LES of fluid and heat flow over a wall-bounded short cylinder at different inflow conditions

D. Borello\textsuperscript{1} and K. Hanjalić\textsuperscript{2}

\textsuperscript{1}Dipartimento di Ingegneria Meccanica e Aerospaziale, Sapienza University of Rome, Italy
\textsuperscript{2}Department of Multi-scale Physics, Delft University of Technology, The Netherlands

E-mail: borello@dma.ing.uniroma1.it

Abstract. We report on LES studies of flow patterns, vortical structures and heat transfer in flows over a short single cylinder of diameter $D$ placed in a plane channel of height $h=0.4D$ in which the bottom wall is heated. The Reynolds number of 6150, based on $D$, corresponds to the water experiments reported by Sahin et al. (2008). For the basic computational domain of $24 \times 14 \times 0.4D$ three different inflow conditions have been considered: a non-turbulent flow with a uniform initial velocity developing along the channel (NT), a fully developed channel flows (FD) (generated a priori) and periodic conditions (PC). The latter boundary conditions have also been considered for two shorter domain lengths of $6D$ and $3D$ corresponding to a cylinder in a compact matrix. For the long domain, despite the length of the channel of 9.5 $D$ before (and after) the cylinder, the inlet conditions show strong effects on the formation and evolution of the multiple vortex systems both in front and behind the cylinder, influencing significantly also friction and heat transfer. Simulations show some agreement with experimental data though the comparison is impaired by the uncertainty in the experimental inflow conditions. For the shortest cylinder spacing the wake never closes and the flow shows enhanced unsteadiness and turbulence level. Interestingly, the comparison for the same short domain ($3D \times 3D$) using the mean temperature at the inflow to this domain as a reference shows the lowest average base-wall Nusselt number in the PC $3D$ case that corresponds to compact heat exchangers.

1. Introduction

Flows over circular cylinders bounded on one or both sides with plane walls are encountered in a number of industrial and environmental situations of practical relevance. Because of rich vortex morphology both upstream and downstream of the cylinder, these flows are also of fundamental interest for studying the dynamics of coherent vortex systems and their interactions with the bounding walls. The focus has often been on the junction of the cylinder
and the bounding wall, which has served as paradigm of a broader family of “junction flows” involving bodies other than cylinder (Simpson, 2001) as found in turbomachinery, external aerodynamics, heat transfer devices, bridge and coastal hydrodynamics. The main feature of such flows is the formation of horseshoe vortices (HSV) in front of the obstacle arising as a consequence of increasing adverse pressure gradient (Escauriaza & Sotiropoulos, 2011) and their interaction with the separating shear layer and the wake behind the cylinder. Such a configuration (with suction at an upstream location) was also used to simulate and investigate some fundamental phenomena in wall boundary layers such as bursting and eruptions which intermittently feed wall-adjacent concentrated vortices into the outer layer (Seal & Smith, 1999).

The topology of the vortex systems in junction flows is known to depend on a number of factors, primarily on whether the incoming flow is laminar or turbulent, on the free-stream turbulence in the case of turbulent inflow, Reynolds number, boundary layer development (i.e. displacement thickness) on the base wall, conditions of the surface and possible vortical structures generated upstream of the obstacle. Placing a short cylinder between closely spaced parallel plane walls, such as found in plate-fin-and-tube heat exchangers or similar devices, may change the vortex topology in the junction corners. The closely spaced bounding walls tends to suppress natural instabilities in front and behind the cylinder and limit the size of HSV, a feature that distinguishes such configuration from the common bundles of long tubes found in other types of heat exchangers. In most industrial applications the focus is on heat transfer and one is dealing with matrix of multiple cylinders (pins, tubes) in different arrangements (in-line or staggered) rather than a single cylinder, and their spacing is another factor that governs the flow pattern, formation and evolution of vortical structures and the dynamics of heat transfer. Still, one can usually observe several distinct vortex systems both in front and behind the cylinder, which differ depending on the inflow conditions and the cylinder arrangement and spacing. Identifying such structures, their wall imprints and their role in local convection is thus the prime prerequisite for understanding heat transport and optimising the configuration geometry for a particular purpose. Experimental and computational studies of flow and heat transfer over a circular cylinder bonded by closely spaced parallel walls (“fins”) relevant to plate-and-fin heat exchanges have been reported in the literature, e.g. Kundu et al. (1992), Tiwari et al (2003), Kim & Song (2003), all confirming the anticipated increase in heat transfer in the stagnation region in front of the obstacle and its decrease in the wake region. However the effects are different depending on the flow configuration and other parameters mentioned above.

