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NUMERICAL INVESTIGATION OF HEAT TRANSFER ON THE OUTER SURFACE OF POLYMERIC HOLLOW FIBERS

NUMERIČNA RAZISKAVA PRENOSA TOPLOTE NA ZUNANJI POVRŠINI VOTLIH POLIMERNIH VLAKEN

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Polymeric hollow-fiber heat exchangers (PHFHEs) are heat exchangers that use polymeric microchannels as the heat-transfer surface. The main purpose of this work was to determine the optimal arrangement of polymeric hollow fibers in a PHFHE. The object of study was a bundle of parallel polymeric hollow fibers with an outer diameter of 0.8 mm. The heated working fluid flew inside the hollow fibers. The heat was transferred by a cooling medium from the outside of the hollow fibers. Heat-transfer coefficients on the outer surface of the hollow fibers were obtained using computer modelling. For the case of water cooling, a sharp heat-transfer-coefficient increase was observed as the angle α between the hollow-fiber position and the direction of the cooling-medium flow increased to 30°; however, its further increase to 45° had no significant effect. With the water cooling, the total heat-transfer coefficient of the PHFHE increased by 35 % and with the air cooling, it increased by 15 %. The angle of 30° can be considered as the optimal angle for manufacturing such PHFHEs.

Keywords: heat transfer, heat exchanger, polymeric hollow fibers, numerical modelling

Toplotni izmenjevalniki z votlimi polimernimi vlakni (PHFHE; angl.: Polymeric hollow fiber heat exchangers) so izmenjevalniki toplote, ki uporabljajo polimerne mikrokanale kot površine za prenos toplote. Glavni namen pričujočega dela je bil določiti optimalno razporeditev votlih polimernih vlaken v PHFHE. Predmet študije je bil svežen vzporednih votlih polimernih vlaken z zunanjim premerom 0,8 mm. Ogrevano delovno sredstvo (voda ali zrak) se je pretakalo znotraj votlih vlaken. Toplota se je prenašala z ohlajanjem s hladilnim medijem, ki se je nahajal na zunanji strani vlaken. Koeficiente prenosa toplote na zunanji površini votlih vlaken so avtorji tega članka dobili z računalniškim modeliranjem. V primeru vodnega hlajenja so zaznali močno naraščanje koeficienta prenosa toplote, ko je kot med položajem votlih vlaken in smerjo pretoka hladilnega medija α narastel na 30°, nadaljnje naraščanje kota do 45° pa ni imelo pomembnega vpliva. Z vodnim hlajenjem je celotni koeficient prenosa toplote PHFHE narastel za 35 %, pri zračnem hlajenju pa za 15 %. Avtorji so kot optimalen za proizvodnjo takšnih PHFHE ocenili kot 30°.

Ključne besede: prenos toplote, toplotni izmenjevalnik, votla polimerna vlakna, numerično modeliranje

1 INTRODUCTION

Nowadays the use of polymeric heat exchangers is becoming more and more popular in different fields, such as heat-recovery system, evaporative cooling system, desiccant-cooling system, electronic-device cooling and water-desalination system.[1] D. Rousse et al.[2] presented the theoretical and experimental analysis of using a plastic heat exchanger as a dehumidifier in a greenhouse for the agriculture industry.

Due to the corrosive resistance of the polymer, heat exchangers can be used in the chemical industry where aggressive media are used with metals. L. Jia et al.[3] present an experimental study on the heat-transfer performance of a wet flue-gas heat-recovery system, using a plastic, longitudinal, spiral-plate heat exchanger.

Another important advantage of a polymer heat exchanger is that its production uses twice less energy than the production of metal heat exchangers. W. Liu et al.[4] tested two types of heat exchangers (tube-in-shell and immersed tube) made either from high-temperature nylon or cross-linked polyethylene for solar water-heating systems. Authors reported that the polymer heat exchangers can provide thermal outputs equivalent to the conventional copper heat exchangers at a lower cost. The cost of a nylon tube-in-shell heat exchanger is about 80 % of that of a copper tube-in-shell heat exchanger.

