SHELL-AND-TUBE HEAT EXCHANGER GEOMETRY MODIFICATION: AN EFFICIENT WAY TO MITIGATE FOULING

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ABSTRACT

Crude oil fouling of a shell-and-tube heat exchanger sized according to TEMA standard is compared to a *No-Foul* design under industrial operating conditions. For similar operating conditions, TEMA and *No-Foul* heat exchangers have the same behavior regarding fouling. Since the *No-Foul* one has less tubes by design for the same heat duty, shear stress is increased. Consequently, the *No-Foul* heat exchanger is less prone to fouling at the same throughput. Impact of tube bundle geometry is then investigated. Helically-finned tubes are compared to plain tubes in the *No-Foul* heat exchanger. Under similar operating conditions, fouling rates measured are up to an order of magnitude lower than plain tubes (respectively 10^{-11} and 10^{-10} m^2 K/J). However, pressure drop across the tube-side in both *No-Foul* plain and finned setup are increased in comparison to the TEMA heat-exchanger.

INTRODUCTION

Fouling of heat exchangers is the build-up of fouling layers on the heat transfer surface. Crude oil fouling in refineries causes several operating, financial and safety issues. Before entering the atmospheric column, crude oil is usually heated in the preheat train by recovering heat from hot streams. But thermal performance is reduced by fouling due to an increase of the thermal resistance leading to additional fuel consumption and CO_2 emissions. Moreover, the gradually decrease in tube cross-sectional area due to the fouling layer growth requires more pumping power to maintain the throughput. Ultimately fouling can cause a fluid blockage. Before reaching this stalemate, cleaning of heat exchangers is required. It could be performed using chemical or by dismantling and water-jetting the equipment. All these consequences are the root of numerous additional expenditures. Costs linked to fouling are estimated to USD3.6 billion a year (Coletti and Hewitt, 2014).

Unfortunately, fouling phenomenon remains still not well understood. Therefore, several mitigations techniques have been explored to manage fouling in preheat train such as optimization of operating conditions, preheat train monitoring to optimize cleaning cycles (Müller-Steinhagen et al., 2011; Ishiyama et al., 2010), use of chemicals (Baxter et al., 2004; Brant et al., 2009), tube inserts (Aquino et al., 2007; Krueger and Pouponnot, 2009), etc. This study focuses particularly on the technique of enhanced heat exchanger geometries.

Shell-and-tube heat exchangers are commonly set up in hydrocarbon services since it is perceived in industry as a safer option (Coletti and Hewitt, 2014). Nevertheless, fouling of other heat exchanger technologies have also been investigated so far. Compact and spiral heat exchangers successfully mitigate fouling (Tamakloe et al., 2013; Wilhelmsson, 2005) by providing higher shear stress at a given flowrate. Shell-and-tube heat exchanger geometry modification (helical baffle, EM-Baffle or twisted tubes) can also help to mitigate fouling but improvement is limited to the shell-side where dead zones are suppressed (Master, 2003; Brignone et al., 2015).

However, tube-side geometry modification has received little attention up to date. Yang et al. (2013) and Crittenden et al. (2015) studied the fouling of an enhanced surface (by the mean of an incorporated wire) and compared it to a smooth one. For the same operating conditions, fouling resistance measured were lowered with the enhanced surface. Provost et al. (2013) and Sippel et al. (2015) published a feedback on fouling of internally helically finned tube with quench water of an industrial naphtha cracker. Enhanced tubes did not undergo any significant fouling after 24 months of unit operation. To the authors' knowledge, no fouling study of these tubes has been carried out with crude oil.

That's why we will study the influence of the sizing method and tube technology on fouling. Hence, fouling propensity of a shell-and-tube heat exchanger sized according to TEMA standards (TEMA, 1968) is compared to a heat exchanger sized with a *No-Foul* method as described by Bennett and Nesta (2004). Impact of tube bundle technology is secondly addressed by testing successively plain and helically-finned tubes in the *No-Foul* heat exchanger.

Helically-finned tubes – whose geometry features are illustrated in Fig. 1 – are provided by WIELAND. Compared to the plain tubes, internal area is increased by 30% (*i.e.* $A_{i,fin} / A_{i,plain} = 1.30$).

