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The Effect of Plate Corrugations Geometry on Performance of Plate Heat Exchangers Subjected to Fouling

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The novel approach of estimating the influence of plate corrugations geometry on plate heat exchanger (PHE) operation in conditions of fouling is proposed. It is based on the presented mathematical model of the PHE with commercially produced plates. To account for fouling on the heat transfer surface, the fouling model of reaction and transport type presented in dimensionless form is employed. The influence of plate corrugations geometry on PHE performance is discussed on results of modelling PHE installed for thin juice heating at evaporation station of the sugar factory. The corrugations inclination angle to the main flow direction is considered as the main influencing parameter. The effect of this parameter on fouling intensity is shown for the cases when the plates with different corrugations angle are used in one PHE. The measures to mitigate fouling in PHE by optimal selection of plate corrugations geometry are discussed.

1. Introduction

The sustainable development of modern society requires the efficient use of energy. Nowadays most of it is generated by the combustion of fossil fuels. The use of fossil fuels leads not only to exhaustion of the limited natural resources but also to the emission of harmful substances, including greenhouse gases and carbon dioxide (Klemeš et al., 2010). To limit the consumption of energy obtained from primary sources is possible by efficient heat recuperation with the use of efficient heat transfer equipment like compact heat exchangers (Klemeš et al., 2015).

Plate heat exchangers (PHEs) are one of the most widely used in industry types of compact heat exchangers. The construction and operation principles of PHEs are well described in the literature, e.g. book by Wang et al. (2007). It is much smaller in size and internal volume than conventional shell and tube heat exchangers, it uses much less material for heat transfer area and has smaller ecological footprints. Among PHE’s advantages are a lower cost with the use of expensive materials (Hajabdollahi et al., 2016) and close temperature approach down to 1 K that is very important for the effectiveness of heat recuperation. It stipulated the efficient use of PHEs in different applications at chemical industry (Kapustenko et al., 2009), crude oil refineries (Arsenyeva et al., 2016), food (Arsenyeva et al., 2016b) etc.

The heat transfer surface of PHE is formed by corrugated plates assembled in a pack with multiple contact points of adjacent plates. It creates a rigid structure capable of withstanding high difference in pressures of heat exchanging streams. The channels between the plates have intricate complex geometry that promotes intensive turbulence stipulating heat transfer enhancement. It also leads to much higher level of shear stress at the channel’s wall that creates the effect of fouling mitigation, as was shown experimentally by Crittenden et al. (2015) for the model of fouling formation on the PHE channel surface.

Much lower fouling tendency in PHEs than in shell and tube heat exchangers is well recognised as it is illustrated by recommended thermal resistances of fouling for different media by Wang et al. (2007). Those approximate

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values for PHE design are up to ten times smaller than for shell and tube in standards of Tubular Exchangers Manufacturers Association (TEMA, 2007). However, such approximate averaged values of fouling thermal resistance can lead to significant error in determining PHE’s performance at fouling conditions, as overall heat transfer coefficient in PHEs can be by two times and even more higher than in shell and tubes. The local features of the fouling process in PHE with flat plates were accounted in a model presented by Guan and, Macchietto (2018). For a correct prediction of water fouling in PHE from commercial corrugated plates the thermal model based on process, local parameters were proposed by Kapustenko et al. (2018). It was further developed with the introduction of the equation for fouling rate in the dimensionless form in a paper by Kapustenko et al. (2019). The geometry of plate corrugations is significantly influencing PHE performance. The novelty of the present paper is in presenting an approach to study the influence of plate corrugations geometry on the thermal and hydraulic performance of PHE in fouling conditions. It is illustrated by the example of PHE operating in the industry.

