

PARTICULATE FOULING OF POLYMERIC HOLLOW FIBER HEAT EXCHANGER

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ABSTRACT

Thermal effectiveness of heat exchangers is negatively affected by tube fouling caused by working in contaminated conditions. Surface of polymers is smooth and low-adhesion which provide additional antifouling resistance and defouling possibility. We prepared a polymeric hollow fiber heat exchanger and tested it to cool down water contaminated by titania particles. Thermal performance tests showed its high effectiveness and good fouling resistance. It was found that the particle deposit developed rapidly and achieved its asymptotic values during hours. It should be mentioned that the high rates of fouling evolution were associated with low liquid velocities. The asymptotic (maximal) fouling level was low enough causing a small influence on heat transfer. The associated decrease of the overall heat transfer coefficient (HTC) was 5-20 % depending on flow regimes. Moreover it was found that particle deposit is not strong and can be easily detached and removed from the fiber surface by local increase of liquid velocity. The injection of air bubbles in the heat exchanger shell was tested as the defouling procedure and showed good efficiency.

INTRODUCTION

The photocatalytic degradation of organic pollutants is based on the ability of TiO₂ particles to perform oxidation and reduction on their surfaces. This process is used to treat different types of polluted water. Fig. 1 shows scheme of continuous photo-degradation of polluted water.

The basic precondition is that TiO₂ particles have the adequate crystallographic form and they must be activated by suitable ultraviolet (UV) radiation, for details see Sarasidis and Patsios (2011). However, it is a well-known fact that the efficiency of UV radiators is relatively low and therefore a significant fraction of the total electricity input is transferred not into UV radiation but into heat. The heat generation is so intensive that coolers are required to prevent the water from boiling. The heat exchangers are used either in the separation or degradation sections. Fouling is an integral part of the work of these heat

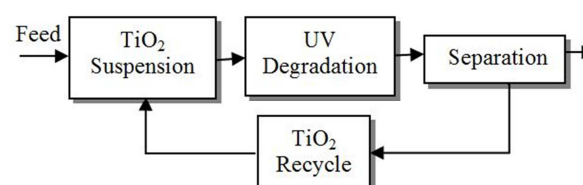


Fig. 1 Continuous photo-degradation scheme

exchangers. It is combined in its nature and depends on the type of polluted water that has to be cleaned. However, in every case it is associated with particulate fouling of heat exchanger by TiO₂ particles.

The nature of particulate fouling is described in detail by Grandgeorge et. al. (1998). Particle accumulation on surfaces from liquid suspension flow can be considered to be a continuous process. Particles are transferred from the suspension bulk at the concentration c_b to the wall surface and then adhere to it. The deposition flux Φ_d can be expressed as:

$$\Phi_d = k_d c_b \quad (1)$$

$$\frac{1}{k_d} = \frac{1}{k_m} + \frac{1}{k_a} \quad (2)$$

where k_d , k_m and k_a are, respectively, the overall deposition coefficient, the mass transfer coefficient and the adhesion coefficient. According to the respective values of k_m and k_a , the global process is limited by the mass transfer or by the adhesion.

During particle deposition, the deposit thickness generally reaches a constant asymptotic value R_f^* . (Bott, 1995). On the one hand, it is possible that a continuous removal process occurs during the deposition process. On the other hand, the asymptotic behavior can be explained if the deposition flux decreases with the deposit thickness. If the deposit thermal conductivity and density are constant, time evolution of the fouling thermal resistance can be expressed through asymptotic value R_f^* and fouling process time constant t_c as (Gradgeorge et. al., 1998):

