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COMPARISON OF FOULING DETECTION METHODS USING EXPERIMENTAL DATA

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ABSTRACT

Two heat exchangers of the plate and frame type have been tested in a DANFOSS research laboratory under fouling conditions. Flow rates, pressure drop on the fouling side, inlet and outlet temperatures are available. Three detection methods are compared. The first one is based on the study of the evolution of the dimensionless overall heat transfer coefficient. The second one is based on the evolution of the pressure drop. The third one is based on the comparison of the experimental and estimated outlet temperatures. The results obtained by two statistical tests applied to the dimensionless heat transfer coefficient are presented. A first conclusion drawn from the pressure drop evolution is that the plate pattern might have a significant influence on fouling. Finally the results obtained using the estimated values are studied. All these results show that fouling is detected quite early and that there is no "best" method.

INTRODUCTION

In the recent years, the online detection of fouling in heat exchangers by taking account of input and output data only has been addressed using various techniques. It is possible to mention LPV models (Mercère et al., 2013), wavelets (Ingimundardóttir and Lalot, 2011), neural networks (Riverol and Napolitano, 2005) (Lalot and Pálsson, 2010), a physical model (Gudmundsson et al., 2009), a subspace based method (Lalot and Mercère, 2008), fuzzy observers (Delmotte et al, 2008), and Extended Kalman filters (Jonsson et al., 2007). The common goal is to predict when a given fouling factor is reached so that it is possible to start a maintenance process. This helps managers to move from systematic or curative maintenance to predictive maintenance. This reduces the maintenance costs as mentioned in chapter 5 of (Müller-Steinhagen, 2000).

Although the data used in these works come from experiments or from validated simulators (Lalot et al., 2011), no general conclusion can be drawn concerning the relative efficiency of these methods. The fact is that none has been tested when actual fouling occurs and/or on two different heat exchangers.

The aim of the present study is to compare three methods on data recorded using heat exchangers manufactured by two companies. One of these companies is DANFOSS. For confidentiality reasons the second one is not mentioned and differences given here between the heat exchangers are limited to the fact that the plate patterns are different. Note that the exchangers will be numbered 1 and 2 without the possibility to link the number to the company.

Both heat exchangers are plate heat exchangers, having the same size and capacities. They are tested on the same test bench (Fig. 1) using the same fouling fluid. The durations of the tests are 54 and 70 hours.

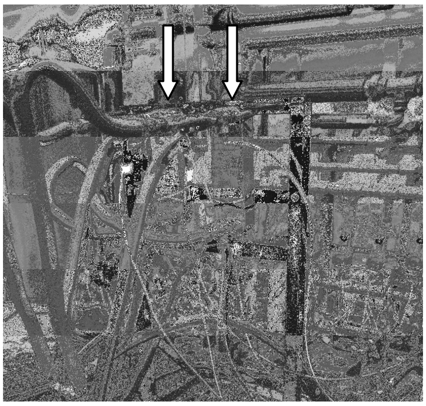

Fig. 1 Partial view of the test rig showing the heat exchangers (below the arrows)

EXPERIMENTAL VALUES

The tests have been carried out in a DANFOSS research lab. The flow rate for the first heat exchanger is given in Fig. 2 and the temperatures in Fig. 3.

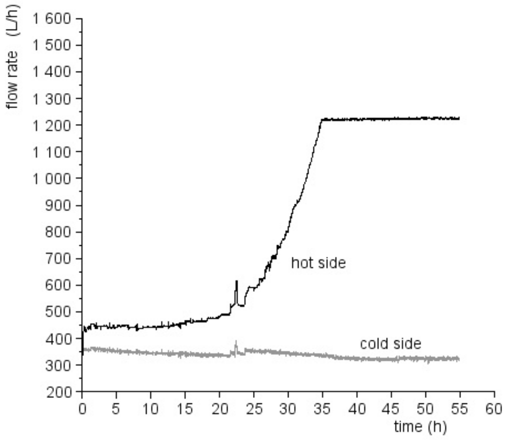

Fig. 2 Flow rate for heat exchanger #1

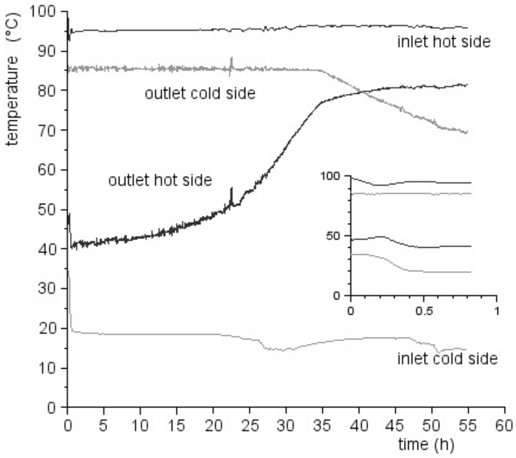

Fig. 3 Inlet and outlet temperatures for heat exchanger #1

Figures 4 and 5 give the same data for the second heat exchanger.

