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Selection of Heat Transfer Enhancement Technique for Compact Mini and Micro Heat Exchangers Design

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The need to decrease the sizes and masses of heat exchangers while preserving their performance has stipulated the development of compact heat exchangers with mini and micro channels (MCHE). It is supported by the need for increased recuperation of heat energy, facilitating better energy efficiency with strict limitations for space, material and cost. The adequate substitution of conventional heat exchangers with MCHE requires maintaining the same heat load, not exceeding the allowable pressure losses. The different ways to increase the compactness are analysed, including the change of hydraulic diameter, and the use of various methods of heat transfer intensification by changing the channel geometry and flow structure. The Nusselt numbers and friction factors correlations for plane tubes and criss-cross flow channels of plate heat exchangers are compared. A newer form of the core velocity equation has been developed, which allows comparison of the performance of MCHE heating surfaces with different enhancement techniques in specific process conditions. The results of the calculations illustrate the influence of the channel's hydraulic diameter and length on the MCHE performance for channels with the considered methods of heat transfer intensification. The ways to decrease channel size to mini and micro scales are determined. The recommendations on choosing the best channel geometry and size, depending on specified process conditions and stream nature, are formulated.

1. Introduction

Increased heat recuperation is the main prerequisite for efficient energy use in thermal systems in chemical and many other industrial applications. The main components responsible for this and for the total size and weight of such a system are heat exchangers. With a further rise in demand for energy, limited space and material resources necessitate further developments in heat exchangers' construction and design principles. In that view, compact heat exchangers with mini and micro channels can be regarded as the next generation of heat transfer equipment, accounting for enhanced heat transfer and reduction of weight, volume and materials for their construction.

According to Kandlikar and Grande (2003) minichannels can be classified by hydraulic diameters D_h from 3 mm to 0.2 mm, and microchannels as having D_h from 0.2 mm to 0.01 mm. The hydraulic diameters outside these ranges characterise conventional channels with D_h exceeding 3 mm and transitional microchannels and nanochannels with D_h smaller than 0.01 mm.

Mini and microchannel heat exchangers have found a wide range of applications in microelectronics (Bhandari et al., 2024), micro process systems (Sharifian et al., 2024), waste heat utilisation involving supercritical CO2 and Micro-Electromechanical Systems (MEMS) technology (Ejaz and Zubair, 2025), air conditioning systems (Han et al., 2012), communal sector and pasteurisation (Tuckerman et al., 2011), solar thermal energy and nuclear power (Ma et al., 2022), recuperators for micro gas turbines (Wang et al., 2024), refrigeration (Başaran

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and Benim, 2024), aerospace and defense aviation applications (Marseglia et al., 2024). It stipulated a large number of studies in this field. For 6 years only, from 2017 to 2023, the Scopus database contained 3877 papers on microchannel heat exchangers (Dwivedi et al., 2023). During that period, about 25 review papers were published only on microchannel heat sink studies (Bhandari et al., 2024), and approximately the same number for other applications. With such a large number of studies on heat transfer and friction in minichannels and microchannels, there are considerable differences in the results of experiments and the views on the applicability of conventional correlations. It was observed more than two decades ago by Sobhan and Garimella (2001), and not much changed later with the appearance of newer research results surveyed by Morini (2004) and later by Singh et al. (2019). It creates difficulties in designing real compact heat exchangers with mini and micro channels. The selection of the proper geometry of such channels requires an efficient method of estimating their thermal and hydraulic performance in specific conditions of operation in different applications. It becomes even more important with current trends of developing new methods of heat exchangers production with additive manufacturing, and heat transfer intensification using nanofluids.

The objective of this paper is to present a method for comparing the efficiency of heat exchangers operating with different minichannels and microchannels, based on a developed equation for optimal fluid velocity inside the channels. This approach enables the full utilisation of the available pressure drop in the heat exchanger with minimal heat transfer surface area *F*. These values of heat transfer areas *F*, obtained for the different channels, can be compared by the proposed Specific Performance Index.

