

Imperial College London
Department of Mechanical Engineering

**Simulation, Modelling and Feedback Control
of the flow around a simplified
Square-Back Road Vehicle.**

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Doctor of Philosophy in Mechanical Engineering of Imperial College London.

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Declaration of originality

I hereby declare that the work presented in this thesis is the result of my original research, except where acknowledged in the text. The findings reported throughout this thesis have been presented at several conferences and published or submitted for publication in scientific journals.

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Laurent Dalla Longa

Abstract

The present work investigates the use of wall-resolved *Large Eddy Simulations* (LES) to compute the flow around simplified *square-back road vehicles*. The objective being to simulate numerically the *unsteady* flow past such blunt bluff bodies and to assess the use of a *linear feedback control* technique for drag reduction. Wall-resolved large-eddy simulations are employed as a test-bed, but the control strategy could be transferrable to moving vehicle experiments as both the sensing (base pressure sensors) and the actuation (synthetic jets) are *body-mounted*. The control strategy attempts to reduce the pressure drag by acting on the wake rather than manipulating the flow separation location which is fixed for square-back road vehicles. This technique is known as *direct wake control*. The control technique exploits the link between wake flow fluctuations and mean drag reduction. The flow response to the synthetic jets is characterised using system identification, and controller design is via shaping the frequency response to achieve wake fluctuation attenuation.

The first bluff body studied is an infinite spanwise blunt bluff body called D-shaped body. It exhibits two interacting shear layers and a large recirculation area. The designed controller successfully attenuates base pressure fluctuations, increasing the time-averaged pressure on the body base by 38%.

The second bluff body is a three-dimensional simplified lorry in presence of a fixed floor. The wake flow is composed of four detached shear layers, which form the envelope of a large low-pressure recirculation area. For this case, unforced simulations captured numerically, for the first time, the wake *bi-modality*. Bi-modality manifests as a random displacement (switching) of the wake between preferred off-center locations. The unsteady flowfield is studied in great details using modal decomposition. High-frequency snapshots of the switching sequence allow us to propose an explanation for the triggering of the bi-modal switching. Finally, various feedback control configurations are assessed. Base pressure fluctuations are significantly reduced as targeted but it appears challenging to obtain a clear mean base pressure recovery. The best configuration yields to a mean base pressure increase of 2.1%.

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- Evstafyeva, O., Morgans, A. S. & Dalla Longa, L. 2017 Simulation and feedback control of the Ahmed body flow exhibiting symmetry breaking behaviour. *J. Fluid Mech.* **817**.
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Conferences

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- ERCOFTAC Spring School on Modelling and Control. Montestigliano, Italy, 2015.
- 68th American Physical Society's Division of Fluid Dynamics Meeting. Boston, USA, 2015.
- Second International Conference in Numerical and Experimental Aerodynamics of Road Vehicles and Trains, AEROVEHICLES 2, Göteborg, Sweden 2015.
- UK Turbulence Consortium Conference, Imperial College London. UK, 2017.
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- UK Fluids Conference. Leeds, UK, 2017.

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Chapter 1

Overview

1.1 Background

Carbon dioxide emissions (CO_2) have significantly increased over the past decades and represent a major concern for the environment. It has been shown that the transportation sector and more specifically road vehicles are responsible for a major part of world carbon dioxide emissions (Figure 1.1, International Energy Agency (2017)).

Indeed most vehicles are powered by fossil-fuel based engines that produce CO_2 emissions to overcome the driving resistance. To minimise the impact of vehicle emissions, European Union states have implemented various eco-tax systems based on CO_2 emissions (European Automobile Manufacturers Association (2017)). The CO_2 emissions of a vehicle are measured over a driving cycle meant to represent real vehicle usage. Driving resistance is caused by three main sources: the mechanical energy losses arising from the lack of efficiency of the engine and from the power chain between the engine and the wheels, the friction of the tyres on the road and the air resistance or aerodynamic drag. At highway speeds, the majority of the energy losses are caused by the latter source since the aero-

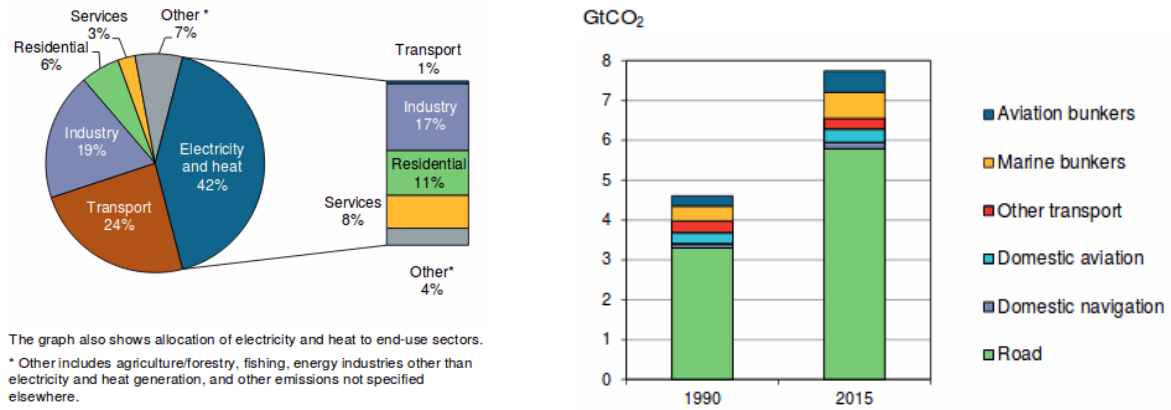


Figure 1.1: (a) World CO₂ emissions by sector in 2015. (b) Comparison of CO₂ emissions from transport between 1990 and 2015 (International Energy Agency (2017)).

dynamic drag is related to the square of the velocity (Hucho (1998)). Hence reducing the aerodynamic drag is very relevant to vehicle manufacturers seeking to comply with emission limits. The recent emergence of electric vehicles also highlighted the importance of aerodynamics. Current battery technologies limit the range to which electric vehicles can operate compared to fuel-based vehicles. Therefore reducing the aerodynamic drag extends the range of a given vehicle, making electric vehicles more competitive.

Because of capacity, comfort and aesthetic requirements, most vehicles are not streamlined but are blunt bluff bodies. Pressure drag dominates over friction drag - they represent respectively 85 % and 15 % of the the total aerodynamic drag for most road vehicles (Gilliéron & Kourta (2011)). The flow past a road vehicle exhibits a large low-pressure recirculation area at the back of the vehicle. The difference in pressure between the high pressure front face and the low pressure wake represents the main source of aerodynamic drag for most road vehicles. Other large sources of drag are the under-body flow, the cooling flow and the flow inside the wheel housings (Hucho (1998)). The time-averaged features of the flow around road vehicles and their impact on the aerodynamic drag have been extensively studied in the past, recent work has focused on wake unsteady dynamics. In particular it was found that the wake past such road vehicle could exhibit a symmetry-breaking behaviour called bi-modality (Grandemange *et al.* (2013b)).

Techniques for reducing the aerodynamic drag of vehicles can be decomposed into two categories. If they are just based on geometrical modifications and do not need any energy input, they are termed “passive” control. If they do require the use of additional energy to achieve drag reduction, the term “active” control is used (Gad-el-Hak (2000)). Passive techniques have been extensively studied over the past decades. However in most cases, they are not practically applicable because of the vehicle design constraints mentioned before. The technique used in this work belongs to the second category and employs synthetic jets also called zero-net-mass-flux (ZNMF) jets. Experimentally, these jets are produced by the vibrating membrane of a loudspeaker placed inside a small cavity which pulses alternately air in and out of this cavity. This technique is promising because not only it does not affect the vehicle geometry, but it also does not necessitate any pumping system to function which saves weight and energy. The solution has been already transferred to real concept vehicles (Volvo Great Dane Trailer, Renault Altica). Interesting gains were achieved but development is still required before the technique could be fully employed on production vehicles.

1.2 Project Goals

The goals of this project are to study numerically the dynamics of simplified square-back vehicle wake flows and to assess the use of active feedback control to reduce their aerodynamic drag. A Large Eddy Simulation solver developed by Lardat & Leschziner (1998) called STREAM-LES is used for the flow simulations. Synthetic jets and body mounted pressure sensors are incorporated to perform feedback control and achieve drag reduction. Two bluff bodies are considered in this work:

- a D-shaped body, with an infinite extent in the spanwise direction. The flow around such a blunt body exhibits a large low pressure wake and two shear layers interacting with each other. This simple case helps to understand the mechanism of vortex shedding and to test the control solutions.
- a simplified lorry geometry in presence of a fixed floor. This bluff body exhibits a three-dimensional wake similar to real square-back road vehicles. Experiments performed on a $\sim 15\%$ scale model of the lorry in the (3.0m x 1.52m) Honda wind tunnel at Imperial College London by Cabitza (2013) serve to validate numerical simulations.

1.3 Outline of the thesis

Chapter 2 reviews some aspects of road vehicle aerodynamics and in particular the sources of aerodynamic drag. Control solutions aiming at reducing aerodynamic drag of such vehicles are also presented. A section is dedicated to synthetic jets, which are used in this work.

Chapter 3 introduces numerical methods to model the flow around such road vehicles. Large Eddy Simulations (LES) are then introduced and the solver used in this work called *Stream-LES* is detailed. This chapter also provides an overview of the previous use of LES to compute the flow around simplified road vehicles.

Chapter 4 presents the numerical results and features of the flows past a D-shaped body. In this chapter, the linear feedback control method used in this work is first introduced. Two actuation strategies are then presented.

Chapter 5 presents the simulations of the flow around a simplified lorry. First, the unforced flow is presented. Unsteady features of this flow and in particular bi-modality are then highlighted using modal decomposition and high-frequency snapshots. To finish, four feedback control strategies are tested.

Chapter 2

Introduction

2.1 Road vehicle aerodynamics

The flow around a road vehicle travelling at highway speeds ($>100\text{km/h}$) is highly turbulent and composed of complex separated flow structures generated by the vehicle features such as the air intakes, the A-pillars, the underbody geometry, the wheels and wheel housings, the side mirrors, the different gaps and steps around the side windows, and also the blunt back face responsible for the large low pressure wake (Figure 2.1). As a result, road vehicles exhibit high aerodynamic drag which represents the main source of energy loss at such speeds (Hucho (1998)).

The action of the fluid flow on a vehicle (D) can be decomposed into two components:

$$D = \left[\int_{S_{veh}} (p_0 - p) \underline{\underline{I}} \vec{n} dS \right] \vec{x} - \left[\int_{S_{veh}} (\underline{\underline{\tau}}_\mu + \underline{\underline{\tau}}_t) \vec{n} dS \right] \vec{x} \quad (2.1)$$

where S_{veh} is the vehicle surface of outwards normal vector $\vec{n}$, p_0 is the pressure far away from the body and p the pressure on the vehicle surface, $\underline{\underline{\tau}}_\mu$ and $\underline{\underline{\tau}}_t$ are the viscous and turbulent strain tensors and $\vec{x}$ is the streamwise vector.

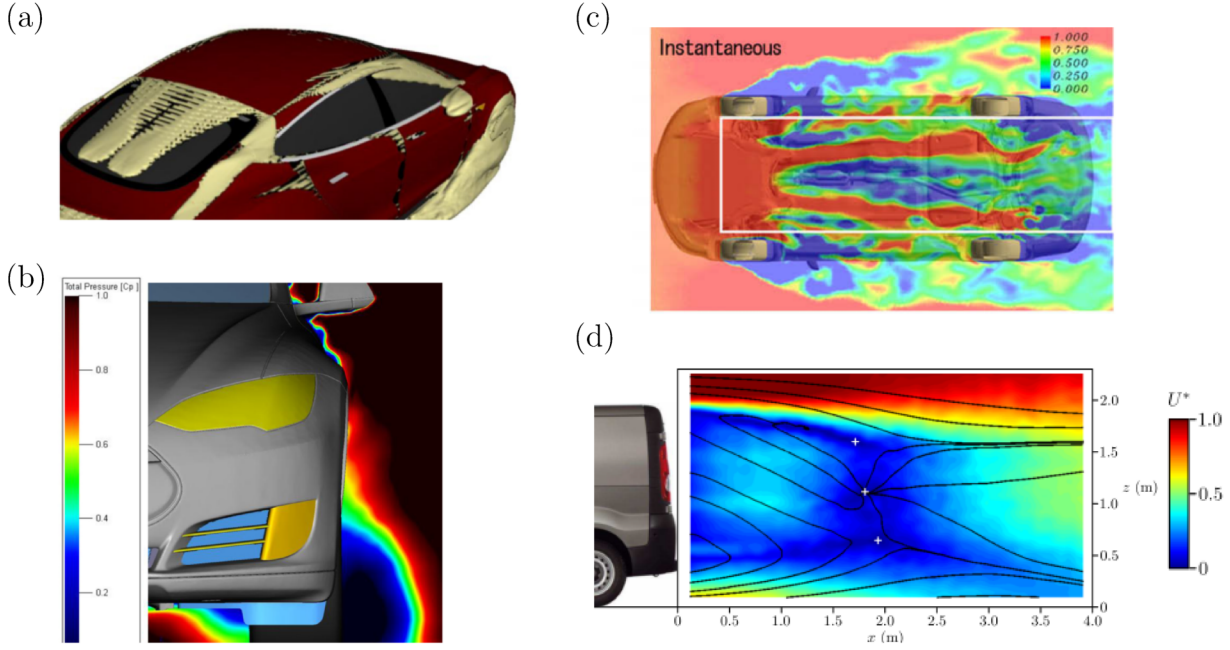


Figure 2.1: Sources of aerodynamic drag for road vehicles. (a) Isosurfaces of total pressure coefficient over a Tesla Model S (Palin *et al.* (2012)). (b) Total pressure coefficient contours behind the front wheel of a Tesla Model S (D’Hooge *et al.* (2012)). (c) Total pressure contours of the underbody flow of a Mazda Atenza (Tsubokura *et al.* (2009)). (d) Velocity in the wake of a Renault Trafic (Grandemange (2013)).

- a pressure drag term, which represents the integral of pressure forces applied on the surface of the vehicle and projected along the streamwise direction, $\vec{x}$
- a friction drag term, which comes from the viscous forces acting in the boundary layer around the vehicle. As mentioned in the introduction, this term is small compared to the pressure drag in the case of a road vehicle.

The sum of both terms represents the aerodynamic drag and is the force obtained in a wind tunnel using the aerodynamic balance.

The aerodynamic drag can also be studied by considering a stream-tube containing the vehicle and looking at the pressure and velocity on the boundaries of the volume. Onorato *et al.* (1984) showed that using this approach, the aerodynamic drag force can be

decomposed as following:

$$D = \underbrace{-\frac{\rho}{2}U^2 \int_S (1 - \frac{u}{U})^2 dS}_{(a)} + \underbrace{\frac{\rho}{2}U^2 \int_S (\frac{v^2}{U^2} + \frac{w^2}{U^2})}_{(b)} + \underbrace{\int_S (p_{i0} - p_i) dS}_{(c)} \quad (2.2)$$

where ρ is the air density, S is the downstream transverse surface where the measurements are done, U is the upstream flow velocity, (u, v, z) are the cartesian components of the flow velocity on the section S , p_{i0} is the upstream stagnation pressure and p_i the stagnation pressure measured at S . The first term of the equation (2.2) (a) represents the loss of streamwise velocity in the wake. This term quantifies the strength of the transverse vortices in the wake which generates back flows. This term tends to zero further downstream of the body as the streamwise velocity u grows to become similar to U . The second term (b) represents the drag induced by longitudinal vortices and is usually called vortex-drag (Gilliéron & Kourta (2011)). This term can be particularly large in the case of an Ahmed body with a slant angle between 12 and 30 degrees. At these slant angles, two large longitudinal vortices appear at the top of the body back face. Finally (c) represents the part of the drag induced by the total pressure losses between the upstream and the vehicle wake. This last term dominates and is responsible for the majority of the drag for most road vehicles. Therefore, an effective way to reduce the aerodynamic drag is to increase the pressure in the vehicle wake or to reduce the wake cross-sectional area.

2.1.1 The relationship between mean drag and flow fluctuations

The drag reduction strategy used in this work is based on the premise that a reduction in the mean pressure drag can be achieved by manipulating temporal fluctuations of the flow. The link between wake fluctuations and mean aerodynamic drag has been observed in several experimental settings (Heenan & Morrison (1998); Pastoor *et al.* (2008)), and was exploited using feedback control by Dahan *et al.* (2012). A theoretical underpinning

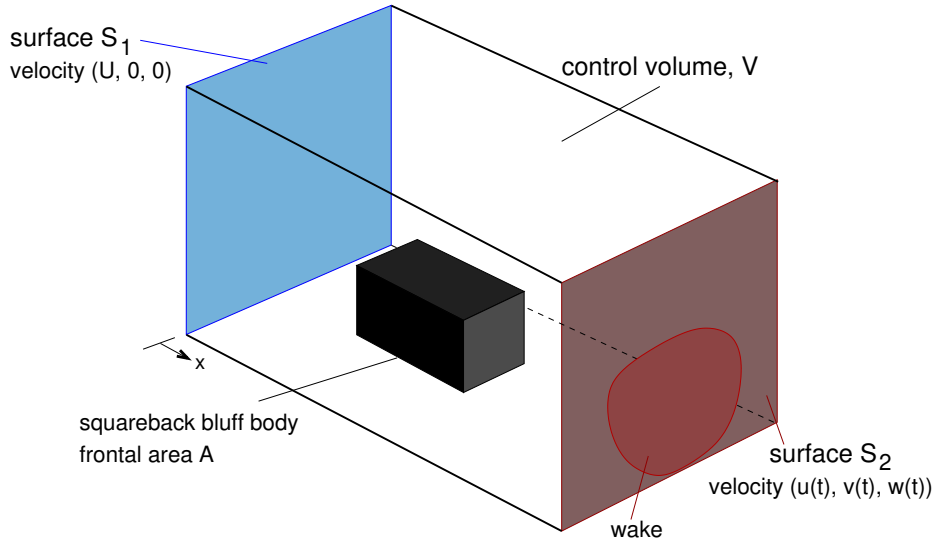


Figure 2.2: Control volume analysis of the flow around a bluff body. The side, top and bottom walls are assumed impermeable.

for the link, using an *unsteady* control volume analysis is here presented (Dalla Longa *et al.* (2017)).

A control volume containing incompressible flow around a bluff body is considered, as shown in Fig. 2.2. The flow is assumed longitudinal and undisturbed at the side, top and bottom boundaries, and the volume cross sectional area is constant, so that $\int_{S_1} dS = \int_{S_2} dS$. The flow entering is assumed steady, but is unsteady through the exit surface. The *unsteady* flow conservation equations, specifically the equations describing continuity of mass and x -momentum, for the control volume can be written,

$$\frac{\partial}{\partial t} \int_V \rho u(t) dV = -D(t) + p_1 S_1 - \int_{S_2} p_2(t) dS + \rho U^2 S_1 - \int_{S_2} \rho u(t)^2 dS + \int_{S_2} \tau_{xx} dS \quad (2.3)$$

$$U S_1 = \int_{S_2} u(t) dS \quad (2.4)$$

where $D(t)$ is the drag of the bluff body. The last term of Eq.(2.3) represents the viscous stress term acting in the x -direction on the exit surface. Although often neglected, it can be significant for near-wake flows (Van Dam (1999)).

The equations defining stagnation pressure are $p_{01} = p_1 + \frac{1}{2}\rho U^2$ and $p_{02}(t) = p_2(t) + \frac{1}{2}\rho(u(t)^2 + v(t)^2 + w(t)^2)$ (Greitzer *et al.* (2007)). For high Reynolds numbers flows, the viscous stress term can be assumed to be dominated by the Reynolds stress component, such that $\tau_{xx} = -\rho \overline{u'(t)^2}$ (Van Dam (1999)). It is then possible to express $D(t)$ as

$$D(t) = \int_{\text{wake}} \Delta p_0(t) dS + \frac{1}{2}\rho \int_{S_2} (U^2 - u(t)^2 + v(t)^2 + w(t)^2 - 2\overline{u'(t)^2}) dS + \rho \frac{\partial}{\partial t} \int_V u(t) dV \quad (2.5)$$

The term on the left-hand side represents overall aerodynamic drag. The first right-hand side term represents stagnation pressure loss in the wake; the exit and upstream stagnation pressures differ only over the wake component of the exit plane. The second term includes a longitudinal velocity deficit contribution, a vortex-induced drag contribution and a Reynolds stress contribution. The last term represents the rate of change of x -momentum in the control volume. The steady version of this equation is used widely in road vehicle aerodynamics as a tool for understanding drag (Hucho (1998); Van Dam (1999); Fourrié *et al.* (2011)).

Insights are drawn from the new, unsteady form of Eq. (2.5), when all variables are represented as the sum of a time-averaged $\overline{(\)}$ and fluctuating $(\)'$ component (Flinois (2015)). Because $D(t)$ depends linearly on stagnation pressure loss and x -momentum control volume unsteadiness, fluctuations in these will not alter the time-averaged $\overline{D}$. However, the term $(v'(t)^2 + w'(t)^2 - u'(t)^2 - 2\overline{u'(t)^2})$ comprises only velocity fluctuations, but due to the quadratic nature of its components has both a mean and fluctuating part. Reducing the size of this term reduces $\overline{D}$. That is, reducing velocity fluctuations normal to the streamwise direction, or increasing velocity fluctuations in the streamwise direction, results in a time-averaged drag reduction. This link seemingly provides an explanation for the impressive drag reductions achieved in high Reynolds number experiments using unsteady longitudinal slot jet actuation (Oxlade *et al.* (2015)). Of course, the link is a first-step approximation: altering the mean drag may in turn affect the mean and

fluctuating values of other terms, complicating the final relation. Thus it is probably best used as an insight or guidance tool.

2.1.2 Recent findings on wake dynamics

Recently, the relevance of symmetry breaking behaviour in the wakes of blunt bluff bodies to the aerodynamic drag has emerged. It is known that the laminar symmetric wake flow behind a flat disk becomes unstable above some Reynolds number, then undergoing a sequence of bifurcations which break the wake symmetry (Fabre *et al.* (2008)). This behaviour can be predicted via a weakly nonlinear analysis of the Navier–Stokes equations (Meliga *et al.* (2009)). Subsequent laminar flow experiments on an Ahmed body revealed very similar symmetry-breaking behaviour (figure 2.3, Grandemange *et al.* (2012)). Above a critical Reynolds number, the wake flow is no longer instantaneously symmetric, instead choosing one of two possible asymmetric states and remaining in that state – no switching between asymmetric states occurs and different experiments (initial conditions) can lead to a different choice of state.

A very recent finding is that this symmetry-breaking behaviour persists to high, turbulent Reynolds numbers (Grandemange *et al.* (2013b, 2014); Rigas *et al.* (2014)), but with the flow now exhibiting switching between different asymmetric states. This switching occurs over slow, random time scales, typically three orders of magnitude slower than the vortex shedding mode associated with separation at the blunt end of the body. This is known as bi-modality for squareback blunt bluff bodies and multi-modality for axisymmetric blunt bluff bodies (Grandemange *et al.* (2013b, 2014); Rigas *et al.* (2014); Varon *et al.* (2017)). The system can be described by a Langevin model with the stochastic forcing term associated with turbulent flow fluctuations – thus turbulent forcing appears to perturb the flow causing it to switch from one asymmetric state to another (Rigas *et al.* (2015); Brackston *et al.* (2016)). Grandemange *et al.* (2013a) showed that bi-modality can occur for blunt

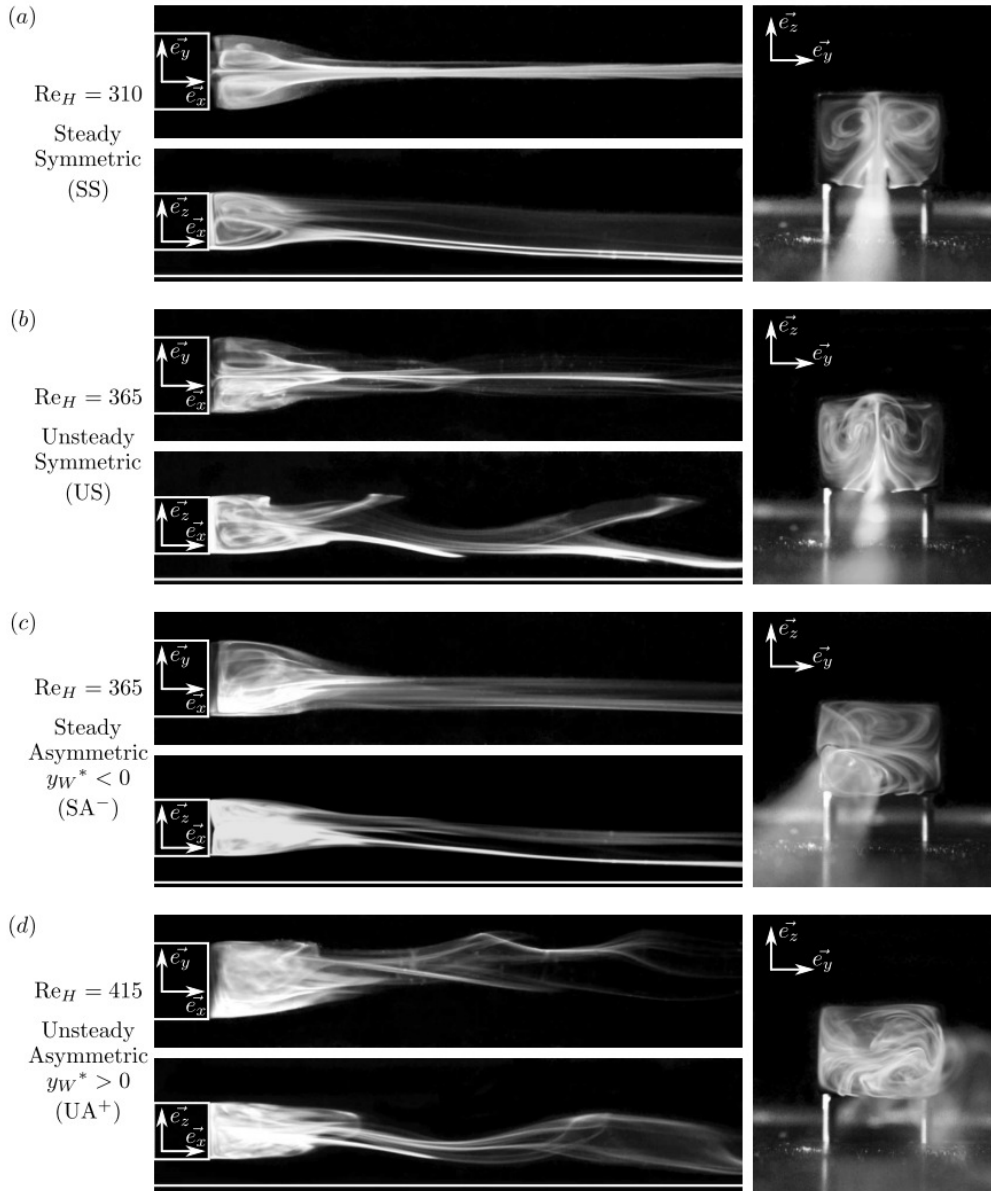


Figure 2.3: Symmetry-breaking behaviour of the wake behind an Ahmed body in the laminar regime at various Reynolds numbers (experiments from Grandemange *et al.* (2012)).

bluff bodies with a rectangular base across a wide range of aspect ratios. The existence of bi-modality and, if present, its axis (ie. whether it occurs with top–bottom asymmetry or side–side asymmetry) was found to be sensitive to the bluff body aspect ratio, ground proximity and yaw angle. Bi-modal switching has very recently been observed on real full-sized road vehicles (Bonnaïon *et al.* (2017)). Its presence affects the forces and moments on the body, including the aerodynamic drag.

Thus far, studies on bi-modality of blunt bluff body wakes have employed wind tunnel experiments. Simulations which accurately capture the wake unsteadiness at high Reynolds number, and which furthermore extend over the large time-scales of the bi-modal switching, are very costly. Unsteady simulations of Ahmed body flows have been performed using Partially-Averaged Navier–Stokes (Mirzaei *et al.* (2015)), Unsteady Reynolds-Averaged Navier–Stokes simulations (Khalighi *et al.* (2012)), Detached Eddy Simulations and Large Eddy Simulations (Krajnović & Davidson (2003); Serre *et al.* (2013); Aljure *et al.* (2014); Östh *et al.* (2014)), some also employing a Lattice–Boltzmann formulation (Roumeas *et al.* (2009); Lucas *et al.* (2017)). Recent Lattice–Boltzmann simulations by Lucas *et al.* (2017) successfully captured wake asymmetry, but not the presence of two asymmetric states nor any wake switching. To our knowledge, no simulations have yet captured a blunt bluff body wake exhibiting bi-modal switching.

