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FOULING DETECTION IN A CROSS-FLOW HEAT EXCHANGER BASED ON PHYSICAL MODELING

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ABSTRACT

The aim of this paper is to derive an off-line method for detecting fouling in a cross-flow heat exchanger by using measurements that are gathered in normal operation of the heat exchanger. Measured values are the inlet and outlet temperatures and the mass flow of the hot and cold fluid. The model is derived by satisfying physical energy balance equations and chosen empirical heat transfer equations.

In the method, the heat exchanger is divided into equal number of sections on each side (hot and cold side) where the temperatures in these sections represent the states of the model. The model is finally written in a state space form. The model parameters are then estimated by a least squares method, with the help of state estimates from a standard Kalman filter, i.e. the temperature in each section. Fouling detection is done by monitoring the model parameters that are a priori known to depend on the heat transfer and hence the fouling.

Because of difficulty to access data from a cross-flow heat exchanger, simulated data was used in this study. Using simulated data has the advantage that it is possible to know exactly when the fouling begins, which gives the opportunity to compare the fouling detection of the model to the actual fouling in the data from the simulation.

The conclusion is that it is possible to detect fouling in cross-flow heat exchanger with the derived method, with a reasonable accuracy and consistency.

INTRODUCTION

The dynamics of heat exchangers can be described by physical laws concerning mass, energy and momentum. By using those laws the heat exchanger can be modeled with physical equations that depend on the mass flows, inlet and outlet temperatures of the fluids that go through the heat exchangers. It is also possible to model other factors that can influence the effectiveness of the heat exchanger, for example heat transfer to or from the surroundings.

When heat exchangers are in use it is always a possibility that fouling will occur, so that the metal that is placed between the hot and cold fluids in the heat exchanger a) accumulates deposits from the fluids, b) builds up biofilm and c) starts to corrode. As fouling accumulates the overall heat transfer coefficient decreases in most cases, which results in a less efficient heat exchanger and consequently increased energy cost. Studies have shown that the total cost

of fouling in heat exchangers in highly industrialized countries are approximately 0.25% of gross national production, (Nejim et al., 1999), (Steinhagen et al., 1993).

Fouling has also impacts on the environment since an increased resistance from fouling increases the power consumption of the heat exchanger. For instance in a 550 MW coal-fired power plant a fouling biofilm of 200µm resulted in an increase of about 12 tons of CO₂ per day (Casanueva-Robles and Bott, 2005).

Due to the cost and environmental issues introduced with fouling it is preferred to take steps to detect or reduce fouling if possible. Fouling mitigation techniques include, (Sanaye and Niroomand, 2007) a) reducing rate of fouling by adding chemical materials into the flow, b) increasing the heat transfer area of the heat exchanger and c) clean the heat exchanger when it gets fouled.

Research of fouling, both causes and detection, has been studied and still is extensively studied. Studies of causes of fouling are for example (Ramachandra et al., 2005) and (Nejim et al., 1999) where modifications of the heat transfer surface is studied and ion implantation of fluorine and silicon ion into the heater alloys are studied respectively. Studies of detection of fouling are for example (Jerónimo et al., 1997) and (Nema et al., 2005) where the thermal efficiency and temperature drop in the outlets are studied respectively. Research of fouling in heat exchangers is a challenge and conferences on the subject are regularly organized.

The classical detection methods are, (Jonsson et al., 2007) a) examination of heat transfer coefficient, b) simultaneous observations of pressure drops and mass flow rates, c) temperature measurements, d) ultrasonic or electrical measurements and e) weighing of heat exchanger plates.

All of those detection methods have faults, for example methods a-c) require that the system has shown steady state behavior for some period of time, method d) can only detect local fouling and method e) requires the process to be stopped. It can be costly and therefore uneconomical to apply these methods.

Another approach is to model the heat exchanger and look for discrepancy between the model predictions and what is actually measured. The aim of this paper is to show how a physical state space model can be used to detect fouling in a cross-flow heat exchanger. The model will be based on the mass flow rates and the inlet/outlet temperatures.

In cases like heat exchangers where it is not possible to observe the model states of interest, the Kalman filter can be used to estimate the unobservable states by using the measurable states. That is by using the inputs and outputs of the process. The Kalman filter has been used in many different areas of studies where the state of interest is not observable. As an example, in (Jonsdottir et al., 2006) the Kalman filter was used in flood forecasting for the river Fnjóská in Northern Iceland. In that study the inputs to the model were precipitation, either snow or rain, and the output was the water flow in the river. The states of the models were the water that was bound in the snow and the water on the watershed of the river.

