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PROCEDURE FOR APPLYING FOULING MODELS TO PREDICT OVERALL FOULING RATES IN INDUSTRIAL HEAT EXCHANGERS

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ABSTRACT

The application of a fouling model in either heat exchanger design or in the simulation of pre-heat train performance would appear to require integration over both space and time. Such analysis is prohibitive in terms of computer time. In this paper the authors present a simple approach that eliminates the need to integrate over exchanger space.

The predictions of this procedure are compared with a full space-time analysis of the performance of an industrially sized multi-pass heat exchanger. The results demonstrate the validity of the approach within expected accuracy of the predictions of a fouling model.

The procedure has been applied to the difficult task of a fouling model that has a sudden limiting condition (such as that observed in exchangers fitted with Turbotal inserts). The results have been compared with experimental data.

INTRODUCTION

Fouling models are generally derived from the analysis of rate data obtained under closely defined shear and temperature conditions. Consequently, they relate fouling rate to local velocity and local wall temperature.

Wall temperatures within industrial exchangers are not uniform. There is often very marked variation between inlet and outlet positions.

Variation in wall temperature will give rise to a variation in local fouling rate. This, in turn, will result in variations in deposit thickness throughout an exchanger. Since, wall shear is a strong function of the diameter of the tube this parameter can also exhibit large variation across a heat exchanger.

As fouling develops the overall heat transfer coefficient will vary from point to point within the exchanger. So, the calculation of the heat transfer that occurs under fouled conditions would require that the basic heat transfer equation be applied at number of points within the exchanger and the results summed over the length of the exchanger (integration over "space").

Since, the thermal resistance presented by the deposit affects the temperature at the surface of the deposit the rate of fouling occurring at individual points within the exchanger will change as the fouling progresses.

Therefore, application of a fouling model in either heat exchanger design or in the simulation of pre-heat train performance would appear to require integration over both space and time. As found in conducting such simulations in the course of this work, such analysis is prohibitive in terms of computer time.

Recognition of this situation led to the development of an integrated form (Polley et al, 2004,2007) of the Ebert-Panchal Model (Ebert & Panchal, 1997, Panchal et al, 1999) for the prediction of fouling rates during the heating of crude oils. This integration was based on the assumption that the variation of wall temperature within heat exchangers was approximately linear. Whilst this assumption may be valid for pure counter-flow, industrial exchangers usually employ two or more tube passes. In this situation the integration may prove invalid. Consequently, the integrated model was tested against exchanger simulation that incorporated integration of performance over both space and time. A range of exchanger geometry (various tube pass arrangements and more than one shell-in-series) was covered (Gonzales-Garcia, 2009) and the integrated model was found to give predictions of acceptable accuracy.

Fouling in crude oil systems is the subject of much research. It is likely that the Ebert-Panchal Model will soon be superseded by one having a better theoretical base and better accuracy. It may prove difficult to produce an integrated from of such a model. So, in this paper we develop an alternative approach that is based upon the direct use of "point" models.

PREDICTION OF THE PERFORMANCE OF MULTI-PASS EXCHANGERS

Roetzel & Spang (1989) have analysed the performance of multi-pass shell-and-tube heat exchangers in which either the amount of surface area contained within the individual passes or the overall heat transfer coefficients encountered in the individual passes differed.

The analysis showed that the overall effectiveness of the exchanger was a function of the Number of Transfer Units contained in the individual passes (parallel and counter-current flow) and is given by:

$$\frac{1}{\varepsilon} = \nu + R + \frac{1}{Ntu_T}\frac{m_1 e^{m1} - m_2 e^{m2}}{e^{m1} - e^{m2}} \qquad (1)$$

