BRNO UNIVERSITY OF TECHNOLOGY
VYSOKÉ UČENÍ TECHNICKÉ V BRNĚ

FACULTY OF MECHANICAL ENGINEERING
FAKULTA STROJNÍHO INŽENÝRSTVÍ

HEAT TRANSFER AND FLUID FLOW LABORATORY
LABORATOŘ PŘENOSU TEPLA A PROUDĚNÍ

OPTIMIZATION OF HEAT TRANSFER SURFACES OF HEAT EXCHANGERS
OPTIMALIZACE TEPLOSMĚNNÝCH PLOCH TEPELNÝCH VÝMĚNÍKŮ

SHORT VERSION OF DOCTORAL THESIS
TEZE DIZERTAČNÍ PRÁCE

AUTHOR
AUTOR PRÁCE
Ing. Erik Bartuli

SUPERVISOR
ŠKOLITEL
prof. Ing. Miroslav Raudenský, CSc.

BRNO 2019

KEY WORDS
hollow fibers, radiator, heat exchanger, optimization, intensification

KLÍČOVÁ SLOVA
dutá vlákna, radiátor, tepelný výměník, optimalizace, intensifikace

CITATION
BARTULI, Erik. *Optimization of heat transfer surfaces of heat exchangers*. PhD thesis. Brno: Brno University of Technology, Faculty of Mechanical Engineering, Heat transfer and fluid flow laboratory, 2019. 91 p. Supervisor: prof. Ing. Miroslav Raudenský, CSc.

CONTENT

INTRODUCTION

Devices intended to transfer heat from one fluid to another are commonly called heat exchangers. Different thermal processes such as temperature change, evaporation, boiling, condensation, melting and finally more complex combined processes can occur in heat exchangers. Number of fluids involved in these processes can be more than two, for example, heat can be transferred from one body to another several bodies or conversely from several bodies to the one. These bodies which give and receive heat are called the working fluid. Working fluids may be either gases or liquids.

Heat exchangers are widely used in various industrial manufacturing processes, building heating and conditioning systems, in engines with internal and external combustion, in refrigerating system and in many other fields. The most frequent material from which these devices are manufactured is metal (copper, aluminum and so on). One of the most common type of metal heat exchangers is a heat exchanger made of copper coil pipe with aluminum fins. But these days exist a lot of heat exchangers which are made of other materials such as graphite and polymers. Polymeric heat exchangers become more and more popular in different engineering fields because of their important features to compare with metal heat exchangers.

Polymeric hollow fiber heat exchanger (PHFHE) is a heat exchanger which uses small hollow fibers with outer diameter less than 1.5 mm as a heat transfer aria. The wall thickness of this polymeric fibers is approximately 0.1 mm. Small diameter of the hollow fibers and it thick wall reduces the influence of polymeric thermal resistance on heat transfer performance of all device. Also, these heat exchangers are very compact and have a big heat transfer area to compare with its dimensions.

For improving a heat transfer performance of heat exchangers exist a lot of different methods. Usually they are connected with an optimization of heat transfer surfaces of heat exchangers.

1 INVESTIGATION OF HEAT PULSES INFLUENCE ON METAL HEAT EXCHANGER

The purpose of this work was a study of heat pulses influence on the heat exchanger efficiency reduction. A thermal cycling tests of a metal heat exchanger were carried out to simulate its actual operating conditions and find out a potential deterioration of contact between fins and the coil pipe. Using achieved data to find a way for design optimization and a method of fins fixing.

In this work, the commercial fancoil that is used in building air-conditioning systems was studied. Overall view of a metal finned tube heat exchanger placed under the fancoil external casing is presented in Fig. 1. Heat exchanger consists of 3 rows of copper tubes with an external diameter of 10 mm. Aluminum fins were pressed on copper tubes with the fin pitch of 2 mm. The overall heat exchanger dimensions were 250x250x65 mm. A fan was installed at the bottom of the heat exchanger to provide a directed air flow throw the radiator. According to technical characteristics of the heat exchanger, specified by the manufacturer in the device certificate, in the heating mode this heat exchanger has a heat transfer rate of 1000 W at the water inlet temperature to the heat exchanger equal to 50 °C, the water flow rate through the heat exchanger equal to 136 l/h and the ambient air temperature equal to 20 °C.

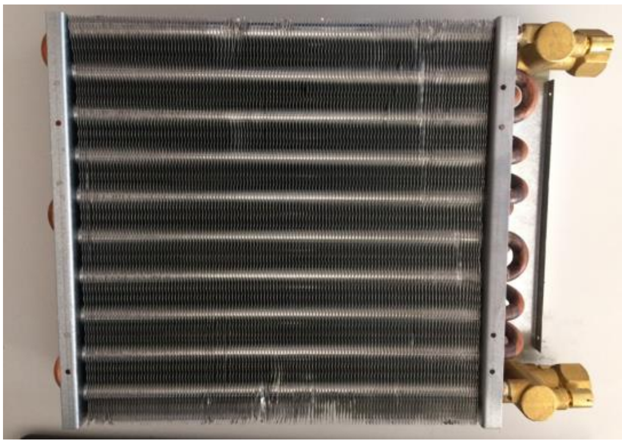

Fig. 1 Overall view of the metal finned tube heat exchanger inside the fancoil

The working body of the fancoil is made of two different materials: the radiator coil pipe is made of copper and fins are made of aluminum plates. The thermal expansion coefficient for copper is $16{,}6*10^{-6}$ °C^{-1} and for aluminum is $22{,}2*10^{-6}$ °C^{-1}. It means that during cooling and heating these two metals will expand differently. It can impair the contact between fins and copper coil pipe of the heat exchanger. This can lead to increase in thermal resistance in the contact area between fins and pipe, which will decrease the heat transfer rate of the heat exchanger.

To evaluate the influence of thermal stress on the fin-coil contact the following experiments were conducted:

1) heat transfer rate of the new fancoil sample was measured;
2) thermal cycle tests were conducted;
3) heat transfer rate of tested fancoil was measured to compare with initial values.

All the tests were conducted in the calorimetric chamber. The chamber allows precise control and measurement of all parameters (temperature, flow rate, etc.) and simulate a real working conditions of heat exchangers.

1.1 HEAT TRANSFER RATE MEASUREMENT OF THE FANCOIL

At the first step of experiments with the fancoil, tests to measure its heat transfer rate in the calorimeter chamber were performed for its further comparison with the characteristics declared by the manufacturer.

For testing in heating mode, the inlet water temperature of the heat exchanger was 50 °C, the water flow rate through the heat exchanger was 136 l/h and the ambient air temperature was 20 °C. For testing in cooling mode, the inlet water temperature of the heat exchanger was 7 °C, the water flow rate through the heat exchanger was 136 l/h and the ambient air temperature was 27 °C.

The measured data in comparison with the certificate data of the device are presented in Tab. 1.

Tab. 1 Heat transfer rate comparison between measured data and manufacturer's data.

Heat transfer rate, heating mode, certificate	Measured heat transfer rate, heating mode	Heat transfer rate, cooling mode, certificate	Measured heat transfer rate, cooling mode
1000 W	942 W	790 W	602 W

The table shows that the heat transfer rate in heating mode agrees with the manufacturer data, however, in cooling mode, these two values had a difference of 24%. This is primarily because the manufacturer had his data for a relative air humidity of 45%, but in our measurements the air humidity was 13-14%, which leads to a decrease heat transfer rate of heat exchanger.

1.2 THERMAL CYCLE TESTS OF METAL HEAT EXCHANGER

The second stage of experimental research was to conduct thermal cycle tests with switching between heating and cooling modes. The purpose of these tests was to determine the effect the useful life of a copper heat exchanger with aluminum fins on its thermal performance.

For providing a life-tests a special stand was developed and assembled. This stand allows to carry out thermal cyclic tests. One cycle consists of a quick heat exchanger heating and then its rapid cooling. Duration of one cycle is regulated by the water flow rate. Experimental setup allows to make tests at pressures up to 10 bar.

The temperature plot of resistance thermometer on the outer surface of the radiator for several cycles is shown in Fig. 2. The heat exchanger was heated up to the temperature of 76 °C and then rapidly cooled to a temperature of 17 °C. Duration of one cycle was about 35 seconds. The pressure in the system was equal to atmospheric. To identify the effect of different thermal expansion coefficients on the heat exchanger heat transfer rate 100000 heating and cooling cycles of the metal heat exchanger were carried out.

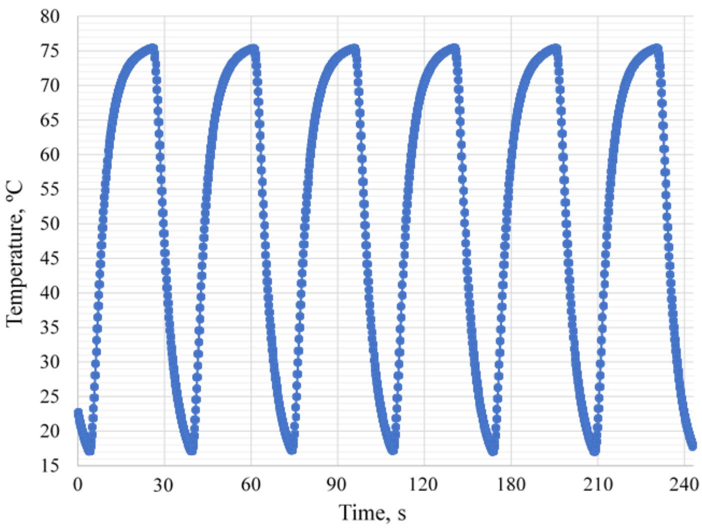

Fig. 2 Thermal cycles of tested heat exchanger

1.3 HEAT TRANSFER RATE MEASUREMENT OF THE FANCOIL AFTER THERMAL CYCLE TESTS

The third stage of the study was to conduct tests to re-measure the power of the heat exchanger using a colorimetric room in order to find out how thermal cycles, typical for actual operating conditions, influence on operating characteristic of the metal heat exchanger. The obtained data are presented in Tab. 2.

Tab. 2 Comparison of heat transfer rate measurements before and after thermocycles.

