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Heat transfer enhancement by layering of two immiscible co-flows

A.-R.A. Khaled [a], K. Vafai [b,*]

[a] King Abdulaziz University, Mechanical Engineering Department, Jeddah 21589, Saudi Arabia
[b] University of California at Riverside, Mechanical Engineering Department, Riverside, CA 92521, USA

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ABSTRACT

Enhancement of heat transfer in minichannels due to co-flowing of two immiscible fluids in a direct contact is investigated in this work. Different fluid combinations are analyzed. The momentum and energy equations for both flows are solved analytically and numerically. The numerical and analytical solutions are found to be in good agreement. A parametric study including the influence of fluids relative viscosity, thermal conductivity, thermal capacity and height ratios is conducted for various Peclet numbers. Different ranges of the parameters that augment the heat transfer are obtained, and different physical aspects of the problem are discussed. For practical fluid combinations with small Peclet numbers, the enhancement factor can increase up to 2.6 folds. However, that increase is about 1.2 folds when the Peclet number is increased by two orders of magnitude. This work establishes the mechanisms for heat transfer enhancement utilizing two immiscible co-flows.

1. Introduction

A review of recent heat transfer literature [1–3] reveals that large number of investigations was devoted to the topic of heat transfer enhancement. Most of these works considered at least one of the following enhancement mechanisms: (a) stream profiling [4,5] such as using twisted tapes, spirals, and wire coils, (b) fins such as slotted and louvered fins [6,7], (c) electrohydrodynamic effects [8], (d) surface coatings [9], (e) additives [10,11], (f) acoustic streaming [2], (g) turbulators [2] and, (h) flow and velocity amplifications [12–14]. Additional reviews in this area have shown an increased interest in enhancement technology [15,16]. A number of these works introduce new concepts in this area [17–22].

Heat transfer Enhancement utilizing an immiscible fluid co-flowing with the coolant flow in a direct contact manner has not received much attention by researchers [1–3,15,16,22]. As such, this subject is considered as the main topic of the present work. Immiscible fluids having smaller viscosity or larger specific heat than the coolant can enhance the heat transfer. This is because they can enhance the velocity near the heated plate, widening the thermal entry region or amplifying the coolant flow rate. These effects are the major causes for heat transfer enhancement by laminar flows. Therefore, the present work is additionally aimed to specify ranges for thermophysical properties and flow conditions that can result heat transfer enhancement by co-flowing two immiscible fluids.

One of the main advantages of heat transfer enhancement by co-flowing of two immiscible fluids in direct contact manner is that both flows are subject to the same pressure gradient. In contrast, the pressure gradient of the counter-flowing system decrease to zero then changes sign and starts to decrease below zero. As a result, the fluid flow rates in the co-flowing system are expected to be larger than those in the counter-flowing system. Moreover, for short channels with co-flowing system, the secondary fluid temperatures at the inlet of the primary flow are much smaller than those of the counter-flowing system. This effect elevates the convection heat transfer coefficient between the heated boundary and the adjacent primary flow due to thermal entry region. Finally, as the micro-channel geometry may cause inadequate operation of efficient enhancement methods [23–25], co-flowing of two immiscible fluids can become an attractive alternative.

In the present work, heat transfer through two layered immiscible fluid flow within a horizontal minichannel is analyzed. For all analyzed cases, the density and the thermal conductivity of the primary fluid residing on the heated boundary are considered to be larger than those of the other fluid, referred to as the secondary fluid. The dynamic viscosity and the volumetric thermal capacity of the secondary fluid are allowed to be either larger or smaller than those of the primary fluid. Both momentum and energy transport equations are solved for each flow using analytical and numerical methods. Various analytical solutions for the temperature field under applicable constraints are obtained and validated against the numerical solution. A parametric study of the heat transfer enhancement is made to identify ranges of controlling parameters that reveal favorable enhancement attributes.

