Energy 174 (2019) 1049–1059

Contents lists available at ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

Accounting for local thermal and hydraulic parameters of water fouling development in plate heat exchanger

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A R T I C L E I N F O

Article history:
Received 10 January 2019
Received in revised form
16 February 2019
Accepted 6 March 2019
Available online 8 March 2019

A B S T R A C T

The complex phenomenon of fouling when heating water solutions can significantly hamper the performance of heat exchangers. The fouling process is considered in an application for a plate heat exchanger (PHE) with enhanced heat transfer that proved to have much lower fouling tendencies than conventional shell and tube heat exchangers. To eliminate the drawbacks of the dimensional fouling model forms the dimensionless its form is developed. It is based on the equation for transport and chemical reaction fouling mechanism initially proposed for other types of fouling media. Thermo-hydraulic mathematical model of PHE under fouling conditions accounting for the distribution of local process parameters along heat transfer surface is presented. It enables to predict not only thermal performance of PHE, but also pressure losses. The mathematical model consists of the system of differential equations with the nonlinear right-hand side. Its solution is implemented with software for the personal computer. The model application is demonstrated with two practical examples. It confirms models' validity and its acceptable accuracy for practical calculations of PHE in industry and also the possibility of proposed dimensionless model application for different fouling substances with the similar types of fouling mechanism.

1. Introduction

The development of modern society is characterised by a permanent increase in demand for energy to satisfy the needs of a growing population and increasing standards of comfort for individuals. Even the widening use of renewables cannot completely curb the detrimental effect of energy generation on environment and depletion of fossil fuel resources, the combustion of which still is and will be in foreseeable future the main source of energy for the planet as a whole. To curb such detrimental effects the increase of energy usage efficiency can help, which require the enhancement in heat recuperation that is possible with the use of compact heat exchangers in which are implemented the principles of heat transfer intensification [1]. Such heat exchangers are usually characterised by much high overall heat transfer coefficients than conventional shell and tube heat exchangers. In these conditions, any additional thermal resistance to heat transfer, such as thermal resistance of fouling on the heat transfer surface, can cause relatively much high adverse effect than the same value in shell and tube. Beside deterioration of heat transfer, the growing fouling layer is leading to a reduction of channels cross-section and increase of wall roughness that stipulate the rise of pressure losses in heat exchanging stream that can follow by final clogging of the channel with a significant increase of energy required for stream pumping. At the same thickness of the fouling layer in channels of smaller hydraulic diameter, this effect can become even more harmful. It emphasizes the significance of accurate accounting for fouling in compact heat exchangers.

Nowadays one of the widely used in industry types of compact heat exchangers is plate heat exchanger (PHE). The construction and operation principles of PHE are well described in the literature e.g. Ref. [1]. This type of heat exchanger is considerably smaller in size, uses much less material for production, has a smaller footprint on the environment than conventional shell and tube units. The

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https://doi.org/10.1016/j.energy.2019.03.026
0360-5442/

plate-and-frame PHE advantages are also low cost, flexibility in changing the heat transfer surface area, accessibility, and what is important for efficient energy usage, a close temperature approach down to 1 K. The main feature of PHE construction is heat transfer surface formed by a stack of corrugated plates stamped from thin metal sheets. The channels for heat media movement between the plates of plate-and-frame PHE are sealed by elastic gaskets. The multiple contact points between plates with inclined corrugations create channels of complex geometry and robust construction capable to withstand high difference in pressure of hot and cold streams in PHE. The intricate form of the channels is promoting high levels of turbulence and considerable heat transfer intensification. A much higher level of wall shear stress than in shell and tube heat exchangers at the same overall thermal and hydraulic process conditions stipulate fouling mitigation, as is discussed in the paper [2]. It was confirmed by laboratory tests results for calcium carbonate water fouling on the model of PHE plate surface and tube with wire insert for enhancement of heat transfer. It permits to assume for fouling prediction in PHEs the same approaches and fouling mechanisms as for shell and tube heat exchangers with proper estimation of wall shear stress.

The adverse effects of fouling on the operation of individual heat exchangers, heat recuperation systems and process industries as a whole are well recognized. In industrial countries fouling phenomenon is responsible for the reduction of GDP (gross domestic product) on about 0.25%, by data of Malayery et al. [3]. By the rough estimation fouling on heat transfer surfaces increases on up to 2.5% the CO_2 generation caused by industry. It has stipulated the researches on different types of fouling, among which are sedimentation of particulate solids, crystallisation, biological, chemical reaction, corrosion and freezing forms of fouling layer deposition [4]. The type of fouling mechanism is dependent on the nature of flow and fouling precursor in it and the way of fouling material creation. However, the development in time of fouling layer for all that types can be divided into four stages (some of which can be absent or negligible):

(1) Initiation period (or terms induction and delay period are used).
(2) Fouling grows period, that in many cases can end with stabilization of fouling layer thickness.
(3) Stabilization period, when fouling behaviour is the asymptotic and stable thickness of the fouling layer is reached.
(4) Fouling ageing period for deposited materials which properties are changing with time.

All these periods are characterised by stochastic nature in which process parameters can fluctuate around certain average values due to instabilities of fouling substances deposition and removal. For different types of fouling and periods of its development a number of fouling models were proposed, which describe the effect on fouling rate of influencing its parameters such as the concentration of fouling precursor in fluid flow, physical properties, temperature, shear stress, wall surface roughness etc. The brief analysis of these models can be found in Ref. [1]. The fouling models proposed by different researchers are corresponding to different stages of fouling development, forms and conditions of fouling studied by model authors. Those models are mostly empirical and have a limited range of application. To develop the comprehensive fouling model for fouling from heated water solutions in PHEs it is important to analyse developments achieved for different fouling media with the similar type of fouling mechanism on heat transfer surfaces available for all types of heat exchangers and their applicability to process under consideration.