THE MODEL

The model is based on dividing the heat exchanger into *ns* section on each side. The following assumptions are used a) the heat exchanger is perfectly insulated, b) there is no heat conduction in the direction of the flow in the medium between the fluids or in the fluid themselves, c) there is uniform temperature in each section of the heat exchanger and d) the specific heat capacities are constant through the heat exchanger. Using the above assumptions the differential energy balance Eq. (1) is derived for each section of the heat exchanger

$$Mc \frac{dT_{ij}(t)}{dt} = \underbrace{\dot{m}(t)c[T_{i-nj-m}(t) - T_{ij}(t)]}_{ii} \pm \underbrace{AU_{ij}(t)F\Delta T_{ij}(t)}_{iii} \quad (1)$$

where i) is the energy change in section *ij* at a given time *t*, ii) is the energy flow in the fluid in section *ij* (where *n* = 1 and *m* = 0 for the cold side, and *n* = 0 and *m* = 1 for the hot side) and iii) is the energy that is transferred from or to section *ij*. A positive sign is used for the cold side and negative sign for the hot side.

For the cross-flow heat exchanger (Cengel and Turner, 2005) recommend to use the representation of $\Delta T_{ij}(t)$ for the counter-flow heat exchanger with a correction factor *F*.

In (Jonsson et al., 1992) where counter flow heat exchanger where studied it is suggested to use arithmetic mean temperature difference since the log mean temperature difference will introduce extra nonlinearities in the model. The arithmetic mean temperature difference for counter flow heat exchanger is calculated with.

$$\Delta T_{ij}(t) = \frac{[T_{h,ij-1}(t) + T_{h,ij}(t)]}{2} - \frac{[T_{c,i-1j}(t) + T_{c,ij}(t)]}{2} \quad (2)$$

The correction factor depends on the geometry of the cross-flow heat exchanger and the inlet and outlet temperatures of the hot and cold fluid streams. In this study a heat exchanger with both fluids unmixed was studied. The correction factor *F* has a value less than or equal to 1, where the case *F* = 1 corresponds to the counter-flow heat exchanger. Tables for the correction factor are readily available and can be seen for example in (Cengel and Turner, 2005). In this study the correction factor was calculated from the equations for the number of transfer

units (NTU) method (Cengel and Turner, 2005). The heat transfer in heat exchanger is explained by

$$q = UAF\Delta T_{LMTD} \quad (3)$$

and also by

$$q = C_{\min}\Delta T(\text{minimum fluid}) \quad (4)$$

The minimum fluid is the fluid that has the minimum value of the production of massflow and specific heat. The NTU is explained by

$$NTU = \frac{UA}{C_{\min}} \quad (5)$$

By combining Eqs. (3) and (4) and inserting the relation in Eq. (5) the correction factor is found to be

$$F = \frac{\Delta T(\text{minimum fluid})}{NTU\Delta T_{LMTD}} \quad (6)$$

Since the overall heat transfer is usually unknown it is not possible to calculate the NTU directly. Therefore it is necessary to estimate the NTU from the relation between the NTU and the effectiveness. The effectiveness can be calculated with

$$\epsilon_1 = \frac{\Delta T(\text{minimum fluid})}{\Delta T(\text{max})} \quad (7)$$

and

$$\epsilon_2 = 1 - \exp\left[\frac{\exp(-NTU^{0.78}C_{\min}/C_{\max}) - 1}{C_{\min}/C_{\max}NTU^{0.22}}\right] \quad (8)$$

Since it is not possible to get explicit equations for NTU with relation to the effectiveness for heat exchanger with both fluids unmixed, a minimization algorithm was used to estimate the value of NTU by minimizing the score function $V = (\epsilon_2 - \epsilon_1)^2$ with respect to NTU.

Parametrization

By introducing the following parameters

$$\alpha(t) = \frac{FA_h U_{ij}(t)}{\dot{m}_h(t)c_h} \quad (9)$$

$$\beta(t) = \frac{FA_c U_{ij}(t)}{\dot{m}_c(t)c_c} \quad (10)$$

$$\tau_h(t) = \frac{M_h}{\dot{m}_h(t)} \quad (11)$$

$$\tau_c(t) = \frac{M_c}{\dot{m}_c(t)} \quad (12)$$

and by inserting Eqs. (9-12) into Eq. (1) the model equations for the hot and cold side respectively become

$$\frac{dT_{h,ij}(t)}{dt} = \left(1 - \frac{\alpha}{2}\right)\tau_h^{-1}T_{h,ij-1}(t) - \left(1 + \frac{\alpha}{2}\right)\tau_h^{-1}T_{h,ij}(t) + \left(\frac{\alpha}{2\tau_h}\right)T_{c,i-1j}(t) + \left(\frac{\alpha}{2\tau_h}\right)T_{h,ij}(t) \quad (13)$$

$$\frac{dT_{c,ij}(t)}{dt} = \left(1 - \frac{\beta}{2}\right) \tau_c^{-1} T_{c,i-1j}(t) - \left(1 + \frac{\beta}{2}\right) \tau_c^{-1} T_{c,ij}(t) + \left(\frac{\beta}{2\tau_c}\right) T_{h,ij-1}(t) + \left(\frac{\beta}{2\tau_c}\right) T_{h,ij}(t) \quad (14)$$