Although the configuration considered originates from fin-and-tube heat exchangers, its versatile topological patterns with several vortex systems make it suitable for studying separated flows and heat transfer around bluff bodies confined with closely spaced side walls with different thermal conditions. Because the Reynolds number encountered in fin-and-tube heat exchangers is usually moderate, wall-resolved large-eddy simulation (LES) can serve as a suitable tool for tracking and investigating the dynamics of the dominant vortical structures and their role in heat transport.

In this paper we report on LES study of flow and heat transfer in a simple configuration consisting of a single short cylinder placed between parallel walls of different extents, of which the bottom wall is heated. The hydrodynamic properties of the setup corresponds to one of the configuration (for Re=6150, based on $D$) investigated experimentally with water by Sahin et al. (2008). This work is a continuation of the earlier reported studies (Borello at al. 2010a, 2010b) with focus on effects of the inflow conditions by which a number of above mentioned parameters can be varied. In the course of studies we considered three types of inflows that can be encountered in a real situation: no-turbulent undisturbed incoming flow (NT), fully developed channel flow (FD), and periodic conditions (PC). The first two cases correspond to an isolated cylinder or the ones in the first row in a matrix washed by non-
turbulent and turbulent incoming fluid respectively, whereas the third case mimics a row or in-line matrix, i.e. a representative cylinder placed deeply in a matrix sufficiently far downstream where the full periodicity is established. The periodic conditions (PC) have been considered for three different sizes of the solution domain, corresponding to different streamwise spacing of the pins. The available space does not allow to cover all cases and we presented only a selection of results, focusing on the two limiting cases, NT and PC, anticipating that other configurations of interest falling in between.

2. Flow description and computational issues

2.1. The flow configuration

The flow configuration studied, Fig 1, consists of a single, thermally passive, short cylinder of diameter $D$ placed in the middle of a plane channel having a height of 0.4 $D$ of which one of the walls was heated. The base computational domain extending over $x \times y \times z = 20 \times 14 \times 0.4 D$ correspond to the (isothermal) water experiment by Sahin et al. (2008). In addition, two other, shorter domains have also been considered, with $x=6D$ and $3D$ corresponding to a cylinder placed in an array with shorter streamwise spacing. The Reynolds number based on the cylinder diameter, bulk velocity and fluid viscosity was 6150, was also chosen to correspond to one of the experimental cases of Sahin et al. (2008). The fluid is heated by imposing a constant non-dimensional heat flux of 0.00243 on the bottom wall. All the other solid walls are treated as adiabatic.

Figure 1. Flow configuration and computational mesh.

2.2. Mathematical formalism

The filtered incompressible Navier-Stokes and energy equations were closed by the dynamic Smagorinsky sub-grid scale (sgs) model, free from any empirical coefficients and wall damping functions. The subgrid scale stress and heat flux (SGS) were calculated using eddy viscosity/diffusivity as functions of the characteristic grid size $\Delta = (\Delta x \times \Delta y \times \Delta z)^{1/3}$ and the module of the deformation tensor of the filtered field. The turbulent Prandtl number $Pr_t$ is obtained from the expression of Kays and Crawford (1993). In order to prevent unphysical heat accumulation in the fluid when using periodic inflow-outflow conditions (PC), the energy equation is solved for the “periodic” filtered temperature $\hat{T}$ (Patankar et al., 1977) containing a sink term equivalent to head addition in each cycle. The actual temperature can be deduced for each periodic domain in a matrix $T = \hat{T} + \gamma Ln$, where $L$ denotes the domain period (distance between the two cylinders in a row), $n$ is the period (cylinder) number and $\gamma$
is the correction for the heat input defined as $\gamma = Q / \dot{m} c_p L$, where $Q$ stands for the rate of heat addition (per unit span) to the fluid in the period length $L$, $\dot{m}$ is the mass flow rate and $c_p$ is the specific heat.

2.3. Numerical details
The simulations were performed using the finite volume unstructured code T-FlowS (Niceno and Hanjalić, 2005). The second order central differencing is applied to the discretisation of diffusive and convective terms in the equations for all the variables. A fully implicit three-level scheme was used for the time marching. The coupling of the velocity and pressure fields to ensure mass conservation was achieved by using the SIMPLE algorithm. The Conjugate Gradient solver coupled with Incomplete Cholesky preconditioner is used for the solution of the linearized algebraic system obtained from the discretization of the flow equations. The simulations were conducted using an unstructured 650k cells grid, clustered in the vicinity of the channel wall and the cylinder for the NT and PC 20D cases. The PC 3D and 6D cases show strong unsteadiness and consequently they required much denser grids to develop numerically stable LES solutions (0.9 and 1.7 M cells respectively). The nondimensional time considered ranges from 450 for the PC 20D case to 750 for the NT case covering 22.5 and 37.7 flow through times respectively. The turbulence statistics has been gathered over about 2/3 of the simulation period for each case considered. The computations of shorter PC cases 3D and 6D required many more flow through times to reach a statistically convergence (80 and 60 respectively). The convergence threshold for the mass and momentum conservation was kept at $10^{-8}$.

