

Accounting for fouling in plate heat exchanger design

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The fouling removal mechanism for plate heat exchanger (PHE) heat transfer surface controlled by shear stress is considered. The semi-empirical formula to predict asymptotic value of fouling thermal resistance velocity dependence based on heat and momentum transfer analogy is proposed. The correlation is in a good agreement with literature data and was checked for PHEs installed in District Heating systems of a number of cities in Ukraine. As a result the modified method for PHE heat transfer surface area calculation is proposed, which takes into account the influence of flow velocity and heat transfer enhancement in channels on fouling thermal resistance. The obtained correlation can also be used for estimation of fouling rate in process PHEs where one media is water, e.g. in distilleries, nitric and phosphoric acids, fertilizers productions.

1. Introduction

The fouling on heat transfer surface can significantly deteriorate the performance of any heat exchanger and should be correctly predicted in calculation of required surface area at the design stage. This is even more important for PHEs with enhanced heat transfer which as a rule have much high film heat transfer coefficients than conventional shell and tube units. The analysis of data available in literature has shown that thermal resistance of fouling in PHEs is up to ten times lower than recommended by TEMA for shell and tubes working at same conditions, but clear guidelines and correlations to predict the exact value of fouling in PHE on designing stage are absent.

The fouling in PHEs working with water as both heating and cooling media was investigated. There are conditions where precipitation and particulate solids fouling mechanisms are predominant. The rate of fouling removal can be theoretically estimated based on modified analogy of heat and momentum transfer. The correlation predicting asymptotic value of fouling thermal resistance dependence of flow velocity will be obtained here.

2. Semi-empirical modeling

The detailed analysis of fouling mechanisms and theories for fouling modeling can be found elsewhere (e.g. see Hesselgreaves, 2001, Shah and Seculic, 2003). For fouling at cooling water side of heat exchanger, mostly scaling and precipitating solids mechanisms, the deposition and reentrainment models are applied. By these models the change of fouling thickness δ with a time t is a difference between fouling deposition φ_d and fouling removal φ_r rates:

$$\frac{d\delta}{dt} = \varphi_d - \varphi_r \quad (1)$$

To estimate φ_r at shear stress controlled fouling mechanism the Kern and Seaton approach is mostly used (see Panchal et al, 1997). According to it:

$$\varphi_r = b * \tau_w * \delta, \quad (2)$$

where τ_w is wall shear stress, b - coefficient depending of deposit and fluid properties.

When fouling removal rate equals its deposition rate, we have $d\delta/dt=0$ and no fouling layer grows. It comes after long time of heat exchanger operation and asymptotic value δ^+ of deposited fouling thickness is achieved. At this conditions from (1) and (2) we have:

$$\delta^+ = \varphi_d / b\tau_w \quad (3)$$

In channels of PHEs the modified analogy of heat and momentum transfer can be used, as it was shown earlier by Tovazshnyanskyy and Kapustenko, 1984. Here we will use extended Reynolds analogy, as equationed by Shah and Bhatti, 1988:

$$\frac{Nu}{Re} Pr^{-1/3} = \frac{f}{2} \phi_w, \quad (4)$$

where $Nu = hD_e / k$ - Nusselt number; $Re = wD_e \rho / \mu$ - Reynolds number; $Pr = \mu c_p / k$ - Prandtl number; ϕ_w - function that modifies analogy depending on duct geometry, flow type and boundary conditions; f - friction factor; ρ - fluid density; μ - fluid dynamic viscosity; k - fluid thermal conductivity; w - flow velocity; D_e - equivalent diameter of channel; h - film heat transfer coefficient.

The wall shear stress can be calculated as:

$$\tau = f \rho w^2 / 2 \quad (5)$$

For fully developed turbulent flow in straight tube this analogy also holds true at $\phi_w = 1$ and friction factor can be estimated by Blazius equation

$$f = 0.0791 Re^{-1/4} \quad (6)$$

Excluding velocity w from equations (4), (5) and (6), one can obtain relationship between shear stress and film heat transfer coefficient:

$$\tau_w = h^{7/3} D_e^{1/3} A_h, \quad (7)$$

where
$$A_h = \left(\frac{2}{0.0791} \right)^{4/3} \frac{\mu^2}{(\phi_w k)^{7/3} Pr^{7/9} \rho}.$$

In channels of complex geometry, like PHE passages, the friction part constitutes only fraction of total hydraulic resistance and it is much more difficult to obtain equation similar to (7). Here we will assume that the relationship (7) holds true also in PHE channels at $\phi_w = 1$ and later compare the results with data available in literature.

