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CONSIDERING IN-TUBE CRUDE OIL BOILING IN ASSESSING PERFORMANCE OF PREHEAT TRAINS SUBJECT TO FOULING

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ABSTRACT

Oil refinery preheat trains can exhibit unwanted two-phase flow behaviour. An example is boiling of crude oil inside heat exchangers, when the local pressure is not high enough to keep crude in a liquid state. This often arises when the pump is under-sized. Understanding the two-phase behaviour and assessing the boiling heat transfer coefficients would result in a better prediction and estimation of exchanger fouling. Where single-phase modelling is used under boiling conditions, the anomalous behaviour leads to unrealistic estimates of fouling resistance, and can severely under-predict the increased pressure drop and consequent loss of crude throughput.

There is little public information on fouling in two-phase flows as laboratory experiments are very costly, despite the importance of this in refinery heat exchangers and furnaces. Indeed, the importance of crude boiling is likely to increase as lighter crudes such as shale oils are processed. These lighter crudes are often blended with heavier crudes to maintain an appropriate refining average density.

This manuscript consists of two sections. The first section uses industrial monitoring data to illustrate fouling behaviour for a heat exchanger that undergo both boiling and fouling. The second section discusses simulations to evaluate thermo-hydraulic behaviour when the crude undergoes boiling. The analysis requires coupled heat transfer, hydraulic and surface fouling aspects, and a commercial preheat train network simulator, SmartPM, was used for this study.

INTRODUCTION

Crude oils are a complex mixture of hydrocarbons and impurities; their physical and chemical characteristics vary widely from oil reserves and also within the same reserve. The mixture includes hydrocarbon components with a range of volatilities, water, inorganic salts and various chemicals that assist in extraction. Processing of crude oil in refineries involves steps such as washing the crude with water to remove inorganic salts (desalting) and heating the crude oil through a network of heat exchangers (known as the preheat train, PHT), to raise its temperature in preparation for fractional distillation. The crude is processed at high pressures to maintain single-phase flows (liquid state); in a typical refinery the local crude pressure could experience values as high as 30 bars (IHS ESDU 2007). During crude heating in PHT's, unwanted two-phase flow (boiling) could occur when the local pressure is less than the vapour pressure of the most volatile component at the local operating temperature. Such industrial example was reported by Liporace and De Oliveira (2007), where the vaporization of crude occurs in heat exchangers downstream of the desalter.

Water carry over during a desalting process could also result in water boiling in heat exchangers downstream of the desalter. This is often apparent from seemingly very large fouling resistance values in those exchangers. Controlling the desalting operation is also of great importance to the PHT (Ishiyama *et al*., 2010a); phase behaviour in the PHT relating to the desalter operation is not considered in the scope of this paper.

The main effect of crude oil boiling on fouling rate is through the high turbulence created by the bubbles at the solid-fluid interface. This turbulence enhances both heat and mass transfer. Boiling reduces the boundary layer resistance and promotes the flow of fouling precursor towards the heated surface. Experimental studies of hydrocarbon fouling under boiling conditions were reported by several research community: *e.g*. Huasler and Thalmeyer (1975), Fetissoff *et al*. (1982) and Crittenden and Khater (1984).

This paper discusses a methodology to evaluate thermo-hydraulic behaviour of a heat exchanger undergoing tube-side boiling. This methodology is used to illustrate three case studies. First case study is on 'rating' the performance of an industrial heat exchanger. The second and third case studies explore thermo-hydraulic behaviour for a hypothetical heat exchanger assuming a linear fouling rate; its effect on a simple network is discussed.

MODEL FORMULATION

For a heat exchanger undergoing crude boiling on the tube-side, the vapour quality present at the tube inlet and at the outlet could be significantly different due to variation in temperature and pressure along the tube. An example of such flow in a horizontal tube is illustrated in Figure 1, where the liquid is initially sub-cooled and heated in single-phase flow. Nucleation begins at point A, and bulk boiling at B. When the volume flow rate vapour considerably exceeds that of the liquid, the flow becomes 'annular', as at C in the sketch. Most of the liquid flows along the wall, whilst the vapour plus some of the liquid, in the form of entrained droplets, flows at considerably higher velocity in the centre of the tube. At the point marked D the wall becomes dry, because the supply of liquid to the wall by impingement of drops is less than the rate of evaporation.

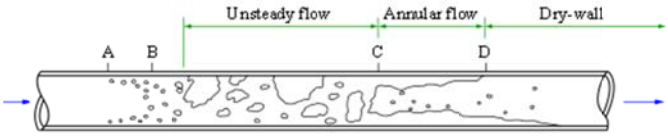

Figure 1: Sketch of flow patterns in boiling flow along a horizontal tube (IHS ESDU, 1985).

Widely different conditions could exist at various points along a heat exchanger tube under tube-side boiling conditions; deriving a lumped (or an average) operating condition is not easily achieved. Another complication is the likelihood of vapour-liquid separation occurring in the heat exchanger headers when several tube-side passes are present; thus resulting in flow mal-distribution. In this manuscript, boiling is evaluated based on 'local' thermo-dynamic conditions at the inlet and/or the outlet of the heat exchanger.

