



Correct interpretation of nanofluid convective heat transfer

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ARTICLE INFO

Keywords:

Convective heat transfer
Newtonian nanofluids
Pipe
Twisted-tape
Coil heat exchanger
Counterflow heat exchanger
Plate heat exchanger

ABSTRACT

Engineers and scientist have a long tradition in trying to improve the thermophysical properties of convective heat carriers such as water and transformer oil. Technological developments of the last decades allow the dispersion of particle of sizes ranging between 10 and 100 nm in these liquids. In a large number of recent studies the resulting nanofluids have been reported to display anomalously high increase of convective heat transfer. The present study compiles experiments from five independent research teams investigating convective heat transfer in nanofluid flow in pipes, pipe with inserted twisted tape, annular counter flow heat exchanger, and coil and plate heat exchangers. The results of all these experiments unequivocally confirm that Newtonian nanofluid flow can be consistently characterized by employing Nusselt number correlations obtained for single-phase heat transfer liquids such as water when the correct thermophysical properties of the nanofluid are utilized. It is also shown that the heat transfer enhancement provided by nanofluids equals the increase in the thermal conductivity of the nanofluid as compared to the base fluid independent of the nanoparticle concentration or material. These results demonstrate that no anomalous phenomena are involved in thermal conduction and forced convection based heat transfer of nanofluids. The experiments are theoretically supported by a fundamental similarity analysis of nanoparticle motion in nanofluid flow.

1. Introduction

Heat transfer is involved in countless industrial applications, including nuclear reactors, electronic devices, chemical reactors, engines, etc. Appropriate thermal management is a must in these systems to maintain reliability and prevent premature failure.

Consider electronic cooling area as an example. Technology advancements result in miniature micro-devices with higher packing density and therewith higher heat generation. This miniaturization leads to an increase in heat flux density that then needs to be dissipated, which is one of the most limiting barriers for technology advancement. Proper thermal management ensures reliable, high performance operation and maximizes the mean time between failures of microelectronic devices. There are a variety of solutions that can be implemented for cooling high power electronic devices using either air or liquid cooling. It has been shown that forced convection liquid cooling in microchannels (both single-phase and multi-phase flows) can provide a

solution for the cooling rate requirements of Microelectromechanical systems (MEMS) [1]. Clearly, the highest rate of heat dissipation can be attained by phase change. However, challenges associated with this solution such as high pressure drop, possible dryout, etc. have hampered the applicability of the phase change heat transfer in electronic cooling [1]. Although heat dissipation capability of single-phase forced convection is much smaller than multi-phase forced convection, it is more reliable and does not suffer from the aforementioned deficiencies related to multi-phase flow.

The effectiveness of single-phase convective heat transfer between a solid surface and a Newtonian fluid depends on fluid's thermophysical properties, its velocity, and system geometry. Hence, the generic functional form of all empirical correlations for prediction of single-phase forced convection heat transfer reads