Other significant advantages of plastic heat exchangers are low weight, smooth surface, simplicity of shaping and producing, resistance to fouling.[5] Most commercially available polymer heat exchangers can be used in low-temperature applications.[6]

One type of polymer heat exchanger is the polymeric hollow-fiber heat exchanger (PHFHE).[7] Using thin-wall polymeric hollow fibers as the heat-exchanger tubes, it was first proposed by D. Zarkadas et al.[8] 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 mm. The heat exchanger is made of hundreds of such fibers, resulting in a very large heat-transfer area compared to the size of

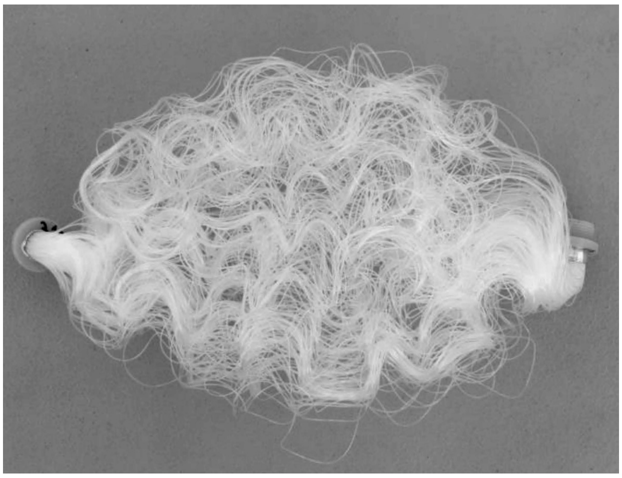

Figure 1: Twisted polymeric hollow-fiber bundle

the entire heat exchanger (**Figure 1**). This design allows the PHFHE to compete with metal heat exchangers in terms of heat-transfer intensity. I. Krasny et al.[9] conducted comparative tests of PHFHE and a conventional automobile aluminum finned-tube radiator and observed that the heat-transfer rates (up to 10.2 kW) and the overall heat-transfer coefficients (up to 335 $W/(m^2 K)$) of the PHFHE were competitive with the metal heat exchanger.

The theory of the heat transfer inside a polymeric hollow fiber has been well-described.[10] However, the description of the heat-transfer processes on the outer surface of the fibers is more complex and currently still has not been sufficiently explored.

This work was devoted to the numerical modelling of the heat transfer on the external surface of polymeric hollow fibers. The main purpose of this work was to determine the optimal configuration of polymeric hollow fibers in a PHFHE allowing it to achieve the maximum heat-transfer coefficients on the external surface of polymeric hollow fibers and of the total device as a whole.

The hollow-fiber position with respect to the direction of the cooling-medium flow, determined by the angle α, was examined as a variable parameter. **Figure 2** shows a real PHFHE with the angle α of about 20°.

2 EXPERIMENTAL PART

The polymeric hollow-fiber heat exchanger studied in this paper was designed and manufactured in our Heat Transfer and Fluid Flow Laboratory (**Figure 3a**). It is based on the tube-in-tube type heat exchanger. Inside the outer tube is a polymeric hollow-fiber bundle made of polypropylene. Individual fibers were weaved with a thread that made it possible to put them in order (**Figure 3b**). Hot water is used as the working fluid inside the fibers. Outside of these fibers, heat is removed by a cold medium, which can be liquid or gaseous and should be compatible with polypropylene. In this work, pairs of working fluids of water-water and water-air were considered.