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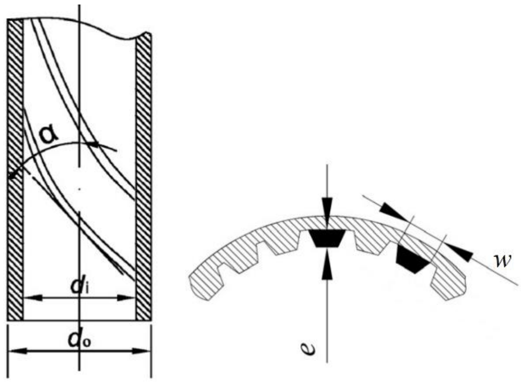

Fig. 1 Geometry description of a helically-finned tube (Ji et al., 2012).

EXPERIMENTAL SETUP

Fouling rig

The heat exchanger to be tested (the test section) is settled in a fouling rig described successively by Ratel et al. (2013) and Chambon et al. (2015) during past issues of this meeting. So, the features of the rig are not recalled here and the reader is strongly invited to refer to these former papers to learn about it.

As in the industrial process, atmospheric tower bottom (ATB) flows through the shell side to heat up the crude. Oil used for experiments was sourced from the Black Sea area. Physical properties and chemical composition of both fluids are respectively reported in Table 1 and 2.

Table 1. Physical properties of the oil and the ATB.

Physical properties at 300°C	Oil	ATB
Density (kg/m^3)	675	789
Dynamic viscosity (10^4 N s/m^2)	3.69	7.98
Thermal conductivity (10^2 W/m/K)	9.14	9.11
Heat capacity (J/kg/K)	2 990	2 884

Table 2. Chemical composition of the oil and the ATB.

Fluids used for testing:	TEMA		*No-Foul*	
	Oil	ATB	Oil	ATB
Asphaltenes (%w)	1.1	2.5	1.1	0.46
CCR	3.92	5.96	4.45	5.19
Sulphur (%w)	1.22	2.25	1.37	0.56
Coking factor	0.81	2.05	0.77	1.25
Cracking factor (%w)	1.92	2.20	1.99	2.37
Σ C aromatics (%w/w)	17.5	24.5	16.2	17.5
Σ C saturates (%w/w)	82.5	75.6	83.8	82.5

Crude oil and ATB used for *No-Foul* tests come from a different batch than that used for TEMA tests, nevertheless their composition shows a very strong similarity. However, the ATB used for the *No-Foul* runs would appear to contain less sulfur and asphaltenes which are known as fouling precursors. As a consequence, the ATB should have, *a priori*, a less fouling propensity than the one used for TEMA tests.

Heat exchanger designs

The sizing method originally proposed by the TEMA is known to promote fouling. Fouling resistances listed by the TEMA are fixed values whereas the fouling is a dynamic phenomenon which depends on the operating conditions. These inappropriate values lead to oversize heat exchanger which, finally, amplifies their fouling. Indeed, the addition of extra heat transfer surface tends to reduce flow velocities and the convective heat transfer coefficients and finally increase the wall temperatures. That's why Bennet and Nesta (2004) have developed a sizing method that takes better account of fouling.

The *No-Foul* test section is sized to fulfill their recommendations:

- tube-side wall temperature below 300°C and flow velocity higher than 2 m/s at rated throughput;
- on the shell side, leakage currents are minimized and cross-flow (B-stream) promoted;
- over-sizing is reduced to a 15% extra heat transfer area and no fouling resistance is considered in the design.

A comparative datasheet in Table 3 gives the geometrical characteristics of the exchangers tested. Each tubular heat exchanger tested is made with carbon steel and has a similar heat flux level and the same tube gauge than the heat exchanger of a refinery pre-heat train.

Table 3. Geometrical characteristics of the shell-and-tube heat exchangers compared.