Ntu_T is the total number of transfer units contained in the exchanger:

$$Ntu_T = Ntu_{cf} + Ntu_{pf} \quad (2)$$

and,

$$\nu = \frac{Ntu_{pf}}{Ntu_T} \quad (3)$$

$$R = \frac{CP_t}{CP_s} \quad (4)$$

$$m_1 = \left(\frac{Ntu_T}{2}\right) \left(\left[\{R+2\nu-1\}^2 + 4\nu(1-\nu)\right]^{1/2} - [R+2\nu-1]\right) \quad (5)$$

$$m_2 = \left(\frac{Ntu_T}{2}\right) \left(-\left[\{R+2\nu-1\}^2 + 4\nu(1-\nu)\right]^{1/2} - [R+2\nu-1]\right) \quad (6)$$

If we have a suitable means of specifying a mean fouling resistance in each pass then these equations can be used to predict overall exchanger performance without the need to integrate detailed equations over "space".

Simulation of Multi-pass Exchanger Performance

Consider the two pass shell-and-tube heat exchanger illustrated in Figure 1. The hot and cold inlet temperatures are known. The temperature distribution of the hot and cold temperatures throughout the exchanger can be determined using the following procedure.

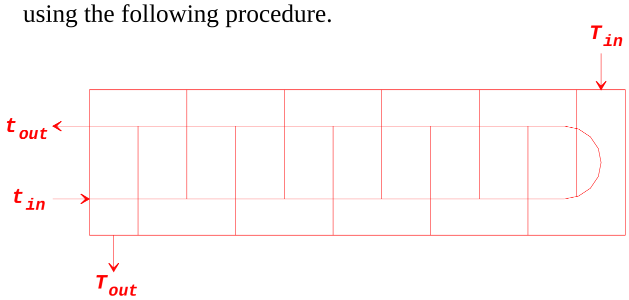

Figure 1. Two-pass Shell-and-tube Exchanger

The exchanger is divided into a number of individual sections. Provided the baffle count is reasonably high (greater than ten) the sections could be the individual baffle spaces. Then the temperature at which the shell-side fluid leaves the exchanger is set at an initial value. The individual film heat transfer coefficients can be calculated using suitable heat transfer correlations. In the case of the tube-side coefficient changes in physical properties may result in the two passes having different film coefficients.

With the fouling resistance for each pass being set at an initial value, the overall heat transfer coefficients for each pass are determined. Since, the stream flow-rates are known the number of transfer units presented by each pass can be calculated. Then using Effectiveness-Ntu relationship for parallel flow for the exit pass and the counter-flow relationship for the entry pass the temperatures at the boundary of the section can be calculated (Figure 2).

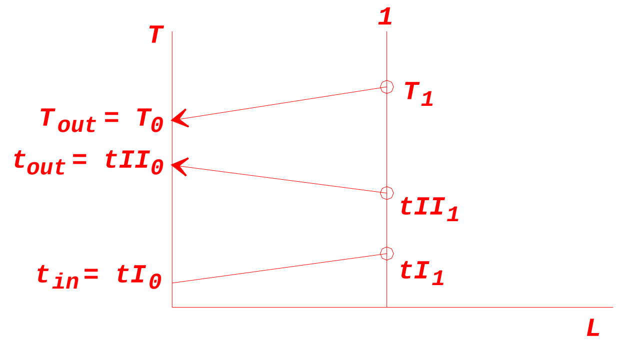

Figure 2. Determination of temperature change across first section

The bulk temperatures for the tube-side are taken to be the mean of the inlet and outlet temperatures for each pass. That for the shell-side is taken to be the mean of the inlet and outlet shell-side temperatures for the section. The wall temperature for each pass is obtained from the difference between the bulk values and the ratio of the pass overall heat transfer coefficient to the inside film coefficient.

The sequence of calculations is repeated for the next area section and for subsequent sections until the end of the exchanger is reached. If the temperature at which the hot stream leaves the exchanger has been set correctly the boundary temperatures at the end of the unit will equate. The hot stream exit temperature is adjusted until this condition is achieved.