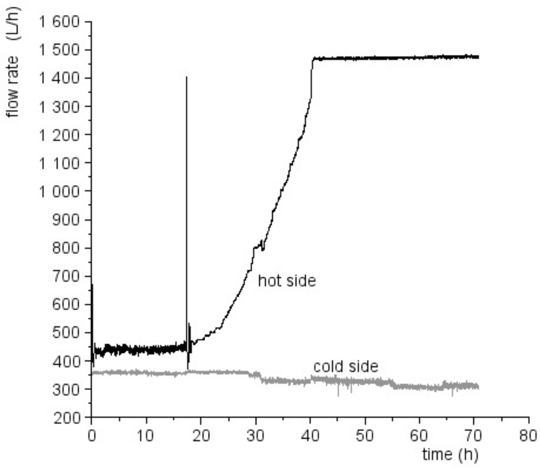

Fig. 4 Flow rate for heat exchanger #2

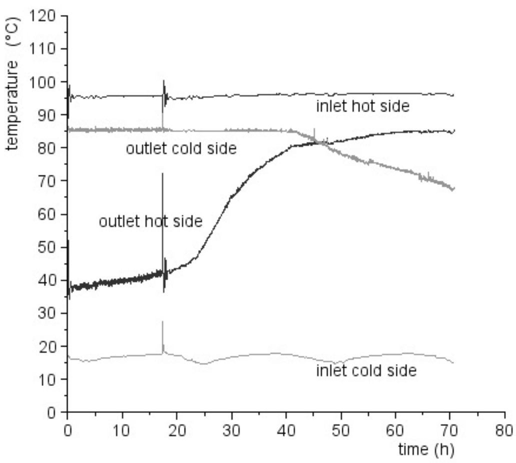

Fig. 5 Inlet and outlet temperatures for heat exchanger #2

It can be observed that at first sight both heat exchangers exhibit a similar behavior.

FIRST METHOD (Overall heat transfer coefficient)

It can also be seen that, after a short period of time where the temperatures vary quite quickly, the inlet condition vary about a quite stable mean value for a very few hours. So, in a first step, it is considered that the heat exchangers are in a steady state and thus that it is possible to compute an overall heat transfer coefficient (in fact the product of the overall heat transfer coefficient by the convective surface area). For confidentiality reasons this values is divided by the value during a reference period. Figures 6 and 7 show this dimensionless parameter for heat exchanger #1 and #2 respectively.

