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PLASTIC HEAT EXCHANGER USING TWISTED HOLLOW FIBERS

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Abstract: *Metallic heat exchangers have a number of well-known disadvantages, such as heavy weight and cost and low resistance to corrosion. Polymeric hollow fiber heat exchangers were proposed about a decade ago as an alternative for low temperature applications. A large-scale heat exchanger was prepared using hollow fibers with a special, "curly" shape. The heat exchanger's thermal performance was studied using hot (40-90°C) ethyleneglycol-water brine flowing inside of the fibers and cool air (20-30°C) flowing across the fibers. Experiments showed that the heat exchanger can achieve a significant heat transfer rate and have high values of overall heat-transfer coefficients (60-230 $Wm^{-2}K^{-1}$). Moreover, obtained results showed that the classical theory is not suitable to estimate outer convection and that special approach should be used.*

Keywords: Hollow fiber, Plastic heat exchanger, Overall heat transfer coefficient.

1. Introduction

Polymers have many advantages, such as having smooth surface, resistance to corrosion and low weight. They are non-toxic; easy to shape, form and machine; and can be blended with different fillers to obtain additional properties (Han & Chung, 2011). According to Malik & Bullard (2005), most commercially available polymer heat exchangers are used in corrosive environments or in low temperature applications. Many different fields of plastic heat exchanger applications (desalination, heat recovery, cooling and cryogenics, humidification, solar energy, microelectronics and the computer industry) were noted by Zaheed & Jachuck (2004). Using thin-wall polymeric hollow fibers as heat exchanger tubes was first proposed by Zarkadas & Sirkar (2004). Heat transfer of PHFHEs for desalination applications was studied (Song et. al., 2010). It was shown that such devices have overall heat transfer coefficients up to 2100 W $m^{-2}K^{-1}$ and compactness (conductance per volume) up to $3.5{\cdot}10^6$ W $m^{-3}K^{-1}$.

Air heat transfer coefficients (HTC) are low and fins are usually used to increase the air-side heat transfer area in metal heat exchangers. However, polymer thermal conductivity is low so such polymer fins are ineffective and the primary heat transfer area of should be large enough. Hollow fibers are tubes of small diameter which can be closely packed within a small volume. According to Song, the packing density (heat transfer area to volume ratio) of hollow fibers can be as much as 3060 m^2/m^3. However, this means that many fibers need to be connected, uniformly distributed and supplied with fluid. Poor fiber distribution causes a so-called bypassing of liquid and decreases heat transfer. Using twisted fiber bundles is a simple method for distributing the fibers and preparing large-scale hollow fiber heat exchangers.

2. Theory

According to Zarkadas & Sirkar (2004), heat transfer inside a hollow fiber can be considered a conventional convection problem. It should be noted that only laminar-flow solutions are considered because PHFHEs normal operation are going in a laminar regime. Zarkadas & Sirkar (2005) proposed a relationship to calculate the mean Nusselt number of a thermal developing region based on limiting inside Nusselt number and incremental heat transfer number:

$$Nu_{T3} = \frac{\left(\frac{48}{11}\right)+Nu_w}{1+\left(\frac{59}{220}\right)Nu_w} + \left\{0.0499 - \frac{0.06487}{[1+\exp(0.45895\cdot Nu_w+2.46887)]}\cdot(4Gz/\pi)\right\} \quad (1)$$

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The geometrical configuration of twisted fiber bundles is difficult to describe because it is chaotic. Nonetheless, two approaches can be tested to estimate outer heat transfer. The Churchill & Bernstein (1977) relation can be used to calculate a Nusselt number across separate cylinders for a wide range of Reynolds and Peclet numbers ($10^2 < Re < 10^7$, $Pe > 0.2$):

$$Nu_d = 0.3 + \frac{0.62 Re^{1/2} Pr^{1/3}}{\left[1+(0.4/Pr)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{Re}{282{,}000}\right)^{5/8}\right]^{4/5} \tag{2}$$

The heat-transfer characteristics of an in-line bank of tubes can be estimated as proposed by Grimson (1937):

$$Nu = C \cdot Re^m \cdot Pr^{1/3} \tag{3}$$

The values of the constant *C* and the exponent *m* depend on the geometric parameters used to describe the tube-bundle arrangement ($C = 0.317$ and $m = 0.608$ values were used). Reynolds numbers were calculated based on the mean velocity occurring in the frontal cross section.

3. Experimental Study

Fig. 1 shows a hollow fiber bundle (a) and heat exchanger consisting of 4 bundles (b). Its parameters are as follows: the fiber amount is 1880 mm, length is 650 mm, outer/inside diameter is 0.7/0.6 mm, overall heat transfer area based on outer fiber surface is 2.7 m^2 and the frontal air flow area is 0.087 m^2. Thermal performance tests were performed in a certified calorimeter room designed for testing automotive radiators. 50% by volume aqueous ethylene glycol (EG) solution was used as a hot medium circulated inside of the PHFHE and air was used as the cooling media. The inlet and exit temperatures and volume flow rates of the two streams were measured with calibrated equipment and observed in real time (frequency of 1 Hz). Temperatures of air were measured with sets of PT 100 temperature sensors placed in a regular pattern upstream (8 sensors) and downstream (19 sensors) of the heat exchanger. This gave the adequate approximation of air temperature fields to estimate mean temperatures. The inlet temperature of EG was 60°C and air inlet temperature was 20°C in 12 tests and 90°C and 30°C in other 9 tests.