The results of a typical calculation for a two pass exchanger are shown below. The exchanger studied had 606 tubes of 19.05 mm otside diameter and 6.1 m length. There were twelve baffles on the shell-side (giving thirteen individual spaces). This baffle count was found to be sufficient to ensure the validity of the numerical integration used whilst allowing the simulation to be undertaken in reasonable computer time. Crude oil flowed through the tubes at a rate of 59.1 kg/s. It had an inlet temperature of 200.9 C. Hot liquid flowed through the shell-side of the exchanger at a rate of 66.1 kg/s and had an inlet temperature of 276.4 C. The individual film heat transfer coefficients were taken from the results of analysis undertaken using ESDU's EXPRESS ™ computer program as being 2175.6 W/m^2K on the tube-side and 1864.8 W/m^2K on the shell-side. The shell-side of the unit was assumed to be clean. The initial fouling resistance for the tube-side of the unit was assumed to be 0.00148 m^2K/W. This value was based on measurements obtained on the plant immediately following cleaning.

The outlet temperatures were identical with those predicted by the EXPRESS ™ computer program.

Space-time integration with a fouling model

Since, the simulation described above provides information on both velocity and temperature fields it is possible determine the rate of fouling occurring at different locations the exchanger at a given time using a fouling model. For purposes of demonstration the reaction fouling model proposed by Ebert & Panchal is used. This model is given by the equation:

$$\frac{dR_d}{d\theta} = \frac{a}{\mathrm{Re}^{0.66}\,\mathrm{Pr}^{0.33}} \exp\left(\frac{-E}{R_g T_f}\right) - \gamma \tau_w \qquad (7)$$

The value of the deposition constant (a) was set at 50.3, the Activation energy (E) was set at 42 kJ/mole and the suppression constant (γ) was set at $3\text{x}10^{-08}$.

Fouling was found to have a significant influence not only on the temperatures at which the hot and cold fluids exited the exchanger but on the distribution of bulk and wall temperatures within the exchanger. In Figure 3 we show the variation of wall temperature across the exchanger as a function of both location and time. Each set of lines relates to a given time after commencement of operation (e.g. top line shows conditions after 100 hours operation). The exchanger was 6 m in length. We observe that the variation across each pass is approximately linear. However, the gradient for the counter-flow pass is greater than that for the co-current pass. The gradients reduce as fouling level increases.

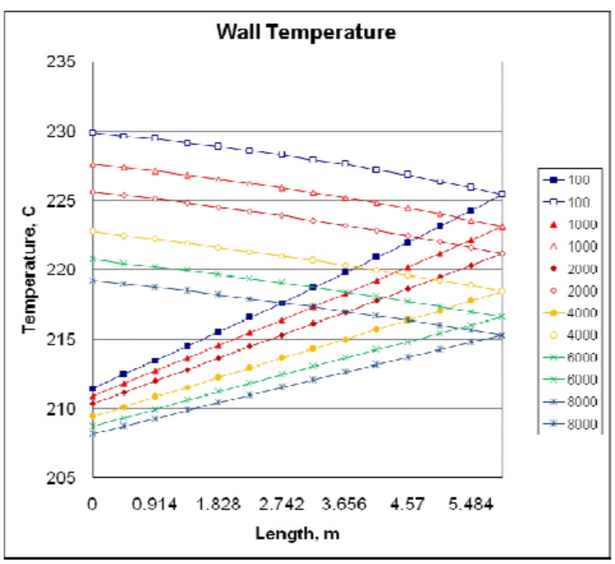

Figure 3. Exchanger Simulation: Wall temperature Variation

In Figure 4 we show how the rate at which fouling occurs varies with both location and time. The variations are shown over total flow length (12 m) rather than distance from inlet. Again we see that the variation is approximately linear in each pass but the gradient for the counter-flow pass is significantly different to that for the co-current pass.

Application of Roetzel-Spang Equations with Fouling Model

The observations that both wall temperatures and fouling rates appear to vary linearly with regard to exchanger location at any moment of time suggests that the application of a point fouling model at a mean pass temperature would yield good predictions for the overall fouling rate.

