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Thermal Performance Study of Polymeric Hollow-Fiber Heat-Exchangers

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Abstract

Polymeric hollow-fiber heat-exchangers (PHFHEs) have recently attracted great attentions for their superior characters, from light weight to high chemical resistance, as compared with that of metal-based heat exchangers (MBHEs). In this article, a PHFHE, which can be used for car radiators, is built to test its thermal performance in an experimental wind tunnel. A MBHE, which uses stainless steel tubes, has also been built with the major geometrical parameters equivalent to that of the PHFHE tested, so that the results from these two heat exchangers can be compared with each other. Experimental results indicate that the heat flow of the PHFHE can be more than 11% higher than that of the stainless-steel heat exchanger at similar operation conditions.

Keywords: *heat exchanger; polymer hollow-fiber; similarity system, thermal analysis*

Nomenclature

A	area (m^2)
A_r	Aspect ratio
C	geometric parameter
F	factor
N	number of tube
NTU	number of transfer units
Nu	Nusselt number
P	pitch (m)
Pr	Prandtl number
Q	heat flow (W)
Re	Reynolds number
R^2	coefficient of determination
T	temperature (°C)
U	overall heat transfer coefficient (W/m^2-K)
V	velocity (m/s)
c	specific heat (J/kg-K)
d	diameter
h	local heat transfer coefficient (W/m^2-K)
k	thermal conductivity (W/m-K)
$\dot{m}$	mass flow rate

Greek symbols

ε	thermal effectiveness factor
ρ	density (kg/m^3)
μ	viscosity (Pa-s)

Subscripts

h	horizontal direction
I	inner
m	mean or geometric parameter
o	reference or outer
r	ratio or tube row
t	transversal direction or tube side
s	shell side
w	wall

1. Introduction

It is well-known that polymer possesses many attractive engineering properties, such as high corrosion resistance, excellent manufacturability, cost-effectiveness, light-weight, dual transport property, and less fouling ability. To take the advantages of these superior properties, a large number of various types of polymer-based heat exchangers have been developed and adopted by a wide range of industry [1,2]. However, since the thermal conductivity of polymer is much lower than that of metals, polymer heat exchangers have relatively low heat transfer efficiency and less

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popular, as compared with their metallic counterparts.

In order to increase the heat transfer performance, micro-sized polymeric hollow fibers (PHF) instead of macro-sized polymer tubes have recently been used for making heat exchangers. Because PHFs have very thin walls, the fiber wall thermal resistance can be reduced and because a high fiber density can be achieved by using micro-sized fibers, the contact surface area can be enlarged. As a result, the thermal performance in a PHF made heat exchanger can be greatly enhanced [3,4]. Consequently, the adoption of micro-sized PHFs to compensate to the decrease of the thermal performance due to the low thermal conductivity of polymer becomes very attractive for making different types of heat exchangers, which are also known as polymer hollow-fiber heat exchangers (PHFHEs) [3-6].

A cross-flow PHFHE used for engine radiators is fabricated to evaluate its thermal performance. A wind-tunnel type of experimental facilities is used to perform the thermal performance evaluation, where the overall heat transfer coefficients (OHTC) are specifically estimated. Also, a stainless-steel based heat exchanger (SSBHE), which is equivalent to the cross-flow PHFHE fabricated, is built and its thermal performance is experimentally evaluated. By performing a comparison study between the PHFHE and SSBHE, the pros and cons of PHFHEs are established. Finally, conclusions are provided to summarize the finding of the present study and to suggest the directions for future developments.

2. Wind Tunnel Testing of Polymeric Hollow-Fiber Heat-Exchangers

In this section, a cross-flow type of PHFHEs is built and tested to assess its thermal performance in an experimental wind tunnel.

