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# THE STUDY OF POLYMERIC HOLLOW FIBER HEAT EXCHANGERS

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**Abstract:** *The polymeric hollow fiber heat exchanger (PHFHE) is a modern type of apparatus which uses polymeric fibers, with a small diameter around 1 mm, for separation of the heat transfer mediums. The main goal of this work is to study different factors which affect heat transfer in polymeric hollow fibers (diameters, length and material of fibers, liquids temperatures and velocities) and to obtain conclusions concerning hollow fiber application. The values of external, internal as well as overall heat transfer coefficient, heat transfer rate, number of transfer units (NTUs), efficiency and pressure drops were obtained for both water-water and water-air applications. The special performed Delphi-based software was prepared to accelerate the computation process.*

**Keywords:** ***Polymeric hollow fibers, heat exchanger, heat transfer.***

## 1. Introduction

The first attempts to use polymers as a material for heat exchange equipment started over 40 years ago because of some benefits in comparing them with conventional metals (Whitley, 1957). Polymers as common are less expensive and easier to shape, form and machine than metals. Moreover, the energy required to produce a unit mass of plastics is about 2 times lower than that of common metals, making them environmentally attractive. The smoothness of the polymer surface begets low friction, hence a drop in pressure as well as to good fouling characteristics. Long-term fouling data of polymer heat exchangers is not available, but 76 h test with hard water was performed by Githehens (1965). Polymers have excellent chemical resistance and moreover, their hydrophobic surface promotes dropwise instead of filmwise condensation and, consequently, much higher heat transfer coefficients (Bigg et al., 1989).

In accordance with Malik (2005) most of the commercially available polymer heat exchangers are being used in a corrosive environment or in low temperature applications (ice storage or solar heating of domestic hot water and swimming pools). In particular, plastic heat exchangers were also used for heat recovery in greenhouses (Rousse et al., 2000), in superfluid Stirling refrigerators or 3He-4He dilution refrigerators (Patel & Brisson, 2000). Lately, an interest in the application of polymeric materials in solar water heating systems has also emerged (Tsilingiris, 2000). Also a lot of different fields of plastic heat exchangers application (in the desalination industry, heat recovery, cooling and cryogenic industry, humidification, solar energy industry, microelectronics and computer industry) were quoted by Zaheed (2004).

On the other hand, polymer materials thermal conductivity is low, usually between 0.1 and 0.4 W/m·K which is 100-300 times lower than that of metals and considerably limits the use of polymers for heat exchanges equipment because of big magnitude of wall thermal resistance (Zarkadas & Sirkar, 2004). In order to overcome this deficiency two approaches exist. The first one is to increase thermal conductivity of material. Excellent review of current state of the art of polymers utilizing was performed by T'Joen et al. (2008). In this review the material properties of polymers and polymer matrix composites were examined. It was shown that these materials do hold promise for use in the construction of heat exchangers, but that a considerable amount of research is still required into material properties and life-time behavior. The status of worldwide research in the thermal

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conductivity of carbon nanotubes (CNT) and their polymer composites was done by Han & Fina (2011) as well as Guarded Hot Plate Method of increasing the through-thickness thermal conductivity of CNT composites was described by Han & Chung (2011). The investigation of finned-tube heat exchangers produced from pure and modified polypropylene was performed by Chen (2008). For the heat exchangers designed in that paper, when the plastic thermal conductivity can reach over 15 W/m·K, it can achieve more than 95% of the titanium heat exchanger performance and 84% of the aluminum or copper heat exchanger performance with the same dimension. Using of increased conductivity polypropylene was investigated by Qin et al. (2012). Heat-conducting medium was developed by melt-mixing polypropylene with graphite particles (PP-g-MA). The overall heat transfer coefficient of graphite modified polypropylene hollow fiber heat exchangers reached 1228.7 W/($m^2$K) and the overall conductance per unit volume $1.1 \cdot 10^6$ W/($m^3$K).

The second approach to decrease wall thermal resistance is to use thin walls between heat transfer mediums. The study of different construction thin wall polymeric heat exchangers was done by Scheffler (2008). Cross-flow plate heat exchangers made of crosscorrugated films of poly(ether ether ketone) (PEEK) were used to study air-air and water-water flow configurations by Zaheed (2004). The heat transfer and the hydrodynamic response of microcapillary films (MCFs) within round and elliptical channels (diameter 30 up to 500 μm) were investigated by Hornung et al. (2005). MCFs are a class of novel, extrusion-processed, polymer films containing an array of continuous, parallel, capillaries that run along the film's length. These studies were added by Hallmark et al. (2005).

Polymeric hollow-fiber based heat exchangers (PHFHEs) are also a type of thin-wall polymer heat exchangers, which were firstly proposed by Zarkadas & Sirkar (2004) as a useful alternative for lower temperature applications. The small devices containing a few hollow PP-based fibers with the liquids in parallel flow at temperatures up to 74°C were studied. The overall heat transfer coefficients of these devices were 647-1314 W/$m^2$K for the water-water application. As more, proposed heat exchangers had very low values of the height of transfer, large numbers of transfer units (NTUs) for so comparably short devices, high values of heat exchanger effectiveness and overall conductance/volume rate. However, a number of important questions remained unanswered. Part of them were answered by Song et al. (2010) which study a lot of different PHFHEs with emphasis of application for thermal desalination processes and Qin et al. (2012) which study PHFHEs constructed from the modified thermal conductivity polymers.