Measured heat transfer rate, heating mode before cycling	Measured heat transfer rate, heating mode after cycling	Measured heat transfer rate, cooling mode before cycling	Measured heat transfer rate, cooling mode after cycling
942 W	945 W	602 W	593 W

The experimental data showed that the thermal characteristics of the fancoil after thermocycling remained the same under measurement accuracy. According to this data can be concluded that the contact between aluminum fins and the copper coil pipe of the heat exchanger created by pressed method remained the same and the longtime of using does not have influence on heat exchanger. Since the contact between the fins and the tube has not changed, therefore, the occurrence of additional vibration of finning is excluded. Due to this fact there is no need to conduct noise tests.

2 OPTIMIZATION OF AIR-WATER PHFHE

Despite all the advantages of metal heat exchangers, there fields of application where due to several restrictions (weight, aggressive environment, impossibility of shaping, etc.) metal heat exchangers can’t be used. For such applications heat exchangers made of non-metal material play significant role. Most common materials for non-metal heat exchangers manufacturing are polymers and graphite.

In this work will be considered polymeric heat exchangers which are becoming more and more popular in different engineering fields, because of their advantages compared with metal heat exchangers.

Polymeric hollow fiber heat exchangers [1] described and investigated in this work are the type of polymer heat exchanger. Using thin-wall polymeric hollow fibers as heat exchanger tubes was first proposed by Zarkadas & Sirkar [2] as a new type of heat exchanger for lower temperature/pressure applications. This heat exchanger utilizes polymeric microchannels as the heat transfer surface. The outer diameter of these microchannels is smaller than 1.5 mm and the wall thickness is about 0.1 mm. The heat exchanger is made of hundreds of such fibers, which result in a very large heat transfer area compared to the size of the entire heat exchange Fig. 13.

Fig. 3 Overall view of a PHFHE with a straight hollow fiber bundle

2.1 HEAT EXCHANGER BASED ON THREE HOLLOW FIBER BUNDLES

A PHFHE made of twisted hollow fiber bundle with chaotic and semi-chaotic structure (shown in Fig. 4) was designed and manufactured in Heat Transfer and Fluid Flow Laboratory. Compared with straight hollow fibers bundels (presented in Fig. 3) this kind of bundle is more efective in water-air application becouse fibers can uniformly fill the volume of heat exchanger ensuring a homogenium hydraulic resistanse in the cross section. It means that air passes through all hollow fibers and whole heat transfer surface is involved in heat transfer process.

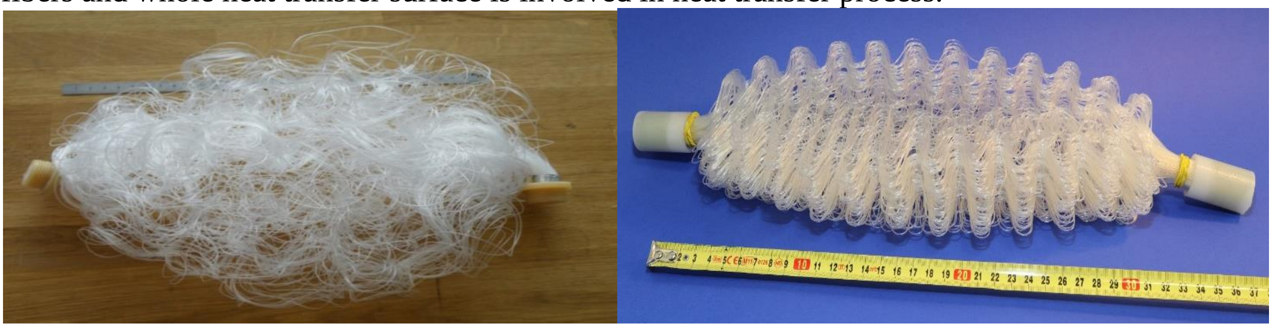

Fig. 4 Overall view of polymeric hollow fiber bundle

with chaotic (left) and semi-chaotic (right) structure

Experimental research of natural convection in PHFHE based on polymeric hollow fibers bundle with chaotic structure, also called twisted hollow fibers bundle, is presented in this work. Overall view of first heat exchanger prototype is shown in Fig. 5 (left). Three twisted hollow fiber bundles with chaotic structure were placed inside the duct. Distance between bundles was equal to 250 mm. From the both sides of each hollow fiber bundle a metal net was placed to prevent bundles from dipping and to uniformly distribute the hollow fibers over the cross section of the heat exchanger. A cross-section area of the heat exchanger was equal to 0.044 m^2 and height was 1.2 m. Resistance thermometers Pt100 RTD were used to measure the temperature distribution inside the heat exchanger. Sensor 1 was placed in the inlet of the heat exchanger. Sensors 2, 3 and

4 were placed 70 mm above each hollow fiber bundle. Top view of three hollow fiber bundles heat exchanger is shown in Fig. 5 (right).

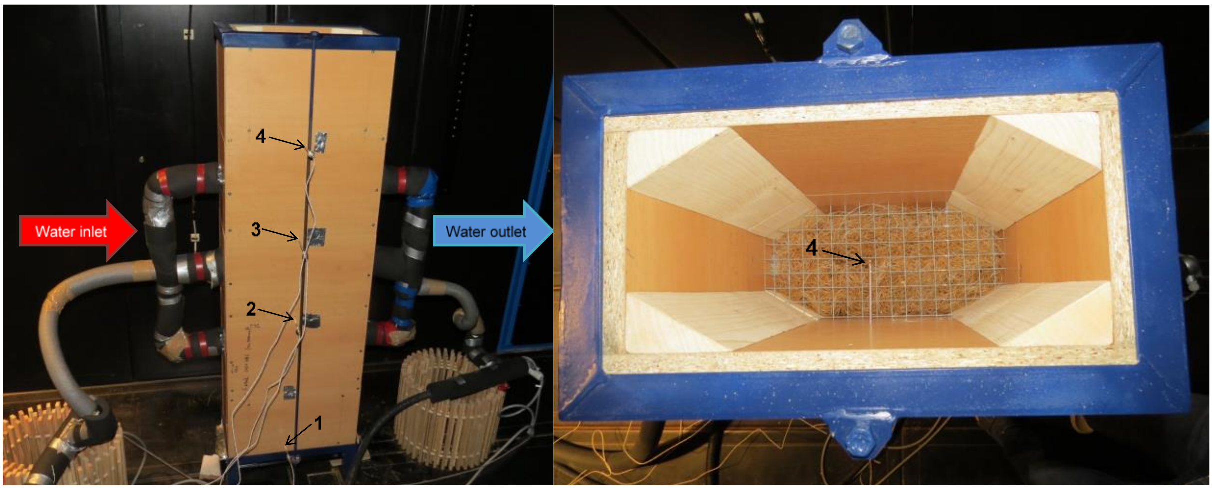

Fig. 5 Overall view of a three hollow fiber bundles heat exchanger (left). Top view of the three hollow fiber bundles heat exchanger (right)

Five experiments were conducted in calorimeter chamber with this heat exchanger. The testing conditions for the first experiment were chosen as the inlet water temperature of 45 ˚C, the outlet water temperature of 35 ˚C and the ambient temperature of 20 ˚C. Required water flow rate was adjusted based on temperature values. When the steady state was reached temperatures at each the heat exchanger section was measured and the heat transfer rate was calculated. Values of temperatures and heat transfer rates are given in Tab. 3. In the second experiment the initial conditions were the following: the inlet water temperature was equal to 75 ˚C, the outlet water temperature was equal to 65 ˚C and the ambient temperature was 20 ˚C.

Tab. 3 Experimental results

Experiment number		1	2	3	4	5
Inlet/Outlet/Room temp.	˚C	45/35/20	75/65/20	75/65/20	75/65/20	75/65/20
Temp. 1	˚C	20.0	20.0	20.5	20.0	19.5
Temp. 2	˚C	34.5	64.0	59.0	52.5	48.5
Temp. 3	˚C	39.4	68.0	69.5	66.7	61.2
Temp. 4	˚C	35.6	60.0	70.5	70.3	68.5
Air velocity	m/s	-	0.2*	0.5	0.9	2
Heat transfer rate	W	145	550	1260	2020	≈ 4000
Heat transfer coefficient	W/m^2K	16.5	34.6	65.3	77.8	134.1

* air flow rate at natural convection, the value was calculated.

Experiments № 3, 4 and 5 were conducted at the same initial conditions that experiment 2, but a fan installed from the bottom side of heat exchanger was used. The air flow velocity through the heat exchanger was set with the help of a voltage regulator. Experiments were conducted at air flow velocities equal to 0.5, 0.9 and 2 m/s.

From the table it can be seen that heat transfer rates for the experiments conducted at natural convection are quite low, especially for the case when the inlet water temperature was equal to 45˚C, the outlet water temperature was 35˚C and the ambient temperature was 20 ˚C. With increasing of water temperatures, the air velocity also increases and achieves the value of 0.2 m/s. Fig. 6 presents a dependence of the heat transfer rate on the air velocity inside the heat exchanger. It can be seen that the heat transfer rate increases linearly with the air velocity increasing and reaches a value of 4000 W at the air velocity of 2 m/s.

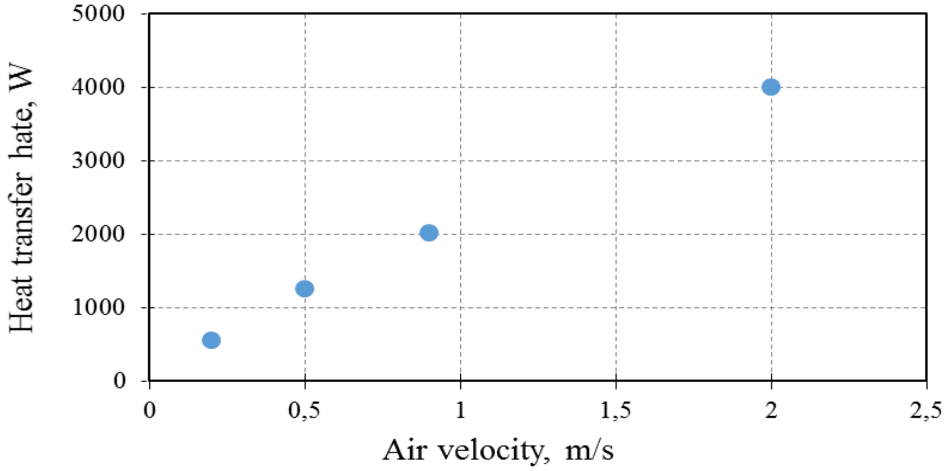

Fig. 6 Dependence of the heat transfer rate of PHFHE on the air velocity through the three hollow fiber bundles heat exchanger

Fig. 7 presents a temperature distribution inside the heat exchanger for different air flow velocities.