Heat exchanger pressure drop calculation

Most process equipment design is based on generalized pressure drop correlations that do not explicitly account for the two-phase flow regime (Lestina and Serth, 2010). We consider a generalized approach here.

Two general models of two-phase flow are common; 1) homogenous flow and 2) separated flow. The homogenous model assumes that each phase has a same local velocity. In the separated flow model, each phase flows in separate zones and has different velocities, but could interact with each other. Separated flow models are known to provide a better representation of the pressure drop in pipe flows. A model of this type, presented by Lockhart and Martinelli (1949), was utilised in the study, where a two-phase flow multiplier was introduced. The two-phase flow multiplier was defined as the ratio of the pressure gradient during two-phase flow, ΔP_{TP}, and the pressure gradient due to friction if total fluid flows as liquid, ΔP_L. The two-phase multiplier used in the study is based on Chisholm's Equation (Chisholm, 1973) and given by:

$$\frac{\Delta P_{TP}}{\Delta P_L} = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{1}$$

where X is the Lockhart-Martinelli parameter given by:

$$X = \left[\left(\frac{dP}{dz}\right)_L \Big/ \left(\frac{dP}{dz}\right)_V\right]^{0.5} \tag{2}$$

The pressure gradients are those if the whole mixture flows as vapour or liquid:

$$\left(\frac{dP}{dz}\right)_L = -2(1-x)^2 \frac{M^2 C_{f,L}}{\rho_L d} \tag{3}$$

$$\left(\frac{dP}{dz}\right)_V = -2x^2 \frac{M^2 C_{f,V}}{\rho_V d} \tag{4}$$

where x is the vapour quality, M is the total mass velocity, d is the tube-internal diameter and C_f is the friction factor. Subscripts V and L denote states when vapour and liquid alone were present, respectively. C_f is obtained based on the fluid Reynolds number, Re:

$$\begin{aligned} C_{f,i} &= \frac{16}{Re_i} \quad if\ Re_i < 2{,}000 \\ C_{f,i} &= 0.079 {Re_i}^{-0.25} \quad if\ 2{,}000 < Re_i < 20{,}000 \\ C_{f,i} &= 0.046 {Re_i}^{-0.25} \quad if\ Re_i > 20{,}000 \end{aligned} \tag{5}$$

Subscript 'i' indicate either 'vapour (V)' or 'liquid (L)' state.

Parameter C in equation (1) is related to the local ratio of the densities of the liquid and vapour phases:

$$C = \left(\frac{\rho_L}{\rho_V}\right)^{0.5} + \left(\frac{\rho_V}{\rho_L}\right)^{0.5} \tag{6}$$

Shell-and-tube exchangers usually consist of a bundle of tubes with several tube-side passes. Under single (liquid)-phase flow, the pressure drop across on the tube-side of a shell-and-tube heat exchanger could be presented as (Sinnott et al., 2005):

$$\Delta P_{HEX,L} = 4N_{pass} C_{f,L} \frac{l}{(d-2\delta)} \frac{\rho_L u_L^2}{2} + 2.5 N_{pass} \frac{\rho_L u_L^2}{2} \tag{7}$$

where l is the tube length, N_{pass} is the number of tube-side passes, ρ_L is the liquid crude density, and u_L is the mean tube-side velocity of the liquid crude. δ is the deposit layer thickness. For tube-side fouling, Yeap *et al.* (2004) have shown that, for typical crude oil systems, δ, may be related reasonably well to the 'thin-layer' approximation as

$$\delta = R_f\, \lambda_f \tag{8}$$

Here λ_f is the deposit thermal conductivity and R_f is the thermal resistance due to fouling. The relationship of R_f to the heat transfer coefficients are described later in equation (19).

An approximation is now made to obtain the two phase pressure drop of the heat exchanger, $\Delta P_{HEX,TP}$, through multiplying $\Delta P_{HEX,L}$ by the two-phase flow multiplier.

$$\Delta P_{HEX,TP} = \Delta P_{HEX,L} \left(1 + \frac{C}{X} + \frac{1}{X^2}\right) \tag{9}$$

Heat transfer coefficient

Among the earlier methods for predicting the heat transfer coefficient in in-tube boiling, the method of Chen (1966) was frequently used. He studied experimental results obtained with the saturated boiling of water and several organic liquids. Both convective and nucleate boiling contributed to the transfer of heat. The correlation was developed based on experimental data for vertical tubes; its applications in horizontal tubes are also discussed for saturated wet-wall heat transfer. Since, boiling heat transfer correlations have evolved and new correlations were developed such as described by Shah (1976, 1982) and IHS ESDU (1985, 1991). Crude oil is a complex mixture and limited information is available on correlations for crude boiling heat transfer coefficients. In this manuscript, the convective heat transfer term of Chen (1966) correlation will be considered to estimate the tube-side heat transfer coefficient; the study is not limited to this correlation and other heat transfer correlations could readily be applied. Chen (1966) correlation is given by:

$$h_{i,TP} = h_{i,co} + Sh_{i,n} \tag{10}$$

Here, $h_{i,TP}$ is the two-phase heat transfer coefficient, $h_{i,co}$ is the convective heat transfer coefficient, $h_{i,n}$, is the nucleate boiling coefficient and S is the suppression factor. The convective coefficient is based on a momentum - heat transfer analogy, and is related to that for the case in which the liquid phase flows alone by:

$$\frac{h_{i,co}}{h_{i,L}} = \left(\frac{\Delta P_{TP}}{\Delta P_L}\right)^{0.445} \tag{11}$$