Equations (13) and (14) can be written in state space form as

$$\frac{d}{dt} \underline{T} = \mathbf{A}(\underline{\dot{m}}, \underline{\theta}, F) \underline{T} + \mathbf{B}(\underline{\dot{m}}, \underline{\theta}, F) \underline{T}_{in} \quad (15)$$

where the elements in the matrices **A** and **B** are functions of the massflows and temperatures. In this study the focus was on the case where there are 4 sections on each side of the heat exchanger. On a matrix form the model can be seen in Eq. (16). The outlet temperatures are calculated with

$$T_{out} = \mathbf{H} \underline{T}(t) = \begin{bmatrix} 0 & 0.5 & 0 & 0.5 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0.5 & 0.5 \end{bmatrix} \begin{bmatrix} T_{h,11} \\ T_{h,12} \\ T_{h,21} \\ T_{h,22} \\ T_{c,11} \\ T_{c,12} \\ T_{c,21} \\ T_{c,22} \end{bmatrix} \quad (17)$$

It is easy to increase the number of section on each side in this model. But the size of the **A** matrix will always be double the number of the section on each side. That is if there are 9 sections on each side **A** will be an 18x18 matrix.

Empirical relations in parameters

In (Jonsson and Palsson, 1991) it is pointed out that the main reason for nonlinearity in heat exchangers is the strong massflow and temperature dependence of the heat transfer coefficient. In order to account for this it is suggested to use empirical relations of the heat transfer coefficient and incorporated it in the model parameters. The coefficient can then be estimated along with the parameters over the

operating area of the heat exchanger. By using the empirical relations it is possible, according to (Jonsson and Palsson, 1991), to use a very low model order.

Since the model derived is a mass dependent model, it is assumed that the convection coefficient, h , is a function of the massflow

$$h(t) = C' \dot{m}^y(t) \quad (18)$$

It is assumed that the thermal resistance in the separating metal is negligible, this is a reasonable assumption since the metal is usually thin and has high thermal conductivity. The overall heat transfer coefficient, U , is a function of the heat coefficients with following relation

$$U^{-1} = \frac{1}{h_h} + \frac{1}{h_c} \quad (19)$$

which is only valid for flat plate heat exchanger. By assuming that Eq. (18) applies to both of the heat transfer coefficients, h_h and h_c , the overall heat transfer coefficient is given by

$$U(t) = \frac{h_h(t)h_c(t)}{h_h(t)+h_c(t)} = \frac{C'(\dot{m}_h(t)\dot{m}_c(t))^y}{(\dot{m}_h^y(t)+\dot{m}_c^y(t))} \quad (20)$$

Since the massflow is present in the denominator in all of the model parameters it is practical to normalize them by some reference massflow \dot{m}_{ref} . By using the reference massflow the overall heat transfer coefficient can be written as

$$U_{ref} = \frac{C'(\dot{m}_{h,ref}\dot{m}_{c,ref})^y}{(\dot{m}_{h,ref}^y + \dot{m}_{c,ref}^y)} \quad (21)$$