$$R_f(t) = R_f^* (1 - e^{-t/t_c}) \quad (3)$$

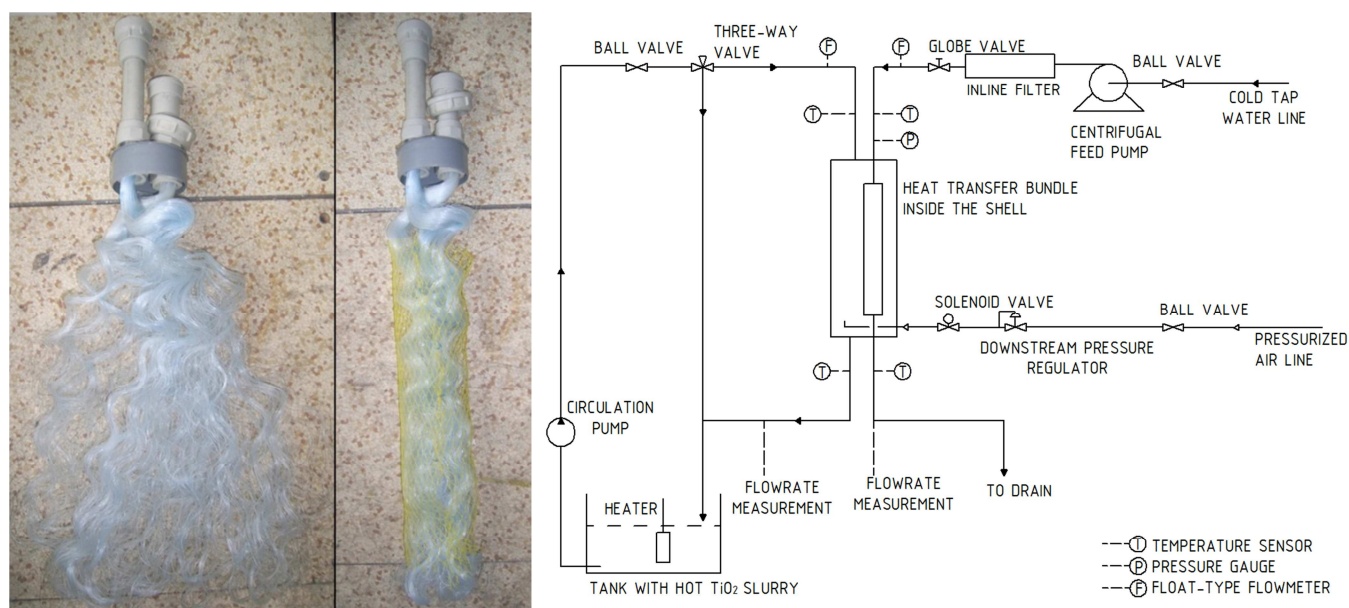


Fig. 2 Hollow fiber heat transfer bundle (at the left) and experimental setup scheme

Value of R_f^* shows the maximum level of fouling and its influence on heat transfer and t_c indicates the rate of its development. Both parameters are influenced by parameters of flow such as foulant concentration, particle size distribution, pH (measure of the activity of the hydrogen ion) and liquid velocity.

The heat exchanger from the polypropylene (PP) hollow fibers was prepared to cool TiO_2 suspension and tested under fouling conditions. Such heat exchangers were proposed as an alternative for low-temperature application by Zarkadas and Sirkar (2004). More information is also provided by Song et. al. (2010) and Qin et. al. (2012). It is a low weight, low cost and corrosion resistant heat exchange system. It is expected that this heat exchanger has a high thermal performance as well as a high fouling resistance and defouling ability. A previous study showed that small-diameter fiber has a superior overall heat transfer coefficient even in low-velocity flow (Astrouski and Raudensky, 2012). Moreover polypropylene fiber has a smooth surface and low surface energy (29 mJ/m²) causing chemical inertness and low adhesion (James, 1999). Furthermore polymer fibers is flexible enough to move in a flow. In accordance with Wicaksana et. al. (2006) bubbling is commonly used to control the fouling of submerged hollow fiber membranes. In the same manner injecting of pressurized air can be used to increase liquid velocity causing the movement of fibers and deposit removal. The results of fouling tests and the efficiency of the applied defouling procedure are described in this paper.

EXPERIMENTAL

Heat Exchange Element

The heat transfer bundle (see fig. 2) was built from PP fibers whose ends were collected in plastic screwed pipe fittings. Used fibers were twisted with different curvatures (around 50-100 mm) to ensure equal distribution of these fibers in volume. To produce this curvature the fibers were

wound around 50 mm o.d. steel tubes. Tubes with fibers were heated up to 120° C for 5 minutes and then they were cooled naturally to stabilize this twisted shape. Epoxy resin was used to fill the space between fibers in the fittings. The ends of the fittings were milled and heated by hot air (around 120° C) to remove burrs and open all ends of fibers. In addition, the bundle was pressurized by water up to 4 bar gauge pressure to ensure an absence of leakages in fittings and fibers. All damaged fibers (which showed leakages during the test) were eliminated. Moreover, colored water (with a small addition of ink) was used to determine fibers through which water can flow. The total amount of functional fibers in the bundle was 262 and effective immersed length was 1300 mm. It should be mentioned that extruded polymeric fiber has relatively high oscillation of both diameters along the length. The relative difference of outer diameter measured in two places can exceed 5 %. Inner and outer fiber diameters (0.461 and 0.523 mm respectively) were statistically obtained by studying light-microscope photos of milled cross-sections of the fittings as a mean values. Thus, the total heat transfer area based on the inner fiber surface was 0.493 m².