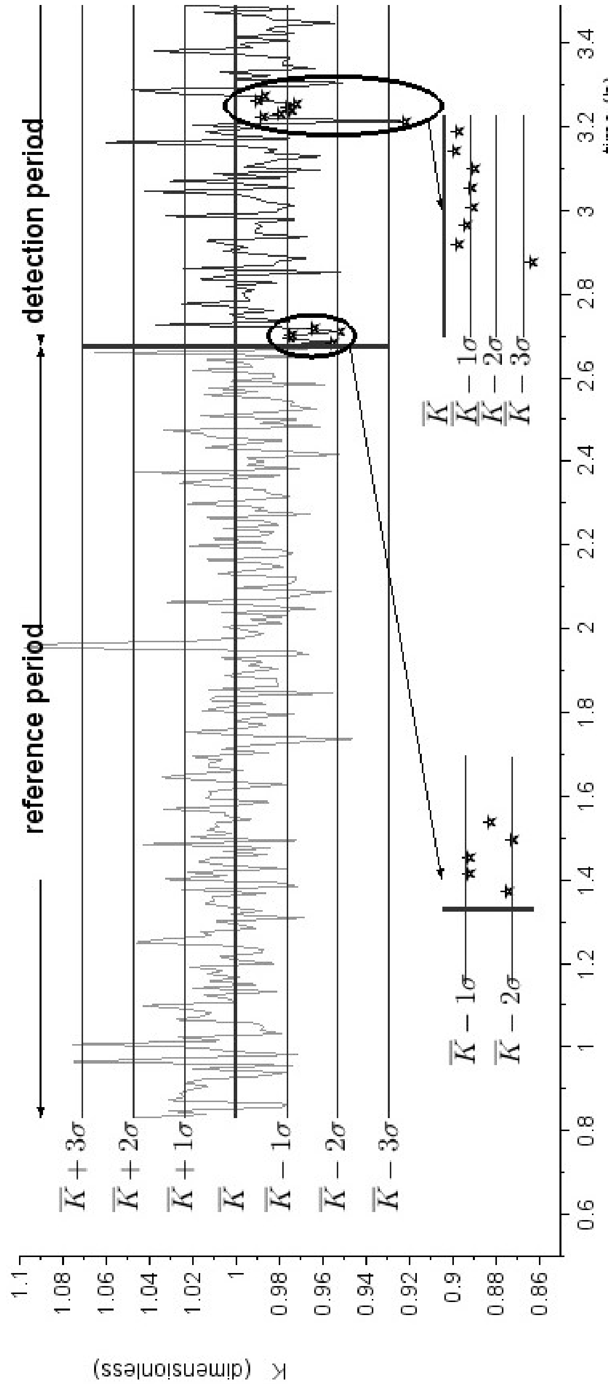

Fig. 6 Dimensionless overall heat transfer coefficient for heat exchanger #1

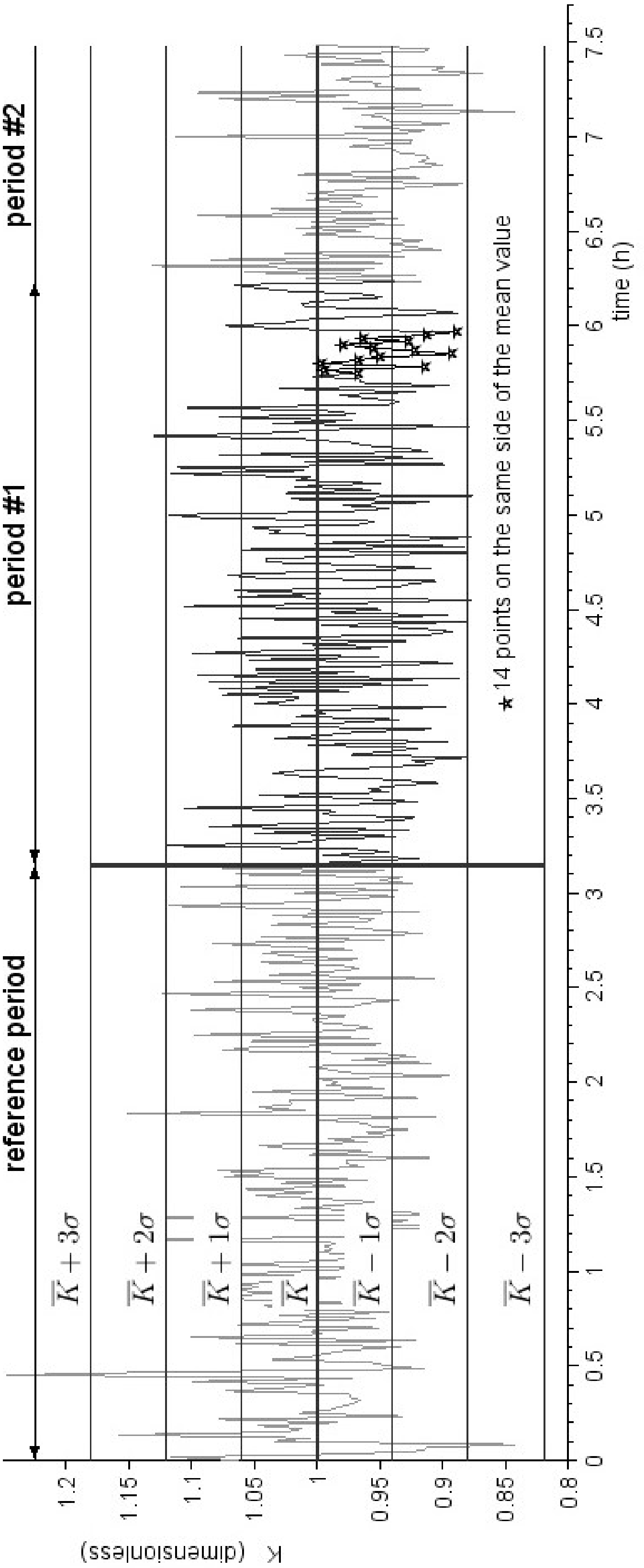

Fig. 7 Dimensionless overall heat transfer coefficient for heat exchanger #2

During the reference period it is possible to compute the mean value of the dimensionless overall heat transfer coefficient $\overline{K}$ and its standard deviation σ. To be able to visualize the simplest statistical test used in the study, namely the WECO rules (NIST, 2013), the $\overline{K} \pm i\sigma$ have been plotted for $i \in \{1,2,3\}$.

The first detection occurs right after the reference period (5 consecutive points over the one standard deviation limit). At this point the average value over the past 20 minutes of the dimensionless overall heat transfer coefficient is 0.99

Even if the first detection is disregarded, the result of the WECO rules (8 consecutive points on the same side of the mean value) on the first heat exchanger leads to the detection of the deviation of the dimensionless overall heat transfer when its value over the past 20 minutes is also 0.99.

Applied to the second heat exchanger, the detection of the deviation of the overall heat transfer coefficient occurs when its dimensionless value is 0.95.

It is well known that these WECO rules are very sensitive and can easily lead to false alarms. It is sometimes recommended to prefer the Cusum test (NIST, 2013). Figure 8 shows the value obtained by the Cusum test for both heat exchangers..