The convective heat transfer coefficient for the liquid phase flowing alone, $h_{i,L}$, is given by Gnielinski (1976). A conservative approximation for the two-phase flow heat transfer was made in this study by neglecting the nucleate boiling term in equation (10):

$$h_{i,TP} = h_{i,co} \tag{12}$$

The shell-side stream was assumed to be single-phase. The shell-side film transfer coefficient is obtained through stream analysis method (IHS ESDU, 1984).

Pump hydraulics

The pumps providing the hydraulic driving force are normally centrifugal devices operating at constant rotational speed, so that the flow rate depends upon the operating head, H, and therefore the pressure drop across the network. The pressure flow characteristic curve is approximated here by

$$\varphi = \varphi_s - pV^2 \tag{13}$$

where V is the liquid volumetric flow, p is a dimensional constant. Throughout, it is assumed that the crude is single-phase liquid at the pump. φ_s is the pump shut-off head based on the pump design:

$$\varphi_s = \frac{(r\omega)^2}{2g} \tag{14}$$

where r is the impeller radius and ω is the rotational speed. A typical centrifugal pump characteristic curve is presented in Figure 2. The pump head provides the required pressure drop for the crude to flow across the PHT, ΔP_{PHT}, and also the pressure to keep the crude at the liquid state, P_{sp}. ΔP_{PHT} is described by three components, pressure drop across the control valve, pressure drop across the heat exchanger and the pressure drop across piping and fittings.

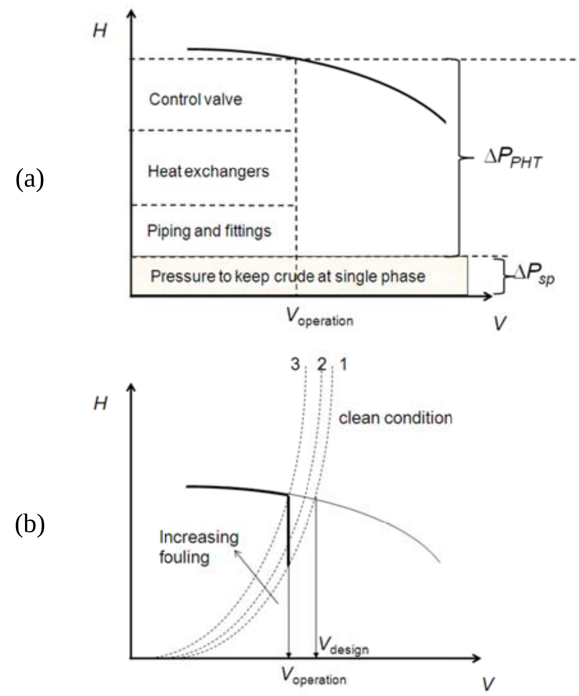

Figure 2: Illustration of a pump characteristic curve. (a) Pressure distribution of the pump head, across the preheat train; (b) centrifugal pump characteristic; dashed line PHT characteristic, with fouling increasing from 1 to 3; bold solid line, combined characteristic curve. At position 3 the control valve is fully open (Ishiyama *et al*., 2009).

The pump characteristic is not the sole determinant of the flow, as the flow is also subject to control valve action (Ishiyama et al., 2009). Figure 2(b) shows the characteristic of a pump that has been overdesigned (for the combination of target flow and clean pressure drop. Initially, under clean conditions, the control valve will partly close to maintain the flow at the set point, $V_{operation}$. As the pressure drop across fouled HEXs increases, the control valve will open to compensate, until it is fully open. Thereafter, the flow rate will decrease as fouling proceeds. The hydraulic effect of fouling is therefore masked by the operation of the control valve: a period of constant throughput is followed by a decline in throughput determined by the pump characteristic curve. The changes in the crude feed rate are matched by proportional changes to the hot stream flow rate.

Calculating the change in stream local pressure

Consider a shell-and-tube heat exchanger illustrated in Figure 3, with the cold stream on the tube-side. The exchanger could consist of an active bypass and in such instances the pressure measurements are likely to be made before the splitting of the cold stream (P_1) and after mixing of the bypass to the main stream (P_2). In this instance, the pressure drop obtained from the measurements ($P_1 - P_2$) is the sum of the pressure drop across the heat exchanger, ΔP_{HEX}, and the additional pressure drop incurred at the junction of the splitter and the mixer, ΔP_{P}.

$$P_1 - P_2 = \Delta P_{HEX} + \Delta P_p \tag{15}$$

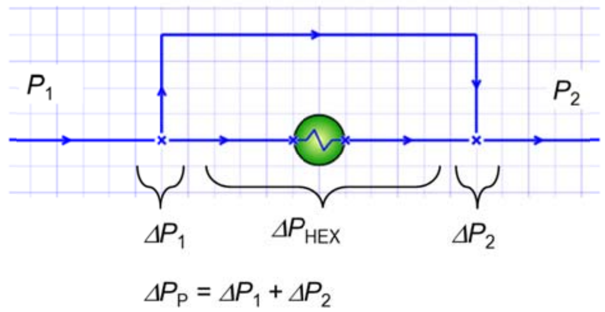

Figure 3: Heat exchanger with an active bypass; gauge pressure measurements are made before stream split (P_1) and after stream mixing (P_2), respectively. ΔP_1 and ΔP_2 represent the pressure drops associated with the splitter and the mixer, respectively. Only the cold stream on the tube-side is presented for simplicity.