$$\frac{d}{dt} \begin{bmatrix} T_{h,11} \\ T_{h,12} \\ T_{h,21} \\ T_{h,22} \\ T_{c,11} \\ T_{c,12} \\ T_{c,21} \\ T_{c,22} \end{bmatrix} = \begin{bmatrix} -\left(\frac{1+\frac{\alpha}{2}}{\tau_h}\right) & 0 & 0 & 0 & \frac{\alpha}{2\tau_h} & 0 & 0 & 0 \\ \left(\frac{1-\frac{\alpha}{2}}{\tau_h}\right) - \left(\frac{1+\frac{\alpha}{2}}{\tau_h}\right) & 0 & 0 & 0 & \frac{\alpha}{2\tau_h} & 0 & 0 & 0 \\ 0 & 0 & -\left(\frac{1+\frac{\alpha}{2}}{\tau_h}\right) & 0 & \frac{\alpha}{2\tau_h} & 0 & \frac{\alpha}{2\tau_h} & 0 \\ 0 & 0 & \left(\frac{1-\frac{\alpha}{2}}{\tau_h}\right) - \left(\frac{1+\frac{\alpha}{2}}{\tau_h}\right) & 0 & \frac{\alpha}{2\tau_h} & 0 & \frac{\alpha}{2\tau_h} & 0 \\ \frac{\beta}{2\tau_c} & 0 & 0 & 0 & -\left(\frac{1+\frac{\beta}{2}}{\tau_c}\right) & 0 & 0 & 0 \\ \frac{\beta}{2\tau_c} & \frac{\beta}{2\tau_c} & 0 & 0 & 0 & -\left(\frac{1+\frac{\beta}{2}}{\tau_c}\right) & 0 & 0 \\ 0 & 0 & \frac{\beta}{2\tau_c} & 0 & \left(\frac{1-\frac{\beta}{2}}{\tau_c}\right) & 0 & -\left(\frac{1+\frac{\beta}{2}}{\tau_c}\right) & 0 \\ 0 & 0 & \frac{\beta}{2\tau_c} & \frac{\beta}{2\tau_c} & 0 & \left(\frac{1-\frac{\beta}{2}}{\tau_c}\right) & 0 & -\left(\frac{1+\frac{\beta}{2}}{\tau_c}\right) \end{bmatrix} \begin{bmatrix} T_{h,11} \\ T_{h,12} \\ T_{h,21} \\ T_{h,22} \\ T_{c,11} \\ T_{c,12} \\ T_{c,21} \\ T_{c,22} \end{bmatrix} + \begin{bmatrix} \left(\frac{1-\frac{\alpha}{2}}{\tau_h}\right) & 0 & \frac{\alpha}{2\tau_h} & 0 \\ 0 & 0 & 0 & \frac{\alpha}{2\tau_h} \\ 0 & \left(\frac{1-\frac{\alpha}{2}}{\tau_h}\right) & 0 & 0 \\ 0 & 0 & 0 & 0 \\ \frac{\beta}{2\tau_c} & 0 & \left(\frac{1-\frac{\beta}{2}}{\tau_c}\right) & 0 \\ 0 & 0 & 0 & \left(\frac{1-\frac{\beta}{2}}{\tau_c}\right) \\ 0 & \frac{\beta}{2\tau_c} & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} T_{h,10} \\ T_{h,20} \\ T_{c,01} \\ T_{c,02} \end{bmatrix} \quad (16)$$

After normalizing, the parameters become

$$\alpha(t) = \alpha_* \frac{\dot{m}_{h,ref} U(t)}{\dot{m}_h(t) U_{ref}} \quad (22)$$

$$\beta(t) = \beta_* \frac{\dot{m}_{c,ref} U(t)}{\dot{m}_c(t) U_{ref}} \quad (23)$$

$$\tau_h(t) = \tau_{h*} \frac{\dot{m}_{h,ref}}{\dot{m}_h(t)} \quad (24)$$

$$\tau_c(t) = \tau_{c*} \frac{\dot{m}_{c,ref}}{\dot{m}_c(t)} \quad (25)$$

where α_* , β_* , τ_{h*} and τ_{c*} are from Eqs. (9-12) respectively. The parameter set in the heat exchanger model is $\underline{\theta} = [\alpha_*, \beta_*, \tau_{h*}, \tau_{c*}, \gamma]$ where γ is from Eqs. (20) and (21).

PARAMETER ESTIMATION

The heat exchanger model presented in Eq. (15) is deterministic. To compensate for deviations from the correct temperature values it is necessary to add a noise term, $w(t)$, to the model, where $w(t) \in N(0, \mathbf{Q}(t))$, i.e. independent normal distributed white noise process with zero mean. The model then becomes

$$\frac{dT}{dt} = \mathbf{A}(\underline{\dot{m}}, \underline{\theta}, F)T + \mathbf{B}(\underline{\dot{m}}, \underline{\theta}, F)T_{in} + \underline{w} = f(\underline{\dot{m}}, \underline{\theta}, F, T, T_{in}) + \underline{w} \quad (26)$$

Since the model is continuous discretization is needed before the standard Kalman filter can be used to estimate the states of the model. By assuming that the temperatures and massflows are constant between sampling instants it is possible to write the model in discrete time as

$$T(t + \Delta t) = \Phi(\underline{\dot{m}}, \underline{\theta}, F, \Delta t)T(t) + \Gamma(\underline{\dot{m}}, \underline{\theta}, F, \Delta t)T_{in}(t) + \underline{w}(t + \Delta t) \quad (27)$$

where

$$\Phi(\underline{\dot{m}}, \underline{\theta}, F, \Delta t) = e^{\mathbf{A}(\underline{\dot{m}}, \underline{\theta}, F)\Delta t}$$

$$\Gamma(\underline{\dot{m}}, \underline{\theta}, F, \Delta t) = \left[\int_0^{\Delta t} e^{\mathbf{A}(\underline{\dot{m}}, \underline{\theta}, F)s} ds \right] \mathbf{B}(\underline{\dot{m}}, \underline{\theta}, F)$$

In the model the covariance of the noise term should be a function of the sampling interval and the massflow, but in this case the sampling time is constant and massflow dependence is neglected for simplification.