The tested heat exchanger contained about 1800 hollow fibers. Due to the large number of fibers, the simulation of the whole internal volume of the heat exchanger took a lot of computing power. Therefore, a rectangular parallelepiped domain with a square cross-section of 10 mm × 10 mm and a length of 100 mm was selected as the object of investigation. The fibers in the model, which were considered to be hollow cylinders, passed through the cooling medium's domain. The outer diameter of each fiber was 0.8 mm and the interaxial distances between adjacent fibers were 2.4 mm. As the fiber wall temperature at this length changes slightly in reality, in the model, it was set as a constant temperature of 60 °C. Air and water were chosen as the cooling media with the velocities equal to 0.3 m/s and 2 m/s, respectively. 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.

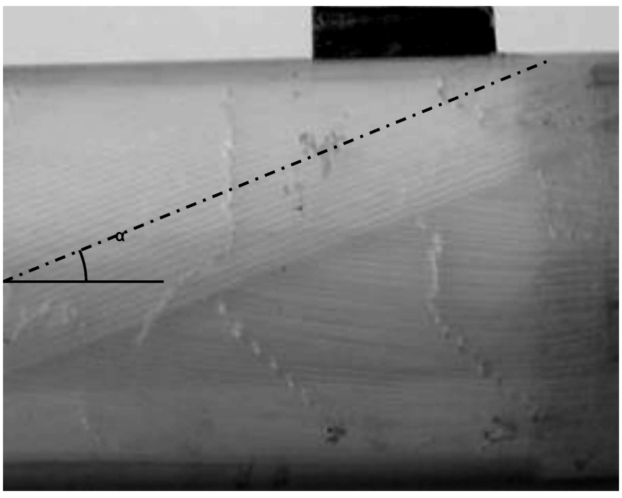

Figure 2: Angle α in a real PHFHE

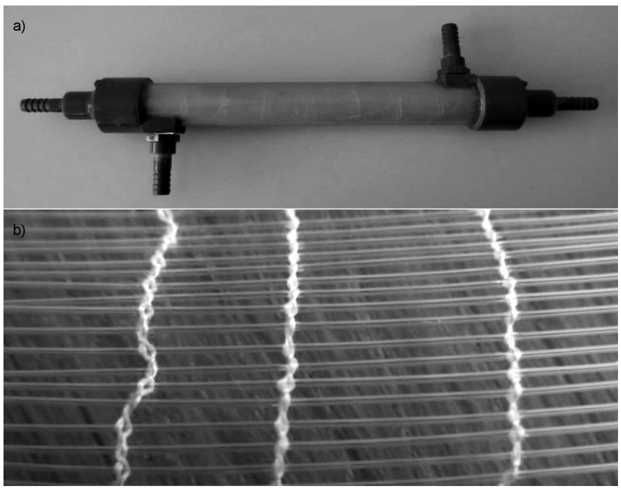

Figure 3: Polymeric hollow-fiber heat exchanger

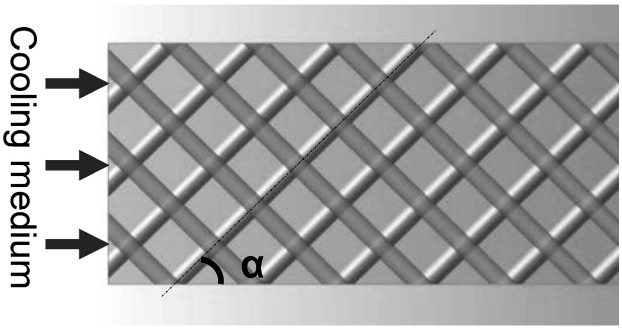

Figure 4: Angle α between hollow fibers and the cooling-medium flow

Angle α in the model was set according to **Figure 4**. The largest value of the angle α that can be set in a real heat exchanger is 45°. Therefore, in this simulation, this parameter varied from 0° to 45°.

The heat-exchange process was simulated using the finite-volume method in ANSYS CFX.[11,12] Because the aim of the work was to compare the heat-transfer coefficients for the hollow-fiber surfaces, the domain size and boundary conditions for each case were identical.

