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Cooling of flue gas by cascade of polymeric hollow fiber heat exchangers

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A R T I C L E I N F O

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A B S T R A C T

Polymeric hollow fibers heat exchangers can be effectively used for flue gas cooling. These heat exchangers are made of hundreds of hollow fibers with an outer diameter of 1.3 mm and a wall thickness of 0.15 mm. Selection of material allows to work with aggressive gasses and condensed liquids without risk of corrosion or damage by chemicals. These fibers may seem to be fragile, but they can stand up the inner pressure above 100 bar at room temperature and have burst pressure of 60 bar at temperature of 80 °C. Their surface is very smooth and that positively contributes to low fouling and its capability of self-cleaning. The flowing condensate is cleaning the fibers during its operating and it carries ash particles size up to 0.22 mm. No marks of abrasion or corrosion were observed on the fibers. Due to the low inner diameter of the fibers, the internal flow is laminar and heat transfer coefficient is velocity independent and high (2317 W/m^2K). Values of overall heat transfer coefficient are over 100 W/m^2K and are dominantly determined by heat transfer on the gas side. Measured volume heat performance of the tested heat exchangers varied from 4.2 MW/m^3 to 2.2 MW/m^3.

1. Introduction

Heat exchangers are commonly made of metallic materials such as various steel grades, aluminum alloys, and copper. Heat exchangers for cooling of flue gas are made typically of stainless steel especially when cooling is connected with condensation of aggressive liquids. Recently, polymeric materials entered the scene offering several outstanding features such as being chemically stable, corrosion/erosion resistant, electrically non-conductive. Polymers are cheaper than metals and exhibit a reduced ecological footprint [1]. Corrosive resistance is the major parameter for the use of these heat transfer surfaces for cooling of flue gas [2].

Microchannels have been used mainly in heat and mass transfer applications. Hollow fiber membranes are applied in air and liquid treatments for separation processes [3,4]. The non-porous polymeric hollow fibers are used to construct heat exchangers. As demonstrated in Ref. [5], polymeric hollow fibers are an alternative to finned tubular or plate-fin heat exchangers as the tiny fibers (outer diameter about 1 mm) have high surface-to-volume ratios. Exceptional heat transfer rates are certainty and that explains why polymeric fibers have been already used with success in liquid/liquid and liquid/gas systems. The fibers must be separated to guarantee a high heat transfer rates [6]. Special attention to fibers arrangement for shell-and-tube heat exchanger was paid in Refs. [7,8]. The authors claim that tilting the fibers by an angle of 22.5° across the layers (from the parallel position) results in the increase of the

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overall heat transfer coefficient by 12.5%.

In spite of the low thermal conductivity of polymers, the thermal resistance of the wall of a fiber is very small thanks to its thinness about 0.1 mm. A hollow fiber with a diameter of around 1 mm provides a very high heat transfer coefficient (HTC) independent of the flow velocity (constant Nusselt number) because of the laminar liquid flow inside the fibers. A low flow velocity is beneficial as the pressure loss becomes more favorable.

The durability of polymeric hollow fibers was tested from two aspects. The temperature-dependent burst failure was studied for six different materials (one of them is used as material of hollow fibers in this study). All tested materials proved that they are suitable for heat exchanger applications [9]. Experimental work in laboratory of authors of this paper showed that fibers can withstand an inner pressure loading that simulates a fatigue. The tests were done according to ISO 19892:2011. The results of fatigue testing of whole heat exchangers in the form of chaotic bundles were reported in Ref. [10]. The heat exchangers showed no sign of damage after reaching one million of pressure cycles.

Polymeric hollow fibers proved their suitability for construction of heat exchangers and were already tested in several applications. Their alternative for automotive industry is described in Refs. [11,12]. The cooling of the Li-ion cells by polymeric hollow fiber heat exchangers is studied in Refs. [13,14]. The fully polymeric membrane destilation unit that combines porous and non-porous hollow fibers is described in Ref. [15]. Their fouling by wastewater was investigated in Ref. [16].

