Accounting for the thermal resistance of cooling water fouling in plate heat exchangers

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HIGHLIGHTS

- Heat transfer enhancement of PHEs has a potential to mitigate fouling.
- The fouling formation in PHE can be described by the "threshold model".
- The model predicts the effects of wall shear stress and surface temperature.
- The model to predict cooling water fouling development over time was proposed.
- Data obtained for smooth channels can be used directly for plate heat exchangers.

ARTICLE INFO

ABSTRACT

Of the many different approaches available to mitigate fouling, the use of enhanced heat transfer surfaces is a principal method. The flow in channels of plate heat exchangers (PHEs) has high levels of turbulence due to the channels intricate geometry. In principle therefore, this type of heat transfer augmentation should lead to fouling mitigation. The effects of process parameters on fouling in PHE channels are studied in this article. The asymptotic behaviour of water fouling is examined specifically and the net rate of fouling accumulation is described as the difference between the fouling deposition rate and the fouling removal rate. An equation is proposed to account for how the fouling resistance varies with time. The conclusion is that for scaling there exist threshold conditions on wall shear stress, wall temperature and salt content, beyond which fouling deposition starts. The fouling deposition rate expression proposed for tubes incorporating heat transfer enhancement as used by Yang and Crittenden is adapted to be used in a new model for PHEs. Comparison with experimental data published in the literature reveals good agreement with the proposed model. The model can then be used to predict cooling water fouling development over time for all the heat exchangers in a chosen industrial circuit.

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1. Introduction

As shown by Fodor et al. [1], when integrating renewables, polygeneration and CHP units with traditional sources of heat in both industrial and public sector applications, there is a requirement to consider minimal temperature differences in heat exchangers of reasonable size. Such conditions can be satisfied by using plate heat exchangers (PHEs). The design and operation of PHEs are well described in the literature, e.g. [2]. It is observed that high heat transfer coefficients and low fouling tendencies are typical characteristics of PHE channels of complex geometry because of high levels of turbulence, effects that are similar in principle to those observed in enhanced tubes, see e.g. [3]. Analysing the different methods available, Panchal and Knudsen [4] emphasised the use of enhanced heat transfer surfaces as one of the important methods to mitigate water fouling. Heat transfer enhancement and mitigation of fouling renders further advantages to the use of PHEs in different applications in the process industries. As emphasised by Wang et al. [5], the reliable prediction of fouling in enhanced heat exchangers is of great importance for heat exchanger network optimization. It is very important for the design of coolers for use in an existing cooling water networks, see [6], and also for the design of welded PHEs used in refinery processes, see [7].

To select a heat exchanger for the cooling of a process fluid, the design engineer must strictly satisfy the thermal and hydraulic

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Nomenclature

b	a proportionality coefficient, 1/(Pa s)
D_{eq}	equivalent diameter, m
Re	Reynolds number
R_f	fouling thermal resistance, m² K/W
t	time, s
u	flow velocity, m/s
β	corrugation inclination angle to the plate's longitudinal axis
γ	aspect ratio of the corrugation doubled height to pitch
δ	fouling thickness, mm
ζ	hydraulic resistance coefficient
λ_f	fouling deposit thermal conductivity, W/(m K)
ρ	flow density, kg/m³
τ_w	wall shear stress, Pa
φ_d	fouling deposition rate
φ_r	fouling removal rate
ψ	share of friction losses

conditions on the process side. The initial temperature of the cooling water as well as its quality also cannot be influenced by the designer. These parameters would be the same for any type of heat exchanger which could be utilized for this particular process situation, whether it be a tubular exchanger, a PHE (for which many different designs are available), or any other type of exchanger. In the general case it is possible to vary the flow rate and hence the outlet temperature of the cooling water, but this parameter is frequently specified strictly due to constraints on the overall cooling system for the particular application. For a PHE, therefore, only the internal parameters of an exchanger's construction can be varied, such as the number, size and corrugation pattern of the plates, as well as the number of passes of the fluid streams. These choices lead to different channel geometries, different flow velocities and different wall temperatures inside the plate heat exchanger. Accordingly, the designer has good scope by varying these parameters to influence the fouling accumulation rate and its control.

