Vortices in turbulent curved pipe flow—rocking, rolling and pulsating motions

by

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To Johan and Christo
This thesis is motivated by the necessity to understand the flow structure of turbulent flows in bends encountered in many technical applications such as heat exchangers, nuclear reactors and internal combustion engines. Flows in bends are characterised by strong secondary motions in terms of counter-rotating vortices (Dean cells) set up by a centrifugal instability. Specifically the thesis deals with turbulent flows in 90° curved pipes of circular cross-section with and without an additional motion, swirling or pulsatile, superposed on the primary flow. The aim of the present thesis is to study these complex flows in detail by using time-resolved stereoscopic particle image velocimetry to obtain the three-dimensional velocity field, with complementary hot-wire anemometry and laser Doppler velocimetry measurements.

In order to analyse the vortical flow field proper orthogonal decomposition (POD) is used. The so called “swirl-switching” is identified and it is shown that the vortices instantaneously, “rock” between three states, viz. a pair of symmetric vortices or a dominant clockwise or counter-clockwise Dean cell. The most energetic mode exhibits a single cell spanning the whole cross-section and “rolling” (counter-)clockwise in time. However, when a honeycomb is mounted at the inlet of the bend, the Dean vortices break down and there is strong indication that the “swirl-switching” is hindered.

When a swirling motion is superimposed on the incoming flow, the Dean vortices show a tendency to merge into a single cell with increasing swirl intensity. POD analysis show vortices which closely resemble the Dean cells, indicating that these structures co-exist with the swirling motion. In highly pulsating turbulent flow at the exit of a curved pipe, the vortical pattern is diminished or even eliminated during the acceleration phase and then re-established during the deceleration.

In order to investigate the effect of pulsations and curvature on the performance of a turbocharger turbine, highly pulsating turbulent flow through a sharp bend is fed into the turbine. Time-resolved pressure and mass-flow rate measurements show that the hysteresis loop in the pressure-ratio-mass-flow plane, may differ significantly between straight and curved inlets, however the mean operating point is only slightly affected.

Descriptors: Turbulence, curved pipes, swirling flow, pulsatile flow, time-resolved stereoscopic particle image velocimetry, hot-wire anemometry, proper orthogonal decomposition, turbocharger
Sammanfattning

Motivet bakom avhandlingen är en önskan att bättre förstå destörre strukturer som uppträder i turbulent strömning i krökta rör något som förekommer frekvent i olika tekniska sammanhang såsom i t.ex. värmeväxlar, reaktorer och förbränningsmotorer. Strömningen i en krökt rör karakteriseras av en mer eller mindre stark sekundär strömning i form av sekundära virvlar s.k. Deanvirvlar drivna av tryckfältet kopplade till de kröktas strömlijnorna. Arbetet fokuserar på turbulent strömning genom cirkulära 90° krökrökar med olika krökningsradiärer och strömning som både är stationär, pulserande och har olika grad av swirl i inloppet. Målet med avhandlingen är att analysera strömningsen i detalj genom att använda tidsupplöst stereoskopisk ”particle image velocimetry” (PIV) för att bestämma det tredimensionella hastighetsfältet. Hastighetsfältet bestäms också i vissa ögonblick m.h.a. varmträdsanemometri och ”laser-Doppler velocimetry” (LDV). För att analysera hastighetsfältet används ”proper orthogonal decomposition” (POD). Arbetet beskriver fenomenet ”swirl-switching” och det visas att virvlar växlar mellan två olika konfigurationer: två symmetriska Deanvirvlar, en dominerande medurs-virvel och en dominerande moturs-virvel. Konfigurationen med medurs- eller moturs-virvel som växlar i tid och som spänner över hela rörvärsnittet är den mest energerika. När inloppet till kröksen förses med en strömningsriktare (hexagonstruktur) undertrycks virvelstrukturen och det finns en stark indikation att ”swirl-switching” upphör.

När inloppströmmningen har ett rotationsbidrag kring strömningensaxeln ändras Deanvirvelstrukturen till en enda förstärkt virvel. POD visar virvlar som är mycket lika Deanvirvlar vilket indikerar att dessa strukturer samexisterar med den pålagda roterande strömningen. Vid kraftigt pulserande turbulent strömning minskar virvelintensiteten i rörkörens utlopp. Under accelerationsfasen elimineras virvlar med tiden för att under decelerationsfasen återupptas

För att undersöka hur en rörkök och pulserande strömning påverkar funktionen hos en turbin monterades en 90° rörköp genom inloppet till turbinen och systemet matades med ett pulserande turbulent flöde. Tidsupplösta massflödes- och tryckmätningar visar kurvor med hysteres i tryck-massflödesplanet vilka kan skilja sig markant mellan raka och krökt inlopp medan medelvärdet bara påverkas nåmnvärt.

Nyckelord: Turbulens, rörkök, roterande flöde, pulserande flöde, tidsupplöst stereoskopisk particle image velocimetry, varmträdsanemometri, proper orthogonal decomposition, turboladdare
Preface

This doctoral thesis in fluid mechanics deals with turbulent flows in curved pipes with special emphasis on the gas-exchange system of the internal combustion engine. The results in this thesis are from experimental work. The thesis is divided into two parts, with Part I including an introduction to the field and a literature review as well as a section where the experimental setups and techniques used in the present work are presented. Part I ends with a section where the results and conclusions from this study are summarised and a section where the respondent's contributions to the results are stated. Please note that the respondent changed name during this doctoral project, therefore “Kalpakli” and “Kalpakli Vester” refer to the same person. Part II consists of six papers, three of which are published but have been adjusted to comply with the present thesis' format for consistency. Their content, however, is unchanged except from minor corrections. This research project was initiated by Prof. P. Henrik Alfredsson who has acted as main supervisor, whereas Dr. Ramis Örlü and Dr. Nils Tillmark were co-advisors.

May 2014, Stockholm

Athanasia Kalpakli Vester

Related publications and presentations

Below is a list of related work that has been partially performed by the author during the PhD study, but it is not included in this thesis.


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Part I

Overview and summary
CHAPTER 1

Introduction

“The water just rushes out against the outer bank of the river at the bend and so washes the bank away [...] it allows deposition to occur at the inner bank [...] The question arose to me: Why does not the inner bank wear away more than the outer one?”

James Thomson (1822–1892)

Indeed, many of us might have observed a similar behaviour of the water flowing in a river when sitting close to a river bank, as described by Thomson (1876). The water rushes against the outer bank of the river whereas at the inner bank the velocity of water is much slower, allowing deposition of, for example granular material. This is a real situation that researchers and engineers have to face often, even nowadays\(^1\), when for example bank erosion in meandering rivers causes failure in constructions which are built near an outer river bank.

This is a common example from nature which marks the striking effects of streamline curvature on the motion of fluids. However, curved geometries—either open ones such as rivers or confined such as conduits\(^2\)—do not only exist in nature. Almost everything surrounding us is curved and how could it be otherwise, with the limited space we have been given to live in. From the circulatory systems of humans and other mammals that consist of curved veins, arteries and capillaries to technical applications such as the internal combustion engine with its branches and conduits, curved geometries are greatly involved in our everyday life.

On the other hand, the greek saying “\(\tau\alpha \pi\acute{a}v\tau\alpha \varphi\acute{e}\nu\)”, meaning that everything is constantly in κίνησις (kinesis) could not be more true. From the air

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\(^1\)For example, geologists and engineers in Farmington, Franklin County, Maine, USA are still making major efforts to stop the erosion caused by the outer bank of the Sandy river by using “root wads”. The erosion is causing failures on one part of the Whittier road placed at the outer river bank. Source: personal communication with Prof. Thomas Eastler, University of Maine, Farmington, Franklin County, Maine, USA.

\(^2\)A clear distinction needs to be made between those since the effects of curvature are more striking in open channels.

\(^3\)Heraclitus of Ephesus (c. 535–c. 475 BCE).
1. INTRODUCTION

around an aeroplane to the water in a river and air and fuel mixtures in engines, everything is flowing. To be able to understand fluid flows, is not always straightforward but gaining insight into the fluid dynamics of a flow is often of vital technological importance.

The present thesis, focuses on the dynamics of turbulent flow in curved pipes which is encountered in a number of technical applications such as heat exchangers, nuclear reactors, gas pipelines, internal combustion engines etc. Flows through bends are associated with strong secondary flow created due to unbalanced centrifugal forces, whereas turbulent flow has been described as unsteady, irregular, chaotic and unpredictable, which are the most difficult concepts in fluid dynamics. Statistical and modelling tools have been developed in order to predict and describe some of the aspects of turbulence but a complete description of turbulence is not possible, at least to date. A fact that makes this problem even more complex, is that sometimes turbulence might co-exist with a motion superimposed on the main flow. This can for example be a swirling motion (hydraulic plants, combustion chambers) or a pulsatile motion (internal combustion engine, blood flow). Those superimposed motions bear their own dynamical characteristics and together with the peculiarities of turbulence and flows in bends, lead to complex flow phenomena. As stated in literature: “[...] unsteady flows in curved conduits are considerably more complex than those in straight conduits and exhibit phenomena not yet fully understood” (Hamakiotes & Berger 1988), “[...] pulsatile flow through a curved tube can induce complicated secondary flow with flow reversals and is very difficult to analyse” (Kundu et al. 2012).

Within the present work, highly pulsatile and swirling turbulent flows through curved pipes are studied experimentally. Studying such flows numerically has been proven in the past to be problematic due to lack of accurate turbulence models and sensitivity to grid resolution (Pruvost et al. 2004; Hellström 2010). Numerical approaches—sometimes even direct numerical simulations—need validation against experiments (for example see the large-eddy simulations performed in Rütten et al. 2005; Fjällman et al. 2013). Therefore, the motivation of this study, is to gain a better understanding of the fluid dynamics of turbulent flows in curved pipes which are prevalent in almost any industrial process involving fluid motion and to provide a quality experimental database to the CFD community.
1.1. Towards increased engine efficiency: can fundamental research help?

The Internal Combustion (IC) engine is still the most common source for powering both light- and heavy-duty road vehicles. With the increasing cost and decreasing availability of fossil fuels as well as increasing concerns of green-house gases on the climate, focus has recently been set on increasing the efficiency of the IC engine without sacrificing performance. Similar concerns are relevant also for engines running on alternative fuels.

The gas-exchange system has a prominent role in the development towards more efficient engines, where downsizing is, at least for light-duty vehicles, the name of the game. The gas-exchange system should efficiently provide the intake of fresh air to the engine as well as utilising the energy (heat) in the exhaust gases, where an important, if not crucial, component is the turbocharger. However, the gas-exchange system is always a compromise between performance and what is possible from a packaging viewpoint, e.g., the piping system cannot be designed with straight pipes, the manifolds have complex geometry resulting in non-ideal flow profiles etc. The design of such systems is usually made with the aid of one-dimensional models although it is known \textit{a priori} that such models cannot give an accurate description of the flow dynamics. Testing in engine test benches together with empirical knowledge, rather than scientifically based experimentation, are also used to a large extent for developing the design. Although one should not downgrade the importance of the experienced engineer, the following statement is clearly valid: "\textit{The challenge of internal combustion require a broad collection of research discoveries to make the transition from hardware intensive, experienced based fuel development and engine design to simulation intensive, science-based design}" (Manley et al. 2008).