Hollow fibers in gas – liquid application are typically arranged in layers where each layer is formed by fibers with constant pitch as shown in Fig. 1. An important question arises over what the optimal pitch between the hollow fibers should be so that the cooling performance of the heat exchanger in W/m^2 is maximized. The cooling performance is scaled by the active surface area (m^2) of the heat exchanger i.e., an area through which the air flows perpendicularly to the fibers. Consider a bank of parallel fibers arranged in an in-line square array with uniform pitches between two adjacent fibers in the horizontal and vertical direction. If the fibers are too few, the surface area of the heat exchanger is simply wasted because the fibers are overly distant from each other. Significant difference in design is for heat exchangers (HX) with natural flow of gas [17] and for forced convection. Comprehensive experimental and numerical study of air natural convection on ceiling cooling unit is in Ref. [18] where question of optimal geometrical configuration is discussed as well.

Each application of HX of this type require different spacing of fibers and different number of fiber layers to get optimal balance between heat transfer, size of heat transfer surface a performance on one side and pressure gas losses on the other side. The thermal performance (the Nusselt number) and the pumping power (the pressure drop) are among the most important parameters of optimization. Both parameters are typically calculated using well-established formulas [19,20] referenced in many handbooks [21–24]. The formulas were constructed from experimental data, in which rigid steel wires and tubes were used. With the emergence of flexible polymeric fibers, the applicability of the original formulas is opened to doubt. Pliable HXs wildly interact with the flow in the form of intrinsic high-amplitude vibrations, which affects heat transfer as well as pressure conditions. Four types of flow-induced vibration mechanisms are recognized in the cross-flow over a bank of tubes: turbulent buffeting [25], acoustic resonance [26], vortex-shedding [27] and fluid-elastic instability [28]. Experimental work in laboratory of authors of this paper showed for some configurations pressure losses on flexible fibers about 20–50% higher than predicted by models for rigid structure. Effect of fiber vibration is subject of study.

Pressure losses inside heat exchanger is important operating parameter. A polymeric hollow fiber with inner diameter of 1.0 mm can be classified as a microchannel [29]. According to Ref. [30], only a few solutions of the laminar flow forced convection problem and experimental investigations are available in the literature with some variations of the associated thermophysical properties. Variations of the density, thermal capacity and thermal conductivity have no significant influence. On the other hand, the viscosity of liquids has a strong dependence on viscous forces that play an important role in the microscale. For example, water viscosity changes more than twice for a temperature change from 20 °C to 60 °C [31]. This example shows that the variation of the fluid viscosity cannot be neglected in the analysis of micro-flows.

The role of surface roughness on fluid flow and heat transfer in micro-channels has been emphasized by several authors. According to Ref. [32], tubes can be considered smooth or rough depending on roughness size. The authors proposed the following equation to estimate the boundary that subdivides the flow in smooth and rough channels:

$$k_s \;/\; r_i = 5 \;/\; \left(1.41 \,*\, Re^{0.5}\right) \tag{1}$$

Fig. 1. Typical configuration of heat transfer surface from polymeric hollow fibers for gas-liquid HX.

where k_s – height of roughness, r_i – microchannel radius, Re – Reynolds number. This equation correlates well with [30], which noted that for Re ~ 2,000, the relative roughness that corresponds to the boundary between the smooth and rough channels is about 0.08.

The Nusselt number is a dimensionless characteristic of convective heat transfer. It is a ratio of convective to conductive heat transfer through the fluid layer. A laminar flow is characterized by constant Nusselt number and for circular tube it is close to 4 [11].

A formula of Nusselt number for laminar flow in a circular tube was introduced in Ref. [33]. The authors are using the Hickman's and Hsu's approach to obtain the formula for the mean Nusselt number of the tube

$$Nu = \frac{Nu_w + 48/11}{1 + Nu_w * 59/220} + \left(0.0499 - \frac{0.06487}{1 + exp\left(\frac{Nu_w + 5.37935}{2.17887}\right)}\right)\frac{4Gz}{\pi} \tag{2}$$

where Gz is the Graetz number which characterizes the laminar flow in a conduit, and Nu_w is the external resistance [33].

