20th International Conference
ENGINEERING MECHANICS 2014
Svratka, Czech Republic, 12 – 15 May 2014

HEAT TRANSFER SIMULATION OF HEAT EXCHANGERS MADE BY POLYMERIC HOLLOW FIBERS

V. Hřibová[*], J. Komínek*, I. Astrouski*, M. Raudenský*, A. A. Tseng[**]

Abstract: *Heat exchangers have been used for a broad range of industrial applications. Due to its good thermal conductivity and mechanical strength, metal has been widely applied for making the heat exchangers. Recently, however, because of their superior characters, including corrosion resistance, cost-effectiveness, light-weight, high ratio of surface-area to volume, dual transport ability, and less fouling ability, polymeric hollow fibers (PHFs) have been used for fabricating heat exchangers for many applications. The purpose of this paper is to present a versatile numerical model, which can be conveniently used for designing PHF heat exchangers (PHFHEs) and reliably used to predict the thermal characteristics of the heat exchangers, including fluid outlet temperatures, external, internal, and overall heat transfer coefficients, and total heat transfer rate as well as the thermal efficiency*.

Keywords: Numerical model, Heat exchanger, Heat transfer, Polymeric hollow fibers.

1. Introduction

Polymers began to be used in the construction of heat exchangers over 40 years ago (Whitley, 1957). Polymers offer several advantages over metal. Their lower price; ease of shaping, forming and machining; and lower densities are the reasons for their much lower construction, transportation and installation costs. They are environmentally attractive because the energy required to produce a unit mass of plastics is about 2 times lower than that of common metals. Because of the smooth surface of polymers, the friction factors and thus pressure drops are smaller and there is less fouling than with commercial metal tubes. Polymers have excellent chemical resistance to acids, oxidizing agents, and many solvents. Moreover, drop-wise condensation caused by the smooth surface of hydrophobic plastics instead of film-wise condensation leads to a much higher heat transfer coefficient.

The main disadvantage of using polymers is their low thermal conductivity, which is usually between 0.1 and 0.4 $Wm^{-1}K^{-1}$ and thus 100-300 times lower than thermal conductivity of metals. This limits the use of polymers for heat exchangers due to high magnitude of wall thermal resistance (Zarkadas & Sirkar, 2004). Additionally, the high thermal expansion of plastics requires special design considerations. On the other hand, this expansion can also be a benefit because repeated expansion and contraction of the plastic tubes can lead to scale detachment (Githens, 1965).

Two main approaches exist to achieve performance comparable with metal heat exchangers. The first is to increase the thermal conductivity of the material and the second one is to decrease the wall thermal resistance by using thin walls between heat transfer mediums. That is why PHFHEs are proposed as a new type of heat exchanger for lower temperature/pressure applications (Zarkadas & Sirkar, 2004).

Besides the advantages mentioned above, PHFHEs are easily and inexpensively formed even into complex shapes, which enables their mass production. They are recyclable, which is a benefit from an ecological point of view.

[*] Bc. Veronika Hřibová, Ing. Jan Komínek, Ing. Ilya Astrouski, prof. Ing. Miroslav Raudenský, CSc.: Heat Transfer and Fluid Flow Laboratory, Faculty of Mechanical Engineering, Brno University of Technology, Technicka 2896/2; 616 69, Brno; CZ; kominek@lptap.fme.vutbr.cz

[**] Dr. Ampere A. Tseng: School for Engineering of Matter, Transport and Energy, 501 E. Tyler Mall, ECG301, Arizona State University, Tempe, AZ 85287-6106 USA; E-mail: ampere.tseng@asu.edu

2. Calculation Method

A simple iterative analytical model is used to calculate the estimation of the heat transfer between the hot fluid in the tubes of the heat exchanger and the air flow in the shell side. It is very important to have an approximation of the behavior of polymers for designing an appropriate PHFHE.

2.1. Equations

For any other considerations the subscript t is used for values related to the tube inside, s for values related to the shell side, i for inputs and o for outputs.