In the present work certain aspects of the gas-exchange system are approached from a basic scientific, rather than an engine application, viewpoint. Three specific aspects have been addressed, namely the turbulent flow through curved pipes, swirling turbulent curved pipe flow as well as highly pulsating flow through bends, all features that are apparent within the gas-exchange system. As already mentioned in the previous section, such aspects on flows in piping systems are not only dominant in internal combustion engines, but also in a number of other flow systems, and quite some efforts have been done earlier, but with other motivations, to investigate such conditions. On the other hand the parameter ranges for the IC engine flows are quite specific and it is therefore necessary to make studies for the relevant values of the parameters. The present studies have been performed through idealised experiments, and the fact that the quote from Manley et al. (2008) above states that one should strive towards simulation intensive methods, such methods also need
qualified boundary data and verification through quantitative scientific experiments. The aim of the present study is therefore to allow the reader mainly interested in IC engines \textit{per se} to realise that it is both important and rewarding to go outside the immediate neighbourhood of the engine aspects to get a better understanding of the physical processes important for engine performance.

1.2. Layout of thesis

The thesis is organised as follows: Part I continues with chapter 2 where a literature review is presented for flows in curved pipes with special focus on turbulent flow with and without superimposed pulsatile and swirling motion. Thereafter, in chapter 3, the experimental setups and techniques which were used during this project are described while in chapter 4 a summary of the most important results and authors’ contributions is made. Part I ends with a list of publications and conferences where some of the results have been presented as well as a comprehensive reference list. Part II of the thesis, contains the main results organised in the form of six papers. Two of them (of which one is a short paper) deal with turbulent curved pipe flow without any superimposed motion whereas one of them deals with swirling and two with pulsatile turbulent flow in curved pipes. The last paper is presenting results from measurements performed in a turbocharger rig with and without a pipe bend mounted at the inlet of the turbine.
CHAPTER 2

Turbulent flows in curved pipes

“Learn from yesterday, live for today, hope for tomorrow. The important thing is not to stop questioning.”

Albert Einstein (1879–1955)

The following chapter aims primarily to make the reader familiar with the subject of the present thesis i.e. flows in curved pipes, and secondarily to “learn from yesterday” and pose the questions which mark the need for new research within the area of curved pipe turbulent flows in the future. The chapter is divided into four parts, starting with a section where the basic characteristics of curved pipe flow are presented together with some historical facts. The second section refers explicitly to turbulent flow through bends, including an additional part on the behaviour of the secondary motion under turbulent flow conditions. The third and last sections deal with the case when a swirling and pulsatile motion is additionally superimposed on the main turbulent flow field, respectively. It should be noted that the intention of this chapter is not to make an extended summary of previous work on flows through curved pipes (this has already been done by the author in Kalpakli 2012) but rather make an assessment of the achievements made in the field of curved pipe flows, in earlier and recent years. In that respect, special focus is given on turbulent flow in spatially developing bends, e.g. 90° or U-bends, upstream and downstream of which, straight pipe sections are connected. The flow in a 90° bend is the case studied in this thesis, but some studies referring to other flow regimes (laminar, transitional) as well as flows in coiled pipes are mentioned for historical purposes.

2.1. “Learn from yesterday ...”

Early in the 20th century, Eustice (1910) observed that by even slightly bending a pipe, the resistance of the water flowing through it increases. He also introduced coloured liquid through capillary nozzles in bends (Eustice 1911) and visualised the streamline motion of the fluid. An uneven motion of the fluid was observed and the author compared it with the motion in straight pipes: “But in a curved pipe the water is continually changing its position with respect to the sides of the pipe, and the water which is flowing near the centre...”
at one part approaches the sides as it moves through the pipe and flowing near the sides it exerts a ‘scouring’ action on the pipe wall”.

Later, Dean (1927) noted that: “The motion of the fluid as a whole can be regarded as made up of what are roughly screw motions in opposite directions about these two circular stream-lines” referring to Fig. 2.1a. In this pioneering work, Dean (Fig. 2.1b) provided a first approximation to the laminar flow in an infinitely long curved pipe of circular cross-section. Even though this early study had weaknesses—limited agreement with the experiments by Eustice and no dependency between pressure gradient and curvature—it set the basis for other studies (Dean 1928; Dean & Hurst 1959) and contributed greatly to our understanding of flow through curved pipes. Dean introduced a new parameter for laminar flows through curved pipes of small curvature (for the exact definition see Dean 1928). This variable—known today as the Dean number—took various forms in different studies (White 1929; Taylor 1929; McConalogue & Srivastava 1968), causing some confusion in the interpretation of the literature (see a discussion on this matter in Berger et al. 1983). The Dean number in this thesis\(^1\) is defined—based on the bulk velocity which can easily be obtained through experiments—as:

\[
De = \sqrt{\gamma Re}, \tag{2.1}
\]

where \(Re\) denotes the Reynolds number equal to: \(Re = W_b D/\nu\), with \(W_b\) being the bulk velocity, \(D\) the pipe inner diameter and \(\nu\) the kinematic viscosity,

\(^1\)Even though the Dean number is used sometimes in the literature for turbulent flow (see for example Anwer & So 1993; Rütten et al. 2001) as well as in this thesis for comparative purposes, \(\gamma\) and \(Re\) should be viewed as independent parameters for turbulent flows.

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whereas the curvature ratio is $\gamma = R/R_c$ with $R$ the pipe radius ($= D/2$) and $R_c$ the radius of curvature of the centreline.

From an early stage (White 1929; Taylor 1929; McConalogue & Srivastava 1968; Greenspan 1973) it was appreciated that the flow through curved pipes is significantly more complex than the flow through straight pipes. If a fluid is moving along a pipe which is initially straight but after some point becomes curved, the bend will force the fluid particles to change their main direction of motion. The flow motion will not be parallel to the pipe axis anymore but secondary flow must be present, such as the one described by Dean (1927). A lateral force ($\sim W^2/R_c$) is experienced by the fluid particles as they move with a velocity $W$ along the curved path with radius $R_c$. The particles close to the pipe walls are moving with the lowest velocity due to no-slip condition, whereas the particles close to the pipe centreline move with the highest velocity and thus are subjected to the largest centrifugal force. This will cause the faster moving fluid near the centre to be swept towards the outer wall of the bend. This kind of flow cannot be regarded as a linear superposition of two separate motions but rather as an interaction between the streamwise and transverse velocity components which makes the study of such flows complicated.

Visualisations\(^2\) of how the secondary flow establishes itself at the exit of a 90° curved pipe of circular cross-section are shown in Fig. 2.2. The fluorescein dye is injected at the bend inlet through a probe placed approximately at the centre of the pipe. The Reynolds number based on pipe diameter ($D = 40$ mm) and bulk velocity is $Re = 1500$. A fluorescein dye is following, and at the same time, colouring the fluid motion, thus revealing the secondary flow. It is obvious that due to the fluid with the higher velocity close to the outer wall, the fluorescein starts its path from the outer wall creating a “C-shaped” pattern.

Depending on the applications, the flow in a pipe bend can be laminar, transitional or turbulent, whereas sometimes motions such as swirl or pulsations can be induced or superposed on the main flow field, adding new physics and parameters. For example, if the flow is turbulent, the Dean vortices do not stay symmetrical with respect to the symmetry plane but rather oscillate in time (Tunstall & Harvey 1968; Brücker 1998) whereas in a laminar pulsatile flow, the vortex pattern varies from single- to multiple-cell constellations during one pulsation cycle (see for example the studies by Timité et al. 2010; Jarrahi et al. 2010; Glenn et al. 2012).

A few textbooks including chapters on flows through bends, for the interested reader, such as the ones by Goldstein (1938); Schlichting (1979); Ward-Smith (1980); Pedley (1980); Kundu et al. (2012).

\(^2\)These experiments were performed during the author’s visit to Princeton University, Princeton, NJ, USA under the supervision of Prof. Alexander J. Smits.
2. TURBULENT FLOWS IN CURVED PIPES

Figure 2.2. Visualisations of the formation of the secondary motion at the exit of a 90° curved pipe at $Re = 1500$, corresponding to a bulk velocity of about 4 cm/sec. From left to right and from top to bottom: equidistant snapshots ($dt = 0.8$ sec) from the moment the fluorescence enters the cross-section until it fills up the measurement plane. Courtesy of A. Kalpakli-Vester, L. H. O. Hellström & A. J. Smits.
2.2. “... never stop questioning”

“I am an old man now, and when I die and go to heaven there are two matters on which I hope for enlightenment. One is quantum electrodynamics, and the other is the turbulent motion of fluids. And about the former I am rather optimistic.”

Horace Lamb (1849–1934)

Turbulence has been characterised from chaotic and random to a flow condition that “drives engineers nuts” (Vergano 2006). That is, as Vergano points out: “[...] because turbulence disrupts and drags air, gas and liquids that flow in and on everything from pipelines to airplane wings to artificial heart valves [...] in ways both costly and unpredictable.”

Even though it is rather obvious that in most engineering equipment involving fluid flow, turbulence co-exists with streamline curvature, much of the interest of scientists has long been focussed on turbulence in straight conduits. Turbulent curved pipe flow is encountered in many technical applications ranging from nuclear reactors (Ono et al. 2010) to reciprocating engines. Despite their direct practical relevance (Vashishth et al. 2008), such flows have not been investigated within the fluid dynamics community as much as turbulent flows in straight pipes—which of course are also of great importance. One of the difficulties that an experimentalist has to face when performing measurements in curved pipe flows is that all velocity components need to be obtained since the secondary flow might contaminate the data e.g. obtained with a single hot-wire. On the other hand, even though numerical work might be favoured, direct numerical simulations are computationally expensive and usually restricted to low Reynolds numbers, whereas turbulence models (including Reynolds-averaged Navier-Stokes) as well as large-eddy simulations are sensitive to grid resolution and have been found problematic in capturing the secondary flow in the past (see for example Hellström 2010; Fjällman et al. 2013), thus validation through comparison with experimental data is necessary.

2.2.1. The characteristics of turbulence in curved pipes?

The transition to turbulence is known to occur at higher Reynolds numbers in a curved pipe, compared to a straight one, as shown early by Taylor (1929). However, no universal critical Reynolds number for curved pipes has been found so far, as it exists for straight pipes. Some relations for the critical Reynolds number for curved pipes proposed by different studies can be found in Ward-Smith (1980) and Spedding & Benard (2004), whereas a more recent one can be found in Piazza & Ciofalo (2011). Much effort has also been done on determining the relation between wall friction and flow rate in turbulent curved pipe flows. One of the most well-known studies on this matter is the one by
Ito (1959) where the correlation between the friction factor and flow rate was deduced through a series of experiments.

Between the early 70’s and late 90’s, a series of studies were performed on mean quantities (velocity, pressure) and turbulence statistics along bends. Rowe (1970) measured the total pressure variation along a 180° bend with $Re = 2.36 \times 10^5$ and $\gamma = 0.042$. It was shown that the secondary motion is strongest 30° from the bend inlet, whereas for larger angles it reduced in strength until it reached a steady value at 90°. However, this work provided only information on the mean flow behaviour and nothing on the turbulence characteristics. Later, Patankar et al. (1975) used the $k - \epsilon$ model to simulate the experiments by Rowe and concluded that “[...] the turbulence modelling requires improvement”. This would not be the only time when turbulence models provided unsatisfactory agreement with experimental data for curved pipe flows (see e.g. Pruvost et al. 2004; Hellström 2010). Enayet et al. (1982) studied the mean streamwise velocity and corresponding turbulence intensity as well as static-pressure variation across a 90° bend by means of laser Doppler velocimetry (LDV) for $\gamma = 0.17$ and Reynolds numbers up to $Re = 4.3 \times 10^4$. The secondary flow was found to strongly depend on $Re$. Azzola et al. (1986) investigated the flow through a U-bend for $Re = 5.74 \times 10^4$ and $11 \times 10^4$ and $\gamma = 0.15$. The three velocity components were measured by means of LDV and the experimental results were compared with results from simulations (eddy-viscosity model). As the flow passes through the bend, the r.m.s of the longitudinal and circumferential velocities was found to increase due to the additional mean strain associated with the turning of the primary flow and the secondary-flow velocity gradients created. The agreement between numerical and experimental data was found to be somewhat acceptable by the authors. The largest disagreement between experimental and numerical data was found for the flow field downstream of the bend, even though secondary velocities are gradually decreasing in strength as the flow recovers from the bend.

