lower than the reference (uncoated) case,
249 respectively. Both of the coated units transfer about 32% more heat over a year than the uncoated unit.
250 This heat does not need to be provided elsewhere, *e.g.* in a furnace. Assuming straight line

251 depreciation period of ten years, the maximum price of the coating per unit area ranges from (i) 642 to
252 844 US\$ m⁻², for a greenfield application, where the unit is new, and (ii) 272 to 464 US\$ m⁻² for a
253 revamp. This sum is the ‘value price’ and represents the maximum benefit which needs to be shared
254 between the operator and the coating vendor. If the coating cannot be provided at this price or less
255 there would be no incentive for the operator to install such a unit.

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

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

268 4.2 Sensitivity analysis

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

274 Figure 5 shows the impact of t_{f} and R_{f}^{∞} , on the optimal operating cost. Increasing R_{f}^{∞} (more severe
275 fouling) and reducing t_{f} (faster fouling) both increase ϕ_{op} . The dash-dot and dashed lines indicate the
276 special cases when the characteristic fouling time constant approaches zero and infinity, respectively.
277 If t_{f} is small, fouling occurs very quickly and cleaning is not attractive. The dash-dot line shows where
278 the criterion $\chi < 0$ holds. In this region, the corresponding minimum in Figure 4 is shallow or
279 practically non-existent. Figure 5 shows that the operating cost is then only dependent on four
280 variables, given by Equation (12). In contrast, if fouling is slow, *i.e.* t_{f} tends towards infinity, cleaning
281 is not attractive either. The figure confirms what might be an obvious result, that cleaning is therefore

282 only economically sensible if fouling is neither too fast nor too slow. The value of this analysis is that
283 it allows the terms ‘too fast’ and ‘too slow’ to be quantified.

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

289 **5 Conclusions**

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

294

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

300

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

303

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

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311

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- 339

340 **Nomenclature**

341 **Roman**

a	group of parameters	/
A, A_{coat}	heat transfer area, coated unit	m^2
b	linear fouling rate, Equation (2)	$\text{m}^2 \text{K/J}$
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	$\text{W/m}^2 \text{K}$
NTU	number of transfer units	-
Q, Q_{cl}	heat duty, clean value	W
Q_{max}	maximum feasible heat duty	W
R_f	fouling resistance	$\text{m}^2 \text{K/W}$
R_f^∞	asymptotic fouling resistance	$\text{m}^2 \text{K/W}$
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	$\text{W/m}^2 \text{K}$
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	-

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343

344 **Greek**

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δ_{coat}	coating thickness	m
ε	effectiveness	-
ϕ_{op}	annualised operating cost	US\$/day
ϕ_{cap}	amortised capital cost	US\$/day
ϕ_{Γ}	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

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347

348 **Tables**349 *Table 1: Case Study Parameters. Source: experiments [8]; case study II in [5]*

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

350 ¹ set by the author² data taken from [8]

* differs from case study in [5]

351

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

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

355
356

357 **List of Figure Captions**

358

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

363

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

365

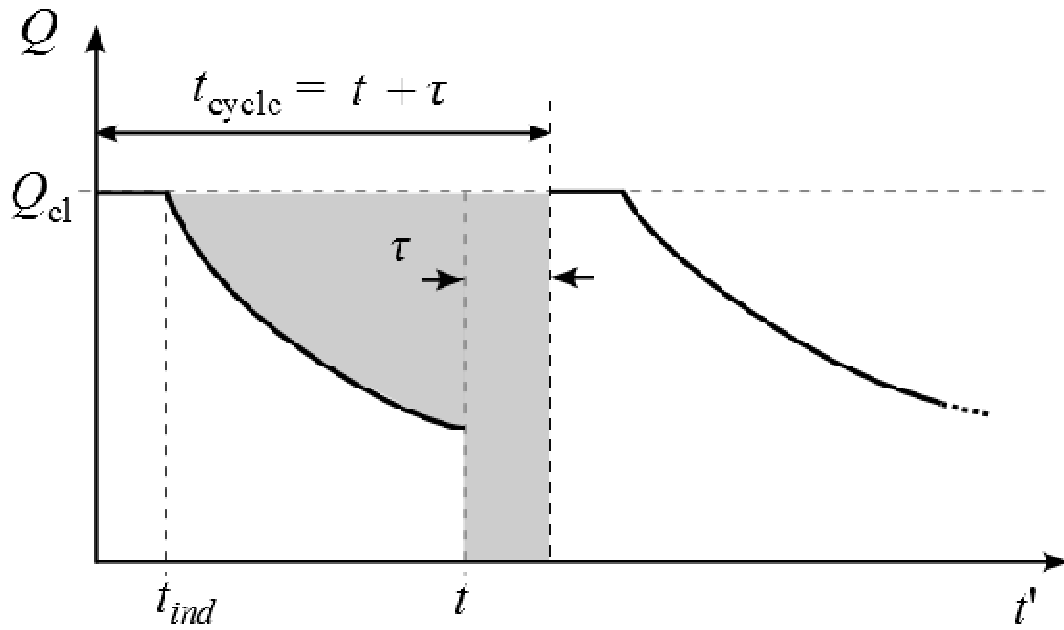
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367 Loci show the fit of the Kern-Seaton model (Equation (3)) to the data, with parameters given
368 in Table 1. Dashed lines show simultaneous 95% confidence bounds. R^2 for the coated and
369 uncoated fits were 0.319 and 0.378, respectively.

