

SIMPLIFIED ANALYSIS OF INFLUENCE OF PRE-HEAT TRAIN PERFORMANCE AND FIRED HEATER DESIGN ON FUEL EFFICIENCY OF FIRED HEATERS

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ABSTRACT

This paper explores the complexity of the problem of fouling in the design and operation of fired heaters and presents a procedure that could ultimately be used in fired heater design. The aspects of performance that are considered are as follows: a). determination of flow conditions at the inlet to the heater; b). determination of local heat transfer coefficients across the whole length of the heater; c). determination of fouling rates across the whole length of the heater; d). interaction between hot and cold side conditions. Two phase flow patterns are an important consideration when the tubes have a horizontal orientation. A procedure for determining the pattern present at any location is reported. Fouling on the crude-side is assumed to follow a model that has been widely applied to heat exchanger fouling. This is for the purposes of demonstration alone. The shear rates encountered in two phase flow are significantly higher than those in shell-and-tube heat exchangers. There is no basis for assuming that the model will apply in two phase flow situations. Furthermore, changing the parameters of the model could also result in different conclusions. Much work is being conducted on the development and refinement of fouling models. It is the authors' view that the short comings stated above are likely to be removed through this work.

INTRODUCTION

Fouling occurs on both hot and cold sides of fired heaters. On the hot side it mostly occurs in the convective section of the heater and can be controlled using soot blowers and steam lances (Taler et al, 2009). On the cold side the fouling rates are influenced by the flow conditions within the furnace tubes and by wall temperatures. Wall temperatures are affected by the design of the furnace and by the conditions on the hot side. Consequently, cleaning of the hot side can lead to increased fouling on the cold side.

On the cold-side of the furnace the flow conditions depend upon the temperature at which the feed enters the unit. So, with crude oil the amount of heat recovery achieved within

the pre-heat train could have effect upon the fouling that occurs within the fired heater.

Fouling of the heat transfer surface of fired heaters providing the boil-up for crude oil distillation columns can have a highly detrimental effect not only on the fuel economy of the pre-heat train but in some cases on the economic efficiency of the refinery as a whole. For instance, fouling on the crude-oil side of the heater generally leads to the formation of coke. In extreme cases, particularly with heaters used on the feed to vacuum distillation units, coke can be carried over from the heater into the column itself and form deposits that affect the efficiency of the column (Koch-Glitch estimate that 15% of refinery distillation columns experience problems of this nature (2010)).

The modelling of the fouling on the crude side of the fired heater is complicated by the effect that this fouling has upon the temperature profile on the hot-side of the heater. As time progresses the temperature of hot gas leaving the radiant section of the heater increases.

The pressure drop across a fired heater is high and both two phase flow conditions and the saturation temperature of the crude oil is dependant on local pressure. So a further possible complication considered here is the effect that the fouling on the crude side has upon the flow resistance presented by the furnace tubes. The thicker the deposit the greater is this flow resistance.

The higher the pressure drop through the tubes the higher is the pressure immediately after the pressure let-down valve at the entrance to the heater. This, in turn can result in less flash across the valve and poorer heat transfer in the tubes situated in the convective section of the unit. Poorer heat transfer leads to higher wall temperatures and higher fouling rates.

The work reported in this paper involves the combination of a simplified model of a furnace coupled with modelling of

the thermal and hydraulic behaviour on the crude oil side of the heater. Predictions of local wall temperature and fluid shear are then used in a prediction of local fouling rates.

FURNACE MODEL

Furnaces come in several configurations as presented by Berman (1978). The use of a convective heat recovery section following the radiant chamber is common to most configurations.

It is assumed that the heat transfer to the tubes positioned in the “fire box” is solely by radiation and a uniform radiant heat flux can be used. Since the quantity of heat transferred in the radiant section is known, the temperature at which hot gas leave this section and enter the convective section are easily calculated.

In the “convective section” it is assumed that the heat transfer is solely by convection. The temperature of the hot gas is reduced as it passes over each individual row of tubes and it is assumed to be thoroughly mixed between tube rows (this is a significant approximation).