Experimental Setup

The experimental setup scheme is shown in Fig. 2. The heat transfer bundle was placed into a transparent Plexiglas 100 mm i.d. tube which was the shell of the heat exchanger. The shell was under atmospheric pressure and hot foulant fluid circulated through the shell naturally (circulating pump lifted the slurry up and then it flowed down through the shell). The foulant fluid was a suspension of TiO_2 in tap water (with concentration 1-5 g/l). Suspension particle size distribution was measured by laser diffraction method by Malver Mastersize 2000 granulometer. 90% (by mass) of TiO_2 particles have size of 1 – 10 μm . The level of the liquid in the shell was adjusted by siphon piping and the shell flow rate was controlled by a three-way valve installed upstream. The slurry returned from the shell was re-heated

in a bath having three heaters with a total capacity of 6 kW to temperature around 28-30 °C.

The cold tap water was used as a cooling medium. It was pumped through the fiber bundle by means of a centrifugal pump. The flow rate of cooling water was adjusted by means of a globe valve but the temperature was not controlled and was 8-10° C. It was not controlled because temperature of water flowing inside the fiber as coolant has no influence on particulate fouling of external fiber surface. An inline filter (50 µm particle size) was installed upstream from the fiber bundle to protect lumen path of fibers from fouling.

The shell and tube (lumen path of fiber) flow rates were measured by float-type flowmeters calibrated for water and checked at the two ends by measuring the time required to collect a certain volume of liquid. Inlet and exit temperatures of both streams were measured with an accuracy of ± 0.1 °C by PT100 resistance sensors with 3-wire connections. They were recording with frequency 0.1 Hz by datalogger throughout the measurements.

The pressurized air line was connected to the bottom side of the shell to inject air bubbles inside the shell liquid and remove the collected particulate deposit. It consisted of a downstream pressure regulator to control air injection pressure and solenoid valve to control time of injecting.

Fouling Test Procedure

Experiments consisted of three sets. The goal of the first part was to determine values of asymptotic fouling resistance R_f^* relative to the foulant velocity. At the beginning overall heat transfer coefficient (HTC) of pure water was determined. Then pre-diluted concentrated titania solution was added to the circuit and fouling started. When fouling had achieved its asymptotic value overall HTC was measured one more time. This procedure was repeated for different shell velocity. The tube flow rate was maintained constant (around 0.1 kg/s) and the shell flow rate was varied (0.025 – 0.4 kg/s). The concentration of titania particles was maintained constant (1g/l) as well. Between the measurements for different velocity the heat transfer bundle was removed and mechanically cleaned by pressurized water.

The goal of the second part was to study the rate of deposit growth (fouling process time constant t_c) and the influence of TiO₂ suspension concentration on it. Flow conditions of both (tube and shell) liquids were maintained constant for all runs (tube flow rate was 0.1 kg/s and shell flow rate 0.25 kg/s). A typical experiment started with pure water flowing through the experimental section. After the stationary regime was achieved, pre-diluted concentrated TiO₂ slurry was added to the heat bath and fouling started. Outlet temperature change was observed in-real time and used to determine the changing HTC.

Defouling was the subject of study in the third part of experiment. When the HTC had decreased significantly a cleaning procedure was used to remove the deposit. The cleaning process was the same for all runs. It consisted of 5 one-second air injection pulses with a two-second pause between pulses. The injection pressure was set-up to 2 bars by a downstream pressure regulator. Volume of injecting air

was limited by the open cross-section of solenoid valve seat and was around 20 l (at atmospheric pressure) for the whole set of defouling.

REDUCTION OF DATA

Calculation of Physical and Transport Properties

Thermophysical properties of the suspension were estimated as properties of pure water because of low TiO₂ concentration of suspension. Experimentally observed inlet and outlet temperatures for the shell-side and tube-side liquids were used to determine the density, viscosity and thermal conductivity of the liquids using the following correlations (Song et. al., 2010). The averages of the inlet and outlet of the tube-side and shell-side liquid temperatures T_{av} were substituted in the following correlations. For density, kg/m³

$$\rho = 10^{-5}T_{av}^3 - 6 \cdot 10^{-3}T_{av}^2 - 3 \cdot 10^{-3}T_{av} + 1003 \quad (4)$$

dynamic viscosity, kg/m s

$$\mu = (-0.4607 \ln T_{av} + 2.3669)/1000 \quad (5)$$

thermal conductivity, W/m K

$$\lambda = (-0.0097T_{av}^2 - 2.1662T_{av} + 559.2)/1000 \quad (6)$$

Constant value of water specific heat $C_p = 4180$ J/kg K was used for the whole temperature range of test.