The mesh was set using the ANSYS meshing program. For α = 0°, the mesh had quadro elements. The number of these elements was equal to 3.7 mL. One main parameter for the mesh quality is the Y^+ coefficient. It represents the non-dimensional wall distance for the wall-bounded flow and can be described with Equation (1):

$$Y^+ = \frac{u \cdot y}{v} \tag{1}$$

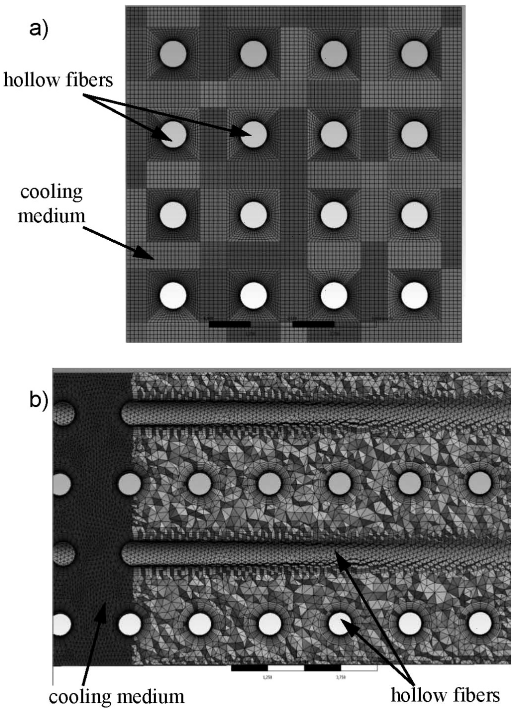

Figure 5: Computational mesh for the cases of: a) α = 0° and b) α = 45°

where u is the friction velocity at the wall, y is the distance to the wall and v is the local kinematic viscosity of the fluid. For a heat-transfer calculation, Y^+ near the wall must usually be less or equal to 1. When α = 0°, Y^+_{max} is equal to 0.94.

When α > 0°, a mesh with tetra elements was used in the volume and the near-wall layer was defined by the quadro elements. For different values of α, the amount of elements varied from 5 to 7 million and the Y^+_{max} never exceeded 0.98. General views of each mesh are shown in **Figure 4**.

At angle α = 0°, R_e < 1000 and the water-cooling flow regime was laminar. A laminar model for modelling this flow was used for this case. The heat from the fiber outer surface was transferred to the cooling medium only by the thermal conductivity of the water.

For α > 0°, the SST-turbulence model was selected as a combination of the k-ε and k-ω models. The k-ω model was used for simulating the near-wall layer turbulence. The turbulence in the flow volume is described by the k-ε model.[13] The LES and DES tabulating models facilitate a simulation of the flow turbulence in more details, but they require significant computing-power resources and for the case under discussion, they are redundant.[14]

For all the angle-α values, the air-cooling flow regime was turbulent and the SST-turbulence model was used.

3 RESULTS

3.1 Total thermal resistance of the heat exchanger

Equation (2) was used for the calculation relating to the heat-transfer coefficient and the thermal resistance was inversely proportional:

$$R = \frac{1}{h} \tag{2}$$

where h is the heat-transfer coefficient. Total thermal resistance of the heat exchanger can be presented as a sum of the resistances:

$$R_{total} = R_{in} + R_{wall} + R_{out} \tag{3}$$

where R_{in} is the thermal resistance of the heat transfer from the hot fluid inside the fiber to its wall, R_{wall} is the thermal resistance of the fiber wall, R_{out} is the thermal resistance of the heat transfer from the fiber wall to the cooling medium outside the fiber.

Each fiber is a capillary tube and the heat exchange inside the tubes with a small diameter is very intensive (**Figure 5**), so R_{in} is a relatively small part of R_{total} and attempts for its further reduction do not have a significant impact on R_{total}.