In a straight duct with non-circular cross-section, it is known that streamwise vortices are formed due to local variation in Reynolds stresses and are known as Prandtl’s secondary flow of the second kind or “turbulence-driven” secondary flow (Bradshaw 1987)—whereas the secondary flow of first kind is induced by skewing of the mean flow in curved channels. For curved pipes, the existence of turbulence-driven secondary flow has been investigated to some extent. Experimental and numerical studies investigated the existence of a turbulence-driven cell, appearing within the bend, in conjunction with the Dean cells and a separation cell at the inner bend (Anwer et al. 1989; Azzola et al. 1986; Lai et al. 1991). For example, Lai et al. (1991) performed computations using a Reynolds stress model and showed the existence of three vortex pairs within a U-bend at $Re = 5 \times 10^4$ and $\gamma = 0.077$. The primary vortex pair was the Dean vortices which is driven by the centrifugal forces and is always present in a curved pipe. A second pair existed near the pipe core as a consequence
of local imbalance between the centrifugal force and the pressure gradient and last, a third pair of vortices was found near the outer bend embedded in the Dean vortices. This pair of vortices started to appear around 60° from the bend entrance, reached a maximum strength at the bend exit and disappeared at about 7D downstream distance from the bend. This additional pair of vortices was not found for developing laminar curved-pipe flows. Therefore, it was believed that this is a secondary flow driven due to the anisotropy of the turbulent normal stresses and their gradients. Those results agreed qualitative with experiments (Anwer et al. 1989; Azzola et al. 1986) but no quantitative comparison was made by the authors.

Hot-wire measurements of the three velocity components in a 90° bend at \( Re = 6 \times 10^4 \) and \( \gamma = 0.5 \) were performed by Sudo et al. (1998). They showed that at the inlet part up to 30° bend angle the high flow velocity was shifted towards the inner wall, and at 30° bend angle the Dean vortices were formed. Due to the action of the Dean vortices the high velocity fluid was moved towards the outer wall. Finally at the exit of the 90° bend, the main flow was non-uniform with the high velocity towards the outer wall and the low velocity towards the inner wall. The distribution of the streamwise velocity became smoother as the distance from the bend increased whereas the Dean vortices started to break down. However, the influence of the bend on the flow persisted even at 10D distance from the bend exit. The turbulence intensity increased in the outer part of the cross section between 0° and 30°, in response to an increase of the longitudinal velocity gradients in the radial direction. However, high values of turbulence intensity were found at the inner pipe wall at 60°. At 90° bend angle, the maxima of the turbulence intensity—a value of approximately 18%—was settled at the inner bend.

This section—although focusing on spatially developing bends—would be incomplete without mentioning some selected studies on infinitely long bends, including some recent efforts to study turbulent curved pipe flows by direct numerical simulations (DNS). However, the differences between the two geometries i.e. 90° bends and torus as well as the possibility that those might affect turbulence differently, should be kept in mind.

A first effort to employ large-eddy simulations (LES) considering an infinitely long bent pipe using periodic boundary conditions was done by Boersma & Nieuwstadt (1996). The authors motivated the study by the fact that up to that point, mainly data from single-point measurements were available with exception of some numerical studies, which however were restricted to small curvature ratios (Patankar et al. 1975; Lai et al. 1991). Results for the mean flow as well as r.m.s. fluctuations were presented for \( \gamma = 0.01 \) and 0.05. The simulations showed that the r.m.s. is enhanced at the outside of the bend and suppressed at the inside, whereas in contrast to straight pipe flow, the Reynolds stresses were found to be large at the centre of the pipe. However, this work
Turbulent flows in curved pipes could only be validated against experiments regarding the mean axial velocity profile, since no data to validate turbulence statistics were available.

A similar study was performed by Hüttl & Friedrich (2001) where turbulent statistics in an infinitely long bent pipe were studied by DNS. The Reynolds number based on the bulk velocity was given as $Re = 5.632 \times 10^3$, and the curvature ratio was $\gamma = 0.1$. The main conclusion from that work was that the turbulence intensity in a curved pipe is lower compared to a straight pipe. Fully developed turbulent flow in straight and curved pipes ($\gamma = 0.01$ and 0.1) at moderate Reynolds numbers ($Re = 5.3 \times 10^3$ and $11.7 \times 10^3$) was studied in Noorani et al. (2013) using DNS. The local axial wall shear stress was found to be significantly higher at the outer wall compared to the inner side. For the highest $Re$ and $\gamma$ case studied, the wall shear showed a plateau at the inner wall indicating relaminarisation, whereas distinct oscillations were found near the outer wall. The Dean vortices were observed for $\gamma = 0.01$ but for the higher curvature ratio case, the core of the vortices moved towards the side walls and a distinct bulge region appeared in the pipe centre. More recently, Di Liberto et al. (2013) investigated the turbulent flow in curved pipes for $\gamma = 0.1$ and 0.3 and for $Re = 1.5 \times 10^4$ and $1.2 \times 10^4$, respectively, also by DNS. The overall turbulence levels decreased for a curved pipe compared to the straight one, whereas increasing the curvature ratio led to a further reduction of the turbulence levels.

The aforementioned studies provided mainly mean velocity profiles and turbulence statistics. Even though they reveal some interesting flow phenomena in curved pipes, they address the need for more detailed data which will expand our understanding on how the instantaneous secondary flow is affected by different conditions i.e. curvature ratio, Reynolds number, upstream conditions etc.

2.2.2. “Swirl-switching” — rocking or rolling?

As it has already been mentioned in § 2.1, if the flow in the bend is turbulent, the Dean vortices are symmetric with respect to the symmetry plane only in a time-averaged sense, whereas instantaneously they exhibit an oscillatory character. Tunstall & Harvey (1968) (in the following referred to as TH68) were the first to show that the turbulent flow through sharp bends differs from the classical vortex pattern existing for laminar curved pipe flow. At first, they performed a series of flow visualisations in an elbow ($\gamma \to \infty$) and showed the existence of a separated region at the inner corner of the bend which was not symmetrical but displaced to either side of the plane of symmetry. At the same time a single vortex was switching (counter)clockwise in the azimuthal direction with a frequency corresponding to a Strouhal number (defined as: $St = f D/W_b$, where $f$ the frequency) of 0.002. A flag placed between golden-plated contacts was additionally mounted where the switching of the vortex was detected earlier. The output from a transducer placed somewhat upstream of
the flag also showed that the flow was dominated by two distinct states, an anti-clockwise and a clockwise state. Spectra of the axial velocity component and wall-pressure close to the outer and at the inner pipe wall showed the existence of two peaks at frequencies of 65 Hz and 120 Hz, corresponding to Strouhal numbers equal to 0.189 and 0.35, respectively. The mean frequency of the switching was found to correspond to \( St = 0.002 \), agreeing with the visualisations. It was suggested that this flow is bi-stable and the cause for this motion is the separation bubble at the inner bend which can change position along the pipe circumference and thus biasing the main flow in one half of the pipe. Furthermore, it was suggested that the origin of the switching is related to the turbulence of the flow upstream of the bend and that a sufficiently large perturbation is needed in order for the flow to switch. To account for Reynolds and curvature ratio effects, a series of measurements were performed for \( \gamma \) up to unity and a Reynolds number range between \( Re = 4 \times 10^4 \) and \( 22 \times 10^4 \). The switching occurred for all cases with the switching frequency increasing with the Reynolds number. The Strouhal numbers corresponding to the switching frequencies for the aforementioned \( Re \) range, were between below 0.001 and 0.004.

In Brücker (1998), two-dimensional, two-component (2D2C) particle image velocimetry (PIV) measurements were performed in a sharp bend (\( \gamma = 0.5 \)) for \( Re = 2 \times 10^3 \) and \( 5 \times 10^3 \). For the lowest Reynolds number the switching of the secondary flow as described by TH68 was not observed. However, for the higher Reynolds number case, at which the flow is expected to be turbulent, the Dean vortices were found to “rock” with respect to the plane of symmetry (Fig. 2.3) with a frequency corresponding to \( St = 0.12 \). Spectral peaks at lower (\( St = 0.03 \)) and higher (\( St = 0.2 \)) frequencies were also present. This supported that such oscillatory motion exists only for turbulent flows as suggested in TH68.

In more recent years, the interest in the behaviour of the Dean vortices in turbulent flows has increased successively, in the fluid dynamic community. Until the beginning of the 21st century, only the two aforementioned studies were available on the matter and the swirl-switching—with the one by Brücker (1998) being the only one providing quantitative visualisations of the phenomenon—was thought to be an abrupt domination of one of the two Dean vortices and to be related to the inner-corner separation region.

LES by Rütten et al. (2005) in a sharp (\( \gamma = 0.5 \)) and mild bend (\( \gamma = 0.17 \)) at \( Re = 2.7 \times 10^4 \) showed that the swirl-switching is not caused by the separation at the inner bend corner. Spectra of the forces onto the pipe walls showed distinct peaks for both bends even though separation was not observed for the mildly curved pipe. Spectral analysis was also performed on the time series of the position of the stagnation point to check whether a correlation exists between the forces on the walls, the stagnation point movement and the alternating domination of the Dean vortices. Both spectra from the forces and
the stagnation point showed low-frequency oscillations of $St = 0.01$. In an earlier work (Rütten et al. 2001), spectra on the same data for the bend with $\gamma = 0.5$ showed a low-frequency peak at $St = 0.0055$. That frequency was found to agree with the switching frequencies found in TH68 by extrapolating those measurements to the Reynolds number employed in the LES, however, this comparison is misleading since the Reynolds number was misinterpreted by Rütten et al. Higher frequency peaks at $St = 0.2$ and 0.3, were also present in the LES spectra which were interpreted as shear layer instabilities. It was additionally found in Rütten et al. (2005) that the stagnation point—and therefore the Dean vortices—was not abruptly switching between two stable positions (as suggested by TH68 and Brücker) but rather moving smoothly between angular positions within $\pm 40^\circ$.

Proper orthogonal decomposition (POD) applied on time-resolved stereo-scopic PIV (TS-PIV) data was used in Sakakibara et al. (2010) in order to

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure2.3.png}
\caption{Velocity field and contours of the streamwise vorticity for two moments within the cycle of the swirl switching. “S” marks the saddle point and the red line indicates roughly the plane of symmetry of the flow; the vorticity field is shown as contour-lines with constant difference, blue broken lines = negative value, red solid lines = positive value. Reprinted from Brücker (1998).}
\end{figure}
identify the switching motion of the Dean vortices. Data were acquired at 2D-15D distances downstream the bend exit for \( Re = 12 \times 10^4 \) and \( \gamma = 0.75 \). The swirl switching was reconstructed using the most energetic modes, that are the mode representing the mean and the first two modes related to the fluctuating part of the flow. It was shown that the most energetic mode related to the fluctuations in the flow at the \( 2D \) station, resembles the switching motion as described in TH68 i.e. a single vortex switching direction alternatively from anti-clockwise to clockwise rotation. However, this mode decayed as the flow evolved from \( 2D \) further downstream and became weaker than the mode which was the second most energetic at \( 2D \). This second mode was found to be responsible for a rotation of the plane of symmetry of the Dean cells as described in Brücker (1998). Furthermore, spectra of the second temporal mode at \( 2D \) showed a peak at \( St = 0.07 \).