The understanding of the fouling starting stage mechanism is important for the description of the whole process. The experimental data obtained by a number of researchers studying the induction period of fouling and its time length have shown the influence of a number of factors. It included process parameters, properties and composition of the mainstream, nature of fouling precursor, the nature and conditions of the heat transfer wall material and its roughness. In this initial period, the thermal resistance of fouling is very low or even show "negative" trend explained by heat transfer enhancement with small initial patches of deposit acting as turbulence promoters near the surface of the channel wall. It looks promising to extend this period and such attempts were made by different researchers using some coatings or treatments of the surface, as e.g. described in the paper [5]. For calculation of the time span for initiation period, the proposed fouling models are mainly empirical with the range of application limited to specific conditions studied, as e.g. model presented in paper [6] for boiling of calcium sulphate solution. In a paper by Yang et al. [7] is proposed induction period model using the concept of particles sticking probability for fouling substances describing the influence of temperature and stream velocity on the length of the induction period. The model was experimentally checked for fouling of crude oil, calcium sulphate and whey protein. As it was noted, the study of the initial period of fouling formation can explain the nature of "threshold fouling" concept introduced by Ebert and Panchal [8] for fouling of crude oil inside tubes. By this concept at some conditions the fouling not starting at all. However, in a number of fouling studies, no induction period was observed. The existence of the induction period is rare for particulate fouling, as was observed in the paper [9]. Such conclusion was confirmed for flue gas fouling at heat exchanger with finned tubes in the paper [10] and for the flow of air in finned plate heat exchanger [11]. There was not found induction period in experiments on water fouling of some researchers, e.g. Teng et al. [12] not observed it in stainless steel double pipe heat exchanger for water calcium carbonate fouling. Bai et al. [13] have experimentally investigated river water fouling in pipes of the surface water heat pump. For new tubes, the induction period from 17 to 86 h was observed. However, after tubes were for the first time cleaned by different methods (with a brush or by increased water velocity) no induction periods were reported. For surfaces with enhanced heat transfer, like helically ribbed copper tubes, no induction period was reported in the analysis of experimental results in the paper [14]. The same observation was made in experiments with water fouling in dimpled enhanced tubes [15]. In experiments on double pipes, a heat exchanger with coiled wire inserts Hasan et al. [16] have not found any induction period for crystallisation fouling. The results of laboratory experiments with calcium carbonate crystallisation fouling on enhanced surfaces were reported in the paper [2]. Induction period was reported neither for modelled tubes with wire inserts nor for the simulated surface of the plate heat exchanger. As plate heat exchanger (PHE) is concerned, in results of field tests [17] of PHEs installed in district heating system induction period was also not observed. As is discussed in the paper [18], the accounting for initiation period can be made by introducing a time lag in fouling development model, but not accounting for it in the calculation of heat exchangers operating in the industry is leading to some increase in estimated values of fouling deposits quantities that however keeps the design on a safe side. In a study presented in the current paper, this approach is also kept.

In course of induction period model reported in the paper [7] the particles of depositing material are sticking to the surface of the heat transfer wall and gradually cover it. In patches of the covered surface the already stuck foulant is attracting more foulant increasing the deposited material quantity with some grows rate. But at the same time, it is subjected to removal forces with the total removal rate proportional to the share of covered surface. When all surface area becomes completely covered by foulant the stage of fouling layer growth begins. To calculate the fouling growth rate in this period a significant number of models are proposed in the literature, which corresponds to the specific nature of the fouling mechanism. The most widely applied by different researchers is an approach proposed by Kern and Seaton [19] that is using the material balance of the fouling layer with thickness δ_f deposited on the surface. The fouling grows rate with time is described as the difference between deposition and removal rates, which can change with the growth in a time of deposited layer thickness. The process has stochastic character and when some amount of fouling is deposited on the surface, at the same time some already deposited particles are removed back to the main flow by hydrodynamic forces. When the time-averaged rates of both deposition and removal processes become equal to each other the amount of deposited material stabilises and fouling layer thickness is coming close to some constant value. It corresponds to the asymptotic fouling mechanism.

Panchal and Knudsen [4] came to the conclusion that for particulate and crystallisation types of fouling such character of its development in time was most frequently observed in experimental researches. Such time dependence during crystallisation water fouling was reported in the paper [20], for calcium sulphate fouling in flow boiling in Ref. [21], in some experiments of fouling from media with natural fibres [22] and in a number of other publications. For water fouling asymptotic behaviour is also common in experiments with enhanced heat transfer surfaces, as e.g. for double pipe heat exchanger with cold wire inserts [16], in helical-rib tubes [14], for models of tube with wire insert and of PHE surface [2], for different enhanced tubes surveyed by literature publications [23]. The experiments with particulate $CaCO_3$ fouling in plate-and-frame PHE [24] and in PHE [25] with TiO_2 and $CaCO_3$ particulate fouling also revealed an asymptotic character of fouling behaviour. The method of using some fixed value of fouling thermal resistance for specific heat media recommended for tubular heat exchangers design by TEMA [26] can be grounded for asymptotic fouling mechanism. For heat exchangers with enhanced heat transfer in some papers are proposed simplified approaches for prediction of water fouling asymptotic thermal resistance, like for tubes with heat transfer intensification [27] or PHEs [28]. Some approaches based on neural network predictions for PHE asymptotic fouling [29] were also proposed. However simplified models are limited in accuracy and neural network require extensive data for its training and not giving incite view into the details of process mechanism. For accurate modelling of heat exchangers operation in conditions of fouling the reliable and accurate enough equations for estimations of the deposition and removal rates are required.

A large number of different equations for fouling deposition rate at its development period calculation have been proposed in the literature for specific fouling mechanisms. For water fouling some of the researches were concentrated on the estimation of the effect of water solution concentration, pH and dissolved substance saturation properties, like e.g. in the paper [30] for $CaCO_3$ fouling in stainless steel PHE. For water fouling as well as for wider class of fouling mechanisms, the reaction and transport models are frequently used. Epstein [31] has formulated the general form of such a model for the process of styrene polymerization from a kerosene solution. The important feature of this model is accounting for the influence of wall shear stress on fouling deposition rate. The application of this model for PHE was analysed in the paper [32]. For crystallisation type of fouling in the paper [33] was proposed to use in this model the difference of fluid bulk concentration and concentration of saturation at surface conditions, such approach was used in the paper [16]. The equation of such type can be also helpful for analysis of particulate fouling. According to paper [33], Arrhenius-type Equation can be applied for calculation of the attachment rate constant with known surface temperature T_s as an exponent, that in form is similar to reaction term of the model proposed in paper [31] and enable assumption of using such model also for particulate fouling. That conclusion was also made in the literature survey of publications on particulate fouling [34]. A number of fouling models were analysed in the paper [35] to study their suitability for accounting "threshold fouling" phenomenon, by which at some temperature and shear stress levels fouling is not starting at all. It was shown that all those models can be deduced from Equation proposed in paper [31] with some assumptions on terms in the denominator and relative influence of these terms at specific fouling mechanism. As it is discussed in paper [36], the application of such model type has proved its validity in researches of different fouling substances, like chemical reaction fouling [37], fouling of the protein solutions [38], colloidal particle fouling [39] and precipitation fouling from calcium sulphate solutions [40]. It can be also concluded for a number of fouling models reviewed in the paper [41]. The modelling of water fouling in PHE including crystallisation and particulate fouling mechanism based on expression derived in a similar way was validated in the paper [28] for averaged process parameters and in the paper [32] for accounting the distribution along heat transfer surface of local process characteristics. It confirms the validity of the considered approach in different fouling conditions. However, the model of paper [31] relates dimensional parameters of the fouling process and includes empirical constants, which at some model modifications have rather complicated units. Some researchers have made attempts to introduce thermodynamic Biot number *Bi* but like e. g in the paper [41] for milk fouling in similar Equation with omitted diffusivity term their dimensional constants were still employed. It reduces the model generalization power and not permits to use assumptions of dimensional analyses and theory of similarity that have proved useful in fluid dynamics and heat-and-mass transfer theory. In paper [32] the validity of the proposed fouling model based on the approach of paper [31] was demonstrated by thermal simulation of water fouling in PHE by local process parameters. It was shown the advantages in the accuracy of such calculations. The importance of accounting for local process parameters in simulation of fouling in heat exchangers was demonstrated also in paper [42] for water fouling in shell-and-tube heat exchanger, in paper [43] for individual shell and tube heat exchanger and heat exchanger network in crude oil preheat train [44] and for PHE in dairy application [45].

