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Effect of buoyancy on turbulent convection heat transfer in corrugated channels – A numerical study

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ABSTRACT

Convection heat transfer of turbulent flow between two corrugated plates in the presence of buoyancy is numerically investigated. Three different channel geometries, with width-to-pitch ratios equal to 0.2, 0.32 and 0.5 are investigated for Reynolds numbers ranging between 2200 and 13,500.

The results show that, for buoyancy-aided flows, the heat transfer decreases with Grashoff number until a certain point where it starts to recover. For buoyancy-opposed flows, heat transfer monotonically increases with Grashoff number. It is the same trend as has already been reported in the literature for vertical pipes; however, for the same Reynolds number, the critical Grashoff number at which buoyancy can affect the heat transfer in a corrugated channel is higher. These Grashoff numbers increase with the width-to-pitch ratio of the channel. If the wall heat flux (thus Grashoff number) is kept constant, Reynolds number must be 3–7 times smaller in corrugated channels with aspect ratios of 0.2–0.5, respectively, compared to a vertical pipe so that buoyancy can affect heat transfer.

1. Introduction

Supercritical fluids are attractive working fluids for various industrial applications including renewable energy [1–4], conventional thermal power plants [5,6], refrigeration [7,8] and heat pumps [9].

A fluid in a pressure higher than the critical pressure (supercritical fluid, in brief) differs from a normal fluid in that when it is heated at constant pressure, there are never two phases in equilibrium. Rather, there is a continuous, but sharp, variation of density from the low-temperature liquid-like fluid to the high-temperature gas-like one [10]. This transition happens within a narrow temperature range in the neighbourhood of a temperature called pseudocritical temperature (T_{pc}). This abrupt change of density gives supercritical fluids the unique feature of having significant buoyancy forces in the conditions where, for other fluid flows, buoyancy is negligible. High buoyancy forces arise when a supercritical fluid exchanges heat with the walls of the conduit in which it flows. It has been observed that whenever buoyancy force and mean flow are in the same direction (upward heating or downward cooling) the presence of buoyancy can impair heat transfer; and vice versa [11–13]. This phenomenon can be explained as a result of buoyancy changing the level of turbulence production through its influence on the mean velocity profile. For more details see Refs. [11,14,15]. As turbulence diffusion plays a key role in heat transfer, any change in turbulence production will be influential on the heat transfer coefficient. It must be mentioned that although this phenomenon has been widely studied in the context of supercritical fluids, it can happen in any flow with significant buoyancy [15,16].

An increasing number of experimental [13,17–20] and numerical [21–29] investigations on the effect of buoyancy on supercritical heat transfer have been reported in the last decade. The major challenge of the numerical studies in this field is the right choice of turbulence models. It has been shown that conventional turbulence models (i.e. k-ε and k-ω with empirical wall functions) are not effective due to the complexity of the near-wall region in this kind of flow [30,31]. Mikielewicz et al. [21] examined a number of eddy viscosity turbulence models against then-available empirical heat transfer data for supercritical CO_2 (SCO_2) and suggested the Low-Reynolds k-ε model of Launder and Sharma [32] to be the most successful model. Bae and Yoo [23] used Direct Numerical Simulation (DNS) to simulate flow and heat transfer of SCO_2 in vertical pipes with the inlet Reynolds number of 5400. He et al. [24] assessed various eddy viscosity turbulence models for the same flow conditions as those of Bae and Yoo, and recommended the V2F model [33]. Du et al. [27] used FLUENT to model SCO_2 heat transfer in horizontal tubes using different Low-Reynolds k-ε models and achieved satisfactory results. Bazargan and Mohseni emphasized on the significance of turbulence buffer layer [26] in supercritical heat transfer. In another paper they suggested the choice of damping function to be determining in the capability

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Nomenclature

Latin symbols

b	channel width
Bo	Buoyancy parameter
C_p	specific heat at constant pressure
$C_{\mu,\varepsilon,1,2,\eta,L}$	model constant
d_h	hydraulic diameter
f	source of turbulence velocity scale
G_k	production of turbulence energy due to density fluctuation
$\vec{g}$	gravitational acceleration vector
Gr	Grashoff number
h	enthalpy
$\hat{i},\hat{j}$	unit vectors
k	turbulence kinetic energy
L	length scale; see Eq. (10)
n	normal-to-wall direction
Nu	Nusselt number
P_k	production of turbulence energy due to shear
P	pressure
p	channel corrugation pitch
Pr	Prandtl number
q	heat flux
Re	Reynolds number ($=d_hU_{mean}/\nu$)
S	strain tensor
T	temperature
$\tilde{t}$	time scale; see Eq. (4)
$\vec{U}$	velocity vector
U	velocity magnitude
u	component of the velocity vector
V	channel volume
$\overline{v^2}$	turbulence velocity scale
x, y	coordinates

