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Laminar flow field in a viscous liquid impinging jet confined by inclined plane walls

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ABSTRACT

An experimental and numerical investigation was carried out to characterize the isothermal laminar flow of a liquid impinging jet confined by inclined plane walls and emanating from a rectangular duct, where the flow was allowed to become fully-developed. The rectangular duct has an aspect ratio of 13, the plane walls opposite to the impinging jet have an inclination of 12° and the nozzle-to-plate distance (D) is very short, $D/H = 0.8$. The presence of the impact plate is felt upstream the nozzle, inside the rectangular duct, up to $x/H = -0.4$. The flow in the cell is symmetric relative to the x-y and x-z center planes and near the inclined walls the flow separates for Reynolds numbers higher than 208, except close to the side walls where the flow remains attached. The length of the separated flow region, L_R, measured along the inclined wall, is constant in the central portion of the channel with $L_R/H = 0.35$ for $Re = 275$ and dropping to zero before reaching the side walls. The recirculation length increases with the Reynolds number and with the thickness of the outlet channel. There is a three-dimensional effect associated with the finite slenderness of the geometry. It consists of spiraling secondary motions away from the central symmetry plane and toward the side walls, where the fluid merges with the main flow creating a local wall jet, as is discussed in detail.

1. Introduction

Confined impinging jets are frequently used to enhance heat and mass transfer since the high velocity fluid impinging the solid surface creates a very thin boundary layer in addition to the beneficial effect of convective transport. Impinging jets are found in various industrial processes and systems such as in paper and textile drying, steel mills, cooling of turbine blades, tempering of glass and cooling of electronic components. Hence, they have been the subject of many investigations to characterize the flow, as well as the corresponding heat and mass transfer fluxes, and the relation between these fluxes, geometric parameters and inlet flow conditions in particular for the turbulent regime. Early experimental contributions were those of Gardon and Akfirat [10,11] and Korger and Krizek [15], amongst others. With the advent of accurate optical diagnostics in fluid mechanics such as laser-Doppler anemometry (LDA), and more recently particle image velocimetry (PIV), detailed experimental programs have extensively documented some basic flows.

The most obvious way of increasing heat and mass transfer is the promotion of turbulence. Recent investigations on turbulent impinging jets are the extensive turbulence statistics characterization of a circular jet flow impinging on a standing plate by Nishino et al. [23], of Sakakibara et al. [30], for a plane jet and of Senter and Solliec [31] for a slot jet impinging on a moving surface. These flows are also extensively used in the context of two-phase fluids as in spray cooling or in liquid fuel delivery in internal combustion engines, which has fostered a large body of research on dilute and dense spray atomization. A state of the art review in spray cooling is found in Kim [14], whereas Panão [26] reviews spray atomization. Cotler at al. [8] provided a study on the modern application of two-phase impinging jets to electronic cooling systems.

Liquid single-phase impinging jets are usually either submerged or free-surface jets. In submerged jets, very common with air, the liquid fluid issues into a region containing the same fluid at rest, whereas in free-surface jets the liquid jet is surrounded by ambient air or gas. Submerged jets can be unconfined or confined by a surface; in the latter case, there is usually a plate attached to the

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Nomenclature

AR	aspect ratio
D	nozzle-to-plate distance
D_H	hydraulic diameter
g	standard earth gravity
h	height of the channel exit
H	height of rectangular duct
LDA	Laser Doppler Anemometer
Re	Reynolds Number based on the hydraulic diameter of the rectangular duct and on the bulk velocity
U	area-averaged bulk velocity in the rectangular duct
u	fluid velocity along the x direction
v	fluid velocity along the y direction
W	width of the rectangular duct

Greek symbols

α	angle of the sloping wall
μ	fluid viscosity
ρ	fluid density
σ	area ratio between the entrance and the exit channels $\sigma = A_1/A_2$