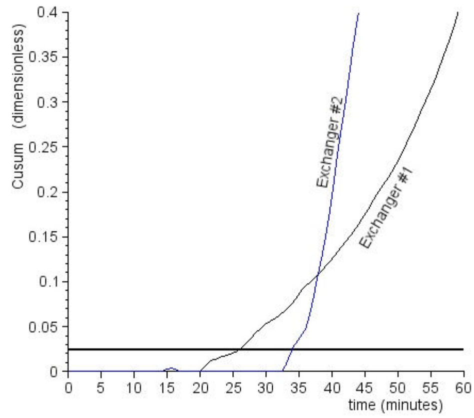

Fig. 8 Cusum value computed for heat exchanger #1 for the dimensionless overall heat transfer coefficient

It can clearly be seen that the variations are very smooth. Hence, it can be concluded that the test is very trustful (a process not having a clear trend does not lead to a smooth Cusum curve). It shows that 27 minutes and 35 minutes after the reference time the deviation of the dimensionless value of the overall heat transfer coefficient is detected for the first heat exchanger and the second heat exchanger respectively. At this time the mean value of the overall heat transfer coefficient are 0.99 and 0.96 for the first heat exchanger and the second heat exchanger respectively.

SECOND METHOD (The pressure drop)

This is a very popular method to try to detect fouling in heat exchangers. It must be noted that due to variations of the flow rates, the pressure drop varies also with time. Thus, it is necessary to compute a sliding average value of the pressure drop and to determine the number of samples needed to compute this value. It has been chosen to compute this average value over periods of 10 minutes (Fig. 9) and 30 minutes (Fig. 10) for the first heat exchanger.

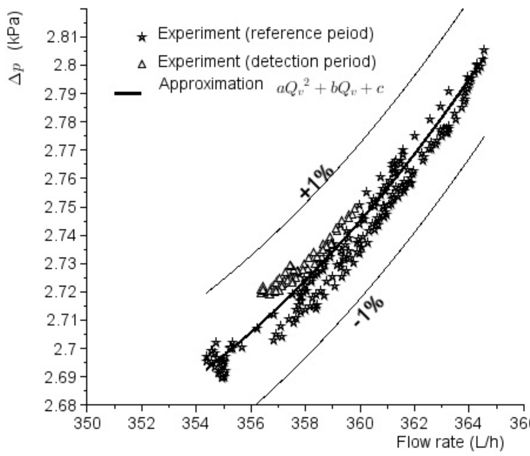

Fig. 9 the 10 minute sliding average value of the pressure drop for heat exchanger #1

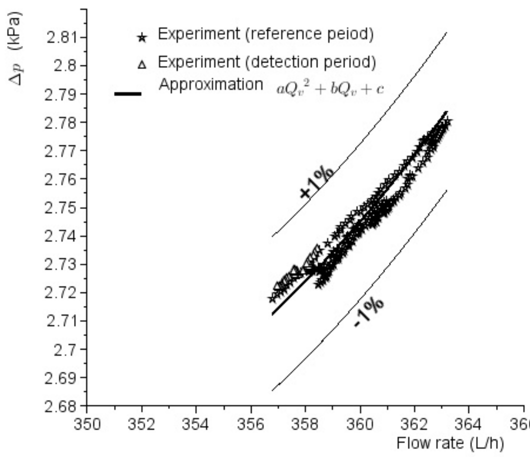

Fig. 10 the 30 minute sliding average value of the pressure drop for heat exchanger #1

These figures show that it is preferable to choose 30 minutes to get more stable values. This is what is done for the second heat exchanger (Fig. 11).

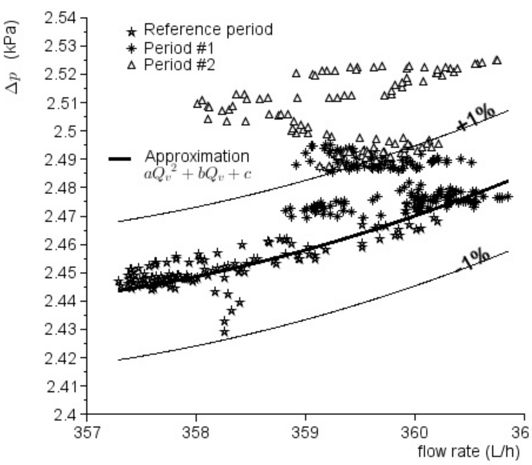

Fig. 11 the 30 minute sliding average value of the pressure drop for heat exchanger #2

These figures clearly show that the effect of fouling on the pressure drop strongly depends on the heat exchanger, and more specifically on the plate pattern.

This dependency is even clearer when the evolution of the pressure drop is plotted over a longer period of time. Figures 12 and 13 show this evolution for the first and second heat exchanger respectively.