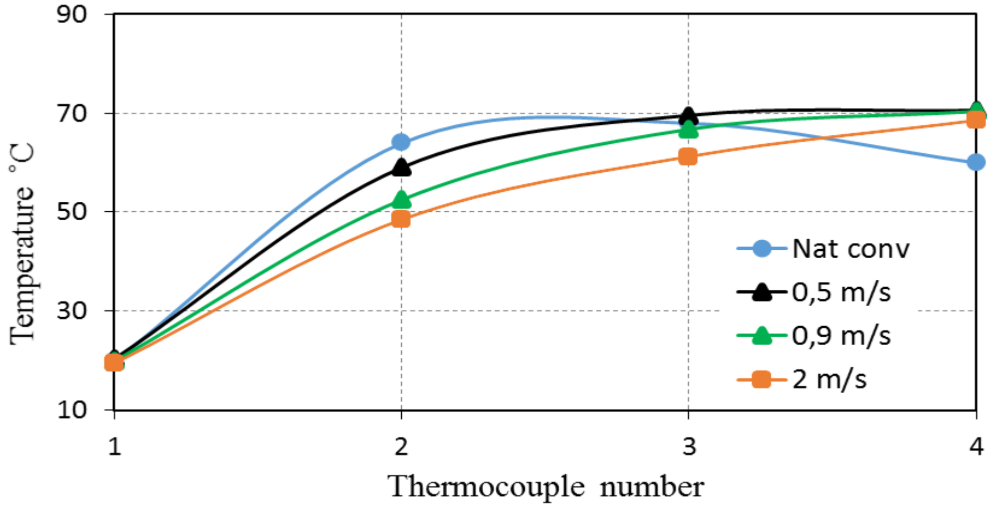

Fig. 7 Temperature distribution inside the three hollow fiber bundles heat exchanger

It can be seen that at air velocity equal to 0.9 and 2 m/s a substantial contribution to the air heating is made only by the first two bundles. The use of the third bundle is unreasonable. At natural convection and forced convection with the air velocity of 0.5 m/s the main air heating is realized by the first bundle. In the case of natural convection, can be seen that air after third bundle had a lower temperature than air after first one. It can be explained by very slow air flow rate through the third bundle and by the mixing of the surrounded air above the third bundle with lower temperature with the hot air after third bundle in the top of the heat exchanger. It means that is no need to use second and third bundles because they almost don't participate the air heating but provide an additional hydraulic resistance in the system.

2.2 HEAT EXCHANGER BASED ON ONE HOLLOW FIBER BUNDLES

Based on experimental results with three bundle heat exchanger, tests of the heat exchanger consisting of one hollow fiber bundle were carried out. At the experiment № 6 a free hollow fiber bundle was tested. The initial conditions during experiment were the following: the inlet water temperature was equal to 75 ˚C, the outlet water temperature was equal to 65 ˚C and the ambient temperature was 20˚C. Heat was transferred by natural convection. Pt100 RTD sensor № 1 registered temperature on the bottom side of the bundle, sensors № 2-4 measured temperature above the bundle.

At the experiment № 7 the heat exchanger containing one hollow fiber bundle was tested. Bundle was placed between two metal nets, protecting it from dipping. Pt100 RTD sensor № 1 registered temperature on the bottom of the heat exchanger, sensor № 2 - on the bottom side of the bundle, sensors № 3-4 measured temperature above the bundle. Experiment № 7 was conducted at the same conditions as experiment № 6.

The overall view of both experimental setups is shown in the Fig. 8.

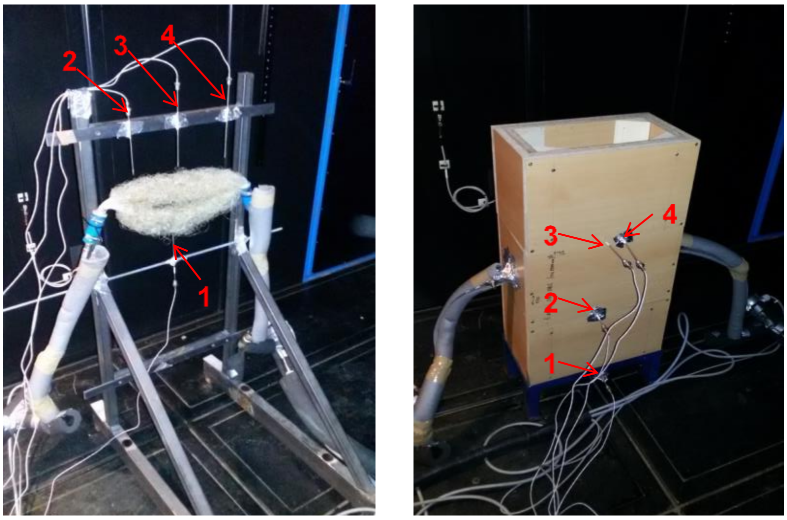

Fig. 8 Overall views of experimental setup for a free hollow fiber bundle (left) and for the heat exchanger with one hollow fiber bundle (right)

Values from resistance thermometers and heat transfer rates are given in Tab. 4. Base on the equation of thermal balance values of air velocities for each type of experiment were calculated.

Can be seen that in experiment with free hollow fiber bundle values of temperature sensors 2-4 are differ. It indicates nonuniformity of the air flow, the air flow rate is small. Experimental data for one hollow fiber bundle heat exchanger show that the use of a case results in air flow rate increase by 60%. Heat transfer rate also increased by 54%. It can be explained by the chimney effect.

Tab. 4 Experimental results

Experiment number		6	7	8
Inlet/Outlet/Room temp.	˚C	75/65/20	75/65/20	75/65/20
Temp. 1	˚C	19.1	19.3	18.6
Temp. 2	˚C	65.8	20.4	19.4
Temp. 3	˚C	65.6	56.5	64.6
Temp. 4	˚C	31.7	60.0	62.4
Heat transfer rate	W	292	450	1085
Heat transfer coefficient	W/m^2K	21.2	29.7	75.9
Air velocity	m/s	0.13	0.21	0.48

Air velocities for each type of experiment are demonstrated in Fig. 9.

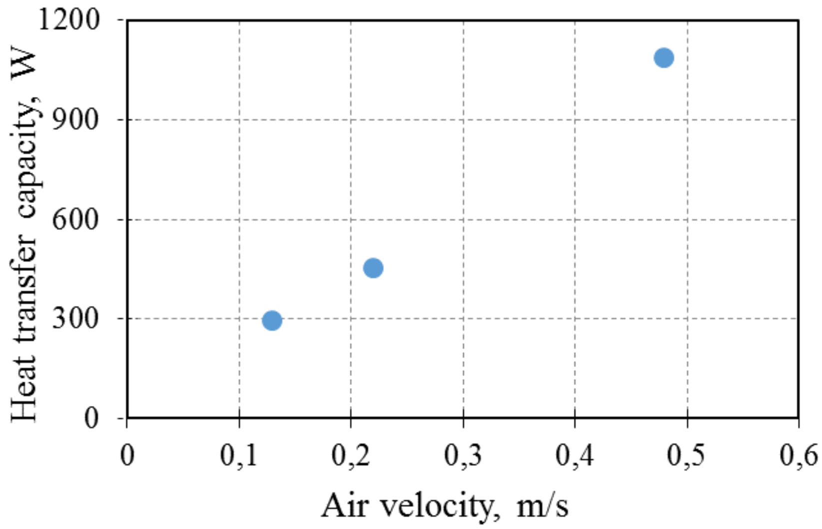

Fig. 9 Dependence of the heat transfer rate on the air velocity for one hollow fiber bundle

In order to increase the chimney effect one hollow fiber heat exchanger equipped with a 1.6 m tube installed on the top was tested. Experiment № 8 was conducted at the same conditions like experiments № 6 and 7. The use of a tube strengthened the chimney effect and resulted in the heat transfer rate increase by 2.5 times and reached a value of 1085 W. Results obtained for one hollow fiber bundle heat exchanger with tube on the top are compared with a heat transfer rate of series-produced metal radiators.

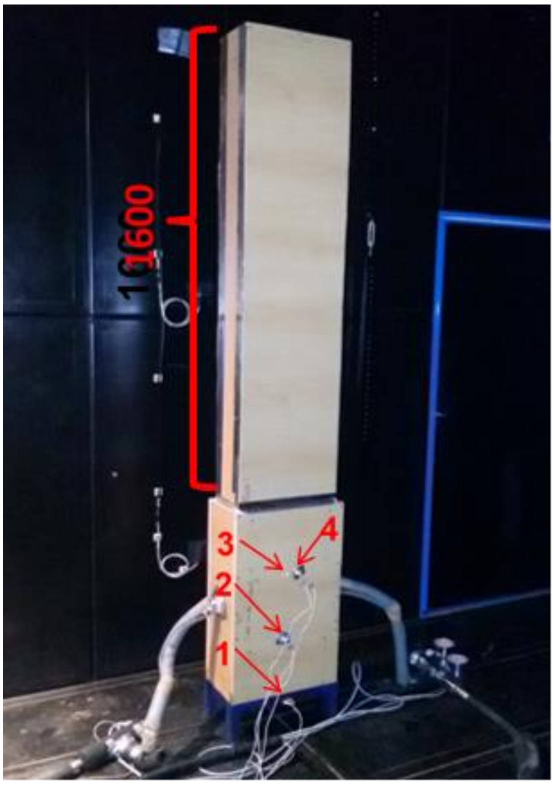

Fig. 10 Overall view of the heat exchanger with one hollow fiber bundle and a tube.