## 2. Theoretical consideration

As usual polymeric hollow fibers and films have internal diameters around 0.05 – 2 mm and can be classified as the so-called microdevices (Herwig, 2001). The development of micro-mechanics during the last decades stimulated a great interest in heat transfer studies in micro-channels. A lot of theoretical and experimental investigations devoted to this problem were performed. There is no convincing explanation of the difference between experimental and theoretical results for laminar flow, and between experimental and semi-empirical results for turbulent flow (Yarin et al., 2009). On the one hand, several researchers argue that some new effects exist in micro-channels, e.g., Tso & Mahulikar (2000), Gad-el-Hak (2003). On the other hand, the phenomenon can be related to the discrepancy between the actual conditions of a given experiment and theoretical or numerical solution obtained in the frame of conventional theory (Herwig 2001; Herwig & Hausner 2003).

Thus, there are some additional factors which possible can influence the heat transfer in microchannels can be determined in comparing with conventional theory: dissipation effects which determined by Eckert number (or Brickman number in accordance with Tso & Mahulikar (2000)), axial conduction which determined by Peclet number, conjugate effects and variable properties effects (Herwig, 2001). However, in accordance with Song et al. (2010) and Zarkadas & Sirkar (2005), the axial heat conduction, flow work and viscous dissipation are negligible for laminar flow in polymeric hollow fibers. Two different methods of determining the temperature profile inside the polymeric hollow fiber heat exchanger or the inside, outside and wall heat transfer coefficients were proposed. These methods are a simplified correlation suggested by Hickman and a rigorous solution of the extended Graetz problem by Hsu (Zarkadas & Sirkar, 2004). The simple relationship to calculate internal mean Nusselt number of thermal developing region was proposed by Zarkadas & Sirkar (2005) based on Hickman's one and incremental heat transfer number calculated by Hsu's approach.

**2.1. Calculation of fiber heat transfer performance**

Firstly, the physical properties of the air and water were calculated based on inlet temperatures of liquids. The formulas obtained from ThermalSpreadsheets were used for determine the specific heat, dynamic viscosity, kinematic viscosity, thermal conductivity and density of air:

$$Cp = 1.22295 \cdot 10^{-10}\, T^4 - 5.35621 \cdot 10^{-7}\, T^3 + $$
$$+ 8.27169 \cdot 10^4 T^2 - 0.295423 \cdot T + 1032.1 \quad (1)$$

$$\mu = 5.41836 \cdot 10^{-15} T^3 + 2.52496 \cdot 10^{-11} T^2 + 5.87333 \cdot 10^{-8} T + 2.80339 \cdot 10^{-6} \quad (2)$$

$$\nu = -1.14681 \cdot 10^{-14} T^3 + 8.87916 \cdot 10^{-11} T^2 +$$
$$+ 4.55037 \cdot 10^{-8} T + 5.43395 \cdot 10^{-6} \quad (3)$$

$$k = \frac{0.0316 - 0.0243}{100} \cdot t + 0.0243 \quad (4)$$

$$\rho = 353.179/T \quad (5)$$

where $t$ and $T$ are temperatures on a scale of Celsius and Kelvin respectively. The results of calculations in adequate range were compared and matched with data from Incropera & DeWitt (1996) so these formulas were chosen for use. The required physical properties of water were taken from the NIST Chemistry WebBook. The Prandtl and Reynolds numbers were calculated as followed:

$$Pr = \frac{c_p \cdot \mu}{\lambda} \quad (6)$$

$$Re = \frac{u \cdot D}{\nu} \quad (7)$$

where $c_p$, $\nu$ and $\mu$, $\lambda$ are specific heat, kinematic and dynamic viscosities, thermal conductivity respectively.

The outside and inside diameters of fibers $D_o$ and $D_i$, the outside bulk flow velocity $u_o$ and average inside velocity $u_i$ were used respectively for calculation of outside (around the fibers) and inside (lumen) Reynolds numbers. Average inside velocity was calculated as followed:

$$u_i = \frac{Q_{s,t}}{0.25\pi \cdot N \cdot D_i^2} \quad (8)$$

where $Q_{s,t}$ and $N$ are volumetric flow rate through the tube (fiber) side and number of fibers respectively. The outside Nusselt number was calculated in according with Hilbert formula for single circular tube in cross-flow (Incropera & DeWitt, 1996):

$$Nu_o = C \cdot Re^m \cdot Pr^{0.3333} \quad (9)$$

where $C$ and $m$ are constants were determined in accordance with respective value of Reynolds number (Incropera & DeWitt, 1996). The outside convective heat transfer coefficient was calculated as:

$$h_o = \frac{Nu_o \cdot k_o}{D_o} \quad (10)$$

where $k_o$ is thermal conductivity (W/m·K) of external liquid (water or air with the relevant temperature).

