Journal of the Taiwan Institute of Chemical Engineers 62 (2016) 259–266

Contents lists available at ScienceDirect

Journal of the Taiwan Institute of Chemical Engineers

journal homepage: www.elsevier.com/locate/jtice

An innovative nanofluid-based cooling using separated natural and forced convection in low Reynolds flows

A. Raisi[a,*], S.M. Aminossadati[b], B. Ghasemi[a]

[a] Engineering Faculty, Shahrekord University, PO Box 115, Shahrekord, Iran
[b] School of Mechanical and Mining Engineering, The University of Queensland, QLD 4072, Australia

ARTICLE INFO

Article history:
Received 27 August 2015
Revised 28 December 2015
Accepted 3 February 2016
Available online 24 February 2016

Keywords:
Forced convection
Natural convection
Channel
Cavity
Heat source
Nanofluid

ABSTRACT

This article numerically studies the thermal performance of a cavity filled with the alumina-water nanofluid and with a discrete heat source at its bottom. The cavity is located on the bottom wall of a horizontal channel; and it is cooled by an external air flow entering the channel with a low Reynolds number and a temperature relatively lower than the heat source temperature. The significance of this study is the unique heat transfer mechanism due to both forced convection of main fluid in the channel and natural convection of nanofluid within the cavity without any mixing between forced and natural convection. The pertinent parameters analysed in this study are the Reynolds number of main fluid in the channel ($1 \le \mathrm{Re} \le 20$), the Rayleigh number ($10^4 \le \mathrm{Ra} \le 10^7$) and the solid volume fraction of alumina-water nanofluid ($0 \le \phi \le 0.04$) in the cavity. A numerical model was generated to simulate the forced convection in the channel, heat conduction in the cavity walls and the natural convection in the cavity. The corresponding governing equations were solved using the SIMPLE algorithm in Fortran. The results were presented in terms of flow and temperature patterns, velocity profile at different cross-sections of the channel and the local and average Nusselt numbers. It was found that the proposed cooling method was effective in removing heat from the heat source and the heat transfer rate increased at higher Reynolds and Rayleigh numbers. It was also found that the nanofluid with a higher solid volume fraction corresponds to a higher heat transfer rate; the rate of heat transfer increase, however, was less pronounced at $\mathrm{Ra} = 10^5$ compared to other values of Rayleigh number.

1. Introduction

Heat Transfer performance of natural and forced convection flow in channels has been an area of investigation over decades due to its importance in various engineering applications such as electronic cooling mechanisms, solar systems and heat exchangers [1,2]. Convection heat transfer in channels has been studied for low to high Reynolds numbers. Low Reynolds number flows have been identified in various engineering problems. Hossain and Floryan [3] studied the mixed convection in a channel with flow driven by a pressure gradient and subject to spatially periodic heating along one of the walls. They argued that three patterns of secondary motion could occur at low Reynolds numbers: longitudinal rolls, transverse rolls and oblique rolls. Sang et al. [4] numerically studied the two-dimensional flow around a circular cylinder at low Reynolds numbers in finite channels. Elatar and Siddiqui [5] experimentally investigated the development of low Reynolds number channel flow during mixed convection. They found that the buoyancy-driven secondary flow was generated right from the upstream tip of the channel heated section and was enhanced in the downstream direction.

Recent technology advancement has enabled manufacturers to significantly reduce the size of thermal devices. The size reduction poses challenges in the heat removal process of these devices. Therefore, most of studies on heat transfer in channel flows have been motivated by a need for enhanced cooling systems. Various passive and active techniques have been proposed, tested and/or implemented in order to increase the heat transfer rate of miniaturised components [6]. These include utilisation of rough or ribbed surfaces, coolant fluid additives, surface vibration, mechanical aids and electrical and acoustic fields [7,8].

Utilisation of nanofluids to improve the heat transfer performance of thermal devices has recently received considerable attention [9]. The main objective of nanofluid studies, where the base coolant fluid is replaced by a nanofluid, is to investigate the improvements of heat transfer in various geometries by using

* Corresponding author. Tel./fax: +983832324438.
E-mail addresses: raisi@eng.sku.ac.ir, afraisi@gmail.com (A. Raisi), uqsamino@uq.edu.au (S.M. Aminossadati), behzadgh@yahoo.com (B. Ghasemi).