3 OPTIMIZATION OF SHELL-AND-TUBE PHFHE

3.1 PHFHE WITH PARALLEL HOLLOW FIBERS PA-72

The first object of study was a plastic shell-and-tube type heat exchanger PA-72, shown in Fig. 11. The heat exchanger 240 mm long had an external transparent shell with a diameter of 50 mm, made of plastic. Inside the shell 1700 parallel polyamide hollow fibers 180 mm long with outer diameter of 0.8 mm were placed. The maximum internal pressure of this heat exchanger was 3 bar.

Fig. 11 Overall view of PA-72 PHFHE

Water was used as working fluids. The experiment was carried out under the following conditions: water flow inside the fibers had a temperature of 11 °C, shell water flow had a temperature of 50 °C and 70 °C. The measurements were carried out for two values of water flow rate inside the fibers equal to 338 l/h and 675 l/h. Shell water flow rate varied from 50 to 240 l/h with the step of 50 l/h. During the experiments, temperatures at the inlet and outlet of the fibers, as well as at the inlet and outlet of the shell were measured. The differential manometer was used to measure the water pressure loss in the heat exchanger shell.

The dependence of the PA-72 heat transfer rate on the shell water flow rate is shown in Fig. 12. Data are given for the shell water temperature of 50 °C. From the graph it can be seen that at low shell water flow rates the heat transfer rate of heat exchanger does not depend on the flow rate of the liquid inside the fibers. Only with an increase in the shell water flow rate more than 150 l/h, the difference in heat transfer rate becomes more than 10%. The maximum heat transfer rate reached for a PA-72 heat exchanger with a shell water temperature of 50°C is 3.8 kW and maximum flow rates for both working fluids.

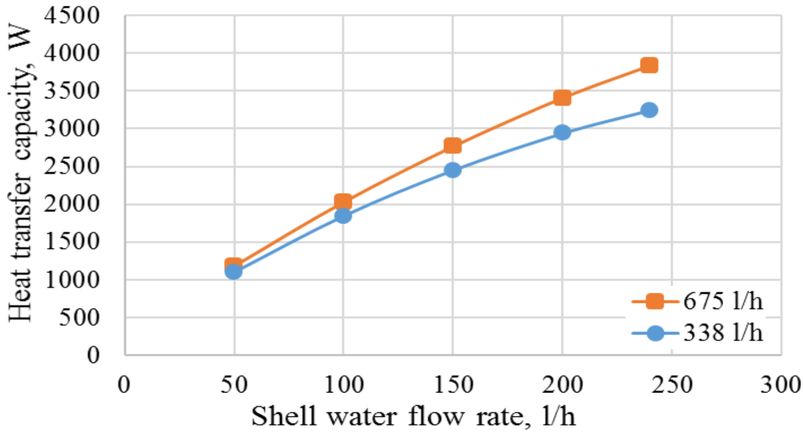

Fig. 12 Dependence of the PA-72 heat transfer rate on the shell water flow rate for the shell water temperature of 50 °C

Fig. 13 shows the dependence of the PA-72 heat transfer rate on the shell water flow rate for the shell water temperature of 70 °C.

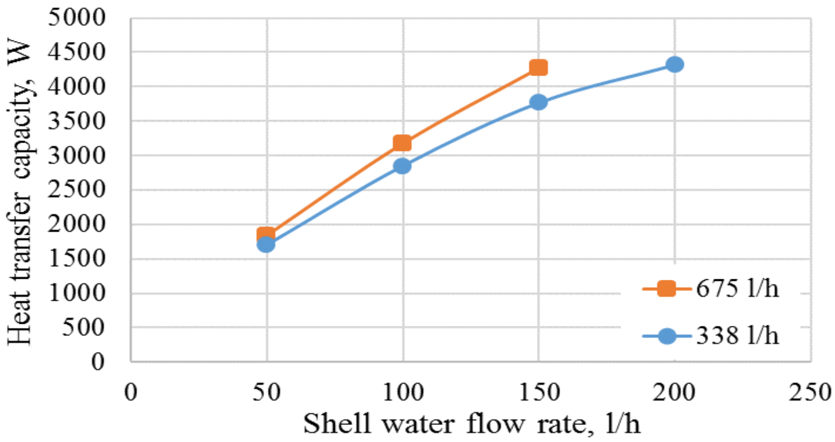

Fig. 13 Dependence of the PA-72 heat transfer rate on the shell water flow rate for the shell water temperature of 70 °C

It is seen that the nature of the thermal rate change with the increase in the water flow rate inside the fibers is similar to that observed at 50 °C. The maximum value of the heat transfer rate reaches 4.3 kW with a shell water temperature of 70°C and maximum flow rates for both working fluids.

Fig. 14 presents a dependence of the pressure losses in the PA-72 heat exchanger shell on the shell water flow rate. It can be noted that already at relatively small flow rates of 200 l/h the pressure loss is 50 kPa. This happen due to the fact that a large number of fibers inside the heat exchanger create a significantly hydraulic resistance to fluid flow. However, with this manufacturing technology it is impossible to reduce the number of fibers keeping the uniformity of their distribution in the volume of the heat exchanger.

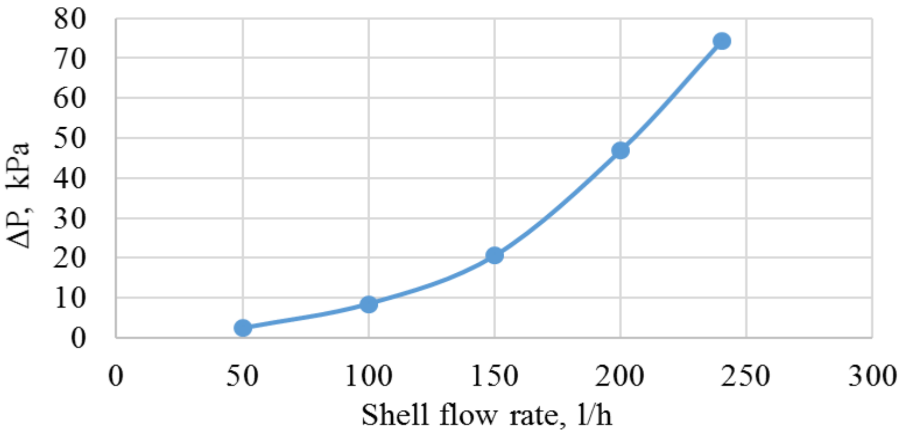

Fig. 14 Dependence of the pressure drop in the heat exchanger shell on the shell water flow rate

If you pay attention to the graph of the overall heat transfer coefficients of the heat exchanger (Fig. 15), it is clear that $h_{overall}$ for both water shell temperatures are quite low. The maximum $h_{overall}$ was equal to 180 W/m^{2}K at the shell water temperature of 50 °C and the water flow rate inside the fibers of 675 l/h. It can be explained by the large number of fibers in the shell and specialty of hollow fibers location along each other. In this case the fibers can easy overlap and adhere to each other, reducing the active heat transfer area.

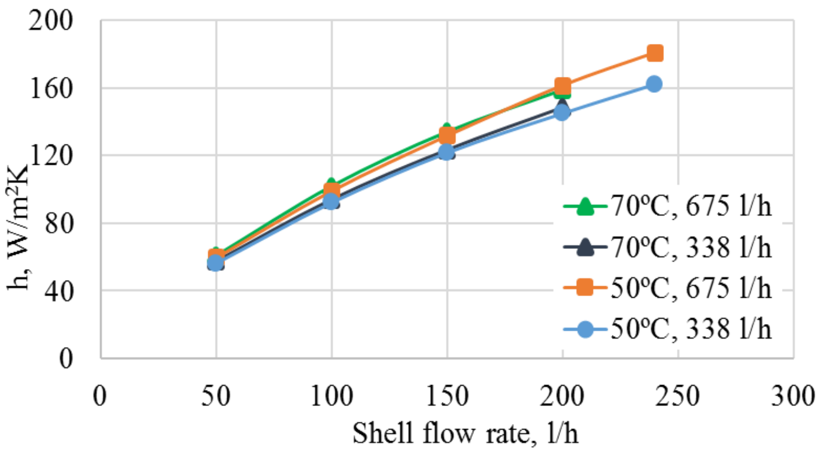

Fig. 15 Dependence of the overall heat transfer coefficient of the PA-72 heat exchanger on the shell water flow rate

Changes in the heat exchanger structure connected with adhesion of hollow fibers can be seen in Fig. 16 presenting a photo of the heat exchanger PA-72 after experiments. Fibers stick to each other, blocking the active area of heat transfer. It can also be noted that separate compact structures were formed from the fibers. Gaps with a low hydraulic resistance appeared between these structures. So, the part of shell water goes through these gaps and does not penetrate between the fibers. Than this working fluid almost not participate in heat transfer. Both of these factors leads to a significant decrease in the overall heat transfer coefficient of the PA-72 heat exchanger.

Fig. 16 External view of PA-72 PHFHE after tests

3.2 MAIN METHODS OF HEAT- AND MASS- TRANSFER MODELING

To avoid sticking of the fibers during operation of the heat exchanger, was considered a method of optimizing the design of the heat exchanger by placing fibers in the shell at an angle (Fig. 17). Supposed that such a change in the heat exchanger structure will allow the fibers to be evenly positioned and to avoid sticking of the fibers. Fiber will have a contact to each other only in one point and not in the whole length. It can lead to a uniform fluid flow in the volume of the heat exchanger shell and the whole entire surface will participate heat transfer processes. Also, when hollow fibers will lay at an angle to each other it may lead to an increasing in the thermal performance of the device due to intermixing of the liquid flow and avoiding of laminar flow forming inside the shell.

Before making a prototype, a numerical simulation of the heat exchange process on the outer surface of polymeric fibers was carried out. The main purpose of such a modeling was to optimize the heat exchanger geometry and to determine the best hollow fibers position inside the heat exchanger. For this purpose, a program ANSYS was used.

3.3 DESCRIPTION OF PHFHE DOMAIN

This study was already published in Materiali in tehnologije journal in 2018 [3]

Described above heat exchanger with parallel hollow fibers PA-72 had about 1700 fibers. Due to the large number of fibers the simulation of the whole internal volume of the heat exchanger will take a lot of computing power. Therefore, a rectangular parallelepiped domain with a square cross section of 10x10 mm and length of 100 mm was selected as the object of investigation.