In a follow-up paper, Sakakibara & Machida (2012) addressed the connection between upstream conditions and the swirl switching mechanism for \( Re = 2.7 \times 10^4 \) and \( \gamma = 0.5 \). The azimuthal displacement of the stagnation point was found to be correlated to streaks existing upstream of the bend with lengths of 7–8D. Those streaks were thought to be related to the very large-scale motions (VLSM, see for example Guala et al. 2006), though being much shorter in length. The authors concluded that the streaks are responsible for the oscillatory motion of the vortices, which opened new possibilities for explaining the origin of the swirl switching.

A similar effort as the one made in Sakakibara et al. (2010) i.e. to identify the origin of the swirl switching through POD analysis, was done in Hellsström et al. (2013). The Reynolds number in this study was \( Re = 2.5 \times 10^4 \), whereas the curvature ratio was 0.5. Flow fields at three downstream distances from the bend exit (5D, 12D and 18D) were acquired by means of TS-PIV. In that case the reconstruction of the swirl switching was done considering only the most energetic modes related to the fluctuating part of the flow i.e. the mode representing the mean was not included. This was based on the fact that in an earlier work (Hellsström et al. 2011) the Dean vortices were seen only in the mean vorticity maps and not in the instantaneous ones. Therefore, the authors concluded that the Dean vortices are a very weak feature in the flow field and they were, therefore, not considered in the reconstruction process in Hellsström et al. (2013). Additionally, the Dean vortices in the mean vorticity map were not symmetrical and as the authors remarked: “the mean vorticity [...] shows the tilted Dean motion as mentioned by Brücker (1998). The mean flow structures are, however, not evident among the much higher levels of instantaneous vorticity” (Hellsström et al. 2013). It should, however be kept in mind, that in any turbulent flow the levels of instantaneous vorticity will always be higher than in the mean. Furthermore, small turbulent scales which cannot be resolved due to the limited spatial resolution of the PIV, will appear as noise in the instantaneous vorticity maps. Therefore in order to obtain a
meaningful representation of the flow field, appropriate filtering of the data, in order to suppress the small scales, is important.

The first POD mode at all downstream stations, depicted a single vortex spanning the entire cross-section and rotating in the clockwise and anticlockwise direction as the flow evolved downstream the bend (Fig. 2.4). This mode was found to be associated with \( St = 0.33 \) through spectra of its temporal counterpart. The second and third mode depicted two Dean cells tilted with respect to the plane of symmetry with one of the cells being considerably suppressed. The second mode was associated with \( St = 0.16 \). The authors conclude that the tilting of the Dean cells is a transitional state between two states of the swirl switching, during which one of the cells is being suppressed resulting in a single vortex (shown as mode 1 in the POD analysis). The time scale of those transitions corresponded to structures of length 0.25–2.5\( D \). Finally, it was surmised that the suppression of one of the Dean cells was related to the shear flow region at the inner bend corner which is in return sensitive to upstream conditions, supporting the findings by Sakakibara & Machida (2012).

![Figure 2.4. First three POD modes for \( Re = 2.5 \times 10^4 \).](image)

\( (a, d, g) \) 5\( D \), \( (b, e, h) \) 12\( D \) and \( (c, f, i) \) 18\( D \); \( (a-c) \) Mode 1, \( (d-f) \) Mode 2 and \( (g-i) \) Mode 3. The inner and outer stagnation point is indicated with open and fill circles, respectively. *Reprinted from Hellström et al. (2013).*
2.3. SWIRLING FLOW IN CURVED PIPES

<table>
<thead>
<tr>
<th>Author (year)</th>
<th>Re × 10⁻⁴</th>
<th>γ</th>
<th>z/D</th>
<th>St</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tunstall &amp; Harvey (1968)</td>
<td>4.22</td>
<td>∞</td>
<td>2.2</td>
<td>0.002</td>
</tr>
<tr>
<td>Brücker (1998)</td>
<td>0.5</td>
<td>0.5</td>
<td>0.15</td>
<td>0.03, 0.12</td>
</tr>
<tr>
<td>Rütten et al. (2001)</td>
<td>2.7</td>
<td>0.5, 0.17</td>
<td>2.5</td>
<td>0.0055, 0.014, 0.2, 0.3</td>
</tr>
<tr>
<td>Rütten et al. (2005)</td>
<td>2.7</td>
<td>0.5, 0.17</td>
<td>2.5</td>
<td>0.01, 0.2, 0.3</td>
</tr>
<tr>
<td>Sakakibara et al. (2010)</td>
<td>12</td>
<td>0.75</td>
<td>2,10,15, 25</td>
<td>0.07</td>
</tr>
<tr>
<td>Sakakibara &amp; Machida (2012)</td>
<td>2.7</td>
<td>0.5</td>
<td>2.5</td>
<td>-</td>
</tr>
<tr>
<td>Hellström et al. (2011)</td>
<td>1.2, 1.8</td>
<td>0.5</td>
<td>5</td>
<td>-</td>
</tr>
<tr>
<td>Hellström et al. (2013)</td>
<td>2.5</td>
<td>0.5</td>
<td>5,12, 18</td>
<td>0.16, 0.33</td>
</tr>
</tbody>
</table>

Table 1. Previous studies on the “swirl-switching”. The Strouhal number associated with the frequency of the swirl-switching is shown in bold.

A list of the aforementioned studies dealing with the “swirl-switching” including some of the parameters under which they were conducted are summarised in Table 1. Although the number of studies that provide a characteristic frequency for the swirl-switching is limited and no definite conclusions can be made at this point, it is made clear that—with the exception of one study (Hellström et al. 2013)—the majority of frequencies correspond to low Strouhal numbers of the order 10⁻². Those studies underline the need for more investigations of the behaviour of the Dean vortices in a turbulent flow. For example, it is still not clear at this point if the swirl-switching consists of two different states i.e. one where the flow field is dominated by a single cell “rolling” (counter)clockwise in time and one where the Dean vortices “rock” with respect to the plane of symmetry, whereas there appears to be some ambiguity in literature when it comes to the frequency of those two motions. Furthermore, a relation between the switching mechanism and the separation region at the inner bend is implied in literature but is not conclusive. Finally, the suggestion made in Sakakibara & Machida (2012), that the displacement of the Dean cells is caused by elongated structures existing upstream from the bend, opens new possibilities for the control of the vortex switching and emphasises the need for more measurements on both sides of the bend.

2.3. Swirling flow in curved pipes

For the case where a swirling motion is superimposed on the main flow—or induced due to, for example, double bends (Yuki et al. 2011)—a Coriolis force acts on the fluid and the balance between centrifugal, inertial and viscous forces in the curved pipe changes. In cases where the swirling motion is introduced by imposing a secondary flow or by means of passive methods (guiding vanes) one needs to calculate an integral swirl number, defined as the ratio between the fluxes of angular momentum to streamwise momentum (Örlü 2009). However
in the present work the mean velocity both in axial and tangential direction are well defined by rotating the whole pipe around its streamwise axis (see § 3). The intensity of the imposed swirling motion can then be expressed by the swirl number which is defined as:

\[ S = \frac{V_w}{W_b}, \] (2.2)

where \( V_w \) is the rotational speed of the pipe wall.

The case of a turbulent swirling flow through a 180° curved pipe with \( \gamma = 0.077 \) and \( Re = 5 \times 10^4 \) was examined experimentally in Anwer & So (1993). The effects of swirl of intensity \( S = 1 \) on the secondary flow were examined through a comparison with non-swirling flow data acquired in the same setup. Wall-static pressure distributions between the swirling and non-swirling cases showed opposite results in the two cases: the static pressure at the outer bend was found to be lower than the inner for the swirling flow whereas the opposite was found for the non-swirling case. The total wall shear for \( S = 1 \) did not show differences between the outer and inner wall whereas for \( S = 0 \) there was a significant difference between the shear at the inner and outer wall due to the existence of the Dean cells in that case. From the mean velocity profiles along the horizontal planes, it was shown that the velocity distribution is more uniform and symmetric for the swirling flow case. The aforementioned observations suggested that a single dominating cell exists for \( S = 1 \) and that the curved-pipe flow becomes fully dominated by the imposed “solid-body” rotation\(^3\). An increase in turbulence production by the imposed swirling motion, resulted in more uniform distributions of the normal stresses in the radial and tangential directions. As a consequence, the radial and circumferential gradients of these stresses are reduced and do not provide enough vorticity, a fact which explains why the secondary flow created by the bend can not be sustained.

The follow-up work by So & Anwer (1993) showed that in swirling flow, the distance needed for the flow to become fully developed is shorter than that needed in the case of a straight pipe. Furthermore it was found that the bend accelerates the decay of the swirl compared to swirling flow in a straight pipe.

Pruvost et al. (2004) studied numerically the same flow case as in Anwer & So (1993) but extended the investigation for a range of swirl numbers between \( S = 0 \) and 4. An additional case for \( S = 5 \) was also studied but no experimental data were available for validation. Different numerical approaches were tested and the results were compared to the experimental data by Anwer & So both for the swirling and non-swirling case. However, all models gave poor agreement

\(^3\)Note that a solid-body rotation can only be obtained in laminar flow. For fully-developed turbulent flow, it has been shown, that the azimuthal velocity lags behind a velocity distribution corresponding to solid body rotation due to the influence of the cross-stream Reynolds stress (Facciolo et al. 2007).
with the experiments for the mean velocity profile and turbulent kinetic energy profile for $S = 0$. For $S = 1$ the low-$Re$ $k-\epsilon$ model gave satisfactory results for the mean velocity, but again the agreement with the experimental data for the turbulent energy profile was poor. Nevertheless, it was shown that there exists a complex interaction between the Dean vortices and the swirling motion for small $S$. As the swirling motion intensifies the Dean cells tend to merge in a single cell until the flow field becomes completely swirl dominated (Fig. 2.5).

The effects of swirl on the secondary flow field along a bend were also studied in Chang & Lee (2003) through 2D2C PIV measurements for $Re = 1 \times 10^4$, $1.5 \times 10^4$, $2 \times 10^4$ and $2.5 \times 10^4$. In that study the swirling motion was created in a swirl chamber connected to the setup and decayed along the bend. Therefore the conditions in Chang & Lee (2003) differ from the aforementioned studies and the present thesis and no direct comparison can be made. A two-cell pattern was observed at the entrance of the bend which diminished as the flow developed along the bend with a decrease in swirl intensity.

![Flow structures at increasing swirl intensities](image)

**Figure 2.5.** Flow structures at increasing swirl intensities ($S_n = S/2$). *Reprinted from Pruvost et al. (2004).*
As apparent from the aforementioned studies, the effects of a superimposed swirling motion on turbulent curved pipe flow has been studied only to a limited extent. Turbulent swirling flow is, however, encountered in many industrial applications such as in hydraulic plants, combustion chambers and any machine that involves a turbine or fan or a combination of bends e.g. double bends, and it is important to understand the physics underlying such mechanisms.