2. The mathematical model

The thermo-hydraulic mathematical model of fouling formation on plate-and-frame PHE heat transfer surfaces is developed based on the following assumptions:

i. The process in one pass PHE is considered with counter-current flow.

ii. The process conditions in all channels for one stream are the same.

iii. The heat losses to the environment are neglected.

iv. The process parameters are uniformly distributed across the channel, and their changes along the channel length are accounted.

v. The fouling at the hot stream side is not considered.

vi. The PHE channel can be regarded consisting of the main corrugated field (4 in Figure 1), distribution zones (2, 5) and flow entrance-exit (1).

vii. The dominant heat transfer process is happening on the main corrugated field of plates and for the whole plate can be calculated by uniform correlations presented in a paper by Arsenyeva et al. (2012).

viii. The pressure losses at flow entrance and distribution zones are accounted as local hydraulic resistances.

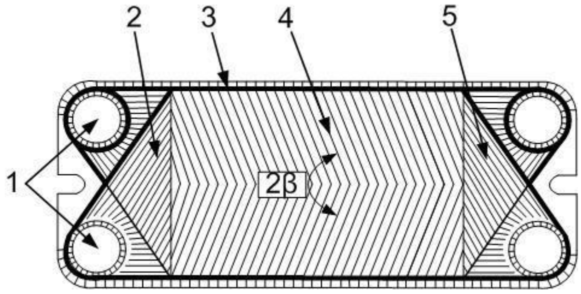


Figure 1: Schematic drawing of PHE plate area: 1 - holes forming collectors at inlet and outlet of streams; 2, 5 - flow distribution zones; 3 - elastomeric gasket; 4 - the main area of heat transfer surface field.

The mathematical model is based on a system of differential equations presented in a paper by Kapustenko et al. (2018). For estimation of fouling rate, the fouling model based on reaction and transport fouling mechanism is used, as presented in a paper by Kapustenko et al. (2018b) in the following dimensionless form:

$$\Phi_f = \frac{\partial \delta_f}{\partial \theta}\frac{d_e \cdot \rho_2}{\mu_2} = \frac{1}{c_D \cdot \dfrac{K_D^{2/3} \cdot Pr_2^{1/3}}{Nu_2} + c_R \cdot K_R \cdot \exp\left(\dfrac{E}{R \cdot T_s}\right)} - c_{rm} \cdot Re^{*2} \cdot Pr_2 \frac{\delta_f}{d_e} \qquad (1)$$

Where

$$Re^* = \frac{\sqrt{\tau_w \cdot \rho_2} \cdot d_e}{\mu_2};\ K_D = \frac{\mu_2^2 \cdot r_m}{(T_s \cdot \rho_2 \cdot k_B)};\ K_R = \frac{\tau_w}{(\rho_2 \cdot d_e \cdot g)}.$$

here $Nu_2 = h_2 \cdot d_e / \lambda_2$ is the Nusselt number of heated stream; $Pr_2 = c_{p2} \cdot \mu_2 / \lambda_2$ is the Prandtl number; ρ_2 is liquid density, kg/m^3; λ_2 is thermal conductivity of the liquid, J/(m·°C); c_{p2} is specific heat capacity of the liquid, J/(kg·°C); d_e is channel equivalent diameter, m; $k_B = 1.38048 \cdot 10^{-23}$ J/K is Boltzmann constant; T_s is the surface temperature, K; μ_2 is dynamic viscosity, Pa·s; $r_m = 1.36 \cdot 10^{-10}$ is Van der Vaals water molecule radius, m; δ_f is the thickness of the deposited fouling layer, m; ϑ is time, s; R is universal gas constant, J/(mol K); E is reaction activation energy,

J/mol; c_D, c_R and c_{rm} are dimensionless model parameters determined by experimental data for specific fouling media, as is illustrated in cited paper.
For estimation of the friction factor on the main corrugated field of PHE channel, the Eq(2) is used. It was proposed by Arsenyeva et al. (2012) for criss-cross flow channel formed by PHE plates with a different form of corrugations and is used with the accounting for surface roughness by analogy with equation proposed for flow inside tubes (Churchill, 1977) in the same form.