In the majority of the papers published on fouling modelling in heat exchangers, the thermal performance and thermal resistance of fouling is the main subject. Much less number of researches is concerned with the prediction of pressure drop due to fouling which, however, in some applications can be extensive and cause an increase of electricity consumption by pumping equipment, finally leading to shutdowns for heat exchangers cleaning. However, for water fouling the publications aimed for accurate enough prediction of fouling influence on pressure drop in heat exchangers are not found by authors. The ways of accounting deposited fouling layer in hydraulic design of shell and tube heat exchangers for crude oil preheat train were discussed by Yeap et al. [35]. It was concluded that the model should take into account the increase of flow velocity with the decrease of free flow cross section area by fouling deposits and also roughness on the deposited layer surface. Respectively two kinds of hydraulic models linking thermal and hydraulic performance of heat exchanger are analysed [35] using as dimensionless thermal resistance of fouling Biot number *Bi*. The analysis is made on an assumption of uniform fouling layer distribution along the heat transfer surface, but it was emphasized that in practice the pressure drop depends on the distribution of deposited fouling layer within the heat exchanger. Accounting for this effect was made in thermo-hydraulic modelling of the shell and tube heat exchanger under crude oil fouling by Coletti and Macchietto [43] and of tubular heat exchangers network in crude oil preheat train in the paper [44].

According to the presented state of the arts analysis, the fouling process in different heat media used in heat exchangers can have significant differences, but for the same types of fouling mechanism including chemical reaction, crystallisation and particulate fouling the similar approach based on model presented in paper [31] can be used for prediction of fouling tendencies. However, the generalization of available fouling models for different heat media and geometry of heat transfer surfaces is limited due to their dimensional empirical character. The objective of the current paper is to present the thermo-hydraulic model of the plate heat exchanger with intensified heat transfer operating under fouling conditions of heating water solutions. First is described the derivation of dimensionless fouling model based on approach initially proposed for fouling of crude oil and its products. After that, the mathematical model of PHE based on the use of local process parameters is considered. The model includes the prediction of water pressure drop in PHE under fouling conditions accounting for recommendations developed for the case of crude oil fouling. Finally, the model validity is checked by comparison with available industrial tests data for two cases of heating water solutions in PHEs.

2. Dimensionless fouling model

Dimensional analysis and similarity theory are the powerful tools for generalization of experimental results and providing insight into the investigated phenomenon. The successful applications of these methods in fluid mechanics and heat-and-mass transfer are well proved by the historical experience of developments in these fields of science. It is also successfully used in modern researches at very different areas including analysis of pumps operation [46], natural gas hydrate experimental modelling [47], two-phase convective flows [48], the gas-liquid-solid system [49] and a number of others. To develop a dimensionless form of fouling model its dimensional analogue is first to be considered.

The approach proposed by Kern and Seaton [19] is mathematically presented in form of following Eq. (1):

$$\frac{\partial \delta_f}{\partial \theta} = \varphi_d - \varphi_{rm} \tag{1}$$

The deposition term according to the model proposed in paper [31] for the chemical reaction of the first order can be expressed as:

$$\varphi_d = \lambda_f \cdot \frac{C_b}{\frac{k_{1M}}{k_m} + \frac{k_{2R} \cdot \tau_w \cdot e^{E/R \cdot T_s}}{\mu}} \tag{2}$$

In Eq. (2) k_m is coefficient of mass transfer, m/s; μ is dynamic viscosity of fluid, Pa s; τ_W is shear stress at the channel surface, Pa; E is chemical reaction activation energy, J/mol; R is universal gas constant, J/(mol K); T_s is temperature at the surface, K; C_b is a concentration of fouling substance at the main flow, kg/m^3; λ_f is the fouling layer thermal conductivity, W/(m·K); k_{1M} and k_{2R} are empirical parameters.

According to approach originally proposed by Kern and Seaton [19] the fouling removal rate is directly related to the amount of deposited material (or deposited layer thickness δ_f) and to shear stress on its surface τ_W with some proportionality coefficient B. The strength σ of deposited layer material is making an inverse influence. All these effects can be summarised in the Eq. (3):

$$\varphi_{rm} = \frac{B \cdot \delta_f \cdot \tau_W}{\sigma} \tag{3}$$

The direct proportionality of the right part in Eq. (1) to δ_f on its integration is leading to exponential function inherent to asymptotic fouling mechanism [33]. In some models for crude oil fouling, according to an analysis published in Ref. [35], removal rate depends only from wall shear stress in power less than unity. In such case, the fouling layer thickness is influencing mostly through decreasing of free cross-section area for the flow leading to an increase of flow velocity and shear stress on the surface. Such models are corresponding to falling fouling rate mechanism at which asymptotic fouling layer thickness usually is not achieved due to the excessive rise of pressure drop that would require cleaning of heat exchanger long before stabilization of fouling layer thickness. In such conditions, stable fouling layer thickness usually cannot be reached, as it would require excessive pressure drop in the heat exchanger and cleaning for this reason. By comparison with experimental data for water fouling in PHE channels in the paper [28] it was shown the direct proportionality of fouling removal rate to wall shear stress in first power and for removal rate proposed the following Equation:

$$\varphi_{rm} = B^* \cdot \delta_f \cdot \tau_W \tag{4}$$

The dimensional empirical parameter B^* is accounting for the strength of fouling deposit material in the assumption that it is not changing during the fouling layer grows. The simulation of water fouling in PHE with the use of Eq. (1), deposition term by Equation derived from Eq. (2) and removal term by Eq. (4) was reported in the paper [28]. By comparison with experimental data available in the literature, it was confirmed the validity of the model for averaged by the whole heat exchanger process parameters.