The diameter-based shell-side Reynolds number Re_{sh} was calculated based on the shell interstitial velocity u_{sh} and equivalent hydraulic diameter D_h :

$$Re_{sh} = \rho u_{sh} D_h / \mu = ((Q_{f,sh} / A_{sh}) \rho D_h) / \mu \quad (7)$$

$$A_{sh} = \pi D_{sh}^2 / 4 - \left(\frac{1300}{500} \cdot \pi N D_o^2 / 4 \right) \quad (8)$$

where $Q_{f,t}$ is tube volume flow rate, A_{sh} is shell free cross-section, D_{sh} is internal shell diameter, D_o is outer fiber diameter and D_h is equivalent hydraulic diameter calculated as:

$$D_h = 4A_{sh} / P = Q_{f,sh} / \left(\pi D_{sh}^2 / 4 + \left(\frac{1300}{500} \cdot \pi N D_o^2 / 4 \right) \right) \quad (9)$$

where P is wetted perimeter consisted of internal shell surface and external fiber surface. It was assumed that the occupied cross-section and wetted perimeter for 1300 mm twisted fiber bundle can be calculated as 500 mm straight fiber multiplied by 1300/500 ratio.

Determination of Overall Heat Transfer Coefficient and Fouling Resistance

Fouling causes a decrease of overall HTC and can be expressed as fouling thermal resistance R_f (Holman, 2010). It is given by

$$R_f = \frac{1}{U_f} - \frac{1}{U_0} \quad (10)$$

where U_f and U_o are heat transfer coefficients for clean and fouled conditions. In the stationary test set we fitted experimentally observed HTC values by the curve $U = f(Re_{sh})$ and found resulting fouling resistance. Moreover we used Hickman's approach to determine and compare inside, outside, wall and fouling deposit thermal resistances. Sequence of calculations is described below.

The total rate of heat transfer between the shell-side and tube-side liquids, Q , was obtained from

$$Q = C_{sh}(T_{sh,in} - T_{sh,out}) = C_t(T_{t,in} - T_{t,out}) \quad (11)$$

where $C = Q_f \rho C_p$ is capacity rate of flowing liquid. The differences in the Q values between the shell-side and tubes-side liquids were around 5% usually. The overall heat transfer coefficient is assumed constant and is calculated:

$$U = Q / \int_{A_i} (T_{sh,b} - T_{t,b}) dA_i \quad (12)$$

where $(T_{sh} - T_t)$ is mean difference of bulk temperatures. Effective mean temperature difference averaged over the total heat exchange area was determined by using logarithmic mean temperature difference (LMTD) and appropriate LMTD correction factor F . Thus the overall heat transfer coefficient was calculated based on inside fiber surface area as

$$U = Q / (\Delta T_{lm} \cdot F \cdot A_i) \quad (13)$$

where LMTD was determined as

$$\Delta T_{lm} = \frac{(T_{sh,in} - T_{t,out}) - (T_{sh,out} - T_{t,in})}{\ln((T_{sh,in} - T_{t,out}) / (T_{sh,out} - T_{t,in}))} \quad (14)$$

and correction factor was obtained from ChemSOF LMTD Correction Factor Charts and checked by diagrams for one-shell heat exchanger with a one tube path (Holman, 2010). The Q value for the tube-side stream is used since the latter does not present any heat losses to the surroundings.

On the other hand the overall HTC can be expressed as a combination of convection, wall and fouling thermal resistances:

$$U = 1 / \left(\frac{1}{h_i} + \frac{D_i}{\lambda} \ln \frac{D_o}{D_i} + R_f + \frac{D_i}{h_o D_o} \right) \quad (15)$$

where h_i and h_o are convective HTCs, R_f are fouling thermal resistance $(D_i/\lambda)(\ln(D_o/D_i))$ is thermal resistance of wall. We modified and used this expression to fit experimental obtained HTC data.