2.4. Pulsating motions in turbulent curved pipe flows ...

Pulsating (or pulsatile) flow, i.e. the unsteady flow composed of a periodic and a steady component, is typical in biological flows such as the respiratory and cardiovascular system. For instance, the heart is probably the most well-known pump in nature, distributing the blood to the whole body with a distinct frequency rate (of the order of 1 Hz). Therefore, the main body of unsteady flow research has been related to physiological flows (McDonald 1955; Chandran & Yearwood 1981; Glenn et al. 2012). However, pulsatile flow occurs in and plays a major role in the performance of mechanical systems such as the internal combustion engine. For a pulsating flow, transient inertial forces counteract the viscous forces. This is expressed by the Womersley number (Womersley 1957), which needs to be taken into account in pulsatile curved pipe flow, apart from the effect of curvature. The Womersley number is defined as:

\[ \alpha = R \sqrt{\frac{\omega}{\nu}} \]  

(2.3)

where \( \omega \) is the angular frequency of the pulsations equal to \( \omega = 2\pi f \) with \( f \) denoting the frequency of pulsations. As can be seen from eq. (2.3), \( \alpha \) is a composition of the Reynolds number and the Strouhal number. The latter can be seen as a ratio between the time scale inherent to the flow motion \( (D/W_b) \) and the time scale of oscillations \( (\omega^{-1}) \) as:

\[ St = \frac{\omega D}{W_b} \]  

(2.4)

For physiological flows, the Womersley number is small (of order one) as is the Reynolds numbers whereas in technical applications (e.g. internal combustion engine) those numbers can be of the order of \( 10^2, 10^5 \), respectively. According to Carpinioğlu & Gündoğdu (2001), for Womersley numbers smaller than approximately one, the flow can be considered quasi-steady since the turbulent structures have time to accommodate to the slowly varying flow rate. For \( 1 < \alpha < 30 \) the flow is intermediate (passage between steady and pulsatile flow) whereas for \( \alpha > 30 \) the frequency of the pulsatile motion is high enough.

\footnote{It should be noted that unsteady flows are divided into two categories: pulsating flow when the periodically time-averaged velocity is non-zero and oscillating flow in which the periodically time-averaged velocity is zero.}
2.4. PULSATING MOTIONS IN TURBULENT CURVED PIPE FLOWS ...

for turbulence not to be able to respond to the rapid changes in the flow and the flow becomes inertia dominated. Hence, turbulence becomes independent of the phase angle of the pulsations. This can be explained also with the aid of boundary-layer approximation, see for example the review in He & Jackson (2009) where the effects of high pulsations on turbulence are described as: “ [...] the shear waves propagating into the fluid from the wall [due to fact that higher vorticity needs to be generated to satisfy no-slip condition] [...] will be strongly attenuated and mainly confined to the viscous sub-layer region [...] the modulation of the velocity field varies across the region where the shear stress is being attenuated but becomes uniform in the region further out. [...] the inner and outer layers are completely decoupled [...]. Such a flow can be viewed as one in which the turbulence is 'frozen' ”.

Even though the above provide a general description of the effects of pulsations on turbulence, there are—to the author’s knowledge—no studies dealing with the effects of high pulsatile ($\alpha > 30$) flow on turbulent curved pipe flow with regard to the secondary flow evolution. As already mentioned in § 2.1, studies in laminar, low-frequency pulsatile flow showed that under one cycle, the secondary flow patterns vary due to competition between the centrifugal, inertial and viscous forces (e.g. Timité et al. 2010; Jarrahi et al. 2010; Glenn et al. 2012). There is no information, how these patterns would be affected if the flow is turbulent and highly pulsatile, as it is for example in the flow environment of the internal combustion engine.

2.4.1. ... and their effect on the turbocharger performance

With passenger cars contributing today to 12% of the total european emissions of carbon dioxide, EU emission legislations are gradually becoming stricter and engine manufacturers are called to develop technological solutions in order to reduce pollutant emissions and enhance fuel economy.

Different methods to increase the efficiency of automotive engines and reach the EU requirements for reducing toxic emissions, have been developed through the years such as exhaust gas recirculation (EGR) (Millo et al. 2008; Reifarth 2014) and turbocharging (Hellström & Fuchs 2008). The working principle of a turbocharger is that energy of the exhaust gas is extracted by expanding it through the turbine and the inlet air is compressed so that more air enters the cylinders during the intake stroke. Whereas turbocharging is not a new concept, the use of it together with new technologies such as downsizing—i.e. the

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6The invention of the turbocharger is credited to Alfred Büchi, a Swiss engineer who received a patent in 1905 for using a compressor driven by exhaust gasses to force air into an internal combustion engine to increase power output.
implementation of smaller engines providing the same power as larger engines—is, with regard to EU emission legislations, as stated in Marelli et al. (2014): “[...] turbocharging technique in conjunction with the downsizing concept seems to be the most promising way to achieve this target.”

Highly pulsating turbulent flow is created by the movement of the cylinder pistons and it is the flow condition in the intake and exhaust circuit of engines. The frequency of pulsations depends on the number of cylinders and rotational speed of the engine and can typically exceed 100 Hz (Olczyk 2009). As mentioned in § 2.4, for such high pulsatile frequencies—and therefore Womersley numbers—the flow can not be regarded as quasi-steady. Furthermore, as remarked in Hellström & Fuchs (2008): “In addition to effects of the pulsatile flow there is also a cross-sectional motion added to the axial flow velocity component at the inlet to the turbine. This secondary motion may consist of swirl, Dean vortices and other non-axial velocity components”. That is because, most—if not all—of the pipes comprising the gas intake and exhaust system are significantly curved. As already mentioned in the previous sections, such flow is complicated involving flow phenomena such as oscillating vortices, reverse flow and non-uniform velocity distributions. It is thus essential that such flows are investigated in detail for real-engine-operating conditions. However, as it has been described earlier in this chapter, the three-dimensionality of such flows as well as reverse flow, temperature variations during a pulse cycle and other peculiarities make the investigation of such flows difficult not only experimentally (see for example, Nabavi & Siddiqui 2010) but also numerically. On the other hand, there is more need for studies on generic geometries relevant to the engine flow environment, since on-engine tests not only limit the results in specific engine models but they also hinder the use of sophisticated and delicate experimental techniques (such as particle image velocimetry), which are necessary if detailed information on the flow is desired.

Under inlet pulsating flow conditions, the turbine is known to yield a “hysteresis” type performance characteristic, i.e. the mass-flow versus pressure-ratio curve follows a so-called “hysteresis” loop surrounding the steady state curve (see e.g. Capobianco & Marelli 2009). Turbine maps supplied by the manufacturers are, however, based on steady flow conditions covering, at the same time, only a small area of operational points. Furthermore, it is common practice that turbine performance is modelled under pulsating flow conditions using 1-D simulations (Chen et al. 1996; Piscaglia et al. 2007; Tabatabaei et al. 2012; De Bellis et al. 2014; Chiong et al. 2014). However those have been proven to be insufficient and experimental data are needed for their improvement (Chiong et al. 2012), whereas if simulations need to be performed, 3D unsteady flow simulations are necessary (Tabatabaei et al. 2012)—even though computationally expensive—in order to model three-dimensional phenomena at the inlet of the turbine. As remarked in Baines (2009): “[...] A more rigorous engine
2.4. PULSATING MOTIONS IN TURBULENT CURVED PIPE FLOWS

Simulation will require a dynamic approach to turbine modelling, in place of current steady-state map interpolation and extrapolation."

Szymko et al. (2005) introduced a modified Strouhal number, that incorporated also the pulse length, and based on that, three regimes in the mass-flow versus pressure-ratio-map were identified: quasi-steady, steady encapsulated hysteresis and fully unsteady. That could be useful as a rule of thumb when the quasi-steady-assumption for modelling is valid. The velocity field within the turbine was found—through 3D unsteady flow simulations—to be highly distorted by the pulsations at the inlet in Palfreyman & Martinez-Botas (2005), leading to substantial fluctuation to the incidence angle indicating poor flow guidance throughout the pulse period. A reduction of turbine efficiency for a pulsating inlet flow was found in Marelli & Capobianco (2011), through instantaneous pressure and mass-flow measurements, supporting that the steady flow characteristic curves do not represent realistically the turbine behaviour. Also a quasi-steady approach appeared inadequate to predict the unsteady flow operation of the turbine. Tabatabaei et al. (2012) performed experiments and 3D unsteady simulations on the inlet conditions of an SI-engine turbocharger turbine and found a difference of almost 10% in pressure ratio in turbine maps produced under steady and unsteady operating conditions, obtaining higher pressure ratios for the steady flow case. The “hysteresis loop” was found to widen and shift away from the quasi-steady operation with increase in pulsatile frequency in Copeland et al. (2012). Padzillah et al. (2014) showed that there is a potential loss of information on the turbine efficiency if the performance parameters, such as the incidence angle at the leading edge of the turbine rotor, are averaged without considering the instantaneous effects.

The aforementioned studies constitute an important database for engine flows and provide useful information on the effect of pulsations on turbine performance, with results induced mainly from simulations and single-point measurements. There is, however lack of a detailed description of the flow field entering the turbine, that is taking into account not only the effect of pulsations but also streamline curvature. That means that single-point experimental techniques are not ideal for such studies, since the flow field entering the turbine is three-dimensional and all velocity components need to be measured, thus whole field measurement methods need to be implemented. A study which combines a detailed description of the three-dimensional flow field and the effects of bends on turbine performance is—to the author’s knowledge—missing from the literature. If the target is to reach emission legislation requirements by increasing the performance of the turbocharger and enhancing fuel economy, it is very important that, first, a better understanding of the performance of the turbocharger is established. In order to succeed that, flow phenomena occurring at the inlet of the turbine, such as the ones described in § 2.1–2.4, need to be fully integrated into the design stage so that correct performance maps can be obtained.
CHAPTER 3

Experimental setups and techniques

“A scientist in his laboratory is not a mere technician: he is also a child confronting natural phenomena that impress him as though they were fairytales.”

Marie Curie (1867–1955)

In the following chapter the experimental setups and techniques that have been used for the purposes of the current study are presented. Two experimental setups have been used. The first one is part of the rotating-pipe facility at KTH Mechanics and consists of a long axially rotating-pipe to create a swirling motion before the flow is fed into a pipe bend. The second one, is located in the CICERO laboratory at KTH CCGEx (Competence Centre for Gas Exchange), where pulsatile flow can be created through a rotating valve located upstream of the pipe test section.

Due to the complexity of turbulent curved pipe flow, different techniques had to be used in order to fully investigate the flow field both in terms of statistical quantities and large-scale structures. Therefore, time-resolved stereoscopic particle image velocimetry (TS-PIV) was employed with the aim to both visualise and quantify the coherent structures existing at the exit of the bend, whereas hot-wire anemometry (HWA) was employed due to its high frequency response and temporal resolution in order statistically analyse the flow field. Finally, laser-Doppler velocimetry (LDV) has been applied for further investigation of some of the results from the two aforementioned techniques. The principles of the three techniques and how they have been applied in the present study are explained in detail in the following sections.