2.4. Inflow and boundary conditions
For the first of the two generic cases of inflow conditions, denoted as a non-turbulent, (NT), the incoming flow was presumed to have a constant uniform velocity, allowing development of a laminar boundary layer over a short distance of 9.5D, whereas the convective boundary conditions were applied at the outlet cross-section. This case was supposed to mimic the experimental setup of Sahin et al. (2008), which was placed in an open water channel, but, as discussed below, due to insufficient information about the free stream turbulence entering the experimental setup, we are not sure that our computational inflow conditions for the NT case correspond exactly to those in the experiment.

In the second case the inflow conditions were of a fully developed plane channel. This was supposed to be a fully developed (FD) counterpart to the NT case, though, in view of the relatively low channel flow Reynolds number (Re$_h$=2460, corresponding to Re$_D$=6150), the turbulence level is relatively low.

In the third case, fully periodic conditions, (PC), at the inlet and outlet of the flow domain were imposed in the streamwise direction for all variables, accounting for the temperature corrections as discussed above. Because the periodic conditions (PC) correspond to a cylinder place deep in a matrix or array, we considered three sub-cases corresponding to different streamwise spacing of the cylinders: the original domain of 20D and the two shorter domains of 6D and 3D. In most wall-bounded cylindrical pins or tubes encountered in practice will correspond to the ones here considered or to cases in between.

4. Results

4.1. Flow patterns and vortex morphology
Junction flows are characterised by a strong primary horseshoe vortex (HSV) created in the junction corner due to the deceleration of the boundary layer on the base wall. The primary
HSV has the same rotation direction as the vorticity in the approach boundary layer. Several distinct patterns of (usually) multiple separated HSV in front of the obstacle have been identified, consisting of six, four or two vortices. The primary HSV has always the same rotation (vorticity direction) as does the approach boundary layer (Simpson, 2001). Other, usually weaker horseshoe vortices are created upstream from the body with alternating rotation directions to preserve the streamlines continuity. As noted in Introduction, the number of vortices, their size and strength depend on several factors, primarily on the Re number, flow regime, free stream turbulence, surface conditions and inflow conditions. If the incoming boundary layer is laminar, the HSV systems are well organised and periodic. Rouland et al. (2005) noted that in laminar flow the size of the HSV and the base-wall shear seem to increase with the Re number, whereas the opposite occurs if the incoming flow is turbulent. With a turbulent entry usually one dominant HSV appears with a range of associated winding/braiding vortices of a broad spectrum of frequencies, though of weaker intensity. These vortex systems are highly unstable, enhance the generation of turbulence affecting thus the shedding of the vortices behind the obstacle. Strong flow acceleration on the obstacle side stretches the horseshoe vortices which in turn, depending on their nature and number, influence the separation behind the body and, consequently, the formation and extent of the wake.

Figure 2 illustrates the flow and vortical structures for the nonturbulent inflow (NT) and for the streamwise periodic conditions (PC) for three domain lengths, 20, 6 and 3 cylinder diameters. The flow patter is indicated by a set of streamlines originating at selected upstream locations, whereas the vortex structures are identified by the contours of the constant pressure Laplacian, \( \nabla^2 p = 20 \) (which can be related to the Q criterion, but appears to be more informative for flows bounded by compact walls). The structures are coloured by temperature.

![Non-turbulent entry (NT)](image1)

![Periodic conditions (PC) 20D](image2)

![Periodic conditions (PC) 3D](image3)

![Periodic conditions (PC) 6D](image4)