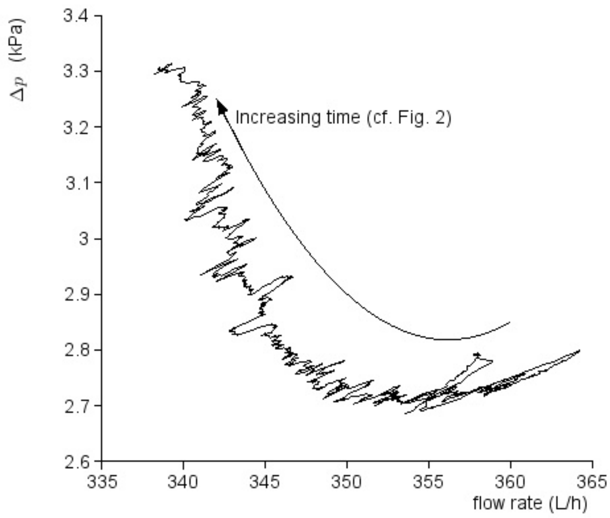

Fig. 12 the long term evolution of the sliding average value of the pressure drop for heat exchanger #1

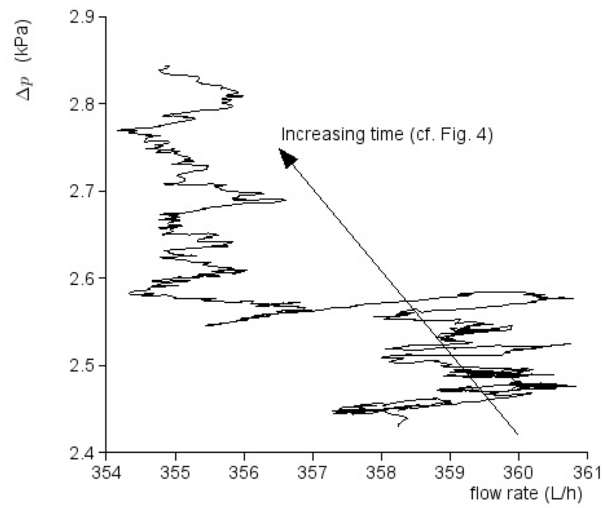

Fig. 13 the long term evolution of the sliding average value of the pressure drop for heat exchanger #2

It can be seen that the effect of fouling in the first heat exchanger is much higher than the effect of fouling in the second heat exchanger: the flow rate decreases in a much larger amount and the pressure drop increases in a much larger amount for heat exchanger #1 than for heat exchanger #2.

From these last five figures, it can be concluded that it would be quite impossible to detect fouling for the first heat exchanger as early as when using the first method: the evolution of the pressure drop is not significant. An easy detection (threshold value of 1% for the increase of the pressure drop) would lead to a detection time of 5.8 hours for the second heat exchanger. The dimensionless overall heat transfer coefficient would then be higher than 0.96.

It can then be concluded that this method is as sensitive as the first method for the second heat exchanger.

THIRD METHOD (The state space model)

This method is based on a model of the counter flow heat exchangers. This model is directly derived from the lumping of the channels and of the separating plates. When considering two sections parallel to the plates, the heat exchanger can be represented as shown in Figure 13. A fully detailed procedure is given in (Jonsson and Palsson, 1991).

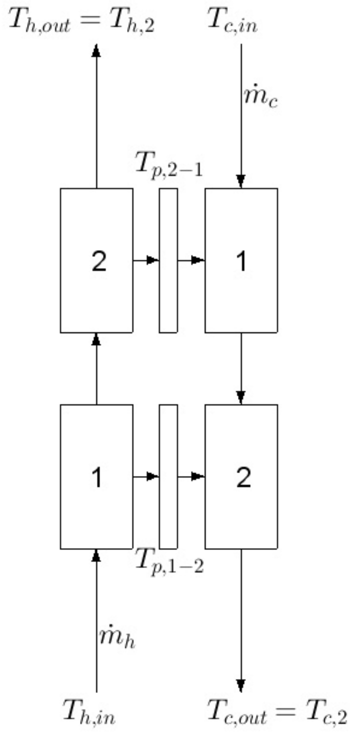

Fig. 13 2 section model of a counter flow heat exchanger

The heat balance is written in each section for the hot fluid, for the cold fluid and for the separating plate. For the 2 section configuration, the following set of equations is obtained:

$$M_h c_h \frac{dT_{h,1}}{dt} = \dot{m}_h c_h \left(T_{h,in} - T_{h,1}\right) - A_h h_{h,p}\left(T_{h,1} - T_{w,1-2}\right)$$

$$M_h c_h \frac{dT_{h,2}}{dt} = \dot{m}_h c_h \left(T_{h,1} - T_{h,2}\right) - A_h h_{h,p}\left(T_{h,2} - T_{w,2-1}\right)$$

for the hot side.

$$M_c c_c \frac{dT_{c,1}}{dt} = \dot{m}_c c_c \left(T_{c,in} - T_{c,1}\right) - A_c h_{c,p}\left(T_{c,1} - T_{w,2-1}\right)$$

$$M_c c_c \frac{dT_{c,2}}{dt} = \dot{m}_c c_c \left(T_{c,1} - T_{c,2}\right) - A_c h_{c,p}\left(T_{c,2} - T_{w,1-2}\right)$$

for the cold side

$$M_p c_p \frac{dT_{p,1-2}}{dt} = A_h h_{h,p}\left(T_{h,1} - T_{p,1-2}\right) + A_c h_{c,p}\left(T_{c,2} - T_{p,1-2}\right)$$

$$M_p c_p \frac{dT_{p,2-1}}{dt} = A_h h_{h,p}\left(T_{h,2} - T_{p,2-1}\right) + A_c h_{c,p}\left(T_{c,1} - T_{p,2-1}\right)$$

for the plate.