The measurement model is described in Eq. (17) with added measurement noise term, $v(t)$, where $v(t) \in N(0, \mathbf{R}(t))$, i.e. independent normal distributed white noise process with zero mean.

$$T_{out}(t) = \mathbf{H}T(t) + v(t) \quad (28)$$

The Kalman filter

During the parameter estimation the Kalman filter is used to estimate the states of the model. When the states have been estimated the model outlet temperatures are calculated from the states estimates. The aim is therefore to find the parameter set that minimize the sum of squares of

the two residuals, from the hot and the cold side. A good introduction to the Kalman filter can be found in (Welch and Bishop, 2006).

The score function

The Kalman filter is used to estimate the states of the model, that is the temperatures in the sections of the heat exchanger, which is not possible to measure. The parameters of the model are then estimated with the method of least squares and the parameter set is found by minimizing the score function Eq. (29) which is the sum of the squares of the deviations of the observed outlet temperatures and the estimated outlet temperatures, this is done by using minimization routing.

$$V(\underline{\theta}) = \sum_{t=t_1}^N \xi^T(t) \xi(t) \quad (29)$$

When estimating the parameters it is important to allow the process to tune in before the score function is calculated, if the process is not allowed to tune in there will be constant error in the estimated parameters.

To obtain an estimate of the parameter values the minimum routine, **fmincon**, in Matlab (MathWorks, 2009) was used. In addition to estimate the parameter values the function estimates the Hessian matrix of the parameters. By using the Hessian matrix it is possible to estimate the uncertainty of the parameters (Gelman et al., 2003)

$$\text{cov}(\hat{\underline{\theta}}) = \frac{2\mathbf{V}(\hat{\underline{\theta}})}{N-(t_1+p)} H^{-1} \quad (30)$$

where p is the number of estimated parameters.

FOULING DETECTION

When fouling accumulates in a heat exchanger the resistance to heat transfer increases. The increased resistance will decrease the overall heat transfer coefficient, U . It can be hard to detect changes in U in heat exchangers where there are frequent massflow changes since U is correlated with the massflow through the Reynolds number, (Cengel and Turner, 2005).

$$U = \frac{1}{\frac{1}{A_c h_c} + \frac{1}{A_h h_h}} \quad (31)$$

where

$$h = \frac{\text{Nu} k}{D_h}$$

$$\text{Nu} = C' \text{Re}^x \text{Pr}^y$$

$$\text{Re} = \frac{v_m D_h}{\nu} = \frac{\dot{m} D_h}{\rho A v}$$

Since it is not possible to observe U directly the model parameters are used to observe changes in U indirectly. By looking on Eqs. (22) and (23) it can be seen that if U decreases then α and β will also decrease. By using this relation of the parameters to U it is possible to detect fouling through a shift in the parameters.

To detect the shift in the parameters the Cumulative sum control chart (CuSum) is used. According to NIST

online engineering statistics handbook CuSum is very efficient when detecting small shift in mean of a process (NIST/SEMATECH, 2009).

When analyzing the parameters the average value of the parameters is calculated over a window of a specific length, which is chosen in accordance with the process being studied, the CuSum chart is used to monitor the average value of the parameters for a shift from their reference value. If the process is under control the cumulative sum should fluctuate around the zero, if on the other hand there is a shift in the value of the parameters the cumulative sum should shift either up- or downward depending on the shift in the parameters. In the case of fouling the parameters will decrease and it is therefore sufficient to do a one-sided test. If the cumulative sum is higher than some predefined threshold, a drift in the parameters is detected. The test is defined as follows

- a) Compute the cumulative sum:

$$\text{Cus}(i) = \max[0, \mu_0 - \bar{x}_i - K + \text{Cus}(i - 1)]$$
- b) If $\text{Cus}(i) > H^*$ then a drift is detected.

The moving average value, \bar{x} , of the process is calculated over a window of a specific length. The parameters K and H^* are used to assess if the process is going out of control.

THE DATA

Since it was difficult to get access to data from a cross-flow heat exchanger during this study a simulation program was developed to simulate the effect of fouling. The model is based on a mathematical representation of the flow, where temperature is defined as a position dependent field for both the cold and hot fluid in the exchanger. General conditions in the heat exchanger can therefore be defined as two planar functions, for the cold side and for the hot side. Figure 1 shows a graphical layout of the model with the relevant dimensions, in the x - y plane.