2. Method

To estimate heat transfer area for comparison of different heat exchangers, consider the heat exchanger strictly satisfying specific process conditions (with specified flow rates, temperature program and pressure losses), operating at pure counter current flow of heat exchanging streams. The heat transfer load *Q* calculated at the cold fluid side must be equal to the heat transferred through the heat transfer surface, following Eq(1).

$$Q = U \cdot \Delta T_{ln} \cdot F = W_2 \cdot F_{cs} \cdot c_{p2} \cdot \rho_2 \cdot (T_{22} - T_{21}), \tag{1}$$

where W_2 is the cold stream flow velocity, m/s; F is the heat transfer area, m^2; F_{cs} is the channel cross-section area, m^2; T_{21} is the cold stream entrance and T_{22} is the cold stream exit temperature, K; c_{p2} is the cold fluid specific heat, J/(kg K); ρ_2 is the fluid density, kg/m^3; ΔT_{ln} is the Logarithmic Mean Temperature Difference.

The number of heat transfer units *NTU* from Eq(1) can be written as Eq(2):

$$NTU = \frac{U \cdot F}{W_2 \cdot F_{cs} \cdot c_{p2} \cdot \rho_2} = \frac{T_{22} - T_{21}}{\Delta T_{ln}} \tag{2}$$

On the left-hand side, it is *NTU* satisfied by the heat exchanger, and on the right-hand side, it is NTU_0 required by the specified process. The heat transfer area at the cold side can be expressed as the product of its length *L* multiplied by the wetted perimeter Π. Then the hydraulic diameter D_h of the channel is calculated by Eq(3):

$$D_h = 4 \cdot \frac{F_{cs}}{\Pi} \tag{3}$$

The left part of Eq(2) can be expressed by Eq.(4), and for specific conditions must be equal to NTU_0:

$$NTU_0 = \frac{U \cdot 4}{W_2 \cdot c_{p2} \cdot \rho_2} \cdot \frac{L}{D_h} \tag{4}$$

The heat exchanger must strictly satisfy the specified pressure drop ΔP_2 for the cold fluid expressed by Eq.(5).

$$\Delta P_2 = 2 \cdot f_2 \cdot \rho_2 \cdot W_2^2 \cdot \frac{L}{D_h} \tag{5}$$

where f_2 is the Fanning friction factor. Dividing Eq(5) on Eq(4) gives:

$$\frac{\Delta P_2}{NTU_0} = \frac{f_2 \cdot W_2^3 \cdot c_{p2} \cdot \rho_2^2}{U \cdot 2} \tag{6}$$

For a known friction factor and overall heat transfer coefficient *U*, this Eq(6) determines the flow velocity in the channel that strictly satisfies the specified process conditions as equalities. It corresponds to a minimal heat transfer surface for these conditions, as all pressure drop is used and no margin on heat load. However, the values of *U* and f_2 are not known, but can be determined by empirical correlations for the considered channel when thermal resistances for the hot side of the heat exchanger and its wall are known. For comparison of two different channels in the same conditions, it can be assumed that the thermal resistances on the hot side are the same for both channels, and the main resistance to heat transfer is concentrated on the cold side. In that case overall heat transfer coefficient *U* is equal to the heat transfer coefficient at the considered channel h_2 and can be determined by the correlation for Nusselt number (*Nu*) in this channel, as well as the friction factor f_2 by its correlation. For the same fluid and temperature program h_2 and f_2 depend only on fluid velocity in the channel under consideration and channel hydraulic diameter D_h, and Eq(6) can be transformed to Eq(7), as it is shown in more detail by Kapustenko et al. (2025).

$$W_2 = \sqrt[3]{\frac{2 \cdot \Delta P_2 \cdot h_2(D_h, W_2)}{c_{p2} \cdot \rho_2^2 \cdot NTU \cdot f_2(D_h, W_2)}} \tag{7}$$

For the fixed value of hydraulic diameter D_h, Eq(7) can be solved by iterations, with the initial value of W_2 equal to 0.5 m/s. The calculations are repeated until the difference between two consecutive values of W_2 becomes smaller than 0.001 m/s. After that, heat transfer area F is determined by Eq(1). This value corresponds to the channel of the considered geometry, with the length of the heat transfer area strictly satisfying the specified process conditions. It is valid for the ranges of Reynolds (Re) and Prandtl (Pr) numbers at which there are accurate correlations for *Nu* and *f.* With maintained geometrical similarity of the channel, the change of D_h does not affect these correlations until the continuum approach with no wall slip is valid for the flow of the considered fluid (Kandlikar and Grande, 2003). With diminishing hydraulic diameter of microchannels, such a conventional approach will not be satisfied starting with some lower value of D_h. By the estimates of different scholars, this value can lie somewhere between 0.01 and 1 mm. It will be checked later by comparing experimental results published by the different authors. As the basis of conventional correlations for smooth tubes, the following applies.