2. Experiment configuration

2.1. Heat transfer surfaces

Polymeric hollow fibers are used as heat transfer surface in this study. Hollow fibers are made by extrusion when tube exiting extrusion head is elongated in semi-solid state and its diameter shrinks while proportion between diameter and wall thickness is constant. Fibers can be made from wide range of polymers from polypropylene (PP) to polyether ether ketone (PEEK). Operation temperature of polymeric heat exchangers is from 80 °C for PP to 230 °C for PEEK. Inner and outer surface of hollow fibers is very smooth because of technology of production. Smoothness of the surface causes good fouling resistance [34]. An example of extruded fiber is shown in Fig. 2. The polymeric fibers are very flexible and the photo in Fig. 2 right shows the wall of polyamide fiber broken in liquid nitrogen. Fibers are typically extruded with outer diameter from 0.3 to 1.3 mm and thickness of the wall from 5% to 25% of the outer diameter.

Our studies of available hollow fibers show that their internal surface is very smooth and has an absolute roughness less than 1 μm (see Fig. 2). For 1 mm inner-diameter hollow fibers it gives a value of relative roughness less than 0.001, so the hollow fiber can be considered to be a hydrodynamically smooth tube. Such roughness does not influence fluid flow and heat transfer.

As hollow fibers can be considered smooth microchannels, the conventional Poiseuille number Po = 64 can be used to calculate pressure drop. On the other hand, variable liquid viscosity along the fiber length needs to be considered. Thus, pressure drop can be calculated as

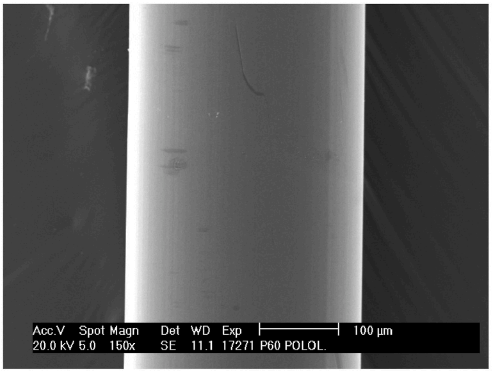
$$\Delta p_{it} = 128 * \mu_{av} * l * Q_{f,t} / \left(\pi * D_i^4 * N\right) \tag{3}$$

where $Q_{f,t}$ is a volumetric flow rate through the tube side (fibers), l is length of fibers, D_i is internal diameter, N is number of fibers and μ_{av} is average value of viscosity along the fiber.

Based on the expected temperatures of the flue gas (in this study), hollow fibers from polyamide were used for heat transfer surfaces. Polyamide itself is name for wide range of materials, the most frequent in technical applications is PA6 and PA11. Fibers in this study were extruded from PA 612 (Zytel 6159). PA 612 is a grade of long-chain polyamides (LCPA) flexible polymer that combines heat, chemical, and hydrolysis resistance. The various grades of LCPA are designed to meet specific performance needs, including high temperature resistance and stiffness to lower flue and gas permeability.

PA 612 is easily processable by extrusion and by followed drawing of melt to required dimension of hollow fiber. Melt temperature range by extrusion is 230–250 °C. Mechanical and thermal properties of the used PA 612 are listed in Table 1.

The size of fiber used in this study (outer diameter 1.3 mm and wall thickness of 0.15 mm) can lead to belief that heat transfer surface is delicate but opposite is true. Fig. 3 shows polyamide fiber of outer diameter 1.3 mm and wall thickness of 0.15 mm after burst



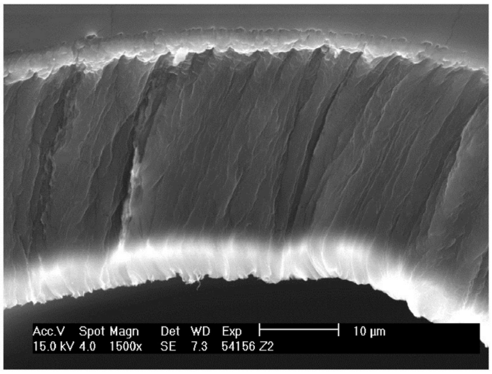


Fig. 2. Hollow fiber after extrusion in electron microscope (left), broken wall of fiber (right).

test. This fiber can withstand internal pressure over 100 bar at room temperature and pressure of 60 bar at temperature of 80 °C.

2.2. Polymeric hollow fiber heat exchangers

Three identical heat exchangers were made and are used in cascade configuration. Heat exchanger has cross section 200 × 200mm. Heat transfer surface of one heat exchanger is formed by 402 fibers configured in 6 layers each with 67 fibers (see Fig. 4). Pitch of fibers is 3 mm both in direction of flow and in direction perpendicular to the flow. Outer diameter of the fibers is 1.3 mm and inner diameter is 1.0 mm. Total heat transfer surface on gas side is 0.33 m^2.