The state of the art analysis shows that fouling models based on different modifications of Eq-s (1), (2) and (4) are giving good results in correlating experimental data of quite a number of researches concerning fouling in heat exchangers. Here it is represented in dimensionless form maintaining its structure and main variables with following assumptions.

(I) Diffusional Sherwood number in the channel with a heated stream can be determined by the analogy of heat and mass transfer processes in a form:

$$Sh_2 = \frac{k_m \cdot d_e}{D} = Nu_2 \cdot \left(\frac{Sc_2}{\Pr_2}\right)^{1/3} \tag{5}$$

here Nu_2 is Nusselt number in heated stream; $\Pr_2 = c_{p2}\ \mu_2/\lambda_2$ is Prandtl number; $Sc_2 = D \cdot \rho_2/\mu_2$ is diffusional Schmidt number; ρ_2 is liquid density, kg/m^3; λ_2 is thermal conductivity of liquid, W/(m·K); c_{p2} is specific heat capacity of liquid, J/(kg·K); d_e is channel equivalent diameter, m; D is coefficient of diffusion, m/s.

(ii) The dependence of the diffusion coefficient from dynamic viscosity for certain substance can be expressed by a modified Stocks-Einstein equation [50] as follows:

$$D = \frac{\chi \cdot T_s \cdot k_B}{\mu_2 \cdot r_m} \tag{6}$$

here $k_B = 1.38{,}048 \cdot 10^{-23}$ J/°C is Boltzmann constant; T_s is the surface temperature, K; μ_2 is dynamic viscosity, Pa·s; r_m is Van der Vaals molecule radius, m, which is used as a scale of molecule radius; χ is a coefficient influenced by on the nature of solution and accounting for discrepancies with Stocks-Einstein equation for a certain solution with specific content. This form of the link between D, μ and T is confirmed experimentally in the paper [51] for various solutions as a result of a study establishing the influence of proportionality scale on solute properties. This result is also confirmed in more recent papers, like e.g. Ref. [52]. For the specific media, the coefficient χ can be supposed as not depending on the solution concentration and temperature. The molecule radius for different substances is rather difficult to estimate and it is taken the radius for water molecule $r_m = 1.36 \cdot 10^{-10}$ m from the paper [53] being introduced as a scale factor. The possible difference in r_m is accounted in coefficient χ value.

By substituting D from Eq. (6) into Eq. (5) and expressing k_m from that Eq. (5) the Eq. (2) after rearranging its components can be rewritten in dimensionless form as follows:

$$\Phi_d = \frac{\varphi_d \cdot d_e \cdot \rho_2}{\mu_2} = \left\{ c_D \cdot K_D^{\frac{2/3 \cdot Pr_2^{1/3}}{Nu_2}} + c_R \cdot K_R \cdot \exp\left[\frac{E}{(R \cdot T_s)}\right] \right\}^{-1} \tag{7}$$

where:

$$K_D = \frac{\mu_2^2 \cdot r_m}{(T_s \cdot \rho_2 \cdot k_B)}; \tag{8}$$

$$K_R = \frac{\tau_w}{(\rho_2 \cdot d_e \cdot g)}. \tag{9}$$

The first dimensionless complex K_D can be regarded as characterizing the interrelation between impulse and mass transport properties of media. The second K_R expressing the relation between shear and gravity forces for flow near the heat transfer surface. Both dimensionless complexes include variables that can change with temperature and flow conditions.

Other two dimensionless complexes c_D and c_R are depending from constants k_{1M} and k_{2R} in Eq. (2), the concentration of the solution C_b and other properties of the solution and fouling deposit which are not changing for the same media:

$$c_D = \frac{k_{1M}}{\left(\chi^{2/3} \cdot C_b \cdot \lambda_f\right)}; \tag{10}$$

$$c_R = \frac{k_{2R} \cdot g}{C_b \cdot \lambda_f}. \tag{11}$$

For the same water solution c_D and c_R can be taken as constants that are possible to estimate by experimental data.

In the same way, the expression for the dimensionless removal term is derived from Eq. (4) as follows:

$$\Phi_{rm} = \frac{\varphi_{rm} \cdot d_e \cdot \rho_2}{\mu_2} = \frac{B^* \cdot \delta_f \cdot \tau_w \cdot d_e \cdot \rho_2}{\mu_2} = c_{rm} \cdot \mathrm{Re}^{*2} \cdot \mathrm{Pr}_2 \frac{\delta_f}{d_e} \tag{12}$$

Here

$$\mathrm{Re}^* = \frac{w^* \cdot d_e \cdot \rho_2}{\mu_2} \tag{13}$$

It is Reynolds number calculated with the wall shear stress velocity w^* determined according to Ref. [54] as:

$$w^* = \sqrt{\tau_w/\rho_2} \tag{14}$$

In Eq. (12) is introduced the dimensionless parameter of the model:

$$c_{rm} = B^* \cdot \frac{\lambda_2}{c_{p2}} \tag{15}$$

For the certain liquid solution, the variation with temperature of such properties as heat capacity c_{p2} and thermal conductivity λ_2 are small, this parameter c_{rm} can be supposed as constant for certain fouling media and can be determined by the experimental data.

Following the procedure applied for fouling deposition (Eq. (7)) and fouling removal (Eq. (12)) terms, the dimensionless fouling rate can be expressed as follows:

$$\Phi_f = \frac{\partial \delta_f}{\partial \theta} \frac{d_e \cdot \rho_2}{\mu_2} \tag{16}$$

This dimensionless complex by its form is similar to Reynolds number and can be regarded as characterizing the ratio of forces caused by nonlinear effects of fouling grows to viscous forces in fluid flow. As a result, the fouling model of Eq. (1) in dimensionless form can be presented by Eq. (17):

$$\Phi_f = \frac{1}{c_D \cdot K_D^{\frac{2/3 \cdot Pr_2^{1/3}}{Nu_2}} + c_R \cdot K_R \cdot \exp\left(\frac{E}{R \cdot T_s}\right) - c_{rm} \cdot \mathrm{Re}^{*2} \cdot \mathrm{Pr}_2 \frac{\delta_f}{d_e}} \tag{17}$$

The dimensionless parameters in Eq. (17) are c_D, c_R, c_{rm} and E. These parameters can be determined by experiments on fouling for specific media in PHE with calculations on local process parameters as is shown in paper [32] on example of modelling thermal performance of PHE with the use of dimensional form of fouling model.