The inside Nusselt number was calculated in accordance with asymptotic solution for thermal developing region proposed by Hickman (Song et al., 2010):

$$\frac{1}{U_w} = \frac{D_i}{D_o \cdot h_o} + \frac{D_i}{2k_w} \cdot ln\left(\frac{D_o}{D_i}\right) \quad (11)$$

$$Nu_w = \frac{U_w D_i}{k_i} \quad (12)$$

$$Nu_{T3} = \frac{(48/11) + Nu_w}{1 + (59/220) Nu_w} \quad (13)$$

where are $k_w = 0.18$ W/m·K is thermal conductivity of isotactic polypropylene which quoted by Mark (1999) in the 0.12 – 0.22 range. Equation (13) yields $Nu_{T3}$ values that fall between 3.66 and 4.364. The lower limit of this Nusselt number range is the limiting Nusselt number corresponding to the constant wall temperature boundary condition ($Nu_w = \infty$) and the upper limit is the limiting Nusselt number corresponding to the constant heat flux boundary condition ($Nu_w = 0$) (Song et al., 2010). This simplified formula doesn't take into account the influence of developing flow region, but have enough good accuracy for relatively long ducts (hollow fibers of small diameters, for example). Thus, the internal heat transfer coefficient was calculated based on internal Nusselt as:

$$h_i = \frac{Nu_{T3} \cdot k_i}{D_i} \tag{14}$$

where $k_i$ is tube liquid thermal conductivity. Linear thermal resistance and linear overall heat transfer coefficient were calculated as followed:

$$R_l = \frac{1}{D_o \cdot h_o} + \frac{\ln(D_o/D_i)}{2k_w} + \frac{1}{D_i \cdot h_i} \tag{15}$$

$$h_l = \frac{1}{R_l} \tag{16}$$

Furthermore, to estimate the influence of local thermal resistances on overall one the inside, wall and outside local to overall resistance ratios ($R_i/R_l$, $R_w/R_l$ and $R_o/R_l$) were calculated:

$$R_i/R_l = \frac{1}{D_o \cdot h_o}/R_l \cdot 100\% \tag{17}$$

$$R_w/R_l = \frac{\ln(D_o/D_i)}{2k_w}/R_l \cdot 100\% \tag{18}$$

$$R_o/R_l = \frac{1}{D_i \cdot h_i}/R_l \cdot 100\% \tag{19}$$

Overall heat transfer coefficient was calculated based on heat transfer area of outside surface of fibers as:

$$U_{ov} = \frac{h_l}{D_o} \tag{20}$$

To obtain a thermal performance characteristics and predictions of outlet liquid temperatures the effectiveness-NTU method was used. Both outside $C_o$ and inside $C_i$ heat capacity rates of liquids were calculated as:

$$C = C_p \cdot \rho \cdot Q_f \tag{21}$$

where $C_p$, $\rho$, $Q_f$ are isobaric specific heat, density and volumetric flow rate respectively, and heat capacity ratio:

$$C_r = \frac{C_{min}}{C_{max}} \tag{22}$$

where $C_{min} = \text{minimum}(C_o,\ C_i)$ and $C_{max} = \text{maximum}(C_o,\ C_i)$. The number of transfer units (NTU) was obtained by the following relationships:

$$NTU = \frac{h_{ov} \cdot A}{C_{min}} \tag{23}$$

where $A$ is a heat transfer area of outside surface of fibers and calculated based on external surface of fibers:

$$A = \pi \cdot D_o \cdot l \cdot N \tag{24}$$

Because tube liquid flows through a large amount of small diameter fibers we must assume that tube liquid is unmixed. On the other hand the outside liquid flows without separation and so should be considered mixed. If the outside flow heat capacity rate value $C_o$ was bigger than inside one $C_i$ then effectiveness was calculated as (Incropera & DeWitt, 1996):

$$\varepsilon = \left(\frac{1}{C_r}\right)(1 - exp\{-C_r[1 - exp(-NTU)]\}) \tag{25}$$

And vice versa, for bigger inside unmixed flow heat capacity rate value $C_i$ effectiveness was calculated as:

$$\varepsilon = 1 - exp(-C_r^{-1}\{1 - exp[-C_r(NTU)]\}) \tag{26}$$

The maximum possible and actual heat transfer rate were obtained by

$$Q_{max} = C_{min} \cdot (T_{t1} - T_{s1}) \tag{27}$$

$$Q = \varepsilon \cdot Q_{max} \tag{28}$$

Thus outlet temperatures for inside and outside liquids were calculated as:

$$T_{i2} = T_{i1} + \frac{Q}{C_t} \tag{29}$$

$$T_{o2} = T_{o1} + \frac{Q}{C_s} \tag{30}$$

Excellent review of laminar fluid flow in micro-channels was done by Hestroni et al. (2004). They quoted that in case of hydraulic diameters 50-254 μm and Re < 500 and in case of hydraulic diameters 620-1067 μm and Re = 500..2600 Poiseuille number is independent from Reynolds number and equal to 64 for the laminar flow in circular channels. This statement was extended based on investigations of Maynes & Webb (2002) for all range of calculation. Thus the conventional Poiseuille number was used to obtain a prediction of pressure drops in tube side (fibers). Pressure drop of tube side liquid was calculated as

$$\Delta p_{it} = \frac{128 \cdot \mu_{av} \cdot l \cdot Q_{f,t}}{\pi \cdot D_i^4 \cdot N} \tag{31}$$

where $Q_{f,t}$ is a volumetric flow rate through the tube side (fibers), $l$ is length of fibers, $D_i$ is internal diameter and $N$ is number of fibers. Because of strong dependence of dynamic viscosity of water from temperature the average value of viscosity along the fiber $\mu_{av}$ was used. This value was obtained as mean of ten local dynamic viscosity values calculated for local temperatures of sub-regions. The local temperatures of sub-regions were obtained based on the assumption of linear distribution of temperatures along the channel and predetermined inlet and outlet temperatures. This assumption is rough for heat transfer calculation (Herwig & Hauser, 2003) but sufficiently accurate for calculation of local dynamic viscosity.