HEAT DEMAND CURVES

Heat transfer on the crude-side is greatly enhanced once a two-phase flow has formed and film heat transfer coefficients increase with increase in the vapour mass quality of the stream. Calculation of the vapour mass quality requires knowledge of the vapour-liquid equilibrium. This knowledge is best provided in the form of two individual relationships. The first relationship is that between the fraction of feed vaporised and the quantity of heat transferred to the oil. The second relationship is between the fraction of feed vaporised with saturation temperature.

Such relationships are easily generated for a given pressure. Unfortunately they are dependant upon pressure which varies along the flow path. The approach adopted here was to use the facilities provided in the ASPEN+ program to develop a series of curves at differing pressures (between the pressure at the inlet to the let-down valve and the distillation column pressure). A polynomial having constants that were functions of pressure were then developed from these individual relationships.

HEAT TRANSFER EQUATIONS

Single phase convective heat transfer was calculated using the Dittus-Boelter equation.

Two-phase convective heat transfer was calculated using the momentum-heat transfer analogy proposed by Chen (1966).

$$\frac{\alpha_{TP}}{\alpha_L} = \left(\frac{\Delta P_{TP}}{\Delta P_L} \right)^{0.44} \quad (1)$$

α_{TP} and α_L represent the two phase and the heat transfer coefficient that would have been encountered if the liquid phase flowed alone respectively.

Nucleate boiling heat transfer was ignored.

The vapour phase mass transfer resistance has a major effect upon surface temperature. This effect was calculated using the Bell & Ghally method (1972) described by Collier & Thome (1996).

PRESSURE DROP EQUATIONS

Standard equations were used for the estimation of single phase pressure drop.

The Chisholm equation:

$$\frac{\Delta P_{TP}}{\Delta P_L} = 1 + \frac{C}{X_{tt}} + \frac{1}{X_{tt}^2} \quad (2)$$

where

$$C = \left(\frac{\rho_L}{\rho_g} \right)^{1/2} + \left(\frac{\rho_g}{\rho_L} \right)^{1/2} \quad (3)$$

was used for the calculation of two-phase pressure drop. A uniform core model (Yeap et al, 2004) was used for the calculation of the effect of fouling on pressure drop.

TWO PHASE FLOW PATTERN

A variety of flow patterns can occur during two phase flow through horizontal pipes. Inside fired heaters it is important that the surfaces are fully wetted at all times. This means that three of the reported flow patterns are unacceptable. These are stratified, wavy and spray.

Flow patters during flow through pipes can be predicted using the flow map proposed by Taitel & Dukler (1976) and reproduced in Figure 1.

The regions can be approximated using a series of straight line equations (the dotted lines in the figure). Then, the determination of flow pattern can be made using the following algorithm.

Annular flow is present if the following criteria are met:

$$X_{ii} < 1.0 \quad \text{and} \quad Fr > 1 - 0.8X_{ii}$$

Where:

$$X_{ii} = \sqrt{\frac{\Delta P_L}{\Delta P_G}} \quad (4)$$

$$Fr = \frac{m_G^*}{(\rho_G(\rho_L - \rho_G)dg)^{1/2}} \quad (5)$$

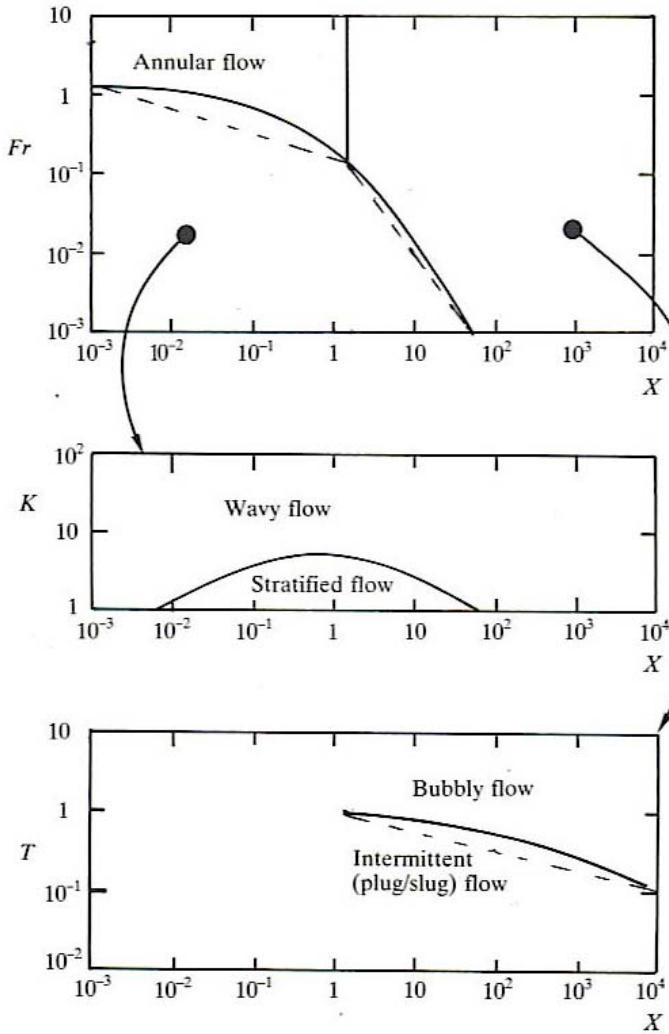


Figure 1. Two-Phase Flow Patterns in Horizontal Pipes

The flow pattern will be intermittent if the following criterion are met:

$$X_{ii} > 1$$

$$Fr > 0.2 - 0.0028 X_{ii}$$

Otherwise the flow will be wavy or stratified.

FOULING MODEL

A generalised form (Polley 2011) of the Ebert-Panchal Model (1997) was used for the calculation of local fouling rates within the fired heater:

$$\frac{dR_d}{d\theta} = \frac{A}{\alpha} \exp\left(\frac{-E}{RT_f}\right) - \gamma\tau_w \quad (6)$$

The heat transfer coefficient was taken to be the single phase coefficient if the crude was below saturation temperature, or the two phase convective heat transfer coefficient if a two phase flow was present.

The wall shear stress is calculated from the local pressure gradient (which also depends on fluid state).

The following model parameters were assumed:

Activation Energy	42	kJ/mole
Deposition Constant	20	m ² K/W.hr
Suppression constant	3e-9	m ² K/W.hr.Pa

These are simply representative values. In applications these parameters would be determined from analysis of pre-heat train behaviour.

The sensitivity of behaviour to changes these parameters is demonstrated below using a case in which the deposition constant is halved.

CALCULATION PROCEDURE

Since, the objective of the study is to determine the effect of inlet conditions on the fouling occurring within the fired heater the analysis has been simplified by assuming that the fuel consumption is not adjusted as the outlet conditions deteriorate. In practice we could expect the firing to increase as fouling progresses in order to provide fixed boil-up for the column.

Given fixed fuel consumption the temperature at which the hot gases exit the last row of tubes in the convective section is a direct measure of fuel efficiency. The lower the temperature the higher is the fuel efficiency.

In this analysis it has been assumed that the crude oil enters the heater through the row positioned at the exit from the heater as shown in Figure 2.

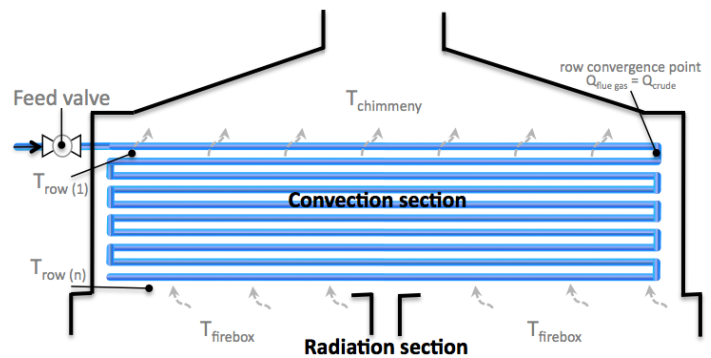


Figure 2. Contacting Arrangement in Convective Section
It is assumed that the following parameters are known: crude oil flow rate, temperature at exit from pre-heat train, pressure at exit from pre-heat train, pressure at exit from fired heater, radiant heat flux in heater, fuel consumption and heat flux in the radiant section. Flue gas flow rate is determined from a heat balance and the addition of 15% excess air.

From the fuel flow rate, and furnace geometry, the hot gas temperature in the radiant chamber is determined using a

well stirred furnace model, as described by Shires et.al. (1994). This temperature is assumed to be the inlet condition to the convection section.