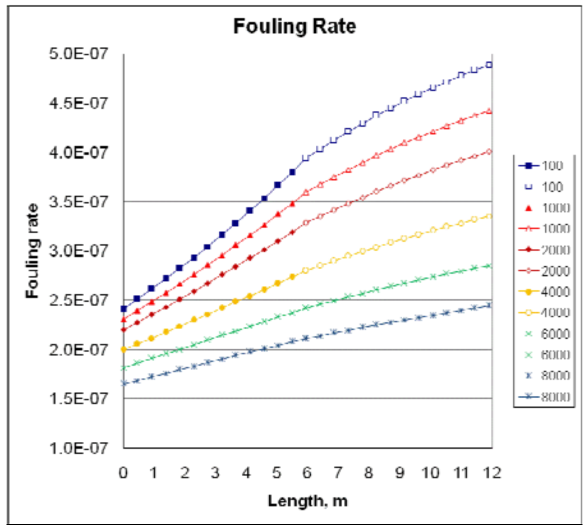

Figure 4. Exchanger Simulation: Fouling Rates

For a two pass exchanger, Roetzel & Spang presented the following equation for the temperature in the return header:

$$t_x = 1 - \frac{1}{RN_{tu}}\left(\frac{m_1 - m_2}{\exp(m_1) - \exp(m_2)}\right)\frac{T_{in} - T_{out}}{T_{in} - t_{in}} \qquad (8)$$

At any given fouling level the exchanger outlet temperatures are obtained by solving equation 1. The fouling rate for that condition is then obtained by applying the chosen fouling model at the mean pass temperatures obtained from these temperatures and the header temperature given by Equation (8).

Comparison of rapid procedure with space-time integration

The predictions of the method based upon the Roetzel-Spang Equations has been compared with the results of a full space-time simulation over a range of operating conditions. The results have been found to be virtually identical.

Prediction of fouling behaviour of exchanger fitted with Turbotal ^{TM} inserts

The higher the heat recovery in a pre-heat train the higher the bulk temperature of the crude oil and the greater the wall temperatures encountered in the heat recovery exchanger. At the highest bulk temperatures the fouling rate can get so high that regular cleaning of an exchanger is required in order to maintain production rates.

An alternative to regular cleaning, which is an expensive exercise, is the use of tube inserts (Turbotal^{TM}) that initially reduce the rate at which fouling occurs and subsequently control the thickness of the deposit formed such that the fouling resistance is maintained at a constant value.

Aquino et. al. (2007) presented a modified form of the Ebert-Panchal Model for the prediction of the initial fouling rates in tubes fitted inserts. The equation is:

$$\frac{dR_d}{d\theta} = \frac{h_{plain}}{h_{enhanced}} \frac{a}{\mathrm{Re}^{0.66}\,\mathrm{Pr}^{0.33}} \exp\left(\frac{-E}{R_g T_f}\right) - 0.7 \frac{\Delta P_{enhanced}}{\Delta P_{plain}} \gamma \tau_w \quad (9)$$

The modification is a simple one. The deposition term has been multiplied by the ratio of the plain tube heat transfer coefficient to the enhanced coefficient. This modification quantifies the reduced thickness of the thermal film – and hence reduction in “reaction volume” assumed by Ebert & Panchal (1997).

The suppression term has been multiplied by the ratio of the pressure drop encountered in the enhanced tube to that in the plain tube. This term is multiplied by 0.7 to reflect that only 70 % of the pressure drop encountered during flow through tubes fitted with inserts is associated with wall friction. The remainder is caused by form drag.

Once, the deposit has filled the space between the tube wall and the rotating insert the fouling resistance is maintained at a constant value. Materials that would otherwise deposit on the surface are removed by mechanical action of the insert. The limiting fouling resistance can be determined from the insert clearance divided by the thermal conductivity of the deposit. A typical value is 0.0035 m^2K/W.