## 3. Results and discussion

As was previously mentioned the calculating program was performed to calculate heat transfer values and pressure drop of hollow fiber in cross-flow. The work window of calculating program is presented in Figure 1. The input data include external and internal diameter, length of fibers and height of cross-section in which they are located, total number of fibers, material of fibers wall, types and temperatures of liquids inside and outside of fibers, volumetric flow rate for inside liquid and bulk velocity around the fibers for outside one.

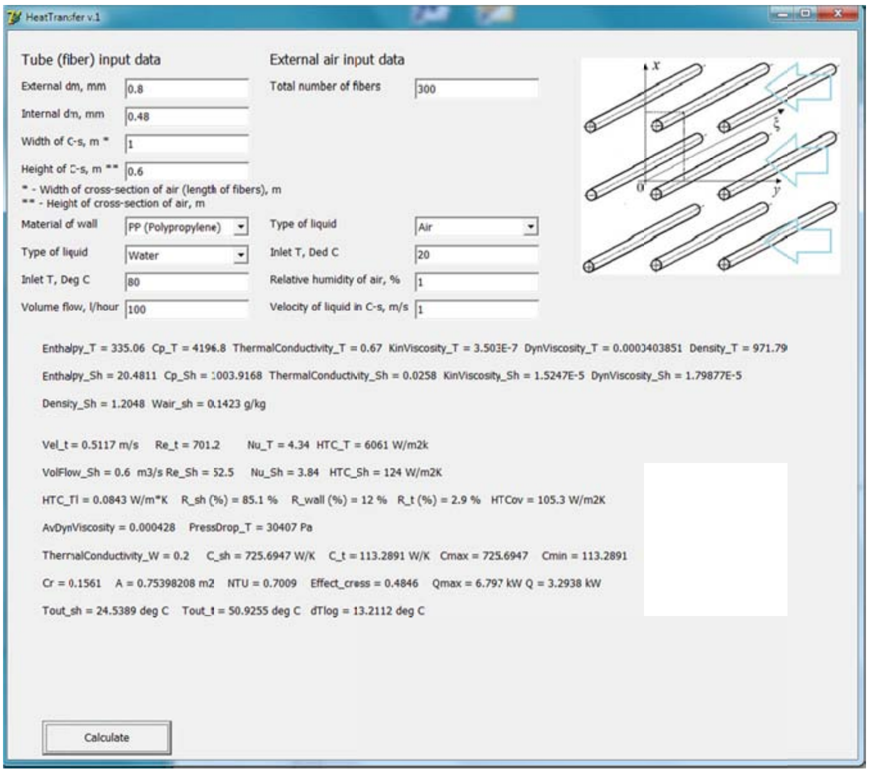


*Fig. 1: Work window of calculating program*

The output data include the physical properties of liquids, inside tube velocity and outside volumetric flow rate, Reynolds numbers, Nusselt numbers and heat transfer coefficients for both inside and outside, linear heat transfer coefficient, overall heat transfer coefficient, local to overall resistance ratios and pressure drop in tube (fiber) side. The NTU-analysis section has the following output date: heat capacity rates of both flow, heat capacity ratio, heat transfer area based on external surface of fibers, NTU, effectiveness, actual rate of heat transfer, inside and outside liquid output temperatures. The algorithm of calculation and formulas used were described before in Calculation subsection. Program was used to determine different factors magnitude of the influence on heat transfer by varying input parameters. Moreover a number of different fibers comparisons were performed.

### 3.1 Influence of liquid velocity around the fibers

Overall heat transfer coefficient (HTC_Ov, W/m$^2$K), linear heat transfer coefficient (HTC_l, W/m·K) and heat transfer rate (Q, kW) graphs are presented in the Figure 2 and 3 for both water-water and water-air applications. The multiply coefficients (for example 1/10, 100, 1000 etc.) were used to present different magnitude values in same graphs here and further during the article. To determine outside liquid velocity degree of influence on this parameters it was varied in the range 0.005 – 2 m/s and 0.005 – 20 m/s respectively for outside water and air. Please take note that the kinks in the graphs exist because of utilizing non-linear scale on horizontal axis which contains values of external liquid velocities. Other input conditions were constant and presented in captions of figures. It can be seen that velocity of cross-flow water around the fibers has no strict influence on heat transfer performance and even at very low water velocities (0.005 m/s, for example) values of heat transfer coefficient is not inapplicable small (see Figure 2). This conclusion isn't true for water-air application (see Figure 3) because of external wall-air thermal resistance has the main influence on the overall heat transfer rate and has a strong dependence from velocity of air.

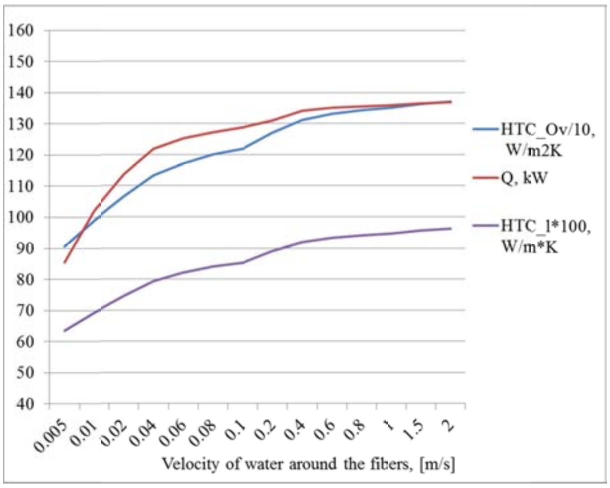


*Fig. 2: Influence of external water velocity. 5400 polypropylene fibers with length 0.6 m and external/internal diameters 0.7/0.56 mm are in water cross-flow. Velocity of water inside is 1 m/s, external/internal inlet temperatures of water are 20°/50° C.*