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

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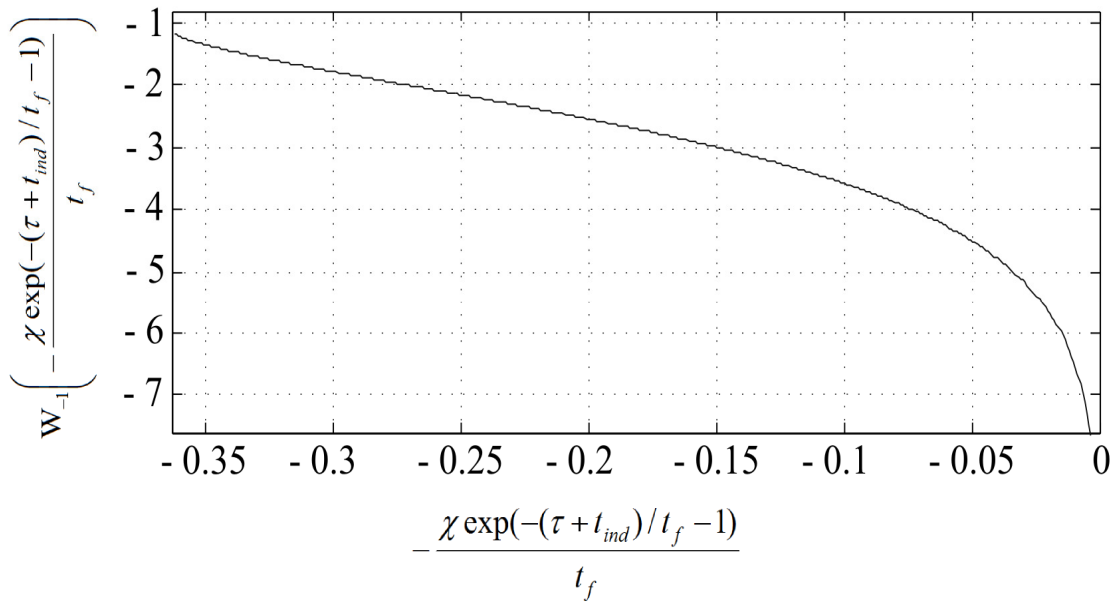