2.1 Fabrication of cross-flow polymeric hollow-fiber heat-exchangers

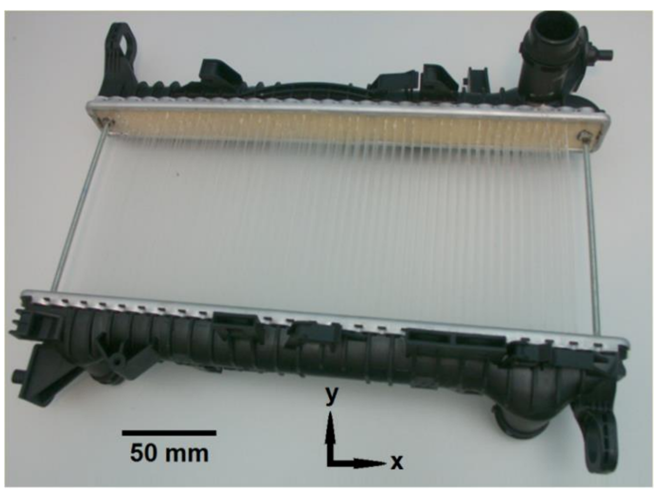

Figure 1. cross-flow PHFHE for wind tunnel test used for auto radiator application
(laid on x-y plane where air flow in z direction)

The cross flow PHFHE shown in Figure 1 was designed and was fabricated for the thermal performance evaluation, where the PHFHE was laid on the x-y plane. The PHFs were made of polypropylene (PP) by a plastic extrusion process with subsequent axial stretching to obtain the required diameter or to increase their strength to satisfy the designed requirement. The outer diameter (d_o) is controlled at 0.50 mm with an inner diameter (d_i) of 0.43 mm, where the corresponding aspect ratio, A_r, which is equal to $d_i/(2t_w)$, is 6.1, where t_w is the fiber wall thickness.

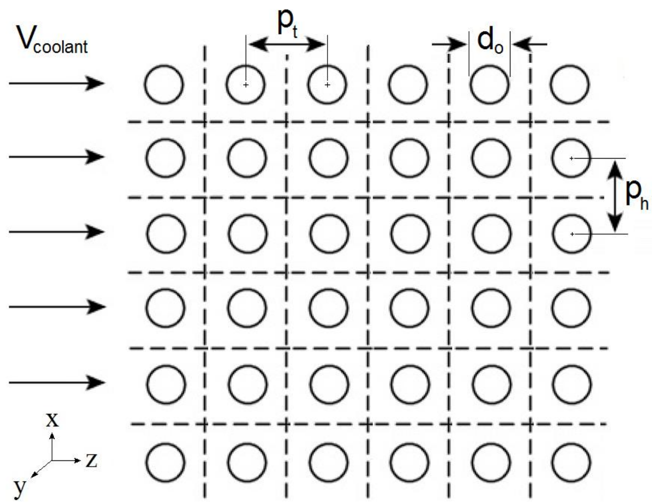

Figure 2. cross-section of aligned hollow-fibers lay-out on z-x plane (air flow in z-direction)

The PHFHE fabricated is a single-pass cross-flow heat exchanger and consists of 510 straight fibers. Using the coordinates defined in Figure 1, all fibers are vertically oriented in y-direction, where 51 fibers are aligned in the horizontal direction (x-direction) and 10 fibers are in the transverse or thickness direction (z direction). The pitches in the horizontal (x) direction (p_h) and in the transverse (z) direction ($p_{t,}$) are all 4.2 mm. Based on the core dimensions, the total effective volume of the PHFHE is 1.71x10^6 mm^3, which is also the design requirement or constraint. The parameters, d_o, p_h, and p_t, are also defined in Fig, 2, which is the z-x cross-section view of the PHFHE. The corresponding geometric parameters are summarized in Table 1, where P_h and P_t are the dimensionless pitches and are normalized by the outer diameter. Moreover, based on the coordinates defined in the figure, the air velocity, V_{air}, is flowing towards z direction during experiment.