With change in ΔP_{HEX} due to fouling, P_2 is recalculated by re-arranging equation (15):

$$P_2 = P_1 - \left(\Delta P_{HEX} + \Delta P_p\right) \tag{16}$$

Change in ΔP_p due to the change in mass flow rate is approximated by assuming it to be proportional to the square of total volume flow rate.

Data reconciliation (rating)

Data reconciliation is a key step in processing industrial measurements as raw industrial data usually have errors and missing information. For crude oil refineries, it is common to have missing hot stream flow measurements, which are inferred through heat and mass balance based on the crude stream. For an exchanger undergoing boiling, the heat duty of the crude stream, Q_c, is given by

$$Q_c = m_c\left[H\left(T_{c,out}, P_{c,out}\right) - H\left(T_{c,in}, P_{c,in}\right)\right] \tag{17}$$

Here, Q_c is the heat duty, m_c is the cold stream mass flow rate, H is the specific enthalpy of the crude, T is the temperature and P is the operating pressure. The subscripts 'in' and 'out' denotes conditions at the inlet and outlet streams, respectively.

The hot stream mass flow rate is inferred by:

$$m_h = \frac{Q_c}{C_{p,h}\left(T_{h,in} - T_{h,out}\right)} \tag{18}$$

Here $C_{p,h}$ is assumed to be a linear function of temperature (as described in the form in Table 3); the bulk temperature, which was obtained as the arithmetic mean of the stream inlet and outlet temperatures were used to evaluate $C_{p,h}$.

The thermal resistance, R_f, of an heat exchanger is obtained from plant monitoring data through solving the following equation using data reconciliation methodology detailed in (Ishiyama et al., 2011):

$$\frac{1}{UA_o} = \frac{1}{h_o A_o} + \frac{1}{h_{i,TP} A_i} + \frac{R_f}{A_{i,cl}} + \frac{R_w}{A_o} \tag{19}$$

Here U is the overall heat transfer coefficient, h_o is the external film transfer coefficient, A_o is the external heat transfer area, A_i is the internal heat transfer area, and R_w is the wall resistance. Subscript 'cl' denotes clean conditions.

Fouling rate (simulation)

A linear fouling rate, $\dot{R}_f$, is assumed in the simulation case studies.

$$\dot{R}_f = constant \tag{20}$$

Solving equations (1) to (20) for a heat exchanger (and associated network) required coupling of heat transfer, hydraulic and surface fouling aspects. A commercial preheat train network simulator, SmartPM, was used in this study to perform data reconciliation and predict thermal and hydraulic performance under crude boiling and fouling.

CASE STUDIES WITH IN-TUBE BOILING

Three case studies were explored. First case study (case study 1), investigates a single industrial heat exchanger undergoing crude boiling. Plant monitoring data were collected and the fouling resistance profiles were extracted for a period of six months of operation.

Second case study (case study 2), illustrates a hypothetical heat exchanger operation under single-phase and two-phase flow. This exchanger is taken to a third case study (case study 3) where the impact of in-tube boiling on a downstream exchanger is explored. The single (liquid) phase thermo-physical property of the streams for case studies 2 and 3 are detailed in Table 3. Crude heat release curve data were used to generate vapour quality and specific enthalpy data.

Case study 1: Industrial heat exchangers undergoing tube-side boiling

Operation of an industrial shell-and-tube heat exchanger with crude boiling on the tube-side was studied. The inlet and outlet pressures of the crude stream were monitored daily for a period of 5 months (Figure 4(a)). Both the inlet and outlet temperatures of the crude and the hot streams were also monitored (Figure 4(b)). The heat demand curve for the crude was obtained from a thermo-dynamic package, where comparing the inlet pressure and the temperature with vapour quality indicated that the crude is evaporated within the unit (i.e. the vapour quality was greater than zero as shown in Figure 5(c)). Only the crude stream volumetric flow was available; the crude flow was measured at a location, upstream of the heat exchanger where the stream was single-phase. The crude had a fluctuating mass flow ranging between 100 – 140 kg s^{-1}. This unit is single segmental baffled with two tube-side passes.