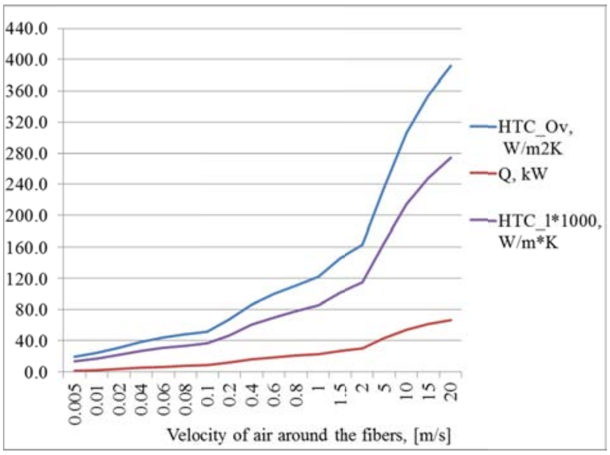


*Fig. 3: Influence of external air velocity. 5400 polypropylene fibers with length 0.6 m and external/internal diameters 0.7/0.56 mm are in air cross-flow. Velocity of water inside is 1 m/s, external/internal inlet temperatures of air are 20°/50° C.*

### 3.2 Comparison of different diameter of fibers (for constant external surface and constant mass of fibers)

An amount of fibers which needed to create a 1 m$^2$ of heat transfer area was calculated. The external diameter of fibers was varied in the range 0.1 – 2.0 mm and other conditions were the constant. Please take note that the internal diameter of fibers was 80% of external one in all calculations described in this article because it was providing the constant value of wall thermal resistance $\ln(D_o/D_i)/2k_w$ during all calculations except of different conductivity cases. Overall heat transfer coefficient (HTC_Ov), linear heat transfer coefficient (HTC_l), heat transfer rate (Q), pressure drop and number of fibers graphs are presented in Figure 4. The working conditions are described in the caption of figure. It can be seen from graphs that linear heat transfer coefficient has very small dependence on diameter and the same length of different diameter fibers has around the same heat transfer possibility. Big difference in overall heat transfer coefficient is based on big difference in external surface of different diameter fibers. Other side 1 $m^2$ of heat transfer surface which created by small fibers has bigger heat transfer possibility then the surface created from bigger ones. Thus, it can be concluded that the main advantage of small fibers is a possibility to use more fibers with bigger total length. However, the pressure drops also is much bigger for small fibers and it is important factor which bordered the using of relatively small fibers. Please take note that the kinks in the graphs exist because of utilizing non-linear scale on horizontal axis which contains values of external diameters.

The same conclusions as previous can be done based on comparison of different fibers which can be produced from 1 kg of material. As before the external diameter of fibers and number of fibers was varied and other conditions were the constant and described in the Figure 5 caption. Graphs of the overall heat transfer coefficient (HTC_Ov), linear heat transfer coefficient (HTC_l) and pressure drop is very similar as in previous case but the number of fibers (and total heat transfer rate Q) is different. As well as it is evident that using of smaller fibers gives a possibility to use less of material for construction of heat exchange equipment.

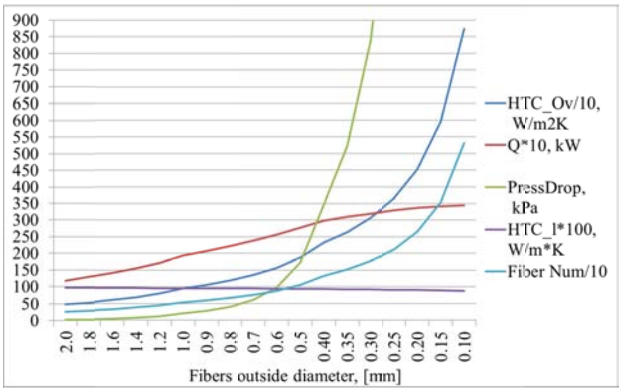


*Fig. 4: Comparison of polypropylene fibers with different diameters and constant total external surface. Water-water cross-flow with volumetric flow rate in tube side 1000 l/h, external flow velocity 1 m/s, external/internal inlet temperatures of water are 20°/50° C.*

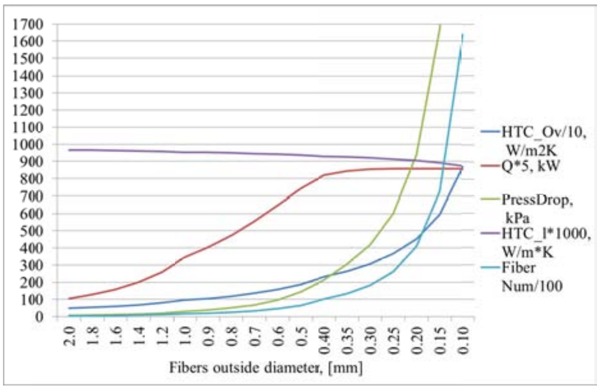


*Fig. 5: Comparison of polypropylene fibers with different diameters and constant mass of material. Water-water cross-flow with volumetric flow rate in tube side 5000 l/h, external flow velocity 1 m/s, external/internal inlet temperatures of water are 20°/50° C.*