The values of the inner and outer diameters reported in Table 1 were the mean of ten measurements from the magnified cross-section photos took from a microscope at different axial locations. The corresponding standard deviation (SD) is approximately 5 % of its mean; such a high SD indicates that the PHFs have relatively large differences in diameter along its length. The cause for the size differences may be due to the non-isothermal temperatures arisen from the fiber extrusion or from the non-uniform axial stretching. Nonetheless, the original PP fibers before extrusion were obtained from Zena Membrane (www.zena-membranes.cz).

Table 1. Geometrical parameters of heat exchangers considered

	Tube count, N	Tube row, N_r	Tube inner dia [mm]	Tube outer dia [mm]	Normalized Pitch, P_t, P_h (p_t/d_o, p_h/d_o)	Aspect ratio, A_r [$d_i/(2t_w)$]	Length [mm]	Exchanger volume [mm^3]
PHFHE*	510	51	0.430	0.500	8.4	6.1	190.0	1.71 x 10^6
SSBHE$^+$	140	28	0.775	0.902	8.4	6.1	212.7	1.71 x 10^6

* Polymeric hollow fiber heat-exchanger; $^+$ Stainless-steel based heat-exchanger.

The PHFHE and PP fibers were tested on leaking by air pressurizing and by immersing to colored water before the wind-tunnel tests. Also, the colored water was used to determine whether the fibers were plugged or not. The unplugged fibers were considered to be functional or active. About 1% of fibers were inactive.

2.2. Wind tunnel testing and thermal performance evaluation

To study the effect of the air speed (vehicle speed in car radiators) on the heat transfer performance, the PHFHE fabricated was loaded on the test chamber of a wind tunnel as shown in Figure 3. In testing, airflow was brought to the desired flow rate (or velocity) and the equipped sensors measured the temperatures of air and coolant at different locations. The coolant or hot fluid in the PHFHE experiment is a 50 vol% ethylene-glycol/water solution. Since its freezing temperature is -37 °C, this solution is widely used as a radiator fluid. The inlet and outlet temperatures, flow rate, and pressure drop were measured by calorimeters, flow meters, and pressure gauges, respectively. The inlet temperatures were maintained around constant in both coolant and air during testing. Temperatures of air were also measured by a separate set of thermocouples placed in the cross-section upstream and downstream of the wind tunnel. The layout of the wind tunnel and sensor locations are shown in Figure 4.

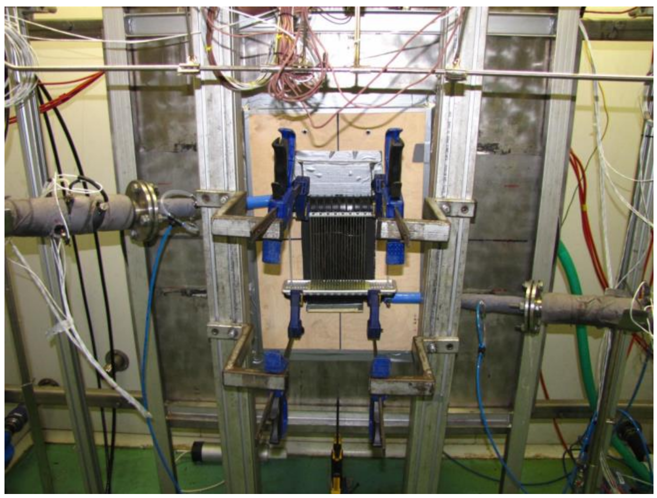

Figure 3. PHFHE fabricated is loaded on test section of wind tunnel for thermal performance assessments