3. A thermo-hydraulic mathematical model of PHE

According to Eq. (17), the fouling grows rate is determined mainly by such factors as temperature and shear stress at the heat transfer surface. These factors can change significantly along the heat transfer surface following the change of streams temperatures in the heat exchanger and also undergo variation in time with developing of deposited fouling layer. The growth of fouling deposit is reducing the free flow cross-section area and creates a roughness on the surface of deposit that creates modification in flow conditions. To account for all these features in calculations of PHE is possible by creating detailed enough mathematical model that includes equations describing the variation of process parameters along the heat transfer surface and their development in time. Such a mathematical model for PHE in which two liquid streams are exchanging heat is presented here. It is based on assumptions:

(I) The heat transfer process between two streams in one pass PHE with counter-current flow is considered.
(II) The maldistribution of streams is not considered and conditions in channels for one stream are the same.
(III) The losses of heat to ambient can be neglected.
(IV) The model variables are uniformly distributed along the channel width and the changes only along the length of the channels are considered.
(V) The thermal resistance of fouling at the heating stream side is not considered.
(VI) The PHE channel is considered according to Ref. [55] as consisting of the main corrugated field (4 in Fig. 1), distribution zones (2, 5) and flow entrance-exit (1). The major part of the heat transfer process is going according to correlations for the main heat transfer field. The flow entrance and distribution zones can be accounted for by their influence on total pressure drop in PHE as local hydraulic resistances.

vii. The shear stress at the fouling layer surface can be determined by correlations for clean PHE channel with velocity calculated for a decreased cross-section area because of fouling deposition.
viii. At the small period of time averaged heat and material balances are the same as for stable conditions, not accounting for fouling development during this period.

For countercurrent streams movement in two adjacent PHE channels the heat balance in differential form for the cold stream:

$$\frac{\partial T_2}{\partial x} = \frac{q \cdot 2 \cdot F_{pl}}{g_2 \cdot c_{p2} \cdot L_{pl}} \tag{18}$$

For hot stream

$$\frac{\partial T_1}{\partial x} = \frac{q \cdot 2 \cdot F_{pl}}{g_1 \cdot c_{p1} \cdot L_{pl}} \tag{19}$$

$$p1 = \exp(-0.157 \cdot \beta); p2 = \frac{\pi \cdot \beta \cdot \gamma^2}{3}; p3 = \exp\left(-\pi \cdot \frac{\beta}{180} \cdot \frac{1}{\gamma^2}\right); p5 = 1 + \frac{\beta}{10};$$
$$p4 = \left(0.061 + \left(0.69 + tg\left(\beta \cdot \frac{\pi}{180}\right)\right)^{-2.63}\right) \cdot \left(1 + (1-\gamma) \cdot 0.9 \cdot \beta^{0.01}\right) \tag{24}$$

Here T_1 and T_2 are temperatures of hot stream and heated stream, K; g_1 and g_2 are mass flowrates of hot and heated streams in one channel, kg/s; c_{p1} and c_{p2} are specific heat capacities of hot and heated streams, J/(kg·K); F_{pl} is the heat transfer area of one plate, m^2; x is the coordinate along plate length, starting at heated stream inlet to the channel, m; q is the flow of heat through unit of area, W/m^2.

$$q = \left(\frac{1}{h_1} + \frac{1}{h_2} + R_f + \frac{\delta_w}{\lambda_w}\right)^{-1} \cdot (T_1 - T_2) \tag{20}$$

Here h_1 is film heat transfer coefficients for hot stream, W/(m^2·K); h_2 is a film heat transfer coefficient for heated stream, W/(m^2·K); δ_w is the thickness of the plate wall, m; λw is thermal conductivity of plate material, W/(m·K), $R_f = \delta_f/\lambda_f$ is thermal resistance of fouling layer, (m^2·K)/W.

The temperature at the fouling layer surface is calculated as follows:

$$T_s = \frac{q}{h_2} + T_2 \tag{21}$$

The fouling deposition rate, as well as heat transfer intensities in PHE channel, depend on shear stress at the channel wall [56], which value is linked with pressure drop in PHE. But the pressure drops in a whole PHE assembled with commercial plates has different components, some of which are not linked to the friction forces at the main heat transfer area. There are distributing collectors, channel entrance and exit (positions 1 in Fig. 1), flow distribution zones (2, 5 in Fig. 1) with pressure drops not making a significant contribution to considered shear stresses. For calculation of friction factor at the main heat transfer area in the paper [57] is proposed an empirical correlation accounting for a geometrical form of plate corrugations with Reynolds number Re defined with channel equivalent diameter as double plate spacing $d_e = 2b$. This correlation was received based on equation form proposed in paper [60] for correlation of pressure drop in tubes at different flow regimes. Its parameters p1, p2, p3, p4 and p5 are obtained in the paper [57] with statistical analysis of data from four publications on hydraulic performance of criss-cross flow channel models with different corrugations forms and sizes. It was confirmed later by a comparison with data from another three additional publications in the paper [58]. The correlation is as follows

$$\zeta = 8\left\{\left(\frac{12+p2}{Re}\right)^{12} + \left[A + \left(\frac{37,530p1}{Re}\right)^{16}\right]^{-\frac{3}{2}}\right\}^{\frac{1}{12}} \tag{22}$$

where

$$A = \left[p4 \cdot \ln\left(p5 \cdot \left(\left(\frac{7 \cdot p3}{Re}\right)^{0.9} + 0.27 \cdot 10^{-5}\right)^{-1}\right)\right]^{16} \tag{23}$$

The parameters in these Eqs. (22) and (23) are calculated depending on the geometrical form of corrugations:

The parameters characterizing the geometry of plate corrugation are: $\gamma = d_e/S$ is the ratio of equivalent diameter to corrugations pitch; β is the angle of corrugations to the main flow direction, degrees.

The total pressure drop in the PHE channel of complicated geometrical form is caused not only by action of friction forces at the walls but also by form drag due to complicated flow pattern. With a known share of friction losses ψ in total pressure drop the wall shear stress can be calculated as:

$$\tau_w = \zeta \cdot \psi \cdot \frac{\rho \cdot w^2}{8} \tag{25}$$

Here w is the average velocity of flow in PHE channel, m/s.