Figure 2: Selected streamlines and vortical structures identified by pressure Laplacian \( \nabla^2 p = 20 \), coloured with temperature. NT: non-turbulent inflow; PC: Periodic streamwise conditions for the domain length of 20D, 6D and 3D.
The characteristic horseshoe vortices (HSV) develop upstream of the cylinder in all cases, Fig. 1, but show a large diversity of strength and morphology for different entry conditions. The number of distinct HSV, their braiding and/or inter-wining, are all very different from one case to another. Whereas with the NT inflow no less than four counter-rotating unsteady HSVs next to each wall can be discerned, in the fully turbulent PC situations the number is reduced to two systems of which one vortex is clearly dominant. In the non-turbulent (NT) inlet the horseshoe vortex structures are initially laminar with the dominant primary and secondary vortex of the same rotation direction and of almost the same strength, with a counter-rotating wall-attached vortex in between as well as additional small HSVs upstream (see also Figs 3 and 4). Their spiralling, clearly visible in Fig. 2, extend long downstream and affect the wake by entraining its fluid into the side mainstream and transiting to turbulence behind the cylinder as a consequence of flow separation and with the wall. The contours of the turbulence kinetic energy in a layer very close to the wall (not shown here), which was used to detect the turbulence level, indicates that the transition occurs in the side separation shear layers interacting with the wall boundary layer beneath the stretched horseshoe vortices immediately after separation. The wake is atypical and very long stretching over two cylinder diameters. Transition occurs also in the wake but a little further downstream from the cylinder. In the recirculation region of the NT case, the Laplacian isosurfaces show no significant vortical structures apart from the main recirculation and spiralling HSV.

Figure 3. Streamlines coloured with velocity magnitude in two characteristic vertical planes: the cylinder midplane and at 3D downstream from the cylinder centre corresponding to the exit of the shortest domain (PC 3D)
In contrast, the PC cases show more chaotic, typically turbulent structures, with only two distinct HSVs, much thinner and closer to the cylinder. The shorter the solution domain (cylinder spacing) the more vigorous is the turbulence in front of the cylinder, with a broader spectrum of vortical structures. The primary vortex is much stronger than the secondary one in all cases.

Figures 3 and 4 illustrate further the above observations. The instantaneous snapshots show for the NT case that the flow is steady with an almost symmetric vortical pattern near the upper wall. For the FD case the pattern is similar (note a very low channel Re number), though with the secondary HSVs stronger than in the NT case. In contrast, the PC simulations show a rich structure morphology, much more unsteadiness and asymmetry between the lower and upper junctions and much shorter recirculation zones close to both walls.

The time averaged streamline patterns show also very different vortex systems, as shown in Fig. 5 for the horizontal plane at 0.05h above the bottom wall. The saddle point (denoted by a circle) is much closer to the cylinder in the fully turbulent PC case (here only 20D case is shown) than in the NT situation. The pattern in the FD case is in between the other two, showing closest agreement with the experiments. In the NT case a nodal point detached from the wall is clearly visible very close to the stagnation point.

Downstream from the cylinder the simulations show even larger variety of flow patterns, as illustrated in Fig. 6. The averaged streamlines (again at 0.05h from the bottom wall) show the typical recirculation bubbles only for the two turbulent inflow cases, the FD and PC. Both patterns show two saddle points (one upstream and the other downstream from the foci). However, due to relatively low Re number, the separation points are typical of laminar flows. The recirculation bubbles are slightly asymmetric and notably different in length. Here the PC pattern is closest to the experimental one. In both cases the recirculations entrain fresh fluid into the wake thus enhancing heat transfer behind the cylinder. In contrast, the streamline pattern with the NT inflow shows a non-standard pattern with a “dead” zone immediately after the cylinder, enveloped by the separated shear layer and practically with almost no entrainment of the downstream fresh fluid.
4.2. Thermal field and heat transfer

The temperature plumes (identified by surfaces of constant temperature) downstream from the cylinder show a relative similarity between the NT and FD case in the sense that the main effect of heat removal are felt upstream from the cylinder. The high temperature fluid, entrapped by the vortices is especially visible in the NT case and less, though noticeable in the FD situation. In contrast, the remnants of the large structures, originating from vortex shedding in the PC cases, show continuous heat transfer especially on sideways along the whole channel, thus also upstream from the cylinder.

A closer look at the Nusselt numbers, Fig. 8 (computed for comparison over the 3D×3D domains for all cases, using the mean temperature along the inlet section of this domain as the reference for each case), confirms the above finding, except that for some reasons a high heat transfer is detected in the stagnation region for the PC 6D case. Interestingly, the highest Nusselt number is achieved for the PC
20D case and the lowest for the PC 3D, corresponding to closely packed cylinders in an array as usually found in fin-and-tube compact heat exchangers.

Figure 8 Nusselt number on the base wall normalized with the corresponding reference temperature, $Nu = D(\partial T / \partial n) / (T_w - T_{ref})$.