Five different types of fouling mechanism have been recognized for tubular exchangers [8] and these are expected to arise in PHEs as well: precipitation, particulate deposition, chemical reaction fouling, corrosion fouling and biological fouling. Corrosion fouling in PHEs can be mitigated by using appropriate corrosion resistant steels. This can be a much more economical thing to do for PHEs than for shell and tube heat exchangers due to the much smaller surface areas and thinner walls (down to 0.3–0.5 mm) needed for PHEs for the same process operating conditions as for shell and tube exchangers. Most models describing the other types of fouling mechanism are based on the presumption that the net fouling accumulation rate is equal to the difference between a fouling deposition rate φ_d and a fouling removal rate φ_r. This presumption is adopted in this article:

$$\frac{d\delta}{dt} = \varphi_d - \varphi_r \tag{1}$$

In this equation, δ is the fouling thickness, in mm, at any instance in time t, in s. When φ_d is equal to φ_r the fouling deposit thickness does not grow. This is possible for one of the two reasons. In the first, the removal mechanism can be sufficiently strong to always overcome the deposition. In this case, adhesion of fouling to the heat transfer surface can only start after certain threshold

conditions, such as those of surface temperature and surface shear stress, have been met, as shown by Yang and Crittenden [9]. In present paper the influence on fouling of surface finish is not analysed, all examined surfaces have finish of metal generally used for heat exchange surface in process industries, without polishing. In the second situation, once fouling has commenced the removal rate could become directly proportional to the thickness of deposits δ, or the deposition rate could be inversely proportional to δ. In this second situation, after a certain time t^*, the deposition thickness stabilizes to a certain asymptotic value δ^* when the deposition and removal rates become equal. Such asymptotic behaviour during crystallisation fouling was observed by Hasan et al. [10], for calcium sulfate under subcooled flow boiling by Peyghambarzadeh et al. [11], as well as in some cases of fouling with media containing natural fibres by Kazi et al. [12].

The above analysis leads to the conclusion that for selection of PHEs for use as coolers and as components of heat exchange networks in the process industries the correct prediction of water fouling is extremely important. To define operational limits and the optimal design of a PHE to minimize fouling it becomes necessary to estimate the effects of process parameters on the fouling accumulation rate in PHE channels of complex geometry which promotes higher levels of turbulence and enhancement of heat transfer. This is the scope of the present article.

2. Asymptotic values of fouling thermal resistance

The asymptotic behaviour of water fouling on heat transfer surfaces has been observed by many researchers, see Panchal and Knudsen [5]. It usually happens when the stream velocity is high enough to ensure a certain level of shear stress τ_w on the wall. It is now assumed that at asymptotic fouling conditions all effects contributing to fouling growth are accounted for by the deposition rate term φ_d^* and all mitigation effects are accounted for by the removal rate term φ_r^*. A further assumption now made is that φ_r^* is proportional to shear stress at the wall raised to a certain power m and to the deposit thickness δ^*. Hence:

$$\varphi_r^* = b \cdot \tau_w^m \cdot \delta^*, \tag{2}$$

here b is a proportionality coefficient [1/(Pa s)]. When the deposition thickness reaches its asymptotic value, its time derivative equals 0 and from Eq. (1) it follows that:

$$\delta^* = \varphi_d^* / (b \cdot \tau_w^m), \tag{3}$$

Knowing the fouling deposit thermal conductivity, λ_f, the asymptotic value of fouling thermal resistance can be expressed as follows:

$$R_f^* = B^* \cdot \tau_w^{-m}, \tag{4}$$

here $B^* = \varphi_d^* / (b^* \cdot \lambda_f^*)$.

Below is given the analysis of data reported in four publications on water fouling in different channels, illustrated in Fig. 1 with designation by Roman numerals.

(I) A comprehensive study of particulate water fouling in PHE channels has been reported by Karabelas et al. [13]. Experiments were conducted for PHE channels formed using commercial plates with corrugation inclination angles of 60° and 30°. Asymptotic behaviour of the fouling resistance with time was observed, as well as a strong influence of flow velocity. In this study, the wall shear stress on a main corrugated field of inter-plate channels has been estimated using Eq. (5):