Fibers in the model, which were considered as hollow cylinders, pass through the cooling medium’s domain. The outer diameter of each fiber was 0.8 mm and the interaxial distances between adjacent fibers was 2.4 mm. As the fiber wall temperature at this length changes slightly in reality, in the model it was set as a constant and were equal to 60 °C. Water was chosen as a cooling medium with the velocity equal to 0.3 m/s. The pressure inside the domain was 1 bar. Plastic fibers have a low surface roughness (less than 1 μm), therefore it was not considered in this model.

The model of model makes possible to change the orientation of the hollow fibers inside the heat exchanger. The hollow fiber position with respect to the direction of the shell water flow, determined by the angle α, was concerned as a variable parameter.

Angle α in the model was set according to Fig. 17 (left). The largest value of the angle α is restricted by opportunities of production equipment and is equal to 45°. Therefore, in this simulation, this parameter varied from 0° to 45°. Fig. 17 (right) shows angle α determination in a real PHFHE.

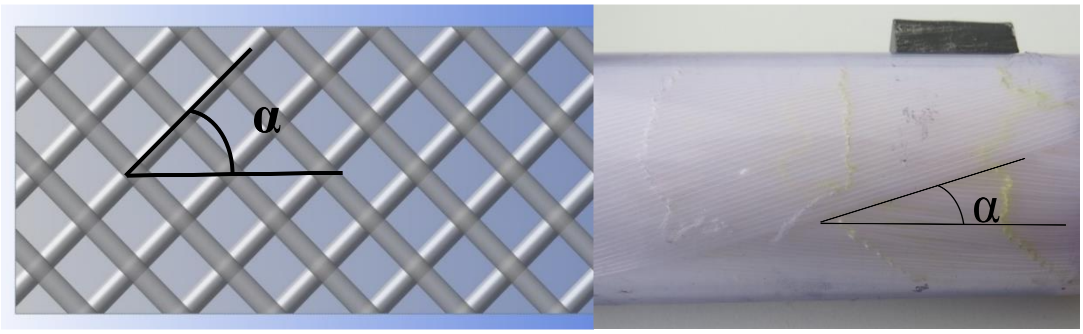

Fig. 17 Angle α in in the model (left) and a real PHFHE (right)

The heat exchange process was simulated using the finite volume method in ANSYS CFX [4], [5]. Because the aim of the work was to compare the heat transfer coefficients on the hollow fiber surface, the domain size and boundary conditions for each case of α were identical.

3.4 RESULTS OF PHFHE MODELING

The computer simulation results in the view of the dependence of the heat transfer coefficient on the outer fibers surface on the angle α between the cooling flow direction and the fibers is shown on the. Fig. 18 The graph shows that even an increase in angle α up to 5° results in the heat transfer coefficient increase by two times. This fact is explained by the change of the laminar cooling flow regime in the heat exchanger with the angle α=0 to the turbulent cooling flow regime for the exchangers with α > 0. Hollow fibers cross the water flow and water starts to intermix, leading to intensification of heat exchanging on the outer surface of hollow fibers.

It can be seen from the graph that heat transfer coefficients increase almost linearly when angle α changes from 0° to 30°. Further increases in α, from 30° to 45°, lead to very slight increase in heat transfer coefficients. Decrease in the growth rate of heat transfer coefficients can be explained by the achievement of the regime of advanced turbulence. The maximum heat transfer coefficient was equal to 10 563 W/m^{2}K.

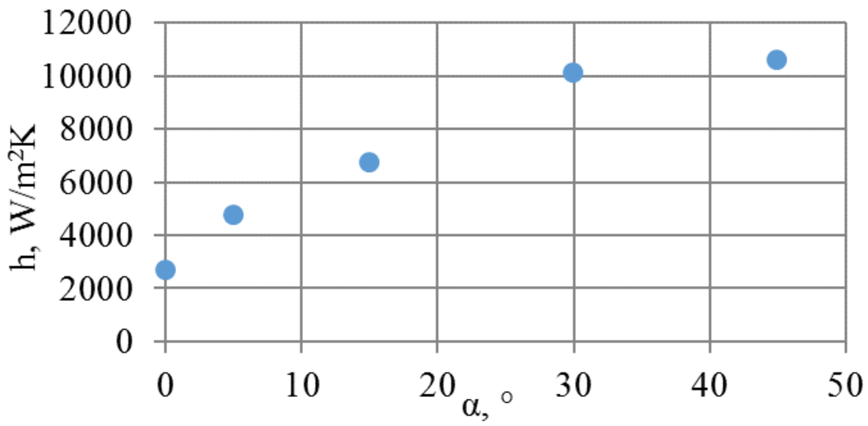

Fig. 18 Heat transfer coefficient on the outer surface of hollow fiber vs the angle α.

Fig. 19 illustrates the dependence of the total heat transfer coefficient of the whole heat exchanger on angle α. Even an increase in α to 5° increases the heat transfer coefficient by nearly 20%. At the maximum angle α= 45°, the heat transfer coefficient is increased by 35% and is equal to 1267 W/m^{2}K.

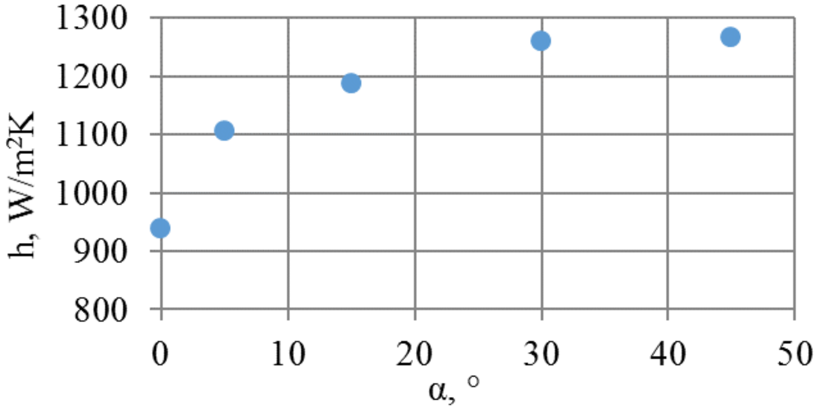

Fig. 19 Overall heat transfer coefficient of the PHFHE vs the angle α.

3.5 PHFHE WITH CROSSED HOLLOW FIBERS

Based on computer modeling results two similar PHFHEs shown in Fig. 20 were produced. Heat exchangers were constructed as shell-and-tube with a water as a working fluid [6]. Hollow fibers with the diameter of 0.8 mm were made of polycarbonate. Potting area diameter of the shell was equal to 40 mm and the effective length of the PHFHEs was 280 mm.

In PHFHE-1 hollow fibers were placed parallel to the liquid flow. PHFHE-2 had hollow fibers crossed at the angle of 22.5°. Such hollow fibers orientation was realized by hollow fibers interweaving by a synthetic filament and its further twisting, see Fig. 21 and Fig. 17 (right). Fibers amount of potting area for PHFHE-1 was 30% and the heat transfer area was 0.52 m^2, for PHFHE-2 these values were equal to 33% and 0.54 m^2 correspondently.

In comparative experiments the water temperature inside the fibers was equal to 27 ºC and the shell water temperature was set as 80 ºC. The liquid flow rate inside the fibers was 660 l/h, and the shell water flow rate varied from 200 to 1000 l/h.

Fig. 20 PHFHE-1 and PHFHE-2 in stainless steel shell

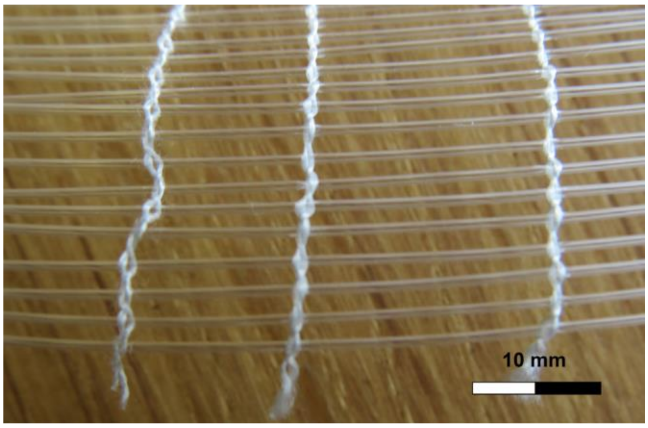

Fig. 21 Layer of hollow fibers separated by weaving technology [7].

The dependence of the PHFHE overall heat transfer coefficients on the shell water flow rate is presented in the Fig. 22. It can be seen, that at low flow rates heat transfer coefficients are almost the same for both of heat exchangers. However, the heat transfer coefficients for the PHFHE-2 with crossed hollow fibers are more than 10% higher compared with the PHFHE-1 with parallel fibers at the shell water flow rate increasing over 800 l/h. At the shell water flow rate of 1000 l/h $h_{overall}$ for PHFHE-2 achieves the maximum value of 1091 W/m^{2}K, wherein the maximal heat transfer coefficient of the PHFHE-1 was 970 W/m^{2}K.

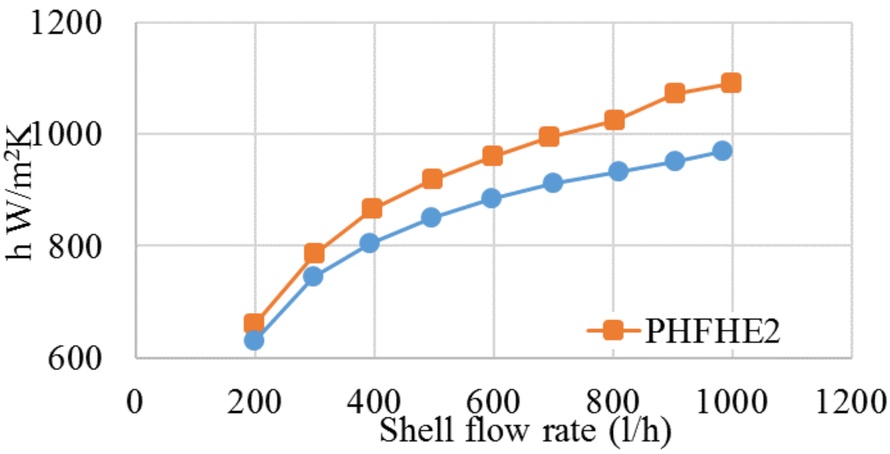

Fig. 22 Dependence of the PHFHE-1 and PHFHE-2 overall heat transfer coefficients on the shell water flow rate.