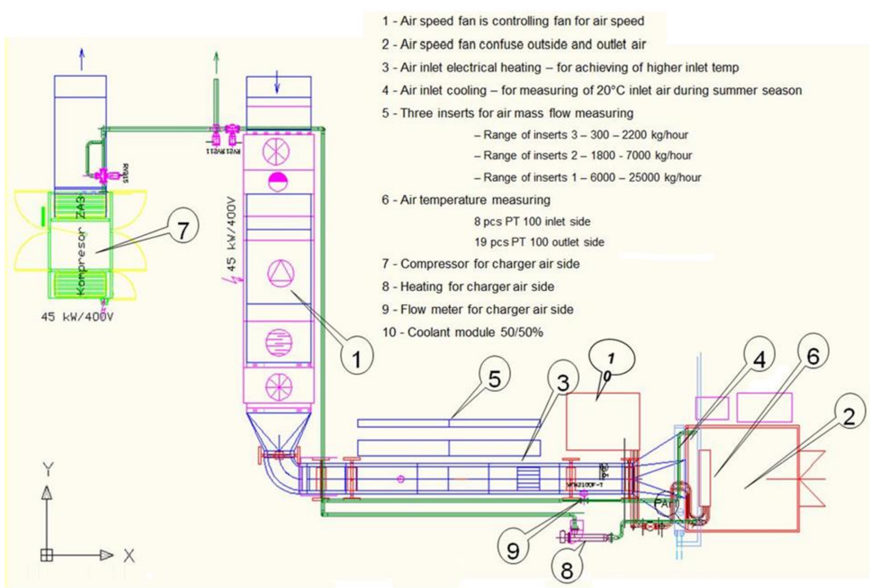

Figure 4. layout of wind turnnel set-up and sensors locations.

During testing, the coolant (hot) fluid inlet temperature was kept at 70°C and the air inlet temperature was controlled at 25°C, while the outlet temperatures of both coolant and air were measured and recorded. The coolant flow rate was kept at 1.56 mL/min (or 2.60×10^{-5} m^3/s), while the air speed was varying from 5 to 40 m/s at several preset testing conditions. The corresponding Reynolds numbers based on the properties reported in Table 2 were calculated. Based on the data obtained from Engineering ToolBox (www.engineeringtoolbox.com), all the material properties listed in Table 2 are extrapolated to the values at the mean temperatures of the fluids during testing. Then, the Overall Heat Transfer Coefficient (OHTC) can be estimated based on Eqs. (1 & 2) described in the next subsection.

Table 2. material properties used in thermal performance calculations*

	k [W/m-K]	ρ [kg/m³]	c [J/kg-K]	μ[Pa-s]	**Pr**=$c_p\mu/k$	T_m [°C]
Coolant	0.412	1037.5	3509	1.052×10^{-3}	8.960	67
Air	0.0262	1.179	1005	1.847×10^{-5}	0.708	27
PP fiber	0.17	900	1800	--		47
SS 314	17.5	7800	500	--		47

* Properties are evaluated at mean temperature (T_m) and based on data reported in www.engineeringtoolbox.com & www.stainless-structurals.com

3. Thermal Performance Evaluations

For a cross-flow heat exchanger, the associated heat flow, Q, can be expressed as [7]:

$$Q = UAF_c(\Delta T_a - \Delta T_b)/\ln(\Delta T_a/\Delta T_b) \quad (1)$$

where U is the Overall Heat Transfer Coefficient (OHTC), A is the heat transfer area, F_c is a correction factor for cross-flow heat exchangers, $\Delta T_a = T_{coolant,in} - T_{air,out}$, and $\Delta T_b = T_{coolant,out} - T_{air,in}$, Here, the subscripts coolant, air, in, and out refer to hot coolant flow, cold air flow, inlet, and outlet temperatures, respectively. The OHTC or U is a measure of the overall ability of a series of conductive and convective barriers (resistances) to transfer heat between two fluids. Since the airflow is relatively fast and the thickness (size in z-direction, see Figure1) of the PHFHE is relatively small, the temperature change of the air flow during testing is negligible and the correction factor can be assumed to be 1 [7,8].

If the mass flow rate of the coolant, $\dot{m}_{coolant}$, is available, the value of Q can be determined by

$$Q = Q_{coolant} = \dot{m}_{coolant} c_{p,coolant}(T_{coolant,in} - T_{coolant,out}) \quad (2)$$

where c_p is the mean specific heat of the coolant. In the present testing, the inlet and outlet temperatures of the cold air and hot coolant are measured directly as described in the preceding section. Then, with the value of F_c equal to 1, U can be computed from E_q. (1) without difficulties.