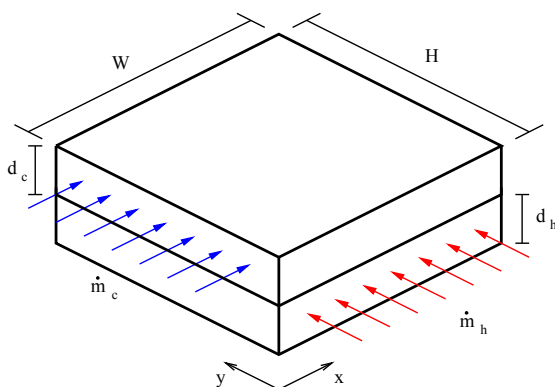


Figure 1. Graphical layout of the model

In conjunction with Figure 1, consider a plate heat exchanger with cross flow along x and y directions. The width of the exchanger in the x direction is W and the height is H . Furthermore, the thickness of the hot and cold passages are d_h and d_c , respectively. It is assumed that the cold stream travels only in the x direction and the hot stream in the y direction, so no internal mixing takes place inside

the exchanger. It is also assumed that there is no diffusion (or thermal conduction) along the fluid streams and thus only pure convection is considered.

The model mathematical representation is formulated in a computer program and solved numerically, resulting in simulated data which represent an actual heat exchanger. For details of similar modeling procedure, see (Mercere et al. 2009). The data used in this study were simulated without fouling for the first 25% of the data points and then fouling was introduced. The simulation model was validated by using steady state conditions and known theoretical solutions of such cases.

Fouling factor

Performance of heat exchangers deteriorates with time as a result of fouling. Fouling can be accumulation of mineral deposits, rust or presence of micro-organism on the heat transfer surfaces. These deposits increase the resistance of heat transfer and cause the heat transfer to decrease. The resistance because of fouling is usually represented by a fouling factor, R_f , which measures the thermal resistance introduced by fouling. The development of fouling depends on number of things, major groups of fouling dependents are (Bansal and Chen, 2005) and (Cengel and Turner, 2005) a) composition of the fluids, b) operating conditions in the heat exchanger, c) type and characteristics of the heat exchanger, d) location of fouling and e) presence of micro-organism. According to (Bansal and Chen, 2005) and (Rizzo, 2005) there is usually an induction time before a noticeable amount of mineral deposits has formed so the overall heat transfer coefficient changes noticeably. In (Bohnet, 2005) it is shown that fouling grows with increased rate during the fouling period.

During heat exchanger design the effect of fouling is addressed by designing the heat exchanger such that it can withstand the effect of fouling up to a certain point without becoming harmful for the intended process. According to (Lalot et al., 2007) typical fouling factor for water the heat exchanger can sustain is in the range $[0.0001, 0.0007]$, typical fouling factors can also be found at the (Engineering page). Fouling detection should therefore be detected on that interval. The fouling evolution used in this study can be seen in Figure 2, to save computing time the fouling factor was allowed to be maximum $R_f = 0.0004$.

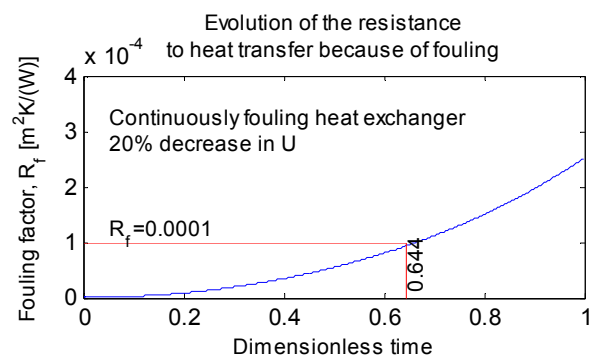


Figure 2. Evolution of the fouling factor through the time series

RESULTS

The simulated data used was of cross-flow heat exchanger with water on both sides. It was assumed that the specific heat was constant, $c_h = c_c = 4200 \left[\frac{\text{J}}{\text{kg}^\circ\text{C}} \right]$, the density of the fluids where $\rho = 998 \text{ [kg/m}^3\text{]}$. During the study the temperatures and massflows were allowed to vary randomly on certain intervals, the hot temperatures were on the interval $[54.7, 99.4]^\circ\text{C}$ and the cold temperatures $[13.3, 27.6]^\circ\text{C}$, the mass flow rates were on the interval $[0.24, 1.7]\text{kg/s}$. The parameters were varying both on the clean period as well as when fouling was occurring. The average massflows was $\dot{m}_h = \dot{m}_c = 1 \text{ [kg/s]}$. The dimensions of the simulated heat exchanger where 0.5 m wide and height and had a depth of 0.002 m for both the hot and the cold side, the reasons for these dimensions were to insure a turbulent flow in the heat exchanger.

