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Thermal analysis of a solar collector absorber plate with microchannels

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ABSTRACT

Experimental and theoretical analyses were carried out to investigate the absorber plate temperature distribution for compact (thin and light-weight) solar thermal collectors. An analytic model combining convective heat transfer with axial conduction in the metal plate was developed. Forced convection experiments were then performed on an instrumented metal plate with micro-channels 0.5 mm × 2 mm × 270 mm long, at various flow rates; the heat transfer fluid was Tyfocor® LS. Reynolds numbers were in the range 10–100 and fluid inlet temperatures ranged from 5 to 60 °C. The predicted plate temperature profiles from the analytic model were in close agreement with the measured profiles. Thermal entry lengths were found to be significant and resulted in slight variations at the entry portion of the plate at higher flow rates. The model was used to study the effects of varying design/operating parameters and showed that axial conduction can significantly alter the temperature profile in the plate.

1. Introduction

Compact solar thermal collectors are an architecturally attractive option for incorporation in buildings as an alternative to conventional flat panel or evacuated tube collectors. Several studies dedicated to improving the thermal performance of flat plate collectors have been done, some of these have focused on achieving better performance by a low temperature difference between plate and fluid [1] for example, Matrawy and Farkas [2] investigated parallel and serpentine tube arrangements in conventional sheet and tube absorber plates while Rommel and Moock [3] analytically studied solar absorbers with rectangular fluid ducts. Alvarez et al. [4] investigated a flat-plate collector with a surface contact with the heat transport fluid. The microchannel absorber plate design proposed in this study offers the potential of high heat transfer coefficient (due to the small hydraulic diameters of the channels) [5] and a fairly uniform temperature distribution in the transverse direction. These accomplish the aim of a low temperature difference between plate and fluid; it also eliminates the challenge of hotspots encountered in the conventional solar collector arrangement with tubes bonded to sheet.

The correlations and experimental results for heat transfer in conventional sized channels are well established; for fully developed laminar flow, a constant Nusselt number, whose value depends on the cross-sectional geometry and boundary conditions, is expected. For example, in circular tubes with constant axial heat flux, the average Nusselt number, Nu = 4.364 while Nu = 3.657 for a constant axial wall temperature boundary condition [6]. In fully developed flow through rectangular channels, depending on the number of walls transferring heat and the aspect ratio, Nu ranges from 0.457 to 7.541 for constant wall temperature, and 0.538 to 8.235 for constant wall heat flux [7]. Several correlations for estimating the Nusselt number in developing flow are available, Wibulswas [8] presented correlations for non-circular ducts. Most of the results are obtained when the assumptions that the heat transfer in the duct wall is negligible and that a simplified boundary conditions (constant wall temperature or constant wall heat flux) exists.

Tuckerman and Pease [9] pioneered the study of heat transfer in microchannels over 3 decades ago by designing and testing a very compact, water-cooled integral heat sink for silicon integrated circuits. Since then, heat transfer in micro-channels has been extensively studied, however, published results are still inconsistent; some studies have found the average Nusselt number to be Reynolds number dependent in the laminar regime [10–12], some recorded lower Nusselt numbers [13–15] while some recorded higher Nusselt numbers [16–18]. Reviews on experimental and numerical studies of heat transfer in microchannels published by Mokrani et al. [19], Hetsroni et al. [20], Rosa et al. [21] and Sobhan and Garimella [22] confirm the very large scatter in published results and attribute this to "scaling effects", which arise from neglecting phenomenon which are insignificant in conventional sized channels but become significant with the high channel wall surface to fluid volume ratio in microchannel flow. Some examples of these include surface roughness [14], entrance and exit effects