Heat exchanger design		TEMA	*No-Foul*	
Tube bundle		Plain	Plain	Finned
Tube length	m	2.0	2.0	
Tube outer diameter (d_o)	mm	25.4	25.4	
Tube inner diameter (d_i)	mm	19.9	19.9	21.4
Tube count	—	32	28	
Tube pitch	mm	31.75	31.75	
Tube arrangement	—	square	square	
Tube pass	—	4	4	
Shell pass	—	1	1	
Shell internal diameter	mm	260	248	
Number of baffle	—	17	13	
Baffle spacing	mm	100	115	
Baffle window	%	20	24	
Tube-to-baffle clearance	mm	0.9	0.4	
Shell-to-baffle clearance	mm	2.0	0.9	
Heat transfer area (A_o)	m^2	5.1	4.3	
Heat duty	W	5 10^4	5 10^4	
Heat flux	W/m^2	10^4	1.2 10^4	
Fouling for sizing	m^{2}K/W	1.9 10^{-3}	0	

Consequently, at constant flowrate, the *No-Foul* heat exchanger requires less heat transfer area than the TEMA in order to transfer the same duty. This results in fewer tubes. Hence, the flow velocity and the wall shear stress are increased, promoting suppression of fouling deposit. However, the *No-Foul* exchanger will generate more pressure drop than the TEMA due to the higher flow velocities.

Operating conditions range

Crude oil velocities and film temperature investigated respectively have been selected to be representative of industrial operating conditions.

Tests are performed in order to investigate effect of film temperature and velocity separately. After each fouling run, both sides of the test heat exchanger are flushed with gasoil for at least two days in order to remove as fouling material as possible.

Data reduction

Fouling resistances and fouling rates are derived from measurements of the overall heat transfer coefficient U(t). The latter is figured out with Eq. 1 by continuous monitoring of inlet and outlet temperatures. More details are given in the previous study (Chambon et al., 2015).

$$Q = F\, U(t)\, A_o\, LMTD \tag{1}$$

In Eq 1, Q is the mean duty derived from hot and cold-side heat balances. LMTD is the logarithmic mean temperature difference and F the correction factor. In order to make reliable fouling propensity comparisons fouling rate and resistances must be related to the same heat transfer surface. Since plain and helically-finned tubes have the same outer diameter (1”), the external heat transfer area (A_o) is chosen. Hence, all the fouling resistance is affected outside the tubes albeit it occurs on both sides of the heat exchanger.

Tube wall shear stress, τ, is figured out with Eq. 2, where ρ and u are respectively crude oil density and velocity.

$$\tau = \frac{1}{2} f \rho u^2 \tag{2}$$

The Fanning friction factor *f* is respectively calculated with Blasius equation (Eq. 3) for plain tubes and with the Zdaniuk et al. (2008) correlation (Eq. 4) for helically-finned tubes. In these equations, Re refers to the crude Reynolds number and N, e, α respectively to the fin height, the number of fin starts and the helix angle (see Fig. 1).

$$f = 0.0791 Re^{-0.25} \tag{3}$$
$$f = 0.128 Re^{-0.035} N^{0.235} (e/d_i)^{0.319} \alpha^{0.397} \tag{4}$$

The Zdaniuk et al. (2008) correlation has been chosen since it gives the better agreement with experimental pressure drop measurements made during a preliminary study.

According to Ebert and Panchal (1995), the film temperature (T_f) is a weighted average between wall (T_w) and bulk (T_b) temperature (Eq. 5).

$$T_f = T_b + 0.55(T_w - T_b) \tag{5}$$

Bulk temperature is assumed to be the average between crude inlet and outlet temperature during the whole run duration. Wall temperature is derived from Eq. 6.

$$T_w = T_b + \frac{Q}{h_i A_i} \tag{6}$$

Where h_i is the internal convective heat transfer coefficient and A_i, the effective internal heat transfer area. Convective heat transfer coefficient is respectively figured out with Dittus-Boelter (Eq. 7) and Zdaniuk et al. (2008) correlation (Eq. 8) for plain and helically-finned tubes.

$$\frac{h_i d_i}{k} = 0.023 Re^{0.8} Pr^{0.4} \tag{7}$$
$$\frac{h_i d_i}{k} = 0.029 Re^{0.653} N^{0.253} (e/d_i)^{0.0877} Pr^{0.33} \tag{8}$$

In these equations, k is the thermal conductivity of crude oil (see Table 1) and Pr, the crude Prandtl number.