2.2 Flow control techniques

As mentioned in the introduction, flow control in order to achieve drag reduction has been extensively studied and can be decomposed into two categories depending on whether control techniques require an input of energy or not (Figure 2.4). They are called respectively active and passive control techniques (Gad-el-Hak (2000)).

- Passive control techniques

In the context of bluff bodies, passive control techniques are characterized by shape modifications or addition of aerodynamic elements generally in order to achieve lower drag. Passive devices are usually designed to delay flow separation or to modify the pressure distribution on the body surface. In the case of squareback geometries, for which the separation point is fixed, passive devices generally act on the wake size or on the wake flow structures. For instance, Tanner (1972) performed experiments to investigate the effects

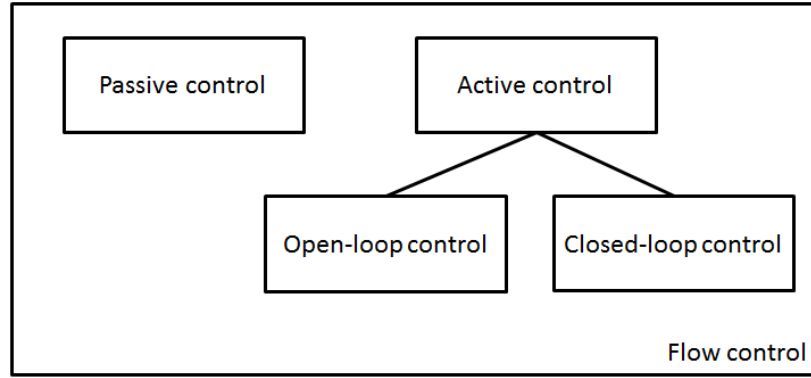


Figure 2.4: Flow control techniques

of a discontinuous trailing edge on the drag of bluff bodies of infinite span. He found that by segmenting the trailing edge of a blunt bluff body, drag reduction up to 64% could be obtained. Tombazis & Bearman (1997) carried out experiments on a three-dimensional (3D) D-shaped bluff body with a wavy trailing edge in the span-wise direction. The wavy surface proved to be able to affect the location of break-up of large roll-up structures in the wake and to increase the base pressure thanks to the three-dimensional structures added to the wake. Bearman & Owen (1998) performed comparable experiments with a rectangular cross-section cylinder with a wavy front face. They were able to completely suppress vortex shedding and to achieve drag reduction of up to 30 %. 3D forcing has been identified as a way to reduce the aerodynamic drag of bluff bodies (figure 2.5). Unfortunately these 3D forcing techniques are only viable for two-dimensional (2D) bluff bodies or quasi-2D bodies for which one dimension is much larger than the others (Choi *et al.* (2008)).

Another solution to reduce the aerodynamic drag of a bluff body is to use porous surfaces. Bruneau *et al.* (2008) performed numerical simulations to investigate the effect of the presence of porous surfaces on a 2D Ahmed body. They tried different configurations and demonstrated that applying a porous layer on the upper part of the nose and on the top surface of the Ahmed body was the most effective setting. The aerodynamic drag was reduced by 45% for this configuration. A porous layer decreases the vorticity in the near

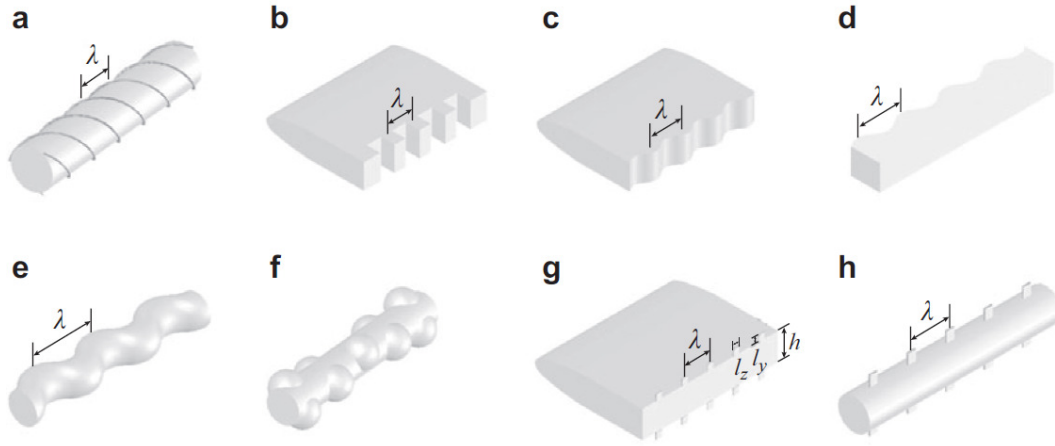


Figure 2.5: Passive 3D forcing techniques: (a) helical strake, (b) segmented trailing edge, (c) wavy trailing edge, (d) wavy front face, (e) sinusoidal axis, (f) hemispherical bump, (g,h) tabs (Choi *et al.* (2008))

wake and reduces also the cross-sectional area of the wake resulting in a decrease of the overall aerodynamic drag.

A different drag reduction method is to add physical extensions at the back of a bluff-body (figure 2.6). These extensions - forming a base cavity - extend the body surface and push the flow separation line further downstream from the base surface. These extensions lead to a mean base pressure recovery and therefore to a reduction of the aerodynamic drag (Khalighi *et al.* (2012); Evrard *et al.* (2016)). The base cavity can also be designed with a slant angle to diminish the cross-sectional area of the wake, decreasing the aerodynamic drag. In this case, it is called a “boat tail”. Numerous studies employed have boat tails with drag reductions of up to 31% achieved in the case of a square back Ahmed body (Verzicco *et al.* (2002), Balkanyi *et al.* (2002), Khalighi *et al.* (2012) Lee & Lee (2017)). Gilliéron & Kourta (2010) performed experiments using a vertical splitter plate placed either at the front or in the wake of an Ahmed body, obtaining a drag reduction of up to 12% when the device was placed in the vehicle wake. Finally, García de la Cruz *et al.* (2017) used a thin slit located on the perimeter of the base of an axi-symmetric bluff body. The slit was connected to an internal cavity allowing the air to flow in and out. The device resulted in an increase in total base pressure of up to 20% by connecting the

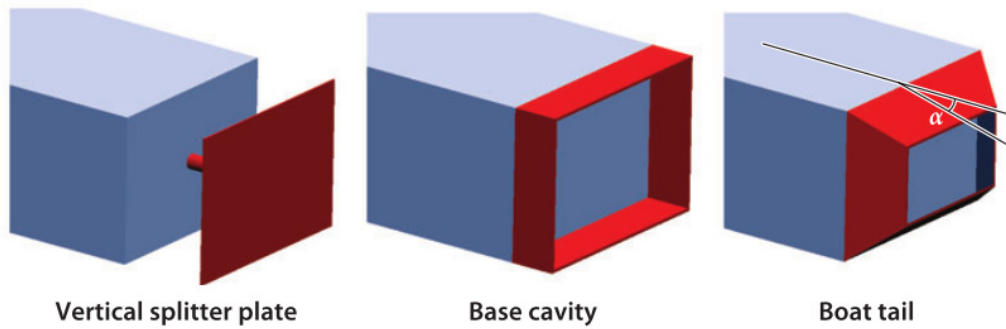


Figure 2.6: Passive control devices used for drag reduction. (Choi *et al.* (2014))

lower and higher pressure areas of the wake.

- Active control techniques

Active techniques require an input of energy to operate. Common active devices are deformable or moving surfaces such as flaps or also various type of jets (steady blowing/suction, pulsed jets, synthetic jets). Although active control systems are usually more expensive than passive devices because of their complexity, they present several advantages. First, they are usually tunable and able to function over a wider range of flow conditions than passive devices designed for one operating scenario. Moreover, in most cases, they do not require changes in the vehicle shape, making their use attractive for vehicle manufacturers seeking to achieve aerodynamic drag reduction without affecting the vehicle aesthetic properties. Active techniques can be subdivided (Figure 2.4): open loop forcing techniques, for which the actuation scenario is predetermined and no sensing is used, and closed loop active techniques, for which online measurements of flow physical properties are “fed-back” in real-time into a control loop in order to adapt the actuation scenario to transient or changing flow conditions. Open-loop control systems have been applied to bluff bodies in order to achieve aerodynamic drag reduction in a large number of cases. For instance, Choi *et al.* (2002) and Poncet (2004) investigated the effect of rotational oscillations on the drag of a cylinder. Arcas & Redekopp (2004) performed nu-

merical simulations on the effects of steady blowing/suction on the wake of a rectangular body. They were able to suppress the vortex shedding in the near wake. Roumeas *et al.* (2009) performed numerical simulations on the effects of continuous blowing on the back face of an Ahmed body and achieved aerodynamic drag reduction up to 29 %. Kourta & Leclerc (2013) used synthetic jets on an Ahmed body with a slant angle of 25 degrees. They achieved a drag reduction of 8.5% by re-attaching the flow on the slanted surface. Oxlade *et al.* (2015) performed high frequency forcing using synthetic jets located at the back of an axisymmetric bullet-shaped body, obtaining base pressure increase of 35%. Cabitza (2013) employed the same method on a simplified lorry, also increasing the base pressure, in this case by 27.3%. For both cases, the high-frequency forcing caused the break-down of the large vortex structures usually shed by the shear layers. To summarise, a large number of open-loop forcing studies were successful in reducing the drag of bluff bodies. However, in many cases, the energy consumed by the actuation exceeded the amount of energy saved by drag reduction (Littlewood & Passmore (2012)).

Finally, active feedback control has been applied for drag reduction in a number of studies at low Reynolds number, but few cases achieved drag reduction at higher turbulent Reynolds number, mainly because of the difficulties in modelling the turbulent flow due to its strong non-linear behavior and the infinite degrees of freedom of the flow solution. Active feedback control necessitates sensing of the flow and a choice of a control law. The actuation then reacts to sensing according to the control law chosen (Brunton & Noack (2015)). Control laws can be decomposed into two categories.

The first is a model-free approach, popular for complex flows as it does not necessitate any underpinning mathematical model for the flow. One commonly used technique is the slope-seeking or extremum seeking control. This consists of slowly tuning open-loop control forcing parameters (frequency, amplitude, etc.) while looking at the variation of the desired quantity to be reduced, such as the bluff body drag (Beaudoin *et al.* (2006); Pastoor *et al.* (2008); Gautier & Aider (2013)) until reaching a minimum. Recently, ma-

chine learning techniques have been used to optimise such control laws. They consist of creating a population of different control laws and to testing to evolve towards the best individual. Selection of the few best individuals, mixing between individuals or mutation of individuals are used at each iteration. These techniques are very expensive as they require a large number of experiments or simulations to converge towards an optimum solution - one experimental/simulation run represents one individual. For instance, Gautier *et al.* (2015) and Li *et al.* (2016) used a genetic algorithm to reduce the drag of respectively a back-facing-step and a square-back Ahmed Body. In both cases, although promising results were obtained, several hundred experiments were necessary until reaching the final control law.

A more common approach is to use a model-based control law representing the physics of the flow. Models can be categorised based on their level of complexity and the required level of knowledge of the flow. The terminology white-box, gray-box, black-box is often used to describe decreasingly the level of details captured by the model (Brunton & Noack (2015)).

For simpler flows at low-Reynolds number it is possible to linearise the Navier-Stokes equations to obtain the control law (white-box model). This approach has been employed to control boundary layer flows (Bagheri *et al.* (2009)) and cavity flows (Rowley *et al.* (2006)). However for more complex flows at higher Reynolds number, control laws are usually obtained from a simpler reduced order model (ROM) or gray-box model. The idea is to represent the flow with a finite number of modes containing most of the flow “physics” and from which the flow can be reconstructed. Various techniques can be found in the literature to obtain these modes, often based on Galerkin-derived models (Noack *et al.* (2003); Barbagallo *et al.* (2009)).

Finally “black-box” models are obtained by looking at the input-output response of the system to actuation and extracting a transfer function or state-space model of the re-

sponse. One way to extract the flow response to actuation, is to perform a flow impulse - via a Dirac forcing function - and to use a Eigensystem Realization Algorithm (ERA) (Juang & Pappa (1985)). ERA uses snapshots of the flow to extract a linear state-space model. Flinois & Morgans (2016) used this method to control the flow past a D-shaped body. Another approach is to use a set of single frequency inputs as forcing signal and to observe the response of the flow. Thus, a state-space-model of the flow response to actuation can be obtained. Note that this approach also allows the degree of non-linearity to be tested and, in the case of weak non-linearities, accounted for (Li & Morgans (2016)). This is the approach used in this work. It has been successfully applied to a range of two-dimensionnal and three-dimensional bluff bodies (Dahan *et al.* (2012); Flinois (2015); Dalla Longa *et al.* (2017); Evstafyeva *et al.* (2017)) and more details about this method are presented in chapter 4.

2.3 Synthetic jets

Synthetic jets, also called zero-net-mass-flux (ZNMF) jets are used in this work. These jets can be obtained experimentally using a loudspeaker placed in a small cavity. By vibrating, the loudspeaker membrane creates a periodic pressure drop which pulses alternately air in and out of the cavity through a slot (Glezer & Amitay (2002)). The advantage of this device is that it does not require any pumping system. Hence the low weight but also the small needs in energy to operate this type of jet, makes it an interesting solution in term of energy balance for aerodynamic drag reduction. Over one period of membrane oscillation, the net mass flow through the slot is null but the momentum injected to the outside flow is not.

Numerically, various geometries and boundary conditions can be used to represent synthetic jets (Leschziner & Lardeau (2011)). Due to the periodicity of the jets and their transient behavior, Reynolds Averaged Navier-Stokes (RANS) simulations are ill-suited

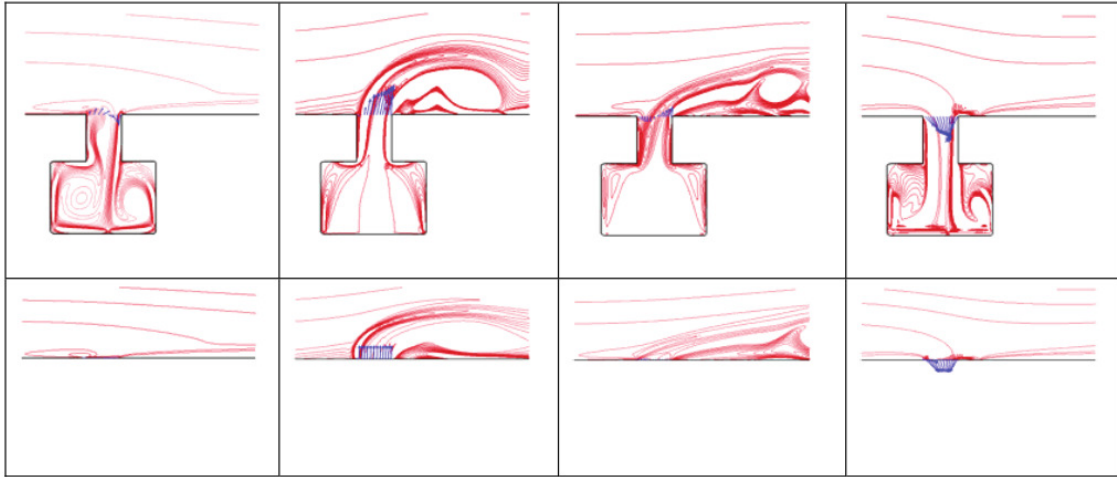


Figure 2.7: Velocity vectors (blue) and spanwise vorticity contours (red) during a synthetic jet actuation cycle. (top row) Full cavity simulation, (bottom row) velocity condition imposed on a plane (Aram *et al.* (2010))

to represent such actuators. Unsteady RANS simulations were shown to be sufficient to obtain a first-order representation of the jets (Dandois *et al.* (2006)), though Large-Eddy-Simulations (LES) appear to be a better option to capture the unsteady characteristics of the jet-flow. For LES, one of the challenge lies the range of scales required to model such jets. Indeed, the forcing region - in particular when the cavity is represented - is very small compared to the bulk of fluid affected by the jet. Therefore simulations require a very fine grid, able to resolve the high speed flow through the cavity and a large outer domain to capture the jets, which make the simulations expensive.

Leschziner & Lardeau (2011) and Aram *et al.* (2010) (figure 2.7) showed that it is possible to represent reasonably accurately the properties of slot jets using a velocity condition imposed on a single spanwise plane. Using this technique, meshing and computations are much easier.

In our simulations, synthetic jets are modelled by adding an artificial horizontal and vertical velocity condition at the desired locations. The cavity of the jets is not represented. More details and explanations about the various jet configurations are provided in chapter 4 and 5.

Chapter 3

Large Eddy Simulations

3.1 Introduction

Computational fluid dynamics (CFD) is extensively used in the current automotive industry to investigate the effect of design changes on the aerodynamic drag or on the cooling efficiency of a car. The interest for this tool comes from the ability to test a lot of designs without having to build several physical models. Moreover the data obtained provide rich information about the flow around a vehicle compared to wind-tunnel tests (flow properties being available everywhere in the flow). However, due to a the lack of computational power, direct numerical simulations (DNS) are not possible for industrial applications and models are still required. The lack of fidelity of the numerical models compared to reality in terms of vehicle geometry is also an important problem. Usually, vehicle geometry details are extremely difficult to represent accurately and simplified parts are used which add additional errors in the calculations. Examples are the gaps between parts, the engine compartment geometries, the tyre profiles and the interaction between the tires and the road. Therefore CFD is often not able to discriminate small changes in the car design or to correctly represent flow properties compared to wind-tunnel tests

for complex configurations. This is why today's use of CFD in the automotive industry is often only complementary to wind-tunnel experiments and provides additional data to support engineering decisions. A large number of codes used for industrial applications solve the Reynolds Averaged Navier-Stokes equations. This technique consists of decomposing the flow variables into a mean and a fluctuating part. The flow is then solved only for the mean velocity and pressure fields and a turbulence model is used for closure. Although this technique is relatively computationally inexpensive and the codes have been extensively developed and tuned to give good results compared to wind-tunnel experiments, for instance by using elaborate turbulence models, it does not represent correctly flow unsteadiness nor describe transient phenomena, which are particularly relevant in detached regions of the flow such as the vehicle wake. An intermediate solution between DNS and RANS is called Large-Eddy Simulation (LES). The idea is to separate the flow into large structures that are fully resolved and smaller structures that will be modelled using a sub-grid scale model. To achieve this, a filtering operation is necessary such that: $f(x, t) = \bar{f}(x, t) + f'(x, t)$ with f the flow function to be filtered, $\bar{f}$ being the part of f which will be computed and f' the filtered part which needs to be modelled by the sub-grid scale model. This filtering operation can be explicitly chosen by adding a spatial filter, but in our simulations, the numerical grid acts as an implicit filter. To summarise, LES is less expensive than DNS since only the large scales of the flow are resolved. The importance of the sub-grid scale model is very dependent on the grid coarseness. If the grid is coarse, the sub-grid scale model has to represent a larger range of flow scales and as a consequence, the accuracy of the result will be very dependent on the quality of the subgrid model employed.

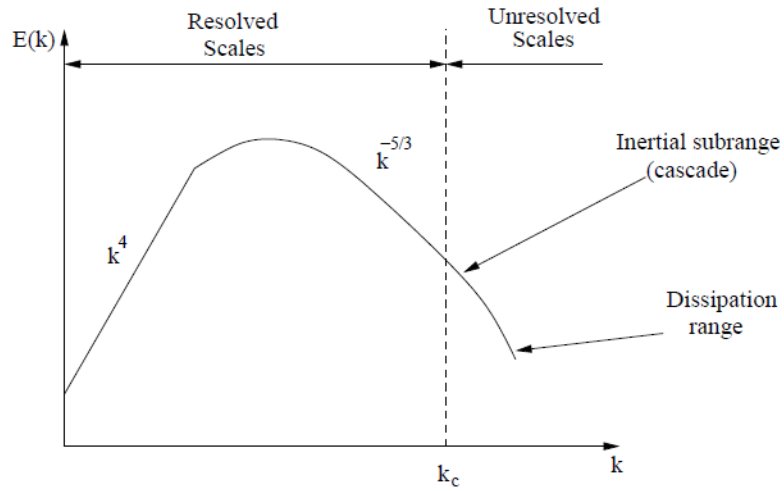


Figure 3.1: Scale separation in the energy spectrum. k is the wavenumber and k_c its the cut-off value. Temmerman *et al.* (2003)

3.2 *Stream-LES*

The present work uses an in-house flow solver called *Stream-LES*. This Large Eddy Simulation code was developed by Lardat & Leschziner (1998). Temmerman (2004) providing details of the solver and code validation. It has been successfully used to compute the flow on a flat plate (Lardeau *et al.* (2012)), around a compressor blade (Lardeau *et al.* (2012)), a turbulent channel flow (Touber & Leschziner (2012)) and the flow over two-dimensional and three-dimensional back facing steps (Dahan (2013); Flinois (2015)). Very recently, it was used to compute the flow around an Ahmed body at low-Reynolds number, successfully capturing the wake-bifurcation scenario (Evstafyeva *et al.* (2017)). It is a parallel code, which uses a non-orthogonal, block structured grid and is based on a finite-volume method. The flow domain is decomposed into small cells or control volumes, with each term of the Navier-Stokes equations integrated over each control volume. The incompressible Navier-Stokes equations to be solved can be written in their finite volume discretised formulation as follows:

$$\int_{\Omega} \frac{\partial \bar{u}_i}{\partial x_i} d\Omega = 0 \quad \text{continuity eq.} \quad (3.1)$$

$$\int_{\Omega} \frac{\partial \bar{u}_i}{\partial t} d\Omega = - \int_{\Omega} \frac{\partial \bar{u}_i \bar{u}_j}{\partial x_i} d\Omega - \int_{\Omega} \frac{\partial \bar{p}}{\partial x_i} d\Omega + \int_{\Omega} \left(\frac{2}{Re} \frac{\partial \bar{S}_{ij}}{\partial x_j} + \frac{\partial \bar{\tau}_{ij}}{\partial x_j} \right) d\Omega \quad \text{momentum eq.} \quad (3.2)$$

where $\bar{u}_i$ and $\bar{u}_j$ are the i^{th} and j^{th} components of the velocity, x_i and x_j are the i^{th} and j^{th} components of the spatial coordinates, $S_{ij} = \frac{1}{2}(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i})$ is the strain rate tensor, τ_{ij} is the subgrid-scale stress tensor and Re the Reynolds number. Ω represents the finite volume over which the flow is computed. The pressure $\bar{p}$ and velocity components of $\bar{u}$, averaged over a cell, are stored in the cell centroid node. This method, called colocated arrangement, is easier to implement and requires less memory resources than staggered discretisation, for which only the pressure is stored in the cell center whereas the velocity is stored at the boundaries (or faces) of the cells. An issue with the colocated formulation is that it leads to spurious oscillations due to the linear interpolation required to impose mass-conservation between cells. Here, a Rhie-Chow interpolation scheme, introducing some artificial dissipation, is implemented to correct these (Rhie & Chow (1983)).

Computations were performed using both Imperial College High Performance Computing (HPC) facilities and ARCHER, the UK HPC clusters.

3.2.1 Sub-grid scale model

A Wall-Adapted Local Eddy-viscosity (WALE) model is employed as sub-grid model in the simulations. The goal of this turbulent model is to represent the effects of turbulent structures not captured by the grid because of their smaller scale and which are responsible for the energy dissipation in the flow. This model introduces an eddy-viscosity, ν_t , in the traceless part of the subgrid-scale stress tensor, τ_{ij} , as follows:

$$\tau_{ij} - \frac{\delta_{ij}}{3}\tau_{kk} = -2\nu_t \overline{S_{ij}} \quad (3.3)$$

This model was first proposed by Ducros *et al.* (1998). Temmerman (2004) illustrated its improved performance over various other models in the case of turbulent channel flow simulations. It is an improved version of the Smagorinsky model (Smagorinsky (1963)), the eddy viscosity behaving as y^3 near the wall to reproduce the proper vanishing behaviour (Nicoud & Ducros (1999)). This behaviour is missing in the regular Smagorinsky model, with non-physical damping functions usually applied near the walls to compensate. The eddy-viscosity of the WALE model is then determined by:

$$\nu_t = C_w \overline{\Delta}^2 \frac{(\overline{S_{ij}^d S_{ij}^d})^{3/2}}{(\overline{S_{ij} S_{ij}})^{5/2} (\overline{S_{ij}^d S_{ij}^d})^{5/4}} \quad (3.4)$$

where Δ is the filter width defined implicitly by the grid size, S_{ij}^d is the traceless strain rate tensor and C_w is a constant parameter, which is set to 0.1, assuming that the cut-off wave number k_c lies within the inertial sub-range of the boundary layer (Temmerman (2004)). Due to the eddy-viscosity definition, the WALE model is sensitive to rotation and strain rate at the smallest resolved scales but vanishes for planar/shear flows and therefore avoids over-dissipation. Due to these characteristics, the model was shown to be able to capture turbulent transition (Nicoud & Ducros (1999); Mistry *et al.* (2015)).

3.2.2 Spatial discretisation

By replacing the subgrid-scale stress tensor in the momentum equation (3.2) by the expression for eddy-viscosity models (eq. 3.4) we can re-write the momentum equation as:

$$\int_{\Omega} \frac{\partial \bar{u}_i}{\partial t} d\Omega = - \int_{\Omega} \frac{\partial \bar{u}_i}{\partial x_j} \bar{u}_j d\Omega - \int_{\Omega} \frac{\partial \bar{p}}{\partial x_i} d\Omega + \int_{\Omega} \frac{\partial}{\partial x_j} \left(2 \left(\frac{1}{Re} + \nu_t \right) \bar{S}_{ij} - \frac{\delta_{ij}}{3} \tau_{kk} \right) d\Omega \quad (3.5)$$

By merging the deviatoric part of the subgrid-scale stress tensor with the pressure gradient, this can be written more compactly as:

$$\int_{\Omega} \frac{\partial \bar{u}_i}{\partial t} d\Omega = - \int_{\Omega} \frac{\partial \bar{u}_i}{\partial x_j} \bar{u}_j d\Omega - \int_{\Omega} \frac{\partial \bar{p}}{\partial x_i} d\Omega + 2 \int_{\Omega} \frac{\partial (\nu_{tot} \bar{S}_{ij})}{\partial x_j} d\Omega \quad (3.6)$$

where $\bar{p}$ is $\bar{p} + \delta_{ij} \tau_{kk}/3$ and ν_{tot} is $1/Re + \nu_t$.

The convective term of equation (3.2) is computed as a surface integral by using the Gauss-Ostrogradsky divergence theorem:

$$\int_{\Omega} \frac{\partial \bar{u}_i}{\partial x_j} \bar{u}_j d\Omega = \int_S (\bar{u}_i \bar{u}_j) n_j dS \quad (3.7)$$

The surface integral is then approximated using the second-order midpoint rule as:

$$\int_S (\bar{u}_i \bar{u}_j) n_j dS = \sum_{c=e,w,n,s,t,b} \bar{u}_i^c \bar{u}_j^c S_j = \sum_{c=e,w,n,s,t,b} \bar{u}_i^c C^c \quad (3.8)$$

where c represents the cell face of the control volume considered ($c = e$ - east, w - west, n - north, s - south, t - top, b - bottom) and n_j is the j^{th} component of the vector normal to the cell face. $C^c = \bar{u}_j^c S_j^c$ is the mass flux through the cell c .

The double-derivative diffusive term of the momentum equation (3.2) is computed initially using the same approach (i.e. Gauss-Ostrogradsky divergence theorem and second-order midpoint rule):

$$\int_{\Omega} \frac{\partial^2 \bar{u}_i}{\partial x_j^2} d\Omega = \int_S (\bar{u}_i \bar{x}_j) n_j dS = \sum_{c=e,w,n,s,t,b} \left(\frac{\partial \bar{u}_i}{\partial x_j} \right)^c S_{x_j} \quad (3.9)$$

Subsequently, the Gauss-Ostrogradsky divergence theorem is re-applied to the resulting gradient centered on the cell face c (staggered):

$$\left(\frac{\partial \bar{u}_i}{\partial x_j} \right)^c S_{x_j} = \frac{1}{\Omega_c} \int_{\Omega_c} \left(\frac{\partial \bar{u}_i}{\partial x_j} \right)^c d\Omega_c = \frac{1}{\Omega_c} \int_{S_c} \bar{u}_i^c S_{x_j}^c \quad (3.10)$$

where Ω_c is a staggered control volume. The geometric parameters at the cell center are computed by averaging the cell parameters, for instance $S_x^M = (S_x^e + S_x^w)/2$ where $S_{x_j}^c$ is the x component of the surface vector oriented in the direction $w - e$ and located at the cell center M.

