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Numerical investigation of developing convective heat transfer in a rotating helical pipe☆

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ABSTRACT

Numerical simulation is carried out for heat transfer characteristics of flow in rotating helical pipes. A good agreement has been achieved compared with experimental data from literature. The impacts of both co-rotation and counter rotation on local heat transfer enhancement are discussed in detail. Different developing modes of heat transfer enhancement in laminar and transitional regions are observed. Streamwise variation of circumferential distribution of heat transfer enhancement by rotation exhibits sensitivity to rotation speed, rotation direction and curvature ratio. Within the range of *De* and *Ro* under discussion, the impact of streamwise inertial force is the major factor of heat transfer enhancement for co-rotational cases while the effect of reversed flow and accompanied Dean vortex for counter rotational cases cannot be neglected.

1. Introduction

Helical heat exchangers have been widely applied in many industrial fields including power generation, food processing, chemical engineering, HVAC & refrigeration, and aerospace equipment, due to their compactness, high heat transfer coefficient and low fabrication cost compared to many other heat exchangers with either active or passive design for convective heat transfer intensification [1–6], for which the most significant factor contributing to such enhancement is the formation of secondary flow in the cross-section perpendicular to the bulk flow direction, incurred by the combinatory effect of the Coriolis force and centrifugal force [7] in the rotating frame of reference.

Experimental results on heat transfer and fluid flow characteristics in a curved channel with or without rotation have been reported by many previous investigators. Yildiz et al. [8] analyzed the compressed air flow in a helical pipe with inside spring-shaped wires which yields a 30% heat transfer enhancement at the cost of a 10 fold pressure drop compared with a traditional plain helical tube for which the effect of rotation on heat transfer and pressure drop is tested as well. Semi-empirical correlations are obtained in both cases by incorporating a dimensionless geometric parameter and rotational speed to the conventional form which expresses *Nu* as a power law function of *De* and *Pr*. Chang et al. [9] proved that the transition from the inlet developed turbulent flow to laminar downstream by examining the variation of streamwise local Nusselt number for both the inner and outer walls of the helical pipe with and without a ribbed wall surface. The local Nusselt number correlation depicting this region is developed on the experimental data for smooth-wall-tube cases. Wu et al. [10] compared the effect of the concentration of alumina/water nanofluid on the heat transfer of an annular-channel helical pipe with the conclusion that 1) nanoparticles have negligible impact on laminar–turbulent flow transition, 2) augmentation on heat transfer is not obvious due to the weakened secondary flow by the larger density and viscosity of the nanofluid tested even with the advantageous thermal conductivity, and 3) nanofluids used in the experiments can be treated as homogeneous fluids with the trivial effect of the Brownian motion, thermophoresis and diffusiophoresis on heat transfer and can be well predicted with correlations from literature. Kumar et al. [11] tested the variation of heat transfer and pressure drop with a Dean number for both inner and annulus channels in a tube-in-tube helical heat exchanger with a semicircular baffle in the annular channel for structural support and heat transfer enhancement. The experimental data is compared with those from literature in which a more practical test setup configuration, i.e. counter directional flow in the inner and outer tubes, is believed to be one of the major reasons for the discrepancy. Wael et al. [12] experimentally investigated the heat transfer reduction and drag reducing effect of surfactant solution indicating that flow in helical tubes thermally fully developed after two turns of the coil for both water and surfactant solution flow. And such reduction for either heat transfer or pressure drop is due to the turbulence suppression effect of surfactant solute on self-induced vortices by fluid dynamic instability which is believed to be dominant in the flow being tested.

Numerous numerical analyses have also been issued in regard to curved pipe flow and heat transfer. Ishigaki [13] numerically studied the orthogonally rotating pipe and helical pipe case and noted that temperature profiles of the two cases are similar to each other for the same

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Nomenclature

C_{ij}, Wm^{-3}	Convection term
c_p, J kg^{-1} K^{-1}	Specific heat capacity
C_μ	Empirical constant for modeling of turbulent dissipation rate at the inlet boundary
D, m	Curvature diameter, $D = 2R$
d, m	Diameter of helical tube
De	Dean number
$D_{L,ij}$, W m^{-3}	Molecular diffusion term
$D_{t,ij}$, W m^{-3}	Turbulence diffusion term
E, J kg^{-1}	Total internal energy
F_{ij}, W m^{-3}	Production by system rotation
g, ms^{-2}	Gravity acceleration
G_{ij}, W m^{-3}	Buoyancy production term
I	Turbulence intensity
k, m^2 s^{-2}	Turbulent kinetic energy
l	Turbulent length scale
N	Rotation number, $N = \Omega d^2/\nu$
Nu	Nusselt number
p, Pa	Pressure
P_{ij}, W m^{-3}	Stress production term
Pr	Prandtl number
q, W m^{-2}	Heat flux
Re	Reynolds number
Ro	Rossby number
S, W m^{-3}	Source term
S_h, W m^{-3}	Heat source term
u, m s^{-1}	Velocity
u', m s^{-1}	Fluctuating component of flow velocity in Reynolds decomposition
V^*	Normalized velocity magnitude for rotating cases, $V^* = 1 - V_s/V$
V, m s^{-1}	Velocity magnitude
V_s^*	Normalized velocity magnitude for stationary cases, $V_s^* = (V_s - \mathrm{V_{min}}) / (\mathrm{V_{max}} - \mathrm{V_{min}})$, where $\mathrm{V_{max}}$ and $\mathrm{V_{min}}$ are maximum and minimum velocity magnitude respectively within the computational domain of the current case being investigated
x	Cartesian coordinate