* Corresponding author. Tel.: +1 951 827 2135; fax: +1 951 827 2899.
E-mail address: vafai@engr.ucr.edu (K. Vafai).

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Nomenclature

a	channel aspect ratio ($a = H/L$)
a_1, a_2	(primary, secondary) flow aspect ratio ($a_1 = \delta/L$, $a_2 = [H-\delta]/L$)
C_1, C_2	(primary, secondary) flow thermal capacity (W K^{-1})
c_{p1}, c_{p2}	(primary, secondary) fluid specific heat (J kg^{-1} K^{-1})
H	channel height (m)
h_1	convection coefficient at the hot boundary (W m^{-2} K^{-1})
h_I	convection coefficient between interface and secondary flow (W m^{-2} K^{-1})
k_1, k_2	(primary, secondary) fluid thermal conductivity (W m^{-1} K^{-1})
L	channel length (m)
$\dot{m}_1$, $\dot{m}_2$	(primary, secondary) fluid mass flow rate (kg s^{-1})
Nu_1	Nusselt number between hot boundary and primary flow ($Nu_1 = h_1\delta/k_1$)
Nu_2	Nusselt number between interface and secondary flow, Eq. (20)
P_1, P_2	(primary, secondary) flow ideal pumping power (W)
P_t	total ideal pumping power (W)
p_1, p_2	(primary, secondary) fluid pressure (kg m^{-1} s^{-2})
$\overline{p}_1$, $\overline{p}_2$	(primary, secondary) fluid dimensionless pressure, Eqs. 4(h, i)
Pe_1, Pe_2	(primary, secondary) flow Peclet number ($Pe_{1,2} = Re_{1,2}Pr_{1,2}$)
Pr_1, Pr_2	(primary, secondary) fluid Prandtl number ($Pr_{1,2} = \mu_{1,2}(c_p)_{1,2}/k_{1,2}$)
q_s''	heat flux at hot boundary (W m^{-2})
q_I''	heat flux at interface (W m^{-2})
Re_1, Re_2	(primary, secondary) flow Reynolds number, Eqs. 5(d, e)
T_1, T_2	(primary, secondary) fluid temperature (K)
u_1, u_2	(primary, secondary) fluid axial velocity (m s^{-1})
$\overline{u}_1$, $\overline{u}_2$	(primary, secondary) fluid dimensionless velocity ($\overline{u}_{1,2} = u_{1,2}/u_{(1,2)avg}$)
v_1, v_2	(primary, secondary) fluid transverse velocity (m s^{-1})
$\overline{v}_1$, $\overline{v}_2$	dimensionless (v_1, v_2) transverse velocity ($\overline{v}_{1,2} = v_{1,2}/[(\delta/L)u_{(1,2)avg}]$)
x, $\overline{x}$	(dimensional, dimensionless) axial distance ((m), $\overline{x} = x/L$)
y_1, y_2	normal (primary, secondary) system coordinate (m)
$\overline{y}_1$, $\overline{y}_2$	dimensionless (y_1, y_2) coordinate ($\overline{y}_1 = y_1/\delta$, $\overline{y}_2 = y_2/[H-\delta]$)
Greek symbols	
γ	second thermal performance factor, Eq. (23)
λ	heat transfer enhancement factor, Eq. (22)
μ_1, μ_2	(primary, secondary) fluid dynamic viscosity (kg m^{-1} s^{-1})
θ_1, θ_2	dimensionless (primary, secondary) fluid temperature, Eqs. 4(j, k)
ρ_1, ρ_2	(primary, secondary) fluid density (kg m^{-3})
Subscripts	
avg	average quantity
I	quantity at interface
i	quantity at inlet
o	quantity at outlet
m	mean bulk value of the quantity
r	quantity when primary flow occupying the channel height
W	quantity at the hot boundary