Greek symbols

α	angle of pipe with horizontal plane
β	volumetric expansion coefficient
ε	dissipation rate of turbulence energy
θ	angle with y axis
λ	thermal conductivity
μ	kinetic viscosity
ν	kinematic viscosity
ξ	friction factor
ρ	density
σ	turbulent Prandtl number
ϕ	enlargement factor

Subscripts

ave	Area averaged quantity
b	bulk temperature
d	deterioration
FC	forced convection
k	turbulent kinetic energy
$mean$	mass averaged quantity
ref	reference value
t	turbulence
w	wall/measured on the wall
ε	dissipation rate of turbulent energy

of a Low-Reynolds k-ε model in supercritical heat transfer problems with significant buoyancy influence [29].

Instead of using real thermophysical properties of a fluid, some numerical works assumed constant properties along with the Boussinesq approximation for the variations of density with temperature [22,25]. Such an approach facilitates investigating buoyancy effects, as they are isolated from other property variation effects on the thermohydraulics of the problem, and leads to a better understanding of the underlying physics. You et al. [22] adopted this approach for their DNS study. Kim et al. [25] investigated various turbulence models and found the V2F model as the most reliable one. Altogether, most of the previous works in this area have focused on the flows in circular tubes. There are, however, a few reports on other simple geometries such as annuli [34] or rectangular ducts [35], but no study has been done on corrugated channels or other more complicated geometries so far. As will be discussed later, such studies are useful in understanding and modelling the performance of Plate Heat Exchangers (PHEs).

PHEs have been in use since the beginning of the 20th century. On account of their compactness and high thermal efficiency, they have gained an increasing popularity in various industries including HVAC, refrigeration and food processing since the 1980s [36,37]. Although – due to leakage issues – PHEs have been traditionally regarded as low-pressure heat exchangers, development of the advanced brazing/welding technologies and, in particular, new designs such as Plate-and-Shell Heat Exchangers (PSHEs) have made way for their usage in high-pressure applications [36,37]. The present research is, particularly, motivated by the benefits of using PHEs in a supercritical geothermal cycle [1,2].

A plate heat exchanger is composed of a number of plates packed together. On two sides of each plate hot and cold streams flow while exchanging heat through the plate. It is a common practice in PHEs to use corrugated plates and, in order to produce more effective turbulence transport, adjacent plates have different – often opposite – angles of corrugation. Therefore, the fluids flow in the channels formed by corrugated plates with crossing furrow lines (see Fig. 1 for more illustration). First systematic attempts for the prediction of hydraulic and thermal performance of PHEs were made in the early 1970s [38,39]. Since then a rather large number of experimental investigations have been carried out aiming to find reliable correlations for this kind of heat exchangers [40–43]. In spite of all available data, finding a generalized theoretical approach for the prediction of pressure drop and heat transfer in PHEs is hardly possible because of the variety of designs and the wide range of

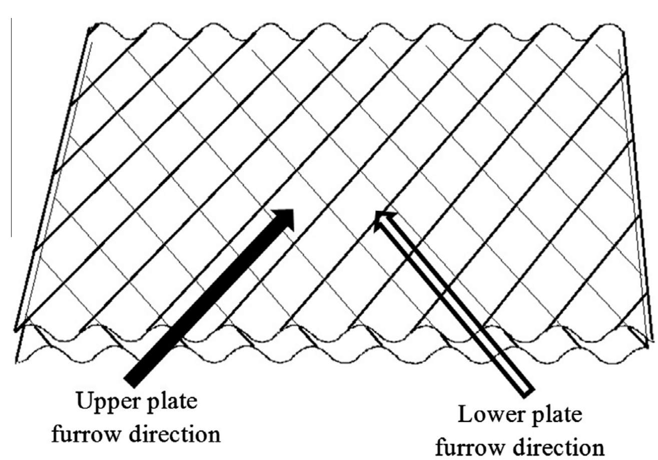

Fig. 1. Channel formed with two corrugated plates with crossing furrow directions. Narrow lines are corrugation lines of the lower plate.