$R^*, m^2K/W$
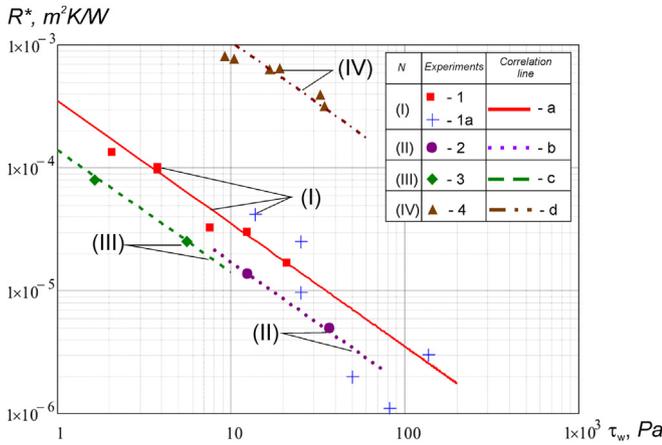
Fig. 1. The influence of wall shear stress on asymptotic value of fouling thermal resistance: I (1 and correlation line a − PHE channel $\beta = 30°$; 1a − PHE channel $\beta = 60°$, data by Karabelas et al., [13]); II (2 and line b − PHE channel $\beta = 60°$) data by Bansal et al. [16], III (3 and line c − annular channel) data by Zhenhua et al. [18]; IV (4 and line d − PHE channel $\beta = 60°$) data by Chernyshov [19].

$$\tau_w = \zeta_s \cdot \psi \cdot \rho \cdot w^2 / 8 \qquad (5)$$

The friction factor for the total pressure losses (due to friction on the wall and form drag) has been calculated using the formula proposed by Arsenyeva et al. [14]. The share of friction losses ψ has been estimated using the relationships in Eq. (6) proposed by Kapustenko et al. [15]:

$$A = 380/[tg(\beta)]^{1.75}; \quad \text{at} \quad Re > A\psi$$
$$= (Re/A)^{-0.15\sin(\beta)}; \quad \text{at} \quad Re \leq A\psi = 1. \qquad (6)$$

here β is the corrugation inclination angle to the plate's longitudinal axis. The Reynolds number Re is calculated for velocities presented in the paper by Karabelas et al. [13] for water properties at a temperature of 40 °C, and for an equivalent diameter $D_e = 0.005$ mm. The corrugation aspect ratio is assumed to be $\gamma = 0.581$. The experimental data on fouling from Karabelas et al. are plotted against wall shear stress in Fig. 1. The influence of wall shear stress on asymptotic thermal fouling resistance can be seen clearly. The data are correlated (solid line in Fig. 1) using Eq. (4) with $m = 1$ and $B^* = 3.5 \cdot 10^{-4}$ K s/m. For wall shear stresses higher than 40 Pa, the thermal resistance becomes very small (less than 10^{-5} m^2 K/W) and the data become unstable due to experimental errors and instability of fouling deposit and removal processes.

(II) The process of calcium sulfate fouling in PHEs was investigated by Bansal et al. [16], and in more detail [17] using a PHE made from commercial plates with chevron corrugations. The corrugation inclination angle was $\beta = 60°$. To investigate the effect of velocity, three experimental runs were made with velocities of 0.183, 0.352 and 0.667 m/s. All other conditions were kept the same for the three runs. In the current paper, wall shear stresses have been calculated using Eqs. (5) and (6), as described above for a water temperature of 61 °C, $D_e = 0.005$ mm, and $\gamma = 0.581$. Two points on Fig. 1 correspond to fouling thermal resistances at the end of runs (after 7000 min) for the velocities of 0.352 and 0.667 m/s. The test run at the lowest velocity finished after only 2000 min and so this run does not permit estimation of R_f^*. The data have been correlated by Eq. (4) using $m = 1$ and $B^* = 2.05 \cdot 10^{-4}$ K s/m.

(III) Fouling by calcium carbonate on the surface of the inner tube in an annular channel was investigated experimentally by

Zhenhua et al. [18]. Two test runs were carried out at velocities of 0.6 and 1.2 m/s with all other conditions being the same. Estimating the wall shear stress as for a smooth annular duct, the data are presented in Fig. 1. They are correlated using Eq. (4) with $m = 1$ and $B^* = 1.45 \cdot 10^{-4}$ K s/m.

All the above-mentioned investigations were performed under laboratory conditions using small PHEs or experimental models of channels.