$$Nu = f(GEO, Re, Pr) \quad (1.1)$$

where Nu denotes the Nusselt number ($h l/k$), GEO is some geometry

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Nomenclature

A	surface area, m^2
a	thermal diffusivity, $a = k \rho^{-1} c_p^{-1}$, $m^2 s^{-1}$
a, b	constant parameters
b_p	plate depth, m
c_p	specific heat capacity, $J kg^{-1} K^{-1}$
d	diameter, m
D	diffusion coefficient, $m^2 s^{-1}$
D_c	coil diameter, m
De	Dean number, $De = \rho u d \mu^{-1} (d/D)^{0.5}$
f	friction factor
$ Fo$	Fourier number, $ Fo = \tau k d^{-1} \rho^{-1} c_p^{-1}$
G	heat exchanger conductance, $W K^{-1}$
GEO	geometrical parameter
h	heat transfer coefficient, $W m^{-2} K^{-1}$
HTC	heat transfer coefficient, $W m^{-2} K^{-1}$
I	rotational momentum of inertia
k	thermal conductivity, $W m^{-1} K^{-1}$
K_{pe}, K_{Nu}	weighting factors
l	length, m
L	plate length, characteristic length, m
$LMTD$	logarithmic mean temperature
\dot{m}	mass flow rate, $kg s^{-1}$
n	number of turns in coil heat exchanger
Nu	local Nusselt number, $Nu = h d k^{-1}$
\overline{Nu}	Nusselt number averaged over device
Δp	pressure loss, Pa
p	coil pitch, m
Pe	Péclet number, $Pe = u d \rho c_p k^{-1}$
Pr	Prandtl number, $Pr = \mu c_p k^{-1}$
Q	heat amount, W
q	heat flux, $W m^{-2}$
Re	Reynolds number, $Re = \rho u d \mu^{-1}$
RH	ratio of heat transfer coefficient
RNu	ratio of Nusselt numbers
Sc	Schmidt number, $Sc = \nu D^{-1}$
T	temperature, K
U	overall heat transfer coefficient, $W m^{-2} K^{-1}$
u	velocity in streamwise direction, $m s^{-1}$
V	volume, m^3
v	velocity in wall normal direction, $m s^{-1}$
w	velocity in crossflow direction, $m s^{-1}$
x	streamwise/axial coordinate, m
y	wall normal/radial coordinate, m
z	crossflow coordinate, m

Greek symbols

ϕ	volumetric concentration of nanoparticles, %
Φ	surface enlargement ratio
φ	mass concentration of nanoparticles, %

δ	thickness (of wall etc.), mm
Δ	indicates relative differences of compared pairs, %
ζ	normalised pressure loss coefficient,
μ	dynamic viscosity, $kg m^{-1} s^{-1}$
ν	kinematic viscosity, $m^2 s^{-1}$
Π	similarity number
ρ	density, $kg m^{-3}$
τ	time scale, s

Subscripts

a	annulus side
b	bulk
con	convection
B	Brownian motion
bf	base fluid
c	cold side
DB	Dittus-Boelter
dif	diffusion
e	equivalent
exp	experimental
h	hot side
H ₂ O	water
i	inner
in	inlet
nf	nanofluid
np	nanoparticle
o	outer
out	outlet
R	scale basis
ref	reference case
RePr	with respect to Reynolds and Prandtl number
s	sedimentation
T	thermophoresis
w	wall
proc	process

Superscripts

*	dimensional
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Abbreviations

EMT	effective medium theory
ILK	Institut für Luft-und Kältetechnik Dresden
MIT	Massachusetts Institute of Technology
MWCNT	multi-walled carbon nanotube
NF	nanofluid
PdI	polydispersity index of the particle size distribution
PHE	plate heat exchanger
UJI	Universidad Jaume I

factor, Re is the Reynolds number ($\rho V l \mu^{-1}$) and Pr is the Prandtl number ($c_p \mu k^{-1}$).

The simplest way to increase the amount of heat transferred per unit volume is to enlarge the surface area where heat transfer takes place. However, this method leads to an undesirable increase in size and cost of unit operations. As a result, the most common practices currently are roughening the surface or increasing fluid velocity. While with the first technique the viscous sublayer is disturbed and a wall-normal velocity component is generated, the second approach aims to enhance flow dynamics by increasing the Reynolds number. All of these approaches usually result in greater pumping power. Practical applications show

that these strategies have been already pushed to their limits.

To push the limits further, we are left with the option of changing the thermophysical properties of the working fluid which means changing the Prandtl number Pr and the Reynolds number Re . One obvious option is to replace the working fluid, which is not an easy solution, because in most of industrial applications there are many factors such as safety concerns that dictate the choice of the working fluid and replacement is not trivial. An alternative option is to modify the thermophysical properties of the original fluid by adding dispersed particles.

Dispersing particles with higher thermal conductivity than that of

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