### 3.3 Influence of wall material thermal conductivity

To study the influence of wall material thermal conductivity two types of fiber materials were studied. The first one was isotactic polypropylene with thermal conductivity $k = 0.18$ W/m·K (Mark, 1999) and the second was arbitrary material with conductivity 2 W/m·K (around the 11 times higher). The graphs of local to overall thermal resistance ratios and linear heat transfer coefficient are presented in Figures 6 and 7 for both materials (polypropylene and material with conductivity 2 W/m·K). *R_o/R_l, R_w /R_l* and *R_i/R_l* are, respectively, outside, wall and inside to overall linear thermal resistance ratios which were calculated in accordance with formulas (17) – (19). The constant test conditions were followed: fibers in water-water cross-flow, volumetric flow rate inside is 5000 l/hour, velocity around the fibers is 0.05 m/s, external/internal inlet temperatures of water are 20°/ 50° C. The external diameter was varied in 2.0 – 0.4 mm range. It can be seen that the wall thermal resistance plays the main role in the overall one (around the 55 %) in polypropylene application case. This conclusion can be generalized to a lot of other polymers, which have similar thermal conductivity values. The opposite situation takes a place in the case of 2 W/m·K conductivity material (see Figure 7). The wall resistance in this case has a minimal influence on overall one and heat transfer performance can be increased by intensification internal and external convection. Moreover it can be seen that by increasing (from 0.18 to 2 W/m·K) material conductivity eleven times the linear heat transfer coefficient doubles.

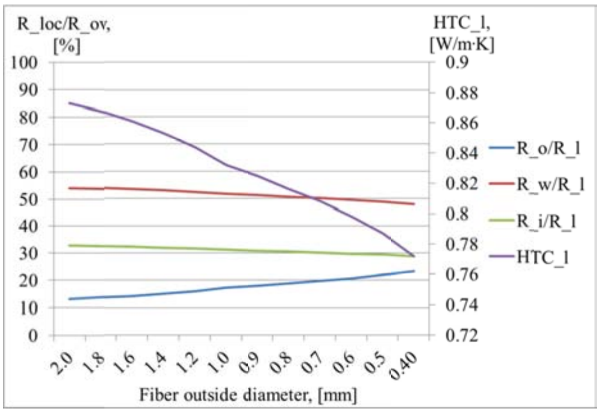


*Fig. 6: Local to overall resistance ratios and linear heat transfer coefficient graphs (polypropylene fibers)*

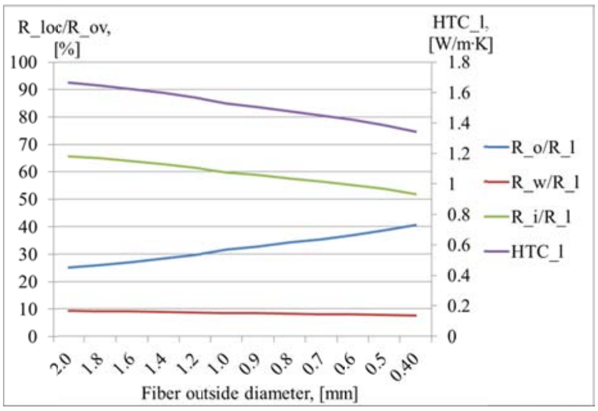


*Fig. 7: Local to overall resistance ratios and linear heat transfer coefficient graphs (fibers from increased conductivity material)*

### 3.4 Comparison of different fibers for water-air and water-water application

The main goal of this section is to compare three diameters of fibers for water-to-air heat exchanger design. The standard performance and working conditions of heat exchanger were chosen for comparison: summary heat transfer rate of fibers should be 29 kW, pressure drops is less the 20 kPa, velocity of air around the fibers 2 m/s, inlet temperatures of water/air are 90°/30° C, material of fibers is isotactic polypropylene with density 0.9 g/cm$^3$ and thermal conductivity 0.18 W/m·K (Mark, 1999). The number of fibers needed, their diameters, length and inside volumetric flow rate were varied such a way that pressure drop, total heat transfer rate and tube volumetric flow rate were similar to required. The results obtained are partly presented in the Table 1. There are the following columns in the table: external diameter *Dex*, a length of fibers *L*, volumetric flow rate trough single fiber *Qfi*, velocity of liquid in fiber *Vi*, inside, outside, linear and overall heat transfer coefficients (*HTCi*, *HTCo*, *HTCl* and *HTC*), efficiency *E*, total heat transfer area *A*, heat transfer rate of single fiber *Q*, inside outlet temperature *Toi*, pressure drops in tube side *dP*, number of fibers *N*, total volumetric flow rate *Qf*, total mass of fibers *M*. They were combined into three groups in accordance with efficiency values: the first one is around 0.15, the second – 0.34, the third – 0.54. It is evident that the smaller values of efficiencies were obtained for lower tube volumetric flow rates and correspond to smaller temperature difference of tube water. The comparison of linear heat transfer coefficients, heat transfer rate, number and mass of fibers for two values of efficiency (0.15 and 0.54 respectively) is presented in Figures 8 and 9. It shows that the linear heat transfer coefficient (and thus thermal performance) is slightly better for bigger fibers (with 0.8 mm outside diameter). The heat transfer rate is several times higher for these fibers based on its larger length and so less fiber needed to obtain required heat transfer capacity. We can see also that bigger fibers are longer for the same pressure drop limit and permit to pump bigger amount of heat transfer medium (they have relatively smaller pressure drops). Other side the small fibers (with 0.4 mm outside diameter) have bigger values of overall heat transfer coefficients and smaller values of total surface needed. As more by utilizing of small fibers the same heat transfer rate can be released in smaller volume (Song et al., 2010). Thus created on the basis of smaller fibers heat exchangers should have a tendency to be more compact and lighter.