Such an increase in $h_{overall}$ for the PHFHE-2 with crossed hollow fibers can be explained by the fact that the declination of the hollow fiber axis with respect to the shell water flow leads to the its turbulization and as a consequence to the heat transfer coefficients on the hollow fibers outer surface increase.

Moreover, despite hollow fibers crossing the pressure losses for both of heat exchangers were identical as can be seen from the Fig. 23. Such a PHFHE design doesn't affect the device hydraulic resistance.

It should be noted that active heat transfer areas were almost the same for both heat exchangers. So, the method of hollow fibers crossing inside the PHFHE results on the heat exchanger efficiency increase and can be applied for producing the shell-and-tube PHFHEs.

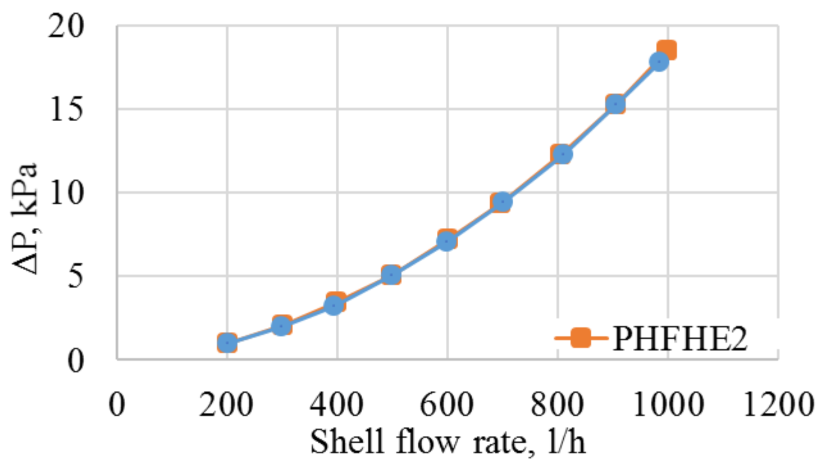

Fig. 23 Dependence of the hydraulic losses inside the PHFHE-1 and PHFHE-2 shell on the shell water flow rate

4 DEVELOPMENT OF TECHNOLOGY FOR PRODUCING PHFHE WITH OPTIMIIZED STRUCTURE

The requirements for manufacturing methods of polymeric hollow fiber heat exchangers and for pilot machine itself can be summed up as follows:

1. should allow automated production of the PHFHE;
2. should be easily adapted for mass production;
3. should allow to set the dimensions of the heat exchanger (length, diameter and so on);
4. should allow to regulate angle of slope of the hollow fibers and their number.

Taking into account these requirements, the pressure tanks manufacturing technology was taken into account. The winding principle on these machines was adapted and adjusted for the needs of PHFHEs manufacturing.

4.1 WINDING SETUP X-WINDER AND ITS MODERNIZATION

For the production of PHFHE with crossed fibers, the X-Winder was used. This machine is a smaller version of the filament winding machines, which were invented during the 1960's and were used to winding of airframes and missiles [8]. These machines are typically the size of a school bus and are very expensive. The X-Winder considered as a desktop version of these machines and it is mainly used to create small pressure tanks by winding technology. The main advantage of this device is the ability to simulate the process of manufacturing a product on real production capacity for reasonable price. External view of the X-Winder is shown in Fig. 24.

Fig. 24 External view of the X-Winder

To use this system for the shell-and-tube heat exchanger production, changes were made to its design. New feeding heads have been designed and manufactured using 3D printer shown in Fig. 25: the lower one was used for winding plastic fiber, the upper one - for winding glass fiber for the PHFHE shell. New heads made it possible to more accurately set the position of the plastic fiber in the finished product and prevent the glass fiber of slipping during the winding of the outer shell.

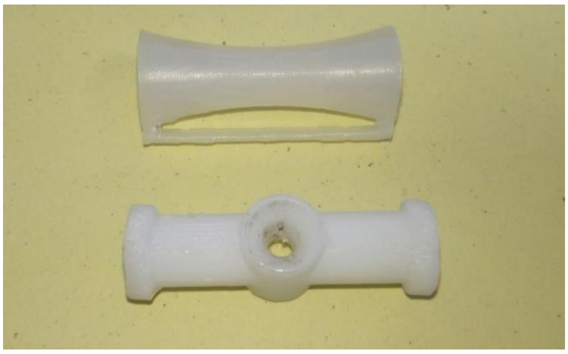

Fig. 25 Improved feeding heads

In order to avoid slipping of the plastic fibers at the ends of the PHFHE during their winding, as well as to set the exact location of the fibers in the necessary positions, special sprockets were designed and manufactured. They were fixed at the ends of the device. During the winding process the fiber lays in the groove and due to this remains fixed in the right position. The use of different number of a sprocket beam allowed to create a constant volume of fibers with increasing winding diameter. The External view of the sprockets and their use in the pilot machine are shown in Fig. 26. For different diameters of the winding samples, various sprockets were used. These sprockets allowed their reuse for production of the next piece of the PHFHE.

Fig. 26 External view of the sprockets with different diameters.

The third change concerned the management program. It was necessary to keep the fiber volume at the end sections constant and to fit each hollow fiber to exact sprocket groove. So, the G-code was completely rewritten. It took into account different speeds of feeding head movement while forming the linear and ends sections, turns of the feeding head, the speed of the created PHFHE rotation and so on.

The production of the heat exchanger is divided into three stages. The first is polymeric fibers winding and its sealing at the ends of the device. At this stage it was possible to set the number of fibers in the heat exchanger and the angle of their inclination relative to the axis of the heat exchanger. Simultaneously with the winding process, flooding of the hollow fibers took place to create the end sections. This method of casting is highly effective, because epoxide is applied to the end sections of the fiber gradually and evenly, better fills all cavities between the fibers. It prevents the formation of through microcracks between the fibers and epoxy, which can damage the tightness of the device. Also, one of the most important advantages of this manufacturing method is that during the entire cycle of heat exchanger winding one continuous filament is used. Appropriately, when the hollow fibers are flooded with epoxide, the penetration of the epoxide into the fibers is impossible, and when cutting off the ends of the heat exchanger, 100% of the fibers are open. Overall view of the heat exchanger inner part at the end of the first creation stage is shown in Fig. 27.

Fig. 27 Inner part of PHFHE after first stage of creation

At the second stage a shell of the heat exchanger is wound. The shell is created by winding fiberglass strip directly onto polymer hollow fibers. The fiberglass strip from the supply coil passes through the bath with epoxide, extra amount of epoxide is cut off with a special knife, and glass fiber is wound onto the heat exchanger through the delivery head. Winding of the heat exchanger shell is realized by several layers of winding fiberglass strip. The first layer consists of fiberglass wound at an angle close to 90º to the axis of the heat exchanger. This layer ensures the tightness of the heat exchanger. The next layer is wound at an angle of 20-30º to the axis of the heat exchanger and ensures rigidity and strength of the shell. At the end of the PHFHE shell production it consists of 3-4 layers of fiberglass interchanging each other. The heat exchanger shell made in such way is very durable and quite thin, about 2-3 mm. The main advantage of this method is to ensure reliable adhesion of the shell structure with the end-filled areas of the heat exchanger that eliminated a possible leak. External view of the PHFHE at the end of the second stage of creation is shown in Fig. 28.

Fig. 28 External view of the PHFHE at the end of the second creation stage

At the third stage the side sections are cut, and side flanges are glued. Side flanges have holes for the input and output of working fluids. The overall view of the PHFHE at the end of the third stage of creation is shown in Fig. 29.

Fig. 29 External view of PHFHE in the end of the third part of the creations

A scheme of the PHFHE wound by X-Winder is shown in Fig. 30. The first working fluid, passing through the central hole in the PHFHE side flange flows into the central tube, around which PHFHE was wound. The tube had special holes inside the heat exchanger. Separation walls preventing liquid to flow inside the tube were made. Thus, the shell water flows through the side flange hole inside the shell of the heat exchanger, flows around the fibers from the outside and goes out from the other end of the heat exchanger. The second working fluid, passing through an additional hole in the PHFHE side flanges, flows inside the polymeric hollow fibers.

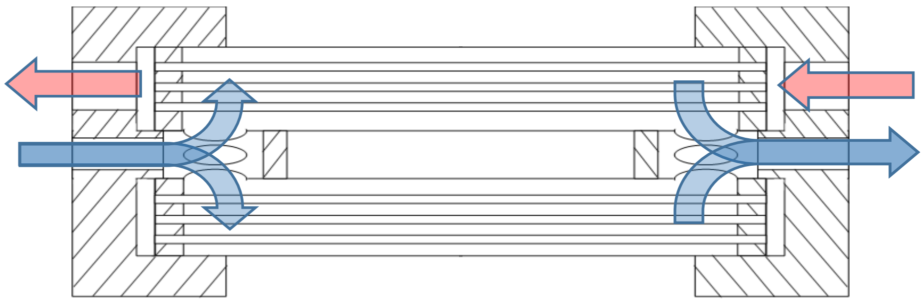

Fig. 30 Scheme of the PHFHE

The main advantage of the described method of polymeric heat exchanger creating is the following: based on this principle, it is possible to create serial production of shell and tube PHFHE. This production method can be fully automatized and one winding machine can create several heat exchangers at a once.

4.2 STRENGTH PRESSURE TESTS OF PROTOTYPE WIND-1

Before conducting a heat transfer experiments, it was necessary to verify the reliability of the created heat exchanger and evaluate its performance in terms of leak tightness. The heat exchanger shell was made of 4-layers of fiberglass strip. The main purpose of tests was to check whether this number of layers is capable to ensure sufficient body strength. PHFHE pressure tests were carried out with static and cyclic loading of increased pressure. For these purposes, a special stand was designed and created. The stand allowed pressure testing with air or water at the given pressure. The typical pressure profile during the cycle tests in dependence on time is shown in Fig. 31.