The first iteration loop relates to the temperature profile across the convective section. Starting from the chimney temperature (first guessed), heat balance between flue gas and crude oil in each row is carried out to determine the flue gas in the next row (closer to the firebox). Chimney temperature is obtained when convergence between last row flue gas temperature and firebox temperature match (calculated from the well stirred model mentioned above).

For operating time of zero, a pressure at the entrance to the fired heater is assumed. A flash calculation is performed in order to determine the feed conditions at entry (temperature, vapour mass quality).

In the convective section of the heater the hot gas contacts the cold stream in a cross-flow manner. It is therefore assumed that each point along the tube encounters the same hot side conditions. The hot-side heat transfer coefficient is low. This results in a constant wall temperature condition.

The heat transfer along the first tube row in the convective section is calculated by numerical integration of the heat transfer and pressure drop equations. Local fouling rates are calculated. The gas-side thermal changes are then bulked and applied in the analysis of the behaviour of the next tube row in the convective section. The integration procedure is applied to each tube row in the convective section. The integration is then conducted across each tube row in the radiant section. Here the heat flux at each location is assumed to be the same.

The pressure at the exit form the heater is then known and compared with the specified value. If there is a significant difference the inlet pressure is adjusted and the procedure is repeated.

A time interval is then applied and the change in local fouling resistances calculated from the predicted local fouling rates. The analysis is then repeated until a specified operating period has been covered.

CASE STUDY

The fired heater was assumed to handle 88 000 barrels/day of light Arabian crude. The furnace geometry is given in Table 1.

Table 1. Geometry of Fired Heater

	Convective Section	Radiant Section
Tube i.d./o.d m	0.135/0.141	0.145/0.152
Length/pass m	16.64	18
Total length m	166.4	324
Coils	8	8 (4 per box)
Passes/coil	10	18
Passes/row	1	1

The pressure at the end of the pre-heat train was 12 bar. The pressure at the exit from the fired heater was 3 bar. The affect that three separate conditions have on fired heater behaviour have been examined. The first is associated with the furnace design and how well the gas side conditions are controlled and involves the assumed gas-side heat transfer coefficient. The second of these is the effect of crude fouling model parameters. The third is the effect of the amount of heat recovery achieved in the pre-heat train itself.

Gas Side Conditions

Modelling of the gas side behaviour in the convective section is difficult. This is because velocities across the coils are difficult to predict, the coils are usually fitted with fins in order to enhance the gas side heat transfer coefficient but ash tends to deposit on these surfaces.

The problem is simplified by looking at two individual cases: a gas in which it is assumed that the gas-side behaviour is what would be expected with a constant plain surface heat transfer coefficient of 150 W/m²K and this value is maintained throughout the operating period. This case provides insight in the value of cleaning the gas side of a fired heater. In the second case it is assumed that the behaviour is equivalent to a plain surface heat transfer coefficient of just 50 W/m²K.

Crude Inlet Condition

When the crude oil enters the expansion valve at a temperature of 280 C a two phase flow forms at the entry of the coil. The vapour mass quality of this flow is around 0.04.

At an inlet temperature of 250 C there is insufficient superheat in the liquid to generate vapour at the inlet to the fired heater. So, in the convective section of the fired heater the flow is initially single phase.

Influence of Gas Side Behaviour on Fuel Efficiency

The temperature profiles predicted for the convective section of the furnace are presented in Figures 3 and 4.

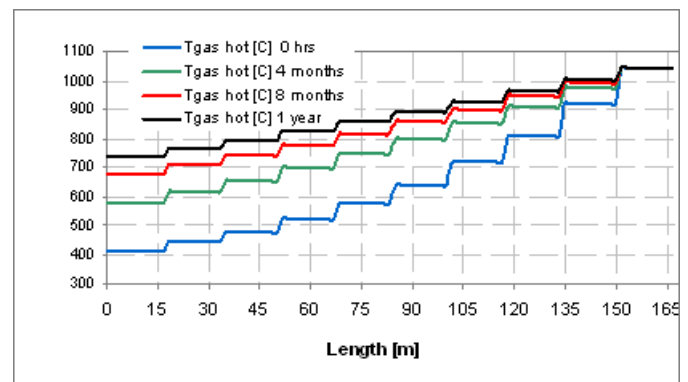


Figure 3. Gas Temperature Profiles: Pre-heat Train Exit of 280 °C, Gas Side Coefficient 150 W/m²K.