Hickman's formula was proposed to evaluate the asymptotic Nusselt number for laminar flow inside hollow fiber (Song et. al., 2010). It is given by

$$Nu_{T3} = \frac{(48/11) + Nu_w}{1 + (59/220)Nu_w} \quad (16)$$

$$Nu_w = \frac{U_w D_i}{\lambda_i} \quad (17)$$

$$\frac{1}{U_w} = \frac{D_i}{D_o \cdot h_o} + \frac{D_i}{2\lambda_w} \cdot \ln \left(\frac{D_o}{D_i} \right) \quad (18)$$

$$h_i = \frac{Nu_{T3} \lambda_i}{D_i} \quad (19)$$

The overall Nusselt number can be expressed and calculated as

$$\frac{1}{Nu_{ov}} = \frac{k_i}{UD_i} = \frac{1}{Nu_{T3}} + \frac{1}{Nu_w} \quad (20)$$

The expression for Nu_{T3} in Eq. 16 then can be substituted into Eq. 20 and a second order polynomial expression in terms of Nu_w is then obtained:

$$\left[1 - \frac{59}{220} Nu_{ov} \right] Nu_w^2 + \left[\frac{48}{11} - 2 Nu_{ov} \right] Nu_w - \frac{48}{11} Nu_{ov} = 0 \quad (21)$$

This quadratic equation is then solved and it produces a positive and a negative root. Only the positive root is considered as the wall Nusselt number. It will be shown later that the inside HTC values, which were derived from experimental results by mean of Hickman's formula, are in good agreement with the assumption of constant inside HTC.

Gr/Re^2 complex was calculated to estimate the influence of natural convection in the shell (Holman, 2010). It was found that the natural convection contribution is negligible even at the lowest velocity. Thus the shell-side convection was considered as forced. Shell-side HTC can be expressed through Nusselt number by the following common equation (Holman, 2010):

$$h_o = Nu \frac{\lambda}{D_o} = C Re_{sh}^m Pr^{1/3} \frac{\lambda}{D_o} \quad (22)$$

where Pr is Prandtl number which can be considered as function of temperature and assumed as constant (temperature differences were small). C and m are constants which depend on geometrical configuration of flow around fibers. All parameters in Eq. 22 were maintained constant during experiments except of Reynolds number. Thus substituting h_o values to Eq. 15 gives us following function:

$$U = f(Re_{sh}) = \frac{1}{C_2 + (1/C_1 Re_{sh}^m)} \quad (23)$$

where $1/C_1 Re_{sh}^m$ is shell convection thermal resistance and C_2 is lumped resistance consisting of all other components including fouling resistance. We calculated parameters C_1 , C_2 and m and fitted appropriate curves in Origin 9 software.

RESULTS AND DISCUSSION

Firstly, it should be mentioned that obtained values of overall HTC are relatively high for both clean (up to 2100 W/m² K) and dirty (up to 1750 W/m² K) conditions. Table 1 compares the overall HTC values of different hollow fiber heat exchangers with a regular water-to-water heat

exchange system. It can be concluded that the system used has high, competitive HTC values.

Figure 3 shows variation of clean and fouled overall HTC with a change of the shell Reynolds number. It is seen that HTC has a strong dependence on the shell Reynolds number. It means that shell-side thermal resistance is dominant. It was assumed that fouling resistance was independent from the shell velocity and results have been correlated by using Eq. 23. For clean HTC: $C_1 = 35.21$, $C_2 = 3.34 \cdot 10^{-5}$ and $m = 0.558$ and for fouled condition HTC: $C_1 = 35.21$, $C_2 = 1.15 \cdot 10^{-4}$ and $m = 0.558$. Asymptotic value of fouling resistance R_f^* of $8.2 \cdot 10^{-5} \text{ m}^2 \text{ K/W}$ was obtained in this way. This value and Hickman's approach were used to determine the contribution of separate thermal resistances to the overall one (see figure 4). It is clearly seen that deposit layer resistance is relatively low in comparison with the resistance of wall, tube convection resistance and especially shell convection resistance. In fact it has no significant influence on heat transfer for Reynolds numbers lower than 600. Grandegeorge et. al. (1998) claims values of TiO_2 particulate fouling R_f^* up to $5.5 \cdot 10^{-4} \text{ m}^2 \text{ K/W}$ for liquid flowing with a velocity 0.2 m/s. Extrapolation of this result on lower velocities used in the present study (0.007-0.045 m/s) causes R_f^* one order less.

On the other hand the assumption of fouling resistance independent on shell velocity is not strict because dependence on liquid velocity is very important and R_f^* depends on liquid velocity as u^{-2} (Grandegeorge et. al., 1998). We correlated the HTC test results by second-order polynomials and used Eq. 10 to determine R_f^* dependence on velocity (see fig. 5). These values were fitted by the following function:

$$R_f^* = (1.075 \cdot 10^{-7})/u^2 \quad (24)$$

It can be seen that the inclination of the curve changes at the critical velocity value of around 0.1 m/s. Having a strong dependence from velocity asymptotic fouling resistance R_f^* sharply increases below this value. This R_f^* behavior is in good agreement with the results obtained by Grandegeorge et. al., 1998. Unfortunately our results cover a narrow range of velocities (range of $\text{Re} = 800\text{-}1600$ in Fig.3) and this is not enough to characterize R_f^* behavior precisely.

