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Polymeric hollow fiber heat exchanger as an automotive radiator

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HIGHLIGHTS

- Polymeric hollow fiber heat exchanger as an automotive radiator is proposed.
- The mechanism of heat transfer (HT) relies on diameter of polymeric hollow fiber.
- Grimson equation is sufficient for approximate prediction of the heat transfers.

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ABSTRACT

Nowadays, different automotive parts (tubing, covers, manifolds, etc.) are made of plastics because of their superior characteristics, low weight, chemical resistance, reasonable price and several other aspects. Manufacturing technologies are already well-established and the application of plastics is proven. Following this trend, the production of compact and light all-plastic radiators seems reasonable. Two plastic heat exchangers were manufactured based on polypropylene tubes of diameter 0.6 and 0.8 mm (so-called fibers) and tested. The heat transfer performance and pressure drops were studied with hot (60 °C) ethyleneglycol-water brine flowing inside the fibers and air (20 °C) outside because these conditions are conventional for car radiator operation. It was observed that heat transfer rates (up to 10.2 kW), overall heat transfer coefficients (up to 335 W/m^2 K), and pressure drops are competitive to conventional aluminium finned-tube radiators. Moreover, influence of fiber diameter was studied. It was observed that air-side convective coefficients rise with a decrease of fiber diameter. Air-side pressure drops of plastic prototypes were slightly higher than of aluminium radiator but it is expected that additional optimization will eliminate this drawback. Experimentally obtained air-side heat transfer coefficients were compared with the theoretical prediction using the Grimson equation and the Churchill and Bernstein approach. It was found that the Grimson equation is sufficient for approximate prediction of the outer HTCs and can be used for engineering calculations. Further work will concentrate on optimizing and developing a polymeric hollow fiber heat exchanger with reduced size, weight and optimized performance and pressure drops.

Contents

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Nomenclature

A	heat transfer area or area of cross-section, m^2
D	diameter, m
Gz	Graetz number, DRePr/L, dimensionless
h	heat-transfer coefficient, W m^{-2} K^{-1}
L	fiber length, m
N	number of fibers, dimensionless
Nu	Nusselt number, hD/k, dimensionless
Pr	Prandtl number, $C_p\mu/k$, dimensionless
Q	rate of heat transfer, W
R	thermal resistance, m^2 K W^{-1}
Re	Reynolds number, Du/μ, dimensionless
T	temperature, K
ΔT	temperature difference, K
U	overall heat-transfer coefficient, W m^{-2}

Subscripts

f	frontal
i	inner
o	outer
ov	overall
t	tube
T3	convection boundary condition (the third type)
w	wall

0. Introduction

Polymeric material and their processing technology [1] have been of growing importance over the last decades. They are applicable across all branches including chemical [2], pharmaceutical [3], and automotive industries [4,5]. From the point of view of polymeric material properties, development of new polymeric material has led to modified polymers being used as alternative automotive materials [6]. These new materials should be more ecologically and economically competitive. For example, investigations of glass-fiber reinforced polyamide composites applicable in the automotive industry were reported by Rudzinski et al. [7], Njuguna et al. [8], Teixeira et al. [9], Thomason [10], Mouti et al. [11]. The polymeric hollow fiber heat exchanger [12] is a novel technology with the potential not only to significantly improve the current products but to enable entirely new applications and markets. The specific objectives leading to production are characterized as: design, testing and technology. This new approach to the enhanced functionality of polymeric fiber materials used in heat transfer surfaces already has the potential in its current development stage to be competitive in large segments of commercial heat exchangers concerning cost [13], weight, mass scalability, recyclability, resistance to corrosion [14] and low fouling. This technology by itself applied in the car-radiator market has the potential to save globally over one hundred thousand tons of aluminium annually. The main goal of this paper is to study thermal performance and compare it with existing aluminium radiators with the aim to develop completely new product on high innovative level.

1. Theoretical background

An evaluation was conducted of potentially replacing metallic heat exchangers by considering thermal-hydraulic performance, mechanical strength, size, weight and material cost which was investigated by Park and Jacobi [13]. It was stated that a polymer-tube-bundle heat exchanger could be a potential replacement for a conventional metallic plate-fin-and-tube heat exchanger for water-to-air applications. The lower thermal conductivity of polymeric material was overcome by employing a larger number of thin-walled small-diameter tubes. The use of hollow metallic particles for thermal conductivity enhancement and lightening of a filled polymer was studied by Garnier and Boudenne [15]. As mentioned earlier [12], the overall concept of heat transfer is based on heat transfer features of micro-objects [16], where the intensity of convective heat flow through a given material grows with the square of the object's size. However, the heat transfer, thermal stability, pressure limits [17] and toughness of a hollow polymeric material is still a major problem in the development sphere in the automotive industry. Thus, engineers are focusing their scientific interests to improve these properties.

The basic parameter of convective heat transfer is the Nusselt number. It is viewed as the dimensionless convection heat transfer coefficient characterized as the ratio of convective to conductive heat transfer through the fluid layer [18]. Lamilar flow is characterized for similar magnitudes of convection and conduction. The Nusselt number is independent of fluid flow velocity and is close to 4 for a circular tube.

Zarkadas and Sirkar [19] provided an analytical solution for laminar flow inside circular tubes with boundary conditions described by HTC and ambient temperature. The solution is based on the asymptotic Nusselt number calculated by Hickman's approach and the incremental heat transfer number calculated by Hsu's approach. The incremental heat transfer number evaluated as a function of the external resistance Nu_w and the length of the thermal developing region expressed by the Graetz number Gz [19]. These values were used to construct a simple relationship for the mean Nusselt number of the tube, which is only a function of the external resistance and the hydrodynamic conditions prevailing in the tube [19]:

$$Nu_{T3} = \frac{\left(\frac{48}{11}\right) + Nu_w}{1 + \left(\frac{59}{220}\right)Nu_w} + \left(0.0499 - \frac{0.06487}{1 + \exp\left(\frac{Nu_{w+5.37935}}{2.17887}\right)}\right)\frac{4Gz}{\pi} \quad (1)$$

Considering flow on the outer surface of the hollow fiber bunch, the Nusselt number for a given geometry can be expressed in terms of the Reynolds number and the Prandtl number. We decided to evaluate two models for predicting the outer HTC. The first model is for a single tube and the second is for a bunch of tubes with a cross-flow of air. The average Nusselt number for a cross-flow over a single tube can be evaluated using the equation by Churchill and Bernstein [20]:

$$Nu_d = 0.3 + \frac{0.62Re^{1/2}Pr^{1/3}}{\left[1 + (0.4/Pr)^{2/3}\right]^{1/4}}\left[1 + \left(\frac{Re}{282.000}\right)^{5/8}\right]^{4/5} \quad (2)$$

For a bunch of tubes, depending on fiber width and depth pitches, the Grimson approach can be recommended [21]:

$$Nu = CRe^{m}Pr^{1/3} \quad (3)$$

Experimentally determined constants C and exponents m represent a geometric description of the tube-bundle arrangement