The share ψ of friction pressure loss in channels of PHE with inclined corrugations can be estimated by the correlation presented in paper [58] and later confirmed by data of CFD modelling [59]:

$$\psi = \left(Re/_{Aa}\right)^{-0.15 \cdot \sin(\beta)} \text{ at } Re > Aa;\ \psi = 1 \text{ at } Re \leq Aa \quad \text{where } Aa = 380\Big/[tg(\beta)]^{1.75} \tag{26}$$

Using this Eq. (26) in the paper [58] was derived Equation for calculation of film heat transfer coefficients based on modified Reynolds analogy that is used here for estimation of heat transfer intensity in both hot and heated streams:

$$Nu = \frac{h \cdot d_e}{\lambda} = 0.065 \cdot Re^{6/7} \cdot \left(\psi \cdot \zeta/_{F_X}\right)^{3/7} \cdot Pr^{0.4} \cdot \left(\mu/_{\mu_w}\right)^{0.14} \tag{27}$$

where μ and μ_w are dynamic viscosities of heat media at main fluid flow temperature and at wall temperature, respectively, Pa·s; λ is the thermal conductivity of heat media, W/(m·K); F_x is area

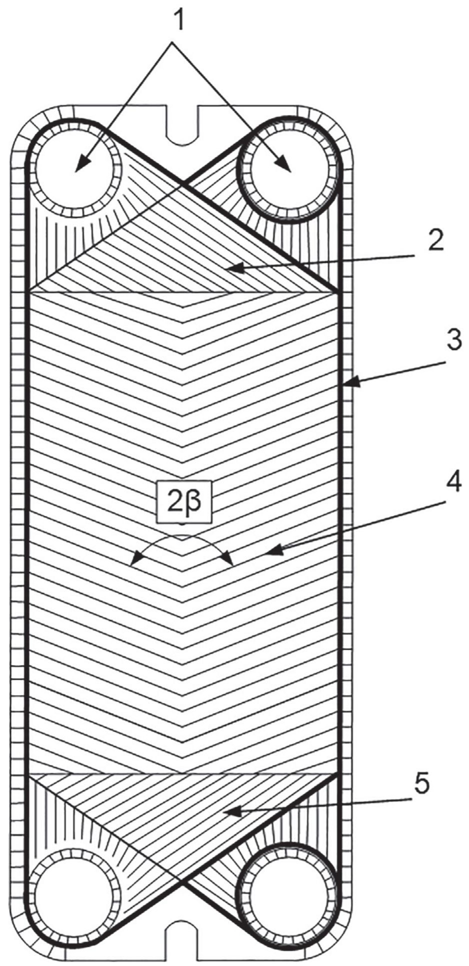

Fig. 1. Sketch of PHE plate area: 1 inlet and outlet of streams; 2, 5 flow distribution zones; 3 elastomeric gasket; 4 the main area of the corrugated field.

enlargement factor due to corrugations.

The free area of flow for the heated stream is being reduced in time with grows of fouling layer thickness δ_f. Here it is accounted by reduction of channel height to $b - 2\cdot \delta_f$ and a corresponding increase of flow velocity:

$$w_2 = \frac{g_2}{\left(f_{ch} - 2\cdot\delta_f\cdot F_{pl}/L_{pl}\right)\cdot\rho_2} \tag{28}$$

For a hot stream with no fouling, the flow velocity is determined by full channel cross-section area f_{ch}, m^2.

The thermophysical properties of liquid streams in heat exchangers can be regarded as not dependent on pressure variation due to its change in heat exchanger channels. In this view, the problem of fouling development can be solved without accounting for variations of local pressure. It is enabling to receive information on fouling deposition and its distribution along the channel length by the solution of thermal part of the heat exchanger model prior for estimation of pressure losses in the heat exchanger.

As it was discussed in the paper [55], the total loss of pressure in PHE can be determined by the summation of following pressure losses: at the main heat transfer field ΔP_{mf}, at distribution zones on inlet ΔP_{DZin} and outlet ΔP_{DZout} of the channel, in ports and collectors ΔP_{pc}. Keeping this order, the total pressure loss of the heated stream can be calculated as follows:

$$\Delta P_2 = \int_0^{L_p} \zeta_2\cdot\frac{\rho_2\cdot w_2^2}{2\cdot de}dx + \zeta_{DZin}\cdot\frac{\rho_2\cdot w_{2DZin}^2}{2} + \zeta_{DZout}\cdot\frac{\rho_2\cdot w_{2DZout}^2}{2} + 1.3\cdot\frac{\rho_2\cdot w_{2p}^2}{2} \tag{29}$$

Here w_{2DZin}, w_{2DZout} and w_{2p} are velocities at inlet zone of the channel, its outlet zone and PHE ports, respectively, m/s.

As is discussed in the paper [57], the correlation (22) with its parameters (23) and (24) was developed based on the structure proposed by Churchill [60] and experimental data for different models of PHE channel corrugated field with various geometries of corrugations. All models considered there were made from metal sheets that could be considered technically smooth, so wall roughness was not accounted for. However, the original correlation for straight tubes presented in paper [60] is accounting for surface roughness by introduction it in parameter A of Eq. (23). It is assumed here that the influence of fouling layer on pressure loss in PHE channel can be accounted in the same way by the introduction in correlation the ratio of fouling layer thickness to channel equivalent diameter and using parameter A in the following form:

$$A^* = \left[p4\cdot\ln\left(p5\cdot\left(\left(\frac{7\cdot p3}{\mathrm{Re}}\right)^{0.9} + 0.27\cdot\frac{\delta_f}{de}\right)^{-1}\right)\right]^{16} \tag{30}$$

For estimation of pressure drop in the channel with fouling the calculation of friction, factor ς is made according to Eq. (22) and its parameters by (30) and (24). For distribution zone at the inlet of the channel, the coefficient of local hydraulic resistance in inlet distribution $\zeta_{DZin} = 38$ is taken according to data of paper [55]. For collecting zone at channel outlet for ζ_{DZout} the value 38 is corrected for deposit roughness for the ratio of friction factors at the end of the main corrugated field with and without the fouling. The velocity w_{2DZout} is also determined accounting for the reduction of cross-section area by deposited fouling.

The Equations (17) – (30) with correlations for calculation of thermo-physical properties of flow media accompanied by PHE geometrical relations are composing the system of differential equations with nonlinear right-hand sides, which is not permitting an analytical solution. The main independent variables of the system are T_1, T_2, δ_f and P. The boundary conditions are taken as temperature T_{2in} and pressure P_{2in} at heated stream inlet and temperature at the inlet of hot stream T_{1in}. The pressure of the hot stream is not considered in this paper. The initial value of fouling thickness is taken as zero $\delta_{f0} = 0$. The solution of the system with these boundary and initial conditions is performed numerically with the finite difference method and implemented on PC using Mathcad. It allows calculation of local and averaged parameters of PHE subjected to fouling and development of these parameters in time, including streams temperatures, fouling thermal resistance and pressure drop. The values of dimensionless coefficients in fouling model Eq. (17) c_D, c_R, c_{rm} and activation energy E can be estimated using the data of monitoring PHE operation with specific media, as is described in the paper [32]. Two examples of model application in practice are presented in the following section.