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

where

$$A = \left[\mathrm{p4}\cdot ln\left(\mathrm{p5}\cdot\left(\left(\frac{7\cdot\mathrm{p3}}{\mathrm{Re}}\right)^{0.9} + 0.27\cdot\frac{\varepsilon}{d_e}\right)^{-1}\right)\right]^{16} \quad (3)$$

The parameters in these Equations are calculated depending on the geometrical form of corrugations, which is determined by the inclination angle of corrugations to the main flow direction β, degrees, and the ratio of equivalent diameter to corrugations pitch $\gamma = d_e/S$:

$$\mathrm{p1} = \exp(-0.157\cdot\beta);\quad \mathrm{p2} = \frac{\pi\cdot\beta\cdot\gamma^2}{3};\quad \mathrm{p3} = \exp\left(-\pi\cdot\frac{\beta}{180}\cdot\frac{1}{\gamma^2}\right);\quad \mathrm{p5} = 1+\frac{\beta}{10};$$
$$\mathrm{p4} = \left(0.061+\left(0.69+\mathrm{tg}\left(\beta\cdot\frac{\pi}{180}\right)\right)^{-2.63}\right)\cdot\left(1+(1-\gamma)\cdot 0.9\cdot\beta^{0.01}\right) \quad (4)$$

This Eq(4) is valid in the range of geometrical parameters: β from 14° to 65°; γ from 0.5 to 1.5; the coefficient of area enlargement because of corrugations F_X from 1.14 to 1.5. For calculation of friction factor for technically smooth surface of stainless steel PHE plates the parameter accounting for roughness ε, m, in Eq(3) is taken as $\varepsilon/d_e = 1\cdot 10^{-5}$. For fouled PHE channel it is taken as $\varepsilon = \delta_f$.
For calculation of heat transfer is used the equation Eq. (5) obtained based on heat and momentum transfers analogy in PHE channels in a paper by Kapustenko et al. (2011).

$$Nu = \frac{\mathrm{h}\cdot\mathrm{d_e}}{\lambda} = 0.065\cdot\mathrm{Re}^{6/7}\cdot\left(\psi\cdot\zeta/F_X\right)^{3/7}\cdot\mathrm{Pr}^{0.4}\cdot\left(\mu/\mu_w\right)^{0.14} \quad (5)$$

Here Ψ is the share of friction losses in total pressure losses in the channel of PHE, determined as:

$$\psi = \left(\mathrm{Re}/\mathrm{A}\right)^{-0.15\cdot\sin(\beta)} \text{ at } Re > A;\ \psi = 1 \text{ at } Re \le A \quad \text{where} \quad A = 380/\left[tg(\beta)\right]^{1.75} \quad (6)$$

The shear stress at the channel surface is calculated with accounting for an increase of flow velocity with a decrease of channel cross-section area due to fouling:

$$\tau_w = \zeta\cdot\psi\cdot\frac{\rho\cdot w^2}{8} \quad (7)$$

The total pressure drop of heated media in the PHE is calculated as a sum of pressure losses on the corrugated field, pressure losses in the inlet and outlet distribution zones with local coefficients of hydraulic resistances $\zeta_{DZin} = \zeta_{DZout} = 38$ and pressure losses in port and collectors:

$$\Delta P_2 = \int_0^{L_p}\zeta_2\cdot\frac{\rho_2\cdot w_2^2}{2\cdot\mathrm{d_e}}dx + \zeta_{DZin}\cdot\frac{\rho_2\cdot w_{2in}^2}{2} + \zeta_{DZout}\cdot\frac{\rho_2\cdot w_{2out}^2}{2} + 1.3\cdot\frac{\rho_2\cdot w_{2p}^2}{2} \quad (8)$$

The use of the presented Eq(1) –Eq(8) together with differential Equations from paper by Kapustenko et al. (2018), accompanied by correlations for thermo-physical properties of substances involved, gives the system of partial differential equations describing the process of heat transfer with fouling formation in the PHE, which accounts the influence of plate corrugations geometry. The system solution with finite difference method is implemented using Mathcad (2019). The results of the modelling allow estimating the influence of plate geometry on PHE performance in specific process conditions, as is illustrated in the following section.

