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Numerical investigation of turbulent aided mixed convection of liquid metal flow through a concentric annulus

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ABSTRACT

Turbulent aided mixed convection of a liquid metal with $Pr = 0.021$ in a concentric heated annulus is investigated by solving the Reynolds-Averaged-Navier-Stokes equations. This geometry approximates rod bundles heat exchangers at high pitch-to-diameter ratios. Two inner-to-outer radius ratios of 0.13 and 0.5 are considered and a constant uniform heat flux is applied only to the inner wall, only to the outer wall or to both walls. Constant thermo-physical properties are assumed and buoyancy is accounted for in the momentum equation using the Boussinesq assumption. Four different eddy-viscosity models are first assessed against the few available experimental data for a pipe flow. The turbulent heat fluxes are modeled with the Simple-Gradient-Diffusion-Hypothesis and the turbulent Prandtl number is locally evaluated either with a correlation or by solving one additional transport equation for the temperature variance and one for its dissipation rate. The first approach gives a better agreement with the experimental data. It is found that, compared to medium-to-high Prandtl number fluids, the Reynolds number has a much greater influence on the onset and magnitude of heat transfer impairment. Its extent and degree are less than for ordinary fluids. It is shown that, contrarily to a pipe flow where liquid metals with $Pr \approx 0.025$ behave similar to air or water, in the concentric annulus big differences exist. The reason is the considerable contribution of molecular heat transfer in liquid metals that compensates the reduced turbulent mixing due to buoyancy.

1. Introduction

The demand of reliable and clean energy at low prices poses a great challenge to the world. The study of liquid metals is very important for this purpose since they are considered excellent coolant fluids for many advanced applications. Their very large molecular conductivity, resulting in $Pr \sim \mathcal{O}(10^{-2} \div 10^{-3})$, makes them able to exchange energy more efficiently and with smaller surfaces than conventional fluids. Therefore they are attractive when the size and weight of the heat exchange devices should be limited and when high thermal loads are present. The underlying physical mechanism of heat transfer to liquid metals significantly differs from that of gases or ordinary liquids. Indeed, the contribution of the molecular thermal conduction to the total heat transfer is much higher for liquid metals than for order one and higher Prandtl number fluids.

Convective heat transfer can be classified depending on which mechanism generates the flow motion.

- *Forced convection* occurs when external forces induce the flow
- *Natural convection* occurs when the flow is induced by gravitational forces due to density non-uniformity caused by temperature variations
- *Mixed convection* occurs when the previous modes act together, not as a simple superposition of effects but in a complex modification of flow and turbulence field

Mixed convection is encountered in many engineering applications, among others heat exchangers, nuclear and solar reactors, chemical plants and cooling of electronic components.

Depending on flow direction and thermal boundary conditions it can be classified in *aided* or *opposed* mixed convection. The first occurs for a vertical upward heated or downward cooled flow, i.e. when the buoyancy forces act in the same direction of the flow. Contrarily, opposed mixed convection occurs for a vertical downward heated or upward cooled flow, i.e. when the buoyancy forces act in the opposite direction of the flow. In laminar flows only the

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Nomenclature

Roman letters

Bo	Buoyancy number, Eq. (1) (–)
C_D	Darcy–Weisbach friction factor (–)
C_f/C_{f0}	mixed-to-forced friction factor ratio (–)
d_h	hydraulic diameter (m)
f	re-distribution function (–)
g	acceleration of gravity (m/s^2)
Gr_q	Grashof number, Eq. (2) (–)
k	turbulent kinetic energy (m^2/s^3)
k_θ	variance of temp. fluctuations, Eq. (16) (K^2)
Nu	Nusselt number (–)
Nu/Nu_0	mixed-to-forced Nusselt number ratio (–)
P^*	non-dimensional pressure $\frac{P-gx_i\delta_{i1}}{\rho u_b^2}$ (–)
Pe	Péclet number $Pe = RePr$ (–)
Pr	Prandtl number (–)
Pr_t	turbulent Prandtl number $Pr_t = \frac{\nu_t}{\alpha_t}$ (–)
q^*	non-dimensional molecular or turb. heat flux, (see Section 6.2) (–)
q_w^*	surface averaged wall heat flux, Eq. (25) (W/m^2)
q_w	wall heat flux (W/m^2)
r	radial coordinate (m)
r^*	non-dimensional radial coordinate $r^* = \frac{r-r_i}{r_o-r_i}$ (–)
Re	Reynolds number $Re = u_b d_h/\nu$ (–)
r	pipe radius (m)
S_{ij}	strain rate tensor $S_{ij} = 0.5\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)$ (s^{-1})
T	temperature (K)
T_{b0}	inlet bulk temperature (–)
u_i	velocity component (m/s)
u_i^*	non-dimensional velocity component u_i/u_b (–)
u^*	non-dimensional streamwise velocity (–)
u_b	bulk velocity (m/s)
$\overline{u'v'}^*$	non-dimensional Reynolds stresses (–)
$\overline{v^2}$	wall normal turbulent stress (m^2/s^2)
$\overline{v'T'}^*$	non-dimensional wall normal turbulent heat flux (–)
x_i	dimensional coordinate (m)
x_i^*	non-dimensional coordinate x_i/d_h (–)
y_w	distance from the nearest wall (m)