Data reduction was performed as proposed by Zarkadas & Sirkar (2004). The effective mean temperature difference averaged over the total heat exchange area was determined by using the logarithmic mean temperature difference (LMTD) and appropriate LMTD correction factor *F*. The correction factor was calculated as suggested by Jeter (2006). Air-flow was considered as completely mixed and the flow inside the tubes was considered as completely unmixed. Additionally, we used the iterative procedure proposed by Zarkadas & Sirkar (2004) to determine simultaneously both the tube- and air-side heat-transfer coefficients based on our experimental data. In all calculations, liquids properties were evaluated at the average temperature between the inlet and outlet. The wall thermal conductivity needed for the calculations was taken to be equal to 0.18 W m^{-1}K^{-1} for isotactic polypropylene (Mark, 1999).

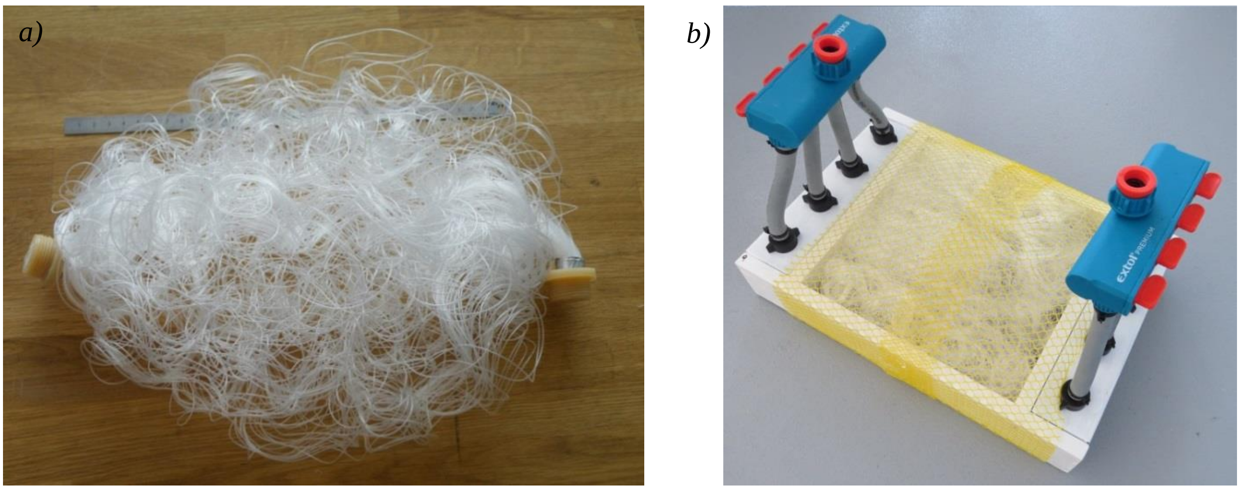

Fig. 1: a) Hollow fiber heat exchange bundle; b) Heat exchanger based on 4 bundles.

4. Results and Discussion

Tab. 1 gives the range of results obtained for the 21 heat-transfer tests performed with PHFHEs. It presents the tube- and air-side Reynolds numbers, the heat-transfer rates, the overall HTCs based on the outside fiber area, the heat exchanger effectiveness, and the number of transfer units (NTU). Tab. 2 shows that the heat-transfer rate of this device can reach high values of up to 21.6 kW (the device is about 30x30x10 cm in size and several hundred grams in weight). Obtained overall HTCs (up to 230 W m^{-2} K^{-1}) are reasonably high compared to finned-tube metal heat exchangers. In the literature (Holman, 2010), design values of 25-55 W m^{-2} K^{-1} are quoted for liquid-gas applications of finned-tube heat exchangers. These values include a total dirt factor which needs to be incorporated in the coefficients quoted in Tab. 1. However, the incorporation of the same fouling factor for polymer surfaces is questionable because limited experimental data about polymeric hollow fiber fouling is available. However, based on extremely high values of overall HTC it can be stated that PHFHEs have high thermal performance. Values of the thermal effectiveness and the NTU are high enough, up to 0.9 and 2.54, respectively. However, some experimental runs showed low NTU and effectiveness because the thermal effectiveness and NTU are functions of the relative heat capacity rates of both streams, overall HTC and heat transfer area. It can be concluded that the accomplishment of high NTUs and efficiency is primarily a problem of proper rating.