Computed (equation (2)) and measured pressure losses of the three heat exchangers are shown in Fig. 5. The computed pressure drop is for flow inside fibers only without flow inside manifold. The measured values are higher because include pressure losses of manifolds.

2.3. Configuration of experiment

Configuration of laboratory setup with main hardware parts and measured points is shown in Fig. 6. Three heat exchangers (HX) described above (see Fig. 4) are located in wind channel. Each HX is cooled by water. Input flow and water temperature is measured. Output temperature is measured. Based on equal pressure loss for each HX, it is expected that coolant flow is equal for all HX. Amount of condensed liquid is measured for all HX. Coal boiler is source of flue gas. The boiler is computer controlled on the side of intake air and coal supply. Temperature and humidity of flue gas at the entry to the first HX is measured. Temperature of gas is measured behind each HX and gas humidity is measured after third HX. For measurement of temperatures is used Pt100 (1/3 DIN Class A precision, Omega Engineering Inc., Norwalk, CT, USA) sensors with an accuracy of ±0.1 °C. Water flowmeter (ifm electronic GmbH, Essen, Germany) has precision of 1.5% of the flowrate, i.e., maximum value of ±18 l/h. Humidity probe (Ahlborn Mess-und Regelungstechnik GmbH, Holzkirchen, Germany) is calibrated with precision of ±1.3%. Pressure difference before first HX and after third HX was measured by differential manometer (GHM GROUP - Greisinger, Regenstauf, Germany) with precision of 1.5 Pa. Photo of experimental setup is shown in Fig. 7.

3. Results

Typical laboratory experiment with described configuration last 5 h. First hour was necessary for setting and stabilization of all parameters. The remaining 4 h of test provided stable results. The measured parameters are summarized in Table 2. Total cooling water flow is 1380 l/h. Pressure difference between first and last HX was measured and is 40 Pa. Heat exchangers used in this study work in all cases with laminar flow inside the fibers with Reynolds number in range from 20 to 200. For laminar flow is constant Nusselt number and heat transfer coefficient inside the tubes is velocity independent. Heat transfer coefficient (HTC) is high even for low velocity of liquid inside the fibers. Computed HTC for fibers of inside diameter 1.0 mm and water temperature of 11 °C is 2317 W/m^2K. Flue gas temperature distribution along the tunnel is shown in Fig. 8.

Table 3 summarize performance and heat transfer details of the tested HXs. The overall heat transfer coefficient is influenced by the heat transfer conditions on the outer surface of fibers. Thermal resistance caused by heat transfer inside the fibers and thermal resistance of the polymeric wall is identical for all HXs. The main thermal resistance, over 80%, is on the outer surface. It should be noticed that heat transfer on the outer surface is caused both by forced convection from flue gas and by condensation of the liquid from cooled gas.

Additional tests with single HX were done with elevated gas temperatures and with heavy fouling conditions. Five hours test was done with flue gas temperature of 160 °C and relative humidity of 2.3%. The output gas temperature was 101.5 °C and relative humidity of 5.3%. Increased water flowrate of 1080 l/h was used for tests. Intake water temperature was 11.1 °C and output temperature was 15.0 °C. This gives HX heat performance of 4890 W. Amount of condensed liquid was 405 ml/h. Boiler in the experiment with elevated temperature did not work in optimal regime and large amount of solid particles was carried by the gas flow. Heat exchanger in this boiler regime was seriously fouled by solid products of burning coal after 5 h of test. Pressure loss on the gas side increased by 34% during the experiment. It was interesting that the pressure losses did not grow continuously. There were observed some stem falls of pressure loss value. Explanation was found after test when removed the HX from tunnel. Flexible fibers shake in gas flow. If the layer of ash on the surface is thicker, it becomes unstable, and segments of ash layer are taken away by gas. View of the fibers with layer of ash is shown in Fig. 9. It should be noticed that the ash layer was easily removed from the fibers by shaking and by pressurized air.

Table 1
Mechanical and thermal properties of the used PA 612.