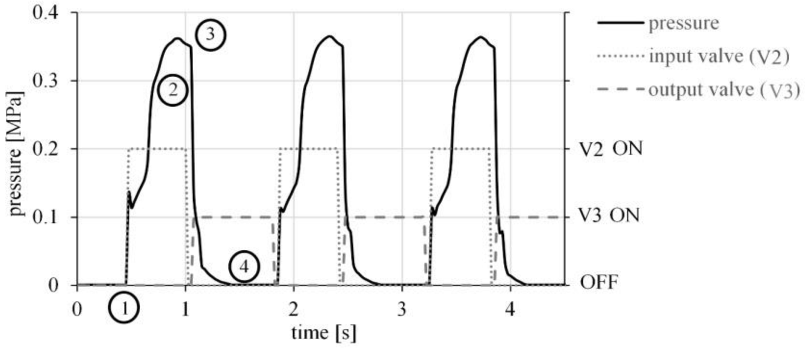

Fig. 31 Sample of pressure test cyclogam [9]

The cycling testing facility also can be used for static pressure tests of heat exchanger. It means that a constant pressure will be maintained in the sample throughout the experiment.

Prototype Wind-1 was made with the help of the modified winding system X-Winder especially for pressure test aiming to check the shell reliability and PHFHE leak tightness. A 350 mm long heat exchanger with a diameter of 80 mm was made of polyamide hollow fibers that were crossed at the angle of 22.5º to the axis of the heat exchanger. The flanges of PHFHE were made of metal, one of them did not have holes for the input/output of the working fluid, so that it was possible to make a pressure tests of the heat exchanger. The first pressure test was carried out at the air constant pressure of 3 bar, while the heat exchanger was immersed in a water tank with the water temperature of 60 °C.

After 10 hours of Wind-1 testing at the junction of the shell and the metal flange a leak occurred (Fig. 32).

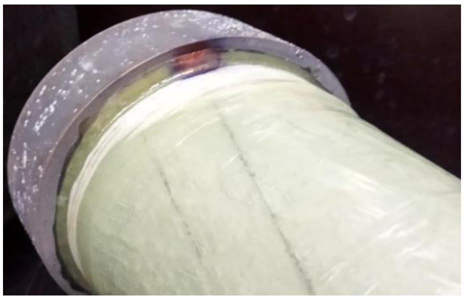

Fig. 32 Leak location in PHFHE Wind-1 after static pressure tests

It can be explained by the difference in the metal and fiberglass expansion coefficients. When Wind-1 was heated, in the junction of the shell and the flange can be form microcracks, which disrupted the tightness of the device. Based on this experiment, it was concluded that flanges of the PHFHE should be made of a material with a similar coefficient of expansion, like the fiberglass.

Since one of the tests purposes was to check the fiberglass shell reliability, so the leak was resealed, and the heat exchanger was tested again. In this case, water was used as the test fluid, and the pressure inside the heat exchanger was 3.5 bar throughout the test. Wind-1 was no longer placed in water but was in air at a temperature of about 20 °C to avoid leak occurred due to the different expansion coefficient of materials. After 287 hours of testing, no leaks were detected. The test was stopped, and the test results were regarded as successful.

The next step was cyclic pressure tests. Tests were carried out at the air pressure with the peak value of 3 bar. The Wind-1 was immersed in water with a temperature of 20 °C. The heat exchanger was tested for a week, during this time 1747364 cycles were made, and no leaks were identified.

After this, a test was carried out with Wind-1 immersed in water with temperature of 60 °C and air pressure inside was equal to 6 bar in the maximum peak. After 100 cycles a small leak was found in the same place as shown in Fig. 32. Since the heat exchanger shell was still leak-proof, the tests were continued, and only after 7609 cycles, when more leaks between the shell and the metal side sections began to occur, the tests were stopped. No leaks were found in the shell.

These tests showed that the shell is reliable and ruggedized enough to withstand an impact of internal pressure in 6 bar. In this regard, it can be concluded that 4 layers of fiberglass provide the necessary strength and tightness of the PHFHE shell. The same tests showed that even at the prototype stage it is impossible to use metal flanges, due to the difference in expansion coefficients of the metal and fiberglass which leads to the occurrence of microcracks at the junction of the shell and the flanges and the leakage of the heat exchanger.

4.3 PROTOTYPE WIND-2 AND ITS TESTING

With the help of the modified winding system X-Winder the second shell-and-tube PHFHE prototype Wind-2 shown in Fig. 33 was made.

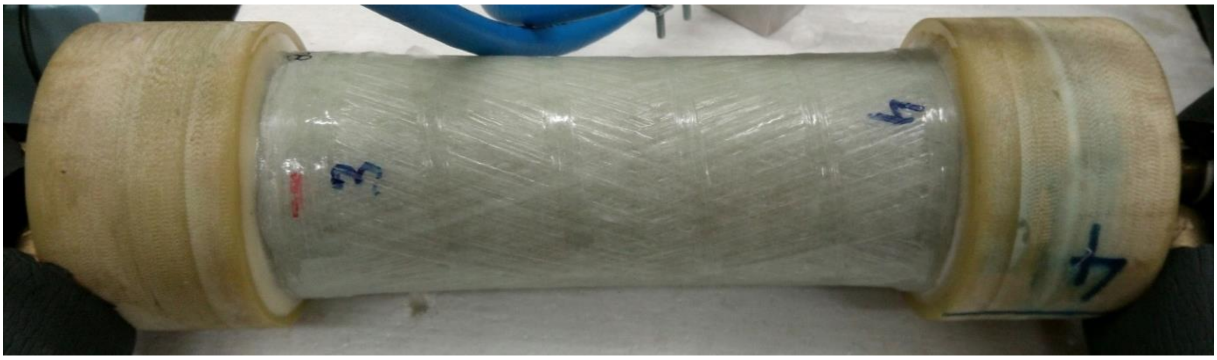

Fig. 33 Hollow fiber heat exchanger prototype Wind-2

The outer diameter of the heat exchanger body was equal to 80 mm, length was equal to 350 mm. Inside the heat exchanger were 2200 PA fibers 206 mm long with an outer diameter of 0.8 mm, that were crossed at angle of 22.5 ° to the axis of the heat exchanger. Despite the relative compactness of devise, the heat exchange surface of the Wind-2 was 1.14 m^2.

Considering the negative test results of the prototype Wind-1 with metal flanges, for prototype Wind-2 they were made from fiberglass.

Experiments were carried out under the following conditions: water flows into fibers with a temperature of 13°C and with a constant flow rate of 2160 l/h. The shell water flow rate varied from 390 l/h to 800 l/h with a step of 200 l/h. Experiments were carried out at two shell water temperatures - 70°C and 50°C. During the test, temperatures at the inlet and outlet of the heat exchanger shell, as well as at the inlet and outlet of the fibers, were measured. Pressure drops in the heat exchanger shell were measured by a differential manometer.. The dependence of the pressure drop inside the heat exchanger on the shell water flow rate is shown in Fig. 34.

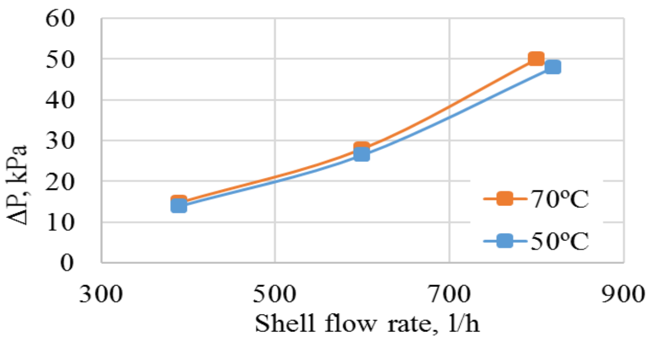

Fig. 34 Dependence of the pressure drop inside the shell of Wind-2

on the shell water flow rate.

It can be seen from Fig. 34 that pressure drop does not depend on the temperature of the liquid inside the fibers and at a maximum shell water flow rate of 800 l/h reaches a value of 0.5 bar. This is an acceptable value for shell-and-tube heat exchangers. The pressure drop in the Wind-2 heat exchanger is higher than in the previously tested PHFHE-2 (Fig. 23), in which polymeric hollow fibers were located at an angle of 22.5º to the axis of the heat exchanger and were fixed with the help of threads. However, it should be noted that the active heat transfer area in Wind-2 is more than two times higher than in the heat exchanger PHFHE-2. It means that inside the shell of Wind-2 is a greater amount of fibers and, correspondingly, a greater hydraulic resistance. However, this production method allows percussive setting of fibers amount inside the heat exchanger to achieve the optimal number for each application.

Fig. 35 (right) shows the dependence of the Wind-2 heat transfer rate on the shell water flow rate. Curves are given for shell water temperature of 50°C and 70°C. Difference in heat transfer rate values for both water temperatures is about 27% and a maximum reached at shell water temperature of 70 °C is equal to 38 kW at shell water flow rate of 800 l/h.

Fig. 35 (left) shows dependence of the Wind-2 heat transfer coefficient on the shell water flow rate. It can be seen that heat transfer coefficients do not depend on shell water temperature and reach a maximum value of 1950 W/m²K at maximum shell water flow rate 800 l/h. This is a relatively high value. It means that the whole surface of hollow fibers in the shell is involved in heat transfer process. Moreover, hollow fibers crossing inside the shell leads to better shell water intermixing and allows to achieve better heat transfer coefficients on an outer surface of hollow fibers.