It is possible to calculate the value of the parameters in a steady state condition. In this case the average temperatures were $T_{h,in} = 75.0^\circ\text{C}$, $T_{h,out} = 65.3^\circ\text{C}$, $T_{c,in} = 20.2^\circ\text{C}$, and $T_{c,out} = 29.9^\circ\text{C}$. The correction factor was estimated from Eq. (6) to be $F = 0.95$ and the overall heat transfer coefficient can be calculated from the relation $\dot{m}_h c_h (T_{h,in} - T_{h,out}) = AU\Delta T_{LMTD}$ to be $U = 3.62 \frac{\text{kW}}{\text{m}^2^\circ\text{C}}$. By using this information it is possible to calculate the expected value for the overall heat transfer coefficient and the expected values of the parameters using Eqs. (9-12), see Table 1.

Table 1. Calculated and estimated values of the model parameters.

Parameters	Calculated values	Estimated values	Standard deviation
θ	$E[\theta]$	$\hat{\theta}$	$\hat{\sigma}(\theta)$
α	0.13	0.112	1e-7
β	0.13	0.126	2e-7
τ_h	0.25	0.007	9.2
τ_c	0.25	0.04	0.09
y		0.8	1e-4

The literature recommends to use $y = 0.8$ for a turbulent flow (Cengel and Turner, 2005).

Table 1 shows the estimated values of the parameters using model with 4 sections on each side and U massflow dependent estimated the model parameters to be. Experiments showed that the model has problem to estimate the τ_h and τ_c parameters correctly, this can also be seen from Table 1. The reason for this is that the values of the τ_h and τ_c parameters that minimize the score function are so small that the score function is insensitive to the exact value of the parameters. To compensate for that τ_{h*} and τ_{c*} in Eqs. (24) and (25) where kept constant at their calculated values. Keeping the τ_h and τ_c parameters fixed has little concerns for the estimation of the other parameters, as can be seen in Table 2.

Table 2. Estimation of the parameters without τ_h and τ_c .

Estimated values of the parameters		
θ	$\hat{\theta}$	$\hat{\sigma}(\theta)$
α	0.1117	2e-7
β	0.1261	3e-7
y	0.8013	2e-4

In order to test if the method is sensitive to initial values, one hundred random initial values in the interval $[0, 2]$ were tried. In all cases the score function was minimized with approximately the same parameter vector, which indicates that the obtained minimum is global.

Different sampling steps

In effort to minimize computing time, analysis of sensitivity of the model to different sampling steps was done, the results indicated that the sampling time does not affect the ability of the model to estimate the model parameters.

Fouling detection

As stated above, the fouling detection process is done by estimating the model parameters over a window of specific length. In this study the window size was chosen to be 200 observations and the window was moved 20 observations ahead between estimations. Both the computing time and the variation of the parameters depend on the window size. The computing time increases with increased window size and the parameter variation will decrease with increase window size. As for the steps between windows they should be chosen with accordance with expected fouling, if fast fouling is expected small steps between adjacent windows might be reasonable. In Figure 3 typical parameter estimation over the whole dataset can be seen. The first 246 values correspond to a clean heat exchanger.

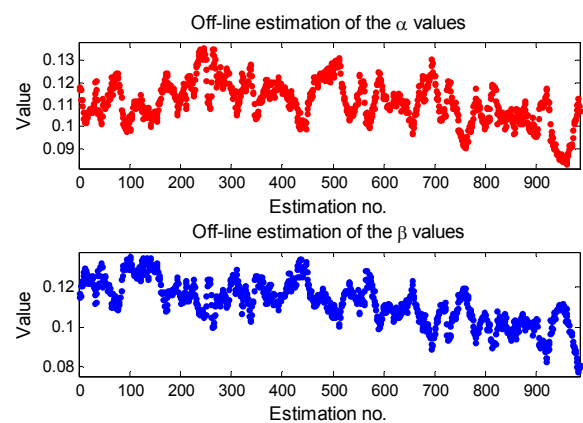


Figure 3. Evolution of the parameters through the sampling time.

As mentioned above the fouling is detected by monitoring for shifts in α and β . If their value decreases it can be a sign of accumulated fouling. To monitor for changes in the mean of the parameters the CuSum chart mentioned above is used, the parameters K and H were

chosen to be $2\sigma_{\theta, \text{clean}}$ and $3\sigma_{\theta, \text{clean}}$ respectively in this study. The CuSum chart was used to monitor the moving average of 40 values of the parameters, if too few values are used to calculate the average a possibility of Type I error on the other hand if too many values are used to calculate the average a possibility of Type II error can occur.