R_{wall} depends on the wall thickness. For this PHFHE, hollow fibers with a wall thickness of 0.1 mm were used. This value was determined on the basis of the strength properties of the fibers. A further reduction in the wall

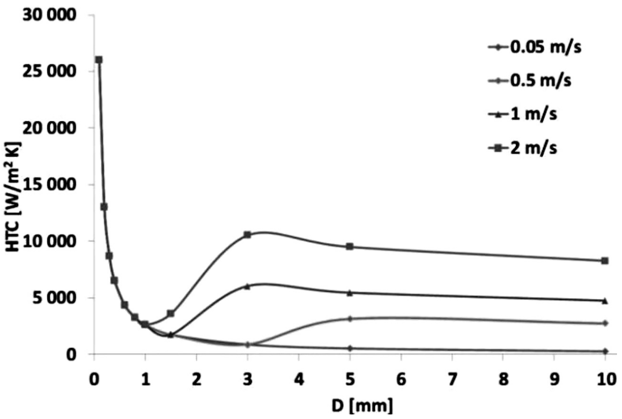

Figure 6: Dependence of heat-transfer coefficients inside the tubes on the tube diameter

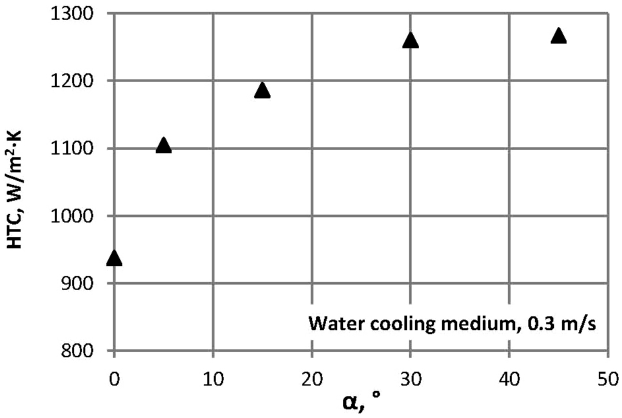

Figure 8: Total heat-transfer coefficient of the hollow-fiber heat exchanger vs angle α

thickness would have adversely affected the reliability of the heat exchanger. So, for this type of PHFHE, value R_{wall} could be considered as a constant.

That is why the main attention in this paper is focused on the intensity of the heat transfer on the outer surface of the fibres (R_{out}). The contribution of R_{out} to the total thermal resistance of the heat exchanger at $\alpha = 0°$ is about 35–40 %.

3.2 Water-cooling medium

The computer-simulation results indicating the dependence of the heat-transfer coefficient on the outer-fiber surface on the angle α between the cooling-flow direction and the fibers is shown on **Figure 6**. The graph shows that even an increase in angle α by up to 5° results in a twofold increase in the heat-transfer coefficient. This fact is explained with the change of the laminar cooling-flow regime in the heat exchanger with the angle of $\alpha = 0$ to the turbulent cooling-flow regime for the exchangers with $\alpha > 0$. Hollow fibers in the water flow and the water start to intermix, leading to an intensification of the heat exchange on the outer surface of the hollow fibers.

It can be seen from the graph that heat-transfer coefficients increase linearly when angle α changes from 0° to 30°. Further increases in α, from 30° to 45°, lead to a very slight increase in the heat-transfer coefficients. A decrease in the growth rate of the heat-transfer coefficients can be explained with the regime of advanced turbulence. The maximum heat-transfer coefficient was 10,563 W/(m^2 K).

Figure 7 illustrates the dependence of the total heat-transfer coefficient of the whole heat exchanger on angle α. Even an increase in α by 5° increases the heat-transfer coefficient by nearly 20 %. At the maximum angle of $\alpha = 45°$, the heat-transfer coefficient is increased by 35 % and is 1267 W/(m^2 K).