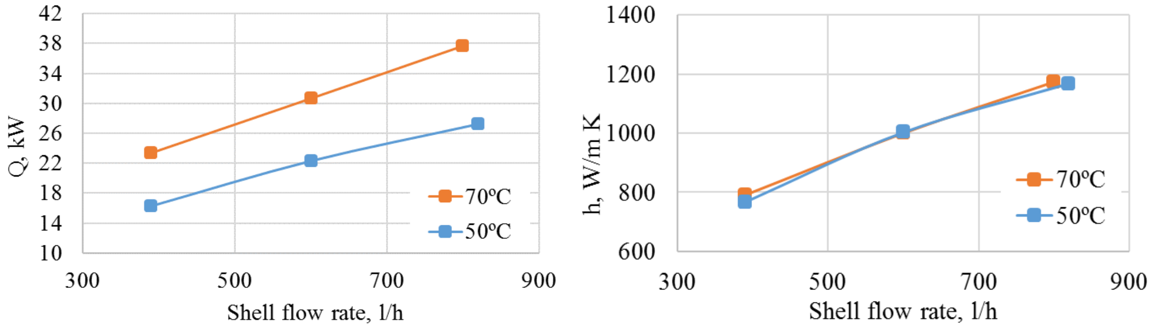

Fig. 35 Dependence of the heat transfer rate (right) and heat transfer coefficient (left) of Wind-2 on the shell water flow rate.

5 CONCLUSION

This thesis is dedicated to the optimization of heat exchangers in order to increase their thermal performance. Metal and polymeric heat exchangers intended for operation in air-water and water-water systems were considered.

The influence of heat pulses on the efficiency reduction of the commercial heat exchanger in fancoil used in building air-conditioning systems was investigated. The metal finned tube heat exchanger consisting of copper coil pipe with pressed aluminum fins as a working body of the fancoil was thermally cycled to simulate its actual operating conditions and find out a potential deterioration of the contact between the fins and the coil pipe. Test have shown that even 100 000 cycles of metal heat exchanger heating and its subsequent rapid cooling did not have a negative impact on the fancoil heat transfer rate. Has been confirmed that the contact between fins and the tube remains constant during heat exchanger’s useful life and the vibration of finning is excluded.

In this regard, the technology of manufacturing metal-heat exchangers with a copper coil pipe with pressed aluminum fins can be considered as optimal.

Most of the work is devoted to the optimization of a relatively new type of polymeric heat exchangers - polymeric hollow fiber heat exchangers (PHFHEs). These heat exchangers utilize hundreds of polymeric microchannels with the outer diameter smaller than 1.5 mm and the wall thickness about 0.1 mm as the heat transfer surface.

Experimental investigation has shown that the use of twisted hollow fiber bundles in water-air applications is perspective. For vertical heat exchangers with natural convection even the use of one twisted hollow fiber bundle can be enough to obtain high values of the heat transfer rate. Chimney effect can significantly increase the heat transfer rate of the heat exchanger.

Wide investigation of PHFHEs design of shell-and-tube type used for water-water application and their optimization were conducted. Study of PHFHE with parallel hollow fibers brought to light problems with fibers sticking and their irregular spacing in the shell due to the formation of compact structures. Such changes in the heat exchanger structure had led to decrease in the heat transfer efficiency of the PHFHE. Computer modeling with the help of ANSYS was used for searching the optimal hollow fiber arrangement inside the PHFHE. Modeling results have shown that the hollow fiber placement at the angle about 20º-30º leads to heat exchange intensification on the outer surface of the hollow fibers. Another advantage of such an arrangement of fibers consist in their volume separation what prevents fibers from sticking.

Tests of PHFHE with hollow fibers placed at the angle to the heat exchanger axis and interweaved by synthetic filament have shown that a strict arrangement of the fibers in the heat exchanger and the possibility of crossing them gives a significant positive effect on the thermal characteristics of the device. However, the use of such a synthetic filament is a quite expensive technology. Besides, during the operation of the heat exchanger, there is a possibility of the fibers wiping in the contact point with the filament.

To solve this problem the technology of pressure tanks winding was modified for producing crossed-wound hollow fiber heat exchanger – a new type of the PHFHE. The prototype produced by the developed winding technology with hollow fibers placed at the angle to heat exchanger axis have shown very good results. At relative compactness this device had a large heat exchange surface that led to a high heat transfer performance of the PHFHE. The prototype shell wound by this technology had withstood the pressure up to 6 bar. This value can be increased by increase in number of shell layers during winding.

The possibilities of assembled winding setup make it possible to wind several heat exchangers at the same time. The developed technology can by applied for crossed-wound PHFHEs mass production.

6 BIBLIOGRAFIE

[1] ASTROUSKI, Ilja a Miroslav RAUDENSKY. The study of polymeric hollow fiber heat exchangers. *18th International Conference ENGINEERING MECHANICS, Svratka, Czech Republic*. 2012, , 47-57.

[2] ZARKADAS, Dimitrios, Baoan LI a Kamalesh SIRKAR. Polymeric Hollow Fiber Heat Exchangers (PHFHEs): A New Type of Compact Heat Exchanger for Lower Temperature Applications. *Heat Transfer: Volume 4*. ASME, 2005, , 429-438. DOI: 10.1115/HT2005-72590. ISBN 0-7918-4734-9. Dostupné také z: http://proceedings.asmedigitalcollection.asme.org/proceeding.aspx?articleid=1586589

[3] BARTULI, E. a M. RAUDENSKY. Numerical investigation of heat transfer on the outer surface of polymeric hollow fibers. *Materiali in tehnologije*. 2018, **52**(4), 459-463. DOI: 10.17222/mit.2016.221. ISSN 15802949. Dostupné také z: http://mit.imt.si/Revija/izvodi/mit184/bartuli.pdf

[4] LEWIS, R., P. NITHIARASU a K. SEETHARAMU. *Fundamentals of the finite element method for heat and fluid flow*. Hoboken, NJ: Wiley, 2004, xi, 341 p. ISBN 978-0-470-84789-3.

[5] FERZIGER, Joel a Milovan PERIć. *Computational methods for fluid dynamics*. 3rd rev. ed. Berlin: Springer-Verlag, 2002, xiv, 423 s. ISBN 35-404-2074-6.

[6] BARTULI, E., T. BROZOVA a M. RAUDENSKY. The Influence of Hollow Fibers Orientation Inside the Polymeric Hollow Fiber Heat Exchanger on the Heat Transfer Intensity. *Engineering Mechanics 2018*. Prague, 2018, , 61-64.

[7] RAUDENSKÝ, Miroslav, Ilya ASTROUSKI a Miroslav DOHNAL. Intensification of heat transfer of polymeric hollow fiber heat exchangers by chaotisation. *Applied Thermal Engineering* [online]. 2017, **113**, 632-638 [cit. 2016-11-23]. DOI: 10.1016/j.applthermaleng.2016.11.038. ISSN 13594311. Dostupné z: http://linkinghub.elsevier.com/retrieve/pii/S1359431116330770

[8] *X-Winder* [online]. b.r. [cit. 2019]. Dostupné z: http://www.xwinder.com/

[9] BROŽOVÁ, Tereza. *Fázové změny na povrchu tepelných výměníků s dutými vlákny*. Heat Transfer and Fluid Flow Laboratory, 2018, 106 s. PhD thesis. Faculty of Mechanical Engineering, Brno University of Technology. Vedoucí práce Prof. Ing. Miroslav Raudenský, CSc.

AUTHOR'S BIOGRAPHY

PERSONAL INFORMATION

Birthday	30.07.1987
Place of birth	Ekaterinburg, Russia
Permanent residence	620043 Ekaterinburg, Zavodskaya 79-8
Correspondence address in ČR	Božetěchova 102, 61200 Brno
Telephone	(+420) 773 218 954
E-mail	Erik.Bartuli1@vut.cz

EDUCATION

2014-2019 **Ph.D., Brno University of Technology,** Faculty of mechanical engineering, Heat transfer and fluid flow laboratory, Department of Engineering mechanics
PhD thesis: Optimization of heat transfer surfaces of heat exchangers

2010-2013 **Ph.D., Institute of Thermal Physics**, Ural branch of the Russian Academy of Sciences, Ekaterinburg, Russia.
Post-graduate course on specialty: Thermal physics and theoretical thermal engineering
Theme: "Investigation of heat-mass exchange and hydro dynamical processes during condensations in constrained conditions"

2008-2010 **Mgr.,** Ural State University, Ekaterinburg, Russia, Physical faculty, Department of General and molecular physics (finished with a state master's exam)
Master's thesis: Investigation of heat transfer during condensation in constrained conditions

2007-2009 Physics teacher, Ural State University, Ekaterinburg, Russia, Faculty of political science and sociology

2004-2008 **Bc.,** Ural State University, Ekaterinburg, Russia, Physical Faculty, Department of general and molecular physics (finished with a state master's exam)
Bachelor's thesis: Investigation of structural properties of nickel capillary-porous structures

COMPUTER SKILLS

Microsoft Office – advanced
Solid Works, AutoCAD, Inventor, – advanced
ANSYS CFX, Fluent – advanced
COMSOL – beginner

LANGUAGE SKILLS

Russian – native
Czech – B2
English – B2

ABSTRACT

The thesis focuses on metal and plastic heat exchangers. The main object of the research is the optimization of heat transfer surfaces for increase of heat transfer characteristics of heat exchangers. A computer modeling via ANSYS and experimental research were used for this purpose. A cross-wiring technology for plastic hollow fiber heat exchangers (PHFHE) producing was developed based on modeling and experimental results. An experimental setup used for pressure tanks winding was modified for PHFHE mass production.

Heat exchangers for air condition systems were also considered in this work. Example of twisted hollow fiber heat exchanger use in such systems is presented. Influence of thermal cycling test to heat transfer performance of common metal fin-type heat exchanger was also studied.

ABSTRAKT

Disertační práce je zaměřena na kovové a polymerní výměníky tepla. Hlavním předmětem zkoumání je optimalizace teplosměnných ploch za účelem zvýšení účinnosti výměníku tepla. Tyto cíle byly dosaženy experimentálně a numericky pomocí modelování v ANSYS. Na základě dosažených výsledků byla rozpracována technologie křížového navíjení polymerních výměníků z dutých vláken. Experimentální zařízení původně určené pro navíjení tlakových nádrží bylo modifikované pro automatizovanou výrobu polymerních výměníků z dutých vláken, ježto může být použita při jejich masové výrobě.

Tato práce se také zabývala výměníky tepla pro klimatizační systémy. Byly zkoumány možnosti využití polymerních výměníků z dutých vláken v těchto systémech. Mimo jiné byla provedena studie vlivu cyklického tepelného zatížení standardního kovového žebrovaného tepelného výměníku.