3. Modelling results and discussions

To illustrate the influence of plate corrugations geometry on PHE performance accounting for fouling the conditions of the example presented in a paper by Demirskiy et al. (2018) are considered The M15M PHE type of AlfaLaval production is installed for thin juice heating before the evaporation station at the sugar factory. The modelling is performed for the inlet juice temperature equal to 102 °C. It is heated by the steam condensate with temperature 124 °C, coming from the first evaporation effect. The flowrate of juice is 290 m^3/h and flowrate of condensate, which heats it, is 65 m^3/h. The installed PHE has 151 plates with one pass arrangement. The height of PHE is 1941 mm, width 610 mm and frame length 3700 mm. The length of the plate pack is 700 mm. The used plates are from AISI 316 stainless steel with the corrugations inclination angle to the main flow direction β = 35°, plate spacing 4 mm and the ratio of equivalent diameter to corrugations pitch γ = 0.58. After carrying out the initial retrofit of the evaporation station purposing to decrease flow velocities and pressure drop in ports and collectors, the PHE was connected to the thin juice line both on fixed and moving frame plates. The dimensionless parameters in fouling model, presented by Eq(1), are taken equal to the values determined by the data of PHE onsite monitoring, and presented in paper by Kapustenko et al. (2019): $c_D = 2.291 \cdot 10^6$, $c_R = 0.1259$, $c_{rm} = 0.451 \cdot 10^{-15}$, the activation energy E = 52,100 J/mol. The simulated results of the dynamic behaviour in time of the fouling thermal resistance R_f and change of heat load Q are presented in Figure 2 by corresponding curves. The thermal fouling resistance is going up to 0.0003 W/($m^2 \cdot$K), and heat load drops from 1,500 kW for clean PHE down to 1,300 kW. There are also presented the results of modelling for PHEs consisting of plates forming medium channels with average angle β_m = 50° and from plates with β_h = 65°. For PHEs with plates of higher angle β the thermal fouling resistance is much smaller and heat load much more significant than for the PHE with β_l = 35°. However, it also involves bigger loss of pressure.