During fouling evolution tests we were watching as TiO_2 particles were accumulating in the shell. It was seen that fiber surface collects a deposit irregularly which was caused by non-uniform velocity field and random fiber distribution inside the shell. In some places the surface of fibers was relatively clean whereas deposit was seen in other places. The most significant fouling agglomerations were observed in locations where fibers were touching each other. Fig. 6 shows fouling evolution for two concentrations: 1 g/l and 5 g/l.

We used Eq. 3 to correlate experimental results and obtained Eq. 25 (for 1 g/l concentration) and Eq. 26 (for 5 g/l).

$$R_{f1}(t) = 1.336 \cdot 10^{-4} (1 - e^{-t/141.86}) \quad (25)$$

$$R_{f2}(t) = 1.336 \cdot 10^{-4} (1 - e^{-t/53.04}) \quad (26)$$

In both cases fouling develops rapidly and achieves asymptotic values in just a few hours. It can be associated with very low liquid velocities in the shell and, as it was mentioned before, low level of maximum fouling value R_f^* . The previous study (Grandegeorge et. al., 1998) showed that liquid velocity strongly influences deposition coefficient k_d value which sharply increases with a decrease of velocity below 0.1 m/s. Fig. 6 shows that concentration of foulant c_b has influence on the rate of deposit development (deposition coefficient k_d) but has no influence on its maximum value R_f^* .

We used air bubbling as the defouling procedure to clean the fiber surface from the deposit. It was seen through a transparent shell that fibers were moving and particles were being removed from the fiber surface during the cleaning. It is worth noting that defouling was not absolutely complete. Bubbles destroyed particle agglomerates and removed the main mass of deposit particles but some particles remained. The graph on Fig. 7 shows the typical overall HTC behavior during defouling. The plateau on the left part of the graph corresponds to maximum fouling state ($U = 1480 \text{ W/m}^2 \text{ K}$). It is around 75% of the initial clean value ($U_{\text{clean}} = 2020 \text{ W/m}^2 \text{ K}$). Steep HTC rises at 13:50 and 14:50 corresponds to air injecting. It should be noted that HTC sharp peaks (abrupt heat transfer intensification) are associated with the rise of mean liquid velocity during the defouling. The growth of the mean velocity was caused by local liquid velocities. They were created with bubbles flowing upward significantly faster than the liquid moving downward. It was observed that the defouling with bubbles recovered HTC to $1800\text{-}1900 \text{ W/m}^2 \text{ K}$ (90% of initial value). The evolution of the deposit after the defouling was similar to clean surface fouling evolution but it started from a higher level (the defouling was incomplete). Moreover it developed faster than the fouling of the initially clean (or mechanically cleaned) surface. It can be explained by the difference in properties of initially clean surface and hydrodynamically cleaned one. This fact was also documented by Grandegeorge et. al. (1998) who quoted that the surface cleaned by liquid velocity change collected the deposit faster than the initially clean surface. We also expect that the defouling efficiency will decrease during long-term operation.

Table 1. Overall HTCs of polymeric hollow fiber heat exchangers

Heat exchanger type	U, W/m ² K
Water-to-water heat exchanger ^a	850-1700
PP ^b	1314
HEPP3 ^c	2076
HEPEEK2 ^c	1929
Membrana 041938 ^c	1100
PES ^c	2109
PP-based hollow fiber M4 ^d	1229
PP, clean ^e	400-2100

^a (Holman, 2010); ^b Zarkadas and Sirkar, 2004; ^c Song et. al., 2010; ^d Qin et. al., 2012; ^e present study

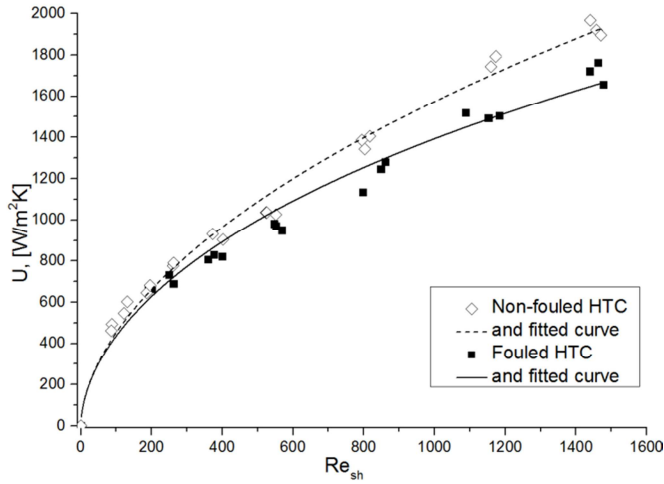


Fig. 3 Overall heat transfer coefficient versus shell-side Reynolds number for clean and fouled ($c_b = 1$ g/l) conditions.