For the case in which the gas side coefficient is $150 \text{ W/m}^2\text{K}$ under a clean condition the overall heat transfer coefficient for the coils is around $110 \text{ W/m}^2\text{K}$. The heat extraction is efficient and the temperature of the hot gas falls rapidly. The initial stack temperature is around 400°C (gas temperature at length zero). However, the fouling within the coils builds up rapidly and after four months operation the stack temperature has risen sharply. After one years operation the stack temperature has risen to 740°C .

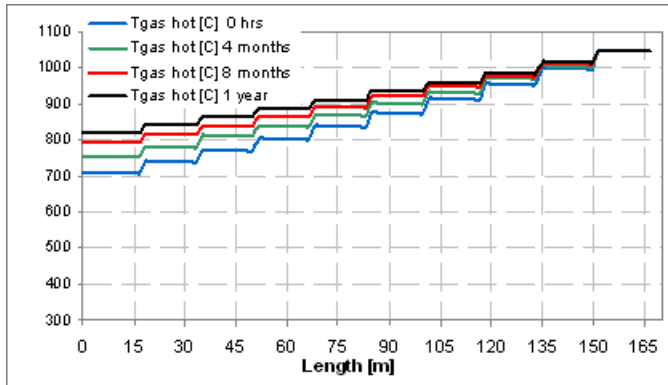


Figure 4. Gas Temperature Profiles: Pre-heat Train Exit of 280°C , Gas Side Coefficient $50 \text{ W/m}^2\text{K}$.

With a gas-side heat transfer coefficient of $50 \text{ W/m}^2\text{K}$ the extraction of heat from the hot gas is much less efficient and the stack temperature for the clean condition is much higher (at 700°C). The stack temperature after one year operation is 820°C .

An exhaust temperature in excess of 600°C would generally be considered unacceptable. For the lower gas side coefficient we find that this happens in the “clean” state. Therefore, the convective section should be engineered to provide a higher gas side coefficient (through closer spacing of tubes and use of extended surface area).

At the higher gas side coefficient the initial exhaust temperature is just 400°C but fouling inside the tubes results in an increase to 600°C after 4 months operation.

The convective section is under-surfaced for heating a crude oil having such fouling behaviour.

Influence of fouling rate on behaviour

Proceeding on the basis of the higher plain surface heat transfer coefficient the effect of fouling rate on fuel efficiency was examined.

The temperature profiles predicted for a crude oil characterised by a deposition constant that is half of that assumed above are shown in Figure 5.

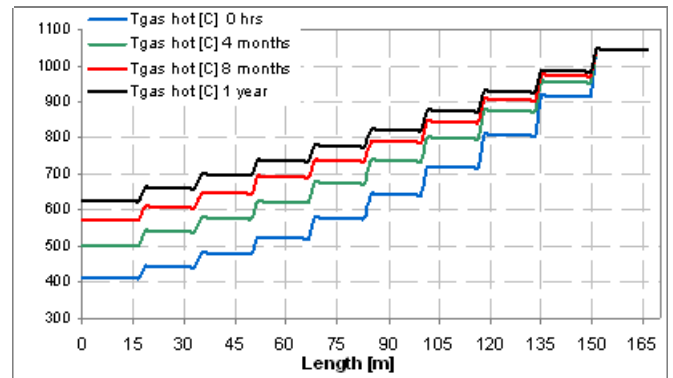


Figure 5. Gas Temperature Profiles when deposition constant is 10. Pre-heat train exit temperature 280°C , Gas Side Coefficient $150 \text{ W/m}^2\text{K}$.

The fouling rate has a very marked effect upon efficiency. The stack temperature after one year operation is now 620°C as opposed to 740°C . The fired heater geometry is acceptable for a crude oil having this fouling behaviour.

Influence of Coil Inlet Condition

If fouling rate has a significant effect on fuel efficiency then it is necessary to consider how the conditions at the inlet to the coil will affect behaviour. With a crude oil inlet temperature of 280°C a two phase flow is present at the coil inlet and the variation of fouling rates along the length of the heater is shown in Figure 6.