RESULTS AND DISCUSSION

TEMA vs. *No-Foul* design

Comparison at similar operating conditions. Results are shown in a temperature-velocity plot in Fig. 2. Tests performed with the TEMA heat exchanger are labelled with ‘T-P’ whereas ‘NF-P’ refers to the *No-Foul* one. The label is followed with the run identification number. If fouling is detected during a run, relative fouling rate (to the TP-4 test) is indicated into brackets. Moreover, the size of the circle is proportional to the measured fouling rate.

The curve plotted represents the threshold conditions calculated using the Panchal et al. (1999) correlation. Parameters have been optimized with the fouling rates measured on both TEMA and *No-Foul* heat exchanger. This graph allows to delimit two areas: a fouling area above the curve and a no-fouling area below the curve.

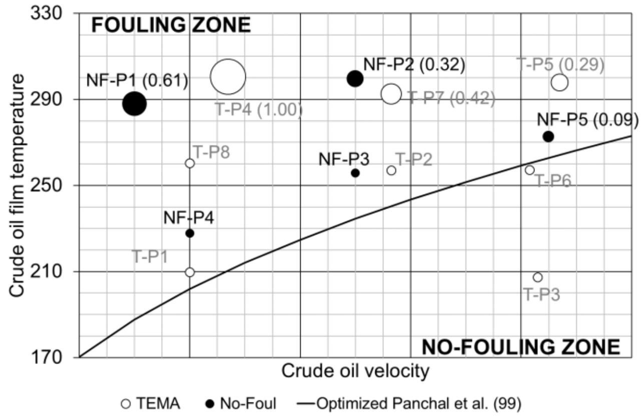

Fig. 2 Fouling experiment results for the TEMA and *No-Foul* heat exchangers as a function of crude oil velocity.

Only tests NF-P2 to NF-P5 can be compared to the results of the T-P series. Flow velocity and film temperature of NF-P1 run is too far from T-P4 and T-P8.

The fouling rate of tests NF-P2 and T-P7 are comparable. Although on NF-P2 the film temperature is higher, the crude velocity (wall shear stress) is lower and the ATB cross-flow velocity is 20% lower. The gap could be explained by a little bit lower fouling propensity of the ATB (see Table 2).

The absence of fouling for the NF-P3 and T-P2 runs proves that they are comparable. On test T-P2 a fouling resistance close to uncertainty was detected while no fouling was detected on test NF-P3. These observations seem to be consistent with the lower crude oil temperature and with the lower fouling propensity of the ATB.

Although test NF-P4 is carried out at a higher temperature than test T-P1, no fouling is noticed. The threshold temperature was probably not reached. Referring to

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Fig. 2, the *real* threshold curve would be positioned above NF-P4.

For a similar wall shear stress (*i.e.* oil velocity) on tests NF-P5 and T-P5, a lower fouling rate was observed for the NF-P5 test. This is in accordance with the lower film temperature and highlights the strong impact of temperature on fouling growth. It is confirmed by comparing NF-P5 and T-P6 (carried out at lower temperature and similar shear stress). Therefore, the *real* threshold curve is assumed to be located between these two points which is in accordance with the Panchal et al. (1999) forecast (see Fig. 2).

At low flow velocities and film temperatures, the equilibrium between deposition and suppression rates is slower to be reached (slow dynamics). Conversely, at higher velocities and film temperatures, the equilibrium of the two mechanisms is more rapidly set up (fast dynamics). Hence, the Panchal et al. (1999) correlation delineates the fouling and non-fouling areas poorly when the balance between the deposition and suppression rate is more difficult to set up. Forecasts are more accurate for high flow velocities when the dynamics is more pronounced.

To sum it up, for similar operating conditions (velocity and film temperature), the TEMA and *No-Foul* heat exchangers have the same behavior regarding fouling. Fouling rate and resistance measured are all included in measurement uncertainties.

Comparison at same throughput. Similarly to Fig. 2, results are gathered in Fig. 3 where horizontal axis is now the crude oil flowrate. This graph allows to compare the fouling propensity of a shell-and-tube heat exchanger according to the method used for its sizing.