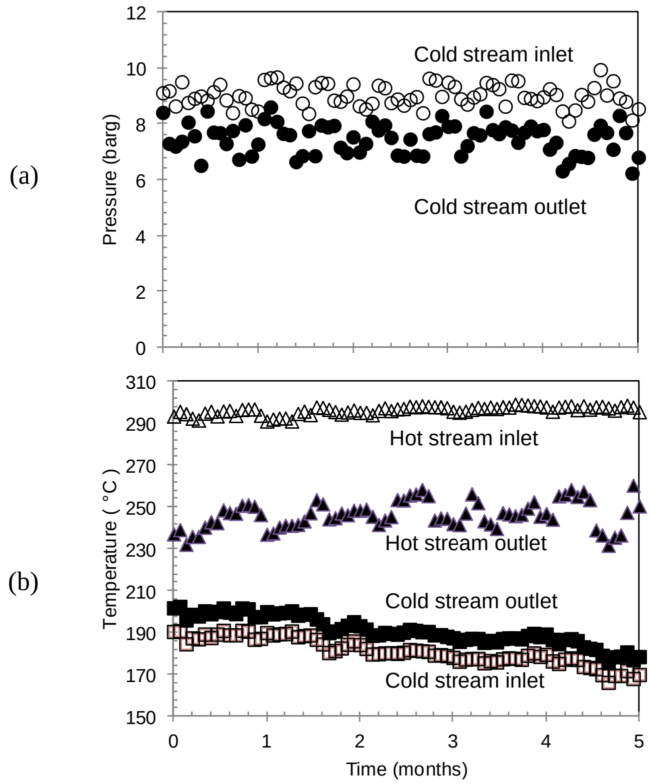

Figure 4: Case study 1. Monitoring data for a 5 months period; (a) local pressure, (b) stream temperature. The hollow and filled circles in (a) denote inlet and outlet conditions, respectively. The hollow and filled triangles in (b) denote inlet and outlet hot stream, respectively. The hollow and filled squares in (b) denote inlet and outlet crude streams, respectively.

h_o was calculated based on the inferred mass flow rate from equation (18). $h_{i,TP}$ was calculated at the inlet and the outlet conditions of the exchanger, separately. Hence two resistance profiles were obtained based on the inlet and outlet $h_{i,TP}$ values (Figure 5(a)). The R_f profile based on $h_{i,TP}$ at the tube-outlet (filled circles in Figure 5(a)), showed to be higher compared to that R_f based on $h_{i,TP}$ at the tube-inlet (hollow circles).

The obtained R_f profile consists of noise; still in the first 2.8 months an increasing trend is observed (Region A, Figure 5(a)). For this region, linear fouling rates could be extracted for the two R_f profiles, 6.2×10^{-11} $m^2K\ J^{-1}$ and 1.5×10^{-10} $m^2K\ J^{-1}$, respectively. The trend in R_f is unclear and consists of high disturbance in the region marked B; this disturbance is not reflected through the fluctuations in calculated $h_{i,TP}$ or the vapour quality profile and requires further analysis. $h_{i,TP}$ exhibits high fluctuations at the first two months of the monitoring period (Figure 5(b)). Both the cold stream inlet and outlet temperature shows a gradual decrease (Figure 4(b)). The decrease in the cold stream inlet temperature is likely to be linked to the reduction in heat duty recovery from upstream exchangers due to fouling. The reduction in vapour quality (Figure 5(c)) at the inlet and the outlet of the exchanger is the result of the reducing crude temperature.

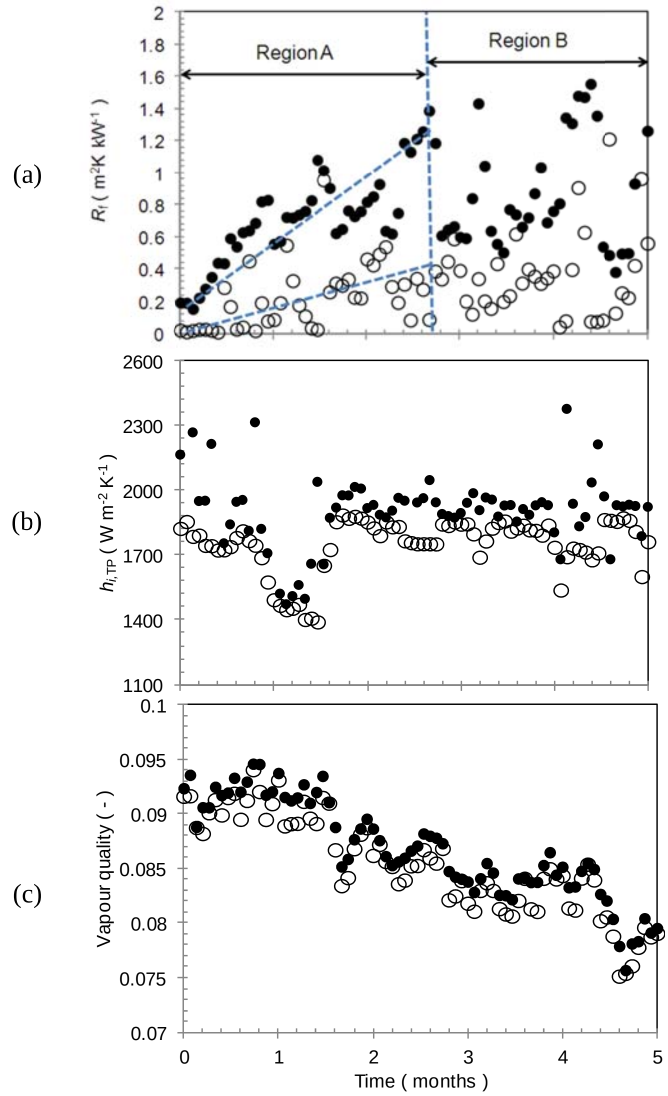

Figure 5: Case study 1. Reconciled data for an industrial exchanger undergoing boiling; (a) fouling resistance, (d) tube-side heat transfer coefficient and (c) vapour fraction. The filled and hollow circles represent reconciled values based on calculated boiling condition at the heat exchanger outlet and inlet conditions, respectively.