*Tab. 1: Summary of calculation results for water-air application*

| *Dex, mm* | *L,m* | *Qfi, l/h* | *Vi, m/s* | *HTCi, W/m²K* | *HTCo, W/m²K* | *HTCl, W/m·K* | *HTC, W/m²K* | *E* | *A,m²* | *Q,W* | *Toi,°C* | *dP, kPa* | *N* | *Qf, l/h* | *M, kg* |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0.8 | 0.60 | 1.20 | 1.04 | 4572 | 171 | 0.121 | 151 | 0.149 | 3.61 | 12.1 | 81.1 | 15.3 | 2397 | 2876 | 0.234 |
| 0.6 | 0.40 | 0.74 | 1.14 | 6101 | 199 | 0.107 | 179 | 0.144 | 3.02 | 7.2 | 81.1 | 19.9 | 4028 | 2981 | 0.148 |
| 0.4 | 0.20 | 0.29 | 1.00 | 9160 | 247 | 0.090 | 228 | 0.155 | 2.44 | 3 | 80.7 | 19.8 | 9667 | 2803 | 0.079 |
| 0.8 | 1.05 | 0.82 | 0.71 | 4572 | 171 | 0.121 | 151 | 0.339 | 4.07 | 18.8 | 69.7 | 19.9 | 1543 | 1265 | 0.264 |
| 0.6 | 0.65 | 0.42 | 0.64 | 6101 | 199 | 0.107 | 179 | 0.359 | 3.49 | 10.2 | 68.4 | 20.1 | 2843 | 1194 | 0.169 |
| 0.4 | 0.30 | 0.18 | 0.62 | 9160 | 247 | 0.090 | 228 | 0.334 | 2.67 | 4.1 | 70 | 19.9 | 7073 | 1273 | 0.086 |
| 0.8 | 1.40 | 0.56 | 0.48 | 4572 | 171 | 0.121 | 151 | 0.553 | 4.87 | 21 | 56.8 | 20.1 | 1381 | 773 | 0.315 |
| 0.6 | 0.80 | 0.31 | 0.48 | 6101 | 199 | 0.107 | 179 | 0.524 | 3.98 | 11 | 58.6 | 19.8 | 2636 | 817 | 0.193 |
| 0.4 | 0.40 | 0.12 | 0.41 | 9160 | 247 | 0.090 | 228 | 0.556 | 3.24 | 4.5 | 56.6 | 19.7 | 6444 | 773 | 0.105 |

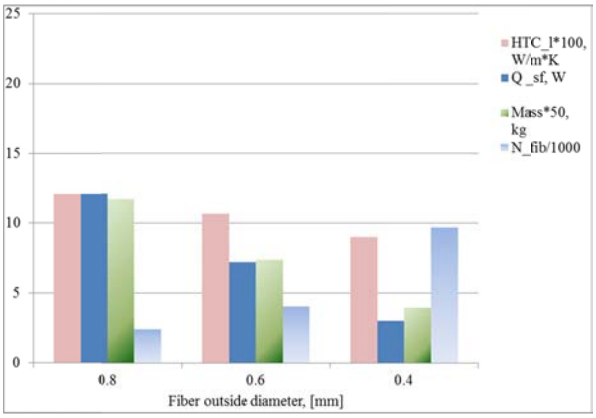


*Fig. 8: Three diameters of fibers comparison for $E \approx 0.15$ ($Dex_1$= 0.8 mm $L_1$=0.6 m, $Dex_2$=0.6 mm $L_2$=0.4 m, $Dex_3$ = 0.4 mm $L_3$=0.2 m)*

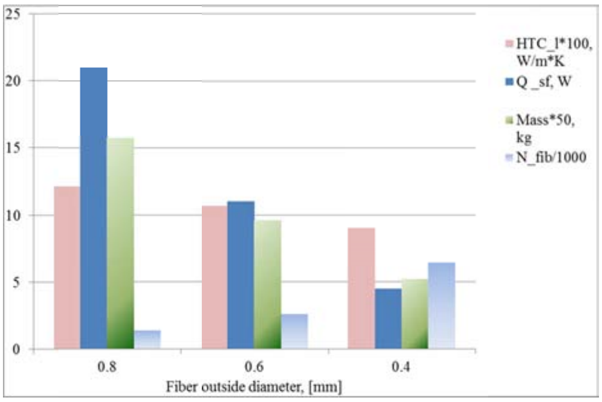


*Fig. 9: Three diameters of fibers comparison for $E \approx 0.53$ ($Dex_1$= 0.8 mm $L_1$=1.4 m, $Dex_2$=0.6 mm $L_2$=0.8 m, $Dex_3$ = 0.4 mm $L_3$=0.4 m)*

The same work was done for water-water application. Performance and working conditions were following: summary heat transfer rate of fibers should be 29 kW, pressure drops was less than 60 kPa, velocity of water around the fibers was 0.05 m/s, inlet temperatures of inside/outside water were 4°/30° C, material of fibers was isotactic polypropylene. The results obtained are presented in Table 2. They were grouped in accordance with efficiency values: the first was 0.65, the second – 0.95, the third – 0.99. The comparison of linear heat transfer coefficients, heat transfer rate, number and mass of fibers for two values of efficiency (0.65 and 0.99 respectively) is presented in Figures 10 and 11. Around the same conclusions as for water-air application can be done for water-water application: the linear heat transfer coefficient is similar for 0.8, 0.6 and 0.4 mm outside diameters fibers, the bigger heat transfer rate value allows to use less of fibers but it's based on large length of fibers, fibers with larger diameter permit to pump more liquid through tube side (have smaller relative pressure drops), the smaller fibers have bigger values of heat flux and contribute to improving the compactness.