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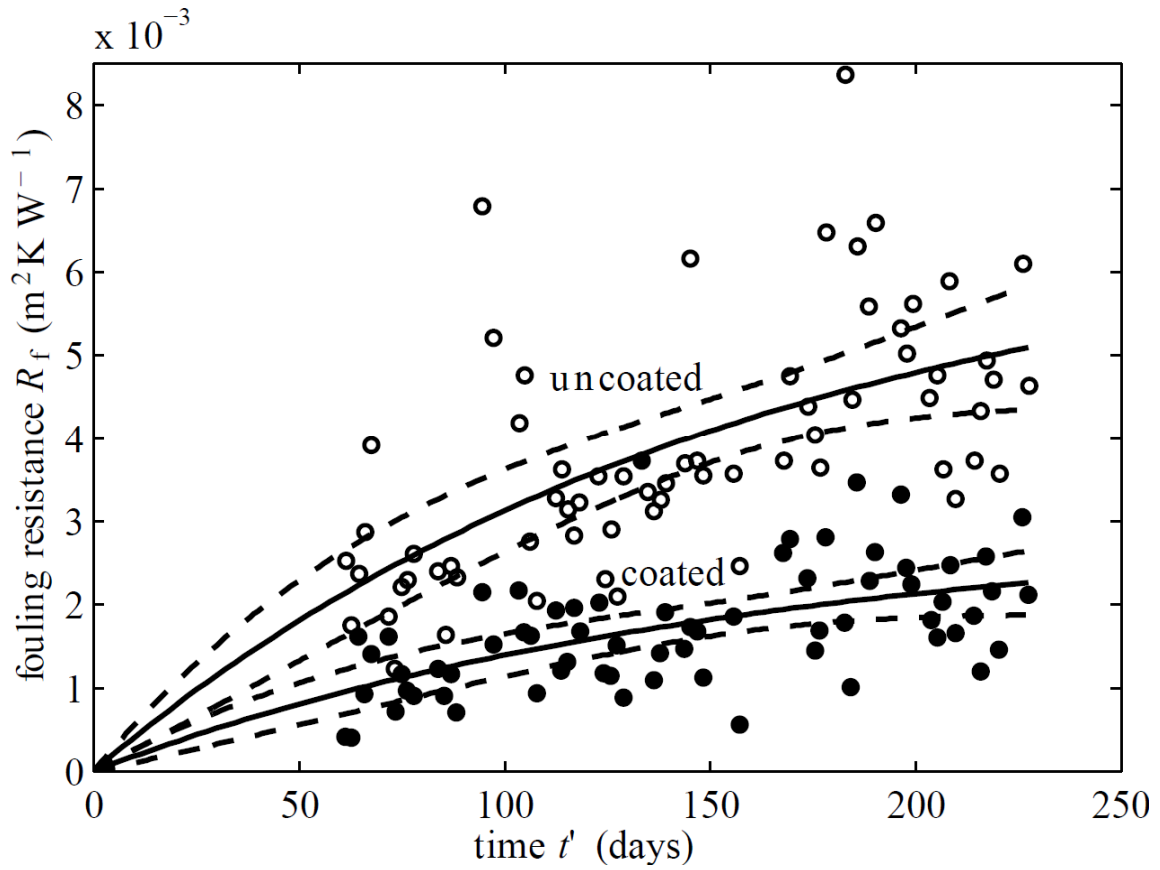
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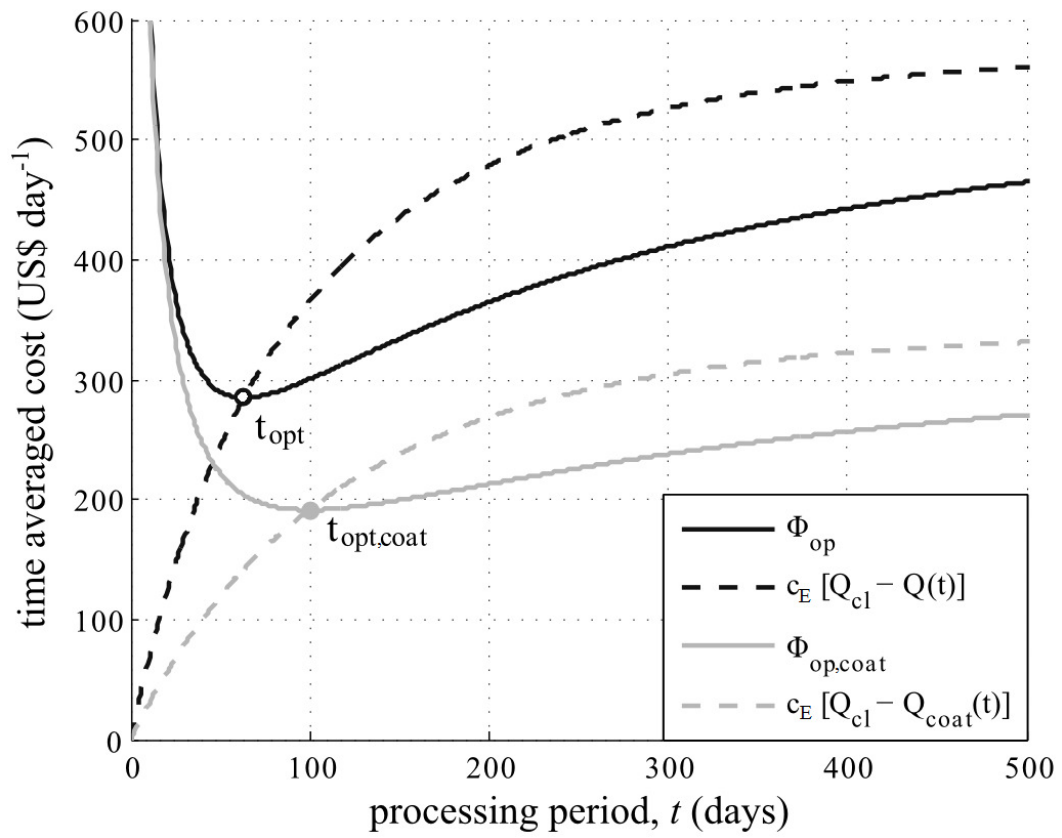