Case study 2: Single heat exchanger – comparison of thermo-hydraulics in single-phase and two-phase simulations

The aim of this case study is to compare the thermo-hydraulic behaviour of a hypothetical shell-and-tube heat exchanger undergoing tube-side boiling with that of a single-phase operation (Figure 6). Tables 1 and 2 summarizes the exchanger geometry and the pump hydraulics in this study. λ_f in equation (8) was assumed as 2 W m^{-1}K^{-1}. This lies in the upper bound of the expected value of λ_f (0.5 – 2 W m^{-1}K^{-1}, Ishiyama *et al*. (2010b)). The thermo-hydraulic behaviour of a exchanger (and PHT) is highly sensitive to λ_f (Ishiyama *et al*., 2008); exploring the sensitivity of λ_f is an ongoing research and not covered in this paper.

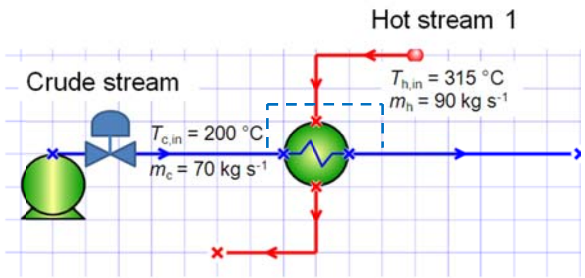

Figure 6: Hypothetical heat exchanger with inlet conditions for case study 2.

Table 1: Heat exchanger design data

Parameter	Value
Tube length	6 m
Tube internal, external diameter	17, 19 mm
Total number of tubes	1290
Number of tube-side passes	2
Shell diameter	1.2 m
Tube layout	30°
Baffle spacing	0.4 m
Baffle cut	23 %

*Linear fouling rate of 1 × 10^{-10} m^{2}K J^{-1}

Table 2: Pump data

Parameter	Value
Impeller diameter	0.75 m
Rotational speed	700 rpm
ΔP_{valve}, ΔP_{PHT} (at 70kg s^{-1})	0.05, 0.1 bar
Point on centrifugal pump (head, flow rate)	(20 m, 0.3 m s^{-1})

