International Journal of Heat and Fluid Flow 50 (2014) 431–444

Contents lists available at ScienceDirect

International Journal of Heat and Fluid Flow

journal homepage: www.elsevier.com/locate/ijhff

# Large-eddy simulation of the interaction of wall jets with external stream

Iftekhar Z. Naqavi [a,*], Paul G. Tucker [a], Yan Liu [b]

[a] Whittle Laboratory, Department of Engineering, Cambridge University, Cambridge, UK
[b] Department of Energy and Power Engineering, Dalian University of Technology, Dalian 116024, China

ARTICLE INFO

*Article history:*
Received 17 September 2012
Received in revised form 8 October 2014
Accepted 14 October 2014
Available online 8 November 2014

*Keywords:*
Wall jet
Turbulent boundary layer
Cutback trailing edge
Film cooling
Boundary layer wall jet interaction

ABSTRACT

Large eddy simulations are performed for a wall jet with an external stream. The external stream is in the form of a heated boundary layer. This is separated from a cold wall jet by a thin plate. The Reynolds number based on the displacement thickness, for the incoming boundary layer is 2776. A series of jet velocity ratios in the range $M = U_j/U_\infty = 0.30-2.30$, is considered. The wall jet and outer stream velocities are $U_j$ and $U_\infty$, respectively. The jets with $M \leqslant 1.0$ develop von-Karman type shed vortices in the wake region. The higher velocity ratio jets with $M > 1.0$ undergo Kelvin–Helmholtz instability and develop closely spaced counter-clockwise rolling structures. These structures determine the mean flow field behaviour and near wall heat transfer. At any given streamwise location adiabatic film-cooling effectiveness for $M < 1.0$ increases rapidly with increasing $M$. For $M > 1.0$ it decays slowly with further increase in $M$. For $M < 1.0$ heat transfer from the hot outer stream to the wall depends on two factors; mean wall normal velocity and wall normal turbulent heat flux. For $M > 1.0$ only a wall normal turbulent heat flux is responsible for heat transfer to the wall. The scaling behaviour shows that the near wall flow scales with wall parameters for all values of $M$. However, scaling in the outer region is highly dependent on $M$. The flow develops towards a boundary layer in the farfield for $M < 1.0$ and towards a wall jet for the highest velocity ratio $M = 2.30$.



## 1. Introduction

Plane, two dimensional wall jets have been studied extensively (Launder and Rodi, 1983; Schneider and Goldstein, 1994; Eriksson et al., 1998; Dejoan and Leschziner, 2005; Ahlman et al., 2007). Wall jets have a complex behaviour. The near wall region, called the inner layer, acts more like a turbulent boundary layer flow. The region away from the wall, or outer layer, acts like a free shear layer. They are also an idealised model for the outflow region of impinging jets and some meteorological phenomena (Lin and Savory, 2010).

In most practical situations wall jets usually have an external stream. Bradshaw and Gee (1962) and Verhoff (1963) made early fundamental studies on wall jets with external streams. They showed that for thin incoming boundary layers with no wake, the jet shear layer absorbs the boundary layer in a short distance. However, the presence of an external stream results in the involvement of several parameters. These include, the ratio of the wall jet bulk velocity, $U_j$, and the external stream velocity $U_\infty$, i.e. $M = U_j/U_\infty$. Also, there are the thickness of the wake plate separating the two streams, incoming turbulence levels and the direction of incoming flows. These parameters determine the evolution of the wall jet. They can be controlled to produce the desired effects in wall jets, depending on their application.

The two major applications of wall jets with external streams are cutback trailing edge (TE) film cooling in gas turbines and the control of the boundary layers over high lift bodies, for example, Coanda jets (Nishino et al., 2010; Dunham, 1968). In both of these cases, wall jets interact with the external stream. However, the desired outcome of the interactions are completely opposite. In the case of TE film cooling a cold stream is introduced as a wall jet along the trailing edge. The objective is to keep the external hot stream (combustion gases) away from the wall, and, hence to avoid the mixing of the two streams as far downstream as possible. For the Coanda jet, to prevent the boundary layer separation a strong mixing of two streams is required. $M$ is usually around 1.5 or less for TE film cooling and $M > 2.0$ for Coanda jet based flow control (Nishino et al., 2010).