3.1. The rotating-pipe facility

The turbulent flow field downstream a curved pipe with and without a superimposed swirling motion was studied by means of TS-PIV measurements at the rotating-pipe facility in the Fluid Physics Laboratory at KTH Mechanics (see papers 2 & 3). In the following paragraphs, some basic information about the facility will be provided. For a more detailed description of the apparatus the reader is referred to Facciolo (2006).
3.1. THE ROTATING-PIPE FACILITY

Figure 3.1 shows the facility schematically. The air is provided by a centrifugal fan and the mass flow rate can be controlled by means of a butterfly valve monitored through the pressure drop across an orifice plate. A distribution chamber is installed in order to minimise the vibrations created by the fan while a honeycomb installed in a stagnation chamber, distributes the air evenly. The air is first led into a one meter long stationary pipe section (the diameter of the pipe is 60 mm) which is connected to the rotating pipe. A 12 cm long honeycomb is located at the inlet of the rotating pipe and brings the flow into more or less solid body rotation. The pipe can rotate around its streamwise axis to speeds up to 2000 rpm by means of a DC motor which is connected to the pipe through a belt. The total length of the rotating pipe section is $100D$.

Figure 3.2 shows the mean velocity profiles at the exit of the pipe for different swirl intensities at a Reynolds number based on the pipe diameter of $Re = 2.4 \times 10^4$. The profile for the non-swirling case depicts a fully-developed turbulent flow profile while as the swirl number increases, the profile shape approaches that of a laminar one (Sattarzadeh 2011).

At the exit of the $100D$ long straight pipe section a curved pipe was mounted (see Fig. 3.3). The curved pipe has an inner diameter of $D = 60$ mm and curvature radius of $R_c = 95.3$ mm, yielding a curvature ratio, $\gamma$ of around

![Figure 3.1. Schematic of the rotating-pipe facility. A) Centrifugal fan, B) flow meter, C) electrical heater, D) distribution chamber, E) stagnation chamber, F) coupling between stationary and rotating pipe, G) honeycomb, H) DC motor, J) ball bearings, K) rotating pipe, L) circular end plate, M) pipe outlet.](image)
Figure 3.2. Mean velocity profiles at the exit of the 100$D$ pipe (see point M in Fig. 3.1) for $Re = 2.4 \times 10^4$ and for different swirl numbers ($S = 0, 0.1, 0.3, 0.5$). Reprinted from Sattarzadeh (2011).

Figure 3.3. a) The curved pipe which was mounted at the outlet of the 100$D$ long straight pipe (M in Fig. 3.1). b) Dimensions of the curved pipe.

0.31. The length of the straight section after the 90° bend is 0.67$D$. Note that while the straight pipe is rotating, the bend is remaining still.

Due to technical restrictions, all measurements were taken at the immediate vicinity of the bend exit i.e. approximately 0.5 mm from the outlet of the pipe. Even though this is not common practise, it was found to provide satisfactory results for the purposes of this study, after comparison with available hot-wire data acquired in the same setup but inside the pipe (see paper 3).
3.2. The CICERO rig

Turbulent highly pulsatile flow downstream a curved pipe was studied at the CICERO Laboratory at KTH CGEx, see papers 1, 4 & 5. A compressor installation facility with two Ingersoll Rand screw compressors (Laurantzon et al. 2010b, 2012b) can deliver up to 500 g/s air flow at 6 bar. The CICERO rig can be operated under both steady and pulsatile flow conditions. The mass flow rate is being monitored by a hot-film type mass flow meter (ABB Thermal Mass Flowmeter FMT500-IG) which is located around 10 m upstream from the measurement site. The pulsations are supplied by a rotating valve, consisting of a sphere with a tight fitting in a 55 mm pipe, which is located upstream of the pipe test section. The sphere is cut off at two sides, thereby the valve opens twice per revolution. The rotation rate of the valve can be set by a frequency-controlled AC motor and the maximum open area is approximately 15% of the pipe area (see Fig. 3.4).

The total entrance length before the flow is fed into a pipe bend was $20D$ and $80D$ for two experimental campaigns, respectively. Three pipe bends with different curvature ratios have been used for the experiments performed in the CICERO laboratory and their geometrical details are listed in Table 2.

![Figure 3.4](image)

**Figure 3.4.** a) The rotating valve with its sphere and housing. Reprinted from Laurantzon et al. (2010b). b) The projected open valve area as function of the revolution angle.
In order to study the effect of steady and pulsatile flow through a sharp curved pipe on the turbine map of a turbocharger an additional set of measurements was performed (see paper 6 for more details) with the pipe denoted as **Bend II** mounted upstream of a Garrett turbocharger. Figure 3.5 shows the experimental configuration used for these measurements. It has been designed, built and taken into operation in conjunction with the work by Laurantzon et al. (2012a). Instantaneous pressure and mass flow rate measurements were performed across the turbocharger by means of fast response pressure transducers (Kistler) and a vortex mass flow meter for unsteady flow measurements described in Laurantzon et al. (2010a), respectively.

**Table 2.** Geometrical details of the pipe bends used in the Cicero Laboratory. The diameter of the pipe \((D)\), the curvature radius to the pipe diameter \((R_c/D)\) and the downstream distance from the bend exit to the pipe diameter \((z/D)\) are shown.

<table>
<thead>
<tr>
<th></th>
<th>(D) [mm]</th>
<th>(R_c/D)</th>
<th>(z/D)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bend I</td>
<td>39</td>
<td>1.17</td>
<td>0.8</td>
</tr>
<tr>
<td>Bend II</td>
<td>40.5</td>
<td>1.3</td>
<td>0.2, 1, 2, 3</td>
</tr>
<tr>
<td>Bend III</td>
<td>40.5</td>
<td>3.6</td>
<td>1.2, 3</td>
</tr>
</tbody>
</table>

**Figure 3.5.** Setup for the instantaneous pressure and mass flow rate measurements across the turbocharger (Garrett) showing the sharp bend mounted at the inlet of the turbine.
3.3. Time-resolved stereoscopic particle image velocimetry

“A man is not a dog to smell out each individual track, he is a man to see, and seeing, to analyse. He is a sight tracker with each of the other senses in adjunctive roles. Further, man is a scanner, not a mere looker. A single point has little meaning unless taken with other points and many points at different times are little better. He needs the whole field, the wide view.”

Prof. F. M. N. Brown, University of Notre Dame

The measurement technique that has been used in order to obtain the main part of the results presented in this thesis is time-resolved stereoscopic particle image velocimetry (TS-PIV). It was chosen as the main experimental technique to investigate the three-dimensional turbulent flow field downstream pipe bends. PIV is capable of providing simultaneously the three-dimensional flow field with a reasonable temporal and spatial resolution (sufficient for the purposes of the present study i.e. resolve and track in time the large-scale structures in the flow field), it is non-intrusive and it is quite robust which is important when measurements in harsh flow environments (such as high pulsations) take place. It would therefore be time consuming, complicated and also restricting the results to statistical information and integral quantities if a single-point technique was used instead.

The principle of particle image velocimetry is rather simple. The flow is seeded with particles which should follow the flow as realistically as possible. The relaxation time of the particles is given by: \[ \tau_s = \frac{D_p^2}{18 \mu} \left( \frac{\rho_p}{\rho} \right) \], where subscript \( p \) denotes particle quantities, whereas \( \rho \) is the density and \( \mu \) the dynamic viscosity of the fluid, respectively. The Stokes number, \( Sk \) gives an estimate of the tracking ability of the particles and is the ratio of the relaxation time of the particle to the flow time scale (see for example Samimy & Lele 1991). Ideally, \( Sk < 0.1 \), in order for the particles to follow the flow closely.

The measurement plane is illuminated twice using a double-pulsed laser\(^1\) in a short time interval and the light scattered by the particles is recorded on double-frame\(^2\) via a camera which is focussed on measurement plane. If all three velocity components are to be measured, as in the flow case described in the present thesis, two cameras need to be employed in a stereoscopic configuration. In that way, distinct off-axis views of the same region can provide the out-of-plane motion of the particles through a reconstruction scheme (Prasad 2000). Finally, the acquired images are divided into small areas (Interrogation Areas,

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\(^1\)For example Nd:YAG or Nd:YLF lasers are broadly used nowadays with repetition rates of the order \( O(10) \) & \( O(10^3) \) Hz, respectively.

\(^2\)Here we describe only the method used within the context of this work which is the double-frame/single-exposure technique which preserves the temporal order of the PIV recordings and is usually preferred if the appropriate equipment is available.
IA) and the local displacement of the particles in the IA is determined by means of cross-correlation, i.e. the peak of the cross-correlation is used to determine the average particle displacement from one frame to the other. For more details on the fundamentals of PIV the reader is referred to the textbooks by Raffel et al. (2007) and Adrian & Westerweel (2010).

The TS-PIV system used for the purposes of this study was supplied by LaVision GmbH and consists of two high-speed CMOS cameras (Fastcam APX RS, Photron, 3 kHz at full resolution of 1024 × 1024 pixel, 10 bit dynamic range) and a Nd:YLF laser with 10 kHz repetition rate (Pegasus laser, New Wave Research). The cameras were usually positioned at angles between 70-90 degrees between their viewing axis at forward-backward scatter mode. The lenses mounted on the cameras were Nikon Nikkor lenses with 105 mm focal length and aperture range from $f/1.8$ to $f/22$. For each camera a Scheimpflug adapter was employed in such a way that the image, lens and object plane intersect in a common line (Prasad & Jensen 1995). Before each new recording, dark images with the lenses covered were acquired from both cameras in order to subtract the dark noise (or dark current) of the CMOS cameras which is usually in the order of a few counts and increases with time.

The calibration of the cameras was done using either a two-level calibration plate provided by LaVision GmbH or a home-made calibration target resembling the one shown in van Doornel & Westerweel (2007). The latter was translated 0.5 mm in the streamwise direction using an accurately machined ring in order to provide images at two different planes for the reconstruction of the out-of-plane velocity component. To eliminate the errors arising from any misalignment between the light sheet and the calibration plate, a self-calibration scheme was applied on the recordings as described in Wiencke (2005). The flow was seeded using either standard olive oil or a water-based solution (Jem Pro smoke Super ZR-Mix) atomised through a high volume liquid seeding generator (10F03 seeding generator, Dantec Dynamics).

Depending on the quality of the particle images (which depends to a great extent on the kind of flow being measured, setup and technical specifications), different post-processing schemes need to be applied on the acquired snapshots in order to obtain meaningful and accurate vector fields. Sources that will introduce errors are typically large velocity gradients in the flow field or particles which move outside the IA between two consequent frames. A general rule exists in order to avoid the latter, i.e. choose the time between the two illuminations such that the particle displacement is between the accuracy of the system and $1/4$ of the IA. The flow field studied at the exit of a curved pipe, is non-uniform and it is characterised by high velocities towards the outer pipe wall and low velocities towards the inner wall. The difference between the highest and lowest mean velocity in the flow field is typically $0.9W_b$, where $W_b$ is the bulk velocity. Therefore the time between the two illuminations ($dt$) had to be chosen carefully in order to obtain a good signal-to-noise ratio (S/N) of
3.3. TIME-RESOLVED STEREOSCOPIC PARTICLE IMAGE VELOCIMETRY

the correlation peak. The \( dt \) for each measurement was typically of the order of few microseconds and it was set after trial-and-error until a good S/N ratio of the correlation peak was obtained. The acquisition and evaluation of the particle images were performed via a commercial software, DaVis 7.2 from LaVision GmbH. An “online”, i.e. at the time of the image acquisition, evaluation of the particle images’ quality as well as the vector fields is offered by the specific software which has been used in order to select the optimal measurement settings.