2.1.1. Heat Transfer in the Tube Side

The average Nusselt number of a circular tube at a laminar condition under a constant wall temperature $T_{t,w}$ can be correlated as (Whitaker, 1972):

$$\overline{Nu_t} = \frac{\overline{h_t} d_t}{k_t} = 1.86 \left[\frac{Re_t Pr_t}{(\frac{L}{d_t})}\right]^{1/3} \left(\frac{\mu_t}{\mu_w}\right)^{0.14}, \qquad (1)$$

where $Re_t = \rho_t V_t d_t / \mu_t$ is the Reynolds number and $Pr_t = c_{p,t}\mu_t / k_t$ is the Prandtl number. All properties, density ρ_t, dynamic viscosity μ_t, specific heat at constant pressure $c_{p,t}$, thermal conductivity k_t and velocity V_t, are evaluated at the arithmetic mean of the tube fluid inlet and outlet temperatures $T_{t,am} = (T_{t,mi} + T_{t,mo})/2$. Dynamic viscosity μ_w is calculated at $T_{t,w}$. The above equation (1) can be used if both assumptions $0.48 \leq Pr_t \leq 16.7$ and $0.0044 \leq \mu_t/\mu_w \leq 9.75$ are fulfilled.

If it is true that $[Re_t Pr_t / (\frac{L}{d_t})]^{1/3} (\frac{\mu_t}{\mu_w})^{0.14} < 2$ the Nusselt number can be considered as a constant and the equation (2) can be applied (Incropera & DeWitt, 1996):

$$Nu_t = \frac{h_t d_t}{k_t} = 3.66 \text{ or } h_t = 3.66 \frac{k_t}{d_t}. \qquad (2)$$

The equation (3) (Dittus and Boelter, 1930) should be used for smooth tubes with a fully-developed turbulent flow, i.e. $Re_t > 4{,}000$:

$$Nu_t = \frac{h_t d_t}{k_t} = 0.023 Re_t^{0.8} Pr_t^{0.3}. \qquad (3)$$

The equation (2) can often be used for calculations of PHFHEs because d_t is less than 1 mm, which means the entry region and Re_t are relatively small under normal operating conditions.

The outlet temperature can be found as

$$T_{t,mo} = T_{t,w} + (T_{t,mi} - T_{t,w}) \exp[-\pi d_t L \overline{h_t} / (\dot{m} c_{p,t})], \qquad (4)$$

where L is the length of the tube and $\dot{m} = \rho_t V_t \pi d_t^2 / 4$ is the mass flow rate.

The heat transfer rate from the tube can be found as:

$$q_t = \dot{m} c_{p,t} \left(T_{t,mi} - T_{t,mo}\right) = \overline{h_t} \pi d_t L \Delta T_{t,lm}, \qquad (5)$$

where $\Delta T_{t,lm}$ is the log-mean temperature difference in the tube side:

$$\Delta T_{t,lm} = \frac{(T_{t,w} - T_{t,mi}) - (T_{t,w} - T_{t,mo})}{\ln[(T_{t,w} - T_{t,mi})/(T_{t,w} - T_{t,mo})]}. \qquad (6)$$

The total heat transfer rate from all tubes in the heat exchanger should be:

$$Q_t = N_t q_t, \qquad (7)$$

where N_t is the total number of tubes.

2.1.2. Heat Transfer in the Shell Side

The most common correlation for the average heat transfer coefficient of the air flow across tube bundles in an in-line arrangement is (Zukauskas, 1972):

$$\overline{Nu_s} = \frac{\overline{h_s} d_s}{k_s} = C_{Z1} Re_{s,max}^m Pr_s^{0.36} \left(\frac{Pr_s}{Pr_w}\right)^{0.25}, \qquad (8)$$

where properties k_s, Pr_s and $Re_{s,max}$ are evaluated at the arithmetic mean of air inlet and outlet temperature $T_{s,am} = (T_{s,i} + T_{s,o})/2$ and Pr_w is evaluated at $T_{s,w}$. Calculation of $Re_{s,max}$ is based on the maximum air velocity which is for aligned arrangement approximately equal to $V_{s,max} = P_T V_s/(P_T - d_s)$, where $P_T = p_{tr}/d_s$ and p_{tr} is a distance between centers of the tubes in the perpendicular direction to the air flow.