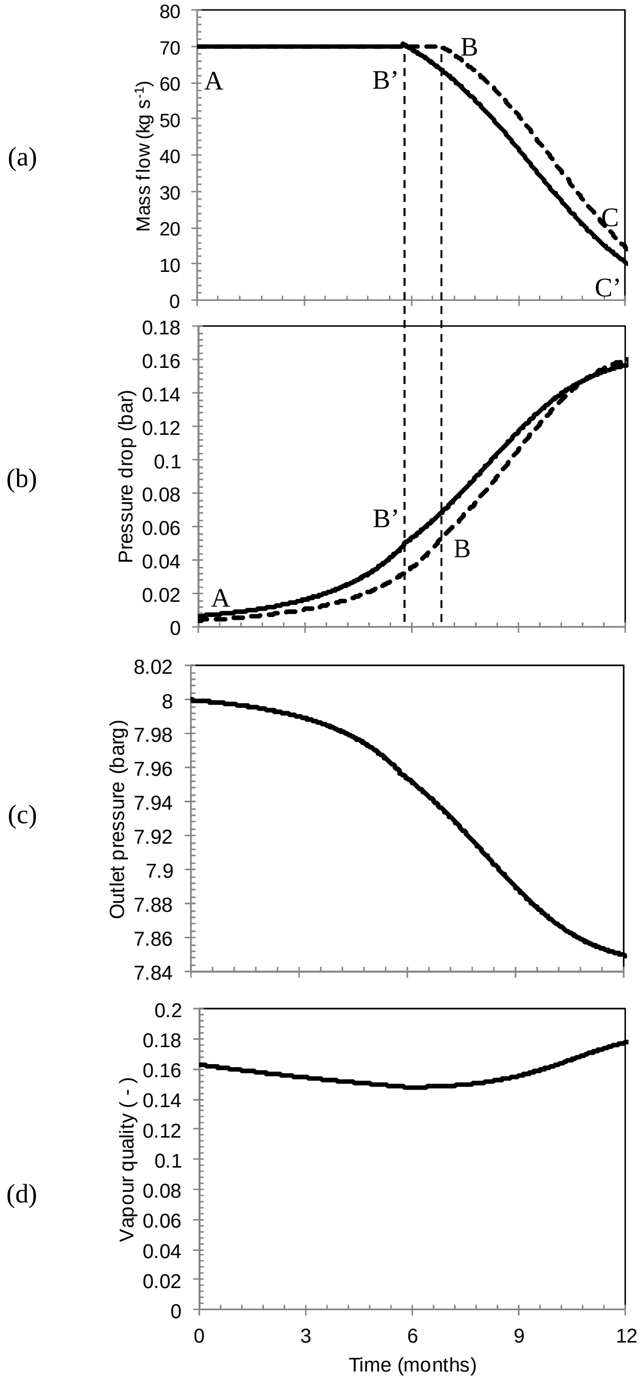

Figure 7: Case study 2. Simulated profiles of (a) throughput, (b) exchanger pressure drop, (c) outlet pressure and (d) vapour quality at the exchanger outlet. Solid line presents simulation with crude boiling. The dashed lines in figures (a) and (c) presents single phase operation.

Simulation result assuming single (liquid) phase operation shows that the exchanger initially operates under constant flow (section AB in Figure 7(a) – dashed line). During this operating period the pressure drop of the heat exchanger increases rapidly (Figure 7(b)). At point B the pumping limit is reached and constant throughput is no longer achieved. The reduction in throughput will occur with progress in fouling (BC in Figure 7(a)).

When the exchanger is undergoing boiling, it initially operates with a pressure drop, higher than the equivalent single-phase flow pressure drop (Figure 7(b)). With fouling, the pump reaches its hydraulic limit at an earlier stage compared to the single-phase operation (B' in Figure 7(a)). Once the pump has reached its pumping limit, reduction in throughput occurs with fouling following the curve B'C' in Figure 7a. The reductions in the outlet vapour quality and outlet pressures are plotted in Figure 7(c) and (d), respectively. The outlet vapour quality initially decreases due to the reduction in outlet crude temperature with fouling. The vapour quality starts to increase with fouling when the reduction in outlet pressure dominates the crude vaporization.

The reduction in the outlet pressure could have a significant effect on the exchangers downstream of this unit. This effect is considered in the next case study (case study 3).

Case study 3: Two heat exchangers in series

Two heat exchangers with identical geometries were arranged in series (Figure 8). Both units were assumed to have the same overall linear fouling rate (1×10^{-10} m^{2}K J^{-1}). The outlet crude stream pressures for each exchanger, E1 and E2 are marked P_1 and P_2, respectively, in Figure 8. P_1 and P_2 were assumed to be 8 and 7 barg at the beginning of the simulation: i.e. the difference in P_1 and P_2 denote the total pressure drop across E2 and that of associated piping and fitting (as described in Figure 3). For simplicity, the additional piping and fitting are not included in Figure 8, but accounted in this study. Simulation was performed assuming a constant throughput over the simulation period (*i.e.* assuming no pumping limit). $h_{i,TP}$ was calculated for each heat exchanger based on the thermo-dynamic condition at the heat exchanger crude stream outlet. A deposit thermal conductivity of 2 W m^{-1}K^{-1} was assumed in this study.

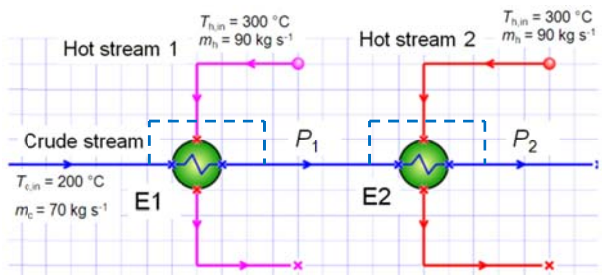

Figure 8: A simple sketch of two heat exchangers in series. The difference (P_1 – P_2) indicate the sum of pressure drop across E2 and the piping and fittings as described in Figure 3.

The simulated P_1 and P_2 are plotted in Figure 9(a)). Near the last month of the simulated period, the reduction in P_1 has a strong effect on the reduction in P_2. With fouling, the thermal performance of the unit decreases and both units exhibits a reduction in vapour quality, due to the reduction in outlet temperature (curves 'ab' (E1) and 'de' (E2) in Figure 9(b)). The Vapour quality tends to rise when the reduction in local pressure dominates the reduction in crude temperature (curves 'bc' (E1) and 'ef' (E2) in Figure 9(b)). The increase in exchanger pressure drop Figure 9(c)) is significantly higher compared to Figure 7(c); this is expected as the simulation was performed assuming constant mass flow rate. In reality, the network is likely to exhibit hydraulic limit as described in case study 1, when the pump capacity is limited. In this example the difference in the selected initial values for P_1 and P_2 are much greater than that of the pressure drop of the heat exchanger alone and require further analysis of industrial data to obtain realistic values; this is an ongoing work.