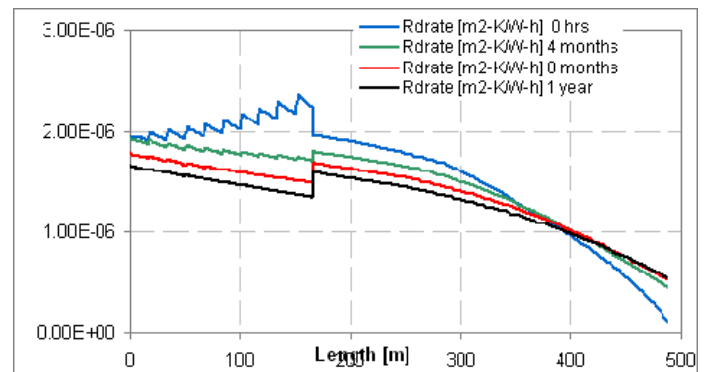


Figure 6. Fouling rate distribution. Pre-heat train exit temperature 280°C , Gas Side Coefficient $150 \text{ W/m}^2\text{K}$.

Under clean conditions the large changes in hot gas temperature across the convective section lead to a large variation in fouling rate across the section. Individual coils show a step increase in fouling rate as the start of the section is approached. Within each coil the rate is partially mitigated by increasing two-phase heat transfer coefficient and increasing wall shear stress. A step change in behaviour is observed at the end of the convective section, start of radiant section. As operating time increases the thickness of the deposit increases which leads to a reduction in fouling rate.

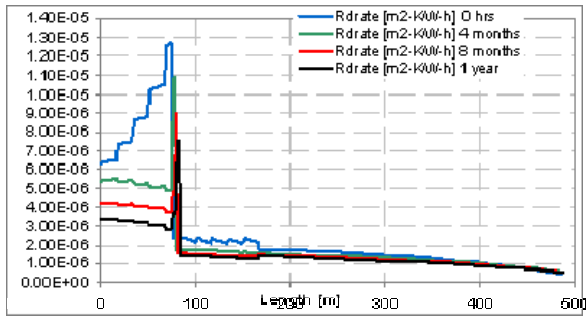


Figure 7. Fouling rate distribution. Pre-heat train exit temperature 250C, Gas Side Coefficient 150 W/m²K.

With a crude oil temperature at the end of the pre-heat train of 250 C the crude enters the coil as a sub-cooled liquid. The fouling behaviour under these conditions is shown in Figure 7.

The sub-cooled length extends to 90 meters. Under clean conditions the fouling rates are very high. However, once a two phase flow is formed they fall rapidly.

The rates fall rapidly as time progresses. However, throughout the operating period the fouling rates are greater than those encountered when the flow enters as a two-phase fluid. The stack temperature after one year operation is predicted to be 800 C (an increase of 60 C over that for a two phase inlet condition).

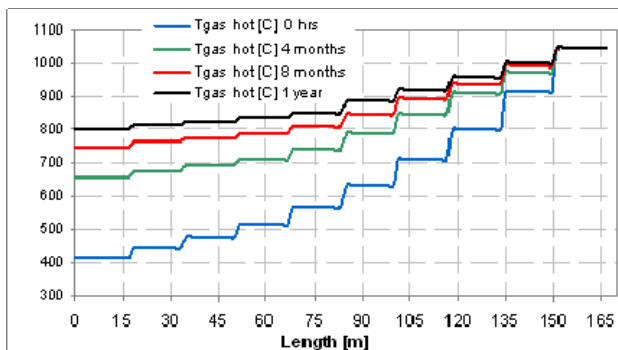


Figure 8. Gas Temperature Profiles for pre-heat train exit temperature 250C, Gas Side Coefficient 150 W/m²K.

Interaction of Hot and Cold Side Behaviour

The fouling rates predicted for a single phase inlet to the coil when the gas side behaviour is equivalent to a constant plain coefficient of 150 W/m²K is shown in Figure 9. Here, under clean conditions, the fouling rate in the first coil is greater than that shown in Figure 7. This is because the surface temperature is greater due to the gas side conditions being hotter. However, this increase in fouling rate is accompanied by an increase in heat flux. The crude heats up more rapidly and rather than the sub-cooled length extending to 90 metres it only extends to 48 metres.