http://dx.doi.org/10.1016/j.jtice.2016.02.014
1876-1070/

Nomenclature

A	area, m^2
c_p	specific heat, $J\,kg^{-1}\,K^{-1}$
d	diameter, m
g	gravitational acceleration, $m\,s^{-2}$
h	channel height, m
h_c	cavity height, m
k	thermal conductivity, $W\,m^{-1}K^{-1}$
l	channel length, m
l_1	channel length before the cavity, m
l_2	channel length after the cavity, m
l_c	cavity length, m
l_s	heat source length, m
L_s	dimensionless heat source length (l_s/h)
Nu	Nusselt number
Nu_m	average Nusselt number
p	fluid pressure, Pa
$\overline{p}$	modified pressure $(p + \rho_a g y)$
P	dimensionless pressure $(\overline{p}/\rho_a u_c^2)$
Pe	Peclet number $(u_s\, d_s/\rho_f)$
Pr	Prandtl number (ν_a/α_a)
Ra	Rayleigh number $((g\,\beta_f\,h^3\,(T_h - T_c))/\nu_f\,\alpha_f)$
Re	Reynolds number $(u_c\,h/\nu_a)$
t	cavity thickness, m
T	temperature, K
T_h	heat source temperature, K
T_c	inlet air temperature, K
u, v	velocity components in x, $y$directions, $m.s^{-1}$
U, V	dimensionless velocity components $(u/u_c, v/u_c)$
u_c	inlet flow velocity, $m.s^{-1}$
u_s	Brownian motion velocity $((2\,\kappa_b\,T)/(\pi\,\mu_f\,d_s^2))$, $m.s^{-1}$
x, y	Cartesian coordinates, m
X, Y	dimensionless coordinates $(x/h,\ y/h)$

Greek symbols

α	thermal diffusivity, $m^2\,s^{-1}(k/\rho\,c_p)$
β	thermal expansion coefficient, K^{-1}
ϕ	solid volume fraction
κ_b	Boltzmann constant, JK^{-1}
θ	dimensionless temperature $(T - T_c)/(T_h - T_c)$
μ	dynamic viscosity, $N\,s\,m^{-2}$
ν	kinematic viscosity, $m^2\,s^{-1}$ (μ/ρ)
ρ	density, $kg\,m^{-3}$

Subscripts

a	air
c	cold
eff	effective
f	pure water
h	hot
m	average
nf	alumina-water nanofluid
out	channel outlet
s	nanoparticle
w	cavity wall

nanofluids with different thermophysical properties [10–14]. Microscale cooling devices such as microchannels using nanofluids have been considered as good candidates for heat removal applications [15,16]. It is known that there is turmoil among the research community because of many conflicting measurement results reported after the first exciting report by Choi [17]. Kumar et al. [18] argued that the nanofluid inconsistencies among the models may be due to the assumption made, depending factors such as particle size, shape, volume fraction, temperature, static, and dynamic conditions of nanoparticles in developing the thermal conductivity models.

Research studies on nanofluid heat transfer in channel flows have mainly introduced the use of nanofluid as the working fluid in order to increase the thermal effectiveness of the design [19–23]. Fan et al. [24] argued that the mixed convection heat transfer in a horizontal channel can be enhanced when nanofluids are used. Khoshvaght-Aliabadi and Hormozi [25] compared the heat transfer performances of the base fluid flow in a plain channel and the nanofluid flow in a pin channel. They found noticeable performance enhancement by utilising pin channel and nanofluid inside the plate-fin heat exchanger. Although many studies on natural, forced or/and mixed convection of pure fluid as well as nanofluids have been reported in the literature, to the best knowledge of authors, there has not been any study reported in literature that introduces the use of a combination of the natural convection of nanofluid in a cavity and the forced convection of fluid in the main channel without any mixing in the design of cooling systems. Thus, the aim of this study is to investigate the heat removal performance of a horizontal channel in which air with a relatively low temperature enters and flows over a nanofluid filled cavity where a heat source with a relatively high constant temperature is located in the centre of the cavity bottom wall.

2. Problem description

Fig. 1 displays the schematic diagram of a two-dimensional horizontal channel with a closed cavity located on its bottom wall.

The geometrical parameters of the channel are: $l/h = 20$, $l_1/h = 8$, $l_2/h = 10$ where, h is the channel height, l is the channel length, l_1 is the channel length before the cavity, l_2 is the channel length after the cavity.

The cavity has the following dimensions: $t/h = 0.05$, $h_c/h = 0.5$, $l_c/h = 2$ where, t is the thickness of cavity walls, h_c is the cavity height and l_c is the cavity length.

There is a constant temperature heat source ($T_h = 310\,K$) located in the centre of the bottom wall of the cavity. The heat source has the following dimension: $l_s/h = 0.5$ where, l_s is the heat source length.

The cavity is filled with the alumina-water nanofluid with various solid volume fractions ($0 \le \phi \le 0.04$). The air with the Prandtl number of Pr=0.707 and a low Reynolds number ($1 \le \mathrm{Re} \le 20$) enters the channel at a temperature of $T_c = 298\,K$. The thermophysical properties of air, water and alumina nanoparticles are presented in Table 1 [26,27]. The nanofluid is considered to be a single-phase and homogenous mixture of pure fluid and nanoparticles in thermal equilibrium with the same velocity. The nanoparticles are assumed to be spherical with equal diameters. The properties of nanofluid depend on the properties of pure fluid and nanoparticles, the size of nanoparticles and the temperature of nanofluid.

Table 1
Thermo-physical properties of water, alumina and air [26,27].

	Pr	ρ (kg/m^3)	C_P (J/kg K)	K (W/m K)	β (K^{-1})
Water	6.2	997.1	4179	0.613	2.1×10^{-4}
Alumina (Al_2O_3)		3970	765	40	8.5×10^{-6}
Air	0.707	1.1614	1007	0.0263	3.33×10^{-3}