Introducing the following parameters:

$$\alpha = \frac{A_h h_{h,p}}{\dot{m}_h c_h}, \quad \tau_h = \frac{M_h}{\dot{m}_h}, \quad \beta = \frac{A_c h_{c,p}}{\dot{m}_c c_c}, \quad \tau_c = \frac{M_c}{\dot{m}_c},$$

$$\gamma_h = \frac{M_p c_p}{A_h h_{h,p}} = \frac{M_p c_p}{M_h c_h}\frac{1}{\alpha}\tau_h, \; \gamma_c = \frac{M_p c_p}{A_c h_{c,p}} = \frac{M_p c_p}{M_c c_c}\frac{1}{\beta}\tau_c$$

the so called state space model representation is obtained:

$$\frac{d}{dt}\begin{bmatrix} T_{h,1} \\ T_{h,2} \\ T_{p,1-2} \\ T_{p,2-1} \\ T_{c,1} \\ T_{c,2} \end{bmatrix} = \begin{bmatrix} -\frac{(1+\alpha)}{\tau_h} & 0 & \frac{\alpha}{\tau_h} & 0 & 0 & 0 \\ \frac{1}{\tau_h} & -\frac{(1+\alpha)}{\tau_h} & 0 & \frac{\alpha}{\tau_h} & 0 & 0 \\ \frac{1}{\gamma_h} & 0 & -\frac{\gamma_c+\gamma_h}{\gamma_c\gamma_h} & 0 & 0 & \frac{1}{\gamma_c} \\ 0 & \frac{1}{\gamma_h} & 0 & -\frac{\gamma_c+\gamma_h}{\gamma_c\gamma_h} & \frac{1}{\gamma_c} & 0 \\ 0 & 0 & 0 & \frac{\beta}{\tau_c} & -\frac{1+\beta}{\tau_c} & 0 \\ 0 & 0 & \frac{\beta}{\tau_c} & 0 & \frac{1}{\tau_c} & -\frac{1+\beta}{\tau_c} \end{bmatrix} \begin{bmatrix} T_{h,1} \\ T_{h,2} \\ T_{p,1-2} \\ T_{p,2-1} \\ T_{c,1} \\ T_{c,2} \end{bmatrix} + \begin{bmatrix} \frac{1}{\tau_h} & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & \frac{1}{\tau_c} \\ 0 & 0 \end{bmatrix} \begin{bmatrix} T_{h,in} \\ T_{c,in} \end{bmatrix}$$

To get accurate estimated outlet values, it is necessary to increase the number of sections as is done for standard discretization as shown in (Lalot and Palsson, 2010). In the present study 20 sections have been considered.

It must be noted here that the sample period is quite large (30 seconds and 60 seconds for the first and second heat exchanger respectively) and that it is therefore quite normal that the results are not as accurate as they could have been if the sample period would have been shorter.

All the parameters have to be determined before being able to estimate the outlet temperatures from the time series of the inlet temperatures and flow rates. Some of them just come from tables (heat capacity, density) or from the geometry (mass, area, residence time). Some parameters are determined in steady states (convection heat transfer coefficient correlations).

Note that other methods exist to determine these parameters such as using Extended Kalman Filters (EKF) as done in (Jonsson et al., 2007).

Figures 14 and 15 show for the first heat exchanger the comparison of estimated and actual outlet temperatures (cold and hot sides) obtained when the parameters of the models are well defined.

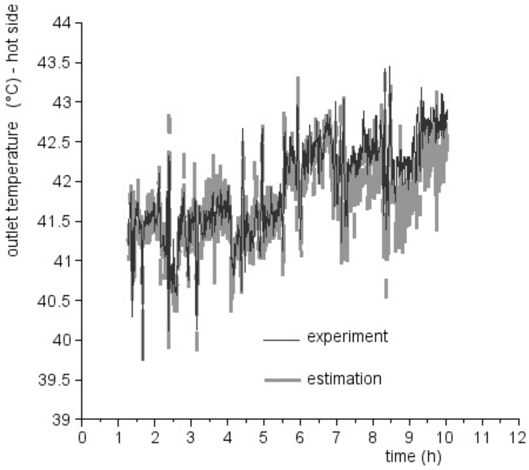

Fig. 14 Comparison of the estimated outlet temperature of the hot side for a 20 section model of the first heat exchanger

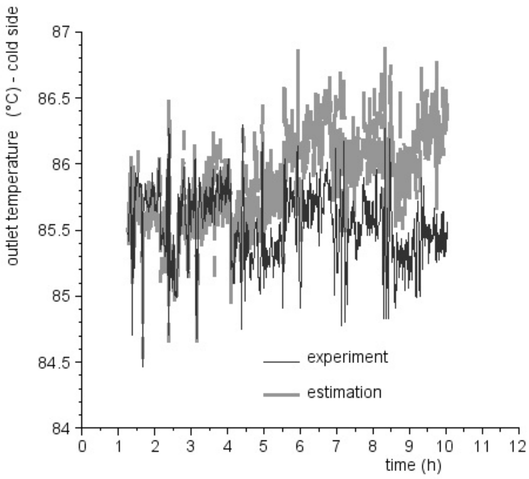

Fig. 15 Comparison of the estimated outlet temperature of the cold side for a 20 section model of the first heat exchanger

These figures show that, for the given inlet values, the effect of fouling is much more important on the cold side than on the hot side for this heat exchanger; the difference between estimated and actual values are larger for the cold side. This is in a good agreement with the fact that the mass flow of the cold side is varying in a large amount when fouling occurs.