4. Results of mathematical modelling and discussion

For correct application of the developed mathematical model, it requires the estimation of empirical parameters in Eq. (17) for fouling rate calculation. It can be done by data of monitoring certain PHE under fouling conditions heating the specific media for work with which other PHEs should be optimised or the plate pack of tested PHE modernised to achieve its better performance. Another important area is PHEs for tap hot water heating in District Heating networks where fresh water in a certain region has constant properties not much changing in time. In this section are presented two case studies based on industrial tests illustrating the model application and validation of obtained results by comparison with tests data.

4.1. Case study 1. PHE at evaporation station of the sugar factory

One of the applications where fouling from water solution can cause significant problems with heat exchangers is thin sugar juice heating before feeding to the first effect of evaporation station of the sugar factory. The thin sugar juice is water solution about 15% of sugar which does not cause fouling on heating having direct solubility. The main fouling component is dissolved calcium carbonate of inverse solubility with some solid particles remaining after sugar extraction from beet chips and juice purification process, where other organic substances are removed. In paper [18] are presented the data of monitoring PHE thermal performance on such position on one of beet sugar factories in Eastern Europe. The PHE of Alfa Laval production type M15 M is installed. The steam condensate after the first effect is used as heating media. This condensed water is clean and practically not causing fouling on a hot side of heat exchanger. The geometrical parameters of plates and corrugations on the main heat transfer area of plates are as follows: corrugations angle $\beta = 35°$; corrugation height b = 4 mm; equivalent diameter $d_e = 8$ mm; channel cross-section area $f_{ch} = 0.00176$ m^2; heat transfer area of plate $F_{pl} = 0.62$ m^2; corrugation aspect ratio $\gamma = 0.58$; area enlargement factor $F_x = 1.2$. In Table 1 are presented the data of recording flowrates and temperatures of streams at inlet and outlet of PHE during 13 days of operation before first cleaning of the heat exchanger. The cleaning was needed because the pressure drops of thin sugar juice in PHE became up to 1.0 bar that was excessive for installed pumping equipment. With the recorded values of inlet temperatures for both streams, the calculation of temperatures at the exits of streams from PHE are performed according to a presented mathematical model with developed software and the results are presented in Table 1.

The values of dimensionless empirical coefficients are obtained by least squire method using tests data and are as follows: $c_D = 2.291 \cdot 10^6$; $c_R = 0.1259$; $c_{rm} = 0.451 \cdot 10^{-15}$. The value of activation energy is taken as $E = 52{,}100$ J/mol according to data of paper [18] on averaged parameters for the same fouling media. The determined by model outlet temperatures of streams are in good agreement with measured values, the error is not more than 0.3 °C for outlet temperature of condensate and not more than 0.2 °C for thin sugar juice. It confirms the validity of the model for prediction of PHE thermal performance. The predicted by model results of calculation local fouling thermal resistances for two different times of operation are presented by graphs in Fig. 2. It is showing the change of local thermal resistance and correspondingly fouling layer thickness not more than 22% with an increase to exit from the channel. It led to the conclusion that in considered conditions of relatively small temperatures changes the calculation on local parameters gives rather small advantages in thermal design and averaged calculations can be used like it is made in the paper [18]. However, it can be more significant in the calculation of pressure losses.

The analysis of pressure loss structure according to Eq. (29) revealed that up to 0.2 bar is lost in ports and collectors of PHE that has for juice two standard connections on fixed frame plate. These pressure losses are not adding anything to heat transfer performance and just leading to more energy for pumping. It was proposed to add two more connections on the movable frame-plate for thin juice. Such frame plate was installed and at the next year sugar season, PHE with the same plate pack worked for 93 days without cleaning. In Fig. 3 are presented data of pressure drop monitoring in that period and comparison with results of pressure loss modelling by the developed model. The modelling is made for thin juice flow rate G2 = 290 m^3/h, which however changed ±7% during tests. The discrepancies of test data and calculated results are up to 30%, but it can be considered as acceptable accuracy in industrial conditions of tests with fluctuations of flowrate and other parameters. Besides, the dynamic nature of fouling development with simultaneous deposition and removal processes results in the unstable surface of the fouling layer and fluctuations of

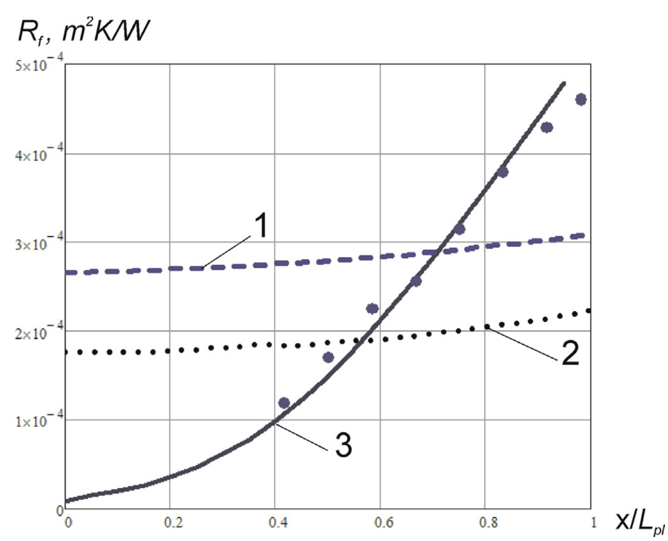

Fig. 2. Calculated local thermal resistance of fouling in PHE channel. Case study 1: (1) –28 d of operation; (2) –13 d of operation. Case study 2: (3) –28 d of work at flow velocity $w_2 = 0.26$ m/s; experimental values are shown by dots.

Table 1
Test data and calculated outlet temperatures for PHE heating thin sugar juice.

Time θ, h		144	216	264	312
Thin sugar juice:					
the mass flowrate, kg/s		71.80	74.50	76.50	73.20
inlet temperature, °C		101.0	100.5	102.0	101.7
outlet temperature:	experimental$_p$, °C	105.0	106.0	107.0	106.0
	calculated, °C	104.89	105.91	106.81	105.91
Condensate of the first effect steam:					
the mass flowrate, kg/s		16.72	16.21	17.48	17.02
Inlet temperature, °C		123.49	123.51	123.51	123.51
outlet temperature:	experimental, °C	102.8	104.8	106.1	104.8
	calculated, °C	102.61	104.42	105.83	104.61

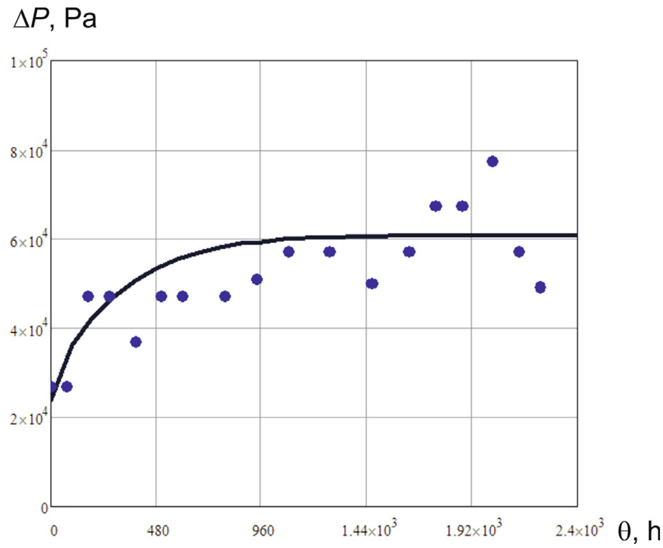

Fig. 3. Pressure loss in PHE for thin sugar juice heating of Case study 1: solid curve is calculated; dots are test results.