The quality of the “raw” images was good since no optical or other kind of obstacle existed between the object and image plane and no reflections were present. However, unfocussed particles due to the much higher velocities close to the outer pipe wall (compared to the inner wall) were unavoidable and this resulted in small intensity fluctuations across the images. In order to improve the quality of the results, a particle intensity correction was applied on the original snapshots by means of a min/max filter with a \( 5 \times 5 \) px window size.

The spatial resolution and accuracy of the obtained vector-field depend on a number of parameters such as the size of the IA, the thickness of the light sheet, the resolution of the camera sensor, the correlation scheme etc. It is common practise to repeat the correlation process for decreasing IA sizes and deforming windows (Scarano 2002) in order to maintain a good spatial resolution but also high accuracy. Here, an iterative cross-correlation procedure was applied with decreasing and deforming interrogation windows with 50% overlap, after which the vector fields were smoothed with a \( 3 \times 3 \) averaging filter. The final smallest IA size was chosen based on the mean statistics of the obtained velocity fields after different IA sizes (ranging from \( 64 \times 64 \) px to \( 8 \times 8 \) px) were tested during the cross-correlation procedure. Whereas no big differences were seen for the mean velocity field when comparing vector fields obtained with different IA sizes, with \( 32 \times 32 \) px giving the same results with \( 16 \times 16 \) px, the root-mean-square (RMS) of the streamwise velocity component was very sensitive in the selection of the IA size. The so-called bias error “peak-locking” (see for example Angele & Muhammad-Klingmann 2005), which was shown in Christensen (2004) to affect statistical moments such as the RMS, was not found to be an issue in the present data. Following the guidelines in Overmars et al. (2010) the degree of peak-locking in the data was in all cases less than 0.3 (with a value of 1 and a value of 0 denoting strong or no peak-locking, respectively). It should be noted that Reynolds number effects were not studied with respect to higher order moments and wherever RMS results are shown (see paper 4) it is only at the same Reynolds number and of course same spatial resolution. The resulting data resolution (around 2 vectors/mm\(^2\)) was found sufficient in order to resolve large-scale structures (which is the aim of the present work) but the smallest dissipative scales (\( \eta \sim 0.085 \) mm) were not resolved (see for example paper 3).
Outliers, i.e. data deviating strongly from neighbouring data, were detected and removed by means of a median test (Westerweel 1994). The percentage of accepted vectors was in all cases more than 90% and the spurious vectors could be replaced through a linear interpolation scheme. It must be noted here, that data validation was performed before any post-processing on the images was applied, through a single-pass cross-correlation scheme with an IA size of $32 \times 32$ px.

3.3.1. Processing of instantaneous vector fields

One of the most challenging tasks for the user of PIV is to analyse the information obtained from PIV measurements. As noted in Raffel et al. (2007): “PIV being the first technique to offer information about complete instantaneous velocity vector fields allows new insights in old and new problems of fluid mechanics”. The development of PIV revived the interest of fluid dynamicists in existing analysis tools for the identification of turbulent coherent structures, such as proper orthogonal decomposition (POD, see for example Lumley 1971). Nowadays, POD is a standard analysis tool for PIV data being part of commercial PIV softwares (e.g. DaVis by LaVision GmbH & Dynamic Studio by Dantec Dynamics).

Through POD the most significant structures in the flow field can be identified and ranked by energy content (Holmes et al. 2012). It has been used extensively for the analysis of vector fields within the context of the present thesis in order to reveal the large-scale structures. For a detailed description on POD, its mathematical background as well as its implementation in MATLAB the reader is referred to Chatterjee (2000). Other works on POD which have been used as guidelines in order to implement POD in the present work are the ones by Meyer et al. (2007) and Semeraro et al. (2012), where a convergence test of the POD modes is presented, among others. Here, a short description on the theory of POD will be given as follows.

For an existing ensemble of velocity fields: $u_1, u_2, \ldots, u_N$, where $u_N$ the $n$-th snapshot, all velocity components for each snapshot are reorganised in a matrix $U$ with dimensions $M \times N$. The $M$ dimension contains the spatial information whereas the temporal information is stored in $N$. The eigenvalue problem of the covariance matrix $U^T U$ is then considered. The eigenvalues are real and positive and represent the energy contained in the corresponding eigenfunctions or modes which are orthogonal and real-valued. Hence, if those are ranked in descending order, the first modes are the most important physically, with the first mode representing the mean field. In flow fields with dominating flow structures, only few of the POD modes are needed to reconstruct and obtain an optimal representation of the initial data (Berkooz et al. 1993).
3.4. Hot-wire anemometry

Hot-wire anemometry (HWA) was implemented at the start of this study (paper 1) in order to obtain a good statistical representation of the flow under focus due to its higher temporal resolution and frequency response compared to TS-PIV. Hot-wire data by Sattarzadeh (2011) have also been used in order to compare and address the pros and cons from implementing HWA and TS-PIV in turbulent curved pipe flows (paper 3).

The use of a heated wire with temperature dependent resistance exposed in air flow to measure the fluid velocity is the basic principle of HWA. It is a relatively cheap, easy-to-use technique with high frequency response and has greatly contributed to our understanding of turbulence, probably more than any other technique, taking into account the long time (almost 100 years) it has been available for the measurement of turbulent flows.

The hot-wire is simply made by a probe which holds two metal cylinders, so-called prongs, on which the sensor element is soldered or welded. Typically, a wire element made of tungsten or platinum with a length-to-diameter ratio of 200 is used. In the present study, a tungsten wire with a diameter of 5 \( \mu m \) and a length of 1 mm was used. The probe was manufactured at the Fluid Physics Laboratory at KTH Mechanics at the available “hot-wire corner” (for the reader who is interested on this facility, the fragile process of manufacturing a probe holder and soldering a hot-wire at KTH Mechanics, is documented in Ferro 2012).

The hot-wire is connected to a Wheatstone bridge and for the purposes of this study it was operated in the constant temperature (CTA) mode, where the wire is kept, as the name implies, under constant temperature by means of a servo amplifier. This keeps the bridge in balance by controlling the current to the sensor and keeps the resistance constant, independent of the cooling of the wire element due to the fluid flow. The change in voltage gives a measure of the flow velocity through the heat transfer. The interested reader is referred to the textbooks by Perry (1982) and Bruun (1995) for more information on the principles of HWA.

The single hot-wire used in this study, was operated by means of a DISA 55M01 main frame with a 55M10 standard CTA channel. The calibration of the wire was done ex-situ in a calibration nozzle facility against a differential pressure transducer (MPX2010DP, Freescale Semiconductor Inc., 0-10 kPa). Due to instantaneous velocities exceeding the range of the pressure transducer, a modified King’s law relation (Johansson & Alfredsson 1982) was preferred for the calibration relation—instead of a polynomial calibration function—which enables extrapolation beyond the calibration range. A thermocouple was incorporated in order to measure possible temperature variations inside the stagnation chamber of the calibration unit.
A known drawback of HWA is the so-called “forward-reverse” ambiguity (Bruun 1995) i.e. it cannot sense the direction of the velocity. This makes it a non-ideal experimental technique for the measurement of pulsatile flows (Berson et al. 2010), where reverse flow is a common feature. Additionally, the flow under pulsatile conditions is non-isothermal (Laurantzon et al. 2012b) and in order to obtain the temperature variations in the flow the wire (called cold-wire in such case) needs to be operated in constant current mode. In such a case the length-to-diameter ratio of the wire needs to be three times larger compared to a wire operated in CTA in order to achieve a good frequency response. Therefore, the wire becomes even more fragile. Although one must be aware of these limitations of the hot-wire technique in pulsating flows, it is still possible to obtain a representative description of the flow behaviour (e.g. paper 1).

Secondary flows in curved pipes are additionally present and contaminate the hot-wire readings as well. However, based on the concept of effective velocity, the secondary components will not assert a too strong effect on the readings of a single hot-wire probe (Bruun 1995); see for example paper 3 where data obtained by means of hot-wire and particle image velocimetry in a turbulent curved pipe flow are compared and show a good agreement for the mean horizontal velocity profile.

3.5. Laser-Doppler Velocimetry

Laser-Doppler velocimetry (LDV) is a single-point, non-intrusive experimental technique which, since its first appearance in the 60’s (Yeh & Cummins 1964), is often used to measure the flow velocity. For the interested reader, a textbook on LDV is the one by Albrecht et al. (2003).

Two beams—obtained by splitting a single beam—of monochromatic and coherent laser light intersect at a point in the measurement volume. A transmitting optics focuses the beams at their waists where they interfere and create a fringe pattern. Particles which have been added in the flow, pass through the fringes and scatter light. This is then received by a photodetector. The reflected light from each particle fluctuates in intensity and the frequency of those fluctuations is equivalent to the Doppler shift between the incident and scattered light, and is thus proportional to the velocity of the particle.

In the present study, LDV experiments were conducted as a complement to either hot-wire (see paper 1) or TS-PIV (see paper 4) data due to its ability to sense backflow and obtain time series with higher temporal resolution than PIV, respectively. The measurements were performed with a single component Dantec FlowLite system in conjunction to a BSA 60 processor. The emitting light source is a 10 mW He-Ne laser with wavelength of 632.8 nm. The lens mounted on the laser has a focal length of 400 mm. The liquid used for seeding was either olive oil or a Shell Ondina 27 oil.
CHAPTER 4

Main contribution and conclusions

In this chapter the main contributions and conclusions from the papers presented in Part II are given (A–G). For details on the results the reader is referred to the appended papers.

A. Highly pulsating turbulent flow downstream a straight and bent pipe—statistical analysis
(Paper 1)

- Highly pulsating ($\alpha = 80$) turbulent ($Re = 2.4 \times 10^5$) flow at $1D$ distance downstream a pipe bend has been examined by means of a single hot-wire probe traversed along the radial symmetry axis of a straight and bent pipe. The effects of the pulsatile motion on turbulent flow were investigated by means of statistical and singular value decomposition (SVD) analysis.
- The mean velocity profile was shown not to be significantly affected by the pulsations (compared to the steady flow case) while the r.m.s. being dominated by the pulsations. This was furthermore supported by a trimodal probability density function (PDF) distribution caused by the pulsations.
- Back flow was encountered, indicated from the weighted PDF distributions towards zero but also from the phase-averaged signal at the pipe centreline where a mirrored “dimple” is depicted in the hot-wire signal. Additional LDV measurements in the straight pipe were performed and back flow of a magnitude of almost 50% of the bulk velocity was substantiated.
- The pulsatile flow was decomposed into its periodic and turbulent parts by means of SVD and a high- and low-pass filter. It was shown that the pulsatile motion is superposed on the turbulence with the high-pass filtered/reconstructed signal using the fluctuating part of the flow (from the SVD), being not only qualitatively, but also to some degree quantitatively identical to the r.m.s. distribution of the steady flow, confirming that the flow is inertia dominated.
B. Visualisation of the “swirl-switching” (Paper 2 and 3)

- The unsteady behaviour of the Dean vortices in turbulent flow—the so-called “swirl-switching”—was visualised for a bend with $\gamma = 0.31$ and for $Re = 3.4 \times 10^4$. The swirl-switching was reconstructed by means of POD applied on TS-PIV data. The analysis showed that only a few modes were needed to reconstruct the flow field and reveal the unsteady vortical motion while the inhomogeneous filtering that the POD is applying on the flow field helped to study the phenomenon further by means of spectral analysis. Two distinctive peaks in the spectra corresponding to $St = 0.04$ and $0.12$ were shown to be associated with the switching mechanism in agreement to Brücker (1998).