3.2.3 Temporal discretisation

The time derivative term of equation (3.2) is computed numerically using a second order backward (implicit) Euler scheme:

$$\frac{\partial \bar{u}_i}{\partial t} = \frac{3\bar{u}_i^{n+1} - 4\bar{u}_i^n + \bar{u}_i^{n-1}}{2\Delta t} \quad (3.11)$$

Therefore we can re-write the equation (3.2) as:

$$\frac{3\bar{u}_i^{n+1} - 4\bar{u}_i^n + \bar{u}_i^{n-1}}{2\Delta t} = -\frac{\partial \bar{p}^{n+1}}{\partial x_i} + CD^{n+1} \quad (3.12)$$

where C is the convective term, D the diffusive term and Δt the time-step. C and D are discretised in time using a third-order Gear-like scheme (Fishpool & Leschziner (2009)).

Stream-LES employs a fractional step method (Chorin (1968)), which consists of two steps in order to march the equation in time. First the momentum equation is solved without considering the pressure term to obtain an intermediate velocity $\overline{u}_i^{int}$:

$$\frac{3\overline{u}_i^{int} - 4\overline{u}_i^n + \overline{u}_i^{n-1}}{2\Delta t} = CD^{n+1} \quad (3.13)$$

The second step consists of an iterative procedure to compute the pressure $\overline{p}^{n+1}$ at the next time step, $n + 1$, in order to satisfy the continuity equation. The problem results in solving the following Poisson equation for the pressure:

$$\int_{\Omega} \frac{\partial^2 \overline{p}^{n+1}}{\partial x_i^2} d\Omega = \int_{\Omega} \frac{\partial^2 \overline{u}_i^{int}}{\partial x \partial t} d\Omega \quad (3.14)$$

Once the pressure is computed, the true velocity at the next time step ($\overline{u}_i^{n+1}$) is updated from the intermediate velocity:

$$\int_{\Omega} \frac{\partial \overline{u}_i^{n+1}}{\partial t} d\Omega = \int_{\Omega} \frac{\partial \overline{u}_i^{int}}{\partial t} d\Omega + \int_{\Omega} \frac{\partial \overline{p}^{n+1}}{\partial x_i} d\Omega \quad (3.15)$$

The size of the time-step Δt is controlled by two parameters, ensuring the solver stability and convergence. The Courant-Friedrichs-Lewy (CFL) number imposes that $\Delta t \leq CFL_{max} \Delta l / u$ and ensures that the convection term is correctly resolved, while the viscous condition $\Delta t \leq D_{max} \Delta l / \nu$, ensures correct diffusion. Here Δl is a local cell size and u the local velocity. $CFL_{max} = 0.25$ and $D_{max} = 0.1$ are the upper bounds for the CFL and the diffusion parameters chosen in *Stream-LES*.

3.2.4 Pressure solver

The computation of the pressure Poisson equation is very expensive and represents up to 80% of the total computational cost (Temmerman (2004)).

The elliptic Poisson equation (3.14) can be written in matrix form as:

$$LP = S \tag{3.16}$$

where L is the matrix containing the discretisation scheme coefficients, P the pressure matrix containing the pressure values of the system at each node and S being the right-hand-side term of equation (3.14). Therefore the problem consists of inverting L to obtain P . However, this technique is too expensive and instead an iterative method is used. Sweeps are computed from an initial guess until reaching the desired level of residual error. *Stream-LES* uses by default a Successive Line Over-Relaxation (SLOR) technique which improves efficiency compared to the original Gauss-Seidel method it is based on. The SLOR method solves P simultaneously for multiple adjacent grid points. For the case of the simplified lorry (chapter 5), for which the grid is highly non-orthogonal near the nose, the regular Gauss-Seidel method was used for stability. To speed-up convergence and eliminate low-frequency residuals, *Stream-LES* uses a V-cycle multigrid algorithm. The solution is computed on successive coarser meshes using only a sub-multiple of the grid nodes and on which the sweeps are computed very quickly until the low-frequency residuals are suppressed. Up to 5 grid sub-levels were used in the current simulations. Finally the solution is transferred back to the original grid.

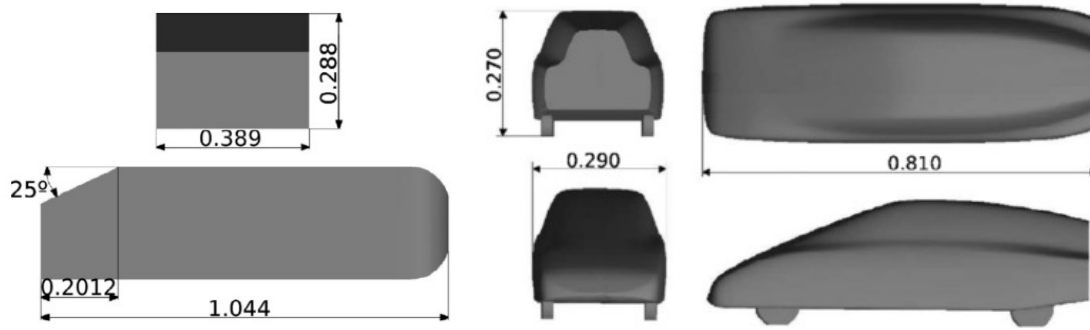


Figure 3.2: (left) Ahmed body with a slanted back face, (right) Asmo road vehicle. (Aljure *et al.* (2014))

3.3 LES for road vehicles, from simplified bluff bodies to real vehicles

Although hybrid methods such as Detached Eddy Simulations (where LES is only used in separated regions such as the bluff body wake to reduce the computational cost) are becoming popular, “pure” Large Eddy simulations are still computationally expensive and are rarely used for industrial applications, especially at high Reynolds number and on complex real vehicle geometries.

A large number of studies on simplified road vehicles using Large Eddy Simulations can be found in the literature. The most popular simplified road vehicle is the Ahmed body (Ahmed *et al.* (1984)). It exhibits massive flow separation at the rear and exists in two configurations: square-back or with a slanted back. Ahmed studied the effect of the slant rear angle on the aerodynamic drag (Ahmed *et al.* (1984)). It was found that for slant angles near 25-30 degrees, a large recirculation region appears on the slanted face as the flow separates. The difference of pressure between the bluff body sides and the slanted surface generates two large counter-rotating vortices responsible for a significant drag increase (figure 3.4).

LES was used in a number of studies to compute the flow in the wake of the square-back Ahmed body (Östh *et al.* (2014); Evstafyeva *et al.* (2017)).

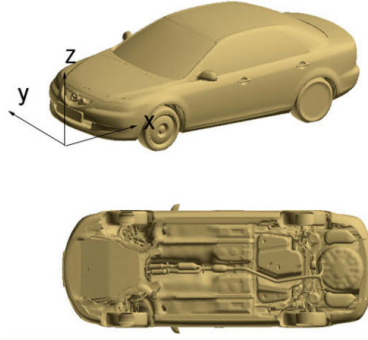


Figure 3.3: Full scale sedan geometry, Mazda Atenza. (Tsubokura *et al.* (2009))

The 25 degrees slant angle case is also particularly challenging for simulations, the flow separation and re-attachement on the slanted surface being difficult to obtain accurately. A number of Large Eddy Simulations were performed on this particular configuration. Krajnović & Davidson (2005*a,b*) computed the flow around an Ahmed body (Figure 3.2(a)) at a Reynolds number $Re = 200,000$. Minguez *et al.* (2008) and Serre *et al.* (2013) computed the flow around a similar configuration at $Re = 768,000$ using an high-order spectral method.

Another simplified road vehicle geometry that can be found in the literature is the Asmo car model (Figure 3.2(b)). This model is more complex than the Ahmed body and was designed by Daimler-Benz in the 90s to test CFD codes. It is a square-back model equipped with simplified wheels, boat tailing and a diffuser. The front end of vehicle is streamlined and separation occurs only at the square-back end. Experiments on it were performed by both Volvo and Daimler-Benz. Nakashima *et al.* (2008) carried out numerical simulations to compare RANS and LES drag predictions for an Asmo geometry, showing that their LES better predicted the flow structures and the drag coefficient than the RANS. Tsubokura *et al.* (2009) evaluated different turbulence models in their simulation of the Asmo car and compared their LES results with RANS. They obtained more accurate pressure distribution on the vehicle and flow structures in the wake using LES, successfully obtain-

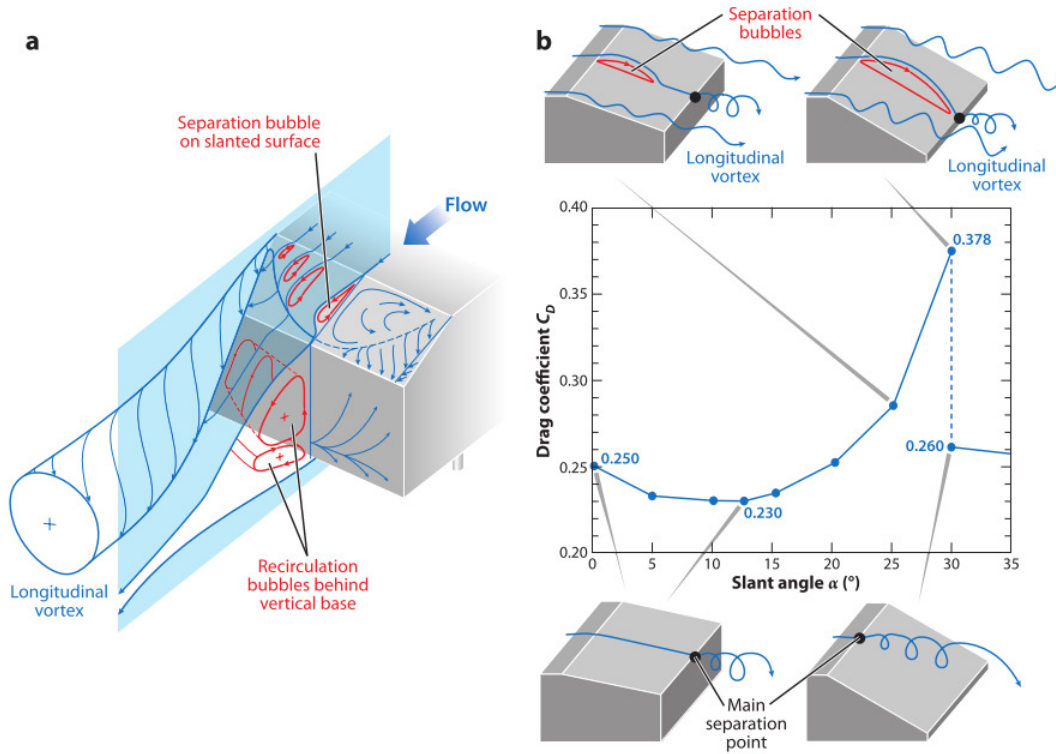


Figure 3.4: Aerodynamics of the Ahmed body (Choi *et al.* (2014)). (a) Time-averaged flow structures in the wake. (b) Drag coefficient (C_D) versus the rear slant angle (α).

ing the large coherent structures even with a coarser grid than that of Nakashima *et al.* (2008).

A few examples of the use of LES for real-vehicle geometries can be found in the literature. For instance, the flow past a full scale sedan car has been computed using LES by Tsubokura *et al.* (2009). They used a relatively coarse mesh ($y^+ \simeq 150$ wall-units, 38 million elements) and a wall model. Comparison of the velocity fields between the experiments and the simulations at different locations (underbody, wake) showed good correlation.

An issue with realistic car geometries is that most of them are not open-source and therefore rarely available for academia. In response to this, a set of realistic open-source road car geometries was proposed: the DrivAer car (Heft *et al.* (2012)). The geometries are based on the shapes of the Audi A4 and BMW 3 series and the three configurations



Figure 3.5: Iso-surface of Q-criterion, fastback DrivAer model. (Aljure *et al.* (2018))

represent fastback, notchback and estate vehicles. Aljure *et al.* (2018) computed the flow around the fastback version of the DrivAer car using wall-modelled LES. However they were not able to run long enough the simulations at high-Reynolds number to capture the lowest-frequency dynamics. As a result, the lift and drag coefficients exhibited large differences compared to the experimental values. Nevertheless, simulations successfully captured the vortex structures created by the mirror and the wheel housings.

Chapter 4

Simulation and Feedback control of the flow around a D-Shaped body

4.1 Introduction

In the present section, numerical simulations of the flow past a D-shaped blunt bluff body are used as a control test-bed. The aim is to use linear feedback control to achieve an increase in mean pressure on the back face or base of the geometry, and therefore to achieve drag reduction. This approach has previously been successful for backward facing step flows (Dahan *et al.* (2012)). The present control strategy is based on the underpinning link between mean drag and fluctuations presented in chapter 2, and extends application to bluff bodies exhibiting interacting shear layers. A notable feature of the present control strategy is that it should be implementable in real experiments outside of the wind tunnel, even though we are using computational flow simulations as a test-bed. The use of “adjoint” computational solvers is thus avoided, and *body-mounted* sensing and actuation are used. With the emphasis being on strategies which can be applied in real experiments, we design our controllers based on characterisation of the flow response

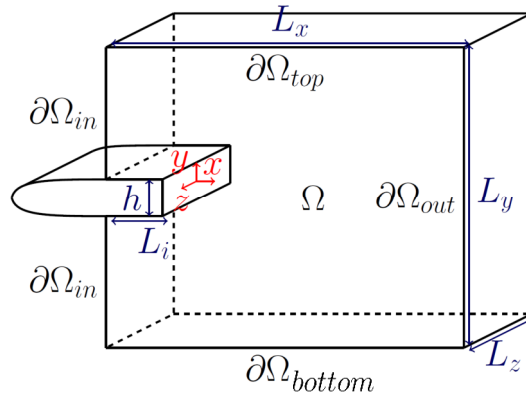


Figure 4.1: Computational domain, only the back of the bluff body is represented.

to actuation via system identification (“black-box” model), rather than any attempt to model the equations governing the system at hand, in this case the infinite dimensional Navier–Stokes equations.

4.2 D-shaped body configuration

The present work considers the simplified flow past a blunt bluff body of infinite spanwise extent, sometimes termed a D-shaped body. The Reynolds number based on the body height is $Re_h = 10\,000$. The boundary layer at separation is laminar with $Re_\theta = 270$; transition to turbulence occurs downstream of separation. Here Re_θ is the Reynolds number based on the momentum thickness θ .

The computational domain is shown in figure 4.1. It has dimensions $(L_i, L_x, L_y, L_z) = (4h, 24h, 9h, 4h)$. The inflow boundary condition, on both the upper and lower surfaces of the body, is a laminar Blasius boundary layer profile of thickness $\delta = 0.1h$, superimposed with random perturbations to encourage transition to turbulence. The top and bottom boundaries have free-slip conditions, with a periodic condition imposed in the spanwise direction. Consequently, the flow is statistically homogeneous in the spanwise direction

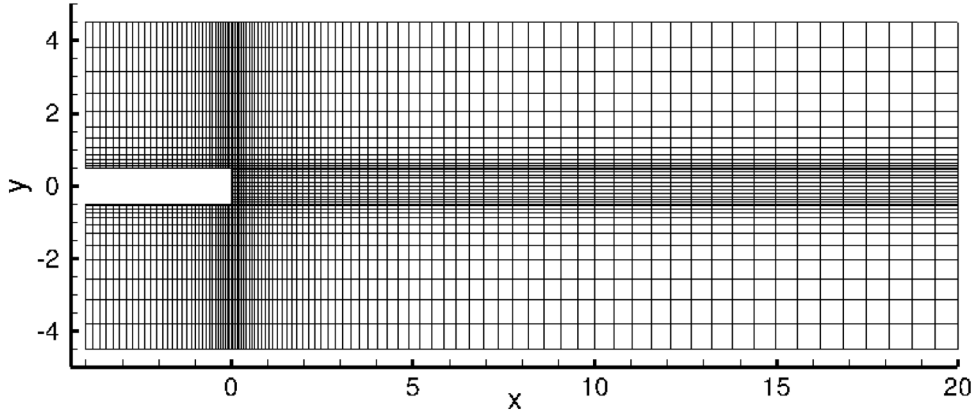


Figure 4.2: Side view of the baseline grid used in the simulations. Only 1/4 of the nodes are displayed for clarity.

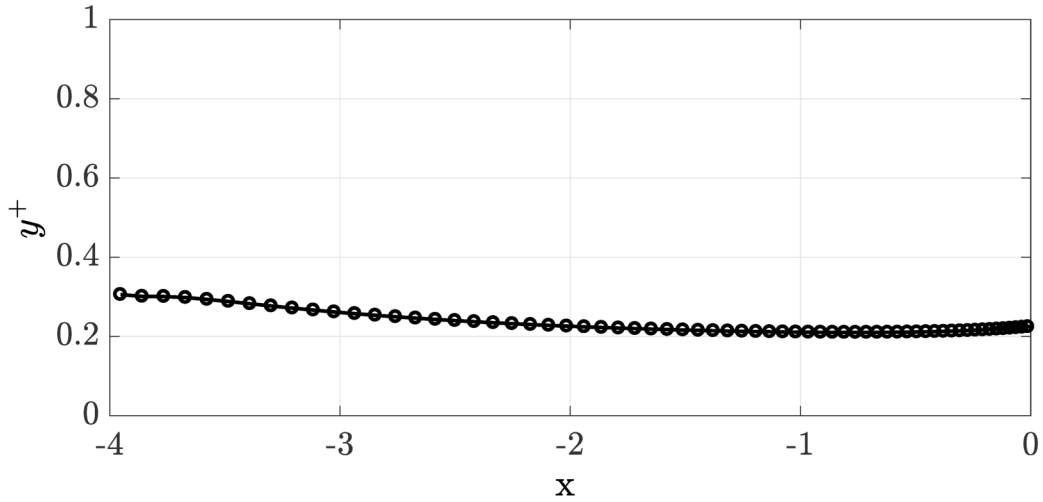


Figure 4.3: y^+ along the surface above the D-body.

and the time averaging is coupled with a spanwise averaging to accelerate statistical convergence. Grid checks were performed by increasing the resolution in each direction until consistency was achieved (Dahan (2013)). The final mesh comprises 9.4 million cells, with local refinement near walls and in the upper and lower actuation region (figure 4.2). The spanwise spacing was $\Delta z^+ = 12.4$ and $\Delta y_{max}^+ < 1$ along the body (figure 4.3).

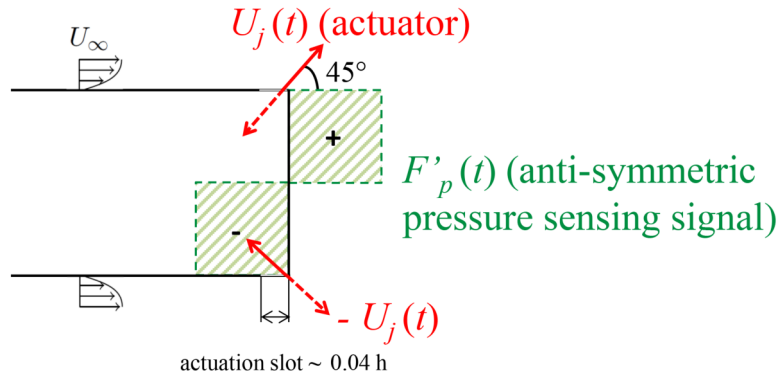


Figure 4.4: Side view of the setup showing actuation and sensing, flow left to right (the front part of the D-body is not shown).

4.3 Linear feedback control strategy for fluctuation attenuation

4.3.1 Sensing and Actuation

Our feedback control strategy exploits the link between mean drag and flow fluctuations presented in equation 2.5. This equation links variations in mean drag to a term containing the wake velocity fluctuations $(v'(t)^2 + w'(t)^2 - u'(t)^2 - 2\overline{u'(t)^2})$.

The issue of how to best capture this term using body mounted pressure sensing is an open question. Clearly, sensing of plane-integrated wake velocity fluctuations would require measurement at multiple locations in the flowfield and would not satisfy our requirement of body-mounted sensing. Reducing the above bracketed term reduces the mean drag, $\overline{D}$, and also tends to reduce drag fluctuations, $D'(t)$. Sensing of $D'(t)$ (equivalent to temporal fluctuations in the base pressure force for blunt bluff bodies exhibiting large separation at the back - Roshko (1993)) was the method used for BFS flows (Dahan *et al.* (2012)). For flows exhibiting strong wake asymmetry, this dynamic term is likely to be better captured by the asymmetric component of the base pressure distribution. Therefore the chosen

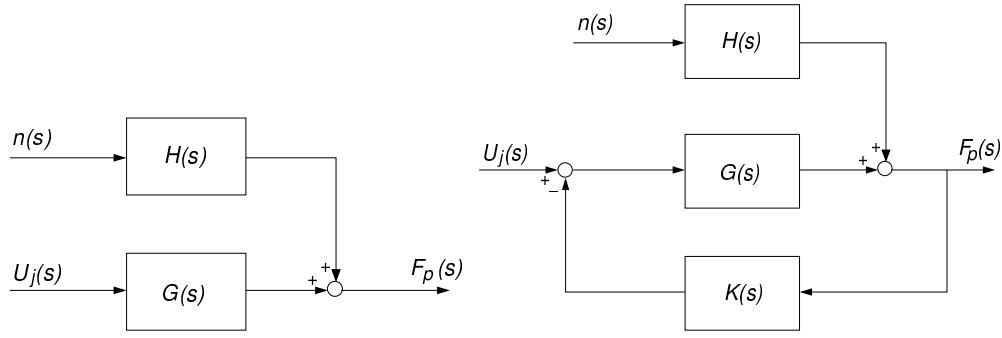


Figure 4.5: Frequency domain models for the open loop system (left) and the feedback control system (right). $s = i2\pi St_h$ is the Laplace transform variable, where St_h represents non-dimensional frequency.

sensor signal is the unsteady component of the spatially integrated base pressure, taking the anti-symmetric component in the y-direction, as shown in figure 4.4. It consists of taking the integrated value of the pressure on the base and counting as negative the values on the lower half. This accounts for the antisymmetric nature of the vortex shedding. Actuation is via slot-jet velocity forcing with zero mean, located just ahead of separation at an angle of 45° . This is out of phase on the upper and lower edges, and is constant in the span-wise direction.

4.3.2 Linear feedback control strategy

Given the above sensing and actuation pairing, a strategy for designing a linear feedback controller which ensures attenuation of the sensor signal must be chosen. Denoting the chosen sensor signal as $F_p'(t)$, the frequency domain models shown in figure 4.5 can be used. When actuating the flow with a varying slot jet velocity, $U_j'(t)$, $F_p'(t)$ can be assumed to vary both due to the actuation, $U_j'(t)$, and due to other disturbances in the flow (for example boundary layer and shear layer disturbances), lumped together as “noise”, $n(t)$. This captures the fact that even in the absence of actuation, the flow is fundamentally unsteady. Note that $s = i2\pi St_h$ is the Laplace transform variable, where St_h represents non-dimensional frequency.

In the absence of feedback control, shown on the left of figure 4.5, the response of the sensor signal, $F_p(s)$ is

$$F_p(s)_{\text{without control}} = H(s)n(s) + G(s)U_j(s) \quad (4.1)$$

By adding a feedback loop, in which the actuator responds to sensing of the base pressure force, $F_p'(t)$, via a control law, $K(s)$, shown on the right of figure 4.5, the response changes to

$$F_p(s)_{\text{with control}} = \frac{H(s)n(s) + G(s)U_j(s)}{1 + G(s)K(s)} \quad (4.2)$$

It can be seen that the ratio of sensor fluctuations with and without control is given by the magnitude of the sensitivity transfer function, $S(s) = 1/(1 + G(s)K(s))$ (Franklin *et al.* (1994)).

$$\frac{|F_p(s)|_{\text{with control}}}{|F_p(s)|_{\text{without control}}} = \frac{1}{|1 + G(s)K(s)|} = |S(s)| \quad (4.3)$$

Thus the aim of feedback control is to return a sensitivity magnitude smaller than unity, so that fluctuations in the chosen sensor signal are attenuated. It is known from Bode's integral law (Aström & Murray (2010)) that the sensitivity cannot be small over the entire frequency range, and so the dominant frequency range of the sensor fluctuations in the unforced flow is prioritised.

The control approach is then to identify $G(s)$, the transfer function describing the response of the sensor signal to actuation. Based on this, the feedback controller, $K(s)$, is designed to yield a sensitivity magnitude of less than unity, over the most dynamically relevant frequencies to the unforced flow. Note that although $H(s)$, the transfer function describing the response of the sensor signal to lumped "noise" is present in the model, its identification is never needed in order to achieve fluctuation attenuation.

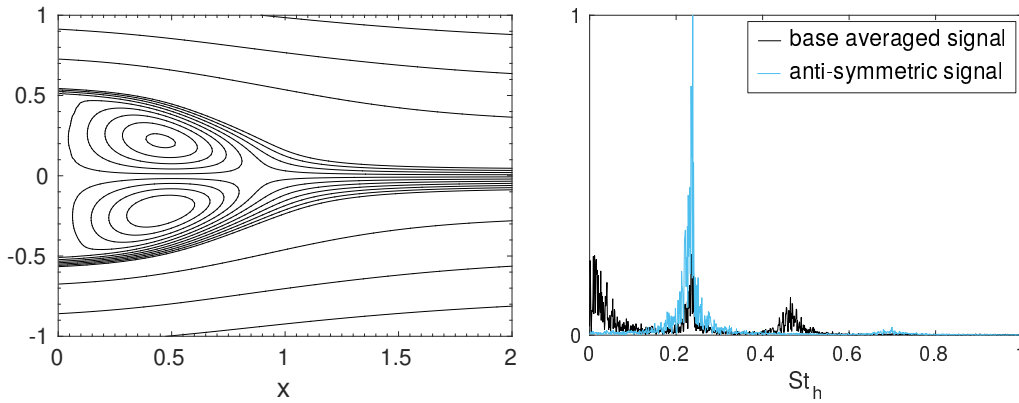


Figure 4.6: Time-averaged streamlines (left) and spectra of the normalised spatially-integrated base pressure (right) for the unforced flow.

4.4 Implementation and Results

4.4.1 Unforced flow field and integrated base pressure spectrum

The time-averaged streamlines and integrated base pressure spectra (both for the base pressure force and the antisymmetric component) for the unforced flow are shown in Figure 4.6. The base pressure spectra exhibit a large narrow peak at $St_h \approx 0.24$, corresponding to the vortex shedding mode. This frequency agrees well with the literature (Henning & King (2007); Pastoor *et al.* (2008)), and the first harmonic is also observed. The antisymmetric component of the base pressure is seen to exhibit the vortex shedding mode more strongly, confirming that as a sensor signal, it is capturing the wake fluctuations efficiently. This spectrum identifies the dominant frequencies of the flow over which feedback control needs to provide attenuation.

Figure 4.7 shows vorticity magnitude and instantaneous isosurfaces of pressure coefficient. The large-scale spanwise vortices, or rollers, that are shed downstream of the bluff body are evident, arranged in the well-known von Kármán vortex street pattern. These vortices are produced by the alternate roll-up of the upper and lower shear layers. Isosurfaces of negative pressure coefficient $C_P = -0.1$, extracted from the same snapshot, are shown in

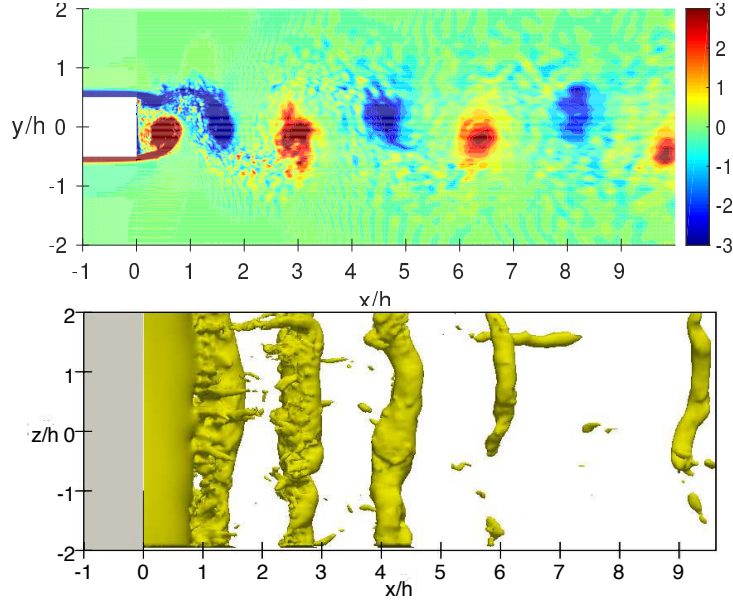


Figure 4.7: (top) Vorticity, ω_z , side view at $z/h=0$. (bottom) Instantaneous isosurfaces of pressure coefficient $C_P = -0.1$, top view at $y/h=0$.

figure 4.7. The large-scale vortical structures resulting from the roll-up are seen to have a low pressure core.