In general, for heat exchangers with cross-flow at the shell side, F_c normally falls within the range of 0.75 to 1.00. For various shell-and-tube and cross-flow heat exchangers, the values of F_c can be found in many sources, for example, in a typical textbook [7] and a Heat Exchanger Handbook [8]. If the factor value is not available for a special designed heat exchanger, the factor F_c can also be calculated, for example, by a formula provided by Song et al. [9]:

$$F_c = \frac{\mathrm{NTU}}{NTU_c} = \frac{\varepsilon(T_{t,in} - T_{s,in})}{\mathrm{NTU}\,\Delta \mathrm{T_{lm}}} = \ln\left(\frac{1-\varepsilon C_r}{1-\varepsilon}\right)/[NTU(1-C_r)] \quad (3)$$

where NTU is the number of transfer units, NTU_c is the corrected number of transfer units, ε is the thermal effectiveness factor, C_r is the heat capacity rate ratio (should not be 1.0), and the subscripts t and s refer to the tube and shell side temperatures, respectively. The above equation has been used to validate the assumption that $F_c = 1$.

Based on the experimental measurement, the calculated OHTCs against the air Reynolds number (**Re**$_{air}$) are depicted in Figure 5. As shown in Figure 5, the OHTC increases linearly with **Re**$_{air}$ at a constant coolant Reynolds number (**Re**$_{coolant}$ =152.4). Each of the OHTC data points shown in Figure 5 is the mean of 5 experiment measurements based on five wind-tunnel tests at the same testing condition. The corresponding standard deviation (SD) is varying from 3 to 5 % of its mean where the vertically extended "I" bars from the data points represent the magnitude of the standard deviation. It seems that the higher the air Reynolds number, the higher the SD. This may implies that the data deviations are caused by the higher wind speed, i.e., the higher the speed, the larger the SD.

The linear correlation of the data presented and the corresponding coefficient of determination, R^2, of the correlation equation are also reported in Figure 5. The associated coefficient R^2 is 0.9896, which implies the linear correlation fits the data very well and the correlation should be accurate and reliable.

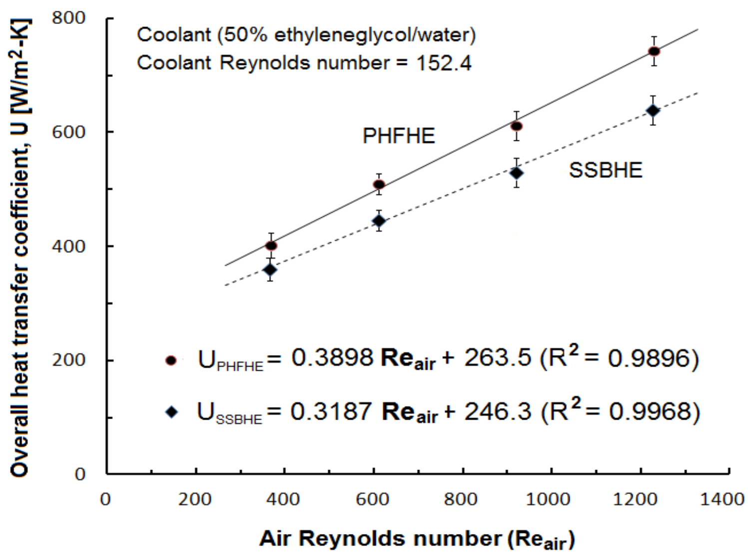

Figure 5. effects of air Reynolds number on overall heat transfer coefficient (U) for two heat exchangers studied.

4. Equivalent Stainless-Steel-Tube Heat Exchanger

In this section, an equivalent SSBHE is fabricated and tested, while a discussion on the equivalent design of heat exchangers is also provided.