An Equation Eq(8) developed by Churchill (1977) correlates the Fanning friction factor for all flow regimes (laminar, transitional and turbulent).

$$f = 2 \cdot \left\{ \left(\frac{8}{Re}\right)^{12} + \frac{1}{\left(\left[2.457 \cdot \ln\left(\frac{1}{\left(\frac{7}{Re}\right)^{0.9} + 0.27 \cdot \frac{0}{D}} \right) \right]^{16} + \left(\frac{37{,}530}{Re}\right)^{16} \right)^{\frac{3}{2}}} \right\}^{\frac{1}{12}} \tag{8}$$

For the Nusselt number *Nu* in turbulent flow inside tubes is used the Eq(9) of Gnielinski (1975):

$$Nu = Nu_T = \frac{f \cdot Pr \cdot (\mathrm{Re} - 1000)}{2 \cdot \left[1 + 12.7 \cdot \sqrt{f/2} \cdot \left(Pr^{2/3} - 1 \right) \right]} \qquad \text{for Re > 4,000} \tag{9}$$

At lower Reynolds numbers the Eq(10) and Eq(11) of Gnielinski (2013) are used.

For Re≤2300 (laminar flow)

$$Nu_L = \left[3.66^3 + 0.7^3 + \left(1.615 \sqrt[3]{Re \cdot Pr \cdot \frac{D}{L}} - 0.7 \right)^3 + \left(\frac{2}{1 + 22 \cdot Pr} \right)^{\frac{1}{6}} \cdot \left(Re \cdot Pr \cdot \frac{D}{L} \right)^{\frac{1}{2}} \right]^{\frac{1}{3}} \tag{10}$$

For 4000≥Re>2,300 (transitional flow regime)

$$Nu = Nu_{L.2300} + \frac{(Nu_{T.4000} - Nu_{L.2300}) \cdot (Re - 2{,}300)}{4{,}000 - 2{,}300} \tag{11}$$

Here, *Re* is the Reynolds number; *Pr* is the Prandtl number; *D* is the tube diameter, m; *L* is the tube length, m. The effect on heat transfer of a temperature change from the mainstream to the wall is neglected.

3. Results and discussion

The method is illustrated by the example of heating an incompressible liquid. For microchannels, this process usually occurs at Re numbers below 4,000. The flow of gases is not considered in this paper. It is necessary to heat water at a flow rate of 2.9 kg/s from 5 °C to 55 °C at the exit of a heat exchanger, using vapour at temperature 60 °C exhausted with the temperature 60 °C. The available pressure difference on the cold water side is ΔP_2 = 20,000 Pa. The results of calculations by Eq(7), using Eqs(8-11) for smooth tubes, are presented in Figure 1. The character of the heat transfer area changes with the decrease in hydraulic diameter is complicated in the region corresponding to the transition between laminar and turbulent flow regimes. In the considered example, when D_h becomes smaller than 3 mm, F grows up to four times. This peak in F at $D_h \approx$ 1–3 mm arises from deterioration of convective heat transfer in transition from turbulent flow regime at 4,000>*Re*>2,300, with flow becoming laminar at Re<2,300. With further decrease of D_h, the heat transfer surface area F decreases with a smaller hydraulic diameter. It diminishes again to the values at turbulent flow regime

(D_h>3 mm) only at D_h about 0.3 mm. However, even such a reduction of F becomes doubtful, accounting for the possible change of correlations in microchannels with a hydraulic diameter less than 1 mm. According to Wu and Cheng (2003), in straight channels of hydraulic diameters from 25.9 to 209.8 μm, the Nu number can be from 10 % to 90 % smaller than the value calculated by the equation for a smooth tube.