Superscripts, subscripts and special symbols

a^*	nondimensional streamwise advective acceleration
α_1	profile shape factor for energy at plane 1(cf. Fig. 2(b))
α_2	profile shape factor for energy at plane 2 (cf. Fig. 2(b))
f_1	friction factor at inlet channel (cf. Fig. 2(b))
f_2	friction factor at outlet channel (cf. Fig. 2(b))
fx	expansion (or contraction) factors for mesh spacing in the x direction
fy	expansion (or contraction) factors for mesh spacing in the y direction
fz	expansion (or contraction) factors for mesh spacing in the z direction
L_1	length of the computational inlet channel (cf. Fig. 2(b))
L_2	length of the computational outlet channel (cf. Fig. 2(b))
L_R	recirculation length measured in the inclined plane wall
Nx	number of internal cells in the x direction
Ny	number of internal cells in the y direction
Nz	number of internal cells in the z direction
P_0	stagnation pressure
$\dot{Q}_{in}$	volumetric flow rate
ΔP	dimensionless excess pressure loss
Δp_{F1}	wall friction pressure loss at the entrance channel
Δp_{F2}	wall friction pressure loss at the exit channel
P_{01}	pressure upstream of the cell at plane 01(cf. Fig. 2(b))
P_{02}	pressure downstream of the cell at plane 02(cf. Fig. 2(b))
P_1	pressure at plane 1(cf. Fig. 2(b))
P_2	pressure at plane 2(cf. Fig. 2(b))
$\dot{\varepsilon}^*$	nondimensional variation of the streamwise velocity also denote normalized strain rate
U_0	fully-developed velocity value
U_1	bulk velocity in the rectangular channel, also plane 1 (cf. Fig. 2(b))
V_2	bulk velocity in plane 2 (cf. Fig. 2(b))
u_c	velocity along the cell axis
τ_{xy}	shear stress

nozzle and in the most frequent configuration the plate is parallel to the impinging surface.

Regarding free jets Quinn [29] studied the influence of aspect ratio (AR) for turbulent rectangular free jets with $AR = 2$, 5 and 10. His results show that as the aspect ratio increases, the mixture velocity (liquid and ambient air) increases, while the length of the jet core decreases. Recently, Zhou and Lee [42] measured the flow field and the heat transfer coefficient in a jet with an aspect ratio of 4, at Reynolds numbers between 2715 and 25,000 and for ejector clearances between 1 and 30 heights of the rectangular duct. They quantified the variation of the average and local Nusselt numbers with the Reynolds number, the clearance and the turbulence intensity. In particular, they observed dramatic changes in the stagnation zone. Yang et al. [41] studied experimentally the effect of the shape of the impinging surface; they found higher heat transfer rates for impingement on semi-cylindrical concave surfaces than on planar surfaces on account of the different flow patterns especially away from the stagnation region.

Wolf et al. [40] studied free impinging water jets with a uniform velocity distribution exiting the ejector. This inlet velocity distribution was found to enhance significantly the heat transfer, but this increase was also attributed to increasing turbulence levels. Narayanan et al. [21] also investigated the flow pattern and heat transfer of plane impinging jets addressing specifically the effect of ejector clearance, which was varied between 3.5 and 5 hydraulic diameters. They measured the mean and turbulent velocity fields, the pressure fluctuations and the heat transfer on the impinging plate to show a high heat transfer rate in the impinging zone, followed by a local minimum and then a second maximum at the wall region, at 1.5 and 3.2 hydraulic diameters downstream of the jet center, respectively. The position of the second heat transfer rate peak coincided with a local maximum of velocity fluctuations. Chen and Modi [7] investigated also rectangular impinging turbulent jets aimed to quantify the mass transfer rate.

However, in many cases the fluids are very viscous, or the geometries are small, so the flow cannot sustain turbulence and the flow regime is inevitably laminar. Our ultimate aim is the investigation of the flow characteristics of impinging rectangular jets for non-Newtonian fluids and a comparison with the corresponding Newtonian flow cases. The jet confinement is by inclined plane walls, which extend from the exit of a rectangular duct to a short distance above the impinging plate. In the present paper, we focus exclusively on Newtonian fluids in the laminar regime and we compare the results with turbulent data for Newtonian fluids. Since the viscous fluids are usually solutions of viscous additives or suspensions of particles, their viscosities can be significantly higher than those of the pure solvent, unless the solvent is already a viscous fluid (such as Boger fluids). However, if the flow geometries are of small size, the fluids do not need to be very viscous. With the advent of cheap micro-manufacturing and the consequent widespread use of microfluidics interest on laminar flows is growing again. Here, even flows of water take place at low Reynolds numbers, well inside the laminar flow regime [22]. For instance, advanced computing systems dissipate so much energy that classical gas cooling is insufficient to remove the dissipated energy and liquid flows in microchannels are used for new compact heat exchangers.

An early study on confined impinging jets with inclined walls is by Garimella and Rice [12], who investigated circular jets confined by a conical wall. They divided the impinging jet flow field into three regions: the free-jet region, the impingement region and the wall region. The flow in the free-jet region is axial and it is not