Mechanical properties	
Tensile Strength, Yield	65.0 MPa
Elongation at Yield	4.3%
Tensile Modulus	2.40 GPa
Charpy Impact, Notched	5 kJ/cm^2 (23.0 °C)
Thermal properties	
Melting Point	216 °C
Temperature of deflection under load 0,45 MPa	150 °C

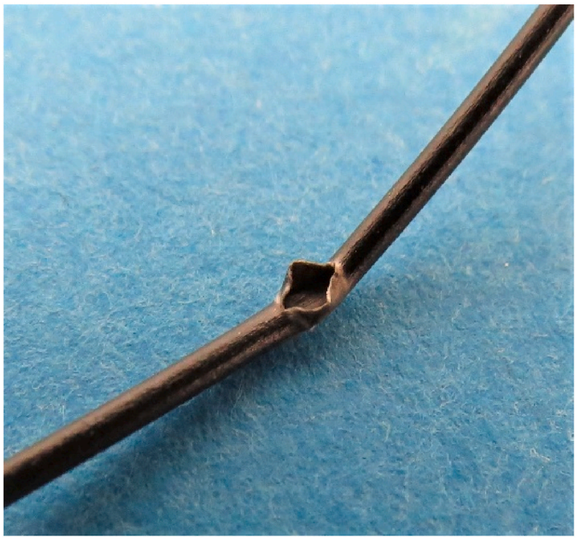


Fig. 3. Polyamide capillary made of PA 612 after burst pressure test (at temperature of 80 °C and burst pressure of 60 bar).

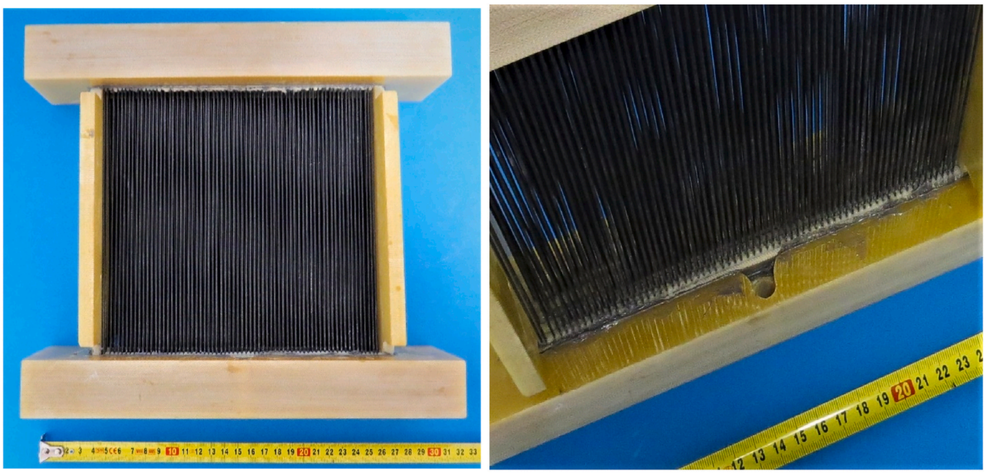


Fig. 4. Heat exchanger for cooling of flue gas, front view (left), outlet for condensed liquid (right).

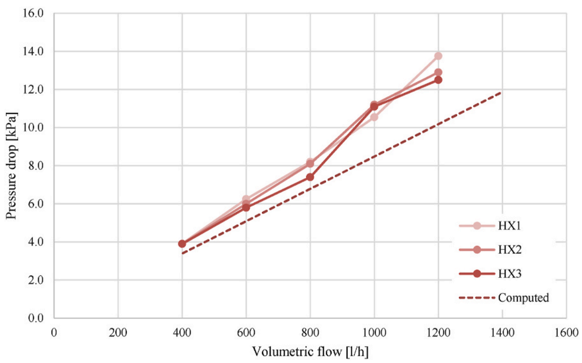


Fig. 5. Computed and measured pressure losses of the used heat exchangers.