4.4.2 System identification

The aim of feedback control is to attenuate temporal fluctuations in the antisymmetrically integrated base pressure. The linear feedback controller, $K(s)$, will be designed so as to shape the gain of the sensitivity transfer function, $S(s) = 1/(1 + G(s)K(s))$, ensuring its magnitude is less than unity over the dynamically relevant frequencies. To accomplish this, it is necessary to characterise $G(s)$, the open loop response of the system to actuation.

The flow is governed by non-linear fluid mechanics. However, we make (and later check) the assumption of dynamic linearity – that the response of the change in the flow due to actuation is approximately linear. If the system is linear, when a purely harmonic actuation signal is applied, the sensor signal eventually settles into a response at the same frequency. The gain and phase shift of the sensor signal compared to the actuation are

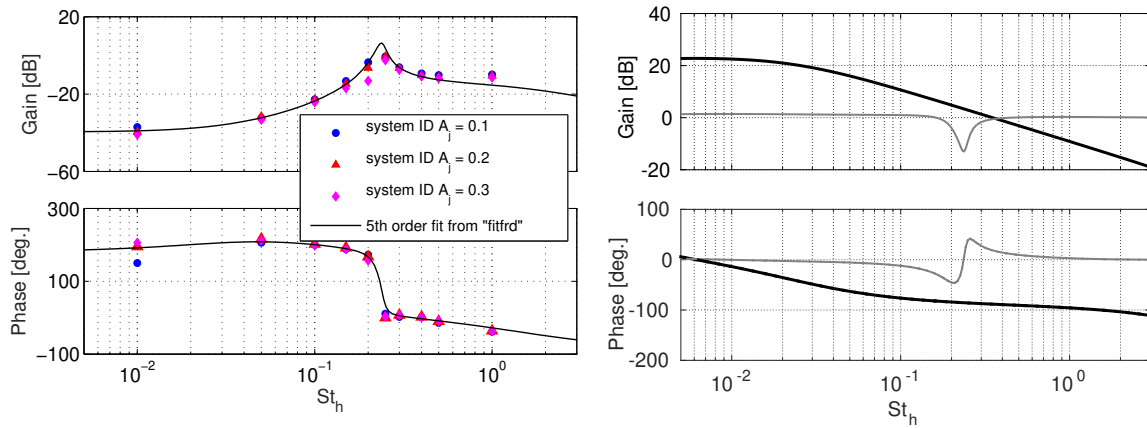


Figure 4.8: Left: System identification of $G(iSt_h)$. Right: Controller design: $K(iSt_h)$ (black) and $S(iSt_h)$ (gray).

then used to deduce the gain and phase of $G(iSt_h)$ (Ljung (1999)). Because the forcing frequency is clearly identifiable in the very clean output sensing signal, the integration of the product of the output signal and the relevant harmonic over at least 8 periods was sufficient to extract the gain and phase shift. No further signal processing tools were required. This is repeated across different frequencies, spanning the range for which the unforced flow exhibits dynamics in figure 4.6. The assumption of dynamic linearity is checked by ensuring that (i) the main flow response is at the forcing frequency with no significant scattering into harmonics and (ii) there is no significant dependence of the gain and phase shift on forcing amplitude.

In the simulations, actuation is performed with a simple “top hat” slot jet velocity profile (Leschziner & Lardeau (2011)) across the last five points of the grid upstream of separation, with slot width $s = 0.04h$, extending along the full span on both top and bottom as shown in figure 4.4 .

The forcing frequency is found to dominate the response across the range of forcing frequencies applied. The amplitude and phase variations of $G(iSt_h)$ as a function of non-dimensional frequency, St_h , are shown in figure 4.8, with the effect of forcing amplitude included. It can be seen that the dynamic response varies little with the forcing amplitude,

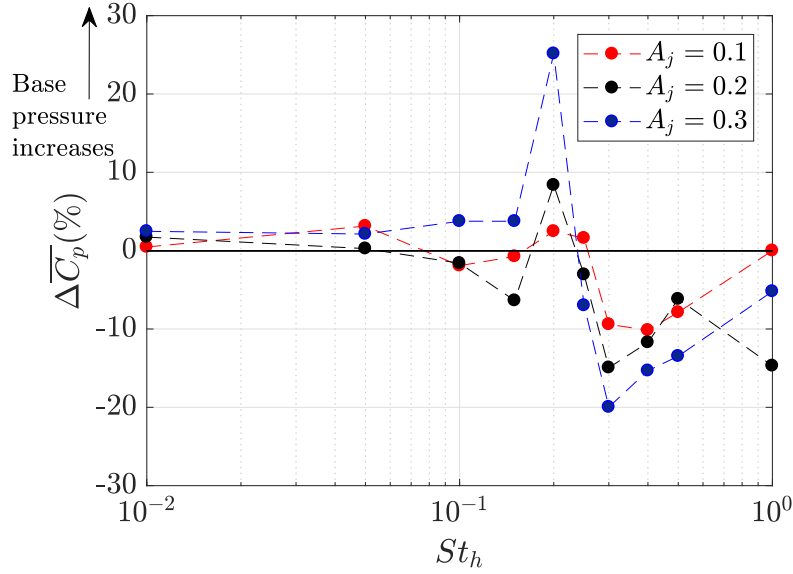


Figure 4.9: Effect of open-control control on the time-averaged base pressure.

A_j (the actuation signal being defined as $U_j(t) = A_j \cos(2\pi t)$), for the forcing range considered. Hence as long as control actuation remains within this range, the assumption of dynamic linearity appears to be justified. The Matlab command “fitfrd” is used to fit the frequency response data with a state-space model, using the “average” frequency response across the three forcing amplitudes in the fitting procedure; a fifth order fit is chosen, as shown in figure 4.8. The effect of the open-loop forcing on the mean base pressure is summarised in figure 4.9. The best case ($A_j = 0.3$, $St_h = 0.2$) led to a base pressure recovery of 25%. The resulting energy balance of the system is discussed in section 4.4.5.

4.4.3 Controller Design

Design of the linear feedback controller, $K(s)$, is performed manually in the frequency domain, with the objective of yielding a sensitivity magnitude less than unity over dominant frequency range of the spectrum in figure 4.6. Controller iteration to achieve a good compromise between the size and bandwidth of sensitivity, closed loop stability and

robustness to model uncertainty at high frequencies was performed (Zhou *et al.* (1996)). The unforced sensor spectrum consists of a main peak, and so classical loop-shaping via a fourth order controller is used to shape the gain of the sensitivity function (Zhou *et al.* (1996)). The resulting controller is $K(s) = 2.2 \cdot 10^4 s / (s^4 + 200s^3 + 1 \cdot 10^4 s^2 + 1600s + 15)$. Its frequency response, along with the resulting sensitivity transfer function, are shown on the right of figure 4.8. The sensitivity magnitude is less than unity (i.e. $0dB$) for $0.15 \lesssim St_h \lesssim 0.3$, hence the dominant frequencies in figure 4.6 should be attenuated by the action of feedback control.

4.4.4 Effect of feedback control on the base pressure

The controller was implemented in discrete-time format in the flow simulations, with the discrete time controller coefficients obtained via a bilinear transform. The results are shown in figure 4.10, where the spatially-integrated base pressure has been non-dimensionalised as a pressure coefficient, $C_p(t)$. The controller achieves its primary objective of reducing the antisymmetric component of the base pressure fluctuations over their most dynamic frequency range; the symmetric base pressure fluctuations are also attenuated. The ultimate and indirect aim of feedback control was to increase the base pressure. This was successful; the mean pressure force is increased by 38%.

4.4.5 Effect of feedback control on the flow-field

The changes to the flowfield accompanying effective feedback control are shown in figures 4.11 and 4.12. The feedback control acts to delay the roll-up of the shear layer vortices. Shed vortices are pushed further downstream, consequently, the recirculation region is extended in the x-direction. This is a similar observation to the 2-D back-facing-step work performed by Dahan *et al.* (2012), where the effect of control was also to push the roll-up of the shed vortices further downstream.

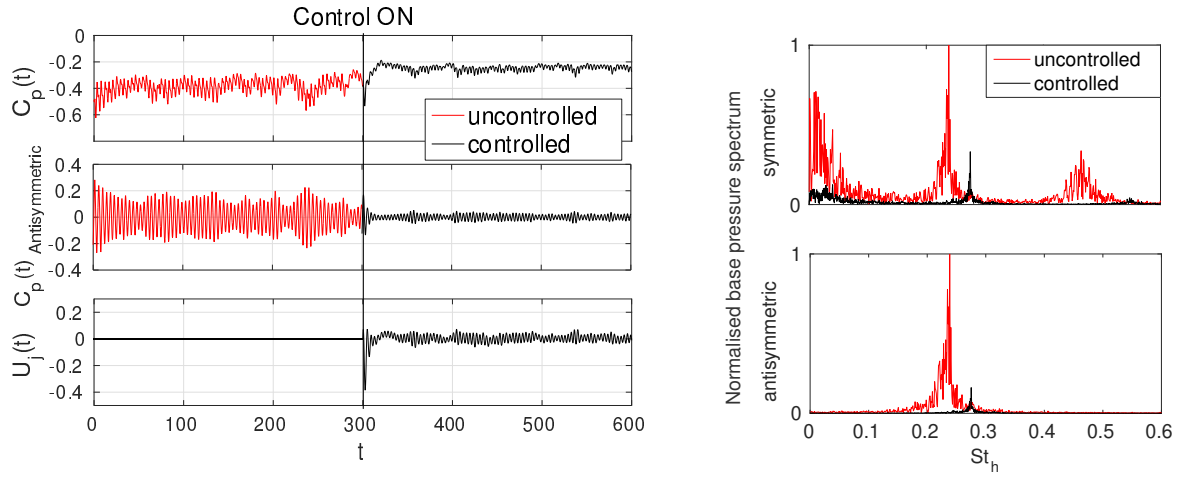


Figure 4.10: Effect of control, showing both the antisymmetric and symmetric components of the spatially-integrated base pressure. Time domain (left), and frequency domain (right). Control is switched on at $t=300$.

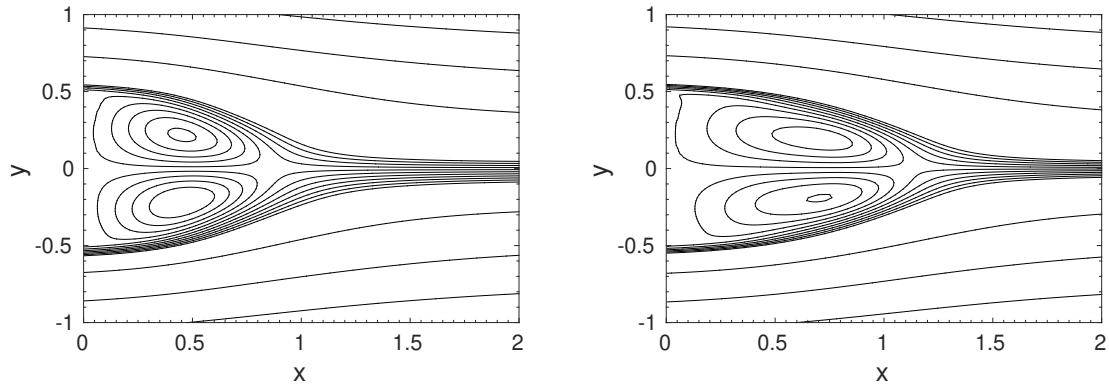


Figure 4.11: Contours of time-averaged streamfunction (left) without control and (right) with control.

To verify the unsteady wake analysis findings presented in section 2, the relevant squared velocity fluctuations integrated over a plane $x = 0.5h$ downstream of the body are shown in figure 4.13, both prior to and after feedback control is applied.

It is clear that although the control target is reducing spatially integrated base pressure fluctuations, it also indirectly reduces the wake flow velocity fluctuations term identified in section 2.1.1. This is consistent with the reduction in the pressure drag observed.

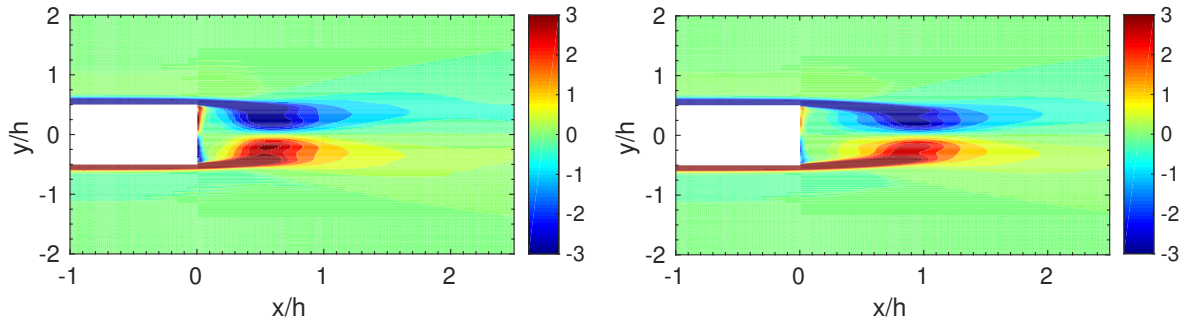


Figure 4.12: Contours of time-averaged vorticity ω_z , (left) without control and (right) with control.

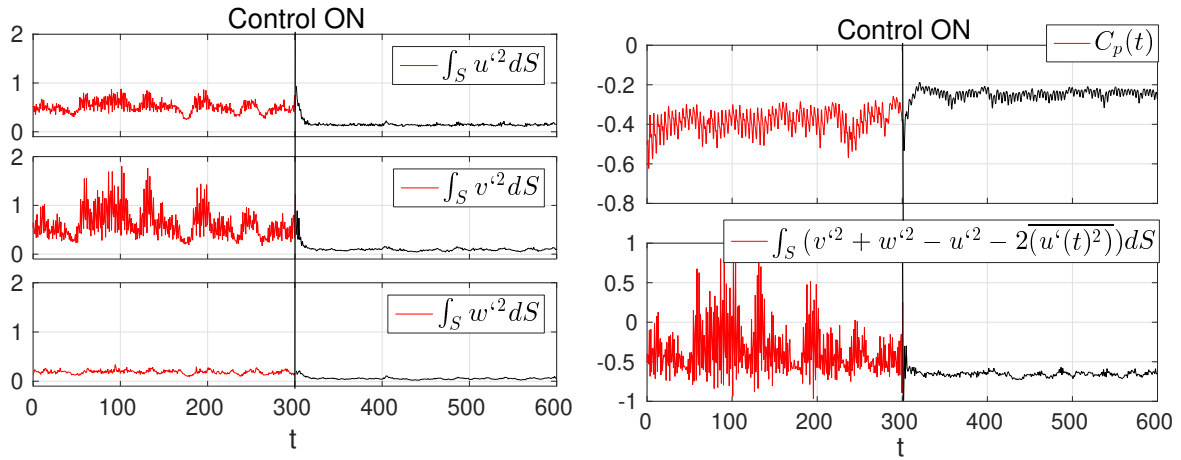


Figure 4.13: Effect of control, integrated values of squared velocity fluctuation components over a plane $x = 0.5h$ downstream of the body (left). Comparison between the sum of the plane integrated velocity fluctuations and the base pressure signal (right). Control is switched on at $t=300$.

Finally, as it is net energy gain that is important in drag reduction, the efficiency of the control method has been evaluated. The power injected into the jet fluid, P_{jet} , is compared to the power saving associated with the drag reduction, P_{saved} . The relevant expressions are

$$P_{jet} = \frac{1}{\eta_{jet}} \rho \frac{U_{jet}^3 S_{jet}}{2} \quad P_{saved} = \Delta F_x U \quad (4.4)$$

where η_{jet} is actuator efficiency, U_{jet} jet velocity, U free stream velocity, S_{jet} slot-jet area and ΔF_x the drag reduction. Initially, the actuator delivers a fairly large amount of momentum to obtain control of the flow, and the momentum coefficient peaks at

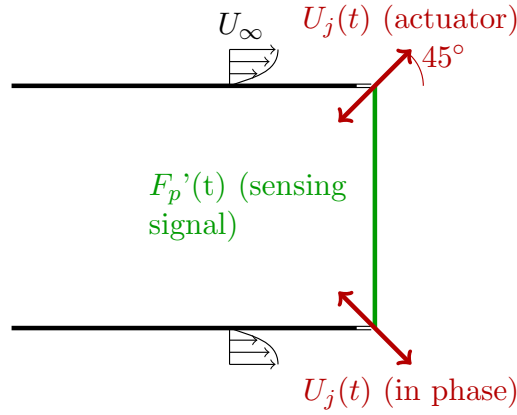


Figure 4.14: Side view of the setup showing in-phase actuation and sensing, flow left to right.

$\max(C_\mu) = s \max(U_{jet})^2 / (h U^2) = 1.18 \cdot 10^{-2}$ where s is the slot width and $\max(U_{jet})$ the maximum jet velocity. However, once the controlled state has been achieved, much less actuation is required to maintain the low drag state and $C_\mu = s U_{jet,rms}^2 / (h U^2) = 2.60 \cdot 10^{-4}$. Thus if we consider an ideal synthetic jet (i.e. $\eta_{jet} = 1$) we obtain $P_{saved} / P_{jet} = 8947$. This then provides a lower bound on the required actuator efficiency for net energy gain as being 0.01%, which is easily achievable by most of the synthetic jets devices available (Seifert (2007)).

4.4.6 Alternative actuation and feedback control strategy

Pastoor *et al.* (2008) showed that for a D-body, a base pressure increase can be achieved by synchronising the shedding of vortices coming from the upper and lower shear layers. The vertical symmetry then means that the shear layers are unable to pull each other and the Von Karman vortex street instability only appears further downstream of the body. The recirculation bubble is consequently extended and the base pressure increases.

In order to achieve this flow physics, our actuation set-up is modified so that the actuation ahead of the lower and upper shear layers is in-phase, rather than out-of-phase as used previously (figure 4.14). Concomittantly, we design a linear feedback controller aimed at

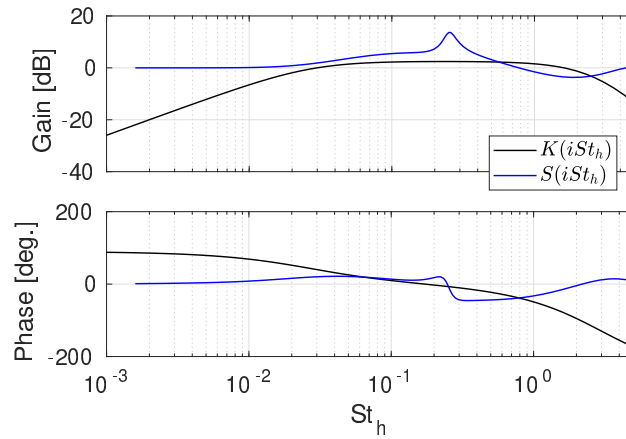


Figure 4.15: Controller design for in-phase forcing: $K(iSt_h)$ and sensitivity $S(iSt_h)$.

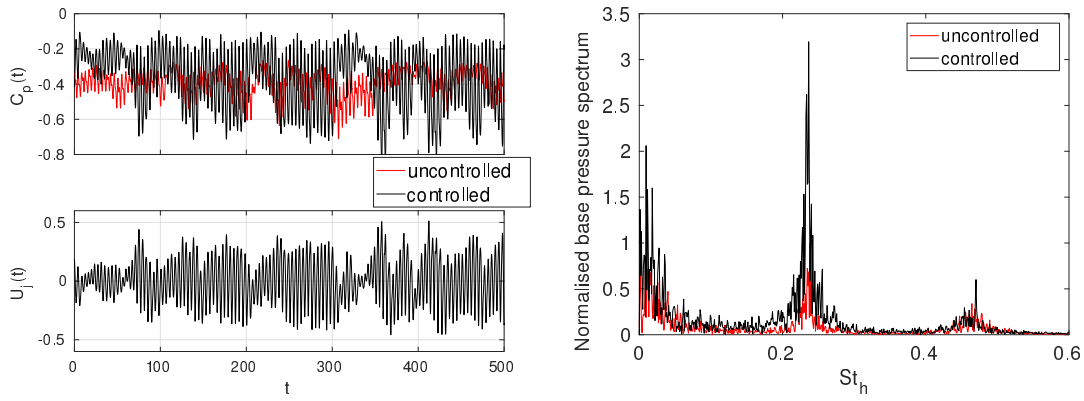


Figure 4.16: Effect of control, showing the spatially-integrated base pressure. Time domain (left), and frequency domain (right).

enhancing, rather than attenuating, the vortex shedding mode. This will involve insuring that the sensitivity $S(s)$ is above 1 at the vortex shedding frequency. The integrated base pressure was chosen for sensing. The “open-loop transfer function”, $G(s)$, for this new sensing actuation pairing must be identified. This follows the same approach as used previously, performing system identification using harmonic forcing across 10 different frequencies, $St_h = [0.01 - 1.00]$, with checks across different forcing amplitudes, $A_j = [0.1 - 0.3]$, for linearity. A controller for which the sensitivity is higher than one at the vortex shedding frequency is then designed in the frequency domain, using loop shaping principles ($St_h = 0.24$ - figure 4.15). The chosen controller is $K(s) = 5500/(s^2 + 30s + 275) \cdot 2s/(s^2 + 30s + 5)$.

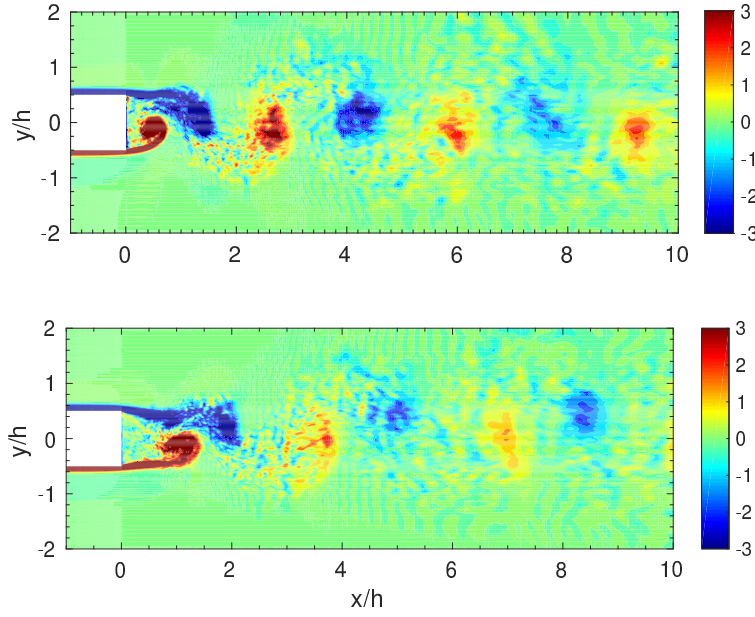


Figure 4.17: Contours of instantaneous vorticity ω_z , (top) without control and (bottom) with in-phase control.

The controller achieved its primary goal of increasing the base pressure fluctuations and amplifying the magnitude of the vortex shedding mode (figure 4.16). As a result, the shear layers became synchronised (figure 4.17). The delayed vortex roll-up resulted in an extended recirculation bubble and a significant increase in the mean base pressure (9%). We note, however, that this strategy is much less energy efficient than the anti-symmetric forcing presented before. It requires a larger amount of energy to enhance the vortex shedding mode - the momentum coefficient in this case being $C_{\mu, InPhase} = 2.7 \cdot 10^{-3}$ an order of magnitude larger than for the previous anti-phase forcing strategy. Furthermore, the base pressure recovery is not as large as for the anti-symmetric case, leading to $P_{saved}/P_{jet} = 11.1$ for this in-phase forcing strategy. Nonetheless, it is interesting to record that a second approach for linear feedback control can be successful.

Chapter 5

Simulation and Feedback control of the flow around a Simplified Lorry

The present chapter considers a more complex geometry which is closer to road vehicles. The body is taller than it is wide, with an aspect ratio corresponding to European lorries. Wind tunnel experiments by Cabitza (2013), performed on an almost identical geometry at higher Reynolds number, were used for flow correlation (figure 5.1). Additionally, the bi-modal switching behaviour (also called bi-stability) observed experimentally by Grandemange et al., was captured for the first time in simulations. Therefore an extended analysis of the unforced flow field is presented. The wake data is analysed, including using modal decomposition, and highlights the energy content and the dominating frequencies of the wake flow structures associated with bi-modal states. The switching sequence is captured via high-frequency snapshots and provide a plausible explanation of the physical phenomenon responsible for triggering bi-modal switches. Finally the feedback control approach is presented and several forcing configurations are assessed.



Figure 5.1: Simplified lorry model used for wind tunnel experiments (Cabitza (2013)).

5.1 Simulations set-up and validation

Simulations were performed on a body whose dimensions correspond to a 15 % scaled model of a European lorry (see figure 5.2). Its aspect ratio (height H over width W) is $H^* = H/W = 1.16$ and the gap between the body and the fixed road is $C^* = C/W = 0.68$. The nose is rounded to minimise front separation. These settings were chosen to match the experimental work of Cabitza (2013). The domain for the simulations was chosen according to ERCOFTAC recommendations for the Ahmed body, but scaled by the body height H instead of the body length L to keep the stream-wise domain size reasonable. As a result, dimensions were chosen such that $(L_{inlet}, L_{domain}, H_{domain}, Z_{domain}) = (7H, 31H, 5H, 5W)$ where L_{inlet} is the length in the x -direction of the upstream section before the body and L_{domain} , H_{domain} and Z_{domain} are respectively the length, height and width of the domain. A steady uniform velocity was set at the inflow, a no-slip condition was used on the floor and on the body surfaces. Free slip conditions were prescribed on lateral walls and on the roof and a convective outflow condition was set at the outlet. The flow is started from an initial inlet velocity condition and develops naturally without the addition of external perturbations. Note that all the simulations are wall-resolved.

The grid used in this work was obtained using Matlab and was an iteration of the code developed by Evstafyeva (2018). The choice of Matlab for mesh generation was due to the specific format of the mesh files required by the Stream-LES solver. The grid has

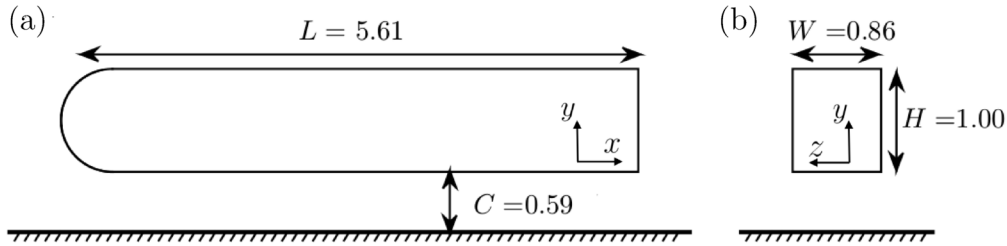


Figure 5.2: Schematic of the simplified lorry geometry. (a) Side view. (b) Rear view.

to be decomposed into binary files for each grid block and arranged in a manner not supported by commercial softwares. The grid obtained is a block-structured non-uniform Cartesian mesh. To obtain the curvature around the nose, two curved surfaces are added and projected onto the front plane of the bluff body. The edges of the rounded grid are then linearly interpolated to merge smoothly with the rest of the grid in the domain.

Grid checks were performed by increasing the number of cells in all directions until reaching convergence of the results. Eventually, a baseline mesh composed of 56.6 million cells was chosen for the simulations (figure 5.3). This baseline mesh is already refined compared with similar numerical studies performed at higher Reynolds number (Östh *et al.* (2014); Krajnović (2014); Lucas *et al.* (2017)), and is sufficiently refined near the body walls using an hyperbolic tangent distribution such that $y^+ \simeq 1$ (figure 5.4), the widely-held benchmark for which the turbulent boundary layer can be taken to be properly resolved. Additional finer meshes composed of up to 103.6 million cells were also used for comparison, these having the same first cell height (and hence y^+) as the baseline grid, but more closely spaced cells in the boundary layer region beyond this, around the bluff body and in the wake. Note that the maximum streamwise and spanwise cell size at separation for the baseline grid was $\max(x^+) = 6.50$, $\max(z^+) = 10.81$. The CFL number in the simulations is dynamic and bounded below 0.25. All grids were symmetric around the body to avoid introducing asymmetry in the flow.