4.1 Test setting of equivalent stainless-steel-tube heat exchanger

The SSBHE is made of 140 stainless steel (304) tubes, where twenty-eight tubes along the horizontal (x) direction (Figure 2). The outer and inner diameters are 0.902 mm and 0.775 mm, respectively. These 304 stainless-steel tubes are typically used for making hypodermic needles and are provided by Vita Needle of Needham, Massachusetts (Gauge 20XX, www.vitaneedle.com/small-diameter-tubing.htm). The normalized pitches in both the horizontal and transverse directions are also 8.4. The geometric parameters are also listed in Table 1. Figure 6 shows the image of the equivalent SSBHE, which is used for testing and assessments.

Using the measurement data and the material properties shown in Table 2, the U coefficients were estimated based on Eqs (1 & 2) described in the preceding section. During the wind tunnel test, the coolant flow rate was kept at 0.86 liter/min (1.43 m^3/s), while the air speed was varying from 5 to 40 m/s. The inlet temperatures of the coolant and air were 70°C and 25°C, respectively, same as those set for the PHFHE case. The results of the U coefficient plotted against $\mathbf{Re}_{air}$ are shown in Figure 5 to study the effects of air Reynolds number. As shown, the OHTC also increases linearly with $\mathbf{Re}_{air}$ but at a lower increasing rate (slope); the difference of the OHTC between the PHFHE and SSBHE increases from 11% to 16 % for the $\mathbf{Re}_{air}$ changing from 300 to 1200 at $\mathbf{Re}_{coolant}$ = 152.4. Note that the corresponding linear-correlated equation and the coefficient of determination, R^2, which is 0.9887, for the SSBHE, are also shown in Figure 5. Since R^2 equals 0.9968, the correlation is almost perfect.

4.2. Similarity equivalent

As shown in Table 1, all the governing geometric parameters, including the aspect ratio (A_r), normalized pitches (P_h and P_t), and volume size, of the SSBHE are respectively equal to those of PHFHE. As the result, based on the similarity theory [7,10], if the operation conditions are the same, the heat transfer phenomena involved in these two heat exchangers (PHFHE and SSBHE), which have the geometric similarity, should be equivalent [11,12].

For the sake of discussion, a similarity solution related to the geometry considered is presented here. It is well-known that, based on a great amount experimental data, Grimison [13] developed a correlation to determine the average heat transfer coefficient, $\bar{h}_{air}$, from the tubes in cross flow for a single-pass aligned fiber bundle (tube bank):

$$\overline{Nu}_{d_o} = \bar{h}_{air} d_o / k_{air} = C[P_h \mathrm{Re}_{d_o}/(P_h - 1)]^m \mathrm{Pr}_{air}^{0.33}$$

or

$$\bar{h}_{air} = C k_{air}[P_h \mathrm{Re}_{d_o}/(P_h - 1)]^m \mathrm{Pr}_{air}^{0.33} / d_o \tag{4}$$

where the geometric parameters, C and m, are functions of the normalized pitches, P_h and P_t; $\mathbf{Pr}_{air}$ is the Prandtl number of air; $\mathbf{Re}_{do}$, and $\mathbf{Nu}_{do}$ are the Reynolds, and Nusselt numbers based on the outer diameter, d_o, respectively. The above correlation has three constraints on its suitability, i.e., $N_r \geq 10$, $\mathbf{Pr}_{air} \geq 0.7$, and $2000 \leq [P_h \mathrm{Re}_{d_o}/(P_h - 1)] < 40{,}000$, where N_r is the number of the tube row [7]. Also, for relatively low-temperature applications, such as the one considered here, car radiators, the changes of the Prandtl number are insignificant and the Prandtl number term in Eq. (4) can be dropped. In the present wind-tunnel tests, the air Prandtl number is 0.708 (Table 2) and N_r is larger than 10 (Table 1).