3.3 Air-cooling medium

For the air-cooling medium, the main contribution to the total thermal resistance of the heat exchanger is the resistance on the outer wall of the fibers (R_{out}) due to poor air conductivity. **Figure 9** shows that the heat-transfer coefficient on the outer surface of the fibers is almost equal to the total heat-transfer coefficient of the heat exchanger. At $\alpha = 0°$, the air flow is already turbulent, so

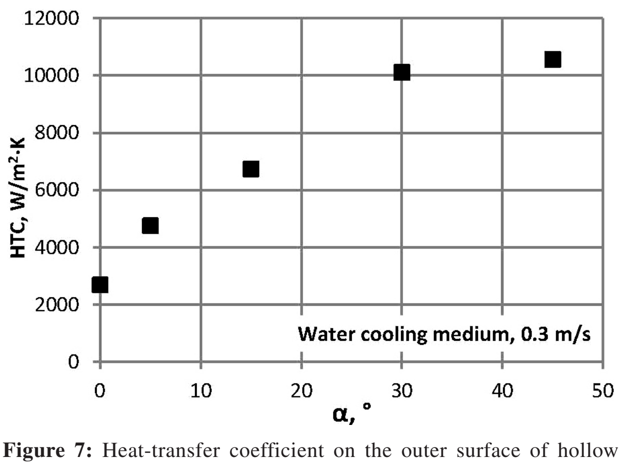

Figure 7: Heat-transfer coefficient on the outer surface of hollow fibers vs angle α

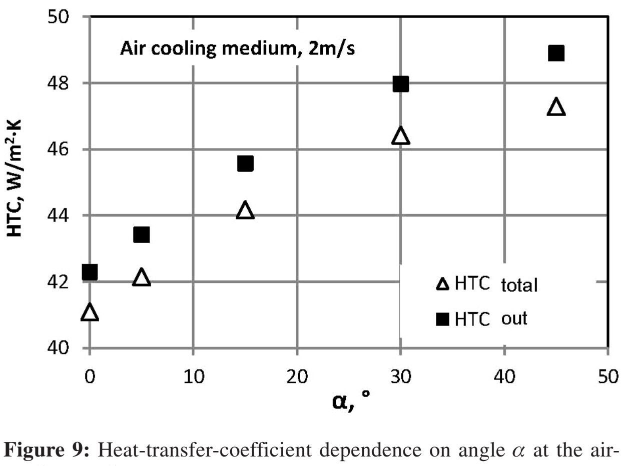

Figure 9: Heat-transfer-coefficient dependence on angle α at the air-cooling mediu

an increase in angle α does not greatly increase the heat-transfer coefficient. When angle α changes from 0° to 45°, the total heat-transfer coefficient changes from 41.1 W/(m^2 K) to 47.3 W/(m^2 K).

4 CONCLUSIONS

A computer simulation was used for determining the optimal arrangement of the polymeric hollow fibers in a PHFHE. Air and water were examined as cooling mediums.

It was found that with the water cooling, even an increase by 5° in the angle α between the fiber axis and the cooling-medium flow direction leads to a twofold increase in the heat-transfer coefficient on the outer surface of the hollow fibers.

A sharp heat-transfer-coefficient increase was observed as angle α increased to 30°. Changing angle α to 45° had no significant effect. The angle of 30° can be considered as optimal for manufacturing such PHFHEs.

The maximum value of the heat-transfer coefficient on the outer surface of the fibers with water as the cooling medium was 10,563 W/(m^2 K). The maximum heat-transfer coefficient of the full PHFHE was 1267 W/(m^2 K). These values are comparable to the ones achieved in metal heat exchangers.

For the air cooling, a linear increase in the heat-transfer coefficient was observed when angle α was increased to 45°. The total heat-transfer coefficients of the PHFHE increased by 15 % when angle α changed from 0° to 45°. However, the process of heat exchange with air is non-intensive, so the heat-transfer coefficients increased from 41.1 W/(m^2 K) to 47.3 W/(m^2 K).

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