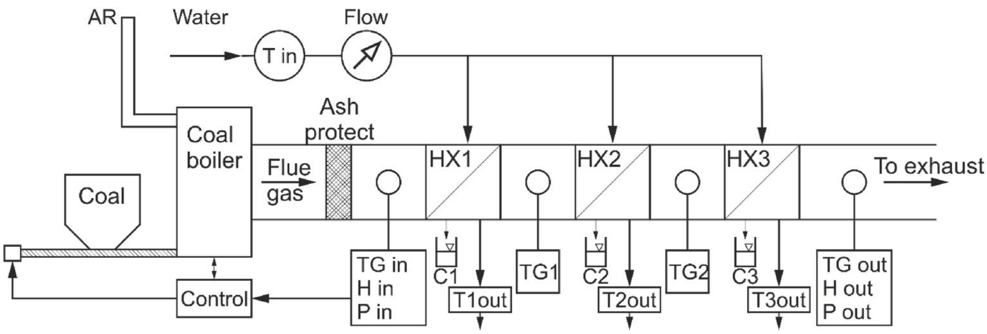


Fig. 6. Scheme of experiment configuration, TG is gas temperature, H is relative humidity, P is gas pressure, C is quantity of condensed liquid, AR is air intake.

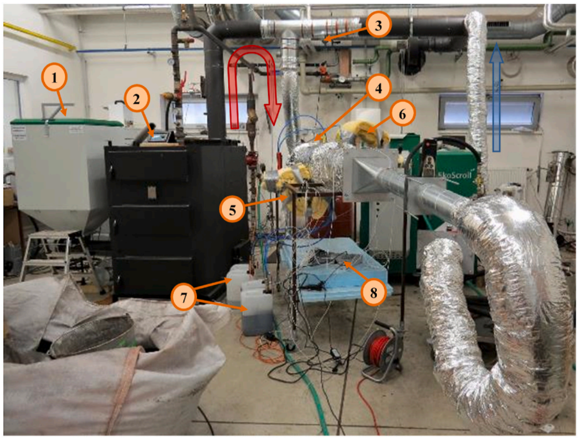


Fig. 7. Experimental setup, 1 – coal bin with screw feeder, 2 – coal boiler, 3 – exhaust gas bypass, 4- wind tunnel with heat exchangers, 5 – insulated cooling water inlet, 6 – insulated water outlet, 7 – condensed liquid, 8 – measuring cards.

Table 2
List of measured parameters in stable regime.

POSITION (along tunnel)	Gas temperature TG (°C)	Gas relativeHumidity H (%)	Water temperature in (°C)	Water temperature out (°C)	Condensed liquid C (ml/h)	HX performance (W)
Gas prior HX1	126.0	3.2	–	–	–	
HX1	–	–	11.0	17.3	420	3364
Between HX1 and HX2	90.2	–	–	–	–	
HX2	–	–	11.0	15.6	850	2456
Between HX2 and HX3	64.4	–	–	–	–	
HX3	–	–	11.0	14.3	900	1762
Behind HX3	44.5	38.8	–	–	–	

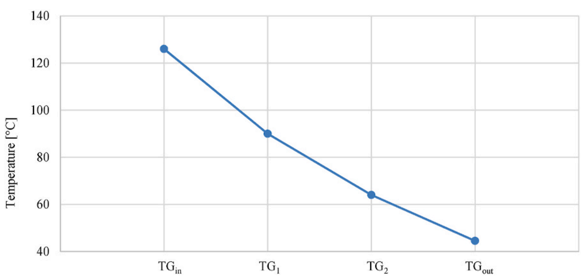


Fig. 8. Flue gas temperature distribution along the tunnel.

Table 3
Performance and thermal resistance distribution at heat exchangers.

	Q	U_o	h_i	h_o	R_i	R_w	R_o	R_{tot}	R_i/R_{tot}	R_w/R_{tot}	R_o/R_{tot}
	[W]	[W/m^{2}K]	[W/m^{2}K]	[W/m^{2}K]	[K/W]	[K/W]	[K/W]	[K/W]	[%]	[%]	[%]
HX1	3364	105	2317	122	0.00171	0.00236	0.02492	0.02899	5.9%	8.1%	86.0%
HX2	2456	119	2317	141	0.00171	0.00236	0.02162	0.02569	6.7%	9.2%	84.2%
HX3	1762	131	2317	158	0.00171	0.00236	0.01924	0.02331	7.3%	10.1%	82.5%

Rem: Q is thermal performance, U_o is overall heat transfer coefficient, h_i is heat transfer coefficient inside fibers, h_o is heat transfer coefficient at outer surface of fibers, R is thermal resistance.