Later, many others, including Zhukauskas [14] and Morgan [15], provided additional correlations to relax the constraint of Reynolds numbers to cover the range from 10 to $2x10^5$ with the basic geometric parameters, C and m, are still functions of P_h and P_t only. Consequently, these correlations discussed above clearly indicate that $\bar{h}_{air}$ for a single pass aligned fiber bundle in cross flow, which is similar to the tube bundle arrangements of the PHFHE and SSBHE considered, should be the same as long as P_h, P_t, $\mathbf{Re}_{do}$ and $\mathbf{Pr}_{air}$ are the same. The dependence of the heat transfer coefficient on P_h and P_t has also been verified by several theoretical studies [16,17]. Therefore, the comparison shown in Figure 5 for the two equivalent heat exchangers considered should be reliable and meaningful for thermal performance assessments.

It is noteworthy that the correlations discussed (Eq. 4, and those developed by Zhukauskas [14] and Morgan [15]) cannot be directly applied to determine the OHTC, because, in developing those correlations, the outer-surface temperature of the tube (fiber) was assumed to be constant (condition associated with evaporation or sublimation). Moreover, $\bar{h}_{air}$ in the correlations is a part of U, which should also include the heat conduction across the fiber walls and the heat convection from the coolant to the fiber inner surfaces. The portions of the wall conduction and inner tube convection should be dependent on the fiber wall thickness (t_w) and the inner diameter (d_i) or be governed by the aspect ratio, A_r, as reported in Table 1.

5. Conclusions

A polymeric hollow-fiber heat-exchanger (PHFHE) has been built to test its thermal performance using a wind tunnel while an equivalent stainless-steel based heat exchanger (SSBHE) has also been built and tested. The thermal performance results of the PHFHE and SSBHE have been compared to each other and have indicated that the overall heat transfer coefficient (OHTC) of PHFHE can be 11 % to 16% higher than that of SSBHE for the operation conditions considered.

Since the thermal conductivity of stainless steel (~15 W/m-K) at room temperature, is about 25 times lower than that of copper (~400 W/m-K), which is a popular metal utilized for making metal based heat exchangers (MBHEs) [16,18,19], the thermal performance of a copper or copper-alloy based heat-exchanger (CBHE) should be expected to be much better than that of PHFHE. As a result, a thermal-performance comparison study between a PHFHE and CBHE should be conducted in the future to provide the benchmark information on the performance of PHFHEs.

It is understood that the PHFHE and SSBHE considered are designed for the convenience to be loaded on the wind tunnel for thermal performance testing and their designs are not optimized for the best thermal efficiency. For example, based on the correlations developed by Grimison [13] and Zhukauskas [14], the U coefficient is dependent on P_h and P_t, and letting $P_h = P_t = 8.4$ is obviously not an optimal design [16-18]. A parametric study of the influence of P_h and P_t on U should be encouraged. Also, the development of the methodology for the optimal design of PHFHEs is important and should also be encouraged, especially, because the design approaches for conventional MBHEs may not be appropriate for PHFHE design [16,20]. Moreover, the analytical software to assess and analyze an optimal PHFHE should be developed in understanding the associated mechanical and transport behaviors, such as temperature, pressure, and velocity variations in different operation conditions as well as the mechanical and structure integrity of the PHFHE in different service temperatures. Furthermore, a more effective way to fabricate PHFHEs should be developed, since a cost-reduction in fabrication could attract more developments of PHFHEs for widening their applications. Currently, the batch manufacturing process used in the present study is certainly not cost effective.

As a final remark, the PHFHE, which is a new type of heat exchanger, has a great potential to provide many industrial applications; it would be beneficial to have a widespread communication and cooperation among the researchers in this field to have a more systematic approach, especially on those recommendations or encouragements mentioned.

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Biographical information

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Professor Miroslav Raudenský graduated from the Faculty of Mechanical Engineering of the Technical University of Brno in 1981 and received his doctor degree in 1992. In 1995, he finished his senior lectureship and was awarded the title associate professor. Professional attention, during the last five years is on new methods of inverse task solving and experimental research of cooling with application, especially, for material processing and is on the design and fabrication of polymeric heat exchanger Author of over hundred papers in journals and at conferences. During the last five years, he was responsible for dealing with research projects for domestic and foreign industrial companies of a financial value of 800.000 €.