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Nomenclature

A	longitudinal cross sectional area (m^2)
A_{pc}	ratio of top surface to channel surface (–)
a	channel depth (m)
b	channel width (m)
c_p	specific heat capacity (J/kg K)
D_h	hydraulic diameter (m)
h	heat transfer coefficient (W/m^2 K)
k_p	thermal conductivity of metal (W/m K)
k_f	thermal conductivity of fluid (W/m K)
L	length of channel (m)
$\dot{m}$	mass flow rate (kg/s)
N_c	number of channels in plate (–)
P	width of plate (m)
p	channel pitch (m)
Q	heat supplied (W)
q	heat flux from channel walls (W/m^2)
q_t	heat flux on top of plate (W/m^2)
Re	Reynolds number (–)
s	channel surface area per unit length (m^2)
S	plate surface area per unit length (m^2)
S_c	collector channel surface area (m^2)
$T(x)$	plate temperature (K)
T_f	average fluid temperature (K)
T_{in}	fluid temperature at inlet (K)
T_{out}	fluid temperature at outlet (K)
T_p	average plate temperature (K)
$U(x)$	difference of plate & fluid temperature (K)
x	position in flow direction (m)
δ	plate thickness (m)
$\theta(x)$	fluid temperature (K)
μ	dynamic viscosity (Pa s)

[23,24], axial conduction effects [15,25], thermal boundary conditions [26], viscous dissipation effects [27], electric double layer [28] and increased measurement uncertainties [29]. Therefore, it may be difficult to apply traditional correlations to adequately describe the heat transfer in this microchannel absorber plate.

Though heat transfer in micro-channels has been extensively studied over the past 3 decades, this was largely in the context of micro-electronic components; there have been relatively few studies of micro-channel solar absorber plates. Celata's [30] definition of a microchannel as a channel whose hydraulic diameter lies between 1 μm and 1 mm is adopted in this study. Oyinlola and Shire [31] showed that absorber plates with micro/mini-channels, instead of the conventional arrangement, were a viable option for compact solar collectors. Sharma and Diaz [32] are some of the few researchers that have published studies on solar collectors based on mini-channels; they modeled an evacuated tube collector based on mini-channels. This paper details recent analytical and experimental study into the temperature distribution in an absorber plate with microchannels. The objective of this study is to adequately model the heat transfer in this plate to allow for easy investigation of optimum design/operation combination.

2. Analytical model

The experimental apparatus (Section 3) has Tyfocor passing along rectangular passages in an aluminum plate. The fourth side of the passage is formed by a cover plate that is clamped in place hard against the inter-passage ribs. A heating element supplied heat to the channeled plate. The following assumptions are made

1. The flow can be approximated as incompressible and steady state.
2. The system is perfectly insulated thus thermal losses are negligible.
3. Axial thermal conduction in the metal plate is in the flow direction only. The metal temperature is constant over a cross-section and heat transfer occurs through 4 sides of the channel.
4. The thermal properties of the plate and fluid are constant.
5. Typical flow rates for solar collectors usually yield laminar flow with very short entry regions; therefore, the heat transfer coefficient is constant from inlet to outlet.

Fig. 1a shows a cross section of the microchannel plate with thickness δ, at temperature $T(x)$, subjected to a uniform heat flux q_t. A working fluid flows through the channel with a mass flow rate $\dot{m}$ and temperature $\theta(x)$. For analysis purposes, the geometry was modeled as shown in Fig. 1b, it shows a schematic of the heat and mass interaction in an elemental volume with length Δx, a unit width and plate thickness δ. For simplicity, the channel has been modeled as a flat plate having a surface area, s, equivalent to the four sides of the channel. Thermal resistance in the channel walls is negligible due to their small size (0.5 mm × 2 mm). The number of channels, N_c in a unit width of plate is given by

$$N_c = 1/p \tag{1}$$

The surface area available for heat transfer per unit length in one channel is

$$s = 2(a + b) \tag{2}$$

Therefore, the total heat transfer surface area per unit length of plate is given by

$$S = sN_c \tag{3}$$

The ratio of top surface area to the heat transfer surface area is given by

$$A_{pc} = \frac{p}{s} = \frac{P}{S} \tag{4}$$

The heat flux from the channel walls per unit length of collector is

$$q = q_t A_{pc} \tag{5}$$

The cross sectional area for thermal conduction in the longitudinal direction is given by

$$A = \delta P - (N_c ab) \tag{6}$$

The study aims to predict the plate temperature T and fluid temperature θ as a function of distance x in the flow direction. The analysis is commenced by applying the conservation of energy principle on the elemental volume shown in Fig. 1b. Taking an energy balance on the plate and simplifying yields

$$\frac{d^2T}{dx^2} - \frac{Sh}{kA}(T - \theta) + \frac{Sq}{kA} = 0 \tag{7}$$

Similarly, taking an energy balance on the fluid and simplifying yields