(IV) On the other hand, an experimental study of precipitation fouling in an industrial PHE for fresh water heating has been reported by Chernyshov [19]. An AlfaLaval M10B PHE of 78 plates was installed in the District Heating (DH) system in the Russian city of Tula. This DH system had a now obsolete "open circuit" of hot water supply, where the hot water is taken from the radiator circuit. It required the fresh water to be heated to the temperatures of the radiator circuit, these being much higher than in "closed radiator circuit" schemes. Because of the high salt content in the fresh water (780 mg/L by data of Chernyshov [19]) as well as the necessity to heat it to temperatures of 60−63 °C and more, the heat exchanger experienced severe fouling. The fouling depends strongly on the temperature of the heated fresh water and hence varied along the plate length. The data on average asymptotic fouling thermal resistance reported by Chernyshov [19] at three different flow velocities (from 0.26 m/s to 0.57 m/s) are presented in Fig. 1. The fouling resistance was calculated from data on fouling thickness and the value of fouling thermal conductivity, namely 1.03 m K/W. In the current paper, the wall shear stresses have been estimated using Eqs. (5) and (6), as described above for a water temperature of 61 °C, $D_e = 0.005$ mm, and $\gamma = 0.581$. The data have been correlated using Eq. (4) with $m = 1$ and $B^* = 115 \cdot 10^{-4}$ K s/m as estimated by Chernyshov [19] for his tests conditions. Slightly lower data for the smallest velocity can be explained by a relatively large fouling thickness (up to 0.8 mm) compared with the plate spacing (2.5 mm). Such significant fouling leads to a decrease in cross-sectional area for the flow of water. In turn, this leads to an increase in water velocity and to a higher surface shear stress than that calculated for the clean channel cross section.

The data presented above were obtained at four different conditions of water fouling. For the same level of wall shear stress the asymptotic values of the fouling thermal resistance are seen to be quite different. The asymptotic resistances depend critically on a number of different factors including concentration, nature and size distribution of suspended particles, soluble salt concentration, the salt itself, the water temperature and the channel wall [20]. Nonetheless, for the same water and the same temperature conditions, the dependence of asymptotic fouling thermal resistance on wall shear stress should be described by the same relation and is inversely proportional to wall shear stress. Counting on the assumptions made in deriving Eq. (4) it can be concluded that the removal term φ_r^* in Eq. (1) is proportional to wall shear stress raised to the power $m = 1$.

On the other hand, determination of the deposition term φ_d^* at the conditions when asymptotic fouling is reached is a very complicated task. Many parameters influence the deposition. Nevertheless, for a large enterprise with a particular water circuit, the quality of the cooling water remains more-or-less the same for all coolers. In this case, when the asymptotic fouling rate in one heat exchanger can be determined, it becomes feasible to calculate the coefficient B^* in Eq. (4) and then to use this value to determine

the cooling water fouling for all the other PHEs in the circuit for the particular enterprise.

In many practical applications, the fouling becomes stabilized after some period of operation, thereby showing asymptotic character. Fig. 2 shows two sides of a plate which has been in operation for about one year at the DH system of Kiev (Ukraine) for tap water heating. In this case scaling is the predominant fouling mechanism. Side (a) was in contact with DH water. Here only particulate fouling takes place. The plate has some deposits which are almost evenly distributed along its length, with some increase in amount towards the entrance of the hot medium. This can be attributed to the higher strength of the deposits at higher temperatures. Side (b) of the plate was in contact with fresh water heated from 5 to 10 °C up to 57 °C for the domestic hot water supply. On this side the plate surface near the entrance of the cold fresh water was clean. The deposits then start to grow significantly towards the hot water exit with the increase in water and wall temperatures. The maximal thickness of the deposit at the corrugation valley was about $\delta^* = 0.05$ mm.

The fouling can be judged more clearly by examining photographs of fouling deposit distributions on small plate areas along the plate as presented in Fig. 3. It can be concluded that for scaling fouling there exist certain threshold conditions on water temperature and salt content, beyond which fouling deposition starts. So, when scaling is possible for the cooling water of an enterprise circuit, the outlet temperature should be kept below a certain level, the level of which depends on the quality of the cooling water for the enterprise.

3. The time dependence of fouling deposits

It is assumed now that the removal term φ_r throughout the duration of the fouling deposit formation is consistent with that proposed for Eq. (2) and with $m = 1$, as follows:

$$\varphi_r = b \cdot \tau_w \cdot \delta \tag{7}$$
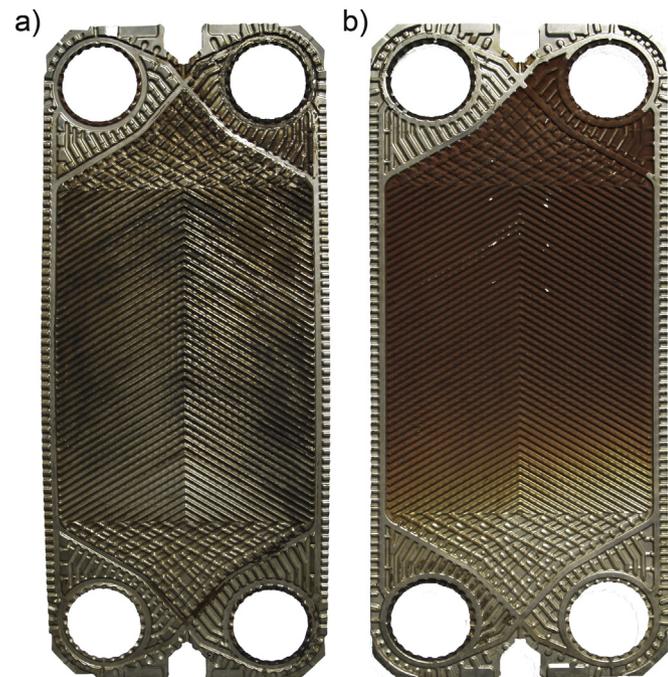
Fig. 2. Two sides of the plate after one year in operation for tap water heating in DH system (fresh water coming from the bottom, heating DH radiator water from the top).