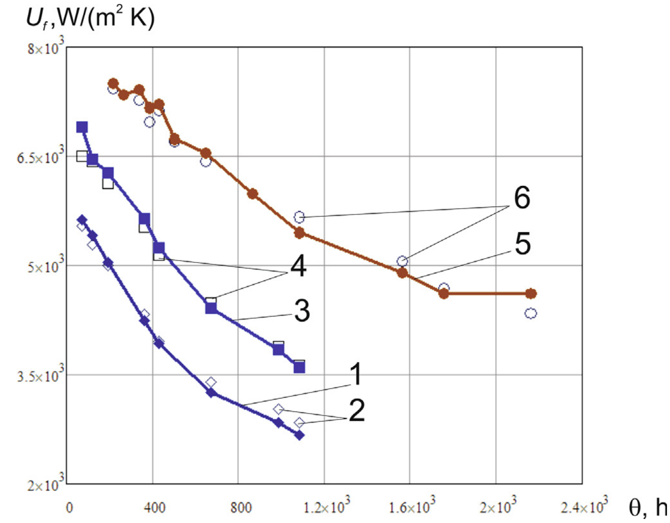

Fig. 4. Tests data (empty dots) and calculated by model results (solid dots) for overall heat transfer coefficients in PHE of Case study 2 at different flow velocities: 1,2 – 0.26 m/s; 3,4 – 0.4 m/s; 5,6 – 0.57 m/s.

its thickness, that leads to uncertainty of process parameters measurement even in laboratory conditions by many researchers, e.g. Refs. [16,24], and even up to 50% like in paper [62].

4.2. Case study 2. PHE for water heating in the DH system

District heating (DH) is widely used and at the same time fast-developing technology that enables to integrate for buildings space heating and tap hot water supply different energy sources including cogenerated heat and power, waste to energy, different kinds of renewables [61]. Some now obsolete systems were initially designed without the need in heat exchangers, as some "open" systems in Russian Federation [61]. Such systems provide the same pipes of the network water for space heating and hot domestic water. To compensate for hot water supplied by the system they need the same big amount of make-up water added to the distribution network. In dissertation [17] was tested the PHE installed on this position at DH system of city Tula in Russian Federation. The tests were made with Alfa Laval M10B plate-and-frame heat exchanger. It was heating fresh water supplied with temperature from 7.9 to 9.5 °C up to 60 °C with water from the boiler circuit having a temperature in the range from 74 to 98 °C. The data of corrugations geometry for a plate of M10B heat exchanger the angle $\beta = 60^\circ$ and parameter $\gamma = 0.56$ are taken by measurement on a commercial plate. The channel height b = 2.93 mm as given in Ref. [17]. The tests data on temperatures and flowrates are from appendixes in Ref. [17] for three sets of experiments at constant flow velocities in channels: 0.57 m/s, 0.4 m/s and 0.26 m/s. The tested fresh water can be considered as a weak solution of calcium carbonate salt of inverse solubility, that was the main component of fouling deposit.

The values of dimensionless coefficients in Equation (17) are determined by the least squares method for tests results with a flow velocity of 0.57 m/s. There are: $c_D = 8.181 \cdot 10^5$; $c_R = 0.0451$; $c_{rm} = 0.171 \cdot 10^{-16}$ with activation energy $E = 52{,}100$ J/mol. The experimental values of overall heat transfer coefficients and predictions by the proposed mathematical model are presented in Fig. 4. The discrepancies for tests at all flow velocities do not exceed ±7% that is confirming the model validity in the simulation of PHE thermal performance. The results of calculation for local values of fouling thermal resistance presented in Fig. 2 are demonstrating the significant variation in fouling depositions along the channel length that emphasizes the need to account for local process parameters distribution in conditions of tap hot water heating in the heat exchanger. The data on pressure losses are not presented in publication [17], but calculations with a proposed mathematical model for the last tests in series for constant flow velocity give the values from about 1.0 to 1.4 bar which is usually the cause to stop operation for PHE cleaning.

5. Conclusions

The development of water fouling on the heat transfer surface of PHE can be with acceptable for practical applications accuracy simulated by the presented mathematical model. The model is represented by the system of differential equations accounting for the distribution of local process parameters along the heat transfer surface and their development in time. The important features of mathematical modelling are (a) the use of fouling model derived in a dimensionless form that includes dimensionless coefficients for its adaptation for specific media with fouling tendencies and (b) simulation of pressure drop based on recommendations for fouling in crude oil heat exchangers. The values of empiric coefficients can be determined with data of monitoring PHE thermal performance in industrial conditions. The application of the developed modelling method is illustrated in two examples for data of different PHEs tests in the sugar industry and District Heating. In these examples is confirmed model validity and good for industrial applications accuracy of simulated results on temperature distribution, fouling development and pressure drop in tested PHEs. The error on overall heat transfer coefficient is not more than ±7% and on pressure drop not more than ±30%. The calculations with the proposed modelling approach enhance the accuracy of PHE design and give the tool to develop fouling mitigation measures by properly adopting construction elements of the heat exchanger. The further development of proposed approach requires more data of PHE monitoring at different fouling conditions and checking its application in a bigger class of fouling media, including different sorts of cooling water, that can be subject of further researches.

Acknowledgements

This research has been supported by the EU project "Sustainable Process Integration Laboratory – SPIL", project No. CZ.02.1.01/0.0/0.0/15_003/0000456 funded by EU "CZ Operational Programme Research, Development and Education", Priority 1: Strengthening capacity for quality research in a collaboration agreement with National Technical University "Kharkiv Polytechnical Institute" and AO Spivdruzhnist-T LLC, Czech Republic. Olga Arsenyeva has been grateful to the Alexander von Humboldt Foundation, Germany for financial support.

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