In the case of TE film cooling, a major focus is the measurement and prediction of film-cooling effectiveness. Martini and Schulz (2004) performed measurements showing the importance of incoming turbulence. Recent large eddy simulations (LES) of TE

* Corresponding author.

http://dx.doi.org/10.1016/j.ijheatfluidflow.2014.10.014
0142-727X/© 2014 The Authors. Published by Elsevier Inc.

film cooling (Schneider et al., 2010; Schneider et al., 2012) for a series of blowing ratios $M$, in the range of 0.35–1.4, showed large coherent structures shed from the plate separating the two streams. With increasing $M$ three different kinds of coherent structures appear. These are clockwise structures (CS), counter-clockwise structures (CCS) with CS and CCS only, for $M \leqslant 0.95, M \sim 1.25$ and $M = 1.40$, respectively. Moreover with increasing velocity ratios there are three distinct regions $[0.35; 0.65], [0.65; 0.95]$ and $[0.95; 1.40]$, where the effectiveness of film cooling increased, decreased and then increased again. This unusual behaviour was associated with the strength and the rotation of the coherent structures present in the flow.

However, apart from jet velocity ratios the wake plate thickness has a strong influence on the film cooling effectiveness (Taslim et al., 1992). In previous studies (Schneider et al., 2010; Schneider et al., 2012) the wake plate thickness is of the order of the slot height. In the current work, extending the study of Taslim et al. (1992), a LES is performed on a simplified geometry. In the current work the wall jet and outer stream are parallel and separated by a thin wake plate around a tenth of the slot height.

Another important aspect, from the turbulence modelling perspective and a basic physical understanding, is the determination of self-similar behaviour of wall jets. Glauert (1956) has shown that complete self-similarity does not exist. Different velocity and length scales are required for inner and outer layers. Various parameters have been suggested to scale the mean velocity and Reynolds stress profiles. Narashima et al. (1973) suggested the jet momentum flux and kinematic viscosity as scaling parameters. These have been used to determine skin friction and show self similarity for a variety of jet nozzle Reynolds numbers (Wygnanski et al., 1992). George et al. (2000) have shown that with appropriate scaling, profiles collapse at infinitely large Reynolds number. For the inner layer, friction velocity $u_\tau$ and $\nu/u_\tau$ are the velocity and length scales. The outer layer is scaled with maximum velocity $U_{max}$ and $y_{1/2}$. The local maximum velocity in the wall jet at any given streamwise location is defined as $U_{max}$, where $y_{1/2}$ is the distance from the wall at which the velocity drops to half of its maximum value for a given streamwise location. The Reynolds shear stress in the outer layer scales with friction velocity $u_\tau^2$, whereas normal stresses and mean velocities scale with maximum velocity $U_{max}$. In the presence of an external stream, a wall jet does not show self-similarity with the velocity and length scales noted above. Hence, in this paper we also present a scaling analysis to explore the similarity of the wall jet with an external stream.

## 2. Problem formulation

The interaction of wall jet with external stream is simulated with filtered conservation of mass and momentum equations for incompressible flow:

$$\frac{\partial \bar{u}_j}{\partial x_j} = 0 \tag{1}$$

$$\frac{\partial \bar{u}_i}{\partial t} + \frac{\partial \bar{u}_j \bar{u}_i}{\partial x_j} = \frac{1}{Re_{\delta_o^*}} \frac{\partial^2 \bar{u}_i}{\partial x_j \partial x_j} - \frac{\partial \bar{p}}{\partial x_i} - \frac{\partial \tau_{ij}}{\partial x_j}; \tag{2}$$

The $\bar{u}_i$ represents the filtered streamwise $\bar{u}$, wall-normal $\bar{v}$, and spanwise $\bar{w}$ velocity components respectively, in a Cartesian coordinate system. $\bar{p}$ is the filtered pressure. The Reynolds number is $Re_{\delta_o^*} = \frac{U_\infty \delta_o^*}{\nu}$, where $\delta_o^*$ is the displacement thickness of the inlet boundary layer and $\nu$ is the kinematic viscosity. The subgrid-scale (SGS) stresses, $\tau_{ij} = \overline{u_i u_j} - \bar{u}_i \bar{u}_j$, are modelled using the Lagrangian-averaged dynamic eddy-viscosity model (Meneveau et al., 1996).