The equation (8) is valid only for tube banks having 17 or more rows of tubes in the air flow direction ($N_r \geq 17$), otherwise a correction factor C_{Z2} has to be added (Zukauskas, 1972):

$$\overline{Nu_s}|_{N_r<17} = C_{Z2}\overline{Nu_s}|_{N_r\geq 17}. \tag{9}$$

The equation (8) can be used only for $10 \leq Re_{s,max} \leq 100$ and $1.000 \leq Re_{s,max} \leq 2 \cdot 10^6$. For $100 \leq Re_{s,max} \leq 1.000$ the equation (10) for a single tube can be applied (Zukauskas, 1972):

$$\overline{Nu_s} = \frac{\overline{h_s} d_s}{k_s} = 0.51 Re_s^{0.5} Pr_s^{0.37} \left(\frac{Pr_s}{Pr_w}\right)^{0.25}. \tag{10}$$

The values of all the constants can be found in tabular form in most heat transfer books (e.g. Incropera & DeWitt, 1996).

The total heat transfer rate from the tube bank to the air can be calculated as:

$$Q_s = N_t q_s = N_t \overline{h_s} \pi d_s L \Delta T_{s,lm}, \tag{11}$$

where $\Delta T_{s,lm}$ is the log-mean temperature difference:

$$\Delta T_{s,lm} = \frac{(T_{s,w}-T_{s,i})-(T_{s,w}-T_{s,o})}{\ln[(T_{s,w}-T_{s,i})/(T_{s,w}-T_{s,o})]}. \tag{12}$$

Based on the principle of conservation of energy, $T_{s,o}$ can be found as:

$$T_{s,o} = T_{s,w} - (T_{s,w} - T_{s,i})\exp\left[-\frac{\pi d_s N_t \overline{h_s}}{\rho_s V_s N_p p_{tr} c_{p,s}}\right]. \tag{13}$$

Using area of the tube side or area of the shell side, the overall heat transfer coefficients can be evaluated and $U_t A_t = U_s A_s = UA$ (Incropera & DeWitt, 1996):

$$U_t = \frac{1}{\frac{1}{h_t} + r_f^t + \frac{d_t \ln\left(\frac{d_s}{d_t}\right)}{2k_w} + r_f^s\left(\frac{d_t}{d_s}\right) + \frac{\left(\frac{d_t}{d_s}\right)}{h_s}}, \tag{14}$$

$$U_s = \frac{1}{\frac{1}{h_s} + r_f^s + \frac{d_s \ln\left(\frac{d_s}{d_t}\right)}{2k_w} + r_f^t\left(\frac{d_s}{d_t}\right) + \frac{\left(\frac{d_s}{d_t}\right)}{h_t}}. \tag{15}$$

In the equations (14) and (15) r_f represents a fouling factor. Its value is negligible for PHFHEs and does not need to be considered in the present calculation.

2.2. Iterative Model

An iterative model is based on the equations from the above subsection. The superscript denotes the iteration.

Step 1: Start with the assumption that $T_{t,w}^1 = (T_{t,mi} + T_{s,i})/2$ for the first iteration so all h_t^1, $T_{t,mo}^1$ and q_t^1 can be calculated using equations (1-6).

Step 2: Assume $T_{s,w}^i = T_{t,w}^i - q_r \ln(\frac{d_s}{d_t})/(2\pi L k_w)$ and h_s^i, q_s^i and $T_{s,o}^i$ can be calculated using equations (8-13). Note that q_r=q_t for i=1.

Step 3: Evaluate next temperatures of the wall as $T_{t,w}^{i+1} = T_{t,w}^i + T_{t,w}^i(q_t^i - q_s^i)/[s(q_t^i + q_s^i)]$ and $T_{s,w}^{i+1} = T_{s,w}^i - T_{s,w}^i(q_s^i - q_t^i)/[s(q_s^i + q_t^i)]$ and using them, recalculate h_t^{i+1}, h_s^{i+1}, $T_{t,mo}^{i+1}$, $T_{s,o}^{i+1}$, q_t^{i+1} and q_s^{i+1}.

Step 4: If both $\left|T_{t,w}^{i+1} - T_{t,w}^i\right| < \delta$ and $\left|T_{s,w}^{i+1} - T_{s,w}^i\right| < \delta$, where δ is a small non-negative number, stop and go to step 5. Otherwise, $q_r = (q_t^{i+1} + q_s^{i+1})/2$ and i = i+1. Then go to step 2 using the updated $T_{t,w}^{i+1}$.

