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The influence of flow maldistribution on the performance of inhomogeneous parallel plate heat exchangers

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ABSTRACT

The heat transfer performance of inhomogeneous parallel plate heat exchangers in transient operation is investigated using an established model. A performance parameter, denoted the Nusselt-scaling factor, is used as benchmark and calculated using a well-established single blow technique. A sample of 50 random stacks having equal average channel thicknesses with 20 channels each are used to provide a statistical base. The standard deviation of the stacks is varied as are the flow rate (Reynolds number) and the thermal conductivity of the solid heat exchanger material.

It is found that the heat transfer performance of inhomogeneous stacks of parallel plates may be reduced significantly due to the maldistribution of the fluid flow compared to the ideal homogeneous case. The individual channels experience different flow velocities and this further induces an inter-channel thermal cross talk.

1. Introduction

Microchannel heat exchangers show promise for applications that require high cooling power density. It has been suggested that liquid cooling using a microchannel heat exchanger is the best suited technique for cooling electronics as transistor density continues to increase [1]. Microchannels have also been applied to a wide variety of applications, such as cryocoolers, dehumidifiers, Stirling engines, solar power, electronics cooling and magnetic refrigeration [2–8].

Despite the high theoretical performance, microchannel heat exchanger performance in actual devices has often been reported in experiments to be lower than expected [9]. Several explanations for the relatively large deviations observed have been suggested, such as whether the fluid continuum assumption breaks down, the influence of surface roughness in the channels etc. In Ref. [9] these issues are reviewed and it is concluded that for incompressible, single phase laminar flows with aqueous fluids no new physical phenomena occur in microchannels. This is supported by careful experiments performed on single-channel tubes and square channel heat exchangers in the microchannel range [10,11].

It was shown that flow maldistribution in fluid manifolds can reduce the microchannel heat transfer performance in a microchannel heat exchanger [12], as channels near the edges of the heat exchanger do not receive as much fluid flow as those in the center. The flow and temperature distributions in two parallel microchannels were studied numerically when obstructions such as bubbles or debris were placed in one of the channels [13]. The outlet temperature profile was shown to be affected by an obstruction in one of the channels, but the accompanying change in heat transfer performance was not quantified.

Another explanation of the observed heat transfer degradation is non-uniform plate spacing [14] in the microchannel stack. As heat transfer performance is inversely proportional to plate spacing, the plate spacing is reduced as much as possible for many applications. As dimensions become smaller the relative manufacturing tolerance can become significant. It was shown that variations in fluid channel heights can dramatically reduce the effective heat transfer in a heat exchanger consisting of a stack of microchannels because larger fluid channels receive a disproportionately high fluid flow while smaller channels are starved for flow [14]. The effect was demonstrated through numerical modeling as well as experiments on passive regenerators. It was also shown that thermal cross talk, heat transfer between adjacent fluid channels through channel walls, equalizes temperature differences between fluid channels caused by differences in channels heights. Therefore, cross talk between channels generally reduces the negative impact on heat exchanger performance due to a non-uniform distribution of fluid channels.

This article studies the effects of thermal cross talk in more detail and shows its effects on microchannel heat exchanger performance under transient operation. It will be shown that the effect of the cross talk can vary greatly with geometry, material properties, and operating conditions. We show that a non-uniform

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microchannel stack can fall into three categories from a thermal modeling standpoint: isolated channels, cross talk dominated, and effectively uniform. In the case of isolated channels, the thermal interaction between the channels is so low in relation to other phenomena that the channels can be modeled as individual isolated channels operating in parallel with an acceptable accuracy. In cross talk dominated operation, a detailed model of a stack of non-uniform flow channels is needed in order to obtain accurate results. In the last category the cross talk is sufficiently high in relation to other heat transport, the heat exchanger can be assumed to be a uniform stack. It is shown that a given microchannel stack can fall into any of these three categories of cross talk depending on the fluid flow rate or thermal conductivity of the solid. A detailed numerical model of a microchannel stack is used to calculate the temperature response when the heat exchanger is subjected to a step change in temperature at its inlet. A technique developed for characterizing heat transfer in porous beds [15] is used to calculate the bulk heat transfer in the entire regenerator and compare it to an ideal microchannel stack with uniform fluid channel heights. All results generated here use a transient response technique that was developed primarily for regenerators rather than steady state heat exchangers such as heat sinks. However, the general conclusions should also apply to heat exchangers in general.

2. Heat transfer in inhomogeneous parallel plate stacks

The general problem of determining the heat transfer characteristics of a heat exchanger, here specifically parallel plate stacks, involves thermal conduction in a solid material (of which the parallel plates are made), thermal conduction and convection in a heat transfer fluid in intimate contact with the solid and the heat transfer between the two media. In the idealized case the stack is assumed perfectly homogeneous, and it is therefore sufficient to consider half a solid plate and half a fluid channel due to symmetry. Numerical models of such systems have been reported including one-dimensional models where the direction of the flow (or axial direction) is the spatial dimension resolved and, in certain cases, heat transfer transverse to this direction is accounted for via a Biot-Fourier number approach [16]. In other, 2-dimensional, models both the axial direction and the direction transverse to the flow (the y-direction in Fig. 1) are numerically resolved for the purpose of studying, e.g., low-thermal conductivity materials [17,18].