To study heat transfer effects, a temperature transport equation is considered:

$$\frac{\partial \overline{T}}{\partial t} + \frac{\partial \overline{T} \bar{u}_i}{\partial x_i} = \frac{1}{Re_{\delta_o^*} Pr} \frac{\partial^2 \overline{T}}{\partial x_i \partial x_i} - \frac{\partial \overline{T}^{sgs}}{\partial x_i}; \tag{3}$$

where $\overline{T}$ is the filtered temperature, $Pr$ is the Prandtl number and $\overline{T}^{sgs} = \overline{Tu_i} - \overline{T}\bar{u}_i$, is the sub-grid temperature flux. $\overline{T}^{sgs}$ is modelled following a dynamic eddy-diffusivity model (Moin et al., 1991).

These governing equations are discretised with a second-order collocated finite volume method, with a semi-implicit time advancement scheme. The convection term in Eq. (3) is discretized with the QUICK (Quadratic Upstream Interpolation for Convective Kinetics) scheme (Leonard, 1979). The Crank–Nicolson scheme is used for wall normal viscous terms and Adam–Bashforth scheme is applied to all other terms in the momentum equation. The solution of the Poisson equation is achieved by applying Fourier transforms in the spanwise direction and solving the resulting penta-diagonal system iteratively with a stabilised Bi-Conjugate gradient method. The code is parallelized with MPI and has been used in various LES studies previously (Radhakrishnan et al., 2006, 2008).

In this study, the focus is on the interaction of the wall jet with an external stream for a range of blowing ratios $M$ = 0.30, 0.45, 0.60, 0.75, 0.90, 1.05, 1.20, 1.35, 1.50 and 2.3. The external stream is in the form of a fully developed turbulent boundary layer separated from the wall jet with a thin plate. This case has been studied experimentally by Kacker and Whitelaw (1971). At the inlet plane of the computational domain, the mean streamwise velocity profile for the wall jet exit and external boundary layer are available from the experiment. This data is used for the current simulations. The experimental boundary layer profile is used to determine $\delta_o^*$ and all lengths are normalised by this. The slot height of the wall jet is $h = 2.767\delta_o^*$ and the thickness of the plate separating the two streams is $t_w = 0.126h = 0.349\delta_o^*$. The inlet Reynolds number, based on the displacement thickness, is $Re_{\delta_o^*} = 2776$. Two different grids are used in this simulation and their details are summarised in Table 1. $L_x/h, L_y/h$ and $L_z/h$ are the normalised streamwise, wall normal and spanwise domain dimensions, respectively. The fine grid is used for all the velocity ratios, the coarse grid is used only for $M = 0.75$ and 2.30 to allow a grid resolution study.

Fig. 1 shows a representative flow field in the domain. The mean profiles for the external stream and the wall jet at the inlet are available from the experiment and are shown in the Fig. 1, however, the time dependent turbulent information is missing. The experimental wall jet mean profile is slightly skewed and close to a channel flow profile. In order to add turbulent fluctuations a separate channel flow simulation is performed at a Reynolds number based on wall jet velocity and slot height. The instantaneous flow fields are saved for that simulation and mean streamwise velocity profile for the channel is removed from them. This results in fluctuating flow field $u'$. The level of turbulence at inlet is not known from Kacker and Whitelaw (1971). The fluctuations from the channel flow simulations are scaled and added into the mean experimental profile for the wall jet to provide a time dependent inlet boundary condition at the slot. Different values for that scaling factor are tested and a value of 0.2 is found suitable to reproduce experimental mean flow profiles, shown in the following section. Similarly, to generate turbulent fluctuations for outer stream a boundary layer simulation is performed with the recycling/rescaling method of Lund et al. (1998). The instantaneous flow fields are again saved for this case. Then again the mean boundary layer profile is removed and instantaneous fluctuations

**Table 1**
Summary of the domain and grid.

| Grid | $L_x/h$ | $L_y/h$ | $L_z/h$ | $N_x$ | $N_y$ | $N_z$ |
|---|---|---|---|---|---|---|
| Coarse | 80.0 | 16.0 | 5.5 | 512 | 145 | 64 |
| Fine | 45.0 | 16.0 | 5.5 | 1026 | 220 | 128 |