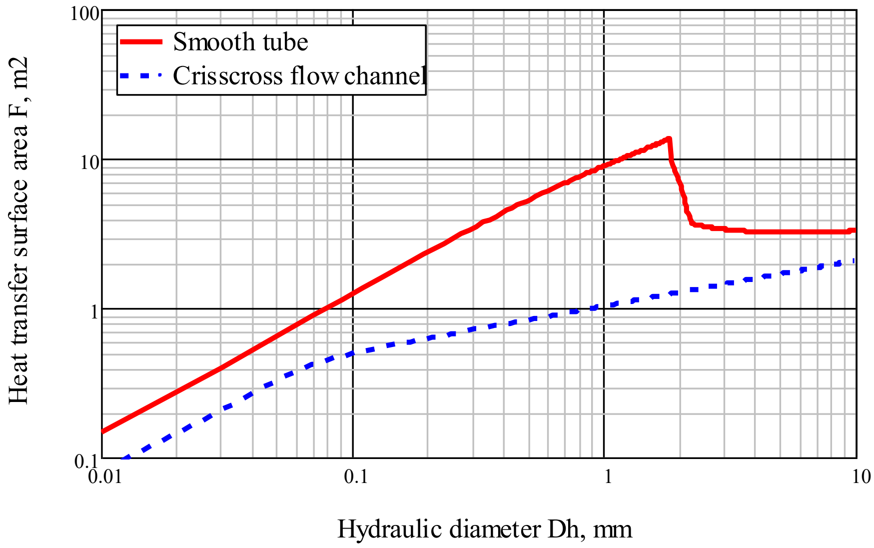

Figure 1: Heating area F at different hydraulic diameters D_h

Comparing different channels, the heat transfer area F_{s3}, of a heat exchanger with smooth tubes at D_h = 3 mm is taken as a benchmark, and the specific performance index (SPI) can be written by Eq(12):

$$SPI = \frac{F}{F_{s3}} \tag{12}$$

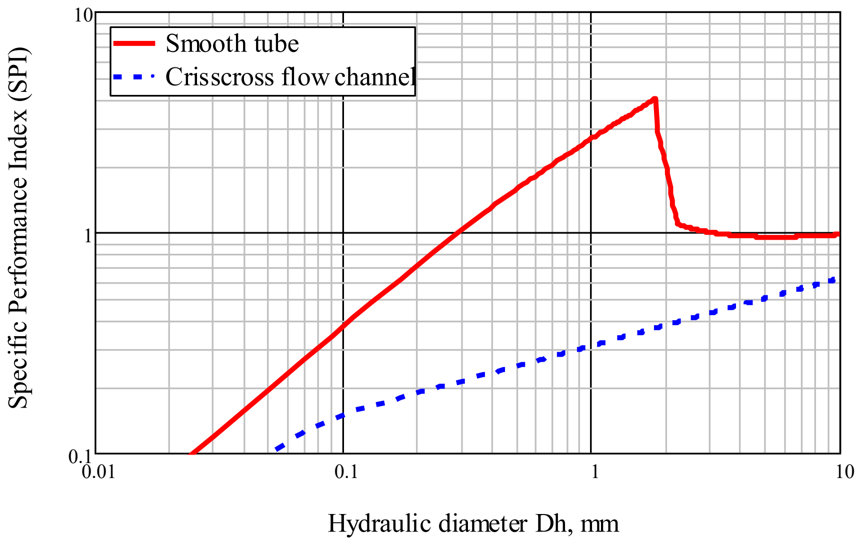

Figure 2: Performance evaluation index (SPI) at different hydraulic diameters D_h

The SPI index enables a more rigorous estimation of the channel performance, as it is nondimensional and, for the examined channel, depends only on the temperature program, fluid properties and pressure drop. The heat transfer area F is otherwise affected by the heat load Q. The evolution of the SPI of the smooth tube with the D_h is depicted in Figure 2. Counting for uncertainty with D_h smaller than 1 mm, there is little chance to decrease the heat transfer area based on channels with smooth surfaces just by decreasing the hydraulic diameter. It would require employing other methods of heat transfer enhancement. One such method is used in crisscross flow channels of plate heat exchangers and primary surface recuperators of micro gas turbines (Figure 3). The generalised correlations for friction factor and Nu number in such channels were proposed by Arsenyeva et al. (2012).