2. Problem formulation

2.1. Generalized two-dimensional model

Consider a two-dimensional horizontal minichannel of height H which is much smaller than its length L. Consider two immiscible fluids co-flowing inside the minichannel. The primary fluid has specific heat c_{p1}, thermal conductivity k_1, dynamic viscosity μ_1 and density ρ_1. The corresponding properties of the secondary fluid are c_{p2}, k_2, μ_2 and ρ_2. The primary fluid is considered to be denser than the secondary fluid ($\rho_1 > \rho_2$) so it will line up against the lower boundary. This boundary is considered to be heated with a constant heat flux q_s''. The upper boundary of the minichannel is considered to be adiabatic so that both co-flows advect the total heat transfer from the source. In addition, the secondary fluid is considered to flow between the upper boundary of the minichannel and the upper boundary of the primary fluid flow in a co-current direction as shown in Fig. 1. The resulting system has a perfect direct contact interface between the fluids. This results in the maximum heat transfer enhancement ratio.

The height of the primary fluid flow is δ while that of the secondary fluid flow is $H-\delta$. Both fluids have the same inlet pressure p_i and the same exit pressure p_o. The x-axis is taken along the minichannel starting from its inlet as shown in Fig. 1. The y_1-axis is taken along the minichannel transverse direction starting from its lower boundary. The y_2-axis is taken along the minichannel transverse direction starting from the upper boundary of the primary fluid flow at $y_1 = \delta$ as shown in Fig. 1. The momentum and energy equations for internal flows are applicable to the present problem [26,27]. These equations for both fluids have the following dimensionless forms:

$$\frac{\partial \overline{u}_{1,2}}{\partial \overline{x}} + \frac{\partial \overline{v}_{1,2}}{\partial \overline{y}_{1,2}} = 0 \qquad (1a, 1b)$$

$$a_{1,2}Re_{1,2}\left[\overline{u}_{1,2}\frac{\partial \overline{u}_{1,2}}{\partial \overline{x}} + \overline{v}_{1,2}\frac{\partial \overline{u}_{1,2}}{\partial \overline{y}_{1,2}}\right] = -12\frac{d\overline{p}_{1,2}}{d\overline{x}} + (a_{1,2})^2\frac{\partial^2 \overline{u}_{1,2}}{\partial \overline{x}^2} + \frac{\partial^2 \overline{u}_{1,2}}{\partial \overline{y}_{1,2}^2} \qquad (2a, 2b)$$

$$a_{1,2}Re_{1,2}Pr_{1,2}\left[\overline{u}_{1,2}\frac{\partial \theta_{1,2}}{\partial \overline{x}} + \overline{v}_{1,2}\frac{\partial \theta_{1,2}}{\partial \overline{y}_{1,2}}\right] = (a_{1,2})^2\frac{\partial^2 \theta_{1,2}}{\partial \overline{x}^2} + \frac{\partial^2 \theta_{1,2}}{\partial \overline{y}_{1,2}^2} \qquad (3a, 3b)$$

where

$$\overline{x} = \frac{x}{L}; \quad \overline{y}_1 = \frac{y_1}{\delta}; \quad \overline{y}_2 = \frac{y_2}{H-\delta}; \qquad (4a\text{-}c)$$

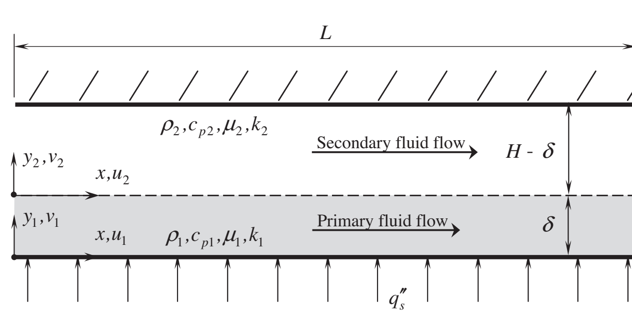

Fig. 1. Schematic diagram the coordinates system for the two dimensional channel with two co-flows of immiscible fluids.