Greek letters

α	molecular thermal diffusivity (m^2/s)
α_t	turbulent thermal diffusivity (m^2/s)
β	thermal expansion coefficient (K^{-1})
δ	inner-to-outer radius ratio (–)
δ_{ij}	Kronecker delta (–)
ε	dissipation rate of k (m^2/s^3)
ε_θ	dissipation rate of k_θ, Eq. (17) (K^2/s)
$\tilde{\varepsilon}$	modified diss. rate of k, (see Section 4) (m^2/s^3)
λ	molecular thermal conductivity (W/m K)
ν	kinematic viscosity (m^2/s)
ν_t	turbulent kinematic viscosity (m^2/s)
θ	non-dimensional temperature $\theta = \frac{T-T_{b0}}{q^* d_h/\lambda}$ (–)

Subscripts

i	inner
o	outer
wi	inner wall
wo	outer wall

modifications of the velocity field due to buoyancy affect the heat transfer. For aided mixed convection heat transfer is enhanced, while it is impaired for opposing mixed convection. In turbulent flows not only the distortion of the velocity field (direct effect) due to buoyancy influences the heat transfer but also the consequent modifications of the turbulence field (indirect effect). A qualitative picture for fluids with Pr of about unity and above, for which diffusion of heat by turbulence is of dominant importance, can be given by referring to the flow inside a pipe [1,2]. When buoyancy forces oppose the flow, the velocity close to the pipe wall is retarded but at the same time the production of turbulent kinetic energy is enhanced. The second effect of increased turbulent mixing prevails and heat transfer is enhanced. For buoyancy aided mixed convection the flow is accelerated close to the wall. The modifications of the turbulence field initially imply a decrease of turbulent diffusion resulting in a less effective heat transfer. It then recovers at sufficiently high wall heat fluxes when the turbulence production increases again.

For fluids with a Prandtl number much lower than unity, diffusion of heat by turbulence can be of secondary importance and so, even if buoyancy significantly modifies the turbulence field, its effect on heat transfer can be no longer dominant. As shown experimentally and numerically by Jackson et al. [3], for liquid sodium ($Pr \sim \mathcal{O}(10^{-3})$) enhancement of heat transfer occurs with upward flow due to the increased advection and impairment occurs with downward flow. In fact the reduced influence of turbulence on energy transport across the pipe is the reason for the behavior analogous to that of a laminar flow.

Anyway, according to the experimental work of Buhr et al. [4] and to the numerical one of Cotton et al. [5], liquid metals with a higher Prandtl number like mercury ($Pr \sim \mathcal{O}(10^{-2})$) behave similarly to moderate Prandtl number fluids, such as air or water: heat transfer is impaired with modest buoyancy influence but recovers as buoyancy forces are increased.

Based on semi-empirical considerations, Jackson et al. [1] proposed the buoyancy number, Bo, defined in Eq. (1) to correlate the experimental data of mixed-to-forced Nusselt number ratios in a pipe or channel. Subsequently it has been used by many authors [3,6–10] to define the regions of pure forced, mixed and natural convection for fluids with $Pr \geqslant 0.7$.

$$Bo = 8 \cdot 10^4 \frac{Gr_q}{Re^{3.425} Pr^{0.8}} \tag{1}$$

$$Gr_q = \frac{g\beta d_h^4 q_w}{\nu^2 \lambda} \tag{2}$$

In obtaining Eq. (1), the Dittus-Boelter equation for forced convective heat transfer has been used. The latter is valid for medium-to–high Prandtl number fluids. For low Prandtl number fluids other correlations [5] and approaches [11] should be used, which lead to Bo number expressions different from Eq. (1). Anyway, the latter will be used throughout the present work in order to make a comparison with the results available for ordinary fluids.

The vast majority of experimental and numerical investigation of turbulent mixed convection have been done for air or water flowing inside a uniformly heated pipe [2,3,5,6,10,12–14], as this geometry finds widespread use in practical applications. The only studies the authors are aware of on mixed convection in concentric annuli are the experimental one with water of Wu et al. [8] and the numerical one for CO_2 close to the pseudo-critical point of Forooghi et al. [15] and Bae et al. [16]. They consider a single radius ratio of $\delta = 0.5$ and a heat flux applied on the inner wall. This geometry is particularly interesting and important for the analysis of more complex geometries like rod bundles at high pitch-to-diameter ratios. For the latter the friction factor and Nusselt number values