here, b is a proportionality coefficient, [1/(Pa s)]. The development of the fouling thermal resistance R_f with time for the deposit of thermal conductivity λ_f can now be described by Eq. (8), which follows directly from Eq. (1):

$$\frac{dR_f}{dt} = \frac{\varphi_d}{\lambda_f} - b \cdot \tau_w \cdot R_f \tag{8}$$

when φ_d, b, λ_f and τ_w depend neither on the time during deposit formation t nor on the deposit thickness δ, then integration of the linear ordinary differential Eq. (8) is facile. For an approximate solution at time t it is possible to use time-averaged values of these parameters for the time period $0-t$. Then the fouling thermal resistance at time t after integrating Eq. (8) is as follows:

$$R_f(t) = \frac{B}{\tau_w} \cdot \left[1 - \exp\left(1 - \frac{\varphi_d}{B} \cdot \tau_w \cdot t \right) \right] \tag{9}$$

In this equation, $B = \varphi_d/(b \cdot \lambda_f)$. To estimate the coefficient B, it is possible to take its value at asymptotic fouling conditions, i.e. $B = B^*$. As was observed on examining the fouling deposit distribution along the plate, water fouling in PHEs exhibits the threshold behaviour. Over the past decade, the concept of "threshold fouling" has become a focus of research into fouling of the tube side of shell and tube heat exchangers when crude oil is preheated before it enters the atmospheric distillation column on an oil refinery. First introduced by Ebert and Panchal [21], this threshold concept has been developed further by Polley et al. [22] and by Young et al. [23]. Yang and Crittenden [9] have, furthermore, proposed the application of their threshold model for tubes fitted with heat transfer enhancement devices. The deposition term in their model was expressed as follows:

$$\frac{\varphi_d}{\lambda_f} = \frac{A_m \cdot C_f \cdot u \cdot T_s^{2/3} \cdot \rho^{2/3} \cdot \mu^{-4/3}}{1 + B_m \cdot u^3 \cdot C_f^2 \cdot \rho^{-1/3} \cdot \mu^{-1/3} \cdot T_s^{2/3} \cdot \exp(E/(R \cdot T_s))} \tag{10}$$

In this equation T_s is the surface temperature, K; ρ is the fluid density, kg/m³; μ is the fluid dynamic viscosity, Pa s, R is the universal gas constant equal to 8.314 J/(mol K), C_f is the fanning friction factor, and u is the average flow velocity, m/s. In the crude oil study reported elsewhere [9] the parameter values were found to be $E = 52{,}100$ J/mol, $A_m = 7.93 \cdot 10^{-10}$ kg²/³ K¹/³ m⁵/³ (kW)⁻¹ s⁻¹/³ h⁻¹ and $B_m = 1.8 \cdot 10^{-5}$ m¹³/³ kg²/³ s⁸/³ K⁻²/³. Eq. (10) was developed for bare tubes. For intensified heat transfer situations, the velocity u is defined as the equivalent velocity. This is the velocity in a bare tube that gives the same wall shear stress as in a tube of same internal diameter but with heat transfer enhancement and operating probably at a different average fluid velocity [9]. It is now assumed here that the above method is justified also for PHE channels in the current paper such that the equivalent velocity can be determined for a bare tube of the same equivalent diameter D_e. This would then allow Eq. (10) to be used. For fully developed turbulent flow in a straight bare tube with internal diameter D_e the Fanning friction factor can be calculated from the Blasius equation:

$$C_f = 0.0791 \cdot (u \cdot \rho \cdot D_e/\mu)^{-0.25} \tag{11}$$

The wall shear stress in a straight bare tube is related to the friction factor as follows:

$$\tau_w = C_f \cdot \rho \cdot u^2/2 \tag{12}$$

For a known shear stress in a channel with enhanced heat transfer, the equivalent velocity in a bare tube can be calculated from Eq. (12), substituting C_f from Eq. (11):
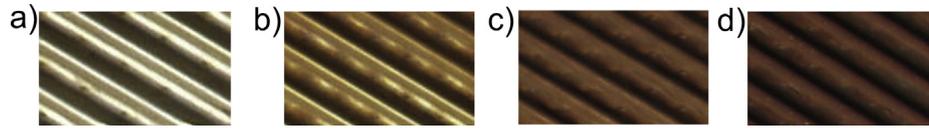
Fig. 3. Photographs of four parts of plate surface a − heated water inlet $\delta^* \approx 0$; b − heated water 2/4 of plate from inlet $\delta^* \approx 0.01$ mm, c− heated water 3/4 of plate from inlet $\delta^* \approx 0.03$ mm, d − heated water near channel exit $\delta^* \approx 0.05$ mm.

$$u = \left(\frac{\tau_w \cdot D_e^{0.25} \cdot 2}{\mu^{0.25} \cdot \rho^{0.75} \cdot 0.0791} \right)^{1/(2-0.25)} \tag{13}$$

The product of C_f and u from Eq. (12) is as follows:

$$P_{cu} = C_f \cdot u = \frac{2 \cdot \tau_w^{1-(1/1.75)}}{\rho} \left(\frac{D_e^{0.25} \cdot 2}{\mu^{0.25} \cdot \rho^{0.75} \cdot 0.0791} \right)^{-\frac{1}{1.75}} \tag{14}$$

Eliminating C_f and u from Eq. (10), using Eqs. (12) and (14), an expression for the deposition term through wall shear stress is now obtained:

$$\varphi_d = \frac{A_m^* \cdot P_{cu} \cdot T_s^{2/3} \cdot \rho^{2/3} \cdot \mu^{-4/3}}{1 + B_m \cdot P_{cu} \cdot 2 \cdot \tau_w \cdot \rho^{-4/3} \cdot \mu^{-1/3} \cdot T_s^{2/3} \cdot \exp(E/(R \cdot T_s))} \tag{15}$$

In Eq. (15), $A_m^* = A_m \cdot \lambda_f$. Using Eq. (15) and the value of B^* which was presented earlier by analysis of the asymptotic fouling thermal resistance data, it is now possible to calculate the fouling thermal resistance at any time t by using Eq. (9). As the chemical and physical properties of crude oil and water are considerably different, the empirical parameters A_m^*, B_m and E must now be re-evaluated to fit the experimental data for water fouling.

To fit the experimental data for precipitation of calcium sulfate in PHE channels as presented by Bansal et al. [16], only one parameter A_m^* has been adjusted for the work presented in this paper. For the value $A_m^* = 6.5 \cdot 10^{-12}$ kg$^{2/3}$ K$^{-2/3}$ m$^{2/3}$ s$^{-1/3}$ h^{-1} the computed thermal resistances of fouling at different flow velocities are presented as curves in Fig. 4 and the predictions are compared with the experimental data taken from a graph in the paper. There is, additionally, later data for calcium sulfate fouling in a PHE presented for one flow velocity by Bansal et al. [17]. Taken from a graph in Bansal et al. [17], the experimental data are compared with predicted values in Fig. 5. Considering the
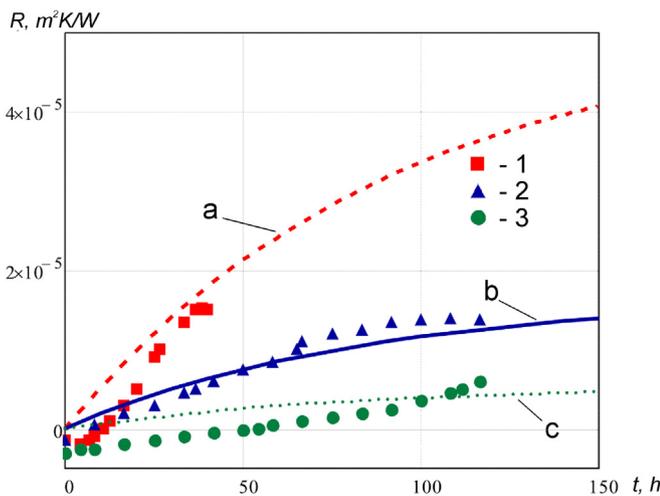
experimental scatter of the data, the agreement between fouling resistances calculated from the model developed in the present paper (solid line in Fig. 5) is rather good. The lower (and negative) experimental values during the initial period can be explained in the same manner as by the authors of the original experimental paper. In essence, the initial delay period may be part of the physico-chemical process for the formation of the deposit on the heat transfer surface. Negative fouling resistance values may be obtained since roughening of a clean surface may actually enhance the film heat transfer coefficient when the surface is almost clean.