*Tab. 2: Summary of calculation results for water-air application*

| Dex, mm | L,m | Qfi, l/h | Vi, m/s | HTCi, W/m²K | HTCo, W/m²K | HTCl, W/m·K | HTC, W/m²K | E | A,m² | Q,W | Toi,°C | dP, kPa | N | Qf, l/h | M, kg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0.8 | 0.60 | 1.19 | 1.03 | 3674 | 5701 | 0.791 | 989 | 0.6570 | 1.84 | 23.7 | 21.1 | 60.0 | 1224 | 1456 | 0.120 |
| 0.6 | 0.35 | 0.650 | 1.00 | 4906 | 6617 | 0.771 | 1286 | 0.6710 | 1.43 | 13.3 | 21.4 | 60.10 | 2180 | 1417 | 0.070 |
| 0.4 | 0.15 | 0.300 | 1.04 | 7371 | 8491 | 0.747 | 1868 | 0.6330 | 0.95 | 5.8 | 20.4 | 60.90 | 5000 | 1500 | 0.031 |
| 0.8 | 1.05 | 0.74 | 0.64 | 3674 | 5701 | 0.791 | 989 | 0.9500 | 3.58 | 21.4 | 28.7 | 59.7 | 1355 | 1003 | 0.232 |
| 0.6 | 0.6 | 0.410 | 0.63 | 4906 | 6617 | 0.771 | 1286 | 0.9510 | 2.77 | 11.8 | 28.7 | 59.80 | 2458 | 1008 | 0.135 |
| 0.4 | 0.25 | 0.195 | 0.67 | 7371 | 8491 | 0.747 | 1868 | 0.9230 | 1.66 | 5.5 | 28 | 60.40 | 5273 | 1028 | 0.054 |
| 0.8 | 1.35 | 0.59 | 0.51 | 3674 | 5701 | 0.791 | 989 | 0.992 | 5.53 | 17.8 | 29.8 | 60.5 | 1629 | 961 | 0.359 |
| 0.6 | 0.75 | 0.33 | 0.51 | 4906 | 6617 | 0.771 | 1286 | 0.9900 | 4.13 | 9.9 | 29.8 | 59.5 | 2929 | 967 | 0.201 |
| 0.4 | 0.35 | 0.140 | 0.48 | 7371 | 8491 | 0.747 | 1868 | 0.9920 | 3.05 | 4.2 | 29.8 | 59.50 | 6905 | 967 | 0.098 |

## 4. Conclusions

As it shown in Introduction section polymer heat exchangers have a lot advantages and problems concerned with its application are slowly addressing in different studies of polymers. Thus we can assume significant improvement of polymer heat exchanger performance and their wider application in different fields. We have studied theoretically some factors affected heat transfer of polymeric hollow fibers with respect to cross-flow water-water and water-air application. Performed conclusions and comparisons show tendencies which have place in polymer fibers heat transfer and can help to choose adequate fibers and flow conditions.

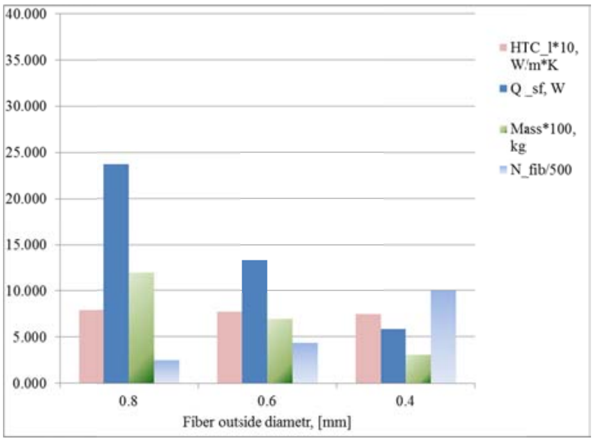


*Fig. 10: Three diameters of fibers comparison for E ≈ 0.65 ($Dex_1$= 0.8 mm $L_1$=0.6 m, $Dex_2$=0.6 mm $L_2$=0.35 m, $Dex_3$ = 0.4 mm $L_3$=0.15 m)*

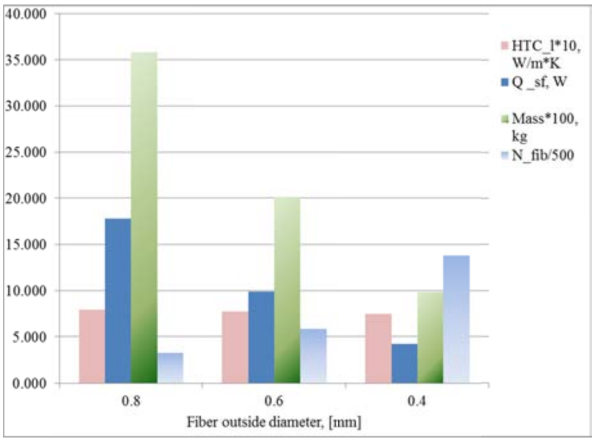


*Fig. 11: Three diameters of fibers comparison for E ≈ 0.99 ($Dex_1$= 0.8 mm $L_1$=1.35 m, $Dex_2$=0.6 mm $L_2$=0.75 m, $Dex_3$ = 0.4 mm $L_3$=0.35 m)*

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