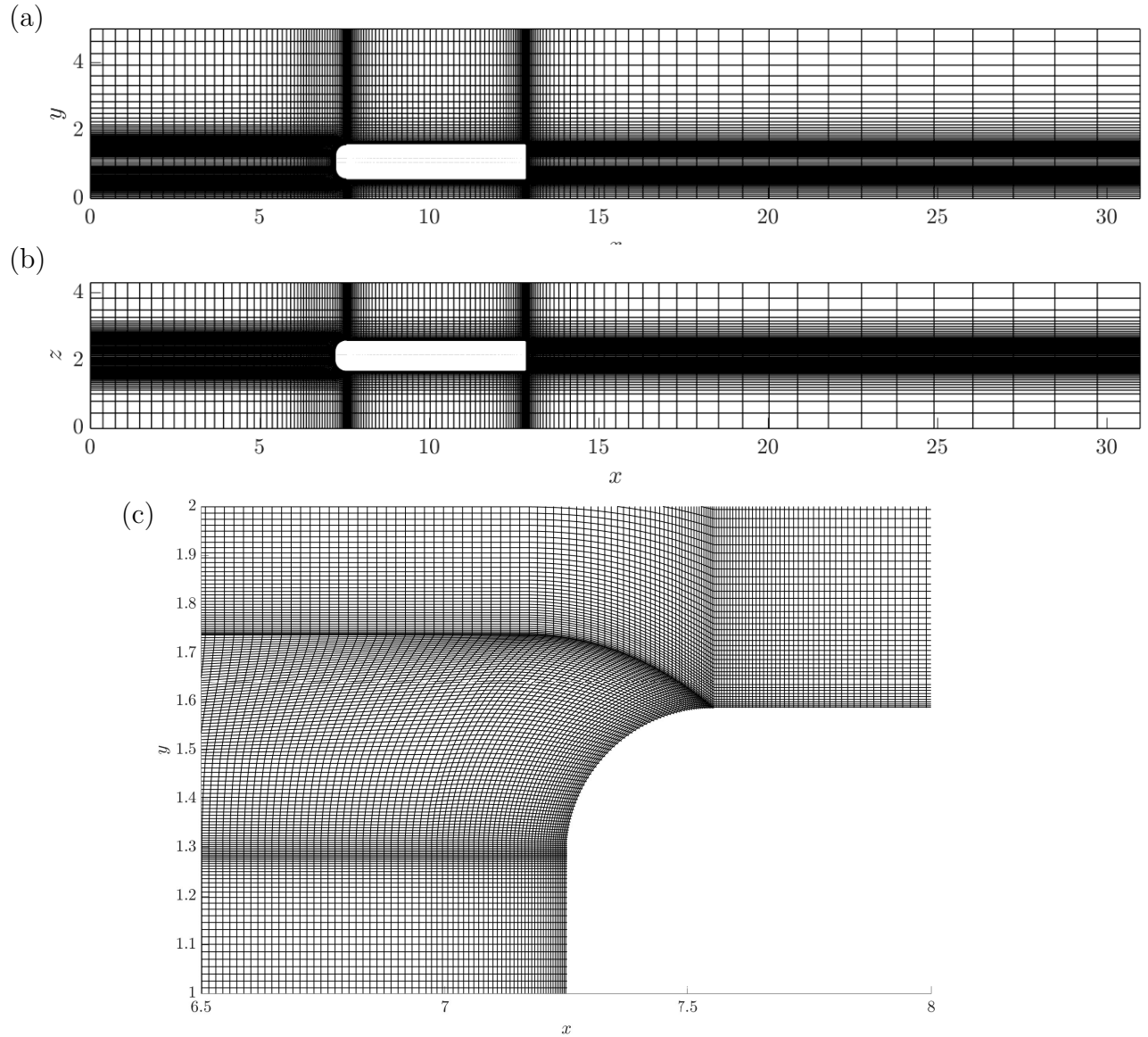


Figure 5.3: Baseline grid used in the simulation. (a) XY slice, side view. (b) XZ slice, top view. Only showing 1/4 of the points for clarity. (c) Grid refinement around the nose, all grid points are displayed.

The Reynolds number based on body height in the simulations was $Re_H = 20\,000$. This value was found to be sufficient to ensure a fully turbulent boundary layer at separation. The boundary layer profile and turbulent fluctuations within the boundary layer were measured above the body and compared qualitatively with values measured in a turbulent channel flow (figure 5.5). The boundary layer profile matches the channel flow data in the inner region of the boundary layer. However the outer part is different and is influenced by

Case	Mesh Size	Grid points on the bluff body	$\overline{C_{p_b}}$	$St_{H-b.p.}$
Coarse	25.3m	101 376	-0.15	0.07
Baseline	56.6m	245 760	-0.13	0.08
Fine	82.6m	245 760	-0.12	0.08
Very Fine	103.6m	270 336	-0.12	0.08
Exp.	<i>n.a.</i>	<i>n.a.</i>	-0.13	0.08

Table 5.1: Summary of the grid refinement study. The total mesh size and the number of grid points adjacent to the body are presented. The time-averaged base pressure $\overline{C_{p_b}}$ and the bubble pumping frequency $St_{H-b.p.}$ are compared. Experimental values from Cabitza (2013) are also included.

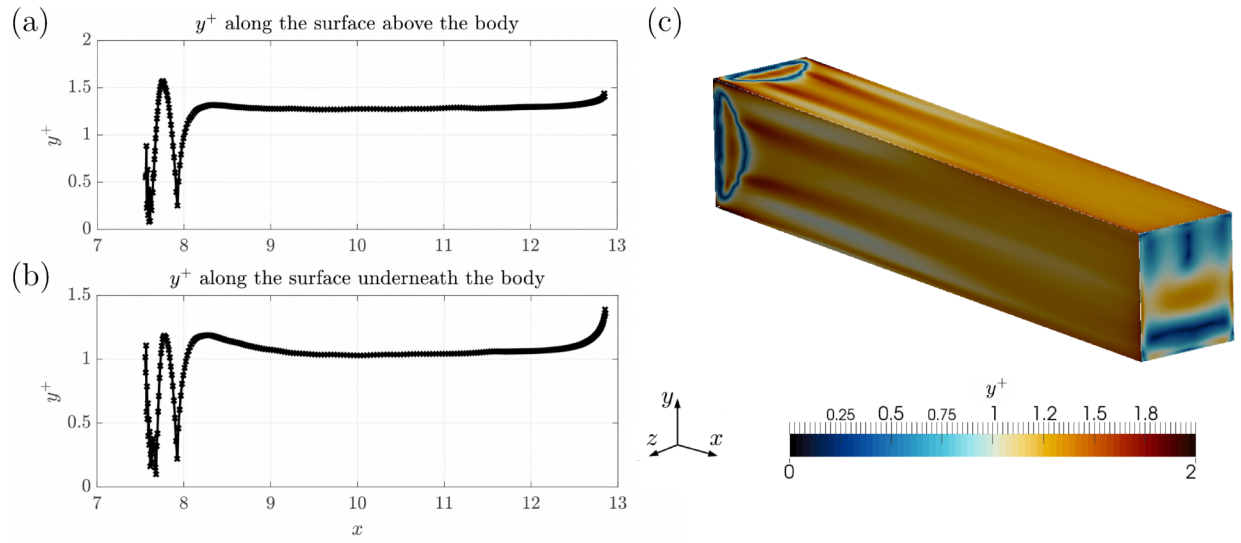


Figure 5.4: y^+ measured along the centerline of the surfaces (a) above the body, (b) underneath the body. (c) y^+ contours projected on the bluff body surfaces.

the geometry of the simplified lorry. A small recirculation bubble near the nose sheds low pressure structures downstream which probably affects the boundary layer. Additionally the presence of the lorry wake squeezes the boundary layer and its height is reduced near separation. Stronger velocity fluctuations level were measured in the case of the lorry indicating that the boundary layer is indeed fully turbulent. The Reynolds number based on the momentum thickness θ is $Re_\theta \simeq 2160$ near separation.

In order to capture any large time-scale bi-modal behaviour, simulations were run on Imperial College and ARCHER (the UK National Supercomputing service) clusters for up to 8-10 days on several thousand cores (from 1728 cores for the baseline grid to 3168

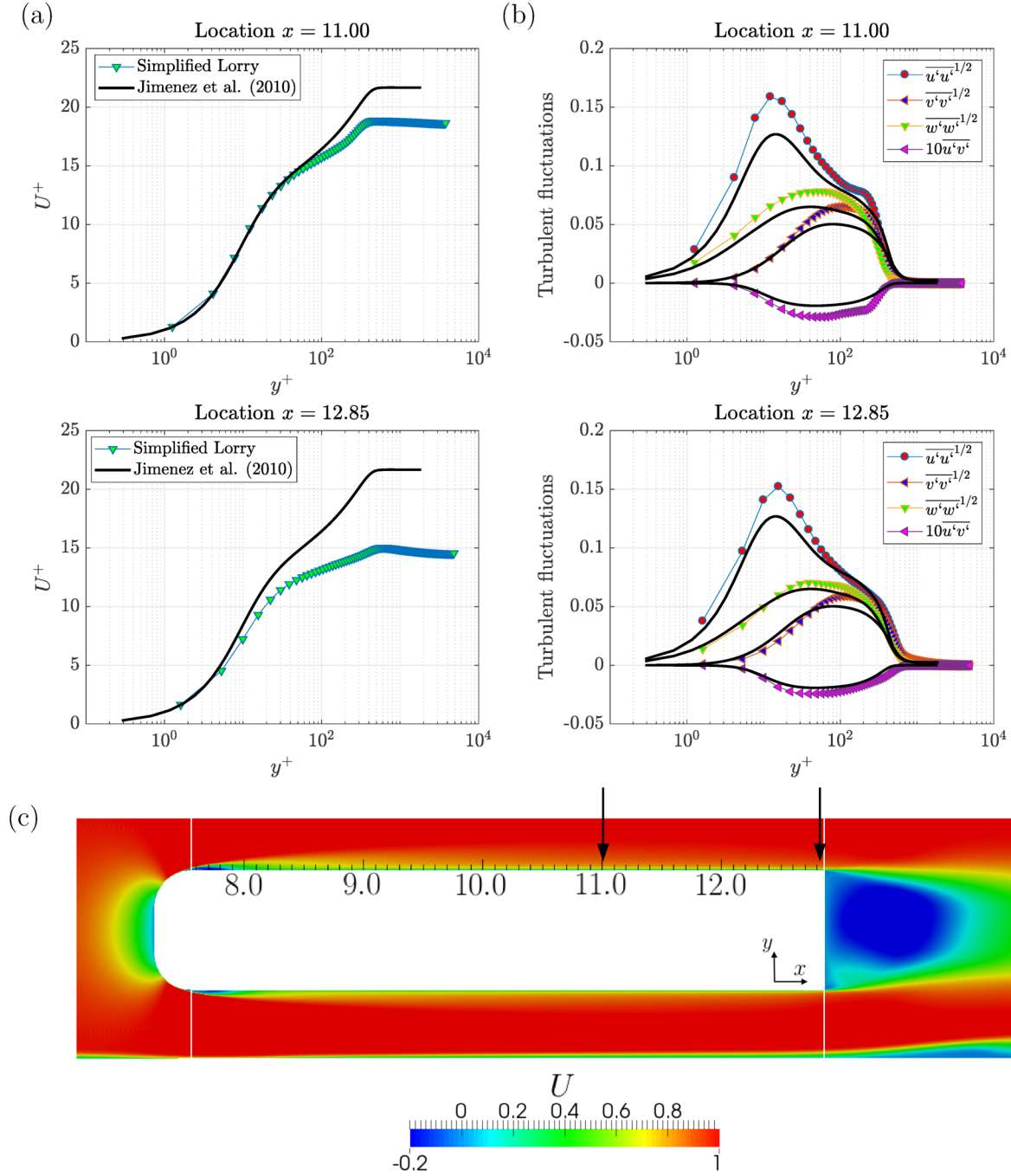


Figure 5.5: Time-averaged boundary layer characteristics measured on the centerline of the surface above the body. Current LES data at various x locations of the simplified lorry are compared with the DNS results of a turbulent channel flow at $Re_\theta = 1551$ (Jiménez *et al.* (2010)). (a) Streamwise velocity profile, (b) velocity fluctuations. (c) Time-averaged velocity field around the simplified lorry. The black arrows indicate the locations of the boundary layer measurements. In the current simulations $Re_\theta \simeq 2160$ near the separation point.

cores for the very fine grid).

Numerical simulations were compared to experimental results from Cabitza (2013), these having been performed at a higher Reynolds number of $Re_H = 4.1 \cdot 10^5$. Results from different grid refinements are summarised in table 5.1 and show the correlation of the numerical results and the experimental data for both integrated base pressure and dominant wake frequency. The base pressure coefficient is here defined as $C_{p_b} = \int ((p - p_0)/(\frac{1}{2}\rho U_0^2)) dS$ integrated over the base surface $S = HW$.

5.2 Results

5.2.1 Time-averaged flow

The time-averaged flow downstream of the bluff body is shown in figure 5.6. The flow separates over the blunt end of the body and forms a large recirculation area, with the low pressure bubble extending up to $x = 1.17H$ downstream. The time-averaged wake is symmetric in the horizontal direction but slightly asymmetric in the vertical direction due to the presence of the floor. A small recirculation region appears on the fixed floor due to the upwash flow. Figure 5.7 shows the energy spectra of the base pressure force obtained using the “pwelch” function of Matlab. Two main modes are captured. The main peak at a frequency $St_H \sim 0.08$ corresponds to the bubble pumping mode, caused by the wake oscillation. The second peak at $St_H \sim 0.18$ corresponds to the vortex shedding mode generated by the detached shear layers at the rear end of the body.

5.2.2 Bi-modality

As described in section 2.1.2, a recent finding is that turbulent wake flows can exhibit bi-modality. The present study achieves what we believe are the first simulations of wake

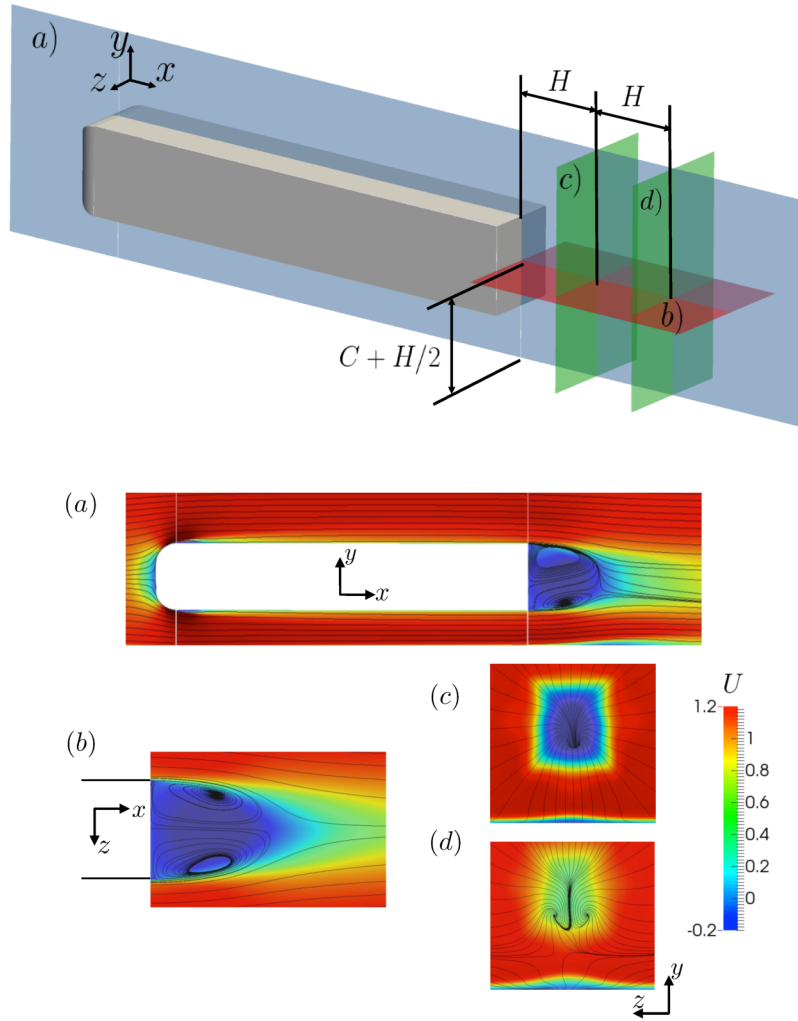


Figure 5.6: Time-averaged streamwise velocity field U and projected streamlines. (a) Side view in the symmetry plane $z = 0$, (b) top view in the symmetry plane $y = C + H/2$, (c-d) downstream planes at $x = H$ and $x = 2H$. The locations of the planes are shown in the upper schematic. Results are obtained using the baseline grid.

bi-modality, capturing both asymmetric states, for a blunt bluff body. No artificial forcing nor inlet perturbations are applied to trigger bi-modality; the flow develops naturally from a steady inlet velocity condition, with bi-modal switches occurring randomly

In order to assess the asymmetry of the body wake and ascertain whether the wake exhibits bi-modality, spatial gradients in base pressure were analysed in both the vertical ($\partial C_p / \partial y$) and horizontal ($\partial C_p / \partial z$) directions. Grandemange *et al.* (2013a) mapped the presence of vertical or horizontal bi-modality as a function of bluff body aspect ratio and

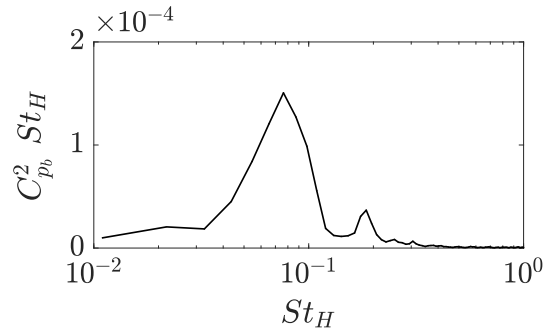


Figure 5.7: Energy spectrum of the pressure signal integrated over the body base. Filtering is applied for clarity.

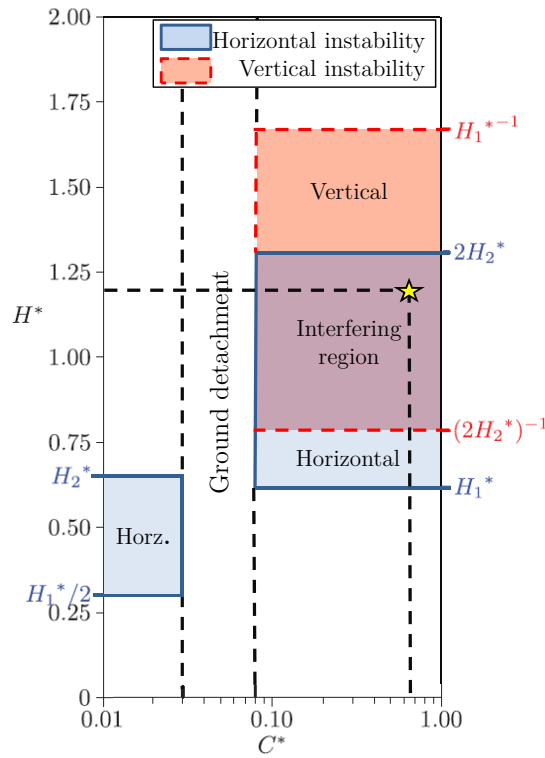


Figure 5.8: Regions of bi-modality from Grandemange *et al.* (2013a), the limits H_1^* and H_2^* were obtained in the experiments. The yellow star indicates the location of the current geometry.

underbody gap size. The current geometry is shown in the context of their mapping in figure 5.8, and is seen to lie within the boundaries of the interfering region. This means that both vertical or horizontal bi-modality can occur. Experimentally it has been observed that even when a geometry can exhibit both bi-modalities over a series of different experiments/runs, only one is observed in any given experiment/run - they are

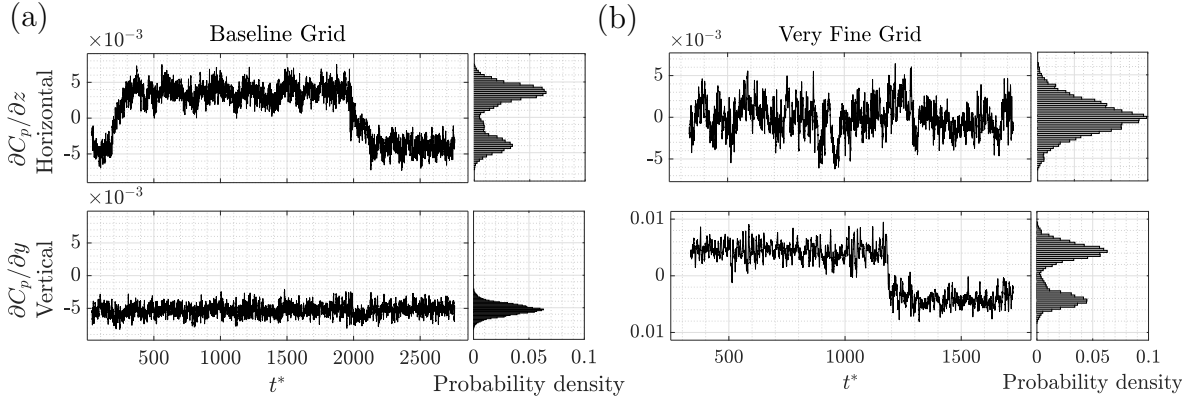


Figure 5.9: Variation of the base pressure gradients in the y and z directions over time and their corresponding probability distribution. (left) Baseline grid, (right) very fine grid.

never both observed simultaneously. Note that the Reynolds number in the Grandemange *et al.* (2013a) experiments was $Re_W \simeq 45\,000$ (equivalent to $Re_H \simeq 52\,200$ for the corresponding lorry aspect ratio $H^* = 1.16$) whereas in the current lorry simulations $Re_H = 20\,000$. It is highly probable that the boundaries of appearance of bi-modality are dependent on the Reynolds number. At lower Reynolds number, the floor is expected to have a stronger influence on the wake as the underbody flow is slower and the boundary layers are thicker. Indeed, even with the current large underbody gap, we observe some vertical asymmetry in the wake. This vertical asymmetry reduces the likelihood of the vertical bi-modality. It is therefore possible that the horizontal bi-modality exists for a wider range than in the configuration of Grandemange *et al.* (2013a) and thus their mapping should be used only qualitatively in the lorry case.

For the simplified lorry case, two behaviours are observed. We can see in figure 5.9 that for the baseline grid case, two peaks are present in the horizontal (z) gradient probability density, meaning that the wake is bi-modal in the horizontal direction. The wake which is initially located on the left, moves to the right at $t^* \simeq 200$ and stays in this position until $t^* \simeq 2000$ when it goes back to its original left location (here t^* is the non-dimensional time defined as $t^* = tU_0/H$). We expect the two locations to become equally probable as the simulation is extended. Due to the large time-scale of the phenomenon, only two bi-modal

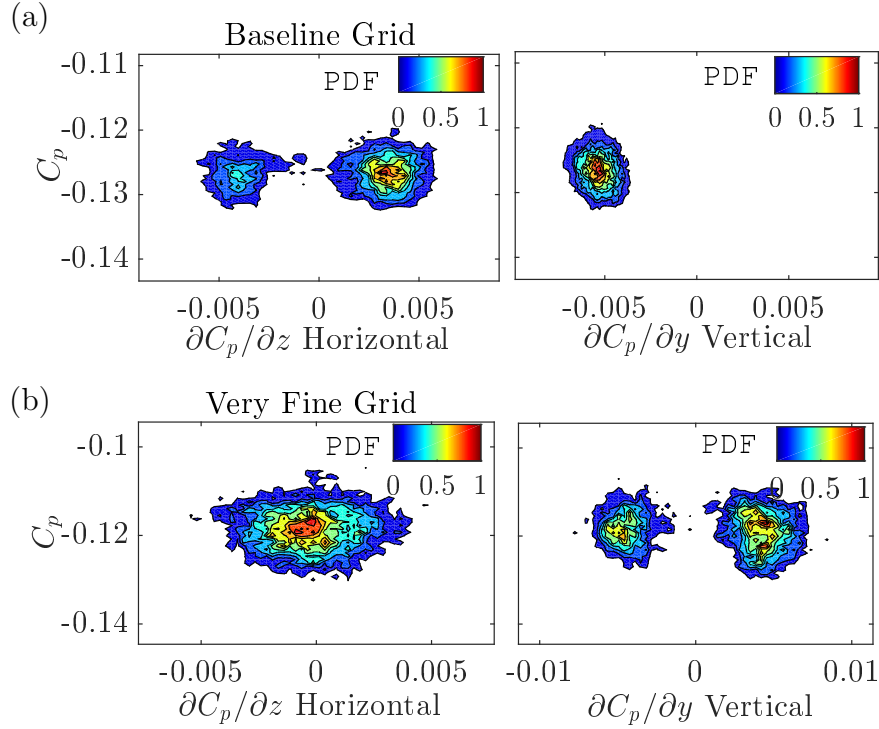


Figure 5.10: Normalized probability distribution of the horizontal and vertical base pressure gradients vs. the base pressure. (a) Baseline grid, (b) very fine grid.

switches were captured in this simulation. Note that the wake is slightly asymmetric in the vertical direction due to the presence of the floor, and fluctuates around this shifted position exhibiting a Gaussian distribution. It can also be observed that when the wake undergoes a horizontal switch, the vertical pressure gradient is affected, decreasing as the wake rotates towards the bottom to reach the opposite location.

Using the very fine grid (figure 5.9), two peaks are now present in the vertical (y) gradient probability density, meaning that the wake is bi-modal in the vertical direction. The wake undergoes a change of location at $t^* \simeq 1180$, moving from its original top state (ie. upper state, far away from the ground) towards the bottom state (ie. lower state, closer to the ground), where the wake remains for the rest of the simulation. Horizontally, large oscillations occur. Because of the large computational cost of running the very fine grid, the simulation could not be sufficiently extended to obtain a second switch.

It therefore appears that even though the mean and main unsteady flow quantities are

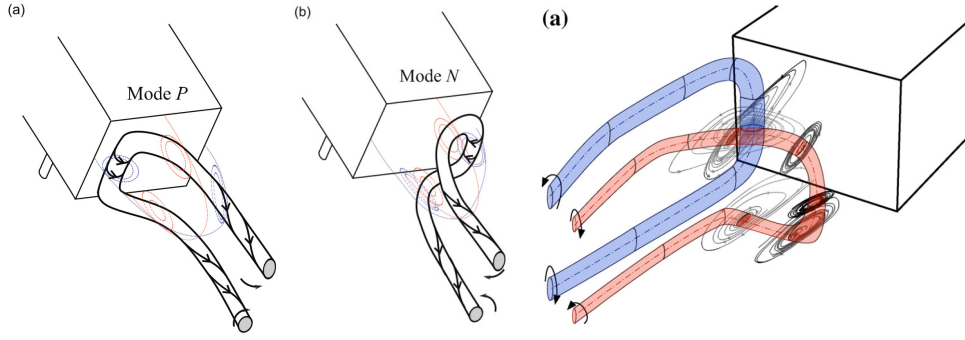


Figure 5.11: Schematics of the vortex system in the wake of a square-back Ahmed Body. (left & center) Mirrored bi-modal states from Evrard *et al.* (2016). (right) Bi-modal state from Perry *et al.* (2016). Note that the current simulation results disagree with these schematics and show that the toroidal structure remains throughout bi-modality.

correctly converged between the two grids, over-refinement of the grid affects the direction of the bi-modality. In the case of the lorry, either vertical or horizontal bi-modality can appear in the simulations depending on the grid refinement chosen. As for the Ahmed body experiments of Grandemange *et al.* (2013a), one bi-modality direction dominates over the other, and bi-modality in both directions is not observed within a single simulation. At this point, the nature of the mechanism that establishes and maintains the direction of the bi-modality remains an open question.

Figure 5.10 shows the phase diagrams of the probability density functions, $PDF(\partial C_p / \partial z, C_p)$ and $PDF(\partial C_p / \partial y, C_p)$. For the baseline grid, we observe the presence of two attractors located an equal distance from the center of symmetry. Both states are mirrored and have similar base pressure (i.e similar drag). We can see also that within each state, large oscillations in base pressure occur. For the very fine grid, the vertical bi-modality results in two attractors in the vertical gradient plane, as shown in figure 5.10.