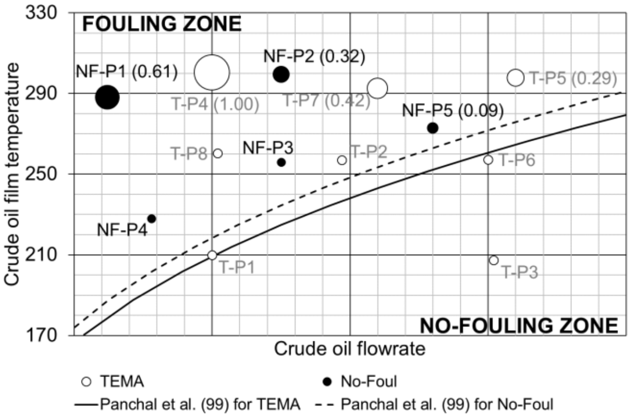

Fig. 3 Fouling experiment results for TEMA and *No-Foul* heat exchangers as a function of crude oil flowrate.

The two threshold curves are plotted with the same correlation (Panchal et al., 1999) and the same optimized parameters. They are not superimposed because the threshold condition depends on the wall shear stress but not directly to the crude oil flow rate. Indeed, for the same flowrate, wall shear stress is greater in the *No-Foul* heat exchanger because crude oil velocity is higher due to fewer tubes. Consequently, the *No-Foul* heat exchanger threshold curve is translated horizontally to the left-hand side of the graph.

By replacing the TEMA heat exchanger with a *No-Foul* one, fouling operating loci located between the two thresholds curves are moved into the non-fouling area. Thus, fouling is reduced because operating conditions are now in the non-fouling zone of the dashed curve.

Cost considerations. *No-Foul* heat exchangers prove to be more economical to produce because they are sized as tightly as possible. Since both exchangers are tubular heat exchangers, replacing a TEMA with a *No-Foul* avoid heavy modification of the unit because the bulk is similar and nozzle locations can be kept in place. Finally, OPEX costs are reduced since the *No-Foul* is less prone to fouling. The main drawback of the latter design is pressure drop due to the higher flow velocities at same throughput.

Helically finned vs. plain tube

In this section, the plain tube bundle of the *No-Foul* heat exchanger was replaced by a bundle of helically finned tubes. The two bundles have the same features: same tube count, same baffle spacing, same outer diameter, etc. The tube internal geometry is the only modified parameter. The goal is to compare the fouling propensity of these tubes with respect to the plain tubes at similar operating conditions.

Results of fouling tests are summarized as a function of the wall shear stress in Fig. 4. Fouling runs are named with the 'NF-F' label. Similarly, fouling rates are expressed into brackets relatively to the T-P4 value. For readability reasons, only T-P4 and T-P8 are plotted for the TEMA-series.

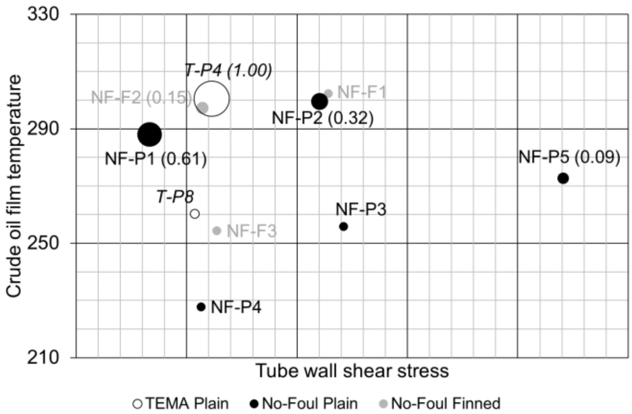

Fig. 4 Fouling experiment results for plain and helically finned tubes.

Fouling was detected during NF-P2 run carried out with plain tubes while no fouling is measured at the end of the test NF-F1. Both tests are performed at a similar wall shear stress. Nevertheless, shear stress figured out with the Zdaniuk et al. (2008) correlation (Eq. 5) is derived from experimental pressure drop measurements which include losses caused both by friction and drag. Thus, the shear stress assessed is an *overall* wall shear stress. Overall wall shear stress cannot explain the absence of fouling at the end of NF-F1 test. This suggests that local (rather than overall) wall shear stress acts in helically-finned tubes to prevent fouling.