Tab. 2 shows the range of tube- and air-side coefficients and the percent contribution of each resistance to the total resistance. It is evident that the tube-side coefficients are very high and air-side coefficients are significantly lower. The results shown reveal that tube-side resistance would be the smallest of the three and that little improvement in the overall heat-transfer performance should be anticipated by increasing the tube-side Re number. The contribution of wall thermal resistance is also low despite the low thermal conductivity of polypropylene. These results agree with the calculated data from Astrouski and Raudensky (2012).

Results of 12 experimental runs (60 ºC inlet temperature of EG) were used to create a plot of overall HTC versus tube- and air Reynolds numbers (see Fig. 2a). It was shown that HTC have small dependence on tube Reynolds numbers, which is in agreement with Equation 1. The same tendency was found for 9 tests with 90ºC inlet temperature of EG. Therefore, PHFHEs can be operated with low tube-side velocities in order to achieve a combination of high thermal performance and low pressure drop provided by low inside velocity. On the other hand, HTC has a strong dependency on the air Reynolds number (see Fig. 2a) because air-side thermal resistance is dominant.

At this point, our experimental results will be examined against the theory presented earlier in the paper. Fig. 2b shows a graph of outer (air-side) HTC versus air Reynolds number. A high discrepancy exists between the experimental data and the theoretical prediction by both the Churchill & Bernstein (1977) and Grimson (1937) models. Both of these models overestimate heat HTC twice or more. This can be explained by the following facts: fiber orientation significantly differs from strict geometrical cross-flow (fibers are twisted and directed in various directions), some fibers are in touch between each other (this decreases effective heat transfer area), and fiber distribution is not uniform enough, so bypassing effects are significant. It can be concluded that a special approach should be developed to define fiber distribution and orientation and predict HTC based on these data.

Tab. 1: Thermal performance of heat exchanger from twisted hollow fibers.

number of runs	Re_t	Re_{air}	Q (kW)	U_o (W m^{-2} K^{-1})	ε	NTU
21	33-407	43-692	2.97-21.6	58-230	0.35-0.90	0.50-2.54

Tab. 2: Tube- and Air-Side Heat-Transfer Coefficients.

h_t (W m^{-2} K^{-1})	h_o (W m^{-2} K^{-1})	R_t/R_{ov} (%)	R_w/R_{ov} (%)	R_o/R_{ov} (%)
2983-3077	58-230	2-7	2-10	84-96

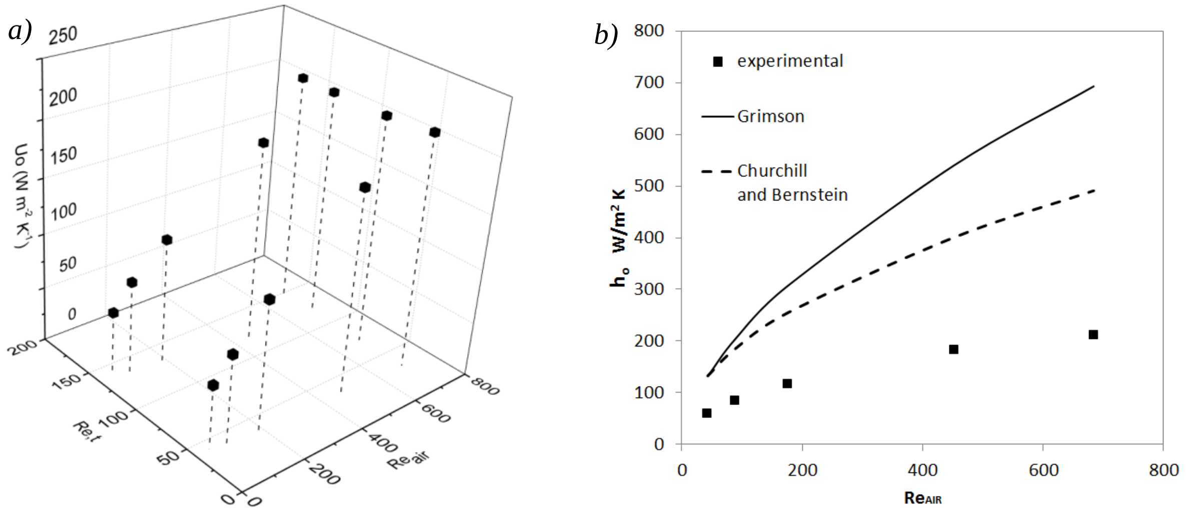

Fig. 2: a) Overall HTC vs tube- and air Reynolds numbers; b) Outer HTC vs air Reynolds number.

5. Conclusions

It was proven that the use of twisted hollow fibers to create large-scale gas-to-liquid heat exchangers can be successful. Experimental data showed that such device can achieve substantial values of heat transfer up to 21.6 kW) and have high values of overall HTC (up to 230 $Wm^{-2}K^{-1}$). It was shown that air-side thermal resistance is dominant in all cases and defines heat transfer. On the other hand, it was shown that heat transfer across fiber bundles strongly depends on effects which are not described enough by existing theory. This issue requires further study which will include both: introduction of parameters describing fiber bundle geometry and experimentation establishing dependence of heat transfer on these parameters.

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