This fouling model was incorporated into the full space-time simulation. The fouling rate was predicted using equation (10). Once the fouling resistance reached a value of 0.0035 it was maintained at this value. An overall fouling resistance for the exchanger is calculated from the differences between the inverse of the clean overall heat transfer and that of a duty coefficient calculated from the performance of the fouled unit.

The same model was applied at the mean temperature of each pass and the Roetzel-Spang equations used to calculate fouling rates.

The case involved two shells operating in series (the handling of more than one shell in series and more than two tube passes is described in detail by Gonzales Garcia (2009)).

A comparison between the overall fouling resistances is shown in Figure 5. The predictions of the full simulation are slightly higher than those from the more rapid method.

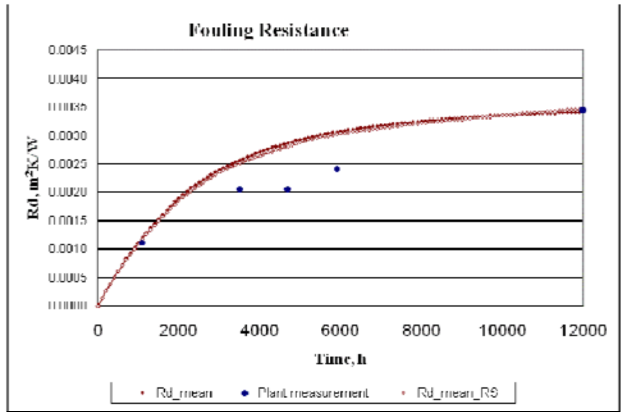

Figure 5. Fouling Rates Predicted for Exchangers Fitted with Turbotal

Finally, experimental measurements of fouling resistances observed on the operational exchanger are superimposed on these plots. The time at which the exchanger was brought on-line was unknown. So, the first observation was directly superimposed on the predicted line (time of around 1200 hours). The other points then indicate how fouling developed in the unit.

CONCLUSIONS

1. A new method for the determination of the behaviour of industrial exchangers using fouling models has been developed. This methodology is independent of the fouling model that is to be used.
2. The predictions of the model have been compared with a full space-time simulation of the behaviour of a shell-and-tube heat exchanger. And found to give valid results.
3. It has been tested against a model that provides a rapid fouling rate followed by a sharp limit on fouling resistance. It has been found to provide valid results under these circumstances.
4. The Ebert-Panchal Model has been used in this study. Research aimed at developing models having a better theoretical base and better accuracy is being undertaken in various laboratories around the World. When such a model is finally developed it may prove difficult to produce an integrated from of such a model. Hopefully, the approach described here will allow the direct use of such a model. However, this will require action on the part of companies providing software for heat exchanger design and analysis.

NOMENCLATURE

a	Deposition constant
CP_s	Heat Capacity Flowrate - shellside, W/K
CP_t	Heat Capacity Flowrate - tubeside, W/K
E	Activation Energy, kJ/mole
h	heat transfer coefficient, W/m^2 K
m_1	defined in equations 5
m_2	defined in equations 5
Ntu_T	Total Number of transfer units
Ntu_{cf}	Number of transfer units – counter flow pass
Ntu_{pf}	Number of transfer units – parallel flow pass
Pr	Prandtl number
R	Ratio of tubeside to shellside CP
Re	Reynolds number
R_g	Universal Gas constant, kJ/mol K
t	Cold temperature, C
T	Hot temperature, C
T_f	Fouling Side Film Temperature, K
ΔP	Pressure drop, Pa
γ	Removal constant m^{2}K/W.hr.Pa
ε	Overall heat transfer effectiveness
θ	Time, hours
τ_w	Wall shear stress Pa

Subscripts

in	Exchanger inlet
out	Exchanger outlet
plain	empty tube
enhanced	tube fitted with insert

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