C. The effect of a swirling motion on the Dean vortices (Paper 3)

- Turbulent swirling flow through a curved pipe bend with $\gamma = 0.31$ was studied for a wide range of swirl numbers ($S = 0.1-1.2$). The Dean vortices are perturbed even for an imposed weak swirl motion, with the lower vortex being more sensitive to the motion since it is rotating in the opposite direction (counter-clockwise) as compared to the applied motion (clockwise direction). The upper vortex grows in strength and size as the swirl number increases until the flow becomes fully swirl dominated with a single vortex located at the centre of the pipe. Velocity profiles of the streamwise velocity component for the different swirl numbers showed that the flow field gradually becomes symmetrical and the centrifugal effects become weaker as the swirl number increases.

- The effect of the swirling motion on the secondary flow was further examined by means of POD. It was shown that the swirling motion contributes mostly to the total energy of the flow field, being the most energetic structure. The energy percentage of the 0-mode (mean field) increases from 60% for the lowest swirl number to almost 90% for the swirl dominated flow field. From the first two spatial modes (considering only the fluctuating part of the flow field) it is shown that coherent structures, constituting the most energetic features (regardless the 0-mode which shows the swirling motion) resemble the Dean vortices for all the swirl number cases studied. These structures are not as well structured as the Dean vortices but show the existence of other large scale features in the flow field, co-existing with the swirling motion.

- The instantaneous and reconstructed streamwise velocity fluctuations by using the most energetic POD modes were visualised using Taylor’s hypothesis. Elongated meandering structures were visualised, spanning a streamwise extent of about $5R$. The effect of curvature was seen as an
inclination of the structures as compared to the case of a corresponding flow case in a straight pipe, considered in the literature. Moreover, in a swirling motion the structures are tilted due to the change in the mean flow direction while for the swirl dominated flow motion they are teared up into shorter and wider structures.

D. Curvature ratio effects on turbulent curved pipe flow
(Paper 4)

- Turbulent flow ($Re = 2.3 \times 10^4$) at the exit of two curved pipes with curvature ratios $\gamma = 0.39$ and 0.17, denoting a sharp and mild bend, respectively, was studied by means of TS-PIV. Complementary LDV measurements were also performed. Flow fields were obtained at different stations downstream the two bends, at 1, 2 and 3$D$. Statistical analysis, vortex tracking and POD were performed on the data.
- Vortex tracking of the Dean vortices in the mean fields shows a tendency of the vortices to successively move towards the centre of the pipe with downstream distance from the exit of the bend. They also showed a tendency to locate more towards the centre for the sharp bend compared to the mild one.
- POD applied on the data revealed the existence of two separate motions for the case of the sharp bend for all downstream stations. Those were a single vortex dominating the entire cross-section in mode 1 and the Dean vortices being tilted and rotated in opposite directions in modes 2 and 3. Those modes resembled the ones in Hellström et al. (2013), which were associated with the switching mechanism. For the mild bend, that modal pattern was not so clear.
- LDV performed above the centreline in the vertical axis and along the horizontal axis of the mild and sharp bend provided the PDF of the vertical velocity component. A clear bimodal behaviour of the PDF was observed for the sharp bend whereas for the mild bend, only small wiggles were visible. This indicated that the swirl-switching in that case might not be as profound as for the sharp bend.
- Vortex tracking of the Dean vortices in time for both bends, showed that there is a clear periodic motion of the vortices whereas the upper vortex was anti-correlated to the lower one. It was additionally found that each one of the Dean cells dominated the flow field alternatively and equally in time. However, the vortices exhibited a more oscillatory character for the sharp bend compared to the mild one.
E. Effects of upstream conditions on turbulent curved pipe flow (Paper 4)

- A honeycomb was placed just upstream the inlet of a sharp bend ($\gamma = 0.39$) and the effects of such a setups on the flow field 1, 2 and 3D downstream the bend were studied by means of TS-PIV for $Re = 2.3 \times 10^4$.
- The existence of the honeycomb upstream the bend resulted in a significant delay of the flow field development. This was shown by analysing the mean streamwise velocity distribution at different stations downstream the bend.
- The secondary flow became successively distorted with downstream distance from the bend. The Dean vortices broke down in three cells at $1D$ whereas at $3D$, the upper vortex ceased and only a lower distorted vortex remained.
- The magnitude of the secondary flow was considerably damped when the honeycomb was present with the maximum mean value reaching $0.2W_b$ compared to $0.5W_b$ for the case without the honeycomb mounted.
- The modal pattern when the honeycomb was mounted did not resemble the one associated with the swirl-switching as described by Hellström et al. (2013). It was indicated that the switching phenomenon could be eliminated by the honeycomb, however it is not clear at this stage whether all oscillations were eliminated.

F. Secondary flow under pulsating turbulent flow (Paper 5)

- Pulsating and steady turbulent flow for Womersley numbers $\alpha = 0$, 30 and 71 and $Re = 2 \times 10^4$ was measured by means of TS-PIV at 0.2D and 2D distances downstream from a sharp pipe bend ($\gamma = 0.39$). Complementary LDV measurements were also performed and the agreement with the PIV data was found to be good.
- The flow field at a pipe cross-section under steady conditions exhibits two vortices which on average appear symmetrical to each other in both downstream stations. However, at the 2D station the Dean vortices exhibit an unsteady character in time, oscillating between three states, *viz.* a clockwise, an anti-clockwise and a symmetrical one. In contrast, at the 0.2D station the Dean vortices remain symmetrical in time as in the mean. This behaviour was further substantiated through POD and vortex tracking analysis.
- POD reconstruction of the pulsating flow by using only the four most energetic modes shows that POD provides a better representation of the flow field, compared to phase-averaging, which is the most common way of presenting turbulent pulsatile flows in the literature.
During high pulsations, the vortical pattern appears to depend highly on the cycle phase, with no secondary motions during acceleration while symmetrical vortical structures are formed during deceleration and a strong back flow sets in.

G. The effect of curved pulsating flow on turbine performance (Paper 6)

Time- and phase-resolved mass flow and pressure measurements were performed in highly pulsating turbulent flow, by means of a vortex-shedding flow meter (Laurantzon et al. 2010a) and fast pressure transducers, respectively. Turbine maps at high mass flow rates up to 105 g/s and at a pulsation frequency of 30 Hz are shown for the case when a sharp bend \( \gamma = 0.39 \) is mounted upstream the turbine in order to account for effects of the presence of curvature on the turbocharger performance. The hysteresis loop due to the filling and emptying of the turbine under a pulse period, was observed to expand to greater magnitudes as the mass flow rate increased. A slight effect of the presence of the bend at the inlet of the turbine was shown on the average quantities but a significant changes were observed in the instantaneous results.
CHAPTER 5

Papers and authors contributions

Paper 1

*Highly pulsatile turbulent flow past a straight and bent pipe*
A. Kalpakli Vester (AKV), R. Örlü (RÖ) & P. H. Alfredsson (HAL).
Submitted.

This work deals with hot-wire measurements on pulsating turbulent flow downstream a straight and bent pipe and focuses on the decomposition of the flow into its coherent and incoherent parts. The experiments were performed by AKV under the supervision of RÖ. The data analysis and the writing was done jointly by AKV & RÖ and comments were provided by HAL. This paper has been submitted for publication.

Part of this work has been published in:

*Experimental investigation on the effect of pulsations on turbulent flow through a 90 degrees pipe bend*
3rd *Int. Conf. on Jets, Wakes and Separated Flows*, 27 – 30 September 2010, *Cincinnati, OH, USA*

Paper 2

*Dean vortices in turbulent flows: rocking or rolling?*
A. Kalpakli (AK), R. Örlü (RÖ) & P. H. Alfredsson (HAL).
*J. Vis.*, 15, 37-38, 2012

This work presents clear snapshots of the three-dimensional flow field at a cross-section downstream a curved pipe. The experiments were done by AK. The data analysis and the writing was done jointly by AK & RÖ with input from HAL. This paper has been published in the Journal of Visualization.
Part of this work has been presented and selected to appear online at:

Reynolds and swirl number effects on turbulent pipe flow in a 90 degree pipe bend
A. Kalpakli, R. Örlü & P. H. Alfredsson.
64th Annual Meeting APS DFD,
20 – 22 November 2011, Baltimore, MD, USA

Dancing in the pipe
A. Kalpakli, R. Örlü & P. H. Alfredsson.
APS Gallery of fluid motion, Virtual Pressroom

Paper 3
Turbulent pipe flow downstream a 90° pipe bend with and without superimposed swirl
A. Kalpakli (AK) & R. Örlü (RÖ).
Int. J. Heat Fluid Flow 41, 103-111, 2013

This work deals with swirling turbulent flow downstream a pipe bend. The experiments and the data analysis were done by AK. The writing was done by AK with input from RÖ. This work was initially presented at ETMM9 (see below) and was selected for publication in the International Journal of Heat and Fluid Flow.

Part of this work has been published in:

Experimental investigation on the secondary motion downstream a pipe bend with and without swirl
A. Kalpakli, & R. Örlü
9th International ERCOFAC Symposium on Engineering Turbulence Modelling and Measurements,
6 – 8 June 2012, Thessaloniki, Greece
5. PAPERS AND AUTHORS CONTRIBUTIONS

**Paper 4**

*Turbulent pipe flow past a 90° pipe bend: effects of upstream conditions and curvature ratio*
A. Kalpakli Vester (AKV).

*Internal Report*

This work deals with turbulent flow downstream two pipe bends with different curvature ratios. The effect of upstream conditions on the turbulent flow field downstream a curved pipe is also examined.

Part of this work has been accepted for oral presentation in:

*Turbulent pipe flow past a 90° pipe bend: effects of upstream conditions and curvature ratio.*
A. Kalpakli, R. Örlü & P. H. Alfredsson.


**Paper 5**

*Vortical patterns in turbulent flow downstream a 90° curved pipe at high Womersley numbers*
A. Kalpakli (AK), R. Örlü (RÖ) & P. H. Alfredsson (HAL).

*Int. J. Heat Fluid Flow 44, 692-699, 2013*

This work deals with pulsating turbulent flow downstream a pipe bend. The experiments and the data analysis were done by AK. The writing was done by AK with input from RÖ and HAL. This work has been published in the International Journal of Heat and Fluid Flow.

Part of this work has been published in:

*The characteristics of turbulence in curved pipes under highly pulsatile flow conditions*  
iTi Conference in Turbulence, 30 September – 3 October 2012, Bertinoro, Italy.
Paper 6

Some observations of pulsating, curved pipe flow and its influence on turbine maps
A. Kalpakli Vester (AKV), R. Örlü (RÖ), N. Tillmark (NT) & P. H. Alfredsson (HAL).

Internal Report

This work examines the effect of a pipe bend mounted at the inlet of a turbocharger. The experiments were done by AKV whereas NT contributed to the setup of the turbocharger rig\(^1\). The data analysis was done by AKV (PIV data) & RÖ (turbine maps). The writing was done by AKV, RÖ and HAL.

Part of this work has been published in:

*Experimental investigation on the effect of pulsations on exhaust manifold-related flows aiming at improved efficiency*
10\(^{th}\) Int. Conf. on Turbochargers and Turbocharging,

\(^1\)Dr. Fredrik Laurantzon is acknowledged for his help on the setup of the turbocharger rig and for providing the LabView code for the turbocharger measurements
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