Let's assume also that for the same fluid and deposit material under the same thermal conditions in two channels the rate of fouling deposition is equal. Under such conditions the value of coefficient A_h is also the same for both channels. Then, from equations (3) and (7) can obtained:

$$\frac{\delta_1^+}{\delta_2^+} = \left(\frac{h_2}{h_1}\right)^{7/3} \left(\frac{D_{e2}}{D_{e1}}\right)^{1/3} = \frac{R_{f1}^+}{R_{f2}^+} \quad (8)$$

Here R_{f1}^+ and R_{f2}^+ - asymptotic values of fouling thermal resistances in both channels, respectively. Accounting that film heat transfer coefficient dependent of flow velocity w , from equation (8) it is possible to estimate the influence of w on fouling thermal resistance in one channel ($D_{e1} = D_{e2}$) and to compare the result with data available in literature of Karabelas et al, 1997. From most empirical correlations for plate heat exchangers the influence of flow velocity on heat transfer can be described as:

$$Nu = \frac{hD_e}{k} \propto Re^n \propto \left(\frac{wD_e\rho}{\mu}\right)^n, \quad (9)$$

where $n = 0.65 \div 0.7$. Then the influence of flow velocity on fouling thermal resistance can be estimated as:

$$\frac{R_{f1}^+}{R_{f2}^+} = \left(\frac{w_2}{w_1}\right)^{n7/3} \approx \left(\frac{w_2}{w_1}\right)^{1.6}. \quad (10)$$

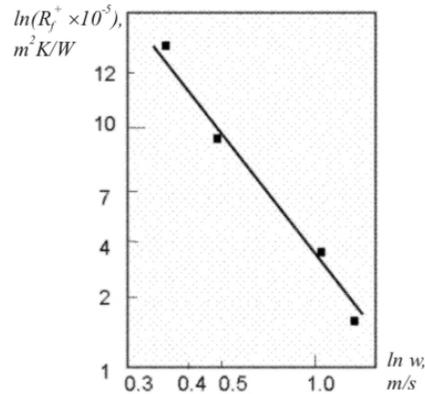


Figure 1. The comparison of velocity influence on asymptotic value of fouling thermal resistance predicted by equation (11) with experimental data of Karabelas et al, 1997.

We can conclude that for the same fluid properties for turbulent flow in the same channel the asymptotic value of fouling thermal resistance is inversely proportional to flow velocity in power 1.6:

$$R_f \approx b_f w^{-1.6},$$

where b_f - coefficient depending of fluid properties.

The comparison with experimental data presented on Figure 1 shows rather good agreement with predicted power of velocity influence, that confirm the validity of analysis above and obtained equations.

3. Results and discussion

The performed semi-empirical analysis has revealed the strong dependance of fouling at heat transfer surface of PHEs on film heat transfer coefficient for investigated type of fouling mechanism. It enables us to conclude that intensification of heat transfer in channels of PHE consist not only in enhancement of film heat transfer coefficient but also in even more significant reduction of fouling on heat transfer surface and its thermal resistance. Therefore this phenomenon should be accounted for in PHE design and in procedure for calculation of heat transfer area. Equation (8) enables to do this in case if there is a reference conditions for the same fluid where film heat transfer coefficient h_0 and corresponding asymptotic value of fouling thermal resistance

R_{f0}^+ are known. Using this assumption and suggesting that fouling of both fluids in PHE channels occurs by the same mechanism described above, we can write equation for calculation of overall heat transfer coefficient U in following form:

$$\frac{1}{U} = \frac{1}{h_1} + R_{f01}^+ \left(\frac{h_{01}}{h_1} \right)^{7/3} \left(\frac{D_{e01}}{D_{e1}} \right)^{1/3} + \frac{\delta_w}{k_w} + R_{f02}^+ \left(\frac{h_{02}}{h_2} \right)^{7/3} \left(\frac{D_{e02}}{D_{e2}} \right)^{1/3} + \frac{1}{h_2}. \quad (11)$$

To use this equation it is necessary to know reference values of h_0 and R_{f0}^+ , which depend on properties of fluid and its fouling deposit. These values can be obtained from site tests on individual enterprises or district heating networks. Such data for heat exchanger installed at one enterprise in Ukraine are presented on Figure 2.