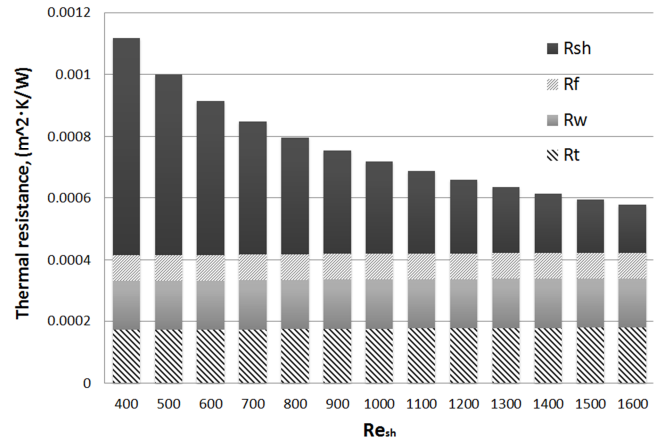


Fig. 4 Contribution of shell (R_{sh}) and tube (R_t) convective, wall (R_w) and fouling (R_f) thermal resistances to overall thermal resistance versus shell-side Reynolds number

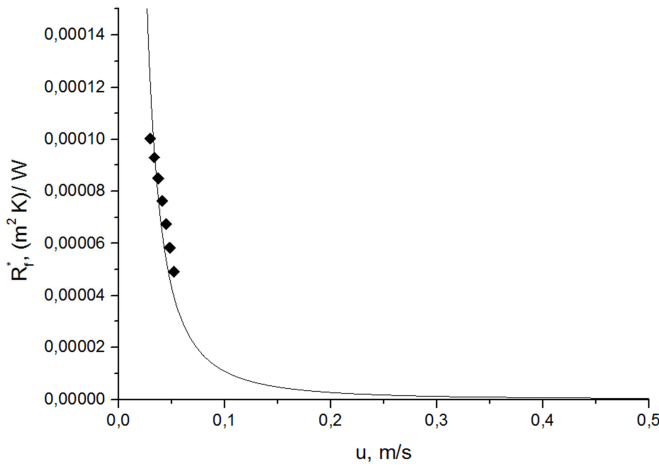


Fig. 5 Asymptotic fouling resistance versus liquid velocity

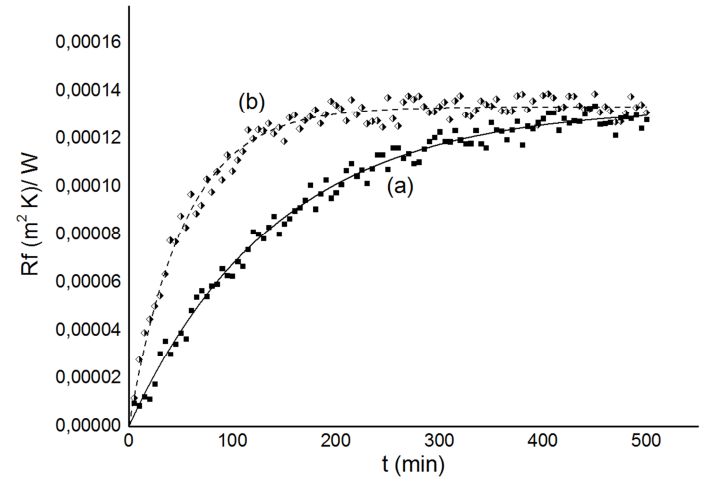


Fig. 6 Fouling evolution in time ($u = 0.04$ m/s): (a) bulk concentration $c_{b1} = 1$ g/l; (b) $c_{b2} = 5$ g/l.