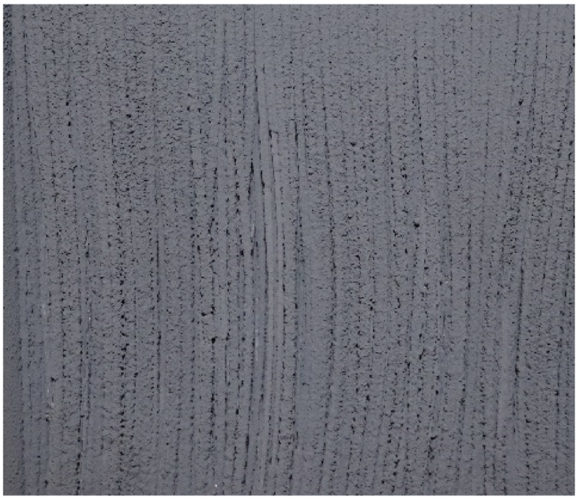


Fig. 9. Detail of polyamide HX after test with heavy fouling.

4. Comparison of results with published data

The pressure drop is an important parameter of each heat exchanger as it highly influences its operational costs. The pressure drop on the flue gas was compared to the pressure drop that was published in Ref. [35] that presents the pressure drop of heat exchanger of a gas-fired boiler. Fig. 10 shows the reported data (marked blue), their fitting (orange), and the pressure drop of the PHFHE (red). Even though the coal boiler was used for measuring with PHFHE, it can be observed that the cascade of three PHFHE achieved comparable pressure drop.

Fig. 11 shows a comparison of flue gas temperature in dependance on the heat transfer area. The data marked orange were presented in Ref. [36] and were measured for the inlet temperature of 127.7 °C with the flue gas flowrate of 199.5 kg/h. The inlet cooling water temperature was 9.7 °C with the flowrate of 274.5 l/h. The boiler used was the oil-fired boiler. The data marked green were achieved by Ref. [37] using the coal boiler with inlet temperature of flue gas 149.5 °C. Flue gas flow rate was 185.7 kg/h. The inlet cooling water temperature was 31 °C with the flowrate of 545.3 l/h. All reported results were achieved by using a cascade of 6 heat exchangers made of U-bend tubes. The data marked red were obtained for PHFHE at the inlet temperature of 126 °C and the flue gas flow rate of 340 kg/h. The inlet cooling water temperature was 11 °C with the flowrate of 1380 l/h. It can be observed that the cooling

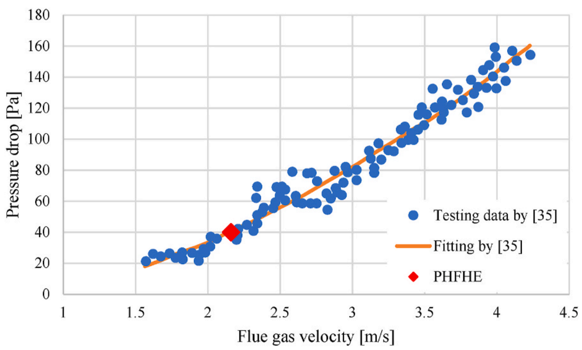


Fig. 10. Comparison of measured pressure drop for PHFHE and reported data in Ref. [35] in dependance on flue gas velocity.

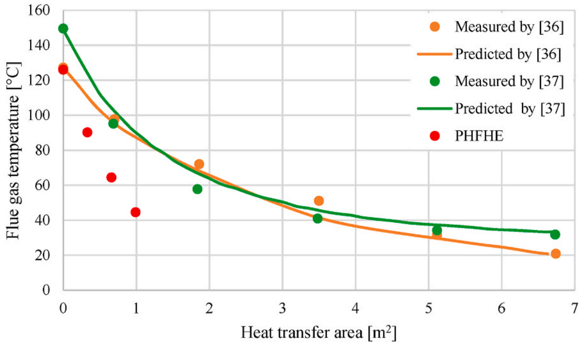


Fig. 11. Comparison of measured flue gas temperature in dependance on heat transfer area: PHFHE, data reported by [36,37].

velocity of PHFHE is larger than for the reported data in Refs. [36,37]. Although, the much larger cooling water flowrate was used the pressure drop of each of three heat exchangers was only 5 kPa due to the large number of the fibers. Also, the flue gas flow rate used for PHFHE was approximately 70% higher. That proves that the PHFHE are effective and durable heat exchangers which provide such benefits as corrosion resistance, low weight, easy shaping, and machining.