Figure 16 shows the evolution of the Root Mean Squared Error (RMSE) for both sides for the first heat exchanger. Note that the RMSE is a sliding value computed over a period of 25 minutes.

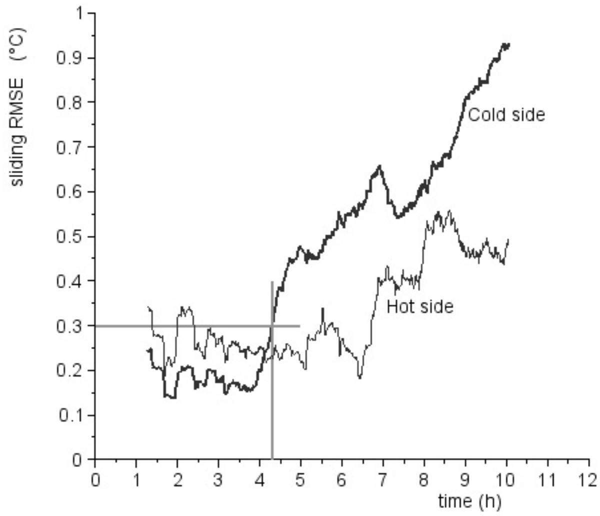

Fig. 16 Evolution of the RMSE for the outlet temperature of the hot side for a 20 section model of the first heat exchanger

It must be noted that the detection has to be carried out using the time series for the cold side. Doing so, the detection of the drift occurs at about 4.3 hours. This is done using the simplest test: the threshold method. At this time the average value (over 20 minutes) of the dimensionless overall heat transfer coefficient is 0.97. It can be seen that a quite large margin is taken into account for the threshold. Reducing this margin would lead to false alarms but would not lead to a large modification of the value of the overall heat transfer coefficient at the detection time.

Figures 17 to 19 show the same results for the second heat exchanger.

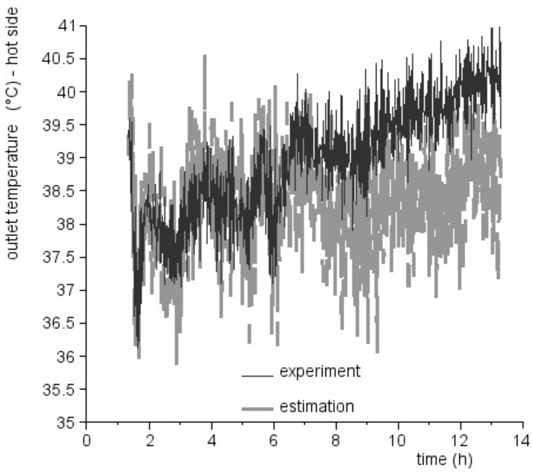

Fig. 17 Comparison of the estimated outlet temperature of the hot side for a 20 section model of the second heat exchanger

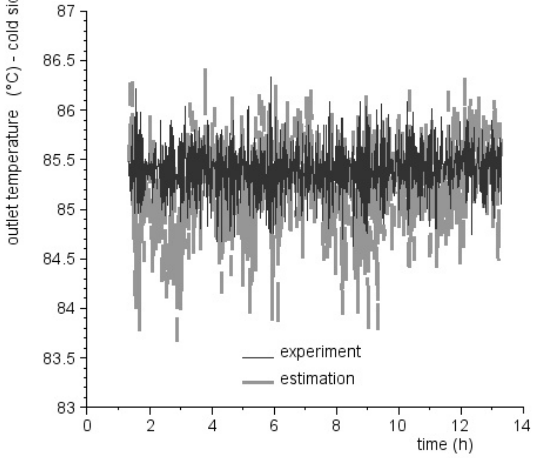

Fig. 18 Comparison of the estimated outlet temperature of the cold side for a 20 section model of the second heat exchanger

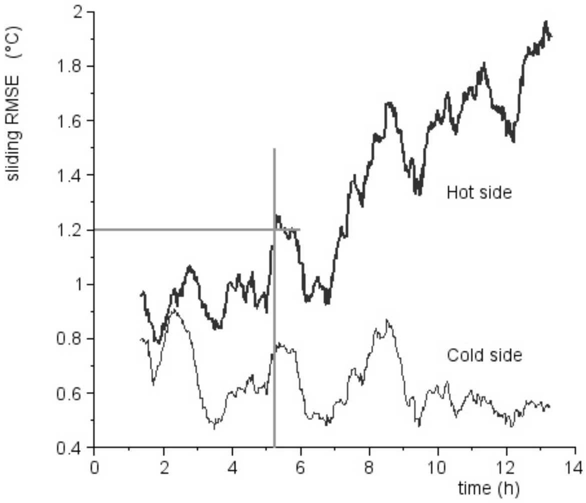

Fig. 19 Evolution of the RMSE for the outlet temperatures for a 20 section model of the second heat exchanger

These figures show that, for the given inlet values, the effect of fouling is much more important on the hot side than on the cold side for this heat exchanger.