The fouling rate and resistances measured on test NF-F2 are an order of magnitude lower than for the plain tubes (T-P4). Temperature monitoring of tests NF-F2 revealed that the tube-side was fouled first. However, it cannot be known whether the shell-side was subsequently involved in fouling.

The ATB processed on test T-P4 is assumed to be more prone to foul and could explain why fouling is more severe in this test but it cannot account for the large difference in fouling rates. The previous hypothesis of higher local shear stress in helically-finned is believed to explain fouling mitigation.

No fouling resistance was detected on the NF-F3 test, while a fouling resistance near of the uncertainty was measured on the T-P8 test. It is unlikely fouling on the T-P8 test is due to the ATB because under similar shell-side conditions, no fouling was measured in the T-P2 and T-P6 tests. Thus, the fouling resistance on the T-P8 test could only be due to crude oil. Since the NF-F3 and T-P8 tests are carried out at the same film temperatures and shear stress, the higher local shear stress seems, once again, to be the most likely hypothesis to explain the fouling mitigation.

Helically-finned tubes would appear to be less prone to fouling than plain tubes, at least for the range of operating conditions investigated. Fouling of helically-finned tubes has only been noticed for very low velocities and high film temperatures. For similar operating conditions, the fouling rate measured are is one order of magnitude lower than for plain tubes. However, the relatively short duration of the tests (300 to 500 h) and the limited number of experimental runs do not allow to conclude about fouling over the long periods of time and on other operating ranges.

The comparison of the tests carried out under similar operating conditions suggests a different rate of foulant growth which could explain the lower fouling propensity of helically-finned tubes. Fins could generate boundary layer detachments and impart swirling flow motion. Thus, at same film temperature and shear stress, the enhanced local turbulence would reduce the fouling rate.

However, the mitigation effect of finned tubes is counterbalanced by a significant increase in pressure drop compared to the plain tubes. At similar flowrate, the higher shear stress causes more pressure drop.

CONCLUSIONS

1. For similar operating conditions (velocity and film temperature), shell-and-tube heat exchangers sized according TEMA standard and with a *No-Foul* design method have the same behavior regarding fouling.
2. At the same throughput, the *No-Foul* sized heat exchanger has proven to be effective to mitigate fouling.
3. Helically-finned tubes have a lower fouling propensity than plain tubes. Fouling rate measured are an order of magnitude lower than for the plain tubes (10^{-11} and 10^{-10} m^2 K/J respectively). However, the relatively short duration of the tests (300 to 500 h) and the limited number of experimental runs do not allow to conclude about fouling over the long periods of time and on other operating ranges.
4. The lower fouling propensity of helically-finned tubes could perhaps be explained by a higher local wall shear stress due to local turbulence enhancement.
5. At the same throughput, pressure drop is higher in the *No-Foul* sized heat exchanger than in the TEMA sized one. Similarly, our finned tubes bundle causes significantly more pressure drop than in the plain tube bundle.
6. Tests carried out with a non-fouling fluid (alternately on the shell-side and on the tube-side) and a hydrocarbon fluid (crude or ATB) would allow to dissociate the contribution of each fluid in the fouling of the heat exchanger. Moreover, a better knowledge of mechanisms involved in crude oil fouling would provide additional evidences for understanding the lower fouling propensity of the helically-finned tubes.

NOMENCLATURE

Latin

A	Heat transfer area, m^2
d	Diameter, m
e	Fin height, m
f	Fanning friction factor, dimensionless
F	Correction factor, dimensionless
h	Convective heat transfer coefficient, $W/m^2/K$
k	Thermal conductivity, W/m/K
LMTD	Log mean temperature difference, K
N	Number of fin starts, dimensionless
Pr	Prandtl number, dimensionless
Q	Heat exchanger duty, W
Re	Reynolds number, dimensionless
T	Temperature, K
u	Flow velocity, m s^{-1}
U	Overall heat transfer coefficient, $W/m^2/K$
w	Fin thickness, m

Greek

α	Fin helix angle, °
ρ	Density, kg/m^3
τ	Wall shear stress, N/m^2

Subscript

b	bulk
f	film
i	inner
o	outer
w	wall

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