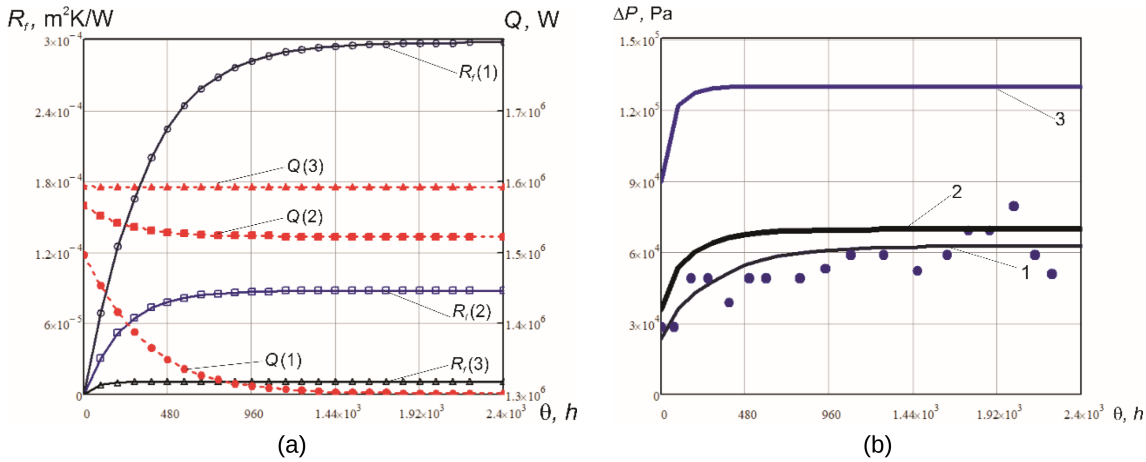
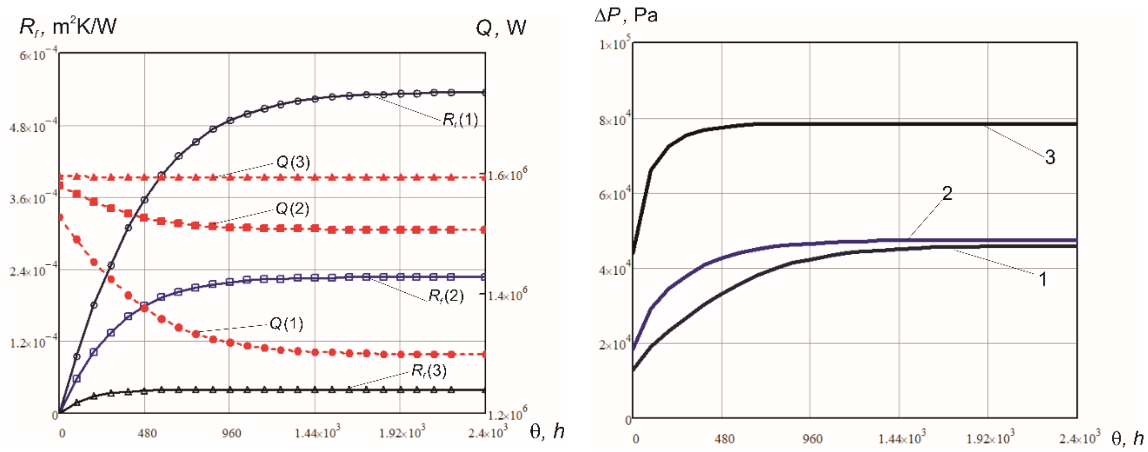


Figure 2: Development in time for PHEs with 150 plates. (a) Fouling thermal resistances (R_f) and heat loads Q: R_f(1) - β_l = 35°; R_f(2) – β_m = 50°; R_f(3) – β_h = 65°; Q(1) - β_l = 35°; Q(2) – β_m = 50°; Q(3) – β_h = 65°. (b) Pressure drops: 1 - β_l = 35°; 2 – β_m = 50°; 3 – β_h = 65°.

The calculated pressure losses for PHEs with differently corrugated plates (Figure 2b) reveals considerable differences, which changes over time with fouling development. The pressure drop in clean PHE with β_l = 35° is 23 kPa but rises to 62 kPa at the asymptotic stage of the process. The measured pressure drops in this PHE are presented by dots (Figure 2b). For clean PHE with β_m = 50°, the pressure drop is 37 kPa what is 60 % bigger, but at the asymptotic stage, it is equal to 69 kPa, which is bigger only on 11 %. However, the heat load at the asymptotic stage is 1,520 kW, that is higher than even value for the clean PHE installed now, but, in comparison to its asymptotic stage, it saves 220 kW of heat energy. The PHE with β_h = 65° can give even extra 70 kW, but the pressure drop for the clean condition is 94 kPa and rising to 130 kPa or on 38 %. Such a high-pressure drop is not acceptable, but the tendency of its relatively small rising with fouling can be accounted for in the analysis of other options.