The evolution of the sliding RMSE is only significant for the hot side. Hence, the detection is possible only on the hot side. Using a threshold with a large margin leads to a detection time of 5.25 hours. At this time the average value (over 20 minutes) of the dimensionless overall heat transfer coefficient is 0.98.

As an example, considering that the convection heat transfer coefficients are 4000 and 2000 W/m^2.K, a decrease of 3% of the overall heat transfer coefficient corresponds to a fouling factor of 2.3 10^{-5} m^2.K/W, which is much smaller than what is usually taken into account, e.g. 9 10^{-5} m^2.K/W for demineralized or distilled water (Engineering Page, 2013).

DISCUSSION

The three methods have shown that it is possible to detect fouling quite early: all methods detect fouling when the overall heat transfer coefficient has decreased by less than 5%. They differ from the easiness to be implemented in an automatic supervision tool.

The simplest one is the study of the pressure drop. The only computation needed is the average value on a sliding period of time. Hence, it is not necessary to use powerful processors. The drawback is that the threshold must be adapted to the heat exchanger plate pattern and certainly to the fluid properties.

On the other hand, it is necessary to use more sophisticated processors to be able to compute the average sliding value of the overall heat transfer coefficient. It is also necessary to measure the mass flow rate since this value is deduced from the Number of Transfer Units. But it must be noted that this method would not lead to accurate results if more variations are encountered. In that case it would be necessary to compute the overall heat transfer coefficient on a much longer period. The problem would then be to determine the correct value of this period.

These two methods are more efficient when the heat exchanger under supervision is in a steady state. One common drawback is that it is necessary to re-estimate the thresholds when the working conditions are changing (new steady state).

Finally, to be able to use the most sophisticated method (the state space model based method), it is necessary to use a computing tool close to a standard PC. As for the method based on the overall heat transfer coefficient, it is necesary to measure the input temperatures, the output temperatures, and the mass flow rates on both sides. A tuning phase is necessary to estimate all parameters of the matrices involved in the model, e.g. the correlation linking the flow rates and the convection heat transfer coefficients. One difficulty is also to find the correct sampling time: a too long one would mask the dynamic behavior of the heat exchanger; a too short sampling time would lead to a very heavy computational load. Nevertheless, it seems that this method would be more "universal" than the others when the heat exchangers are not in perfectly steady states.

This study confirms that it is possible to detect fouling using input and output data only. This an advantage over online sensors that then might be avoided. In fact online sensors either mimic fouling usually using a heated probe, or are localized in a zone where fouling is expected to occur. In both cases it is impossible to be sure that fouling will not occur in a different way or in a different location in the heat exchanger.

CONCLUSIONS

From the results drawn from experiments carried out on two heat exchangers, it is possible to conclude that:

1. The pressure drop increase due to fouling depends on the plate pattern.
2. When a heat exchanger is close to be in a steady state, detection of fouling based on the evolution of the

overall heat transfer is efficient and trustful when using the Cusum test. In this case, the size of the sliding observation window has to be carefully chosen.

3. The model based detection method has to take account of both sides of the heat exchanger (the hot side and the cold side).
4. The sampling period has to be adjusted to the dynamics of the heat exchanger and to the variations of the inlet temperatures and of the flow rates.
5. When well tuned, all detection method detect fouling when the overall heat transfer coefficient decrease is less than 5%. It has been shown that the method based on the overall heat transfer coefficient could detect a 1% decrease; but then, it is necessary to be in a quite steady state.
6. There is no evident "best" method to detect fouling. The model based method seems to be quite "universal", but has a heavy computational cost. The final user has to make a balance between the cost of instrumentation (number of measuring devices) and the cost of energy that is lost (not recovered) due to fouling
7. Future work will address new heat exchangers of a much larger size to check that the methods can be adapted to much thermally heavier heat exchangers.

NOMENCLATURE

A convective surface area per section, m^2
i index, dimensionless
K dimensionless overall heat transfer coefficient
M mass per section, kg
$\dot{m}$ mass flow rate
T temperature, K
t time, s
α equivalent to a Number of Transfer Units on the hot side, dimensionless
β equivalent to a Number of Transfer Units on the cold side, dimensionless
Δp pressure loss, Pa or kPa
σ standard deviation of K, dimensionless
τ residence time per section, s

Subscript

c cold side
h hot side
p plate
in inlet
out outlet

Superscript

$\bar{x}$ mean value of variable x

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