It is clear that if the stack is inhomogeneous, e.g., due to varying channel thicknesses and/or plate thicknesses that the symmetries just described will not apply. Each fluid channel will have a different mean fluid velocity, which will cause varying heat transfer in the transverse direction between adjacent channels and plates. This effect is referred to as thermal cross talk. It is non-trivial to provide a general estimate of the influence of this cross talk and in the following Section 2.1.3 a term that quantifies the significance of cross talk for a given operating condition is proposed.

The other influence that inhomogeneity in a parallel plate stack has on the heat transfer characteristics is, of course, due to the varying heat transfer conditions in each individual channel caused by the varying flow velocities (or flow maldistribution) and thus varying convection coefficients. Larger channels will generally have poorer heat transfer than smaller ones and at the same time significantly larger fractions of the total flow rates thus causing the heat exchanger to perform less efficiently than it would had it been a perfect stack with channel thickness equal to the average spacing of the inhomogeneous stack [14]. These two effects are investigated applying the numerical model described in Section 2.1.2 below.

2.1. Governing equations

2.1.1. 1D homogeneous regenerator equations

Porous medium regenerators are often modeled as a solid phase interacting with a fluid phase with uniform flow. The geometry of the regenerator is characterized by correlations for pressure drop, heat transfer and other correlations rather than by solving detailed conduction and fluid flow equations to determine them directly. The one-dimensional partial differential equations for the solid and fluid are given below. For the fluid the equation is

$$\dot{m}_\mathrm{f} c_\mathrm{f}\frac{\partial T_\mathrm{f}}{\partial x} + hA_\mathrm{HT}(T_\mathrm{f}-T_\mathrm{s}) + \rho_\mathrm{f} A_\mathrm{c}\varepsilon c_\mathrm{f}\frac{\partial T_\mathrm{f}}{\partial t} - k_\mathrm{f} A_\mathrm{c}\frac{\partial^2 T_\mathrm{f}}{\partial x^2} = 0 \quad (1)$$

where T is temperature, ρ is density, c is specific heat, k is thermal conductivity, h, is the heat transfer coefficient, A_HT is the area for heat transfer per unit length, $\varepsilon = \frac{H_\mathrm{f}}{H_\mathrm{f}+H_\mathrm{s}}$ is the porosity, $\dot{m}_\mathrm{f}$ is the fluid mass flow rate, t is time and A_c is the cross sectional area. Subscripts s and f indicate solid and fluid, respectively. The x-direction is defined as the direction of the flow (the axial direction; see Fig. 1). The terms represent (in order from left to right) the enthalpy change of the flow, heat transfer from the fluid to the solid, energy storage, and axial conduction in the fluid. Viscous dissipation due to pumping losses is ignored. Eq. (1) cannot directly account for in-homogeneous regenerator geometries because the fluid flow would be a function of the y direction. However, the correlations used to determine heat transfer, pressure drop etc. can be modified to account for in-homogeneous effects.

The 1D solid material governing equation is

$$hA_\mathrm{HT}(T_\mathrm{f}-T_\mathrm{s}) + k_\mathrm{s} A_\mathrm{c}\frac{\partial^2 T_\mathrm{s}}{\partial x^2} = (1-\varepsilon)A_\mathrm{c}\rho_\mathrm{s} c_\mathrm{s}\frac{\partial T_\mathrm{s}}{\partial t} \quad (2)$$

The terms represent heat transfer from the fluid to the regenerator, static axial conduction and energy storage.

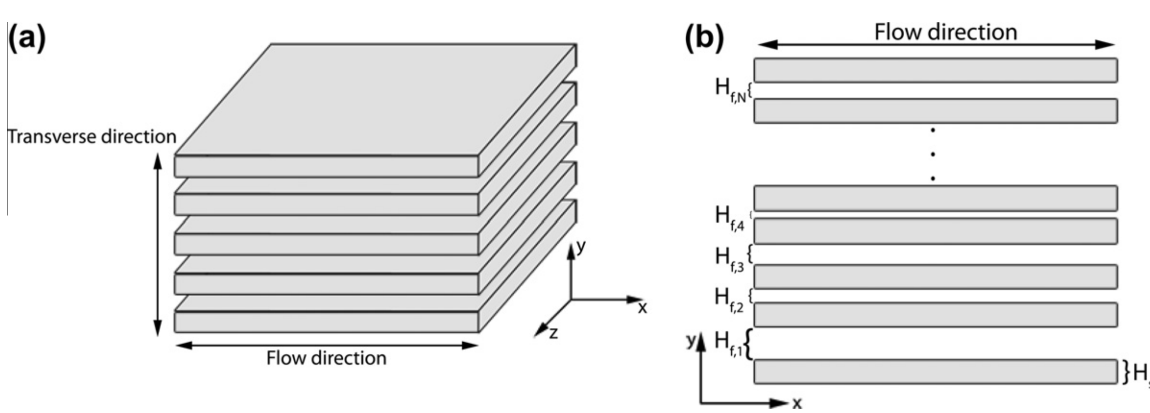

Fig. 1. Schematic of the parallel plate stack geometry: (a) 3D perspective of the stack, (b) the geometry modeled in the detailed model. Each plate has the same thickness whereas the flow channels may have varying thicknesses. The x-direction is the direction of the flow and the y-direction is referred to as the transverse direction.