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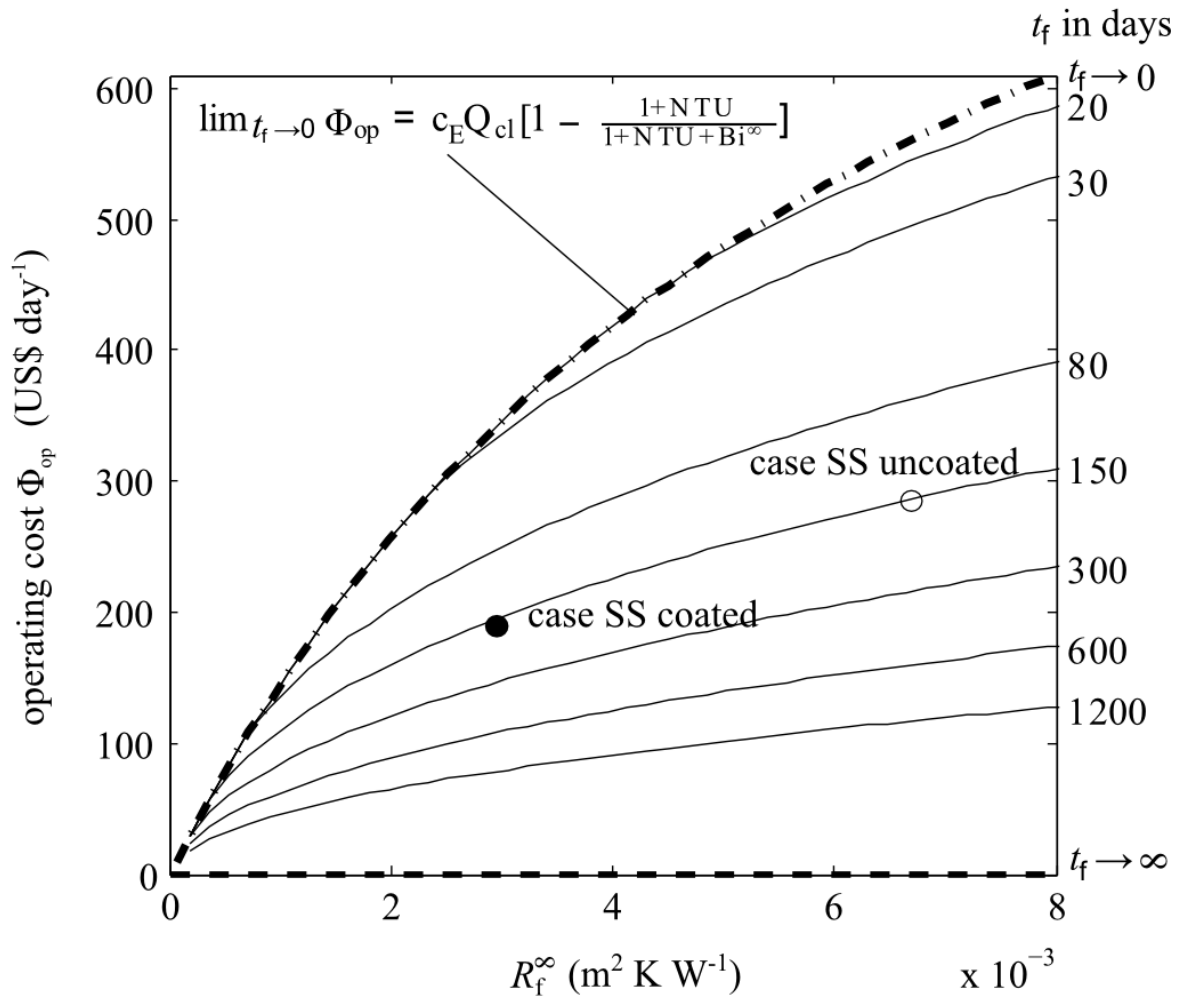
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