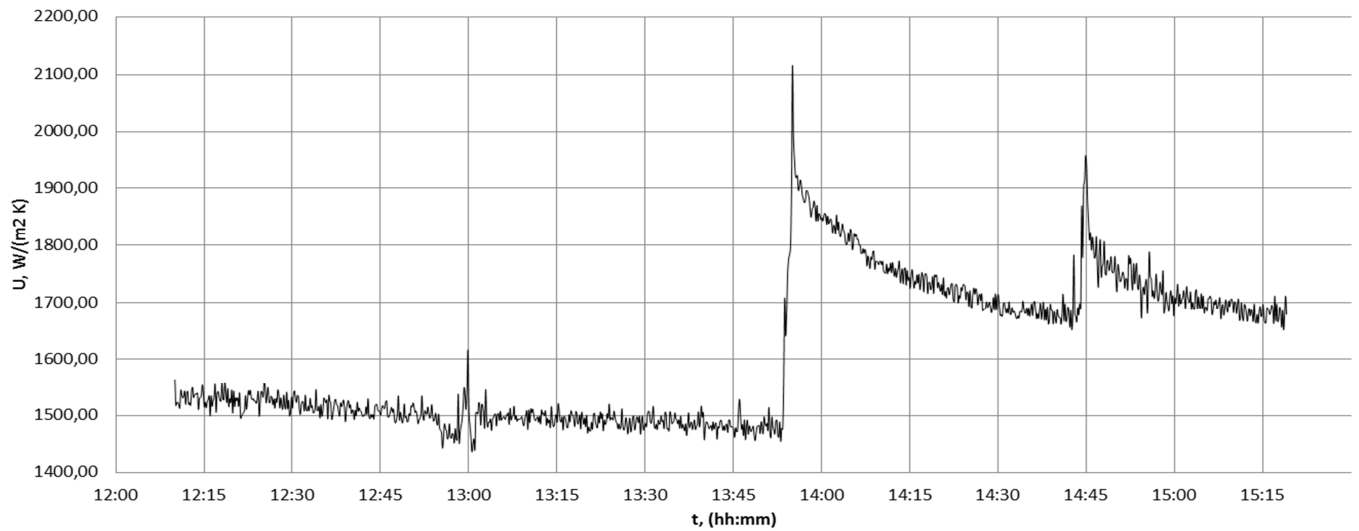


Fig. 7 HTC change during defouling procedure ($u = 0.04$ m/s, $c_b = 5$ g/l)

CONCLUSIONS

The tested polymeric heat exchanger showed high thermal performance. High values of overall HTC (up to 2100 W/m² K) were observed during the tests even at very low flow velocities (about 0.05 m/s).

Particulate fouling tests showed that polymeric fiber surface was quite resistant to fouling. The high rate of fouling evolution was compensated by low fouling asymptotic value and low deposit strength. The asymptotic values of fouling R_f^* were up to $15 \cdot 10^{-5}$ m² K/W causing about 20 % decrease of overall HTC. The high rates of deposit evolution (during hours) were associated with very low liquid velocities performed in the experiments.

The tested defouling procedure was evaluated as efficient enough. It was approved that it caused the movement of fibers and provided the particle removal from the surface. Depending on the frequency of cleaning it can recover HTC to the level of around 90%. On the other hand it requires installation of additional equipment and creates secondary maintenance costs. The expediency of its use should be determined based on economic factors.

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NOMENCLATURE

A	heat transfer area or area of cross-section, m ²
C	heat capacity rate, $Q_f \rho C_p$, W/K
C_p	specific heat, J/kg K
c_b	mass particle concentration, kg/m ³
D	diameter, m
F	LMTD correction factor, dimensionless
Gr	Grashof number, $g\beta(T_b - T_w)L^3$, dimensionless
h	heat-transfer coefficient, W/m ² K
k_a	adhesion coefficient, m/s
k_d	overall deposition coefficient, m/s
k_m	mass transfer coefficient, m/s
N	number of fibers, dimensionless
Nu	Nusselt number, hD/k , dimensionless
P	wetted perimeter, m
Pr	Prandtl number, $C_p \mu / k$, dimensionless
Q	rate of heat transfer, W
Q_f	volume flow rate, m ³ /s
R	thermal resistance, m ² K/W
Re	Reynolds number, $D u \rho / \mu$, dimensionless
t	time, s
t_c	fouling process time constant, s
T	temperature, K
ΔT	temperature difference, K
u	liquid velocity, m/s
U	overall heat-transfer coefficient, W/m ² K
λ	thermal conductivity, W/m K
μ	dynamic (absolute) viscosity, kg/m s
ρ	density, kg/m ³
Φ_d	deposition flux, kg/s m ²

Subscript

0	initial
av	average
b	bulk
f	fouling
h	hydraulic
i	inner
in	inlet
lm	logarithmic
o	outer
out	outlet
ov	overall
sh	shell
t	tube
T3	convection boundary condition (the third type)
w	wall

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