5. Conclusions

LES of flow around a short cylinder bounded by flat walls reproduced multiple horseshoe vortices (HVS) in front and a wake behind the cylinder as found in common junction flows. However, the closely spaced bounding walls tend to suppress the instabilities and the activities of the vortex systems as compared with an unbounded junction. The vortex topology is markedly dependent on the inflow conditions. For $Re_D = 6150$, in the case of a nonturbulent entry the flow is steady with symmetric four counter-rotating vortices next to each wall. In contrast, all three cases with periodic streamwise conditions show unsteady asymmetric patterns with usually one dominant primary HSV. For the shortest periodic domain of 3D, corresponding to a compactly spaced row of cylinders, the wake shows no closed recirculation zone as the next cylinder is encountered before the recirculation bubble is closed. The wake zone is highly turbulent and the HSVs are very thin and weak. None of the cases showed satisfactory agreement with experiments, arguably because of uncertainty in mimicking the experimental inflow conditions. The closest agreement in HSV patterns in front of the cylinder is achieved with the uniform and fully developed nonturbulent inflows, while the closed wake size and morphology is found with periodic conditions for the largest domain (PC 20D).
The temperature field and heat transfer reflect the wall imprints of the vortex morphology. A direct comparison of the Nusselt number is not straightforward because of the ambiguity in defining the reference temperature. Nevertheless, to facilitate a comparison in the area around the cylinder, the Nusselt number was computed for all cases for the compact 3Dx3D domain using the mean temperature along the line at the domain inflow cross-section. For such reference temperature the highest average Nusselt number is achieved for the PC 20D and the lowest for the PC 3D case. Surprisingly the temperature in the wake of the cylinder is very high in the NT and PC 3D cases (though for different reasons) resulting in rather low Nusselt numbers. In contrast the PC 6D and 20D show efficient heat removal in the wake.

6. References

BORELLO D., DELIBRA G., HANJALIĆ K. & RISPOLI F. 2010a, LES and hybrid LES/RANS study of flow and heat transfer around a wall-bounded short cylinder. In: J. Peinke, M. Oberlack and A. Talamelli (Eds) Progress in Turbulence III (Proc. iTi Conf., Bertinoro, Italy, 2008), Springer, ISBN 9783642022241

BORELLO D., HANJALIĆ K., DELIBRA G. & RISPOLI F. 2010b, LES study of the effect of inflow conditions on heat transport over a wall-bounded cylinder. Proc. 8th Intern. Symp. On Engineering turbulence modelling and measurements – ETMM8, Marseille, France, 9-11 June 2010, Vol 1: 282-287.

ESCAURIAZA, C. & SOTIROPoulos, F. 2011, Reynolds number effects on coherent dynamics of the turbulent horseshoe vortex system, Flow, Turbulence and Combustion 86(2): 231-262.

KAYS W. M. & CRAWFORD M. 1993, Convective Heat and Mass Transfer, 3rd Edition, McGraw Hill.

KIM J.Y. & SONG T.H., 2003, Effect of tube alignment on the heat/mass transfer from a plate fin and two-tube assembly: naphthalene sublimation results, International Journal of Heat and Mass Transfer 46: 3051-3059

KUNDU D., HAJISHEIKH A. & LOU D. 1992, Heat-transfer in cross-flow over cylinders between 2 parallel plates, ASME Journal of Heat Transfer 114: 558-564.

NIČENO B. & HANJALIĆ K., 2005, Unstructured large eddy and conjugate heat transfer simulations of wall-bounded flows, In M. Faghri and B. Sunden (Eds) Modeling and Simulation of Turbulent Heat Transfer, (Developments in Heat Transfer Series), WIT Press, 1-40.

ROULAND A., SUMMER B. M. FREDSOEM J., MICHELSON, J. 2005, Numerical and experiemntal; investigation of flow and scour around a circular pile. J. Fluid Mech. 534: 351-401.

PATANKAR S.V., LIU C.H. & SPARROW E.M. 1977, Fully developed flow and heat transfer in ducts having streamwise-periodic variations of cross-sectional area, ASME J. Heat Transfer 99: 180-186.

SAHIN B., OZTURK, N. A. & GURLEK, C. 2008, Horseshoe vortex studies in the passage of a model plate fin and tube heat exchanger, Int. J Heat and Fluid Flow 29: 340-351

SEAL, C. V. & SMITH, C.R. 1999, Visualization of a mechanism for tyhree-dimensional interaction and near-wall eruption, J. Fluid Mech. 394: 193-203.

SIMPSON, R. 2001, Junction flows, Ann. Rev. Fluid Mech. 33: 415-443.

TIWARI S., BISWAS G., PRASAD P.L.N. & BASU S, 2003, Numerical prediction of flow and heat transfer in a rectangular channel with a built-in circular tube, ASME Journal of Heat Transfer 125: 413-421