For blunt bluff body flows, it is well-known that a low pressure toroidal vortex structure (also called “vortex ring”) typically appears in the near-wake, and can be visualised by considering iso-contours of negative pressure coefficient (Krajnović & Davidson (2003); Roumeas *et al.* (2009); Lucas *et al.* (2017)). Figure 5.12 shows high frequency snapshots of these pressure iso-contours for the very-fine grid simulations exhibiting vertical bi-

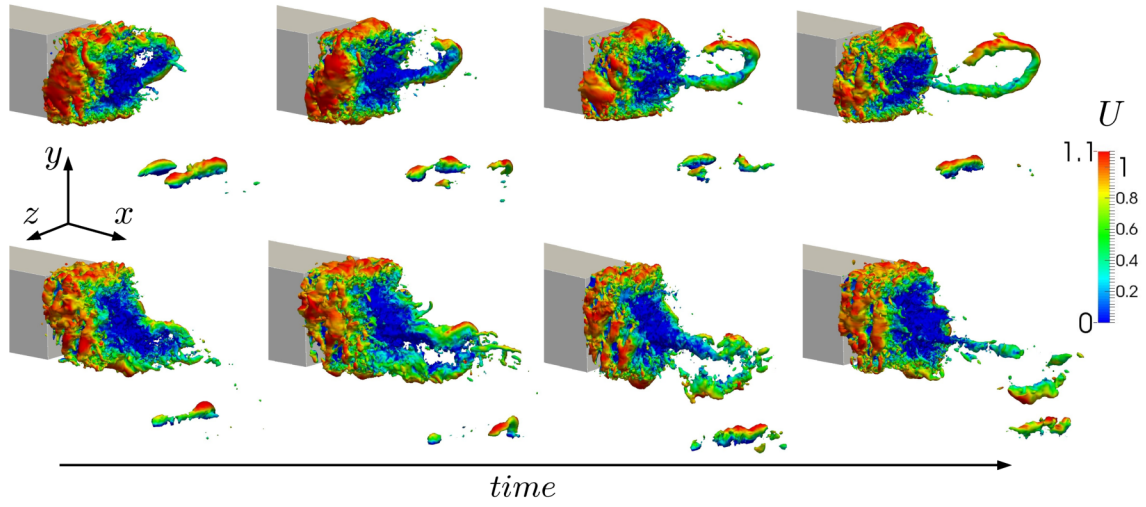


Figure 5.12: Instantaneous snapshots of iso-contours of pressure $C_p = -0.15$ coloured by velocity. (top row): wake located at the top, (bottom row): wake located at the bottom.

modality. It is apparent that the toroidal structure is vertically tilted depending on the bi-modal position of the wake. This observation differs from the schematic drawings proposed by Evrard *et al.* (2016); Perry *et al.* (2016) and deduced from PIV measurements in different planes of the wake, who suggested that the toroidal structure was broken by bi-modality (figure 5.11).

The snapshots also reveal the presence of large horseshoe structures being shed from the longest part of the wake (i.e. they originate from the wake edge located further from the body). These horseshoe structures are different from the toroidal vortex structure located in the wake recirculation region, and are swept downstream

5.2.3 Wake switching event

In the current simulation, bi-modal switching is not triggered by artificial forcing but occurs randomly during the simulations. To further understand the bi-modal switching phenomenon, the flow field during a switch was investigated in more detail by taking very

high-frequency snapshots and by placing probes downstream of the bluff body. Figure 2.3 shows the evolution of the pressure field in the vertical symmetry plane of the body (side view) as the wake moves from the “top” to the “bottom” location, it also shows the three-dimensional iso-pressure contours in the wake. We can see the toroidal low-pressure structure which is initially tilted away from the top. When the wake is in its top bi-modal state, the top of the vortex ring oscillates and its location is highly affected by the shedding of horseshoe vortices that we presented in figure 5.12. These horseshoe vortices randomly vary in size and strength. When one is large enough, it pulls the top boundary of the vortex ring downstream. Soon after, a new low-pressure core appears near the top part of the body base. Subsequently, the vortex ring splits and loses coherence before re-connecting with this new low-pressure core. In this simulation, the shedding of the horseshoe vortices appears to be responsible for initiating the bi-modal switch. Indeed during the switch, a strong enough horseshoe vortex pulls the wake downstream and as the vortex structure “reconnects”, the newly formed vortex ring is now tilted away from the bottom as in the sequence shown in figure 5.13. The sequence duration is very short ($\Delta t_{switch}^* \simeq 20$) and occurs randomly which highlights the difficulty in visualising and investigating it

Additionally probes placed at different locations of the wake measured pressure and velocity components (figure 5.14). Before the switch occurs, the vertical component of the velocity (v) near the top edge of the base (measured by the probe $e1$) exhibits unusually large fluctuations, coinciding with the presence of a stronger horseshoe vortex. The probes $e2$ and $e3$ located further downstream and near the top and bottom shear layers respectively, show the change in vertical velocity fluctuations once the switch occurs. The probe $e2$ located at the top, was initially in the core of the low-pressure wake and recorded intense fluctuations. When the wake moves from top to bottom, the pressure fluctuation level drops. The sequence is the opposite for the sensor $e3$ located at the bottom.

The triggering event and switch sequence are reminiscent of the work of Podvin & Sargent

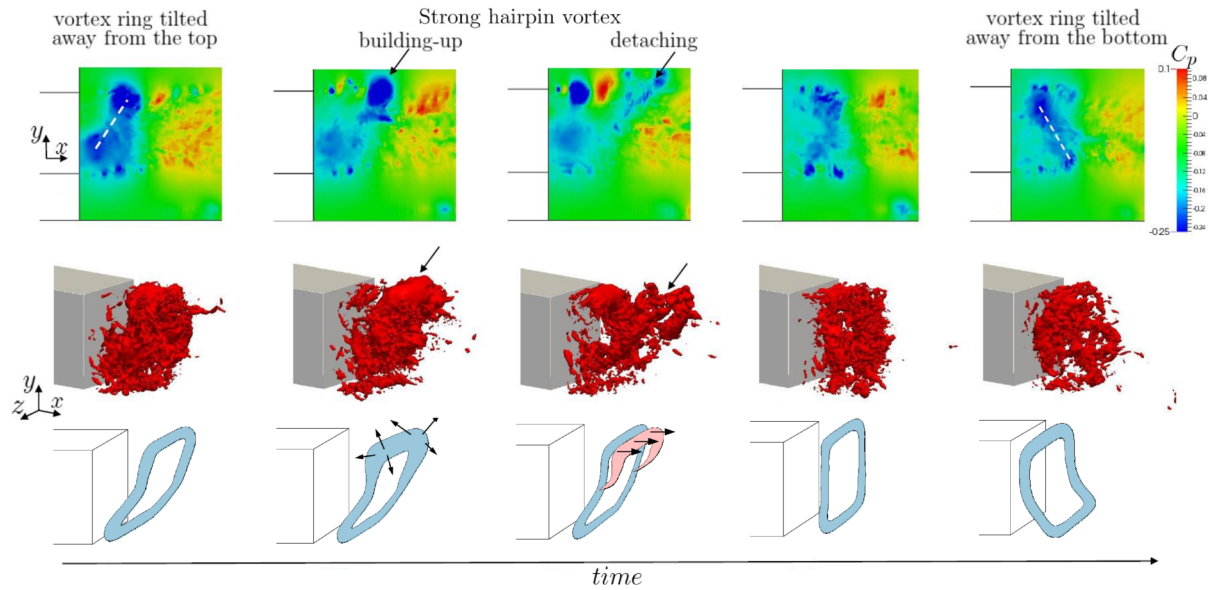


Figure 5.13: Bi-modal switching sequence. Instantaneous snapshots of the wake pressure field in the vertical symmetry plane of the body (top row - side view). Instantaneous iso-contours of pressure taken at $C_p = -0.2$ (middle row). Schematic of the toroidal vortex structure during the switch (bottom row). The wake moves from top to bottom during the switch

(2017) who studied the bi-modal behavior of a two-dimensional Rayleigh-Bénard cell. This flow also exhibits bi-modality in the form of random flow reversals. Their simulations showed that before a bi-modal switch, a precursor event “disconnects” the core vortex region from the boundary layers. During the switch, this main vortex then splits before reconnecting into an oppositely rolled vortex corresponding to the second bi-modal state.

5.2.4 Sensitivity of the bi-modality to numerical set-up

Grandemange (2013) showed that the bi-modal behavior of the wake flow is highly sensitive to external perturbations. Indeed a slight misalignment of the bluff body or an asymmetric wind-tunnel set-up (for example due to mounting arms) are enough to lock the wake in one of the asymmetric state.

Cabita probably encountered a similar issue in her experiments on the flow past the

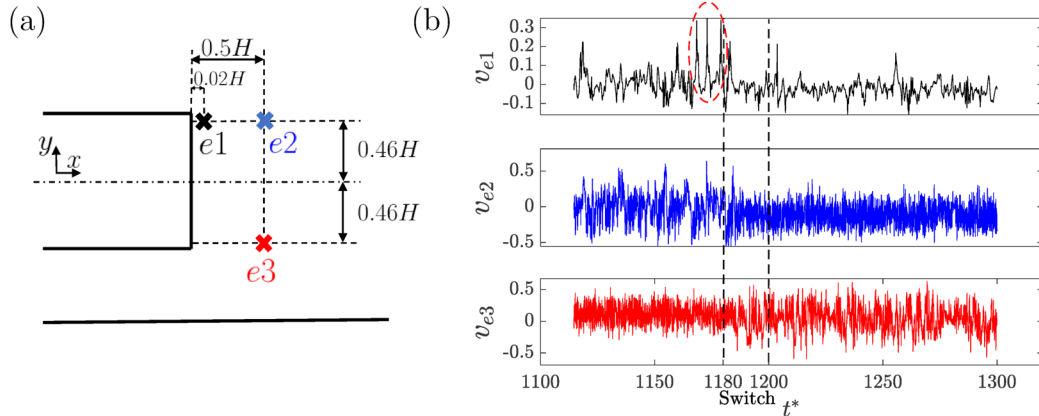


Figure 5.14: (a) Location of the velocity probes ($e1$, $e2$, $e3$) in the bluff body symmetry plane $z = 0$. (b) Vertical velocity v measured by the probes during the switch. The red ellipse highlights the unusually large velocity fluctuations before the switch occurs caused by a large horseshoe vortex.

simplified lorry geometry: the wake appeared to be locked in a shifted state and bi-modal switches were not observed (5.15). In the wind tunnel set-up a vertical mounting pillar was attached to the top side of the bluff body and could have caused the wake to be stuck on the opposite edge. Indeed if we observe carefully the time signal of the location of the center of pressure, we can notice very short switches to the opposite bi-modal state.

The high-sensitivity to set-up conditions and external perturbation is also present in numerical simulations. Indeed bi-modality captured by the current large eddy simulations also appeared to be very sensitive to slight changes in the numerical set-up. It has been shown in current simulations that bi-modality was sensitive to grid resolution changes above and beyond usual best practice requirements to solve turbulent flow and the turbulent boundary layer. This suggested a high sensitivity of bi-modality to the numerical noise caused by spatial refinement and that grid standards for bi-modality are more severe than for usual turbulent flow investigations.

Even though *Stream-LES* is intrinsically deterministic (ie. two simulations with identical settings will go through the same transients and lead to the same flow results), we observed that minor changes such as a decreasing the threshold value for the mass residual ($m_{err} = 1 \cdot 10^{-9}$ to $m_{err} = 1 \cdot 10^{-10}$) were sufficient to alter the occurrence of bi-modality. Another

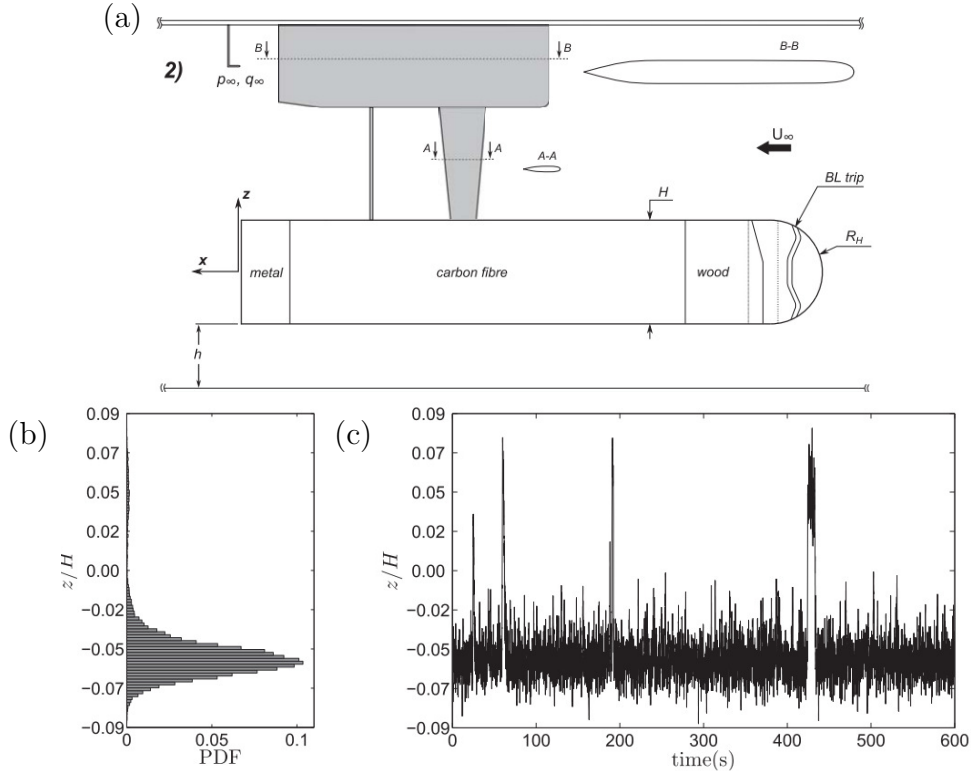


Figure 5.15: (a) Wind tunnel configuration with mounting pillar in grey color. (b) PDF and (c) time signal of the center of pressure in the vertical direction. Note that in this work z indicated the vertical direction (Cabitza (2013)).

observation was that the same simulation on ARCHER or on the Imperial HPC CX2 would not yield similar bi-modal switching times. In this case, a difference of MPI protocol and compiler (Cray on ARCHER, Intel on CX2) caused differences in the last digits (double precision) of the flow variables. These were enough to affect the bi-modal switch occurrence (see figure 5.16), even though the mean and spectral quantities of the flow remained unaffected.

To summarise, numerical results showed that wake bi-modality is highly sensitive to numerical errors or ‘noise’ generated by the flow solver. Therefore even though the turbulent flow variables are fully resolved, bi-modality results will vary across different simulation settings due to both hardware and software.

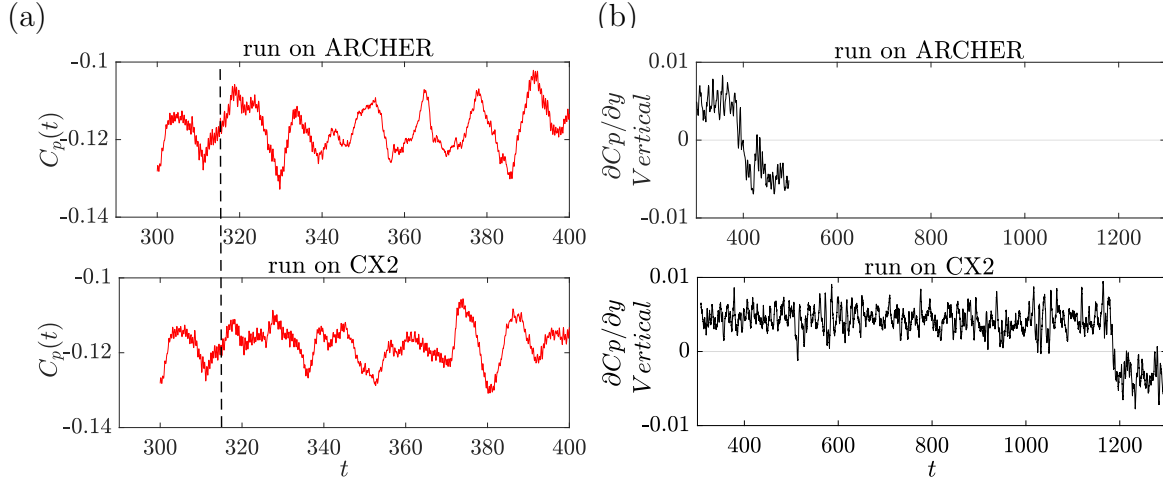


Figure 5.16: (a) Instantaneous base pressure signal and (b) vertical base pressure gradient. Very fine grid case restarted on ARCHER (top row) and CX2 (bottom row) until a bi-modal switch occurs. The dashed line indicates the time when the base pressure signals become “visibly” different.

5.3 Modal analysis

In order to further investigate changes in the wake dynamics and topology induced by bi-modality, modal decomposition is now applied to the bi-modal wake states of the flow past the lorry. Snapshots of the base pressure were taken for each wake state of the baseline and the very fine grid configurations. In total 4 samples of snapshots (composed of about 500 snapshots each), representing the top, bottom, left and right configurations were extracted. Only the wake domain is captured by the snapshots and the resolution is coarser than the simulation grid (half of the points in all directions). This resolution is fine enough to capture the large structures we are looking at using modal decomposition while saving memory and post-processing time.

The mean and the root mean square of the pressure distribution for each state are represented in figure 5.17. We can see the strong gradient in the mean base pressure towards one of the sides, whereas the area of maximum pressure fluctuation is located at the center of the wake.

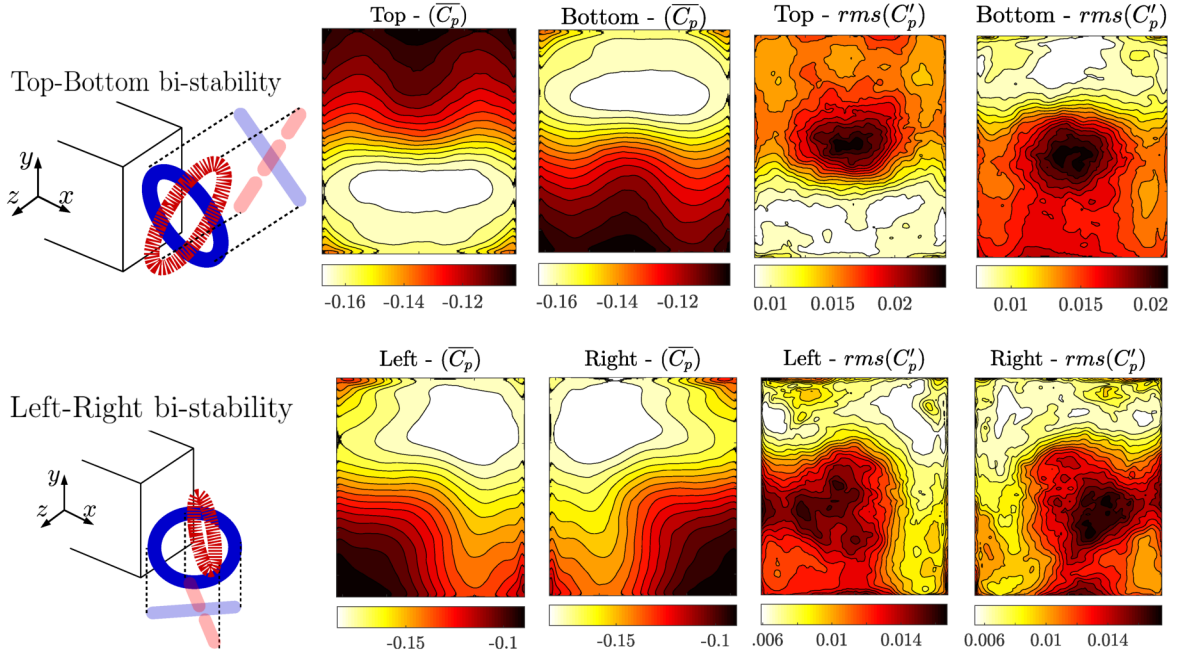


Figure 5.17: Pressure distribution on the body base (time-averaged and root-mean-square of the fluctuations). Top row: vertical bi-modality. Bottom row: horizontal bi-modality. A schematic of the toroidal vortex structure is also presented for both bi-modal directions.

5.3.1 Proper orthogonal decomposition

Proper orthogonal decomposition (POD) was applied separately to the base pressure distribution for the “top” and “bottom” wake states, in order to extract the dominant coherent structures and their energy content (Sirovich (1987); Berkooz *et al.* (1993)). The mean component of the base pressure is removed from the samples. The POD modes for these two states are shown in figure 5.18. Most of the energy is contained in the first three modes – these represent up to 93% of the total energy for both states. By projecting each mode onto the instantaneous pressure field (see figure 5.18), we observe that the first two POD modes correspond to anti-symmetric left and right states, both containing a similar amount of energy. The third mode represents a symmetric state which is less energetic. These POD modes are very similar to the ones observed by Rigas *et al.* (2014) on the axisymmetric body configuration.

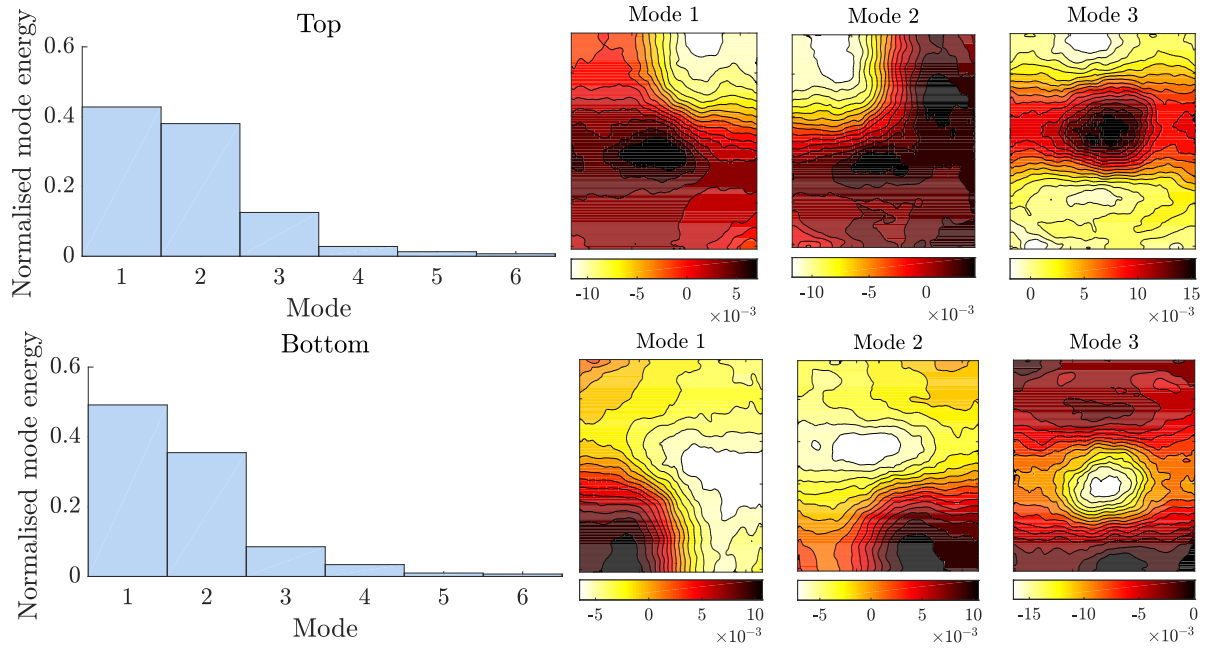


Figure 5.18: First three base pressure POD modes and energy associated: (upper row) wake in “top” state, (lower row) wake in “bottom” state.

5.3.2 Dynamic mode decomposition

In order to analyse the three-dimensional wake structures, Dynamic Mode Decomposition (DMD) was employed using three-dimensional pressure snapshots of the wake (Schmid (2010)). Each eigenmode captured by DMD is associated to a single frequency of the flow, allowing for a better understanding of the flow structures. This method has been applied to a large number of flow studies both experimental and numerical: jet-in-crossflow (Rowley *et al.* (2009)), D-shaped bluff body flow (Tu (2013)) cylinder flow at low-Reynolds number (Bagheri (2013)), Ahmed body flow at low-Reynolds number (Evstafyeva *et al.* (2017)).

As in the previous section, separate DMD analysis was performed for the wake in the “top” and “bottom” bi-modal positions. About 200 pressure snapshots were taken at constant interval every 500 time-step ($ts \simeq 3.5 \cdot 10^{-4}$ time units in the very-fine grid simulation) for each mode. To perform the DMD, the following procedure is employed:

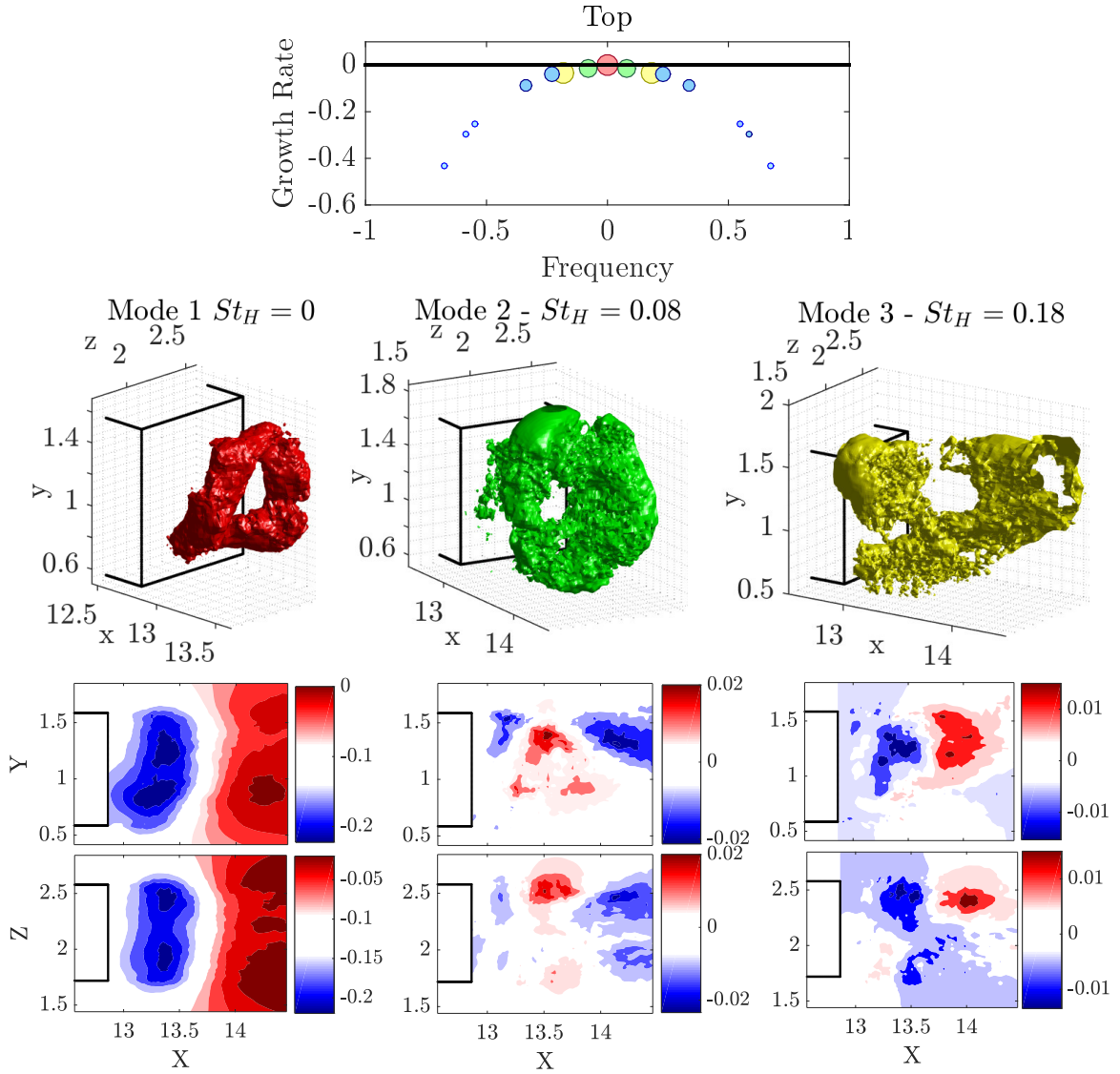


Figure 5.19: Growth rate of the DMD modes in the wake, markers are sized according to the mode amplitudes. The three main modes are coloured specifically and their corresponding spatial structure are plotted using the iso-contour of dominant amplitude. Wake in “top” state.