Calcium carbonate scaling in an annular smooth channel with an outer diameter of 22 mm and an inner diameter of 16 mm has been investigated by Zhenhua et al. [18]. Their data for two different velocities have been fitted by the new model with $A_m^* = 1.5 \cdot 10^{-10}$ kg$^{2/3}$ K$^{-2/3}$ m$^{2/3}$ s$^{-1/3}$ h^{-1} (see Fig. 6). The other two model parameters, B_m and E, have not been changed. Particulate fouling in PHE channels using water with suspended calcium carbonate particles has been investigated by Karabelas et al. [13]. Fig. 7 shows the comparison of these data with the model predictions (solid lines calculated for $T_s = 37$ °C). All parameters in Eq. (15) were set to be the same as for the calculations for Fig. 6. It is possible to conclude that the values of parameters in Eq. (15) for similar salt and impurity contents have close numerical values. Eq. (15), besides the wall shear stress and the physical properties of the fluid, accounts for the effect of surface temperature T_s. It can be postulated that the coefficient B depends of the surface temperature in much the same way. For two surface temperatures T_{s1} and T_{s2}, when all other conditions the same, it is now assumed that:

$$B(T_{s1})/B(T_{s2}) = \varphi_d(T_{s1})/\varphi_d(T_{s2}) \tag{16}$$

The data for calcium carbonate scaling at three different wall temperatures, when both the flow velocity and the salt concentration remain the same, are presented by Zhenhua et al. [18]. These data are shown in Fig. 8. The lines on the graph are calculated from Eq. (9) using Eqs. (15) and (16). The value of coefficient B is taken

Fig. 4. Fouling thermal resistance in a PHE channel with $\beta = 60°$ under calcium sulfate precipitation: for data of Bansal et al. [16]: 1 and line a − $u = 0.183$ m/s; 2 and line b − $u = 0.352$ m/s; 3 and line c − $u = 0.667$ m/s.
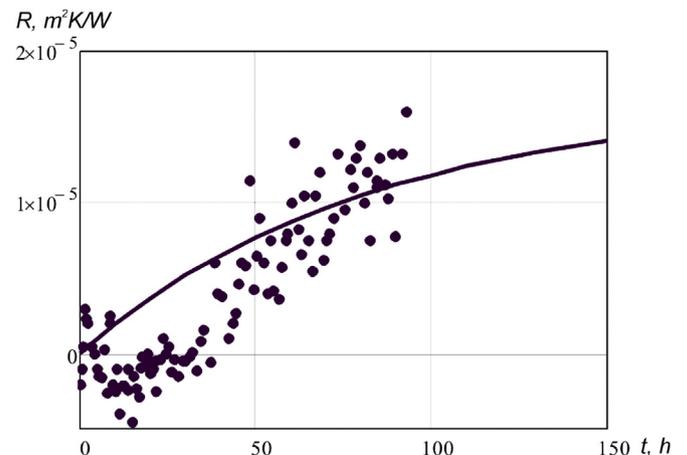
Fig. 5. Fouling thermal resistance in a PHE channel with $\beta = 60°$ under calcium sulfate precipitation: for data of Bansal et al. [17] at $u = 0.352$ m/s.
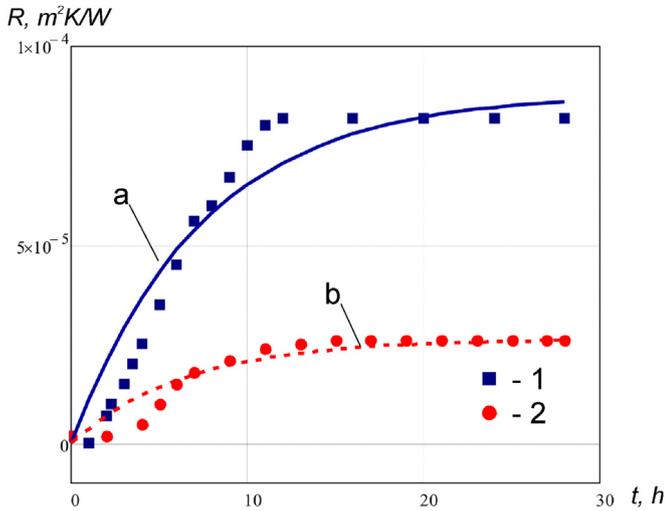
Fig. 6. Fouling thermal resistance at calcium carbonate scaling in annular channel after Zhenhua et al. [18]: 1 and line a − u = 0.6 m/s; 2 and line b − u = 1.2 m/s.