To evaluate the detection sensitivity 50 different time series were monitored, all with the same fouling evolution. The result shows that the average dimensionless fouling detection time was 0.77 for α and 0.76 for β . In 7 cases fouling detection was not made for either α or β and in five cases a possible wrong detection was made for one or both of the two parameters. The results for the detections can be seen in Figure 4. The dimensionless detection time on the y-axis can be compared to evolution of fouling in Figure 2. The comparison shows that detection is in most cases made close to typical lower limit of designed fouling factor.

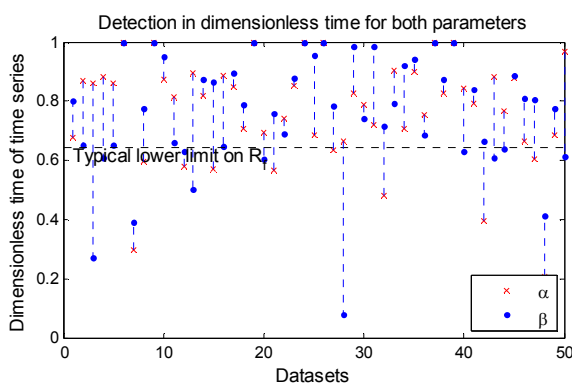


Figure 4. Dimensionless detection time for the parameters. The fouling detection on each dataset through the α and β are connected with a dotted line.

DISCUSSION

The results show that the method derived in this paper can be used to detect fouling in cross flow heat exchangers by using measurements that can be obtained under normal operation. The method is physically based and some of its parameters contain the heat transfer coefficients of the heat exchanger. Since the parameters in the model are time varying it is possible to monitor for changes in the heat transfer coefficient due to fouling. Unlike conventional methods for fouling detection this method does not need the heat exchanger to be operating in a steady state condition.

It has been shown that the method is invariant of initial values and can cope with large sampling steps. Further studies will be made by using data from real heat exchanger to validate the results.

The fouling detection is usually made in the interval of designed fouling factor of heat exchangers, as mentioned above the fouling factor indicates the maximum fouling the heat exchanger can sustain and still fulfill its operational requirements.

The results are in agreement with the results of the studies made in (Jonsson et al., 2006) and (Lalot et al., 2007) that used similar method to detect fouling in counter flow heat exchanger.

CONCLUSION

The main conclusion can be summarized to the following

1. The results show that the method can be used to detect fouling in cross-flow heat exchangers by monitoring the α and β parameters in the model for shift in their means. The means of the parameters dropped on average 18% in this study because of the fouling.
2. The method uses measurements that can be obtained under normal operation.
3. The method is invariant to initial values.
4. The method can cope with large sampling steps.

Further work will include

1. Run the method on a data from a real heat exchanger.
2. Include the metal between the fluids as model state.
3. Use specific correction factor for each section of the model.
4. Increase number of sections.
5. Change the model from being off-line to on-line.

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NOMENCLATURE

A	system matrix
<i>A</i>	surface area for heat transfer if used with subscript, m ²
B	input matrix
<i>c</i>	specific heat, kJ/kg°C
<i>C</i>	heat capacity, kW/°C, constant when used with prime
$\frac{d}{dt}$	derivative with respect to time
<i>D_h</i>	hydraulic diameter, m
<i>F</i>	correction factor
<i>h</i>	convection heat transfer coefficient, kW/m ² °C
H	measurement matrix if bold, otherwise Hessian matrix
<i>H</i>	Hessian matrix or CuSum parameter when used with *
<i>K</i>	CuSum parameter
<i>k</i>	thermal conductivity, kW/m°C
<i>M</i>	mass, kg
\dot{m}	mass flow rate, kg/s
<i>N</i>	number of measurements
NTU	number of transfer units
Nu	Nusselt number
<i>p</i>	number of estimated parameters
Pr	Prandtl number
<i>q</i>	heat transfer rate, kW
Q	covariance matrix
R	measurement noise covariance
<i>R_f</i>	fouling factor, m ² °C/W
Re	Reynolds number
<i>T</i>	temperature, °C
<i>t</i>	time, s
<i>U</i>	overall heat transfer coefficient, kW/m ² °C
<i>V</i>	score function
<i>v</i>	white Gaussian noise or kinematic viscosity, -, m ² /s
<i>V_m</i>	mean flow velocity, m/s
<i>w</i>	white Gaussian noise
<i>x</i>	exponent of Prandtl number

y	exponent of Reynolds number
α	model parameter
β	model parameter
Δ	difference
ϵ	effectiveness
Φ	discrete system matrix
θ	parameter vector
Γ	discrete input matrix
μ	mean value
ξ	residuals
ρ	density, kg/m ³
τ	model parameter, 1/s

Subscript

c	cold side
h	hot side
i	section indicator
in	inlet
j	section indicator
LMTD	log mean temperature difference
m	separating metal
min	minimum fluid
out	outlet
ref	reference

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