- The pressure snapshots $x(t)$ are collected from the flow simulation and split into two shifted matrices:

$$X1 = [x(t_1) \ x(t_2) \ \dots \ x(t_m)] \text{ and } X2 = [x(t_2) \ x(t_3) \ \dots \ x(t_{m+1})] \text{ with } X1 \text{ and } X2 \in \mathbb{R}^{n \times m} \quad (5.1)$$

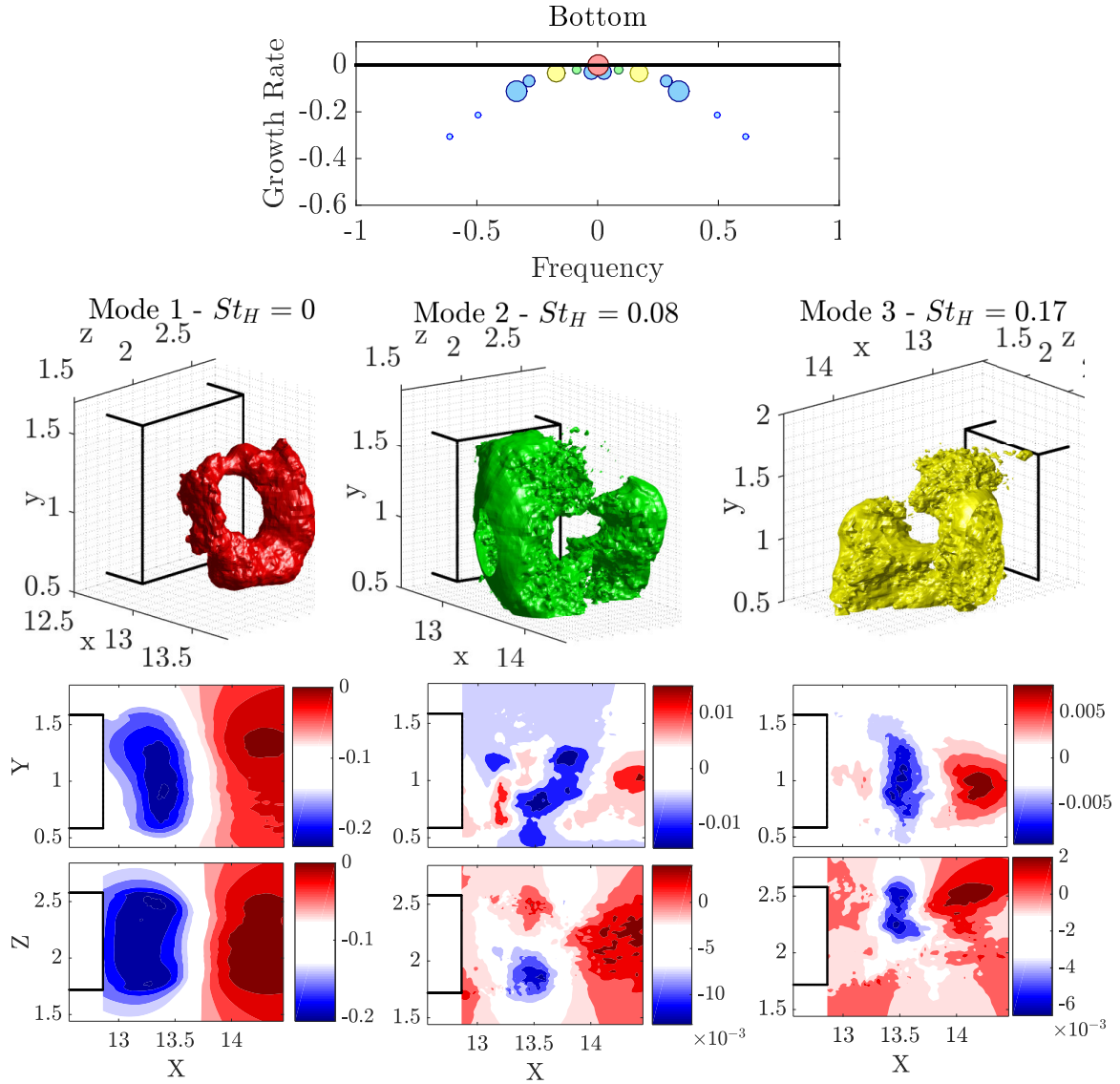


Figure 5.20: Growth rate of the DMD modes in the wake, markers are sized according to the mode amplitudes. The three main modes are coloured specifically and their corresponding spatial structure are plotted using the iso-contour of dominant amplitude. Wake in “bottom” state.

- Then we perform a reduced singular value decomposition (svd) of the snapshot matrix $X1$

$$[U, \Sigma, W] = \text{svd}(X1). \quad (5.2)$$

- The DMD matrix S is computed as the linear combinaison of the matrices previously

obtained using the svd:

$$S = U^H X 2 W \Sigma^{-1} \quad (5.3)$$

- The DMD modes v_i are finally extracted by computing the eigenvectors and eigenvalues of S .

The frequencies and growth rates (resp. the imaginary and real parts of the eigenvalues of S) of the main DMD modes are shown in figures 5.19 and 5.20 for the top and bottom positions of the wake. The first mode, coloured in red, corresponds to zero frequency and is the mean mode. It is composed of a ring-shaped vortex, tilted either towards the top or bottom. The second mode, coloured in green, corresponds to the bubble pumping mode and consists of a larger toroidal structure extending further downstream. It represents the region where the vortex ring oscillates. The third mode, coloured in yellow, is the vortex shedding mode. It highlights the shear layers merging together further downstream and creating a large bulk located either at the top or at the bottom of the base.

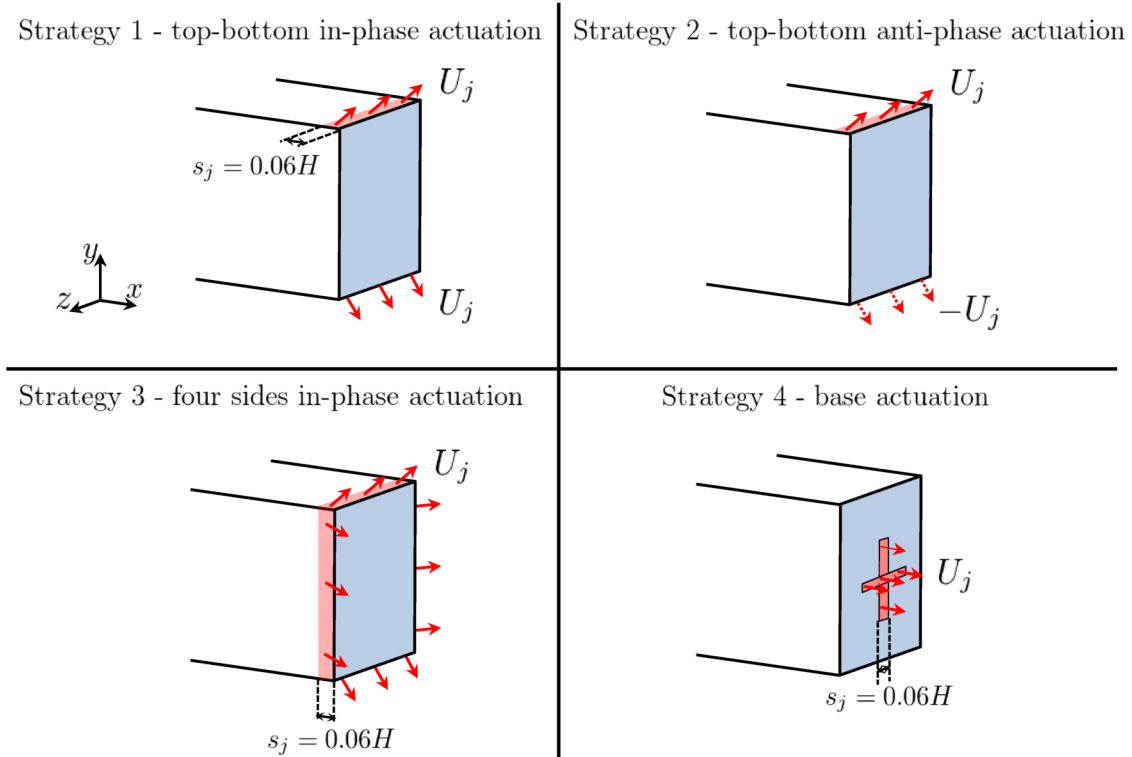


Figure 5.21: Actuation strategies tested. The location of the jets is shown using red arrows, the location of the pressure sensing (i.e. on the base) is represented in blue. s_j is the jet slot width, U_j is the jet flow velocity.

5.4 Investigations into feedback control

The same linear feedback control strategy, as described in section 4.3, is now applied to the lorry geometry across four actuation choices. The actuation is either located on a combination of the four edges of the body to target the development of the shear layers (strategies 1-3) or directly on the base to target the core of the wake (strategy 4). These actuation configurations are summarised in figure 5.21. The width of the actuation slots is $s_j = 0.06H$ in all cases. For strategies 1-3, actuation slots extend along the entire edges of the body. For strategy 4, actuation is located at the center of the body base, forming a cross of height $0.45H$ and width $0.33H$. The pressure sensing on the base was decomposed into five regions, one region covering the entire base surface used for feedback control and four reduced regions of width $(0.2H)$ located near the edges, used to record instantaneously the base pressure gradients and therefore monitor the presence of wake

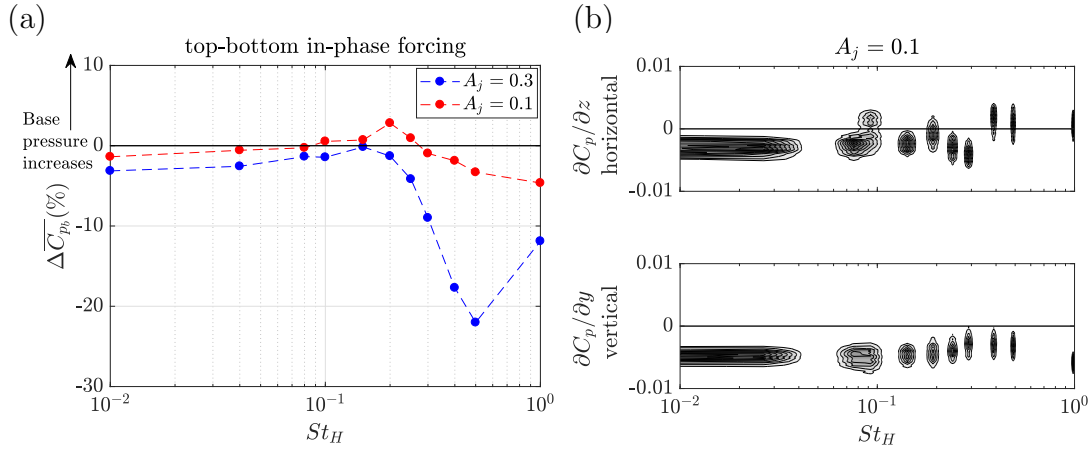


Figure 5.22: Effect of open-loop control. (a) Time-averaged base pressure for various forcing frequencies St_H . (b) Probability density function of the base pressure gradient in the horizontal (z) and vertical (y) directions as a function of the forcing frequency St_H .

bi-modality. Feedback control based on the sensitivity approach is then employed. About 100 simulations were necessary to cover all the open-loop and feedback control cases. Each simulation ran for about 24 hours on 1728 cores. The baseline grid was used for all cases.

5.4.1 Strategy 1 - Top-Bottom in-phase actuation

The first strategy consists of actuation on the top (above) and the bottom (underneath) sides of the body just before separation. Synthetics jets are oriented at 45° outwards to affect both the streamwise and the vertical (wall-normal) components of the flow velocity. The sensing uses the base pressure integrated over the lorry base surface ($S = HW$). To extract the response of the system to actuation we performed 20 open-loop harmonic forcing simulations, across 2 amplitudes, $A_j = [0.1, 0.3]$, and 10 frequencies, $St_H = [0.01 - 1.00]$.

Figure 5.22a shows the effect of the forcing on the mean base pressure. The best open-loop forcing setting is $A_j = 0.1$ and $St_H = 0.20$ (close to the vortex shedding frequency $St_H \sim 0.18$), achieving a base pressure recovery of 2.8%. The worst case involves forcing at $A_j = 0.3$ and $St_H = 0.50$ for which the base pressure is decreased by 22.2%. Note that when forcing at the highest amplitude (i.e $A_j = 0.3$), the base pressure is always

decreased (i.e. the mean drag is increased).

Figure 5.22b shows the effect of forcing (at $A_j = 0.1$) on the base pressure gradients. The contour plot represents the probability density functions of the vertical and horizontal base pressure gradients at various forcing frequencies. Initially, the unforced case (baseline grid) was bi-modal in the horizontal direction and slightly shifted downwards in the vertical direction due to the presence of the floor (figure 5.9). When forcing at low frequency ($St_H < 0.08$) the wake is still shifted vertically due to the influence of the floor as in the unforced case and also shifted horizontally to one side, probably due to the horizontal bi-modal behaviour of the wake. Because of the large number of simulations required, open-loop simulations are run for about 700 time-units (~ 24 hours on 1728 cores) to capture enough forcing cycles but not long enough to capture bi-modal switches. If the forcing frequency matches the bubble pumping frequency ($St_H = 0.08$) then the wake becomes symmetric horizontally and oscillates around the symmetry plane. As we increase again the forcing frequency ($0.08 < St_H < 0.20$), the wake recovers its original shifted state in the horizontal direction. As we reach $St_H = 0.20$ ($\simeq$ vortex shedding frequency) the wake becomes once more symmetric horizontally - it corresponds to the lowest drag state. Another behaviour appears when forcing at $0.20 < St_H < 0.40$. In this case the asymmetric behaviour of the wake appears to be enhanced. Forcing in this range also reduces the asymmetry in the vertical direction. This state exhibits higher drag. Finally at forcing frequencies higher than $St_H > 0.40$, the wake becomes symmetric horizontally and the drag is increased compared to the unforced case.

Since simulations are too short to capture the horizontal bi-modality, figure 5.23 reconstructs the possible horizontal bi-modal behaviour of the wake by adding the mirrored PDF of the horizontal base pressure gradient. By doing so, we assume: (i) that bi-modality occurs only in the horizontal direction for this case (similarly to the unforced flow), (ii) that the actuation does not prevent bi-modal switches from occurring. These assumptions would need to be confirmed by extending the open-loop forcing simulations. Netherthe-

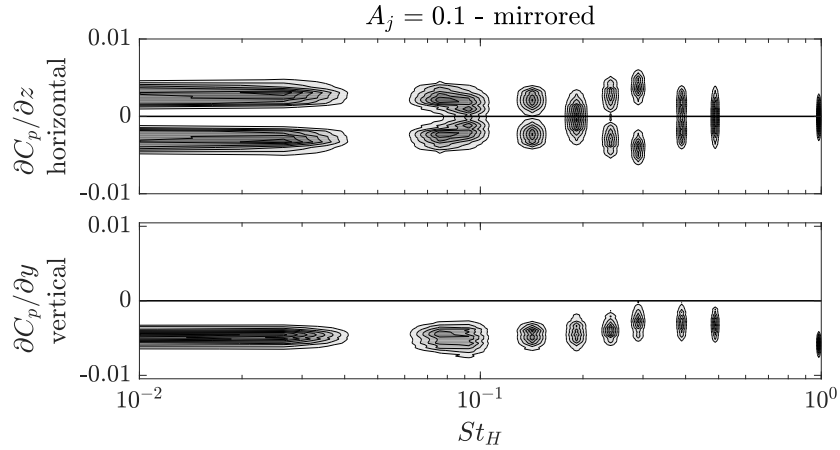


Figure 5.23: Effect of open-loop control. Probability density function of the base pressure gradient in the horizontal (z - the mirrored PDF is also added) and vertical (y) directions as a function of the forcing frequency St_H .

less, the mirroring assumption shows the frequency bandwidths where bi-modality could occur and the frequencies that tend to suppress the wake asymmetry and therefore bi-modality. Indeed the horizontal asymmetry of the wake is suppressed when forcing at $St_H \sim 0.08$, $St_H \sim 0.2$ and at high frequencies $St_H > 0.4$, this suggests that bi-modality is unlikely to occur when forcing at these frequencies.

In summary, even though forcing is only performed on the top and bottom edges of the base, it strongly affects the wake horizontal behaviour. We can identify forcing frequency ranges, bounded by physical frequencies of the wake flow (the bubble pumping, the vortex shedding frequencies), for which actuation influences the wake by either suppressing or enhancing the asymmetric behaviour and possibly bi-modality. The lower-drag case (forcing frequency of $St_H = 0.20$) corresponds to the wake being centered in the horizontal direction. The vertical asymmetry is also affected by this forcing configuration, although over a small range of forcing frequencies (the wake being less asymmetric for $St_H \sim [0.20 - 0.40]$).

Using the same approach as for the D-body, we use these harmonic forcing simulations to extract the gain and phase shift of the sensing signal compared to the actuation (figure 5.24) in order to deduce a state-space model $G(iSt_H)$ representative of the open-loop

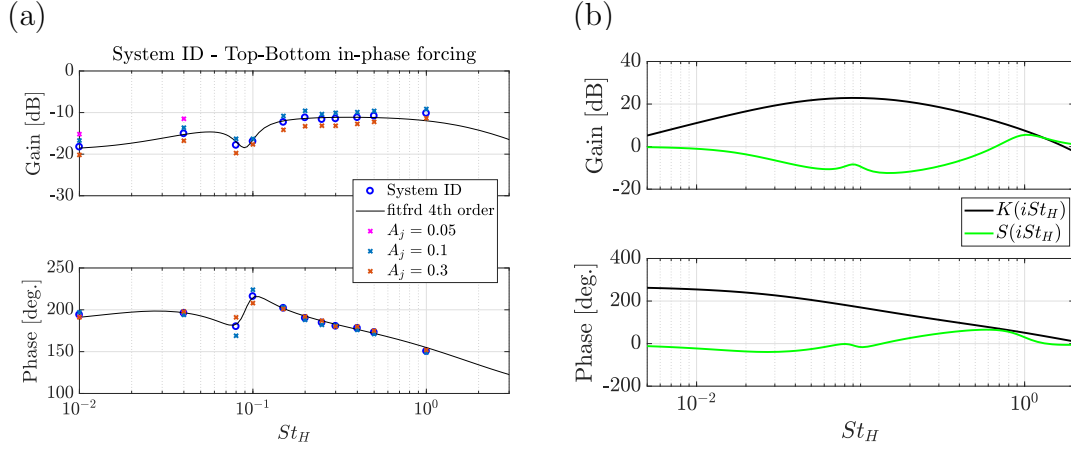


Figure 5.24: (a) System identification of $G(iSt_H)$. (b) Controller design $K(iSt_H)$ and sensitivity $S(iSt_H)$.

response of the flow to actuation. We then chose a fitting function to approximate the averaged frequency response. Using the *fitfrd* Matlab command, we chose a 4th order fitting function.

Subsequently, a controller $K(s)$ is designed manually in the frequency domain to reduce the base pressure fluctuations at the dominant frequencies of the wake flow. A few iterations of the controller are tested to find a good compromise between performance and closed-loop stability. $K(s)$ is formed by multiplying a 2nd order low pass filter K_{LP} , aiming at attenuating the main physical frequencies of the flow, and a 2nd order band pass filter K_{BP} , designed to suppress the mean component of the actuation signal to make it zero-net-mass-flux:

$$K_{LP}(s) = \frac{k\omega_c^2}{s^2 + 2\xi s\omega_c + \omega_c^2}, \quad K_{BP}(s) = \frac{s\omega_h}{(\omega_l + s)(\omega_h + s)} \quad (5.4)$$

The controller gain k , cutoff frequencies $(\omega_c, \omega_h, \omega_l)$ and the damping parameter ξ are tuned manually and iterated.

Various controllers were tested, with the final controller design being:

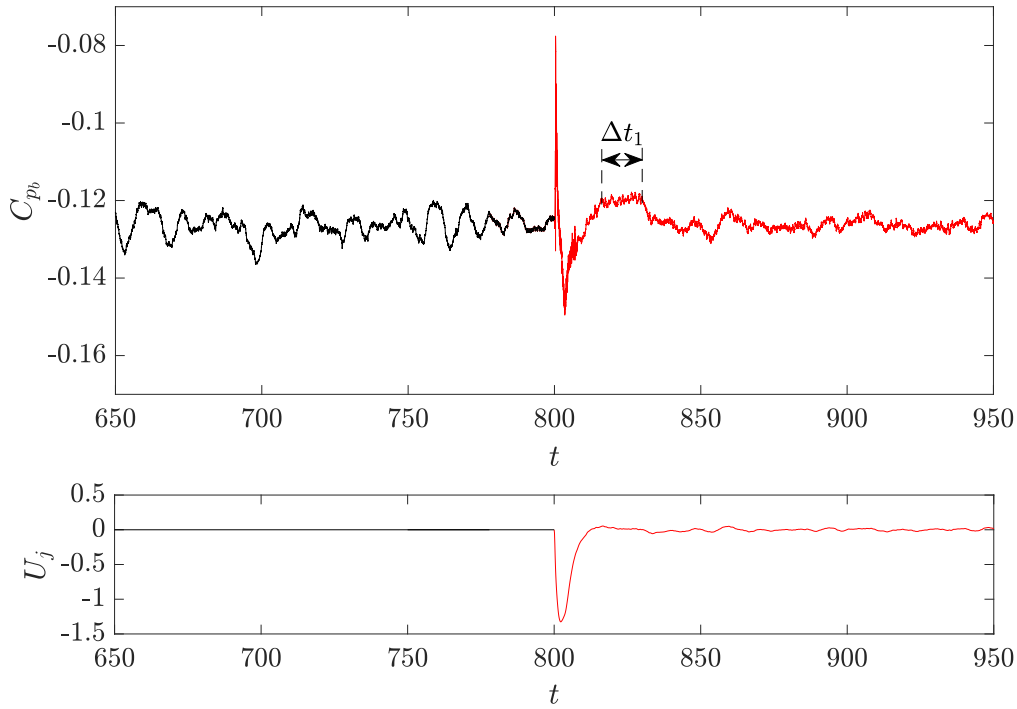


Figure 5.25: Effect of control, showing the instantaneous base pressure signal $C_{pb}(t)$ and the forcing signal $U_j(t)$. Feedback control is switched on at $t = 800$.

$$K(s) = \frac{-3500}{s^2 + 20s + 20} \cdot \frac{s}{s^2 + 10s + 3} \quad (5.5)$$

Using this controller, the sensitivity magnitude is less than 0 dB for $0.01 < St_H < 0.6$ and therefore should attenuate both the bubble pumping and the vortex shedding frequencies.

Figure 5.25 shows the effects of control on the base pressure. Feedback control is turned on at $t = 800$. Initially, the forcing signal exhibits a strong transient before converging as it becomes zero-net-mass-flux. We can see that shortly after converging, the wake reaches a low-drag state for a short period of time shown as Δt_1 . During this period of time, the low-frequency bubble pumping mode is not present in the base pressure signal. The base pressure increases by 5.4%. Unfortunately, the controller is not able to maintain this system in this state and the bubble-pumping frequency re-appears, although it is attenuated. The base pressure is then almost similar to the unforced case (decreased by

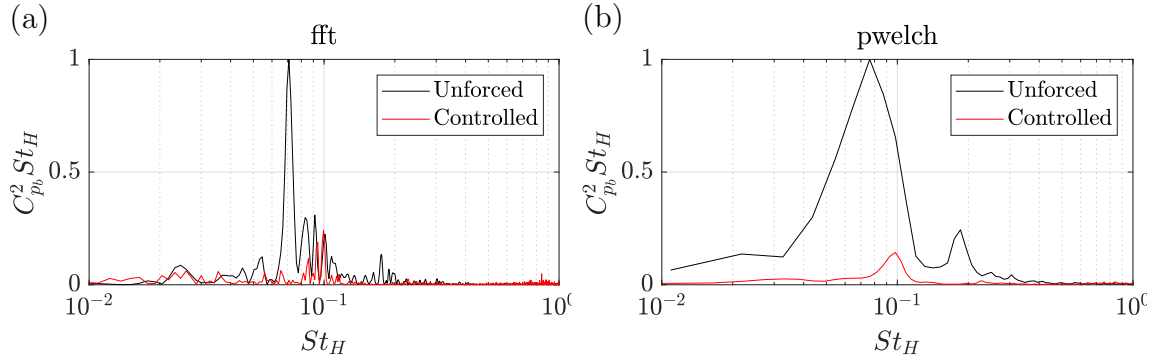


Figure 5.26: Effect of control, showing the base pressure energy spectrum. (left) Standard fft version, (right) Filtered spectrum with pwelch.

0.3%). Multiple iterations of this controller design were tested, but could not maintain the system in the low-drag state or were too unstable to be used. Figure 5.26 shows the energy spectrum of the base pressure. The controller achieves its objective of reducing the base pressure fluctuations with both the bubble pumping and vortex shedding frequencies attenuated. Note that the controller shifts the frequency of the bubble pumping mode ($St_{H-b.p,unforced} \sim 0.08$, $St_{H-b.p,controlled} \sim 0.10$).

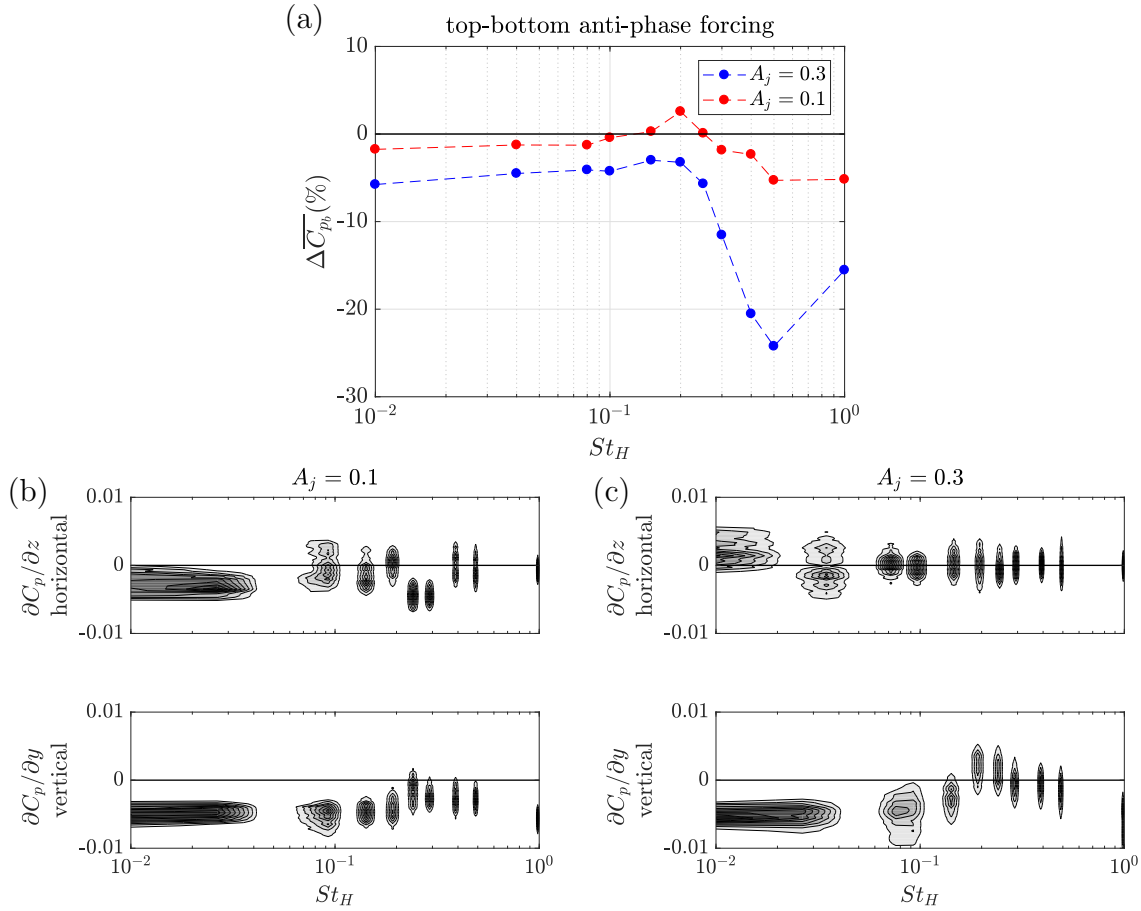


Figure 5.27: Effect of open-loop control. (a) Time-averaged base pressure for various forcing frequencies St_H . (b-c) Probability density function of the base pressure gradient in the horizontal (z) and vertical (y) directions as a function of the forcing frequency St_H .

5.4.2 Strategy 2 - Top-Bottom anti-phase actuation

The second strategy employs the same actuation locations as for strategy 1, but now the jets located on the top ($U_{j,t}$) and bottom ($U_{j,b}$) edges act in anti-phase by simply setting $U_{j,b} = -U_{j,t} = -A_j \sin(2\pi St_H t)$. Here again, harmonic forcing across two amplitudes, $A_j = [0.1, 0.3]$, and 10 frequencies, $St_H = [0.01 - 1.00]$, is performed.

Figure 5.27 shows the effect of the open-loop forcing on the time-averaged base pressure and the base pressure gradients. The results are very similar to those of strategy 1, with the best-open loop actuation setting being $A_j = 0.1$ and $St_H = 0.20$, for which a base pressure recovery of 2.6% is achieved. The worst drag case occurs for $A_j = 0.3$ and

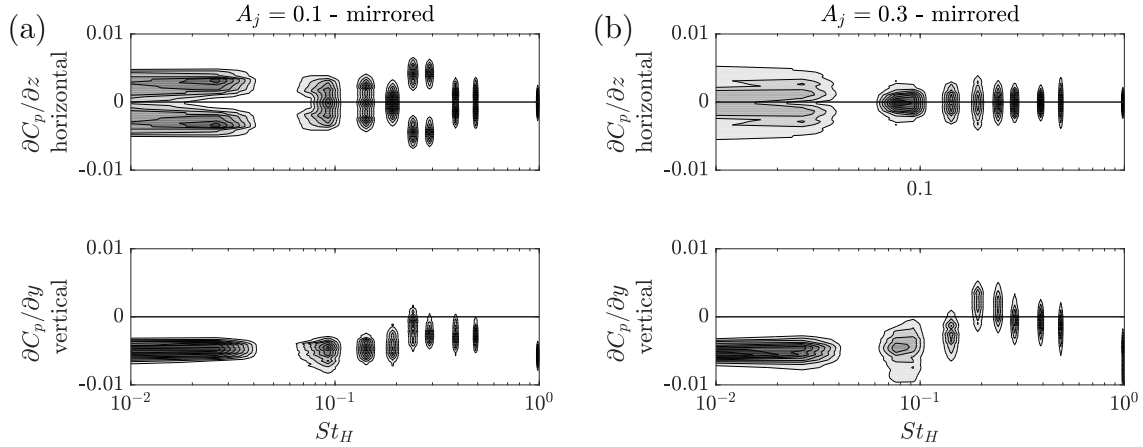


Figure 5.28: Effect of open-loop control. (a-b) Probability density function of the base pressure gradient in the horizontal (z - mirrored pdf is added) and vertical (y) directions as a function of the forcing frequency St_H .