Greek symbols

δ	Curvature ratio, $\delta = d/D$; Kronecker delta (with subscripts, e.g. δ_{ij})
Δ	Absolute deviation
ε	Turbulent dissipation rate, m^2s^{-3}; Levi-Civita symbol (with three subscripts, e.g. ε_{ikm})
ε_{ij}, W m^{-3}	Dissipation term
θ, K	Temperature difference between surface and main flow
λ, W m^{-1} K^{-1}	Turbulent kinetic energy
μ, Pa·s	Dynamic viscosity
ρ, kg m^{-3}	Density
τ, N m^{-2}	Shear stress
φ	Axial angle
ϕ_{ij}, W m^{-3}	Pressure strain
Ω, rad·s^{-1}	Rotation speed
ω, s^{-1}	Specific turbulence dissipation rate

Subscripts

A	Averaged
b	Bulk flow
C	Circumferential
d	Hydraulic diameter
eff	Effective
emp	Empirical
i,j,k,m	Direction index of Cartesian coordinate
pred	Predicted
s	Stationary
t	Turbulence

pair of De and Pr. Nusselt numbers for these two cases are also proved to be almost identical in a wide range of De and Pr. The effect of Pr on heat transfer related contours is also demonstrated. It is indicated that for small Pr, i.e. $Pr = 0.01$, the temperature contours for all cases are nearly concentric which is similar to non-rotational pipe flow. When $Pr \approx 1$ temperature and axial velocity contour are similar to each other and so is the case for temperature contour and secondary flow streamlines at large Pr conditions. Wang [14] proposed a dimensionless parameter describing the ratio of centrifugal-force-based buoyancy to the streamwise inertia force. The corresponding flow regime goes through a multi-pair vortex region and caused a dramatic decrease in heat transfer and frictional pressure drop, where the Nusselt number and friction factor approaches those for forced convection in a stationary straight tube, due to the neutralization of centrifugal, Coriolis and buoyance forces. Chen et al. [15] compared the perturbation solution with the numerical for laminar incompressible flow in a rotating helical pipe with both wall heat flux and peripheral temperature kept constant. The numerical solution is verified by the perturbation solution first. When De is large, the position of the high temperature core in a cross sectional view goes from the outer bend, corresponding to the centrifugal force dominant situation (co-rotational), to the inner bend of the pipeline when the Coriolis force is stronger (counter rotational). The starting point of this shift corresponds to when the effects of Coriolis and centrifugal forces almost counteract each other. The impacts of curvature and torsion on heat transfer are also discussed with the variation of rotation number with the fact that whenever Coriolis and centrifugal forces are well matched in strength the secondary flow becomes weakest and heat transfer is more close to straight pipe flow. Aside from laminar flow, turbulence is also introduced in the discussion by some other authors. Lin and Ebadian [16] applied a standard k-ε model with a standard wall function to a 3/4 turn computational domain. It has been confirmed that helical pitch and curvature substantially affect both circumferential and streamwise distributions of Nu for a thermally developing flow. Sleiti and Kapat [17] carried out a simulation in a rotating square channel with a U-turn. An RSM turbulence model with enhanced wall treatment and PRESTO scheme for pressure interpolation at faces is used with the fluid density approximated under a constant pressure process for ideal gas and piecewise linear functions for viscosity, thermal conductivity and specific heat variations. A constant wall temperature assumption is adopted resulting in a good prediction for the experimental work by Wagner et al. [18] with respect to the Nusselt number ratio on the trailing surface. It is observed at the center of the U bend that increasing Ro will suppress the corner vortices governed by a weakened cross stream Coriolis force due to the fact that streamwise velocity is parallel to rotational angular velocity. On the other hand, at the U-turn exit, increasing the rotation number will enhance the Coriolis force induced vortex, suppressing the rest, given that the density ratio was kept unchanged. Once the density ratio increased, vortices induced by centrifugal buoyancy will reappear at inner corners. Colder fluid near the trailing surface is accelerated by increasing Ro leading to higher Nu and separation of hotter fluid near the leading surface yields enhanced heat transfer as well. At the exit of U-turn, high Ro and density ratio increase the velocity magnitude in both leading and trailing surfaces but the accompanied temperature difference between wall