5. Discussion

The use of polymer heat transfer surfaces for cooling of flue gas seems to be risky. In fact, some problems can appear only if liquid cooling stops. Numerical simulation shows for conditions of measurement, with flue gas temperature of 160 °C, surface temperature only 31 °C. This effect is caused by fact that HTC inside fibers is about ten times higher than HTC outside and the fiber wall thickness is very small.

The boiler in experiment burned coal and no filtering was used at the entrance to the HXs. Experiment with gas temperature of 125 °C was not influenced by fouling of HX. The measured parameters were almost constant during 4 h of measurement. The used HX with polymer fibers proved much better fouling parameters than classic metal HXs in the laboratory study comparing fouling parameters for standard test with Ashrae 52.1 test dust is published in Refs. [32,38]. The extruded fibers can be easily cleaned by shaking or by pressurized air.

All of the three heat exchangers in cascade worked in regime of condensation of liquid from gas. Vertical position of fibers is suitable for condensate flow. Details about influence of fiber orientation and gas velocity on condensation and heat transfer is described in Refs. [39,40] and. An interesting feature of the flowing condensate is the cleaning effect on heat transfer surface. This effect was not measured quantitatively but condensed liquid carried small ash particles. We assume that cleaning effect of condensate helped to hold stable heat transfer conditions and almost constant hydraulic resistance on gas side during all experiments with cascade of heat exchangers.

Experiment with single HX in heavy fouling conditions showed self-cleaning ability of heat transfer surfaces. The cleaning of heavily fouled surfaces is caused by smooth fiber surface and by vibration of fibers in cross flow. Interesting finding is that HX in heavy fouling conditions with layer of ash on the surface lost performance only by about 16%. Explanation of only small performance drop is in fact that ash layer is not dry. The condensed liquid keeps the ash layer on fibers wet with elevated thermal conductivity.

6. Conclusion

Heat exchangers with polymeric hollow fibers can be effectively used for cooling of flue gas and for condensation of liquid phase from the products of burning. Polymeric fibers can withstand high gas temperatures because in configuration with internal liquid cooling and thin polymeric wall is the outside temperature of the fibers low. Polymeric material allows to work with aggressive gasses and condensed liquids without risk of corrosion or damage by chemicals.

Smooth surface of extruded fiber is advantageous for fouling resistance. Cleaning the dry surface by shaking or by pressurized air is easy. Experiments detected positive influence of condenser flow on vertical fibers on cleaning the surface from small particles of ash. Collected condensed liquid carry ash particles size up to 0.22 mm. Experiment with heavy fouling of the heat transfer surface showed ability of self-cleaning of the smooth and flexible fibers. The used capillary from polyamide PA 612 do not report any marks of abrasion or corrosion.

Internal flow inside the fibers is laminar. Heat transfer coefficient for laminar flow is velocity independent and is high. HTC inside the fibers used in experiments (diameter 1.0 mm) was 2317 W/m^{2}K. Laminar flow inside fibers cause linear increase of pressure drop with increasing flowrate. This effect is advantageous in comparison with tubes of bigger diameter and turbulent flow where hydraulic resistance grows with square of velocity.

Measured volume heat performance of the tested HX was 4.2 MW/m^3 for the first HX, 3.0 MW/m^3 for the second HX and 2.2 MW/m^3 for the third HX. Measured values of overall heat transfer coefficient are over 100 W/m^{2}K and are dominantly determined by heat transfer on the gas side.

Credit author statement

Miroslav Raudensky: Conceptualization; Formal analysis; Funding acquisition; Investigation; Methodology; Resources; Supervision; Visualization; Roles/Writing - original draft. Tereza Kudelova: Conceptualization; Data curation; Formal analysis; Investigation; Methodology; Resources; Validation; Visualization; Writing - review & editing Erik Bartuli: Data curation; Formal analysis; Validation; Visualization. Tereza Kroulikova: Data curation; Investigation; Visualization. Ilya Astrouski: Data curation; Visualization.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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