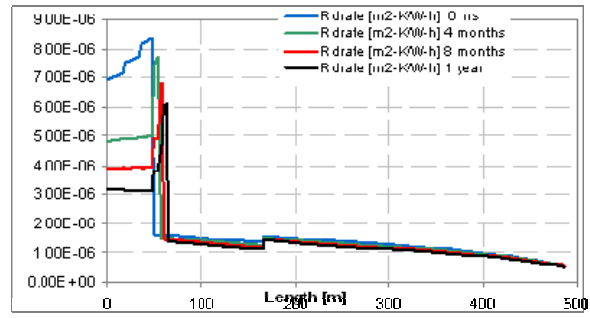


Figure 9. Fouling rate distribution. Pre-heat train exit temperature 250C, Gas Side Coefficient 50 W/m²K

As coil length extends beyond 48 metres hot side temperature gets greater and greater and if a single phase flow is present the fouling rate increases rapidly. So, when a liquid is introduced into the coil better heat transfer on hot side can give rise to increase in sub-cooled length and crude side fouling rates. The effect that this has on fuel efficiency is shown in Figure 10. The increase in fouling rate does not counteract the benefit of cleaning the hot side of the coils.

CONCLUSIONS

Fouling within fired heaters is a complex problem. This study has looked at the effects of crude oil fouling rate on overall behaviour. It was found that halving the crude side rate had a major effect on fuel efficiency.

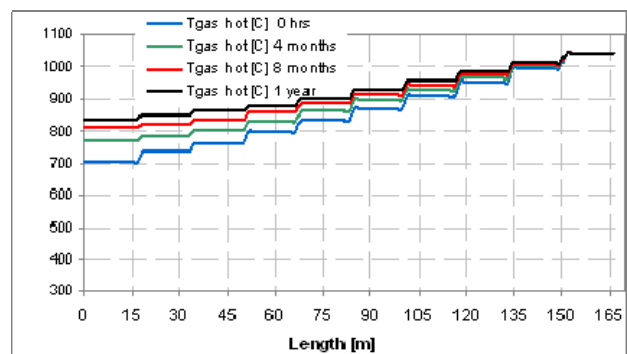


Figure 10. Gas Temperature Profiles when pre-heat train exit temperature 250C, Gas Side Coefficient 50 W/m²K.

Examination of hot side conditions indicated that cleaning of the hot side is important. However, cleaning of the hot side can result in a marked increase in fouling on the crude side if the crude enters the coils as a liquid. Not only is the overall fouling rate across the convective section increased but the length of coil that handles a single phase liquid can increase significantly.

Despite the increase in crude side fouling levels control of fouling on the hot side is still important. The fouling of hot surfaces is not the only problem that can arise from excessive fouling within the convective section. Increased fouling means increased coke formation. This can

subsequently lead to coke being moved from the fired heater and deposited in the distillation column itself.

In design and operation of pre-heat trains engineers should endeavour to ensure that the heat recovery levels achieved within the train are sufficient to ensure that the crude enters the fired heater as a two-phase mixture.

Finally, it is necessary to re-state that fouling on the crude-side has been assumed to follow a model that, whilst having been widely applied to heat exchanger fouling, has not been shown to apply at the levels of wall shear encountered in two phase flow. There is, as yet, no basis for assuming that the model will apply in two phase flow situations. Furthermore, changing the parameters of the model could also result in different conclusions.

Much work is being conducted on the development and refinement of fouling models. It is the authors' view that the short comings stated above are likely to be removed through this ongoing work.

NOMENCLATURE

A	Deposition constant	--
E	Activation Energy	kJ/mole
d	diameter	m
Fr	Froude Number	--
*		
<i>m</i>	mass flux	kg/m ² s
P	Pressure	Pa
R	Universal Gas Constant	
R _d	fouling resistance	m ² K/W
T	temperature	K
X _{tt}	Lockhart-Martinelli Parameter	--
α	heat transfer coefficient	W/ m ² K
γ	suppression constant	m ² K/W.hr.Pa
θ	time	hours
ρ	density	kg/m ³
τ	shear stress	Pa
Subscripts		
g	gas	
L	liquid, or liquid fraction flowing alone	
TP	two phase	
w	wall	

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