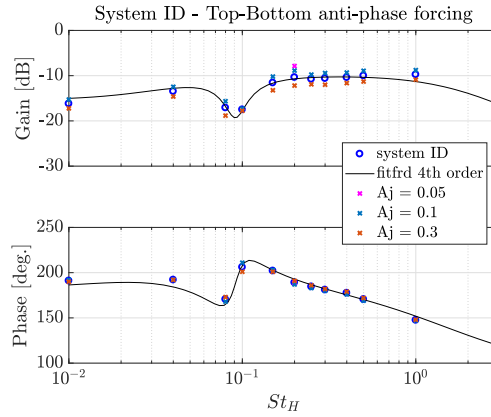


Figure 5.29: (left) System identification of $G(iSt_H)$. (right) Controller design $K(iSt_H)$ and sensitivity $S(iSt_H)$.

$St_H = 0.50$, decreasing the base pressure by 24.2%. Here again, the highest jet amplitude (i.e. $A_j = 0.3$), is too strong and has an adverse effect on the base pressure which is always decreased. Base pressure gradients are affected in a similar manner to strategy 1 (figure 5.27, horizontal-mirrored version: figure 5.28). For this case we also recorded the effect of forcing at $A_j = 0.3$ on the base pressure gradients. The horizontal wake asymmetry is suppressed over most of the forcing frequencies due to the stronger jets. The effect of actuation on the vertical gradients is also much clearer. For forcing frequencies in the range $0.2 < St_H < 0.5$, the wake becomes symmetric vertically, indicating that the influence of the floor is attenuated in this frequency range.

Extracting the open-loop harmonic response from the open-loop forcing simulations (figure 5.29) and using a 4th fitting function, we obtained the averaged state-space model $G(s)$, for which the gain and phase are very similar to those obtained for strategy 1.

Several controllers aiming at reducing the base pressure fluctuations at the dominant frequency of the flow were tested. They produced similar results than those of strategy 1: none of them yield a significant mean base pressure increase.

5.4.3 Strategy 3 - Four sides in-phase forcing

The third actuation strategy consists of actuating over all four edges of the body before separation. Jets are oriented at 45° outwards. The sensing uses the base pressure integrated all over the lorry base surface ($S = HW$). We expect this configuration to have better control authority on the wake-flow compared with strategies 1 and 2 due to the additional jets.

The response of the system to actuation is obtained by computing ten harmonic open-loop simulations. One forcing amplitude $A_j = 0.3$ and ten forcing frequencies in the range $St_H = 0.01 - 1.00$ are tested. Figure 5.30a shows the effect of the forcing on the integrated base pressure and on the base pressure gradients. The lowest drag open-loop forcing setting occurs at $St_H = 0.08$, the bubble pumping frequency of the wake, achieving a base pressure recovery of 1.7%. This actuation configuration therefore affects the flow differently to the top-bottom strategies for which the vortex shedding frequency ($St_H \sim 0.2$) gave the lowest drag flow. When the forcing frequency exceeds $St_H \geq 0.15$, actuation has an adverse effect and the base pressure is significantly decreased (up to 43.0% for $St_H = 0.40$). Figure 5.30b shows that for forcing frequencies in the range $St_H = 0.25 - 0.40$, the base pressure gradients exhibit two possible values, indicating wake bi-modality. The range of forcing frequencies over which bi-modality occurs, correspond to the highest drag state (lowest base pressure). Additionally, we can see that the forcing causes the wake to be bi-modal not only horizontally but also vertically, with bi-modality in the vertical ($St_H \sim 0.25$) and horizontal ($St_H \sim 0.30 - 0.40$) directions occurring at different forcing frequencies. To summarise, even though we are forcing similarly on all four sides of the base, the wake reacts differently to actuation in the vertical and horizontal directions.

Figure 5.31a shows the gain and phase shift measured for each open-loop simulation. A 3rd order linear state-space model $G(s)$ can be fitted through these points.

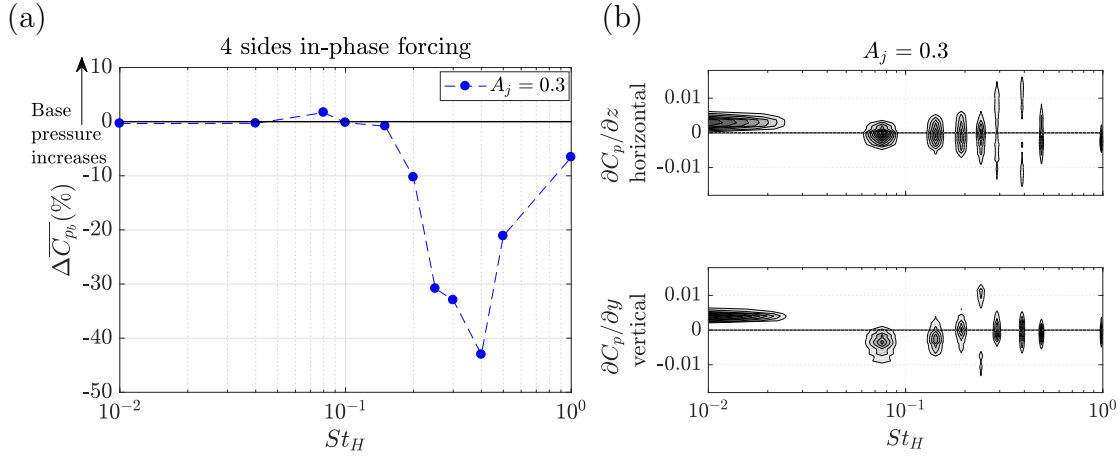


Figure 5.30: Effect of open-loop control. (a) Time-averaged base pressure for various forcing frequencies St_H . (b) Probability density function of the base pressure gradient in the horizontal (z) and vertical (y) directions as a function of the forcing frequency St_H .

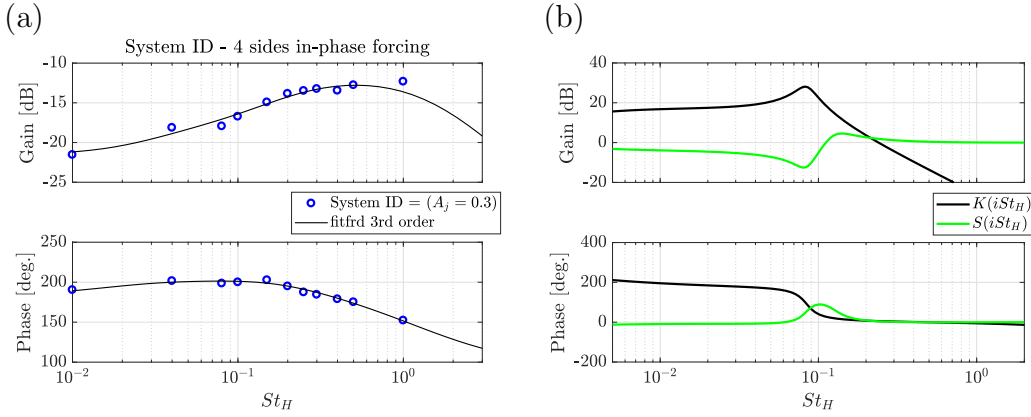


Figure 5.31: (a) System identification of $G(iSt_H)$. (b) Controller design $K(iSt_H)$ and sensitivity $S(iSt_H)$.

A controller, $K(s)$, is designed to reduce pressure fluctuations at the bubble pumping frequency $St_H \sim 0.08$ - the dominant frequency in the wake flow. $K(s)$ is composed again of a 2nd order low-pass filter multiplied by a 2nd order band-pass filter which removes the mean from the actuation signal, with the optimum choices being:

$$K(s) = \frac{-100}{s^2 + 0.15s + 0.28} \cdot \frac{s}{s^2 + 50s + 1} \quad (5.6)$$

The sensitivity of this controller is lower than 0dB for frequencies $St_H < 0.12$. Because of the “water-bed” effect, the gain of the sensitivity is higher than 0dB for $0.12 < St_H < 0.22$ and therefore we expect the frequencies near the vortex shedding mode ($St_H \sim 0.18$) to be

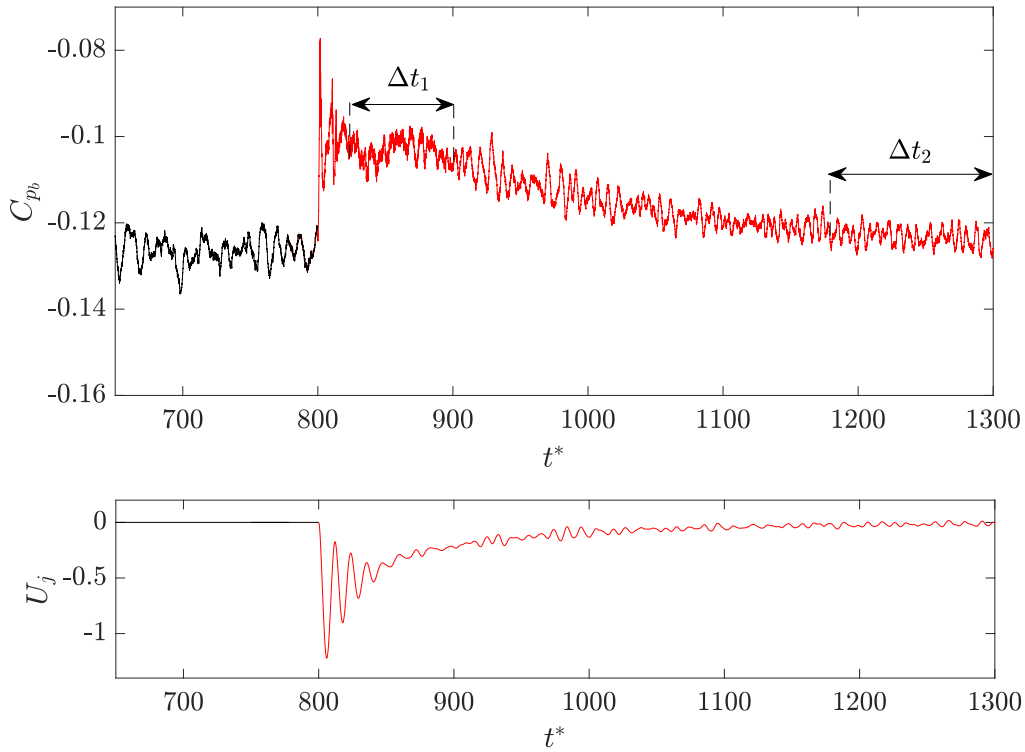


Figure 5.32: Effect of control, showing the instantaneous base pressure signal $C_{pb}(t)$ and the forcing signal $U_j(t)$. Feedback control is switched on at $t = 800$.

amplified. Due to the proximity of the negative and the positive peaks in the sensitivity gain, it is difficult to design a controller targeting the bubble pumping frequency without affecting the vortex shedding frequency.

Figure 5.32 shows the effects of control on the base pressure signal. Feedback control is turned on at $t = 800$. Initially, the forcing signal exhibits a strong transient and the jets operate with negative values of velocity. For a short period of time, shown as Δt_1 , the base pressure is significantly increased (+17.9%). However as the jet signal slowly converges towards zero-mean actuation, the base pressure recovery is diminished. When the jet signal is converged (Δt_2) and the jets operate in zero-net-mass-flux, the base pressure increase compared to the unforced case is 2.1%.

Figure 5.33 shows the energy spectrum of the base pressure. We can see that the controller achieves its objective of reducing the base pressure fluctuations at the bubble frequency

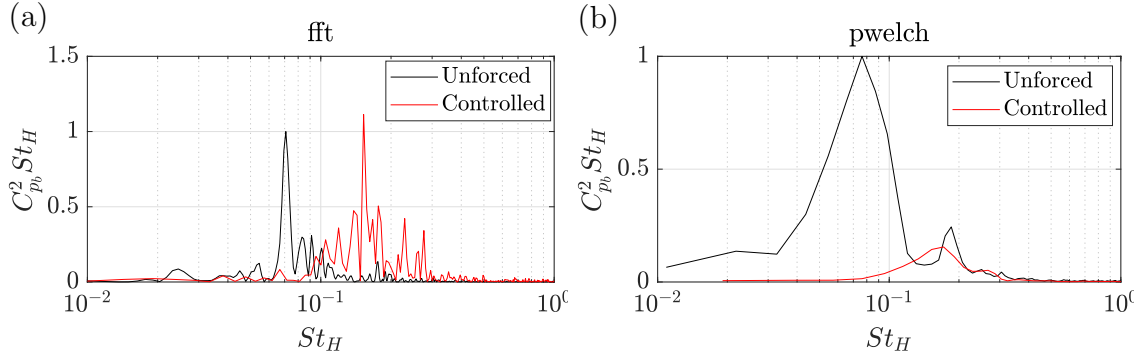


Figure 5.33: Effect of control, showing the base pressure energy spectrum. (left) Standard fft version, (right) Filtered spectrum with pwelch.

($St_H \sim 0.08$). Additionally, a range of frequencies around the vortex shedding frequency ($St_H \sim 0.18$) are significantly enhanced by the controller as expected.

To summarise, strategy 3 results showed that when forcing on all four sides, the flow reacts positively when forcing at the bubble pumping frequency ($St_H \sim 0.08$) at which we obtained the largest drag reduction in open-loop. We designed a controller to target base pressure fluctuations at the bubble bumping frequency. The feedback controller achieved some significant base pressure increase, but exhibited a long transient phase, during which it provided mean-suction making the system unphysical. Several iterations of this controller with an improved zero-net-mass-flux enforcement were tested. These controllers were successful in attenuating the base pressure fluctuations but did not lead to a mean base pressure increase.

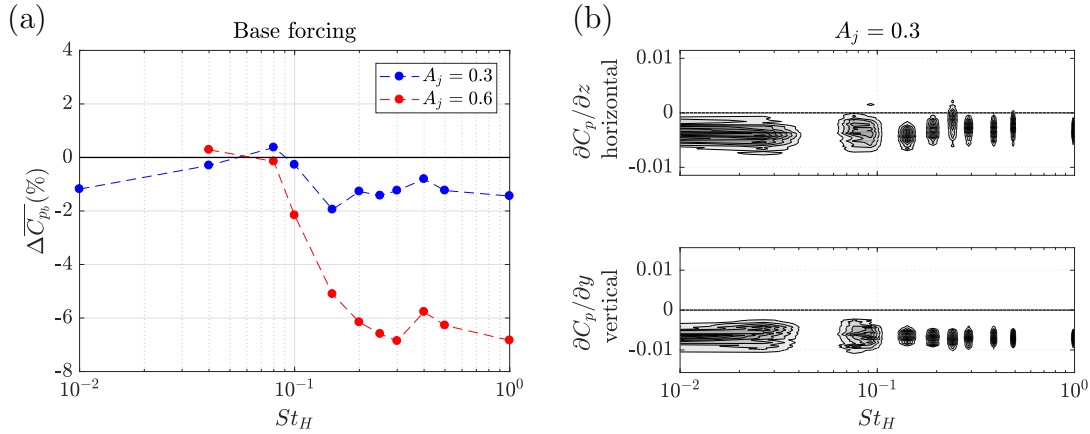


Figure 5.34: Effect of open-loop control. (a) Time-averaged base pressure for various forcing frequencies St_H . (b) Probability density function of the base pressure gradient in the horizontal (z) and vertical (y) directions as a function of the forcing frequency St_H .

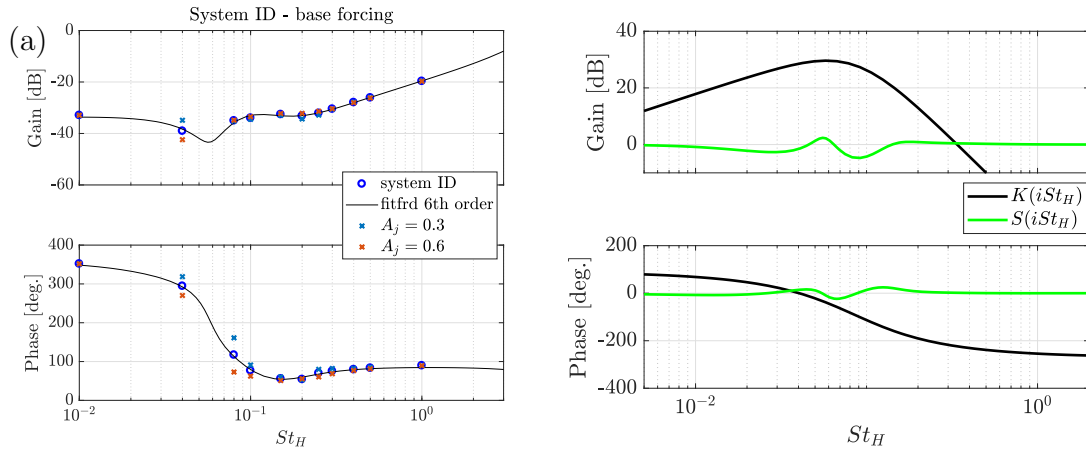


Figure 5.35: (a) System identification of $G(iSt_H)$. (b) Controller design $K(iSt_H)$ and sensitivity $S(iSt_H)$.

5.4.4 Strategy 4 - Base forcing

The last strategy consists of actuation directly on the base surface to affect the core of the wake. Jets are oriented in the streamwise direction (jet angle = 0°) and placed at the center of the base, forming a cross. Harmonic forcing for one forcing amplitude, $A_j = 0.3$, and ten forcing frequencies, $St_H = 0.01 - 1.00$, was applied. Figure 5.34 shows the effect of forcing on the mean base pressure and the base pressure gradients. If we compare the variations of the mean base pressure for this case with the previous strategies, we can see that the effect is very limited. For the best forcing setting ($A_j = 0.3, St_H = 0.08$), the

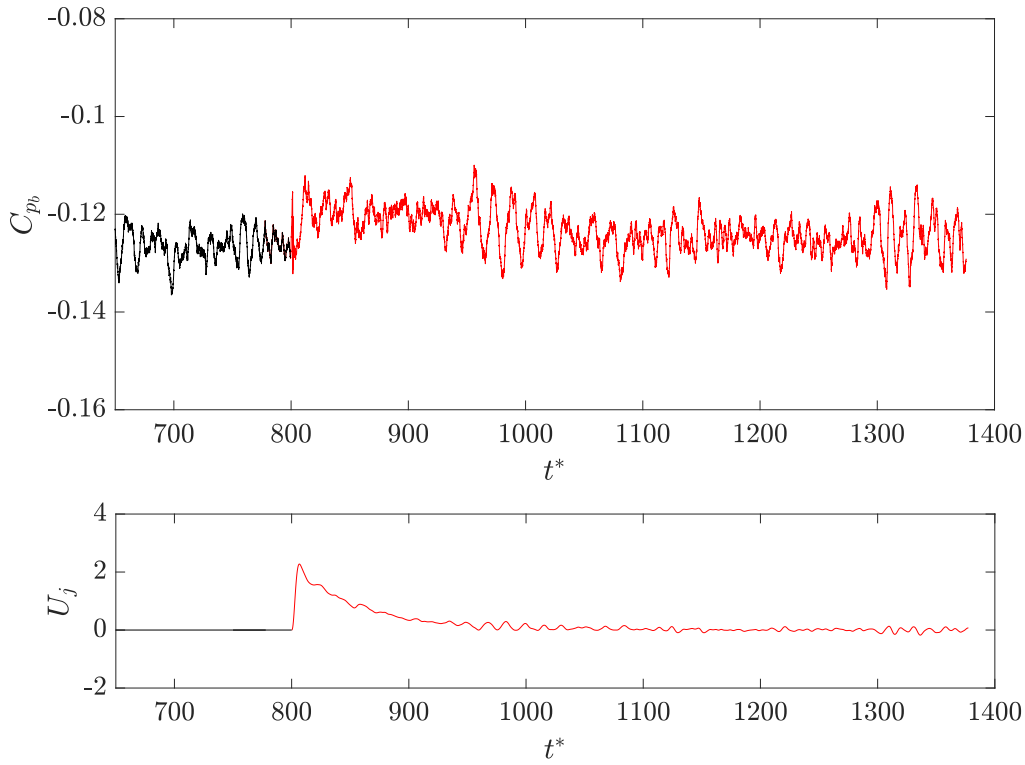


Figure 5.36: Effect of control, showing the instantaneous base pressure signal $C_{pb}(t)$ and the forcing signal $U_j(t)$. Feedback control is switched on at $t = 800$.

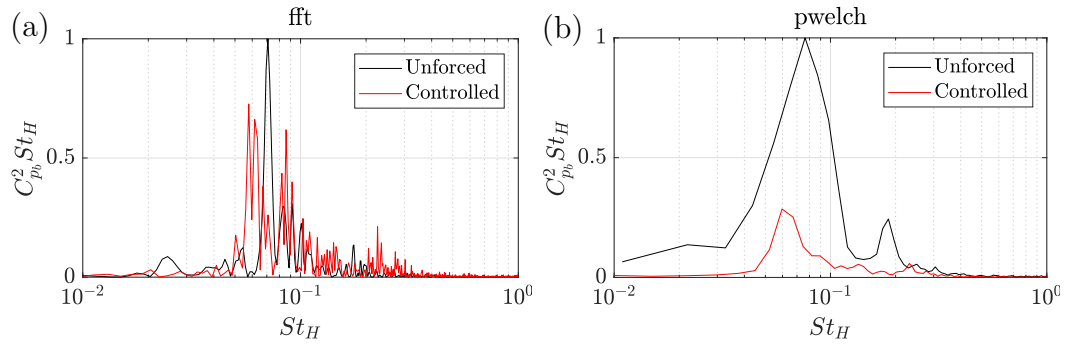


Figure 5.37: Effect of control, showing the base pressure energy spectrum. (left) Standard fft version, (right) Filtered spectrum with pwelch.

base pressure increases by 0.4%, whereas the worst forcing setting ($A_j = 0.3, St_H = 0.15$) leads a base pressure decrease of only 1.9%. The synthetic jet seem to have a weaker effect on the wake flow when they are located directly on the bluff body base. Additional simulations with an increased forcing amplitude of $A_j = 0.6$ were performed, but the mean base pressure decreased for most frequencies tested (figure 5.34).

The open-loop response of the flow to actuation, $G(s)$, was then measured. The “average” frequency response across the two forcing amplitude was then fitted with a 6th order model (figure 5.35).

Then a controller, $K(s)$, targeting the bubble-pumping frequency, was designed manually in the frequency domain yielding a sensitivity magnitude less than unity for $0.05 < St_H < 0.15$. The resulting controller is:

$$K(s) = \frac{10}{s^2 + 0.70 + 0.20} \cdot \frac{s}{s^2 + s + 0.4} \quad (5.7)$$

Figure 5.36 shows the effects of the feedback controller on the mean base pressure. Initially, the controller delivers a large amount of momentum to control the flow, the base pressure increases and the amplitude of the pressure fluctuations is attenuated. But soon after $t^* \simeq 950$, when the jet becomes zero-net-mass-flux, the base pressure recovery vanishes. The controller attenuates the base pressure fluctuations at the bubble pumping frequency (figure 5.37), but is unable to maintain a mean base pressure increase.

5.5 Conclusions

This chapter has presented wall-resolved simulations of the flow around a simplified lorry geometry. Simulations were performed at a turbulent Reynolds number of $Re_H = 20,000$ and were validated against the wind tunnel experiments of Cabitza (2013).

For the first time, the bi-modal switching behaviour of the wake was captured numerically. Using high-frequency snapshots, we provided a plausible explanation of the phenomenon triggering switches. The body geometry and floor distance lay in the interfering region exhibiting vertical and horizontal bi-modality, as defined by Grandemange *et al.* (2013a). Different simulation grids were used, and it was found that even when the mean and

spectral features of the flow were resolved with grid, the direction of the bi-modality was dependent on level of grid over-refinement. It was also shown that bi-modality is highly sensitive to numerical noise and we observed different bi-modality switches between two simulations running on different hardware and software set-ups. Due to the very long timescales associated with bi-modality, only one or two wake switches were captured in each simulation. The simulations provided access to full flow-field data, allowing the wake structures associated with bi-modality to be investigated in a new level of detail. The toroidal vortex structure in the near wake was found to remain intact throughout bi-modality, but tilted according to the wake position. Large horseshoe structures were identified as being shed from the longest part of the wake. These horseshoe structures appear to be responsible for triggering bi-modal switches. Modal analysis of the asymmetric wake states provided further insight into the wake topology changes.

Additionally, the feedback control strategy based on sensitivity was extended to the lorry case. Four different forcing configurations were tested. The link between base pressure fluctuations and mean base pressure increase proved to be unclear for this particular wake flow. Most of the forcing configurations were successful in damping the base pressure fluctuations but did not lead to a mean base pressure recovery. The best case (strategy 3), for which jets were placed on all four sides of the body just before separation, achieved 2.1% of base pressure increase. However, the jet signal exhibited a long non-zero mean transient, making the system unrepresentative of real synthetic jets.

Chapter 6

Conclusion

6.1 Conclusion and Perspective

6.1.1 Conclusions

The present work has presented wall-resolved large eddy simulations of the flow past square-back geometries. It also presented a concept for reducing the aerodynamic drag of bluff body flows, based on sensing and manipulating only fluctuating flow quantities. This is underpinned by theory, and facilitates the use of linear feedback control strategies which target attenuation of temporal base pressure fluctuations.

In a first part, the flow over a simple blunt bluff body (a “D-shaped body”), with upper and lower edges exhibiting interacting shear layers, was considered. Actuation in the form of zero-mean slot jet forcing at 45° just ahead of separation was used, and a relevant form of the spatially integrated base pressure was used for sensing. Large eddy simulations were used as a test-bed. The response to actuation was characterised via harmonic forcing system identification, and the controller was designed in the frequency domain, so as to give fluctuation attenuation over the frequency range in which the flow dynamics were

concentrated. Upon implementing the feedback controller, it achieved both its direct aim of attenuating base pressure fluctuations, and its indirect aim of increasing the mean base pressure, in this case by 38%.

In a second part, the flow over a three-dimensional square-back geometry in presence of a fixed floor was considered. The bluff body was taller than wide with an aspect ratio corresponding to European lorries. Various actuation strategies were tested, synthetic jets placed either on the perimeter of the base, or directly at its center. The spatially integrated base pressure was used for sensing. The unforced flow computations captured for the first time using numerical simulations the bi-modal switching behavior of the wake which had been only observed experimentally. The simulations provided access to the full flow-field data, allowing the wake structure associated with bi-modality to be investigated in a new level of detail. In particular, the wake switching sequence was captured using high-frequency snapshot and provided a plausible explanation of the phenomenon triggering the switch. To finish, various actuation strategies were assessed. Open-loop forcing results showed that it was possible to increase the mean base pressure using synthetic jets - in the best case by 2.8 %. Various feedback controllers, aiming at suppressing the base pressure fluctuations at the main frequencies of the wake flow, were assessed. Base pressure fluctuations were significantly reduced as targeted but it appeared challenging to obtain a clear mean base pressure recovery.

The open-loop and feedback control results show that there is some potential in this active control technique for the lorry case, but the current forcing-sensing and controller implementations are certainly not ideal for this flow configuration and would necessitate additional trials.

6.1.2 Perspectives

The current large eddy simulations proved to be successful in capturing the features of the flow past square-back geometries. The simulations provided additional valuable insights, in particular about wake bi-modality. Active feedback control using a sensitivity approach also showed promising outcomes in reducing the aerodynamic drag of simplified bluff bodies. The control strategy is based only on body-mounted sensing and actuation, does not require solving the adjoint equations, and is not based on approximations of the infinite-dimensional governing flow equations. Thus the strategy would apply to moving bluff body experiments. The simplified lorry geometry showed that the link between base pressure fluctuations and mean base pressure was not as clear for complex three-dimensional geometries. The best way to control such three-dimensional wake to achieve drag reduction remains an open question. The current approach will be extended and assessed on a realistic car geometry through the PhD work of F. Hesse, hopefully leading to an application on real road vehicles in the coming years.

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