Step 5: Evaluate total heat transfer rate Q_t, Q_s using equations (7) and (11) and overall heat transfer coefficient U_t and U_s using equations (14) and (15).

The iterative model is developed to find the outlet temperatures $T_{t,mo}$ and $T_{s,o}$, the heat transfer rate Q_t and Q_s and the overall heat transfer coefficients U_t and U_s. In order to preserve the principle of conservation of energy, i.e. $Q_t \approx Q_s$, several iterations are expected to obtain a convergent value of q_r, which should be close to q_t and q_s and $(q_t - q_s)^2$ should be smaller than a preselected allowed error ε.

3. Results

The iterative model was tested for two same proportional PHFHEs with different diameters of fibers. As you can see in Fig. 1 the total heat transfer rate is much better for PHFHE with a smaller diameter if the same proportional PHFHEs used with constant velocity (0.15 m/s) and the same input temperature (60°C) of 50% ethylene-glycol/water and in tubes and the same input temperature of air (20°C) outside. These results confirm opinions mentioned in the first section.

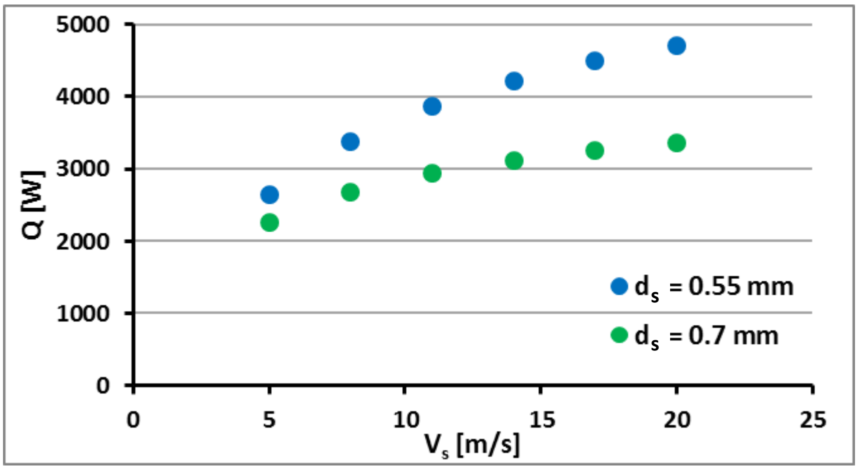

Fig. 1: Dependency of the total heat transfer rate on velocity of air outside.

4. Conclusions

As described in the first section, using polymers for heat exchangers has many advantages but also some problems which require further close study. However, this theoretical iterative model can approximately show the behavior of the PHFHE and processes of heat transfer that take place in polymer fibers. That can help to choose or create the best heat exchanger for the conditions required.

Acknowledgements

The present work has been supported by the European Regional Development Fund within the framework of the research project NETME Centre, CZ. 1.05/2.1.00/01.0002**,** under the Operational Programme Research and Development for Innovation**.**

References

Dittus, F. W., Boelter, L. M. K. (1930) Heat transfer in automobile radiators of the tubular type, University of California Publications on Engineering, 2, 443-461, Berkeley, CA.

Githens, R. E., Minor, R. W., Tomsic, V. J. (1965) Flexible tube heat exchangers. Chem. Eng. Prog., 61 (7), 55.4.

Incropera, F. P. & DeWitt D. P. (1996) Fundamentals of Heat and Mass Transfer, 4th ed., Wiley, New York.

Whitaker, S. (1972) Forced convection heat transfer correlations for flow in pipes, past flat plates, single cylinders, single spheres, and for flow in packed beds and tube bundles, AIChE J., 18(2), 361- 71.

Whitley, D. M. (1957) Plastic heat exchangers gain in severe service. Chem. Eng. 64 (9), 308.

Zarkadas, D. M. & Sirkar K. K. (2004) Polymeric hollow fiber heat exchangers: An alternative for lower temperature applications. Ind. Eng. Chem. Res. 43, pp. 8093-8106.

Zukauskas, A. (1972) Heat transfer from tubes in crossflow, Adv. Heat Transfer, 8, 93-160.