For initial period of fouling growth ($t < t^+$) the following experimental correlation was obtained:

$$R_f(t, w) = (3.74 - 3.075w) \cdot 10^{-6} \cdot \sqrt{t}. \quad (12)$$

We can estimate R_{f0}^+ from this equation at $t=2000$ h and h_0 calculated on respective correlation for this heat exchanger. After determination of reference values h_0 and R_{f0}^+ , they can be used in equation (11) for thermal design of PHEs working on the same enterprise with the same cooling water. More over they can be used to estimate fouling on the process side of heat exchanger from data on overall heat transfer coefficient during on site tests of such equipment.

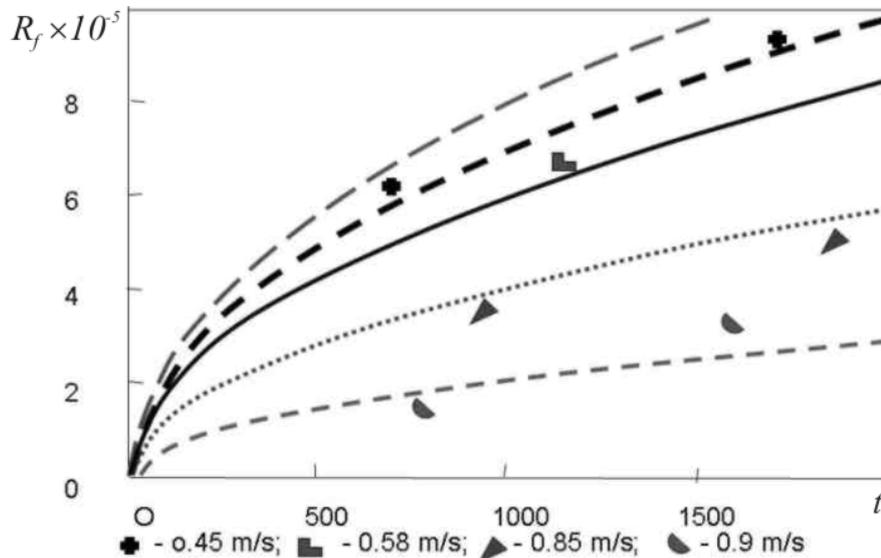


Figure 2. Data for thermal resistance of fouling in plate heat exchanger installed on site.

District heating is the typical application of PHEs where both heat exchanging fluids are water streams. The quality of water in various locations is different, as well as its fouling features. The on site monitoring of PHEs installed in different district heating networks of Ukraine by AO Sodrugestvo-t company enabled us to evaluate reference values of h_0 and R_{f0}^+ for a number of locations. It makes possible to use design procedure based on equation (11) for accounting of fouling when selecting new PHEs for these applications. The predicted results are in good agreement with data obtained from on site tests.

From equation (11) one can conclude, that the influence of film heat transfer coefficient enhancement on overall heat transfer coefficient after long operational time is much higher with fouling then for clean medium. The more intensified is heat transfer the less fouling resistance after long run. Because of this the constant value of fouling thermal resistance can not be used in design of PHE. It will lead to excess increase of surface area, less enhanced heat transfer and excessive grows of fouling. If data on reference values h_0 and R_{f0}^+ are absent we can recommend first to perform design of PHE without fouling to achieve highest heat transfer enhancement. Then to estimate fouling thermal resistance using TEMA values for shell and tube heat exchangers (e.g, see Hesselgreaves, 2001) at $h_0 = 5000 \text{ W/sq.m.K}$ and $D_{e0} = 0.02 \text{ m}$. After that either to correct design, or to establish cleaning schedule. The excessive heat transfer area will lead to increase of fouling tendencies and thus will deteriorate the PHE performance.

4. Conclusions

Fouling by scaling and precipitating solids on heat exchange surface of plate heat exchangers can be predicted using the heat and momentum transfer analogy. For investigated type of fouling its dependence of flow velocity is stronger than that of heat

transfer. But in plate heat exchanger these processes are inversely correlated. The higher film heat transfer coefficient the lower fouling is.

The correct prediction of fouling thermal resistance is very important in PHE thermal design, but adoption of its fixed excessive value can lead to oversized heat transfer area and increased fouling. The equation to account for fouling in PHE design is proposed, which require reference data for heat exchanger working at the same fouling conditions of fluid. Because oversized plate heat exchanger can lead to much increase in fouling, in case of absence of reference data, it is better first to calculate it in clean conditions and then to estimate fouling by proposed correlation and data for shell and tubes heat exchangers. When it is not possible to minimize fouling, the schedule for plate heat exchanger cleaning should be developed.

5. Acknowledgement

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