Fig. 8. Fouling thermal resistance at calcium carbonate scaling in annular channel after Zhenhua et al. [18]: u = 1.2 m/s: 1 and line a − T_s = 64 °C; 2 and line b − T_s = 51 °C; 3 and line c − T_s = 44 °C.

from the estimation for data as presented earlier in this paper, for which T_s = 51 °C. The prediction of the asymptotic fouling thermal resistance is fairly good. The description by the model of the time dependence of fouling is also reasonable, with some over-estimation of the fouling thermal resistance.

4. Discussion of the results

From the predictions and comparisons with data presented in the literature, it can be concluded that the new mathematical model developed in this paper is capable of predicting the fouling thermal resistance for precipitation and particulate fouling at different flow velocities and surface temperatures. The model can be used for PHEs with enhanced heat transfer and also for straight channels without heat transfer intensification. The model can be used to predict the effects of wall shear stress and surface temperature, but the influence of salt concentrations and solid particle contents and sizes cannot be accounted for *a priori*.

Large industrial enterprises usually have quite large numbers of heat exchangers, many of which use water from a centralized cooling water circuit. The salt and solid particle contents in such

Fig. 7. Particulate calcium carbonate fouling thermal resistance at PHE channel with β = 30° after Karabelas et al. [13]: 1 and line a − u = 0.5 m/s; 2 and line b − u = 1.0 m/s; 3 and line c − u = 1.35 m/s.

water remain in principle the same for all heat exchangers. Therefore, by monitoring the development of water side fouling in one heat exchanger (in a PHE or inside tubes of a tubular heat exchanger, for example) it becomes possible to estimate the model parameters B and A_m proposed here in the new mathematical model. Subsequently, cooling water fouling in all the PHEs in a particular enterprise can be accounted for. The threshold values of wall shear stress and surface temperature can be calculated. After this has been done, it becomes important in the design and selection of PHEs for the enterprise that the wall shear stress should be kept higher than the threshold value, or at least as high as possible if to reach the threshold is not practicable. The wall surface temperature should be kept below its threshold value, or at least as low as is practicable. When the conditions to completely prevent fouling cannot be reached, the asymptotic fouling thermal resistance should be calculated by the model and incorporated into the design of the PHE.

Even when asymptotic fouling conditions are reached, the value of fouling thermal resistance can change gradually due to fluctuations in water quality and long term changes in deposit properties. It requires monitoring of PHE performance to make cleaning of the heat transfer surface. By our experience on some process positions PHEs did not require cleaning during 20 years and even more. But for the industry the qualifying time period for asymptotic fouling could be stated as 1 year, counting for possibility of annual maintenance.

5. Conclusions

- The concept of equivalent velocity introduced for tubes with inserts by Yang and Crittenden [9] has been further developed and Eq. (15) establishing the link between deposition term and wall shear stress is proposed. By comparison with experimental data available in literature it is shown that, in this way, fouling models developed for bare round tubes can be extended for use with channels of complex geometries, including the channels of PHEs.
- The concept of "threshold fouling", first introduced by Ebert and Panchal [21] for fouling of the tube side of shell and tube heat exchangers when crude oil is preheated before it enters the atmospheric distillation column on an oil refinery, can be applied for cooling water fouling in PHE channels. The wall

shear stress plays a crucial role not only in the fouling process itself but also in the determination of the threshold conditions below which fouling does not take place.

- For correct predictions of fouling thermal resistance using the fouling model developed in this paper it is necessary to determine experimentally one model parameter for a cooling water of a given quality. For a particular cooling water circuit of a big industrial enterprise this parameter can be determined by monitoring fouling data in just one heat exchanger. After this the model can be used for designing PHEs for retrofitting or for new installations on this enterprise. The threshold values of wall shear stress and surface temperature can be calculated. After this has been done, it becomes important in the design and selection of PHEs for the enterprise that the wall shear stress should be kept higher than the threshold value, or at least as high as possible should reaching the threshold not be practicable.
- The model, so far, cannot account for the effects of salt content and solid particle size and concentrations on the fouling thermal resistance and how it develops over time. To estimate the fouling thermal resistance as a function of water purity and chemical content, more experimental data are needed.

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