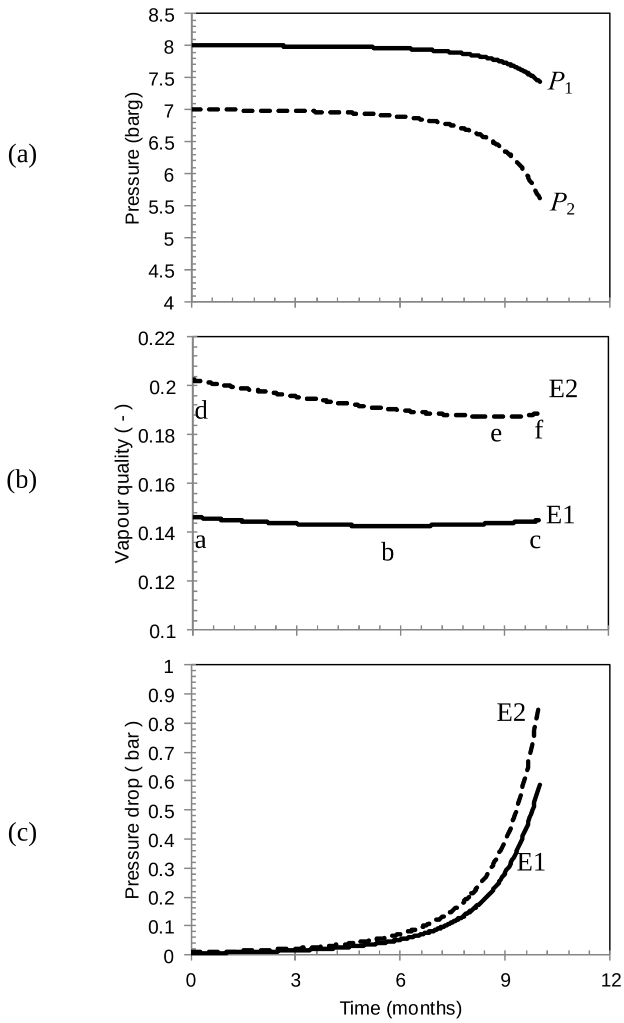

Figure 9: Case study 3. Profiles of (a) outlet gauge pressure, (b) vapour fraction and (c) heat exchanger pressure drop alone. Solid and dashed lines indicate E1 and E2, respectively in Figure 8.

Conclusions

1. A methodology was discussed to extract fouling profiles for heat exchangers undergoing tube-side boiling, using the local pressure and temperature conditions at the tube-inlet and outlet.
2. Thermo-hydraulic simulations were performed for a heat exchanger undergoing tube-side boiling.

Comparison with single-phase simulation indicated that heat exchangers with in-tube boiling reaches its hydraulic limit earlier compared to a single-phase operation.

3. Crude boiling could have considerable effect in a network and a simple illustration described the impact of boiling on a downstream heat exchanger.

Nomenclature

A heat transfer area, m^2
C dimensionless parameter in equation (1), –
C_f friction factor, –
C_p specific heat capacity, J kg^{-1} K^{-1}
d tube diameter, m
H total enthalpy of the crude, J kg^{-1}
h film transfer coefficient, W m^{-2} K^{-1}
g gravitational acceleration, m s^{-2}
l tube length, m
M mass flux, kg s^{-1} m^{-2}
m mass flow rate, kg s^{-1}
N_{pass} number of tube-side passes, –
P pressure, pa
ΔP pressure drop, pa
p dimensional constants in equation (12), s^2 m^{-5}
Q heat duty, W
r pump impeller diameter, m
R_f fouling resistance, m^2K W^{-1}
R_w wall resistance, m^2K W^{-1}
Re Reynolds number, –
S suppression factor, –
T temperature, K
X Lockhart-Martinelli parameter, –
x vapour quality, –
U overall heat transfer coefficient, W m^{-2} K^{-1}
u mean tube-side velocity, m s^{-1}
V volumetric flow, m^3 s^{-1}

Greek

δ deposit thickness, m
φ pump head, m
φ_s shut-off head, m
λ thermal conductivity, W m^{-1} K^{-1}
ρ density, kg m^{-3}
ω pump rotations speed, rpm

Subscripts

c cold (crude) stream
cl clean condition
co convective
design design condition
f foulant
h hot stream
i internal
in at the inlet
HEX heat exchanger
L liquid phase
n nucleate boiling
o external
operation operating condition
out at the outlet
p piping
PHT preheat train
sp single-phase
TP two-phase
V vapour-phase

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Table 3: Liquid-phase thermo-physical properties of the streams in case studies 2 and 3.

	Density, kg m^{-3} = b_1 $(T\text{-}273)$+ b_2	Thermal conductivity, W m^{-1} K^{-1} = c_1 $(T\text{-}273)$ + c_2	Specific heat capacity, J kg^{-1} K^{-1} = d_1 $(T\text{-}273)$ + d_2	Viscosity, cP = e_1 $exp(e_2 /T)$
Stream	b_1 (kg m^{-3} K^{-1}), b_2 (kg m^{-3})	c_1(W m^{-1} K^{-2}), c_2 (W m^{-1} K^{-1})	d_1(J kg^{-1} K^{-2}), d_2 (J kg^{-1} K^{-1})	e_1 (Pa s), e_2 (K)
Crude	-0.890, 860	-0.0004, 0.15	3.46, 1900	6.677E-03, 1952
Hot 1, 2	-0.726, 920	-0.0002, 0.17	3.50, 1890	2.576E-06, 3077