To decrease the pressure drop in PHE and increase heat recuperation, an option of adding plates is considered. The results of modelling thermal behaviour of PHEs with 225 different plates are presented in Figure 3. However, the expected increase in heat recuperation is not observed due to the dramatic rise of fouling thermal resistance. The heat load for all PHEs at the asymptotic stage even slightly dropped on about 1 %. It is only partly compensated by an increase in the heat transfer area. At the same time, the pressure drop in PHEs became much more acceptable for their operation. The pressure drop in clean PHE with β_l = 35° is 12 kPa and rises to 46 kPa at the asymptotic stage of the process. For clean PHE with β_m=50°, the pressure drop is 19 kPa or 60 % bigger, or the same as for PHE with 150 plates. However, at the asymptotic stage, it is 47 kPa or practically the same as for PHE with β_l = 35°. Counting for increased heat recuperation on 220 kW, the option of PHE with β_m = 50° is much more preferable.
From the point of increased heat recuperation, the option of PHE from plates with high corrugations angle β_h = 65° looks lucrative, as it increases the heat load on about 80 kW. However, the pressure drop for PHE with 225 plates is 79 kPa that is too high. To make pressure drop to near 50 kPa, the number of plates has to be increased to 340. It requires much bigger and costly heat exchanger or even two installed in parallel. For PHEs with low and medium angles, β_l = 35° and β_m = 50° such increase of heat transfer area practically not effects recuperated heat with a decrease of pressure drop only to 35 kPa. This is not making such option better than PHE with 225 plates, counting for additional cost involved.



(a) (b)

Figure 3: Development in time for PHEs with 225 plates. (a) Fouling thermal resistances (R_f) and heat loads Q: R_f(1) - β_l =35°; R_f(2) – β_m = 50°; R_f(3) – β_h = 65°; Q(1) - β_l = 35°; Q(2) – β_m = 50°; Q(3) – β_h = 65°. (b) Pressure drops: 1 - β_l = 35°; 2 – β_m = 50°; 3 – β_h = 65°.

To estimate the different options of PHE retrofit, the economic considerations must be involved. The use of PHE consisting of 225 plates with medium corrugations angle β_m = 50° allows to save 220 kW of heat energy compare to basic case with PHE from 151 plates and β_l = 35°. The sugar campaign for this factory is lasting about four months or 120 d. During this period energy saving would be 220x120x24 = 633,600 kW·h or 2,281,000 MJ. It allows using less steam that is generated at the factory by firing natural gas. Counting the efficiency of 0.7 for steam generation and specific heat of natural gas combustion 39 MJ/m^3, it will make the economy of about 84,000 m^3 of natural gas. The price of natural gas for the factory is 0.31 EUR/m^3, and economy for 120 d of beet sugar campaign is 26,000 EUR. According to a quotation from Alfa Laval for this factory from 21.04.2018, the price of one M15M plate with nitrile gasket bought as spare parts is about 98 EUR. The cost of an additional 112 plates for PHE retrofit is 10,976 EUR. That will require reassembly of plates on the same frame that can be easily made for 2000 EUR including any additional costs. That comes to the total cost of such retrofit 13,000 EUR with payback in two months. The other options involving purchasing of the new PHE are less economical as such PHE with 225 plates according to Alfa Laval quotation can cost about 29,000 EUR and payback time can be up to 5 months. The cost of two PHEs with total 340 plates is even more, and additional economy of heat energy in the amount of 80 kW cannot compensate it fully.

4. Conclusions

The presented mathematical model allows analysis of the performance of PHE in conditions of fouling on its heat transfer surface and development of process parameters in time with accounting for the effect of plate corrugations geometry. It has to be accounted for the correct selection of PHE in the conditions of fouling. The increase of heat recuperation by simply adding plates with the same low inclination angle β of corrugations to the main flow direction in certain conditions is not leading to the desired effect, as at the same time thermal resistance of fouling increases and after some time of operation can become much more significant than in smaller PHE. However, such a measure can reduce the pressure drop in the PHE. The use of plates with higher β can be preferable in water fouling conditions, as even at higher pressure drop in clean PHE its increase can be smaller than at lower angle β with fouling development leading to the same pressure drop after some time of PHE operation. At the same time, the amount of recuperated heat can be of a much higher level. However, the final decision must be made based on correct process modelling with consideration of economic factors.

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