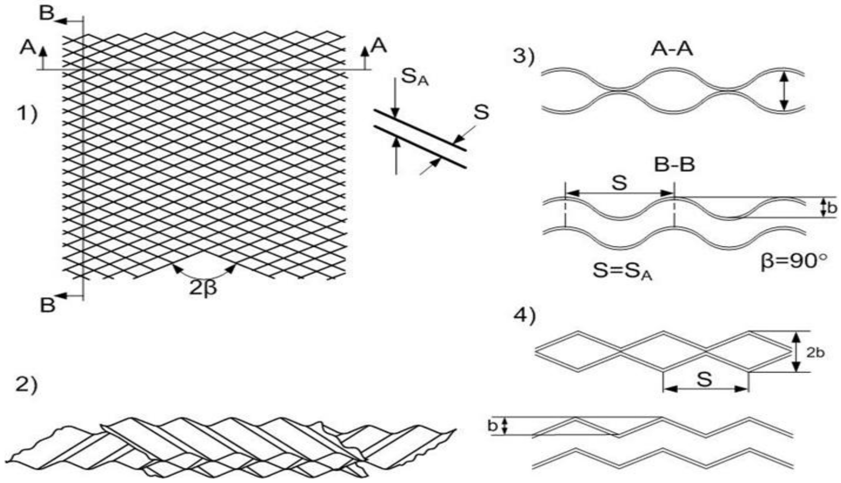

Figure 3: Schematic drawing of a crisscross flow channel

The results of calculations by these correlations for the channel with the corrugation inclination angle 60 degrees and aspect ratio 0.6 are presented in Figure 1, with index SPI in Figure 2 by the dashed lines. The performance index is changing from 0.43 at D_h = 3 mm to 0.307 at D_h = 1 mm, with the possibility of further decrease to 0.148 at D_h = 0.1 mm. While the applicability of such conventional correlations at D_h < 1 mm needs to be studied, at D_h = 1 mm it can be granted. Xi et al. (2018) studied crossflow channels of primary surface recuperators with hydraulic diameters from 0.89 to 0.94 mm. In that case, SPI=0.307, which even theoretically could be achieved with a smooth surface at a much smaller hydraulic diameter D_h = 0.08 mm. In such conditions, the Re number is equal to 82, corresponding to laminar flow in a smooth tube. In a crisscross flow channel at D_h = 1 mm Re number is equal to 685, which corresponds to a turbulent regime in these channels with heat transfer enhancement. It confirms the possibility of using crisscross flow channels for manufacturing efficient micro heat exchangers and emphasises the need for further studies with different heat transfer enhancement techniques.

4. Conclusions

Nowadays, compact heat exchangers with mini and micro channels can be regarded as the next generation of heat transfer equipment, accounting for enhanced heat transfer and reduction of weight, volume, and construction materials. This efficient type of equipment is finding a lot of applications in different industries. A large number of experimental and theoretical studies on heat transfer and hydraulic performance of such channels have been published in recent years. Some results of these studies for channels of hydraulic diameters less than 1 mm are not consistent. A method to estimate the performance of a heat exchanger with different mini and micro channels is proposed. It is based on a developed equation for the estimation of fluid velocity in the channel that allows strict satisfaction of the specific conditions of the temperature program and pressure losses in a heat exchanger with channels of different hydraulic diameters. The Specific Performance Index (SPI) for comparison of channel performance in specific process conditions is introduced. It is shown that simple reduction of hydraulic diameter for channels with smooth walls can lead to an increase of required heat transfer surface area on some intervals of mini and micro sizes from hydraulic diameter about 3 to 0.3 mm. To get a monotonic decrease in heat transfer surface area with the diminishing hydraulic diameter, it is better to use channels with enhanced heat transfer. It is shown by an example of the crisscross flow channel, typically used in plate heat exchangers and primary surface recuperators of gas turbines. The SPI value of 0.30, which corresponds to 70 % reduction in heat transfer area, can be achieved at a moderate value of hydraulic diameter of about 1 mm. In future research, other methods of heat transfer enhancement in microchannels should be investigated and estimated with the proposed methodology.

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